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ADVANCED COAL-FUELED COMBUSTOR/HEAT EXCHANGER TECHNOLOGY STUDY

Contract EF-77-C-01-2612

FIRED HEAT EXCHANGER DESIGN REPORT

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1.0 INTRODUCTION

This report is submitted in partial fulfillment of the requirements of Contract EF-77-C-01-2612, "Advanced Coal Fueled Combustor/Heat Exchanger Technology Study". The contracted effort constitutes the first phase of what is expected to be a three-phase program to advance coal fueled combustor/heat exchanger technology to be utilized in conjunction with high temperature, closed cycle gas turbine/Rankine power conversion systems. The ultimate objective in evaluating such CCGT systems is to permit more efficient conversion of U.S. coal resources to electrical energy. The objective of the three-phase study and development program is to advance the technology of coal fired CCGT/Rankine power conversion systems to a state of technology readiness, i.e., to a point where no major risks remain for full-scale commercial development.

Much of the rationale for studying the coal fired closed cycle gas turbine power generation system lies in two factors. The closed cycle gas turbine power system, which utilizes a non-condensing working fluid, is capable of higher overall thermal efficiencies than is the conventional steam based Rankine system utilized in present day steam power stations. It utilizes higher working fluid temperatures to achieve this higher efficiency. Additionally, the utilization of the closed cycle as opposed to the "open" gas turbine cycle, isolates the turbomachinery from the products of coal combustion. These coal combustion products are so dirty, corrosive, and erosive that no practical system for their direct utilization in gas turbines has yet been devised, and near term achievement of practical coal fired open cycle gas turbines appears unlikely. The direct coal firing of heat exchangers, by comparison, is present day state-of-the-art at the lower working fluid temperatures typical of today's steam based power cycles. The operating requirements for the coal fired combustor/heat exchanger of a closed cycle gas turbine power conversion system differ from those of steam boilers in several respects, particularly as regards maximum heating surface temperatures. But they are sufficiently similar that the attainment of technology readiness for such combustor/heat exchangers is expected to be attainable for modest development costs and in the near future, thus permitting more efficient utilization of the only economically abundant U.S. fuel, coal.

The effort contracted under EF-77-C-01-2612 consisted entirely of studies, analyses, design effort and reporting. No fabrication or test effort was involved. The technical goals for the contracted study effort were:

A. Provide four preliminary designs of coal fueled combustor/heat exchangers suitable for use with high temperature closed cycle gas turbine/Rankine power conversion systems. Two of the four preliminary designs were to be designed for a nominal maximum working fluid temperature of 1550 F and utilize the direct combustion of coal along with metallic heat exchanger surfaces. Two of the preliminary designs were to utilize a nominal working fluid temperature of 1750 F or higher, and additionally could utilize coal derived fuel and/or nonmetallic heat exchanger surfaces. The nominal unit rated capacity was to be 350 MWe.

B. An analysis of the "key features" of the four preliminary designs created in A above was to be conducted and documented, to determine those key features which are areas of uncertainty in the design and critical to the success of the design.

C. A detailed plan was to be prepared to spell out the analysis work, design refinement work, and development test work necessary on the identified problem areas, or areas of design uncertainty, in the combustor/heat exchanger designs (as identified in the key features analysis). This program plan for future work was to be sufficiently detailed to form a basis for planning the next phase of the overall effort necessary to achieve "technology readiness", which is attained when there are no major technology risks in full size commercial development.

This "Closed Cycle Gas Turbine, Coal Fired Heat Exchanger Design" report covers all fired heat exchanger design effort conducted during the period 11 March 1977 through 25 May 1978. It combines the three technical design reports envisioned by the program plan, "Interface Design Criteria and Reference Designs", "Candidate Heat Exchanger Conceptual Designs", and "Heat Exchanger Preliminary Design".

Three additional technical summary reports have been issued:

a. PCS 78-07, the "Working Fluids and Cycle Analysis Report" documents the effort that was conducted in those areas to define appropriate performance requirements for the fired heat exchangers.

b. PCS 78-08, the "Key Features Analysis Report" provides an analysis of the key features of the four preliminary fired heat exchanger designs.

c. PCS 78-09, the "Research and Development Plan" presents recommended plans for the attainment of coal fired CCGT combustor/heat exchanger technology readiness.

The final report RI/RD78-212 summarizes the results of the entire study and includes material on CCGT "balance of plant" equipment, i.e., unfired heat exchangers, turbomachinery, piping, etc., which is not covered in the technical reports. Additionally, the results of "cost of electricity" evaluations are presented.

2.0 ORGANIZATION

The contents of this design report include:

a. The introduction, which describes the relationship of this report and design effort to other reportts and effort on this contract.

b. A "combustor/heat exchanger interface design criteria and reference design" section, (Section 3), which defines the target operating conditions for the combustor/heat exchanger.

c. A "technical discussion of design criteria" section (Section 4), which discusses the technical factors that influence the design of the coal fueled CCGT combustor/heat exchanger. This section discusses the influences upon design of coal combustion methods and pehnomena, emission phenomena and requirements, available material properties, and safety codes; the effects of coal ash on heat exchanger operation and design, the inter-relationship of ash and metallurgy, etc.

d. The "conceptual design of closed cycle gas turbine heaters" section, (Section 5), describes candidate concepts, including those based on pulverized coal, atmospheric fluidized bed combustion, etc. These conceptual designs are examples of designs that have some bearing on the project and which furnished guidance for the selection of the concepts for preliminary design.

e. Each of the four preliminary designs is described in detail in Section 6, including drawings, tables of weights and surfaces, temperature and flow and pressure distributions, material selections, etc. Additional discussion relative to the preliminary designs is contained in the "Key Features Analysis Report", PCS 78-08 as well as in the "R&D Plan", PCS 78-09.

f. A cost estimate for each preliminary design is presented. Those costs included and those omitted are defined, and a comparison of costs with a 350 MWe boiler is presented. The sources of significant costs are discussed.

g. The discussion of results section summarizes the conclusions resulting from the design exercise, i.e., reaches judgments on the practicability of the heat designs relative to the required service, desirable design trends, major advantages and major problems.

3.0 INTERFACE DESIGN CRITERIA, REFERENCE DESIGNS AND CONTRACTUAL GROUNDRULES

The function of the coal fired combustor/heat exchanger of a closed cycle gas turbine power conversion system is to convert the chemical energy present in the raw coal fuel into heat energy in the working fluid of the thermodynamic cycle. The closed cycle gas turbine power conversion system operates in such a way that the working fluid passing through the gas turbine is entirely isolated from the products of combustion of the coal fuel, thus avoiding the problems of turbine erosion, corrosion, and cleanliness that must be reckoned with when the products of coal combustion are passed through a turbine. The closed cycle system thus involves the transfer of heat from the coal combustion gases through the impervious walls of the heat exchanger surface to the gas turbine working fluid.

The combustion and heat exchange process may be visualized as consisting of a number of steps, typically; coal preparation for burning; the mixing of coal and combustion air; the combustion of the coal and the conversion of its chemical energy to sensible heat in the combustion products and of solid fuel substance to gaseous combustion products; the transfer of sensible heat from the combustion products through heat exchanger walls to the working fluids; the removal of excess solid and gaseous constituents, i.e., ash, SOX and NOX, from the combustion products prior to discharge to meet legal emissions limitations; and the discharge of the spent combustion gases to the atmosphere.

3.1 CYCLE INTERFACE DESIGN CRITERIA AND REFERENCE DESIGN

The performance requirements of the combustor/heat exchanger are defined by the thermodynamic cycle. A variety of cycle requirements were considered in the course of the design studies on this contract, and there was an interplay and iteration between thermodynamic cycle requirements and combustor/heat exchanger effects. The "Working Fluids and Cycle Analysis Report", PCS 78-07, discusses the wide variety of cycles that were examined, lists their performance requirements and gives the reasoning behind the choice of cycles for the design effort. For the purposes of this design report, the operating conditions and performance requirements for the cycles which formed the basis for the major design effort are repeated in Tables 3.1.1 thru 3.1.7. As can be seen these cycles involve peak working fluid temperatures of 1450 F, 1550 F, 1750 F, and 2250 F. Performance data on several of the cycles is presented for several working fluids; air; a helium/CO_2/O_2 mixture; and a helium/O_2 mixture. All of the cycles utilize a 2400 psi steam bottoming cycle, with some of the heat from the combustion gases being absorbed directly by the bottoming cycle. All cycles are sized for 350 MWe of electrical output. The cycle schematics of Figs. 3.1.1 thru 3.1.6 provide an aid to comprehension of the thermodynamic cycle tables.

As evident from Tables 3.1.1 thru 3.1.7 all of the combustor/heat exchanger designs studied in detail during this contract were sized and arranged to meet the requirements of a steam bottomed non-regenerative Brayton cycle. However this does not preclude the utilization of the studied design concepts to supply the heat input for regenerated and/or organic bottomed cycles. These adaptations are further discussed in Section 6.

Having defined the thermodynamic cycles of major interest, (Tables 3.1.1 thru 3.1.6), one can now establish "reference" designs applying to specific coal fired combustor/heat exchangers. These are listed in Tables 3.1.8 and 3.1.9. Note that the reference design tables now provide:

complete system energy balances
material flow rates
system state point temperatures and pressures
component and loop pressure drops

Table 3.1.1. NON-RECUPERATED CCGT CYCLE/STEAM BOTTOMING CYCLE

He	2 -Stage Exp.(1550 F/1550 F/Adiab.)1-Stage Comp.(80 F)		
	Comp. Ratio (5)	Exp. Ratio (4.55)	PR Factor (91%)
Steam	Reheat (2400 P/1050 F/1050 F)		Condensing (80 F)

Heater - gas in		644.06F	1368.62H	1000P	HP turbine PR=2.459
gas out		1550.00	2491.98	970	967 P_{in} 393.25 P_{out}
Reheater - gas in		985.93	1792.54	391.6	LP turbine PR=1.85
gas out		1550.00	2491.98	380.6	379.0 P_{in} 204.9 P_{out}
Non-Fired					Comp. PR = 5
Boiler - gas in		1144.69	1989.40	204.4	200.5 P_{in} 1002.5
gas out		790.00	1549.58	203.4	P_{out}
water/steam in		682.19	1102.70	2640.	
steam out		1054.16	1494.00	2480.	
Fired					
Boiler - water in		687.06	781.98	2840.	
water/steam out		685.55	1102.70	2680.	
Economizer - gas in		790.00	1549.58	203.4	
gas out		135.00	737.39	2020.	
water in		83.17	59.39	3040.	
water out		688.53	781.98	2880.	
Cooler - gas in		135.00	737.39	202.0	
gas out		80.00	669.19	201.0	
cooling water in		65.00			
cooling water out		110.00			
Fired Reheater - steam in		638.00	1322.50		
steam out		1050.00	1549.66		
Heat Balance Based on 1 lb Gas Flow					
Heat Input	Heater	1123.36			
	Reheater	699.44			
	Fired Steam Boiler	360.49			
	Fired Steam Reheater	255.33	2438.62		
Energy Output	Non-Fired Boiler	439.82			
	Non-Fired Economizer	812.19			
	Cooler	68.20			
	CCGT Gen. Output	494.04	502.58		
	CCGT Mech. & Gen. Loss	8.54			
Steam Cycle Heat Input		1867.83			
Net Thermal Eff.		.3954			
Generator Output		738.54			
Lb Steam/lb Gas Flow		1.124			
Overall Cycle Efficiency		.5054			
Lb Gas Flow/Hr (376.34 MW)		1041819.			
Lb Steam Flow/Hr		1171005.			
Gas Cycle Output MW		150.84			
Steam Cycle Output MW		225.50			

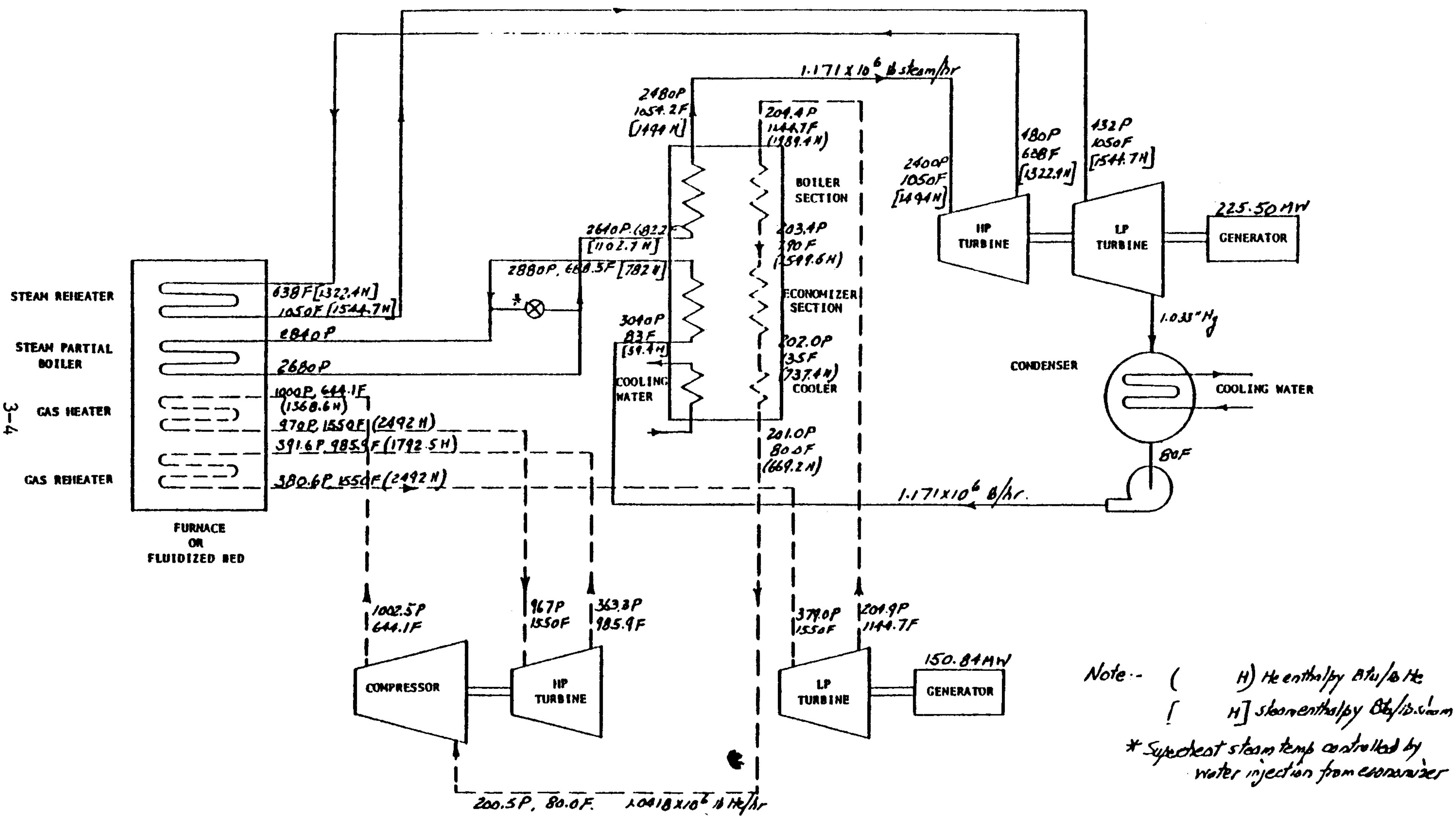


Figure 3.1.1. 1550 F/1550 F Helium CCGT Combined With 2400 P/1050 F/ 1050 F Steam Rankine Cycle

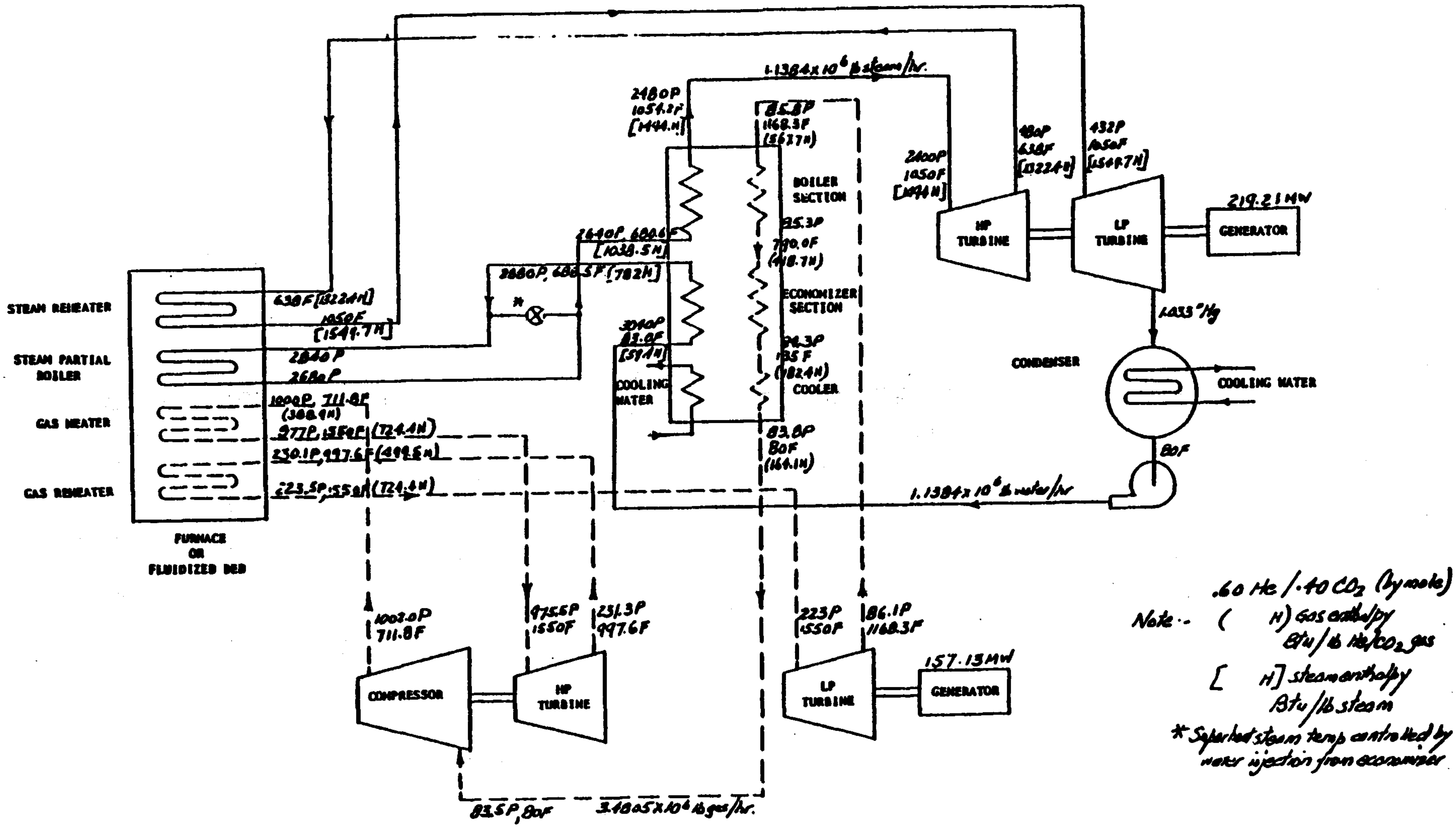
Table 3.1.2. NON-RECUPERATED CCGT CYCLE/STEAM BOTTOMING CYCLE

He/CO_2	2-Stage Exp. (1550 F/1550 F adiab turb)	1-Stage Comp. (80 F)	
Steam	Comp. Ratio (12)	Exp. Ratio (10.92)	PR Factor (91%)
	Reheat (2400 P/1050 F/1050 F)		Condensing (80 F)

Heater - gas in	711.84F	388.90H	1000P	HP Turbine PR = 4.217, 975.5 P_{in}, 231.3 P_{out}, LP Turbine PR = 2.59, 223.0 P_{in}, 86.1 P_{out}, Comp. PR = 12, 83.5 P_{in}, 1002.0 P_{out}
gas out	1550.00	724.36	977	
Reheater - gas in	997.55	499.50	230.1	
gas out	1550.00	724.36	223.5	
Non-Fired				
Boiler - gas in	1168.30	567.66	85.8	
gas out	790.00	418.69	85.3	
water/steam in	676.16	1038.53	2640.	
steam out	1054.16	1494.00	2480.	
Fired				
Boiler - water in	687.06	781.98	2840.	
water/steam out	678.41	1038.53	2680.	
Economizer - gas in	790.00	418.69	85.3	
gas out	135.00	182.35	84.3	
water in	83.17	59.39	3040.	
water out	688.53	781.98	2880.	
Cooler - gas in	135.00	182.35	84.3	
gas out	80.00	164.05	83.8	
cooling water in	65.00			
cooling water out	110.00			
Fired Reheater - steam in	638.00	1322.50		
steam out	1050.00	1549.66		

Heat Balance Based on 1 lb Gas Flow

Heat Input	Heater	335.46	
	Reheater	224.86	
	Fired Steam Boiler	83.91	
	Fired Steam Reheater	74.30	718.53
Energy Output	Non-Fired Boiler	148.97	
	Non-Fired Economizer	236.34	
	Cooler	18.30	
	CCGT Gen. Output	154.04	156.70
	CCGT Mech. & Gen. Loss	2.66	
Steam Cycle Heat Input		543.52	
	Net Thermal Eff.	.3954	
	Generator Output	214.91	
	Lb Steam/lb Gas Flow	.32707	
Overall Cycle Efficiency		.5135	
Lb Gas Flow/Hr (376.34 MW)		3480486.	
Lb Steam Flow/Hr		1138363.	
Gas Cycle Output MW		157.13	
Steam Cycle Output MW		219.21	



STEAM REHEATER
STEAM PARTIAL BOILER
GAS HEATER
GAS REHEATER
FURNACE OR FLUIDIZED BED
1.1384 x 10^6 lb steam/hr.
BOILER SECTION
ECONOMIZER SECTION
COOLER
COOLING WATER
HP TURBINE
LP TURBINE
GENERATOR
219.21 MW
CONDENSER
COOLING WATER
80F
1.1384 x 10^6 lb water/hr
COMPRESSOR
HP TURBINE
LP TURBINE
GENERATOR
157.13 MW
83.5P, 80F
3.1805 x 10^6 lb gas/hr.
.60 He / .40 CO2 (by mole)
Note: (H) Gas enthalpy Btu/lb He/CO2 gas
[H] steam enthalpy Btu/lb steam
* Superheated steam temp controlled by water injection from economizer

Table 3.1.3. NON-RECUPERATED CCGT CYCLE/STEAM BOTTOMING CYCLE

Air	2 -Stage Exp. (1550 F /1550 F Adiab)	1-Stage Comp. (80 F)
	Comp. Ratio (12) Exp. Ratio (10.92)	PR Factor (91%)
Steam	Reheat (2400 P/1050 F/1050 F)	Condensing (80 F)

Heater - gas in	704.45F	278.12H	1000P	HP turbine PR=4.214,
gas out	1550.00	503.75	977P	975.5 P_{in}, 231.5
Reheater - gas in	979.02	349.42	230.5	P_{out}, LP turbine
gas out	1550.00	503.75	224.0	PR = 2.59, 223.0 P_{in}
Non-Fired				86.1 P_{out}, comp.
Boiler - gas in	1154.06	395.89	85.8	PR = 12, 83.5 P_{in},
gas out	790.00	300.12	85.3	1002 P_{out}
water/steam in	676.16	1070.00	2640.	
steam out	1054.16	1494.00	2480.	
Fired				
Boiler - water in	687.06	781.98	2840.	
water/steam out	678.41	1070.00	2680.	
Economizer - gas in	790.00	300.12	85.3	
gas out	135.00	136.91	84.3	
water in	83.17	59.39	3040.	
water out	688.53	781.98	2880.	
Cooler - gas in	135.00	136.91	84.3	
gas out	80.00	123.78	83.8	
cooling water in	65.00			
cooling water out	110.00			
Fired Reheater - steam in	638.00	1322.50		
steam out	1050.00	1549.66		

Heat Balance Based on 1 lb Gas Flow

Heat Input	Heater	225.64	
	Reheater	154.34	
	Fired Steam Boiler	65.06	
	Fired Steam Reheater	51.31	496.35
Energy Output	Non-Fired Boiler	95.77	
	Non-Fired Economizer	163.21	
	Cooler	13.13	
	CCGT Gen. Output	106.03	107.86
	CCGT Mech. & Gen. Loss	1.83	

Steam Cycle Heat Input	375.35
Net Thermal Eff.	.3954
Generator Output	148.41
Lb Steam/lb Gas Flow	.22587
Overall Cycle Efficiency	.5126
Lb Gas Flow/Hr (376.34 MW)	5046869.
Lb Steam Flow/Hr	1139936.
Gas Cycle Output MW	156.83
Steam Cycle Output MW	219.51

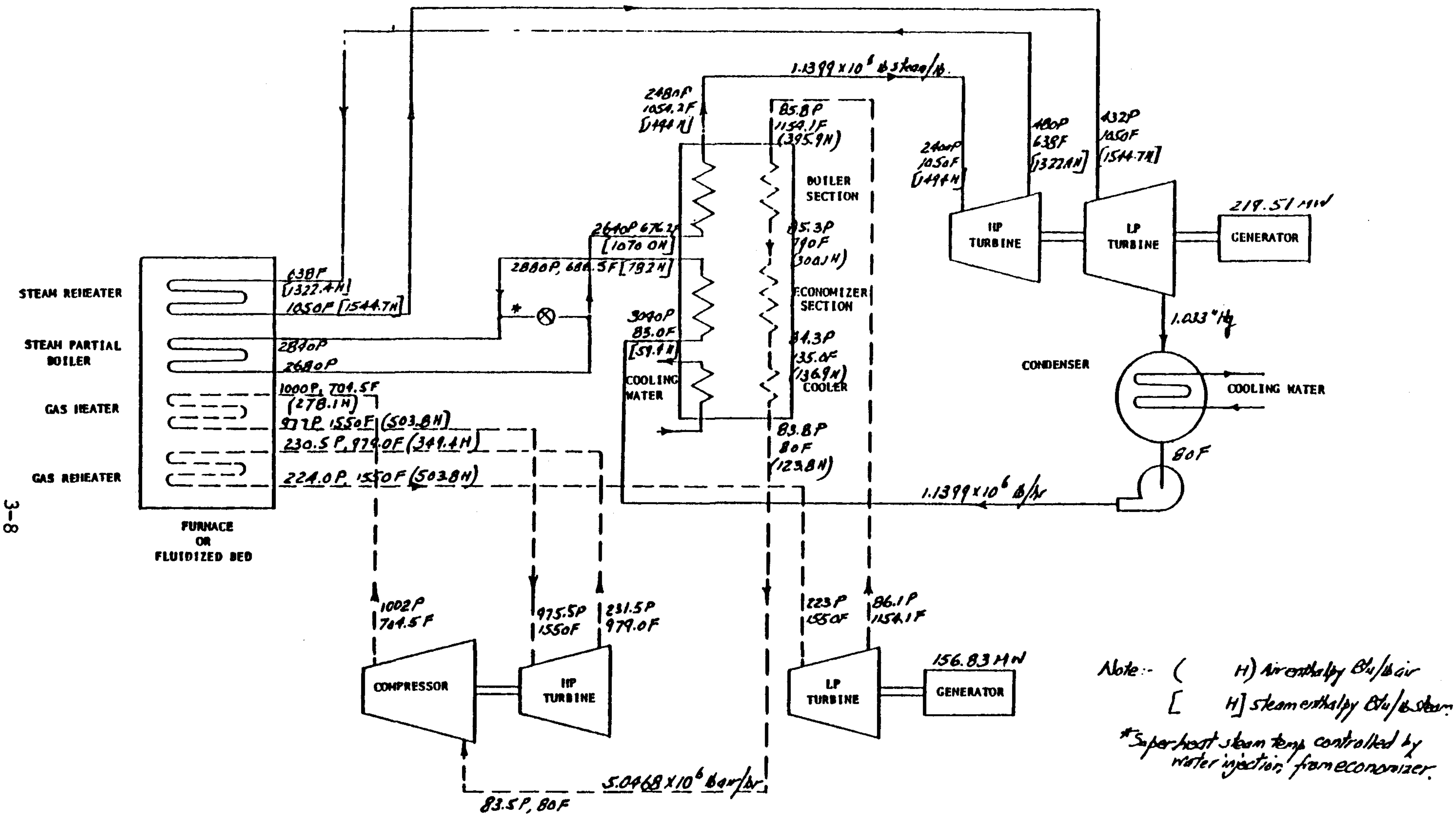


Figure 3.1.3. 1550 F/1550 F Air CCGT Combined With 2400 P/1050 F/ 1050 F Steam Rankine Cycle

Table 3.1.4. NON-RECUPERATED CCGT CYCLE/STEAM BOTTOMING CYCLE

He/CO_2 2-Stage Exp.(1750 F cooled/1550 F adiab) 1-Stage Comp. (80 F)
Comp. Ratio (12) Exp. Ratio (10.92) PR Factor (91%)
Steam Reheat (2400P/1050F/1050F) Condensing (80 F)

Heater - gas in .965 lb	712.05F	388.98H	1000P	HP turbine PR=3.778 975.5 P_{in} 258.21 P_{out}, LP turbine PR=2.89, 248.83 P_{in} 86.1 P_{out}, 85.8 comp. PR=12, 83.5 P_{in} 1002 P_{out}
gas out	1750.00	808.24	977	
Reheater - gas in	1170.70	568.63	257.01	
gas out	1550.00	724.36	249.33	
Non-Fired				
Boiler - gas in	1128.46	551.64	85.8	
gas out	790.00	418.69	85.3	
water/steam in	678.57	1087.51	2640.	
steam out	1054.16	1494.00	2480.	
Fired				
Boiler - water in	687.06	781.98	2840.	
water/steam out	681.93	1087.51	2680.	
Economizer - gas in	790	418.69	85.3	
gas out	135	182.35	84.3	
water in	83.17	59.39	3040.	
water out	688.53	781.98	2880.	
Cooler - gas in	135	182.35	84.3	
gas out	80	164.05	83.8	
cooling water in	65			
cooling water out	110			
Fired Reheater - steam in	638	1322.50		
steam out	1050	1549.66		

Heat Balance Based on 1 lb Gas Flow

Heat Input	Heater	404.58	
	Reheater	155.73	
	Fired Steam Boiler	99.93	
	Fired Steam Reheater	74.30	734.54
Energy Output	Non-Fired Boiler	132.96	
	Non-Fired Economizer	236.34	
	Cooler	18.30	
	CCGT Gen. Output	169.78	172.72
	CCGT Mech. & Gen. Loss	2.94	
Steam Cycle	Heat Input	543.53	
	Net Thermal Eff.	.3954	
	Generator Output	214.91	
	Lb Steam/lb Gas Flow	.32707	
Overall Cycle Efficiency		.5237	
Lb Gas Flow/Hr (376.34 MW)		3.338078×10^6	
Lb Steam Flow/Hr		1.091785×10^6	
Gas Cycle Output MW		166.09	
Steam Cycle Output MW		210.25	

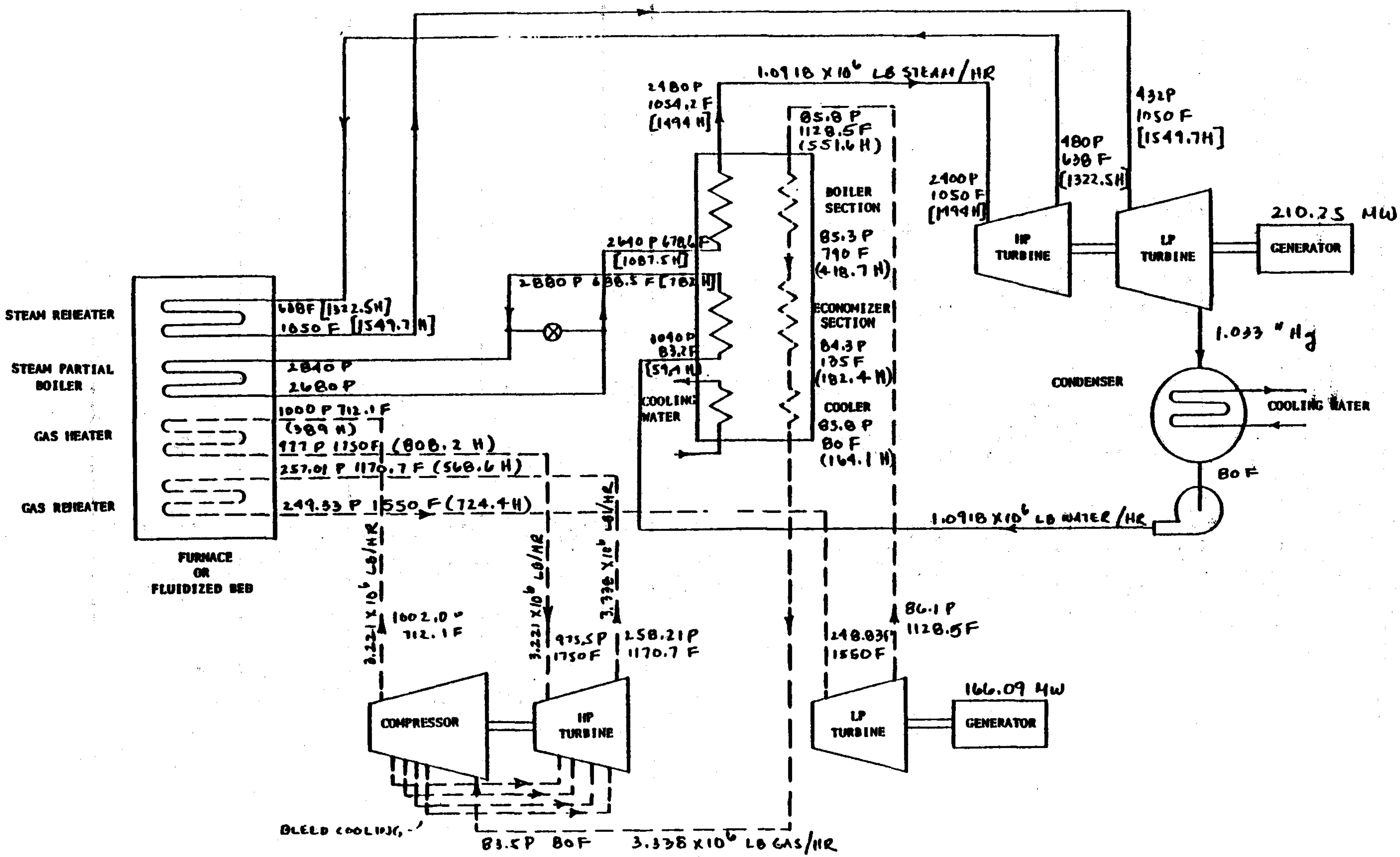


Figure 3.1.4. 1750 F/1550 F He/CO_2 CCGT Combined With 2400 P/1050 F/1050 F Steam Rankine Cycle

Table 3.1.5. NON-RECUPERATED CCGT CYCLE/STEAM BOTTOMING CYCLE

Air 2 -Stage Exp.(1750 F cooled/1550 F adiab) 1-Stage Comp. (80 F)
Comp. Ratio(12) Exp. Ratio(10.92) PR Factor (91%)

Steam -Reheat (2400 P/1050 F/1050 F) Condensing (80 F)

Heater - gas in .9457 lb	704.86F	278.22H	1000P	HP turbine PR=3.797
gas out	1750	559.59	977	975.5 P_{in} 256.91
Reheater - gas in	1131.60	389.88	255.91	P_{out}, LP turbine
gas out	1550	503.75	248.62	PR = 2.876
Non-Fired				247.62 P_{in} 86.1 P_{out}
Boiler - gas in	1115.07	385.47	85.8	Comp. PR=12, 83.5
gas out	790	300.12	85.3	P_{in}, 1002 P_{out}
water/steam in	685.98	1116.13	2640	
steam out	1054.16	1494	2480	
Fired				
Boiler - water in	687.06	781.98	2840	
water/steam out	689.10	1116.13	2680	
Economizer - gas in	790	300.12	85.3	
gas out	135	136.91	84.3	
water in	83.17	59.39	3040	
water out	688.53	781.98	2880	
Cooler - gas in	135	136.91	84.3	
gas out	80	123.78	83.8	
cooling water in	65			
cooling water out	110			
Fired Reheater - steam in	638	1322.50		
steam out	1050	1549.66		
	Heat Balance Based on 1 lb Gas Flow			
Heat Input: Heater	266.11			
Reheater	113.87			
Fired Steam Boiler	75.47			
Fired Steam Reheater	51.31	506.76		
Energy Output: Non-Fired Boiler	85.35			
Non-Fired Economizer	163.21			
Cooler	13.13			
CCGT Gen. Output	116.27	118.28		
CCGT Mech. & Gen. Loss	2.01			
Steam Cycle Heat Input	375.34			
Net Thermal Eff.	.3954			
Generator Output	148.41			
Lb Steam/lb Gas Flow	.22587			
Overall Cycle Efficiency	.5223			
Lb Gas Flow/Hr (376.34 MW)	4.851616×10^6			
Lb Steam Flow/Hr	1.095834×10^6			
Gas Cycle Output MW	165.32			
Steam Cycle Output MW	211.02			

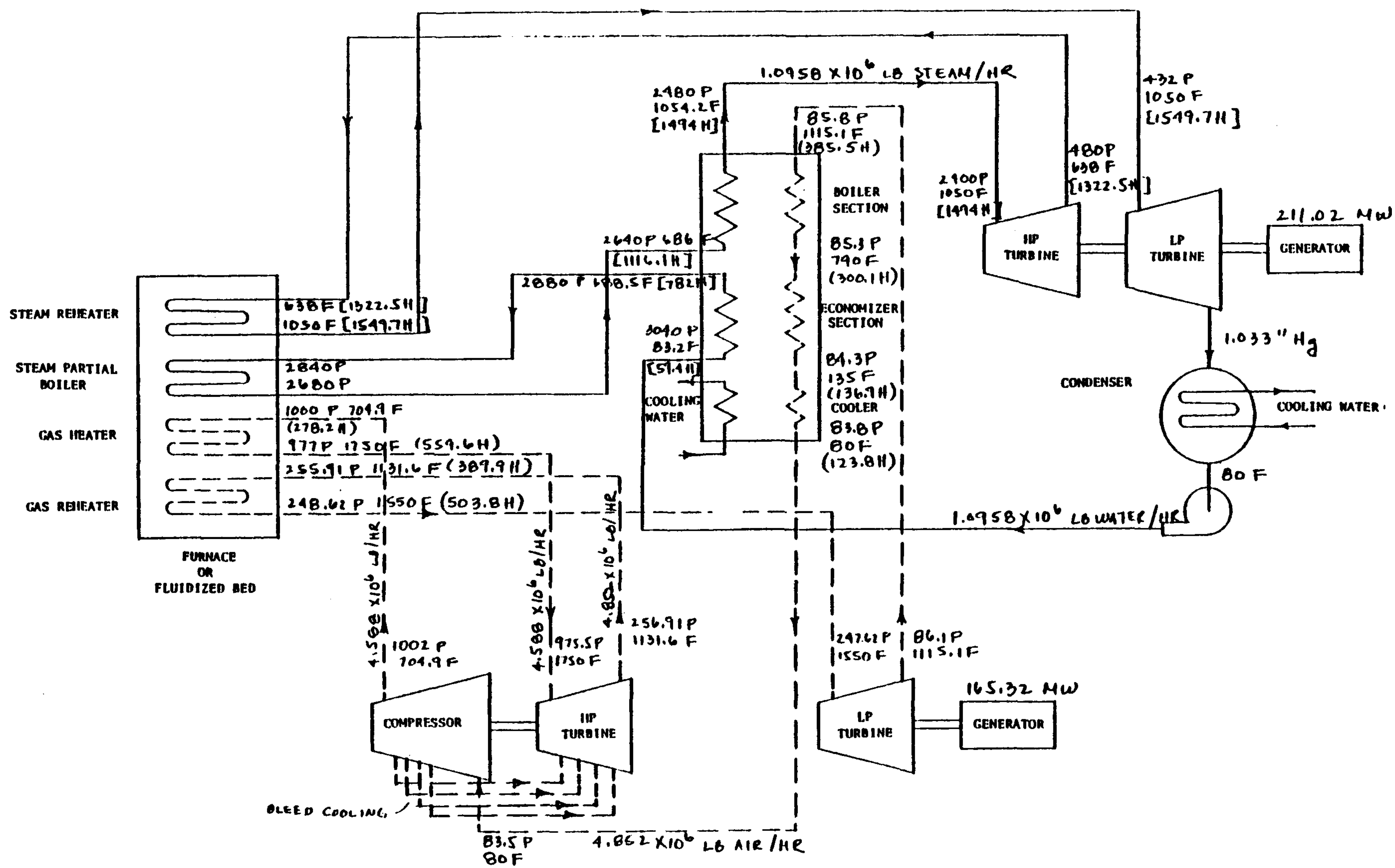


Figure 3.1.5. 1750 F/1550 F Air CCGT Combined With 2400 P/1050 F/1050 F Steam Rankine Cycle

TABLE 3.1.6. NON-RECUPERATED BRAYTON CYCLE/STEAM BOTTOMING CYCLE

He	1-Stage Exp. (2250 F) 1-Stage Comp. (80 F)	
	Comp. Ratio (5) Exp. Ratio 4.55 (91%)	
Steam	1 Reheat (2400 P/1050 F/1050 F	Condensing (80 F)

Heater - gas in	644.06F	1368.62H	1000P	HP turbine PR=4.55
gas out	2250.	3359.98	955	950 P_{in} 208.79
Reheater - gas in	--	---		P_{out}, Comp. PR=5
gas out	--			200.8 P_{in},
Non-fired Boiler - gas in	1096.40	1929.52	207.5	1004 P_{out}
gas out	790	1549.58	206	
water/steam in	701.38	1155.98	2640	
steam out	1054.16	1494	2480	
Fired Boiler - water in	687.06	781.98	2840	
water/steam out	704.6	1155.98	2680	
Economizer - gas in	790.	1549.98	206	
gas out	135.	737.39	203	
water in	83.17	59.39	3040	
water out	688.53	781.98	2880	
Cooler - gas in	135.	737.39	203	
gas out	80.	669.19	201.5	
cooling water in	65.			1.335614×10^6 lb/hr
cooling water out	110			
Fired reheater - steam in	638	1322.5		
steam out	1050	1549.66		

		Heat Balance Based on 1 lb. CCGT Gas Flow	
Heat Input	Heater	1991.36	
	Reheater	---	
	Fired Steam Boiler	420.38	
	Fired Steam Reheater	255.33	2667.07
Energy Output	Non-Fired Boiler	379.94	
	Non-Fired Economizer	812.19	
	Cooler	68.2	
	CCGT Generator Output	718.59	731.02
	CCGT Mech. & Gen Loss	12.43	
Steam Cycle Heat Imput		1867.84	
	Net Thermal Eff.	.3954	
	Generator Output	738.54	
	Lb Steam/lb Gas Flow	1.124	
Overall Cycle Efficiency		.5463	
Lb Gas Flow/Hr (376.34 MW)		881270.	
Lb Steam Flow/Hr.		990548.	
Gas Cycle Output MW		185.59	
Steam Cycle Output MW		190.75	

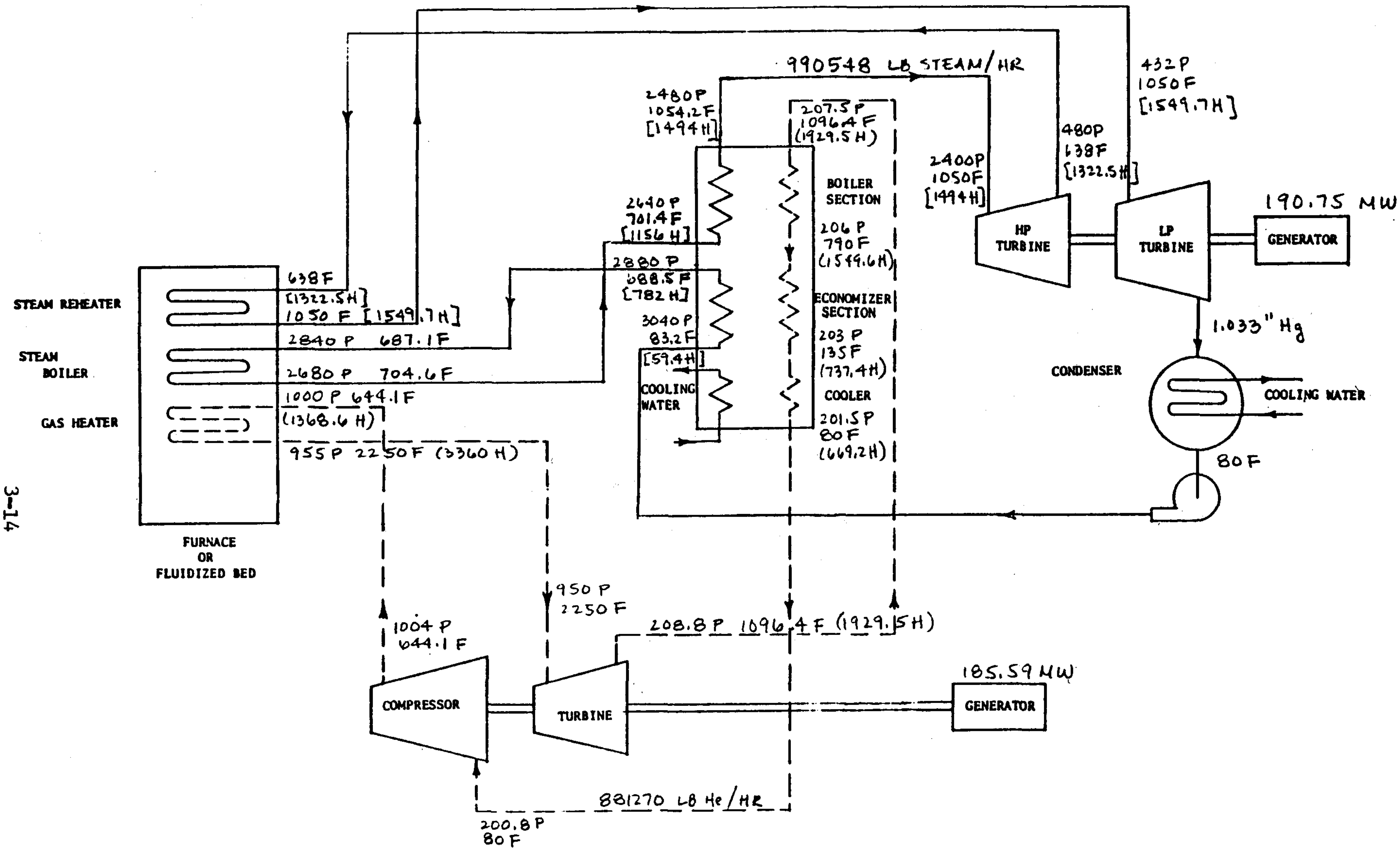


Figure 3.1.6 2250 F Helium Brayton Combined With 2400 P/1050 F/1050 F Steam Rankine Cycle

Table 3.1.7. Non-Recuperated Brayton Cycle/Steam Bottoming Cycle

He/CO_2	2-Stage Exp. (1450F/1450F Adiab) Comp. Ratio (10) Exp. Ratio (9.1)	1-Stage Comp. (80F) PR Factor (91%)
Steam	Reheat (2400 P/1050 F/1050 F)	Condensing (80F)

Heater - gas in	651.15F	366.04H	1000P	HP turbine PR=3.85
gas out	1450.00	682.82	977	975.5 P_{in} 253.2 P_{out}
Reheater - gas in	950.13	480.83	252	
- gas out	1450.00	682.82	245.4	LP turbine PR=2.362
Non-fired				244.9 P_{in} 103.7 P_{out}
Boiler - gas in	1117.16	547.11	103.4	Comp. PR=10
gas out	790.00	418.69	102.7	100.2 P_{in} 1002 P_{out}
water/steam in	681.86	1101.36	2640	
steam out	1054.16	1494	2480	
Fired				
Boiler - water in	687.06	781.98	2840	
water/steam out	685.23	1101.36	2680	
Economizer - gas in	790.00	418.69	102.7	
gas out	135.00	182.35	101.2	
water in	83.17	59.39	3040	
water out	688.53	781.98	2880	
Cooler - gas in	135.00	182.35	101.2	
gas out	80.00	164.05	100.5	
cooling water in	65.00		1499271 lb/hr	
cooling water out	110.00			
Fired Reheater - steam in	638.00	1322.5		
steam out	1050.00	1549.66		

Heat Balance Based on 1 lb Gas Flow

Heat Input	Heater	316.78	
	Reheater	201.99	
	Fired Steam Boiler	104.46	
	Fired Steam Reheater	74.30	697.53
Energy Output	Non-fired Boiler	128.42	
	Non-fired Economizer	236.34	
	Cooler	18.30	
	CCGT Generator Output	133.40	135.71
	CCGT Mech.&Gen.Loss	2.31	

Steam Cycle Heat Input	543.52
Net Thermal Eff.	.3954
Generator Output	214.91
Lb Steam/Lb Gas Flow	.32707
Overall Cycle Efficiency	.4993
Lb Gas Flow/Hr (376.34 MW)	3686731
Lb Steam Flow/Hr	1205819
Gas Cycle Output MW	144.14
Steam Cycle Output MW	232.20

Table 3.1.8. Cycle-Combustor Combination (350 MWe)

Combustor/ Heat Exchanger	Fluidized Bed	Pulverized Coal	Cyclone	Fluidized Bed
Temperatures	1550/1550	1550/1550	2250	1750/1550
Working Fluid	$He-CO_2-O_2$	$He-O_2$	$He-O_2$	$He-CO_2$-02
Mol Fraction	.60 - .39 - .01	.99 - .01	.99 - .01	.60 - .39 - .01
PR	5	12	12	5
η_{cyc} (%)	51.4	50.54	54.6	52.37
η_{comb} (%) *	85.2	86.2	86.2	85.2
η_{plant} (%) **	40.8	38.7†	41.7†	41.7
$\dot{w}_{coal}$, lb/hr	$.271(10)^6$	$.286(10)^6$	$.266(10)^6$	$.266(10)^6$
$\dot{w}_{combustion\ air}$, lb/hr	$2.57(10)^6$	$2.71(10)^6$	$2.41(10)^6$	$2.52(10)^6$
$\dot{w}_{limestone}$, lb/hr	91,000††	45,000†††	42,000†††	89,000††

* Illinois No. 6 coal, 300 F exit temperature

** Include 7% house load

† Include 6% scrubbing loss

†† Ca/S Mol ratio of 2.75

††† Wet limestone scrubbing, Ca/S Mol ratio of 1.3

Table 3.1.9 Cycle Design Parameters

Combustor/ Heat Exchanger	Fluidized Bed	Pulverized Coal	Cyclone	Fluidized Bed
Brayton Cycle	1550/1550	1550/1550	2250	1750/1550
Primary	1x	1x	1x	0.965x
Inlet P (psia)	1000	1000	1000	1000
Inlet T (^oF)	712	644	644	712
ΔP (psia)	23	30	45	23
Reheat	1x	1x	1x	1x
Inlet P (psia)	230	392	NA	257
Inlet T (^oF)	998	986	NA	1171
ΔP (psia)	7	11	NA	8
$\dot{w}_{gas}$ (lb/hr)	$3.480(10)^6$	$1.105(10)^{6*}$	$.934(10)^{6*}$	$3.338(10)^6$
Rankine Cycle				
$T_{throttled\ steam}$	1050/1050	1050/1050	1050/1050	1050/1050
Primary	0.45x	0.63x	1x	1x
Inlet T (^oF)	687	687	676	687
Inlet P (psia)	2840	2840	2640	2840
Exit T (^oF)	678	686	1054	682
Exit P (psia)	2680	2680	2480	2680
Reheat	1x	1x	1x	1x
Inlet T (^oF)	638	638	638	638
Inlet P (psia)	480	480	480	480
Exit T (^oF)	1050	1050	1050	1050
Exit P (psia)	432	432	432	432
$\dot{w}_{steam}$ (lb/hr)	$1.138(10)^6$	$1.241(10)^{6*}$	$1.050(10)^{6*}$	$1.092(10)^6$

* 6% added to Tables 3.1.1 and 3.1.6 values to provide for SOX scrubbing losses

The assumptions on combustor/heat exchanger system losses are discussed and justified in Section 4.3.

3.2 CONTRACTUAL GROUNDRULES

While the definition of the thermodynamic cycles and reference designs was part of the study effort, several of the combustor/heat exchanger performance requirements were defined by the D.O.E. as groundrules for the study. These groundrules included the coal specification, the target availability and load factor, the emissions limitations and several other performance parameters. These contractually set requirements are summarized in Tables 3.2.1 thru 3.2.4, and further discussed below.

The coal specifications supplied as contractual requirements are the same as had been specified by the ERDA for the "Energy Conversion Alternatives Systems" studies, (ECAS I and ECAS II). These specifications cover Illinois No. 6 coal, a Montana sub-bituminous coal, and a North Dakota lignite. The interpretation placed upon these coal specifications during the study was that the types of combustor/heat exchangers defined under this contract should be capable of being designed to handle any of the specified coals in a satisfactory manner, but it was not assumed that the coals would be interchangeable in any one given embodiment of the design. All the specified coals have relatively low ash fusion temperatures, and low BTU content, as can be seen in Table 3.2.1. Because of this factor a fourth coal was added to the list. It is representative of the higher heating value, higher ash fusion temperature coals available in ample supply in the Eastern portions of the United States. The inclusion of this class of coal among those to be capable of being accommodated by the combustor/heat exchanger designs tends to make the designs more universally applicable to the full range of American coals. While utilization of the full range of coals was considered for each of the combustor/heat exchanger design studies, all computations, coal and air flow quantities, etc., were based upon the Illinois No. 6 coal composition.

Table 3.2.1 Coal Specifications

	Illinois No. 6 (Macoupin County)	Montana Subbituminous (Rosebud County)	North Dakota Lignite (Mercer County)	Pittsburgh Seam (Washington County)
Reference material	BOM TP-641	BOM TP-529	BOM RI-7158	
Proximate analysis (as received), %:				
Moisture	13.0	24.3	36.7	3.6
Volatile	36.7	28.6	26.6	37.2
Fixed Carbon	40.7	39.6	30.5	52.5
Ash	9.6	7.5	6.2	10.2
Ultimate analysis (as received), %:				
Ash	9.6	7.5	6.2	
Sulphur	3.9	0.8	0.7	2.1
Hydrogen	5.9	6.1	6.9	
Carbon	59.6	52.2	41.1	
Nitrogen	1.0	0.8	0.6	
Oxygen	20.0	32.6	44.5	
Higher heating value (as received,Btu/lb)	10,788	8,944	6,890	13,320
Gross heating value (dry), Btu/lb	12,600	11,300	10,400	
Average softening temperature, F	1979	2224	2280	2000 to 2900
Initial deformation temperature, F	1990-2130	2120-2410	2190-2400	
Fluid temperature, F	2090-2440	2180-2520	2330-2500	
Ash analysis, %				
SiO_2	46.6	22.1	17.9	
Al_2O_3	19.3	15.5	9.9	
FC_2O_3	20.8	6.4	10.2	
TiO_2	0.8	1.2	0.3	
P_2O_5	0.24	0.11	0.4	
CaO	7.7	18.9	23.6	
MgO	0.9	6.6	6.7	
Na_2O	0.2	1.0	7.4	
K_2O	1.7	0.4	0.4	
SO_3	2.4	26.2	21.8	

Table 3.2.2 Fuels

COAL*	Illinois no. 6	
Higher heating value (Btu/lb)		10788
Cost (delivered) ($/MBtu)		1.00
SEMICLEAN LIQUID*	H-Coal	
Higher heating value (Btu/lb)		16700
Cost (delivered) ($/MBtu)		2.25
Process efficiency		0.74
LOW-Btu GASIFIER		
Advanced fixed bed		

*Specified for use in study

Table 3.2,3 Fuel Composition

COAL

Analysis	% By Weight*	Required Reduction For Emission Limits (%)
C	59.6	—
H	5.9	—
S	3.9	83
N	1.0	77**
O	20.0	—
Ash	9.6	98.8
	100.0	

H-COAL SEMICLEAN LIQUID

Analysis	% By weight	Required Reduction For Emission Limits (%)
C	88.2	—
H	7.4	—
S	0.5	—
N	1.3	92**
O	2.4	—
Ash	0.2	20
	100.0	

Ash

Na + K	4.8 ppm
V	2.0 ppm

*As received
**Removal or combustion control to limit the formation of NO_x

TABLE 3.2.4

EMISSION LIMITS

POLLUTANT	FUEL	LB/MBTU
SOX	SOLID	1.2
	LIQUID	.8
	GASEOUS	.2
NOX	SOLID	.7
	LIQUID	.3
	GASEOUS	.2
PARTICULATES	ALL FUELS	.1

In addition to the direct combustion of the specified coals the program groundrules permitted the utilization of liquid and gaseous fuels derived from coal. The heating value and cost of the liquid fuel is defined in Table 3.2.2 The gaseous fuel permitted is low BTU gas as produced by a gasifier integral with the power station.

The limitations on emissions as defined by the contractual groundrules are summarized in Table 3.2.3. These emission tolerances are based upon the guidelines published by the Federal Environmental Protection Agency (Table 3.2.4) and are not uniformly applied in all jurisdictions of the United States. It was recognized that a trend towards stricter emissions limitations exists and this factor was considered in evaluating the future potential of the designs studied.

The contractually specified groundrules on other than fuels and emissions are summarized in Table 3.2.5. For the convenience of the reader, the table presents a comparison of the current contract groundrules with those utilized during the ECAS studies. In nearly all instances they are identical. The performance requirements of particular significance for the design of the CCGT combustor/ heat exchanger include the following:

Item 1B. "Self-protecting for sudden loss of loads". This requirement implies provision for a rapid loss of cooling on the working fluid side of the heat exchanger and thus requires the rapid cessation of heat input on the combustion products side of the heat exchanger surface to avoid excessive temperatures of the heat exchanger surface.

Item 1D. "Black Start Capability". This requires that the power station be capable of startup while it is completely isolated from electrical connections to other stations on the utility system.

Table 3.2.5. Groundrules for Closed Gas Turbine Heater Program

Parameter	ECAS (GE)	Study
I. Type of Load		
A. Type	Baseloaded	Baseloaded
B. Self-protecting for sudden loss of load	Yes	Yes
C. If electrical system failure, capable of continued service at reduced local load	Yes	Yes
D. "Black Start" capability	No	Yes
II. Power Output		
A. Rated Capacity ≡ Continuous electrical power output from the transmission voltage side of the transformer with rated system pressures, temperatures, flows, and normal makeup working fluid	300 MWe	350 MWe
B. Availability ≡ Fraction of time unit is capable of generating rated capacity	$\geq$0.90	$\geq$0.90
C. Capacity Factor	0.65	0.65 - Base
D. Rated capacity is the output when new or refurbished	Yes	Yes
E. Power condition at output side of the transformer		
(1) Frequency	50 Hz	60 Hz
(2) Phases	3	3
(3) Transmission line voltage	500 kV	500 kV
III. Emission Standards		
A. Thermal Pollution		
(1) Wet or dry cooling towers	Yes	Yes
B. Exhaust Emissions	ECAS-Based Table 3.2.4	ECAS-Based Table 3.2.4

Table 3.2.5 (continued)

Parameter	ECAS (GE)	Study
C. Control Techniques		
(1) Fluidized bed	During combustion/ clean fuels	During combustion
(2) Pressurized furnace	Clean fuels	Clean fuels
(3) Conventional furnace	Stack gas cleanup	Stack gas cleanup and/or Clean Fuel
IV. Extent of the Plant to be Studied		
A. For those systems using direct coal combustion, the following subsystems will be included in the study:		
(1) Coal handling equipment at the central station plant, including facilities for coal unloading from rail cars, storage facilities for a 60-day supply, and conveyor equipment (but not including coal transportation from the mine to the plant site)	Yes	Yes
(2) Combustors and emission control equipment	Yes	Yes
(3) Ash and other waste removal and disposal equipment within the plant site property limits	Yes	Yes
(4) The energy conversion systems, including heat input heat exchangers, and electrical generators	Yes	Yes
(5) All auxiliaries and balance of plant, including buildings, land, offices, shop facilities, special maintenance equipment, water treatment equipment, protective devices, etc.	Yes	Yes
(6) Power and voltage control subsystems suitable for isolated operation or operation in parallel with existing generating units in a utility system	Yes	Yes
(7) Heat rejecting subsystem	Yes	Yes
(8) Transformers to raise voltage to transmission line or distribution voltage, but not the high voltage breaker and switch yard	Yes	Yes

Table 3.2.5 (continued)

Parameter	ECAS (GE)	Study
V. Unit Rating and Sizing		
A. Units rated at Middletown, U.S.A. site with the following average conditions:		
(1) Average daily ambient temperature	59 F	59 F
(2) Relative humidity	0.60	0.60
(3) Ambient air pressure	14.7 psia	14.7 psia
(4) Process and makeup water temperature	57 F	57 F
B. Generator and fuel handling capacity will be sized and costed for the maximum power output which occurs at the following conditions:		
(1) Ambient air temperature	20 F	20 F
(2) Relative humidity	0.60	0.60
(3) Ambient air absolute pressure	14.7 psia	14.7 psia
(4) Process and makeup water temperature	39 F	39 F
C. Heat rejection equipment will be sized and costed to meet the nominal plant output during the average dry conditions and still maintain load for the 5 percent summer conditions which are:		
(1) Heat rejection to air		
(a) Air inlet temperature	94 F	94 F
(b) Specified maximum rise	None	None
(c) Ambient air absolute pressure	14.7 psia	14.7 psia
(2) Heat rejection in an evaporative tower		
(a) Available inlet air wet bulb temperature	76 F	76 F
(b) Ambient air absolute pressure	14.7 psia	14.7 psia
VI. Site Location		
Middletown, U.S.A.	Yes	Yes
VII. Coals		
A. Types to be considered		
(1) Illinois No. 6	Baseline	Illinois No. 6
(2) Montana Sub-bituminous		Montana Sub-bituminous
(3) North Dakota lignite	Baseline	N.D. Lignite, [illegible]
B. Coal Properties	Table 3.2.1	Pennsylvania Bituminous, Pgh. Seam, Wash. Cty
C. Coal costs $\$/10^6$ BTU	.85	1.00

Table 3.2.5 (continued)

Parameter	ECAS (GE)	Study
VIII. Coal-Derived Fuels		
A. Types to be considered		
(1) Low-Btu	Integrated Gasifier	Integrated Gasifier
(2) Intermediate Btu	Over the fence	Not considered at present costs
(3) High Btu	"	"
(4) Solvent refined coal (SRC)	"	"
B. Clean fuel specifications, H-coal	Table 3.2.3	Table 3.2.3
IX. Minimum Stack Gas Temperature	250 F	250 F
X. Working Fluids Preparation		
A. Closed Brayton Cycle		
(1) Helium	Over the fence	Over the fence
(2) XeHe mixtures	--	Over the fence
(3) Air	--	Plant treated
(4) Nitrogen	--	Over the fence
B. Rankine Bottoming Cycle		
(1) Water	Plant treated	Plant treated
(2) Ammonia	--	Over the fence
(3) Other organic fluids	Over the fence	Over the fence
(4) Inorganic fluids	--	Over the fence
XI. Efficiency and Heat Rate		
A. Final Study Definitions - For the final study comparisons the following efficiency definitions will be used:		
(1) Efficiency ≡ Overall efficiency (Coal pile to bus bar efficiency)	Yes	Yes
(2) Thermodynamic efficiency = $\frac{\text{Total gross electrical energy generated by prime cycle and bottoming cycle excluding generation by pressurizing turbines}}{\text{Thermal energy into the cycles}}$	Yes	Yes
(3) η_{NPP} = Net power plant efficiency = $\frac{\text{Brayton Gross Generator Power} + \text{Rankine Gross Generator Power} - \text{Plant Parasitic Losses}}{\text{(HHV of Power Plant Fuel)(Fuel Flow Rate)}}$	Yes	Yes

Table 3.2.5 (continued)

Parameter	ECAS (GE)	Study
(4) For the final study evaluations the plant parasitic losses will include the following:	Yes	Yes
(a) Closed Brayton system		
1. Working fluid storage and/or treatment		
2. Working fluid management system (i.e., loop pressure compressor)		
3. Controls		
(b) Rankine Bottoming System		
1. Working fluid storage and/or treatment		
2. Controls		
3. Working fluid feed pumps		
(c) Combustor/Heat Exchanger System		
1. Controls		
2. Feed system		
3. Recirculation fans		
4. Combustion air fans		
5. Combustion gas fans		
6. Slag and ash removal systems		
7. Emissions management systems		
(d) Balance of Plant		
1. Coal handling and supply		
2. Sorbents handling, storage		
3. Stack		
4. Transformer		
5. Plant controls		
6. Closed Brayton intermediate cooling loop pump (if required)		
7. Ranking intermediate cooling loop pump		
8. Cooling tower - wet or dry		
(5) Plant parasitic losses do not include plant and office lighting, heating, and cooling	Yes	Yes
XII. Final Detailed COE Model		
A. COE, mills/kWh = Fuel Costs + Fixed Charges + Operating and Maintenance Costs	Yes	Yes

Table 3.2.5 (continued)

Parameter	ECAS (GE)	Study
B. Fuel Costs	Yes	Yes
C. Fixed Charges	Yes	Yes
D. O & M Charges	Yes	Yes
E. Fixed Charge Rate	Yes	Yes

Category	Percent of Capital, year	Study
Cost of Money	7.5	18% ECAS Base
Federal Income Tax	4.1	
Depreciation (30 yr)	3.3	
Other Taxes	2.8	
Insurance	0.1	
Working Capital	0.2	
Total	18.0	

F. O & M Rates

(1) Cycle Base Values

	Mills/kWh	Study
(a) Closed Brayton Cycle (For all working fluids)		Independent assessments of the cycle base values made for each of the four final systems.
T_6 = 1550 F	1.6	
T_6 = 1750 F+	1.9	
(b) Rankine Cycle (for all operating conditions)		
Water	2.0	
Ammonia	2.5	
Other organic fluids	2.5	

(2) Special Component Maintenance Adders

Component	Annual Adder as % of Initial Capital Cost of Component	Study
Low-Btu Gasifier	6.0	6.0
Pressurized Fluidized Bed	4.0	4.0
Pressurized Furnace	2.0	2.0
Conventional Furnace	0.0	0.0
Emissions Control Equip.	3.0	3.0

(3) Power Split = $\dfrac{\text{Closed Brayton Output}}{\text{Closed Brayton Output} + \text{Rankine Cycle Output}}$ — Study: Yes

Table 3.2.5 (continued)

Parameter	ECAS (GE)	Study
XIII. Initial Study COE Definitions		
A detailed COE evaluation will be accomplished on four power plant systems during Task 2 of the program. During Task I, comparative COE models will be used to evaluate various cycle and heater concepts. Only those elements of the final COE model that are a function of the variables being considered during any given evaluation will be included in that study's limited COE model. Results from these studies will be labeled "comparative COE" and those elements included will be defined.		COE Based on ECAS Costs "Comparative COE" concept not used
XIV. Total Power Plant Cost Distribution		
A. Plant Capital Cost, \$ = Major Components + Balance of Plant + Contingency + Escalation Costs	Yes	Yes
B. Major Components Categories (1) Prime Cycle (2) Bottoming Cycle (3) Primary Heat Input and Fuel System		As per GE Breakdown in ECAS Phase I
C. Balance of Plant Categories (1) Cooling Tower (2) All other component costs (3) Site Labor		As per GE Breakdown in ECAS Phase II
D. Contingency = (0.2) All major components and balance of plant costs		0.2
E. Escalation costs, %/year	6.5	6.5
F. Interest during construction, %	10	10

Item IIA. "Rated Capacity 350 MWe". This specified size of the power generating unit is toward the low end of the range of steam station unit sizes presently being installed. Power Magazine, November 1977 issue, reports that of 28 new projects in 1977 four units had lower capacity than 350 MWe, ranging down to as low as 125 MWe, and 23 units had capacities greater than 350 MWe, ranging up to a maximum of 850 MWe, and with 18 units having capacity greater than 500 MWe. The implication is that commercial closed cycle gas turbine unit sizes may well be expected to be 350 MWe or greater, and that one may anticipate that the designs synthesized in the study should be capable of extrapolation to substantially larger sizes if they are to be commercially feasible. One must recognize at the same time that at 350 MWe the unit size is very large as compared with CCGT heaters that have been operated to date. German experience is primarily with unit capacities of 50 MWe or less.

Item IX. "Minimum Stack Gas Temperature - 250 F". This requirement relates to the minimum stack gas temperature when utilizing a wet scrubber for cleanup of sulphur-oxides from the flue gases. The gas temperature leaving the last heat trap, (the air heater), was assumed to be 300 F on the basis of sulphur corrosion in the air heater with high sulphur coals. Additional discussion of appropriate stack gas temperatures when utilizing SOX scrubbers is presented in Section 4.5.2.

4.0 TECHNICAL DISCUSSION OF FIRED HEAT EXCHANGER DESIGN CRITERIA

Much of the rationale for studying the closed cycle gas turbine power generation concept is that such CCGT systems are capable of accepting their input from the direct combustion coal, with all of the difficulties that entails in coping with the dirty, corrosive, erosive, and polluting combustion products produced. That is one of the fundamental pluses of the CCGT system when compared with open cycle gas turbine systems. The CCGT system is suitable for the direct firing of coal and avoids all the enormous expense and energy losses associated with the conversion of coal into the clean liquid or gaseous fuels that are required for open cycle systems.

This section of this design report will discuss the criteria and boundary conditions that govern the design of coal fired heat exchangers suitable for supplying the heat input to closed cycle gas turbine systems.

4.1 BASIC COAL COMBUSTOR TECHNOLOGY

The technology of coal fired combustor/heat exchangers for the working fluids of closed cycle gas turbine systems would be expected to be similar but not identical to the features and technology of steam boilers. The steam boiler technology thus serves as a base for developing the CCGT fired heater technology. Some of the significant constraints and boundary conditions existing in the direct firing of coal as illustrated by steam boilers of the size range of interest are summarized in the following material. Many of these constraints and boundary conditions are caused by the nature of the coal itself. Coal is a solid fuel whose combustion process involves a complex sequence of events, which may be superficially listed as drying, destructive distillation of volatile content, and finally combustion of solid carbon. The inorganic content of the coal, i.e., the ash, may range up to 10-15% of the weight of the fuel as fired and exercises a strong influence upon many aspects of combustor/heat exchanger configuration and operation. The design responses to the challenges of direct coal firing of steam boilers have developed over many years, and are still in the process of active development at this date.

4.1.1 Description of Coal Combustion Alternatives

The methods of burning coal can be classified as:

- Fixed Bed
- Fluidized Bed
- Suspension Burning
- Pre-combustion Gasification or Liquefaction
- Miscellaneous (Such as Cyclone Firing and Spreader Stoker Firing)

4.1.1.1 Fixed Bed. Fixed bed combustion of coal is the historic method and is still used in small hand fired combustors and larger underfeed and chain grate stoker fired units. However, its zone of economic application is to units very much smaller than those under consideration on this contract. Fixed bed type stokers may find application to combustors of up to approximately 100 million BTU per hour input in the coal, whereas a 350 MWe CCGT combustor would require a coal input equivalent to approximately 3 billion BTU per hour. Thus, there is little prospect for the application of fixed bed coal combustion to the 350 MWe CCGT combustor, and none of the designs studied on this contract utilized fixed bed combustion.

4.1.1.2 Fluidized Bed. The fluidized bed combustion of coal is a fairly recent development in coal combustion, having been in development for only approximately 20 years. A fluidized bed is one in which the solid bed particles are fluidized by passing a gas upwards through them at a velocity sufficient to cause them to be stirred about and "boiled" in a manner strongly analogous to fluid behavior. The gas velocity however is insufficient to entrain and carry away the major portion of the bed particles. Fluidized bed operations have been used extensively in chemical processing for many years, particularly in oil refining operations. Interest in the fluidized bed combustion of coal was greatly increased by the discovery that much of the sulphur present in the coal could be captured as calcium sulphate if the fluidized bed was formed of limestone or dolomite particles. Research has shown that there is an optimum temperature, typically in the range of 1400 F to 1550 F, for sulphur capture. At the optimum temperature calcium to sulphur molal ratios of on the order of 2:1 to 4:1 are required to reduce the emission of sulphur-dioxide to the current United States limit of 1.2 lbs of sulphur-dioxide per million BTU of input.

Heat is released in the fluidized bed as the coal particles are combusted. Bed temperature is maintained at the desired level by heat exchanger surface immersed in the bed. The external heat transfer coefficient to this tube surface is rather high, on the order of 40 to 50 BTU/hr-ft^2-F as compared to the 10 BTU/ft^2-hr-F typically encountered on the convection surface of coal fired heat exchangers. Thus the required bed surface area is reasonable even though the restricted combustion temperature limits the mean temperature difference available for driving the heat transfer.

Typical fluidization velocities in coal fired fluidized beds range from 3 to 12 ft/sec, (calculated as though there were no particles or tubular surface in the cross-sectional area of the bed). The fluidization rate is a compromise influenced by the necessity to maintain the velocity high enough to continue to have fluidization at lower loads, low enough to avoid excessive entrainment and carryover, and high enough to be economical as regards sizing. As the entire combustion air flow passes through the bed the typical fluidization velocities result in coal burning rates in the range of approximately 20 to 80 lbs of coal per hour/ft^2 of bed plan area. The higher fluidization velocities are of most interest, as the required bed area to burn a given quantity of coal is inversely proportional to the fluidization velocity. Carryover of unburned carbon from the fluidized bed is a serious design consideration, particularly at the higher velocities. High carbon utilization is pursued by collecting the unburned carbon and either reinjecting it into the combustor or feeding it to a separate fluidized bed, (carbon burnup cell), which is optimized to burn the finer carbon particles.

The fluidized bed combustion of coal is still in the development stage, there are no commerical installations yet. Its major attraction with respect to large utility sized combustors is that it promises to avoid the requirement for the scrubbing of the flue gases prior to their discharge to the atmosphere and thereby avoid the capital cost, operating costs, and cycle inefficiencies that accompany the application of present day state-of-the-art scrubbers for sulphur-dioxide removal.

The application of fluidized bed combustion to coal fired heat exchangers for CCGT systems is attractive for the same reasons that apply to steam boilers. Additionally, there is an attractive self-limiting metal temperature feature which exists when bed temperature is relatively close to final working fluid temperature, excursions in tube metal temperature are limited. There are however, limitations in that the bed temperature cannot be operated at much higher than 1650 or 1700 F and yet absorb sulphur with an efficient use of limestone. Also an absolute upper limit on bed operating temperature is reached at the temperature where the ash particles begin to soften and agglomerate and destroy the fluidization. For Illinois No. 6 coal this limit is probably about 2000 F. Notwithstanding its limitations the attractions of the fluidized bed concept for combusting the coal of the CCGT systems is sufficiently great that much attention was given to it during the design phase of this contract and several fluidized bed concepts were evaluated. They are discussed in detail later in the report.

4.1.1.3 Suspension Burning. Suspension burning typically refers to combustion of finely divided coal while it is suspended in the combustion air stream. While fluidized bed combustion is a specialized case of suspension burning, we will limit our discussion here to the combustion of pulverized coal, where the coal particle sizes are typically such that 70% of the particles will pass through a 200-mesh screen, (and the size distribution is consistent with a Rosin-Rammler plot). Pulverized coal combustion for power generation has been under development for about 50 years and has reached a state of considerable maturity. Recent developments in pulverized coal combustion have been primarily concerned with optimization to limit the production of the nitrogen-oxides, which are pollutants whose quantity must be controlled by law.

Pulverized coal furnaces have been designed and operated as slag tapping or "wet bottom" furnaces and also as non-slagging or "dry bottom" furnaces. With the wet bottom furnace, the ratio of heat input to containment surface provided is so great that the equilibrium temperature in the furnace is higher than the melting temperature of the ash. The molten ash accumulates on the walls and floor of the furnace and drains off into a slag collecting tank. The wet bottom pulverized coal furnace removes about half of the coal ash as molten slag. However, nearly all pulverized coal furnaces installed in the last 20 to 30 years have been dry bottom furnaces. The dry bottom furnaces are generally less sensitive to coal type, (primarily as reflected in the ash properties), and to operating load, and thus they offer greater operational flexibility. Dry bottom pulverized coal furnaces are designed to cool the suspended ash particles by radiation to the containment surface to a temperature below the ash softening point before the ash particles can be deposited on the walls. Thus the ash which does reach the walls, (which is a small fraction of the total ash), is dry and does not strongly adhere to the walls. The deposits of ash are commonly removed by sootblowers which dislodge the accumulation by jets of air, steam, or water. Because of the method of cooling the ash particles and combustion gases in the furnace, dry bottom pulverized coal furnaces are also called "radiant furnaces". Successful design practive here requires a high ratio of furnace wall surface to heat release rate and thus dry bottom furnaces are typically much larger than slag tapping furnaces.

While attempts have been made to control the emission of sulphur-dioxide from pulverized coal furnaces by effecting a sulphur-dioxide/calcium reaction in the furnace or in the convection passes of the heat exchanger, the only proven method presently available for sulphur-dioxide control is the use of a flue gas scrubber.

Nitrogen-oxide emissions from dry bottom pulverized coal furnaces appear to be controllable by limiting the maximum flame temperature through delayed combustion, or low air temperature, or recirculation of spent flue gas, or staged combustion, or a combination of several methods. Wet bottom pulverized coal furances inherently operate at higher temperatures than dry bottom furnaces and are thus at a further disadvantage because of the nitrogen-oxide problem.

Suspension burning via pulverized coal combustion appears to be a strong candidate for application to the combustors of CCGT fired heaters. Several variations upon the suspension burning concepts were considered during the design phase of the contract and one of the preliminary designs incorporates a dry bottom pulverized coal furnace.

4.1.1.4 Pre-combustion Gasification or Liquefaction. All of the present large commercial processes for coal utilization to supply heat input to power generating cycles presently involve the direct combustion of the coal as opposed to a two-step process of first gasifying or liquefying the coal and then burning the resulting "clean" liquid or gaseous products in combustor/heat exchangers. The groundrules of the present study permit the utilization of coal derived liquid fuels and coal derived low BTU gas, as defined in Table 4.1.1.4, for CCGT system working fluid temperatures of 1750 F or higher. Such coal derived fuels would

TABLE 4.1.1.4

CHARACTERISTICS OF COAL DERIVED FUELS

ECAS PHASE 1	SEMI-CLEAN FUEL (SRC)	INTERMEDIATE-BTU GAS	LOW-BTU GAS (FREE-STANDING)	HYDROGEN	COED	HIGH BTU GAS
HIGHER HEATING VALUE (BTU/LB)	15,682	6350	2535	61,070	17,041	22,674
COST DELIVERED ($/MILLION BTU)	1.80	2.10	2.08	2.50	2.60	2.60
CONVERSION EFFICIENCY (PERCENT)	74	70	68	61	56	50

ECAS PHASE 2

INTEGRATED LOW BTU GAS

$$\sim 85\% = \frac{\text{LHV + SENSIBLE HEAT OF GAS}}{\text{LHV OF COAL}}$$

Source: Energy Conversion Alternatives Study (ECAS) Summary Report, Prepared by NASA-Lewis Research Center, Cleveland, Ohio, September 1977.

be particularly attractive if they were available with very greatly reduced ash and sulphur contents as compared with the original coal from which they were derived. Removing all the ash, including its alkali metal constituents (which are frequently present in only small amounts) would greatly improve the corrosive environment to which the high temperature heat exchanger materials are subjected and further permit those heat exchanger surfaces to be arranged with clearances and in configurations not possible when utilizing direct coal combustion. The optimum combustor and heat exchanger configuration for utilization of a really clean coal derived fuel would likely be quite different than that required for a direct combustion concept. However, there are several factors which discourage strong consideration of utilizing coal derived fuel for the closed cycle gas turbine combustor/heat exchanger:

a. All of the conversion processes involve a substantial energy loss. A listing of some typical conversion efficiencies is presented in Table 4.1.1.4. Even integrated low BTU gasifiers with hot gas cleanup, (systems not yet operational), are projected to involve energy losses on the order of 10 to 20% of the heating value of the original coal. When these coal conversion efficiencies are applied to the efficiency available from candidate closed cycle gas turbine systems, the resulting overall efficiency is too low to be competitive with direct fired steam cycles utilizing existing technology.

b. A partially cleaned coal derived fuel, unsuitable for open GT systems, will probably entail substantial energy losses and capital costs in the conversion process and yet not permit the utilization of combustor/heat exchanger design concepts that are significantly cheaper than those suitable for direct coal firing.

For the above reasons, the combustor/heat exchanger concepts tailored to the special advantages offered by coal derived fuels were not given detailed evaluation in this study.

4.1.1.5 Miscellaneous (Such as Cyclone Firing and Spreader Stoker Firing). Cyclone firing is a unique method of coal combustion that was developed by the Babcock & Wilcox Co. during the 40's and 50's. Crushed rather than pulverized coal is admitted to a relatively small cyclinder along with hot combustion air in such a manner that the crushed coal particles, (commonly passing a four-mesh screen), are thrown to the walls of the cylinder where they are trapped in a molten slag layer and burned by contact with the rapidly swirling air mass. The cyclone furnace is thus a species of slag tapping furnace and subject to its problems with load range, adaptability only to certain coals, and high nitrogen-oxide emissions. The concept worked very well with some low fusion temperature coals but has not found application in recent years, presumably because its sponsors have not been able to modify it to meet the required nitrogen-oxides emissions limits. Slagging or wet bottom furnaces are attractive for CCGT fired heaters in those instances where the working fluid design temperature is so high that the heat exchanger hot wall temperature will be higher than the fusion temperature of the coal ash, thus precluding the use of dry bottom furnaces and fluidized beds. As a rough approximation it is believed that this condition will be reached when the working fluid temperature is much higher than 1750 F. Naturally, this temperature will vary with the coal ash composition, and the 1750 F applies to the Illinois No. 6 coal. Also there are some coals not suitable for cyclone firing. The cycles examined during this study included one whose maximum working fluid temperature is 2250 F, and cyclone firing is a strong candidate for this application.

Spreader stoker firing is a hybrid between fixed bed and suspension burning in that randomly sized coal is flipped into the furance at some height above a grate. The fines content of the coal is burned in suspension and the larger pieces fall to the grate where they burn in the "fixed bed" manner. Spreader stoker firing has pretty much displaced underfeed stokers and chain grate stokers in its range of economic application, but the upper limit of that range is on the order of 400 million BTU of input per hour. Above that size, the furnace required for a spreader stocker usually results in increased costs as compared to pulverized coal or cyclone furnace fired units with narrower and higher furnaces. A single 350 MWe CCGT unit will require about 8 times the 400 million maximum capacity of spreader stoker units so spreader stoker firing was not evaluated in detail for application in this design study. It should be recognized however that the design principles and configurations appropriate for dry bottom pulverized coal unit are not drastically different from those required for spreader stoker fired units, so that it is likely that the dry bottom pulverized coal designs developed during this study could be modified to accept spreader stoker firing should unit size or some other special circumstance make that advisable.

4.1.2 Summary of Coal Combustion Alternatives

To summarize the previous material, the coal combustion alternatives that are available for the 350 MWe CCGT combustor/heat exchanger on the basis of existing technology include suspension burning, i.e., (pulverized coal), fluidized bed burning and slag tap pulverized coal or cyclone furnace combustors. Gasifiers and liquefiers are eliminated because they are uneconomic as applied to 350 MWe CCGT cycles. Fixed bed combustors, chain grate stokers and underfeed stokers are inappropriate for the size range. All of the contractually specified coals appear to be suitable for firing in dry bottom pulverized coal furnaces or fluidized beds, although lignite has caused considerable difficulty in some existing dry bottom pulverized coal boilers and fluidized bed combustion has not been sufficiently developed to state with certainty that it will be adequate for combustion of all the contractually specified coals. High ash fusion temperature coals are not suitable for "wet bottom" units. A major modifier upon the applicability of the various coal combustion methods to the CCGT systems under study is the maximum working fluid temperature. The zones of applicability of the various combustion systems as a function of maximum CCGT working fluid temperatures are summarized in Fig. 4.1.1.

4.2 PRESSURIZED vs UNPRESSURIZED COMBUSTOR/HEAT EXCHANGER OPERATION

The pressurized combustion of coal has several attractive aspects. Pressurized combustion permits intensification of the heat transfer characteristics, both in the furnace and in the convective heat transfer regions. Combustion is intensified so that the volume required for combustion may be reduced. Pressurized fludized beds seemingly absorb sulphur better than AFB's. Additionally, cycles have been devised which incorporate gas turbine components in the combustion products stream, and thereby improve the overall efficiency of the cycle. Considerable attention was given to this type of hybrid closed cycle/open cycle system in some of the ECAS studies. For the present contract study, the Department of Energy specified that none of the combustion products of the direct combustion of coal were to be passed through any turbomachinery. It is believed that this was a wise decision in that the hybrid closed/open cycle has some inherent economic disadvantages as compared to fully open/combined cycles. Thus, if it eventually proves possible to overcome the terribly difficult problems of cleaning up the product gas stream resulting from direct coal combustion, it will make more economic sense to apply that clean up technology to coal fired open/combined cycles than to hybrid closed/open cycles. The coal fired CCGT cycles and technology by contrast are expected to find near-term application, providing an improvement in coal pile to bus bar efficiency while at the same time avoiding the problems of turbomachinery corrosion and erosion by isolating the turbine from the products of combustion.

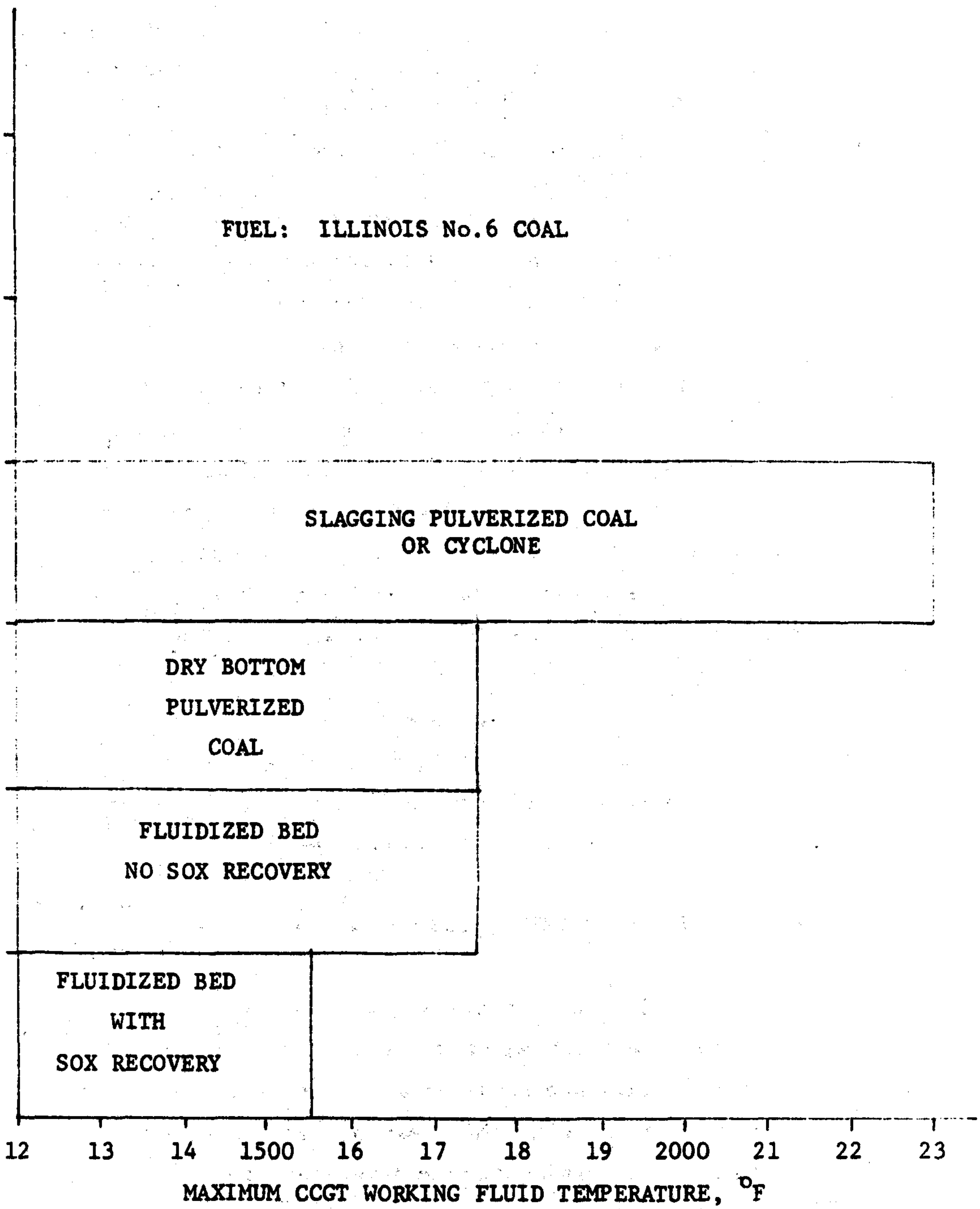


Figure 4.1.1 Zones of Combustor Applicability

Having disposed of those situations in which the products of coal combustion are utilized in the turbomachinery loop the question still remains as to whether pressurized combustion is economically justified, even without passing the products through a turbine to extract power, because of the reduction afforded in heat transfer surface. This option has been available to the steam boiler industry for many years, and has not been applied, implying that it is uneconomical in that service. An elementary analysis of the economic effects of pressurized operation upon the overall CCGT cost of electricity is contained in Tables 4.2.1 and 4.2.2. These analyses cover only combustor/heat exchanger cost effects and air compressor power requirements. The balance shown here is sufficient to indicate that pressurized operation is uneconomical. On the basis of the groundrules and these analyses, all of the combustor/heat exchanger designs studied under this contract have been designed for operation at atmospheric pressure. This does not preclude the pressurization of windboxes sufficiently to force the combustion air into the combustion process, nor does it preclude the so-called "pressurized operation" of furnaces in which combustion takes place at a sufficiently high pressure, usually a few inches of water gauge, so that the combustion gases can be forced across the convection surfaces and out the stack without having to use an induced draft fan.

4.3 COMBUSTOR/HEAT EXCHANGER ENERGY LOSSES AND THEIR CONTROL

The cost of fuel supplied to the generating station is minimized as the losses incurred in converting the chemical energy content of the fuel into heat energy content of the working fluid are minimized. The minimization of these losses has been a continuing objective in the steam boiler and other fossil fuel utilization industries. An explanation and methodology for calculation of energy conversion efficiency is contained in many standand texts and is well explained for instance in the publication "Steam" issued by the Babcock & Wilcox Co. As in all conversion processes the efficiency of conversion falls short of 100 percent. The major sources of energy loss are summarized below and the losses for

Table 4.2.1 Sample Calculation of Combustion Air Compression Loss

$$\text{Compressor work required} = \dot{m}\, C_p\, T_1 \left[\left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} - 1\right] \frac{1}{\eta_c}$$

where:

$\dot{m}$ = 8.0 lb/10^4 BTU

C_p = 0.24 BTU/lb$^{\circ}$R

T_1 = 540°R

γ = 1.4

η_c = 0.8

For $\frac{P_2}{P_1} = 2$

$$\text{Compressor work} = 8.0\ (0.24)\ 540\ \left[2^{0.286} - 1\right] \frac{1}{0.8} = 283.84\ \text{BTU}/10^4\ \text{BTU input}$$

For cycle efficiency, $\eta = 0.5$, compression loss = 283.84 $(1 - \eta)$

$$\text{Net cycle efficiency} = \frac{10^4 \eta - 283.8\ (1 - \eta)}{10^4} = 0.486$$

RESULTS

COMPRESSION RATIO	BASIC CYCLE EFFICIENCY				
	0.500	0.450	0.400	0.350	0.300
	← NET EFFICIENCY →				
1	0.500	0.450	0.400	0.350	0.300
1.5	0.492	0.441	0.390	0.340	0.289
2.0	0.486	0.434	0.383	0.331	0.280
2.5	0.480	0.428	0.375	0.325	0.273
3.0	0.476	0.424	0.371	0.319	0.266

Table 4.2.2 Pressurization Effect on Heat Exchanger and Fuel Cost (mill/kw-hr)

(+) Savings (-) Losses

PR	Heat Exchanger Cost	FUEL COST $1/10^6 BTU	FUEL COST $2.0/10^6 BTU
1.0	No Change	No Change	
1.5	+0.52	-0.33	-0.66
2.0	+0.81	-0.59	-1.18
2.5	+0.98	-0.87	-1.74
3.0	+1.11	-1.08	-2.16

several candidate combustion systems are summarized in Table 4.3.1. These losses were utiilized in defining the reference design data of Tables 3.1.7 and 3.1.8.

4.3.1 Basis for Combustor/Heat Exchanger Efficiency Calculations

Conventional practice in the United States is to consider that the entire higher heating value of the fuel as fired is available for conversion into heat energy in the working fluid. The higher heating value of a fuel is determined experimentally in such a way that all of the products of combustion are oxidized to their most oxidizable state and all of the products of combustion are returned to ambient condition. Thus nearly all of the water produced by combustion, and the moisture content of the fuel, is condensed during the determination of the higher heat of combustion. The latent heat of this condensation is considered as part of the heat available from the combustion of the fuel. The practical effect of this base for calculation is that fuels having high moisture and/or a high hydrogen content are inherently penalized in the efficiency calculations because it is not practical in the state-of-the-art heat exchangers to condense the moisture content of the flue gases and thereby recover the latent heat of evaporation of the water.

4.3.2 Efficiency Losses Associated with Discharge of Hot Flue Gases to the Stack

In fired heat exchanger practice it is not practical to cool the products of combustion to room temperature prior to their discharge to the atmosphere through the smoke stack. This comes about for two reasons. First, there is typically no working fluid available at a low enough temperature to accomplish such cooling. Secondly, if the flue gases are cooled much below 300 F some of the sulphur trioxide present in the flue gases will condense on the heating surfaces as sulphuric acid, typically resulting in their rapid corrosion. The

Table 4.3.1 Combustor/Heat Exchanger Efficiency Comparison *

	Fluidized Bed	Pulverized Coal	Cyclone Comb.
Sensible Heat Loss	5.7	5.7	5.5
Latent Heat Loss	5.9	5.9	5.9
Unburnt Carbon Loss	1.0	0.5	0.1
Radiation	0.3	0.3	0.3
Unaccounted For	1.5	1.5	1.5
Calcination	0.5	0.0	0.0
Combustor/Heat Exchanger Efficiency (%)	85.1	86.1	86.7

* Illinois No. 6 coal

boiler industry has developed correlations and design procedures which permit the optimization of the design heat exchanger train exit gas temperature as a function of the nature and sulphur content of the fuel. For the purposes of this study it has been assumed that all heat exchanger trains would be designed for an exit gas temperature of 300 F. This is a representative temperature typically suitable for the coals specified. The sensible heat loss in the flue gas leaving the heat exchanger train thus is the product of the gas weight flow, the average specific heat of the flue gas, and the 220 F temperature differential between 300 F and the 80 F base temperature. Of the 3 quantities involved in this relation, only the weight flow of flue gases is subject to control by the designer, and that only modestly controllable. The economics of combustion require that more air be supplied to the coal combustion equipment than is theoretically necessary to accomplish complete combustion of all combustible material in the fuel. This is because the losses associated with an excess supply of air are less than those associated with the incomplete combustion and loss of available chemical energy which ensues if insufficient air is supplied. Typical values of excess air needed by combustion equipment are 5% for oil or gas burners, 10% for cyclone furnaces, and 15% for pulverized coal burners. The excess air required to be supplied to a fluidized bed so as to optimize energy release is as yet incompletely determined, but for purposes of this study is assumed to be 15%. In addition to the air supplied to the burning equipment, the operation of most combustor/heat exchangers is such that much of the convection surface operates at sub-atmospheric pressure, i.e., the flue gases are sucked over the heating surfaces rather than blown over them, and some air leaks through the containment walls of the heat exchanger into the flue gases, is heated to 300 degrees and discharged at that temperature from the last heat exchanger. This also represents an efficiency loss. A typical value for air infiltration is 10%, although this obviously may vary widely depending upon the size of the equipment relative to the gas flow and the design and condition of the containment walls. For purposes of this heat exchanger study, the total excess air leaving the last heat exchanger was assumed to be as shown in Table 4.3.2, and the sensible heat losses associated

Table 4.3.2. Operating Flowrates

	1550/1550 FB	1550/1550 PC*	1750/1550 FB	2250 Cyclone*
Coal Flow	271,000	286,000	266,000	266,000
% Excess Air at Burners	15	15	15	10
Air Flow	2,570,000	2,710,000	2,520,000	2,410,000
Flue w/Leak, (Adds 10% Excess Air)	3,040,000	3,200,000	2,980,000	2,870,000

* Includes 6% extra coal to cover energy losses in wet SOX scrubber system

with the discharge of these gases to the atmosphere are shown on line 1 of Table 4.3.1.

4.3.3 Latent Heat Losses

The heat loss assigned to the combustor heat exchanger on the basis of the higher heating value calculation base had been discussed in 2.3.1. These losses are indicated on line 2 of Table 4.3.1 for the base Illinois No. 6 coal

4.3.4 Unburned Combustible Losses

None of the combustion processes are perfect, there is always some combustible material that is not completely oxidized. With the combustion equipment considered for application to the CCGT system these unburned combustible losses consist almost entirely of unreacted carbon. The losses in the form of carbon monoxide and/or hydrocarbons are very small. The magnitude of the unburned combustible loss varies with the coal and with the type of combustion equipment. Typical values assumed for this study are indicated on line 3 of Table 4.3.1. The losses associated with the pulverized coal and cyclone furnace combustion are well established. The loss assumed for fluidized bed combustion is less well established and at the typical fluidization velocity of 12 ft/sec is strongly dependent upon being able to separate the carbon bearing fly ash from the spent ash in the cyclone dust collectors, with the carbon bearing ash being returned to one or more beds for completion of combustion.

4.3.5 Radiation Loss

The radiation loss covers the heat losses from the walls of the combustor/heat exchanger. Its magnitude thus depends on the temperature of the walls and their area relative to the capacity of the unit. For large fired heaters, such as a single 350 MWe unit, the surface to volume ratio is quite favorable and this loss is very low. A simple approximate method is provided by the chart of Fig. 4.3.5 which was published by the American Boiler Manufacturers Association. This chart was utilized to calculate the radiation losses of line 4 of Table 4.3.1. Where the 350 MWe output of the CCGT unit was divided among several combustor/heat exchangers the radiation loss was calculated by entering the chart with an output equal to the total output divided by the number of heat exchangers.

4.3.6 Unaccounted for Loss

The bulk of the energy losses associated with the conversion of chemical energy to heat energy are covered in the items above, but there remain some not covered. These include, for instance, the losses associated with removing the ash from the combustor in a hot or even molten state and not recovering its heat, and the losses associated with drips, and sootblowing. These losses by convention in steam boiler practice are usually assigned a value of 1.5% and a similar value has been assigned for the CCGT design studies.

4.3.7 Calcination Loss

The fluidized bed combustor provides an additional reaction between the limestone, calcium carbonate, and the sulphur content of the coal. This reaction when carried to $CaSO_4$ actually gives off additional heat. However, much of the excess limestone provided is calcined to calcium oxide, with the absorption of heat. This heat loss is relatively minor but is entered on line 6 of Table 4.3.1 for the fluidized bed combustor/heat exchanger.

4.3.8 Overall Combustor/Heat Exchanger Efficiency

The overall combustor/heat exchanger efficiencies as calculated on the conventional basis are shown on line 7 of Table 4.3.1. These values are typical

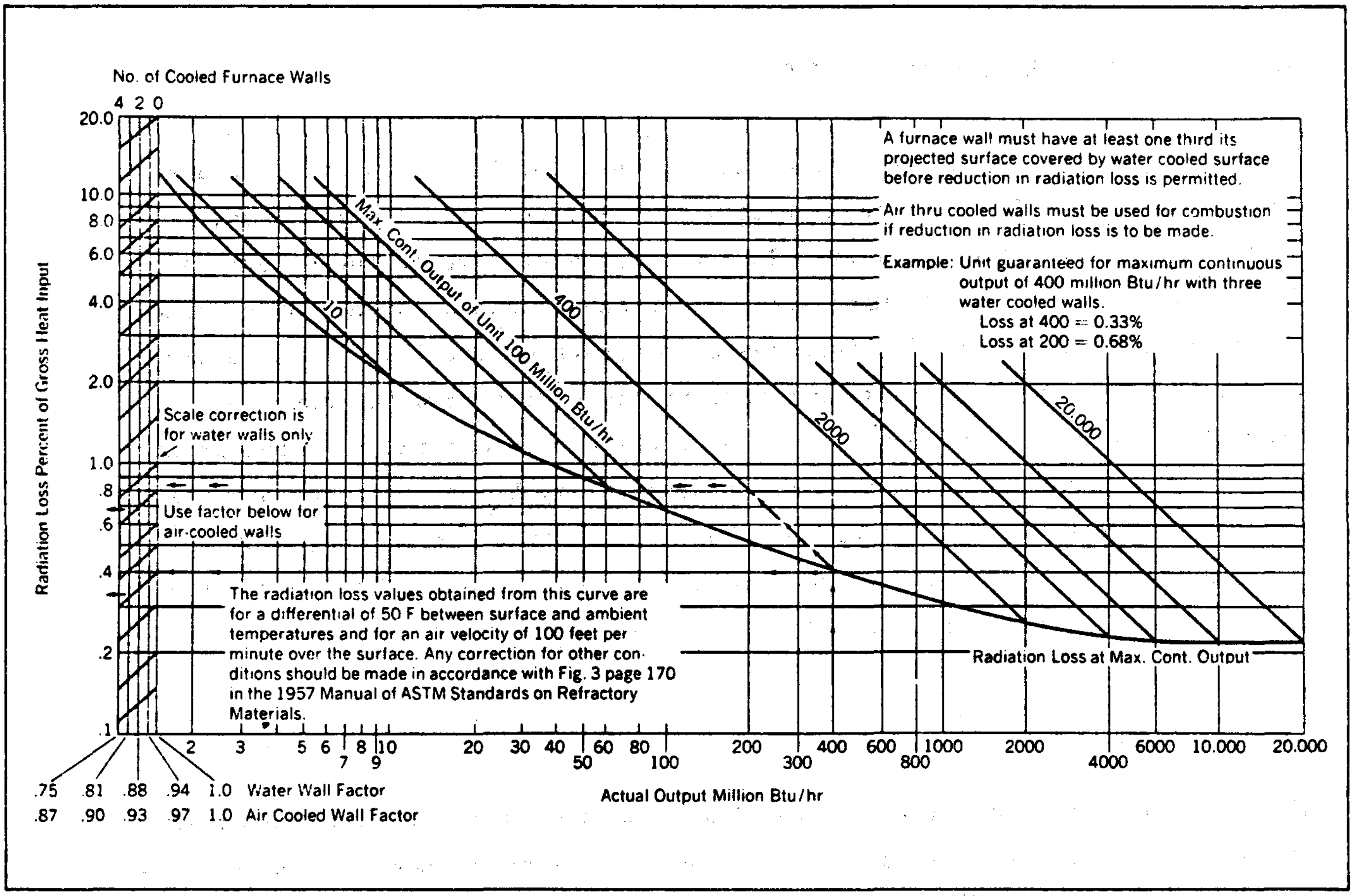


Figure 4.3.5 Radiation Loss in Percent of Gross Heat Input (American Boiler Manufacturers Association)

of what is attainable from steam boilers of comparable size. It will be noted that the losses entailed in crushing, pulverizing, and forced and induced draft fans are not included; these are typically accounted for in the "house load" station account. The loss in drying the coal is included as the calculation implicity assumes that it takes place during the combustion process.

4.4 HEAT EXCHANGER SURFACE SIZING

The sizing of the combustor/heat exchanger is a complex operation in which judgmental trade-offs must be made between installed cost, operability, operating costs, and durability. The influences of some of these factors are discussed in the material immediately below and in the sections which follow. Additionally, correct sizing requires a knowledge of the laws of convective, radiative and conductive heat transfer, pressure drop computations, and elementary structural mechanics. Much of the basic computational background necessary for sizing combustor/heat exchangers may be found in the publication "Steam" by the Babcock & Wilcox Co. The heat transfer and pressure drop correlations contained within that publication were used extensively during the design phase of this contract.

4.4.1 Air Heater Sizing

The air heater is typically the last heat trap in a steam boiler installation and will fulfill the same function in the CCGT combustor/heat exchangers studied under this contract. With a steam boiler installation, the feedwater temperature to the economizer is typically in the range of 450 F to 550 F, being set by the regenerative feedwater heating arrangements of the turbine cycle. Thus, the gas temperature entering the air heater of such a steam boiler installation can easily be arranged to fall in the region between 600 F and 800 F, and it is relatively simple to proportion an air heater to provide a 300 F exit gas temperature under those circumstances. The capacity of an air heater as a heat trap is limited by two factors. The weight flow of flue gas is greater than the weight flow of air because much of the coal substance has been gasified and added to the flue gas, and because of the in-leakage of air through the boundary walls of the combustor/heat exchanger. Additionally, the higher temperature and higher moisture content of the flue gas as compared with air give it a higher specific heat. Thus, the air temperature rise in an air heater is greater than the flue gas temperature drop, and if a given flue gas exit temperature is to be maintained, (i.e., the 300 F of the groundrules), there is a limitation on the inlet gas temperature that can be accommodated by an air heater. The relationship among these factors for a typical air heater installation is shown in Fig. 4.4.1. It is seen from Fig. 4.4.1 that the air heater surface requirement begins to escalate rapidly at about 800 F inlet gas temperature and becomes asymptotic before 1100 F is reached. A normally sized air heater represents about 10% of the cost of a steam boiler installation, and is constructed entirely of mild steel. If the entering gas temperature is designed to be much higher than 800 F, the surface requirement will escalate rapidly and it will also be necessary to use alloy steel in the air heater surface and ducting. On the basis of this comparison, the designs of the CCGT combustor/heat exchangers studied under this contract were generally constrained to provide a gas temperature entering the air heater no greater than 800 F.

4.4.2 Design Flue Gas Temperatures and Mass Fluxes

The designer of a CCGT combustor/heat exchanger system is naturally faced with questions regarding the appropriate criteria to which to size the heat transfer flow passages and combustion spaces onthe flue gas side of the heat transfer surfaces as well as on the working fluid side of the heat transfer surface. The considerations involved in choosing the working fluid, and in choosing the pressure drops on the working fluid side, are discussed in detail in the "Cycle Analysis and Working Fluid" report. For purposes of this report, we may accept that the working fluids have been defined and that the allowable

30% OF TOTAL COAL MOISTURE REMOVAL
BY SEPARATE DRYER

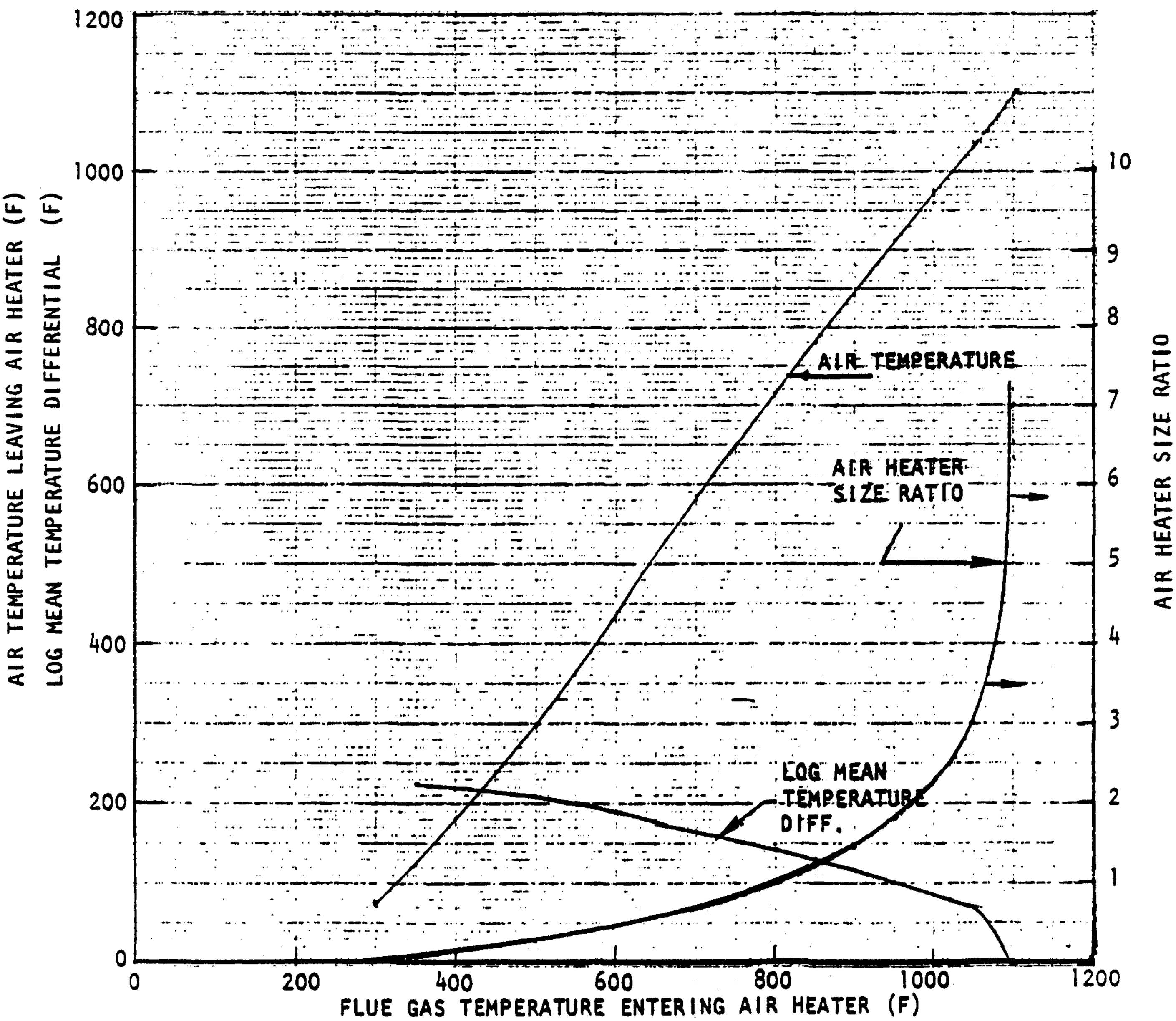


Figure 4.4.1 Air Heater Characteristics Based on Ambient Inlet Air

pressure drops are specified in the interface design criteria of Tables 3.1.1 thru 3.1.4. The challenge remaining with respect to working fluid then is to arrange the circuits to achieve the necessary heat transfer, maintain surface temperature within acceptable boundaries and provide both acceptable working fluid pressure drops and fluid distribution among the parallel circuits involved.

On the flue gas side of the heat exchanger surface the definition of appropriate design values for gas temperature and gas mass fluxes throughout the combustor/heat exchanger is more complex. The overriding considerations here when firing coal directly are the requirements to be able to operate continuously by keeping the flue gas passages open, (i.e., avoidance of plugging by slag or ash deposits), and to avoid excessive erosion of heat exchanger surfaces by ash particles entrained in the flue gas stream. These criteria are discussed in Section 4.4.3 below. The economic balance between surface furnished, (and thus capital cost), as against fan power required is a secondary consideration. General steam boiler practice would indicate that application of solely the economic balance, (no consideration of ash), would result in only moderately smaller installations, with less combustion volume and higher gas mass fluxes over the convection heating surface.

4.4.3 Coal and Coal Ash Design Affects

Influences of coal type upon design were briefly discussed in Section 4.1. All of the coals specified for study under this contract have relatively high volatile content and appear to be suitable for firing in the type of combustors enumerated in 4.1. The ash content of all coals exercises a profound influence upon the design criteria for coal fired combustor/heat exchangers in this size range. The combustion process strongly influences the nature of the residual ash present in the flue gases as they pass through the combustor and convection surfaces of the fired heater.

4.4.3.1 Pulverized Coal and Cyclone Furnaces. If the combustion process is one which results in coal particle temperatures high enough to melt the non-combustible ash, (as for instance with dry bottom pulverized coal and with cyclone fired furnaces), then the design must provide for the transition of the ash particles from the molten to the dry solid state without the accumulation of large and unmanageable deposits of semi-molten sticky ash, which can plug up gas passages, upset heat transfer balances, and force shutdowns. For dry bottom furnaces, it is thus necessary to design for a furnace exit temperature that will avoid these unmanageable conditions. For wet bottom furnaces, (such as the cyclone fired furnace), an additional constraint exists, the gas temperature in the slag tapping portion of the furnace must be maintained high enough so that freely flowing slag can be tapped from the furnace at the design low load. Consistent with what is believed to be acceptable practice, the CCGT combustor/heat exchangers were constrained to provide a gas temperature entering the convection surface of both the dry bottom and wet bottom furnaces of no greater than 2100 F. Additionally, the gas temperature leaving the freely flowing slag region of the cyclone combustor fired heat exchanger was designed to be no less than 2600 F at full load. These design conditions are believed entirely appropriate for the Illinois No. 6 and the Montana sub-bituminous coals. Experience with firing North Dakota lignite is limited and lignite firing of dry bottom furnaces has sometimes resulted in unmanageable deposits on furnace walls and convection surfaces. A more conservative furnace design may therefore be indicated for the North Dakota lignite.

4.4.3.2 Atmospheric Fluidized Bed Combustors. The atmospheric fluidized bed combustor is constrained to operate at a nominal bed temperature low enough so that the residual ash particles are below the temperature at which they will stick together and form large enough lumps to be a problem. Thus most of the AFB's considered involved operation at temperatures in the neighborhood of 1650 F and no bed was designed to operate at a bed temperature higher than 2000 F.

4.4.3.3 Convection Surface. The convection surfaces of pulverized coal and cyclone combustor fired heat exchangers are subject to ash accumulations even at temperatures well below the coal ash initial deformation temperature. Such accumulations are sometimes bonded to the heating surface by alkaline sulfates whose sublimation and liquefaction temperatures are well below the equivalent temperature for the bulk of the ash. The steam boiler design practices that have evolved to cope with this "fouling" tendency have been to provide relatively wide spacing of tubes in the higher temperature zones of the convection surface, say above 1400 F, and to provide tube bank depths that can be readily cleaned by powerful traveling sootblowers. This practice appears appropriate as well for CCGT combustor/heat exchanger surface and was adopted for the wet and dry bottom designs studied under this contract. Less experience is available on the fouling effects of the ash that passes over the convection surface of fluidized bed CCGT combustor/heat exchangers. The flow rate of coal ash and limestone residue is expected to be as great or greater than that existing in a dry bottom pulverized coal unit but the particle size may be somewhat larger and will contain a considerable excess of CaO. Early indications are that this ash is relatively non-fouling and shows little tendency to accumulate on the convection surface. In the interests of conservative design the convection surface arrangement of the AFB combustor/heat exchangers has been constrained to be consistent with that of pulverized coal fired units in the same temperature range. Provision for the installation of sootblowers on a pattern similar to that typical for above coal units is also made.

4.4.3.4 Erosion. When the coal ash particles entrained in flue gas reach a temperature where they are truly hard and solid, they become erosive. This is especially true for the coal particles resulting from the combustion of pulverized coal, but is also true for cyclone combustor particles and expected to be true for fluidized bed particles. Experience with steam boilers has shown that if excessive erosion of tubular heating surface is to be avoided the gas velocity over the tubes must be limited to 75 ft/sec or less, and that configurations tending to concentrate the ash in narrow bands must be avoided. It is this erosion limitation which governs the sizing of the flue gas flow passages in the temperature range from above 1400 F to 300 F. All of the designs synthesized for this contract are constrained to avoid flue gas velocities higher than 75 ft/sec in this temperature region.

4.5 EMISSIONS CONTROL

Coal firing has always tended to produce unpleasant pollutants, sulphur oxides, soot, and ash particles. The coal combustion processes as they were developed were guided at least in part by efforts to minimize such emissions. Over the last 10 or 15 years, however, the influence of emissions control upon acceptable coal combustion processes has greatly strengthened. The Federal Government now sets stringent emissions limitation on many pollutants, and some state, county, or city jurisdictions impose even stricter limitations. The emissions limitations specified by the groundrules for this design study are summarized in Section 3. The pollutants covered include particulate matter, sulphur oxides and nitrogen oxides. One should recognize however that most jurisdictions also require that the appearance of the smoke from the stack not to exceed that of a No. 1 Ringleman chart, (which is a relative measure of the opacity of the stack plume and in effect requires that the plume be no darker than very light grey). Additionally, some jurisdictions regulate carbon monoxide and hydrocarbons. Additional pollutants that may be troublesome include trace poisons, particularly heavy metals that may be present in the coal ash, acid smut which consists of very fine carbon particles upon which sulfuric acid has condensed, and condensed water mist.

The design constraints imposed upon the CCGT combustor/heat exchangers studied under this contract are summarized in the paragraphs below. As will be seen, it is expected that it will be possible to design CCGT combustor/heat exchangers to comply with existing and probable future emissions control requirements. However, it seems important to recognize that the whole area of emissions controls is undergoing continuous change, and in fact the U.S. Congress has mandated that the legal limits be re-examined every four years. The regulatory agencies recognize that the electric utility industry cannot be shut down by some quirk of the emissions regulatory process, and emissions limits must be set consistent with some reasonable interpretation of the state-of-the-art in attaining those limits. However, it is possible for some particular technology, (as for instance, a particular combustion method), to be found wanting in its emissions control capabilities with respect to some new or modified limitation. Thus it is conceivable that if CCGT combustor/heat exchanger technology is developed exclusively along the lines of one particular combustion process, technical developments in emissions control and regulations could obsolete that combustion process and require a return to steam based technology, where there are many alternative combustion processes available.

4.5.1 Particulates

The particulate emissions from coal fired CCGT combustor/heat exchangers may include ash particles; plus limestone, calcium oxide, calcium sulfite, and calcium sulfate particles if a fluidized bed is utilized; smoke, i.e., very fine particulate matter; droplets of water or acid; or acid condensed upon particles. For the purposes of this study, it has been assumed that all these emissions will be controllable with any of the combustion systems so far described.

Fly ash will be controllable with electrostatic precipitators on pulverized coal, cyclone combustor and spreader stoker fired combustors. The fabric filter (bag house) will be applicable to fluidized bed combustors, (for which fly ash emissions tend to be very high because of the relatively large amount of limestone fed to the combustor along with the coal). As all of these particulate control concepts are "add-on types" they tend to exercise little constraint upon the design of the combustor/heat exchanger.

Smoke control is typically obtained by providing enough combustion volume, and sufficient mixing and retention of coal particles in that volume, so that the fine carbon particles are burned. For this design study, it has been assumed that all of the combustion concepts discussed will be capable of meeting No. 1 Ringleman chart requirement without requiring special design configurations. The quantity of excess air also plays a role in controlling smoke production and it is conceivable that the 15% excess air to the fluidized bed that has been assumed in the design studies will not be sufficient to prevent objectionable smoke under all circumstances and with all coals.

Trace minerals, acid smut, and superfine particulates have not been considered in the design. Condensed mist produced by a scrubber, if applied, is evaporated by reheating prior to discharge.

4.5.2 Emissions of Sulphur Oxides

The groundrule requirements for sulphur oxides emissions is given on Sec. 3.2 as 1.2 lbs of sulphur oxides, (expressed as sulphur dioxide), per million BTU of heat input to the combustor/heat exchanger. This is the present requirement of the Federal Environmental Protection Agency and is a minimum required by most jurisdictions. On this basis all of the specified coals would require some cleanup of sulphur oxides if one assumed that all of the sulphur content were converted to sulphur dioxide. It is conceivable under this rule that the Montana coal, with its low sulphur content, might squeak by with minimum attention to sulphur removal. However, the U.S. Congress in 1977 mandated that "best available technology" be utilized to minimize the sulphur content of the exhaust fuel gases regardless of the initial sulphur content in the coal as mined. The process of defining the allowable emissions is presently underway. It is anticipated that the eventual rule will require that at least 85% of the sulphur be removed between the time the coal is mined and the time the stack gases are discharged to the atmosphere, and that as an absolute limit no more than 1.2 lbs of sulphur oxides emissions expressed as sulphur dioxide be emitted per million BTU of heat input to the combustor/heat exchanger. It has been assumed that this regulation will apply to the CCGT combustor/heat exchangers at the time they go into service. The effect is that positive means for sulphur oxides control will be required on all combustor/heat exchangers regardless of the coal they burn.

Sulphur oxides emissions control for the conventional coal firing methods, pulverized coal, stoker, and cyclone combustor firing, has so far been achievable only with add-on wet scrubbers. These scrubbers wash the flue gases leaving the last heat trap, subjecting them to intimate and prolonged contact with an alkaline solution and chemically combining the sulphur dioxide and trioxide with the alkaline solute. The add-on scrubber exercises little or no constraint upon the basic design for the combustor/heat exchanger. Its effects are upon the economics of the entire installation. The scrubbing requires substantial capital and operating costs and degrades the overall thermal efficiency of the installation. Many scrubber concepts are available for adoption and it is believed that to date no one concept has clearly demonstrated overall economic superiority. For the purposes of this study, it was assumed that limestone scrubbing would be utilized.

The limestone scrubbing solution operates at adiabatic equilibrium temperature, which is largely a function of the inlet temperature of the flue gases and their moisture content and in this study is expected to be the area of 125 F to 130 F. While many American installations successfully discharge their stack gases at this low adiabatic equilibrium temperature, and other installations reheat the stack gases to approximately 175 F, the groundrules for this study have specified that the stack gases be reheated to a minimum temperature of 250 F, (see Section 3), and this groundrule has been followed in the study. The scrubber then makes its influence felt upon the basic thermodynamic cycle in its requirement for power for pumping and flue gas movement, and for reheating the stack gases. The reheat function especially may be worked into the thermodynamic cycle as it can utilize low grade heat from which power has already been extracted. For the purposes of this study, however, it was simply assumed that the power and reheat demands of a SOX scrubber system required that 6% more coal be burned and 6% more heat be transferred to the cycle working fluids than would have been the case without scrubbing. This level of penalty appears to be well supported by information available in the literature, (see pages A7 and A9)*. However, it is not consistent with the penalty assigned by the General Electric Co. in their ECAS studies where they assigned an overall coal pile to bus bar efficiency for their reference steam plant of 36.2% with no stack clean-up and 31.8% with a scrubber, (and 250 F stack temperature). This requires that the coal input for the plant supplied with a scrubber be approximately 14% higher than a plant without a scrubber, and is believed to assign much too high a penalty to scrubbing. They used hot air for reheating, which is very wasteful of energy.

The fluidized bed combustion concept is attractive for large combustor/heat exchangers primarily because it is expected that the operators will be able to control sulphur dioxide emissions to meet legal requirements without using tail-end scrubbing. The absorption will take place in the fluidized bed and the sulphur will be removed as a solid calcium and/or calcium magnesium compound. Two advantages are foreseen, the efficiency penalty accompanying tail-end scrubbing will be largely eliminated, and the disposal of the dry residue resulting from the fluidized bed operations will be much simpler and cheaper than the disposal of the wet sludge resulting from tail-end sulphur scrubbing. Much of the fluidized bed experimentation that has been done to date has been small scale, conducted under a wide variety of operating conditions and by many different investigators. As a result the data at this time is not sufficient to define exactly what operating conditions, especially as regards the ratio

* "Final Report on Economic Evaluation of Stack Gas Desulfurization for a Power Plant Located in the Mohawk Valley Region of New York State" by J. N. Genco and H. J. Rosenberg of Battelle Columbus Laboratories, 2/28/77, to F.E.A.

of calcium supplied in the limestone to sulphur supplied in the coal, will be required in the fluidized beds to attain the legally required sulphur emissions reductions. For the purposes of this study it has been assumed that sulphur removal can be effected provided that the molal ratio of calcium supplied in the limestone to sulphur supplied in the coal is 2.75 or greater. It should be recognized that this is an approximate number and may be as low as 2 or perhaps as high as 4 in actual installations.

While the fluidized bed combustor is not burdened with the cycle efficiency degradation inherent in the power requirement and stack gas reheating requirement of tail-end flue gas scrubber, it should be recognized that there are some off-setting costs accompanying sulphur removal via fluidized bed combustion. These off-setting costs include the greater amount of limestone feed requirement expected to be required by the fluidized bed and the higher forced draft fan power also anticipated with fluidized bed combustion. Additionally, it is likely that the fluidized bed installation will be made up of a number of modules, requiring additional operating personnel as compared with the single unit installation typical of pulverized coal, cyclone combustor, or spreader stoker firing. An elementary comparison of the energy and consumables costs of fluidized beds versus tail-end scrubbers is presented in Table 4.5.2.1. It can be seen that many factors are involved in identifying the most economical system. Considering that both tail-end scrubbing and fluidized bed combustion are developing technologies, one must accept that their relative standing is subject to change.

4.5.3 Nitrogen Oxides Emissions

Concern with nitrogen oxides emissions is a relatively new event, having developed in the 1960's, when it became apparent that nitrogen oxides played a strong role in the generation of photochemical smog, especially in areas subject to temperature inversions, such as the Los Angeles County basin. The

TABLE 4.5.2.1

COMPARISON OF COMBUSTOR/HEX BOILER EFFICIENCIES

	PULVERIZED COAL	CONVENTIONAL FLUIDIZED BED
"BOILER EFFICIENCY" %	86.1	85.1
CA/S MOLAL RATIO	1.3	2.75
BURNER PRESSURE, INCHES H_2O	5.0	100
TOTAL AIR FLOW, SCFM @ 350 MWe	596,000	565,000
FAN POWER TO BURNERS, MWe	.5	8.2
COAL AND LIMESTONE PULVERIZING, MWe	2.0	0.5
TOTAL	2.5	8.7
MWe LOST = $(1-\eta)$ x ABOVE = .60 ABOVE =	1.5	5.2
EFFY RATIO DUE TO SOX SCRUBBING, (BASED ON 250 STACK)	.943	= 1
RELATIVE INPUT TO COAL	1.236	1.193
LBS LIMESTONE/LB COAL	.16	.34
FINAL COMPARISON, EQUIVALENT COAL INPUT ($ LIMESTONE = $ COAL/3)	1.29	1.31

groundrule permissible concentration of nitrogen oxides is specified in Sec. 3.2 as 0.7 lbs of nitrogen oxides (expressed NO_2) per million BTU of heat input to the combustor/heat exchanger. This is the present EPA limit for solid fossil fuel firing. Many existing pulverized coal installations experience difficulty in attaining this target, and to the best of our knowledge cyclone combustor installations find it impossible. NOX emissions evidently derive from two sources, some of the nitrogen in the combustion air is converted to nitrogen oxides in the combustion process, and some of the nitrogen present in the fuel, (i.e., fuel bound nitrogen), is also so converted. The conversion of air supplied nitrogen to nitrogen oxide is seemingly a function of the temperatures reached during combustion, high temperature favoring NOX formation, the time at temperature, and the excess air. In dry bottom pulverized coal furnaces, the NOX emission level has been found to be reasonably controllable through a variety of mechanisms including burner design and adjustment, the use of a large number of relatively small burners, the spreading out of the combustion to permit radiation to reduce flame temperature prior to completion of combustion, the recirculation of spent flue gases, either within the furnace or from the heat exchanger exit, to temper the combustion and reduce the temperature, and the like. For the purposes of this study, it has been assumed that the NOX emissions from dry bottom pulverized coal fired CCGT combustor/heat exchangers would be controllable by a combination of the above means and the only special constraint on the design for NOX control is the provision for recirculation of flue gases from the heat exchanger exit.

Slag tap furnaces, either pulverized coal or cyclone combustor fired, evidently entail more intensive combustion and/or more residence time for the flue gases at high combustion temperature and tend thereby to have excessive NOX emissions. No combustion or design fix for this problem is presently known. For purposes of this design study, it has been assumed that the developing technology involving the mixing of ammonia gas into the flue gases near the exit of the heat exchanger train will prove technically and economically feasible, and will be applicable to wet bottom type pulverized coal or cyclone combustors.

Most fluidized bed combustion concepts involve bed temperatures in the neighborhood of 1450 F to 2000 F. One would expect nitrogen oxides formation due to nitrogen fixation from the air to be minimal at these temperatures, and this does appear to be the case. However, while the nitrogen oxides emissions displayed by fluidized beds are generally well within the existing 0.7 lbs per million BTU limit, they are not as low as might be anticipated solely on the basis of bed operating temperature. It appears that in fluidized beds a larger proportion of the fuel bound nitrogen is converted to nitrogen oxides. As a consequence the nitrogen oxidies emissions may be on the order of 50% of the present allowable. This is the source of some apprehension that legal limits may be exceeded if the allowable values are reduced or bed operating conditions and coal compositions vary substantially from those already tested. For the purposes of this design study, however, it was assumed that nitrogen oxide emissions from coal fired CCGT combustor/heat exchangers would be below groundrule requirements, and no special design constraint in the fluidized bed would be necessary to attain that condition.

4.6 MATERIALS OF CONSTRUCTION

The application of materials of construction to the CCGT combustor/heat exchanger equipment may be expected to differ from conventional steam boiler combustor/heat exchanger equipment for several reasons. The working fluid differs from the water and steam working fluids used in steam boilers, some portions of the heat exchanger surface will operate at temperatures much higher than those existing in steam boilers, and some novel combustion systems are to be studied which may present a different environment on the flue gas side of the heat exchanger surface.

The materials picture with respect to the working fluid side of the heat exchanger surface is covered in the "Cycle Analysis and Working Fluids" report, and will not be repeated here. This section will cover materials with respect to their strength to withstand the rated internal pressures, for the desired heat exchanger life, at the temperature and pressure conditions existing in the heating surface walls, and under the corrosive and erosive environment existing on the flue gas exposed surfaces of the heat exchanger materials.

4.6.1 Metals

4.6.1.1 Metals Strength. The groundrules of the design study require that CCGT combustor/heat exchangers be investigated for the condition where an all-metal heat exchanger surface is supplied for working fluid temperatures up to a nominal 1550 F. This implies that in practice excursions to a maximum of 1650 F can probably be expected. The ASME Boiler and Pressure Vessel Code presently provides approved metals and their maximum allowable stress values only up to 1500 F, so it is necessary to evaluate other metals than those contained in the codes. As indicated in Fig. 4.6.1, the choice of materials typically considered for the 1500 F to 1700 F temperature range is limited. The metallic materials must, moreover, possess a combination of strength and corrosion resistance to the fireside gases and to the working fluid, so that alloy selection is rendered more difficult since optimum high-temperature strength and maximum high-temperature corrosion resistance are not always compatible. Because of the high temperatures and the novel environments in some of the advanced heat exchanger loops, specific corrosion rate information is insufficiently developed to allow materials selection with great confidence.

In the following sections, the available metallic materials with high-temperature strength, and the various corrosion zones in the advanced heat exchanger designs, are reviewed together with the usual practices for combatting corrosion. Current corrosion rate data is also included. The alloys available for design on the basis of these considerations reflect conventional boiler practice where possible, and where there is no proven practice, the most economic and practical choice,

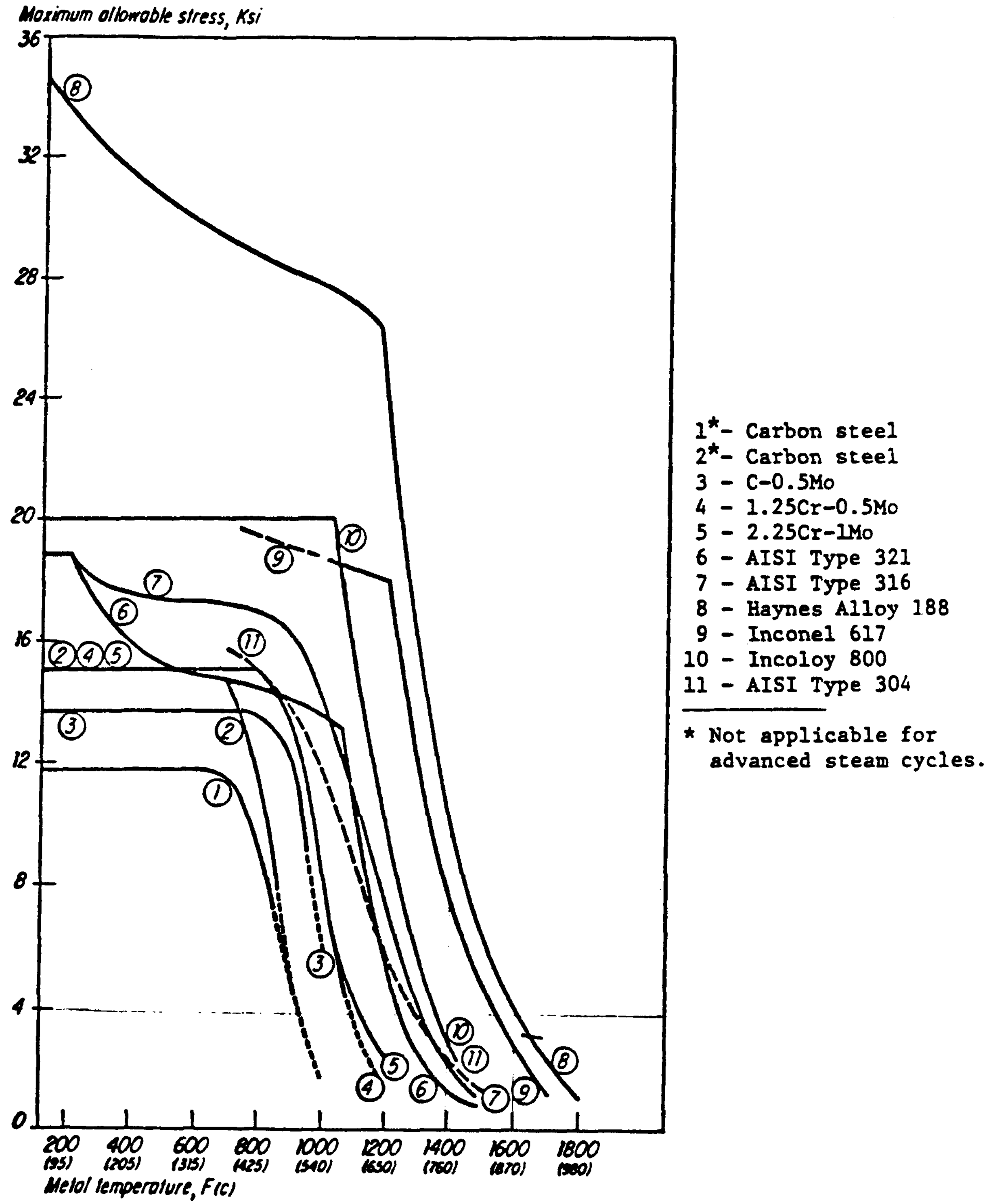


Figure 4.6.1 Estimated Relative Mechanical Strength for Some Candidate Heat Exchanger Materials Based on ASME Boiler Code (Criteria for Design Allowable Stresses) (General Electric Co.)

From: J. E. Mesko, Metal Progress, 112(2), 30 (1977).

based on available data, with a systems approach used in the most severe corrosive applications. By this systems approach, the alloy is chosen primarily for its mechanical properties, and the corrosion resistance is a secondary consideration which may be taken care of by cladding, for instance.

Alloys which are considered as candidates for high temperature applications because of their combination of high temperature strength and corrosion resistance are listed in Table 4.6.1. None of the refractory metals, i.e., molybdenum, tungsten, columbium and the like have been listed as candidate alloys. This follows from their frequently brittle nature at ambient and moderately high temperatures, the rapidity with which many of them oxidize at high temperatures, and their extraordinarily high cost. The four strongest candidates on Table 4.6.1 are:

- Inconel 617, a wrought, solid-solution-strengthened Ni-base alloy which has excellent creep and rupture strength to 1800 F, good tensile elongation after extended elevated temperature service, and is weldable.
- Haynes 188, a carbide- and solid-solution-strengthened cobalt-base alloy with similar apparent creep and rupture properties to Inconel 617, although data are available only for up to 10,000 hours, which is weldable and possesses excellent high-temperature oxidation resistance.
- HK-40, a carbide-dispersion-strengthened cast Fe-Cr-Ni alloy with somewhat lower high-temperature strength than the preceeding alloys, but for which a large body of data exists for extended service up to 1800 F, with some also available at higher temperatures.
- Hastelloy X, a wrought, solid-solution-strengthened nickel-base alloy, with excellent oxidation resistance to 2200 F and moderately high strength above 1450 F. It age-hardens somewhat after long exposure at 1200-1800 F, after which it loses some formability. It can be welded by most fusion welding procedures.

Table 4.6.1 Typical Properties of Alloys Chosen for High-Temperature Strength

Alloy	Strength: UTS(ksi)	Strength: Stress Rupture, ksi: 1800 F	Strength: Stress Rupture, ksi: 1600 F	Strength: Stress Rupture, ksi: 1450 F	Availability	Remarks	Relative Price	Refs.
Inconel 617	11 @ 2000 F	2.4; 10,000 hrs	6.2; 10,000 hrs	9; 100,000 hrs	Std. mill forms		9.1	2,3
Haynes 188	19 @ 2000 F	2.0; 10,000 hrs	6.4; 10,000 hrs	9.6;100,000 hrs	No tube, sheet plate, etc.		27.4	3,4
HK-40	6 @ 2000 F	1.9; 10,000 hrs	4.0; 10,000 hrs	—	Centrifugally cast tube	Min. O.D.3"; very well characterized	1.0	5
Hastelloy X	12 @ 2000 F	1.7; 10,000 hrs	5.0; 10,000 hrs	—	Std. mill forms		14.5-15.9	6,7,8
RA-333	18 @ 1800 F	1.2; 10,000 hrs	5.0; 10,000 hrs	—	Std. mill forms		9.8	9
Inconel 600	11 @ 1800 F	1.15;10,000 hrs (0.73;100,000 hrs)	1.9; 10,000 hrs	—	Std. mill forms		4.2	10
Incoloy 800	8.9 @ 1800 F	1.0; 10,000 hrs (0.62;100,000 hrs)	2.3; 10,000 hrs (1.5;100,000 hrs)	— (4.5;100,000 hrs)	Std. mill forms	Very well characterized	2.8	3,11
RA-330	10.7 @ 1800 F	0.7; 10,000 hrs	2.0; 10,000 hrs	—	Std. mill forms		4.2	12
L-605	19 @ 2000 F	4.6; 1,000 hrs	6 @ 1650 F 10,000 hrs	—	Sheet, plate, bar	Extremely well characterized	—	13,14,15
Inconel 625	10 @ 2000 F	—	5; 10,000 hrs	—	Std. mill forms		8.7	16,17,18
Hastelloy C	30 @ 1800 F	4.0; 1,000 hrs	6; 1,000 hrs (12 @ 1500 F, 1,000 hrs)	—	Std. mill forms		15.6-20.4	19,20
Stellite 31	29 @ 1800 F	10; 1,000 hrs	18; 1,000 hrs	—	Cast rod, castings		—	21,22,23
IN-718	90 @ 1400 F	—	30 @ 1200 F, 1,000 hrs	—	Bar, Sheet, strip	Extremely well characterized	—	24
Inconel 671	12 @ 1800 F	0.8; 1,000 hrs	1.8; 1,000 hrs	—	Std. mill products	Corrosion resistant; used for cladding	9.0	25

The creep-rupture properties of all four alloys have been studied over wide temperature ranges, typically from 100 F to 2000 F.* Results obtained at 1600 F and 1800 F are presented in Fig. 4.6.2 as applied stress versus time to rupture. Inconel 617 and Haynes 188 appear to have comparable creep rupture properties at these temperatures, as do Hastelloy X and HK-40, while for a given time to rupture HK-40 and Hastelloy X fail at about half the stress as compared with the former alloys.

It can be seen that creep-rupture data are available for time periods extending to more than 20,000 hours so that extrapolation to 100,000 hours, which is the basis for Code compatible design, can be made with some confidence. The creep-rupture strength of HK-40, which is superior to that of many high temperature alloys, has been ascribed to its coarse-grained structure strengthened with primary eutectic carbides, reinforced at lower temperatures by a fine dispersion of precipitated carbides.**

The temperature dependencies for the ultimate tensile strength and the 0.2 percent offset yield strength of HK-40, Hastelloy X, Haynes 188, and Inconel 617 are presented in Figs. 4.6.3 and 4.6.4, respectively. Over the entire temperature range of 1400 F to 2000 F, the Haynes 188 is seen to have the highest strength while the HK-40 alloy is the weakest of the group. Little difference is found between tests performed on cyclinders and on sheets of Inconel 617.

The estimated design stress values, according to the rules of the ASME Boiler and Pressure Vessel Code which is discussed in Section 4.7.1, are presented in Table 4.6.2. These are the values available for utilization in the design study.

4.6.1.2 Fireside Corrosion Considerations - Fluidized Bed Combustor. Corrosion data for materials exposed in fluidized bed combustors exhibit a fair amount of scatter and in some cases apparent contradictions, which obviously reflect

* See page 4-42.1
** See page 4-42.1

References

* "Inconel Alloy 617, Data for Use in Design of Gas-Cooled and Liquid-Metal Fast Breeder Reactors", D.J. Tillack, Compilation, Huntington Alloys, (1977).

"The Effect of Impurities in Coal-Derived Fuels on Service Life of Boiler Tubes for Advanced Power Cycle Applications", A.L. Plumbley and J.I. Accortt, Reports on ERDA Contract E(49-18)-2045, (1976).

"Haynes Alloy No. 188", Stellite Div., Cabot Corp. Brochure, F-30,361B, (1973).

"Short-Time Tensile and Long-Time Creep Rupture Properties of the HK-40 Alloy and Type 310 Stainless Steel at Temperatures to 2000°F", J.A. VanEcho, D.B. Roach, and A.M. Hall, Trans. ASME - J. Basic Eng., ___, 465 (1967).

"Hastelloy Alloy X", Alloy Digest, Alloy Filling Code N-14, Eng. Alloy Digest Inc., (1954).

"Hastelloy Alloy X", Stellite Div., Cabot Corp. data, F-30,037 (1961).

"Hastelloy X", C.E. Best and H.C. Popp, GE Technical Data reported in Aerospace Structural Materials Handbood, Code 4112.

** "Short-Time Tensile and Long-Time Creep Rupture Properties of the HK-40 Alloy and Type 310 Stainless Steel at Temperatures to 2000°F", J.A. VanEcho, D.B. Roach, and A.M. Hall, Trans. ASME - J Basic Eng., ___, 465 (1967).

Table 4.6.2. Metal Design Stress Values

MAT'L TEMP	INCO 617	INCO 800H SB-163 Case 1325-9	347H SA-213 TP 347H	316H SA-213 TP 316H	304H SA-213 TP 304H	$2\frac{1}{4}$Cr-1Mo SA-213-T22	$\frac{1}{2}$Cr-$\frac{1}{2}$Mo SA-213-T2	C. Steel SA-210-A1
1600°F	2,700							
1500°F	4,200	2,500	1,300	1,300	1,400			
1400°F	8,500	3,600	2,500	2,300	2,300			
1300°F	12,500	5,400	4,400	4,100	3,700			
1200°F	12,500	8,400	7,900	7,400	6,100			
1150°F		11,200	10,500	9,800	7,700			
1100°F		13,500	13,000	12,400	9,800			
1050°F				14,500	12,200	5,800		
1000°F				15,300	13,800	7,800		
900°F				15,600	14,700	13,100	12,500	800°F LIMIT
800°F				15,900	15,200	15,000	14,400	10,800
700°F				16,300	15,900	15,000	15,000	14,400

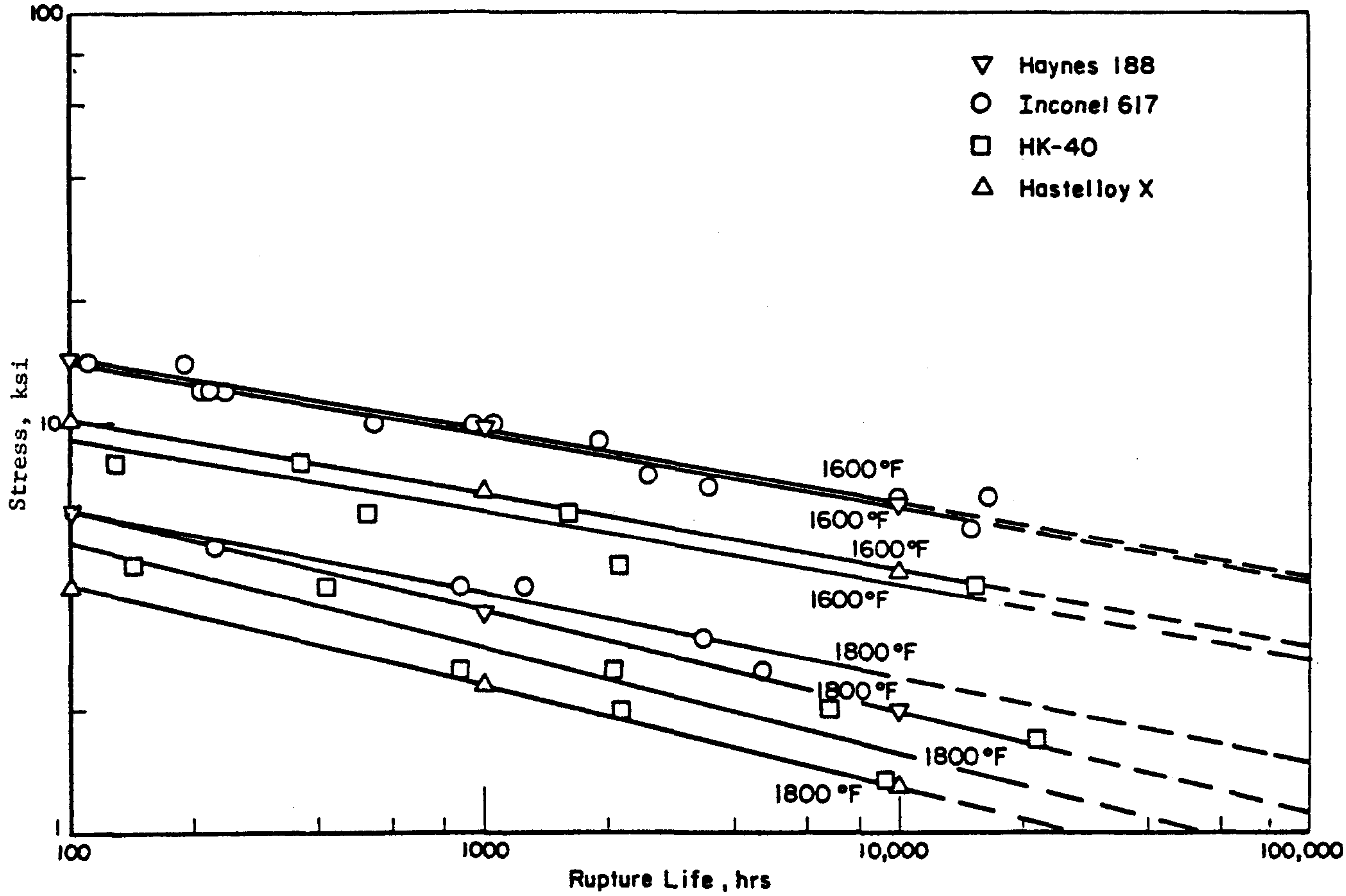


Figure 4.6.2 Creep Rupture Data for Four Strongest Alloys

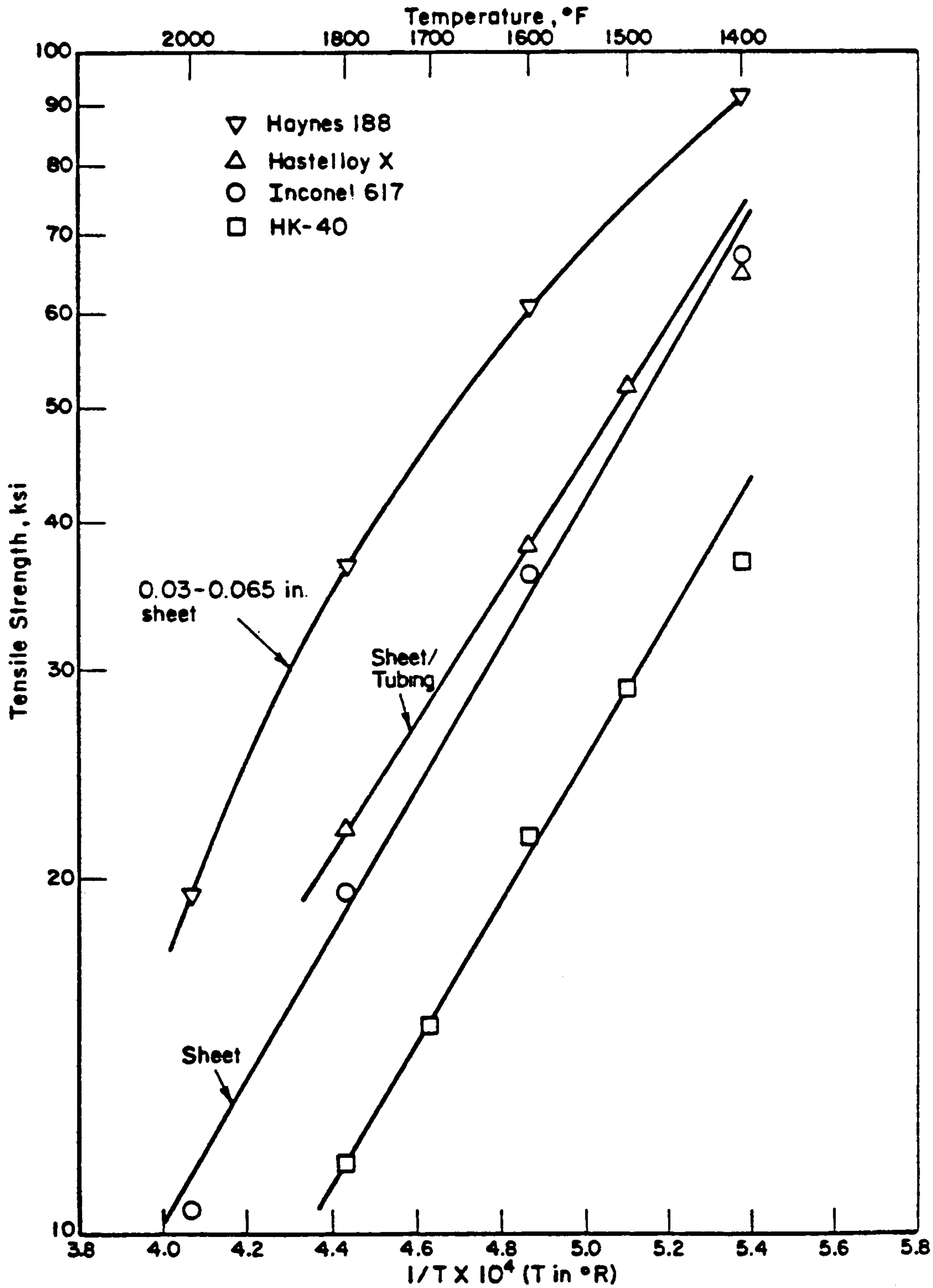


Figure 4.6.3 Ultimate Tensile Strengths of Four Strongest Materials

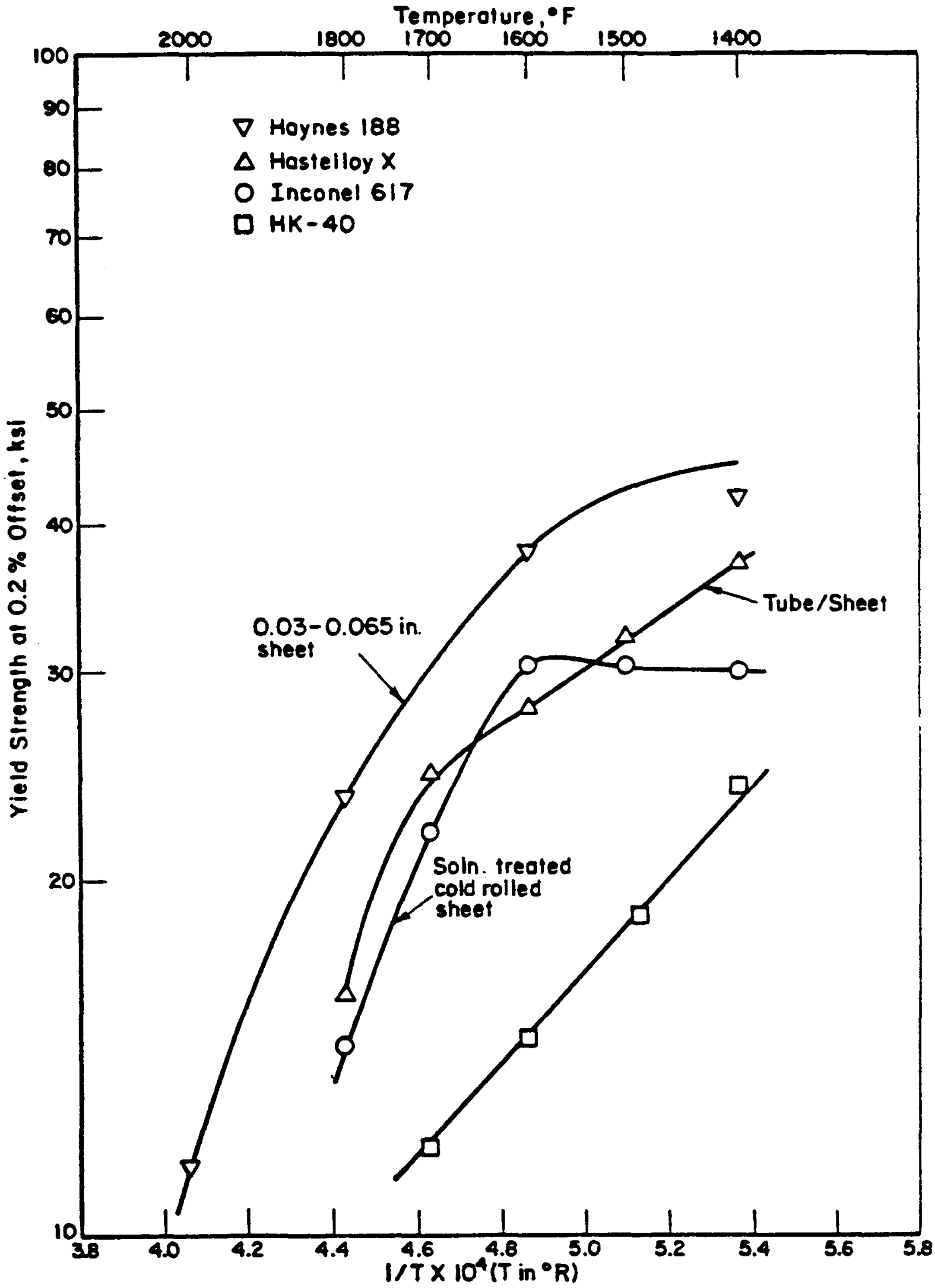


Figure 4.6.4 Yield Strengths of Four Strongest Materials

differences in coal and sulphur sorbent feed, combustion conditions, and length of the various tests. Maximum exposure times have been limited to 2000 hours. While the corrosive environment in an FBC seems at first to be similar to that in a pulverized coal-fired boiler, there are significant differences which are now becoming understood. An important feature of the fluidized bed is the low oxygen partial pressure condition created, and the relatively slow combustion of coal in the bed. At any given position in the fluidized bed, the partial pressure of oxygen apparently oscillates between a value consistent with the expected stoichiometry of the combustion conditions, and remarkably low values* often in syncronization with the pulsed coal feed.

The sulphur from the burning coal is released into the bed before reaction with the sorbent, and it has been suggested* that the effective oxygen and sulphur partial pressures in the bed (in all regions except the bubbles or pulses of burning coal, coal representing approximately 1 percent of a typical bed) are fixed by the equilibrium between CaO and $CaSO_4$. At the usual bed operating temperatures, the conditions set by this equilibrium represent an environment that is sulfidizing to many common alloys, and there is evidence that accelerated in-bed corrosion is often associated with the deposition of a layer of $CaSO_4$ on the alloy surface. In addition, low oxygen-sulfidizing conditions may prevail near the coal feed ports, depending on the mixing characteristics of the bed, as the coal starts to burn.

Corrosion in the freeboard has been investigated mainly with a view to the application of a gas turbine to expand the off-gas, so that most of the data refer to turbine-type alloys. The deposits which form on materials exposed in the freeboard were found to be generally loosely adherent, not fused, and problems of localized accelerated corrosion have not been encountered. Rates of corrosion of freeboard metals have been found to be greater than in air-oxidation at similar temperatures, and this was attributed to erosion by

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* Reference : "Oxygen Measurements in Flue Gases with a Solid Electrolyte Probe", M.J. Cooke, A.J.B. Cutler, and E. Raask, J. Inst. Fuel, 45, 153 (1972)

the fly ash. Signs of incipient sulfidation-assisted corrosion have, however, been found in some gas turbine alloys after the longer exposure.

Corrosion data from all the FBC projects known to have materials testing programs have been collected and analyzed in some detail. Actual corrosion rates, or measured losses due to corrosion exposures for fixed times are listed by in Tables 4.6.3 and 4.6.4. The corrosion rates are presented both in the original units, and when converted to the common basis of μ/hr. Figures 4.6.5 and 4.6.6 are maps of these results, with lines drawn to connect points for the same alloy, to aid legibility, rather than to suggest trends of corrosion rate with temperature. However, where data for several temperatures were obtained from a single source, the line joining these may represent a trend (see, for example, the results for AISI 316,347). For a specific alloy, reference to Appendix will indicate the source of the individual data points, and the essentials of the bed conditions used. A suggested maximum corrosion rate of 0.038 μm/hr, or 13 mil/yr, is indicated on the map in Figs. 4.6.5 and 4.6.6. This value is used as a general rule of thumb by the British National Coal Board for corrosion of utility boiler and superheater tubes.

The in-bed corrosion data suggest that medium carbon steel and 2-1/4 Cr-1 Mo are acceptable up to about 850 F, which is similar to the application range of medium carbon steel in normal boiler practice, but somewhat lower for the 2-1/4 Cr-1 Mo alloy. Alloys for which the corrosion results appear reasonably consistent are the austenitic stainless steels, AISI 316 and 347, and Esshete 1250, and even here there is one data point for 347 (Nb) which is completely out of line with the others. Similarly, the data for AISI 321 (Ti) do not fit the pattern. The two strongest high temperature alloy candidates, Inconel 617 and Haynes 188, do not appear to possess satisfactory corrosion resistance in this environment. Inconel 617 exhibited excessive surface scaling and pitting (for a total surface recession of 340 to 380 x 10^{-3} μ/hr at 1400 to 1650 F), while Haynes 188 underwent increasing amounts of pitting corrosion with

Table 4.6.3. SUMMARY OF REPORTED CORROSION DATA IN FLUIDIZED-BED COMBUSTORS
IN-BED CORROSION

Alloy	T°F	t,hrs	% S in coal	Sorbent	Reported Corrosion Rate	Rate Converted to μ/hr	Ref.*
Medium C-steel	752	1000	2.8	Limestone	20 mg/cm^2	0.025	50
	932	1000	2.8	"	40 mg/cm^2	0.0506	50
Corten-B	1004	1000	3.0	Limestone	S 0.83 mm P 0.025 mm	0.855	51
	1202	1000	3.0	"	1.10 mm max	1.1	51
1Cr-1/2Mo	390-700	100	3.5	Limestone	100 $\mu g/cm^2$ hr	0.127	52
2-1/4Cr-1Mo	752	1000	2.8	Limestone	20 mg/cm^2	0.025	50
	932	1000	2.8	"	40 mg/cm^2	0.050	50
	1112	1000	2.8	"	270 mg/cm^2	0.342	50
	1004	1000	3.0	Limestone	S 0.6 mm P 0.03 mm	0.630	51
	1202	1000	3.0	"	S 1.1 mm P 0.035 mm	1.135	51
	700-1000	100	3.5	Limestone	260 $\mu g/cm^2$ hr	0.329	52
	500-1250	100	3.5	"	400 $\mu g/cm^2$ hr	0.507	52
9Cr-1Mo	1004	1000	3.0	Limestone	S 0.09 mm P 0.05 mm	0.14	51
	1202	1000	3.0	"	S 0.13 mm P 0.06 mm	0.06	51
	1202	2000	3.0	"	S 0.19 mm P 0.05 mm	0.095	51
AISI 304	1417	1500	4.1	Limestone	S 42μ P 76μ	0.079	54
	1550-1610	500	3.0	"	S 5μ P 0	0.01	53
AISI 310	1417	1500	4.1	Limestone	S 34μ P 38μ	0.048	53
	1620	1500	4.1	"	S 23μ	0.015	53
	1400	1000	3.0	Limestone	S 0.02 mm P 0.11 mm	0.13	51
	1544	1000	3.0	"	P 0.04 mm	0.04	51
	1550-1610	500	3.0		S 2.5 P 0	0.005	54

Table 4.6.3 (continued)

Alloy	T°F	t, hrs	% S in coal	Sorbent	Reported Corrosion Rate	Rate Converted to μ/hr	Ref.*
AISI 316	752	1000	2.8	Limestone	1 mg/cm^2	0.0013	50
	932	1000	2.8	"	1 mg/cm^2	0.0013	50
	1112	1000	2.8	"	2 mg/cm^2	0.0025	50
	1292	1000	2.8	"	7 mg/cm^2	0.0089	50
	1562	1000	2.8	"	13 mg/cm^2	0.0165	50
AISI 321	1150-1380	100	3.5	Limestone	70 μg/cm^2 hr	0.0887	52
	1430-1530	100	3.5	"	40 μg/cm^2 hr	0.0507	52
AISI 329	1000	1000	3.0	Limestone	S 0.002 mm P 0	0.002	51
	1202	1000	3.0	"	P 0.04 mm	0.040	51
AISI 347	752	1000	2.8	Limestone	1 mg/cm^2	0.0013	50
	932	1000	2.8	"	1 mg/cm^2	0.0013	50
	1112	1000	2.8	"	1 mg/cm^2	0.0013	50
	1292	1000	2.8	"	5 mg/cm^2	0.00635	50
	1400	1000	3.0	Limestone	S 0.06 mm P 0.07 mm	0.130	51
	1620	1500	4.1	"	S 56μ	0.037	54
AISI 405	1004	1000	3.0	Limestone	S 0.1 mm P 0.05 mm	0.15	51
AISI 410	752	1000	2.8	Limestone	1 mg/cm^2	0.0013	50
	932	1000	2.8	"	2 mg/cm^2	0.0025	50
	1112	1000	2.8	"	2 mg/cm^2	0.0025	50
	1292	1000	2.8	"	4 mg/cm^2	0.0051	50
	1562	1000	2.8	"	320 mg/cm^2	0.405	50
	970-1220	100	3.5	Limestone	160 μg/cm^2 hr	0.203	52
	1240-1430	100	3.5	"	170 μg/cm^2 hr	0.215	52
Ebrite 26-1 (AISI 446)	1004	1000	3.0	Limestone	S 0.018 mm P 0.008 mm	0.026	51
	1202	1000	3.0	"	P 0.08 mm	0.080	51
Nitronic 50	1004	1000	3.0	Limestone	S 0.02 mm P 0.02 mm	0.040	51
	1202	1000	3.0	"	P 0.024	0.024	51
	1202	2000	3.0	"	S 0.02 mm P 0.03 mm	0.025	51

*See Page 4-48g

TABLE 4.6.3 (continued)

Alloy	T°F	t, hrs	% S in coal	Sorbent	Reported Corrosion Rate	Rate converted to μ/hr	Ref.*
Esshete 1250	752	1000	2.8	Limestone	1 mg/cm^2	0.0013	50
	932	1000	2.8	"	1 mg/cm^2	0.0013	50
	1112	1000	2.8	"	3 mg/cm^2	0.0038	50
	1292	1000	2.8	"	9 mg/cm^2	0.0114	50
	1562	1000	2.8	"	14 mg/cm^2	0.0177	50
21-6-9	1004	1000	3.0	Limestone	{S 0.02 mm P 0.03 mm	0.050	51
	1202	1000	3.0	"	{S 0.06 mm P 0.06 mm	0.120	51
Incoloy 800	1202	1000	3.0	Limestone	{S 0.05 mm P 0.108 mm	0.158	51
	1202	2000	3.0	"	{S 0.14 mm P 0.17 mm	0.230	51
	1400	1000	3.0	"	{S 0.08 mm P 0.19 mm	0.440	51
	1550	1000	3.0	"	{S 0.006 mm P 0.17 mm	0.176	51
	1380-1520	100	3.5	Limestone	45$\mu g/cm^2$ hr	0.057	52
	1550-1610	500	3.0	"	S 3.5μ P 3μ	0.013	53
Inconel 600	1550-1610	500	3.0	Limestone	6.5μ	0.013	53
Inconel 601	1400	1000	3.0	Limestone	{S 0.2 mm P 0.35 mm	0.55	51
	1544	1000	3.0	"	{S 0.05 mm P 0.2 mm	0.25	51
Inconel 617	1400	1000	3.0	Limestone	{S 0.14 mm P 0.2 mm	0.34	51
	1544	1000	3.0	"	{S 0.16 mm P 0.2 mm	0.36	51
	1650	1000	3.0	"	{P 0.38 mm, or S 0.112 mm	0.38	51

*See Page 4-48g

TABLE 4.6.3 (concluded)

Alloy	T°F	t, hrs	% S in coal	Sorbent	Reported Corrosion Rate	Rate Converted to μ/hr	Ref.*
Inconel 671	1340	1500	4.1	Limestone	S 299μ P 572μ	0.531	54
	1417	1500	4.1	"	S 80μ P 114μ	0.129	54
	1482	1500	4.1	"	S 107μ P 114μ	0.147	54
	1544	1000	3.0	Limestone	750+μ	0.750	51
	1650	1000	3.0	"	830+μ	0.830	51
Hastelloy X	1400	1000	3.0	Limestone	S 0.04 mm P 0.08 mm	0.120	51
	1544	1000	3.0	"	S 0.015 mm P 0.096 mm	0.111	51
Haynes 188	1202	1000	3.0	Limestone	S 0.023 mm P 0.03 mm	0.053	51
	1202	2000	3.0	"	S 0.10 mm P 0.05 mm	0.075	51
	1400	1000	3.0	"	S 0.09 mm P 0.09 mm	0.180	51
	1544	1000	3.0	"	S 0.05 mm P 0.134 mm	0.184	51
	1650	2000	3.0	"	P 0.38 mm	0.019	51
FSX 414	1340	1500	4.1	Limestone	S 46μ P 57μ	0.069	54
USS 18-18-2	1340	1500	4.1	Limestone	P 118μ	0.078	54
	1417	1500	4.1	"	P 57μ	0.038	54

*See Page 4-48g

TABLE 4.6.4. SUMMARY OF REPORTED CORROSION DATA IN FLUIDIZED BED COMBUSTORS
FREEBOARD CORROSION

Alloy	T°F	t, hrs	% S in coal	Sorbent	Reported Corrosion Rate	Rate Converted to μ/hr	Ref.*
2-1/4Cr-1Mo	1202	1000	3.0	Limestone	1061μ(oxdtn)	1.061	51
AISI 304	1594	1003	3.5	Dolomite + Burgess clay	17μ	0.017	55
AISI 309	1594	1003	3.5	Dolomite + Burgess clay	24μ	0.024	55
AISI 310	1594	1003	3.5	"	27μ	0.027	55
AISI 316	1594	1003	3.5	"	18μ	0.018	55
AISI 446	1594	1003	3.5	"	3μ	0.003	55
RA 330	1594	1003	3.5	"	12μ	0.012	55
RA 333	1594	1003	3.5	"	30μ	0.030	55
	1650	1000	3.0	Limestone	103μ(oxdtn)	0.103	51
Incoloy 800	1594	1003	3.5	Dolomite + Burgess clay	47μ	0.047	55
	1670	568	3.5	Dolomite + Burgess clay + MgO	50μ(80μmax)	0.141	55
Incoloy 825	1594	1003	3.5	Dolomite + Burgess clay	35μ	0.035	55
	1670	568	3.5	Dolomite + Burgess clay + MgO	24μ(55μmax)	0.097	55
Inconel 600	1594	1003	3.5	Dolomite + Burgess clay	70μ	0.070	55
	1670	568	3.5	Dolomite + Burgess clay + MgO	15-20μ	0.035	55
Inconel 601	1594	1003	3.5	Dolomite + Burgess clay	65μ	0.065	55
	1670	568	3.5	D+clay+MgO	20μ(30 max)	0.053	55

*See Page 4-48g

Alloy	T°F	t, hrs	% S in coal	Sorbent	Reported Corrosion Rate	Rate Converted to μ/hr	Ref.*
Inconel 671	1650	1000	3.0	Limestone	102μ (oxdtn)	0.102	51
Inconel 690	1594	1003	3.5	Dolomite + Burgess clay	50μ	0.050	55
	1670	568	3.5	D+clay+MgO	17μ (20 max)	0.035	55
Inconel 706	1594	1003	3.5	Dolomite + Burgess clay	18μ	0.018	55
Nimonic PE16	1292	1000	2.8	Limestone	3 mg/cm^2	0.0038	50
	1562	1000	2.8	"	15 mg/cm^2	0.019	50
IN 738	1594	1003	3.5	Dolomite + Burgess clay	45μ	0.045	55
	1670	568	3.5	D+clay+MgO	25μ (40 max)	0.070	55
U-700	1594	1003	3.5	Dolomite + Burgess clay	30μ	0.030	55
IN 713C	1594	1003	3.5	"	18μ	0.018	55
	1670	568	3.5	D+clay+MgO	13μ (25 max)	0.044	55
Rene 77	1594	1003	3.5	Dolomite + Burgess clay	28μ	0.028	55
Nimonic 80A	1594	1003	3.5	"	70μ	0.070	55
	1670	568	3.5	D+clay+MgO	50μ (80 max)	0.141	55
N 155	1590	1003	3.5	Dolomite + Burgess clay	20μ	0.020	55
Hastelloy X	1650	1000	3.0	Limestone	P81μ	0.081	51
HS 31	1594	1003	3.5	Dolomite + Burgess clay	30μ	0.030	55
FSX 414	1594	1003	3.5	"	60μ	0.060	55
	1670	568	3.5	D+clay+MgO	50μ (60 max)	1.056	55
Haynes 188	1650	2000	3.0	Limestone	S62μ	0.031	51
MAR M 509	1594	1003	3.5	Dolomite + Burgess clay	22μ	0.022	55

S: Surface attack
P: Penetration attack
*See Page 4-48g

REFERENCES

(50) Fluidized Combustion Control Group, NCB, "Reduction of Atmospheric Pollution", Final Report of National Coal Board to EPA, Ref. No. DHB 060971 (Sept 1971).

(51) E.A. Rogers, A.J. Page and R.D. LaNauze, Work in progress on EPRI Project RP 388-1, "The Investigation of Corrosion Performance of Boiler and Gas Turbine Alloys in Fluidized Combustion Systems", and "The Corrosion Performance of Heat Exchanger Alloys in Fluidized Combustion Systems", presented at Sixth European Corrosion Congress, London, Sept. 1977.

(52) NRDC, "Pressurized Fluidized-Bed Combustion", R&D Reprt. No. 85, Interim No. 1 for OCR on Contract NO. 14-32-0001-1511 (1974).

(53) R.H. Cooper, "Preliminary Report on Corrosion Analysis of Heat Exchanger Tubes from a Fluidized-Bed Coal Combustor", presented at Int. Conf. on Coal Ash Deposits, Henniker, New Hampshire, July 1977. NBS IN 480.

(54) A.M. Hall, et al., "Evaluation of Heat Exchanger Materials for Use in Coal-Fired Fluidized Bed Combustion Environment", Final Report on ERDA Contract No. E949-18)-2325, Rept. No. FE-2325-19, Dec. 1977.

(55) K.E. Phillips, "Energy Conversion from Coal Utilizing CPU-400 Technology", Annual Report for July 1975 - June 1976 to ERDA on Contract No. EX-76-C-01-1536 (Nov. 1976).

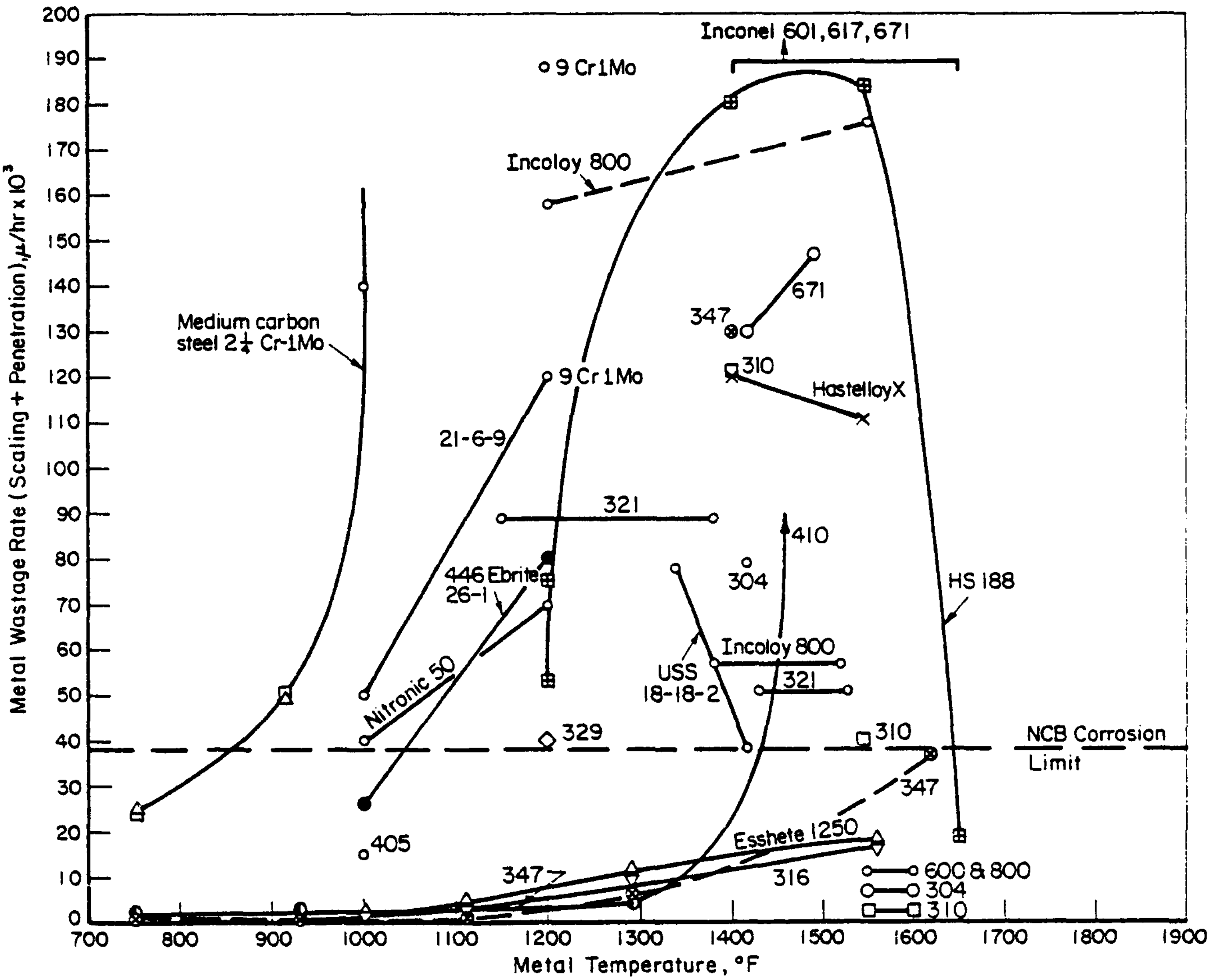

Figure 4.6.5 Map of Known Corrosion Results for Exposure in the Bed of a FBC

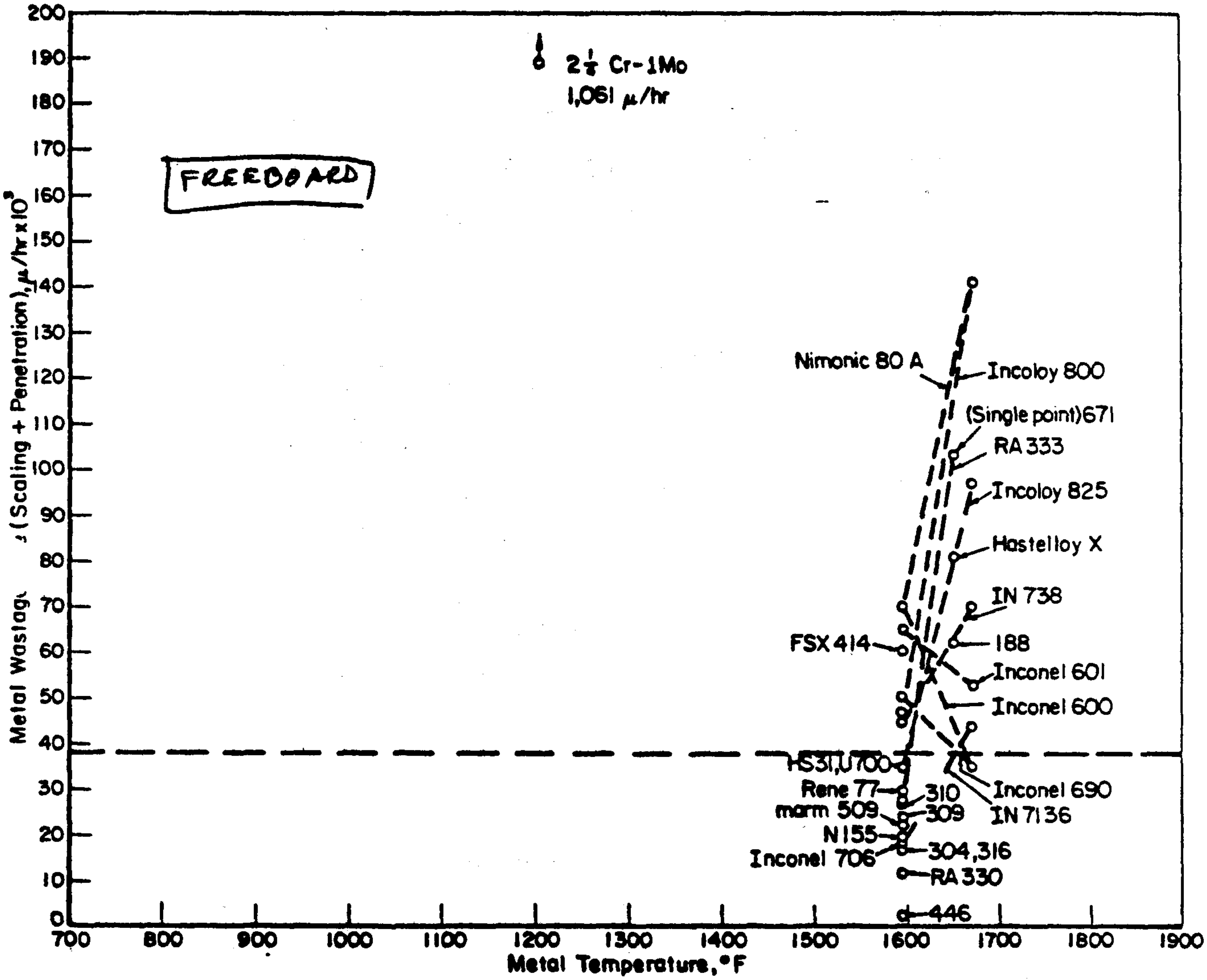

Figure 4.6.6 Map of Known Corrosion Results for Exposure in the Freeboard of a FBC

increasing temperature until 1650 F, where the alloy showed quite protective behavior. The validity of this apparent temperature dependence of Haynes 188 (all data from one source) has not yet been corroborated.

In the absence of any longer term comparative corrosion data or of any data from an operating unit, a reasonable choice of alloy for corrosion resistance up to ∿1600 F appears to be an austenitic stainless steel such as AISI 316 or 347, with 347 being the safer choice because of similar data from two separate sources.

The corrosion data for the freeboard in Fig. 4.6.6 indicates the preoccupation with turbine applications. At ∿1600 F, AISI 445 and the austenitic stainless steels appear perfectly satisfactory. In the absence of corrosion rate data in the lower temperature regions, the logical procedure is to follow the normal materials selection procedure used by the electric utilities and follow the Boiler Code recommendations as far as possible.

The small size of current FBC's used for corrosion testing has meant that some corrosion coupons have inevitably been situated in locations (such as close to the coal feed port) which would be avoided by heat transfer surfaces in a full-sized plant, and as a consequence, higher corrosion rates have been observed than would perhaps occur in practice. Additionally, since the local environment in the bed depends on the mixing characteristics of the bed, the corrosion rates are also somewhat dependent on the physical configuration of the bed. Thus some of the variability in corrosion rates reported so far for a given alloy in different beds can probably be related to these two agencies: location in the bed and bed configuration, as well as to the sulphur content, etc., of the coal used.

4.6.1.3 Fireside Corrosion Considerations Pulverized Coal-Fired Combustor/ Heat Exchanger. The corrosive conditions existing in the various forms of present day pulverized coal-fired boilers are understood sufficiently well that severe and recurrent fireside corrosion problems are relatively uncommon. This state of affairs is the result of intensive laboratory and field research efforts over a period of 20-30 years, which culminated in the late 1950's and early 1960's in a halt to the trend of rising steam temperatures, and a general acceptance of 1050 F or 1000 F steam.

The major corrosion problem is related to the formation of complex deposits on the superheater tubes, (principally of convection superheaters), in which the formation of alkali iron trisulfates, $(K, Na)_3 Fe(SO_4)_3$, has been associated with observations of accelerated attack. These salts have relatively low melting points, in the range 1030-1150 F. The exact mechanisms of deposit-related fireside corrosion are the source of active discussion, and the detailed arguments are too involved and voluminous to be succinctly summarized here, so that reference is made to the several compilations available in which the state of knowledge is detailed.* Reid considers the basic steps in the reaction chain to be:

a. An oxide film forms on the metal surface

b. Alkali sulfates, originating from the alkalies in the fuel ash and the sulphur oxides in the atmosphere, are deposited on this oxide surface. As an example, this deposit could be K_2SO_4.

c. The outer surface of the sulfate becomes sticky because of the increasing temperature gradient, capturing fly ash. There is consequently a further increase in the temperature of the fly ash, until eventually SO_3 is released by the thermal decomposition of sulphur compounds in the ash, and this SO_3 migrates toward the colder metal surface. A layer of slag forms on the outer surface.

* Reference: "The Mechanism of Corrosion by Fuel Impurities", H.R. Johnson and D.J. Littler, eds., Proc. of the CEGB Marchwood Conference, Butterwoths, London (1963.

"External Corrosion and Deposits, Boilers and Gas Turbines", W.T. Reid, American Elsevier Publishing Co., Inc. N.Y. (1971).

d. As the ash accumulates and reaches an equilibrium thickness, the temperature of the sulfate layer falls and the sulfate layer reacts with the oxide scale and the SO_3 to form an alkali iron trisulfate. With this removal of the oxide scale, the metal oxidizes further.

e. Deslagging occurs because of normal temperature excursions in the furnase exposing the alkali iron trisulfate to temperatures high enough to dissociate it, in part, releasing some SO_3, which again moves toward the cooler parts of the deposit, repeating the sequence. Further oxidation of the metal occurs to provide the normal equilibrium thickness of scale.

At the temperatures of interest, at least 250 ppm SO_3 is required for the trisulfates to be stable, whereas the actual SO_3 content of the boiler atmosphere is seldom more than 50 ppm.* However, catalytic oxidation of SO_2 beneath a layer of deposit has been found to provide essentially equilibrium concentrations of SO_3, which can locally exceed 2000 ppm.

The temperature dependence of the deposit-related corrosion is in the form of a "bell-shaped" curve, illustrated in Fig. 4.6.7. Little attack is observed at metal temperatures below 950 F or at above 1350 F, while a maximum rate is observed at 1250 F. These temperatures can vary by as much as ±50 F, depending on deposit composition, operating conditions (ambient SO_3) and, to some extent, alloy type, but there is general agreement about the shape of the temperature dependence. The corrosion rates of alloys ranging from T-22 (2-1/4 Cr-1 Mo) to AISI 321 (18 Cr-10 Ni-Ti) have exhibited this type of behavior, although alloys capable of forming Cr_2O_3-rich scales in this temperature range (high-Cr alloys) should be less susceptible to this type of attack because alkali chromium trisulfates have not been found in the deposits.

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* Ref. "High-Temperature Corrosion in Fluidized Bed Combustors", J. Stringer and S. Ehrlich, ASME 76-WA/CD-4 (1976).

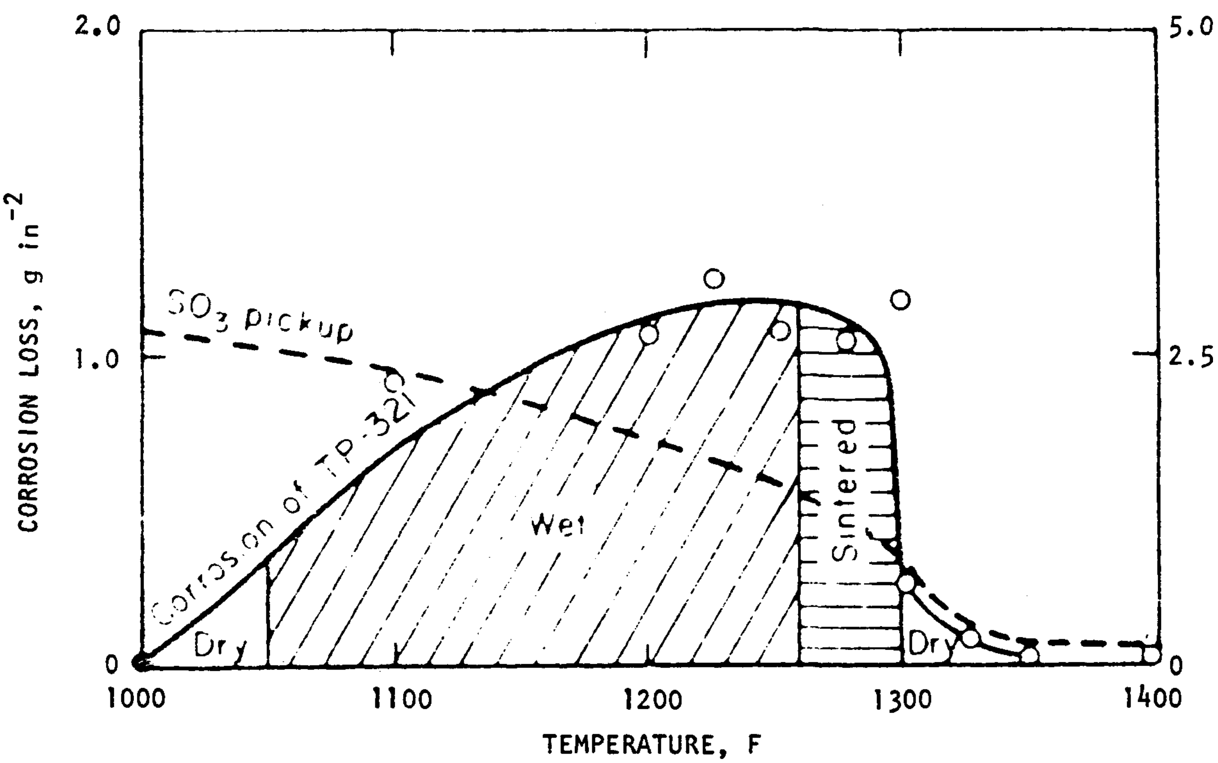

Figure 4.6.7 Corrosion of Chrome-Nickel Alloy (TP-321)
By Molten Trisulfates

The measures adopted to overcome these superheater fireside corrosion problems have included:

1. Reduction of the steam temperature to $\sim$1000 F, so moving the maximum metal temperatures down to 1100 F or lower, away from the peak of the "bell-shaped" curve.

2. Use of shields on the lead tubes of the superheater and reheater bundles. Since the shields are not cooled by the steam flow, the metal temperature can rise to 1350 F or higher, which is above the upper limit of the "bell-shaped" curve. The shield material must be more oxidation-resistant than the tubes, but results have been encouraging.

3. Careful control of the coal feed stocks to ensure that the intake of alkali metals is balanced by a sufficient intake of alkaline earth metals which can inhibit the formation of alkali iron trisulfates. This is a very promising method of corrosion control, and a methodology has been developed for evaluating coals in terms of their corrosion potential and for determining the levels of inhibitors to be introduced by feedstock blending or as additives.*

Alloy selection for steam tubes in conventional boilers is dominated by economic considerations so that, for instance, a 2-1/4 Cr-1 Mo superheater offering a life of 50,000 hours is sometimes preferred to an austenitic stainless steel superheater which could give a 100,000 hour life. Thus for boiler tubes (450-650 F metal temperature) carbon steel tubes are used almost without exception. Superheater and reheater tubes (750 - 1080 F metal temperature) are mainly 2-1/4 Cr-1 Mo, while the last few rows of the superheater and the reheater are often austenitic stainless steels, particularly AISI 304 and 321, to combat corrosion at the higher metal temperatures (up to 1150 F).

Ref: "The Control of High-Temperature Fire-Side Corrosion in Utility Coal-Fired Boilers", R. W. Borio, et al., OCR R&D Rept. No. 41, April 1969. PB 183716

Recent information indicates that the high-Cr alloy Inconel 671 (Ni-48Cr) can provide good corrosion resistance to the superheater coal ash deposit corrosion. Composite superheater tubes of Inconel 671 clad on to Incoloy 800H (INCOCLAD 671/800H) operating at a nominal temperature of 1150 to 1175 F in a location where problems were previously experienced with ash corrosion have been in service in Muskingum (Ohio Power Co.) power station for about 9 years with minimal corrosion.* The usual fuel in this station is Ohio coal (5% S). Similar composite tubes are still in service at Electric Energy, Inc., Joppa, Illinois, after 9 and 11 years, respectively.

There has been little incentive in recent years to investigate tube materials for higher service temperatures than required for 1050 F steam. Results available from an on-going DOE-sponsored research program to determine the service life of boiler tubes for advanced power cycle applications** indicate satisfactory corrosion behavior for austenitic stainless steels 316, 320 12R72 (15 Cr-15 Ni-1.2 Mo-Ti,B), Inconels 617 and 671, Incoloys 800 H and 802, and Haynes 188 in the temperature range 1200 to 1700 F. These data, essentially based on the results of pilot-scale laboratories tests of 300 hr duration, are apparently confirmed by 4,000 hours probe exposures in the convection sections of operating boilers.*** The coals burned in these tests, Appalachian bituminous (1.4% S); Rocky Mountain Sub-bituminous blend (0.4% S); Illinois bituminous - high volatile B (3.6%) and Texas lignite (0.7% S) gave calculated corrosion potentials ranked as low, very low, moderate and very low, respectively. The corrosion results from the 300 hour tests are summarized in Table 4.6.5.

The absence of any accelerated corrosion in the temperature range below 1300 F, expected from the usual "bell-shape" curve considerations, was attributed to the low corrosive indices of the coals used, and to judicious blending of the coals and additives by the power station in which the longer term tests were conducted.

Refs. * "Controlling Fuel-Ash Corrosion with Nickel Alloys", D.W. Rahio, Met. Eng. Quarterly, ______(1974).

Private communication with T.F. Lemke, Huntington Alloys (1978).

** "The Effect of Impurities in Coal-Derived Fuels on Service Life of Boiler Tubes for Advanced Power Cycle Applications", A.L. Plumbley and J.I. Accortt, Reports on ERDA Contract E(49-18)-2045, (1976).

*** Private communication with A.L. Plumley, Combustion Engineering Co. (1978).

Table 4.6.5 Average Corrosion Data from High-Temperature PC-Fired Corrosion Tests

Alloy	Average Temperature (°F)	Average Weight Loss Data* ($mg/cm^2/hr x 10^3$) Appalachian Bituminous (1.45 %S)	Rocky Mountain Subbituminous (0.45 %S)	Illinois Bituminous (3.65 %S)	Texas Lignite (0.75 %S)
AISI 316	1350	9.9	18.7	12.3	6.7
	1650	--	--	--	--
AISI 310	1350	44.3**	24.0	9.8	10.2
	1650	12.2	15.7	18.3	12.7
12R72	1350	20.2	19.0	26.7	12.6
	1650	--	--	--	--
Incoloy 802	1350	5.6	5.2	6.8	6.4
	1650	9.0	17.3	52.3**	4.1
Inconel 617	1350	4.3	4.8	0.7	4.7
	1650	2.2	40.0**	14.0	--
Inconel 671	1350	7.8	89.0**	10.0	4.3
	1650	26.7	15.7	26.0	6.5
Haynes 188	1350	--	--	--	--
	1650	6.3	4.0	35.0*	4.1

* 13 mil/yr $\cong$ 30 x 10^{-3} mg/cm^2 hr

** Mechanical damage.

Corrosion of wall tubes in PC furnaces has caused occasional problems* even though in most cases the wall tubes are run at relatively low temperatures. Essentially, the serious corrosion problems generally occur under layers of slag and the corrosion pattern is often related to the flame configuration along the furnace walls. Tube metal temperatures are usually found to be little different from adjacent areas where no corrosion occurred. In one well documented case, deposits found beneath the slag layer in the areas where corrosion had occurred were essentially mixtures of alkali sulfates. Even though the tube temperature never exceeded 750-850 F, these deposits were tightly adherent to the tubes and glassy in appearance. Yet, the melting point of the deposit was ~1000 F.

A significant portion of the metal heat exchanger surface of a PC fired CCGT combustor/heat exchanger will operate within the 950-1350 F temperature region, and thus be exposed to the possibility of corrosive attack as discussed above. The design constraints on metals selection for this service appear to include:

a. Use of Inconel 671 for corrosion resistance, either as a cladding on stronger metals, or as the entire tube constituent.

b. Application of coal blending and additive techniques to protect against corrosion. This practice will permit utilization of less expensive but otherwise suitable alloys, such as austenitic stainless steel and Incoloy 800.

4.6.2 Ceramic Materials

Silicon carbide is presently the only ceramic heat exchanger material that meets the criteria of this program. These criteria include:

1. Superior resistance to thermal shock
2. High strength

*Ref.: "External Corrosion and Deposits, Boilers and Gas Turbines", American Elsevier Publishing Co., Inc., N.Y. (1971).

3. High creep strength
4. High thermal conductivity
5. Excellent resistance to erosion
6. Long term resistance to coal slag
7. Impermeability
8. Availability in the form of long tubes
9. Low cost
10. Nonstrategic raw materials

The term silicon carbide refers to a family of materials, rather than to a single material. There are basically six members of the family, hot pressed, siliconized, sintered, chemical vapor deposited, recrystallized, and composite silicon carbide materials. Of the six types of silicon carbides, two cannot be considered for advanced heat exchanger tubing. Hot pressed silicon carbide is relatively expensive and limited to small, simple shapes. Hot pressing is not a practical fabrication method for making heat exchanger tubes. A silicon/silicon carbide/carbon composite is being developed by at least two producers but this material is still in the experimental stage. It will probably be several years before this type of silicon carbide can be considered as a candidate heat exchanger material. The four types of silicon carbide that are candidate heat exchanger materials are described below.

4.6.2.1 Sintered Silicon Carbide. Only recently has SiC been pressureless sintered to a high density, 95-98% of theoretical. Until this time, covalently bonded materials, such as SiC were considered unsinterable. However, at least three investigators have successfully sintered SiC powder by using small amounts of additives (usually less than 1% of B and C) and by starting with a very reactive (high surface area) SiC powder. None of these sinterable powders are commercial but one manufacturer, the Carbonundun Company, markets sintered alpha SiC products. This manufactured alpha SiC product is relatively new and still under development. Alpha is the stable, high temperature phase.

The submicron powder used by Carborundum is an Acheson furnace product consisting entirely of alpha silicon carbide. This powder, mixed with the additive, can be formed into shapes using conventional ceramic forming processes but tubes are made exclusively by extrusion. The extrusion process is an economical process for mass producing a product to any length but with a constant cross-section. Thus the extrusion of circumferentially finned tubes would not be feasible. U-shaped tubes are made by carefully bending the "green" extended tube, which at this point contains a large amount of plasticizer and lubricant. These unfired preforms possess enough strength when dried to be easily machined. Sintering is then performed in a reducing atmosphere at over 2000 C (3632 F). Initial results indicate the sintering process occurs in the solid state because there is no evidence of a liquid. Linear shrinkage is 18%. The microstructure is homogenous with an average grain size of 7 microns and porosity, which is distributed uniformly in the grain boundaries, is not connected. The as-sintered surface finish is satisfactory so that grinding or cleanup is not required.

Manufacturing large quantities of ceramic tubes by extrusion and sintering is an established process, but high volume manufacturing of long SiC heat exchanger tubes is in its infancy. Heat exchanger tubes are longer and have a larger diameter than conventional products, the sintering process is more complicated, and control of tolerances and the quality of these tubes is many times more critical than in present production practices. None of these problems are insurmountable but a good deal of effort will be required to develop high volume production of high quality extruded and sintered heat exchanger tubes. To begin with, the extruded tubes are plastic so there is a problem in keeping them straight and round until they are dried into a rigid condition. Large diameter tubes and headers may even deform under their own weight so that removable inserts may have to be used. Handling U-tubes and large headers will be difficult and sintering will require large furnaces.

The most significant advantage of sintered alpha SiC is that it can be used to higher temperatures than the siliconized type of silicon carbide. The mechanical properties of sintered alpha SiC are invariant with temperature to 3000 F, well above wall temperatures anticipated in coal-fueled heat exchangers. Moreover, there is no cost penalty expected for using sintered alpha SiC. The producer, who also manufactures siliconized SiC, projects that the alpha type of SiC will be less expensive than the siliconized type of SiC due to the high cost of high purity silicon required for manufacturing siliconized SiC.

4.6.2.2 Chemical Vapor Deposited SiC. In the chemical vapor deposition (CVD) process for manufacturing SiC tubes, mixtures of gases containing Si and C are reacted at elevated temperatures to form SiC. The tube is formed by a SiC buildup, atom by atom and the temperature of this reaction is lower than that of any other process for making SiC shapes, about 1300 C (2400 F). SiC is deposited at a rate of 6 to 30 mils per hour on graphite mandrels having a very close match in thermal expansion to the deposited SiC. SiC can be deposited on the outside or inside surface of the graphite mandrel, depending on where a smoother surface is desired. The graphite mandrel is removed by leaching.

There are several inherent advantages in the CVD formed SiC. The CVD SiC is highly pure. The only contamination is a slight excess of Si, less than 1%. The grain structure is also very fine. The strength of CVD SiC is relatively very high, the highest of the candidate types of SiC, due to this high purity, high density and fine grain structure. Another significant advantage of the CVD process is dimensional control. Since the heat exchanger tube is deposited on a rigid graphite mandrel, the SiC tube is as straight and true as the graphite mandrel. This is a manufacturing feature that can not be duplicated by any of the manufacturing processes of other candidate SiC materials. The inner surface of the CVD tube will also be as smooth as that of the graphite mandrel.

At present there are one, and possibly two, manufacturers with the technology and the capability for mass production of CVD SiC heat exchange tubes. The most succesful of these two manufacturers has shown, by cost analysis studies, that the CVD process will be economically competitive with the other candidate processes.

The only significant limitation of the CVD process is that shapes other than straight tubes are not presently practical. Fabrication of tubes with fins could be accomplished but the cost would be relatively high. A small U-shaped tube has been made by CVD, but mass production of large U-tubes at a competitive price would require the design and installation of new facilities. It also appears that making tubes longer than 7 feet by CVD may not be practical, but that may be true of all manufacturing methods. Shipping longer tubes may simply cost too much due to their fragility.

4.6.2.3. Siliconized SiC. Siliconized SiC materials are considerably different than sintered alpha and CVD SiC. As a result of the densification process, siliconized SiC materials exhibit a wide variation of microstructures and they contain about 10% free silicon metal. This large amount of silicon has a strong influence on properties at high temperatures because silicon melts at 1410 C (2570 F).

Siliconized SiC obtains its bonding by either, or both, of two mechanisms. In once case, alpha SiC is mixed with carbon powder and a plasticizer. The mixture is extruded into tubes which are dried and then fired at about 2200 C (4000 F). The tubes are set in a pool of molten silicon during the firing process. Silicon vapor and liquid, which permeates the shape by capillarity, react with the carbon powder and the carbon from the pyrolyzed plasticizer to form beta SiC. It is this beta SiC that bonds the original SiC grains together. Refel (TM), KT (TM), and SKT (TM) types of SiC obtain their strength primarily from this reaction.

The other bonding mechanism is recrystallization. A mixture of SiC grains, usually a bimodal mixture of coarse and fine, are suspended in water with the aid of agents. The suspension, or slip, is poured into porous plaster molds where the excess water from the slip is absorbed by the porous wall of the mold. After the desired thickness is obtained, the remaining slip is poured off. The slip-cast tube is removed from the mold after it dries. The dried tube is then fired between 2100 C and 2450 C (3700 F and 4440 F) where the fine SiC grains recrystallize, thereby developing a self-bonded structure. The tube is also in contact with molten silicon during the firing operation. The silicon reacts with any residual carbon and fills in the voids to make the tube impermeable. This bonding mechanism typifies the Noralide (TM) 400 series of materials.

Slip casting has two noteworthy advantages over extrusion for the fabrication of heat exchanger tubes. One, tubes with fins or changes in cross section can be made, and two, straightness can be controlled easier because the tube is not removed from the mold until it is rigid. On the other hand, wall thickness, concentricity, and taper are difficult to control whereas the features are constant in the extrusion process. It should be pointed out that all prototype hardware made to date, i.e., long tubes (7 feet), and manifolded U-tube modules, has been produced by slip casting the Noralide type of SiC.

Since the microstructure and the nature of the bonding can vary considerably in different types of siliconized SiC, it follows that the strength can also vary considerably from type to type, and from batch to batch. But the most significant characteristic of siliconized SiC is the precipitous drop in strength starting at about 1240 C (2200 F) (Fig. 4.6.8). Thus, tensile overloading at high temperatures may be a serious problem when designing with siliconized SiC. One potential advantage in designing with siliconized SiC compared to pure SiC, on the other hand, is that it has a lower elastic modulus and the modulus decreases with temperature. The elastic modulus of sintered alpha and CVD SiC is higher and it does not decrease at temperature increases. The lower elastic modulus leads to lower thermal stresses.

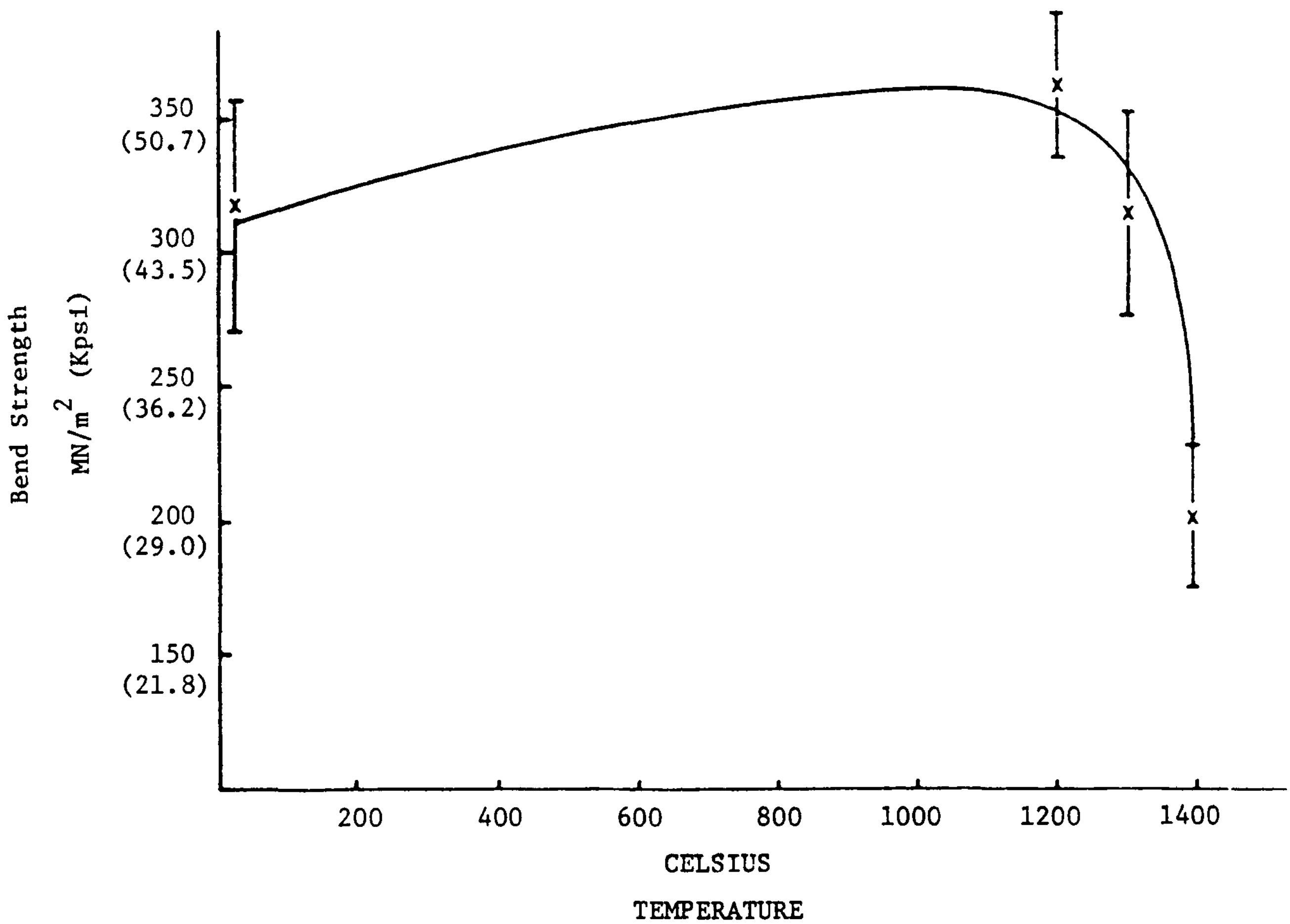

Figure 4.6.8 NC-430 Strength Retention After 100 Hour Exposure in Air

Besides the loss in strength above 1240 C (2200 F) there is a potential corrosion problem. Silicon metal reacts with molten coal slag. Although the reaction is rapid on the surface of the tube, depending on microstructure, it may proceed slowly in the thickness direction. Nevertheless, long-term life of siliconized SiC tubes will be adversely affected by coal slag corrosion.

4.6.2.4 Recrystallized SiC. Recrystallized SiC shapes are fabricated in the same way described in the last section on siliconized SiC except that the porous structure is not filled with molten silicon during the firing operation. The reason for using this material is because the properties are not degraded by molten silicon at elevated temperatures. However, the strength level of recrystallized SiC is much lower, about one-half that of siliconized SiC, because of the 10 to 15 percent porosity and because some of the strength developed in the siliconized SiC is due to reaction of the residual carbon and the molten silicon. The porous recrystallized body would have to be coated with a dense layer of CVD SiC to make it impervious to the heat exchanger working fluid. Thus, that type of heat exchange tubing would have half the strength of other candidate materials at almost double the cost.

4.6.2.5 Fabrication of Ceramic Structures. Fabrication processes are intimately tied to the characteristics of the SiC material. For this reason, the basic fabrication processes have already been discussed in the materials description section and the key features of the three fundamental fabrication methods considered are compared in Table 4.6.6. Table 4.6.6 shows that each process has advantages over the others but it also shows that the pluses and minuses are equally distributed. Thus, no process is clearly superior to the other for all applications. Of course, materials selection will also be based on materials properties and manufacturing costs. At this point, neither of the materials or processes is clearly more economical than the other. The purpose of this section will be to address manufacturing problems that apply to all types of SiC materials.

Table 4.6.6 Comparison of Manufacturing Methods

	Tolerance Control								
	OD	ID	CONCENTRICITY	ROUNDNESS	TAPER	STRAIGHTNESS	VARIABLE GEOMETRY	LARGE U-SHAPES	SURFACE FINISH
Extrusion									
Siliconized SiC	+	+	+	-	+	-	-	0	0
Sintered SiC	+	+	+	-	+	-	-	0	+
Slip Casting	+	-	-	+	-	+	+	0	0
Chemical Vapor Deposition	-	+	0	+	0	+	-	0	+

\+ A positive feature of the fabrication process

\- A negative feature or a potential problem.

0 A feasible feature of the fabrication process but needs to be demonstrated

The problem that comes to mind first is length. Heat exchanger designers, who demand long tubes for design efficiency, are considering tubes as long as 40 or even 80 feet. Such lengths are not practical for ceramics unless the tube is made of a series of smaller tubes joined in at the site. The question then is, what is a practical tube length for manufacturing? Seven foot SiC tubes, the longest SiC ever made, are presently being supplied for a solar power heat exchanger. This may turn out to be the largest practical size because problems will be compounded for the production of longer tubes. For examples, tolerance will be much more difficult to hold, special handling will result in higher manufacturing cost on a weight basis, firing will require larger furnaces and longer schedules, the scrap rate will be higher, quality assurance will be more complicated and, finally, shipment to the site will be very expensive due to the required special handling and packaging. Moreover, quality assurance would have to be repeated on site to assure there was no damage to the long brittle tubes during transit.

Holding tolerances, including I.D., O.D., roundness, concentricity, straightness, and taper will be difficult. Tolerances will have to be maintained, however, if for no other reason than because anomalies in shape and in wall thickness will result in excessive stresses.

Surface condition is important from two standpoints. One, ceramics are highly sensitive to surface defects and, two, impurities on the surface (such as silicon metal) will react with the coal slag on one side, or, on the other side they could spall and cause erosion in the turbomachinery. The manufacturing process will have to generate acceptable surfaces in the as-fired tubes because it will probably be impractical and too expensive to process the total surface area of a large heat exchanger.

The fabrication of large headers and U-tubes with large bends is yet another large problem in scaling up the manufacturing process without losing the high strengh properties of the SiC material. Scaling up will require larger equipment, especially high-temperature furnaces, and new techniques in handling the green shapes. For example, large headers will have thicker walls. Thick walls lead to two new problems. One, the green strength of the shape may be inadequate to support itself and, two, wall thickness is limited by the fabrication process. Residual stresses may be generated in thick walls by CVD processes, molten silicon may not be wicked deep enough into thick-walled bodies to complete the SiC bonding reaction, and extruded plasticized mixtures may not retain their shape as they come out of the extrusion die. At least in the case of sintered SiC, cold pressing or isostatic pressing may have to be used instead of extrusion.

The refractory ceramics industry, without exception, is not set up to measure and monitor the dimensions of SiC heat exchanger tubes, nor is it even presently possible to perform the necessary NDE (nondestructive evaluation) for detecting flaws. In terms of quality assurance, the ceramic industry, possibly with the exception of the electronic ceramics industry, lags years behind the metal industry. After all, very little, or no, precision inspection is required for bricks, hearths, sagers, combustion tubes, and the like. This capability will have to be developed starting from an almost zero baseline. The same is true for flaw detection except that the techniques and the equipment are nonexistent or in the development stage. Existing techniques are of minimum value because the flaws of importance in ceramics are about three orders of magnitude smaller than those in metals. Ultra-high frequency ultra-sonics and microfocus X-radiography are two of the more useful methods used at present. But applying these methods to SiC tubes produced in a high volume will require considerable development. It may be possible to supplement NDE with proof testing, but some NDE for detecting flaws will certainly be required. Proof tests cannot be devised to completely simulate conditions in the heat exchanger environment.

SiC-to-SiC joints are the most critical technological challenge for constructing a large advanced heat exchanger. A satisfactory method of joining SiC to

SiC is essential because (1) shipping full-sized tubes, especially U-tubes, to the site is impractical, (2) the tubes delivered to the site must be joined to a header, and (3) headers must be joined on site because they could not be shipped full length. Headers may be as long as the entire heat exchanger complex.

Strong joints can be produced by present technology but the process has severe limitations. Joints must be fired at the same temperature as the original part and the entire assembly must be placed in the furnace. Within this limitation no heat exchanger module larger than existing SiC sintering furnaces can be fabricated. The volume of these furnaces is small, in the order of 10 cubic feet, due to the high-temperature and controlled atmosphere requirements. Obviously, techniques for forming strong joints formed by local heating must be developed. One inherent problem with localized heating is the thermal stresses that are developed adjacent to the joint. Thus joints formed at lower temperatures, such as by CVD, would have an advantage. Provided the surface ends up recessed as in metal welds, resultant stress risers will require severe safety factors in design stress, and allowable design stress is already marginally low.

Glass braze joints are being evaluated at Solar Turbines International. A successful glass joint would have the attributes of a low joining temperature and stress relaxation during service. These joints may be developed to where they can be used in specific applications but glass joints will not provide the ultimate solution. There are seemingly unsurmountable technical problems for their use under a wide range of conditions. For one thing, glasses are dynamic systems at elevated temperatures. Glasses are excellent solvents and they devitrify over long periods particularly as they dissolve impurities. As crystals form, the properties, such as the thermal expansion coefficient, change, causing thermal stresses to develop between the glass and the crystalline phases and between the braze joint and the SiC tubing. As temperature increases, the crystalline phases dissolve and the glass becomes less viscous, more reactive, and the joint becomes weaker. As temperature gradually decreases, devitrification processes increase, viscosity increases, and the brittle, poly phase joint becomes highly susceptible to the thermal shock. Design of a given glass system for a narrow temperature range may be feasible, but it is unlikely that any glassy system will have all of the necessary properties of a stable, strong braze joint that will retain these properties over a wide temperature band for a long duration. The temperature of joints will fluctuate widely as a function of position and with time in a real heat exchanger environment.

Methods for forming strong, reliable, stable joints made by spot heating must be developed before large advanced heat exchangers can be designed and manufactured.

Methods of metallizing SiC and then brazing it to metal are reported but the useful service temperature of the joint, and cyclic life of the joint may not be adequate for advanced heat exchanger applications. This type of joint needs development. Another method of joining SiC to metal is use of a mechanical joint. A mechanical joint offers the advantage of easy assembly and disassembly, both at ambient temperature. This type of joint is certainly more practical than brazing or chemical diffusion and is within the state-of-the-art. One design is reported in the recent EPRI report on advanced ceramic heat exchangers.

4.6.2.6 Design Properties of SiC Ceramics. All available property data for sintered, CVD, and siliconized SiC was collected from the literature and vendor publications. These data were reviewed from the standpoint of reliability and applicability, and the best data was used to generate design curves. Data is scarce and many of the materials tested were not well characterized, or the material was not the same as that being considered. Much more materials property testing will be required for designing advanced ceramic heat exchangers. The mechanical properties that are required include modulus of rupture, tensile strength, stress rupture, creep, elastic modulus, Poisson's ratio, K_{IC}, and slow crack growth rate. All tests must be performed on material representive of the production heat exchanger tubes, and tests must be conducted under simulated heat exchanger conditions (at least as many conditions as possible).

Calculation of an allowable design stress is important in this program because it indirectly determines the unit cost of power. The allowable stress is used to determine the wall thickness of the heat exchanger tubes, which in turn dictates the volume of heat exchanger material required, which in turn dictates the cost of the heat exchanger facility, which is a significant factor in the final cost analysis of unit power generation. Determining the working stress for ceramics is very complicated, much more so than for metals. There are more variables for determining working stress for ceramics and the data, as stated at the beginning of this section, is inadequate.

Unlike metals, even so-called brittle metals, ceramics are completely brittle and unforgiving and, consequently, they exhibit a very low fracture toughness. Since ceramics have a low toughness, they are highly susceptible to flaws. The critical flaws in ceramics are extremely small compared to those in metals; they are in the order of microns compared to millimeters for most structural metals. Flaws always exist in ceramics because they are smaller or in the same size range as microstructural features, such as pores, inclusions, unbonded grains, atomic scale defects in the crystal lattice, and surface cracks. Since the strength of the ceramic body, among other things, depends on the size and distribution of flaws in the volume of material and on the stress distribution, ceramics exhibit considerable scatter and they must be treated on a statistical basis rather than on a deterministic basis. The statistical relationship most commonly used is the Weibull equation:

$$P = 1 - \exp\left\{ - \int_v \left[\frac{\sigma - \sigma_u}{\sigma_o}\right]^m \right\} dv$$

where:

σ = maximum tensile stress

σ_u = stress at zero probability of failure

σ_o = a normalizing constant

P = probability of failure

v = stressed volume

m = Weibull modulus

The Weibull modulus is a measure of failure stress. Large values (e.g., greater than 30 for metals) indicate little scatter and a small effect of size on the material's strength. Smaller values indicate large scatter and a large effect on the material's strength.

While some strength data are available for the candidate SiC materials, the data were obtained from small bend specimens. The maximum stressed volume in a small bend specimen is orders of magnitude smaller than the stressed volume in a long heat exchanger tube. Also the stress in the tube is biaxial rather than uniaxial like it is in the small test bar. Thus the best design data available, although extremely limited, are the burst test data.** The data for the strongest tubes tested* were used to generate an iterative study shown in Table 4.6.7. There was inadequate data to determine the Weibull modulus so a realistic value was assumed to be 10. This was bracketed by a low value of 5 and a high value of 20. The mean stress is represented by the failure probability of 0.5. The other failure probabilities represent one failure in 1000, 10,000, and 100,000 tubes. It should be noted that this is the probability each time

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* CVD SiC tubes made by Materials Technology Corp.

** "Preliminary Evaluation of Tubular Ceramic Heat Exchanger Materials", Final Report, AiResearch Casting Co., Report No. 77-14570, ERDA Contract EX-76-C-01-213, 8 December 1977.

Table 4.6.7 Parametric Study of Allowable Design Stress (KSI) for CVD SiC

Probability of Failure	SiC Tube Length*					
	8 Inches			20 Feet		
Weibull Modulus	5	10	20	5	10	20
0.5	23.2	23.2	23.2	11.8	16.6	19.6
10^{-3}	6.3	12.1	16.7	3.2	8.6	14.1
10^{-4}	4.0	9.6	15.0	2.0	6.8	12.6
10^{-5}	2.5	7.6	13.4	1.3	5.4	11.2

* 1" O.D., 3/4" I.D.

the tube is stressed, time dependent failure modes (see Table 4.6.8) and temperature effects were not included in determining the allowable stress. These factors would in all likelihood decrease the stress. Another assumption was that the flaw distribution in long mass-produced tubes will be the same as that in the 8-inch tubes used for testing. The values in Table 4.6.7, consequently, must be considered speculative. Much more work clearly needs to be accomplished before reliable design stresses can be calculated.

As an approximation, a value of 6 KSI was selected as a guideline in designing ceramic heat exchanger tubes. This value is based on a Weibull modulus of 10 and a failure probability between 10^{-4} and 10^{-5}. The 6 KSI design stress is about as low a value as can be used to design feasible heat exchangers, (considering that no yielding is permissible). In effect, its adoption says that if SiC ceramics technology is developed sufficiently to justify that design stress then the CCGT combustor/heat exchanger designs synthesized on this contract provide some probability of success. If the 6 KSI is never attained then it is unlikely that SiC ceramics will be successful in any CCGT combustor/heat exchanger.

4.7 SAFETY

The design, fabrication and operation of pressurized equipment such as steam boilers and fired CCGT combustor/heat exchangers are governed by safety codes which have the force of law in nearly all jurisdictions of the United States. These codes apply specifically to the pressure parts of the combustor/heat exchanger system, i.e., those parts that may explode due to working fluid pressure. The safety codes are designed to avoid loss of life and property in the operation of such equipment. The codes are devised and maintained by the American Society of Mechanical Engineers, their enforcement is systematized by the National Board of Boiler and Pressure Vessel Inspectors, and their application is enforced by various governmental jurisdictions, which may be for instance, state, county or city.

TABLE 4.6.8 TIME DEPENDENT FAILURE MODES

- SUBCRITICAL (SLOW) CRACK GROWTH
    - STATIC FATIGUE
    - DYNAMIC FATIGUE
- HIGH TEMPERATURE CREEP
    - STRESS RUPTURE
    - THERMAL LOW CYCLE FATIGUE
- MATERIAL INSTABILITY
    - CHANGE IN PHYSICAL PROPERTIES
        - ALTERS STATE OF STRESS WITH TIME
    - STRENGTH DEGRADATION
        - INCREASES TIME DEPENDENT FAILURE PROBABILITY

Additional safety codes have been formulated to govern the safety of electrical installations; steam and fuel gas and other pressurized piping; ignition equipment; and the like. These codes constrain the design of equipment other than "combustor/heat exchanger pressure parts".

4.7.1 Boiler and Pressure Vessel Code

The pressure parts portion of the fired CCGT combustor/heat exchanger would presently fall within the jurisdiction of Section 8, Division 2 of the ASME Boiler and Pressure Vessel code. This judgment derives from reading of the scope of the various sections of the code. It is found that Section 1 is inapplicable because the fluid being processed in the heater is neither water nor a boiling organic fluid. Section 3 is intended for cases where nuclear processes are involved so it is inapplicable. Section 8, Divison 1, is inapplicable because of the specific disclaimer on direct fired process heaters. Section 8, Division 2, is applicable because of paragraph A100-C under "Scope" which states that pressure vessels not covered by Section 1, Section 3, or Section 4 and subject to direct firing may be constructed in accordance with the general rules of Divison 2 of Section 8 of the code.

An alternative to the application of Section 8, Division 2, is a modification of Section 1 to authorize the applicability of its provisions to direct fired CCGT combustor/heat exchangers. This change is probably justified by the close similarity between a CCGT fired combustor/heat exchanger heating helium or air and a separately fired steam superheater which uses dry steam as the working fluid.

The metals planned for application to the heat exchanger surfaces of the CCGT combustor/heat exchanger have been discussed in Section 4.6 above. The allowable stress values utilized inthe design are in each case consistent with the values permitted under the rules of Section 1 of the ASME Boiler and Pressure Vessel code. In the case of Inconel 617 this material is not yet authorized for utilization by the code in boilers and pressure vessels. It will be necessary to have it accepted and incorporated in the code, or alternatively special dispensation received from the safety agency having jurisdiction for its use as a non-coded material. All of the other metals, and their permissible operating temperatures and stress levels as utilized in the design studies are consistent with the rules of Section 1 and Section 8 of the Boiler and Pressure Vessel code. It is not anticipated that severe difficulties would be encountered in obtaining approval for design, fabrication, and operation of CCGT combustor/heat exchangers utilizing those metals.

The safety code acceptability of ceramic pressure vessel materials appears very different from that applying to metals. The ceramic's strenths and weaknesses have been described in detail in Section 4.6 above. Their most severe weakness with respect to safety codes is the brittle nature of ceramics and the always present probability that a sudden fracture will occur, with destructive release of pressure and the flying about of destructive pieces of the pressure vessel. It seems very likely that obtaining the acceptance of ceramic materials as authorized portions of boilers and pressure vessels will be a long and difficult struggle, and there is no guarantee of success. In deference to this safety aspect, all of the designs studied which utilize ceramic materials for containing pressure are arranged so that the ceramics are completely contained within metal walls, which thereby prevent the release of pressure, or pressure propelled pieces of ceramic, within the normal working spaces of the power plant. Thus ceramic heat exchanger surface is always contained within furnace or heat exchanger walls that are fabricated of metal, and any exterior ceramic manifolding is contained within a metal shell that will prevent fragmentation. These constraints upon design are believed a necessary step towards obtaining authorization for installation and operation of experimental CCGT combustor/heat exchangers utilizing ceramic heat exchanger surface.

4.8 ECONOMIC FACTORS

Many of the constraints and design criteria applying to large coal fired combustor/heat exchangers, such as utility boilers and CCGT heaters, are economic rather than physical. It is economic considerations for instance that call for the application of the cheapest alloy that will meet the pressure and temperature requirements at each station along the heat exchanger flow path. It will be economic considerations that will largely determine the outcome of the competition between pulverized coal and fluidized bed combustion methods. Economic considerations also enter into the decision as to whether the 350 MWe system heat input should be supplied from a single combustor/heat exchanger or from a group of smaller exchangers. The general economic law that applies here may be called the "power law" of costing. It has been explained and applied to a wide variety of equipment in many references.* The law correlates an observed phenomenon, which is that when comparing process installations it is found that the installed costs of the installations vary roughly as their output capacities raised to some fractional power, usually on the order of 0.6. This power law relationship of installed cost has been found to apply to complete boiler installations, as indicated in Fig. 4.8.1. While there may be deviations from the law, as for instance, if each of a number of multiple units can be factory fabricated while a large single unit must be field fabricated, it is obvious that the law applies in the case of steam based utility stations in the size range we are studying. There has been a continuous trend towards larger and larger single boiler installations, (although this has somewhat slowed recently due to unreliability encountered with super-size units of 1000 MWe capacity and higher). In the light of this economic relationship, the CCGT combustor/heat exchanger designs studied under this contract were in general arranged to provide the heat input to the thermodynamic cycle while using as few separate combustor/heat exchanger units as was technically feasible.

Ref: "Modern Cost Engineering Techniques", H. Popper, McGraw Hill, 1970, and K. M. Guthrie, Chemical Engineering, March 24, 1969

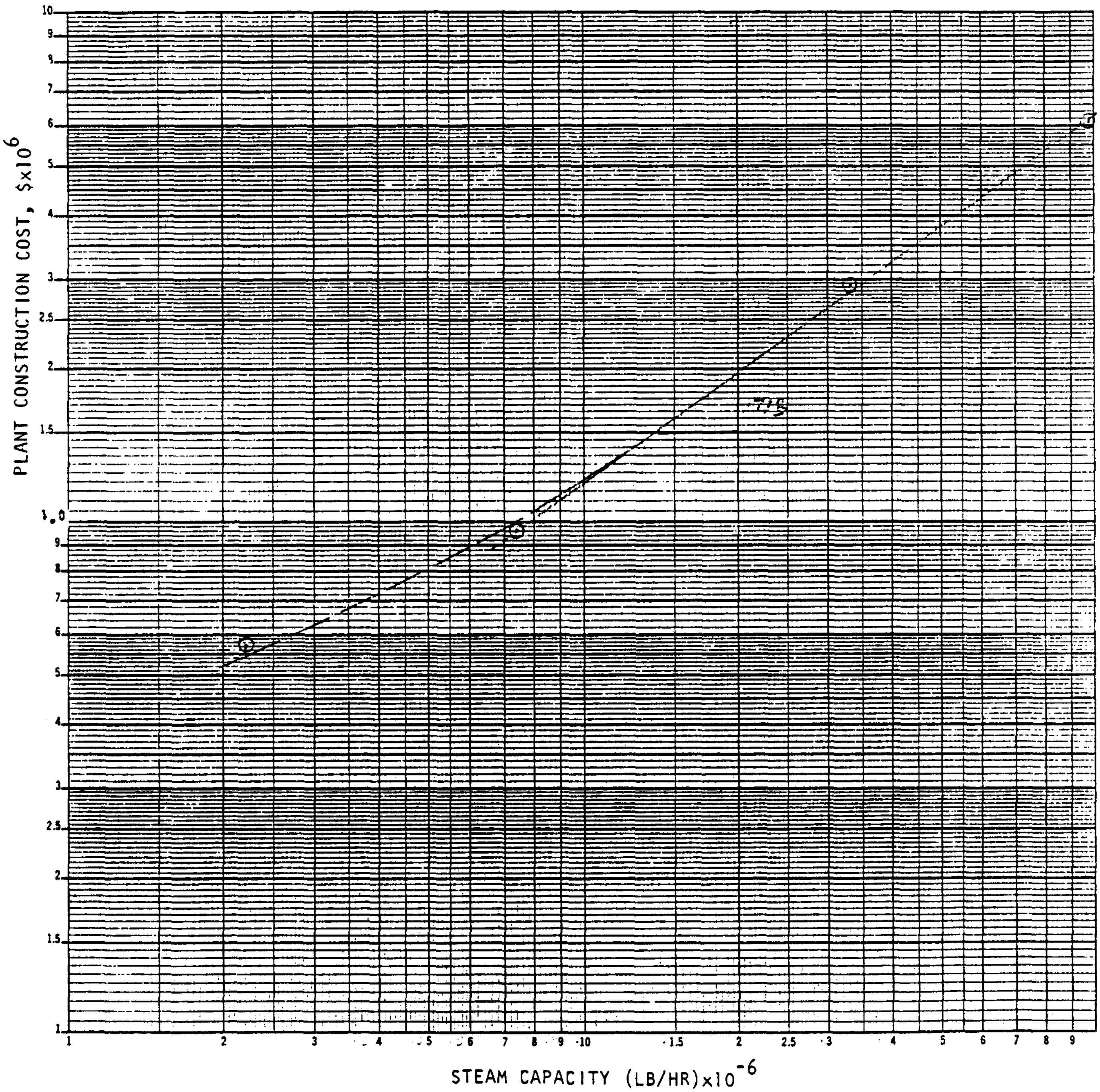


Figure 4.8.1 Plant Construction vs Installed Capacity

Based on Figure 11/5 of "Report on Total Energy Feasibility Criteria", for U.S. Army, Belvoir, Contract DAAK 02-73-C-0370

5.0 CONCEPTUAL DESIGNS

The material presented in Section 3.0 discussed the reference designs of the coal fired combustor/heat exchangers studied on this contract. The reference design material listed the required operating conditions and contractual groundrules which govern the design of the combustor/heat exchangers. Section 4.0 discussed the design criteria and constraints which govern the design of the coal fired CCGT/heat exchangers, defining available materials properties, combustion methods and the like. This section 5.0 discusses and describes the conceptual designs of coal fired CCGT combustor/heat exchangers which were synthesized in the course of coming to a decision upon which concepts would be carried through the preliminary design stage.

It will be seen from Section 3.0 that the reference design conditions were chosen so that the overall study would evaluate a wide range of combustor/heat exchanger operating conditions. This was done consciously, so as to cover the probable range of operating conditions that might be encountered during the development of CCGT systems, from their first applications to the maturity occurring after high temperature materials technology has become available.

The reference design conditions are tabulated in Section 3.1 and may be summarized as requiring the following:

a. A nominal 1550 F working fluid temperature design which represents the highest working fluid temperature likely to be available from all metal heat exchangers. This reference design may also be extended downward to lower operating temperatures to optimize the capital cost versus fuel cost relationship.

b. A nominal 1750 F working fluid design representing an operating condition that calls for the utilization of ceramic materials, but which might be considered an early application of ceramics in that the temperatures are well within the ceramics capability.

c. A 2250 F working fluid design representing an advanced application of ceramics, utilizing them at close to the upper limit of their projected capabilities, and thereby attaining the highest CCGT cycle coal pile to bus bar efficiency.

The conceptual designs considered for each of the above applications will be discussed in turn.

5.1 CONCEPTUAL DESIGNS FOR 1550 F AND LOWER WORKING FLUID TEMPERATURES

All of the combustion concepts described under "Design Criteria" in Section 4.0 appear to have some element of suitability for coal fired CCGT combustor/heat exchangers whose maximum working temperature is 1550 F or lower. Thus, there is a wide variety of choices among the conceptual designs available for application to such heaters. The concepts considered during the conceptual design phase of this contract study are discussed below.

5.1.1 Dry Bottom Pulverized Coal Fired Combustor/Heat Exchangers - European Practice

All of the direct coal fired combustor/heat exchangers for CCGT application that have actually been built and operated have been dry bottom pulverized coal installations with radiant furnaces. A description of the state-of-the-art of design and operation of these combustor/heat exchangers is contained in a report by R.A. Harmon to the Department of Energy.* A summary of the design conditions for the installations is presented in Table 5.1.1 and sectional views of two installations are presented in Fig. 5.1.1 and 5.1.2. All this material is extracted from the Harmon report.

As is evident from the table and figures all of the units are very small as compated to the target capacity on this study, the largest being only 4% of

* Ref.: "Current European Technology for Design of Direct Primary Combustion Heat Exchangers for Closed Cycle Gas Turbine Power Plants", by R. A. Harmon, DOE Contract EX-76-C-01-2453, November 1977

Table 5.1.1 Main Data for Coal-Fired Air Heaters

	Plant		A	B	C	D	E
	Place		Ravensburg	Coburg	Oberhausen	Moscow	Haus Aden
	Maximum continuous output	MW	2.3	6.6	13.75	12.0	6.37
1	Air heater as in Fig.	—	10	11	12	14	16
2	Fuel	—	hard coal	hard coal	hard coal	brown coal	mine gas and hard coal
3	Number of combustion chambers	—	1	1	2	1	1
4	Shape of combustion chamber	—	round	octagonal	octagonal	octagonal	octagonal
5	Ignition muffle	—	with	without	without	without	with
6	Number of burners	—	3	4	2 x 4	6	5
7	Ignition	—	light oil	town gas	town gas	light oil	mine gas
	Working air						
8	Throughput	kg/s	26.5	86.5	129.3	110.4	66.6
9[1])	Inlet temperature	C	397	434	419	388	424
10[1])	Inlet pressure p_1	ata	32.2	38.7	36.86	36.0	32.46
11[2])	Outlet temperature	°C	660	680	710	680	681
12[2])	Outlet pressure p_2	ata	31.07	36.83	35.2	34.92	31.1
13[3])	Pressure loss in air heater	%	3.5	4.8	4.5	3.0	4.2
	Combustion chamber						
14[4])	Volume loading	Gcal/m³ h	0.133	0.112	0.137	0.125	0.137
15[5])	Cross-sectional loading	Gcal/m² h	1.09	1.28	1.30	1.497	1.37
16[6])	Inside width between tube banks	m	3.2	5.0	4.65	5.8	4.3
17	Length of irradiated tubes	m	7.5	10.15	9.5	12.0	10.0
18	Number of tubes	—	136	320	2 x 240	320	320
19[7])	Tube dimensions	mm	32 x 2.5	32 x 3.0	32 x 2.75	40 x 4.0	31.8 x 2.5-3.0
20	Shape factor	m^{-1}	1.46	0.98	1.06	0.82	1.12
21[8])	Tube arrangement ratio t/d	—	2.31	1.62	2.01	1.5	1.34
22[9])	Preheating of the combustion air	°C	20–430	30–445	20–420	20–420	30–440
23	Exhaust gas temperature	°C	160	160	160	190	180

1 Temperature and pressure at the air heater inlet (in the inlet header of the convection part).
2 Temperature and pressure at the air heater outlet (in the end header of the radiant part).
3 Total pressure loss of the working air in the air heater $\Delta p = (p_1 - p_2)/p_1 \cdot 100$.
4 The volume loading means the ratio of the total heat supplied to the combustion chamber by the fuel, hot air and cold air to the combustion chamber volume.
5 The cross-sectional loading is the ratio of the total heat supplied to the cross-sectional area of the combustion chamber.
6 The inside width between tube banks relates to the centre of the tube.
7 The first figure is the external tube diameter d, the second and any third figure is the wall thickness.
8 The tube arrangement ratio is the tube spacing t/external tube diameter d.
9 Temperatures of the combustion air before and after the preheater.

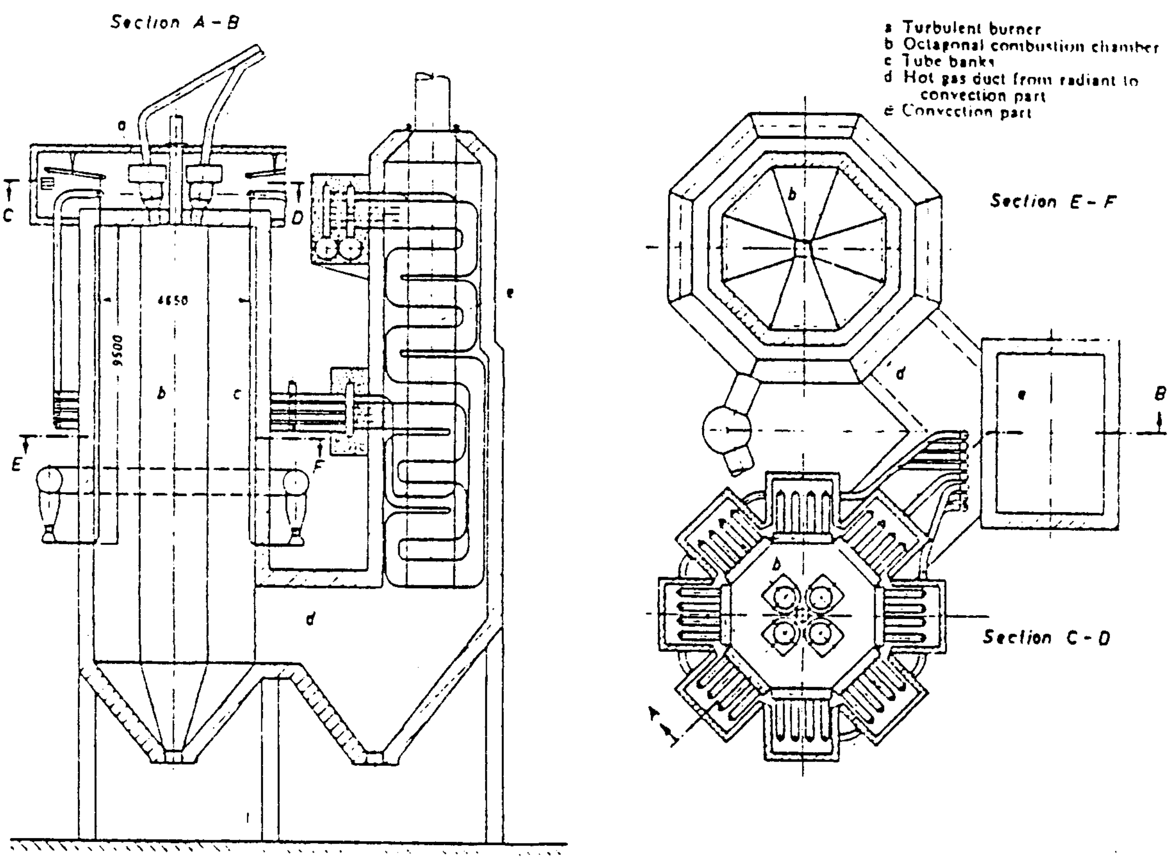


Figure 5.1.1 Coal-Fired Air Heater of the Oberhausen Hot-Air Turbine Plant (Dimensions in mm)

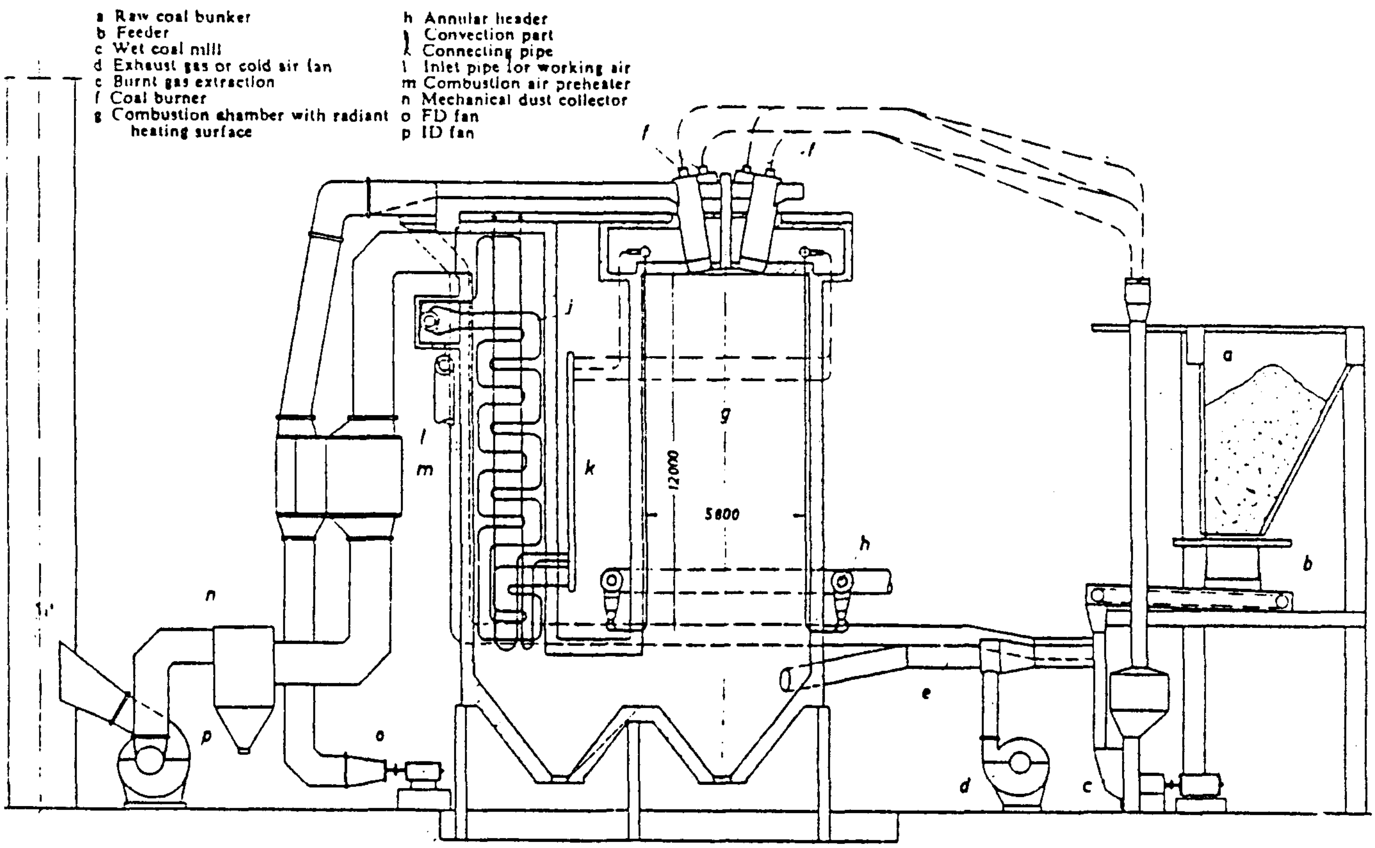


Figure 5.1.2 Air Heater of the Moscow Hot-Air Turbine Plant Fired by Brown Coal (Dimensions in mm)

the target capacity and the average being on the order of 2%. The units are typically designed with circular or octagonal refractory furnaces having standoff radiant heater tubes positioned in front of the refractory. Tube spacing is about 1-1/2 times the tube diameter. The burners are typically located in the refractory roof of the furnace and fired downwards. The Harmon report states an opinion that "at the present stage of development, it is believed that coal fired plants could be built to a high level of confidence with capacities of up to 50 MWe and turbine inlet temperatures up to 800 C".

The existing combustor/heat exchanger design concept is not directly scalable to the 50 megawatt and certainly not directly scalable to 350 megawatts. The economics of power plant installations as discussed in Section 4.8 are such that the use of a large number of separate units in parallel is generally economically unacceptable, as for instance, seven 50 megawatt units or fourteen 25 megawatt units. Additionally, the "German designs" make extensive use of refractory walls, partially protected as in the radiant tube section, and unprotected as in the ash hoppers. Such construction is basically obsolete for large central station units because of problems with slagging, air infiltration, refractory deterioration, and load changing limitations. For all of these reasons, it was decided not to follow the German design closely as one of the concepts to be carried through the preliminary design stage.

5.1.2 Conventional Radiant Furnace Pulverized Coal Concepts

Modern practice in dry bottom pulverized coal fired steam boilers in the size range of interest, i.e., 350 MWe, is illustrated by Fig. 5.1.3.
In such a boiler all radiant surface is "cold" surface, there are no refractory walls in the furnace area, and the hopper area is also fully cooled. Such construction has been found to be economically viable in the boiler industry and to provide the basis for continuous and reliable operation. An adaptation of

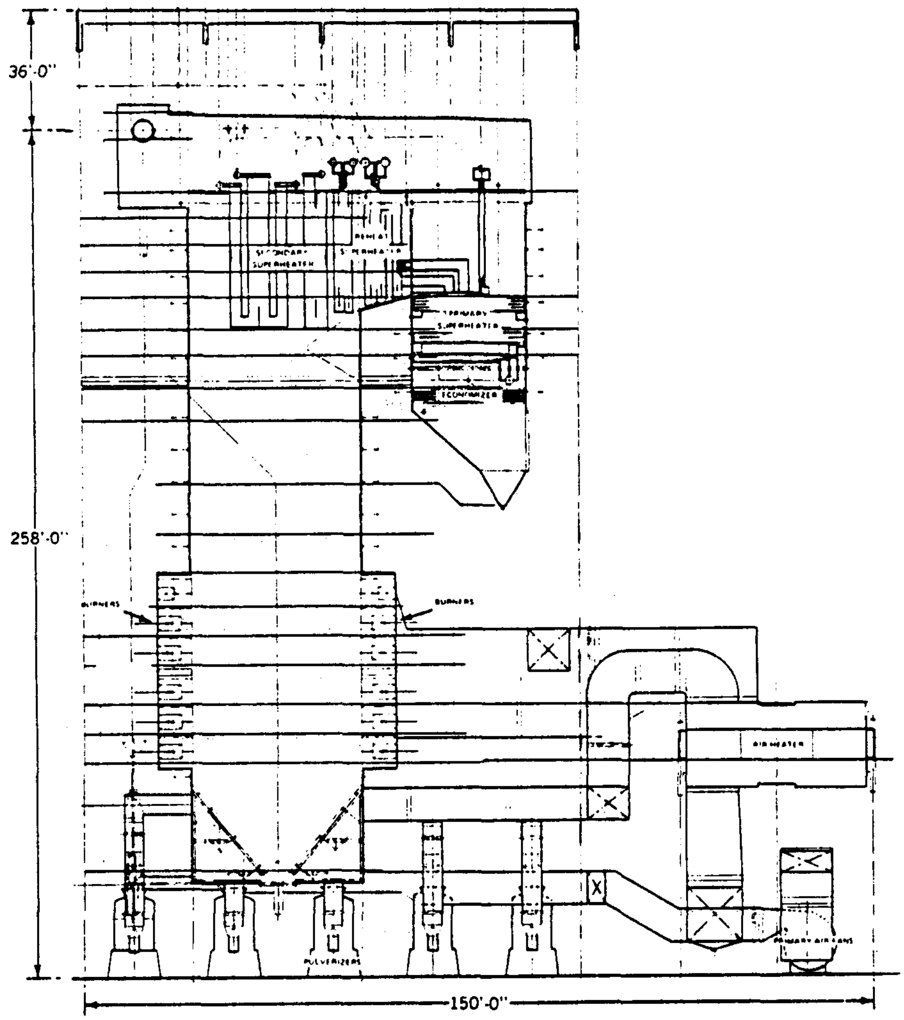


Figure 5.1.3 600 MW Pulverized-Coal-Fired Natural Circulation Boiler

Source: Proceedings of the American Power Conference, Vol. 38, 1976, p. 291.

these concepts to a service similar to that required of the CCGT combustor/heat exchanger is illustrated in Fig. 5.1.4. This hopper bottom unit operated on oil firing for a number of years, and never fired coal. Its general design configuration however, including the hopper bottom, is consistent with coal firing. The heat input to the working fluid of this separately fired heat exchanger was about 1/4 of what would be required for a 350 MWe CCGT combustor/heat exchanger. It is believed that this unit represents the largest separately fired steam superheater constructed to date.

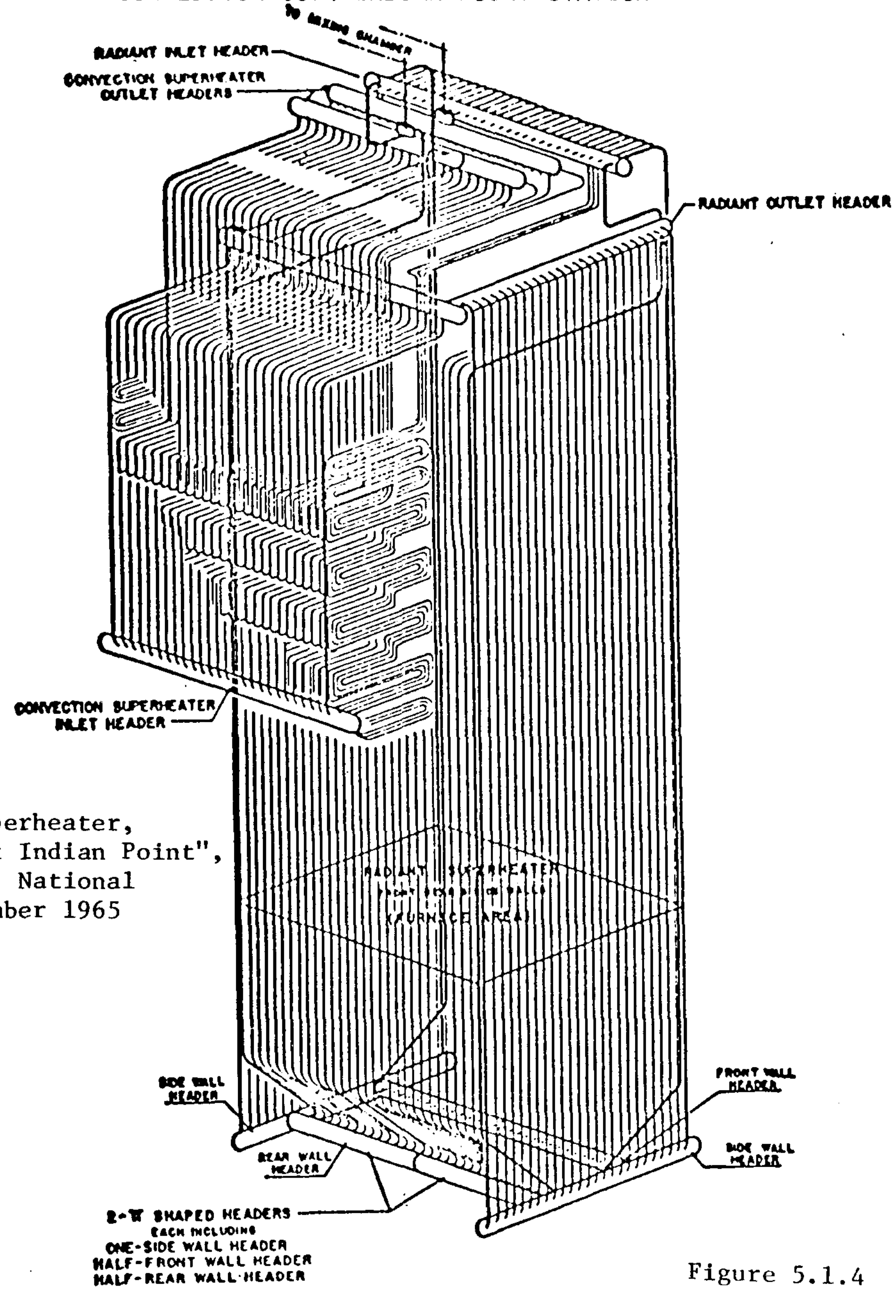
The dry bottom pulverized coal radiant furnace concepts presented by Figs. 5.1.3 and 5.1.4 appear the most suitable for extrapolation to the requirements of a 350 MWe CCGT combustor/heat exchanger. It was therefore decided that this would be one of the design concepts that would be carried through the preliminary design stage, (to assess the key features of the design and to estimate its cost). Figure 5.1.5 is constructed to depict the heat transfer characteristics of a 1550 F PC fired CCGT furnace as a reasonable compromise among the sizing, operability, and materials in the design. The plot is based on equilibrium calculations and also reflects the flue gas recirculation.

5.1.3 Fluidized Bed Combustion Concepts

The design criteria applying to the fluidized bed combustion concept were discussed in Section 4.1.1.2, where the limiations on bed operating temperatures were discussed. One of the most advanced embodiments of the fluidized bed combustor process is the 30 megawatt multi-cell fluidized bed boiler plant installation at Rivesville, West Virginia. This installation has been the subject of numerous technical papers, of which Mesco* presents an overall view. Figure 5.1.5 shows the cutaway views of the system. The 30 MWe Rivesville boiler is designed to produce 300,000 lbs of steam per hour at 1300 psi and 925 F steam temperature. The approximate dimensions of the 4 cells together are 12 ft wide by 25 ft high by 38 ft long. This Rivesville installation has approximately one-tenth of the heat input capacity required for a 350 MWe CCGT combustor/heat exchanger. The experimental installation has been operated intermittently over the past year and has successfully produced steam. Its operators find the optimum combustion bed temperature to be approximately 1550 F.

* Ref.: "Atmospheric Fluidized Bed Steam Generators for Electric Power Generation", Mesco & Gamble, American Power Conference, 1974

FOSTER WHEELER SEPARATELY FIRED SUPERHEATER
CON EDISON CO., INDIAN POINT STATION



"The Separately Fired Superheater,
a Nuclear Application at Indian Point",
by McCormick & Gorzengo, National
Power Conference, September 1965

Figure 5.1.4

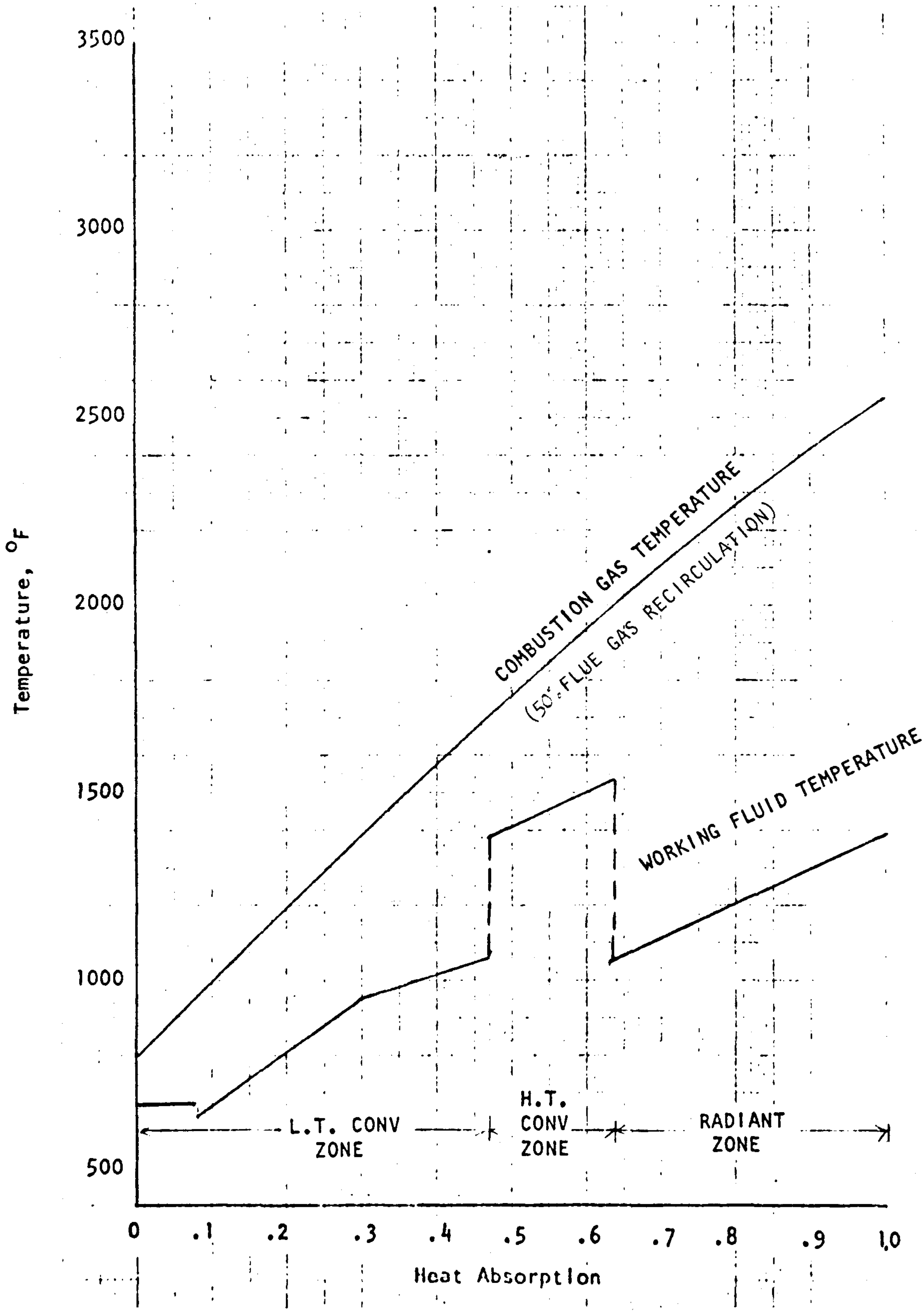


Figure 5.1.5. Heat Transfer Characteristics of 1550F/1550F PC System

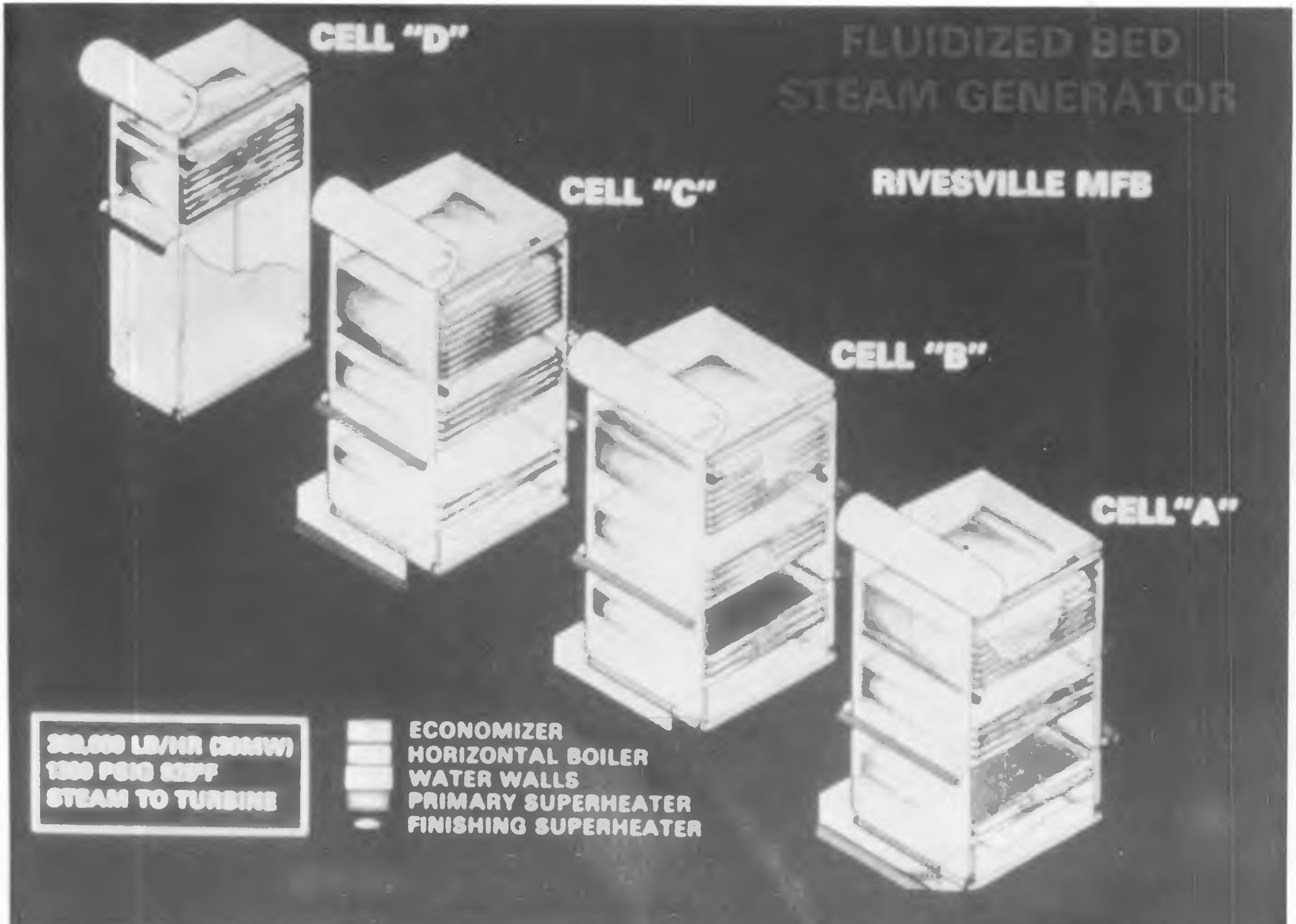

FLUIDIZED BED
STEAM GENERATOR
RIVESVILLE MFB
CELL "D"
CELL "C"
CELL "B"
CELL "A"
STEAM TO TURBINE
ECONOMIZER
HORIZONTAL BOILER
WATER WALLS
PRIMARY SUPERHEATER
FINISHING SUPERHEATER

Figure 5.1.6

When designing for 1550 F maximum working fluid temperature, the minimum required bed operating temperature for an economically viable design is anticipated to be in the neighborhood of 1650 F. The effect of operating at 1650 F is to require more input of limestone for acceptable sulphur capture than would otherwise be the case.

The Battelle Columbus Laboratory has developed a concept called the "multi-solids fluidized bed combustor" which experimentation has indicated can operate at higher bed temperatures than the "conventional fluidized bed" without as much penalty in added limestone consumption. The design parameters for multi-solids fluidized bed combustor are shown in Table 5.1.2 and a diagram illustrating its mode of operation for a steam boiler heat exchanger is shown in Fig. 5.1.7. The multi-solids bed combustion takes place in a bed of dense inert material. There is no heating surface present in that bed to absorb the heat of combustion. The heat of combustion is absorbed in the dense bed by circulating through it an entrained flow of lighter solid material, which is carried through the dense bed where it absorbs heat and then through the banks of heating surface above the dense bed where it gives off its heat to the surface. With the mode of operation, it is possible to provide much higher superficial gas velocities and therefore smaller beds than is the case with a conventional concept. Additionally, it has been found experimentally that adequate sulphur removal can be obtained in the multi-solids bed at temperatures higher than those normally possible with the more conventional beds. However, a screening analysis of heating surface requirements for pulverized coal, conventional atmospheric fluidized bed, and multi-solids bed concepts for CCGT service showed that the heating surface requirements for the multi-solids bed concept is substantially greater than that required for the others. The combination of a lower available heat transfer coefficient for the heating surface exposed to the entrained recirculating bed material, as compared to the heat transfer coefficient for the "in-bed" surface of a conventional fluidized bed, plus the lower log mean temperature difference available with the multi-solids fluidized bed concept,

TABLE 5.1.2

MULTISOLIDS FLUIDIZED-BED COMBUSTOR DESIGN PARAMETERS

• SUPERFICIAL VELOCITY	32 FT/SEC
• STATIC BED DEPTH	18 INCHES
• EXPANDED BED DEPTH	48 INCHES
• DENSE BED MATERIAL	SPECULITE (HEMATITE), -8 + 16 MESH
• ENTRAINED BED MATERIAL	ROUNDED SILICA SAND, -20 + 70 MESH
• COAL FEED	MINUS 1/4 INCH
• LIMESTONE FEED	MINUS 325 MESH
• EXCESS AIR	15%
• COMBUSTION EFFICIENCY WITH FINES RECYCLE	96%
• BED TEMPERATURE	
BASELINE	1650 F
ALTERNATE	1750 F
• Ca/S MOLE RATIO	
BASELINE	2.0
ALTERNATE	4.0
• ENTRAINED BED HEAT TRANSFER h	25BTU/HR FT^2 ^{o}F
• MAX BED SIZE	12' X 30'

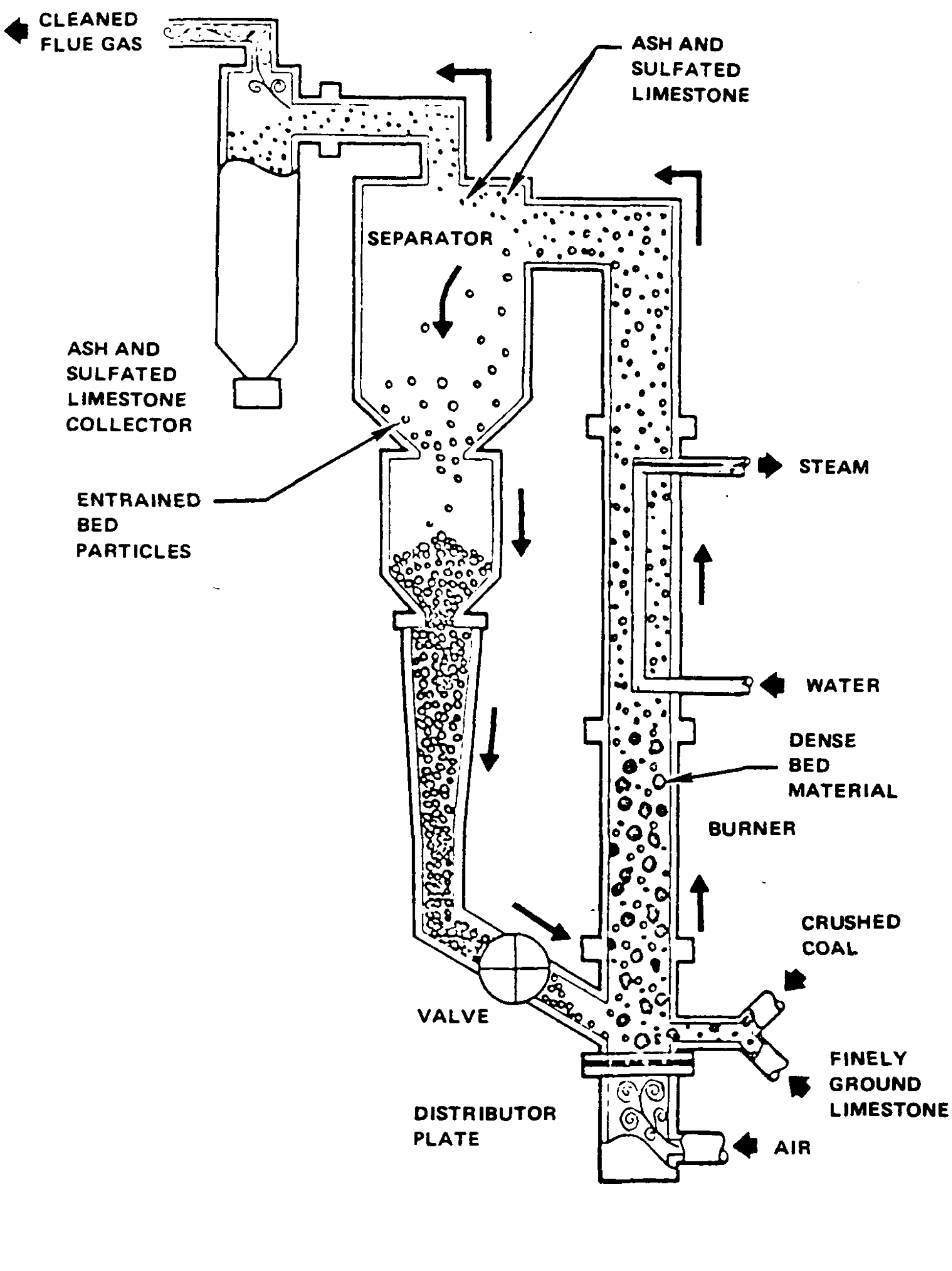


Figure 5.1.7 Battelle Multisolid Fluidized Bed Integral Boiler

led to an inherent disadvantage for this concept with respect to total surface when it is applied to the requirements of the advanced heater. Because of these disadvantages for the multi-solids bed concept in this particular application, it was decided to eliminate it from the competition, i.e., not conduct a preliminary design on its concept.

The screening estimates of total surface requirements for the conventional fluidized bed operating at 1650 F indicated that its total surface would roughly be comparable to that of a dry bottom pulverized coal unit. This comes about because the gas side heat transfer coefficient in the fluidized bed is approximately 40, which offsets the low log mean temperature difference occasioned by the relatively low bed temperature.

On the basis of the economic advantages foreseen for fluidized bed combustion due to its eliminating the need for an add-on sulphur scrubber, it was decided that one of the preliminary designs to be carried out for evaluation of design concepts, key features, and costs would be a 1650 F fluidized bed combustor concept.

5.1.4 Slagging Combustors

The slagging or wet bottom combustor, either pulverized coal or cyclone fired, was discussed in Section 4.1.1.5 as regards its design criteria. There appeared to be no technical reasons to prevent the application of this slagging combustor concept to the 1550 F CCGT systems, however, slagging combustors as discussed in Section 4.1.1.5, are inherently limited in the range of coals that they can satisfactorily handle. Coal ashes with too high a melting point will not provide satisfactory performance. Because both dry bottom pulverized coal and atmospheric fluidized combustion gave promise of satisfactory operation with the full range of American coals it was decided that slagging combustors would not be carried through the preliminary design stage for the 1550 F operating condition.

5.2 CONCEPTUAL DESIGNS OF COMBUSTOR/HEAT EXCHANGERS FOR 1750 F WORKING TEMPERATURE

At 1750 F the choices among available combustion concepts become more constricted. The outside surface temperatures of the heat exchanger surfaces may be expected to operate in the neighborhood of 1850 F. For fluidized bed combustion, the bed temperature would have to be in the neighborhood of 1950 F to 2000 F, i.e., above the temperature at which sulphur recovery appears to be economically practical. With dry bottom pulverized coal units, the 1850 F surface temperature makes for an uncomfortably small differential between surface temperature and slagging temperature of Illinois No. 6 coal, which implies a probably excessive schedule of soot-blowing inorder to maintain the surfaces in a reasonably clean condition. The concepts evaluated and the rationale pursued in selecting the preliminary design for evaluation at 1750 F working fluid temperature under these increasingly constrained conditions are explained below.

5.2.1 Dry Bottom Pulverized Coal Radiant Furnace Construction

On evaluation of the projected operating conditions in a combustor/heat exchanger design for 1750 F working fluid temperature and a lignite or Illinois No. 6 coal firing, the judgment was made that this operating condition is probably too close to marginal to be comfortable. Additionally, the dry bottom pulverized coal concept had been previously selected for the 1550 F operating condition and it was felt that that preliminary design for that working fluid temperature would be adequate to represent the dry bottom pulverized coal concept. The 1550 F design concept could be modified to handle the 1750 F condition when a coal of suitable ash slagging characteristics was to be fired. In the light of these considerations, it was decided against providing a preliminary design of the 1750 F working fluid case utilizing the dry bottom pulverized coal combustion concept.

5.2.2 Fluidized Bed Combustor Concepts

As mentioned previously, it is evident that at 1750 F working fluid temperature at least a portion of the heat supplied to the working fluid must come from a bed or other source operating at a temperature on the order of 1950 F to 2000 F, i.e., well above the temperature at which it is expected a significant recovery of sulphur can be obtained by the use of limestone in the fluidized bed. It is possible however to break up the heat input process into several steps, i.e., the high temperature working fluid could be heated by a high temperature fluidized bed, or a slagging type combustor/heat exchanger, or a dry bottom combustor/heat exchanger. It is also possible to visualize the high temperature portion of the working fluid heat addition being supplied by a clean gaseous or liquid fuel combustor/heat exchanger. The low temperature heat absorption portion is provided by a fluidized bed. The heat exchanger design studied by the General Electric Co. during ECAS II was of the nature of a high and low temperature bed in series, and is an example of this concept, which we have labeled "mixed mode".

With the mixed mode concept, a part of the combustion takes place at a sufficiently high temperature that an effective heat exchanger arrangement can be configured to attain the design maximum working fluid temperature. The flue gases from this high temperature combustor may then be routed through a fluidized bed operating at a temperature favorable for sulphur oxides absorption, i.e, in the neighborhood of 1450 F to 1650 F, where some of the coal is combusted along with additional air, and where sulphur oxides contained in the gases from the high temperature combustor are absorbed. Alternatively the high temperature combustor gases may be separately discharged. There are a great many variations in configuration that may be constructed about this concept. One can easily visualize wet bottom pulverized coal or cyclone fired units for the high temperature heat input, high temperature fluidized beds, or coal derived clean fuel.

In evaluating the choices among these concepts for the 1750 F maximum working fluid temperature cycle, it quickly becomes apparent that a parallel flow arrangement, i.e., one in which the flue gases from the high temperature combustor do not pass through the low temperature fluidized bed combustor, is impractical. Sufficient sulphur oxides recovery would not be possible with the high sulphur coals to meet the present EPA limitations, and additionally, it would probably fail to meet the future limitation which will require SOX reduction via the best available technology.

The slagging combustor concept for the high temperature combustor was rejected because its utilization would require the coordinated operation of two very different combustion technologies for the same unit.

In plant production of coal derived clean fuel for the high temperature combustor was rejected for the same reasons as the slagging combustor, and for the economic reasons discussed in Section 4.1.1.4.

A concept similar to the series bed arrangement of GE's ECAS Phase II is attractive for its simplicity in directly feeding flue gases from the high temperature combustor to the SOX recovery bed, but that system has several very serious practical drawbacks. It will be impossible to operate a fan between the two combustors to supply the windbox pressure required by the SOX absorbing fluidized bed, the temperature is too high and the gas too dirty. A jet pumping concept for this job is theoretically available but the pumping power penalty is much too high. Thus, the only "practical" concept is to operate the high temperature combustor under pressurized conditions, and this entails severe economic and operating penalties.

Considering the penalties associated with high temperatures entering the windbox of the sulphur oxides absorbing fluidized bed, other concepts were evaluated in which the gas temperature leaving the high temperature combustor is reduced to about 800 F prior to being mixed with the combustion air and admitted to the sulphur oxides absorbing bed. Fan conditions at 800 F are expected to be severe but within present state-of-the-art, and both high and low temperature combustors could operate at atmospheric pressure. The proportioning of heat absorptions by high and low temperature combustors operating under these conditions is illustrated in Fig. 5.2.1.

After evaluating the combustor/heat exchanger concepts available for application to the selected 1750 F working fluid temperature of Table 3.1.4 , it was decided to evaluate a mixed mode concept consisting of a high temperature fluidized bed operating at 2000 F which exhausts its flue gases through a sulphur oxides absorbing fluidized bed operating at 1650 F. This concept has these advantages; the system involves only one combustion technology, fluidized beds; the system could logically grow out of fluidized bed operations for 1550 F or lower working fluid temperatures; and one can visualize the system's technical feasibility.

5.3 2250 F MAXIMUM WORKING FLUID TEMPERATURE CCGT HEAT EXCHANGER CONCEPTUAL DESIGNS

The heat transfer relationships applying to the selected 2250 F cycle of Table 3.1.6 are shown in Fig. 5.3.1 . With this working fluid temperature, the combustion concepts available for application are much narrowed as compared to previous cycles. The heating surfaces will operate at maximum temperatures up to 2300 F or 2400 F. About one-third of the heat transferred to the working fluid occurs at working fluid temperatures higher than 1550 F. A dry bottom pulverized coal fired concept is obviously impractical for all but the very highest ash fusion temperature coals, and perhaps not for those. A mixed mode system involving a slagging combustor/heat exchanger concept operating in series with a 1650 F sulphur oxides absorbing fluidized bed is theoretically possible but due to the large amount of heat transfer at high working fluid temperature requires combusting about 50% of the coal in the high temperature

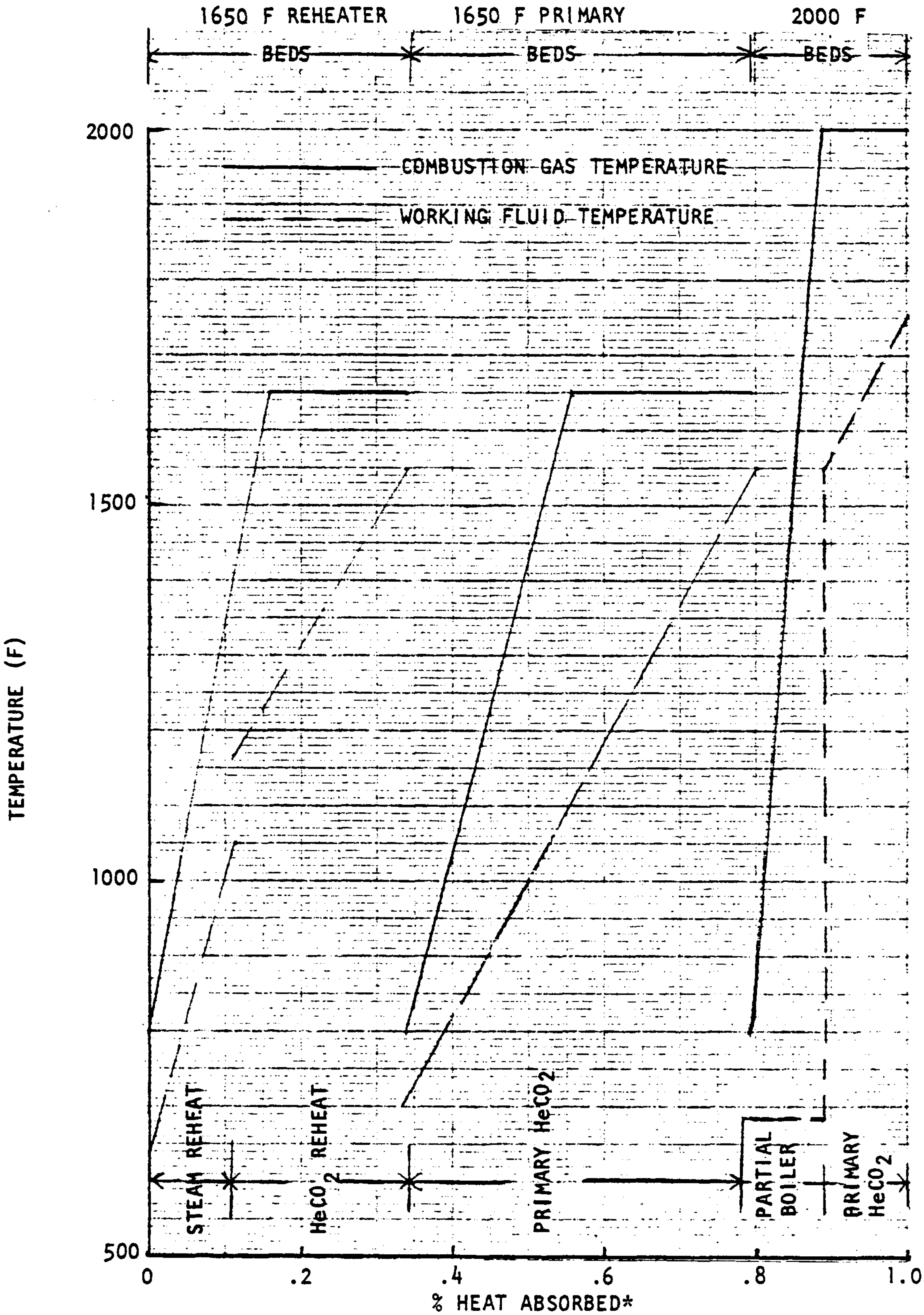


Figure 5.2.1. Heat Transfer Characteristics For 1750 F/1550 F Mixed Mode System

* Excludes Carbon Burnup Cells

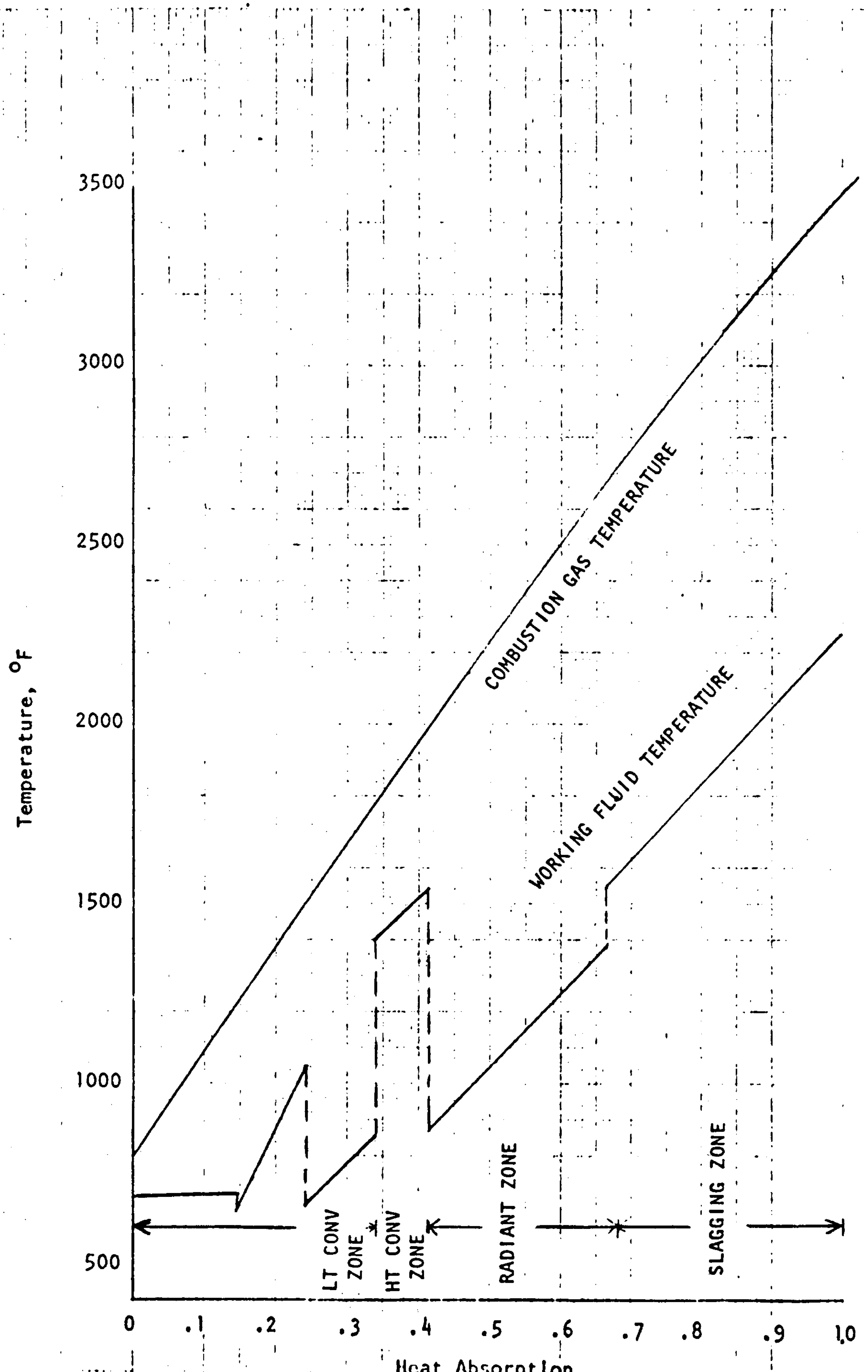


Figure 5.3.1 Heat Transfer Characteristics for 2250 F Cycle

combustor and then providing the equivalent of a full-size fluidized bed installation in the colder beds. Such an arrangement appears economically handicapped. The slagging combustor concept, on the other hand, appears capable of coping with the heat absorption conditions required by this cycle. Such a concept might be arranged to operate for instance with running slag at all times on the really high temperature ceramic heat absorbing surfaces and dry ash on the lower temperature metal surfaces.

Several slagging combustor concepts were evaluated for application to the 1750 F cycle. Several novel slagging combustor concepts involving the patented Rocketdyne Variflux tube and illustrated in Figs. 5.3.2 and 5.3.3 were available. Additionally, the pulverized coal or the cyclone fired slagging combustor/heat exchanger concepts as available from steam boiler practice and illustrated in Fig. 5.3.4 were evaluated. As discussed in Section 4.7.1, the design constraint was adopted that all combustion gas containment surfaces be constructed of metal to prevent escape of hot combustion gases and/or fragments of ceramic heat exchanger in the event of ceramic heat exchanger material failure. On the basis of these considerations the slagging cyclone combustor/heat exchanger was selected for preliminary design evaluation. It has already demonstrated its economic superiority in the marketplace over the slagging pulverized coal combustor/heat exchanger concept.

CONCENTRIC-TUBE ELEMENT HEAT EXCHANGER

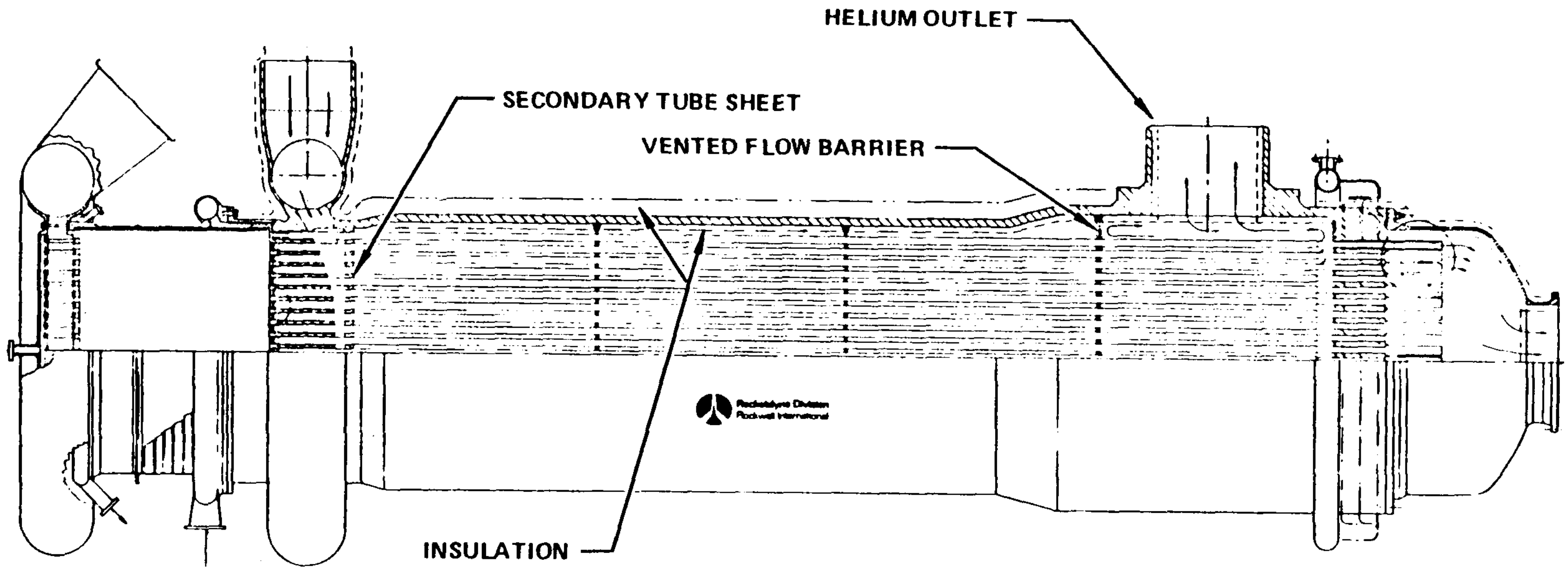


Figure 5.3.2

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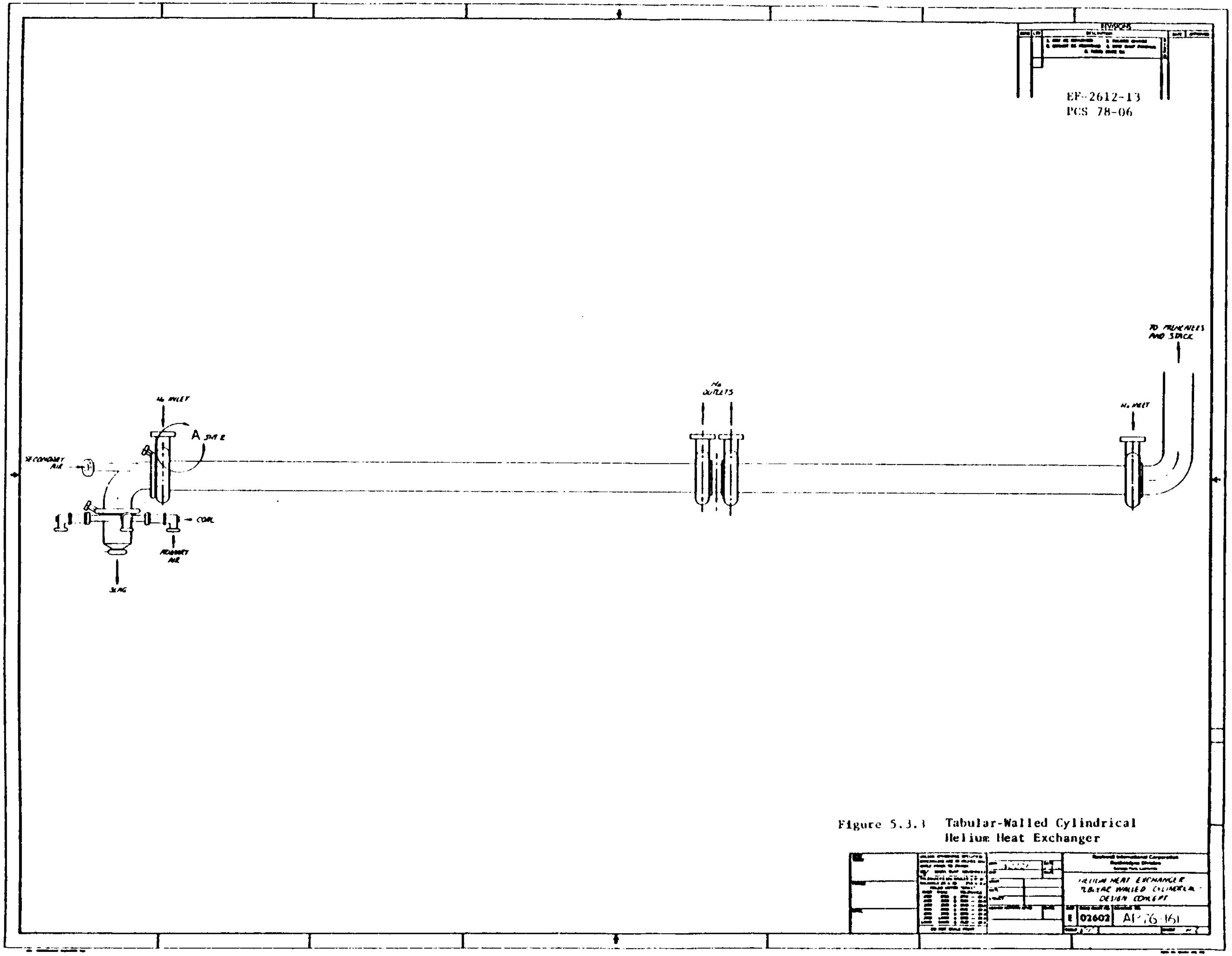


Figure 5.3.3 Tabular-Walled Cylindrical Helium Heat Exchanger

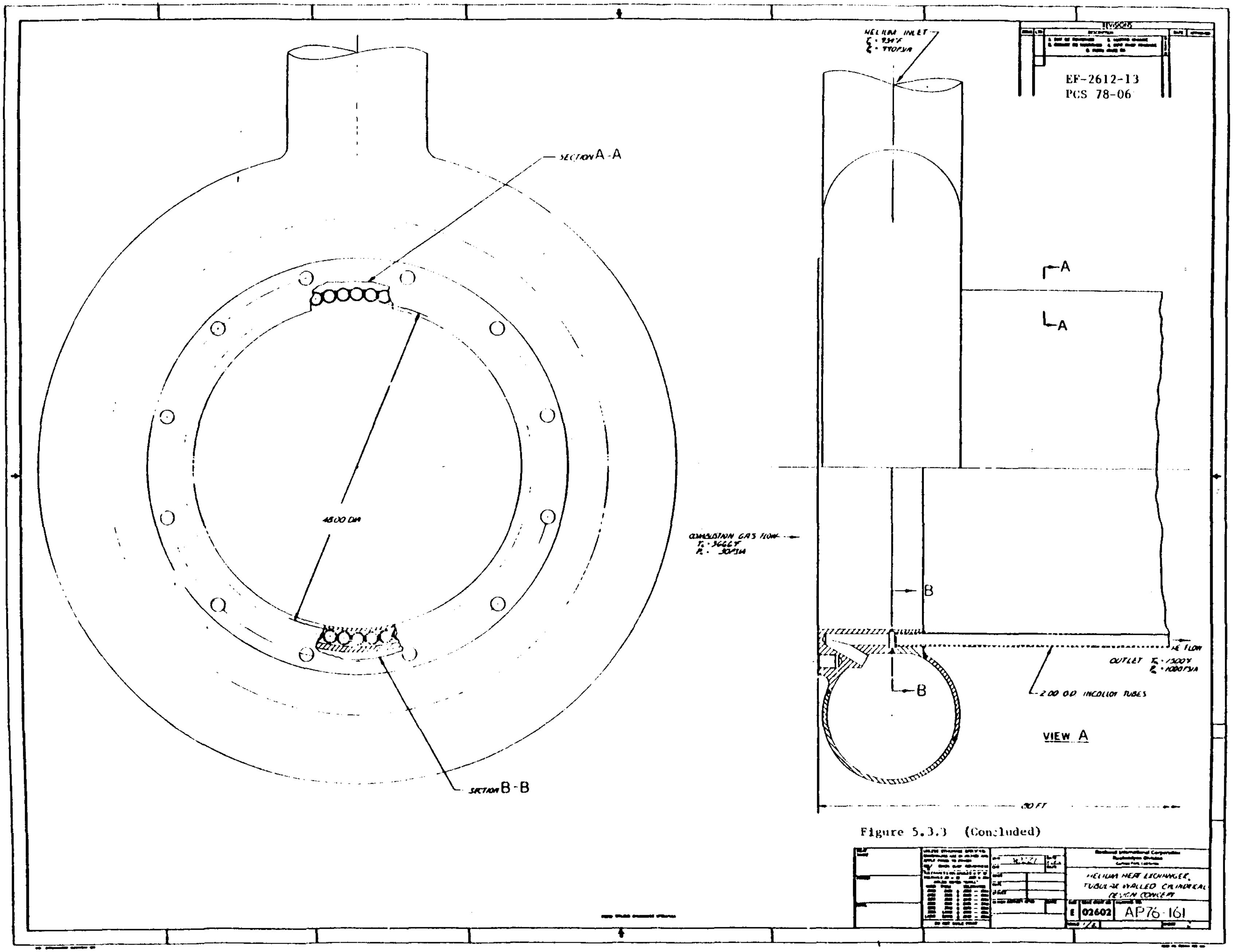


Figure 5.3.3 (Concluded)

RADIANT BOILER

CYCLONES

SLAG TAP

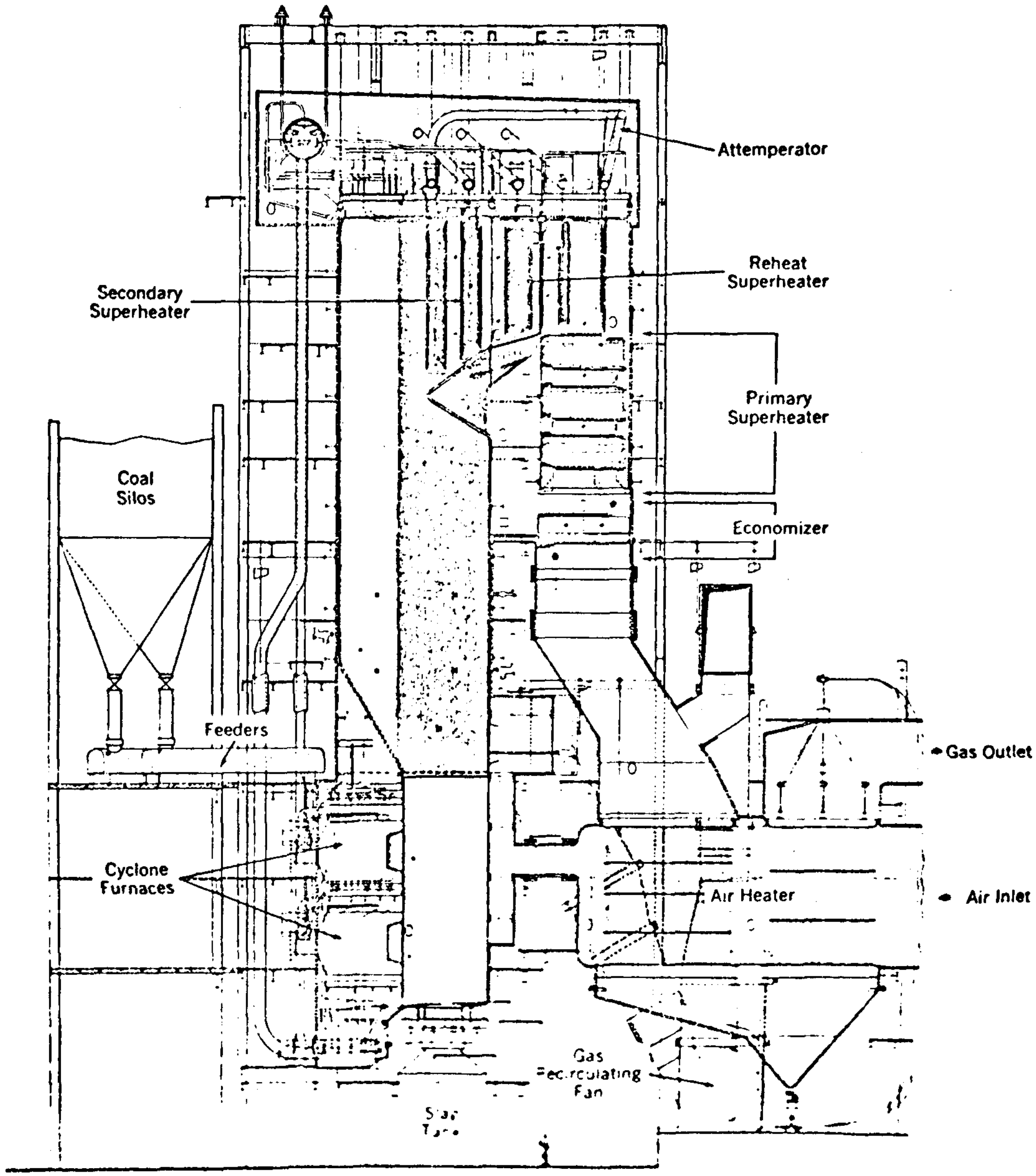


Figure 5.3.4

5.4 COMBUSTOR/HEAT EXCHANGER CONCEPTS SELECTED FOR PRELIMINARY DESIGN

Four combustor/heat exchanger concepts were chosen for preliminary design and analysis. The concepts are listed in Table 5.4.1. along with their cycles and expected ranges of application.

Table 5.4.1. Combustor/Heat Exchanger Concepts Selected For Preliminary Design

Concept	Cycle Application in Study	Expected Range of Application
"Dry Bottom" Pulverized Coal	1550F/1550F, steam bottomed, helium working fluid	1300F to 1750F, helium or helium/CO_2, air with some difficulty
Atmospheric Fluidized Bed	1550F/1550F, steam bottomed, helium/CO_2 working fluid	1300F to 1550F, helium, helium/CO_2, or air
Mix Mode Atmospheric Fluidized Beds, (Series Beds)	1750F/1550F, steam bottomed, helium/CO_2 working fluid	1550F to 1750F, helium, helium/CO_2, or air
Slagging Cyclone	2250F, steam bottomed, helium working fluid	1750F-2250F, helium or helium/CO_2, air with some difficulty

6.0 PRELIMINARY DESIGNS

Previous sections of this report have included Section 3, which presented "reference design conditions", i.e., cycle conditions, flowrates, pressures, temperatures, and other operating conditions which were to be satisfied by the coal fired combustion heat exchanger. Section 4 discussed the design criteria and constraints that apply to large coal fired combustor/heat exchangers of the general nature involved in this project. Section 5 discussed the conceptual combustor/heat exchanger designs that were considered in arriving at a decision as to which concepts would be carried through the preliminary design stage. Section 5 also discusses the reasons for the selection of the particular concepts for preliminary design. This Section 6 of the report describes the four coal fired combustor/heat exchanger preliminary designs that were created during the study.

In creating the preliminary design for each selected concept the required heating surfaces were calculated and proportioned to meet the heat transfer and working fluid pressure drop requirements while at the same time satisfying the design constraints enumerated in Section 4. The physical arrangement of coal and air supplies, combustion equipment, heat transfer surfaces, ash disposal equipment, etc., were designed to be consistent with operational requirements and design constraints. A layout drawing was prepared for each concept to define the arrangement. A primary objective of this preliminary design exercise was to permit the analysis of the designs, to identify those "key features" which present problems or questions so that an appropriate R&D program could be prepared. Additional objectives included the visualization of the design concept as applied to the CCGT heater service so as to be able to critique its probability of providing satisfactory results, both economic and technical. Additionally, the preliminary designs provided the basis for an economic analysis of the 350 MWe CCGT systems and the cost of electricity. Thus, the preliminary designs represent an attempt to identify and resolve the major challenges in design, fabrication, economics, and operation of the combustor/heat exchangers.

Many of the design details of steam boiler construction can be adapted to the CCGT combustor/heat exchanger with considerable confidence. However, there are many other design details whose service in the CCGT system is new, and will require the creation of novel design details configurations for satisfactory behavior. These will need thorough examination at a later date, as the preliminary designs are necessarily "broad brush".

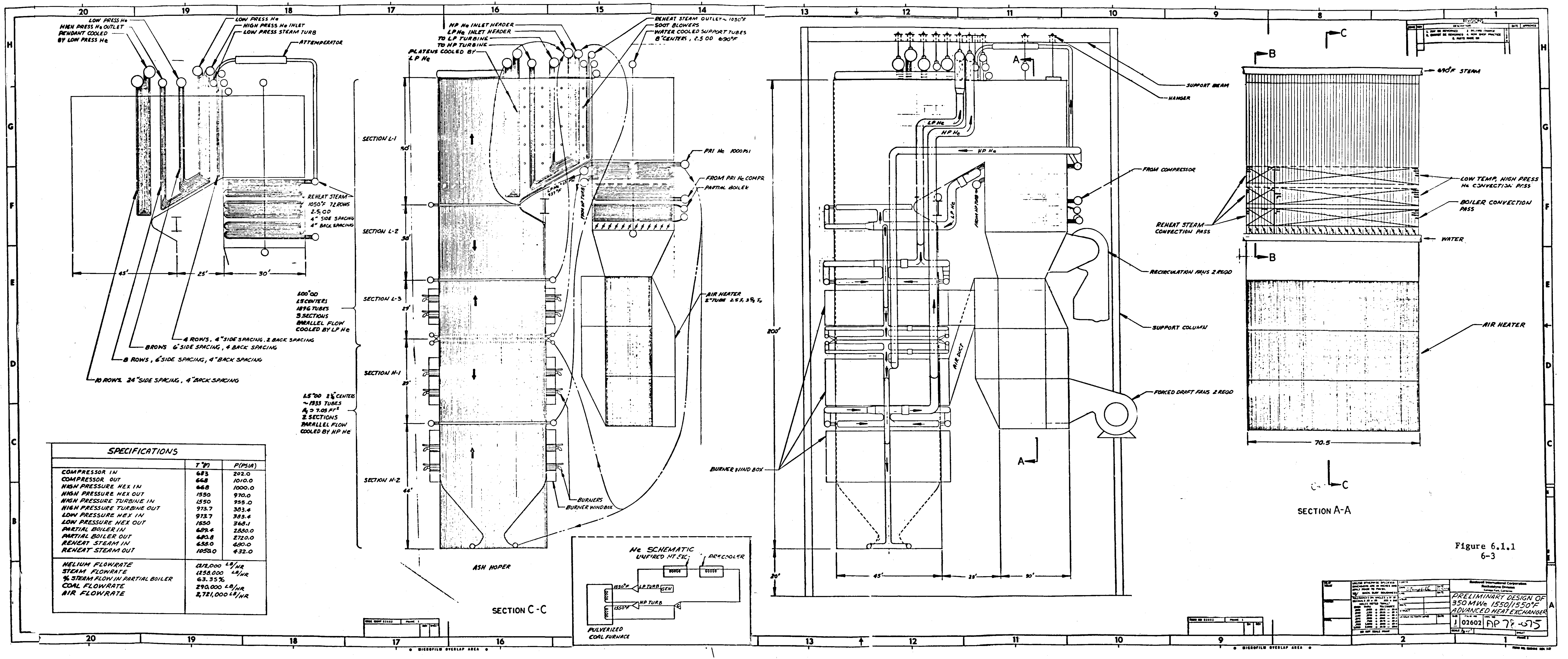
Each of the four preliminary designs created during the course of the contract is separately described and discussed below.

6.1 PRELIMINARY DESIGN OF PULVERIZED COAL FIRED DRY BOTTOM FURNACE COMBUSTOR/HEAT EXCHANGER FOR 1550 F MAXIMUM WORKING TEMPERATURE

This preliminary design embodies concepts and configurations that are closest to the existing steam boiler state-of-the-art. The concept is the "dry bottom radiant furnace" in which it is endeavored to maintain combustion and heat transfer conditions such that no running slag accumulates anywhere on the combustor/heat exchanger surface. The concept is described in Section 5.1.2. The design operating condition for this preliminary design are given in Section 3. The design constraints which tend to affect the configuration are described in Section 4 .

A drawing of the preliminary design is presented in Fig. 6.1.1 , and an artist's isometric rendition in Fig. 6.1.2.

The preliminary design represents a 350 MWe capacity combustor/heat exchanger embodied in a single unit. The single 350 MWe unit incorporates all of the different heater functions of high pressure helium, low pressure helium, steam



SPECIFICATIONS

	T °(F)	P(PSIA)
COMPRESSOR IN	683	202.0
COMPRESSOR OUT	668	1010.0
HIGH PRESSURE HEX IN	668	1000.0
HIGH PRESSURE HEX OUT	1550	970.0
HIGH PRESSURE TURBINE IN	1550	955.0
HIGH PRESSURE TURBINE OUT	973.7	383.4
LOW PRESSURE HEX IN	973.7	383.4
LOW PRESSURE HEX OUT	1550	368.1
PARTIAL BOILER IN	689.4	2880.0
PARTIAL BOILER OUT	680.8	2720.0
REHEAT STEAM IN	638.0	480.0
REHEAT STEAM OUT	1050.0	432.0
HELIUM FLOWRATE	1212,000 LB/HR	
STEAM FLOWRATE	1258.000 LB/HR	
% STEAM FLOW IN PARTIAL BOILER	63.35%	
COAL FLOWRATE	290,000 LB/HR	
AIR FLOWRATE	2,721,000 LB/HR	

Figure 6.1.1
6-3

350 MWe PULVERIZED COAL FIRED HELIUM HEATER

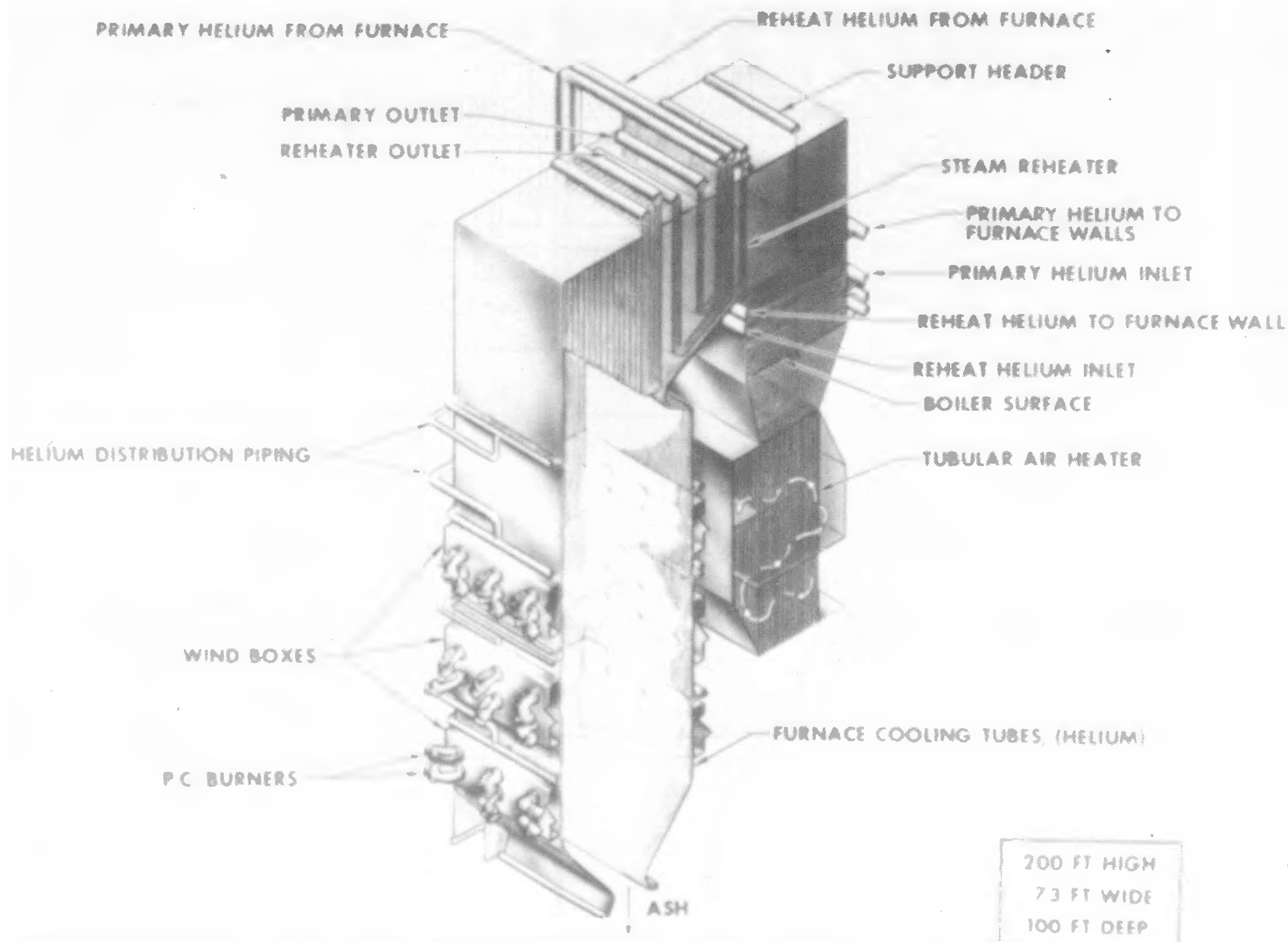


Figure 6.1.2

boiling and steam reheat. The unit is approximately 200 ft high, 75 ft wide and 100 ft deep. A single hopper bottom radiant furnace is supplied and proportioned to cool the products of combustion sufficiently that slagging and pluggage will not occur in the convection surfaces. Conventional dry bottom pulverized coal steam boiler design practices have been followed insofar as practical in the design of the combustion system and gas side heat transfer and flow arrangements. A more detailed description of the preliminary design is given by category below.

6.1.1 Combustion System

The combustion system shown consists of circular burners arranged in both front and rear walls of the furnace and covering a substantional portion of the height of the furnace. This arrangement is consistent with modern steam boiler design practice with circular burners, and with the CCGT combustor/heat exchanger objective of spreading out the flame for uniformity of heat absorption and minimization of hot areas. It also provides for burner adjustments for temperature control. The burners and pulverizers and coal handling equipment for this installation would be the same as used in steam boiler practice. While circular burners are shown, the dry bottom pulverized coal fired concept is probably adaptable as well to corner firing.

6.1.2 Heat Transfer and Pressure Drop Considerations

In most combustor/heat exchangers of this nature the major influence upon the quantity and arrangement of heating surface is exercised by the design criteria applying to the combustion gas side of the heat exchanger surface, i.e., available heat transfer coefficients, erosion, etc. The working fluid side available delta P's and heat transfer coefficients influence the arrangement of the circuits, may have some influence on tube diameters, and strongly influence metal temperatures and thereby material utilization. In the preliminary design of this pulverized coal fired combustor/heat exchanger, it was found that while the gas side heat transfer conditions exercised by far the major influence on the overall heat transfer coefficients available, and thereby the amount of surface necessary, the cooling capability of the CCGT working fluid was limited by the available working fluid pressure drops, (see Section 3.1). A completely conventional steam boiler type radiant furnace design appears to subject the wall cooling tubes to radiant heat fluxes higher than can be reasonably handled by the cooling capability of the CCGT working fluid. The design solution employed was to provide for somewhat greater exhaust gas recirculation from the economizer exit to the furnace than is normally provided in steam boiler practice. In this instance, provision is made to recirculate as much as 50% of the flue gas throughput from the economizer exit to the furnace hopper apperture and/or to recirculating ports provided between and above the burner windboxes. The effect of such exhaust gas recirculation is to reduce the average radiant heat flux in the furnace to a value that can be handled by the CCGT working fluid without encountering excessive metal temperatures, and at the same time to provide a much more uniform heat flux throughout the furnace than would otherwise be the case. The furnace surface provided is based upon performance information contained in "Steam". The furnace exit temperature is based upon the curve of Fig. 6.1.3 and is little affected by the amount of flue gas recirculation employed. A typical relationship between furnace exit temperature and percent gas recirculation is shown in Fig. 6.1.4, also adapted from "Steam". Figure 6.1.5 then indicates the expected variation of average heat flux with percent of recirculation and distribution of heat flux as affected by flue gas recirculation. Sufficient surface has been provided in the furnace walls and in the 24-inch side spacing pendant platens at the furnace exit to provide a furnace exit gas temperature of approximately 2100 F at full load. This temperature is believed to be low enough to avoid excessive slagging accumulations in the more closely spaced convection banks that follow the furnace exit.

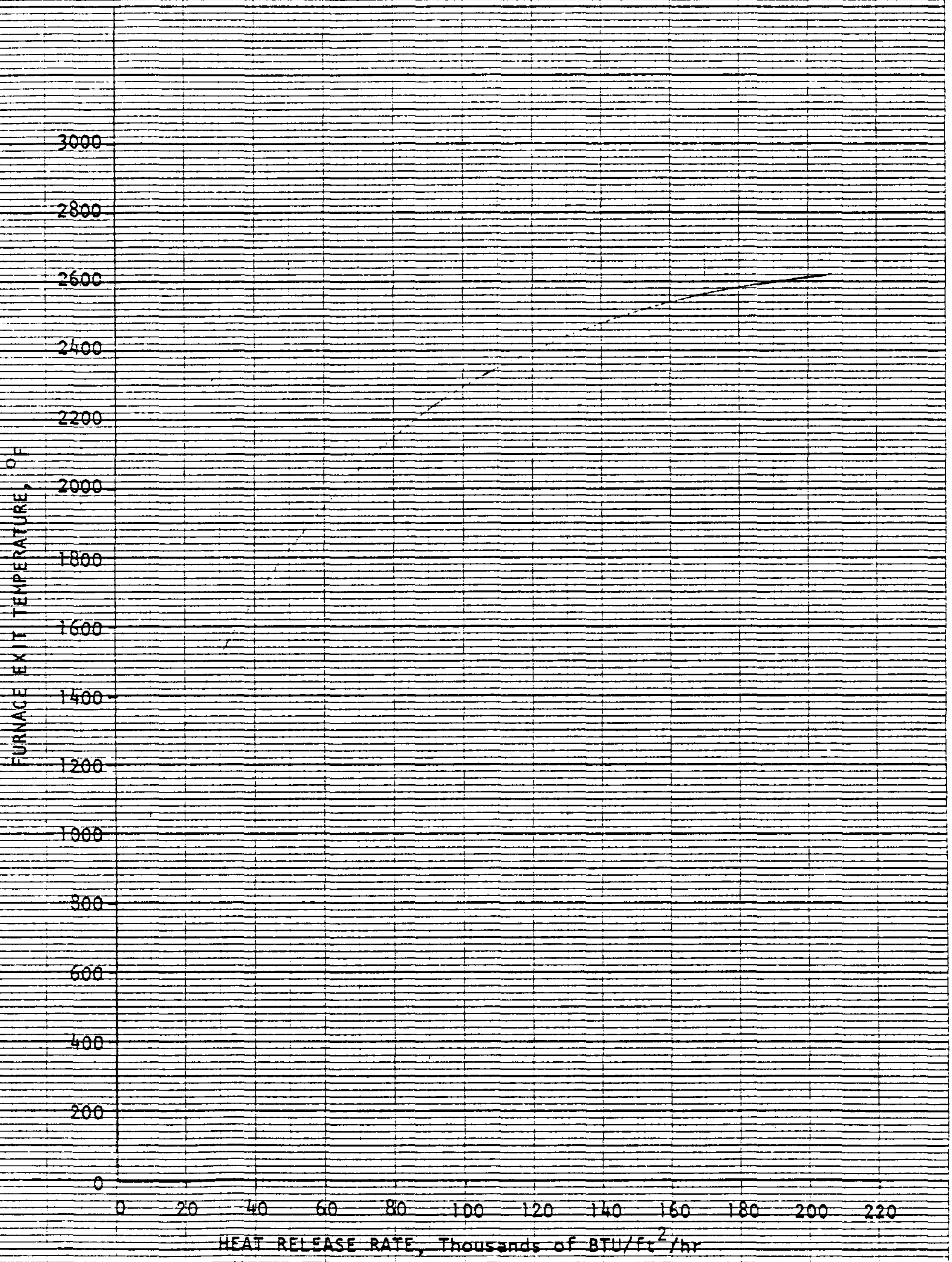


Figure 6.1.3 Dry Bottom PC Furnace Exit Temperature

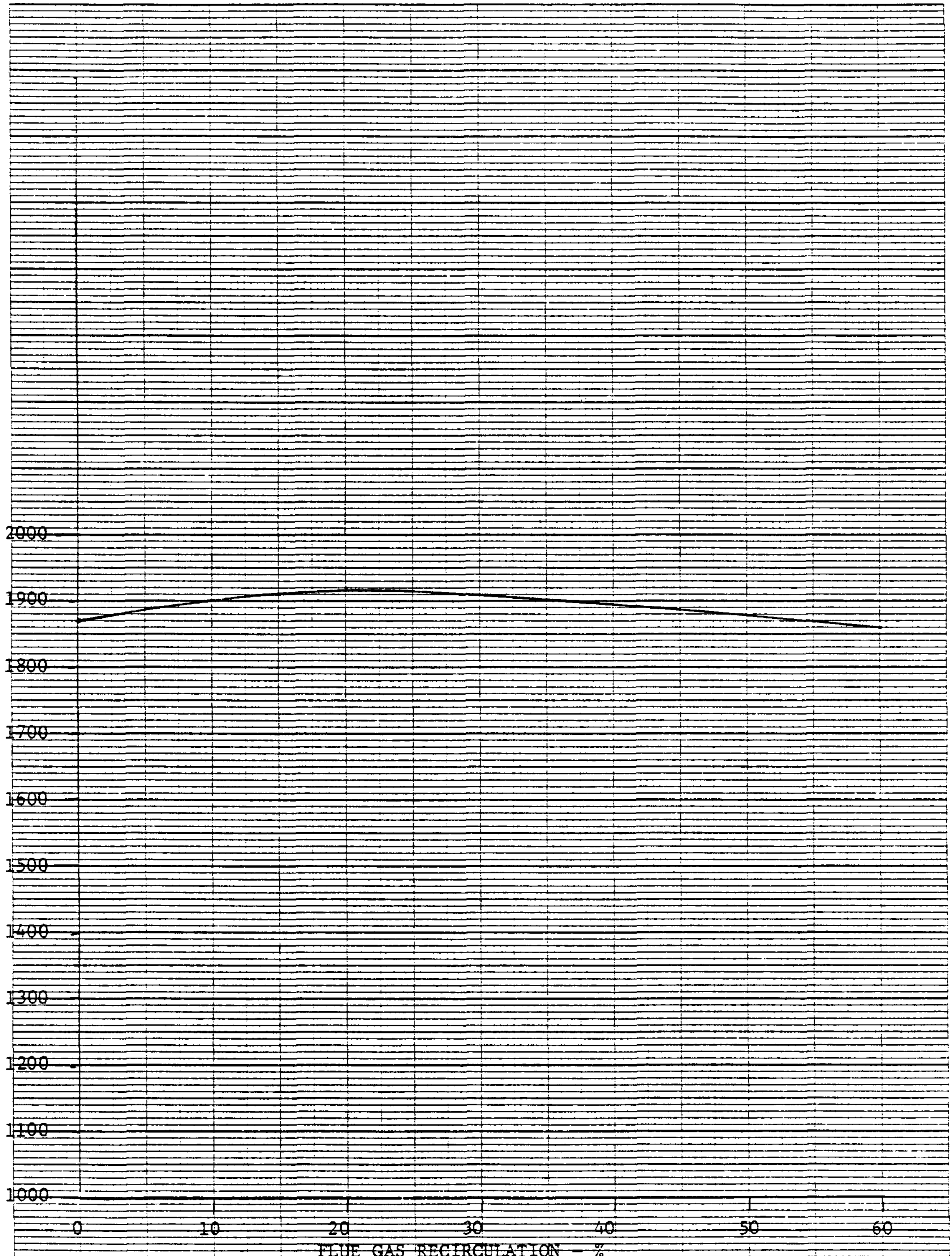
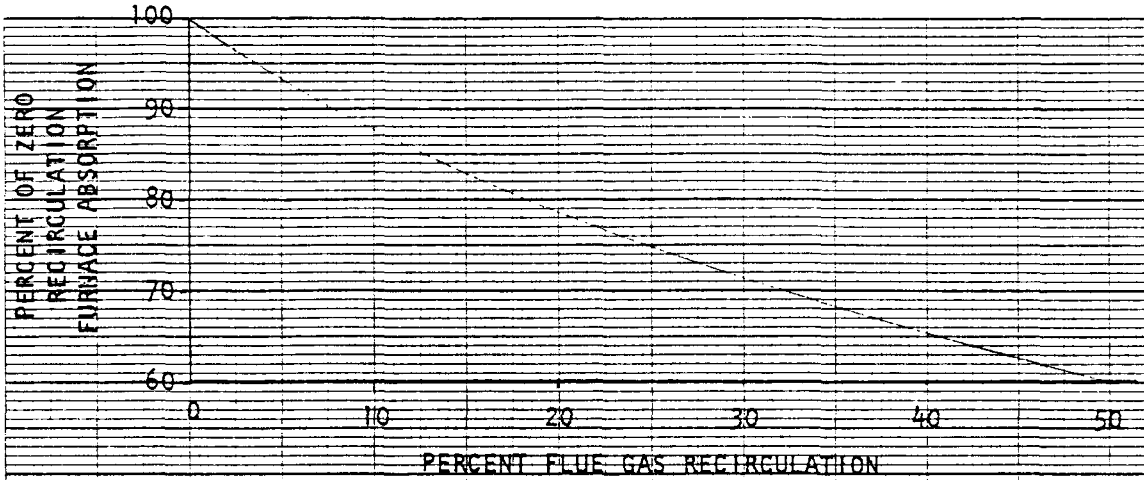


Figure 6.1.4. Dry Bottom P.C. Furnace Exit Temperature, °F
vs % Flue Gas Recirculation



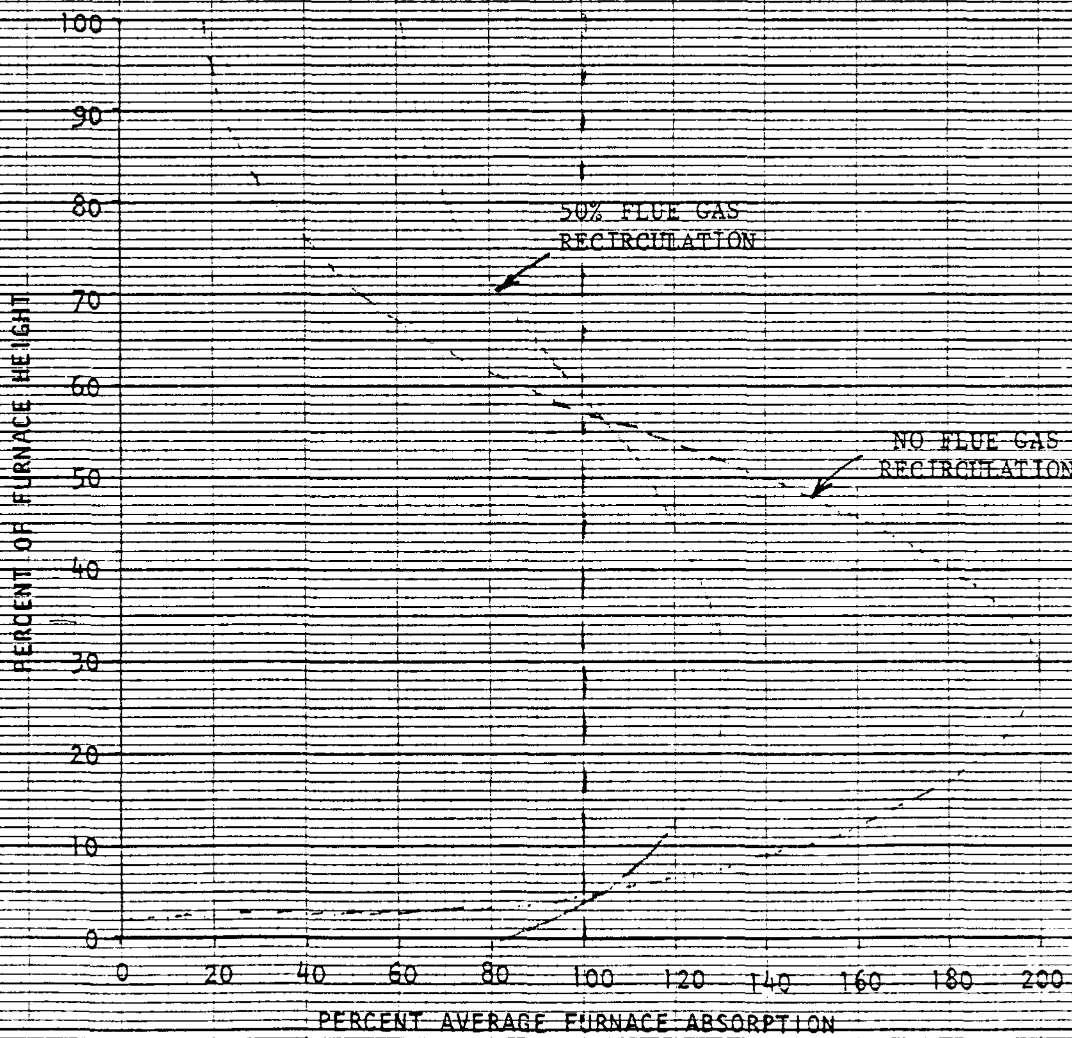


Figure 6.1.5 Flue Gas Recirculation Effects

The influence of working fluid characteristics upon furnace design is further felt in the circuitry provided for cooling the furnice walls. In steam boiler practice these walls are normally cooled with boiling water, and one long unbroken circuit is employed from bottom to top of the furnace. With the CCGT working fluid flow rates and temperature conditions available for the reference design, it is not practical to cool these walls with water or steam. The closed cycle gas turbine working fluid must be employed for cooling, and that imposes limitations on pressure drop and circuit arrangement. In the preliminary design of Fig. 6.1.1 the lowest temperature CCGT working fluids have been employed to cool the heating surface near the exit end of the heat exchanger train, so as to provide the desired 800 F gas temperature entering the air heater. The highest temperature CCGT working fluid would suffice to absorb the radiant heat in the furnace, but it is expected that there will be relatively large differences in the absorption from tube to tube in the furnace and this would lead to excessive temperatures of individual tubes if the furnace cooling constituted the last heat exchanger pass. As a result the circuitry employed in the preliminary design is to cool the furnace with intermediate temperature working fluid, designing for a working fluid exit temperature from the furnace circuits of approximately 1450 F. The 1450 F working fluid from the furnace circuits is then mixed and transported to the pendant convection surface just downstream of the furnace exit, where the final increment of heat is added to raise both the primary and the reheat CCGT flows to the 1550 F design exit temperature.

The furnace height is so great that it is not feasible, working fluid pressure drop-wise, to cool the walls with tubes that run from bottom to top in one uninterrupted circuit. With reasonably sized tubes, on the order of 1-1/2 inch O.D., the L/D would produce much too much pressure drop. While the tubes might in theory be made large enough to provide an acceptable L/D, their diameter would be so great that the wall thickness would be unacceptable. The design solution employed in the preliminary design is to arrange the furnace wall cooling circuitry as parallel circuits of both primary and reheat helium. The furnace wall is divided into two halves with approximately equal total heat absorption. The top half is devoted to the low pressure helium circuit, having three parallel flows, sections L1, L2, and L3 plus a fourth parallel circuit which consists of the widely spaced pendant platens at the furnace exit. The bottom portion of the furnace is divided into two circuits for the high pressure helium. The manifolds and interconnecting piping are complicated by this arrangement but the CCGT working fluid pressure losses remain within the cycle allowances of Table 3.1.1. For convenience of manifolding two of the cirucits in the furnace are arranged for down flow of the working fluid. Down flow circuits have given difficulty with stagnation of flow at low loads on some separately fired steam superheaters, but it is believed they will provide satisfactory performance in this instance because the method of control for the CCGT system will be inventory control, which will maintain relatively high pressure drops through the circuit at low loads. In any event, the down flow circuits could be converted to up flow for a relatively modest cost in additional manifolding. The multiple parallel circuit furnace cooling concept may be viewed as having six separate combustor/heat exchangers on the working fluid side, which are joined together on the combustion gas side for economic and operational reasons.

Pendant convection surface between the furnace exit and the turn into the down flow convection surface is provided with three banks of 2-1/2 inch O.D. tubes on 6 inch side spacing and a final bank of 2-1/2 inch tubes on 4 inch side spacing. The combination of wide side spacing and adequate provisions for soot-blowers is believed to be sufficient to avoid difficulties with fouling accumulations, i.e., plugging of the pendant sections. The pendant sections are designed to contain sufficient surface to reduce the gas temperature entering the horizontal convection surface to approximately 1500 F.

The horizontal convection banks are constructed of 2-1/2 inch O.D. tubes on 3-3/4 inch side spacing. The flue gas velocity is kept below 75 ft per second at all times. The horizontal banks consist of primary and reheat CCGT working fluid surfaces, steam reheater surface, and steam boiler surface. For purposes of steam and CCGT working fluid temperature control the down pass is divided into three separate sections, and biasing of gas flow over each section is possible by positioning the dampers shown at the horizontal convection pass exit. The control concepts are explained below under Section 6.1.3.

The preliminary design of Fig. 6.1.1 has been sized for 99% helium/1% oxygen working fluid because of the superior properties of that fluid, and its better cooling at the allowable delta P. It is probable that the design could be easily adapted to the 60% helium, 39% CO_2, 1% oxygen synergistic mixture which provides about 80% of the cooling effect of the pure helium while at the same time permitting operation at a greater pressure ratio and about 1 point higher cycle efficiency. The cooling capability of air or nitrogen working fluids, however, is so far below that of helium that the design would need substantial modification for those fluids. The metals of the furnace cooling circuits would tend to operate at much hgiher working fluid side film temperature drops, with additional hazard of tube overheating and burnout. It is believed that the design of Fig. 6.1.1 could be adapted to air or nitrogen working fluid, but the circuitry would need modification to supply the furance walls with cooler fluid, and the convection banks surface would increase.

A tubular air heater to provide the final temperature drop from 800 F to 300 F is shown on Fig. 6.1.1. A regenerative type air heater, along the lines of the Ljungstom concept, would serve equally well.

A summary of the heat exchanger surface size, quantity and material requirements is presented on Fig. 6.1.1 and Table 6.1.2.1. A summary of the working fluid side pressure drop allocations is presented in Table 6.1.2.2.

6.1.3 Control Considerations

The control requirements for the CCGT combustor/heat exchanger are relatively complex in that four working fluids are handled simultaneously, primary CCGT,

Table 6.1.2.1 Tube Materials

1550 F/1550 F PC System

Location	Material	T max F	Diameter	Length
Steam Boiler Convection	C. Steel	700 F	2.50 In.	125,000 Ft
Steam Reheat Convection	1/2Cr - 1/2Mo	900 F	2.50 In.	44,000 Ft
	2 1/4 C4 - 1 Mo	1025 F	2.50 In.	23,000 Ft
	304 Cres	1075 F	2.50 In.	9,000 Ft
H.P. He Convection	1/2Cr - 1/2Mo	875 F	2.50 In.	63,000 Ft
	2 1/4Cr - 1 Mo	1025 F	2.50 In.	54,000 Ft
	304 Cres	1150 F	2.50 In.	45,000 Ft
H.P. He Wall	INCO 800	1500 F	1.50 In.	107,000 Ft
H.P. He Convection	INCO 800	1500 F	1.75 In.	57,000 Ft
	INCO 617	1600 F	1.75 In.	86,000 Ft
L.P. He Convection	304 Cres	1150 F	2.50 In.	56,000 Ft
L.P. He Platens	INCO 800	1450 F	2.50 In.	19,000 Ft
L.P. He Walls	INCO 800	1500 F	1.00 In.	213,000 Ft
L.P. He Convection	INCO 800	1500 F	1.75 In.	57,000 Ft
	INCO 617	1600 F	1.75 In.	86,000 Ft

Table 6.1.2.2 He Pressure Drop Allocation for 1550 F/1550 F Pulverized Coal System, ΔP, psi

Compressor to High Pressure Heater	2.5
High Pressure Heater	30.0
High Pressure Heater to High Pressure Turbine	3.0
High Pressure Turbine to Low Pressure Heater	1.65
Low Pressure Heater	11.0
Low Pressure Heater to Low Pressure Turbine	1.6

reheat CCGT, boiler and reheat steam. The temperature of three of these fluids must be closely controlled as their flowrate varies in response to load changes, (the steam boiler outlet temperature is constant). The preliminary design makes provision to effect heat input changes in the coal feed by exactly the same means used in steam boilers, reduction of coal throughput of the pulverizers and burners and removal from service of pulverizers and burners as the load gets down below about half load. Variations in load from 25% to 100% should be relatively easily handled by these means. The maintenance of proper CCGT working fluid and steam reheat temperature with variations in load and/or variations in furnace cleanliness is provided for by the incorporation of baffles in the horizontal convection pass with dampers which permit biasing of the gas flow over the primary CCGT circuit, the reheat CCGT circuit, and the steam reheater surface. A spray type desuperheater is also provided between the horizontal and pendant steam reheater banks. A final adjustment to permit proportioning of heat absorption between the steam reheater and the CCGT working fluid circuits is to vary the amount of exhaust gas recirculation to the furnace, which will affect the proportioning of heat input between the furnace circuits, (which are entirely CCGT working fluid cooled), and the convection circuits which have both steam and CCGT working fluid cooling. The sum of these control methods is believed adequate to provide for accurate temperature control and operation over the load range of 25 to 100% of rated capacity.

6.1.4 Emissions Considerations

Sulphur oxides and particulates emissions control with the preliminary design of Fig. 6.1.1 will be provided entirely by add-on devices, i.e., electrostatic precipator or bag house and a sulphur scrubber. These devices were discussed in some detail in Section 4 and will not be further described here. Nitrogen oxides emissions are expected to be readily controlled by burner adjustment and through the provisions for flue gas recirculation. Smoke, carbon monoxide emissions and unburned carbon or acid smut should present no more difficulties than with an equivalent size steam boiler, i.e., no special problem.

6.1.5 Materials of Construction

The constraints applying to metal materials of construction were discussed in Section 4.6.1. The materials selection for this preliminary design are consistent with that discussion and have been applied insofar as possible in accordance with the rules of the ASME Boiler and Pressure Vessel Code. The tubing size, quantity and material requirements for the various heat exchanger surfaces are called out in Table 6.1.1. The conventional boiler alloys, carbon steel, chrome-moly low alloy steel, and stainless steel are used at metal temperature levels up to 1150 F. These alloys are well tested and have proven to be an economical choice for applications at those temperatures. The furnace walls and the 24 inch side spaced platens at the furnace exit are fabricated of Incoloy 800. This is a boiler code approved alloy having acceptable strength levels to 1500 F metal temperature. This temperature region from 1150 F to 1500 F encompasses the region in which difficulty has been encountered in steam boiler superheaters (the bell shaped curve region). This region of frequently observed excessive corrosion rate is traversed in the furnace wall cooling circuits. Incoloy 800 has been selected for these circuits on the basis of its strength and corrosion resistant properties. It may be helped in avoiding excessive corrosion by the absence of convection effects and, if necessary, by the utilization of coal additives to control the virulence of the attack. Incoloy 800 is relatively modest in cost as compared with Inconel 671 clad Incoloy 800. The material used for the finishing convection pass is Inconel 800 for metal temperatures from 1400 to 1500 F and Inconel 617 for metal temperatures from 1500 to 1600 F. Although these tubes are located in the zone frequently experiencing excessive corrosion rates in a conventional boiler, it is expected that the metal operating temperatures are too high to be subject to that particular corrosion mechanism. Inconel 617 is used at temperatures of 1500 F and above because of its high temperature creep strength. The design values for the strength of materials utilized is consistent with provisions of the ASME Boiler and Pressure Vessel codes. These values were tabulated in Section 4.6.1.

The materials of construction of the non-pressure part items of the combustor/heat exchanger, i.e., casing, insulation, refractories, and the like will be the same as utilized in steam boiler practice for equivalent size units.

6.1.6 Design Details

The mechanical details of the Fig. 6.1.1 preliminary design are sufficiently similar to that of large steam boilers that many of the configurations of steam practice will carry over to the CCGT combustor/heat exchanger. This will include for instance support practices, insulation, casing and buck stays, tube welding, header fabrication, and the like. The furnace walls of the Fig. 6.1.1 design are nominally a welded tube and fin configuration, which has wide usage in the boiler industry. Considerations of thermal strain, however, may indicate a change to bare tube and refractory tile construction on the same tube centers so as to bring more heat into the rearward face of the tube and minimize thermal strains. This decision would be made after a more thorough analysis.

6.1.7 Safety

The preliminary design of Fig. 6.1.1 is believed to be consistent with the intent of all of the "pressure part" safety codes of the ASME Boiler and Pressure Vessel codes. The Inconel 617 material is not presently permitted by the code but it appears reasonable to expect that its admission could be secured if properly applied for. The details of the construction of the pressure parts and design principles involved should be capable of meeting code requirements.

On the combustion side of the heat exchanger the pulverizers, burners, fans, electrical equipment, etc., are no different than present day boiler applications and should be capable of meeting all of the applicable safety codes.

6.2 PRELIMINARY DESIGN OF COAL-FIRED CCGT COMBUSTOR/HEAT EXCHANGER, ATMOSPHERIC FLUIDIZED BED, FOR 1550 F MAXIMUM WORKING FLUID TEMPERATURE

The fluidized bed combustor concept for the simultaneous combustion of coal, transfer of heat, and absorption of sulphur, was described in Section 4.1.1.2 The application of fluidized bed combustion to large combustor/heat exchangers is very much in its infancy, but promises significant advantages. It was decided to use it as the basis of a preliminary design for 1550 F working fluid operating conditions, permitting comparison with the dry bottom pulverized coal design of Section 6.1. The 1550 F maximum CCGT cycle working fluid temperature is appropriate for an atmospheric fluidized bed design in that bed temperature can be low enough to attain sufficient sulphur absorption and yet high enough to permit heat transfer to the highest temperature working fluid.

This preliminary design was carried out for the reference design conditions of Table 3.1.2, i.e., 60% helium/39% carbon dioxide/1% oxygen working fluid, 1550 F primary and 1550 F reheat working fluid temperature. The design constraints that apply in general to fluidized bed/heat exchangers are discussed in Section 4.1.

The preliminary design drawings created during the study are shown in Figs. 6.2.1, 6.2.2, and 6.2.3 As can be seen, the 350 MWe combustor/heat exchanger is made up of four primary high pressure working fluid modules of Fig. 6.2.1, four low pressure modules of Fig. 6.2.2 and four of the carbon burnup cell modules. An understanding of the arrangement and interrelationship of the modules may be gained from the station layout drawing of Fig. 6.2.4. An artist's isometric of a high pressure module is presented in Fig. 6.2.5.

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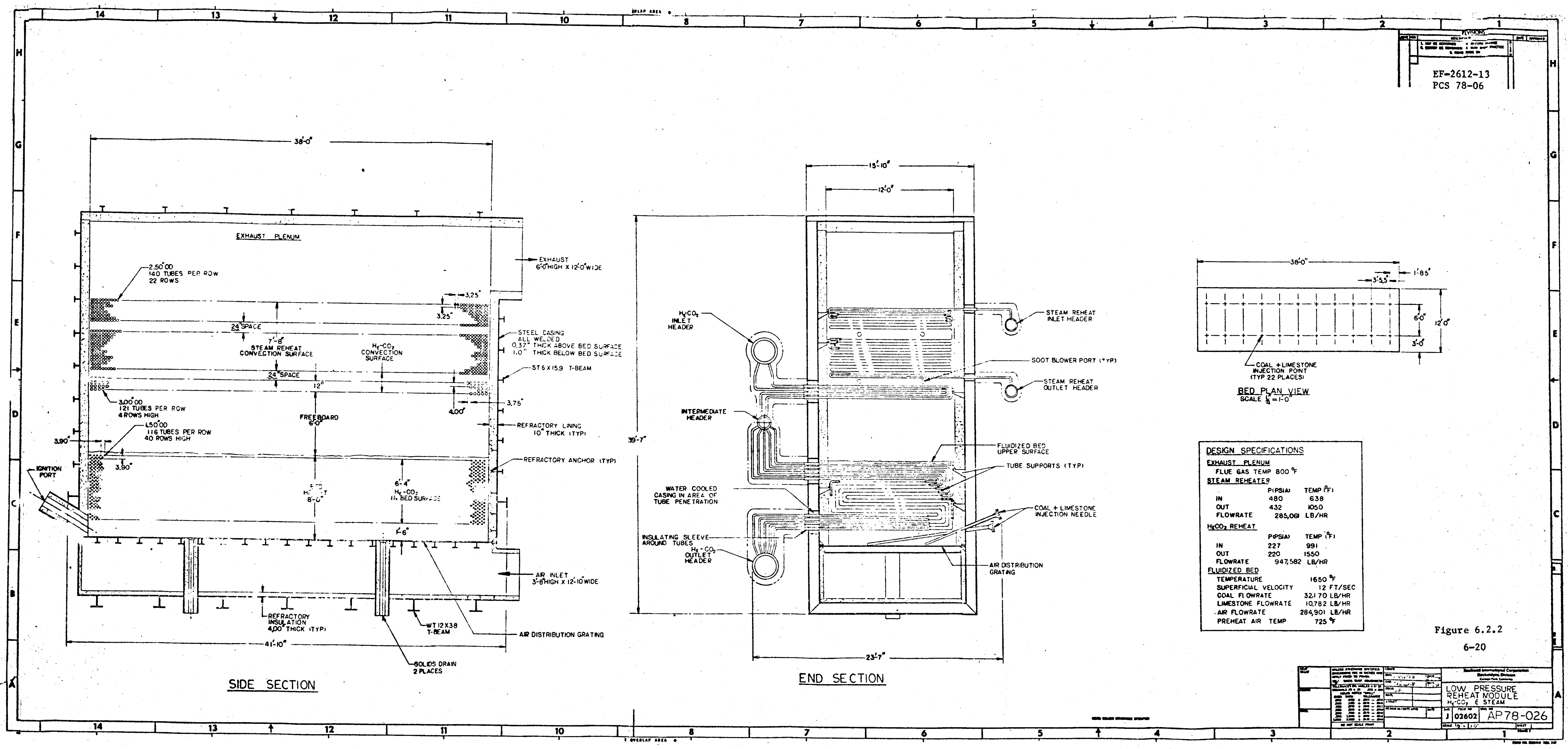


DESIGN SPECIFICATIONS

EXHAUST PLENUM

FLUE GAS TEMP 800 °F

STEAM REHEATER

	P(PSIA)	TEMP (°F)
IN	480	638
OUT	432	1050
FLOWRATE	285,061 LB/HR	

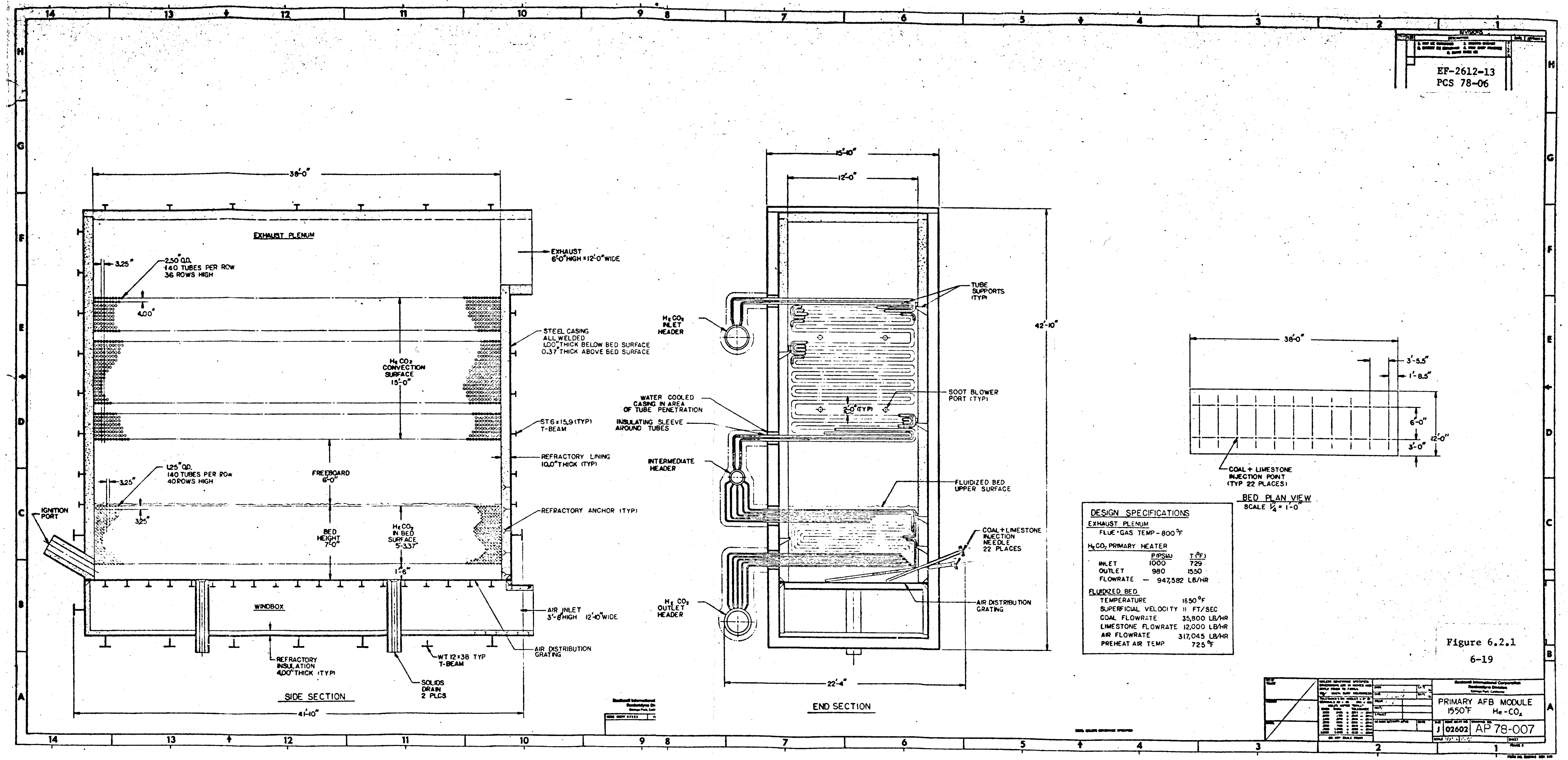
H_2-CO_2 REHEAT

	P(PSIA)	TEMP (°F)
IN	227	991
OUT	220	1550
FLOWRATE	947,582 LB/HR	

FLUIDIZED BED

TEMPERATURE	1650 °F
SUPERFICIAL VELOCITY	12 FT/SEC
COAL FLOWRATE	32,170 LB/HR
LIMESTONE FLOWRATE	10,782 LB/HR
AIR FLOWRATE	284,901 LB/HR
PREHEAT AIR TEMP	725 °F

Figure 6.2.2



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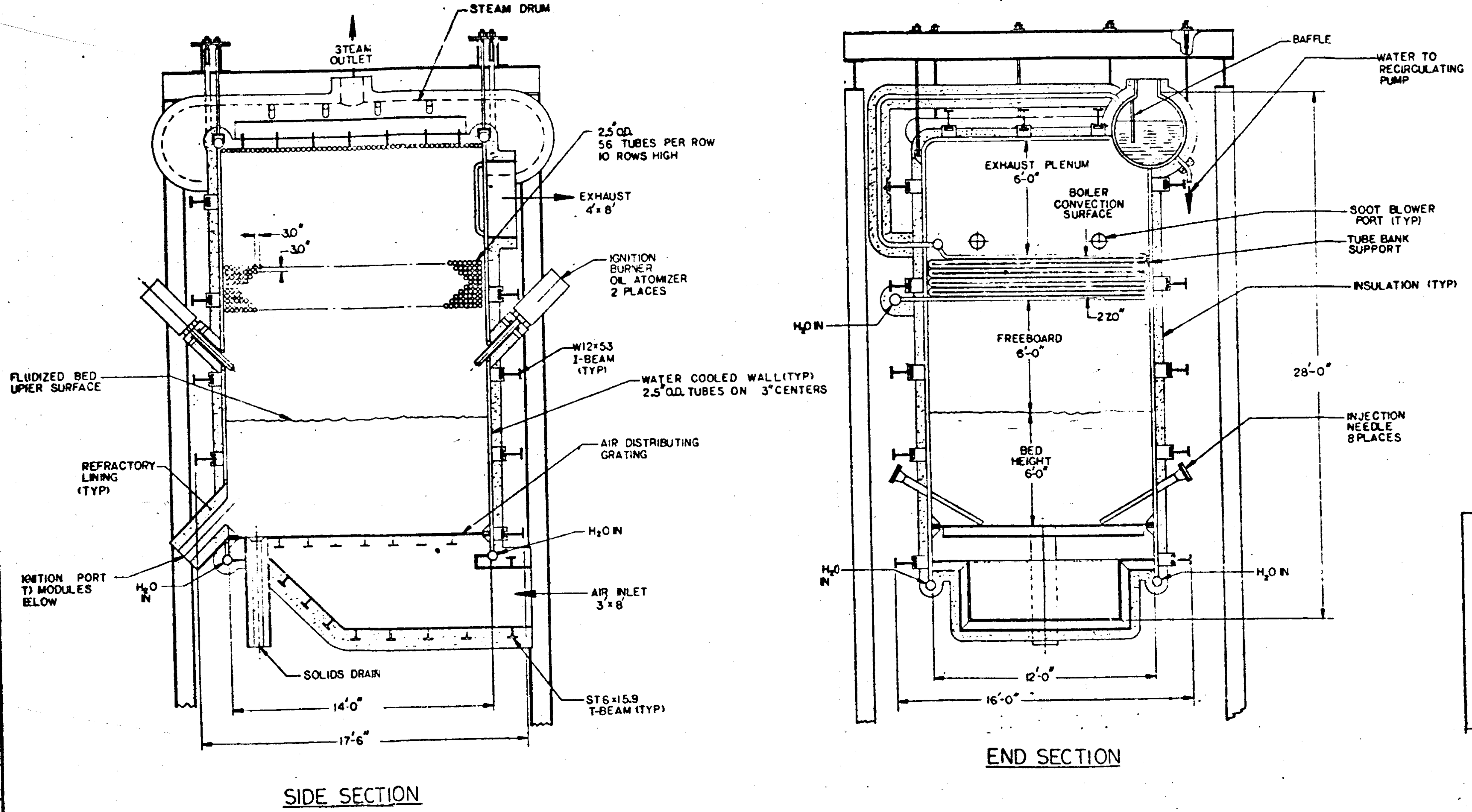
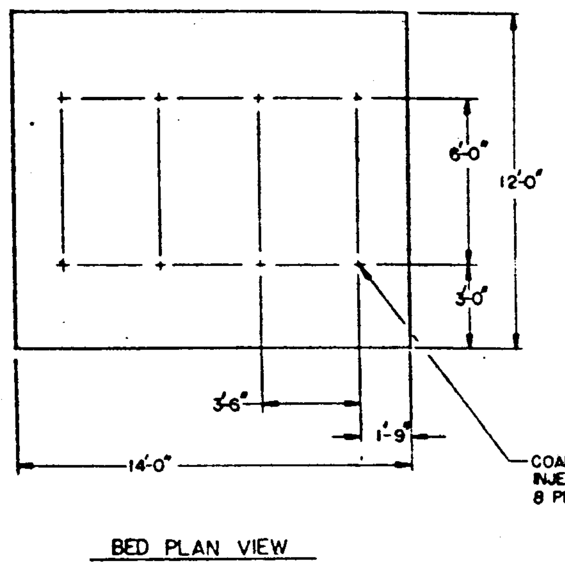
EXHAUST PLENUM
38'-0"
EXHAUST 6'-0" HIGH x 12'-0" WIDE
2.50" O.D. 140 TUBES PER ROW 36 ROWS HIGH
3.25"
4.00"
STEEL CASING ALL WELDED 1.00" THICK BELOW BED SURFACE 0.37" THICK ABOVE BED SURFACE
He CO2 CONVECTION SURFACE 15'-0"
ST6 x 15.9 (TYP) T-BEAM
REFRACTORY LINING 10.0" THICK (TYP)
1.25" O.D. 140 TUBES PER ROW 40 ROWS HIGH
FREEBOARD 6'-0"
IGNITION PORT
REFRACTORY ANCHOR (TYP)
BED HEIGHT 7'-0"
He CO2 IN BED SURFACE 5'-3.37"
1'-6"
WINDBOX
AIR INLET 3'-6" HIGH 12'-10" WIDE
AIR DISTRIBUTION GRATING
REFRACTORY INSULATION 4.00" THICK (TYP)
WT 12 x 38 TYP T-BEAM
SOLIDS DRAIN 2 PLCS
SIDE SECTION
41'-10"
15'-10"
12'-0"
42'-10"
TUBE SUPPORTS (TYP)
He CO2 INLET HEADER
SOOT BLOWER PORT (TYP)
WATER COOLED CASING IN AREA OF TUBE PENETRATION
INSULATING SLEEVE AROUND TUBES
2'-0" (TYP)
INTERMEDIATE HEADER
FLUIDIZED BED UPPER SURFACE
COAL + LIMESTONE INJECTION NEEDLE 22 PLACES
He CO2 OUTLET HEADER
AIR DISTRIBUTION GRATING
22'-4"
END SECTION
38'-0"
3'-5.5"
1'-8.5"
6'-0"
12'-0"
3'-0"
COAL + LIMESTONE INJECTION POINT (TYP 22 PLACES)
BED PLAN VIEW
SCALE 1/4" = 1'-0"
DESIGN SPECIFICATIONS
EXHAUST PLENUM
FLUE-GAS TEMP - 800°F
He CO2 PRIMARY HEATER
P(PSIA) T(°F)
INLET 1000 729
OUTLET 980 1550
FLOWRATE — 947,582 LB/HR
FLUIDIZED BED
TEMPERATURE 1550°F
SUPERFICIAL VELOCITY 11 FT/SEC
COAL FLOWRATE 35,800 LB/HR
LIMESTONE FLOWRATE 12,000 LB/HR
AIR FLOWRATE 317,045 LB/HR
PREHEAT AIR TEMP 725°F
Rockwell International Corporation
PRIMARY AFB MODULE 1550°F He-CO2
02602 AP 78-007

Figure 6.2.1

EF-2612-13
PCS 78-06





DESIGN SPECIFICATIONS

EXHAUST PLENUM			
FLUE GAS TEMP			800 °F
STEAM BOILER			
	P (PSIA)	TEMP (°F)	
IN	2,880	689	
OUT	2,720	681	
FLOWRATE			128,135 LB/HR
FLUIDIZED BED			
TEMPERATURE			2,000 °F
SUPERFICIAL VELOCITY			3.9 FT/SEC
UNBURNED CARBON FLOWRATE			2,766 LB/HR
AIR FLOWRATE			36,550 LB/HR
PREHEAT AIR TEMP			725 °F

Figure 6.2.3

Rockwell International Corporation
Rocketdyne Division

CARBON BURNUP CELL AFB

J 02602 AP 78-032

EF-2612-13
PCS 78-06

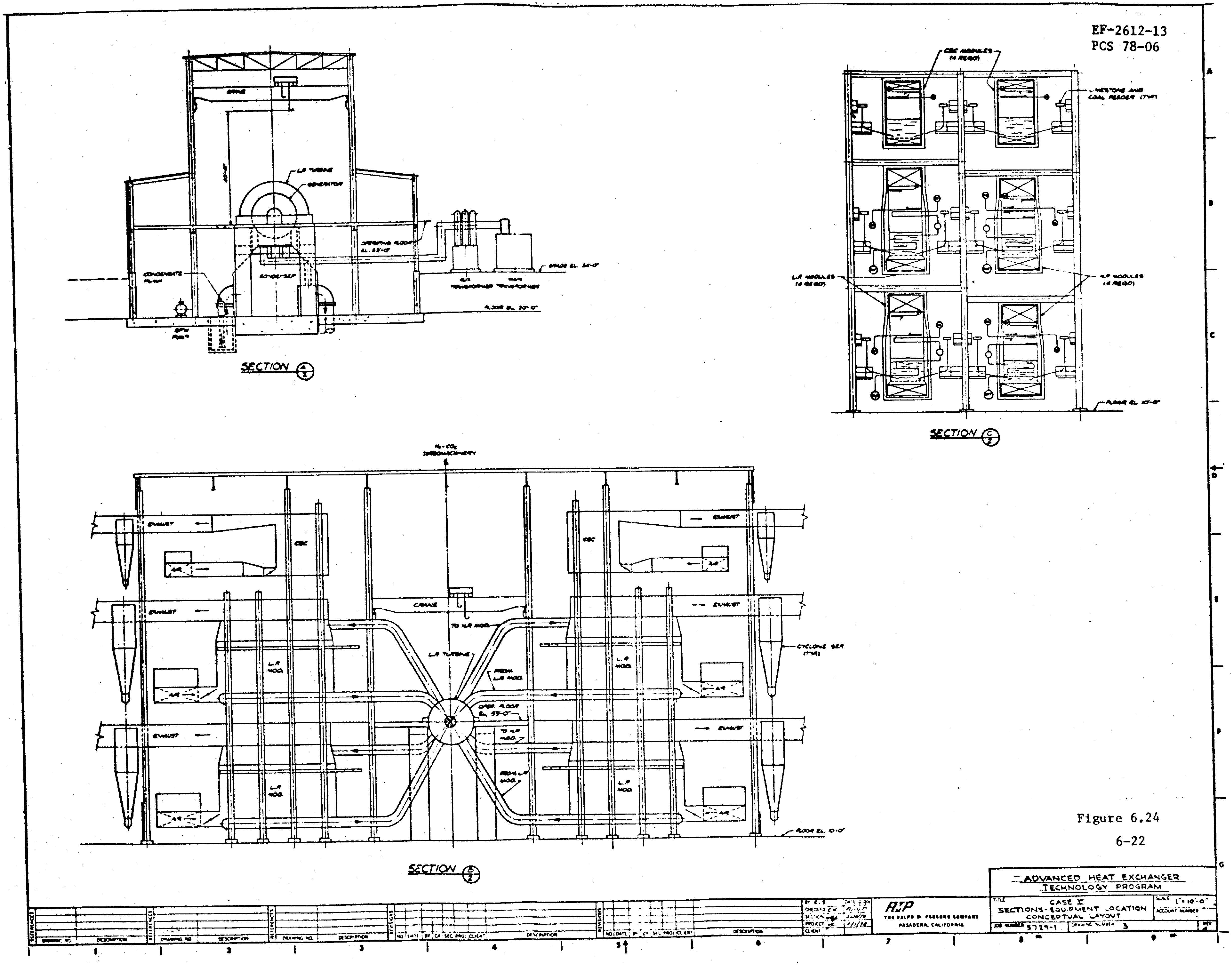


Figure 6.24

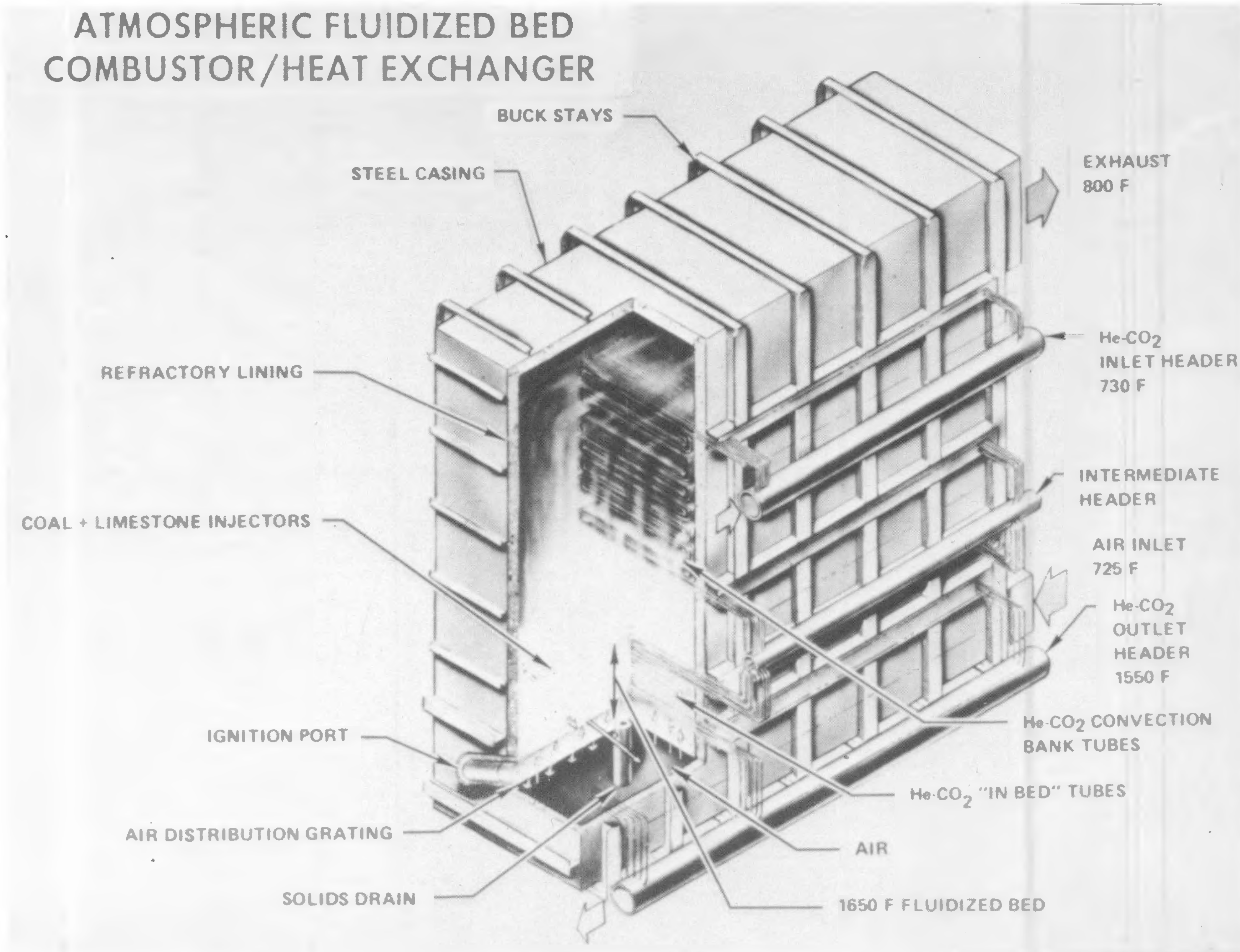

ATMOSPHERIC FLUIDIZED BED
COMBUSTOR/HEAT EXCHANGER
BUCK STAYS
STEEL CASING
EXHAUST
800 F
He-CO_2
INLET HEADER
730 F
REFRACTORY LINING
INTERMEDIATE
HEADER
COAL + LIMESTONE INJECTORS
AIR INLET
725 F
He-CO_2
OUTLET
HEADER
1550 F
He-CO_2 CONVECTION
BANK TUBES
IGNITION PORT
He-CO_2 "IN BED" TUBES
AIR DISTRIBUTION GRATING
AIR
SOLIDS DRAIN
1650 F FLUIDIZED BED

Figure 6.2.5

6.2.1 Combustion System and General Arrangement

An obvious difference between the fluidized bed preliminary design and the dry bottom pulverized coal concept discussed in Section 6.1 is the modular nature of the fluidized bed design. This modular concept was selected for reasons that are primarily aligned with the combustion system, i.e., feeding of coal and limestone to the beds and the removal of spent bed material. The design parameters that were applied to the fluidized bed combustion system are listed in Table 6.2.1. Of primary importance to the layout of the bed is the requirement that a coal and limestone feed be supplied for each 20 sq. ft. of bed plan area. The 12 ft width of the bed was selected to permit installation of downwardly feeding coal and limestone injection needles and is believed to be about the maximum width feasible. The 38 ft length represents a compromise between number of modules and a reluctance to place reliance upon one very long bed. It seems clear that economic forces would tend toward gigantism in fluidized bed combustors as it has in other combustor/heat exchanger concepts. The use of large modules has several economic advantages. Large units are less expensive to build per unit of service than a greater number of small units. The balance of plant equipment and installation costs are reduced by dealing with a few large units. Plumbing, coal feed, controls, etc., are simplified. Finally, operating costs are reduced by having fewer fired units for operator attention. The preliminary design presented here represents a recognition that economic forces press towards a minimization of the number of separate modules, but the state-of-the-art of fluidized bed combustion tempers that force.

The primary and reheat modules are designed to operate at a bed temperature of 1650 F. This is expected to permit attainment of the sulphur oxides emissions goals with a limestone feed rate consistent with a calcium to sulphur ratio of 2.75. Bed operating temperature was not optimized to balance heat exchanger surface requirement as against limestone feed requirements, and it is possible that a bed temperature slightly above or below 1650 F would provide a more economical installation. As shown in Table 6.2.1 primary and reheat beds are designed for a superficial velocity of 12 ft/sec. This is believed to represent a reasonable compromise between bed area requirements and particle carryover at rated load. It also provides the capability of maintaining fluidization at loads as low as 25%. The operating bed depth is determined by the depth required to cover the heating surface that is necessary to maintain the bed at the equilibrium temperature. The 6 and 7 ft bed heights indicated are relatively high as compared with experience in coal fired fluidized bed heat exchangers, but not out of the range of operation that has been demonstrated in other operating fluidized beds. Bed height for the CCGT service tends to be higher than that required for boiler service because the CCGT log mean temperature difference is smaller, so more heating surface is required to maintain bed equilibrium temperature.

A flat plate air distributor has been shown for each module as it is the simplest and seemingly gives adequate service. Advancements in air distribution plates may be incorporated as they come to light. Two solids drains are provided to permit maintaining bed height at the proper level, and draining on shutdown. Drainage through the solids drain is adjusted to make up the difference between feed material and elutriated and gasified material.

In operation the pre-heated combustion air enters the windbox at the bottom of the module at about 725 F and is admitted to the underside of the air distribution grating. The windbox and its inlets are proportioned so that the pressure available at all areas of the distribution grating is as nearly identical as possible. The air distribution grating separates the windbox from the fluidized bed and provides a controlled flow resistance to the combustion air which meters the air uniformly to the bottom of the bed and maintains stable and uniform air flow. A pressure drop of 20 inches of water at 725 F has been provided to

Table 6.2.1 Fluidized Bed Coal Combustion System Design Parameters

	Primary & Reheat Beds	Carbon Burn-up Cells
Bed Temperature	1650 F	2000 F
Superficial Velocity	12 fps	4 fps
Static Bed Depth	24 inches	24 inches
Expanded Bed Height	75-86 inches	70 inches
Coal Feed	-1/4 inch	--
Limestone Feed	-8 mesh	--
Ca/S Mole Ratio	2.5-4	--
Excess Air	15%	25%
Combustion Efficiency	90%	90%
In-Bed Heat Transfer Coefficient	$\sim$35 Btu/hr-ft^2-F	$\sim$60 Btu/hr-ft^2-F
Freeboard Height Above Bed	6 feet	6 feet

accomplish that function. The air pressure drop through the bed itself is expected to be on the order of 80 inches of water. It represents the weight of the particles in the bed divided by the square feet of distributor plate area.

Coal and limestone in proper proportion having been pre-mixed in the coal and limestone feed system are supplied uniformly to the bed through the "injection needles" at discrete points above the air distribution grating. All of the needles penetrate the casing from one side rather than both sides, thus eliminating interference between the needles and heater tubes and also simplifying the coal feed system. A small portion of the total air flow, on the order of 5%, is utilized to aid in the injection of the coal and limestone. Coal and limestone are supplied at the bottom of the bed and have an opportunity to mix with the bed materials in the 18 inch space provided between the air distribution grating and the bottom tube of the heating surface. This mixing space is intended to avoid non-uniformities in the bed material occupying the interstices between the tubes, as such non-uniformity may lead to smoking, or localized fuel rich regions that would be more corrosive to tube materials. No provision is made in the primary and reheat modules for the reinjection of fly ash. All fly ash collected inthe cyclone separators arranged at the exits of the 1650 F beds is passed to the carbon burnup cells of Fig. 6.2.3.

The function of the carbon burnup cells with respect to combustion is to burn the residual carbon in the fly ash collected from the 1650 F beds. The carbon burnup cells are supplied with windboxes, and distribution gratings, and "injection needles", in a manner similar to the cells just described. The combustion conditions designed for the carbon burnup cells are shown in Table 6.2. Bed operating temperature has been selected as 2000 F, which is about the maximum believed possible without encountering clumping of the ash particles due to incipient melting. For operating reasons that will be explained below, the carbon burnup cells have not provided with "inbed" surface. The maintenance of an equilibrium bed temperature depends upon the transfer of heat to the surrounding water cooled walls. Thus equilibrium bed temperature will be a function of operating bed height. The normal bed height has been calculated to be 6 feet. It is the incorporation of the carbon burnup cells in the combustion system which permits the prediction of sufficient recovery of carbon carry over from the primary and reheat beds to attain an unburned combustible loss of 1% or less, as shown in Table 4.3.1.

The physical arrangement of the fluidized bed modules as shown in Fig. 6.2.4 was influenced by the requirements for igntion and control of the combustion system. As shown in that figure each stack of primary or reheat modules is topped by a carbon burnup cell. In lighting off the system it is planned that an operating bed will first be established in each carbon burnup cell, utilizing coal and limestone for bed material and oil burners in the freeboard for light-off. The oil burners are seen in the "side section" of Fig. 6.2.3. The absence of "inbed heating surface" in the carbon burn-up cells will make it possible to establish ignition of the bed materials. Once having established a burning bed in the carbon burnup cell, provision is made to drain some burning bed material from the carbon burnup cell to the primary and reheat beds positioned below them. This burning bed material will serve to ignite the bed material and coal being fed to the lower beds. In this manner it is expected that ignition of the primary and reheat beds, with their large "inbed" surface allocations, can be reliably accomplished.

6.2.2 Heat Transfer and Pressure Drop Considerations

The heat transfer environment for the "inbed" surface of the fluidized bed combustor/heat exchanger is very different from that of the above bed surface. The inbed surface heat transfer coefficient is discussed in Section 4.1.1.2, and as explained there, is largely independent of tube surface arranement and combustion gas velocity. It appears to be primarily influenced by the size of the bed particles, and as bed particle size is related to acceptable superficial bed velocity so too the inbed gas side heat transfer coefficient may be regarded as a function of design superficial velocity. For this preliminary design, with its nominal 12 ft/sec superficial bed velocity, the inbed heat transfer coefficient has been assumed to be 35 BTU/ft^2-hr-F. Further it is assumed to be largely independent of operating load.

The heat transfer surface arrangement in the fluidized bed was governed by the desire to incorporate as much surface as possible per foot of bed height, so as to minimize bed height, while at the same time providing an arrangement that would permit free and uniform fluidization. The tube spacing adopted is similar to that incorporated in the Rivesville steam boiler demonstration plant, where successful fluidization operation appears to be achieved. Having fixed bed operating temperature, and inbed gas side heat transfer coefficient, and tube spacing proportions, the inbed surface requirement is determined by providing enough to pick up the total heat released between the adiabatic flame temperature and the effective gas temperature leaving the bed. Based on evidence at the Battelle Columbus Laboratories the heat absorbed in the bed was calculated to be 95% of the heat available between the adiabatic flame temperature and the 1650 F bed temperature. Bed tube sizing, i.e., diameters, was designed to stay within the allowable working fluid side pressure drops. Working fluid inlet temperature to the bed heating surface is based on heat balance considerations. Thus, all factors are known to calculate the log mean temperature difference that is available in the fluidized bed and thus the heating surface requirement, heating surface arrangement, and operating bed height.

The above bed surface of the fluidized bed combustor/heat exchanger is subject to the same heat transfer laws as the surface of the pulverized coal combustor/heat exchanger discussed in Section 6.1. The surface here was spaced to provide no more than 75 ft/sec flue gas velocity at rated load and the tubes were arranged in an in-line pattern to permit adequate sootblower installation. The above bed convection surface was arranged in counterflow to permit attainment of the desired 800 F gas temperature entering the air heater.

The tube bank configuration for the primary module is shown in Fig. 6.2.1. The working fluid mixture enters at the inlet header near the top of the unit and flows downward through the convection pass tubes, providing counter-flow. An intermediate header is provided to permit transition from the 280 convection pass tubes with their inline pattern to the 1400 inbed tubes with their staggerd pattern. The intermediate header also provides a mixing station to level out the working fluid temperature maldistributions prior to entering the inbed tubes, thus minimizing the alloy requirements of the inbed tubes.

The tube bank for the reheat module is shown in Fig. 6.2.2 and is similar to the primary module except that the higher design inlet temperature for the reheat working fluid permits a smaller convection bank and the addition of a steam reheat convection bank. The steam reheat tubes reduce the gas temperature leaving the reheat module to the same 800 F as applies to the primary module.

The preliminary design computations and surface arrangements were sized for the 60/39/1 mol percent mixture of helium/carbon dioxide/oxygen. However, the overall heat transfer coefficient is only moderately affected by heat transfer conditions on the working fluid side of the surface. Following the reasoning of Section 3.0 of the "Working Fluid and Cycle Analysis" report, it is expected that a similar heat exchanger could be designed for air or nitrogen working fluids at a very modest penalty in suface requirements. The premium on high working side heat transfer coefficient is not nearly so important with the fluidized bed combustion concept as it is with the pulverized coal radiant furnace combustor because the maximum temperature in the fluidized bed system is only 1650 F, so the metal temperature is relatively unaffected by working fluid side conductance.

The same design principles as already discussed for heat transfer also apply to the carbon burnup cell.

For reasons to be discussed under Section 6.2.3 below, the carbon burnup cell heat transfer surface is cooled entirely by boiling water. The much smaller bed plan area of the carbon burnup cells together with the greater temperature differential afforded by the water cooling make it possible to control carbon burnup cell temperature at 2000 F entirely with the water wall construction, omitting all other inbed surface. The working fluid side of the carbon burnup cell is arranged consistent with normal steam boiler practice. A recirculating water pump is provided for the horizontal convection bank to ensure adequate cooling at all times.

A summary of the design criteria and results with respect to heat transfer and pressure drop in all beds is contained in Tables 6.2.2.1 thru 6.2.2.3.

6.2.3 Control

The control requirements for the fluidized bed combustor/heat exchanger system are similar to those already discussed for the pulverized coal system, i.e., load control over the range from approximately 25% to rated load while maintaining close control of the temperature of three working fluids, high pressure primary CCGT, low pressure reheat CCGT, and reheat steam.

Control of a fluidized bed system is complicated by the characteristic of fluidized beds that the heat transfer coefficient to the inbed surface does not drop off as load is decreased, it may in fact increase. It is necessary that the heat pickup by the inbed surface drop off approximately proportional to load so as to reduce heat output consistent with load. Additionally, it is necessary that the heat removed from the bed by the heating surface be approximately proportional to load in order that bed temperature be maintained high enough to continue to burn the coal, (the bed will tend to go out if its temperature drops to much below 1400 F). This problem of shedding load with fluidized bed installations has been studied by several investigators with

Table 6.2.2.1 Atmospheric Fluidized Bed Combustor/Heat Exchanger

Unit	4 High Pressure Units	4 Low Pressure Units
Plan Area (Ft x Ft)	38 x 12	38 x 12
Fludiizing Vel (Ft/Sec)	11	10
Wall Construction	Refractory Lined	Refractory Lined
Bed Height (In)	75	86
Bed Temperature (F)	1650	1650
In-Bed Heat Exchanger	He-CO_2 Circuit	He-CO_2 Circuit
Free Board Height (Ft)	6	6
Convection Bank	He-CO_2 Circuit	He-CO_2 Circuit and Steam Reheat

TABLE 6.2.2.2

FBC/HEAT EXCHANGER OPERATING CONDITIONS

IN-BED	4 HIGH PRESSURE PRIMARY HEATERS	4 LOW PRESSURE REHEATERS
V (FT/SEC)	11	10
TOTAL BED DEPTH	84"	94"
U_T (BTU/FT^2 HR^{o}F)	35	34
TUBE OD (IN)	1.25	1.5
TUBE MATERIAL	INCO 617/18-8 CLADDING	INCO 617/18-8 CLADDING
ΔP (PSI)	6.70	3.86

ABOVE BED		
U_T (BTU/FT^2 HR^{o}F)	10.6	10.1
GAS VELOCITY, FPS	50	50
TUBE OD (IN)	2.5	3.0
TUBE MATERIAL	CROLOY T2/ CROLOY T22	CROLOY T22
ΔP (PSI)	3.03	.70

TABLE 6.2.2.3

CARBON BURN-UP CELL

4 CELLS

PLAN AREA (FT x FT)	14′ x 12′
FLUIDIZING VEL (FT/SEC)	3.9
WALL CONSTRUCTION	2.5″ OD ON 3.0″ CENTERS MEMBRANE WALL WATER COOLED
MATERIAL	SA 210 C
BED HEIGHT (FT)	6
BED TEMPERATURE (F)	2000
IN-BED HEAT EXCHANGER	NONE
FREE BOARD HEIGHT (FT)	6
CONVECTION BANK	56 TUBES/ROW @ 2.5 OD ON 3.0″ CENTERS SQUARE PATTERN 10 ROWS - 5 PASSES

respect to steam boiler conditions, and the conclusion reached is that it will be necessary to remove modules from service in order to operate at low loads, i.e., typically below two-thirds load. This arises from the condition with steam boiler installations that the inbed surface temperature is so low that the beds can be cooled to a temperature at which they will go out.

The preliminary designs of this CCGT study provide three options for achieving load control:

a. The bed temperature can be allowed to drop moderately while the working fluid outlet temperature is maintained at 1550 F. The log mean temperature differential is strongly affected by modest temperature drops in the bed which will reduce the heat transfer to the inbed surface and permit low load operation at feasible bed temperatures and constant bed height. This concept is illustrated graphically in Fig. 6.2.3.1. This process will cause the lower temperature loops of the in-bed heat exchanger surface to operate at higher temperature at part load than they do at full load, Fig. 6.2.3.2, but should be achievable with judicious alloy selection. The temperature rise in the convection section of the primary and reheat modules will be relatively unaffected by load changes so the extra alloy requirements are confined to the inbed surface.

b. An option also available without removing beds from service is to operate with reduced bed levels, exposing some of the upper rows of inbed surface to conditions more closely approximating the convection surface and thus reducing the heat pickup.

c. A final option when operating at really low loads for prolonged periods of time is to remove one reheater and one primary module from service. This can be achieved by installing a butterfly valve in the working fluid connections to those modules.

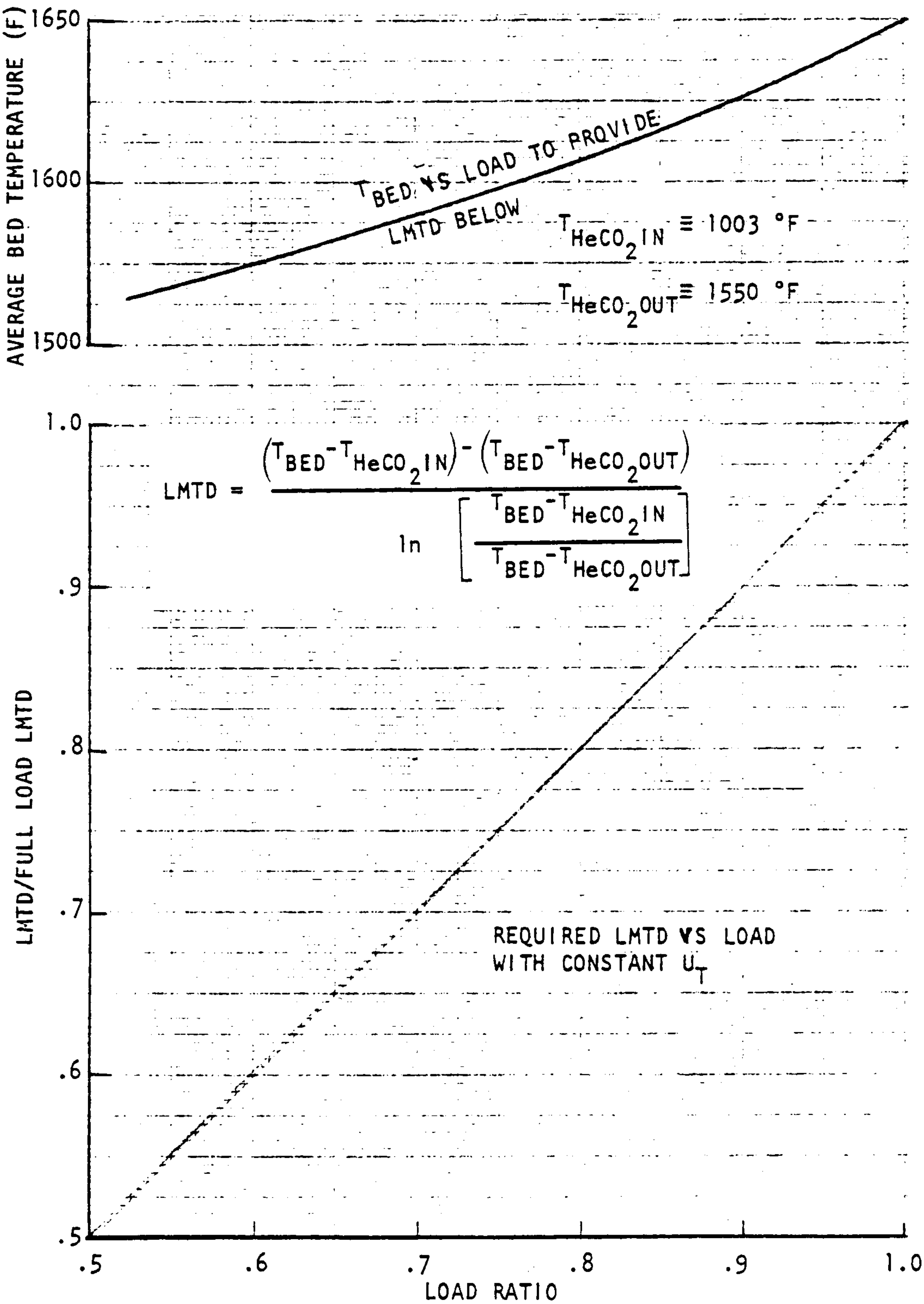


Figure 6.2.3.1 Primary Bed Temperature Profile and In-Bed Hex Equilibrium Log Mean Temperature Difference Ratio as Functions of Operating Load

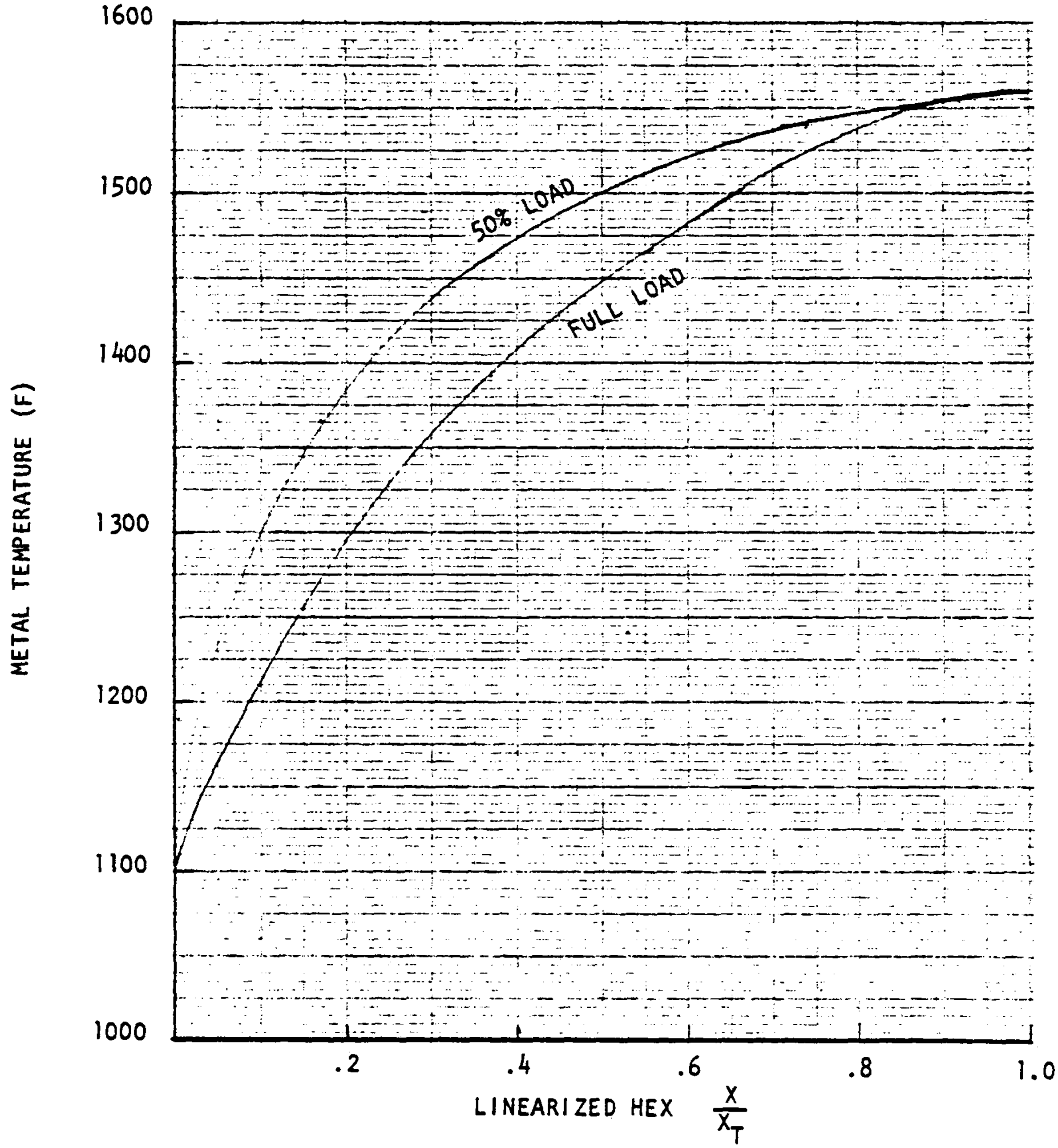


Figure 6.2.3.2. Part Load Effect on In-Bed Hex Metal Temperature Profile (High Pressure $He-CO_2$)

High pressure primary working fluid CCGT temperature is easily controllable by proportioning the coal and air flows to the working fluid flow. This is simplified by the construction of the primary fluidized bed which contain only primary surface. The reheat modules contain both low pressure CCGT reheat working fluid and steam reheater surface. Simultaneous control of CCGT reheat working fluid temperature and steam reheat temperature is planned by proportioning fuel and air flows to CCGT working fluid flow and temperature, thus controlling final CCGT working fluid temperature. Primary control of steam reheat temperature will be obtained by properly proportioning the quantity of steam reheat surface to CCGT reheat surface. Adjustment of steam reheat temperature will be obtained by a spray type desuperheater in the steam line prior to its entrance to each low pressure reheat module.

Operating conditions in the carbon burnup cell will be fundamentally controlled by proportioning the air flow to the input of recycled fly ash so as to maintain a stable bed condition. If recycled fly ash flow is so disproprtionately great that a stable bed cannot be maintained then some of the fly ash will be recycled to the high pressure modules. If recycled fly ash flow is insufficient to provide an appropriate quantity of steam generation in the carbon burnup cells then raw coal will be added to the carbon burnup cell. These latter two adjustments are expected to be of a "trim" nature and may be occasioned by changes in the coal or limestone composition over the life of the station.

6.2.4 Emissions Implications

The major advantage of the fluidized bed combustor/heat exchanger is its capacity for sulphur oxides absorption in the bed, obviating the need for the expense and energy losses associated with tail-end SOX scrubbers. Under the 1650 F operating condition, it is anticipated that the ratio of calcium to sulphur in the feed streams will need to be maintained at approximately 2.75 in order to achieve the 1.2 lbs of SO_2 per million BTU input target condition with Illinois No. 6 coal. No provisions for SOX scrubbing are provided in this preliminary design.

The particulate carry over from the fluidized bed installation is expected to be even greater than that accompanying pulverized coal combustion in that a greater quantity of limestone is carried out. Substantial amounts of this limestone may have been calcined to calcium oxide, giving a whitish appearance to the stack and perhaps tending to destroy painted surfaces downstream of the plume, especially on wet days. A highly efficient bag house installation is included in the balance of plant equipment to support this preliminary design.

Nitrogen oxides emissions from the fluidized bed combustors are expected to be well within the presently allowed minimums. The low nitrogen oxide emissions result from the low rate of nitrogen fixation from the combustion air because of low combustion temperatures. No special design provisions have been made to control nitrogen oxides.

Emissions of smoke, carbon-monoxide, hydrocarbons, and the like are not expected to be a problem with the excess air used for rated conditions. However, one must recognize that much is to be learned about coal fired fluidized beds and adjustments to the design may be necessary to achieve satisfactory operation in these respects.

6.2.5 Materials of Construction

The design constraints applying to pressure part construction for fluidized bed applications were discussed in Section 4.6.1. These constraints are adhered to in the preliminary designs. A listing of tube material, sizes, quantities and composition is given in Tables 6.2.5.1, 6.2.5.2, and 6.2.5.3. The principle followed has been to utilize the cheapest material that will meet the needs of the service with respect to both strength requirements and corrosion resistance.

The preliminary design utilizes refractory walls for the primary and reheat modules. Refractory was selected as opposed to cooled tubular walls both

TABLE 6.2.5.1

TUBE MATERIALS

4 PRIMARY HE-CO_2 MODULES

1550°F/1550°F AFB SYSTEM

LOCATION	MATERIAL	T_{MAX} °F	DIA.	LENGTH
ABOVE BED	2 1/4 Cr-1Mo	1025°F	2.50	185,000 FT
	304 CRES	1100°F	2.50	65,000 FT
IN BED	347 CRES	1450°F	1.25	174,000 FT
	INCO 617 +347 CLAD	1575°F	1.25	106,000 FT
				530,000 FT (Total 4 Modules)

TABLE 6.2.5.2

TUBE MATERIALS

4 LOW PRESSURE $He-CO_2$ AND STEAM MODULES

1550°F/1550°F AFB SYSTEM

LOCATION	MATERIAL	T MAX	DIA	LENGTH
ABOVE BED	1/2 Cr-1/2 Mo	900°F	2.50	81,000 FT
(STEAM)	2 1/4 Cr-1 Mo	1025°F	2.50	47,000 FT
	304 CRES	1075°F	2.50	21,000 FT
ABOVE BED ($He-CO_2$)	304 CRES	1100°F	3.00	23,000 FT
IN BED ($He-CO_2$)	347 CRES	1500°F	1.50	131,000 FT
	INCO 617 +347 CLAD	1600°F	1.50	103,000 FT
				406,000 FT (Total, 4 Modules)

TABLE 6.2.5.3

TUBE MATERIALS

4 CARBON BURNUP CELLS

1550°F/1550°F AFB SYSTEM

LOCATION	MATERIAL	T_{MAX} °F	DIA	LENGTH
ABOVE BED	C. STEEL	700°F	2.50	27,000 FT
IN BED	NONE			
WATER WALLS	C. STEEL	700°F	2.50	17,000 FT
				44,000 FT (Total, 4 Cells)

because it was felt that the operating conditions were such that refractories would give satisfactory service, and because of the rather awkward design problem in providing for the cooling of tubular walls. There is not enough steam boiling requirement in the reference design cycle to provide the cooling, and cooling with CCGT working fluids complicates the circuits. While a cycle change to provide more boiling is possible, the refractory lined furnace construction inherently puts more high grade heat in to the high temperature CCGT cycle and puts less in the lower efficiency Rankine cycle. The adiabatic refractory wall design also provides a higher log mean temperature difference for the CCGT cycle heating surface, thus reducing the required surface area and cost.

The refractory walls are constructed of a gunnited monolithic refractory such as a high duty fire clay in an inner layer about 4 inches thick, backed up by a layer of lightweight insulating fire clay for a total thickness of 10 inches. The high duty fire clay is required here for its high strength and chemical resistance to both the limestone and the coal combustion product. The insulating layer will be thick enough to provide an outside wall temperature of approximately 180 F and the entire enclosure will be steel cased for leak tightness. The gunnited refractory walls are tied to the outer structure and casing to prevent fracture and collapse.

6.2.6 Design Details

Much less is known about the appropriate design details for the fluidized bed combustor/heat exchanger than for the pulverized coal type combustor/heat exchangers. This is because of the early stage of development of fluidized bed combustion. There is a natural tendency to utilize the details available from the Rivesville experimental installation, and details are sure to be modified as greater operating experience is obtained.

The preliminary designs of this study have been configured to cope with differential thermal expansions between the tubes and the refractory walls and headers. A sticky design detail appears to occur where the tubes penetrate the wall, and without some additional cooling provisions the casing temperature would tend to approach the tube temperature in those areas. To alleviate this problem, a water cooled box manifold is provided where tubes penetrate the wall. Each heater tube is surrounded by a water cooled shield tube where it penetrates the manifold. The function of the tube shields is to prevent the direct contact between the heater tubes and cooling water thus minimizing the heat loss from the CCGT working fluid mixture, and providing a gradual temperature transition between the high temperature tube and the water manifold. The main tube bank supports are carried from the outer structure through the refractory into the combustion spaces. These supports will be constructed of cast high temperature resisting super alloy such as HK-40 to provide satisfactory service life under essentially adiabatic temperature conditions. Additional tube to tube supports or ties are used to transfer the loads within the tube banks and carry them to the wall supports. These in-bank tube supports are designed consistently with normal steam boiler practice. All of the tube bank supports are designed to provide for free horizontal thermal expansion of the tubes while supporting their weight. The details of the design of the tube supports are critical to the success of the heater, and additional attention to this area is warranted in the future.

Sootblower ports are provided in the convection banks in a manner consistent with pulverized coal firing. Present experience at the Rivesville power plant appears to indicate that no cleaning is necessary so sootblowers themselves have not been included in the preliminary design pricing. There is no provision for in-service cleaning of the inbed surface. The action of the bed and the low operating temperature are expected to prevent heat insulating deposits from accumulating on the inbed tubes. Inbed surface can be washed or mechanically cleaned during shutdowns through access ports provided in above bed and below bed cavities.

Consideration was given to the installation of the intermediate and exit manifolds of the primary and reheat beds within the refractory lining, perhaps in a shielded location. This construction would result in significant alloy material saving as the extra length and 90 degree bend provided for expansion purposes would not be present. The construction utilized however will permit rapid plugging of tube leaks, i.e., a leaking tube can quickly be plugged at both ends and the unit put back in service. The units are capable of running with several tubes out of service without a significant effect on capacity or efficiency.

The coal injection needles are designed similarly to an oil atomizing gun, i.e., the coal and air supplied to a plugged needle can be rapidly shut off entirely and the needle removed and replaced with a functional one.

The water cooled walls of the carbon burnup cells are assembled in an all-welded membrane wall fashioned to form a leak-tight shell around the unit. The tubes operate at the water saturation temperature of about 680 F and are top supported, allowing free expansion. I-beam shaped buck stays are provided to prevent the walls from deformation under internal pressure. The buck stays are mounted outside of the insulation and are attached to the membrane wall with sliding joints to allow for thermal growth of the wall. This entire construction of the carbon burnup cells resembles standard boiler practice.

6.2.7 Safety

With the exception of the stainless clad INCO 617 tubing, all materials utilized for pressure part construction are consistent with the ASME Boiler Pressure Vessel code. Fire side safety, electrical installation, power piping and the like are also code consistent.

There remains a question about the reaction of the fluidized bed system to a sudden trip of the CCGT working fluid turbines, i.e., loss of coolant to the inbed and above bed surfaces. Some of the CCGT literature indicates that German users of CCGT equipment have shied away from fluidized bed combustors because of possible tube material overheating under these circumstances. The plan with this preliminary design is to immediately cut off the combustion air flow to each module in the event of loss of coolant. The bed will slump, with much of it being contained in the 18-inch clear space under the bank tube. As the bed consists almost entirely of inert material, with only about 1% combustible material present, combustion will cease rapidly and the temperature excursion above rated bed operating temperature will be minimized. The materials of construction of the "inbed" heating surface are able to withstand short periods of heating to higher temperatures than design, and this is shown in Fig. 4.6.4 which shows short term tensile strength versus temperature. Additionally, the working fluid pressure in the heating surface will drop rapidly during a sudden turbine trip because the bypass valve between the turbine inlets and outlets will be opened, so the tube materials will not be subjected to rated stresses during any temperature excursion that does take place. As a result, it is expected that it will be possible to handle turbine trips without damaging the inbed surface. It may however prove difficult to restart the system, i.e., re-establish fluidization. Clean out ports providing access to the space between the air distribution grating and the lower most tubes will be provided. Additionally, experience may dictate a greater or lesser distance between the tubes and the grate due to some influence of bed slumping during a loss of cooling accident. Experimental verification of these loss of cooling procedure is probably necessary to see whether dangerous quantity of combustible gases will be distilled off of the coal content of the slumped bed, giving rise to the possibiity of explosion in the module or downstream ducting and dust separation equipment.

6.3 PRELIMINARY DESIGN OF COAL FIRED CCGT COMBUSTOR/HEAT EXCHANGER FOR 1750 F MXAIMUM WORKING FLUID TEMPERATURE

The combustor/heat exchanger concept which was carried through the preliminary design stage for 1750 F maximum working fluid temperature was the "mixed mode" concept which was described in Section 5.2 . The concept is applicable when the maximum working fluid temperature is so high that it is not practical to input the entire heat requirement via fluidized bed combustion, (the fluidized bed operating temperature would have to be too high to provide satisfactory sulphur oxide removal). The combustion process is carried out instead in two stages, the first stage at a high enough temperature to input heat to the high temperature portion of the heat exchanger and the second stage in a fluidized bed at a temperature consistent with sulphur oxides recovery and heat input to the lower working fluid temperature regions. All of the flue gas generated in the high temperature combustor is passed through the fluidized bed of the low temperature combustor so as to remove its sulphur oxides content to a degree consistent with emissions regulations.

The cycle selected for combustor/heat exchanger design was a 1750 F primary temperature 1550 F reheat temperature, 60% helium/39% CO_2/1% oxygen, (mol %), working fluid,steam bottomed cycle. The cycle operating conditions and the performance requirements of the combustor/heat exchanger are summarized in Table 3.1.4. A discussion of the design constraints applying to mixed mode combustor/heat exchanger systems and an explanation of the reasons for the choice of this particular combustor/heat exchanger concept for preliminary design effort is given in Section 5.2.

The 1750 F preliminary design was planned to be an evolutionary out-growth of the technology of lower working fluid temperature all-metal atmospheric fluidized bed based heat exchangers. As explained in Section 4.6.1 , it is believed that the upper limit of feasible all-metal heat exchangers will be reached at about 1550 F working fluid temperature. Thus, the subject combustor/heat exchanger requires the application of ceramic heat exchanger tubes to all surfaces whose working fluid operate at higher 1550 F. The design produced is partially shown in Fig. 6.3.1. This drawing depicts the 2000 F bed temperature fluidized bed combustor module which heats the high pressure CCGT working fluid from 1550 to 1750 F. The convection surface of Fig. 6.3.1 then reduces the exit gas temperature of these modules to 800 F, a temperature suitable for passage through a fan and introduction to the windboxes of the fluidized beds which operate at 1650 F bed temperature. The 1650 F operating temperature beds are nearly identical in plan area to the beds described in Section 6.2 above, because they handle approximately the same total flue gas flow as the Section 6.2 design. The heating surface arrangements in these low bed temperature modules vary only slightly from the arrangements in the design discussed in Section 6.2 . Drawings of the heating surface arrangements of the low temperature beds of the 1750 F design have not been prepared. However, the surface arrangements have been sized and are tabulated and discussed in Section 6.3.2 below.

The total combustor/heat exchanger installation for the 1750 F preliminary design consists of four 1650 F bed temperature 1550 F working fluid temperature high pressure modules, four 2000 F bed operating temperature 1750 F working fluid temperature high pressure modules, four 1650 F bed operating temperature low pressure reheat modules, and four carbon burnup cells. An arrangement of the modules in the power station is shown in Fig. 6.3.2, and an artist's isometric of a 2000 F module is shown in Fig. 6.3.3.

A description of the design features of the 1750 F working fluid temperature combustor/heat exchanger preliminary design is presented below.

6.3.1 Combustion System and General Arrangement

The combustion system of the 1750 F preliminary design differs from that of the 1550 F fluidized bed preliminary design in two respects. The Fig. 6.3.1

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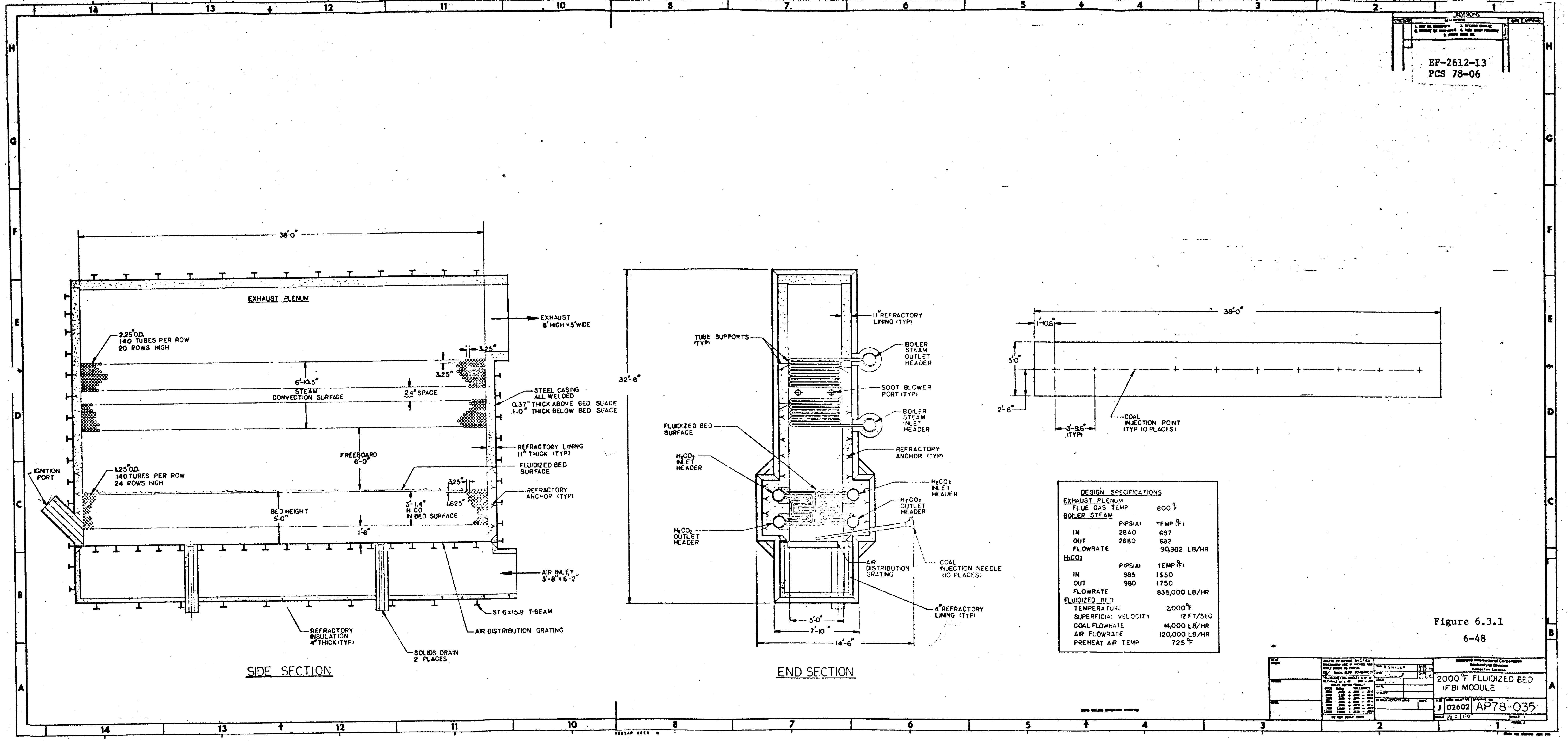

Figure 6.3.1

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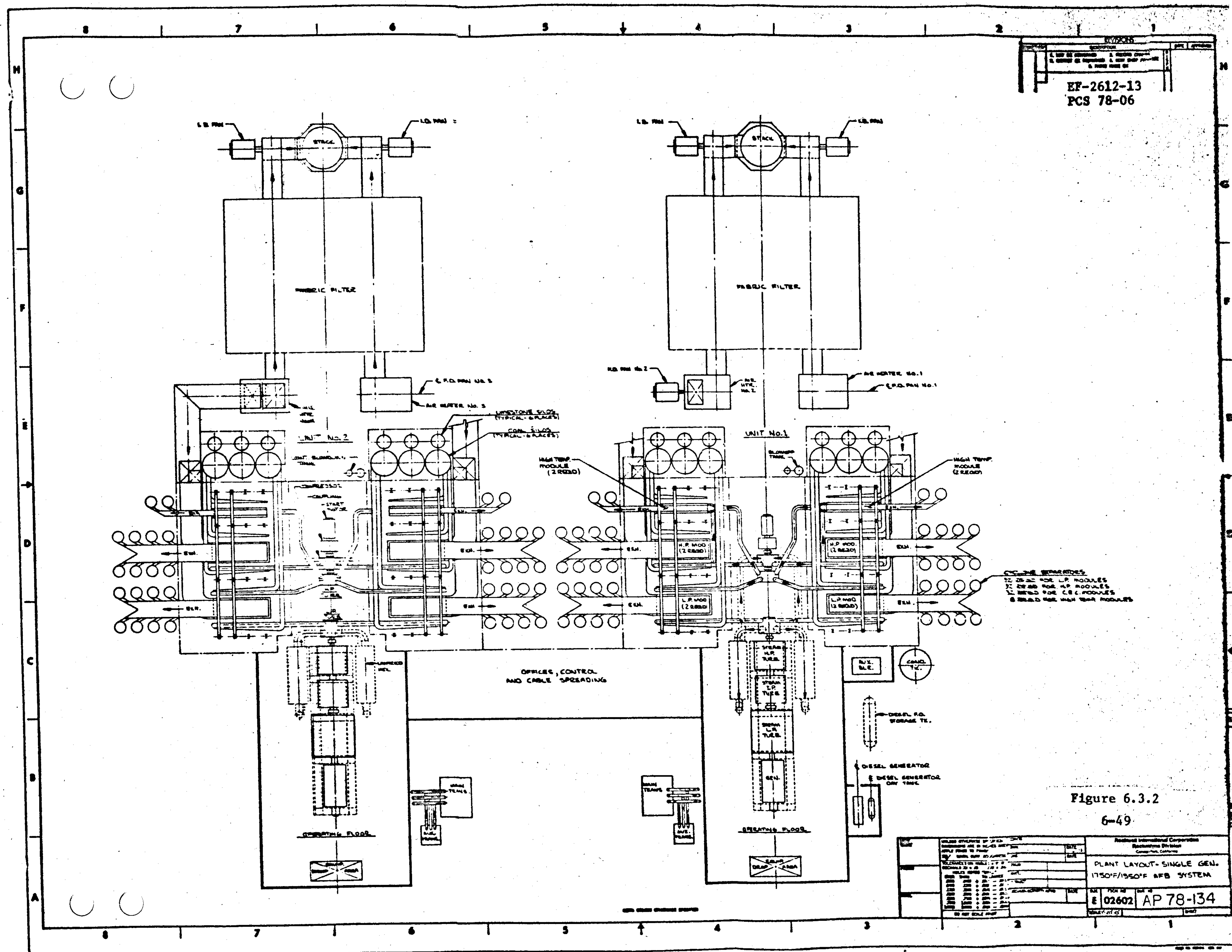

Figure 6.3.2

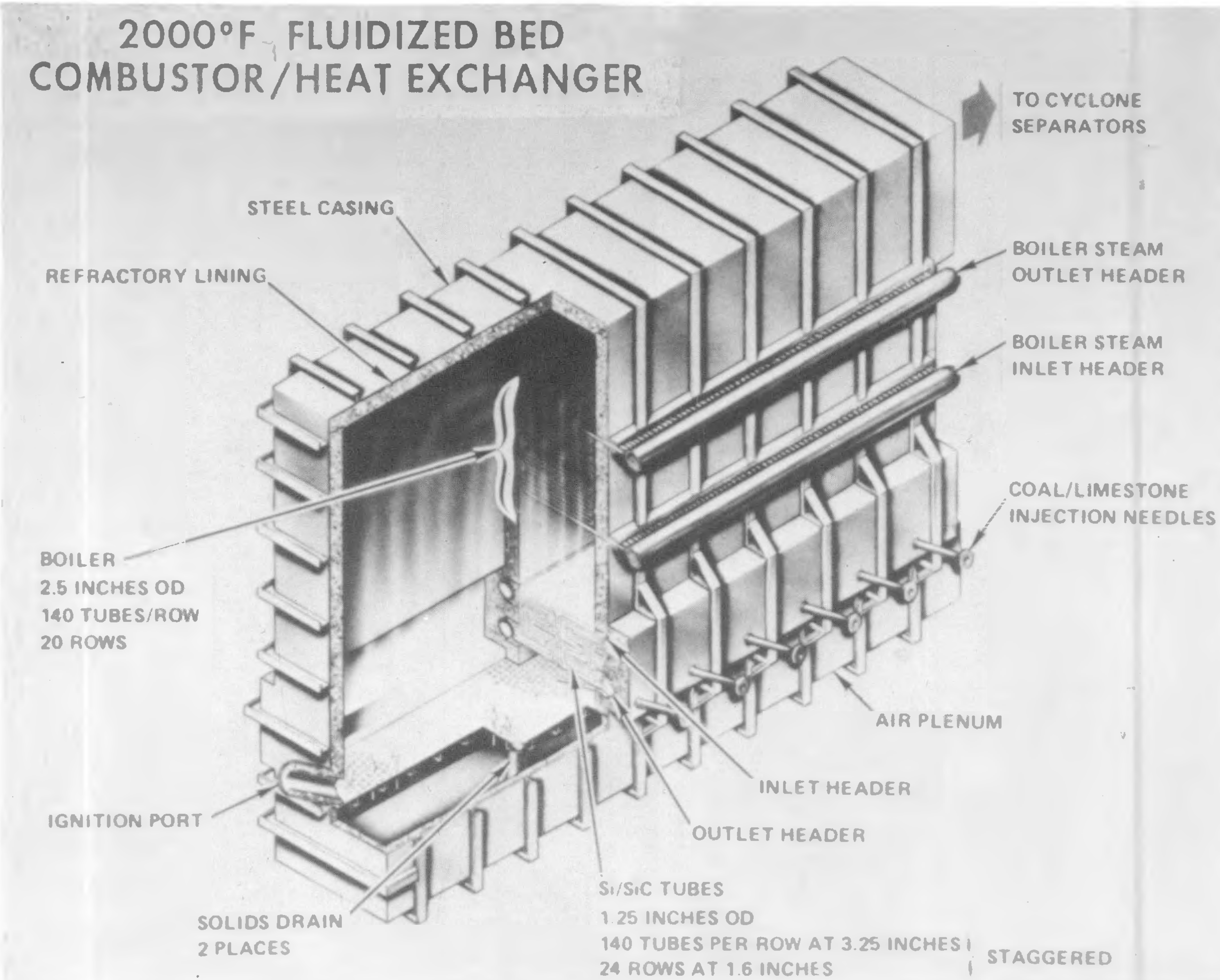
2000°F FLUIDIZED BED
COMBUSTOR/HEAT EXCHANGER
TO CYCLONE
SEPARATORS
STEEL CASING
REFRACTORY LINING
BOILER STEAM
OUTLET HEADER
BOILER STEAM
INLET HEADER
COAL/LIMESTONE
INJECTION NEEDLES
BOILER
2.5 INCHES OD
140 TUBES/ROW
20 ROWS
AIR PLENUM
INLET HEADER
IGNITION PORT
OUTLET HEADER
Si/SiC TUBES
1.25 INCHES OD
140 TUBES PER ROW AT 3.25 INCHES
24 ROWS AT 1.6 INCHES
STAGGERED
SOLIDS DRAIN
2 PLACES

Figure 6.3.3

1750 F working fluid temperature beds are designed to operate at approximately 2000 F, and the 1650 F operating temperature bed will be fed with a mixture of air plus the gaseous combustion products resulting from the combustion of 21% of the total coal input. The percentage of coal burned in the 2000 F beds is minimized when the working fluid leaves the 1650 F all-metal beds at the highest feasible temperature. Two factors make it desirable to minimize the extent of the heat addition in the 2000 F operating temperature beds. One cannot be sure at this point that the sulphur oxides removal efficiency of the 1650 F bed will be entirely satisfactory with respect to absorbing sulphur oxides from combustion gases entering it through its windbox. And secondly, the 1650 F operating bed plan dimensions are fixed by the total coal burned for the 350 MWe capacity. It does not matter whether 50% of the coal is burned in 2000 F beds or 10% of the coal, the 1650 F beds have the same plan dimensions. Thus, economic forces favor minimizing the coal burned in the 2000 F beds and thereby the plan dimensions of those beds. However, it is possible that a 1550 F working fluid temperature is not the economically optimum dividing line between metal tubes and ceramic tubes. The clad Inconel 617 alloy required for that temperature is extraordinarily expensive so it may turn out that the economically optimum dividing line is lower than 1550 F, i.e., 1500 F or 1450 F. The preliminary design Fig. 6.3.1 is capable of implementation with other dividing lines between metal and ceramic working fluid temperatures.

The 2000 F beds will be fed with a mixture of coal and limestone in the same proportions as is fed to the low temperature bed but it is not expected that very much, if any, of the sulphur in the coal will be reacted with that limestone. The 800 F combustion products leaving the 2000 F bed module will be passed through cyclone spearators prior to their being mixed with the combution air for the low temperature beds. The bottoms from the cyclone separators will be injected into the low temperature beds to utilize their lime content for SOX capture.

Limestone is used for make-up bed material for the 2000 F beds to provide simplicity of operation. Alternative inert bed materials for the high temperature bed might also find application but would seemingly complicate the bed handling problem.

The 2000 F operating temperature of the bed is believed to be feasible with the Illinois No. 6 coal feed. The 1650 F operating temperature beds combustion system is similar to that of the Section 6.2 preliminary design except that the incoming air stream will carry with it the gaseous products of combustion of 21% of the coal. While it seems reasonable that these 1650 F beds will perform the function of removing the sulphur oxides from the incoming gas stream experimental verification appears in order before extensive commitment is made to this concept.

The general arrangement of the fluidized bed systems, their light off and operation will be similar in nearly all respects to that described in Section 6.2 above. The arrangement drawing of Fig. 6.3.2 permits short connections between the modules and between modules and turbomachinery. The position of the carbon burnup cells above all of the modules which contain in-bed surface permits light off by bed material drained from the carbon burnup cells. The design specifications for the modules are presented in Tables 6.3.1 through 6.3.4.

An alternative considered but rejected was using the fly ash allocated to the carbon burnup cells as input for the 2000 F temperature cells. It is expected that there will be insufficient energy available in this returned fly ash to supply all the needs from the 2000 F module, the unburned carbon available being only about 5% of heat input while the 2000 F modules require 21%. Additionally, the carbon burnup cells must operate at low fluidization velocities, on the order of 4 ft/sec, to complete the combustion of the fly ash and this would make for an excessively large ceramic module. The possibility remains however that means can be found to combine the functions of the carbon burnup cells and the ceramic bed modules, thereby reducing the number of modules required and the expense of the combustor/heat exchanger.

Table 6.3.1 Design Specifications of Each of Four 2000 F Bed Modules

-1750/1550 F Mixed Mode System

<u>Exhaust Plenum</u>

Flue Gas Temperature - 800 F

	P (psia)	T (F)
He-CO_2 High Temperature Heater		
Inlet	985	1550.0
Outlet	977	1750.0
Flowrate	835,000 lb/hr	
Steam Boiler		
Inlet	2840	687.0
Outlet	2680	682.0
Flowrate	91,000 lb/hr	

<u>Fluidized Bed</u>

Temperature	2000 F
Superficial Velocity	12 ft/sec
Coal Flowrate	14,700 lb/hr
Limestone Flowrate	5,000 lb/hr
Air Flowrate	126,300 lb/hr
Preheat Air Temperature	725 F
Bed Plan Size	5 ft x 38 ft
Bed Height	38 inches

Table 6.3.2 Design Specifications of Each of Four Primary Heater Modules

-1750 F/1550 F Mixed Mode System

Exhaust Plenum

Flue Gas Temperature - 800 F

He-CO_2 Primary Heater

	P (psia)	T (F)
Inlet	1000	712.1
Outlet	985	1550.0
Flowrate	805,250 lb/hr	

Fluidized Bed

Temperature	1650 F
Superficial Velocity	12 ft/sec
Coal Flowrate	32,500 lb/hr
Limestone Flowrate	11,000 lb/hr
Air Flowrate	305,500 lb/hr
H.T. Bed Comb. Products Flowrate	90,000 lb/hr
Preheat Air Temperature	725 F
Bed Plan Size	38 ft x 12 ft
Bed Height	7 ft

Table 6.3.3 Design Specifications of Each of Four Low Pressure Reheater Modules

-1750 F/1550 F Mixed Mode System

Exhaust Plenum

Flue Gas Temperature - 800 F

Steam Reheater

	P (psia)	T (F)
Inlet	480	638.0
Outlet	432	1050.0
Flowrate	273,000 lb/hr	

$He-CO_2$ Reheat

	P (psia)	T (F)
Inlet	258	1170.7
Outlet	250	1550.0
Flowrate	834,500 lb/hr	

Fluidized Bed

Temperature	1650 F
Superficial Velocity	9 ft/sec
Coal Flowrate	24,470 lb/hr
Limestone Flowrate	8,200 lb/hr
Air Flowrate	230,200 lb/hr
H.T. Bed Comb. Products Flowrate	62,000 lb/hr
Preheat Air Temperature	725 F
Bed Plan Size	38 ft x 12 ft
Bed Height	8 ft

Table 6.3.4 Design Specifications of Each of Four Carbon Burnup Cells

-1750 F/1550 F Mixed Mode System

Exhaust Plenum

Flue Gas Temperature - 800 F

Steam Boiler

	P (psia)	T (F)
Inlet	2840	687
Outlet	2680	682
Flowrate	105,100 lb/hr	

Fluidized Bed

Temperature	2000 F
Superficial Velocity	4 ft/sec
Unburned Carbon Flowrate	3,000 lb/hr
Air Flowrate	38,100 lb/hr
Preheat Air Temperature	725 F
Bed Plan Size	14 ft x 12 ft
Bed Height	6 ft

6.3.2 Heat Transfer and Pressure Drop Considerations

The working fluid chosen for the preliminary design was the 60% helium/39% CO_2/ 1% oxygen, (mol %). The pressure drop allocations were revised slightly as compared to the 1550 F design to provide for the extra ducting and heat exchange surface in the high pressure circuits. The working fluid side heat transfer coefficients in general do not exercise a strong effect on the surface requirements for the reasons explained in Section 6.2.2 above. Conditions in the bed of the 2000 F module are such that the ceramic tube material has a large margin of temperature safety so that it can accommodate low working fluid mass fluxes and pressure drops. In the light of these considerations, this 1750 F preliminary design concept can be adapted to air or nitrogen working fluids with relatively modest changes in circuit arrangements and total surface required.

The combustion product side of the heat exchanger surface was designed on the basis of the same heat transfer considerations as described in Section 6.2.2 above for the 1550 F preliminary design.

6.3.3 Control

The system of control for working fluid temperatures and load for the 1750 F preliminary design will be similar in most respects to that already discussed for the 1550 F design in Section 6.2.3 above. The modular bed arrangement for the primary CCGT working fluid, with 1650 F beds raising the temperature to 1550 F and the 2000 F beds raising temperature to the final 1750 F, will permit separate control of the heat inputs in each of these modules and maintenance of the final working fluid temperatures at the design condition.

Modular bed arrangement will permit isolation of the 2000 F beds from the circuit in the event that special difficulties are experienced with the ceramic heat exchanger tubes which require long term modification efforts. The high temperature primary helium can then be bypassed directly from the 1550 F primary beds to the higher pressure turbine. The cycle efficiency would be slightly reduced by this expedient and the system pressures would balance out at a slightly different condition than the design, but it appears feasible to provide structural strength in the original design to cope with the unbalanced condition and the capability for continuing operation may well be worth the small extra expense involved.

6.3.4 Emissions Considerations

The emissions considerations in the 1750 F preliminary design are similar to those of the 1550 F fluidized bed preliminary design except for the addition of spent flue gas to the windbox of the 1650 F beds. NOX emissions of the 2000 F beds are expected to be acceptable.

6.3.5 Materials of Construction

The heating surface, tube sizing, materials of construction, etc., of the modules making up the 1750 F preliminary design are presented in Tables 6.3.5 thru 6.3.8 . The materials of construction of the 1650 F operating temperature fluidized beds and the carbon burnup cell were selected according to the same rationale explained in Section 6.2.5 for the 1550 F preliminary design.

The 1750 F preliminary design materials were selected so that ceramic tubing was used for the heating surface at all points where the design temperature of the working fluid was greater than 1550 F. The available ceramic materials and the criteria applied to the design are discussed in Section 4.6.2. The 1750 F operating temperature design represents a relatively modest demand upon the working strength of silicon carbide tubing. Virtually any of the silicon carbide material forms discussed in Section 4.6.2 appear suitable for this

Table 6.3.5 Tube Materials
Four High Temperature Modules

1750 F/1550 F Mixed Mode System

Location	Material	T max F	Diameter	Length
Above Bed (Steam)	C. Steel	700 F	2.25 inch	56,000 ft
In-Bed (He CO_2)	SiC	1800 F	1.25 inch	67,000 ft
				123,000 ft (Total, 4 Modules)

Table 6.3.6 Tube Materials
Four Primary He-CO_2 Heaters

1750 F/1550 F Mixed Mode System

Location	Material	T max F	Diameter	Length
Above Bed	2-1/4Cr-1Mo	1025 F	2.50 inch	166,000 ft
	304 CRES	1100 F	2.50 inch	58,000 ft
In-Bed	347 CRES	1450 F	1.25 inch	141,000 ft
	INCO 617 (347 CLAD)	1575 F	1.25 inch	86,000 ft
				451,000 ft (Total, 4 Modules)

Table 6.3.7 Tube Materials
Low Pressure He-CO_2 and Steam Reheaters

1750 F/1550 F Mixed Mode System

Location	Material	T max F	Diameter	Length
Above Bed	1/2Cr-1/2Mo	900 F	2.50 inch	56,000 ft
	2-1/4Cr-1Mo	1025 F	2.50 inch	32,000 ft
	304 CRES	1075 F	2.50 inch	14,000 ft
Above Bed (He-CO_2)	304 CRES	1210 F	2.50 inch	23,000 ft
In-Bed (He-CO_2)	347 CRES	1500 F	1.50 inch	98,000 ft
	INCO 617 (347 CRES)	1600 F	1.50 inch	90,000 ft
				313,000 ft (Total, 4 Modules)

Table 6.3.8 Tube Materials
Four Carbon Burnup Cells

1750 F/1550 F Mixed Mode System

Location	Material	T max F	Diameter	Length
Above Bed	C. Steel	700 F	2.50 inch	27,000 ft
In-Bed	None			
Water Walls	C. Steel	700 F	2.50 inch	17,000 ft
				44,000 ft (Total, 4 Modules)

application. The brittle and unforgiving nature of silicon carbide tubing did strongly influence the configuration of the 2000 F bed temperature modules. The self-supporting serpentine tubes seen in the end section were provided in order to minimize the thermal strains to which the tubing is exposed. The maximum length of self-supporting tubing was calculated to be approximately 30 inches and this set the width of the 2000 F fluidized bed module. The design concept envisions a ceramic outlet header for the high temperature module and ceramic lined metal piping from the module to the high pressure turbine. Further discussion of materials influence upon the design is presented in Section 6.3.6 below.

6.3.6 Design Details

The design details of the 1650 F bed temperature modules will be similar in nearly all respects to those of the 1550 F fluidized bed preliminary design discussed in Section 6.2 above. The flue gas from the 2000 F operating temperature beds will be passed through dust separating cyclones to remove most of the fly ash and limestone particles elutriated from the 2000 F bed modules, but some portion of the fines will continue through the cyclones, on through the fan, and into the windbox of the 1650 F operating temperature beds. A relatively high pressure drop, $\sim 20''$ H_2O, of the air and gas mixture through the distributor plate is necessary for proper functioning of the low temperature beds and erosion of the distributor plate due to entrained abrasive ash and limestone products may be a problem that will need design attention. Additionally, it is possible that there will be a tendency towards blockage of this distributor plate as a result of dust deposits containing calcium oxide and calcium sulphate, especially if high humidity during periods of inoperation permits hydration of the products. Provision is made in the design for access to the lower face of the distributor plate to assess and correct these conditions.

The design details of the 2000 F bed temperature module contain differences from 1650 F bed operating temperature modules that are a function of the unique properties of the silicon carbide heating surface material. Because of the brittle nature of the silicon carbide, careful consideration must be given to the total combined stress level, i.e., the sum of each and every stress resulting from pressure, and thermal strains, and externally applied loads, and vibrations, etc. The design response to this requirement embodied in Fig. 6.3.1 is the self-supporting serpentine loop arrangement. All of the tubular heating surface is supported from the upper header. The lower header is also supported from the upper header, being tied to it by a relatively few bypass tubes which operate at 1550 F, thus avoiding thermal strains caused by the difference in temperature between the refractory walls and the ceramic tubing. The inlet and outlet manifolds of the inbed surface are recessed into the refractory wall for several reasons. By mounting the manifolding inside the enclosure only the inlet and outlet feed pipes have to be brought through the enclosure and thermal strain at the penetration points that would otherwise occur if tubing were brought through is eliminated. An additional safety reason is discussed in Section 6.3.7 below. While the ceramic tubing configuration of this preliminary design is believed to be technically feasible, it is entirely possible that additional design effort and sophisticated analyses would provide improvements.

6.3.7 Safety

The safety aspects of the 1750 F preliminary design are similar to those of the 1550 F all-metal design in all respects except those concerning the utilization of ceramic heat exchanger materials. The safety aspects of ceramic material and their inter-relationship with legally required construction codes, such as the ASME Boiler and Pressure Vessel code, were discussed in Section 4.7. The preliminary design of Fig. 6.3.1 is responsive to the safety design criteria in that all ceramic pressure parts are entirely enclosed within the refractory and steel cased walls of the combustor. All pressurized working fluid piping exterior to the wall is refractory lined metal, and thus not subject to sudden fracture and explosion.

6.4 2250 F WORKING FLUID CCGT COMBUSTOR/HEAT EXCHANGER PRELIMINARY DESIGN

The fourth preliminary design created under this study contract is intended to supply the requirements of a CCGT cycle operating with a maximum working fluid temperature of 2250 F. The cycle design and combustor/heat exchanger performance requirements are listed in Table 3.1.6. The combustor/heat exchanger concept selected to supply these requirements is the cyclone combustor fired slagging or "wet bottom" furnace. The constraints applying to combustor/heat exchangers required to accommodate such high working fluid temperature as 2250 F are discussed in Section 4.7.1 and the rationale supporting this selection of the cyclone combustor concept for this application is presented there.

The selected cycle is a non-reheated CCGT system, non-recuperated, and steam bottomed. The working fluid selected is a 99 mol % helium/1 mol % oxygen mixture. The preliminary design represents the embodiment of advanced ceramic technology in the combustor/heat exchanger area and would require advanced ceramics in the turbine area. The design is thus acknowledged at the start to be "far out" and was intended to provide information on one way the combustor/heat exchanger technology might grow with advances in the state-of-the-art of ceramic utilization. While the cycle selected is non-reheat there are no intrinsic reasons why reheat to an intermediate temperature, or even to 2250 F, could not be incorporated in the combustor/heat exchanger.

A drawing of the 2250 F preliminary design is presented in Fig. 6.4.1, and an artist's illustration in Fig. 6.4.2. The single 350 MWe unit is approximately 175 feet high, 60 feet wide, and 125 feet deep. Crushed coal and hot air are admitted to the cyclones where combustion is intensive enough to raise the gas temperature to over 3000 F. These hot gases pass over ceramic tube elements in which the CCGT working fluid is heated from 1550 F to 2250 F. Molten slag accumulates on the boundary walls and ceramic heater surfaces and is drained

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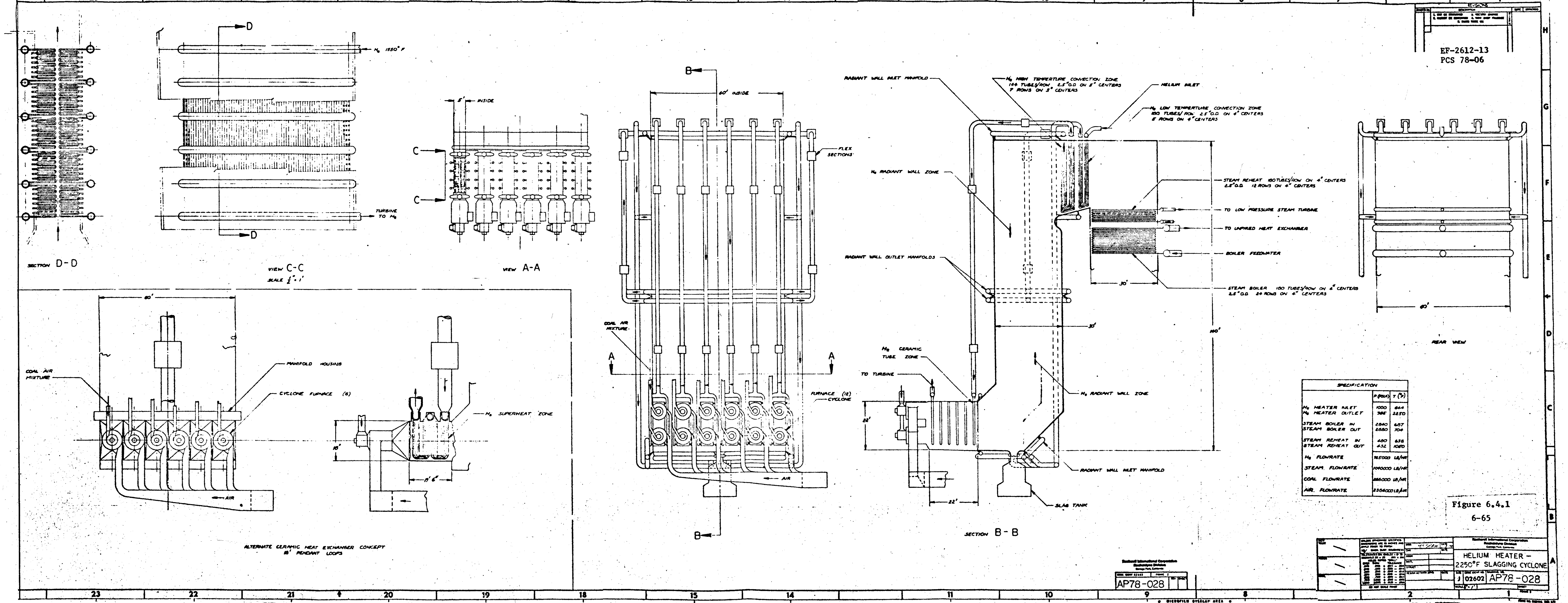

SPECIFICATION	P (PSIA)	T (°F)
He HEATER INLET	1000	644
He HEATER OUTLET	985	2250
STEAM BOILER IN	2840	657
STEAM BOILER OUT	2680	704
STEAM REHEAT IN	460	638
STEAM REHEAT OUT	432	1060
He FLOWRATE	925000 LB/HR	
STEAM FLOWRATE	1010000 LB/HR	
COAL FLOWRATE	266000 LB/HR	
AIR FLOWRATE	2364000 LB/HR	

Figure 6.4.1

2250°F CYCLONE FIRED COMBUSTOR/HEAT EXCHANGER

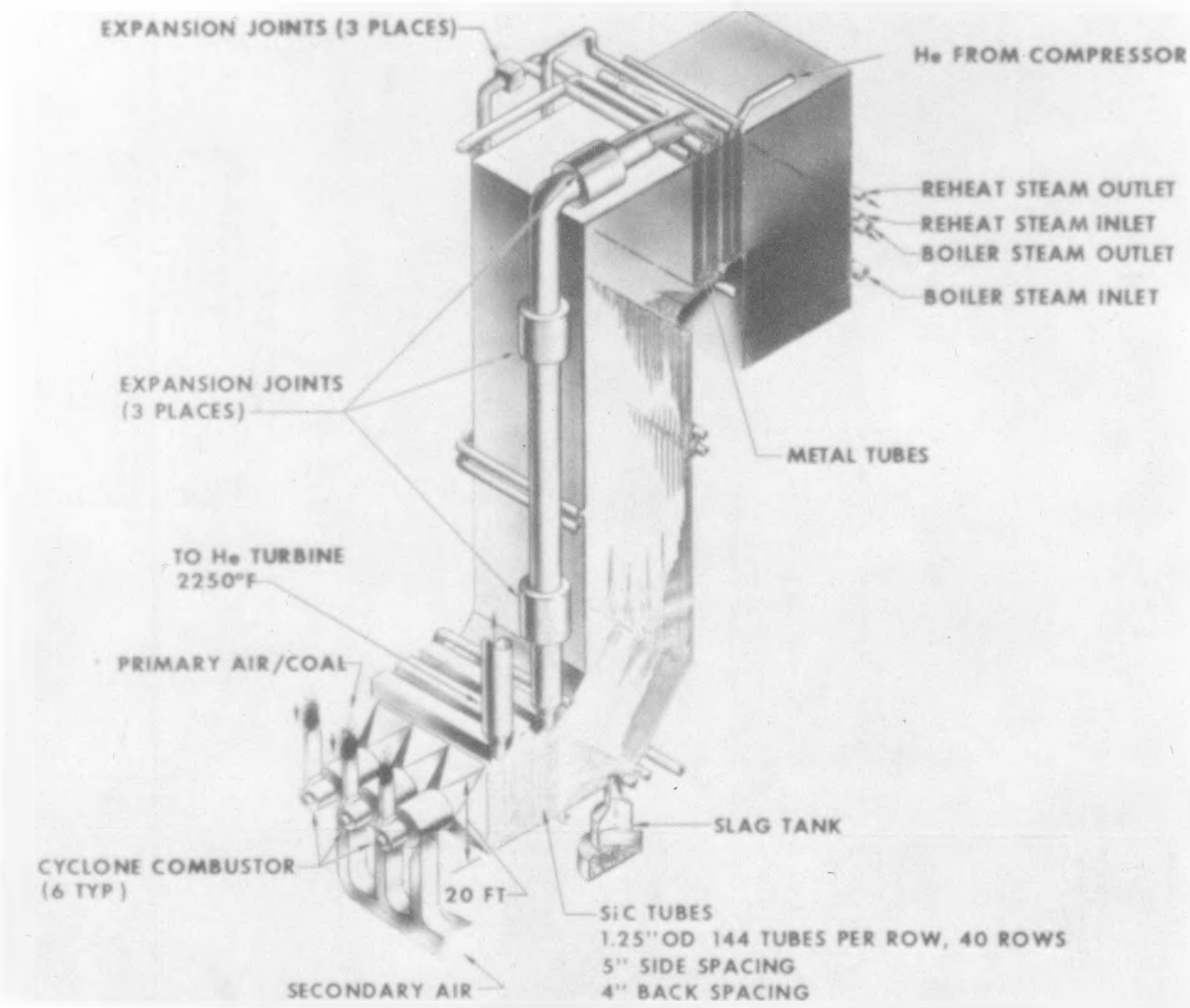

Figure 6.4.2

off to the slag tank. Approximately 50% of the ash content of the coal is collected this way. The hot combustion gases, after having been cooled to approximately 2600 F, are discharged into the tall radiant furnace and travel up the furnace to the exit at the top, by which time their temperature has been reduced to 2100 F. Conventional metal CCGT and steam working fluid convection heating surfaces are provided in pendant and horizontal banks to cool the gaseous products of combustion to approximately 800 F prior to their discharge to the air heater. Either a tubular or a Ljungstrom type air preheater may be supplied to bring the gaseous products of combustion to 300 F, at which point they will be passed through an electrostatic precipitator and a wet sulphur dioxide scrubber prior to being exhausted to the atmosphere through the smoke stack.

This design is dominated by the requirement for 2250 F working fluid temperature, and the inability of non-slagging combustion concepts to supply that temperature. It is recognized that the slagging combustor has limitations with respect to the coals it will accept, high ash fusion temperature and/or high ash viscosity coal being inappropriate. However, it is believed that all of the groundrule coals can be acceptably combusted with the exception of the high ash fusion temperature eastern bituminous coal represented by the Pennsylvania coal. A more detailed description of the preliminary design and its rationale is presented by category below.

6.4.1 Combustion System and General Arrangement

The cyclone combustor concept has been explained in Section 4.7.1. It is well developed and no unusual demands are made upon its application to this combustor/heat exchanger.

Two different high temperature ceramic heat exchanger surface configurations are presented in Fig. 6.4.1, but both operate on the same principle. The principle is that CCGT working fluid is heated from 1550 F to 2250 F in the ceramic heat exchanger and this ceramic heat exchanger is located in a flue gas temperature zone whose exit temperature is 2600 F at full load. The combination of high heat exchanger surface temperature and high operating gas temperature is expected to ensure that the surface of the ceramic heat exchanger tubes will be covered with running slag at all times, and there will be no opportunity to accumulate large deposits of solidified ash which would impede the gas flow. No sootblowers are provided, thus avoiding thermal strain. The 2600 F flue gas leaving the ceramic high temperature section of the combustor/heat exchanger enters the radiant furnace where it is cooled as it flows upwards to the outlet. At the furnace outlet, the combustion gas temperature has been reduced to about 2100 F and is suitable for transferring heat to metal convection surface in a manner similar to that employed in the dry bottom pulverized coal furnace of Section 6.1. Water cooled metal tubes are used to construct the walls of the cyclone and of the enclosure for the ceramic heat exchange surface. Heat transferred to these metal walls is minimized by employing a "studded tube" construction which places a coating of slag resistant refractory between the metal wall and the running slag. As the gas temperature has been reduced to 2600 F prior to entering the radiant furnace, which has CCGT working fluid cooled walls, it is not necessary to employ exhaust gas recirculation to moderate and equalize the heat transfer rates in the radiant furnace. As a consequence of the reduced gas flow rate over the pendant and horizontal convection surface, it is possible to employ a furnace and gas path width of only 60 feet as compared to the 70 feet chosen for the dry bottom PC design of Section 6.1. Overall design specifications are presented in Table 6.4.1.

6.4.2 Heat Transfer and Pressure Drop Considerations

The heat energy available from the combustion gases and required by the various working fluids is illustrated in Fig. 5.3.1. Computation of heat transfer in the slagging regions of the combustor/heat exchanger was based upon temperature/surface relationships available in "Steam", as was the computation in the radiant

Table 6.4.1 Design Specifications
2250 F Cyclone System

	P (psia)	T (F)
He Heater Inlet	1000	644
He Heater Outlet	955	2250
Steam Boiler Inlet	2640	676
Steam Boiler Outlet	2480	664
Steam Reheat Inlet	480	638
Steam Reheat Outlet	432	1050
He Flowrate	934,000 lb/hr	
Steam Flowrate	1,050,000 lb/hr	
Coal Flowrate	266,000 lb/hr	
Air Flowrate	2,410,000 lb/hr	
Air Preheat Temperature	725 F	

furnace. Heat transfer in the convection surface was based upon the same relationship utilized in the dry bottom pulverized coal designs. The 99 mol % helium/1 mol % oxygen working fluid was selected for this preliminary design for the same reasons that governed its utilization on the dry bottom 1550 PC unit, the difficulty of providing adequate cooling in the radiant furnace region with less powerful coolants than helium. A tabulation of the required surface areas and tube parameters is presented in Table 6.4.2. Gas side cleanliness in the radiant furnace and convection surface will require the same tube spacing and sootblowing procedures that were outlined in Section 6.1.2 for the dry bottom PC preliminary design. It is likely that a heavy concentration of wall blowers will be necessary in the region of transition from slagging to dry ash.

6.4.3 Control

Slag tap furnaces have typically encountered difficulties with tapping of slag at lower loads. This preliminary design endeavors to maintain running slag on the ceramic surfaces by designing for 2600 F exit gas temperature from that cell at full load, and since the heat that must be extracted from the gases by the ceramic section is directly proportional to load, and the water cooled boundary surface is a relatively small portion of the total surface, it should be possible to maintain high gas temperatures leaving the ceramic slagging zone even as load drops. The consequences of any accumulation of solidified slag in this area might be fairly severe as the flow areas could easily be plugged and the weight might be hazardous to the integrity of the ceramic. A counterflow arrangement of CCGT working fluid versus combustion gas is shown for the ceramic heat exchanger section in order to minimize the amount of ceramic surface required. More thorough future analyses may possibly indicate that the surface should be arranged for parallel flow to help ensure running slag conditions at all times.

Table 6.4.2 Tube Materials
2250 F Cyclone System

Location	Material	T max F	Diameter	Length
Steam Boiler Convection	C. Steel	700 F	2.50 inch	123,000 ft
Steam Boiler Wall	C. Steel	700 F	2.50 inch	36,000 ft
Steam Reheat Convection	1/2Cr-1/2Mo	900 F	2.50 inch	35,000 ft
	2-1/4Cr-1Mo	1025 F	2.50 inch	19,000 ft
	304 CRES	1075 F	2.50 inch	8,000 ft
Helim Convection	C. Steel	800 F	2.50 inch	33,000 ft
	1/2Cr-1Mo	875 F	2.50 inch	19,000 ft
Helium Wall	INCO 800	1500 F	2.50 inch	98,000 ft
Helium Convection	INCO 617	1575 F	1.75 inch	82,000 ft
	SiC	2300 F	1.25 inch	109,000 ft

There are only two working fluid temperatures which must be controlled, CCGT working fluid at 2250 F and the steam reheat at 1050 F. CCGT working fluid temperature will be controlled by proportioning coal and air input to CCGT working fluid flow. The steam reheat temperature will be controlled by the method discussed for the 1550 F dry bottom PC unit, i.e., a combination of spray attemperation and the provision of dampers to adjust the proportion of combustion products passing over the steam reheat and the CCGT working fluid convection surfaces.

6.4.4 Emissions

Sulphur oxide emissions are controlled by a wet scrubber and particulate emissions by electrostatic precipitator. As discussed in Section 4.5.3, nitrogen oxides emissions may prove particularly troublesome with cyclone combustor firing. The tentative nitrogen oxide emissions control visualized is the utilization of small quantities of gaseous ammonia in the air heater exhaust.

6.4.5 Materials of Construction

Selection and application of metal materials of construction was governed by the same criteria discussed above for dry bottom pulverized coal design. The same apprehensions with respect to the corrosion rate on metal materials operating in the bell curve region, i.e., from about 1100 to 1350 F, apply to their design.

The ceramic materials will operate at temperatures up to approximately 2300 F and will be continuously exposed to a running slag condition at that temperature. This application will presumably be considerably more severe than the application to the fluidized bed concept discussed in Section 6.3 . The application is obviously a challenging one for any material. The design choice is sintered or CVD silicon carbide. It seems evident that considerable experimental verification of the suitability of this material will be required prior to an extensive commitment.

6.4.6 Design Details

Many of the design details will share the characteristic of the 1550 F dry bottom PC preliminary design in that they may be taken directly from acceptable steam boiler practice. The cyclone combustors and the boundary walls of the ceramic heat exchanger section are studded boiler tubes no different from conventional boiler practice. The specific design details utilized to guard against the accumulation of excessive tensile stresses in the ceramic heat exchanger material deserve great care. Two concepts are shown on Fig. 6.4.1 The principle concept, seen in views AA, CC, and DD, provides short self-supporting serpentine loops of ceramic tubing whose design principle is similar in some respects to that described in Section 6.3.6 above for the serpentine loops of the 1750 F preliminary design. Note however that in the Fig. 6.4.1 concept, the ceramic tubes will be brought out through the water cooled walls of the flue gas tunnels that are provided between the cyclone exits and the common radiant furnace. These wall penetrations may well present some difficulties. The serpentine ceramic loops themselves, however, are calculated to be self-supporting and sufficiently flexible to avoid the accumulation of excessive thermal strain. In keeping with the principles discussed under Safety below, the exterior ceramic headers would be enclosed in sufficiently strong metal enclosures that their fracture would not endanger life. The alternate ceramic heat exchanger concept shown on Fig. 6.4.1 provides a single large rectangular gas passage between the cyclone exits and the radiant furnace. Pendant ceramic loop tubes are suspended in the gas passage. Their inlet, intermediate, and outlet manifolds are recessed into the roof of the gas passage. In this manner, all ceramic pressure parts are contained within the water cooled passage walls, and the necessity for a large number of ceramic penetrations of the water cooled walls, with all their possible difficulties of leakage and thermal strain, is avoided. Additionally, only half the number of cyclone combustors are required as with the baseline design. A major question with the alternate pendant loop design is the possibility of problems due to vibration and knocking together of the loops. The available design response to this challenge is the provision of uncooled ceramic spacer bars which are held by gravity in the bottoms of each loop.

7.0. COSTING OF COMBUSTOR/HEAT EXCHANGERS

Cost estimates were prepared for each of the four preliminary combustor/heat exchanger designs. Items costed under the category of fired heat exchangers include the fired heaters themselves plus some auxiliary equipment. The auxiliary equipment includes burners, air heaters, soot blowers, air and gas ducting, structural steel, platforms, and I.D. and F.D. fans. Cost estimates for the fired heaters are based on material cost, shop labor cost and field erection cost. Costs for external insulation, instrumentation and controls, pulverizers, coal and ash handling, etc., are included in the B.O.P. estimates. The cost of the cyclone separators for the fluidized bed systems is included in "other equipment" in the B.O.P. costs. All costs are in January 1978 dollars and contingencies are not applied at this point.

All cost estimates are for the "Nth item" production unit after all necessary technology has been developed and well proven in pilot and full-scale plants. Amortization of research and development costs is not included.

In addition to the four preliminary design systems, a 1450 F, AFB system was priced in order to determine the potential cost advantage of lowering temperature far enough to eliminate the very expensive super-alloy and clad materials. Also, a comparison of the costs of all of the fired heat exchanger systems with a conventional 350 MWe P.C./steam system is presented.

The cost estimates for the four systems plus the low-temperature AFB system are presented in Tables 7-1 through 7-5. They are compared with a conventional pulverized coal single reheat steam system in Table 7-6.

Boiler manufacturers do not generally "break down" their bids. We were fortunate enough, however, to obtain the cost breakdown estimates of two large boiler manufacturers on a 350 MWe coal-fired, 2400 psi/1000 F/1000 P boiler. The breakdowns are included as Tables 7-7 and 7-8. They provide some insight on the makeup of costs in a typical steam boiler installation. The

Table 7.1 1550F/1550F PC System - Fired Heat Exchanger Cost - 350 MWe - Drawing No. AP78-075

Pressure Parts	$21,865,000
Structural Steel + Platforms	3,000,000
Air Heaters	1,539,000
Soot Blowers	750,000
Air + Gas Ducting	1,750,000
I.D. + F.D. Fans	1,300,000
Freight	396,000
Erection	9,000,000
	$39,600,000

Table 7.2 1550F/1550F AFB System - Fired Heat Exchanger Cost - 350 MWe
Drawing Nos. AP78-007, AP78-026, AP78-032

	4 High Pressure Modules	4 Low Pressure Modules	4 Carbon Burnup Cells	Total
Pressure Parts	14,045,000	7,612,000	770,000	22,427,000
Casing, Refractory + Misc	627,000	627,000	48,000	1,302,000
Injection Needles, Air Distribution Grating	218,000	218,000	76,000	512,000
Support Steel and Platforms				400,000
Air Heaters				1,474,000
Air & Gas Ducting				2,360,000
I.D. and F.D. Fans				1,600,000
Freight				378,000
Erection				7,327,000
	Total Fired Heat Exchanger Cost			37,780,000

Table 7.3 1450F/1450F AFB System - Fired Heat Exchanger Cost - 350 MWe Proportioned from 1550F Design

	4 High Pressure Modules	4 Low Pressure Modules	4 Carbon Burnup Cells	Total
Pressure Parts	6,540,000	6,159,000	770,000	13,469,000
Casing, Refractory + Misc	627,000	627,000	48,000	1,302,000
Injection Needs, Air Distribution Grating	218,000	218,000	76,000	512,000
Support Steel & Platforms				400,000
Air Heaters				1,474,000
Air and Gas Ducting				2,360,000
I.D. and F.D. Fans				1,600,000
Freight				378,000
Erection				6,709,000
		Total		$ 28,204,000

Table 7.4 1750F/1550F Mixed Mode System - Fired Heat Exchanger Cost - 350 MWe - Drawing No. AP78-035

	4 High Temperature Modules	4 High Pressure Modules	4 Low Pressure Modules	4 Carbon Burnup Cells	Total Total
Metal Pressure Parts	677,000	12,300,000	7,420,000	770,000	21,167,000
Ceramic Pressure Parts	2,550,000	N/A	N/A	N/A	2,550,000
Casing, Refractory + Misc.	471,000	627,000	627,000	48,000	1,773,000
Injection Needles, Air Distribution Grating	94,000	218,000	218,000	76,000	606,000
Support Steel & Platforms					560,000
Air Heaters					1,448,000
Air and Gas Ducting					2,950,000
I.D. and F.D. Fans					1,800,000
Freight					435,000
Erection					9,911,000
			Total		$ 43,200,000

Table 7.5 2250F Cyclone System - Fired Heat Exchanger Cost - 350 MWe
Drawing No. AP78-028

Pressure Parts	19,266,000
Structural Steel & Platforms	3,000,000
Air Heaters	1,474,000
Soot Blowers	750,000
Air & Gas Ducting	2,000,000
I.D. and F.D. Fans	1,400,000
Freight	400,000
Erection	11,270,000
Total	39,560,000

Table 7.6 Fired Heat Exchanger Cost Comparison - 350 MWe System

	Conventional Steam 2400 psi 1000/1000	1550F/1550F PC	1550F/1550F AFB	1450F/1450F AFB	1750F/1550F Mixed Mode	2250F Cyclone
Pressure Parts	14,525,000	21,865,000	22,427,000	13,469,000	23,717,000	19,266,000
Casing, Structural Steel, Platforms + Misc.	3,000,000	3,000,000	2,214,000	2,104,000	2,939,000	3,000,000
Air Heaters	1,680,000	1,539,000	1,474.000	1,474,000	1,448,000	1,474,000
Soot Blowers	750,000	750,000	N/A	N/A	N/A	750,000
Air & Gas Ducting	1,750,000	1,750,000	2,360,000	2,360,000	2,950,000	2,000,000
I.D. + F.D. Fans	1,420,000	1,300,000	1,600,000	1,600,000	1,800,000	1,400,000
Freight	325,000	396,000	378,000	378,000	435,000	400,000
Erection	9,000,000	9,000,000	7,327,000	6,709,000	9,911,000	11,270,000
Total	32,450,000	39,600,000	37,780,000	28,204,000	43,200,000	39,560,000

Table 7.7 Estimated Cost Breakdown
350 MWe Coal-Fired Power Boiler

Boiler Manufacturer B

Item	Percent of Total	$X1000
Pressure Parts	35.9	$12,500
Air Preheaters	6	2,000
Pulverizer Systems	7	2,500
Soot Blowers	2	750
Refractory	0.1	50
Coal Piping (Starts at Bunkers)	1.4	500
Structural Steel and Platforms	9	3,000
Insulation and Lagging	1	300
Supervision and Start-Up Services	1	300
Freight	3.6	1,300
Controls (Burner Management Only)	2	800
Erection	31	11,000
Total	100	$35,000

Total Cost $35,000,000* erected

*This price includes coal pulverizers and piping to the burners. Erection cost based on 1978 labor rates.

Table 7.8 Estimated Cost Breakdown
350 MWe Coal-Fired Power Boiler

Boiler Manufacturer A

Item	Percent of Total	$X1000
Structural Steel	8	$ 2,680
Pulv. Equip. & Coal Piping	7	2,350
Air Preheaters	5	1,680
Soot Blowers	1	330
Freight	2	670
Boiler Proper	46	15,400
Erection	31	10,390
Total	100	$33,500

Total Cost $33,500,000* erected

*This price includes coal pulverizers and piping to the burners. Erection cost based on 1978 labor rates.

pressure parts at the manufacturer's plant amount to about one-third the installed cost of the entire boiler. The major differences between CCGT combustor/heat exchanger costs and steam boiler costs lie in the pressure part category. The CCGT heater has no drums, but it does have many manifolds, and much expensive alloy heating surface, so in general, the pressure parts are more costly than the equivalent boiler design. The 1978 costs of tubing materials used in this study are shown in Table 7.9.

A comparison of the conventional steam and 1550/1550 P.C. CCGT combustor/heat exchanger shows that the main cost difference lies in the pressure parts. The 1550/1550 P.C. is more expensive because of its use of larger quantities of high alloy metals , as seen in Table 7-10. A large percentage of the weight of conventional steam boiler, furnace walls, etc., is low-cost carbon steel. The major portion of the 1550/1550 P.C. CCGT unit is high alloy steels, 304 stainless, Incoloy 800, and the very expensive Inconel 617. As can be seen from Table 7-11, 50 percent of the cost of the conventional unit lies with low-cost carbon steel, while 52 percent of the cost of the 1550/1550 P.C. unit is in the high-cost Inconel 617. The high operating temperature of the Brayton cycle requires high-cost materials which result in an estimated 50 percent increase in pressure parts cost, as compared to the conventional steam boiler.

A similar comparison exists with the 2250 cyclone unit. Almost half of the material is high alloy or SiC, resulting in 91 percent of the pressure part costs being for these materials. The air and gas ducting cost is higher because of the multiple cyclone burners that must be manifolded to. The erection cost is higher due to the ceramic heat exchanger. Costs of air heaters and fans change slightly due to capacity differences.

A comparison of the 1550/1550 AFB and P.C. units shows almost the same pressure parts costs, with both units containing about the same total percent of high-alloy material as seen in Table 7-11. A large portion of the material cost, 55 percent, of the AFB system is in the Inconel 617 and Inconel

Table 7.9 Basic Tubing Material Costs (1978 Values)

Material	Purchased Cost, \$/lb
Carbon Steel SA210-A1	0.50
1/2 Cr-1/2 Mo SA213-T2	0.80
2-1/4 Cr-1 Mo SA213-T22	1.00
304H SA213-TP304H	2.50
Incoloy 800H SB-163	3.25
Inconel 617	10.00
Inconel 617 + 310 Cladding	10.00
SiC	7.00

Table 7.10 Materials Weight Comparison
Pressure Parts Only, weight percent

	Conventional Steam	1550F/1550F P.C.	1550F/1550F AFB	1450F/1450F AFB	1750F/1450F Mixed Mode	2250F Cyclone
C. Steel	71	24	15	37	24	46
1/2Cr-1/2Mo (T2)	9	8	4	11	3	5
2-¼ Cr-1Mo (T22)	15	12	25	10	20	2
304 Stainless	5	10	12	20	14	1
347 Stainless	--	--	20	22	16	--
Incoloy 800	--	29	--	--	--	27
Inconel 617	--	17	12	--	11	14
Inconel 617 + 347 Cladding	--	--	11	--	9	--
SiC	--	--	--	--	3	5

Table 7.11 Materials Cost Comparison
Pressure Parts Only, Dollar Percent

	Conventional Steam	1550F/1550F P.C.	1550F/1550F AFB	1450F/1450F AFB	1750F/1450F Mixed Mode	2250F Cyclone
C. Steel	50	4	3	5	5	7
1/2Cr-1/2Mo (T2)	10	2	1	6	1	1
2-1/4Cr-1Mo (T22)	22	4	7	7	6	1
304 Stainless	18	8	8	37	10	.5
347 Stainless	--	--	16	45	13	--
Incoloy 800	--	30	--	--	--	27
Inconel 617	--	52	33	--	31	42
Inconel 617 + 347 Claddin	--	--	32	--	27	--
SiC	--	--	--	--	8	22

617 clad tubing required for the high-temperature manifolds and bed tubes. For the AFB systems, the items included under casing, structural steel, platforms, etc. in Table 7-6 include the following items shown on the sheets for the individual systems: casing and refractory, injection needles and air distribution grating, support steel and platforms. This item is less for the 1550 AFB system than for the P.C. because less steel is required to support the module stacks which are not suspended from 200 feet above the floor as with the P.C. unit. Also, less steel is required to support the short spans of the AFB module walls as compared to the very large P.C. furnace walls. Air and gas ducting cost is higher for the AFB system because of the 12 modules that must be ducted to and from.

Since a large amount of the cost of the 1550/1550 AFB modules is in the Inconel 617 and clad materials, a lower temperature design was priced to see if a more cost-effective design would result. The temperature chosen was 1450 F for the Brayton cycle. The highest alloy required, as seen on Tables 7-10 and 7-11, is 347 stainless. The resulting pressure parts cost is roughly two-thirds that of the 1550 F system, with a total savings for the fired heat exchanger items of $9,000,000.

The 1750/1550 mixed mode costs are similar to the 1550/1550 AFB except for the addition of four high-temperature ceramic tube modules and their associated ducting, fans and support structure. The pressure parts cost is higher than the 1550 AFB system even though the 1650 F operating temperature high-pressure and low-pressure modules are slightly less expensive. The cost of the support steel and platforms and air and gas ducting increases to accommodate the four high-temperature modules. Additional fans are required to bring the exhaust pressure from the high-temperature modules back up for reinjection into the high-pressure and low-pressure beds for sulfur removal. Erection cost is also higher due to the additional high-temperature modules with their ceramic pressure parts.

8.0 RESULTS

The purpose of this section is to discuss the results of the design studies, and the conclusions that may be drawn therefrom. It is the general results of the design studies that are discussed in this section. Specific results in the form of an analysis of the key features of each of the preliminary designs created during the course of the contract are contained in report PCS 78-08, the "Key Features Analysis". The key feature analyses constitute a detailed critique of each of the preliminary designs. The key features material will not be repeated here.

The cycle operating conditions for which preliminary designs were to be undertaken were purposely chosen to cover a wide range of maximum CCGT working fluid temperature, from 1550 F to 2250 F. This wide range was chosen so as to provide for evaluation of designs to cover the maximum range of possible interest, and not necessarily from a belief that applications for CCGT stations operating over this whole temperature range will be immediately forthcoming. While the contract required only that applications at 1550 F and above be examined, some attention was also given to the possible application of the designs to working fluid temperatures of 1450 F and lower. During the course of the study combustor/heat exchanger design concepts were synthesized to cover the full range of target working fluid operating temperatures, and combustor/heat exchanger preliminary designs, each believed to be fundamentally technically viable, were created for 1550 F, 1750 F, and 2250 F maximum working fluid temperature operating cycles.

The provisions of the contract permitted the use of coal derived fuels, either liquid or gaseous, for those designs providing 1750 F or higher CCGT working fluid temperature. The conclusion was reached early in the design study that the utilization of gaseous or liquid coal derived fuels would be inappropriate in any of the combustor/heat exchangers operating at maximum CCGT working fluid temperatures of up to 2250 F. The energy losses entailed in manufacturing really clean coal derived fuels are greater than the efficiency gains, (relative to direct coal fired steam) available from CCGT cycles at up to 2250 F.

An examination of the economics of combustor/heat exchangers with duty requirements similar to those under study indicated that at a 350 MWe unit size, the least expensive combustor/heat exchanger installation, with respect both to first cost and operating costs, was a single large combustor/heat exchanger rather than several smaller combustor/heat exchangers which together made up the required total 350 MWe capacity. The design effort was guided by this relationship. In the case of pulverized coal and cyclone fired combustor/heat exchangers we were successful in meeting the combustor/heat exchanger operating requirements with a single unit. In the case of fluidized bed heat exchangers, this economic relationship resulted in minimizing the number of discrete fluidized bed modules required. The fluidized bed preliminary designs however may well be faulted for involving an excessive number of discrete modules, and it is expected that future design trends will be in the direction of reducing the number of those modules.

The work conducted under the cycle analysis and working fluids studies brought out that there are several strong candidate working fluids, including helium, helium/carbon dioxide mixtures, and air or nitrogen. There are small differences in the costs of the working fluids themselves and in the storage and handling facilities. Additionally, there are substantial differences in optimum pressure ratios, working fluid state pressures, and working fluid flowrates. The choice of working fluid impacts the design of the combustor/heat exchanger in that the various fluids provide different relationships between heat transfer coefficient and pressure drop. This influences the amount of heating surface that must be provided, and its operating temperature. This study did not attempt to provide a complete optimization of working fluid choice. That optimization would basically involve complex economic trade-offs between stations designed for each working fluid, including the working fluid effects upon the fired and unfired heat exchanger costs, the turbo-machinery costs, the CCGT working fluid ducting costs, the storage and handling facilities costs, the costs of initial supply and make up of the working fluids themselves, and the differentials in cycle operating conditions which produce differentials in coal consumption. Additionally, subtle variations in corrosion of CCGT plant equipment on the working fluid side may exist. The choice among working fluids was simplified by the assumption that the differentials in economic feasibility of the overall CCGT/Rankine cycle, as affected by the selection of helium, a helium, CO_2 mixture, or an air working fluid, would be relatively small. On this basis, helium working fluid was chosen for those combsutor/heat exchangers which involved radiant furnaces, as it was felt that helium provided substantially higher working fluid side heat transfer coefficients, and smaller overall combustor/heat exchangers. Additionally, there was less hazard that furnace circuit heat input unbalances would result in dangerously excessive metal temperatures. Helium/CO_2 mixture was chosen for fluidized bed heat exchanger concepts because it provided slightly improved overall cycle efficiency with only a small penalty in additional heat exchanger surface. The nature of the fluidized bed operation is such as to minimize the danger of excessive overheating of the highest working fluid temperature circuits. The preliminary designs were based upon the operating conditions suitable for the working fluid choices. On examining the preliminary designs after their completion, it is evident that the fundamental concepts involved can be adapted to any of the three prime working fluid choices, with varying degrees of impact. The fluidized bed designs, which are based upon the 60% helium, 39% CO_2, 1% oxygen mixture, could be adapted to either helium or air with relatively small changes in surface and circuitry. The pulverized coal and cyclone combustor/heat exchangers, which are designed for the 99% helium, 1% oxygen mixture, could be adapted to the helium/CO_2/oxygen mixture, or air. While significant rerouting of circuitry would be involved, along with some additional heating surface, the fundamental technical and economic viability of the concepts would be relatively unaffected. Thus, the position of the preliminary designs relative to working fluid choice is that while the working fluid selections made are believed to be technically and economically defensible, the preliminary designs of the combustor/heat exchangers themselves are adaptable to other working fluid choices should such changes be indicated by more thorough optimization studies in the future.

The design study effort was successful in synthesizing what are believed to be technically viable preliminary designs for CCGT working fluid temperatures from well below 1550 F to as high as 2250 F. The study indicated that a wide choice of viable combustor/heat exchanger concepts is available in the lower operating temperature range, up to approximately 1550 F. For this range, two preliminary design concepts were synthesized, one based upon non-slagging pulverized coal combustor technology and the other based upon the emerging technology of coal combustion in atmospheric fluidized beds. Both concepts appear to be technically feasible, although much remains to be learned with respect to the combustion of coal and the absorption of sulphur in fluidized beds. Insofar as first cost and operating costs of the combustor/heat exchangers themselves are concerned it seems likely that the two concepts are roughly equivalent. The advantage of the fluidized bed concept lies almost entirely in its anticipated capabilities for eliminating the requirement for the capital cost and operating expense of tail-end sulphur oxides scrubbing.

As the CCGT working fluid temperature is increased, to 1750 F for instance, the choice among available combustion concepts is narrowed. This is because the higher working fluid temperature requires higher heat exchanger surface temperature, and this affects the combustion products side of the operation. With fluidized bed combustors for instance, it is believed that it will not be practical to extract sufficient sulphur-Oxides from the combustion products in the bed (with economically acceptable limestone to coal feed ratios), if the bed temperature is much above 1650 F or 1700 F. With non-slagging pulverized coal combustors, the surface temperature of the heating surface begins to get uncomfortably close to the ash softening temperature as the working fluid temperature approaches 1750 F. This latter effect, of course, is a strong function of the coal burned. High ash softening temperature coals would be relatively unaffected.

The mixed mode combustor concept was selected for preliminary design effort at the 1750 F operating temperature because it was believed to be suitable for all of the contractually specified coals. The mixed mode concept, as embodied in the 1750 F preliminary design, provides that about 20 percent of the coal is combusted in a 2000 F operating temperature fluidized bed, with the heat released in that bed being absorbed by CCGT working fluid as it is heated from 1550 F to 1750 F. The remainder of the coal is combusted in 1650 F operating temperature fluidized beds, which provide heat input for CCGT working fluids and other cycle working fluids at temperatures below 1550 F. Additionally, all of the gaseous products of combustion resulting from the 2000 F bed operations are passed through the 1650 F operating temperature beds to provide for absorption of the sulphur oxides contained therein. This "mixed mode" combustor/heat exchanger concept may be visualized as an extension of the fluidized bed combustion technology necessary for the 1550 F CCGT operating temperature designs, perhaps following after the development of the lower temperature technology.

The design effort brought out that both the technical and economic success of the CCGT combustor/heat exchangers will be strongly dependent upon metals technology. The examination of available metals materials of construction indicates that the 1550 F CCGT working fluid temperature is about the maximum that can reasonably be expected to be provided with metal heat exchanger surface. The available design stress falls off much too rapidly at higher temperatures. This judgment was based upon examining the available metals, and did not include the probability that some miracle metal would be invented which would drastically alter this picture. The strongest contenders for application to the combustor/heat exchangers were found to be the conventional heat exchanger metals, i.e., carbon steel and low alloy chrome moly steels at the lower operating temperatures and austenetic stainless steel, Incoloy 800, Inconel 617 and Inconel 671 as the temperatures approach the limiting 1550 F. The factors influencing the choice of metal materials of construction included working fluid side corrosion effects, combustion products side corrosion effects, long term resistance to creep and stress rupture effects at operating temperature, fabricability and cost.

The subject of metals applications is discussed at some length in the Key Features and R&D reports and it may be briefly stated here that it is evident that much development effort will be required in the area of combustion products side corrosion effects versus economics prior to extensive commerical applications of CCGT technology. It is possible that the corrosive environment for high temperature metals will be found to be more benign with the fluidized bed combustor/heat exchanger concept than it is with the pulverized coal combustor. This could well be an added economic benefit for the fluidized bed concept in that it may be possible to survive with much cheaper alloys.

The metals chosen for application in each preliminary design are believed in each case to be the least expensive material suitable for the service. Metals properties affected the technically feasible tube sizes that went to make up the heating surface and boundary walls but did not significantly affect the overall configuration of the final combustor/heat exchanger designs.

As metals materials of construction are believed to be unavailable for working fluid temperatures much above 1550 F, the only alternative available for those temperatures appears to be the "ceramic" materials of construction. The investigation of available ceramics identified only the silicon carbide based materials as viable candidates for the coal fired CCGT combustor/heat exchanger system. The overriding consideration in designing with ceramic heat exchanger materials of construction was found to be their complete lack of ductility, and the necessity to provide configurations that never result in exceeding the available tensile strength under any possible circumstance of pressure stress, thermal strain, vibration strains, etc. An additional design consideration was the brittle nature ceramic pressure containment surfaces. When failures do occur there is a sudden release of pressure and possible break-up of the pressure containment surface and flying about of the pieces. These two considerations did significantly affect the configurations of the combustor/heat exchangers containing the ceramic heating surface. The basic decision was taken to protect against operator injury and possible exterior property damage by ensuring that all ceramic heat exchanger surface is contained within metal walls, and is also not essential to the structural integrity of the overall heat exchanger. Ceramic materials were not utilized for the containment of the products of combustion, and their application was such that ceramics failures would not result in collapse of the combustor. The constraints on heating surface configuration imposed by the absence of ductility in ceramics are seen in the arrangement of the manifolding and tube loops of the mixed mode and cyclone combustor designs.

The 2250 F preliminary design further illustrates the narrowing of combustor/heat exchanger concept choices that occurs with increasing working fluid temperature. At that temperature the heating surface will be operating at approximately 2300 F, which is well above the fusion temperature of the specified coals. Under these conditions, fluidized bed combustion,even without regard to sulphur oxides absorption, is believed to be impractical because of clinkering in the bed. Non-slagging pulverized coal combustor operation is impractical because the high temperature surfaces will inevitably accumulate running slag. Thus, the choice appears to be narrowed to the slagging type combustor. Slagging combustor operation has been well established for many years. The choice made for the 2250 F preliminary design is the slagging cyclone combustor, which has the advantage of minimizing the quantity of slag forming material carried over on to the convection heating surface. The underlying rationale of the 2250 F slagging cyclone combustor preliminary design is the provision of tubular ceramic convection surface to operate in the temperature region between the adiabatic flame temperature and 2600 F. This surface is expected to be continually covered with a layer of running slag, making sootblowing and the thermal strains it involves unnecessary. The basic safety principle of surrounding any ceramic surface with a safety shield of metal, and avoiding strucutral dependence upon ceramic is carried out in this design as well as the 1750 F design.

The study groundrules set the maximum permissible emissions rates, sulphur oxides, nitrogen oxides, and particulates, at no more than the values presently specified by the Federal Environmental Protection Agency. The two preliminary designs handling 1550 F CCGT working fluid appear well capable of meeting the limitations. The pulverized coal design will meet NOX limitations by a combination of burner adjustment and flue gas recirculation, sulphur oxides emissions by wet limestone scrubbing, and particulates emissions with an electrostatic precipitator, all reasonably well proven techniques. The 1550 F fluidized bed preliminary design will meet emissions requirements with respect to SOX and NOX by combustion in a relatively low temperature limestone bed. This technology must be regarded as unproven but is so central to the economic viability of the fluidized bed combustion concept that one can assume that if fluidized beds are to be used at all, they will be able to meet the SOX and NOX requirements. Particulates emissions of the fluidized bed combustor/heat exchanger are controlled via baghouse, and this is reasonably well proven technology. The 1750 F preliminary design requires a portion of the combustion to take place at bed temperatures higher than can be reasonably expected to control SOX emissions. The design expedient of passing the high temperature combustor effluent gases through the bed of the low temperature combustors sounds logical but needs experimental confirmation. The slagging cyclone combustor designed for the 2250 F application is expected to exceed the NOX emissions limitation as designed. In proposing this design reliance is placed upon the concurrent development of NOX abatement concepts that employ the addition of ammonia to the heat exchanger exhaust flue gases to oxidize the NOX content of the flue gas to nitrogen and oxygen. While such technology is under development, there are no present guarantees that it will be both technically successful and economically viable. An alternative NOX control method for such slagging combsutors is to design the combustor for staged combustion, a slight deficiency of air being utilized for the high temperature combustion section with the air required to make up the deficiency added to the combustion products after their temperature has been substantially reduced by heat transfer to the heating surface. This concept has considerable promise for NOX control but was not incorporated in the preliminary design because of added complications with fire side corrosion of the heat exchanger material.

Cost estimates were prepared for each of the combsutor/heat exchanger preliminary designs and comparisons were made with the costs of a 2400 psi/1000 F/ 1000 F steam boiler sized for 350 MWe output. The CCGT combustor/heat exchangers, when compared to the steam boiler, showed modestly less cost in many auxiliary areas, i.e., pulverizers, burners, air heaters, fans, etc., because they burned less coal. The combustor/heat exchanger pressure parts however were more expensive, principally because the higher operating temperatures of the CCGT equipment required more expensive heating surface materials to resist those temperatures. The metals costs used in the study are believed to be realistic. The ceramics costs are much less predictable. The 7 $/lb cost assumed for SiC tubes in an optimistic prediction based on inputs from potential suppliers. It is justified by the argument that the basic materials are cheap and finished costs must be reasonable if the material is to be competitive. The "pressure parts" of the CCGT combustor/heat exchangers were found to be more than 50 percent more expensive than the pressure parts of the steam boiler. The net effect is that the 350 MWe CCGT heat exchanger is anticipated to be on the order of 10-20 percent more expensive overall than the 350 MWe steam boiler. As the combustor/heat exchanger is only a relatively small portion of the cost of the entire generating station, on the order of 20 percent, it seems unlikely that the cost of the CCGT combustor/heat exchanger will be the sole determining factor with respect to the economic viability of coal fueled CCGT/Rankine generating systems.

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