Preliminary Design and Assessment of Circulating-Bed Boilers

CS-1426
Research Project 1028-2

Final Report, June 1980
Work Completed, December 1979

Prepared by

PULLMAN KELLOGG
Research and Development Center
A Division of Pullman Incorporated
16200 Park Row, Industrial Park Ten
Houston, Texas 77084

Principal Investigators
L. D. Fraley
K. H. Hsiao
L. N. Do

Prepared for

Electric Power Research Institute
3412 Hillview Avenue
Palo Alto, California 94304

EPRI Project Manager
T. E. Lund

Fluidized Combustion and Nonconventional Fuels Program
Coal Combustion Systems Division

ORDERING INFORMATION

Requests for copies of this report should be directed to Research Reports Center (RRC), Box 50490, Palo Alto, CA 94303, (415) 965-4081. There is no charge for reports requested by EPRI member utilities and affiliates, contributing nonmembers, U.S. utility associations, U.S. government agencies (federal, state, and local), media, and foreign organizations with which EPRI has an information exchange agreement. On request, RRC will send a catalog of EPRI reports.

NOTICE

Prepared by
Pullman Kellogg
Houston, Texas

ABSTRACT

The circulating bed boiler (CBB) represents an alternative, fluidized bed combustor (FBC) technology which offers distinct advantages over both the current FBC systems, and pulverized-coal boilers with scrubbers. This report describes the findings of a study undertaken to evaluate these advantages. The information obtained made it possible to identify potential CBB design and operating problems and to propose further plans for developing this technology.

Several significant determinations resulted from the study. The circulating bed boiler capital costs should not exceed the cost for a conventional atmospheric fluid bed combustor, primarily due to the reduced combustor size; however, any cost advantage for a pressurized circulating bed boiler is questionable. Overall efficiency for an electric utility power plant using an atmospheric CBB should be increased by at least 1% over using a pulverized-coal boiler and the increase would be at least 3% using a pressurized CBB. The circulating bed boiler has several of the advantages of an FBC over pulverized coal, and in addition, it has turndown capabilities, greater throughput, and simplified feeding. Both the atmospheric and the pressurized CBB's can be designed with technology currently available in the process industry, but only after additional study and development has been completed for cyclones, pollution control, solids attrition, feed systems, and combustion reactions. Pilot plant studies are required for these investigations.

EPRI PERSPECTIVE

PROJECT DESCRIPTION

This final report describes attempts at the conceptual design of a 600-megawatt (electric), circulating-bed boiler without developing actual pilot plant data. The circulating-bed boiler represents an alternative, fluidized-bed combustion (FBC) technology that potentially offers important advantages over current, more-familiar FBC boilers as well as pulverized coal (PC) boilers with flue gas desulfurization systems. This report describes the findings of a study undertaken to evaluate these advantages. Both atmospheric pressure and pressurized designs are considered, and general economic conclusions are drawn.

PROJECT OBJECTIVES

Prior to undertaking this project, evidence existed from the process industries (e.g., petroleum, petrochemical, etc.) that a circulating-fluid-bed boiler could offer a number of operating and performance advantages over other FBC systems now under development, as well as existing PC systems. In particular, greater throughput due to higher velocity should be obtained; fuel feeding should be simplified; higher efficiencies should be achieved due to greater solid residence time; and configurational advantages could offer better load-following capabilities. The long-range objective of this project was to prove these advantages via experimental testing. Another objective was the development of conceptual designs for large units on which sound economic projections could be based. Pilot testing could not be carried out, and only this paper study was performed.

PROJECT RESULTS

Pullman Kellogg developed conceptual atmospheric and pressurized circulating-bed boiler designs and attempted to evaluate their expected performance based on published correlations of key system components plus internal Pullman Kellogg know-how. This project ultimately suffered from the lack of specific pilot plant data. Pullman Kellogg's overall conclusion was that the circulating bed would be simpler to design and fabricate (much of the technology is currently available in the process industry) and would be more responsive to load following while maintaining high efficiency. Pullman Kellogg's primary concern was the need to design a high-efficiency cyclone for the dense solids loadings inherent in this design. Future pilot plant studies are now planned to validate the key assumptions and to prove that high-efficiency cyclone designs are possible with high-dust loadings.

T. Lund, Project Manager
Fluidized Combustion and Nonconventional Fuels Program
Coal Combustion Systems Division

CONTENTS

CONTENTS (CONTD)

CONTENTS (CONTD)

FIGURES

FIGURES

FIGURES

APPENDICES

FIGURES

APPENDICES

FIGURES

FIGURES

TABLES

TABLES

SECTION 1

SUMMARY

1.1 INTRODUCTION

1.1.1 Circulating Bed Boiler (CBB) Description

In a conventional fluidized bed combustor (FBC), a bed of granular particles is supported by an upward flow of gas that is introduced uniformly through a grid (air distributor plate). The gas velocity, in an ideal device, is sufficient to float or suspend the particles, but it remains below the transport velocity of the solids. In actual operation, however, there is always a range of particle sizes due both to attrition within the combustor as well as an inherent feed size distribution due to solids crushing. In any case, the gas stream inevitably elutriates an appreciable quantity of solids from the bed as fines. In fact, a conventional FBC boiler may lose as much as 15% unburned fines due to premature elutriation. Since this loss is very detrimental, researchers have turned to using high efficiency cyclones to capture the unburned carbon for injection into a high temperature carbon burnup bed, or for reinjection into the main fluidized bed. In the latter situation, the fines remain in the recycle stream until attrition permits the material to escape from the high efficiency cyclone. Because the quantity of solids in the recycle stream can be large and tends to dominate the chemistry and physics occuring in the bed, the operation becomes much like that of a regenerator in an oil refinery's fluidized-catalyst cracking unit.

The circulating bed combustor (CBC) is characterized by having a gas velocity high enough to transport all particles introduced into the system upward. The solids are separated from the overhead gas by cyclone, and they return to the bottom of the combustor. In principle, this operation is similar to the fines recycle stream in a conventional FBC, but is different since the design parameters for the circulating bed are selected to maximize the gas-solids reactions. It is significant that the solids particle size and the gas residence time can be manipulated independently and this provides an additional degree of freedom over conventional FBC systems. High gas velocities (up to 60 fps) in CBC Systems offer compact designs; it is anticipated that a circulating bed of reasonable size and cost can be developed with an unburned coal loss of less than 1%. A similar argument shows superior SO_2 retention by the limestone sorbent in a circulating bed. It is anticipated that greater than 90% sulfur capture will be economically achievable in CBC Systems.

A circulating bed boiler (CBB) concept is illustrated in Figure 1-1. The following discussion describes the operation of the CBB. Coal and limestone (100μ) are fed by gravity into the combustor at Location A. This location is a highly-turbulent region where combustion air, feed coal and limestone, and recirculating solids are mixed. The feed solids constitute less than 1% of

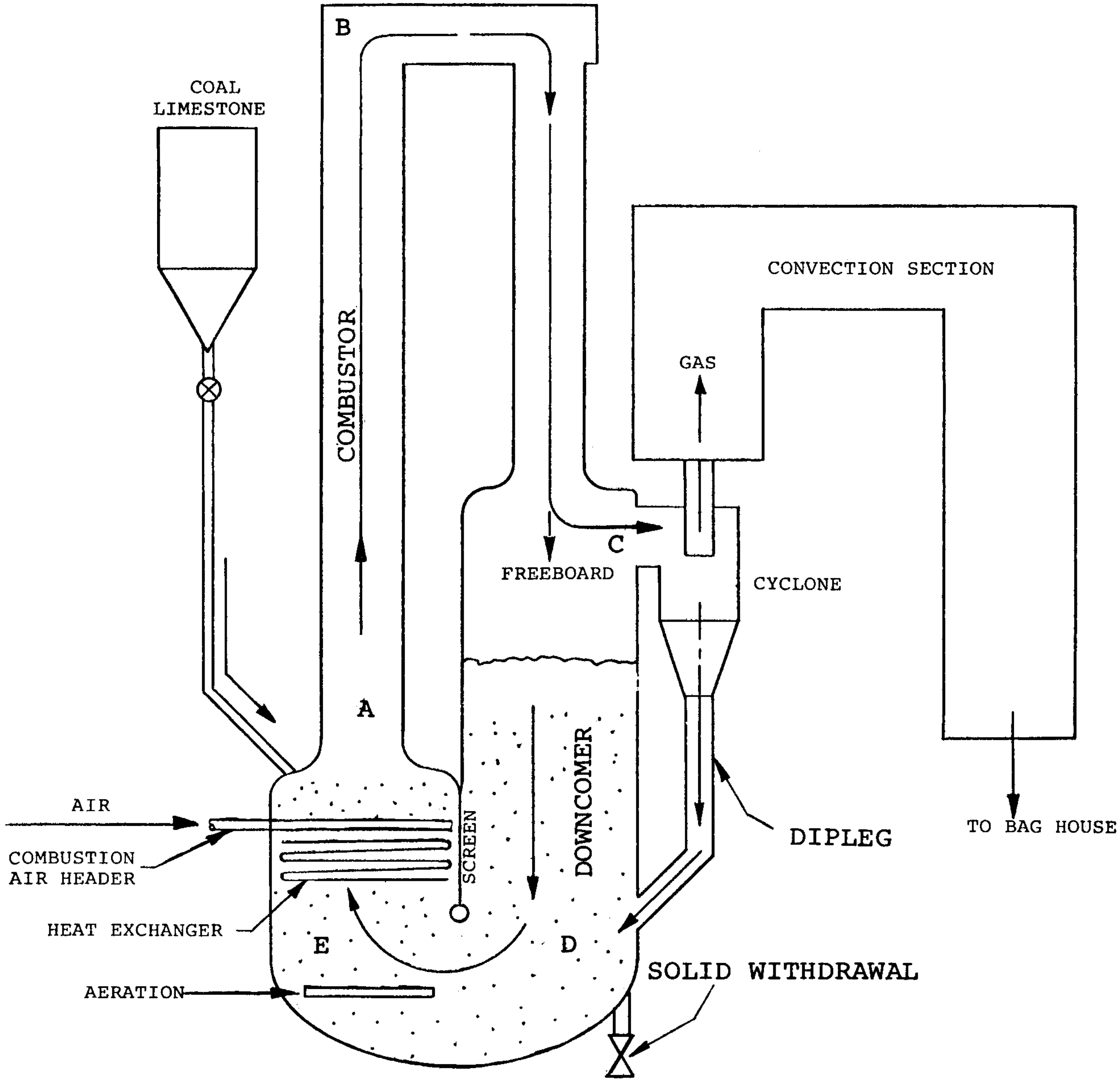


Figure 1-1. - Atmospheric Circulating Bed Boiler.

the recirculating solids. The recirculating solids, at 1650°F, heat the air and feed solids to the operating temperature in a fraction of a second. The residence time in Region A is greater than one second, and devolatization of the feed coal occurs here.

The mixture of solids and air flow upward into a cylindrical cross section where the gas and solids are accelerated to as high as 60 fps although, depending on the solid density in the gas, the initial solids velocity may only be half that of the gas. The volatiles burn immediately, followed by the slower burning coal particles. Sulfur dioxide formed during combustion reacts with the "in-situ", calcined limestone.

The reacting gas and solids flow upward, and then reverse direction through two, ninety-degree bends (Location B), in what is commonly termed a folded riser in the process industries. The turbulent inlet, the high gas velocity, and the two bends insure excellent gas-solid contacting throughout the system.

The folded riser discharges into a disengager (Location C), and the gas accelerates here to higher velocities (100 to 150 fps) as it enters the cyclones. Approximately 50% of the solids are disengaged prior to the turn into the cyclone and are decelerated in the downcomer. The solids entering the cyclone are separated and returned to the bottom of the downcomer (Location D). The solids flow into the dense bed from the downcomer (Location E). Fluidizing (aeration) air is supplied at 0.5 fps to promote solids flow and to enhance heat transfer as the solids flow upward through the heat exchanger surface. The fluidization (aeration) is sufficient to achieve high, solids-to-surface, heat-transfer coefficents (60 to 100 Btu/hr $ft^2 {}^\circ F$) and yet is sufficently mild to minimize errosive effects.

The mass input is balanced via solids leaving with the flue gas discharge from the cyclones and via solids withdrawn from the downcomer. The large mass of solids that recirculate consist primarily of unreacted lime (CaO) and sulfated limestone ($CaSO_4$).

The flue gas leaves the cyclones at the temperature of the combustor (1650°F) and is cooled to 300°F by flowing over a conventional convection bank and tubular, air preheater. A bag house provides the final particle removal before the gas is discharged to the stack.

The air is supplied by a compressor. The pressure drop in the combustor (8 psi) is dominated by two elements, the static head of solids in the folded riser and the acceleration of solids to high velocities.

When compared to a conventional pulverized coal (P.C.) boiler with a scrubber, the circulating bed boiler offers the same design and operating advantages that are offered by current fluidized bed boiler designs. These advantages include the ability to burn a broad range of fuels economically, operation at low combustion temperature to remove ash slagging fouling as well as reduce NO_x emissions, and the use of an "in-situ" sulfur sorbent, eliminating the need for a flue gas desulfurization system. There are, also, several additional advantages that the circulating bed boiler (CBB) offers which should improve upon current economics and operability significantly. The following list describes the additional advantages:

1. Two to three times the fuel throughput per unit base area due to higher combustion air velocities;
2. Combustion efficiencies greater than 99% along with required Calcium to Sulfur (Ca/S) ratios of less than 2.25;
3. Turndown ratios of near 5 to 1;
4. Greatly simplified fuel feeding equipment.

Major questions confronting the circulating bed boiler include:

1. The required cyclone efficiency and design;
2. The auxilliary power requirements;
3. The erosion characteristics of the system.

1.1.2 Purpose And Scope of Study

The initial purpose of this study was to prepare and evaluate the conceptual design of a circulating bed system, and to compare its projected capital and operating costs both with those for pulverized coal boilers with scrubbers, and with those for currently proposed (conventional) FBC systems. This information is to be used in deciding whether the circulating bed combustor warrants a significant development program.

The five tasks described in the following list were established to fulfill the requirements of the study:

1. Establish a data base;
2. Complete conceptual designs of circulating bed coal

fired boilers both for atmospheric and pressurized systems;

3. Develop conceptual designs of power plant systems using atmospheric and pressurized circulating bed boilers;
4. For each plant, estimate capital and operating costs; develop cost of electricity; and compare these costs to conventional FBC and pulverized coal plants;
5. Evaluate the study results and make recommendations concerning the development of the circulating bed.

The concept was to select a conventional, coal-fired, power plant and replace the pulverized coal system with a circulating bed boiler. The rest of the plant, including the coal source, remained unchanged. The cost of electricity was to be calculated and a comparison was to be made with the base P.C. boiler with scrubber case.

As the study progressed it became evident that the original plan was not capable of supplying the desired information for the following two reasons:

1. Insufficient design and operating data existed for the circulating bed system for a "reasonable" cost comparison to be made with a P.C. boiler, as well as a conventional FBC design.

2. Experimental testing was excluded from this project. Therefore, no means of obtaining the above information was available.

In retrospect, the major contributions of this study are:

1. Identifying the basic design and operating features which can be manipulated to make the circulating bed combustor an attractive device; and,
2. Establishing conceptual circulating bed combustor design arrangements which merit further consideration.

1.2 METHODS AND FINDINGS

1.2.1 Selection Of Ground Rules

At the start of the study, a plant size, steam cycle, types of coal and sorbent, and set of emission standards were chosen.

The selection of design conditions for the base case, pulverized coal fired power plant was complicated by the various philosophies concerning excess air that exist in the power industry. A Bechtel study (EPRI Report AF-342) was chosen initially but was rejected eventually due to excess air considerations.

The pulverized coal plant eventually selected was one being used for comparison in the DOE/TVA conventional FBC design studies. The steam cycle was 2500 psi/ 1000°F/1000°F with 4 inches Hg condenser pressure. Plant output is 525 MW_e net (1).

The coal selected was one used by EPRI in its internal comparative studies, and is nearly identical to the DOE/TVA coal. Table 1-1 presents the ultimate analysis.

TABLE 1-1.- EPRI COAL SPECIFICATION

TYPE: Deep-mined Illinois Coal

ULTIMATE ANALYSIS:

	Weight
Moisture	12.0
Carbon	57.5
Hydrogen	3.7
Nitrogen	0.9
Chlorine	0.1
Sulfur	4.0
Ash	16.0
Oxygen	5.8

HEATING VALUE:

H.H.V. = 10,100 BTU/lb

The selection of sorbent was made on the basis of kinetic data availability as well as close similarity to the sorbent utilized in the DOE/TVA studies. The Greer limestone selected has been investigated extensively by Westinghouse in its thermogravimetric analyses.

The emission standards selected were based on anticipated amendment to the 1977 Clean Air Act in which the following New Source Performance Standards (NSPS) are expected:

• particulates	0.5 lb/MM Btu
• SO_2	90% Removal of S
• NO_x	0.6 lb NO_2/MM Btu

1.2.2 Data Base

A survey was made of the technology available to select performance parameters for the circulating bed boiler. Major parameters are identified below along with comments concerning their end use in the designs.

1.2.2.1 SO_2 removal.- Westinghouse has developed a methodology for predicting the reaction rate of limestone during calcination and SO_2 absorption during coal combustion. Westinghouse data was used to predict the effects of particle size and gas residence time on SO_2 removal in the circulating bed. The residence time served to establish the size of the atmospheric combustor (the heat transfer surface set the size of the pressurized combustor). It should be pointed out that the results from the above predictive method have not been verified in a pilot plant simulating circulating bed conditions.

1.2.2.2 Stack gas temperature.- The stack gas temperature was set at 275°F based on normal power plant practice with P.C. boiler operation.

1.2.2.3 Coal combustion.- Westinghouse has also developed a kinetic model for combustion of coal which was applied to the circulating bed combustor. Results showed that if the combustor is sized for 90% SO_2 removal, then sufficient residence time will exist for over 99% combustion efficiency with 20% excess air. Again these results have not been verified in a CBB pilot plant operation.

1.2.2.4 NO_x emission.- Very little NO_x data were available that could be applied directly to the study. However, data from selected, conventional FBC tests indicate that a circulating bed should meet the emission standards.

1.2.2.5 Heat transfer coefficients.- Heat transfer coefficients from the circulating bed solids to heat exchange surface submerged in the bed were evaluated from literature, Battelle data, and from Kellogg field data; i.e., fluidized, catalytic cracking units. A value of 60 Btu/hr ft^2°F can be used for a conservative design. However, by selecting a small particle size and using 0.5 fps in the dense bed it is possible to achieve coefficients greater than 100 Btu/hr ft^2°F at bed temperatures of 1600°F.

1.2.2.6 Bed particle size distribution (and carryover of particles from cyclones).- The equilibrium particle size distribution within the CBB system will be determined by a number of design and operating parameters such as;

solids attrition, fractional efficiency of the cyclones, and physical effects resulting from particle burning and SO_2 removal. No information was available about limestone attrition, and it was necessary to select cyclones that are only currently being developed. Particle size distribution assumptions were based therefore on attrition and cyclone performance data taken from fluidized catalyst cracking units. Size distribution assumptions are critical to predicting whether a circulating bed boiler is technically feasible, and, therefore, experimental work is needed to verify these assumptions.

1.2.2.7 Gas solids fluid flow.- Kellogg experience with gas solid flow in the process industry has shown that flow calculations can be made confidently using published techniques.

1.2.3 Atmospheric Circulating Bed Combustors

Four design concepts were evaluated for the atmospheric circulating bed boiler (ACBB), and one was chosen to detail in this report. In addition, cycle studies were performed to determine the efficiency of power plants employing an ACBB.

The investigation of atmospheric circulating bed boiler conceptual designs was dominated by cyclone considerations. This study, as initially conceived, was to use only the technology which was readily available. When conventional process industry high efficiency cyclones were employed, eighty were required for the 525 MW_e plant. This requirement made two of the four ACBB designs impractical. The plumbing interfacing with the cyclones became so complex for these designs that they were eliminated immediately.

High capacity/efficiency cyclones were assumed for the designs on the basis that they could be developed commercially and would reduce the number of cyclones by a factor of four. This appears technically feasible although the expected fractional efficiency for small particles is difficult to estimate. A third design was developed assuming the use of twenty cyclones, and assuming inefficient collection of small particles. This produced a desirable combustor design, but, essentially, it moved the particle collection problem downstream of the combustor to a point in the convection bank where the flue gas and solids had cooled to 800°F. It was found necessary to recycle 800°F solids back to the combustor to maintain heat balance. This design may merit consideration in the future since it has no heat transfer surface in the circulating bed. Heat is carried into the convection bank by the excess particulate carryover.

The fourth design, and the one detailed in this report, (Figure 1-2), assumed that with development work, the high capacity cyclones would also have a high collection efficiency for small particles.

The major operating parameters assumed for this design were:

Bed temperature	1650°F
Gas residence time	3.5 sec
Average bed particle size	100 μ
Combustor solids density	4 lb/ft^3
Ca/S	2.25
Gas pressure drop	8 psi

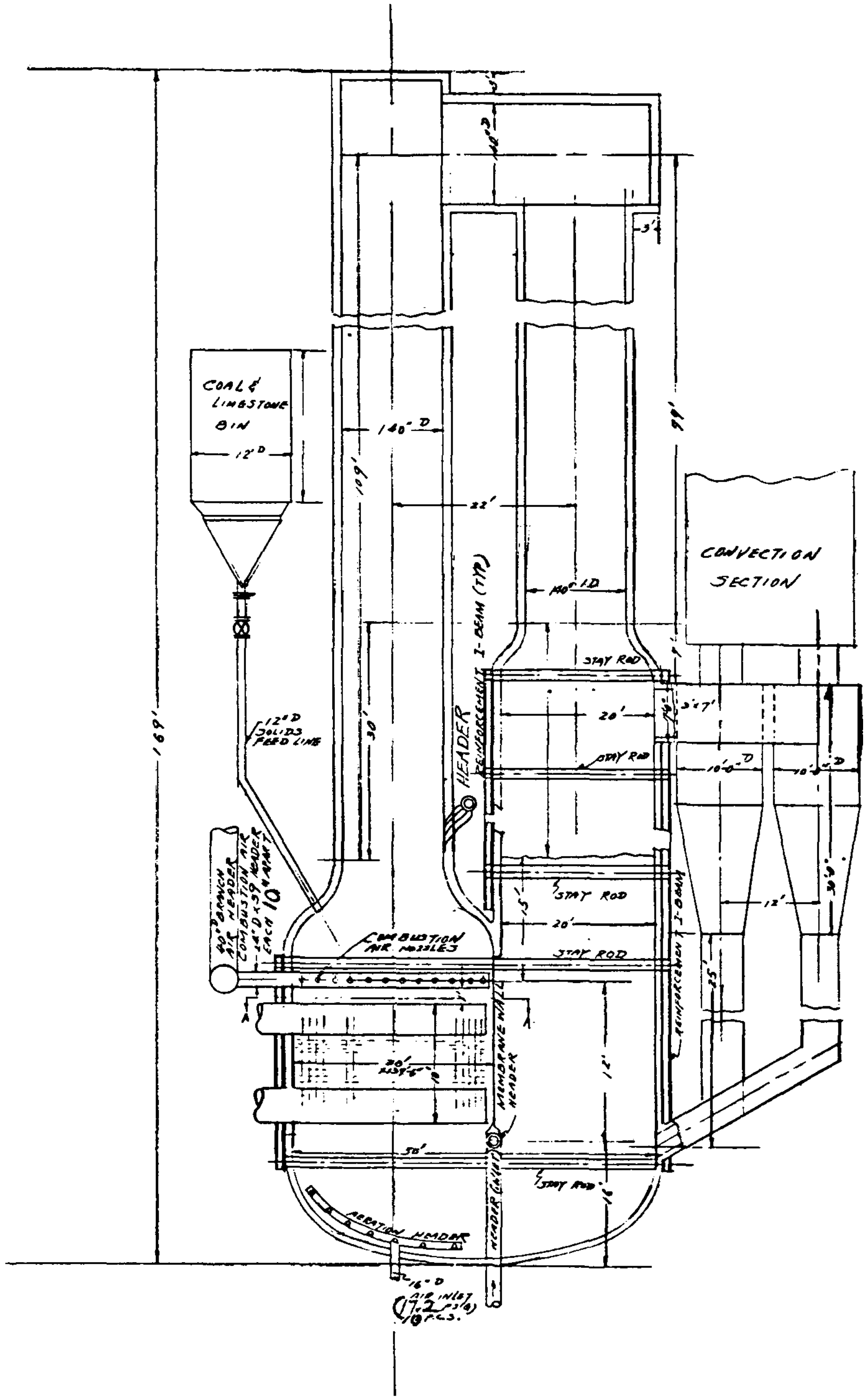


Figure 1-2. - Elevation View - Atmospheric Circulating Bed Boiler.

A T-Q diagram (Figure 1-3) was employed to determine the heat exchanger surface distribution within the combustor and convection sections. The steam plant driven by the combustor is essentially the same as the one employed in the DOE/TVA study with one major exception.

Due to the high pressure drop in the circulating bed (8 psi), the air leaving the forced draft compressor is maintained at 140°F. In a conventional plant this temperature is 80°F. The extra heat in the incoming air to the circulating bed is used to reduce extraction steam from the turbine to the feedwater heater. The result is that part of the compression work is recovered as power output from the steam cycle.

The plant efficiencies (η) for the ACBB and the base case P.C. fired plant were calculated based on the following equation:

$$\eta = \frac{\text{bus bar power out}}{\text{flowrate of coal x HHV}} \times 3413,$$

The ACBB showed an improved efficiency:

Atmospheric Circulating Bed = 34.9%
Pulverized Coal With Scrubber = 33.7%

An additional, cursory evaluation indicated that a high efficiency supercritical steam cycle (5000/1100/1050/1050) used with the atmospheric circulating bed combustor would increase the efficiency to 38.7%, which is five percentage points greater than that possible with pulverized coal firing.

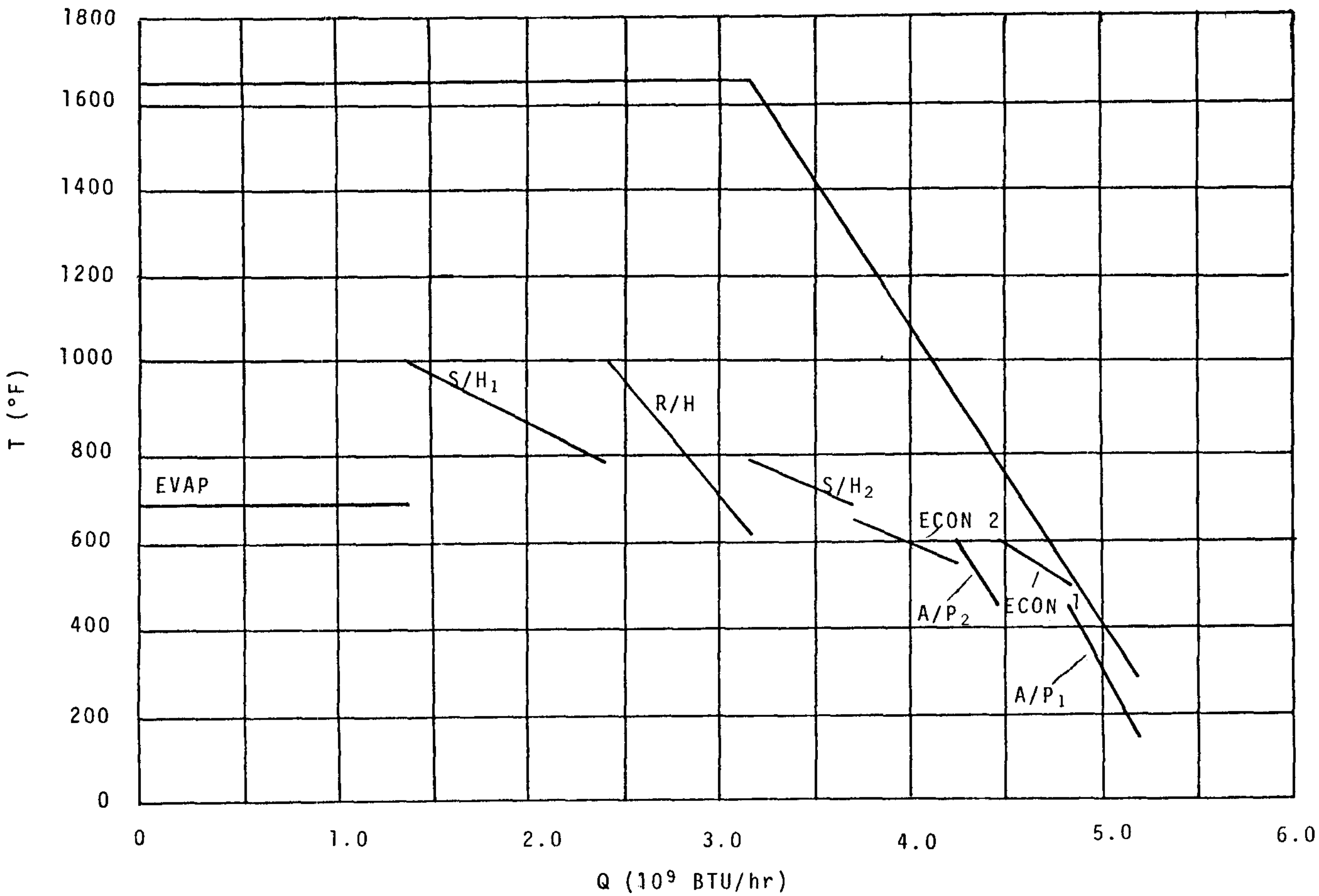


Figure 1-3. - T-Q Diagram for Atmospheric Circulating Bed Boiler.

The operation of the design shown in Figure 1-2 was previously described in the introduction. Key features are:

- Turndown - large turndown ratios may be possible by adjusting solids inventory and varying the fluidization air to the dense bed.
- Temperature control - separate "firing" is provided for the superheater, reheater, and evaporators.
- Fuel flexibility - there are no known limits on fuel characteristics. It is expected that any coal, liquid, or gas can be burned.

1.2.4 Pressurized Circulating Bed Combustor

While the basic principles are the same for atmospheric and pressurized fluidized bed boilers, reducing the concepts to design produces entirely different results.

The pressurized system must utilize both the gas and steam cycle. In the design used here, the combustor pressure was set at 8 atm by cycle analysis. Once this was established, it was found that the lowered gas volume made it relatively easy to meet the rest of the basic requirements of combustor design, including gas residence time, for combustion and SO_2 retention, and pressure drop.

The cycle that was analyzed is illustrated in Figure 1-4. It was derived from what is commonly termed a supercharged boiler, combined cycle. In this cycle, part of the heat generated by combustion is used to

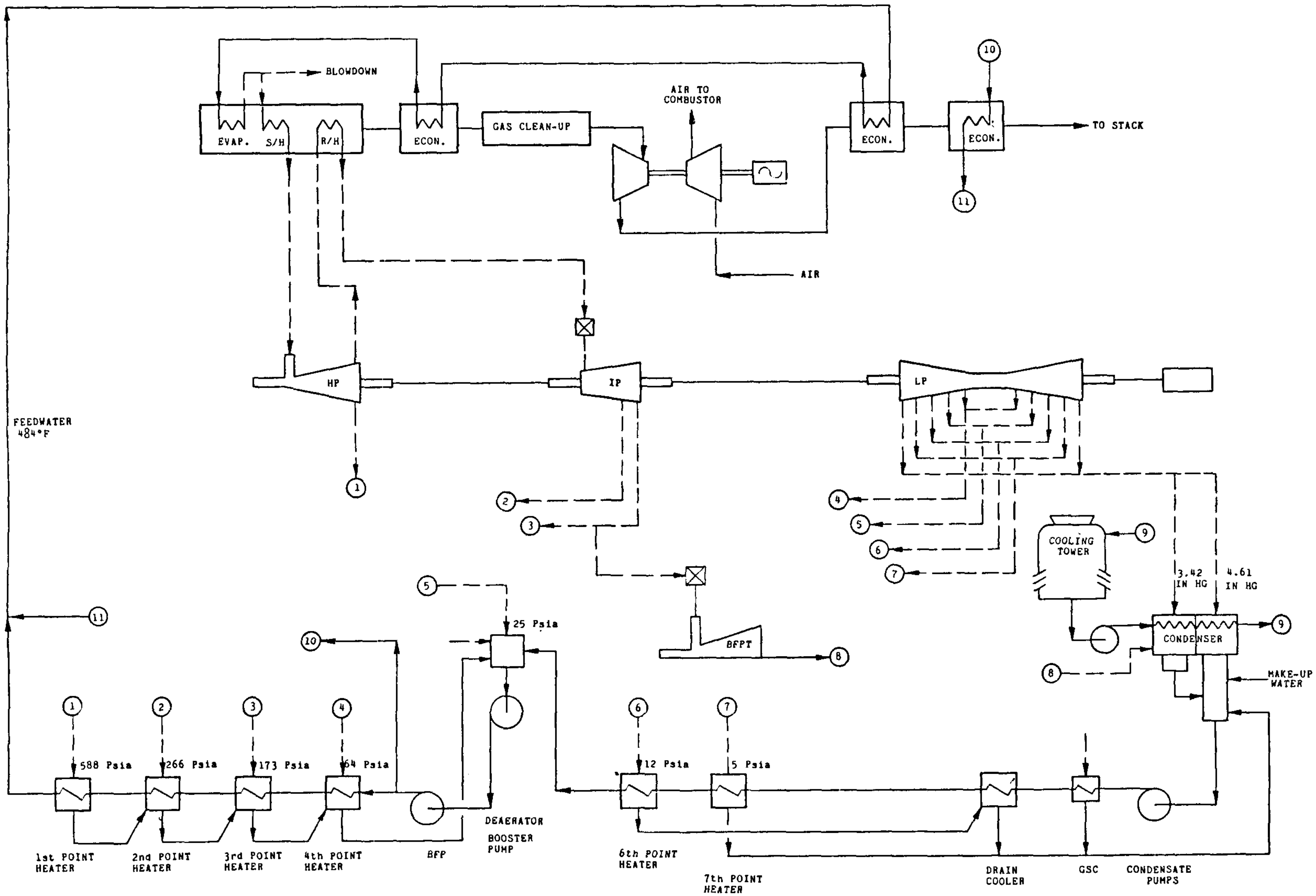


Figure 1-4. - Flow Diagram Combined Cycle for Pressurized Circulating Bed Boiler.

generate steam for a steam turbine cycle, and part is used by a gas cycle which incorporates flue gas expanders. The temperature of the inlet to the expander determines the split in the amount of heat available to the two cycles. The higher the inlet temperature, the higher the combined cycle efficiency will be.

For this study, again following the ground rule to use available technology, an inlet temperature of 1250°F was selected. The process industry has sucessfully operated expanders at this temperature downstream of fluidized catalyst cracking units for many years. It is reasonable to expect that similar successful operation can be achieved downstream of a pressurized circulating bed combustor. The major problem encountered is in capacity. For a 550 MW_e plant, fifteen expanders of the design used in the process industry are required. This number of units along with the piping to and from, and the complexity in control present a significant cost burden.

The steam cycle selected was similar to the others in this study (2500/1000/1000). The extraction steam for feedwater heating was reduced by using flue gas for this purpose. The heat in the flue gas is normally used to preheat combustion air, but the compressor discharge temperature (600°F) was high enough not to require an air preheater.

The heat load distribution is shown in the T-Q diagram, (Figure 1-5). The combined cycle efficiency, using the same definition as for the atmospheric combustor is:

Pressurized combustor efficiency = 36.7%.

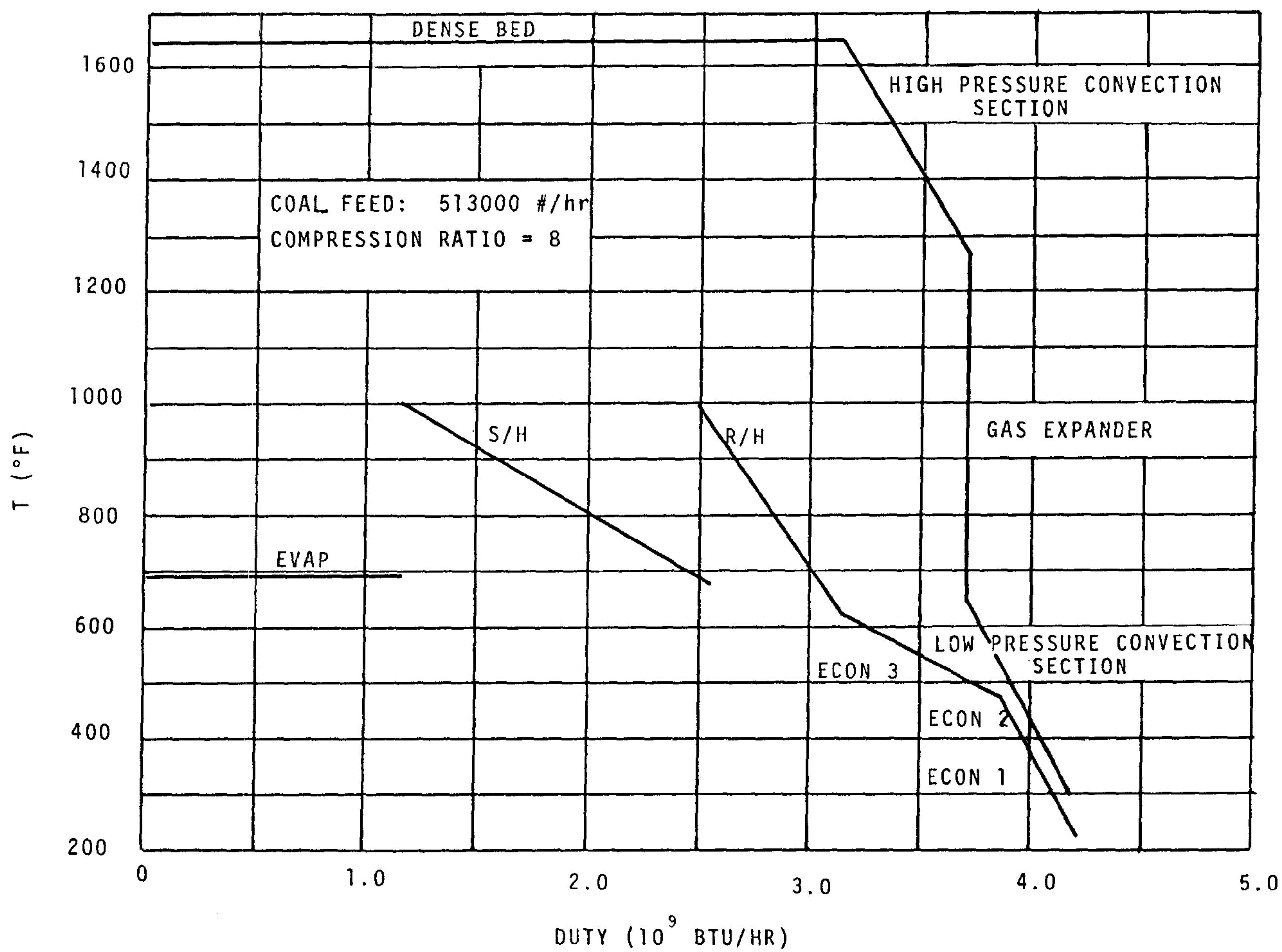


Figure 1-5. - T-Q Diagram Pressurized Circulating Bed Boiler.

This value is three percentage points higher than projected for a pulverized coal fired plant.

Two pressurized circulating bed boilers were evaluated. These differed by where the heating surfaces were located. In one, Figure 1-6, the heating surface was located in the tandem dense bed, similar to the atmospheric design. In the second design, Figure 1-7; the heating surface was located in the combustion zone. The design conditions were:

Bed temperature	1650°F
Gas residence time	4 sec
Average bed particle size	100 μ
Combustor solids density	12 lb/ft^3
Ca/S	2.25
Gas pressure drop	15 psi

In both designs, the size of the units were limited by the quantity of heat exchange surfaces.

The separator problems encountered with the atmospheric design were eliminated in the pressurized design because the elevated pressure reduced the gas volume. However, the lock hopper method for feeding coal and limestone at eight atmospheres has not been demonstrated and will have to be developed commercially.

1.2.5 Cost of Electricity (COE)

Initially, it was planned to estimate the capital and a operating costs for a pulverized coal fired plant, a conventional FBC system, and both an atmospheric, and a pressurized, circulating bed system. The final goal

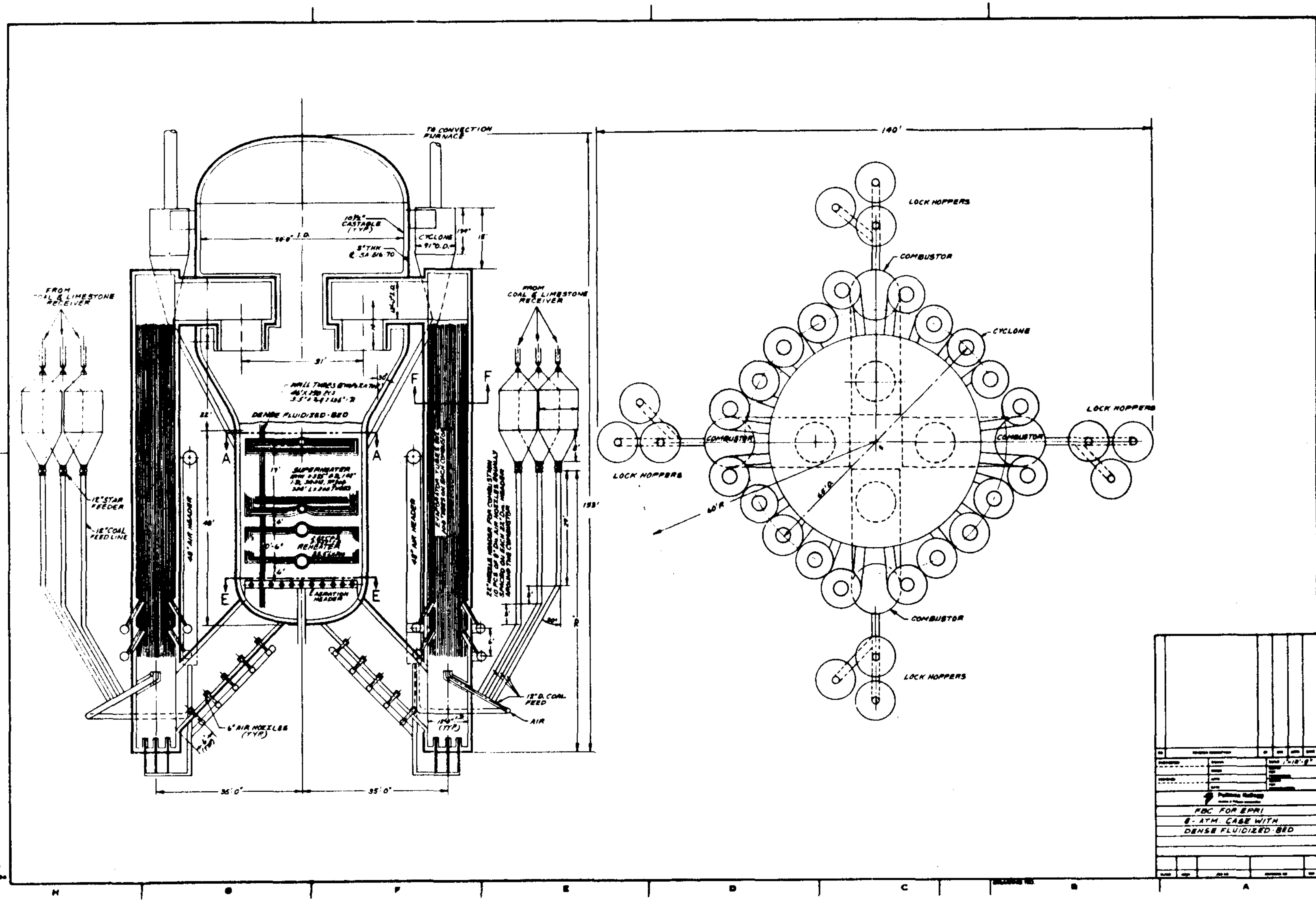


Figure 1-6. - Pressurized Circulating Bed Boiler with Dense Bed.

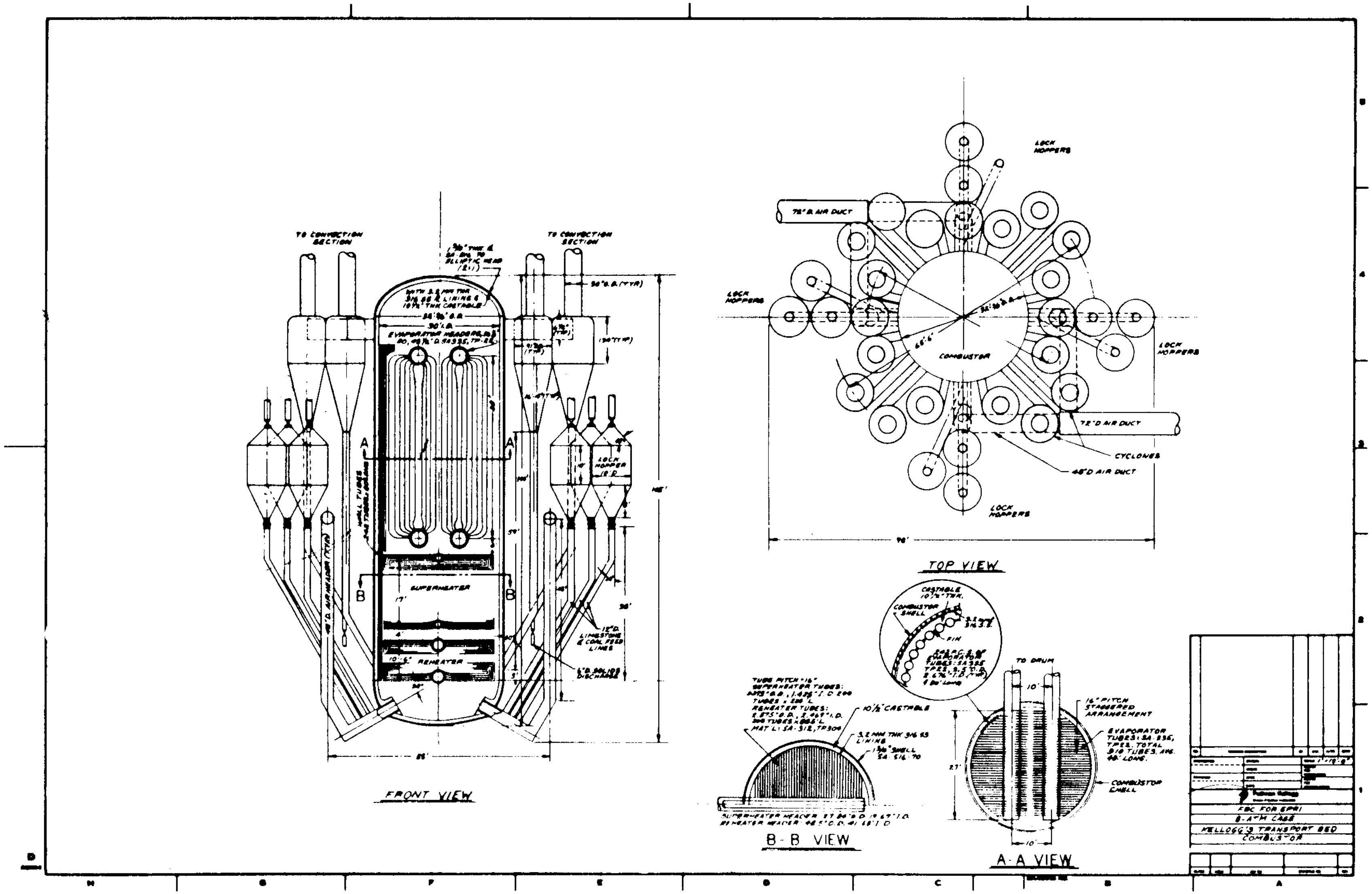


Figure 1-7. - Pressurized Circulating Bed Boiler without Dense Bed.

was to compare all four concepts on the basis of a single number, the cost of electricity (COE). The COE was to be calculated using an algorithm developed by EPRI which combines all costs in a manner representative of utility accounting. This concept became too difficult to execute.

The first problems developed when an attempt was made to establish a common method for estimating the capital cost of all four designs. After lengthy discussions with boiler manufacturers, it was determined that a pulverized coal boiler is normally estimated from what can be termed macro-design parameters; that is, type of coal and steam cycle conditions. Such an estimate is considered very accurate. This is possible because of the years of economic design studies, fabrication studies, erection, and general experience with a variety of real situations. Once the macro-design parameters are established the most economical design is known to very fine detail.

As a matter of contrast, if this study were to achieve a comparable level of accuracy with the FBC or circulating bed designs a micro-design would have to be estimated. This would require detailing such items as tube supports, tube bends, headers, insulation details, etc. Such work was beyond the scope of this study.

1.3 CONCLUSIONS AND RECOMMENDATIONS

1.3.1 Conclusions

1. Circulating bed boiler should not exceed conventional atmospheric fluidized bed combustor in capital cost, and will probably be less, due to reduced combustor size. The cost advantage of pressurized circulating bed boiler is questionable.

2. The overall efficiency of an electric utility power plant should be increased by at least 1% over pulverized coal - using atmospheric circulating bed boiler, and by at least 3% using pressurized circulating bed boiler.

3. Circulating bed boiler offers all the operating advantages of conventional fluidized bed combustor over pulverized coal, and offers the following additional advantages:

   - turndown capabilities
   - greater throughput
   - simplified feeding

4. Actual capital costs for circulating bed boiler cannot be determined due to lack of data.

5. An atmospheric circulating bed combustor can be designed with technology available in the process industry, and with a reasonable amount of development work. Development areas are cyclones, solids attrition, NO_x, SO_2, and combustion reactions.

6. A pressurized circulating bed combustor can also be designed using process industry technology. Major developmental areas are solids attrition, NO_x, SO_2 and combustion studies, and solids feed hardware.

1.3.2 Recommendations

The following items are recommended for the development of a circulating bed:

1. Detailed design investigations should be performed to reduce costs and to select the most feasible mechanical design.

2. High capacity cyclones should be developed. The number of cyclones should be twenty or less for a 570 MW_e atmospheric system.

3. A pilot plant scale, circulating bed apparatus should be built. The apparatus should investigate;

   - Solids attrition
   - Coal combustion rate
   - SO_2 removal rate
   - NO_x emissions
   - Fuel flexibility

SECTION 2

DESIGN SPECIFICATIONS

2.1 BASE PLANT: PULVERIZED COAL PLANT

The base case plant chosen for comparison with the circulating bed boiler is a pulverized coal boiler (570 MW_e) and stack gas scrubber, presented in a recent study by Stone & Webster Engineering Corporation (1) for DOE. Figure 2-1 illustrates the steam cycle used in this plant. Briefly, the cycle is a 2500 psi, 1000°F superheat and 1000°F reheat temperature cycle. It has seven stages of extraction heating, a bi-level condenser at 3.36 and 4.43 inches Hg, a deaerator at 60 psia, and final feedwater temperature of 484°F.

This steam cycle was adapted, with minimum modification to the circulating bed combustor plant.

2.2 COAL SPECIFICATIONS

The type of coal selected for use in this study (see Table 2-1) is a high sulfur (four weight percent), Illinois coal with a 10,100 Btu/lb higher heating value.

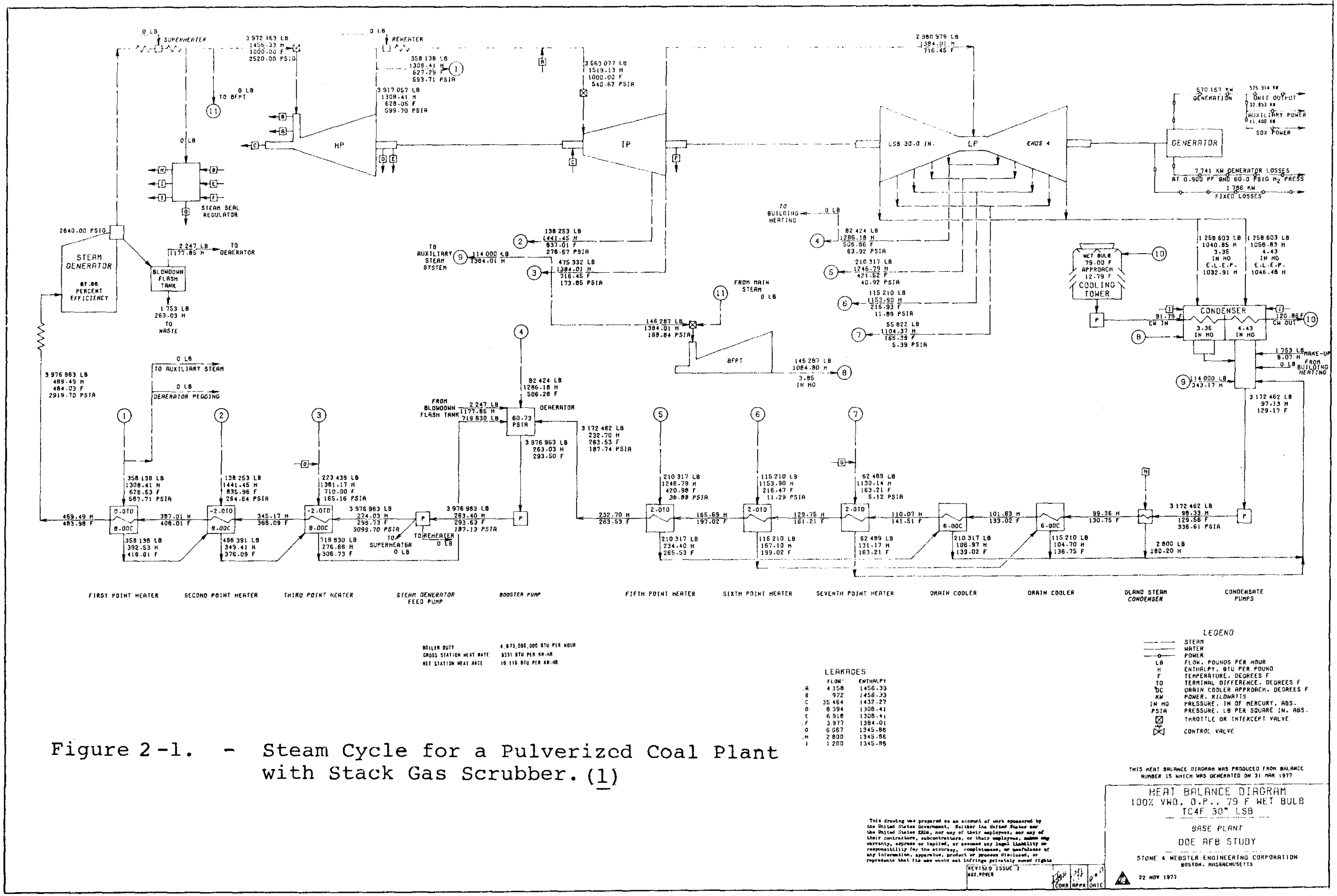


Figure 2-1. - Steam Cycle for a Pulverized Coal Plant with Stack Gas Scrubber. (1)

TABLE 2-1.- EPRI COAL SPECIFICATION

Type: Deep-mined Illinois coal

ULTIMATE ANALYSIS:

	Weight %
Moisture	12.0
Carbon	57.5
Hydrogen	3.7
Nitrogen	0.9
Chlorine	0.1
Sulfur	4.0
Ash	16.0
Oxygen	5.8

HEATING VALUE:

H.H.V. = 10,100 BTU/lb

2.3 SORBENT SELECTION

The selection of a sorbent depends on various considerations including availability, composition, performance, and cost (see the discussion on SO_2 removal in Appendix I).

For this study, the following limestone was selected; it is the same one used in the pulverized coal plant (1).

Moisture	:	2% by weight
$CaCO_3$	:	94.1%
$MgCO_3$	:	3.3%
Other	:	0.6%

2.4 EMISSION CONTROLS

The designs in this study were required to meet EPA current and projected New Source Performance Standards (NSPS) for emission control. These standards are listed in Table 2-2 for NO_x, SO_2 and particulate emissions. Other rules or specifications used in the study will be mentioned and discussed when they become appropriate to the text.

TABLE 2-2.- EMISSION STANDARDS

	CURRENT	New (NSPS)
SO_x	1.2 lb SO_2/MM Btu	90% SO_2 removal efficiency
NO_x	0.7 lb NO_2/MM Btu	0.6 lb NO_2/MM Btu
Particulate	0.1 lb/MM Btu	0.05 lb/MM Btu

2.5 DESIGN ASSUMPTIONS

Different designs for circulating bed boilers are considered in the subsequent sections of this report. They are based on the assumptions discussed in detail in Appendix I and are highlighted below:

A. The circulating bed boiler is sized to meet the sulphur dioxide removal requirement. The gas residence time required has been calculated based on a model developed by Westinghouse.

B. The residence time dictated by the sulphur dioxide removal, is sufficient to provide high combustion efficiency. A reasonable balance between thermal efficiency and combustion efficiency can be obtained at 20% excess air.

C. NO_x emission does not appear to be a problem.

D. The heat transfer coefficient to tubes submerged in a dense fluidized bed and in a circulating fluidized bed were characterized. Conservative designs would use values of 60 B/hr ft^2°F while more optimistic designs could use values exceeding 100 B/hr ft^2°F.

E. The solid carryover is a function of solid feed, attrition, bed density and cyclone efficiency. To maintain heat and material balance in the combustor, high efficiency cyclones (above 99.99% collection efficiency) are necessary.

F. Pressure drops in the combustor are calculated from gas/solid flow formulation available in standard texts (2).

SECTION 3

ATMOSPHERIC CIRCULATING BED BOILER SYSTEM

3.1 GENERAL

The selection of combustor conditions, the design of the combustor, the steam plant arrangement, and the overall performance of the circulating bed combustor plant are all requisite considerations in designing an atmospheric, fluidized-bed-boiler, power plant. These factors and their associated problems are evaluated and presented here.

3.2 DESIGN RATIONALE

This section discusses the factors and considerations for determining an attractive combustor design that results in high plant efficiency and relatively low capital cost.

The principal design variables discussed here are bed density, gas velocity, and solid particle size. The selections made among these variables were based on considerations of heat transfer, system pressure drop, reaction kinetics, overall plant efficiency, equipment size, and operability.

3.2.1 Bed Density

The selection of bed density for the combustor affects both the performance and operation of the combustor, and, therefore it affects the final plant efficiency.

A. The pressure drop in the combustor increases with bed density. Figure 3-1 shows this effect in a 100 foot high bed, operating at gas velocities from 20 to 100 ft/sec. As density increases from 0 to 10 lbs/ft^3 both the static head and acceleration head (resulting from accelerating the solids to velocity) increase resulting in a total combustor pressure loss of from 12 to 30 psi. The higher pressure drop increases the design pressure, increases the compressor horsepower, and results in a penalty to cycle efficiency. Figure 3-2 shows the plant efficiency lost due to the increase in auxiliary power consumption. As pressure drop increases from 2 to 15 psi, from 1 to 5.5 percentage points in efficiency are subtracted from the gross plant efficiency.

B. Higher bed densities improve the heat transfer for the heating surfaces installed in circulating beds (see Appendix I, Basic Studies - Heat Transfer). If the combustor design uses this type of heating surface installation, a high bed density would improve the heat transfer performance and reduce the tube surface area requirements.

C. A high bed density reduces the residence time requirement for the SO_2 removal reaction. The gas residence time versus bed density is shown in Figure 3-3 (also refer to Appendix I, Basic

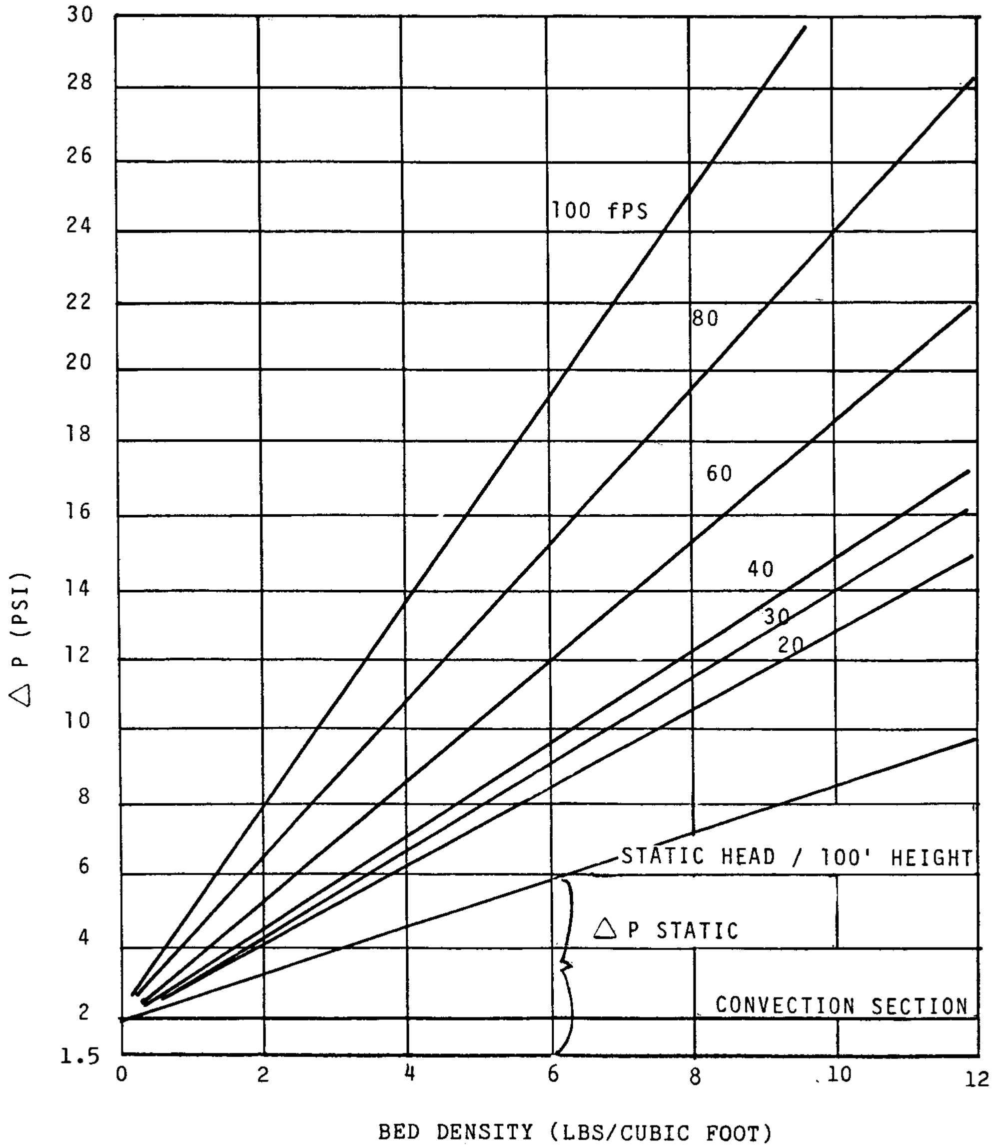


Figure 3-1. - System Pressure Drop Distribution vs Solids Density in Combustor.

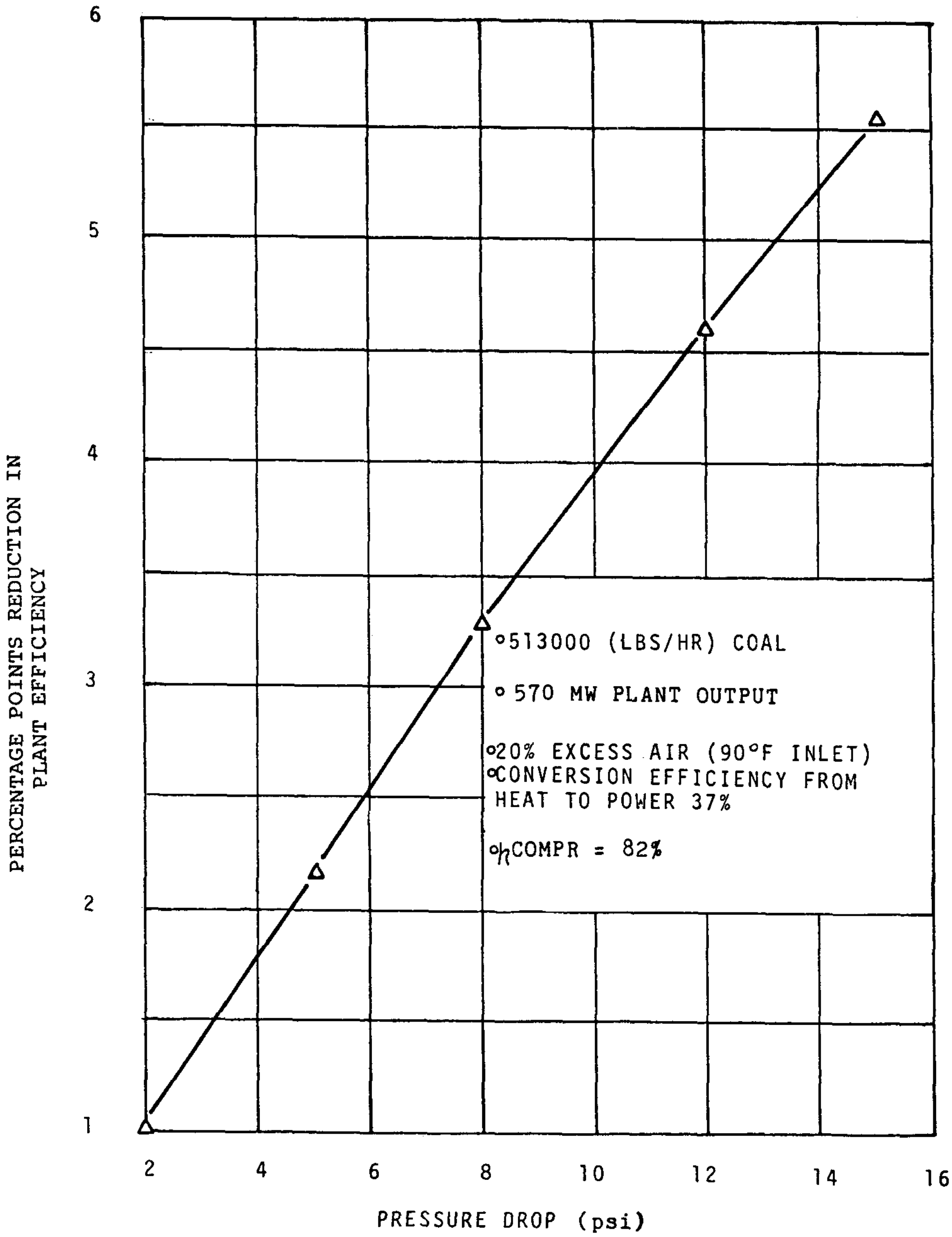


Figure 3-2. - Reduction in Plant Efficiency vs Pressure Drop in Combustor.

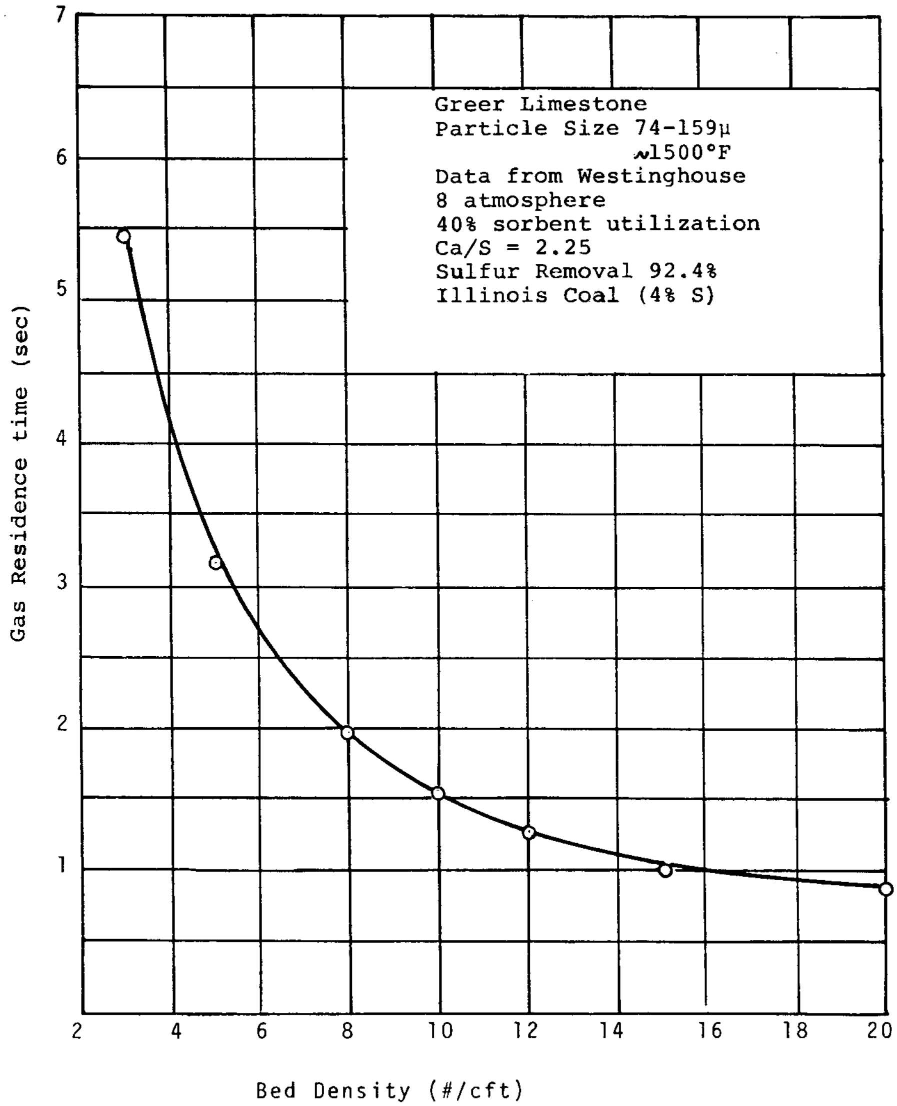


Figure 3-3. - Design Curve - Gas Residence Time vs Circulating Bed Density.

Studies - SO_2 removals). The residence time for SO_2 removal to the NSPS projected standard (0.6 #SO_2/MM Btu) decreases from 5.5 to 1.5 seconds as bed density increases from 3 to 10 lbs/ft^3. A low residence time would reduce the combustor height requirement.

D. Another effect to be considered when selecting bed density is the solid carryover and loading to the plant solid collecting system. A bed operating at high density would increase the solid loading to particle cleaning equipment, and high solid loadings would require high performance equipment to clean particulate emissions to the NSPS level. This requirement adds constraints to the design and selection of the plant solid cleaning system and it will be reflected in the equipment cost.

Heat loss associated with the solids entrained out of the combustor at high temperature was an aspect of the solid carryover problem investigated in this study. Figure 3-4 shows the magnitude of this heat loss for different bed densities and cyclone performance. As shown, high solid densities would require high performance of the cyclone (above 99.9% collection efficiency) to limit the heat loss from the combustor. Because the cyclones selected in this study were assumed to meet these performance levels (see Appendix I, Basic Studies - Solid Carryover), this heat loss will be assumed negligible.

However, for a case when the cyclones cannot achieve these performances, and the heat loss becomes

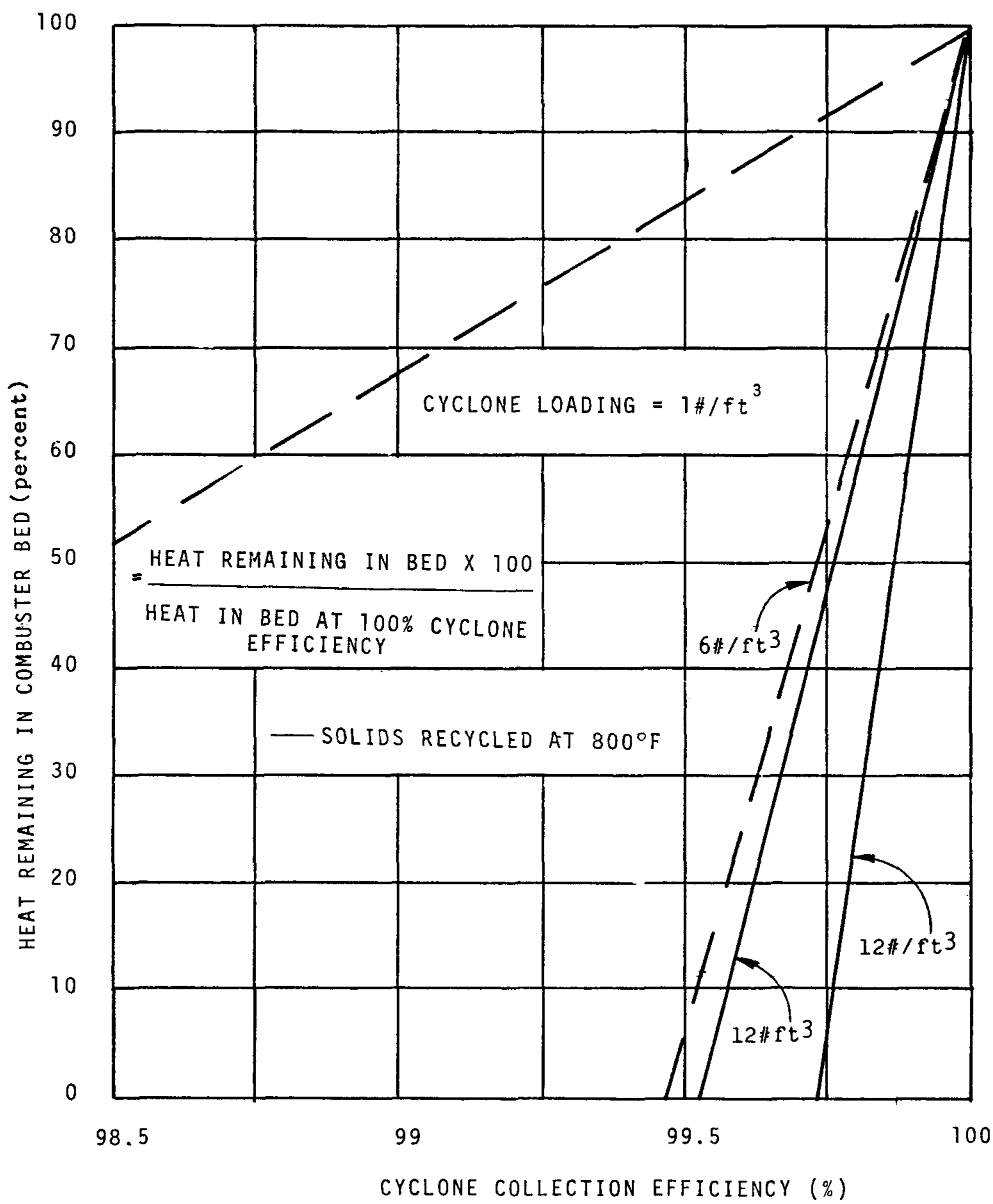


Figure 3-4. - Heat Losses from Cyclone Carryover.

significant, a different combustor design was worked out and included in Appendix II.

3.2.2 Gas Velocity

The gas velocity selection affects reaction kinetics, system pressure drop, combustor plot plant, and equipment erosion. These functions are discussed in the following paragraphs.

A. The degree of gas/solid mixing in the combustor may vary with the gas velocity in the combustor, and this will have effect on the chemical reaction kinetics. Tests have been conducted to measure gas and solid backmixing in the pilot plant scale reactor at gas velocities up to 25 ft/sec (References 3 and 4). The results indicated that, at gas velocities corresponding to fast fluidized bed flow regime, the gas is essentially in plug flow while the solids show some degree of backmixing (and slip). This solid backmixing is believed to be beneficial to the gas solid reactions inside the combustor. At higher gas velocities, the flow regime switches to a dilute phase flow with the gas still in plug flow but with much less solid backmixing. For a combustor operating in this flow regime (for example around 60 ft/sec), the reduced backmixing of solids may or may not affect the chemical reactions that are occurring. This question must be investigated in future development work.

B. The selection of gas velocity affects the pressure drop in the combustor as shown in Figure 3-1. The acceleration component of pressure drop increases when the solids in the combustor are accelerated to higher velocities. As mentioned earlier, this increase in system pressure drop would result in a loss in cycle efficiency.

C. The major advantage in having high gas velocities in the design of the combustor is in the potential for savings in the base area for the combustor. Because there are high gas flow volumes in power plant applications (compared to process applications), there is added incentive to investigate the high range of gas velocities to achieve more economically attractive combustor designs. Although there is some increase in combustor height with higher gas velocity (based on a fixed gas residence time requirement), this increase is largely offset by the reduction in combustor diameter and overall size. Moreover, the reduction in diameter achieves a more uniform flow distribution. These arguments play an important role in the final selection of gas velocity for the combustor.

D. Another consideration in the selection of gas velocity is erosion of equipment. The higher the velocity in the combustor, the higher the potential for erosion damage by solid particle impact.

3.2.3 Solid Particle Size

Another variable to be established in combustor design is the solids particle size. The factors involved in this selection are chemical reaction kinetics, heat transfer performance, and particle size availability.

Small particle sizes (around 100 microns average) were considered for this application primarily to achieve high reactivity (combustion and SO_2 removal) as well as good heat transfer rates. The penalties paid using fine particle sizes would be primarily in the additional cyclone costs as well as the power consumption for crushing the solid feeds to finer sizes.

3.2.4 Selection of Design Parameters

The principal features emphasized in establishing attractive design are low system pressure drop, good plant efficiency, and a compact boiler design.

The requirement for a low pressure drop influences the selection of bed density, and for this design, density of around 4 lbs/ft^3 was chosen. At this density, the gas residence time requirement for SO_2 removal from the bed is 3.5 seconds. Next, to achieve a compact combustor design, a 60 ft/sec gas velocity was chosen. In this case, the chosen velocity was limited to maintain a low system pressure drop (around 8 psi) while maintaining the combustor height within practical limits.

3.3 CYCLE STUDY

The cycle study and plant performance for the circulating combustor design included evaluation of the steam cycle, the heat load for the steam generator, and the plant efficiency. These studies are consistent with the design parameters chosen in Section 3.2.4.

3.3.1 Power Cycle

Figure 3-5 shows the flow diagram for the steam plant. As mentioned earlier, the steam cycle selected to evaluate the fast bed combustor is a 2500 psia, 1000°F superheat temperature and 1000°F reheat temperature similar to the cycle used in the pulverized coal boiler plant of Reference 1. A steam balance for the plant is shown in Figure 3-6. The cycle uses seven stages of extraction heating to preheat feedwater to a final temperature of 484°F. The deaerator pressure is set at 60 psia and the bilevel condenser at 3.36 and 4.43 inches Hg. The total feedwater flow to the steam generator is 3.977×10^6 lbs/hr. A detailed description of the steam plant and equipment (power turbine, generator, condenser, feedwater pumps, deaerator) is provided in Reference 1.

3.3.2 Heat Load For The Circulating Bed Combustor

The circulating bed boiler system replaces the pulverized coal boiler in a conventional steam power plant as the source of heat. The heat load arrangement for the combustor is shown on the T-Q diagram of Figure 3-7. The combustor, burning coal in a fluidized bed at 1650°F, supplies heat at that temperature to a steam evaporator, a superheater and a reheater. After the solids in the cyclone system are disengaged, the waste heat in the flue gas is recovered in a convection section. The arrangement of the convection section shown on the T-Q diagram, includes one bank of superheater tubes and two banks of economizers and two banks of air preheater tubes. The combustion air leaves the

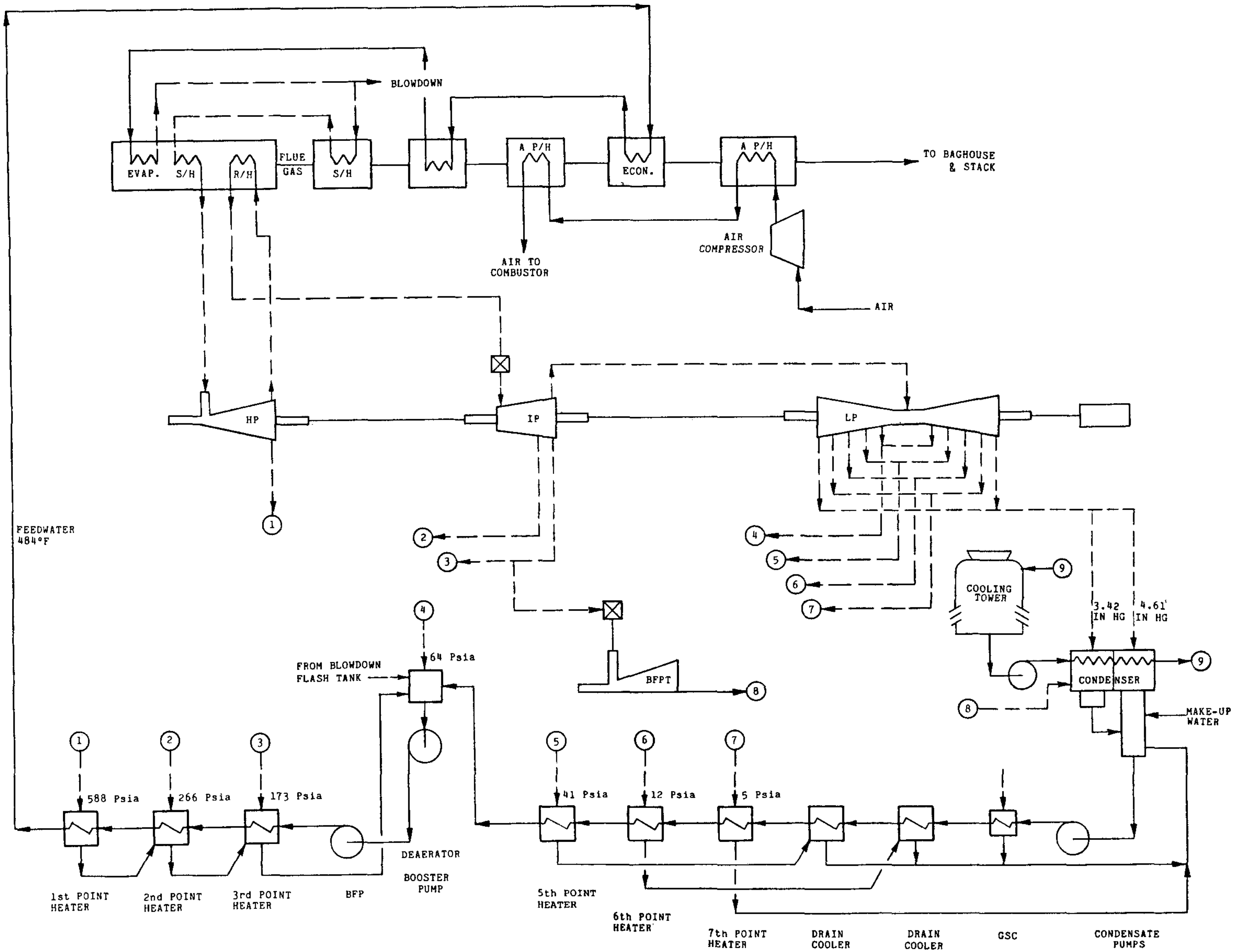


Figure 3-5. - Atmospheric Circulating Bed Boiler - Flow Diagram.

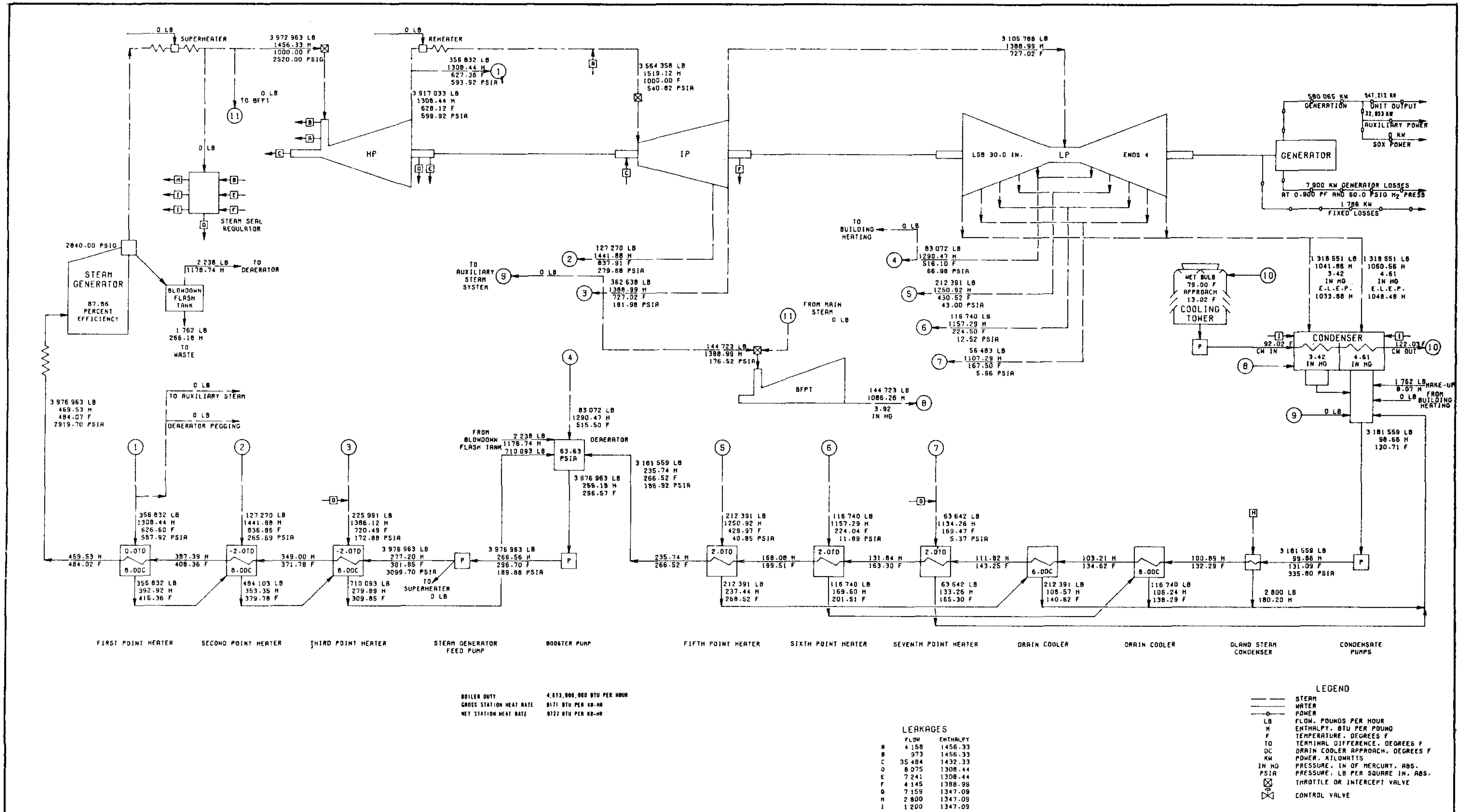


Figure 3-6. - Steam Balance for Atmospheric Circulating Bed Combustor Power Plant. (1)

THIS HEAT BALANCE DIAGRAM WAS PRODUCED FROM BALANCE NUMBER 19 WHICH WAS GENERATED ON 9 SEP 1977

HEAT BALANCE DIAGRAM
100% VWO, O.P., 79 F WET BULB
TC4F 30" LSB

BASE PLANT WITHOUT FGD

DOE AFB STUDY

STONE & WEBSTER ENGINEERING CORPORATION
BOSTON, MASSACHUSETTS

28 NOV 1977

REVISED ISSUE 2
AUX. POWER

CORR APPR DATE

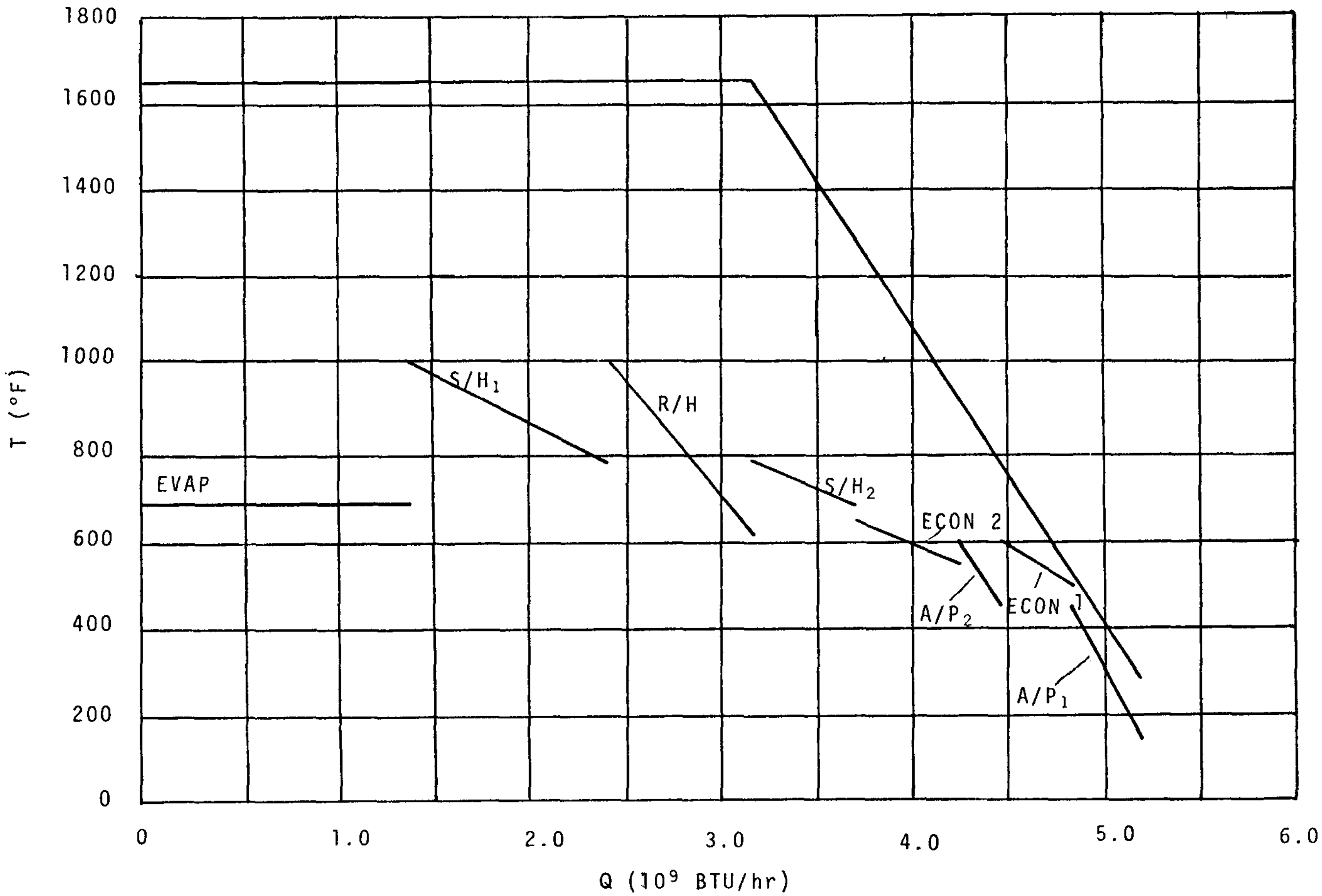


Figure 3-7. - T-Q Diagram for Atmospheric Circulating Bed Combustor.

final air preheater at about 600°F. The flue gas leaves the convection section at 280°F.

3.3.3 Plant Efficiency

The overall performance of the circulating bed combustor plant compared to one using a pulverized coal boiler is summarized in Table 3-1.

TABLE 3-1.- CIRCULATING BED VERSUS PULVERIZED COAL BOILER PERFORMANCE

	Circulating Bed Combustor Plant	Pulverized Coal Plant (Ref. 17)
Heat Input From Coal (HHV) BTU/Hr	5.18×10^9	5.32×10^9
Gross Plant Output (MW)	580.06	570.17
Aux. Power Consumption (MW)		
Steam plant	8.49	8.49
Fans	34.20	15.74
Scrubber		11.40
Others	7.65	8.62
Total	50.34	44.25
Net Plant Output (MW)	529.72	525.92
Net Plant Efficiency	34.90	33.74

The main difference between the two plants is in auxiliary power consumption. The circulating bed combustor plant shows a higher fan power consumption (due to higher system pressure losses), 34.20 MW compared to 15.74 MW for the pulverized coal boiler plant. On the other hand, since the circulating bed combustor plant does not require stack gas scrubbing, the plant realizes a saving in auxiliary power over the pulverized coal boiler plant equipped with stack gas scrubber. The final plant efficiency is 34.90% for the fast bed combustor plant compared to 33.74% for the pulverized coal boiler plant with scrubber.

3.4 COMBUSTOR DESIGN

3.4.1 Design Conditions

The design conditions presented below are incorporated in the design concept described next and are consistent with the plant output and efficiency presented in Table 3-1.

Coal Feed	:	513,000 lbs/hr
Air	:	4,810,000 lbs/hr
Limestone	:	137,000 lbs/hr
Flue Gas	:	5,254,700 lbs/hr
Bed Temperature	:	1,650°F
Total System Pressure Drop	:	8 psi Gas
Residence Time	:	3.5 sec
Combustor Solids Density	:	4 lbs/ft^3

3.4.2 Combustor Sizing and Construction

The governing parameters in sizing the combustor are: the gas velocity in the combustor; the flow rate of the flue gas which determines the diameter and the number of combustors; the required gas residence time which determines the combustor height; and the recirculated solids flow rate which determines the cross-sectional area of the downcomer as well as the dense bed. Once these parameters are set, the physical size of combustor and bottom housing is fixed.

For the flue gas weight flow rate of 5,754,700 lbs/hr, the corresponding volume flow rate is 2.28 (10^8) ft^3/hr. The solids flow rate through the combustor is 9.12 (10^6) lbs/hr. The elevation of the combustor is shown in Figure 3-8 and plan view shown in Figure 3-9. There are ten combustors in parallel, each 140 inches I.D., and 240 feet in overall centerline length which provides 4 seconds gas residence time in the combustor.

The bottom of each combustor is connected to the common housing for heat exchangers and dense-bed. Both the combustor and the housing wall are constructed from 10-1/2 inch thick, Harbinson-Walker castable lining - "Castolast G" and SA-516-70 C.S. shell. The castable lining provides both the resistance to abrasion and insulation. Reinforcement I-beam steel structure on the outside wall of the flat surface together with hollow stays with circulating cooling water inside are designed to hold the housing against the operating pressure. The solids flow cross-sectional area of 20 feet x 140 feet gives 100 lbs/sec-ft^2 mass flux in the dense-bed section. The fluidizing air header is installed in the bottom of the chamber which contains heat exchanger tubes. The finned evaporator tubes

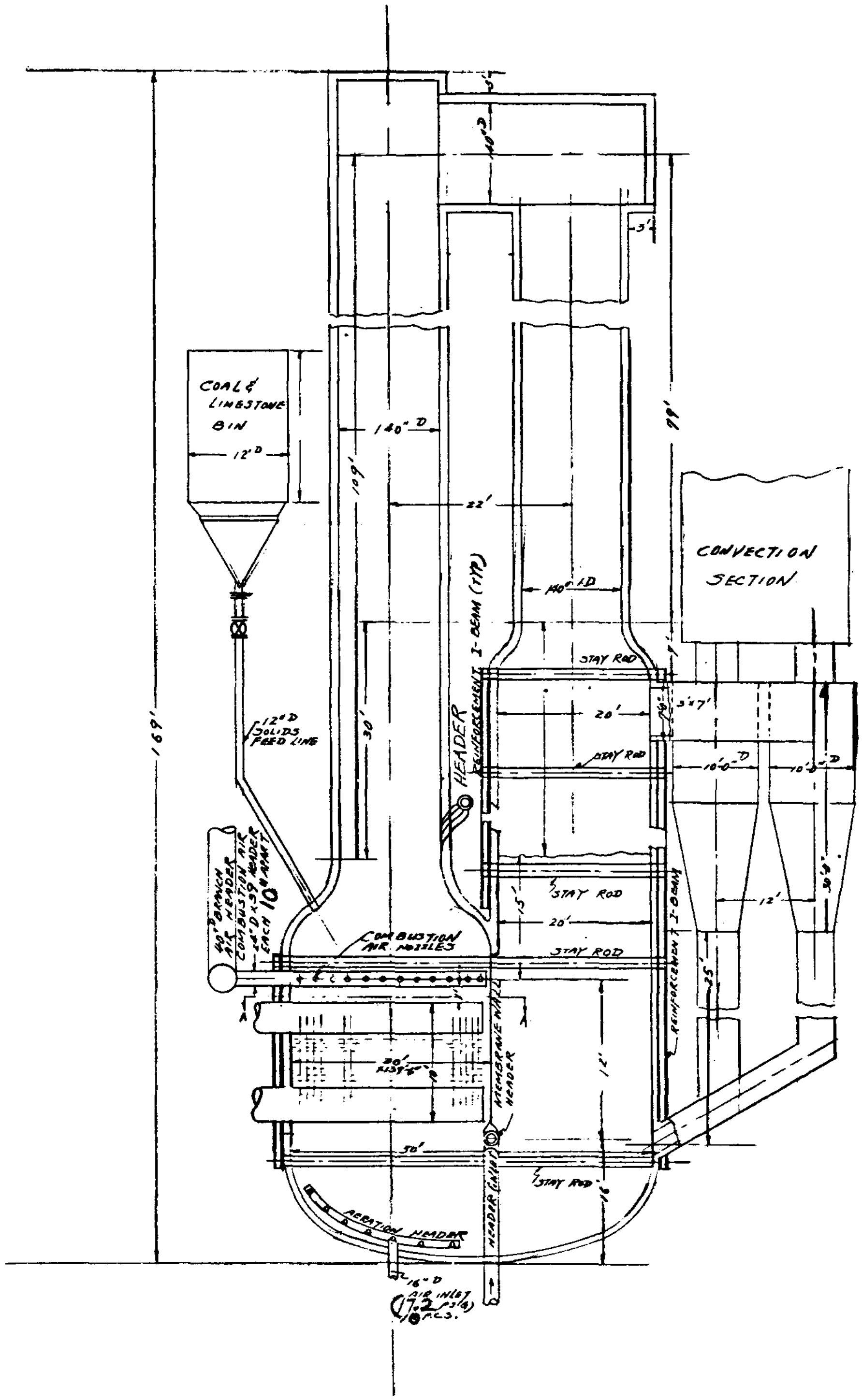


Figure 3-8. - Elevation View - Atmospheric Circulating Bed Boiler.

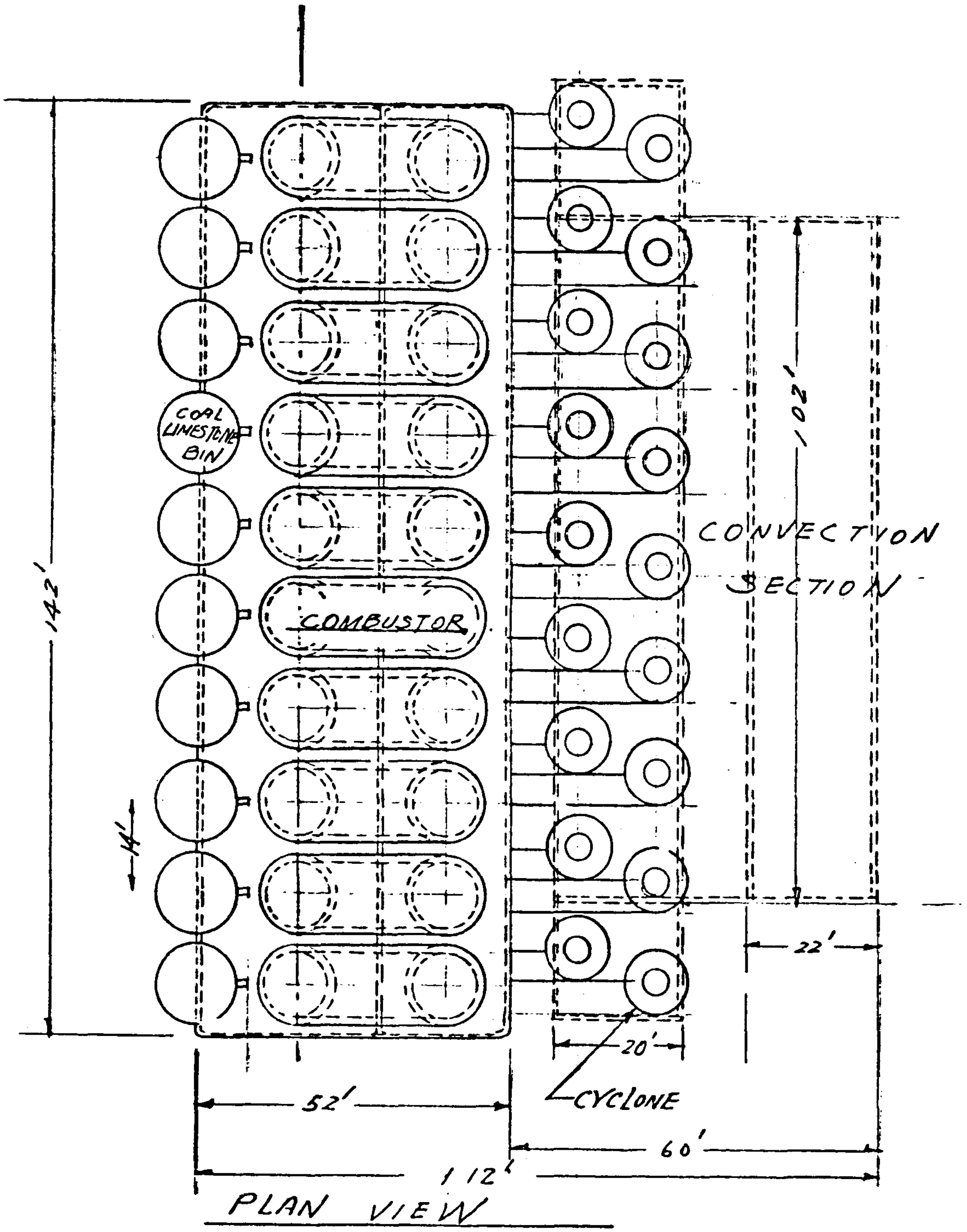


Figure 3-9. - Plan View - Atmospheric Circulating Bed Combustor.

constitute the membrane wall which acts as a partition wall to confine the fluidized space and fluidizing air to flow upwards within this space. The combustion air header supplies air at 8.5 psig. Ten individual headers are distributed over the bed area and air flows are adjusted independently to control the combustion so that the output of the evaporator, reheater and superheater can be adjusted. Coal and limestone are fed into the bed above the air header and are mixed with recycled solids and accelerated upwards into the combustors. The solids-gas mixture turns two 90° turns and flows downwards to the freeboard in the disengager where about half of the solids are separated before entering the cyclone inlet.

The cyclones have an inlet port with a cross sectional area of 21 ft^2. The solids-gas velocity at the inlet to the cyclones is 150 fps. Disengaged solids from cyclones enter the bottom of the bed housing through cyclone dip-legs.

3.4.3 Evaporator, Superheater, and Reheater

As shown in Figure 3-10, the evaporator, superheater, and reheater are installed separately over a section of the bed floor. The cross-sectional area occupied by each heat exchanger is proportional to their heat load. The inlet header for each heat exchanger enters into the bottom housing. The tube coils extend upwards to the outlet header. Table 3-2 presents a description of evaporator, superheater and reheater.

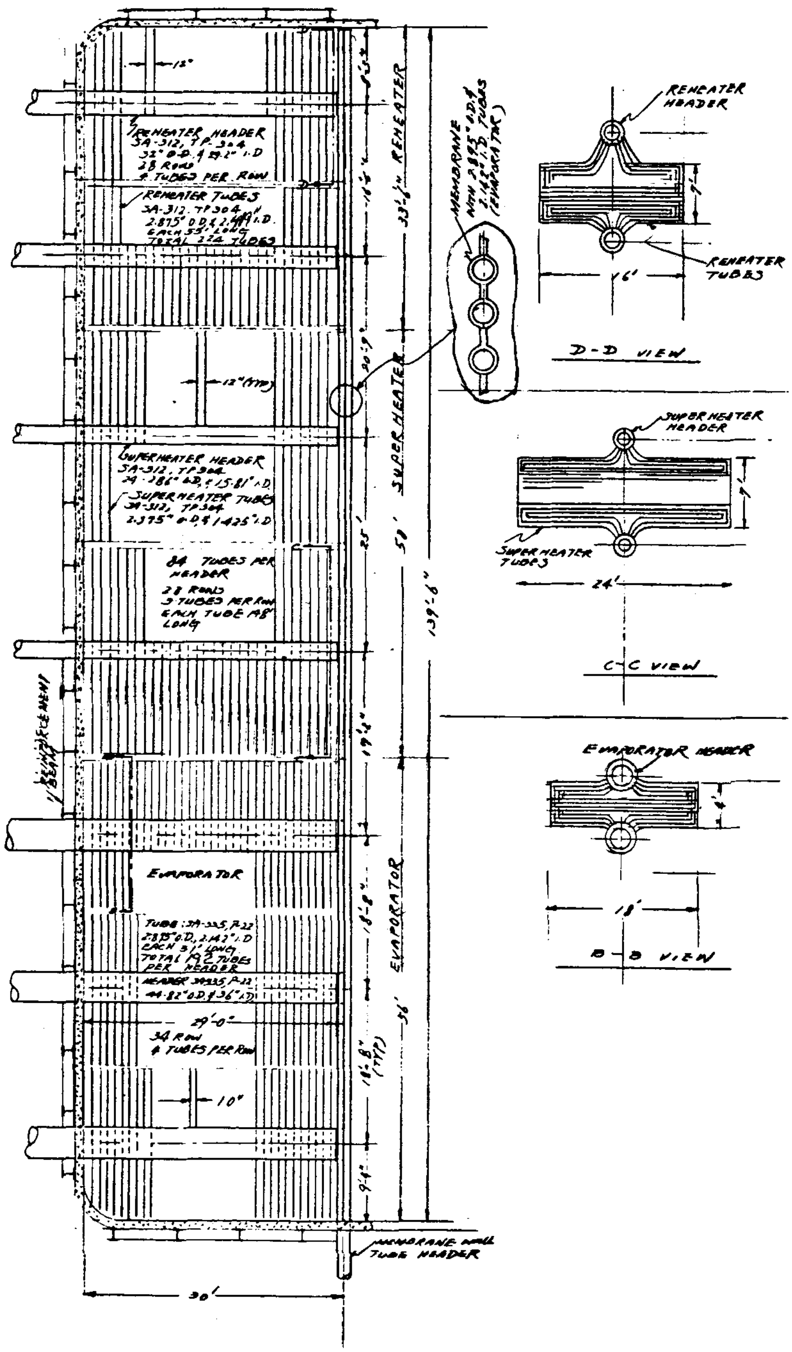


Figure 3-10. - Heat Exchanger Arrangement -
Atmospheric Circulating Bed Combustor.

TABLE 3-2.- DESIGN PARAMETERS FOR THE ATMOSPHERIC CIRCULATING BED COMBUSTOR EVAPORATOR, SUPERHEATER AND REHEATER

	Evaporator	Superheater	Reheater
Heat Load (10^9 Btu/hr)	1.276	1.124	0.755
Design Pressure (psig)	3250	2910	610
Design Temp. (°F)	800	1150	1150
LMTD (°F)	960	750.1	823.3
Bed Temp. (°F)	1650	1650	1650
Steam Temp. (°F)			
Inlet	650	790	625
Outlet	690	1000	1000
Tube Material	SA-335, P-22	SA-312, P-304	SA-312, TP-304
Tube Size (in)			
O.D.	2.875	2.375	2.875
I.D.	2.142	1.425	2.465 pcs
Tube Length	31'	148'	55'
No. Tubes	575 pcs	163 pcs	224 pcs
Overall Heat Transfer Coefficient (BTU/hr ft^2 °F)	------------------100---------------------		
Heat Transfer Surface Area (ft^2)	$1.33(10^4)$	$1.5\ (10^4)$	$9.17(10^4)$
Header Material	SA-335, P-22	SA-312, TP-304	SA-312, TP
Header Size			
O.D.	44.82"	24.29"	32.0"
I.D.	36.0"	15.810"	29.2"

3.4.4 Convection Section

The following table describes the convection section for the combustor.

TABLE 3-3.- DESIGN PARAMETERS FOR ATMOSPHERIC COMBUSTOR CONVECTION SECTION

		Superheater	Economizer		Air Preheater	
		#2	#1	#2	#1	#2
Heat Load (10^6 BTU/hr)		500	385	600	347	200
Tube Side Temp	in	692	485	555	150	450
(°F)	out	790	555	650	450	600
Flue Gas Temp	in	1650	805	1327	530	940
(°F)	out	1330	530	940	280	805
Heat Transfer Surface Area (ft^2)		49,950	221,600	69,500	1,189,760	96,920

3.5 OTHER ATMOSPHERIC CASES

Other atmospheric cases include:

A. Circulating bed boiler with dense beds and 96 high efficiency cyclones.

B. Circulating bed boiler without dense bed and with 96 high efficiency cyclones.

C. Circulating bed boiler with low efficiency cyclones.

Detailed descriptions of these cases are included in the Appendix II.

3.6 SUPERCRITICAL STEAM CYCLE

The steam cycle selected for the fast bed combustor design is a 2520 psia/1000°F/1000°F cycle similar to the one used in the pulverized coal boiler plant study. The reason for this selection was to simplify the comparison of the circulating bed boiler with the pulverized coal boiler.

However, additional steam cycles were evaluated to determine their ability to achieve higher plant efficiency. In pulverized coal fired boilers, an upper limit on steam temperature is imposed by the ash slagging and fouling characteristics, i.e., on superheater tubes. This material is part of the mineral impurities in the coal which, following heating to 2500-3000°F in the flame, form complex salts, some of which are liquid above 1100°F and can be highly corrosive.

In a fluidized bed, the maximum temperature is about 1600°F, which is low enough to inhibit formation of corrodants and to prevent ash melting, so that these particular problems may be avoided. On this basis, it has been suggested that the fluidized bed boiler might open the way to more efficient steam cycles, using higher steam temperatures (5).

The study in Reference 5 discusses the potential values of advanced cycles in fluidized bed technology. One cycle recommended in that study is the efficient, supercritical steam cycle. The sections that follow present examples of plants already built with this cycle and the performance that can be expected with further use of this highly efficient cycle.

3.6.1 Examples of Supercritical Plants

Supercritical plants built in the United States include the 120 MW plant installed at Philo (Ref. 6, 7, 8) and the 325 MW unit at Eddystone (Ref. 9, 10, 11).

The Philo station was designed for 4500 psig, 1150°F initial steam with double reheat at 1050°F and 1000°F. The net heat rate reported for this plant is about 8530 BTU/Kw hr.

The Eddystone plant was designed for supercritical steam condition of 5000 psig, 1200°F at throttle with double reheat both to 1050°F. The heat rate reported for this plant is around 8000 BTU/Kw hr.

It is understood that neither plant can operate at the design steam temperature because of slagging and corrosion problems.

3.6.2 Performance With Supercritical Cycles

These high efficiency steam cycles when used with the circulating bed combustor would improve the plant performance. The plant efficiencies that can be achieved for the atmospheric circulating bed combustor design are summarized in Table 3-4.

TABLE 3-4.- PLANT EFFICIENCIES

Steam Cycle	Reference	Net Plant Efficiency with Circulating Bed Combustor
2520 psia/1000°F/1000°F	(1)	34.90 %
4500 psig/1150°F/1050°F/ 1050°F	(6),(7),(8) (Philo)	37.70
5000 psig/1200°F/1050°F/ 1050°F	(9),(10),(11) (Eddystone)	38.40
5000 psig/1100°F/1050°F/ 1050°F	(5)	38.73

The last steam cycle included in the table is the most efficient double reheat, supercritical cycle recommended in Reference 5. The heat rate for this cycle is 7389 BTU/Kw hr.

The results show that, with the use of high efficiency supercritical steam cycles, the atmospheric circulating bed combustor plants can achieve net efficiencies of 38-39%.

SECTION 4

PRESSURIZED CIRCULATING BED BOILER SYSTEM

4.1 DESIGN RATIONALE

Selection of design variables for the pressurized combustor follows the same rationale as for the atmospheric combustor design. The major considerations are combustion efficiency, SO_2 removal, heat transfer, overall plant efficiency, system pressure drop, combustor size, and plot area. Operating pressure, bed density, gas velocity, and gas residence time are the principal design variables for the combustor.

4.1.1 Operating Pressure

The selection of operating pressure for the pressurized combustor affects the final size of the combustor and auxiliary equipment - the higher the pressure the more compression of the volume gas flow, and the smaller the diameter of the combustor, cyclones, and related equipment. Other considerations that affect this selection are the availability, performance, and cost of air compressors; the air temperature at the discharge of these compressors (whether intermediate cooling is required or not), and the possible effect on combined cycle efficiency.

4.1.2 Bed Density

As mentioned earlier, the selection of bed density affects the heat transfer, SO_2 removal, combustion rate, and pressure drop in the combustor.

While there is a strong incentive in the atmospheric CBB System to minimize the overall pressure drop, this consideration is not as significant in the case of the pressurized plant where the efficieny of the combined gas and steam cycle is much less sensitive to the pressure drop in the combustor. Heat transfer considerations rather than pressure drop become the controlling factor in the selection of bed density for the pressurized combustor. As the combustor design becomes more and more compact with the selection of higher operating pressures, the amount of space available for heating surfaces becomes a limiting parameter. If the design calls for installation of heating surfaces in the combustor, a high circulating bed density becomes necessary to insure a good heat transfer coefficient and minimize heating surface requirements. An advantage of selecting high bed density is to lower the gas residence time requirement for SO_2 removal (Figure 4-1) thereby reducing the required height. Thus, for the pressurized combustor, there are incentives to operate the combustor at high bed density for good SO_2 removal, good heat transfer performance, and convenience in the installation of heating surfaces in the combustor. All of these are obtained without incurring a large penalty in plant efficiency.

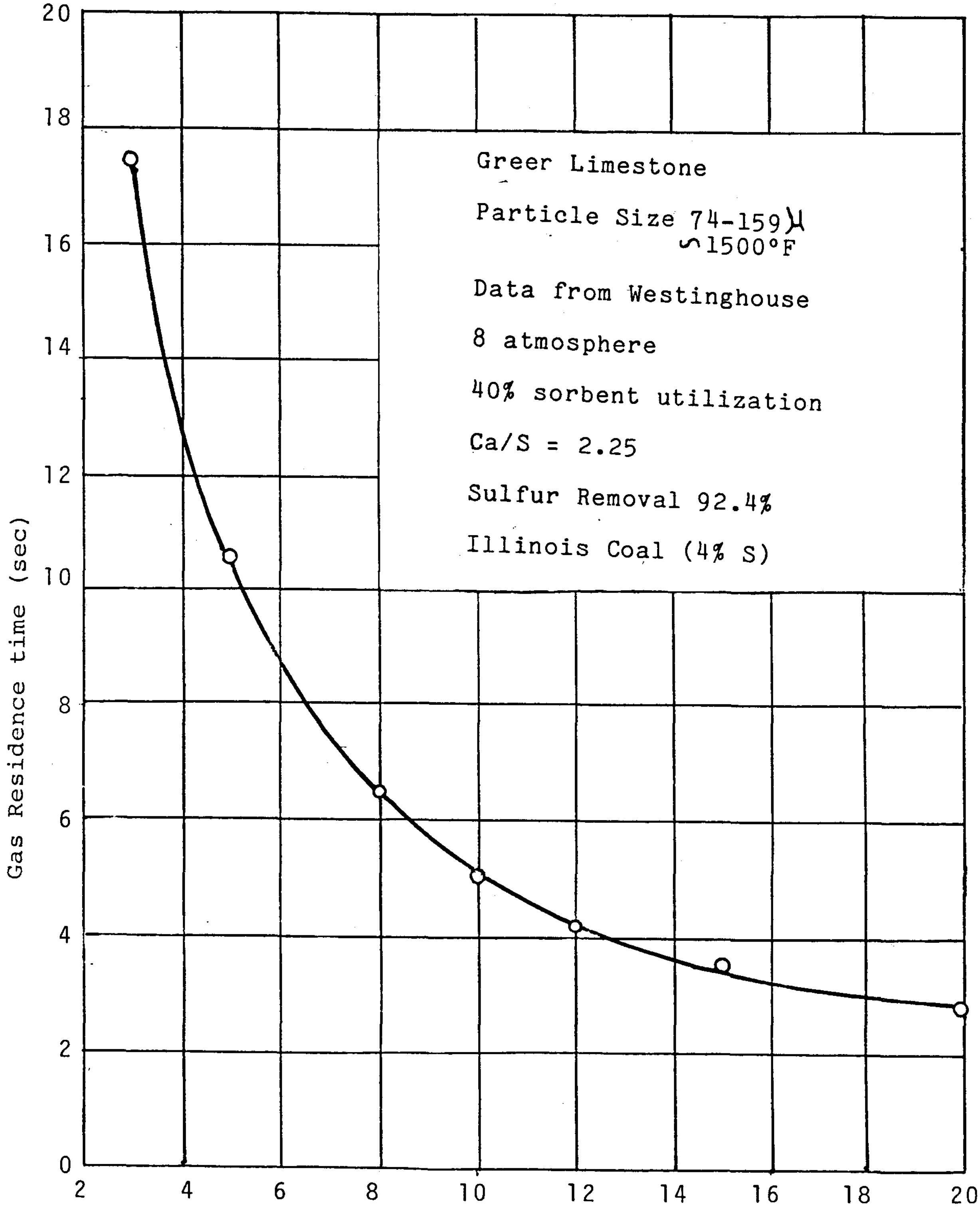


Figure 4-1. - Design Curve - Gas Residence Time vs Circulating Bed Density at Eight Atmospheres.

4.1.3 Gas Velocity

In the pressurized combustor, with the total volume reduced by compression, a compact design can be achieved at low gas velocities. Velocities on the order of 25 to 30 ft/sec can be used which, combined with high bed densities (around 12 lbs/cft in the combustor), achieve good gas/solid mixing and benefits the chemical reactions in the combustor. These lower velocities have the additional advantages of lowering the combustor height and reducing the combustor pressure drop.

4.2 CYCLE STUDY

The purpose of the cycle study is to select a combined gas and steam turbine cycle for the pressurized combustor plant. The bases for selection will be the overall performance of the plant and considerations of equipment technology, particularly for the gas expanders operating with "dirty" flue gases.

The materials in the following sections will include a combined cycle study to choose a power cycle for the plant, a description of the power plant selected, and the final performance obtained for the pressurized combustor plant.

4.2.1 Combined Cycle Study

The combined cycle system selected for this study is the type used for the supercharged boiler shown in Figure 4-2. In this cycle, part of the heat generated by combustion in the pressurized combustor is used to generate steam for the steam turbine cycle and part is used by the gas cycle in the flue gas expanders. The

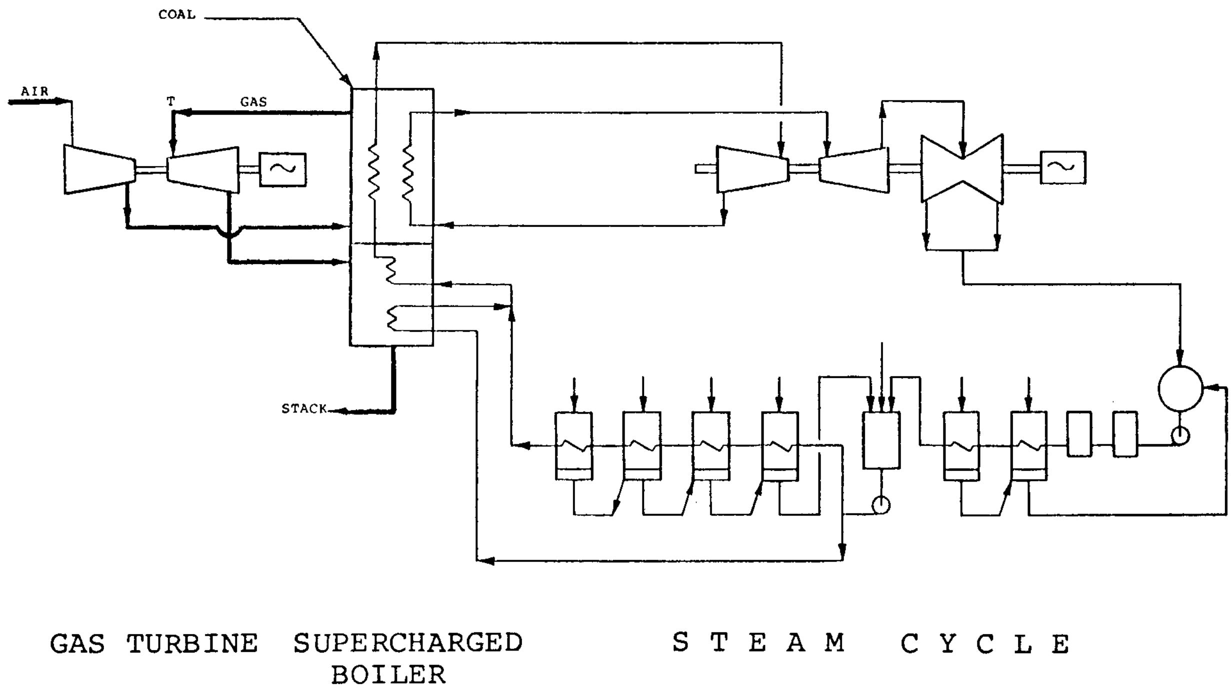


Figure 4-2. - Combined Cycle for Supercharged Boiler.

temperature of the flue gas inlet to the gas expanders and the pressure ratios across the air compressors and gas expanders in the gas cycle determine the split in the amount of heat available to steam and gas cycles.

This distribution of the heat among the gas and steam cycles, as much as the performance of each cycle independently will affect final plant efficiency. This study investigated the performance of the gas, steam, and combined cycles to select the operating pressure and gas expander inlet temperature.

4.2.1.1 Gas cycle.- The performance of the gas cycle depends on the performance of the air compressors and gas expanders used and on the amount of heat supplied to the cycle to be converted to power.

Figure 4-3 shows the compressor work as a function of compression ratio and the expander work as a function of compression ratio and the flue gas temperature to the expanders. The following conditions were used for the study:

- Air compressors
  - inlet: ambient air at 14.7 psia, 60°F
  - efficiency: 82% polytropic efficiency
- Gas Expanders
  - inlet: compressor discharge pressure less 15 psi for combustor and system pressure losses
  - outlet: 15 psia (to low pressure convection section)
  - efficiency: 82% polytropic efficiency

The graph shows that, for compressor discharge pressures up to 10 atmospheres, an expander inlet

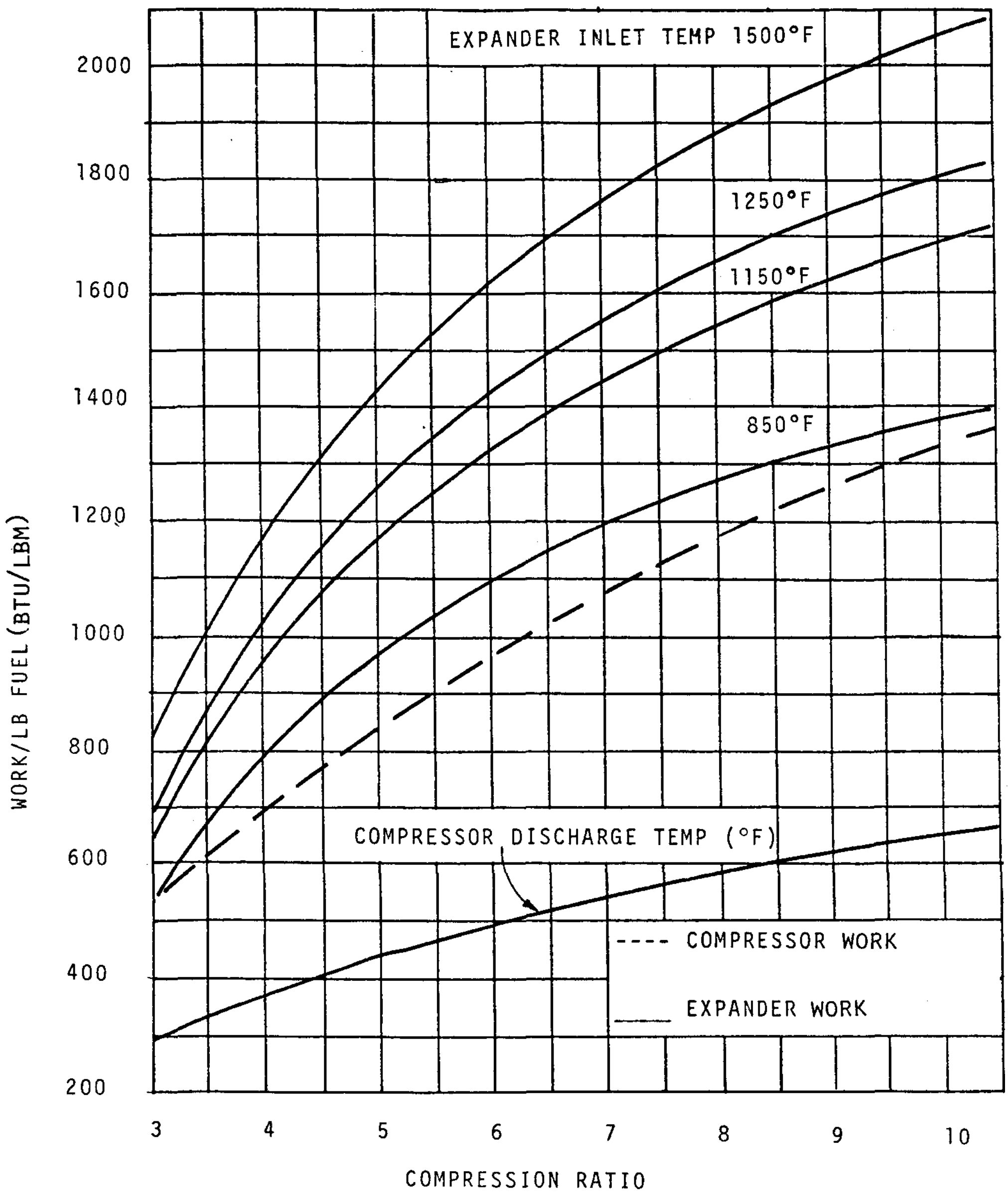


Figure 4-3. - Compressor Work and Expander Work vs Compression Ratio and Expander Inlet Temperature.

temperature slightly less than 850°F would be needed to balance the power requirements of the compressor. Above this temperature, the gas cycle generates net electrical power.

The net heat absorbed by the open gas cycle to be converted to power, is shown in Figure 4-4 as a function of compressor discharge pressure (expressed as compression ratio), and gas temperature to the expanders. This net heat is defined as the heat delivered to the gas expanders (as sensible heat in the hot flue gases) less the power consumption of the air compressors (expressed as sensible heat increase of air between the compressor inlet and discharge). The graph shows that for compression ratios between 4 and 10, as the temperature of the gas to the expanders increases from 850°F to 1500°F, the fraction of the total heat generated from burning coal in the combustor that can be converted to power by the gas cycle increases from 1% to 7%.

These figures are important in terms of the efficiency of conversion from heat to power by the gas cycle. The 1% to 7% figures represent the net percentage heat supplied by the combustor to the gas cycle. This net heat is converted to power directly at close to 100% efficiency (minus generator losses and leakages in the system). The gas cycle efficiency is, therefore, much higher than the steam cycle counterpart, and it provides incentive to supply as much heat from the combustor as possible to the gas cycle, primarily by operating the gas expander at as high a temperature as possible.

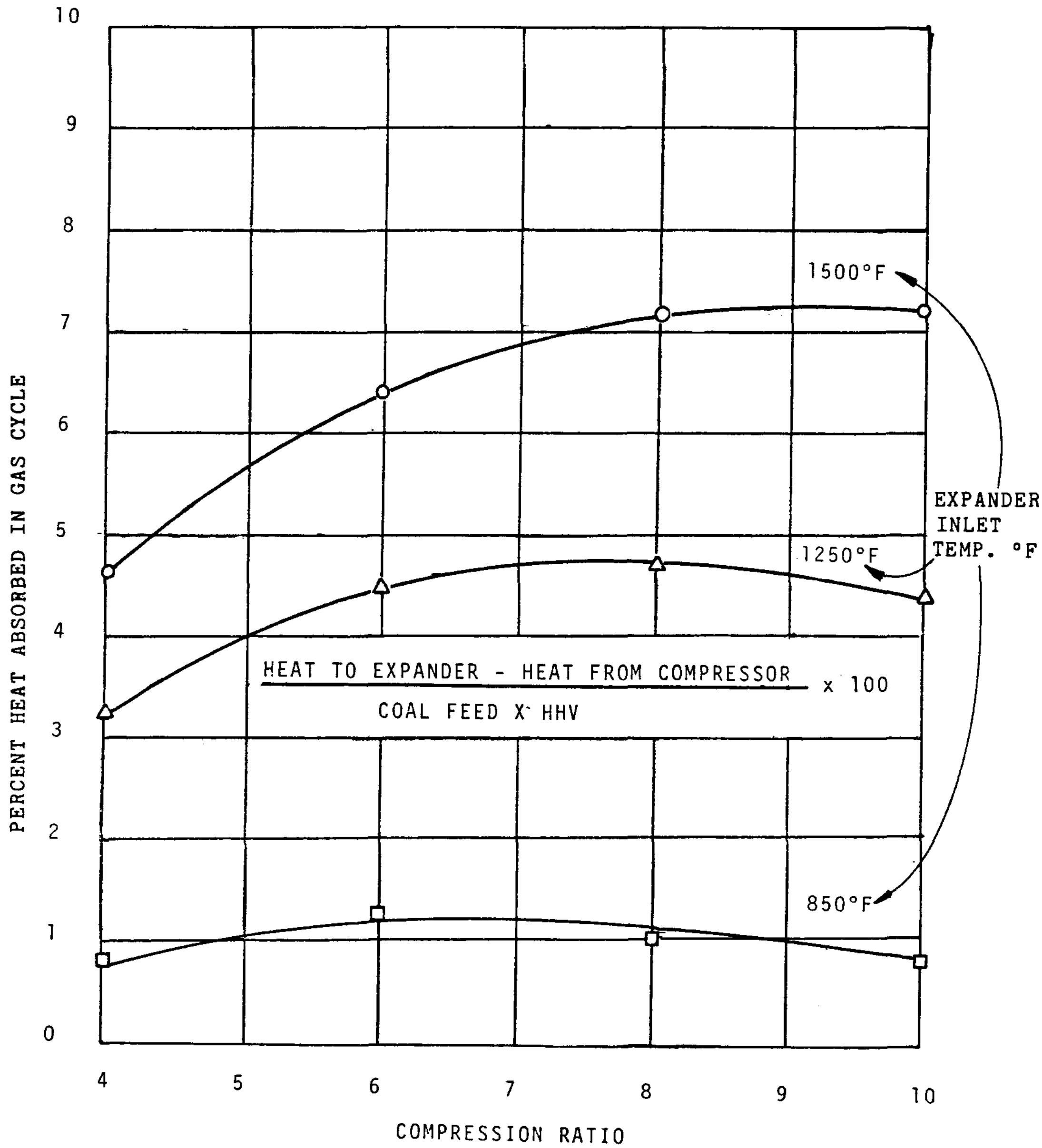


Figure 4-4. - Net Heat Absorbed by Gas Cycle vs Compression Ratio and Expander Inlet Temperature.

In summary, the gas cycle, for the conditions used in this pressurized combustor design, operating at compressor pressures up to 10 atmospheres and at gas expander temperatures from 850°F to 1500°F, can use from 1% to a maximum of 7% of the heat generated in the combustor to produce power at a rate of efficiency close to 100%.

4.2.1.2 Steam cycle.- As discussed earlier, the steam cycle used in this study will be a 2500 psig/1000°F/1000°F cycle similar to the ones used in the pulverized coal plant and the atmospheric combustor plant. The power production of the steam cycle will depend on the amount of heat from the combustor available to the cycle for steam generation and for feedwater heating. This heat includes high temperature level heat from the combustor bed and hot flue gases upstream of the gas expanders, and it includes lower temperature level waste heat in the gases after expansion through the gas expanders.

A. Heat supplied to steam cycle - Figure 4-5 shows the total heat available to the steam cycle at different operating conditions of the gas cycle. At an 850°F expander inlet temperature, most of the heat from combustion of coal in the combustor (85% to 87%) will be used by the steam turbine cycle (the combustor efficiency is around 88%). As this temperature increases and hotter gases are delivered to the gas cycle, the heat remaining to the steam cycle will drop. At 1500°F expander inlet temperature, with the gas cycle using up to 7% of the total heat input to the combustor (as discussed in gas cycle), 77% to 78% of the total heat in the combustor will be delivered to the steam cycle for power production.

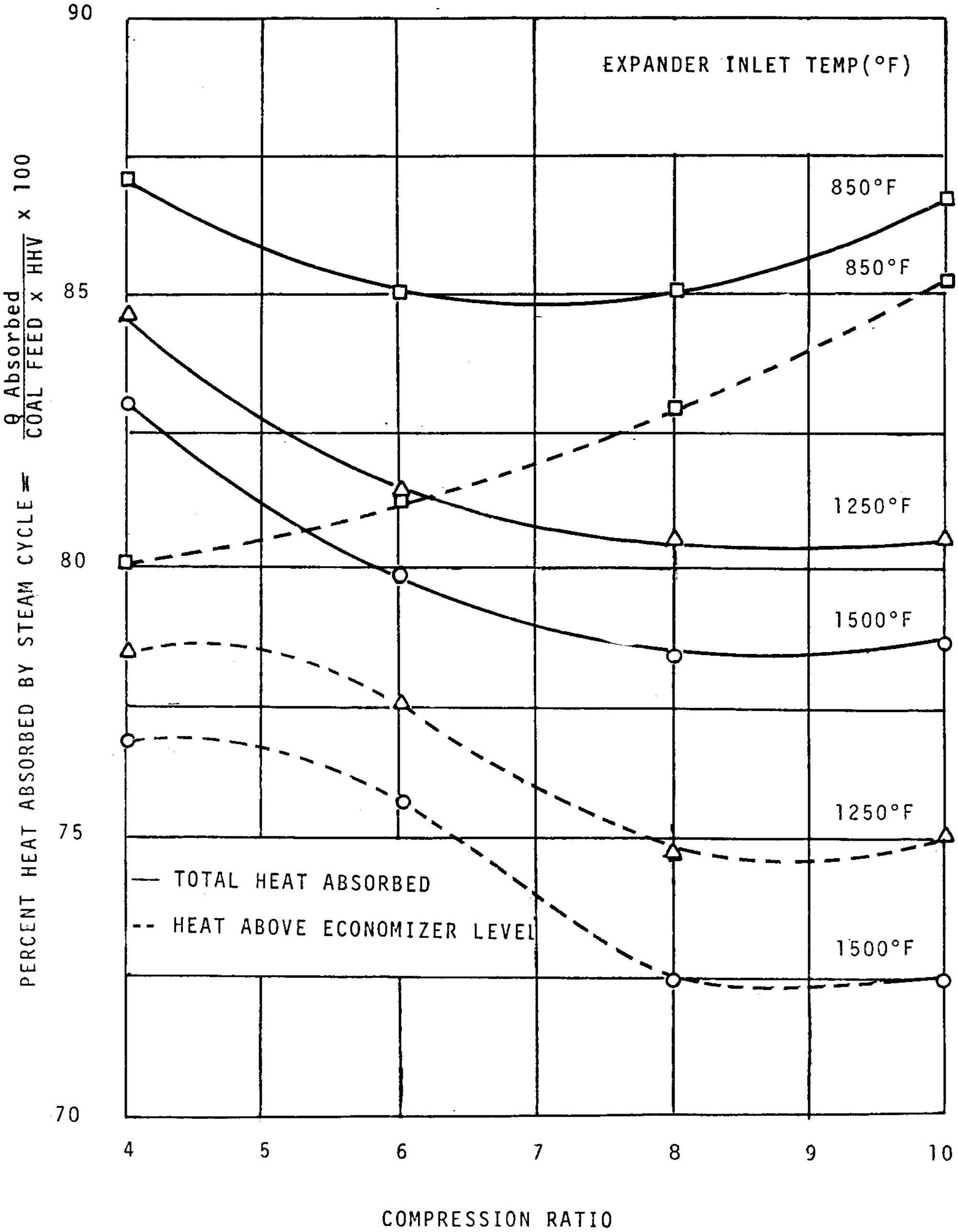


Figure 4-5. - Heat Absorbed by Steam Cycle vs Compression Ratio and Expander Inlet Temperature.

Figure 4-5 also shows the amount of heat available to the steam cycle at high temperature level that is suitable for steam generation, superheating, and reheating. The remaining heat supply will be lower temperature level heat that can be used in economizers to preheat feedwater.

B. <u>Power generation from the steam cycle</u> - The power produced by the steam turbine cycle can be determined from the amount of heat that is consumed in the steam cycle. The procedure is as follows:

1. The high temperature level heat can be used directly by the steam cycle to produce power at steam cycle efficiency. This efficiency depends on the steam cycle used, specifically it depends on the turbine throttle conditions, the extraction heater arrangement, the condensor pressure, the turbine performance and the leakage. For a 2500 psig/1000°F/1000°F steam cycle with seven-stage extraction heating, of the type considered thus far in this study, a typical cycle eficiency is 42%. Using this number, all the heat absorbed by the steam cycle at high temperature level, shown in Figure 4-4, will be converted to electrical power at 42% conversion efficiency.

2. The remaining heat at lower temperature level will be used in economizers to preheat feedwater going to the steam evaporator. This displaces extraction steam from the turbine in the extraction heater arrangement used by the cycle. The steam can continue to expand through the turbine to generate additional power. Therefore, the low level temperature heat used in the economizers can generate

additional power for the steam cycle through savings in the turbine extraction steam. The "efficiency" of power production with this economizer heat is primarily a function of the temperature level of the heat, and it is discussed in References 12 and 13. Figure 4-6 shows this efficiency plotted against the feedwater temperature level. The efficiency varies from less than 10% for steam saved at the first extraction heater level (lowest pressure heater), to about 38% for the higher level steam feeding the seventh stage extraction heater.

Based on these values, the heat at low level supplied to the steam cycle for preheating feedwater, can generate power for the cycle at efficiencies between 10% and 40%.

The final power production of the steam turbine cycle will consist of all the contributions made by having heat absorbed into the cycle at different temperature levels.

4.2.1.3 Combined cycle.- Using the results obtained for the gas and steam cycles, the overall performance of the combined gas and steam cycle as a function of compression ratio and gas expander temperature is shown in Figure 4-7. The combined cycle efficiency (gross) varies from 36.5% to 39.5% as the gas expander temperature is increased from 850°F to 1500°F. The effect of pressure is not as pronounced, at least up to the 10 atmospheres considered in the study. As discussed earlier, the improvement in the combined cycle efficiency at high expander inlet temperature results from the better distribution and utilization of the heat generated in

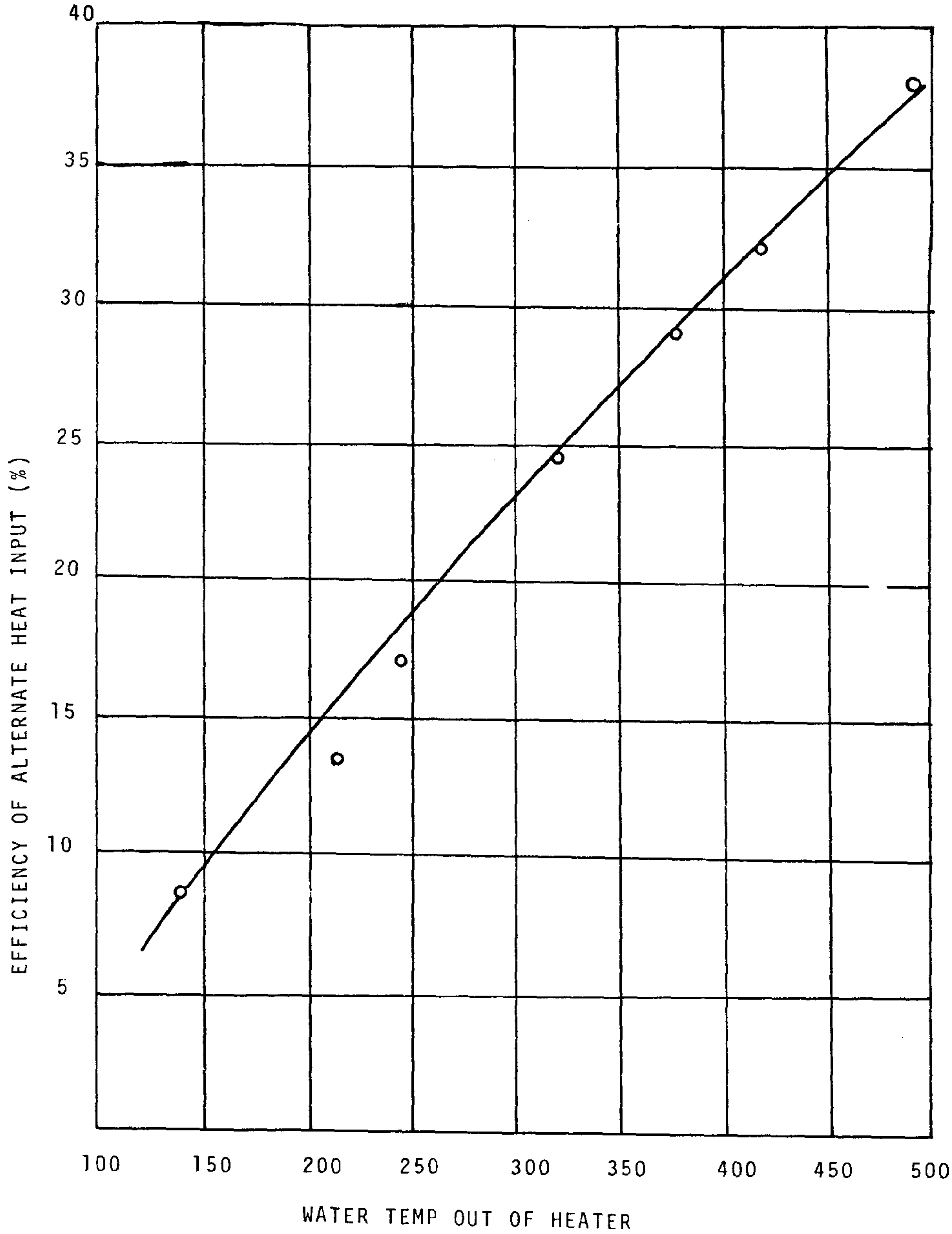


Figure 4-6. - Conversion Efficiency of Low Temperature Level Heat When Supplied as Alternate Heat Input to Economizer.

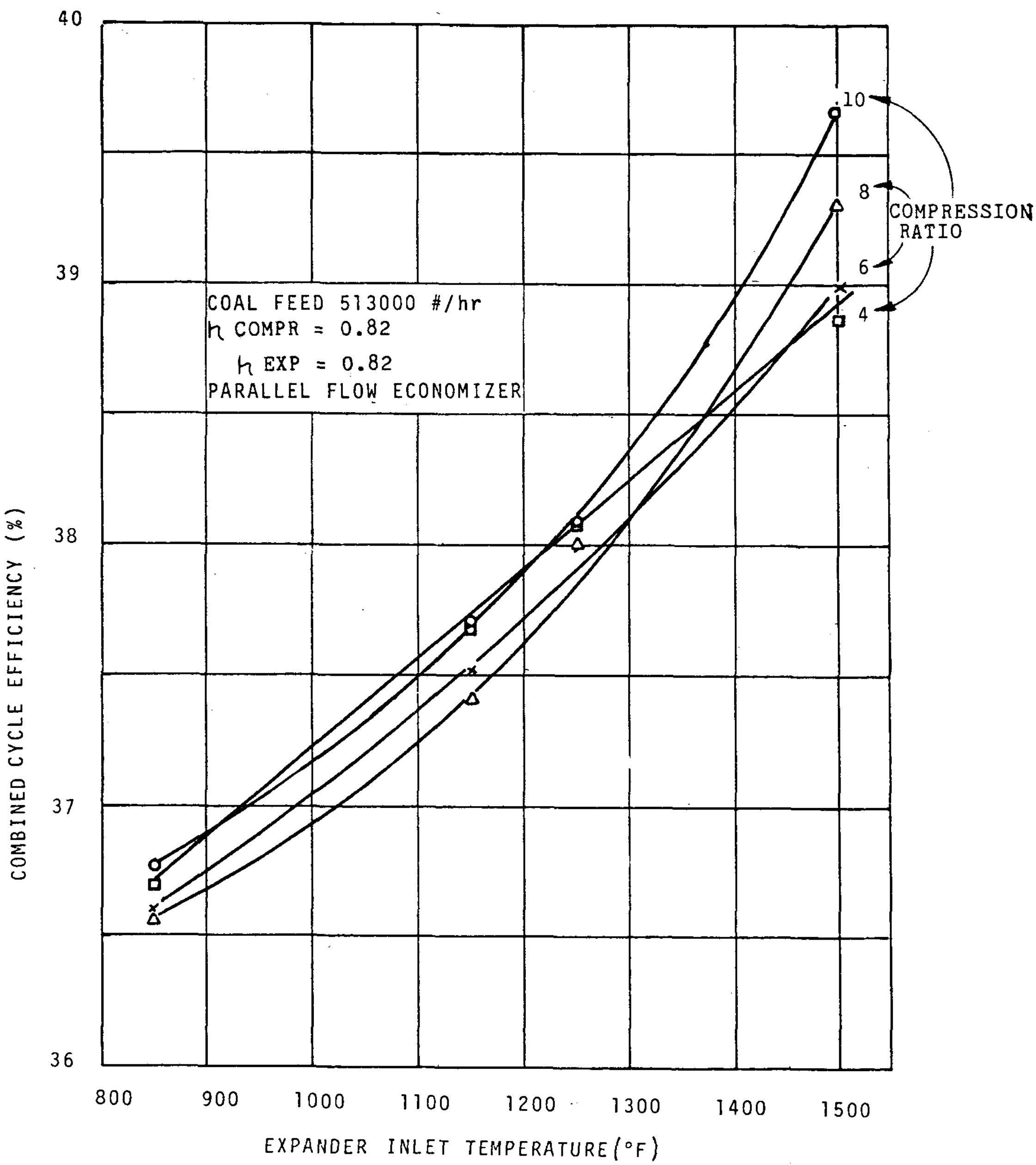


Figure 4-7. - Combined Cycle Efficiency vs Expander Inlet Temperature and Compression Ratio.

the combustor. At 1500°F expander inlet temperature, 7% of the total heat is used by the gas cycle for power production, but only about 1% is used at an 850°F expander temperature. Because of the higher thermodynamic efficiency of the gas cycle (approaching 100%) compared to the steam cycle (42% or less), this fraction of heat will be converted to power in the gas cycle much more efficiently than in the steam cycle operating at heavy condenser thermal losses.

4.3 PRESSURIZED COMBUSTOR POWER CYCLE

4.3.1 Power Cycle

The combined steam and gas turbine cycle selected for the pressurized combustor plant is shown in Figure 4-8. The steam cycle is a 2500 psig/1000°F/1000°F with seven stages of heat extraction similar to the one used for the atmospheric combustor plant. The difference is the deaerator pressure level, which is reduced to 25 psia (60 psia in the atmospheric plant), to permit a convenient arrangement for a low temperature waste heat economizer installed in parallel with the remaining extraction heaters (1st to 4th stage heaters as shown in the diagram). A steam balance for the cycle is included in Figure 4-9.

The gas cycle operates at 8-atmospheres compressor discharge pressure and 1250°F expander inlet temperature. The bases for these selections are as follows:

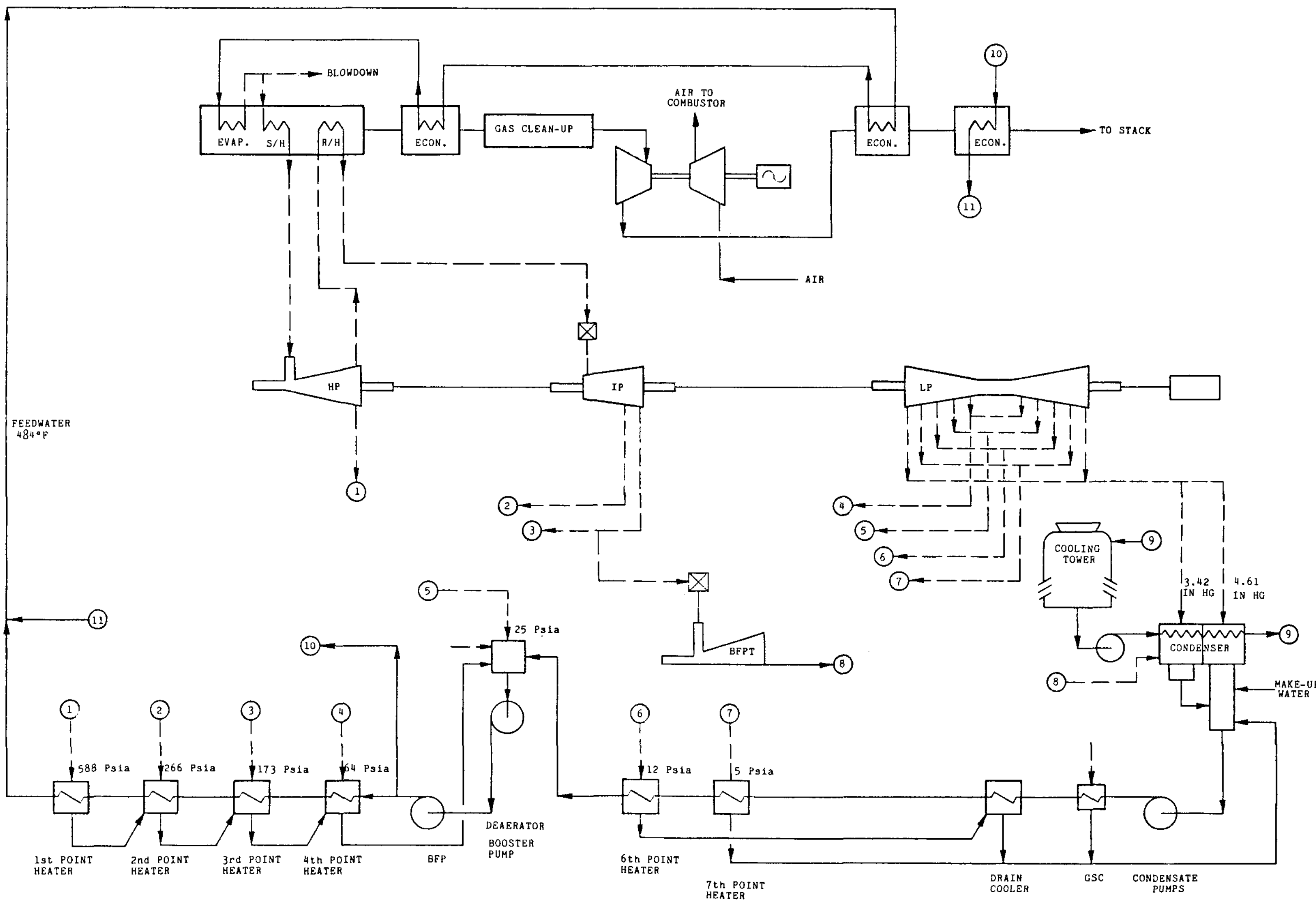


Figure 4-8. - Flow Diagram Combined Cycle for Pressurized Circulating Bed Boiler.

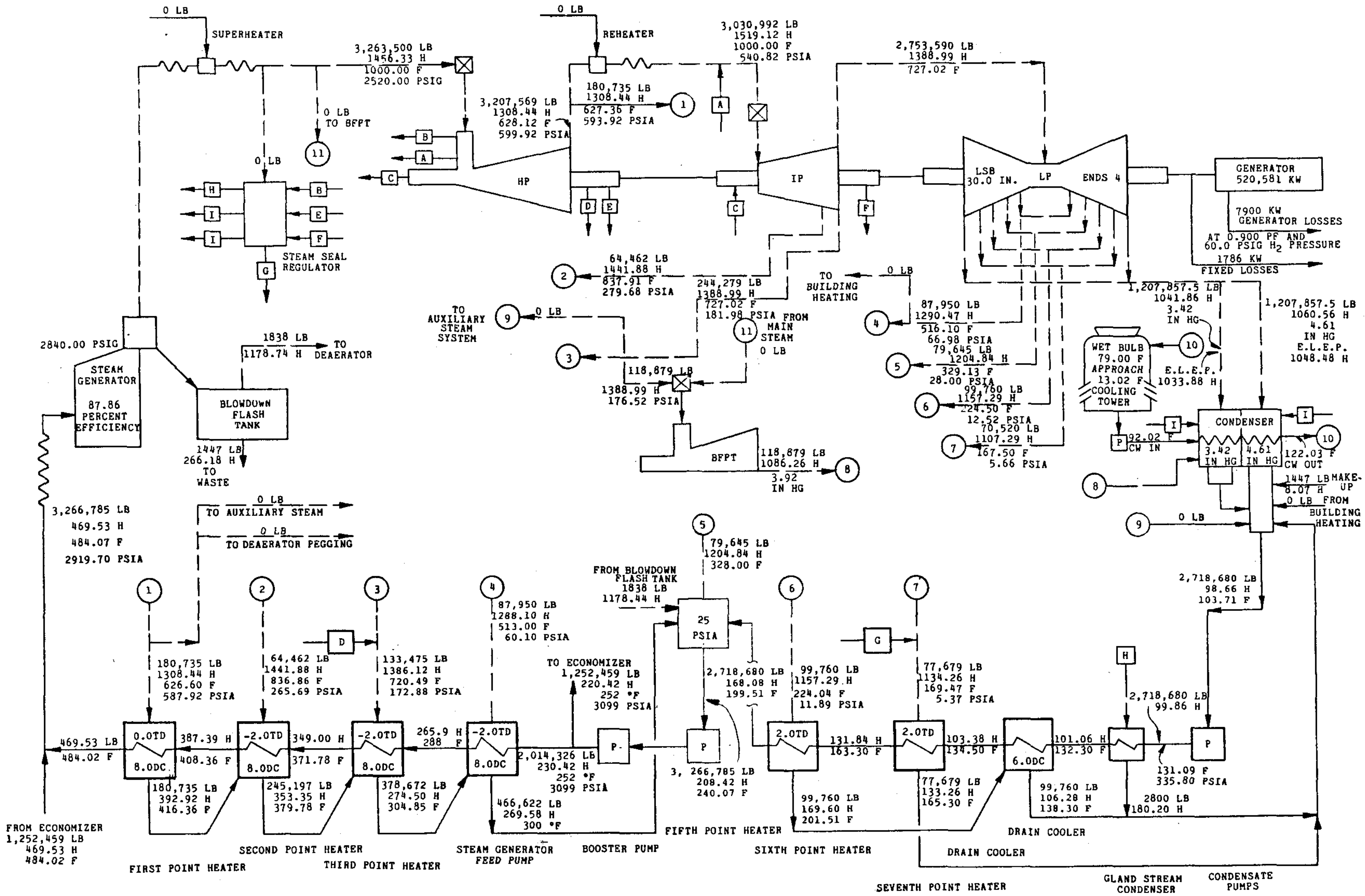


Figure 4-9. - Steam Balance for Pressurized Circulating Bed Boiler Power Plant.

A. <u>Operating Pressure</u> - A compressor discharge pressure of 8 atmospheres was selected based on the considerations of combined cycle performance, size reduction in the design of the combustor and supporting equipment, and on the air temperature at the discharge of the compressor. Figure 4-3 shows this compressor discharge temperature as a function of compression ratio. Without intermediate cooling, the compressor discharge may reach and exceed 600°F as air is compressed to 8 atmospheres and above. This high air temperature may or may not create problems in the design of the coal injection system to the combustor using this primary air. The selection of 8 atmospheres for the compressor discharge pressure would limit the temperature to below 600°F without requiring intermediate compressor stage cooling.

B. <u>Expander Inlet Temperature</u> - The primary limitation in selecting the expander inlet temperature is in the technology of the gas expanders required to operate at high solid particulate loadings. High temperatures combined with the heavy solid loadings may create excessive damage to the machines. Pullman Kellogg's experience in catalytic cracking, with similar conditions of solid loadings, has shown satisfactory gas expander operation at temperatures less than 1250°F. This temperature is in compliance with the recommendations provided by the gas expander manufacturers. As a consequence, the selection of a 1250°F temperature for gas expander operation in this pressurized combustor plant is based on available and demonstrated technology for gas expander machines operating at heavy solid loadings. The overall combined cycle efficiency for this temperature, based on Figure 4-7, is 37.8% gross.

Integration of this gas cycle, operating at 8 atmospheres and 1250°F gas expander inlet, into the overall power plant is shown in Figure 4-8.

4.3.2 Heat Load Arrangement for the Pressurized Combustor

The heat load arrangement for the pressurized circulating bed combustor design is shown by the T-Q diagram in Figure 4-10. The combustor bed, at 1650°F, will distribute heat to a steam evaporator, a superheater, and a reheater. After a first stage solid disengagement in the cylcone units, the heat in the flue gas is recovered in a high pressure convection section (upstream of the gas expanders) to preheat feedwater. The gas exhaust of this economizer, at 1270°F, will go through a second stage solid cleaning before being delivered to the gas expanders. After expansion through the gas expanders, the remaining waste heat in the flue gas is recovered to preheat feedwater in two low pressure economizers. The final flue-gas stack temperature is 300°F.

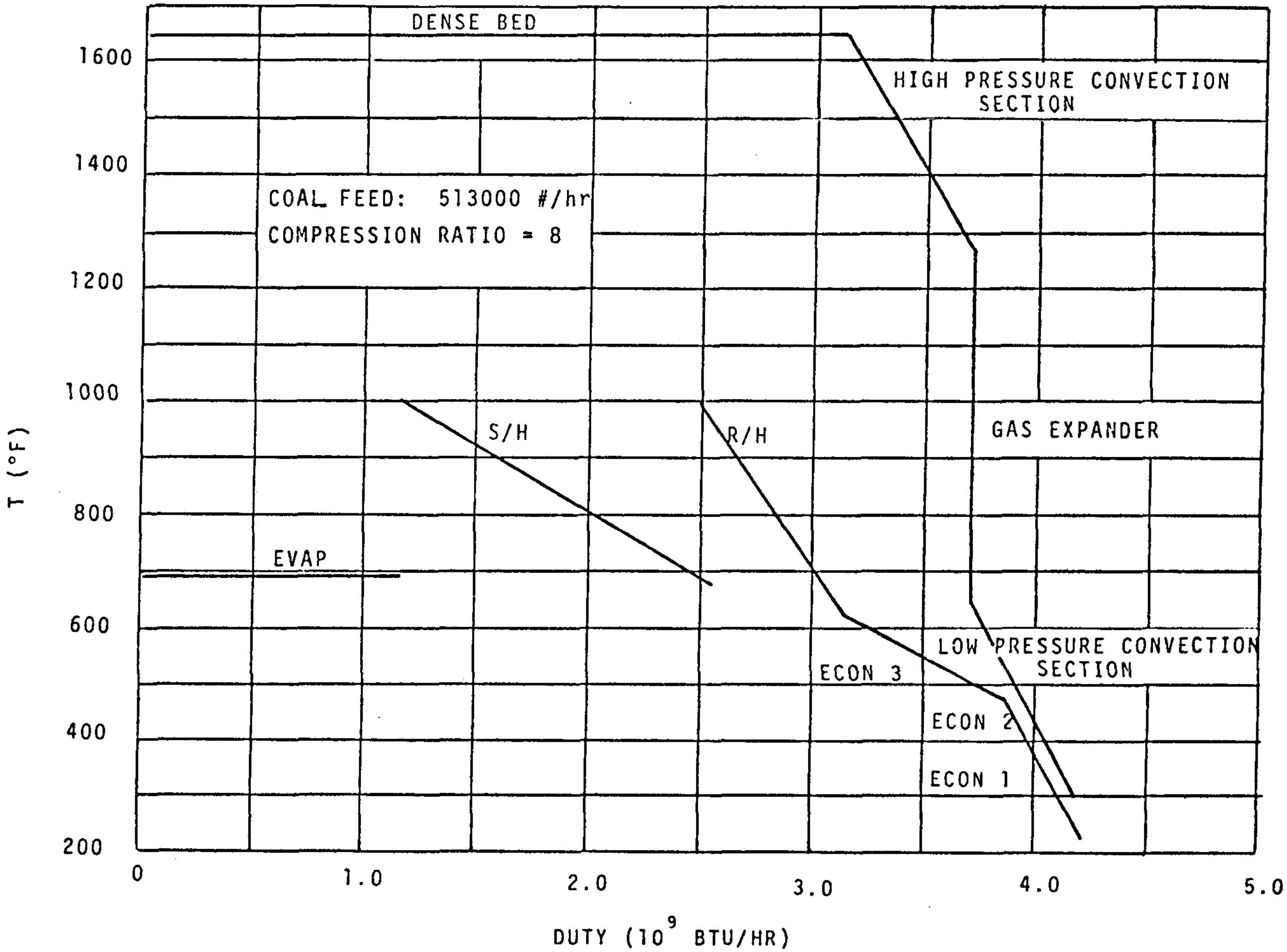


Figure 4-10. - T-Q Diagram Pressurized Circulating Bed Combustor.

4.3.3 Power Plant Efficiency

Overall performance of the pressurized combustor plant is summarized in Table 4-1. Net Plant efficiency is 36.71%

4.4 COMBUSTOR DESIGN

The design conditions presented in Table 4-2 are incorporated in the design concepts described next and are consistent with the plant output and efficiency shown in Section 4.2.2.3.

Two pressurized designs were investigated and reported as Cases 1 and 2 in the following sections.

Case 1 has a dense fluidized bed which contains most of the heat transfer surface. Case 2 does not have a dense bed and the heat transfer surface is contained in the dilute phase riser.

TABLE 4-1.- NET PLANT EFFICIENCY PRESSURIZED CIRCULATING BED COMBUSTOR POWER PLANT

	Pressurized Circulating Bed Combustor Without Dense Bed	Pressurized Circulating Bed Combustor With Dense Bed	Pulverized Coal Plant
Heat input from coal (HHV) BTU/hr	5.18×10^9	5.18×10^9	5.32×10^9
Gross plant output (MW)	575.2	575.2	570.17
Aux. power consumption (MW)			
steam plant	7.70	7.70	8.49
fans	2.50	9.28	15.74
scrubbers			11.40
others	7.65	7.65	8.62
Total	17.85	24.63	44.25
Net plant output (MW)	557.35	550.57	525.9
Net plant efficiency %	36.71	36.27	33.74

The difference between the two circulating bed cases shown in the table, and discussed in more detail in the combustor design sections, is in the additional air consumption needed to fluidize the dense bed.

The final efficiency for the pressurized combustor plant is between 36.27 and 36.71% (net), compared to 33.74% for the pulverized coal plant.

TABLE 4-2.- COMBUSTOR DESIGN CONDITIONS

Parameter		Design Condition
Coal Feed	:	513,000 lbs/hr
Air	:	4,810,950 lbs/hr
Limestone	:	137,000 lbs/hr
Flue Gas	:	5,254,700 lbs/hr
Bed Temperature	:	1650°F
Gas Residence Time	:	4 sec
Combustor Solid Density	:	12 lbs/ft^3
Total System Pressure Drop	:	15 psi
Combustor Operating Pressure	:	8 atm

4.4.1 Case 1 - Pressurized Circulating Bed Combustor with Dense Bed

As shown in Figure 4-11, this system consists of four circulating bed combustors, surrounding a dense fluidized-bed. The combustors have a 12-foot I.D., and they are 119 feet high. The dense-bed vessel has a 30-foot I.D. at the dense bed, and it has a 50-foot I.D. at the freeboard section. There are four clusters of coal and limestone lock hoppers, each consisting of three hoppers. Each cluster of lock-hoppers feeds coal and limestone to a combustor. The lock-hoppers are 10 feet in I.D. and 20 feet high, and they are equipped with 12-inch rotary valves connected to 12-inch feed pipes.

The end of each feed pipe is connected to a common air supply where air from the compressor entrains solids into the combustor. Recycled solids flow into the combustor from the dense fluidized bed through a 10-foot I.D. duct. There are aeration nozzles on the bottom of the combustor and the solid return duct to enhance the solids flow. Combustion is initiated, and the returned solids, which are at 1650°F, flow upward through the combustor and come into contact with the fresh coal feed. Evaporator tubes are installed in the combustor wall. As the solids-gas flows upward, part of the generated heat is transferred to the evaporator tubes to produce steam. The solids-gas mixture turns 180°F at the top of the combustor and then enters the freeboard space of the dense bed. Approximately half of the solids loading separates from the flue gas before entering the cyclones and the remaining solids are separated from the gas in the cyclones. Twenty

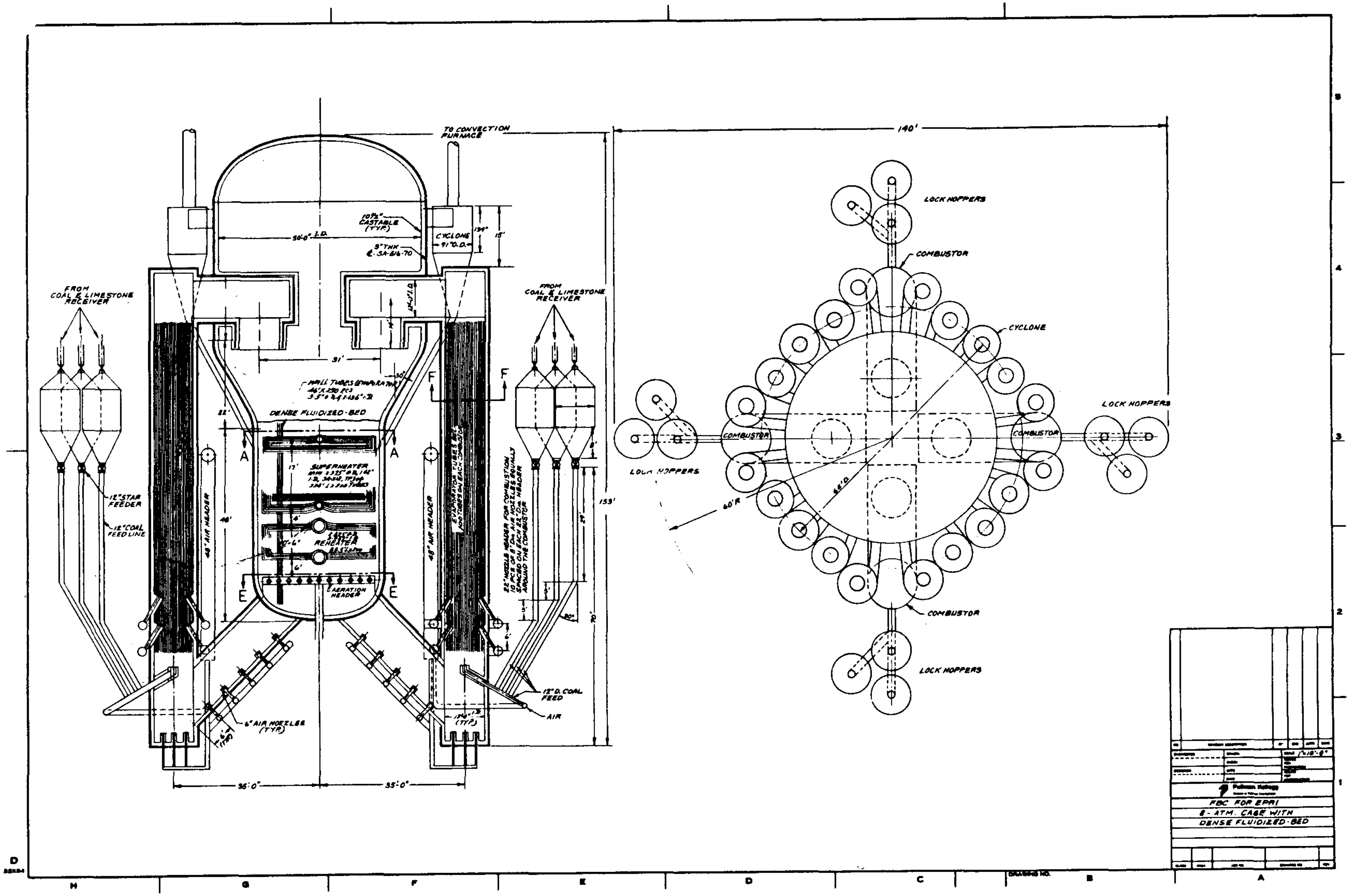


Figure 4-11. - Pressurized Circulating Bed Boiler with Dense Bed.

high efficiency cyclones are required. The separated solids are discharged into the dense bed.

The flue gas leaves the vessel and flows to the pressurized convection section. The pressurized convection section sits on top of the dense-bed vessel. The flue gas from the convection section enters gas cleaning equipment and then is delivered to the gas expanders. The flue gas exhaust from the gas expanders enters a low pressure convection section in which economizers are installed. After the economizer, the flue gas enters the baghouse (or electrostatic precipitator) prior to discharge from the stack.

The solids which were separated and discharged into the dense bed, flow downward in the dense bed. As the solids pass downward, heat is transferred to the evaporator (wall tubes), as well as the reheater and the superheater which are located in the dense bed itself. Aeration nozzles at the bottom of the dense bed fluidizes the solids. The solids flowing from the dense bed are then divided into four streams. Each stream flows into one of the four combustors.

Combustion air is supplied to the combustor through air nozzles situated around the combustor that draw air from two common ring-type headers. The inlet air is at a pressure of 8 atmospheres. All solids feed hoppers are situated at sufficient elevation to prevent backflow of solids-gas from the combustor. Also, the air header is raised to a height that will prevent plugging by solids due to backflow.

The gas velocity in the dense-bed is 0.5 to 1 fps and 5 fps in the freeboard space. Inlet velocity into the cyclones is 72 fps.

The calculated system pressure drop distribution is:

• Acceleration loss	4.5 psi
• Static head loss in combustor	9.9 psi
• Total	14.4 psi

4.4.1.1 Combustor.- The combustor size is determined by the space requirement for installing the required heat exchanger tubes. The following table presents the design criteria.

TABLE 4-3.- DENSE BED COMBUSTOR DESIGN DATA

Parameter	Design Data
No. of Combustor	4
Inside Diameter	12'-0"
Height	119'-0"
Length	126'-0" (effective length)
Solids Density	12 lbs/ft^3
Gas Velocity	21 fps (at combustor inlet)
Actual Gas Residence Time	6.0 sec
Required Gas Residence Time	4.0 sec (dictated by SO_2 removal
Flue Gas Flowrate	5,254,700 lbs/hr
Inlet Air Pressure	8 atm
Combustor Operating Temp.	1,650°F
Material of Construction;	
Shell	7/8" t, SA 517-70 C.S. Plate clad with 316 SS
Lining	10-1/2" Harbinson-Walker Castolast "G"

TABLE 4-4.- HEAT EXCHANGERS

Item		Evaporator	Superheater	Reheater
Heat Load (10^9 Btu/hr)		1.156	1.328	0.656
Design Pressure (psig)		3250	2910	610
Design Temp (°F)		800	1150	1150
Bed Temp (°F)		1650	1650	1650
Steam Temp (°F)	Inlet	690	690	625
	Outlet	690	1000	1000
Tube Material		SA-335,P-22	SA-312,TP-304	SA-312,TP-304
Tube Size (in.)	O.D.	3.50	2.375	2.875
	I.D.	2.636	1.425	2.469
Tube Length x No. of Tube		80'x400 46'x250	224'x200	88.5'x200
Heat Transfer Surface Area (Ft^2)		20,070	27,840	13,280
Header Material		SA-335,P-22	SA-312,TP-304	SA-312,TP-304
Steam Flow Rate (10^6) lb/hr		3.267	3.264	3.027

4.4.1.2 Heat exchanger.- The evaporator section consists of 400 tubes that are 80 feet long and 250 tubes that are 46 feet long. 100 lengths of 80-foot tube are installed in each combustor inside wall. The 250 lengths of 46-foot tube are installed in the dense bed around the wall. These tubes provide the necessary heat transfer surface for the evaporator. The superheater consists of 200 tubes, each 224 feet long, installed in the dense bed. The reheater is installed in the dense bed below the superheater (See Table 4-4).

4.4.1.3 Convection section.- The convection section consists of three economizers, called Economizer 1, 2 and 3. Economizer 1 is shown in Figure 4-12, and Economizers 2 and 3 are shown in Figure 4-13. Economizer 1 is situated upstream of the gas expanders, and Economizers 2 and 3 are downstream of the expanders. The heat loads and heating surfaces are listed in Table 4-5.

TABLE 4-5.- CONVECTION SECTION

Item	Economizer 1	Economizer 2	Economizer 3
Heat Load (10^9) Btu/hr	0.3245	0.163	0.56
Heat Transfer Surface Area (Ft^2)	372,000	395,000	18,200
Tube Size: O.D.	4.5"	4.5"	2.5"
I.D.	3.438"	3.438"	1.833"
Tube Length	97'-9"	97'-9"	260'
x No. of Tubes	360	385	110
Tube Material	SA-335,P-22	SA-335,P-22	SA-335,P-22

The total economizer surface area is 785,220 ft^2.

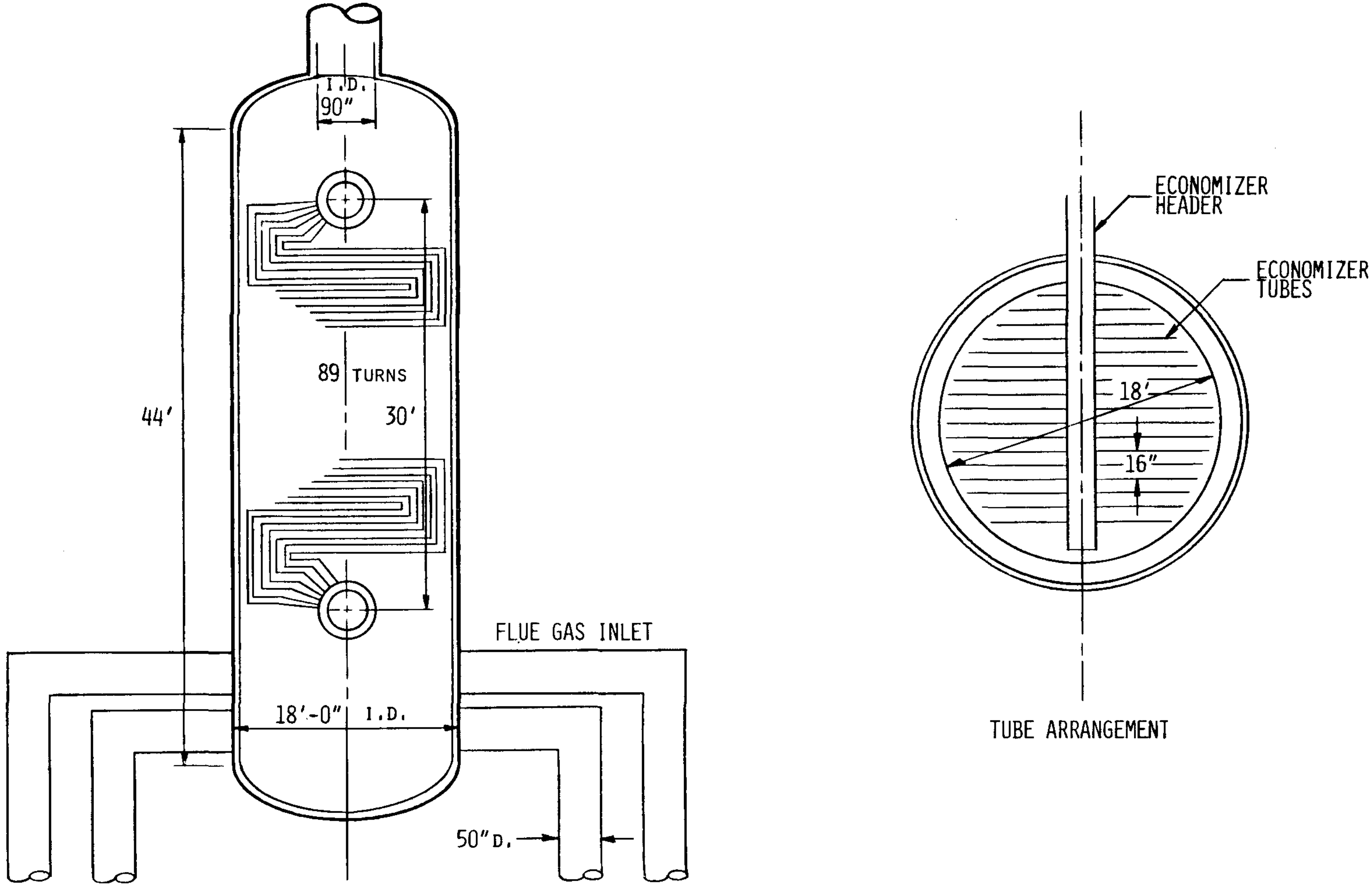


Figure 4-12. - Pressurized Convection Section.

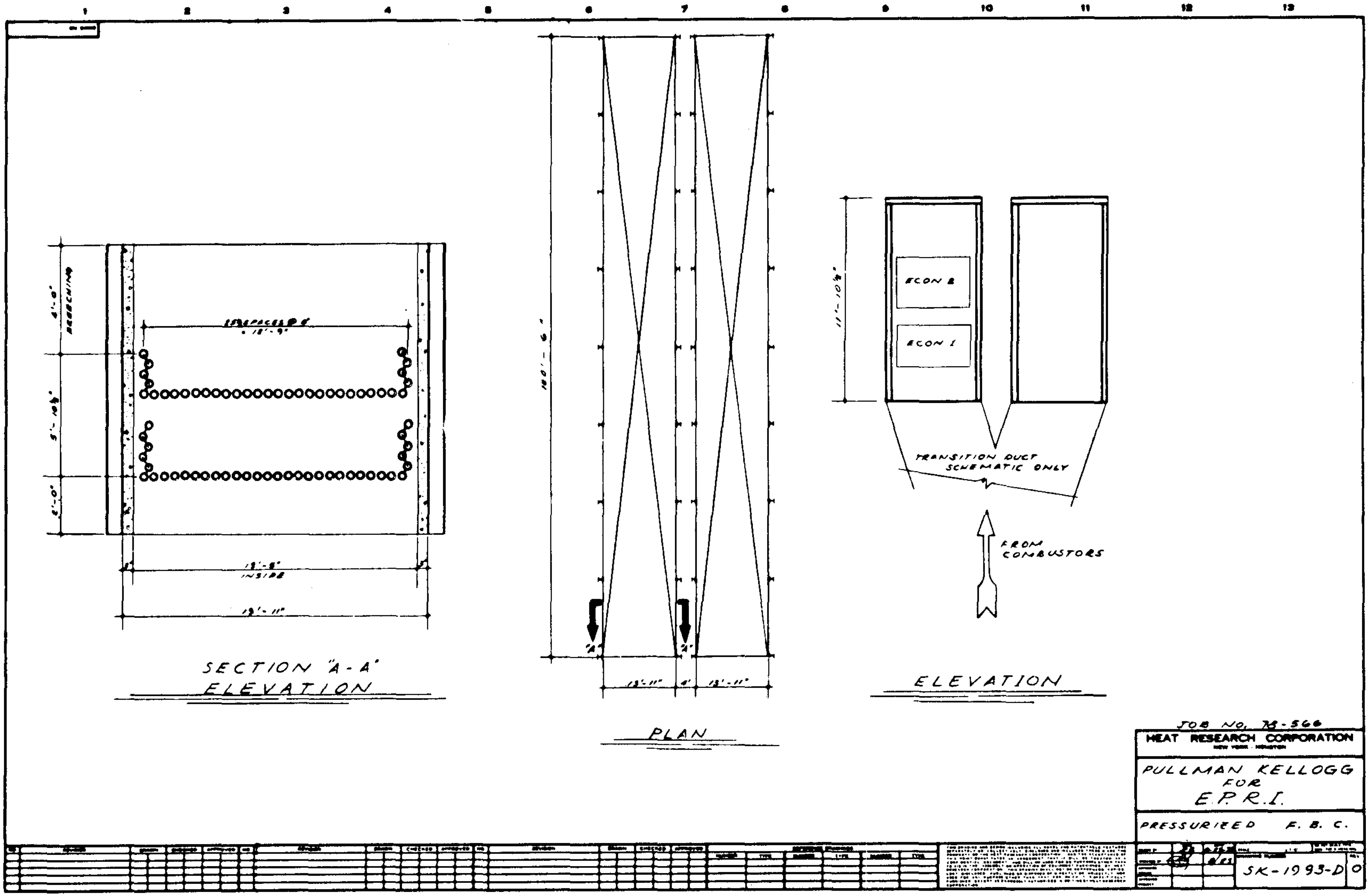


Figure 4-13. - Low Pressure Convection Section Economizers for Pressurized Circulating Bed Combustor.

4.4.2 Case 2 - Pressurized Circulating Bed Combustor Without Dense Bed

As shown in Figure 4-14, this case differs from Case 1 in that it does not have a dense-bed. Therefore, all heat exchanger tubes are installed in the combustor, itself. The fresh limestone and coal feeding hoppers, cyclones, and the convection furnaces are essentially identical to those in Case 1. Four streams of coal and fresh limestone are fed into the bottom of the combustor where they are mixed with the solids returning from the cyclones. Since both the returned solids and the combustor are at 1650°F, combustion of the fresh coal is initiated at the bottom. As the solids-gas mixture flows upward, it comes into contact with the reheater, superheater and evaporator tubes. The tube spacing is designed in such a way that a number of channels, five inches wide, spanning the diameter, and running vertically through all the tube bundles between each row of tubes, creates a smooth flow of the fluidized solids stream. This enhances the chemical reaction, heat transfer and flow distribution. The solids-gas mixture enters the cyclones at the top of the combustor. Solids are separated from the flue gas and returned to the bottom of the combustor. The flue gas leaves the cyclones and enters the pressurized convection section, flue gas cleaning system, gas expanders, low pressure economizer, baghouse, and stack as depicted in Case 1. The calculated flue gas pressure drop distribution is:

• Acceleration loss	8.05 psi
• Static head loss in combustor	9.60 psi
• Total	17.65 psi

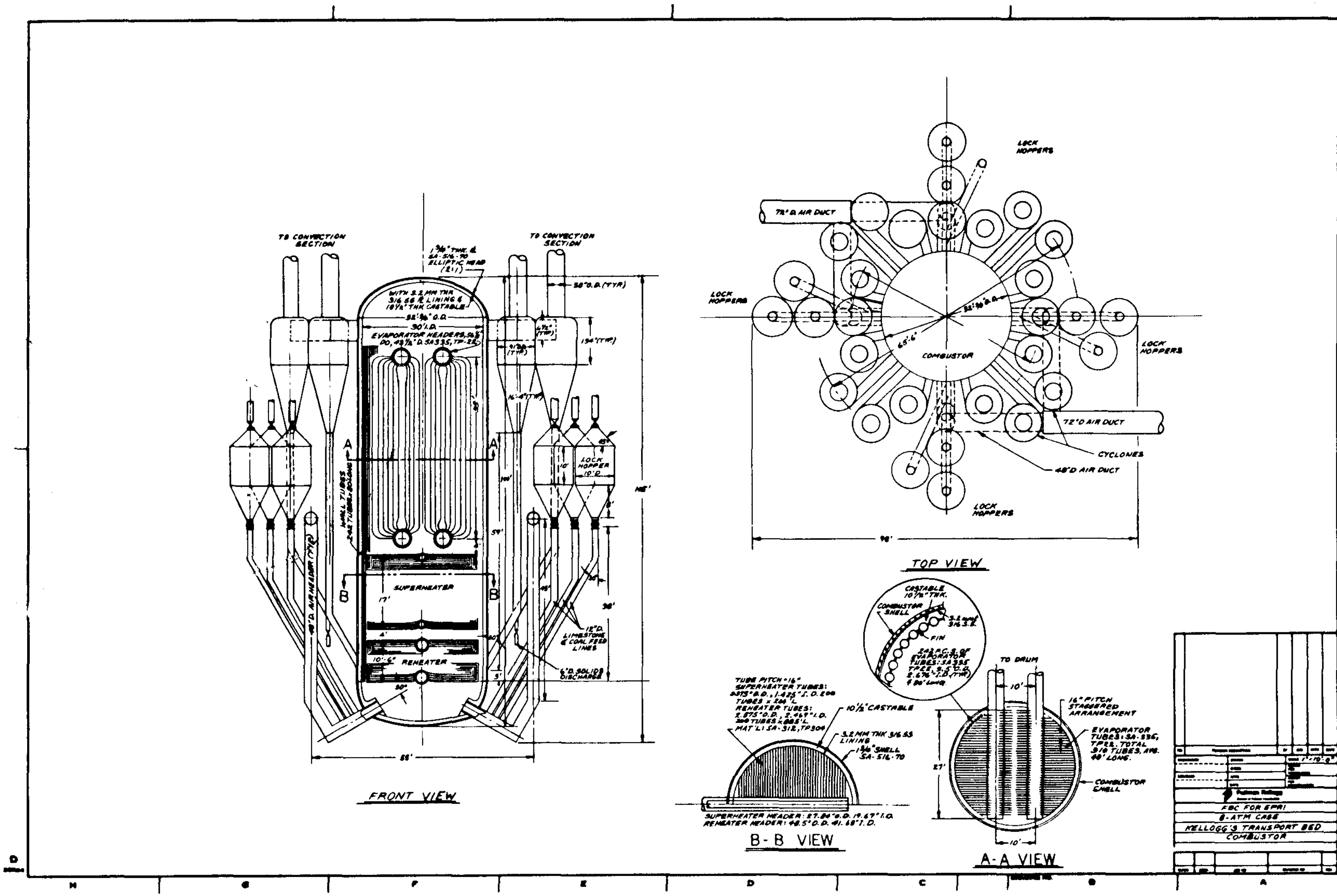


Figure 4-14. - Pressurized Circulating Bed Boiler without Dense Bed.

4.4.2.1 Combustor.- In designing the fast-fluidized-bed combustor, consideration was given to accommodating the entire heat transfer surface required in the combustor since there is no additional heat transfer space like the dense bed in Case 1. In this case, sizing based on heat exchanger installation resulted in a conservative design for the chemical reaction. The combustor design data are shown in Table 4-6.

TABLE 4-6.- COMBUSTOR DESIGN DATA

Item	Design Data
Number of Combustors	1
Inside Diameter	30'0"
Height	115'-0"
Solids Density	12 lbs/ft^3
Gas Velocity	22 fps (at bottom of combustor)
Gas Residence Time	5.4 sec
Required Gas Residence Time	4.0 sec (for SO_2 removal)
Flue Gas Flowrate	5,254,700 lbs/hr
Inlet Air Pressure	8 atm
Combustor Operating Temperature	1650°F
Material of Construction;	
Shell	1 3/4" SA-517-70 C.S. Plate clad with 316 SS
Lining	10 1/2"Harbinson-Walker Castolast "G"

4.4.2.2 Evaporator, superheater and reheater.- The evaporator, superheater and reheater heat duty is the same as in Case 1 except for the evaporator tube arrangement. The heat transfer tube surfaces and their arrangement are the same as in Table 4-4. There are 242 evaporator tubes, each 80 feet long, incorporated as a membrane wall on the inside diameter of the combustor. The remainder of the evaporator consists of 310 tubes, 40 feet long, installed in the fast bed on top of superheater and reheater as shown in Figure 4-14. The rest of the design data are shown in Table 4-4.

4.4.2.3 Convection section.- The heat duty and design of the convection section is identical to that of Case 1.

REFERENCES

"Technical Notes for the Conceptual Design for an Atmospheric Fluidized-Bed Direct Combustion Power Generating Plant," Stone and Webster Eng. Co, Volume 1, March 1978.

"Fluidization Engineering," Kunii & Levenspiel, J&W & Sons, NY, 1969.

Yerushalmi, J., et al., "Flow Regimes in Vertical Gas-Solid Contact Systems," paper presented at the 65th Annual Meeting of the American Institute of Chemical Engineers, Chicago, IL, November 29 - December 2, 1976.

Cankurt, N.T., et al., "Gas Backmixing in High Velocity Fluidized Beds," Fluidization, Cambridge University Press, 1978.

"Advanced Steam Cycles Using Fluidized Bed Steam Generation and Heating - Task I," EPRI FP-317, Project 582-1, December 1976.

Fiala, S.N., "First Commercial Supercritical-Pressure Steam Electric Generating Unit for Philo Plant," Trans. ASME, Volume 79, 1957.

Rowand, W.H., et al., "First Commercial Supercritical-Pressure Steam Turbine Built for the Philo Plant," Trans. ASME, Volume 79, 1957.

8. Elston, C.W., et al., "First Commercial Supercritical-Pressure Steam Turbine Built for the Philo Plant," Trans. ASME, Volume 79, 1957.

9. "Eddystone Plant Employs Supercritical-Pressure Cycle," Mechanical Engineering, Volume 79, No. 2.

10. Campbell, C.B., et al., "The Eddystone Superpressure Unit," Trans. ASME, Vol 79, 1957.

11. Powell, E.M., "Engineering the Eddystone Steam Generator for 5000 psig, 1200°F Steam," Trans. ASME, Vol. 79, 1957.

12. Sheldon, R.C., et al., "Performance Characteristics of Combined Steam-Gas Turbine Cycles," Proceedings of the American Power Conference, 1962, Vol. XXIV.

13. Foster-Pegg, R.W., "Steam Bottoming Plants for Combined Cycles," Journal of Engineering for Power, Vol. 100, April 1978.

APPENDIX I

BASIC STUDIES (With References)

1.0 GENERAL

Several basic studies were required to design the fast fluidized bed combustor. They considered sulphur dioxide (SO_2) removal, combustion efficiency, NO_x emission, solid carryover, heat transfer, erosion, corrosion, materials of construction, and pressure drop in circulating beds.

2.0 SO_2 REMOVAL

When developing new combustor designs, a principal objective is to provide adequate SO_2 removal to meet current and projected EPA standards. The material in the following discussion will review the experimental data available concerning SO_2 removal from different fluidized bed test units, and it will discuss the variables that affect the efficiency of SO_2 removal. Also a procedure for sizing the combustor for SO_2 removal will be described.

2.1 FIELD DATA

2.1.1 External data

Experimental data from various organizations were reviewed for application in SO_2 removal in this design. The units included a Babcock and Wilcox (B&W) 3-foot x 3-foot unit (1), and two Pope, Evans & Robbins (PER) units (2); one of which was 12 feet x 16 feet in cross section, and the other was 2 feet x 20 feet in cross section.

2.1.2 Operating Parameters

The effects of various operating parameters that may affect SO_2 removal in a combustor bed are significant enough to be considered separately. The following paragraphs discuss each parameter.

A. Effect of sorbent type and pretreatment conditions (see Figures I-1, I-2, and I-3) - Generally dolomite sorbent is more effective than limestone. However, on a weight basis, limestone is more economical to use since limestone contains about 97% $CaCO_3$, while dolomite has only about 51%.

Fine sorbents in hydrate form appear to be the most reactive. High activity can also be obtained by precalcination under controlled conditions.

B. Effect of sorbent particle size - Desulfurization is strongly dependent on sorbent particle size. This effect can be seen in Figure I-2. The minus 325 mesh size sorbents indicate markedly improved SO_2 removal over the larger size particles.

C. Effect of bed height and gas velocity - Generally the SO_2 removal would improve with longer gas residence time in contact with the sorbent solids. This effect can be observed in Figures I-4 and I-5.

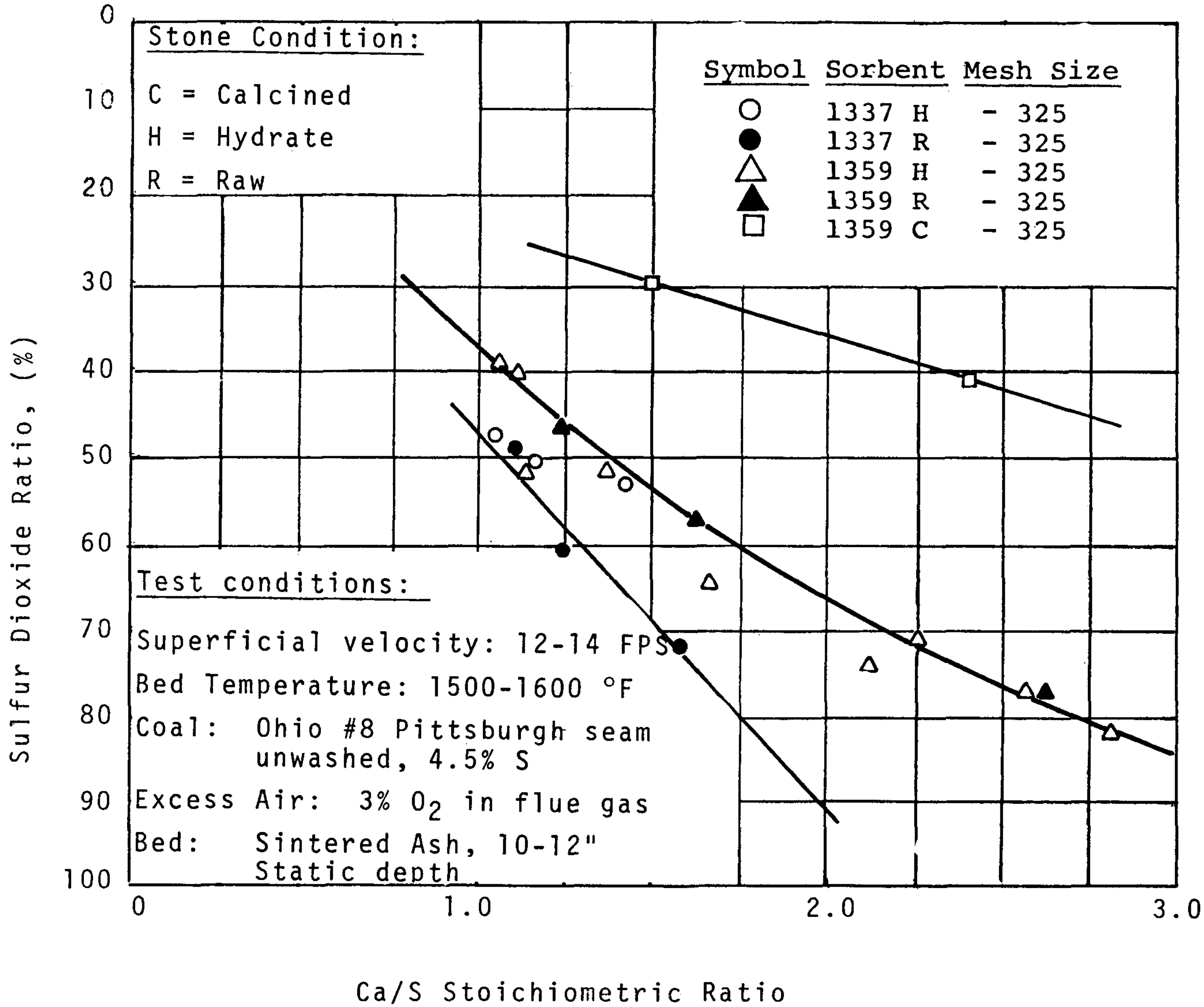
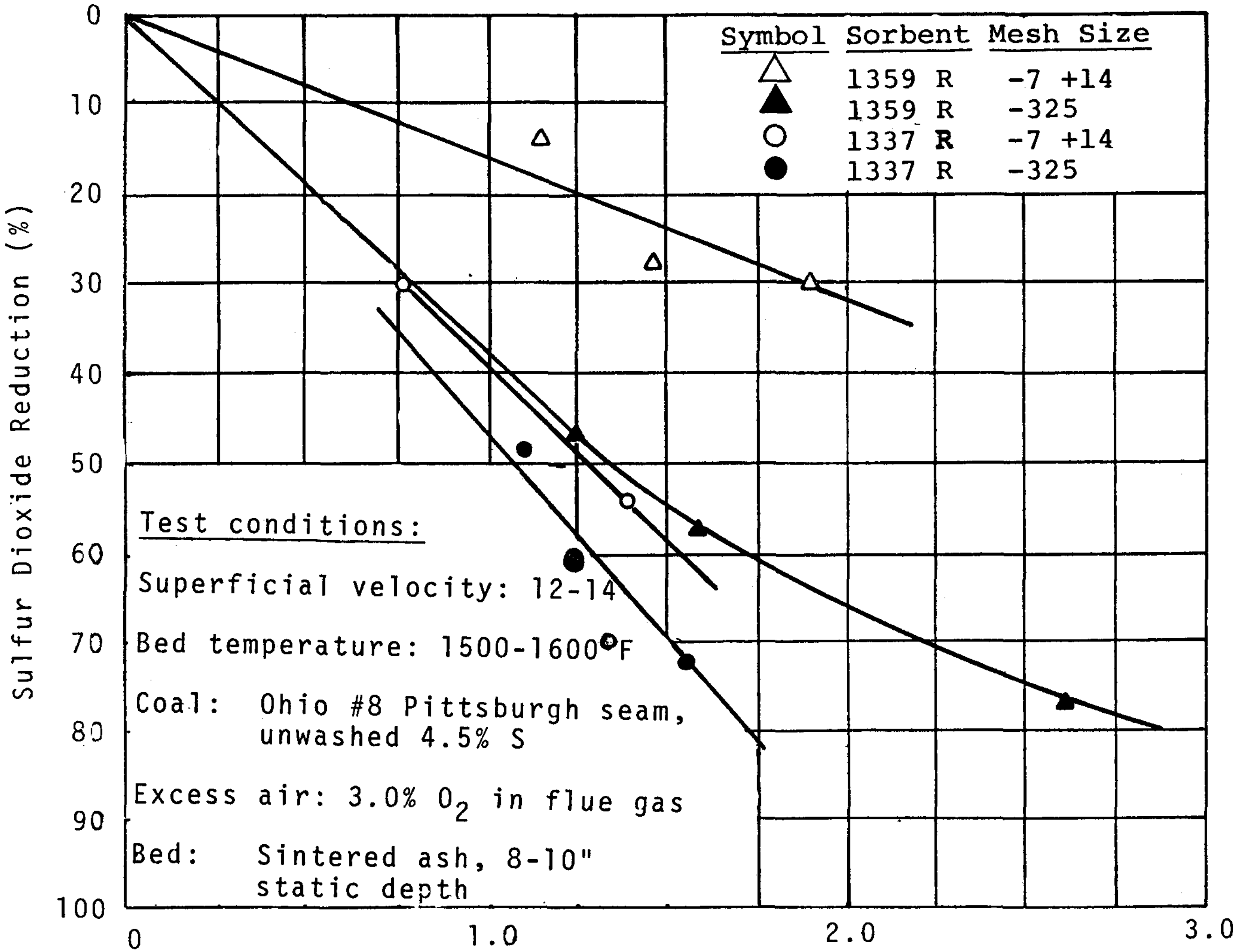


Figure I-1. - Sulfur Dioxide Reduction with Fine Sorbent Addition to the FBC Burning a 4.5% Sulfur Coal. (Ref. 2)


Sulfur Dioxide Reduction (%)
0
10
20
30
40
50
60
70
80
90
100
0
1.0
2.0
3.0
Symbol Sorbent Mesh Size
1359 R -7 +14
1359 R -325
1337 R -7 +14
1337 R -325
Test conditions:
Superficial velocity: 12-14
Bed temperature: 1500-1600°F
Coal: Ohio #8 Pittsburgh seam, unwashed 4.5% S
Excess air: 3.0% O_2 in flue gas
Bed: Sintered ash, 8-10" static depth

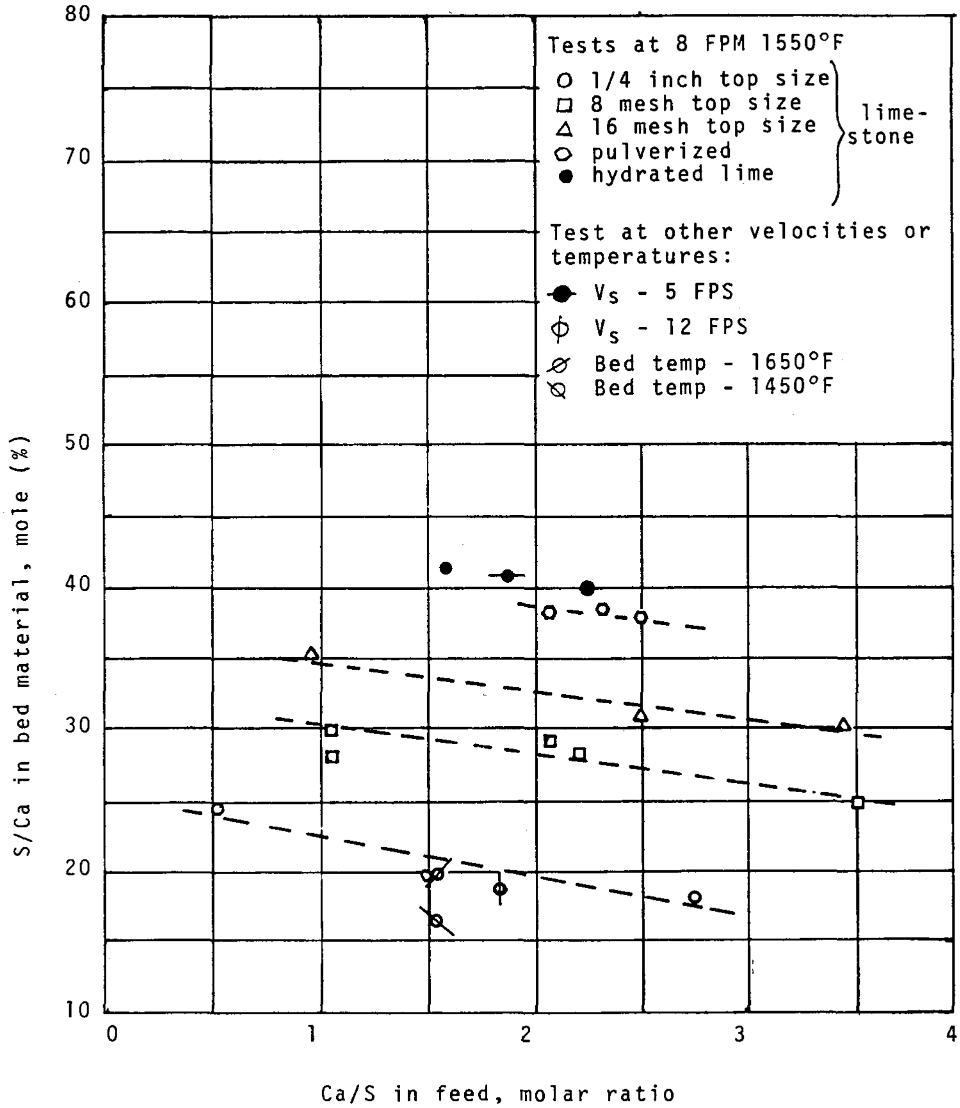


Figure I-3. - Utilization of Calcium in the Bed.
(Ref. 1)

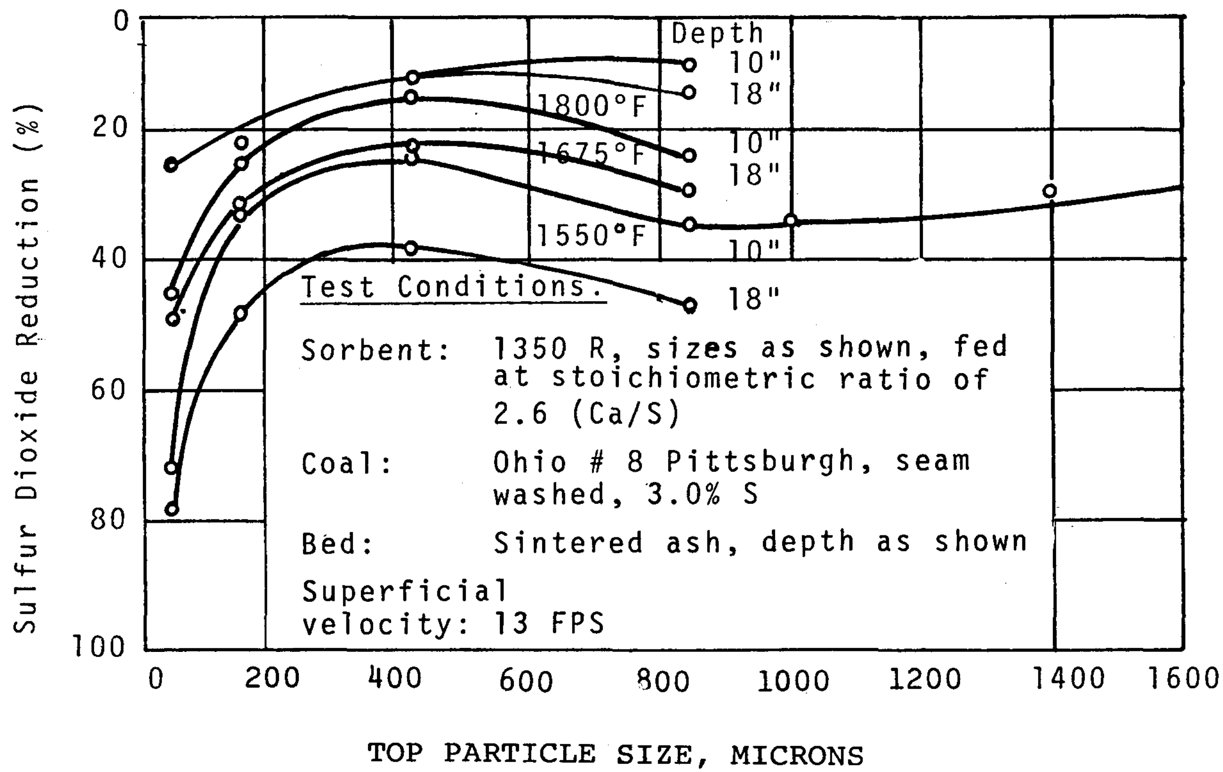


Figure I-4. - Variation in Sulfur Dioxide Reduction with Sorbent Particle Size, Bed Depth, and Bed Temperature. (Ref. 2)

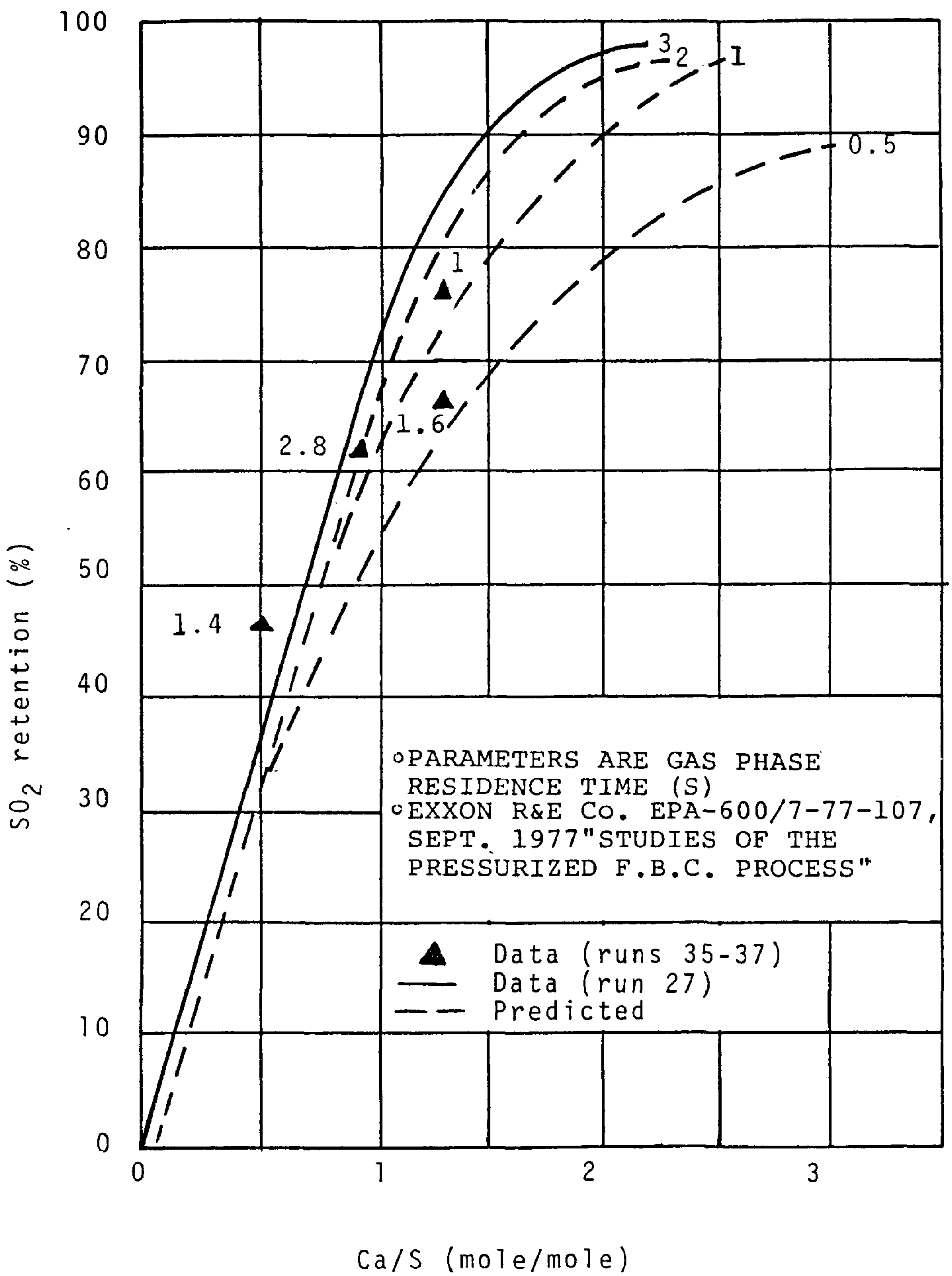


Figure I-5. - Effect of Gas Residence Time on SO_2 Retention Dolomite No. 1337 (90% Between 8-20 Mesh). (Ref. 4)

D. Effect of temperature - Desulfurization increases with temperature to a maximum, then decreases as indicated in Figure I-6. Figure I-7 illustrates that SO_2 sorption increases with temperature in a differential reactor for 30 mg of dolomite 135.

E. Effect of pressure - SO_2 retention data for various test units operating at atmospheric and elevated pressures are summarized in Table I-1. The effects of pressurized vs. atmospheric operation upon the sulfation of limestone and dolomite are discussed in detail in the footnoted reference[1]. Increasing the pressure from one atmosphere to ten atmospheres for a calcined dolomite increased the reaction rate by a factor of about four to five in the sulfation range of interest (40 to 50% sulfation).

Operation of fluidized bed combustors at elevated pressures increase the partial pressure of CO_2. When the CO_2 partial pressure exceeds the equilibrium pressure for calcination of calcium carbonate, the calcination of the limestone within the fluidized bed combustor is more difficult. In

[1] Ulerich, N.H., et.al., "A Thermogravimetric Study of the Sulfation of Limestone and Dolomite- Prediction of Pressurized and Atmospheric Fluidized Bed Desulfurization", 8th North American Thermal Analysis Society Conference, Atlanta, Georgia, October 15-18, 1978.

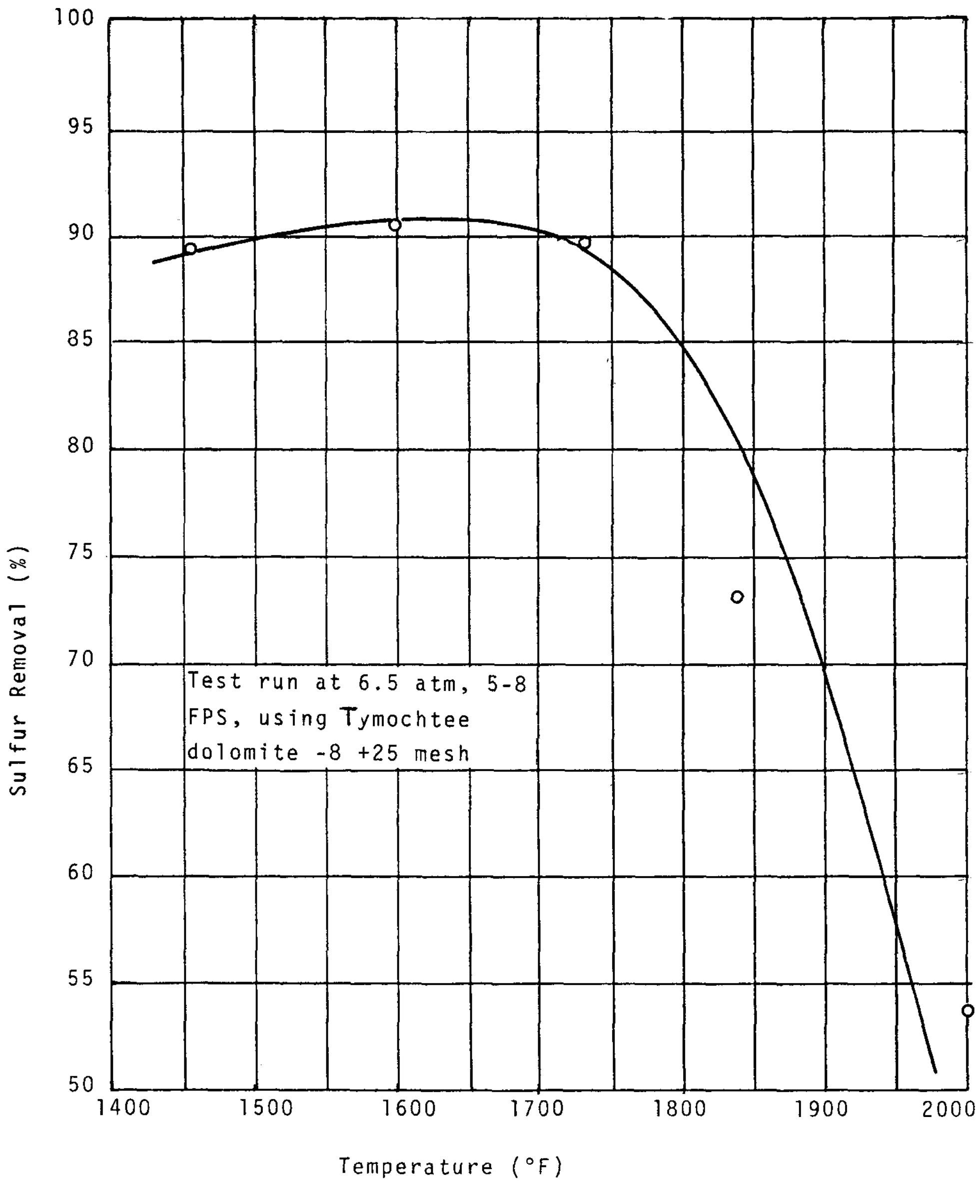


Figure I-6. - Effect of Temperature on Desulfurization. (Ref. 5)

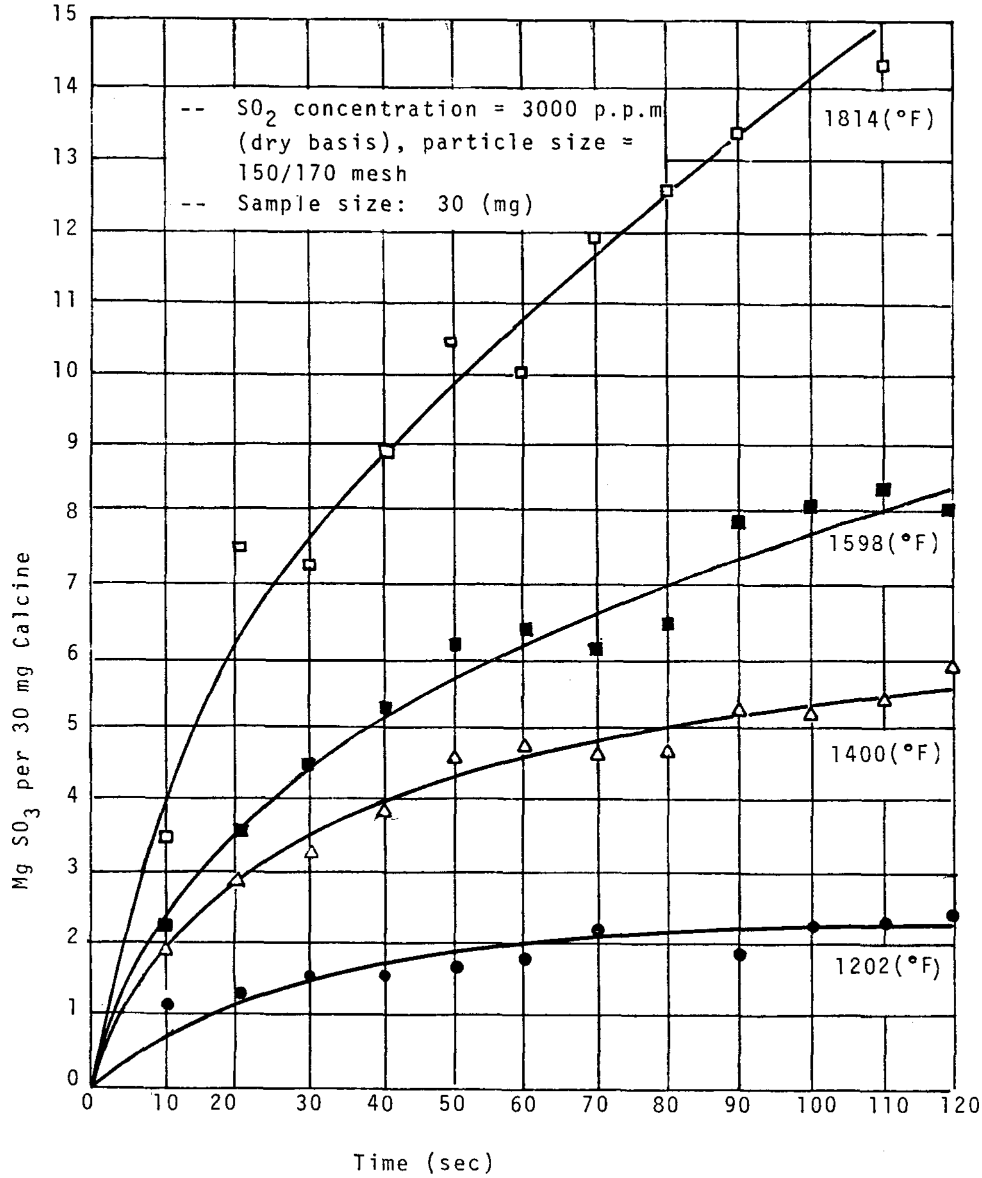


Figure I-7. - Sorption of Sulfur Dioxide by Dolomite 1351 at Various Reactor Temperatures. (Ref. 6)

TABLE I-1.- SO_2 REMOVAL - EXPERIMENTAL DATA

Research Organization	Sorbent	C_a/S	Temp. (°F)	Pressure (Atm.)	%SO_2 Retention
PER	Limestone & Dolomite	3/1	1500-1800	1.	90
B&W	Limestone	3/1	1450-1650	1.	
OCR	Dolomite	2/1			95
ANL	Dolomite	2/1	1550	10	90
			1550	10	95
ESSO R&E	Dolomite	3/1	550-16505	6.	90
Consolidation Coal Co.	Dolomite	1/1	1800	1.	78
		2/1	1800	1.	90

this case the calcination step must be completed external to the combustor (precalcination). Generally, pressurized operation calls for the use of dolomite as the sorbent, since dolomite is less affected by increased CO_2 partial pressures.

F. Effect of Ca/S feed ratio - SO_2 removal improves strongly with an increasing calcium to sulfur ratio in the feed. This can be seen in Figures I-1 to I-3 and Figure I-5. Figure I-8 reports some of the data for the following conditions of interest to this study:

- minus 325 mesh particle size
- 8-13 fps gas velocity
- 1550-1650°F, 1 atm

As shown in Figure I-8, the Ca/S molar feed ratio that would be required to achieve 90% SO_2 removal in the combustor would be about 2 to 2.5 with dolomite and 2.5 to 3.5 with limestone sorbent.

2.2 CALCULATION PROCEDURE

2.2.1 The Westinghouse Model

A. Assumptions and formulation - Westinghouse (3) developed the following model to calculate the degree of SO_2 retention in a fluidized bed combustor:

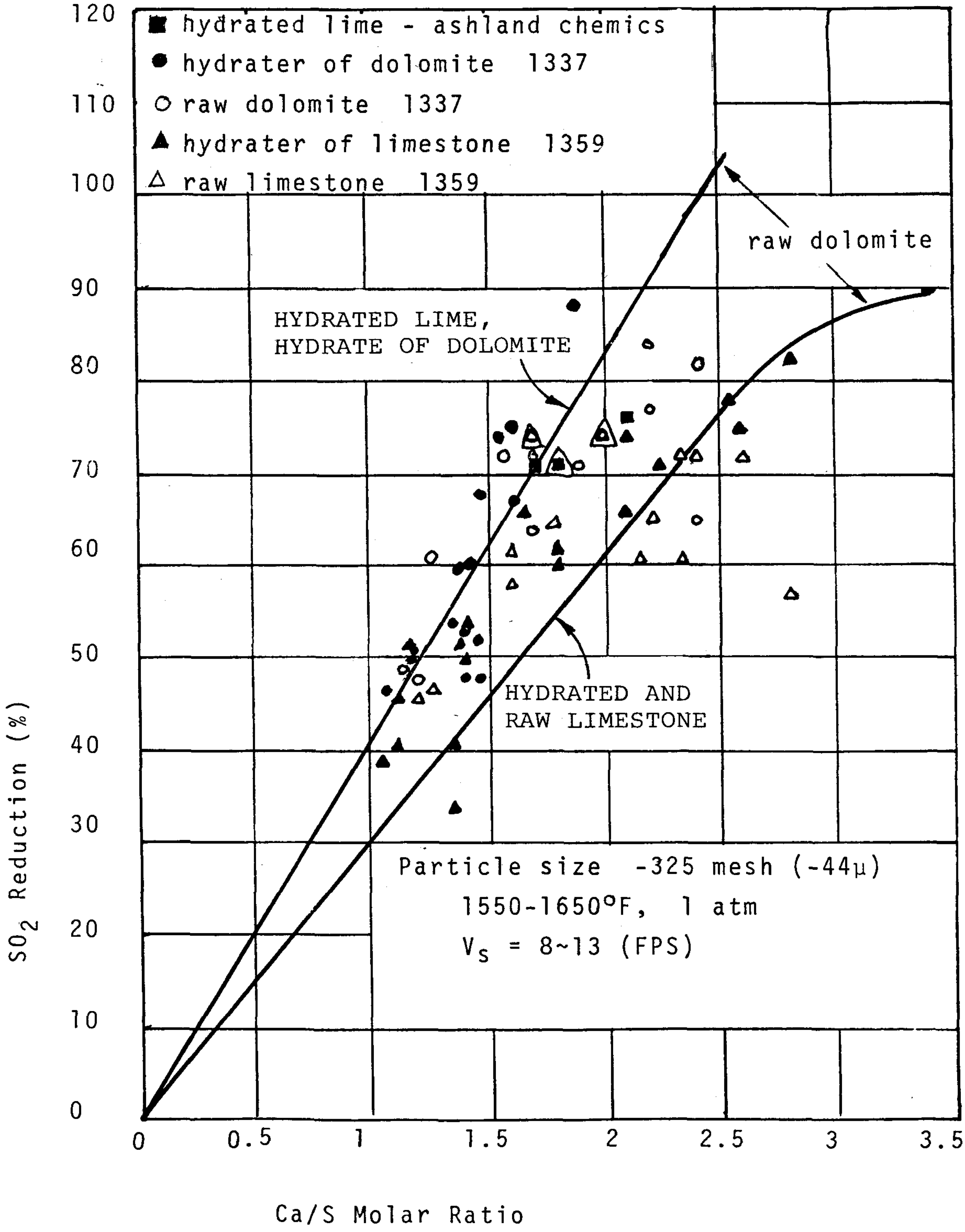


Figure I-8. - Literature Survey SO_2 Reduction vs Ca/S Molar Ratio.

Fraction of SO_2 removal R (1)

$$= 1 - \frac{1}{kt}(1-e^{-kt})$$

where t = gas residence time (sec)

$$= \frac{\text{bed height } H}{\text{gas velocity } V_g}$$

and k = SO_2 - CaO reaction rate constant.

This model uses a first order kinetic expression with two variables, gas residence time t, and reaction rate constant, k. All the variables including temperature, pressure, type of sorbent, history of sorbent (pretreatment, sulfation level), particle size, concentration of sorbent, etc. are lumped under the rate constant, k.

The assumptions used to derive the model are:

1. Gas is plug flow.

2. Sorbent particles are uniformly distributed within the bed and can be represented by a mean particle size and a single sulfation level. This assumption is based both on a large recycle to feed ratio and on good solid recirculation within the combustor.

3. SO_2 is generated uniformly within the combustor. This implies that the coal particles burn uniformly throughout the combustor. Although it does not conform to the exact SO_2 generation pattern, such an assumption leads to conservative SO_2 removal predictions.

B. <u>Experiment: Thermo-Gravimetric-Analyzer (TGA)</u> - As mentioned earlier, the rate constant k is dependent on a number of parameters such as the type of sorbent, its precalcining conditions, its particle size, the sulfation level, and bed temperature and pressure, etc. Although this constitutes a complex function of all these variables, k can be determined by a single experiment with the Thermo-Gravimetric-Analyzer. In this experiment, a stream containing SO_2 flows through a sample of the sorbent of interest at the desired test conditions. As the SO_2 gas is absorbed, the weight change of the sorbent sample with time is recorded by a micro-balance. This rate of weight change can be converted into reaction rate data for this set of operational information. The equation used to relate the experimental data to the rate constant is:

$$k = \underline{d\alpha} \times \rho \times \frac{1-\varepsilon}{\varepsilon} \cdot \frac{1}{C_{SO_2}} \qquad (2)$$

where: C_{SO_2} = SO_2 concentration used in the TGA experiment [g moles/cm^3 gas]

ε = void fraction in bed (expanded)

ρ = solids mole density [g mole Ca/cm^3 solids]

$\frac{d\alpha}{dt}$ = fraction of total Ca converted per unit time [sec^{-1}]

C. Results - Typical data (7) for $\frac{d\alpha}{dt}$ are shown in Figure I-9 for one type of limestone (pretreated Greer limestone) and one set of test conditions (815°C, atmospheric pressure). Reference 3 develops additional data concerning different sorbents and sets of operating conditions.

Westinghouse reported good agreement between the SO_2 removals predicted by this method and actual test results from the Exxon and Argonne National Lab FBC units. The comparison is shown in Figure I-10, and it is taken from Reference 3.

2.2.2 Application to Design

The calculation procedure discussed in the previous paragraphs can be applied to the design of the combustor. The size of the combustor should provide adequate gas residence time to remove SO_2 to the required level. This size (or residence time) is determined by the rate of the SO_2 removal reaction, which in turn depends on the selection of the sorbent and operating conditions for the bed.

A. Selection of sorbent - When selecting a sorbent, factors such as type and availability of sorbent, pretreatment, particle size, and sulfation level all have to be considered. Also, the highly reactive forms of sorbent, such as hydrated lime, may not be available generally, or the cost of these materials may preclude this on FBC applications.

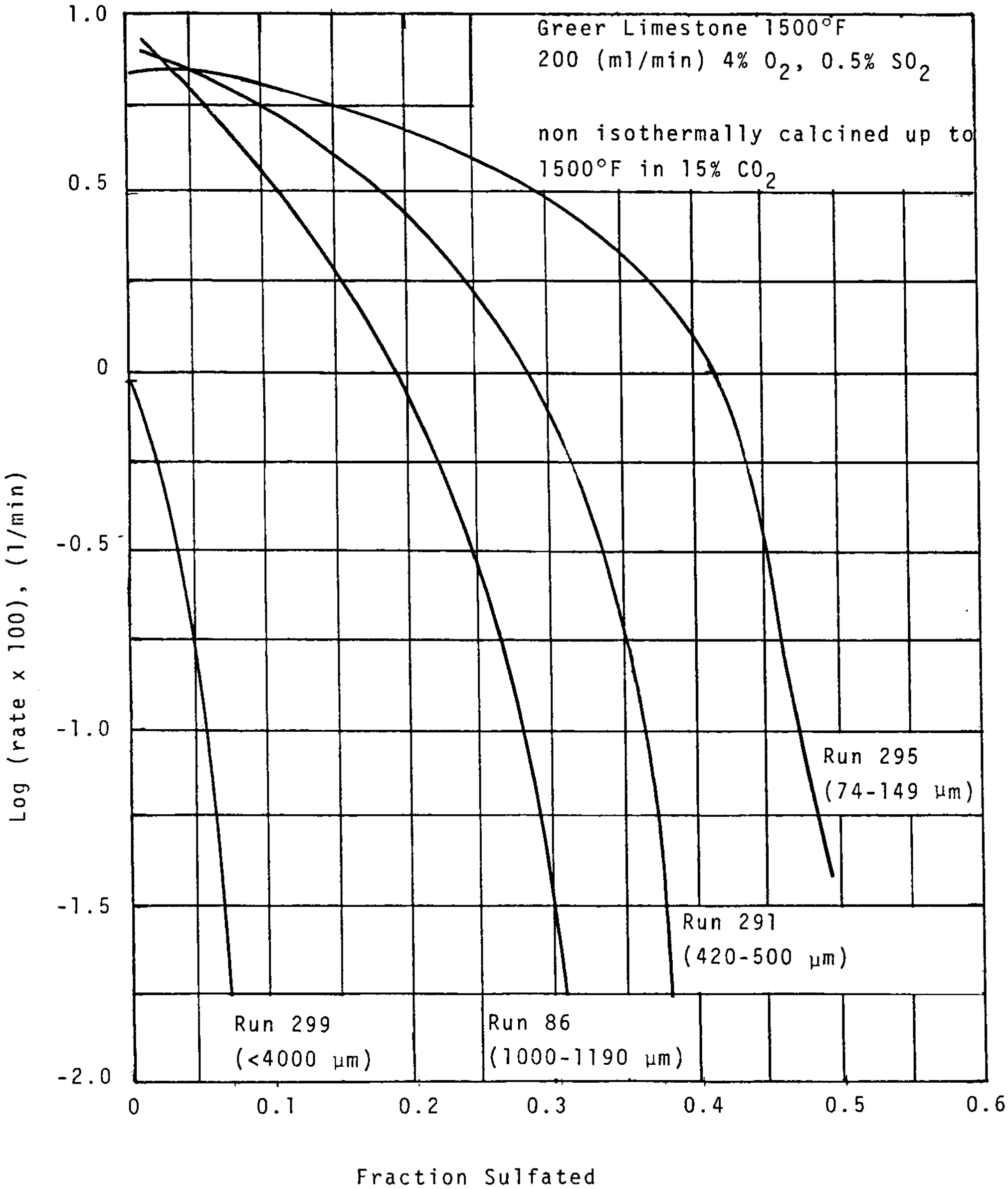


Figure I-9. - The Effect of Particle Size on Greer Limestone Sulfation.

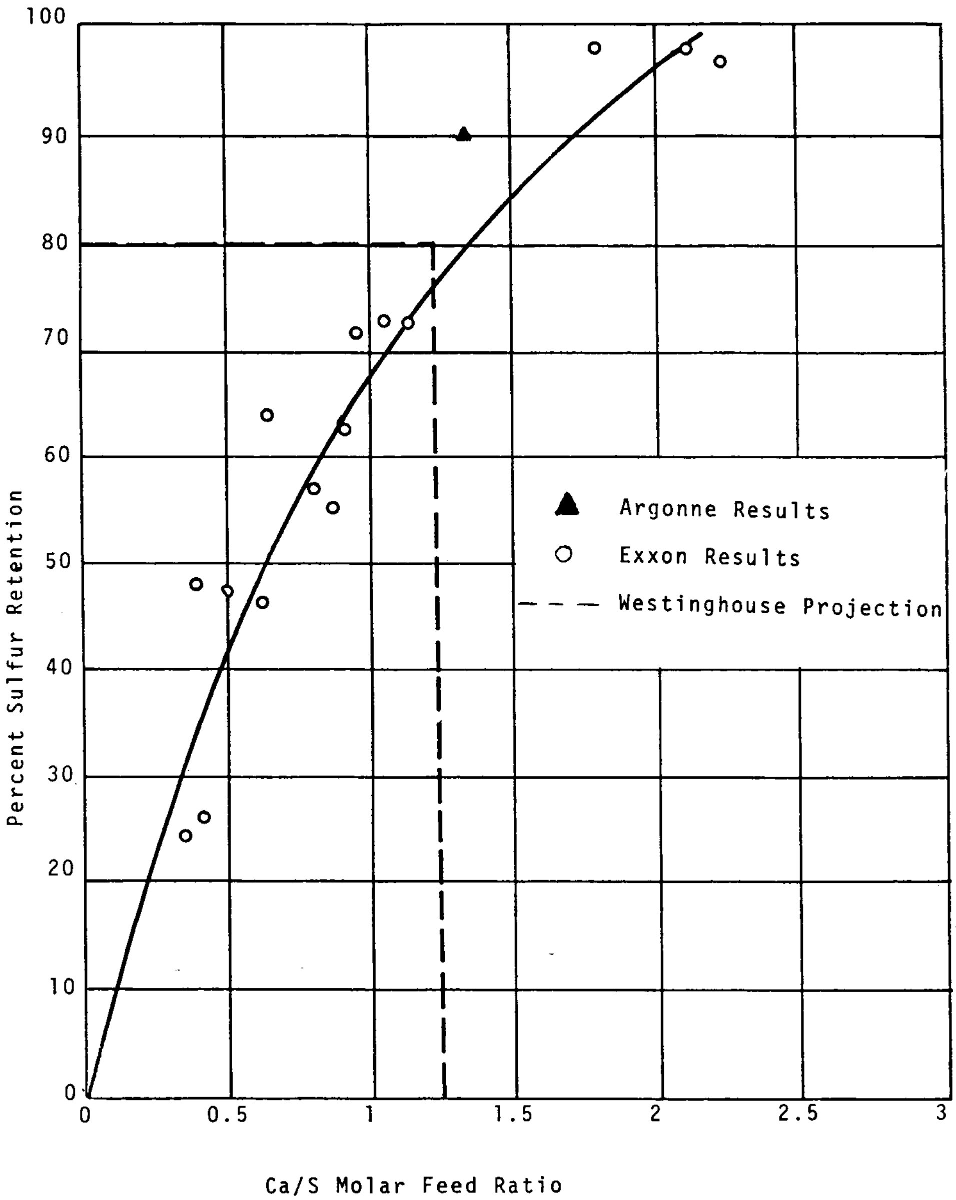


Figure I-10. - Comparison of Exxon Miniplant Results with Dolomite 1337 and Projections from Westinghouse TG Data.

The sorbent sulfation level (or degree of calcium utilization) dictates the Ca/S feed ratio. For about 90% sulfur removal, at 40 to 50% sorbent utilization, the Ca/S feed ratio required would be 1.8 to 2.25. With less sorbent utilization, between 10 and 20%, the Ca/S feed ratio required would increase to 4.5 to 9, raising boiler operating costs.

Controlled pre-calcination outside of the FBC unit would enhance the sorbent's reactivity and utilization. The penalties however are the need for additional processing equipment and thermal losses.

The sorbent assumed for this study was of small particle size (100 microns or less), and was assumed not to require external calcining (1% Ca converted per minute) at reasonable calcium utilization levels (40% or better). These assumptions require using dolomite as the sorbent rather than limestone for pressurized operation.

B. Sizing the combustor for SO_2 removal - Once a sorbent is selected, its reactivity can be measured by TGA experiments. The reaction rate constant, k, can be determined for each set of conditions. Equation (1) can then be used to find the gas residence time needed to achieve the fraction of SO_2 removal required.

Applying this calculation procedure for atmospheric pressure operation and using the reactivity data of Figure I-9 (for precalcinating Greer limestone, 74 to 149 micron particle size), the calculated residence times required to achieve 92.4% SO_2 removal can be plotted as a function of bed density. This has been done in Figure I-11. For 40% sorbent utilization (corresponding to Ca/S feed ratio of 2.25), the residence time for 92.4% sulfur removal decreases from 5.3 to 0.5 seconds when bed density increases from 3 to 25 lbs/ft. Less residence time would be required at a higher Ca/S feed ratio.

Figure I-11 indicates the residence time to provide SO_2 removal resulting in 0.6 lbs SO_2/MM Btu (or about 92.4% removal) and 40% sorbent utilization. For a low bed density (about 4 lbs/ft^3), up to 4 seconds gas residence time may be required to meet the SO_2 removal requirement.

Figure I-12 shows a similar relationship for 8 atmosphere operation. At the higher pressure, it appears that higher bed densities and longer gas residence time will be required in order to achieve the targeted SO_2 removal of 92.4%.

For this study, and until selection of a sorbent is made final (along with reactivity data), the results presented in Figures I-11 and I-12 will be used as design guidelines to size the combustor for SO_2 removal.

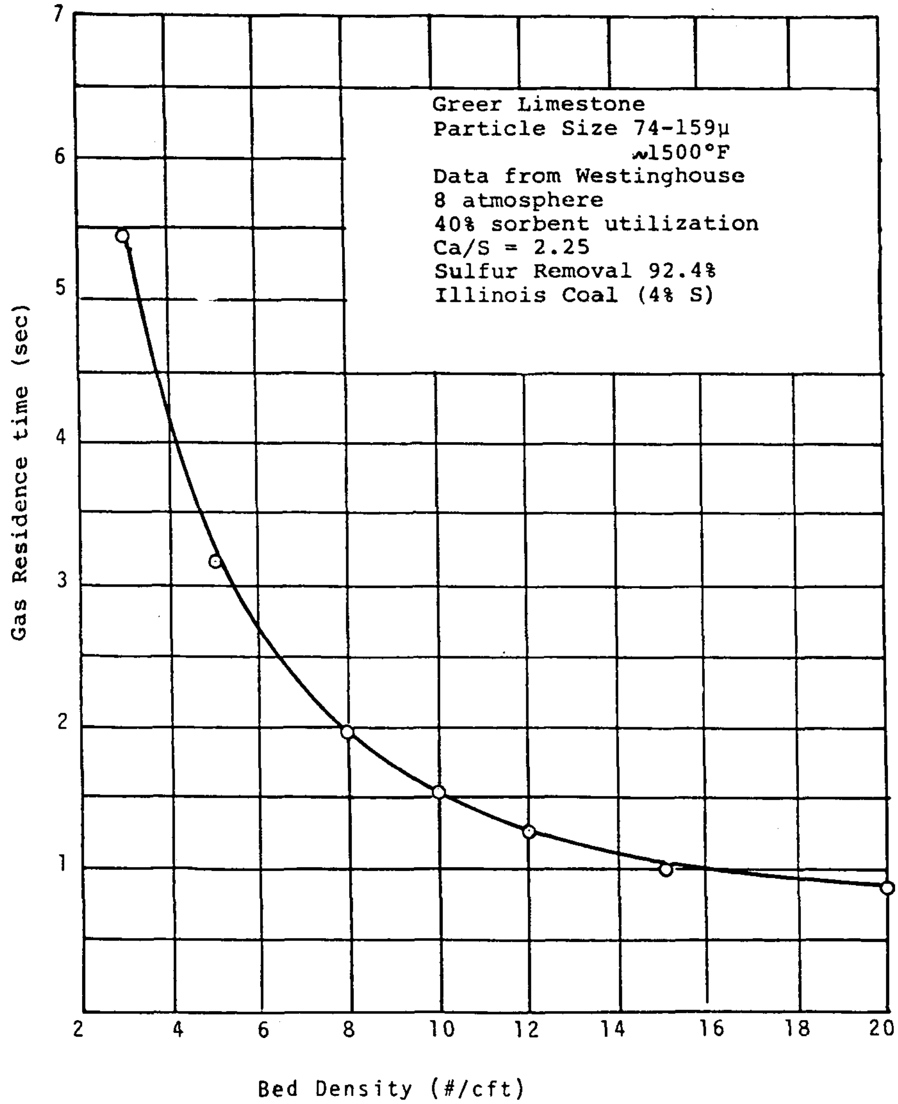


Figure I-11. - Design Curve - Gas Residence Time vs Circulating Bed Density.

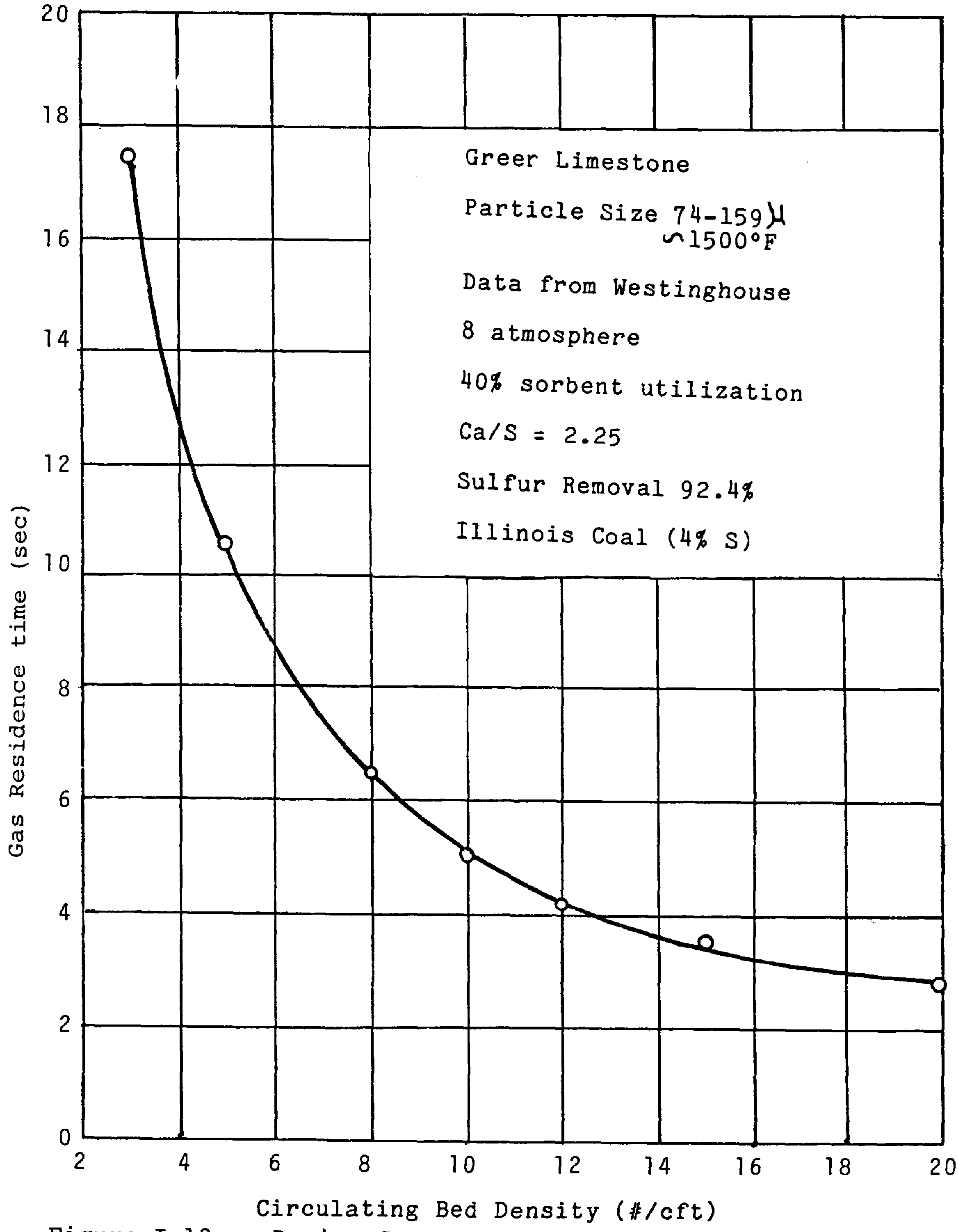


Figure I-12. - Design Curve - Gas Residence Time vs Circulating Bed Density at Eight Atmospheres.

3.0 COMBUSTION EFFICIENCY

The materials in this section deal with the combustion efficiency in the circulating bed combustor. As with the question of SO_2 removal, the approach taken in this study of combustion efficiency in the circulating bed combustor will be to review the experimental data on combustion efficiency from fluidized bed test units, to discuss the major variables affecting the efficiency, and to describe the calculation procedure used in the design of the combustor.

3.1 FIELD DATA

3.1.1 Combustion efficiency

Combustion efficiency for fluidized bed systems has been studied extensively, and Table I-2 lists typical data reported by investigators in the field. Generally it is true that combustion efficiency increases with temperature and excess air, and typical test results indicating this are shown in Figures I-13 and I-14. In cases where the reported values for combustion efficiency were not clearly defined, the following definition was assumed:

Combustion efficiency E (%) =

$$100 - \left\{ \begin{array}{l} \text{percent of combustible} \\ \text{carbon from the feed} \\ \text{lost with flyash and} \\ \text{with purged bed} \\ \text{materials} \end{array} \right\} + \left\{ \begin{array}{l} \text{percent of combustible} \\ \text{carbon from the feed} \\ \text{lost as CO} \end{array} \right\}$$

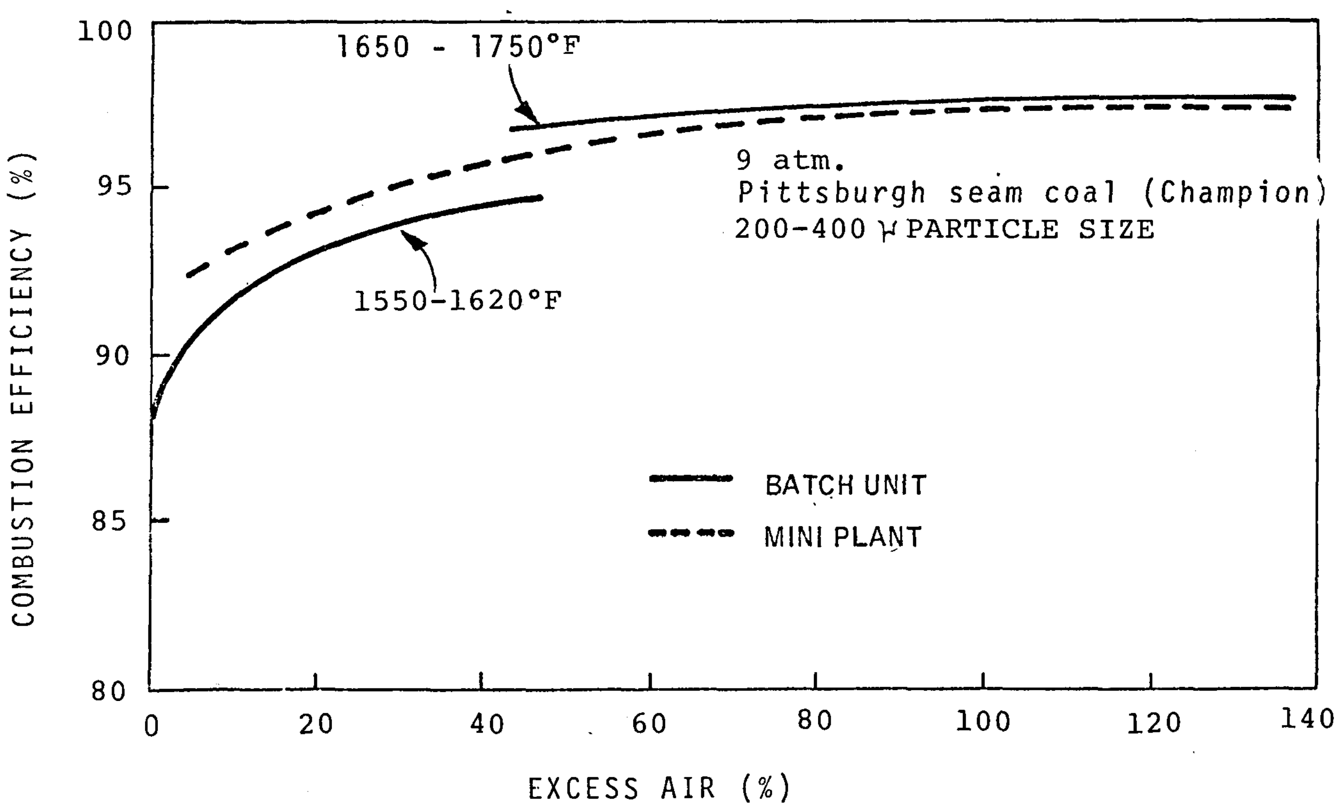


Figure I-13. - Typical Literature Data - Fluidized Bed Combustion Efficiency vs Excess Air.

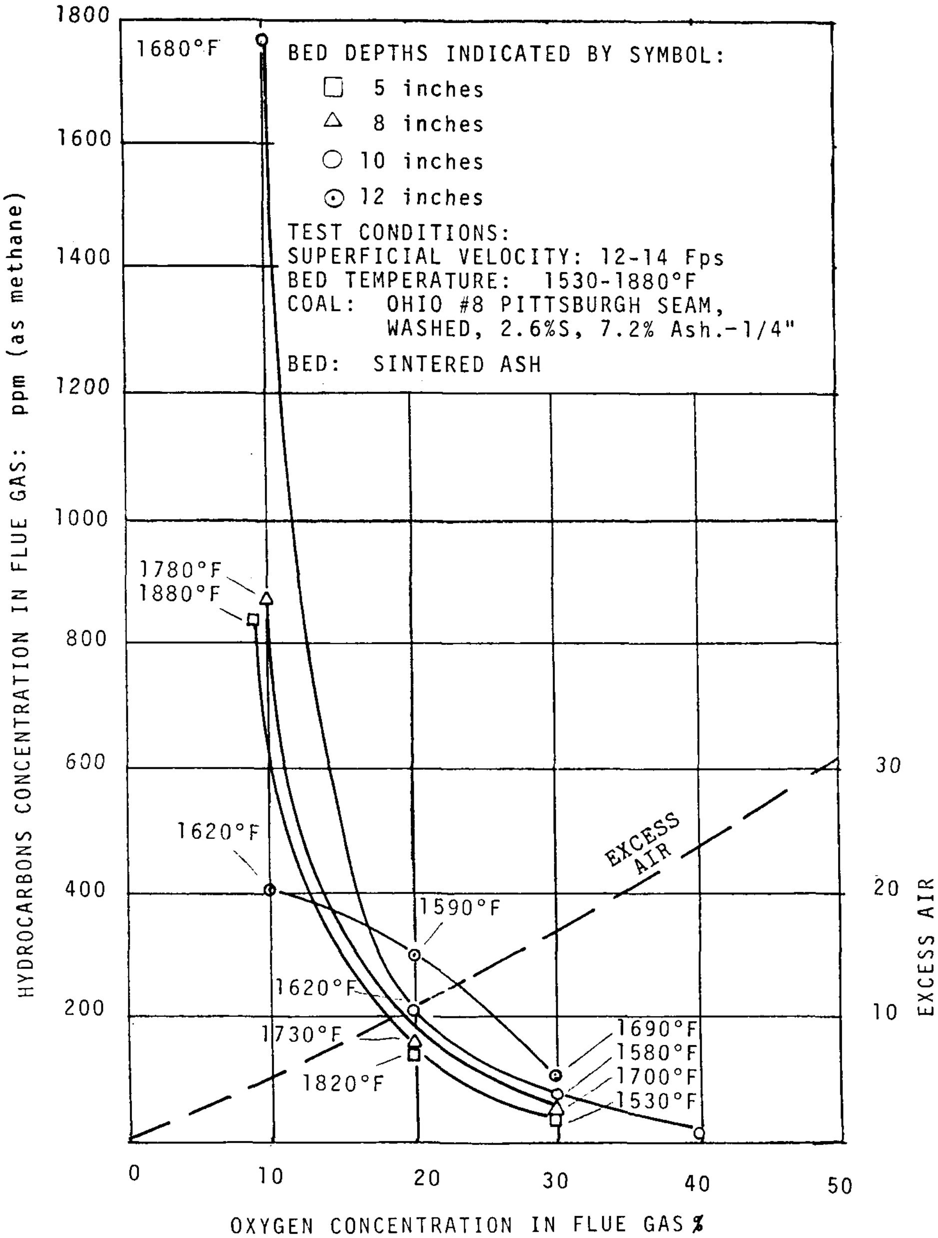


Figure I-14. - Typical Literature Data - Fluidized Bed Combustion Hydrocarbon Concentration in Flue Gas vs Oxygen Concentration.

TABLE 1-2.- COMBUSTION EFFICIENCIES

Research Organization	Temperature (°F)	Pressure (Atm)	Excess Air (%)	Combustion Efficiency
Consolidation Company	1800	1.5	15-20	97
Esso R&E Co.	1500-1800	9		91-97
OCR	1750	Pressurized		99
ANL	1450	8		89
	1650	8		97
Exxon Miniplant	1650-1750	9	+20	92-98
Batch Unit	1550-1600	9	+20	87-98

3.1.2 Sources of Lost Efficiencies

A. Unburned carbon - Unburned carbon particles carried out of the bed represent an important source of loss in combustion efficiency. This loss can be controlled by (%) selection of efficient first stage cyclones to recycle the unburned carbon fines back to the combustor, and/or by allowing more residence time for combustion.

B. Hydrocarbon and CO emissions - Hydrocarbon and CO emissions generally are low. Both are controlled by the degree of excess air used in combustion and by bed temperature.

At temperatures between 1550 and 1650°F, with excess air higher than 15%, the measured CO concentration is about 150 to 250 ppm, and hydrocarbon concentration (measured as methane) is less than

100 ppm. (These data are included in Figures I-14 and I-15.) While high excess air improves combustion efficiency and reduces hydrocarbon and CO emissions, it also results in thermal losses for the boiler because the excess heat cannot be recovered economically. The cost of cooling flue gas beyond 250 to 300°F is too high. If maximum thermal efficiency were the sole objective, an optimum operating point might be around 11% excess air. Usually, a good balance between thermal and combustion efficiency requires 15 to 20% excess air.

C. Solid purges- The solids withdrawn from the combustor bed normally contain much less than 1% unburned carbon.

3.2 CALCULATION PROCEDURE

3.2.1 Assumptions and Formulation

The rate of reaction of coal char particles with oxygen is expressed as follows (8):

$$q = \frac{AP_{O_2}}{\frac{1}{K_g} + \frac{1}{K_s}} \tag{3}$$

where q = rate of consumption of carbon, g-moles/sec

K_s = surface reaction rate coefficient, g-moles/cm^2-s-atm

K_g = 0 mass transfer coefficient, g-moles/cm^2-s-atm

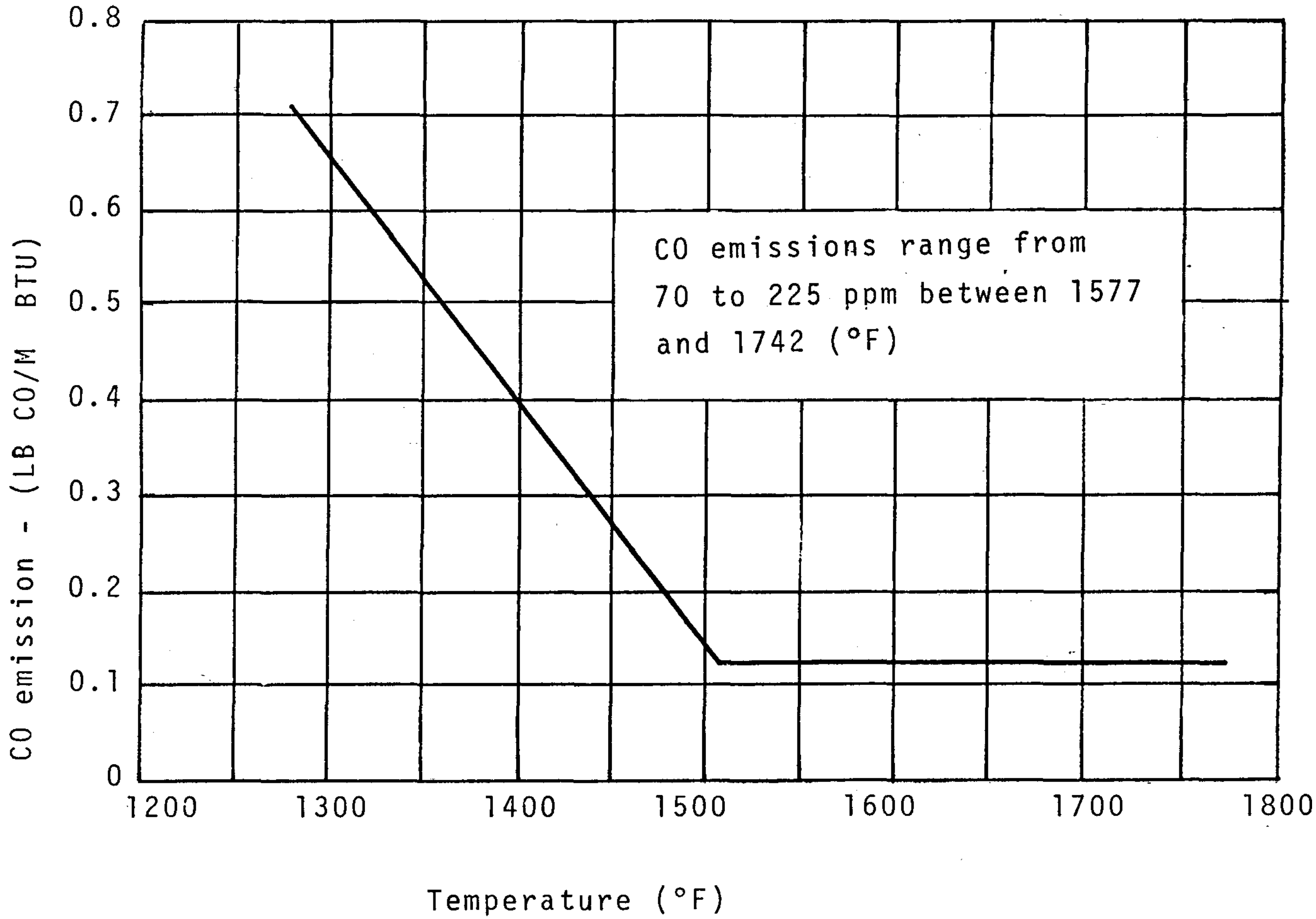


Figure I-15. - Typical Literature Data - Fluidized Bed Combustion
Carbon Monoxide Emissions vs Bed Temperature.

A = total available carbon surface area, cm^2

P_{O_2} = log-mean oxygen partial pressure, atm

$$= \frac{Pin - Pout}{\ln(Pin/Pout)}$$

furthermore,

$$K_S = 726\ e^{-\frac{18,000}{TS}}\ \text{g-moles/cm}^2\text{-s-atm}$$

T_S = temperature at particle surface, °K

and Kg may be estimated by assuming the Sherwood number is equal to two,

$$Sh = 2 = \frac{Kg\ RTmd_c}{D} \tag{4}$$

where d_c = average diameter of the coal char particle in the bed, cm

R = gas constant, 82 cm^3-atm/g-mole-°K

$Tm = \frac{Ts + Tg}{2}$, Tg gas temperature, °K

D = O_2/N diffusion coefficient, cm^2/s

$$= \frac{3.49\ (Tm/1600)^{1.75}}{P}$$

P = total pressure of the system, atm

The total carbon surface available may be calculated by the following relation:

$$A = \frac{VD\ f}{\rho_c} \cdot \frac{6}{d_c} \tag{5}$$

where V = bed volume of the combustor, cm^3

D = bed density, g-solids/cm^3-bed volume

f = average fraction of carbon in the bed material

ρ_c = density of the char particle, gm/cm^3

In using equation (3) to calculate the coal combustion rate, the following assumptions were made:

A. Gas is plug flow.

B. The time required for devolatization of coal particles is rather short relative to the time required for burning the char particles.

C. Char particles are uniformly distributed throughout the bed and can be represented by a mean particle size and an average fraction of carbon in the bed material.

3.2.2 Application to Prediction of Combustion Efficiency

The calculation procedure was applied to the following design case:

A. Coal feed: 513,000 #/hr, average particle size 100 microns

B. 20% excess air

C. Limestone feed: 137,000 #/hr

D. Bed temperature: 1650°F

E. Bed pressure: one atmosphere

F. Bed density: 4 lbs/ft^3

G. Gas velocity in combustor: 60 ft/sec

H. Gas residence time in combustor: 4 sec

The following additional assumptions were used:

A. Burning volatiles in the coal is fast relative to burning the char, and it consumes one half of the stoichiometric oxygen requirement. The time required for devolatilization of the 100 micron coal particles was estimated to be roughly 1 second.

B. The remaining char is 50% of the initial coal particle.

C. Surface temperature of the char particle is 500°F above gas temperature.

D. All unburned carbon particles are recycled to bed.

E. Gas residence time is 4 seconds (established from the SO_2 removal calculation).

The combustion efficiency calculated under the above assumptions was 99% per pass. This calculated efficiency level resulted primarily from the 4-second residence time assumption, which was established above by the SO_2 removal requirement.

3.3 NO_x FORMATION

NO_x emission was found to increase with the amount of excess air. Typical data are shown in Figures I-16, 17 and 18. An increase in temperature (in the 1400 to 1800°F range used in most FBC situations) or nitrogen content of the coal also tends to increase the NO_x emission. However, these effects are secondary compared to the effect of excess air. The presence of the sorbent used to control SO_2 emission in the bed, ($CaSO_4$) has been reported to reduce the NO_x emission (9), but the results of testing to verify this were not conclusive.

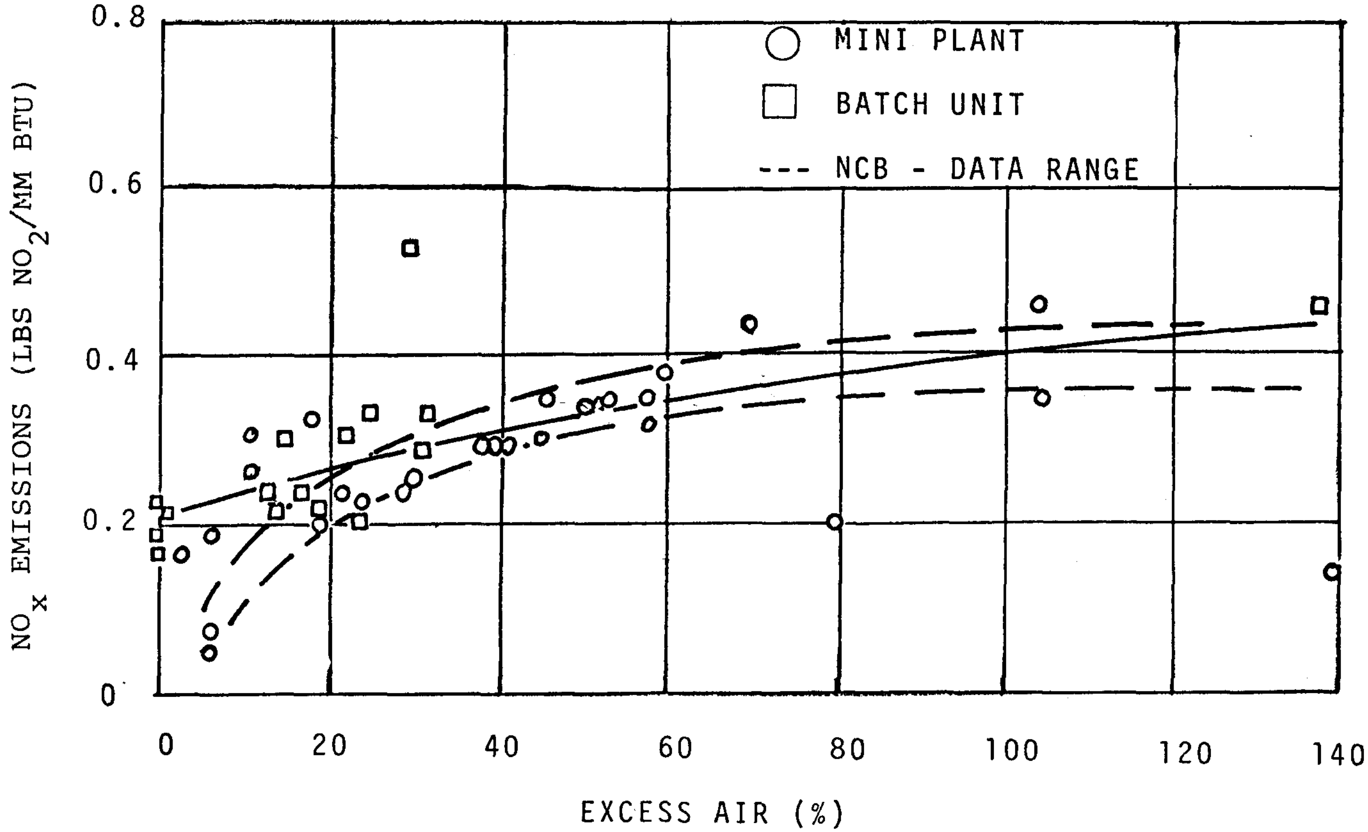


Figure I-16. - Typical Literature Data - Fluidized Bed Combustion NO_x Emissions vs Excess Air.

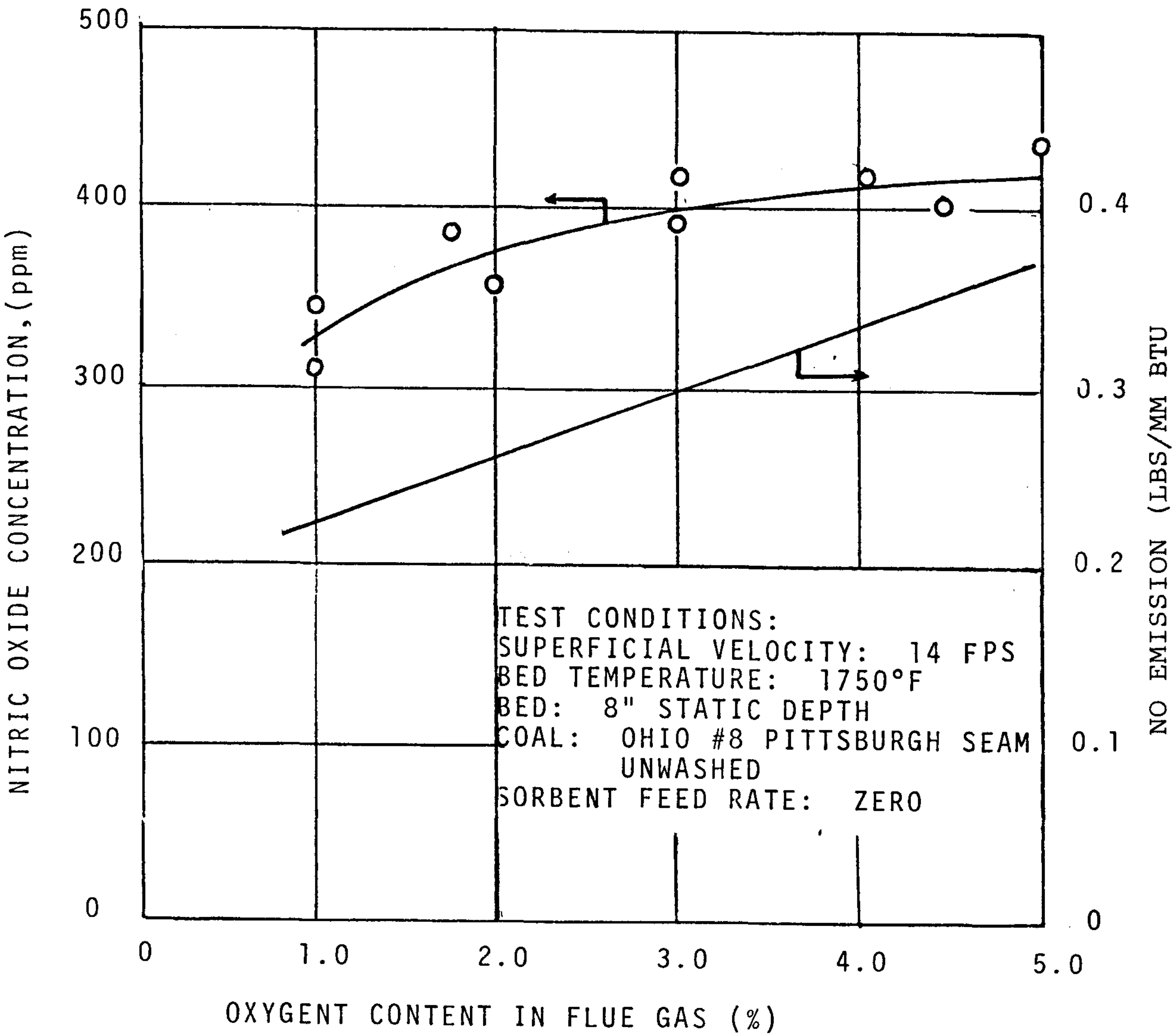


Figure I-17. - Typical Literature Data - Fluidized Bed Combustion NO_x Emissions vs Excess Air.

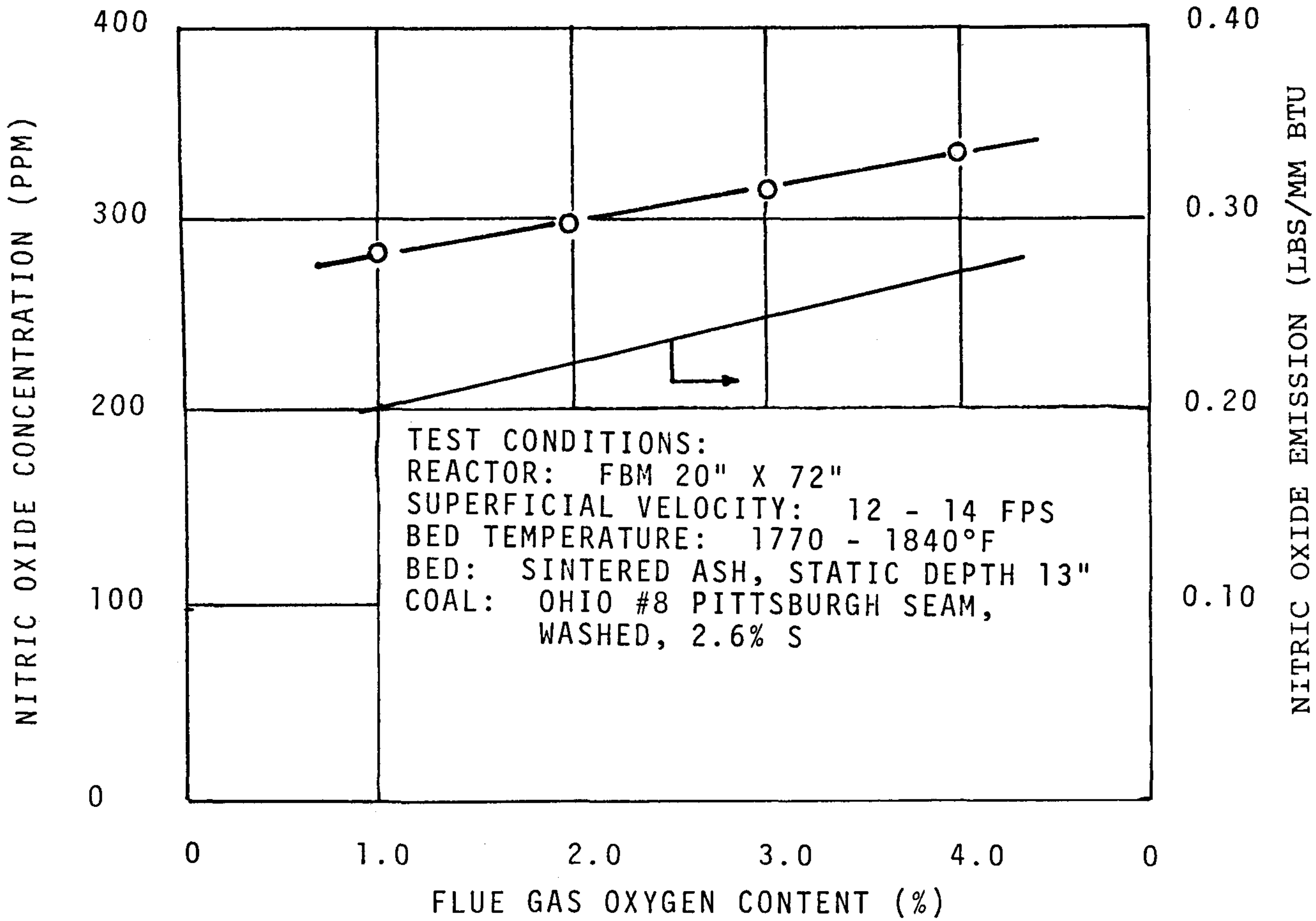


Figure I-18. - Typical Literature Data - Fluidized Bed Combustion NO_x Emissions vs Flue Gas Oxygen Content.

Generally, FBC data indicated that NO_x emissions are within the currently considered New Source Performance Standards (NSPS) established at 0.6 lb NO_x/MM Btu. The available data on NO_x emissions have been summarized in Table I-3.

4.0 HEAT TRANSFER

In the combustor designs that this study proposed, the heat exchanger surfaces were located in areas of dense fluidized beds (or bubbling beds), or they were located in less dense zones of circulating beds. For each type of bed, a specific heat transfer coefficient was assumed in order to size the heating surfaces.

4.1 HEAT TRANSFER COEFFICIENT IN DENSE FLUIDIZED BED

Dense fluidized beds are characterized by low gas velocities and a bed volume with distinct separation between bed and freeboard.

Data on the heat transfer coefficient for this type of bed have been generated and an example is shown in Figure I-19 (taken from Reference 10). These data were obtained in cold test measurements, in a 4-inch diameter bed using a cracking catalyst around 70 microns in average particle size. In the dense bed zone, a convective coefficient up to 65 Btu/hr $ft^2{}^\circ F$ was obtained. This coefficient varies with gas superficial velocity. Hot test data from Exxon (11) show values from 55 to 85 BTU/hr $ft^2{}^\circ F$ in pressurized units (batch and miniplant), with solids of about 200 micron particle size.

TABLE I-3.- NO_x EMISSIONS IN A FLUIDIZED BED COMBUSTOR

Research Organization	T (°F)	P (atm)	% excess air	NO_x Emissions lb NO /MMBtu
OCR	1600-1750	Pressurized	-	0.2
	1600-1750	Pressurized	5	0.07
ANL	1630-1665		-	0.15 - 0.4
PER				0.22 - 0.33
Consolidation Coal Company	1800	1.5	20	<0.2
Exxon R&E Co.	1460-1700	5-9	15	0.2 - 0.3

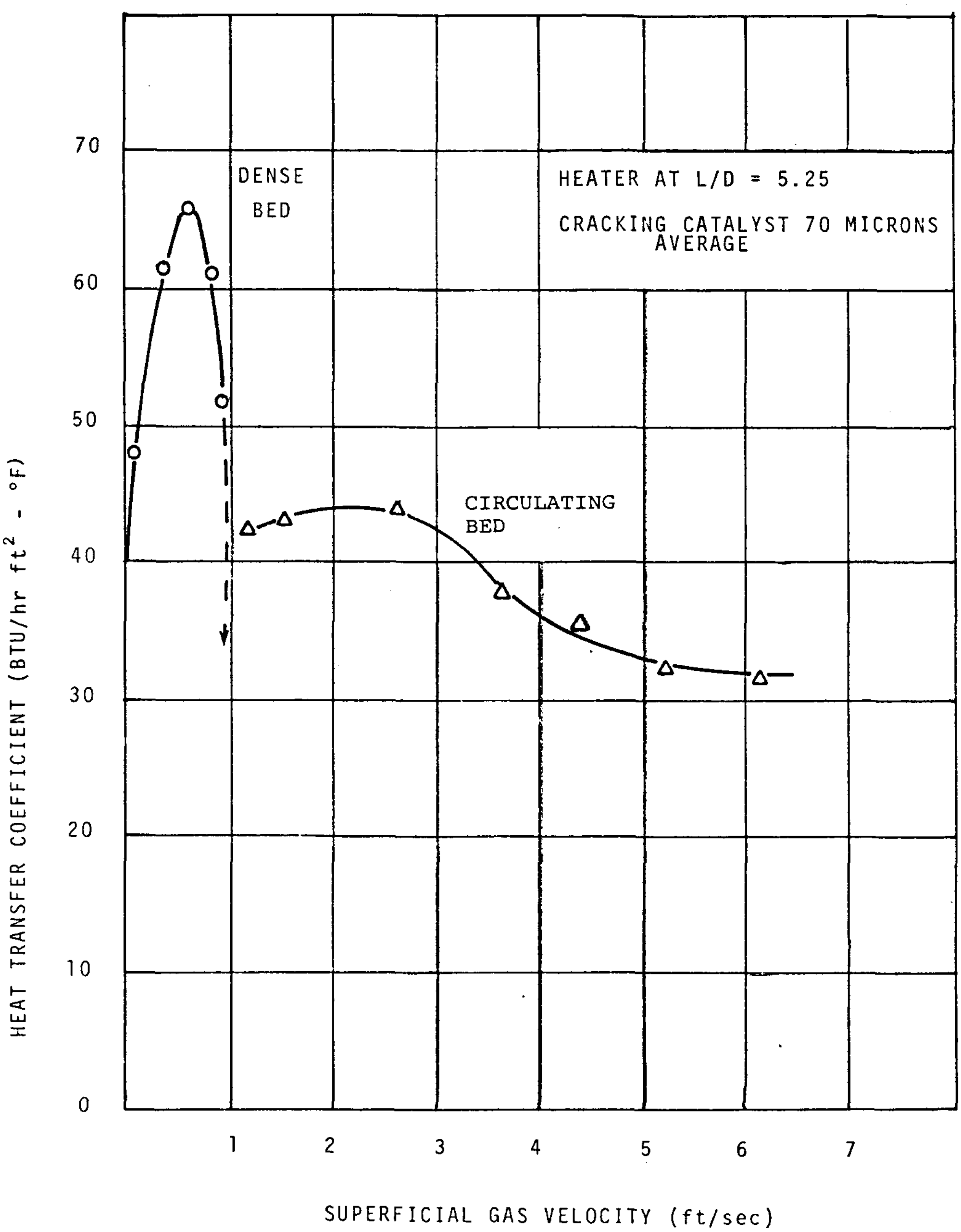


Figure I-19. - Typical Literature Data - Fluidized Bed Combustion Heat Transfer Coefficient vs Gas Velocity.

Some data (12) show even higher values for convective heat transfer coefficients. Jacob & Osberg reported coefficients up to 200 Btu/hr ft^2°F in cold test using glass beads of around 300 microns and fluidized in air (12).

Analysis of the available heat transfer data in dense fluidized beds, operating in the 50 to 300 micron range of particle size, underlines the fact that there is a problem in choosing an accurate design value for this coefficient. A conservative design could use a value of about 60 Btu/hr ft^2°F for the coefficient, or, a more optimistic approach could use up to 100 Btu/hr ft^2°F to reduce the total heating surfaces.

4.2 HEAT TRANSFER COEFFICIENT IN A CIRCULATING BED

Circulating beds operate at higher gas velocities than dense beds. The solids are entrained by the gas, and they circulate with no distinct separation between bed and freeboard.

Data about heat transfer coefficient from particle to a submerged heat transfer surface for this type of bed are shown in Figure I-20. A convective coefficient of up to 45 Btu/hr ft^2°F was reported in Battelle data. Other data are shown in the figure for particles averaging 50 microns including Kellogg data from the catalytic cracking reactors (13). The figure shows a dependency of heat transfer coefficient on bed density, the coefficient increases with higher solid concentration in the bed.

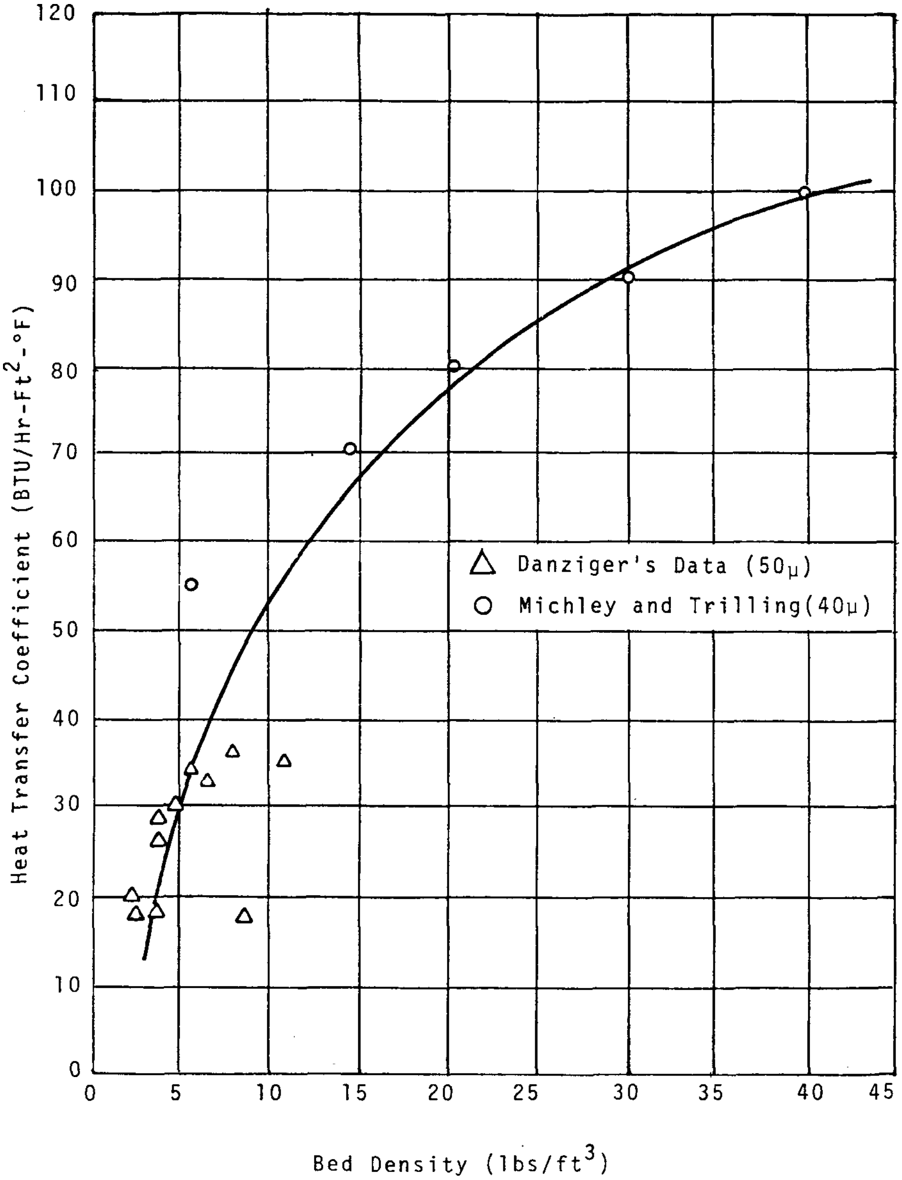


Figure I-20. - Typical Literature Data - Circulating Bed Heat Transfer Coefficient to Submerged Heat Transfer Surface vs Bed Density.

Typically, for a bed operating at 15 lbs/ft^3 density, a convective heat transfer coefficient of about 60 Btu/hr $ft^2{}^\circ F$ is used.

The convective coefficient should be combined with a radiant coefficient. There is no good procedure for calculating the negligent component. A first estimate is that at 1650°F the radiant heat transfer coefficient plus other temperature effects will add more than 40 Btu/hr $ft^2{}^\circ F$ to the convective coefficient.

5.0 SOLID CARRYOVER

Particulate emissions from the power plant, based on the New Source Performance Standard (NSPS) considered for this study, were restricted to 0.1 lb solid/10^6 Btu to satisfy the projected EPA standard. These requirements establish the performance criteria demanded of the solid collection system (cyclone, baghouse, etc.) used to control the plant particulate emissions.

The degree of solids removal achieved by solid collection equipment depends on the solid loading and particle size distribution fed to the equipment, and on the equipment's fractional collection efficiency.

5.1 SOLID LOADINGS TO CYCLONES

The first-stage cyclone handles materials entrained from the combustor itself. The particle size distribution of the solids carryover out of the bed is determined by the equilibrium particle size distribution reached in the bed. The solid loading depends on the bulk density selected for operation of the bed and on the design of the combustor outlet section.

5.1.1 Particle Size and Particle Size Distribution of Solid Carryover

The equilibrium particle size distribution reached in the bed depends on the particle size of the solid feed to the bed; on the performance of the cyclones; and on the change of particle size due to attrition and/or chemical reactions.

A. Solid feed - Typical particle size distributions of the coal feed from crusher and pulverizer mills are:

	Size	Weight Percent
Pulverizer	-100 U.S. Mesh (less than 150 microns):	95%
	-200 U.S. Mesh (less than 74 microns):	90%
	-325 U.S. Mesh (less than 40 microns):	84%
	-400 U.S. Mesh (less than 35 microns):	73%
Crusher	-20 U.S. Mesh (less than 800 microns):	90%
	-60 U.S. Mesh (less than 250 microns):	50%
	-150 U.S. Mesh (less than 90 microns):	20%

These distributions are plotted in Figure I-23. For the present it is assumed that limestone will demonstrate a similar behavior.

B. Attrition - Attrition is the break-up of materials by impact, usually at high velocities. Tests have been conducted to measure attrition in fluid beds (e.g., Westinghouse, Argonne National Laboratory, Particulate Solid Research Inc., and American Cyanamid). The results of these tests (mainly bench scale units), however, are still linked to the test set-up, and the attrition rates measured cannot be applied to general design, especially to large commercial units. The results are best used as a comparative index to rate the resistance to attrition of different materials. An example of this comparison, found in Reference 15, is presented below:

Tymochtel Dolomite; gas velocity 2.2 ft/sec. Attrition measured as weight percent loss from the bed (as fines) in 10 hrs - 47%.

Sulfated Dolomite; gas velocity from 2.2 to 3 ft/sec - 4 to 25 weight percent loss in 10 hrs.

Catalyst support material; gas velocity 2.2 ft/sec - 40% weight loss in 10 hrs.

Reference 16 contains data concerning attrition of a cracking catalyst taken over a three-month start-up period with a commercial fluid bed reactor that had two stages of cyclones. The reference also proposes a correlation for attrition (measured in terms of the amount of fines below 325 mesh present in the bed at equilibrium condition) versus cyclone inlet conditions (Figure I-21).

C. Prediction of equilibrium particle size in the bed - Because the particle size distribution in the bed at equilibrium is a complex function of feed size,

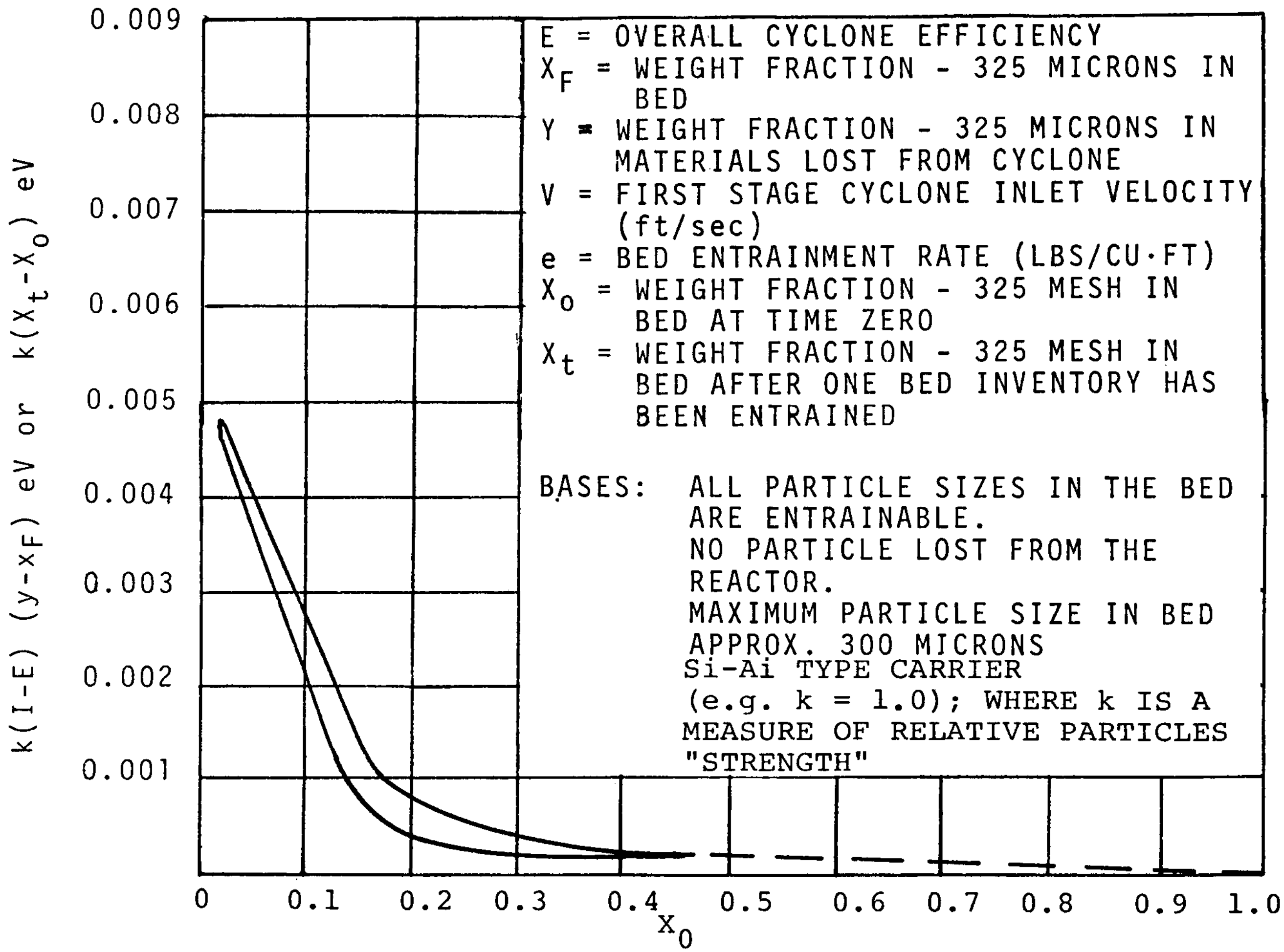


Figure I-21. - Typical Correlation of Rate of Fines Production in a Fluidized Bed. (Ref. 16) recycle, attrition, chemical reactions, etc., prediction of this distribution is not simple. However, the size distribution is required to determine the solid loading and performance of cyclones, and it is possible to approximate the equilibrium particle size distribution by assuming a similarity of behavior between the solids in the circulating bed combustor and the catalysts in catalytic cracking reactors. This assumption becomes more valid if the same types of cyclones are used in both applications (same ranges of inlet velocities and collection efficiencies); if the attrition resistance of both materials is not too different; and, if the feed particle sizes are similar. This becomes less important if the ratio of recycle solids to fresh feed solids in the circulating bed combustor is high. Using this assumption, the equilibrium particle size distribution in the circulating bed combustor can be related to the data obtained on the particle size history of catalysts in commercial catalytic cracking reactors. Figure I-22 develops typical data concerning feed and equilibrium particle size distributions in catalytic cracking, plotted on normal probability paper. As seen in the graph, the effects of attrition and elutriation result in rotating the line representing the particle size distribution of the fresh feed to a line representing the equilibrium particle size distribution of the bed. The average particle size (at 50 cumulative weight percent) of the bed at equilibrium remains at 70 microns, the same as the average size of the solid

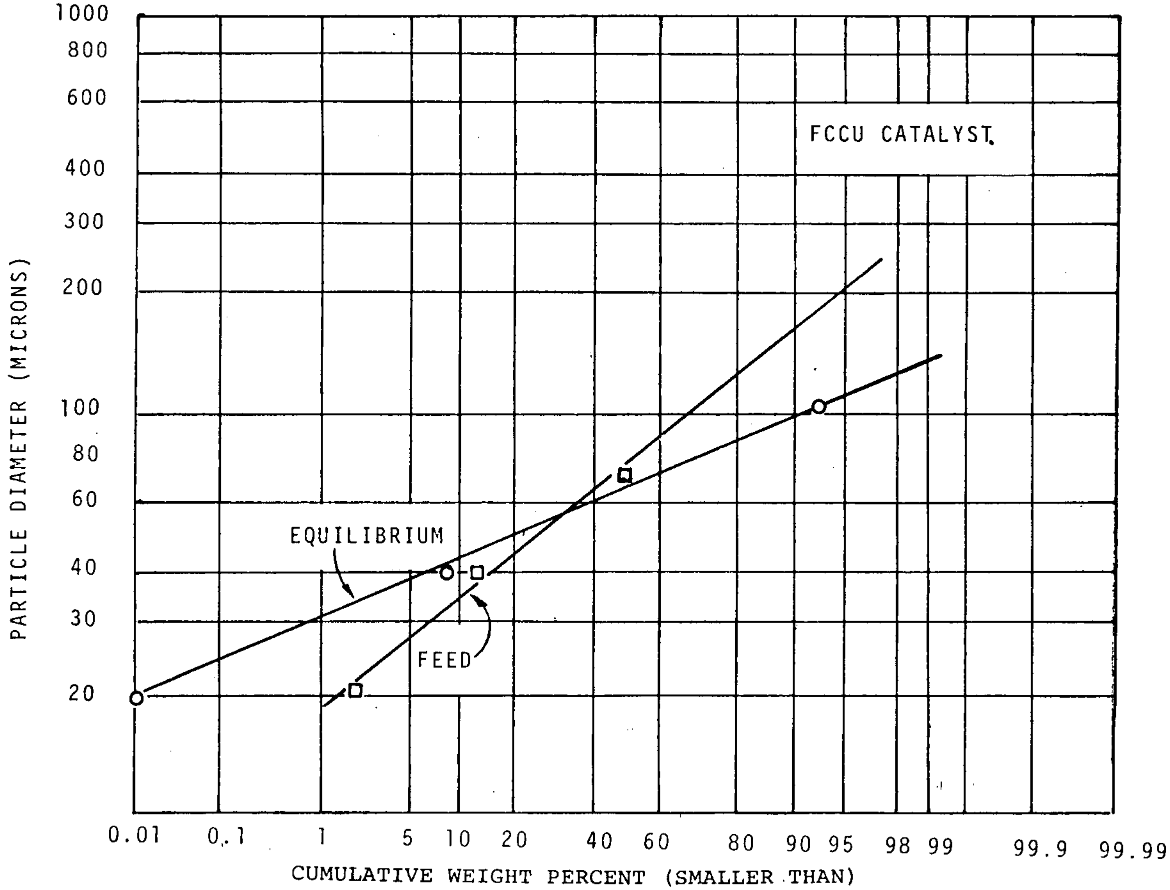


Figure I-22. - Particle Size Distribution for Fluidized Catalytic Cracking Catalyst - Feed Catalyst and Equilibrium Bed Catalyst.

feeds. The change in slope indicates that, although the average particle size does not change, the distribution has become more narrow. The net effect of attrition and elutriation has been to purge out fines, and break down heavy particles to reach a more uniform particle size distribution at equilibrium.

The same effect should exist in the circulating combustor, although the extent may or may not be the same. Starting from an initial feed distribution, the bed should achieve a more uniform particle size distribution at equilibrium. For a desired average particle size of about 100 microns, an initial feed particle size distribution should be as shown in Figure I-23, where it falls between the crushed and pulverized feed distributions. Because of attrition, elutriation and recycling, the line would rotate with the average particle size remaining at 100 microns to reach the equilibrium distribution line.

The amount of fines (-325 mesh) remaining in the bed at equilibrium would be very small. The resulting line represents an approximation of the particle size distribution at equilibrium in the combustor. The slope of the line was maintained the same as the equilibrium line in Figure I-22 for catalytic cracking and it indicates that the standard deviation or degree of uniformity is the same in both distributions.

The particle size distribution represented by the equilibrium line and Figure I-23 can be used as an approximation for the particle size loading to the

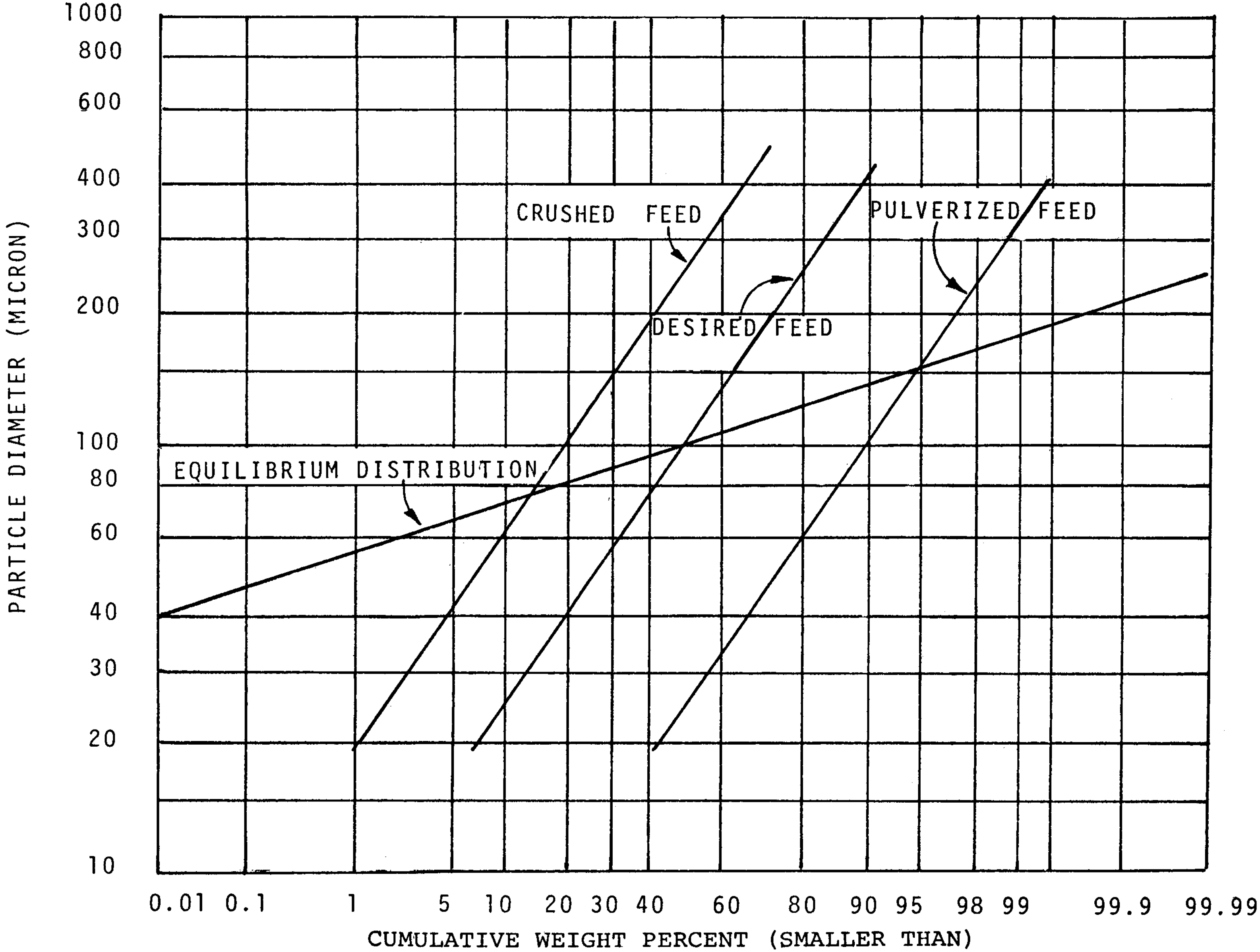


Figure I-23. - Calculated Particle Size Distribution for Circulating Bed Combustor-Limestone and Coal Feed and Equilibrium Bed.

first stage collection cyclones. This will be used until data can be developed to allow a better definition of this complex problem.

5.1.2 Density of Carryover to Cyclones

The density or amount of solids per unit volume of gas carried into the cyclone inlet sections depends both on the density of the circulating bed, and on the design of the combustor outlet before the cyclones. If a disengagement zone is provided prior to the cyclone inlets such as in Kellogg's catalytic cracker design where a freeboard zone with 180 degree change in direction of gas flow is provided just before the cyclone inlets, some partial solid separation may be possible (50% is a common figure), prior to the cyclones, thus reducing the solid density loading to the cyclones.

The bed densities considered in this study range from 4 lbs solid/ft^3 gas to 12 lbs/ft^3. The design cases corresponding to these densities are:

TABLE I-4.- SOLIDS LOADING TO THE CYCLONES

	Atmospheric Designs			Pressurized Designs	
Bed density (#/ft^3)	4	12	12	12	12
With or without disengagement zone	with	with	without	with	without
Solid density to cyclones (#/ft^3)	2	6	12	6	12

As indicated in the table, when the design includes a solid disengagement zone prior to the cyclone, a 50% pre-cyclone solid cleaning is assumed. These solid densities to the cyclones, and the volume flow of the gas corresponding to each case, determine the solid loadings (in lbs/hr) to the cyclones.

5.2 PERFORMANCE OF CYCLONES

Overall collection efficiencies of cyclones can vary from 80% to 90% in low performance cyclones to as high as 99.995% in high performance units such as the types used in catalytic cracking. The high performance cyclones are necessary for the circulating bed boiler application, to deal with the relatively high solid loadings associated with circulating beds.

5.2.1 Cyclone Fractional Efficiency Curves

The collection efficiences of cyclones as functions of particle size loading are given by the cyclone fractional efficiency curves. Examples of the curves are shown in Figures I-24 and I-25 for two high performance cyclones. The cyclone in Figure I-24 shows 100% collection efficiency for all particle sizes above about 120 microns. The cyclone in Figure I-25 is even more efficient, it shows 100% efficiency for everything above 80 microns.

5.2.2 Cyclone Performance and Carryover

The performance of the combustor first-stage cyclones can be determined by applying the collection efficiency curves of Figure I-24 or Figure I-25 to the particle size loading to the cyclones. This is shown by the

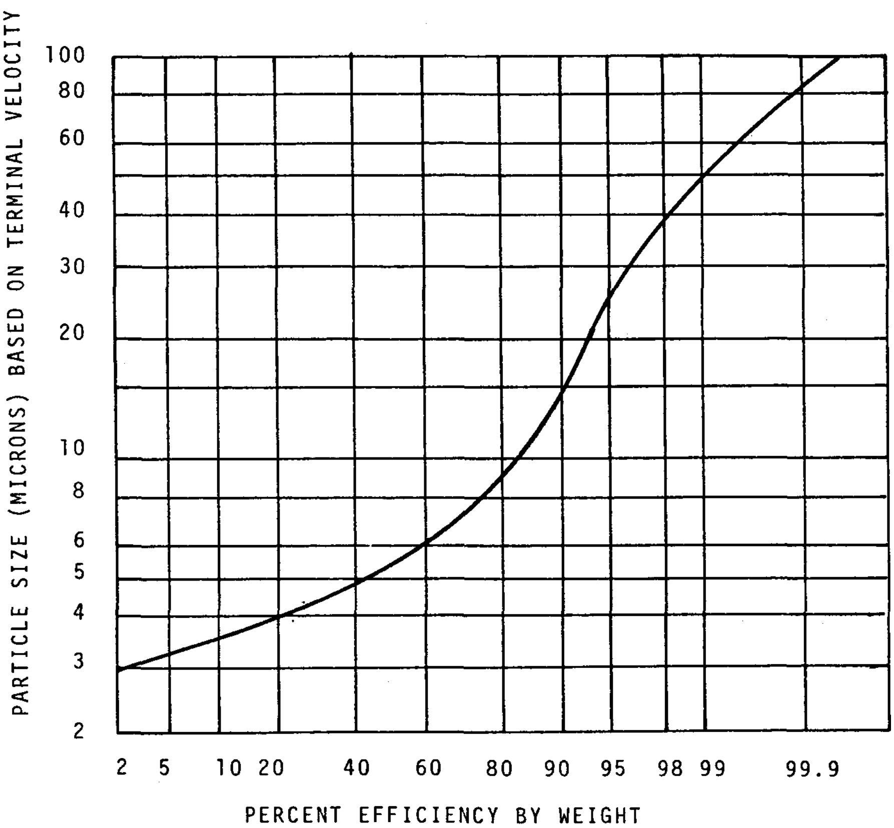


Figure I-24. - Typical FCCU Cyclone Fractional Efficiency Curve - Cyclone A.

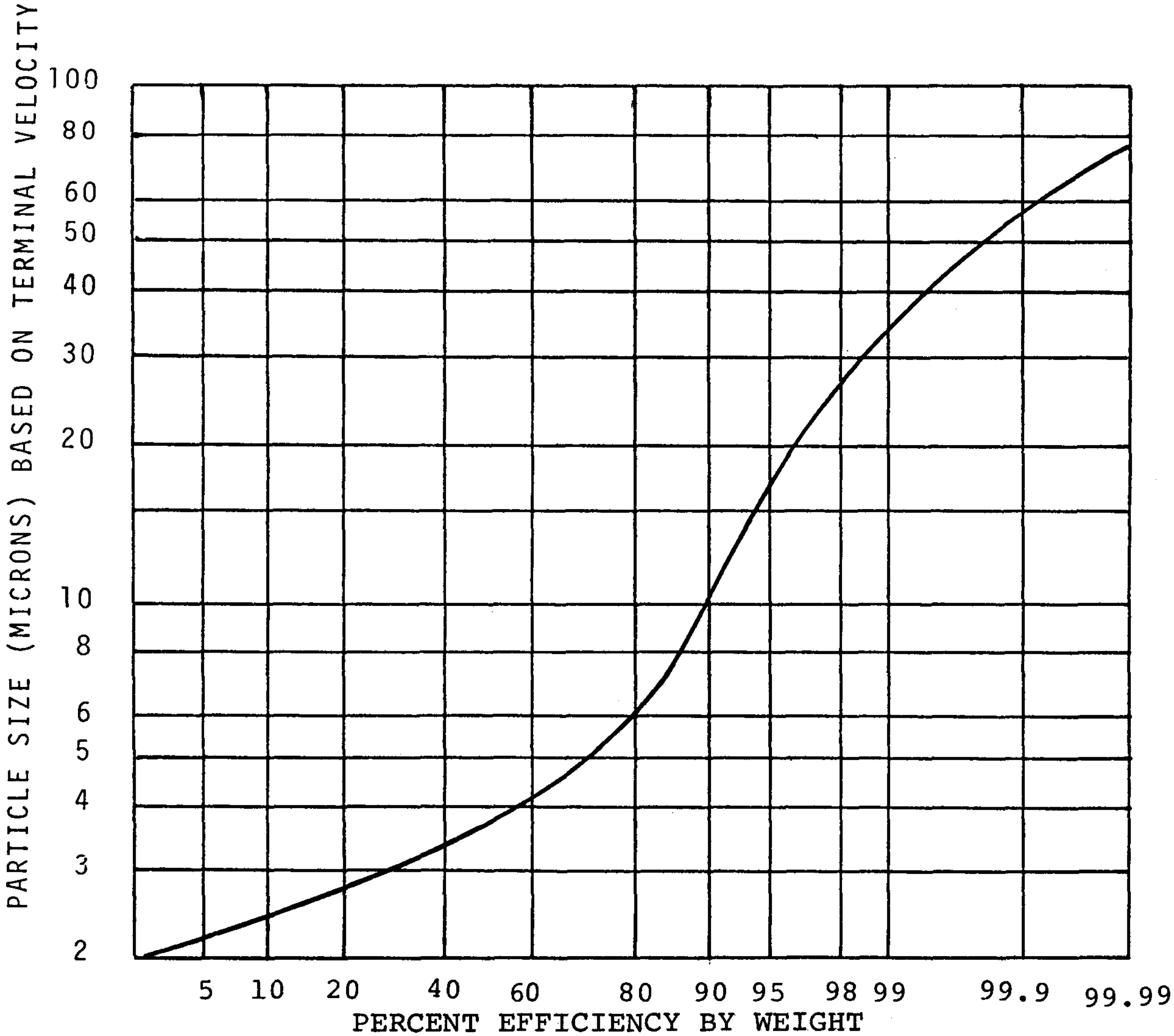


Figure I-25. - Typical Cyclone Fractional Efficiency Curve - Cyclone B.

equilibrium line of Figure I-23 and redrawn in Figure I-26. The results of this operation are significant to this study and are described in the following paragraphs:

A. Overall collection efficiencies - The overall solid collection efficiencies obtained for the cyclones studied at these particle size loadings are:

-99.95% efficiency for the cyclone of Figure I-24
-99.995% efficiency for the cyclone of Figure I-25

B. Cyclone carryover - The carryover from the combustor first stage cyclones (both amount and particle size distribution) will constitute the solids loading to subsequent solids cleaning equipment.

1. Particle size distributions - The particle size distribution of the solids carried out of the first stage cyclone system can be determined from the particle size loading to the cyclone and the cyclone fractional efficiency curve. The particle size distributions obtained are shown in Figure I-26 for the cyclones of Figure I-24 and I-25. As the figure shows, although the cyclone carryover contains some amount of fines the bulk consists of particles between 40 and 100 microns in size. This small amount of fines present in the cyclone carryover is a direct result of the very small fraction of fines in the bed equilibrium mixture.

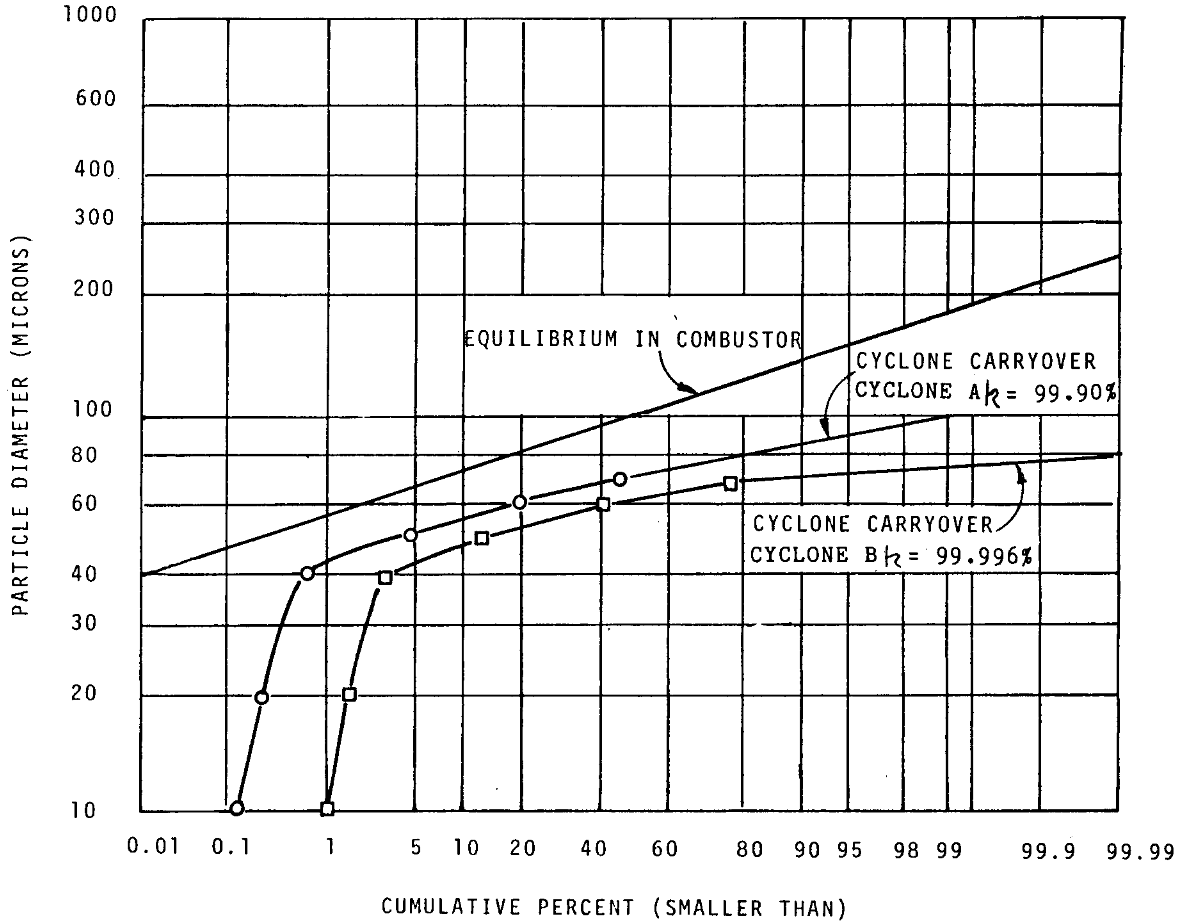


Figure I-26. - Particle Size Distribution Leaving Cyclones.

2. Density of cyclone carryover - The solids loss from the combustor first stage cyclone system can be established directly from the density loadings to the cyclones and the cyclone overall collection efficiency. The results are summarized in Table I-5.

5.3 SUBSEQUENT SOLID CLEANING EQUIPMENT

The solid loading after the first set of cyclones, as shown in Table I-5, still exceeds the particulate emission requirements of 0.1 to 0.05 lb solid/10^6 Btu fuel. To meet these standards, additional solid collecting equipment has to be provided. Also, in the pressurized combustor plant design, special solid cleaning equipment is required to meet the particulate loading limitations of the power recovery gas expander.

5.3.1 Particulate Emission Control for Atmospheric Combustor Plants

Because of the high solids densities involved in atmospheric combustors, only the very high performance cyclones (99.995%) will be considered for particulate emission control. Using these cyclones, the solid entrainment after the cyclones for loadings between 2 and 12 lb/ft^3 will be between 4.95 and 29.73 lb solid/ 10^6 Btu (as shown in Table I-5). The particle size distribution of the entrainment is as shown in Figure I-26. The collection efficiencies required of additional solids cleaning equipment to achieve the particulate emission standard of 0.05

TABLE I-5.- SOLIDS ESCAPING FIRST STAGE CYCLONE

	Atmospheric Design			Pressurized Design	
Density loading to cyclones (from Table 3) (lbs/ft^3)	2	6	12	6	12
Cyclone carryover (#/MM Btu fuel) with 99.95% collection efficiency	49.5	148.7	297.3	22.3	44.6
Cyclone carryover (#/MM Btu fuel) with 99.995% collection efficiency	4.95	14.87	29.73	2.23	4.46

lb/10^6 Btu (projected NSPS) are tabulated in Table I-6, and vary between 98.99% and 99.85%. These collection efficiencies fall in the range of efficiencies achievable in baghouse.

For particle sizes from 0.1 to 10 micron the baghouse provides collection efficiencies from 99 to 99.9%. Therefore, the final particulate emission level of 0.05 lbs solid/10^6 Btu can be achieved by installing baghouses before the stack gas exhaust.

5.3.2 Particulate Control in a Pressurized Combustor Plant

When power recovery gas expanders are used in the pressurized combustor plant, the solid loadings in the gas flow to these expanders (Table I-5) have to satisfy the particulate limitations of the machines. Typical particulate requirements for these machines are:

Particle distribution	
Size (microns)	% less than
80	100
40	99.9
20	96.9
15	95.5
12	94.0
10	92.6
5	84.4

Solid concentration: 3×10^{-4} # solid/#gas (To reach this concentration would require cleaning the cyclone carryover to 0.3 # solid/10^6 Btu).

A proprietary inertial separator was investigated in this study to achieve the required degree of solid cleaning. A typical performance curve for equipment of this type is shown in Figure I-27. For the particle loading shown in Table I-5 (using the 99.995% efficiency cyclone) and the size distribution of Figure I-26, the performance obtained with the inertial separator is:

- overall collection efficiency; about 96%. The solid carryover from this separator would be from 0.09 to 0.18 # solid/10^6 BTU (this is summarized in Table I-5).
- particle size distribution out of the separator;

Size (microns)	% less than
30	99.99
20	97.14
10	87.0

Use of the inertial separator should achieve the degree of solid cleaning required by the gas expander's manufacturer.

After expansion through the gas expanders, the gas can be cleaned to the final particulate level using a baghouse. Table I-6 gives a summary of the different stages of solid cleaning for both atmospheric and pressurized boiler plants.

6.0 MISCELLANEOUS STUDIES

The life of heat exchanger tubes immersed either in dense fluidized beds or in circulating beds of

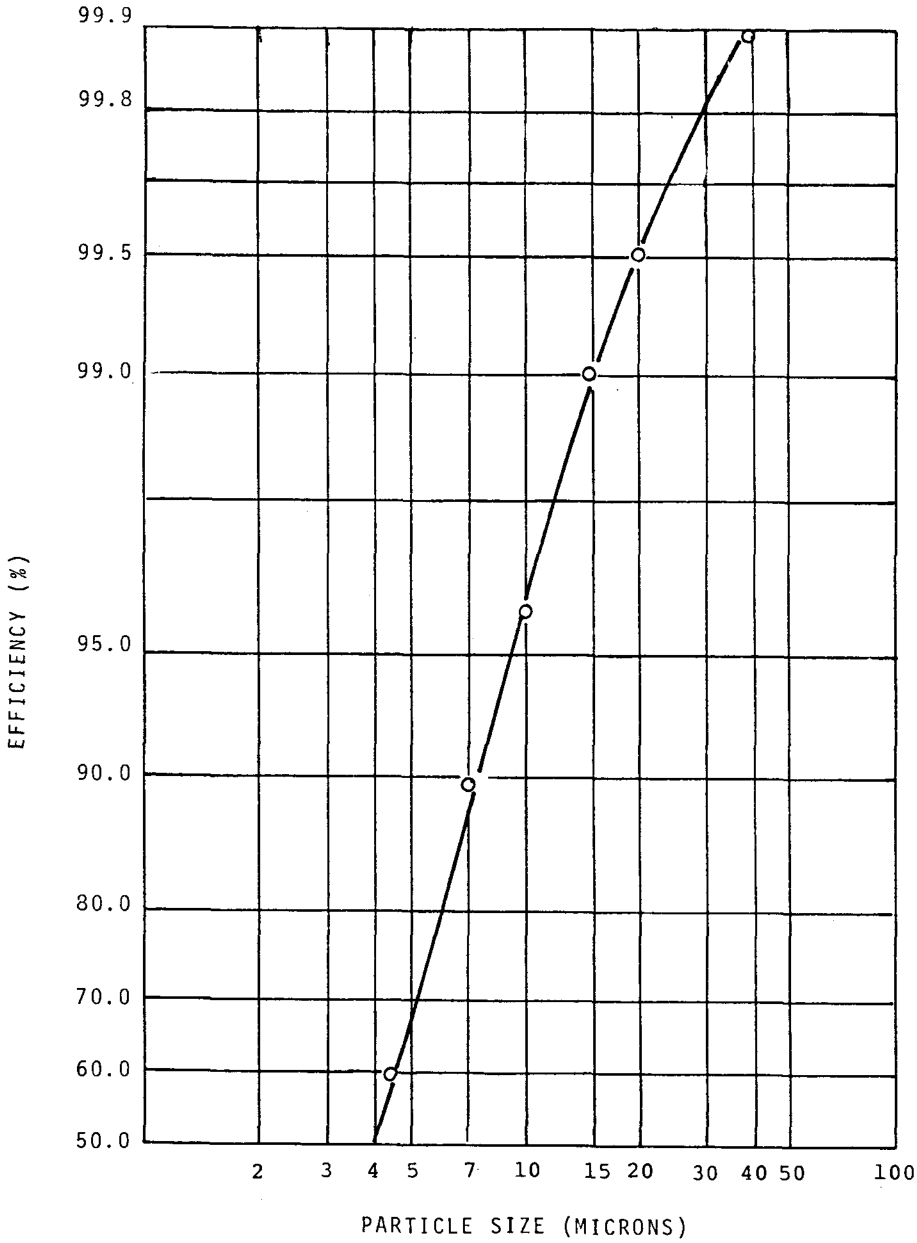


Figure I-27. - Typical Fractional Efficiency Curve of Proprietary Inertial Separator.

TABLE I-6.- SUMMARY OF SOLID CLEANING STAGES

	Atmospheric Boiler Plant			Pressurized Boiler Plant	
First Stage Cyclones:					
• density loadings ($\#/ft^3$)	2	6	12	6	12
• efficiency (%)	99.995	99.995	99.995	99.995	99.995
• carryover (#/MM Btu)	4.95	14.87	29.73	2.23	4.46
Proprietary inertial separator					
• efficiency (%)		not used		96.0	96.0
• carryover				0.09	0.18
Baghouse					
• efficiency (%) (req'd)	98.99 (req'd)	99.66 (req'd)	99.83 (req'd)	45.0 (req'd)	72.3 (req'd)
• carryover	0.05	0.05	0.05	0.05	0.05

limestone or dolomite and various coals depends on their resistance to corrosion, erosion, and fouling of the heat transfer surfaces. Some experimental data concerning these problems are available (17). The experimental data indicated that conventional boiler tube materials are suitable for fluidized bed combustion boiler design without encountering exceptional corrosion, erosion, or fouling problems.

APPENDIX I

REFERENCES

1. Lange, H.B., "Fluidized Bed Combustion Test Facility," paper presented to FBC Technology Exchange Workshop, Reston VA, April 1977.

2. Robinson, E.B., et al., "Characterization and Control of Gaseous Emissions from Coal-Fired Fluidized-Bed Boilers," Interim Report to Department of HEW, October 1970, NTIS-PB 198413.

3. Keairns, D.L., et al., "Fluidized Bed Combustion Process Evaluation-Phase II-Pressurized FBC Development," Westinghouse Research Laboratories, EPA 650/2-75-027C, September 1975.

4. Hoke, R.C., et al., "Studies of the Pressurized FBC Process," Exxon Rand EG., EPA-600/7-77-107, September 1977.

5. Hoke, R.C., et al., "Combustion and Desulfurization of Coal in a Fluidized Bed of Limestone," Combustion, January 1975.

6. Borgwardt, R.H., "Kinetics of the Reaction of SO_2 with Calcined Limestone," Environ., Scien. & Techn., p.59, 4, 1, January 1970.

7. "Atmospheric Fluidized Bed Combustion EPRI Workshop," Meeting Report, August 25-26, 1977.

8. Field, M.A., et al., "Combustion of Pulverized Coal," Cheney & Sons Ltd., Banbury, England, 1967.

APPENDIX I

REFERENCES

9. Hammons, G. and Skopp, A., "A Regenerative Limestone Process in FBC and Desulfurization," Esso R. and E., Linden, NJ.

10. Kiang, K.D., et al., "Heat Transfer in Fast Fluidized Beds," paper presented at International Conference on Fluidization, Asilomar, California, June 1975.

11. Hoke, R.C., et al., "Studies of the Pressurized Fluidized-Bed Coal Combustion Process," Exxon R. and E. Co., NTIS, PG-260478, September 1976.

12. Jacob, A. and Osberg, G.L., Cand. Jour. Chem. Engng., 35, 5-9, June 1957.

13. Danziger, W.K., I & EC Process Design and Development, Volume 2, No. 4, October 1963.

14. Mickley, H.S., Trilling, C.A., Ind. Eng. Chem., 41, 1135, 1949.

15. Vogel, G.J., et al., "A Development Program on Pressurized Fluidized-Bed Combustion," ANL, NTIS, ANL/ES-CEN-1016, June 1976.

16. Zenz, F.A., "Find Attrition in Fluid Beds," Hydrocarbon Processing, February 1971.

17. Thomas, D.E., et al., "Energy Conversion Alternatives Study Le Cast," Volume II - Materials Considerations, Westinghouse Electric Corporation, Pittsburgh, PA, February, 1976.

APPENDIX II

OTHER ATMOSPHERIC CASES

1.0 ATMOSPHERIC CIRCULATING BED BOILERS WITH DENSE BEDS

When combustors operate at atmospheric pressure, the volume flow rate of flue gas is large, and they require a large number of high efficiency cyclones. Hopefully, in the near future, a high capacity and efficiency cyclone will be developed for this application. The immediate problem associated with the large number of cyclones is the difficulty associated with the plant arrangement to accommodate the cyclones, and their inlet duct and leg connections. As shown in Figures II-1 and II-2, the size of the dense bed is increased compared to the single dense bed described in the main text to accommodate the space necessary for connection and installation of the cyclones.

The system described has 96 cyclones. Also, there are 10 combustors, each situtated between dense beds. An individual limestone and coal bin is provided for each combustor. Fresh limestone and coal are fed into the bottom of the combustor. The air from the compressors flows into the bottom of the combustor and entrains solids as it flows upward into the combustor. The returned solids from the dense beds flow into the combustor from both sides. The combustion starts at the bottom and continues as the solids-gas mixture flows upward. There are no heat exchanger tubes inside the combustor since there is ample space available in the dense-beds for accommodating the required heat exchanger tubes. The solids-gas stream is divided into

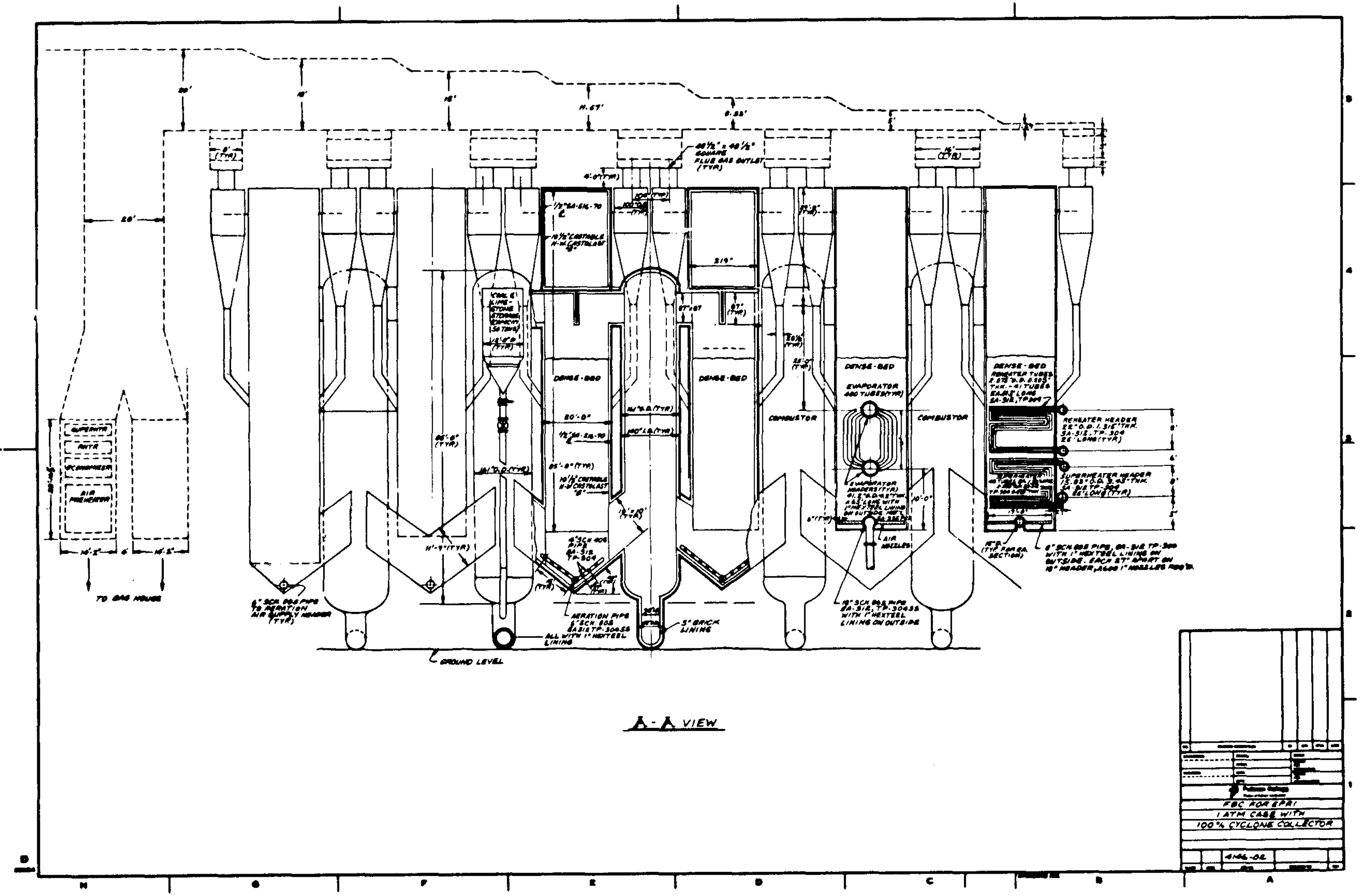


Figure II-1. - Circulating Bed Combustors with Dense Bed - Elevation.

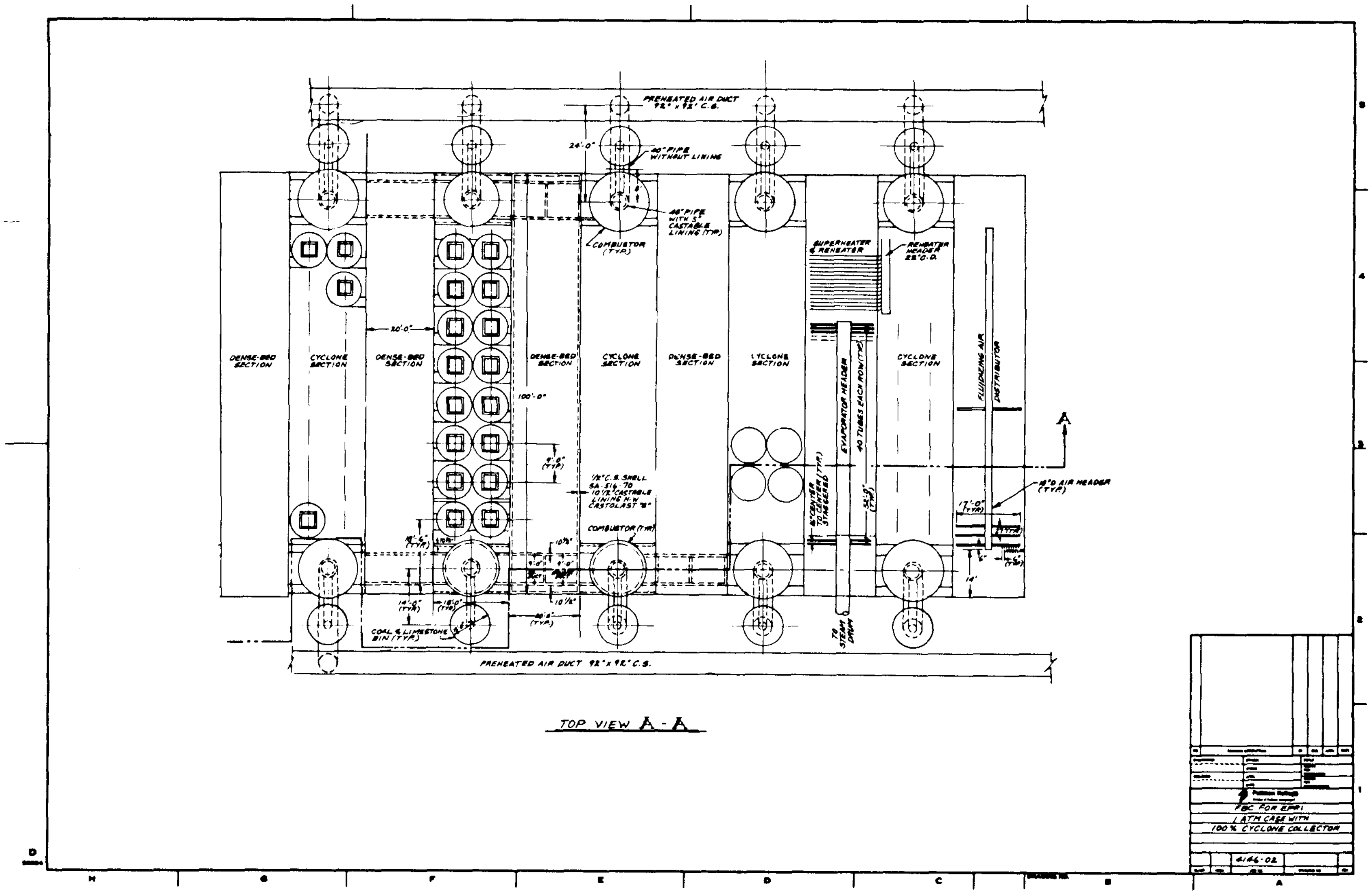


Figure II-2. - Circulating Bed Combustors with Dense Bed - Top View.

two streams at the combustor top. Each stream enters the freeboard space in the separate dense-bed housing where approximately half of the solids are disengaged and discharged to the dense bed. The flue gas and the remaining solids enter the cyclones. The flue gas continues it's flow to the convection furnace and the solids are returned to the dense bed through the cyclone legs. Solids in the dense bed are recycled back to the combustor through two ducts. Aeration nozzles at the bottom of the dense bed and the ducts are used for fluidizing the solids. The gas velocity in the dense bed is 0.5 to 1 fps. The flue gas leaving the cyclones is collected in a common duct and then conveyed to the convection section, baghouse and finally the stack. The dense beds are 20 feet in width by 100 feet in depth. The air used in fluidizing the dense beds adds an additional load both to the air compressors and the cyclones. The calculated system pressure drop distribution is:

Acceleration loss		4.85 psi
Static head loss in combustor		8.3 psi
	TOTAL	13.2 psi

This calculation does not include the 1 psi pressure head loss down stream of the combustor.

1.1 COMBUSTOR

The combustor design data are shown in Tables II-1 through II-3. The number of combustors in this design is determined principally by the necessity to match the arrangement with the dense bed. The number of dense beds is determined by the space needed for the cyclones. Once the number of combustors is determined and the flue gas flow rate is established, the diameter of the combustor is set by selecting a gas velocity. In selecting the velocity, the factors left to be considered are; (1) low system pressure drop, (2) reduction of the combustor diameter, and (3) avoidance of extreme erosion of combustor internals.

TABLE II-1.- ATMOSPHERIC COMBUSTOR DESIGN WITH DENSE BED DATA

Item	Design Data
Number of Combustors	10
I.D.	12'-0"
Height	100'-0"
Solids Density	12 lbs/ft^3
Gas Velocity	34 fps
Actual Gas Residence Time	2.85 sec
Required Gas Residence Time	1.2 sec
Flue Gas Flowrate (lbs/hr)	5,254,700
Inlet Air Pressure	14 psig
Combustor Operating Temperature	1,650°F
Material of Construction;	
Shell	3/4" t SA-517-70 C.S. plate clad with 316 SS
Lining	10 1/2" Harbinson-Walker Castolast "G"

1.2 HEAT EXCHANGERS

As shown in Figures II-1 and II-2, the evaporator, superheater and reheater tubes are equally distributed among the six dense beds. Consequently, each dense bed contributes one sixth of the total heat duty of each heat exchanger. The design data are shown in Table II-2.

TABLE II-2.- HEAT EXCHANGER DESIGN DATA FOR ATMOSPHERIC COMBUSTOR WITH DENSE BED

Parameter	Evaporator	Superheater 2	Reheater 2
Heat Duty (10^9 Btu/hr)	1.275	1.225	0.525
LMTD (°F)	960	759.2	772.7
Steam Temp. (°F) Inlet	690	770	740
Outlet	690	1000	1000
Overall Heat Transfer Coefficient (Btu/hr-Ft^2 °F)	----------60----------		
Heat Transfer Surface (Ft^2)	22,140	26,900	11,330
Tube Size I.D.	2.875"	2.375"	2.875"
O.D.	2.142"	1.425"	2.465"
No. of Tubes	2400	288	246
Length of Tubes	14'	150'	61.5'

1.3 CONVECTION

The design data of the convection section is shown in Table II-3, and the configuration is shown in Figure II-3.

TABLE II-3.- DESIGN DATA FOR ATMOSPHERIC COMBUSTOR WITH DENSE BED-CONVECTION SECTION

Parameter	SH#1	RH#1	ECON	APH
Heat Duty (10^9 Btu/hr)	.400	.25	1.025	.308
LMTD (°F)	781	609.3	205.5	92.8
Heat Transfer Surface Area (Ft^2)	16,900	77,000	481,000	1,300,000
No. of Tubes x Length of Tubes	147 97'-9"	72 97'-9"	468 97'-9"	17,280 100'
Tube Size	4" XX STG	4" SCH40*	4" SCH160*	4" 9 GA
Tube Material	316 SS	C.S.	316 SS	C.S.

*1/2" x 0.05" C.S. Helical Serrated Fins, 6/inch

2.0 <u>CIRCULATING BED BOILER WITHOUT DENSE BED</u>

The CBB without a dense bed case differs from the previous case in that there is no dense bed. Since there is no need to consider the relative position of

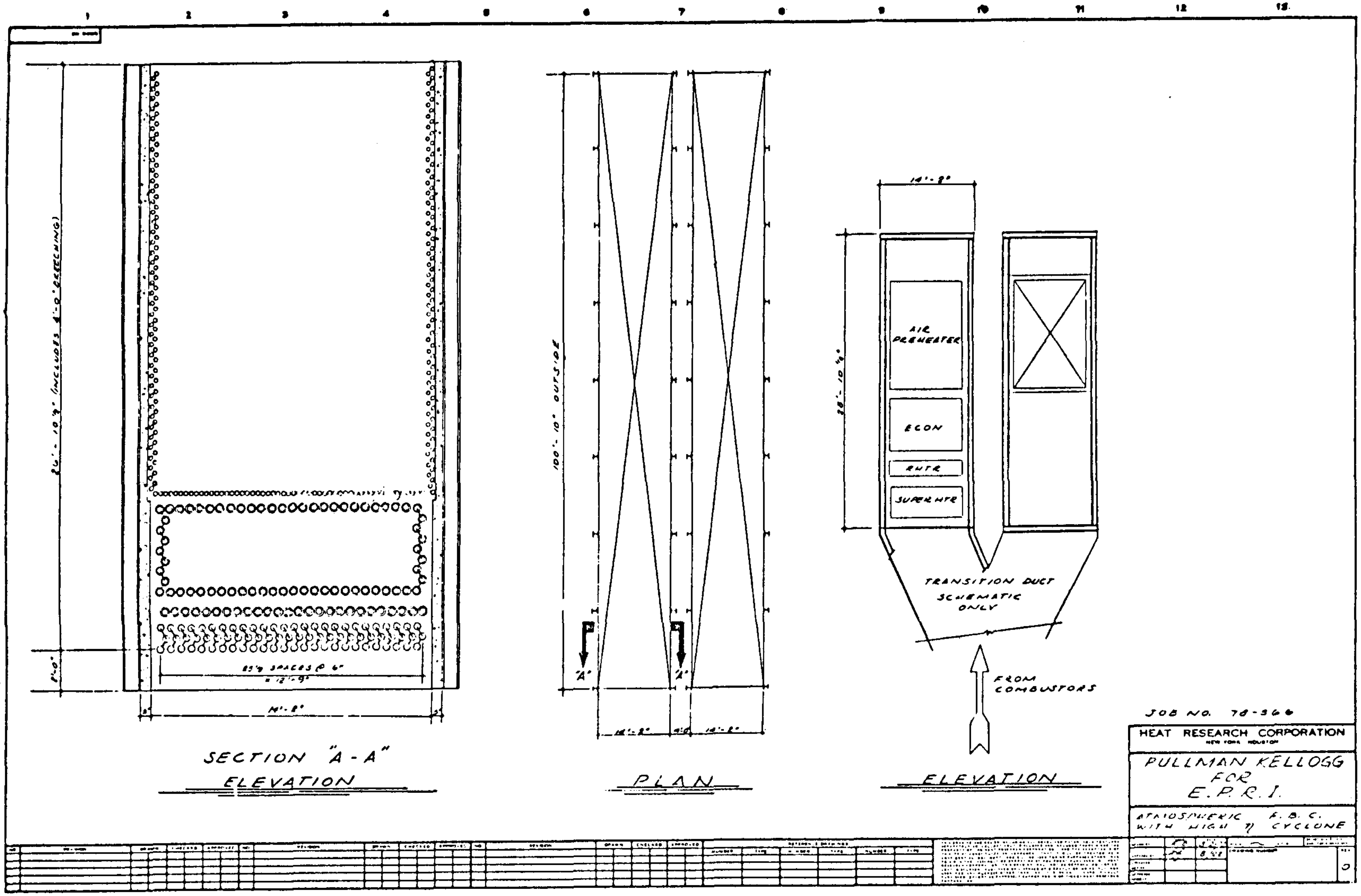


Figure II-3. - Convection Section for Atmospheric Circulating Bed Combustor with Dense Bed.

the combustor to the dense bed, the number of combustors is decreased to four. However, the same 96 cyclones are needed since the flue gas flow rate is the same. As shown in Figure II-4, four combustors operate in parallel. All required evaporator, superheater, and reheater tubes are equally distributed in the combustors. Each combustor is connected to two limestone and coal feed lines. Air ducts are connected to the combustor bottom. The limestone and coal feed pipes are connected to the air duct so that solids feed is entrained by the air and blown into the combustor. The combustion begins and continues as the solids-gas mixture flows upward through the combustor. The mixture passes the heat exchanger surfaces and then leaves the combustor at the top discharging into the cyclones. The flow is the same as explained in the previous design and will not be repeated here. Flue gas leaving the cyclones is collected in a common duct which leads to the convection section. The calculated system pressure drop distribution is:

Acceleration		8.0 psi
Static head loss in combustor		9.6 psi
	TOTAL	17.6 psi

2.1 COMBUSTOR

There are four combustors operating in parallel. The design data are shown in Table II-4.

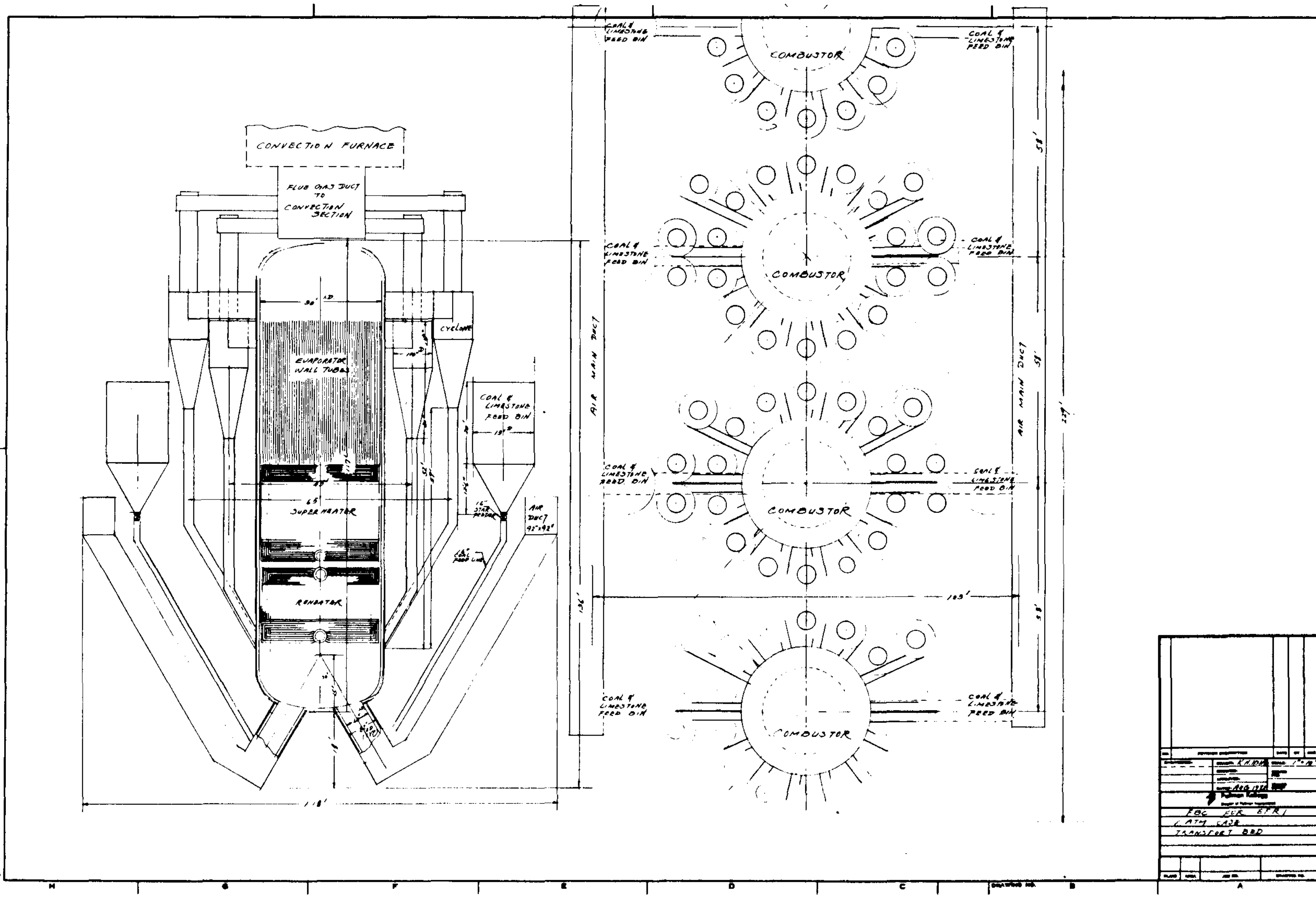


Figure II-4. - Circulating Bed Combustors without Dense Bed.

TABLE II-4.- ATMOSPHERIC COMBUSTOR WITHOUT DENSE BED DESIGN DATA

Item	Design Data
Number of combustors	4
I.D.	30'-0"
Height	115'-0"
Gas Velocity	19.5 fps
Solids Density	12 lbs/ft^3
Actual Gas Residence Time	6.1 sec
Required Gas Residence Time	1.2 sec
Flue Gas Flow rate	5,254,700 lbs/hr
Inlet Air Pressure	18 psig
Combustor Operating Temperature	1,650°F
Material of construction	
Shell	1" t SA-517-70 C.S. plate clad with 316 SS
Lining	10 1/2" t Harbinson-Walker Castolast "G"

2.2 EVAPORATOR, SUPERHEATER AND REHEATER

As shown in Figure II-4, the evaporator tubes are installed in the combustor wall. The superheater and reheater tubes are installed in the circulating bed. Detailed data of these heat exchangers are shown in the Table II-5.

TABLE II-5.- HEAT EXCHANGER DATA FOR ATMOSPHERIC COMBUSTOR WITHOUT DENSE BED

Item		Evaporator	Superheater 2	Reheater 2
Heat Duty (10^9 Btu/hr)		1.275	1.225	0.525
LMTD (°F)		960	759.2	772.7
Steam Temp. (°F)	Inlet	690	770	740
	Outlet	690	1000	1000
Overall Heat Transfer Coefficient (Btu/hr-Ft^2°F)		-----------------60-----------------		
Heat Transfer Surface (Ft^2)		22,140	26,900	11,330
Tube Size	I.D.	2.875"	2.375"	2.875"
	O.D.	2.142"	1.425"	2.465"
No. of Tubes		210	288	246
Length of Tube		80'	150'	61.5'

2.3 CONVECTION SECTION

The convection section is identical to that described in the previous design.

3.0 CIRCULATING BED BOILER WITH LOW EFFICIENCY CYCLONES

In this design solids carryover from low-efficiency cyclones cannot be neglected. They must be collected and returned to the combustor through an additional solids collection system, downstream of the cyclones. In this design, multiclones are used. Also for this design, heat transfer surfaces are not installed in the circulating bed combustor due to a conclusion derived from energy balance studies. Instead, they are installed in the dilute solids-gas stream between the cyclones and the multiclones. The details of the design is shown in Figures II-5, 6, 7 and 8. Figure II-9 shows the corresponding T-Q diagram. The primary deficiency in this system is that all heat transfer surfaces are installed in a low solids density stream which results in a lower heat transfer coefficient when compared to the corresponding 60 Btu/hr-ft^2°F heat transfer coefficient inside the combustor. This increases the size of the heat exchanger surfaces, but only by 12%. The other drawback with this system is, as can be seen from the T-Q diagram, the decrease in the logarithmic mean temperature difference between the flue gas and the heat transfer surfaces. This also contributes to the larger heat exchanger size. A final drawback is in the need for additional solids collection equipment and the solids return line which increases both the capital cost and cost of operation.

The advantages of operating a combustor without tubes should not be ignored. There are 12 combustors and 24 cyclones in this system. Twelve limestone and coal

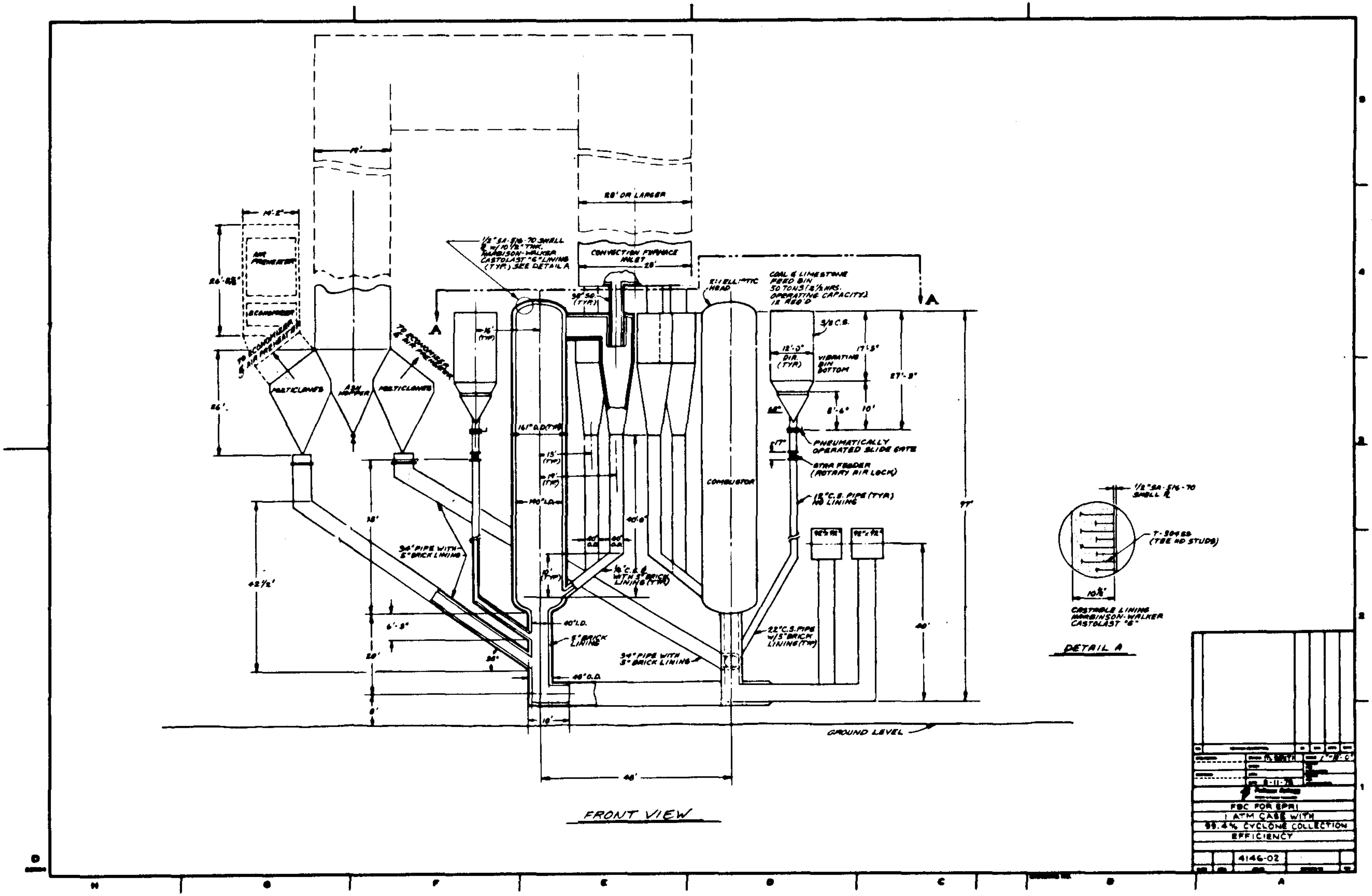


Figure II-5. - Atmospheric Circulating Bed Combustor with Low Efficiency Cyclones - Elevation.

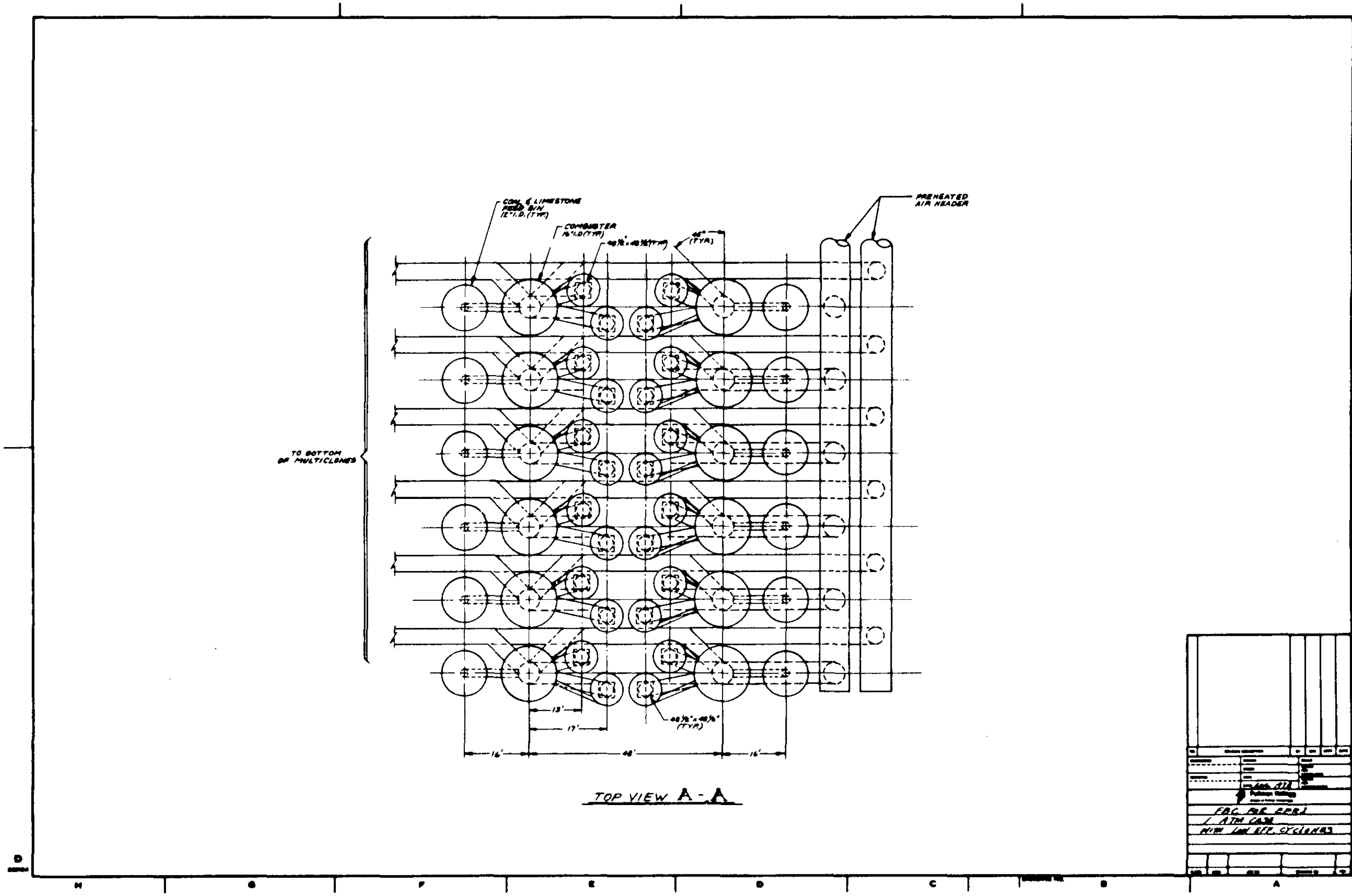


Figure II-6. - Atmospheric Circulating Bed Combustor with Low Efficiency Cyclones - Top View.

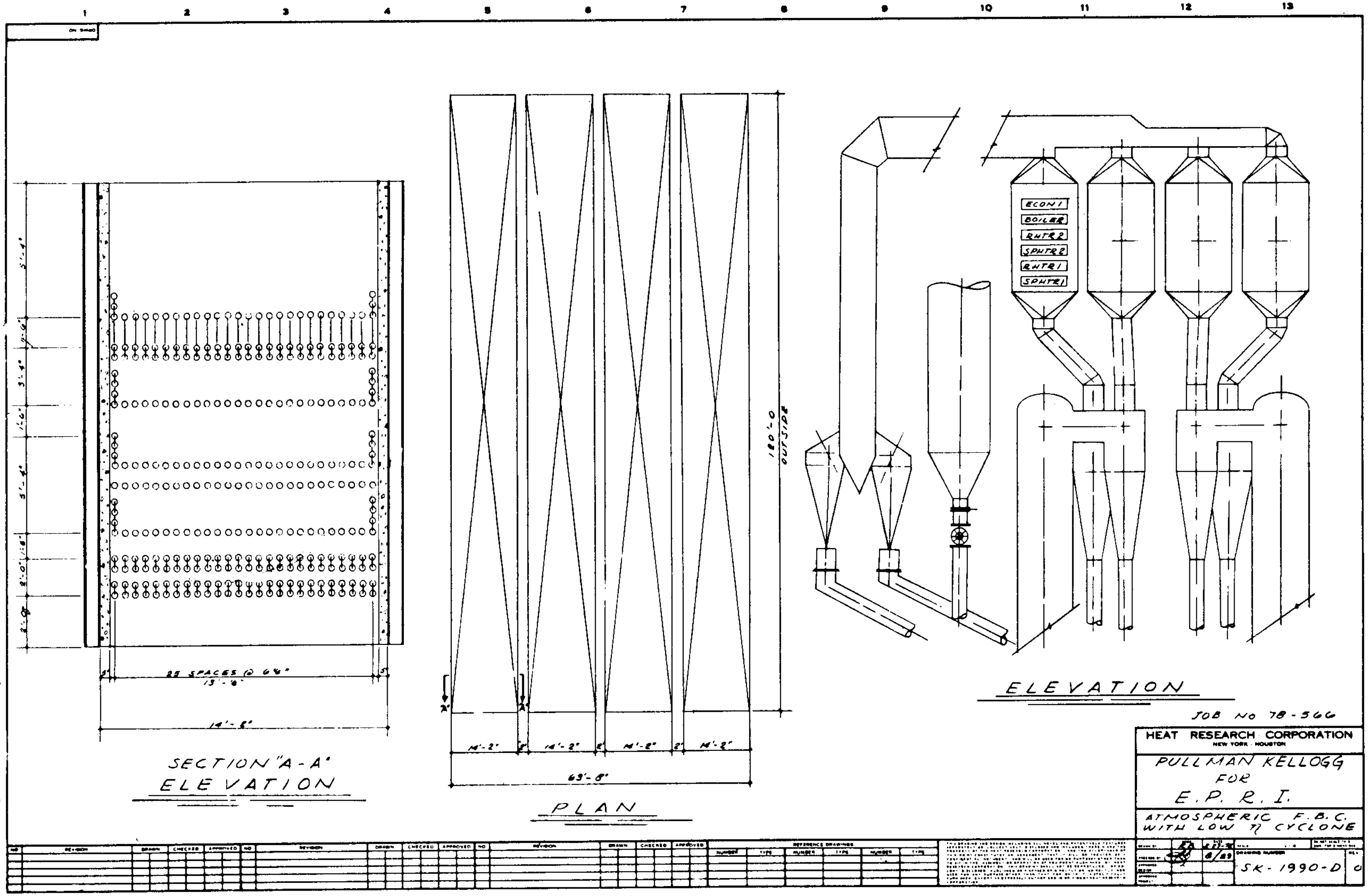


Figure II-7. - Atmospheric Circulating Bed Combustor with Low Efficiency Cyclones - Dilute Phase Heat Transfer Arrangement.

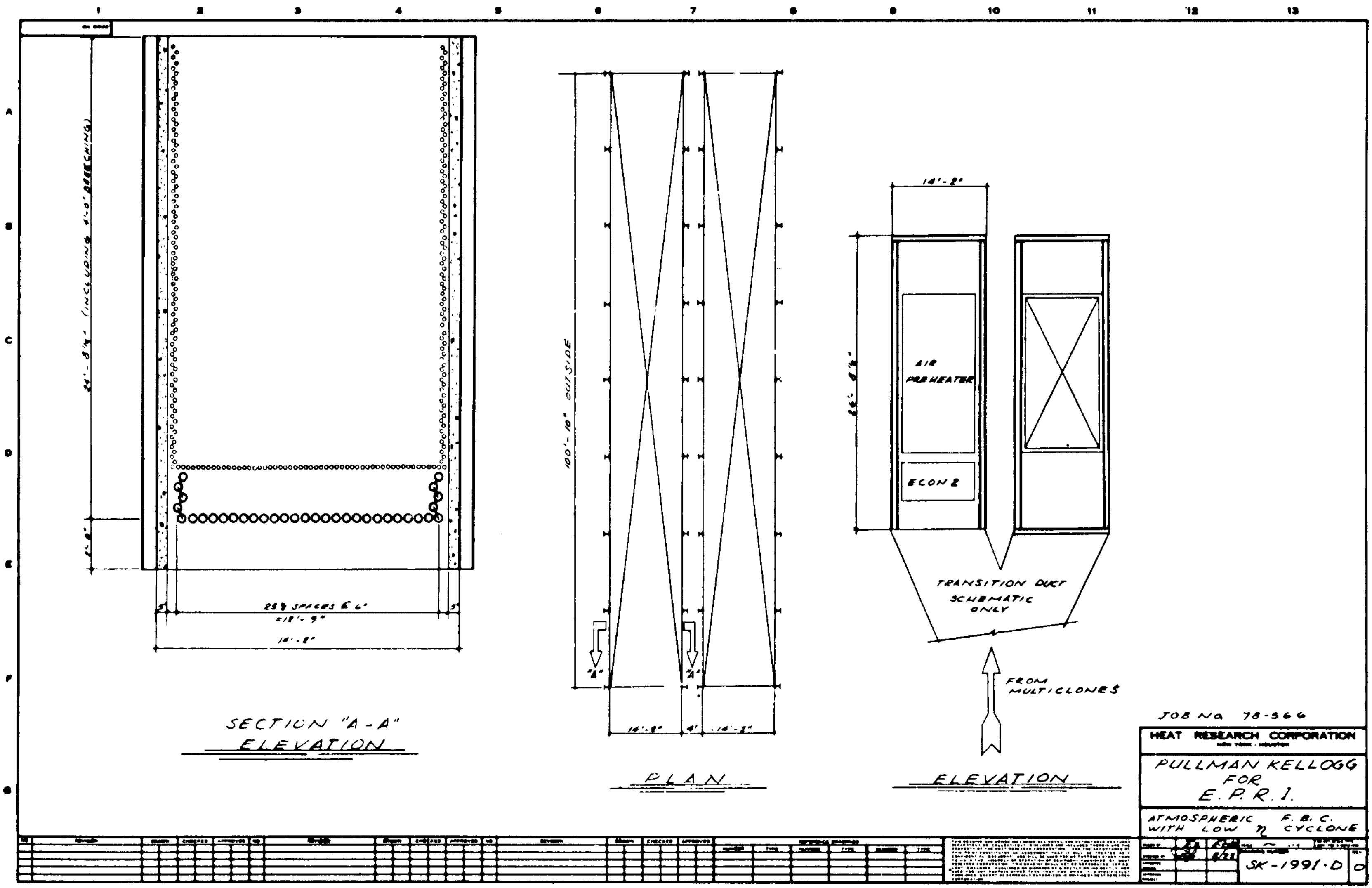


Figure II-8. - Atmospheric Circulating Bed Combustor with Low Efficiency Cyclones - Convection Section Arrangement.

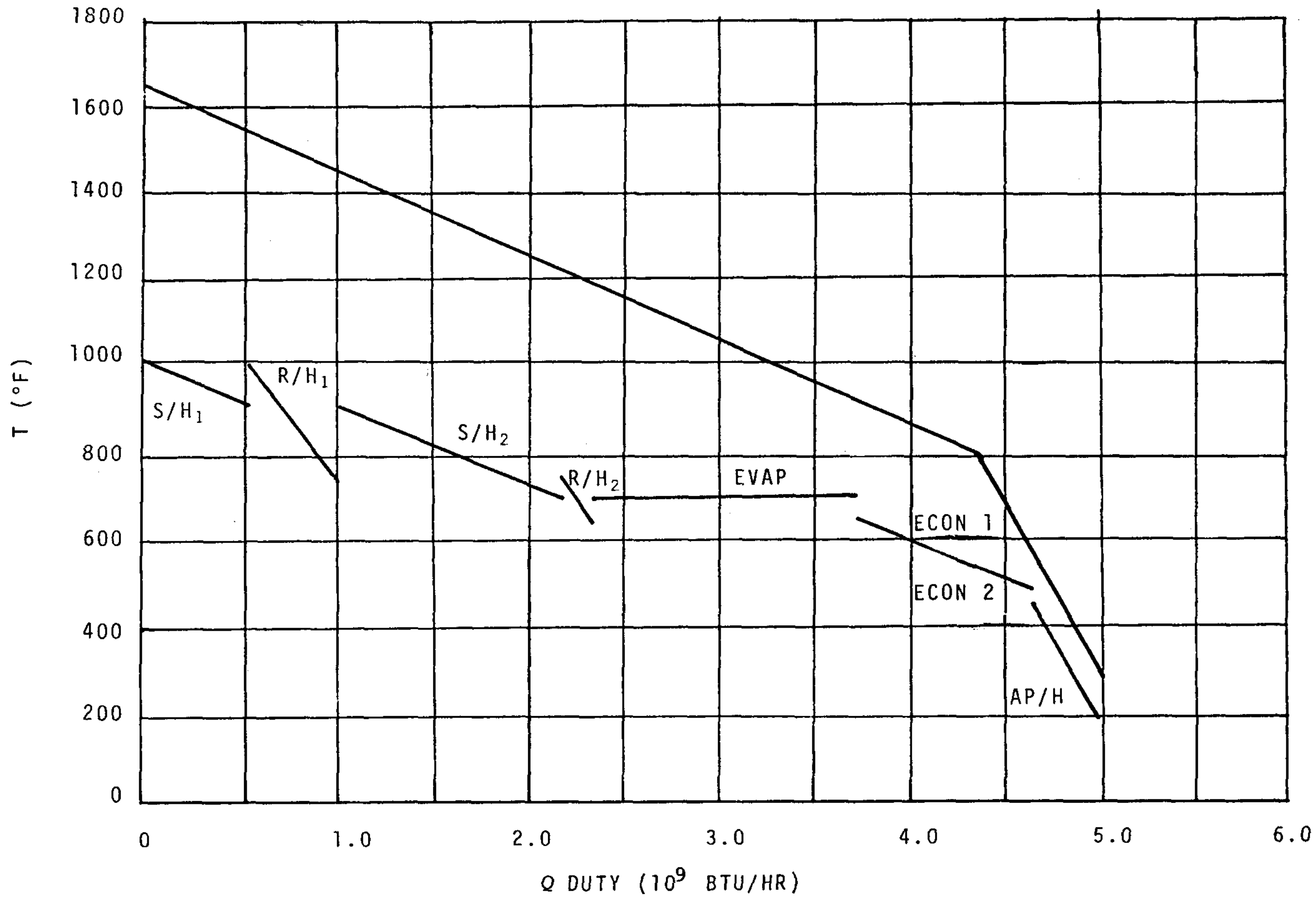


Figure II-9. - T-Q Diagram for Atmospheric Circulating Bed Combustor with Low Efficiency Cyclones.

feed bins supply fresh limestone and coal to individual combustors. The return solids, coal and fresh limestone are entrained by the air from the compressors into the bottom of the combustor where combustion begins. The solids-gas enters the cyclones at the top of the combustor where about 99.4% of the solids are separated from the flue gas inside the cyclones and flow back through cyclone legs to the bottom of the combustors. The rest of the solids and flue gas flow out from the cyclones, entering a common collecting duct and then pass over the heat exchanger tubes. After passing the heat exchanger tubes, the solids-gas mixture enters the multiclones where the remaining solids in the flue gas are separated and returned to the bottom of the combustor. Next, the flue gas enters the convection section and baghouse, and, finally, the stack. The multiclones have to be installed at a height that will ensure the returning solids have enough pressure head to overcome the air pressure at the combustor inlet. The calculated system pressure drop distribution is:

Acceleration loss		7.0 psi
Static head loss in combustor		7.5 psi
Loss in dilute-bed section		0.6 psi
	TOTAL	15.1 psi

3.1 COMBUSTOR

There are 12 combustors operating in parallel. The combustor design data are shown in Table II-6.

3.2 EXCHANGERS

Details of design data for the evaporator, superheaters, economizer and air preheater are shown in Table II-7 and in Figures II-7 and II-8.

TABLE II-6.- COMBUSTOR DESIGN DATA FOR CIRCULATING BED WITH LOW EFFICIENCY CYCLONES

Item	Design Data
Number of combustors	12
I.D.	11'-8"
Height	90'-0"
Solids Density	12 lbs/ft^3
Gas Velocity	28 fps
Actual Gas Residence Time	3.1 sec
Required Gas Residence Time	1.2 sec
Combustor Operating Temperature	1650°F
Flue Gas Flow Rate	5,254,700 lbs/hr
Inlet Air Pressure	15 psig
Material of construction:	
Shell	3/4" t SA-517-70 C.S. plate clad with 316 SS
Lining	10 1/2" Harbinson-Walker Castolast "G"

TABLE II-7.- HEAT EXCHANGERS, CIRCULATING BED COMBUSTOR WITH LOW EFFICIENCY CYCLONES

	Superheater #1	Reheater #1	Superheater #1	Reheater #1
Heat Duty (10^9 Btu/hr)	.505	.500	1.125	.250
LMTD (°F)	645	698	639	514
Steam Temp. (°F) Inlet	900	750	690	628
Outlet	1000	1000	900	750
Heat Transfer Are (Ft^2)	29,960	29,960	59,920	14,424
Tube Size	4" XX STG	4" SCH 40	4" XX STG	4" SCH 40
Tube Material	316 SS	C.S.	316 SS	C.S.
No. of Tubes	216	216	432	104
Length of Tubes	117.75'	117.75'	117.75'	117.75'

TABLE II-7.- HEAT EXCHANGERS, CIRCULATING BED COMBUSTOR WITH LOW EFFICIENCY CYCLONES (Continued)

	Evaporator	Economizer #1	Economizer #2	Air Preheater
Heat Duty (10^9 Btu/hr)	1.274	647	372	308
LMTD (°F)	514	264.7	137	149.5
Steam Temp. (°F) Inlet	650	540	484	200
Outlet	690	650	540	460
Heat Transfer Are (Ft^2)	119,850	72,130	267,100	1,301,200
Tube Size	4" XX STG	4" XX STG	4" SCH 160	4" 9 GA
Tube Material	316 SS	316 SS	C.S.	C.S.
No. of Tubes	864	520	260	17,280
Length of Tubes	117.75'	117.75'	97.75'*	100'

*1/2" x 0.05" carbon steel helical serrated fins, 6/inch.

APPENDIX III

TABULATION OF PLANT PARAMETERS

Selected items, important for their comparative value for all the circulating bed boilers, have been tabulated in the two tables that constitute the body of this appendix. Table III-1 describes the heat transfer surface area, and Table III-2 lists plant efficiencies, pressure drops, and boiler space requirements.

TABLE III-1.- TABULATION OF HEAT TRANSFER SURFACE AREA

HEAT TRANSFER SURFACE AREA (FT^2)

CASE	SUPERHTR	REHTR	BOILER	ECONOMIZER	STEAM GENERATOR TOTAL	AIR PREHTR	AIR PREHTR & STEAM GENERATOR
Atmospheric Dilute Riser	$6.5(10^4)$	$9.17(10^4)$	$1.33(10^4)$	$29.11(10^4)$	$46.11(10^4)$	$128.67(10^4)$	$174.78(10^4)$
Atmospheric W/slow bed	$4.38(10^4)$	$8.83(10^4)$	$2.21(10^4)$	$48.1(10^4)$	$63.52(10^4)$	$130.1(10^4)$	$193.62(10^4)$
Atmospheric W/O Slow Bed	$4.38(10^4)$	$8.83(10^4)$	$2.21(10^4)$	$48.1(10^4)$	$63.52(10^4)$	$130.1(10^4)$	$193.62(10^4)$
Atmospheric Low Efficiency Cyclone	$9.0(10^4)$	$4.44(10^4)$	$12.0(10^4)$	$45.9(10^4)$	$71.34(10^4)$	$130.1(10^4)$	$201.44(10^4)$
Pressureized W/slow bed	$2.81(10^4)$	$1.33(10^4)$	$2.0(10^4)$	$78.5(10^4)$	$84.63(10^4)$	----------	84.63
Pressuirzed	$2.81(10^4)$	$1.33(10^4)$	$2.0(10^4)$	$78.5(10^4)$	$84.63(10^4)$	----------	84.63
B&W FBC	103,622	161,868	180,248	46,727	492,465	1,691,600	2,183,000
PARADISE*	204,878	67,276	33,288	114,227	419,669	1,422,000	1,842,000
BULL RUN**	159,000	95,415	46,517	168,380	469,312	1,211,000	1,680,000

* Paradise is 700 MW_e gross. Surfaces were multiplied by 570/700 for this comparison.
** Bull Run is 914 MW_e gross. Surfaces were multiplied by 570/914 for this comparison.

TABLE III-2.- TABULATION OF PLANT EFFICIENCY, PRESSURE DROP AND BOILER SPACE REQUIREMENT

Case	Plant Efficiency (%)	System Pressure Drop (psi)	Boiler Space (ft) (W) x (D) x (H)
Atmospheric Dilute Riser	34.90	6.5*	100 x 140 x 170
Atmospheric W/Slow Bed	32.76	12.5	270 x 160 x 160
Atmospheric W/O Slow Bed	33.46	13.6	150 x 120 x 180
Atmospheric Low Efficiency Cyclone	33.46	17.1	120 x 240 x 180
Pressurized W/Slow Bed	36.27	12.9	140 x 140 x 220
Pressurized W/O Slow Bed	36.71	15.5	100 x 100 x 180
Pulverized Coal Plant (Paradise)	33.74	2.4	216 x 428 x 222

*Combustor solids density is 4 $\#/ft^3$ for this case. It is 12 $\#/ft^3$ for the other cases.