FY-1977 Progress Report Compressed Air Energy Storage Advanced Systems Analysis

by
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M. A. McKinnon**

March 1978

Pacific Northwest Laboratory
Richland, Washington 99352
Operated for the
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ADVANCED SYSTEMS ANALYSIS

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SUMMARY

The goal of the Compressed Air Energy Storage (CAES) Advanced Systems Analysis task is to accelerate the development of new technologies that will reduce the consumption of natural gas and oil. The immediate overall objectives of this program are:

- Provide a screening cost assessment for thermal energy storage (TES) systems that are suitable for CAES applications.
- Establish the potential fuel savings of hybrid CAES cycles that incorporate TES for recovery of the heat of compression and estimate the economic incentive for using TES in CAES systems.
- Investigate modified CAES cycles that eliminate the use of gas and oil by the use of alternative fuels.

This document is the FY-1977 Annual Report of the Pacific Northwest Laboratory (PNL) for work performed in the Advanced Systems Analysis program. This is an interim report of work in progress; therefore, results and conclusions should be considered tentative at this time.

TES ASSESSMENT

The TES assessment identifies three alternatives that would be suitable for CAES applications. In order of estimated overall suitability, these are:

- pebble beds,
- pressurized hot water, and
- high temperature porous reservoirs (aquifers).

The estimated sums of the major construction costs for all three of these systems were very nearly the same based on 2000 MW-hr(t) and 600°F storage exclusive of all auxiliaries. Pebble beds appear to be the most practical TES systems to implement. Although estimated costs are all approximately equal, both hot water and aquifer systems involve greater uncertainties and, therefore, greater risks.

Cost estimates were also prepared for TES systems based on heat storage in hot oil and in fusible salts. Both of these alternatives are estimated to cost substantially more than the recommended alternatives and both involve numerous technical difficulties. For these reasons, neither hot oil nor molten salts are considered to be attractive alternatives for CAES applications.

ASSESSMENT OF CAES CYCLES WITH TES

The major effort in this task was devoted to a parametric thermodynamic analysis of CAES systems with TES. Three distinct designs were considered. They are:

- the base system (Huntorf style cycle, but with exhaust heat recuperation),
- the single-stage system (base system with a single TES), and
- the double-stage system (base system with two stages of TES).

The base system represents state-of-the-art technology and is therefore the standard against which the cycles with TES are compared. The cycle analysis was performed to establish the degree of fuel savings that could be achieved with TES and the performance trade-offs that this would entail.

For single-stage systems, the addition of TES can result in turbine fuel savings of up to 20% in an optimized CAES system. For an optimized double-stage CAES system, fuel savings can be as high as 46%. For the two-stage adiabatic cycle, no fuel is used. However, increased pumping requirements and increased physical size of the system mitigate against the adiabatic concept at this time.

Effective utilization of TES results in a number of design and performance penalties or trade-offs. These are:

- The input compressor power is increased.
- The turbine airflow requirement is increased.
- The latter results in larger compressors, turbines, reservoirs, etc.
- The system becomes complex, subject to severe operational constraints.

The conclusion is that fuel savings can be attained by the addition of TES, but only at the price of increased system size and complexity, reduced overall energy efficiency, and increased cost.

In terms of economics, the value of the turbine fuel saved must be sufficient to pay for the additional compression energy plus the additional costs of the TES reservoir(s) and other changes or additions to the CAES facility. A detailed economic assessment of systems with TES has not been completed. However, rough economic comparisons, based on costs reported in previous work, have been done.

The tentative result is that the economic value of the turbine fuel conserved minus the cost of additional compression energy can account for no more than half of the cost of the TES reservoir alone without regard for the additional costs for other elements of the CAES system. Therefore, the tentative conclusion is:

- There is currently no apparent economic incentive for the inclusion of TES in a CAES system.

Because this is a preliminary conclusion based on incomplete data, the following qualification and recommendation is given:

- The economics of including TES should be given more thorough treatment and costs should be extrapolated to future conditions.

FLUIDIZED BED COMBUSTION SYSTEMS

A preliminary analysis of a Fluidized Bed Combustion (FBC)/CAES system was performed. When operated as an air heater, the FBC has airflow requirements that are compatible with CAES, and the system should have excellent turndown (load following) capability. The conceptual design suggested by this study would have a turbine/combustor system which runs continually, driving air compressors or generators as needed. The system provides a clean, efficient means for burning coal or other fuels, thereby eliminating the need for natural gas or oil.

The conclusion of this study is that the FBC/CAES concept has excellent potential and should be pursued. The plan for the Advanced Systems Analysis program for FY-1978 is to shift emphasis to further evaluate this concept. Work planned will concentrate on cyclic performance analysis and a technical and economic feasibility assessment.

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1.0 INTRODUCTION

Large scale energy storage systems are useful in an electric utility grid as a means for load leveling, i.e., transferring energy from periods of excess capacity to periods of excess demand. In most utility systems, storage can provide greater flexibility, more efficient utilization of capacity, conservation of premium fossil fuels, and ultimately, lower cost for the consumer. In the future, provision for storage will be even more important for the effective utilization of alternative energy technologies. For example, the intermittent availability of solar energy could result in fluctuations in production and usage that might be catastrophically disruptive to electrical power generation systems as they now exist. Since the U.S. is committed to the development of alternative resources, there is a definite need for parallel development of storage systems.

Pumped Hydro Energy Storage (PHES) is the only large scale storage concept in broad use at this time. PHES is a proven load leveling system and will certainly continue as a prime candidate for those utilities which have suitable sites for development. However, problems related primarily to site availability and licensing are severely impeding broader implementation of PHES. Thus, alternative energy storage options are needed and are actively being sought.

Underground Pumped Hydro Storage (UPHS) and Compressed Air Energy Storage (CAES) are currently considered to be the only other viable candidates for utility applications, at least for the near future. In recent years, several detailed technical and economic analyses of CAES have been performed⁽¹⁻⁴⁾ which conclude that this technology is technically feasible and economically attractive at the present time and will be in the foreseeable future. However, implementation of CAES has only recently begun, with the first plant⁽⁵⁾ scheduled to go on line near Huntorf, West Germany in late 1977. A second plant in West Germany is in the planning stage.⁽⁴⁾ In the U.S., a joint DOE/EPRI program⁽⁶⁾ is currently being negotiated which may result in as many as three contracts for detailed design and analysis of CAES systems.

Assuming positive results, this program could ultimately lead to one or more CAES demonstration projects with construction beginning as early as 1980 or 1981. Therefore, it seems highly likely that development and implementation of CAES technology will proceed at an accelerated pace in the very near future.

There is, however, a potential weakness in the CAES concept that makes its long-term prospects open to serious question. Since the CAES concept is basically a modified gas turbine (Brayton) cycle, a significant portion of the power produced by the system is derived from the combustion of liquid or gaseous fossil fuels. The CAES system is a step in the right direction in that it reduces the consumption of turbine fuel to about one-third to one-half of that required by conventional gas turbine generators without storage. However, because of increasing demand, decreasing reserves, and the continuing threat of politically motivated embargo, it is questionable whether fossil fuels (other than coal) will long remain a viable option for the production of electricity. Therefore, the continued, although reduced, dependence on turbine fuels in the "conventional" CAES concept is a potential flaw that could discourage its future development and implementation and might eventually cause the abandonment of plans to construct facilities that require these fuels.

The need for advanced CAES concepts which reduce and ultimately eliminate the dependence on fossil fuels⁽⁷⁻⁹⁾ is thus apparent. The purpose of the work summarized in this report is to examine some of the advanced CAES concepts which promise to reduce strategic fuel consumption through more efficient operation and/or by substitution of alternative fuels and technologies. We retain the fundamental features of the CAES concept while considering alternatives for various elements in the cycle. The strategy of the work is to first provide conceptual designs for advanced cycles and to assess the potential improvement of thermodynamic performance. As a source of comparison, we define a base design which adequately represents the state of the art for conventional CAES technology. We then provide preliminary economic analyses of those particular modifications which appear to provide significant progress toward the reduction of fossil fuel consumption.

In the analysis which follows, we discuss the major advanced CAES concepts which currently appear to have potential merit. Each concept is briefly described, the current status of work is summarized, and the organizations responsible for each task are identified. Work currently in progress at the Pacific Northwest Laboratory (PNL) is, of course, emphasized.

2.0 CONCLUSIONS AND RECOMMENDATIONS

The primary effort in the CAES Advanced Systems Analysis task for FY-1977 was concentrated in three subject areas:

- economic assessment of TES options for CAES,
- assessment of CAES cycles with TES for recovery of the heat of compression, and
- alternative fuels and technologies.

This report summarizes the principal conclusions and recommendations for each of the subject areas. As this is an interim report of work yet in progress, results and conclusions are preliminary.

2.1 ECONOMIC ASSESSMENT OF TES OPTIONS

A survey of the available literature on TES technology yielded a relatively small number of candidates which appear to have potential for CAES applications. To limit the scope of the problem, the best candidates were screened according to gross technical compatibility with CAES requirements. This narrowed the field to five technologies which appear to have the greatest potential.

A conceptual design was established for each TES system, based primarily on the design of systems described in the literature, but adapted to CAES applications. A rough cost estimate was then computed for each system. A screening of the candidate systems, based on technical and economic considerations, yields the following overall ranking:

- pebble beds,
- pressurized hot water,
- high temperature porous reservoirs (aquifers),
- hot oil, and
- fusible salts.

Pebble beds, at an estimated cost of \$3/kWh(t), appear to be the best all around TES system for a CAES application. A pebble bed would probably be simplest to build and operate with a minimum of safety problems (i.e., as compared with hot water or hot oil).

The upper bound on the estimated cost of TES in hot water was found to be only slightly higher than that of a pebble bed. However, the cost difference between the pebble bed and hot water systems is well within the error band of the cost estimates produced in this study. Therefore, no definite conclusions regarding the relative costs of these systems can safely be drawn at this point. However, the hot water system would be more complex to build and operate, and more stringent safety precautions would probably be required. With these considerations, it is concluded that hot water TES systems are less attractive than pebble beds but should not be ruled out at this time.

The estimated incremental cost for TES by hot air injection into an aquifer is essentially the same as for pebble beds and hot water. Thus, when the air reservoir is an aquifer, thermal storage by hot air injection would probably cost about the same as storage in a separate (i.e., pebble bed) TES reservoir. Separate TES systems might conceivably be used with an aquifer air reservoir; but, in view of the added complexity, direct hot injection in the aquifer would probably be more practical. Therefore, we conclude that for a system where the air reservoir is an aquifer, heat storage by hot air injection appears to be economically competitive with the best separate TES options considered.

The estimated cost for TES in hot oil is significantly higher than for the other options discussed above, due primarily to the high cost of the oil. In addition, an air to oil heat exchange process at temperatures relevant to CAES poses a potential fire hazard which would not be easy to eliminate. For these reasons, TES in oil does not appear attractive for CAES, and further consideration of this option is not recommended.

The estimated costs for TES in fusible salts is significantly higher than for the best options discussed above, due primarily to the estimated cost of heat exchangers. This technology is still so undeveloped that more

R&D is required before a meaningful assessment is possible. However, the problem of heat transfer from air to a two-phase bed is not likely to be resolved without considerable complexity and cost. Thus, barring a technological breakthrough, TES in fusible salts is not likely to be practical for CAES.

2.2 ASSESSMENT OF CAES CYCLES WITH TES

The major effort in this task was devoted to a parametric thermodynamic analysis of CAES systems with TES. For this phase of the analysis, three distinct designs were considered. They are:

- the base system (Huntorf style cycle, but with exhaust heat recuperation),
- the single-stage system (base system with a single TES), and
- the double-stage system (base system with two stages of TES).

The base system represents state-of-the-art technology with which the advanced systems would have to compete. The base system is therefore the standard with which the cycles with TES are compared for both performance and cost. The double-stage cycle in the extreme case represents the adiabatic or no-fuel cycle where the only source of energy to the cycle is the electrical input to the compressors. All other cycles considered lie between these extremes.

The cycle analysis was performed to establish the degree of fuel savings that could be achieved with TES and the performance trade-offs that this would entail. In each case, the fuel savings is computed relative to a base cycle which operates under essentially the same cyclic parameters but without TES. For single-stage systems the addition of TES can result in turbine fuel savings of up to 20% in a properly optimized CAES system. For the double-stage system, fuel savings can be as high as 46%. For the two-stage adiabatic cycle, no fuel is used. However, increased pumping requirements and increased physical size of the system mitigate against the feasibility of the adiabatic cycle at this time.

Effective utilization of TES results in a number of design and performance penalties or trade-offs. These are:

- The input compressor power is increased approximately in proportion to turbine fuel saved.
- The turbine expansion ratios are altered with the consequence that the airflow requirement is increased.
- The latter results in larger compressors, turbines, reservoirs, etc.
- The system becomes complex, subject to severe operational constraints; and considerable new equipment development may be required.

The conclusion to be reached is that fuel savings can be attained by addition of TES, but only at the price of increased system size and complexity and reduced overall energy efficiency.

In terms of economics, the value of the turbine fuel saved must be sufficient to pay for the additional compression energy plus the additional fixed costs. The fixed costs include the costs of the TES reservoir(s) and other changes or additions to the CAES facility. A detailed economic assessment of systems with TES has not been completed. However, rough economic comparisons, based on costs reported in the literature, have been done. The calculations indicate that the value of the turbine fuel saved minus the cost of additional power consumed in the compressors can account for less than half of the cost of the TES reservoir alone, disregarding additional costs that would be incurred in the surface plant. Therefore, the preliminary conclusion is:

- There is currently no apparent economic incentive for the inclusion of TES in a CAES system.

Because this is a preliminary conclusion based on incomplete data, we include the following qualification and recommendation:

- The economics of including TES should be given more thorough treatment and costs should be extrapolated to future conditions.

2.3 ALTERNATIVE FUELS AND TECHNOLOGIES

Synthetic fuels derived from coal, waste or other sources should be directly applicable to CAES applications with little or no modification. The problems in this case relate to the production and cleanup of the fuels. The economics of CAES with synthetic fuels versus other options for peaking will depend primarily on the relative costs of fuel and capital at the time such options become competitive.

Certain coal utilization technologies, such as coal gasification, fluidized bed combustion (FBC), and magneto-hydrodynamics (MHD) may be effectively integrated with CAES technologies. Of these, the FBC/CAES option was determined to have the best near-term potential and preliminary analysis of this concept was initiated in FY-1977.

When operated as an air heater, the FBC has airflow requirements that are compatible with CAES, and the system should have excellent turndown (load following) capability. The conceptual design suggested by this study would have a turbine/combustor system which runs continually, driving air compressors or generators as needed. The system provides a clean, efficient means for burning coal or other fuels, thereby eliminating the need for gas.

The conclusion of this study is that the FBC/CAES concept has excellent potential and should be pursued. The plan for the Advanced Systems Analysis program for FY-1978 is to shift emphasis to this area. Work planned will concentrate on cyclic performance analysis and a technical and economic feasibility assessment.

3.0 BACKGROUND

The primary objective of the Advanced Systems Analysis project is to minimize the dependence of CAES plants on premium fuels such as gas or oil. This may be done through improved cycle design, the use of alternate fuels, and the utilization of alternate technologies.

3.1 IMPROVED CYCLE DESIGN

The first logical step in reducing the consumption of fossil fuel is to better utilize the available energy by improved cycle design. If we use the Huntorf CAES design as an example, there are several obvious ways of improving the basic design.

- Provide an exhaust heat recuperator.
- Provide thermal energy storage to conserve the heat of compression.
- Minimize the throttling losses during charge and discharge.
- Optimize turbine and compressor designs, cycle pressures and temperatures.

All of the above developments can lead to improved fuel efficiency regardless of the fuel employed. However, optimization must include the overall economics of the system. A primary task of the Advanced Systems Analysis program at PNL is to investigate the performance characteristics of advanced CAES cycles because this data will be needed to properly assess the economics of the various alternatives.

There is additional work on thermodynamic cycle analysis currently in progress at the University of West Virginia. The design characteristics and economics of advanced turbomachinery systems for CAES applications are primary concerns of work currently in progress at ANL. The ongoing work at PNL concerning the overall economic aspects of advanced CAES systems will draw on these studies where possible. This is an interim report that deals primarily with the cycle thermodynamic performance. The economic analysis is in progress and will be dealt with more thoroughly in the FY-1978 annual report.

3.2 ALTERNATIVE FUELS

The second major class of advanced CAES systems would eliminate the need for fossil fuels by substitution of alternative fuels. In this category we recognize the possibility of synthetic fuels derived from coal, oil shale, biomass and waste. Although modifications may be required to adapt combustors and turbines for the use of synthetic fuels,⁽¹¹⁾ the bulk of the CAES system should be conventional and its potential viability would depend primarily on fuel cost and resultant delivered cost of power produced.

One important alternative fuel possibility is the combustion of low Btu gas produced from coal. This is an attractive option from the user's point of view because gas is extremely convenient to work with. In addition, the availability of synthetic gas would guarantee extended service for extensive facilities now in place that utilize the existing natural gas distribution system. There is little doubt that existing gas turbines may be modified to burn low Btu gas. However, the crucial question is whether or not it will be acceptable to burn synthetic gas for electric power generation in a CAES system.

The price of synthetic fuels is still highly speculative and significantly new technology is not required (other than that related to fuel production). For these reasons, this aspect of Advanced CAES Systems Analysis is not being directly addressed in current work. However, we may consider this option indirectly by including the effects of fuel costs in the general economic assessment of CAES technology.

3.3 ALTERNATIVE TECHNOLOGIES

The third major class of advanced CAES systems under consideration draws upon several radically different technologies which would utilize direct combustion of alternative fuels. We retain many of the basic features of the CAES concept. However, the major components of the surface plant, including compressors, combustors and expanders, may be modified to best suit the fuel (or alternative energy source) being considered. In this way we may propose a number of technological "hybrids" which incorporate the essence of the CAES concept while eliminating the dependence on fossil fuels.

3.3.1 Coal Combustion and Gasification Technologies

The most important alternative fuel for the U.S. in the near-term and beyond is unquestionably coal. Nuclear energy will probably continue to grow in volume and as a proportion of our total generation base, but is likely to occupy a secondary position relative to coal at least through the end of this century. Commercial implementation of fusion reactors or solar electric generators is probably at least 30 years away and no other alternatives are likely to make significant contributions in the foreseeable future. It is apparent that base loaded generation capacity will soon be predominantly coal fired, probably before the end of this century. Therefore, it is sensible to give high priority to alternative CAES technologies that could also utilize coal.

The possibility of firing gas turbines directly with coal in various forms has been tried unsuccessfully in the past.⁽¹¹⁾ The principal problem has been corrosion and erosion of the blading due to several different impurities and the highly abrasive ash in the combustion products. Although work is continuing on the hot gas cleanup problem, there is little reason to believe that direct coal firing of gas turbines will soon be a practical option. This aspect of the problem is not currently under direct consideration in this program.

The use of gasified or liquified coal as a fuel should require little or no additional development of gas turbine technology. When these fuels become available, their suitability for CAES applications will primarily be a question of fuel cost, as noted in Section 3.2.

3.3.1.1 CAES/Coal Gasification

It has been suggested that a hybrid CAES/Coal gasification facility might be practical.⁽¹²⁾ This system might produce synthetic gas for general distribution in addition to electric power production for peaking duty. This cycle may or may not draw compression power from the grid. Analysis of this concept will be performed in programs being initiated by the Electric Power Research Institute.

3.3.1.2 Magneto-Hydrodynamics

Magneto-hydrodynamics (MHD) is a possible means for the direct combustion of coal which might integrate well with certain aspects of the CAES concept. The throughput of air in a utility scale MHD generator will probably be comparable to that of a similar gas turbine facility. If so, MHD has the potential for CAES applications. However, the development of MHD technology has not yet reached the stage where its possibilities for utility applications may be assessed. Therefore, an MHD/CAES hybrid is at best a long range option that offers no potential for the near term. A brief preliminary assessment of this option will be performed by PNL in FY-1978.

3.3.1.3 Fluidized Bed Combustion

Fluidized bed combustion (FBC) systems offer a promising new technology for the combustion of coal (and other fuels as well). Fluidized beds are especially attractive for the direct combustion of coal because they provide the potential for high combustion efficiency and direct sulfur removal without the need for stack gas scrubbing. Fluidized beds are widely used for various industrial processes and in waste processing applications. Research on FBC systems has been modest, until quite recently, with most of the activity in England. In the past few years, R&D on FBC systems has accelerated and there are now a number of demonstration/research systems in various stages of planning, construction, and testing. Although no utility scale applications are definitely planned for construction in the near future, the apparent potential and state of development are sufficient to classify FBC systems as a near- to mid-term option for the combustion of coal.

Most FBC systems proposed to date⁽¹³⁾ are essentially boilers that would provide steam to a conventional Rankine cycle. Combined Rankine/Brayton cycles have been proposed, but the power would still be produced primarily by the Rankine cycle. Because the combined cycles are rather complex with limited turndown potential, they are best suited to base load applications.

FBC systems have also been proposed⁽¹¹⁾ which would function as air heaters in a modified Brayton cycle. These would improve turndown potential (i.e., load following capability) and would be better suited to peaking duty.

Since the airflow requirements of an FBC/Brayton cycle are comparable to flow rates in a conventional gas turbine, the FBC offers excellent potential for CAES application. This concept has been briefly considered⁽¹¹⁾ but a complete analysis of the performance and economics of this option has not yet been performed.

In Section 7.0 we report on our preliminary investigation of the performance of several FBC/CAES design options. This work will be expanded and extended to include the system economics in FY-1978. Part of this work will be done at PNL with assistance from an outside contractor. Details of the contract arrangements will be worked out in early FY-1978.

3.3.2 Solar Technologies

Use of solar energy, whether for utility scale solar electric generators or for consumer-based space and water heating, poses potentially severe problems to the operation of a utility system. Conventional CAES systems could serve well in a utility which incorporates a significant amount of solar energy use.

In addition, CAES technology might be applied more directly to solar systems. For example, one might envision the use of solar or wind powered air compressors in direct analogy to solar powered irrigation pumps recently demonstrated⁽¹⁴⁾ in the southwest. It may also be conceivable to utilize solar concentrators to preheat the air in a Brayton cycle with a resultant savings of fuel. However, both of these options would probably have to incorporate additional thermal energy storage with a resultant increase in complexity and cost. Solar based CAES systems will be investigated in FY-1978 in a separately funded program recently begun at the Massachusetts Institute of Technology (MIT).

3.3.3 Other Technologies

Another concept which falls into the alternative technologies classification is a novel CAES system proposed by researchers at ANL.⁽¹⁵⁾ The proposed "comander" system is based on a modified diesel cycle which would utilize a piston engine for both compression and expansion. The primary

potential advantages of this cycle are reported to be a high output specific work (reduced reservoir volume) and increased thermal efficiency (fuel economy). However, the concept requires a liquid or gaseous fuel which may lead to essentially the same fuel cost/availability problems that limit the conventional CAES systems. This concept will be subjected to a technical and economic feasibility analysis at ANL in a separately funded program beginning early in FY-1978.

We mention briefly a CAES concept which would utilize decay heat from spent nuclear reactor fuel. This possibility was proposed and briefly investigated at PNL as an alternative energy source for preheating air in a Brayton cycle. However, questions relating primarily to environmental acceptability were raised in an earlier study of a similar concept⁽¹⁶⁾ that tend to rule out this possibility for CAES. A brief analysis of this concept will be reported in FY-1978.

4.0 THERMODYNAMIC ANALYSIS OF IMPROVED CYCLES

We now turn to the consideration of advanced cycles which offer the potential for reduction of fuel consumption by more efficient cycle design and operation. This option is based on the use of essentially conventional CAES technology with improvements, such as compressor heat recovery and exhaust heat recuperation. In the extreme, this concept leads to the "adiabatic" CAES cycle which was the subject of an earlier study at PNL.⁽⁹⁾

The first step in assessing various options for reduction of fuel consumption is to investigate their effect on the CAES thermodynamic cycle. In this way we can identify which options offer potential thermodynamic improvement and we can quantify their potential for fuel savings or other benefits. This data will be needed for economic assessment and comparison of the various options.

In this phase of the analysis, we will consider the Huntorf cycle,⁽⁴⁾ as illustrated by Figure 4-1, and three advanced cycles that incorporate three levels of increasing complexity. We designate these as the base system, the single-stage system and the double-stage system, respectively. Abbreviated cycle diagrams for each of the three orders of complexity are shown in Figures 4-2, 4-3, and 4-4. Briefly, the three levels of complexity are characterized as follows:

- base system--Huntorf design with exhaust recuperator, but no TES capability (Figure 4-2)
- single-stage system--base system with single-stage TES (Figure 4-3)
- double-stage system--base system with two stages of TES (Figure 4-4).

These specific designs were selected from a large field of possible variants employing similar components in different sequential order. Narrowing the field of alternatives was based on the following criteria:

FIGURE 4-1. Huntorf System Design Schematic

FIGURE 4-2. Base System Advanced CAES Cycle, Huntorf System with Exhaust Recuperation

FIGURE 4-3. High Pressure Advanced CAES Cycle with Single Stage of Heat Storage

- The system should have minimized consumption of oil in the turbines.
- The system should have maximized recovery of the heat of compression.
- The system should have minimized total fuel consumption.
- The system design will probably have to be based on state-of-the-art turbines and compressors with modest modifications (as opposed to total redesign).
- The system should entail a minimum of safety, social and environmental problems.
- Unit costs (mil/kWh) for CAES should be competitive with conventional peaking facilities.

It is possible to propose a number of potential CAES configurations which, from a purely thermodynamic point of view, are equivalent to or potentially more efficient than the systems chosen for this analysis. Many such

FIGURE 4-4. High Pressure Advanced CAES Cycle with Two Stages of Heat Storage

alternatives were considered in the preliminary analysis which led to the selection of the systems described here. Some of the alternatives and the reasons for their elimination are described briefly in the following discussion. First, however, we will discuss certain common features of these systems and various simplifications employed in the figures and the analysis.

The schematic diagrams shown in Figures 4-1 through 4-4 are simplified for clarity of presentation. The following comments apply in varying degrees to all four figures.

- Valving and other control systems are not illustrated.
- Compressor coolers may be a combination of intercoolers, aftercoolers and/or jacketed coolers in configurations which might differ significantly from those shown in the figures.
- Mass flow matching requirements may require two or more booster compressors (and high pressure turbines) in parallel instead of single units as illustrated.
- Direct coupling of the system (bypassing storage) may be desirable, but will not generally be considered as a design criterion.
- The same ducting could probably be used for air injection and withdrawal by appropriate valving, as shown in Figure 4-2. The separate ducting shown in Figures 4-3 and 4-4 is for illustration only.
- The air reservoirs are illustrated in a general sense. The possibility of hydrostatic compensation is not ruled out.
- The heat reservoirs (Figures 4-3 and 4-4) are illustrated in a general sense. In the case of an aquifer, the heat reservoir could be integral with the air reservoir.
- Provision for air dehydration is not specifically illustrated. However, dehydration could be accomplished in one of the intercoolers or aftercoolers.

4.1 BASE SYSTEM

The system illustrated schematically by Figure 4-2 is the recommended first generation, high-pressure CAES system. With the exception of the exhaust recuperator, this design is essentially that of the Huntorf facility, Figure 4-1. Multiple stages of compression are required with intercooling

and aftercooling to minimize the input work and to allow air injection at or near the ambient temperature of the reservoir media. The heat of compression is thus rejected to the atmosphere in cooling towers.

As in the Huntorf design, the air recovered from storage is heated and expanded in two combustor/turbine-generator stages, producing power. However, the design described here also includes an exhaust recuperator that recovers heat from the high temperature exhaust gases to preheat the air at the inlet to the high pressure combustor.

The recuperator was not included at Huntorf because of high cost estimates and relatively low fuel prices at the time the design was finalized.⁽⁴⁾ A recuperator adequate for this application is still not readily available, but manufacturers are now ready to bid on them at projected costs much below previous estimates.⁽⁵⁾ If Huntorf were to be redesigned today with current cost data, it is highly probable that a recuperator would be included in the design. This system can be built with existing technology and "off the shelf" components requiring relatively minor alterations. Since the Huntorf system is due to go on line late in 1977, excellent data should soon be available to assess its cost and performance. The system of Figure 4-2 provides a logical basis for comparison with other advanced CAES systems.

A computer code has been prepared to model the performance of the cycles shown in Figures 4-1 through 4-4. The analysis is based on a modified Brayton cycle where each element in the system is described by a performance parameter as defined in Table 4-1 and in the nomenclature.

The performance of the turbines and compressors is defined by their respective isentropic efficiencies. Pressure losses in the various system components, including the TES systems and the air reservoir, are accounted for by pressure recovery factors. The thermal performance of the combustors, including combustion efficiency and heat loss, is characterized by a lumped combustor efficiency. Losses of heat in storage are accounted for by a reservoir temperature recovery factor. The reservoirs are assumed to be air

TABLE 4-1. Component Properties and Performance Parameters

System or Component	Parameter Identification, Description	Symbol	Value
Compressor	Isentropic compression efficiency	η_c	0.87
	Intercooler and aftercooler pressure recovery factors	δ_i	0.98-0.99
Turbine	Isentropic expansion efficiency	η_t	0.87
Combustor	Overall combustor efficiency	η_h	0.98
	Pressure recovery factor	δ_h	0.98
Recuperator	Effectiveness	ϵ_e	0.80
	Pressure recovery factor	δ_e	0.98
Motor/Generator	Input efficiency of M/G set	η_m	0.98
	Output efficiency of M/G set	η_g	0.98
Air Reservoir	Mass recovery factor	ζ_m	1.0
	Pressure recovery factor	δ_s	0.6-0.9
	Temperature recovery factor	β_s	1.0
Thermal Energy Reservoir (TES)	Pressure recovery factor (charge and discharge)	δ_r	0.95-0.98
	Temperature recovery factor	β_r	0.95
Base Plant	Cycle thermal efficiency	η_b	0.35
Air Properties	Specific heat	C_a	0.25
	Polytropic exponent	n	0.30

tight so that the mass recovery factor is 1.0. The air reservoir pressure recovery factor includes piping losses and throttling losses on each end as required to maintain the specified turbine inlet and compressor outlet pressures. The operating pressure range of the reservoir depends on its design (i.e., aquifer versus mined cavern, compensated versus uncompensated, etc.). The values employed for the performance parameters are tabulated in Table 4-1.

4.1.1 Comparison of Base System with Huntorf

The code developed to model the thermodynamic cycles may be used for all of the systems illustrated in Figures 4-1 through 4-4 by appropriate identification of control parameters. As a test case, the code was checked against published data for Huntorf⁽⁴⁾ (Table 4-3) with reasonable agreement. In addition, numerous hand computations have been done for code verification.

The primary variables in the parametric analysis of the three cycles are the inlet pressure and temperature of the high pressure turbine (HPT), and the inlet temperature of the low pressure turbine (LPT). The overall pressure ratio of the compressor train is computed for each case to give the designated reservoir delivery pressure allowing for all of the assumed pressure losses.

For the systems of Figures 4-1 and 4-2, the compressors are cooled as much as possible to minimize the work input. When thermal storage is used to recover the heat of compression, the cooling is adjusted to keep the compressor outlet temperatures at or below the maximum limits presented in Table 4-2. Discussions with the manufacturers indicate maximum allowable temperature limits of about 650°F for axial compressors⁽³⁾ and 400°F for centrifugal compressors for currently available equipment. We have optimistically assumed that these could be stretched to 700°F and 450°F, respectively, with minor modifications. However, the manufacturers indicate that further increases in operating temperature would require major changes in design, especially because of the intermittent duty required. Studies planned at ANL for FY-1978 should establish reasonable estimates of the limits of feasibility and costs for high temperature compressors.

Table 4-3 is a map of the thermodynamic states and the computed input and output works, and the heat transfer and flow rates for design conditions which approximate certain variations on the Huntorf cycle. For comparison, computations are presented for this cycle with and without exhaust heat recuperation and hydro-compensation. The published results for Huntorf⁽⁴⁾ are

TABLE 4-2. Assumed Limiting Operational Criteria for System Components

<u>System or Components</u>	<u>Design Criteria</u>	<u>Symbol</u>	<u>Assumed Limiting Values</u>
<u>Compressors</u>			
Primary (C1 and C2, combined, axial flow)	Maximum outlet pressure	P_4	~15 atm
	Maximum outlet temperature	T_4	~700°F
Booster (C3, multi- stage centrifugal)	Maximum outlet pressure	P_7	~150 atm
	Maximum outlet temperature	T_7	~450°F
<u>Turbines</u>			
High Pressure (T1, modified steam turbine)	Maximum inlet pressure	P_{12}	~100 atm
	Maximum inlet temperature	T_{12}	~1500°F
Low Pressure (T2 axial gas turbine)	Maximum inlet pressure	P_{15}	~15 atm
	Maximum inlet temperature	T_{15}	~2000°F
<u>Air Reservoir</u>			
	Maximum storage temperature		200°F
	Maximum pressure, cavern		0.7 psia/ft
	Porous reservoir		0.43 psia/ft

TABLE 4-3. Comparison of Huntorf Simulation with Base System
with Recuperation and/or Hydro-Compensation

State	Huntorf Simulation Without Recuperation or Compensation					Base System With Recuperation but Without Compensation					Base System With Recuperation and Compensation				
	P, atm	T, °R	W, Btu/lb	Q, Btu/lb	Results	P, atm	T, °R	W, Btu/lb	Q, Btu/lb	Results	P, atm	T, °R	W, Btu/lb	Q, Btu/lb	Results
1	1.0	520				1.0	520				1.0	520			
2	4.2	838	81.1			4.2	838	81.1			3.8	811	74.1		
3	4.1	560		-69.5		4.1	560		-69.5		3.7	560		-62.7	
4,5	16.8	903	87.4			16.8	903	87.4			13.8	873	79.8		
6	16.5	560		-85.6		16.5	560		-85.6		13.5	560		-78.2	
7	68.4	903	87.4			68.4	903	87.4			50.0	873	79.8		
8	67.8	610		-73.1		67.8	610		-73.1		49.5	610		-65.7	
			255.9	-228.2				255.9	-228.2				233.7	-206.6	
				M_c (lb/kWh) = 13.3					M_c (lb/kWh) = 13.3					M_c (lb/kWh) = 14.6	
11	45.0	610				45.0	610				45.0	610			
12	(No recuperator)			216.8		44.1	990		119.8		44.1	990		119.8	
13	44.1	1460	104.8			43.2	1460	102.3			43.2	1460	102.3		
14	11.2	1032		236.6		11.5	1043		234.0		11.5	1043		234.0	
15	11.0	1960	214.3			11.2	1960	214.3			11.2	1960	214.3		
16	(No recuperator)					1.02	1085				1.02	1085			
17	1.0	1085	319.1	453.4		1.0	705	316.6	353.8		1.0	705	316.6	353.8	
				M_t (lb/kWh) = 10.7					M_t (lb/kWh) = 10.8					M_t (lb/kWh) = 10.8	
				H_t (Btu/kWh) = 4851					H_t (Btu/kWh) = 3815					H_t (Btu/kWh) = 3815	

$$M_c \approx 13.7 \text{ lb/kWh}$$

$$M_t \approx 11.2 \text{ lb/kWh}$$

$$H_t \approx 5500 \text{ Btu/kWh}$$

The code predicts these data accurately with the exception of the heat rate. The choices for the turbine and combustor efficiencies used in the present computation are quite optimistic, especially for the HPT. Also, the specific heat and the polytropic exponent of the exhaust gas are assumed to be the same as for pure air. These differences could easily account for the predicted difference in H_t . However, since the same assumptions are used for all computations in the present case, the difference in heat rates should be an accurate prediction of the potential fuel savings attributable to the recuperator.

The result for the Huntorf simulation with and without recuperation indicates a potential fuel savings of better than 1000 Btu/kWh if a recuperator with 80% effectiveness is used. If the LPT in these two systems were fired to 2000°F instead of 1500°F, the fuel savings attributable to the recuperator would be about 1400 Btu/kWh. Similar results have been obtained by others^(2,3,5) who have shown that the recuperator is indeed economic for CAES systems.

Table 4-3 also contains the results for simulation of the Huntorf cycle for the case where both recuperation and hydro-compensation are employed. The same compression data apply for the two cases which do not employ compensation. The same expansion data apply whether or not compensation is employed. We note that although hydro-compensation reduces the pressure range in the reservoir, a small differential between fully charged and discharged conditions will exist. In addition, losses in the pipes, valves and manifolds must be accounted for. We have assumed a 2% pressure loss (~ 0.5 atm) for these losses in each case.

We see from Table 4-3 that hydro-compensation results in savings in compressor input. For this example, the compressor specific mass flow goes from 13.3 to 14.6 lb/kWh for a savings of about 10%. More significantly, however,

the air reservoir volume can be greatly reduced since a "cushion bubble" need not be maintained. This can significantly impact the overall system cost.

The computations presented in Table 4-3 are presented here as an example of what can be accomplished in a given system through two relatively straightforward cycle modifications. We will assume that our base system (advanced concept) incorporates recuperation since this feature is almost certain to be included in all systems which will be built in the future. However, hydro-compensation will not always be possible so it will not be included in the base system.

4.1.2 Parametric Analysis of Base System

The thermodynamic code was first used for a parametric study of the performance of the base cycle of Figure 4-2. The results of these computations are shown in Figures 4-5 through 4-8. The variable parameters in these and succeeding computations are the reservoir delivery pressure and the turbine inlet temperatures. The performance parameters investigated are the turbine fuel consumption (heat rate), the total fuel consumption, the turbine and compressor specific mass flows, and the ratio of the mass flows.

The total fuel consumption is the sum of the heat equivalent of fuel consumed in the turbines and the fuel burned in the base plant to generate the electric energy required to run the compressors. The base plant is assumed to have a net thermal efficiency of 35%. The total fuel consumption is a measure of the overall energy conversion efficiency for the process.

The compressor specific mass flow in lb/kWh is the input parameter which defines the amount of air compressed per unit of input electrical power. This figure gives us the motor and compressor ratings for a given mass flow rate of air. Similarly, the turbine specific mass flow tells us the amount of air required per unit of energy output or the flow rate required for a given output power. We refer to the ratio of specific mass flows as the cycle coefficient of performance (COP),

$$COP = \frac{M_c}{M_t} = \frac{W_t}{W_c} \left[\frac{(\text{kWh}) \text{ out}}{(\text{kWh}) \text{ in}} \right]$$

FIGURE 4-5. Turbine Fuel and Total Fuel Consumption for a 50 atm Base System Cycle

This term is sometimes called the charging energy ratio since it represents the ratio of the electrical energy output to the electrical energy input. This ratio can be less than or greater than 1.0, depending on the design of the cycle. For pumped hydro or for an adiabatic CAES system, the COP is always less than 1.0 because the pump or compressor input is the only energy consumed in the cycle.

The computed energy consumption by the system in Figure 4-2 is shown parametrically in Figures 4-5 and 4-6. The data of Figure 4-5 illustrate the effects of the HPT and LPT inlet temperatures for a system with an HPT

FIGURE 4-6. Turbine Fuel and Total Fuel Consumption for Base System with HPT Fixed at 1000°F

inlet pressure of 50 atm. The data of Figure 4.6 illustrate the effect of the HPT inlet pressure for a cycle where the HPT is fired to 1000°F. In each case, the data are plotted with the LPT firing temperature as abscissa, the turbine fuel consumption as the left ordinate, and the total fuel consumption as the right ordinate.

For the computations shown in Figures 4-5 through 4-8 and for the other advanced cycles as well, we have used the maximum operating variables defined in Table 4-2. We assumed that the HPT may be operated unfired or fired to

FIGURE 4-7. Turbine Specific Mass Flow and COP as a Function of LPT Firing Temperature for a 50 atm Base System Cycle

the assumed maximum of 1500°F . Where the HPT is unfired, its inlet temperature is controlled strictly by the LPT outlet condition and the recuperator effectiveness. The HPT inlet of 1000°F is the value used in the Huntorf design.

The LPT inlet conditions are assumed to be at 15 atm and temperatures in the range of 1000°F to 2000°F . The 15 atm level is at the limits of the current state of the art for U.S. gas turbine technology.⁽³⁾ Most gas

FIGURE 4-8. Turbine and Specific Mass Flow and COP for Base System with HPT Fixed at 1000°F

turbines of American manufacture operate at lower pressures than this. The upper temperature limit of 2000°F is representative of present equipment capabilities. Some European equipment may be available with higher pressure ratios. However, they tend to operate at more conservative temperatures (as illustrated by the 1500°F firing temperature used at Huntorf).

4.1.3 Discussion of Base System Results

We may immediately draw several conclusions from the data of Figures 4-5 and 4-6. First, we note that the turbine fuel consumption H_t Btu/kWh increases with an increase in the LPT inlet temperature. This is, of course, opposite to the behavior of a simple gas turbine cycle where the heat rate decreases with increasing turbine inlet temperature. The apparent anomaly is due to the effect of decoupling the turbine from the compressor in the CAES cycle. We note that the total fuel consumption by the system, H_o (including turbine and base plant fuel), decreases with increasing LPT firing temperature, just as in the simple gas turbine cycle.

Next, we note that the turbine fuel consumption decreases only slightly with increasing temperature at the HPT inlet because the higher temperature allows more work per pound of air in the HPT. However, the reduction in the turbine heat rate is very small because the higher HPT inlet temperature results in a higher HPT exhaust temperature and thus less heat recovery in the recuperator. The reduction in the total fuel consumption, H_o , is more dramatic because the increased temperature results in more work per pound of air, giving a significant reduction in air flow and compression energy requirements.

Figure 4-6 illustrates the effect of pressure on the energy consumption for an HPT firing temperature of 1000°F. We find that the turbine heat rate decreases with increasing pressure while the total fuel consumption increases with increasing pressure. The reduction in the turbine heat rate is due to expansion to lower temperatures (due to higher pressure ratios available), which causes more efficient use of the available energy. The higher system energy consumption is due to higher compression input, most of which is subsequently lost in the intercoolers and aftercoolers.

Figures 4-7 and 4-8 illustrate the effect of the design conditions on the specific mass flow M_t lb/kWh, the compressor specific mass flow M_c lb/kWh, and the cycle coefficient of performance (COP) which is the ratio M_c/M_t . We find that the turbine mass flow requirement is reduced by increasing the cycle

pressure or increasing the inlet temperature at either or both of the turbines. The reduction of M_t diminishes with increasing pressure. Similarly, the reduction of turbine mass flow requirements diminishes somewhat with increasing temperature. The effect of the HPT temperature T_{12} is least pronounced because the majority of the output is derived from the LPT.

The cycle COP increases essentially linearly with the turbine inlet temperature. The COP decreases with increasing pressure because of compressor cooler heat losses. For this cycle we note that the COP is greater than 1.0 for nearly all of the conditions shown on the figures.

4.1.4 Base System Conclusions

A number of conclusions may be reached by reference to the data in Figures 4-5 through 4-8. First, we note that increasing the temperature of the HPT inlet has very little effect on the turbine heat rate H_t , but dramatically reduces both the total fuel consumption H_o and the turbine mass flow requirement M_t . Therefore, increasing the HPT inlet temperature is certainly desirable for the cycle configuration.

Second, we note that increasing the LPT inlet temperature reduces M_t and H_o but increases H_t . Thus, a minimum must be sought in the sum of the costs for investment, maintenance, compression energy and turbine fuel in order to determine the optimal LPT firing temperature. This is in contrast to the conventional gas turbine cycle where it has been cost effective to increase the turbine inlet temperature as far as possible.

Finally, we note that increasing the storage pressure decreases H_t and M_t , but increases the compressor work which increases H_o . A minimum in the total costs must again be sought to identify the optimum pressure for this configuration. In this case, reservoir costs may be a dominant factor in the optimization.

The data of Figures 4-5 through 4-8 represent the primary performance characteristics of the base system which we will use for comparison. The single- and double-stage systems will be compared to the base system to quantify potential improvements that might be attained by the use of TES systems.

4.2 SINGLE-STAGE SYSTEM

The system shown in Figure 4-3 includes a single stage of heat storage to partially recover the heat of compression. This requires a few changes in the compressor train and in the expansion side of the system. The primary change in the compression system is the elimination of the aftercooler and the substitution of an interstage heat recuperator. The intercooler heat loss is also reduced to produce the maximum allowable temperature at the outlet of the last stage of the axial flow compressor.

The purpose of the interstage recuperator is to recover heat from the outlet of the axial flow compressor. An efficient recuperator allows heat storage at a temperature which approaches the maximum temperature limit of the last stage of the axial flow compressor. (This temperature is higher than that of the centrifugal compressors.)

We have assumed from the outset that the maximum pressures and temperatures allowable in each compressor stage will be limited to the capabilities of currently available equipment with only minor modification. The first stage (or stages) of compression will be accomplished with an axial flow compressor of the type employed in large turbogenerator sets used for peaking duty. Compressors of this type can currently be obtained with overall (multi-stage) compression ratios approaching 16:1 and peak outlet temperatures approaching 650°F. According to previous studies, the best available choice for the booster compressors would be large, industrial-grade centrifugal compressors. These units are available with peak outlet pressures well in excess of 100 atm and a maximum outlet temperature of about 400°F. We will, however, assume that we may extend these to 700°F and 450°F for the purpose of this analysis. Further increases would almost certainly require major equipment development.

The compressor recuperator should not be a difficult device to build or maintain, as compared to the exhaust heat recuperator. The pressure and temperature levels and differentials will be moderate and there should be no corrosion problems of the type experienced in the exhaust system. The ultimate

deciding factor on the use of this recuperator will be a trade-off between the economic advantages and disadvantages of the recuperator. The advantages are the saving of compressor energy and turbine fuel and economic savings due to increased compressor life. The disadvantages are the increased investment and maintenance costs for the recuperator. Alternatively, if it is possible to increase the temperature limit of the booster compressors to approach that of the axial flow compressor, the interstage recuperator could be eliminated.

The expansion side of the cycle in Figure 4-3 is schematically similar to that in Figure 4-2. The differences are that the air entering the first combustor is preheated in the heat reservoir and that the recuperator follows the first rather than the last stage of expansion. It is conceivable that the HPT could be operated adiabatically. This would maximize the heat recovery in the exhaust recuperator and possibly reduce fuel consumption. In general, however, we have assumed that the HPT can be fired at any temperature up to 1500°F.

4.2.1 Parametric Analysis of Single-Stage System

A parametric analysis of the cycle in Figure 4-3 closely parallels the procedure used for the base cycle. However, for the single-stage system, optimization of the design requires that we investigate additional design variables. The primary new variable is the apportioning of the available expansion ratio between the HPT and the LPT.

For the case of the base cycle (without TES), the parametric studies found that it is advantageous to keep the expansion ratio of the LPT as high as possible since this minimizes the turbine fuel and airflow requirements. For the base system, constrained by the assumptions in this analysis, the LPT expansion ratio was assigned the assumed maximum value of 15:1. For the systems with TES, the relative location of the recuperator is changed to best utilize the available waste heat in the exhaust. In this case, the turbine fuel consumption can be reduced by using an LPT expansion ratio which is less than the assumed maximum of 15:1. This effect is illustrated by the data and the discussion which follows.

The results of the general parametric analysis of the single-stage system are given in Figures 4-9 through 4-14. Again, as in the analysis of the base system, the primary performance parameters are assumed to be the turbine heat rate, the total fuel consumption, the turbine and compressor specific mass flows and the ratio of the latter, the coefficient of performance. Selected data are presented in the figures to illustrate the effects of the reservoir delivery pressure, the two turbine inlet temperatures and the relative expansion ratio split between the two turbines. We will briefly discuss the principal features of the data in the figures and will then compare specific cycles.

The data of Figures 4-5 and 4-9 are for comparable systems employing the same turbines, compressors and auxiliary equipment. For this example, the HPT inlet pressure is 50 atm and the LPT expansion ratio is fixed at 15:1. In both cases, three values of the HPT inlet temperature are considered up to 1500°F for a range of LPT inlet conditions up to 2000°F.

A comparison of the data for these conditions indicates that the potential savings of turbine fuel due to addition of the TES reservoir are not dramatic. In fact, at the higher turbine inlet temperatures, there is essentially no potential for fuel savings in this configuration. We can improve the fuel savings relative to the base system by reducing the HPT inlet temperature, thereby lowering its outlet temperature and improving the recuperator effectiveness. In addition, reducing the LPT inlet temperature reduces the heat rate. Both of these options result in an increase in the turbine airflow requirement, however, as shown by Figures 4-7 and 4-12. Since the potential for fuel savings in this configuration is minimal, the trade-offs required are probably not very practical, at least for the pressures and temperatures used in this example.

We look next at the effects of pressure on the cycle performance. The data of Figures 4-6 and 4-10 illustrate the effect of the cycle pressure. Again, the various cycle components of the base system and the single-stage system are assumed to be the same. The data shown assume $T_{12} = 1000^{\circ}\text{F}$ and $r_{t2} = 15:1$.

FIGURE 4-9. Turbine Fuel and Total Fuel Consumption for Single-Stage System with HPT Inlet Pressure at 50 atm and LPT Expansion Ratio at 15:1

We observe that the variations of H_t with P_{11} are similar at high values of T_{15} , where similar reductions in heat rate can be achieved by going to higher pressures. At lower values of T_{12} , the reduction in heat rate at higher pressures disappears in the single-stage system. In either case, the relative benefit (reduction in H_t) diminishes with increasing pressure.

The overall heat rate is not dramatically affected by the cycle pressure. In the base system, H_o increases with pressure because the increased compression energy is essentially lost. In the single-stage cycle, H_o is essentially unaffected by the cycle pressure because the compression energy is partially recovered.

FIGURE 4-10. Turbine Fuel and Total Fuel Consumption for Single-Stage System with HPT Inlet at 1000°F and LPT Expansion Ratio at 15:1

The turbine specific mass flow M_t is given by Figures 4-7 and 4-8 for the base cycle and Figures 4-12, 4-13 and 4-14 for the cycles with TES. For the same values of turbine inlet pressures and temperatures, we observe that the turbine specific mass flows are essentially the same. The compressor specific mass flow data are shown in the figures, in addition to the cycle COP. The cycles with TES suffer greater pressure losses, producing larger values of W_c and smaller M_c . Thus, the COP for systems with TES are uniformly lower than for the base system. We note that an LPT expansion ratio of 15:1 gives the lowest mass flow requirement at high values of T_{15} and the highest

FIGURE 4-11. Turbine Fuel and Total Fuel Consumption for Single-Stage System as a Function of LPT Expansion Ratio with HPT Inlet at 10 atm, 1000°F

values of M_t at low values of T_{15} . The trend in the variation of M_t is opposite to that for H_t . In fact, we find that whatever reduces H_t tends to increase M_t and vice versa.

By comparison of the data in Figures 4-5 and 4-9, we conclude that systems with TES offer little potential for improved heat rate if the design of the turbines is assumed to be the same as in the base system cycle. Therefore, we must consider cycles with different expansion ratios which allow more effective use of the recuperator. The data of Figures 4-11 and 4-14 illustrate the effects on the cycle performance by varying the expansion ratio split from that assumed in the base cycle.

FIGURE 4-12. Specific Mass Flow and COP for Single-Stage System with HPT Inlet at 50 atm and LPT Expansion Ratio at 15:1

From Figure 4-11 we see that reducing the LPT expansion ratio (and thus increasing the HPT expansion ratio accordingly) generally results in decreased turbine fuel consumption H_t but increased total energy consumption H_o . However, we note that at very low LPT inlet temperatures ($T_{15} \lesssim 1500^\circ\text{F}$), $r_{t2} = 15$ can give the lowest heat rate. The overall fuel consumption is not dramatically affected by the pressure ratio split until r_{t2} gets very low (i.e., less than 5.0).

FIGURE 4-13. Specific Mass Flow and COP for Single-Stage System with HPT Inlet at 1000°F and LPT Expansion Ratio at 15:1

From Figure 4-14 we see that varying the pressure ratio split will change the turbine airflow requirements. For example, at an LPT inlet temperature of 1800°F going from $r_{t2} = 15$ to $r_{t2} = 3$ causes an increase in M_t from 9.5 to 11.5 lb/kWh. This corresponds to an increase in the reservoir volume by 20% and larger turbines and compressors will be needed to handle the increased flow rates.

FIGURE 4-14. Turbine Specific Mass Flow for Single-Stage System as a Function of LPT Expansion Ratio r_{t2} for $P_{12} = 50$ atm $T_{12} = 1000^{\circ}\text{F}$

From the above discussion, we conclude that the single-stage CAES cycle can achieve significant fuel savings relative to the base cycle performance. However, the fuel savings is achieved at the expense of increased compression energy requirements, larger equipment, larger reservoirs, and new designs in the turbomachinery, especially in the turbines. This is, of course, where economics must affect the assessment since the relative costs of charging power, capital and turbine fuel will ultimately decide how much capital investment is feasible and how much fuel can economically be burned.

4.2.2 Cycle Performance Comparisons

The general performance data of Figures 4-5 through 4-14 illustrate parametrically the overall cycle performance characteristics for the alternative cycles. However, for a more quantitative comparison we need to study some direct comparisons of specific cycles. Therefore, we have selected several representative cycles for comparison which illustrate the effects of the primary design variables. The summary of the assumed design variables and resultant performance data for these cycles are given in Tables 4-4 and 4-5.

TABLE 4-4. Compressor Cycle Thermodynamic State and Process Map for 50 atm Systems^(a)

State	Base System				Single-Stage System			
	P, atm	T, °R	W, Btu/lb	Q, Btu/lb	P, atm	T, °R	W, Btu/lb	Q, Btu/lb
1	1.0	520			1.0	520		
			79.9				80.9	
2	4.1	833			4.1	837		
				-68.3				-29.1
3	4.0	560			4.0	721		
			86.1				112.1	
4	16.3	897			16.7	1160		
								(-48.0)
5	16.3	897			16.4	960		
				-84.4				-98.7
6	16.0	560			16.1	565		
			86.1				87.9	
7	65.6	897			65.6	910		
				-71.9				(+48.0)
8	65.0	610			65.0	1100		
TOTAL			252.1	-224.6			280.9	-127.8

(a) These data apply to both high temperature and low temperature options and for both pressure ratio combinations used in Table 4-5.

TABLE 4-5. Comparison of Performance Potential of Advanced CAES Cycles With and Without Thermal Energy Storage

System/State Parameter	Symbol	Units	Option A, Low Temperature Turbines					Option B, Mixed System					Option C, High Temperature Turbines				
			Base System	Single-Stage Systems				Base System	Single-Stage Systems				Base System	Single-Stage System			
4.5a Summary of Primary Operating Variables																	
LPC (Axial) Outlet Temperature	T ₄	°F	437	700	700	700	700	437	700	700	700	700	437	700	700	700	700
HPC (Centrifugal) Outlet Temperature	T ₇	°F	437	450	450	450	450	437	450	450	450	450	437	450	450	450	450
HPT Inlet Temperature	T ₁₂	°F	1,000	1,000	1,000	1,000	1,000	1,000	1,000	1,000	1,000	1,000	1,500	1,500	1,500	1,500	1,500
LPT Inlet Temperature	T ₁₅	°F	1,500	1,500	1,500	1,500	1,500	2,000	2,000	2,000	2,000	2,000	2,000	2,000	2,000	2,000	2,000
HPT Expansion Ratio	r _{t1}		3.1	3.1	5.8	9.2	3.0	3.1	3.1	5.8	9.2	3.0	3.1	3.1	5.8	9.2	3.0
LPT Expansion Ratio	r _{t2}		15.0	15.0	8.0	5.0	15.4	15.0	15.0	8.0	5.0	15.4	15.0	15.0	8.0	5.0	15.4
4.5b Summary of Primary Performance Variables																	
Compressor Specific Mass Flow	M _c	lb/kWh	13.5	12.2	12.2	12.2	12.2	13.5	12.2	12.2	12.2	12.2	13.5	12.2	12.2	12.2	12.2
Turbine Specific Mass Flow	M _t	lb/kWh	10.6	10.5	10.6	11.0	11.7	8.97	8.97	9.20	9.87	10.6	8.27	8.23	8.24	8.45	8.96
Coefficient of Performance	COP	$\frac{(kWh)_{out}}{(kWh)_{in}}$	1.27	1.16	1.15	1.11	1.04	1.51	1.36	1.33	1.24	1.15	1.63	1.48	1.48	1.44	1.36
Turbine Heat Rate	H _t	Btu/kWh	3,771	3,368	3,279	3,129	2,986	3,855	3,650	3,456	3,343	3,243	3,830	3,951	4,003	3,863	3,757
System Heat Rate	H _o	Btu/kWh	11,420	11,820	11,810	11,920	12,380	10,310	10,840	10,850	11,120	11,770	9,790	10,550	10,620	10,640	10,950

Table 4-4 illustrates compressor cycles and performance data for the base and single-stage systems. In the base system, cooling is provided to minimize the compression work and to cool the air before it enters the air reservoir. As a consequence, 90% of the compression energy is dissipated as heat.

In the single-stage system, the compressors are cooled just enough to keep the compressor outlet temperature from exceeding the limits defined in Table 4-2. The compressor heat recuperator recovers heat from the exhaust of the axial flow compressor and allows air injection into the TES reservoir at very nearly the maximum compression temperature. As a consequence, 55% of the compression energy is retained and transmitted to storage. If the recuperator is eliminated only 87.9 Btu/lb, or 35%, of the compression heat could be recovered because of the lower temperature limit for the booster compressor.

The data of Table 4-5 deal primarily with variations on the expansion side of the single-stage cycle with comparative data for the base system. We have grouped the designs by options. Option A is for systems in which both turbines are operated at relatively conservative temperatures, Option C is for systems in which both turbines run at high temperatures, and Option B is a mixed option which uses high LPT temperature and low HPT temperature. We note that Option A closely parallels the Huntorf design and is thus fairly conservative. Option B is the same as A but the LPT inlet is raised to 2000°F, which more nearly matches the GE, Westinghouse and UTC systems. Option C requires that the HPT be fired to 1500°F, a level which will require turbine development since nothing is currently available for the assumed operating conditions. These options are grouped to fall in the order of increasing complexity which also happens to be in order of decreasing airflow requirement and decreasing overall energy consumption. Within each option we give the primary performance criteria for a base system cycle and for single-stage systems where four different pressure ratio splits are used for the latter. We note that all systems are based on 50 atm HPT inlet and that the LPT in the base system is assumed to have a fixed expansion ratio of 15:1.

We have already noted the general reduction in M_t and H_o in going from Options A to C. The variation in the heat rates is not so simple. For Option A (the low temperature/Huntorf style design), the best fuel economy is achieved by the single-stage system which has an expansion ratio split of 15.4/3.0 for the HPT and LPT, respectively. In this case the potential savings in fuel is approximately 785 Btu/kWh or a 21% reduction compared to the base system. The overall heat rate of this cycle would be increased by about 960 Btu/kWh or 10%, and the mass flow requirement is increased by about 1.1 lb/kWh or 11%, compared to the base cycle. The COP is reduced by about 20% compared to the base system.

At the high turbine inlet temperatures employed in Option C, TES offers no improvement in any of the performance parameters, as compared to the base system. Thus, there is no reason to consider this type of system any further since the addition of TES accomplishes nothing that cannot be done easier with the recuperator alone.

The higher LPT inlet temperature used in Option B results in increased turbine fuel consumption but reduced airflow and compression energy requirements when compared to the equivalent lower temperature cycles in Option A. As in Option A, the 15.4/3.0 HPT/LPT expansion ratio split gives the lowest heat rate of the five alternatives considered. At the higher inlet temperatures, the fuel savings attributable to addition of TES is about 512 Btu/kWh or 13% compared to the base cycle. However, the penalty paid is a 10% increase in the airflow requirement and a 25% increase in the compression energy requirement relative to the base cycle for the same inlet conditions.

4.2.3 Single-Stage System Performance

We summarize the above discussion by stating the general design alternatives which tend to minimize turbine fuel consumption in a system with TES.

- The HPT inlet temperature should be low and its expansion ratio should be relatively high. This allows maximum utilization of the heat recovered from storage, maximum heat recovery in the recuperator, and minimum fuel consumption in the HPT.

- The LPT inlet temperature and expansion ratio should be relatively low. The heat rates for all of the cycles decrease with decreasing LPT inlet temperature. In addition, the reduction in heat rate due to TES is greatest at lower temperatures.

Unfortunately, the above conditions that tend to reduce the heat rate result in increased airflow and compression energy requirements. The increased airflow requires larger air reservoirs, larger turbines, compressors, heat exchangers and auxiliary equipment. Furthermore, the systems with TES require that most of the expansion occur in the HPT, the opposite of the base system design. Since there are currently no available gas turbines suitable for high pressure, high expansion ratio duty, steam turbines would probably have to be employed. Due to aerodynamic mismatch for air versus steam operation, the HPT isentropic efficiency will probably be lower than current state of the art for gas and steam turbines. This implies even greater performance penalties than estimated here.

4.2.4 Economic Incentives for Addition of TES

To make an assessment of the feasibility of CAES systems with TES, more than just fuel savings must be considered, although that is the prime focus of this study. The ultimate measure of merit will be decided by an economic trade-off in which additional costs for the plant and additional compression energy, are balanced against the economic value of the turbine fuel saved.

A first order estimate of the economic incentive to employ (or not to employ) TES in advanced CAES systems is currently being addressed at PNL but is not sufficiently complete to report. However, we will make some order of magnitude comparisons based on the performance data presented here which are instructive if not quantitatively precise. A more detailed cost analysis will be completed in FY-1978.

For the order of magnitude economic assessment, we use the data of Table 4-5 for Options A and B which we feel represent state-of-the-art technology with modifications which can reasonably be expected considering the ultimate market. For cost data we use results reported in previous assessments of conventional CAES systems.

First, we point out that previous studies of CAES systems without TES (but with recuperators) indicate the following approximate breakdown in the sent-out cost of power,⁽¹⁻⁵⁾

capital fixed charges	50%
turbine fuel	25%
compression energy	20%
operation and maintenance	5%

The results of the preceding analysis illustrate that a modestly conservative CAES system with TES (Option A) can achieve a fuel savings of about 21% as compared to the same cycle with only a recuperator. However, because of increased airflow requirements and pressure losses, the compression energy requirement is increased by about 20%. Operation and maintenance (O&M) costs will be assumed to be the same in both cases. The sum of turbine fuel, compression energy and O&M costs is then $(0.79)(25) + (1.2)(20) + 5 = 49\%$ of total costs, a decrease of only 1%. Therefore, the capital fixed costs may increase by only 1% of the sent out power costs before the economic incentive to add TES is lost. In other words, addition of TES becomes uneconomic when it increases total capital expenditures by more than 2%, based on the cost of an equivalent CAES system without TES.

For Option B, the fuel savings is only 13% and the compression energy increase is 25%. Again assuming fixed O&M costs and the same sent-out cost, the total for fuel, compression energy and O&M costs is $(0.87)(25) + (1.25)(20) + 5 = 52\%$, or 2% higher than the base plant. Therefore, there is no potential for cost savings in Option B since a reduction in capital investment would be required to achieve a net savings in power costs. A reduction is unlikely because of the added costs of the TES system and larger reservoirs and equipment.

The cost of a conventional CAES system will depend on many factors, including reservoir type and design, location, utility needs, etc. Previous studies⁽¹⁻⁵⁾ have shown that total costs for a facility resembling the base cycle will probably be approximately \$200 to \$250/kW of installed capacity

for 8 hr/day duty. Using this estimate as a base, the above analysis indicates that the total costs attributable to the addition of TES may account for no more than \$4 to \$5/kW of installed generation capacity. The system of Figure 4-3 conserves about 50% of the available heat of compression with a cycle COP of about 1.15. Thus, the thermal storage capacity required per unit of output is $0.5/1.15 \approx 0.44 \text{ kWh(t)/kWh(e)}$. The maximum allowable incremental cost per unit of thermal capacity is then about \$1.14 to \$1.42/kWh(t) assuming 8 hr/day of generation. From Section 5 we observe that this is less than half of the estimated cost for the cheapest TES system considered, not including additional costs for turbomachinery and the air reservoir. Therefore, there appears to be no economic incentive to attempt to recover the heat of compression by use of a single-stage TES reservoir system.

4.3 DOUBLE-STAGE SYSTEM

The analysis in Section 4.2 has shown that the single-stage system (with a single TES reservoir) is capable of storing only about half of the available compression energy with the balance being lost in the intercoolers. Furthermore, the analysis shows that use of a single TES, even with the compression heat recuperator, results in only slight to moderate savings in turbine fuel (compared to the base system) with no real potential for adiabatic (no-fuel) operation. To increase the recovery of the compression energy and, hopefully, to further reduce the fuel requirement, we now consider the possibility of including a second TES reservoir in the system. The result is the double-stage concept illustrated schematically by Figure 4-4. Variations of the basic configuration in Figure 4-4 represent the highest degree of complexity that will be considered in this analysis.

The cycle of Figure 4-4 is similar to the cycles in Figures 4-2 and 4-3 in that it uses many of the same components in approximately the same order. However, on the compression side, the recuperator between the compressors is replaced by a second TES reservoir and the compressor cooling is reduced

accordingly. On the expansion side, the primary change is the elimination of the combustor at the HPT inlet. As a result, the design and operating constraints of many of the system components are interrelated and inflexible compared to the much simpler design of the single-stage system.

The cycle configuration illustrated in Figure 4-4 was selected after considering several alternatives. Each alternative was studied to determine the potential performance improvement and feasibility. For example, the optimal airflow path through the TES reservoirs during discharge was found to be the reverse of the path followed during charge-up. This path was chosen to insure that both TES reservoirs would always be subjected to approximately constant pressure. Presumably, inverting the order of the reservoirs from that shown in Figure 4-4 would subject the TES containment vessels to excessive stress due to the large daily pressure cycles which would occur.

The elimination of the HPT combustor is due primarily to constraints imposed by TES #1. First, we note that the exhaust from the HPT passes through TES #1 where unburned hydrocarbons and other contaminants from the combustor could be deposited. During compression, these contaminants could be picked up and transported through the booster compressors and into the air reservoirs with potential damage to both systems.

Second, we note that the temperature at the cold end of TES #1 is determined by the HPT outlet temperature T_{13} . Therefore, anything that causes an increase in T_{13} reduces the heat storage capacity of TES #1 and increases the heat loss in the booster compressor precoolers. If the overall pressure ratio and the HPT outlet temperature T_{13} are both fixed, an increase in the HPT inlet temperature T_{12} requires an increase in the HPT expansion ratio and a compensating decrease in the LPT expansion ratio. This will result in greater power output from the HPT and reduced output from the LPT. However, if T_{15} is greater than T_{13} (as will almost always be the case), the loss in the LPT power output will be greater than the gain in the HPT output for the same fuel input and airflow rate. Therefore, it is not advantageous to attempt to increase the HPT inlet temperature T_{12} beyond that which is available from the

outlet of TES #2. These considerations contributed both to the decision to eliminate the HPT and to the placement of the exhaust heat recuperator as shown in Figure 4-4.

The analysis established that alternative configurations to the cycle of Figure 4-4 which tend to increase the HPT inlet temperature above the outlet temperature from TES #2, do so at the expense of the cycle performance. The optimal configuration is found to be that in Figure 4-4 where the HPT is assumed adiabatic, the exhaust heat recuperator is at the LPT inlet and the only fuel burned is in the LPT inlet combustor. These conclusions follow from the assumed capabilities of the compressors, turbines and TES reservoirs. Although it is possible to propose other cycles which theoretically offer improved cycle performance, the potential improvement is slight and the design changes required in many of the system components appear to be impractical.

4.3.1 Parametric Analysis of Double-Stage System

The basic configuration shown in Figure 4-4 is used to illustrate a "family" of possible cycles that are derived from the same configuration. On the compressor side, limited cooling will normally be required to stay within specified design limits. However, in the limiting cases, all compressor cooling might be eliminated causing the compressor outlet and TES inlet temperatures to correspond to adiabatic compression processes. Similarly, on the expansion side, the turbine fuel consumed in the LPT may theoretically be adjusted to achieve any desired turbine inlet temperature. In the limiting case where no fuel is burned, the LPT inlet temperature T_{15} would then be determined by the temperature of TES #1 and the recuperator would be inoperative. This complete range of potential turbine and compressor operating conditions will be treated parametrically to assess the potential fuel savings of this cycle as compared to the base and single-stage technologies.

The primary independent variables are assumed to be the reservoir delivery (HPT inlet) pressure and the LPT inlet temperature. The HPT inlet temperature is assumed to be governed by the outlet temperature from TES #2 (i.e., the booster compressor temperature limit). The HPT expansion ratio is set by

the required low end temperature of TES #1. In addition, we consider two compressor design scenarios, Case I and Case II, corresponding to state of the art and advanced adiabatic technologies, respectively. The primary performance parameters are the turbine fuel and total fuel consumption, the compressor and turbine specific mass flows and the ratio of the latter which we call the COP.

Most of the performance criteria and limitations used for the analysis in this section are the same as were used for the base and single-stage systems as given in Tables 4-1 and 4-2. The state-of-the-art compressor system, identified in Case I, is the same as in Section 4.3. In Case II, the compressors are assumed to be totally adiabatic so that the full potential of the adiabatic cycle may be examined. The compressors and TES reservoirs are assumed to be capable of sustaining the temperatures thus attained. Since these assumptions exceed current capabilities in most cases, the costs involved will be difficult to predict. Nevertheless, the predicted performance of this option will allow us to ascertain whether the potential for improvement is sufficient to warrant further consideration.

The data given in Figures 4-15 through 4-21 are representative of the results of the parametric performance analysis for the "family" of cycles represented by Figure 4-4. Figures 4-15 through 4-18 are for Case II where totally adiabatic compression is assumed. These data illustrate the best fuel economy attainable with this configuration. Figures 4-19 through 4-21 illustrate the performance trade-offs between Case I, in which the compressors are partially cooled, and Case II, in which they are not. These comparisons indicate the approximate level of improvement in heat rate and other parameters that might be achieved through development of high temperature adiabatic compressors.

The data of Figure 4-15 illustrate the turbine fuel and total energy requirements of the double-stage CAES cycle (with adiabatic compression and two stages of TES). The data illustrate the effects of cycle pressure and LPT inlet temperature. As noted above, the HPT is unheated and its expansion ratio is fixed by its assumed inlet and outlet conditions.

FIGURE 4-15. Fuel Consumption for System with Double-Stage TES (Case II)

The parametric analysis indicates that the total energy requirements are relatively unaffected by the cycle pressure. The turbine fuel consumption decreases with increasing pressure for a given LPT inlet temperature. These characteristics are quite similar to the behavior of the base and single-stage systems, but the heat rates are generally lower. The data highlighted by circles illustrate the adiabatic cycles for which the only source of energy to the cycles is the power input to the compressors. For the range of 25 to 100 atm, the overall heat rate H_o is about 14,500 to 13,800 Btu/kWh, corresponding to net conversion efficiencies of 24 to 25%, respectively.

FIGURE 4-16. Turbine and Compressor Specific Mass Flows for System with Double-Stage TES (Case II)

The total energy requirement decreases and the turbine fuel requirement increases with increasing LPT inlet temperature. The dependence of H_t on T_{15} is very pronounced at lower temperatures. For high LPT inlet temperatures H_t is lower and H_o is higher than for either the base or single-stage systems for comparable conditions.

The data of Figure 4-16 illustrate the turbine and compressor mass flow requirements. The behavior is essentially similar to the data for the base and single-stage cycles. Again, the circles highlight the mass flow requirements of the adiabatic cycles. This figure illustrates the principal

FIGURE 4-17. COP for System with Double-Stage TES (Case II)

disadvantage of the adiabatic cycles: very high mass flow requirements and thus large reservoirs, compressors, etc. However, the mass flow requirement of the adiabatic cycle decreases significantly with increasing pressure. The compressor specific mass flow in Figure 4-16 is somewhat lower than for the single-stage system (i.e., more work is required per lb) since complete adiabatic compression is assumed in this case.

The data of Figure 4-17 illustrate the cycle COP for Case II, double-stage systems. The COP is uniformly lower than for the other cycles considered, because the adiabatic compression requires more work input and the HPT is unfired, giving lower output. The COP for the adiabatic cycles is the

FIGURE 4-18. Illustration of Fuel Consumption for 50 atm Cycle, Case II. Example for $T_{15} = 1500^{\circ}\text{F}$.

ratio of total energy out to total energy in (which, in this case, is simply the cycle energy storage efficiency). We note that the storage efficiencies predicted are in the range of 0.66 to 0.71, which compares very favorably with pumped hydro efficiencies. We note that the adiabatic cycle energy storage efficiency increases only slightly with increasing pressure.

The data of Figure 4-18 illustrate how the addition of turbine fuel reduces compression energy requirements. The top line is total fuel, the bottom line is base fuel and the difference between them is turbine fuel. At an LPT inlet of 1500°F we see that the compression fuel equivalent input is

FIGURE 4-19. Comparison of Fuel Requirements for Case I and Case II

reduced by 3193 Btu/kWh or 23%, compared to the adiabatic cycle for a turbine fuel input of 2033 Btu/kWh. From Figure 4-16, the turbine mass flow for the same comparison would go from $M_t = 15.97$ to 12.35 lb/kWh for a reduction of 23%. Thus, the fuel input sufficient to fire the turbine inlet to a very moderate temperature allows reductions of 23% in compression energy and reservoir volume, plus similar reductions in the size and capacity of many other elements in the system.

The data of Figures 4-19 through 4-21 illustrate the potential advantages and disadvantages of employing totally adiabatic compression (Case II) as compared to systems which utilize essentially state-of-the-art compressors

FIGURE 4-20. Comparison of Mass Flow Requirements for Nonadiabatic (Case I) Versus Adiabatic (Case II) Compression

(Case I). We observe in Figure 4-19 that Case II (totally adiabatic compression) provides the potential for turbine fuel savings but at the expense of increasing the total energy requirements. At high values of the LPT inlet T_{15} , the turbine fuel requirement of Case II approaches that of Case I, and the total energy requirement of Case II exceeds that of Case I by an increasing amount. Adiabatic compression appears potentially advantageous at low turbine inlet temperatures, but loses its appeal as the LPT inlet temperature increases.

FIGURE 4-21. Comparison of COP for Nonadiabatic (Case I) Versus Adiabatic (Case II) Compression

In Figure 4-21 we observe that the cycle COP of Case II is substantially lower than that of Case I. For the 50 atm adiabatic cycles (where no fuel is burned), Case I gives a COP of about 65% and Case II has a COP of about 67%. Thus, the use of totally adiabatic compressors (and the required high temperatures in the TES reservoirs) results in only a 2% gain in energy storage efficiency, compared to Case I based on state-of-the-art compressors. Even at 100 atm, the COP is barely over 70% for Case II versus 66% for Case I (see Figure 4-20), only a 4% improvement. For the adiabatic cycle, it is questionable whether the development of adiabatic compressors could be justified.

4.3.2 Cycle Performance Comparisons

In order to assess the potential merit of the double-stage CAES system, it is necessary to compare its performance with the base and single-stage systems. To do so we have tabulated cycle thermodynamic process data for representative cycles in Tables 4-6 and 4-7. In Table 4-6 we show the process/state data for five cycles based on 50 atm HPT delivery pressure and 1500°F at the LPT inlet. However, other aspects of the cycles vary considerably in order to realize the best fuel economy in each cycle configuration. For example, the base cycle assumes the HPT/LPT expansion ratio is 15.0/3.1, respectively, whereas the single-stage systems use 3.0/15.4 and the double-stage systems use 6.0/7.6. The base cycle, single-stage cycle and double-stage cycle for Case I use the same compressor specifications. However, the double-stage, Case II cycles assume adiabatic compression. For comparison, the last column in Tables 4-6 and 4-7 also give the equivalent data for the adiabatic, double-stage cycle at 50 atm. The cycle specifications and performance data are summarized in Table 4-7.

From Table 4-7, we note that the double-stage systems result in turbine fuel savings, relative to the base cycle, of about 1180 Btu/kWh for Case I and 1740 Btu/kWh for Case II or savings of about 31% and 46%, respectively. The adiabatic, double-stage cycle uses no turbine fuel and thus provides 100% fuel reduction. However, the system overall energy consumption increases by about 9% and 13%, respectively, for Cases I and II and by 23% for the adiabatic cycle. Therefore, in terms of overall energy efficiency, retention of the compression energy by TES is self-defeating.

From Table 4-7 we see that the turbine mass flow requirement is increased, relative to the base cycle, by 15% for Case I, 17% for Case II, and 51% for the adiabatic cycle. The size of the air reservoir and associated equipment must be increased proportionately to maintain the same output. In addition, we observe that the compressor specific mass flow is decreased by 11% for Case I and by 23% for both Case II and the adiabatic cycle. The combined effects of increased turbine specific mass flow and reduced compressor specific mass flow result in significant reductions in the cycle COP. For example, the COP of the adiabatic cycle is reduced by 84% compared to the base cycles.

TABLE 4-6. Comparison of Thermodynamic Cycle Maps, Advanced CAES Concepts
Using Thermal Storage for 50 atm, 1500°F Systems

State	Base System				Single-Stage System, $r_{t2} = 3.0$				Double-Stage System, Case I				Double-Stage System, Case II			
	P, atm	T, °R	W, Btu/lb	Q, Btu/lb	P, atm	T, °R	W, Btu/lb	Q, Btu/lb	P, atm	T, °R	W, Btu/lb	Q, Btu/lb	P, atm	T, °R	W, Btu/lb	Q, Btu/lb
1	1.0	520			1.0	520			1.0	520			1.0	520		
			79.9								81.6				81.6	
2	4.1	833			4.1	837			4.2	840			4.2	840		
				-68.3				-29.1				-30.4				
3	4.0	560			4.0	721			4.1	718			4.1	840		
			86.1				112.1				112.7				131.8	
4	16.3	897			16.7	1160			17.1	1160			17.1	1356		
				-84.4												
5	16.0	560			16.4	960			15.9	610			16.2	610		
								-98.7				-11.6				
6	16.0	560			16.1	565			15.9	563			16.2	610		
			86.1				87.9				88.4				95.7	
7	65.6	897			66.2	910			66.3	910			66.3	985		
				-71.9												
8	65.0	610			65.6	1110			65.0	660			65	610		
			252.1	-224.6			280.9	-127.8			282.7	-42.0			309.1	0
11	50	610			50	1081			50	891			50	962		
								96.8								
12	49.5	1138			49.5	1460			50	891			50	962		
				135.9			174.1				68.5				86.0	
13	49.0	1460			3.3	749			11.2	611			8.2	611		
			89.0													
14	15.7	1097			3.2	1340			10.7	1128			7.8	1315		
				220.2				158.1				212.2				203.3
15	15.6	1960			3.1	1960			10.5	1960			7.6	1960		
			232.4				117.3				211.2				190.4	
16	1.04	1012			1.04	1481			1.0	1098			1.0	1183		
17	1.0	690			1.0	890			1.0	1098			1.0	1183		
			321.4	356.1			291.4	256.9			279.7	212.2			276.4	203.3

TABLE 4-7. Comparison of Cycle Performance for Advanced CAES Cycles
Using Thermal Storage for 50 atm, 1500°F System

Parameter	Symbol	Units	Base System $r_{t2}=15$	Single-Stage System $r_{t2}=3$	Double-Stage System Case I	Double-Stage System Case II	Double-Stage System Adiabatic
4-7a Summary of Primary Operational Parameters							
LPC (Axial) Outlet Temperature	T_4	°F	437	700	700	896	896
HPC (Centrifugal) Outlet Temperature	T_7	°F	437	450	450	525	525
HPC Inlet Temperature	T_{11}	°F	1,000	1,000	431	502	502
LPT Inlet Temperature	T_{15}	°F	1,500	1,500	1,500	1,500	855
HPT Expansion Ratio	r_{t1}		3.1	3	4.5	6.1	6.1
LPT Expansion Ratio	r_{t2}		15	15.4	10.5	7.6	7.6
4-7b Summary of Primary Performance Parameters							
Compressor Specific Mass Flow	M_c	lb/kWh	13.5	12.2	12.1	11.0	11.0
Turbine Specific Mass Flow	M_t	lb/kWh	10.6	11.7	12.2	12.4	16.0
Coefficient of Performance	COP	$\frac{(\text{kWh})_{\text{out}}}{(\text{kWh})_{\text{in}}}$	1.27	1.04	0.99	0.89	0.69
Turbine Heat Rate	H_t	Btu/kWh	3,771	2,986	2,590	2,033	0
System Heat Rate	H_o	Btu/kWh	11,420	12,380	12,440	12,940	14,100

4.3.3 Performance Analysis Conclusions

The preceding results and discussion illustrate the performance benefits and penalties which would be experienced in a CAES system with two TES reservoirs. This system is more complex and operationally more constrained than the systems with one TES reservoir. The performance analysis shows that addition of a second TES allows storage and recovery of essentially all of the compression energy which results in approximately double the fuel savings achieved with a single TES. However, the performance penalties, including increases in compression energy and turbine mass flow, also increase roughly in the same proportion. Thus, the trade-offs in performance are similar to those described for a system with a single TES, but greater in magnitude. If the incentive for conservation of turbine fuel is very strong, the reduced turbine fuel consumption may justify the other losses in performance. However, the bottom line is usually the cost of power produced compared to competing systems.

4.3.4 Preliminary Economic Assessment

A detailed economic assessment of the double-stage CAES concept is beyond the scope of the present work. However, a gross comparative evaluation of the relative economic incentives can be given, similar to the discussion in Section 4.2. For our example, we will use the double-stage CAES cycle, Case I, which uses the state-of-the-art compressor technology and most of the design specifications that were used for the base and single-stage cycles (see Tables 4-1 and 4-2). The heat rate of the Case II cycle is lower, but not substantially, and the trade-offs in performance do not, in our judgment, justify development of very high temperature compressors and TES systems.

Since most of the components of the base, single-stage and double-stage systems are essentially the same, it is probably legitimate to use as a guide the relative cost breakdowns for conventional CAES systems, as given in Section 4.2. For convenience we repeat the data here. The sent-out cost of power may be broken down approximately as follows:

capital fixed charges	50%
turbine fuel	25%
compression energy	20%
operation and maintenance	5%

From Table 4-7, comparing the base cycle and the double-stage, Case I cycle we find that the potential saving in turbine fuel is about 31% but that the compression energy requirement increases by 22%. Assuming that the resultant cost of power in the two systems is the same (i.e., the breakeven point) and that O&M costs are proportionally the same in both systems, the operating costs, less fixed charges, account for 47% of sent-out costs. Thus, for breakeven, the proportional costs due to fixed charges may rise from 50% to 53% in the double-stage system. In other words, total capital investment in the double-stage system may not exceed that for the base system by more than 6%.

To estimate the allowable increase in capital costs we will use the same data employed in Section 4.2. Previous estimates of capital costs for conventional CAES systems (our base system) are approximately \$200 to \$250/kW for 8 hr/day operation. The allowable 6% increase in cost thus translates to about \$12 to \$15/kW of installed generation. This figure must allow for additional costs due to addition of two stages of TES, a larger air reservoir, larger compressors, modification of compressors to allow maximum outlet temperatures and modification of the turbine to allow a different expansion ratio split. If we assume that 85% of the heat of compression is stored and reclaimed and that the cycle COP is 0.99 (see Tables 4-6 and 4-7), the required TES capacity per unit of electrical output is $0.85/0.99 = 0.86 \text{ kWh(t)/kWh(e)}$. The breakeven costs for addition of TES in the double-stage system for 8 hr/day operation are thus about \$1.75 to \$2.05/kWh(t). However, from the data in Section 5, we see that these maximum allowable costs are significantly lower than the estimated costs of just one TES reservoir. We thus conclude that there is no economic incentive for recovery of the heat of compression in CAES systems with two stages of TES.

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5.0 ASSESSMENT OF TES OPTIONS FOR CAES APPLICATIONS

The preceding analysis has shown that turbine fuel can be saved if all or part of the compression energy in a CAES system can be reclaimed by use of regenerative TES reservoirs. The analysis also shows that trade-offs exist that ultimately produce an economic balance of the costs for fuel, compression energy and plant investment. Thus, to assess the potential economic feasibility of advanced CAES systems incorporating TES, we need to assess the technical and economic feasibility of TES systems that might be suitable for this application.

TES devices are used to store a quantity of thermal energy at high availability until it can be efficiently used. The energy may be stored as sensible heat, latent heat, heat of fusion, or as a chemical potential. TES devices do not produce energy nor are they conservative. Energy is lost due to the inefficiencies associated with transferring the energy to and from the TES medium. However, TES devices can increase the utilization of the available energy for processes that are intermittent or operate in a batch mode.

Most of the literature currently available on TES technology is oriented toward high temperature technologies (such as iron, steel, cement, aluminum, glass, and ceramics), or low temperature technology (such as solar residential space and hot water heating).⁽¹⁷⁾ Some effort has also been expended to investigate ways to store the energy associated with boiler feedwater or boiler outlet steam. The stored energy would be used to swing the power output of a baseload plant.⁽¹⁸⁻²³⁾ An advanced concept of storing energy in chemical reactions is being studied by Martin Marietta Aerospace and Boeing Engineering and Construction.

The TES concepts for storing the CAES heat of compression considered in this preliminary assessment are pebble beds, high temperature water, high temperature oil, fusible salts, and aquifers. Containment costs, material costs, and heat exchanger costs associated with the various concepts are estimated. The concepts are separated into those storing sensible heat and those storing latent heat.

5.1 SENSIBLE HEAT STORAGE

Sensible heat is generally stored in a liquid or a solid. Gases or vapors could be used but their volumetric energy storage capacity is too low to be of practical value. A major advantage of sensible heat storage is that the materials used experience small volume changes associated with their heating or cooling. A disadvantage is that they require large temperature differences in order to store substantial quantities of energy.

One of the major design considerations for TES devices applied to CAES systems is the pressure of the compressed air. The exit conditions of the compressor will be around 600°F and between 400 and 1500 psia. These pressures require either pressure vessel containment of the TES materials or a high pressure heat exchanger to transfer heat from the air to the storage medium. Whether a heat exchanger and/or a pressure vessel is needed depends on the TES material selected.

The three methods of sensible heat storage considered are pebble beds, high temperature water, and high temperature oil. The CAES system used in the analysis has 2000 MW-hr of thermal storage which is charged at a rate of 200 MW for 10 hours and discharged at the same rate. A 400°F air temperature change is assumed with an air pressure of 1000 psia at the outlet of the compressor.

5.1.1 Pebble Beds

Glendenning⁽²⁴⁾ indicates that a suitable regenerative heat store should ideally meet the following requirements:

1. It should be capable of storing all the compression energy.
2. The temperature of the air entering the air reservoir should be close to the geothermal temperature.
3. The temperature of the air leaving the store during regeneration should be close to the compressor delivery temperature.
4. The pressure drop through the store should be minimal.
5. It should be simple to minimize cost and operational complexity.

Glendenning then states that a stationary packed column or pebble bed heat store would seem to be ideal.

General Electric Company, in a preliminary report at the ERDA Thermal Energy Storage Program Information Exchange Meeting,⁽²⁵⁾ indicated that rock pebble beds looked more attractive than brick checkerworks or iron pebble beds. The costs of the three systems as reported by General Electric Company are shown in Table 5-1. All three systems were considered safe and require little maintenance. As a result of General Electric's results, rock pebble beds were selected for further study in this report.

A schematic of a possible pebble bed (heat reservoir) incorporated in a CAES system is shown in Figure 4-2. The cost of the pebble bed system can be estimated from the cost of the fill material and the method of containment. The total cost must consider the extra volume of material required to insure that the moving thermal gradient remains in the pebble bed. Another practical consideration which should be addressed is the inclusion of a small heat exchanger to remove the heat retained by the bed due to the inevitable irreversibilities of the regenerative process.

Based on $V\rho C_p \Delta T = 2000 \text{ MW-hr}$, the volume of rock required would be about one million cubic feet. Glendenning⁽²⁶⁾ included an additional 20% of volume allowance on top of the working volume in order to minimize the possibility of hot air breaking through to the storage caverns at the end of the storage period. Stephens⁽⁸⁾ indicates that this allowance or margin factor is typically 20 to 50%. Using these allowances, the total TES volume required would be between 1.2M and 1.5M ft^3 . The current local cost of 1-1/2 in. round rock is \$2.50/yd which would give a total rock cost of \$110K to \$140K.

TABLE 5-1. Screening Comparison of Alternative TES Concepts

Concept	Operating Temperature Range, °F	Million Btu's Recovered	Device Volume, Ft^3	Device Cost, \$	Unit Cost, \$/MM Btu
Brick Checkerwork	400-900	171	27,000	81,500	477
Iron Pebble Bed	400-900	171	15,000	630,000	3684
Rock Pebble Bed	400-900	171	26,000	68,000	397

The containment cost depends on whether the pebble bed is to be located above or below ground. A pressure vessel is required for above-ground storage whereas excavation costs are associated with underground storage. A cost comparison of the two methods of storage will be made.

If a safety factor of four is used to determine the thickness of steel required for the pressure vessel used to contain the pebble bed, approximately 60M lb of steel would be required to provide one million cubic feet of storage. A conservatively low price for carbon steel is 20¢/lb. Fabrication costs can be approximated by assuming they will equal the cost of material. Based on the above, the fabrication and material costs for 1.2M to 1.5M ft³ of storage would be \$30M to \$36M.

Based on a conversation with Bill Massica of NASA Lewis Laboratory, the cost of the pressure vessel could be reduced by the use of prestressed cast iron vessels in place of steel. However, no codes exist for construction of containment vessels using this technology. Even if prestressed cast iron could be used, the saving in pressure vessel costs would be no more than 50%. This would give a minimum cost between \$15M and \$18M for above ground storage vessels.

An alternate method of containment is the use of an underground cavern. The costs can range from those associated with modifying and lining an existing cavern or abandoned mine to those associated with excavating and lining a new cavern in virgin rock. The method of excavation and the type of rock excavated further impacts the potential cost of an underground cavern. Giramonti⁽³⁾ reports cost of excavation of underground caverns from \$30 to \$50/yd. Unsworth and others in a review of excavation costs for storage systems found them to be between \$1.08/yd and \$54.00/yd depending on the contractor, location and type of rock assumed.^(19,22,27) The above excavation costs depend on blast and haul mining procedures. Excavation by boring is more expensive than the blasting method. However, boring allegedly introduces fewer rock fractures than blasting. The estimated cost of using a boring machine in excavating a cavern of the configuration shown in Figure 5-1 are given in Table 5-2.

FIGURE 5-1. Pebble Bed Mined Containment Configuration for Use with CAES

TABLE 5-2. Excavation Costs for 1M ft^3 of Thermal Storage at Depth of 1000 ft

Item	Cost/ft, \$	Length, ft	Total Cost, \$M
15- to 20-ft lined access shaft	2500	1000	2.5
4-ft vent shaft, raised bored	100	4000	0.4
1M ft^3 of storage, 15-ft diameter drifts	350	6000	2.1
Drift lining with 10-gauge carbon steel	100	6000	0.6
		Subtotal	5.6
		Additional Costs ^(a)	0.07
		Total	5.67

(a) Rental of hoist equipment to hoist muck costs \$1500/month for 3 months and muck handling at the surface costs \$65,000.

For our particular case, the storage volume required will be from 1.2M to 1.5M ft³. The additional cost can be scaled from vent shaft requirements, storage requirements, and storage lining requirements. The estimated cost for 1.2M to 1.5M ft³ will be from \$6.8M to \$8.5M. This cost represents an expected maximum cost for an underground excavated cavern suitable for thermal storage since it is based on the more costly boring procedure. Based on this analysis, above-ground storage of pebble beds pressurized to 1000 psi will be at least five times as costly as underground storage.

The total cost of the pebble bed, material plus containment, is \$6.9M to \$8.6M. Pebble beds do not need a heat exchanger to transfer heat from the air to the rocks. However, precautions will need to be taken to insure that particles are not entrained in the air stream, since they could cause corrosion and/or erosion problems in the valves and turbines.

5.1.2 Hot Water Storage

The properties of water make it particularly attractive as a fluid used for TES. It has high specific heat and good thermal conductivity. It is inexpensive and has low pumping requirements. Its limitations are its freezing point and its requirement for pressurized containment at high temperature. In addition, in order to lessen corrosion, potential dissolved oxygen in the water should be limited.

Two methods are available for using hot water thermal storage techniques. In the first, two storage volumes are required, one for hot water and one for cold water. The second requires only one volume in which water undergoes temperature stratification. In either case a heat exchanger is required to transfer heat from the compressed air to the water as the air goes to storage and then from the water to the air as the air leaves storage. Figures 5-2 and 5-3 show possible system and storage configurations that could be used with hot water TES.

For the purposes of this assessment the air leaving the compressor was assumed to be at 600°F, the stored hot water was assumed to be at 550°F, and the returning air was assumed to be heated to 500°F. Air entering and leaving the air reservoir was assumed to be at 150°F.

FIGURE 5-2. Two Reservoir Hot Water Thermal Storage

FIGURE 5-3. Stratified Reservoir Hot Water Thermal Storage

Since underground storage was found to be less expensive than above-ground storage, only underground storage will be considered for the hot side. The water storage working volume is about 250,000 ft³. An additional 20% increase in volume will be added to minimize the risk of overrunning the storage volume. The total hot water storage volume becomes 275,000 ft³.

The cost of excavating this volume was determined in the same manner as for a pebble bed. A summary of the costs is shown in Table 5-3.

If an above-ground pond was required for the cold water storage, its cost could be estimated based on a cost of \$2.50/m³ (\$0.0708/ft³) for excavating and lining.⁽²⁸⁾ This would make the cold water storage costs about \$20K. The total cost for the hot and cold water storage becomes \$3.4M.

Storage pressure must be greater than 1000 psia since it must exceed the saturated pressure of water at 550°F. Hence the heat exchanger and hot storage vessel must be designed to withstand high pressure. The potential energy associated with the pressurized cold water leaving the heat exchanger can be recovered by use of a turbine generator system. However, no attempt was made to estimate costs of the turbine generator as a part of this study.

The heat exchanger cost can be estimated based on a cost of \$10 to \$15/ft² of surface,^(a) an overall air side limited heat transfer coefficient of 100 Btu/ft²hr°F, and inlet and outlet temperatures specified above. The heat transfer area can be determined from

$$Q = UA \Delta T_{lm}$$

where $U = 100 \text{ Btu/ft}^2\text{hr}^\circ\text{F}$

$Q = 2000 \text{ MW-hr}$

$\Delta T_{lm} = 50^\circ\text{F}.$

(a) The value of \$10 to \$15/ft² of heat exchanger area is based on information received from Dan Hettrich, a consultant for Bos Hatten, Inc. and from Cliff Webb of Thermxchanger, Inc.

TABLE 5-3. Excavation Costs for a 275,000 ft³ Storage Volume at a Depth of 1000 ft

Item	Cost/ft, \$	Length, ft	Total Cost, \$M
15- to 20-ft lined access shaft	2500	1000	2.5
4-ft vent shaft, raise bored	100	1000	0.1
275,000 ft of storage, 15-ft drifts	350	1600	0.56
Drift lining with 10-gauge carbon steel	100	1600	0.16
Muck handling equipment			<u>0.06</u>
		Total	3.38

Therefore, the required heat exchanger area will be 140,000 ft² and at a cost of \$10 to \$15/ft² the cost of the heat exchanger will be \$1.4M to \$2.0M. If we assume that the one heat exchanger can be used to deliver heat to the water when air goes to storage and is later used to return heat to the air as it returns from storage, then the cost of the hot water storage system becomes \$4.8M to \$5.4M. If two heat exchangers are required, one on the compression side of storage and one on the turbine side of storage, the systems cost will be increased to \$6.2M to \$7.4M.

5.1.3 High Temperature Oil

Oil is a feasible candidate as a TES material. It can be stored at relatively high temperatures and low pressures. The low pressure permits above-ground storage with currently available storage systems. The systems illustrated in Figures 5-2 and 5-3 for water would be equally applicable for oil. In addition, we could consider a system like that in Figure 5-3 where a solid material (rocks) could be added to reduce the required volume of oil.

The cost of a high temperature oil TES system can be estimated based on the cost of oil, the containment vessels and heat exchangers. Based on ρC_p of 28 Btu/ft³°F and a ΔT of 370°F it would require 660,000 ft³ of oil to store 2000 MW-hr of thermal energy. The ΔT of 370°F represents a change in temperature of the oil from 150°F to 520°F. The temperature of 520°F was obtained from an Exxon study⁽²¹⁾ where oil was studied as a TES medium for

storing feedwater heat. Above 520°F the rate of degradation of the oil is too rapid to insure long term stability. Exxon is presently working on the development of oils that can withstand higher temperature and the development of oil purification devices.

The present cost of heat transfer oils varies from about 90¢/gal for Caloria HT 43 to over \$16/gal for silicon oils. Table 5-4 lists several heat transfer fluids with their costs and maximum recommended operating temperatures. All of the heat transfer fluids will degrade at the temperature proposed for the CAES application. The degradation will require makeup fluid and either reprocessing or purification. The cost of the makeup fluid will depend on the resale value of the degraded fluid. Vendors indicate that the worth of spent fluid depends on the degree of degradation and can be anywhere from a few percent to about 60% of the cost of new fluid. Used Caloria HT 43 can probably be resold for 30 to 40% of the existing price. The life of Caloria HT 43 is expected to be between 5 and 10 yr.

If Caloria HT 43 is used as the heat transfer fluid, the initial supply of fluid for the working volume would cost \$4.4M. If we add 20% to that volume to be sure that we have a margin of safety built into the system, the oil will go up to \$5.3M. Based on the Exxon study, the storage dependent facilities will cost about two-thirds of the oil costs or about \$3.5M. Their storage facilities consist of two hot oil storage tanks, two cold oil storage tanks, surge tanks, and an inert cover gas system.

TABLE 5-4. A Sampling of Costs and Maximum Operating Temperatures of Heat Transfer Fluids

<u>Fluid</u>	<u>Cost/gal, \$</u>	<u>Cost for 660,000 ft³, \$M</u>	<u>Maximum Recommended Operating Temperature</u>
Caloria HT 43	0.90	4.44	600°F bulk liquid 630°F to 650°F metal surface
Amaco 4199	1.10	5.43	400°F (thermal cracking above 500°F)
Dow Corning 200	16.00	79	400°F to 550°F
Dowtherm G	7.00	34	650°F

A heat exchanger will also be required to transfer heat from the air to the oil. Based on input from the vendors, the cost of the heat exchanger for air/oil applications can be approximated at \$7.5 to \$10/ft² of surface area. This will give a heat exchanger cost of \$1M to \$1.4M. This cost will be doubled unless the heat exchanger on the compressor side of storage and the heat exchanger on the turbine side of storage can use valving to control the source and destination of the fluid going through the heat exchanger.

If the safety considerations make it necessary to include an intermediate water loop between the air and oil loops, then additional costs will be incurred due to the need for an air/water heat exchanger and an oil/water heat exchanger. The initial costing of the oil system assumes that the intermediate system is not required. Based on this assumption the estimated costs for the oil system are summarized in Table 5-5.

TABLE 5-5. Cost Associated with an Oil TES System Used to Augment CAES

<u>Item</u>	<u>Cost, \$M</u>
Oil (Caloria HT 43)	5.3
Oil Storage	3.5
Heat Exchanger	<u>1.0 to 2.8^(a)</u>
Total	9.8 to 11.6

(a) Depends on the number of heat exchangers required.

5.2 LATENT HEAT STORAGE

The latent heat associated with a phase change can be used to store thermal energy as the heat of evaporation, heat of fusion, heat of solution, or heat associated with a solid-solid phase transition. Latent heat systems are used when large quantities of energy are to be stored and recovered over a small temperature span. There are several disadvantages associated with latent heat devices.

1. Large volume changes usually accompany the phase change. The volume changes have major impact on the design of containment, piping and heat exchange equipment compatability problems.
2. Salts are one of the primary candidates for latent heat storage. They pose severe corrosion problems especially if there is the possibility of moisture and oxygen contamination of the salt.
3. Some of the materials used in latent heat systems have been observed to degrade when thermally cycled.
4. Perhaps the major disadvantage to using a phase change material is a lack of engineering experience associated with its use.⁽²³⁾

To be effective, the material used for TES should have a melting point in the proper temperature range and have a high heat of fusion. It should melt congruently, not supercool, and should be stable, inert, nonflammable, nontoxic and inexpensive. Costs of selected salts are shown in Table 5-6. Some materials having melting points in the temperature range needed for CAES systems are listed in Table 5-7. A possible CAES system augmented with a phase change TES system is shown schematically in Figure 5-4.

TABLE 5-6. Prices of Selected Salts (Technical Grade).
Cost neglects shipping charges.

<u>Salt</u>	<u>Price, \$/lb</u>
KNO_3	0.14 to 0.23
NaNO_3	0.7
NaNO_2	0.23
NaOH	0.16
ZnCl_2	0.56
KCl	0.03
LiNO_3	1.60
$\text{Ba}(\text{NO}_3)_2$	0.26

TABLE 5-7. Summary of Data for Selected Salts⁽²⁹⁾

Heat Storage, cal/g	Salt or Salt Mixture	Average Molecular Weight	Density Solid	Liquid, g/cc	Melting Point, °C	Specific Heat, (cal/°K mole)/(cal/g)		Heat of Fusion, cal/g
						solid	liquid	
107	NaNO ₃ /KNO ₃	93.06	2.18	1.88	220	~30/0.32	33.3/0.35	32.8
80	KCl/ZnCl ₂	107.91	2.61	2.2	228	~17/0.16	~20/0.19	39.6
38	SnCl ₂	189.6	3.95	3.36	246	19-21/0.1	--	15.8
68	KCl/ZnCl ₂	118.39	2.74	2.34	262	~18/0.15	22.5/0.10	30.4
	NaCl/ZnCl ₂	104.76	2.74	2.3	262	--	--	39.9
171	LiNO ₃	68.95	2.38	1.77	264	20-28.5/0.35	26.6/0.39	88.5
	NaNO ₂	69.0	2.170	1.8	271	--	--	--
53	ZnCl ₂	136.28	2.910	2.52	283	18.3-20.3/0.14	24.1/0.18	18.0
94	KNO ₃ /Ba(NO ₃) ₂	120.98	2.41	--	287	~37/0.3	~38	27.7
130	NaNO ₃	84.99	2.26	1.9	307	22-38/0.35	37.0/0.43	43.5
146	NaOH	40.00	2.13	1.78	318	19.2/0.48	20.4/0.51	38.0

FIGURE 5-4. CAES System Augmented with Salt TES

Based on the density and specific heat values in Table 5-7, it will probably require between 30M and 60M lb of salt in order to store 2000 MW-hr of thermal energy using a salt temperature change of 400°F. Based on an assumed salt price of 15¢/lb, this represents a salt cost of \$4.5M to \$9.0M.

In order to get the heat into the salt, a heat exchanger must be used. Due to the lack of engineering experience associated with designing a heat exchanger for a change of phase material the cost of the salt/air heat exchanger is at best an educated guess. In this study we assumed that the transfer process is air side limited and used 100 Btu/ft²hr°F as a heat transfer coefficient. For 200 MW of energy transfer, 140,000 ft² of surface area will be required. Vendors indicate that the cost of carbon steel heat exchangers for water/air would be \$10 to \$15/ft². For oil/air, the cost would be \$7.50 to \$10/ft². The pressure conditions of the air/salt heat exchanger system are probably similar to the air/oil; however, salt will probably require the use of stainless steel instead of carbon steel for the heat exchangers. If carbon steel can be used then the heat exchanger cost might be as low as for the air/oil unit (\$1M to \$1.4M), but would be expected to be substantially more due to thermal expansion considerations. If stainless steel must be used, the cost of the heat exchanger surface area will be

increased to \$35 to \$50/ft² due to material selection alone. Additional costs will be incurred to allow the heat exchanger design to accommodate the volume change that accompanies the phase change. Therefore, the cost of the stainless steel heat exchanger is estimated to be at least \$7M. Regardless of what construction material is used, it should be noted that a salt/air heat exchanger is not considered to be near-term technology. A significant amount of developmental work will be required to make the system workable.

Based on values given in Table 5-7, the volume of salt required to provide the 2000 MW-hr of storage is in the range of 0.2M to 0.4M ft³. This is about half the storage volume required for an oil TES system. The containment cost would be expected to be at least as great as the oil system on a volume basis. This would make the containment costs \$0.8M to \$1.4M.

The salt TES system cost of containment, material, and heat exchanger for 2000 MW-hr of storage is estimated to be \$6M to \$17M. The large spread of cost reflects the uncertainty in material selection and related heat exchanger costs. A summary of the costs is given in Table 5-8.

TABLE 5-8. Cost Associated with a Phase Change Material TES System to Augment CAES

<u>Item</u>	<u>Estimated Cost Range, \$M</u>
Salt	4.5 to 9.0
Storage	1.0 to 7.0
Heat Exchanger	0.8 to 1.4
Total	6.3 to 17.4

5.3 SUMMARY

The estimated costs of each of the four TES systems that might be used to augment CAES are shown in Table 5-9. Based on this assessment, the most promising near-term technologies are pebble beds and high temperature water. The factor having significant influence on costs of pebble beds and high

TABLE 5-9. Summary of Estimated Costs of TES Concepts
Required for 2000 MW-hr of Thermal Storage

TES System	TES Material			Containment System		Heat Exchangers			Total Investment, \$M	Storage Cost, \$1000/10 ⁶ Btu
	Quantity, 1M ft ³	Unit, Cost	Total Cost, \$M	Storage Volume, 1M ft ³	Total Cost, \$M	Area, ft ²	Unit, \$/ft ²	Total, \$M		
Pebble Bed										
Above-Ground Storage	1.2-1.5	\$2.50/yd ³	0.11-0.14	1.2-1.5	30-36	--	--	--	30-36	4.4-5.3
Underground Storage	1.2-1.5	\$2.50/yd ³	0.11-0.14	1.2-1.5	5.7	--	--	--	5.8-6.0	0.85-0.88
High Temperature Water	0.275	--	--	0.275 hot 0.275 cold	3.38 0.02	140,000 ^(a) 280,000 ^(b)	10-15	1.4-2.0 2.8-4.0	4.8-5.0 6.2-7.4	0.700-0.730 0.900-1.0
High Temperature Oil	0.8	90¢/gal	5.3	0.8 hot 0.8 cold	3.5	140,000	7.5-10	1-1.4	9.8-10.2	1.4-1.5
Fusible Salt	0.2-0.4 30-60 x 10 ⁶ lb	15¢/lb	4.5-9.0	0.2-0.4	0.8-1.4	140,000	7.5-50	1-6.7	6.3-17	0.92-2.5

(a) Assumed only one heat exchanger. Heat exchanger used for storage and extraction of heat.

(b) Assumed two independent heat exchangers used; one on the compression side of storage, one on the expansion side of storage.

temperature water TES systems are the costs of heat exchangers and the TES material volume and cost. The pebble bed interacts directly with air, whereas separate high pressure heat exchangers are required for the water system. The energy storage density of the water system is greater than that of the pebble bed system; hence, the volume requirements and excavation costs are less for the water system. The trade-offs between material cost, volume and heat exchanger requirements make the cost of pebble bed and hot water systems look about the same.

In order to be competitive with other systems, both pebble bed and hot water storage must be done underground. Above-ground storage would require pressure vessels, the cost of which would make the system unattractive. Analysis done for pebble beds shows that above-ground storage is over five times more expensive than underground storage.

The major cost disadvantage associated with oil systems is the cost of the oil. If the oil cost could be ignored as part of the storage cost then oil systems could be attractive. Alternatively, the oil could be circulated through a fixed bed of solids (pebble bed), thus reducing the volume of oil required. However, this option was not considered in this study.

The major cost associated with salt systems is the cost of the salt. Also, the technology is not well developed and any use of phase change TES systems will require major and possibly lengthy technology development programs.

6.0 COMBINED THERMAL/AIR STORAGE IN AN AQUIFER

Aquifers, or more generally, underground porous media, have excellent potential for use as CAES reservoirs. If the air is injected at the compressor outlet temperature, the compression heat would be stored in the rock matrix in the vicinity of the wells where it would be recovered upon discharge. The TES reservoir and air reservoir would thus be combined in a single system in this case. Although the concept is simple, its technical feasibility depends on many factors relating primarily to reservoir stability. These questions are being addressed in ongoing work through the Reservoir Stability Program and will not be dealt with here.

The purpose of this analysis is to assess the economic merits of combined thermal/mechanical energy storage in an underground porous reservoir such as that illustrated by the system in Figure 4-3. For brevity, we will refer to all porous structures as "aquifers" even though that term may not properly describe all possibilities. To compare costs of combined CAES/TES in an aquifer with the costs of the other options, we seek to compute the additional investment cost for the high temperature reservoir in \$/kWh of thermal storage capacity. However, TES in an aquifer cannot be accomplished by the "add-on" approach which is possible with the other TES options proposed for CAES applications. Rather, the entire aquifer reservoir system must be designed specifically for the particular storage temperature chosen for each application. Therefore, the only practical means of computing the costs of the addition of TES in an aquifer is to compute separately the costs of the reservoirs for high temperature and low temperature storage. The additional cost of TES is then the difference between these two figures.

If the cost estimate for TES is to be meaningful, we must optimize the designs of both systems. Even then, however, the cost estimate will be highly speculative since it is computed as the difference between two estimates, both of which have a considerable margin of error.

Our cost estimate for the addition of TES in an aquifer will be performed as an extension of the previous work by Katz^(30,31) in which he analyzed a CAES system based on the Belle River Mills natural gas storage reservoir. We will borrow heavily on this work for data and will employ similar optimization procedures. The system design will be altered for our present needs and costs will be modified where necessary.

6.1 BELLE RIVER MILLS DESIGN WITH TES

The design and operating characteristics of the Belle River Mills natural gas storage reservoir are described by Katz in his general work on the storage of fluids underground⁽³⁰⁾ and in his more recent study relevant to CAES applications.⁽³¹⁾ The characteristics of the field, as given by Katz, are summarized in Table 6-1. The conceptual design of a CAES system based on this structure is described in Table 6-2 and Figures 4-3 and 6-1.

TABLE 6-1. Characteristics of the Belle River Mills Gas Storage Reservoir

Type	Niagaran Reef
Depth	2150 ft
Surface Area	750 acres
Length	~2 mi
Width	~1/2 mi
Max Closure	~360 ft
Discovery Pressure	1132 psia
Discovery Temperature	64°F
Permeability	320 md
Porosity	10%
Max Storage (at 1500 psia)	49×10^9 scf
Total Storage Space	556×10^6 ft ³

TABLE 6-2. Thermodynamic Cycle Design Data

Symbol	Parameter	Storage Conditions	
		150°F	625°F
P_{11}	HPT inlet pressure, psia	588	588
T_{12}	HPT inlet temperature, °F	1000	1000
P_{15}	LPT inlet pressure, psia	206	76
T_{15}	LPT inlet temperature, °F	2000	2000
W_t	Turbine specific work, kWh/lb	0.107	0.10
H_t	Turbine heat rate, Btu/kWh	3883	3426
P_8	Maximum compressor outlet pressure, psia	732	832
W_c	Compressor specific input, kWh/lb	0.069	0.080
$(P_f)_{\max}$	Maximum formation pressure	745	806
$(P_f)_{\min}$	Minimum formation pressure	683	720
$(M_r)_{\max}$	Stored air mass at maximum pressure, lb x 10 ⁹	1.835	1.428 ^(a)
$(M_r)_{\min}$	Stored air mass at minimum pressure, lb x 10 ⁹	1.682	1.275 ^(a)
ΔM_r	Working air requirement, lb x 10 ⁹	0.153	0.153

(a) Computed assuming $T_{\text{ave}} = \frac{625 + 150}{2} = 388^\circ\text{F}$.

The Belle River Mills structure is a reef which shows very little evidence of water drive so that this reservoir behaves essentially as a constant volume porous reservoir. After completion of bubble development, many aquifers (except those with extreme water drive) should also behave essentially as constant volume reservoirs over the short term. Therefore, if we include costs of bubble development (which Katz neglected), the cost estimate for CAES development of the Belle River Mills structure should also be approximately relevant to other aquifer systems at about the same depth, pressure,

WELL SPACING 500'
WELLS ARE 7" NOMINAL ID
TOTAL NUMBER OF WELLS 105

FIGURE 6-1. Conceptual Design of the Air Distribution System for Belle River Mills Field

permeability, porosity and capacity. However, we emphasize that the design and ultimate cost of an aquifer reservoir is highly site-specific so that generalizations of results must be made with care.

In computing the costs of TES in an aquifer, we employ two similar designs that are based on the same site-specific data. It is thus conceivable that some of the site-specific aspects of the cost estimates may cancel to some degree. Hopefully then, our estimate for the incremental cost of TES in an aquifer will be less site-specific than the total cost of the reservoir systems upon which the estimate is based.

To perform the reservoir design optimizations, we must identify which elements of the design are to be optimized. The logical choices for optimization, in order of decreasing cost, are as follows:

- the well field,
- the surface distribution system,
- bubble development, and
- land and storage rights.

The drilling and completion of the wells is by far the greatest single cost in development of an aquifer reservoir so that optimization of the well field will be given greatest emphasis. The surface collection field comprises the second greatest cost. However, a complete optimization of the surface field is beyond the scope of this study and will not be attempted. In the design used by Katz for Belle River Mills and in the present design, the large, primary distribution pipelines comprise a significant portion of the surface costs. The pipeline design is readily amendable to optimization, so we chose to optimize this element of the system, as did Katz. We will use Katz's estimate for the cost of the balance of the surface lines, adjusted in proportion to the number of wells.

The cost of bubble development was not included in the analysis by Katz. In developing a porous reservoir, it will be necessary to grow the air bubble by displacement of the fluid present at discovery. In the case of Belle River Mills, all of the gas present at discovery would have to be produced and replaced with air. In an aquifer, the water would have to be displaced. In either case, the cost of pumping energy and the associated operational costs for equipment and manpower should be included in the reservoir costs. Therefore, we will include estimates of bubble development costs in the present analysis but we will not attempt to optimize the development procedure.

In summary, we will optimize only the well field cost and the pipeline cost for the two storage temperatures. Since these two items account for over half the system cost, the resulting designs should be sufficiently optimal for our purposes.

The design to be analyzed in this study, as illustrated in Figures 4-3 and 6-1 and Tables 6-1 and 6-2, will differ somewhat from that proposed by Katz. The main difference is the length of the pipelines which connect the field to the surface plant. Reasonable caution would probably require siting the surface plant at or slightly beyond the fringe of the storage field. There are no apparent constraints inherent in the CAES concept which would require remote siting of the surface plant. In the design analyzed here, we assume that the surface plant is sited at a distance of 1/2 mi from the edge of the structure, as illustrated in Figure 6-1. Since the field is only about 1/2-mi wide, this would place the surface plant well off the reservoir.

In the present design, the two pipelines are assumed to turn at the midpoint of the field, such that the primary collection branches extend in opposite directions--a distance of about 3/4 mi each way following the crest of the field. The total length of each of the pipeline branches will thus be only about 1-1/4 mi, as opposed to 4 mi in the design used by Katz. The collection lines from the wells are assumed to connect into the pipelines at uniformly spaced intervals as shown.

The comparative design data for storage at 150°F and 625°F are given in Table 6-2. The present design for 150°F is essentially the same as that analyzed by Katz with the exception of shorter pipelines, as noted above. Although Katz has already performed an optimization of the system for 150°F storage, we will repeat his optimization with data relevant to the present design to provide a direct comparison with the high temperature system analysis.

6.2 SYSTEM DESIGN OPTIMIZATION

The basis of the optimization procedure is very straightforward. We note that the friction in the system is paid for in two ways: higher equipment costs and greater use of compression energy. The sum of these costs will be referred to as the incremental costs of compression. Friction can be reduced by increasing the size and/or the number of wells and distribution

lines which comprise the major costs of the reservoir system. The optimum design is then taken as that which minimizes the sum of the reservoir investment costs and the incremental costs of compression. We apply this procedure to determine the optimum number of wells and the optimum pipeline diameter. The details of the optimization are included in the Appendix.

The results of the systems optimizations in the Appendix indicate that the optimal pipeline diameter and the optimum number of wells are essentially the same for storage at 150°F and 625°F. The analysis yields an optimum pipeline diameter of 48 in. for both systems. This is in agreement with Katz's results in spite of the difference in the length of the pipelines employed. The optimum number of wells for 150°F storage was found to be 105 (versus 100 by Katz). For 625°F storage, the optimum number of wells is found to be 110. These differences are probably insignificant considering the nature of the optimization procedure. We thus conclude that the same basic design is optimal for both temperatures, providing the appropriate modifications in materials and construction are incorporated to account for the higher temperature.

Initially, it is somewhat startling to discover that the same basic design could be optimal for both high and low temperature storage so a few words of explanation are in order. We again point out that the optimization balances the operational costs attributable to frictional losses against the additional investment costs for increasing the "size" of the system. At high temperatures, frictional losses increase for a given well configuration as does the cost of the wells and distribution lines. For the assumptions employed in this analysis, these costs very nearly compensate so that the same design is optimal for both cases. If the relative costs of charging power and capital is changed, this balance will work out somewhat differently.

To check the sensitivity of the optimization to charging power costs, the incremental costs of compression for the 625°F case was increased by 50% and a new optimization was performed. The optimal pipeline for this case was closer to 54 in. and the optimal well number rose to about 125 wells. This would result in a cost increase of about 10% which is significant but not overwhelming.

We will estimate the costs of the reservoir for 150°F storage using the 1975 cost data provided by Katz, escalated by 20% to account for inflation. The costs of the surface air distribution system for 625°F will be escalated an additional 20% to allow for the costs of insulation (or burying) and for expansion joints required by the higher temperatures. Surface work, grading, roads and buildings will be included in the cost of the CAES surface plant.

The cost of the high temperature wells is difficult to predict without specific design data. We assume that multiple casings will be required with insulation packing between inner and outer strings. High temperature cementing will also be required. Due to the multiple strings and packing, the nominal well bore size and the outer casing will be larger than for a cold well. Casing and completing the well normally account for about 80% of the well cost. Since each well is essentially a well within a well, we should expect approximately two times 80%, or a factor of 1.6 times the base well cost. To account for the extra materials and labor associated with the high temperature completion, we will adopt a well cost estimate of twice the base cost for 625°F storage.

The above estimate of high temperature well costs is in rough agreement with previous estimates, but no hard data is currently available for verification. Further work on hot well design is needed. Since the well costs account for over half of the total costs of an aquifer reservoir, our cost estimate for the high temperature reservoir system must be considered highly speculative.

The cost of the 48-in. pipelines at 150°F is estimated from data provided by Katz (escalated by 20% to account for inflation). The pipeline cost for 625°F is based on the same figure, escalated an additional 20% to allow for insulation and/or burying of the pipe. It may also be necessary to provide for expansion of the pipe, but this is not specifically provided for in this estimate.

The conceptual design of the collection field is illustrated in Figure 6-1. The primary collection lines which connect to the 48-in. pipe are assumed to be 20-in. diameter and the secondary lines which connect to

the wells are assumed to be 10-in. diameter. This is by no means optimal but the results should provide a reasonable estimate of the costs. The length of the lines is estimated based on 500-ft well spacing, 105 wells and a field which measures 1/2-mi wide by 2-mi long. The cost of the pipe is estimated from the data provided by Katz (escalated for inflation and insulation as was done for the pipeline). The cost of valves and fittings is estimated to be about equal to the cost of the pipe for 150°F storage. For 625°F storage, the valves are more expensive due to the need for high temperature packing and seals. For consistency, the 20% escalation for high temperature is also used for the valves and fittings.

The estimated cost of the collection field for 150°F agrees well with the results of Katz (see Table 6-3). Further comparison is impractical, however, since the details of the design used by Katz are not provided. The most speculative item in the collection field cost estimate is for the valves and fittings. It is not clear whether or not each well needs individual valving, or if welded or bolted connections can be used. Detailed design of a specific system is required to adequately specify these items.

The costs of acquiring the reservoir are based on the estimates provided by Katz. There is no need to assume that high temperature storage will require greater land costs in this case. The costs of site location, exploration and field testing are not included in our estimate since no reliable data are available. This aspect of the cost is highly site-specific and is potentially a very significant part of the system cost. For this reason, a large contingency (25%) is provided in our estimate in each case.

Finally, we must estimate the cost of bubble development. The major element in the bubble development cost is for power to run the compressors. From the data given by Katz, the actual storage volume of the Belle River Mills reservoir is about 556M ft³. At 745 psia, 150°F the volume contains about 1.84×10^9 lb of air. For these conditions, we may assume a compression specific work of about 0.069 kWh/lb. Bubble development thus requires about 1.27×10^8 kWh of compression energy. We will adopt a charging power cost of \$0.025/kWh which we will assume includes operational costs for equipment, manpower and maintenance attributable to bubble development. The total

TABLE 6-3. Summary of Reservoir Cost Estimate

6-3a Capital Cost Estimates

Cost Element	Quantity	Estimated Costs, \$M	
		150°F	625°F
Wells, 2150-ft long x 7-in. diameter	105	10.5	21.0
Surface Distribution System		5.0	6.0
Bubble Development	1.27×10^8 kWh	3.2	3.2
Land and Storage Rights			
Purchase in Full	400 acres	1.2	1.2
Storage Rights	1000 acres	0.4	0.4
Contingency ^(a)	25%	<u>5.1</u>	<u>8.0</u>
TOTALS		25.4	39.8

6-3b Capacity Ratings and Costs

Parameter	Unit	Storage Conditions	
		150°F	625°F
Power Rating ^(b)	MW	818	764
Power Related Cost	\$/kW	31.1	52.0
Rated Generation Capacity ^(c)	kWh $\times 10^7$	1.64	1.53
Capacity Related Cost	\$/kWh	1.55	2.60
Rated Thermal Capacity ^(d)	kWh(t) $\times 10^6$	--	4.84
TES Capacity Related Cost	\$/kWh(t)	--	3.0
	\$/ 10^6 Btu	--	880

(a) Includes site exploration.

(b) For 7.64×10^6 lb/hr

(c) For 1.53×10^8 lb working air.

(d) Estimated for 1.53×10^8 lb and 460°F effective ΔT .

cost for bubble development at 150°F to 50 atm is then estimated to be about \$3.2M. This estimate may be conservative since the pressure is assumed constant at 50 atm. It may be possible to increase the compression ratio gradually as the bubble develops, thus requiring less power.

Computation of the bubble development costs for storage at high temperature is dependent on the bubble development strategy employed. We will assume that the bubble is developed at 150°F and dehydrated before hot air injection begins. The initial bubble development costs will thus be about the same as that computed for the low temperature system. The dehydration will be accomplished by cycling the plant cold with warm dehydrated air during charging, just as would be done for a system without TES. This may take several months or possibly a year or more. The thermal development will subsequently occur over an extended time period during which the maximum pressure will be slowly increased to maintain the required flow rate and delivery pressure. The costs of dehydration and thermal development will thus be spread out over the early years of operation and will be reflected as higher operating costs. It is impractical to assess these costs of the initial bubble development. We thus assume that the bubble development costs for 150°F storage are also applicable to the 625°F storage system.

The cost estimates for the major system elements are summarized in Table 6-3. The total costs for the reservoirs are transformed to power-related costs (\$/kW) by division with the rated power outputs of the two systems. The capacity-related costs (\$/kWh) are computed by dividing the total cost by the maximum continuous generation capacity. For the 150°F system,

$$\text{maximum output} = (1.53 \times 10^8 \text{ lb})(0.107 \text{ kWh/lb}) = 1.64 \times 10^7 \text{ kWh}$$

$$\text{capacity related costs} = \frac{\$31.1 \times 10^6}{1.64 \times 10^7 \text{ kWh}} = \$1.90/\text{kWh}$$

where a similar computation holds for the high temperature system.

The incremental cost for adding TES is the difference between the capacity costs which equals \$1.5/kWh of continuous generation capacity. Alternatively, we need to compute the cost in terms of thermal capacity. The maximum heat stored and recovered is equal to the air enthalpy difference between the assumed delivery temperature of about 600°F and the reservoir temperature of 150°F.

$$\begin{aligned}\text{Heat required} &= (1.53 \times 10^8 \text{ lb})(0.24 \text{ Btu/lb}^\circ\text{F}) \left(\frac{450^\circ\text{F}}{3412 \text{ Btu/kWh}} \right) \\ &= 4.84 \times 10^6 \text{ kWh(t)}\end{aligned}$$

The incremental cost for TES is then

$$\text{TES cost} = \frac{\$39.8 - 25.4 \times 10^6}{4.84 \times 10^6} = \$3.0/\text{kWh(t)}$$

These costs may be compared to the costs of the separate TES systems considered. However, it is important to note that the aquifer thermal capacity is essentially unlimited. If longer compression and expansion times are used at the same flow rates, the capacity is expanded at essentially no additional cost. This is a distinct improvement over the separate TES systems, where TES capacity has to be added in proportion to generation capacity. It is difficult to fairly assess the economic value of this feature.

Finally, we note that the above cost is only slightly higher than the cost of the least expensive competitor. Although aquifer TES has no apparent cost advantage, it might be the best approach where the air reservoir is to be an aquifer.

6.3 CONCLUSIONS

The analysis in Section 6.1 and in the Appendix yields several interesting results. The optimization of the well field indicates that the basic geometry of the well field and collection lines would be very nearly the same

for storage at 625°F as it is for 150°F. The optimum pipeline size and the optimum number of wells were only slightly greater for 625°F storage. This conclusion was reached by balancing the incremental costs of compression against additional capital costs.

The cost of the system with storage at 625°F is estimated to be substantially higher. This is due primarily to higher costs for high temperature wells, estimated at double the cost at 150°F. The cost of compressors and turbines is also assumed to be greater for 625°F storage, but this cost is not included in this estimate.

From Table 6-3, the incremental cost for the reservoir (due to addition of TES by storage at 625°F) is about \$3.0/kWh(t) or \$880/10⁶ Btu. From Table 5-9, this is about the same as the cost of the most favorable options considered here. Although hot storage in an aquifer holds no apparent advantage over the other TES options considered, the cost is comparable to the cost of the best separate TES system considered.

Finally, we note that the incremental cost of \$3.0/kWh(t) is three times greater than the breakeven cost for TES of \$1.14/kWh(t) estimated in Section 4.2. The latter figure includes the additional costs for turbines and compressors. The cost estimate for the aquifer includes only those costs directly associated with the reservoir. Thus, there appears to be no economic incentive for hot storage in an aquifer at this time.

7.0 ASSESSMENT OF FLUIDIZED BED COMBUSTION SYSTEMS FOR CAES APPLICATIONS

Fluidized Bed Combustion (FBC) offers a promising new technology for the clean, efficient combustion of coal without the need for stack gas scrubbing. In addition, FBC combustors are compact and offer other design and operational advantages. There are a number of FBC operational problems yet to be resolved, but the developmental effort is intense and there is reason to believe that FBC systems will be practical in the near future.

There appears to be an excellent match between FBC and CAES system requirements. FBC systems require relatively large airflows which CAES provides. The ability to burn coal in a CAES system would certainly be advantageous considering the present energy supply picture. When used as air heaters, FBC systems should have excellent turn-down capability⁽¹¹⁾ and thus deserve serious study.

The material presented here is a summary of a preliminary study initiated at PNL late in FY-1977. This work will continue in FY-1978 and will be expanded to include a technical and economic feasibility analysis. Some of this work will be done through contractual commitments yet to be arranged.

7.1 ANALYSIS

The basic CAES/FBC concept considered is illustrated schematically in Figure 7-1. An atmospheric FBC (AFBC) system was selected for the analysis since it represents a nearer-term technology than does a pressurized FBC system. The principal features and potential advantages of the proposed cycle are summarized below.

The cycle has multifuel capability thereby eliminating the dependence on natural gas and/or oil. This dependence is a fundamental weakness inherent in conventional CAES cycles.

The turbine and FBC run continually in the proposed cycle, as opposed to intermittent duty in conventional CAES systems. This results in a number of significant improvements, including:

FIGURE 7-1. Conceptual Design of a Peaking Facility Using AFBC and CAES

- fewer losses due to off-design conditions experienced during startup and shutdown,
- better control of effluents,
- simplified control systems,
- reduced thermal cycling stress, reduced maintenance, and longer life in the turbines and combustors.

The FBC is ideally suited to combustion of coal for several reasons:

- FBC systems can directly utilize a wide range of high sulfur and low rank fuels.

- Sulfur is removed in the bed with high efficiency, minimizing exhaust cleanup. NO_x emissions are also minimized.
- Exhaust particulates do not sinter, simplifying exhaust gas cleanup and minimizing potential erosion of turbines and exhaust handling equipment.

Finally, it is certain that a significant share of our future power generation capacity will use coal and it is likely that FBC technology will be used to an increasing extent. It will thus be advantageous to have base load and peaking facilities dependent on the same fuel and the same combustion technology.

The CAES/FBC system consists of five primary elements: 1) the motor/generator (M/G) set, 2) the compressors with their auxiliary coolers, 3) the turbine set, 4) the AFBC and auxiliaries, and 5) the air storage reservoir.

The turbine set, consisting of a high pressure turbine T1 and a low pressure turbine T2, operates continually, as does the AFBC. The turbine output drives either the primary compressor set or the M/G set. When the turbine drives the primary compressor set, part of the compressed air is used to operate the cycle with the remainder being delivered to the air reservoir. During compression the M/G set may be used as much as desired to drive the auxiliary compressors, thereby utilizing excess power from the grid to charge additional compressed air into the reservoir. This allows for longer generation times, or alternatively, higher power during generation. When peak power is needed, the compressors are decoupled and the turbine is coupled to the M/G set, delivering electrical power to the grid. During generation, process air is discharged from the air reservoir since the compressors are inactive.

We now trace the air around the heating-expansion cycle. Upon leaving the reservoir (or from the compressor discharge), the air is preheated in the exhaust gas recuperator R1, heated in the first pass through the AFBC and expanded in the high pressure turbine T1. Upon leaving T1 the air may be reheated in the compressor heat recuperator R2 (optional), further heated in a second pass through the AFBC and delivered to the low pressure turbine

T2. The air is then expanded in T2 where the fluidizing/combustion air is extracted at an intermediate pressure. The remaining air is expanded, passed through the low temperature end of the exhaust gas recuperator, and exhausted to the atmosphere.

In order to perform a preliminary thermal analysis of the combined CAES/FBC system, several simplifying assumptions were made. These are listed in Table 7-1. In addition to these, the pressure ratio of the compressors and turbines were chosen to minimize the work of compression and to maximize work output.

TABLE 7-1. Major Assumptions Used in Performing Thermal Analysis

Parameter	Value
P_1 (compressor inlet pressure)	1 atm
T_1 (compressor inlet temperature)	60°F
η_c (compressor efficiency)	87%
T_3 (intercooler temperature)	100°F
P_4 (maximum air pressure)	50 atm
T_5 (storage temperature)	150°F
$T_9 = T_{12}$ (air exit temperature--FBC)	1600°F
η_t (turbine efficiency)	90%
η_{rc} (recuperator efficiency)	80%
C_p (specific heat)	0.24 Btu/lbm°F
γ (isentropic coefficient)	1.4
$\left. \begin{matrix} P_{out} \\ P_{in} \end{matrix} \right\} \begin{matrix} \text{Storage} \\ \text{Fluidized Bed Heat Exchanger} \\ \text{Cyclone Separator} \\ \text{Recuperator} \end{matrix}$	 0.9 0.98 0.75 0.97
ΔP fluidizing flow = bed height x bed density	

A maximum pressure of 50 atm was assumed for the basic system. Pressure losses through various system components were assumed to be a fraction of that component's inlet pressure. The assumed pressure ratios are also given in Table 7-1.

The analysis assumed three cases. In Case I the delivery pressure of the air passing through the recuperator and FBC heat exchanger was equal to the pressure of air returning from storage. This assumption requires throttling of the air coming directly from the compressor to the FBC part of the system.

In Case II the pressure of the air delivered to the FBC side of the system was assumed to be equal to the pressure of air leaving the compression side of the cycle. In order for this to happen, a small booster compressor has to be included in the air storage system to compensate for the pressure losses in that system. The booster compressor could be used either during the delivery of air to storage or upon its retrieval. This analysis assumed the booster compressor was used when air was taken from storage.

Case III assumed a single pass of air through the FBC (i.e., no reheat). The results of the analysis are given in Table 7-2.

The results of the calculations done for Case I indicated that the recuperator located between the first and second stages of the turbine was not practical. With an assumed air temperature of 1600°F at the entrance of the turbines, a compression ratio greater than 10 must exist across the turbine before the temperature of the air leaving the turbine would be less than the temperature of the air leaving the compressors. However, a comparison of Cases I and II with Case III indicated that reheating the air is desirable in order to lower the overall heat rate and increase the storage time. Also, in Cases I and II the exit temperature of the air leaving the first stage of expansion (T_{10}) is greater than that leaving compression (T_4) by several hundred degrees. Even if full expansion of the air occurs (Case III), the air leaving the turbine is only about 200 degrees less than the temperature of the air leaving the compressor. Hence, recuperation of the compressor heat is not practical.

TABLE 7-2. Analysis Results

<u>Location in Cycle</u>	<u>Symbol</u>	<u>Unit</u>	<u>Case I</u>	<u>Case II</u>	<u>Case III</u>
<u>Compression Side</u>					
Inlet Air, 1st Stage Compression	T ₁	°F	60	60	60
	P ₁	atm	1	1	1
Inlet Air, 2nd Stage Compression	T ₃	°F	100	100	100
	P ₃	atm	7.07	7.07	7.07
Compressor Exit	T ₄	°F	581	581	581
	P ₄	atm	50	50	50
Air Storage Temperature	T ₅	°F	150	150	150
<u>Expansion Side</u>					
Air Delivery	P ₇	atm	45	50	50
During Storage	T ₇	°F	581	581	581
During Generation	T ₇	°F	150	172	172
Recuperator Exit	P ₈	atm	44.1	49	49
During Storage	T ₈	°F	798	785	--
During Generation	T ₈	°F	680	673	--
FBC Exit	T ₉	°F	1600	1600	1600
	P ₉	atm	43.2	48.0	48.0
First Stage Expansion	T ₁₀	°F	877	860	bypassed
	P ₁₀	atm	7.67	8.08	--
Inlet Second Stage	T ₁₂	°F	1600	1600	bypassed
	P ₁₂	atm	7.5	7.9	--
Fluidizing Air (Bleed)	T ₁₃	°F	1003	984	486
	P ₁₃	atm	1.93	1.93	1.93
Turbine Exit	T ₁₄	°F	877	860	359
	P ₁₄	atm	1.33	1.33	1.33
<u>Fluidizing Flow</u>					
Total Flow					
Storage	$\dot{m}$		0.3274	0.326	0.219
Generation	$\dot{m}$		0.3112	0.3313	--
Air Exit Temperature					
Storage	T ₁₈	°F	635	632	642
Generation	T ₁₈	°F	282	297	232
Heat Rate					
During Generation		Btu/kWh	3814	3767	2955
Overall		Btu/kWh	12,640	11,830	13,800

The calculations performed indicate that of the configurations analyzed, Case II is the best. In this case we assumed a constant pressure equal to the compressor output at the inlet of the FBC. The constant pressure was maintained by including a small booster compressor in the return leg of the air storage system. The power required to operate this compressor reduced the peaking power output of the turbines by about 2%. Even with this reduction the analysis indicated an overall heat rate of 11,800 Btu/kWh. The booster compressor could have been included on the compression side of storage, but this would have decreased the volume of air stored during the compression cycle.

The thermal analysis conducted to date indicates that FBC holds promise as a method for heat addition to a CAES system. However, before the combined system can be totally assessed, an economic evaluation of the system must be made. The economic evaluation was considered to be beyond the scope of this work. It would require the estimation or projection of costs of many components which are not readily available now. The economic assessment of the system will be done as a follow-on study.

7.2 SUMMARY AND CONCLUSIONS

A preliminary thermal analysis was conducted on a CAES system using FBC and an exhaust gas recuperator. The FBC and turbines in this system were designed to run continuously so that they deliver power either for air compression or electrical generation. Additional air storage can be accomplished by taking power directly from the electrical power grid. The analysis indicated that compression heat recuperation is possible but probably impractical. Based on this analysis system overall heat rates of 11,830 Btu/kWh and peaking power heat rates of 3,000 to 3,800 Btu/kWh can be obtained. This system (or variations on this basic concept) appears to be a promising new technology for CAES applications and continued study of this option is recommended.

REFERENCES

1. T. R. Schneider et al., An Assessment of Energy Storage Systems Suitable for Use by Electric Utilities. EPRI EM-264, Electric Power Research Institute, Palo Alto, CA 94304, 1976.
2. J. B. Bush, Jr. et al., Economic and Technical Feasibility Study of Compressed Air Storage. ERDA 76-76, National Technical Information Service, Springfield, VA 22151, 1976.
3. A. R. Giramonti, Preliminary Feasibility Evaluation of Compressed Air Storage Power Systems - Final Technical Report. R-76-952161-5, United Technologies Research Center, East Hartford, CT 06108, December 1976.
4. W. Mattick, H. G. Haddenhorst, O. Weber and Z. S. Stys, "Huntorf - The World's First 290 MW Gas Turbine Air Storage Peaking Plant." Proceedings of the American Power Conference, 37:322-330, 1975.
5. K. Vosburgh, Conceptual Design for a Pilot/Demonstration Compressed Air Storage Facility Employing a Solution Mined Salt Cavern. EPRI EM-391, General Electric Company, Schenectady, NY 12301, June 1977.
6. "Compressed Air Storage Underground Pumped Hydro Demo Plant." RFP(49-18)2159, Issued by U.S. Department of Energy (formerly Energy Research and Development Administration), September 1, 1976.
7. I. Glendenning, Compressed Air Energy Storage by Incorporating Heat Storage. Technical Disclosure Bulletin No. 235, Central Electricity Generating Board, Marchwood, Southampton, 504 42B, England, 1974.
8. T. Stephens, "Adiabatic Compressed Air Energy Storage." Presented at the ERDA CAES Workshop, Airlie House, VA, December 1975.
9. D. K. Kreid, Technical and Economic Feasibility Analysis of the No-Fuel Compressed Energy Storage Concept. Available from Battelle, Pacific Northwest Laboratories, Richland, WA 99352, November 1975.
10. N. F. Sather et al., Compressed Air Energy Systems Studies. Available from Argonne National Laboratory, Argonne, IL 60439, 1976.
11. Henrick Harboe, Importance of Coal for Heat and Power Generation. 354E04.74, Second Edition, Stal Laval, Ltd, Finspong, Sweden, 1974.
12. G. T. Kartsounes and N. F. Sather, Compressed Air Energy Storage System with Coal Gasification Reheat. Available from Argonne National Laboratory, Argonne, IL 60439, April 1976.

13. Proceedings of the Fluidized Bed Combustion Technology Exchange Workshop. ERDA/EPRI Conf. - 770447-P-1, Vol. 1 & 2, April 1977.
14. "Solar Power Irrigates Range." Industrial Research, 19-21, August 1977.
15. F. W. Ahrens and G. T. Kartsounes, Compressed Air Energy Storage Plant Using an Internal Combustion Compressor/Expander Engine. Available from Argonne National Laboratory, Argonne, IL 60439, January 1977.
16. W. E. Sande, W. J. Bjorkland and N. A. Brooks, Cost Analysis for Application of Solidified Waste Fission Product Canisters in U.S. Army Steam Plants. BNWL-2266, Battelle, Pacific Northwest Laboratories, Richland, WA 99352, April 1977.
17. D. R. Glenn, Technical and Economic Feasibility of Thermal Energy Storage. COO 2558-1, RCA Advanced Technology Laboratories, Camden, NJ 08102, February 1976.
18. J. L. Dooley, G. P. Frost, L. A. Gore, R. P. Hammond, D. L. Rawson and S. L. Ridgeway, A Feasibility Study of Underground Energy Storage Using High-Pressure, High-Temperature Water, RDA TR-7100-001, R and D Associates, Marina del Ray, CA 90291, August 1976.
19. G. N. Unsworth, A Review of Pumped Energy Storage Schemes. AECC 4926, Whiteshell Nuclear Research Establishment, Pinawa, Manitoba, Canada, July 1975.
20. An Assessment of Energy Storage Systems Suitable for Use by Electric Utilities. EPRI EM-264, Vol. II, Electric Power Research Institute, Palo Alto, CA 94304, July 1976.
21. R. P. Cahn and E. W. Nicholson, "Storage of Off-Peak Thermal Energy in Oil." A 76 326-9, presented to the IEEE PES Summer Meeting, Portland, OR, July 1976.
22. P. H. Margen, "Thermal Energy Storage in Rock Chambers." Proceedings of the Fourth International Conference on Peaceful Uses of Atomic Energy, UN/IAEA, A/Conf 49-P-798, Vol. 4, September 1971.
23. Solar Thermal Electric Power Systems, Systems Studies and Economic Evaluations. PB 243 836, Colorado State University, prepared for the National Science Foundation, Washington, DC 20550, November 1974.
24. I. Glendenning, "Long Term Prospects for Compressed Air Energy Storage." Proceedings of the Workshop on Compressed Air Energy Storage Systems, Airlie House, VA, 358-389, December 1975.
25. ERDA Thermal Storage Program Information Exchange Meeting, NASA Lewis Research Center, Cleveland, OH 44135, September 1976.

26. I. Glendenning, Compressed Air Storage Symposium on Long Term Studies, Marchwood Engineering Laboratories, Marchwood, Southampton, SO4 4ZB, England, October 1974.
27. E. K. A. Olsson, "220 MW Air Storage Power Plant." ASME 70GT34, presented to the ASME Gas Turbine Conference and Products Show, Brussels, Belgium, May 1970.
28. A. F. Clark, J. A. Day, W. C. Dickinson and L. F. Winters, The Shallow Solar Pond Energy Conversion System: An Analysis of a Conceptual 10 MWe Plant. UCRL-51533, Rev. 1, Lawrence Livermore Laboratory, Livermore, CA 94550, January 1974.
29. Alina Berucka, Survey and Selection of Inorganic Salts for Application to Thermal Energy Storage. ERDA-59, Technical Information Center, Springfield, VA, June 1975.
30. D. L. Katz and K. H. Coats, Underground Storage of Fluids. Ulrich's Books, Inc., Ann Arbor, MI, 1968.
31. D. L. Katz and E. R. Lady, Compressed Air Storage for Electric Power Generation. Ulrich's Books, Inc., Ann Arbor, MI, 1976.

APPENDIX

OPTIMIZATION OF AQUIFER RESERVOIR DESIGN

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OPTIMIZATION OF AQUIFER RESERVOIR DESIGN

The design of the well field and the surface collection pipeline are optimized by seeking the minimum in the sum of the investment costs and the incremental costs of compression. The procedure and data employed are essentially those used by Katz.⁽³¹⁾

A.1 INCREMENTAL COSTS OF COMPRESSION

We first estimate the incremental costs of compression. The additional compressor power required to compress 1 lbm/hr of air by an additional 1 psi may be given by

$$\frac{\Delta W}{\dot{m}_a} = \frac{C_a T_o}{\eta_c} \left[\left(\frac{P_o + 1}{P_o} \right)^n - 1 \right] \quad (A-1)$$

where T_o and P_o are assumed to be nominal design outlet temperature and pressure and η_c is the compressor efficiency. For low temperature storage, all compressor stages will be provided with substantial cooling to keep the average air temperature as cool as possible, since this minimizes compressor work and reduces operation and maintenance costs. Thus, an outlet temperature of about 250°F is probably a reasonable value for a system designed for 150°F storage. For a minimum delivery pressure of 40 atm from storage, the compressor will have to deliver air at 50 atm pressure or higher. Assuming 800 psia outlet with $\eta_c = 0.87$ we then obtain

$$\frac{\Delta W}{\dot{m}_a} = \frac{0.25(710)}{0.87(3412)} \left[\left(\frac{801}{800} \right)^{0.30} - 1 \right] = 22.3 \times 10^{-6} \text{ kWh/lb-psi}$$

To increase the outlet pressure from 800 to 801 psia requires that we increase the pressure ratio of the booster compressor. We estimate the incremental cost of these compressors at 200 \$/kW based on input power, which is slightly higher than the value used by Katz. We assume 2000 hr/yr of generation, capital fixed charges of 18%/yr and yearly operation and maintenance of 5% of capital. The estimated incremental cost for compression attributable to capital fixed charges and maintenance for the compressors may then be computed as follows,

$$\begin{aligned}\Delta C_{\text{cap}} &= (22.3 \times 10^{-6})(1.05)(200)(0.18/2000) \\ &= 0.422 \times 10^{-6} \text{ \$/lb-psi}\end{aligned}$$

We assume that the energy costs for compression power are charged at 0.025 \$/kWh. The incremental cost of compression attributable to energy may then be given by

$$\Delta C_{\text{en}} = (22.3 \times 10^{-6})(0.025) = 0.558 \times 10^{-6} \text{ \$/lb-psi}$$

The total incremental cost for compression is then the sum of costs attributable to capital and energy, $(0.422 \times 10^{-6}) + (0.558 \times 10^{-6}) = 0.98 \times 10^{-6}$ \$/lb-psi assuming storage at 150°F and a compressor delivery pressure of 50 atm. Computations are summarized in Table A-1.

For storage at 625°F, we assume that the same booster compressors are used with cooling jackets to achieve an outlet temperature of about 250°F, just as was assumed for 150°F storage. The high air injection temperature is achieved by use of a recuperator between the axial compressor and the boosters as illustrated in Figure 4-3. The additional compressor power per psi of frictional loss is then about the same as for 150°F storage. Similarly, the cost of the additional capacity may be taken as 200 \$/kW since the same booster compressors may be used in both systems. Changes required in the recuperator will be neglected. The amount and cost of compression energy required per psi

TABLE A-1. Computation of Incremental
Costs of Compression

<u>Storage Concept</u>	
Storage Temperature, °F	150
Compressor Outlet Pressure, psia	735
Compressor Outlet Temperature, °F	250
Additional Compressor Power, kW/lb-psi x 10 ⁻⁶	22.3
Costs for Compressor Capacity, \$/kW	200
Hours of Generation, hr/yr	2000
Fixed Charge Rate, yr ⁻¹	0.18
Operation and Maintenance, % of capital	0.05
Power Costs, \$/kWh	0.025
Capital Charges, \$/lb-psi x 10 ⁻⁶	0.422
Power Costs, \$/lb-psi x 10 ⁻⁶	<u>0.558</u>
Total Incremental Costs of Compression, \$/lb-psi x 10 ⁻⁶	0.98

of frictional loss will be about the same in both systems. Thus, although the investment cost for both equipment and the consumption of energy per lb of air are higher for high temperature storage, the incremental costs of compression should be about the same in both cases. This conclusion is valid for the compressor design illustrated in Figure 4-3, but it may not be valid for other compressor configurations.

A.2 OPTIMUM PIPELINE DIAMETER

We now begin the system optimization by computing the optimum pipeline diameter. We assume that two pipelines are needed for 150°F storage, as was determined optimal for the similar design considered by Katz. We assume that the 625°F system also requires two pipelines to keep the two designs as similar as possible.

The pressure drop in the pipelines, assuming uniform flow, may be given by

$$P_w^2 - P_t^2 = 7.708 \times 10^7 \left(\frac{Q_p^2 T L f}{D_p^5} \right) \quad (A-2)$$

where P_w = well head flowing pressure (psia)
 P_t = pipeline delivery pressure (psia)
 Q_p = pipeline flow rate (MM scf/hr)
 T = air temperature (°F)
 L = pipe length (miles)
 D_p = pipe diameter (inches)
 f_p = Darcy friction factor in pipe.

The friction factor is computed from the result of Nikuradse⁽³¹⁾ for fully turbulent flow

$$\frac{1}{\sqrt{f_p}} = 2 \log \frac{D_p}{e_p} + 1.14 \quad (A-3)$$

where roughness $e_p = 0.0006$ in. is used for the pipe. To check this result we compute the Reynolds number defined by

$$Re_p = \frac{\rho v_p D_p}{\mu} = 4.83 \times 10^5 \frac{Q_p}{\mu D_p} \quad (A-4)$$

where μ is the air viscosity in centipoise.

For two 36-in. ID pipes, 1.25-miles long at an air temperature of 150°F and a total delivery rate of 50 MM scf/hr, we obtain

$$f_p = \left[2 \log \left(\frac{36}{0.0006} \right) + 1.14 \right]^{-2} = 0.0087$$

$$Re_p = 4.83 \times 10^5 \left(\frac{50}{(0.021)(36)} \right) = 3.19 \times 10^7$$

Thus, Re_p is sufficiently large to justify use of Equation A-3. For $P_t = 40 \text{ atm} = 588 \text{ psia}$, the well head pressure is given by

$$P_w = \left\{ 588^2 + 7.708 \times 10^7 \left[\frac{(50)^2 (610) (1.25) (0.0087)}{36} \right] \right\}^{1/2} = 605.7 \text{ psia}$$

The pressure drop in the pipeline on discharge is thus $605.7 - 588 = 17.7 \text{ psia}$. We estimate the loss during compression (the flow rate in compression is here assumed to be 80% of the discharge rate) at $\Delta P_2 = 0.64 \Delta P_1 = 11.3 \text{ psia}$. The total pressure drop in the pipe per cycle is then $17.7 + 11.3 = 29.0 \text{ psia}$. The incremental cost of compression for 36-in. pipe is then computed as follows,

$$\begin{aligned} \Delta C_p &= (0.98 \times 10^{-6} \text{ \$/lb-psi})(7.64 \times 10^6 \text{ lb/hr})(29 \text{ psi}) \\ &= 217.1 \text{ \$/hr} \end{aligned}$$

The cost for 2.5 mi of pipeline is taken from the 1975 data given by Katz, escalated by 20% to allow for inflation. The capital charges for 36-in. pipe assuming fixed charges at 0.18/yr and generation of 2000 hr/yr are then

$$C_c = (\$1.26 \times 10^6) \frac{0.18 \text{ yr}^{-1}}{2000 \text{ hr/yr}} = 113.4 \text{ \$/hr}$$

The sum of the incremental costs of compression and capital fixed charges for the pipeline is then $217.1 + 113.4 = 330.5 \text{ \$/hr}$. The costs for the other diameters are computed in a similar manner and summarized in Table A-2.

The costs of pipe for 625°F storage are escalated by an additional 20% to account for insulation and expansion joints. Otherwise the costs are computed as above. The costs for 54-in. and 60-in. pipe were estimated by extrapolation of data given by Katz. The results are summarized in Table A-2.

From Table A-2 we observe that the 48-in. pipeline diameter is optimal for both systems. For 150°F storage, this result is in agreement with the result obtained by Katz, in spite of the difference in the assumed pipeline lengths. For 625°F storage the pressure loss is a little higher than at 150°F,

TABLE A-2. Optimization of the Pipeline Diameter

n_p	D_p	$Re \times 10^7$	f	P_w , psi	ΔP_1 , psi	ΔP_2 , psi	ΔP_p , psi	ΔC_p , \$/hr	ΔC_c , \$M	ΔC_c , \$/hr	ΔC_{tot} , \$/hr
$T_s = 150^\circ\text{F} (610^\circ\text{R}) \quad C_{\Delta P} = 0.98 \times 10^{-6} \text{ $/lb-psi} \quad \dot{m}_{tot} = 7.64 \times 10^6 \text{ lb/hr} \quad (L_p)_{tot} = 2.5 \text{ mi}$											
2	36	3.19	0.0087	605.7	17.7	11.3	29.0	217.1	1.26	113.4	331
2	42	2.74	0.0085	596.1	8.07	5.17	13.2	99.1	1.62	145.8	245
2	48	2.40	0.0083	592.1	4.06	2.60	6.66	49.8	2.02	181.4	231
2	54	2.13	0.0082	590.2	2.23	1.43	3.66	27.4	2.40	216.0	243
$T = 625^\circ\text{F} (1085^\circ\text{R}) \quad C_{\Delta P} = 0.98 \times 10^{-6} \text{ $/lb-psi} \quad \dot{m}_{tot} = 7.64 \times 10^6 \text{ lb/hr} \quad (L_p)_{tot} = 2.5 \text{ mi}$											
2	36	2.16	0.0087	619.2	31.2	19.9	51.1	382.6	1.51	136.1	519
2	42	1.85	0.0085	602.3	14.3	9.14	23.4	175.2	1.94	175.0	350
2	48	1.62	0.0083	595.2	7.20	4.61	11.8	88.4	2.42	217.7	306
2	54	1.44	0.0082	592.0	3.96	2.53	6.49	48.6	2.88	259.2	308
2	60	1.30	0.0081	590.3	2.31	1.48	3.79	28.4	3.38	304.2	333

but the higher cost of the pipeline makes the larger frictional losses economical. We note that because of the shorter pipeline in the present designs, the pipeline costs are a smaller proportion of the total. The pipeline optimization is thus not as critical as it might be in the design employed by Katz.

A.3 OPTIMIZATION OF THE WELL FIELD

We now proceed to the optimization of the well field. The optimum number of wells will be determined by the same approach used to optimize the pipeline diameter.

The well field optimization by Katz for 150°F storage is quite general in that it determines both the optimum number and size of the wells. For 100 MM scf/hr nominal delivery rate, Katz determined that a configuration of 100 wells, each 7 in. in diameter, is optimal. In an earlier study of CAES system economics, Bush et al.⁽²⁾ also arrived at the conclusion that 7-in. wells were optimal for the aquifer reservoirs they considered. Although these results are by no means general, the 7-in. well was assumed to be optimal for our design for both 150°F and 625°F storage. This reduces our optimization task to determining the optimum number of wells for each system.

We begin by computing the pressure drops in the wells and in the formation as a function of the number of wells for each storage temperature.

The flowing well bottom pressure (including the static or gravitational head) may be computed from the following

$$P_s^2 - e^s P_w^2 = \pm 3.842 \times 10^5 \frac{f Q_w^2 T^2}{D_w^5} (e^s - 1) \quad (A-5)$$

$$s = 0.0375 \frac{h}{T} \quad (A-6)$$

where P_s = well bottom flowing pressure (psia)

f_w = Darcy friction factor in the well

Q_w = flow rate per well (MM scf/hr)

T = temperature ($^{\circ}\text{R}$)
 D_w = well diameter (in.)
 h = well depth (ft)

For an idealized radial well, in a uniform porous media, assuming Darcy flow, the pressure drop in the reservoir may be computed from

$$P_f^2 - P_s^2 = 3.418 \times 10^7 (T\mu Q_w/kh) \ln \left| \frac{D_f}{D_w} \right| \quad (\text{A-7})$$

where P_f = formation pressure (psia)
 P_s = well bottom flowing pressure (psia)
 μ = air viscosity (cp)
 k = formation permeability (md)
 D_f = "effective" diameter of the formation zone drained by the well (ft)

Direct application of this equation to quantitative prediction of flow in an aquifer is difficult because of numerous non-uniformities and non-Darcy effects. However, it will be useful for extrapolating data for 150°F to 625°F storage conditions.

To predict the flow performance of the Belle River Mills field, we use the well back pressure/prediction data provided by Katz.⁽³¹⁾ For 150°F storage, we may use this data directly, where the results are tabulated in Table A-3.

For 625°F storage, we will attempt to adjust the back pressure data by use of Equation A-7. Taking the ratio of Equation A-7 evaluated at 625°F and 150°F we obtain

$$\frac{(P_f^2 - P_s^2)}{(P_f^2 - P_s^2)_o} = \frac{T}{T_o} \left(\frac{\mu}{\mu_o} \right) \frac{\ln(D_f/D)}{\ln(D_f/D)_o} \quad (\text{A-8})$$

where o denotes the 150°F data. We have assumed that all other variables are constant.

TABLE A-3. Computation of Pressure Losses in the Wells and in the Formation

n_w	Q_w , MM scf/m	$\dot{m}_w$, lb/hr x 10 ⁶	Re x 10 ⁶	f	P_s , psia	ΔP_w , psi	$P_r^2 - P_s^2$, psia ²	P_f , psia	ΔP_f , psi	ΔP_{in} , psi	ΔP_{out} , psi	ΔP_{tot} , psi
T = 150°F (610°R)		$P_w = 592.1$ psi	$\Delta P_{stat} = 40.5$ psi									
75	1.33	0.102	4.38	0.0123	653.0	20.4	78	710.2	57.2	77.6	49.6	127.2
100	1.00	0.0764	3.29	0.0125	644.3	11.8	51	682.7	38.4	50.2	32.1	82.3
125	0.80	0.0611	2.63	0.0126	640.2	7.62	36	667.7	27.5	35.1	22.5	57.6
150	0.667	0.0510	2.19	0.0128	637.9	5.38	27	658.7	20.8	26.2	16.8	43.0
175	0.571	0.0436	1.88	0.0129	636.5	3.98	21	652.8	16.3	20.3	13.0	33.3
T = 625°F (1085°R)		$P_w = 592.0$	$\Delta P_{stat} = 22.41$ psi									
75	1.33	0.102	2.97	0.0125	651.0	36.5	176	774.5	123.5	160.0	102.4	262.4
100	1.00	0.0764	2.23	0.0127	635.6	21.1	115	720.4	84.8	105.9	67.8	173.7
125	0.8	0.0611	1.78	0.0129	628.3	13.8	81	689.9	61.5	75.3	48.2	123.5
150	0.667	0.0510	1.49	0.0130	624.2	9.68	61	671.3	47.1	56.8	36.4	93.2
175	0.571	0.0436	1.27	0.0131	621.7	7.17	47	658.4	36.7	43.9	28.1	72.0
200	0.50	0.0382	1.13	0.0132	620.1	5.55	38	650.0	29.9	35.5	22.7	58.3

The effective field diameter D_f varies with T for several reasons. D_f increases with T because the air density is lower, thus requiring a greater volume for a given stored air mass. The required number of wells will also vary with T , so that the well spacing would change, resulting in a change in D_f . The high temperature zone will be concentrated around the well bore, thus accentuating the severe pressure gradient near the well. This will also tend to reduce D_f . It is apparent that certain of these effects may cancel each other. Therefore, we will arbitrarily neglect the effect of D_f in Equation A-8.

For the present problem we obtain

$$\frac{(P_f^2 - P_s^2)}{(P_f^2 - P_s^2)_0} \approx \left(\frac{1085}{510}\right)\left(\frac{0.031}{0.021}\right) \approx 2.26$$

where we have neglected the variation of viscosity with pressure. The data for $(P_f^2 - P_s^2)$ at 625°F are thus estimated by multiplying the values of $(P_f^2 - P_s^2)$ at 150°F by the factor 2.26.

For 75 wells (which are 7 in. in diameter) and a total flow rate of 100 MM scf/hr, the flow per well is 1.333 MM scf/hr or 0.102×10^6 lb/hr. The Reynolds number at 150°F is

$$Re = 4.83 \times 10^5 \left(\frac{1.333}{(0.021)(7)} \right) = 4.38 \times 10^6$$

The friction factor f is obtained from the Moody diagram. Assuming $e/D = 0.00015$ and $Re = 3.48 \times 10^6$ yields $f \approx 0.0123$. From Table A-3 the well head pressure is 592.1 psia for 48-in. pipe at 150°F storage. P_s is then computed from Equation A-5.

$$s = (0.375)(2150)/(710) = 0.1136$$

$$P_s = \left[e^{0.1136} (592.1)^2 + 3.842 \times 10^5 \frac{(0.0123)(1.33)^2 (610)^2}{7^5} \right. \\ \left. \times (e^{0.1136} - 1) \right]^{1/2} = 653.0 \text{ psia}$$

The pressure drop in the well is $653 - 592.1 = 60.9$ psia. This is the sum of the hydrostatic head and frictional loss. The hydrostatic head is given by

$$\Delta P_h = 653.0 \left(e^{\frac{0.1136}{2}} - 1 \right) = 40.5 \text{ psia}$$

Thus the frictional loss in the well is $60.9 - 40.5 = 20.4$ psia.

The frictional loss in the formation is estimated from the back pressure curve for the Belle River Mills field provided by Katz. At 1.33 MM scf/hr, $(P_r^2 - P_s^2) \approx 78 \times 10^3 \text{ psia}^2$ for 150°F storage. We thus obtain

$$P_f = \left[(653.0)^2 + (78 \times 10^3) \right]^{1/2} = 710.2 \text{ psia}$$

The frictional loss in the reservoir is thus $710.2 - 653.0 = 57.2$ psia. The sum of the frictional losses in the wells and in the reservoir during discharge is then $20.4 + 57.2 = 77.6$ psi. The pressure loss during compression, when the mass flow rate is only 80% of the discharge rate, is estimated at $\Delta P_{in} = 0.64 \Delta P_{out} = 49.6$ psia so that the total pressure loss per cycle is $77.6 + 49.7 = 127.2$ psia. The results of these computations are illustrated in Figure A-1 which shows the reservoir pressure distributions for charge and discharge cycles.

The incremental cost for compression, on an hourly basis may now be computed as follows,

$$\Delta C_p = (0.98 \times 10^{-6} \text{ \$/lb-psia})(7.64 \times 10^6 \text{ lb/hr})(127.2 \text{ psia}) = \$952/\text{hr}$$

The cost of the wells is estimated from the 1975 cost data provided by Katz, applying a 20% cost escalation to account for inflation. For 7-in. wells, 2150-ft long at 150°F service, the result is an estimated cost of \$100K/well. The total cost for 75 wells is thus \$7.5M. The hourly fixed charges may be computed assuming 18%/yr and 2000 hr/yr of generation,

$$\Delta C_w = 7.5 \times 10^6 (0.18)/2000 = \$675/\text{hr}$$

FIGURE A-1. Operating Pressure Conditions for the Belle River Mills CAES Design

The sum of the fixed charges for wells and the incremental costs of compression is then $952 + 675 = \$1672/\text{hr}$ for 75 wells at 150°F storage conditions. The above result for 75 wells at 150°F storage and the similar results for the other well numbers are summarized in Table A-4.

For 625°F storage, the pressure losses are computed in a manner analogous to the above and summarized in Table A-3. The hourly incremental cost of compression is computed using the same unit cost as for 150°F storage. The cost of wells at 625°F storage is much more difficult to predict. It

TABLE A-4. Computation of Optimum Number of Wells

Storage Temperature	Number of Wells	C_w , \$M	C_w , \$/hr	P_{tot} , psi	C_p , \$/hr	C_{tot} , \$/hr
150°F	75	7.5	675	127.2	952	1627
	100	10.0	900	82.3	616	1516
	125	12.5	1125	57.6	431	1556 ← (a)
	150	15.0	1350	43.0	322	1672
	175	17.5	1575	33.3	249	1824
625°F	75	15	1350	262.4	1965	3315
	100	20	1800	173.7	1301	3101
	125	25	2250	123.5	925	3175 ← (b)
	150	30	2700	93.2	699	3398
	175	35	3150	72.0	539	3689
	200	40	3600	58.3	811	4411

(a) Minimum at ~105 wells.

(b) Minimum at ~110 wells.

has been estimated that wells can be completed at these temperatures for about double the cost at ambient conditions, so we have used this figure as a guide. The hourly fixed charges and the total hourly costs are computed as above, and all results are tabulated in Table A-4.

The minimum in the sum of the hourly costs is at about 105 wells for 150°F storage. This is in excellent agreement with the value obtained by Katz. However, the agreement of these two results is expected since the basic design and economic assumptions in the present analysis are very similar to those employed by Katz.

For 625°F storage, the minimum in the sum of hourly costs occurs at about 110 wells. The optimization procedure used here is probably not sufficiently precise to predict the economic optimum with 5% accuracy. Therefore, we conclude that the economic optimum number of wells is essentially the same for both storage temperatures. Our optimization of the pipeline design in the preceding section also determined that the same pipeline diameter is

optimum for both systems. Although we have not optimized the design of the collection system, we extrapolate these results to conclude that the economic optimum design of the reservoir system for 150°F and 625°F storage are essentially the same, assuming that the wells, pipes and other hardware are suitably modified for high temperature duty.

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