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Coal-Fired Alkali Metal Power System Design Study

Technical Progress Report for Period
April 1, 1977–June 30, 1977

Garland Samuels

MASTER

OAK RIDGE NATIONAL LABORATORY

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Engineering Technology Division

COAL-FIRED ALKALI METAL
POWER SYSTEM DESIGN STUDY

TECHNICAL PROGRESS REPORT FOR PERIOD
APRIL 1, 1977 - JUNE 30, 1977

Garland Samuels

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Prepared by the
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COAL-FIRED ALKALI METAL POWER SYSTEM DESIGN STUDY
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ABSTRACT

This report is the third of a series of quarterly progress reports for the Coal-Fired Alkali Metal Power System Design Study. The effort during this quarter was devoted to the design and analysis of an atmospheric pressure fluidized bed furnace-boiler system for either cesium or potassium. The tasks receiving the most attention during this period were the selection and analysis of the basic plant arrangement, the design of the furnace-alkali metal boiler, and the analysis of the alkali metal condenser-steam boiler.

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 results in 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.

Most of the work during the second quarter of the program⁴ entailed parametric cycle analyses, various plant arrangements with both cesium and potassium as the working fluid, and for three fluidized bed pressures—atmospheric pressure, intermediate pressure [3 to 5 atm (0.3 to 0.5 MPa)], and high pressure [6 to 10 atm (0.6 to 1.0 MPa)].

Studies were also made of the information available on sulfur retention, combustion efficiency, and heat transfer coefficients in fluidized bed for the various operating conditions studied. The conceptual design of a pressurized fluidized bed furnace using a vertical tube natural circulation alkali metal boiling system was also prepared.

TECHNICAL PROGRESS

The effort during this quarter was devoted to the design and analysis of an atmospheric pressure system. The tasks receiving the most attention during this period were the selection and analysis of the basic plant arrangement, the design of the furnace-alkali metal boiler system, and the analysis of the alkali metal condenser-steam generator.

System Analysis

Several variations of the plant and cycle arrangements of the atmospheric pressure system were analyzed and a new mass and energy balance made for the system. Two variations of the combustion air system that received the most study are shown in Figs. 1 and 2. A flow diagram of the power conversion system that applies to either of these combustion systems is shown in Fig. 3. The difference between the two combustion systems is that the system of Fig. 1 uses electrostatic precipitator to remove particulates from the stack gases while the system of Fig. 2 uses bag filters. The bag filter system has a number of advantages which are discussed later.

Mass and Energy Balance

In making the mass and energy balance for the system, the following assumptions were made.

- a. All materials enter the system at 60°F (289°K) and exit at 300°F (422°K).
- b. The coal is Illinois No. 6.
- c. The combustion air flow rate is 15% in excess of that required for complete combustion of the fuel.
- d. The relative humidity of the inlet air is 60%.
- e. The fluidized bed additive material for sulfur retention is assumed to be 95% CaCO_3 .
- f. The required Ca/S ratio is 4.
- g. All CaCO_3 fed to the furnace is calcined.

Fig. 1. Schematic flow diagram of combustion system with a precipitator.

Fig. 2. Schematic flow diagram of combustion system with bag filters.

Fig. 3. Schematic flow diagram of power conversion system.

- h. Eighty-three percent of the sulfur in the coal is retained in the fluidized bed.
- i. Heat is recovered from the solids rejected from the system by air. The air flow rate for this purpose is equal to 10% of the combustion air flow rate.

The analysis of the plant combustion system was based on the use of Illinois No. 6 coal. This is the fuel being used for the Energy Conversion Alternatives Study (ECAS).⁵ 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

Higher heating value (as received)

Btu/lb	10,788
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Both the mass and energy balance were made on the basis of 1 lb (.454 kg) of dry coal. A tabulation of the material into and out of the system is as follows:

Materials In

Air		
For combustion (dry)	10.809	lb
For combustion (moisture)	.071	
For ash cooler (dry)	1.081	
For ash cooler (moisture)	.007	
Limestone		
CaCO ₃	.559	
Ash	.029	
Coal		
Dry coal	1.000	
Water	<u>.149</u>	
Total	13.705	lb

Materials Out

Gaseous Products		
From Furnace		
Combustion products	11.598	lb
Combustion air moisture	.071	
CO ₂ from CaCO ₃	.246	
Water vapor from coal	.149	
Ash cooler air (dry)	1.081	
Ash cooler moisture	.007	
Solid Products		
Coal ash	.110	
CaSO ₄	.158	
CaO	.248	
Limestone ash	.029	
Unburned carbon	<u>.007</u>	
Total	13.704	lb

The energy balance for the system is tabulated below. This balance includes the energy input from the system blowers and from the sulfation reaction in the bed. All values are based on 1 lb of dry coal.

Energy In

Coal	12,400 Btu/lb (dry coal)
Blowers	115
Sulfation reaction	<u>237</u>
Total	12,752

Energy Out

Calcination reaction	395
Gaseous products	1,433
Solid products	26
Unburned carbon	99
Radiation and convection	25
Power conversion system	<u>10,774</u>
Total	12,752

Cycle Analysis

Two plant arrangements are still being considered for the combustion system. The first, shown in Fig. 1, uses an electrstatic precipitator to clean the stack gases. The second, shown in Fig. 2, uses fabric filters. The disadvantage of the system of Fig. 1 is the 750°F (672°K) limitation on the inlet temperature to the precipitator. This limits the furnace outlet temperature and requires a greater fraction of the furnace heat output to be transferred directly to the steam superheater and reheater than with the bag filter system. This in turn reduces the overall plant efficiency. Another disadvantage of this system is that the flue gases used to dry the coal are used to cool the gas streams from the furnace and ash cooler before entering the precipitator. The water vapor from the dryer adds to the weight flow and heat capacity of the stack gases passing through the air preheater and increase the required effectiveness and size of the preheater.

The plant cycle efficiency of either arrangement (see Fig. 4) depends on the ratio of the heat that is transferred from the furnace to the steam superheater and reheater to that transferred to the alkali metal boiler. The overall plant efficiency also depends on the efficiency of both the alkali metal and the steam system. For the data of Fig. 4, the alkali metal cycle efficiency is 24.2% and the steam cycle efficiency is 44%. The efficiencies correspond to topping cycle conditions of 1540°F (1111°K) to the turbine and 900°F (756°K) to the condenser, and steam cycle conditions of 2400 psia (16.5 MPa) with superheat and reheat of 1000°F (811°K) and a condenser pressure of 1 1/2 in. Hg (5079 Pa).

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Fig. 4. System efficiency and key furnace gas temperatures as a function of the furnace energy input distribution and the percent of carbon burned in the furnace freeboard.

Figure 4 also shows the effects of combustion in the freeboard of the furnace on the temperature of the combustion products to the steam superheater and reheater. Although this freeboard combustion may amount to several percent of the total combustion rate in furnaces operating at very high fluidization velocities and bed operating temperatures of 1500 to 1550°F (1089 to 1117°K), the lower fluidization velocity [7 to 8 ft/sec (2.1 to 2.4 m/s)] and the higher bed temperature [1750°F (1228°K)] of this study should limit this value to less than 2%. Tests conducted at BCURA^{6,7} using a bed pressurized to about 4.5 atm (456 kPa) with a fluidizing velocity of 2.5 ft/sec (0.76 m/s) showed 2 to 3% of the heat input was released in the freeboard at a bed temperature of 1470°F (1072°K). However, at bed temperatures of 1650°F (1172°K) and 1750°F (1228°K) there was no detectable combustion in the freeboard. These same tests, which were run without recycle of the particulate material elutriated from the furnace, also showed an increase in combustion efficiency from 97.5% at 1470°F (1072°K) to 99.5% at 1750°F (1228°K); i.e., a reduction in the energy lost by elutriation by a factor of 5.

Figure 5 shows the effect of varying the alkali metal operating conditions on the topping cycle efficiency while Fig. 6 shows the effect of the topping cycle efficiency on the overall plant cycle efficiency. The values of efficiency shown in Fig. 5 are for cesium, however, the values for potassium are about the same. Figure 6 also shows the effect of increasing the gas temperature leaving the furnace (exit from the steam reheater). The data of Fig. 6 are for the case of no combustion in the freeboard. It should be noted that these cycle efficiencies include blower power requirements for the combustion system.

Several key system temperatures are shown in Figs. 7 and 8 as a function of the ratio of the heat transferred to the steam superheater and reheater to that transferred to the alkali metal system for the system arrangements shown in Figs. 1 and 2. Figure 7, which is for the plant arrangement with a precipitator, shows the close approach temperatures required in the preheater. These approach temperatures can be increased by increasing the stack gas temperature or by increasing the

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Fig. 5. Effect of turbine and condenser temperatures on the topping cycle efficiency (for cesium).

Fig. 6. Effect of the topping cycle efficiency and the furnace outlet gas temperature on the overall binary cycle efficiency.

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Fig. 7. Air and combustion product temperatures as a function of the furnace energy input distribution (with precipitators).

amount of the furnace energy input going to the steam superheater and reheater. However, either of these changes will reduce the plant efficiency.

The preheater approach temperatures are much larger with the plant arrangement using bag filters. As can be seen in Fig. 8, the approach temperatures and required effectiveness of the preheater are reasonable up to a furnace outlet temperature of 900°F (756°K). An increase in the furnace outlet temperature from 775°F (686°K) to 900°F (756°K) will increase the plant efficiency by about 0.5 points (see Fig. 6).

Furnace - Boiler Design

The current design of the furnace and boiler for the plant is shown in Fig. 9. The furnace consists of eight approximately 18 ft by 18 ft (5.49 m by 5.49 m) cells and two approximately 10 ft by 18 ft (3.05 m by 5.49 m) recycle cells. The larger cells each contain two boiler tube bundles while the recycle cells contain only one. All of the cells operate at a bed temperature of about 1750°F (1228°K). With this arrangement most of the particulate material carried from the eight regular cells will be removed by two large cyclone separators located at one end of the furnace. This material, which will contain a few percent of the carbon fed to the eight cells is then recycled to the two small beds located at the end of the furnace near the cyclones. The particulate material carried from the recycle cells is removed from the combustion products from these two cells by a third cyclone separator and rejected from the plant. This arrangement requires a baffle in the upper part of the furnace between the combustion products from the regular cells and the recycle cells to prevent excessive mixing between the two product streams.

The amount of carbon carried from the eight regular cells is not known very well, but with the high bed temperature it will probably be less than 5% of the carbon fed to these cells. The recycle cells have sufficient heat transfer capacity to burn up to 14% of the carbon fed to the regular cells, thus, the recycle cells will operate primarily on coal and will be provided with a coal and limestone feed system.

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Fig. 8. Air and combustion product temperatures as a function of the furnace energy input distribution (with bag filters).

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Fig. 9. Fluidized bed furnace-alkali metal boiler arrangement.

There are several reasons for using the arrangement of Fig. 9 rather than a high temperature carbon burnup cell or a more normal recycle system. For the bed operating conditions of near atmospheric pressure, 1750°F (1228°K) and a coal (Illinois No. 6) with a sulfur content of 3.9%, it may be difficult to meet the EPA emission standard for SO₂ which for this case requires that 83% of the sulfur be retained in the bed. Most atmospheric pressure systems operate with a bed temperature of about 1550°F (1117°K) which is near the optimum for minimizing the required limestone feed rate. There is little data available for systems operating at atmospheric pressure and 1750°F (1228°K) and the design limestone feed rate (a Ca/S ratio of 4) selected for this design is based on extrapolation of lower temperature results. If a high temperature [2000°F (1367°K)] carbon burnup cell is used, then sulfur carried into this bed will be released to the stack. This increased release from the burnup cell then requires a higher sulfur retention in the regular cells. This would be very difficult for the conditions of this study.

Small experimental fluidized beds often recycle the particulate carryover back to the bed. This permits the unburned carbon to make several or many passes through the bed and reduces the carbon loss from the system. However, this type of system also recycles the inert material along with the unburned carbon and leads to a very high particulate loading in the combustion products. With the plant arrangement of Fig. 9 the combustion products pass through the steam superheater and reheater at a velocity of about 50 ft/sec (15.24 m/s) and it was deemed prudent to limit the particulate loading in these products. Also, it is simpler to recycle the material to only two cells located near the cyclone separators than to transport and feed the material to all cells.

In order to obtain a high overall combustion efficiency (>98%) with this concept, it is necessary to obtain a combustion efficiency of over 90% in the regular cells and the recycle cells. Work at both BCURA⁷ and Exxon⁸ have demonstrated combustion efficiencies of 95% to 99.5% without recycle in beds operating in the range of 1650 to 1750°F (1172 to 1228°K). These results were with pressurized systems operating at lower fluidizing velocities. With the system of Fig. 9 a combustion efficiency of 95% or more would be expected from the cells and an overall efficiency of

99% should be obtainable. It should be noted that the superficial velocity from the recycle cells is about 10% less than from the regular cells and this reduction should aid the efficiency of these cells.

The amount of material elutriated from the bed is another quantity which is not known with much precision. For this system, it was assumed that 90% of the coal ash and one-third of the CaO, CaSO₄ and limestone ash was elutriated from the regular cells. For the recycle cells, the above assumptions were made in regard to fresh coal and limestone and all of the recycled material was assumed to be elutriated.

Alkali Metal Condenser-Steam Generator

Three different reentry tube configurations (see Fig. 10) were analyzed in order to determine the effect of design parameter changes on the performance, geometry and cost of the complete boiler-condenser unit. Two larger tubes of nearly the same size were compared with a smaller tube. The tube sizes are given in the table below. Notice that all dimensions correspond to standard schedule 40 pipe.

Size	Inner Tube		Outer Tube		Hydraulic Diameter For Annulus
	ID	OD	ID	OD	
1 1/2 × 2	1.610 in.	1.900 in.	2.067 in.	2.375 in.	0.167 in.
1 × 1 1/2	1.049 in.	1.315 in.	1.610 in.	1.610 in.	0.295 in.
3/8 × 3/4	0.493 in.	0.675 in.	0.824 in.	1.050 in.	0.149 in.

A computer program was used to analyze the performance of each tube. The feedwater had an inlet temperature of 450°F (506°K) and the alkali metal was condensing at 900°F (756°K). Figure 11 shows a typical set of output data for the 3/8 × 3/4 in. tube. The temperature profile throughout the tube, the tube length and the exit steam temperature may be obtained from this output.

It should be evident from Fig. 11 that the exit steam temperature is relatively constant, regardless of length. While it may at first seem surprising, this behavior is reasonable since the tube must operate between the temperature limits set by the condensing alkali metal and the inlet feedwater. As can be seen in Fig. 11, any increase in length

Fig. 10. Section through a reentry tube for the alkali metal condenser-steam boiler.

Fig. 11. Reentry tube temperature profiles for pipe sizes of 3/8 x 3/4 in. and a flow rate of 600 lb/hr.

(and thus surface area) is offset by a smaller temperature difference between the alkali metal and the steam at the top end of the tube. In other words, the degree of superheat at the top of the tube uniquely determines the tube length.

For the sake of consistency all three tube configurations were analyzed by assuming 20°F (11°K) superheat at the top of each tube. The computer results are summarized in Figs. 12 and 13. Figure 12 again shows that the steam exit temperature is relatively constant as the flow rate varies. As a result of the constant enthalpy behavior of the outlet steam, H_v , the heat load for each tube, q , is a linear function of feedwater flow rate, $\dot{m}$:

$$q = \dot{m} (H_v - H_{fw})$$

where H_{fw} = enthalpy of feedwater

Figure 13 shows the tube length as a function of feedwater flow rate. The similar length requirement of the largest and smallest tubes in Fig. 13 may possibly be explained by the character of the convective heat transfer in the annulus, since both tubes have approximately the same hydraulic diameter. Figure 13 also shows the number of tubes, N , required for each boiler-condenser unit based on a total heat load, Q , of 7.5×10^8 Btu/hr (220 MW):

$$N = \frac{Q}{q}$$

In an effort to identify the relative cost for each boiler-condenser unit, the weight of the tube matrix was computed. Based on the cross-sectional area of the tubes, A_x , the tube length L and the number of tubes, N , Fig. 14 was developed. The density of the tube material was taken as 494 lb/ft³ (7914 kg/m³) so that the weight for the total unit was expressed as

$$W = 494 A_x L N.$$

From Fig. 14 it would be more economical to use smaller diameter tubes with higher flow rates, although the higher flow rates may present problems with lengthy tubes.

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Fig. 12. Reentry tube performance for 20°F superheat from central tube, 450°F feedwater and 900°F condensing alkali metal.

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Fig. 13. Reentry tube geometry requirements for 20°F superheat from central tube and a heat load of 7.5×10^8 Btu/hr.

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Fig. 14. Reentry tube total weight for 20°F superheat from central tube and a heat load of 7.5×10^8 Btu/hr.

The steam temperature available from the condenser-steam generator will be about 50°F (28°K) higher than that required for the steam superheater in the furnace. This excess can be reduced by bypassing part of the feedwater around the condenser-steam generator and attemperating the temperature of the steam going to the superheater. This bypass flow can be used to control the furnace energy input distribution between the steam superheater and reheater and the alkali metal boiler, which, in turn, determines the combustion product temperatures downstream of the furnace.

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