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Coal-Fired Alkali Metal Power System Design Study

Technical Progress Report for Period January 1, 1977-March 31, 1977

Garland Samuels

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OAK RIDGE NATIONAL LABORATORY

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COAL-FIRED ALKALI METAL
POWER SYSTEM DESIGN STUDY

TECHNICAL PROGRESS REPORT FOR PERIOD
JANUARY 1, 1977 - MARCH 31, 1977

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COAL-FIRED ALKALI METAL POWER SYSTEM DESIGN STUDY
TECHNICAL PROGRESS REPORT FOR PERIOD
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ABSTRACT

This report is the second of a series of quarterly progress reports for the Coal-Fired Alkali Power System Design Study. The primary effort during this period was the parametric cycle analysis of various plant arrangements with both cesium and potassium as the working fluid, and the design and analysis of a vertical tube natural circulation boiler for a pressurized fluidized bed furnace.

SCOPE OF THE DESIGN STUDY

The objective of the Coal-Fired Alkali Metal Power System Design Study is to establish a reference design for a 200 MW(e) alkali metal vapor/steam Rankine cycle system employing a coal-fired fluidized bed furnace and to prepare a preliminary design of a fluidized bed furnace. This will be accomplished by studying the relative merits of both potassium and cesium vapor cycle systems and making a comparison of these systems. The results will be analyzed and potassium or cesium will be chosen as the cycle fluid and one of the systems will be selected as the reference. The reference system will be used for the preparation of a preliminary design of a fluidized bed furnace-boiler.

The initial scope of this study was curtailed in late March, 1977. The program is now limited to a study of an atmospheric fluidized bed system using either cesium or potassium as the working fluid. The new schedule also changed the study from the two-phase approach which was to be completed by December 31, 1977 with an interim report to be issued by August 31, 1977. The work on the atmospheric system will be covered

in a single report to be completed by September 30, 1977. The scope of work for the revised program includes the following major tasks:

1. Parametric cycle analysis for cesium and potassium power systems.
2. Conceptual designs for an atmospheric pressure fluidized bed furnace.
3. Condenser-steam generator conceptual designs for cesium and potassium at several condenser temperatures.
4. Metal vapor turbine design and analysis.
5. Conceptual layouts of overall system.
6. Preliminary design of an atmospheric pressure fluidized bed furnace for either a cesium or potassium power system.

PROGRESS TO DATE

The primary effort during the first quarter was the compilation of physical and thermodynamic properties of potassium and cesium, the analysis of a series of potassium and cesium turbine designs, and the determination of the cycle efficiency for both fluids for a variety of operating conditions.¹

The basic thermodynamic data used were those published by Ewing, et al., in NRL Reports 6233 and 6246.^{2,3} However, these data do not cover the lower temperatures needed for our current work, and it was necessary to prepare computer codes to extend the range to lower temperatures of interest. Mollier diagrams were prepared for the temperature range of 1050°R (583°K) to 2300°R (1278°K) for cesium and from 1200°R (667°K) to 2300°R (1278°K) for potassium. The wet region for both diagrams extends down to 28% moisture. Tables of the specific heat, viscosity, and thermal conductivity for both the liquid and vapor, and a table of the sonic velocity were prepared for both working fluids. A draft report documenting data sources and calculational procedures was prepared.

The metal vapor turbine design work was limited to analyses to determine the number of stages, blade height and stress, and the turbine

efficiency. The comparison of the turbine designs and efficiencies for cesium and potassium was based on the decision to make the units nearly identical in size and number of stages. It is possible to take advantage of the smaller enthalpy change across the turbine for cesium to reduce the number of stages as compared to potassium. Although this would result in a somewhat less expensive cesium unit it would also reduce the efficiency of the unit by several points. Rather than take this approach the speed of the cesium unit was made one-half that of the potassium unit which gives machines of nearly the same size and number of stages. This lower speed has a very significant effect on the materials requirements for the blades and disks.

The first stage blade height for a cesium turbine operating at 1800 rpm is about the same as for a potassium unit operating at 3600 rpm. This means that the centrifugal stress on the first stage rotor blades, which is proportional to the square of the speed, is four times as great for the potassium turbine as for the cesium unit. At a rotor temperature of 1850°R (1028°K), this factor of four increase in stress would result in a factor of about 4000 in the creep rate of the material. Furthermore, the required flow area, and thus the blade height, increases much more rapidly for the potassium unit, with the last stage requiring about 1.5 times the area of the cesium unit. As the centrifugal stress is directly proportional to the rotor annulus area, the centrifugal blade stress in the last stage will be about six times greater with the potassium unit. Thus, for the turbine, there are significant advantages in using cesium rather than potassium.

TECHNICAL PROGRESS

The primary effort during this period was a parametric analysis of various plant arrangements with both cesium and potassium as the working fluid, and the design and analysis of a vertical tube natural circulation boiler for a pressurized furnace.

Cycle Analysis

The parametric cycle analysis of the plant was completed. This analysis combines the results of studies of three of the plant systems; the combustion air system, the alkali metal system, and the steam system.

Combustion Air System

Three types of combustion air systems were studied for the plant; a high pressure [6 to 10 atm (0.61 to 1.01 Pa)] system using a conventional gas turbine power unit in which the turbine was driven by high temperature combustion gases, an intermediate pressure [3 to 5 atm (0.30 to 0.50 MPa)] system using a turbocharger in which the turbine was driven by combustion gas at 1000°F (811°K) or less with no electrical power generated by the unit, and an atmospheric system in which the combustion air was supplied by a motor driven blower. Schematic diagrams of these systems are shown in Figs. 1, 2, and 3.

Combustion Analysis. The analysis of the plant combustion system and the furnace efficiency was based on the use of Illinois No. 6 coal as the fuel. This is the fuel being used in the Energy Conversion Alternatives Study (ECAS).⁴ The analysis of this coal is as follows:

Proximate analysis, (as received) percent:

Moisture	13.0
Volatile	36.7
Fixed Carbon	40.7
Ash	9.6

Ultimate analysis (as received) percent:

Ash	9.6
Sulfur	3.9
Hydrogen	5.9
Carbon	59.6
Nitrogen	1.0
Oxygen	20.0

Fig. 1. Flow diagram for a coal-fired alkali metal/steam power plant with a high pressure combustion air system.

Fig. 2. Flow diagram for a coal-fired alkali metal/steam power plant with an intermediate pressure combustion air system.

Fig. 3. Flow diagram for a coal-fired alkali metal/steam power plant with an atmospheric pressure combustion air system.

Higher heating value (as received)

Btu/lb 10,788

For the analysis of the combustion system it was assumed that the coal was dried before entering the furnace, and that the energy required for drying was taken from the energy content of the coal; i.e., the energy content of the coal is reduced by 1060 Btu/lb (2.5 MJ/kg) of water. With these assumptions, the moisture does not become a part of the combustion products. It should be noted that assuming the energy for drying comes from the energy content of the coal will increase the plant heat rate in comparison to removing the moisture with the energy in the stack gases. This difference will vary between the different types of systems and conditions being studied, but will be about 1%. This corresponds to a reduction in the overall plant efficiency of about $1\frac{1}{2}$ point.

Another assumption that was used in the parametric analyses is that the combustion products have the same properties as air; however, the actual weight of the products will be used. The error in this assumption is that the heat capacity, and thus the enthalpy, of the combustion gases is several percent higher than air at the higher temperatures of the cycles. At lower temperatures (a few hundred degrees F) the differences are very small.

In making the stoichiometric analysis and the material and energy balances for the furnace, it was assumed that the amount of combustion air used is 15% greater than that required for complete combustion of the coal, limestone is used to control sulfur emissions with the amount of limestone calcined being twice that required to absorb the sulfur content of the coal (two molecules of CO_2 are released for each molecule of SO_3 absorbed), the combustion of sulfur yields SO_3 , no oxides of nitrogen are formed in combustion, and the carbon is burned to CO_2 . The actual amount of limestone fed to the system will result in a Ca to S ratio of about 2.5. However, the assumption of two molecules of CO_2 for each atom of S limits the amount of CO_2 released from the limestone to twice that required for absorption of the sulfur. This assumption has very little effect on the weight of combustion products produced.

The analysis of the efficiency of the various systems of the plant was based on one pound of incoming air. The key quantities of interest, based on one pound of air, that were derived from the combustion analysis and used throughout the plant analysis are as follows: 1.08 lb of combustion products formed, 1145 Btu in the as received coal, 28 Btu lost from the furnace as heat and unburned carbon, 59 Btu lost as latent heat, 1058 Btu available to alkali metal boiler or combustion gases of which 72 Btu is lost to the stack. Energy diagrams for the three combustion air systems are shown in Figs. 4, 5, and 6. No heat losses from the piping or components (except the furnace) are shown. The total of the losses is about 14% of the heat content of the as received coal.

High Pressure Combustion Air System. The assumptions used in analyzing the high pressure air system are as follows:

Compressor efficiency, %	86
Turbine efficiency, %	89
Ratio of combustion products/air	1.08
Generator plus mechanical losses, %	4
Air inlet temperature, °F	60
Air inlet temperature, °K	289
Recuperator heating effectiveness, %	80
Fluidized bed temperature, °F	1750
Fluidized bed temperature, °K	1228
Turbine inlet temperature, °F	1750-1500
Turbine inlet temperature, °K	1228-1089

The performance of the system was calculated for compressor ratios of 6 to 10, for system pressure losses (the summation of the $\Delta P/P$ throughout the system) of 0.1, 0.2, and 0.3, and for turbine inlet temperatures of 1750°F (1228°K) and 1500°F (1089°K). The turbine and compressor efficiencies represent values attainable with current technology while the recuperator heating effectiveness represents a compromise between system efficiency and component cost.

There are a number of design problems with this system that will influence its temperatures and performance. The boiler for the system will have vertical tubes between 20 and 25 ft (6.1 and 7.6 m) long while the depth of the fluidized bed will be between 12 and 15 ft (3.7 and 4.6 m). Thus, there will be about 8 or 10 ft (2.4 or 3.0 m) of tubing above the bed which may act to reduce the temperature of the combustion gases before

Fig. 4. Energy diagram for the high pressure combustion air system.

Fig. 5. Energy diagram for the intermediate pressure combustion air system.

Fig. 6. Energy diagram for the atmospheric pressure combustion air system.

they leave the furnace. However, there may be some combustion taking place above the bed and there will be considerable material from the bed thrown into this space which will tend to keep the gases near the 1750°F (1228°K) bed temperature. The 1750°F (1228°K) turbine inlet temperature case ignores any temperature reduction in this space.

Another problem that is ignored with the 1750°F (1228°K) case is the manner in which the hot gases are piped from the furnace to the turbine. This piping must be protected from the gases by either internal insulation, cooling, or a combination of the two. If cooling is provided by the combustion air from the recuperator, this will increase the air temperature to the furnace which, in turn, reduces the amount of heat removed from the furnace by the combustion products. This results in an equivalent increase in the amount of heat transferred to the alkali metal system. It should be noted that, in regard to the system energy balance, it makes no difference whether the combustion products are cooled by the combustion air from the recuperator or by the boiler tubes in the free-board above the fluidized bed. Thus, in order to study the effects of cooling the combustion gases from the bed, a lower turbine inlet temperature case [1500°F (1089°K)] was also analyzed.

There are also a number of other problems with this system that are not being addressed at this time. The most difficult problem is reducing the particulate matter in the combustion gases to a sufficiently low level to prevent damage to the turbine. This problem will have an important impact on the system efficiency because of the pressure losses in the cleanup system and is one reason for the range of values of system $\Delta P/P$ used in the analysis.

The results of the analysis of the system are summarized in Tables 1 and 2 for turbine inlet temperatures of 1750°F (1228°K) and 1500°F (1089°K), respectively. As can be seen from these tables, the system $\Delta P/P$ influences the system power output more than the compressor pressure ratio. The split in the heat from the combustion system to the alkali metal system and the steam feedwater system is shown in the last two columns and will have an influence on the overall plant efficiency. For best efficiency, the heat should flow to the higher temperature cycle (alkali metal) before flowing

Table 1. High Pressure Combustion Air System - Data for 1750°F Turbine Inlet Temperature

Compressor Pressure Ratio	System $\frac{\Delta P}{P}$	Generator Output Btu lb air	Temperatures				Enthalpies				Recuperator Heat Load Btu lb air	Heat Transferred to	
			Air		Combustion Gases		Air		Combustion Gases			Boiler	Feedwater
			Compressor Outlet °F	Recuperator Outlet °F	Turbine Outlet °F	Recuperator Outlet °F	Compressor Outlet Btu/lb	Recuperator Outlet Btu/lb	Turbine Outlet Btu/lb	Recuperator Outlet Btu/lb			
10	0.1	106.9	612	852	512	691	259.0	320.1	335.5	279.0	61.1	769.9	104.7
	0.2	97.4	612	880	547	701	259.0	327.3	344.8	281.5	68.3	777.1	107.4
	0.3	86.2	612	913	586	712	259.0	335.9	355.6	284.3	76.9	785.9	110.4
9	0.1	106.1	580	871	544	677	250.9	325.0	343.8	275.2	74.1	775.0	100.5
	0.2	96.1	580	900	580	687	250.9	332.5	353.4	277.8	81.6	782.5	103.4
	0.3	84.5	580	934	1023	699	250.9	341.3	364.6	280.9	90.4	791.3	106.7
3	0.1	104.7	544	893	580	661	242.0	330.7	353.4	271.3	88.7	780.6	96.3
	0.2	94.4	544	923	1018	671	242.0	338.5	363.3	273.9	96.5	788.6	99.1
	0.3	82.4	544	958	1062	684	242.0	347.6	374.9	277.1	105.6	797.6	102.6
7	0.1	102.5	505	919	1023	645	232.2	337.4	364.6	267.2	105.2	787.3	91.9
	0.2	91.8	505	951	1062	655	232.2	345.8	374.9	269.7	113.6	795.8	94.6
	0.3	79.7	505	986	1106	667	232.2	355.0	386.6	272.9	122.8	804.9	98.1
6	0.1	98.9	461	951	1074	627	221.3	345.8	378.1	262.8	124.5	795.8	87.2
	0.2	88.0	461	983	1113	638	221.3	354.2	388.7	265.6	132.9	804.1	90.2
	0.3	75.2	461	1020	1160	652	221.3	363.9	401.1	269.1	142.6	813.7	94.0

Table 2. High Pressure Combustion Air System - Data for 1500°F Turbine Inlet Temperature

Compressor Pressure Ratio	System $\frac{\Delta P}{P}$	Generator Output: $\frac{\text{Btu}}{\text{lb air}}$	Temperatures				Enthalpies				Recuperator Heat Load $\frac{\text{Btu}}{\text{lb air}}$	Heat Transferred to	
			Air		Combustion Gases		Air		Combustion Gases			Boiler	Feedwater
			Compressor Outlet °F	Recuperator Outlet °F	Turbine Outlet °F	Recuperator Outlet °F	Compressor Outlet Btu/lb	Recuperator Outlet Btu/lb	Turbine Outlet Btu/lb	Recuperator Outlet Btu/lb			
10	0.1	79.1	612	718	745	647	259.C	285.8	292.6	267.8	26.8	811.0	92.6
	0.2	70.6	612	744	777	655	259.C	292.3	300.7	269.9	33.3	817.7	94.8
	0.3	60.9	612	773	814	665	259.C	299.8	310.1	272.4	40.8	825.1	97.5
9	0.1	79.3	580	735	773	631	250.9	290.0	300.0	263.7	39.1	815.3	88.1
	0.2	70.6	580	761	806	640	250.9	296.7	308.2	265.9	45.7	822.0	90.5
	0.3	60.4	580	792	844	650	250.9	304.5	318.1	268.5	53.6	829.8	93.3
8	0.1	79.2	544	754	806	613	242.0	294.8	308.2	259.3	52.8	820.1	83.4
	0.2	70.2	544	781	840	623	242.0	301.8	317.0	261.6	59.8	827.0	85.9
	0.3	59.6	544	812	879	633	242.0	309.8	327.2	264.3	67.9	835.1	88.8
7	0.1	78.4	505	777	844	595	232.2	300.6	318.1	254.7	68.5	825.9	78.4
	0.2	69.0	505	804	879	605	232.2	307.8	327.2	257.1	75.6	833.1	81.0
	0.3	58.4	505	836	919	616	232.2	315.9	337.4	259.9	83.8	841.2	84.0
6	0.1	76.5	461	804	890	576	221.3	307.8	330.0	249.9	86.5	833.1	73.2
	0.2	66.9	461	833	926	586	221.3	315.1	339.2	252.4	93.8	840.4	75.9
	0.3	55.6	461	866	968	598	221.3	323.7	350.2	255.3	102.4	849.0	79.1

to the steam system. Furthermore, as will be discussed later, the value of the heat used for feedwater heating is relatively low. Thus, the higher compressor pressure ratios will add little if any to the overall plant efficiency. The lower compressor pressure ratios do result in much greater heat loads on the recuperator.

Intermediate Pressure Combustion Air System. With the intermediate pressure system, the turbocharger is used only to pressurize the furnace and not to supply shaft power for electrical generation (see Fig. 2). This has several advantages over that of using a power turbine. The temperature required to drive the turbocharger will be less than 1000°F (811°K). Currently available inertial type separators should be sufficiently efficient in removing particulates from the combustion gas to prevent serious damage to the turbine for turbine inlet temperatures up to about 1100°F (867°K).

Another advantage of the turbocharged system is that the turndown ratio (ratio of design power output to actual output) of the furnace can be increased. This is due to the fact that the fluidization characteristics of the furnace bed are very weak functions of the bed pressure, i.e., they depend primarily on the velocity of the gases through the bed. With the turbocharger, the system power is reduced by decreasing the turbine inlet temperature which in turn decreases the turbocharger speed and the pressure of the air delivered to the furnace. Thus, most of the turndown can be achieved by a reduction in furnace pressure with little change in the furnace bed velocity or fluidization characteristics. It should be noted that, as the speed of the turbocharger is reduced, it will eventually reach an operating point at which the turbine does not deliver sufficient power to drive the compressor. At this point, auxiliary shaft power from a variable speed motor or a steam turbine will be needed. This auxiliary power source will also be needed to start the system. However, in the following analysis, only the full-power-output case is analyzed.

Another interesting aspect of this system is that the efficiency of the turbocharger has little effect on the efficiency of the plant, and, in regard to cycle efficiency, can be considered a heat exchanger that replaces part of the surface area of the air preheater and recuperator. The efficiency of the turbocharger does influence the turbine inlet temperature required to drive the unit and also the compressor outlet temperature.

Rather modest component efficiencies are assumed for the turbocharger — 80% for the compressor and 81% for the turbine.

The performance of the system was calculated for compressor pressure ratios of 3, 4, and 5 and for system pressure losses ($\Delta P/P$) of 0.1, 0.2, and 0.3. The turbine inlet temperature was that required to drive the compressor without any turbine bypass. The turbocharger mechanical losses were accounted for by increasing the turbine output by 2% over that required for compression. The compressor air inlet temperature was 60°F (289°K), the weight ratio of combustion products to air was 1.08, the recuperator heating effectiveness was 80% and the air preheater was sized to give the correct turbine inlet temperature. The only effect of the furnace exit temperature is on the size of the air preheater and thus the air temperature from the preheater. Two furnace exit gas temperatures, 1750°F (1228°K) and 1500°F (1089°K), were analyzed.

The results of the calculations for this system are given in Tables 3 and 4. Table 3 gives data for that part of the system at temperatures below those of the air preheater. All of these data are independent of the furnace outlet temperature. The turbocharger shaft power is the turbine work required to drive the compressor and includes a mechanical loss of 2%. The required turbine inlet temperature for all cases is less than 1000°F (811°K). The inlet temperature shown for the $\Delta P/P = 0.3$ case shows a minimum at a compressor pressure ratio of about 4 and is typical for such equipment. The inlet temperature required for the 0.1 and 0.2 system pressure loss cases will also start increasing at lower pressure ratios. This operating characteristic of the turbocharger will result in a part load control system that requires auxiliary shaft power to operate the system.

All of the data shown in Table 3 favor a low system pressure loss. The recuperator heat loads are lower, the amount of heat transferred to the alkali metal system is higher and the heat transferred to the feedwater heaters is lower. However, a lower system pressure loss will require a larger air preheater.

Table 4 shows the preheater operating conditions, heat loads and required heating effectiveness for furnace outlet temperatures of 1700°F (1228°K) and 1500°F (1089°K). The design requirements are more difficult

Table 3. Intermediate Pressure Combustion Air System Data

Compressor Pressure Ratio	System $\frac{\Delta P}{P}$	Turbo- charger Shaft Power Btu lb air	Temperatures					Enthalpies					Recuperator Heat Load Btu lb air	Heat Transferred to	
			Air		Combustion Gases			Air		Combustion Gases				Boiler	Feedwater
			Compressor	Recuperator	Turbine	Turbine	Recuperator	Compressor	Recuperator	Turbine	Turbine	Recuperator			
			Outlet °F	Outlet °F	Inlet °F	Outlet °F	Outlet °F	Outlet Btu/lb	Outlet Btu/lb	Inlet Btu/lb	Outlet Btu/lb	Outlet Btu/lb		Btu lb air	Btu lb air
5	0.1	93.0	437	450	796	453	441	215.4	218.7	305.6	219.5	216.5	2.29	946.9	37.2
	0.2	93.0	437	520	880	541	454	215.4	236.1	327.3	241.2	222.1	20.63	940.9	43.2
	0.3	93.0	437	615	994	660	496	215.4	259.8	357.1	271.0	229.9	44.35	932.5	51.6
4	0.1	77.4	374	434	735	449	394	200.2	214.7	290.0	218.3	204.9	14.5	959.9	24.6
	0.2	77.4	374	516	834	551	421	200.2	234.9	315.4	243.7	211.5	34.7	952.7	31.8
	0.3	77.4	374	632	973	697	450	200.2	264.1	351.5	280.4	221.2	63.9	942.3	42.2
3	0.1	58.8	299	420	668	450	308	181.9	211.2	273.1	218.6	191.4	29.4	974.8	10.0
	0.2	58.8	299	526	798	583	374	181.9	237.6	306.1	251.7	200.2	55.6	965.3	19.5
	0.3	58.8	299	688	995	785	430	181.9	278.1	357.3	302.9	213.8	96.2	950.6	34.2

Table 4. Air Preheater Operating Conditions, Heat Loads and Required Heating Effectiveness for the Intermediate Pressure Combustion Air System

Furnace Outlet Temp. °F	Furnace Outlet Enthalpy Btu/lb	Compressor Pressure Ratio	System $\frac{\Delta P}{P}$	Preheater Air Outlet Temp. °F	Preheater Air Outlet Enthalpy Btu/lb	Preheater Heat Load $\frac{\text{Btu}}{\text{lb air}}$	Heating Effective- ness
1750	563.4	5	0.1	1513	497.1	278.4	.818
1750	563.4	5	0.2	1491	491.1	255.0	.789
1750	563.4	5	0.3	1460	482.6	222.8	.744
1750	563.4	4	0.1	1559	510.0	295.3	.855
1750	563.4	4	0.2	1533	502.7	267.8	.824
1750	563.4	4	0.3	1498	493.0	228.9	.774
1750	563.4	3	0.1	1612	524.7	313.5	.896
1750	563.4	3	0.2	1579	515.5	277.9	.860
1750	563.4	3	0.3	1529	500.7	222.6	.792
1500	493.6	5	0.1	1237	421.7	203.0	.750
1500	493.6	5	0.2	1214	415.7	179.6	.708
1500	493.6	5	0.3	1183	407.2	147.4	.642
1500	493.6	4	0.1	1284	434.6	219.9	.797
1500	493.6	4	0.2	1258	427.4	192.5	.754
1500	493.6	4	0.3	1221	417.6	153.5	.678
1500	493.6	3	0.1	1338	449.3	238.1	.850
1500	493.6	3	0.2	1305	440.1	202.5	.800
1500	493.6	3	0.3	1251	425.3	147.2	.693

for the lower system pressure loss and the higher furnace outlet temperature where the operating temperatures are higher, and both the heat load and required heating effectiveness are larger, especially at the lower compressor pressure ratios. The lower furnace outlet temperature for this case is highly advantageous. This implies that the combustion gases on leaving the fluidized bed should be cooled (by the alkali metal boiler tubes in the freeboard above the bed) before leaving the furnace.

Atmospheric Pressure Combustion Air System. In the atmospheric pressure combustion air system, the air is compressed only to the pressure required to overcome the pressure losses throughout the system. The drive for the compressor was assumed to be an electric motor with a combined mechanical and electrical efficiency of 92%. The compressor efficiency was assumed to be 86%. The compressor pressure ratios studied were 1.2, 1.4, and 1.6. The compressor air inlet temperature for all cases was 60°F (289°K). Furnace outlet temperatures of 1750°F (1228°K) and 1500°F (1089°K) were analyzed to determine the heat load on the air preheater, the heat flow split between the alkali metal boiler and the feedwater heaters, and the temperature of the combustion gases going to the feedwater heaters. The heating effectiveness of the preheater was assumed to be 80% for all cases.

The results of the calculations for this system are shown in Table 5. As with the intermediate pressure case, the design of the preheater will be much easier with the lower furnace outlet temperature, as both heat load and temperatures are lower. Also, the overall plant efficiency will be higher for the lower furnace outlet temperature since the amount of heat bypassing the alkali metal system and going to the feedwater heaters is less.

Alkali Metal System

The analysis of the alkali metal system was done for both cesium and potassium. Three turbine inlet temperatures — 1540, 1440, and 1390°F (1111, 1856, and 1028°K), and three condenser temperatures — 890, 990, and 1090°F (750, 806, and 861°K) were analyzed for both working fluids.

In order to calculate the performance of this system, it was first necessary to determine the efficiency of the turbine for the various cases considered. In doing so, a decision was made to use approximately an equal

Table 5: Atmospheric Pressure Combustion Air System Data

Furnace Outlet Temp: °F	Furnace Outlet Enthalpy Btu/lb	Compressor Pressure Ratio	Motor Power Required Btu lb air	Temperatures			Enthalpies			Preheater Heat Load Btu lb air	Heat Transferred to	
				Compressor Air Outlet °F	Preheater Air Outlet °F	Preheater Gases Outlet °F	Compressor Air Outlet Btu/lb	Preheater Air Outlet Btu/lb	Preheater Gases Outlet Btu/lb		Boiler Btu lb air	Feedwater Btu lb air
1750	563.4	1.2	8.43	92	1419	574	132.0	471.2	249.4	339.2	921.1	72.7
1750	563.4	1.4	15.91	121	1424	593	138.9	472.8	254.3	333.9	922.6	78.0
1750	563.4	1.6	22.65	147	1429	611	145.1	474.2	258.7	329.1	924.1	82.7
1500	493.6	1.2	8.43	92	1219	496	132.0	416.8	230.0	284.8	942.1	51.7
1500	493.6	1.4	15.91	121	1224	516	138.9	418.3	234.9	279.4	943.6	57.0
1500	493.6	1.6	22.65	147	1229	533	145.1	419.7	239.4	274.6	944.9	61.9

number of stages for the cesium and potassium units, which resulted in units with nearly identical first stages. It is possible to take advantage of the smaller enthalpy change across the turbine for cesium to reduce the number of stages as compared to potassium. Although this would result in a somewhat less expensive cesium unit, it would also reduce the efficiency of the unit by several points. Rather than take this approach, the speed of the cesium unit was made one-half that of the potassium unit, which gives machines of nearly the same size and number of stages. This lower speed has a very significant effect on the materials requirements for the blades and discs.

In analyzing the turbines for the various cycle conditions, it was assumed that the only difference in the turbine efficiency for the various cases was that associated with supersaturation, moisture, and exhaust losses. The moisture losses were assumed to be 0.6% for each percent of moisture and the exhaust loss was assumed to be the kinetic energy in the fluid from the last stage. The supersaturation losses were calculated as a function of the enthalpy drop per stage and were found to range between 2% for a stage enthalpy change of 11 Btu/lb (25.6×10^3 J/kg) to 4% for an enthalpy drop of 15 Btu/lb (34.9×10^3 J/kg) for cesium. Comparable results for potassium are a loss of 2% for a stage enthalpy drop of 35 Btu/lb (81.4×10^3 J/kg) and 4% for an enthalpy drop of 60 Btu/lb (140×10^3 J/kg). These results are based on the assumption that the return to equilibrium occurs along a constant pressure line, probably somewhat conservative. The turbine efficiency was then calculated by subtracting the sum of the supersaturation losses, moisture losses, and exhaust losses from a basic machine efficiency of 88%.

In carrying out both the turbine and the cycle analyses, it was assumed that vapor for regenerative feed heating would be bled from the turbine between all stages except the first and second. Furthermore, it was assumed that moisture removal devices were located at each bleed point. The efficiency of the moisture removal was taken as 25%, i.e., 25% of the moisture in the working fluid at each bleed point is removed with the bleed flow. This value is well within the 20 to 35% efficiency demonstrated on a small potassium turbine tested by General Electric for the space power program.⁵

The alkali metal feed heating arrangement is shown in Fig. 7. The number of feed heaters used varied from 1 to 3 depending on the difference between the turbine inlet and the condenser temperature. The bleed flow to the respective feed heaters was considered to be comprised of the moisture removed plus a quantity of the remaining working fluid. The bleed enthalpy listed in Tables 6 and 7 is that of the portion of the working fluid which was bled. The enthalpy of the moisture removed would be that of saturated liquid at the bleed temperature. The pressure loss between the bleed points and feed heaters was assumed to be a $\Delta P/P$ of 0.1. The effectiveness of the feed heat exchangers was taken into account by imposing the following limits on the temperature changes across any one heater

$$T_4 = T_2 - 10^\circ\text{R}$$

$$T_3 = T_1 - 5^\circ\text{R}$$

where the temperatures are defined as shown in Fig. 7.

Tables 6 and 7 give the cycle state points for the cases considered for cesium and potassium. The vapor at the turbine inlet was saturated for all cases. The bleed flow shown is the fraction of the turbine inlet flow extracted for feed heating. The moisture flow is also the same fraction of the turbine inlet flow and represents the moisture separated from the working fluid on passing through the turbine. The enthalpy identified as h_1 is the enthalpy of the combined bleed and moisture on entering the feed heater. The cesium pressure and enthalpy for turbine inlet temperatures of 1540, 1440, and 1390°F are 57.65, 38.67, and 31.14 psia and 309.5, 308.3, and 307.6 Btu/lb. Corresponding values for potassium inlet temperatures of 1540, 1440, and 1390°F are 29.58, 18.58 and 14.44 psia and 1194.7, 1190.5 and 1188.4 Btu/lb. The cesium pressure and enthalpy for condenser temperatures of 890, 990, and 1090°F are 1.44, 3.16, and 6.28 psia and 73.9, 79.2, and 84.5 Btu/lb. Corresponding values for potassium condenser temperatures of 890, 990, and 1090°F are 0.40, 1.01 and 2.24 psia and 263.6, 281.8, and 300.5 Btu/lb.

The results of the alkali metal system analysis are summarized in Table 8. The number of turbine stages varies from 3 to 5 and the number

Fig. 7. Alkali metal system feed heating arrangement.

Table 6. Cesium Cycle State Points

Turbine Inlet Temp. °F	Condenser Temp. °F	Feed Heater	Bleed Pressure psia	Bleed Temp. °F	Bleed Enthalpy	T ₁ °F	T ₂ °F	T ₃ °F	T ₄ °F	h ₁ Btu/lb	h ₂ Btu/lb	h ₃ Btu/lb	h ₄ Btu/lb	Bleed Flow	Moisture Flow
1540	890	3	3.92	1020	281.0	1005	900	1000	890	159.8	75.7	79.7	75.3	.0180	.0276
		2	8.30	1135	285.5	1120	1010	1115	1000	190.7	80.2	85.9	79.7	.0291	.0265
		1	16.69	1260	290.8	1240	1125	1235	1115	215.3	86.4	93.8	85.9	.0379	.0238
1540	990	2	8.06	1130	284.8	1115	1000	1110	990	188.1	79.7	85.6	79.3	.0285	.0272
		1	16.27	1255	290.3	1235	1120	1230	1110	213.2	86.1	93.5	85.6	.0377	.0246
1540	1090	2	11.12	1185	288.0	1165	1100	1160	1090	182.9	85.1	89.4	84.6	.0218	.0249
		1	19.44	1290	292.0	1270	1170	1265	1160	198.6	90.0	95.6	89.4	.0301	.0227
1440	890	3	3.64	1010	282.8	995	900	990	890	175.4	74.3	79.2	73.9	.0225	.0254
		2	6.90	1105	286.8	1090	1000	1085	990	182.8	79.7	84.3	79.2	.0227	.0242
		1	12.78	1210	291.5	1190	1095	1185	1085	210.8	84.8	90.9	84.3	.0311	.0212
1440	990	2	6.69	1100	286.7	1085	1000	1080	990	180.4	79.7	84.0	79.3	.0215	.0240
		1	12.10	1200	291.5	1185	1090	1180	1080	210.9	84.5	90.6	84.0	.0310	.0210
1440	1090	1	11.12	1185	290.0	1165	1100	1160	1090	190.4	85.1	89.4	84.6	.0230	.0230
1390	890	2	3.92	1020	283.0	1005	900	1000	890	187.5	74.3	79.7	73.9	.0300	.0268
		1	8.30	1135	288.5	1120	1010	1115	1000	197.5	80.2	85.9	79.7	.0288	.0237
1390	990	2	6.08	1085	287.2	1070	1000	1065	990	174.1	79.7	83.2	79.3	.0177	.0223
		1	10.81	1180	291.7	1165	1075	1160	1065	209.5	83.7	89.4	83.2	.0290	.0200
1390	1090	1	11.45	1190	292.1	1170	1100	1165	1090	198.3	85.3	89.7	84.6	.0226	.0198

Table 7. Potassium Cycle State Points

Turbine Inlet Temp. °F	Condenser Temp. °F	Feed Heater No.	Bleed Pressure psia	Bleed Temp. °F	Bleed Enthalpy	T ₁ °F	T ₂ °F	T ₃ °F	T ₄ °F	h ₁ Btu/lb	h ₂ Btu/lb	h ₃ Btu/lb	h ₄ Btu/lb	Bleed Flow	Moisture Flow
1540	890	3	1.30	1020	1082.1	1035	900	1000	890	620.5	265.4	283.7	263.8	.0210	.0291
		2	3.10	1135	1099.1	1120	1010	1115	1000	676.7	285.5	305.2	283.7	.0242	.0278
		1	6.99	1260	1119.9	1245	1125	1240	1115	762.5	307.1	329.7	305.2	.0294	.0245
1540	990	3	2.16	1085	1095.8	1070	1000	1065	990	587.3	283.7	295.8	282.0	.0148	.0262
		2	4.50	1190	1111.5	1175	1075	1170	1065	701.7	297.5	316.1	295.8	.0231	.0248
		1	8.83	1300	1129.6	1250	1180	1275	1170	761.1	318.0	336.5	316.1	.0246	.0216
1540	1090	2	4.50	1190	1111.5	1175	1100	1170	1090	649.6	302.3	316.1	300.6	.0177	.0248
		1	8.83	1300	1129.6	1250	1180	1275	1170	761.1	318.0	336.5	316.1	.0246	.0216
1440	890	3	1.01	990	1085.6	950	900	975	890	585.2	265.4	279.1	263.7	.0164	.0271
		2	2.24	1090	1102.4	1075	985	1070	975	650.2	280.9	296.7	279.1	.0198	.0256
		1	4.96	1205	1121.6	1130	1080	1185	1070	753.6	298.5	319.0	296.7	.0264	.0225
1440	990	2	2.24	1090	1102.4	1075	1000	1070	990	629.4	283.7	297.6	281.9	.0178	.0256
		1	4.96	1205	1121.6	1130	1080	1185	1070	745.6	298.5	319.0	297.6	.0253	.0225
1440	1090	2	3.68	1160	1119.8	1145	1100	1140	1090	583.6	302.3	309.9	300.6	.0106	.0211
		1	6.57	1250	1135.3	1255	1150	1230	1140	764.0	312.2	327.2	309.9	.0213	.0183
1390	890	3	0.93	980	1090.0	970	900	965	890	571.5	265.4	277.2	263.7	.0145	.0258
		2	2.00	1075	1105.0	1050	975	1055	965	648.8	279.1	293.9	277.2	.0187	.0243
		1	4.08	1175	1123.2	1155	1065	1160	1055	744.3	295.8	314.2	293.9	.0239	.0212
1390	990	2	2.00	1075	1105.0	1050	1000	1055	990	586.6	283.7	293.9	281.9	.0136	.0244
		1	4.08	1175	1123.2	1150	1065	1155	1055	735.7	295.8	313.2	293.9	.0228	.0211
1390	1090	1	4.08	1175	1123.2	1150	1100	1155	1090	652.0	302.3	313.2	300.5	.0150	.0211

Table 8. Summary of Alkali Metal Cycle Analysis

Turbine Inlet Temp. °F	Condenser Temp. °F	No. of Turbine Stages	No. of Feed Heaters	Turbine Efficiency %	Cycle Efficiency %	Net System Efficiency %
<u>Cesium</u>						
1540	890	5	3	78.8	25.4	24.6
1540	990	4	2	78.6	21.3	20.7
1540	1090	4	2	80.7	17.2	16.7
1440	890	5	3	79.2	22.9	22.2
1440	990	4	2	79.7	18.4	17.8
1440	1090	3	1	78.8	14.2	13.8
1390	890	4	2	77.9	21.6	21.0
1390	990	4	2	80.4	17.8	17.3
1390	1090	3	1	81.9	13.0	12.6
<u>Potassium</u>						
1540	890	5	3	77.3	25.3	24.5
1540	990	5	3	79.0	21.8	21.1
1540	1090	4	2	78.9	17.7	17.2
1440	890	5	3	78.7	22.8	22.1
1440	990	4	2	78.6	18.5	17.9
1440	1090	4	2	80.6	14.7	14.3
1390	890	5	3	79.0	21.6	21.0
1390	990	4	2	79.5	17.3	16.8
1390	1090	3	1	79.9	12.9	12.5

of feed heaters used in the system was always 2 less than the number of turbine stages. The feed pump energy requirements were based on a pump efficiency of 70% and a head equal to twice the difference between the boiler and condenser pressures. The net system efficiency was based on assumed mechanical and electrical losses of 3%, i.e., 97% of the turbine shaft power. The efficiency of the system varies between 12 and 25%. However, expressed as a percent of the ideal Carnot cycle the variation was from 74 to 78%. The turbine efficiency varied between 77 and 82%.

In the overall plant analysis, the power output from the alkali metal system is the heat input from the combustion air system furnace-boiler multiplied by the net system efficiency. The amount of heat rejected from the alkali metal system to the steam system is the input from the combustion air system multiplied by one minus the cycle efficiency (the use of cycle efficiency rather than the net system efficiency assumes that the mechanical and electrical losses are lost to the plant cycle). As noted previously, part of the energy input to the combustion air system bypasses the alkali metal system and goes to the feedwater heaters in the steam system.

Steam System

For the parametric cycle analysis of the overall plant efficiency the efficiency of the steam system was estimated from that of actual operating plants, theoretical considerations and previous studies done at ORNL. The steam system for a fluidized bed alkali metal will differ from a conventional system in that the heat supply to at least one stage of feedwater heating will be from the combustion products rather than from steam bled from the turbine. The steam cycle efficiency as used in this analysis is defined as the generator output divided by the heat added to the system through both the steam generator from the condensing potassium and the feed water heater from the combustion products.

For a total plant output of 200 MW(e) the output of the steam plant will be between 100 and 150 MW(e) for the range of plant conditions being studied, i.e., alkali metal turbine inlet conditions from 1850°R (1028°K) to 2000°R (1111°K) and condenser temperatures from 1350°R (750°K) to 1550°R (861°K). For the parametric analysis of the plant efficiency, the steam

temperature to the turbine is assumed to be 100°F (56°K) less than the alkali metal condensing temperature. One reheat is used with the reheat temperature also 100°F (56°K) less than the alkali metal condensing temperature. The condenser pressure is assumed to be 1 1/2 in. of Hg (5 kPa).

A review of small fossil-fired central stations completed during the last two years or currently under construction^{6,7} shows that the steam conditions for plant sizes of 100 to 300 MW(e) are a pressure of 2000 to 2600 psi (13.8 to 18.0 MPa) and a temperature of 1000°F (811°K) with a reheat to 1000°F (811°K). The efficiency of these plant varied rather widely from about 32 to 42%. However, the more efficient units, about one-half of the total, have design efficiencies of about 40%. In general there is little difference in the reported plant heat rates of the smaller units to those several hundred to a thousand megawatts in size. The smaller plants had 5 or 6 stages of feedwater heating while the large units generally had 7 or 8 stages. This difference in the number of feedwater heaters would result in about a one point difference in plant efficiency.

The basic cycle efficiency of large modern steam plants operating near or above critical pressure and steam temperatures of 1000°F (811°K) and one reheat to 1000°F (811°K) is about 47%. This value does not include boiler losses, auxiliary power requirements such as coal grinders, nor losses due to SO₂ scrubbers. The auxiliary power losses amount to about 2 to 3% of the gross power output and furnace efficiencies run about 88 to 90%. Thus, the net plant efficiency is about 40 to 41%. In general, the efficiency of smaller plants would run about 2 or 3 points less because of lower steam pressures and fewer feedwater heaters and one would expect a basic efficiency of about 44 or 45% for a steam temperature of 1000°F (811°K) and a net plant efficiency of about 38 to 40%.

A number of studies of the effect of steam pressure and temperature on cycle efficiency have been made at ORNL. A summary of one of these studies is shown in Fig. 8 for regenerative, non-reheat cycles. Although the steam pressures shown are lower than that of the current study, it can be seen from Fig. 8 that the pressure has little, if any, effect on the rate of change of efficiency with temperature. The rate of change is about 1.3% for each 100°F (56°K) change in temperature. In the following analysis an

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Fig. 8. Effect of pressure and temperature on the efficiency of regenerative, non-reheat steam cycles.

efficiency of 44% at 1000°F is used with reductions of 1.3% for each 100°F (56°K) reduction in temperature.

The effect of using the combustion gases for part of the feedwater heating will reduce the cycle efficiency. This reduction will depend on the temperature of the gases and the amount of the heat transferred to the feedwater. In order to estimate this effect, first assume a system with a pressure of 2000 psia (13.8 MPa), a temperature of 900°F (756°K), a reheat to 900°F (756°K), a condenser pressure of 1 1/2 in. of Hg (5 kPa), a turbine efficiency of 85% and 5 stages of feedwater heating which raise the feedwater temperature to its optimum temperature of 550°F (56°K). The effect of the feedwater heating on the steam system heat rate with all heating done by extracted steam is a reduction of 14.2% as compared to a non-regenerative system (this value and parts of the following analysis use data and curves from Chapter 12 of Ref. 8). If all feedwater heating were done by combustion gases, the heat rate would be the same as for a non-regenerative system and 14.2% higher than with steam extraction. If the heat from the combustion gases replaced the highest temperature feedwater heater, the remaining four feedwater heaters would still reduce the heat rate by 13.1% as compared to a non-regenerative system. Replacing the two highest temperature heaters would reduce the savings to 11.6%; replacing the three highest temperature units would reduce the savings to 8.8%; and replacing the four highest temperature units would drop this value to 5%.

In the analysis to follow, the value of the feed heaters will be assumed to be as follows:

<u>Feedwater Heater No.</u>	<u>Percent Reduction in the Plant Heat Rate</u>
1	1.1
2	1.5
3	2.8
4	3.8
5	5.0

In a non-regenerative system the amount of heat required would be as follows:

Boiler	860 Btu/lb
Reheater	211
Feed heating	<u>490</u>
	1627 Btu/lb

where the feed heating is that required to raise the water temperature from the condenser temperature of 92°F (307°K) to the temperature normally leaving the feedwater heaters of 550°F (561°K). Now, we will neglect the effects of the bleed flows on the flow rates of a regenerative cycle and use these values of heat input to the different parts of the cycle to determine the number of feedwater heaters replaced by combustion gases in a regenerative system. The total heat used for feed heating is 490 Btu/lb (1.1 MJ/kg) or 98 Btu/lb (0.23 MJ/kg) per heater (for 5 feed heaters). Thus, each of the feedwater heaters requires about 6% of the total heat input.

The temperature leaving each feedwater heater with a condenser temperature of 92°F (307°K) is

<u>Feed Heater No.</u>	<u>Exit Temperature, °F</u>
1	550
2	458
3	367
4	275
5	184

while the combustion gases going to the stack are 300°F (422°K). To allow a fairly large approach temperature, it will be assumed that the combustion gases will replace feed heater No. 4 and those at higher temperatures as follows:

<u>Percent Heat to Feedwater Heaters</u>	<u>Feedwater Heater Replaced</u>	<u>Percent Increase in Plant Heat Rate</u>
0-6	4	0-3.8
6-12	3	3.8-6.6
12-18	2	6.6-8.1
18-24	1	8.1-9.2

A graph showing the increase in the steam system heat rate as a function of the percent of heat to the feedwater heaters is shown in Fig. 9.

The above type of analysis will give slightly different values for different steam conditions. However, for the parametric analysis, the above values will be used for steam temperatures of 800 to 1000°F (700 to 811°K). The basic steam cycle efficiencies for temperatures of 800, 900, and 1000°F are 41.4, 42.7 and 44% which correspond to system heat rates of 8244 Btu/hr (2.42 kW), 7933 Btu/hr (2.32 kW) and 7757 Btu/hr (2.27 kW) respectively. These basic efficiencies can be combined with the penalties for using combustion gases for feedwater heating to arrive at steam plant efficiencies as a function of steam temperature and percent of heat going to the feedwater heaters. Figure 10 shows both the system heat rate and efficiency as a function of the steam temperature and percent heat for feedwater heating that are used for the steam system and the overall plant analysis. The power output from the steam system is, then, the product of this efficiency times the total heat added to the system from both the alkali metal system and the combustion air system.

Overall Plant

There are a total of 972 combinations of the combustion air system and alkali metal systems analyzed in the preceding sections. As 810 of these influence the system efficiency, no attempt will be made to tabulate all of these combinations. Several are selected to show the range of plant efficiencies that may be expected and to show the effect of the more important parameters on the plant efficiency. Table 9 shows the effect of varying the combustion air system with a constant alkali metal (cesium) system. The most efficient combination is with a high pressure combustion air system, followed by the intermediate pressure system and then the atmospheric pressure system. The peak efficiency, which is not shown on the table, would be at a furnace outlet temperature of 1750°F (1288°K), a compressor pressure ratio of 10, a system $\Delta P/P$ of 0.1, an alkali metal turbine inlet temperature of 1540°F (1111°K) with a condenser temperature of 890°F (750°K) and a steam temperature of 790°F (694°K). The efficiency of this system is 48.5% based on the higher heating value of the as-received coal. All values in both Tables 9 and 10 are based on the higher heating value of

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Fig. 9. Increase in system heat rate as a function of the percent of system heat to feedwater heaters.

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Fig. 10. Net steam system efficiency and heat rate as a function of the percent of system heat to feedwater heaters.

Table 9 Overall Plant Performance with Various Combustion Air Systems and Conditions

Combustion Air System						Cesium System						Steam System						Overall Plant Efficiency %
Furnace Outlet Temp. °F	Compressor Pressure Ratio	System $\frac{\Delta P}{P}$	Generator Output Btu/lb air	Heat Transferred to		Turbine Inlet Temp. °F	Condenser Temp. °F	Cycle Efficiency %	Net System Efficiency %	Generator Output Btu/lb air	Heat to Steam System Btu/lb air	Turbine Inlet Temp. °F	Heat to System Btu/lb air	Heat to Feedwater %	System Efficiency %	Generator Output Btu/lb air		
				Boiler	Feedwater													
				Btu/lb air	Btu/lb air													
1750	6	.2	88.0	804.1	90.2	1540	990	21.3	20.7	166.4	632.8	890	723.0	12.5	39.7	287.0	47.3	
1750	8	.2	94.4	788.6	99.1	↓	↓	↓	↓	163.2	620.6	↓	719.7	13.8	39.5	284.3	47.3	
1750	10	.2	97.4	777.1	107.4	↓	↓	↓	↓	160.9	611.6	↓	719.0	14.9	39.4	283.3	47.3	
1750	8	.1	104.7	780.6	96.3	↓	↓	↓	↓	161.6	614.3	↓	710.6	13.6	39.6	281.4	47.8	
1750	8	.3	82.4	797.6	102.6	↓	↓	↓	↓	165.1	627.7	↓	730.3	14.0	39.5	288.5	46.8	
1500	8	.1	79.2	820.1	83.4	↓	↓	↓	↓	169.8	645.4	↓	728.8	11.4	39.9	290.8	47.1	
1500	8	.2	70.2	827.0	85.9	↓	↓	↓	↓	171.2	650.8	↓	736.7	11.7	39.8	293.2	46.7	
1500	8	.3	59.6	835.1	88.8	↓	↓	↓	↓	172.9	657.2	↓	746.0	11.9	39.8	296.9	46.2	
1500	3	.2	0	965.3	19.5	1540	990	21.3	20.7	199.8	759.7	890	779.2	2.5	42.4	330.4	46.3	
or	4	.2	↓	952.7	31.8	↓	↓	↓	↓	197.2	749.8	↓	781.6	4.1	42.3	330.6	46.1	
1750	5	.2	↓	940.9	43.2	↓	↓	↓	↓	194.8	740.5	↓	783.7	5.5	42.2	330.7	45.9	
	4	.1	↓	959.9	24.6	↓	↓	↓	↓	198.7	755.4	↓	780.0	3.2	42.3	329.9	46.2	
	4	.3	↓	942.3	42.2	↓	↓	↓	↓	195.1	741.6	↓	783.8	5.4	42.2	330.8	45.9	
1750	1.2		- 8.4	921.1	72.7	1540	990	21.3	20.7	190.7	724.9	890	797.6	9.1	40.2	320.6	43.9	
1750	1.4		-15.9	922.6	78.0	↓	↓	↓	↓	191.0	726.1	↓	804.1	9.7	40.1	322.4	43.4	
1750	1.6		-22.7	924.1	82.7	↓	↓	↓	↓	191.3	727.3	↓	810.0	10.2	40.0	324.0	43.0	
1500	1.2		- 8.4	942.1	51.7	↓	↓	↓	↓	195.0	741.4	↓	793.1	6.5	40.7	322.8	44.5	
1500	1.4		-15.9	943.6	57.0	↓	↓	↓	↓	195.3	742.6	↓	799.6	7.1	40.6	324.6	44.0	
1500	1.6		-22.7	944.9	61.9	↓	↓	↓	↓	195.6	743.6	↓	805.5	7.7	40.5	326.2	43.6	

Table 10 Overall Plant Performance with Various Cesium Cycle Conditions

Combustion Air System						Cesium System							Steam System				Overall Plant Efficiency %
Furnace Outlet Temp. °F	Compressor Pressure Ratio	System $\frac{\Delta P}{P}$	Generator Output Btu/lb air	Heat Transferred to		Turbine Inlet Temp. °F	Condenser Temp. °F	Cycle Efficiency %	Net System Efficiency %	Generator Output Btu/lb air	Heat to Steam System Btu/lb air	Turbine Inlet Temp. °F	Heat to System Btu/lb air	Heat to Feedwater %	System Efficiency %	Generator Output Btu/lb air	
				Boiler Btu/lb air	Feedwater Btu/lb air												
1750 ↓	8 ↓	.2 ↓	94.4 ↓	788.6 ↓	99.1 ↓	1540 1540 1540	890 990 1090	25.4 21.3 17.2	24.6 20.7 16.7	194.0 163.2 131.7	588.3 620.6 653.0	790 890 990	687.4 719.7 752.1	14.4 13.8 13.2	38.3 39.5 40.8	263.3 284.3 306.9	48.2 47.3 46.6
1750 ↓	8 ↓	.2 ↓	94.4 ↓	788.6 ↓	99.1 ↓	1440 1440 1440	890 990 1090	22.9 18.4 14.2	22.2 17.8 13.8	175.1 140.4 108.8	608.0 643.5 676.6	790 890 990	707.1 742.6 775.7	14.0 13.3 12.8	38.3 39.6 40.9	270.8 294.1 317.3	47.2 46.2 45.5
1750 ↓	8 ↓	.2 ↓	94.4 ↓	788.6 ↓	99.1 ↓	1390 1390 1390	890 990 1090	21.6 17.8 13.0	21.0 17.3 12.6	165.6 136.4 99.4	618.3 648.2 686.1	790 890 990	717.4 747.3 785.2	13.8 13.2 12.6	38.4 39.6 40.9	275.5 295.9 321.1	46.8 46.0 45.0
1500 to 1750	4 ↓	.2 ↓	0 ↓	952.7 ↓	31.8 ↓	1540 ↓	890 990 1090	25.4 21.3 17.2	24.6 20.7 16.7	234.4 197.2 159.1	710.7 749.8 788.8	790 890 990	742.5 781.6 820.6	4.3 4.1 4.0	41.0 42.3 43.6	304.4 330.6 358.8	47.1 46.1 45.2
1750 ↓	1.2 1.4 1.6		- 8.4 -15.9 -22.7	921.1 922.6 924.1	72.7 78.0 82.7	1390 ↓	890 990 1090	21.6 17.8 13.0	21.0 17.3 12.6	193.4 159.6 116.4	722.1 758.4 804.0	790 890 990	794.8 836.4 886.7	9.1 9.3 9.3	39.0 40.2 41.5	310.0 336.2 368.0	43.2 41.9 40.6

the as-received coal which is equivalent to a heat input to the plant of 1145 Btu/lb (2.7 MJ/kg) of air. The furnace, latent heat and stack gas losses are 14%; i.e., only 86% of the heat enter the systems of the plant. The negative values shown under generator output for the atmospheric pressure combustion air system represent the electrical power required to drive the compressors.

Table 9 shows the sensitivity of the plant efficiency to the combustion air system variables. The compressor pressure ratio has no effect on the plant with a $\Delta P/P$ of 0.2. However, the plant efficiency does depend on the system $\Delta P/P$ — decreasing about 1/2 point for each 0.1 change. Another interesting aspect of the high pressure system is that the difference in plant efficiency for furnace temperatures (also gas turbine inlet temperatures) of 1750°F (1228°K) and 1500°F (1089°K) is only 0.6 points. The high pressure system at 1750°F (1228°K) and a $\Delta P/P$ of 0.2 is 1.0 to 1.4 points better than the intermediate pressure system, and 3.4 to 4.3 points better than the atmospheric pressure system.

Table 10 shows the sensitivity of the plant efficiency to the alkali metal system operating conditions. There is very little difference in the efficiencies of the cesium and potassium systems and cesium was used here for comparison purposes. As would be expected, the higher turbine inlet temperature gives the best efficiency with each 100°F (56°K) being worth about 1 point in plant efficiency. Table 10 also shows that a lower cesium condenser-steam boiler temperature is also beneficial with each 100°F (56°K) reduction being worth from 0.7 to 1.3 points in efficiency. The last row of numbers in Table 10 shows the poorest plant efficiency (40.6) of all those considered. This is an atmospheric pressure air system with a furnace outlet temperature of 1750°F (1228°K) and a compressor pressure ratio of 1.6, a cesium turbine inlet temperature of 1390°F (1028°K) with a condenser temperature of 1090°F (861°K), and a steam temperature of 990°F (806°K). In general, the atmospheric pressure combustion air system is rather poor in comparison to the other two air systems.

Furnace-Boiler Analysis

There are many factors that influence the selection of the design parameters for the furnace-boiler. The most important of these design parameters are the operating temperature and pressure of the fluidized bed as they will have the most influence on the efficiency and capital cost of the plant. The efficiency of the plant will depend on the temperature of the alkali metal vapor generated — a 100°F (56°K) change in boiler temperature results in about a one point change in overall plant efficiency over the temperature range of 1390 to 1540°F (1028 to 1111°K). However, the cost of the furnace-boiler will depend primarily on the boiler tube material, the surface area required, and the general arrangement of the boiler. These costs will increase as the boiler temperature increases.

With an alkali metal boiling system, the upper bed temperature restriction of 1750 to 1800°F (1228 to 1256°K) places a rather tight limit on the temperature difference between the bed and the boiler which in turn limits the heat transfer rate to the boiler and becomes the controlling factor in determining the surface area requirements and cost of the furnace-boiler. Thus, for reasons of both efficiency and economy a fluidized bed temperature of 1750°F (1228°K) will be used as the base case for this study. A temperature of 1800°F (1256°K) would be better but there is little operating experience at temperatures above 1750°F (1228°K). It should be noted that the use of a bed temperature of 1750°F may be incompatible with the use of limestone with an atmospheric bed. Based on available data on sulfur retention, a bed operating at atmospheric pressure may be restricted to lower temperatures or require that dolomite be used to absorb the sulfur.

The most attractive feature of the fluidized bed is its ability to reduce sulfur dioxide emission by absorption with limestone or dolomite. However, the reaction of SO_2 with either material is limited to a rather narrow temperature range — from a lower temperature of 1350 to 1400°F (1006 to 1033°K) to an upper temperature of 1750 to 1800°F (1228 to 1256°K). This temperature range depends on the operating pressure of the bed and whether limestone or dolomite is used.

Sulfur Retention

Although there are many variables that affect the degree to which sulfur is retained within a fluidized bed, the variables that are most important to this study are the bed operating temperature and pressure and the type of additive used to retain the sulfur, i.e., limestone or dolomite.

Combustion Temperature. Studies of the effect of the combustion temperature of the fluidized bed on sulfur retention by the National Coal Board⁹ indicate that at atmospheric pressure there is an optimum temperature between 1400°F (1033°K) and 1600°F (1144°K) at which the percent sulfur retention is a maximum, decreasing from that value at both lower and higher temperatures. The influence of bed temperature appears to be related to findings from laboratory studies of Limestone 18 that the reaction rate constant varies with temperature in a manner that depends on the fraction of the lime utilized. With less than 20% of the lime converted, the reaction rate constant continues to rise with temperature up to at least 1650°F (1072°K), but as the fraction of lime reacted increases, a peak develops in the curve, and becomes more pronounced as the percent lime utilized is increased as shown in Fig. 11. The percent sulfur retention vs temperature is shown for Limestone 18 and Dolomite 1337 for a fluidizing velocity of 4 fps (1.22 m/s) in Fig. 12. It may be observed that the curve peaks much more sharply for limestone than for dolomite.

It was found that the decrease in sulfur retention above the optimum temperature is somewhat less for lower fluidizing velocities and for higher values of Ca/S ratio. The latter effect is probably due to the decrease in lime utilization with higher values of Ca/S ratio.

It should be noted that these results are for atmospheric pressure conditions and are much different at higher pressures. Workers at Westinghouse Research Laboratories^{10,11} attribute this decrease in sulfur retention at high temperature and atmospheric pressure to too rapid a calcination of the stone. Their laboratory tests indicate that when calcination takes place in nitrogen, which permits fast rates of calcination, the resulting oxide has a low capacity for sulfate formation. However, when pressurized systems are used, the partial pressure of the carbon dioxide prevailing is

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Fig. 11. Variation of reaction rate constant with temperature and percent utilization of Limestone 18 for a particle size of 1000 - 2060 μm .

Fig. 12. Sulfur retention vs bed temperature for Limestone 18 and Dolomite 1337 with Pittsburgh coal. (Ref. 9).

relatively high at temperatures up to 1742°F (1223°K) and calcination is slowed down. Thus, the resulting material retains its sulfur removal capability to higher temperatures.

These Westinghouse results suggest that good sulfur retention may be possible at high temperature and atmospheric pressure if the stone is calcined under controlled conditions prior to being fed to the furnace.

Operating Pressure. An increase in operating pressure can be expected to inhibit calcination. This is because at atmospheric pressure the partial pressure of CO_2 is below the equilibrium value with CaCO_3 at temperatures above about 1380°F (1022°K). At pressures greater than 2 atm, the partial pressure of CO_2 with MgCO_3 is very high, so that the MgCO_3 in dolomite can be expected to calcine at increased pressure, giving access to internal surfaces and better sulfur retention for dolomite than for limestone.

Results from tests reported by the National Coal Board⁹ on the effects of pressure showed a reduction in sulfur retention for Pittsburgh coal with Limestone 18 from 77 to 66% when the pressure was increased from 1 to 5 atm (0.1 to 0.5 MPa). For the same increase in pressure, the sulfur retention increased from 80 to 98% with Dolomite 1337 without recycle of carryover fines. With recycle, the sulfur retention was reduced from 99 to 94% with Dolomite 1337 going from 1 to 5 atm (0.1 to 0.5 MPa) pressure. Dolomite 1337 was more effective on both a molar and total weight basis than Limestone 18 when operating at 5 atm (0.5 MPa) pressure.

These results were all based on relatively low bed temperatures [1500°F (1089°K) and less]. As noted earlier, laboratory results from Westinghouse^{10,11} show that higher operating pressures are beneficial for higher temperature operation. At higher pressures, the higher partial pressure of carbon dioxide slows the rate of calcination and results in a material with a much better sulfur removal activity.

The good high pressure-high temperature sulfur retention properties of dolomite and limestone have been demonstrated in tests at Exxon.¹² These experimental results were obtained from a system operating at 8 to 9 atmospheres (0.8 to 0.9 MPa) and burning an eastern coal containing 2.5% sulfur. Fluidizing velocities were 4 to 6 ft/sec (1.2 or 1.8 m/s) in

expanded beds of 3.5 or 10 ft (1.1 or 3.0 m). Temperatures were varied from 1500 to 1750°F (1089 to 1228°K) with excess air from 0 to 140%. The results of these tests are shown in Fig. 13. These results are presented as emissions in ppm rather than as percent retention in the previous curves. For 2.5% sulfur in the coal, the retention would have to be in the 70 to 75% range to meet EPA standards and the corresponding emissions for these tests would be about 600 ppm.

The conclusions presented in Ref. 12 are that limestone is an adequate sorbent at high pressure with a Ca/S ratio of 2.5 being sufficient, and that dolomite is a more effective sorbent with a Ca/S ratio of 1.5 being sufficient. However there may be operating problems with dolomite because of its much greater attrition rate.

Experimental results of sulfur retention using both dolomite and limestone by Argonne National Laboratory¹³ differ somewhat from those of Exxon.¹² In the ANL tests the sulfur retention capability of Tymochtee dolomite and Limestone No. 1359 were compared at different Ca/S ratios and fluidized bed temperatures of 1650 and 1750°F (1172 and 1228°K).

"At a combustor operating pressure of 8 atmospheres, the MgCO_3 in the dolomite is calcined to MgO . The CaCO_3 does not calcine at 1650°F but at 1750°F it does, probably producing a more porous structure. The sulfur retention was better at 1750°F than at 1650°F in experiments with Tymochtee dolomite at different Ca/S ratios. In a single experiment with limestone at a Ca/S ratio of ~ 1.5 , sulfur retention was the same at 1650°F as at 1750°F. Preheating the limestone at an adequate temperature before feeding it into the combustor did not improve the sulfur retention capability in a single experiment made at a combustion temperature of 1650°F and a Ca/S ratio of ~ 1.4 . The sulfur retention capability of Tymochtee dolomite at 1750°F was superior to that of Limestone No. 1359, both on a molar basis and on a mass basis."

Attrition Rate. Limestone does have a very significant advantage over dolomite in that its attrition rate is much less. Experiments in a high pressure [8 to 9 atm (0.8 to 0.9 MPa)] fluidized bed at Exxon¹² showed limestone to have low attrition rates while dolomite was found to be marginal or unacceptable unless special provisions were made for startup conditions. For example when the unit was started with a bed of Pfizer dolomite, the

Fig. 13. Comparison of SO_2 emissions with Grove limestone and Pfizer dolomite as a function of Ca/S mole ratio. (Ref. 12)

attrition rates were so high that "when the bed reached coal ignition temperatures, the stone attrited and was entrained from the bed at such a rate that after about 15 to 30 minutes, the bed was essentially gone." "However, it was found that if a run were started with a stable bed, for example, Grove limestone and then a fragile dolomite was fed into the bed continuously, a good portion of the dolomite was retained in the bed." Even for this type of operation the attrition rate and loss from the bed was high.

Bed Operating Pressure

The fluidized bed operating pressure, in addition to its effect on sulfur retention, has an important effect on the design and cost of the furnace boiler. For example, the bed pressure will be the controlling factor in determining the bed power density and the type of boiler used.

The power output of a bed is limited by two factors; the rate at which coal can be burned, and the rate at which heat can be removed from the bed. The combustion is limited by the weight flow of air that can be fed to the unit. The fluidization characteristics and elutriation of particles from a bed are very weak functions of the pressure, being primarily a function of the air velocity. Thus, the weight flow of air and the combustion rate increase almost directly with the system pressure.

The operating depth of the bed also depends on the pressure. This results from the fact that the pressure loss penalty or pumping power required to operate the system depends on the system $\Delta P/P$ (actually the summation of the ratios of pressure drops to the pressure throughout the system) and not just the ΔP (pressure drop). The ΔP across the bed is equal to the weight of the bed per unit area and, therefore, proportional to the bed depth. Also, to insure uniform flow through the bed, the pressure drop through the air distribution plates must be some fraction of (and proportional to) the pressure drop through the bed. Thus, the two largest pressure losses in the system are directly proportional to the bed depth. In view of this dependency, a general rule-of-thumb is that the maximum depth of a bed should not exceed about 3 ft (0.9 m) per atm (0.1 MPa) of operating pressure.

The maximum allowable operating bed depth will also influence the type of boiler used. One boiler which is currently favored by this study is a vertical tube natural circulation unit. For this unit to be practical, a

bed depth of about 10 ft (3.1 m) is required to minimize the number of tubes, the tube to header welds, and headering and manifold problems. However, if the bed pressure is restricted to 1 atm (0.1 MPa) then the bed would be much larger in area and shallow. This would require a horizontal tube boiler and, in turn, require a forced circulation type unit.

A deep bed will also enhance combustion efficiency and reduce the amount of unburned coal carried from the bed. Atmospheric units, in order to obtain a reasonable bed power density, operate with rather high velocities and, thus, carry from 10 to 15% of the coal from the bed. This coal is usually recycled to a special burnup cell for final combustion. A pressurized system permits one to obtain a high power density without high velocities and also permits the use of a deeper bed. Exxon experiments¹² showed combustion efficiencies of 92 to 99%. It was concluded from these tests that recycling fines collected in the first stage cyclone separator to the bed will yield combustion efficiencies of 99% without a special burnup cell.

Bed Operating Temperature

As noted in the introduction to this section, high bed temperatures are very important to the alkali metal system and a temperature of 1750°F (1228°K) was specified. A somewhat higher temperature, perhaps 1800°F (1256°K) would be better but there is very little operating data available at this temperature. Thus, the choice of 1750°F (1228°K) may be conservative and represent a penalty to the system under study.

In addition to the advantage of higher temperatures in reducing the surface area requirements of the boiler, the higher temperatures also improve the combustion efficiency. This effect is shown in Fig. 14. These data are from tests run by BCURA and reported in Ref. 10. These results were obtained with velocities of 0.67 to 0.76 m/s (2.2 to 2.5 ft/sec), a pressure of 456 kPa (4.5 atm), a bed depth of 1.22 to 1.43 m (4.0 to 4.7 ft) and excess air of 14 to 20%.

Bed Fluidization Characteristics and Heat Transfer

The analytical methods used in this section are those reported by Gelperin and Einstein.¹⁴ The heat transfer equations and analysis which

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Fig. 14. Effect of bed temperature on combustion efficiency with 17% excess air. (Ref. 10)

follow are based on vertical tubes in the bed. The reported heat transfer coefficients for horizontal tubes are about the same as for vertical tubes and are discussed briefly at the end of this section. The vertical tube heat transfer data were obtained from experimental studies in beds ranging from 7 to 20 in. (18 to 51 cm), with tube sizes from 0.75 to 1.5 in. (1.9 to 3.8 cm) OD, arranged in a triangular array with ratios of the tube pitch to tube diameter of 1.25 to 5.0.

The heat transfer rate in a fluidized bed is higher than in either a fixed bed or an empty tube with single phase gas flow. The general dependence of the heat transfer coefficient on the superficial velocity in a fluidized bed is shown in Fig. 15.¹⁵ The heat transfer coefficient increases slowly in the fixed bed until the superficial velocity attains a value that will initiate fluidization. A further increase in the velocity results in a rapid increase in the heat transfer coefficient. At some "optimum velocity" the heat transfer coefficient reaches a maximum and then decreases slowly with further increase in the superficial velocity. This decrease is apparently caused by a progressively larger fraction of the bed being occupied by gas bubbles that form in the course of pulsations so that conductive heat transfer from the solid particles to the tubes is reduced. The rate of increase of the heat transfer coefficient after fluidization is initiated and the difference between the velocity at this minimum fluidization and that at the optimum for heat transfer depends on the Archimedes number.

Gelperin gives the following equations to relate the important heat transfer variables for the transfer of heat to a bundle of vertical tubes from a fluidized bed.

$$Re_{mf} = \frac{Ar}{1400 + 5.22(Ar)^{1/2}} \quad (1)$$

$$Re_t = \frac{Ar}{18 + 0.61(Ar)^{1/2}} \quad (2)$$

$$Re_{opt} = \frac{Ar}{18 + 5.22(Ar)^{1/2}} \quad (3)$$

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Fig. 15. Typical dependence of heat transfer coefficient on the superficial velocity in fluidized beds.

$$Nu_{\max} = 0.75 Ar^{0.22} \left(1 - \frac{D_t}{S_t} \right)^{0.14} \quad (4)$$

where

- Ar = Archimedes number, $gd^3 \rho_f (\rho_s - \rho_f) / \mu_f^2$
- Re = Reynolds Number, $d V_f \rho_f / \mu_f$
- Re_{mf} = Value of Re at minimum fluidization
- Re_t = Value of Re at single particle terminal velocity
- Re_{opt} = Value of Re at maximum heat transfer coefficient between bed and heat transfer surface
- Nu = Nusselt number for heat transfer between bed and heat transfer surface, hd/k_f
- Nu_{max} = Maximum value of Nu
- d = Equivalent spherical diameter of solid particle
- D_t = Tube outside diameter
- g = Acceleration of gravity
- h = Heat transfer coefficient between bed and heat transfer surface
- k_f = Thermal conductivity of fluidizing fluid
- S_t = Horizontal (transverse) pitch of tube bundle
- V_f = Superficial fluid velocity
- μ_f = Fluidizing fluid viscosity
- ρ_f = Fluidizing fluid density
- ρ_s = Solid particle density

Fluidization Characteristics. The most important parameter is the Archimedes number which depends primarily on the mean particle size in the bed pressure. This relationship is shown in Fig. 16 for a bed operating temperature of 1750°F (1228°K). Figure 17 shows the values of

Fig. 16. Archimedes number as a function of particle size and system pressure.

Fig. 17. Terminal, minimum fluidizing and optimum Reynolds number and maximum Nusselt number as a function of the Archimedes number.

Re_{mf} , Re_t , Re_{opt} , and $Nu_{max}/(1 - D_t/S_t)^{0.14}$ as a function of the Archimedes number while Fig. 18 shows the ratios of the various Reynolds numbers as a function of the Archimedes number. It is obvious that the Reynolds number for all planned ranges of operation should be between that for the minimum fluidization and the terminal velocity. The relation between the design point Reynolds number and the optimum Reynolds number will depend on several factors. For example, with a bed using small particles with a low Archimedes number, the value of Re_{opt}/Re_{mf} is large and the design point may be specified at the Re_{opt} and still have sufficient operating range below the value of Re_{opt} for turndown to lower power operations. However, for beds operating at high power densities (higher superficial velocities, large particles and higher pressure), the value of Re_{opt}/Re_{mf} becomes smaller and the full power design value of the Reynolds number should be higher than the value of Re_{opt} . This then will allow "operating room" for turndown without getting too close to the point of minimum fluidization.

The effects can also be seen in Fig. 19 which shows the terminal, minimum fluidizing and optimum velocity as a function of both bed pressure and mean particle size. For example, for small particles and low pressures, the ratio of the optimum to minimum fluidizing velocity is large while for large particles and higher pressures the ratio becomes much smaller. Thus, the design point selected for the plant will depend on the Archimedes number, the plant turndown specified and also on the method used to obtain the turndown; i.e., the plant control system.

In order to understand these effects, let us examine several possible control modes for a high power density bed in which the full-power operating Reynolds number is greater than the value of Re_{opt} . The simplest control method is to keep the air flow rate and pressure constant and reduce the coal feed rate. This method can be used with a high temperature boiling system, such as the alkali metals, in which a small change in the bed temperature gives a large percentage change in the temperature difference between the bed and the boiler, and, therefore a large change in the heat transferred to the boiler. However, with a lower temperature boiling system such as water, this method of control will lead to a very large reduction in bed temperature, which, in turn, can reduce the bed temperature so low that sulfur retention

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Fig. 18. Ratios of Reynolds numbers as a function of the Archimedes number.

Fig. 19. Terminal, minimum fluidizing, and optimum velocity as a function of pressure and mean particle size.

is impossible. With low temperature boiling systems, the effective surface area between the bed and the boiler tubes must be reduced to maintain a proper bed temperature. The most serious disadvantage of this control mode is that the excess combustion air becomes very high at reduced loads which increases stack gas losses and reduces the plant operating efficiency. However, the bed fluidization characteristics remain fairly uniform during the turndown.

Another control method would be to decrease the air flow rate to match the power output while keeping the furnace pressure and the fuel/air ratio constant. This method will affect the fluidization characteristic of the bed. The Archimedes number is independent of the velocity and the values of Re_{mf} , Re_t , and Re_{opt} will remain fairly constant (with the exception of the effects of temperature). However, the actual operating Reynolds number will decrease in direct proportion to the reduced air flow rate. For this case the full power design Reynolds number should be well above the Re_{mf} and Re_{opt} so that good fluidization and heat transfer coefficients are maintained at the lower flow rates. This method is also more suited to high temperature boiling systems than for steam. With an alkali metal boiler a large turndown can be achieved and still maintain reasonable bed temperatures. With steam, the surface area of the tubes in contact with the bed must be decreased to maintain reasonable bed temperatures.

A third method of control which can be used with a pressurized system is to vary the system pressure while maintaining a nearly constant velocity through the bed and a constant fuel/air ratio. The advantage of this type of control is that both the Reynolds number and Archimedes number are proportional to the pressure. During a turndown, the operating velocity remains constant while the minimum fluidizing velocity changes very little (see Fig. 19). The optimum velocity will rise during a turndown but at a rate less than the power change. For this case, the ratio of the actual velocity to the optimum and minimum fluidizing velocities change much less than for the case in which the turndown is achieved by a reduction in the velocity. Again, the control is much simpler with an alkali metal boiler than with steam.

In summary, the full power design point will depend on the control mode used for turndown. For a system in which turndown is achieved

primarily with reduced pressure, a design point Reynolds number of 1.5 to 2.0 times the Re_{opt} would be satisfactory. For this case, the design point heat transfer coefficient would be about 95% of the maximum. For a system in which turndown is achieved by reducing the velocity, the design point Reynolds number should be about 3 times the Re_{opt} and the heat transfer coefficient would be somewhat less, perhaps 90% of the maximum.

Heat Transfer. The current knowledge concerning heat transfer in fluidized beds is far from precise. This was emphasized by Gelperin¹⁴ with the statement, "there are as yet no reliable formula for calculating h_{max} at high temperature (even more so at high pressure)." However, as noted previous, the Gelperin correlations are being used in this study.

The relation between the heat transfer coefficient and the bed parameters is shown in Figs. 20, 21, and 22 for vertical tubes. Figure 20 shows the heat transfer coefficient as a function of the mean particle size for a bed temperature of 1750°F (1228°K), a pressure of 60 psia (0.41 MPa) and a ratio of the tube spacing to tube diameter of 2. Figure 21 shows the effect of pressure on the coefficient. The reference point for this figure is 60 psia (0.41 MPa) and the abscissa represents a correction factor for other pressures. Figure 22 shows the effect of the ratio of the tube spacing to tube diameter with the abscissa representing a correction factor for values other than 2. The effects of bed temperature, using Gelperin's correlation, is very small — decreasing about 1% as the bed temperature decreases from 1750°F (1228°K) to 1500°F (1089°K). It should be noted that these are the maximum value and for most operations the coefficient will be 5 to 10% lower.

Exxon¹⁶ measured heat transfer coefficients with vertical tubes in a pressurized bed during one of their runs and reported values of 55 to 65 Btu/hr-ft²-°F (312 to 368 W/m²-°K). These tests were run at 8 to 9 atm (0.8 to 0.9 MPa) and temperatures of 1700 to 1750°F (1200 to 1228°K). The mean particle size of the bed material was not given. However, they reported that limestone sized to 2800 - 700 μ (0.110 - 0.028 in.) and dolomite sized to 3300 - 1200 μ (0.130 - 0.047 in.) was used during the testing period when the coefficients were obtained. Based on the Gelperin correlation, this range of values for the coefficients would be obtained with mean particle sizes of from 0.060 to 0.10 in. (1540 to 2540 μ) which is in the range reported by Exxon.

Fig. 20. Heat transfer coefficient as a function of particle size for 1750°F (1228°K), 60 psia (0.41 MPa) and a tube spacing to tube diameter ratio of 2.0.

Fig. 21. Heat transfer coefficient pressure correction factor.

Fig. 22. Heat transfer coefficient correction factor for the ratio of tube spacing to tube diameter.

Gelperin¹⁴ also gives correlations for horizontal tube banks with the tubes both in-line and staggered. For the in-line arrangement the equation is

$$Nu_{\max} = 0.79 Ar^{.22} \left(1 - \frac{D_t}{S_h} \right)^{0.25}$$

and for staggered tubes the equation is

$$Nu_{\max} = 0.74 Ar^{.22} \left[1 - \frac{D_t}{S_h} \left(1 + \frac{D_t}{S_v + D_t} \right) \right]^{.25}$$

where

D_t = tube OD

S_h = horizontal spacing of tubes

S_v = vertical spacing of tubes

Tests conducted at BCURA with horizontal tubes and reported in Ref. 10 gives values for the heat transfer coefficient of 67 Btu/hr-ft²-°F (380 W/m²-°K) at 1650°F (1172°K) and 71 Btu/hr-ft²-°F (403 W/m²-°K) at 1750°F (1228°K). The fluidizing velocity for the tests was 2.2 to 2.5 ft/sec (0.67 to 0.76 m/s), the expanded bed depth was 4.0 to 4.7 ft (1.22 to 1.43 m) and the pressure was 4.5 to 5.0 atm (.45 to .50 MPa). The particle size was not given.

Preliminary results of heat transfer experiments made at FluidDyne (under subcontract with ORNL) with an atmospheric pressure bed gave maximum coefficients of about 90 Btu/hr-ft²-°F (510 W/m²-°K). These tests were run with horizontal tubes in the bed. The tubes — part were air cooled and part were water cooled — were 1/2 in. (1.2 cm) OD and staggered with a horizontal spacing of 3 in. (7.6 cm) and a vertical spacing of 2.5 in. (6.4 cm). The maximum fluidizing velocity in these tests was about 2.2 ft/sec (0.67 m/s) and the mean particle size in the bed was about 460μ (0.018 in.). The heat

transfer measured was about 25% above that predicted by the Gelperin equation.

With the small particle size used in these tests, the ratio of Re_{opt} -to- Re_{mf} is large and operation at velocities below the Re_{opt} would be expected. The variation in the heat transfer coefficient with fluidizing velocity is shown in Fig. 23. The value of the ratio of Re_{opt} -to- Re_{mf} was about 10 for these tests while the Archimedes number was a little over 300.

Figure 24, from Ref. 17, shows the variation of the heat transfer coefficient as a function of particle size. The test conditions for these data were not given but it is believed that they are for atmospheric pressure. Figure 24 also shows a dependence of the coefficient on the temperature of the tubes. This is not surprising as many heat transfer correlations are based on evaluating the fluid properties on the film temperature (usually on the average of the fluid and the surface temperature). In general, these results agree well with that predicted from Gelperin's correlation for particle sizes in the range of 0.04 to 0.08 in. (1016 to 2032 μ). However, they are higher than predicted by Gelperin for smaller particle sizes and smaller for larger particles as the dependence on particle size is greater than the 0.34 power used by Gelperin. These results agree rather well with the FluidDyne tests.

Furnace Power Density

The furnace power density, measured in power output per unit area, depends on the combustion air passing through the bed and the heat transfer rate to the boiler tubes to maintain the desired bed operating temperature. In analyzing the power output of the furnace-boiler, the following assumptions are made: the bed temperature is 1750°F (1228°K), the ratio of the weight of combustion products to the weight of air is 1.08, the products have the properties of air, the amount of excess combustion air is 15%, and the heat released (to the boiler tubes and combustion products) in the furnace is 1058 Btu/lb (2.5 MJ/kg) of air or 980 Btu/lb (2.3 MJ/kg) of products. The velocity, unless noted otherwise, is the superficial velocity through the furnace and is evaluated on the basis of the weight of combustion products.

Fig. 23. Heat transfer cefficient vs fluidizing velocity.

Fig. 24. Effect of mean particle size and tube temperature on fluidized bed heat transfer coefficients. (Ref. 17)

The total heat given to the boiler tubes and the combustion air is directly proportional to the weight of air available to the combustion process and can be expressed in terms of the system pressure and the superficial velocity. For the assumptions listed above, the total heat released to the boiler and combustion products can be expressed as

$$Q_F = 4300 PV$$

where

Q_F = heat to boiler and combustion products (Btu/hr)

P = pressure at the top of the bed (psia)

V = superficial velocity at the top of the bed (ft/sec)

The maximum velocity that can be used is limited by the elutriation rate of bed material, including unburned fuel, and by the ability of the boiler tubes to maintain the proper bed temperatures. For an atmospheric pressure bed, the upper limit on the velocity is about 10 ft/sec (3 m/s) and on the maximum power density is in the range of 150 to 200 kW/ft² (13.9 to 18.6 kW/m²). For such high velocities the unburned carbon carried from the bed may be in the range of 10 to 15%, requiring collection and feeding to a special burnup cell to recover the heating value of this material.

For a pressurized system, the power density can be increased without serious elutriation problems. For example, at a pressure of 4 to 5 atm (0.4 to 0.5 MPa) and a velocity of 5 ft/sec (1.5 m/s) one can obtain a power density in the range of 375 to 475 kW/ft² (34.8 to 44.1 kW/m²). Furthermore, the higher pressure permits a deeper bed which, in turn, gives a larger bed volume and thus permits a larger surface area of boiler tubes in contact with the bed.

The heat transfer to the boiler tubes is determined primarily by the heat transfer coefficient between the bed and the tubes — about 95% of the temperature difference between the bed and the alkali metal boiling temperature is between the bed and the outer surface of the tubes. The following analysis is based on the use of a deep bed [>10 ft (>3 m)] and, therefore, a pressurized system. In this system, it is also assumed that the boiler

consists of vertical tubes, 1.0 in. (2.54 cm) OD, arranged in a triangular pitch throughout the furnace. With this type of boiler, there will be 10 to 15 ft (3.0 to 4.6 m) of tubes in the volume above the bed which will drop the temperature of the combustion products from 1750°F (1228°K) to about 1625°F (1158°K) with a 1540°F (1111°K) boiling temperature. Of course, for lower alkali metal boiling temperatures, the combustion products temperature will also be lower.

The heat generated that goes to the boiler and that going to the combustion gases will depend on the type of combustion air system. For example, with a high pressure combustion air system with power extracted by a gas turbine system, about 75 to 80% of the heat generated goes to the boiler. For the intermediate pressure system, 88 to 92% of the heat goes to the boiler. The analysis which follows will use a value of 85%. It should be noted that the high pressure combustion air system requires only 85% of the boiler surface area required for the intermediate pressure system.

As noted previously the power generation rate in the furnace is 4300 PV, and for this example, 85% or 3655 PV goes to the boiler. The heat transfer to the boiler tubes per square foot of bed area can be expressed as

$$Q_B = UALN\Delta t$$

where

Q_B = heat transferred to the boiler per square foot of bed

U = overall heat transfer coefficient

A = outside tube surface area per ft of length

L = fluidized depth of bed

N = number of tubes per square foot of bed

Δt = temperature difference between the bed and the alkali metal

thus,

$$UALN\Delta t = 3655 \text{ PV}$$

and

$$UL\Delta t = 13,970 \text{ PV}$$

The number of tubes per square foot of the bed is a function of the ratio of the tube spacing to the tube diameter and for 1 in. (2.54 cm) OD tubes can be expressed as

$$N = 166.3 \frac{D_t^2}{S_t}$$

The required value of $UL\Delta t$ then becomes

$$UL\Delta t = 84 \text{ PV} \frac{S_t^2}{D_t}$$

Figure 25 shows the bed power density in kW/ft^2 as a function of PV while Fig. 26 shows the required value of $UL\Delta t$ as a function of PV and S_t/D_t to maintain the bed temperature. These curves give a quick method of evaluating a set of design conditions. For example, with a pressure of 60 psia (0.41 MPa) and a superficial velocity of 5 ft/sec (1.5 m/s), the furnace power density is 380 kW/ft^2 . For an alkali metal boiling temperature of 1540°F (1111°K) the Δt is 210°F (117°K). The heat transfer coefficient will depend on the pressure, the mean particle size and the value of S_t/D_t and will require an iterative solution. However, for the conditions described here, an overall coefficient of about $60 \text{ Btu/hr-ft}^2\text{-}^\circ\text{F}$ ($340 \text{ W/m}^2\text{-}^\circ\text{K}$) would be reasonable. The required length for a value of $S_t/D_t = 2$ would be 7.9 ft (2.4 m). This would not be the most economical choice for the boiler as the pressure of 60 psia (0.41 MPa) warrants a deeper bed. Thus, the value of PV could be increased or the tube spacing increased. Increasing the value of PV would reduce both the size of the furnace and the number of boiler tubes, while the second option keeps the furnace size constant and reduces the number of boiler tubes.

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Fig. 25. Bed power density as a function of the product of bed pressure and superficial velocity.

Fig. 26. Required value of $UL\Delta t$ as a function of the product of bed pressure and superficial velocity.

Summary of Furnace-Boiler Design Considerations

The most important design variables influencing the economics and performance of the furnace-boiler are the bed operating temperature and pressure, operating bed depth, SO₂ absorbent material, mean particle size in the bed, and the ratio of the spacing-to-diameter of the boiler tubes.

Temperature. With the high boiling temperature of the alkali metal systems, the temperature difference between the bed and the boiler is restricted to a rather narrow limit and becomes the controlling factor in determining the boiler surface area required. The economic incentive is to specify as high an operating temperature as possible. However, there is little operating experience above 1750°F (1228°K), and this temperature is used as the full-power design point.

Pressure. The system operating pressure is the controlling factor in determining the furnace power density and the permissible operating bed depth. It also influences the bed heat transfer coefficient, the type of boiler design, the control system, and in combination with the temperature has a significant effect on the required Ca/S ratio required to control SO₂ emissions. The atmospheric pressure system is far less attractive than the intermediate or high pressure systems. The power density is low, the bed depth is restricted to about 3 ft (0.91 m), a forced circulation boiler would be required, the furnace turndown, although probably not a critical item, would be more limited than with a pressurized system, and at bed temperatures of about 1600 to 1650°F (1144 to 1172°K) the Ca/S ratio for sulfur retention may become excessive. Also, as noted in the cycle analysis, the atmospheric pressure system has a significantly lower plant efficiency.

Bed Depth. The largest pressure loss in the system is that associated with the fluidized bed; i.e., the pressure drop across the bed which is equal to the bed weight per unit area, and the pressure drop across the air distributor plate which is proportional to the pressure drop across the bed. The effect of these losses on the system performance is measured by the value of $\Delta P/P$, thus the operating depth of the bed will be limited by the system pressure. In general, the maximum depth of the bed should not exceed about 3 ft (0.91 m) per atm (0.1 MPa) of operating pressure. Most operating experience, even at pressures up to 10 atm (1.0 MPa), has been limited to depths of 10 to 15 ft (3.0 to 4.1 M). In this study the maximum

depth to be considered will be approximately 3 ft (0.91 m) per atm (0.1 MPa) with an upper limit of 15 ft (4.1 m).

Absorbent Material. In an actual plant the choice of absorbent material will probably depend on local resources available. In this study limestone with a Ca/S feed ratio of 2.5 will be used. This selection is based primarily on the results of experiments by Exxon⁵ which showed limestone to have far superior resistance to attrition with adequate SO₂ retention at a Ca/S of 2.5. Although their results were with a 2.5% sulfur coal, the data indicate that the Ca/S ratio of 2.5 should be sufficient for the 3.9% sulfur in the Illinois No. 6 coal selected for this study.

Mean Particle Size. The mean particle size is the controlling variable in the Archimedes number which, in turn, determines the bed fluidization and heat transfer characteristics. The selection of the particle size represents a tradeoff between having a low minimum fluidizing velocity and a high heat transfer coefficient with small particles and having a high terminal velocity and a low elutriation rate with large particles. The final selection of particle size will depend on the system pressure and method of control during turndown and will probably be in the range of 0.04 to 0.05 in. (1016 to 1270 μ m).

Ratio of Tube Spacing-to-Diameter. The ratio of the tube spacing-to-diameter will influence the bed power density by its effect on the heat transfer coefficient and the heat transfer area per square foot of bed. The limitations on how close the tubes may be spaced are not known. Although Gelperin¹⁴ reports data on geometries with tube spacing-to-diameter as small as 1.25, this closeness of spacing may limit radial mixing and require very uniform coal feed to the bed. For the current study the minimum tube spacing-to-diameter will be about 2.

Alkali Metal Boiler Design and Analysis

The initial design work on the furnace-boiler has envisioned three pressurized units for a 200 MW(e) plant. Assuming a furnace efficiency of 86% and an overall plant efficiency of 46%, the heat load on each of the three boilers is about 120 MW(t). Each unit will have a fluidized bed about

20 ft in diameter. The boiler will consist of vertical tubes arranged into bundles or modules (see Fig. 27). The modular concept is used for headering purposes, to facilitate assembly of the units, and for ease of maintenance and repair, as necessary. The boiler tubes, which will probably be Incoloy 800, are 1.0 in. (2.54 cm) OD, 20 to 25 ft (6.1 to 7.6 m) long, and located on approximately 2.0 (5.08 cm) centers throughout the furnace.

A parametric study was made of the effects of the furnace-boiler configuration on the performance of the alkali metal boiler. Examples of the performance characteristics of interest are the exit quality of the boiling mixture, the natural circulation flow rate, the exit temperature of the combustion gases, and flow velocities in various parts of the boiler circuit. Since this part of the study was to be of a parametric nature, the calculation scheme was necessarily devised to be as general as would be practical. The complexity of the resulting general calculation scheme required that it be coded for computer evaluation. Although the code can accommodate a wide range of variables, it is restricted to a vertical tube, natural circulation unit.

Performance maps of the cesium and potassium boilers are presented in Figs. 28 and 29 for 1540°F (1111°K). The probable operating range for the respective boilers are indicated on the figures. By varying the heat transfer coefficient in the fluidized bed (which varies the heat flux to the boiler tubes), an upper limit for the boiler output with stable natural circulation was not found for exit quality less than saturated vapor. The stability criterion was the Leddineg criterion²⁰ which essentially states that at a given operating point, any decrease in mass flow would result in a decrease in pressure drop and, likewise, an increase in mass flow would produce an increase in pressure drop around the loop. It is believed that the large liquid driving head contributes to the high degree of stability of the boiler.

Alkali Metal Condenser-Steam Boiler

The reentry type tube shown in Fig. 30 is currently under study as an alkali metal condenser-steam boiler unit. Feedwater is admitted to the bottom of the central tube where it boils as it rises to the top. Vapor

Fig. 27. Conceptual design of a pressurized fluidized bed furnace-alkali metal boiler unit.

Fig. 28. Performance map for cesium boiler.

Fig. 29. Performance map for potassium boiler.

Fig. 30. Section through a reentry tube for the alkali metal condenser-steam boiler.

emerges from the top of the central tube, flows vertically downward in an annulus between the inner and outer tubes, and exits at the bottom as superheated steam. The alkali metal vapor will condense on the outside of the outer tube, and a film of liquid metal will flow down the tube to a sump at the base of the condenser.

A computer program has been prepared to analyze a reentry tube for this application. Presently the unit is being designed for a steam pressure of 2000 psi (13.8 MPa). The design procedure yields the overall tube height corresponding to given temperatures of the inlet feedwater, the outlet steam and the condensing alkali metal. One important result of the reentry tube analysis will be the acceptable temperature differences between the feedwater, steam, and alkali vapor at the bottom of the tube.

Analysis of the alkali metal condensation will result in outer tube wall temperature, liquid metal film thickness and condensation coefficient as functions of tube length. The tube wall temperatures must be known for an accurate evaluation of the thermal stresses in the outer tube. It may be necessary to place fins on the outside surface of the central tube in order to transfer more heat from the condensing alkali metal to the inner tubes.

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