Technical and Economic Assessment of Fluidized Bed Augmented Compressed Air Energy Storage System

**Volume III
Preconceptual Design**

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September 1981

**Prepared for
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TECHNICAL AND ECONOMIC ASSESSMENT
OF FLUIDIZED BED AUGMENTED COMPRESSED
AIR ENERGY STORAGE SYSTEM
VOL III - PRECONCEPTUAL DESIGN

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Foreword

Compressed air energy storage (CAES) is a technique for supplying electric power to meet peak load requirements of electric utility systems. A CAES plant uses low-cost power from base load plants during off-peak periods to compress air in an underground reservoir -- an aquifer, solution mined salt cavity or mined hard rock cavern. During subsequent daytime peak load periods, the compressed air would be withdrawn from storage, heated, and expanded through turbines to generate peak power.

Studies have shown that the CAES concept is technically feasible and, with a proper utility power generation mix is economically viable. Replacement of current oil-fired gas turbine peaking units by CAES systems could result in an annual savings of more than 100,000,000 barrels of oil. Already, a CAES plant is being operated by Nordwestdeutsche Kraftwerke AG. in Germany and other plants are being planned or considered by U.S. utilities.

In view of the potential benefits the CAES concept offers, the Department of Energy (DOE) has undertaken a comprehensive program in order to accelerate commercialization of this technology. The Pacific Northwest Laboratory (PNL) was selected by DOE as lead laboratory for the CAES Technology Program. As such, PNL is responsible for assisting the DOE in planning, budgeting, contracting, managing, reporting, and disseminating information. Under subcontract to PNL are a number of companies, universities and consultants that are responsible for various research tasks within the Program.

This report describes the results of a study subcontracted by PNL to the United Technologies Research Center on the engineering feasibility and economics of a CAES concept which uses a coal fired, fluidized bed combustor (FBC) to heat the air being returned from storage during the power production cycle. By burning coal instead of fuel oil, the CAES/FBC concept can completely eliminate the dependence of compressed air energy storage on petroleum fuels.

This study is a part of the "Second Generation CAES Systems" task of the CAES Technology Program. The general objective of the Second Generation task is to develop advanced CAES technologies that would reduce or eliminate the dependence of CAES systems on petroleum fuels. Other concepts receiving preliminary attention in this task are:

- Adiabatic CAES utilizing mined hard rock caverns for compressed air storage.

- Adiabatic CAES utilizing aquifers for storage of compressed air and part of the heat of compression
- CAES cycle integrated in a coal gasification plant
- Solar assisted CAES
- Combined CAES and coal-fired steam plant.

At the present time PNL is directing the limited research and development funding resources to advancement of CAES concepts utilizing thermal energy storage. The CAES/FBC concept appears technically feasible and economically competitive with conventional CAES. However, significant advancement is required in FBC technology before serious commercial commitment to CAES/FBC can be realized. At present, other elements of DOE, industrial groups, and other countries are performing the required R&D for advancement of FBC technology. The CAES/FBC will be reevaluated at a later date when FBC technology has matured and many of the concerns now plaguing FBC are resolved.

Foreword

This technical and economic assessment of fluidized bed combustion augmented compressed air energy storage systems was performed by United Technologies Research Center with subcontract assistance from Coal Processing Consultants Limited of the United Kingdom and Acres American Incorporated. The study program has been prepared under subcontract No. B-51299-A-L for Pacific Northwest Laboratory which is operated for the U.S. Department of Energy by Battelle Memorial Institute. The work described herein was initiated in November 1978 and completed in April 1980.

The results of this assessment effort are presented in three volumes. Volume I - Executive Summary provides a concise overview of all major elements of the assessment including the conclusions. Volume II - Introduction and Technology Assessment presents a discussion of program background and an in-depth coverage of both fluid bed combustion and turbomachinery technology pertinent to their application in a CAES power plant system. Volume III - Preconceptual Design contains the system analysis which led to the identification of a preferred component configuration for a fluidized bed combustion augmented compressed air energy storage system, the results of the effort which transformed the preferred configuration into preconceptual power plant design, and an introductory evaluation of the performance of the power plant system during part-load operation and while load following.

The United Technologies Research Center provided overall program management and major contributions in the areas of turbomachinery technology assessment, systems analysis, turbomachinery preconceptual design, power plant economics, part-load operation, and load following. This effort was performed under the general guidance of F. L. Robson, Chief, Utility Power Systems and involved the following people:

- A. J. Giramonti, Program Manager (November 1978 to October 1979)
- R. D. Lessard, Program Manager (November 1979 to April 1980)
- W. A. Blecher
- E. R. Fisher
- R. L. Sadala

Coal Processing Consultants Limited provided contributions in the areas of fluid bed combustor technology and fluid bed combustor preconceptual plant design. These efforts involved the following people:

- D. Merrick, Project Manager
- P. C. Finlayson
- H. R. Hoy
- R. Lanauze
- C. W. Maude

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W. B. N. Rowlands
J. Stanton
C. B. Tydd
D. A. Whiteman

Acres American Incorporated provided contributions in the areas of balance-of-plant preconceptual design and institutional factor identification. These efforts involved the following people:

M. J. Hobson, Project Manager
J. S. Murray
C. C. Richard

Additional assistance was provided by the Department of Energy, Electric Power Research Institute, and Potomac Electric Company in making available data and conceptual design information from their ongoing CAES assessment study.

The study program was conducted under the general direction of W. V. Loscutoff of Pacific Northwest Laboratory.

The support and assistance of all of the aforementioned individuals and organizations in contributing to the successful completion of this study program is acknowledged with sincere appreciation.

TECHNICAL AND ECONOMIC ASSESSMENT OF
FLUIDIZED BED COMBUSTION AUGMENTED COMPRESSED
AIR ENERGY STORAGE SYSTEMS

VOLUME III - PRECONCEPTUAL DESIGN

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Chapter 3

Systems Analysis

Systems Analysis

The objective of the systems analysis effort in this study program was to identify a preferred coal-fired FBC/CAES power plant system configuration -- one which could serve as the basis for a preconceptual design. The process of transforming an equipment characterization into a power plant system that can produce electric power both efficiently and cost effectively is one that requires both inventiveness and strong analytical capability. In this study program, this transformation process involved system integration, parametric performance analysis, system screening, and commercial application evaluation before selection of a preferred FBC/CAES power plant system configuration could be made.

The integration of a power plant system incorporating fluid bed combustors, gas turbines, air storage and heat recovery equipment is a complicated task, but one which was necessary in this program. The purpose of a system integration study is to identify practical power plant configurations wherein the characteristics of the major equipment components are compatible. Conducting the study requires the practical application of thermodynamics, design principles, and engineering judgment in order to obtain high levels of performance without increasing the complexity of the system to a level which would ultimately lead to excessive capitalization requirements. With this as a guideline, a number of integrated FBC/CAES power plant system configurations incorporating either pressurized or atmospheric fluid bed combustors were synthesized.

A parametric performance analysis was then performed for the more promising FBC/CAES configurations which had been synthesized. An appropriate duty cycle was chosen as representative of the utility industry as a whole; this cycle was based on a typical estimated annual usage rate and duration of peak relative to off-peak load periods. For different FBC operating pressure and exit temperature levels, estimates were made of the compressor power input, turbine power output, and coal consumption rate for each of the promising configurations by using United Technologies Corporation's SOAPP (state-of-the-art performance program) computer simulation model. The data produced were used to calculate the compression, generation, and round trip heat rates of the power plant and to prepare schematic process diagrams indicating the mass and energy flows.

To assess the commercialization potential of these FBC/CAES power plant system configurations, a preliminary screening analysis was performed to identify those configurations that possessed relatively efficient operation combined with low cost. Using approximate cost trends, correlations, extrapolations, and preliminary estimates for the major components, the relative capital cost for each configuration was estimated. Operating costs were also estimated based on projected fuel costs, off-peak compression energy costs, and heat rate. These costs then were used to provide a measure of the cost potential of the FBC/CAES configurations.

Each configuration which had both favorable economic and performance characteristics was further evaluated by comparing the relative technical risk associated with its major components. Thus, from the several configurations which were identified as having a good potential for commercial application, it was possible to select the open-bed pressurized FBC/CAES power plant system configuration as the one system with the greatest potential for early commercialization. This power plant system was then carried forward into the pre-conceptual design effort.

The elements of this system analysis effort summarized above provide the basis for the commentary in this chapter of the report.

SYSTEM INTEGRATION

Integrated power systems incorporating fluid bed combustors, gas turbines, air storage, and heat recovery equipment can become very complicated. As a result, performance of these studies require considerable ingenuity to postulate practical power plant system configurations which recognize the technological limitations of the various system components. This section contains a description of a) the approach used in synthesizing the integrated fluid bed combustor/compressed air energy storage (FBC/CAES) systems and b) the resulting integrated systems.

Basic CAES System Components

In its most simple terms, a coal-fired FBC/CAES power plant system could be considered to be similar to an oil-fired CAES system except for the substitution of a coal-fired fluid bed combustor for the oil-fired conventional combustor (Fig. 3-1). This means that the remaining elements of an FBC/CAES system configuration could be comparable to those found in an oil-fired CAES system configuration. These latter systems have been described in previous CAES power plant system studies conducted by UTRC (i.e., Refs. 3-1, through 3-4). Briefly, the remainder of the system configuration consists of: 1) a motor/generator which drives the compressor through a clutch 2) a booster compressor followed by an aftercooler; 3) an air storage reservoir; 4) an expansion turbine with a recuperator preceeding to heat the air; and 5) a low-pressure turbine which drives the motor/generator through a clutch.

The reasons for using the CAES configuration arrangement in the previous studies are valid in this study. The CAES system configuration is composed of existing hardware which would be modified for CAES applications and, therefore, should require a minimum of development. In addition, separating the CAES system configuration into low- and high-pressure components provides the opportunity to tailor the system to the air storage reservoir pressure by changing the pressure ratio across the booster compressor and the expansion ratio across the expansion turbine. This will also provide a degree of CAES system standardization, thus reducing modification costs for a specific application.

Description of FBC/CAES System Configurations

Once the balance of the CAES system configuration components has been adequately outlined, it is then possible to introduce the components associated with the fluid bed combustor and any other components which might be unique

SIMPLIFIED COAL-FIRED FBC/CAES CONFIGURATION

FIG. 3-1

to a specific FBC/CAES system configuration. In this study, these components included either a PFBC or an AFBC with the appropriate operating conditions and performance limitations identified in the preceding chapter. In addition to the combustor itself, they also included various heat transfer equipment items which are added to the system configuration with the goal of increasing performance.

In total, five PFBC/CAES system configurations and three AFBC/CAES system configurations were considered in detail. Various ranges of the operating conditions associated with these system configurations were examined. As a result, a total of twenty-one cases were simulated.

It is recognized that the FBC/CAES system configurations and operating conditions which were investigated represent only a fraction of the total number of options available. However, it was felt that the cycle configurations and their operating conditions which were selected for detailed consideration offer suitable promise for future commercial implementation. In addition, they reflect an appropriate regard for the various technological design constraints (covered in the preceding chapter of this report) which are associated with both the fluid bed combustor and the turbomachinery. Lastly, the system configurations that were investigated were relatively uncomplicated and possessed the potential for minimizing capital costs.

Pressurized FBC/CAES System Configurations

Several PFBC/CAES system configurations were considered. For the purpose of discussion, however, these configurations will be classified as having either an open bed or tubes immersed in the bed. In each of these configurations, the compression system remains the same and the only variations among the configurations are related to the manner in which the stored air is heated and expanded.

Open-Bed PFBC/CAES System Configurations

The simplest approach to integrating a coal-fired FBC and a CAES power plant system is to directly substitute a PFBC for the conventional oil-fired combustor and expand the combustion gases.

There are two combustors in the oil-fired CAES system configuration of Refs. 3-1, 3-2, and 3-3 -- one prior to the high-pressure or expansion turbine (ET) and one prior to the low-pressure turbine (LT). With the current state of PFBC technology limiting PFBC bed pressure, substitution for only one of these combustors could be considered practical for the near term, i.e., the combustor prior to low-pressure turbine. A PFBC/CAES system configuration synthesized within this pressure constraint is depicted in Figure 3-2. Air from the recuperator

COAL FIRED PFBC/CAES SYSTEM CONFIGURATION

OPEN BED

FIG. 3-2

at approximately 60 atm (bar) would be expanded through the high-pressure turbine and combusted in a moderate-pressure (10-20 atm (bar)) open-bed PFBC. The exhaust gas, after cleaning, would then enter the low-pressure turbine at essentially bed exit temperature (1562 to 1652 °F (850 to 900 °C)) and be expanded to almost atmospheric pressure, whereupon, it would be cooled in the recuperator.

The simplicity of this system configuration carries with it the potential for low capital cost. However, because the recuperator is the only source of thermal energy prior to the expansion turbine, the turbine inlet temperature would be relatively low (500 to 600 °F (260 to 315 °C)) and the output power from the expansion turbine would be relatively low. To increase the power output from the expansion turbine, other system configuration arrangements were considered.

One approach which was explored was to pass the storage air through a high-temperature heat exchanger after it leaves the recuperator and before it enters the expansion turbine (Fig. 3-3). While in the heat exchanger, the storage air would receive thermal energy from the PFBC exhaust gas which is at bed exit temperature. Of course, the increase of expansion turbine inlet temperature which could be achieved in this system configuration would be done at the expense of the low-pressure turbine inlet temperature.

It is possible to obtain significant increases in performance if the technology were extended beyond the levels anticipated in the near term. A promising advanced system configuration which was investigated would utilize a very high pressure (VHP) PFBC that would be operated at approximately 60 atm (bar). The VHP PFBC would be substituted for the oil-fired conventional combustor prior to the expansion turbine (Fig. 3-4). Air from the recuperator would be combusted in the VHP PFBC, introduced to the expansion turbine at bed exit temperature and expanded. It would then be combusted again in the moderate-pressure PFBC and admitted to the low-pressure turbine at bed exit temperature. This system configuration is highlighted despite its advanced technology characteristics and additional PFBC development requirement because it exhibited the best performance of all FBC/CAES system configurations that were studied.

Immersed-Tube PFBC/CAES System Configurations

Other methods of increasing the temperature into the expansion turbine, and, thus, the output power, were also explored. These would require placement of immersed heat transfer tubes in the PFBC. The simplest of these system configurations utilizes immersed tubes as shown in Fig. 3-5. The storage air

COAL FIRED PFBC/CAES SYSTEM CONFIGURATION

OPEN BED

HIGH TEMPERATURE HEAT EXCHANGER

FIG. 3-3

COAL FIRED PFBC/CAES SYSTEM CONFIGURATION

OPEN BEDS
VERY HIGH PRESSURE PFBC

FIG. 3-4

COAL FIRED PFBC/CAES SYSTEM CONFIGURATION

TUBES IMMERSED IN BED

FIG. 3-5

upon leaving the recuperator passes through tubes immersed in the bed prior to admission to the expansion turbine. This arrangement would permit an expansion turbine inlet temperature of around 1022 °F (550 °C). The temperature would be limited to this level rather than the bed exit temperature of 1562 °F (850 °C) to keep the tube metal temperature below those levels (approximately 1200 °F (650 °C)) at which corrosion problems could potentially occur with state-of-the-art tube materials in the PFBC environment.

One other system configuration was studied which incorporates tubes in the bed (Fig. 3-6). In this configuration, a very high-temperature heat exchanger is added prior to the expansion turbine to increase the temperature of the air to 1382 °F (750 °C). Exhaust gas from the fluid bed combustor would be passed through this heat exchanger prior to entering the low-pressure turbine, and heat would be transferred to the air which is on its way to the expansion turbine. As with the high-temperature heat exchanger open-bed system configuration of Fig. 3-3, it is recognized that the increase in expansion turbine inlet temperature occurs at the expense of the low-pressure turbine inlet temperature and at the cost of introducing a heat exchanger into the system.

Atmospheric FBC/CAES System Configurations

The use of an AFBC would do away with the potential problems of turbomachinery material corrosion and erosion since there would be no combustion gases going through the turbines. As with the PFBC/CAES system concepts, several configurations were investigated. Similarly, the compression system remains the same among the configurations with the only variations related to the manner in which the stored air is heated and expanded.

Moderate-Pressure Tube AFBC/CAES System Configurations

A system configuration incorporating moderate-pressure tubes in the AFBC bed is depicted in Fig. 3-7. Air from the recuperator would be expanded to about 15 atm (bar) in the expansion turbine, then heated in immersed tubes in the AFBC to 1022 °F (550 °C), and further heated in a very high-temperature heat exchanger located in the freeboard of the AFBC to 850 °F (1562 °C) prior to expansion in the low-pressure turbine. Upon leaving the low-pressure turbine, the air is preheated to 1022 °F (550 °C) and then combusted in the AFBC. The combustion gases would be used to heat the low-pressure turbine inlet air, to preheat the combustion air, and to provide for heating the storage air in the recuperator.

This system was synthesized to produce a simple AFBC/CAES system configuration with the potential of achieving low-pressure turbine inlet temperatures of the same magnitude as those which could be achieved with the immersed tube PFBC/CAES

COAL FIRED PFBC/CAES SYSTEM CONFIGURATION

TUBES IMMERSED IN BED
HIGH TEMPERATURE HEAT EXCHANGER

FIG. 3-6

COAL FIRED AFBC/CAES SYSTEM CONFIGURATION

TUBES IMMERSED IN BED
VERY HIGH TEMPERATURE HEAT EXCHANGER

FIG. 3-7

systems of Fig. 3-5, i.e., 1562 °F (850 °C). To do this, it was necessary to extend technology by postulating very high-temperature heat exchangers.

Although the resulting configuration in Fig. 3-7 appears more complex than the PFBC/CAES system configuration of Fig. 3-5, it is comparable in thermodynamic sophistication. The storage air passes only through the recuperator on its way to the expansion turbine. Because the exhaust gas from the fluid bed is at atmospheric pressure, it cannot be expanded in the low-pressure turbine. Therefore, it releases a portion of its thermal energy to the air, which has been previously heated in tubes immersed in the bed to 1022 °F (550 °C), in a very high-temperature heat exchanger located in the AFBC freeboard. This location has been chosen for the heat exchanger because it is possible to have gas temperatures of 1832 °F (1000 °C) due to combustion of the elutriated coal in the freeboard of the bed. This heated air is then used in the low-pressure turbine. The remaining thermal energy in the AFBC exhaust gas stream is utilized by first reheating the combustion air in a preheater and, then the storage air in the recuperator before being exhausted. Because there is still high-temperature thermal energy left in the AFBC exhaust gas stream when it reaches the recuperator, the expansion inlet temperature is relatively high.

Another approach using only moderate-pressure tubes would be to reheat the air in the low-pressure turbine. In Fig. 3-8, the air leaving the recuperator on its way to the expansion turbine bypasses the AFBC but is heated in the high-temperature heat exchanger located in the AFBC freeboard. The AFBC bed was bypassed because the air exhausting from the recuperator was already close to 1022 °F (550 °C). After leaving the expansion turbine, the air is heated in tubes immersed in the bed and further heated in the freeboard high-temperature heat exchanger. The heated air is expanded to about 4 atm (bar) in the low-pressure turbine whereupon it is reheated in the bed and further heated in the freeboard heat exchanger before being further expanded to approximately ambient pressure. Combustion air reheat is also used in this system configuration and the thermal energy transfer in the recuperator from the AFBC exhaust gas is relatively large.

Moderate- and High-Pressure Tube AFBC/CAES System Configurations

System configurations incorporating both moderate- and high-pressure tubes immersed in the bed were also synthesized. In these configurations (Fig. 3-9), storage air is passed through the recuperator, then through tubes in the bed whereupon it exits at 1022 °F (550 °C). It then passes through a high-temperature heat exchanger located in the freeboard. Both parallel and counterflow heat exchangers were investigated to obtain as high a gas temperature as possible for the turbines while limiting the maximum heat exchanger metal temperature. After passing through the expansion turbine, the air again passes through the AFBC in a separate set of immersed tubes and through the freeboard high-temperature heat exchanger before entering the low-pressure turbine. This is followed by a combustor air reheater and the recuperator.

COAL FIRED AFBC/CAES SYSTEM CONFIGURATION

TUBES IMMERSED IN BED
DUAL TURBINE REHEAT

FIG. 3-8

COAL FIRED AFBC/CAES SYSTEM CONFIGURATION

TUBES IMMERSED IN BED
HIGH TEMPERATURE HEAT EXCHANGER

FIG. 3-9

PARAMETRIC PERFORMANCE ANALYSIS

All of the FBC/CAES system performance estimates presented in this section were prepared using SOAPP.* Although the output from the SOAPP computer program is in sufficient detail to provide mass flow, temperature, pressure and enthalpy characteristics for every station point in a system configuration, for the purposes of this analysis only the pertinent performance parameters related to each of the system configurations will be presented. These performance parameters include output power, compression, generation and round trip heat rates, expansion and low-pressure turbine inlet temperatures, and bed exit conditions.

Operating Conditions and Performance Assumptions

The level of performance that a CAES power plant system achieves depends greatly on the component operating conditions and performance assumptions selected. The pertinent component values associated with the FBC and turbo-machinery technology documented in the preceding chapter provided the basis for this performance analysis. Operating conditions and performance assumptions of the remaining CAES equipment were based on previous studies conducted by UTRC (Refs. 3-1, 3-2, 3-3, and 3-4). The following sections highlight some of the more important component assumptions.

CAES Turbomachinery

The low-pressure compressor characteristics are based on those of modern, heavy-frame industrial gas turbines. The two compressor sections of seven stages and ten stages, with one stage of intercooling to keep the temperature in the compressor below 500 °F (260 °C), will compress 761.8 lb/sec (345.5 kg/sec) of air from ambient pressure to approximately 16 atm (bar). The booster compressor is modeled after an existing vertically-split, multi-stage, centrifugal compressor extended to seven stages. It is preceded by an inter-cooler and has a single stage of intercooling located after the fourth stage, both of which keep the temperature in the compressor below the design limit of 410 °F (210 °C). The booster compressor will compress the air from 16 atm (bar) to 62.9 atm (bar). (The pressure head of air in the vertical air shaft will finish the compression process by raising the air to the storage pressure of 66.3 atm (bar).)

The expansion turbine characteristics are modeled after an existing intermediate-pressure steam turbine unit modified for gas expansion application. Although this turbine is limited to 1000 °F (538 °C) because

*SOAPP - State-of-the-Art Performance Program. The chief simulation modeling tool used within United Technologies Corporation (UTC) for power plant system modeling.

of material considerations, the inlet temperature was permitted to exceed this value under the assumption that appropriate materials could be incorporated into the turbine to withstand the additional temperature. This unit has one governing and one expansion stage.

The low-pressure turbine is characteristic of modern, heavy-frame industrial gas turbine which has an expansion ratio of 16:1 and four expansion stages. It should be noted, however, the inlet temperature capability of modern gas turbines is well above the temperature level encountered in this study. The low-pressure compressors and turbine have efficiencies on the order of 90 percent (adiabatic).

The high-pressure turbomachinery has an efficiency range of 75-85 percent which is significantly lower than that for the low-pressure turbomachinery. This occurs for two reasons: 1) the performance characteristics are based on existing hardware modified to CAES applications; and 2) even if the equipment were designed specifically for CAES applications, the low-volume flow resulting from the high pressure magnifies the effects of wall friction losses, leakage at blade tips, etc.

It should be mentioned that although each of the components has technological characteristics similar to the appropriate existing hardware described above, the physical size of the equipment was not necessarily constrained to existing hardware. For example, the low-pressure turbomachinery is representative of modern, heavy-frame gas turbine aerodynamic design; consequently, the appropriate values for expansion gas flow, efficiency and cooling air flow were used. However, the expansion ratio across this low-pressure turbomachinery was allowed to vary from 10 to 1 up to 20 to 1 during the parametric performance analysis. Physically, this would change the number of expansion stages. Although this change is obviously significant from a design viewpoint, from a systems analysis viewpoint, it is significant only in that the correct number of stages must be noted along with changes in any of the other characteristics describing the performance of the low-pressure turbine.

Fluid Bed Combustor

Perhaps the most critical of the FBC operating conditions is the maximum temperature of 1022 °F (550 °C) which was used as a current technology limitation on the air inside the immersed tubes in the bed. This same constraint is also applicable to the air entering the FBC distributor plate. This restriction was brought about by a concern for the integrity of the metal which, while being required to contain pressurized air, is being subjected to a high-temperature corrosive environment in the fluid bed. By maintaining the air temperature no greater than 1022 °F (550 °C), it is possible to keep the outside tube metal temperature below a level which can be considered to be within the corrosion limit, thus inhibiting corrosion. However, maintaining the air temperature at this level severely restricts the number of viable system configurations

which could be considered because of the reduced performance that would be associated with a turbine inlet temperature of this magnitude.

The maximum exit air temperature from the convection heat exchangers placed in the freeboard of the AFBC was generally restricted to 1382 °F (750°C). (For two simulations of the moderate-pressure tube configuration, it was allowed to go to 1562 °F (850 °C)). This exit temperature from the heat exchanger is higher than the exit temperature from tubes immersed in the bed even though a similar type limitation exists on the metal temperature of the heat exchanger, i.e., 1472 °F (800 °C) because the environment in the freeboard is not as corrosive and there are different characteristics associated with the heat transfer in the freeboard as opposed to that in the fluid bed.

Additional performance characteristics include the radiation loss from the combustor and the carbon loss, both of which were assumed to be 1 percent. A sulfur retention of 85 percent was selected to meet the environmental requirements of less than 150 ppm at 15-percent excess air (corresponds to about 75-to 80-percent sulfur capture).

Remaining CAES Equipment

In previous studies performed by UTRC (Refs. 3-1, 3-2, 3-3, and 3-4), considerable attention was paid to equipment pressure losses and heat exchanger characteristics to ensure that the values used were reasonable. After reviewing the conditions associated with the operation of the remaining CAES equipment in the FBC/CAES system configurations considered, it was concluded that the main characteristics previously used were directly applicable to the present study.

The air storage temperature in the underground storage reservoir was selected to be 120 °F (49 °C). Also assumed was a 4-percent leakage loss of storage air. This resulted in a reduction of mass flow to the turbines of 4 percent relative to that of the mass flow going through the compressors so that the compression and generation times would remain equal. A minimum stack temperature of 300 °F (149 °C) was maintained to inhibit condensation of sulfuric acid in the flue gas.

The motor/generator operating both as a motor and as a generator was assumed to be 98.7-percent efficient. The overall grid system heat rate (includes base-load power plant plus transmission and distribution losses) supplying electrical power for compression was assumed to be 10,000 Btu/kWh (10,550 kJ/kWh).

PFBC/CAES System Parametric Performance

The performance for those PFBC/CAES systems described in the section entitled SYSTEMS INTEGRATION was estimated based upon the operating conditions

and performance assumptions discussed above. The performance results are summarized in Table 3-1 and are discussed in the following two sections.

Open-Bed PFBC Systems

Four sets of operating conditions were simulated for the open-bed system configuration in Fig. 3-2. First, the bed operating pressure which could be expected from current PFBC technology and which would be applied to a near-term CAES application, i.e., 10 to 15 atm (bar), was bracketed. Next, the effect on performance of an evolution in technology was simulated; first, by increasing bed temperature 90 °F (50 °C) above existing temperatures to 1652 °F (900 °C), then by increasing the bed pressure to 20 atm (bar) from 15 atm (bar) along with the elevated temperature of 1652 °F (900 °C). In reviewing both the output power and round trip heat rate resulting from this exercise, it can be seen that the increase in bed temperature produced a more marked improvement in performance than did the increase in bed pressure.

Only one set of operating conditions was simulated for the system configuration of Fig. 3-3 (open-bed with heat exchanger). The performance of this system was sufficiently unimpressive, despite the higher bed temperature, to preclude further investigation of this system configuration.

As noted before, the system configuration of Fig. 3-4 (very high-pressure open bed) produced very attractive performance characteristics. Two sets of operating conditions were investigated for the moderate-pressure PFBC representative of near-term technology (15 atm, 1562 °F (15 bar, 850 °C)) and slightly advanced technology (20 atm, 1652 °F (20 bar, 900 °C)). The bed temperature for the VHP PFBC was taken to be the same as the moderate-pressure PFBC, while the bed pressure corresponded to the air pressure at the exit of the recuperator (58 atm (bar)).

Immersed-Tube PFBC/CAES Systems

The system configuration of Fig. 3-5 with immersed tubes was the most extensively investigated system configuration, with seven sets of operating conditions simulated. Five of these seven sets were used to bracket the current technology bed operating condition of 1562 °F (850 °C) and 15 atm (bar). First, it was bracketed by bed pressure, 10 atm (bar) and 20 atm (bar), then it was bracketed by bed temperature, 1472 °F (800 °C) and 1652 °F (900 °C). An effect similar to that observed for the simple open bed resulted; i.e., a variation in bed temperature produced more significant improvements in performance than variations in bed pressure.

To evaluate the effect that air storage pressure would have on system performance, the storage pressure was alternately increased then decreased

TABLE 3-1

PFBC/CAES SYSTEM PERFORMANCE SUMMARY

	<u>Bed Exit Conditions</u>		<u>Power, MW</u>	<u>Heat Rate, Btu/kWh</u>			<u>Turbine Temperature, °F</u>	
<u>Fig.</u>	<u>Press., atm</u>	<u>Temp., °F</u>		<u>Comp.</u>	<u>Gen.</u>	<u>R.T.</u>	<u>ET</u>	<u>LT</u>
Open-Bed								
3-2	10	1562	220	9093	4335	13428	639	1562
	15	1562	224	8926	4322	13248	511	1562
	15	1652	235	8507	4309	12816	573	1652
	20	1652	238	8413	4302	12715	494	1652
3-3	15	1652	199	10023	4392	14415	674	1200
3-4	58/15	1562/1562	273	7310	4328	11638	1562	1562
	58/20	1652/1652	281	7112	4389	11501	1652	1652
Immersed Tubes								
3-5	10	1562	242	8277	4203	12480	1022	1562
	15	1562	247	8111	4191	12302	1022	1562
	20	1562	247	8080	4187	12267	1022	1562
	15	1427	234	8528	4200	12728	1022	1427
	15	1652	255	7855	4186	12041	1022	1652
	(1)	15	1562	254	8210	4178	12388	1022
(2)	15	1562	237	8025	4204	12234	1022	1562
3-6	15	1562	240	8659	4227	12886	1382	1203

(1) Storage pressure +20% (72.7 atm)

(2) Storage pressure -20% (52.3 atm)

by 20 percent. The results of this exercise could also be used to approximate the performance that could be expected in constant volume or porous media storage reservoirs wherein the pressure during compression and generation varies over a similar range (see Ref. 5-3). In viewing the performance resulting from these two simulations, it can be concluded that increased storage pressure slightly improves performance. In addition, it can be shown that the performance of a variable pressure air storage reservoir would be comparable to that obtained from a constant pressure air storage reservoir when the constant air storage pressure is equal to the average storage pressure of the variable pressure air storage reservoir.

The last PFBC/CAES configuration investigated (Fig. 3-6, immersed tube with heat exchangers) produced results comparable to, but slightly less impressive than those of the simple immersed tube PFBC/CAES configuration with similar bed operating conditions. As a result, the concept was not investigated further.

AFBC/CAES System Parametric Performance

The performance for the AFBC/CAES systems, are summarized in Table 3-2 and discussed in the following two sections.

Moderate-Pressure Tube AFBC/CAES Systems

Two sets of operating conditions were simulated for the system configuration in Fig. 3-7 (moderate-pressure tubes with very high-temperature convective heat exchanger). For the first simulation, the expansion turbine air temperature was set at 1022 °F (550 °C) with the exit temperature from the freeboard heat exchanger at 1562 °F (800 °C). For the second set, the expansion turbine inlet temperature was permitted to reach 1231 °F (666 °C). This was accomplished by preheating the air less, thus diverting more thermal energy to the recuperator. The results of the exercise revealed that the increased expansion turbine inlet temperature contributed to an increase in output power. It also contributed to producing system performance which, unlike the other AFBC/CAES system investigated, is comparable to the PFBC/CAES systems. However, it should be noted that the level of performance attained was achieved by incorporating very high temperature heat exchangers. These heat exchangers represent a significant advance in technology. It should be further noted that comparable PFBC/CAES performance was obtained from systems incorporating near-term technology components.

Only one set of operating conditions was simulated for the system configuration of Fig. 3-8 (moderate-pressure tubes with dual turbine reheat). The system configuration is more complex, but the system does attain fair performance at low heat exchanger temperatures.

TABLE 3-2

AFBC/CAES SYSTEM PERFORMANCE SUMMARY

	<u>Bed Exit Conditions</u>		<u>Power, MW</u>	<u>Heat Rate, Btu/kWh</u>			<u>Turbine Temperature, °F</u>	
<u>Fig.</u>	<u>Press., atm</u>	<u>Temp., °F</u>		<u>Comp.</u>	<u>Gen.</u>	<u>R.T.</u>	<u>ET</u>	<u>LT</u>
Moderate Pressure Tubes								
3-7(1)	1	1832	235	8510	4510	13020	1022	1562
(1)	1	1832	245	8171	4327	12498	1231	1562
3-8(2)	1	1832	224	8910	4374	13284	1269	1206/1206
Moderate and High-Pressure Tubes								
3-9(1)	1	1832	218	9161	4358	13519	1022	1382
(1)	1	1832	225	8873	4321	13194	1366	1366
(2)	1	1832	213	9398	4373	13771	1267	1267

(1) Counterflow heat exchanger in freeboard

(2) Parallel-flow heat exchanger in freeboard

Moderate-and High-Pressure Tube AFBC/CAES Systems

In these system configurations, the storage air was allowed to pass through immersed tubes in the bed on its way to the expansion turbine (Fig. 3-9). Three sets of operating conditions were simulated. The first two simulations provided for counterflow heat exchangers in the convective zone above the freeboard, the last, for a parallel-flow heat exchanger. In the first of the counterflow simulations, the expansion turbine air bypassed the convective heat exchangers. Consequently, the expansion turbine inlet temperature reflected the immersed tube air exit temperature of 1022 °F (550 °C). In the second of the counterflow simulations, the expansion turbine air did pass through the heat exchanger where it picked up thermal energy. Because this additional heat transfer would be done at the expense of the low-pressure turbine inlet temperature, the coal-feed rate was increased to maintain the low-pressure turbine inlet temperature at about the same level (resulting temperature differential was 16 °F (9°C)). For the parallel-flow simulation, the expansion turbine air also passed through the heat exchanger.

The counterflow simulations produced the better performance results for this system configuration because of the higher turbine exit conditions. However, the heat exchanger metal temperature will also be higher. As a result, the counterflow systems have a greater element of risk associated with them. Of the two counterflow system configurations, the one in which the heat exchanger was utilized produced the better performance results.

PRELIMINARY SCREENING ANALYSIS

The screening analysis involved an evaluation of the performance, cost, and relative risk associated with each of the system configurations investigated. A summary of the parametric performance results was furnished in the previous section. This section will present the results of the preliminary cost analysis and of the risk assessment. In addition, it will also provide the preliminary screening analysis wherein the systems were placed in generic groups and ranked accordingly. This narrowed the number of potential system configurations down to just two -- the open-bed PFBC/CAES (Fig. 3-2) and the immersed-tube PFBC/CAES system configuration (Fig. 3-5).

Cost Characteristics

Cost characterizations were provided for each of the major subsystems of the FBC/CAES system: the fluid bed combustor, the heat transfer equipment, and the CAES aboveground and below-ground equipment. To these direct costs were added approximations of the indirect costs and the owning and operating costs. The cost values used in this analysis are preliminary in nature but are internally consistent and suitable for comparison purposes within the guidelines of the current effort. Accordingly, the results are presented in relative terms. During the preconceptual power plant design effort, more detailed cost estimates were developed based on EPRI guidelines (Ref. 3-5) so that consistent comparisons can be made with independent cost results from other studies. Some of the key power plant design parameters used during the screening analysis were altered for the subsequent preconceptual design work.)

Fluid Bed Combustor

Both PFBC and AFBC costs expressed in 1978 dollars were obtained from the work accomplished during the technology assessment effort. Interpolation and extrapolation of these cost values were performed to provide cost estimates when the FBC characteristics did not exactly coincide with the characteristics previously developed. Although the thermal power capacity of the FBC varied from system to system, no economies of scale were assumed; the FBC cost was simply proportional to output.

Heat Transfer Equipment

For heat-exchanger cost, a simple correlation relating cost per unit of thermal power to metal temperature was used. Again no economies of scale were assumed. The cost of the intercoolers and the aftercooler were based upon a previous study (Ref. 3-1) wherein the costs were expressed in 1976 dollars.

CAES Power Plant

The costs of both the aboveground and below-ground equipment were obtained from the Ref. 3-1 study; these costs were also expressed in 1976 dollars. A weekly cycle requiring a twenty-hour storage capacity was selected to size the air storage reservoir. A power plant installation with two units was selected. This would provide between 400 MW and 550 MW of output power depending on the system configuration and operating conditions.

Indirect Costs

The year 1980 was selected as both the first year of plant operation and as the base year for costing purposes with the recognition that all of the advanced concepts described herein could not be developed until sometime after 1980. A four-year construction period, based on the estimate of Ref. 3-1, was selected, consequently, all capital costs not expressed in 1976 dollars were converted to that year by using an escalation rate of 7.5 percent. The capital cost of the FBC, heat transfer equipment, and CAES power plant were then totaled and increased by 25 percent to account for the support facilities. Escalation and interest during construction were assumed to be 7.5 percent per annum for the construction period. The appropriate factors reflecting the escalation and interest rates were used to estimate the capital cost of the power plant in the year of operation.

Owning and Operating Costs

An annual fixed charge rate of 20 percent was selected. This was combined with an annual capacity factor of 25 percent (2190 hours per year) to obtain the capital cost contribution to electrical energy cost. A coal cost of \$1.75/10⁶ Btu (\$1.66/GJ) was combined with the generation heat rate to estimate the fuel cost contribution. A value of 16 mills/kWh was chosen for the off-peak electrical energy cost. This was combined with the compression heat rate to estimate the electricity cost contribution.

Maintenance costs were estimated for the FBC, the heat transfer equipment, and the CAES power plant. For the FBC, a value equal to 4 percent of the capital cost was chosen for the annual maintenance cost. For the heat transfer equipment, a value of 2 percent was selected. The CAES power plant maintenance cost was expressed as a variable cost of 6 mills/kWh. All three of these values were treated appropriately and combined with the capital cost, fuel cost, and electricity cost to obtain a busbar electrical energy cost.

Cost Comparison

The costs developed for the screening analysis are summarized in Table 3-3. All estimated costs are given relative to the simple immersed-tube PFBC/CAES

TABLE 3-3
FBC/CAES SYSTEM RELATIVE COSTS

Fig.	Description	Relative Capital Cost			Total	Relative Electrical Energy Cost				Total
		FBC System	Heat Exchangers	CAES System		Capital	Fuel	Electricity	Maint.	
3-2	PFBC - Open 10a - 639/1562 F	0.576	N/A	0.684	1.260	0.858	0.088	0.170	0.110	1.226
	PFBC - Open 15a - 511/1562 F	0.442	N/A	0.664	1.106	0.753	0.088	0.166	0.102	1.109
	PFBC - Open 15a - 573/1652 F	0.454	N/A	0.635	1.089	0.742	0.088	0.159	0.092	1.081
	PFBC - Open 20a - 494/1652 F	0.396	N/A	0.623	1.019	0.693	0.088	0.156	0.086	1.023
3-5	PFBC - Tubes 10a - 1022/1562	0.507	N/A	0.620	1.127	0.766	0.085	0.154	0.101	1.106
	PFBC - Tubes 10a - 1022/1562	0.398	N/A	0.602	1.000 (base)	0.681	0.085	0.149	0.085	1.000 (base)
	PFBC - Tubes 20a - 1022/1562	0.348	N/A	0.594	0.942	0.643	0.085	0.149	0.080	0.957
	PFBC - Tubes 15a - 1022/1427 F	0.411	N/A	0.628	1.039	0.707	0.085	0.160	0.090	1.042
	PFBC - Tubes 15a - 1022/1652 F	0.342	N/A	0.586	0.978	0.665	0.085	0.149	0.085	0.984
3-6	PFBC - Tubes - HX 15a - 1382/1203 F	0.410	0.066	0.630	1.106	0.754	0.085	0.160	0.096	1.095
3-7	AFBC - Tubes - CFHX 1022/1562 F	0.459	0.098	0.663	1.220	0.831	0.091	0.159	0.101	1.182
	AFBC - Tubes - CFHX 1231/1562 F	0.475	0.094	0.648	1.217	0.829	0.088	0.152	0.102	1.171
3-8	AFBC - Tubes - PFHX 11269/1209/1206 F	0.492	0.114	0.658	1.264	0.861	0.089	0.166	0.106	1.222
3-9	AFBC - Tubes - CFHX 1022/1382 F	0.408	0.109	0.692	1.209	0.825	0.090	0.170	0.090	1.175
	AFBC - Tubes - CFHX 1366/1366 F	0.495	0.159	0.691	1.345	0.915	0.090	0.165	0.111	1.281
	AFBC - Tubes - PFHX 1267/1267 F	0.514	0.153	0.719	0.136	0.941	0.090	0.176	0.112	1.319

system configuration operating at 15 atm (bar) with a 1022 °F (550 °C) expansion turbine inlet temperature and a 1562 °F (850 °C) low-pressure turbine inlet temperature. No cost estimates have been made for the open-bed system configuration with heat exchanger (Fig. 3-3) because of the poor performance relative to all other configurations. No cost estimate is provided for the VHP PFBC/CAES system configurations because no cost information presently exists for the VHP PFBC. Lastly, no cost estimate is provided for the simple immersed-tube PFBC/CAES system operating at ± 20 percent of air storage pressure, because this would not contribute to the screening analysis.

It can be noted from Table 3-3 that the various PFBC/CAES systems have electrical energy costs ranging from -4 percent to +23 percent of the base cost, whereas the various AFBC/CAES system energy costs range from +17 to +32 percent. Within either the PFBC/CAES or AFBC/CAES system configuration, the cost usually decreases with the incorporation of advanced technology because of improved performance.

Risk Assessment

A major concern when investigating a novel power system is a proper evaluation of the risks associated with its commercialization. Not only is there a technological risk in that the power system may not perform up to expectation, but there is also economic risk in that the developer may not recoup the investment required. Despite the vital nature of this evaluation, no methodology has yet been proposed which unerringly predicts the outcome of a business venture, nor will one ever be proposed as long as the future remains unpredictable. Nonetheless, risk estimates must be attempted.

The following section documents the work performed in assessing the relative risk inherent in the FBC/CAES power plant system. The basis for the evaluation is the information presented on the FBC and CAES turbomachinery in the preceding chapter.

Risk Areas for FBC/CAES Systems

In the preceding chapter, there is a listing of areas in which R&D on fluid bed combustion is required. These include: coal and sorbent feeding, coal distribution, combustor design, hot gas cleanup, solid waste disposal, tube bundle design, load following and control, and regeneration of sorbent. The existence of unknowns in these areas poses a potential risk to the commercialization of the FBC/CAES power plant system concept. Obviously, unqualified success in some areas is more vital to the concept than in other areas. For the open-bed PFBC/CAES system, hot gas cleanup is important. For the FBC/CAES systems with immersed tubes, tube bundle design and load following are important. Consequently, in assessing these systems greater weight was subjectively assigned to those areas.

Also contained in the preceding chapter is a listing of areas in which R&D on CAES turbomachinery is required. This list includes: hot gas cleanup, particulate erosion control, corrosion tolerance, combined erosion/corrosion control, desposition avoidance, combined deposition/erosion/corrosion tolerance, damage control techniques, and alkali metal control. All of these areas relate to operation with a PFBC. Consequently, operation with an AFBC is considered to have little risk relative to the turbomachinery.

Many of the concerns expressed about the turbomachinery when operating with a PFBC are shared with the heat transfer equipment located in or above the bed. These concerns are present for both the AFBC and PFBC systems.

It should be noted that most of these areas are being addressed as part of an extensive DOE combined-cycle development effort (Ref. 3-6). However, the rapid cycling requirement for CAES systems in peaking applications is significantly different from the intermediate- and base-load requirements of the combined-cycle program. Accordingly, some of the load-following requirements of FBC/CAES systems require special attention.

Risk Evaluation

The approach used in this study to evaluate risk was to quantify the subjective ratings of the risk associated with each system with respect to the three developmental areas mentioned above -- the FBC system, the heat exchangers and the CAES system. A numerical ranking between 0 and 10, with 10 representing high risk, was assigned to the risk associated with equipment development in each of three development areas. A description of the ranking guidelines is provided in Table 3-4. The result of the numerical ranking is presented in Table 3-5. Reviewing this table, it can be seen that the AFBC/CAES system with low-temperature tubes and the open-bed PFBC/CAES systems were assessed as having the lowest risk, while the VHP PFBC and high-temperature systems have the highest risk.

FBC/CAES System Screening

The parametric performance analysis, cost characteristics and risk assessment provided the tools necessary to identify promising FBC/CAES system configurations for the preconceptual power plant design. This identification process will be explained in the following sections.

FBC/CAES System Configuration Generic Groupings

All the FBC/CAES systems which were analyzed were placed into five generic groups. Included within these groups were: 1) PFBC/CAES systems with an open-bed

TABLE 3-4

RISK ASSESSMENT GUIDELINES

<u>Numerical Rank</u>	<u>Equipment Risk</u>
0	No risk - Equipment available
2	Slight risk - Equipment can be designed and built on commercial basis
4-6	Moderate risk - Design verification and testing needed
8-10	High risk - Design unknowns, R&D required

TABLE 3-5

RISK ASSESSMENT
(High numbers imply high risk)

<u>Fig.</u>		<u>FBC System</u>	<u>Heat Exchangers</u>	<u>CAES System</u>	<u>Overall</u>
3-2	PFBC-Open-10a-639/1562F	4	-	6	10
	PFBC-Open-15a-511/1562F	4	-	6	10
	PFBC-Open-15a-573/1652F	5	-	6	11
	PFBC-Open-20a-494/1652F	5	-	6	11
3-3	PFBC-Open-HX-15a-674/1200F	4	2	7	13
3-4	PFBC-VHP 58/15a-1562/1562F	10	-	6	16
	PFBC-VHP 58/20a-1652/1652F	10	-	7	17
3-5	PFBC-tubes-10a-1022/1562F	7	-	5	12
	PFBC-tubes-15a-1022/1562F	8	-	5	13
	PFBC-tubes-20a-1022/1562F	9	-	5	14
	PFBC-tubes-15a-1022/1427F	8	-	4	12
	PFBC-tubes-15a-1022/1652F	10	-	6	16
	PFBC-tubes-15a-1022/1562F	8	-	5	13
	+ 20% storage pressure				
	PFBC-tubes-15a-1022/1562F	8	-	5	13
	-20% storage pressure				
3-6	PFBC-tubes-HX-15a-1382/1203F	8	4	4	16
3-7	AFBC-tubes-CFHX-1022/1562F	5	6	2	13
	AFBC-tubes-CFHX-1231/1562F	5	6	2	13
3-8	AFBC-tubes-PFHX-1269/1209/1206F	5	3	3	11
3-9	AFBC-tubes-CFHX-1022/1382F	5	3	1	9
	AFBC-tubes-CFHX-1366/1366F	5	4	1	10
	AFBC-tubes-PFHX-1267/1267F	5	3	1	9

PFBC (Fig. 3-2 and 3-3); 2) PFBC/CAES systems with an open-bed VHP PFBC (Fig. 3-4); 3) PFBC/CAES systems with low-temperature tubes (Fig. 3-5); 4) PFBC/CAES and AFBC/CAES system with high-temperature tubes (Figs. 3-5 through 3-9); and 5) AFBC/CAES systems with low-temperature tubes (Figs. 3-5 through 3-9). For each of these systems, the heat rate, electrical energy cost, and risk were described on a relative basis as high, moderate, or low.

FB/CAES Candidate System Configurations

The results of the assessment are provided in Table 3-6. The open-bed PFBC/CAES system was chosen as a possible candidate system for the preconceptual design. This selection was based mainly on the relatively low risk for the PFBC design (no tubes in bed), the moderate cost, and the respectable round trip heat rates which these systems exhibited.

The open-bed VHP PFBC/CAES systems look attractive due to the superior performance that was exhibited by these systems. This concept may be a future candidate once better design and costing data become available. PFBC/CAES systems with low-temperature air tubes would also be viable candidates for the preconceptual design. Their cost and performance are better than the open-bed PFBC/CAES systems; however, because they have tubes in the bed, the risk is increased.

Both AFBC/CAES and PFBC/CAES systems with high-temperature air tubes were judged questionable due to the high risk of tube corrosion and questionable cost even though their performance would be attractive.

The AFBC/CAES systems with low-temperature air tubes were also judged to be questionable. These systems would have low risk resulting from the lack of turbine corrosion/erosion problems, but they also have relatively high cost and poor performance.

TABLE 3-6

FBC/CAES SYSTEM SCREENING RESULTS

<u>Generic System</u>	<u>Risk</u>	<u>Cost</u>	<u>Heat Rate</u>	<u>Prognosis</u>
PFBC/CAES - open bed	L	M-L	H-M	Possible candidate
PFBC/CAES - open bed; VHP PFBC	H	?	L	Future candidate
PFBC/CAES - low-temperature tubes	M	L	L	Possible candidate
AFBC/CAES } PFBC/CAES } - high-temperature tubes	H	?	L	Questionable
AFBC/CAES - low-temperature tubes	L	H	H	Questionable

H = High
M = Moderate
L = Low

CANDIDATE CONFIGURATION IDENTIFICATION

Having identified two potentially viable system configurations, it became necessary to select the candidate configuration and the operating conditions for that candidate configuration.

FBC/CAES System Comparison

To provide a comparison, the performance, cost, and risk characteristics for each of the two candidate configurations were examined. The relevant data are summarized in Table 3-7.

Performance, Cost, and Risk Screening

In reviewing this table, it is possible to eliminate at least several operating conditions - the open bed PFBC/CAES system at 10 atm (bar) and 15 atm (bar)/1652 °F (900 °C) because better performance and cost characteristics exist for other operating conditions for the same risk. It is also possible to eliminate the 10 atm (bar) immersed-tube PFBC/CAES system for the same reason. After doing this though, it was not really possible to select a preferred system from among the remaining systems because the attractiveness of all the remaining systems based on the combined parameters of performance, cost and risk were all about equal.

Commercialization

To provide a means for differentiating between the configurations and their operating conditions, the criterion of commercialization date was introduced. For the purpose of this study, this term is defined as the earliest date at which both the buyer (electric utility) and the seller (equipment manufacturer) would agree to sign a contract based on standard commercial terms and conditions. Typically, the first date of commercial operation would follow the contract signing by three to ten years, depending on the time required to obtain the necessary permits and construct the plant.

Estimates of the commercialization date are provided in Table 3-8 for the FBC/CAES generic systems considered. These estimates are based on the development of FBC technology which was determined to be the pacing item. These projections, which were generated in this study, are consistent with recent DOE Fossil Energy Division projections (Ref. 3-6)

System Selection

Based on the data in Tables 3-7 and 3-8, the open-bed PFBC/CAES system configuration was selected as the preferred system configuration for the pre-conceptual design because it has the potential for earlier commercialization.

TABLE 3-7

CANDIDATE FBC/CAES SYSTEM COMPARISON

<u>System</u>	<u>Round Trip Heat Rate, Btu/kWh</u>	<u>Relative Costs</u>		<u>Risk</u>
		<u>\$/kW</u>	<u>mills/kWh</u>	
PFBC-Open-10a-639/1562F	13428	1.25	1.25	10
PFBC-Open-15a-511/1562F	13248	1.10	1.10	10
PFBC-Open-15a-573/1652F	12816	1.10	1.10	11
PFBC-Open-20a-494/1652F	12715	1.00	1.00	11
PFBC-Tubes-10a-1022/1562F	12480	1.15	1.10	12
PFBC-Tubes-15a-1022/1562F	12302	1.00	1.00	13
PFBC-Tubes-20a-1022/1562F	12267	0.95	0.95	14
PFBC-Tubes-15a-1022/1427F	12728	1.05	1.05	12

TABLE 3-8

COMMERCIALIZATION DATE PROJECTIONS

<u>Generic System</u>	<u>Tube Metal Temperature, ° F</u>	<u>Date</u>
PFBC/CAES - open bed	-	~ 1985+
PFBC/CAES - open bed, VHP PFBC	-	~ 1995+
PFBC/CAES - low-temperature tubes	1200	~ 1990-
AFBC/CAES } - high-temperature tubes PFBC/CAES }	1600	~ 1995+
AFBC/CAES - low-temperature tubes	1200	~ 1985+

Of the two surviving open-bed PFBC/CAES systems, the system with the 15-atm (bar) bed pressure and 1562 °F (900 °C) bed temperature was chosen because it possessed less risk and should offer earlier commercialization than the system with the higher pressure (20 atm (bar)), higher temperature (1652 °F (900 °C)) bed conditions.

Description of Selected System

A schematic of the selected open-bed PFBC/CAES system is presented in Fig. 3-10. The station numbers coincide with the station numbers for the heat and mass balance data provided in Table 3-9.

The final selection of this system configuration and its operating conditions was based on early commercialization. As a result, some performance and cost penalties were incurred compared to potentially more advanced systems. However, the selection which was made did not negate the possibility of benefiting from these advances at some future date. The growth potential of the recommended open-bed system is graphically displayed in Fig. 3-11. Two future development paths are possible. Evolution of this concept could occur by first increasing bed temperature and then increasing pressure. This could be followed by introduction of the very high pressure PFBC with its attractive performance characteristics. Evolution could also occur with immersed tubes; first by adding low-temperature tubes, followed by high-temperature tubes as material technology advances.

SELECTED COAL-FIRED PFBC/CAES SYSTEM

SYSTEM ANALYSIS DESIGN

FIG. 3-10

TABLE 3-9

HEAT AND MASS BALANCE FOR SELECTED COAL-FIRED FBC/CAES SYSTEM

Note - Stations 1 through Cavern for Compression Mode
Stations 10 through 17 for Generation Mode

PFBC Pressure = 15 Atmospheres
LT Turbine Inlet Temperature = 1562 F

Station No. ⁽¹⁾	1	2	3	4	5	6	7	8	9	Cavern	10	11
W, lb/sec	761.8	761.8	761.8	761.8	761.8	761.8	761.8	761.8	761.8	761.8	708.6 ⁽²⁾	708.6
T, F	58.7	345.0	160.9	500.0	200.0	377.6	200.0	409.3	120.0	120.0	120.0	511.2
P, psia	14.7	60.2	55.8	230.4	225.8	440.3	430.1	925.2	906.7	973.7	902.9	859.0
H, Btu/lb	124.0	193.0	148.5	231.1	157.9	201.0	157.9	208.8	138.7	138.7	138.7	233.8
Station No. ⁽¹⁾	12	13	14	15	16							
W, lb/sec	708.6	22.4	728.8	745.7	745.7							
T, F	255.5	--	1562.0	660.7	300.0							
P, psia	220.5	--	216.0	15.2	14.9							
H, Btu/lb	171.3	--	510.8	272.4	181.9							

Compressor Power, MW		Turbine Power, MW		Gross Heat Rate, Btu/kWh		Heat Exchanger Heat Transfer, Btu/sec	
				HHV			
LC	56.2	ET	45.5	Generation	4322	IC1	33929
HC	67.2	LT	178.4	Compression	8926	IC2	55730
BC	76.5	Total, gross	223.9	Round trip	13248	IC3	32819
Total	199.9					Aftercooler	53387
						Recuperator	67441

(1) Station numbers refer to locations identified in Fig. 3-10 (all at surface except for cavern).

(2) Plus 22.7 lb/sec of cooling air for total of 731.3 lb/sec withdrawal rate.

This is 4 percent less than compressor flow rate to adjust for cavern leakage.

GROWTH POTENTIAL OF RECOMMENDED PFBC/CAES SYSTEM

FIG. 3-11

REFERENCES

- 3-1 Giramonti, A. J. et al.: Preliminary Feasibility Evaluation of Compressed Air Storage Power Systems. United Technologies Research Center Report R76-952161-5, Supported by National Science Foundation Grant AER74-00242, Technical Oversight by Energy Research and Development Administration, December 1976.
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Chapter 4

PFBC/CAES Power Plant Preconceptual Design

PFBC/CAES Power Plant Preconceptual Design

The goal of the work performed during this phase of the study program was to transform the open-bed PFBC/CAES system configuration selected from the system analysis into the preconceptual design of a PFBC/CAES power plant. The design effort was conducted in three phases.

The objective of the initial phase was to establish the basic design approach to be used. This approach mandated liberal utilization of existing designs for major power plant equipment items developed during previous investigations. Consequently, it was necessary to conduct a review of past study efforts to identify commonality between past equipment design efforts and the requirements of equipment for the selected configuration. The adoption of existing designs for an item of major equipment combined with a critical examination of the system lead to minor changes in operating characteristics from those previously identified for the selected configuration. The impact of these changes on the system was estimated and the resulting characteristics of the power plant were defined in sufficient detail to serve as the basis for the preconceptual design effort.

The next phase was to implement the design approach and prepare the preconceptual design. To produce the design, each of the three participating organizations concentrated upon that portion of the power plant for which its background and experience made it most suited to prepare the design (i.e., UTRC-CAES turbomachinery, CPC-CAES fluid bed combustor, and AAI-balance of plant and underground facility). The preconceptual design effort included itemization of the major equipment items along with descriptions of their operation and design features. Also included were layout sketches displaying major equipment location and an assessment of institutional factors which could affect a PFBC/CAES power plant.

The last phase involved estimating the costs of the major equipment items and the use of these costs to estimate busbar cost of electricity. Although beyond the scope of this study program, the costing phase was extended slightly to include a simple comparison of the PFBC/CAES power plant system with alternative peaking systems.

These three phases - power plant definition, preconceptual design, and cost estimating - are the subjects of this chapter.

POWER PLANT DEFINITION

Once the open-bed PFBC/CAES system configuration was selected and approved by PNL and DOE for the preconceptual design effort, the selected system configuration was carefully reviewed. This examination was necessary to incorporate as much of the applicable previously existing design data available to the participating organizations into the preconceptual design and to ensure that all relevant equipment operating behavior had been identified. The incorporation of updated data and operating behavior required minor modification of the system configuration and its heat and mass balances relative to those produced in the system analysis effort. This task, performed in preparation for the preconceptual design, is discussed in this section.

Design Approach

The scope of this study program mandated the implementation of a design approach that made maximum utilization of existing designs which had been developed in previous PFBC and CAES power plant studies (e.g., Refs. 4-1 through 4-4). Acceptance of this approach required a detailed review of the operating conditions of major equipment items in the PFBC/CAES system configuration selected during the systems analysis.

The purpose of this review was to ascertain whether these major equipment items had features common to those of existing designs. Where commonality did exist, it was possible to directly employ the existing design in the PFBC/CAES power plant preconceptual design effort. Substantial commonality was found to occur with respect to some of the turbomachinery components and balance-of-plant items.

Obviously, because the PFBC/CAES concept is unique, direct substitution was not always possible. When the incompatibility was of a minor nature, modification was made to both the PFBC/CAES requirements and the existing design characteristics to provide a suitable compromise. Examples of major equipment item designs which were modified are the heat transfer equipment and piping.

If neither direct substitution nor modification of major equipment items were possible, then an original design was developed. The prime example of this is the open-bed, pressurized fluid bed combustor for CAES applications. Although the PFBC preconceptual design is quite unique, the limited scope of this activity required that this effort reflect an extension of existing design work for combined cycle applications. Accordingly, a careful design optimization for CAES applications was not possible.

Throughout the design approach implementation, whenever a modification or an original major equipment item design was required, the resulting performance characteristics of the design were held as close as possible to the characteristics produced in the system analysis. This guideline was essential so as not to compromise the intent of the system analysis selection process. Because most modifications produced cascading changes in the operating conditions of the other components, these operating conditions were incorporated in a revised steady-state simulation of the PFBC/CAES system. These performance revisions which are incorporated in the preconceptual design are described in the next section.

Preconceptual Design Characteristics

Concurrently with the implementation of the design approach, the system configuration was examined more critically than was possible during the system analysis effort. As a result of this examination, equipment operational behavior not considered in the systems analysis was identified. This behavior was also incorporated in the simulation of the PFBC/CAES system to produce a redefinition of the power plant characteristics in sufficient detail to serve as the basis for the preconceptual design.

Design Heat and Mass Balance

The implementation of the design approach, along with a critical examination of the system configuration yielded modification of the data which was used to develop Table 3-9; these are presented in Table 4-1.

Pressure drops were accounted for in the following piping: the surface pipe from the aftercooler to the shaft head, the surface pipe from the shaft head to the recuperator, the pipe from the expansion turbine (ET) exhaust to the pressure vessel inlet, and the pipe from the pressure vessel exhaust to the low-pressure turbine (LT). A control valve was also inserted between the recuperator and ET to provide off-design control.

Other revisions were also incorporated. The pressure drops required in the compressor intercoolers and aftercooler were recalculated and revised downward. The pressure differential in the shaft was modified to more closely agree with the pressure-depth relationships developed in Ref. 4-1. The turbine cooling airflow was withdrawn from the ET exit instead of from storage, thus making that airflow available to increase the power output from the ET. The Ca/S ratio was changed from 1.2:1 to 1.5:1. Lastly, the storage pressure was revised upward to cover the increased pressure losses in the system. A schematic diagram of the revised system is presented in Fig. 4-1.

The heat and mass balances resulting from these revisions are presented in Table 4-2. The increased airflow through the ET has increased the ET power output by 2.5 MW from 45.5 MW (Table 3-9) to 48.0 MW. The pressure drop through the PFBC and cleanup equipment has increased, thus providing a lower pressure into the LT. Because the back pressure of the LT is fixed at a value

Table 4-1

REVISIONS TO PFBC/CAES SIMULATION

<u>Component</u>	<u>Pressure Losses, ΔP, psi</u>	
	<u>Base</u> ⁽¹⁾	<u>Revised</u>
IC1	4.4	1.2
IC2	4.6	2.9
IC3	10.2	5.0
AFCL	18.5	10.0
Surface pipe from AFCL	0	11.75
From surface to storage	-67.0	-67.6
From storage to surface	70.8	97.6
Surface pipe to recuperator	0	10.0
Recuperator (cold side)	43.9	20.0
Control valve	0	30.0
From ET exit to pressure vessel	0	5.0
Pressure vessel (PFBC and three stages of cyclones)	4.5	19.1
From pressure vessel to LT	0	2.0
<u>Miscellaneous</u>		
LT cooling air extraction	From storage	From ET exit
Ca/S Ratio	1.2:1	1.5:1
Storage pressure, psia	973.7	1045.6

⁽¹⁾Used to develop Table 3-9.

COAL-FIRED PFBC/CAES SYSTEM

PRECONCEPTUAL DESIGN BASIS

FIG. 4-1

Table 4-2

HEAT AND MASS BALANCE FOR COAL-FIRED PFBC/CAES CONFIGURATION
Preconceptual Design Basis
Per Unit

Note - Stations 1 through Cavern for Compression Mode
Stations 12 through 22 for Generation Mode

PFB Pressure = 15 Atmospheres
LT Turbine Inlet Temperature = 1562 F

Station No. ⁽¹⁾	1	2	3	4	5	6	7	8	9	10	11	Cavern
W, lb/sec	761.8	761.8	761.8	761.8	761.8	761.8	761.8	761.8	761.8	761.8	761.8	761.8
T, F	58.7	58.7	345.	160.9	500.0	200.	377.6	200.	409.3	120.	120.	120.
P, psia	14.7	14.6	60.2	59.0	243.8	240.9	469.8	464.8	999.8	989.8	978.1	1045.7
H, Btu/lb	124.0	124.0	193.0	148.5	231.1	157.9	201.0	157.9	208.8	138.7	138.7	138.7
Station No. ⁽¹⁾	12	13	14	15	16	17	18	19	20	21	22	
W, lb/sec	731.3 ⁽²⁾	731.3	731.3	731.3	709.5	709.5	730.4	730.4	746.6	746.6	22.88	
T, F	120.	120.	524.6	524.6	263.9	263.9	1562.	1562.	684.1	300.	--	
P, psia	948.1	938.1	918.1	888.1	225.4	220.4	201.3	199.3	15.2	14.9	--	
H, Btu/lb	138.7	138.7	237.2	237.2	173.3	173.3	510.8	510.8	278.4	181.8	--	
Compressor Power, MW		Turbine Power, MW		Gross Heat Rate, Btu/kWh			Heat Exchanger Heat Transfer, Btu/sec					
LC	56.2	ET	48.0	Generation	HHV							
HC	67.2	LT	174.8	Compression	4437		IC1		33929			
BC	76.5	Total, gross	222.8	Round trip	8976		IC2		55730			
Total	199.9				13413		IC3		32819			
							Aftercooler		53387			
							Recuperator		72025			

(1) Station numbers refer to locations identified in Fig. 4-1 (all at surface except for cavern).

(2) This is 4 percent less than compressor flow rate to adjust for cavern leakage.

large enough to allow the flow to escape up the stack, the expansion ratio and, thereby, the power output of the LT is reduced. The LT power output has been reduced 3.6 MW from 178.4 MW to 174.8 MW. As a result of these changes, the overall power output of the plant is reduced by 1.1 MW, and the round trip heat rate is increased by 1.2 percent.

The values presented in Table 4-2 were used as the basis for the preconceptual design effort.

Ensuing Modifications

During the preconceptual design effort, further minor modifications of the system characteristics were undertaken. An estimate of the auxiliary power requirements for the PFBC was developed. This amounted to 2.0 MW per CAES unit. The effect of these requirements is to reduce the net power from the CAES plant and also to increase the heat rate.

Bleed air was provided for use in drying the coal and dolomite. This air was extracted after the LT and before the recuperator. Bleeding part of the flow before entering the hot side of the recuperator reduces the amount of energy to be recovered in the recuperator by the air flow withdrawn from the cavern. As a result, there is a temperature reduction in the flow prior to entering the ET which, in turn, reduces the power output of the ET (the LT output is not changed).

The effects of these additional modifications are presented in Table 4-3. A schematic drawing of the system with the bleed air indicated is depicted in Fig. 4-2. The air bleed reduces the ET turbine output by 0.9 MW from 48.0 MW (Table 4-2) to 47.1 MW. The PFBC auxiliary power demand reduces net power output by an additional 2.0 MW. The total net power output of the PFBC/CAES power plant is, therefore, 219.1 MW and the round trip heat rate is 13633 Btu/kWh.

The differences that result in the equipment design from using the heat and mass balance of Table 4-2 instead of the data in Table 4-3 are insignificant. Consequently, it was deemed not necessary to iterate on the preconceptual design effort to incorporate the Table 4-3 values. However, the data in Table 4-3 were used for the economic analysis.

Table 4-3

HEAT AND MASS BALANCE FOR COAL-FIRED PFBC/CAES CONFIGURATION
Modified Preconceptual Design
Per Unit

Note - Stations 1 through Cavern for Compression Mode
Stations 12 through 23 for Generation Mode

PFB Pressure = 15 Atmospheres
LT Turbine Inlet Temperature = 1562 F

Station No. ⁽¹⁾	1	2	3	4	5	6	7	8	9	10	11	Cavern
W, lb/sec	761.8	761.8	761.8	761.8	761.8	761.8	761.8	761.8	761.8	761.8	761.8	761.8
T, F	58.7	58.7	345.	160.9	500.0	200.	377.6	200.	409.3	120.	120.	120.
P, psia	14.7	14.6	60.2	59.0	243.8	240.9	469.8	464.8	999.8	989.8	978.1	1045.7
H, Btu/lb	124.0	124.0	193.0	148.5	231.1	157.9	201.0	157.9	208.8	138.7	138.7	138.7
Station No. ⁽¹⁾	12	13	14	15	16	17	18	19	20	21	22	23
W, lb/sec	731.3 ⁽²⁾	731.3	731.3	731.3	709.5	709.5	730.4	730.4	746.8	713.5	23.10	33.3
T, F	120.	120.	507.5	507.5	250.7	250.7	1562.	1562.	684.1	300.	--	684.1
P, psia	948.1	938.1	918.1	888.1	225.4	220.4	201.3	199.3	15.2	14.9	--	15.2
H, Btu/lb	138.7	138.7	232.8	232.8	170.1	170.1	510.8	510.8	278.4	181.8	--	248.4
<u>Compressor Power, MW</u>		<u>Turbine Power, MW</u>		<u>Gross Heat Rate, Btu/kWh</u>			<u>Heat Exchanger Heat Transfer, Btu/sec</u>					
LC	56.2	ET	47.1	Generation	<u>HHV</u> 4538			IC1	33929			
HC	67.2	LT	<u>174.8</u>	Compression	<u>9095</u>			IC2	55730			
BC	<u>76.5</u>	Total, gross	221.9	Round trip	13633			IC3	32819			
Total	199.9	Aux.	<u>- 2.0</u>					Aftercooler	53387			
		Total Net	<u>219.1</u>					Recuperator	68815			

(1) Station numbers refer to locations identified in Fig. 4-2 (all at surface except for cavern).

(2) This is 4 percent less than compressor flow rate to adjust for cavern leakage.

4-9

79-11-2-1

CAES FLUID BED COMBUSTOR PRECONCEPTUAL DESIGN

The objective of the fluid bed combustor preconceptual design effort was to produce a design description of a fluid bed combustor subsystem which could be utilized in CAES applications. To achieve this objective, it was necessary to first synthesize a fluid bed combustor plant which would satisfy the design basis identified in the preceding section. This section presents the results of that effort and includes a description of the fluid bed combustor, an identification of the main equipment items, and an illustration of the arrangement of these equipment items in several layout sketches.

Scope of PFBC Design Effort

The limited magnitude of the present study precluded the synthesizing of alternative designs or the utilization of optional methods of construction which could possibly optimize the combustor concept for CAES applications. Instead, existing PFBC design expertise for conventional combined-cycle power generation systems was applied with modifications incorporated when the CAES requirements differed from those of conventional systems.

The battery limits for the PFBC preconceptual design are defined as follows:

- (1) the base-plate of the major equipment;
- (2) the receiving bunker for coal and dolomite below the rail cars;
- (3) the ash storage bunker outlet;
- (4) the inlet flanges of the air duct on the combustor;
- (5) the outlet flanges of the hot gas duct on the combustor; and
- (6) the discharge from the fuel oil storage tank.

The following items are excluded:

- (1) the central control system, because this is integrated with the balance of plant; however, provisions have been made for the valves, actuators and integral instruments associated with the combustor system; and
- (2) all civil works.

PFBC Design Basis

The following parameters were adopted as the basis of the design for the pressurized fluid bed combustor:

- (1) The flow rates, temperatures and pressures at the design point are as given in Table 4-2.
- (2) Four parallel combustors, each with a coal input of 283.6 MW(th) (HHV) and each feeding a single turbine train are included on the site with no provision made for further extensions.
- (3) Removal of sulfur was selected to be at least 85 percent of the sulfur inventory in the coal to meet the environmental emission requirement of less than 150 ppm at 15-percent excess air.
- (4) Based upon the peak-load duty cycle analysis performed in Ref. 4-2, a yearly generation time of 2190 hours was chosen. If a 50-week operating year were assumed, this would correspond to 43.8 hours per calendar week. Because the PFBC/CAES power plant will generally operate only five days per week, the number of hours of operation per day is approximately 8.8 hours.
- (5) Storage facilities are sufficient to supply the four fluid bed combustors with coal and dolomite for 60 calendar days (42.9 operating days).
- (6) A green-field, east-coast site was assumed.
- (7) Coal and dolomite were assumed to be rail delivered to the site uncrushed. Removal of ash and spent sorbent will also be by rail.

PFBC Process Description

A schematic diagram showing the main process units and flows is given in Fig. 4-3. A brief overview of the fluid bed combustion system and its ancillaries is given below; more detailed descriptions of the process units are included in the section on PFBC Equipment Specifications.

Coal and dolomite are withdrawn from stockpiles and fed to separate crushing and drying mills. A sidestream of the hot gases from the exhaust of the LT is used to dry the coal and dolomite and also serves to transport the prepared materials to cyclones which discharge into a common buffer hopper. The gases (and water vapor) leaving the cyclones are filtered before being vented to the atmosphere with the captured fine solids being returned to the buffer hopper. The pressure drops in the drying, transport, and solids disengagement systems are such that booster fans are required to increase the pressure of the turbine exhaust gases.

The mixture of crushed coal and dolomite is fed in batches to the storage injector after being screened to remove oversize material (e.g., agglomerates).

PROCESS FLOW DIAGRAM PFBC PLANT

FIG. 4-3

This oversized material could be returned to the coal crushing and drying mill for recycling. The storage injector is pressurized and depressurized cyclically. In the pressurized state, it discharges to the primary injector; in the depressurized state, it receives the screened feed from the buffer hopper. A side stream of air from the exhaust of the ET is used for pressurizing the storage injector. From the primary injector, the mixture of coal and dolomite is split equally into twenty-four streams and conveyed pneumatically to the fluidized bed using another side stream of air from the exhaust of the ET.

The main flow of air from the exhaust of the ET is used as combustion air in the fluidized bed. The coal and dolomite mix rapidly with the bed material and are heated to the bed temperature. The operating conditions are chosen so that high combustion efficiency is achieved on a single pass and retention of sulfur dioxide by the dolomite exceeds 85 percent. The combustion gases from the bed are cleansed using three stages of cyclones which remove most of the particulate matter carried out from the bed. The combustion gases are then expanded through the LT.

Coal ash and spent dolomite (calcined or sulfated) accumulate in the bed and are withdrawn to maintain a constant bed level. The surplus bed material, together with the fines collected by the cyclones, are conveyed pneumatically to a water-cooled ash cooler using still another side stream of air from the exhaust of the ET. The solids are then conveyed to disengagement cyclones followed by filters to remove the fine dust. The conveying air (which is mixed with a small amount of underflow gas from the combustor cyclones) is returned to the combustor using a booster compressor. The material collected by the disengagement cyclones and filters is removed from the system through pressure lock vessels and sent to storage ready for disposal.

Raw Materials and Utilities Requirements

To operate the fluid bed combustor, fuel, sorbent, and combustion air are required. The fuel is coal when the bed is operating at temperatures above about 1382 F (750 C). During startup, however, fuel oil would be used to heat the fluid bed up to temperature. In addition, auxiliary power and cooling water are required.

Coal

The fluid bed combustor design has been based on the use of Illinois No. 6 coal, an analysis of which is given in Table 4-4.

Table 4-4

COAL ANALYSIS

Moisture Content:	13% w/w, as received
	5% w/w, as fired

Proximate Analysis:

Volatile matter	37.30%, w/w, dry basis
Ash	10.40%, w/w, dry basis
Carbon dioxide	0.18%, w/w, dry basis
Fixed carbon	<u>52.12%</u> , w/w, dry basis
Total	100.00%

Ultimate Analysis:

Ash	10.40%, w/w, dry basis
Carbon	71.35%, w/w, dry basis
Hydrogen	4.60%, w/w, dry basis
Oxygen	8.90%, w/w, dry basis
Nitrogen	1.40%, w/w, dry basis
Sulphur	3.30%, w/w, dry basis
Chlorine	<u>0.05%</u> , w/w, dry basis
Total	100.00%

The higher heating value (dry basis) of the coal as fired is 12,600 Btu/lb (29.29 MJ/kg).

At the design point, the coal input rate (lb/sec (kg/s)) is:

	<u>Per Unit</u>	<u>Total Plant</u>
dry basis	21.8 (9.9)	87.1 (39.5)
as fired	22.9 (10.4)	91.7 (41.6)
as received	24.9 (11.3)	100.1 (45.4)

The above rates are based on a maximum continuous rating of 283.6MW(th) (HHV) net input of coal per fluid bed combustor unit (i.e. excluding energy rejected by the combustor as unburned carbon or thermal losses).

Dolomite

The dolomite specification assumed for the design is based on BCR No. 1337 dolomite and is given in Table 4-5.

The dolomite requirements for 85-percent sulphur retention using the specified coal have been estimated to correspond to a Ca/S ratio of 1.5:1. Because of the possibility of variations in coal and dolomite specifications, storage and reclamation facilities have been sized assuming a Ca/S ratio of 1.75:1. The preparation and feeding systems have been sized assuming a Ca/S ratio of 2.0:1 to allow for additional acceptor requirements during transients, such as startup.

At the design point, the dolomite feed rate (lb/sec (kg/s)) is:

	<u>Per Unit</u>	<u>Total Plant</u>
dry basis	6.4 (2.9)	25.4 (11.5)
as fired	6.4 (2.9)	25.4 (11.5)
as received	6.6 (3.0)	26.7 (12.1)

Combustion Air

For the design point conditions, the combustion air supply rate per unit (lb/sec (kg/s)) is as follows:

direct supply	681.2	(309.0)
transport air for coal and dolomite feeding	7.3	(3.3)
transport air for coal ash and spent dolomite	<u>20.1</u>	<u>(9.1)</u>
total air input	708.6	(321.4)

Table 4-5

SORBENT ANALYSIS

Moisture Content:	5.0% w/w, as received
	0.2% w/w, as fired

Analysis:

CaO	29.8%, w/w, dry basis
MgO	21.4%, w/w, dry basis
CO ₂	46.9%, w/w, dry basis
Other	<u>1.9%</u> , w/w, dry basis
Total	100.0%

In addition, 11.0 lb/sec (5.0 kg/s) of combustion gases are withdrawn with the cyclone fines and are recirculated to the bed with coal ash and spent dolomite handling system transport air.

The pressure at the combustor inlet is 15.2 atm (bar), and the temperatures (F(C)) are as follows:

direct supply	255 (124)
transport air for coal and dolomite feeding	122 (50)
transport air for coal ash and spent dolomite handling system	214 (101)

In all cases, the air is assumed to have a zero moisture content.

Auxiliary Fuel

Auxiliary fuel oil will be required during startup to heat the fluidized bed to the minimum temperature required for combustion of the coal (about 1382 F (750 C)). Various types of startup are possible, depending on the initial temperature. It is recommended that for the purpose of sizing the oil storage facilities for the total plant, allowance be made for 6.6 tons (6 t) of No. 2 fuel oil per start per unit.

Other Services

In addition to the items discussed above, the combustion system requires auxiliary power and cooling water. These are specified in Tables 4-6 and 4-7, respectively. Hot gases for drying the coal and dolomite will also be required. A side stream of the exhaust gases from the LT is used to dry the coal and dolomite. The side stream is taken before the recuperator where the gas temperature is 684 F (362 C). The total gas flow per unit taken for drying is 33.3 lb/sec (15.1 kg/s). No auxiliary firing is required. The gas composition is similar to that at the combustor exit (approximately 78-percent N₂ and 15-percent O₂).

PFBC Effluents

The effluents from the fluid bed combustor system occur at three points in the process. Coal ash and spent sorbent are removed from the fluidized bed, the combustion gases are ducted to the turbomachinery, and vent gases are released from the drying plant.

Coal Ash and Spent Sorbent

The total production rate of coal ash and spent dolomite (including both material withdrawn from the bed and fines collected by the cyclones) is 7.9

Table 4-6

PFBC AUXILIARY POWER REQUIREMENTS

<u>Process</u>	<u>Power Requirement, MW⁽¹⁾</u>
Coal mill feeder	0.015
Coal mill	0.120
Milling system vent fan	0.120
Dolomite mill feeder	0.010
Dolomite mill	0.050
Coal and dolomite transport air compressor	0.150
Coal ash and spent sorbent transport air booster compressor	0.550
Coal ash and spent sorbent transport air return compressor	0.750
Drying gas booster fan	<u>0.260</u>
Total	2.025

(1) per unit

Table 4-7

PFBC COOLING WATER REQUIREMENTS

<u>Process</u>	<u>Cooling Requirements, lb/sec (kg/s)⁽¹⁾</u>
Coal and dolomite transport air compressor suction cooler	11.9 (5.4)
Coal and dolomite transport air compressor aftercooler	9.5 (4.3)
Ash transport air booster compressor suction cooler	21.4 (9.7)
Bed ash cooler	47.0 (21.3)
Primary cyclone ash cooler	79.1 (35.9)
Secondary cyclone ash cooler	56.0 (25.4)
Tertiary cyclone ash cooler	34.2 (15.5)
Ash transport air cooler	<u>44.8 (20.3)</u>
Total	303.8 (137.8)

(1) per unit

lb/sec (3.6 kg/s) per unit at the design point. The composition of this material (mass basis) is as follows:

CaSO ₄	32.5%
MgO	17.0%
Inert mineral matter impurities present in the dolomite	1.5%
Ash and unburned carbon	30.8%
CaCO ₃	<u>18.2%</u>
Total	100.0%

During operation at other than the design point (e.g., with a different coal or during transients) it may be necessary to increase the dolomite feed rate. Provision has been made in the design of the coal and spent sorbent handling system for a maximum production rate of spent sorbent of 9.7 lb/sec (4.4 kg/s). Operation of the combustor at low pressure (e.g., during startup or at part load) may result in calcination of some of the CaCO₃ so that the discarded material contains CaO.

Combustion Gases

The flow rate per unit of the combustion gases at the design point condition is 730.0 lb/sec (331.1 kg/s) per unit. The gases have the following approximate composition (volume basis):

CO ₂	5.34%
O ₂	14.7%
N ₂	77.7%
HCl	0.001%
SO ₂	0.013%
H ₂ O	<u>2.246%</u>
Total	100.0%

The temperature and pressure of the gases at the outlet from the pressure vessel are 1562 F (850 C) and 13.9 atm (bar) respectively.

Vent Gases

The flow rate per unit of vent gases (including water vapor) from the coal and dolomite drying plant is 35.5 lb/sec (16.1 kg/s) per unit. The gases have the following approximate composition (volume basis):

CO ₂	4.687%
O ₂	13.2%
N ₂	69.5%
HCl	0.001%
SO ₂	0.012%
H ₂	<u>12.6%</u>
Total	100.0%

The temperature of the vent gases is 199 F (93 C).

PFBC Mass and Energy Balances

The mass and energy balances for a single combustor operating at the design point conditions are summarized in Tables 4-8 and 4-9, respectively. The mass balance data exclude the intermittent flows which occur during pressurization and depressurization. Estimated peak rates for these flows are 2.4 lb/sec (1.1 kg/s) for the storage injector vent air and 1.3 lb/sec (0.6 kg/s) for the ash and spent dolomite lock hopper vent air.

PFBC Plant Layout

The plot layout proposed for the PFBC plant is illustrated in Fig. 4-4. In this layout the CAES plant consists of four identical combustion systems and turbomachinery sets, fed from common stockpiles of coal and dolomite. These stockpiles are serviced by two rail sidings. The combustors are arranged in pairs on opposite sides of the milling plants. Each turbine train takes gases from the adjacent combustor. Solid effluent from each combustor is transported to the ash storage silos located at the ash handling rail siding.

PFBC Equipment Specifications

The function of the major fluid bed combustion equipment components has been briefly discussed in the section entitled PFBC Process Description and identified in Fig. 4-3. The following subsections present a more detailed discussion of the main features associated with the fluid bed combustor equipment components and a listing of these components.

Coal and Dolomite Handling

The equipment components for coal and dolomite handling are common to all four units on the site; consequently, they have been sized to service the total plant.

Table 4-8

PFBC MASS BALANCE

Single Combustor Design Point Operation

<u>Inputs</u>	<u>Rate, lb/sec (kg/s)</u>	
Air (including transport air for coal/dolomite feeding and ash handling)	708.6	(321.4)
Coal (as received)	24.9	(11.3)
Dolomite (as received)	6.6	(3.0)
Turbine exit gas for coal and dolomite drying	<u>33.3</u>	(15.1)
Total	773.4	(350.8)
<u>Outputs</u>		
Combustion gases to turbine	730.0	(331.1)
Coal ash, spent dolomite and unburned carbon	7.9	(3.6)
Vent gases from coal and dolomite drying (including water vapor)	<u>35.5</u>	(16.1)
Total	773.4	(350.8)

Table 4-9

PFBC ENERGY BALANCE

Single Combustor Design Point Operation

<u>Inputs</u>	<u>MW(th)</u>
Combustion air (sensible heat)	35.3
Coal (heat of combustion and sensible heat)	289.1
Dolomite (heat of reaction and sensible heat)	<u>0.8</u>
Total	325.2
<u>Output</u>	
Combustion gases (sensible heat)	316.7
Ash and reacted dolomite (sensible heat)	2.9
Carbon loss (heat of combustion and sensible heat)	2.8
Thermal losses from combustor	<u>2.8</u>
Total	325.2

PFBC PLANT PLOT PLAN

4-24

80-1-64-30

FIG. 4-4

Based on 43.8 hours of operation per week and the feed rates of 100.1 lb/sec (45.4 kg/s), the weekly requirements of coal and dolomite are approximately 7716 tons (7000 t) and 2480 tons (2250 t), respectively. In order to provide 60-calendar days storage, a coal stockpile of approximately 66138-ton (60000-t) capacity and a dolomite stockpile of 22046-ton (20000-t) capacity are required. The delivery rate and stockpile size given above for dolomite are based on a Ca/S ratio of 1.75:1, rather than the design point value of 1.5:1. This is to allow for possible variations in the coal and dolomite specifications.

It is envisaged that coal will be delivered to the site by 40-car train sets, each car having a capacity of 99.2 tons (90 t). Two such trains per week are required; sidings to take these are included in the specifications. One siding will have under-rail receiving hoppers of approximately 110.2-ton (100-t) capacity.

The coal handling system follows conventional practice for installations of this scale. A car puller is used to position a loaded car over the hopper into which the coal is discharged. The hopper is fitted with a fixed grizzly screen to exclude dunnage and trash. An apron feeder takes the coal from the hopper and discharges it into a covered belt conveyor fitted with a belt weigher and totalizer. The conveyor is carried in a gantry with a service walkway and discharges into an open stockpile via a moving tripper.

Two coal stockpiles, each 985-ft (300-m) long, and 78-ft (24-m) wide with a maximum height of 28 ft (8.5 m) are required. The spread of the stockpiles is minimized by 6.6-ft (2-m) high concrete retaining walls (an angle of repose of 35° and a bulk density of 42.8 lb/ft³ (685 kg/m³) are assumed).

Coal is reclaimed from the stockpile by a reclaimer/conveyor system which can deliver it into each of the four coal mill storage bunkers. The conveying system is sized to operate at a maximum rate of 253.5 tons/hr (230 t/h) with the maximum coal consumption rate for the plant being 181.9 tons/hr (165 t/h). The coal mill storage bunkers each have a capacity of 771.6 tons (700 t) to provide operation for two days at the design rating.

The dolomite is delivered to the site and conveyed to the stockpile using the same system as for the coal. For reclaiming the dolomite from the stockpile, it is proposed that a front-end loader be used to transport the dolomite into a receiving hopper. A conveyor belt would then deliver the dolomite to the four dolomite mill storage bunkers. The conveyor is sized to operate at a maximum rate of 82.7 tons/hr (75 t/h) and the dolomite mill storage bunkers to hold 248.0 tons (225 t).

The main equipment items required for coal and dolomite handling are as follows:

- (1) train puller
- (2) under-rail receiving hopper
- (3) apron feeder
- (4) coal conveyor belt to stockpile, complete with flap gate, belt weigher, and tripper
- (5) coal conveyor belt to coal mill bunkers, complete with reclaimer and magnetic separator
- (6) dolomite conveyor belt to stockpile, complete with tripper
- (7) front end loader
- (8) receiving hopper, and
- (9) dolomite conveyor belt to mill bunkers, complete with magnetic separator.

Coal and Dolomite Preparation

The coal and dolomite will be delivered to the site with a top size of 1 1/4 in. (32 mm). Separate mills for the coal and dolomite dry and crush the material to the top size of 1/8 in. (3 mm) required for feeding to the combustor.

The hot gas for each drying circuit is taken from the LT exhaust with the temperature being controlled by a tempering air fan and the necessary pressure provided by a booster fan. The coal and dolomite are dried in the mills which are swept by the hot gas. The hot gas also operates the mill classifying systems and transports the prepared feed to disengagement cyclones. The disengagement cyclones for both the coal and dolomite discharge into a common buffer hopper, and the gases (and water vapor) leaving the cyclones pass to bag filters before being vented to the atmosphere. The fine particles collected in the bag filters are also returned to the common buffer hopper.

The layout of the milling plant and fuel injection equipment is shown in Fig. 4-5. A list of the main equipment items required for coal and dolomite preparation is as follows:

COAL AND DOLOMITE MILLING PLANT AND FUEL INJECTOR LAYOUT

ELEVATION

P L A N

FIG. 4-5

- (1) coal mill feed hopper
- (2) coal mill feeder
- (3) coal mill
- (4) coal collection cyclones
- (5) coal drying gas booster fan
- (6) dolomite mill feed hopper
- (7) dolomite mill feeder
- (8) dolomite mill
- (9) dolomite collection cyclones
- (10) dolomite drying gas booster fan
- (11) buffer hopper
- (12) drying gas vent filter, and
- (13) rejects screen

Coal and Dolomite Feeding

Each combustor will have one injection system for feeding the coal/dolomite mixture to the combustors. The system selected is one developed and supplied by Petrocarb, Inc. for feeding solids into pressurized vessels. In this system, the mixture of coal and dolomite received in the buffer hopper would be fed in batches to the storage injector after being screened to remove oversize material (e.g., agglomerates). The oversize material may, if appropriate, be returned to the coal mill. The storage injector discharges the feed in batches to the primary injector following this four-step cycle:

- (1) The storage injector is filled with coal/dolomite mixture.
- (2) It is then isolated from ambient pressure and brought to the same pressure as the primary injector.
- (3) The primary injector inlet valve is opened and the feed transferred to the primary injector. Feed terminates when the primary injector reaches a minimum preset weight as indicated by the load cells in which it is mounted.

- (4) The empty storage injector is then isolated from the primary injector and depressurized with the vented air being released through the bag filter in the milling and drying system. When at ambient pressure, the storage injector inlet valve is opened and step (1) is repeated with the storage injector being refilled with screened feed from the buffer hopper.

Feed from the primary injector is continuous, there being no interruption when a fresh charge is received from the storage injector. Because it has been found from experience that it is unnecessary to use inert gas as a precaution against spontaneous combustion in the injector vessels, each primary injector outlet (twenty-four outlets per combustor in the present design) is fluidized by bleed air. The feed rate is regulated by changing the differential pressure between the inside of the primary injector and the combustor. This is achieved by varying the quantity of air vented from the primary injector to the milling and drying system bag filter.

The air required to pressurize the injector vessels and to transport the feed to the combustor is taken from the exhaust of the ET. The air is cooled to 122 F (50 C), compressed to 20.7 atm (bar) by a booster compressor, and cooled again to 122 F (50 C) for use in the injectors. The air velocity through the conveying lines is maintained constant through the plant operating range. The fuel feed rate from the primary injector can be reduced to one-third of the full-load rate by changing the pressure differential as described above. Further turndown can be achieved by concurrently shutting off the feed flow to individual conveying lines and the fluidizing air flow to the isolated injector outlets. The transport air is allowed to continue flowing through the conveying lines, however, to prevent settling of residual feed in the line and backflow of bed material into the line.

A list of the main equipment items required for coal and dolomite feeding is as follows:

- (1) storage injector
- (2) primary injector
- (3) transport air compressor suction cooler
- (4) transport air compressor, and
- (5) transport air compressor aftercooler.

Combustor and Gas Cleanup

The combustor design chosen is a refractory-lined tapered bed contained within a horizontal refractory-lined pressure vessel. The main dimensions

of the combustor bed and pressure vessel are given in Table 4-10. A tapered bed with a 2-to-1 area ratio (top-to-bottom) has been chosen because the higher fluidizing velocity at the base of the bed improves coal and air distribution, and the area of the distributor plate is reduced.

The operating conditions are:

fluidizing velocity (at freeboard section)	3.0 ft/sec (0.9 m/s)
bed temperature	1562 F (850 C)
pressure at inlet to plenum chamber	15.2 atm (bar)

The pressure drops (atm (bar)) across the system are as follows:

distributor plate	0.20
bed	0.25
cyclones	<u>0.85</u>
total	1.30

The combustor design and layout is illustrated in Fig. 4-6.

The combustion air is introduced through the outer part of the co-axial connecting pipe and is allowed to fill the pressure vessel, thereby cooling the internal components. The air then passes upward through the distributor plate into the fluidized bed. The coal/dolomite mixture is fed pneumatically through twenty-four feed points located immediately above the distributor plate. The combustion gases leave the bed and enter the freeboard section where most of the particles carried out of the bed by the turbulence of fluidization disengage from the gas stream and return to the bed. The gases are then ducted from the top of the freeboard to the cyclone system. The inlets to the cyclones are arranged to minimize transverse gas velocities in the freeboard.

The combustor and cyclones are both contained in the same pressure vessel. This has the advantage of reducing the amount of external hot gas ducting and simplifying the use of a co-axial ducting system, but results in a pressure vessel of dimensions which require site fabrication. Existing experience in the construction of large pressure vessels includes vessels of the size envisaged by the present application. A list of relevant pressure vessel construction contracts carried out recently by Babcock and Wilcox Ltd. of the UK is given in Table 4-11.

Table 4-10

PFBC DIMENSIONS

Pressure Vessel:

Diameter	38.1 ft	(11.6 m)
Length	126.2 ft	(38.45 m)

Combustor Bed:

Bed length	88.1 ft	(26.85 m)
Bed width (maximum)	9.5 ft	(2.91 m)
Bed area (maximum)	840.7 ft ²	(78.1 m ²)
Bed height	10.0 ft	(3.05 m)
Freeboard height	10.0 ft	(3.05 m)

FLUID BED COMBUSTOR GENERAL LAYOUT

FIG. 4-6

Table 4-11

PRESSURE VESSELS CONSTRUCTED RECENTLY BY BABCOCK & WILCOX, LTD.

Pressure Vessel Dimensions -----				
<u>Year</u>	<u>Client</u>	<u>Length, ft (m)</u>	<u>Internal Diameter, ft (m)</u>	<u>Operating Pressure, atm (bar)</u>
1977	Kellogg for Turkey	15.1 (4.60)	15.1 (4.60)	34
	(2 vessels)	16.7 (5.10)	15.1 (4.60)	34
1977	Davy Power Gas	54.8 (16.70)	14.4 (4.38)	95
	for USSR			
1975	BDP Group for	100.0 (30.48)	12.0 (3.66)	30
	Danks			
1975	Badger	44.3 (13.49)	10.0 (3.05)	39
1974	CJB (Proj.) Ltd.	87.0 (26.51)	24.0 (7.32)	24
	for USSR			
	(3 reactors)			
1974	BDP Group (various	126.7 (38.63)	15.0 (4.57)	29
	pressure vessels)			
1974	Ko-Ri Power Station	228.7 (69.70)	105.3 (32.08)	1.7
	Korea			
1973	BDP (Oldbury)	109.2 (33.28)	13.0 (3.96)	46

Table 4-11 (Cont'd)

PRESSURE VESSELS CONSTRUCTED RECENTLY BY BABCOCK & WILCOX, LTD.

Pressure Vessel Dimensions				

<u>Year</u>	<u>Client</u>	<u>Length, ft (m)</u>	<u>Internal Diameter, ft (m)</u>	<u>Operating Pressure, atm (bar)</u>
1971	GEC/REL	144.0 (43.89)	108.0 (32.92)	3.7 internal 1.1 external
1971	Procon (GB) Ltd.	46.1 (14.06)	10.0 (3.05)	183
1971	Humphreys & Glasgow	31.6 (9.64)	12.8 (3.90)	57
1969	James Howden & Co.	32.7 (9.98)	10.0 (3.05)	45
1968	Fluor for BP Grangemouth (4 reactors)	67.6 (20.60)	12.0 (3.66)	179
		66.3 (20.22)	12.0 (3.66)	179
		77.3 (23.57)	12.0 (3.66)	172
		68.6 (20.91)	12.0 (3.66)	179
1966	Edwin Danks	67.3 (20.50)	12.0 (3.66)	33
1966	Bechtel Int. Ltd. for Gulf Oil, Milford Haven	157.5 (48.00)	20.0 (6.10)	4.6 to 5.3
1965	Lummus for ICI Wilton (2 fractionators)	270.8 (82.55)	16.0 (4.88)	21
		249.3 (75.98)	13.0 (3.96)	21

The proposed gas cleanup system consists of three stages of high-efficiency cyclones. The first two stages are of conventional design; the third stage is an ultra-high efficiency cyclone currently in the development stage. The specification of the third stage has not been fixed in the present study; however, the view is taken that advanced cyclone gas cleaning will produce a gas of acceptable quality for use in a gas turbine operating at an inlet temperature of 1562 F (850 C).

A list of the main equipment items required for the combustor and gas clean up is as follows:

- (1) pressure shell
- (2) refractory lining
- (3) refractory lined combustion bed containment including plenum chamber and air distributor plate
- (4) gas cleanup cyclones
- (5) cyclone interconnecting ducting
- (6) cyclone support system
- (7) startup burners
- (8) coal distribution system, and
- (9) ash and spent dolomite withdrawal system.

Ash and Spent Dolomite Handling

Excess bed material is released through a bed level control system into a pneumatic conveying system. The conveying air and bed material are transported through water-jacketed conveying lines, which act as coolers. The system is designed to maintain constant conveying velocities in the conveying lines, irrespective of changes in plant pressure levels. High air-to-dust ratios are used to minimize the possibility of conveying line blockages. The solids are disengaged using cyclones and discharged to surge hoppers. The conveying air is filtered and cooled before being returned to the combustor via a booster fan. The solids are discharged through pressure locks and conveyed by a standard low-pressure pneumatic handling system to storage hoppers for delivery to final disposal.

The fines collected by the cyclones are handled in a similar way to that of the bed material, except that a small underflow of gas from each cyclone is

arranged to minimize the possibility of conveying air entering the cyclone through the dust discharge leg. This type of pneumatic transport system has been used successfully for a range of temperatures and pressures and no difficulties are expected in its application to the FBC/CAES system.

The coal and spent dolomite handling equipment has been sized on the basis of a Ca/S ratio of 1.75 to allow for variation in the feed specification, and is designed for the disposal of 661 tons/day (600 t/d) of coal and spent dolomite from the site. Four storage hoppers each capable of holding a one-day production of 165 tons (150 t) are located near the combustors. The solid waste is conveyed pneumatically from these storage hoppers to six hoppers positioned over the rail track for discharge into rail cars. The capacity of each of these hoppers is 331 tons (300 t), providing storage capacity for three days.

A list of the main equipment items required for ash and spent dolomite ("ash") handling is as follows:

- (1) "ash" transport air booster compressor suction cooler
- (2) "ash" transport air booster compressor
- (3) bed "ash" cooler
- (4) primary cyclone "ash" cooler
- (5) secondary cyclone "ash" cooler
- (6) tertiary cyclone "ash" cooler
- (7) bed "ash" collection cyclone
- (8) primary cyclone "ash" collection cyclone
- (9) bed "ash" transport air filter
- (10) primary cyclone "ash" transport air filter
- (11) secondary cyclone "ash" transport air filter
- (12) tertiary cyclone "ash" transport air filter
- (13) "ash" transport air return compressor
- (14) bed "ash" surge hopper

- (15) bed "ash" lock vessel
- (16) primary cyclone "ash" surge hopper
- (17) primary cyclone "ash" lock vessel
- (18) secondary cyclone "ash" surge hopper
- (19) secondary cyclone "ash" lock vessel
- (20) tertiary cyclone "ash" surge hopper
- (21) tertiary cyclone "ash" lock vessel, and
- (22) "ash" storage bunkers.

Controls and Instrumentation

The combustor control and instrument equipment have been designed to form part of a centralized control room, which will also include the control and instrument equipment for the turbomachinery, etc. The control systems and instrumentation required for the main combustor equipment items are described in the following sections.

Coal and Dolomite Handling

A full set of conveyor controls will be provided for this section. Stockpile control is achieved by using belt weighers for the input and output conveyors with these signals being registered in a centralized control panel.

Coal and Dolomite Preparation

The control panels for this section could either be installed in the central control room or in an area near the mills. Instrumentation includes twin air damper control loops for mill conveying air flow and temperature. During normal running, hot flue gas is mixed with tempering air to provide the conveying and drying functions for the coal mills. During startup, a gas heater system is required to obtain the desired air temperature for milling. Similar control loops are applied to the dolomite mill except that cold air would be used, with a bleed of flue gas to the dryer if required.

Fuel feed and blending control are achieved by variable speed drives on the screw feeders to the mills. Control loops adjust the feed rates to maintain an adequate level in the hoppers, and a ratio control proportions the coal and dolomite feed rates to a value set by the operator in response to flue gas SO₂ readings.

Coal and Dolomite Feeding

The main feature of the control equipment for this section is a set of automatic sequence controls to operate the transfer of blended coal and dolomite from the buffer hopper to the storage injector and then from the storage injector to the primary injector. Transfer cycles are initiated by low material level signals obtained from load cell instruments attached to injector vessels.

Combustor and Gas Cleanup

The power output of the combustor can be controlled by changing the pressure at the inlet to the expansion turbine. This is achieved by throttling the airflow through the combustor using a control valve located at the inlet to the expansion turbine. The mass flow of the air entering the combustor is measured and the fuel feed rate automatically controlled relative to the airflow rate. In order to maintain the combustor outlet temperature at a constant value, the ratio of coal to air is changed continuously by a control loop acting from bed temperature signals. Limiting values are set into the control loop to ensure that adequate excess air levels are maintained at all times.

The rate of fuel injection can be adjusted by changing the differential pressure existing between the primary injector vessel and the combustor vessel. A differential control loop is required to maintain the pressure difference between the vessels by operating a control valve in the vent line from the primary injector vessel. Changing the set point of the controller will change the feed rate of the fuel.

Because the above power and fuel compensation functions are highly interactive, additional compensation functions are likely to be required. Feed forward signals from the plant pressure or load controls can be considered for enhancing the action of the fuel feed control to avoid large temperature transients in the combustor outlet temperature as the combustor airflow is changed. Conversely, some limiting values may be required to be applied to the air control valve action to override the load control action if this were driving the combustor outside preset limits of temperature and pressure.

Instruments will signal the bed differential pressure which is then computed with the air mass flow to obtain a measurement of the mass of material in the fluid bed. This measurement will be used to adjust the frequency of operation of coal and spent dolomite removal valves in order to maintain a constant bed mass.

Instruments required to monitor stack emissions will include flue gas analyzers for continuous monitoring of oxygen and sulphur dioxide in the flue

gas. Manual control of SO₂ levels is proposed by adjusting the coal/dolomite ratio at the blending stage. An online continuous particulate monitor is proposed so that action can be taken to protect the turbines from accidentally high dust levels.

Automatic sequence control equipment is included in the combustor control panel to supervise fluidizing air and startup burner operations. Override action is applied to the main air control valve and the load control signals during this mode of operation. Similar actions are applied by the system when shutdown is selected by the operator during a plant-trip.

The combustor control section includes all field and panel instrumentation necessary for the operation of the combustor as outlined above, with the exception of the main air control valve and the plant load control equipment which will be part of the balance-of-plant control section. Arrangements will be made for the normal signals to this valve to be routed via the combustor control panel for monitoring the override actions to be performed.

Ash and Spent Dolomite Handling

A full set of control instruments is proposed for the ash handling equipment. Central control room panels will be compatible with the computer control panels and sequencing controls will be interlocked with the combustor bed control systems.

CAES TURBOMACHINERY PRECONCEPTUAL DESIGN

The objective of the turbomachinery preconceptual design effort was to produce a design description of a turbomachinery system which could be utilized in CAES applications. Therefore, as with the fluid bed combustor, a turbomachinery plant was synthesized which would satisfy the design basis presented in the section entitled Preconceptual Design Characterization. This section presents the results of the turbomachinery design effort and includes a description of the turbomachinery, an identification of the main equipment items, and an illustration of the arrangement of these equipment items in several component and layout sketches.

Scope of Turbomachinery Design Effort

Liberal utilization was made of existing CAES turbomachinery preconceptual designs because of the limited magnitude of the present study. During the study, however, it was possible to improve upon existing designs described in Refs. 4-2 and 4-3.

The battery limits for the CAES turbomachinery preconceptual design were defined as follows:

- (1) the inlet and exhaust flanges of the compressor units within the compressor section;
- (2) the inlet and exhaust flanges of the turbine units within the turbine train;
- (3) the motor/generator;
- (4) the clutches; and
- (5) the baseplate of the major equipment.

The following items are excluded:

- (1) the intercooler and aftercoolers;
- (2) the central control system; and
- (3) all civil works.

Turbomachinery Design Basis

The following parameters were adopted as the basis for design for the turbomachinery:

- (1) The flow rates, temperature and pressure at the design point are those given Table 4-2.
- (2) Four parallel turbomachinery trains with a turbine section power output of 222.8 MW and a compressor section power input of 199.9 MW were utilized. Each of these trains is fed by its own fluid bed combustor.
- (3) Each turbomachinery train has a compressor section consisting of a low compressor, a high compressor, and a booster compressor. It also has a turbine section consisting of an expansion turbine (high-pressure turbine) and a low-pressure turbine. These are mounted on a common centerline with the compressor and turbine sections straddling a motor/generator and connected to the motor/generator by clutches.
- (4) All turbomachinery components operate at 3600 rpm.
- (5) Because of the peak-load duty cycle chosen, wherein the time during which charging occurs approximately equals the time during which generation occurs, the compressor section and turbine section are sized for approximately the same mass flow.

Unit Operation Description

A schematic diagram showing the main turbomachinery units and flows is given in Fig. 4-3. A brief review of the turbomachinery and its ancillaries is given below, more detailed descriptions of the units are included in the section on Turbomachinery Equipment Specifications.

The operating sequence of a CAES power plant requires that the turbine section and compressor section be alternately connected and disconnected from the motor/generator. During the air storage mode of operation, the compressor section is clutched into the motor/generator which is acting as a motor and the turbine section is declutched. Ambient air is drawn into the low compressor. The air is compressed to an intermediate pressure level and then directed to an intercooler. It then proceeds to a high compressor for further compression. The intercooler location is selected to minimize compression input power requirement for compressing air from ambient pressure level to that level associated with the high compressor discharge pressure. The high compressor discharge pressure level corresponds to a level which can be obtained with demonstrated state-of-the-art technology.

Additional intercooling occurs prior to the air entering the booster compressor. In the booster compressor, the air is compressed still further, intercooled again, and then compressed to a pressure level compatible with the air storage facility. The booster compressor intercooler is located so as to keep the temperature of the compressed air from exceeding those levels which can be tolerated by state-of-the-art booster compressor units.

Prior to storage, the air is cooled again in an aftercooler which contains a moisture separator. The level of cooling in the aftercooler is dictated by air storage reservoir requirements. The air is then directed to the underground air storage wherein it undergoes further compression due to the weight of the column of air in the air shaft. While in storage, there will be some loss of air to the storage media.

During the generating mode of operation, the compressor section is declutched from the motor/generator and the turbine section is clutched to the motor/generator which will now act as a generator. Air is directed from storage, incurring a slight loss in pressure as it rises to the surface. It then travels to the recuperator where it picks up thermal energy from the low-pressure turbine exhaust gases. The heated air is then expanded in an expansion turbine from approximately air storage pressure down to a level compatible with that of the low-pressure turbine.

After leaving the expansion turbine, a side stream of air is removed from the main stream to provide turbine blade cooling in the low-pressure turbine. Additional side streams of air are also removed prior to the fluid bed combustion to serve as transport air for the coal and dolomite fuel system and transport air for the coal ash and spent dolomite handling system. The main flow of air goes to the fluid bed combustor system where it is used as combustion air in the fluidized bed. The transport air is reunited with the main stream when the coal ash and spent dolomite transport air is returned to the main stream flow just prior to the fluid bed combustor, and when the coal/dolomite mixture is fed into the fluid bed combustor. The hot exhaust gases from the fluid bed are cleansed and are expanded in the low-pressure turbine. During the expansion process, the cooling air, which had been extracted after the expansion turbine is reintroduced to the main stream to provide blade, rotor disc, and bearing cooling.

The gas stream then travels to the recuperator. Prior to arriving, however, another side stream of air is removed to provide drying for the coal and dolomite feed. The remaining stream proceeds to the recuperator to release a portion of its thermal energy to the storage air traveling to the expansion turbine. The relatively cool gases are then directed to the stack.

Turbomachinery Layout

The proposed layout for the turbomachinery is illustrated in Fig. 4-7. This layout represents an update from the layouts in previous reports (Refs. 4-2

TURBOMACHINERY TRAIN LAYOUT

FIG. 4-7

and 4-3). These changes have been aimed at reducing the complexity and projected costs of the turbomachinery by consolidating and simplifying individual units, as well as the total train.

The most obvious of the simplifications has been the elimination of the separate expansion turbine/generator set in favor of mounting the expansion turbine on a common centerline with the other turbomachinery. This should eliminate the potential control problem which existed by having the expansion turbine on a separate shaft.

The clutches have been integrated into the compressor and turbine units. In order to expedite starting, a separate clutch is located between the redesigned high compressor and the booster compressor.

All units have been arranged on the aforementioned single centerline in an elevated position to facilitate piping between the various individual units and the support equipment which is located below it. The fluid bed combustor will be located to one side of the turbomachinery on the same approximate level, while the various intercoolers, as part of the support equipment, will be on the lower level. The recuperator will be incorporated into the exhaust stack and, along with the PFBC, will be located external to the machinery gallery.

The principles of modular construction have been retained and expanded on throughout the turbomachinery package. Whole compressor and turbine sections may be removed and/or replaced by the simple uncoupling of the drive shaft on either end of the unit and the disconnection of the section mounting points. Individual sections may be further disassembled by modules: low compressor, high compressor, low-pressure turbine, etc. By removing the whole turbine section to one side, for instance, it becomes possible to remove the motor/generator rotor axially without disturbing the stator and outer case. In this way, no realigning need be done of the motor-generator and compressor units. Only the turbine unit has been disturbed; and since it has not been further disassembled, it should be self-aligning at reassembly.

The proposed revisions to the various turbomachinery units have served to reduce the overall length (as compared to Refs. 4-2 and 4-3) of the total power generating package (excluding recuperator duct) to approximately 126 feet (38.4 m). This is a reduction of approximately 100 feet (30.5 m).

It is felt that a major savings in structural cases, scroll work, and number of bearings necessary throughout the turbomachinery package has been made in this re-examination. Further, less apparent, savings are also possible in a reduction in the size of the building housing and the turbomachinery gallery due to the reduction in overall machinery length.

Compressors

The low-pressure axial compressors (Fig. 4-8) have been unified by a new double scroll support structure which incorporates provisions for low-compressor discharge and high-compressor inlet air to one of a series of intercoolers. This structure also provides support structure and oil pressure and scavenge services to the bearings located at one end of each compressor. The support parts on the other end of each unit is unchanged from previous designs. Mount points on these parts are utilized for mounting the redesigned unit. An additional mount point, which would be limited to vertical and side load reaction may be utilized on the new central case, if so desired.

The booster compressor (Fig. 4-9) remains essentially unchanged. The inlet duct to the compressor from the second intercooler has been placed vertically as is the outlet duct to the aftercooler. The inlet duct to the third intercooler and exhaust duct from that intercooler has been placed horizontally.

Turbines

The expansion turbine has been extensively modified from previous CAES designs (Fig. 4-10). Dual inlet ports have been manifolded to a single inlet to facilitate connection to the recuperator. The power shaft aft end has been adapted to a splined connection on the low-pressure turbine shaft to continue the modular repair or replacement concept into the turbine area. The air exiting from the expansion turbine has been ducted into a new structural, outer-split case to the fluid bed combustor in such a way as to surround the extremely hot pipe arriving at the low turbine inlet from the fluid bed combustor. This coaxial pipe arrangement serves to keep the external, structural case relatively cool and, thus, minimizes thermal distortion due to uneven stress buildup in the structural members of the turbine unit. It also serves to eliminate any major pressure differential across the walls of the high temperature fluid bed exhaust pipe because the pressure of the air entering the fluid bed combustor in the outer duct is essentially the same as the hot gases within the inner duct. The inner duct is also free to expand on the turbine end, thus minimizing thermal expansion stresses.

Some re-blading and re-statoring, together with new spacing parts in the rotor and stationary structure of low-pressure turbine, is necessary; but the aerodynamics, discs, etc. remain essentially unchanged. Mount points are provided on both the forward and aft cases with an auxiliary mounting position available near the midspan for vertical and/or side load reaction, as in the axial compressor.

It should be noted that the low-pressure turbine exit scroll shown in the detail drawing (Fig. 4-10) has been eliminated in the assembled drawing (Fig. 4-7) in favor of a straight diverging duct extending to the externally situated recuperator and stack.

CAES COMPRESSOR UNITS

FIG. 4-8

0 5 10
SCALE - FEET

4-46

79-11-80-4

BOOSTER COMPRESSOR

COMPRESSOR
EXIT TO INTERCOOLER # 3

FIG. 4-9

TURBINE UNITS

FIG. 4-10

Turbomachinery Equipment Specification

The function of the major CAES turbomachinery equipment components has been discussed briefly in the section entitled Unit Operation Description. These equipment components have been extensively discussed in Refs. 4-1, 4-2 and 4-3, consequently only a summary of the major points is deemed appropriate here. The following subsections will present that summary.

Low-Pressure Compressor

The low compressor and high compressor which comprise the low-pressure compressor are 3600-rpm, multi-stage axial flow units of seven and ten stages, respectively, separated by an intercooler stage (Fig. 4-8). Intercooling is provided to reduce work and to maintain the design value of corrected airflow through the high compressor. These machines were originally derived from the three-shaft United Technologies FT-50 industrial gas turbine generator but have subsequently been updated to incorporate recent design developments. The low and high compressors are housed in a single casing and have been redesigned for CAES applications.

These modifications include:

- 1) replacing an inlet case thrust bearing by a 90,000-lb capacity bearing;
- 2) replacing a compressor intermediate case by a low-compressor diffuser and high-compressor inlet duct to accommodate compressor intercooling;
- 3) using a new high-compressor diffuser case with the burner section deleted; and
- 4) coupling the high-and low-compressors and operating them at 3600 RPM.

With these modifications, the low-pressure compressor is capable of compressing 761.8 pounds of air per second (345.5 kg/s) from ambient to 16.6 atm (bar). This requires 123.4 MW of shaft power.

Booster Compressor

The booster compressor (Fig. 4-9) would involve extending the heavy-duty industrial Ingersoll-Rand 87RE series, vertically-split, multistage, centrifugal compressor to seven stages and incorporating intercooling after the fourth stage. The booster compressor is preceded by an intercooler and followed by an aftercooler. The booster compressor will consume 76.5 MW of shaft power while operating at 3600 rpm to compress 761.8 pounds of air per second (345.5 kg/s) from 16.4 atm (bar) to 68.0 atm (bar) (the pressure head of air in the vertical air shaft will raise the air to the net storage pressure of 71.1 atm (bar)).

Expander Turbine

The expander turbine (Fig. 4-10) is a 3600-rpm, Stal Laval Model TMI-19, two-stage (one governing, one expansion) steam turbine modified for use with compressed air. The turbine will expand 731.3 pounds of air per second (331.7 kg/s) from 60.4 atm (bar) down to 15.3 atm (bar) while producing 48.0 MW of shaft power.

Low-Pressure Turbine

The low-pressure turbine (Fig. 4-10) is also derived from the FT-50 gas turbine, with a number of recent design modifications. These modifications include:

- 1) reversing the direction of rotation of the high- and low-turbines from the present direction;
- 2) joining the high-, low-, and free-turbine rotors to form a four-stage, 3600-rpm turbine; and
- 3) designing first- and second-stage turbine vanes and blades (i.e., new high- and low- turbines) to accommodate the reversed direction of rotation.

These modifications will allow the low-pressure turbine to generate 174.8 MW while expanding the 746.6 pounds of air per second (338.7 kg/s) of both combustion gases and turbine cooling air from 13.6 atm (bar) to essentially ambient pressure.

Clutches

Three clutches are required for each CAES unit (Fig. 4-7). Two are required for the compressor train, one adjacent to the motor/generator and rated at 200 MW and one between the axial and centrifugal compressor rated at 77 MW. The remaining unit, located between the motor/generator and the low-pressure turbine is rated at 223 MW. The SSS Gears Ltd. clutches (supplied under license by Waukesha Bearings) are based upon designs commonly used in gas turbine-powered navy destroyers, and are being used in the West German CAES plant. The incorporation of the smallest clutch allows the booster compressor to be isolated from the system for ease of starting.

A clutch would be installed between the output shaft of the driving component (i.e., from the motor/generator during compression or from the turbine section during power generation) and the input shaft of the driven component. Engagement of the two shafts would occur whenever the rotational speed of the output shaft overtakes the rotational speed of the input shaft. Once engaged, torque is transmitted from the output shaft to the input shaft. Disengagement would occur whenever the output shaft rotational speed drops

below that of the input shaft. Both engagement and disengagement proceed automatically when the appropriate commands have been signaled to the hydraulic locking systems.

Motor/Generator

The design used in this study includes a motor/generator operating between the compressor train and the turbine train.

The motor/generator is nominally rated at 250 MVA/225 MW, 17 KV, 60 Hz, and is sized based upon compressor train power requirements. This 3600-rpm machine is available from several manufacturers and would typically be a hydrogen-cooled unit incorporating static excitation.

BALANCE OF PLANT PRECONCEPTUAL DESIGN

This section addresses the design of the components required to support the operation of the main CAES turbomachinery and the pressurized fluidized bed combustion system, and those systems required to make the PFBC/CAES plant into a workable utility power generation system. Most of these components and systems are common in both function, size, and design to other facilities. However, some are unique to the PFBC/CAES system and reflect a departure from standard power plant design and configuration.

As part of an oil-fired CAES development study (Ref. 4-4) presently being conducted for the Department of Energy and the Electric Power Research Institute by the Potomac Electric Power Company and their subcontractors, extensive design information has been developed which describes the balance-of-plant systems for a plant similar in size and duty cycle to the PFBC/CAES power plant which is the subject of this study. In compliance with the design approach philosophy, this existing information was used, with appropriate modifications where necessary, as the basis for the PFBC/CAES balance-of-plant design.

For the purpose of discussion the balance-of-plant is separated into four distinct sections:

- 1) energy storage system - those items in the FBC/CAES plant required to deliver, store, and regulate the storage pressure of the compressed air during off-peak periods in the utility schedule;
- 2) plant mechanical system - the mechanical equipment and systems required to support the CAES turbomachinery and PFBC system;
- 3) plant electrical system - the electrical equipment and systems required to effect a workable PFBC/CAES power plant; and
- 4) PFBC/CAES site arrangement - the overall arrangement of the power plant site and the interaction of various components within the site layout.

Energy Storage System

The energy storage system includes those major surface and underground structures which store the compressed air. Included in the energy storage system are the main air and water shafts, the underground cavern, and the surface water reservoir. The energy storage system must provide storage

capability for the daily operation of the four-unit PFBC/CAES plant. It must also be designed to allow for a variety of operating modes with air storage and release rates available from zero to one hundred percent of rated air flow.

Compensating Reservoir

The compensating reservoir maintains the compressed air storage pressure at an essentially constant value during charging, storage, and discharging operations by using a constant head of water to the air storage cavern. The reservoir must provide a live storage volume equivalent to the volume of air being stored, as well as sufficient dead water storage volume to prevent vortices from forming over the reservoir intake structure.

The selected design consists of a diked reservoir of approximately circular shape. The dike cross section is constructed of compacted rock fill composed principally of material which was excavated from the storage cavern. There is an impervious lining of asphalt on the dike slope and reservoir bottom to control the tendency for erosion of the dike slopes due to fluctuations of the reservoir level and to minimize seepage. The reservoir intake structure is located at the inside toe of the dike, with a water conduit connected to the water shaft located outside of the dike area.

The level of the surface reservoir will fluctuate with the changes in stored compressed air volume. Since the storage pressure is equal to the hydrostatic head provided by the water column, the storage pressure will vary as the water levels in the surface reservoir and underground cavern fluctuate. The change in the water level in the surface reservoir plus the change in the water level in the cavern gives the total pressure variation. This pressure variation is defined within the turbomachinery limitations. Accordingly, the surface reservoir is designed to provide a live volume height fluctuation of approximately twenty-five feet in the reservoir. The total dike height is approximately thirty-five feet, to accommodate the 780,000 cubic yards (596,000 m³) of air storage volume required for the plant.

Intake Structure

The intake structure provides unrestricted water movement between the compensating reservoir and the main water shaft. This structure consists of isolation gates for use during dewatering the storage cavern, and trash racks to prevent entry of debris into the water shaft.

Water Shaft

The water shaft acts as a water column which maintains constant pressure in the air storage cavern. This shaft must transmit the water which moves

between the surface reservoir and underground caverns during compression and power generation cycles. The water shaft is sized to accommodate lowering of the construction equipment used in cavern excavation.

At the bottom of the water shaft is a U-tube bend. This bend is required to prevent loss of the water seal during cavern charging due to the champagne effect. This phenomena could occur when the cavern air which has gone into solution in the cavern water comes out of solution as the compensating water rises in the water shaft and hydrostatic pressure decreases. The air bubbles rising in the shaft could create an unstable condition leading to water shaft evacuation and rapid depressurization of the cavern. Previous studies (e.g. Ref. 4-2) indicated that a U-tube depth equal to 12 percent of the cavern depth should be adequate to prevent occurrence of a failure due to the champagne effect.

The water shaft is 20 feet (6.1 m) in diameter and is a concrete-lined structure extending approximately 2200 feet (670 m) between the surface reservoir and cavern. Shaft construction must be such that the shaft can sustain both outward pressures during operation and inward pressures during dewatering for cavern inspection. The water velocity in the shaft is restricted to approximately 4 ft/sec (1.2 m/s) under design flow rates for the turbomachinery.

Air Shaft

The air shaft must transmit compressed air at varying flow rates between the surface air piping and underground cavern without allowing the compressed air to escape. The shaft is designed to withstand the system operating pressures (up to 75 atm (bar)), and air temperatures up to 150 F (65 C). The velocity of the air in the shaft will be approximately 50 ft/sec (15 m/s).

The air shaft is constructed of stainless steel, approximately four feet (1.2 m) in diameter and supported by an outer concrete casing approximately two-feet (0.6-m) thick. Physically, the air shaft is located at the opposite end of the cavern from the water shaft. This allows for proper emptying of the cavern and prevents trapped air volumes within the cavern.

Underground Cavern

The underground cavern provides storage space for the compressed air and compensating water volumes. As with the surface compensating reservoir, the underground cavern design influences the storage pressure variation. Because this variation is a function of the height of the cavern, the cavern is made as low as practical within the constraints of turbomachinery performance and cavern construction economics.

The underground cavern consists of four hard rock tunnels interconnected by collection tunnels at the two ends. Each tunnel is approximately 60-feet (18.3-m) wide, 1400-feet (425-m) long, and 85-feet (26-m) high. The cavern occupies an overall area approximately as shown in Fig. 4-11.

The cavern layout dimensions were determined by the minimum cavern spacing and by the relationship between cavern length and total volume required. Cavern spacings and size must take into consideration the specific rock properties of the site. Both theoretical and experimental investigations have shown that a significant change in the state of stress of the material surrounding the cavern does not occur beyond a distance of two or three times the width of the cavity (Ref. 4-5). Thus, the maximum spacing required by an opening 60-feet (18.3-m) wide would be 240 feet (73 m) from center to center. The properties of the rock in which the caverns are being constructed also determine the maximum width of the cavity which will remain stable with a minimum of additional support. Poor rock characteristics may limit the width of the opening to less than the design width of 60 feet (18.3 m), and thus result in increased length.

In addition to the required 780,000 cubic yards (596,000 m³) of live storage volume, the underground cavern, as designed, provides adequate free-board to provide a margin against complete air depletion. There is also dead water storage to provide a margin against overcharging with compressed air.

Plant Mechanical System

Several mechanical systems are needed to support the operation of the PFBC/CAES plant and the main turbomachinery systems within it. These systems include the special equipment unique to the CAES cycle, and the more standard equipment which can be found in most fossil-fueled power plants of this size range.

Turbomachinery Buildings

Each of the turbomachinery trains and their support systems are housed in a two-story, rigid frame structural steel building. The building provides support and protection for the turbomachinery systems and includes adequate space for equipment laydown and maintenance. A general arrangement of the building and equipment within the building is presented in Figs. 4-12 and 4-13.

The building grade floor houses the compressor system intercoolers and aftercoolers, turbomachinery lubrication and cooling systems, and electrical auxiliaries. A maintenance bay is also provided on this floor. The building

SITE ARRANGEMENT PLAN

FIG. 4-11

TURBOMACHINERY SYSTEM GENERAL ARRANGEMENT

PLAN VIEW

PLAN

VIEW D-D

FIG. 4-12

TURBOMACHINERY SYSTEM GENERAL ARRANGEMENT

ELEVATION

FIG. 4-13

machinery floor is located 26 feet (7.9 m) above the grade floor. This floor supports the main turbomachinery train and provides a maintenance laydown area for turbomachinery disassembly. The turbomachinery building is serviced over its full length by a 125-ton (113.4-t) capacity bridge crane. This crane is capable of lifting and transporting the single heaviest maintenance lift anticipated within the turbomachinery building.

Compressor Intercoolers/Aftercooler

The air compression cycle utilizes three stages of air intercooling and an aftercooling stage to condition the air for cavern storage. A total heat flow of approximately 176,000 Btu/sec (185,700 kJ/sec) per unit must be absorbed from the compressed air by the intercooler/aftercooler system.

The shell-and-tube intercoolers and aftercoolers are designed to use water delivered from a wet cooling tower as the air cooling medium. Cooler bodies, located in the turbomachinery building grade floor level, are provided for each unit in the following quantities:

<u>Cooling Stage</u>	<u>Heat Rejection, Btu/sec (kJ/s)</u>	<u>Number of Coolers</u>
intercooler #1	34,000 (35,900)	1
intercooler #2	56,000 (59,100)	3
intercooler #3	33,000 (34,800)	1
aftercooler	<u>53,000 (55,900)</u>	12
Total	176,000 (185,700)	

Intercoolers have been sized to provide for cooling the air to 200 F (93.3 C). The aftercooler stage must cool the air down to 120 F (48.9 C) for delivery to the underground cavern.

Moisture Separators

Moisture separators are located after the first compressor intercooling stage. These separators serve to protect the second axial compressor body from water impingement damage by removing up to 95 percent of the moisture droplets formed during the first stage of intercooling. Additional moisture removal will take place after the air aftercooling stage, using simple in-line, knock-out pots.

Exhaust Gas Recuperator

The exhaust gas recuperator extracts heat from the low-pressure turbine exhaust gas stream and imparts it to the high-pressure air being released from

the cavern. It is positioned in a vertical section of the exhaust stack, supported by a structural steel framework approximately 80 feet (24 m) above grade. The recuperator and stack are located immediately outside of the turbomachinery building.

The recuperator is designed as a multi-pass U-tube system, with the high-pressure air contained within the tubing and the exhaust gas passing outside of the tubes. The complete assembly is made up of five distinct recuperator modules tied into common inlet and outlet headers. The recuperator is designed to heat 732 lbs/sec (389 kg/s) of dry air at upwards of 68 atm (bar) from 120 F (48.9 C) to 525 F (274 C) by using the exhaust gas from the low-pressure turbine. The recuperator is designed for rapid startup and cyclic operation over the expected range of turbine air flow rates and temperatures.

Ducting and Silencing

Inlet and exhaust systems are provided to supply air to the compressor inlet and discharge turbine exhaust gasses to the atmosphere. The primary features of these systems include:

- 1) An inlet air filtration system containing prefilters and high-efficiency cartridge-type filters. High-efficiency filters will prevent coal and ash contaminants from reducing compressor performance. Filters are located on the roof of each turbomachinery building.
- 2) Inlet and exhaust silencing sufficient to reduce turbomachinery system noise impact on surrounding areas to an acceptable level.
- 3) Transition ducting which directs inlet air and exhaust gases at minimum pressure losses.

Combustor/Turbomachinery Concentric Duct

A concentric duct is provided to allow expansion turbine exhaust air to enter the combustor and to allow hot combustion gases to travel to the low-pressure turbine. The cooler expansion turbine exhaust air is in the outer case while the hotter combustion gases are in the inner case. Two arrangements are shown of the ducting in Fig. 4-13. In section B-B, an elevated combustor is shown with a downward sloping duct. This corresponds to the PFBC design of Fig. 4-6. However, the elevated turbomachinery arrangement which resulted from the balance-of-plant preconceptual design and which features elevated turbomachinery may make the upward sloping duct and foundation-mounted PFBC layout of section C-C more desirable.

Plant Cooling System

A group of forced draft wet cooling modules make up the plant cooling system. Four modules (one per turbomachinery train) service the cooling water loops for the four separate turbomachinery trains, while a fifth module provides cooling for common plant services.

The plant cooling system is located away from the turbomachinery buildings and switchyard so as to eliminate possible plume interaction with the switchyard and turbomachinery inlet air systems. The location of the cooling system in the overall site plan is illustrated in Fig. 4-11.

The maximum water circulation through the total wet tower system is approximately 60,000 gallons per minute ($227.1 \text{ m}^3/\text{min}$) during full load operation in the compression cycle.

Water Supply

The water supply system consists of facilities for supplying water to the compensating reservoir and other plant systems. A preliminary water supply system has been selected after defining the operational water requirements and evaluating potential water sources.

The plant water requirements consist of the following:

- 1) compensating reservoir initial fill and make-up;
- 2) plant cooling water system make-up;
- 3) service water system;
- 4) domestic water system; and
- 5) fire protection system

The water demands have been estimated, where possible, for each of the above systems. These demands amount to approximately 203 million gallons ($768,400 \text{ m}^3$) for the initial filling of the reservoir and service systems. In the normal daily operation of the plant, water is expected to be supplied at a rate of approximately 3 million gallons per day ($11,400 \text{ m}^3/\text{day}$). To provide the quantities of water required, the following sources of water would be considered depending on the site:

- 1) deep wells,
- 2) water impoundment of natural on-site sources,
- 3) direct precipitation,
- 4) storm water runoff, and
- 5) equipment generated water.

Waste Treatment

The waste treatment system will be utilized to process plant generated wastes in order to render them suitable for disposal. Wastes generated by the balance of plant will include solid refuse, water contaminated by coal, oils, detergents, and solids, and oily wastes. These wastes are typical of those found in most fossil-fueled generating plants. The design of this system is therefore standard.

Safety and Fire Protection

The plant safety and fire protection system will consist of various fire detection, suppression and alarm systems. The fire protection systems selected for servicing the plant facilities are:

- 1) water system for servicing the plant grounds;
- 2) foam system for protecting the fuel oil storage area and other areas where fuel oil is located;
- 3) halon/CO₂ system for servicing the various internal hazards of the machinery hall and control building;
- 4) portable fire extinguishers for personnel use throughout the station;
- 5) smoke detection and pull box system for serving areas not protected by special hazard systems; and
- 6) alarm transmission for alerting local fire departments.

Plant Electrical System

The PFBC/CAES plant requires all of the electrical features and capabilities normally found in a fossil-fueled electric power generation station of similar size. In addition, the electrical system must have the capability to start very large synchronous motors during the CAES plant compression cycle and deliver large amounts of electric power to them once started. The equipment required to perform these functions is described below.

Electrical Switchyard

The electrical switchyard ties the four unit motor/generators and the plant services to the existing electric power grid. The switchyard provides protection and reliability for the FBC/CAES plant site by 500-kV switching of the following circuits through a breaker-and-a-half scheme:

- 1) (2) 500-kV transmission lines,
- 2) (4) 500-kV motor/generator unit tie lines, and
- 3) (2) 500-kV station auxiliary transformer feeders.

Design of the switchyard equipment and layout incorporates 12 gas-insulated breakers connected with an air-insulated 500-kV bus, insulators, lightning arrestors, a 13.8-kV station service bus switchgear, and all necessary control, metering, and protection equipment. From the switchyard, two 500-kV transmission lines carry the plant power to the site boundary.

The switchyard occupies an area of approximately 15 acres (61,000m³) on the CAES plant site. This area is designed for proper drainage, access, and clearances in accordance with standard switchyard design practices. The switchyard location is illustrated in Fig. 4-11.

Electrical Power and Control System

The design of the plant electrical power and control system follows standard utility power plant practice and incorporates all of the electrical systems necessary to support the operation of the FBC/CAES power plant.

Static Motor Starting System

To provide for start up for the compressor/motor system upon initiation of each air compression cycle, a variable frequency static starting system is provided. This system eliminates the need to start up the turbine to drive the compressor/motor up to synchronizing speed. Turbine life is greatly increased, with a corresponding decrease in maintenance costs, through the use of the variable frequency static starting system.

A single system, rated at approximately 33-MVA input, is provided for the complete four-unit plant. The static starting system can consecutively start each of the four compressor/motor units within a one hour period without experiencing thermal restriction. The static starting system is located in the grade floor level of the main control building.

Main Control and Administration Building

The control and administration building provides a common center for the control of the four CAES turbomachinery trains and the coal handling, treatment, and combustion systems. This building also houses the static motor starting system, maintenance and parts storage facilities, and employee facilities.

The control and administration building measures approximately 220 feet (67m) by 80 feet (24m) and contains three levels of working space in most areas. It is located in close proximity to the turbomachinery buildings but as far as practical from the compressor air inlets and coal processing equipment to minimize the impact of noise and dust on control building design. The arrangement and location of the building relative to power block components is illustrated in Figs. 4-14 and 4-15.

PFBC/CAES Power Plant Site Arrangement

The arrangement of the complete four-unit FBC/CAES plant has been shown in Fig. 4-11. This arrangement was developed after giving careful consideration to the following factors:

- 1) physical size of surface and underground structures,
- 2) operational interaction of major components,
- 3) construction requirements, and
- 4) environmental impacts and interactions.

To provide an illustrative example of the difference between land requirements of a PFBC/CAES plant and a conventional oil-fired CAES plant, the site boundaries used in the Reference 4-4 study were imposed on the PFBC/CAES site arrangement.

The following sections describe the major characteristics of the site arrangement as presented in Fig. 4-11.

Underground Cavern

The underground cavern is positioned on the upper border of the site. Since it requires a long construction period, it has been positioned so as not to interfere with other plant construction. No plant components are positioned above the cavern structure reflecting concern for plant safety as a result of a highly improbable cavern failure. Land area above the cavern may also be required for excavated rock storage.

Compensating Reservoir

The compensating reservoir is located adjacent to the underground cavern, in the upper right corner of the site. Close proximity of the cavern and compensating reservoir reduces the effort required to construct the compensating reservoir using the rock excavated from the cavern. More importantly, however, the reservoir must be located close to the cavern to reduce the amount of water passage between the reservoir and cavern water shaft.

ALTERNATIVE 1

FIG. 4-14

POWER PLANT GROUP GENERAL ARRANGEMENT

ALTERNATIVE 2

FIG. 4-15

Coal and Fuel Oil Unloading and Storage

Location of the rail line access to the site determines the position of coal and fuel oil unloading and storage facilities. For the purposes of this study, rail access has been established at the lower left corner of the plant site. Other plant facilities have been located at a reasonable distance from the coal and fuel oil storage facilities to minimize coal dust contamination.

Power Block

The layout of the four-unit power block was presented in Figs. 4-14 and 4-15. The power block is made up of the turbomachinery building, fluidized bed combustors, coal and dolomite processors, and the central control and administration building. To reduce the impact of noise on surrounding properties, the power block is located toward the center of the site, away from boundaries.

Two arrangements of the power block were presented - Alternative 1 in Fig. 4-14 and Alternative 2 in Fig. 4-15. Location of the coal processing system either outboard or inboard, respectively, of the turbomachinery buildings, as shown on these two arrangements, does not significantly effect the overall size, configuration, or cost of the power block. The control and administration building is located adjacent to the four turbomachinery buildings. This location provides aesthetic appeal and minimizes noise and dust effects on the control building.

Electrical Switchyard

The electrical switchyard is located adjacent to the power block, since it is tied directly to it through several sets of power conductors. The switchyard is also positioned toward the center of the site to reduce the aesthetic impact of the plant on adjacent properties.

Cooling Towers

The forced draft wet cooling towers are located on the bottom side of the site, adjacent to the power block. Their location in close proximity to the power block reduces the cost of the large piping systems required to direct cooling water between the power block and cooling towers. Their location over 1200 feet (365 m) away from the switchyard is necessary to prevent cold weather icing of switchyard components caused by cooling tower plumes.

Construction Time Schedule

During the study, it became apparent quite early that installation of the above ground equipment would be the pacing item for plant construction. As a result, a detailed construction timescale for the above ground equipment was developed. The timescale for the design, manufacture, and construction of the four-unit CAES power plant is illustrated in Fig. 4-16. The first activity for both the fluid bed combustor and turbomachinery would comprise test work and development. This would be followed by engineering design, which continues to year 3.5. Also during this same period the necessary permits needed by the utility would be obtained.

Manufacture of the combustors begins at year 2.0, with the first combustor being delivered at year 3.5 and subsequent combustors delivered at six-month intervals. Erection of the first combustor starts at year 3.75 and finishes at year 5.0, with subsequent completions at six-month intervals.

Gas turbine manufacture begins at year 2.5, with the first turbine being delivered at year 4.0 and subsequent turbines at six-month intervals. Erection of the first gas turbine begins at year 4.25, with completions at six-month intervals.

Construction of the cavern begins at year 2.0. By year 3.5 the shaft will be completed and by year 5.0, the cavern. Construction of the compensating reservoir begins at year 4.5 and with its completion at year 5.5 the energy storage system is completed. Completion of the mechanical plant occurs at year 6.25 and the electrical plant at year 7.0.

Commissioning and plant shake-down commences in year 5.25, with completion of the first unit at year 6.25, the second unit at year 6.75, the third unit at year 7.25, and the fourth unit at year 7.75.

PFBC/CAES CONSTRUCTION TIMESCALE

FIG. 4-16

INSTITUTIONAL FACTORS

As with the construction and operation of any electric power generation facility, the PFBC/CAES plant should be designed to have as small an environmental impact on the surrounding area as possible. The environmental impacts and the safety of both the general public and plant operating personnel are therefore of major importance to successful development of the PFBC/CAES concept.

This section addresses those environmental and institutional factors which are expected to play an important role in the acceptance of a PFBC/CAES facility in any area of the United States. In addition to these major factors, additional environmental topics will surely come to bear in site specific areas where CAES plants may be economically viable.

Noise

The impact of PFBC/CAES plant noise on the area surrounding the plant site is primarily dependent on site location. In all likelihood, the plant will be constructed in a rural area or in an industrially-zoned site, both of which will have low residential population densities.

The noise associated with a PFBC/CAES plant stems from different operations in two distinct time frames -- plant construction and plant operation.

Plant Construction

During the construction of a PFBC/CAES plant, noise will be generated both local to the site area and in a large area servicing the site. Access roads will experience frequent large equipment traffic, especially if extensive amounts of site spoils must be removed from the plant boundaries. If rail access and power transmission lines must be brought to the site boundary, these activities will also create noise impact away from the immediate plant site. A third source of construction noise may arise from requirements to perform blasting operations in an air storage cavern mined from hard rock. Schedule requirements would probably require that such a cavern be constructed both during the day and night, adding to the impact of construction noise on surrounding areas.

Most noise generated during construction of the PFBC/CAES plant can be controlled to within expected government noise regulations. OSHA and mining

industry noise regulations for construction personnel can be achieved using standard noise control procedures for the construction techniques employed. Off-site noise limits along highways, and transmission and railroad right-of-ways may restrict the use of or construction in these areas to daylight hours when allowable noise levels are typically 10 dBA greater than nighttime allowable levels.

Plant Operation

Operation of the PFBC/CAES plant will generate noise from several sources including the rail siding, compressor inlets, turbine exhausts, cooling towers, and switchyard. Compliance with both daytime and nighttime noise regulations at the plant boundaries will be required.

Proper design of equipment buildings and gas path ducting, as well as judicious placement of equipment within the site boundaries, should enable a PFBC/CAES plant to achieve operating noise levels of approximately 55 dBA, measured at the site boundary. This level is equivalent to typical rural residential, nighttime noise regulation requirements. Noise generated by rail delivery of coal to the site may, however, require special attention in the form of access berms and possibly the restriction of coal deliveries to daylight hours.

Air Emissions

Operation of an FBC/CAES plant will generate air emissions from four major sources: combustion products emanating from the turbine exhaust stack, vent gases from fuel drying, fogging from the reservoir surface and the forced draft cooling towers, and dust emissions from coal and dolomite processing and ash disposal operations. Each of these air emission sources is expected to be controllable to within regulatory values established for the plant site location, as the discussion below indicates.

Air emissions during site construction will be typical of any industrial or commercial project of similar size.

Combustion Products

The PFBC/CAES power plant would burn pulverized coal using dolomite as a sorbent to remove sulfur. A multi-stage cyclone removes particulates from the combustion gas stream prior to entry into the low-pressure turbine. The combustion gas is therefore relatively clean, exhibiting the following properties (volume basis):

CO ₂	5.34%
O ₂	14.7 %
N ₂	77.7 %
HCl	0.001 %
SO ₂	0.013%
H ₂ O	<u>2.246%</u>
Total	100.0%

The emission values for the PFBC/CAES combustion system are within most regional limits for SO₂ (.013 percent vs. the .015 percent limit) and CO (limits not usually established). Because of the low firing temperatures used in the PFBC system, formation of NO_x can be expected to be within allowable limits for a coal burning installation. Concentration of particulates in the exhaust gas stream is also expected to be typical for a coal burning installation, provided that the PFBC cyclone system performs as designed. Monitoring of the exhaust gas stream will probably be necessary to insure adequate removal of particulates.

It is not a simple matter to establish whether the combustion products from a PFBC/CAES plant, or any other fuel burning process, can comply with air emission standards for a selected site. For even though air emission limits as a function of combustion air flow rate are achieved, the degree of application of such regulations as the Prevention of Significant Degradation (PSD) and National Ambient Air Quality Standards (NAAQS) become a function of the total absolute emissions in any specific area. Consideration of these two regulations must therefore be weighed when siting a four-unit PFBC/CAES facility.

Vent Gases

The drying of dolomite and coal results in the discharge of vent gas exhibiting the following volumetric composition:

CO ₂	4.687%
O ₂	13.2 %
N ₂	69.5 %
HCl	0.001%
SO ₂	0.012%
H ₂ O	<u>12.6 %</u>
Total	100.0%

Since the composition of the vent gas is very similar to that of the main turbine exhaust gas, the concerns relative to environmental impact are the same for both the vent and main combustion gases.

Fogging

Water vapor from the plant cooling system and the surface of the compensating reservoir for the air storage cavern will cause a visible fogging to occur in the power plant area. Since the main plant cooling load occurs during the nighttime compression cycle, periods of worst visible fogging will go unnoticed. Daytime fogging will be minimal. The fogging process could cause localized icing of roads during cold weather unless care is taken during plant design to insure that prevailing winds and obstructions do not force plant cooling system exhaust and reservoir fog across roadways.

Processing Dusts and Particulates

Coal and dolomite processing systems will generate dusts during drying, crushing, and transport operations. The coal processing system therefore incorporates filtering systems in the crushing and drying mills to remove dusts before they can escape to the ambient air. Other operations in the plant coal and dolomite handling system, including the main unloading and storage area and the main conveyor system, will generate dusts which cannot be easily controlled. These dust emissions will be minimal, however, and will probably not leave the immediate site area. Intermittent coal unloading and ash disposal operations will also generate dusts in the immediate rail siding area.

Solid Waste Disposal

Solid waste products will be generated on the PFBC/CAES plant site during both facility construction and operation. If these wastes cannot be contained at the site due to either space or environmental restrictions, then suitable arrangements for off-site disposal will have to be made.

Construction Solid Waste

Plant construction will produce large quantities of rock, up to a million cubic yards (765,000 m³) from the excavation of an underground storage cavern, which can be used for site leveling, dike construction, and possibly for concrete aggregate. It is expected that a large percentage of the excavated material will have to be either stored permanently at the site, or trucked away for use or disposal. Potential off-site uses of excess rock include highway and building construction fill, concrete aggregate, and crushed stone for erosion control or decorative application. Off-site disposal could take place without significant environmental impact, since the solid spoil is of contaminant-free quality.

Operational Solid Waste

In the daily operation of the CAES plant, the fluidized bed combustor system will be the largest single source of solid waste. In a normal day of operation in the electric power generation mode, approximately 650 tons (590 t) of coal ash and spent dolomite with the following chemical composition will require disposal:

CaSO ₄	32.5%
MgO	17.0%
CaCO ₃	18.2%
ash and unburned carbon	30.8%
inert mineral matter	<u>1.5%</u>
Total	100.0%

Disposal of the coal ash/spent dolomite waste will utilize methods similar to those being used to dispose of solid wastes from conventional coal-fired power plants. Disposal will be regulated by the Environmental Protection Agency Solid Waste Disposal Act, and the Resource Conservation and Recovery Act (RCRA), both of which serve to protect the areas in the vicinity of the disposal area from environmental damage.

Other solid wastes associated with the operation of the PFBC/CAES plant include defective or worn-out plant components and operation-generated organic matter. The effect of these wastes on the environment and area population is negligible.

Safety

In general, the design, construction, and operation of an PFBC/CAES plant will follow standard power plant practices to assure the safety of both operating personnel and the general public. These practices include proper development of railways, transmission lines, plant entrances, and in-plant systems. However, some aspects of the PFBC/CAES concept do present concerns for safety which are over and above those associated with a conventional power plant.

The presence of a large, pressurized air storage cavern over 2000 feet (600 m) below grade raises concern for surface stability in the immediate plant-site area. Cavern collapse or subgrade faults may cause surface settlement or allow the pressurized air to escape at the surface. Both events raise the possibility of resultant injury to the public, remote as they appear to be from initial analyses.

A second unique safety concern is associated with plant construction. Hard rock air storage cavern construction will require the use of almost 1500 tons (1350 t) of explosives. These explosives will be delivered to the site over public roadways in the site area, an operation which raises concern for the safety of the general public. The use of these explosives, and other underground mining techniques also raises the level of risk to which plant construction personnel are exposed.

Plant Security

Site security at the PFBC/CAES plant will be established through the use of fencing, security lighting, and personnel guarding at plant entrances in a manner similar to conventional fossil-fueled power stations. The plant does not require the strict security common to nuclear power plants since plant sabotage would not present a serious health or safety risk to intruders or the public residing in the site area.

Other Institutional Factors

The location, construction, and operation of an PFBC/CAES plant will also impact the public and the environment in other ways. Although these impacts are minor in comparison to those discussed above and are common to most large plant sitings, they should still be considered as factors affecting PFBC/CAES plant development.

Aesthetic Impact

Impacts include changes in elevation and the addition of man-made landforms and plant structures. The four exhaust stacks, high voltage transmission lines, and cooling tower exhaust will all be visible from land areas surrounding the site. Little could be done to make the plant aesthetically compatible with a rural background.

Surface Water Impact

The operation of the PFBC/CAES plant requires that large amounts of heat be dissipated to the atmosphere. If wet cooling towers are selected as the cooling mechanism, the site will require a large quantity of water as makeup for evaporative and blowdown losses. This water would have to be taken from reservoirs or flowing streams, and may impact site area groundwater, aquatic ecology in water bodies being used, and availability of water in areas downstream from the site point of use. The use of dry cooling towers will decrease water use rates at the expense of plant capital and operating costs.

Economic and Demographic Impacts

The construction of a PFBC/CAES plant would take place over a 7.75-year period, employing a labor force of from 100 to 500 people at all times. Plant construction will require that some special skills be brought into the plant site area for short time periods, creating a need for rental housing within a 20 - 30 mile (30-50 km) radius of the selected site.

PFBC/CAES POWER PLANT ECONOMIC ANALYSIS

An important aspect of the FBC/CAES power plant preconceptual design effort was an estimation of the busbar electric energy cost. A credible estimate of busbar energy cost is essential in order to permit a meaningful comparison with alternative peaking systems and establish the commercial viability of the FBC/CAES concept. Because the FBC/CAES concept is rather novel, reliable estimates for busbar electric energy cost require equally reliable estimates for the owning and operating costs which, for the most part, did not exist.

Accordingly, a prime objective of the preconceptual design effort was to prepare an economic analysis for the entire FBC/CAES system. This economic analysis focused initially on developing equipment cost estimates for the FBC, CAES turbomachinery, and balance of plant. Although equipment cost is a fundamental element in the definition of capital investment, in this period of rapidly escalating prices and high interest rates it is not the only element. The length of the construction period must also be considered so that the effect of escalation and interest on borrowed monies can be included in the total capital investment.

Once the power plant has been constructed and is in operation, appropriate escalation factors must be considered in estimating the contributions to busbar electric energy cost due to fuel consumption, compression energy, and operating and maintenance expenses. Although these expenses vary over the lifetime of the plant, an accepted approach to handling them is: 1) to assume that they remain constant (level) in real dollars throughout the lifetime of the power plant with the only change in dollar magnitude of the expenses resulting from escalation; then 2) to discount the series of expenses to a base year to produce a present value of the levelized costs. Levelization of costs must be done not only for the FBC/CAES power plant but, if comparisons with alternative peaking power plant options are to be performed, it must also be done on a consistent basis for the alternative power plant systems.

The results of the economic analysis are discussed in this section, starting with the direct capital costs and progressing through the indirect and operating costs to the resulting busbar energy cost estimates for the FBC/CAES plant. Finally, cost comparisons are presented for selected alternative power plants.

Direct Capital Costs

A capital cost estimate was prepared for a four-unit, hydraulically-compensated, hard-rock mined CAES plant using fluidized bed coal combustion

as the energy source during the power generation mode. The plant is nominally rated at 876-MW output (total for four units), with a total air storage capacity equivalent of approximately 7550 MW-hr at the turbine design flow rate. The important plant operating characteristics are summarized in Table 4-12.

The capital cost estimates were prepared using information received from several sources. Manufacturers' budgetary quotations were used whenever available, with engineering estimates used in those areas where quotes could not be provided. The cost estimates are presented in mid-1979 dollars for a plant constructed on an east coast site and include shipping and erection costs. The capital investment requirement in January 1980 dollars then was developed, based upon a construction time schedule for the plant and the economic factors contained in Appendix B.

Major Equipment Costs

Budgetary cost estimates were prepared for the FBC and CAES turbomachinery based upon the preconceptual designs presented in previous chapters.

Fluid Bed Combustor

The estimated capital cost of the combustor plant equipment (see Fig. 4-4) in July 1979 money values is given in Table 4-13. The cost of \$195.9 million refers to the four parallel units on the site within the defined battery limits (see Scope of Design Effort section for the FBC). The capital cost estimates were prepared using quoted and manufacturers' estimated prices for similar equipment used elsewhere, cost estimations using similar equipment of different size or capacity, and in some cases, engineering judgment.

It should be reiterated here that the limited scope of the present study has not permitted the design concept of the fluid bed combustor to be optimized to meet the specific requirements of CAES applications. It is, therefore, expected that more detailed studies would result in improved designs and lower costs.

CAES Turbomachinery

The total estimated selling price for the CAES turbomachinery (see Fig. 4-7) is provided in Table 4-14. The estimate of \$63.1 million refers to all four units. The price estimate for the modified low-pressure components and their supports are based upon producing 50 units of a single standard design. The importance of this assumption should be emphasized. As was shown in Ref. 4-1, production of fewer units as was assumed in Ref. 4-4, such as 20 or 10 units, could easily raise the selling price by a factor of two or four, respectively, as the development costs are amortized over fewer units. All other

TABLE 4-12

KEY PFBC/CAES POWER PLANT CHARACTERISTICS

Turbine output, MW - 4 units	876.4
Compressor input, MW - 4 units	799.6
Turbine output, MW-1 unit	219.1
Compressor input, MW-1 unit	199.9
Generation heat rate, Btu/kWh	4538
Compression heat rate, Btu/kWh	9095
Round trip heat rate, Btu/kWh	13633
Annual utilization, hr/yr	2190
Capacity factor	0.25
Storage pressure, atm (bar)	71.2 (71.2)
Expansion turbine temperature, F(C)	507.5 (264.2)
Low-pressure turbine temperature, F(C)	1562 (850)
FBC bed temperature, F(C)	1562 (850)
FBC bed pressure, atm (bar)	15.0 (15.0)

TABLE 4-13

CAES FLUID BED COMBUSTOR PLANT EQUIPMENT COST
Budgetary Estimate
July 1979\$
4 Units

<u>Item</u>	<u>Cost, 10⁶\$</u>
Coal and dolomite handling preparation and feeding	38.3
Combustors and gas cleanup	112.9
Ash and spent dolomite handling	19.6
Controls and instrumentation	<u>8.8</u>
Subtotal	179.6
Building and structures	<u>16.3</u>
Total	195.9

TABLE 4-14

CAES TURBOMACHINERY EQUIPMENT COST
Budgetary Estimate
July 1979\$
4 Units

<u>Item</u>	<u>Cost, 10⁶ \$</u>
Low-pressure compressors	13.7
Booster compressor	6.6
Turbines	16.1
Supports	0.1
Clutches	1.2
Motor/Generator	<u>25.4</u>
Total	63.1

components (booster compressor, expansion turbine, clutches, and motor/generator) are based upon vendor quotes for their machinery modified for CAES applications. Items not included in the component estimate are freight, foundation, and assembly, i.e., they are FOB factory.

The estimated selling price in July 1979 dollars for the four low-pressure compressor units (see Fig. 4-8) is estimated to be \$13.7 million. This estimate is based on Refs. 4-1 through 4-3 escalated to July 1979 dollars and includes all basic compressor components plus the casing, the inlet plenum preceding the first stage of compression, the interstage exit and readmission plenum, and the exhaust plenum. In addition, the clutch housing and the shaft between the compressor and the motor/generator are incorporated in this estimate. Not included, however, are the special clutches and supports.

The estimated selling price of \$6.6 million for the four booster compressor units (Fig. 4-9) is based upon the estimates in Refs. 4-1 through 4-3, escalated to July 1979 dollars. It includes the basic components plus the casing, the inlet plenum preceding the first stage of compression, the interstage exit and readmission plenum, and the exhaust plenum. The booster compressor clutch is not included.

The estimated selling price of \$16.1 million for the four expansion turbine and low-pressure turbine units (Fig. 4-10) are also based upon the estimates in Refs. 4-1 through 4-3, escalated to July 1979 dollars. It includes the basic requirements plus the casings, the inlet and exhaust plenums, and the plenum between the expansion and low-pressure turbines. It does not include the supports or clutches.

An allowance of \$0.1 million has been provided for the pedestal supports.

The estimated selling price of \$1.2 million for the four sets of three clutches are based upon an estimate in Ref. 4-2 escalated to July 1979 dollars. Of the total, the turbine clutches account for approximately \$0.5 million and the booster compressor clutches and low-pressure compressor clutches \$0.35 million each.

The estimated selling price for the motor/generator of \$25.4 million is based upon an estimate in Ref. 4-3, escalated to July 1979 dollars.

Plant Cost Estimate

A summary of all direct costs for constructing a four-unit FBC-augmented CAES plant is presented in Table 4-15. The capital cost estimate of \$458.9

TABLE 4-15

PFBC/CAES POWER PLANT TOTAL DIRECT COST SUMMARY

Budgetary Estimate

July 1979\$

4 Units

876.4 MW

<u>Account</u>	<u>Description</u>	<u>Cost, 10⁶\$</u>
10	Construction preparation	11.4
20	Surface facilities	13.5
30	Storage system	58.3
40	Generation/compression system	115.5
50	Mechanical systems	20.6
60	Switchyard	27.8
70	Plant electrical systems	15.3
80	Coal handling and combustion	<u>196.5</u>
	Total direct cost	458.9
	Specific capital cost (\$/kW)	523.6

million (\$523.6/kW) was prepared using a system of accounts similar to that used by the Federal Power Commission. The estimate is presented in account format (Table 4-16) with a description of the composition for each account code appearing below. For a more detailed description of the components included in the balance-of-plant capital cost estimate, refer to the Balance of Plant Preconceptual Design section of the report.

Construction Preparation - Code 10

This code includes the purchase of approximately 500 acres (202 hectares) of land, residences, and rights-of-way, preparation of site surface, and provision of construction utilities. The reference site is zoned as rural-residential, but possesses the necessary rail access to the site. Costs are derived from Ref. 4-4.

Surface Facilities - Code 20

This code includes all surface buildings and structures exclusive of the FBC plant (see Fig. 4-4) which are included in the following Code 80 account, and the equipment within these items which is necessary to maintain the major mechanical and electrical equipment. Major items included within this code are:

- 1) turbomachinery buildings and concrete pedestals;
- 2) 125-ton cranes; and
- 3) control and administration building.

Costs are taken from Ref. 4-4.

Storage System - Code 30

This code includes the surface and underground facilities and structures for a hard-rock, water-compensated compressed air storage system. Major items represented are:

- 1) surface compensating reservoir;
- 2) 4-foot diameter main air shaft;
- 3) 20-foot diameter main water shaft;
- 4) water control gates and intake structure; and
- 5) underground cavern

Costs are taken from Ref. 4-4.

Generation/Compression System - Code 40

This code includes the major turbomachinery components required for air compression and power generation, (Fig. 4-7) as well as the auxiliary equipment

TABLE 4-16

PFBC/CAES POWER PLANT TOTAL DIRECT COST
 Budgetary Estimate
 July 1979\$
 4 Units
 876.4 MW

<u>Account</u>	<u>Description</u>	<u>Quantity</u>	<u>Cost, 10⁶\$</u>
10	<u>Construction Preparation</u>		<u>11.396</u>
11	Land and land rights	-	4.756
12	Site preparation	-	6.640
20	<u>Surface Facilities</u>		<u>13.504</u>
21	Turbine buildings	4	8.148
22	Control building	1	1.000
23	Miscellaneous building	-	0.837
24	Final grading	-	0.100
25	Permanent roads	-	0.300
26	Site utilities	-	3.119
30	<u>Storage System</u>		<u>58.263</u>
31	Compensating reservoir	1	6.504
32	Intake structure	1	0.432
33	Water shaft	1	9.252
34	Air shaft	1	10.577
35	Underground cavern	1	31.498
40	<u>Generation/Compression System</u>		<u>115.494</u>
41	Turbomachinery system	4	70.100
42	Intercoolers/recuperator	4	24.464
43	Primary ducting	4	4.262
44	Main air piping	4	16.668

TABLE 4-16 (Cont'd)

FBC/CAES POWER PLANT TOTAL DIRECT COST
 Budgetary Estimate
 July 1979\$
 4 Units
 876.4 MW

<u>Account</u>	<u>Description</u>	<u>Quantity</u>	<u>Cost, 10⁶\$</u>
50	<u>Mechanical Systems</u>		<u>20.616</u>
51	Fuel oil system	1	1.102
52	Cooling towers	4	13.244
53	Water supply system	1	1.423
54	Waste treatment system	1	2.399
55	Fire protection	-	1.644
56	Plant air	-	0.157
57	Miscellaneous services	-	0.590
58	Bulk storage	-	0.057
60	Switchyard		<u>27.768</u>
61	500-kV Power Equipment	-	12.044
62	500/13.8-kV station	-	1.000
63	13.8-kV service	-	0.340
64	Switchyard control	-	1.150
65	Other electrical systems	-	0.469
66	Civil works	-	6.120
67	Buildings	-	0.410
68	Static starting system	1	5.000
69	500-kV transmission line	-	1.235
70	Plant Electrical Systems		<u>15.298</u>
71	Generator/motor bus	4	1.044
72	Main power transformers	4	5.380
73	Raceways/trenching	-	0.650
74	Cabling	-	3.880
75	Unit substations	-	1.520
76	Motor control centers	-	0.528
77	DC systems	-	0.936
78	Other electrical systems	-	0.790
79	Backup power systems	-	0.570

TABLE 4-16 (Cont'd)

FBC/CAES POWER PLANT TOTAL DIRECT COST
 Budgetary Estimate
 July 1979\$
 4 Units
 876.4 MW

<u>Account</u>	<u>Description</u>	<u>Quantity</u>	<u>Cost, 10⁶\$</u>
80	<u>FBC Combustion System</u>		<u>196.521</u>
81	Coal & dolomite delivery storage, preparation, and feeding	-	38.333
82	FBC Coal combustors	4	112.916
83	Ash and spent dolomite handling	-	19.580
84	Instrumentation	-	8.750
85	Buildings	-	16.330
86	Combustor piping	4	0.412
87	Expansion joints	8	0.200

required to support the turbomachinery and thermodynamic requirements of the power generation and compression cycles. Costs for the power train (turbines, compressors, motor/generator, and power train auxiliaries) are taken from Refs. 4-1 through 4-3, escalated to mid-1979 dollars. Clutch costs are taken from Ref. 4-2, with appropriate escalation. The \$7 million difference existing between the \$63.1 million estimate of Table 4-14 and the \$70.1 million estimate of Table 4-16 for the turbomachinery system (Account 41) results from shipping plus installation and the inclusion of the turbomachinery control system in the latter estimate. Costs for compressor intercoolers, exhaust gas recuperator, ducting, controls, and main air piping are based on modifications to costs developed in Ref. 4-4.

Mechanical Systems - Code 50

This code includes those mechanical systems which indirectly support the operation of the CAES turbomachinery train. Costs for these items are taken from Ref. 4-4. Fuel oil, plant cooling water supply, waste treatment, fire protection, plant air, miscellaneous services, and bulk storage are addressed in this code.

Switchyard - Code 60

A transmission voltage of 500 kV has been selected for this plant to facilitate cost estimating based on costs established in Ref. 4-4. Although based on Ref. 4-4, the switchyard costs are approximately \$10 million less for the PFBC/CAES power plant. This is due to the complex site-specific nature of the Ref. 4-4 switchyard being located along an existing 500-kV intertie. This code, therefore, includes all power transmission equipment at 500-kV, as well as 13.8-kV station service equipment, switchyard control, and the motor static starting system.

Plant Electrical Systems - Code 70

This code includes those electrical items connecting the 13.8-kV generator with the 500-kV switchyard. Also included are all electrical systems being fed from the 13.8-kV station service and the emergency power generation equipment for the plant. Costs for equipment presented within this code are taken from Ref. 4-4.

FBC Combustion System - Code 80

This code addresses the coal processing and combustion equipment (Fig. 4-4), plus combustor/turbine piping. Costs for coal and dolomite delivery, storage, preparation and feeding, coal combustors, ash and spent dolomite handling, and controls were developed during the study. Combustor/turbine piping costs are engineering estimates.

Indirect Capital Costs

Once the total direct costs have been evaluated and a construction time schedule adopted, it is possible to estimate the indirect capital costs associated with:

- 1) contingency allowance;
- 2) engineering and management fees;
- 3) escalation of the total direct cost prior to construction;
- 4) escalation of the cost of equipment purchased during construction above the original total direct cost estimate; and
- 5) interest on the borrowed monies incurred during construction.

The total direct cost budgetary estimate of Table 4-15 is based on July 1979 dollars. According to Ref. 4-6, all CAES power plant systems should be costed based upon a January 1990 date of commercialization. During the study, it became apparent quite early that installation of the surface equipment rather than the storage facility would be the pacing item. Thus the construction timescale for the surface equipment calling for a 7.75-year construction period was adopted. This means an April 1982 construction start date and requires escalation of the July 1979 total direct cost estimate to a total direct cost for that date. A six-percent escalation rate was used (Ref. 4-6).

The engineering plus management fee and contingency allowances are each typically quoted as between 10 and 20 percent. For the purposes of this study a factor of 15 percent was chosen for both the engineering and management fee and contingency allowance. As directed in Ref. 4-6, the contingency allowance was applied to the total direct cost at the time of construction commencement while the engineering and management fee was applied to the sum of the total direct cost and contingency fee. It should be noted, however, that the resulting sum would be the same regardless of the order of application of the two factors because of their multiplicative application.

The escalation during construction and interest during construction was estimated by applying the factor provided in Ref. 4-6 to the total direct cost at the time of construction commencement. These factors are based upon a six-percent escalation rate during construction, a ten-percent cost of capital, and a S-shaped cash flow curve during construction with half the costs being incurred at sixty-percent of the construction period. However, because the factors supplied in Ref. 4-6 did not extend beyond seven years; it was necessary to utilize the methodology developed by UTRC as part of Ref. 4-7 to obtain the appropriate factors for 7.75 years. In addition, the methodology of Ref. 4-6

assumed that no revenue was provided during the construction period. Because the four-unit PFBC/CAES power plant is constructed in stages, three of the four units will actually be generating power and earning revenue prior to construction of the total plant. A rigorous treatment of this effect was beyond the scope of this study. However, as part of the sensitivity analysis the effect of early generation on total capital investment was approximated by varying the construction period.

The result of this procedure is displayed in Table 4-17. Because the total capital investment is stated in January 1990 dollars, it is necessary to discount it back to January 1980 dollars. Six percent was used for this discounting (Ref. 4-6). This results in a total capital investment of \$706.7 million for the 876.4 MW power plant which corresponds to a plant cost of \$806.4/kW.

Operating Costs

Three other cost items are necessary to estimate busbar electric energy cost: fuel cost, compression energy cost, and operating labor plus maintenance costs.

Fuel Costs

The cost utilized for the coal was that recommended in Ref. 4-6. This value is \$1.34/10⁶ Btu in January 1980 dollars. However, because the plant uses sizable quantities of dolomite as a sorbent, it was necessary to include an allowance for this consumable.

Exact costing of the dolomite proved difficult. At the source, it costs about \$1.50 to 2.00/ton. Conveying it to a nearby facility could double this figure. From this point on, it incurs a transportation charge which depends on the distance and an unloading charge at the destination which depends on the unloading facility. Costing of the spent dolomite and ash disposal also proved to be equally inexact. A beneficial application of the waste could reduce the disposal fee from a typical value of \$5/ton when the waste is used solely as landfill.

Weighing these factors led to the application of a \$10/ton dolomite fee to cover both feed and waste disposal. Because of the fixed mass flow rate between the calcium in the sorbent and the sulfur in the coal, it was possible to calculate the dolomite cost per 10⁶ Btu of coal used. This value is \$0.12/10⁶ Btu of coal feed. Thus, the coal plus sorbent plus waste disposal cost was taken to be \$1.46/10⁶ Btu of coal feed in January 1980 dollars.

TABLE 4-17

FBC/CAES POWER PLANT
CAPITAL INVESTMENT SUMMARY
Start Construction - April 1982
Commercial Operation-January 1990
4 Units
876.4 MW

<u>Description</u>	<u>Cost, 10⁶\$</u>
Total direct cost (July 1979\$)	458.9
Escalation prior to construction (April 1982, @ 6%/yr)	<u>79.7</u>
Subtotal I	538.6
Contingency (@ 15% of subtotal I)	<u>80.8</u>
Subtotal II	619.4
Engineering plus management (@ 15% of subtotal I)	<u>92.9</u>
Subtotal III	712.3
Escalation during construction (@ 6%/yr applied to subtotal III)	215.1
Interest during construction (@ 10%/yr applied to subtotal III)	<u>338.3</u>
Total capital investment as of January, 1990	1,265.7
Total capital investment (January 1980, @ 6%/yr)	706.7
Specific capital cost (\$/kW)	806.4

In addition to coal and dolomite, the PFBC/CAES power plant will also consume No. 2 fuel oil to reheat the bed after periods of shutdown. Estimating the magnitude of this consumption was beyond the scope of the study program because it involves the establishment of a shutdown/startup sequence during which the plant would be operating at reduced power. However, it is anticipated that the auxiliary fuel requirement should be minor adding no more than five percent to the generation heat rate.

Compression Energy Costs

The compression energy cost was taken to be 11.4 mills/kWh in January 1980 dollars as recommended in Ref. 4-6. This cost is based upon the incremental generating cost for power using a capacity mix of one-third nuclear and two-thirds coal.

Operating Labor Plus Maintenance Costs

The operating plus maintenance costs were developed in three steps. First the manpower requirements for the FBC plant were estimated. These requirements were then matched with those of the CAES plant to eliminate overlap. Lastly, the annual expense was calculated for these requirements.

Operating Labor Expenses

The annual expenses associated with operating the power plant are listed in Table 4-18. The annual expense for each labor grade was calculated based upon the estimation provided in Ref. 4-2 for comparable labor grades escalated to January 1980 dollars. As was noted in that reference, it should be cautioned that the actual labor rates are significantly affected by company and union policies and local ordinances. To arrive at the annual expense estimate, it was assumed that the FBC plant would require only one full shift per day during the peak-load period. The remainder of the plant was assumed to operate on a 24-hour basis.

Maintenance Expenses

The annual expenses associated with maintaining the power plant are listed in Table 4-19. Again, the labor grade expenses were based on Ref. 4-2, escalated to January 1980. The majority of the materials and supplies are related to the FBC plant operation. The remainder of the material and supplies requirement was derived from Ref. 4-2.

Busbar Electric Energy Cost

The calculation of the busbar electric energy cost was performed using the levelized costing methodology described in Ref. 4-6 and summarized in Appendix B. The values used as economic inputs are listed in Table 4-20.

TABLE 4-18

PFBC/CAES POWER PLANT
ANNUAL OPERATING LABOR EXPENSE
Budgetary Estimate
July 1979\$
4 Units
876.4 MW

<u>Title</u>	<u>Requirement</u>	<u>Expense, \$</u>
Plant supervisor	4	240,000
Operator	4	200,000
Operator's helper	4	160,000
Plant engineering	4	200,000
Engineering aides	2	66,000
Utility staff*	-	66,000
Feed handling operator	6	300,000
Feed preparing operator	6	300,000
Combustor operator	8	400,000
General yardmen	4	132,000
Plumbers	2	100,000
Secretary, clerical	2	66,000
Guards	8	264,000
Janitorial	16	528,000
Groundskeeping	<u>13</u>	<u>429,000</u>
	83	3,451,000

*i.e., extra men for overhaul work

TABLE 4-19

PFBC/CAES POWER PLANT
ANNUAL MAINTENANCE EXPENSE
Budgetary Estimate
July 1979\$
4 Units
876.4 MW

<u>Title</u>	<u>Requirement</u>	<u>Expense, \$</u>
Mechanic	13	650,000
Mechanic's helper	13	520,000
Electrician	13	650,000
Electrician's helper	13	520,000
Laborers	10	330,000
Foremen	3	120,000
Engineering	3	150,000
Engineering aides	<u>2</u>	<u>66,000</u>
	69	\$3,006,000
Materials & supplies		<u>\$5,506,000</u>
		<u>\$8,512,000</u>

TABLE 4-20

INPUTS FOR PFBC/CAES COST ANALYSIS

Base Case

<u>Parameter</u>	<u>Description</u>	<u>Value</u>
CI_x	Capital investment in January 1980\$ (Table 4-15 value of \$458.857 x 10 ⁶ escalated @ 6% for 0.5 years)	472.422 x 10 ⁶
sc	Year of start of construction	April 1982
co-sc	Construction period in years	7.75
f_{EDC}	Escalation during construction factor	0.302 ⁽¹⁾
f_{IDC}	Interest during construction factor	0.475 ⁽¹⁾
f_{EM}	Engineering & management factor	0.15
f_{CONT}	Contingency factor	0.15
FP_x	Price of fuel in \$/10 ⁶ Btu (includes \$0.12/10 ⁶ Btu allowance for dolomite)	1.46
FHR	Fuel heat rate (Btu/kWh)	4538
CF	Capacity factor (2190 hrs/yr)	0.25
RP	Rated power in MW	876.4
N	Power plant lifetime in years	30
CEP_x	Compression energy cost (mills/kWh)	11.4
EER	Compression energy divided by generation energy (kWh/kWh)	199.9/219.1
OMC_x	Operating and maintenance cost in January 1980\$ (Values from Table 4-18 and 4-19 escalated @ 6% for 0.5 years)	12.316 x 10 ⁶
FCR	Fixed charge rate	0.18

(1) Derived from Ref. 4-7 work

The resulting busbar energy cost estimate is depicted in Fig. 4-17. Note that the dominant cost item is the capital investment which accounts for about 57 percent of the total power cost. The next largest contribution is from the compression energy which accounts for about 20 percent. Fuel and operation and maintenance contribute about equal amounts, 13 and 10 percent, respectively.

For comparison, busbar costs estimated using the traditional nonlevelized method are also illustrated. Of obvious note is the difference in magnitude between the two estimates. The capitalization contribution is the same because they are calculated in the same manner. The operating costs are considerably higher for the levelized estimate, however, because they incorporate the effect of escalation during the lifetime of the power plant.

Economic Comparison with Alternative Power Systems

Although it was beyond the scope of the present program, it became apparent that terminating the economic investigation without a comparison among alternative systems on a consistent basis would produce an incomplete evaluation. Consequently, it was decided to conduct a preliminary screening analysis wherein the sensitivity of the costs to variations in key parameters could be assessed. The alternative systems chosen for comparison were all oil-fired -- a CAES power plant, a gas turbine power plant, and a combined-cycle power plant. The base values for the economic parameters are shown in Table 4-21.

The oil-fired CAES values are taken from Ref. 4-3 and represent the Shultz Mountain installation. Both the gas turbine and combined cycle values are taken from Ref. 4-8.

Method of Comparison

For the purpose of this preliminary economic comparison, simple screening curves were used. Screening curves are graphs of the levelized busbar electric power cost (\$/kWyr or the equivalent) versus hours per year of operation or capacity factor for competing types of new power generation equipment. The annual levelized costs for each power plant type are calculated as the sum of the annualized capital cost (the capital cost times an annual cost recovery factor), the operating plus maintenance cost, the fuel cost (the heat rate times the price of fuel), and for CAES plants, the compression energy costs. The result is a series of comparative curves with the plant producing lowest cost power (at the busbar) on the bottom. The bottom-line plant will vary according to the number of hours the plant is run per year. A low capital cost plant will produce lowest cost power for a small number of hours per year while a low fuel cost plant will produce lowest cost power for a large number of hours per year. Generally speaking, a plant must be at or near the bottom line in order to be seriously considered by a utility. Additional factors such as environmental impact, operating flexibility, fuel supply, and siting modify a strict bottom-line decision. In summary though, a screening curve provides a preliminary estimate of whether a new power plant is economically competitive with alternative types of generation.

BUSBAR ENERGY COST ESTIMATE

BASE CASE
0.25 CAPACITY FACTOR
1980 DOLLARS

TABLE 4-21

INPUTS FOR PFBC/CAES POWER PLANT COST COMPARISON WITH ALTERNATIVE POWER PLANTS

Parameter ⁽¹⁾	Oil-Fired ⁽²⁾ CAES	Oil-Fired (3) Gas Turbine	Oil-Fired (3) Combined Cycle
CI_x (10^6 1980\$)	181.715	10.993	68.767
sc (year)	1983	1988	1987
co-sc (years)	7	2	3
f_{EDC}	.275	.069 ⁽⁴⁾	.109
f_{IDC}	.401	.091 ⁽⁴⁾	.138
f_{EM}	0.15	0.15	0.15
f_{CONT}	0.15	0.15	0.15
FP_x (\$/ 10^6 Btu)	5.24	5.24	5.24
FHR (Btu/kWhr)	4383	11,500	8,400
RP (MW)	7605	75	250
N (years)	30	20	30
CEP_x (mills/kWhr)	11.4	--	--
EER (kWhr/kWhr)	0.835	--	--
OMC_x (10^6 1980\$)	4.317	0.290	2.044
FCR	0.18	0.19	0.18

(1) See Table 4-20 for definitions

(2) Values from Ref. 4-3 with dollar values escalated to January 1980 dollars

(3) Values from Ref. 4-8 with dollar values escalated to January 1980 dollars

(4) Derived from Ref. 4-7 work

Base Case

The screening curves for the base case are displayed in Fig. 4-18. From this comparison, it appears that the coal-fired PFBC/CAES system is quite competitive against the oil-fired alternatives. The PFBC/CAES power plant does have the highest capital cost (corresponding to y-intercept), but it also has the lowest operating cost (corresponding to slope of line) due to its lower fuel cost and low fuel heat rate.

For capacity factors below about seven percent, the oil-fired gas turbine produces the lowest cost power. Between seven percent (615 hr/yr) and about twenty-two percent (1925 hr/yr), the oil-fired CAES plant produces the lowest cost power. Above twenty-two percent capacity factor (1925 hr/yr) the PFBC/CAES plant emerges as the lowest cost alternative. Even at capacity factors down to thirteen percent (1140 hr/yr), the PFBC/CAES system is very competitive with gas turbines.

Sensitivity Analysis

While contemplating the significance of the comparison in Fig. 4-18, it must be recognized that several of the key economic parameters used in the analysis contain an inherent and unquantifiable degree of uncertainty. Unfortunately, it is impossible to predict precisely what future equipment, labor, fuel, and off-peak electricity costs will be. Accordingly, a series of selected parametric comparisons were made to determine the sensitivity of the economic comparisons to the value of these key parameters.

Parametric Variations

The key FBC/CAES parameters chosen for the sensitivity studies are listed in Table 4-22. The capacity factor was varied over a fairly wide range (from zero to fifty percent) with the knowledge that, in all probability, the utility load characteristics would not permit the PFBC/CAES system to be utilized over the full range. Most of the other parameters were varied ± 20 percent relative to their base value (see Tables 4-20 and 4-21). The compression energy was allowed to vary over a wider range because of the wide range of incremental electric energy costs actually experienced (Ref. 4-2). Generally, the lower the capacity factor, the greater the probability will be that the compression energy costs would be low because the shorter compression period could more easily be made to correspond to a period of low incremental electric energy cost. Obviously other parameters could also have been varied, such as inflation, escalation, cost of capital, etc.; however, these parameters are more likely to be dictated by prevailing economic conditions than the parameters which were varied.

FIG. 4-18

BASE CASE ANNUAL COST COMPARISON

TABLE 4-22

SCREENING ANALYSIS PARAMETER VARIATIONS

<u>Parameter</u>	<u>FBC/CAES Variation⁽¹⁾</u>	<u>Alternative System Variation</u>
Capacity factor	0 → .5	0 → .5
Fuel price	.8 → 1.2 ⁽¹⁾	1
Contingency	.8 → 1.2 ⁽¹⁾	
Engineering & management		
Operating & maintenance	.8 → 1.2 ⁽¹⁾	1
Compression energy	.5 → 2.0 ⁽¹⁾	1
Capital investment	.8 → 1.2 ⁽¹⁾	1
Construction start date	1985 → 1985.25	None

⁽¹⁾ These variations are expressed as a multiplicative factor to be applied to the base case values (see Tables 4-20 and 4-21).

Capital Cost Comparison

The effect of a capital cost variation of ± 20 percent ($\pm \$161.3/\text{kW}$) is shown in Fig. 4-19. The three parallel lines which result indicate the different total capital investments. Because capital cost is the dominant contribution to busbar cost (refer to Fig. 4-17), a change of twenty percent in capital cost would be expected to significantly alter the economic comparison. An increase of twenty percent in cost pushes the crossover point with an oil-fired CAES out to twenty-nine percent capacity factor (2540 hr/yr), whereas a comparable decrease in cost brings the crossover down to fifteen percent capacity factor (1315 hr/yr).

Capital cost variations can be obtained by variations of several of the parameters listed in Table 4-22 (i.e., contingency, engineering plus management, and construction start date to reflect revenue generation during construction period). The effect of these variations were also calculated. In all cases, the variation from the base case were less than that displayed in Fig. 4-19, consequently, separate illustrations were not prepared. Obviously, because the fixed charge rate is multiplied by capital investment, the illustration can also be interpreted as the effect resulting from a ± 20 -percent variation in fixed charge rate or a ± 20 -percent variation in the product of fixed charge rate times capital investment.

Coal Cost Comparison

The effect of a ± 20 -percent coal cost variation is shown in Fig. 4-20. The oil cost for the alternative plants were assumed to remain unchanged. The crossover point of the FBC/CAES power plant with the next best alternative (oil-fired CAES) varies by ± 1 points in capacity factor (± 85 hr/yr). Hence, the annual costs are substantially less sensitive to coal cost variations than capital cost variations. The real sensitivity may be even less than indicated in Fig. 4-20 because an increase in coal price would undoubtedly be accompanied by an increase in oil price.

Multiple interpretations of the fuel-cost sensitivity are possible because the fuel cost is a function of the product of fuel cost times fuel heat rate. Consequently, Fig. 4-20 could be considered to represent a ± 20 -percent variation in either of the two multiplicands with the other held constant, or of the product of the two.

FIG. 4-19

CAPITAL COST VARIATION EFFECT ON ANNUAL COST COMPARISON

COAL COST VARIATION EFFECT ON ANNUAL COST COMPARISON

Compression Energy Cost Comparison

The effect of a large variation in compression energy cost is shown in Fig. 4-21. This variation was applied to both the coal-fired FBC/CAES power plant and the oil-fired CAES plant. Although this produces a considerable variation in the cost for both power plant systems, the capacity factor at the intersection between the two lines for comparable compression energy costs does not experience a significant variation in that it remains around twenty-two percent (1925 hr/yr).

This illustration also has a multiple interpretation with compression energy and the compressor input power being the multiplicands. The term electric energy ratio ($\text{kWh}_{\text{in}}/\text{kWh}_{\text{out}}$) cannot be considered to be a multiplicand because a change in output energy would affect all costs, not just compression energy cost.

Operating Plus Maintenance Cost Comparison

The variation of annual cost with a ± 20 percent variation in O&M cost was also calculated. The variation was so small as to be indistinguishable from the base case when plotted with the scale used, consequently, another illustration is not presented.

Discussion of Economic Results

The simple economic comparisons presented in the preceeding sections indicate that coal-fired FBC/CAES systems should be economically competitive against oil-fired CAES gas turbines and combined-cycle power plants. Although major uncertainties in the estimated capital cost estimates could have a significant impact on the comparisons, corresponding uncertainties in fuel and compression energy costs do not.

It must be noted that the economic comparison presented herein considers only a few selected alternative power plants. A comprehensive analysis should include other power plant options such as: heavy oil-fired cycling steam plants and pulverized coal-fired cycling steam plants with flue gas desulfurization. Other energy storage plants such as underground pumped hydro and electric storage batteries should also be included. Finally, utility-specific production cost models should be used to identify the compression energy costs and the annual utilization which could be expected under actual utility operating conditions.

COMPRESSION ENERGY COST VARIATION EFFECT ON ANNUAL COST COMPARISON

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Chapter 5

Part-Load and Load-Following Capability

Part-Load and Load-Following Capability

All power plants must have some degree of load-following capability. This implies the ability to operate in steady-state and transient part-load conditions. The open-bed PFBC/CAES power plant system would achieve part-load operation by concurrently and proportionately varying the mass flow of both the air and coal to the PFBC to produce a varying thermal output. Operation in this manner will also:

- 1) produce a constant bed temperature and, therefore, a constant low-pressure turbine inlet temperature because the air and fuel flows remain proportional,
- 2) provide a bed pressure that is directly proportional to the flow through the bed because the low-pressure turbine nozzle area is fixed resulting in choked flow, and
- 3) yield a constant superficial velocity in the bed over the entire load range because the flow rates are proportional to pressure and the bed temperature remains constant.

Despite this recognized procedure for operating an open-bed PFBC, there were some reservations about the part-load and load-following capabilities of the PFBC system. These reservations were based partly on the absence of documented evidence regarding the part-load capabilities of open-bed PFBC designs and partly on the extrapolation of the characteristics of PFBC designs for combined cycles which incorporate immersed tubes and have severe part-load constraints associated with the massive amount of tube material immersed in the bed. Consequently, an investigation was initiated to address these reservations.

This investigation had three tasks: 1) identify the utility load-following requirements, 2) provide a preliminary assessment of the part-load characteristics of open-bed PFBC/CAES power plants, and 3) evaluate alternative load-following concepts wherein the PFBC could be operated continuously. For the first task, current utility practices for dealing with system load variations were explored to develop an understanding of how utility system requirements affect the response goals of individual plants. Since the second task was very preliminary in nature, the principal effort involved acquiring an appreciation of the steady-state, part-load performance associated with open-bed PFBC/CAES power plants. It was then possible to provide a preliminary assessment of the factors affecting transient operation and to estimate the general order of magnitude of response rate. The last task required a technical and cost evaluation of alternative PFBC/CAES operating modes and cycle configurations which would not require the PFBC to operate at part load, but which would still permit variations in power plant output.

These three topics - utility requirements, PFBC/CAES part-load characteristics, and alternative load-following PFBC/CAES system concepts - are covered in this chapter.

UTILITY SYSTEM REQUIREMENTS

An important characteristic of any peak- or intermediate-load generating unit, such as a PFBC/CAES power plant, is the ability to respond to rapid changes in load demand. Generally the more responsive the unit, the more beneficial it is to the utility system. Before examining the part-load and load-following characteristics of the PFBC/CAES plant, it is desirable to review the demands that may be placed on a generating unit when operating in a electric utility power system. Because no two utility systems are alike, however, it is not possible to establish firm specifications for response requirements. Nonetheless, an appreciation of the generation control process can be helpful in understanding the significance of the requirements that do exist.

System Load-Following Control

Typically, when the electric generator in a power plant is connected to the utility distribution network, it will respond to frequency deviation and to load change commands from the centralized automatic generation control (AGC). These two inputs are shown schematically in Fig. 5-1. Also shown in this diagram are two key internal controls that can affect dynamic response. Load changes due to frequency error (which are sensed at the generator) are referred to as primary control actions while those from the AGC are considered secondary control actions. In addition to these system operating functions, plant control systems exist to govern operation during startup, synchronization, and failures within the plant.

During normal operation, the plant must respond to meet the needs of the normal variations in load demand and to changes in frequency caused by sudden changes in demand or in generating capability. These disturbances or rapid load changes cause a nearly instantaneous change in the electric and magnetic fields of generators in the system which are followed in a short period of time (seconds to minutes) by the longer term response (speed change) of the rotating elements of the generating sources and loads. The faster electrical transients are very important in system control. In considering prime mover control, however, only the longer term response is of interest.

TYPICAL GENERATION CONTROL SCHEMATIC

FIG. 5-1

Interconnections

One of the most effective approaches available to a utility for handling disturbances is to provide for interconnections (tie lines) with other utilities. For discussion purposes, a diagram representing three interconnected automatic generation control areas is shown in Fig. 5-2. Each of the three areas consists of a number of generation sources (which may belong to more than one utility) that supply the loads in their area. These loads vary on daily, weekly, and seasonal cycles and, while the loads are not controlled by the utility (except in rare instances where load must be dropped to maintain system integrity following a severe disturbance), they are quite predictable. Because of this predictability, generating units within each area can be scheduled to match output to the load with fairly good accuracy. When unexpected disturbances in an area occur either in the load or generating equipment, however, a relatively large imbalance can occur between the load demand and the generator output. When this happens the interconnections perform an important function. Because of these interconnections, all of the large synchronous generators in the interconnected areas are tied together electrically and must rotate at identical speeds. Thus, the frequency measured at any point in the three interconnected areas will be exactly the same at any point in time, and all of the units in the interconnected areas participate in absorbing the impact of a disturbance. Utilities have found these interconnections to be quite beneficial, with a large part of the U.S. east of the Rockies and parts of Canada forming a large (400,000 MW (Ref. 5-1)) interconnection.

Interconnection Operation

When a disturbance occurs within one of the interconnected areas, it causes a rapid change in power flowing through the tie lines. Because these lines are used for normal interchange of power and for possible emergencies, it is necessary that these fluctuations be corrected by appropriate changes in power generation in the area in which the disturbance occurred. The North American Power Systems Interconnection Committee (APSIC) currently requires that the control area which suffered the loss of a generating unit return tie line currents to normal values within 10 minutes. This is generally done by measuring tie line currents in each area and comparing the tie line current error with a frequency error. This procedure is used because if an area were to lose generating capacity, the tie line current flows into the area would increase while the frequency would decrease. By applying the proper sign and proportionality factors, an area control error is produced that is zero in all but the area in which the disturbance occurred. Based on the area control error, the AGC loads or unloads the operating units in that area.

FIG. 5-2

GENERATING SYSTEM INTERCONNECTIONS

Response Requirements

Response requirements are highly dependent on the characteristics of an individual utility system and its generation equipment mix. Daily load following, for example, may require that the power generation within an interconnected area increase at a rate of 0.5 percent of peak load per minute. However, because some generation units may be base loaded and/or not capable of rapid response, the remaining units in that area, irrespective of utility boundaries, must carry an increased share of the load change requirement. Other considerations, such as economic dispatch may also place constraints on the sequential loading of individual generator units.

A graphical representation (Fig. 5-3) has been developed (Refs. 5-2 and 5-3) to estimate typical generation unit response requirements over the major operating regimes encountered in a large interconnection. The desired response is most rapid when the unit is being used for frequency regulation. Even then, however, the total load excursion (measured as percent of full-load power) is only on the order of one to five percent. If the unit is being used for tie line thermal backup, the size of the disturbance will affect the response rate. The loading on the tie lines will also affect the response rate with heavy loading requiring rapid response. Thus, the desired response rates can vary from nearly ten percent per minute to three percent per minute with excursions of from three percent to fifteen percent of full load, respectively. For daily load following, response rates are lower but excursions larger.

As mentioned earlier, the above values determined by the desired response curve will vary with different utilities. However, the advantages of rapid response capability coupled with a large excursion capability is quite clear. For example, it is generally necessary to have spinning reserves equal to approximately 150 percent of the capacity of the largest single generator in operation to enable restoration of tie line flows in the event that that unit is shut down. To comply with the APSIC requirement mentioned previously, the most desirable type of reserve would be one that would respond with full load to such a large disturbance in ten minutes or less.

Based on considerations such as those described above, individual utilities will specify performance capabilities for their equipment. A typical set of performance capabilities is presented in Table 5-1.

FIG. 5-3

DESIRED GENERATING UNIT RESPONSE

LARGE INTERCONNECTION

TABLE 5-1

TYPICAL REQUIRED OPERATING CAPABILITIES FOR A GENERATING UNIT

Minimum Generation

- a) Each unit must be capable of maintaining synchronous speed and carrying station service load while not connected to the electric system.
- b) Each unit must be capable of operation as a synchronous condenser.

Load Following

- a) Each unit must be able to transfer from synchronous condenser operation to maximum emergency generation in under ten minutes. This capability will be utilized solely under emergency conditions when reduction of machinery life expectancy is acceptable.
- b) From any point between its minimum and maximum generation points, each unit must be capable of changing load at the rate of 3 1/3 percent of nameplate generator rating per minute under normal conditions.

Regulation

- a) From a variable set point between fifteen and eighty-five percent of generation nameplate rating, each unit must be capable of maintaining a regulation band plus or minus 15 percent of generator nameplate rating, with a rate of change of 5 percent of generator nameplate rating per minute.

Turnaround

- a) Each unit must be capable of instantaneous electrical disconnect when pumping at motor nameplate rating under emergency conditions when reduction of unit life expectancy is acceptable.
- b) Each unit must be capable of transfer from pumping at motor nameplate rating to maximum emergency generation within thirty minutes under emergency conditions when reduction of unit life expectancy is acceptable.

Comment:

These criteria present an ideal from a system operating standpoint. They are not absolute, however, and are subject to trade-offs in costs and plant reliability.

PFBC/CAES SYSTEM PART-LOAD CHARACTERISTICS

When a PFBC/CAES power plant (Fig. 5-4) responds to a load disturbance, the major equipment will experience significant changes in operating conditions. It was beyond the scope of the present study program to directly relate the utility load-following requirements to those changes in power plant operating conditions. Rather, this effort was aimed at providing a preliminary identification of the open-bed PFBC/CAES power plant steady-state, part-load performance and an indication of the limits placed upon transient operation by the physical and thermal characteristics of the equipment. Thus, although the transient path to a part-load operating point are not defined, the acceptable boundaries of the path are indicated.

Steady-State Part-Load Performance

The open-bed PFBC/CAES power plant has the capability of operating at part load. This capability exists for both the compression and generation modes of operation. Although part-load operation during generation is the prime subject of this discussion, a brief commentary on part-load operation during compression is appropriate at this point.

Part-Load Operation During Compression

Part-load operation during the compression period could be accomplished by the use of variable geometry guide vanes in the compressor which would permit quite rapid loading and unloading of the motor by changing the airflow rate. A single variable flow compressor would typically be limited to operation between 70 percent and 100 percent of design flow with an almost proportional change in the required compression power. While the load is limited to a thirty percent change, this change exceeds the typical load-following rapid response requirements previously identified in the section on Response Requirements which are on the order of fifteen percent. Larger load variations can only be accommodated by using multiple compressors.

Part-Load Operation During Generation

During the peak load period when the PFBC/CAES plant is generating electricity, the most direct means of changing output power is by throttling the airflow to the expansion turbine as shown in Fig. 5-4. However, an alternative method was also investigated based on the use of a partial admission expansion turbine which contains a governing stage and throttle valve within the turbine.

COAL-FIRED PFBC/CAES SYSTEM

FIG. 5-4

The steady-state, part-load performance estimates for both part-load operation approaches are presented in Fig. 5-5. The partial admission turbine leads to better overall system performance, although the difference is very slight. The primary differences between these two control methods is in the part-load operating conditions of the major components. For example, the expansion ratio across the expansion turbine is quite different for the two control schemes. This variation in expansion ratio occurs primarily because of the different pressure at the inlet to the expansion turbine (Fig. 5-6). For the throttle valve control scheme, the pressure at the inlet to the expansion turbine decreases linearly with flow. Without the throttle valve, such as is the case for the partial admission control scheme, the pressure at the inlet to the expansion turbine remains high at part load, and even increases slightly because of the reduced pressure drops in the ducting and recuperator at reduced flow. At the same time, the pressure in the PFBC experiences a linear decrease with flow (Fig. 5-7) because that pressure is determined by the choked nozzles in the low-pressure turbine. The resulting higher expansion ratio produces a higher temperature difference across the partial admission expansion turbine and a lower exit temperature (corresponds to PFBC inlet temperature of Fig. 5-7). This increased expansion ratio, combined with the lower airflow requirement, yields a round trip heat rate with the partial admission turbine that is about six percent lower than with the simple throttling control scheme (Fig. 5-5). The heat rate profiles for either method are quite similar, however, and it appears that there would be little difference in terms of transient response. Therefore, in the next section on transient response, attention was focused on the simple throttling approach to part-load operation.

As a result of throttling, the low pressure turbine exit temperature (Fig. 5-8) rises to nearly 1000 F (540 C) at twenty-five percent output. This will affect the design of the last turbine stages as well as the recuperator. This increase in exhaust temperature at part load is opposite from the trend experienced in conventional gas turbines where the exhaust temperature continually declines in going from peak-load to idle. As a result, conventional turbine blade designs are based on peak-load conditions. Higher part-load temperatures, such as those estimated for PFBC/CAES systems, will require careful evaluation relative to the specific turbine being considered for PFBC/CAES applications. Similarly, the recuperator materials must be selected for the part-load conditions, or other means devised to limit gas temperature into the recuperator.

While it would be possible to reduce the low-pressure turbine exit temperature at part load by passing some air around the PFBC, such action would affect operation of the PFBC. As presently configured, the constant low-pressure turbine inlet temperature and the fixed low-pressure turbine nozzle area results in a PFBC pressure that is directly proportional to the flow through the bed. This results in a constant superficial velocity in the

FIG. 5-8

PFBC over the entire load range. Bypass of air around the PFBC would require a re-evaluation of PFBC design which was outside the scope of the study. Nevertheless, it is anticipated that such a scheme would be workable and that the variations in low-pressure turbine exit temperature would be less than shown in Fig. 5-8.

Transient Response

Several physical limitations primarily related to thermal cycling and the resulting thermal stresses affect the transient response of a power plant under changing loads. In the CAES power plant, these limitations would be applied to the expansion and low-pressure turbines plus the recuperator because they would be subjected to thermal cycling during load changes and startup. The PFBC, however, would not experience thermal stresses under normal operation because it would be operated at constant temperature. The output from the PFBC would be changed by varying its operating pressure with corresponding changes in air and coal feed rates. The coal inventory in the bed could, however, limit the PFBC response rate. A discussion of these limitations is presented in the following sections along with a preliminary estimate for transient response rate and a commentary on prospective procedures that permit startup to proceed within the limitations.

Turbomachinery Thermal Cycling Limitations

The expansion and low-pressure turbines differ in that the former is based on steam turbine design practice while the latter is a gas turbine derivative. As was shown in Figs. 5-6 and 5-8, both turbines could experience a 300 F (167 C) increase in temperature while load is decreased from 100 percent to 25 percent. While this change could be reduced by passing air around the fluid bed combustor, its existence would not create an undue constraint on either the expansion or low-pressure turbine design.

The gas turbine which is used for the CAES low-pressure turbine is capable of relatively rapid load changes. Aircraft derivative peaking units are brought from synchronous idle to full load in as little as three minutes, while the heavier-duty industrial machines usually take 15 to 20 minutes. The temperature changes which occur in current peaking applications at these response rates are actually much more severe than those that would be anticipated for PFBC/CAES applications. Consequently no thermal cycling limitations are expected for the low-pressure turbines.

The steam turbine-derivative expansion turbine is generally less tolerant of temperature changes. Rapid changes could cause thermal stresses in the heavy turbine sections which could result in fatigue failures. Typical limits for steam turbines are shown in Fig. 5-9 (Ref. 5-4). This figure shows the

ROTOR SURFACE CYCLIC DAMAGE

CYCLE DAMAGE OF 0.02 PERCENT PER CYCLE

difference between high-performance, base-load designs and those that would be used for load-following service with more modest inlet conditions. The latter would be the type used in this application and could tolerate temperature changes of up to 200 F (93 C), at essentially unlimited rate. Such a temperature change would correspond to a load change in the PFBC/CAES power plant on the order of 50 percent of design. With an operating characteristic this liberal, it is unlikely that the steam derivative expansion turbine will impose any undue limitations on system response.

Fluid Bed Combustor Cycling Limitations

The considerations associated with combustion and sulfur removal in the PFBC as well as the combustor mechanical design factors make it highly desirable for the PFBC to operate at constant temperature. For such operation, transition from one operating point to another point should not involve any net thermal transients. However, changing the load in the combustor by merely maintaining the proper relationship between air and coal flow will not necessarily assure constant temperature operation. Because the open-bed PFB combustor operates at approximately 400-percent excess air, at any airflow rate in the turbine operating envelope, the heat released by the combustion process is determined largely by the carbon inventory of the bed. Therefore, simply changing air and coal flow will not produce an immediate, commensurate change in burning rate, nor will it result in a constant combustor exit temperature.

Combustor Transient Model

To obtain a preliminary estimate of the dynamics associated with load change, a simplified transient PFBC model was used. It is based on the equations shown in Fig. 5-10 and the following assumptions:

- 1) the fluid bed is assumed to be well mixed and all of the heat of combustion appears as sensible heat in the air stream;
- 2) the heat of combustion is assumed to come from two sources, carbon which burns relatively slowly in the bed and volatile matter which is assumed to be given off rapidly and burned quickly as compared to the carbon; and
- 3) the rate of carbon combustion is assumed to be directly proportional to the amount of carbon in the bed.

After taking partial derivatives, the model can be described by the block diagram of Fig. 5-11 which illustrates, using Laplace transform notation, the effect on bed temperature resulting from changes in coal feed and airflow rates. A change in the coal feed rate changes the heat released in the bed

COMBUSTOR EQUATIONS

HEAT BALANCE

$$M_B C_B \frac{dT_B}{dt} = W_G C_G (T_{IN} - T_B) + Q_C$$

HEAT RELEASE

$$Q_C = K_{CB} M_{CB} (HHV_C) + K_V W_C$$

CARBON INVENTORY

$$\frac{dM_{CB}}{dt} = K_C W_C - K_{CB} M_{CB}$$

- $M_B C_B$ = HEAT CAPACITY OF BED MATERIAL — Btu/F
- T_B = BED TEMPERATURE — ALSO OUTLET GAS TEMPERATURE — F
- $W_G C_G$ = COMBUSTION GAS FLOW TIMES SPECIFIC HEAT — Btu/sec-F
- T_{IN} = AIR TEMPERATURE INTO BED — F
- Q_C = HEAT RELEASED BY COMBUSTION IN BED — Btu/sec.
- K_{CB} = RATE OF CARBON COMBUSTION/CARBON IN BED — 1/sec.
- M_{CB} = AMOUNT OF CARBON IN BED — lb.
- HHV_C = HIGHER HEATING VALUE OF CARBON — Btu/lb.
- K_V = HEATING VALUE OF VOLATILES PER POUND OF COAL — Btu/lb.
- W_C = COAL FEED RATE — lb./sec.
- K_C = WEIGHT FRACTION OF FIXED CARBON IN COAL

COMBUSTOR DYNAMICS

FIG. 5-11

via two mechanisms. In the first mechanism, a change in the coal rate changes the volatile matter availability. The effect on the bed temperature of this change is experienced directly through the coefficient K_v . In the second mechanism, a change in the coal rate changes the carbon inventory in the bed. The effect on bed temperature of this change is delayed through the time constant τ_1 (estimated from equation in Fig. 5-11 to be approximately 4 min) which reflects the carbon inventory in the bed. A change in the airflow directly affects the rate of heat removed from the bed. The difference between heat released by combustion and that removed by the air stream results in a change in bed temperature through the time constant τ_2 (estimated from equation in Fig. 5-11 to be approximately 5 to 6 min at design conditions).

To maintain constant bed temperature at steady-state conditions with constant air inlet temperature, both fuel and airflow rates must be changed by the same fractional amount. When they are changed, however, the airflow rate affects bed temperature through the bed thermal time constant while the coal feed rate affects bed temperature through the lag associated with the change in carbon inventory as well as the bed time constant.

Combustor Transient Response Estimate

The transient response estimate for a simultaneous ten-percent increase in both coal feed and airflow rates is shown in Fig. 5-12. This response shows a temporary temperature excursion peaking approximately 5 minutes after the change in feed rates. The maximum temperature excursion during this period is approximately 25 F (14 C). Since the response equations are linear, a fifteen-percent increase in feed rate, which is the greatest that typically would be required in a short time, would be expected to result in a temperature disturbance of less than 38 F (21 C). Should greater load changes be desired, however, it would be necessary to add dynamic compensation so that a change in coal feed rate would lead to a change in airflow, thereby minimizing the thermal disturbance to the bed. It should be noted that although Fig. 5-12 illustrates the response of bed temperature to an increase in coal and air flow rates, a decrease in both the coal and air flow rates would produce an almost symmetrical curve about the initial bed temperature.

It should be cautioned that while the general conclusion that the PFBC will be capable of relatively rapid response is undoubtedly valid, the estimated disturbances are based on a preliminary analysis. As noted earlier, the primary cause of the thermal disturbance is the time lag associated with changing the carbon inventory. Unfortunately, this is one of the least quantified parts of the combustion process and, while the assumed relationship appears reasonable, it is possible that the measured response could be significantly different. It is expected that the general trend is correct, however, and that a more refined analysis, combined with small-scale test data for correlation of the model, would be helpful in improving confidence in the ability of the PFBC to respond.

BED TEMPERATURE RESPONSE

10-PERCENT INCREASE IN COAL & AIR FEED RATE

FIG. 5-12

Startup

It is expected that the PFBC will be the limiting factor in system startup. For the PFBC, the design of the startup system and the procedures followed during startup depend mainly on the initial temperature of the bed material.

During commissioning or after a prolonged shutdown, a cold start from ambient temperature will be necessary. Under the normal operating regime of the PFBC/CAES power plant, two methods of startup are possible:

- (1) hot start - this involves maintaining the bed near the operating temperature (typically 1470 F (800 C)) during the overnight or weekend periods during which operation of the combustor is not required; and
- (2) warm start - this involves cooling the bed to an intermediate temperature (typically 570 F (300 C)) on shutdown, and then re-heating the bed using auxiliary oil burners during startup.

The main advantages of the hot start concept are that coal can be fed to the combustor from the beginning of the startup period and the plant can be brought to maximum continuous output conditions comparatively fast. If no fluidizing air is supplied during the shutdown period, however, the temperature of the combustor internals will approach the bed temperature. Consequently, the maintenance of a hot start state requires the internals to be designed to withstand the temperature.

The warm start concept enables the combustor internals to be kept cool during the shutdown period without the consumption of cavern air or supplementary oil. However, the startup period is comparatively long and requires cavern air during both the transition to shutdown and during startup, plus supplementary firing during startup.

It was beyond the scope of the present study to evaluate the various PFBC startup operations in the context of CAES requirements. For the PFBC/CAES preconceptual design, the warm start concept was assumed as this is a well-established design feature in PFBC studies for conventional combined cycle power generation applications. As applied to the PFBC/CAES preconceptual design, the startup procedure is actually simpler than for PFBC/combined cycle applications because of the availability of an independent supply of compressed air. Air from storage is used to fire No. 2 fuel oil in standard gas turbine combustors. The hot gasses enter the fluidized bed through the distributor plate and the bed is heated to between 1290 F (700 C) and 1470 F (800 C). Coal injection to the bed is then started, the bed temperature is raised to 1560 F (850 C), and the airflow to the bed increased to bring the plant on load.

LOAD-FOLLOWING WITH CONTINUOUS PFBC/CAES OPERATION

As mentioned in the previous section, preliminary indications suggest that the open bed, PFBC/CAES power plant selected for the preconceptual design should be capable of wide load variations at relatively fast response rates. Unfortunately, there is yet no hard evidence of this load-following capability in large-scale combustor systems envisioned for PFBC/CAES applications. Consequently, an investigation was conducted into alternative methods of load following intended to maintain the combustor at or near its full-load operating point.

Operation of the combustor at full-load conditions implies that the turbine must also be operated continuously. Because the turbine would be continuously producing power, the opportunity exists to use this power to drive the CAES compressors rather than relying on grid power to drive them. Consequently, one load-following scheme investigated featured compressor powering solely by the CAES power plant turbines. In this scheme, load following would be achieved by varying the compressor power demand by using variable geometry guide vanes in the compressor. The differential between the varying compressor power demand and continuous turbine power output would represent the load-following capability of a single CAES turbocompressor unit. This method of load following was investigated for the four-unit PFBC/CAES power plant preconceptual design previously presented. Accordingly, the resulting load following capability of the plant includes that attainable by using variable geometry plus that attainable by sequentially declutching each of the four compressors trains from their respective turbines.

Another load-following scheme featured a variation of the previous continuous-fired scheme in which supplementary compressors would be added to each primary turbocompressor unit. The power to drive these supplementary compressors would be taken from the grid, and the air would be put into storage. This scheme would provide a two-fold benefit: one, by consuming off-peak power from the grid it would provide a load-leveling benefit leading to a wider range of net plant load following capability, and two, because of the additional compressed air put into storage, the plant would be capable of generating peak power for a longer period of time.

This section covers the two continuous firing load following schemes and describes their operation under steady-state, part-load operating conditions. The results of preliminary performance and economic analyses, similar to those performed during the system analysis effort, are presented.

System Description

The PFBC/CAES system configuration selected in the system analysis phase uses a cycling open-bed PFBC operating at a pressure of 15 atm (bar) and an exit temperature of 1562 F (850 C). For convenient reference, a schematic diagram of the selected system configuration is repeated in Fig. 5-13 (same as Fig. 3-10), the heat and mass balance is repeated in Table 5-2 (same as Table 3-9), and a brief summary of its operation is given below. This same basic system configuration was specified for investigation of the continuous-firing schemes.

Cycling Open-Bed PFBC/CAES System

When the cycling PFBC system is operating in the storage mode, air would be compressed to approximately 63 atm (bar) by a train of three compressors, (LC, HC, and BC) incorporating three stages of intercooling. Power to drive the compressors would be taken solely from the grid while the turbines are unclutched, and the PFBC would be turned down to banked or idle condition. During the generation mode, high-pressure air from storage would be released through a valve and heated in the recuperator prior to expansion through the expansion turbine to 15 atm (bar). The PFBC would have to be brought up to full load so the air could be heated to 1562 F (850 C) and expanded again through the low-pressure turbine (LT). A consequence of this type of cycling operation is that the PFBC must cycle between turn down and full-load conditions.

Continuous Firing System

In the continuous firing PFBC/CAES system (Fig. 5-14), the PFBC and turbine would be operating at full load at all times. During the compression period, the compressors would be driven solely by the turbines. Air required for the combustors and turbines would flow through bypass lines connecting the booster compressor exit with the recuperator inlet. The balance of compressor discharge air would go into storage. To switch to power generation, the airflow into the compressors would be reduced by using variable geometry stator vanes so that the power requirement to drive the compressors would be reduced. In this manner the excess turbine output could go to the grid. During this period, compressor discharge air must be supplemented by air from storage because the compressor discharge would no longer be sufficient to operate the turbine. To further increase the power to the grid, the compressors would be sequentially declutched from the turbines, permitting the entire power output to go the grid. Thus, this continuous firing operating mode would provide a means of load following by switching from compression to generation without requiring turn-down of the combustor.

SELECTED COAL-FIRED PFBC/CAES SYSTEM

CYCLING OPERATION

FIG. 5-13

TABLE 5-2

HEAT AND MASS BALANCE FOR COAL-FIRED PFBC/CAES CONFIGURATION

Cycling Combustor

Note - Stations 1 through Cavern for Compression Mode
Stations 10 through 16 for Generation Mode

PFB Pressure = 15 Atmospheres
LT Turbine Inlet Temperature = 1562 F

Station No. ⁽¹⁾	1	2	3	4	5	6	7	8	9	Cavern	10	11
W, lb/sec	761.8	761.8	761.8	761.8	761.8	761.8	761.8	761.8	761.8	761.8	708.6 ⁽²⁾	708.6
T, F	58.7	345.0	160.9	500.0	200.0	377.6	200.0	409.3	120.0	120.0	120.0	511.2
P, psia	14.7	60.2	55.8	230.4	225.8	440.3	430.1	925.2	906.7	973.7	902.9	859.0
H, Btu/lb	124.0	193.0	148.5	231.1	157.9	201.0	157.9	208.8	138.7	138.7	138.7	233.8
Station No. ⁽¹⁾	12	13	14	15	16							
W, lb/sec	708.6	22.4	728.8	745.7	745.7							
T, F	255.5	--	1562.0	660.7	300.0							
P, psia	220.5	--	216.0	15.2	14.9							
H, Btu/lb	171.3	--	510.8	272.4	181.9							
Compressor Power, MW		Turbine Power, MW		Gross Heat Rate, Btu/kWh		Heat Exchanger Heat Transfer, Btu/sec						
						HHV						
LC	56.2	ET	45.5	Generation	4322	IC1		33929				
HC	67.2	LT	178.4	Compression	8926	IC2		55730				
BC	76.5	Total, gross	223.9	Round trip	13248	IC3		32819				
Total	199.9					Aftercooler		53387				
						Recuperator		67441				

(1) Station numbers refer to locations identified in Fig. 5-13 (all at surface except for cavern).

(2) Plus 22.7 lb/sec of cooling air for total of 731.3 lb/sec withdrawal rate.

This is 4 percent less than compressor flow rate to adjust for cavern leakage.

CONTINUOUS FIRING OPERATION

Alternate, Continuous Firing System Configuration

Prior to initiating the continuous firing investigation, it was recognized that the base cycling PFBC/CAES plant, modified for continuous operation as described in the proceeding section, would have relatively high fuel costs. During the compression mode, the continuous-fired PFBC/CAES plant would essentially be operating as an open-cycle gas turbine with the net shaft power being used to compress air for storage. Open-cycle gas turbines are known to have relatively high heat rate. Accordingly, several alternate continuous fired configurations were suggested with the intent of maximizing the thermal efficiency. These alternate configurations generally involved the addition of steam cycles to convert the basic open-cycle gas turbine into a high-performance combined cycle during the compression period. These alternate configurations were subsequently judged by DOE to be inappropriate for inclusion in the present study.

Full-Load Operating Conditions

The analysis of the continuous firing concept was initially based on the PFBC/CAES plant preconceptual design discussed earlier. The key guideline during this investigation was maintaining the combustor at its full-load operating condition. This condition corresponds to a constant coal consumption rate of 22.4 lb/sec (10.2 kg/sec) as in the preconceptual design. Another important guideline was requiring zero net electrical power generation during the off-peak period so that no power would be delivered to the grid. A consequence of this was to increase the compressor airflow rate slightly so that the compressor power requirement was equal to the turbine output power.

Using these guidelines, the performance of the continuous fired system was simulated in both the full generation mode (Table 5-3) and for the full compression mode (Table 5-4).

The operation of the continuous firing plant during the generation mode would be unchanged relative to the cycling simulation of Table 5-2. With the turbines and combustors operating at their respective full load conditions. Several changes in the operation of the plant during the compression mode would occur as a result of continuous firing. The major change, noted in Table 5-4, is that the turbine airflow and power output has increased compared to the generation mode (Table 5-3) operation. These increases resulted from two effects. First, the turbine air would now be supplied from the booster compressor exit where the air temperature would be 408 F (209 C) instead of 120 F (49 C) as from the cavern. This increase in sensible heat would be translated to the combustor where the inlet temperature would be 82 F (46 C) hotter than during the generation mode, thus reducing the coal rate for the same inlet airflow and combustor exit temperature. To increase the coal rate to the full load value, the airflow through the turbines must be increased by about six percent. The second effect

TABLE 5-3

HEAT AND MASS BALANCE FOR COAL-FIRED PFBC/CAES CONFIGURATION
Continuous Firing
Full Generation Mode Operation

PFB Pressure = 15 Atmospheres
LT Turbine Inlet Temperature = 1562 F

Station No. ⁽¹⁾	Cavern	10	11	12	13	14	15	16
W, lb/sec	761.8	708.6 ⁽²⁾	708	708.6	22.4	728.8	745.7	745.7
T, F	120.0	120.0	511.2	255.5	--	1562.0	660.7	300.0
P, psia	973.7	902.9	859.0	220.5	--	216.0	15.2	14.9
H, Btu/lb	138.7	138.7	233.8	171.3	--	510.8	272.4	181.9

<u>Turbine Power, MW</u>		<u>Gross Heat Rate, Btu/kWh</u>		<u>Heat Exchanger Heat Transfer, Btu/sec</u>	
			<u>HHV</u>		
ET	45.5	Generation	4322	Recuperator	67441
LT	<u>178.4</u>	Compression	<u>20715</u>		
Total, gross	223.9	Round trip	25037		

- (1) Station numbers refer to locations identified in Fig. 5-14 (all at surface except for cavern).
- (2) Plus 22.7 lb/sec of cooling air for total of 731.3 lb/sec withdrawal rate.
This is 4 percent less than compressor flow rate to adjust for cavern leakage.

TABLE 5-4

HEAT AND MASS BALANCE FOR COAL-FIRED PFBC/CAES CONFIGURATION
Continuous Firing

Full Compression Mode Operation

PFB Pressure = 15 Atmospheres
LT Turbine Inlet Temperature = 1562 F

Station No. ⁽¹⁾	1	2	3	4	5	6	7	8A	8B	9	Cavern	10	11
W, lb/sec	932.9	932.9	932.9	932.9	932.9	932.9	932.9	152.5	780.4	152.5	152.5	752.7	752.7
T, F	58.7	345.0	160.9	500.0	200.0	377.6	200.0	409.3	409.3	120	120	409.3	624.8
P, psia	14.7	60.2	55.8	230.4	225.8	440.3	430.1	925.2	925.2	906.7	973.7	925.2	880.2
H, Btu/lb	124.0	193.0	148.5	231.1	157.9	201.0	157.9	208.8	208.8	138.7	138.7	208.8	262.2

Station No. ⁽¹⁾	12	13	14	15	16
W, lb/sec	752.7	22.4	772.9	793.5	793.5
T, F	337.4	--	1562.	662.9	462.6
P, psia	220.5	--	216.1	15.2	14.9
H, Btu/lb	191.2	--	510.8	272.8	222.2

<u>Compressor Power, MW</u>		<u>Turbine Power, MW</u>		<u>Gross Heat Rate, Btu/kWh</u>		<u>Heat Exchanger Heat Transfer, Btu/sec</u>	
				HHV			
LC	68.8	ET	55.0	Generation	4322	IC1	41514
HC	82.4	LT	189.9	Compression	20715	IC2	68288
BC	93.7	Total, gross	244.9	Round trip	25037	IC3	40208
Total	244.9					Aftercooler	10690
						Recuperator	40194

(1) Station numbers refer to locations identified in Fig. 5-14 (all at surface except for cavern).

that led to the increases in turbine output and flow is the decreased pressure drops experienced by the flow in reaching the expansion turbine inlet. An increased inlet pressure level of almost 1.4 atm (bar) would be experienced by the expansion turbine compared to operation during the generation mode because the flow would bypass several major plant components (aftercooler, cavern, piping, etc.) and the pressure losses associated with these components would not be incurred. Both the higher turbine flow rate and pressure level would increase the turbine power from 223.9 MW to 244.9 MW (per unit).

To match the turbine power output, the compressor power requirement would have to be increased from 199.9 MW (Table 5-2, cycling plant) to 244.9 MW by increasing the compressor flow capacity from 761.8 lb/sec (345.5 kg/sec) to 932.9 lb/sec (423.2 kg/sec). Of course, the overall pressure ratio would be maintained.

As a result of these modifications, the compression heat rate would be changed dramatically (compare Table 5-4 with Table 5-2) while the generation heat rate remains unchanged. If it were assumed that the continuous system operates in the full compression mode or the full generation mode and nowhere in between, then it will only supply power to the grid during the generation mode when the compressors are declutched. During this period of full generation, the plant will be operated on compressed air taken from storage. The capacity of the equipment is such that for every five hours of operation in the compression mode, enough air would be stored (152.5 lb/sec/unit; 69.2 kg/sec/unit) to provide approximately one hour of operation during the generation mode while consuming 761.8 lb/sec (345.5 kg/sec) of compressed air from storage. This means that the continuous firing system would be producing little grid power during the generation mode relative to the coal consumed during the compression mode, and leads to the high compression heat rate of 20715 Btu/kWh and a round trip heat rate of 25037 Btu/kWh.

If the plant is assumed to be available to generate and pump (at full generating or compression operation) five days a week with the weekends left just for pumping, the daily power generation time would be 5.8 hr/day for a total plant utilization time of 1508 hr/yr (52 weeks/yr plant operation).

Load Following Using Variable Geometry Compressors

A PFBC/CAES power plant must be able to smoothly follow changes in load demand. The following section indicates how a continuous firing PFBC/CAES system could load follow yet maintain full-load operation of the PFBC. For this investigation, the total plant is assumed to contain four PFBC/CAES turbo-machinery power blocks and to have a total gross power output of 896 MW (four times output from Table 5-3) at full generation capacity.

Increasing Power Plant Output from Full Compression Operation

The utilization of variable geometry compressors in the continuous firing PFBC/CAES system would allow the compressor flow rate and power requirement to vary, yet still allow the compressor to maintain storage pressure level and rotor speed. An overall compressor map for a representative compressor with variable stator blades is shown in Fig. 5-15. The pressure and flow characteristics for this map have been normalized relative to the design point for the preconceptual design. As can be seen from the figure, storage pressure can be maintained down to seventy percent of the compressor inlet airflow by varying the stator blade angle. By reducing the compressor flow rate, the power required to drive the compressor would be reduced and the turbine power output could be reallocated from the compressor to the electric generator. Below seventy percent air flow, the discharge pressure would have to be reduced to maintain adequate compressor surge margin.

The variation in net power which could be delivered to the grid by a single turbine/compressor train is depicted in Fig. 5-16 in terms of the net airflow into storage. In the full compression mode (stator blade wide open), zero net power would be generated and the difference between compressor airflow and turbine airflow (152.5 lb/sec (69.2 kg/sec)) would be delivered to storage. At an intermediate stator setting where the compression flow has been reduced to about 83 percent, the compressor and turbine flows are equal so no air would flow to storage. At this point approximately 40 MW of net power could be delivered to the grid. As the stator blades are reduced to the minimum flow setting, about seventy percent, the compressor airflow would be only 653 lb/sec (296.2 kg/sec), so that approximately 118 lb/sec (53.6 kg/sec) of air must be taken from storage to operate the PFBC and turbine. Complete shutdown of the compressor requires it to be disconnected from the air storage system because the discharge pressure cannot be maintained. This shutdown could be accomplished by either an almost instantaneous unclutching of the compressor and allowing it to coast to a stop or by leaving it coupled to the main shaft so that it acts as a brake while the compressor air is vented. This latter approach would be very costly (in terms of energy required to operate the compressor) and should be avoided if at all possible.

When all four turbomachinery units are operating in the full compression mode, a total of 610 lb/sec (276.7 kg/sec) or 152.5 lb/sec (69.2 kg/sec) per unit of airflow would be sent to storage and power would not be available to the grid. A smooth transition from zero net power to a power contribution equal to approximately eighty percent of gross turbine output could be accomplished by sequentially reducing airflow rate through each of the four compressors from 100 percent of flow to 70 percent of flow. Referring to Fig. 5-17, the first compressor airflow rate would be progressively reduced to seventy percent flow; this

VARIABLE GEOMETRY COMPRESSOR OPERATING CHARACTERISTICS

RPM = 3600

SINGLE TURBINE/COMPRESSOR TRAIN OUTPUT POWER VARIATION

FULL COMPRESSION TO FULL GENERATION

POWER PLANT PART-LOAD OUTPUT POWER VIARATION

FULL COMPRESSION TO FULL GENERATION

FIG. 5-17

would change the power requirement for that compressor from 244.9 MW at 100-percent flow to approximately 171.4 MW at 70-percent flow. With the turbine running at full load (244.9 MW), the difference (73.5 MW) between the reduced compressor power requirement and the turbine output would be available to the grid. After the first compressor is at its seventy-percent flow operating point, the second, third, and fourth compressors would then have their flows decreased in sequence until all four compressors are operating at their minimum flow condition at seventy percent. At this point the power plant output to the grid will correspond to approximately 33 percent of the combined turbine capacity.

To increase power output still further one compressor would be shut down with the remaining three compressors brought to full flow capacity to maintain approximately the same net power output to the grid. Next, the second compressor would have its flow rate decreased by progressively closing the stator vanes. This would be followed in sequence by compressors three and four until all three compressors are again at minimum flow. At this point the second compressor would be shut down by declutching.

This scenario would continue until the first three compressors are shut down and only the last compressor train would be operating at seventy-percent flow. The power plant would then be generating at approximately eighty-percent capacity.

The operation of the continuous firing PFBC/CAES power plant between 80-percent and 100-percent power output presents problems because the compressor cannot be efficiently used to load the turbine. Recall that below the seventy-percent operating point, the compressor discharge pressure would be too low to be put into storage or used in the expansion turbine. However, below the seventy-percent point the compressor could be used as a dummy load by keeping it clutched to the turbines and opening exhaust valves to dump the compressor discharge airflow. In this manner, the net CAES plant output could be smoothly increased from about 80 percent to almost 100 percent. Unfortunately, this mode of operation would be very inefficient because the energy used in compressing the air flow could not be utilized by the power plant as presently configured.

It might be possible to modify the configuration to permit bypassing part of the air from the high compressor discharge directly to the combustor as shown in Figure 5-18. Bypassing the air away from the booster compressor would reduce the booster compressor power requirement approximately proportional to the flow change. As flow is continuously bled from the high compressor exhaust, the overall pressure ratio of the compressor train would decrease (due to a shift in compressor operating point) so that the pressure of the flow from the booster compressor exhaust would be lower than during normal operation. As this lower pressure exhaust flow is mixed with flow from storage, the overall pressure into the expansion turbine would be lower than the design pressure. Consequently, the exit pressure from the expansion turbine would be lower than

than the design value and, after mixing with the booster compressor bypass, the PFBC inlet pressure would be below its design value causing the bed fluidization characteristics to be altered. The temperature of the combustor inlet flow would also be different from the design value. Combustor inlet flow conditions could vary considerably as the booster compressor bypass flow is increased. Although this scheme is presented as a possible means of load following in the 80 to 100 percent output range, it is unclear at this time how the combustor would operate at different pressures and temperatures without performing an extensive study into this problem.

Decreasing Power Plant Output From Full Generation Operation

To decrease the plant power output as grid demand decreases, the compressors would be sequentially engaged to draw power from the turbines. A compressor could be started by using a starting electric motor to bring the compressor from rest to synchronous speed (3600 rpm). The electric motor would require approximately 55 MW of power. The compressor would then be clutched into the turbine and the electric motor disconnected. As the compressor discharge pressure is increased to storage pressure, the airflow through the compressors would be vented through valves to the atmosphere. When the vent valves are closed, the compressor pressure ratio would increase as would the airflow rate and compressor power requirement. A typical compressor operating line from compressor start to operating pressure is shown in Fig. 5-19.

The net power to the grid as a function of net flow to storage for a single turbine/compressor train is shown in Fig. 5-20. Upon reaching the storage pressure level, the compressor discharge flow would be approximately 653 lb/sec (246.2 kg/sec). This flow would be bypassed to the expansion turbine to offset some of the airflow previously extracted from storage. As the compressor stator vanes are opened, the airflow through the compressor increases, thereby decreasing the power to the grid. When the compressor flow reaches 771.1 lb/sec (349.8 kg/sec), storage air would no longer be required to drive the turbines because the compressor would be supplying the total required amount of air to keep the combustor at the full-load operating condition. Increasing the flow rate further would allow airflow greater than 771.1 lb/sec (349.8 kg/sec) to be sent to storage.

Startup of the remaining compressor trains would be accomplished in a similar manner. Approximately twenty-four percent of the total plant generating capacity can be absorbed by the compressor starter motors. Another fifty percent of plant generating capacity can be absorbed by increasing the compressor pressure up to the storage level. The remaining power output would be absorbed by increasing the compressor flow rate from 70 to 100 percent where the plant would be operating in the full compression mode.

COMPRESSOR DISCHARGE PRESSURE VARIATION

SINGLE TURBINE/COMPRESSOR TRAIN OUTPUT POWER VARIATION

FULL GENERATION TO FULL COMPRESSION

The plant may be operated from full generation to full compression by either simultaneous startup or by sequential startup of the compressor trains. Approximately 74 percent of the plant capacity can be absorbed by simultaneously starting the compressors and by bringing them to the 70-percent flow condition (see Fig. 5-21). However, this mode of operation would be very inefficient. The sequential startup method involves starting one compressor and bringing that compressor up to its full operating condition before starting the next compressor train. The resulting plant output power as a function of storage airflow rate is presented in Fig. 5-22.

The sequential method of compressor startup would be more efficient than the simultaneous method because less storage air would be consumed over the time it would take the plant to respond from full generation to full compression modes. A prospective storage airflow consumption/time relationship is shown in Fig. 5-23. The total elapsed time from full generation mode operation to full compression mode operation would be the same for both methods. Integrating the area under the curve would indicate that the sequential method of compressor startup for load following would consume less storage air than the simultaneous method of compressor startup.

Load Following Using Supplementary Compressor Trains

A severe limitation of the continuous firing PFBC/CAES system discussed above is the constraint placed upon the duration of the full-load generation period. This constraint occurs because the compressors are driven directly by the power plant and have an airflow capacity approximately twenty percent greater than the turbine flow requirements. This means that the ratio of generation to compression times for any given duty cycle will be relatively low. To facilitate load following and increase generation time, additional compressor trains powered by the grid were considered to be added to the CAES plant. These compressor trains would be operated according to the same schedule as would compressors driven by the gas turbines.

Increasing the number of compressor trains would enable the plant to store more air during any given off-peak period, provided of course, that sufficient off-peak energy is available from the grid. As a result, the generation time per day and overall plant utilization would be increased over that for a plant that would utilize only turbine-driven compressors. The magnitude of this increase is indicated in Table 5-5. The total plant power output relationship with storage air flow which results from the addition of two 100-MW compressor trains powered by the grid and which uses the sequential method of starting and full compressor operation is shown in Fig. 5-24.

POWER PLANT OUTPUT POWER VARIATION

FULL GENERATION TO FULL COMPRESSION
SIMULTANEOUS COMPRESSOR TRAIN OPERATION

POWER PLANT OUTPUT POWER VARIATION

FULL GENERATION TO FULL COMPRESSION
SEQUENTIAL COMPRESSOR TRAIN OPERATION

FIG. 5-22

STORAGE AIRFLOW CONSUMPTION

FULL GENERATION TO FULL COMPRESSION

TABLE 5-5

EFFECT OF SUPPLEMENTAL COMPRESSOR UNITS ON PLANT UTILIZATION

Additional Com- pressor Units, MW	W_g/W_c ⁽¹⁾	Generation Time ⁽²⁾ , hr/day	Compression Time ⁽³⁾ , hr/day	Utiliza- tion ⁽⁴⁾ , hr/yr.
(Continuous-Firing)	4.80	5.8	18.2	1508
100	1.37	14.2	9.8	3692
200	.80	18.7	5.3	4862
300	.56	21.5	2.5	5590
400	.436	23.4	0.6	6084

- (1) Ratio of turbine airflow requirement during generation mode 731.3 lb/sec (331.7 kg/sec) to the airflow to the cavern (station 9 in Fig. 5-14) in the compression mode.
- (2) Generation would be five days a week.
- (3) Weekday compression time; the total compression time will also include weekend compression of 48 hours.
- (4) Utilization is based on 52 weeks per year, five days a week.

POWER PLANT OUTPUT VARIATION

GRID POWERED COMPRESSOR TRAINS
SEQUENTIAL COMPRESSOR TRAIN OPERATION

FIG. 5-24

If the additional compressors were installed, the round trip heat rate of the overall plant would be significantly improved for two reasons - more efficient base-load power plants would be used to supply compressor power, and power plant generation mode operating time would be significantly increased. Round trip heat rate is defined by the following relationship:

$$\begin{aligned} \text{RTHR} = & \frac{(W_f \text{ HHV}) (3600) (N_g t_g + N_c t_c + N_{wec} t_{wec})}{N_g t_g \text{ PWR}_{out}} \\ & + \frac{\text{HR}_{bl} \text{PWR}_{cp} (N_c t_c + N_{wec} t_{wec})}{N_g t_g \text{ PWR}_{out}} \end{aligned} \quad (5-1)$$

where, RTHR = round trip heat rate (Btu/kWh)
 W_f = coal rate (lb/sec)
 HHV = higher heating value of coal, as fired (12000 Btu/lb (27910 kJ/kg))
 3600 = conversion factor (secs/hr)
 N_g = days of generation per week (5)
 t_g = hours of generation per week day
 N_c = days of compression per week (5)
 t_c = hours of compression per week day
 N_{wec} = days of compression on weekend per week (2)
 t_{wec} = hours of compression per weekend day (24)
 HR_{bl} = heat rate of baseload power plant (10000 Btu/kWh (10550 kJ/kWh))
 PWR_{cp} = compressor power requirement (kW)
 PWR_{out} = power output of PFBC/CAES power plant (kW)

The effect on performance of adding supplementary compressors in 100-MW increments is displayed in Table 5-6. The addition of the first 100-MW compressor markedly improves both the compression heat rate and the round trip heat rate. The second 100-MW compressor also produces a significant improvement, although, it is not as large as the first 100-MW addition.

The primary reason behind the increase is depicted in Fig. 5-25 which shows the power plant utilization time as a function of additional compressor power. The utilization time is increased almost 2.5 times with the first 100-MW compressor addition. The utilization is almost 3.25 times greater with a 200-MW compressor addition. Adding another 100-MW increment would probably not be advantageous, however, because the PFBC/CAES power plant would not likely be used to such high utilization potential. Furthermore, it is questionable whether the grid would have sufficient off-peak energy to drive this much compressor capacity.

TABLE 5-6

TOTAL POWER PLANT ESTIMATED PERFORMANCE

Additional Compressor Units Powered by Grid

<u>Additional Com-</u> <u>pressor Units, MW</u>	<u>Total Power</u> <u>Output, MW</u>	<u>Generation Heat</u> <u>Rate, Btu/kWh</u>	<u>Compression Heat</u> <u>Rate, Btu/kWh</u>	<u>Round Trip Heat</u> <u>Rate, Btu/kWh</u>
(Continuous-Firing)	223.9	4322	20715	25037
100			12006	16328
200			9913	14235
300			9409	13731
400			9153	13475

EXTENDING POWER PLANT UTILIZATION WITH ADDITIONAL COMPRESSOR POWER

COMPRESSOR POWER FROM GRID

Cost Analysis

A preliminary economic comparison was made of the continuous firing options to evaluate their relative economic potential. The analysis included an estimate of the cost of the continuous firing plant, the compressor trains for the supplemental grid-powered compressor option, and the different sized caverns that would be required as plant storage capacity varied. This cost analysis, which was completed prior to the time when the preconceptual design leveled cost data became available, was prepared using the same nonlevelized costing methodology that was used for the system analysis.

Continuous Firing Power Plant

The cycling and the continuous firing PFBC/CAES power plants would have basically the same capital costs except for the compressor train and the storage cavern.

Compressor Cost Adjustment

The compressors would have a higher flow rate in the continuous firing plant than the cycling plant; therefore, the cost of the compressor and associated auxiliary equipment would be correspondingly higher. The cost of this enlarged compressor equipment was estimated by scaling the cycling compressor costs with compressor power. It was estimated that the incremental plant cost for continuous firing due to the compressors and associated auxiliary equipment would add almost three percent to the capital cost of the cycling power plant.

Cavern Cost Adjustment

The storage capacity for the continuous firing plant would not have to be as large as the twenty hour storage capacity needed for the cycling plant. This is a direct result of the reduced annual utilization (1508 vs 2190 hr/yr). Storage capacity is defined as follows:

$$\text{Storage capacity} = (t_{\text{wec}} + t_{\text{wc}}) W_c / W_g \quad (5-2)$$

where, t_{wec} = compression time during a weekend (48 hr)
 t_{wc} = compression time during a weekday (hr)
 W_c = compression flow rate (lb/sec)
 W_g = generation flow rate (lb/sec)

Based upon this relationship, a storage capacity of 13.8 hours would be required for the continuous firing plant as compared to the 20 hours for the cycling plant. This translates to a smaller and less expensive cavern. Using data from the Ref. 5-5 study, a cost correlation of relative cavern cost versus storage capacity was derived and is shown in Fig. 5-26. It was estimated that the lower storage capacity of the cavern would have the effect of reducing the capital cost of the continuous firing power plant by about four percent relative to the cost of the cycling power plant.

Continuous Firing Plant Total Costs

As a result of the increased compressor and reduced storage costs the capital cost of the continuous firing plant was estimated to be about one percent less than that for the cycling PFBC/CAES power plant. This difference is essentially negligible. Maintenance costs were assumed to be the same for both plants. Because the continuous firing plant would not draw power from the grid, no cost for electricity need be included in the operating costs. Even though the capital cost was essentially the same as for the continuous firing plant, the additional fuel required to operate the combustors continuously caused the total owning and operating cost of the plant to be about seven percent higher than for the cycling plant (see Table 5-7). This comparison is not strictly consistent, however, because of the different annual utilizations for the two plants. This point is addressed again in the next section.

Power Plants With Grid-Powered Compressor Unit

If additional grid-powered compressor units were added to the continuous firing PFBC/CAES power plant to increase the annual utilization, the cavern storage capacity would have to be increased, resulting in further increases in the capital cost of the power plant. The capital costs of the additional compressor units were again estimated by scaling the cycling compressor costs with compressor power requirements, and the cavern costs were estimated using Fig. 5-26. The resulting estimates for relative annual cost of electricity are presented in Fig. 5-27 in terms of the amount of supplemental compressor capacity added to the continuous firing plant. Also shown in the figure is the base cost for the cycling PFBC/CAES plant. As illustrated in the figure, adding supplemental grid-powered compression units to the continuous firing plant tend to increase the relative annual electricity cost estimates. At the same annual utilization as the selected cycling PFBC/CAES plant (2190 hr/yr), the continuous firing system with a small amount of supplementary, grid-powered compressors would have an electricity cost estimate approximately 13 to 14 percent higher than the cycling plant.

CAVERN CAPITAL COST VARIATION

TABLE 5-7

ESTIMATED RELATIVE COST FOR CYCLING AND CONTINUOUS
FIRING PFBC/CAES SYSTEM

	<u>Cycling</u>	<u>Continuous Firing</u>
Annual Utilization, hr/yr	2190	1508
Relative Capital Cost (\$/kw)	100 (base)	99
Relative Electric Power Cost (\$/kwyr)		
Capital	68	67
Fuel	8	31
Electricity	15	-
Maintenance	<u>9</u>	<u>9</u>
Total annual electricity cost	100 (base)	107

(1) Cycling plant is plant selected for preconceptual design

CONTINUOUS FIRING PFBC/CAES COST COMPARISON

COMMENTS ON PFBC/CAES POWER PLANT SYSTEM OPERABILITY

The proceeding remarks documented the investigation of three topics: 1) the identification of utility load-following requirements, 2) the preliminary assessment of the part-load capability of the open-bed PFBC/CAES power plant system, and 3) the evaluation of alternative load-following schemes wherein the PFBC is operated continuously rather than cycled. The findings associated with the first two topics indicate that the recognized procedure for operating an open-bed PFBC in the part-load condition, i.e., constant superficial velocity and bed temperature, should be applicable to the PFBC/CAES power plant system and that it should supply the power plant system with the capability to comply with utility load variation requirements. These findings in a sense make the discussion of schemes for continuous firing of the PFBC academic. However, even though this conclusion can be drawn, the preliminary nature of the investigation must be emphasized. Additional extensive and sophisticated analytical modeling of the PFBC/CAES systems is necessary before this initial encouraging result can be substantially documented.

REFERENCES

- 5-1 Musick, C. R. and R. W. Knapp: C-E Meets Utility Load Change Capability Needs with System 80(TM) NSSS. Combustion, Sept. 1979 pp. 40-52.
- 5-2 IEEE Working Group: MW Response of Fossil Fueled Steam Units. IEEE Transactions on Power Apparatus and Systems, Vol. PAS-92, No. 2, pp. 455-463.
- 5-3 Ahner, D. J., et al: Integrated Low Btu Gasification Combined Cycle Plant Interface and Power System Control Considerations. ASME Paper No. 76-JPGC-GT-3, Sept. 1976.
- 5-4 Ispen, P. G. and D. P. Timo: The Design of Turbines for Frequency Starting. Proceedings of the American Power Conference, Vol. 31, pp, 314-320, (1969).
- 5-5 Giramonti, A. J. et al: Preliminary Feasibility Evaluation of Compressed Air Storage Power Systems. United Technologies Research Center Report R76-652161-5, Supported by National Science Foundation Grant per 74-00242, Technical Oversight by Energy Research and Development Administration, December 1976.

Chapter 6

Concluding Remarks

consideration to changes in combustion efficiency, changes in sulfur acceptor requirements, changes in fluidization behavior, changes in elutriation of particulates from the bed, and changes in the operation of the solids handling and preparation systems. Since startup time is an important index of the performance of a cycling plant, studies of startup constitute an important first step in the evaluation of load following. These studies must address the degree of bed cooling during overnight and weekend no-load conditions, and they must also address limitations to startup rate arising from thermal stresses in refractories and other materials of construction.

Parallel studies are necessary to estimate the part-load performance at the integrated open-bed PFBC/CAES power plant complex, initially at steady state and then under load-following conditions. Several operational constraints are anticipated at part load that might dictate changes from the original preconceptual design. Example problems that could arise include excessively low recuperator metal temperature which could cause sulfuric condensation and corrosion during rapid startup, icing in the control valve or expansion turbine during rapid startup, and excessively high recuperator metal temperature which could cause oxidation of low cost heat exchanger materials during extended operation at low load. These operational constraints would not prevent operation at part load, but they do suggest corrective design modifications, such as incorporating bypass lines or protective auxiliary oil burners, might be necessary.

Eventually, a complete transient simulation of the integrated power plant must be developed so that power plant control strategies could be identified which satisfy utility system transient specifications. This task must start with efforts to model the individual components like the PFBC, recuperator, storage facility, long pipes, and turbomachinery. The individual component response characteristics can be interconnected to develop a comprehensive transient model for the complete system. The model could then be used to conduct dynamic studies to identify basic limitations to power plant load following response and investigate alternative methods of load control.

Finally, detailed component design studies should be conducted to address design concepts which are best suited for CAES applications. For example, the open-bed PFBC preconceptual design presented herein, while not resembling traditional PFBC designs for combined cycles, did incorporate many of the design data and procedures used for base-load PFBC designs. Future design efforts should focus on approaches and techniques intended to yield rapid response rates desired for cycling applications. Following these component studies, detailed site-specific power plant designs could be prepared.

Concluding Remarks

The objective of the study reported herein was to assess the technical and economic feasibility of coal-fired FBC augmented CAES power plants. The results of this study indicate that prospective PFBC/CAES power station designs based on the use of open-bed pressurized fluid bed combustors and heavy-duty industrial gas turbines could become technically feasible and economically attractive in the near future for peaking and intermediate applications. These plants appear capable of producing electricity at competitive costs for annual utilizations above 1925 hr/yr.

The primary advantage of these systems is the ability to save oil. Oil-fired CAES plants are capable of reducing oil consumption to about forty percent of the amount that would otherwise be consumed for peaking applications. Coal-fired PFBC/CAES plants are potentially capable of completely eliminating oil consumption for peaking. Other potential advantages are the operating benefits to the utility system attributable to having load-leveling capacity connected to the grid and the siting flexibility relative to conventional pumped hydro storage plants. The environmental impact of a prudently designed PFBC/CAES plant should be no greater than for alternative power plants.

With suitable incentives, a first generation, coal-fired open-bed PFBC/CAES power plant could become commercial some time during the latter half of the 1980 decade. In this context, the date of commercialization is taken to be the earliest date at which the buyer and seller would agree to sign a contract based on standard commercial terms and conditions. The first date of commercial operation would follow by 6 to 8 years, depending on the time required to obtain necessary permits and the size of the plant.

Even though the first generation open-bed PFBC/CAES plants have attractive performance and economic characteristics, future growth versions have an even greater potential to further reduce costs and lower fuel consumption. Promising advancements for second and, possibly, third generation plants include increasing bed temperature by about 90 F (50 C), increasing bed pressure by about 5 atm (bar), adding a very high pressure bed (60 atm (bar)) to increase the expansion turbine inlet temperature, and adding air heater tubes within the bed.

Because of the major benefits associated with these near-term and long-term options, the research and development efforts needed to expedite early commercialization should be energetically pursued. This preconceptual design effort uncovered several critical areas where key design information is lacking. One such area concerns part-load operation of the PFBC. Additional studies are required to define the performance of the open-bed combustor at reduced load. First, the steady-state part-load operation must be investigated with particular

In closing, it appears that coal-fired, open bed PFBC/CAES power plants could become an attractive alternative for future power generation. No insurmountable technical problem areas remain, and much of the technology required for this application is being developed as part of the PFBC programs for combined cycle applications. Appropriate refocusing of selected technology aspects could accelerate commercialization of this concept and lead to the most attractive power plant design.

Appendixes

APPENDIX A

CONVERSION FACTORS

The source of these conversion factors from English to System International (SI) units is Ref. A-1. To calculate the magnitude of a quantity in SI units multiply the quantity expressed in English units by the factor indicated e.g. 1 foot x .3048 = .3048 meters.

Area

square meter (m^2) = acres x 4.047×10^3

hectare = acres x .4047

Energy

kilojoule (kJ) = British thermal unit (Btu) x 1.055

Energy Density

kilojoule per kilogram (kJ/kg) = British thermal unit per pound mass (Btu/lb)
x 2.326

kilogram per gigajoule (kg/GJ) = pound mass per 10^6 British thermal unit
(lb/ 10^6 Btu) x 0.4299

Heat Rate

kilojoule per kilowatt hour (kJ/kWh) = British thermal unit per kilowatt hour
(Btu/kWhr) x 1.055

Length

meter (m) = foot (ft) x 0.3048

millimeter = inch (in.) x 25.40

= mil (mil) x 25.40×10^{-3}

micron = inch (in.) x $25.40 \times 10^{+3}$

= mil (mil) x 25.40

Mass

$$\text{kilogram (kg)} = \text{pound mass (lb)} \times 0.4536$$

$$\text{metric ton (t)} = \text{short ton (ton)} \times 0.9072$$

Mass Density

$$\text{kilogram per cubic meter (kg/m}^3\text{)} = \text{pound mass per cubic foot (lb/ft}^3\text{)}$$

$$\times 16.018$$

$$\text{gram per cubic meter (g/m}^3\text{)} = \text{grain per cubic foot (grains/ft}^3\text{)} \times 2.288$$

Mass Flux

$$\text{kilogram per second} \quad \text{pound mass per second}$$

$$\text{per square meter (kg/s-m}^2\text{)} = \text{per square foot (lb/sec-ft}^2\text{)} \times 4.8825$$

Power

$$\text{watt (W)} = \text{British thermal unit per hour (Btu/hr)} \times .2931$$

Power Density

$$\text{megawatts per cubic meter (MW/m}^3\text{)} = \text{British thermal unit per hour cubic foot}$$

$$(\text{Btu/hr-ft}^3) \times 10.35 \times 10^{-6}$$

Power Flux

$$\text{megawatts per square meter (MW/m}^2\text{)} = \text{British thermal unit per hour}$$

$$\text{per square foot (Btu/hr-ft}^2\text{)} \times$$

$$3.15 \times 10^{-6}$$

Power Flux/Temperature

$$\text{watt per square meter} \quad \text{British thermal unit per square foot}$$

$$\text{per kelvin (W/m}^2\text{-K)} = \text{per Farenheit (Btu/hr-ft}^2\text{-F)} \times 5.678$$

Pressure

$$\text{bar (bar)} = \text{atmosphere (atm)} \times 1.0133$$

Speed

$$\text{meter per second (m/s)} = \text{foot per second (ft/sec)} \times .3048$$

Temperature

$$\text{Celsius (C)} = [\text{Fahrenheit (F)} - 32(\text{F})] \times 5/9$$

$$\Delta \text{ Celsius (C)} = \Delta \text{ Fahrenheit (F)} \times 5/9$$

$$\Delta \text{ Kelvin (K)} = \Delta \text{ Fahrenheit (F)} \times 5/9$$

Volume

$$\text{cubic meter (m}^3\text{)} = \text{gallon (gal)} \times 3.785 \times 10^{-3}$$

Prefixes

$$\text{giga (G)} = 10^9$$

$$\text{mega (M)} = 10^6$$

$$\text{kilo (k)} = 10^3$$

$$\text{milli (m)} = 10^{-3}$$

Reference

A-1 Mechtly, E. A.: The International System of Units - Physical Constants and Conversion Factors. NASA SP-7012, 1973.

APPENDIX B

LEVELIZED ENERGY COST

In recent years, it has become common to express electric energy costs as levelized costs over a selected time period. Levelized costing arose from a need by the electric utilities to account for rapidly escalating investments and fuel costs in their cost comparisons of alternative power plant systems and the inability of the more simplistic nonlevelized approach, which had been utilized prior to the present period of rampant inflation, to meet this need. At least two utility economic methodologies (Refs. B-1 and B-2) have been developed which perform this function. These different, but closely related, economic methodologies take advantage of convenient closed-form mathematical relationships to transform annual costs and investments into levelized costs. In this study program, the methodology of Ref. B-2, as interpreted by Ref. B-3, was utilized. Prior to presenting the methodology, it is appropriate to briefly explain the levelized cost concept as applied in an electric utility context.

Levelized Cost Methodology

In evaluating the possible addition of a power plant to the utility system, the utility must consider that over the lifetime of the power plant it will be liable for the payment of the expenses associated with generating power from that power plant. These expenses will vary from year to year because of many factors, one of which is escalating costs. If the available data on annual expenses are merely tabulated, it is difficult, if not impossible, to compare the cost effectiveness of adding the power plant relative to adding an alternative power plant. A solution does exist to this problem, however, wherein a common business approach is applied to quantify the series of unequal expenses which occur over the power plant lifetime, thus providing a basis for comparison. This approach discounts each of the annual expenses at a suitable rate (usually the cost of capital for a utility) back to a base time period. The summation of all these discounted annual expenses is called the present value of the series of expenses. By comparing the present value associated with each of the alternative power plant options, the utility could select the most cost effective option, i.e., the option with the lowest present value of expenses.

The approach as stated above is inconvenient for preliminary analyses because quite often a schedule of expenses throughout the lifetime of the power plant is unavailable. Typically only the capital investments and a representative annual cost have been estimated. However, if the assumption is made that the costs experienced during representative year are equal to the real costs experienced every year during the lifetime of the power plant (i.e., the only

variation in the dollar magnitude of the annual costs is due to escalation) then it is possible to construct a series of annual costs and discount that series of costs to yield a present value. Because the real annual costs are equal (held level), a present value estimated in this fashion is called a levelized cost. This levelized cost, because it is a present value, can then be used to compare alternative power plant systems.

Levelized Busbar Energy Cost

The levelized busbar energy cost, measured in terms of base-year price levels (mills/kWh) is calculated according to the following equation:

$$\overline{\text{BBEC}} = \left[\frac{1}{1+i} \right]^{(co-x)} \frac{(FCR \times CI_{co}) + (F_{co} + CE_{co} + OM_{co}) \left[\frac{k}{1-(1+k)^{-N}} \right]}{.001 \times CF \times RP \times 8760} \quad (B-1)$$

where,

$\overline{\text{BBEC}}$ = levelized busbar energy cost in base-year price levels (mills/kWh)

i = general annual inflation rate (1/yr)

co = first year of commercial operation

x = base year for price estimates

FCR = fixed charge rate (1/yr)

CI_{co} = present value of capital investment as of January 1 of the first year of commercial operation (\$)

F_{co} = present value of total fuel costs as of January 1 of the first year of commercial operation (\$)

CE_{co} = present value of total compression energy costs as of January 1 of the first year of commercial operation (\$)

OM_{co} = present value of total operating and maintenance costs as of January 1 of the first year of commercial operation (\$)

k = cost of capital (1/yr)

N = system operating lifetime in years

.001 = conversion factor (\$/mills)

CF = capacity factor

RP = rated power output (kW)

8760 = conversion factor (hrs/yr)

A listing of standard economic assumptions are supplied in Table B-1.

Capital Investment Cost

The present value of the total capital investment as of January 1 of the year of first commercial operation (CI_{CO}) is the measure used for total capital investment for the power plant system under evaluation. It is calculated as the sum of three components, as follows:

$$CI_{CO} = (CI_x + EPC) + f_{EDC} (CI_x + EPC) + f_{IDC} (CI_x + EPC) \quad (B-2)$$

where, CI_x = capital investment in base-year price levels (\$)

$$\begin{aligned} &= TDC + f_{CONT} TDC + f_{EM} (TDC + f_{CONT} TDC) \\ TDC &= \text{total direct cost in base-year price levels (\$)} \\ f_{CONT} &= \text{contingency factor} \\ f_{EM} &= \text{engineering and management factor} \end{aligned} \quad (B-3)$$

EPC = escalation prior to construction (\$)

$$= CI_x [(1+g_c)^{(sc-x)} - 1] \quad (B-4)$$

g_c = escalation rate for capital items (1/yr)

sc = year of start of construction

x = base year for price estimates

$(CI_x + EPC)$ = total capital investment in the year at start of construction (\$)

f_{EDC} = escalation during construction factor

f_{IDC} = interest during construction factor

TABLE B-1

ECONOMIC ASSUMPTIONS FOR ANALYSIS⁽¹⁾

<u>Parameter</u>	<u>Units</u>	<u>Value</u>	<u>Source</u>
Fixed charge rate, FCR	1/yr	0.18	(2)
Discount rate, k	1/yr	0.10	(2)
Inflation rate, i	1/yr	0.06	(2)
Escalation rates			
petroleum fuels, g_{fp}	1/yr	0.08	(2)
coal, g_{fc}	1/yr	0.07	(2)
baseload electricity, g_{ce}	1/yr	0.07	(3)
capital equipment/construction, g_c	1/yr	0.06	(4)
operating & maintenance costs, g_{om}	1/yr	0.06	(4)
Energy prices			
distillate #2, FP_{xd}	$\$/10^6$ Btu	5.24	(5)
residual oil, FP_{xr}	$\$/10^6$ Btu	4.33	(5)
coal, FP_{xc}	$\$/10^6$ Btu	1.34	(5)
baseload electricity, CEP_x	mills/kWh	11.4	(6)
System lifetime, N	years	30	
Construction period, (co-sc)	years	5	
Shape of construction payout curve		bell-shaped right-skewed	(7)
Start of operations, co		1990	
Base year for cost estimates, x		1980	

-
- (1) From Ref. B-3
- (2) Based upon information in Ref. B-2.
- (3) Based upon expected escalation rate for the incremental generating cost of electricity from a mix of 1/3 nuclear and 2/3 coal.
- (4) Assumed the same as the general inflation rate.
- (5) 1 January 1980 purchase prices in 1980 dollars. Ref. B-2 escalated at 10% for one year.
- (6) 1 January 1980 incremental generating cost in 1980 dollars. Based upon the incremental generating cost for power using a 1/3 nuclear, 2/3 coal mix. Ref. B-2 escalated for one year at 10%.
- (7) Half-cost flow at 60% of construction period.

The f_{EDC} and f_{IDC} factors account for escalation and interest during construction, respectively, for a right-skewed payout curve with half-cash flow occurring at 60 percent of the construction period. Values for both f_{EDC} and f_{IDC} are listed in Table B-2. Note that f_{IDC} already accounts for the additional interest required to finance the incremental escalation during construction.

Fuel Cost

The present value (as of 1 January of the first year of commercial operation) of the total fuel costs over the system's operating lifetime is the measure used for fuel cost for the system under investigation. It is calculated as follows:

$$F_{CO} = AFC_x (1+g_f)^{(co-x)} \left[\frac{1+g_f}{k-g_f} \right] \left[1 - \left(\frac{1+g_f}{1+k} \right)^N \right] \quad (B-5)$$

where, F_{CO} = present value of total fuel costs as of 1 January of the first year of commercial operation (\$)

AFC_x = annual cost of fuel in base-year price levels (\$)

$$= FP_x \times FHR \times CF \times RP \times 8760 \quad (B-6)$$

FP_x = base-year price of fuel (\$/10⁶ Btu)

FHR = fuel heat rate (Btu/kWh)

CF = capacity factor

RP = rated power output (kW)

8760 = conversion factor (hrs/yr)

g_f = escalation rate for fuel prices (1/yr)

co = first year of commercial operation

x = base year for price estimates

k = cost of capital (1/yr)

N = system operating lifetime in years

TABLE B-2
VALUES FOR f_{IDC} AND f_{EDC}

Construction Period (Year)	Escalation Rate (%/Yr)	f_{EDC}	f_{IDC}						
			Interest Rate (%/Yr)						
			6%	7%	8%	9%	10%	11%	12%
3	4	0.072	0.080	0.093	0.107	0.121	0.135	0.148	0.162
	5	0.090	0.081	0.095	0.109	0.123	0.136	0.150	0.165
	6	0.109	0.082	0.096	0.110	0.124	0.138	0.153	0.167
	7	0.128	0.083	0.097	0.112	0.126	0.140	0.155	0.169
	8	0.146	0.085	0.099	0.113	0.126	0.142	0.157	0.172
	9	0.166	0.086	0.100	0.115	0.129	0.144	0.159	0.174
	10	0.185	0.087	0.102	0.116	0.131	0.146	0.161	0.176
4	4	0.097	0.110	0.129	0.149	0.168	0.188	0.207	0.227
	5	0.122	0.112	0.132	0.151	0.171	0.191	0.211	0.232
	6	0.148	0.115	0.134	0.154	0.174	0.195	0.215	0.236
	7	0.174	0.117	0.137	0.157	0.178	0.198	0.219	0.240
	8	0.200	0.119	0.139	0.160	0.181	0.202	0.223	0.245
	9	0.227	0.121	0.142	0.163	0.184	0.206	0.227	0.249
	10	0.255	0.123	0.145	0.166	0.188	0.210	0.232	0.254
5	4	0.123	0.143	0.168	0.193	0.219	0.245	0.272	0.299
	5	0.155	0.146	0.172	0.198	0.224	0.251	0.278	0.306
	6	0.189	0.150	0.176	0.202	0.229	0.257	0.285	0.313
	7	0.223	0.153	0.180	0.207	0.235	0.263	0.291	0.320
	8	0.257	0.157	0.184	0.212	0.240	0.269	0.298	0.328
	9	0.293	0.160	0.188	0.217	0.246	0.275	0.305	0.355
	10	0.329	0.164	0.193	0.222	0.252	0.282	0.312	0.343
6	4	0.149	0.177	0.209	0.241	0.274	0.308	0.342	0.377
	5	0.190	0.182	0.215	0.248	0.282	0.316	0.351	0.387
	6	0.231	0.188	0.221	0.255	0.290	0.325	0.361	0.398
	7	0.273	0.193	0.227	0.262	0.298	0.335	0.372	0.409
	8	0.317	0.198	0.234	0.270	0.306	0.344	0.382	0.421
	9	0.362	0.204	0.240	0.277	0.315	0.354	0.393	0.433
	10	0.408	0.210	0.247	0.285	0.324	0.363	0.404	0.445
7	4	0.177	0.214	0.253	0.293	0.334	0.375	0.418	0.462
	5	0.225	0.221	0.261	0.303	0.345	0.388	0.432	0.477
	6	0.275	0.229	0.270	0.313	0.356	0.401	0.446	0.493
	7	0.327	0.236	0.279	0.323	0.368	0.414	0.461	0.509
	8	0.380	0.244	0.288	0.334	0.380	0.428	0.476	0.526
	9	0.435	0.252	0.298	0.345	0.393	0.442	0.492	0.543
	10	0.493	0.261	0.308	0.356	0.405	0.456	0.508	0.561

Compression Energy Cost

The present value (as of January 1 of the first year of commercial operation) of the total compression energy cost over the CAES system's operating lifetime is the measure used for compression energy cost for the CAES system under investigation. It is calculated as follows:

$$CE_{co} = ACEC_x (1+g_{CE})^{(co-x)} \left[\frac{1+g_{CE}}{k-g_{CE}} \right] \left[1 - \left(\frac{1+g_{CE}}{1+k} \right)^N \right] \quad (B-7)$$

where, CE_{co} = present value of total compression energy cost as of
1 January of the first year of commercial operation (\$)

$ACEC_x$ = annual cost of compression energy in base-year-x price
levels (\$)

$$= CEP_x \times .001 \times EER \times CF \times RP \times 8760 \quad (B-8)$$

CEP_x = base-year incremental cost of generating compression
energy (mills/kWh)

.001 = conversion factor (\$/mills)

EER = electric energy ratio (kWh-in/kWh-out)

CF = capacity factor

RP = rated power (kW)

8760 = conversion factor (hrs/yr)

g_{CE} = escalation rate for the incremental cost of baseload
electricity generation (1/yr)

co = first year of commercial operation

x = base year for price estimates

k = cost of capital (1/yr)

N = system operating lifetime in years

Operating and Maintenance Costs

The present value (as of January 1 of the first year of commercial operation) of the operating and maintenance (O&M) costs is the measure for O&M costs for the system under investigation. It is calculated as follows:

$$OM_{co} = OMC_x (1+g_{OM})^{(co-x)} \frac{1+g_{OM}}{k-g_{OM}} \left[1 - \frac{1+g_{OM}}{1+k} \right]^N \quad (B-9)$$

where,

OM_{co}	= present value of total operating and maintainance costs as of 1 January of the first year of commercial operation (\$)
OMC_x	= annual cost of operating and maintenance in base-year price levels (\$)
g_{OM}	= escalation rate for operations and maintenance costs (1/yr)
co	= first year of commercial operation
x	= base year for price estimates
k	= cost of capital (1/yr)
N	= system operating lifetime in years

Levelized Annual Energy Cost

The levelized annual energy cost, expressed in terms of base-year price levels is calculated according to the following equation:

$$\overline{AEC} = \overline{BBEC} \times \frac{CF \times 8760}{1000} \quad (B-10)$$

where, $\overline{AEC}$ = levelized annual energy cost in base-year price levels (\$/kWyr)

$\overline{BBEC}$ = levelized busbar energy cost in base-year price levels (mills/kWh)
 CF = capacity factor
 8760 = conversion factor (hrs/yr)
 1000 = conversion factor (mills/\$)

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