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Conceptual Design and Engineering Studies of Adiabatic Compressed Air Energy Storage (CAES) with Thermal Energy Storage

M. J. Hobson et al.

November 1981

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CONCEPTUAL DESIGN AND ENGINEERING STUDIES OF ADIABATIC COMPRESSED AIR ENERGY STORAGE (CAES) WITH THERMAL ENERGY STORAGE

Columbia, MD

M. J. Hobson et al.
Acres American Incorporated

November 1981

Prepared for
the U.S. Department of Energy
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Pacific Northwest Laboratory
Richland, Washington 99352

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Foreword

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 During subsequent daytime peak load periods, the compressed air would be withdrawn from storage, heated, and expanded through turbines to generate peak power.

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

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

This report describes the results of a study subcontracted by PNL to Acres American, Incorporated to perform a conceptual design and engineering study of adiabatic CAES with thermal energy storage. Adiabatic CAES is a cycle variation which eliminates all hydrocarbon based fuel input within the CAES plant. The heat of compression is stored in thermal energy storage regenerators for later use in the CAES expansion cycle.

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

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

- CAES utilizing coal-fired fluidized bed combustors (FBC)
- CAES cycle integrated in a coal gasification plant
- Solar assisted CAES
- Combined CAES and coal-fired steam plant.

At the present time PNL is directing the limited research and development funding resources to advancement of CAES concepts utilizing thermal energy storage. The adiabatic concept appears to be the most attractive candidate for utility application in the near future. It is operationally viable, economically attractive compared with competing concepts, and will require relatively little development before the construction of a plant can be undertaken. It is estimated that a utility could start the design of a demonstration plant in 2 to 3 years if research regarding TES system design is undertaken in a timely manner.

CONCEPTUAL DESIGN AND ENGINEERING STUDY OF ADIABATIC CAES WITH THERMAL ENERGY STORAGE

EXECUTIVE SUMMARY

BACKGROUND

Compressed air energy storage (CAES) is a technology for large-scale centralized storage of off-peak electricity wherein surplus power from the utility system is utilized to compress air using motor generator driven compressors. The compressed air is stored underground in an excavated hard rock or salt cavern, or in an aquifer formation. The compressor discharge must be cooled to prevent thermal damage to the host geological formation, wasting considerable useful heat. During the generation mode, the stored air is released from the cavern, heated with fuel in combustors, and then expanded in turbines. The turbines drive the motor-generator in the generator mode, which supplies peaking and possibly intermediate electric power to the utility system.

This concept has been used in the construction of a CAES plant by Nordwestdeutsche Kraftwerke AG (a West German utility) at Huntorf, near Oldenburg, which has operated better than expected since 1978. The Huntorf equipment design has since been modified for application to the 60 Hertz systems of U.S. utilities and has been used as the basis of engineering studies for the Middle South Utilities and the Potomac Electric Power Company (PEPCO) systems.

Adiabatic CAES, the subject of this report, is a cycle variation which eliminates all hydrocarbon based fuel input within the CAES plant. Instead, the heat of compression is reused in the CAES expansion cycle through the use of thermal energy storage regenerators. This approach was first patented in 1972 and studied in some depth in an Electric Power Research Institute (EPRI) funded effort performed by the Central Electricity Generating Board (CEGB) of England. The study concluded that high temperature (1200 to 1500°F) adiabatic CAES cycles should only be considered as long term developments and recommended further study of hybrid cycles which incorporated fuel topping. Earlier studies had come to similar conclusions. However, a subsequent study performed by the MIT Lincoln Laboratory in 1979 concluded that adiabatic CAES was economically competitive in the near term with other forms of centralized electric utility storage. The results of the MIT study reflected a number of differences from previous work, most notably the effects of fuel escalation in recent years. This report covers a conceptual engineering design and feasibility study based upon use of commercially available machinery to construct an adiabatic CAES plant design, evaluate near term feasibility and identify design aspects which require further investigation.

THE ADIABATIC CAES CYCLE

Comparison of the basic adiabatic CAES concept to the oil-fired plant design reveals several changes. The fuel supply and combustion systems and the compressor intercooling system of the oil-fired design are replaced by one or more thermal energy storage (TES) systems. Allowable compressor exit temperature dictates the cycle operating pressure, and exit temperature and pressure are a function of compressor efficiency. As no fuel is used, high efficiency combustion cannot be employed to make up for compressor and piping losses as is the case with the combustion CAES cycle. Incorporation of a thermal storage device in the cycle also adds unavoidable losses to the system and would be expected to be more capital intensive than the combustion and intercooling system equivalents of the fuel-fired cycle.

Although high temperature heat storage materials are available, the compressor heat source is limited to temperatures below 900°F by the use of commercial technology. Such temperatures are substantially less than those used in the fuel-fired CAES expansion cycle, with the result that both turbine output and potential cycle efficiency are reduced. Performance of the thermal energy storage system is therefore very important, and pressure drop, temperature variation and overall cost should be minimized.

Single, double and triple compression/thermal storage intercooling stages are possible. Minimum capital cost was used in this study to select the two-stage arrangement as the design cycle.

THERMAL ENERGY STORAGE SYSTEM SELECTION

Four basic TES systems were reviewed for the adiabatic CAES application (i.e, direct and indirect contact sensible heat storage, latent heat storage, and thermochemical energy storage) and direct contact sensible heat storage in pebbles or checkers was preferred because of lower costs and greater commercial readiness. Latent heat and thermochemical energy storage systems are still in the developmental stage, while indirect sensible heat systems, which employ heat exchangers, were found to be undesirable due to relatively high capital cost and low thermal performance (i.e., effectiveness).

Extensive evaluations were carried out for direct contact sensible heat systems of the pebble or checker bed types. Both material properties and cost were used to select the most promising materials. These were crushed rock, sintered iron oxide pebbles, and a fireclay pebble commercially known as Denstone™, manufactured by the Norton Company. Selected materials were modeled to simulate performance in various pebble bed arrangements using computerized routines provided by the Massachusetts Institute of Technology and the Central Electricity Generating Board. The computer simulation activities allowed development of a TES system configuration with acceptable performance. With the final design arrangement, average temperature drop between compressor exit and turbine inlet temperatures was designed for 25°F, with pressure losses of a few psi within the storage beds.

The TES system was contained in underground hard rock caverns with the rock formations used for pressure containment. The heat storage materials were placed in silos constructed inside of excavated caverns.

TURBOMACHINERY SELECTION

As the study involved conceptual design of a plant based on commercially available machinery, subcontract arrangements were made with an industrial equipment manufacturer having product lines that encompassed all the turbomachinery requirements. The Dresser Clark Division of Dresser Industries collaborated in both the development of cycle parameters representative of available technology and identification of suitable machinery.

A combination of axial and centrifugal compressors was selected for the two-stage study cycle following analysis of several machinery arrangements. Axial machinery was selected for the first compression stage, with three machines required. Arrangements with two commercially available machines in series were not capable of achieving desired output conditions, whereas a two-into-one arrangement was found to achieve desired efficiency and exit conditions. Barrel type centrifugals were selected for the second stage of compression. Casing and rotor material modifications are required for the high-temperature machines, but these were considered within the limits of present technology and typical of orders for special applications.

The turbine section incorporates all axial machinery with a tandem low pressure turbine arrangement. Machinery selections were made on the basis of maximum efficiency. As a result, turbine section flow rate was increased over that of the compressor section by 20 percent (650 to 780 lbs/sec). This reduced clearance losses in the high pressure turbine, allowing prediction of 85 percent efficiency (isentropic) for the high pressure machine and 88 percent for each low pressure machine.

PLANT PERFORMANCE

The machinery selections and thermal storage system configurations are projected to produce a per unit power output of 200 MW, with a compressor input power requirement of 243 MW. Plant operation requires 12 hours of charging to provide 10 hours of operation in the discharge mode. Including losses for a normal daily cycle, each kilowatt hour of output requires approximately 1.48 kilowatt hours of input energy. Ten hours of generation storage capacity was selected for this study to permit comparison of the adiabatic plant and PEPCO oil-fired CAES plant (Acres, 1979). Since many utilities do not have 12 hours of charging time available to provide the design storage capacity, this design criterion would be modified on a site-specific basis.

Part load operation capability is projected as 63 to 104 percent flow for the compressor train (74 to 105 percent power input) and 0 to 100 percent output from the turbine section (20 to 100 percent of design flow). Seasonal variation in plant output is unlikely to be significant, as inlet air to the compressors is preheated using waste heat.

PLANT LAYOUT

A plant arrangement was developed based upon the layout developed for the PEPCO preliminary engineering fuel-fired CAES design study. Drawings are included in Section 8 of this report. The most noticeable differences between this arrangement and the PEPCO design are the absence of fuel storage tanks and large air-cooled exchangers required for the conventional design. The turbomachinery hall arrangement differs from the PEPCO design in that the centrifugal compressors are driven by a separate motor. As the centrifugal train occupies an area used to enclose compressor intercoolers in the fired design, no significant building changes were required.

The arrangement of the underground facilities included placement of the TES facilities at minimum depth. This minimizes the length of hot vertical piping and, consequently, pipe thermal expansion. An air shaft pressure vessel cap was provided to allow pipe growth while maintaining containment.

The TES system was divided into pebble-filled steel cylinders. Cylinders are simply added or subtracted to vary storage capacity. Eleven cylinders were required for each of the low pressure and high pressure TES systems. The cylinders are free standing within the pressurized TES caverns.

CAPITAL AND OPERATING COSTS AND CONSTRUCTION SCHEDULE

Capital cost estimates were prepared based upon preliminary estimates prepared for the PEPCO study. Direct cost for a four unit 800 MW plant (without contingencies) is estimated to be approximately $449 million (July 1980 dollars). In comparison, the equivalent cost of the PEPCO study design was approximately $380 million. It should be noted that the adiabatic CAES estimates include the cost of one complete set of spare machine rotors and overhaul parts ($14,000,000), an item not included in PEPCO CAES study estimates.

Operating costs were developed for plant staff (an estimated 68 persons for a fully manned station), turbomachinery maintenance and general plant maintenance. These costs were estimated at $4.00/kW-yr for fixed items and 0.28 mills/kWh for variable items.

A preliminary construction schedule for the adiabatic plant was also prepared. The construction of the thermal storage system and deeper caverns was projected to add approximately one year to the period required for construction of a fuel-fired plant, for a total of 6 years. As no fuel would be consumed air emissions may be negligible, and the licensing process would probably be simplified in comparison to fuel-fired CAES.

SYSTEM ECONOMICS

An economic comparison was performed between adiabatic CAES, the underground pumped hydro (UPH) and oil-fired CAES designs of the PEPCO study, and combustion turbines. The evaluation was performed on the basis of levelized energy (production) cost analysis, and was based upon fuel costs and escalation rates supplied by Battelle-PNL. The results suggested a significant economic advantage to adiabatic CAES when compared to combustion turbines, yet only marginal competitiveness when compared to the oil-fired CAES design depending on the heat storage materials used. On the other hand, adiabatic CAES was not competitive with the UPH design when only economic considerations were taken into account. The sensitivity of these results to reasonable changes in capital cost or charging energy cost were small. Such variances only change the relative economic comparison between adiabatic and oil-fired CAES, further reinforcing the conclusion that these technologies compete with each other on an economic basis.

Adiabatic CAES cycle optimization studies, based on the thermal limits of commercially available turbomachinery and system economics, identified the two-stage cycle as the most desirable. With development of higher temperature limits for compressors a single-stage low pressure approach may also be acceptable. Elimination of the centrifugal compressors and the high pressure turbine would improve plant reliability, increase machinery efficiency, and noticeably lower the ratio of power input to power output. For compressor exit temperatures in the range of 1200-1400°F, the levelized busbar cost of a low pressure design may be roughly equivalent to that of the high pressure design when operating between 15 and 25 percent capacity factor.

If the air storage system cost could be further reduced, perhaps through conversion of an existing mine, levelized busbar costs approaching those of a comparably sized UPH plant may be feasible. The disadvantage of increased capital cost with low pressure storage would be largely avoided by the use of an existing low cost storage cavity. The economics of the low pressure storage approach would become even more favorable for utilities with charging power costs greater than the 11.4 mills/kWh base value (1980 dollars), due to the more favorable performance of the low pressure cycle. The mine conversion approach using commercially available (low temperature) axial machinery can potentially bring single-stage cycle economics below combustion turbines and within range of the two-stage design. The identification of existing cavities capable of conversion to air storage at pressures of around 250 psi could therefore be important to moving adiabatic CAES technology towards commercialization.

CONCLUSIONS

Technical Feasibility - The plant configuration developed in this conceptual design study does not appear to include any design problems which would prohibit plant construction, however, a number of design areas require more detailed investigation before detailed engineering can proceed.

A number of uncertainties exist with thermal energy storage material properties and behavior in the conditions of the adiabatic CAES application. Unknown are such items as cyclic life, and particulate generation rate and size distribution for the various storage materials. Pebble bed containment vessel wall stresses due to thermal expansion are unknown, but do not appear to be severe or uncontrollable. Prediction of the exact behavior of such large quantities of materials in single containments is somewhat uncertain as comparably sized systems that operate under similar thermal conditions have never been built. Performance prediction is, however, largely a function of how well air distribution within the bed can be predicted, and therefore depends upon construction features which ensure as-designed conditions. Part-load and partial-cycling TES system behavior were not investigated to any depth in this study. Although no significant deviations are expected, behavior under such conditions must be defined to clarify operating procedures and limitations.

Design of the high temperature air shafts received considerable attention during this study. The arrangement of the pipe to accommodate startup expansion is, of course, critical to successful containment of the stored air. The design approach selected should be adequate, and, although unusual in construction, appears well within the capability of construction technology.

The availability of suitable valves was investigated by both Acres and Dresser Clark and does not appear to be a problem at this stage. The large piping required in the low pressure plant sections will involve custom fabrication, but adequate facilities are available.

The availability of turbomachinery based upon commercially offered designs appears good. Some modification to standard product designs would be required to achieve the temperature limits proposed in this study (870°F max), but no insuperable problems are anticipated. The use of temperatures above those proposed would require, however, extensive compressor redesign or development of new designs. The design configuration developed in this study does include a high horsepower gearbox in the centrifugal compressor train, which is a state-of-the-art component that would require design evaluation. Other compressor arrangements are possible which could eliminate this gearbox.

Plant operating characteristics need further evaluation and definition, particularly in light of utility system needs. The adiabatic plant design does not appear quite as flexible as combustion turbines or fuel-fired CAES, but does appear capable of supplying peaking power in a load following mode. More detailed evaluation of TES system behavior should include definition of any restrictions on partial load operation that may exist. (At present, the only problem foreseen is possible heat buildup in the TES requiring occasional heat purging with a short cold air charge period. This could be a weekend maintenance procedure, if required at all.)

The unusually large number of turbomachinery components for the conceptual design, raising some concerns regarding system reliability, is the direct result of the study requirement to use commercially available designs and a limited effort towards optimized machinery selection. Actual machinery for an adiabatic CAES plant would likely involve fewer rotating components, with some development of new designs, regardless of whether supplied by Dresser, Sulzer/BBC or any other manufacturer. Reduction of the number of machines would benefit reliability, should benefit plant cost and physical layout, appears well within technological limits, and may be performed by the manufacturers themselves if a market is shown to exist.

The general conclusion of this conceptual design study is that no significant barriers are foreseen to technical feasibility, but that plant reliability, TES material properties, and system behavior need further study before technical feasibility is certain.

Economic Feasibility - The levelized energy (production) cost evaluation showed that adiabatic CAES is marginally competitive with oil-fired CAES. The economic evaluation concluded that the cost and durability of the heat storage materials for the TES are critical to the viability of the adiabatic concept. Expensive heat storage materials that are likely to last longer (such as white cast iron balls) place adiabatic CAES at an economic disadvantage compared to oil-fired CAES. On the other hand, relatively inexpensive heat storage materials such as iron oxide pebbles give adiabatic CAES a cost advantage over the oil-fired concept, but the longevity and reliability of these cheaper materials are more in doubt. If several replacements of TES materials are required over the life of the plant, then the economic advantage of adiabatic CAES with low cost TES materials could possibly be lost.

Consideration of factors other than those incorporated in the levelized cost analysis approach indicates that adiabatic CAES could likely only complement the fuel-fired design as an air storage alternative. The operating characteristics of the fuel-fired design (less compressing load, greater generation capability) make it more compatible with the limited availability of charging power in many utilities, but the independence from petroleum based fuels in the adiabatic design may become increasingly attractive.

RECOMMENDATIONS

Acres recommends that further development of the adiabatic CAES cycle be pursued through:

(1) an intensive testing program to determine feasibilty of low cost heat storage materials.

(2) development of pebble bed design models to permit optimal design of TES beds with regards to bed configuration, thermal and mechanical stress conditions, and thermal performance.

(3) review of the general turbomachinery approach selected for this study design by an independent turbomachinery specialist, with particular attention given to the high temperature compressors, the problem of blading erosion/separator efficiency requirements, and system reliability, by an independent turbomachinery specialist.

If the above proves materials properties are acceptable, the program should continue with:

(4) preliminary design of a demonstration plant based upon commercially available equipment in association with an interested utility, and

(5) development of more advanced high efficiency axial compressors capable of discharge temperatures in the range of 1000 to 1200°F or higher.

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1 - INTRODUCTION

1 - INTRODUCTION

1.1 GENERAL

Recent studies have shown that the compressed air energy storage (CAES) concept, which includes near- and mid-term technologies for central station electric utility applications, is technically feasible and economically viable. The present study is a part of the Second Generation CAES Program which has an overall aim to develop and assess advanced CAES technologies that have minimal or no dependence on petroleum fuels.

The study has been performed under Subcontract No. B-82284-A-E, supported by the U.S. Department of Energy (DOE) through the Pacific Northwest Laboratory (PNL) operated for DOE by Battelle Memorial Institute under Prime Contract No. DE-AC06-76RLO 1830.

1.2 OBJECTIVE OF STUDY

The objective of the study was to perform a conceptual engineering design and evaluation study to develop a design for an adiabatic CAES system using water-compensated hard rock caverns for compressed air storage.

1.3 BACKGROUND

Compressed air energy storage (CAES) is a technology for large-scale centralized storage of off-peak electricity wherein surplus power from the utility system is utilized to compress air by driving compressors via a motor-generator operating in the motor mode (Figure 1-1A). The compressed air is stored underground in an excavated hard rock or salt cavern, or in an aquifer formation. Before delivery to storage, the air is cooled to prevent thermal damage to the host geological formation. This involves rejection of considerable useful heat to the environment at the compressor coolers. During the generation mode, the stored air is released from the cavern, heated with fuel in combustors, and then expanded in turbines. The turbines drive the motor-generator in the generator mode, which supplies peaking and possibly intermediate electric power to the utility system.

This concept has been used in the construction of a CAES plant by Nordwestdeutsche Kraftwerke AG (a West German utility) at Huntorf, near Oldenburg. The construction and operation of the Huntorf CAES plant was a milestone achievement which demonstrated the commercial availability of compressed air storage. The Huntorf equipment design has since been modified for application to the 60 Hertz systems of U.S. utilities and has been used as the basis of engineering studies for the Middle South Utilities and the Potomac Electric Power Company (PEPCO) systems.

Although CAES promises dramatic reductions in the use of liquid or gas fuels to produce peaking power when compared to combustion turbines, there has been no dramatic rush to install these plants in the United States. Part of this hesitancy may be attributed to utility overcapacity in many regions and a traditionally cautious approach to new technology. Possible other factors may be concerns regarding price and availability of gas or oil throughout the lifetime of a CAES plant, the U.S. natural gas shortages of the mid-seventies, the Fuel Use Act of 1978, deregulation of oil and gas, and the present instability of the Middle East. Concerns regarding petroleum fuel supplies have prompted the study of numerous variations which involve substitution of indigenous fuels into the basic CAES cycle. These variations have included renewable energy concepts as well as several fluidized bed and coal gasification CAES concepts.

Adiabatic CAES, the subject of this report, is a different approach which eliminates all hydrocarbon based fuel input within the CAES plant. Instead, the heat of compression is reused in the CAES expansion cycle through the use of thermal energy storage regenerators (Figure 1-1B). This approach was first patented in 1972 (Koutz) and studied in some depth in an Electric Power Research Institute (EPRI) funded effort performed by the Central Electricity Generating Board (CEGB) of England (Glendenning, 1979). The study concluded that high temperature (1200 to 1500°F) adiabatic CAES cycles should only be considered as long term developments and recommended further study of hybrid cycles which incorporated fuel topping. Earlier studies performed by other researchers, including Acres, had come to similar conclusions. However, a subsequent study performed by the MIT Lincoln Laboratory in 1979 concluded that adiabatic CAES was economically competitive in the near term with other forms of centralized electric utility storage. The results of the MIT study reflected a number of differences from previous work, most notably the effects of fuel escalation in recent years.

1.4 PROJECT SCOPE

The scope of this study included the development of an adiabatic CAES system requiring no addition of fuel for firing into or heating of turbine fluids. The conceptual plant design was to feature underground containment for thermal energy storage and water-compensated hard rock caverns for high pressure air storage. Other design constraints included the selection of turbomachinery designs that would require little development and would therefore be available for near-term plant construction and demonstration. The design was to be based upon the DOE/EPRI/PEPCO funded 231 MW/unit conventional CAES plant design prepared by Acres for a site in Maryland.

The required work effort for this Adiabatic CAES study was divided into six tasks as follows:

- Task I - Review Thermal Energy Storage Technology
- Task II - Review Turbomachinery System Technology
- Task III - Cycle Arrangement for Conceptual Design
- Task IV - Conceptual Design
- Task V - Evaluation of Costs and Plant Economics
- Task VI - Final Report

1.5 PROJECT TEAM

The Central Electric Generating Board (CEGB) of the United Kingdom and the Massachusetts Institute of Technology (MIT)/Energy Laboratory provided assistance to Acres in selection, evaluation, and design of the thermal energy storage system. NASA/Lewis Research Center provided peer review early in the project.

The Dresser Clark Division of Dresser Industries assisted Acres in the selection of a turbomachinery system and the development of the adiabatic CAES cycle.

1.6 ORGANIZATION OF REPORT

This report is organized into ten sections, with a breakdown by major topics as follows:

Section 2 summarizes the project, its findings, and the recommendations of the study team.

Section 3 presents the general study assumptions.

Section 4 presents the development and optimization of the plant heat cycle.

Section 5 presents the selection and thermal design of the thermal energy storage system.

Section 6 discusses the selection of turbomachinery.

Section 7 discusses estimated plant performance and operational capability and describes the control system concept.

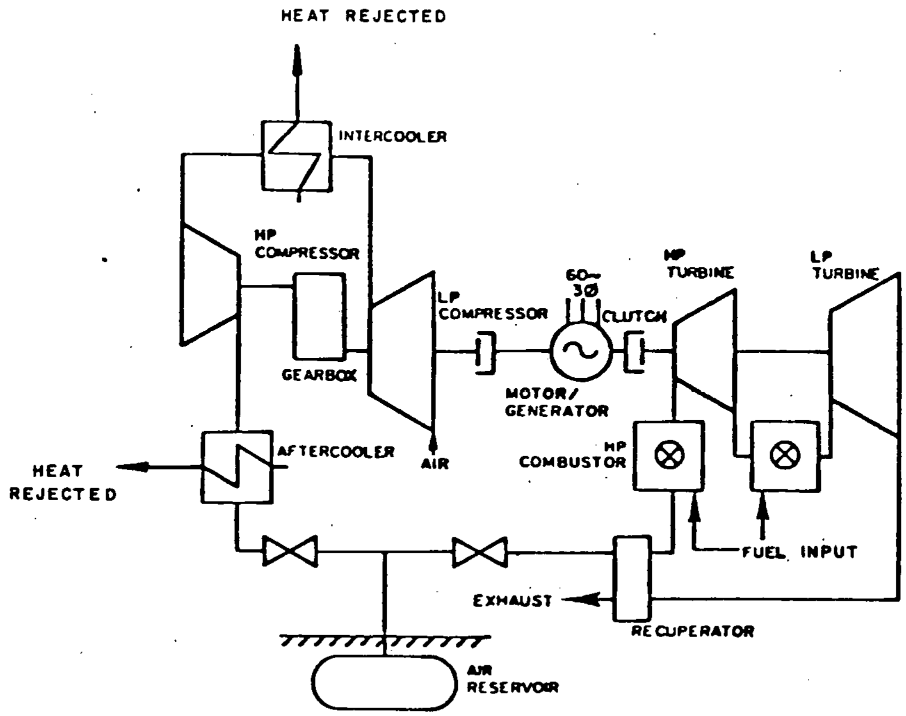
Section 8 presents the conceptual design of the adiabatic CAES plant.

Section 9 presents the cost estimates and the economic evaluation results.

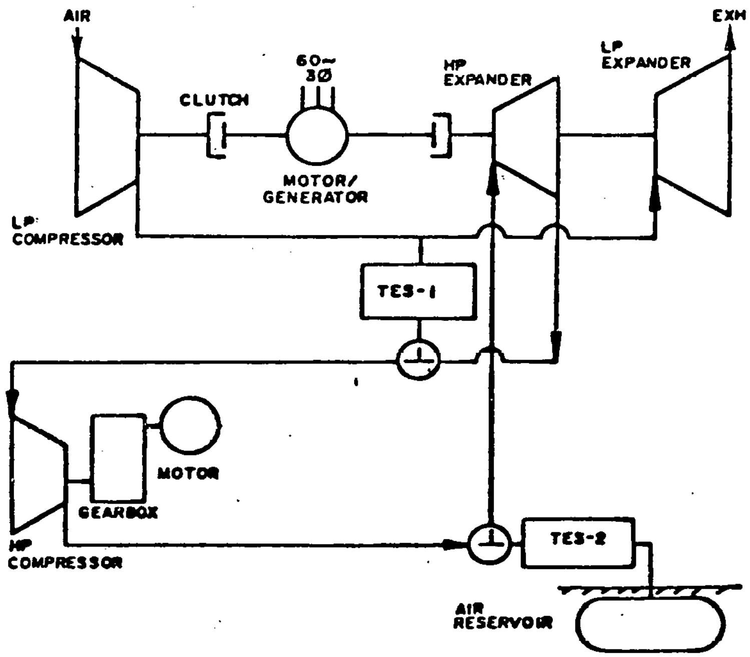
Section 10 presents an assessment of technical and economic feasibility, and contains a discussion of particular areas in the plant design requiring further development or investigation.

As noted in Section 1.6, the project design is based upon construction at a specific site to meet the needs of the Potomac Electric Power Company (PEPCO) system. PEPCO has not, however, directly participated in this study and no endorsement regarding the results or conclusions has been expressed or implied by PEPCO.

Design of the surface plant would be very similar for other utility systems, and any changes required would most likely be in design of the switchyard. This report does not contain detailed dicussions of the site, the switchyard design, or the general plant facilities. For detailed information regarding these features, the reader is referred to the fourteen volumes covering CAES plant design aspects prepared for the PEPCO oil-fired CAES preliminary engineering study (Acres, 1980). These reports are available from the Electric Power Research Institute (reference project number RP 1081-1).



A. CONVENTIONAL CAES



B. ADIABATIC CAES

ACRES	BATTELLE PACIFIC NORTHWEST LABORATORY CONCEPTUAL DESIGN AND ENGINEERING STUDIES—ADIABATIC CAES
MJ Hobson ACRES AMERICAN INCORPORATED	FIGURE 1-1

CAES CONFIGURATIONS

2 - CONCLUSIONS AND RECOMMENDATIONS

2 - CONCLUSIONS AND RECOMMENDATIONS

2.1 TECHNICAL FEASIBILITY

The plant configuration developed in this conceptual design study does not appear to include any design problems which would prohibit plant construction. However, a number of design areas require more detailed investigation before detailed engineering can proceed.

A number of uncertainties exist with thermal energy storage material properties and behavior in the conditions of the adiabatic CAES application. Unknown are such items as cyclic life and particulate generation rate and size distribution for the various storage materials. Pebble bed containment vessel wall stresses due to thermal expansion are unknown, but do not appear to be severe or uncontrollable. Prediction of the exact behavior of such large quantities of materials in single containments is somewhat uncertain as comparably sized systems that operate under similar thermal conditions have never been built. Performance prediction is, however, largely a function of how well air distribution within the bed can be predicted, and therefore depends upon construction features which ensure as-designed conditions. Part-load and partial-cycling TES system behavior were not investigated to any depth in this study. Although no significant deviations are expected, behavior under such conditions must be defined to clarify operating procedures and limitations.

Design of the high temperature air shafts received considerable attention during this study. The arrangement of the pipe to accommodate startup expansion is, of course, critical to successful containment of the stored air. The design approach selected should be adequate and, although unusual in construction, appears well within the capability of construction technology. Other approaches may also be feasible and more economic.

The availability of suitable valves was investigated by both Acres and Dresser Clark and does not appear to be a problem at this stage. The large piping required in the low pressure plant sections will involve custom fabrication, but adequate facilities are available.

The availability of turbomachinery based upon commercially offered designs appears good. Some modification to standard product designs would be required to achieve the temperature limits proposed in this study (870°F max), but no insuperable problems are anticipated. The use of temperatures above those proposed would require, however, extensive compressor redesign or development of new designs. The design configuration developed in this study does include a high horsepower gearbox in the centrifugal compressor train, which is a state-of-the-art component that would require design evaluation. Other compressor arrangements are possible which could eliminate this gearbox.

Materials selection, particularly for valve trim and TES containment walls, will require more in-depth review than was possible in this study. Selections for these two items will be affected by TES material behavior. Although the TES containment was priced on the basis of carbon steel plate, selection of a 300 series stainless would increase costs but would not significantly alter the economic analysis results.

Plant operating characteristics need further evaluation and definition particularly in light of utility system needs. The adiabatic plant design may not be quite as flexible as combustion turbines or fuel-fired CAES, but does appear capable of supplying peaking power in a load following mode. More detailed evaluation of TES system behavior should include definition of any restrictions on partial load operation that may exist. (At present, the only problem foreseen is possible heat buildup in the TES requiring occasional heat purging with a short cold air charge period. This could be a weekend maintenance procedure, if required at all.) Plant startup may be more rapid than the oil-fired design and potentially less subject to start delays because of the lack of a combustion system. These aspects also require further study.

The unusually large number of turbomachinery components for the conceptual design, raising some concerns regarding system reliability, is the direct result of the study requirement to use commercially available designs and a limited effort towards optimized machinery selection. Actual machinery for an adiabatic CAES plant would likely involve fewer rotating components, with some development of new designs, regardless of whether supplied by Dresser, Sulzer/BBC or any other manufacturer. Reduction of the number of machines would benefit reliability, should benefit plant cost and physical layout, appears well within technological limits, and may be performed by the manufacturers themselves if a market is shown to exist.

Although not directly addressed in this study, there appears to be a potential for shorter overall construction schedule than oil- fired CAES because no fuel is required and emissions are minimal. These benefits should simplify and perhaps shorten the licensing process considerably.

The general conclusion of this conceptual design study is that no significant barriers are foreseen to technical feasibility, but that plant reliability, TES material properties, and system behavior need further study before technical feasibility is certain.

2.2 ECONOMIC FEASIBILITY

The levelized energy cost evaluation showed that adiabatic CAES is marginally competitive with oil-fired CAES. The economic evaluation concluded that the cost and durability of the heat storage materials for the TES are critical to the viability of the adiabatic concept. Expensive heat storage materials that are likely to last longer (such as white cast iron balls) place adiabatic CAES at an economic disadvantage compared to oil-fired CAES. On the other hand, relatively inexpensive heat storage materials such as iron oxide pebbles give adiabatic CAES a cost advantage over the oil-fired concept, but the longevity and reliability of these cheaper materials are more in doubt. If several replacements of TES materials are required over the life of the plant, then the economic advantage of adiabatic CAES with low cost TES materials could possibly be lost.

Consideration of factors other than those incorporated in the levelized cost analysis approach indicates that adiabatic CAES could likely complement the fuel-fired design as an air storage alternative. The operating characteristics of the fuel-fired design (less compressing load, greater generation capability) make it more compatible with the limited availability of charging power in many utilities, but the independence from petroleum based fuels in the adiabatic design may become increasingly attractive.

The economic evaluation contained in Section 9 and the studies performed in the Potomac Electric Power oil-fired CAES/UPH project found oil-fired CAES and UPH to be substantially more economic than combustion turbines. This result is significant as it is common to both a simple levelized cost approach and an operational simulation within a real system. Thus, economic feasibility of both the basic oil-fired CAES and the UPH designs would appear to be fairly certain.

The analysis in Section 9 also shows that the conceptual design of adiabatic CAES as presented in this study appears to be competitive with the oil-fired CAES design and more economic than combustion turbines on a levelized cost basis. The economic analysis further indicates, however, that the two-stage adiabatic CAES design is less economically attractive than a much larger UPH plant, primarily because of higher capital cost.

Although UPH is more economically attractive on a strictly levelized cost basis than adiabatic CAES (or oil-fired CAES), utilities may still prefer CAES plants even if it means higher power costs. The major disadvantages of the UPH plant are the much larger storage and generating capacities required for the system to be practical. The large size of UPH plants presents two major problems for a single utility: (1) insufficient charging power available to justify the large storage capacity and (2) up-front capital investments would be considerably higher, increasing economic risks for installation of the plant. CAES designs, on the other hand, have more flexibility in meeting a power utility's needs on a smaller scale.

The operating characteristics of the adiabatic CAES cycle developed in this study are such that some utilities which may be able to utilize the oil-fired CAES design may not be candidates for adiabatic CAES installations. This limitation, the ratio of charging power demand to unit output, is nearly the same as both UPH and conventional pumped hydro, and therefore is not considered a serious obstacle. The conclusion drawn is that some applications may better suit fuel-fired designs, and others will be suitable for an adiabatic CAES design.

As discussed in Section 10.1.2, specific sites may exist for application of a low pressure, single-stage adiabatic CAES design. These would use abandoned mines as the storage reservoir, with potentially substantial capital cost savings.

2.3 GENERAL OBSERVATIONS

Adiabatic CAES, first patented in 1972, offers the possiblity of a thermal storage type peaking plant which operates without any direct need for oil or gas fuels. In the period shortly after patent issue, the concept was not able to compete with either the basic oil-fired CAES concept or, very likely, with combustion turbines for applications in most U.S. utilities (based upon studies by Acres and CEGB). Oil price escalation since that period has, however, been far higher than many of the "high" predictions of the period.

As a result, adiabatic CAES (much to the surprise of even those of us who have performed this study) is now approaching the point at which economics look potentially favorable. Once this first step is achieved, and not before, new concepts generally receive far more serious technical scrutiny.

This study has been performed on a conceptual design basis and has attempted to produce a plant design which could potentially be built within the next decade. A general conclusion of this study is that no insurmountable barriers are forseen regarding technical feasibility. However, as mentioned above, there are a number of design areas which must be further investigated before design feasibility (technical and economic) is certain.

By far, the most serious adiabatic CAES design uncertainties lie with the thermal storage material. The thermal storage system is the key to the adiabatic CAES plant concept, and the only feature which makes it potentially more desirable than the fuel-fired design. Without a material which can survive the operating environment for something approaching 6,000 cycles with an acceptable material attrition rate, and which produces a complete TES cost comparable to the Denstone or rock estimates, any of the other questions regarding plant design are largely insignificant by comparison.

Turbomachinery design effort in this study was intentionally limited to consideration of technology which could be available in the near term (i.e., 1985). This approach has served to bring adiabatic CAES machinery requirements into better perspective, while highlighting major differences between adiabatic CAES needs and the design requirements for the fuel-fired CAES design. The machinery selections made for this study represent optimized selections within the combined constraint of Dresser Clark's near term commercial equipment line. Dresser engineers were among the first to recognize the control and reliability problems associated with eight large pieces of equipment operating as a single unit. Although not specifically discussed within this report, Dresser has examined some of the requirements of simpler arrangements in light of new machine development as extensions of their existing product line. The company appears enthusiastic regarding a commitment to both adiabatic and fuel-fired CAES machinery development and supply. However, development of new machinery by Dresser, Brown Boveri, or any other manufacturer will, without doubt, only occur once adequate thermal storage materials are identified and an adiabatic CAES market appears to exist.

This study has been limited to pure adiabatic CAES. Preceeding work by CEGB concluded that a "hybrid CAES" cycle incorporating the high temperature compressor discharge and thermal storage of the adiabatic concept in combination with the turbine section of the fuel-fired cycle might prove more economic. That study was based upon fuel escalation rate assumptions which proved to be significantly low long before this study began.

However, during this latest adiabatic CAES effort, a parallel conceptual design study has been performed for the CEGB recommended hybrid cycle. This cycle offers both reduced dependence on (and hence lessened vulnerability to shortages of) fuel than the full fuel-fired design, and retains the full unit output of the latter. Thermal storage material availability, however, is as key to the hybrid cycle as it is to the pure adiabatic design. If it is not available, the hybrid design is not feasible either.

2.4 RECOMMENDATIONS

Acres recommends that further development of the adiabatic CAES cycle be pursued through:

(1) an intensive testing program to determine feasibilty of low cost heat storage materials.

(2) development of pebble bed design models to permit optimal design of TES beds with regards to bed configuration, thermal and mechanical stress conditions, and thermal performance.

(3) review of the general turbomachinery approach selected for this study design by an independent turbomachinery specialist, with particular attention given to the high temperature compressors, the problem of blading erosion/separator efficiency requirements, and system reliability.

If the above proves materials properties are acceptable, the program should continue with:

(4) preliminary design of a demonstration plant based upon commercially available equipment in association with an interested utility, and

(5) development of more advanced high efficiency axial compressors capable of discharge temperatures in the range of 1000 to 1200°F or higher.

3 - STUDY ASSUMPTIONS

3 - STUDY ASSUMPTIONS

Conceptual design of the adiabatic CAES system as presented in this study was based on the following assumptions and constraints:

- 10-hour storage capacity.
- 4 unit plant of maximum generating capacity.
- Project features common with the DOE/EPRI/PEPCO conventional CAES plant design at the Maryland site.
- Compressed air storage in water-compensated, underground caverns excavated in granitic gneiss rock.
- Only pure adiabatic CAES systems to be examined.
- High pressure TES to be located underground.
- Utilization of thermal storage types with proven reliability.
- Maximum compressor train exit temperatures within the limits of existing technology.
- Plant design to incorporate machinery available in the near term and requiring little or no development.
- General plant arrangement to be adapted from the DOE/EPRI/PEPCO conventional CAES study.
- Economic analysis to be performed using cost factors and levelized production cost techniques consistent with other studies performed for and by Battelle Pacific Northwest Laboratory.

4 - CYCLE DEVELOPMENT

4 - CYCLE DEVELOPMENT

4.1 INTRODUCTION

The basic concept behind large-scale centralized energy storage is to store off-peak energy from coal-fired and nuclear base load capacity for reuse during peak demand periods and thereby:

(1) take advantage of the substantial cost difference between coal or nuclear fuel and the price of premium fuels which are normally used to produce peak energy.

(2) displace premium fuel usage in the utility system, and

(3) keep base load units constantly operating in their most efficient load range.

Both adiabatic CAES and fuel-fired CAES cycles appear capable of providing these goals. However, significant design alternatives exist for each of these cycle types.

4.1.1 Conventional CAES - The "conventional" CAES cycle (Figure 4-1A) is in fact a hybrid storage design. Off-peak energy is used to compress air for storage in underground reservoirs. Since the air storage reservoir cannot accept high temperature air, the compressor discharge is cooled to nearly ambient temperature. However, as the air cannot subsequently be expanded efficiently through the power recovery turbine fuel must be burned in the discharge mode to replace the discarded heat. Although this fuel input is minimal, the fuel requirement prevents the conventional CAES design from completely displacing the premium fuel requirements of combustion turbine peaking units and achieving independence from premium fuels. Therefore, the production of peaking power remains sensitive to both the cost and escalation of fuels (although far less sensitive than combustion turbines) as well as to the availability of these or substitute fuels in the future.

The conventional CAES compression train is designed to require minimum work input through maximum use of compressor intercooling. The largest available machines are used to maximize flow rate and reduce unit costs. These machine trains represent commercial technology and incorporate normal thermal growth clearances and materials used in standard air service applications.

Discharge pressures to storage for fuel-fired CAES designs typically fall between 600 and 1000 psia. Pressure selections significantly above or below this range do not generally appear economic, except perhaps in the case where a suitable abandoned mine is available. Such pressures are relatively common in compressor applications for the chemical industries.

The combination of maximum flow rates and 1000 psia discharge pressure do, however, constitute unusual design conditions. Satisfaction of these requirements in a single compressor train (to reduce plant complexity and cost) results in an upper flow limit of some 700 pounds per second. Several machines are required in series, as no one machine is capable of the duty.

Figure 4-2 presents the process diagram for the machinery train design developed in the PEPCO study (Acres, 1980). The compressor train is composed of one axial and two centrifugal machines, with an air flow through these units of 661 pounds per second or some 530,000 scfm.

Centrifugal machines are unsuitable for the extrememly high initial flow volume, as several machines would be required. Axials are unsuited for the high pressure discharge conditions as they are basically high volume low pressure ratio type equipment. As axials do exhibit greater efficiency than centrifugals, all axial trains have been proposed in the past. These were considered feasible but required all new designs that were more expensive than the centrifugal approach.

The intercooler/aftercooler system for this train is a major plant system. The fuel-fired cycle cooling system heat load nearly equals the power input to the machinery. The cooling system capacity is therefore large and requires extensive equipment at substantial cost.

Conventional CAES turbine section designs, as a result of the fuel use requirement, normally incorporate modified combustion turbine equipment. This technology provides high efficiency machinery based upon designs developed for operation in temperatures greater than required by the basic CAES cycle. As a result, the dependence upon premium fuel to replace discarded heat can be offset somewhat through selection of an operating temperature which maximizes turbine power available for peak use, thereby minimizing installed cost per unit of capacity.

Development of peaking gas turbine and combustor designs for coal-derived fuels and temperatures in the range of 2500 to 3000°F may be followed by incorporation of this technology in the conventional

CAES design. The use of increasingly higher temperatures will, however, shift the cycle emphasis away from the basic energy storage function.

4.1.2 Adiabatic CAES - The adiabatic CAES plant design approach utilizes a different path in the development of the basic CAES concept. The adiabatic approach converts the conventional CAES design into a true energy storage cycle.

Adiabatic CAES involves, on a very general basis, the same operational steps as occur in fuel-fired CAES. Air is compressed, cooled, stored, reheated and expanded (Figure 4-1B). There are, however, some major differences in the method of performing each of these operations, which are reflected by the significantly different operating requirements and design of the plant equipment.

The intent of the adiabatic cycle is to make maximum use of charging energy. The compression process is performed without intercooling, with compressor discharge temperature limited by the temperature capabilities of the compression equipment. The heat contained in the discharged air must be removed (as in the fuel-fired cycle) to reduce storage volume and protect the air storage reservoir. However, rather than reject this heat to the atmosphere, as in the conventional cycle, it is transferred to a high capacity thermal storage material before the air is placed in storage. This thermal storage system is later used in place of fuel to reheat stored air before expansion in the power recovery turbine. The thermal storage system therefore replaces both the aftercooler and combustor of the conventional cycle.

As in the case of fuel-fired CAES, basic system economics require the use of the largest machinery available. This requirement leads to compressor operating conditions which are more severe than those of the conventional design. The initial compression process of both designs is very similar. However, once operating temperature exceeds the upper limits of the intercooled compressor train design, a new set of machinery design constraints are involved.

Temperature increases produce larger volumetric flows, resulting in a need for larger machinery. Higher temperatures typically lead to lower allowable stresses in the blades of an axial machine or impellers of a centrifugal unit. Higher temperatures also produce greater thermal expansion in the gas path parts. The lack of intercooling leads to greater power requirements and larger machine shafts. The combined need for larger capacity machinery and larger shafts produces greater blade and impeller diameter, compounding the allowable stress problem. Thus for the high temperature machinery, adiabatic CAES compression equipment design or selection is likely to be noticeably different from conventional CAES machinery trains.

The adiabatic CAES turbine operating requirements are also significantly different from those of conventional CAES machines. As no combustion is involved, turbine operating temperatures (defined by compressor limits and thermal storage exchange effectiveness) are likely to be substantially lower than exist in fuel-fired cycle machinery. Consequently, expected unit power output for the same mass flow rate is noticeably reduced.

As flow density through the adiabatic CAES turbine is expected to be significantly greater than in a combustion turbine, gas path changes may be required if high machine efficiency is to be maintained in a modified combustion machine. Combustion products and high temperatures are absent; therefore, advanced turbine technology and costly blade materials may be unnecessary. Construction of suitable turbine machinery may therefore be possible by machinery suppliers outside the general category of gas turbine manufacturing.

The impact of thermal storage system performance could be critical. Poor thermal performance, in terms of a large loss of temperature between charge and discharge, directly affects overall cycle performance. Temperature recovery, or effectiveness, should be as high as is practical. If the system exhibited a typical industrial heat exchanger effectiveness of 0.8 (80 percent recovery of available temperature) with a charging temperature of 1000°F, turbine inlet temperature would be limited to some 820°F. Increasing effectiveness to 95 percent would increase turbine inlet temperatures, increasing power output by perhaps 4 percent and storage cycle efficiency by some 6 percent. Pressure losses through the storage exchanger have a similar effect on cycle performance.

4.1.3 General - Cycle development involved an investigation of the problems associated with use of existing machinery designs, as well as an evaluation of the impact of thermal storage system performance on overall cycle performance. The use of existing machinery designs was required by the scope of work. The intent of this approach was to identify whether such a plant design appeared feasible for near term demonstration. This would allow much more rapid commercialization compared to adiabatic CAES designs requiring extensive machinery development. Cycle development began with a comparison of variations on the basic adiabatic cycle.

4.2 BASIC CYCLE ARRANGEMENTS

The basic design parameters of the adiabatic CAES plant developed in this study were selected to allow a direct comparison with the preliminary engineering design prepared for fuel-fired CAES in the recent PEPCO/DOE/EPRI study conducted by Acres. In summary these were:

- utilize the hard rock mined water compensated cavern storage concept
- design for 10 hours of storage
- design for four units of maximum size

Initial investigation indicated that each unit would likely involve peak operating temperatures between 700 and 900°F and air flows between 600 and 800 pounds per second. Cycle development commenced based on a preliminary outlet air stream temperature limitation of 850°F and three basic cycle configurations. These are shown on Figure 4-3.

The single stage cycle involves compression to a discharge pressure corresponding to maximum allowable machine exit temperature. This may be possible with one machine for temperatures up to 850°F, but is more likely to involve two machines (axial) in series because of pressure ratio and differential thermal growth limitations. Higher temperature machines have previously been identified as long term development items (Glendenning, 1979; and Giramonti, 1979). This cycle is therefore limited to storage pressures of 200 to 230 psi depending upon machine efficency at the present time.

The two-stage cycle builds upon the single-stage approach to increase operating pressures towards the level used in conventional CAES. In both fuel-fired and adiabatic concepts, higher operating pressures allow substantial reduction of required air storage cavern volume. This produces a substantial cost saving which must be weighed against increases in equipment cost and cycle losses.

A three-stage cycle was also considered. This approach further reduces storage volume, and potentially could show relatively good overall cycle performance as a result of the intercooling and multiple reheat effects of three thermal storage systems. Full thermal use of a three stage cycle would, however, involve compression to pressures on the order of 5000 psi and would require very deep underground construction (i.e., 12,000 feet), unusual piping and valve designs, and extreme casing design measures for the turbomachinery. Plant efficiency is also likely to be relatively poor, due to compounding of mechanical and gas path losses. Therefore, this concept was excluded from further consideration.

Selection of plant design pressure in the recent PEPCO CAES study was based upon a PEPCO criterion of lowest capital cost, as levelized cost differences between pressures were negligible. As differences between single and two-stage cycle performance were not expected to be great, the minimum capital cost approach was also used in this study to select the design cycle.

A comparison of the single- and two-stage cycles involves a trade off between capital cost and plant output. The low pressure single-stage design requires an air storage system volume some five times that of the two-stage cycle, involving a substantial cost increase only lightly offset by elimination of high pressure section components. Two-stage performance with addition of the high pressure turbine and compressors to form the two-stage cycle will reflect increases in both plant output and required compressor power, and a slight reduction in cycle efficiency would be expected.

Overall, however, the two-stage cycle approach was estimated to result in substantial net savings in storage system costs, as well as a significant increase in unit output. The two-stage cycle was therefore selected for further study, with a nominal storage pressure of 1200 psia.

4.3 TURBOMACHINERY DESIGN PROBLEMS

Several problems were identified in the application of conventional machinery designs to the two-stage cycle requirements. These problems were addressed through a contractual agreement with an industrial compressor and expander manufacturer to: a) provide engineering assessment of machinery needs and limits, and b) identify representative near-term machine performance, selections and cost. Clark Division of Dresser Industries, located in Olean, New York has provided those services to this study. This section reviews the findings of joint efforts by Acres and Dresser Clark to identify and address the basic problems of using commercially available equipment to develop an adiabatic CAES machinery train.

4.3.1 Compressor Design - The combination of high exit temperatures, high pressures and high mass flow rates required by the adiabatic CAES cycle present a unique design condition for a compressor. Consequently, machines expressly designed for this type of service are not at present commercially available. Near-term technical feasibility of adiabatic CAES compressors therefore depends upon the ability to modify existing compressor designs to meet the needs of the adiabatic CAES application.

Conventional applications for axial and centrifugal compressors rarely involve discharge temperatures greater than 450 to 500°F. The thermal growth of rotors and blades (neglecting material changes) which would be expected for the adiabatic CAES compressors is nearly two and a half times the growth experienced with units built for more conventional service. Thus, generous tip and root clearances must be allowed to prevent rubbing of the rotors or blades against seals and diaphragms. This problem is accentuated by potentially uneven growth of the casing and rotor during startup of the machinery. The larger clearance allowances required result in potentially greater tip losses and consequently lower and less certain stage efficiency. This makes machine efficiency prediction more difficult. Optimistic prediction of efficiency during design could result in a compressor discharge flow rate at design conditions significantly lower than expected.

Centrifugal impeller growth over the full temperature range is also seen to be potentially troublesome in Dresser Clark's conventional machine designs in that coupling of shaft and impeller involves shrinkage fits and keys. Excessive growth combined with the power to be transferred could separate the impellers from the shaft, severely damaging the machine. Other rotor assembly techniques which avoid this problem are used by Dresser in other equipment lines. Use of such approaches is possible but would require development of compressor designs which are at present not offered by Dresser Clark.

As the adiabatic design is not intercooled, actual flow volume downstream of the first stage axial increases over the conventional CAES design for the same mass flow rate. As downstream flow volume is increased, the machine gas path and consequently casing size grows, but internal pressure remains the same. Casing temperature also increases. The combined effects produce an increase in casing forces and a reduction in allowable casing stresses. This clearly requires a review of the casing design, and could lead to use of different construction techniques and/or case materials.

The extension of operating temperatures in centrifugal machines to higher levels therefore gives rise to problems with growth of rotating parts and with allowable stresses of materials at elevated temperatures. These involve the design of interstage seals and diaphragms, selection of rotor materials, and impeller mounting methods. Axial machine designs are also subject to these problems and are affected primarily in the area of tip seal and casing design.

4.3.2 Turbine Design - The limitations on exit temperature associated with the compressor system result in relatively low inlet temperatures to the turbine section in the unfired CAES system. As turbine technology typically is capable of temperatures up to 1400 to 1500°F without blade cooling, the anticipated temperatures of 800 to 900°F were not foreseen to present any difficulties in turbine section design for high temperature. Excessively low temperatures at exit could present a problem, but this can be controlled through proper selection of expansion ratio to match operating temperatures if the thermal storage system performance can be defined.

The pressures involved in the high pressure turbine are higher than normally designed for by an industrial expander manufacturer such as Dresser. Without redesign, increased operating pressure in an existing design would produce excessive losses through shaft seals and casing joints, and excessive casing stresses.

The thermal energy storage systems, assuming pebble bed devices were used, were foreseen to be sources of abrasive dust particles. Therefore, erosion of turbine blading (as well as the high pressure centrifugal compressor impellers) was identified as a potentially significant problem associated with the mechanical design of the turbines.

4.3.3 Other Items

4.3.3.1 Gearboxes - The gearbox selection for the fuel-fired CAES plant design and the gearbox used in the Huntorf plant involved gear loads which approached the limits of gear technology. Power transmission loads required with high flow rate compressors operating at speeds greater than 3600 rpm were, therefore, also considered an item of concern in the adiabatic CAES plant design. Whether high gearbox loads would be required for the final compressor, or whether a gearbox would be required at all, was unknown at the beginning of the study.

4.3.3.2 Clutches - No problems were foreseen to exist with clutch design. The power levels and speeds involved with an adiabatic CAES unit were not considered likely to exceed the capability of commercially available equipment.

4.3.3.3 Motors and Motor-Generators - The motor/generator needs of the adiabatic CAES turbomachinery were considered to be very similar to the PEPCO study CAES plant. As no significant problems were foreseen by Brown Boveri in producing these machines for oil-fired CAES, none were anticipated with the machinery for adiabatic CAES designs.

4.3.3.4 Control Systems - Although the basic control system design was recognized as being important, detailed design of the system is beyond the scope of the present engineering development study. A problem which was of concern in the PEPCO study is the relationship of control valve response time to overspeed of the turbines with load rejection. The Adiabatic CAES turbine system does not require the use of large combustors as are found in the Brown Boveri fired CAES systems. In oil fired CAES the combustors are a significant source of energy. This must somehow be dissipated or controlled upon load rejection to prevent overspeed and unit trip.

The adiabatic CAES cycle utilizes TES systems to provide heated air to the turbines. TES systems have very large volumes which pose a severe potential for overspeed if the air supply cannot be cut off rapidly. Large rapid operating shut-off valves are therefore required at both high and low pressure turbine air inlets. Valves can, however, be located so that there is far less piping volume between the control valve and the adiabatic CAES turbine than is possible in the BBC oil-fired design.

The large flow and high pressures required for CAES have resulted in multistage multi-unit compressor trains in all of the CAES design and construction projects. These arrangements typically consist of axial machines for compression up to 150 to 250 psia, followed by centrifugal units to achieve design storage pressure. Series operation of these two machine types may cause some difficulty in matching and limit operating flexibility or part load efficiency of compressors depending on the manufacturer's design approach. For example, Sulzer axials are nominally 50 percent reaction blading designs which when combined with the centrifugals in the PEPCO study design produced a machinery train which exhibited a limited operating stability range between design condition and surge. Clark axials are 100 percent reaction designs which allow slightly greater operating range when combined with centrifugals in series. The axial compressor designs proposed for U. S. CAES plants by both Sulzer Brothers (Brown Boveri CAES design) and Dresser Clark incorporate variable stator vanes, which aid in compressor matching at off-design conditions.

Such machinery trains exhibit a characteristically steep pressure/flow curve, as this is primarily a function of the number of operating stages required and is little affected by temperature. Numerous stages would be required by any designer/manufacturer. Therefore, the problems with surge margins must be faced with all CAES compressor system designs, and the value of a given axial design approach will depend upon the responsiveness of the surge control system employed and the resonance characteristics of the piping/TES system. This problem was recognized but was not addressed in depth due to the preliminary nature of this study.

4.4 THERMAL ENERGY STORAGE PERFORMANCE

The function of thermal energy storage in the cycle is to provide both compressor discharge cooling and turbine air preheating. The use of thermal storage therefore eliminates both the large cooling system and the turbine combustors of the fuel-fired design. A consequence of using this approach is that the turbine inlet temperatures are limited by compressor discharge temperature. As power output for a given turbine flow rate (size) is generally increased with an increase in inlet temperature, maximum recovery of compressor exit temperature is desirable. Poor recovery degrades performance, as reflected by cycle Electric Energy Ratio (kWh in/kWh out; the inverse of cycle efficiency), and has a major effect on overall plant economics.

Several types of thermal energy storage systems were under consideration for the cycle design. Each exhibits different thermal performance properties. Prior to detailed cycle development, evaluation of the available thermal storage system designs indicated that a direct contact sensible heat type exchanger was most suitable for the plant design. Details regarding selection and development of this TES approach are documented in Section 5.

The input temperature to the TES is virtually constant during the charge cycle. However, with the direct contact design, the TES outlet temperature during generation varies as the TES is discharged (the air stream temperature begins to fall off towards the end of the discharge period) which produces a variable exchanger effectiveness over the cycle. Therefore, the outlet temperature of the TES during generation was taken to be a time mean average. A parameter known as End Temperature Difference (ΔT_e) gives the difference between TES input temperature during compression (compressor discharge temperature) and the time mean average TES discharge air stream temperature during generation.

Pressure drop of the process air stream in passing through the thermal storage device also has a significant effect on cycle performance. To maintain a given turbine pressure ratio, increasing pressure drop requires that compressor discharge pressure be increased. This requires greater work input, and therefore also increases electric energy ratio. Excessive pressure loss between turbine stages in a reheat thermal storage unit results in a reduced turbine pressure range. This results in reduced turbine output and, as before, increases electric energy ratio. Factors which affect pressure loss in a direct contact type exchanger are flow channel velocity (a function of the ratio of open passage area to storage material; i.e., void ratio), storage device length and the severity of directional and velocity changes at entrance and exit to the unit. Remotely located units may also show substantial piping losses.

A range of appropriate end temperature differences was established for a preliminary cycle sensitivity investigation. The ideal case of ΔT_e = 0°F was used as the lower limit to provide a comparison case for the effects of more realistic TES performance values. Based on the results of initial literature reviews and previous pebble bed TES modeling, end temperature difference was allowed to vary from 10 to 100°F depending on bed configuration and material selected.

A range of expected TES pressure drops was also established for preliminary cycle sensitivity modeling. A comparison base was again established using the ideal case where ΔP = 0 psi. Various estimates of pressure loss in the TES systems ranged from a fraction of 1 psi to about 100 psi. The upper limit of pressure drop used for preliminary modeling sensitivity analyses was 75 psi.

A more detailed discussion of the factors affecting TES performance is covered in Section 5 of this report.

4.5 PRELIMINARY ANALYSIS CRITERIA

The previously discussed compressor and turbine limitations, and TES preliminary performance criteria, formed the basis of the modeling effort. The preliminary cycle selection was based on these findings until confirmed by more refined data.

Electric energy ratio (EER) and unit output power, were used in the preliminary analysis to gauge relative cycle performance. This combination provides a sound basis for cycle performance comparisons, based upon the expected performance of the turbomachinery and thermal storage system. Turbomachinery performance information was supplied by Dresser Clark and consisted of available pressure ratios and typical efficiencies for the various compressors and expanders.

Table 4-1 summarizes the performance criteria used in the initial modeling effort.

4.6 BASIC CYCLE PERFORMANCE SENSITIVITY

Specific operating performance of the TES system was unknown at the beginning of the project. Therefore, cycle performance sensitivity to storage pressure, TES 1 temperature swing, end temperature difference (ΔT_e) and bed stream pressure loss (ΔP) was investigated.

4.6.1 TES 1 Temperature Swing and Storage Pressure - The TES input temperature was limited to approximately 850°F by compressor capabilities. Also, the average cold end temperature of TES 2 was assumed to be approximately 120°F to ensure cavern (rock) stability. As a result, only the cold end temperature of TES 1 could be varied. Thus, exit temperature to the second stage of compression affects both the TES 1 design and the overall cycle performance. Figure 4-4 illustrates the effects of intercooling (cold end temperature of TES 1) and final discharge pressure selection on cycle performance. Curve PWR/1000 and EER/1000 can be used to demonstrate the effect of varying intercooling in a 1000 psi discharge cycle.

As centrifugal compressor inlet temperature is increased, exit temperature to thermal storage increases and temperature available for reheating low pressure turbine air in the discharge cycle increases. Thus turbine power output increases (curve PWR/1000). However, higher compressor inlet temperatures increase the input work requirement in excess of the gain in turbine power output. As a result, the ratio of compressor input energy to turbine output energy (EER) increases (curve EER/1000) indicating lower cycle efficiency. A similar effect occurs for the storage pressures of 600 and 1200 psi. The 1200 psi case shows the best performance, as the upper temperature limit of 850°F requires extensive intercooling, allowing more efficient compression. Thus greater cycle efficiency would be expected with cycle pressures which permit maximum TES 1 temperature swing.

A discharge pressure of 1200 psia appeared to be the practical limit based upon commercial compressor capabilities within the specified flow range for a two stage cycle. Therefore, 1200 psia was selected as the design pressure for further analysis.

4.6.2 TES Pressure Loss and End Temperature Difference - Based upon the 1200 psia design pressure, cycle sensitivity to TES operating characteristics was investigated.

First examined were the effects of end temperature difference on cycle performance. The results of this analysis, shown in Figure 4-5, indicate that as the end temperature difference is reduced, the cycle output power increases and EER is improved. The magnitude of this effect is, however, rather small; only an 8 percent drop in output power and a 9 percent increase in EER is seen when the end temperature difference of both TES 1 and 2 increases from 10 to 100°F.

The next step was to include TES pressure drop variations with the end temperature difference variations and note the total TES effect on cycle performance. Figures 4-6 and 4-7 illustrate the expected result that as pressure drop through the TES increases, output power falls off and EER increases. It is important to note that the end temperature difference and single pass pressure drop of both TES 1 and 2 have linear effects on both output power and EER. Figures 4-6 and 4-7 show the maximum effect of both pressure drop and end temperature difference are on the order of 12.5 percent for output power and 25 percent for EER.

The TES operating parameters, as shown here, have a significantly greater effect on EER than on output power. Since the relationships are all linear, there is no optimum point on the basis of performance alone. The cost of the machinery train can be considered virtually constant for the range of pressure examined, and the variation in end temperature difference would not significantly change turbine construction cost. Therefore, both pressure drop and end temperature difference should be minimized.

4.7 CYCLE OPTIMIZATION

The compressor and turbine sections were analyzed separately to determine their respective optimum operating points. The results of this analysis are presented in the following sections.

4.7.1 Compressor System Optimization - Compressor system optimization began by investigating compressor operating conditions required to achieve the desired discharge temperature. Figure 4-8 shows the relationship between first stage compressor inlet temperature and discharge pressure to deliver 850°F, based upon a single compressor efficiency. Second-stage centrifugal compressor inlet temperature requirements for 1200 psi discharge pressures are shown on Figure 4-9, as a function of first stage discharge pressures. These curves were based upon a TES pressure loss of 10 psi.

Compressor train input power calculations were performed for a storage pressure of 1200 psi based on discharge temperatures of 850°F and air flow rates of 720 lb/sec per train. Results (see Figure 4-10) show that total compressor train input power is reduced as the first stage discharge pressure increases. This is due to the higher efficiencies of the first stage axials compared to the centrifugal machines. Thus the first stage should utilize all axial equipment to the highest pressures and temperatures possible, up to the point where use is limited by either maximum allowable temperature or pressure, or by a serious drop in machine efficiency.

The analysis also considered the use of inlet air preheating ahead of the axial units to minimize the pressure ratios required. Although power requirements increase per pound of air per unit pressure rise, overall power requirements are reduced. The use of higher inlet temperatures to the second stage compressors (centrifugals) narrows the TES #1 operating temperature swing and potentially simplifies its design. However, as mentioned in Section 4.5.1, maximum cycle performance is achieved with a maximum TES 1 temperature swing. Other factors, such as ensuring thermal equilibrium of the TES system, must be weighed to define the inlet temperatures to the centrifugal units.

4.7.2 Turbine Train Optimization - Turbine train output power was calculated for a range of expander pressure splits and TES properties. With TES input temperatures of about 850°F during compression and mass flow rates of 720 lb/sec per train, TES properties considered were single-pass pressure drop and end temperature difference. Preliminary results indicated that for any given TES property, there is a pressure split between the turbines which yields a maximum combined output power. Figure 4-11 illustrates this result for a 1200 psia storage pressure. This figure reveals that the optimum high turbine exhaust pressure remains constant for constant TES pressure drop with varying TES output temperatures. Temperature changes affect only the output power, while varying TES pressure loss affects both the output power and, more importantly, the optimum pressure split between turbines. Figure 4-11 also shows that as the TES pressure loss increases, the optimum high pressure turbine exhaust pressure also increases.

Comparison with the zero loss case (end temperature difference and the TES pressure loss) shows that about 90 percent of ideal power is available with $\Delta T_e = 50°F$ and $\Delta P = 50$ psi. This percentage decreases slightly with decreasing storage pressures.

The results indicated that the optimum h.p. turbine exhaust pressure lies in the range of 150 to 250 psia for the selected cycle, which set the basis for equipment selections.

Table 4-1

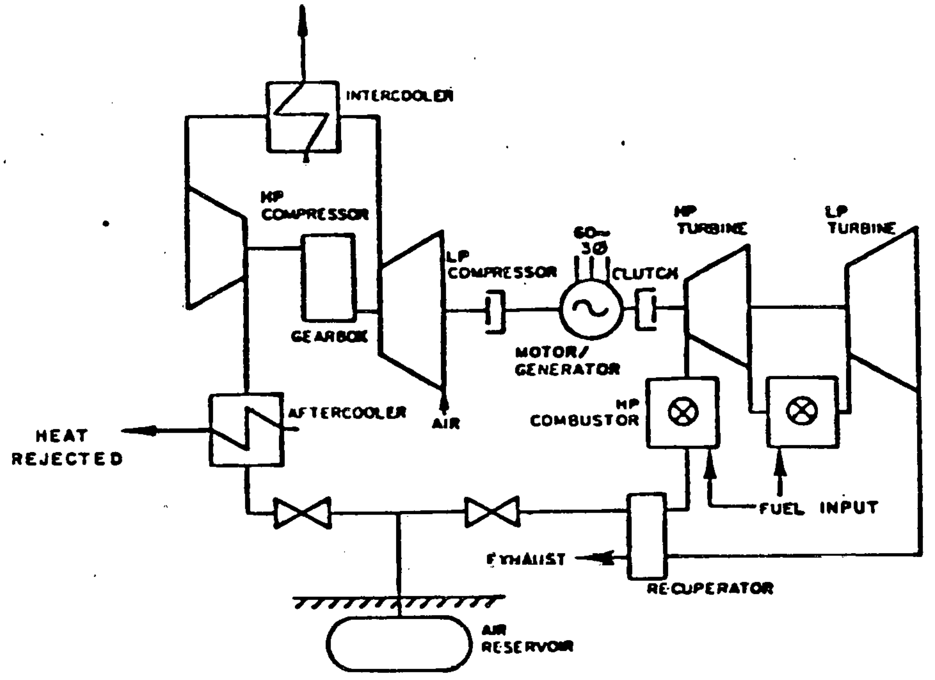
Preliminary Analysis Criteria

Turbomachinery

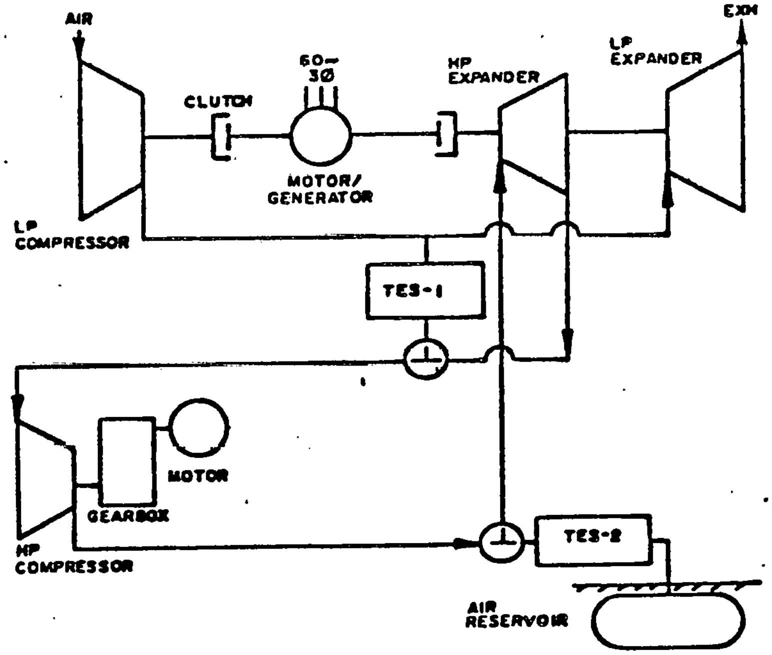
	Flow Rate (lb/sec) per train	Inlet Temperature (°F)	Outlet Temperature (°F)	Adiabatic Efficiency (%)
Compressors	600 - 800	0 - 90	850 (max.)	80 - 90
Turbines	600 - 800	1400 (max.)	---	80 - 90

Thermal Energy Storage Systems

	Flow Rate (lb/sec) per train	Maximum Inlet Temperature (°F)	Maximum Outlet Temperature (°F)	End Temperature Difference T_e (°F)	One Pass Pressure Loss (psi)
TES 1 & 2	600 - 800	850	850	0-100	0-75



A. CONVENTIONAL CAES



B. ADIABATIC CAES

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CAES CONFIGURATIONS

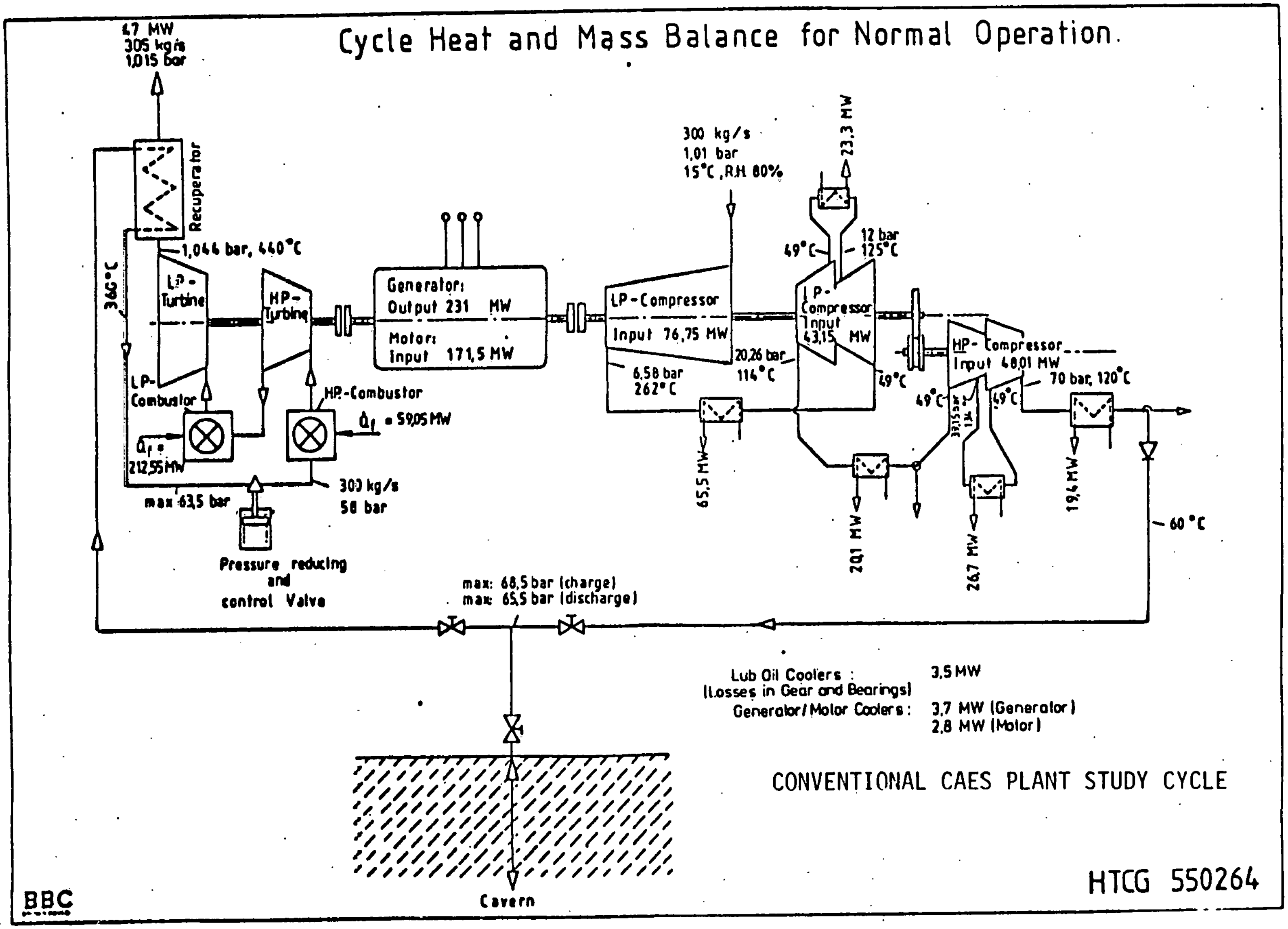
Cycle Heat and Mass Balance for Normal Operation.
47 MW
305 kg/s
1,015 bar
Recuperator
1,044 bar, 440°C
366°C
LP-Turbine
HP-Turbine
LP-Combustor
HP-Combustor
Q̇f = 212,55 MW
Q̇f = 59,05 MW
max 63,5 bar
300 kg/s
58 bar
Pressure reducing and control Valve
Generator: Output 231 MW
Motor: Input 171,5 MW
300 kg/s
1,01 bar
15°C, RH 80%
LP-Compressor
Input 76,75 MW
6,58 bar
262°C
20,26 bar
114°C
65,5 MW
23,3 MW
12 bar
125°C
49°C
IP-Compressor Input 43,15 MW
20,1 MW
HP-Compressor Input 48,01 MW
70 bar, 120°C
26,7 MW
19,4 MW
60°C
max: 68,5 bar (charge)
max: 65,5 bar (discharge)
Cavern
Lub Oil Coolers : 3,5 MW
(Losses in Gear and Bearings)
Generator/Motor Coolers : 3,7 MW (Generator)
2,8 MW (Motor)
CONVENTIONAL CAES PLANT STUDY CYCLE
HTCG 550264
BBC

Figure 4-2

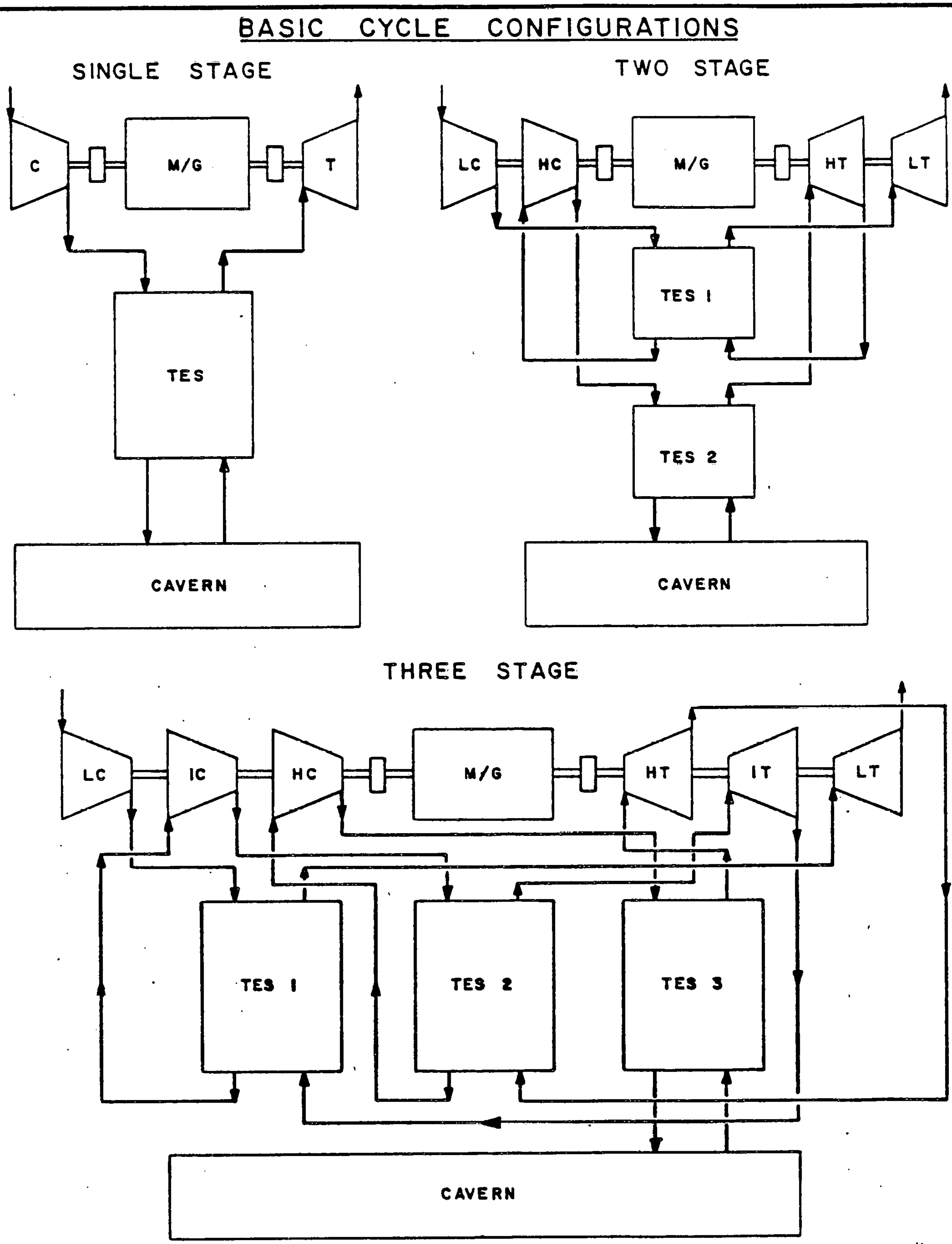

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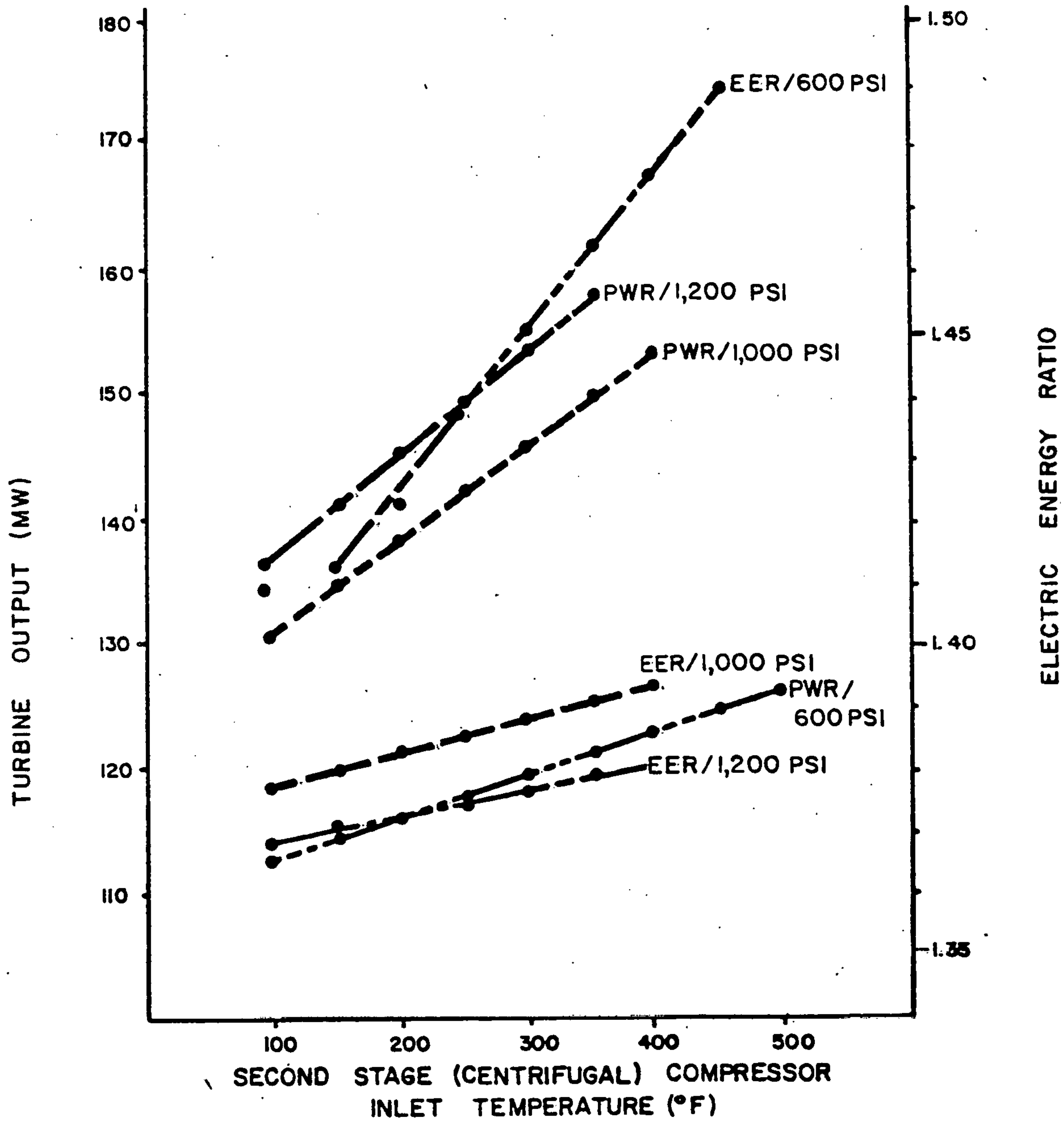
FIGURE 4-3

TWO STAGE CYCLE PERFORMANCE FOR VARIOUS PRESSURES



LEGEND

PWR - TURBINE OUTPUT
EER - ELECTRIC ENERGY RATIO
SEE ASSUMPTIONS (ATTACHED)

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Figure 4-4
Assumptions

- 1st stage Compressor
  P_r = 16:1

  isentropic = 88%

- 2nd stage Compressors
  a. 600 psi. storage
  P_r = 3.1

  b. 1000 psi. storage
  P_r = 2.17 ea. (2 units in series)

  c. 1200 psi. storage
  P_r = 2.33 ea. (2 units in series)

  isentropic = 82% for all 2nd stage units
  FLOW = 660 lb./sec. (all compressors)

- Turbines

  a. High Pressure
  outlet press. = 5 psi. above
  TES #1 charge pressure.
  inlet Temp = TES #2 charge Temp - 50°F

  b. Low Pressure exit press = 15 psi.
  inlet Temp = TES #1 charge Temp - 50°F

  isentropic = 88% for all turbines

  FLOW = 650.8 lb./sec. for all turbines
  (1% per day cavern air loss)

- Mechanical Efficiencies for all Equipment = 100
  (power transfer losses neglected)

END TEMPERATURE DIFFERENCE EFFECTS ON CYCLE PERFORMANCE

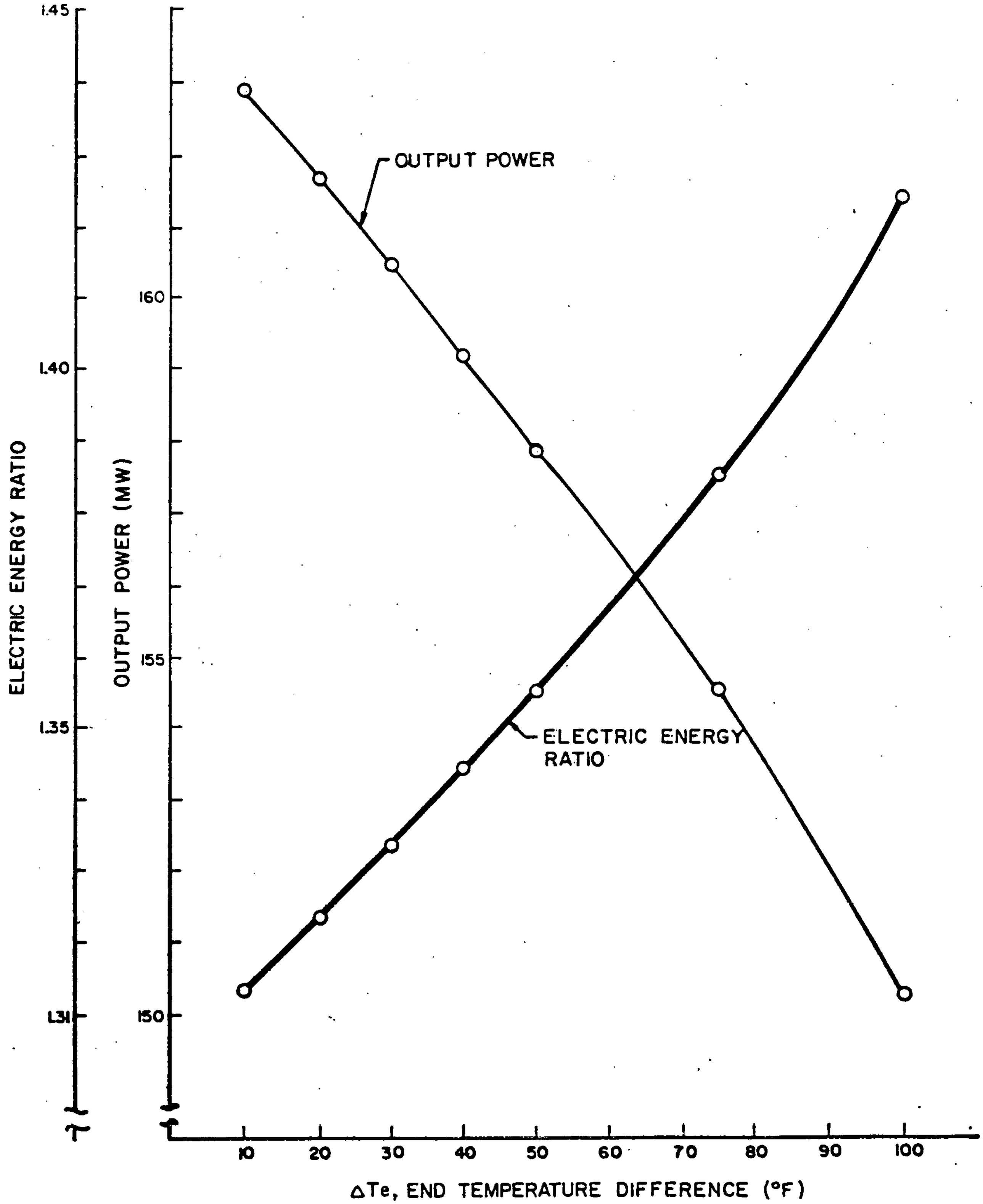

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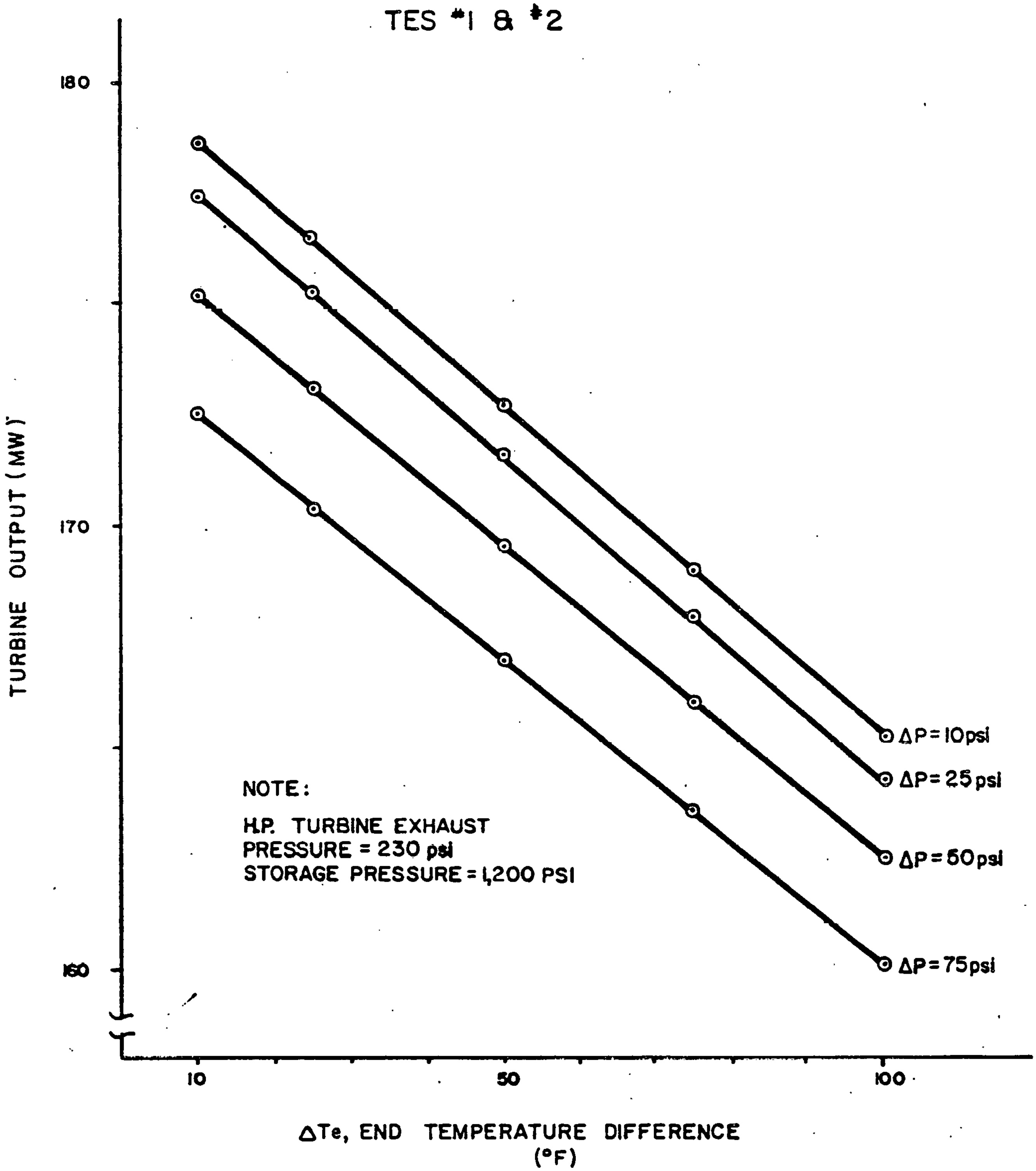
FIGURE 4-5

TURBINE OUTPUT POWER AS A FUNCTION OF BOTH PRESSURE AND TEMPERATURE LOSSES IN TES #1 & #2



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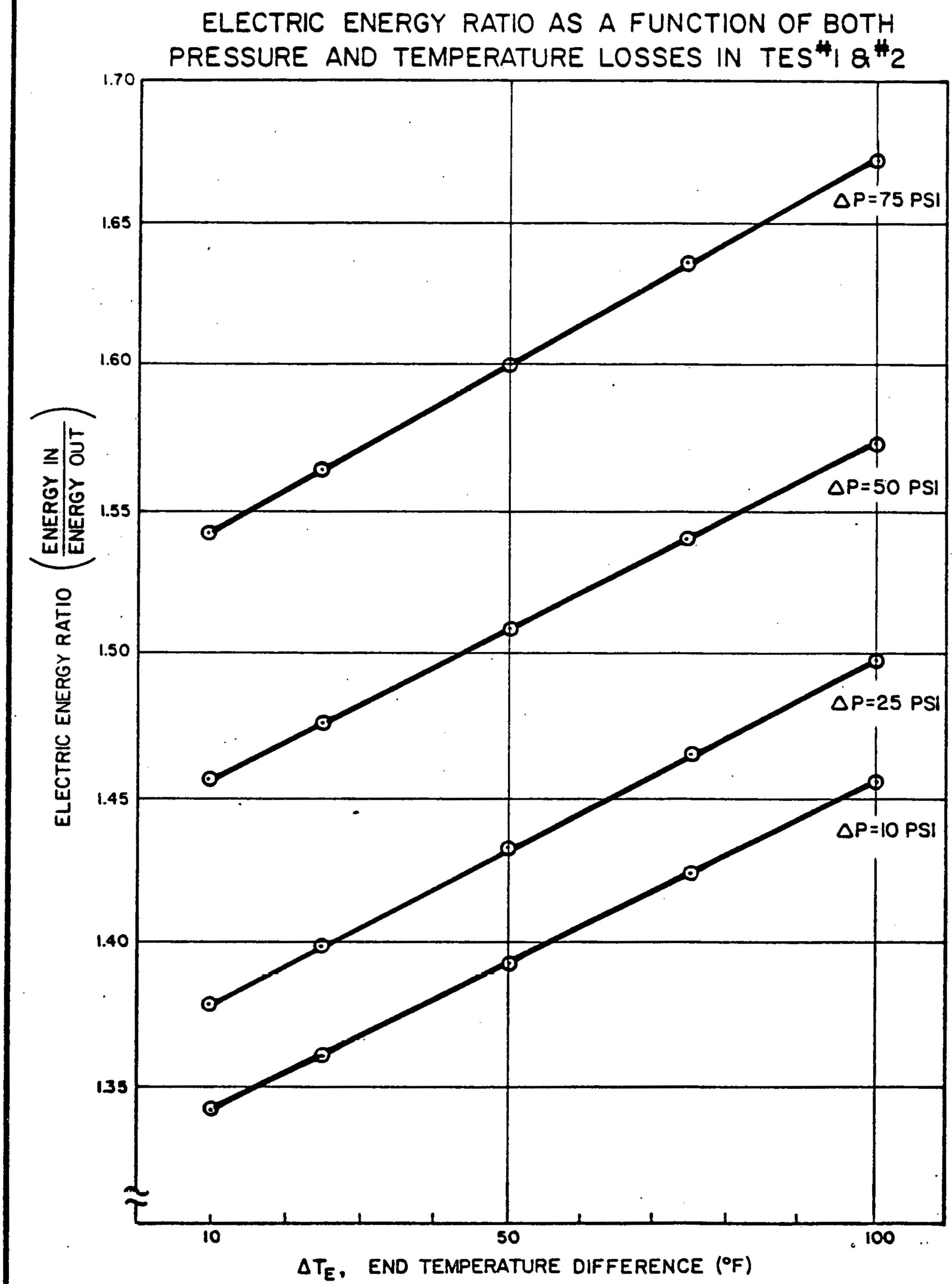

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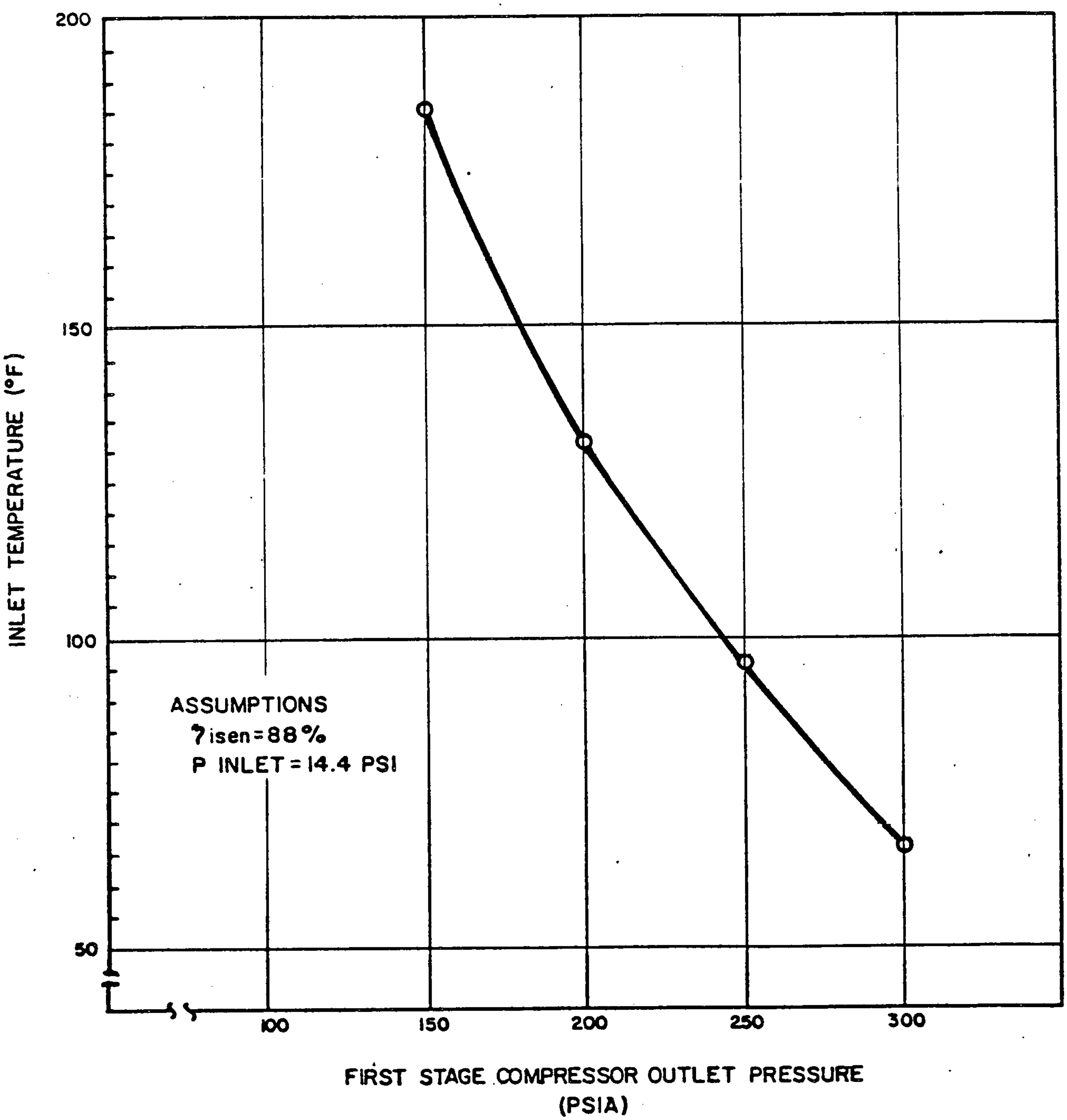
FIGURE 4-7

REQUIRED INLET TEMPERATURE TO FIRST STAGE
COMPRESSOR TO DELIVER 850°F AIR VS.
OUTLET PRESSURE OF FIRST STAGE COMPRESSOR



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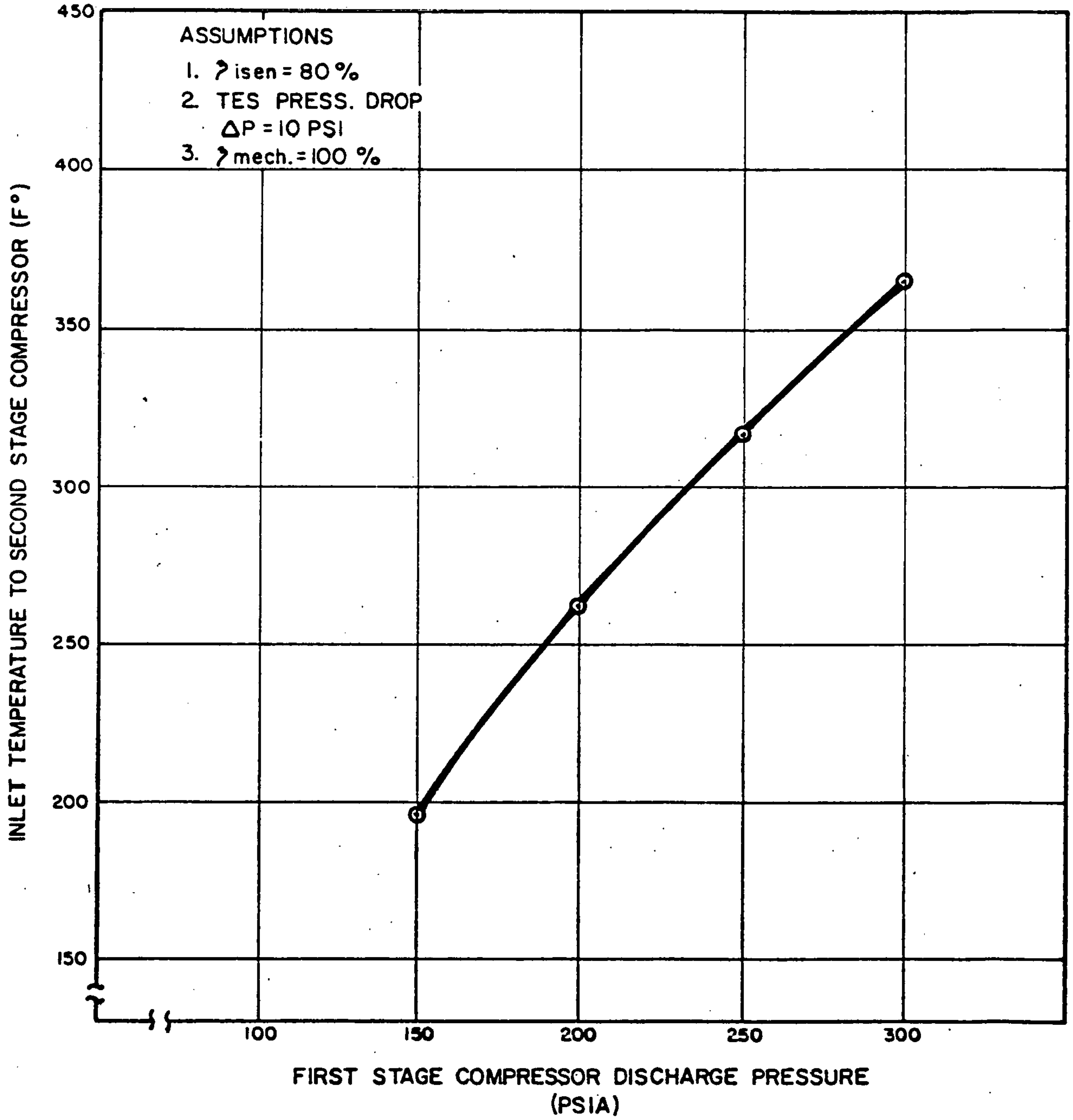
INLET TEMPERATURE (°F) REQUIREMENTS OF SECOND STAGE COMPRESSOR FOR 850° DISCHARGE TEMPERATURE



STORAGE PRESSURE = 1,200 PSIA

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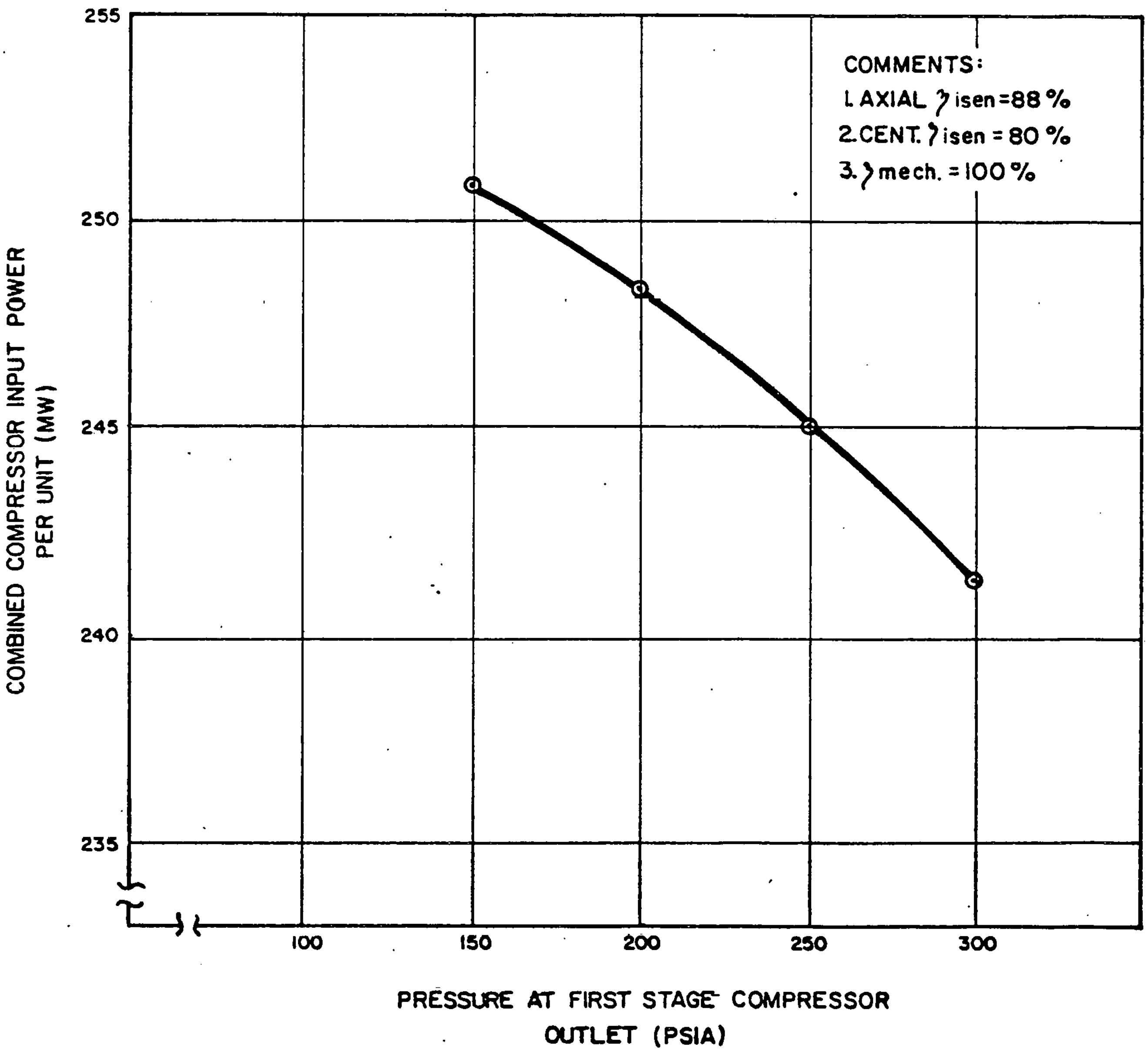
FIGURE 4-9

REQUIRED COMBINED COMPRESSOR TRAIN
INPUT POWER TO DELIVER
850° AIR TO BOTH TES #1 & #2



STORAGE PRESSURE = 1,200 PSIA
MASS FLOW = 660 LBS/SEC

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FIGURE 4-10

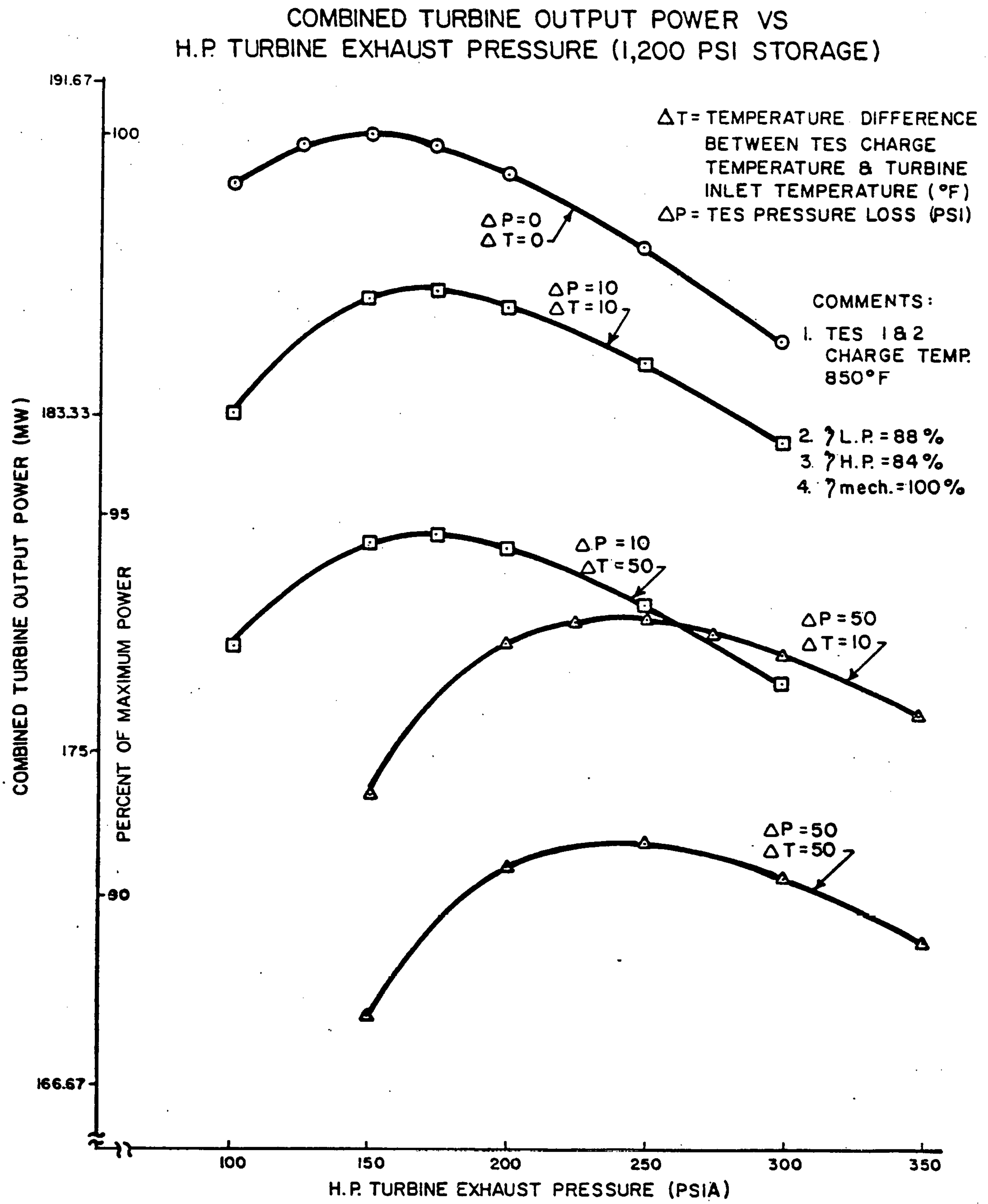

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FIGURE 4-11

5 - THERMAL ENERGY STORAGE SYSTEM

5 - THERMAL ENERGY STORAGE SYSTEM

5.1 GENERAL

The objective of adiabatic CAES is to allow efficient power generation without the oil consumption of conventional CAES. This goal is achieved by storing the heat generated during the compression mode of the CAES cycle and using it for reheating air during the power production mode. Thermal energy storage (TES) regenerators are used to facilitate this heat exchange. These devices are composed of two major components: (1) the fill material that is thermally cycled, and (2) the containment system.

This section summarizes study activities related to determination of TES sizes. These activities included review and evaluation of:

- TES technologies
- TES fill materials
- TES configurations
- TES sizes.

5.2 SYSTEM ARRANGEMENT

As described in Section 4, an adiabatic CAES cycle with two TES regenerators was selected for conceptual design. The first TES regenerator (TES 1) operates as the intercooler between the first and second stages of compression and as the reheater for air entering the low pressure turbines. The second TES regenerator (TES 2) stores the heat generated in the final compression stage for reheat of high pressure turbine inlet air. Figure 5-1 shows the simplified cycle arrangement.

5.3 OPERATING CONDITIONS

The operating conditions used for assessment of the TES are presented in Table 5-1. Equal air mass flow rates for the compressor and expander trains were assumed based on preliminary turbomachinery specifications. Subsequent developments in turbomachinery design showed that increased flow rates through the expander train were preferred to achieve desired efficiency in the high pressure turbines. Therefore air mass flow rates of 650 lb/sec per unit for compression and 780 lb/sec for expansion were adopted (see Section 6.2). As the revised flow conditions were developed too late in the study to be incorporated into the computer simulation activities, the effects of the change have been examined separately and found to be relatively insignificant. This discussion is contained in Section 5.7.6. The discussion of TES sizing as contained in this section is based largely upon analysis performed for the balanced (10-hour charge/10-hour discharge) cycle.

The final adiabatic CAES cycle showed only minor deviation from the other operating conditions given in Table 5-1. The hot air temperature to TES 1 increased to 877°F while the cold air inlet temperature dropped to 438°F. The terminal air temperatures for TES 2 were unchanged and storage pressures changed only slightly as a result of final adjustments to plant pressure drops. These changes, however, affect TES size and performance negligibly. The final temperature balance throughout the cycle was based on a constant compressor inlet temperature of 90°F.

Condensation of water on TES surfaces when storage temperatures fall below the dew point was a potential operating condition of some concern. Figure 5-2 illustrates the relationship between water dew point temperature, and the relative humidity and air temperature at compressor inlet for the operating pressures of TES 1 and TES 2. The chosen compressor inlet temperatures correspond to the yearly average at the site (55°F) and the probable maximum condition (90°F), since charging is normally carried out at night.

Figure 5-2 shows that a minimum cold air return temperature of 200°F is required to avoid condensation in TES 1, and this temperature would always be achieved under the proposed cycle. In contrast, TES 2 will experience condensation under practically all combinations of climatic conditions because of the low temperature (approximately 100°F or less) of the air returning from the cavern.

5.4 TES CONCEPTS

A review of TES concepts was performed to determine the most appropriate heat storage system for adiabatic CAES. Thermochemical, latent heat, and direct and indirect sensible heat storage systems were reviewed. The two major criteria used for the selection were (1) near-term availability, and (2) low cost.

5.4.1 Thermochemical Energy Storage - The storage of energy in chemical bonds involves a reversible process that combines an energy consuming (endothermic) or charging reaction and an energy releasing (exothermic) or discharging reaction. During energy storage, heat is provided to disassociate a chemical compound into reactants which are physically separated and stored. To recover the stored energy, the reactants are recombined in an exothermic reactor and heat is withdrawn for use. The chemical product of the exothermic reaction is then stored to await reuse by the charging process, which completes the storage cycle.

The major advantages of thermochemical systems over sensible and latent heat systems for an adiabatic CAES application are the potential for:

- higher energy storage densities,
- ambient storage temperatures (which can simplify containment), and
- low energy-related costs (such as raw material and storage tank costs).

Measured against these advantages, however, are two predominant drawbacks:

- low state of development, and
- high system complexity.

Currently, thermochemical storge systems are the least developed of candidate thermal energy storage technologies.

Mar and Bramlette present an excellent overview of the current state-of-the-art and prospects for this technology (Mar and Bramlette, 1978). They conclude that significant research and development is required in areas of chemistry, heat transfer, materials, chemical engineering, and system analysis before thermochemical storage systems will advance to the point where feasibility can be fairly assessed. For this reason, the technology was not considered a viable candidate for adiabatic CAES.

5.4.2 Latent Heat Storage - Storage of thermal energy as heat of fusion has received considerable attention in recent years. The storage method is attractive because the heat capacity associated with the phase change of many materials is often greater than the sensible heat storage capacity over a given storage temperature range.

Selection of phase change materials depends primarily on melting point and heat of fusion characteristics, but considerations of material stability, nucleation, and irreversibilities are just as important.

Major problems identified with phase change systems include:

- lack of material stability over many successive melting/freezing cycles,
- poor heat transfer across heat exchanger surfaces because of high film resistances attributed to void formations and adhesion of material to tube surfaces, and
- no proven large scale commercial applications.

Heat transfer and thermodynamic problems have persisted in current research programs such that the energy storage capability of latent heat systems is not significantly more attractive than sensible heat systems. Because of unacceptable performance characteristics and potentially high system costs, phase change systems are not planned in the immediate future for any large TES application. In general, unresolved system problems prevented further consideration of this technology for heat storage in adiabatic CAES.

5.4.3 Sensible Heat Storage - Sensible heat is the oldest and most advanced TES concept in terms of development and demonstrated feasibility. Successful sensible heat systems have employed both liquid and solid materials as the storage media. Gas, however, is an unacceptable heat storage medium because of its low density.

Of the three energy storage forms reviewed, sensible heat storage is the only TES option that is sufficiently developed for consideration in large, near-term commercial applications. Selection of the storage medium depends on its system compatibility, performance characteristics, availability, and overall cost.

Two basic configurations are possible for sensible heat storage in adiabatic CAES:

(1) Indirect system - in which the cycled compressed air transfers heat to and from the storage medium (or to and from an intermediate heat transfer fluid coupled to the storage medium) via a heat exchanger.

(2) Direct contact system - in which the cycled compressed air transfers heat directly to and from a solid medium such as a pebble bed or checker matrix.

Figure 5-3 presents some basic arrangements for sensible heat storage systems that can be used for adiabatic CAES.

In Figure 5-3A, the compressed air flows through small diameter tubes embedded in solid storage material, or submerged in liquid, transferring heat to and from the storage medium across the tube surface boundaries.

Figure 5-3B shows a configuration where a fluid is used as both the heat transport and storage medium. In this system, the fluid cycles from one storage tank to another, passing each time through a heat exchanger coupled to the compressed air duct. Heat is transferred either to or from the compressed air by the storage fluid depending on the operating mode. Design and operation of the system is simplified because each containment tank serves the singular purpose of hot or cold storage.

Figure 5-3C shows a system similar to Figure 5-3B except that containment of the storge medium is confined to one tank. Successful operation of the system depends on establishing a stable and relatively sharply defined thermal gradient between the hot and cold zones within the storage tank. This separation can be accomplished with storage materials of low thermal conductivity and by careful regulation of the fluid flow such that the thermocline region experiences minimal disturbance as the system is thermally charged or discharged. Because only one tank is required, this system is cheaper to construct than the previous system, but operation is somewhat more complex.

The single tank system can also be modified to reduce storage medium costs by displacing expensive storage fluid with solid material of lower cost and higher energy storage density ($\rho \times C$) such as iron oxide pebbles (Burolla, 1979). In this case, the working fluid functions primarily as a heat transfer medium between the TES pebbles and the cycled air.

Figure 5-3D shows a simplified, direct contact pebble bed regenerator. Direct contact systems are characterized by simplicity of operation and high effectiveness (with values above 0.90 easily achievable for pebble bed systems).

5.4.3.1 <u>Direct Contact Systems</u> - An economic comparison of direct and indirect contact sensible heat storage systems concluded that direct contact systems were preferred for adiabatic CAES. This conclusion was consistent with previous studies that examined methods of heat storage on systems with pressurized gas as the working fluid (Glendenning, 1979; Hamilton, June 1978; and Boeing, 1978). The advantages of direct contact systems over indirect systems are higher heat exchange effectiveness yielding greater return temperatures to the turbines, less system complexity, and lower capital and operating costs. The disadvantages of indirect concepts for adiabatic CAES are presented in the next section.

For this study, a direct contact sensible heat storage system was selected for conceptual design development.

5.4.3.2 Indirect Systems - Indirect systems can draw from a wide variety of heat transfer fluids for thermal storage, but a review of fluid mediums did not identify a satisfactory material that could withstand the temperature swing of TES 2. A two-stage heat storage system, however, could accommodate the temperature limitations of materials by dividing the temperature swing into high and low temperature regions. For example, salts could be used for the high temperature stage and oils for the low temperature stage.

For the operating temperatures of TES 1, several heat transfer fluids including molten metals and molten salts are possible storage media. Molten salts, however, are preferred because of their proven reliability, wide use, and greater safety. A single-stage molten salt system is feasible for TES 1 because the cold air inlet temperature to the TES is above the minimum safe operating temperature for a number of established salt mixtures.

The major attraction of indirect TES systems is the potential for reduction of containment cost with nonpressurized storage. This advantage, however, must be weighed against the disadvantages of (1) high heat exchanger and storage medium costs, and (2) reduced thermal performance attributed to the limited effectiveness (by cost) of the heat exchanger. For the basic indirect storage arrangements shown in Figure 5-3, system A can be ruled out when compared to the other systems on first cost alone for tube materials and welding, as shown by CEGB (Glendenning, 1979). The following simple performance and cost analyses show the limitations of systems B and C for adiabatic CAES.

Assuming equal thermal capacity rates through the heat exchanger for the compressed air and heat transfer fluid (which are the probable optimal flow rates), the heat exchanger area required for an indirect system can be expressed as (Kays and London, 1964; Glendenning, 1979):

$$A = \frac{2 W \varepsilon_o}{U (1 - \varepsilon_o)}$$

where A = heat exchange area

ε_o = overall effectiveness of the heat exchanger

U = overall heat transfer coefficient

W = thermal capacity rate ($M \times C_p$)

M = mass flow rate

C_p = specific heat at constant pressure

The overall (roundtrip) effectiveness of the heat exchanger for equal thermal capacity rates is defined by the equation (Kays and London, 1964):

$$\varepsilon_o = \frac{T_i - T_o}{T_i - t_i} = \frac{t_o - t_i}{T_i - t_i}$$

where T_i, T_o = inlet and outlet air temperatures for the charge period

t_i, t_o = inlet and outlet air temperatures for the discharge period

Solutions to the first equation are graphed in Figure 5-4 with heat exchange area presented as a function of overall heat transfer coefficient and overall heat exchanger effectiveness. The thermal capacity rate is taken as the product of the air mass flow rate (2600 lb/sec) and the specific heat at mean air temperature (say 0.25 Btu/lb °F). This figure shows the extreme sensitivity of heat exchanger size to effectiveness at low heat transfer rates, which is to be expected since size varies as a function of $\varepsilon / (1 - \varepsilon)$.

For a compressed air to molten salt tube and shell heat exchanger, the heat transfer rate can be anywhere from 5 to 80 Btu/hr ft^2 °F depending on specific operating conditions and design features. Estimates of heat transfer coefficients for systems with compressed air on the tube side and brine on the shell side typically range between 20 and 40 Btu/hr ft^2 °F (Perry, 1973). Assuming brine is somewhat similar to molten salt in heat transfer characteristics and using a mean value of 30 Btu/hr ft^2 °F for the heat transfer coefficient, the heat exchanger surface area needed to achieve a roundtrip effectiveness of 0.8 (which corresponds to a one-way effectiveness of 0.89) is approximately 0.63×10^6 ft^2.

The total cost of the heat exchanger for this effectiveness is estimated as $13 x 10^6 based on the unit cost of $20/ft^2. The cost assumes stainless steel construction, which is preferred for salt systems operating above 700°F to achieve a long service lifetime. A comparative surface area required for 0.9 effectiveness is shown in Figure 5-4. Figure 5-5 shows the return temperature of the compressed air as a function of overall heat exchanger effectiveness for the TES 1 and TES 2 charging temperatures.

Added to the cost of the heat exchanger is a substantial investment in heat storage fluid. Approximately 80 million pounds of salt would be required for TES 1, based upon a specific heat of 0.37 Btu/lb °F, 10 hours of storage capacity, and 25 percent excess salt in the system. The storage volume for this quantity of salt would be approximately 708,000 ft^3. Because the inlet air temperature to TES 1 is below the minimum safe operating temperature of draw salt (i.e., 480°F), higher grade salts such as HITEC or Partherm 290, which are acceptable at temperatures above 350°F, would be required to prevent problems of solidification at the low end temperature. The cost of Partherm 290 is $0.41/lb or approximately $33 x 10^6 for the quantity required (Radford, 1980).

Therefore, the combined cost of the heat storage fluid and heat exchanger (excluding the cost of containment, piping, pumps, valves, and controls) is roughly $46 x 10^6 for one TES. For a two-stage indirect system these costs would be considerably higher.

A reduction in expensive heat storage fluid can be achieved, for example, with the displacement of salt by iron oxide pebbles (Burolla, 1979). Such modifications can reduce the storage medium cost somewhat, but the total system cost still hovers above direct contact systems, especially when the value of reduced power output as a result of lower return air temperatures is capitalized over the life of the plant.

A summary of various indirect thermal storage systems screened for water/steam and organic fluid solar thermal receiver applications is presented by Copeland (Copeland, 1980).

5.5 - TES Materials

Thermal and physical properties and costs were obtained from manufacturers and literature sources for a number of potential TES media for packed-bed regenerators. Table 5-2 presents the specific heat, apparent density, energy storage density, and cost for some of the storage materials reviewed. Apparent density is defined here as the mass of a particle per unit volume including internal porosity. The energy storage density is simply the product of the apparent density and the specific heat.

5.5.1 Selection Criteria - Operating conditions of the TES require heat storage materials to be able to:

- withstand temperatures up to approximately 850°F,
- cycle over a temperature range of approximately 450°F for TES 1 and 730°F for TES 2,
- resist thermal shock,
- withstand condensation and evaporation of water in TES 2,
- withstand an oxidizing atmosphere, and
- withstand effects of mechanical stresses.

Because of the novel application of the TES in terms of its large size and operating conditions, information was not widely available to substantiate the performance capability of many potential solid TES media, particularly for pebble bed regenerators. The only materials identified that could probably be assured of reliable performance over a long operating lifetime were prohibitively expensive, high-grade materials such as chrome cast iron or alumina grinding balls. However, for this study, unquestionable reliability was not a prerequisite for selection since many materials have what appear to be the appropriate properties but simply remain untested under conditions similar to the subject TES design.

The TES operating conditions presented a difficult materials selection problem because temperatures are too low to justify the use of expensive refractory materials, yet they are sufficiently high that performance data is lacking for a number of low cost materials not typically used under such thermal conditions. Materials with irreversible phase changes or similar deleterious characteristics within the temperature ranges of interest were avoided because of their greater potential to spall and accelerate bed degradation. Magnesite, for example, performs satisfactorily at high temperatures, but tends to react with water at low temperatures (around 500°F), causing fissures in the material that could shorten bed life.

Thermal shock may also cause some materials to break under the TES operating conditions. Glass marbles, for example, tend to break when rapidly cooled because of the high tensile stresses developed in the outer shell (Rayner, 1980).

The presence of condensed water in TES 2, as discussed in Section 5-3, can accelerate corrosion of some metals and eliminate many refractory materials from consideration. One problem linked to the condensation of water in the TES is the deposition of acids from the atmosphere onto packed-bed surfaces. The potential threat of atmospheric pollutants to TES components is currently unknown.

A major requirement of TES materials is their ability to withstand the effects caused by mechanical stresses. These stresses can derive from at least three sources:

- pressure caused by the weight of the stacked bed,
- gas pressure on the particles, and
- differential thermal expansion between the bed particles and the containment wall, and between the bed particles themselves.

The last source of stress represents the largest unknown of the above. If the pebble matrix does not expand en masse in response to stresses induced by heating, settlement may occur as the containment wall expands. When the TES is cooled, contraction of the containment wall can induce stresses in the matrix body and the containment wall. Over repeated heating and cooling cycles, a progressive ratcheting mechanism can be established if there is no slippage between the bed particles. This situation can lead to excessive bed attrition or, possibly, failure of the containment structure. The latter condition is of great importance since failure would be catastrophic. Unfortunately, very little information is known, or at least recorded, on the effects of mechanical loading in packed beds. In pebble bed wind tunnel applications, however, this effect has not been observed to be of major significance (Lindahl, 1980). To lessen the effects of mechanical stresses, materials with a low coefficient of thermal expansion, high wear and abrasion resistance, smooth surface, and high strength are probably desirable.

In addition to satisfying basic conditions set by the operating environment, TES materials should also have the following qualities:

- high density
- high specific heat
- good availability
- long service life, and
- low cost.

Materials with a combination of high density and high specific heat are desired to achieve minimum TES volumes. High thermal conductivity will also reduce the TES volume, but its effect is relatively minor. Availability of materials is important, but given a sufficient lead time, as one would expect for the planning and construction of an adiabatic CAES facility, supply does not present an unreasonable barrier for any of the major materials reviewed. Because transportation costs are significant, the source of supply should probably be as close to the site as possible.

Material life is extremely important for the TES design as replacement costs can be critical to the economic viability of adiabatic CAES. Similarly, the candidate TES materials should have low first cost, since the cost of many materials can dominate the total capital cost of the TES.

5.5.2 Preliminary Selection - Table 5-3 presents a preliminary ranking of selected candidate materials on the basis of material cost, energy storage density, minimum volume (see Section 5.7.5), matrix weight, and matrix cost. All of the materials listed in this table are spherical shaped ranging in diameters from a little less than 1/2 inch to one inch.

Heat transfer salts and oils were eliminated from further consideration because of (1) material limitations over the operating temperature ranges of the TES, and (2) the need for heat exchangers which reduce thermal performance as discussed in Section 5.4.3.2.

Checkers are stackable bricks made for storage of heat. These can be manufactured from a variety of materials such as fireclay, alumina, or magnesite and are widely used in hot blast stove furnaces. Checkers are characterized by low surface area to volume ratios and low heat transfer coefficients when compared to pebbles, thus requiring much larger TES volumes. A 3/4 inch pebble has over five times the surface area to volume ratio of the 1-1/2 inch hole M&P checker (manufactured by G. R. Stein), which has one of the highest surface area to volume ratios commercially available (Glendenning, 1979). Improving the design of checkers by increasing the number of holes and reducing the hole size has been suggested so as to increase the surface area to volume ratio (Chew, March 1980). Manufacturing problems and potentially high checker costs, however, are barriers to development and implementation of these improved checkers. Consequently, commercially produced checkers were eliminated from consideration as a heat storage material for purposes of this study.

Despite these drawbacks, checkers are designed for stacking and, therefore, have the very desirable feature of a built-in allowance for thermal expansion. Thus, improved design could bring checkers to the forefront of candidate TES materials in the future.

Of the materials reviewed, rock, sintered iron oxide, Denstone, tabular alumina, and white cast iron were found potentially suitable as TES media. Further testing, however, is required under the TES operating conditions to confirm suitability, particularly for rock and iron oxide pebbles.

5.5.2.1 Granite - Granite has the attraction of very low cost and good local availability. Thermal and physical properties vary widely depending on the source. Generally, granite has an acceptable energy storage density, good to excellent hardness and strength, and low absorption of moisture. Though little is known about the performance of granite under operating conditions similar to the subject TES, some laboratory experiments have shown encouraging results when cycled 600 times between 600°F and 1100°F (Burolla, 1980). Because of its low cost, granite is a candidate material that commands further testing. The possibility of using excavated site rock is another incentive for serious consideration of this material.

5.5.2.2 Iron Oxide Pebbles - Iron oxide is the material reduced in blast furnaces for the manufacture of iron and steel. Through a sintering and pelletizing process, iron ores such as taconite, magnetite, and hematite are converted to iron oxide pebbles for easy shipping and efficient processing in the blast furnace. The majority of iron oxide pebbles range in size from 1/4 to 5/8 inches. Pebble composition is approximately 94 percent Fe_2O_3 and 5 percent SiO_2 by weight.

The iron oxide pebble has a porosity of approximately 20 to 22 percent, but water penetration is much lower (3 to 5 percent) because of the small pore size (Domingquez, 1980). Iron oxide pebbles are unaffected by oxidation or the presence of water, but its resistance to thermal shock, and wear and abrasion in the TES environment is a major unknown that requires further research. A 1/2 inch pebble crushes under an average weight of approximately 400 lb. Like rock, iron oxide is a promising material because of its relatively low cost and high availability. Its energy storage density is slightly greater than rock.

5.5.2.3 Denstone Pebbles - Denstone, manufactured by Norton Company, is a fireclay pebble designed for use as a catalyst bed support. This material represents a substantial increase in cost from rock and iron oxide. Though Denstone has a lower energy storage density than rock or iron oxide, considerable operational experience exists in environments similar to the TES. Denstone is likely to withstand the TES operating temperatures and the effects of thermal shock. It is reported by Norton to have good resistance to spalling and attrition, and exhibits a low absorption of water (approximately 0.4 percent by weight). The cold crushing strength of Denstone is similar to iron oxide. The material is believed to resist the effects of water condensation. To improve the energy storage capability of Denstone, Norton is currently looking at ways to increase its density through compositional modifications (Vaccareillo, 1980).

5.5.2.4 Tabular Alumina Balls - Tabular alumina balls (sintered alpha alumina) are composed of 99 percent aluminum oxide (Al_2O_3). The material is marketed in sizes ranging from 1/8 to 3/4 inches for chemical reactor and catalyst beds. The properties of alumina balls appear suited for the TES environment. The material resists spalling and corrosion and offers good resistance to thermal shock, high temperatures, and abrasion. The cost of tabular alumina, however, is more than doubled that of Denstone, which is a major barrier to its use on a large scale as a heat storage medium.

5.5.2.5 White Cast Iron Grinding Balls - Cast iron appears in two basic forms (gray or white) depending on the carbon formation. In gray cast iron the carbon appears as graphite flakes that occupy approximately 10 percent of the matrix volume. These flakes give the material its characteristic gray fracture and promote machinability, but they also impair the continuity and lessen the strength of the matrix.

In white cast iron the carbon is fixed in the form of iron carbide or cementite (Fe_3C). The fixed carbon gives the material high compressive strength, hardness, and abrasion resistance, but also low tensile strength and impact resistance. White cast iron is typically used in applications where resistance to wear is desirable and service does not require ductility.

White cast iron is produced by proper adjustment of the chemical composition. The silicon content is reduced to promote formation of iron carbide, thus producing white iron. Generally, alloying elements such as nickel, chromium, and molybdenum are added to stabilize the carbide, increase strength, and provide other special properties depending on the service. For example, white cast iron with a high chromium content ("chrome" cast iron) is often used to combine excellent wear resistance with oxidation and corrosion resistance at elevated temperatures.

For the TES service conditions, the degree of alloying that would be necessary for satisfactory performance is uncertain. The goal should be to minimize the quantity of alloy material to hold costs down, but still achieve acceptable performance. Some alloy, however, would probably be necessary to reduce the effect of graphitization that occurs in cast irons above 800°F.

Of the five materials described, crushed granite, iron oxide, and white cast iron were selected for detailed modeling to determine TES size requirements and performance characteristics. Granite and iron oxide were selected because of their low cost, high availability, and reasonable energy storage density. White cast iron was selected because of its high energy storage density, and hardness and strength qualities. The selection of these three materials, however, is not intended to exclude other materials from future consideration. Denstone, for example, is also a promising material, especially with compositional modifications that could improve energy storage capability.

5.6 TES CONFIGURATIONS

This section discusses the basic arrangement selected for containment of the pebble bed TES system.

5.6.1 Surface Versus Underground Containment of the TES - For the PEPCO site, TES containment could be located on the surface using steel, post-tensioned concrete, or post-tensioned cast iron pressure vessels; or underground in excavated hard rock caverns. One objective of this project was to develop an underground TES containment system if suitable for the selected heat storage system. The intent was to parallel the design of a surface containment for TES as part of a conceptual design of a hybrid CAES plant funded by EPRI (United Engineers and Constructors, 1980).

5.6.2 Vertical Versus Horizontal Arrangement of the TES - Two basic arrangements were perceived for an underground TES - the vertical (or stack) design and the horizontal (or silo-in-cavern) design. Figure 5-6 presents the basic arrangement of the two concepts. The terminology refers to the arrangement of the heat storage material and is not to be confused with the direction of air flow through the bed, which is always vertical.

In the vertical design the air access shaft to the air storage cavern is used for containment of the heat storage medium with appropriate structural modifications. The vertical design imposes no restraint on the height of the TES bed due to the structure of the hard rock pressure containment. The cross-sectional area of the bed can be reduced, with a consequent increase in bed height, up to the limiting condition where the pressure drop becomes unacceptable. The advantage of reducing bed cross-sectional area is that the bed volume can be reduced for a given thermal performance (i.e., given end temperature difference, see Section 5.7.1), though this is achieved at the expense of some increase in thermal breakthrough. The stack design may also provide advantages in terms of mechanical design and cost, such as possible reduced cost of excavation, reduced span of bed support plate, and flexibility to incorporate design modifications such as intermediate bed support plates.

In the horizontal design, the bed matrix is restrained in height and width by allowable cavern dimensions. These are dictated by stress distributions in the cavern walls. The necessary TES volume must be achieved by increasing the bed cross-sectional area, that is, the length of the cavern. Uniform distribution of air over the bed approach area on both charge and discharge cycles is a more significant problem for this arrangement.

The critical factor to the selection of the vertical or horizontal design is the consideration of bed pressure drop. Figure 5-7 shows the variation of bed pressure drop as a function of bed approach area for several materials. This figure indicates the variation in bed length for the vertical TES design. The bed volume required for each material is based on the nominal minimum volume (see Section 5.7.5), assuming complete utilization of the storage capacity of the material, plus a margin of 10 percent. This margin has been found generally sufficient to ensure a high level of thermal performance for small diameter pebbles.

The actual bed volume required will depend on the thermal performance design and the bed cross-sectional area, but the bed length and hence pressure drop for any given approach area within the range considered will be in the ball park of the values shown. The bed approach area can, of course, be subdivided to provide a number of beds in parallel.

The pressure drops of Figure 5-6 were evaluated from the Ergun relationship with a voidage of 0.38. This relationship is widely used for estimating pressure drops in packed-bed systems and is given by the following equation (Ergun, 1952):

$$P = \left[\frac{150\ (1 - e)}{\frac{\rho V d}{\mu}} + 1.75 \right] \left[\frac{(1 - e)}{e^3} \right] \left[\frac{L \rho\ V^2}{d} \right]$$

where:

P = frictional pressure drop
L = bed length
ρ = mean gas density
V = approach gas velocity
d = pebble diameter
μ = mean gas viscosity
e = voidage

At high mass flux, the first term inside the bracket becomes small and pressure drop varies as a function of $(1 - e)/e^3$. Thus, if the voidage were to fall from 0.38 to 0.2 due to the generation and retention of particulates in the bed, the pressure drop rises by a factor of 8.9. Hence, in the vertical design it would be necessary to either use a large safety margin on pressure drop or utilize materials with which particulate generation would likely be minimal. From the viewpoint of particle carry-over to the piping and machinery, the latter course is obviously preferable. However, even with careful material selection and system design, it would still be necessary to employ some safety margin on pressure drop.

After comparison of vertical and horizontal designs, the horizontal TES was preferred, on balance, for the following reasons:

(1) The penalty on bed volume with the horizontal design was shown to be acceptable. Volume penalty in the worst case (one inch cast iron pebbles) was estimated to be less than 15 percent more material than that required of the vertical design.

(2) The pressure drop across the TES bed would be higher with the vertical design.

(3) The vertical concept would be severely penalized by particulate generation, as pressure drop could escalate to many times the values shown in Figure 5-7.

(4) For reasonable levels of pressure drop, the bed approach area required for the vertical TES, while much smaller than that for the horizontal design, is still large. Taking into account possible escalations in pressure drop, the required bed approach area would have to be greater than 3000 ft^2 for TES 2 and much larger still for TES 1.

(5) A number of parallel shafts would probably be necessary to accommodate the required approach area for the vertical TES, leading to similar problems of air flow distribution encountered in the horizontal TES. Excavation costs of multiple shafts would also be high in unit volume compared to horizontal caverns.

5.6.3 <u>Cavern Configuration for the Horizontal (Cavern) TES</u> - Any of a number of rock cavern cross sections are possible for the horizontal design of TES. In the PEPCO study, four air storage cavern configurations were evaluated in detail and these were reviewed for their suitability for containment of TES. The configuration with the highest ceiling was selected because it provided the maximum vertical bed length for the TES and sufficient head room for installation of TES facilities. This cavern has overall dimensions of 106 feet high by 50 feet wide and a roof arch with a radius of 25 feet. The walls of the cavern are curved to improve structural stability. The height of the pebble bed was designed for 65 feet. The cavern can be of any length as required by the thermal design of TES. Of the four cavern designs studied, this configuration was also the least expensive to excavate.

5.7 TES SIZING

The major objectives of the thermal design of the TES matrix are: (1) high thermal performance, (2) smaller mass of heat storage material, and (3) small overall volume. These objectives can be achieved by:

- heat storage materials with high energy storage densities,
- high heat transfer coefficients,
- high surface area to volume ratio of the heat storage material, and
- low bed voidage.

The general relationship of these parameters to the overall TES volume is illustrated by the following expression, which was derived from expressions of heat transfer coefficient and volume presented by CEGB (Glendenning, 1979):

$$V = \left[C_1 \frac{d^{1.3}}{G^{0.7}} + C_2 \frac{d^2}{k} + C_3 \frac{1}{\rho C} \right] \left[\frac{1}{1 - e} \right]$$

where

V = TES volume
d = pebble diameter
G = air mass flux (e.g., lb/hr ft^2)

$$\left.\begin{array}{l} k = \text{thermal conductivity} \\ \rho = \text{density} \\ C = \text{specific heat} \\ e = \text{voidage} \end{array}\right\} \text{ of TES material}$$

C_1, C_2, C_3 = constants dependent on the gas properties, charge/discharge time periods, and design thermal performance

The TES volume is most influenced by the first (convective) and third (heat capacity) terms of this expression, while the effect of the second (wall resistance) term is very small for all cases. The voidage for pebble bed systems is largely fixed regardless of the pebble size. High surface area to volume ratios are achieved with small diameter pebbles, which improve the rate of heat transfer and thereby reduce TES volume. High heat transfer coefficients are also achieved with high air mass fluxes through the bed, which can be increased by reducing bed cross-sectional area. The last term of this expression indicates the advantage of using materials with high energy storage densities to achieve low volumes.

The relative effects on TES volume attributed to the convective and heat capacity terms depend on the extent that the actual bed volume is sized above the true minimum volume (see Section 5.7.5). When the bed is much oversized, convective effects dominate and reduction in bed volume is better achieved by reducing pebble size and increasing air mass flux. As the TES volume approaches minimum volume, however, the sensitivity of the volume to heat transfer effects becomes less dominant as the heat storage material is worked nearer to the limit; that is, each pebble cycles over a wider temperature range. Under this condition the influence of the heat capacity term on bed volume becomes greater than the convective term and reductions in volume are more easily gained by increasing the energy storage density.

5.7.1 TES Performance Parameters - The performance of the TES (i.e., TES 2) under daily thermally cycling conditions is illustrated in Figure 5-8 for the granite case study (Nash-Webber, 1980). This figure shows the temperature variations of the air at the top and bottom ends of the TES for a 12-hour cycle simulation starting from a cold bed. Each cycle includes a 10-hour charge period and 10-hour discharge period with a 2-hour soak separating each period. The temperature variations near the end of each charge and discharge period show the breakthrough of the temperature wave (thermocline). Figure 5-8 also shows that equilibrium (see below) is achieved after only a few cycles.

As illustrated by this example, the return temperature of the air is the same as the charging temperature for approximately the first half or more of the discharge period before thermal breakthrough begins. If desired, the start of thermal breakthrough can be delayed by increasing the air mass flux (G) through the bed, but outlet temperature fluctuation and pressure drop coincidently increase.

Thermal design of the TES can be defined by specifying the end temperature difference (ΔT_e) and the temperature fluctuation (ΔT_f). These parameters are calculated for a given bed matrix and set of operating conditions by the use of numerical models.

End temperature difference is defined as the difference between the hot air inlet temperature during the charge period and the time mean hot air outlet temperature over the discharge period. Similarly, on the cold end of the bed, the end temperature difference is the time mean air outlet temperature during the discharge period minus the cold air inlet temperature during the discharge period. The temperature fluctuation is the total temperature difference of the air outlet temperature between the beginning and end of the charge or discharge periods. Figure 5-9 illustrates the physical interpretation of these temperature terms.

If the total mass flow through the matrix is balanced for the charge and discharge periods and the cold air return temperature from the air cavern is controlled, then a condition of equilibrium can be established for operation of the TES. Equilibrium exists when the end temperature difference and temperature fluctuation for the hot and cold ends of the bed are equal.

Equilibrium TES conditions are achieved very rapidly for most materials and operating conditions, with 12 cycles generally being sufficient. Balanced mass flows, however, cannot always be achieved for CAES applications because of air lost through leakage from the air storage cavern. In this case, thermal equilibrium can be achieved, but the unbalanced mass flow will create a small difference at the hot and cold ends of the TES for the end temperature differences and temperature fluctuations.

End temperature difference is a very sensitive measure of TES thermal performance and is useful for turbomachinery design. From the viewpoint of heat exchanger design, however, end temperature difference does not adequately define TES thermal performance. This is more appropriately given by the effectiveness (ε), which is a measure of the amount of cooling (or heating) done on the air by the TES, and is defined by the following equation (Glendenning, 1979):

$$\varepsilon = 1 - \frac{\Delta T_e}{T - t}$$

where

T = hot air inlet temperature to the TES during the charge period

t = cold air inlet temperature to the TES during the discharge period

For the TES to operate effectively, the hot air (with average temperature $t + \Delta T_e$) that breaks through the TES at the end of the charge cycle must be cooled to TES re-entry temperature (t). Otherwise, the bed will heat up over successive cycles and equilibrium will not be achieved. Therefore, heat must be removed from the system via the water in the air storage cavern and released to the ambient environment at the surface. Some heat loss to the rock will also occur.

5.7.2 TES Models - With support from CEGB and MIT, two models were employed to size the TES. The CEGB computer model (called REGEN) was originally developed by CEGB for the study of regenerators and was revised for the CEGB study of adiabatic CAES concepts using pebble bed TES arrangements. The program solves, for the heat transfer equation between gas and solid surfaces, gas enthalpy change through the bed, and conduction into the solid material. Starting from an arbitrary initial temperature distribution in the matrix, the program calculates gas and solid temperatures through successive charge and discharge periods until the heat lost by the gas in the charging period is within 0.5 percent of the heat gained by the gas in the discharging period. The equations used assume (1) no axial conduction of heat through the bed matrix and (2) that temperature gradients into the solid material are so small that radial thermal conductivity is effectively infinity. Both assumptions were found acceptable for pebble bed regenerators.

The CEGB program solves the heat transfer equations by a predictor-corrector method with a basic time interval specified in the data input. The program subdivides the basic time interval by repeated halving until successive estimates of the temperature at a base time point differ by less than one part in ten thousand.

The MIT computer model was developed as a part of a series of studies performed by MIT on adiabatic compressed air energy storage for the Division of Energy Storage Systems of the Department of Energy. The computational model and code used were basically those described by Hamilton (Hamilton, August 1978), with certain corrections and changes, and minor enhancements to improve running speed, and input/output capabilities (Nash-Webber, 1980). Two auxiliary codes were written to plot the output data.

In addition to basic convective terms, the MIT model includes radiative, conductive, and boundary heat loss terms in the solid phase equation, and a heat storage term in the gas phase equation. This latter term measures the effect of pressure on the thermal behavior of the TES. For comparative runs at 40, 80, and 120 atmospheres, however, the maximum observed change in the predicted temperature profile at any station due to pressure effects was less than 3°F (Nash-Webber, 1980).

Preliminary comparisons of the thermal performance predicted by the CEGB and MIT models of the TES showed a significant difference, with the MIT model producing larger values of end temperature difference. For example, for a datum case of 0.75 inch diameter cast iron pebbles, the MIT model gave a value of ΔT_e of 59°F in comparison to the CEGB result of 18.3°F, a difference in heat transfer effectiveness of the TES of 5.6 percent. Part of the source of this difference was subsequently identified as the limited number of length steps being employed in the MIT model. Increasing the number of steps from 30 to 90 reduced the MIT value of ΔT_e to 38°F, leaving a difference in effectiveness of 2.7 percent in comparison to the CEGB model. Further increasing the number of length steps would reduce ΔT_e by a few degrees, but leads to excessive computer running times. The source of the remaining difference between the models was uncertain, but it is possibly due to differences in the respective numerical solution schemes. Investigation of this difference, however, was considered to be outside the scope of, and of limited significance to, the present study.

For design purposes, it was decided to employ the MIT model, with 90 length steps, to determine the general relationship between bed volume and thermal performance for the candidate bed materials. When the target bed performance was decided upon, the CEGB model was run and the final thermal performance quoted as the mean of the CEGB and MIT results.

Some additional tests were carried out on the MIT model to determine the significance of various terms in the basic differential equations. The tests showed that, even for an extreme case, with cast iron pebbles and employing the average cross-sectional area of the bed material, length-wise conduction of heat would have to be increased by at least two orders of magnitude before it became significant in relation to other heat transport terms. In addition, halving the operational pressure indicated that the effect of the heat capacity term for the air contained within the matrix at any instant, was small. While the latter test should be taken further, the result suggested that the MIT and CEGB programs were modeling the same physical situation, even though the CEGB model does not contain the air heat capacity term.

5.7.3 Heat Transfer Coefficient - The heat transfer coefficient used for sizing the TES was recommended by CEGB based on a survey performed by Meek that examined a number of heat transfer coefficients for randomly packed beds (Meek, 1962). The expression for the heat transfer coefficient used with both the MIT and CEGB models was proposed by Denton, et al (Denton, 1949), and is expressed as:

$$\left(\frac{h}{C_p V \rho}\right)\left(\frac{C_p \mu}{k}\right)^{2/3} = 0.584 \left(\frac{\rho V d}{\mu}\right)^{-0.3}$$

where:

h = heat transfer coefficient
ρ = gas density
C_p = specific heat of gas at constant pressure
V = gas velocity based on the matrix approach area
k = thermal conductivity
d = equivalent spherical diameter of pebble
μ = dynamic gas viscosity

5.7.4 TES Design - Figures 5-10 through 5-12 illustrate, for the three candidate bed materials, the relationship between end temperature difference and mass flux for the cavern design of TES, with a fixed bed length of 65 feet. Also shown is the fluctuation (ΔT_f) in air outlet temperature.

The data presented in Figures 5-10 through 5-12 were derived using the MIT and CEGB models, with hot air inlet temperature (T) to the TES (charge period) of 830°F and cold air inlet temperature (t) (discharge period) of 100°F. For other values of hot and cold inlet temperatures, ΔT_e and ΔT_f can be derived by scaling the values shown in proportion to (T - t). This procedure ignores the second order effect of changes in physical properties with temperature. The air and pebble properties used for the modeling effort are presented in Table 5-4.

From Figures 5-10 through 5-12, the bed cross-sectional area, and hence bed volume and mass of heat storage material, can be derived for any target thermal performance (target ΔT_e) and a given total mass flow of air. Table 5-5 shows the bed dimensions, mass of storage material, and temperature fluctuation taken as a mean of the MIT and CEGB modeling results for a value of ΔT_e of 25°F, a mass flow rate of 2600 lb/sec, and a bed voidage of 0.38.

The bed dimensions presented in Table 5-5 were developed for the operating conditions of TES 2 as specified in Table 5-1. These dimensions were adopted for TES 1, taking a three percent penalty in volume and gaining the benefits of reduced end temperature difference (i.e., ΔT_e = 14°F) and reduced temperature fluctuation. The differences in thermal design are attributable to the smaller temperature swing of TES 1 under the same design bed effectiveness of 0.966. Though the TES operate under vastly different air pressures, the heat transfer coefficient was the same for both cases because the air mass flux (ρ xV) was held constant.

If only the CEGB results were used for sizing the TES beds, then the end temperature difference and outlet temperature fluctuation would be less than those presented in Table 5-5, and the temperature-time curves would be sharper for the same mass flow (G) through the beds. With iron oxide pebbles, air outlet temperature decrease would occur mainly over the last hour of the discharge period rather than the last three hours or so as shown in Figure 5-8 (and Figure 6-8) which displays sample output from the MIT program. Such changes in end temperature difference, temperature fluctuation, and temperature gradient improve thermal performance of the TES.

If the CEGB computer results were used as the sole basis for thermal design and the end temperature difference of 25°F was retained as a design criterion, then higher air mass flows through the TES would be achieved and approximately 15 percent reduction in TES volume for the case of iron oxide as shown in Table 5-5 would be realized. Similar volume reductions should be expected for the case studies of granite and cast iron. The volume reductions would be accompanied by higher outlet temperature fluctuations as indicated in Figures 5-10 through 5-12, while the sharper temperature-time curves would be retained. Greater temperature fluctuation would be tolerable because the fall in air outlet temperature would be delayed such that the turbines would see a constant, high air temperature output from the TES over a larger portion of the discharge period.

5.7.5 True Minimum Volume - A useful measure of the penalty in bed volume associated with a given material under given operating conditions is the comparison of the actual bed volume with the true minimum bed volume.

The true minimum bed volume is defined as the volume of bed required if all the bed material fluctuates in temperature between T and t while the air passed through the TES in the charge period falls in temperature from T to ($t + \Delta T_e$). True minimum volume is to be differentiated from the nominal minimum volume, which is often quoted and is the volume of bed required if the material fluctuates in temperature between T and t and the air passed through the TES in the charge period falls in temperature from T to t. Actual bed volume can be smaller than the nominal minimum volume but must always be greater than the true minimum value. True minimum volume is given by the following expression (Chew, April 1980):

$$\text{True minimum volume} = \frac{M\, c_p\, P}{\rho\, c\, (1 - e)} \left(1 - \frac{\Delta T_e}{T - t}\right)$$

where:

M = mass flow rate of air in charge period
c_p = specific heat of air (at constant pressure)
P = charge period
ρ = density of bed material
c = specific heat of bed material
e = bed voidage
ΔT_e = end temperature difference
T = hot air inlet temperature
t = cold air inlet temperature

A comparison of design bed dimensions (see Table 5-5) to the true minimum volumes for the TES materials under the design operating conditions is presented in the following list as the ratio of actual bed volume to true minimum volume:

0.44" Iron Oxide = 1.10
0.75" Granite = 1.17
1.00" Cast Iron = 1.29

The comparison shows the significant reduction in bed volume that can be achieved as a result of enhanced heat transfer rates with smaller diameter pebbles. Cast iron pays a severe penalty in bed volume because the selection of smaller diameter balls would entail a sharp rise in material cost. Computer runs, however, were not made to optimize the tradeoffs between the higher cost of smaller diameter balls and the lower cost of reduced bed volume.

5.7.6 Alternative Charge/Discharge Schemes - The TES sizes developed in Section 5.7.4 were derived from computer simulation of a balanced 10-hour charge/10-hour discharge cycle. After completion of TES sizing activities, however, air mass flow rates through the expander train were increased from 650 lb/sec per unit to 780 lb/sec to improve efficiency of the high pressure turbines. Consequently, to maintain balanced air flows through the TES based on the same charge mass flow rate and time period assumed for the 10/10 cycle, the discharge period was reduced to 8.3 hours.

If the same bed dimensions developed for the 10/10 cycle are used for the 10/8.3 cycle, then the air mass flux (G) on the discharge period would increase approximately 20 percent with a corresponding 14 percent increase in heat transfer coefficient for each heat storage material. Estimates of the net effect of this modification on TES thermal performance indicate that only minor changes would occur in the temperature fluctuation (less than 4°F) and end temperature difference (less than 0.5°F) for the case of granite. The true thermal behavior of the 10/8.3 cycle, however, should be determined by computer simulation.

The target plant size for the adiabatic CAES study was 10 hours of storage capacity for four power generating units. To achieve the necessary storage for this generating capacity with discharge mass flow rates 20 percent higher than charge mass flow rates, 12 hours of charging time were required. The extended charge period was accommodated by expanding the bed approach area. This expansion reduces air mass flux through the bed (by 17 percent) and lowers the heat transfer coefficient (by 13 percent), but it prevents extensive thermal breakthrough of hot air into the storage cavern. Estimates of changes in thermal performance for this modification indicated that they are on the same order as the 10/8.3 cycle. As with the 10/8.3 cycle, the 12/10 cycle needs verification by computer simulation to fully describe the behavior of the pebble beds with unbalanced air mass flow rates. Total mass flows, however, are in equilibrium for every cycle considered, discounting any air losses from the cavern.

Table 5-1

TES Operating Conditions (Daily Cycle) for a Four-Unit Plant[a]

Item	TES 1	TES 2
Operating Pressure (psia)	226	1215
Hot Air Inlet Temperature Charge Cycle (°F)	870	830
Cold Air Inlet Temperature Discharge Cycle (°F)	460	100
Air Mass Flow Rate per Unit (lb/sec)	650	650
Total Air Mass Flow Rate (lb/sec)	2600	2600
Charge/Discharge Time Period (hr)	10/10	10/10

(a) Compressor inlet air temperature equals 90°F.

Table 5-2

TES Material Properties

Material	Apparent Density (lb/ft^3)	Specific Heat[a] (Btu/lb°F)	Energy Density (Btu/ft^3°F)	Cost ($/ton, F.O.B.)	Source
Granite	168	0.248	41.7	5 to 12	Rockville and St. Cloud, Minn.[b]
Quartzite	165	0.255	42.1	5 to 12	Jasper, Minn.[b]
Limestone	156	0.260	40.6	5 to 12	Bedford, Ind.[b]
Basalt	185	0.253	46.8	5 to 12	Dresser, Wis.[b]
Fireclay	150	0.230	34.5	390	1" - Denstone, Norton Refractories
Tabular Alumina	218	0.237	51.7	820	3/4" - T162, Alcoa
Slag	137	0.120	16.4	5	3/4" - Bethlehem Steel
Iron Oxide	218	0.210	45.8	45	Hannah Mining Company
Forged Steel	490	0.130	63.7	500	1" - Grinding Ball, Coates Steel
Gray Cast Iron	442	0.130	57.5	510	1" - Grinding Ball, American Maggotteaux Corp.
White Cast Iron	480	0.155	74.4	560	1" - Grinding Ball, Estimated Cost
Chrome Cast Iron	480	0.155	74.4	>1000	---
Partherm 290	113	0.370	41.8	810	Park Chemical (similar to HITEC salt, operates above 325°F)
Partherm 430	113	0.370	41.8	580	Park Chemical (similar to draw salt, operates above 480°F)
Fireclay Checker	120	0.230	27.6	---	G. R. Stein

(a) Specific heats are estimated at 550°F.

(b) Source of rock for which thermophysical properties were known.

References: Alcoa, 1976; Lindroth, 1971; Norton, 1974; Metauro, 1980; Touloukian, 1967; Domingquez, 1980; English, 1980; Huffaker, 1980; Walters, 1980; Lowe, 1980; Jolwacki, 1980; Radford, 1980; Strassburger, 1969; Glendenning, 1979; Taylor, 1980; ASM, 1978; Coastal, 1980.

Table 5-3

Preliminary Ranking of Some Potential Heat Storage Materials (Normalized to Iron Ore Pebbles)

Material	Material Cost (F.O.B.)	Energy Storage Density ($\rho \times C_p$)	True Minimum Volume[(a)(b)]	Total Bed Matrix Weight[(b)] (e = 0.38)	Total Bed Matrix Cost[(b)]
Iron Ore	1.00 (\$45.0/ton)	1.00 (45.8 Btu/ft^3°F)	1.00 (0.806 x 10^{6}ft^3)	1.00 (54.5 x 10^3 tons)	1.00 (\$2.45 x 10^6)
Granite Rock	0.16	0.91	1.10	0.85	0.13
Denstone	8.67	0.77	1.33	0.91	7.91
Tabular Alumina	18.22	1.13	0.89	0.88	16.14
Gray Cast Iron	11.33	1.26	0.80	1.61	18.31
White Cast Iron	12.44	1.62	0.62	1.35	16.87
Forged Steel	11.29	1.39	0.72	1.61	18.24

(a) Assumes an effectiveness of 0.966.

(b) Total volume, weight, and cost are based on true minimum volume (see Section 5.7.5) and do not represent actual sizes and costs as predicted by numerical models. Matrix costs exclude the cost of material transportation (which can be significant depending on the distance traveled and mode of transport) and installation.

Table 5-4

Datum Physical Properties

The following air and material properties evaluated at 550°F were used for the modeling efforts performed by MIT and CEGB:

Air:	Dynamic viscosity	= 0.0695 lb/ft hr
	Thermal conductivity	= 0.0255 Btu/hr ft°F
	Prandtl number	= 0.6796
	Specific heat (at constant pressure)	= 0.253 Btu/lb°F (includes allowances for high pressure)
	Gas constant	= 53.34 ft lb/lb°R
Granite:	Specific heat	= 0.248 Btu/lb°F
	Density	= 168 lb/ft^2
	Pebble diameter	= 0.75 in
	Void fraction	= 0.38
Iron Oxide:	Specific heat	= 0.21 Btu/lb°F
	Density	= 218 lb/ft^3
	Pebble diameter	= 0.44 in
	Void fraction	= 0.38
White Cast Iron:	Specific heat	= 0.155 Btu/lb°F
	Density	= 480 lb/ft^3
	Pebble diameter	= 1 in
	Void fraction	= 0.38

Table 5-5

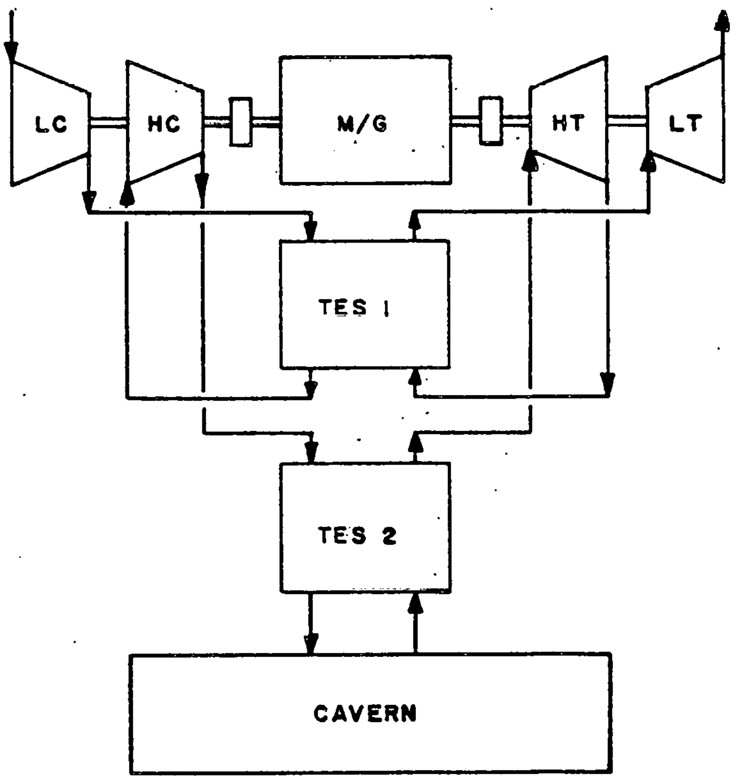
TES Bed Dimensions[a]

Pebble Material (inch dia.)	Cross-sectional Area (10^3 ft^2)	Bed Volume (10^6 ft^3)	Weight of Material (10^3 tons)	Temperature Fluctuation (°F)
0.75 Granite	15.9	1.04	54.0	238
0.44 Iron Oxide	13.7	0.89	60.1	282
1.0 White Cast Iron	9.81	0.64	94.9	198

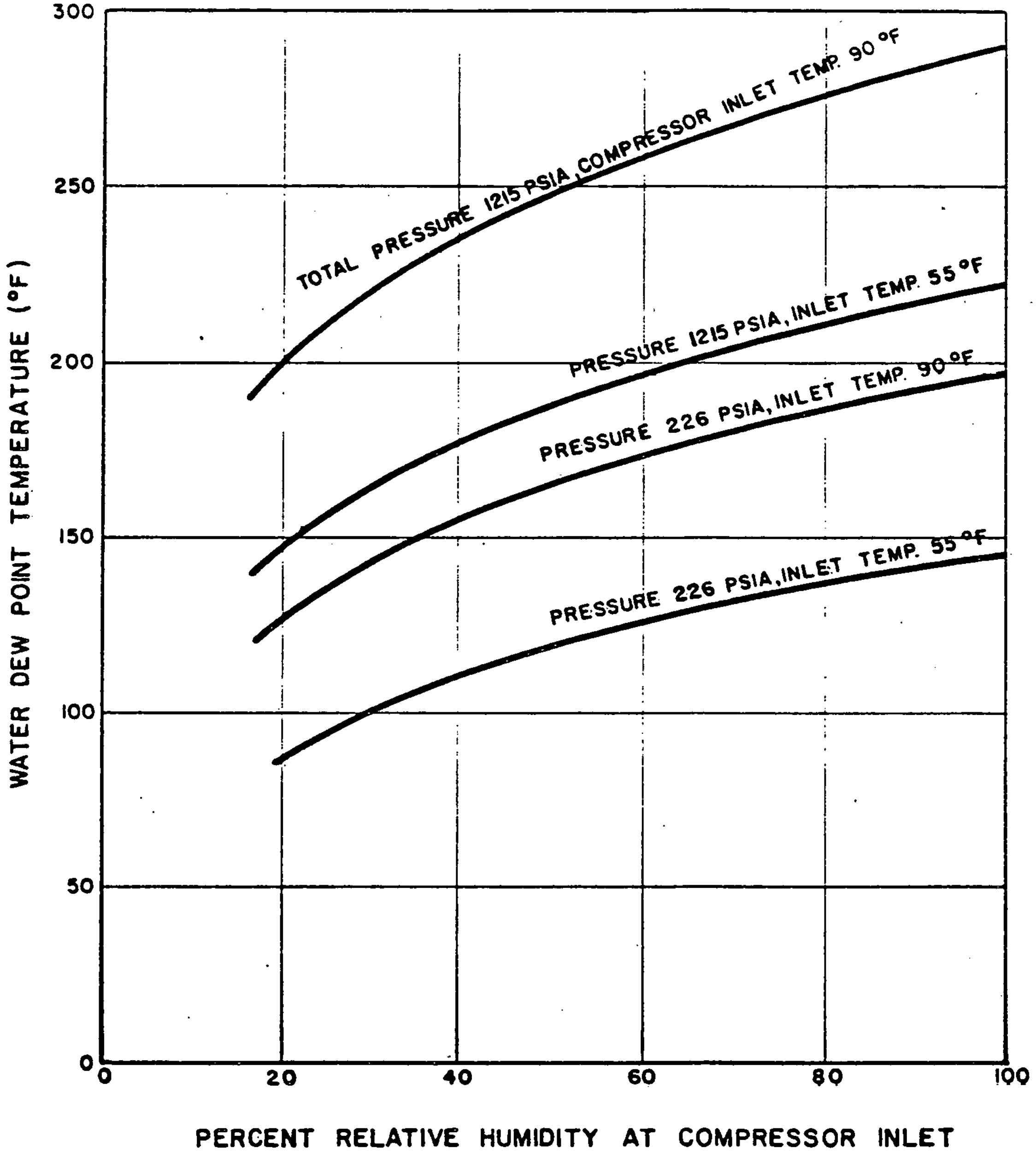
(a) NOTES:

- End temperature difference is 25°F.
- Difference between hot and cold inlet temperatures is 730°F.
- Heat transfer effectiveness of TES is 0.966.
- Bed height is 65 ft.
- Air mass flow rate is 2600 lb/sec.
- Bed dimensions are presented as a mean of CEGB and MIT computer simulation results. If solely CEGB results were used for sizing the TES, bed cross-sectional areas would be reduced approximately 15 percent below those values shown.

SIMPLIFIED CAES CYCLE



ACRES	BATTELLE PACIFIC NORTHWEST LABORATORY
	CONCEPTUAL DESIGN AND ENGINEERING STUDIES—ADIABATIC CAES
M J Hobson ACRES AMERICAN INCORPORATED	FIGURE 5-1



WATER DEW POINT TEMPERATURE

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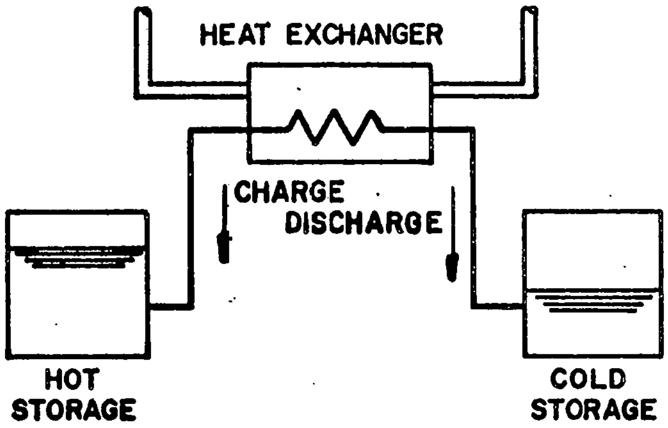
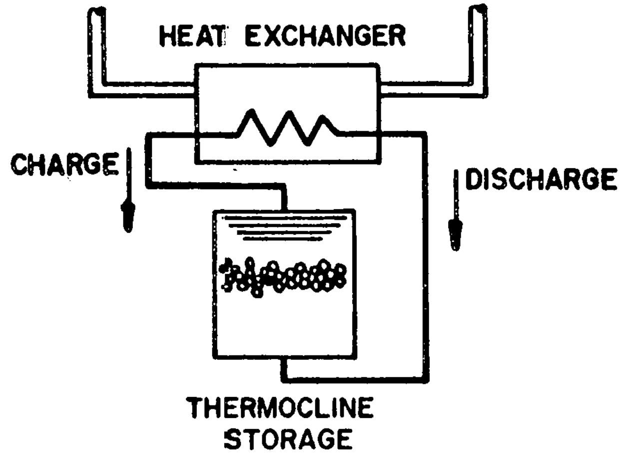
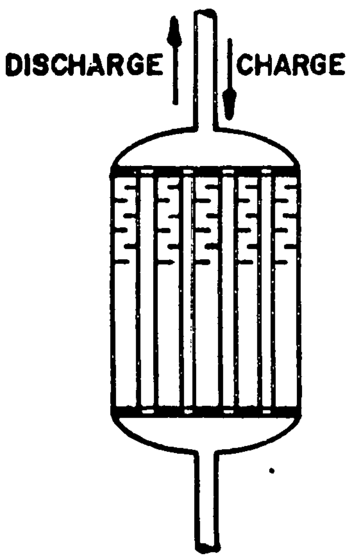


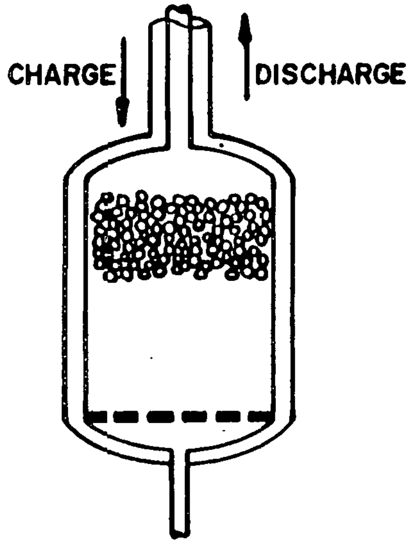
B. INDIRECT SYSTEM WITH HOT AND COLD STORAGE TANKS



C. INDIRECT SYSTEM USING HEAT TRANSFER FLUID AND THERMOCLINE STORAGE IN IRON ORE PEBBLES



A. IMBEDDED TUBES FOR INDIRECT SENSIBLE HEAT STORAGE



D. DIRECT CONTACT SYSTEM

BASIC TES ARRANGEMENTS

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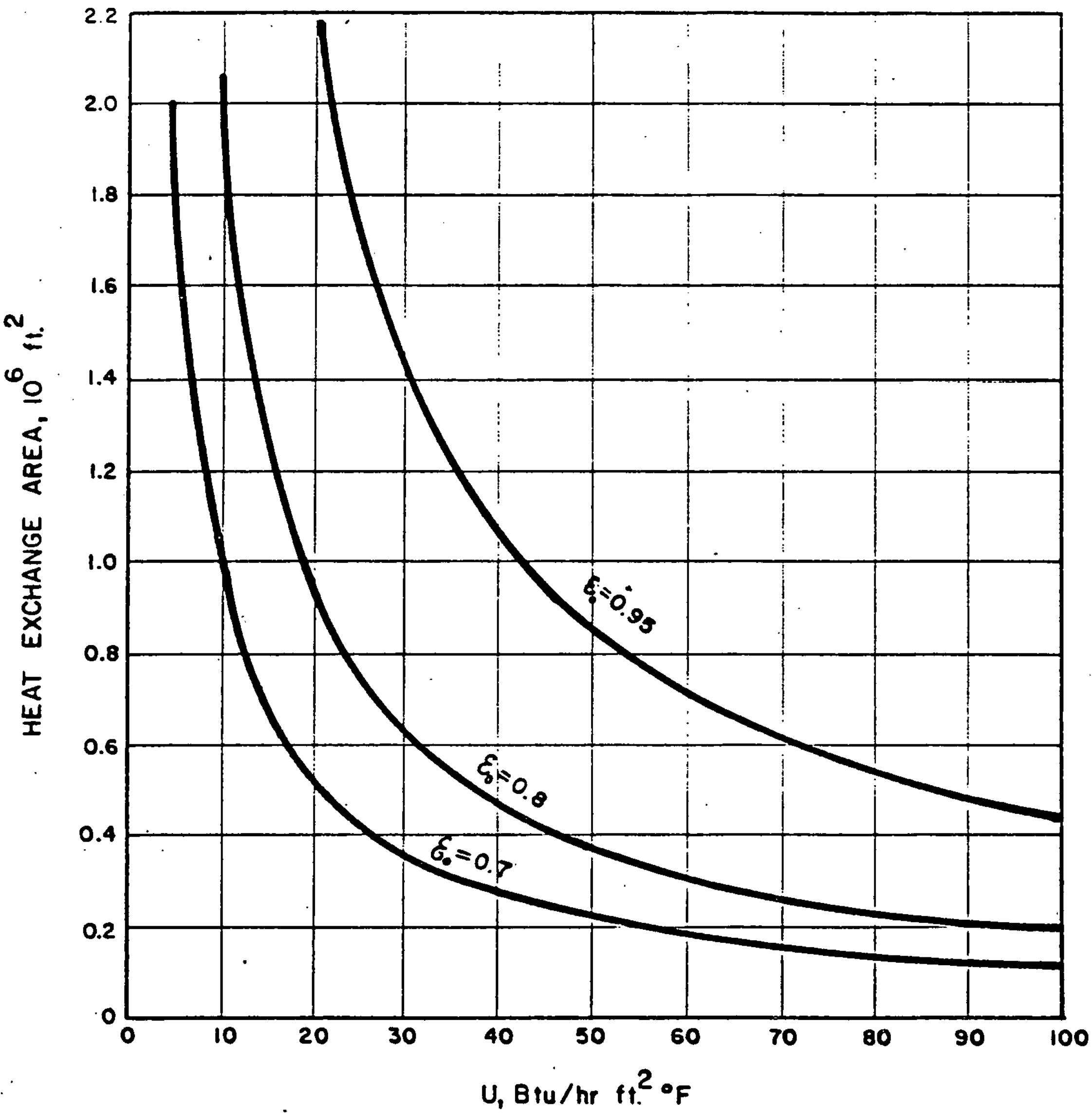
FIGURE 5-3



HEAT EXCHANGE AREA

AIR SPECIFIC HEAT = 0.253 Btu/Lb·°F
AIR FLOW RATE = 2600 Lb/sec.
U = HEAT TRANSFER COEFFICIENT
ε_o = OVERALL EFFECTIVENESS

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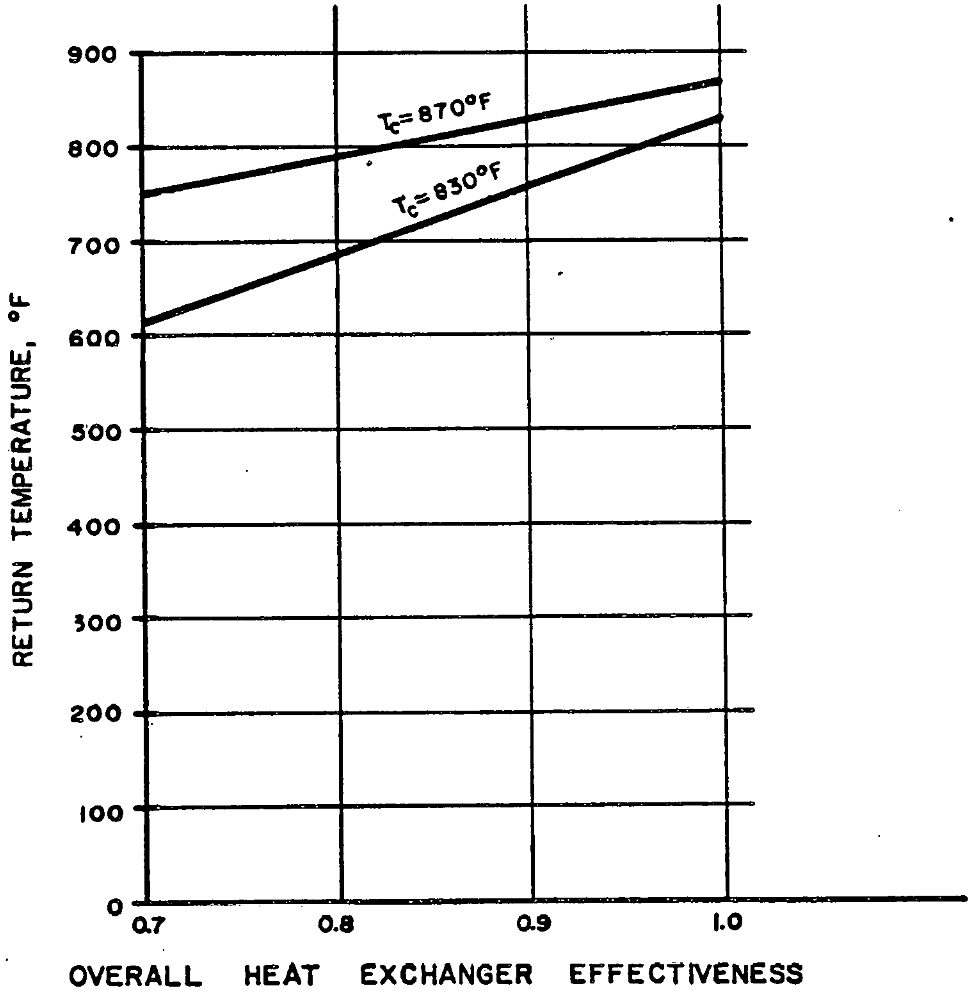

TES
RETURN TEMPERATURE

Tc = CHARGING TEMPERATURE

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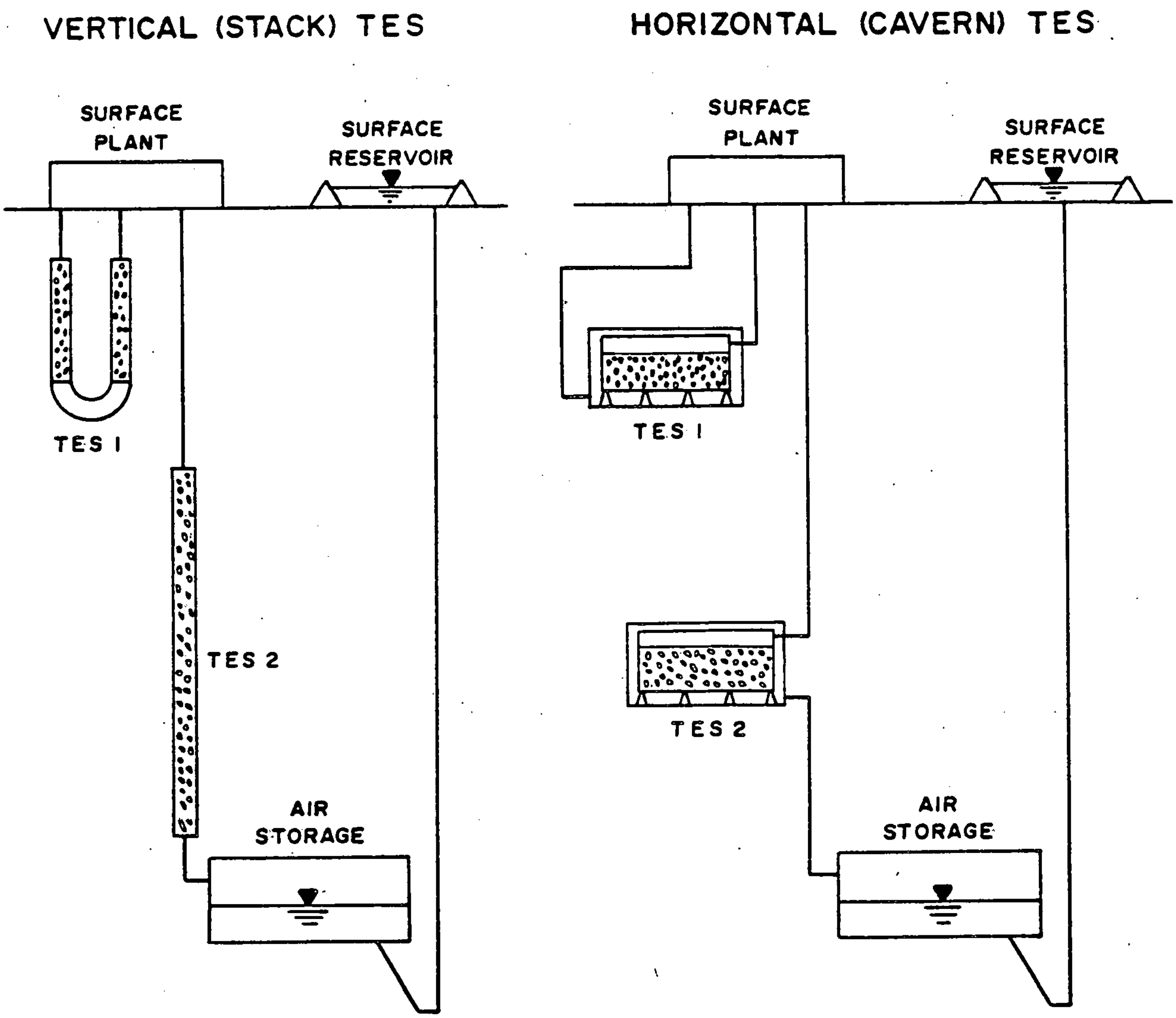
FIGURE 5-5



BASIC PEBBLE BED CONFIGURATIONS

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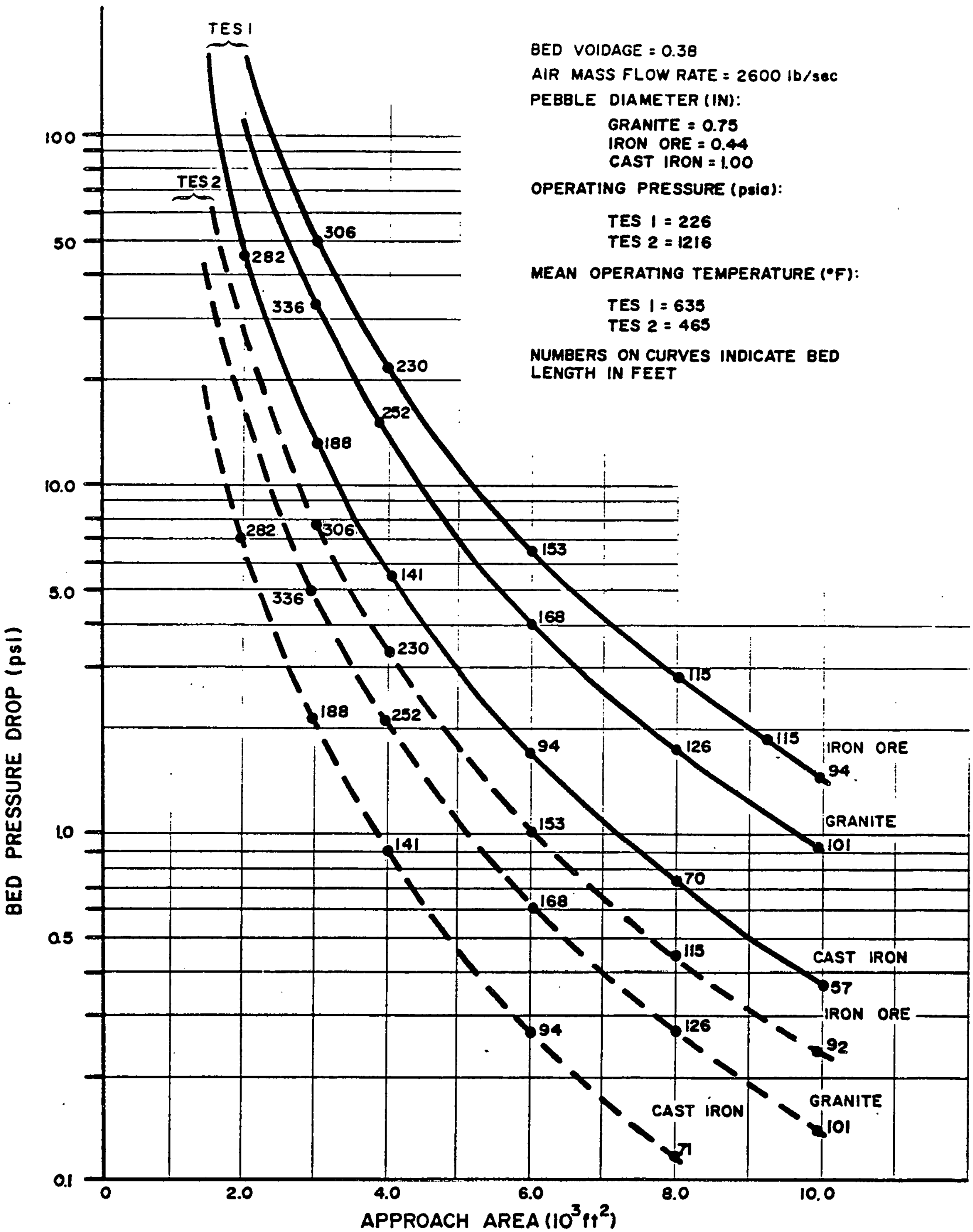

PRESSURE DROPS FOR TES STACK DESIGN

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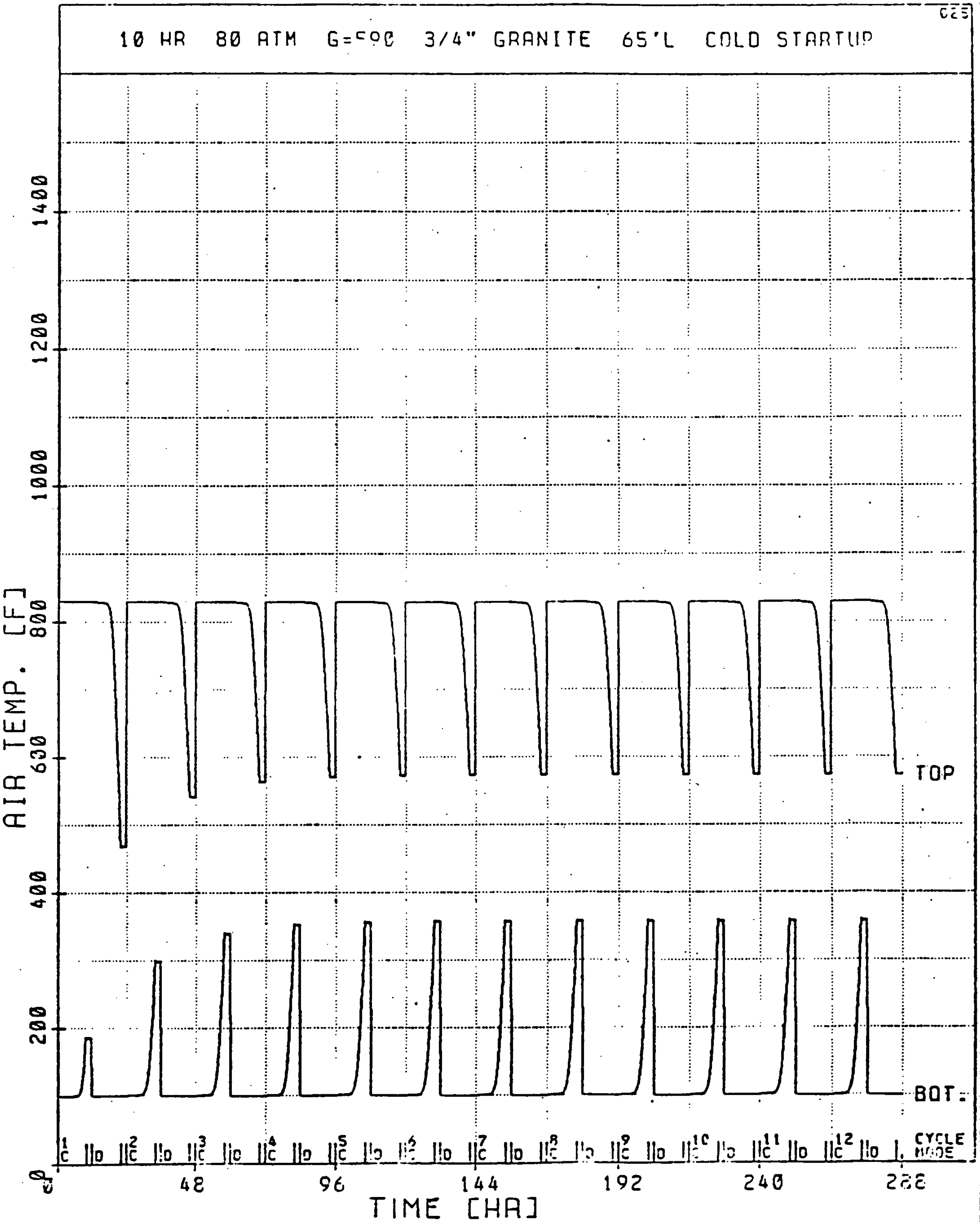
FIGURE 5-7



TYPICAL TES CYCLIC BEHAVIOR

(Ref. Nash-Webber, 1980)

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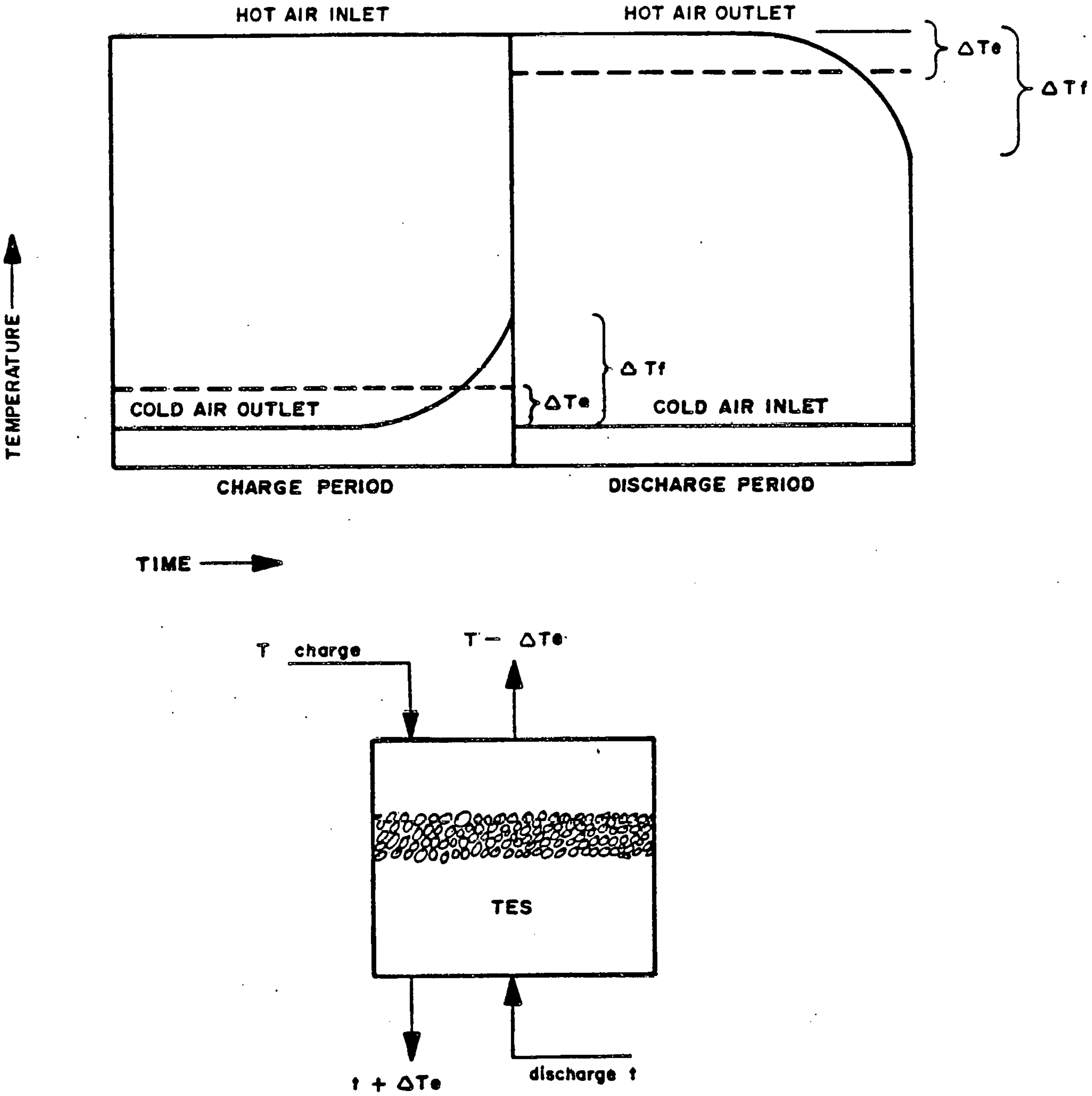
FIGURE 5-8



TEMPERATURE PARAMETERS FOR EVALUATING PERFORMANCE

ACRES	BATTELLE PACIFIC NORTHWEST LABORATORY CONCEPTUAL DESIGN AND ENGINEERING STUDIES—ADIABATIC CAES
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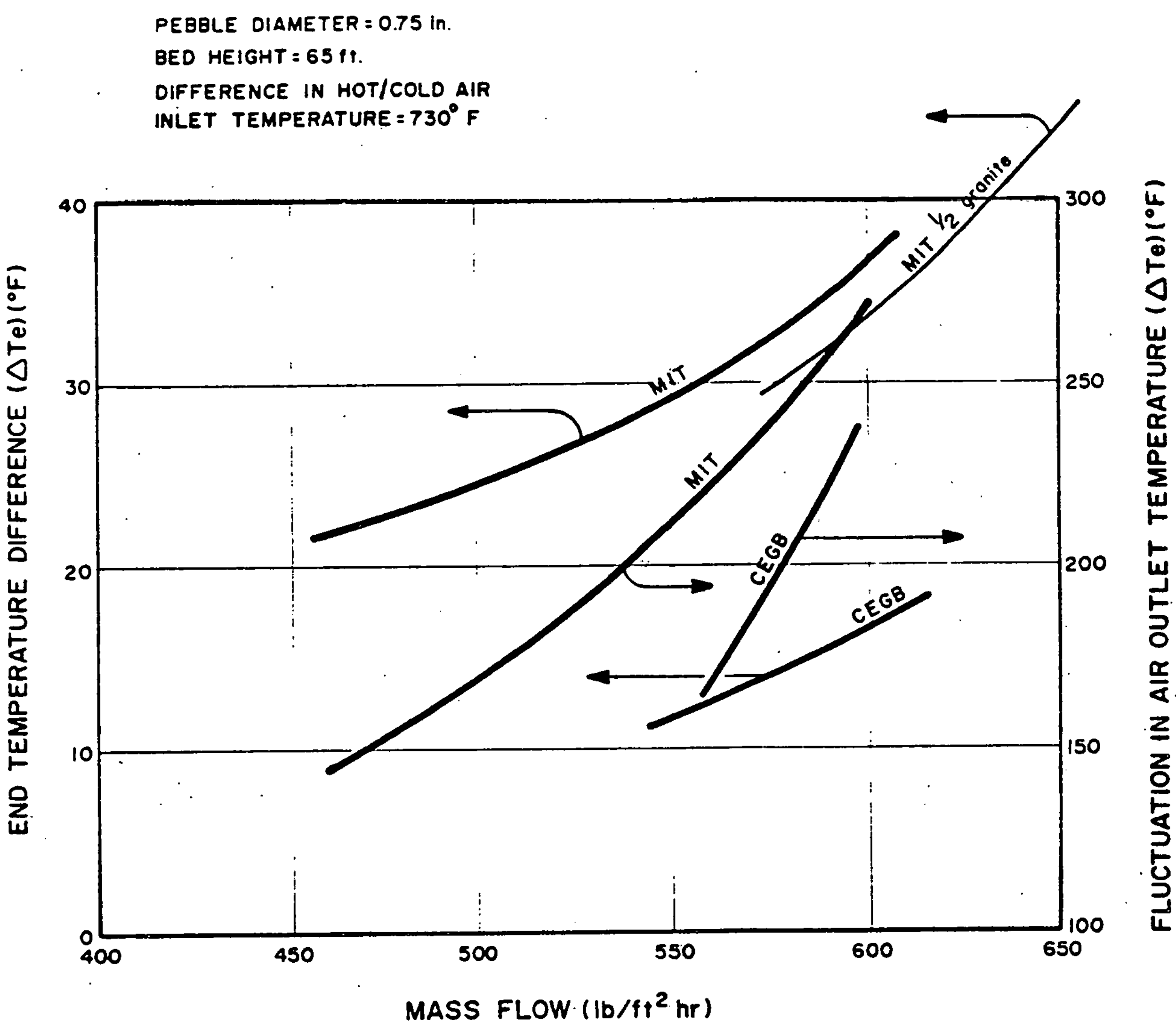

THERMAL DESIGN OF TES WITH GRANITE

ACRES	BATTELLE PACIFIC NORTHWEST LABORATORY CONCEPTUAL DESIGN AND ENGINEERING STUDIES—ADIABATIC CAES
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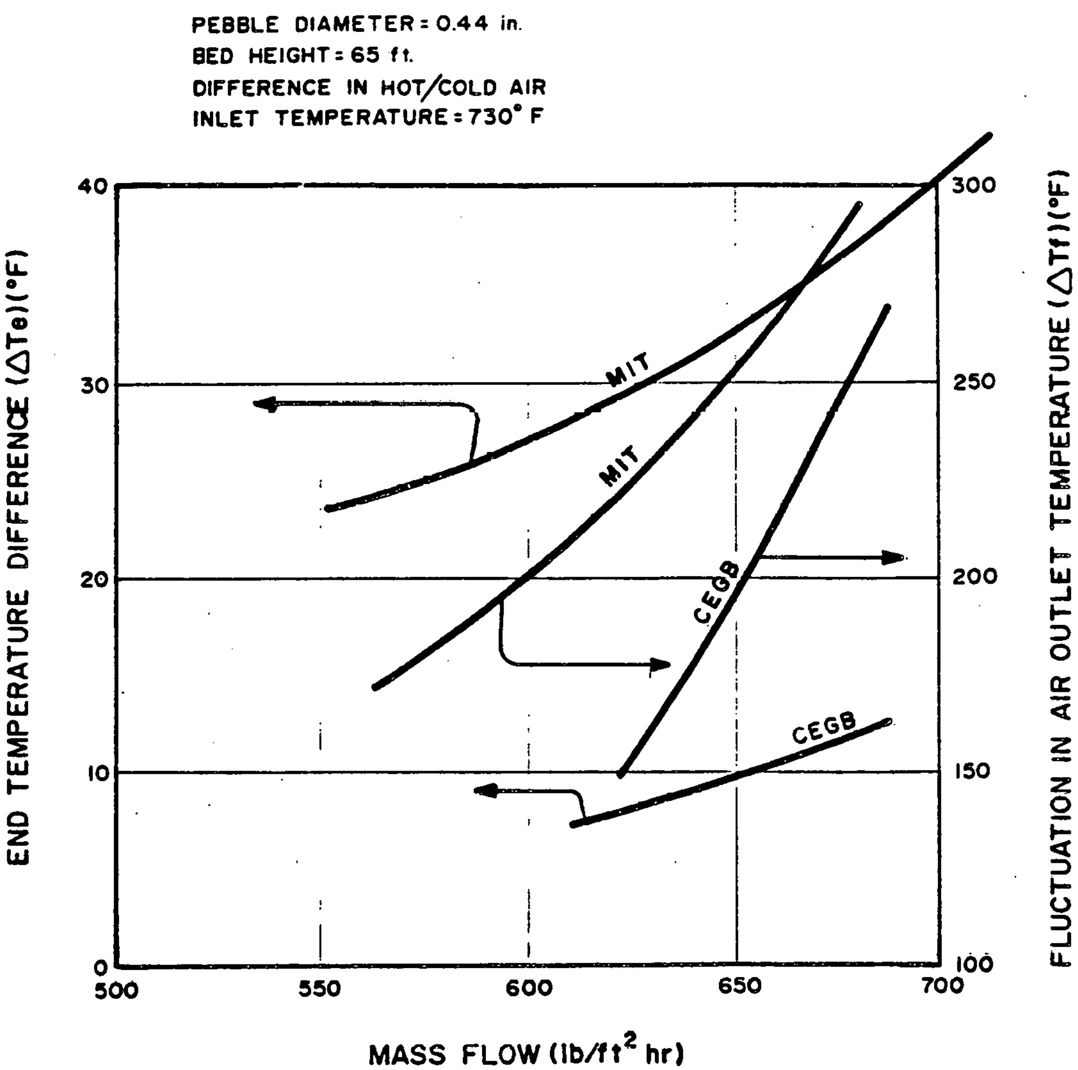

THERMAL DESIGN OF TES WITH IRON OXIDE

ACRES	BATTELLE PACIFIC NORTHWEST LABORATORY CONCEPTUAL DESIGN AND ENGINEERING STUDIES—ADIABATIC CAES
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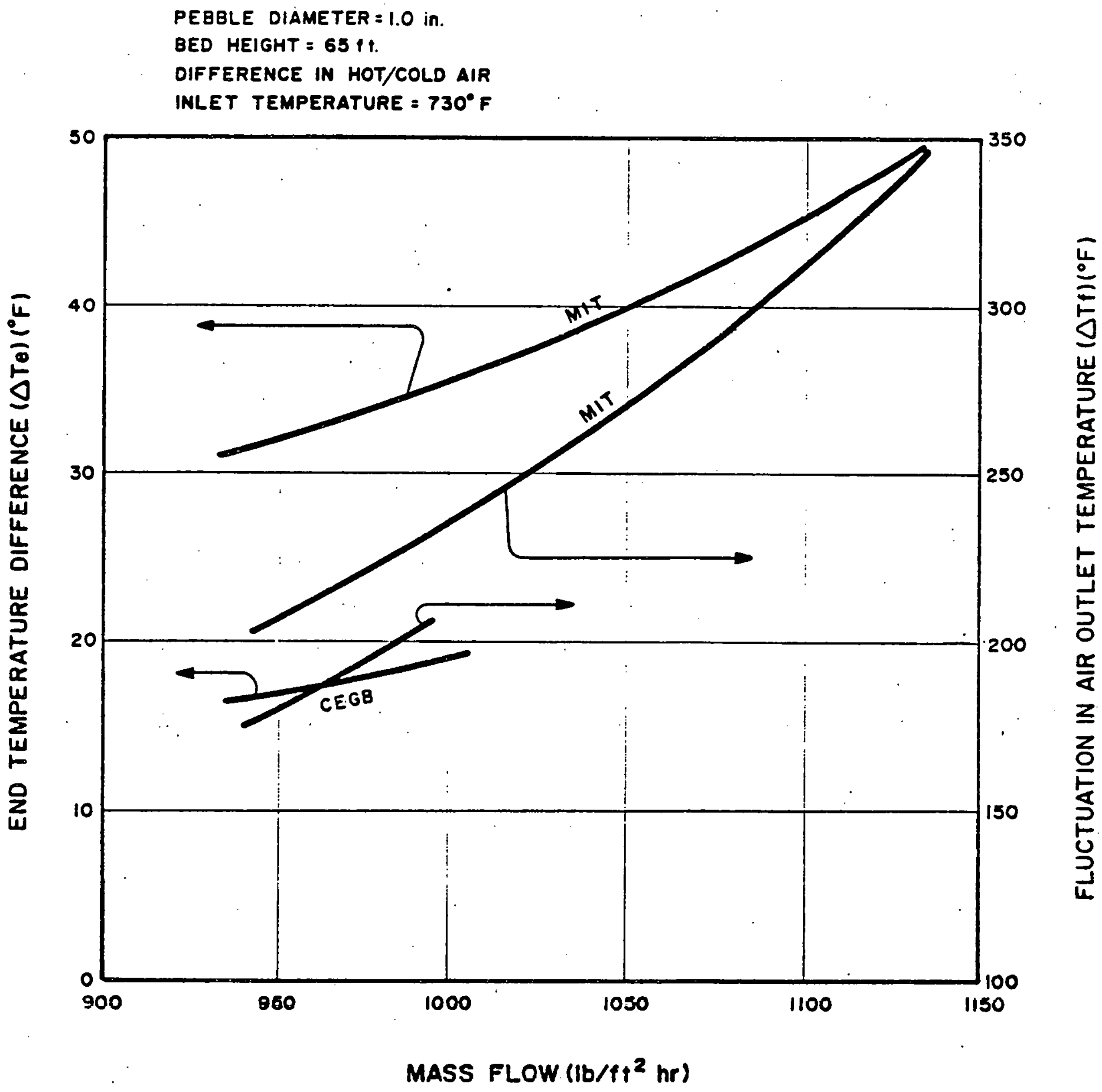

THERMAL DESIGN OF TES WITH CAST IRON

ACRES	BATTELLE PACIFIC NORTHWEST LABORATORY CONCEPTUAL DESIGN AND ENGINEERING STUDIES—ADIABATIC CAES	
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6 - TURBOMACHINERY

6 - TURBOMACHINERY

The turbomachinery equipment selection and performance evaluation incorporated the results of the cycle analysis and the thermal energy storage system investigations. This section describes the machinery selections and arrangement and discusses the expected general performance of the equipment.

6.1 COMPRESSOR SYSTEM SELECTION

Machinery selections from the Dresser Clark product lines were made on the basis of targets shown in Table 4-1 of Section 4 and a nominal final delivery pressure of 1200 psia.

6.1.1 Axial Compressors - After many iterations with various compressor bodies, selections for the first-stage machinery were made which gave discharge conditions of 870°F and 226 psia. Analysis indicated that the best choice for compression from ambient pressure (first stage) was to use two sixteen-stage axial machines in parallel (AGr 11/16) followed by a third axial machine with fourteen stages. Other selections resulted in lower unit efficiencies or discharge conditions outside the desired range.

The third axial body (AGr 11/14) is based upon a standard machine, with a strengthened casing and modified shaft seals. This machine, which would operate between 494°F and the 870°F axial section discharge temperature, is an existing Dresser design and would tentatively be heat traced for preheating to a nominal 494°F temperature before start-up of the train. This approach eliminates most of the concerns with extreme blade growth, clearances and casing expansion during start-up. The design of this machine would require engineering development for:

(1) Selection of case materials for high temperature and pressure.

(2) Review of stresses on all rotor and stator blading.

(3) Selection of rotor and stator materials for the elevated operating temperatures.

The last stage of all AGr machines is a radial wheel. This design achieves a rapid change in flow direction to the exit plenum, minimizes the bearing span of the rotor (improving stability) and reduces construction costs.

6.1.2 Centrifugal Compressors - Centrifugal units are proposed exclusively for use after the TES intercooling step. Return air from the TES, trim cooled (see Section 6.8) to 236°F and 206 psia, is compressed to 462°F, 453 psia, using a standard line vertically split (barrel) machine. This model 848 B4 machine is a four-stage unit, directly coupled to a 3600 RPM synchronous motor. This body exhausts to a model 747.5 B5, five-stage vertically split compressor operating at 5248 RPM.

The 747.5 B5 machine is driven by a step-up gearbox with a power transfer of 66 MW. The gearbox selection was obtained from Philadelphia Gear Corporation (King of Prussia, Pennsylvania). The unit is a double helical design rated for 90,000 horsepower with an AGMA service factor of 1.5, (suitable for moderate shock loads under continuous operation). Although the design is conceptual, it is based upon a number of operating units produced by Philadelphia which have critical parameters of pitch line velocity and horsepower greater or equal to the unit quoted by Maag Gear for the PEPCO project. Philadelphia has already designed for manufacture a similar gear box rated at 90,000 HP with a speed ratio of 1.2:1. Discussion of the gearbox design with Philadelphia indicates that they are fully confident that the unit is within their capability. This equipment is therefore considered to require detailed design review should preliminary engineering design follow this study and incorporate the machinery selection presented.

The 747.5 B5 machine will also tentatively require heat tracing to maintain a minimum temperature of 300°F or greater before starting. Some engineering design development will be required in the areas of interstage diaphragms, impeller to shaft attachment, and casing materials.

The exclusive selection of vertically split (barrel) construction is believed to be a conservative approach to the centrifugal compressor casing design. This approach is based upon concerns that the temperature and pressure gradients involved would cause problems in sealing bolted flanges. Additionally, horizontally split fabricated casing designs would be more expensive to build. Therefore, bolted flanges were eliminated from consideration.

6.2 TURBINE SYSTEM SELECTION

The turbine selections were developed somewhat differently from those of the compressor section. The approach used reflects the independent nature of the gas expander business in which each machine is virtually a custom design for a specific application. Basic frame sizes (casing and shaft diameters, bearing spans) are used by the designer, but blade height, chord and profile, number of stages, rotor cooling, operating speed and pressure ratio are all treated as design variables.

Dresser Clark's basic design, under license from GHH Sterkrade A.G. of West Germany, is a multiple stage axial flow machine. Some 130 Dresser and GHH units have been installed in applications ranging from refinery tail gas expansion to closed cycle helium gas turbines. Dresser Clark also builds a modified design for fluid catalytic cracker gas expander service, with features specifically designed to accommodate a flow stream laden with abrasive particles.

The custom design philosophy allowed development of a preliminary design which maximized unit efficiency and output, with minimum cost penalty.

6.2.1 High Pressure Turbine - The design approach to the high pressure expansion turbine was based upon a desire to maximize efficiency and power output and minimize susceptibility to particulate wear. Early investigation indicated that the use of relatively low pressure ratios per stage of 1.3 to 1.4:1 was practical, would not result in a need for more machines, and would help minimize blade erosion. Higher pressure ratios per stage and thus fewer blade rows would result in unused shaft length in the frame size needed for the initial design flow rate of 650 lbs/sec. A target efficiency range of 84 to 86 percent was established, and development of a gas path design proceeded.

Analysis of the flow path of blading indicated that the initial design flow rate of 650 lbs/sec selected (to match the compressor flow rate) caused problems. The large power output of the unit required a relatively large diameter rotor shaft for the volumetric flow, with the result that initial stage blade heights were very short and tip losses were high, yielding a calculated machine efficiency in the mid-70s, versus the 85 percent target figure. An alternate approach of increasing flow through the turbine section to 780 lbs/sec allowed increased blade height. This change permits the 85 percent efficiency target to be achieved and was adopted as a design basis.

The high pressure machine selection is an 8-stage Dresser Clark model GTHR 8 (frame 8) unit with blading from the smaller frame 7 machine. The design of the machine casing, which would be fabricated from steel plate, involves a double shell approach. Turbine exhaust gas is used to provide an intermediate casing pressure and thus reduce the design pressure differential and thickness of the casing metal. This approach has been used by Dresser Clark and GHH as standard practice and is used in the Brown Boveri CAES turbine design. More severe pressure conditions have been encountered and solved using spherical casings. The overall external dimensions of the H.P. turbine casing are 10 feet long by 6 feet diameter.

Shaft seals will likely be oil film type buffered by nitrogen. This will prevent fires or explosion which would result from contact between 1200 psi/800°F air and lubricating oil. Labyrinth seals are considered inadequate by themselves.

6.2.2 Low Pressure Turbine - The turbine flow rate of 780 lbs/sec required the use of two machines in parallel, as the final exhaust volume exceeded the capability of the largest frame size available. The GTHR 10 (frame 10) machines selected are the largest available from Dresser and represent the next size beyond the frame 8 high pressure machine. Although the frame 10 is an existing design, none have been built to date. The frame 10 is considered to be a minor scale-up from the frame 8 and no difficulties are anticipated in constructing the larger machines. Casings would be fabricated from steel plate and would be strengthened over the existing design.

Flow volume through these machines, and consequently the relationship of blade heights to shaft diameter, is sufficient to allow a machine efficiency of 88 percent.

6.3 GENERAL COMMENT ON MACHINERY SELECTIONS

Dresser Clark has stated that the machinery described above would be required if commercially available machinery were used, consistent with the scope of this study. However, were an actual order involved, Dresser Clark has suggested that they might prefer to redesign the compressor train, incorporating more appropriate centrifugal compressor rotor construction methods. Machine selections would likely change as well, including incorporation of the parallel bodies into single units and development of a single shaft design, resulting in a cheaper and shorter machinery layout. Dresser feels that the new frame sizes that would be needed are well within their capability. These units are not offered at present only because there are no market applications warranting the design development on their part.

6.4 TURBOMACHINERY ARRANGEMENT

A schematic arrangement of the machinery, along with general performance data, is shown in Figure 6-1 and Tables 6-1 and 6-2. Overall, the arrangement is considered generally representative of an adiabatic CAES machinery train using commercially available equipment. Other manufacturers are likely to have different but similar limitations within their machinery lines.

6.5 COMPRESSOR SYSTEM GENERAL PERFORMANCE

Performance curves based upon design inlet conditions have been prepared for the axial section, centrifugal section, and total train (Figures 6-2 through 6-4). Operating conditions are shown in Figure 6-1 and listed in Tables 6-1 and 6-2. Power required by the compressor system from the motor/generator with 90°F ambient temperatures is 237 MW.

Comparison of Figure 6-4 to the compressor system curves of the oil-fired CAES engineering study performed for PEPCO (Volume IX of the PEPCO report series) suggests that the adiabatic CAES compressor train would have a marginally wider operating range at design conditions than the Sulzer oil-fired CAES (minimum/maximum flows of 63/104 percent versus 68/102 percent). Surge at design flow for the adiabatic compressor train is well above the maximum operating pressure set by the storage compensating system, and compressor train stability range is somewhat greater as a result. Relative compressor train efficiency at minimum compressor flow is 90 percent of train efficiency at design operating conditions.

6.6 GENERAL COMPRESSOR SYSTEM OPERATING CONDITIONS

The first stage axial compressors were designed for a maximum nighttime ambient temperature, based on NOAA weather data, of 90°F. At this temperature, a TES 1 charge temperature of 877°F was predicted at a pressure of 226 psia for the low pressure compressor train.

The second stage compressors were projected to deliver air at 830°F, 1215 psia to TES 2, for cooling to 125°F and storage in the air cavern. The 830°F delivery temperature was the limit of the possible centrifugal compressor selections.

Low pressure air piping was sized to provide an air stream velocity of 125 ft/sec. This velocity produces an estimated pressure loss of about 20 psi between the low and high pressure compressor trains. This included losses in piping, equipment, and TES 1.

With an inlet pressure of 206 psi, and the above TES 2 inlet conditions, a required second stage compressor inlet temperature of 236°F was determined based on Dresser Clark machinery efficiencies. This temperature could be obtained in one of two ways. TES 1 could be operated between 877°F and 236°F, or it could be operated at a higher cold end temperature with a trim cooler providing the further cooling required.

6.7 TURBINE SYSTEM GENERAL PERFORMANCE

Non-dimensional performance curves (Figures 6-5 and 6-6), for the adiabatic CAES frame 8 and 10 machines were prepared along with an overall performance curve (Figure 6-7) for the turbine section based upon the conditions of Tables 6-1 and 6-2. Figure 6-5 presents the effect of operating pressure ratio on adiabatic efficiency, η, mass flow parameter $\dot{m}\sqrt{T}/P$, and power parameter $HP/p\sqrt{T}$ for the high pressure (frame 8) machine. Figure 6-6 presents the same information for the low pressure (frame 10) machines. The turbine section is projected to operate from zero output at approximately 20 percent flow up to rated flow and output with virtually linear characteristics (Figure 6-7).

The effect of temperature fall-off from thermal storage in the later stages of the discharge cycle appears to be relatively small (Figure 6-8). Output at inlet temperatures 100°F below design is estimated to decrease by 16 MW or some 8 percent. Figure 6-8 represents the outlet temperature fluctuation as predicted by the MIT code with thermal breakthrough beginning approximately in the seventh hour of the TES discharge period. With the CEGB computer code, thermal breakthrough was not predicted to occur until the last hour of the TES discharge period.

6.8 GENERAL TURBINE SYSTEM OPERATING CONDITIONS

As discussed in Section 6.2.1, the preliminary selection of equal compressor and turbine flow rates led to poor overall cycle performance because of turbine limitations. Therefore, the expander train was selected to operate with a mass flow rate of 780 lb/sec. Consequently, the charge and discharge times must differ with the result that a generation period of 10 hours would require the charging period to be a minimum of 12 hours. These times would vary somewhat with part load operation.

The plant high pressure air piping was sized to provide an air stream velocity of 100 ft/sec. Calculations of losses from storage to the throttle valve indicate a pressure loss of about 90 psi. Therefore, the high pressure expander turbine operates at an inlet pressure of 1125 psi.

Initially, an 825°F inlet temperature to the high pressure expander was selected based on a previous high pressure compressor delivery temperature of 850°F and a TES 2 end temperature difference of 25°F. Subsequent analysis indicated that the high pressure centrifugal compressors were only capable of 830°F delivery temperature to TES 2 during compression. Consequently, the average inlet condition to the high pressure expander turbines during generation was reduced to 805°F. This temperature reduction did not change expander selection.

The exhaust pressure for the high pressure expander was selected at 241 psi to approximately match pressure splits between compressors and expanders. This pressure, with the appropriate losses in the piping, valves and TES 1, will provide an inlet pressure to the low pressure expanders of 225 psi. Thus the H.P. turbine exhaust pressure not only falls within the optimum range, but also matches the compressor operating pressures to minimize design pressure differences for common low pressure piping.

With inlet pressure and temperature of 1125 psi and 825°F respectively, and the outlet pressure of 241 psi, the exhaust temperature of the high pressure expander (dictated by efficiency) will be 454°F. With this exhaust temperature, the cold end temperature of TES 1 can be chosen. TES 1 can either cool the air stream, during compression, to 236°F for input to the high pressure centrifugal compressors or TES 1 can be operated with a cold end temperature of 454°F with the air further cooled by a trim cooler to 236°F. If the cold end temperature of TES 1 is set at 236°F, then the high pressure expander exhaust will have to be cooled by a trim cooler to maintain TES 1 stability. Therefore, in either case, a trim cooler is required for TES and cycle stability. The trim cooler is located between TES 1 and the high pressure compressor. Thus, it is utilized during the compression cycle as this reduces the temperature swing of TES 1. Location in the compressor section also provides a slight improvement in EER over the case with cooler location in the turbine section. The maximum cooling load of the trim cooler is approximately 505 x 10^6 Btu/hr.

Table 6-1

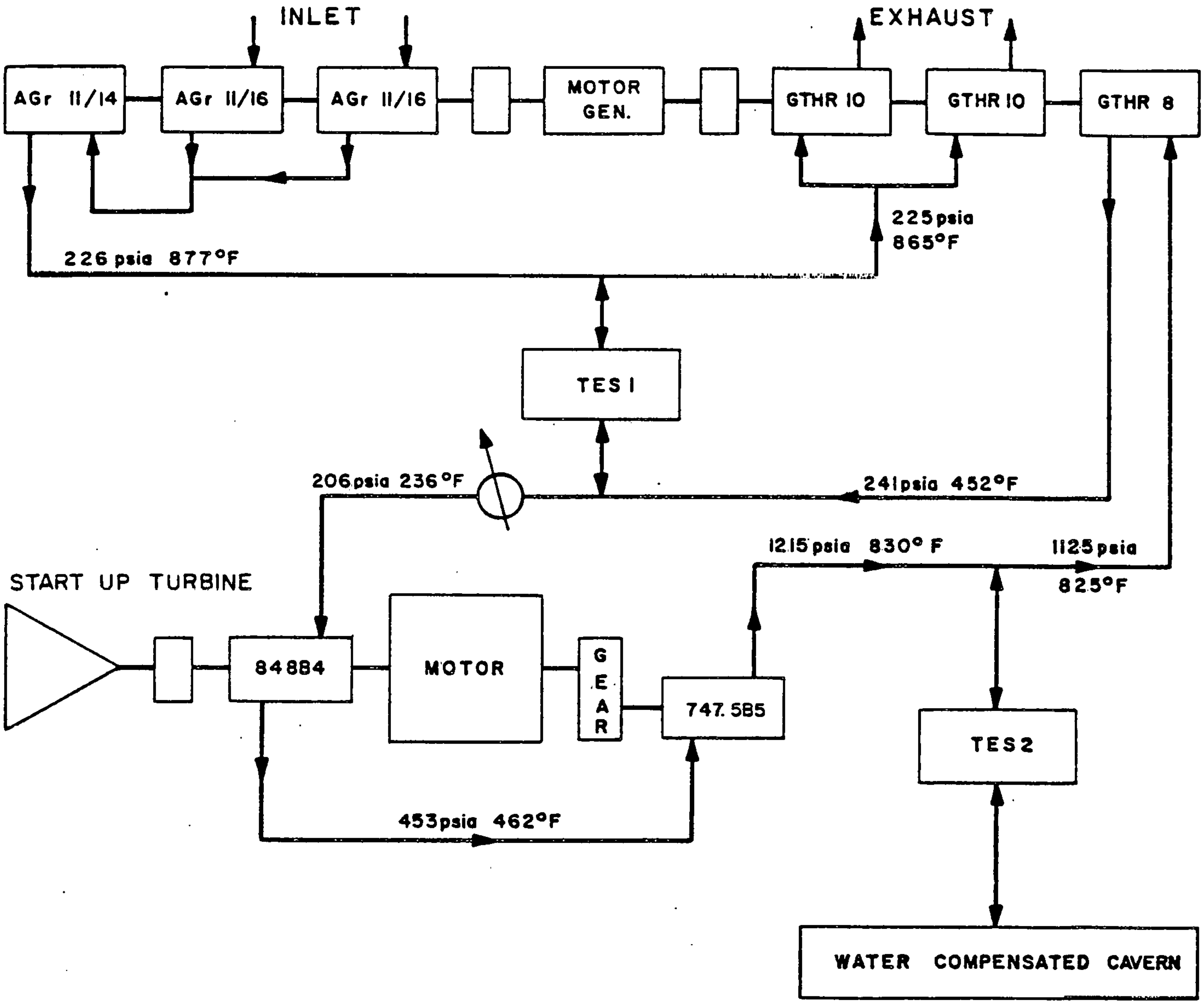
Axial Train

	Compressors		Turbines	
	AGr 11/14	AGr 11/16	GTHR 10	GTHR 8
Wt. Flow #/Sec.	650	325	390	780
P_{in}, psia	79.4	14.5	225	1125
T_{in}, °F	494	90	865	825
P_{out}, psia	226	79.6	15	241
T_{out}, °F	877	494	239	452
K Average	1.3844	1.4044	--	--
Z Average	1.0035	1.002	--	--
Adiabatic Eff. %	87.7	86	88	85
BHP	86,125	45,289	87,000	106,000
Total BHP	--	176,703±4%	280,000	--
MW (power)	--	131.8	208.9	--
RPM	--	3600	3600	--

Table 6-2

Centrifugal Train

	848 B4		747.5 B5
Wt. Flow #/Sec. (Wet)	650		650
P_{in}, psia	206		452
T_{in}, °F	236		462
P_{out}, psia	453		1215
T_{out}, °F	462		830
K Average	1.394		1.377
Z Average	1.010		1.026
Polytropic Eff. %	79.8		80.9
RPM	3600		5248
BHP	51,850		88,790
Total BHP		140,615±4%	
MW (power)		104.9	



COMPRESSORS

MOLECULAR WEIGHT = 28.446
RH = 100%

STARTUP TURBINE

T_{in} = 825°F
P_{in} = 1125 psia
P_{ex} = ATM.
RATE = 15.8 #/HP/HR

MACHINE ARRANGEMENT

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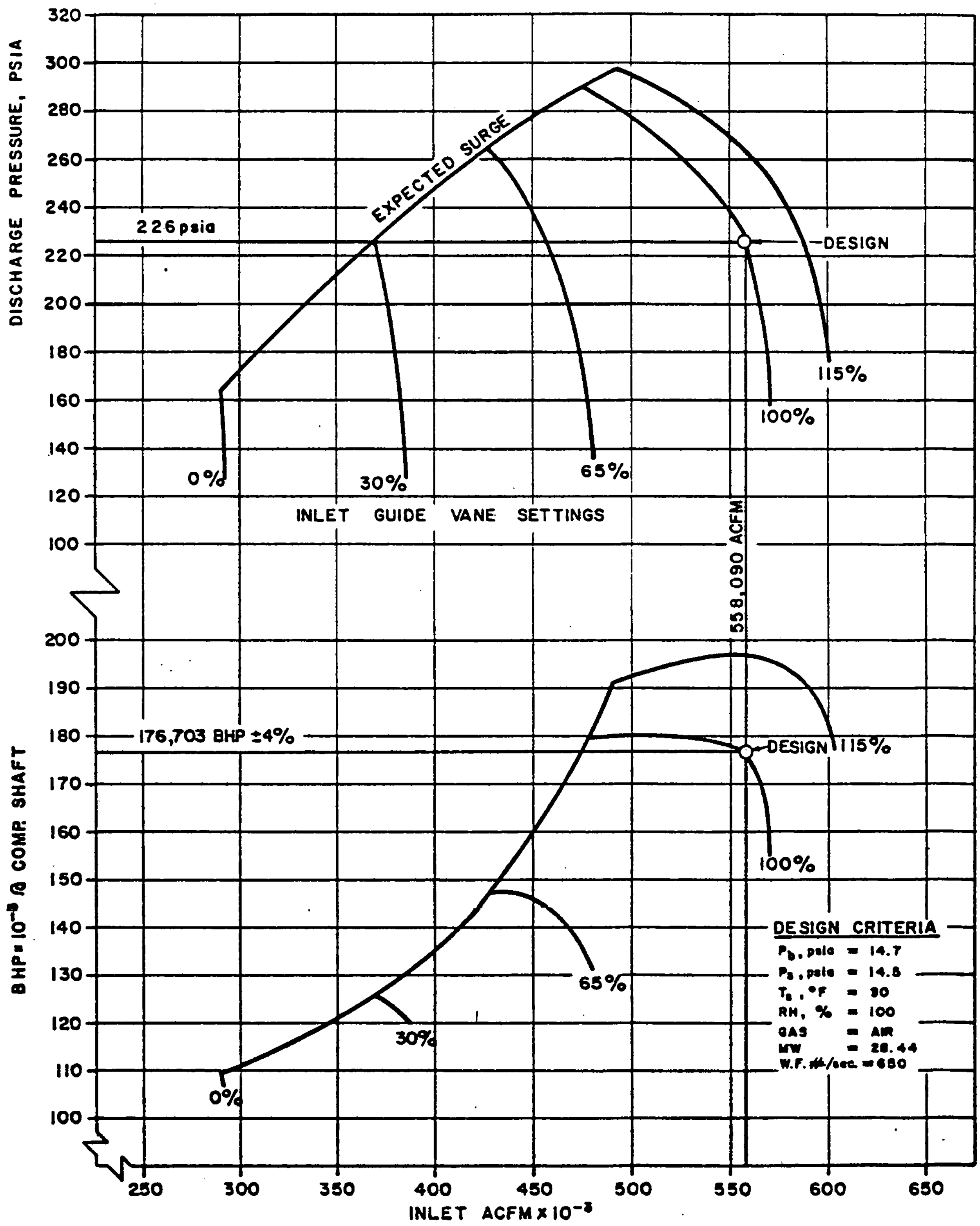
FIGURE 6-1



PERFORMANCE CURVE FOR
AXIAL COMPRESSOR TRAIN
TWO AGr 11/16 AND ONE AGr 11/14

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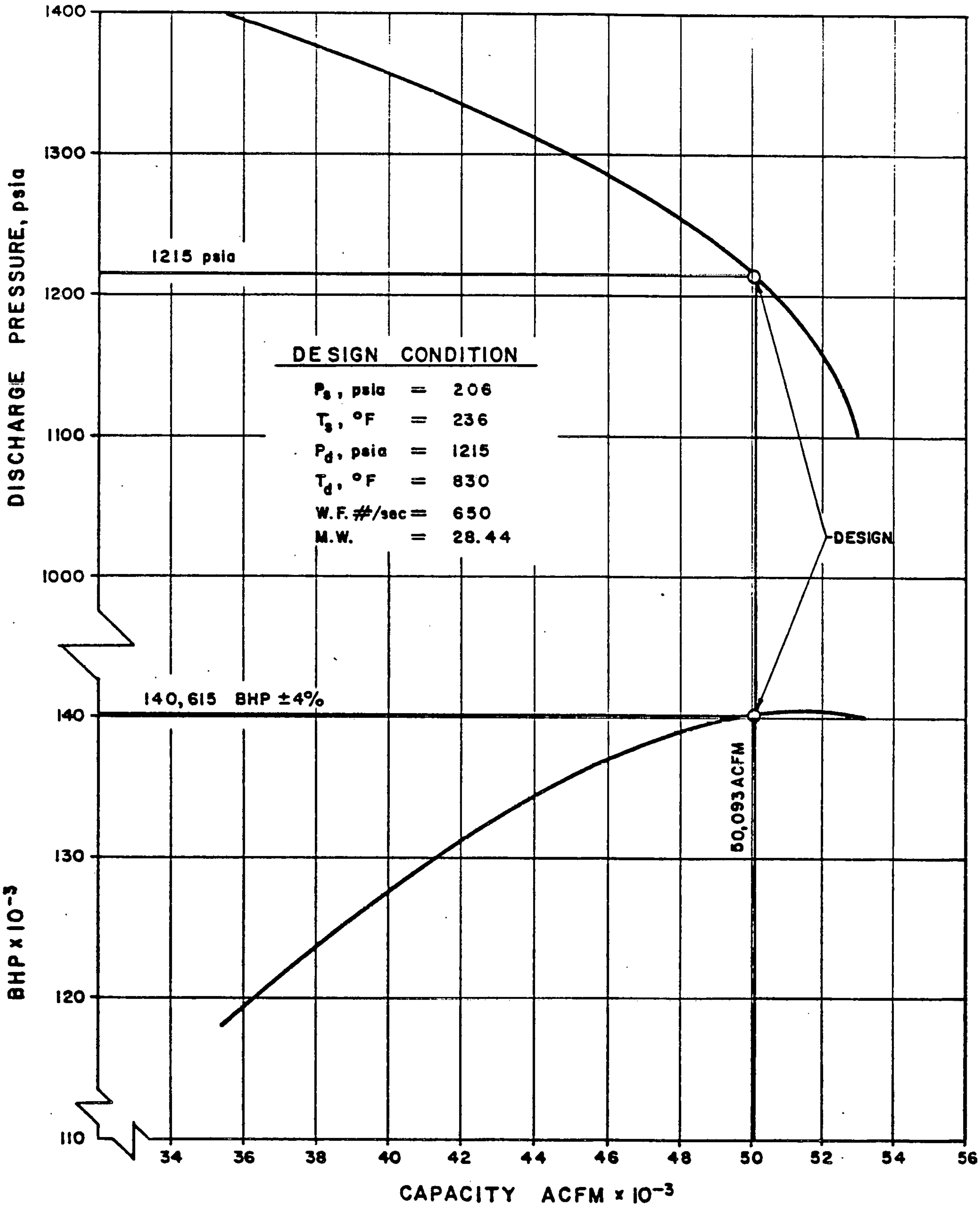
	CONCEPTUAL DESIGN AND ENGINEERING STUDIES—ADIABATIC CAES
M J Hobson ACRES AMERICAN INCORPORATED	FIGURE 6-2



PERFORMANCE CURVE FOR
CENTRIFUGAL COMPRESSOR TRAIN
DRESSER CLARK 848B AND 7475B

ACRES

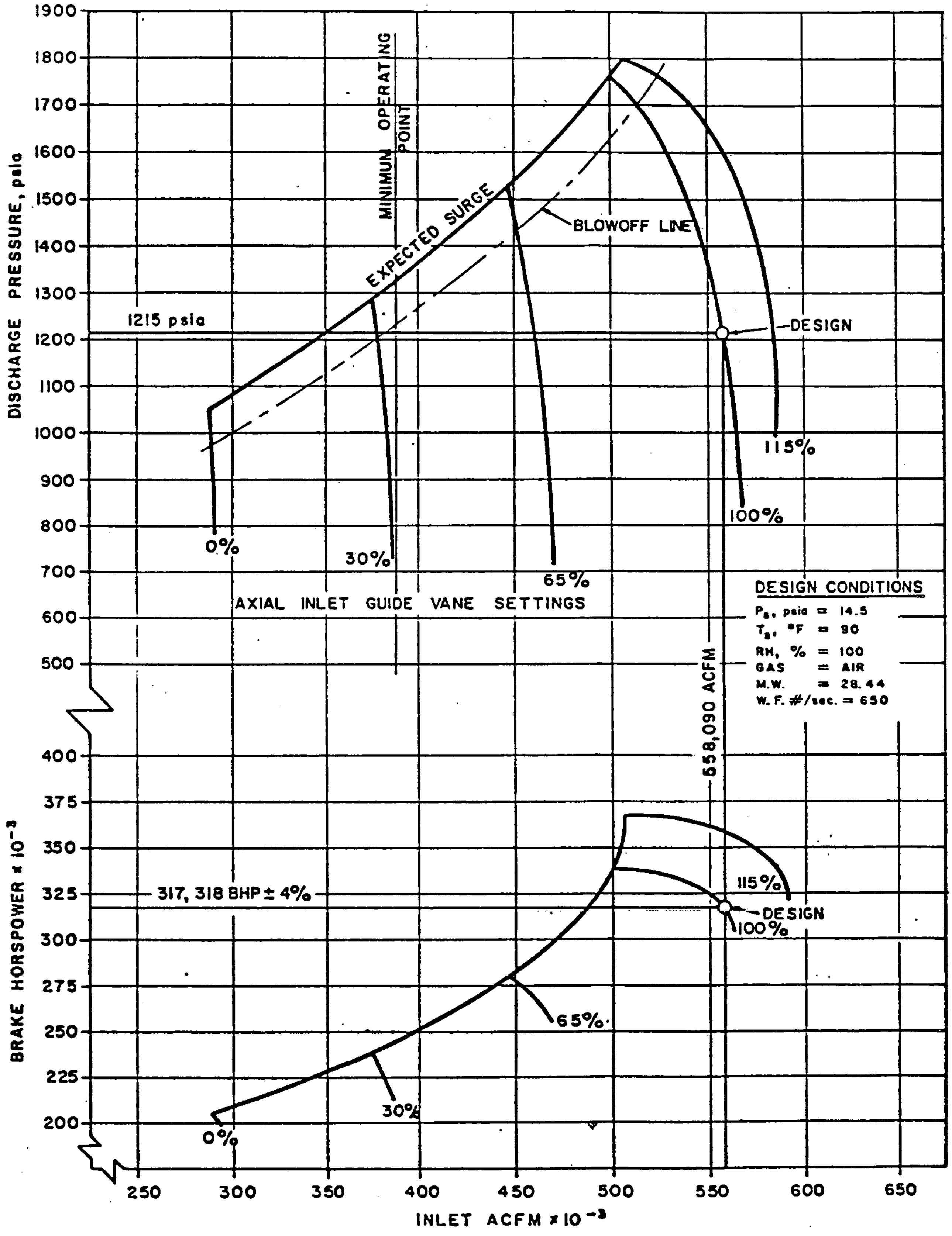
BATTELLE PACIFIC NORTHWEST LABORATORY

CONCEPTUAL DESIGN AND
ENGINEERING STUDIES—ADIABATIC CAES

M J Hobson

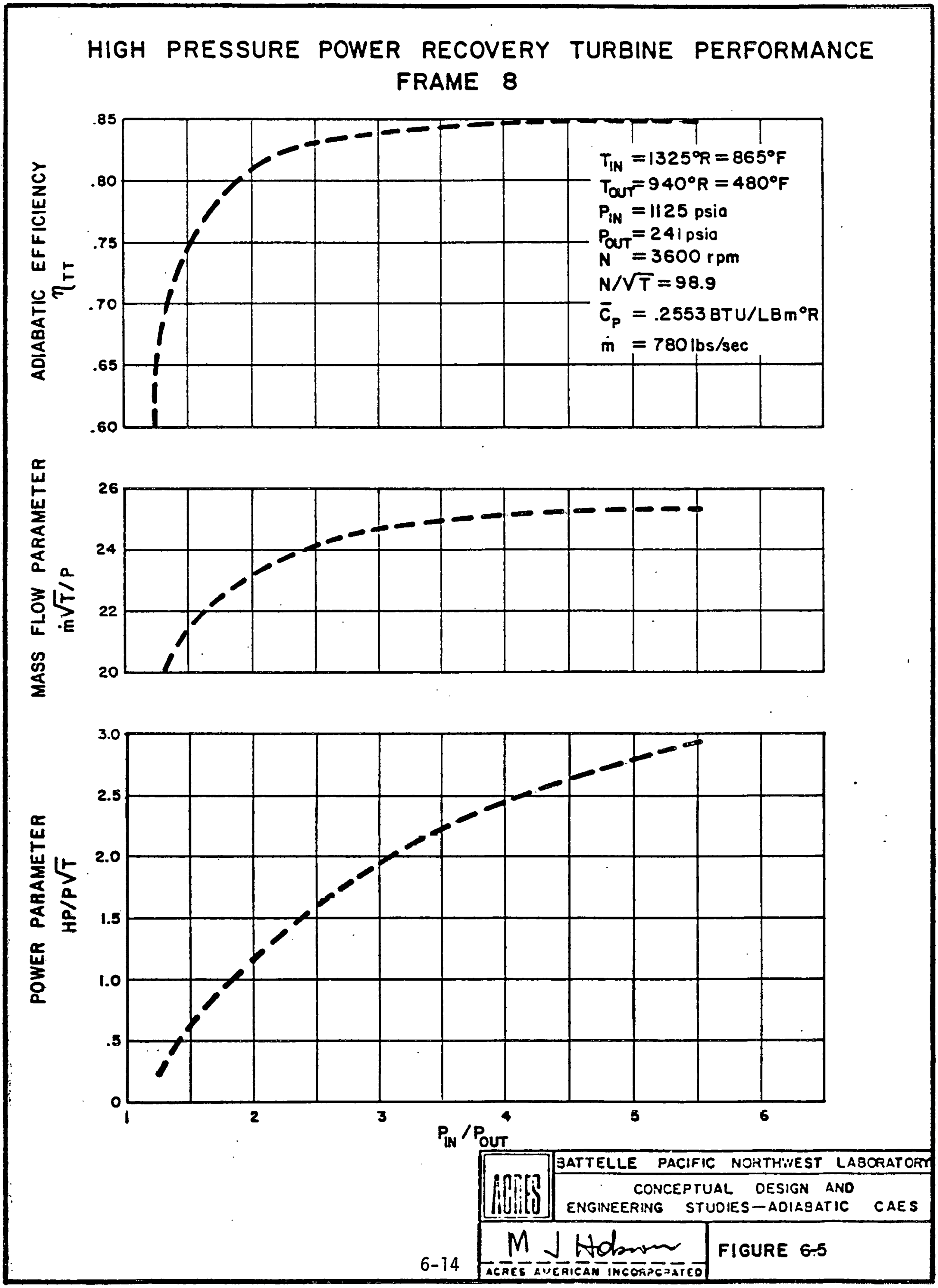
ACRES AMERICAN INCORPORATED

FIGURE 6-3

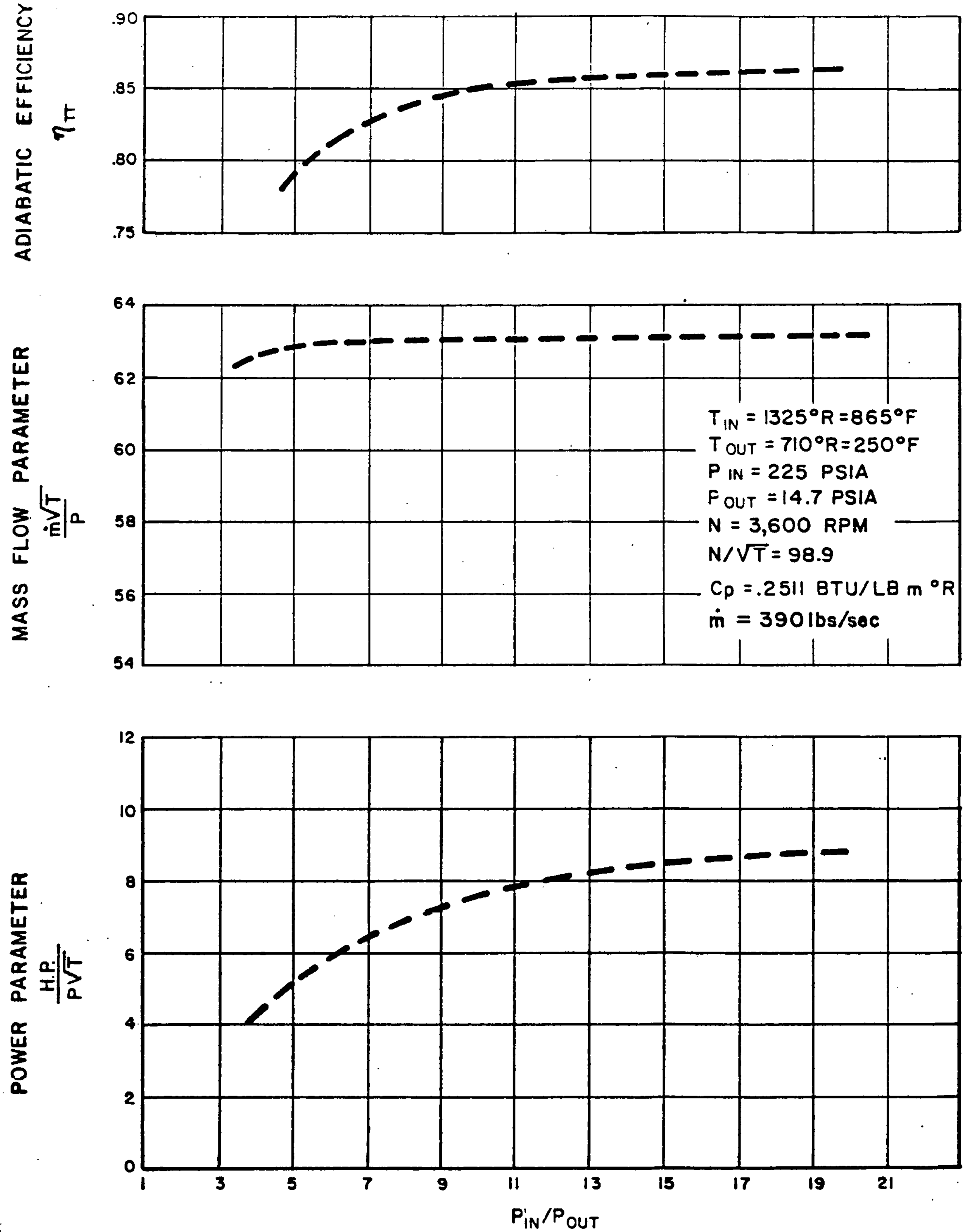


OVERALL COMPRESSOR SYSTEM PERFORMANCE CURVE

ACRES	BATTELLE PACIFIC NORTHWEST LABORATORY CONCEPTUAL DESIGN AND ENGINEERING STUDIES—ADIABATIC CAES
ACRES AMERICAN INCORPORATED	FIGURE 6-4

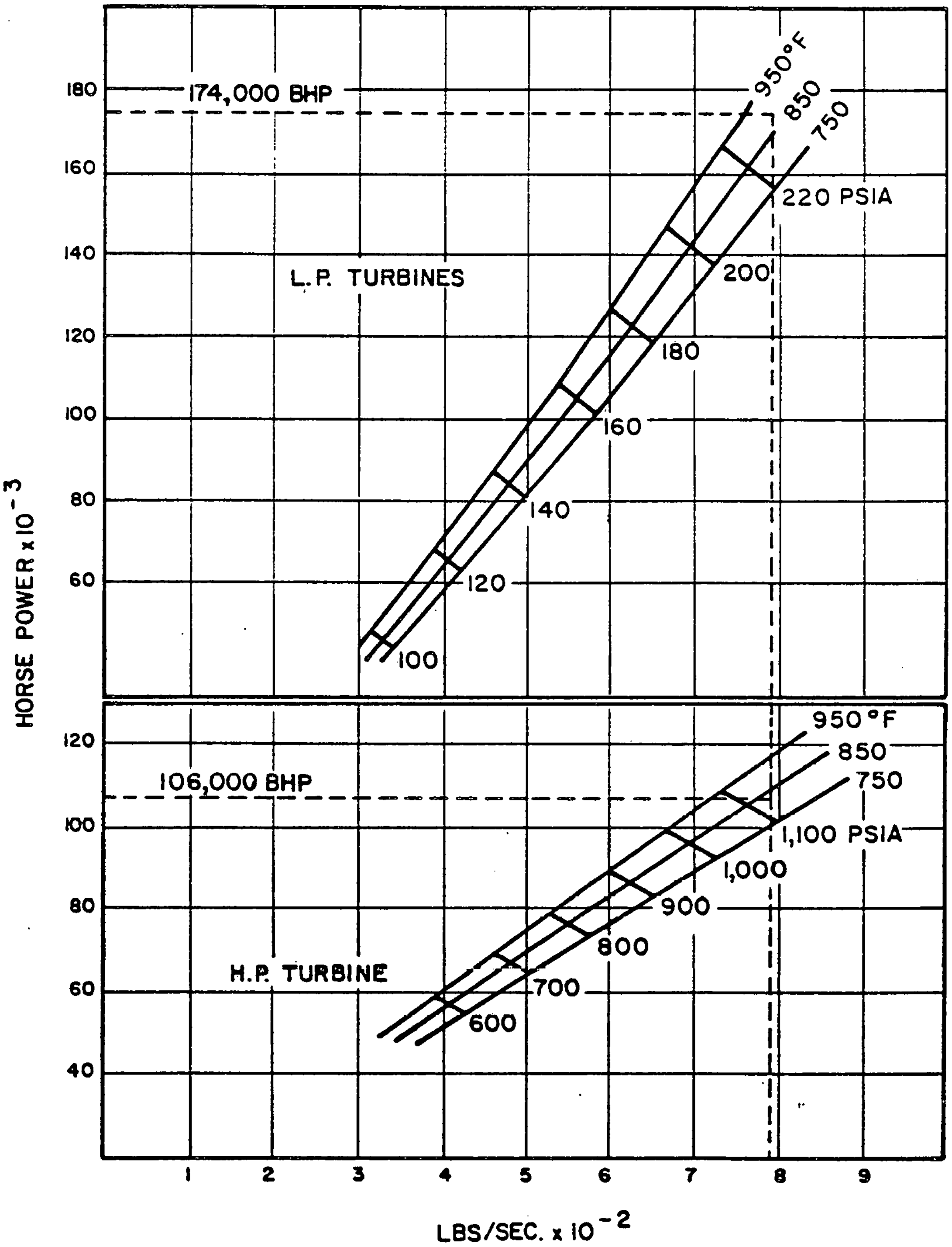

HIGH PRESSURE POWER RECOVERY TURBINE PERFORMANCE
FRAME 8
ADIABATIC EFFICIENCY η_{TT}
.85
.80
.75
.70
.65
.60
T_{IN} = 1325°R = 865°F
T_{OUT} = 940°R = 480°F
P_{IN} = 1125 psia
P_{OUT} = 241 psia
N = 3600 rpm
$N/\sqrt{T}$ = 98.9
$\bar{C}_P$ = .2553 BTU/LBm°R
$\dot{m}$ = 780 lbs/sec
MASS FLOW PARAMETER $\dot{m}\sqrt{T}/P$
26
24
22
20
POWER PARAMETER $HP/P\sqrt{T}$
3.0
2.5
2.0
1.5
1.0
.5
0
1
2
3
4
5
6
P_{IN}/P_{OUT}
BATTELLE PACIFIC NORTHWEST LABORATORY
ACRES
CONCEPTUAL DESIGN AND ENGINEERING STUDIES—ADIABATIC CAES
M J Hobson
ACRES AMERICAN INCORPORATED
FIGURE 6-5

LOW PRESSURE POWER RECOVERY TURBINE PERFORMANCE FRAME 10



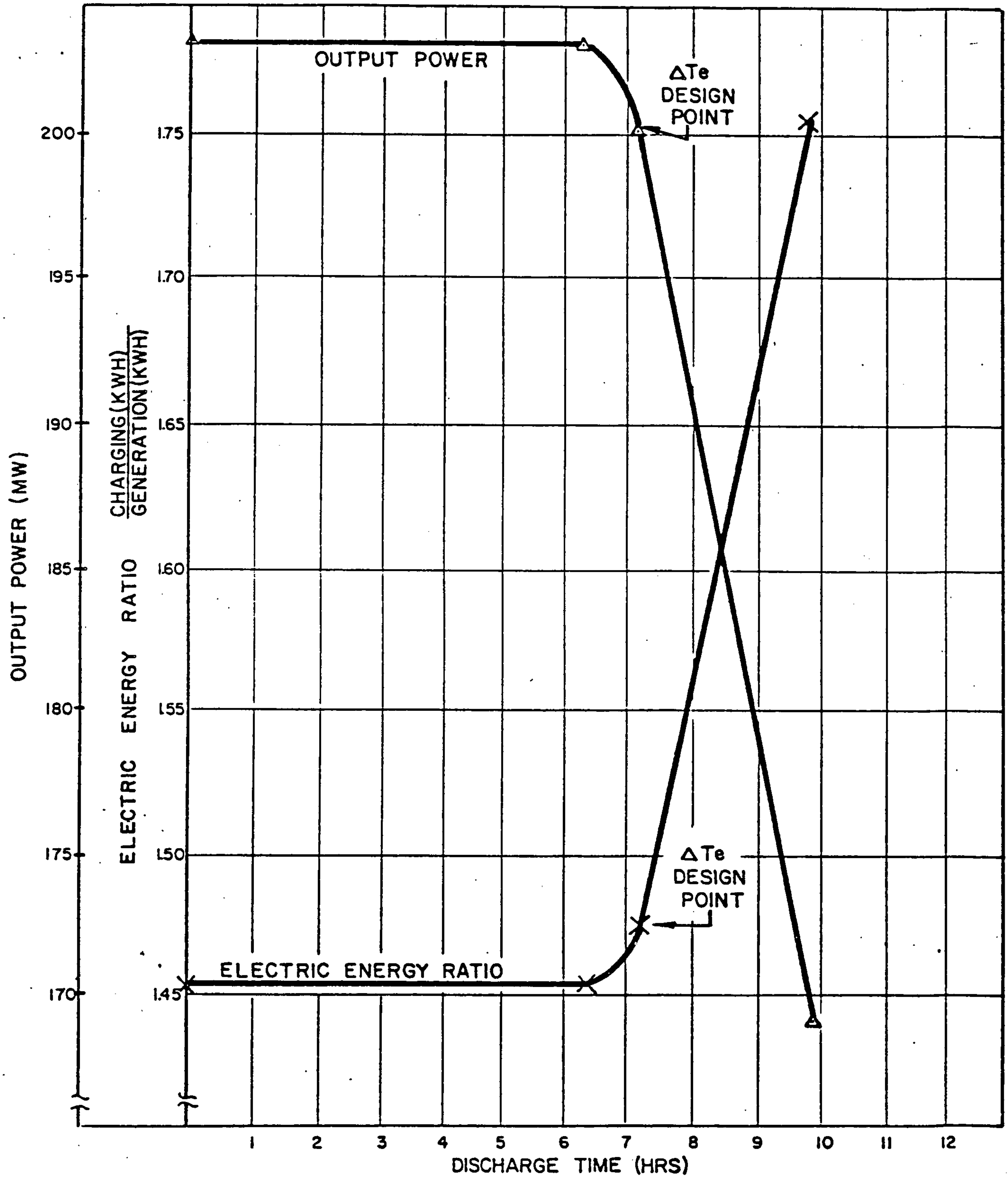
ACRES	BATTELLE PACIFIC NORTHWEST LABORATORY
	CONCEPTUAL DESIGN AND ENGINEERING STUDIES—ADIABATIC CAES
M J Hobin ACRES AMERICAN INCORPORATED	FIGURE 6-6

PART LOAD TURBINE SYSTEM OUTPUT



ACRES	BATTELLE PACIFIC NORTHWEST LABORATORY
	CONCEPTUAL DESIGN AND ENGINEERING STUDIES—ADIABATIC CAES
M J Hdahan ACRES AMERICAN INCORPORATED	FIGURE 6-7

FULL LOAD PERFORMANCE OVER THE DISCHARGE PERIOD



ACRES	BATTELLE PACIFIC NORTHWEST LABORATORY
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ACRES AMERICAN INCORPORATED	FIGURE 6-8

7 - PLANT PERFORMANCE, OPERATION AND CONTROL

7 - PLANT PERFORMANCE, OPERATION, AND CONTROL

This section presents the results of the plant performance evaluation, projected operating procedures, and a general control system concept.

7.1 RATED CONDITIONS

The projected operating conditions of the adiabatic CAES machinery differ slightly from the conditions presented in Section 4. Differences between original and updated rated conditions were caused by such changes as: the addition of intake air preheaters, different TES 1 terminal temperatures, lower high pressure compressor delivery temperature, incorporation of mechanical and electrical losses, and efficiency corrections for air properties at elevated pressure.

7.1.1 Inlet Conditions - Inlet pressure equaled 14.2 psi due to additional losses incurred in the inlet air preheater (See Section 7.3, Off-Design Sensitivity.) Inlet temperature is 90°F.

7.1.2 Efficiency - The process of machine selection involved a preliminary evaluation of gas path losses. The analysis indicated that the efficiency assumptions of the original analysis required slight revisions to represent the final machine selection performance characteristics.

Dresser Clark, as well as many other manufacturers, normally use the properties of air at low pressure in the prediction of machine performance. Consequently, efficiencies based upon low pressure specific heat values are calculated from test stand data.

The Acres developed CAES modeling program incorporates the properties of air, corrected for pressure effects up to 100 atmospheres, according to Vasserman (Vasserman, 1966). Dresser Clark modified their performance prediction routines to incorporate pressure corrected values before final performance estimates were made for the machinery. However, attempts to duplicate the revised Dresser Clark performance predictions with the Acres program revealed discrepancies which are attributed to differences in either air property values for high pressure and temperature or the curve fits used in the two computer code subroutines, or both. Detailed investigation and resolution of the differences were felt to be beyond the scope of the present study. Therefore Acres adjusted input machine efficiencies until the Dresser predictions could be duplicated. A comparison of efficiency values, as well as other efficiency values used in the analysis is presented below to illustrate the differences involved:

- For the compressor train:

The isentropic efficiency differences were based on inlet and outlet conditions of the Dresser Clark design and the Acres computer model.

Machine		Efficiency
AGr 11/16 -	$\eta_{Dresser}$	= 86.0 percent isentropic
	η_{Acres}	= 83.5 percent isentropic
AGr 11/14 -	$\eta_{Dresser}$	= 87.7 percent isentropic
	η_{Acres}	= 81.5 percent
84884 -	$\eta_{Dresser}$	= 79.8 percent polytropic (77.5 percent isentropic)
	η_{Acres}	= 76.5 percent isentropic

- For the expander train:

The isentropic efficiency differences were based on inlet and outlet conditions of the Dresser Clark design and the Acres computer model.

GTHR8 -	$\eta_{Dresser}$	= 85 percent
	η_{Acres}	= 85 percent
GTHR10 -	$\eta_{Dresser}$	= 88 percent
	η_{Acres}	= 90 percent

- Mechanical losses are estimated to be 0.9 percent. Electrical losses (based on the PEPCO study) were estimated at 1.6 percent.

This problem is believed to be the direct result of limited availability of and scatter in data for high pressure/high temperature air properties. Research into high temperature and high pressure thermophysical properties for air would be beneficial for future CAES work.

7.1.3 Thermal Energy Storge System Properties - TES 1 was modeled with an end temperature difference of 15°F. This was based on the design effectiveness of 0.966 and a temperature swing of 422°F. The pressure loss through the bed during one pass was 5 psi. This includes a 1.0 psi loss in the bed, with entrance and exit losses of 3.0 and 1.0 psi respectively.

TES 2 was operated with an end temperature difference of 25°F. This resulted from a design effectiveness of 0.966 and a temperature swing of 730°F. The once through pressure loss of 4 psi, consisted of 0.5 psi in the bed itself with entrance and exit losses of 2.5 and 0.5 psi respectively.

7.2 PERFORMANCE AT RATED CONDITIONS

The performance of the adiabatic CAES plant was evaluated through the incorporation of machinery results and the efficiency corrections discussed in the previous section. Table 7-1 illustrates the flow (lb/sec), temperature (°F), pressure (psi), and enthalpy (Btu/lb) at the exit of each component for a 4-unit adiabatic CAES plant at design conditions. The computer output summary shows input and output power of the compressor and expander trains, as well as electric energy ratio. State points correspond to those shown in Figure 7-1. As mentioned previously, EER is the ratio of electric charging energy to electric energy out and is the reciprocal of storage cycle efficiency for the adiabatic CAES cycle. The "roundtrip" heat rate (Btu/kWh) is an indication of total efficiency of the utility base load plant-storage plant pair for comparison to heat rates of other peaking plant types.

With an electric energy ratio of 1.477, the equivalent design point storage plant efficiency is 68 percent. Round trip heat rate, using grid supplied charging power at 10,000 Btu/kWh is therefore 14,770 Btu/kWh. This is equivalent to a combined generation cycle thermodynamic efficiency of 23 percent.

7.3 OFF-DESIGN SENSITIVITY

While the performance of the cycle at design point is known, it is of interest to evaluate the cycle performance at other than design conditions. The two conditions investigated were:

- TES temperature variation.
- Ambient temperature variation.

7.3.1 TES Temperature Variation - The discharge temperature of the TES beds during generation is a time dependent function which exhibits a rapid fall-off in the latter hours of the cycle. The graph of temperature behavior is similar to the graph of turbine output power over the cycle period presented in Figure 7-2, as temperature directly affects power output. Output power and electric energy ratio are essentially constant for a major portion (6 hours based upon MIT results, 8+ hours based upon CEGB results) of the generation period in a 10-hour cycle (Table 7-2). Afterwards, declining TES delivery temperatures affect power output, with a reduction to about 88 percent of full power at the fully discharged state (Table 7-3) based upon the MIT results. CEGB performance predictions would involve a greater power output reduction at the end of the cycle.

The temperature fall-off at the end of the TES discharge period, and consequent drop in power output, must occur to ensure thermal stability of the TES system. Failure to cycle the TES system through the final hours of the cycle eventually prevents charging of the system as TES discharge temperature to storage will rise beyond acceptable limits (temperature breakthrough) long before the air storage system is fully charged.

7.3.2 Ambient Temperature Variation - The discharge temperature, flow and pressure of a given compressor train are affected by changes in inlet temperature and pressure. As inlet pressure can generally be accommodated by adjustment of axial compressor inlet vane settings with the selected machine train, only temperature effects were examined.

Operation of the compressor train with 45°F and 0°F ambient temperatures produce final discharge and efficiency conditions as follows:

	Axial Discharge Temperature (°F)	Axial Discharge Pressure (psia)	Air Mass Flow (lbs/sec)	Electric Energy Ratio
45 degree F	809.9	254	718	1.533
0 degree F	768.7	306	791	2.072

These results are also presented in Figures 7-3 through 7-5 and Tables 7-4 and 7-5.

Ambient temperature changes will not affect either the high pressure machinery or TES 2 because the inlet temperature to the high pressure compressors can be controlled by the trim cooler. Hence, the input temperature to the high pressure expanders during generation is fixed. However, lower ambient temperatures would involve increased mass flow through the axial machinery. This would increase compressor input power substantially for the 0°F condition. These volumes, when compressed, would be delivered at a higher axial section discharge pressure as a result of centrifugal section flow resistance characteristics. Centrifugal section discharge pressure would be unchanged. Thus, operation of the axial compressor train at low inlet temperature conditions results in discharge pressures to the TES 1 piping which, at some 300 psia, are well above design pressure.

These results suggested that inlet air preheating must be incorporated into the adiabatic cycle to maintain inlet temperature at 90°F and operating pressures within system design levels. This can be accomplished by incorporating part of the trim cooler heat exchange surface in the intake duct to the axial compressor, providing regenerative preheating.

7.4 PART-LOAD PERFORMANCE

The adiabatic CAES plant, in concept, is designed for utility load following and leveling. This type of duty will require part-load operation of the plant. Assuming fully charged thermal and air storage systems, the output power from the expander train can be regulated by throttling the mass flow. Figure 7-6 shows the relationship between mass flow and output power of the expander train with a comparison between performance predictions from Dresser Clark and the adjusted conditions shown in Tables 7-6 and 7-7.

In the event that full power is not available from the utility transmission system during compression, the compressor train will also require throttling or be unable to operate. Figure 7-7 shows that air flow can be modulated to 67 percent of full load flow while maintaining required discharge pressure and avoiding surge. Operating efficiency at minimum flow drops to 91 percent of design point efficiency.

7.5 GENERAL PERFORMANCE COMPARISON

System performance over the full cycle is represented by the design point result (Table 7-1), as this represents a time weighted average operating condition. Thus, estimated average electric energy ratio is 1.477, equivalent to a storage cycle efficiency of 68 percent at the generator terminals. If a charging plant heat rate is assumed such that a round trip cycle efficiency for fuel-fired CAES can be determined, adiabatic CAES, fuel-fired CAES and UPH cycle efficiency can be compared. Based upon a charging plant heat rate of 10,000 Btu/kWh, the following results:

	Round Trip Heat Rate, Btu/kWh	Total Cycle Efficiency
• Adiabatic CAES	14,800	23%
• Underground Pumped Hydro	13,200	26%
• Fuel-Fired CAES	11,600	30%

The effect of the low temperature restrictions of the near-term cycle are clearly shown in the above results. Although improvement of cycle efficiency can be realized through reduction of component losses or modest increases in operating temperatures, a major increase in efficiency (i.e., from 68 to 78 percent) should not be expected.

7.6 CONTROLS CONCEPT

The control concept for the compressor/expander equipment trains is an extension of that employed on equipment with fewer components but no less complex from the operating point of view. Controls must function to provide for starting, operating, and stopping the equipment in a safe, reliable manner. The controls must monitor the rotating and accessory equipment, process equipment, process valves, and operating parameters to provide the operator and automatic control equipment the information essential for the proper modulation of sequential control functions. Detailed controls design would include consideration of the requirements peculiar to each component and the integration of these requirements into an overall control scheme. This section presents only an overview of the required controls and methods of operation.

7.6.1 General - Drawing Figure 7-8 is a block diagram of the major equipment components and functional valve requirements. Approximate design pressure, flow, and temperature for each component is noted. This diagram is simplified for clarity and provides a basis for the following description. Anti-surge controls are not shown, nor is attention focused on the motor, motor/generator or clutch controls.

7.6.2 System Operation - Generation Mode - For this description, the assumption is made that the air storage cavern is pressurized, and the thermal energy storage systems are charged.

7.6.2.1 Component Condition or Position - Pre-Start

(1) Auxiliary systems operating (lube oil) and all monitoring systems in order.

(2) Turning gear (TG-2) operating

(3) Clutch 2 engaged

(4) Clutch 1 disengaged

(5) MV-1 on autospeed control

(6) Block valves

a) BV-1 closed
b) BV-2 closed
c) BV-3 open
d) BV-4 closed
e) BV-5 closed
f) BV-8 closed
g) BO-1, BO-2, BO-3 open
h) BO-4 open

7.6.2.2 Start Sequence

(1) Close BO-4, open BV-2

(2) Open BV-1

(3) Right portion of train turns (driven by GTHR 8, 10&10)-TG-2 disengages and is de-energized.

(4) Unit accelerates to synchronous speed (3600 rpm)

(5) MV-1 modulates expander flow to regulate speed.

(6) Generator switch gear, excitation, etc. in order and synchronizer closes breaker to the line.

Start-up of the turbine system is expected to be relatively simple within a short period after compressor operation. Air at elevated temperature will be available within seconds of start-up. However, if the air in the piping has cooled significantly, turbine exhaust temperature and possibly ice formation will likely limit loading of the turbine for the time required to transport fully heated air from the TES caverns to the turbine, estimated at some 15 minutes at full flow rate, for a single turbine unit. Acceptance of full load should be very rapid after fully heated air becomes available, as thermal gradients in the machinery should be relatively low.

7.6.2.3 Operation - MV-1 is opened to accommodate load requirements according to electrical output instrumentation.

During operation, the process and equipment items and parameters are monitored to sense malfunctions, such as loss of lube pressure, high vibrations, valves out of position, high temperatures, output current reversal, etc. Dependent on the malfunction or out of limit parameter, an alarm or alarm and shutdown would be automatically initiated.

7.6.2.4 Shutdown - Normal

(1) Decrease output to minimum by means of MV-1.

(2) Open generator output breaker.

(3) Cool down as required.

(4) Close BV-1.

(5) After coast down close BV-2 and BV-3.

(6) Energize turning gear TG-2.

7.6.2.5 Shutdown - Abnormal - An abnormal shutdown would be one initiated by a monitoring system sensing a situation which would cause extensive equipment damage if operation were sustained. One example would be loss of lubricating oil. In such an event, an abnormal shutdown in the most expeditious manner takes place. This consists of closing the expander control and block valves (BV-1, BV-3, MV-1 and valves not shown for the low pressure expanders - GTHR-10). The generator breaker should remain closed so that the generator will act as a brake. Upon sensing a current reversal signifying a loss of driving torque, the generator breaker should be opened.

An electrical fault causing opening of the generator breaker is an abnormal condition requiring immediate removal of the driving torque to prevent excessive overspeed. Valve closure would be similar to the case above. However, a comprehensive analysis of the system dynamics would be required to determine the sequence of actions required either to keep the unit operating or to shut down the unit without overspeed. Such an analysis would enable final piping and valving design.

7.6.3 System Operation - Compression Mode - In the compression mode, both compressor trains must operate. The low pressure train is started as in the generation mode by means of the expanders GTHR-10, 10 and 8. The start-up concept for the high pressure train is to use the bus connections and excitation of the motors of both trains in a back-to-back runup of the compressor system. This approach is unusual but is feasible. The method has been used in starting pumped storage units and was used for the first startup of the Huntorf CAES plant compression system. Further analysis is required to confirm suitability of this approach for daily use. The alternatives are use of either a static (variable frequency) starting system or an air start-up turbine in the centrifugal train (shown in Figure 7-8).

When the units are up to speed, the drive motors are synchronized and the expanders are disengaged by means of clutch 2. As in the generation mode, all auxiliary equipment must be operating and all start inhibiting circuitry must be in order. An advantage of using the separate start-up turbine arrangement shown is such that either train may be operated independently for test and check out purposes.

7.6.3.1 Pre-Start Condition

(1) Compressor heating system on and maintaining required pre-start temperature

(2) Turning Gears (TG-1, TG-2) operating.

(3) Clutches 1 and 2 engaged.

(4) MV-1 on auto speed control.

(5) Stator vanes on AGr 11/16 compressors in start position.

(6) Block and blow-off valves set.

a) BV-1, BV-2, BV-4, BV-5, BV-8 closed
b) BV-3 open
c) BO-2, BO-3, BO-4 open
d) BO-1 open
e) MV-2 on 3-speed control
f) BV-5 closed

7.6.3.2 <u>Start Sequence</u>

(1) Close BO-4, BV-2 open

(2) BV-1 open

(3) Entire train rotates (driven by GTHR 8, 10 & 10) TG-1 & TG-2 disengage and are de-energized.

(4) Unit accelerates to synchronous speed and motors are synchronized.

(5) Move AGr 11/16 stator vanes to minimum run position.

(6) Close BV-1, BV-2, and BV-3.

(7) Open BO-4.

The low pressure train and high pressure train are now operating at reduced pressure with blow-off to atmosphere. At this time, the pressure in TES 1 is unknown, but depending on volume, etc., should be less than 225 psia and more than 100 psia.

The high pressure train will be operating at or near design pressure ratio but at reduced pressure.

7.6.3.3 <u>Load Sequence</u>

(1) Open BV-8 (CV-2 may or may not open, dependent on pressure in TES 1)

(2) Open BV-4 (CV-3 will close)

(3) Close BO-2 (Ratio across AGR 11/16 compressor rises)

(4) Close BO-3 (Ratio across low pressure train rises)

(5) Open BV-5

(6) Close BO-1 (See note on auto control)

Note: Anti-surge control action on valves is not shown.

When the compressors are operating, automatic anti-surge controls controlling blow-off valve(s) will maintain the compressors in the stable operating area. The above loading sequence illustrates the method of loading without complicating the explanations. At step (6) the valve BO-1 could not be closed for the overall compression ratio would be low due to the minimum run position of the stator vanes in the first stage axial compressors. The compressor discharge pressure capability would be less than the pressure downstream of CB-1 and the units would be dead ended. With the understanding that blow-off in the appropriate stages would be automatically controlled to prevent surge, the loading sequence is continued.

(7) Open the variable stator vanes on the first stage compressors from the minimum run position.

(8) With the opening of the vanes the discharge pressure will rise. When the discharge pressure exceeds that of TES 2, the discharge check valve CV-1 will open and the units will be operating in the compression mode.

7.6.3.4 <u>Operation</u> - With the opening of the stator vanes in the first stage compressors and the discharge check valve (CV-1), the charging phase is operational. The system is discharging into a system which should present essentially a constant resistance due to the compensating reservoir. Volume flow should remain constant and mass flow will vary with ambient temperature. Flow control is permissible within the overall operating limits (i.e., combined compressor and motor capability).

7.6.4 <u>Shutdown - Normal</u> - Abnormal or emergency situations on either compression train will cause a total compressor system shutdown (i.e., both trains).

In the event of a malfunction shutdown, the object is to prevent compressor surge and to coast down in an orderly fashion. To accomplish this, rapid venting of selected sections of the system is essential. Overriding of the automatic anti-surge controls upon receipt of the shutdown signal is normal to effect rapid blowdown. Total system and sectional volumes dictate the valve placement and sizing criteria. Upon completion of blowdown, the entire compression piping system is de-pressurized, with the exception of TES 1 and TES 2, and the storage cavern is blocked in by closing BV-5. Both trains would be placed on turning gear unless the malfunction was loss of lube oil.

7.6.5 <u>Anti-Surge Controls</u> - Axial and centrifugal compressors are subject to a phenomenon known as surge. Surge occurs when the relationship of flow to pressure rise diminishes such that compressor operation enters a region of instability. This region lies above and to the left of the surge line shown on the compressor performance curve of Figure 7-7. Controls are incorporated to sense the approach of surge conditions, and blow-off valves are modulated to reduce the effective discharge resistance of the compressor system. Minimum flow is maintained such that the system operating point remains to the right of the surge line.

Flow is sensed by measuring the inlet pressure drop in the axial compressors and inserting a flow element in the suction piping of the centrifugal compressors. Pressure rise transmitters are used to measure the pressure rise across the machines. Anti-surge controllers compute operating relationships and modulate the blow-off valves, some of which are not shown.

7.6.6 Protective Instrumentation - Instrumentation is installed on each component to sense lube oil pressures, bearing temperatures, vibration, and discharge temperatures. Alarms indicate approach to out of limit operating conditions, and gross deviations cause shutdown when machine integrity is threatened. Operating conditions would be indicated on an overall control panel to permit monitoring of machine and total system performance. Provisions for interlock and emergency shutdown caused by cavern water level alarms and other plant conditions would also be incorporated in the control system.

7.6.7 Valves - Valves will be hydraulically operated for speed of operation where required.

7.6.8 Initial Plant Startup - Initial start-up presents some difficulties which must be addressed in the preliminary engineering phase. These stem from the fact that when the plant is first completed, there is no supply of stored air in the underground caverns which can be used to operate the turbines. The design of the oil-fired system conceivably allowed development of a small air cushion through careful filling of the lower reservoir with compensating water. This approach presented some risk and contributed to the decision to incorporate a static (variable frequency) starting system into the oil-fired plant design.

The development of a "starting cushion" with the adiabatic system is not possible, as it would involve partial flooding of the high pressure TES cavern. Potential options are charging of the cavern using mobile rental high pressure compressors, installation of the static starting system or use of an isolated transmission back-to-back start as employed at Huntorf.

7.6.9 Additional Considerations - The degree of automation in the operation of compression or generating equipment may be small or large. As stated, the controls must function to provide for starting, operation, and shutdown in a safe reliable manner. As a minimum, the controls must be automatic to protect the equipment where rapid action is required and operator response cannot be assured. Such is the case when equipment malfunction necessitates an abnormal shutdown. The start-stop operation may be semi-automatic or fully automatic. In either event, the sequence must be monitored to inhibit the next step if a previous one has not been satisfied. Thus, operator actions will be inhibited if applied to an inappropriate time in the sequence. Detailed controls design will be a significant effort even though it will be accumulative for each train component and accessory equipment and systems.

Table 7-1

CAES PLANT THERMODYNAMIC ANALYSIS
ACAES STUDY - PNL
ACAES DESIGN POINT

COMPRESSION

COMPONENT NO.	1	2	3	4	5	6	7	8	9	10	11
COMP. NAME	INLET	COMPR	PIPE	COMPR	PIPE	VALVE	PIPE	HEATX	PIPE	VALVE	PIPE
FLOW, LB/S	2600.0	2600.0	2600.0	2600.0	2600.0	2600.0	2600.0	2600.0	2600.0	2600.0	2600.0
TEMP, F	90.0	494.6	494.6	877.9	877.9	877.9	877.9	455.0	455.0	455.0	455.0
PRES, PSIA	14.2	78.0	77.8	221.3	220.7	220.7	221.2	216.2	214.1	214.1	213.7
H, BTU/LB	131.5	229.5	229.8	327.5	327.5	327.5	327.5	219.6	219.6	219.6	219.6

COMPONENT NO.	12	13	14	15	16	17	18	19	20	21
COMP. NAME	PIPE	HEATX	COMPR	PIPE	COMPR	PIPE	VALVE	PIPE	HEATX	PIPE
FLOW, LB/S	2600.0	2600.0	2600.0	2600.0	2600.0	2600.0	2600.0	2600.0	2600.0	2600.0
TEMP, F	455.0	236.0	461.4	461.4	830.0	830.0	830.0	830.0	125.0	125.0
PRES, PSIA	210.2	206.2	453.5	452.5	1216.3	1201.3	1201.2	1201.5	1209.5	1263.0
H, BTU/LB	219.6	165.6	220.9	220.9	315.4	315.4	315.4	315.4	133.7	133.5

GENERATION

COMPONENT NO.	22	23	24	25	26	27	28	29	30	31	32
COMP. NAME	CAVRN	HEATX	PIPE	HEATX	PIPE	VALVE	PIPE	PIPE	VALVE	TURBN	PIPE
FLOW, LB/S	3120.0	3120.0	3120.0	3120.0	3120.0	3120.0	3120.0	3120.0	3120.0	3120.0	3120.0
TEMP, F	125.0	100.0	100.0	805.0	805.0	805.0	805.0	805.0	805.0	437.9	437.9
PRES, PSIA	1263.0	1263.0	1188.4	1184.4	1148.4	1148.3	1125.7	1125.2	1110.6	237.9	237.5
H, BTU/LB	133.5	126.9	127.3	308.8	308.8	308.8	308.9	308.9	308.9	215.3	215.3

COMPONENT NO.	33	34	35	36	37	38	39	40	41
COMP. NAME	VALVE	PIPE	HEATX	PIPE	VALVE	PIPE	PIPE	VALVE	TURBN
FLOW, LB/S	3120.0	3120.0	3120.0	3120.0	3120.0	3120.0	3120.0	3120.0	3120.0
TEMP, F	437.9	437.9	862.0	862.0	862.0	862.0	862.0	860.0	237.6
PRES, PSIA	237.5	238.5	233.5	232.3	232.3	231.5	231.0	229.0	15.2
H, BTU/LB	215.3	215.3	323.3	323.3	323.3	323.3	323.3	323.3	166.9

COMPRESSOR FOWER, MW		MOISTURE REMOVED, LB/S		TURBINE POWER, MW		FUEL CONSUMPTION, LB/S	
1	276.61	1	0.00	1	300.26	1	0.00
2	274.76	2	0.00	2	501.93	2	0.00
3	156.07	3	0.00	3	0.00	3	0.00
4	266.36	4	0.00	4	0.00	4	0.00
TOTAL	973.79	TOTAL	0.00	TOTAL	802.20	TOTAL	0.00

SUMMARY: DAILY STORAGE CYCLE

ELECTRIC ENERGY RATIO (KWHR IN/KWHR OUT) =	1.477
SPECIFIC FUEL CONSUMPTION, BTU/KWHR =	0.0
"ROUNDTRIP" HEAT RATE, BTU/KWHR =	14773.7
HEATING VALUE OF FUEL, BTU/LB =	0.0
AIR STORED, M LB/DAY =	93,600
AIR LEAKAGE, MM LB/DAY =	1.310

COMPRESSION TIME/GENERATION TIME =	1.22
AIR FLOW TO STORAGE CAVERN, LB/S =	2600.0
AIR FLOW FROM STORAGE CAVERN, LB/S =	3120.0
STORAGE PRESSURE, PSIA =	1263.0
COMPRESSION TIME, HRS =	10.00

Table 7-2

CAES PLANT THERMODYNAMIC ANALYSIS
ACAES STUDY - PNL
DISCHARGE BEGINNING

COMPRESSION

COMPONENT NO.	1	2	3	4	5	6	7	8	9	10	11
COMP. NAME	INLET	COMPR	PIPE	COMPR	PIPE	VALVE	PIPE	HEATX	PIPE	VALVE	PIPE
FLOW, LB/S	2600.0	2600.0	2600.0	2600.0	2600.0	2600.0	2600.0	2600.0	2600.0	2600.0	2600.0
TEMP, F	90.0	494.6	494.6	877.9	877.9	877.9	877.9	455.0	455.0	455.0	455.0
PRES, PSIA	14.2	78.0	77.8	221.3	220.7	220.7	221.2	216.2	214.1	214.1	213.7
H, BTU/LB	131.5	229.5	229.8	327.5	327.5	327.5	327.5	219.6	219.6	219.6	219.6

COMPONENT NO.	12	13	14	15	16	17	18	19	20	21
COMP. NAME	PIPE	HEATX	COMPR	PIPE	COMPR	PIPE	VALVE	PIPE	HEATX	PIPE
FLOW, LB/S	2600.0	2600.0	2600.0	2600.0	2600.0	2600.0	2600.0	2600.0	2600.0	2600.0
TEMP, F	455.0	236.0	461.4	461.4	830.0	830.0	830.0	830.0	125.0	125.0
PRES, PSIA	210.2	206.2	453.5	452.5	1216.3	1201.3	1201.2	1213.5	1209.5	1263.0
H, BTU/LB	219.6	165.6	220.9	220.9	315.4	315.4	315.4	315.4	133.7	133.5

GENERATION

COMPONENT NO.	22	23	24	25	26	27	28	29	30	31	32
COMP. NAME	CAVRN	HEATX	PIPE	HEATX	PIPE	VALVE	PIPE	PIPE	VALVE	TURBN	PIPE
FLOW, LB/S	3120.0	3120.0	3120.0	3120.0	3120.0	3120.0	3120.0	3120.0	3120.0	3120.0	3120.0
TEMP, F	125.0	100.0	100.0	830.0	830.0	830.0	830.0	830.0	830.0	456.3	456.3
PRES, PSIA	1263.0	1263.0	1188.4	1184.4	1148.6	1148.4	1125.4	1124.9	1110.1	237.8	237.4
H, BTU/LB	133.5	126.9	127.3	315.4	315.4	315.4	315.4	315.4	315.4	219.9	219.9

COMPONENT NO.	33	34	35	36	37	38	39	40	41
COMP. NAME	VALVE	PIPE	HEATX	PIPE	VALVE	PIPE	PIPE	VALVE	TURBN
FLOW, LB/S	3120.0	3120.0	3120.0	3120.0	3120.0	3120.0	3120.0	3120.0	3120.0
TEMP, F	456.3	456.3	877.0	877.0	877.0	877.0	877.0	877.0	246.1
PRES, PSIA	237.5	238.3	233.3	232.1	232.1	231.3	230.8	228.8	15.2
H, BTU/LB	219.9	219.9	327.2	327.2	327.2	327.2	327.2	327.2	168.9

COMPRESSOR POWER, MW		MOISTURE REMOVED, LB/S		TURBINE POWER, MW		FUEL CONSUMPTION, LB/S	
1	276.61	1	0.00	1	306.45	1	0.00
2	274.76	2	0.00	2	507.83	2	0.00
3	156.07	3	0.00	3	0.00	3	0.00
4	266.36	4	0.00	4	0.00	4	0.00
TOTAL	973.79	TOTAL	0.00	TOTAL	814.28	TOTAL	0.00

SUMMARY: DAILY STORAGE CYCLE

ELECTRIC ENERGY RATIO (KWHR IN/KWHR OUT) =	1.455	COMPRESSION TIME/GENERATION TIME =	1.22
SPECIFIC FUEL CONSUMPTION, BTU/KWHR =	0.0	AIR FLOW TO STORAGE CAVERN, LB/S =	2600.0
"ROUNDTRIP" HEAT RATE, BTU/KWHR =	14554.5	AIR FLOW FROM STORAGE CAVERN, LB/S =	3120.0
HEATING VALUE OF FUEL, BTU/LB =	0.0	STORAGE PRESSURE, PSIA =	1263.0
AIR STORED, M LB/DAY =	93,600	COMPRESSION TIME, HRS =	10.00
AIR LEAKAGE, MM LB/DAY =	1.310		

Table 7-3

CAES PLANT THERMODYNAMIC ANALYSIS
ACAES STUDY - PNL
DISCHARGE ENDING

COMPRESSION

COMPONENT NO.	1	2	3	4	5	6	7	8	9	10	11
COMP. NAME	INLET	COMPR	PIPE	COMPR	PIPE	VALVE	PIPE	HEATX	PIPE	VALVE	PIPE
FLOW, LB/S	2600.0	2600.0	2600.0	2600.0	2600.0	2600.0	2600.0	2600.0	2600.0	2600.0	2600.0
TEMP, F	90.0	494.6	494.6	877.9	877.9	877.9	877.9	455.0	455.0	455.0	455.0
PRES, PSIA	14.2	78.0	77.8	221.3	220.7	220.7	221.2	216.2	214.1	214.1	213.7
H, BTU/LB	131.5	229.8	229.8	327.5	327.5	327.5	327.5	219.6	219.6	219.6	219.6

COMPONENT NO.	12	13	14	15	16	17	18	19	20	21
COMP. NAME	PIPE	HEATX	COMPR	PIPE	COMPR	PIPE	VALVE	PIPE	HEATX	PIPE
FLOW, LB/S	2600.0	2600.0	2600.0	2600.0	2600.0	2600.0	2600.0	2600.0	2600.0	2600.0
TEMP, F	455.0	236.0	461.4	461.4	830.0	830.0	830.0	830.0	125.0	125.0
PRES, PSIA	210.2	206.2	453.5	452.5	1216.3	1201.3	1201.2	1213.5	1209.5	1263.0
H, BTU/LB	219.6	165.6	220.9	220.9	315.4	315.4	315.4	315.4	133.7	133.5

GENERATION

COMPONENT NO.	22	23	24	25	26	27	28	29	30	31	32
COMP. NAME	CAVRN	HEATX	PIPE	HEATX	PIPE	VALVE	PIPE	PIPE	VALVE	TURBN	PIPE
FLOW, LB/S	3120.0	3120.0	3120.0	3120.0	3120.0	3120.0	3120.0	3120.0	3120.0	3120.0	3120.0
TEMP, F	125.0	100.0	100.0	548.0	548.0	548.0	548.0	548.0	548.0	250.7	250.7
PRES, PSIA	1263.0	1263.0	1188.4	1184.4	1145.5	1145.4	1127.4	1126.9	1115.3	238.9	238.6
H, BTU/LB	133.5	126.9	127.3	242.6	242.6	242.6	242.6	242.6	242.6	169.1	169.1

COMPONENT NO.	33	34	35	36	37	38	39	40	41
COMP. NAME	VALVE	PIPE	HEATX	PIPE	VALVE	PIPE	PIPE	VALVE	TURBN
FLOW, LB/S	3120.0	3120.0	3120.0	3120.0	3120.0	3120.0	3120.0	3120.0	3120.0
TEMP, F	250.7	250.7	708.0	708.0	708.0	708.0	708.0	708.0	151.1
PRES, PSIA	238.6	240.2	235.2	233.9	233.9	233.2	232.7	230.7	15.2
H, BTU/LB	169.1	169.1	283.6	283.6	283.6	283.6	283.6	283.6	146.1

COMPRESSOR POWER, MW		MOISTURE REMOVED, LB/S		TURBINE POWER, MW		FUEL CONSUMPTION, LB/S	
1	276.61	1	0.00	1	235.73	1	0.00
2	274.76	2	0.00	2	441.23	2	0.00
3	156.07	3	0.00	3	0.00	3	0.00
4	266.36	4	0.00	4	0.00	4	0.00
TOTAL	973.79	TOTAL	0.00	TOTAL	676.95	TOTAL	0.00

SUMMARY: DAILY STORAGE CYCLE

ELECTRIC ENERGY RATIO (KWHR IN/KWHR OUT) =	1.751	COMPRESSION TIME/GENERATION TIME =	1.22
SPECIFIC FUEL CONSUMPTION, BTU/KWHR =	0.0	AIR FLOW TO STORAGE CAVERN, LB/S =	2600.0
"ROUNDTRIP" HEAT RATE, BTU/KWHR =	17507.0	AIR FLOW FROM STORAGE CAVERN, LB/S =	3120.0
HEATING VALUE OF FUEL, BTU/LB =	0.0	STORAGE PRESSURE, PSIA =	1263.0
AIR STORED, M LB/DAY =	93,600	COMPRESSION TIME, HRS =	10.00
AIR LEAKAGE, MM LB/DAY =	1.310		

Table 7-4

CAES PLANT THERMODYNAMIC ANALYSIS
ACAES STUDY - PNL
75 PERCENT DESIGN FLOW

COMPRESSION

COMPONENT NO.	1	2	3	4	5	6	7	8	9	10	11
COMP. NAME	INLET	COMPR	PIPE	COMPR	PIPE	VALVE	PIPE	HEATX	PIPE	VALVE	PIPE
FLOW, LB/S	2600.0	2600.0	2600.0	2600.0	2600.0	2600.0	2600.0	2600.0	2600.0	2600.0	2600.0
TEMP, F	90.0	494.6	494.6	877.9	877.9	877.9	877.9	455.0	455.0	455.0	455.0
PRES, PSIA	14.2	78.0	77.8	221.3	220.7	220.7	221.2	216.2	214.1	214.1	213.7
H, BTU/LB	131.5	229.8	229.8	327.5	327.5	327.5	327.5	219.6	219.6	219.6	219.6

COMPONENT NO.	12	13	14	15	16	17	18	19	20	21
COMP. NAME	PIPE	HEATX	COMPR	PIPE	COMPR	PIPE	VALVE	PIPE	HEATX	PIPE
FLOW, LB/S	2600.0	2600.0	2600.0	2600.0	2600.0	2600.0	2600.0	2600.0	2600.0	2600.0
TEMP, F	455.0	236.0	461.4	461.4	830.0	830.0	830.0	830.0	125.0	125.0
PRES, PSIA	210.2	206.2	453.5	452.5	1216.3	1201.3	1201.2	1213.5	1209.5	1263.0
H, BTU/LB	219.6	165.6	220.9	220.9	315.4	315.4	315.4	315.4	133.7	133.5

GENERATION

COMPONENT NO.	22	23	24	25	26	27	28	29	30	31	32
COMP. NAME	CAVRN	HEATX	PIPE	HEATX	PIPE	VALVE	PIPE	PIPE	VALVE	TURBN	PIPE
FLOW, LB/S	2340.0	2340.0	2340.0	2340.0	2340.0	2340.0	2340.0	2340.0	2340.0	2340.0	2340.0
TEMP, F	125.0	100.0	100.0	805.0	805.0	805.0	805.0	805.0	805.0	440.4	440.4
PRES, PSIA	1263.0	1263.0	1191.6	1187.6	1157.6	1157.6	1145.0	1144.5	825.0	179.5	179.2
H, BTU/LB	133.5	126.9	127.2	308.8	308.8	308.8	308.9	308.9	309.2	216.1	216.1

COMPONENT NO.	33	34	35	36	37	38	39	40	41
COMP. NAME	VALVE	PIPE	HEATX	PIPE	VALVE	PIPE	PIPE	VALVE	TURBN
FLOW, LB/S	2340.0	2340.0	2340.0	2340.0	2340.0	2340.0	3340.0	2340.0	2340.0
TEMP, F	440.4	440.4	862.0	862.0	862.0	862.0	862.0	862.0	288.7
PRES, PSIA	179.2	179.9	174.9	174.0	174.0	173.4	172.9	170.9	15.2
H, BTU/LB	216.1	216.1	323.3	323.3	323.3	323.3	323.3	323.3	179.3

COMPRESSOR POWER, MW		MOISTURE REMOVED, LB/S		TURBINE POWER, MW		FUEL CONSUMPTION, LB/S	
1	276.61	1	0.00	1	224.09	1	0.00
2	274.76	2	0.00	2	346.51	2	0.00
3	156.07	3	0.00	3	0.00	3	0.00
4	266.35	4	0.00	4	0.00	4	0.00
TOTAL	973.79	TOTAL	0.00	TOTAL	570.60	TOTAL	0.00

SUMMARY: DAILY STORAGE CYCLE

ELECTRIC ENERGY RATIO (KWHR IN/KWHR OUT) =	1.558
SPECIFIC FUEL CONSUMPTION, BTU/KWHR =	0.0
"ROUNDTRIP" HEAT RATE, BTU/KWHR =	15577.7
HEATING VALUE OF FUEL, BTU/LB =	0.0
AIR STORED, M LB/DAY =	93,600
AIR LEAKAGE, MM LB/DAY =	1.310
COMPRESSION TIME/GENERATION TIME =	.91
AIR FLOW TO STORAGE CAVERN, LB/S =	2600.0
AIR FLOW FROM STORAGE CAVERN, LB/S =	2340.0
STORAGE PRESSURE, PSIA =	1263.0
COMPRESSION TIME, HRS =	10.00

Table 7-5

CAES PLANT THERMODYNAMIC ANALYSIS
ACAES STUDY - PNL
50 PERCENT DESIGN FLOW

COMPRESSION

COMPONENT NO.	1	2	3	4	5	6	7	8	9	10	11
COMP. NAME	INLET	COMPR	PIPE	COMPR	PIPE	VALVE	PIPE	HEATX	PIPE	VALVE	PIPE
FLOW, LB/S	2600.0	2600.0	2600.0	2600.0	2600.0	2600.0	2600.0	2600.0	2600.0	2600.0	2600.0
TEMP, F	90.0	494.6	494.6	877.9	877.9	877.9	877.9	455.0	455.0	455.0	455.0
PRES, PSIA	14.2	78.0	77.8	221.3	220.7	220.7	221.2	216.2	214.1	214.1	213.7
H, BTU/LB	131.5	229.8	229.8	327.5	327.5	327.5	327.5	219.6	219.6	219.6	219.6

COMPONENT NO.	12	13	14	15	16	17	18	19	20	21
COMP. NAME	PIPE	HEATX	COMPR	PIPE	COMPR	PIPE	VALVE	PIPE	HEATX	PIPE
FLOW, LB/S	2600.0	2600.0	2600.0	2600.0	2600.0	2600.0	2600.0	2600.0	2600.0	2600.0
TEMP, F	455.0	236.0	461.4	461.4	830.0	830.0	830.0	830.0	125.0	125.0
PRES, PSIA	210.2	206.2	453.5	452.5	1216.3	1201.3	1201.2	1213.5	1209.5	1263.0
H, BTU/LB	219.6	165.6	220.9	220.9	315.4	315.4	315.4	315.4	133.7	133.5

GENERATION

COMPONENT NO.	22	23	24	25	26	27	28	29	30	31	32
COMP. NAME	CAVRN	HEATX	PIPE	HEATX	PIPE	VALVE	PIPE	PIPE	VALVE	TURBN	PIPE
FLOW, LB/S	1560.0	1560.0	1560.0	1560.0	1560.0	1560.0	1560.0	1560.0	1560.0	1560.0	1560.0
TEMP, F	125.0	100.0	100.0	805.0	805.0	805.0	805.0	805.0	805.0	444.0	444.0
PRES, PSIA	1263.0	1263.0	1193.9	1189.9	1164.2	1164.2	1158.6	1158.1	550.0	122.3	122.1
H, BTU/LB	133.5	126.9	127.2	308.8	308.8	308.8	308.8	308.8	308.7	217.1	217.1

COMPONENT NO.	33	34	35	36	37	38	39	40	41
COMP. NAME	VALVE	PIPE	HEATX	PIPE	VALVE	PIPE	PIPE	VALVE	TURBN
FLOW, LB/S	1560.0	1560.0	1560.0	1560.0	1560.0	1560.0	1560.0	1560.0	1560.0
TEMP, F	444.0	444.0	862.0	862.0	862.0	862.0	862.0	862.0	375.4
PRES, PSIA	122.1	122.6	117.6	117.0	117.0	116.6	116.1	114.1	15.2
H, BTU/LB	217.1	217.1	323.2	323.2	323.2	323.2	323.2	323.2	200.5

COMPRESSOR POWER, MW		MOISTURE REMOVED, LB/S		TURBINE POWER, MW		FUEL CONSUMPTION, LB/S	
1	276.61	1	0.00	1	146.93	1	0.00
2	274.76	2	0.00	2	196.93	2	0.00
3	156.07	3	0.00	3	0.00	3	0.00
4	266.36	4	0.00	4	0.00	4	0.00
TOTAL	973.79	TOTAL	0.00	TOTAL	343.86	TOTAL	0.00

SUMMARY: DAILY STORAGE CYCLE

ELECTRIC ENERGY RATIO (KWHR IN/KWHR OUT) =	1.723	COMPRESSION TIME/GENERATION TIME =	.61
SPECIFIC FUEL CONSUMPTION, BTU/KWHR =	0.0	AIR FLOW TO STORAGE CAVERN, LB/S =	2600.0
"ROUNDTRIP" HEAT RATE, BTU/KWHR =	17233.0	AIR FLOW FROM STORAGE CAVERN, LB/S =	1560.0
HEATING VALUE OF FUEL, BTU/LB =	0.0	STORAGE PRESSURE, PSIA =	1263.0
AIR STORED, M LB/DAY =	93,600	COMPRESSION TIME, HRS =	10.00
AIR LEAKAGE, MM LB/DAY =	1.310		

Table 7-6

CAES PLANT THERMODYNAMIC ANALYSIS
ACAES STUDY - PNL
45 DEGREES AMBIENT

COMPRESSION

COMPONENT NO.	1	2	3	4	5	6	7	8	9	10	11
COMP. NAME	INLET	COMPR	PIPE	COMPR	PIPE	VALVE	PIPE	HEATX	PIPE	VALVE	PIPE
FLOW, LB/S	2872.0	2872.0	2872.0	2872.0	2872.0	2872.0	2872.0	2872.0	2872.0	2872.0	2872.0
TEMP, F	45.0	455.2	455.2	809.9	809.9	809.9	809.9	809.9	455.0	455.0	455.0
PRES, PSIA	14.2	91.4	91.2	254.0	226.0	225.3	225.9	225.9	220.9	218.6	218.6
H, BTU/LB	120.8	220.0	220.0	309.8	309.8	309.8	309.8	309.8	219.6	219.6	219.6

COMPONENT NO.	12	13	14	15	16	17	18	19	20	21	22
COMP. NAME	PIPE	HEATX	COMPR	PIPE	COMPR	PIPE	VALVE	PIPE	HEATX	PIPE	PIPE
FLOW, LB/S	2872.0	2872.0	2872.0	2872.0	2872.0	2872.0	2872.0	2872.0	2872.0	2872.0	2872.0
TEMP, F	455.0	455.0	236.0	461.4	461.4	830.1	830.1	830.1	830.1	125.0	125.0
PRES, PSIA	218.2	214.7	206.0	452.9	451.9	1215.0	1196.6	1196.5	1206.6	1202.6	1254.7
H, BTU/LB	219.6	219.6	165.6	220.9	220.9	315.4	315.4	315.4	315.4	133.7	133.5

GENERATION

COMPONENT NO.	23	24	25	26	27	28	29	30	31	32	33
COMP. NAME	CAVRN	HEATX	PIPE	HEATX	PIPE	VALVE	PIPE	PIPE	VALVE	TURBN	PIPE
FLOW, LB/S	3120.0	3120.0	3120.0	3120.0	3120.0	3120.0	3120.0	3120.0	3120.0	3120.0	3120.0
TEMP, F	125.0	100.0	100.0	805.0	805.0	805.0	805.0	805.0	805.0	445.2	445.2
PRES, PSIA	1254.7	1254.7	1180.4	1176.4	1140.5	1140.4	1117.7	1117.2	1102.2	245.1	244.7
H, BTU/LB	133.5	126.9	127.3	308.8	308.9	308.9	308.9	308.9	308.9	217.1	217.1

COMPONENT NO.	34	35	36	37	38	39	40	41	42
COMP. NAME	VALVE	PIPE	HEATX	PIPE	VALVE	PIPE	PIPE	VALVE	TURBN
FLOW, LB/S	3120.0	3120.0	3120.0	3120.0	3120.0	3120.0	3120.0	3120.0	3120.0
TEMP, F	445.2	445.2	795.0	795.0	795.0	795.0	795.0	795.0	215.8
PRES, PSIA	244.6	245.7	240.7	239.5	239.4	238.7	238.2	236.2	15.2
H, BTU/LB	217.1	217.1	306.0	306.0	306.0	306.0	306.0	306.0	161.6

COMPRESSOR POWER, MW		MOISTURE REMOVED, LB/S		TURBINE POWER, MW		FUEL CONSUMPTION, LB/S	
1	308.15	1	0.00	1	294.51	1	0.00
2	279.26	2	0.00	2	463.12	2	0.00
3	172.39	3	0.00	3	0.00	3	0.00
4	294.32	4	0.00	4	0.00	4	0.00
TOTAL	1054.12	TOTAL	0.00	TOTAL	757.63	TOTAL	0.00

SUMMARY: DAILY STORAGE CYCLE

ELECTRIC ENERGY RATIO (KWHR IN/KWHR OUT) =	1.533	COMPRESSION TIME/GENERATION TIME =	1.10
SPECIFIC FUEL CONSUMPTION, BTU/KWHR =	0.0	AIR FLOW TO STORAGE CAVERN, LB/S =	2872.0
"ROUNDTRIP" HEAT RATE, BTU/KWHR =	15329.0	AIR FLOW FROM STORAGE CAVERN, LB/S =	3120.0
HEATING VALUE OF FUEL, BTU/LB =	0.0	STORAGE PRESSURE, PSIA =	1254.7
AIR STORED, M LB/DAY =	******	COMPRESSION TIME, HRS =	10.00
AIR LEAKAGE, MM LB/DAY =	1.447		

Table 7-7

CAES PLANT THERMODYNAMIC ANALYSIS
ACAES STUDY - PNL
0 DEGREES AMBIENT

COMPRESSION

COMPONENT NO.	1	2	3	4	5	6	7	8	9	10	11
COMP. NAME	INLET	COMPR	PIPE	COMPR	PIPE	VALVE	PIPE	HEATX	PIPE	VALVE	PIPE
FLOW, LB/S	3164.0	3164.0	3164.0	3164.0	3164.0	3164.0	3164.0	3164.0	2872.0	2872.0	2872.0
TEMP, F	-.0	412.4	412.4	768.7	768.7	768.7	768.7	768.7	455.0	455.0	455.0
PRES, PSIA	14.2	105.4	105.2	306.0	226.0	225.2	225.2	225.7	220.7	218.3	218.3
H, BTU/LB	0.0	209.3	209.3	299.2	299.2	299.2	299.2	299.2	219.6	219.6	219.6

COMPONENT NO.	12	13	14	15	16	17	18	19	20	21	22
COMP. NAME	PIPE	HEATX	COMPR	PIPE	COMPR	PIPE	VALVE	PIPE	HEATX	PIPE	PIPE
FLOW, LB/S	3164.0	3164.0	3164.0	3164.0	3164.0	3164.0	3164.0	3164.0	3164.0	3164.0	3164.0
TEMP, F	455.0	455.0	236.0	461.4	461.4	830.9	830.9	830.9	830.9	125.0	125.0
PRES, PSIA	217.8	214.3	205.6	452.1	451.1	1215.0	1192.6	1192.5	1200.0	1196.0	1246.4
H, BTU/LB	219.6	219.6	165.6	220.9	221.0	315.6	315.6	315.6	315.6	133.8	133.5

GENERATION

COMPONENT NO.	23	24	25	26	27	28	29	30	31	32	33
COMP. NAME	CAVRN	HEATX	PIPE	HEATX	PIPE	VALVE	PIPE	PIPE	VALVE	TURBN	PIPE
FLOW, LB/S	3120.0	3120.0	3120.0	3120.0	3120.0	3120.0	3120.0	3120.0	3120.0	3120.0	3120.0
TEMP, F	125.0	100.0	100.0	805.0	805.0	805.0	805.0	805.0	805.0	445.2	445.2
PRES, PSIA	1246.4	1246.4	1172.6	1168.6	1132.7	1132.6	1109.7	1109.2	1094.2	243.3	242.9
H, BTU/LB	133.5	127.0	127.3	308.8	308.9	308.9	308.9	308.9	308.9	217.1	217.1

COMPONENT NO.	34	35	36	37	38	39	40	41	42
COMP. NAME	VALVE	PIPE	HEATX	PIPE	VALVE	PIPE	PIPE	VALVE	TURBN
FLOW, LB/S	3120.0	3120.0	3120.0	3120.0	3120.0	3120.0	3120.0	3120.0	3120.0
TEMP, F	445.2	445.2	753.7	753.7	753.7	753.7	753.7	753.7	193.4
PRES, PSIA	242,8	243,8	238.9	237.7	237.6	236.9	236.4	236.4	15.2
H, BTU/LB	217.1	217.1	295.3	295.3	295.3	295.3	295.3	295.3	156.2

COMPRESSOR POWER, MW		MOISTURE REMOVED, LB/S		TURBINE POWER, MW		FUEL CONSUMPTION, LB/S	
1	716.61	1	0.00	1	294.51	1	0.00
2	307.80	2	0.00	2	446.23	2	0.00
3	189.92	3	0.00	3	0.00	3	0.00
4	324.92	4	0.00	4	0.00	4	0.00
TOTAL	1539.25	TOTAL	0.00	TOTAL	740.77	TOTAL	0.00

SUMMARY: DAILY STORAGE CYCLE

ELECTRIC ENERGY RATIO (KWHR IN/KWHR OUT) =	2.078
SPECIFIC FUEL CONSUMPTION, BTU/KWHR =	0.0
"ROUNDTRIP" HEAT RATE, BTU/KWHR =	20781.0
HEATING VALUE OF FUEL, BTU/LB =	0.0
AIR STORED, M LB/DAY =	******
AIR LEAKAGE, MM LB/DAY =	1.595
COMPRESSION TIME/GENERATION TIME =	1.00
AIR FLOW TO STORAGE CAVERN, LB/S =	3164.0
AIR FLOW FROM STORAGE CAVERN, LB/S =	3120.0
STORAGE PRESSURE, PSIA =	1246.4
COMPRESSION TIME, HRS =	10.00

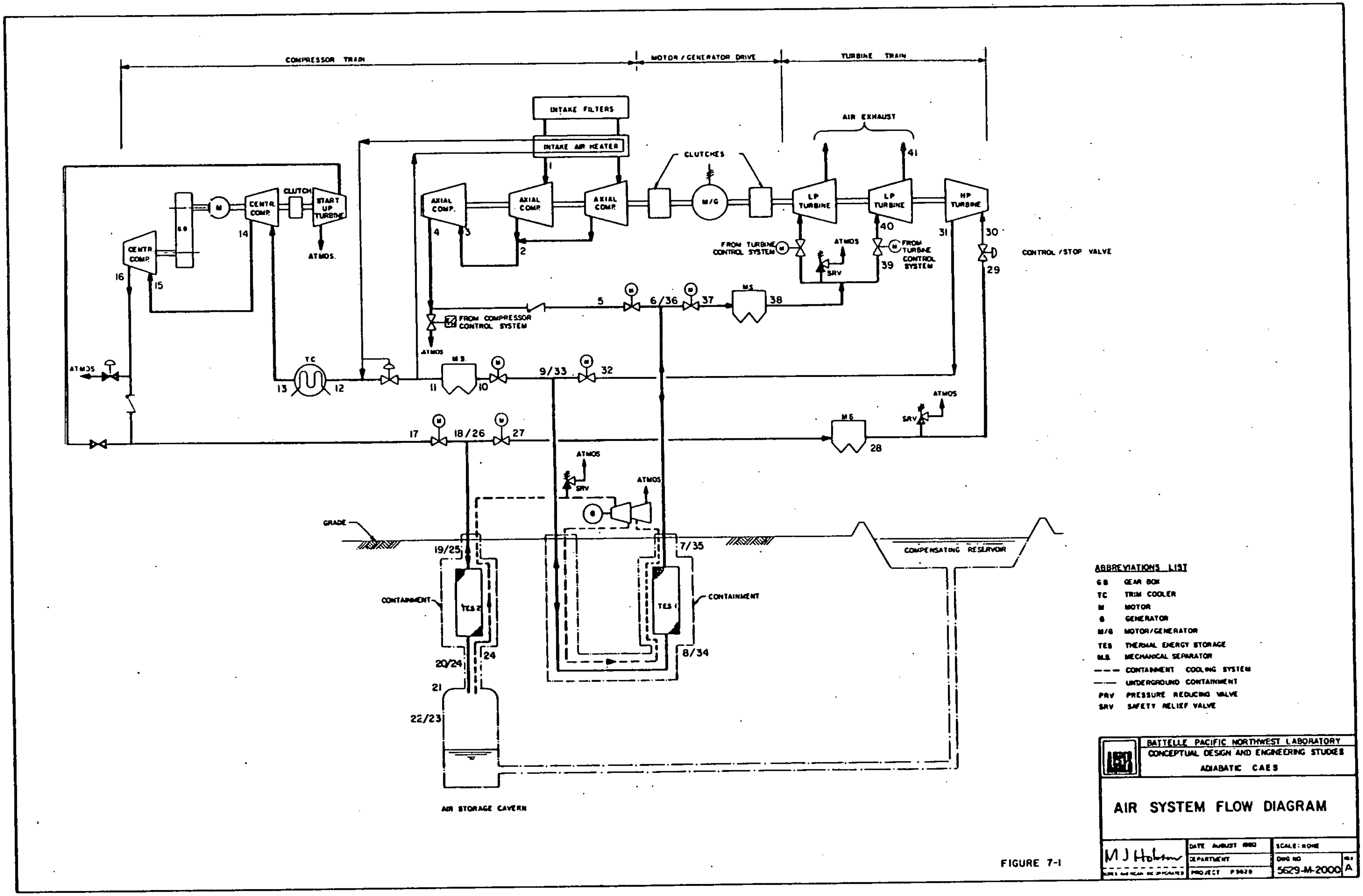
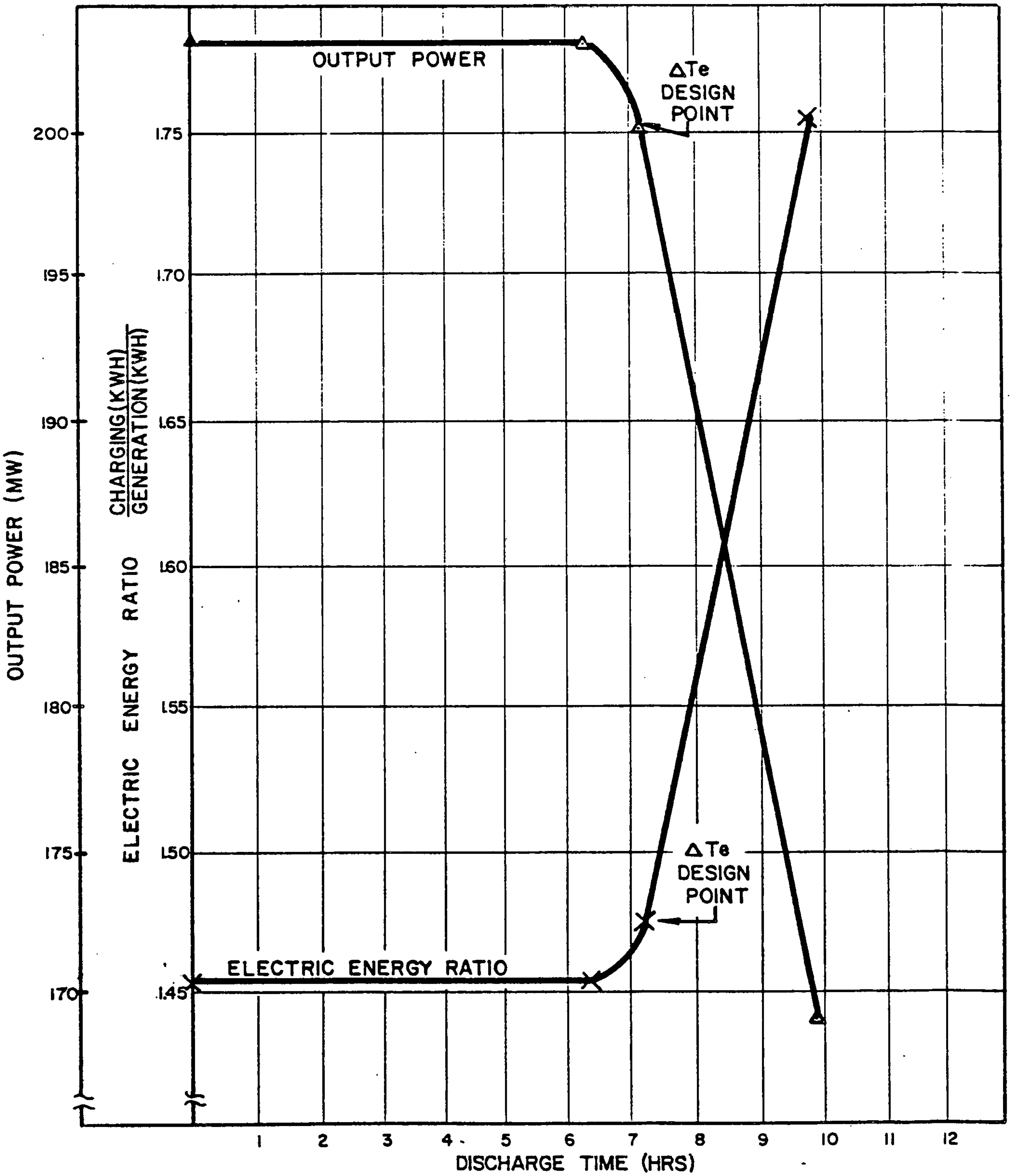

COMPRESSOR TRAIN
MOTOR / GENERATOR DRIVE
TURBINE TRAIN
INTAKE FILTERS
INTAKE AIR HEATER
AIR EXHAUST
CLUTCHES
CLUTCH
START UP TURBINE
CENTR. COMP.
AXIAL COMP.
M/G
LP TURBINE
HP TURBINE
ATMOS.
FROM TURBINE CONTROL SYSTEM
FROM COMPRESSOR CONTROL SYSTEM
CONTROL / STOP VALVE
SRV
TC
MS
GRADE
CONTAINMENT
TES 1
TES 2
COMPENSATING RESERVOIR
AIR STORAGE CAVERN
ABBREVIATIONS LIST
GB GEAR BOX
TC TRIM COOLER
M MOTOR
G GENERATOR
M/G MOTOR/GENERATOR
TES THERMAL ENERGY STORAGE
M.S. MECHANICAL SEPARATOR
CONTAINMENT COOLING SYSTEM
UNDERGROUND CONTAINMENT
PRV PRESSURE REDUCING VALVE
SRV SAFETY RELIEF VALVE
BATTELLE PACIFIC NORTHWEST LABORATORY
CONCEPTUAL DESIGN AND ENGINEERING STUDIES
ADIABATIC CAES
AIR SYSTEM FLOW DIAGRAM
DATE AUGUST 1980
SCALE: NONE
5629-M-2000
A

FIGURE 7-1

FULL LOAD PERFORMANCE OVER THE DISCHARGE PERIOD



BATTELLE PACIFIC NORTHWEST LABORATORY

CONCEPTUAL DESIGN AND ENGINEERING STUDIES—ADIABATIC CAES

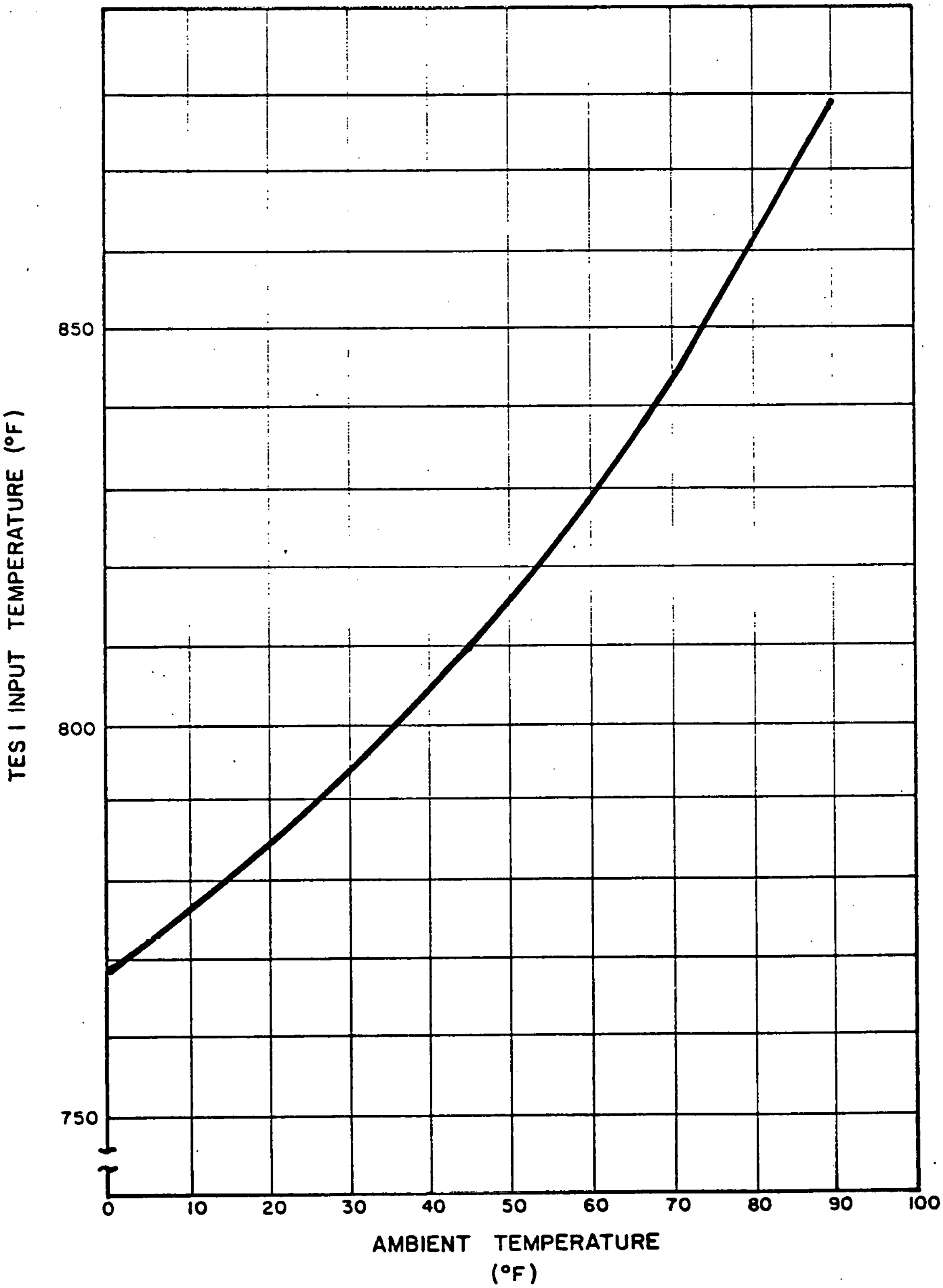
ACRES

M J Hobson

ACRES AMERICAN INCORPORATED

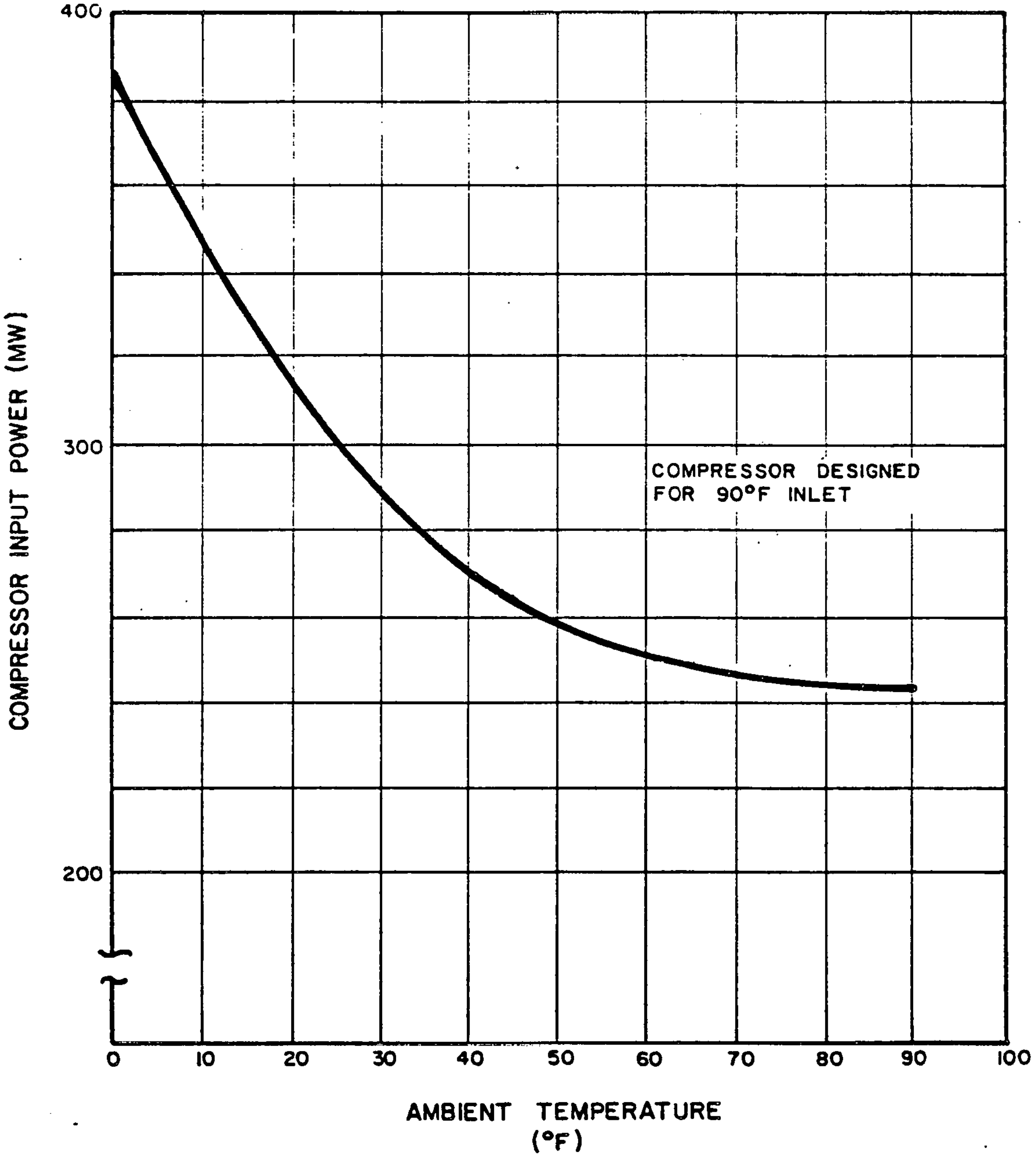
FIGURE 7-2

TES I DELIVERY TEMPERATURE VS AMBIENT TEMPERATURE



ACRES	BATTELLE PACIFIC NORTHWEST LABORATORY
	CONCEPTUAL DESIGN AND ENGINEERING STUDIES—ADIABATIC CAES
M J Hobson ACRES AMERICAN INCORPORATED	FIGURE 7-3

REQUIRED COMPRESSOR POWER
VS AMBIENT TEMPERATURE



ACRES

BATTELLE PACIFIC NORTHWEST LABORATORY

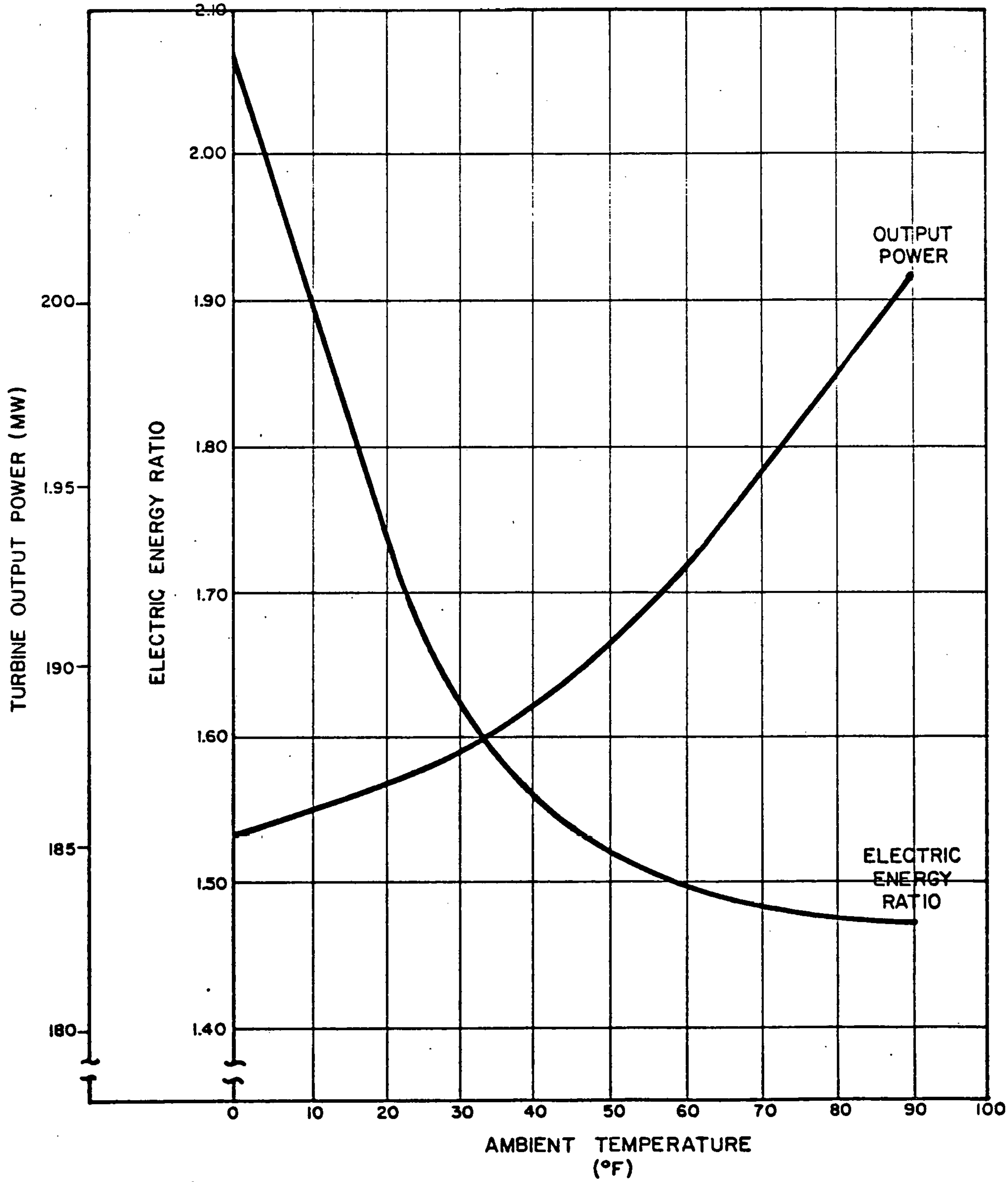
CONCEPTUAL DESIGN AND ENGINEERING STUDIES—ADIABATIC CAES

M J Hobson

ACRES AMERICAN INCORPORATED

FIGURE 7-4

AMBIENT TEMPERATURE VS TURBINE
OUTPUT AND ELECTRIC ENERGY RATIO



ACRES

BATTELLE PACIFIC NORTHWEST LABORATORY

CONCEPTUAL DESIGN AND ENGINEERING STUDIES—ADIABATIC CAES

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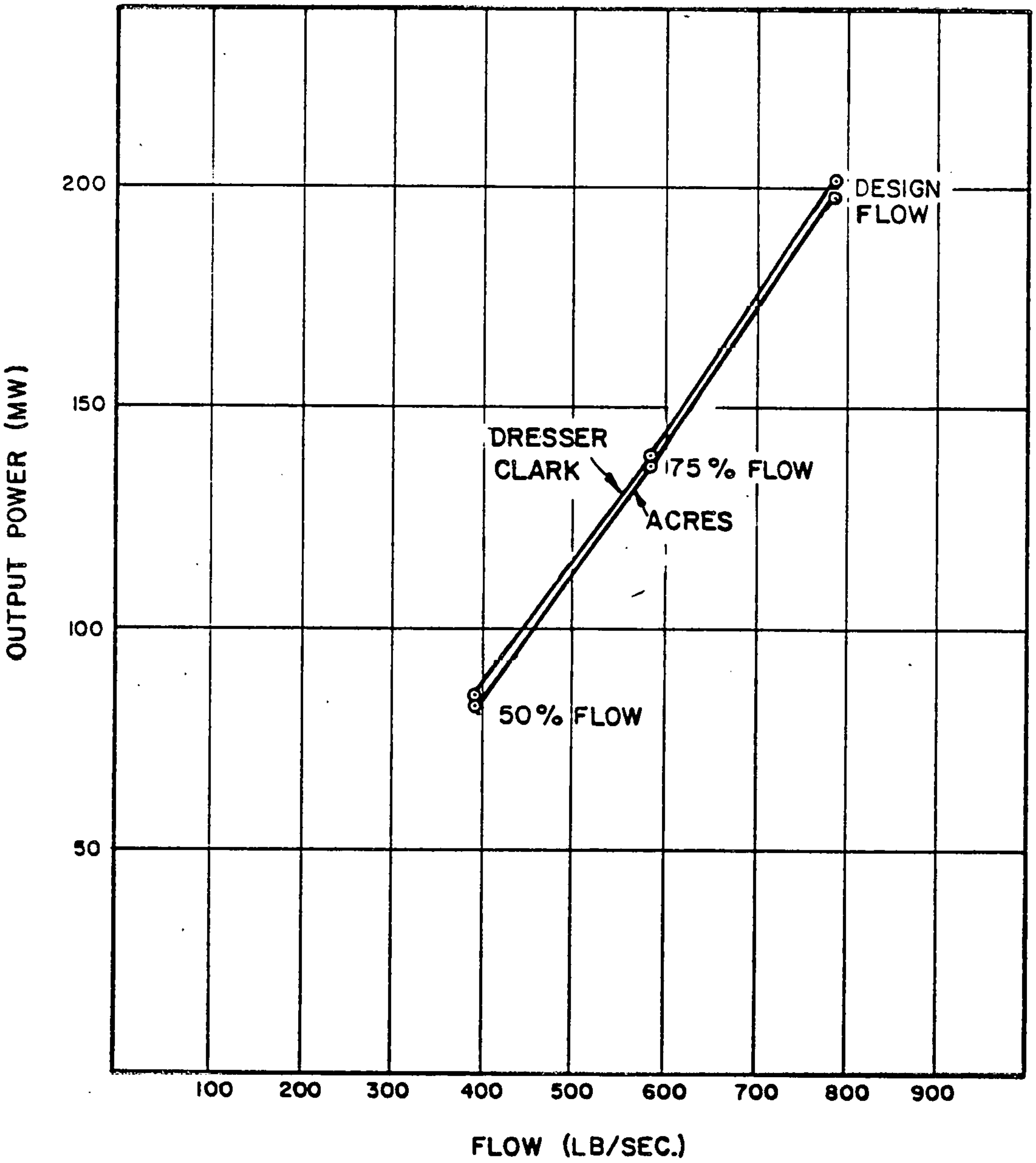
FIGURE 7-5

PART LOAD TURBINE
PERFORMANCE



NOTE: A 97.5 % MECHANICAL/ELECTRICAL EFFICIENCY WAS INCORPORATED INTO DRESSER CLARK ESTIMATES TO FACILITATE THIS COMPARISION

ACRES

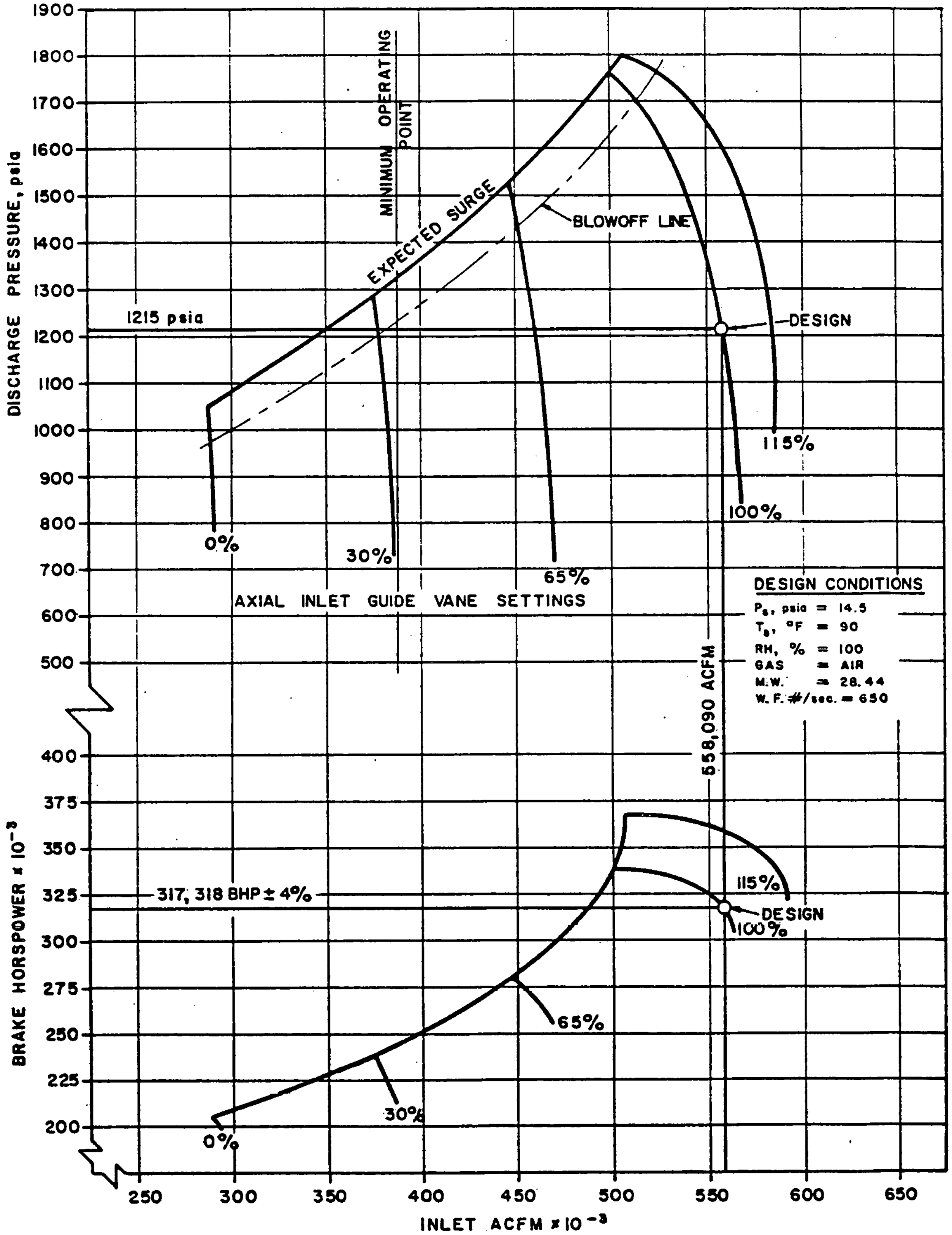
BATTELLE PACIFIC NORTHWEST LABORATORY

CONCEPTUAL DESIGN AND ENGINEERING STUDIES—ADIABATIC CAES

M J Hobson

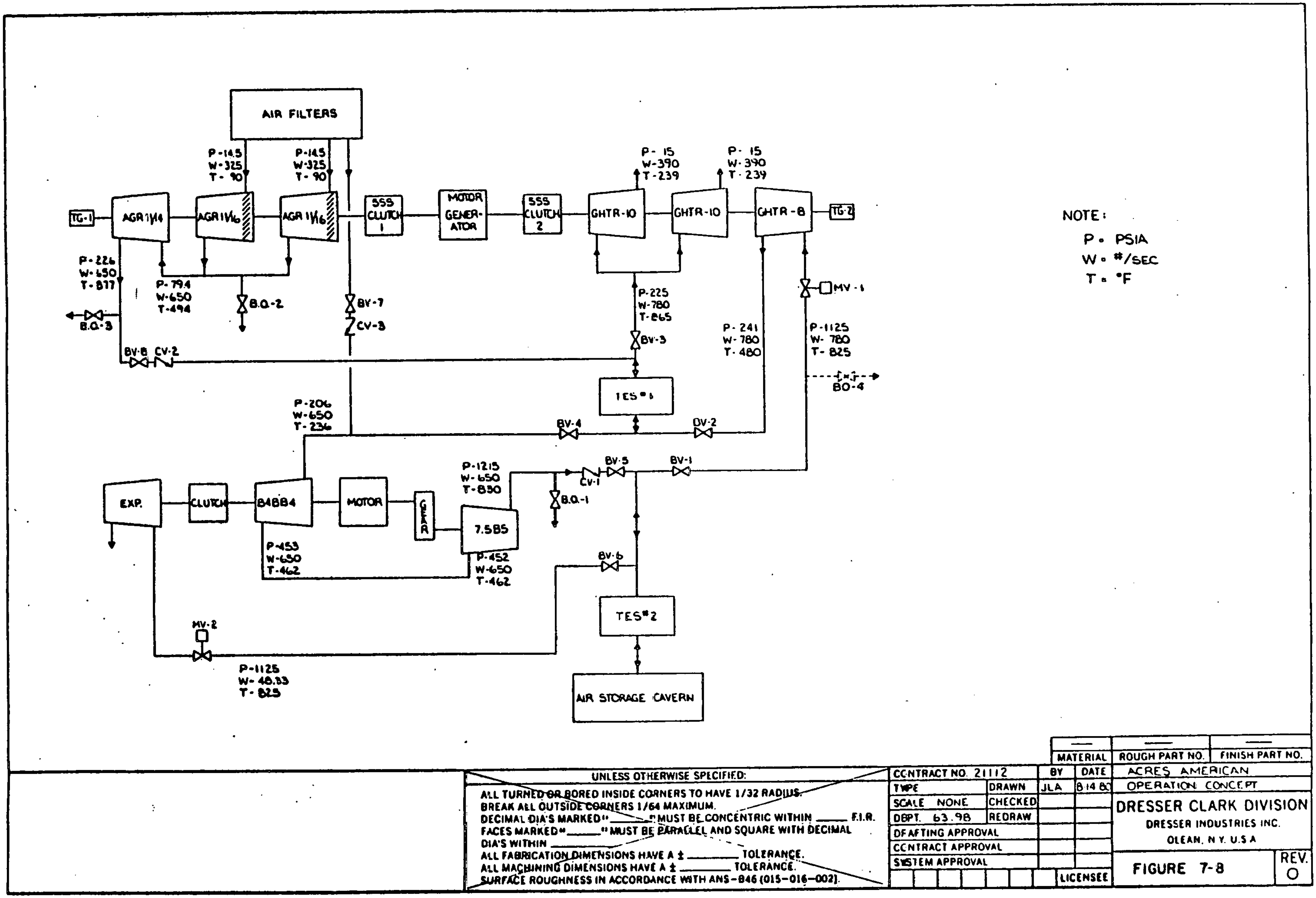
ACRES AMERICAN INCORPORATED

FIGURE 7-6



OVERALL COMPRESSOR SYSTEM PERFORMANCE CURVE

ACRES	BATTELLE PACIFIC NORTHWEST LABORATORY CONCEPTUAL DESIGN AND ENGINEERING STUDIES—ADIABATIC CAES
M J Hobson ACRES AMERICAN INCORPORATED	FIGURE 7-7


AIR FILTERS
P-14.5 W-325 T-90
P-14.5 W-325 T-90
TG-1
AGR 1V14
AGR 1V16
AGR 1V16
SSS CLUTCH 1
MOTOR GENERATOR
SSS CLUTCH 2
GHTR-10
GHTR-10
GHTR-8
TG-2
P-15 W-390 T-239
P-15 W-390 T-239
NOTE:
P = PSIA
W = #/SEC
T = °F
P-226 W-650 T-877
P-79.4 W-650 T-494
B.O.-2
BV-7
CV-3
B.O.-3
BV-8
CV-2
P-225 W-780 T-865
BV-3
MV-1
P-241 W-780 T-480
P-1125 W-780 T-825
BO-4
TES #1
P-206 W-650 T-236
BV-4
BV-2
P-1215 W-650 T-850
BV-5
BV-1
CV-1
B.O.-1
EXP.
CLUTCH
B4BB4
MOTOR
GEAR
7.5B5
P-453 W-650 T-462
P-452 W-650 T-462
BV-6
TES #2
MV-2
P-1125 W-48.33 T-825
AIR STORAGE CAVERN
MATERIAL
ROUGH PART NO.
FINISH PART NO.
UNLESS OTHERWISE SPECIFIED:
ALL TURNED OR BORED INSIDE CORNERS TO HAVE 1/32 RADIUS.
BREAK ALL OUTSIDE CORNERS 1/64 MAXIMUM.
DECIMAL DIA'S MARKED "______" MUST BE CONCENTRIC WITHIN ______ F.I.R.
FACES MARKED "______" MUST BE PARALLEL AND SQUARE WITH DECIMAL DIA'S WITHIN ______
ALL FABRICATION DIMENSIONS HAVE A ± ______ TOLERANCE.
ALL MACHINING DIMENSIONS HAVE A ± ______ TOLERANCE.
SURFACE ROUGHNESS IN ACCORDANCE WITH ANS-B46 (015-016-002).
CONTRACT NO. 21112
BY
DATE
TYPE
DRAWN JLA 8-14-80
SCALE NONE
CHECKED
DEPT. 63.98
REDRAW
DRAFTING APPROVAL
CONTRACT APPROVAL
SYSTEM APPROVAL
LICENSEE
ACRES AMERICAN
OPERATION CONCEPT
DRESSER CLARK DIVISION
DRESSER INDUSTRIES INC.
OLEAN, N.Y. U.S.A.
FIGURE 7-8
REV. 0

8 - CONCEPTUAL PLANT DESIGN

8 - CONCEPTUAL PLANT DESIGN

This section describes the physical and operational relationship of the adiabatic CAES plant components.

8.1 PROCESS ARRANGEMENT

Figure 8-1 shows the operating relationships among the major system groups including the compressor train, turbine train, motor and motor/generator, air storage, and thermal energy storage.

The compressor system includes the air intake filters, air preheater, axial compressors for low pressure compression, and centrifugal compressors and gearbox for high pressure compression. These items operate during periods when electrical demand of the utility system is low, typically at night.

The expansion system includes the high pressure and low pressure turbines, and turbine control valves, and air exhaust ducting. The turbine section converts the energy stored by the compressor section into useful electric power as required by the utility to meet peak demands (typically between 8 am and 10 pm).

The motor/generator system includes the motor/generator, clutches, and motor to drive the high pressure compressors. The motor/generator operates with either the low pressure compressors or the turbines through synchronous clutches, and can also operate as a synchronous condenser.

The air storage system includes the air storage cavern, compensating reservoir, and water shaft. The air storage caverns are pressurized at all times during plant operation and are located underground in solid rock formations.

The thermal energy storage system includes the pebble bed regenerators, air shafts, and TES cooling system. A small continuous flow of cooling air is required for the thermal storage system caverns and high temperature pipe shafts to prevent overheating of the rock walls. Shown in the cooling system is a small turbine generator which allows some power recovery from the throttling process required between TES 2 and TES 1. A pressure reducing valve is also required following the expander, as air exit temperatures would be far too low if all throttling is done in the turbine. The required valve is not shown in the cycle configuration.

Major balance-of-plant items include the trim cooler, mechanical separators, electrical system and switchyard, general plant piping and valving, and instrumentation and control systems.

8.2 SITE DEVELOPMENT

The site used as the design basis of this study is the Sunshine site of the recent PEPCO preliminary engineering oil-fired CAES study. The 500 acre location straddles an existing 500 kV transmission line and lies directly above a formation of granitic gneiss suitable for construction of hard rock caverns for storage of compressed air. A detailed description of the site is presented in the final reports of the PEPCO study (Acres, 1980).

8.3 GENERAL PLANT ARRANGEMENT

The site arrangement for adiabatic CAES is presented in Figure 8-2. This arrangement is similar to the one developed for the oil-fired CAES study, which is shown in Figure 8-3 for comparison. The fuel oil storage and supply system, cooling system, recuperators, and exhaust stacks of the oil-fired CAES plant are major equipment items not required for the adiabatic CAES plant. The turbomachinery buildings, switchyard, auxiliary buildings, compensating reservoir and intake structure, and water impoundment areas are essentially the same design for both plants. In case of plant power outage, a small fuel oil tank is provided to fuel a standby generator.

The high pressure TES system is located underground in a hard rock cavern adjacent to the air pipe that accesses the air cavern. The low pressure TES, which can be located underground anywhere within ready access of surface plant facilities, is shown in this case as close to the turbomachinery buildings as possible to minimize piping costs.

The plant design developed for the PEPCO study was connected to the main system on a 500 kV pool intertie. This connection required a comparatively expensive switchyard and has been carried over in this study to allow direct comparison of the two designs.

8.4 SURFACE FACILITIES

This section describes the major surface facilities of the adiabatic CAES plant. These facilities include the turbomachinery units, switchyard, compensating reservoir, and water impoundment area.

8.4.1 Power Block - The general arrangement of the turbomachinery buildings is shown in Figure 8-4. Main items included in this area are:

- Two main turbomachinery buildings each sized for two units
- Main control and administration building
- Unit 3 and 4 control building
- Inlet and exhaust air ducts
- Unit transformers and 500 kV bus
- Main air pipe runs

8.4.2 Turbomachinery Arrangement - The turbine building for each CAES unit houses the turbomachinery train, piping systems, and mechanical and electrical auxiliaries. Figure 8-5 shows a typical turbomachinery arrangement for one unit.

The turbine buildings designed for oil-fired CAES are of sufficient size to house the equipment of adiabatic CAES. Despite requiring a more complicated machinery train, adiabatic CAES does not require additional building space. Unlike the PEPCO design, exhaust air is vented through the side of the building rather than up a stack and the intercoolers and aftercoolers, which occupy a large portion of the building space for the oil-fired CAES design, are not needed.

Foundation pedestals support each turbomachinery train. Each building is provided with cranes capable of servicing all equipment and maintenance areas.

8.4.3 Switchyard - As with the oil-fired CAES design, adiabatic CAES connects to the 500 kV transmission system which passes through the site. Functions assigned to the switchyard include connection to and protection of the 500 kV transmission system and the interconnection with the motor/generator circuits and station service. The switchyard includes:

- Provisions for switching eight 500 kV lines
- Four motor/generator unit 500 kV tie lines
- Control house and microwave tower

8.4.4 Compensating Reservoir - The compensating reservoir is a circular asphalt-lined rockfill dike constructed above level ground. The reservoir has a bottom diameter of approximately 1000 feet and an embankment height of 35 feet including a live volume height of 25 feet, a dead storage height of 5 feet, and a freeboard height of

5 feet. The intake structure consists of an intake tunnel, trash-racks, and gate structure with an 8-foot diameter sluice gate.

8.4.5 Water Impoundment Area - A water impoundment area is incorporated into the plant design to allow use of the on-site stream for consumptive water needs. The design consists of an asphalt-lined dike, with bottom soil treated with bentonite to minimize ground leakage. A building near the dam houses the waste treatment system and the service and fire water pumps.

8.5 AIR STORAGE SYSTEM

The compressed air storage system is a hydraulically-compensated, hard-rock mined cavern facility based on the oil-fired CAES design. The system consists of a surface compensating reservoir and intake structure, separate air and water shafts, and an underground storage cavern. Figure 8-6 presents the general arrangement of the underground facilities including the TES. Except for design of the air shaft, a larger compensating reservoir, and a larger and deeper cavern, the adiabatic CAES air storage system is identical to the PEPCO study design. In this section the air cavern and water shaft designs are briefly described. The air shaft design is discussed in Section 8.7.2. The surface reservoir was described in the previous section.

8.5.1 Air Cavern - The underground cavern consists of four parallel main tunnels that are joined at each end by cross-connecting tunnels. Figure 8-2 shows the configuration plan of the air cavern and Figure 8-6 shows an elevation view. The water shaft connects to one of the cross-connecting tunnels, and the air shaft connects to the other. The vertical cross section of each main tunnel is 85 feet high and 60 feet wide. The ceilings are arched and the walls are curved to improve structural stability. The length of each main tunnel is approximately 1400 feet. The air and water collector tunnels are approximately 30 feet wide by 40 feet high and 750 feet long, also with arched ceilings and curved walls. The mean depth of the cavern is approximately 2920 feet below ground level. The cavern is constructed with a 1:200 slope to ensure drainage to the water shaft sump.

8.5.2 Water Shaft - The water shaft is an excavated vertical shaft 15 feet in diameter, lined with concrete to a finished diameter of 13 feet. The shaft is connected to the upper compensating reservoir through an intake tunnel and to the cavern through (1) a horizontal tunnel for use during construction which will be blocked with a concrete bulkhead at the end of the construction phase, and (2) a U-tube extending 350 feet below the cavern, used for water flow in and out of the cavern during operation of the plant. The U-tube has an excavated diameter of 10 feet and is lined with concrete to a finished diameter of 8 feet. An 80-foot deep by 8-foot diameter well is provided at the bottom of the U-tube to collect sediment from the water.

8.6 TES SYSTEM

As described in Section 4, two TES regenerators are required for the selected adiabatic CAES cycle. TES 1 is stationed between the first and second stages of compression and expansion, and operates at low pressure (226 psia). TES 2 is stationed between the turbomachinery and the air storage cavern, and operates at high pressure (1215 psia). The cycle arrangement is presented in Figure 8-1.

8.6.1 TES Cavern Depth - An elevation view of the general underground arrangement of the TES is shown in Figure 8-6. TES 1 is accessed by two air ducts extending from the surface to a mean cavern depth of 300 feet. TES 2 is located 1300 feet below the surface and is accessed by the air duct that extends from the surface facilities to the air storage cavern.

The TES cavern depths were selected such that the weight of the rock overburden is sufficient to counterbalance the forces of air pressure within each TES. For safe cavern design, a minimum depth of one foot per psi of pressure confinement was assumed as a reasonable guideline. With this assumption, TES 2 could be placed at any depth from approximately 1250 feet below the surface down to the air storage cavern depth of 2920 feet. The optimum depth would be dependent on the cost of TES cavern excavation, TES construction and maintenance, and hot and cold air access piping design problems. For this study, TES 2 is situated at the minimum depth to minimize the thermal expansion of the hot air pipe when the system is brought from cold start-up to operating temperature. Further discussion of the hot air pipe design is presented in Section 8.6.3.

Because TES 1 is independent of the air storage system, unlike TES 2, shaft placement should be as close to the surface plant facilities as possible to minimize cost. TES placement should also be as close to the surface as possible. Deeper placement of TES 1 would only increase costs in all areas without providing any substantive advantages.

8.6.2 TES Fill Material - As reported in Section 5, a review of potential materials concluded that five had potential as heat storage media - rock, sintered iron oxide, fireclay (Denstone), alumina, and white cast iron. Of these materials, granite, iron oxide, and white cast iron were selected for TES sizing activities. Of the three, iron oxide was selected for thermal storage conceptual design and detailed cost assessment. Estimates of TES costs for other materials were extrapolated from this baseline case.

8.6.3 TES Containment Design - The major elements of the TES containment design are the excavated rock cavern, shell configuration, support structure, and air distribution system. Figure 8-7 presents the conceptual design for these elements.

As discussed in Section 5.6.3, the cavern configuration selected was 106 feet high by 50 feet wide with a 25 foot radius roof arch. The height of the bed fill material was limited to 65 feet to minimize pressure drop and to provide ample room for installation of TES facilities.

The objective of the underground TES design approach was to reduce the cost of the pressurized heat storage systems by using hard rock for pressure confinement rather than thick-walled pressure vessels located on the surface. To achieve this objective, a number of containment design alternatives were developed with the capability of transmitting the pressure load to the rock formation.

Each design variation was generally classified under one of two basic concepts. The first consisted of arrangements that placed the pebble bed containment structure solidly against the rock surface such that all horizontal loads were transmitted through the wall to the rock formation. The second consisted of silo-in-cavern arrangements in which the bed containment structure was surrounded by an annular air space. The air space was pressurized by air bled from the air cavern such that the stresses induced on the silo walls were derived only from the pebble load and not from the air pressure.

Variations of the former concept were discarded based on an analytical heat conduction analysis that showed an unavoidable temperature rise in the rock surrounding TES components (Carslaw and Jaeger, 1962; McAdams, 1954). In previous studies, stress analysis of underground excavations suggested that local rock temperatures should not exceed 150°F for safe cavern design (Acres, 1980). The heat conduction analysis concluded that this temperature would be breached in a relatively short time if the heat transferred through the walls of the TES is not removed from the underground environment. Overheating would occur even for a situation of very high thermal resistance across the boundary between the heat storage medium and the rock surface.

The second design concept allows cooling of the TES components using cooling air released from the air cavern. Therefore, the silo-in-cavern arrangement was preferred.

After investigations of various arrangements, a steel cylinder (silo) configuration was selected for its simplicity, inherent structural strength and stability. The excavated cavern width limits silo diameters to approximately 44 feet, which leaves a minimum annular air space of three feet for cooling and construction purposes. With a maximum bed height of 65 feet designated for heat storage pebbles, eleven silos in parallel are required for each TES to provide 10 hours of generating capacity. To accommodate the silos, the length of each TES cavern would be approximately 515 feet.

The silos are fabricated from one inch carbon steel plate to provide sufficient metal for both structural strength and corrosion/erosion losses. Loss of metal would result from oxidation and possibly pebble abrasion. Because TES temperatures rise above 800°F during the charge cycle, mechanical effects of creep and rupture are major concerns for the carbon steel construction. To ensure longer life of the containment structure, higher grade steels may be preferred in preliminary design.

The silos are enclosed at the top by an insulated steel hood that extends the full length of the silo row. The hood is designed to contain the hot air and is mounted on a structural frame welded to the top of the silos.

A perforated steel duct suspended from the ceiling of the hood is proposed to distribute air evenly over the bed approach area. The air distribution duct is flange connected to the air supply pipe that enters the TES structure through an end wall of the hood enclosure. The duct is unrestrained at the other end so that thermal stresses are minimized. Actual design of the ductwork would likely involve physical modeling studies. A cover plate perforated within the cylinder boundaries spans the length of the silo row to prevent leakage of hot air into the annular space and to assist in the distribution of air through the silos.

The bed fill material rests on a stainless steel mesh, with a grid size smaller than the general size range of the iron oxide pebbles. The mesh is supported by a stainless steel grate with a grid spacing of approximately 1-5/16 inches. The mesh, grate, and silos are supported on structural steel beams mounted on a concrete foundation. The beams are arranged lengthwise so as to form channels for air transport. The support system is enclosed such that air does not escape to the annular space. The total TES structure (hood and silos) is wrapped with six inches of open cell insulation (such as Kaowool) and covered by a protective aluminum jacket. Figure 8-7 shows the arrangement of silos and hood system.

Since the air space is pressurized by the cooling air, the pressure gradient across the cylinder walls is slight. The silos, in effect, are simple containment vessels for the bed fill while the TES caverns are pressure containment vessels. Major stresses developed on the silo walls derive from the lateral forces generated by the weight of the packed bed and differential thermal expansion of the bed particles upon heating and cooling. Information is limited regarding packed-bed behavior under thermally cycling conditions. With regard to quantifying stresses induced on the containment wall, several design options are available to relieve such stresses. For example, stresses could be relieved by lowering the effective height of the bed with the insertion of intermediate bed supports, by using materials with low thermal expansion and hardness characteristics, or by simply increasing bed cross-sectional area such that the bed height requirement is reduced. The latter approach should be discouraged because of the increased bed volume that would be required and potential cavern construction limitations.

Investigation of TES moisture loading indicated that condensation of water occurs only in TES 2 for the inlet air temperature and humidity conditions found in the central Maryland area. Condensation occurs primarily during the charging cycle when air temperatures fall below the dew point temperature. Excess water present in the bed drains from the bottom of the silo and down to the air storage caverns via the main air pipe. Condensation will not occur in TES 1 because working temperatures are far above the dew point temperature at all times. A sump and water removal system, however, should be provided at the floor of the TES 1 cavern to remove water that may collect there from other sources.

8.6.4 Air Pipe Design - The access piping to the underground TES presented a considerable design challenge. Two major problems are of concern. First, the excavated rock shafts require protection from excessive temperature rise. Second, thermal growth of the hot air pipes as the system heats up must be accommodated.

In a heat transfer analysis similar to the one performed for the TES caverns, it was concluded that heat lost from the hot air pipes must be removed from the underground environment to prevent overheating of the rock formations with time. Therefore, a three-foot annular air space between the pipe and the rock shaft walls is provided for ventilation purposes. The air space, as in the cavern arrangement, is pressurized and ventilated by air bled from the air storage cavern. The rock walls are pressure grouted to minimize air leakage. To reduce heat loss, the hot air pipes are wrapped with six inches of open-celled insulation (such as Kaowool) and covered by a protective aluminum jacket.

A pressure vessel, securely anchored in the hard rock, is the approach selected to cap each shaft at the surface. The vessels designed for this purpose are similar in cross section to petrochemical fractionating towers. These vessels are fabricated from 3-1/2 inch carbon steel plate for the high pressure shaft and 2-1/2 inch plate for the low pressure shaft.

The second major problem of the hot air pipe design was thermal growth. The hot air pipe to TES 2, for example, expands approximately eight feet when heated from cold start-up to operating temperature and would expand some sixteen feet if the TES cavern was placed at maximum depth. Several methods to accommodate pipe growth were evaluated including the use of expansion loops and various types of expansion joints.

The preferred solution was to anchor the pipe at the entrance to each TES and allow it to expand upward. At the upper end, the pipe expands through sleeves welded to the pressure caps. A dry lubricated seal between the sleeve and pipe prevents excessive leakage of heated air into the pressurized annular space. This arrangement is shown in Figure 8-7. The only TES access pipe not requiring this design is the pipe connecting TES 2 with the air storage cavern. This is a thin wall duct fixed at the TES cavern exit and allowed to expand freely downward into the air cavern.

A high quality stainless steel (A240-XM19) was selected for the air pipe design based on a preliminary analysis of operating conditions. Two criteria were used to determine thicknesses of TES access pipes: (1) the dead weight of the pipe, and (2) the thermal stress resulting from a design maximum temperature gradient of 50°F across the pipe wall. Because of the pressurized annular air space, the pressure gradient across the pipe wall was assumed to be negligible.

Air shaft pipe diameters were estimated based on a rough tradeoff of pressure drop and plant performance. The inside pipe diameters and pipe thicknesses selected, based on these criteria, are presented below:

	I.D. (Inches)	Thickness (Inches)
TES 1 Entry Pipe (Charge Cycle)	91.4	3/8
TES 1 Exit Pipe (Charge Cycle)	76.7	3/8
TES 2 Entry Pipe (Charge Cycle)	43.7	1/2 (top 500 ft) 5/8 (middle 500 ft) 3/4 (bottom 250 ft)
TES 2 Exit Pipe (Charge Cycle)	43.7	1/2

The irregular pipe sizes are acceptable because the pipes exceed standard sizes for stainless steel and, therefore, would be manufactured by specification. Stainless steels are preferred over carbon steels for piping material in this application because of the potentially corrosive conditions. Premature failure would be difficult and expensive to remedy due to the requirement to depressurize and perform replacement work in confined locations. These large pipe sizes will require significant design analysis for the surface runs to estasblish support and anchor designs.

8.6.5 TES Cooling System - A cooling system is required for safe operation of the TES. The system must cool any excursions in cavern air temperature to prevent cavern damage and to ensure that the return air temperature is consistently lower than the air cavern charge (entry) temperature to achieve effective TES operation. The cooling system must also maintain TES cavern and shaft wall temperatures within safe limits to avoid shaft and TES cavern failures.

In normal TES operation, a thermal breakthrough occurs for a short period near the end of the charge cycle such that elevated temperature air dumps into the air cavern. Heat transfer calculations, however, show that the natural conditions in the water-compensated air cavern sufficiently cool this air such that auxiliary cooling is not required. Most of this heat is eventually conveyed to the surface by the compensating water and dissipated to the environment.

The cooling system (see Figure 8-1 for the air flow arrangement) is critical to the regulation of underground rock wall temperatures. The system operates by continuously bleeding cool air from the air storage cavern through the annular spaces surrounding the air pipe and silos of the high pressure TES system. The cooling air removes heat conducted through the walls of TES components such that rock surfaces do not increase in temperature above 150°F. The air is drawn through the annular space by a throttling process at the surface. The air is throttled to the TES 1 operating pressure and cooled to approximately 70°F. Air flow requirements through the entire system are estimated at some 25 pounds per second.

The combined throttling and cooling activity can be performed conveniently in an air expander provided that flows are sufficient to justify the capital expenditure. Preliminary calculations show that approximately one megawatt could be generated. Figure 8-1 shows the cooling circuit with a two-stage expander exhausting to the atmosphere. Freezing problems in the expander would be avoided by specification of a low efficiency design to maintain exit temperatures above ice formation levels.

8.6.6 Particle Carry-Over - The carry-over of particles from the TES during the discharge period is important from two opposing points of view. The concerns are for, firstly, the size of particles that potentially can be transported through the valving and into the turbine and, secondly, the accumulation of particles in the bed that will reduce bed voidage and increase pressure drop.

Figure 8-8 projects the maximum size of spherical particles that will be transported clear of the top of the bed as a function of velocity and particle material. Particles larger than those shown can percolate upwards through the bed, but these will either fall back onto the bed or hover near the top. During the charge period these particles will reenter the bed. The evaluation of particle size was based on Stoke's law for the viscous flow range and drag coefficients given by Kay for the turbulent range (Kay, 1963).

Irregularity of particle shape will increase the size of particle that will become entrained in the upward flowing air stream. Information suggests that for a sphericity of 0.6, which corresponds to the shape of sharp sand, the maximum particle carry-over size for a given velocity and material will be increased by 70 percent (Chew, April 1980).

The data shown in Figure 8-8 indicates that, even for low mass fluxes on the order of 200 lb/hr ft^2, the size of particle carried over is sufficiently large that gas cleanup cannot be avoided. The only particle material for which this may not be true is cast iron. However, the majority of particles generated in a bed of cast iron pebbles will be iron oxide with a density and hence carry-over behavior similar to the iron ore (iron oxide) demarcation shown in the graph. The use of low mass fluxes to reduce particle carry-over is undesirable, however, as this leads to an increase in bed volume requirements for a given end temperature difference.

The general conclusions concerning particle carry-over and retention are:

- Particle carry-over will remain a problem even with the lowest mass fluxes being considered.
- Upward transport and ejection of particles from the bed will provide only a very limited stabilization of bed voidage and pressure drop.
- If the bed is self-cleaning, this will come almost entirely from the downwards transport of particles during the charge period.

8.7 MAJOR MECHANICAL COMPONENTS

The major mechanical equipment covered in this section includes the turbomachinery, air filtration equipment, piping and valves, cooling systems and plant auxiliary systems.

8.7.1 Turbomachinery - The arrangement of machinery for the adiabatic cycle is shown in Figure 8-5. Unlike the oil-fired machinery design, two separate compressor drives were required, with the result that the centrifugal units were arranged in parallel with the axial compressors. These are located in an area occupied by heat exchangers in the oil-fired intercooled PEPCO plant design. The length of the main train is approximately 230 feet; the centrifugal train is approximately 90 feet. The heaviest item is the assembled motor-generator, at 679,000 lbs. The weight of each low-pressure axial compressor (AGr-16) is 325,000 lbs. The PEPCO building arrangement allows use of dual cranes for motor-generator stator lifts, with a combined capacity of 300 tons. This is also adequate for the adiabatic machinery.

8.7.2 Air Filtration Equipment - Compressor inlet air filtration is required, and is located within housings adjacent to the machinery hall. Inlet air ducts from these housings will be equipped with silencers and will contain the compressor inlet air heaters described in Section 6.3.

Large particulates entrained in the air stream exiting TES 1 and TES 2 during the plant generation mode and exiting TES 1 during the plant compression mode must be removed if the design life of the turbomachinery is to be achieved. Dresser Clark specified that the separators should be capable of removing 95 percent of the particulates above 10 microns with a total allowable concentration of 160 ppm. To sustain the high operating pressures of the systems, the separators would be enclosed in pressure vessels. Discussions with several equipment manufacturers revealed that these design specifications could be met. No particular separator design is shown in the arrangements, as selection of a specific manufacturer for this equipment was left for more detailed investigations following definition of actual service conditions.

8.7.3 Piping and Valves - Stainless steel piping is used throughout the main air system based upon the high temperature, high pressure, and large diameter service requirements. Main air lines connecting the storage system to the plant were sized based on pressure drop criteria to meet cycle optimization requirements.

A number of high temperature valves in large diameters for high pressures are required for control of the machinery and shutoff. Major valve availability was investigated independently by both Acres and Dresser Clark. Several manufacturers were identified with potentially suitable products, and Acres and Dresser selected the same manufacturer for estimates of control valve costs. The valves required are unusual (e.g., 20 inch, 1500 lb ANSI class fast closing), expensive, and involve substantial lead time for fabrication. Additional investigation of service requirements, valve availability and design review is anticipated for these critical operating items in future design efforts. This is consistent with the detail design activities normally seen for any other type of large power project.

8.7.4 Cooling Systems - Cycle optimization showed that, for the operating conditions of the turbomachinery, heat rejection from the cycle is required to achieve equilibrium operation for TES 1 and satisfactory performance of turbomachinery components. A trim cooler to reject this heat is located in the compression system piping following TES 1 (Figure 8-1). This location also allows preheating of compressor inlet air.

As discussed in Section 6.3, preheating compressor inlet air is required to maintain acceptable performance throughout the year. If inlet air temperatures were allowed to fluctuate with the seasons, winter temperatures would result in unacceptable compressor discharge conditions. Approximately one fourth of the air flow through the trimcooler must be diverted to the air preheaters (prior to the second stage of compression, see Figure 8-1) to preheat inlet air to the 90°F design point at 0°F ambient temperature.

Heat exchange equipment is also required for lubricating oil and generator hydrogen cooling. The lube oil/hydrogen system requirements are relatively small and can be handled using either wet towers or air-cooled finned tube units, with the latter assumed for the cost estimate.

Air-to-air finned tube exchangers are proposed for the compressor inlet air heaters and trim cooler units. The trim coolers will require manifolding of the primary air piping, but substantially fewer units are required compared to the PEPCO dry cooling system design. The air inlet heaters are sufficiently small to allow vertical mounting in the compressor inlet duct.

8.7.5 Mechanical Auxiliaries - In general, the mechanical auxiliary systems for adiabatic CAES are similar to the systems developed for the oil-fired CAES plant. The major difference between the two plants is the elimination of most of the fuel oil system for adiabatic CAES. (Adiabatic CAES, however, requires some fuel oil for emergency generators). Other plant system requirements, such as the water supply system, waste treatment system, and fire protection system, are very similar for both adiabatic and oil-fired CAES.

8.8 ELECTRICAL SYSTEM

The electrical system can be divided into two broad categories:

(1) switchyard (which was discussed in the section on surface facilities) and

(2) general station services.

On the whole, the plant electrical system for adiabatic CAES is nearly identical to the system developed for oil-fired CAES. The major departures of adiabatic CAES from the PEPCO study design result from:

(1) the elimination of cooling system and fuel pump drives,

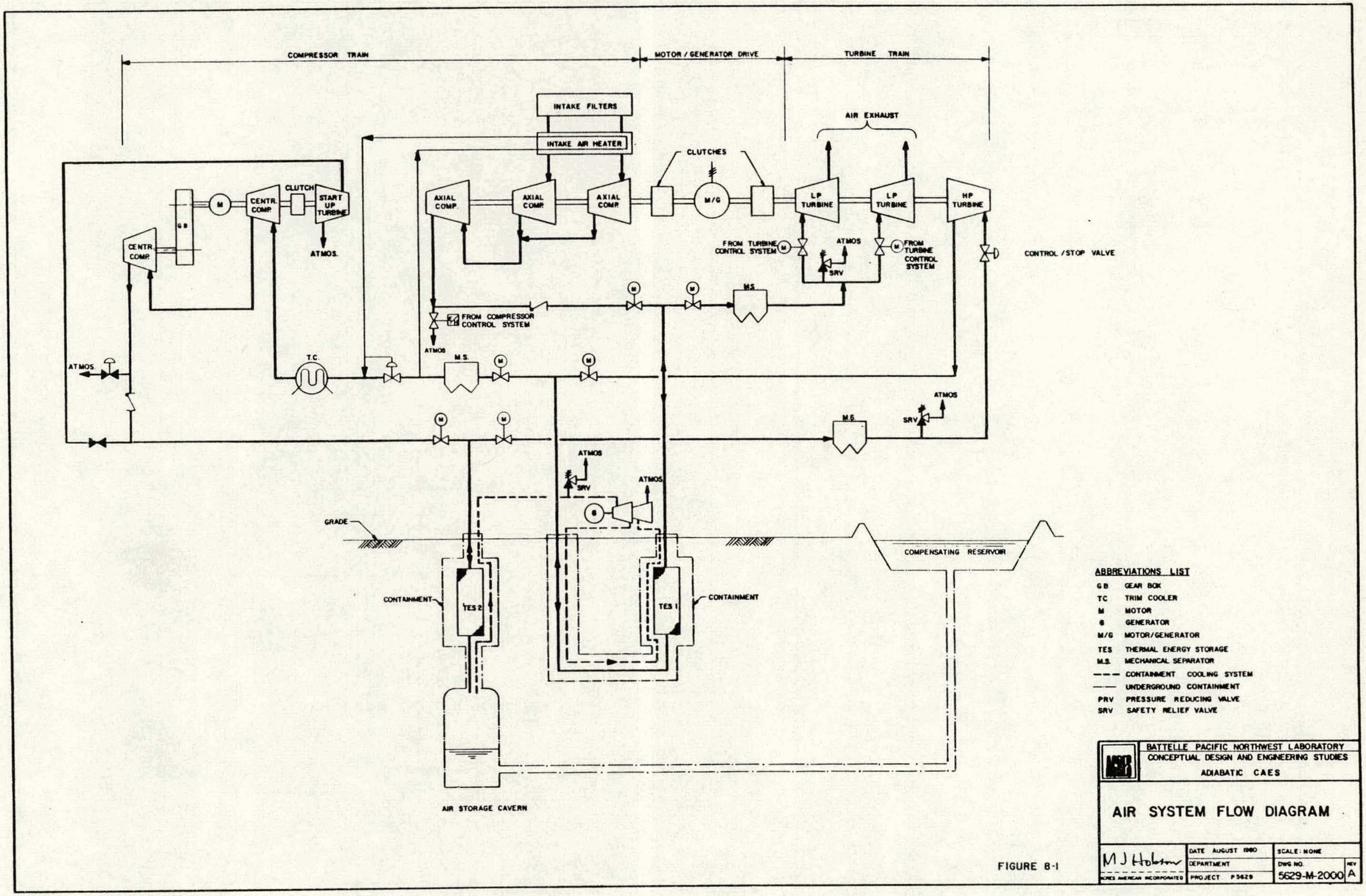
(2) reduction of fan power for the plant cooling systems,

(3) increase of power to drive the larger compressors, and

(4) use of a separate centrifugal compressor motor drive.

Generator voltage is 13.8 kV versus the 18 kV of the Brown Boveri equipment, but this has not resulted in any major change in equipment requirements. The separate motor and motor-generator do require revision to the bus arrangement and equipment needs, altering costs somewhat from the PEPCO study estimate.

The use of a separate motor to drive the centrifugal compressors prevents starting these machines directly with the turbines. Several alternatives are foreseen. The drawings show a separate start-up air turbine in the centrifugal compressor train. Other approaches to start-up include use of variable frequency drives (static starters) and back to back start-up of the motors and motor-generators. The use of a back-to-back (cross-connected windings) approach is foreseen to be the least costly of the three alternatives and is reflected in the cost estimate.


COMPRESSOR TRAIN
MOTOR / GENERATOR DRIVE
TURBINE TRAIN
INTAKE FILTERS
INTAKE AIR HEATER
AIR EXHAUST
CLUTCHES
CLUTCH
START UP TURBINE
CENTR. COMP.
M
GB
ATMOS.
AXIAL COMP.
M/G
LP TURBINE
HP TURBINE
FROM TURBINE CONTROL SYSTEM
FROM TURBINE CONTROL SYSTEM
ATMOS
SRV
CONTROL / STOP VALVE
M.S.
FROM COMPRESSOR CONTROL SYSTEM
ATMOS
T.C.
ATMOS.
ATMOS
SRV
GRADE
G
CONTAINMENT
TES 2
TES 1
CONTAINMENT
COMPENSATING RESERVOIR
AIR STORAGE CAVERN
ABBREVIATIONS LIST
GB GEAR BOX
TC TRIM COOLER
M MOTOR
G GENERATOR
M/G MOTOR/GENERATOR
TES THERMAL ENERGY STORAGE
M.S. MECHANICAL SEPARATOR
CONTAINMENT COOLING SYSTEM
UNDERGROUND CONTAINMENT
PRV PRESSURE REDUCING VALVE
SRV SAFETY RELIEF VALVE
BATTELLE PACIFIC NORTHWEST LABORATORY
CONCEPTUAL DESIGN AND ENGINEERING STUDIES
ADIABATIC CAES
AIR SYSTEM FLOW DIAGRAM
M J Hobson
ACRES AMERICAN INCORPORATED
DATE AUGUST 1980
DEPARTMENT
PROJECT P5629
SCALE: NONE
DWG NO.
5629-M-2000
REV
A
FIGURE 8-1

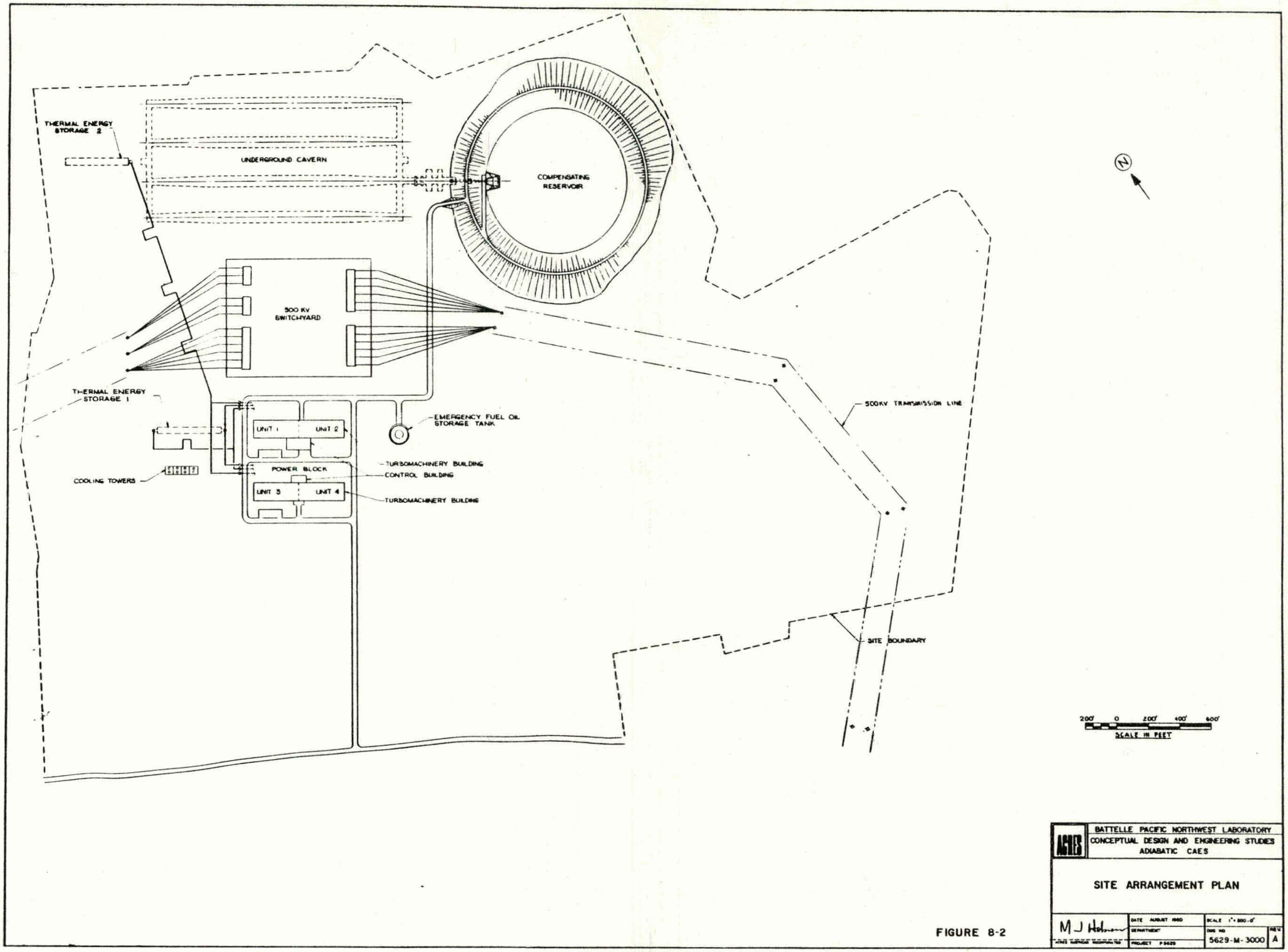

THERMAL ENERGY STORAGE 2
UNDERGROUND CAVERN
COMPENSATING RESERVOIR
500 KV SWITCHYARD
THERMAL ENERGY STORAGE 1
EMERGENCY FUEL OIL STORAGE TANK
UNIT 1
UNIT 2
TURBOMACHINERY BUILDING
CONTROL BUILDING
POWER BLOCK
COOLING TOWERS
UNIT 3
UNIT 4
TURBOMACHINERY BUILDING
500KV TRANSMISSION LINE
SITE BOUNDARY
200' 0 200' 400' 600'
SCALE IN FEET
ACRES
BATTELLE PACIFIC NORTHWEST LABORATORY
CONCEPTUAL DESIGN AND ENGINEERING STUDIES
ADIABATIC CAES
SITE ARRANGEMENT PLAN
DATE AUGUST 1980
SCALE 1" = 200'-0"
5629-M-3000
A

FIGURE 8-2

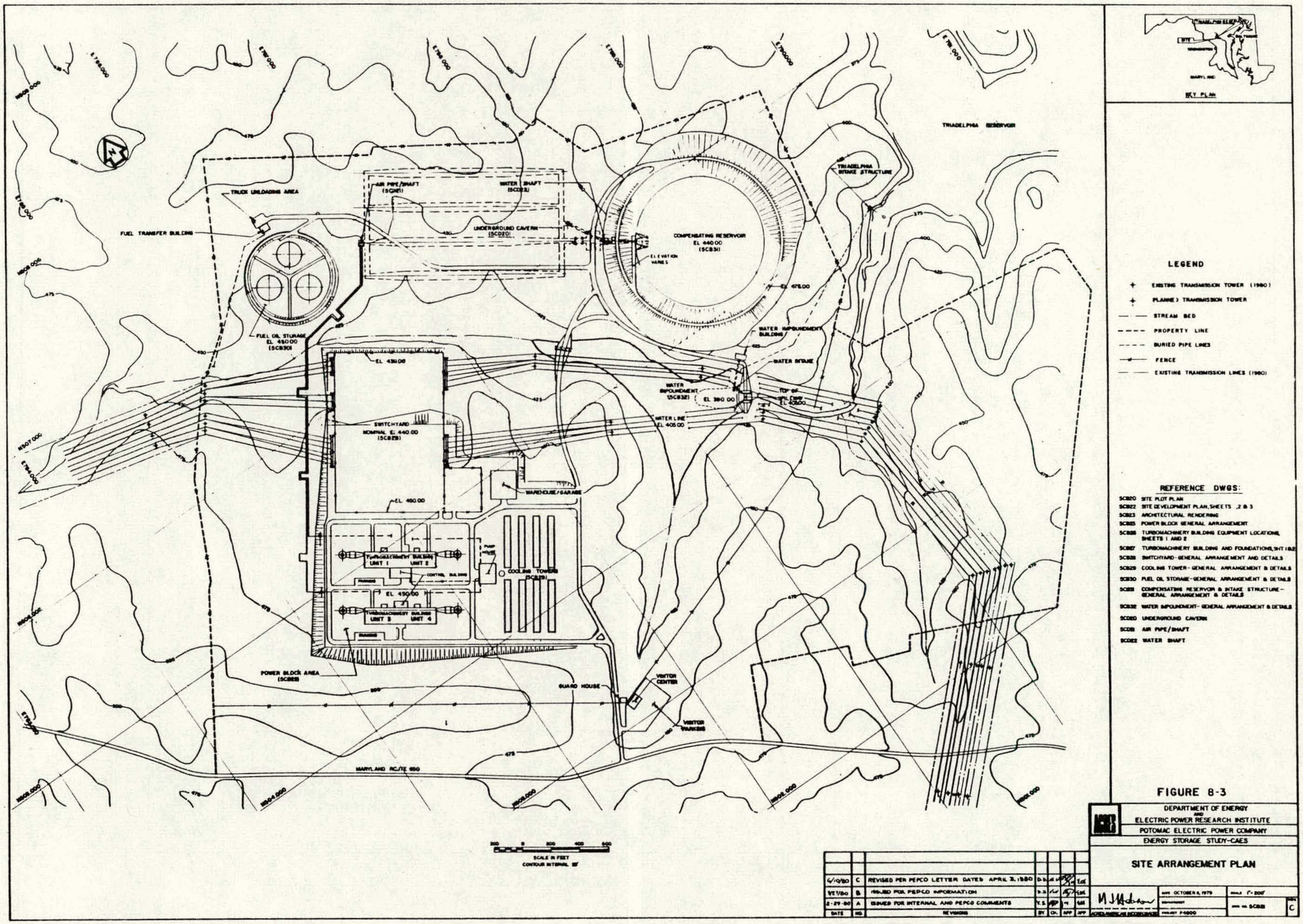

KEY PLAN
TRIADELPHIA RESERVOIR
TRIADELPHIA INTAKE STRUCTURE
TRUCK UNLOADING AREA
FUEL TRANSFER BUILDING
AIR PIPE/SHAFT (SCD21)
WATER SHAFT (SCD22)
UNDERGROUND CAVERN (SCD20)
COMPENSATING RESERVOIR EL 440.00 (SCB31)
ELEVATION VARIES
EL 475.00
FUEL OIL STORAGE EL 450.00 (SCB30)
WATER IMPOUNDMENT BUILDING
WATER INTAKE
WATER IMPOUNDMENT (SCB32)
EL 390.00
WATER LINE EL 405.00
EL 430.00
SWITCHYARD NOMINAL EL 440.00 (SCB28)
WAREHOUSE/GARAGE
EL 460.00
TURBOMACHINERY BUILDING UNIT 1 UNIT 2
CONTROL BUILDING
PUMP HOUSE
COOLING TOWERS (SCB29)
EL 450.00
TURBOMACHINERY BUILDING UNIT 3 UNIT 4
POWER BLOCK AREA (SCB25)
GUARD HOUSE
VISITOR CENTER
VISITOR PARKING
MARYLAND ROUTE 650
SCALE IN FEET
CONTOUR INTERVAL 25'
LEGEND
EXISTING TRANSMISSION TOWER (1980)
PLANNED TRANSMISSION TOWER
STREAM BED
PROPERTY LINE
BURIED PIPE LINES
FENCE
EXISTING TRANSMISSION LINES (1980)
REFERENCE DWGS:
SCB20 SITE PLOT PLAN
SCB22 SITE DEVELOPMENT PLAN, SHEETS ,2 & 3
SCB23 ARCHITECTURAL RENDERING
SCB25 POWER BLOCK GENERAL ARRANGEMENT
SCB26 TURBOMACHINERY BUILDING EQUIPMENT LOCATIONS, SHEETS 1 AND 2
SCB27 TURBOMACHINERY BUILDING AND FOUNDATIONS, SHT 1&2
SCB28 SWITCHYARD-GENERAL ARRANGEMENT AND DETAILS
SCB29 COOLING TOWER-GENERAL ARRANGEMENT & DETAILS
SCB30 FUEL OIL STORAGE-GENERAL ARRANGEMENT & DETAILS
SCB31 COMPENSATING RESERVOIR & INTAKE STRUCTURE-GENERAL ARRANGEMENT & DETAILS
SCB32 WATER IMPOUNDMENT-GENERAL ARRANGEMENT & DETAILS
SCD20 UNDERGROUND CAVERN
SCD21 AIR PIPE/SHAFT
SCD22 WATER SHAFT
DEPARTMENT OF ENERGY AND ELECTRIC POWER RESEARCH INSTITUTE
POTOMAC ELECTRIC POWER COMPANY
ENERGY STORAGE STUDY-CAES
SITE ARRANGEMENT PLAN
6/10/80 C REVISED PER PEPCO LETTER DATED APRIL 3, 1980
3/27/80 B ISSUED FOR PEPCO INFORMATION
2-29-80 A ISSUED FOR INTERNAL AND PEPCO COMMENTS
DATE NO. REVISIONS BY CH. APP APP
DATE OCTOBER 8, 1979
SCALE 1"=200'
PROJECT P4800
C

FIGURE 8-3

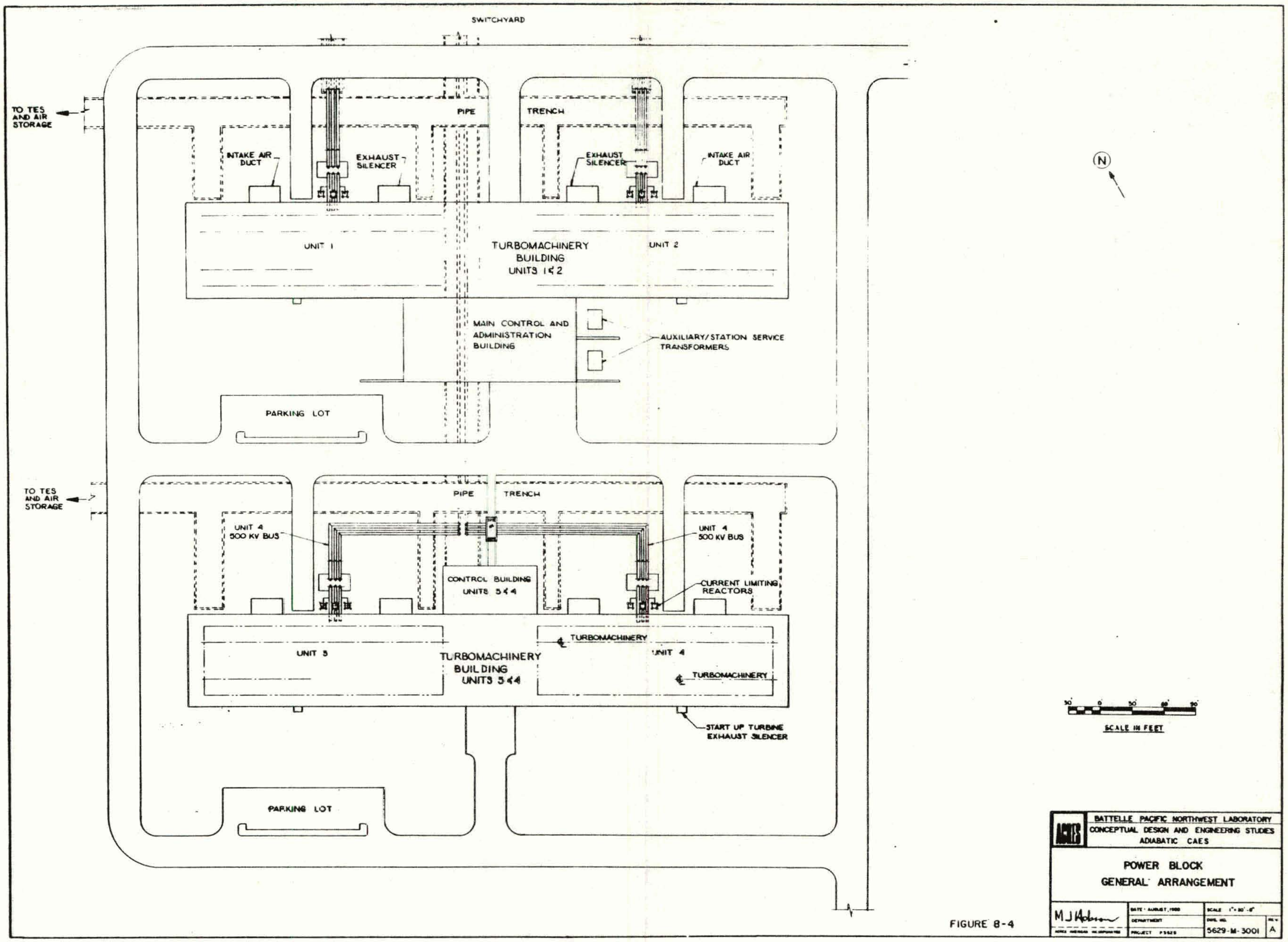
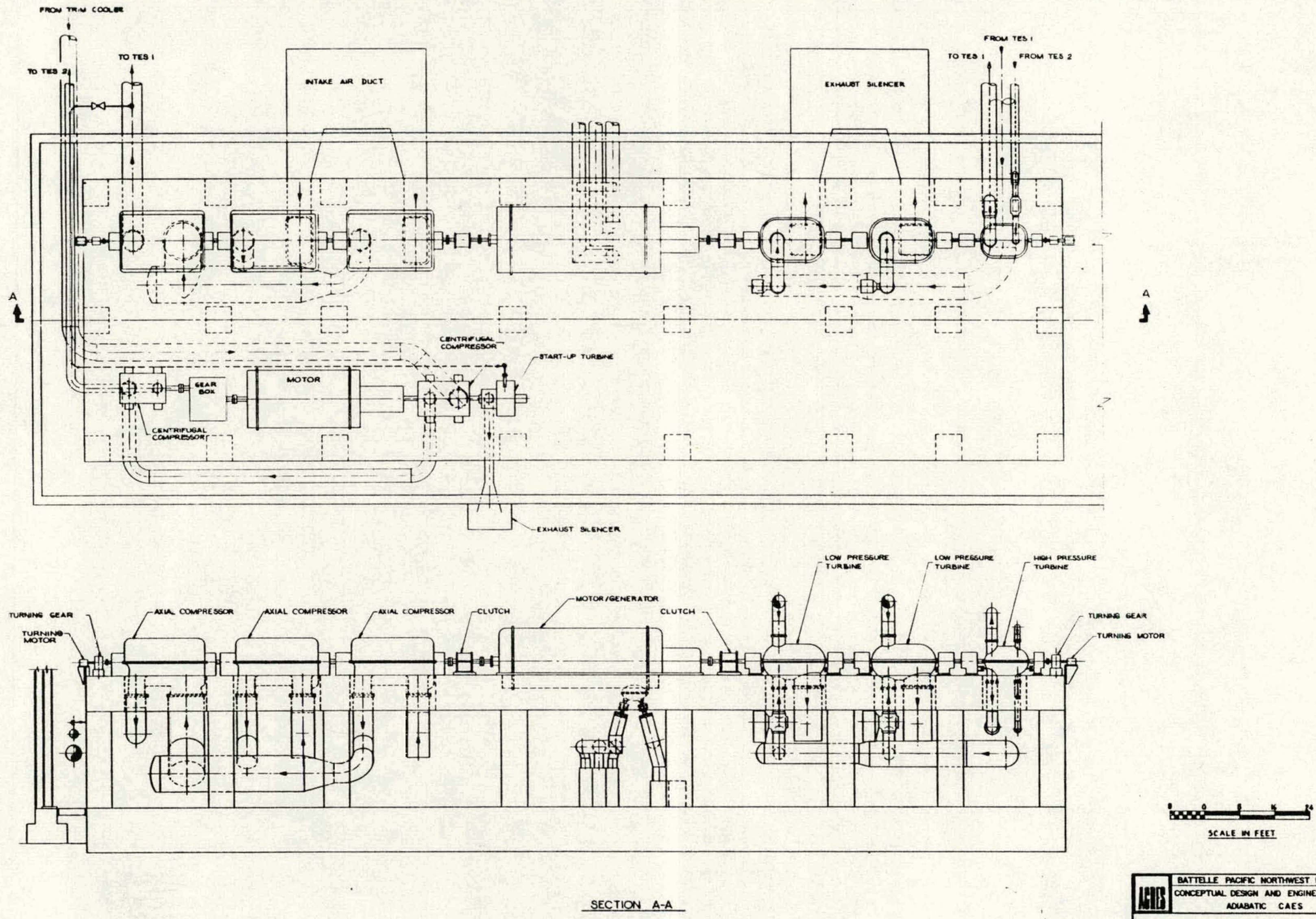

SWITCHYARD
TO TES AND AIR STORAGE
PIPE
TRENCH
INTAKE AIR DUCT
EXHAUST SILENCER
UNIT 1
TURBOMACHINERY BUILDING UNITS 1 & 2
UNIT 2
MAIN CONTROL AND ADMINISTRATION BUILDING
AUXILIARY/STATION SERVICE TRANSFORMERS
PARKING LOT
UNIT 4 500 KV BUS
CONTROL BUILDING UNITS 3 & 4
CURRENT LIMITING REACTORS
UNIT 3
TURBOMACHINERY BUILDING UNITS 3 & 4
UNIT 4
TURBOMACHINERY
START UP TURBINE EXHAUST SILENCER
SCALE IN FEET
N
BATTELLE PACIFIC NORTHWEST LABORATORY
CONCEPTUAL DESIGN AND ENGINEERING STUDIES
ADIABATIC CAES
POWER BLOCK
GENERAL ARRANGEMENT
ACRES
5629-M-3001
A

FIGURE 8-4



BATTELLE PACIFIC NORTHWEST LABORATORY
CONCEPTUAL DESIGN AND ENGINEERING STUDIES
ADIABATIC CAES

TURBOMACHINERY SYSTEM
GENERAL ARRANGEMENT

MJ Hobson

DATE AUGUST 1980 | SCALE: 1/8"=1'-0" | DWG NO. 5629-M-3002 | REV A

FIGURE 8-5

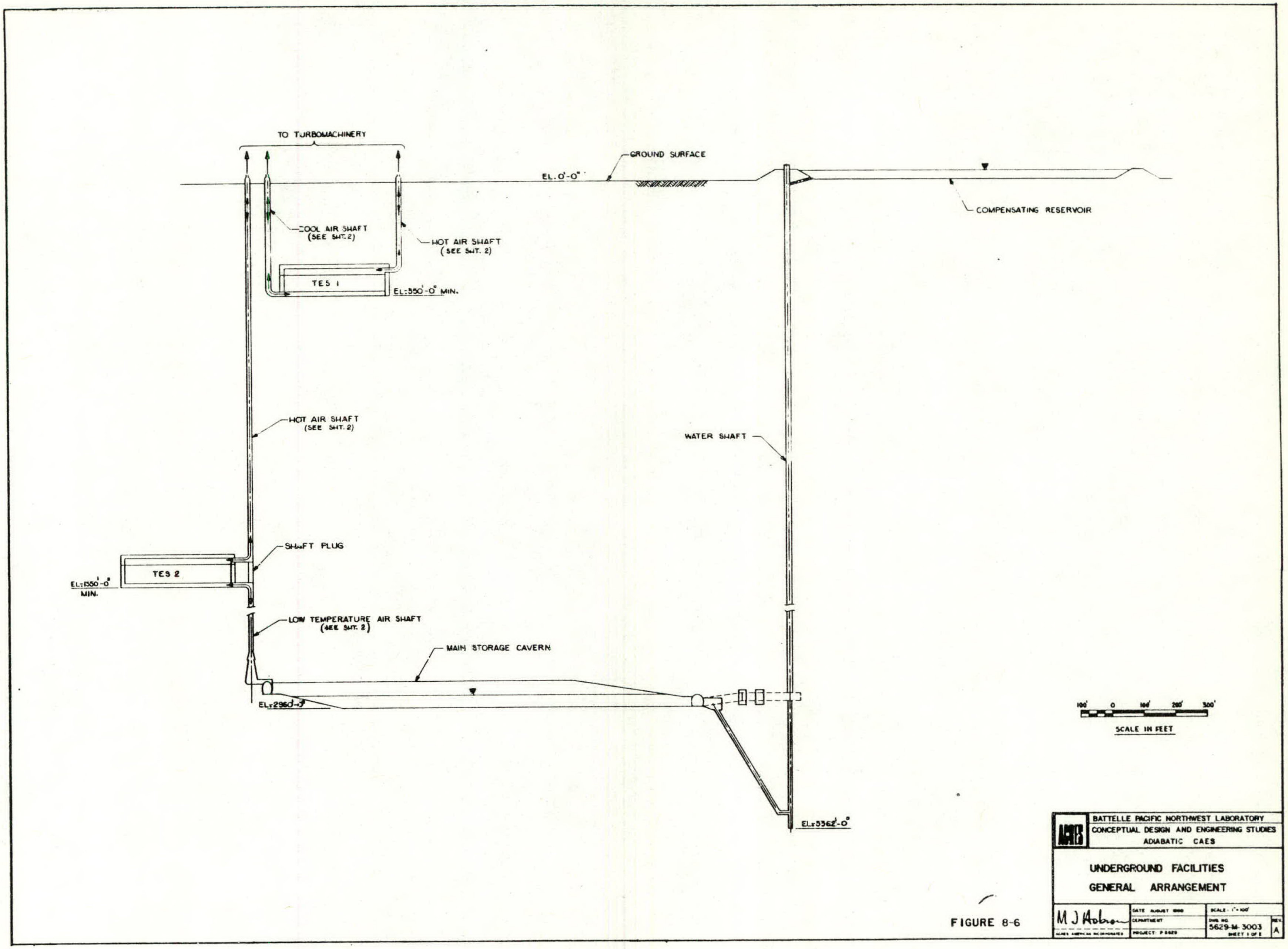
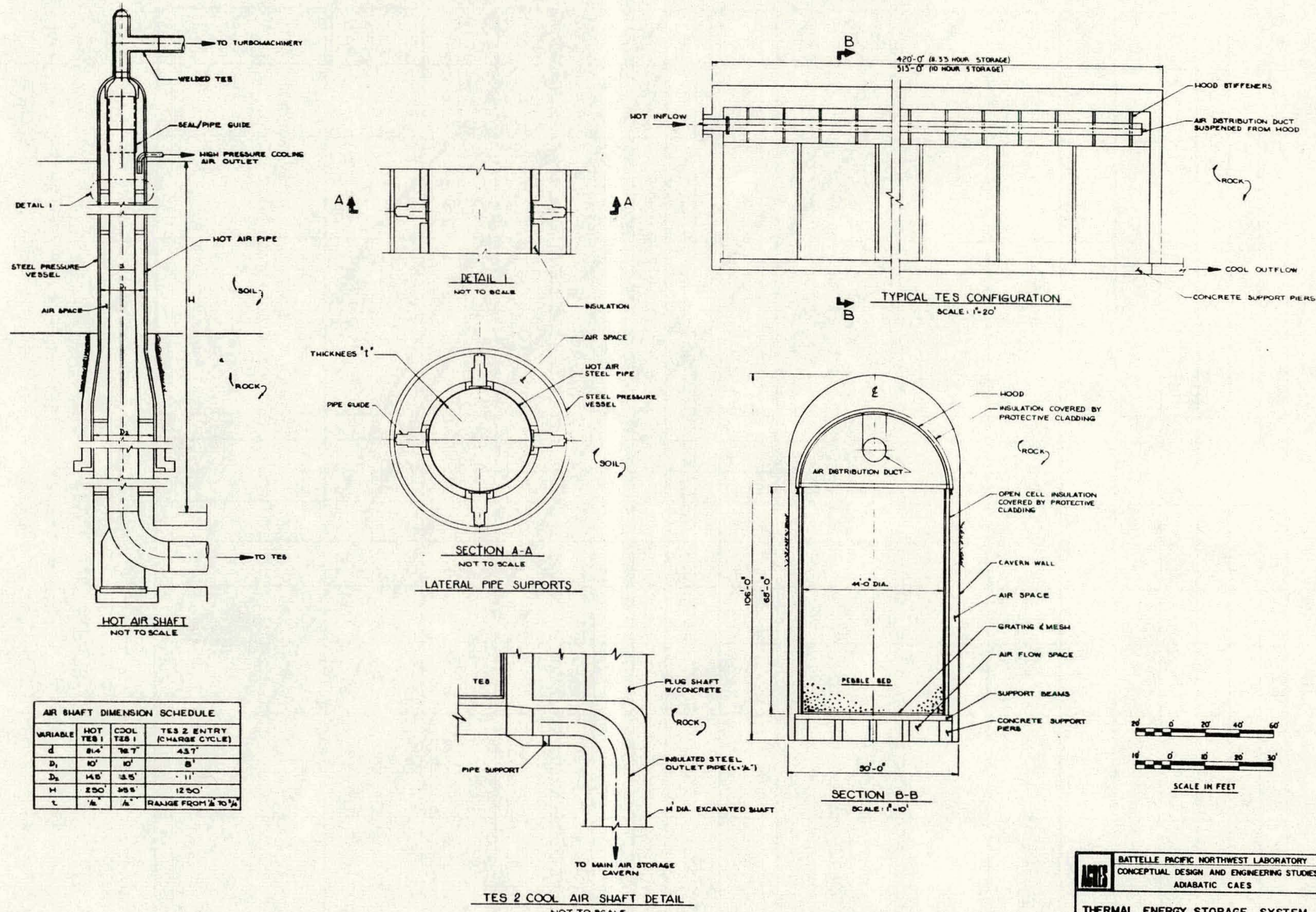

TO TURBOMACHINERY
GROUND SURFACE
EL. 0'-0"
COMPENSATING RESERVOIR
COOL AIR SHAFT (SEE SHT. 2)
HOT AIR SHAFT (SEE SHT. 2)
TES 1
EL: 550'-0" MIN.
HOT AIR SHAFT (SEE SHT. 2)
WATER SHAFT
SHAFT PLUG
TES 2
EL: 1550'-0" MIN.
LOW TEMPERATURE AIR SHAFT (SEE SHT. 2)
MAIN STORAGE CAVERN
EL: 2960'-3"
EL: 3362'-0"
100' 0 100' 200' 300'
SCALE IN FEET
BATTELLE PACIFIC NORTHWEST LABORATORY
CONCEPTUAL DESIGN AND ENGINEERING STUDIES
ADIABATIC CAES
UNDERGROUND FACILITIES
GENERAL ARRANGEMENT
5629-M-3003
A

FIGURE 8-6



AIR SHAFT DIMENSION SCHEDULE			
VARIABLE	HOT TES 1	COOL TES 1	TES 2 ENTRY (CHARGE CYCLE)
d	81.4"	78.7"	43.7"
D_1	10'	10'	8'
D_2	14.5'	13.5'	11'
H	250'	455'	1250'
t	[illegible]	[illegible]	RANGE FROM [illegible] TO [illegible]

ACRES

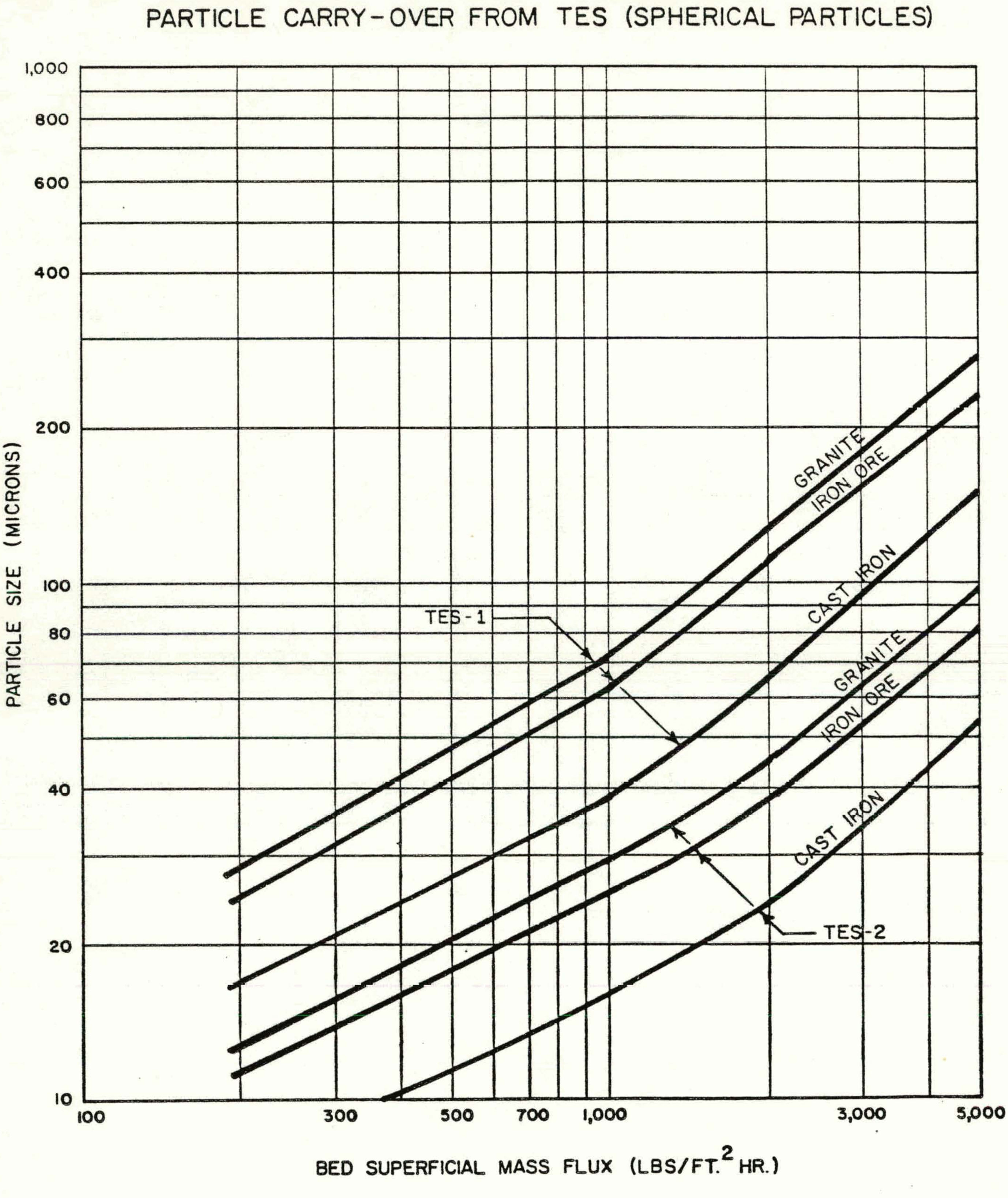
BATTELLE PACIFIC NORTHWEST LABORATORY
CONCEPTUAL DESIGN AND ENGINEERING STUDIES
ADIABATIC CAES

THERMAL ENERGY STORAGE SYSTEM
CONCEPTUAL DESIGN

M J Hobson

DATE AUGUST 1980 · SCALE AS SHOWN · DWG NO. 5629-M-3003 · REV. A

FIGURE 8-7



BATTELLE PACIFIC NORTHWEST LABORATORY
CONCEPTUAL DESIGN AND ENGINEERING STUDIES—ADIABATIC CAES
ACRES
M J Hobson
ACRES AMERICAN INCORPORATED
FIGURE 8-8

9 - COST AND SCHEDULE

9 - COST AND SCHEDULE

This section presents the conceptual estimates for capital and operating cost, a preliminary project schedule, levelized production costs for plant output power, and a discussion of plant economics.

9.1 CAPITAL COST ESTIMATE

Capital cost estimates were developed for the conceptual plant design (described in Section 8) based upon preliminary cost estimates prepared during the PEPCO study. Several variations in plant capacity and TES fill material were examined.

Table 9-1 presents the estimated cost of an 8.3 hour storage capacity adiabatic CAES plant. Air storage is in excavated hard rock caverns with heat storage in iron oxide pebbles. The cost estimate for a 10-hour adiabatic plant are presented in Table 9-2 and comparative costs for the 10-hour oil-fired CAES design are presented in Table 9-3 under a different code of accounts. In all cases for adiabatic CAES, the TES volume requirements are based upon CEGB program results. The changes in thermal energy storage system cost with alternative materials (i.e., granite, Denstone, and cast iron) are also indicated for each plant capacity.

A general breakdown of the TES costs for the 8.3-hour plant design with heat storage in iron oxide pebbles is shown in Table 9-4. In the cases of granite and Denstone fill materials, which are less dense than iron oxide pebbles, the basic structural design of the fill containment cylinders and support structures for the iron oxide system are valid and were retained for costing purposes. The basic difference in TES designs for these fill materials was the size of bed approach area required to achieve equivalent thermal performance (i.e., end temperature difference) for the given bed depth. Cast iron fill, however, would require substantial modifications to the TES containment design to accommodate the much greater structural loads. The TES cost estimate for cast iron fill therefore represents a rough cost extrapolation of the iron oxide case, and reflects a nearly four-fold increase in containment costs in addition to a substantially higher fill cost.

In Table 9-5, direct cost and total cost per kilowatt of power generated are compared for adiabatic CAES, fuel-fired CAES, and underground pumped hydro facilities. The sensitivity of capital costs to the selection of heat storage materials for adiabatic CAES is apparent in this table.

9.2 CONSTRUCTION SCHEDULE

A preliminary construction schedule for the adiabatic CAES plant is shown in Figure 9-1. Construction of the thermal energy storage facilities extends the PEPCO study CAES plant construction period by approximately one year.

9.3 OPERATING COSTS

Plant payroll and maintenance expenses were estimated at $4.00/kW-year fixed costs and 0.28 mills/kWh variable costs (1980 dollars) for adiabatic CAES. These operating costs were based upon estimates of plant staff (68 persons) and maintenance requirements, which incorporated information provided by Dresser Clark together with estimates of additional routine services required to keep the plant on line.

9.4 LEVELIZED ENERGY COSTS

Levelized annual busbar costs were calculated using the methodology presented in the EPRI Technical Assessment Guide (EPRI, 1978) and cost assumptions prepared by Battelle PNL for CAES studies (Table 9-6). The analysis assumed a 1990 start of plant operations. The capital cost estimate for the completed plant incorporated an indirect cost multiplier of 39 percent on direct costs for contingencies, engineering, construction management, and utility costs.

Table 9-7 compares the results of the levelized energy cost analysis for adiabatic CAES, oil-fired CAES, and combustion turbines for various capacity factors and a uniform plant life of 30 years. The results indicate that adiabatic CAES is competitive with oil-fired CAES if low cost materials are used for heat storage.

9.5 SENSITIVITY ANALYSIS

A number of sensitivity tests were performed for such factors as plant life, TES fill material selection, storage capacity, charging energy cost, and cycle arrangement.

9.5.1 Plant Life - At this stage of development, the actual life of an adiabatic CAES plant is difficult to project. The underground facilities (excluding TES) will certainly outlast the surface equipment. The cyclic life of the thermal storage fill material is uncertain. The life of the TES containment and underground piping will depend upon TES material behavior. The mean time between overhauls of the turbomachinery, since no combustion is involved, may be much longer than for combustion turbines because of less severe thermal cycling and gas path corrosion factors. Erosion life of the blades will depend primarily upon the range of particulate size generated by TES materials and particle separator efficiency. It is reasonable to suppose, therefore, that, if storage material life is adequate, particulate erosion can be effectively eliminated by high efficiency separation and cyclic fatigue is reduced by low temperature operation of rotating equipment, the life of an adiabatic CAES plant may extend considerably beyond conventional CAES assumptions.

To reflect this possibility, the CAES plant life assumption of 25 years was supplemented with a 50-year life. The results, as shown in Table 9-8, indicate approximately 10 percent improvement in levelized energy cost. As levelized cost calculations involve present worth analysis, comparisons of plants with differing lives must be evaluated over a common base period. These calculations were performed for a term of 50 years and include capital cost adjustments to reflect complete plant replacement of mechanical components, including TES, for the 25-year plant.

An analysis was also performed to assess the impact of replacing TES fill materials every 10 years. With an operating lifetime of 30 years used as a basis for this analysis, two fill replacements were required. The results indicated that the levelized annual busbar cost of the plant would be increased 6.1 mills/kWh (or 2.7 percent) for an annual capacity factor of 20 percent for an iron oxide fill TES design. The analysis assumed that replacement activity would take six months to access the TES, replace the fill, and return the plant to service. During this period, the cost of replacement power in the form of baseload electricity was factored into the cost of fill replacement since the adiabatic CAES plant would be rendered inoperable. If only combustion turbines were available to generate this make-up power, then the levelized busbar costs would be increased 44.0 mills/kWh over the base case (or approximately 30 percent) at 20 percent capacity factor. The added cost of fill replacement would probably fall somewhere between these two extremes depending on the generation mix of make-up power available, if indeed it is available. These estimates disregard any loss of power output as a result of reduced thermal performance that may be attributed to fill degradation between replacement years. The general conclusion is that the cost impact of TES fill replacement can be small or very large depending on the specific circumstances.

that the cost impact of TES fill replacement can be small or very large depending on the specific circumstances.

9.5.2 Fill Material and Plant Storage Capacity - The selection of fill material affects the size of TES containment required, as different materials exhibit different thermal performance characteristics. Accounting for these differences, calculations were performed to determine the sensitivity of levelized costs to changes in TES fill materials in particular and capital cost variations in general. The effect of fill selection and storage capacity on levelized costs are shown in Table 9-7 for 30-year adiabatic CAES plants. The table indicates that adiabatic CAES with the lower cost fill materials (i.e., iron oxide and Denstone) appears competitive with oil-fired CAES, but that it is marginally competitive with the more expensive cast iron fill.

9.5.3 Charging Energy Cost - In the PEPCO study of oil-fired CAES and UPH, the cost of charging power was defined as 20.9 mills/kWh (1980 dollars) as opposed to the 11.4 mills/kWh (1980 dollars) assumed for the levelized cost calculations of this study. This higher charging power cost was also examined for the case of 10 hours of storage, iron-oxide fill, and a plant life of 30 years. With this figure, levelized energy cost would increase by 129.8 mills/kWh (or 58 percent) for 20 percent capacity factor, while the levelized cost of oil-fired CAES would increase 65.8 mills/kWh (or 25 percent). With the 11.4 mill/kWh base rate, levelized costs for adiabatic CAES are approximately 14 percent less than oil-fired CAES, while adiabatic CAES costs with the higher charging power cost are approximately 8 percent higher than oil-fired CAES at 20 percent capacity factor.

Like adiabatic CAES, the UPH plant design is also very sensitive to charging power costs. The levelized cost of the 2000 MW PEPCO study UPH plant design would increase 79.7 mills/kWh (or 36 percent) using the PEPCO charging power rate. For adiabatic CAES (25 year plant life) the 50-year levelized cost would increase 74.6 mills/kWh (28 percent) at 20 percent capacity factor.

A calculation was also performed to compare oil-fired CAES with adiabatic CAES assuming equal escalation rates (7 percent) for oil and charging energy. Levelized cost of oil-fired CAES at 20 percent capacity factor was reduced 22.8 mills/kWh, or approximately 9 percent below the value presented in Table 9-7 while no effect was produced for the adiabatic CAES results. This calculation suggested that the cost of oil-fired CAES and adiabatic CAES with Denstone fill would be approximately equal if the cost of baseload electricity was assumed to track the cost of oil (i.e., with no escalation rate differential assumed).

Table 9-1

ESTIMATE			
CLIENT	BATTELLE - PNL	TYPE OF ESTIMATE	Conceptual
PROJECT	Adiabatic-CAES in Hard Rock - 8.3 Hr.	APPROVED BY	
JOB NUMBER	P5629.00	FILE NUMBER	" .153
SHEET	1	OF	4
BY	James	DATE	9/80
CHKD	Driggs	DATE	10/80

No.	DESCRIPTION	QUANTITY	UNIT	COST/UNIT	AMOUNT	TOTALS	REMARKS
20.	LAND AND LAND RIGHTS				16,496,000		
21.	GENERATION/COMPRESSION SYSTEM				230,249,000		
22.	COMPRESSED AIR STORAGE				55,120,000		
23.	THERMAL ENERGY STORAGE - IRON OXIDE FILL				54,199,000		
25.	ELECTRIC PLANT EQUIPMENT				67,094,000		
26.	GENERAL PLANT FACILITIES				10,572,000		
	DIRECT COST					433,730,000	July 1980 Dollars
	COST/KW					542	
23.	THERMAL ENERGY STORAGE - ROCK FILL				51,129,000		
23.	THERMAL ENERGY STORAGE - DENSTONE FILL				94,872,000		
23.	THERMAL ENERGY STORAGE - CAST IRON FILL				196,439,000		

57.02.03. Form 134A

Table 9-2

ESTIMATE

CLIENT: BATTELLE - PNL

PROJECT: Adiabatic-CAES in Hard Rock - 10 Hr.

TYPE OF ESTIMATE: Conceptual

APPROVED BY:

JOB NUMBER: P5629.00
FILE NUMBER: " .153
SHEET 2 OF 4
BY: James DATE: 9/80
CHKD: Driggs DATE: 10/80

No.	DESCRIPTION	QUANTITY	UNIT	COST/UNIT	AMOUNT	TOTALS	REMARKS
20.	LAND AND LAND RIGHTS				16,496,000		
21.	GENERATION/COMPRESSION SYSTEM				230,249,000		
22.	COMPRESSED AIR STORAGE				64,410,000		
23.	THERMAL ENERGY STORAGE - IRON OXIDE FILL				59,749,000		
25.	ELECTRIC PLANT EQUIPMENT				67,094,000		
26.	GENERAL PLANT FACILITIES				10,572,000		
	DIRECT COST					443,570,000	July 1980 Dollars
	COST/KW					561	
23.	THERMAL ENERGY STORAGE - ROCK FILL				56,065,000		
23.	THERMAL ENERGY STORAGE - DENSTONE FILL				108,556,000		
23.	THERMAL ENERGY STORAGE - CAST IRON FILL				222,804,000		

57.02.03. Form 134A

Table 9-3

ACRES	**ESTIMATE**		
	CLIENT: BATTELLE - PNL	TYPE OF ESTIMATE: Preliminary	JOB NUMBER: P5629.00
			FILE NUMBER: " .153
	PROJECT: Adiabatic CAES in Hard Rock	APPROVED BY:	SHEET 3 OF 4
			BY James DATE 9/80
			CHKD Driggs DATE 10/80

No.	DESCRIPTION	QUANTITY	UNIT	COST/UNIT	AMOUNT	TOTALS	REMARKS
	COMPARISON COST ESTIMATE - 924 MW/10 HR FUEL FIRED PEPCO CAES PLANT						
10.	CONSTRUCTION PREPARATION				14,996,000		
20.	SURFACE FACILITIES				16,007,000		
30.	STORAGE SYSTEMS				58,263,000		
40.	GENERATION/COMPRESSION				172,580,000		
50.	MECHANICAL B.O.P.				30,751,000		
60.	SWITCHYARD				37,814,000		
70.	PLANT ELECTRICAL EQUIPMENT AND SYSTEMS				14,498,000		
	SUBTOTAL					344,909,000	July 1979 Dollars
	10% ESCALATION				34,491,000		
	DIRECT COST, 7/80					379,400,000	

57.02.03. Form 134A

Table 9-4

ESTIMATE

CRES

CLIENT: BATTELLE - PNL

PROJECT: Adiabatic CAES in Hard Rock

TYPE OF ESTIMATE: Conceptual

APPROVED BY:

JOB NUMBER: P5629.00
FILE NUMBER: P5629.00.153
SHEET 4 OF 4
BY James DATE 9/80
CHKD Driggs DATE 10/80

No.	DESCRIPTION	QUANTITY	UNIT	COST/UNIT	AMOUNT	TOTALS	REMARKS
23.	THERMAL ENERGY STORAGE						
.1	Storage Vessel Construction				18,858,000		Includes rock cavern excavation, structural support, silo fabrication
.2	Storage Medium				8,889,000		Iron oxide fill
.3	Air Shaft Construction				13,635,000		Includes shaft excavation, air shaft piping, and air shaft caps.
.4	Surface Piping/Valving				9,717,000		Excludes main control valves.
.5	Auxiliary Equipment				3,100,000		Includes mechanical separators and instrumentation.
	TOTAL ACCOUNT 23					54,199,000	

57.02.03. Form 134A

Table 9-5

Comparison of Capital Costs

Adiabatic CAES (800 MW)	Direct Cost, $000	1980 Total Cost* $/kW
Iron Oxide Fill		
8.3 hour storage	433,730	542
10 hour storage	448,570	561
Granite Fill		
8.3 hour storage	430,660	538
10 hour storage	444,886	556
Denstone Fill		
8.3 hour storage	474,403	593
10 hour storage	497,377	622
Cast Iron Fill		
8.3 hour storage	575,970	720
10 hour storage	619,257	774
Oil-Fired CAES (924 MW)		
10 hour storage	379,400	411
UPH (2000 MW)		
10 hour storage	915,263	458

All costs in July 1980 Dollars

* Present worth at 10 percent discount rate.

Table 9-6

Economic Assumptions for CAES Analysis

Parameter	Units	Value
Fixed Charge Rate	%/yr	18%
Discount Rate	%/yr	10%
Inflation Rate	%/yr	6%
Escalation Rates		
Petroleum Fuels	%/yr	8%
Coal	%/yr	7%
Baseload Electricity	%/yr	7%
Capital Equipment/Construction	%/yr	6%
Operations/Maintenance Costs	%/yr	6%
Energy Prices - Jan. 1, 1980		
Distillate #2	$/10^6 Btu	5.24
Residual Oil	$/10^6 Btu	4.33
Coal	$/10^6 Btu	1.34
Baseload Electricity	Mills/kWh	11.4
System Lifetime	years	30
Start of Operations		1990
Base Year for Cost Estimates		1980

TABLE 9-7

Comparison of Oil-Fired CAES, Adiabatic CAES, and Combustion Turbines

1990 Levelized Busbar Cost (Mills/kWh)

Capacity Factor (percent)	Oil-Fired CAES[a] (10-hour storage)	Adiabatic CAES[b] Iron Oxide Fill (8.3-hour storage)	Adiabatic CAES[b] Iron Oxide Fill (10-hour storage)	Adiabatic CAES[b] Denstone Fill (10-hour storage)	Adiabatic CAES[b] Cast Iron Fill (10-hour storage)	Combustion Turbine[c]
10	369.9	368.3	378.2	410.6	492.3	446.4
15	297.6	269.4	276.0	297.6	351.8	409.6
20	261.4	219.9	224.9	241.1	281.7	391.2
25	238.6	---	194.2	207.2	239.7	380.0

(a) 924 MW output, 30-year life, water compensated hard rock air caverns.

(b) 800 MW output, 30-year life, water compensated hard rock air caverns.

(c) 50 MW output, 30-year life, 300 $/kW capital cost (1990), 2.0 mills/kWh variable O&M, 140 mills/kW-yr fixed O&M, 12,000 Btu/kWh heat rate (1980 dollars).

TABLE 9-8

Comparison of Adiabatic CAES and UPH for 50-Year Period

1990 Levelized Busbar Cost (Mills/kWh)

Capacity Factor	UPH[a]		Adiabatic CAES[b]	
(percent)	2000 MW	1000 MW	25-Year Life[c]	50-Year Life[d]
10	292.4	355.7	447.7	397.0
15	221.7	263.9	328.6	294.8
20	186.4	218.1	269.1	243.8
25	165.2	190.5	233.1	213.0

a) 10-hour storage capacity, 50-year plant life

b) 800 MW output, 10-hour storage capacity, iron oxide fill

c) Mechanical components are replaced after 25 years of operation to achieve 50-year plant operating period.

d) Mechanical components are assumed to last 50 years before retirement.

PRELIMINARY ACAES CONSTRUCTION SCHEDULE

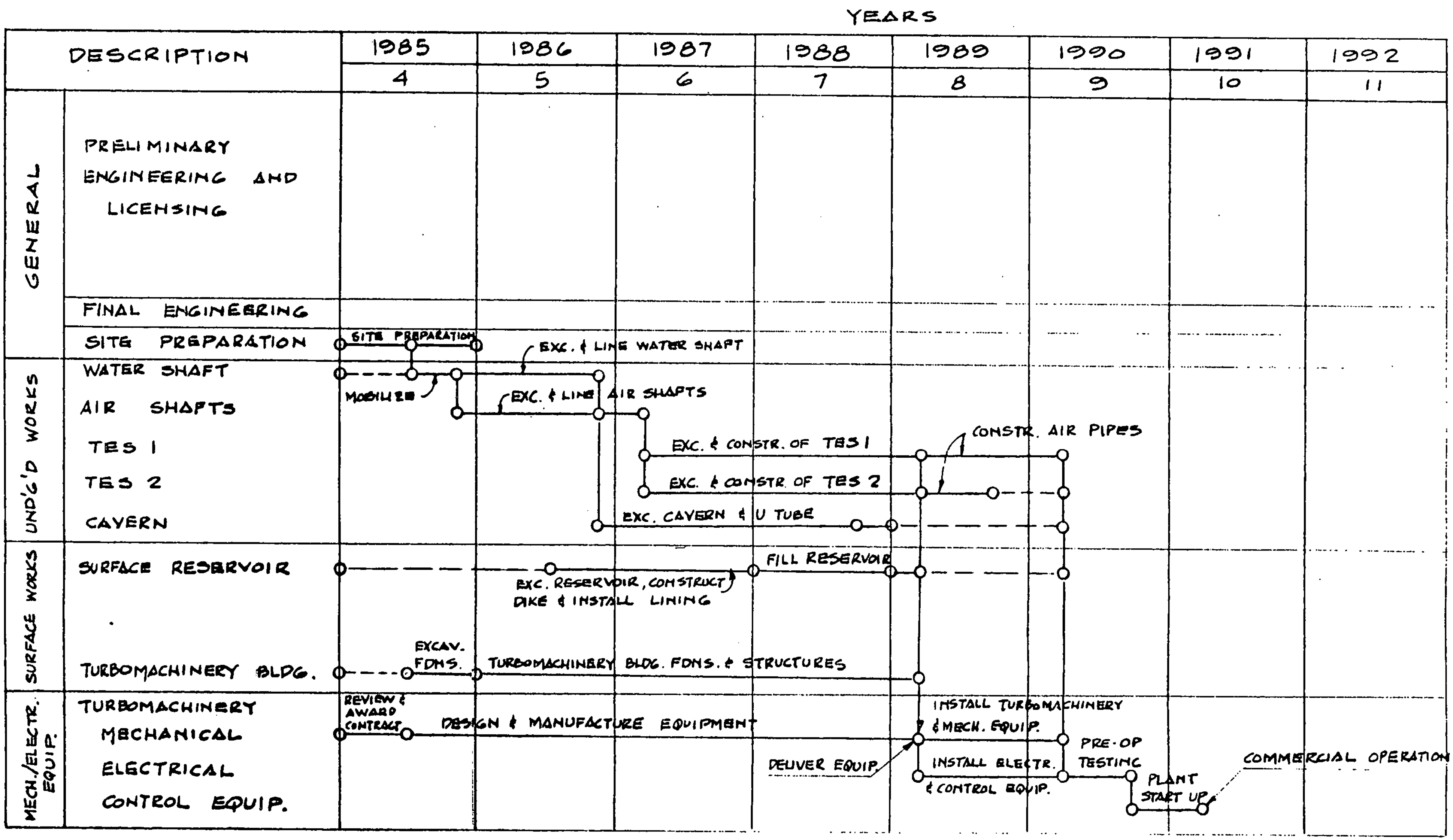


FIGURE 9-1

ACRES

10 - FEASIBILITY ASSESSMENT AND DEVELOPMENT NEEDS

10 - FEASIBILITY ASSESSMENT AND DEVELOPMENT NEEDS

10.1 AREAS REQUIRING FURTHER DEVELOPMENT

The following is a discussion of a number of plant design aspects which should be investigated further if more detailed engineering efforts are undertaken:

10.1.1 TES Behavior - Although heat storage in large pebble bed regenerators remains to be demonstrated, a number of small-scale versions used for preheating air in wind tunnels give credence to the design. To assess the feasibility of using pebbles for large-scale regenerators, as proposed for adiabatic CAES in this and other studies (Hamilton, 1978; CEGB, 1979), a pebble testing program should be initiated.

The goal of the program should be the identification (by experimental evaluation) of pebbles suitable for the bed configuration, operating conditions, and material life expected of a near-term adiabatic CAES facility. Emphasis should be placed first on the evaluation of low cost pebbles (such as iron oxide and various forms of rock) because of the massive bed volumes required. Materials should be tested under thermal cycling conditions in vessels of sufficient size and proportion to permit accurate modeling of full-size regenerators.

The testing program should set out to accomplish the following objectives:

- Establish rates of pebble breakup and attrition resulting from pebble movement, thermal shock, bed stresses, and the presence of moisture.
- Measure stresses induced on the containment cylinder by the bed mass.
- Determine size distribution of elutriated particles in relation to air mass flux.
- Evaluate bed performance with varying flow rate.
- Measure bed voidage.
- Experimentally verify regenerator models.

Bed stresses will result from pebble loads which are dependent on the height of the packed bed and differential thermal expansion. An empirical model should be developed to predict these bed stresses and pebble movements. Once having this model, criteria for the design of bed configurations, in accordance with stress limits of the selected storage media, can be developed based on experimental data. With this capability, a true optimum design for acceptable bed configuration, TES thermal performance, and minimum volume can be achieved.

10.1.2 Cycle Optimization - The two-stage design cycle selected in this study was optimized on the basis of existing equipment designs available from the Clark Division of Dresser Industries. The existing equipment limitation of the study scope has, however proved to be a significant constraint in the development of a plant design. Although the compressor exit temperature limits identified are considered representative of commercial products in the industry, they are far below the capability of the expander turbines offered by Dresser Clark and turbine inlet temperatures typical of the gas turbine industry. If the extreme of 1400 to 1500°F compressor discharge temperatures are considered, turbine output increases substantially; without efficiency changes, a 50 percent increase would be expected on temperature considerations alone. Actual increase in output would be dependent upon a number of other factors. As detailed analysis of cycles requiring extensive compressor development was, however, beyond the scope of this study, a more detailed discussion may be found in references (Glendenning, 1979; Flynn, 1979; and Giramonti, 1979).

Consideration of higher temperatures and correspondingly higher pressures in quest of higher efficiency for the two-stage cycle is limited by economics, risk and technical feasibility. Use of a two-stage cycle with full TES intercooling between 1000°F compression stages would likely involve a storage pressure of some 2000 psia. The economic practicality, much less the technical feasibility, of such a design must be questioned.

Although such piping system pressures are common to coal-fired baseload steam plants, the adiabatic CAES plant requires long runs of large diameter thick wall pipe with the result that such a piping system would be costly. Storage system depth would also increase from 2900 feet to 4800 feet to accommodate the higher pressures, and thermal expansion of the air shaft pipe would become a much greater problem. Electric energy ratio would also be higher (poorer) than for the design system. Unless advances in compressor efficiency are also achieved in the machinery design, centrifugal units with inherently lower efficiency would be required for the high pressure compression. In summary, analytical considerations aside, practical limitations on facility cost and construction feasibility would suggest that development of higher temperature cycles would eventually force the use of a single all-axial stage of compression. This approach was recommended by Glendenning for long-term development.

Adiabatic CAES cycle optimization for the commercially available equipment used in this study incorporated a rough cost tradeoff analysis which identified the two-stage cycle as the most desirable (Section 4.2). Re-examination of this selection in light of the cost assumptions specified by PNL indicated that a single-stage cycle may also be acceptable if initial cost per kilowatt is of less value than overall cost. The reduction of operating pressure from 1200 psia to 240 psia would increase plant capital cost by some $100 million, would double the cost per kW (output), and would reduce plant capacity from 800 to 500 MW. However, elimination of the centrifugal compressors and the high-pressure turbine increases average machinery efficiency to improve the electric energy ratio (EER) to 1.38. Under these conditions the levelized busbar cost of the low pressure design is roughly equivalent to that of the combustion turbines at 20 percent capacity factor. If the air storage system cost can be reduced, perhaps through conversion of an existing mine, levelized busbar costs approaching a comparably sized two-stage plant may be feasible. The economics of the low--pressure storage approach also become more favorable for utilities with charging power costs greater than the 11.4 mill base value due to the more favorable cycle efficiency. The identification of existing cavities capable of conversion to air storage at pressures of around 250 psi could therefore be important to moving adiabatic CAES technology towards commercialization.

If plant output is increased to 700 MW (with no capital cost change), which might be expected with compressors capable of temperatures of 1400°F, levelized cost for the single stage design is approximately 280 mills/kWh. This is roughly comparable to the two stage cycle with cast iron TES fill. Development of axial compressor designs for high temperatures with efficiency equal to or better than the units selected would therefore benefit the single-stage cycle economics. The improvement in cycle efficiency and the increased storage pressure (resulting in storage system excavation cost savings) could significantly reduce levelized energy cost to the point where the low pressure cycle is better than the two stage approach. Such a temperature increase may require substantial engineering development, but should be investigated further if only to quantify the effort required.

10.1.3 Development of Dresser Clark Machinery for Higher Temperatures - Dresser Clark identified the main design features in their machinery that would require engineering redesign and development for higher temperature cycle applications. These are:

10.1.3.1 Axial Compressor Limits - Axial compressor discharge temperatures in the range of 900°F to 1400°F require:

(1) the use of austenitic stainless steels for casing fabrication;

(2) limitation of casing diameter to less than 100 inches to remain in this allowable stress limits;

(3) increased blade tip and seal clearances due to differences in rotor and casing thermal expansion rates;

(4) a revised method of attachment to the shaft for the radial last stage other than shrink fit, or replacement by two axial stages;

(5) redesign of the stator housing in terms of mounting method and materials, most probably with the techniques used in construction of Dresser expander turbines; and

(6) evaluation of the need for cooled rotors, as used in some Dresser expander turbines.

10.1.3.2 Centrifugal Compressors - Centrifugal compressor discharge temperatures in the range of 870°F to 1400°F will require:

(1) the use of austenitic stainless steels for casing fabrication;

(2) heavier wall casings than for the same pressure at lower temperature because of lower allowable stresses;

(3) increased rotor and seal clearances, with attendant reductions in efficiency; and

(4) substantial redesign of machine rotors to incorporate different rotor to shaft attachment methods and different wheel materials.

10.1.3.3 Expander Turbines

Expander turbine inlet temperature increases to 1400°F, and the attendant pressure increases, may involve use of:

(1) spherical casings for the high pressure machines; and

(2) shaft seal modifications.

10.1.4 Gearbox - The gearbox selection, as discussed in Section 6, is a conceptual design based upon operating units of similar design but lower capacity. The manufacturer appears confident that the design is well within manufacturing capability. A detailed investigation of this unit should be performed if use of the present plant design is considered further. Pitch line velocity and the power transfer involved rank this gearbox design near the limit of proven gearbox technology.

10.1.5 Particulate Erosion - Air passage through the TES and piping system will probably result in particulate carryover into the turbines and centrifugal compressors. Separators will be incorporated for particulate removal, but inevitably there will be some percentage of carryover that will reach the stationary and moving blades. Erosion of the turbine blading by solid particles that may be carried over from the TES is therefore a real possibility.

The rate of erosive blade wear is directly proportional to the mass of the particles and to the square of the particle velocity (Smeltzer, 1970). Hardness of the particle is also thought to play a role. Expanders installed in refinery fluid catalytic cracker unit (FCCU) service provide most of the experience with particulate blade erosion.

For FCCU service, the accepted maximum particle concentration is 160 ppm, and 95 percent of the particles must be smaller than 10 microns. For these conditions a blade life of 40,000 hours is commonly achieved (Dziewulski, 1978). However, process upsets, which sometimes drastically increase the particle concentration as well as mean particle size, can significantly reduce blade life.

In order to achieve a 40,000 hour blade life, the turbine blading must be designed for low relative velocity; and certain design features are necessary to protect the blade roots. Chief among these are steps incorporated into the hub and shroud immediately following the blade rows (Jericha, 1972; and Nabors, 1964). These steps are sometimes referred to as stumbling steps and their function is to absorb the energy of the particles passing through the blade clearances. A number of machines with stumbling steps have been installed in Europe in FCCU service, one of which has surpassed 40,000 hours of operation. Dresser Clark's first FCCU unit was undergoing start-up during this project.

Stumbling steps cause a reduction of turbine efficiency of two to three percentage points. If shrouded blades were used, stumbling steps would not be needed as the particles would follow the gas flow path. With shrouds, turbine efficiency is one to two points greater than the same design without shrouded blades. However, the shrouded blade approach is still under development and is as yet unproven. Therefore, if erosive particles in sufficient quantities are generated in the TES beds, a turbine designed for adiabatic CAES will likely incorporate stumbling steps. Stumbling steps have not been included in the proposed design so as to maximize efficiency, as particle generation in the TES beds is not known to be of concern.

A comparison of Dresser's acceptable limits on particle concentration and size distribution to the recommendations of researchers in pressurized fluidized bed combustion turbine development is of interest. General Electric researchers have proposed limits of 100 ppm, with 98 percent removal of particles below 10 micron. These limits are expected to allow blade life (determined by failure of 10 mil thick coatings) in a PFB fired MS 7000 turbine of 25,000 hours.

However, Dresser's design limits gas path velocity to conditions which are lower than conventional combustion turbine practice. Through the final stage of the low pressure turbine (at pitch diameter) exit velocity is 1080 fps, in comparison to some 1300 fps in the last stage of the GE MS 7000 turbine.

Discussions with GE researchers (Gilas, 1980) regarding both reported data and GE internal research has indicated that blade wear is believed to be subject to a threshold particle size for given velocities. The data which led to the previously cited GE proposed limits was taken at 1200 fps, and showed a threshold particle size of 9 microns for fly ash particles. Reportedly, ongoing tests at the Leatherhead (England) PFB combustor test facility appear to corroborate this relationship, but more work is yet to be done. At present, no correlations have been obtained regarding particle hardness, but shape does not appear significant. Although these results are yet to be confirmed with completion and analysis of the tests, some comparisons with the Dresser Clark separation requirements and life projections are of interest.

Assuming particles of equal size, an iron oxide particle would have approximately four times the mass of one of fly ash (300 lbs/ft^3 vs. 75). Thus, if threshold size for a fly ash/PFB ash particle in the GE tests is approximately 9 microns at 1200 fps, threshold size for an iron oxide particle would be some 6 microns, based on particle kinetic energy. Reducing velocity from 1200 to 1000 fps would increase threshold particle size to some 8 microns. Use of Denstone as TES fill material would produce a threshold particle size for the GE machine at 1200 fps of some 7 microns (density 150 lbs/ft^3) to meet a design life of 25,000 hours.

Dresser Clark's expander design for FCCU service (with stumbling steps) is based upon a catalyst laden stream (catalyst specific gravity of 2.0, or 125 lbs/ft^3 density) with a maximum upstream (separator exit) concentration of 160 ppm and no less than 95 percent removal of all particles 10 micron or greater, to produce a blade life of 40,000 hours. Thus, iron oxide TES fill would appear to necessitate removal of some 90 to 95 percent of all particles 8 microns and larger, and Denstone would require removal of particles 9 microns and larger.

The Dresser design also incorporates greater reaction in the stage blading than is practiced in combustion turbine design, although this approach would likely be used as well in the design of a PFB fired combustion turbine. Fifty percent reaction at the mean diameter is reportedly a characteristic approach for cat cracker expander service, as the reduced blade turning angles and relative velocities aid in erosion control (Burton, 1980).

Service life of the turbine section, therefore, will be largely subject to separator removal efficiency. It is important to note, however, that the adiabatic CAES plant, using a 20 percent capacity factor, would need a major overhaul every 12 to 14 years with a 25,000 hour operating life, and 19 to 22 years with a 40,000-hour life.

The high-pressure centrifugal compressors would also be subjected to particulate laden air from the low pressure thermal storage system. The penalty of excessive erosion in these units would be inability to make discharge pressure, with the result that plant operation would be impossible. Hardenable steels could be used for rotor fabrication in these units at added cost, in addition to particle separators. The Dresser Clark designs incorporate backward leaning impeller vanes, which should aid in reducing erosion, and their machines are commonly used in "dirty" air service.

Particle separator performance (and consequently design) will therefore be important in the development of an adiabatic CAES plant design if TES bed particle generation is shown to be significant. As discussed in Section 8, contacts with separator manufacturers indicated that cost was likely to be the limiting factor rather than technical capability in developing separator designs for adiabatic CAES service. Before separators can be adequately designed and the potential severity of machinery erosion can be identified, potential particle generation from the TES bed must first be determined.

10.2 APPLICATION TO UTILITY NEEDS

The economic analysis presented in Section 9 represents a significantly different approach from that taken in the PEPCO/DOE/EPRI oil-fired CAES and UPH study (Acres, 1980) wherein storage plant operation was simulated within a real system. The PEPCO study economic analysis was based not only upon cost factors nearly identical to those used in this levelized cost analysis, but also upon the operating characteristics of both the UPH and CAES plant designs and the PEPCO system.

The analysis of oil-fired CAES and UPH performed by PEPCO incorporated the charging power requirements of each technology in a simulation to determine available charging capacity for each plant type. As a reference, UPH requires 925 MW to operate one 666 MW unit in the pumping mode and is essentially incapable of load variation; fuel-fired CAES requires 510 MW to operate three 230 MW units in the compressing mode, with a turndown capability of 25 percent.

PEPCO is a medium size utility (projected capacity of more than 5000 MW in 1990) with a load dominated by residential and office building customers. This results in significant load variation between day and night. Energy storage was identified in past studies as a plant type which could potentially meet PEPCO's system needs for peaking power while improving base load plant utilization. These findings formed the basis for PEPCO's participation in the recent study.

When modelled as part of the PEPCO system, the UPH plant was penalized by the limited availability of low- cost charging energy. This significantly reduced the contribution of the UPH plant to the PEPCO system relative to the CAES plant. Thus, available charging capacity, cost of charging power, and the performance limitations of the CAES and UPH plants combined to produce nearly equal relative economy with operation of either design within the PEPCO system. Oil-fired CAES was, however, slightly more economic and produced slightly greater oil savings.

These results were achieved despite the fact that the PEPCO analysis used a differential escalation rate between oil and coal of 2 percent (9 vs. 7) and higher capital costs (as a result of a more severe escalation assumption) than used in the analysis of Section 9. Generation costs for CAES and UPH were nearly equal at the beginning of the analysis, but escalated such that CAES operation was some 85 percent more expensive than UPH at the end of the study period. Despite this difference, CAES plant usage remained high, and CAES plant operation required less revenue to be collected from the utility customers than with UPH.

The adiabatic CAES plant design presented in this report would require approximately 974 MW to operate all four units of the 800 MW plant in the compressing mode. This operating relationship is very similar to that of the UPH plant simulated by PEPCO. As a result, simulation within the PEPCO system would be expected to again result in oil-fired CAES being best able to meet PEPCO system needs over the study period.

Ignoring the fact that PEPCO's load growth projections have changed since the PEPCO analysis was performed, selection of a study period which begins five or ten years further into the future might have changed the results obtained. The effects of distillate fuel and charging power cost escalation rate differences would be more noticeable, to the probable benefit of both UPH and adiabatic CAES. Simulation within a different utility system would also likely produce different results.

One conclusion is that comparison of these three storage plant types using the levelized cost approach may be misleading. A second conclusion might be that favorable results for either adiabatic CAES with low cost heat storage materials or UPH require "installation" in a system which has more coal or nuclear capacity than PEPCO, either through overcapacity or through operation within a larger utility system. Thus, the results of Section 9 illustrate that in general adiabatic CAES appears economically competitive with oil-fired CAES, but that a different group of utilities would be candidates for application.

As the charge/discharge characteristics of adiabatic CAES and UPH are similar, and UPH production costs are somewhat lower, selection of an adiabatic CAES plant installation over a UPH design must be based on less clear cut criteria. One major difference between these two plant designs appears to be the relationship of capital cost and optimum plant capacity. Cost characteristics for UPH suggest that optimum plant capacity lies above 1200-1500 MW for the PEPCO design arrangement. If plant capacity is reduced to the 800 MW/10 hr adiabatic CAES size, capital costs are projected to be roughly equal for the two plant designs. Expansion of the adiabatic CAES design to 2000 MW will likely produce a relatively small reduction of cost per unit of capacity. This is a result of the difference in civil and mechanical cost distributions between the two plant types, as far greater proportion of the adiabatic plant cost is tied to the mechanical equipment. A very large portion of the UPH plant cost is tied to the access shafts and general underground construction facilities, and the impact of these costs is substantially reduced by increasing plant size. Thus, a utility or utility partnership capable of supporting a 2000 MW plant would be more likely interested in a pumped hydro design.

However, the fact that such a large portion of the UPH costs are tied to underground construction means that the plant construction is more vulnerable to the delay and cost increase risks which accompany deep underground construction. The UPH design also involves construction at 5000 feet versus 3000 feet for the adiabatic design proposed in this study, with UPH requiring the excavation of approximately five times the rock volume of the adiabatic CAES design. Few utilities can afford to start construciton early to intentionally allow for potential significant delays and still have capacity available when needed. Also, with a base licensing and construction schedule of 10 or 11 years, few utilities can be that certain regarding future peaking capacity needs to allow for a potential schedule increase to 12 to 14 years. Such potential risks with the UPH design would appear to make a somewhat better case for considering an adiabatic CAES approach. The adiabatic CAES concept must first be shown to be technically sound regarding availability of TES materials and reliable equipment.

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