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Compressed Air Energy Storage:

PRELIMINARY DESIGN AND SITE DEVELOPMENT PROGRAM IN AN AQUIFER

MASTER

DOE CONTRACT NO. ET-78-C-01-2159

Final Draft, Task 1

ESTABLISH FACILITY DESIGN CRITERIA AND UTILITY BENEFITS

OCTOBER 1980

Public Service Indiana

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PREFACE

The electric utility industry and the national economy will benefit from the successful integration of economic large-scale energy storage capacity into utility networks. Energy storage permits the substitution of energy from native fossil or nuclear fuels for energy that would otherwise be supplied by premium or strategic fuels. The development of new energy storage technologies will offer important economic advantages to the utilities and expand national energy resources.

It is the corporate policy of the sponsoring utilities to actively support the development of technology that will benefit the utility industry and, in turn, the national interest. The identification of Compressed Air Energy Storage (CAES) as one of the principal technologies worthy of development is compatible with the sponsors' plans for future service expansion and the abundance of potential underground aquifer sites in the Indiana-Illinois service area. A significant impact of such an energy storage facility is added flexibility in levelizing generation patterns of large base-load generating stations. Through the use of energy storage techniques, such as the Compressed Air Energy Storage System, it may be possible to reduce the use of premium fuels required by peaking and intermediate unit capacity generation.

This report presents a specific research and development plan to investigate the behavior and suitability of aquifers as CAES sites. This effort evaluates present uncertainties in the performance of the underground energy storage subsystem and its impact on above-ground plant design and cost. The project is planned to provide the utility industry with a quantitative basis for assessing confidence that financial commitment to a demonstration plant may be justified and poses only acceptable risks.

This project is in conjunction with the Department of Energy's and the Electric Power Research Institute's second phase of a five-phase program to determine the economic and technical feasibility of an aquifer base compressed air energy storage facility. The five phases of the overall development program are the following:

I. Conceptual Design
II. Preliminary Design and Site Exploration
III. Detailed Design and Site Development
IV. Construction
V. Check-out and Operation

This phase II effort is based on the significant body of knowledge accumulated in the already completed Phase I-Conceptual Design and other appraisals of compressed air energy storage technology. The overall goals of Phase II are to provide a detailed and costed preliminary plant design, the identification and selection of suitable and available aquifer sites, and a risk analysis as criteria for a utility decision to proceed with Phase III of the overall compressed air energy storage program. The goals are divided into the following five tasks:

Task 1 Establish facility design criteria
Task 2 Examine selected sites
Task 3 Study system, subsystem, and component designs
Task 4 Make preliminary safety and environmental assessment
Task 5 Prepare preliminary plant design

The work effort associated with these tasks is being provided by a consortium of electric utilities (consisting of Public Service Company of Indiana, Inc.; Central Illinois Public Service Company; Commonwealth Research Company on behalf of Commonwealth Edison Company; Illinois Power Company; and Union Electric Company), the Illinois State Geological Survey, the Indiana Geological Survey, Sargent & Lundy Engineers, and Westinghouse Electric Corporation.

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Compressed Air Energy Storage
Preliminary Design and Site Development Program In An Aquifer
DOE Contract No. ET-78-C-01-2159

Final Draft

Task 1-Volume 1
Establish Facility Design Criteria and Utility Benefits
October, 1980

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SUMMARY

Compressed air energy storage (CAES) has been identified as one of the principal new energy storage technologies worthy of further research and development. The CAES system stores mechanical energy in the form of compressed air during off-peak hours, using power supplied by a large, high-efficiency baseload power plant. At times of high electrical demand, the compressed air is drawn from storage and is heated in a combustor by the burning of fuel oil, after which the air is expanded in a turbine. In this manner, essentially all of the turbine output can be applied to the generation of electricity, unlike a conventional gas turbine which expends approximately two-thirds of the turbine shaft power in driving the air compressor. The separation of the compression and generation modes in the CAES system results in increased net generation and greater premium fuel economy. The use of CAES systems to meet the utilities' high electrical demand requirements is particularly attractive in view of the reduced availability of premium fuels such as oil and natural gas.

This volume documents the Task 1 work performed by Sargent & Lundy in establishing facility design criteria for a CAES system with aquifer storage. For an overall perspective, the Sargent &

Lundy work effort can be divided into four major categories:

- Determination of initial design bases
- Preliminary analysis of the CAES system
- Development of data for site-specific analysis of the CAES system
- Detailed analysis of the CAES system for three selected heat cycles.

The following discussion presents highlights of the work completed and the results obtained under each major category.

INITIAL DESIGN BASES

Several design parameters were developed at the onset of the CAES study to expedite selection of potential aquifer sites (Task 2) and to provide a base for overall cycle and equipment design. These initial design parameters are summarized below.

Design Parameters

Compression-Generation Cycle............	10 hours of compression and 10 hours of generation per day, 6 days per week, for a nominal 1000 Mw plant
Required Aquifer Storage Volume.........	12×10^9 SCF of air
Compressed Air Injection Temperature....	150^oF
Storage Pressure Range..................	200 psi - 1000 psi
Permeability Range......................	750 md - 3000 md

The current literature of compressed air storage substantiates the selection of these parameters as an initial design base for a CAES system with aquifer storage.

PRELIMINARY ANALYSIS OF CAES SYSTEM

Preliminary surface plant data were established under Task 1 to facilitate Task 2 assessment of the environmental impact of the CAES plant at potential sites. These data included initial estimates of compression-generation module dimensions and switchyard sizing, cooling water requirements, and fuel usage and storage needs. In addition, plant waste water flows, potable water requirements, and air effluent flows were approximated. A conceptual diagram of the entire CAES plant was prepared to provide a base for development of refined plot plans as cycle and equipment design progressed.

Typical storage pressure levels of 200 psi, 600 psi, and 1000 psi were selected as base criteria for further CAES system design and analysis. A one-dimensional aquifer flow code developed by Westinghouse was employed to determine the number of air injection-withdrawal wells required at each storage pressure. These preliminary well requirements were incorporated into an above-ground air piping configuration for each selected storage pressure.

Pressure losses through the aquifer porous media, wells, and above-ground piping network were then computed for both storage-charging and storage-discharging conditions at each pressure level. Calculation of these system pressure losses permitted the determination of the external pressure loss ratio, a parameter critical to CAES heat cycle design. The external pressure loss ratio is defined as the ratio of compressor discharge pressure to turbine inlet pressure. A pressure loss ratio range of 1.2 to 1.8 was established for this CAES application, with the base ratio estimated at 1.4.

In addition to the external pressure loss ratio, several other heat cycle parameters were developed as a part of the Task 1 work effort, for use by Westinghouse in the evaluation and optimization of CAES system thermal cycles. The required heat cycle parameters are detailed below:

<u>Supplemental Heat Cycle Parameters</u>

External Pressure Loss Ratio....................	Base	= 1.4
	Range	= 1.2 - 1.8
Compression Cycle Equipment Air Discharge Temperature.................................	T_a	= 150°F
External Plant Air Temperature Loss.............	T	= 10°F
Available Equipment Cooling Water Supply Temperature.................................	T_w	= 95°F
Regenerator Minimum Cold End Average Temperature.................................	T_{ave}	= 205°F

DEVELOPMENT OF DATA FOR SITE-SPECIFIC ANALYSIS OF THE CAES SYSTEM

The piping networks developed for the three typical storage pressures (200 psi, 600 psi, and 1000 psi) served as reference cases for the preparation of order of magnitude piping cost estimates, in current prices. The piping requirements and costs corresponding to specific sites and storage pressures identified in Task 2 were obtained by interpolation from the reference cases.

In addition to piping system cost estimates, several other criteria were established under Task 1 to permit Task 2 assessment of candidate aquifer sites. These criteria included site development and water supply development guidelines, as well as transmission and mechanical equipment cost estimates. The site development criteria provided a base for preparation of site development and plant access costs in Task 2. The water supply development criteria detailed the minimum requirements that must be met before the plant water system can be termed dependable and adequate; these requirements were used in Task 2 identification of the most economical water sources at potential sites.

DETAILED ANALYSIS OF THE CAES SYSTEM FOR THREE SELECTED HEAT CYCLES

Comparison summaries for a total of fourteen possible heat cycles were prepared by Westinghouse for nominal 200 psi, 600 psi, and 1000 psi storage pressure CAES systems. A single heat cycle compatible with each storage pressure level was then selected. In addition, the Sponsoring Utilities and subcontractors chose a single-shaft arrangement as the preferred machinery configuration for each CAES system.

Following the selection of the most viable heat cycle and equipment designs, general plant arrangement and piping and instrumentation diagrams (P&ID's) were developed for the typical 200 psi, 600 psi, and 1000 psi systems. The general arrangement drawings prepared for each case include grade and main floor plans and one cross-section. P&ID's for each nominal storage pressure plant depict the compressed air cycle, fuel oil system, circulating water system, and demineralized and potable water system. A property development layout was also prepared for each storage pressure system. Electrical diagrams, applicable to all three storage pressure cases, were developed to illustrate the basic bus arrangement for the CAES plant, the station single line for a CAES unit, and the CAES plant switchyard requirements.

Preparation of CAES unit data sheets for both a full scale plant and a demonstration plant comprised another aspect of the Task 1 work effort. These data sheets provided the Sponsoring Utilities with information pertinent to development and evaluation of production cost savings for a potential CAES plant.

An investigation of the effects of aftercooler discharge air dehumidification was also initiated as part of the Task 1 detailed CAES system analysis. The general consensus at the conclusion of Task 1 was that some form of dehumidification would be required to prevent reduction of airflow in the storage reservoir. Future task work will continue to assess this problem and will establish design parameters for selection of dehydration equipment.

Task 1 facility design criteria documented in this volume are preliminary, and therefore are subject to refinement in subsequent tasks of the CAES development program.

Section 1

INTRODUCTION

1.1 OBJECTIVE

This report has been prepared for the Public Service Company of Indiana, Inc. (PSI) in compliance with DOE Contract No. ET-78-C-01-2159 to document the results of work conducted for Phase II, Task 1 of the Compressed Air Energy Storage (CAES) Development Program in an Aquifer.

A primary objective of Task 1 was to establish design criteria which define acceptable ranges of performance and operational specifications governing the overall plant design for a CAES system. The criteria established under Task 1, in conjunction with Task 2 results, provide the technical data necessary for specific analysis of plant systems and subsystems in Task 3.

The results documented in this report complement the Westinghouse Electric Corporation Task 1-Volume 2 work effort. Major responsibilities of Sargent & Lundy under Task 1 included the development of preliminary environmental impact data and supplemental heat cycle parameters, the preparation of preliminary above-ground piping networks and piping cost estimates, and the evaluation of plant equipment for development of general arrangement drawings.

Figure 1-1 provides a conceptual representation of the surface plant for the compressed air storage system. A postulated isometric view of the CAES plant and associated well field is shown in Figure 1-2.

1.2 REPORT ORGANIZATION

This volume of the Task 1 report is comprised of six major sections. Section 1 includes the project objective and report organization. Section 2 provides a discussion of the initial design parameters developed as a base for CAES system design. Section 3 is devoted to preliminary analysis of the CAES system. This section describes the development of preliminary environmental impact data, supplemental heat cycle parameters, and preliminary above-ground air piping systems. Section 4 focuses on the preparation of data relevant to site selection in Task 2. Section 5 presents a more detailed analysis of the CAES system for selected heat cycle and machinery configurations. Included in this section are general arrangement drawings, piping and instrumentation diagrams and electrical single line schematics for three typical CAES plants. Section 6 is the report conclusion.

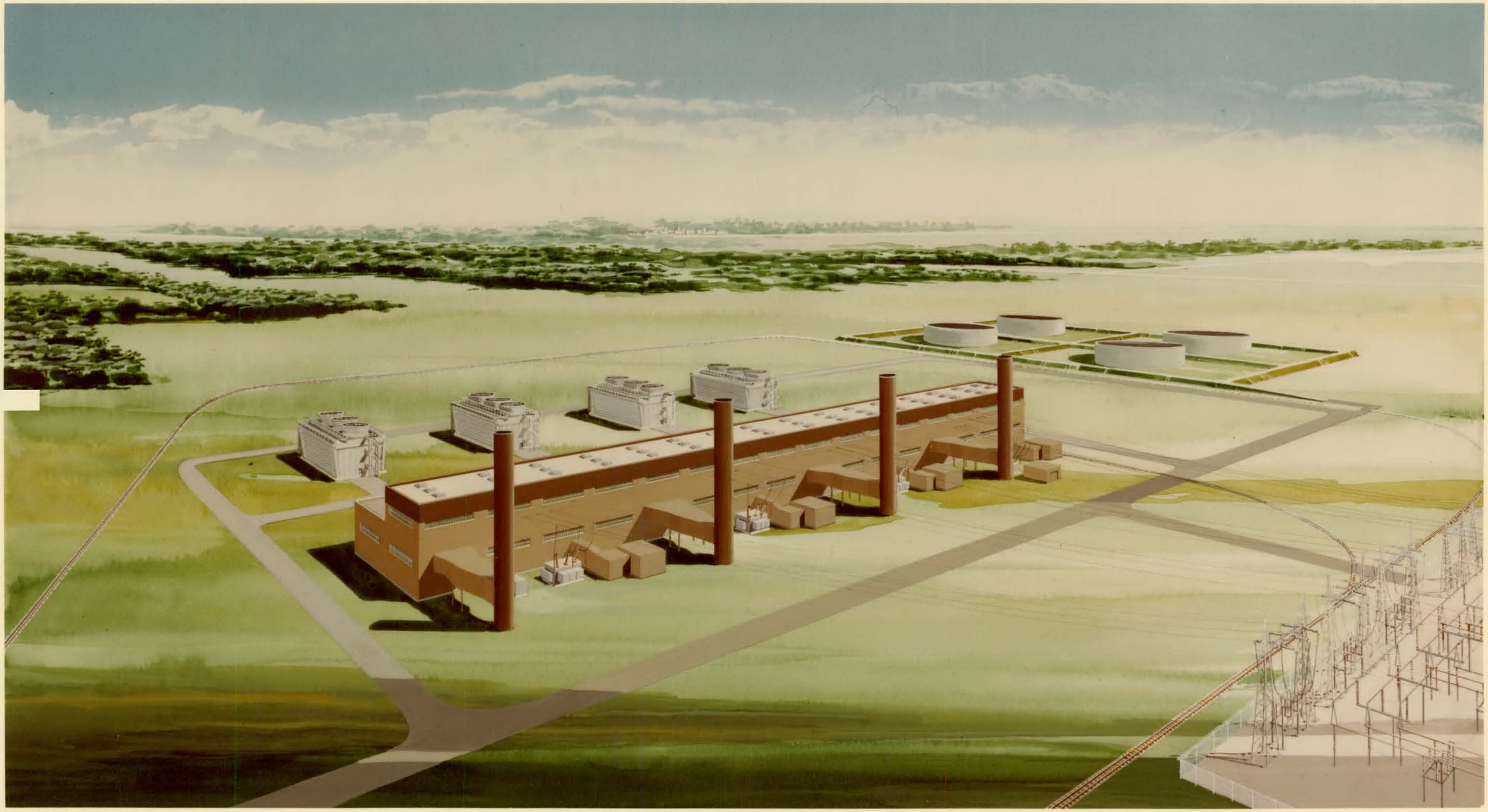

FIGURE I-I
CONCEPTUAL REPRESENTATION
OF
A COMPRESSED AIR ENERGY STORAGE FACILITY

SARGENT & LUNDY
ENGINEERS

PUBLIC SERVICE INDIANA
CAES IN AQUIFER
DOE NO. ET-76-C-01-2159

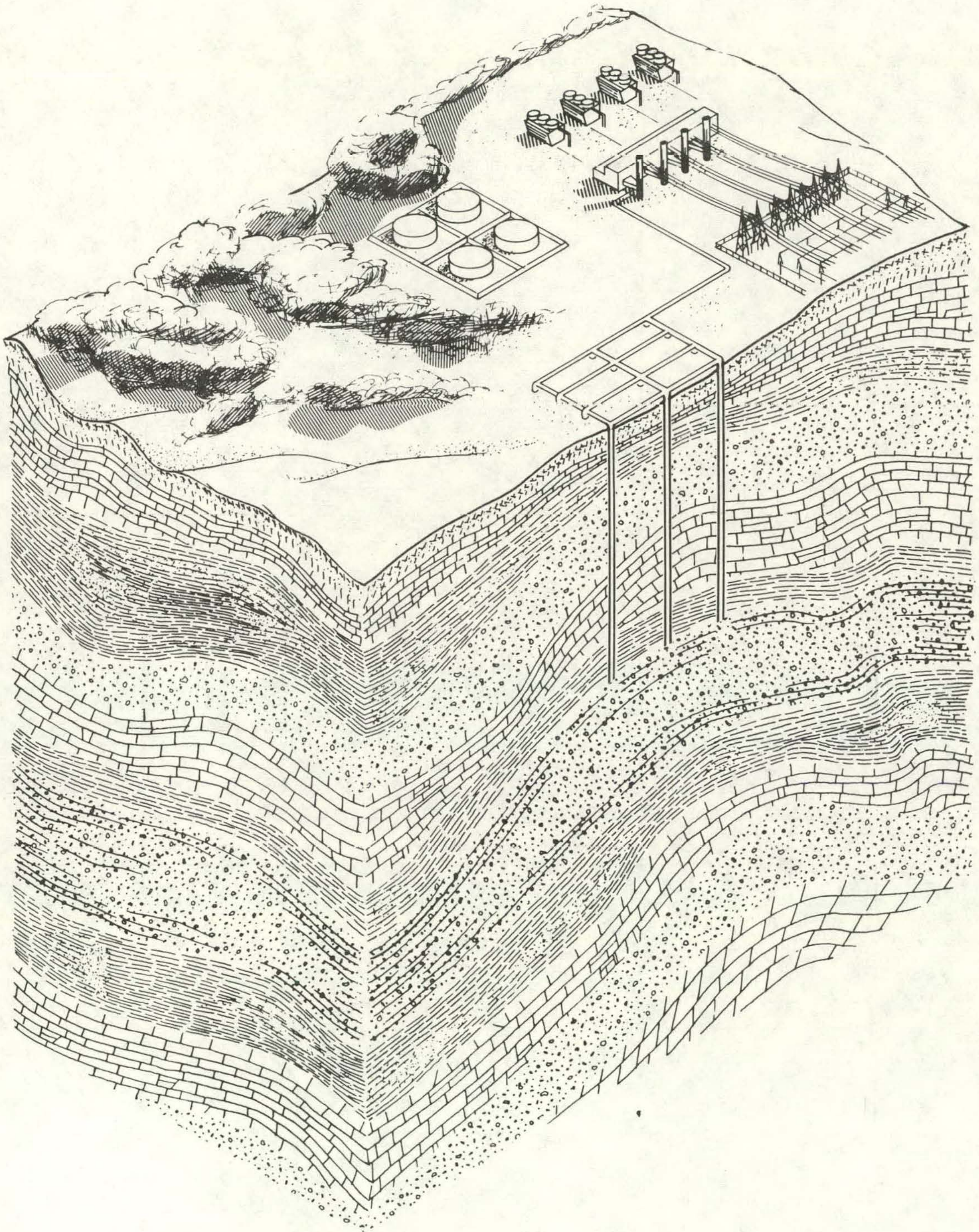

FIGURE I–2
POSTULATED ISOMETRIC VIEW
OF A CAES PLANT AND ASSOCIATED WELL FIELD

SARGENT & LUNDY
ENGINEERS

PUBLIC SERVICE INDIANA
CAES IN AQUIFER
DOE NO. ET–78–C–01–2159

Section 2

INITIAL DESIGN BASES

2.1 GENERAL

In conjunction with the Public Service Company of Indiana and Westinghouse, Sargent & Lundy has established a set of initial design parameters to expedite the selection of potential aquifer sites by the Illinois State Geological Survey (I.S.G.S.) and the Indiana Geological Survey (I.G.S.). These preliminary criteria also provide a basis for overall cycle and equipment design for the Compressed Air Energy Storage System. The parameters identified are as follows:

Design Parameters:

<u>Compression-Generation Cycle</u>.......	10 hours of compression and 10 hours of generation per day, 6 days per week, for production of a nominal 1000 Mw of electricity
<u>Required Aquifer Storage Volume</u>....	12×10^9 Standard Cubic Feet of air.
<u>Compressed Air Injection Temperature</u>....	150^oF
<u>Storage Pressure Range</u>.............	200 psi - 1000 psi
<u>Permeability Range</u>.................	750 md - 3000 md

These criteria serve as input to Task 2 of the CAES work effort. Task 2, which parallels the Task 1 work effort, encompasses the actual site selection process and the evaluation of candidate aquifers, as described in the Task 2 Milestone Report.

The design parameters specified above are discussed in detail in the following subsections.

2.2 COMPRESSION-GENERATION CYCLE

The general consensus of the sponsoring utilities has been that a nominal 1000 Mw compressed air storage plant having equal charging to generation times would best accommodate utility characteristics. The utility sponsors have specified a compression-generation cycle consisting of 10 hours of compression and 10 hours of generation per day, 6 days per week, as a Task 1 plant design criterion.

The plant design therefore assumes equal compression and generation air flow rates. This matching of compressor and turbine flow rate has a significant advantage in that the compressor and turbine components may be operated as a conventional gas turbine. The General Electric <u>Economic and Technical Feasibility Study of Compressed Air Storage</u> (Phase I Report) summarizes the favorable aspects of a system having equal flow-conducting capability:

(1.) The need for a large and expensive controller for the synchronous motor generator during initial system start-up is eliminated if the compressor and turbine have equal flow capacity. The system may be started as a normal gas turbine; that is, a small starting engine may be used to start the shaft turning, after which the turbine can be used to accelerate the system to synchronous speed.

(2.) In the event that the aquifer can not be properly charged during the off-peak period due to an equipment failure, the system can still be operated as a gas turbine (although at reduced output, due to compressor requirements).

(3.) Since compressed air energy storage is a developing technology, the useful life of the storage system is not yet accurately known. In case of premature overall failure of the storage system, the ability to operate the plant as a conventional gas turbine unit (at diminished output) would reduce potential financial risk to the utility.

Further development of plant design parameters, then, has been based on the equal charging/discharging times required by the utility system and on equal flow-conducting capability for turbine and generator.

2.3 REQUIRED AQUIFER STORAGE VOLUME

It is estimated that the potential aquifer must have sufficient closure and volume for storage of approximately 12 billion standard cubic feet of air. This is based on the estimated turbine air mass flow required for production of 1000 Mwe by the surface plant for a full 10 hours each day. Also, an air storage cushion factor (ratio of stored air mass to mass withdrawn each day) of 10 has been applied.

A review of the current literature of compressed air storage substantiates the use of a volume cushion factor of 10 as a reasonably conservative first estimate for a porous media plant configuration (1), (2), (3). General Electric's Technical Feasibility Study of Compressed Air Storage cites storage volumes of up to 12 times the volume change throughout the cycle.

There are a number of advantages inherent in a large air storage factor. A portion of the residual stored air mass may be regarded as energy-in-reserve which can be utilized for power generation on relatively short notice; this permits a more flexible system operation. The volume reserve is also necessary to reduce the pressure range between charged and discharged states to acceptable levels (3). In-situ permeability may be lower than preferred, resulting in high formation resistance. The additional volume is then desirable to reduce air velocities, thereby reducing pressure losses (3). Finally, a large reservoir is advantageous in the event of future plant expansion beyond present estimates.

A cushion factor of ten is felt to be a representative first approximation. However the actual storage volume, and thus the degree of conservatism in the buffer factor, will ultimately depend on the characteristics of the available storage sites.

Other factors which will affect determination of a final storage volume are the utility system's electric load demand curves and plant capacity factor. Previous studies have shown that increased storage reserve can result in higher capacity factors (3). However, as storage capacity continues to rise, capacity factor eventually tends toward a limit. Also, since greater storage capacity results in greater capital cost, the rise in capital cost may offset the lower levelized power costs obtained because of increased capacity factor (3). A sensitivity analysis, to determine the overall economics of compressed air energy storage to the utility, should play an important part in final selection of an appropriate storage volume.

2.4 COMPRESSED AIR INJECTION TEMPERATURE

The initial compressed air injection temperature for the proposed compressed air aquifer has been limited to 150°F. The precise effects of hot, compressed air, in situ, cannot be predicted in advance. However, available literature and current research on compressed air storage suggest that injection temperatures in the vicinity of 150°F should result in minimal disturbance to the mechanical integrity of the reservoir. It is desirable to have the injection temperature as high as possible without causing damage to the aquifer, caprock, or well case grouting.

D. L. Katz and E. R. Lady, in an analysis of compressed air storage (1), point out that potential flow problems within aquifers may involve water blockage of capillaries in the porous media around well bores and the accumulation of water in the bores. The injection temperature can become a factor in alleviation of these water intrusion problems. The injected air warms the rock in the vicinity of the bore and eliminates capillary problems at that point. Katz postulates that acceptable injection temperatures may lie in the range of 150°F-200°F.

The General Electric Feasibility Study of Compressed Air Storage (Phase I Report) also offers a discussion of reservoir injection temperature. According to the General Electric Feasibility Study, a common approach has been to specify temperatures in the vicinity of 125°F, with injection temperatures ranging up to 180°F for several of the aquifers cited. Temperatures much in excess of this range have the potential of causing fractures in the well case concrete lining and grouting within the aquifer system. The imposed temperature limitations also serve to limit the transfer of heat to the water in the lower reservoir; this is advantageous in reducing evaporative losses. In addition, the General Electric Report has suggested that cementation due to mineral deposition, and oxidation of minerals in the groundwater may complicate operation of aquifers at temperatures exceeding the limitations discussed (2).

Research pertinent to compressed air storage temperature limitations has also been conducted by H. J. Pincus, University of Wisconsin-Milwaukee. These investigations focused on the physical properties of various sandstone and limestone rock specimens, upon subjection to dry heated air. No systematic changes in compressive strength and Young's Modulus occurred for sandstone reservoir rock at compressed air temperatures approximating 200^{o}F and 80 psi differential pressure. The compressive strength and Young's Modulus for the Bedford limestone specimens were lowered somewhat under these conditions. These preliminary results lend credibility to the specified aquifer injection temperature of 150^{o}F. Further research proposed by Pincus would evaluate the effects on rock specimens of cyclic air ventilation at higher temperatures and pressures and at varying humidities of air.

The Pincus studies have not yet examined the effects of heated air on shale caprock specimens. Potential damage to caprock from thermal excursions include drying and cracking and thermal stress cracking. However, D. L. Ayers, in association with

Westinghouse Fluid Systems Laboratory, anticipates that a caprock which is sufficiently thick (greater than 20 feet) and which is in intimate contact with structure above and below it, will withstand expected storage temperatures of approximately 200°F. This type of caprock might also have ultimate potential for storage at higher temperatures, in the event that this becomes economically attractive in the future. Of course, before attempting cycling over wider temperature ranges, a careful examination of the rock mechanics of the particular host formation being considered would be necessary.

2.5 STORAGE PRESSURE RANGE

The aquifer discovery pressure is a very important characteristic, since this pressure dictates the maximum allowable storage pressure. An air storage pressure which lies within the range of 200 psi - 1000 psi is considered acceptable for the proposed compressed air storage system; this range is based on the expected pressure capability of the turbo-machinery and on economic considerations.

The lower limit of 200 psi was selected to provide a match with the inlet pressure of the combustor/turbine sections of currently available gas turbines. The lowest inlet pressure deemed practical by Westinghouse was 10 atmospheres. Then, a minimum of 50 psi pressure loss between storage and turbine inlet yields a minimum workable storage pressure of approximately 200 psi. Preliminary Westinghouse calculations indicate that at lower air storage pressures an unreasonably large number of wells are required to service the turbines; consequently, lower storage pressures are not cost effective.

At the upper end of the pressure spectrum, the maximum high pressure turbine inlet pressure used for preliminary calculations by Westinghouse Fluid Systems Laboratory was 750 psi. With allowance for gathering line, well, and aquifer losses, the maximum storage pressure has been set at 1000 psi. The economics of compression become unfavorable as pressures continue to increase above this point.

The current literature of compressed air storage also confirms that the pressure limitations of 200 psi and 1000 psi bracket a technically and economically feasible pressure range. The General Electric Phase I Report, for example, cites storage pressures in the range of 200 psi to 1200 psi, and suggests that the 600 psi to 740 psi range may represent an overall optimum (2). The Consultant Report on Feasibility of Compressed Air Energy Storage As A Peak Shaving Technique in California concludes that pressures in the range of 600 psi to 1000 psi appear to be most attractive.

Within the 200 psi - 1000 psi design criterion specified for this CAES application, the higher storage pressures are preferable since these would tend to provide lower total plant development costs.

2.6 PERMEABILITY RANGE

Permeability, frequently measured in terms of the millidarcy (md), describes the flow of fluid through a porous media; materials vary greatly in their resistance to fluid flow, and thus, in their permeability.

The aquifer system for compressed air storage is comprised of an underground porous media storage reservoir on the top of which rests an impermeable, air-retaining caprock. A layer of water confines the air from below. The porous media storage volume usually consists of sandstone rock or of vugular or porous limestone and/or dolomites (1).

The permeability of the aquifer storage media must be great enough to allow delivery of air to the wells at the mass flow rate required by the turbine. The number of wells necessary and the extent of the air distribution system are determined by this permeability. Pressure losses within the host formation increase considerably at low permeabilities, so that a greater number of wells are required to achieve air deliverability to the turbine. At high permeabilities, the minimized aquifer pressure losses will result in a reduction in number of surface air gathering lines and in compression cost.

It is recognized that there are only a limited number of aquifer sites suitable for compressed air storage, and that available permeabilities may not always be optimal. However, as a preliminary guideline, a permeability range of 750 md - 3000 md is suggested for selection of potential aquifer sites.

Section 3

DESIGN CRITERIA--PRELIMINARY ANALYSIS OF COMPRESSED AIR ENERGY STORAGE (CAES) SYSTEM

3.1 DEVELOPMENT OF DATA FOR PRELIMINARY ENVIRONMENTAL IMPACT STUDY

3.1.1 General

Preliminary surface plant data for the compressed air energy storage system have been prepared for use in the Task 2 assessment of environmental impact of the plant at potential sites. The parameters developed include the following:

- Compressor-turbine-generator equipment dimensions and switchyard sizing
- Cooling water requirements and cooling tower dimensions
- Fuel usage and fuel storage requirements
- Waste water flows
- Potable water requirements
- Air effluent flow.

The plant criteria established at this point in Task 1 represented the first approximation only, and were subject to refinement in later stages of Task work.

3.1.2 Equipment Dimensions & Switchyard Sizing

As an initial estimate, Westinghouse advised that each turbine-compressor-generator module for the CAES system will be capable of 200 MWe output. Therefore, a total of five (5) modules was postulated for a 1000 MWe station. Each module includes one 175 Mwe generator and one 25 Mwe generator acting in combination; the 175 Mwe generator is assumed to start first in each case. The overall length by width dimensions for each module, inclusive of all necessary equipment, were estimated by Westinghouse as 120 feet x 150 feet. For conservatism in the overall plant layout, these dimensions were modified to 140 feet x 150 feet. The number of modules ultimately required for the CAES plant will depend on detailed Task 1 analysis of potential heat cycle and machinery configurations by Westinghouse and Sargent & Lundy.

The overall area required for the main compressor-turbine-generator building was approximated at 700 feet by 180 feet.

It was also estimated that the compressed air storage plant could be serviced by a switchyard having an overall area of 350 feet x 400 feet, (140,000 ft^2).

3.1.3 Preliminary Cooling Water Requirements & Cooling Tower Dimensions

A preliminary assessment of cooling water needs for a nominal 1000 Mw CAES plant has indicated a total cooling water flow requirement of 178,160 gpm. This initial flow estimate is based on heat rejected from the compressor intercoolers and aftercoolers, rotating equipment bearings, and motor generators, with a temperature rise of $40^{\circ}F$. The $40^{\circ}F$ temperature rise has been suggested by heat exchanger manufacturers. State temperatures at the intercoolers and aftercoolers were based on preliminary Westinghouse estimates, and an intercooler approach of 10^{O}F applied. Maximum temperature of available cooling water was set at 95^{O}F.

The preliminary cooling water requirements for the CAES plant are itemized below.

Cooling water requirements (preliminary)

Compressor intercooler water......................	89,290 gpm
Compressor aftercooler water flow..................	78,230 gpm
Rotating equipment bearings & generator...........	10,640 gpm
Total	178,160 gpm

Using the result obtained for total cooling water flow, the required cooling tower make-up water flow rate has been calculated at 7980 gpm. This make-up water flow replaces expected cooling tower losses due to evaporation, drift, and blowdown as follows:

Maximum expected cooling tower make-up water flow

Evaporative losses................................	5,700 gpm
Drift..	180 gpm
Blowdown...	2,100 gpm
Total	7,980 gpm

The make-up water flow rate is based on a maximum cycles of concentration of 3.5.

Typical overall physical dimensions for a cooling tower of the duty described are approximately 324 feet x 73 feet (length x width). These dimensions are based upon use of a single cooling tower having nine cells. If the modular concept is applied to the cooling tower system, a total of five cooling towers having two cells each would be required.

3.1.4 Fuel Usage and Fuel Storage Requirements

Preliminary sizing of fuel storage facilities has been based on a 60-day fuel reserve for the compressed air plant, when operating at a nominal fuel consumption rate of 5289 Btu/Kw-hr, and on a 10 hour per day cycle, 6 days per week. Using a high heating value for distillate fuel oil of 141,000 Btu/gal, the total burn rate required for production of 1000 Mwe is 625 gpm. On these bases, approximately 19,291,000 gallons of reserve fuel oil are required.

A workable storage scheme, therefore, consists of four, 140 foot diameter tanks of 110,000 barrels capacity each. Berms of square configuration with a 6 foot height and a rise-to-run ratio of 1:2 are required for the storage tanks. The berm for each tank must be of sufficient capacity to retain 100% of the tank volume in the event of catastropic failure of the tank. Based on these data, the total fuel storage area enclosed by the berms must be of square configuration with a total overall area of 617,380 ft^2.

3.1.5 Waste Water Flows

Total waste water flow from the nominal 1000 Mw CAES plant is estimated at 2530 gpm; included in this total are contributions from cooling tower blowdown, the compressor aftercooler drain, and the plant cleaning floor drains. Cooling tower blowdown has already been estimated at 2100 gpm. The compressor after-cooler drain flow consists of the total water rejected in the processing of air from 14.7 psia and 60% relative humidity at $60^{O}F$ to 1165 psia and 100% relative humidity at $150^{O}F$; an estimated flow of 130 gpm is rejected during this process. Waste flow from the plant floor drains is on an intermittent basis only, but is estimated at 300 gpm.

3.1.6 Potable Water Requirements

After review of typical plumbing designs for non-coal fired plants, the potable make-up and sewerage water requirement for the proposed compressed air storage plant has been estimated at 35 gallons per day per person. Assuming 40 employees, a total flow of 1400 gal/day, or 1 gpm, is required.

3.1.7 Air Effluent Flow

Preliminary calculations suggest a total effluent air flow from the 1000 Mw CAES plant of 239,590 lb/min. This value represents a summation of exhaust air flow and fuel flow. The exhaust air flow per five modules is approximately 234,900 lb/min, while the total fuel flow requirement is 4,690 lb/min.

Atmospheric pollutant releases from compressed air storage plant operation should be comparable to typical gas turbine emissions. Actual emissions of sulfides and oxides of nitrogen (NO_x) are a function of the fuel oil burned and will be determined during refinement of the CAES plant design.

3.1.8 Preliminary Plant Block Diagram

The space requirements developed in the preceding sections have been incorporated into a general block diagram of the entire compressed air energy storage site (Figure 3-1). This conceptual diagram, along with the design parameters established, initiated the Task 2 environmental impact study. The preliminary block diagram also provided a base for development of refined plot plans as cycle and equipment design data became available.

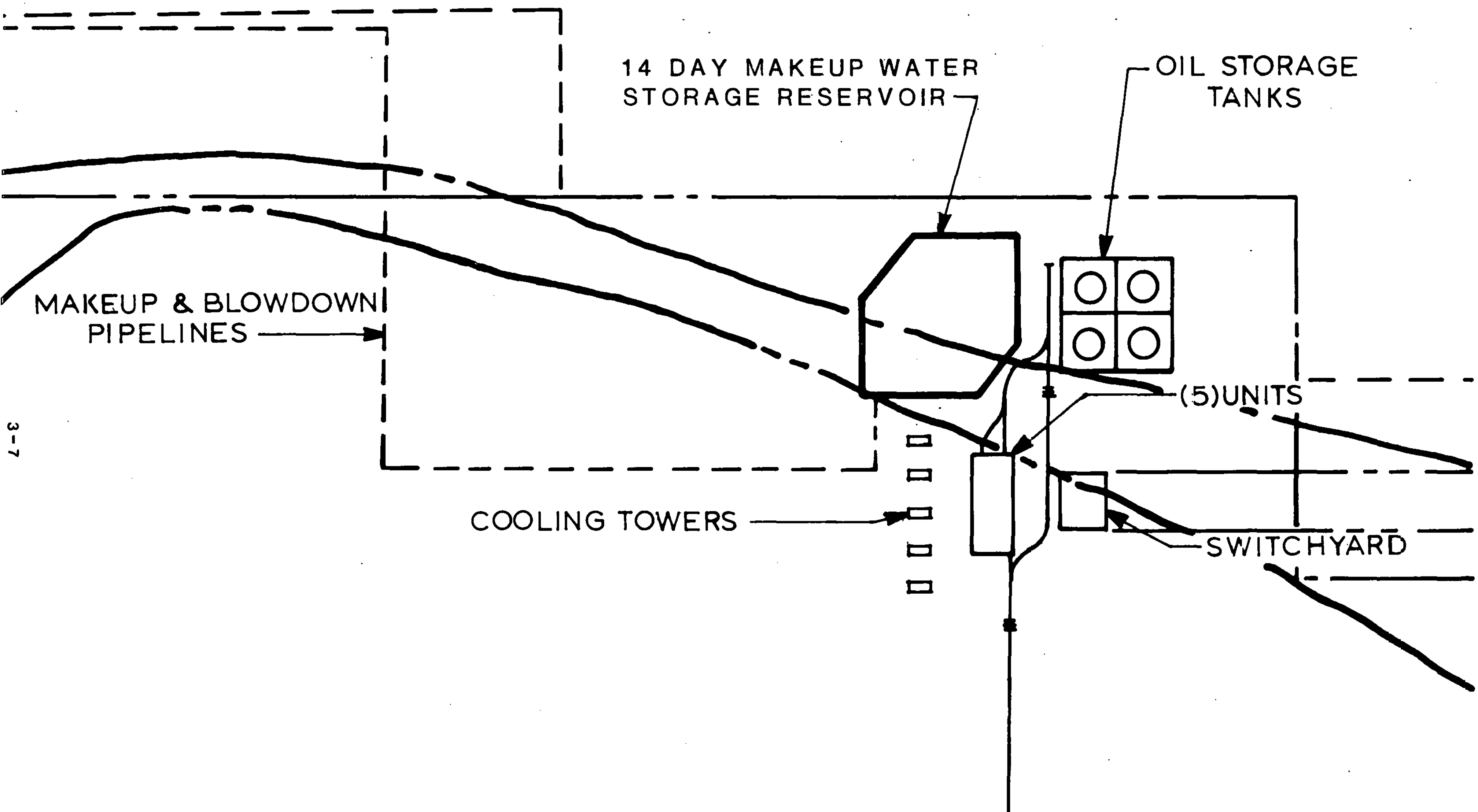

FIGURE 3-1
COMPRESSED AIR
ENERGY STORAGE SYSTEM
PRELIMINARY BLOCK DIAGRAM

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CAES IN AQUIFER
DOE NO. ET-78-C-01-2159

3.2 DEVELOPMENT OF PRELIMINARY ABOVE-GROUND* AIR PIPING SYSTEMS

3.2.1 General

A representative air piping network has been established for each of three typical aquifer pressures, based on the estimated number of wells required to service the generating plant at each pressure. System pressure drops, external pressure loss ratios, and piping system costs have been determined at three feasible flow velocities for each piping network. The results obtained provide a set of reference conditions for development of site specific piping and well costs in later stages of Task 1 (Section 4.3).

3.2.2 Number of Wells Required At Three Typical Storage Pressures

Typical aquifer storage pressures of 200 psi, 600 psi, and 1000 psi, at permeabilities of both 750 md and 3,000 md, have been considered in development of preliminary piping networks. The number of wells necessary to charge or discharge the plant at each storage pressure and permeability has been estimated by D. L. Ayers, Westinghouse Fluid Systems Laboratory. This initial determination of required number of wells employed the simplified one-dimensional aquifer flow code developed by Westinghouse.

*The term "above-ground air piping" used throughout this report refers to the pipelines which transport air between the wellheads and CAES plant. This piping is to be distinguished from the wells themselves. In actuality, the "above-ground" piping may be buried for aesthetic reasons; also, for conservatism in the preparation of piping cost estimates, burial of these pipelines has been assumed.

The one-dimensional flow code is a single well simulation, which uses a simple cylindrical aquifer model to analyze air reservoir--well performance. No account is taken of interference from adjacent wells or of piping frictional losses and well losses. The flow model seeks the finite difference solution to the transient compressible flow equation which governs the flow in the reservoir and single well. For specified boundary conditions, the analysis determines a single well's capacity to inject air into the aquifer and to deliver it to the above-ground piping system. Once the capacity of a single well has been found in this manner, the total number of wells needed is computed according to the required turbine air mass flow rate.

The turbine air mass flow rates corresponding to 200 psi, 600 psi and 1000 psi aquifer storage pressures have been approximated by Westinghouse and are tabulated below.

TABLE 3-1

TURBINE AIR MASS FLOW RATES

Storage Pressure	Turbine Air Mass Flow Rate, M	
psi	lbm/sec	MMCF/hr
200	3759	176.6
600	3096	145.5
1000	2632	123.7

Table 3-2 lists the well requirements for the three typical storage pressures and two permeabilities, based on the preliminary Westinghouse single-well simulation and the specified turbine flow rates. A 12 inch well diameter has been used in all cases, and the discovery pressure has been taken as the storage pressure. Per the CAES system design, the compressor and turbine have been assumed to share the same air-piping and well system. The results shown in Table 3-2 have also been based on the specified plant duty cycle, which consists of 10 hours of compression at night and 10 hours of generation during the day for a 1000 Mw plant.

TABLE 3-2

WELL REQUIREMENTS AT THREE TYPICAL STORAGE PRESSURES

Storage Pressure psi	Permeability md	Flow MMCF/hr	Approximate # of Wells	Flow/Well MMCF/hr
200	750	176.6	500	0.35
600	750	145.5	50	2.90
1000	750	123.7	35	3.50
200	3000	176.6	150	1.18
600	3000	145.5	20	7.30
1000	3000	123.7	10	12.4

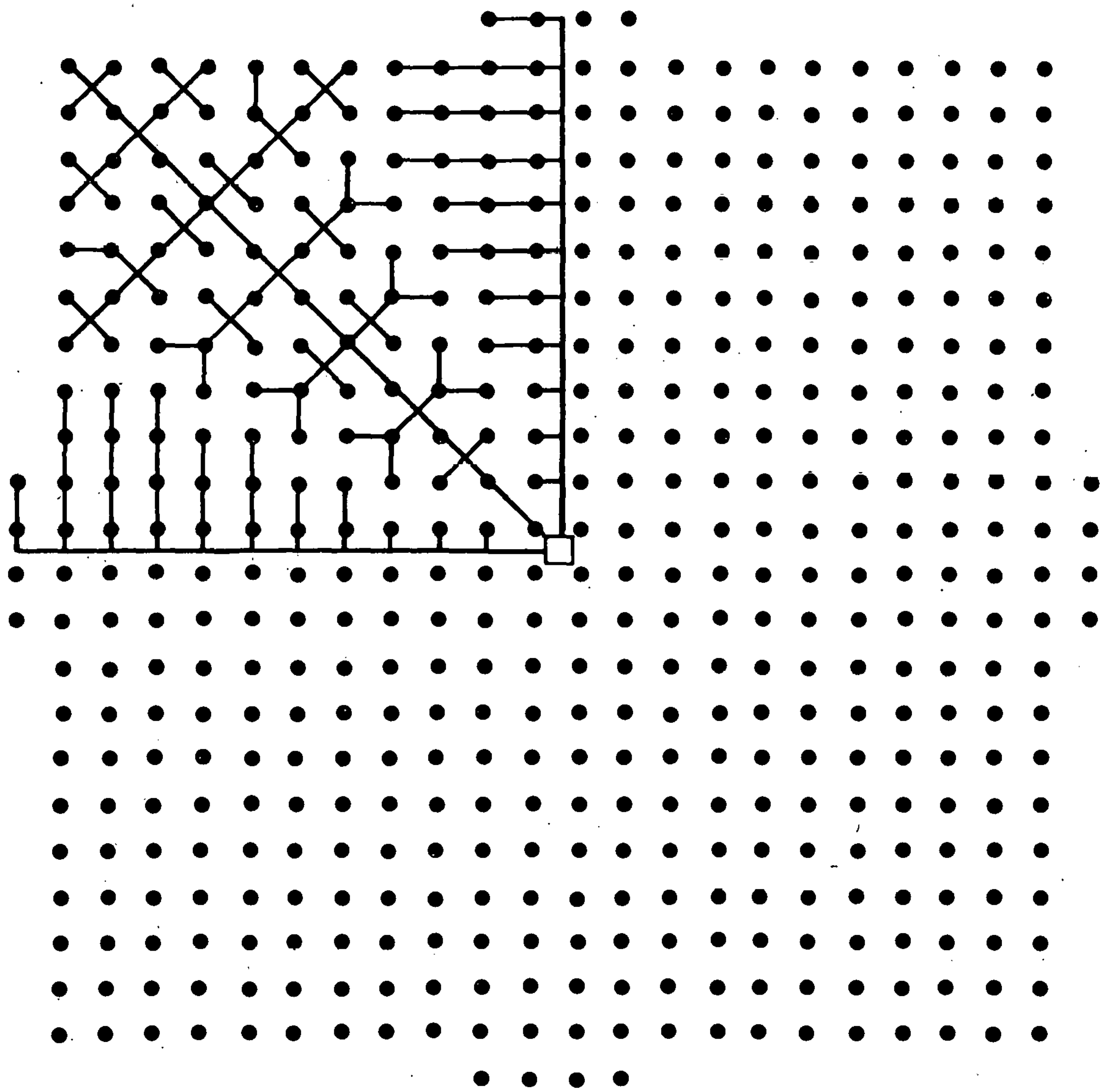
It should be noted that the well requirements specified in Table 3-2 do not reflect in-situ well requirements at the given storage pressures. In fact, actual well requirements may increase significantly if the effects of multiple well operation and frictional losses are considered. The actual number of wells needed at a specific site is expected to be a function of the areal configuration of the reservoir, pressure loss ratio, pipeline design, and aquifer depth, as well as several other factors. Determination of site specific well requirements is discussed in greater depth in the Task 2 Milestone Report.

The well requirements presented in Table 3-2, however, do provide a base for the development of a set of preliminary piping networks and piping system cost estimates. From these reference cost figures a relative comparison of piping costs for the potential aquifer sites has been prepared, as discussed in Section 4.3.

3.2.3 <u>Preliminary Piping Networks for Three Typical Storage Pressures</u>

The preliminary well requirements defined in Table 3-2 have been incorporated into an above-ground air piping configuration for each typical storage pressure. Figures 3-2, 3-3 and 3-4 illustrate the arrangement of wells and the piping network for aquifer storage pressures of 200, 600 and 1000 psi, respectively, at a permeability of 750 md. Each figure presents a plan view of the proposed arrangement of surface piping and wells. The CAES plant has been positioned at the center of each well field configuration for purposes of preliminary analysis. The Task 3 work effort will establish the most practical location for the CAES plant with respect to the well field and other plant structures, based on characteristics of the storage site selected. Figures 3-2, 3-3 and 3-4 show only the pipeline configuration for a single quadrant because of the radial symmetry of the piping network about the centralized plant. The spacing between adjacent wells has been taken as 400 feet in all cases.

Although similar piping networks have also been developed for the 3000 md permeability case, the results have not been included here. It is true that a permeability of this magnitude would constitute a highly desirable aquifer characteristic. As indicated in Table 3-2, well requirements are significantly reduced at 3000 md; consequently, above-ground piping configurations are simplified. However, a cursory inspection of potential aquifer sites throughout



NOTE: SPACING BETWEEN ADJACENT WELLS IS 400 FEET.

PIPELINE

12 IN. DIAMETER WELL

CAES TURBOMACHINERY BUILDING

FIGURE 3-2
COMPRESSED AIR ENERGY STORAGE SYSTEM
- PRELIMINARY PIPING NETWORK -
- 200 PSI STORAGE PRESSURE -
500 WELLS
PLAN VIEW

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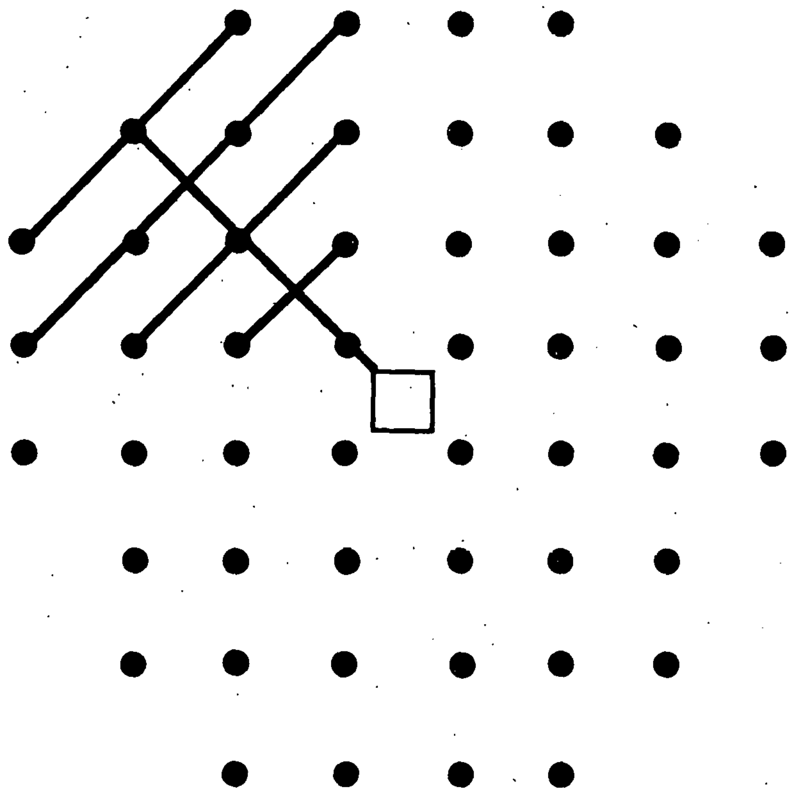

NOTE: SPACING BETWEEN ADJACENT WELLS IS 400 FEET.

FIGURE 3-3
COMPRESSED AIR ENERGY STORAGE SYSTEM
- PRELIMINARY PIPING NETWORK -
- 600 PSI STORAGE PRESSURE -
50 WELLS
PLAN VIEW

PIPELINE
12 IN. DIAMETER WELL
CAES TURBOMACHINERY BUILDING

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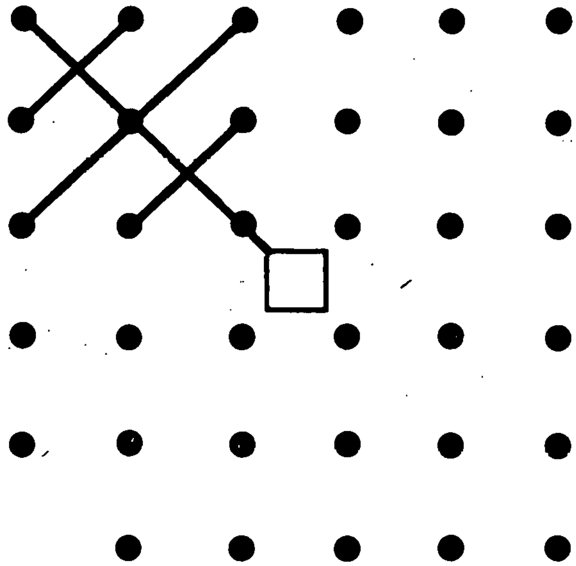

NOTE: SPACING BETWEEN ADJACENT WELLS IS 400 FEET.

FIGURE 3-4
COMPRESSED AIR ENERGY STORAGE SYSTEM
- PRELIMINARY PIPING NETWORK -
- 1000 PSI STORAGE PRESSURE -
35 WELLS
PLAN VIEW

— PIPELINE
● 12 IN. DIAMETER WELL
□ CAES TURBOMACHINERY BUILDING

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Indiana and Illinois has suggested that laboratory determined aquifer permeabilities in these geographic areas will be well below the 3000 md level. To simplify the discussion, therefore, only results applicable to the more relevant 750 md permeability case have been included.

3.2.4 Determination of System Pressure Losses

Pressure losses through the aquifer, wells, and above-ground pipelines have been determined for the 200 psi, 600 psi, and 1000 psi storage pressure conditions, for both the compression and withdrawal cycles. These pressure loss calculations have been used to establish an acceptable range for the external pressure loss ratio.

The pressure loss calculations completed for each typical storage pressure may be summarized as follows:

- Determination of the pressure drop between the bottom of the well and the porous bed comprising the storage media.

- Determination of the pressure change in the well bore.

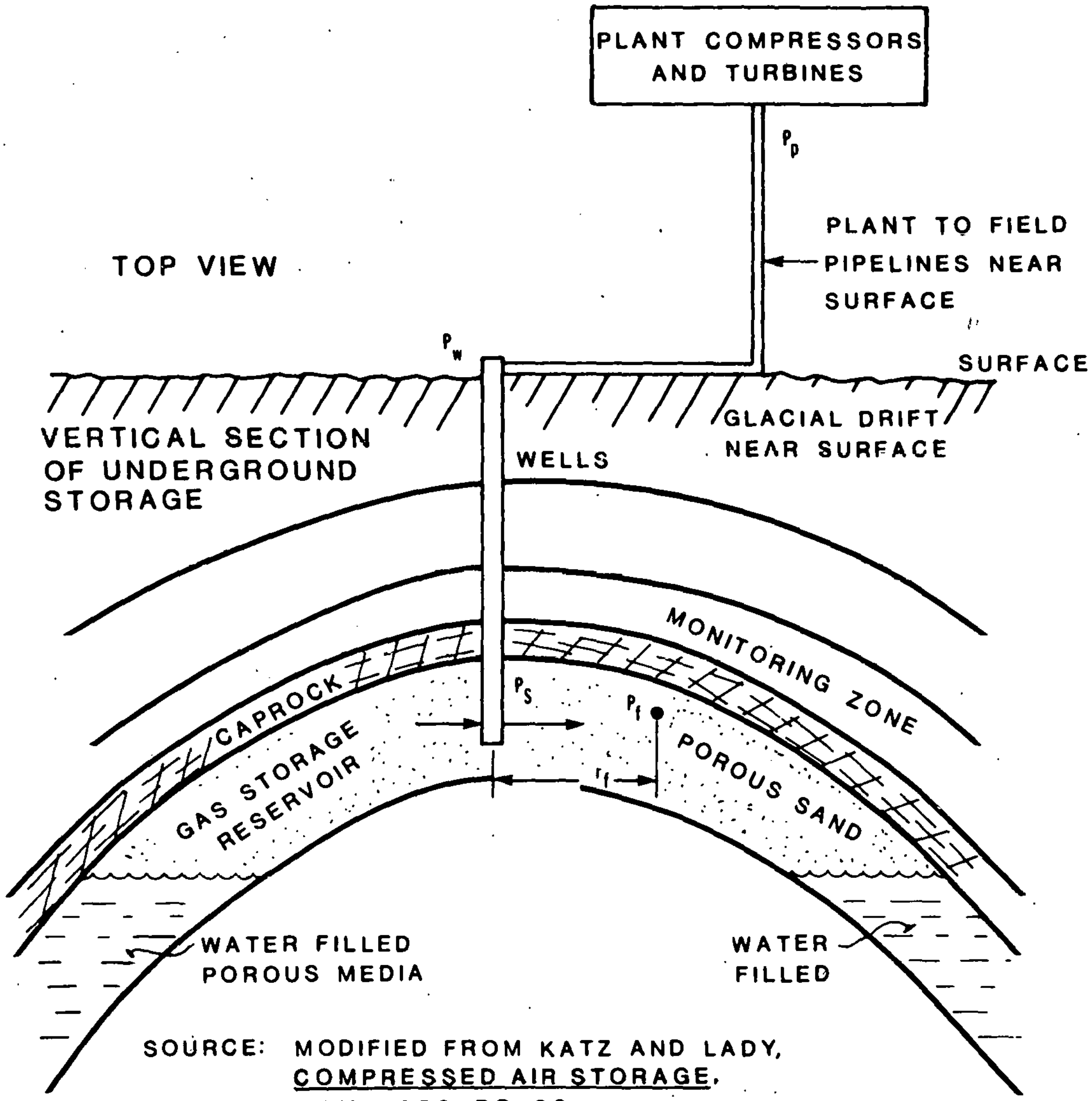
- Determination of the pressure loss in the field lines which transport air to and from the surface plant, for three feasible pipe air flow velocities.

3.2.4.1 Calculation of Pressure Losses In The Aquifer and Well.

Determination of the pressure drops through the aquifer and well bore, at the three typical storage pressures, required the calculation of the flowing sand face pressure and well head pressure for each case, per the procedures delineated by Katz & Lady (1).

The flowing sand face pressure (the pressure at the bottom of the well) P_s, the reservoir pressure P_f, and the well head pressure P_w, are shown schematically in Figure 3-5.

COMPONENTS OF UNDERGROUND STORAGE SYSTEM



SOURCE: MODIFIED FROM KATZ AND LADY, <u>COMPRESSED AIR STORAGE</u>, MAY, 1976, PG. 99.

P_p PLANT PRESSURE

P_w WELL HEAD PRESSURE

P_S FLOWING BOTTOM HOLE (SAND FACE) PRESSURE

P_f RESERVOIR PRESSURE

r_f EXTERIOR RADIUS

FIGURE 3-5

COMPONENTS OF UNDERGROUND AIR STORAGE FLOW SYSTEM WITH NOMENCLATURE

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As discussed by Katz and Lady, air entering the well bore at the bottom of the well flows radially through the porous media. The basic relation between the flow rate and pressure drop through the porous media can be obtained by integration of Darcy's Law. However, since turbulence occurs at the porous media adjacent to the well bore, an additional turbulence term must be introduced into the flow equation to account for non-laminar flow characteristics (1). Determination of flowing sand face pressures for the three typical storage conditions, then, has been based on solution of the differential flow equation describing turbulent flow.

Calculation of the pressure drop in the well bore is somewhat more involved, since the frictional losses between the flowing air and wall must be considered, in addition to the static vertical head between the top and bottom of the well. As documented in Katz and Lady's Compressed Air Storage, a simplified equation has been developed by Cullender & Brinckley for the determination of well head pressure, which combines the static head and friction loss terms. This simplified equation has been used to calculate the well head pressures at the 200 psi, 600 psi, and 1000 psi storage pressure conditions.

The pressure change through the aquifer storage media is just the difference between the storage pressure and the flowing sand face pressure, while the pressure change in the well bore is the difference between the flowing sand face and well head pressures. These pressure differentials have been determined for the 200 psi, 600 psi, and 1000 psi storage systems, for both compression and withdrawal cycles.

3.2.4.2 Calculation of Pressure Losses In Above-Ground Air Piping. The pipeline pressure loss calculations for the 200 psi, 600 psi, and 1000 psi CAES systems have been based on the typical piping networks developed for each pressure in Section 3.2.3. There is a tradeoff between increased pumping cost and decreased piping cost as the pipeline pressure drop is increased. Therefore, at each storage pressure, three different pipe air flow velocities have been considered in order to bracket reasonable combinations of piping costs and pumping power costs.

To determine the pipe sizes for the selected velocities at each pressure, the air mass flow rate required for production of 1000 Mw at that particular pressure level has been used (Table 3-3).

The piping for each pressure case was first sized to obtain the same velocity in the above-ground piping as in the underground wells. This velocity is termed the "well-bore velocity". Since 12-inch diameter wells have been used at all storage pressures, the well-bore velocity for each pressure case corresponded to the velocity of flow in a 12-inch inner diameter (I.D.) pipe.

The well bore velocity provided a lowest case velocity and an upper limit on pipe size at each storage pressure. To obtain a lower limit on pipe size at each pressure, the pipe diameters resulting in a 200 fps air velocity within the above-ground piping were calculated. An intermediate velocity of 100 fps has also been considered.

Table 3-3 summarizes the selected air flow velocities for the three storage pressure cases, at a 750 md permeability.

Table 3-3

VELOCITIES OF FLOW IN AIR PIPELINES FOR THREE TYPICAL STORAGE PRESSURES

Storage Pressure psi	Turbine Air Mass Flow Rate, M lbm/sec	Permeability md	No. of Wells	Velocities of Flow in Pipe fps
200	3759	750	500	13.3,100,200
600	3096	750	50	37.5,100,200
1000	2632	750	35	25.1,100,200

According to procedures described by Katz and Lady, a Weymouth type equation for calculation of pressure drop in horizontal pipes has been applied to determine the piping network pressure losses for the nine cases specified above. Pressure losses have been calculated for both the compression and generation operating modes for each case.

The calculated network pressure drops completed the information required for the determination of external pressure loss ratios.

3.2.5 The External Pressure Loss Ratio

The external pressure loss ratio is defined as the ratio of compressor discharge pressure to turbine inlet pressure (Pc/Pt). This ratio has been determined for each storage pressure and velocity shown in Table 3-3.

The compressor discharge pressure has been computed by starting with the aquifer storage pressure, and then adding the pressure differentials through the aquifer, well, and pipelines for the compression cycle. Similarly, the turbine inlet pressure has been determined by starting with the aquifer storage pressure and subtracting the calculated pressure losses between aquifer and turbine.

Based on the external pressure loss ratios developed for the typical storage pressures and flow velocities, a pressure loss ratio range of 1.2 to 1.8 has been established for the CAES system, with the base ratio estimated at 1.4. These data have been submitted to the Westinghouse Electric Corporation for use in heat cycle and equipment design.

3.2.6 <u>Preparation of Preliminary Piping System Costs</u>

Piping system cost estimates have been developed for the nine reference conditions outlined in Table 3-3.

To establish these cost estimates, the pipe wall thickness, pipe lengths, and required number of fittings have been computed for the typical pipeline networks (Figures 3-2, 3-3 & 3-4) at the flow velocities specified in Table 3-3. From this information, an order of magnitude cost estimate, in current prices, has been prepared for each of the nine reference cases. Included in the piping cost estimates are the costs of pipe, fittings, coating and wrapping, earthwork, field erection and welding. Valves and specialties for the piping, and instrumentation and controls, have not been included.

The cost estimates developed for the reference cases provide a base for site-specific piping cost analysis (Section 4.3).

3.3 DEVELOPMENT OF SUPPLEMENTAL HEAT CYCLE PARAMETERS

3.3.1 General

As a part of the Task 1 work effort, Sargent & Lundy has provided the Westinghouse Electric Corporation with a set of required heat cycle parameters, to be used in the development of curves for the evaluation and optimization of heat cycles. These supplemental heat cycle parameters are summarized in Table 3-4. In addition to the external pressure loss ratio and compression cycle equipment discharge temperature previously determined, the required parameters include the maximum temperature drop for above-ground piping, the cooling water supply temperature, and the regenerator minimum cold end temperature. These additional parameters are discussed briefly in the following subsections.

Table 3-4

TABULATION OF SUPPLEMENTAL HEAT CYCLE PARAMETERS

External plant Air Pressure Loss Ratio (Compression Cycle Equipment Discharge Pressure/Expansion Cycle Equipment Inlet Pressure):
a. Base = 1.40
b. Range = 1.20 to 1.80

Compression Cycle Equipment Air Discharge Temperature:
$T_a = 150^{o}F$

External Plant Air Temperature Loss:
$T = 10^{o}F$

Available Equipment Cooling Water Supply Temperature:
$T_w = 95^{o}F$

Regenerator Minimum Cold End Average Temperature, (Flue Gas Exit Temp. + Expansion Cycle Equip. Inlet Temp.)/2:
$T_{ave.} = 205^{o}F$ (note - stated temp. based on approximately 30% of total regenerator surface at cold end being of corrosion resistant low alloy steel)

3.3.2 Maximum Temperature Drop For Above-Ground Piping

In order to impose worst case conditions (and thus to maximize temperature drop), the determination of maximum temperature loss in the above-ground air piping has been based on the use of uninsulated pipe subjected to weather conditions of 10°F with a 15 mph wind. The pipe routing corresponding to the 500 well, 200 psi CAES system (Figure 3-2) has been applied to this calculation. In addition, the specified air injection temperature of 150°F has been used. As indicated in Table 3-4, these conditions resulted in a maximum expected temperature drop of 10°F for the above-ground pipelines.

This temperature drop could be reduced by approximately 2°F with the addition of 1-1/2 inches of standard calcium silicate insulation. Burial of the air pipelines would also reduce the maximum expected temperature drop.

3.3.3 Regenerator Minimum Average Cold End Temperature

The regenerator minimum average cold-end temperature, T_{ave}, is defined as follows:

$$T_{ave} = \frac{(\text{Flue Gas Exit Temp.} + \text{Expansion Cycle Equipment Inlet Temp.})}{2}$$

T_{ave} has been set at 205°F, for purposes of heat cycle and equipment design. The 205°F temperature is based on approximately 30% of the total regenerator surface at the cold end being of corrosion resistant low alloy steel. The <u>CE Process Equipment Guide for Cold-End Temperature and Material Selection</u> substantiates this choice of minimum cold-end temperature.

The 205^{O}F temperature criterion applies to fuels having a sulphur content of 1% or less. Number 2 distillate fuel oil typically has a sulphur content ranging up to 0.7%. The Standard Oil Company of Indiana also has confirmed the availability of fuel oil with a sulphur content of 1% or less.

3.3.4 Cooling Water Supply Temperature

A maximum expected cooling water supply temperature of 95^{O}F was supplied to the Westinghouse Electric Corporation for the development of curves needed in the evaluation and optimization of heat cycles. This 95^{O}F temperature selection has been based on past experience and is a conservative value; i.e., for a specific site and heat cycle, a heat exchanger-cooling system study performed to determine the most economical combination of design factors based on equipment and operating costs, could result in a lower cooling water supply temperature.

Section 4

DESIGN CRITERIA -- SITE SPECIFIC ANALYSIS OF COMPRESSED AIR ENERGY STORAGE (CAES) SYSTEM

4.1 PREPARATION OF SITE DEVELOPMENT CRITERIA

The site development and plant access criteria established in this section are used in the Task 2 evaluation of potential aquifer sites.

For economic reasons, the station should be located near the geologic formation to be used for the CAES facility. The proposed location of the station area must be selected to meet certain minimum requirements. In addition, certain preferred conditions should also be considered when selecting the station location.

The following minimum requirements for site location have been established:

a. Adequate land must be available for the initial station and possible expansion.

b. The elevation of the station area must be higher than the 100-year flood elevation of any adjacent river, creek, or creek tributary, or higher than the maximum pool elevation of an adjacent lake.

c. The station area property line must be at least 1 mile from the limits of the nearest town of reasonable size, and further away if possible. The mechanical equipment building and cooling tower areas must be at least 2000 feet from the property line, and further away if possible. The limit of 1 mile was selected to minimize the impact on any residential areas. The limit of 2000 feet was established for noise attentuation.

d. The station is to be provided with a railroad spur and an asphalt-paved access road.

In addition, the following criteria for locating the station are to be considered in the evaluation of potential sites.

1. Where practical, the station should be located adjacent to the air storage area so that structures and other facilities do not interfere with well placement. It should also be centered on the long axis of the air storage area to minimize the length of air piping mains.

2. Where possible, the station should be located to minimize the length of the railroad spur.

3. The length of road access from the nearest state, federal, or major county highway should be minimized.

4. The length of the makeup and blowdown pipelines should be minimized.

5. No more than minor relocation of cross-county pipelines, transmission lines, and primary state and federal highways should be required. Relocation of structures and secondary roads should be minimized.

6. The station area should be located on relatively level land to minimize earthwork and drainage requirements. Where the terrain of the area is hilly, the station should be located on the most level area available that best satisfies the minimum requirements and the other preferred conditions.

Estimates of site development and plant access costs have been prepared as a part of the Task 2 work effort, based on the criteria outlined above. These cost estimates are documented in the Task 2 Milestone Report.

4.2 PREPARATION OF WATER SUPPLY DEVELOPMENT CRITERIA

A set of water supply development requirements for the nominal 1000 Mw CAES plant has been defined in Task 1 for inclusion in the Task 2 site evaluation. These requirements provide a basis for identification of the most economical water sources at potential site locations.

The water supply system for the station must be capable of providing an adequate, dependable supply of water. The following minimum criteria must be met by the system before it can be considered dependable and adequate:

1. The system must have a dependable water source, which is defined as one or more points of supply that can provide the station with a minimum of 18 acre-feet of water per day.

2. The 18 acre-feet of water per day must be supplied also during the design drought period, which is a period of low rainfall (and correspondingly low area runoff and low river flow) with a recurrence of once in 100 years.

3. If the water supply is provided by pumping from a river that historical records show may be considered a dependable source, a pond with 14-day's water supply must be provided near the station to supply water in the event of a pipeline or pump malfunction.

4. If the water supply is provided by pumping from a river not considered a dependable source, a storage reservoir near the plant is required that has sufficient capacity

to supply water to the station for the maximum period during the design drought when water cannot be withdrawn from the river.

5. If the water supply is provided by constructing a dam and impounding natural runoff, it must be sized so that the volume of storage plus runoff available from the drainage area during the design drought period is sufficient to provide for station requirements, seepage, evaporation, and downstream releases from the reservoir. The reservoir at maximum pool elevation must not encroach on the spillpoint area such that it impairs the installation of wells.

The preliminary water flow requirements for the nominal 1000 Mw station are summarized in Table 4-1.

Table 4-1

MAKEUP WATER REQUIREMENTS AND BLOWDOWN FOR A NOMINAL 1000 MW CAES PLANT

	Approximate Flow		
Period	Makeup Water (gpm)	Blowdown (gpm)	Duration (hours)
Compression cycle	8,000	2,100	10
Withdrawl cycle	480	125	10
Other needs (Intermittent)	500	500	20
Average total - acre-ft per day	18	6	
Maximum rate-gpm	8,500	2,600	

Water supply and blowdown flows for a station with a capacity of less than 1000 Mw may be determined by reducing the requirements by the ratio of the station capacity in megawatts to 1000 Mw.

For sites supplied by pumping from a river, the quantity of water pumped should be increased to replace evaporation and seepage losses in the 14-day pond or larger reservoir. For sites where a reservoir has been provided to impound natural runoff, computations of the reservoir volume should include allowances for seepage, natural evaporation, and downstream releases. If blowdown has been returned to a reservoir, the reservoir volume should be computed by subtracting the blowdown rate from the makeup rate.

4.3 DEVELOPMENT OF SITE SPECIFIC PIPING COSTS

The preliminary cost estimates for the cases of three typical aquifer storage pressures at selected velocities (Section 3.2.8) have been correlated to the specific aquifer sites identified in Task 2. Correlation of the general estimates to the specific sites required a multi-stage procedure because of the interdependence of storage pressure, external pressure loss ratio, air flow velocity, and pipe size.

The costs for the actual sites were determined by interpolating between the costs for the typical reference cases. For convenience, the reference cases are reiterated below:

Storage Pressure (psi)	No. of Wells	Velocities of Flow In The Pipe (fps)
200	500	13.3,100,200
600	50	37.5,100,200
1000	35	25.1,100,200

Because the proposed sites have discovery pressures between the upper limit of 1000 psi and the lower limit of 200 psi, a family of curves was first developed to relate discovery pressure, velocity of flow, and cost of piping. The costs of piping for the nine reference cases were used to prepare these curves. From the curves a single table of base conditions was derived relating discovery pressure in increments of 100 psi between 200 and 1000 psi, the cost of piping at each pressure and the number of wells at each pressure. The velocity of flow and the number of wells at pressures of 300, 400, 500, 700, 800 and 900 psi were interpolated from the velocities and numbers of wells required for the reference conditions. The estimate of the cost of piping at each site was obtained by first using the discovery pressure at the site to determine the cost of piping for the number of wells in the reference condition. Then the total cost was derived by multiplying this cost times the ratio of the number of wells required at the site to the number of wells in the reference condition.

The site-specific number of wells was obtained from Task 2 computer analyses of the aquifer system, as documented in the Task 2 Milestone Report.

The cost of piping included the cost of pipe, fittings, coating and wrapping, earthwork, and field erection, welding, and testing. Valves and specialities for the piping were not included, and a 3-foot burial depth was used for all piping.

4.4 TRANSMISSION COSTS AND MECHANICAL EQUIPMENT COSTS

Data pertinent to the development of electrical transmission costs and mechanical equipment costs have been gathered as a part of the Task 1 work effort. These data have been incorporated into the Task 2 site suitability comparison, as explained in the Task 2 Milestone Report.

For the potential sites identified in Task 2, the Public Service Company of Indiana, Inc. (PSI) provided the capital costs for 345 Kv transmission lines between the station at each site and the nearest network lines. These capital cost estimates did not include the cost of transmission losses, but an estimate of the equivalent capital investment (ECI) cost of these losses has been established in Task 2.

The Westinghouse Electric Corporation provided the differential costing for mechanical equipment compatible with nominal storage pressures of 200 psi, 600 psi and 1000 psi. In Task 2, the cost of mechanical equipment for each site has been interpolated from these base estimates, according to the compressor rating required for the site.

Section 5

DESIGN CRITERIA -- DETAILED ANALYSIS OF COMPRESSED AIR ENERGY STORAGE (CAES) SYSTEM FOR THREE SELECTED HEAT CYCLES

5.1 SELECTION OF THREE OUT OF FOURTEEN FEASIBLE HEAT CYCLES FOR DETAILED ANALYSIS AND MACHINERY DESIGN

Heat cycle comparison summaries have been prepared by the Westinghouse Electric Corporation for nominal 200 psi, 600 psi, and 1000 psi storage pressure CAES systems. A total of fourteen feasible cycles were investigated, including four cycles compatible with a 200 psi storage pressure aquifer, six cycles compatible with a 600 psi storage pressure aquifer, and four cycles compatible with a 1000 psi storage pressure aquifer. Westinghouse developed the fourteen cycles by analyzing the various practical combinations of number of intercoolers, stage firing temperatures, and choice of whether or not to cool the turbine disc. The Sponsoring Utilities and Sargent & Lundy assisted Westinghouse in selecting a single heat cycle compatible with each storage pressure.

The low permeabilities of Indiana and Illinois aquifers make the 200 psi CAES system appear increasingly unattractive. Parametric studies performed by Sargent & Lundy regarding the development of the compressed air storage bubble in potential Indiana and Illinois aquifers have indicated that very long times are required for air bubble development at low storage pressures. However, the 200 psi heat cycle has been retained in the CAES system analysis. Since the 200 psi cycle may be applicable to aquifer storage in other regions of the country, an analysis of a 200 psi CAES system is of generic value.

5.2 REVIEW OF WESTINGHOUSE COMPRESSOR-TURBINE-GENERATOR FLOW-PRESSURE CURVES FOR COMPATIBILITY

Curves representing the compressor-turbine-generator flow characteristics for typical 200 psi, 600 psi, and 1000 psi storage pressure aquifers have been developed by the Westinghouse Combustion Turbine Systems Division. These curves have been reviewed by Sargent & Lundy for technical compatibility. The curves indicated that the compressors in each case have constant air flow characteristics, this flow being approximately 770 lb/sec.

5.3 EVALUATION AND SELECTION OF COMPRESSOR-TURBINE-GENERATOR MACHINERY CONFIGURATIONS

An initial set of seventeen possible machinery configurations have been developed by Westinghouse, based upon variations of one, two, and three shaft configurations. Three possible configurations were prepared for a 200 psi CAES system; seven candidate machinery configurations were prepared for the 600 psi system; and seven 1000 psi system machinery configurations were established.

At a Mechanical Equipment Review meeting, the following criteria were then developed for final selection of the recommended machinery configuration at each pressure level:

1. Only single shaft configurations are to be evaluated further at each system pressure level (200 psi, 600 psi, 1000 psi). As compared to the two and three shaft configurations, the single shaft configuration, for all pressure levels, has the least capital cost and incorporates the least number of rotating components and auxiliary packages. The single shaft configuration, then, provides a mechanically simple set.

2. Hardware is to be standardized to reduce development, design, manufacturing, and inventory costs.

Final selection of appropriate machinery configurations was then simplified since only one single shaft design had been developed for each pressure level.

The decision to standardize hardware enabled Westinghouse to proceed with standardization of the low pressure and intermediate pressure compressors and intercoolers as well as the low pressure turbine for the nominal 200 psi, 600 psi, and 1000 psi systems.

5.4 PREPARATION OF CAES UTILITY SYSTEM STUDY DATA SHEETS

Upon request of the Utilities System Studies Committee, Sargent & Lundy has completed CAES Unit Data Sheets for both a full scale 1000 Mw plant and a demonstration plant. These sheets provide information relevant to development and evaluation of production cost savings for a potential CAES plant. The completed sheets contain the following data:

- Compressing Capacity
- Storage Energy
- Minimum Available Reservoir Level
- Cycle Efficiency
- Annual Planned Maintenance Hours
- Heat Rate
- Expected Availability Rate
- Forced Outage Rate
- Fixed and Variable Operating and Maintenance (O. & M.) Costs

The CAES Unit Data Sheets for the full scale and demonstration plants are included as Tables 5-1 and 5-2 respectively.

The unit data sheets have been prepared on the basis of information supplied by Westinghouse. Sargent & Lundy also performed a detailed survey of applicable gas turbine availability and O. & M. cost data and verified the Westinghouse values. This survey included an analysis of historic O. & M. costs for fast starting peaking units.

TABLE 5-1

Full Scale Plant

CAES Unit Data Sheet

Compressing Capacity (MW)	821.560
Storage Energy (MWH's Compressing) Energy required to be generated elsewhere to fully charge the reservoir	8215.6
Minimum Allowable Reservoir Level (%) % of total reservoir level	90%
Cycle Efficiency (%) Generation output/compressing input	141.1%
Annual Planned Maintenance Hours (Hrs)	1008 to 3739

Heat Rate and Expected Availability Rates

	1/2 Full Capacity	Full Capacity
Capacity State (MW)	579.500	1159.000
Availability Rate (%)*	90%	71% to 81%
Heat Rate (BTU/KWHR)	3,880	3,880

Forced Outage Rates

Immature (%)	Mature** (%)	Time Required to Mature (Years)
35 to 59	19 to 29	1 1/3 to 1 2/3

Fixed O&M Cost ($/week)***	10,876
Variable O&M Cost ($/week)***	130,000
Maintenance Outage Cost ($/week)***	

*Expressed as follows:

$$\frac{\text{Number of hours unit could operate at this capacity level}}{\text{8760 Hours - Planned Maintenance Hours}}$$

**Expressed as follows:

$$\frac{\text{Forced Outage Hours}}{\text{8760 Hours - Planned Maintenance Hours}}$$

Note - The sum of the mature forced outage rate and the availability rates must add to 100%.

***1979 Dollars.

TABLE 5-2

Demonstration Plant

CAES Unit Data Sheet

Compressing Capacity (MW)	205.390
Storage Energy (MWH's Compressing) Energy required to be generated elsewhere to fully charge the reservoir	2053.9
Minimum Allowable Reservoir Level (%) % of total reservoir level	90%
Cycle Efficiency (%) Generation output/compressing input	141.1%
Annual Planned Maintenance Hours (Hrs)	1008

Heat Rate and Expected Availability Rates

	1/2 Full Capacity	Full Capacity
Capacity State (MW)	144.875	289.750
Availability Rate (%)*	95%	95%
Heat Rate (BTU/KWHR)	4,050	3,874

Forced Outage Rates

Immature (%)	Mature** (%)	Time Required to Mature (Years)
10	5	1 2/3

Fixed O&M Cost ($/week)***	2,719
Variable O&M Cost ($/week)***	32,500
Maintenance Outage Cost ($/week)***	

---

*Expressed as follows:

$$\frac{\text{Number of hours unit could operate at this capacity level}}{\text{8760 Hours - Planned Maintenance Hours}}$$

**Expressed as follows:

$$\frac{\text{Forced Outage Hours}}{\text{8760 Hours - Planned Maintenance Hours}}$$

Note - The sum of the mature forced outage rate and the availability rates must add to 100%.

***1979 Dollars.

5.5 INVESTIGATION OF EFFECTS OF INJECTION AIR DEHUMIDIFICATION

The compressed air leaving the aftercooler in a CAES plant is near saturation and any cooling of the air will cause some liquid to condense. The amount of condensation will depend on the air specific humidity, amount of pressure drop and heat loss in the manifold system and the difference between the reservoir and the injected air temperatures. This condensed liquid can cause pluggage of the porous volume, reducing permeability and thus reducing airflow.

Westinghouse used the Wiles computer code from Battelle Laboratory to determine whether the air leaving the aftercooler could be directly injected into the reservoir without subcooling. Westinghouse made a number of computer runs with the Wiles computer code after checking the mathematics and documentation of this program to insure that the program is correct. First interpretation of the computer results by Westinghouse indicated that the compressed air leaving the aftercooler can be directly injected into the reservoir without subcooling. The maximum calculated reduction in the porous volume for airflow was less than 7 percent, occurring during the first 60 hours of operation.

Review of the Westinghouse results on the reservoir dehumidification by Dr. D. L. Katz indicated that the pressure drop and heat loss in the manifold system should be included in the Westinghouse calculations. These considerations will produce a greater impact on the porous volume, reducing permeability. Dr. Katz recommends some level of dehydration be included in a CAES plant, but he was unable to suggest any specific level of dehydration. Dr. Katz recommends a $20^{o}F$ to $50^{o}F$ dew point depression.

These initial differences of opinion were resolved in further discussions between Westinghouse, Sargent & Lundy and Dr. Katz. The consensus was that the lack of dehydration after the aftercooler will cause reduced airflow into the reservoir and in Task 3, a method will be determined to verify this. Also in Task 3, the design parameters for sizing the dehydration equipment will be chosen along with selection of this hardware.

5.6 DEVELOPMENT OF GENERAL ARRANGEMENT DRAWINGS AND PIPING AND INSTRUMENTATION DIAGRAMS (P&ID's)

5.6.1 General Discussion

After selection of the most suitable heat cycle and preliminary equipment designs for the CAES plant in Task 1, Sargent & Lundy proceeded with the development of general plant arrangement drawings and piping and instrumentation diagrams. Representative drawings have been developed for typical 200 psi, 600 psi, and 1000 psi CAES systems. These general arrangements and P&ID's provide a design base for further development of plant systems and subsystems under Task 3.

CAES system design in Task 1, then, has progressed from preparation of a simple plant block diagram in Section 3.1.8, to completion of general arrangements and P&ID's for three typical storage pressure plants.

The general arrangement drawings compatible with each of the three storage pressures consist of grade and main floor plans and one cross section. A property development layout has also been provided in each case. Four piping and instrumentation diagrams have been developed for each typical storage pressure plant: one each for compressed air, fuel oil, circulating water, and demineralized and potable water.

The general arrangements and piping and instrumentation drawings for the 1000 psi storage pressure case are presented in Figures 5-1 through 5-8; Figures 5-9 through 5-16 comprise the required drawing set for the 600 psi storage pressure plant; and, finally, Figures 5-17 through 5-24 represent the 200 psi storage pressure level.

For easy reference, the drawings prepared for each of the typical CAES systems are summarized in Table 5-3.

Table 5-3

GENERAL ARRANGEMENTS AND PIPING AND INSTRUMENTATION DIAGRAMS FOR THREE TYPICAL CAES SYSTEMS

Drawing	Figure No.		
	1000 psi Case	600 psi Case	200 psi Case
Plant Development	5-1	5-9	5-17
Main Floor Plan	5-2	5-10	5-18
Ground Floor Plan	5-3	5-11	5-19
Cross Section	5-4	5-12	5-20
Main Air Piping	5-5	5-13	5-21
Fuel Oil Piping	5-6	5-14	5-23
Circulating, High Pressure & Low Pressure Service Water	5-7	5-15	5-24
Treated, Potable & Demineralized Water	5-8	5-16	5-25

The following subsections offer a brief discussion of various aspects of CAES plant arrangement and operation for the three storage pressure systems, with reference to applicable general arrangements drawings and P&ID's.

5.6.2 Overall Description of CAES Surface Plant

The proposed surface plant layouts for the 1000 psi, 600 psi, and 200 psi CAES systems are shown in Figures 5-1, 5-9, and 5-17 respectively. Plant facilities are represented by blocks on these plant development drawings; plant access roads and rail spurs have also been shown.

The CAES plant at each storage pressure must be capable of producing a nominal 1000 Mw of electricity. As a first approximation, five 200 Mw compression/generation units were postulated for the CAES plant design. With the detailed performance analyses completed by Westinghouse, the actual required number and capacity of units for the nominal 1000 Mw plant were established at each of the selected storage pressure levels. Heat and material balances for the selected storage pressure systems are presented in Volume 2 of this Task 1 report.

Per the detailed Westinghouse performance analyses, the surface plant for the 1000 psi nominal standardized system is comprised of four 290 Mw capacity compression/generation modules. The four units are arranged in a row, in a "head to tail" fashion. To maintain the modular concept, four cooling towers of three cells each have been provided. This is based on the total cooling water flow requirement of 178,160 gpm for a nominal 1000 Mw plant (Section 3), and on a total flow per cell of 20,000 gpm. The four 290 Mw units share a common fuel storage and delivery system. Four fuel oil storage tanks have been provided, according to fuel storage requirements detailed in Section 3.1.4. Each fuel storage tank is surrounded by a berm which retains 100 percent of the tank volume in the unlikely event of a major spill. In addition, a single switchyard has been located adjacent to the turbo machinery plant building, to service the entire plant complex.

The general surface plant layouts for the 600 psi and 200 psi storage pressure plants are similar to the 1000 psi case. The total plant cooling water and fuel storage requirements are approximately the same in all cases. However, the four compression/generation modules for the 600 psi storage pressure plant each have a capacity of 255 Mw, per Westinghouse design; four cooling towers of three cells each have been provided. The 200 psi CAES system requires six turbine/compressor modules of 178 Mw each. To maintain the modular concept, six cooling towers, with two cells per tower, have been included for the 200 psi system.

5.6.3 CAES Plant Arrangement

The general arrangement drawings and piping diagrams for the three typical CAES plants present the system design for a single unit only. Subsequent units follow essentially the same design and therefore are not shown on the drawings.

The Westinghouse Electric Corporation supplied diagrams to indicate the preliminary equipment outlines for the compressor-turbine-generator machinery compatible with the 200 psi, 600 psi, and 1000 psi aquifer storage pressures. Outlines of the associated regenerators, aftercoolers, and intercoolers were also received, as well as drawings of a typical gas turbine air filter house, inlet silencer, exhaust stack, and exhaust silencer. All equipment outline drawings were carefully reviewed and evaluated prior to incorporation in plant general arrangement drawings.

Main floor and ground floor plant arrangement drawings have been prepared to show the relative placement of the various plant components, for the 1000 psi, 600 psi, and 200 psi CAES systems. For each storage pressure system, the turbine/compressor units include all the necessary turbo-machinery, heat exchangers, valves, and auxilliaries required for compression of air and generation of electricity.

As indicated on the main floor plan drawing for each proposed CAES system, one control room has been provided to service two units; this arrangement reflects the Task 1 control room philosophy of the Sponsoring Utilities. The control room houses the master control board for each unit as well as data log typers. The shift supervisor's office work space has also been located within the control room. The administration offices, locker, washroom and other service facilities are adjacent to the main control room.

The accessories required for the turbomachinery lubrication system, including oil filtering and treatment equipment, have been estimated on the basis of past design experience. These plant accessories have been incorporated into the ground floor general arrangement drawings, along with the necessary water treatment facilities and other plant auxiliaries.

The overall height of the main turbo-machinery building and of the associated auxiliary building are shown in the general arrangement cross-sectional view. The cross-sectional plant view is essentially the same for the three storage pressure plants. The turbo machinery building is of sufficient height to accommodate an 85 ton turbine room crane. Turbine room crane minimum hook height above the main floor has been confirmed by the Westinghouse Electric Corporation at 33'-0". The turbo machinery building has also been sized to accommodate the turbo machinery dismantling set down area required by Westinghouse.

5.6.4 CAES Air System

The units comprising the CAES plant are designed for operation in either compression, generation, or aquifer independent modes. Figures 5-5, 5-13, and 5-21, illustrate the main air pipe routing for the 1000 psi, 600 psi, and 200 psi CAES systems, respectively.

During the compression phase for all three systems, air is drawn into the compressor train adjacent to the motor generator at a rate of 770 lb/sec and proceeds sequentially through the compressor-intercooler stages, exiting to the aftercooler and aquifer storage facilities at the end of the compressor train.

In the generation mode for the three typical CAES systems, air from the aquifer storage reservoir enters a regenerator and passes through the turbine-combustor system. The unit design provides for combustion prior to both the high-pressure and low-pressure turbines. From the low pressure turbine, air passes through the regenerator and out the stack. The regenerator, then, has been incorporated into the plant cycle to recover turbine exhaust heat, thereby increasing turbine cycle efficiency.

Since the plant design assumes equal compression and generation air flow rates, an aquifer storage bypass has been provided to allow operation of the unit as a conventional gas turbine in the event that the storage system is incapacitated.

The compression/intercooler system and the regenerator for the CAES plant cycle are described briefly in the following paragraphs.

5.6.4.1 Intercoolers. The plant cycle for the 1000 psi CAES system (Figure 5-5) incorporates low pressure, intermediate pressure, and high pressure compression with two intercooling stages and a single aftercooling stage. Based on manufacturers' recommendations, six low pressure intercoolers and four intermediate pressure intercoolers have been provided. Entering air, at a pressure of 14.7 psi and at a flow rate of 770 lb/sec, passes through the low pressure axial compressor (pressure ratio = 5.090) where the temperature and pressure of the air are increased before discharge to an intercooler circuit. The process air then reaches the intermediate pressure (IP) compressor (pressure ratio = 3.765) where the compressing/intercooling steps are repeated. Finally, the air is compressed again in the high pressure (HP) compressor (pressure ratio = 4.682) before entering the aftercooler and aquifer storage.

The compression cycle sequence for the 600 psi CAES system is analogous to the 1000 psi case; however, the 600 psi cycle includes two aftercoolers (Figure 5-13). The low pressure (200 psi) CAES system (Figure 5-21) requires only two compression stages and therefore only one intercooler circuit; four aftercoolers are provided for this case, per manufacturers' recommendations.

The intercooler requirements for the 200 psi, 600 psi, and 1000 psi CAES units are summarized in Table 5-4.

Table 5-4

INTERCOOLER AND AFTERCOOLER REQUIREMENTS FOR THREE TYPICAL CAES SYSTEMS

Nominal System Pressure	Number of Coolers	Cooler Size Diameter Ft. In.	Cooler Size Length Ft. In.	Filled Weight (each) LB.
<u>200 PSI</u>				
LP I/C	6	5' - 3"	38' - 0"	51,400
A/C	4	5' - 2"	38' - 0"	53,100
<u>600 PSI</u>				
LP I/C	6	5' - 3"	38' - 0"	51,400
IP I/C	4	5' - 2"	38' - 0"	53,100
A/C	2	5' - 6"	16' - 0"	66,800
<u>1000 PSI</u>				
LP I/C	6	5' - 3"	38' - 0"	51,400
IP I/C	4	5' - 2"	38' - 0"	53,100
A/C	1	5' - 10"	30' - 0"	165,000

LP = Low Pressure
IP = Intermediate Pressure
I/C = Intercooler
A/C = Aftercooler

The incorporation of staged compression with intercooling significantly reduces the power required to compress air.

The intercoolers proposed for the CAES system are conventional shell and tube type heat exchangers, designed to remove the interstage heat of compression. Cooling water flowing through the tubes in a two-pass arrangement cools the process air which is confined to the shell side of the heat exchanger. The intercooler tubes are commonly fabricated from Admiralty metal.

For the three typical CAES systems, the volumetric air flows at each external connection to the compressor and turbine set were calculated by using the state points defined in the Westinghouse Nominal Standardized Heat and Material Balances. This information, in conjunction with a maximum 200 fps air velocity suggested by equipment manufacturers, permitted calculation of the air inlet and outlet connection sizes for intercoolers and aftercoolers. These sizes are detailed on the air piping diagrams for each CAES system.

5.6.4.2 Regenerator. Incoming air from storage passes through the regenerator tubes, where it is heated for the combustion process by hot turbine exhaust gas flow.

The regenerator for the CAES plant cycle is expected to be of approximately the same size for 200 psi, 600 psi, and 1000 psi CAES systems. The regenerator for each CAES unit would consist of seven building block components connected in parallel to provide a complete regenerator. Each regenerator would be constructed of integrally finned self-cleaning stainless steel tubing welded into 5 inch thick Type 304 stainless steel tube sheets, all housed in an insulated, reinforced stainless steel casing. Effectivity of the regenerator is to be 85% for Task 1, based on an average recommended cold end temperature of 205°F.

5.6.5 Fuel Oil System

The fuel oil piping arrangement for the three typical CAES plants incorporates both a rail fill connection and a truck fill connection. The fuel oil is pumped from the rail tank car or highway tank trucks to the four main storage tanks. Each fuel oil storage tank is provided with a weather hood and flame arrestor. Two 100% capacity fuel oil unloading pumps, arranged in parallel, service the four tanks. Isolation valves have been provided on each side of the unloading pumps. This permits pump maintenance during plant operation. A ring header has also been included in the piping system design to allow for bypass of the tank farm in the event of tank failure.

From the main storage tank, fuel is delivered to the day tank for each CAES unit. Two 100% capacity fuel oil transfer pumps, with isolation valves on each side of the pumps, are provided.

Fuel is then supplied to the turbine fuel oil system of each CAES unit by redundant sets of high pressure and low pressure fuel oil pumps. Excess fuel from the unit's fuel oil combustor drains is returned to the day tank to complete the fuel cycle.

5.6.6 Circulating, High Pressure, and Low Pressure Service Water System

The Task 1 circulating and service water system P&ID's for the 1000 psi, 600 psi, and 200 psi storage pressure CAES plants are shown in Figures 5-7, 5-15 and 5-24, respectively.

Each plant design includes two 50% capacity circulating water pumps per turbine-compressor unit. These pumps supply water from the cooling tower basin of each unit to the intercooler and aftercooler circuits.

Each unit also includes two 100% capacity low pressure service water pumps and two 100% capacity service water strainers. The low pressure service water provides cooling for the generator hydrogen coolers, exciter coolers, turbine oil coolers, and hydrogen seal oil coolers. Low pressure service water is also supplied to high head service water pumps for plant fire protection and other miscellaneous plant needs.

Water leaving the intercoolers, aftercoolers and low pressure service water drains is returned to the cooling tower by way of the circulating water return line, to complete the cooling cycle.

Sizings for all major interconnecting pipe routings have been determined and are shown on the water system P&ID's.

5.6.7 Demineralized Water Treatment System

A demineralized water treatment system has been included in the unit design for the typical 1000 psi, 600 psi, and 200 psi CAES systems (Figures 5-8, 5-16, 5-25) to provide distilled quality water for NO_x control. NO_x emissions are reduced by use of water injection. The water treatment system for each CAES unit has been sized to allow 1 pound of water for each pound of fuel oil burned. This 1:1 water/fuel injection ratio has been suggested by the Westinghouse Electric Corporation. The water treatment system also supplies filtered water to the unit's chilled water make-up and potable water systems.

The water treatment process basically consists of a pretreatment step and a demineralization procedure. Deep wells are the water source for the CAES plant, and thus filtration is the only type of pretreatment required. Filtration ensures removal of most suspended solids and turbidity. Sand has been selected as the filter media, since sand is relatively inexpensive and yields effluent water of good quality. Accumulated suspended particles are removed from the sand filter by backwashing when high differential pressure across the filter vessel is realized.

Each CAES unit is supplied with its own filtration equipment and with a filtered water storage tank. Water from the filtered water storage tank is delivered to the unit's chilled water make-up system, potable water system, and demineralization system. Two 100% capacity filtered water supply pumps, arranged in parallel, provide water for both chilled water make-up needs and potable water production. A salt saturator, hypochlorite feed tanks, and sodium zeolite softeners are included for production of potable water. The potable water is stored in a domestic storage tank for each CAES unit.

Delivery of filtered water to the unit's demineralizer equipment is by means of two 100% capacity demineralizer supply pumps. Demineralization removes dissolved solids by an ion exchange process. Cation resin, which has exchangeable hydrogen ions attached to a negatively charged polymer structure, is used to remove such ions as calcium, magnesium and sodium from the influent water. Similarly, anion resin, which consists of hydroxide ions bonded to a positively charged polymeric structure, removes impurities such as sulfides, chlorides, and alkalinity.

The demineralizer train for each CAES unit is comprised of a weak acid cation tank, a strong acid cation tank, an anion tank, and a mixed bed tank. A decarbonator vessel has also been provided to further remove alkalinity by a mechanical procedure; this reduces the amount of costly anion resin required in the anion resin tank.

Since ion-exchange is a reversible process, the demineralizer train can be regenerated as the resins become exhausted to dissolved solids. Regeneration is achieved by passing a strong acid through the cation tank and caustic through the anion vessel to restore the resins to their original hydrogen and hydroxyl forms. Acid and caustic storage tanks, day tanks, and pumps, and a hot water tank have been provided for regeneration of the CAES demineralizer train.

Two 100% capacity supply pumps are provided for each CAES unit to deliver the demineralized water from a storage vessel to the unit's turbine combustors for use in NO_x control.

5.6.8 Unit Equipment Requirements

Based on Westinghouse submittals, preliminary equipment and associated motor requirements have been developed for the CAES plant. Operating parameters have been defined for the various unit pumps, motors, cooling tower fans, control valves, compressors, air conditioning chiller units, travelling screens and turbine turning gear required for the plant.

In addition, the equipment needed for boot strap start-up of a typical CAES unit has been itemized, as shown in Table 5-5.

Table 5-5

EQUIPMENT REQUIRED FOR BOOT STRAP START-UP OF ONE UNIT

System	No.	hp/Motor	Coincident	Non-Coincident
L.P. service water pump	1	400	400	
Turbine main lube oil pump	1	200	200	
Well head control valve oper.	65	5		325
Fuel oil pump	1	100	100	
Bldg. ventilation fans	2	75	150	
Instrument air compressor	1	40	40	
Air conditioning chiller unit	1	40	40	
Turning gear	1	30		30
Turning gear oil pump	1	30		30
Air side seal oil pump	1	20	20	
Turbine room sump pump	1	10	10	
H_2 side seal oil pump	1	3	3	
Fuel tank to bldg. transfer	1	20		20
			963 hp	405 h

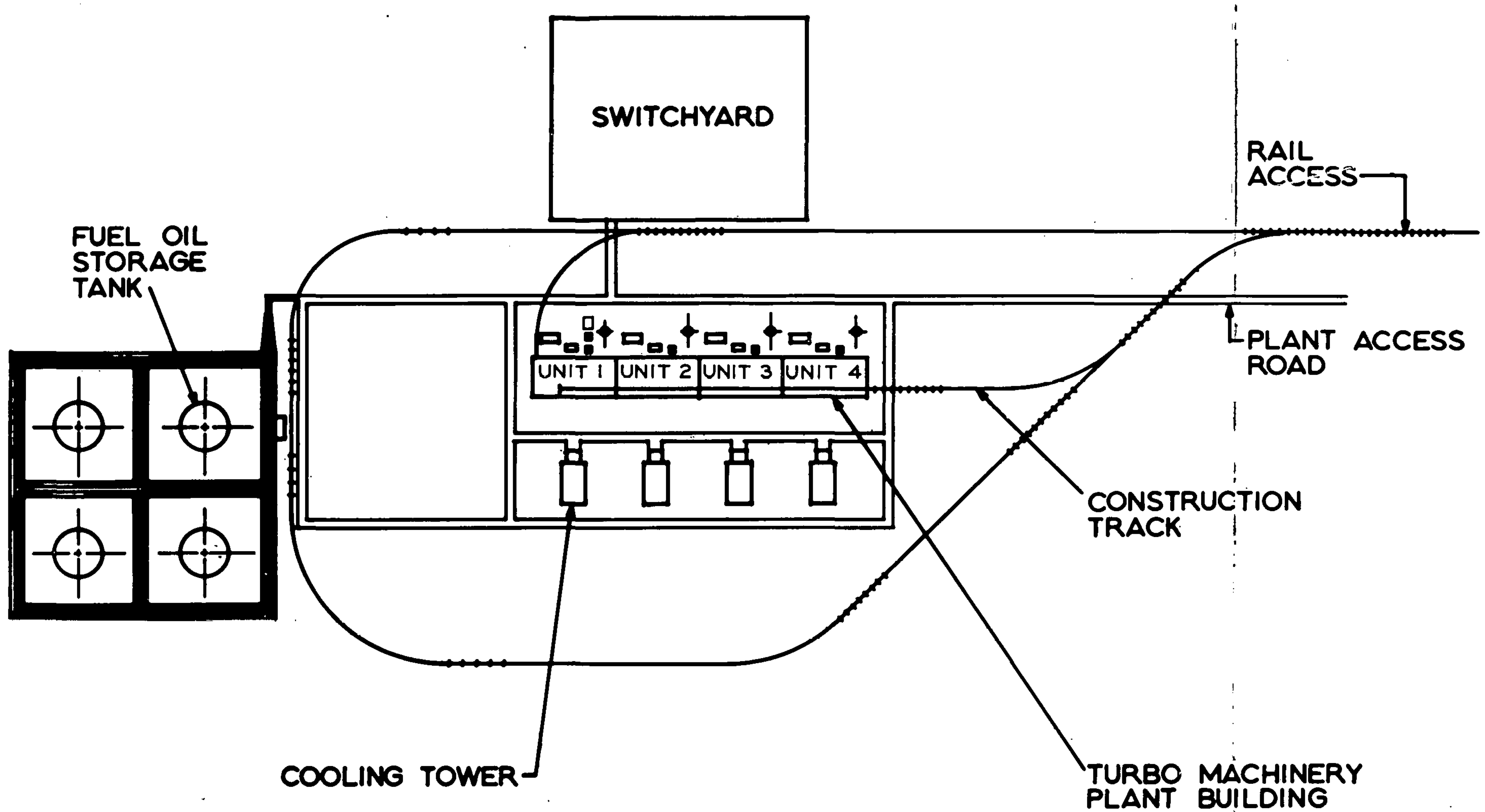


FIGURE 5-1
GENERAL ARRANGEMENT PLANT DEVELOPMENT

1000 PSI SYSTEM

SARGENT & LUNDY ENGINEERS

PUBLIC SERVICE INDIANA
CAES IN AQUIFER
DOE NO. ET-78-C-01-2159

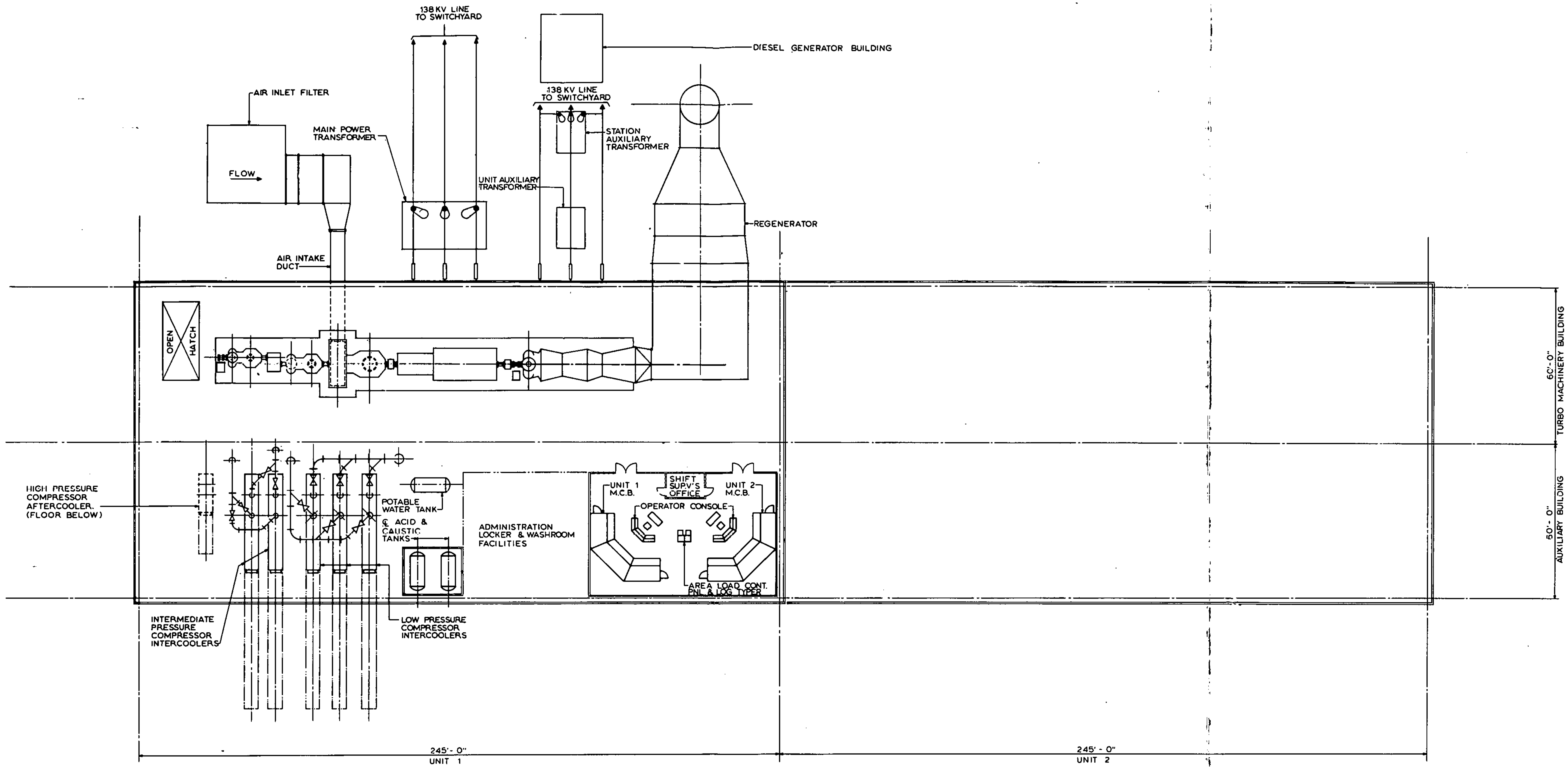


FIGURE 5-2
MAIN FLOOR PLAN

1000 PSI SYSTEM

SARGENT & LUNDY ENGINEERS

PUBLIC SERVICE INDIANA
CAES IN AQUIFER
DOE NO. ET-78-C-01-2159

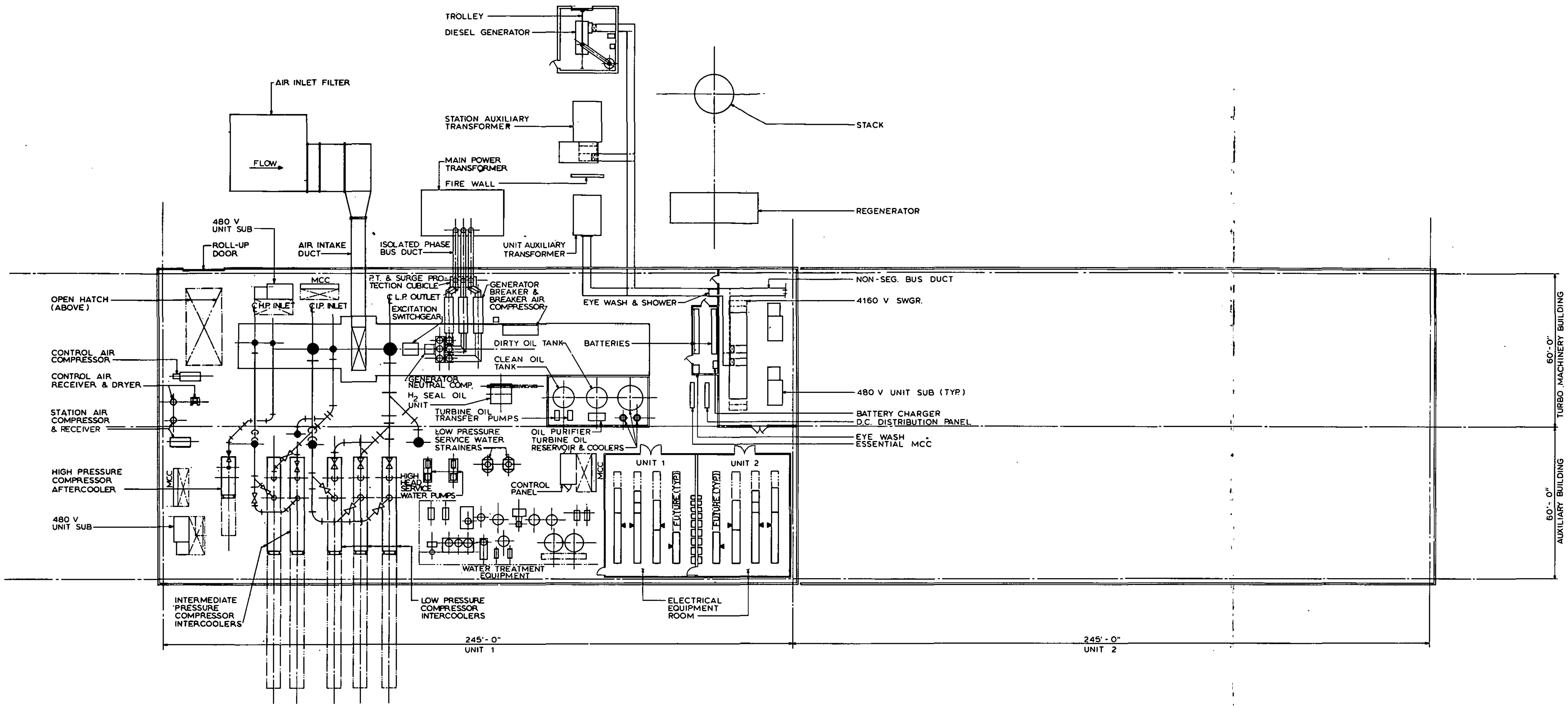


FIGURE 5-3
GROUND FLOOR PLAN

1000 PSI SYSTEM

SARGENT & LUNDY ENGINEERS

PUBLIC SERVICE INDIANA
CAES IN AQUIFER
DOE NO. ET-78-C-01-2159

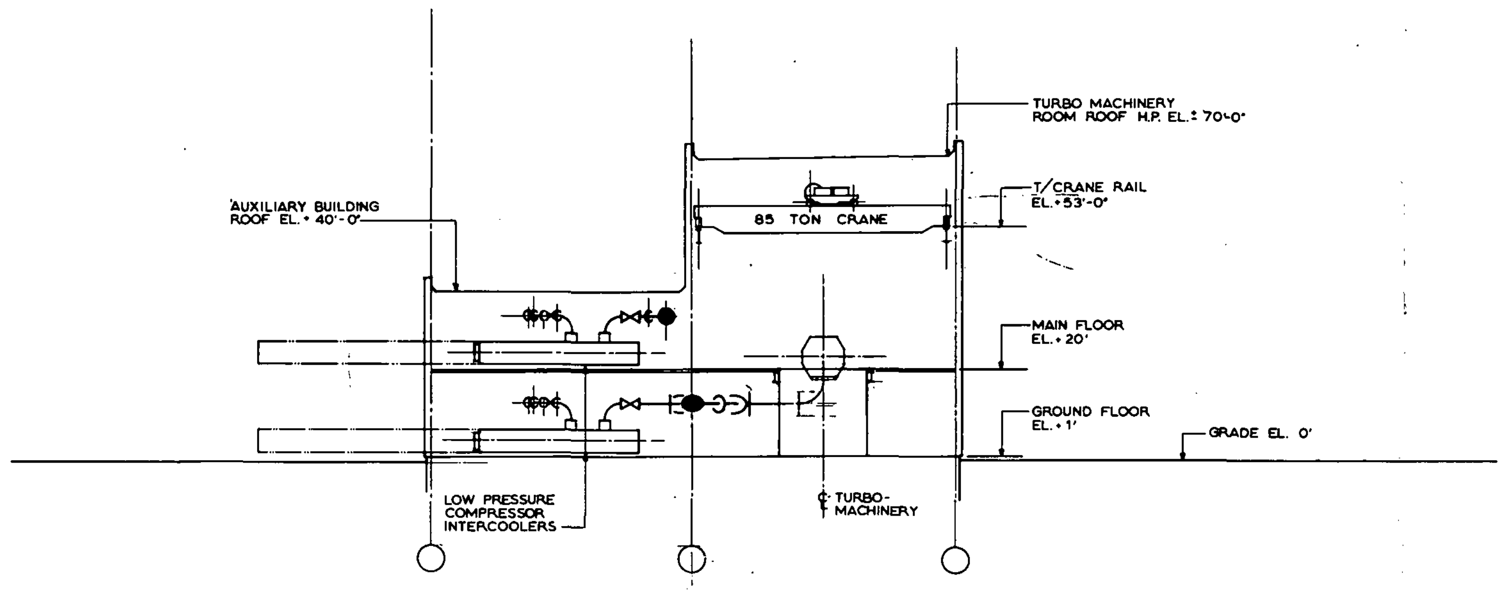


FIGURE 5-4
GENERAL ARRANGEMENT CROSS SECTION

1000 PSI SYSTEM

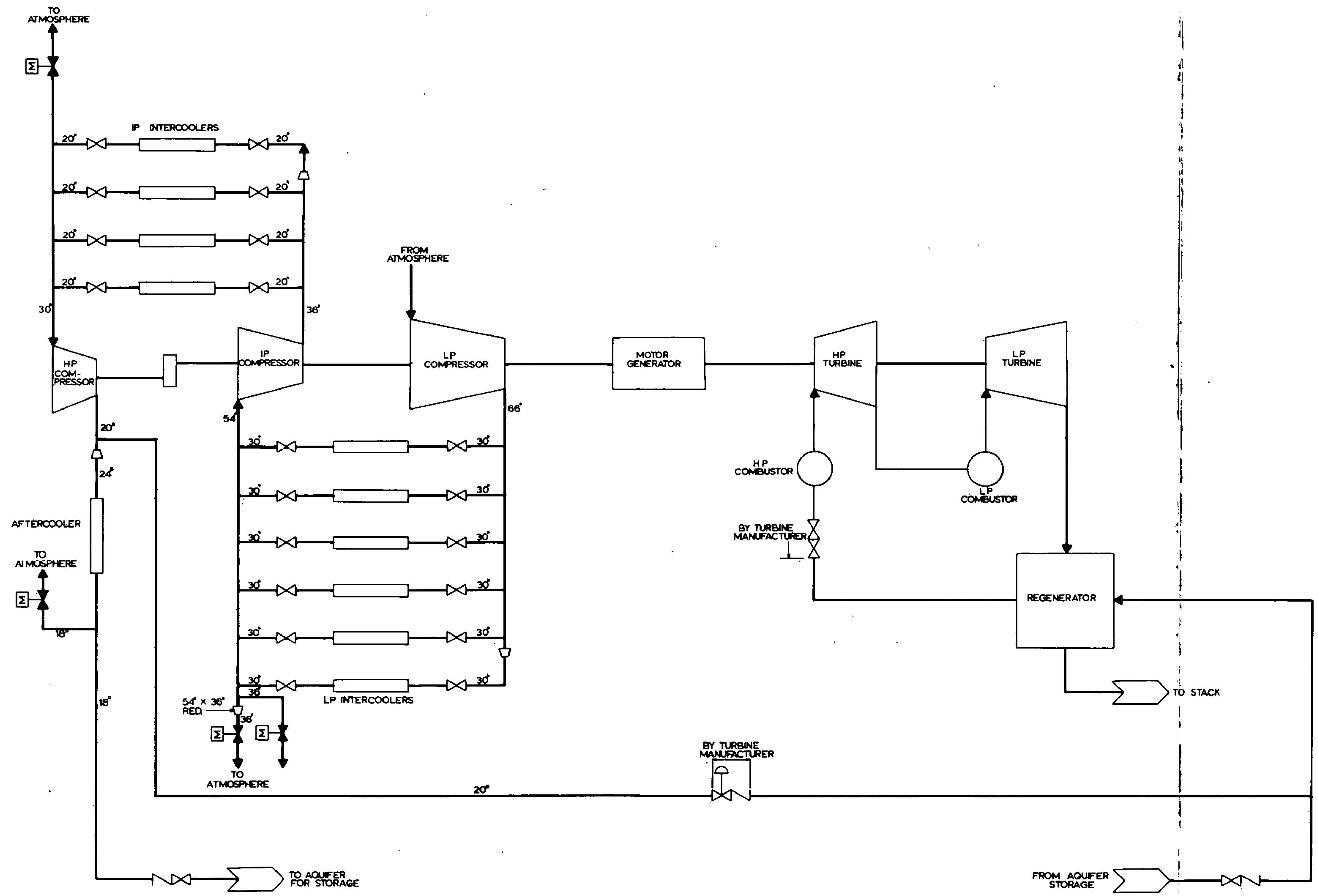
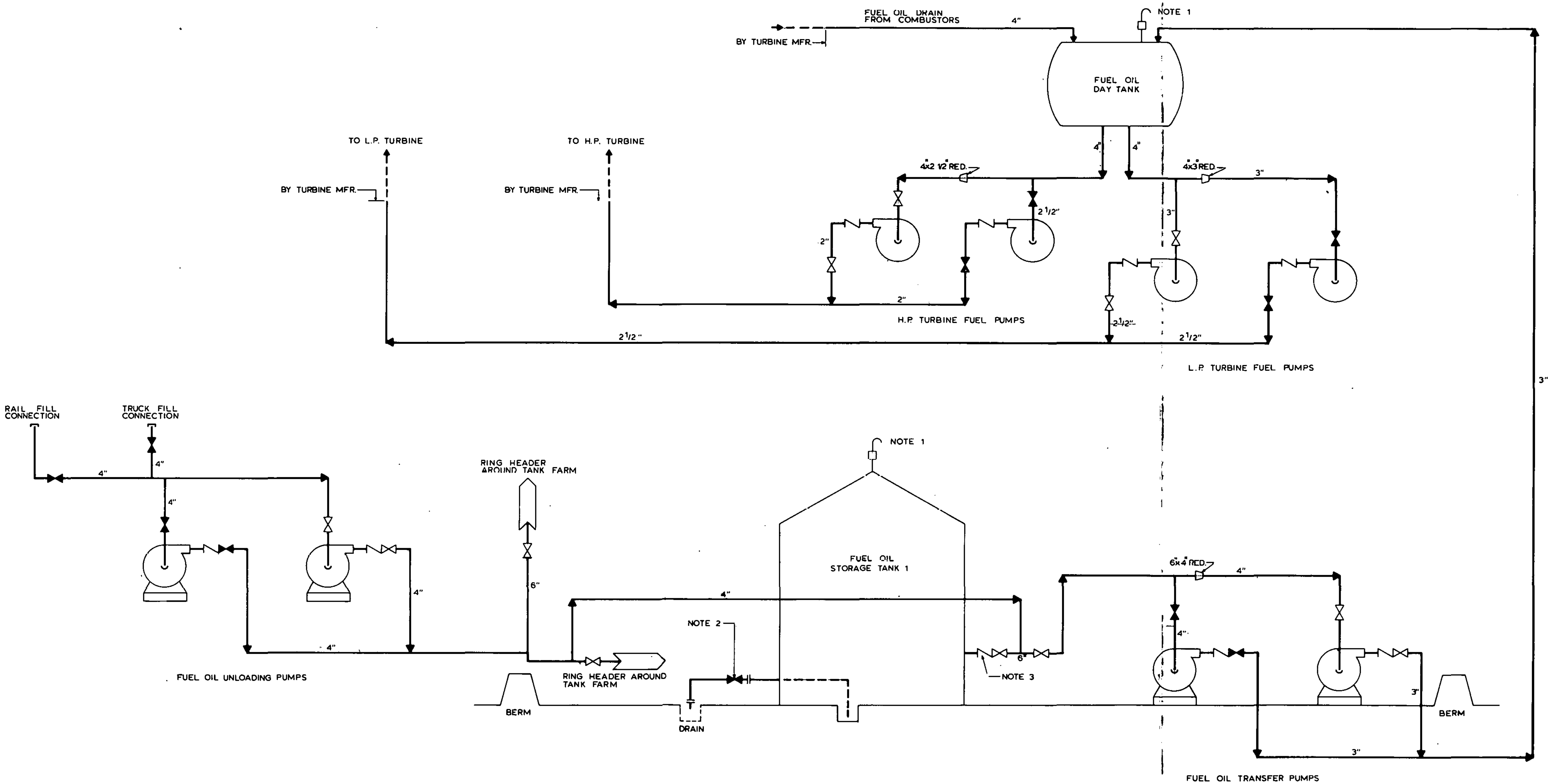


FIGURE 5-5
DIAGRAM OF MAIN AIR PIPING

1000 PSI SYSTEM

SARGENT & LUNDY ENGINEERS

PUBLIC SERVICE INDIANA
CAES IN AQUIFER
DOE NO. FT-78-C-01-2159



NOTES:
1. VAREC FIG. 27 & 50E WEATHER HOOD & FLAME ARRESTOR
2. VAREC FIG. 5000 WATER DRAIN OFF VALVE WITH FIG. 307 MOUNTING FLANGE BY P.C.
3. VAREC FIG. 5070 INTERNAL SAFETY VALVE.

FIGURE 5-6
DIAGRAM OF FUEL OIL PIPING

1000 PSI SYSTEM

SARGENT & LUNDY
ENGINEERS

PUBLIC SERVICE INDIANA
CAES IN AQUIFER
DOE NO. ET-78-C-01-2159

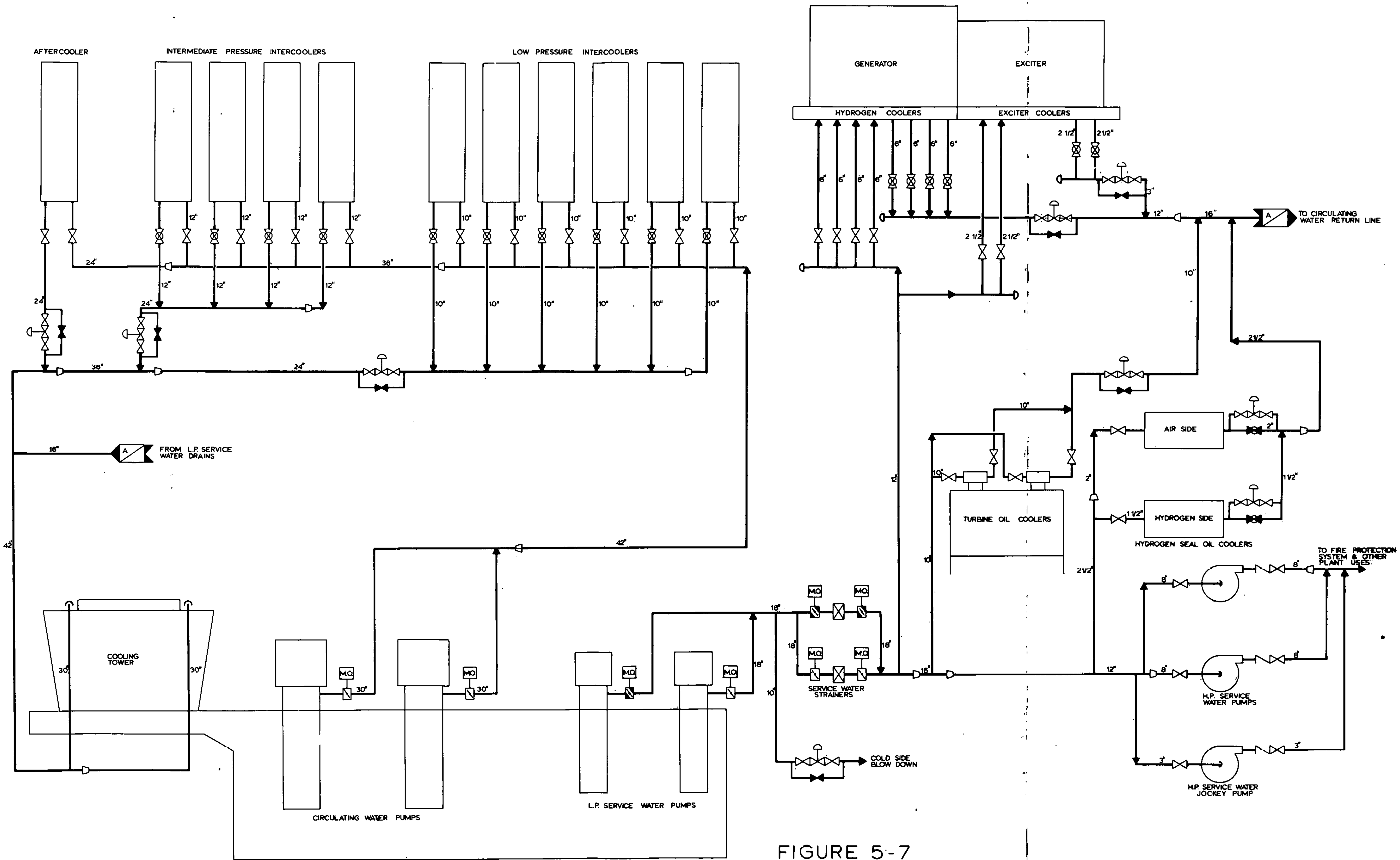


FIGURE 5-7

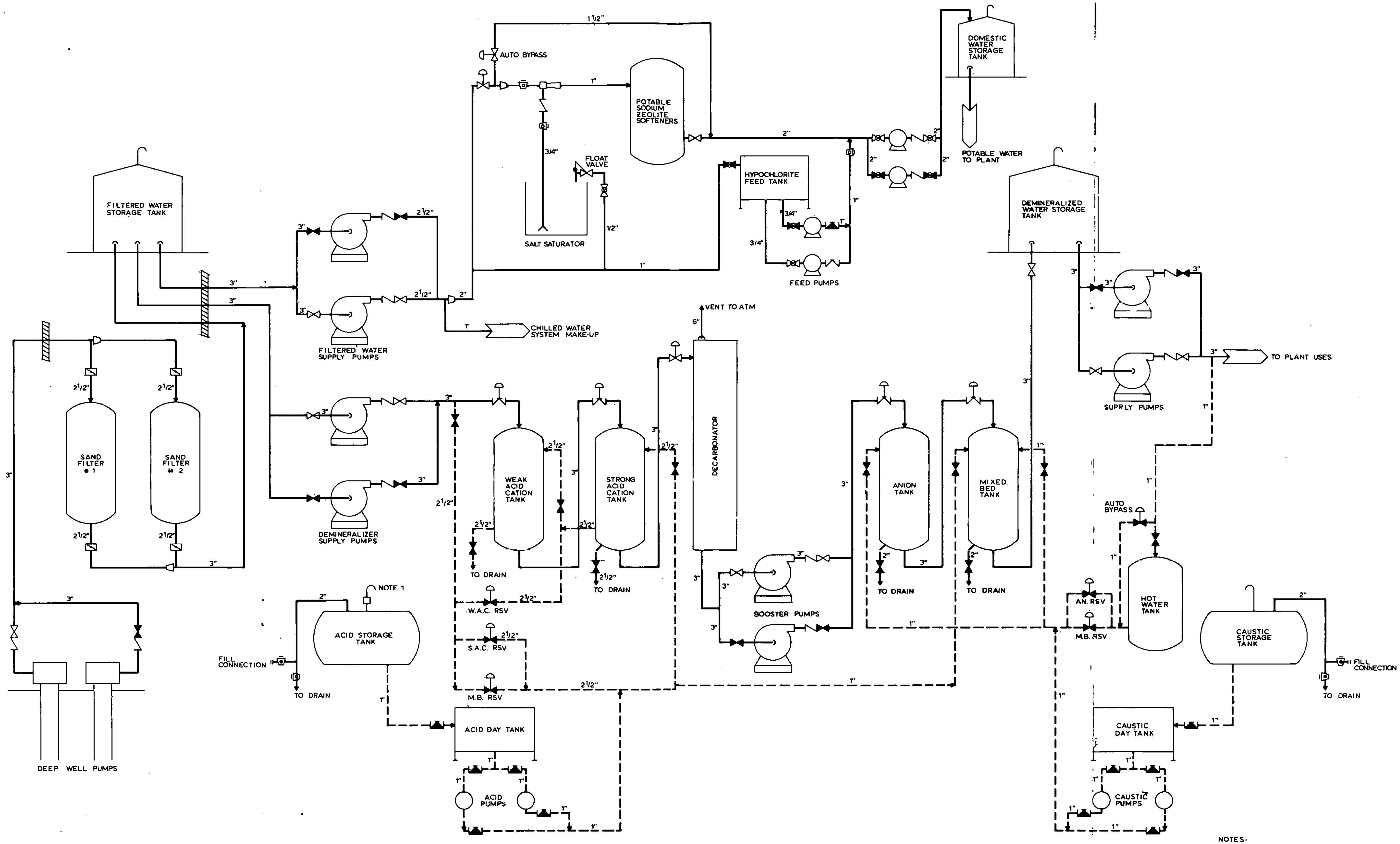
DIAGRAM OF CIRCULATING, HIGH PRESSURE & LOW PRESSURE

SERVICE WATER

1000 PSI SYSTEM

SARGENT & LUNDY ENGINEERS

PUBLIC SERVICE INDIANA
CAES IN AQUIFER
DOE NO. ET-78-C-01-2159



NOTES:
1. DRY BREATHER WITH REVERSE BEND
2. DASHED LINE DENOTES REGENERATION MODE
3. RSV=RATE SET VALVE

FIGURE 5-8
DIAGRAM OF TREATED, POTABLE & DEMINERALIZED WATER PIPING

1000 PSI SYSTEM

SARGENT & LUNDY ENGINEERS

PUBLIC SERVICE INDIANA
CAES IN AQUIFER
DOE NO. ET-78-C-01-2159

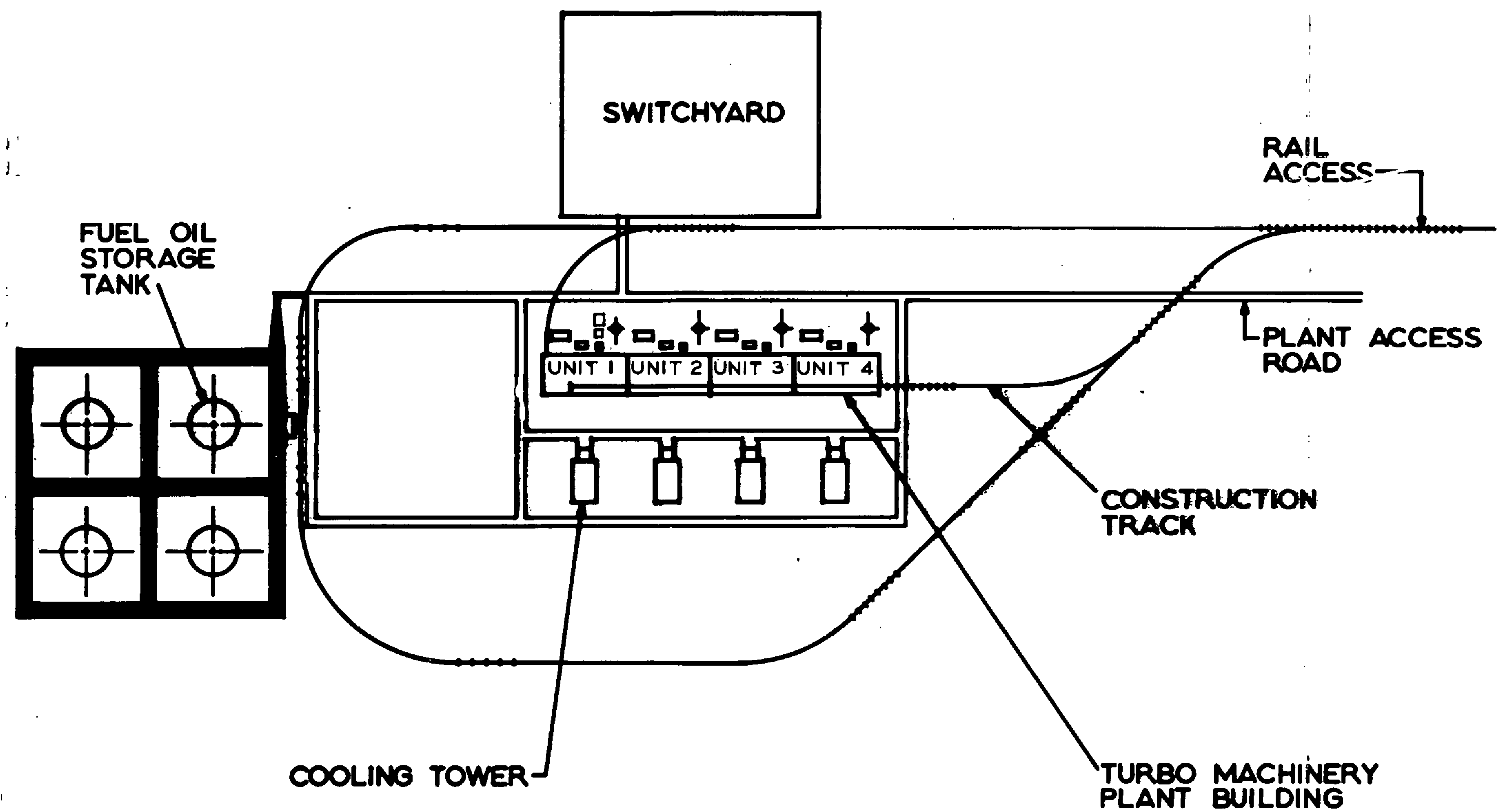


FIGURE 5-9
GENERAL ARRANGEMENT PLANT DEVELOPMENT

600 PSI SYSTEM

SARGENT & LUNDY ENGINEERS

PUBLIC SERVICE INDIANA
CAES IN AQUIFER
DOE NO. ET-78-C-01-2159

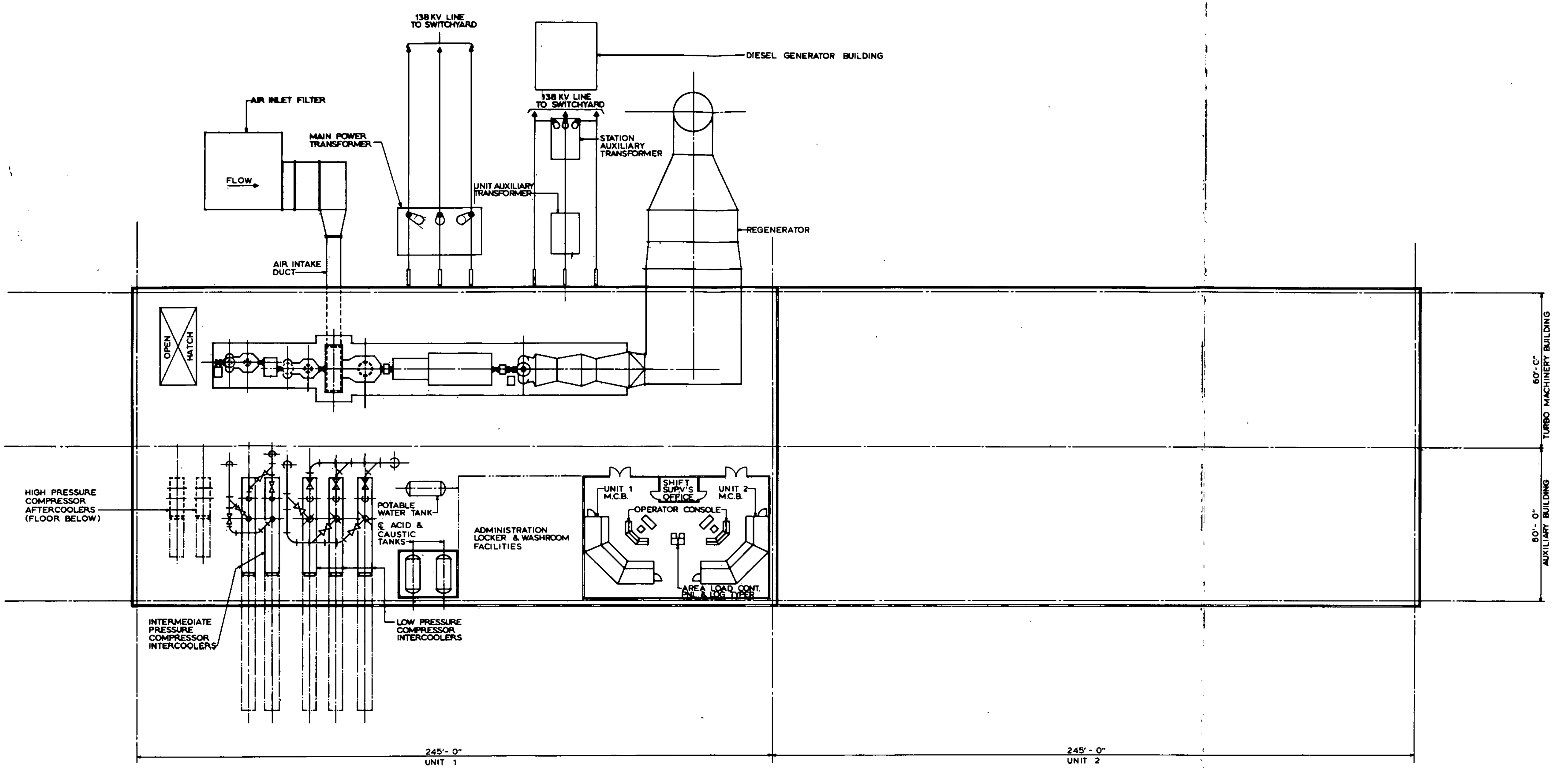


FIGURE 5-10
MAIN FLOOR PLAN

600 PSI SYSTEM

SARGENT & LUNDY
ENGINEERS

PUBLIC SERVICE INDIANA
CAES IN AQUIFER
DOE NO. ET-78-C-01-2159

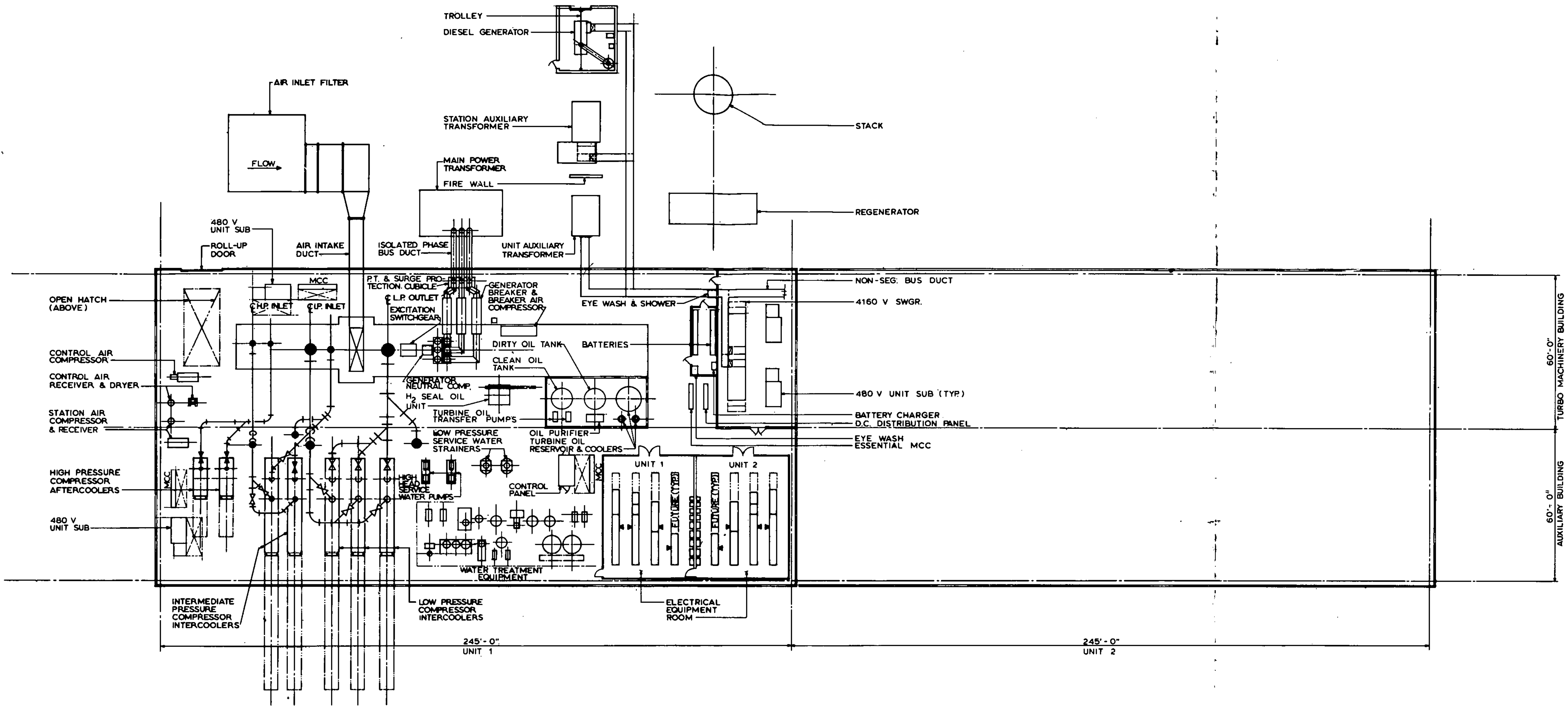


FIGURE 5-11
GROUND FLOOR PLAN

600 PSI SYSTEM

SARGENT & LUNDY ENGINEERS

PUBLIC SERVICE INDIANA
CAES IN AQUIFER
DOE NO. ET-78-C-01-2159

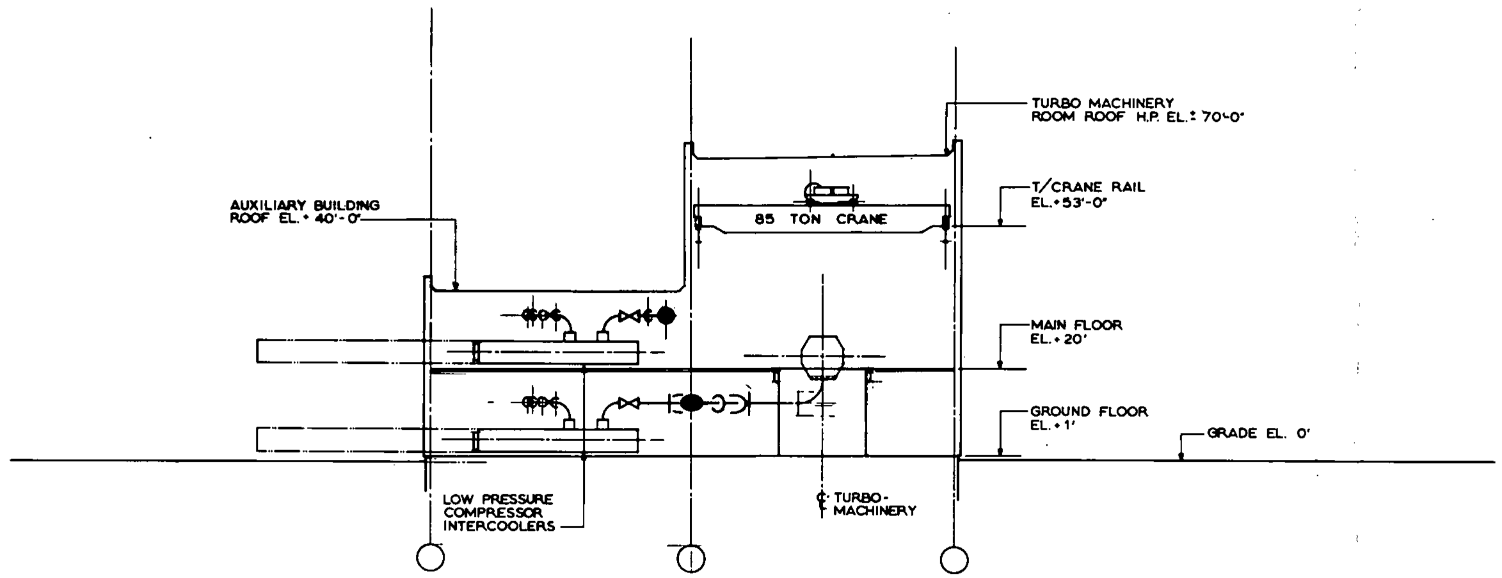


FIGURE 5-12
GENERAL ARRANGEMENT CROSS SECTION

600 PSI SYSTEM

SARGENT & LUNDY
ENGINEERS

PUBLIC SERVICE INDIANA
CAES IN AQUIFER
DOE NO. ET-78-C-01-2159

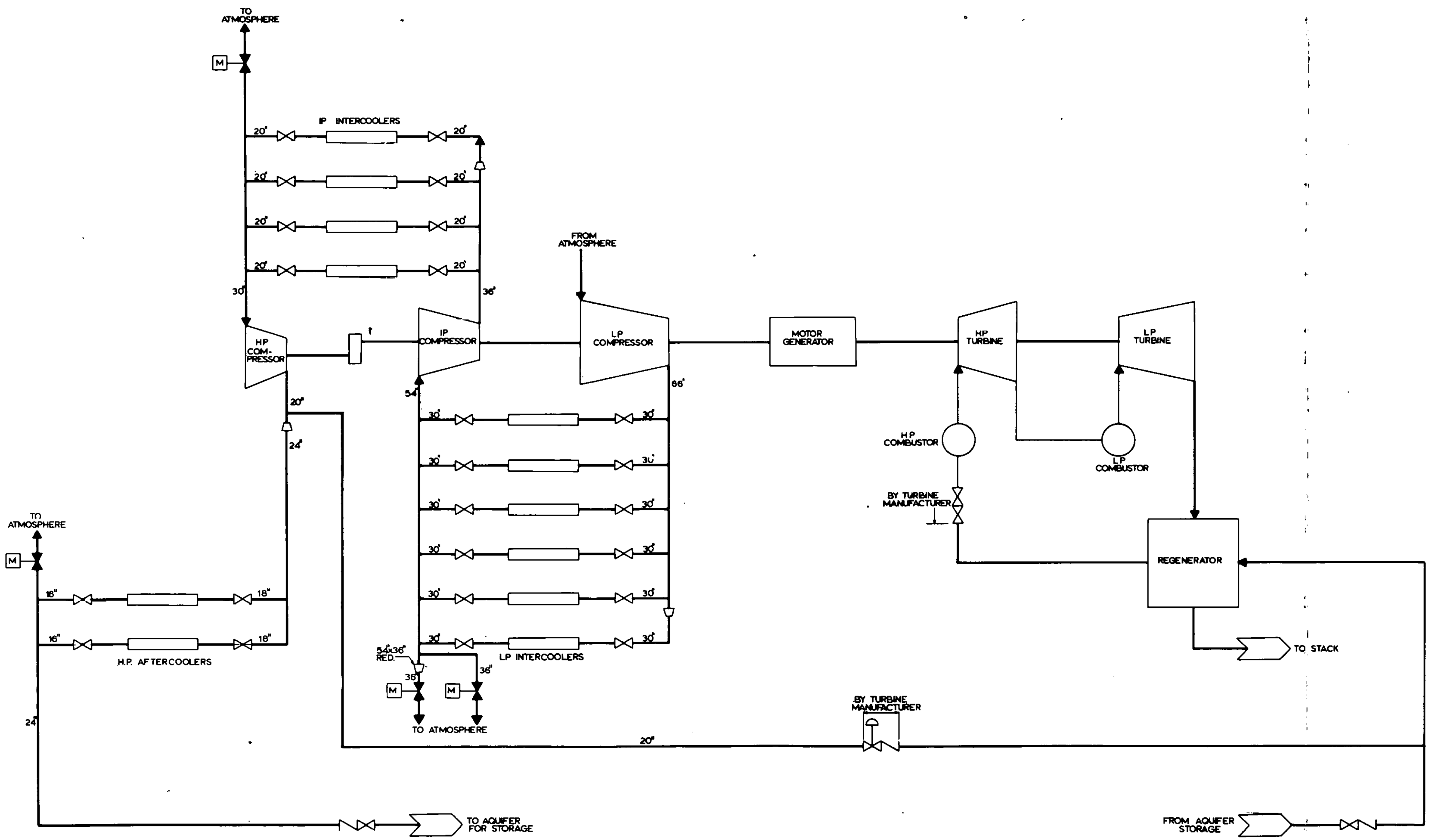


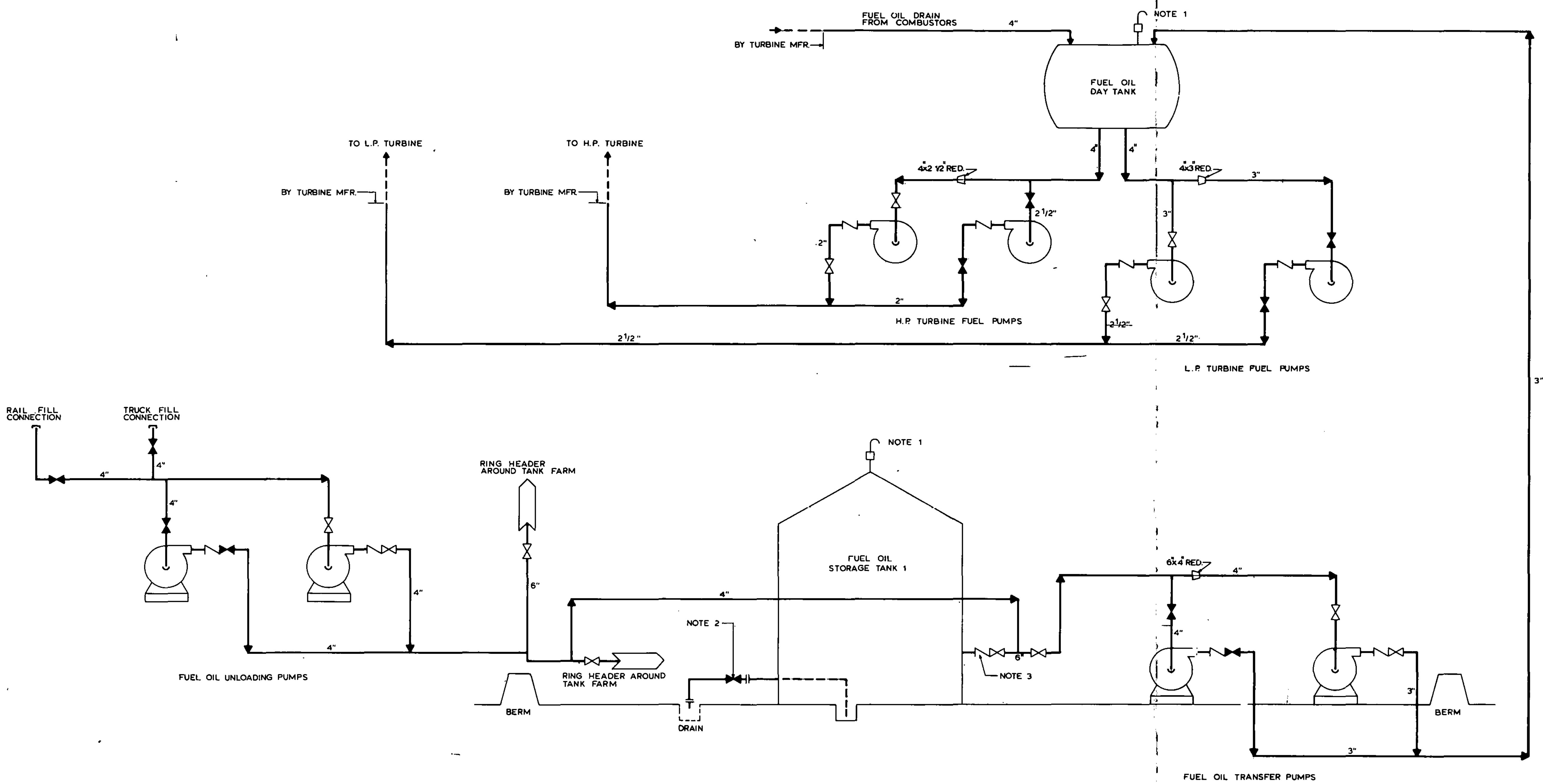
FIGURE 5-13

DIAGRAM OF MAIN PIPING

600 PSI SYSTEM

SARGENT & LUNDY ENGINEERS

PUBLIC SERVICE INDIANA
CAES IN AQUIFER
DOE NO. ET-78-C-01-2159



NOTES:

1. VAREC FIG. 27 & 50E WEATHER HOOD & FLAME ARRESTOR
2. VAREC FIG. 5000 WATER DRAIN OFF VALVE WITH FIG. 307 MOUNTING FLANGE BY P.C.
3. VAREC FIG. 5070 INTERNAL SAFETY VALVE.

FIGURE 5-14

DIAGRAM OF FUEL OIL PIPING

600 PSI SYSTEM

SARGENT & LUNDY
ENGINEERS

PUBLIC SERVICE INDIANA
CAES IN AQUIFER
DOE NO. ET-78-C-01-2159

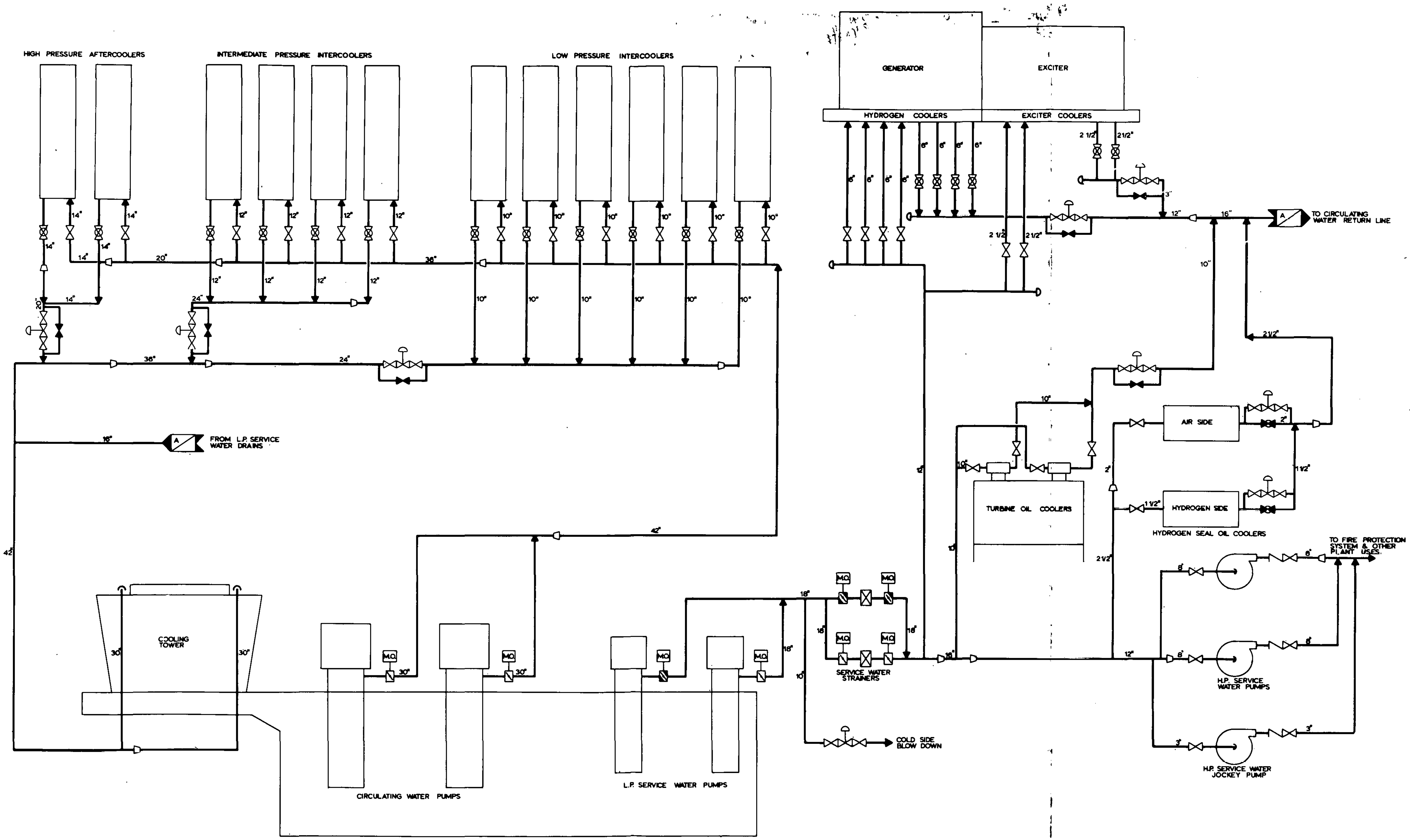


FIGURE 5-15

DIGRAM OF CIRCULATING, HIGH PRESSURE & LOW PRESSURE SERVICE WATER

600 PSI SYSTEM

SARGENT & LUNDY
ENGINEERS

PUBLIC SERVICE INDIANA
CAES IN AQUIFER
DOE NO. ET-78-C-01-2159

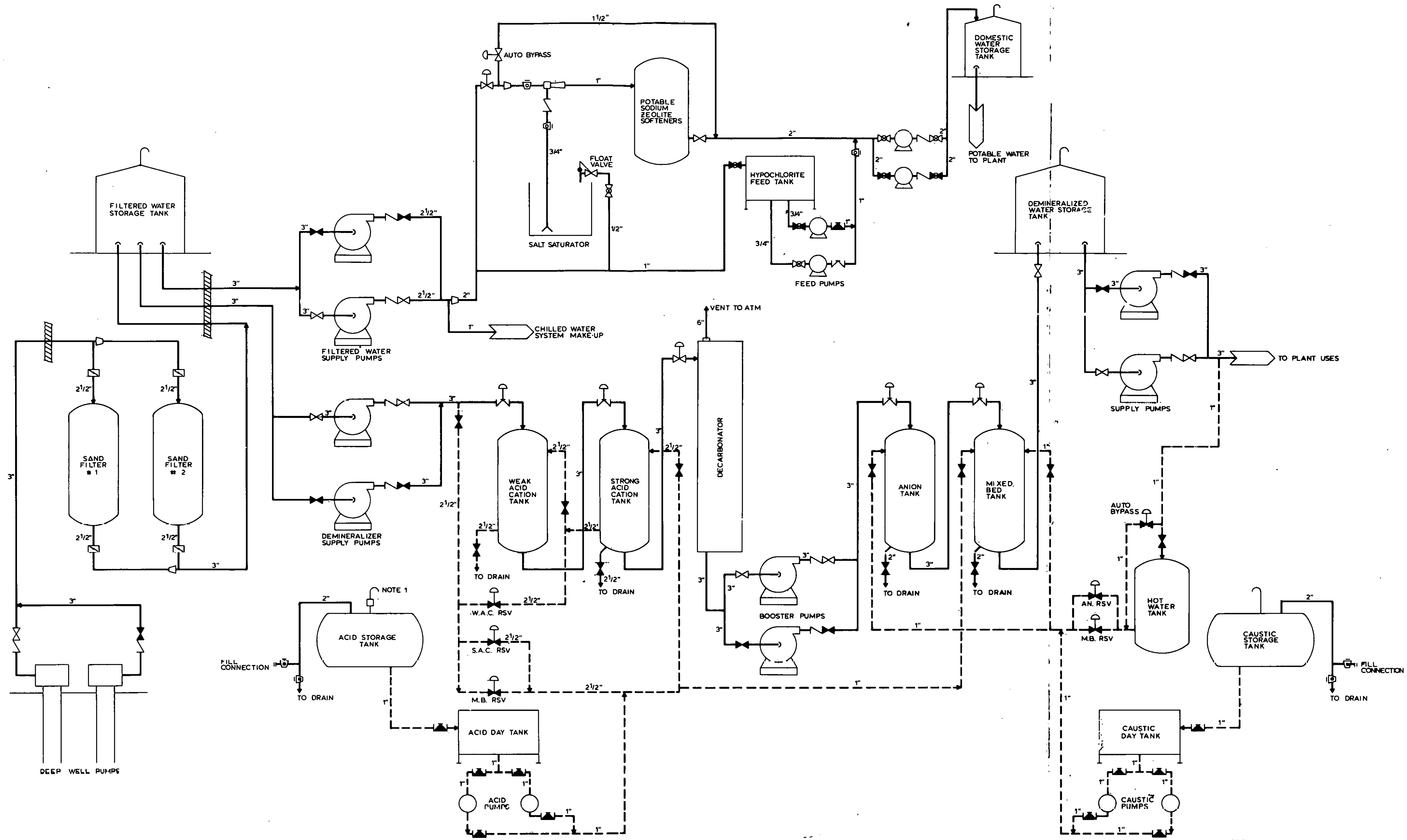


FIGURE 5-16

DIAGRAM OF TREATED, POTABLE & DEMINERALIZED WATER PIPING

600 PSI SYSTEM

NOTES:
1. DRY BREATHER WITH REVERSE BEND
2. DASHED LINE DENOTES REGENERATION MODE
3. RSV = RATE SET VALVE

SARGENT & LUNDY ENGINEERS

PUBLIC SERVICE INDIANA
CAES IN AQUIFER
DOE NO. ET-78-C-01-2159

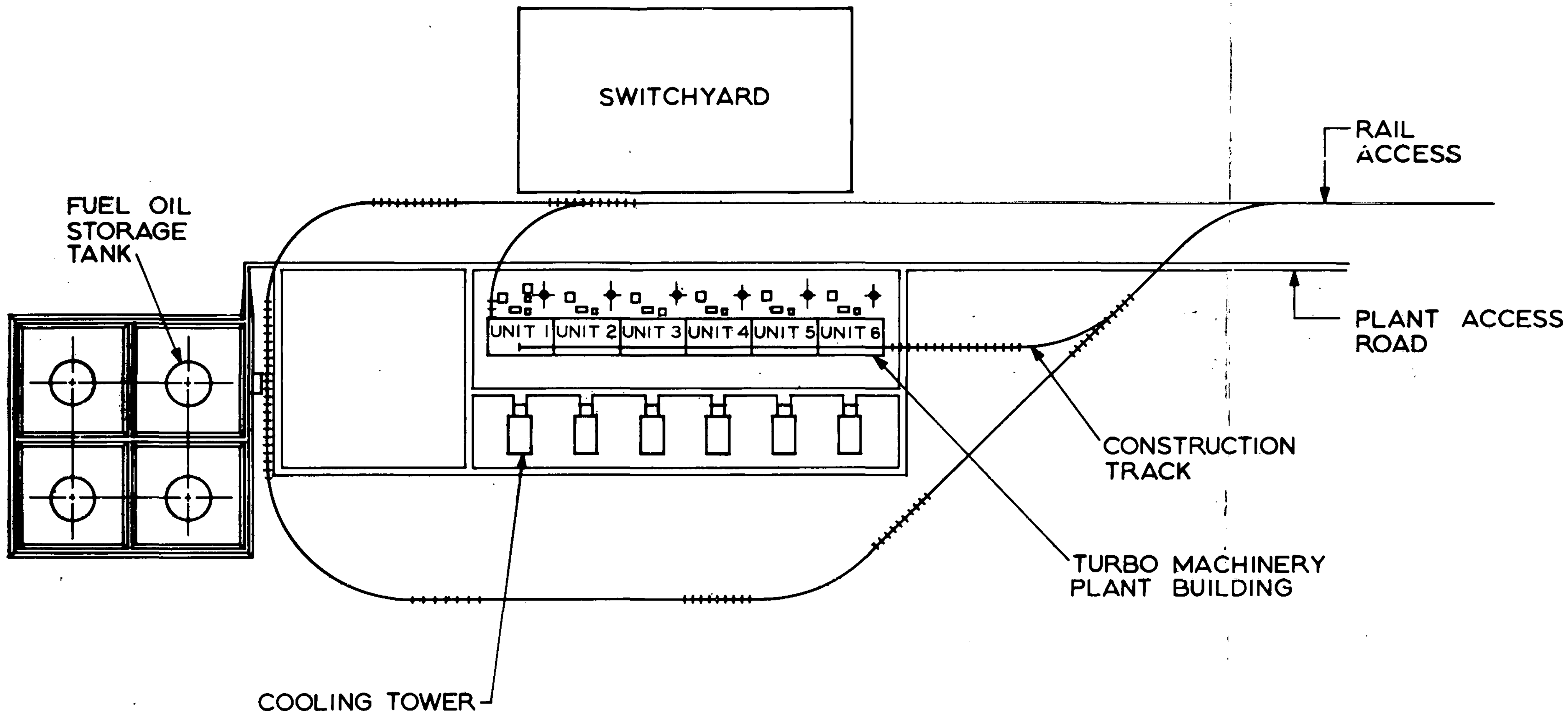


FIGURE 5-17

GENERAL ARRANGEMENT PLANT DEVELOPMENT

200 PSI SYSTEM

SARGENT & LUNDY ENGINEERS

PUBLIC SERVICE INDIANA
CAES IN AQUIFER
DOE NO. ET-78-C-01-2159

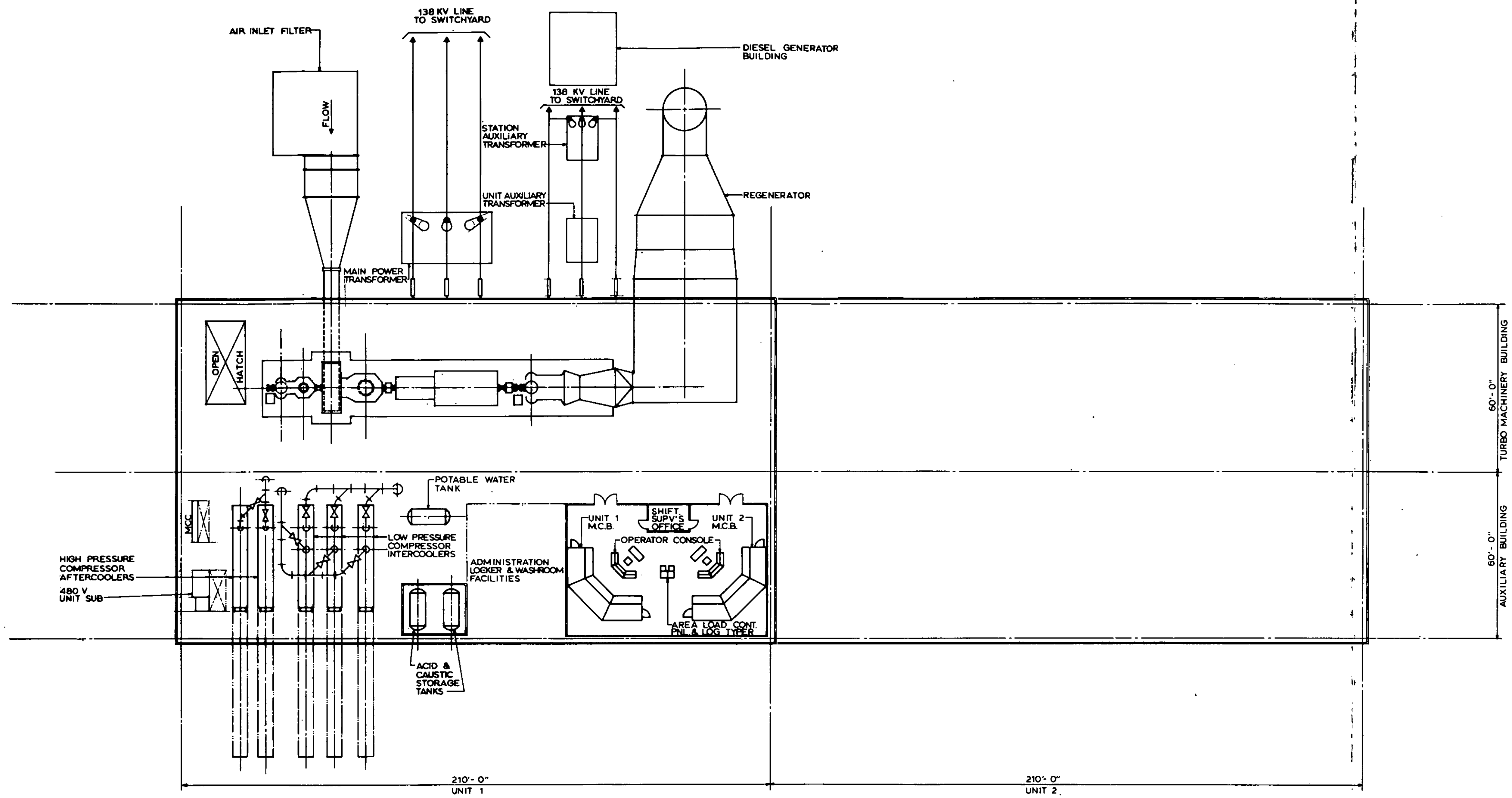


FIGURE 5-18
MAIN FLOOR PLAN
200 PSI SYSTEM

SARGENT & LUNDY ENGINEERS

PUBLIC SERVICE INDIANA
CAES IN AQUIFER
DOE NO. ET-78-C-01-2159

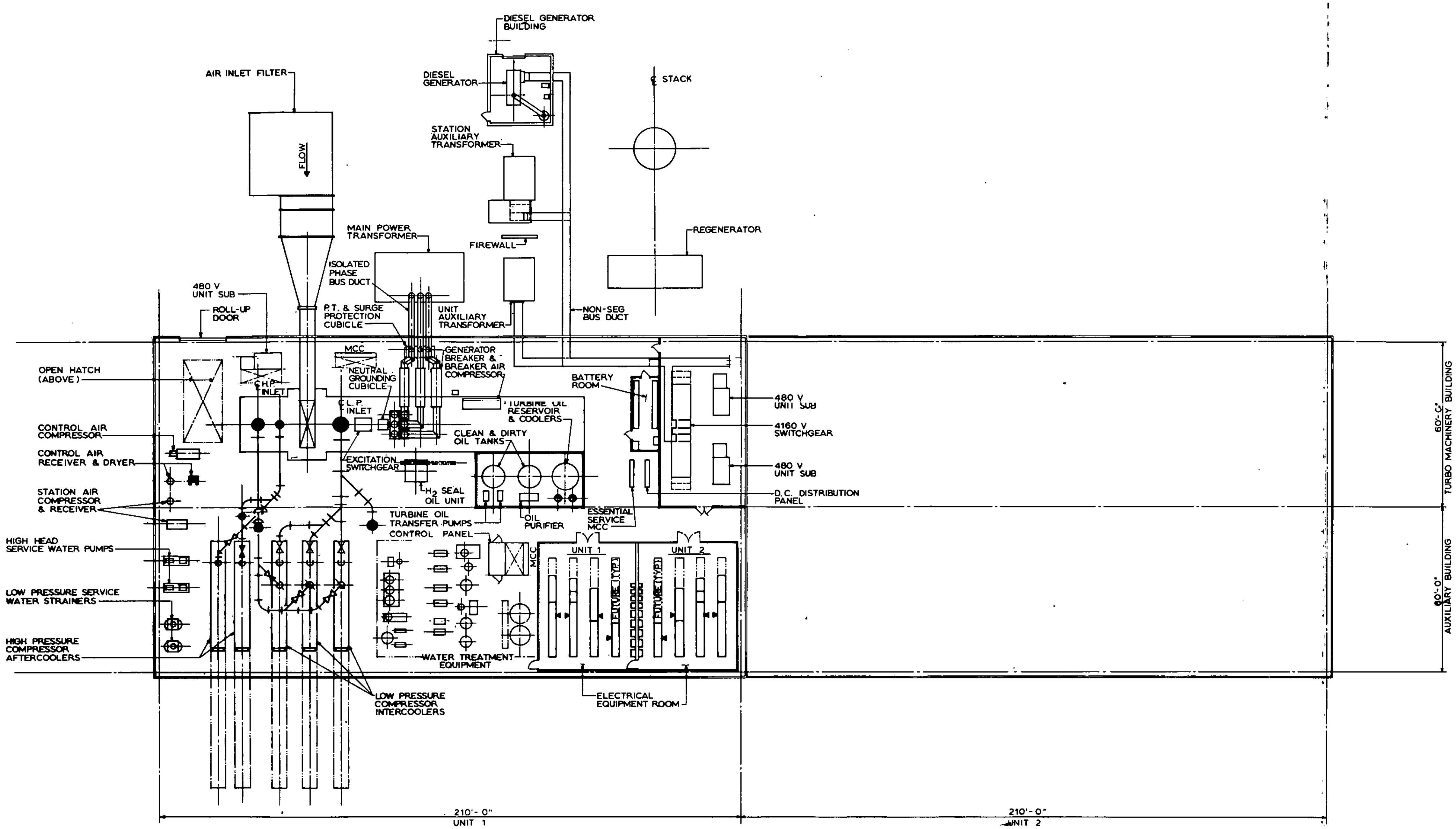


FIGURE 5-19

GROUND FLOOR PLAN

200 PSI SYSTEM

SARGENT & LUNDY ENGINEERS

PUBLIC SERVICE INDIANA
CAES IN AQUIFER
DOE NO, ET-78-C-01-2159

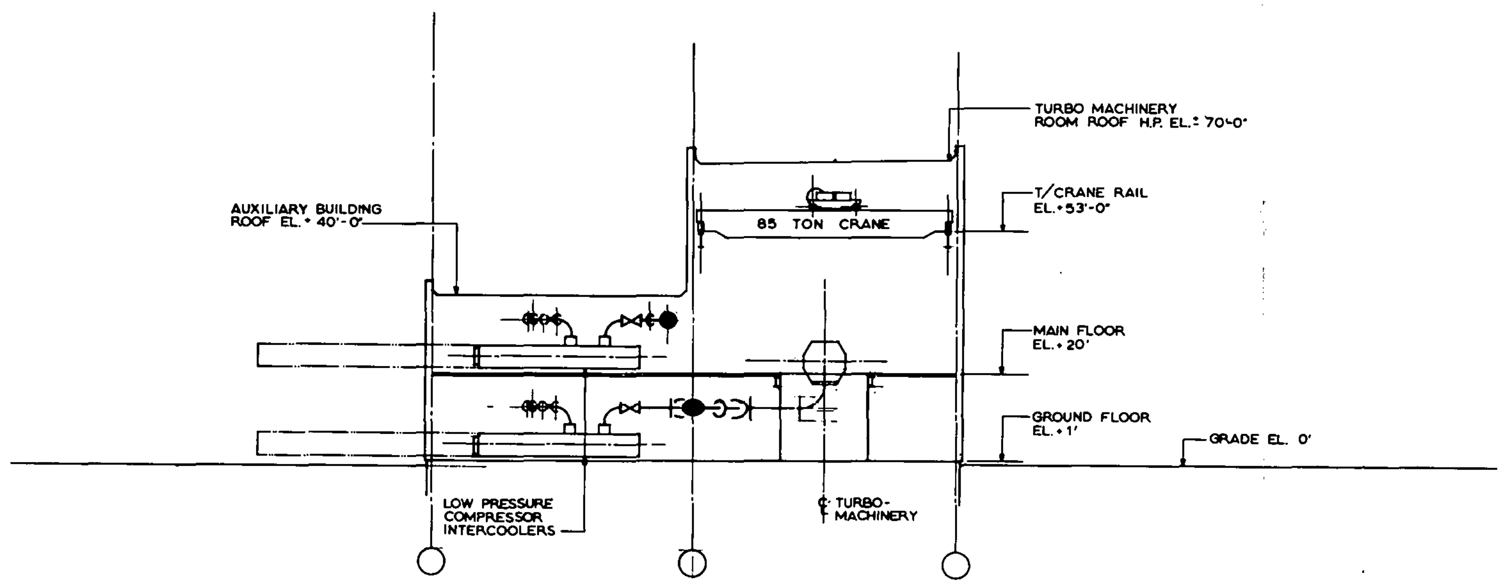


FIGURE 5-20

GENERAL ARRANGEMENT CROSS SECTION

200 PSI SYSTEM

SARGENT & LUNDY ENGINEERS

PUBLIC SERVICE INDIANA
CAES IN AQUIFER
DOE NO. ET-78-C-01-2159

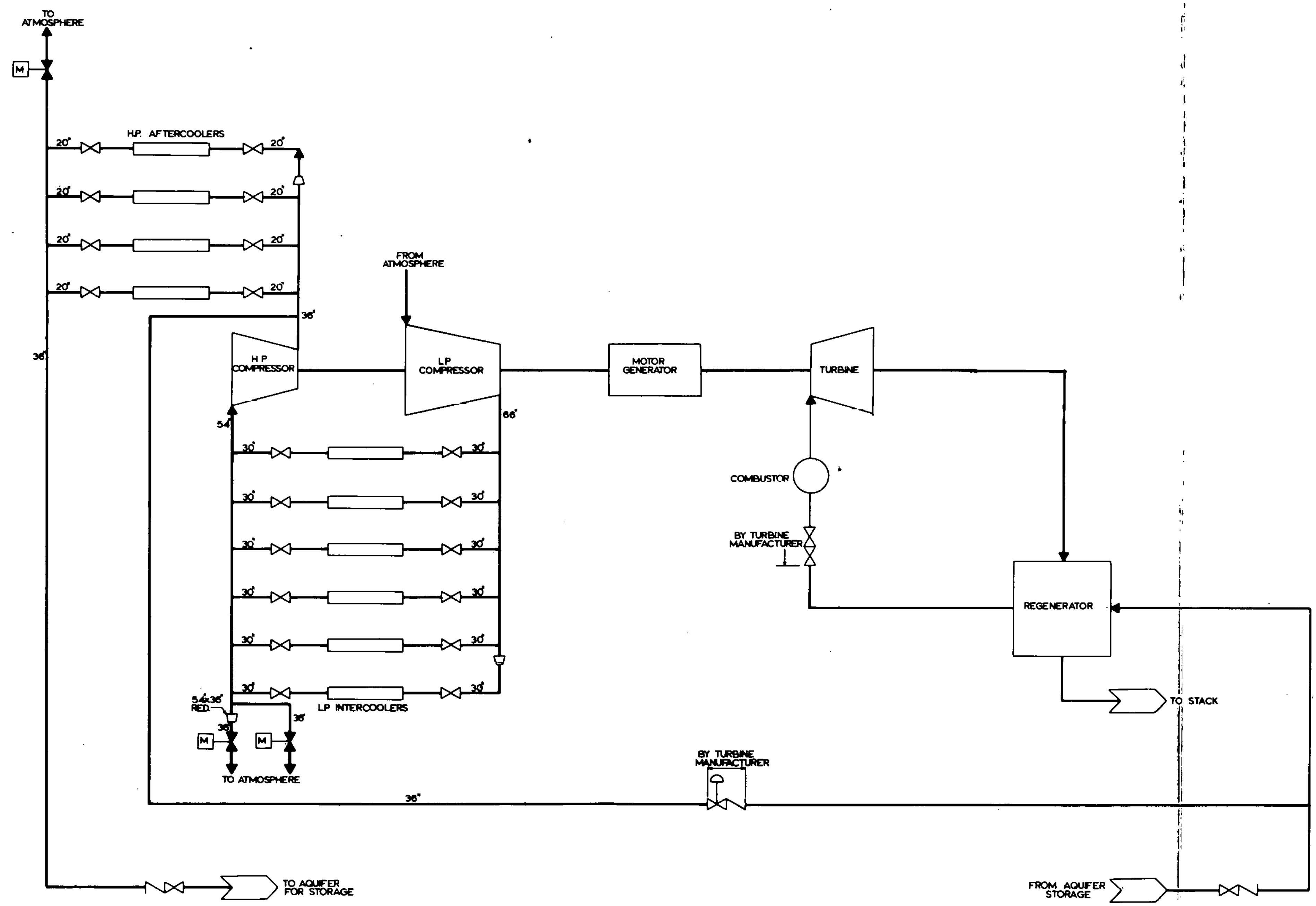


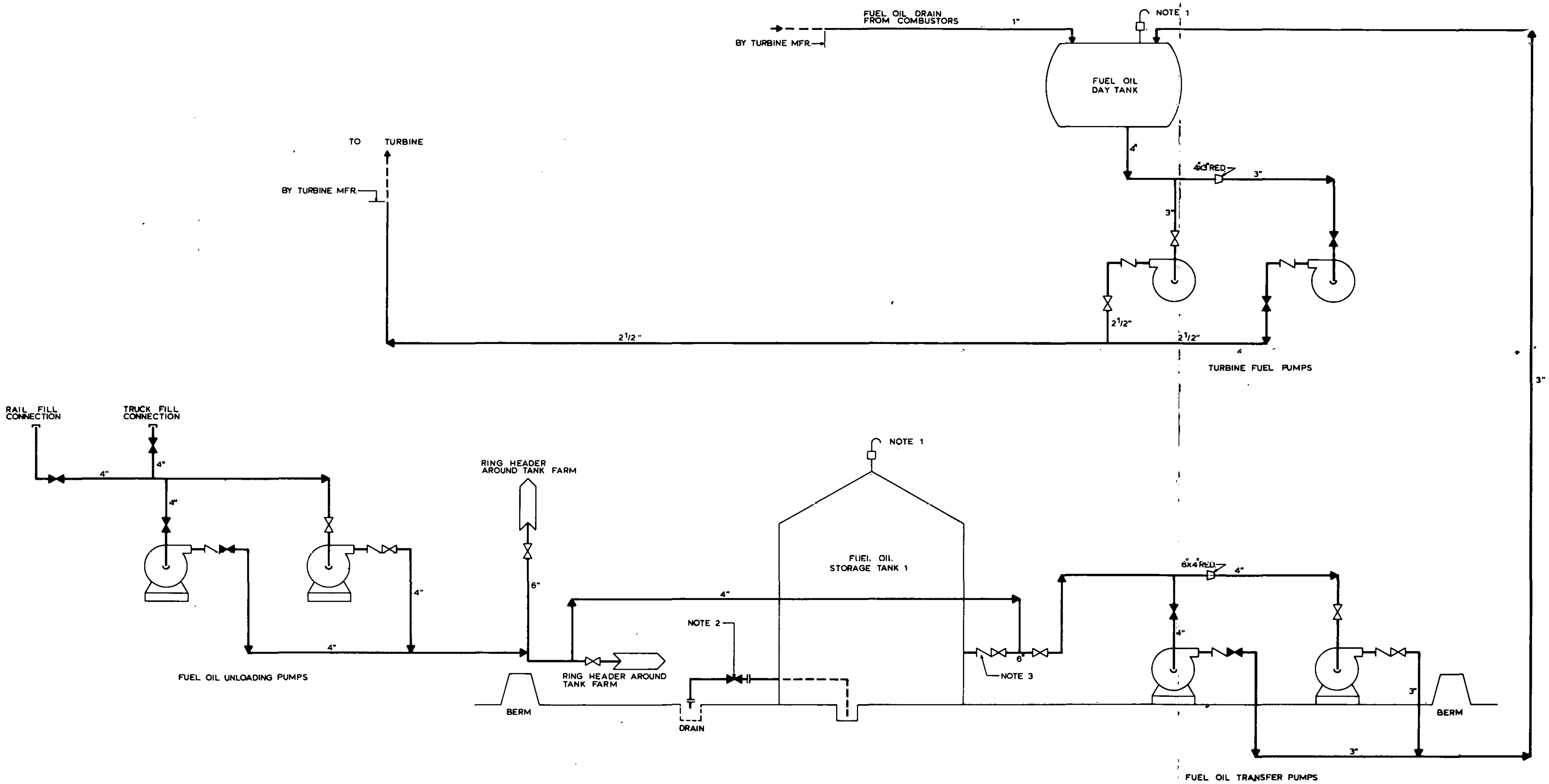
FIGURE 5-21

DIAGRAM OF MAIN AIR PIPING

200 PSI SYSTEM

SARGENT & LUNDY ENGINEERS

PUBLIC SERVICE INDIANA
CAES IN AQUIFER
DOE NO. ET-78-C-01-2159



NOTES:

1. VAREC FIG. 27 & 50E WEATHER HOOD & FLAME ARRESTOR
2. VAREC FIG. 5000 WATER DRAIN OFF VALVE WITH FIG. 307 MOUNTING FLANGE BY P.C.
3. VAREC FIG. 5070 INTERNAL SAFETY VALVE.

FIGURE 5-22
DIAGRAM OF FUEL OIL PIPING

200 PSI SYSTEM

SARGENT & LUNDY ENGINEERS

PUBLIC SERVICE INDIANA
CAES IN AQUIFER
DOE NO. ET-78-C-01-2159

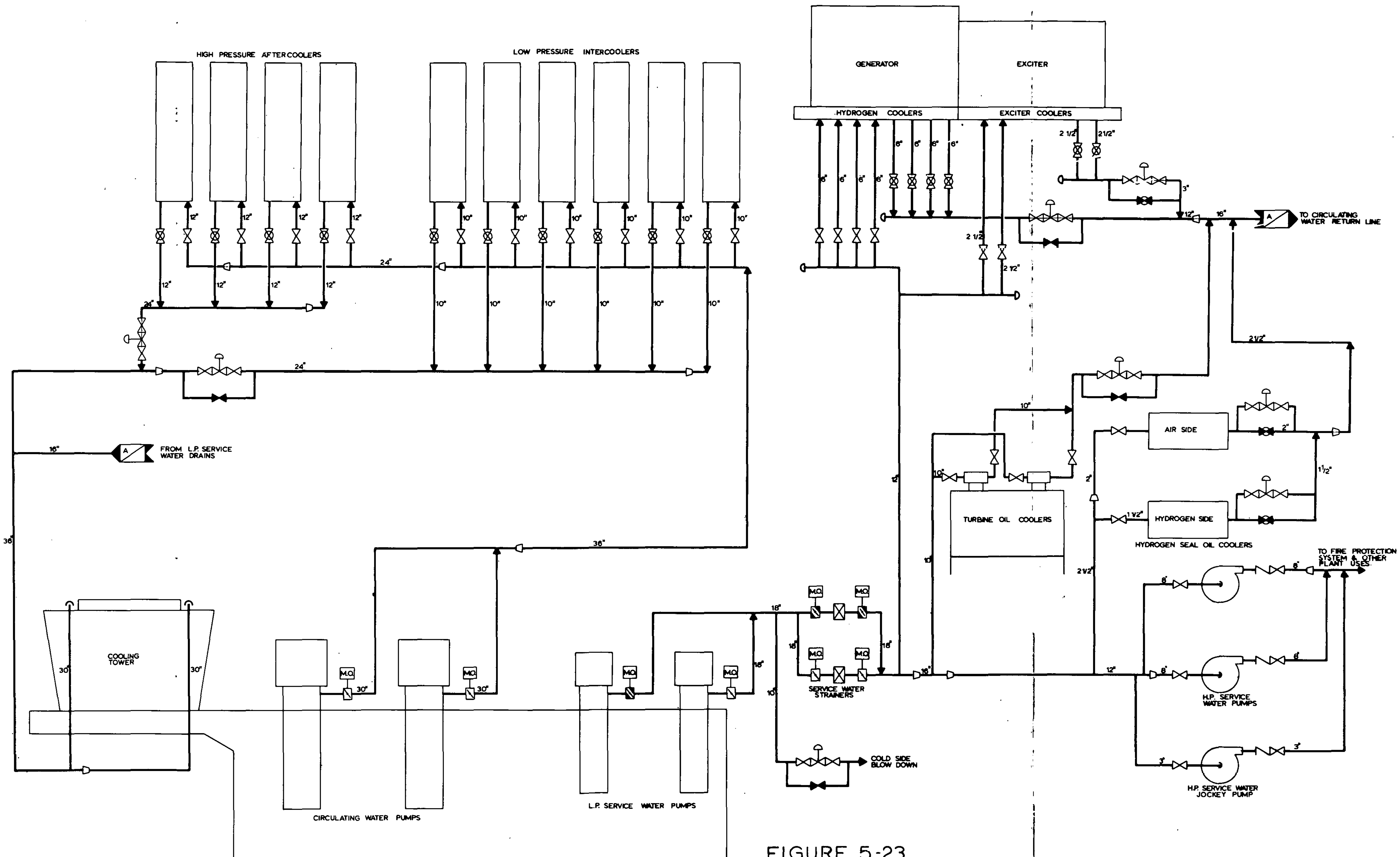


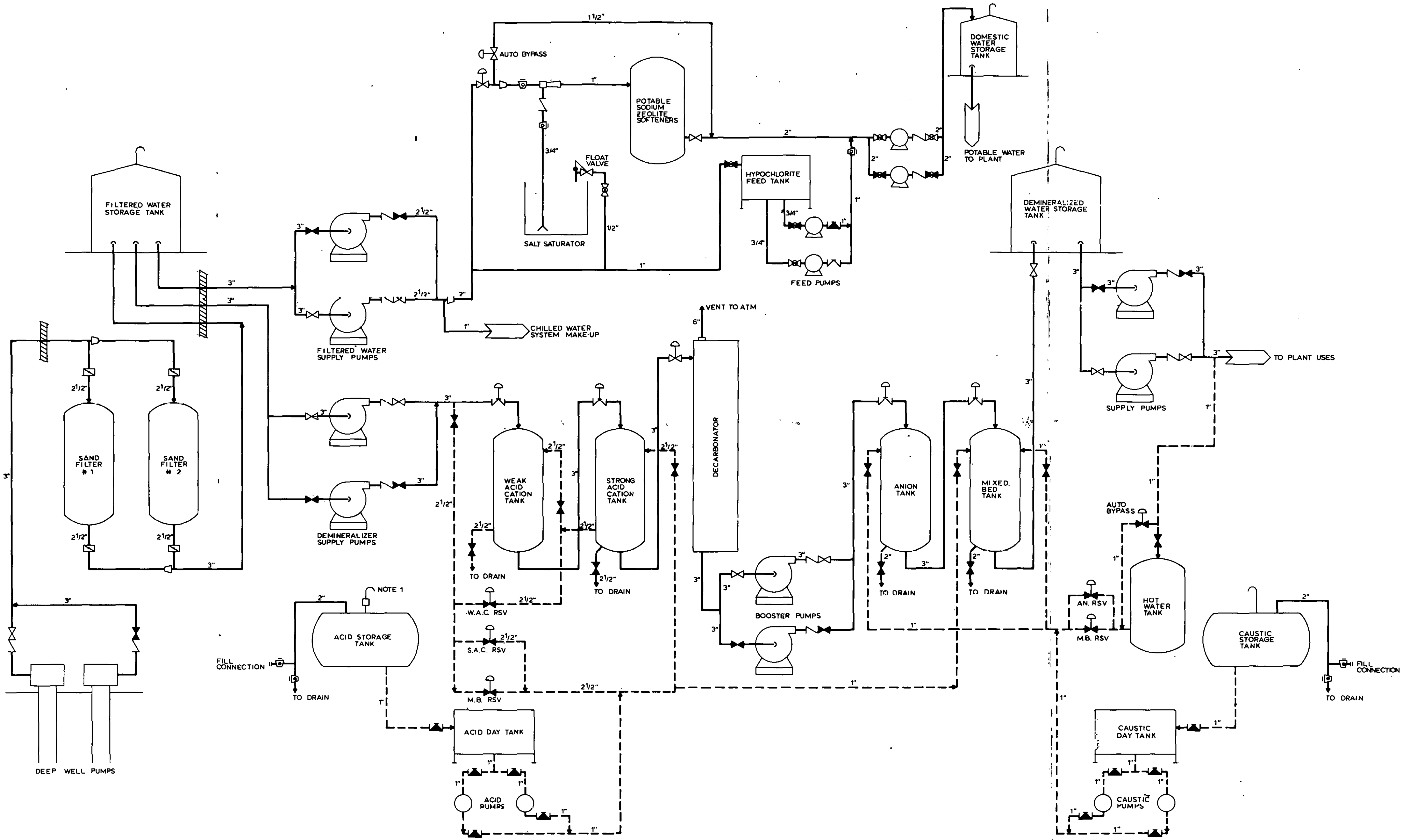
FIGURE 5-23

DIAGRAM OF CIRCULATING, HIGH PRESSURE & LOW PRESSURE SERVICE WATER

200 PSI SYSTEM

SARGENT & LUNDY
ENGINEERS

PUBLIC SERVICE INDIANA
CAES IN AQUIFER
DOE NO. ET-78-C-01-2159



NOTES:
1. DRY BREATHER WITH REVERSE BEND
2. DASHED LINE DENOTES REGENERATION MODE
3. RSV = RATE SET VALVE

FIGURE 5-24
DIAGRAM OF TREATED, POTABLE & DEMINERALIZED WATER PIPING
200 PSI SYSTEM

SARGENT & LUNDY ENGINEERS

PUBLIC SERVICE INDIANA
CAES IN AQUIFER
DOE NO. ET-78-C-01-2159

5.7 PREPARATION OF ELECTRICAL SINGLE LINE DIAGRAMS

A set of preliminary electrical single line diagrams have been developed as part of the Task 1 work effort. These diagrams are based on estimated auxiliary loads developed by Westinghouse and Sargent & Lundy; on an assumed switchyard consisting of 1-138 kV line for Unit 1 and the addition of 2-345 kV lines as Units, 2, 3 and 4 are constructed; and on direction from the Sponsoring Utilities after a review of a number of single line sketches.

Figure 5-25 shows the basic bus arrangement for a four unit CAES plant. Since the prime purpose of this installation is peaking duty, a single 4 kV bus per unit has been assumed. In order to provide back-up for each bus, provisions for electrically tieing all units together with the 4 kV reserve bus have been included.

Figure 5-26 shows the basic station single line for Unit 1. Units 2, 3 and 4 would esentially follow the same design and therefore single lines for these units have not been included.

Since the switchyard has been assumed to have two voltage levels, Figure 5-27 and Figure 5-28 showing the 138 kV yard and the 345 kV yard respectively have been developed. References on Figure 5-28 showing connections to Units 2, 3 and 4 will be cross referenced on the respective station single line drawings.

It should be noted that the single line design has progressed as far as possible based on the information available and reasonable assumptions in many areas where information has not been developed or is specifically site related. Details of the design will be reviewed and finalized as information becomes available as a result of decisions and design progress in subsequent Task work.

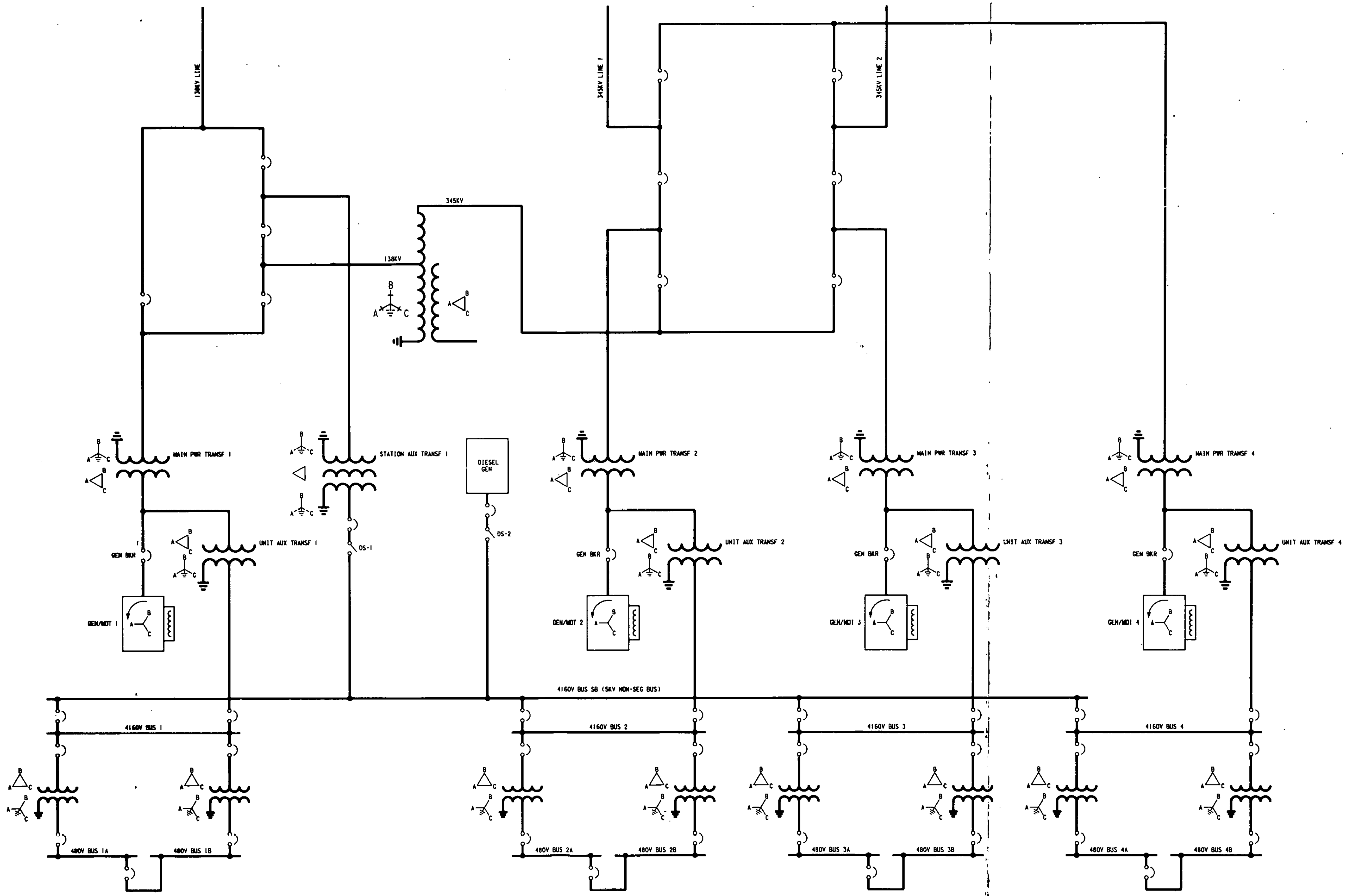
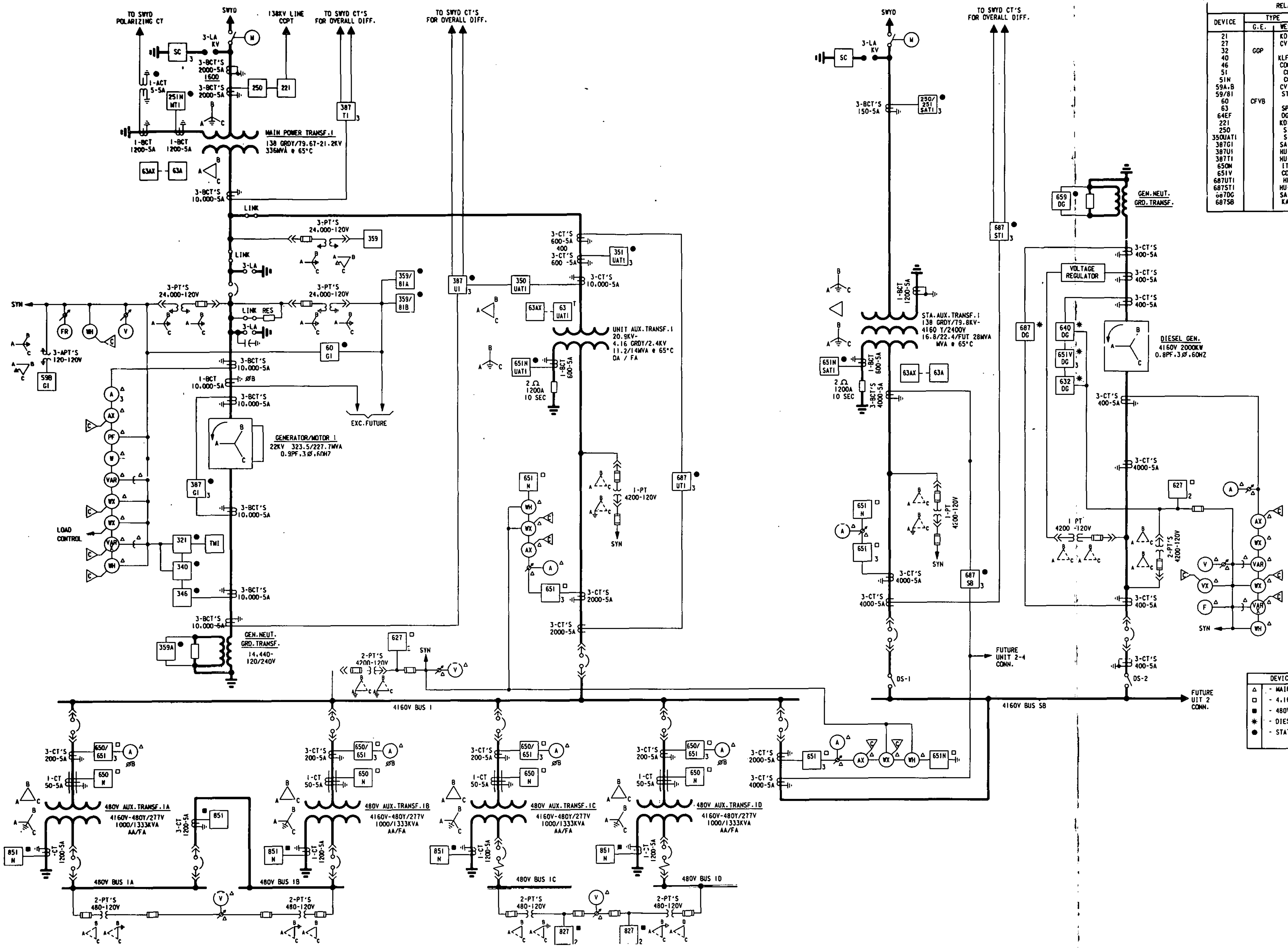


FIGURE 5-25
PRIMARY BUS SINGLE LINE
UNITS 1-4

SARGENT & LUNDY
ENGINEERS

PUBLIC SERVICE INDIANA
CAES IN AQUIFER
DOE NO. ET-78-C-01-2159



RELAY SCHEDULE & DEVICE NUMBER

DEVICE	TYPE G.E.	TYPE WEST.	FUNCTION
21		KD-11	DISTANCE
27		CV-2	UNDERVOLTAGE RELAY
32	GGP		REVERSE POWER RELAY
40		KLF-1	LOSS OF FIELD RELAY
46		COQ	NEGATIVE PHASE SEQUENCE RELAY
51		CO	TIME OVERCURRENT RELAY
51N		CO	NEUTRAL TIME OVERCURRENT RELAY
59A,B		CV-8	OVERVOLTAGE RELAY
59/81		STV	VOLTS / HERTZ
60	CFVB		VOLTAGE BALANCE RELAY
63		SPR	SUDDEN PRESSURE RELAY
64EF		DGF	FILED GROUND DETECTOR RELAY
221		KD10	LOSS OF SYNCHRONISM
250		SI	LOSS OF SYNCHRONISM DETECTOR
350UAT1		SI	INSTANTANEOUS OVERCURRENT RELAY
387G1		SA-1	GENERATOR DIFFERENTIAL RELAY
387U1		HU-4	GENERATOR - MPT DIFFERENTIAL RELAY
387T1		HU-4	TRANSFORMER DIFFERENTIAL RELAY
650N		ITH	INST. OVERCURRENT RELAY
651V		COV	VOLTAGE RESTRAINED OVERCURRENT RELAY
687UT1		HU	TRANSFORMER DIFFERENTIAL RELAY
687ST1		HU-4	TRANSFORMER DIFFERENTIAL RELAY
687DG		SA-1	TRANSFORMER DIFFERENTIAL RELAY
687SB		KAB	[illegible] DIFFERENTIAL RELAY

LEGEND

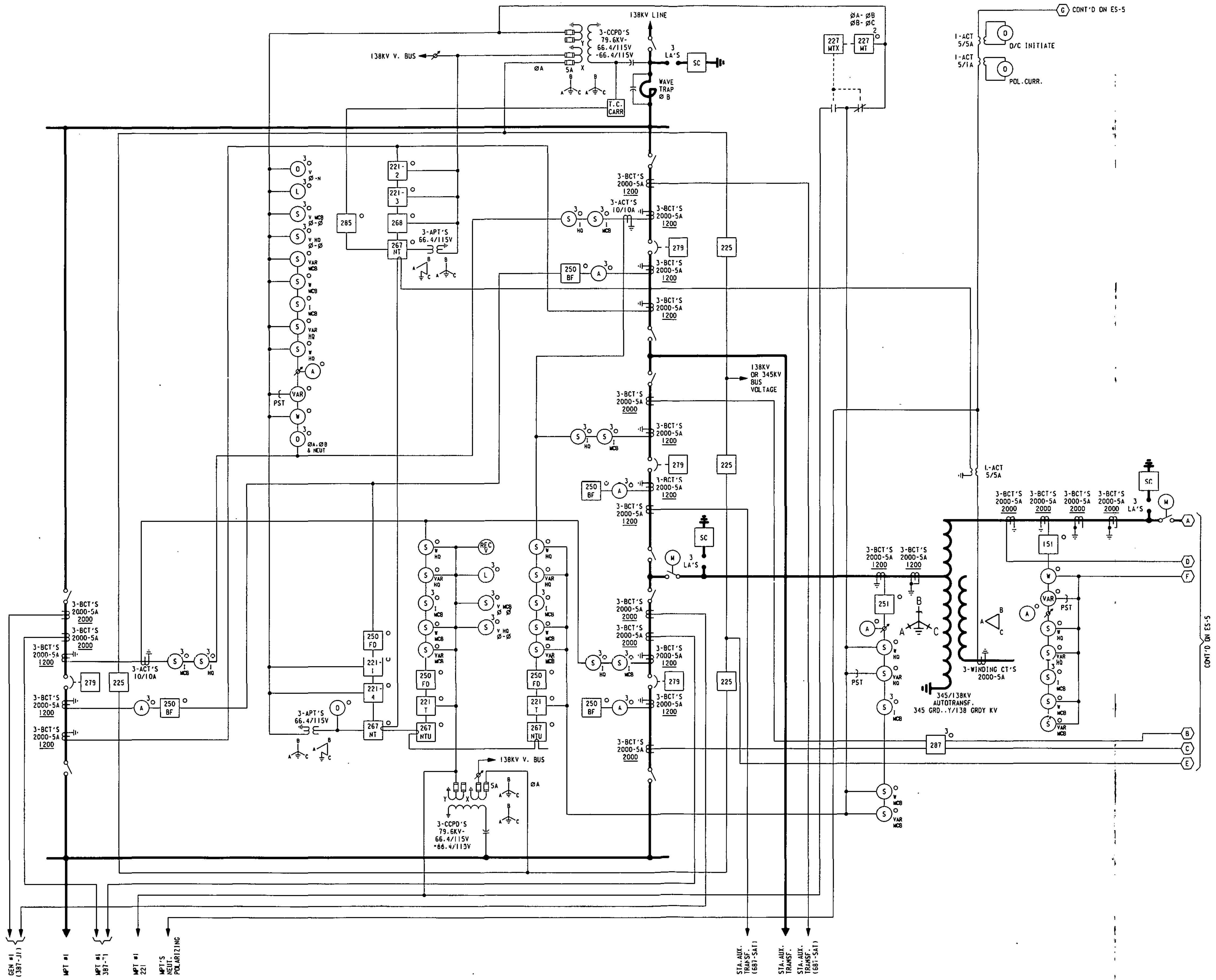
SYMBOL	DESCRIPTION
A	AMMETER
PF	POWER FACTOR METER
V	VOLTMETER
W	WATTMETER
WH	WATTHOUR METER
VAR	VARMETER
VH	VARHOUR METER
AX	CURRENT TRANSDUCER
VAR	VAR TRANSDUCER
VX	VOLTAGE TRANSDUCER
WX	WATT TRANSDUCER
FX	FREQUENCY TRANSDUCER
FR	FREQUENCY RECORDER
	COMPUTER POINT
	WINDOW TYPE 3Ø C.T.
	INSTRUMENT OR CONTROL SWITCH
	PHASE SHIFTING TRANSFORMER

DEVICE LOCATION		DEVICE NO. PREFIX
△	- MAIN CONTROL BOARD	1 - 345,000V
□	- 4,160V SWGR	2 - 138,000V
■	- 480V SWGR	3 - 24,000V
✱	- DIESEL GEN.	4 - 13,800V
●	- STATION RELAY PNL.	5 - 6,900V
		6 - 4160V
		7 - 2400V
		8 - 480V

SARGENT & LUNDY ENGINEERS

FIGURE 5-26
SINGLE LINE DIAGRAM
UNIT 1

PUBLIC SERVICE INDIANA
CAES IN AQUIFER
DOE NO. ET-78-C-01-2159



DEVICE	TYPE G.E.	TYPE WEST.	FUNCTION
21P			CARRIER PHASE FAULT RELAY
21NP			GRD.DIRECTIONAL OVERCURRENT RELAY
21-1			ZONE 1 PHASE FAULT RELAY
21-2			PILOT TRIP PHASE
21-3			CARRIER START
21-4			ZONE 2 PHASE FAULT RELAY
21T			IMPEDANCE RELAY TRANSF.BACK-UP
25			SYNCHRONIZING RELAY
25X			SYNCHRONIZING AUXILIARY RELAY
27MT			UNDERVOLTAGE RELAY
27MTX			AUXILIARY RELAY FOR UNDERVOLTAGE
50BF			BREAKER FAILURE RELAY
50FD			FAULT DETECTOR OVERCURRENT RELAY
51			OVERCURRENT RELAY
67N			DUAL POL.GRD.PILOT
67NT			DUAL POL.GRD.BACK-UP
67NTU			DIRECTIONAL O/C TRANSF.BACK-UP
68			OUT OF STEP BLOCK RELAY
79			RECLOSING RELAY
85			CARRIER AUXILIARY RELAY
87			TRANSFORMER DIFFERENTIAL RELAY
87B			BUS DIFFERENTIAL RELAY

LEGEND

SYMBOL	DESCRIPTION
A	AMMETER
V	VOLTMETER
W	WATTMETER
VAR	VARMETER
REC V	REC.VOLTMETER
L	INDICATING LIGHT
S I	SUPV.-AMPS
S V	SUPV.-VOLT
S W	SUPV.-WATT
S VAR	SUPV.-VAR
O	OSCILLOGRAPH
	INSTRUMENT OR CONTROL SW.
	PHASE SHIFTING TRANSFORMER
	MANUAL OPER.DISC.
M	MOTOR OPER.DISC.
HQ	HEADQUARTER
MCB	MAIN CONTROL BOARD AT STA.
APT	AUX.POT.TRANSF.
ACT	AUX.CURR.TRANSF.
SC	SURGE COUNTER

NO.	NOTES
1.	DEVICE PREFIX NOTES: PREFIX "1" APPLIES DEVICE NUMBER TO 345KV BUS OR LINE PREFIX "2" APPLIES DEVICE NUMBER TO 138KV BUS OR LINE PREFIX "3" APPLIES DEVICE NUMBER TO 24KV BUS OR LINE

DEVICE LOCATION:

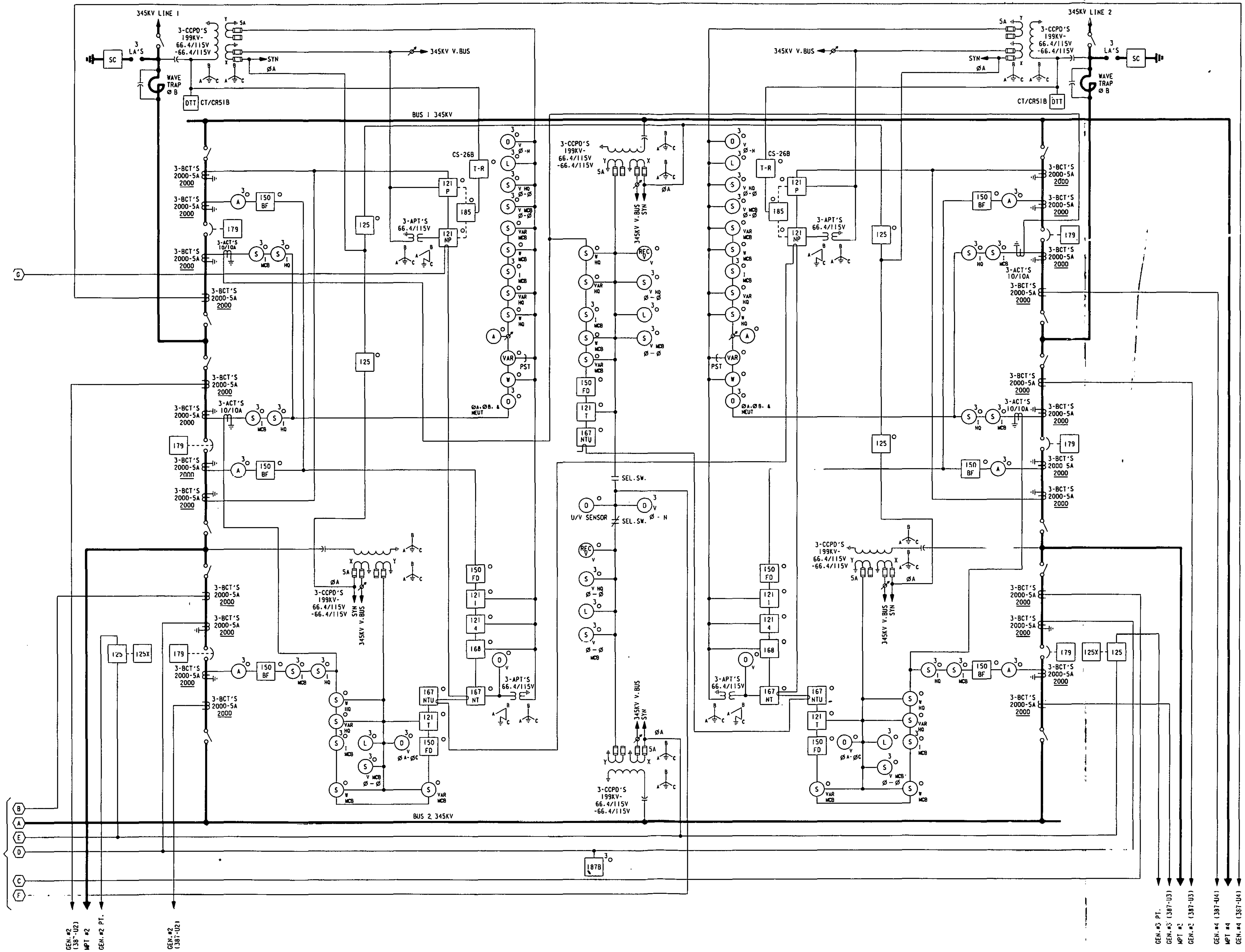
□° OR ○° AT SWITCHYARD RLY.HOUSE

NO.	GENERAL NOTES
1.	SYNCHRONIZIG RELAY APPLICATIONAS SHOWN FOR THE 345KV AND 138KV. SYSTEM ARE REPRESENTATIVE ONLY AND REFLECT USE OF REGULAR SYNCHRONIZING RELAYS TYPE CVE OR EQUIVALENT AND SPECIAL GXS TYPE RELAYS
2.	FINAL DETERMINATION OF THE C.T.TAPS THAT MAY BE REQUIRED WILL BE MADE LATER.
3.	LOCAL SYNCHRONIZING PANEL CONNECTION AT SWYD. RELAY HOUSE WILL BE SHOWN LATER.IF REQUIRED.
4.	345KV BUS 1 AND BUS 2 SELECTED VOLTAGE MONITORING AT THE MCB WILL BE DETAILED LATER

SARGENT & LUNDY ENGINEERS

FIGURE 5-27
SINGLE LINE DIAGRAM
345/138kV SWITCHYARD

PUBLIC SERVICE INDIANA
CAES IN AQUIFER
DOE NO. ET-78-C-01-2159



RELAY SCHEDULE & DEVICE NUMBER

DEVICE	TYPE C.E.	TYPE WEST.	FUNCTION
21P			CARRIER PHASE FAULT RLY
21NP			GRD.DIRECTIONAL OVERCURRENT RLY
21-1			ZONE 1 PHASE FAULT RLY
21-2			PILOT TRIP PHASE
21-3			CARRIER START
21-4			ZONE 2 PHASE FAULT RLY
21T			IMPEDANCE RLY TRANSF.BACK-UP
25			SYNCHRONIZING RLY
25X			SYNCHRONIZING AUXILIARY RLY
27MT			UNDERVOLTAGE RLY
27MTX			AUXILIARY RLY FOR UNDERVOLTAGE
50BF			BREAKER FAILURE RLY
50FD			FAULT DETECTOR OVERCURRENT RLY
51			OVERCURRENT RLY
67N			DUAL POL.GRD.PILOT
67NT			DUAL POL.GRD.BACK-UP
67NTU			DIRECTIONAL O.C. TRANSF.BACK-UP
68			OUT OF STEP BLOCK RLY
79			RECLOSING RLY
85			CARRIER AUXILIARY RLY
87			TRANSFORMER DIFFERENTIAL RLY
87B			BUS DIFFERENTIAL RLY

LEGEND

SYMBOL	DESCRIPTION
(A)	AMMETER
(V)	VOLTMETER
(W)	WATTMETER
(VAR)	VARMETER
(REC V)	REC.VOLTMETER
(L)	INDICATING LIGHT
(S)I	SUPV.-AMPS
(S)V	SUPV.-VOLT
(S)W	SUPV.-WATT
(S)VAR	SUPV.-VAR
(O)	OSCILLOGRAPH
Ø	INSTRUMENT OR CONTROL SW.
	PHASE SHIFTING TRANSFORMER
	MANUAL OPER.DISC.
(M)	MOTOR OPER.DISC.
HQ	HEADQUARTER
MCB	MAIN CONTROL BOARD AT STA.
APT	AUX.POT.TRANSF.
ACT	AUX.CURR.TRANSF.
SC	SURGE COUNTER

NO.	NOTES
1.	DEVICE PREFIX NOTES: PREFIX "1" APPLIES DEV. NO. TO 345KV BUS OR LINE PREFIX "2" APPLIES DEV. NO. TO 138KV BUS OR LINE PREFIX "3" APPLIES DEV. NO. TO 24KV BUS OR LINE

DEVICE LOCATION:

☐ OR ○ AT SWITCHYARD RLY.HOUSE

FIGURE 5-28
SINGLE LINE DIAGRAM
345 kV SWITCHYARD

SARGENT & LUNDY ENGINEERS

PUBLIC SERVICE INDIANA
CAES IN AQUIFER
DOE NO. ET-78-C-01-2159

Section 6

CONCLUSION

CAES system design in Task 1 has progressed from the establishment of basic design assumptions to selection of heat cycle and machinery configurations for three typical aquifer storage pressures. The Task 1 work effort culminated in the preparation of general arrangement drawings, piping schematics, and electrical single line diagrams for the three typical CAES systems. Task 3 will optimize the preliminary plant designs developed by Sargent & Lundy and Westinghouse under Task 1 for plant sites selected in Task 2.

REFERENCES

1. D. L. Katz and E. R. Lady. Compressed Air Storage. Ann Arbor, Michigan, 1976.

2. Economic and Technical Feasibility Study of Compressed Air Storage. General Electric Company, 1976.

3. Feasibility of Compressed Air Energy Storage As A Peak Shaving Technique in California, Volume II. Consultant Report prepared by Acres American Incorporated, 1978.

2

COMPRESSED AIR ENERGY STORAGE PRELIMINARY DESIGN AND SITE DEVELOPMENT PROGRAM IN AN AQUIFER
DOE Contract No. ET-78-C-01-2159

Final Draft Report
TASK 1 — VOLUME 2
Establish Facility Design Criteria and Utility Benefits
October, 1980

Prepared by
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LEGAL NOTICE

ABSTRACT

This study develops combustion turbine heat cycles and machinery configurations for use with aquifer air storage systems by an electric utility. Air is compressed in these systems, by utilizing "off-peak" electric power available from base load plants, and stored underground in an aquifer. During subsequent periods, when intermediate or peaking combustion turbines would normally be employed, the stored air is extracted from the aquifer and serves as an air supply for fired combustion turbine generating units.

Heat cycles are optimized, for nominal storage pressure levels of 200, 600 and 1000 psi, on the basis of minimum power production energy cost. The use of standardized (common) machinery for the low and intermediate pressure components in all three pressure level systems was investigated.

Variations of intercooled compression cycles and regenerative reheat cycles were the basis for the candidate cycles selected for evaluation. Thermal energy storage was not a consideration in this study. Heat was rejected from the intercoolers and the aftercooler in the compression cycle to cooling towers in the balance of plant systems. An aquifer air injection temperature of 150°F was selected for this study. The use of a combustion turbine and steam combined cycle was investigated and it was found that the intercooled, regenerative reheat cycle had a lower power production energy cost than the combined cycle in a CAES application where the stored air temperature was 200°F or less.

The performance of the CAES compressor and turbine machinery, when operating as a generating unit independent of the aquifer system, was calculated and determined to be competitive with conventional combustion turbine generating units at the higher system pressure levels.

ACKNOWLEDGMENTS

Westinghouse Electric Corporation acknowledges the contributions made to this volume by the following persons (listed alphabetically):

- Mr. J.F. Boron analyzed the feasibility of the large gears.
- Mr. W.J. Boyd was responsible for contract administration.
- Mr. T.A. Desimone assisted in obtaining information from vendors required to estimate the equipment cost.
- Mr. K.H. Eagle assisted in projecting the CAES power plant availability and reliability maturing rate.
- Mr. D.M. Handkins provided invaluable assistance in retrieving availability and reliability records.
- Mr. M.M. Hobbs was responsible for the work performed for the control system philosophy.
- Mr. P.C. Holden provided the required turbine cooling flow rates.
- Mr. G.S. Howard provided expander performance and sizing information.
- Mr. N.S. Kosanovich (W East Pittsburgh) was responsible for the electrical rotating machinery related mechanical analyses.
- Mr. J.R. Pipkin (W East Pittsburgh) was responsible for the selection of the required motor/generator frame sizes.
- Mr. J.F. Polhemus projected the manufacturing cost of the turbomachinery.
- Mr. R. Pryor calculated the required sizes of the valves.
- Mr. C.A. Rohr provided compressor performance and sizing information.
- Mr. J.J. Shields was responsible for providing information for the required turbomachinery piping systems.
- Mr. J. Warlow estimated the operating and maintenance costs for the CAES turbomachinery.
- Mr. W.A. Wright assisted in projecting the CAES power plant availability and reliability maturing rate.

Westinghouse Electric Corporation is also grateful to the following companies for supplying initial sizing and costing information:

- Coal Tech Inc.
- Dressor-Clark Compressor Division
- McQuay-Perfex Inc.
- MRM, Ecodyne Heat Exchanger Division
- Philadelphia Gear Corporation
- SSS Clutch Corporation
- Yuba Heat Transfer Corporation

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SUMMARY

Compressed Air Energy Storage (CAES) is a technique being investigated by the electric utility industry, for storing energy from base load power plants during "off-peak" hours, for use during periods of high power demand.

This report describes the work performed in identifying and evaluating candidate compressor and combustion turbine systems for use in a CAES application utilizing aquifer air storage systems. Three system pressure levels were analyzed in this study as the basis for developing optimized heat cycles and machinery configurations.

HEAT CYCLE SELECTION

Compressor trains utilizing one and two intercoolers were optimized for minimum work. The primary considerations in the optimization of the compression cycles were the location and number of intercoolers, the resulting compressor pressure ratios and predicted compressor efficiencies.

Reheat, regenerative turbine cycles were optimized on the basis of the reheat combustor location, turbine firing temperatures and predicted turbine efficiencies.

The optimum compressor trains utilizing one and two intercoolers were combined with optimized turbine cycles and the performance of the overall CAES power plants were evaluated at various turbine firing temperatures and energy cost ratios. Energy cost ratios consist of the ratios of the projected cost of "off-peak" electric power used to drive the compressor train and the projected cost of the fuel oil consumed by the turbine train combustors. The initially selected optimized CAES cycles are summarized in the following table.

Table S-1

SELECTED OPTIMIZED CYCLE SUMMARY
TWO INTERCOOLER CYCLE
REGENERATOR EFFECTIVITY 85%

		Nominal Storage Pressure	
	200 PSI	600 PSI	1000 PSI
Firing Temperature HPT/LPT °F	1500/UF	1500/1500	1500/1500
Turbine Work – KW/LB/SEC*	218.5	331.1	375.7
Air Flow – LB/SEC/Machinery Set**	770.0	770.0	770.0
KW Per Machinery Set*	168,250	254,950	289,300
Machinery Sets Per 1000 MWe (EXACT)	5.94	3.92	3.46
Machinery Sets Per 1000 MWe (ACTUAL)	6	4	3
Total Power Production Energy Costs in Mils/KW HR			
Fuel @ $/10^6 BTU/ELEC PWR in Mils/KW HR 2.50/10	17.80	16.84	16.80
2.50/20	25.40	23.88	23.90
5.00/20	35.59	33.67	33.60
7.50/10	38.17	36.41	36.20
7.50/20	45.78	43.46	43.30
7.75/60	77.23	72.63	72.67

*These Outputs Reflect a 98% Generator Efficiency and Turbine Auxiliary load losses.
**Low Pressure Compressor Inlet Air Flow.

Turbine Output Power values are as calculated and do not include any margin.

COMPONENT STANDARDIZATION

Standardization of hardware is very effective in reducing development, design, manufacturing and inventory costs. A study was performed to determine the performance penalty which might be incurred by standardizing as many components as possible in the low and intermediate pressure range of all three of the nominal pressure level systems. Common, or standardized, components in the 200, 600 and 1000 PSI nominal pressure systems consist of:

- Low pressure compressor
- Low pressure intercooler
- Intermediate pressure compressor
- Intermediate pressure intercooler
- Low pressure turbine and combustor.

A comparison of the performance and total energy cost of cycles utilizing optimized and standardized components is shown on the following table.

Table S-2

OPTIMIZED VS STANDARDIZED
LOW PRESSURE MACHINERY COMPARISON

	200 PSI		600 PSI		1000 PSI	
	OPT	STD	OPT	STD	OPT	STD
LP COMPRESSOR, ρ	3.380	5.090	4.770	5.090	5.470	5.090
IP COMPRESSOR, ρ	2.400	–	3.390	3.765	4.050	3.765
HP COMPRESSOR, ρ	2.400	3.685	3.390	2.861	4.050	4.682
LP TURBINE, ρ	10.700	10.700	11.000	11.000	11.000	11.000
CHARGING POWER, KW/LB/SEC*	166.2	171.5	233.4	233.4	266.7	267.5
NET OUTPUT PER MACHINERY SET, KW/LB/SEC.	218.5	219.0	331.1	330.9	375.7	375.7
NET LHV HEAT RATE, BTU/KWHR*	4075	4075	3915	3915	3880	3880
TOTAL POWER PRODUCTION ENERGY COST IN MILS/KWHR						
FUEL @ $/10^6 BTU / ELEC. PWR. IN MILS/KWHR						
$2.50 / 10	17.80	18.02	16.84	16.84	16.80	16.82
$2.50 / 20	25.40	25.85	23.88	23.89	23.90	23.94
$5.00 / 20	35.59	36.04	33.67	33.68	33.60	33.64
$7.50 / 10	38.17	38.39	36.41	36.41	36.20	36.22
$7.50 / 20	45.78	46.22	43.46	43.47	43.30	43.34
$7.75 / 60	77.23	78.56	72.63	72.66	72.67	72.79

(W 1591)

CROSSHATCHED BLOCKS IDENTIFY COMMON COMPONENTS.
*REFLECTS MOTOR, GENERATOR AND AUXILIARY LOAD LOSSES.

The charging power, turbine output and heat rate values are as calculated and do not include any margin.

In the 200 PSI standardized system, a single intercooler compressor train is compared with the optimized two intercooler train, thus eliminating the intermediate compressor in that system.

This comparison shows that the use of standardized components does not impose a significant performance or total energy cost penalty. The use of the standardized components was selected as the preferred CAES heat cycle for all three pressure levels.

The final estimate for the CAES system performance is presented in Table S-3. These results differ from the above standardized results because the efficiencies for the motors, generators, gearboxes and combustors were refined.

Table S-3

CAES SYSTEM PERFORMANCE
STANDARDIZED COMPONENTS
(REFINED COMPONENT EFFICIENCIES)

	NOMINAL SYSTEM PRESSURE		
	200 PSI	600 PSI	1000 PSI
TOTAL COMPRESSOR INPUT AT MOTOR TERMINALS, KW	130,653	178,540	204,910
TOTAL TURBINE OUTPUT AT GENERATOR TERMINALS, KW	178,505	256,555	291,150
AUXILIARY LOAD, KW	500	500	500
NET TURBINE OUTPUT, KW	178,005	256,055	290,650
NET LHV HEAT RATE BTU/KWHR	3,945	3,855	3,820
FUEL @ $\$10^6$ BTU / ELEC. PWR. IN MILS/KWHR	TOTAL POWER PRODUCTION ENERGY COST IN MILS/KWHR		
2.50 / 10	17.20	16.61	16.60
2.50 / 20	24.54	23.58	23.65
5.00 / 20	34.40	33.22	33.20
7.50 / 10	36.93	35.89	35.70
7.50 / 20	44.27	42.86	42.75
7.75 / 60	74.61	71.71	71.91

REFINED COMPONENT EFFICIENCIES (W 1592)

MOTORS	96%
GENERATORS	98.5%
GEARBOXES	98.5%
COMBUSTION	99%

Work of compression, output power and heat rate values are as calculated and do not include any margin.

MACHINERY CONFIGURATION

Seventeen candidate machinery configurations, based upon variations of one, two and three shaft arrangements, were evaluated. All three arrangements were technically feasible. On the basis of the relative capital cost of the major equipment and overall system complexity, the single shaft arrangement was selected as the preferred configuration. A preliminary outline drawing of the rotating equipment in the 1000 PSI system machinery set is shown.

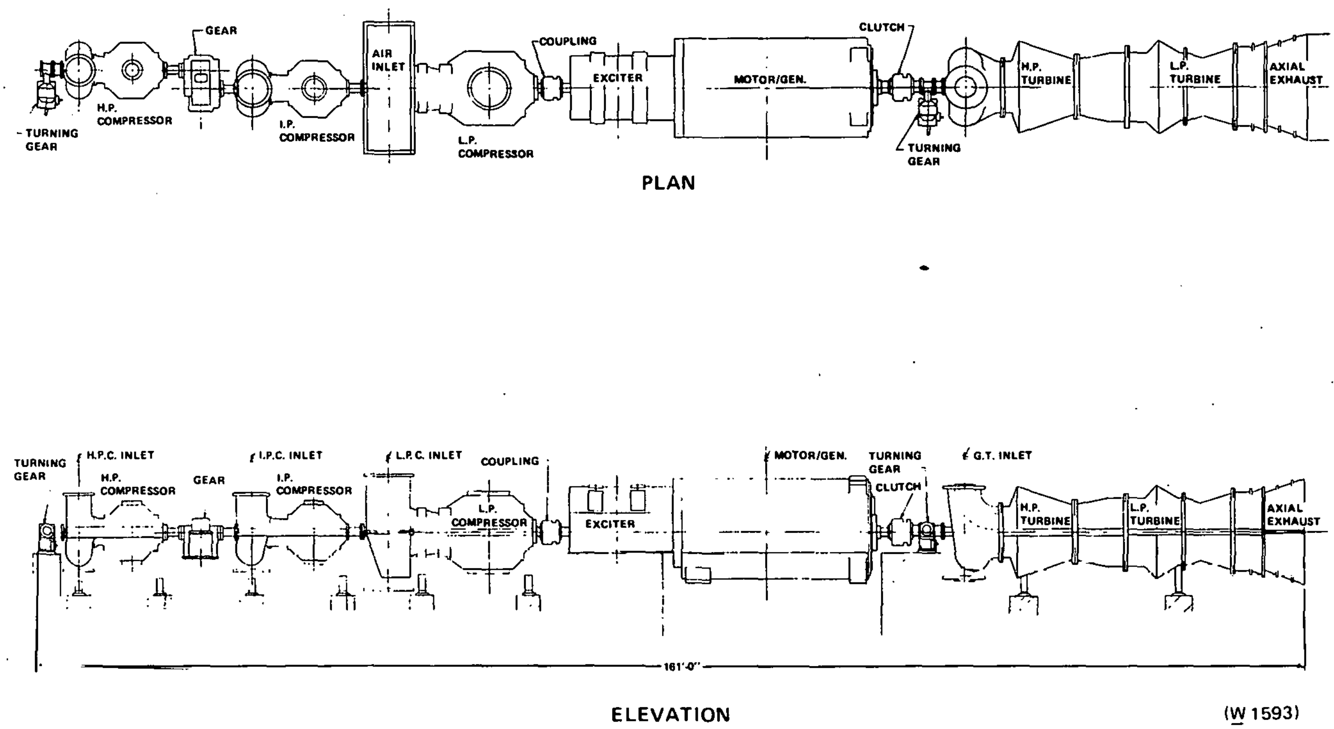
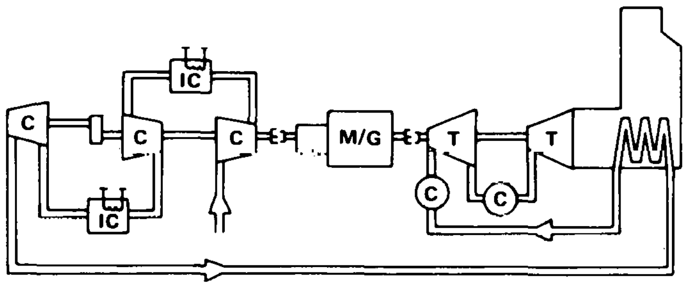


Figure S-1. 1000 PSI System Machinery Outline

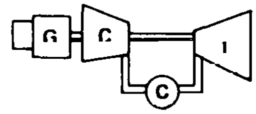
PART LOAD AND AQUIFER INDEPENDENT OPERATION

Flow and discharge pressure characteristic curves for the compressor trains were prepared as an input to the preliminary evaluation of the candidate aquifer sites.

The operation of the selected CAES cycles as aquifer independent generators was analyzed and these results are shown on the following table.



CAES AQUIFER INDEPENDENT GENERATOR



CONVENTIONAL COMBUSTION TURBINE

	CAES SYSTEM AQUIFER INDEPENDENT OPERATION			CONVENTIONAL COMBUSTION TURBINE W501 OPERATION
	200 PSI 1500/UF °F	600 PSI 1500/1500°F	1000 PSI 1500/1500°F	APPROXIMATELY 2000°F
NET TURBINE OUTPUT, KW	69,070	88,436	98,535	94,715
NET LHV HEAT RATE BTU/KWHR	11,465	10,940	10,925	10,660

(W 1594)

NOTES:

1. THE CAES SYSTEM AQUIFER INDEPENDENT PLANT PERFORMANCE INCLUDES MARGINS TO ALLOW COMPARISON WITH TYPICAL PUBLISHED PERFORMANCE DATA FOR A W501 COMBUSTION TURBINE.
2. AN ALLOWANCE HAS BEEN MADE IN THE CAES AQUIFER INDEPENDENT PLANT PERFORMANCE (POWER OUTPUT AND HEAT RATE) FOR THE COOLING TOWER FANS, LOW PRESSURE SERVICE WATER AND WATER PUMPING ELECTRICAL LOADS.
3. RATINGS ARE AT BASE LOAD OPERATION, 14.43 PSIA AND 59°F.

CONTROL SYSTEM PHILOSOPHY

A control system description was prepared together with a main air control schematic. Three modes of operation of the CAES plant are outlined.

- Aquifer Dependent Power Production
- Aquifer Charging
- Aquifer Independent Power Production

Control areas which may require special attention during subsequent phases of the system design, when detailed characteristics of the mechanical and electrical rotating equipment are defined, were identified.

CAES PRICE ESTIMATES

Equipment price estimates were prepared in Task 1 to support the aquifer site evaluation performed in Task 2, "Characterize and Explore Potential Sites and Prepare Site Research and Development Plan."

These price estimates are preliminary since they are based upon the design criteria generated in Task 1, and as such they are subject to refinement as the physical definition of the machinery is undertaken in subsequent tasks in this program. This should be taken into consideration when utilizing these estimates in the Task 1 System Studies projection of plant economics. If possible, these initial studies should also be used to identify target prices which could then be guides in developing the preliminary design of the machinery.

Section 1

INTRODUCTION

This volume describes the work performed by the Westinghouse Electric Corporation in completing Task 1 of a five task study with an overall objective of determining the economic and technical feasibility of an aquifer based Compressed Air Energy Storage (CAES) plant.

The specific objectives of this effort were:

- Identification of preferred heat cycles for compressor and turbine systems applicable to an aquifer based CAES plant.
- Preparing system performance and cost estimates for use in evaluating potential aquifer sites.
- Providing major equipment definition for the preparation of conceptual plant designs.

Concurrent with these Task 1 activities an aquifer site identification and selection process was performed by Sargent & Lundy Engineers (Task 2). A partial objective of this work was the definition of the physical properties of available aquifers. To permit the selection of the preferred heat cycles and the definition of the system performance in Task 1, prior to the selection of the aquifer, a range of aquifer characteristics was defined early in the task which encompassed the known characteristics of potential sites.

Three nominal system pressure levels (200, 600 and 1000 psi) were identified and compressor and turbine cycles were optimized for each pressure level. The use of standardized components in the low and intermediate pressure sections of all three of the optimized cycles was investigated. The results of this work is presented in this volume.

Subsequently, in Task 3, the criteria and characteristics of the aquifer site selected in Task 2 will be integrated with the results of Task 1 and specific design approaches for the CAES systems, subsystems and components will be formulated.

Section 2

SITE AND HEAT CYCLE PARAMETERS

SITE DEPENDENT PARAMETERS

The selection of the air storage aquifer site was conducted in parallel with the identification of the general design criteria for an aquifer based Compressed Air Energy Storage (CAES) plant. These general design criteria will be used as the basis for formulating the site specific design criteria in a subsequent task of this phase of the compressed air storage development program. The site specific design criteria will utilize the general design criteria developed herein and the specific characteristics of the selected aquifer site. The site dependent parameters used throughout this report to develop the general design criteria are the following:

- Site Ambient Parameters

Pressure	14.43 PSIA
Temperature	59°F (dry bulb)
Relative Humidity	60%

- Aquifer Dependent Parameters

  The aquifer dependent parameters listed relate to the storage pressure of the selected aquifer which in turn affects the well and piping air pressure loss.

  The storage pressure range was based upon a preliminary review of the information available on candidate aquifers in the general region of the Illinois Basin in Illinois and Indiana.

  The well and piping pressure losses are expressed in the External Plant Air Pressure Loss Ratio. This ratio is defined as the compression equipment discharge pressure divided by the regenerator inlet pressure.

  Selected Nominal System Pressures

  200 PSIA
  600 PSIA
  1000 PSIA

Selected External Plant Air Pressure Loss Ratios

Base = 1.4
Maximum = 1.8
Minimum = 1.2

External Plant Air Temperature Loss

Aftercooler discharge - regenerator inlet = 10°F

- Heat Cycle Parameters

The development and evaluation of the heat cycles required that the range or limiting values of certain parameters be established. These values are:

- Heat Cycle Development

Cooling water supply temperature	95°F
Cooling water maximum temperature rise	40°F
Maximum aquifer air injection temperature	150°F
Minimum regenerator average cold end temperature	205°F

The average cold end temperature is equal to the sum of the regenerator exit stack temperature and the expansion cycle inlet temperature divided by two.

- Heat Cycle Evaluation

The evaluation of candidate heat cycles requires that a range of predicted liquid fuel costs and electric pumping power costs be established for use in determining the total power production energy cost for each cycle. The following combinations of liquid fuel and electric pumping power costs were used in the evaluation.

Liquid Fuel Cost						
\$ per 10^6 Btu (LHV = 18,055 BTU/LB)	2.50	2.50	5.00	7.50	7.50	7.75
Electric Power Pumping						
Mils per kwh	10	20	20	10	20	60

Section 3

CANDIDATE HEAT CYCLES

GENERAL

The heat cycles investigated were limited to those which represent current technology or near term engineering modifications thereof. In order to minimize the work of compression and remain within pressure ratio limits of current single stage compressor technology, all of the cycles investigated included intercooling with the compressor pressure ratio split adjusted to produce minimum compression cycle work. In addition, aftercooling was utilized to reduce the aquifer air injection temperature.

Two basic thermodynamic cycles were initially considered; the intercooled regenerative reheat cycle shown in Figure 3-1, and the intercooled reheat combined cycle shown in Figure 3-2.

The machinery arrangements shown in these figures are typical and do not identify a preferred arrangement.

While the compression cycles are similar in each, the expansion (power generation) schemes are considerably different. The regenerative reheat cycle utilizes an air to gas heat exchanger to recover heat from the low pressure turbine exhaust; while the reheat combined cycle employs a steam generator and turbine for heat recovery and power generation. Since the air returning from storage will be 140°F as it enters the regenerator, the regenerative cycle is capable of recovering a greater amount of heat from the exhaust gas than in the normal combustion turbine application. As will be shown (Table 3-1) the intercooled regenerative reheat cycle is the more efficient method of improving the cycle heat recovery than is heat recovery through steam generation in a combined cycle.

CANDIDATE CYCLE COMPARISON

Cycle analyses previously performed by Westinghouse (Reference 1) have shown that the intercooled, reheat combined cycle has a higher total energy cost than the

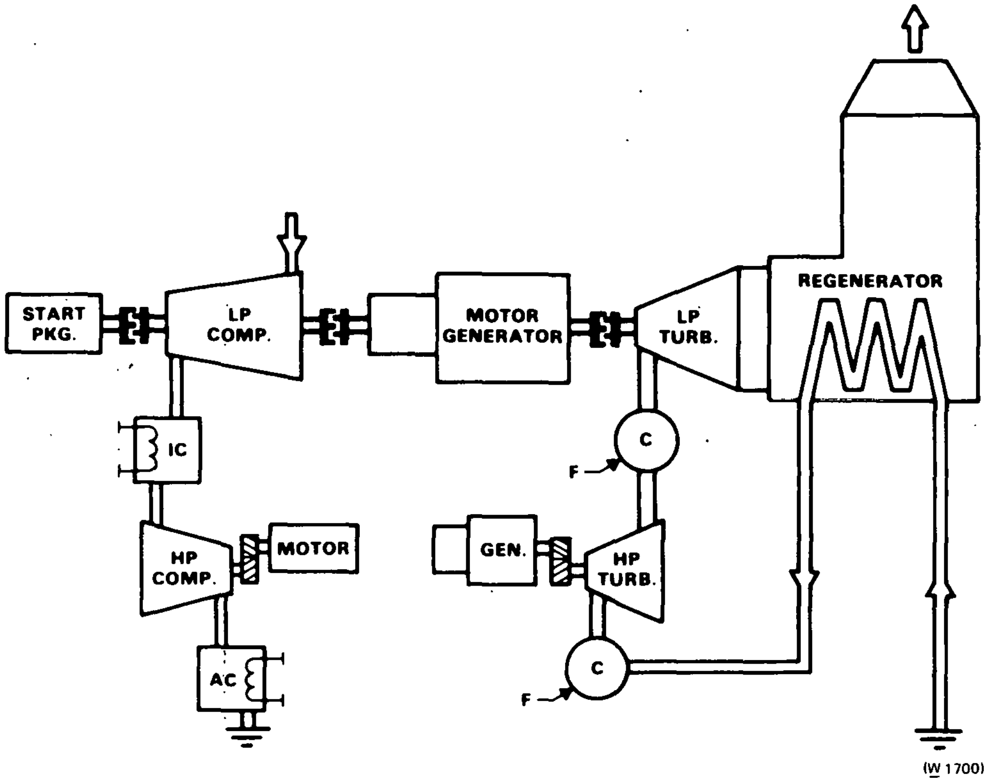


Figure 3-1. Intercooled Regenerative Reheat Cycle for CAES Systems

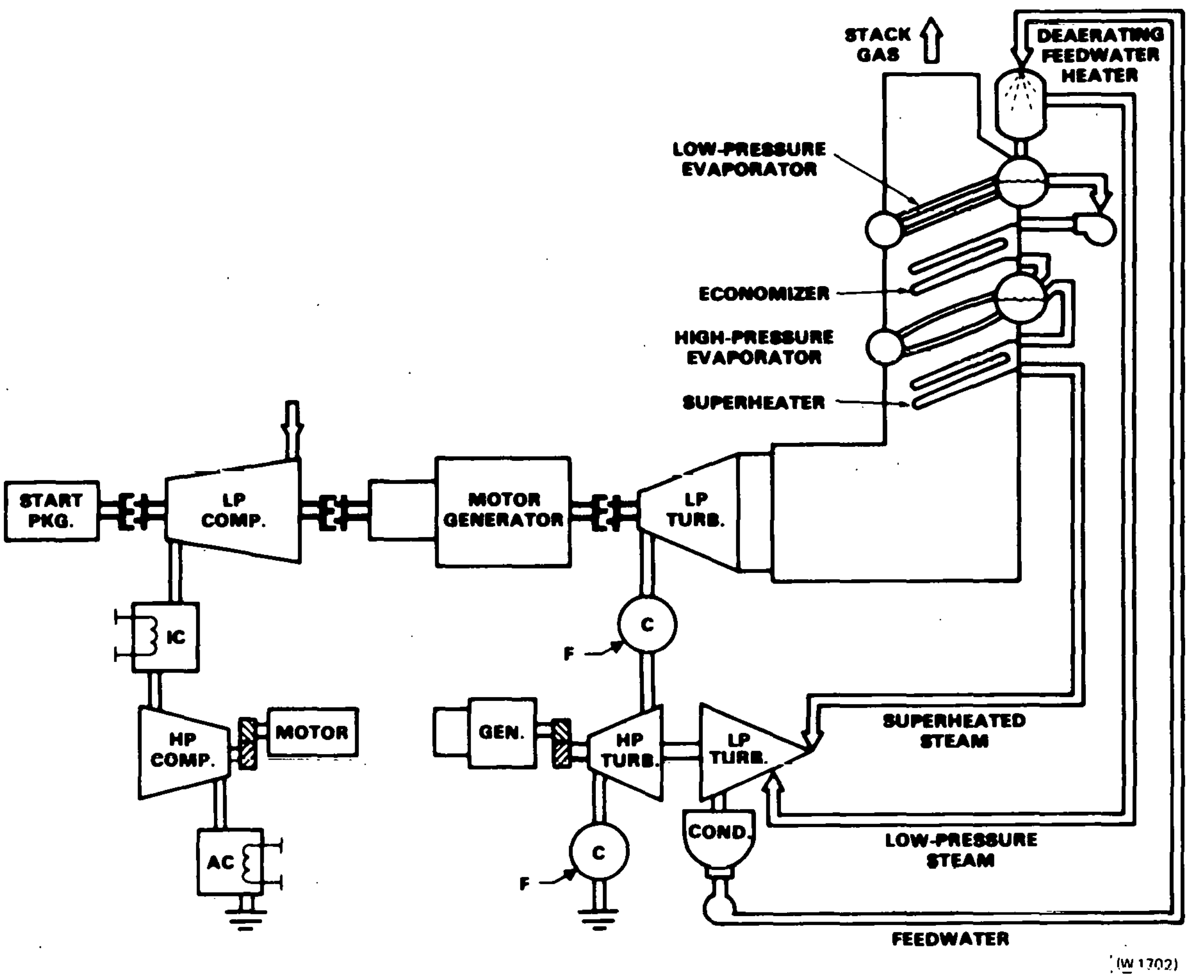


Figure 3-2. Intercooled Reheat Combined Cycle for CAES Systems

intercooled, regenerative reheat cycle, in a compressed air energy storage application where the stored air is 200°F or less. The combined cycle does not become better than the regenerative system until air storage temperature reaches the 500°F range.

The performance of the two cycles was compared on the basis of the total power production energy cost; that is, the cost of the electric power required by the compressors and the cost of the fuel burned in the turbines during power generation. The original analysis performed in reference 1 was updated by inserting the electric pumping power and fuel cost combinations selected for the Task 1 phase of the program. The results of this performance comparison are shown on Table 3-1.

Since the air storage temperature established for the project is 150°F and the return air to the expansion equipment is 140°F, these analyses are applicable. Cycle analyses were also performed in Reference 1 at 700 psi and little difference was noted with regard to the optimum thermodynamic cycle.

CANDIDATE CYCLE SELECTION

In all of the combinations of turbine inlet temperature and electric power/fuel cost ratios shown on Table 3-1, the total power production energy cost of the regenerative reheat cycle was less than that of the combined cycle.

The storage air temperature in the Reference 1 calculations was 200°F whereas the aquifer storage temperature defined in this compressed air energy study is 150°F. Since the exhaust heat recovery performance of the regenerative reheat cycle improves as the storage air temperature decreases, the comparison in Table 3-1 is conservative in that the regenerative reheat cycle advantage will increase at the lower (150°F) air storage temperature.

In view of these results the regenerative reheat cycle was selected for further development in this study.

REFERENCES

1. P.A. Berman. "Compressed Air Energy Storage Turbo-Machinery," London, England. Gas Turbine Conference & Products Show, ASME Publication 78-GT-97, 1978.

Table 3-1

TOTAL POWER PRODUCTION ENERGY COST COMPARISON FOR REGENERATIVE REHEAT AND REHEAT COMBINED CYCLES

Storage Pressure	1000 PSI
Site Elevation	Sea Level
Regenerator Effectiveness	75%
Storage Pressure Loss	1.21
Storage Air Temperature	200°F

Fuel Cost, $/MM BTU		$2.50	$2.50	$5.00	$7.50	$7.50	$7.50
Charging Cost, Mils/KWHR		10	20	20	10	20	60
Turbine Inlet Temperature °F		Total Power Production Energy Cost - Mils/KWHR					
$\frac{HPT}{LPT} \Rightarrow \frac{1000}{1500}$	RC	19.51	27.47	39.03	42.62	50.58	83.57
	RR	18.75	27.45	37.49	38.84	47.54	83.35
$\frac{HPT}{LPT} \Rightarrow \frac{1300}{1500}$	RC	18.83	26.26	37.66	41.62	49.05	79.91
	RR	18.06	26.14	36.13	38.03	46.11	79.43
$\frac{HPT}{LPT} \Rightarrow \frac{1500}{1500}$	RC	18.41	25.51	36.81	41.02	48.12	77.65
	RR	17.65	25.36	35.31	37.54	45.25	77.07

(W 1597)

HPT = High Pressure Turbine Inlet Temperature °F   RR = Regenerative Reheat Cycle

LPT = Low Pressure Turbine Inlet Temperature °F   RC = Reheat - Combined Cycle

NOTE: The above total power production energy costs are as calculated and do not include any margin.

Section 4

COMPRESSION CYCLE OPTIMIZATION

To achieve the system performance required for an economically viable CAES system it is necessary to optimize both the compression and expansion thermal cycles. In the compression cycle, intercooling must be considered. The selection of the number and location of intercoolers defines the pressure ratio of each compressor element in the system. It was necessary, therefore, to develop a technique for identifying the optimum compressor pressure ratio for each of the three aquifer pressures and the related intercooler outlet temperatures selected for the Task 1 analyses. Equations were developed to directly calculate the optimum compressor train pressure ratio split, for one and two stages of intercooling, which results in the minimum required compressor work. The derivation of these equations is contained in Appendix A of this report; however, the compressor train pressure ratio optimizing equations are presented in the main body of this text.

The optimum compressor pressure ratio was calculated for the following site parameters:

<u>Site Ambient Conditions</u>

Pressure - 14.43 psia
Temperature - 59°F

<u>Intercooler Outlet Temperatures</u>

95°F
105°F (base)
115°F

<u>Aftercooler Discharge Pressures</u>

250 PSIA
750 PSIA
1150 PSIA

A CAES COMPRESSOR TRAIN UTILIZING ONE INTERCOOLER

The equation used to calculate the optimum pressure ratio split for a compressor train utilizing one intercooler shown in Figure 4-1 is presented below:

$$\rho_1 = \left(\frac{m_2 cp_2 T_2 \eta_1}{m_1 cp_1 T_1 \eta_2}\right)^{\left(\frac{K}{2(K-1)}\right)} (\rho_T)^{1/2}$$

and

$$\rho_2 = \frac{\rho_T}{\rho_1}$$

where:

- m = the changing mass flow rate
- cp = the constant pressure specific heat of air
- η = compressor adiabatic efficiency
- ρ = compressor pressure ratio
- ρ_T = compressor train total pressure ratio
- K = the specific heat ratio of air
- T = compressor inlet temperature, (^oR)

and the subscripts 1 and 2 refer to the LP and HP compressor, respectively.

The detailed derivation of the above equation which determines the optimum intercooler location that allows the charging of an aquifer with a minimum of compressor work is presented in Appendix A.

Since a constant aftercooler air discharge temperature is maintained, the placement of the intercooler does not effect the turbine cycle. Thus by minimizing the total compressor work by the proper placement of the intercooler, the charging compression cycle can be optimized independent of the turbine cycle.

Data generated from the above equation, for a single stage of intercooling is plotted on Figure 4-2 for three aftercooler discharge pressures. All points on

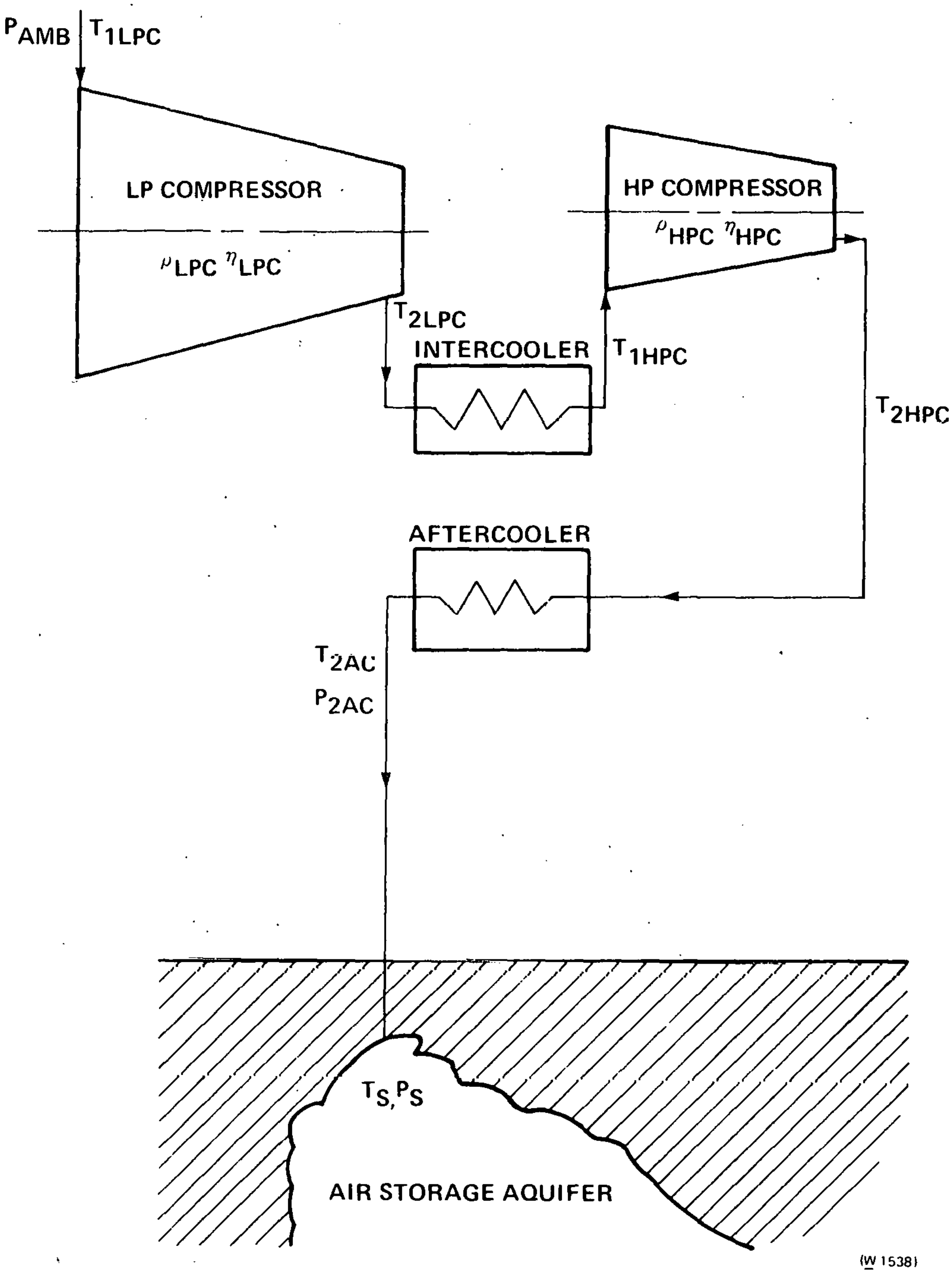
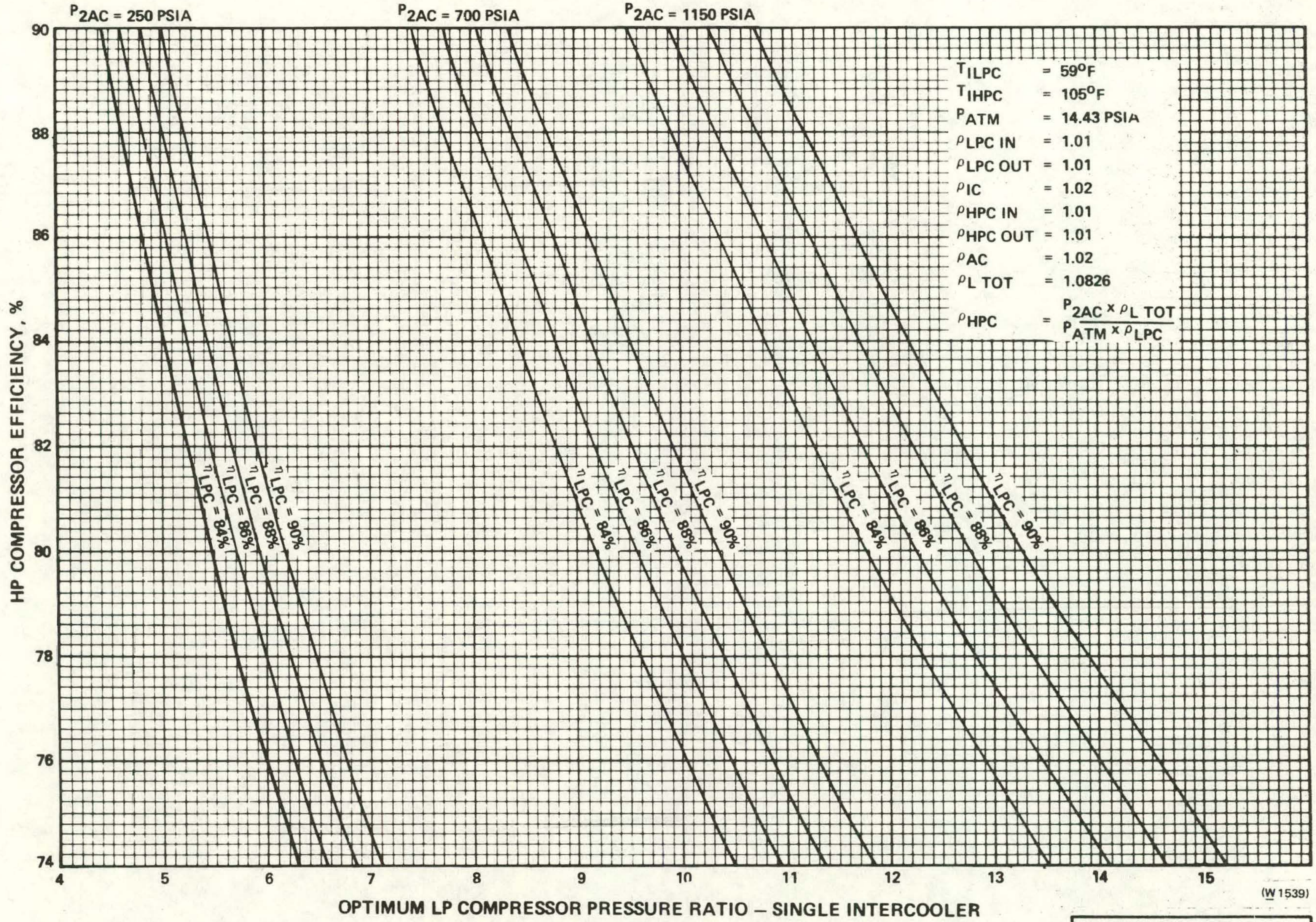


Figure 4-1. Schematic of a CAES Compressor Train Utilizing One Intercooler



EFFECT OF PRESSURE LEVEL

Figure 4-2. Optimum LP Compressor Pressure Ratio - Single Intercooler

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these curves represent the optimum pressure ratio split which leads to the minimum work of compression. At a given high pressure compressor (HPC) efficiency and low pressure compressor (LPC) efficiency, the low pressure compressor pressure ratio is read directly from the curve. The high pressure compressor pressure ratio can be calculated from the following equation:

$$\rho_{HPC} = \frac{P_{2AC} \times \rho_{L\ TOT}}{P_{ATM} \times \rho_{LPC}}$$

where:

ρ_{HPC} = high pressure compressor pressure ratio

P_{2AC} = aftercooler discharge pressure, PSIA

$\rho_{L\ TOT}$ = compressor train inlet and exit total pressure loss coefficient

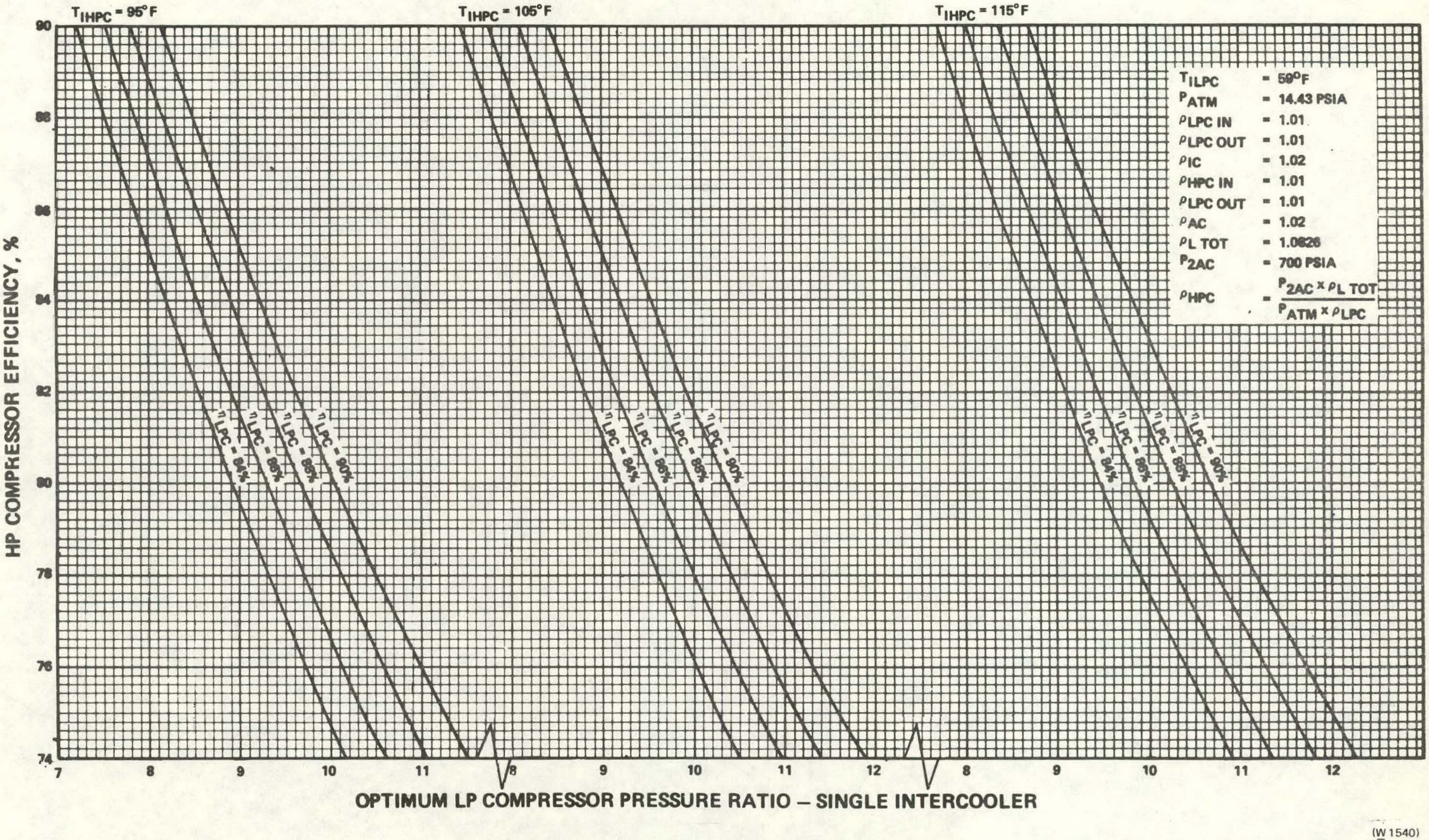
P_{ATM} = site ambient pressure, PSIA

ρ_{LPC} = low pressure compressor pressure ratio

Projected compressor efficiencies were estimated based on Westinghouse machinery experience, which when applied to the curves on Figure 4-2 identified the optimum (minimum work) compressor pressure ratio split between the high and low pressure compressors. It is obvious from the curves that the optimum compressor pressure ratio split is strongly influenced by the component efficiencies. An increase in the efficiency of one compressor body increases the pressure ratio which that component must achieve in order to provide a minimum work of compression system.

A second observation is that the maximum pressure ratio required in any of the optimized systems is approximately 15 to 1 in the low pressure unit. Such pressure ratios are obtainable with present day state of the art axial compressors.

Figure 4-3 is a curve showing the effect of the intercooler outlet temperature on the optimum compressor pressure ratio for a system having one stage of intercooling. The data plotted was generated by utilizing the single intercooler, optimum pressure ratio equation presented above and derived in Appendix A. All three groups of curves shown are for an aftercooler discharge pressure of 700 PSIA. Intercooler air discharge (HP compressor inlet) temperatures of 95°F, 105°F and 115°F were investigated. The optimum compressor pressure ratio is not significantly



EFFECT OF INTERCOOLER OUTLET TEMPERATURE

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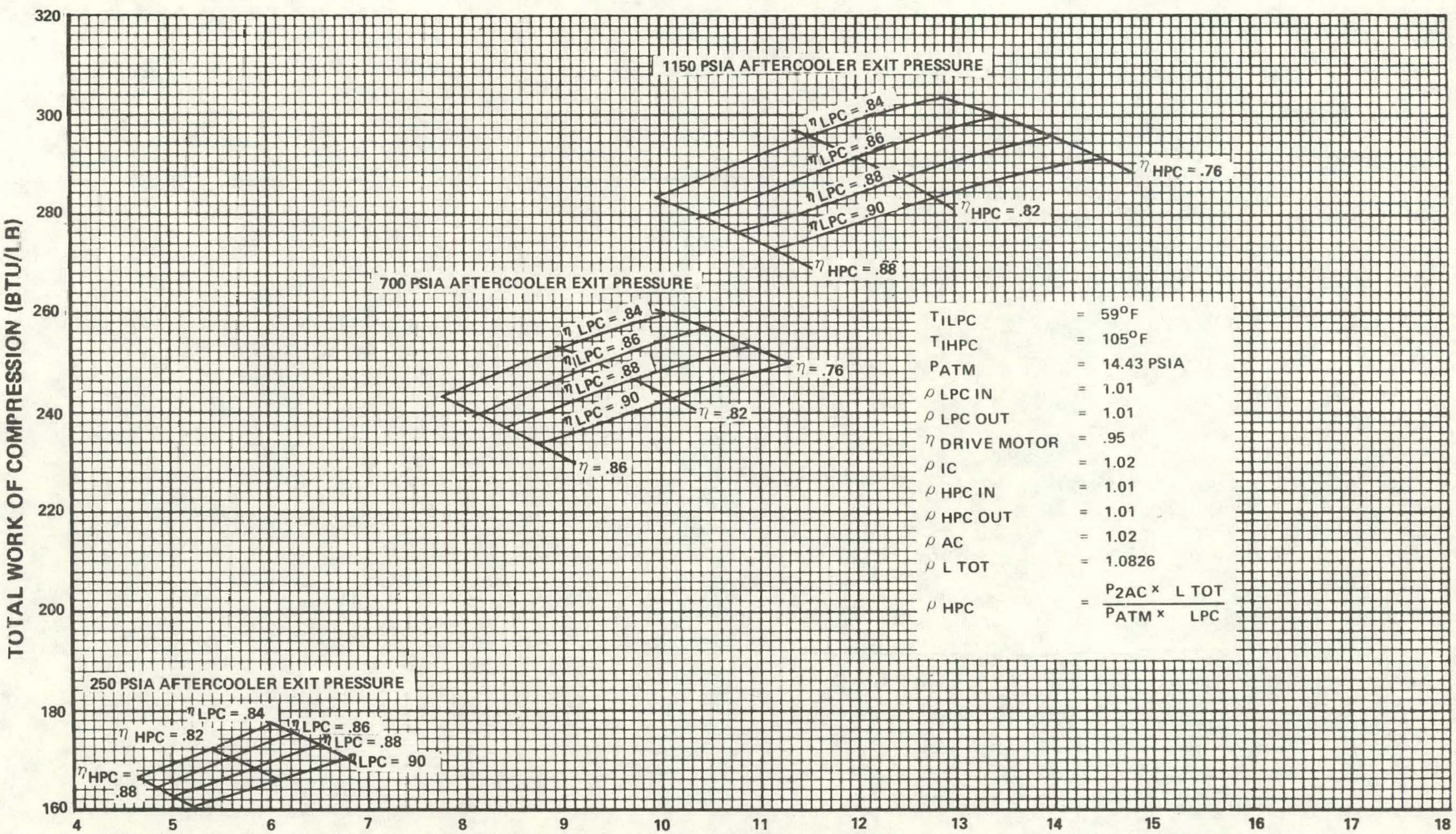
Figure 4-3. Optimum LP Compressor Pressure Ratio - Single Intercooler - Effect of Intercooler Outlet Temperature sensitive to the HP compressor inlet temperature over the 20°F variation shown. The curves also show that as the intercooler outlet air temperature decreases, the high pressure compressor pressure ratio should increase in order to maintain an optimized (minimum work) system. In actual operation, with an axial LP compressor, the pressure ratio across the high pressure compressor will increase as its inlet air temperature decreases. This behavior would tend to automatically maintain an optimized compressor system should the intercooler air discharge temperature vary due to changes in the cooling water supply temperature.

The analyses for a compressor train utilizing one intercooler are summarized on Figure 4-4. This figure displays the low pressure compressor (LPC) pressure ratio and the total work of compression as functions of the low pressure compressor and high pressure compressor efficiencies and three aftercooler exit pressures. The data presented in Figure 4-4 reflects a high pressure compressor (HPC) inlet temperature of 105°F resulting from an assumed 10°F intercooler design point approach temperature.

The two different types of machines that can be used in a large flow rate, high discharge pressure compressor train utilizing a single intercooler are axial or centrifugal units. The low pressure compressor, operating essentially at the constant site ambient inlet conditions should be an axial flow unit because of the inherently higher (than centrifugal) efficiency of this type of compressor. Also the wide availability of such low pressure turbomachinery is a definite plus. Either an axial or a centrifugal unit could be utilized for the high pressure compressor since each has advantages at these operating conditions. A centrifugal unit would be lower in efficiency, but would be more suited for ducting the compressed air to and from aircoolers. However, an axial unit would be more efficient which is a paramount consideration.

Initially both axial and centrifugal HP compressors were analyzed. As stated above, Figure 4-4 displays the correlation between the optimum LP compressor pressure ratio, and the resulting total work of compression required by compressor trains utilizing a single intercooler at the three selected design point aftercooler exit pressures. The optimized compressor pressure ratios and the resulting minimum total work of compression extracted from Figures 4-3 and 4-4 for the initially estimated compressor efficiencies are shown below.



OPTIMUM LP COMPRESSOR PRESSURE RATIO AND TOTAL WORK OF COMPRESSION FOR A COMPRESSOR TRAIN UTILIZING ONE INTERCOOLER

(W 1500)

NOTE: The above total work of compression are as calculated and do not include any margins.

Figure 4-4. Optimum LP Compressor Pressure Ratio and Total Work of Compression for a Compressor Train Utilizing One Intercooler

COMPRESSOR TRAIN UTILIZING ONE INTERCOOLER

		Aftercooler Discharge Pressure, PSIA		
		250	700	1150
Low Pressure Compressor	A/A	5.10	8.38	10.50
Pressure Ratio	A/C	6.36	10.65	13.37
High Pressure Compressor	A/A	3.68	6.27	8.06
Pressure Ratio	A/C	2.95	4.93	6.45
Total Work of Compression	A/A	164	238	276
BTU/LB	A/C	171	251	297

A/A - A compressor train utilizing a LP and HP axial compressor.

A/C - A compressor train utilizing a LP axial compressor and a HP centrifugal compressor.

NOTE: The above total work of compression are as calculated and do not contain any margins.

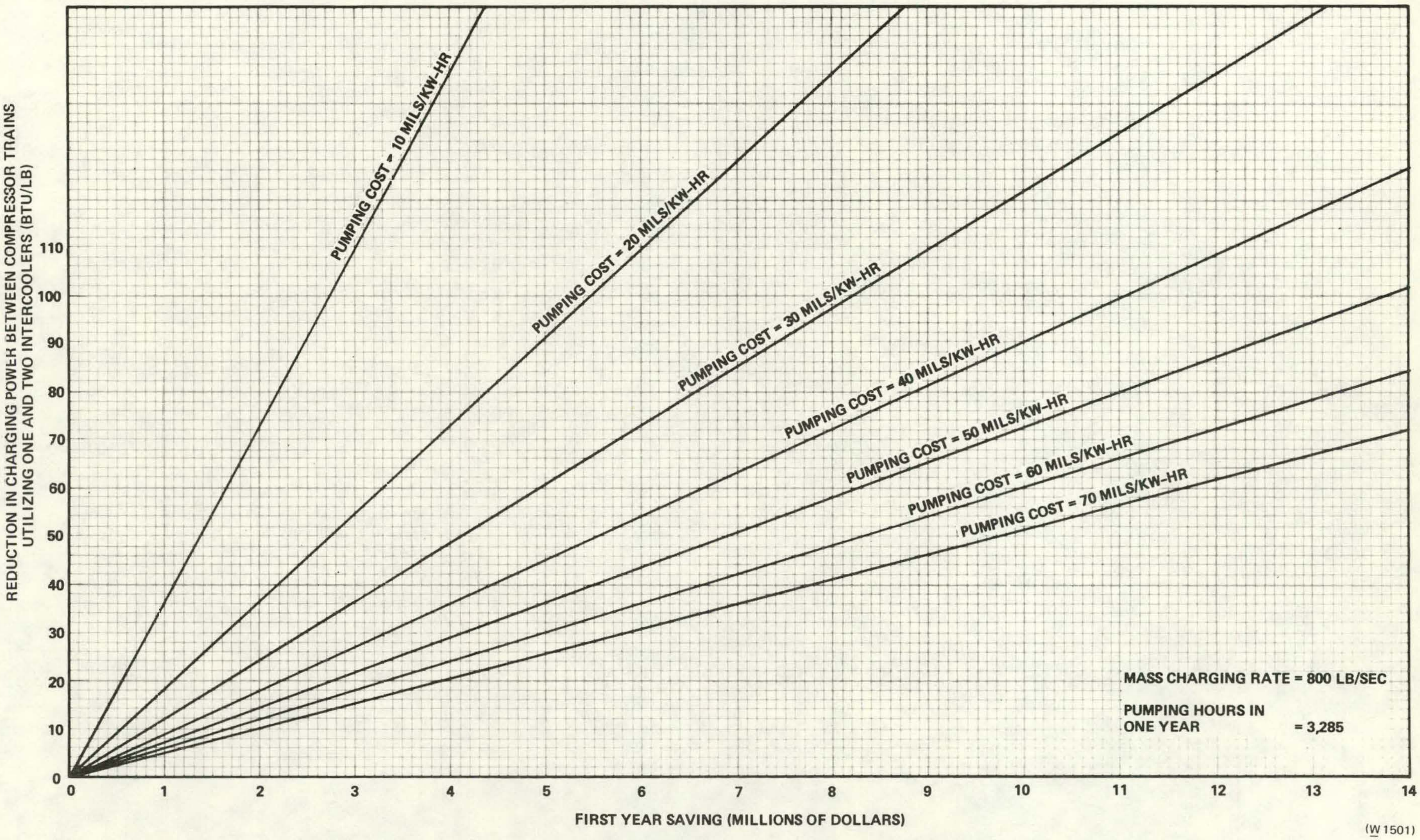
A CAES COMPRESSOR TRAIN UTILIZING TWO INTERCOOLERS

The effect of intercooling is to reduce the total work of compression. The greater the amount of intercooling, the greater the reduction in the compressor work. Additional intercoolers, however, add to the initial plant cost in the form of heat exchangers, pumps, piping and compressor casings, and to the operating cost because the added complexity will cause higher maintenance costs.

In Figure 4-5 the first year savings is shown for various pumping costs, as a function of the reduction in compressor power required to charge an aquifer. The curves are based upon an aquifer charging rate of 800 Lb/Sec for approximately 9 hours/day for 365 days/year. These curves indicate how relatively modest reductions in the required compressor power result in significant operating cost savings.

The attainment of this charging power reduction is more easily realized at the higher aquifer injection pressures than at the lower, because of the greater amount of heat generated in reaching the higher pressure levels.

The derivation for the equation for calculating the optimum intercooler location for a compressor train utilizing two intercoolers is presented in Appendix A. The



REDUCTION OF AQUIFER CHARGING COSTS BY UTILIZING A SECOND INTERCOOLER

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Figure 4-5. Reduction of Charging Costs By Utilizing A Second Intercooler schematic for such a compressor train which consists of three compressor bodies and two intercoolers is presented in Figure 4-6. The equation resulting from the above derivation is presented below:

$$\rho_1 = \left[\left(\frac{m_2 cp_2 T_2 \eta_1 m_3 cp_3 T_3 \eta_1}{m_1 cp_1 T_1 \eta_2 m_1 cp_1 T_1 \eta_3}\right)^{\frac{K}{3(K-1)}} (\rho_T)\right]^{1/3}$$

$$\rho_2 = \left(\frac{m_1 cp_1 T_1 \eta_2}{m_2 cp_2 T_2 \eta_1}\right)^{\frac{K}{K-1}} \rho_1$$

$$\rho_3 = \frac{\rho_T}{\rho_1 \rho_2}$$

where:

m = the charging mass flow rate

cp = the constant pressure specific heat of air

η = compressor adiabatic efficiency

ρ = compressor pressure ratio

ρ_T = compressor train total pressure ratio

K = the specific heat ratio of air

T = compressor inlet temperature (^{o}R)

and the subscripts 1, 2, and 3 refer to the LP, IP and HP compressor, respectively.

The above equation for a compressor train utilizing two intercoolers, which defines the optimum (minimum work) compressor pressure ratios for the high pressure (HPC), intermediate pressure (IPC) and low pressure (LPC), compressors was used to generate the data in Tables 4-1, 4-2 and 4-3. The symbols used on the schematic in Figure 4-6 apply to these tables. Each of the three tables is prepared for a different intercooler exit temperature. They identify the optimum pressure ratio and resulting minimum work of compression for various combinations of compressor efficiencies at three aftercooler discharge pressures.

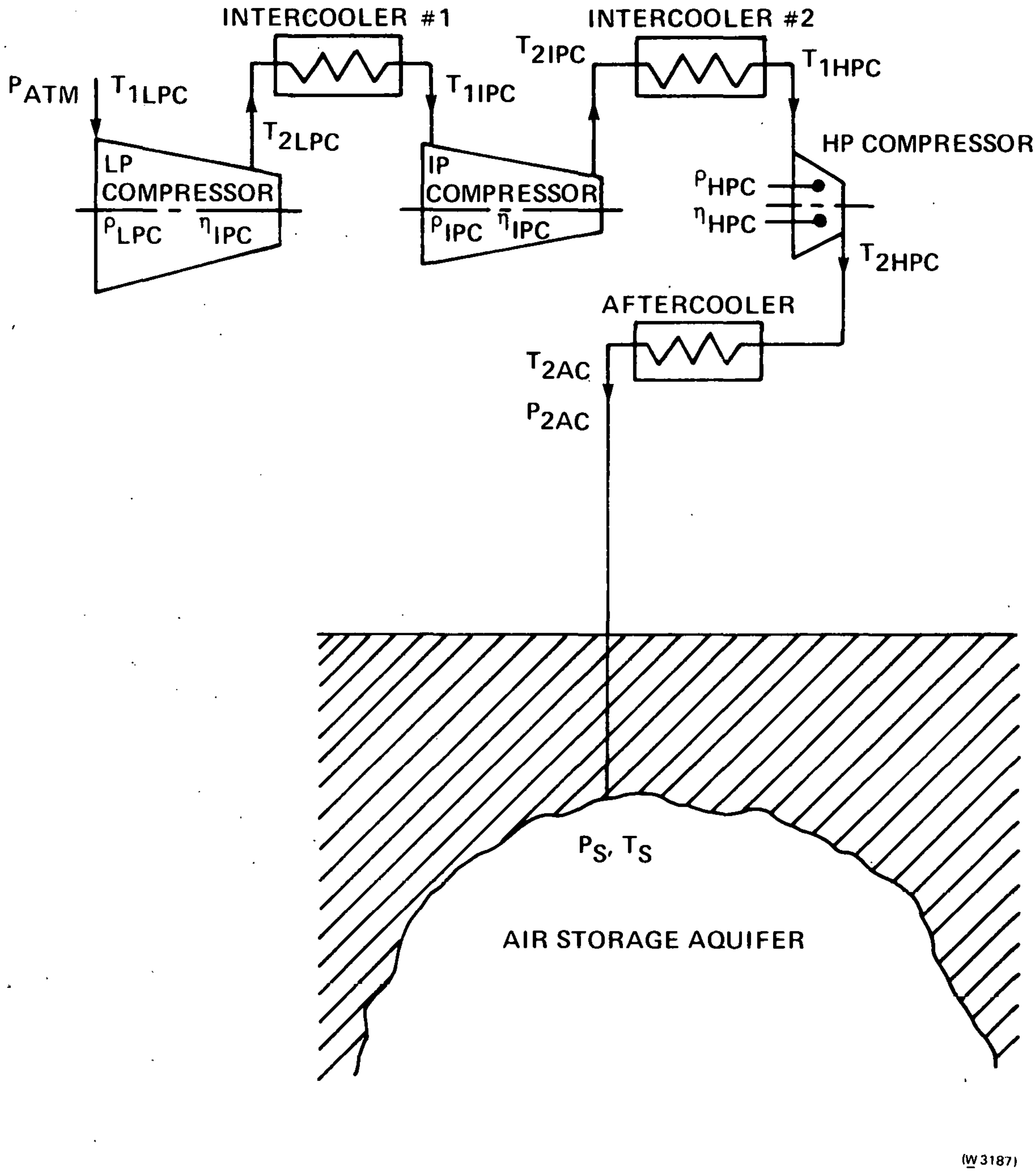

Figure 4-6. Schematic of a CAES Compressor Train Utilizing Two Intercoolers

Table 4-1

PRELIMINARY
OPTIMUM PRESSURE RATIO SPLIT AND WORK OF COMPRESSION
FOR A COMPRESSOR TRAIN UTILIZING TWO INTERCOOLERS
95°F INTERCOOLER OUTLET TEMPERATURE

$T_{1IPC} = T_{1HPC} = 95°F$	P_{2AC} = 250 PSIA	P_{2AC} = 700 PSIA	P_{2AC} = 1150 PSIA
η_{LPC} = .88 η_{IPC} = .85 η_{HPC} = .85	ρ_{LPC} = 3.4205 ρ_{IPC} = 2.3887 ρ_{HPC} = 2.3887 W_{TOT} = 155.66 Btu/Lb	ρ_{LPC} = 4.8209 ρ_{IPC} = 3.3664 ρ_{HPC} = 3.3664 W_{TOT} = 220.88 Btu/Lb	ρ_{LPC} = 5.6885 ρ_{IPC} = 3.9723 ρ_{HPC} = 3.9723 W_{TOT} = 254.76 Btu/Lb
η_{LPC} = .88 η_{IPC} = .85 η_{HPC} = .80	ρ_{LPC} = 3.6712 ρ_{IPC} = 2.5638 ρ_{HPC} = 2.0736 W_{TOT} = 158.01 Btu/Lb	ρ_{LPC} = 5.1746 ρ_{IPC} = 3.6134 ρ_{HPC} = 2.9226 W_{TOT} = 224.91 Btu/Lb	ρ_{LPC} = 6.1058 ρ_{IPC} = 4.2637 ρ_{HPC} = 3.4485 W_{TOT} = 259.46 Btu/Lb
η_{LPC} = .88 η_{IPC} = .80 η_{HPC} = .75	ρ_{LPC} = 4.2484 ρ_{IPC} = 2.3996 ρ_{HPC} = 1.9144 W_{TOT} = 163.26 Btu/Lb	ρ_{LPC} = 5.9882 ρ_{IPC} = 3.3821 ρ_{HPC} = 2.6982 W_{TOT} = 233.31 Btu/Lb	ρ_{LPC} = 7.0658 ρ_{IPC} = 3.9908 ρ_{HPC} = 3.1838 W_{TOT} = 269.25 Btu/Lb
η_{LPC} = .85 η_{IPC} = .75 η_{HPC} = .75	ρ_{LPC} = 4.2245 ρ_{IPC} = 2.1494 ρ_{HPC} = 2.1494 W_{TOT} = 169.65 Btu/Lb	ρ_{LPC} = 5.9544 ρ_{IPC} = 3.0296 ρ_{HPC} = 3.0296 W_{TOT} = 241.42 Btu/Lb	ρ_{LPC} = 7.0260 ρ_{IPC} = 3.5748 ρ_{HPC} = 3.5748 W_{TOT} = 278.54 Btu/Lb

NOTE: The above total work of compression values are as calculated and do not include any margins.

(W 1541)

Table 4-2

PRELIMINARY
OPTIMUM PRESSURE RATIO SPLIT AND WORK OF COMPRESSION
FOR A COMPRESSOR TRAIN UTILIZING TWO INTERCOOLERS
105°F INTERCOOLER OUTLET TEMPERATURE

$T_{1IPC} = T_{1HPC} = 105^oF$	P_{2AC} = 250 PSIA	P_{2AC} = 700 PSIA	P_{2AC} = 1150 PSIA
η_{LPC} = .88 η_{IPC} = .85 η_{HPC} = .85	ρ_{LPC} = 3.5695 ρ_{IPC} = 2.3384 ρ_{HPC} = 2.3384 W_{TOT} = 157.91 Btu/Lb	ρ_{LPC} = 5.0311 ρ_{IPC} = 3.2958 ρ_{HPC} = 3.2958 W_{TOT} = 223.34 Btu/Lb	ρ_{LPC} = 5.9365 ρ_{IPC} = 3.8890 ρ_{HPC} = 3.8890 W_{TOT} = 257.56 Btu/Lb
η_{LPC} = .88 η_{IPC} = .85 η_{HPC} = .80	ρ_{LPC} = 3.8312 ρ_{IPC} = 2.5098 ρ_{HPC} = 2.0297 W_{TOT} = 159.88 Btu/Lb	ρ_{LPC} = 5.3998 ρ_{IPC} = 3.5373 ρ_{HPC} = 2.8610 W_{TOT} = 227.35 Btu/Lb	ρ_{LPC} = 6.3717 ρ_{IPC} = 4.1741 ρ_{HPC} = 3.3755 W_{TOT} = 262.21 Btu/Lb
η_{LPC} = .88 η_{IPC} = .80 η_{HPC} = .75	ρ_{LPC} = 4.4335 ρ_{IPC} = 2.3489 ρ_{HPC} = 1.8741 W_{TOT} = 165.44 Btu/Lb	ρ_{LPC} = 6.2488 ρ_{IPC} = 3.3109 ρ_{HPC} = 2.6413 W_{TOT} = 235.72 Btu/Lb	ρ_{LPC} = 7.3734 ρ_{IPC} = 3.9064 ρ_{HPC} = 3.1168 W_{TOT} = 272.31 Btu/Lb
η_{LPC} = .85 η_{IPC} = .75 η_{HPC} = .75	ρ_{LPC} = 4.4085 ρ_{IPC} = 2.1040 ρ_{HPC} = 2.1040 W_{TOT} = 171.31 Btu/Lb	ρ_{LPC} = 6.2137 ρ_{IPC} = 2.9656 ρ_{HPC} = 2.9656 W_{TOT} = 243.89 Btu/Lb	ρ_{LPC} = 7.3318 ρ_{IPC} = 3.4995 ρ_{HPC} = 3.4995 W_{TOT} = 281.51 Btu/Lb

NOTE: The above total work of compression values are as calculated and do not include any margins.

(W 1542)

Table 4-3

PRELIMINARY
OPTIMUM PRESSURE RATIO SPLIT AND WORK OF COMPRESSION
FOR A COMPRESSOR TRAIN UTILIZING TWO INTERCOOLERS
115^oF INTERCOOLER OUTLET TEMPERATURE

$T_{1IPC} = T_{1HPC} = 115^oF$	P_{2AC} = 250 PSIA	P_{2AC} = 700 PSIA	P_{2AC} = 1150 PSIA
η_{LPC} = .88 η_{IPC} = .85 η_{HPC} = .85	ρ_{LPC} = 3.7223 ρ_{IPC} = 2.2898 ρ_{HPC} = 2.2898 W_{TOT} = 159.56 Btu/Lb	ρ_{LPC} = 5.2463 ρ_{IPC} = 3.2275 ρ_{HPC} = 3.2275 W_{TOT} = 225.80 Btu/Lb	ρ_{LPC} = 6.1904 ρ_{IPC} = 3.8083 ρ_{HPC} = 3.8083 W_{TOT} = 259.41 Btu/Lb
η_{LPC} = .88 η_{IPC} = .85 η_{HPC} = .80	ρ_{LPC} = 3.9951 ρ_{IPC} = 2.4576 ρ_{HPC} = 1.9877 W_{TOT} = 161.62 Btu/Lb	ρ_{LPC} = 5.6311 ρ_{IPC} = 3.4642 ρ_{HPC} = 2.8013 W_{TOT} = 229.76 Btu/Lb	ρ_{LPC} = 6.6444 ρ_{IPC} = 4.0877 ρ_{HPC} = 3.3054 W_{TOT} = 265.09 Btu/Lb
η_{LPC} = .88 η_{IPC} = .80 η_{HPC} = .75	ρ_{LPC} = 4.6233 ρ_{IPC} = 2.3003 ρ_{HPC} = 1.8351 W_{TOT} = 166.85 Btu/Lb	ρ_{LPC} = 6.5164 ρ_{IPC} = 3.2426 ρ_{HPC} = 2.5862 W_{TOT} = 237.76 Btu/Lb	ρ_{LPC} = 7.6891 ρ_{IPC} = 3.8261 ρ_{HPC} = 3.0516 W_{TOT} = 274.95 Btu/Lb
η_{LPC} = .85 η_{IPC} = .75 η_{HPC} = .75	ρ_{LPC} = 4.5973 ρ_{IPC} = 2.0605 ρ_{HPC} = 2.0605 W_{TOT} = 172.98 Btu/Lb	ρ_{LPC} = 6.4796 ρ_{IPC} = 2.9042 ρ_{HPC} = 2.9042 W_{TOT} = 246.42 Btu/Lb	ρ_{LPC} = 7.6456 ρ_{IPC} = 3.4268 ρ_{HPC} = 3.4266 W_{TOT} = 284.36 Btu/Lb

NOTE: The above total work of compression values are as calculated and do not include any margins.

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The above tables were prepared to provide an overview of this sytem, at a broad range of compressor efficiencies, to identify trends and to establish pressure ratios which were used as a first approximation in determining the estimated efficiencies and resulting pressure ratios shown on Table 4-4. The data presented in this table uses optimized pressure ratios generated from the above equation for a compressor train utilizing two intercoolers. The selected design point axial compressor efficiencies were based upon Westinghouse estimates. The centrifugal design point efficiencies were extracted from performance information supplied by the Dressor-Clark Compressor Division.

The calculations shown on Tables 4-1, 4-2, 4-3 and 4-4 utilize the following site and system parameters:

$T_{1\ LPC}$	= 59°F	$\rho_{IPC\ OUT}$	= 1.01
P_{ATM}	= 14.43 PSIA	$\rho_{IC\ \#2}$	= 1.02
$\rho_{LPC\ IN}$	= 1.01	$\rho_{HPC\ IN}$	= 1.01
$\rho_{LPC\ OUT}$	= 1.01	$\rho_{HPC\ OUT}$	= 1.01
$\rho_{IC\ \#1}$	= 1.02	ρ_{AC}	= 1.02
$\rho_{IPC\ IN}$	= 1.01	ρ_{TOT}	= 1.1265
		$\eta_{DRIVE\ MOTOR}$	= .95

As was the case with the compressor train utilizing a single intercooler the above tables show that the work of compression is not significantly affected by the 20°F variation of the intercooler air outlet temperatures. As the intercooler outlet temperatures decrease, the tables show that the pressure ratios across the high and intermediate pressure compressors should increase while the low pressure compressor pressure ratio should decrease in order to maintain an optimized system. In actual operation the high and intermediate pressure compressors will increase their pressure ratios with decreasing inlet temperature, thus tending to automatically maintain the optimized condition as cooling water supply temperatures fluctuate.

The data also shows that an increase in efficiency of one unit increases the pressure ratio which that unit must achieve in order to produce a minimum work of compression system.

Table 4-4

ESTIMATED DESIGN POINT
OPTIMUM PRESSURE RATIO SPLIT AND WORK OF COMPRESSION
FOR A COMPRESSOR TRAIN UTILIZING TWO INTERCOOLERS
105°F INTERCOOLER OUTLET TEMPERATURE

Compressor Element	P_{2AC} = 250 PSIA	P_{2AC} = 700 PSIA	P_{2AC} = 1150 PSIA
LPC, η & ρ	η = .87 (A) 3.38	η = .88 (A) 4.77	η = .88 (A) 5.47
IPC, η & ρ	η = .86 (A) 2.40	η = .87 (A) 3.39	η = .88 (A) 4.05
HPC, η & ρ	η = .86 (A) 2.40	η = .87 (A) 3.39	η = .88 (A) 4.05
Total Work	W_{TOT} = 157 Btu/Lb	W_{TOT} = 221 Btu/Lb	W_{TOT} = 254 Btu/Lb
LPC, η & ρ	η = .88 (A) 3.90	η = .88 (A) 5.50	η = .88 (A) 6.40
IPC, η & ρ	η = .87 (A) 2.62	η = .87 (A) 3.90	η = .88 (A) 4.73
HPC, η & ρ	η = .77 (C) 1.91	η = .77 (C) 2.55	η = .77 (C) 2.97
Total Work	W_{TOT} = 161 Btu/Lb	W_{TOT} = 228 Btu/Lb	W_{TOT} = 262 Btu/Lb
LPC, η & ρ	η = .88 (A) 4.50	η = .88 (A) 6.34	η = .88 (A) 7.48
IPC, η & ρ	η = .77 (C) 2.08	η = .77 (C) 2.94	η = .77 (C) 3.47
HPC, η & ρ	η = .77 (C) 2.08	η = .77 (C) 2.94	η = .77 (C) 3.47
Total Work	W_{TOT} = 166 Btu/Lb	W_{TOT} = 237 Btu/Lb	W_{TOT} = 273 Btu/Lb

(W 1544)

(A) = Axial Compressor

(C) = Centrifugal Compressor

NOTE: The above total work of compression values are as calculated and do not include any margins.

When compared with the performance of a compressor train utilizing one intercooler, the compressor system with two intercoolers has a definite advantage because of the significant reduction in the required compressor power to charge an aquifer.

The discussion of the selection of the type of compressor, the number of intercoolers, and the design point intercooler exit temperature is contained in Section 6.

Section 5

CAES TURBINE TRAIN OPTIMIZATION

The schematic of a reheated CAES turbine expansion cycle with a regenerator is shown on Figure 5-1. An equation maximizing the total turbine work produced by this type of cycle is derived in Appendix B and presented below:

$$\rho_1 = \left[\left(\frac{m_1 cp_1 T_1 \eta_1}{m_2 cp_2 T_2 \eta_2}\right)^{\frac{K}{2(K-1)}} (\rho_T)\right]^{1/2}$$

where:

m = discharging mass flow rate

cp = constant pressure specific heat of air

η = turbine adiabatic efficiency

ρ_1 = the HP turbine pressure ratio

ρ_T = the total turbine pressure ratio

K = the ratio of the specific heats of air

T = turbine inlet temperature (^oR)

and the subscripts 1 and 2 refer to the HP and LP turbine, respectively.

The above equation is also valid for a turbine cycle utilizing a heat storage device instead of a regenerator or for a two stage turbine cycle not utilizing either. In Appendix B the equation is also modified for the case where the high pressure combustor is not utilized.

Figure 5-2 is a plot of data generated from this equation which shows, for given total turbine pressure ratios, the low pressure turbine pressure ratio required for maximum total work for three combinations of turbine efficiencies and four combinations of turbine inlet temperatures. The above data indicates that the optimum turbine pressure ratios are sensitive to turbine inlet temperature combinations and to the turbine efficiencies.

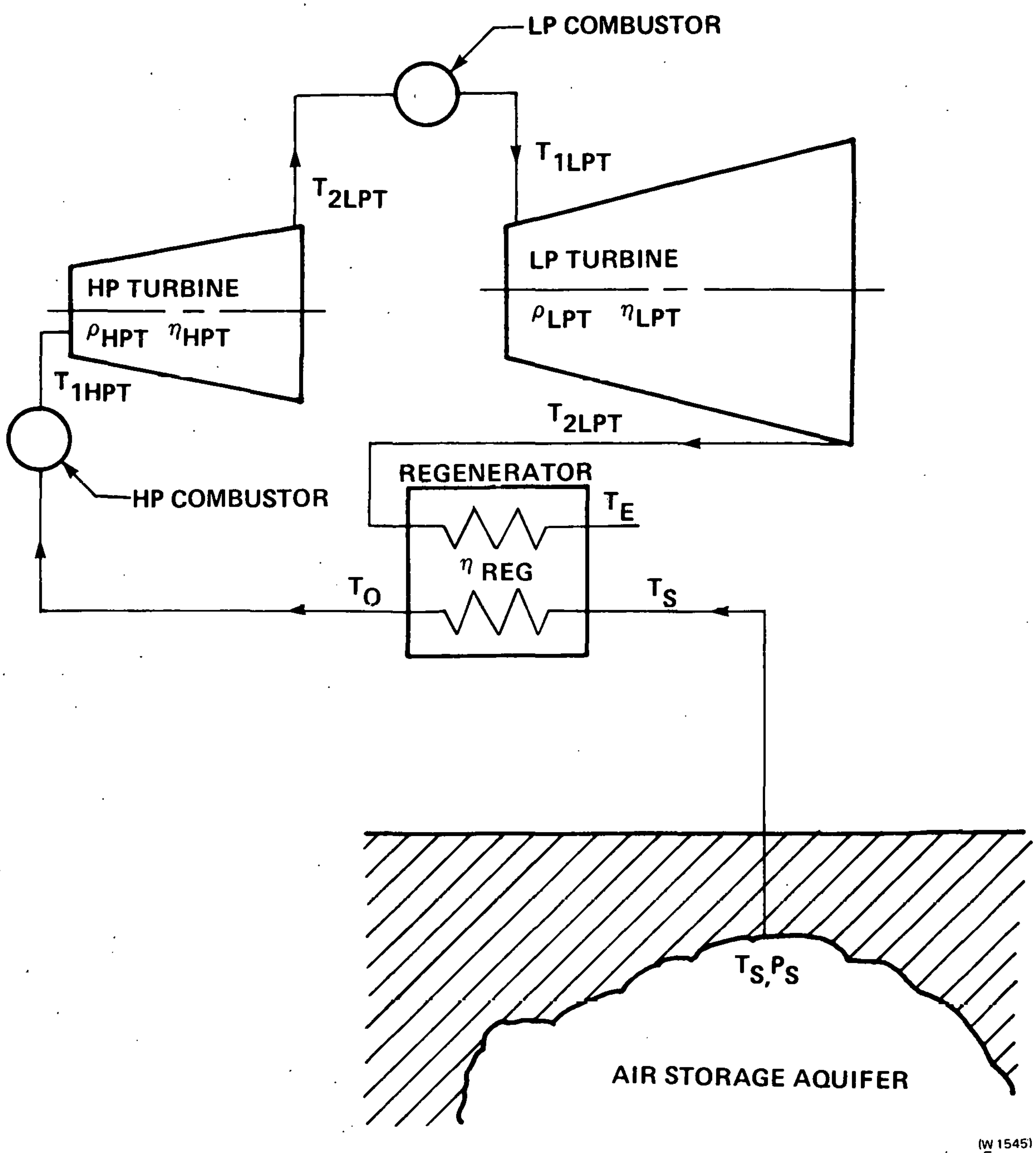

Figure 5-1. Schematic of a CAES Turbine Train With Reheat

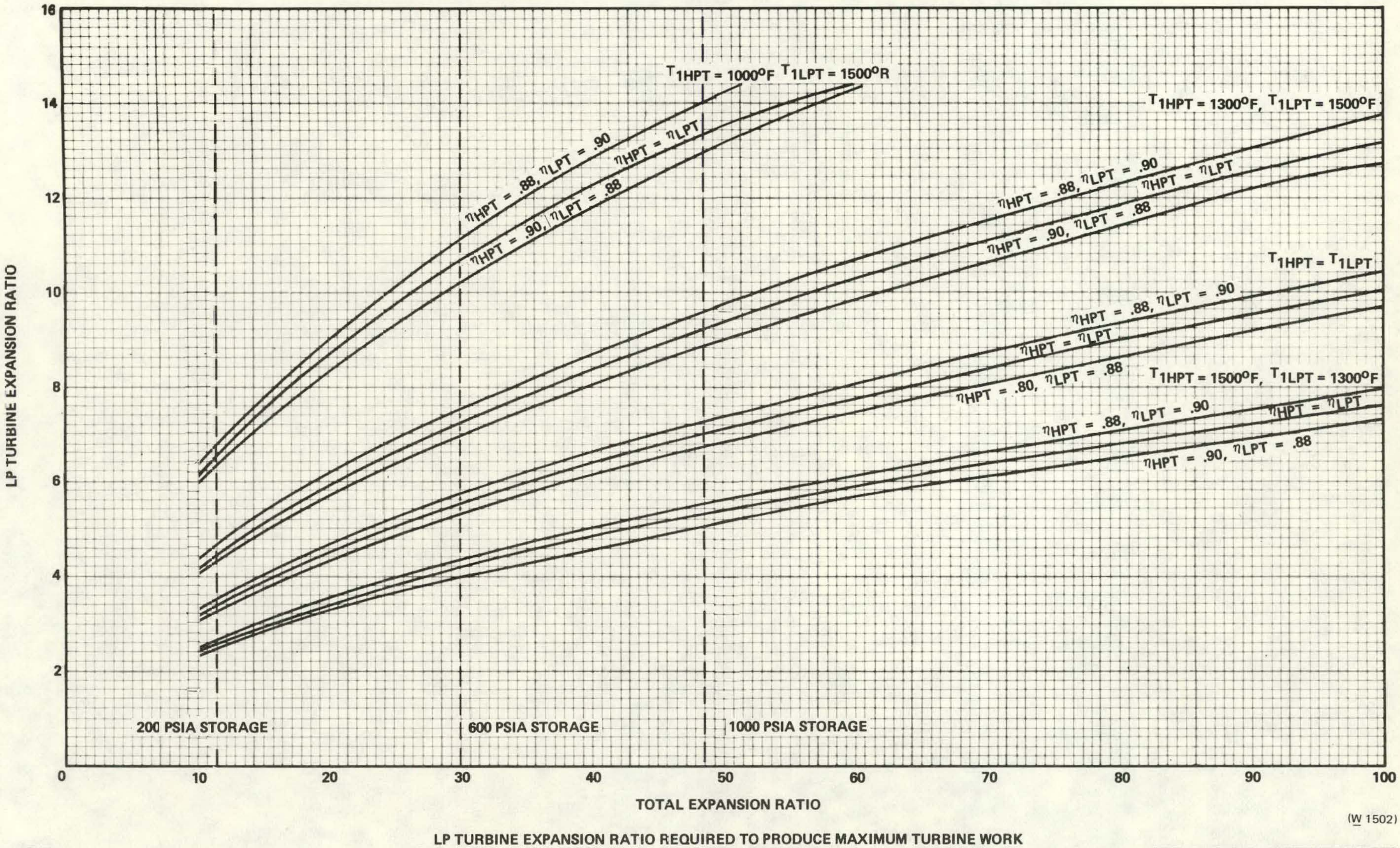

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Figure 5-2. LP Turbine Pressure Ratio Required To Produce Maximum Turbine Work

The curves indicate that as the efficiency of one of the turbines increases the required pressure ratio of that turbine must increase to maintain a maximum work system. By using projected turbine efficiencies, the curves presented in Figure 5-2 identify the turbine pressure ratios required for a maximum turbine work system.

With respect to turbine inlet temperature, Figure 5-2 indicates that as the ratio of the low pressure turbine inlet temperature to the high pressure turbine inlet temperature increases, the required low pressure turbine pressure ratio must increase to maintain a maximum turbine work cycle. In actual operation, the system would automatically tend to adjust in the required direction.

The curves also show that the maximum pressure ratio required by any of the maximum turbine work systems is less than 15 to 1. This requirement does not exceed the present state of the art for turbo-machinery.

However, it must be noted that a CAES turbine train which produces the maximum amount of work is not necessarily the system with the lowest power production energy costs. Unlike the charging cycle, a CAES discharge cycle can not be optimized simply by independently maximizing the total turbine output. The optimum reheat location—the turbine pressure ratio split, which leads to the minimum total power production cost—had to be determined by parametric performance studies where the LP turbine pressure ratio was varied. A complete discussion of this optimizing analysis is presented in Section 6.

To be able to perform the CAES turbine cycle parametric study the amount of cooling air flow and the value of the corresponding turbine efficiency were needed for all of the turbine elements analyzed. For the uncooled turbine disk cases, which included all of the high pressure (HP) turbines, except those with a 2150°F HP turbine inlet temperature, and all of the uncooled low pressure (LP) turbines, the cooling air flow per turbine was initially assumed to be 2% of the aquifer discharge flow. Later, Westinghouse performed a study which indicated large LP turbines with a mass flow rate of approximately 770 lb/sec and all HP turbines with an inlet temperature of 1500°F or less could be manufactured so as to require a cooling flow of 1.9% of the aquifer discharge flow. The turbine efficiency, η, for these cases was found using the following formula.

$$\eta = \frac{1 - \left(\frac{1.03}{\rho}\right)^{.89\left(\frac{K-1}{K}\right)}}{1 - (\rho)^{\frac{1-K}{K}}}$$

where:

η = the turbine efficiency

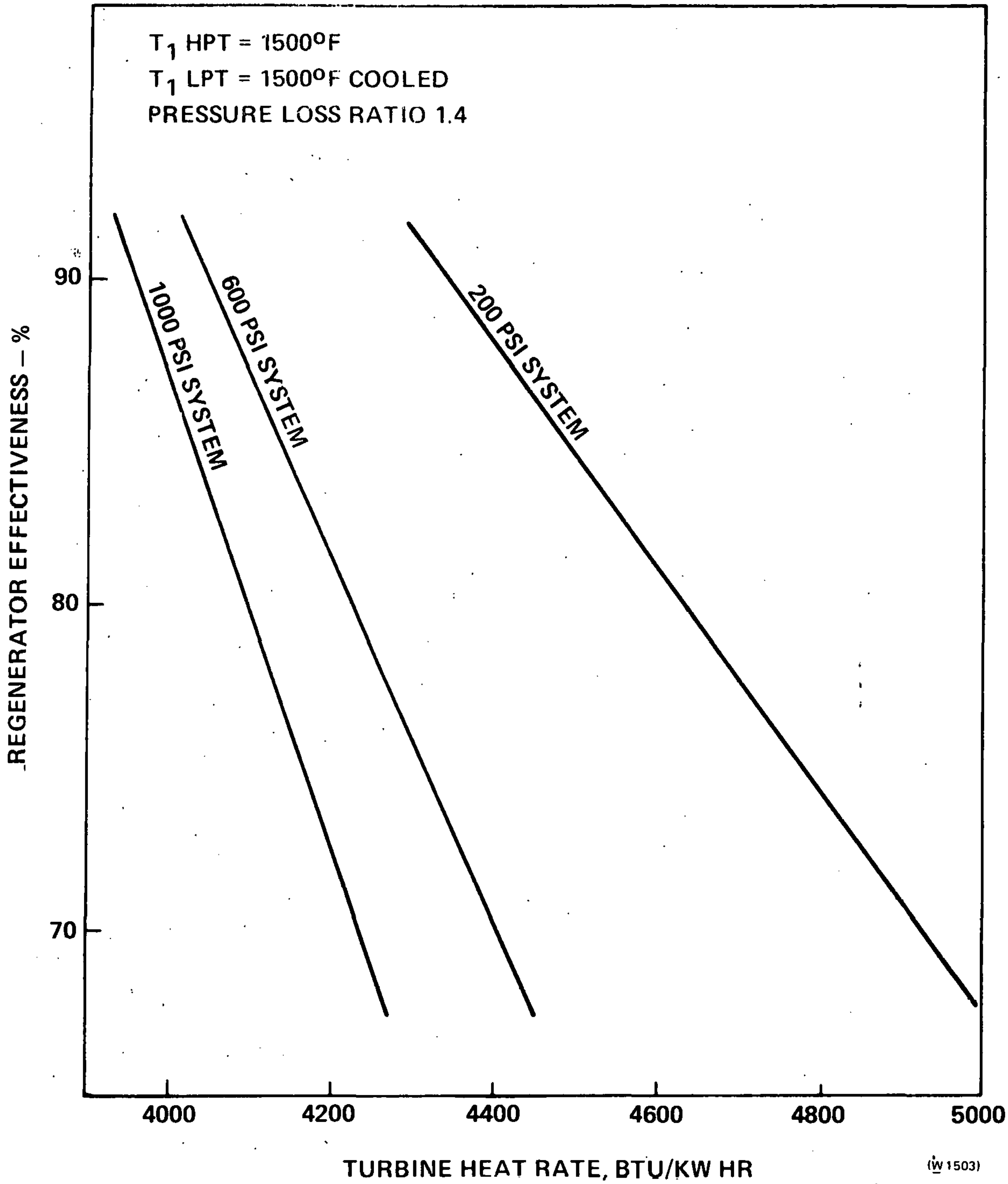
ρ = Turbine pressure ratio

K = Cp/Cv the ratio of the specific heats (based on the turbine inlet temperature)

For all of the turbine elements with a 2150°F inlet temperature both the turbine disk and blades must be cooled. The base efficiency and required cooling air flow for such a turbine was obtained from calculations made for the W501-D turbine at its base pressure ratio. The cooling flow was assumed to remain constant as the turbine pressure ratio was varied. The turbine efficiencies for pressure ratios other than the base pressure ratio were assumed to be proportional to the efficiencies obtained from the above equation.

Low pressure turbines with firing temperatures of 1500°F or lower, assumed to require disk cooling, were considered in the following manner. The performance of the W501-AA was used to establish the base efficiency at its base pressure ratio, and the corresponding disk cooling flow. The required cooling flow was assumed to remain constant as the turbine inlet temperature decreased below 1500°F and/or the pressure ratio varied from the W501-AA base pressure ratio; while the turbine efficiencies were assumed to vary in proportion to the efficiencies predicted by the above equation.

Data resulting from the Westinghouse CAES performance parametric study was used to plot Figure 5-3 which is designed to determine the power production energy cost saving realized by utilizing a more effective regenerator. For a given CAES cycle, as the regenerator effectivity is increased, the only term that changes magnitude in the power production energy cost equation (explained in full detail in Section 6) is the heat rate. The effect on the total power production energy cost resulting from a given change of a regenerator effectivity, can be determined by calculating the net difference in the resulting heat rates and multiplying that value by the cost of the fuel burned.



Turbine heat rate values are as calculated and do not include any margins.

Figure 5-3. Regenerator Effectiveness vs Heat Rate

Section 6

HEAT CYCLE SELECTION

TOTAL POWER PRODUCTION ENERGY COST COMPARISON

In this section the process used to reduce the initial group of proposed cycles to the selected three CAES heat cycles is documented.

Constraints on certain variables were necessary to insure that the cycles analyzed would be feasible using present day technology. As a result the inlet temperature to the LP (reheat) combustor, which is the HP turbine exhaust temperature, was limited to 1100°F to allow conventional cooling of the reheat combustor. The LP turbine exhaust temperature was limited to 1200°F due to mechanical considerations for the last stage of blading and the exhaust manifold.

The last constraint deals with the regenerator which is assumed to be manufactured with approximately 30% of the total cold end surface being corrosion resistant low alloy steel. The regenerator minimum cold end average temperature (flue gas exit temperature + expansion cycle equipment inlet temperature /2) had to be 205°F or higher.

Within the limits of the above constraints Westinghouse performed a parametric study investigating all of the regenerative and regenerative/reheat turbine cycle variables initially selected for Task 1 work. During the parametric study a number of other variables which warranted consideration were also investigated.

For reasons discussed later in this section, Westinghouse evaluated all of the investigated CAES cycles on the basis of total power production energy costs.

The equation used to calculate the total power production energy cost for the CAES power cycle was:

$$(\text{Heat Rate}) \times \left(\frac{\text{Fuel Oil Cost}}{1000}\right) + \left(\frac{\text{Compressor Work}}{\text{Turbine Work}}\right)(\text{Surplus Electrical Power Cost})$$

$$= \text{Total Power Production Energy Costs}$$

where the following units apply:

Variable	Units
Heat Rate	BTU/KWHR
Fuel Oil Cost	$/Million BTU
Compressor Work	KW/Lb/Sec
Turbine Work	KW/Lb/Sec
Surplus Electrical Power Cost	Mils/KWHR
Total Power Production Energy Cost	Mils/KWHR

The results presented in Appendix C reflect the total power production energy costs for the following eight energy cost combinations (which result in seven different energy cost ratios):

Fuel Oil Cost $/Million BTU (Based on a LHV of 18,055 BTU/Lb)	Surplus Electrical Power Cost Mils/KWHR	Energy Cost Ratio
2.50	10	.2500
2.50	20	.1250
4.25	15.10	.2815
5.00	17.06	.2931
5.00	20	.2500
7.50	10	.7500
7.50	20	.3750
7.75	60	.1292

Key cycle parameters for the various CAES cycles investigated are also presented in the Appendix C tables. The first three pages of Appendix C contain a general description of the nomenclature, units, etc. used in the tables plus a Data Index Chart summarizing the tables presented.

During the parametric study the following guidelines were used.

(1) Optimum (minimum work) axial compressor trains utilizing one or two intercoolers were assumed to pressurize the aquifer. (See Table 6-1.)

Table 6-1

MINIMUM WORK (OPTIMIZED) AXIAL COMPRESSOR TRAINS

System Parameters	200 PSI		600 PSI		1000 PSI	
	1 I/C	2 I/C	1 I/C	2 I/C	1 I/C	2 I/C
Aftercooler Discharge Pressure, PSIA	250	250	700	700	1150	1150
Intercooler Outlet Air Temperature, °F	105	105	105	105	105	105
LP Compressor Pressure Ratio/ Efficiency, %	5.09 88	3.38 87	8.43 88	4.77 88	10.80 88	5.47 88
IP Compressor Pressure Ratio/ Efficiency, %	- -	2.40 86	- -	3.39 87	- -	4.05 88
HP Compressor Pressure Ratio/ Efficiency, %	3.69 87.5	2.40 86	6.23 88	3.39 87	7.99 88	4.05 88
Total Work of Compression, KW/LB/SEC	171.4	166.2	249.8	233.4	291.7	266.7

(W 1590)

I/C = Intercooler

The total Work of Compression values are as calculated and do not include any margins.

(2) The reheat location needed to produce maximum turbine work was known, due to prior CAES cycle analyses, to be near the optimum reheat location, therefore initially the maximum work reheat location was employed. The maximum work reheat location was calculated using the equation presented in Section 5.

(3) Once all the proposed CAES cycles were analyzed using the above guidelines, the cases showing the greatest potential were further analyzed. The "maximum work" reheat location used in these high potential cases was fine tuned to determine the LP turbine pressure ratios that produced the lowest possible total power production energy costs.

When the LP turbine pressure ratio was increased above the ratio required to produce the maximum turbine work, the turbine output and the heat rate decreased.

The effects on the total power production energy costs were more complex. For some CAES turbine cycles like the 1500-2150, 2150-1500, and the 2150-2150 cases, the power production energy cost optimum reheat location was not physically feasible because of the 1100°F HP turbine exhaust temperature limitation. In other cases, all of the energy cost combinations investigated, required the same reheat location for optimum total power production energy costs. For the above mentioned cases, just the data for the cycle with the best possible performance is included in the tables found in Appendix C. Thus, data reflecting different reheat location was only presented for the 1500-1500 cases.

In these turbine cycles, the optimum power production energy cost reheat location depended on the energy cost ratio. For such cases the data resulting from these investigations is presented in Tables C-4, C-6, C-8, C-10, C-16 and C-18 of Appendix C. In these tables the LP turbine pressure ratio is listed in the "CASE" row.

It should be noted that the energy cost ratio — not the magnitude of the various energy costs — is the most important factor to consider when optimizing a CAES cycle. The following trends will occur as the energy cost ratio is varied.

(1) As the cost of the fuel relative to the cost of the surplus electrical power increases (as the energy cost ratio increases) the cycles with lower heat rates are favored.

(2) As the cost of the surplus electrical power relative to the cost of fuel increases (as the energy cost ratio decreases) the CAES turbine with more turbine work is favored.

The above trends are valid not only for the vanes turbine inlet temperature cases studied, but also for comparing the CAES turbine trains where the reheat location is varied from the maximum work location to determine the optimum power production energy cost LP turbine pressure ratio. Since as the LP turbine pressure ratio decreases the heat rate increases, the optimum LP turbine pressure ratio will always be equal to or greater than the ratio required to produce maximum turbine work.

During the parametric study, Westinghouse investigated trains utilizing both cooled (these results are presented in Tables C-1 through C-10) and uncooled (these results are presented in Tables C-11 through C-19) turbine elements. Generally, the uncooled turbine trains offered better performance.

Designed to present an overview of all the results of the Westinghouse parametric study, Figures 6-1 through 6-6 illustrate the total power production energy cost of the best performers of the various CAES cycles investigated. Since some of the energy cost combinations had equal or nearly equal energy cost ratios, only the following four cost combinations were included in the above figures:

Energy Cost Combinations	Energy Cost Ratio
$2.50-20	.125
$2.50-10	.250
$7.50-20	.375
$7.50-10	.750

In these figures the various cycles were plotted in the order of increasing heat rate. Since the reheat turbine train cycles with a 2150°F HP turbine inlet temperature did not have favorable cost performance characteristics and were not originally included in the official list of the various cycles to be investigated during the parametric study, their results were not included in the above figures (although their results are presented in Appendix C). The cycles with the lowest total power production energy costs are indicated by squares enclosing the turbine inlet temperatures.

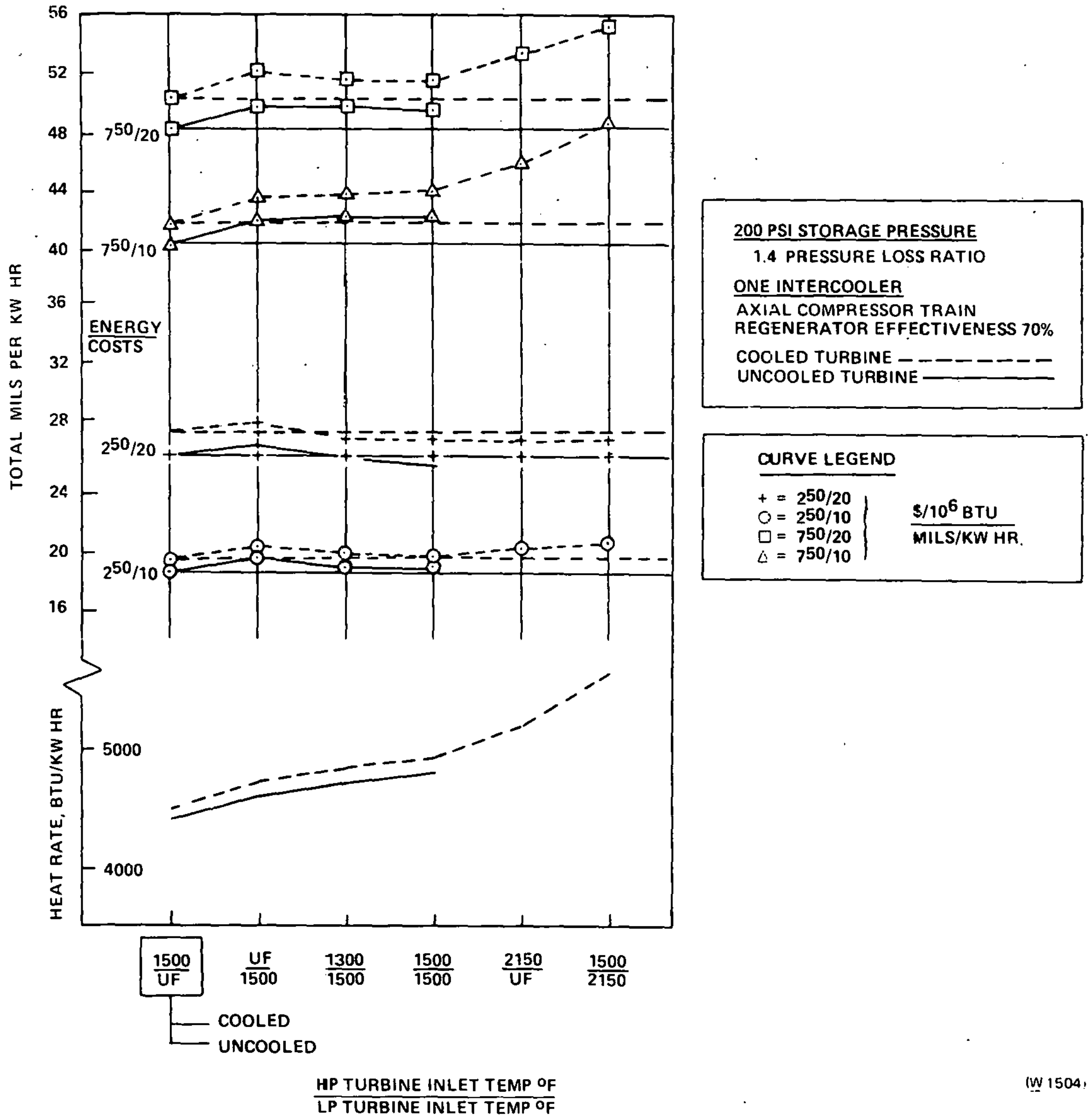
Figures 6-1 through 6-6 graphically illustrate the following points:

(1) The use of a second intercooler in the compressor trains significantly reduced the total power production energy cost of the three CAES systems, especially when the price of the surplus electrical energy was high. Also, the savings realized by using the second intercooler increased as the storage pressure increased.

(2) Except for the 200 PSIA nominal storage pressure case where the optimum reheat location could not be used because of the 1100°F HP turbine exhaust temperature limitation, the reheat CAES cycles were overall better power production energy cost performers than the cycles not utilizing reheat.

(3) Except for the 200 PSIA storage pressure case which could not properly utilize reheat, the overall best power production energy cost cycles did <u>not</u> correspond to the cycles with the lowest heat rate.

(4) As the price of the fuel relative to the cost of the surplus electrical power increased (as the energy cost ratio increased), the cycles with a lower heat rate were favored (see the $7.50/10 results).



NOTE: The above performance values are as calculated and do not include any margins.

Figure 6-1. Total Energy Cost Comparison

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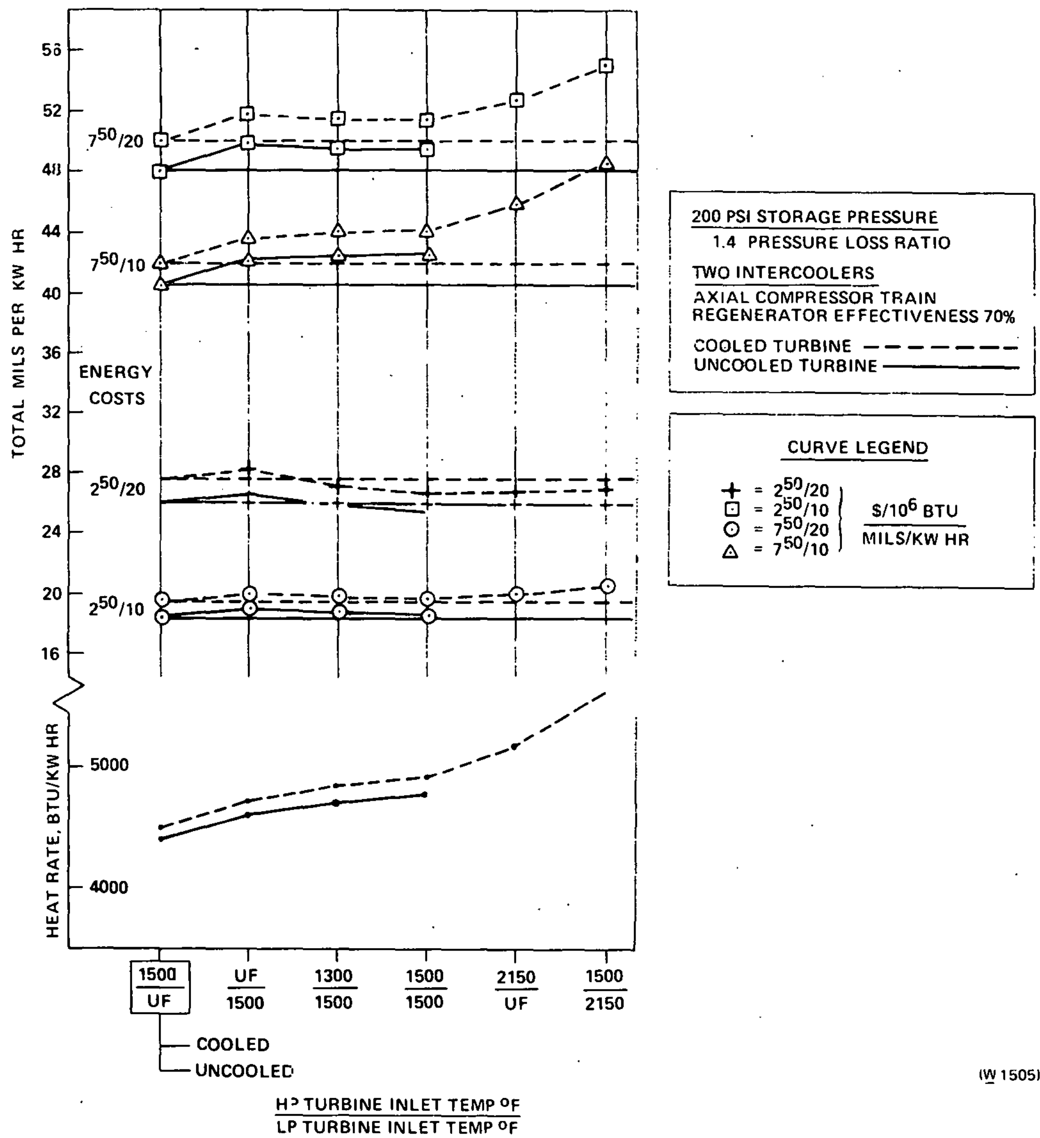

NOTE: The above performance values are as calculated and do not include any margins.

Figure 6-2. Total Energy Cost Comparison

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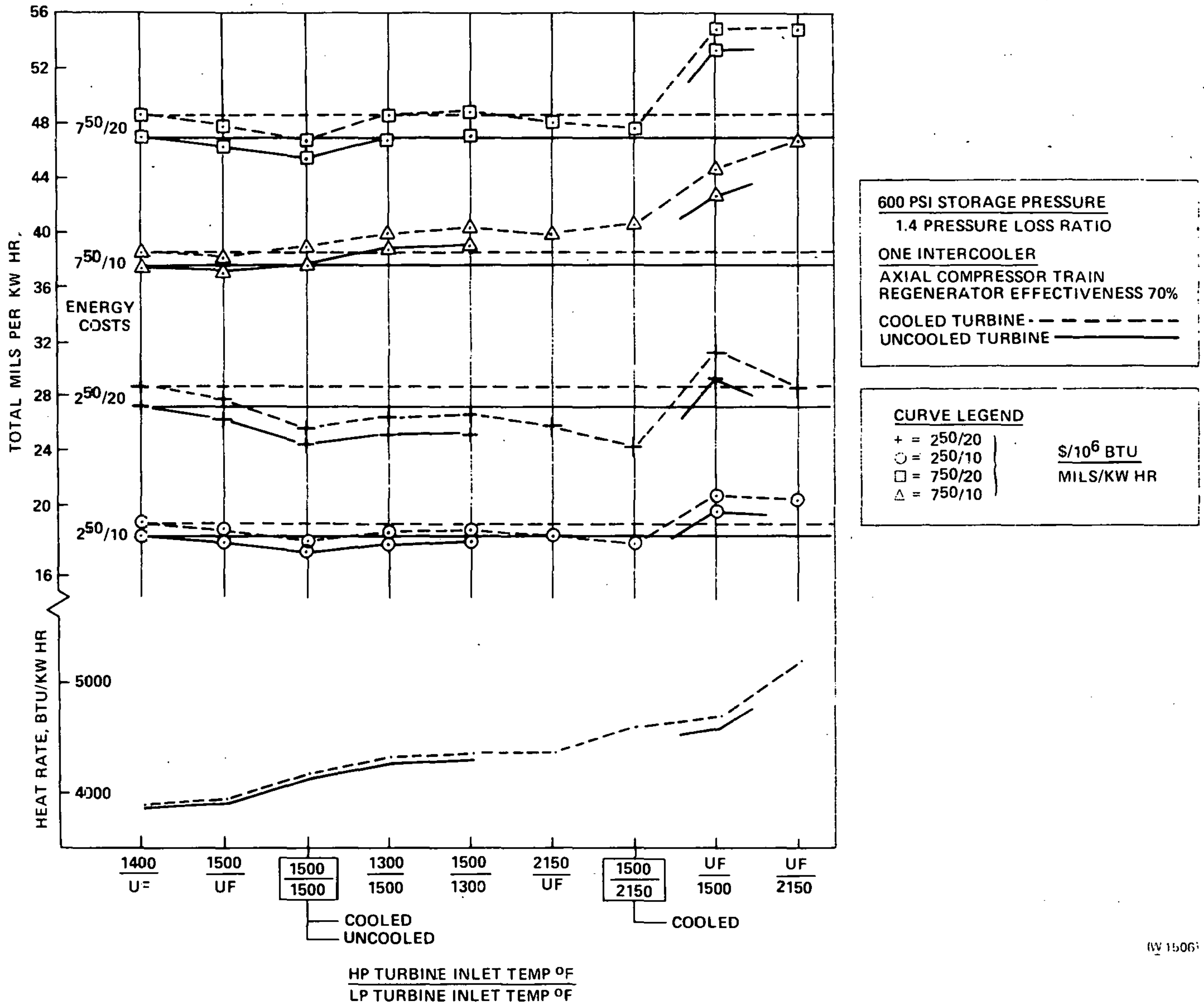

NOTE: The above performance values are as calculated and do not include any margins.

Figure 6-3. Total Energy Cost Comparison

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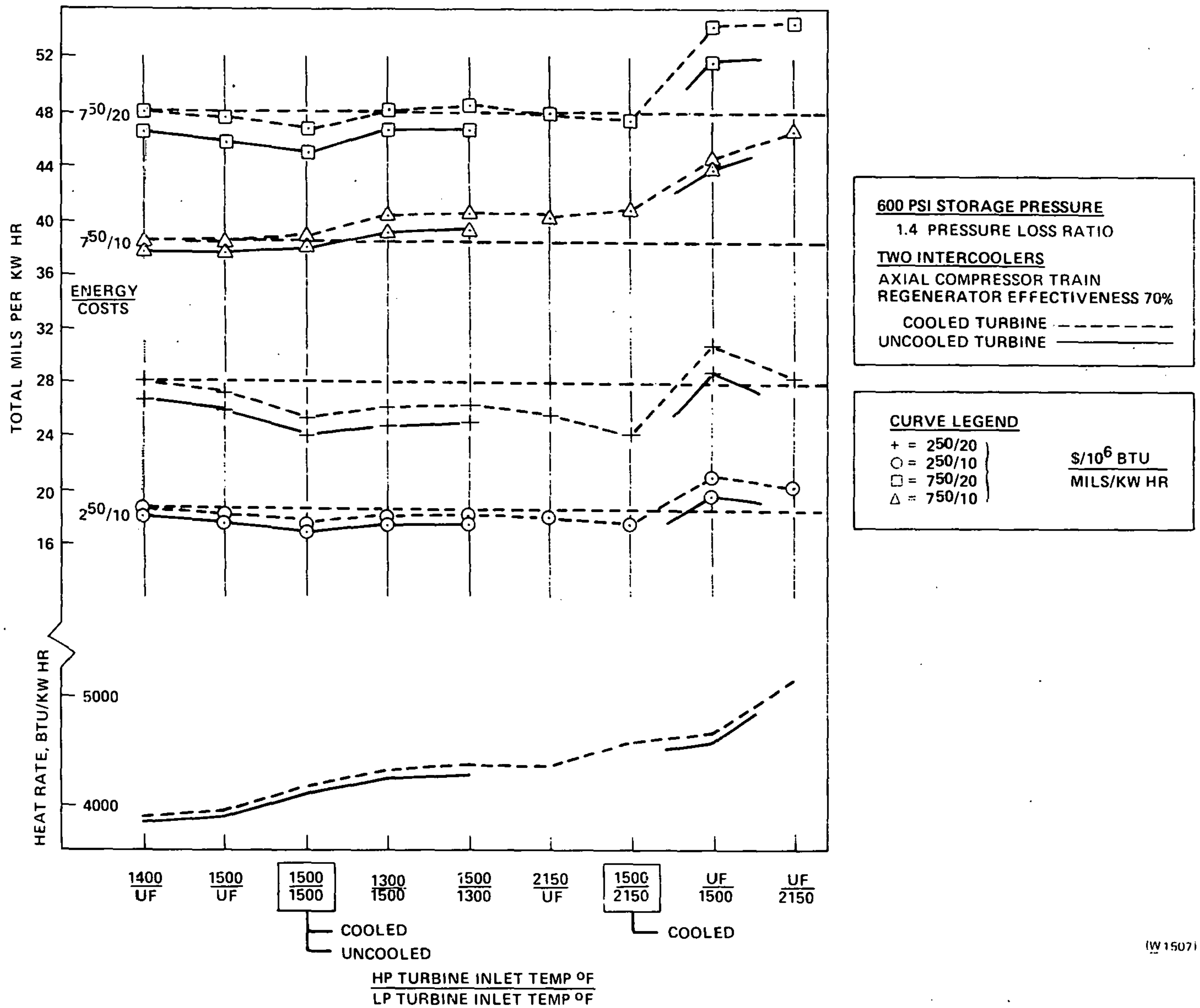

NOTE: The above performance values are as calculated and do not include any margins.

Figure 6-4. Total Energy Cost Comparison

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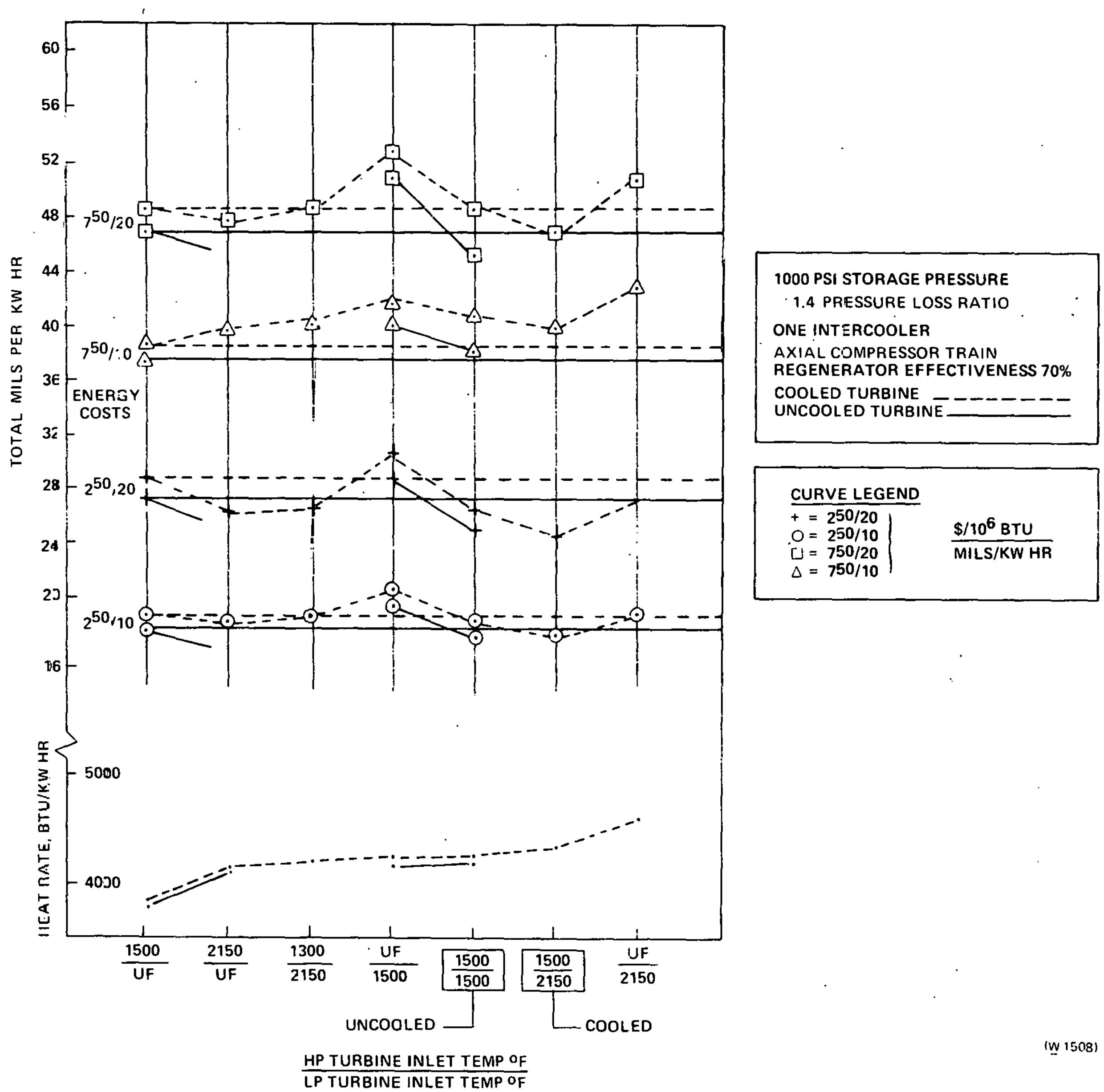
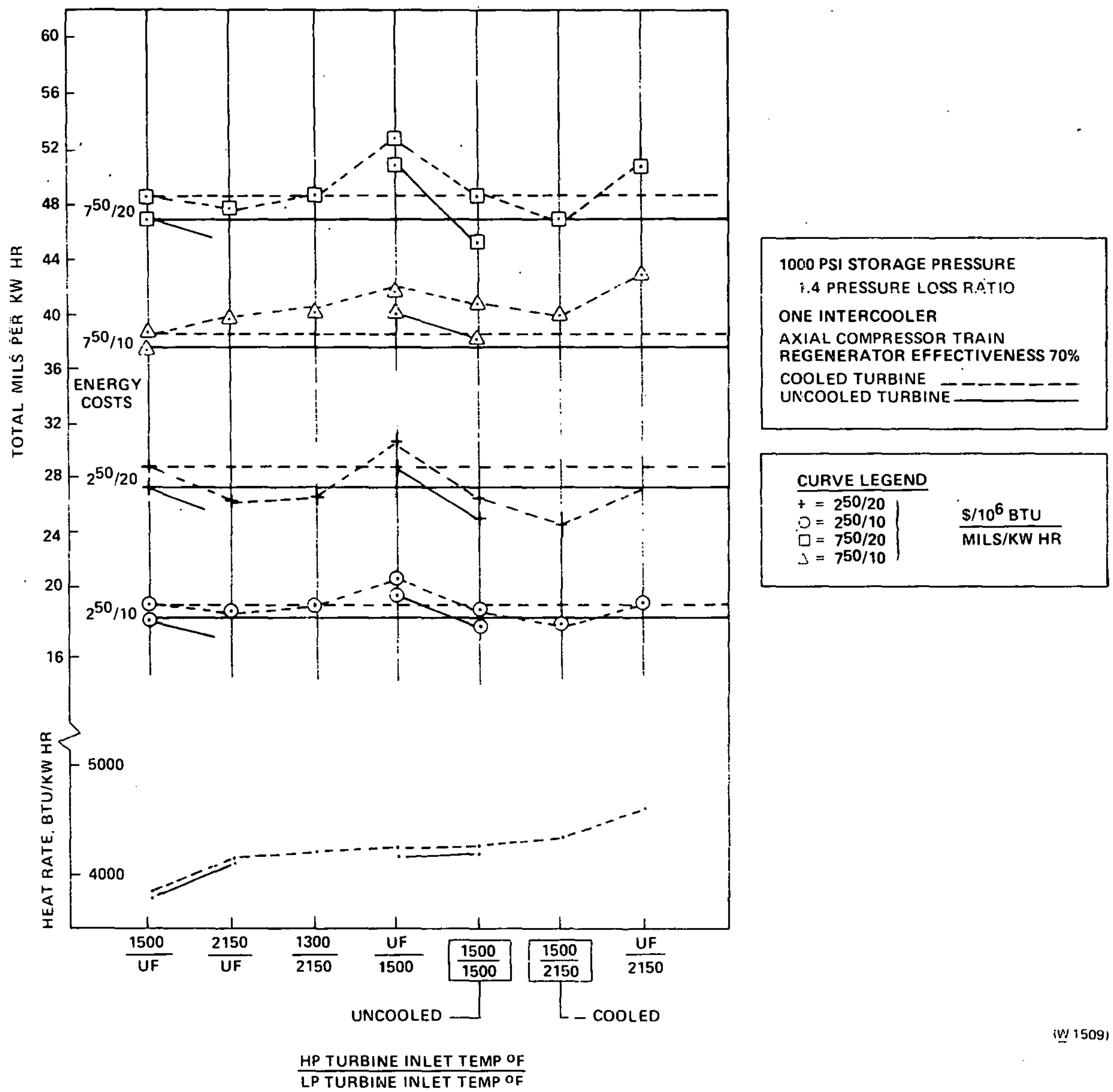


NOTE: The above performance values are as calculated and do not include any margins.

Figure 6-5. Total Energy Cost Comparison



NOTE: The above performance values are as calculated and do not include any margins.

Figure 6-6. Total Energy Cost Comparison

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(5) As the price of the surplus electrical power relative to the cost of the fuel increased (as the energy cost ratio decreased), the CAES turbine cycles with more turbine work were favored (see the \$2.50/20 results).

The cooled and uncooled cycles having the lowest total energy cost for both one and two intercooler trains, at each pressure level, are summarized on Tables 6-2 through 6-4. Based on the results presented in these tables, the following cycles were initially selected for further evaluation.

System Nominal Storage Pressure, PSI	Compressor Train No. of Intercoolers	Turbine Firing Temperature HPT/LPT ^{o}F
200	2	1500/UNFIRED
600	2	1500/1500
1000	2	1500/1500
1000	2	1500/2150

During the parametric study the 1500-1500^{o}F HP-LP turbine inlet temperature cases were rerun at storage pressure loss ratios, from the aftercooler discharge to the aquifer, or from the aquifer to regenerator inlet, of 1.1 to 1.5. In these calculations a storage pressure loss ratio of 1.1 is equivalent to a system storage pressure loss ratio of (1.1 x 1.1) 1.21. The base storage pressure loss ratio used in the majority of the calculations was 1.20. Since varying the storage pressure loss ratio effects both the turbine output and the heat rate, the effect of increasing the storage pressure loss ratio appears as an increase in the total power production energy cost as indicated in the Appendix C tables.

Data resulting from the parametric study was also used to plot Figure 6-7 (presented earlier in Section 5) which is designed to determine the power production energy cost saving realized by utilizing a more effective regenerator. For a given CAES cycle, as the regenerator effectivity is increased, the only term that changes magnitude in the power production energy cost equation is the heat rate. The effect caused by a change in the regenerator effectivity can be determined by calculating the net difference in the resulting heat rates and multiplying that value by the cost of the fuel burned.

As stated earlier, the following trends were observed in Figure 6-7. The steeper the curves in Figure 6-7, the lower the LP turbine exhaust temperature and the

Table 6-2

CYCLE COMPARISON SUMMARY

200 PSI STORAGE PRESSURE
REGENERATOR EFFECTIVITY 70%

		ONE INTERCOOLER		TWO INTERCOOLERS	
		UNCOOLED	COOLED	UNCOOLED	COOLED
Firing Temperature HPT/LPT °F		1500/UF	1500/UF	1500/UF	1500/UF
Turbine Work – KW/LB/SEC		220	204	220	204
Air Flow – LB/SEC/Machinery Set		450	800	450	800
KW Per Machinery Set		99,000	163,200	99,000	163,200
Machinery Sets Per 1000 MWe (EXACT)		10.1	6.13	10.1	6.13
Machinery Sets Per 1000 MWe (ACTUAL)		10	6	10	6
Units Per Machinery Set					
Turbines		2	2	2	2
Compressors		2	2	3	3
Coolers		2	2	3	3
Gear Boxes		2	2	3	3
Units Per 1000 MWe					
Turbines		20	12	20	12
Compressors		20	12	30	18
Coolers		20	12	30	18
Gear Boxes		20	12	30	18
Total Mils/KW HR					
Fuel @ $/10^6 BTU/ELEC PWR	2.50/10	18.81	19.65	18.57	19.39
Mils/KW HR	2.50/20	26.61	28.04	26.13	27.53
	5.00/20	37.61	39.29	37.13	38.78
	7.50/10	40.81	42.15	40.57	41.89
	7.50/20	48.61	50.55	48.13	50.03
	7.75/60	80.93	85.25	79.49	83.71

(W 4145)

For a 1000 MWe Plant Operating 10 HR/Day, 6 Days/Week, 1 Mil/KW HR = $3.12 x 10^6/YR.

NOTE: The above performance values are as calculated and do not include any margins.

Table 6-3

CYCLE COMPARISON SUMMARY

600 PSI STORAGE PRESSURE
REGENERATOR EFFECTIVITY 70%

	ONE INTERCOOLER			TWO INTERCOOLERS		
	UNCOOLED	COOLED	COOLED	UNCOOLED	COOLED	COOLED
Firing Temperature HPT/LPT °F	1500/1500	1500/1500	1500/2150	1500/1500	1500/1500	1500/2150
Turbine Work – KW/LB/SEC*	332	312	362	332	312	362
Air Flow – LB/SEC/Machinery Set	450	800	800	450	800	800
KW Per Machinery Set	149,400	249,600	289,600	149,400	249,600	289,600
Machinery Sets Per 1000 MWe (EXACT)	6.69	4.01	3.45	6.69	4.01	3.45
Machinery Sets Per 1000 MWe (ACTUAL)	7	4	4	7	4	4
Units Per Machinery Set						
Turbines	2	2	2	2	2	2
Compressors	2	2	2	3	3	3
Coolers	2	2	2	3	3	3
Gear Boxes	2	2	2	3	3	3
Units Per 1000 MWe						
Turbines	14	8	8	14	8	8
Compressors	14	8	8	21	12	12
Coolers	14	8	8	21	12	12
Gear Boxes	14	8	8	21	12	12
Total Mils/KW HR**						
Fuel @ $/10^6 BTU/ELEC PWR 2.50/10	17.87	18.58	18.35	17.37	18.05	17.90
Mils/KW HR 2.50/20	25.40	26.61	25.25	24.41	25.57	24.35
5.00/20	35.73	37.15	36.70	34.74	36.09	35.80
7.50/10	38.53	39.54	41.25	38.03	39.01	40.80
7.50/20	46.06	47.64	48.15	45.07	46.57	47.25
7.75/60	77.23	80.91	76.90	74.26	77.78	74.19

For a 1000 MWe Plant Operating 10 HR/Day, 6 Days/Week, 1 Mil/KW HR = $3.12 x 10^6/YR. (W 4148)

*For the 1500/1500 cases, the turbine work is based on an average of the optimum reheat location results.

**For the 1500/1500 cases, the above power production energy costs reflect the fine tuned optimum reheat location for each of the fuel cost combinations.

NOTE: The above performance values are as calculated and do not include any margins.

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Table 6-4

CYCLE COMPARISON SUMMARY

1000 PSI STORAGE PRESSURE
REGENERATOR EFFECTIVITY 70%

	ONE INTERCOOLER		TWO INTERCOOLERS		
	UNCOOLED	COOLED	UNCOOLED	COOLED	COOLED
Firing Temperature HPT/LPT °F	1500/1500	1500/2150	1500/1500	1500/2150	1500/1500
Turbine Work – KW/LB/SEC*	371	402	371	402	344
Air Flow – LB/SEC/Machinery Set	450	800	450	800	800
KW Per Machinery Set	166,950	321,600	166,950	321,600	275,200
Machinery Sets Per 1000 MWe (EXACT)	5.99	3.11	5.99	3.11	3.63
Machinery Sets Per 1000 MWe (ACTUAL)	6	3	6	3	4
Units Per Machinery Set					
Turbines	2	2	2	2	2
Compressors	2	2	3	3	3
Coolers	2	2	3	3	3
Gear Boxes	2	2	3	3	3
Units Per 1000 MWe					
Turbines	12	6	12	6	8
Compressors	12	6	18	9	12
Coolers	12	6	18	9	12
Gear Boxes	12	6	18	9	12
Total Mils/KW HR**					
Fuel @ $/10⁶ BTU/ELEC PWR 2.50/10	17.89	18.03	17.22	17.41	17.89
Mils/KW HR 2.50/20	25.66	25.29	24.34	24.05	25.53
5.00/20	35.79	36.06	34.44	34.82	35.79
7.50/10	37.75	39.58	37.06	38.96	38.01
7.50/20	45.76	46.84	44.39	45.60	45.85
7.75/60	78.02	76.95	74.02	73.22	77.63

For a 1000 MWe Plant Operating 10 HR/Day, 6 Days/Week, 1 Mil/KW HR = $3.12 x 10^6/YR. (W 4151)

*For the 1500/1500 cases, the turbine work is based on an average of the optimum reheat location results.

**For the 1500/1500 cases, the above power production energy costs reflect the fine tuned optimum reheat location for each of the fuel cost combinations.

NOTE: The above performance values are as calculated and do not include any margins.

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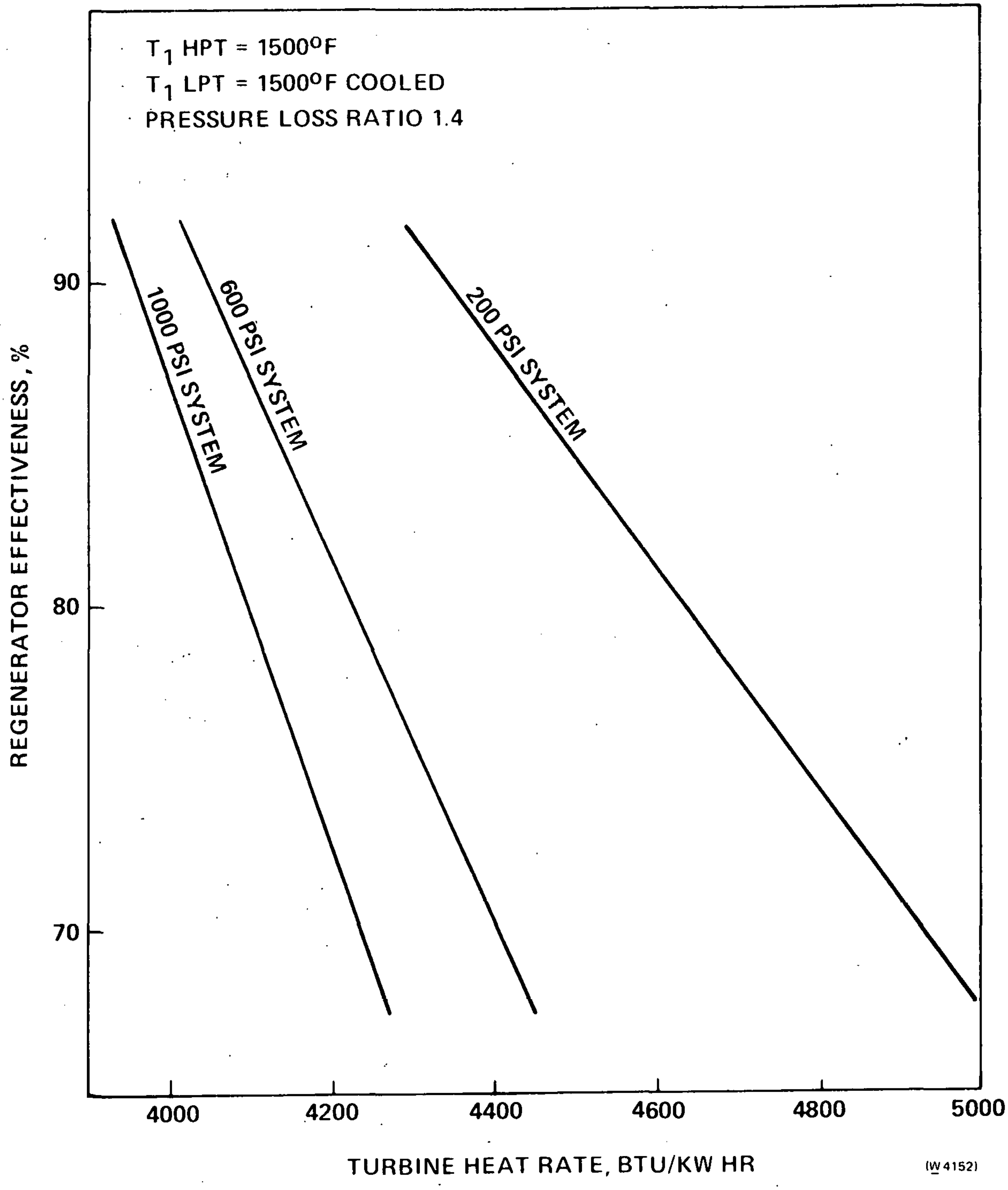


NOTE: The above turbine heat rate values are as calculated and do not include any margins.

Figure 6-7. Regenerator Effectiveness vs Heat Rate lower the return on a more effective (more expensive) regenerator will be. Since all three curves in Figure 6-7 show substantial potential for reducing the power production energy cost, it was later decided that an 85% effective regenerator was justified.

The results presented in Table C-18 of Appendix C reflect the effects of decreasing the HP turbine inlet temperature while maintaining a 1500°F LP turbine inlet temperature. Both a compressor train utilizing a single intercooler (the .11, .21, .31 and .41 cases) and a compressor train utilizing two intercoolers (cases .12, .22, .32 and .42) were investigated. As can be seen, as the HP firing temperature was decreased, the heat rate <u>decreased</u> but the total power production energy cost for all of the energy cost combinations <u>increased</u>. Thus, based on these results, it is obvious that a CAES system can not be optimized by considering only the heat rate.

At first, the above results seem to contradict the heat rate definition, since for most power plant cycles the heat rate, which is expressed as:

$$\text{Heat Rate} = \frac{\text{BTU of Fuel Energy In}}{\text{KWHR of Electrical Energy Out}}$$

is the reciprocal (after appropriate unit changes) of a cycle's thermal efficiency, η_{THERM}, which is traditionally expressed as:

$$\eta_{THERM} = \frac{\text{Useful Energy Out}}{\text{Energy In}}$$

In the above equation the "useful energy out" term represents the generated electrical energy; and the "energy in" term represents - for conventional power plants - the energy of the fuel consumed to operate the plant.

Thus, for conventional power plants the cycle with the minimum heat rate corresponds to the cycle with the maximum thermal efficiency which in turn reflects the cycle that produces electrical power at the lowest possible total power production energy costs. For a CAES power plant, however, the "energy in" term must include not only the fuel energy, but also the energy utilized in the form of compressor work to pressurize the aquifer. Thus the thermal efficiency for a CAES system, η_{CAES}, is expressed as:

$$\eta_{CAES} = \frac{\text{Turbine Energy Out}}{\text{Fuel Energy In + Compressor Work Energy In}}$$

which is <u>not</u> the reciprocal of the (CAES) heat rate. However, maximizing the CAES thermal efficiency will not guarantee that the resulting cycle will have the lowest total power production energy cost unless the cost — per given unit — of fuel energy and the surplus electrical energy used to drive the compressors are equal. The above energy costs are equal when the price of fuel oil is $5.00 per million BTU and the price of surplus electrical power is 17.06 mils per KWHR resulting in an energy cost ratio of .2931. Since the energy cost ratio is calculated by dividing the fuel cost with units of dollars per million BTU by the surplus electrical power cost with units of mils per KWHR, any energy cost combination such as $2.50/8.53 with an energy cost ratio of .2931 will represent a combination where the price — per given unit — of fuel and surplus electrical energy are equal.

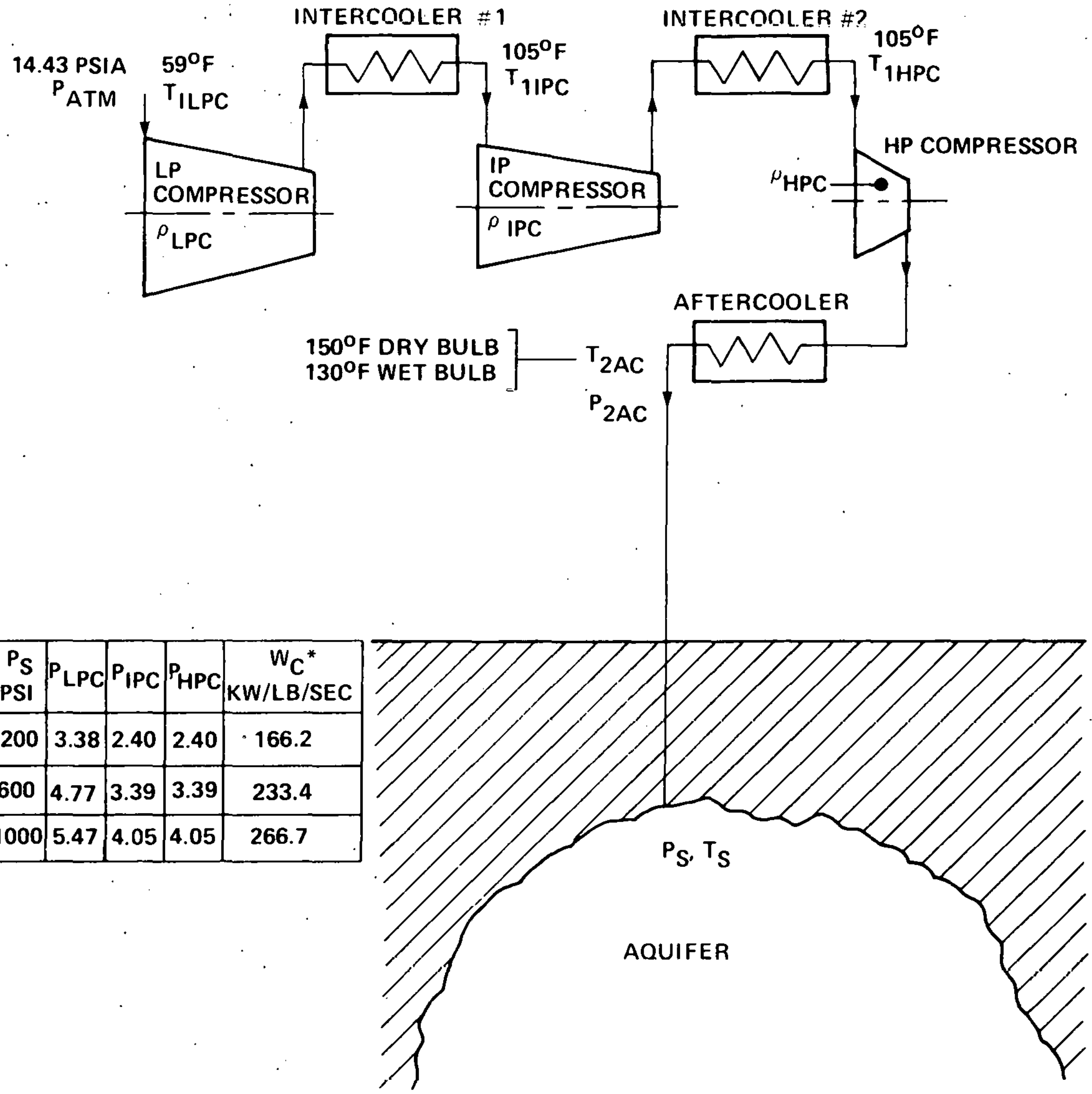
Since all of the proposed energy cost combinations did not have equal fuel and surplus electrical power energy costs, Westinghouse evaluated all of the investigated CAES cycles on the basis of the eight different total power production energy costs presented in Appendix C. The above method also has the advantage that once the projected energy costs are determined, it will be a simple matter to determine the power generation cost competitiveness of the candidate CAES cycles.

Since a reduction in the operating cost of one mil per KWHR is equal to an annual saving of 3.12 million dollars for a 1000 MW plant operating 10 hours per day, six days per week, it was decided by the sponsoring utilities that the optimized, all axial compressor trains utilizing two intercoolers, should be used for further analysis. A summary of the above compressor trains are presented in Figure 6-8.

Westinghouse next performed a study to determine the minimum required cooling flow for a W501 size turbine. The results of the study indicated that a turbine with a mass flow of approximately 770 lbs/sec and an inlet temperature of 1500°F would require a cooling flow of 1.9%.

Table 6-5, Selected Cycle Summary, displays six total power production energy costs and machine capacities for the selected cycles with LP turbine cooling flows of 1.9%, turbine mass flows of 770 lb/sec and with regenerator effectivities of 85%. The key heat cycle parameters and the eight total power production energy costs for the selected cycles are shown in Table 6-6.

The sponsoring utilities selected two heat cycles for the 1000 psi system for additional study. These two cycles were the 1500/1500°F and the 1500/2150°F turbine



P_S PSI	P_{LPC}	P_{IPC}	P_{HPC}	W_C* KW/LB/SEC
200	3.38	2.40	2.40	166.2
600	4.77	3.39	3.39	233.4
1000	5.47	4.05	4.05	266.7

*THE TOTAL WORK OF COMPRESSION VALUES ARE AS CALCULATED AND DO NOT INCLUDE ANY MARGIN.

Figure 6-8. CAES Compression Cycle Selection

Table 6-5

INITIALLY SELECTED OPTIMIZED CYCLE SUMMARY

TWO INTERCOOLER CYCLE
REGENERATOR EFFECTIVITY 85%

	Nominal Storage Pressure			
	200 PSI	600 PSI	1000 PSI	
Firing Temperature HPT/LPT °F	1500/UF	1500/1500	1500/1500	1500/2150
Net Turbine Work – KW/LB/SEC*	218.5	331.1	375.7	400.5
Air Flow – LB/SEC/Machinery Set**	770.0	770.0	770.0	770.0
KW Per Machinery Set*	168,245	254,950	289,290	308,385
Machinery Sets Per 1000 MWe (EXACT)	5.94	3.92	3.46	3.24
Machinery Sets Per 1000 MWe (ACTUAL)	6	4	3	3
Units Per Machinery Set				
Turbines	1	2	2	2
Compressors	3	3	3	3
Cooler Sets	3	3	3	3
Units Per 1000 MWe				
Turbines	6	8	6	6
Compressors	18	12	9	9
Cooler Sets	18	12	9	9
Total Power Production Energy Costs in Mils/KW HR				
Fuel @ $\$/10^6$ BTU/ELEC PWR in Mils/KW HR 2.50/10	17.80	16.84	16.80	16.99
2.50/20	25.40	23.88	23.90	23.65
5.00/20	35.59	33.67	33.60	33.97
7.50/10	38.17	36.41	36.20	37.64
7.50/20	45.78	43.46	43.30	44.30
7.75/60	77.23	72.63	72.67	71.97

(W 1546)

*These Outputs Reflect a 98% Generator Efficiency

**Low Pressure Compressor Inlet Air Flow

The above performance values are as calculated and do not include any margin.

Table 6-6

TOTAL POWER PRODUCTION ENERGY COSTS OF THE FOUR OPTIMIZED CANDIDATE CAES POWER PLANTS

CASE	1.00	2.00	3.00	4.00
T1HPT	1500.	1500.	1500.	1500.
T2HPT	1046.	1110.	948.	1105.
T1LPT	1046.	1500.	1500.	2150.
CAV LOS	1.20	1.20	1.20	1.20
REG EFT	.85	.85	.85	.85
COMP KW	166.2	233.4	266.7	266.7
TURB KW	218.5	331.1	375.7	400.5
H.R.	4075.	3915.	3880.	4130.

FUEL COST = $2.50 PER MILLION BTU
ELECT COST = 10.00 MILS PER KW-HR ENERGY COST RATIO = .2500

10.19	9.79	9.70	10.33
7.61	7.05	7.10	6.66

TOTAL POWER PRODUCTION ENERGY COSTS IN MILS PER KW-HR

17.80	16.84	16.80	16.99

FUEL COST = $2.50 PER MILLION BTU
ELECT COST = 20.00 MILS PER KW-HR ENERGY COST RATIO = .1250

10.19	9.79	9.70	10.33
15.22	14.10	14.20	13.32

TOTAL POWER PRODUCTION ENERGY COSTS IN MILS PER KW-HR

25.40	23.88	23.90	23.65

FUEL COST = $4.25 PER MILLION BTU
ELECT COST = 15.10 MILS PER KW-HR ENERGY COST RATIO = .2815

17.32	16.64	16.49	17.55
11.49	10.64	10.72	10.06

TOTAL POWER PRODUCTION ENERGY COSTS IN MILS PER KW-HR

28.81	27.28	27.21	27.61

FUEL COST = $5.00 PER MILLION BTU
ELECT COST = 17.00 MILS PER KW-HR ENERGY COST RATIO = .2931

20.38	19.58	19.40	20.65
12.98	12.03	12.11	11.36

TOTAL POWER PRODUCTION ENERGY COSTS IN MILS PER KW-HR

33.35	31.60	31.51	32.01

FUEL COST = $6.00 PER MILLION BTU
ELECT COST = 20.00 MILS PER KW-HR ENERGY COST RATIO = .2500

20.38	19.58	19.40	20.65
15.22	14.10	14.20	13.32

TOTAL POWER PRODUCTION ENERGY COSTS IN MILS PER KW-HR

35.59	33.67	33.60	33.97

FUEL COST = $7.50 PER MILLION BTU
ELECT COST = 10.00 MILS PER KW-HR ENERGY COST RATIO = .7500

30.56	29.36	29.10	30.98
7.61	7.05	7.10	6.66

TOTAL POWER PRODUCTION ENERGY COSTS IN MILS PER KW-HR

38.17	36.41	36.20	37.64

FUEL COST = $7.50 PER MILLION BTU
ELECT COST = 20.00 MILS PER KW-HR ENERGY COST RATIO = .3750

30.56	29.36	29.10	30.98
15.22	14.10	14.20	13.32

TOTAL POWER PRODUCTION ENERGY COSTS IN MILS PER KW-HR

45.78	43.46	43.30	44.30

FUEL COST = $7.75 PER MILLION BTU
ELECT COST = 60.00 MILS PER KW-HR ENERGY COST RATIO = .1292

31.58	30.34	30.07	32.01
45.65	42.29	42.60	39.96

TOTAL POWER PRODUCTION ENERGY COSTS IN MILS PER KW-HR

77.23	72.63	72.67	71.07

NOTES:

(1) Case 1.00 has a nominal storage pressure of 200 PSI utilizing its optimized two intercooler compressor train, 1500°F-UF turbine inlet temperature.

(2) Case 2.00 has a nominal storage pressure of 600 PSI utilizing its optimized two intercooler compressor train, 1500-1500°F turbine inlet temperature.

(3) Case 3.00 has a nominal storage pressure of 1000 PSI utilizing its optimized two intercooler compressor train, 1500-1500°F turbine inlet temperature.

(4) Case 4.00 has a nominal storage pressure of 1000 PSI utilizing its optimized two intercooler compressor train, 1500-1500°F turbine inlet temperature.

(W 1547)

The total Work of Compression, Turbine Output Power and Heat Rate values are as calculated and do not include any margin.

inlet temperature cycles shown as Case 3 and Case 4 in Table 6-6. At all fuel oil/electric power pumping cost ratios except two the 1500/1500°F cycle exhibits the lowest power production cost. The two cost ratios where the 1500/2150°F cycle has the lower total power production energy cost are those where the ratio of fuel oil/electric charging power cost is the lowest. These cost ratios are:

Fuel Oil Cost $/Million BTU	Electric Power Charging Cost Mils/KWHR	Energy Cost Ratio
$2.50	20 mils	.1250
$7.75	60 mils	.1292

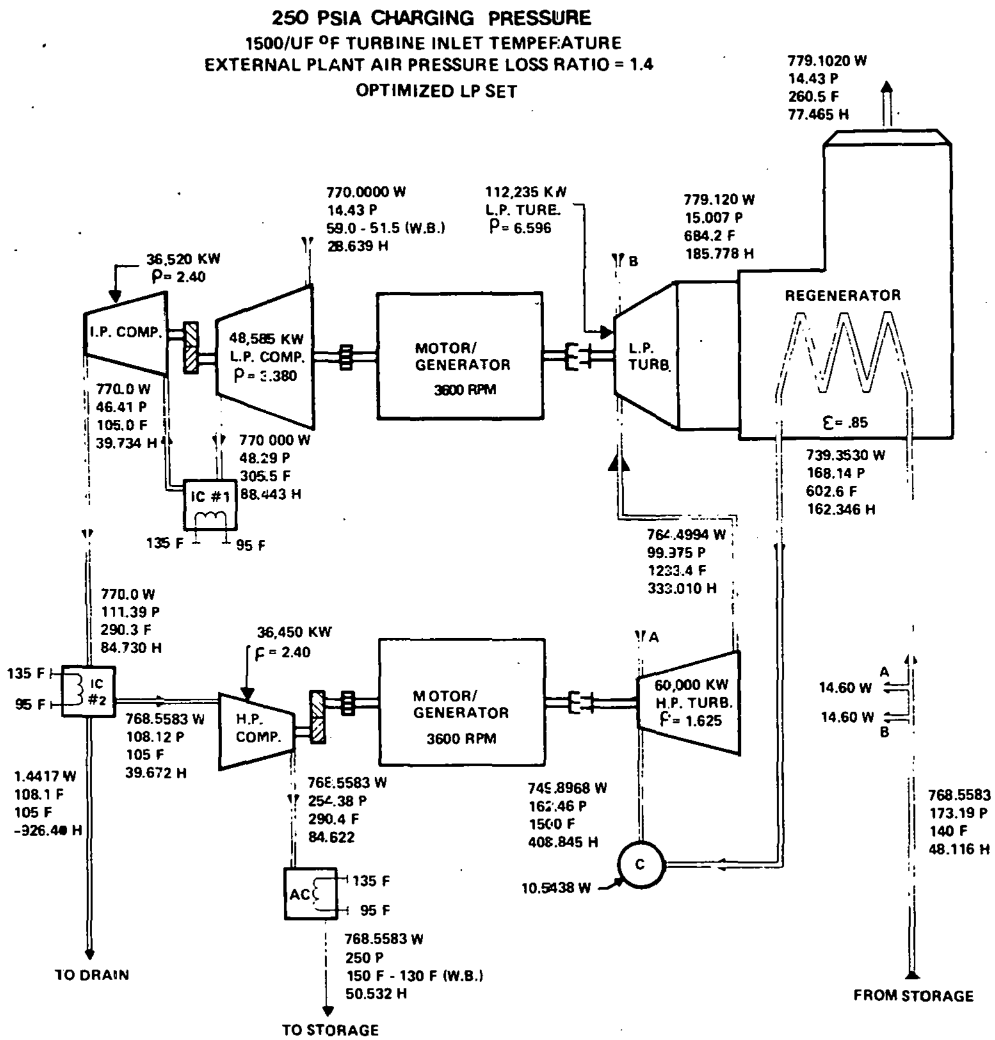
As the cost of fuel oil with respect to electric pumping power increases, the 1500/1500°F cycle has the lower power production cost. Since the present day energy cost ratio for most utilities is already higher than the above ratios and the price of oil is expected to increase at a faster rate than the cost of nuclear fuel or coal, which is expected to generate the surplus electric charging power, the 1500/1500°F cycle was selected over the 1500/2150°F cycle.

The 1500/1500°F cycle also offers the following advantages over the 1500/2150°F cycle:

1. Less NO_x production.
2. A substantial reduction in cooling air requirements.
3. The lower firing temperature will decrease thermal stresses in the turbine.
4. The HP turbine exit temperature is lower, since the optimum HP turbine pressure ratio is increased, allowing conventional cooling of the LP combustor.
5. The low pressure turbine may utilize a standard combustion turbine expander for the 200, 600 and 1000 psi systems.

OPTIMIZED CAES CYCLE HEAT AND MATERIAL BALANCE FLOW DIAGRAMS

Heat and material balances for the four optimized candidate CAES heat cycles initially selected for further evaluation are presented in Figures 6-9 through 6-12. The 1500/2150°F, 1000 PSI nominal storage pressure cycle presented in Figure 6-12 was eliminated for reasons discussed above. The first three diagrams reflect the turbine train performance realized by utilizing a 1.9% per turbine cooling flow, and the effect



COMPRESSED AIR ENERGY STORAGE (CAES) POWER PLANT ENERGY BALANCE

INPUT	BTU/HR
1. ATMOSPHERIC AIR:	79,387,310
2. LP COMPRESSOR POWER (@ THE MOTOR TERMINALS):	174,500,575
3. IP COMPRESSOR POWER (@ THE MOTOR TERMINALS):	131,169,950
4. HP COMPRESSOR POWER (@ THE MOTOR TERMINALS):	130,914,940
5. HP COMBUSTOR FUEL (LHV = 18,055 BTU/LB):	685,325,910
6. LP COMBUSTOR FUEL (LHV = 18,055 BTU/LB):	0
TOTAL INPUT	1.201×10^9

OUTPUT	BTU/HR
1. LP COMPRESSOR DRIVE MOTOR LOSS:	8,726,030
2. IP COMPRESSOR DRIVE MOTOR LOSS:	6,558,500
3. HP COMPRESSOR DRIVE MOTOR LOSS:	6,545,750
4. INTERCOOLER #1 REJECTION HEAT LOSS:	135,021,350
5. INTERCOOLER #2 REJECTION HEAT LOSS:	129,914,805
6. AFTERCOOLER REJECTION HEAT LOSS:	94,320,550
7. AIR COOLER DRAIN LOSS:	-4,808,125
8. CAVITY HEAT LOSS:	6,684,610
9. HP COMBUSTOR LOSS:	13,706,520
10. LP COMBUSTOR LOSS:	0
11. HP TURBINE POWER (@ GENERATOR TERMINALS):	200,633,995
12. HP TURBINE ELECTRICAL LOSS:	4,094,570
13. LP TURBINE POWER (@ GENERATOR TERMINALS):	375,304,920
14. LP TURBINE ELECTRICAL LOSS:	7,659,285
15. EXHAUST GAS LOSS:	217,271,290
TOTAL OUTPUT	1.202×10^9

PLANT PERFORMANCE

TOTAL COMPRESSOR INPUT (@ MOTOR TERMINALS):	127,980 KW
TOTAL TURBINE OUTPUT (@ GENERATOR TERMINALS):	168,720 KW
AUX LOAD:	500 KW
NET TURBINE OUTPUT:	168,220 KW
NET LHV HEAT RATE:	4,075 BTU/KWHR

NOTES:

1. THE ABOVE DATA REFLECTS AS CALCULATED PERFORMANCE. GUARANTEED PERFORMANCE WOULD INCLUDE MARGINS (NOT NECESSARILY THE SAME) FOR THE POWER INPUT AND OUTPUT, AND HEAT RATE.
2. THE ABOVE REFLECTS AN ENTHALPY BASE WHERE THE ENTHALPY OF -60°F GAS EQUALS 0.0.

LEGEND

W – FLOW, LBS/SEC
P – PRESSURE, PSIA
F – TEMPERATURE, °F
H – ENTHALPY, BTU/LB

(W 1511)

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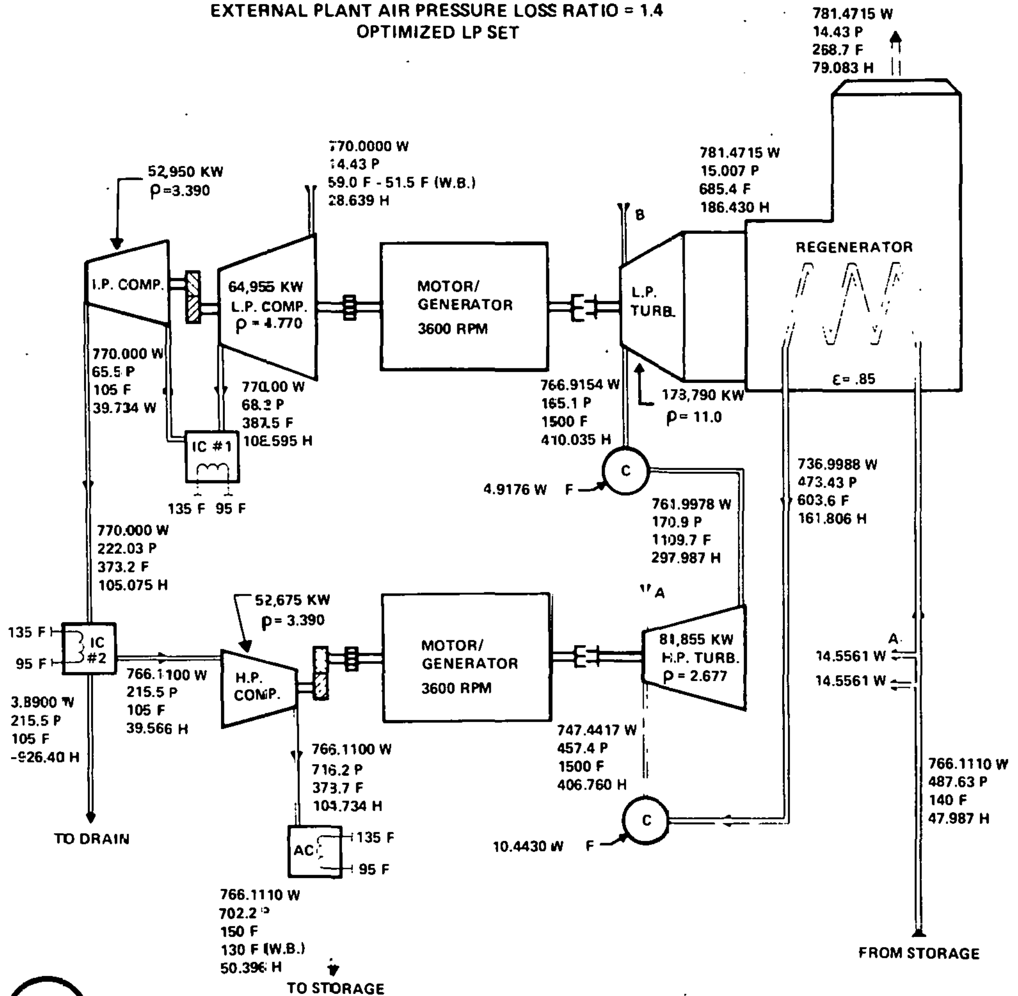
Figure 6-9. 200 PSI Nominal Optimized System Heat and Material Balance

700 PSIA CHARGING PRESSURE
1500/1500°F TURBINE INLET TEMPERATURE
EXTERNAL PLANT AIR PRESSURE LOSS RATIO = 1.4
OPTIMIZED LP SET



COMPRESSED AIR ENERGY STORAGE (CAES) POWER PLANT ENERGY BALANCE

INPUT	BTU/HR
1. ATMOSPHERIC AIR:	79,387,310
2. LP COMPRESSOR POWER (@ THE MOTOR TERMINALS):	233,303,180
3. IP COMPRESSOR POWER (@ THE MOTOR TERMINALS):	190,176,680
4. HP COMPRESSOR POWER (@ THE MOTOR TERMINALS):	189,192,975
5. HP COMBUSTOR FUEL (LHV = 18,055 BTU/LB):	678,774,115
6. LP COMBUSTOR FUEL (LHV = 18,055 BTU/LB):	319,634,165
TOTAL INPUT	1.690×10^9

OUTPUT	BTU/HR
1. LP COMPRESSOR DRIVE MOTOR LOSS:	11,665,160
2. IP COMPRESSOR DRIVE MOTOR LOSS:	9,508,830
3. HP COMPRESSOR DRIVE MOTOR LOSS:	9,459,650
4. INTERCOOLER #1 REJECTION HEAT LOSS:	190,882,690
5. INTERCOOLER #2 REJECTION HEAT LOSS:	195,118,335
6. AFTERCOOLER REJECTION HEAT LOSS:	149,863,985
7. AIR COOLER DRAIN LOSS:	-12,973,305
8. CAVITY HEAT LOSS:	6,644,020
9. HP COMBUSTOR LOSS:	13,575,480
10. LP COMBUSTOR LOSS:	6,392,685
11. HP TURBINE POWER (@ GENERATOR TERMINALS):	273,717,560
12. HP TURBINE ELECTRICAL LOSS:	5,586,075
13. LP TURBINE POWER (@ GENERATOR TERMINALS):	597,861,300
14. LP TURBINE ELECTRICAL LOSS:	12,201,250
15. EXHAUST GAS LOSS:	222,484,000
TOTAL OUTPUT	1.692×10^9

PLANT PERFORMANCE

TOTAL COMPRESSOR INPUT (@ MOTOR TERMINALS):	179,690 KW
TOTAL TURBINE OUTPUT (@ GENERATOR TERMINALS):	255,430 KW
AUX LOAD	500 KW
NET TURBINE OUTPUT:	254,930 KW
NET LHV HEAT RATE:	3,915 BTU/KWHR

NOTES:

1. THE ABOVE DATA REFLECTS AS CALCULATED PERFORMANCE. GUARANTEED PERFORMANCE WOULD INCLUDE MARGINS (NOT NECESSARILY THE SAME) FOR THE POWER INPUT AND OUTPUT, AND HEAT RATE.
2. THE ABOVE REFLECTS AN ENTHALPY BASE WHERE THE ENTHALPY OF -60°F GAS EQUALS 0.0.

LEGEND

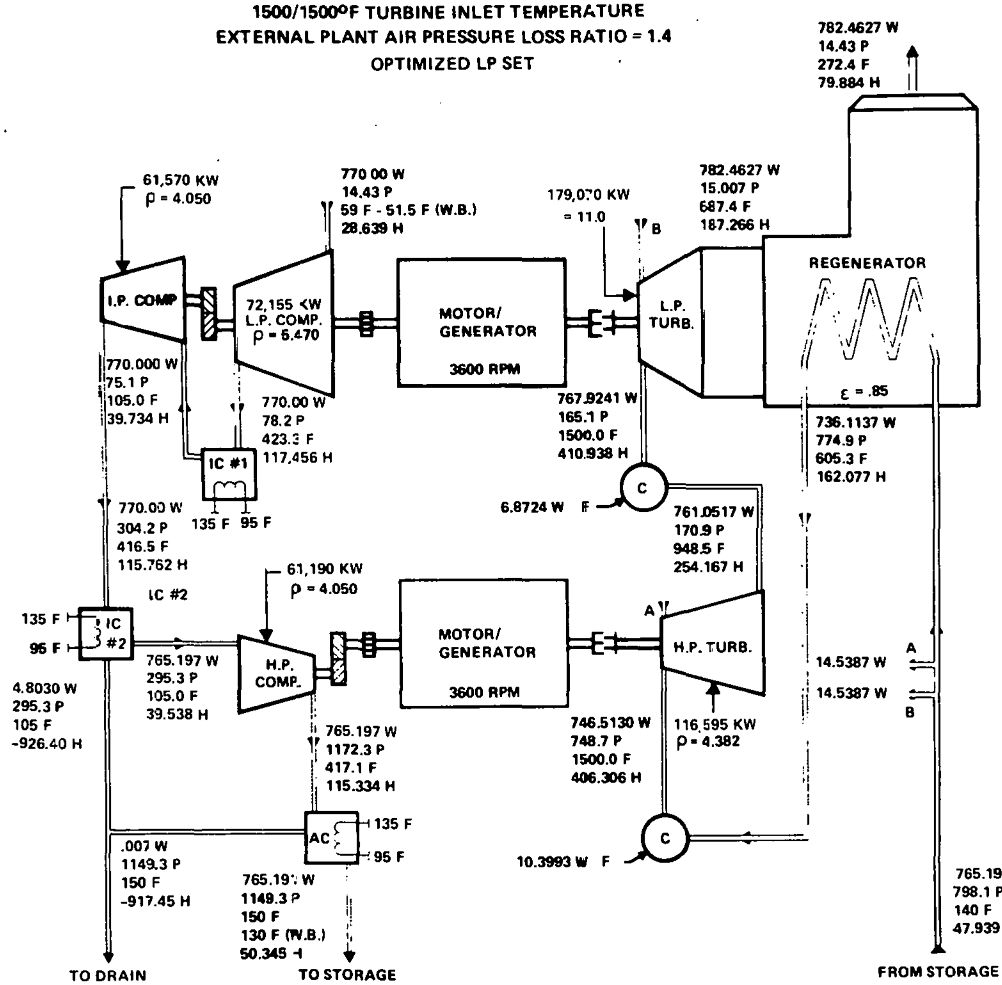
W – FLOW, LBS/SEC
P – PRESSURE, PSIA
F – TEMPERATURE, °F
H – ENTHALPY, BTU/LB

(W 1512)

PUBLIC SERVICE INDIANA
CAES IN AQUIFER
DOE NO. ET-78-C-01-2159

Figure 6-10. 600 PSI Nominal Optimized System Heat and Material Balance

1150 PSIA CHARGING PRESSURE
1500/1500°F TURBINE INLET TEMPERATURE
EXTERNAL PLANT AIR PRESSURE LOSS RATIO = 1.4
OPTIMIZED LP SET



COMPRESSED AIR ENERGY STORAGE (CAES) POWER PLANT ENERGY BALANCE

INPUT	BTU/HR
1. ATMOSPHERIC AIR:	79,387,310
2. LP COMPRESSOR POWER (@ THE MOTOR TERMINALS):	259,159,640
3. IP COMPRESSOR POWER (@ THE MOTOR TERMINALS):	221,148,950
4. HP COMPRESSOR POWER (@ THE MOTOR TERMINALS):	219,783,480
5. HP COMBUSTOR FUEL (LHV = 18,055 BTU/LB):	675,933,700
6. LP COMBUSTOR FUEL (LHV = 18,055 BTU/LB):	446,692,255
TOTAL INPUT	1.902×10^9

OUTPUT	BTU/HR
1. LP COMPRESSOR DRIVE MOTOR LOSS:	12,957,990
2. IP COMPRESSOR DRIVE MOTOR LOSS:	11,057,445
3 HP COMPRESSOR DRIVE MOTOR LOSS:	10,989,175
4. INTERCOOLER #1 REJECTION HEAT LOSS:	215,445,385
5. INTERCOOLER #2 REJECTION HEAT LOSS:	227,994,770
6. AFTERCOOLER REJECTION HEAT LOSS:	179,050,005
7. AIR COOLER DRAIN LOSS:	-16,041,315
8. CAVITY HEAT LOSS:	6,627,780
9. HP COMBUSTOR LOSS:	13,518,675
10. LP COMBUSTOR LOSS:	8,933,845
11. HP TURBINE POWER (@ GENERATOR TERMINALS):	389,875,325
12. HP TURBINE ELECTRICAL LOSS:	7,956,640
13. LP TURBINE POWER (@ GENERATOR TERMINALS):	598,797,895
14. LP TURBINE ELECTRICAL LOSS:	12,220,365
15. EXHAUST GAS LOSS:	225,022,500
TOTAL OUTPUT	1.904×10^9

PLANT PERFORMANCE

TOTAL COMPRESSOR INPUT (@ MOTOR TERMINALS):	205,390 KW
TOTAL TURBINE OUTPUT (@ GENERATOR TERMINALS):	289,750 KW
AUX LOAD:	500 KW
NET TURBINE OUTPUT:	289,250 KW
NET LHV HEAT RATE:	3,880 BTU/KWHR

NOTES:

1. THE ABOVE DATA REFLECTS AS CALCULATED PERFORMANCE. GUARANTEED PERFORMANCE WOULD INCLUDE MARGINS (NOT NECESSARILY THE SAME) FOR THE POWER INPUT AND OUTPUT, AND HEAT RATE.
2. THE ABOVE REFLECTS AN ENTHALPY BASE WHERE THE ENTHALPY OF -60°F GAS EQUALS 0.0.

LEGEND

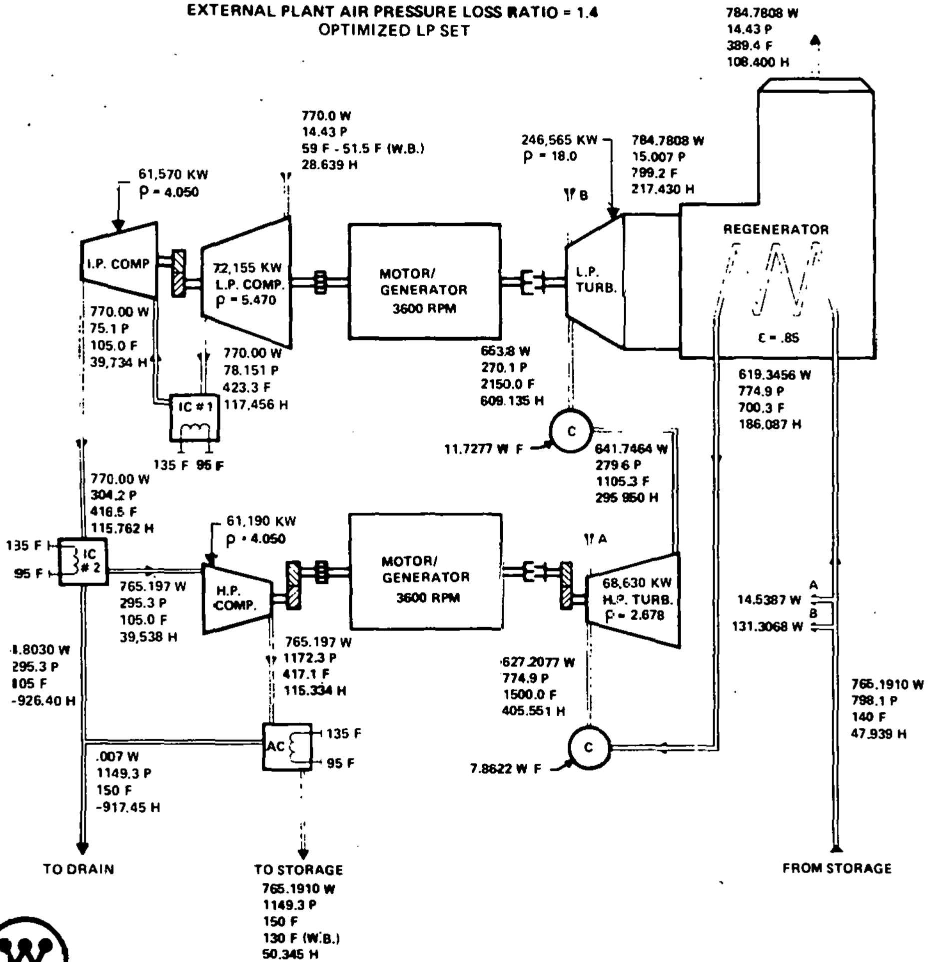
W – FLOW, LBS/SEC
P – PRESSURE, PSIA
F – TEMPERATURE, °F
H – ENTHALPY, BTU/LB

(W 1513)

Figure 6-11. 1000 PSI (1500/1500°F) Nominal Optimized System Heat and Material Balance

PUBLIC SERVICE INDIANA
CAES IN AQUIFER
DOE NO. ET-78-C-01-2159

1150 PSIA CHARGING PRESSURE
1500/2150 °F TURBINE INLET TEMPERATURE
EXTERNAL PLANT AIR PRESSURE LOSS RATIO = 1.4
OPTIMIZED LP SET



COMPRESSED AIR ENERGY STORAGE (CAES) POWER PLANT ENERGY BALANCE

INPUT	BTU/HR
1. ATMOSPHERIC AIR:	79,387,310
2. LP COMPRESSOR POWER (@ THE MOTOR TERMINALS):	259,158,640
3. IP COMPRESSOR POWER (@ THE MOTOR TERMINALS):	221,148,950
4. HP COMPRESSOR POWER (@ THE MOTOR TERMINALS):	219,783,480
5. HP COMBUSTOR FUEL (LHV = 18,055 BTU/LB):	511,027,275
6. LP COMBUSTOR FUEL (LHV = 18,055 BTU/LB):	762,277,045
TOTAL INPUT	2.053×10^9

OUTPUT	BTU/HR
1. LP COMPRESSOR DRIVE MOTOR LOSS:	12,957,930
2. IP COMPRESSOR DRIVE MOTOR LOSS:	11,057,450
3. HP COMPRESSOR DRIVE MOTOR LOSS:	10,989,175
4. INTERCOOLER #1 REJECTION HEAT LOSS:	215,445,385
5. INTERCOOLER #2 REJECTION HEAT LOSS:	227,994,770
6. AFTERCOOLER REJECTION HEAT LOSS:	179,050,005
7. AIR COOLER DRAIN LOSS:	-16,041,315
8. CAVITY HEAT LOSS:	6,627,780
9. HP COMBUSTOR LOSS:	10,220,545
10. LP COMBUSTOR LOSS:	15,245,540
11. HP TURBINE POWER (@ GENERATOR TERMINALS):	229,488,100
12. HP TURBINE ELECTRICAL LOSS:	4,683,430
13. LP TURBINE POWER (@ GENERATOR TERMINALS):	824,496,330
14. LP TURBINE ELECTRICAL LOSS:	16,826,455
15. EXHAUST GAS LOSS:	306,252,860
TOTAL OUTPUT	2.055×10^9

PLANT PERFORMANCE

TOTAL COMPRESSOR INPUT (@ MOTOR TERMINALS):	205,390 KW
TOTAL TURBINE OUTPUT (@ GENERATOR TERMINALS):	308,890 KW
AUX LOAD:	500 KW
NET TURBINE OUTPUT:	308,390 KW
NET LHV HEAT RATE:	4,130 BTU/KWHR

NOTES:

1. THE ABOVE DATA REFLECTS AS CALCULATED PERFORMANCE. GUARANTEED PERFORMANCE WOULD INCLUDE MARGINS (NOT NECESSARILY THE SAME) FOR THE POWER INPUT AND OUTPUT, AND HEAT RATE.
2. THE ABOVE REFLECTS AN ENTHALPY BASE WHERE THE ENTHALPY OF -60°F GAS EQUALS 0.0.

LEGEND

W – FLOW, LB/SEC
P – PRESSURE, PSIA
F – TEMPERATURE, °F
H – ENTHALPY, BTU/LB

(W 1514)

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Figure 6-12. 1000 PSI (1500/2150°F) Nominal Optimized System Heat and Material Balance of a 1.2 pressure loss ratio for the compressed air entering the aquifer, and a 1.2 pressure loss ratio for the compressed stored air exiting the aquifer. The heat balances also show the effect of the assumed 10°F storage air temperature drop.

The machinery configuration depicted in this heat balance diagram does not represent the optimum one shaft machinery layout.

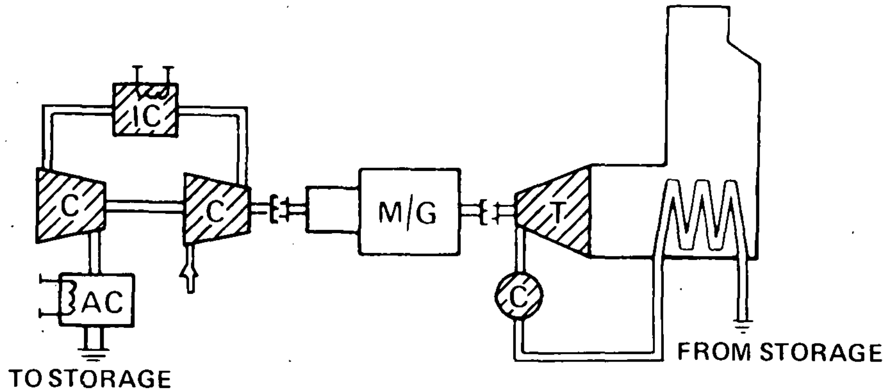
The following is a brief summary of the heat and material balances presented in Figures 6-9 through 6-12:

Nominal System Pressure, PSI	Compressor Train Number of Intercoolers	Turbine Firing Temperatures HPT/LPT in °F	Cycle Power Ratio, KW Charging / Net KW Output	Cycle LHV Heat Rate, BTU/KWHR
200	2	1500/Unfired	.761	4075
600	2	1500/1500	.705	3915
1000	2	1500/1500	.710	3880
1000	2	1500/2150	.666	4130

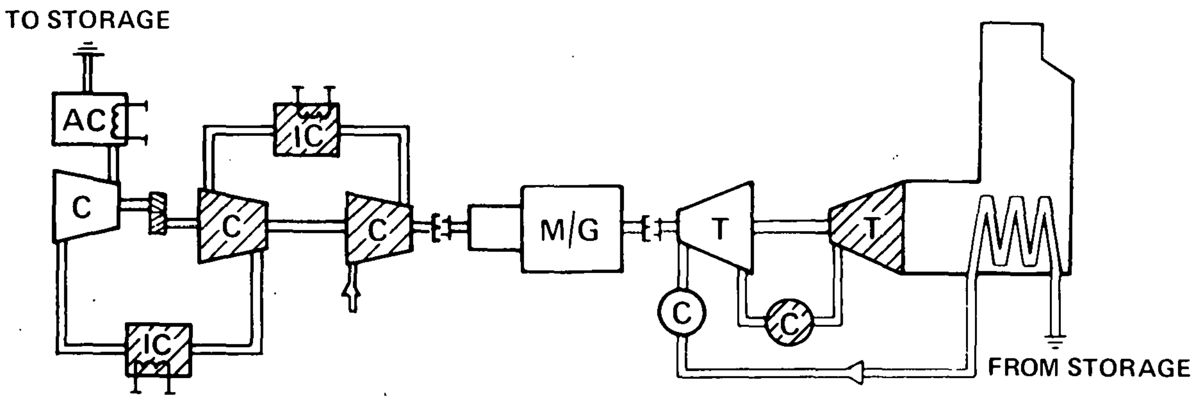
COMPONENT STANDARDIZATION

Standardization of hardware is very effective in reducing development, design, manufacturing and inventory costs. As a result, when possible, the standardization of machinery is very desirable. Westinghouse realized that there was a possibility of standardizing the LP and IP compressors and intercoolers, and the LP turbine for the 200, 600 and 1000 PSI systems. (The standardized IP compressor serves as the 200 PSI system HP compressor.) The possible components that can be standardized are crosshatched in Figure 6-13. The specific changes in the compressor pressure ratios, compressor input power, turbine output power, heat rate and total power production energy cost between the optimized and the standardized machinery configurations are presented in Table 6-7. As can be seen, the maximum change in the total power production energy costs is 3.17%, .01% and .28% for the 200, 600 and 1000 PSI systems, respectively. For a more extensive breakdown of the power production energy costs consult Table 6-8. It should be pointed out that the above two tables reflect the effect of the auxiliary loads.

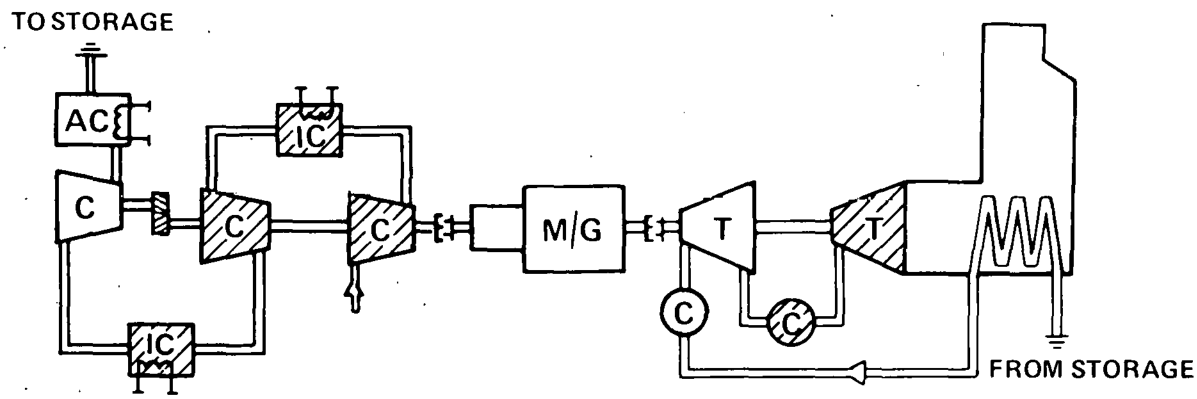
The largest increase (3.17%) in the required compressor power occurred in the 200 PSI system because in the standardized layout there is only one intercooler instead of the two that the optimized configuration contains. The increase in the



250 PSIA CHARGING PRESSURE
1500/UF°F TURBINE INLET TEMPERATURE



750 PSIA CHARGING PRESSURE
1500/1500°F TURBINE INLET TEMPERATURE



1150 PSIA CHARGING PRESSURE
1500/1500°F TURBINE INLET TEMPERATURE

(W 1515)

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Figure 6-13. Common Components in Standardized Machinery Sets

Table 6-7

OPTIMIZED VS STANDARDIZED
LOW PRESSURE MACHINERY COMPARISON

	200 PSI		600 PSI		1000 PSI	
	OPT	STD	OPT	STD	OPT	STD
LP COMPRESSOR, ρ	3.380	5.090	4.770	5.090	5.470	5.090
IP COMPRESSOR, ρ	2.400	–	3.390	3.765	4.050	3.765
HP COMPRESSOR, ρ	2.400	3.685	3.390	2.861	4.050	4.682
LP TURBINE, ρ	10.700	10.700	11.000	11.000	11.000	11.000
CHARGING POWER, KW/LB/SEC*	166.2	171.5	233.4	233.4	266.7	267.5
NET OUTPUT PER MACHINERY SET, KW/LB/SEC*	218.5	219.0	331.1	330.9	375.7	375.7
NET LHV HEAT RATE, BTU/KWHR*	4075	4075	3915	3915	3880	3880
FUEL @ $/10^6 BTU / ELEC. PWR. IN MILS/KWHR	TOTAL POWER PRODUCTION ENERGY COST IN MILS/KWHR					
$2.50 / 10	17.80	18.02	16.84	16.84	16.80	16.82
$2.50 / 20	25.40	25.85	23.88	23.89	23.90	23.94
$5.00 / 20	35.59	36.04	33.67	33.68	33.60	33.64
$7.50 / 10	38.17	38.39	36.41	36.41	36.20	36.22
$7.50 / 20	45.78	46.22	43.46	43.47	43.30	43.34
$7.75 / 60	77.23	78.56	72.63	72.66	72.67	72.79

(W 1548)

CROSSHATCHED BLOCKS IDENTIFY COMMON COMPONENTS.
*REFLECTS MOTOR, GENERATOR AND AUXILIARY LOAD LOSSES.

The charging power, turbine output and heat rate values are as calculated and do not include any margin.

Table 6-8

TOTAL POWER PRODUCTION ENERGY COSTS OF THE OPTIMIZED VS THE STANDARDIZED CAES UNITS

CASE	1.00	1.10	2.00	2.10	3.00	3.10	4.00
T1HPT	1500.	1500.	1500.	1500.	1500.	1500.	1500.
T2HPT	1046.	1046.	1110.	1110.	948.	948.	1105.
T1LPT	1046.	1046.	1500.	1500.	1500.	1500.	2150.
CAV LOS	1.20	1.20	1.20	1.20	1.20	1.20	1.20
REG EFT	.85	.85	.85	.85	.85	.85	.85
COMP KW	166.2	171.5	233.4	233.4	266.7	267.5	266.7
TURB KW	218.5	219.0	331.1	330.9	375.7	375.7	400.5
H.R.	4075.	4075.	3915.	3915.	3880.	3880.	4130.

FUEL COST = $2.50 PER MILLION BTU
ELECT COST = 10.00 MILS PER KW-HR ENERGY COST RATIO = .2500

10.19	10.19	9.79	9.79	9.70	9.70	10.33
7.61	7.83	7.05	7.05	7.10	7.12	6.66
TOTAL POWER PRODUCTION COSTS IN MILS PER KW-HR						
17.80	18.02	16.84	16.84	16.80	16.82	16.99

FUEL COST = $2.50 PER MILLION BTU
ELECT COST = 20.00 MILS PER KW-HR ENERGY COST RATIO = .1250

10.19	10.19	9.79	9.79	9.70	9.70	10.33
15.22	15.66	14.10	14.10	14.20	14.24	13.32
TOTAL POWER PRODUCTION COSTS IN MILS PER KW-HR						
25.40	25.85	23.88	23.89	23.90	23.94	23.65

FUEL COST = $4.25 PER MILLION BTU
ELECT COST = 15.10 MILS PER KW-HR ENERGY COST RATIO = .2815

17.32	17.32	16.64	16.64	16.49	16.49	17.55
11.49	11.82	10.64	10.65	10.72	10.75	10.06
TOTAL POWER PRODUCTION COSTS IN MILS PER KW-HR						
28.81	29.14	27.28	27.29	27.21	27.24	27.61

FUEL COST = $5.00 PER MILLION BTU
ELECT COST = 17.06 MILS PER KW-HR ENERGY COST RATIO = .2931

20.38	20.38	19.58	19.58	19.40	19.40	20.65
12.98	13.36	12.03	12.03	12.11	12.15	11.36
TOTAL POWER PRODUCTION COSTS IN MILS PER KW-HR						
33.35	33.73	31.60	31.61	31.51	31.55	32.01

FUEL COST = $5.00 PER MILLION BTU
ELECT COST = 20.00 MILS PER KW-HR ENERGY COST RATIO = .2500

20.38	20.38	19.58	19.58	19.40	19.40	20.65
15.22	15.66	14.10	14.10	14.20	14.24	13.32
TOTAL POWER PRODUCTION COSTS IN MILS PER KW-HR						
35.59	36.04	33.67	33.68	33.60	33.64	33.97

FUEL COST = $7.50 PER MILLION BTU
ELECT COST = 10.00 MILS PER KW-HR ENERGY COST RATIO = .7500

30.56	30.56	29.36	29.36	29.10	29.10	30.98
7.61	7.83	7.05	7.05	7.10	7.12	6.66
TOTAL POWER PRODUCTION COSTS IN MILS PER KW-HR						
38.17	38.39	36.41	36.41	36.20	36.22	37.64

FUEL COST = $7.50 PER MILLION BTU
ELECT COST = 20.00 MILS PER KW-HR ENERGY COST RATIO = .3750

30.56	30.56	29.36	29.36	29.10	29.10	30.98
15.22	15.66	14.10	14.10	14.20	14.24	13.32
TOTAL POWER PRODUCTION COSTS IN MILS PER KW-HR						
45.78	46.22	43.46	43.47	43.30	43.34	44.30

FUEL COST = $7.75 PER MILLION BTU
ELECT COST = 60.00 MILS PER KW-HR ENERGY COST RATIO = .1292

31.58	31.58	30.34	30.34	30.07	30.07	32.01
45.65	46.98	42.29	42.31	42.60	42.72	39.96
TOTAL POWER PRODUCTION COSTS IN MILS PER KW-HR						
77.23	78.56	72.63	72.66	72.67	72.79	71.97

NOTES:

(1) Case 1.0 has nominal storage pressure of 200 PSI utilizing its optimized two intercooler compressor train, 1500°F-UF turbine inlet temperature.
(2) Case 1.1 has nominal storage pressure of 200 PSI utilizing just the standardized one intercooler LP compressor set, 1500°F-UF turbine inlet temperature.
(3) Case 2.0 has nominal storage pressure of 600 PSI utilizing its optimized two intercooler compressor train, 1500-1500°F turbine inlet temperature.
(4) Case 2.1 has nominal storage pressure of 600 PSI utilizing the standardized LP compressor set, a second intercooler and a customized HP compressor, 1500-1500°F turbine inlet temperature.
(5) Case 3.0 has nominal storage pressure of 1000 PSI utilizing its optimized two intercooler compressor train, 1500-1500°F turbine inlet temperature.
(6) Case 3.1 has nominal storage pressure of 1000 PSI utilizing the standardized LP compressor set, a second intercooler and a customized HP compressor, 1500-1500°F turbine inlet temperature.
(7) Case 4.0 has nominal storage pressure of 1000 PSI utilizing its optimized two intercooler compressor train, 1500-2150°F turbine inlet temperature.

(W 1549)

The total Work of Compression, Turbine Output Power, and Heat Rate values are as calculated and do not include any margin.

required compressor work for the standardized 600 PSI system compressor train over the optimized train is very slight (.01%). The compressor power increase for the standardized 1000 PSI system was .28%.

In the 200 PSI system the turbine train output increased a small amount in the standardized cycle because in the standardized one intercooler 200 PSI system there is less condensation during the charging cycle. Since the CAES cycle calculations are based on equal aquifer injection and extraction mass flow rates; and since the aquifer injection mass flow rate in the standardized cycle is slightly greater (for the same LP compressor inlet air flow) than in the optimized cycle, the turbine mass flow rate in the standardized cycle is also slightly greater. This results in the increase in the net output in the standardized machinery set indicated in Tables 6-7 and 6-8. There is more condensation of the atmospheric moisture in the standardized 600 PSI system compressor train, thus the turbine train output is decreased. There is no appreciable difference in the amount of condensation in the 1000 PSI system.

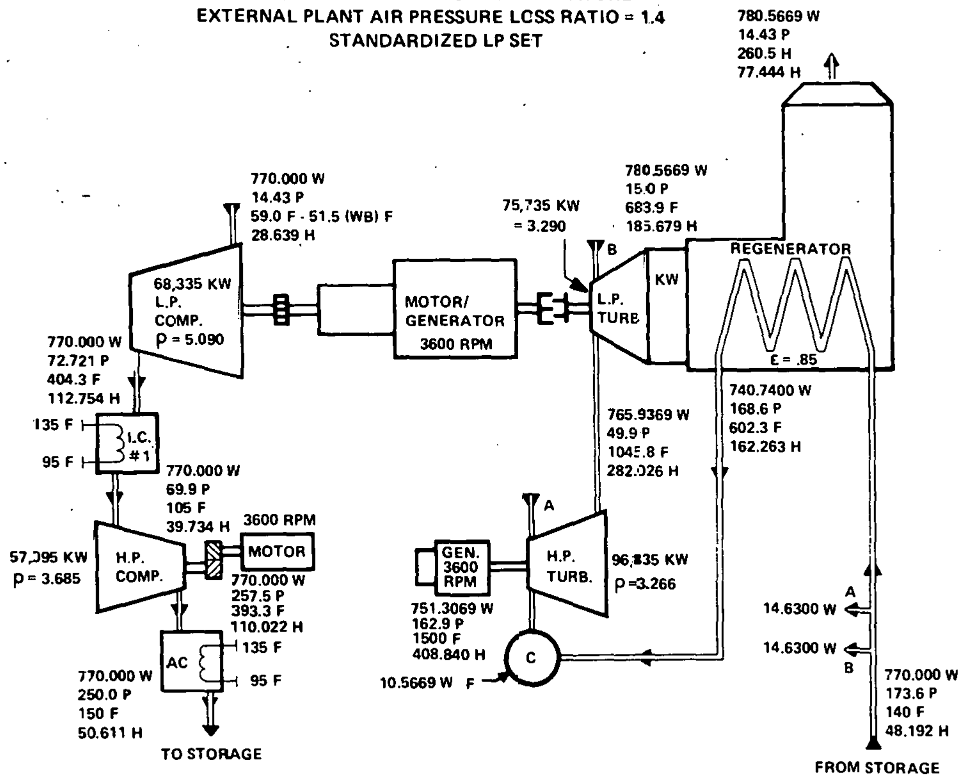
Since the use of the standardized equipment does not significantly increase the total power production energy cost, the Utility Sponsors agreed that standardized components should be utilized.

The 200, 600 and 1000 PSI systems heat and material balances for the standardized configurations are presented in Figures 6-14, 6-15 and 6-16, respectively. It should be noted that these figures do not reflect the actual, physical machinery layouts.

On the above heat balances, the aircooler heat injection and drain losses have been revised since the last publication.

The next set of heat and material balances presented in Figures 6-17, 6-18 and 6-19 reflect the refined performance projections for the motor efficiency, gearbox efficiency, generator efficiency and combustion efficiency; plus the 200 PSI system was analyzed with a single element turbine train thus it required only 1.9% of the total inlet flow for cooling air instead of 3.8% reflected in the earlier heat balances. The new motor efficiency was updated from 95% to 96%. The projected generator efficiency increased from 98% to 98.5%. According to Philadelphia Gear Corporation, the gearbox efficiency will be 98.5%. Finally the projected combustion efficiency increased from 98% to 99%.

250 PSIA CHARGING PRESSURE
1500/UF°F TURBINE INLET TEMPERATURE
EXTERNAL PLANT AIR PRESSURE LOSS RATIO = 1.4
STANDARDIZED LP SET



COMPRESSED AIR ENERGY STORAGE (CAES) POWER PLANT ENERGY BALANCE

INPUT	BTU/HR
1. ATMOSPHERIC AIR:	79,387,310
2. LP COMPRESSOR POWER (@ THE MOTOR TERMINALS):	245,437,230
3. IP COMPRESSOR POWER (@ THE MOTOR TERMINALS):	0
4. HP COMPRESSOR POWER (@ THE MOTOR TERMINALS):	205,069,782
5. HP COMBUSTOR FUEL (LHV = 18,055 BTU/LB):	686,827,365
6. LP COMBUSTOR FUEL (LHV = 18,055 BTU/LB):	0
TOTAL INPUT	1.217×10^9

OUTPUT	BTU/HR
1. LP COMPRESSOR DRIVE MOTOR LOSS:	12,271,860
2. IP COMPRESSOR DRIVE MOTOR LOSS:	0
3. HP COMPRESSOR DRIVE MOTOR LOSS:	10,253,490
4. INTERCOOLER #1 REJECTION HEAT LOSS:	202,411,440
5. INTERCOOLER #2 REJECTION HEAT LOSS:	0
6. AFTERCOOLER REJECTION HEAT LOSS:	164,687,290
7. AIR COOLER DRAIN LOSS:	0
8. CAVITY HEAT LOSS:	6,705,470
9. HP COMBUSTOR LOSS:	13,736,545
10. LP COMBUSTOR LOSS:	0
11. HP TURBINE POWER (@ GENERATOR TERMINALS):	323,806,415
12. HP TURBINE ELECTRICAL LOSS:	6,608,295
13. LP TURBINE POWER (@ GENERATOR TERMINALS):	253,242,310
14. LP TURBINE ELECTRICAL LOSS:	5,168,210
15. EXHAUST GAS LOSS:	217,620,805
TOTAL OUTPUT	1.217×10^9

PLANT PERFORMANCE

TOTAL COMPRESSOR INPUT (@ THE MOTOR TERMINALS):	132,030 KW
TOTAL TURBINE OUTPUT (@ GENERATOR TERMINALS):	169,115 KW
AUX. LOAD:	500 KW
NET TURBINE OUTPUT:	168,615 KW
NET LHV HEAT RATE:	4,075 BTU/KWHR

NOTES:

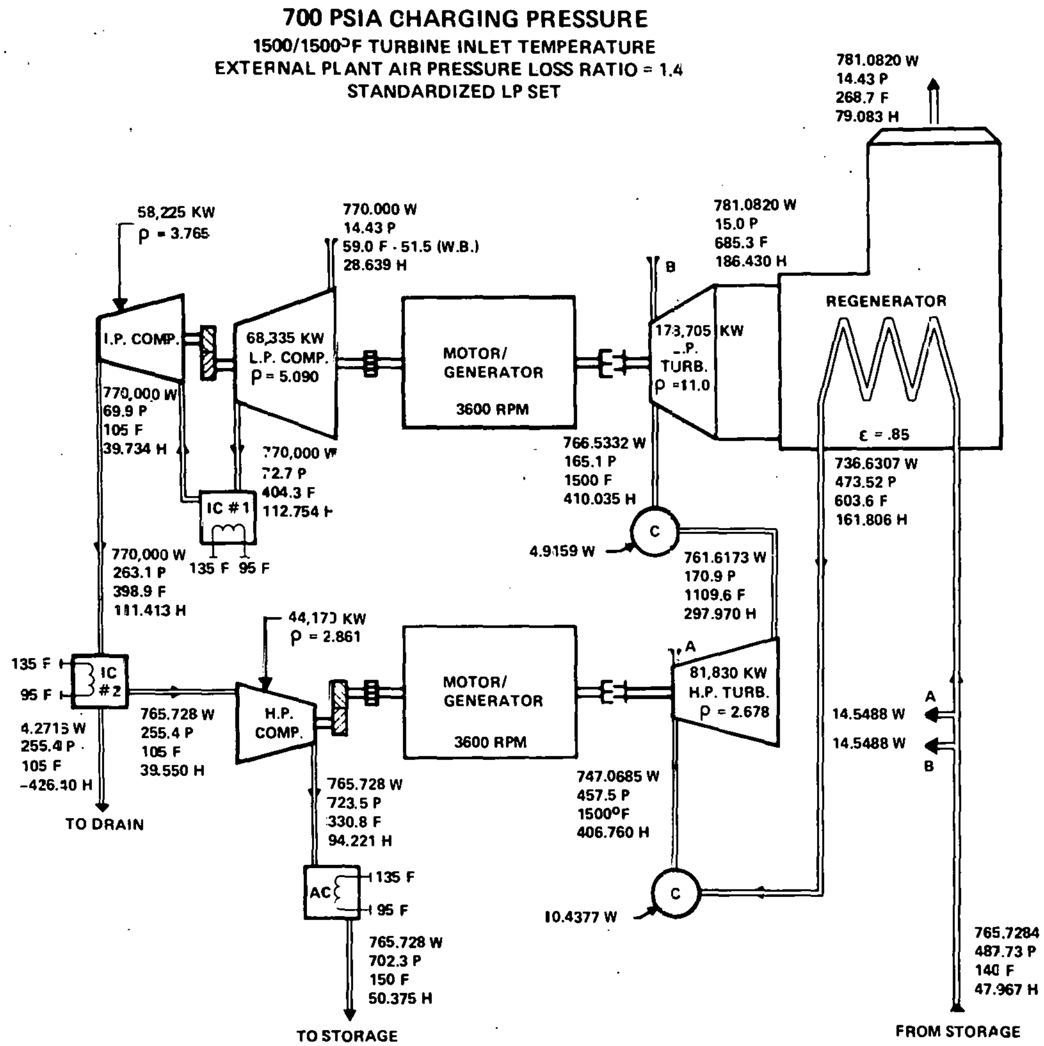
1. THE ABOVE DATA REFLECTS AS CALCULATED PERFORMANCE. GUARANTEED PERFORMANCE WOULD INCLUDE MARGINS (NOT NECESSARILY THE SAME) FOR THE POWER INPUT AND OUTPUT, AND HEAT RATE.
2. THE ABOVE REFLECTS AN ENTHALPY BASE WHERE THE ENTHALPY OF -60°F GAS EQUALS 0.0.

LEGEND

W – FLOW, LBS/SEC
P – PRESSURE, PSIA
F – TEMPERATURE, °F
H – ENTHALPY, BTU/LB

(W 1516)

Figure 6-14. 200 PSI Nominal Standardized System Heat and Material Balance



COMPRESSED AIR ENERGY STORAGE (CAES) POWER PLANT ENERGY BALANCE

INPUT	BTU/HR
1. ATMOSPHERIC AIR:	79,387,310
2. LP COMPRESSOR POWER (@ THE MOTOR TERMINALS):	245,437,230
3. IP COMPRESSOR POWER (@ THE MOTOR TERMINALS):	209,117,660
4. HP COMPRESSOR POWER (@ THE MOTOR TERMINALS):	158,639,500
5. HP COMBUSTOR FUEL (LHV = 18,055 BTU/LB):	678,429,625
6. LP COMBUSTOR FUEL (LHV = 18,055 BTU/LB):	319,523,670
TOTAL INPUT	1.691×10^9

OUTPUT	BTU/HR
1. LP COMPRESSOR DRIVE MOTOR LOSS:	12,271,860
2. IP COMPRESSOR DRIVE MOTOR LOSS:	10,455,880
3. HP COMPRESSOR DRIVE MOTOR LOSS:	7,931,975
4. INTERCOOLER #1 REJECTION HEAT LOSS:	202,411,440
5. INTERCOOLER #2 REJECTION HEAT LOSS:	214,611,985
6. AFTERCOOLER REJECTION HEAT LOSS:	120,866,860
7. AIR COOLER DRAIN LOSS:	-14,245,955
8. CAVITY HEAT LOSS:	6,637,945
9. HP COMBUSTOR LOSS:	13,568,595
10. LP COMBUSTOR LOSS:	6,390,475
11. HP TURBINE POWER (@ GENERATOR TERMINALS):	273,627,375
12. HP TURBINE ELECTRICAL LOSS:	5,584,230
13. LP TURBINE POWER (@ GENERATOR TERMINALS):	597,563,350
14. LP TURBINE ELECTRICAL LOSS:	12,195,170
15. EXHAUST GAS LOSS:	222,372,595
TOTAL OUTPUT	1.692×10^9

PLANT PERFORMANCE

TOTAL COMPRESSOR INPUT (@ THE MOTOR TERMINALS):	179,710 KW
TOTAL TURBINE OUTPUT (@ GENERATOR TERMINALS):	255,320 KW
AUX. LOAD:	500 KW
NET TURBINE OUTPUT:	254,820 KW
NET LHV HEAT RATE:	3,915 BTU/KWHR

NOTES:

1. THE ABOVE DATA REFLECTS AS CALCULATED PERFORMANCE. GUARANTEED PERFORMANCE WOULD INCLUDE MARGINS (NOT NECESSARILY THE SAME) FOR THE POWER INPUT AND OUTPUT, AND HEAT RATE.
2. THE ABOVE REFLECTS AN ENTHALPY BASE WHERE THE ENTHALPY OF -60°F GAS EQUALS 0.0.

LEGEND

W – FLOW, LBS/SEC
P – PRESSURE, PSIA
F – TEMPERATURE, °F
H – ENTHALPY, BTU/LB

(W 1517)

Figure 6-15. 600 PSI Nominal Standardized System Heat and Material Balance

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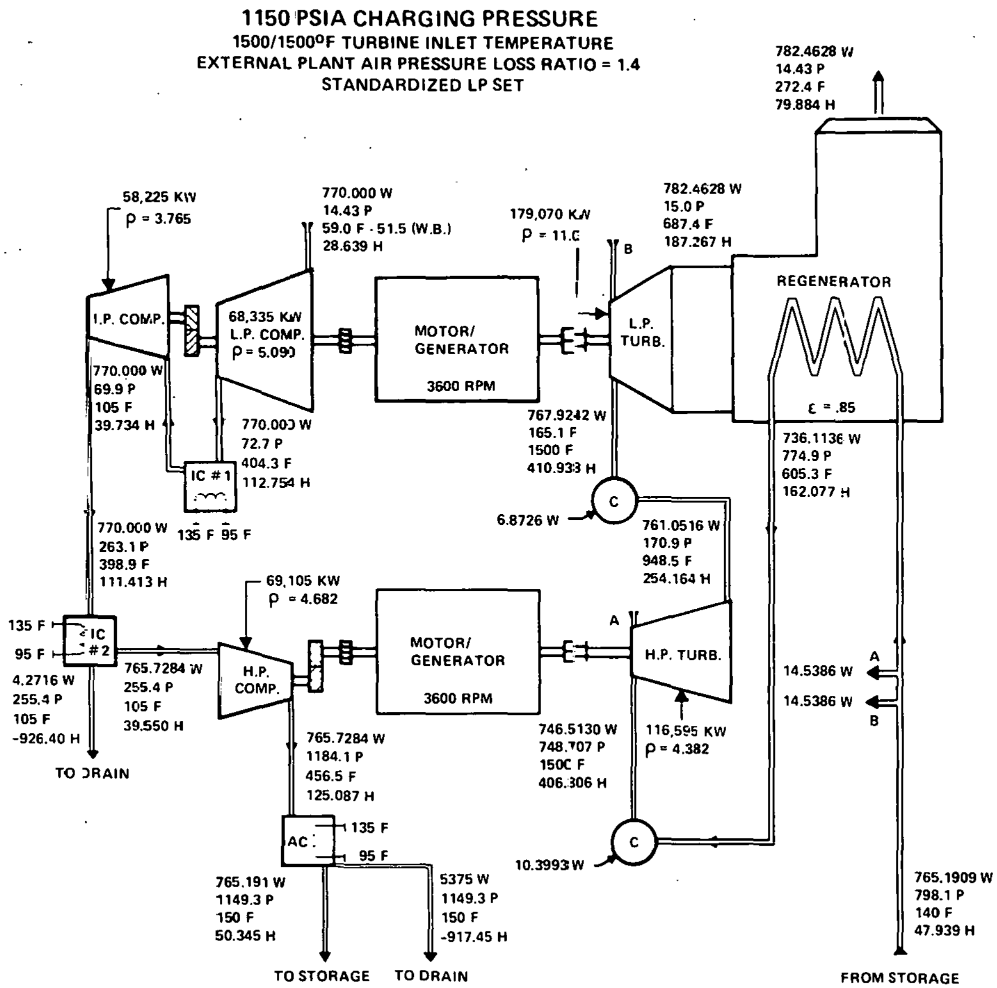
CAES IN AQUIFER
DOE NO. ET-78-C-01-2159



COMPRESSED AIR ENERGY STORAGE (CAES) POWER PLANT ENERGY BALANCE

INPUT	BTU/HR
1. ATMOSPHERIC AIR:	79,387,310
2. LP COMPRESSOR POWER (@ THE MOTOR TERMINALS):	245,437,230
3. IP COMPRESSOR POWER (@ THE MOTOR TERMINALS):	209,117,660
4. HP COMPRESSOR POWER (@ THE MOTOR TERMINALS):	248,202,860
5. HP COMBUSTOR FUEL (LHV = 18,055 BTU/LB):	675,933,700
6. LP COMBUSTOR FUEL (LHV = 18,055 BTU/LB):	446,705,255
TOTAL INPUT	1.905×10^9

OUTPUT	BTU/HR
1. LP COMPRESSOR DRIVE MOTOR LOSS:	12, 271,860
2. IP COMPRESSOR DRIVE MOTOR LOSS:	10,455,880
3. HP COMPRESSOR DRIVE MOTOR LOSS:	12,410,145
4. INTERCOOLER #1 REJECTION HEAT LOSS:	202,411,440
5. INTERCOOLER #2 REJECTION HEAT LOSS:	214,611,985
6. AFTERCOOLER REJECTION HEAT LOSS:	207,908,145
7. AIR COOLER DRAIN LOSS:	-16,574,820
8. CAVITY HEAT LOSS:	6,627,780
9. HP COMBUSTOR LOSS:	13,518,675
10. LP COMBUSTOR LOSS:	8,934,105
11. HP TURBINE POWER (@ GENERATOR TERMINALS):	389,883,720
12. HP TURBINE ELECTRICAL LOSS:	7,956,810
13. LP TURBINE POWER (@ GENERATOR TERMINALS):	598,797,965
14. LP TURBINE ELECTRICAL LOSS:	12,220,370
15. EXHAUST GAS LOSS:	225,022,530
TOTAL OUTPUT	1.906×10^9

PLANT PERFORMANCE

TOTAL COMPRESSOR INPUT (@ THE MOTOR TERMINALS):	205,958 KW
TOTAL TURBINE OUTPUT (@ GENERATOR TERMINALS):	289,750 KW
AUX. LOAD:	500 KW
NET TURBINE OUTPUT:	289,250 KW
NET LHV HEAT RATE:	3,880 BTU/KWHR

NOTES:

1. THE ABOVE DATA REFLECTS AS CALCULATED PERFORMANCE. GUARANTEED PERFORMANCE WOULD INCLUDE MARGINS (NOT NECESSARILY THE SAME) FOR THE POWER INPUT AND OUTPUT, AND HEAT RATE.
2. THE ABOVE REFLECTS AN ENTHALPY BASE WHERE THE ENTHALPY OF -60°F GAS EQUALS 0.0.

LEGEND

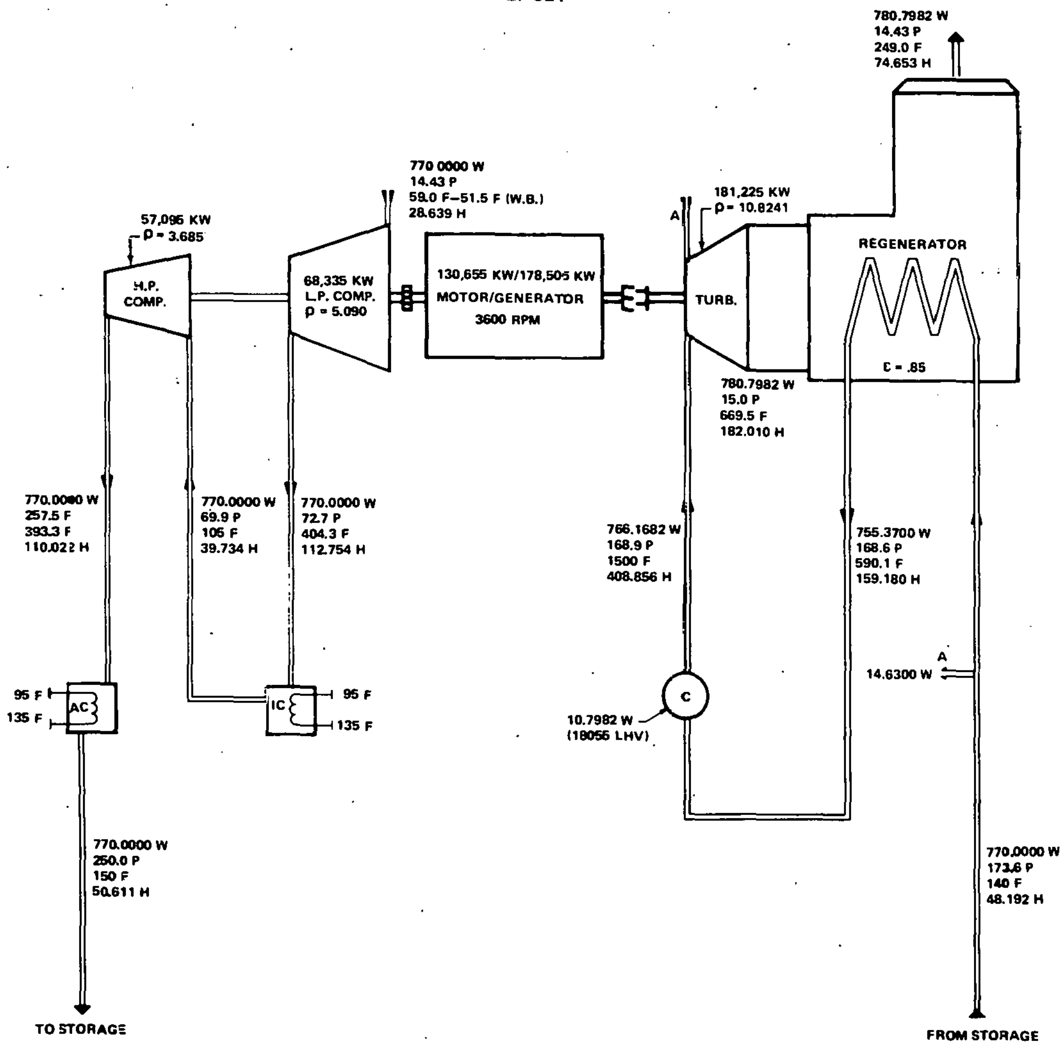
W – FLOW, LBS/SEC
P – PRESSURE, PSIA
F – TEMPERATURE, °F
H – ENTHALPY, BTU/LB

(W 1518)

Figure 6-16. 1000 PSI (1500/1500°F) Nominal Standardized System Heat and Material Balance

PUBLIC SERVICE INDIANA
CAES IN AQUIFER
DOE NO. ET-78-C-01-2159

250 PSIA CHARGING PRESSURE
1500°F TURBINE INLET TEMPERATURE
EXTERNAL PLANT AIR PRESSURE LOSS RATIO = 1.4
REFINED STANDARDIZED LP SET



COMPRESSED AIR ENERGY STORAGE (CAES) POWER PLANT ENERGY BALANCE

INPUT	BTU/HR
1. ATMOSPHERIC AIR:	79,387,310
2. LP COMPRESSOR POWER (@ THE MOTOR TERMINALS):	242,880,590
3. IP COMPRESSOR POWER (@ THE MOTOR TERMINALS):	0
4. HP COMPRESSOR POWER (@ THE MOTOR TERMINALS):	202,933,640
5. HP COMBUSTOR FUEL (LHV = 18,055 BTU/LB):	701,861,405
6. LP COMBUSTOR FUEL (LHV = 18,055 BTU/LB):	0
TOTAL INPUT	1.227×10^9

OUTPUT	BTU/HR
1. LP COMPRESSOR DRIVE MOTOR LOSS:	9,715,225
2. IP COMPRESSOR DRIVE MOTOR LOSS:	0
3. HP COMPRESSOR DRIVE MOTOR LOSS:	8,117,345
4. HP COMPRESSOR GEAR LOSS:	0
5. INTERCOOLER #1 REJECTION HEAT LOSS:	202,411,440
6. INTERCOOLER #2 REJECTION HEAT LOSS:	0
7. AFTERCOOLER REJECTION HEAT LOSS:	164,687,290
8. AIR COOLER DRAIN LOSS:	0
9. CAVITY HEAT LOSS:	6,705,470
10. HP COMBUSTOR LOSS:	7,018,615
11. LP COMBUSTOR LOSS:	0
12. HP TURBINE POWER (@ GENERATOR TERMINALS):	0
13. HP TURBINE ELECTRICAL LOSS:	0
14. LP TURBINE POWER (@ GENERATOR TERMINALS):	609,084,550
15. LP TURBINE ELECTRICAL LOSS:	9,275,400
16. EXHAUST GAS LOSS:	209,840,140
TOTAL OUTPUT	1.227×10^9

PLANT PERFORMANCE

TOTAL COMPRESSOR INPUT (@ THE MOTOR TERMINALS):	130,655 KW
TOTAL TURBINE OUTPUT (@ GENERATOR TERMINALS):	178,505 KW
AUX. LOAD:	500 KW
NET TURBINE OUTPUT:	178,005 KW
NET LVH HEAT RATE:	3,945 BTU/KWHR

NOTES:

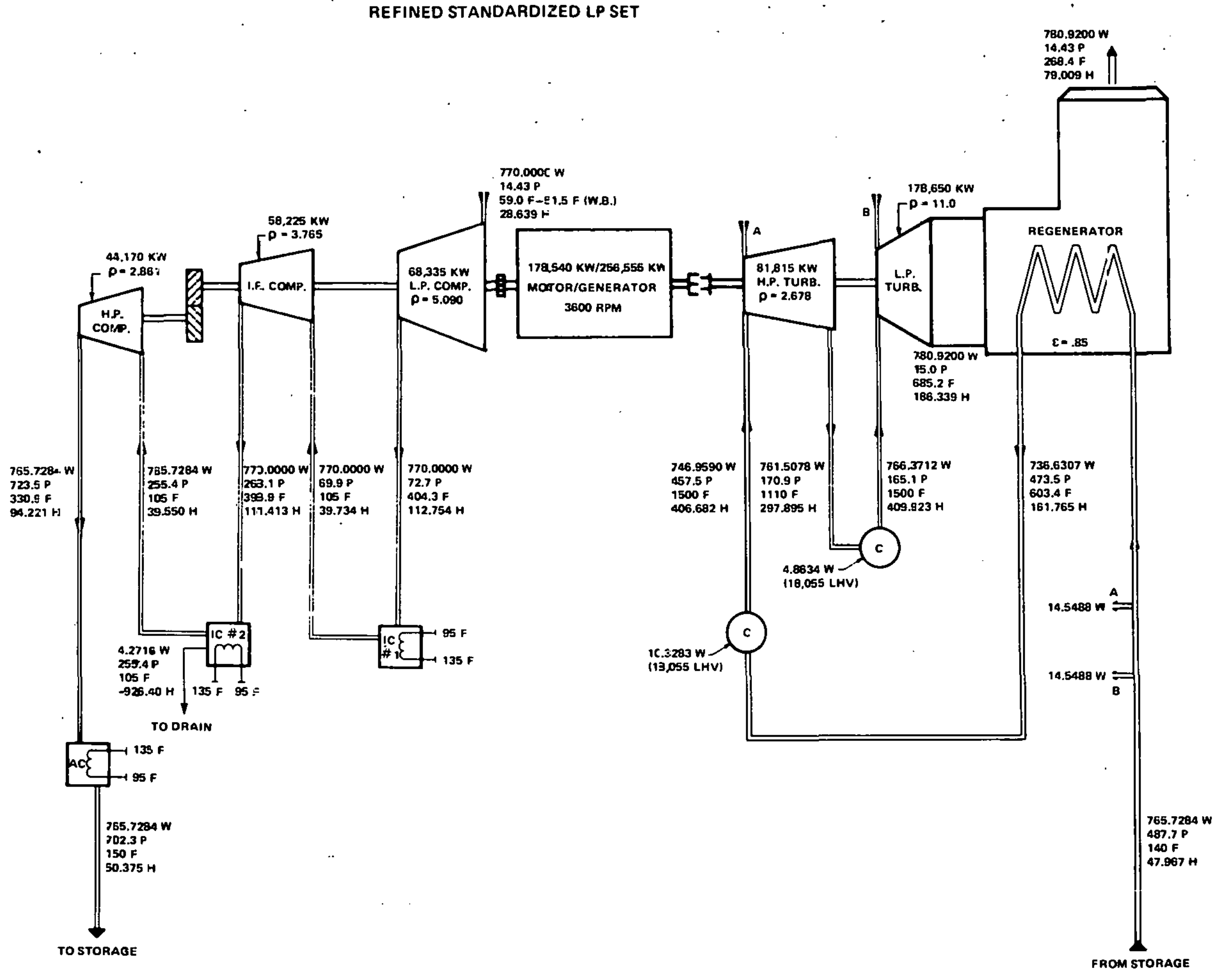
1. THE ABOVE DATA REFLECTS AS CALCULATED PERFORMANCE. GUARANTEED PERFORMANCE WOULD INCLUDE MARGINS (NOT NECESSARILY THE SAME) FOR THE POWER INPUT AND OUTPUT, AND HEAT RATE.
2. THE ABOVE REFLECTS AN ENTHALPY BASE WHERE THE ENTHALPY OF -60°F GAS EQUALS 0.0.

(W 1519)

Figure 6-17. 200 PSI Nominal Refined Standardized System Heat and Material Balance

PUBLIC SERVICE INDIANA
CAES IN AQUIFER
DOE NO. ET-78-C-01-2159

700 PSIA CHARGING PRESSURE
1500/1500°F TURBINE INLET TEMPERATURE
EXTERNAL PLANT AIR PRESSURE LOSS RATIO = 1.4
REFINED STANDARDIZED LP SET



COMPRESSED AIR ENERGY STORAGE (CAES) POWER PLANT ENERGY BALANCE

INPUT	BTU/HR
1. ATMOSPHERIC AIR:	79,387,310
2. LP COMPRESSOR POWER (@ THE MOTOR TERMINALS):	242,880,590
3. IP COMPRESSOR POWER (@ THE MOTOR TERMINALS):	206,939,350
4. HP COMPRESSOR POWER (@ THE MOTOR TERMINALS):	159,377,670
5. HP COMBUSTOR FUEL (LHV = 18,055 BTU/LB):	671,318,845
6. LP COMBUSTOR FUEL (LHV = 18,055 BTU/LB):	316,111,275
TOTAL INPUT	1.676×10^9

OUTPUT	BTU/HR
1. LP COMPRESSOR DRIVE MOTOR LOSS:	9,715,225
2. IP COMPRESSOR DRIVE MOTOR LOSS:	8,277,575
3. HP COMPRESSOR DRIVE MOTOR LOSS:	6,375,105
4. HP COMPRESSOR GEAR LOSS:	2,295,040
5. INTERCOOLER #1 REJECTION HEAT LOSS:	202,411,440
6. INTERCOOLER #2 REJECTION HEAT LOSS:	214,611,985
7. AFTERCOOLER REJECTION HEAT LOSS:	120,866,860
8. AIR COOLER DRAIN LOSS:	-14,245,955
9. CAVITY HEAT LOSS:	6,637,945
10. HP COMBUSTOR LOSS:	6,713,190
11. LP COMBUSTOR LOSS:	3,161,115
12. HP TURBINE POWER (@ GENERATOR TERMINALS):	274,974,550
13. HP TURBINE ELECTRICAL LOSS:	4,187,430
14. LP TURBINE POWER (@ GENERATOR TERMINALS):	600,431,495
15. LP TURBINE ELECTRICAL LOSS:	9,143,630
16. EXHAUST GAS LOSS:	222,118,950
TOTAL OUTPUT	1.678×10^9

PLANT PERFORMANCE

TOTAL COMPRESSOR INPUT (@ THE MOTOR TERMINALS):	178,540 KW
TOTAL TURBINE OUTPUT (@ GENERATOR TERMINALS):	256,555 KW
AUX. LOAD:	500 KW
NET TURBINE OUTPUT:	256,055 KW
NET LVH HEAT RATE:	3,855 BTU/KWHR

NOTES:

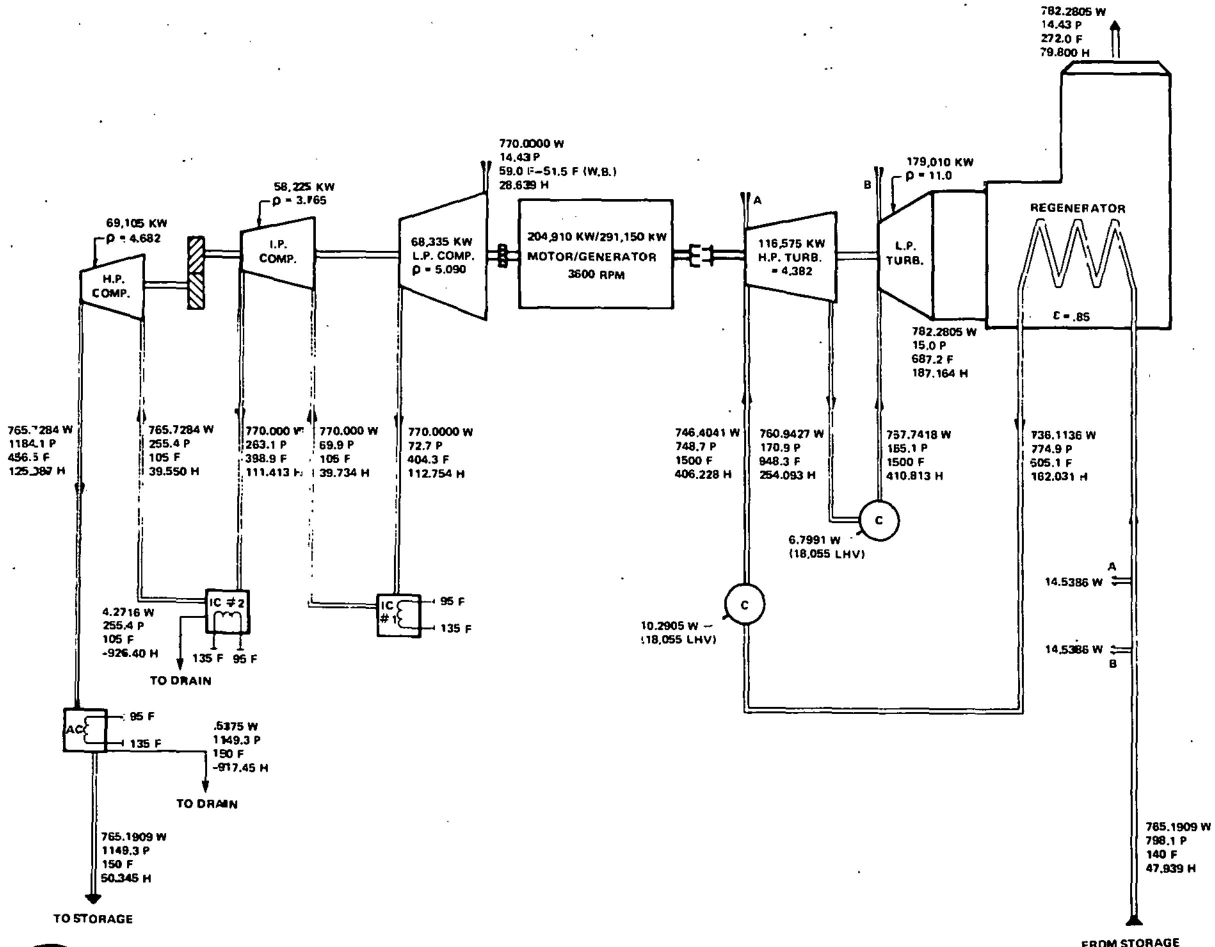
1. THE ABOVE DATA REFLECTS AS CALCULATED PERFORMANCE. GUARANTEED PERFORMANCE WOULD INCLUDE MARGINS (NOT NECESSARILY THE SAME) FOR THE POWER INPUT AND OUTPUT, AND HEAT RATE.
2. THE ABOVE REFLECTS AN ENTHALPY BASE WHERE THE ENTHALPY OF -60°F GAS EQUALS 0.0.

(W 1520)

PUBLIC SERVICE INDIANA
CAES IN AQUIFER
DOE NO. ET-78-C-01-2159

Figure 6-18. 600 PSI Nominal Refined Standardized System Heat and Material Balance

1150 PSIA CHARGING PRESSURE
1500/1500°F TURBINE INLET TEMPERATURE
EXTERNAL PLANT AIR PRESSURE LOSS RATIO = 1.4
REFINED STANDARDIZED LP SET



COMPRESSED AIR ENERGY STORAGE (CAES) POWER PLANT ENERGY BALANCE

INPUT	BTU/HR
1. ATMOSPHERIC AIR:	79,387,310
2. LP COMPRESSOR POWER (@ THE MOTOR TERMINALS):	242,880,590
3. IP COMPRESSOR POWER (@ THE MOTOR TERMINALS):	206,939,350
4. HP COMPRESSOR POWER (@ THE MOTOR TERMINALS):	249,361,390
5. HP COMBUSTOR FUEL (LHV = 18,055 BTU/LB):	668,861,920
6. LP COMBUSTOR FUEL (LHV = 18,055 BTU/LB):	441,927,905
TOTAL INPUT	1.889×10^9

OUTPUT	BTU/HR
1. LP COMPRESSOR DRIVE MOTOR LOSS:	9,715,225
2. IP COMPRESSOR DRIVE MOTOR LOSS:	8,277,575
3. HP COMPRESSOR DRIVE MOTOR LOSS:	9,974,455
4. HP COMPRESSOR GEAR LOSS:	3,590,805
5. INTERCOOLER #1 REJECTION HEAT LOSS:	202,411,440
6. INTERCOOLER #2 REJECTION HEAT LOSS:	214,611,985
7. AFTERCOOLER REJECTION HEAT LOSS:	207,908,145
8. AIR COOLER DRAIN LOSS:	-16,574,820
9. CAVITY HEAT LOSS:	6,627,780
10. HP COMBUSTOR LOSS:	6,688,620
11. LP COMBUSTOR LOSS:	4,419,280
12. HP TURBINE POWER (@ GENERATOR TERMINALS):	391,799,570
13. HP TURBINE ELECTRICAL LOSS:	5,876,995
14. LP TURBINE POWER (@ GENERATOR TERMINALS):	601,650,005
15. LP TURBINE ELECTRICAL LOSS:	9,162,185
16. EXHAUST GAS LOSS:	224,733,540
TOTAL OUTPUT	1.891×10^9

PLANT PERFORMANCE

TOTAL COMPRESSOR INPUT (@ THE MOTOR TERMINALS):	204,910 KW
TOTAL TURBINE OUTPUT (@ GENERATOR TERMINALS):	291,150 KW
AUX. LOAD:	500 KW
NET TURBINE OUTPUT:	290,650 KW
NET LVH HEAT RATE:	3,820 BTU/KWHR

NOTES:

1. THE ABOVE DATA REFLECTS AS CALCULATED PERFORMANCE. GUARANTEED PERFORMANCE WOULD INCLUDE MARGINS (NOT NECESSARILY THE SAME) FOR THE POWER INPUT AND OUTPUT, AND HEAT RATE.
2. THE ABOVE REFLECTS AN ENTHALPY BASE WHERE THE ENTHALPY OF -60°F GAS EQUALS 0.0.

(W 1521)

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CAES IN AQUIFER
DOE NO. ET-78-C-01-2159

Figure 6-19. 1000 PSI Nominal Refined Standardized System Heat and Material Balance

The results of the above changes can be observed by inspecting Tables 6-9 and 6-10, which display the standardized versus the refined standardized power production energy costs.

The single shaft configurations presented in the above figures closely reflect the actual optimum hardware layouts with the exception that the compressor inlet and outlets have been switched to improve the schematic clarity.

Table 6-9

CAES SYSTEM PERFORMANCE

STANDARDIZED COMPONENTS
(REFINED COMPONENT EFFICIENCIES)

	NOMINAL SYSTEM PRESSURE		
	200 PSI	600 PSI	1000 PSI
TOTAL COMPRESSOR INPUT AT MOTOR TERMINALS, KW	130,655	178,540	204,910
TOTAL TURBINE OUTPUT AT GENERATOR TERMINALS, KW	178,505	256,555	291,150
AUXILIARY LOAD, KW	500	500	500
NET TURBINE OUTPUT, KW	178,005	256,055	290,650
NET LHV HEAT RATE BTU/KWHR	3,945	3,855	3,820
FUEL @ $\$10^6$ BTU / ELEC. PWR. IN MILS/KWHR	TOTAL POWER PRODUCTION ENERGY COST IN MILS/KWHR		
2.50 / 10	17.20	16.61	16.60
2.50 / 20	24.54	23.58	23.65
5.00 / 20	34.40	33.22	33.20
7.50 / 10	36.93	35.89	35.70
7.50 / 20	44.27	42.86	42.75
7.75 / 60	74.61	71.71	71.91

REFINED COMPONENT EFFICIENCIES

MOTORS	96%
GENERATORS	98.5%
GEARBOXES	98.5%
COMBUSTION	99%

Work of Compression, Output Power and Heat Rate values are as calculated and do not include any margin.

(W 1550)

Table 6-10

TOTAL POWER PRODUCTION ENERGY COSTS OF THE REFINED VS THE STANDARDIZED CAES UNITS

CASE	1.10	1.20	2.10	2.20	3.10	3.20
T1HPT	1500.	1500.	1500.	1500.	1500.	1500.
T2HPT	1046.	670.	1110.	1110.	948.	948.
T1LPT	1046.	670.	1500.	1500.	1500.	1500.
CAV LOS	1.20	1.20	1.20	1.20	1.20	1.20
REG EFT	.85	.85	.85	.85	.85	.85
COMP KW	171.5	169.7	233.4	231.9	267.5	266.1
TURB KW	219.0	231.2	330.9	332.5	375.7	377.5
H.R.	4075.	3945.	3915.	3855.	3880.	3820.

FUEL COST = $2.50 PER MILLION BTU
ELECT COST = 10.00 MILS PER KW-HR — ENERGY COST RATIO = .2500

10.19	9.86	9.79	9.64	9.70	9.55
7.83	7.34	7.05	6.97	7.12	7.05
TOTAL POWER PRODUCTION COSTS IN MILS PER KW-HR					
18.02	17.20	16.84	16.61	16.82	16.60

FUEL COST = $2.50 PER MILLION BTU
ELECT COST = 20.00 MILS PER KW-HR — ENERGY COST RATIO = .1250

10.19	9.86	9.79	9.64	9.70	9.55
15.66	14.68	14.10	13.95	14.24	14.10
TOTAL POWER PRODUCTION COSTS IN MILS PER KW-HR					
25.85	24.54	23.89	23.58	23.94	23.65

FUEL COST = $4.25 PER MILLION BTU
ELECT COST = 15.10 MILS PER KW-HR — ENERGY COST RATIO = .2815

17.32	16.77	16.64	16.38	16.49	16.24
11.82	11.08	10.65	10.53	10.75	10.65
TOTAL POWER PRODUCTION COSTS IN MILS PER KW-HR					
29.14	27.85	27.29	26.91	27.24	26.88

FUEL COST = $5.00 PER MILLION BTU
ELECT COST = 17.06 MILS PER KW-HR — ENERGY COST RATIO = .2931

20.38	19.73	19.58	19.28	19.40	19.10
13.36	12.52	12.03	11.90	12.15	12.03
TOTAL POWER PRODUCTION COSTS IN MILS PER KW-HR					
33.73	32.25	31.61	31.17	31.55	31.13

FUEL COST = $5.00 PER MILLION BTU
ELECT COST = 20.00 MILS PER KW-HR — ENERGY COST RATIO = .2500

20.38	19.73	19.58	19.28	19.40	19.10
15.66	14.68	14.10	13.95	14.24	14.10
TOTAL POWER PRODUCTION COSTS IN MILS PER KW-HR					
36.04	34.40	33.68	33.22	33.64	33.20

FUEL COST = $7.50 PER MILLION BTU
ELECT COST = 10.00 MILS PER KW-HR — ENERGY COST RATIO = .7500

30.56	29.59	29.36	28.91	29.10	28.65
7.83	7.34	7.05	6.97	7.12	7.05
TOTAL POWER PRODUCTION COSTS IN MILS PER KW-HR					
38.39	36.93	36.41	35.89	36.22	35.70

FUEL COST = $7.50 PER MILLION BTU
ELECT COST = 20.00 MILS PER KW-HR — ENERGY COST RATIO = .3750

30.56	29.59	29.36	28.91	29.10	28.65
15.66	14.68	14.10	13.95	14.24	14.10
TOTAL POWER PRODUCTION COSTS IN MILS PER KW-HR					
46.22	44.27	43.47	42.86	43.34	42.75

FUEL COST = $7.75 PER MILLION BTU
ELECT COST = 60.00 MILS PER KW-HR — ENERGY COST RATIO = .1292

31.58	30.57	30.34	29.88	30.07	29.61
46.98	44.04	42.31	41.84	42.72	42.30
TOTAL POWER PRODUCTION COSTS IN MILS PER KW-HR					
78.56	74.61	72.66	71.71	72.79	71.91

NOTES:

(1) Case 1.1 has a nominal storage pressure of 200 PSI utilizing just the standardized one intercooler LP compressor set, 1500°F-UF turbine inlet temperature.
(2) Case 1.2 is exactly like case 1.1 except a single turbine and refined motor, generator and combustion efficiencies are incorporated.
(3) Case 2.1 has a nominal storage pressure of 600 PSI utilizing the standardized LP compressor set, a second intercooler and a customized HP compressor, 1500-1500°F turbine inlet temperature.
(4) Case 2.2 exactly like case 2.1 except refined motor, gearbox, generator and combustion efficiencies are incorporated.
(5) Case 3.1 has a nominal storage pressure of 1000 PSI utilizing the standardized LP compressor set, a second intercooler and a customized HP compressor, 1500-1500°F turbine inlet temperature.
(6) Case 3.2 is exactly like case 3.1 except refined motor, gearbox, generator and combustion efficiencies are incorporated.

(W 1551)

The total Work of Compression, Net Turbine Output Power and Heat values are as calculated and do not include any margins.

Section 7

MACHINERY CONFIGURATION SELECTION

DESCRIPTION OF CANDIDATE CONFIGURATIONS

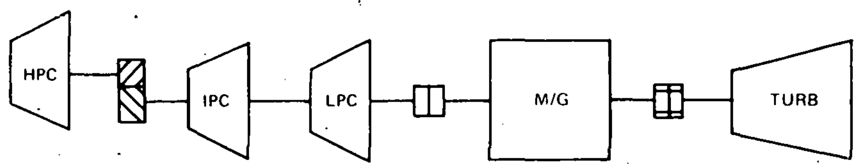
Seventeen candidate machinery configurations were initially identified for evaluation. The selection of these candidates was based upon variations of one, two and three shaft configurations with a gearbox limitation of 70 megawatts. The candidates are shown on Figures 7-1, 7-2, 7-3, 7-4 and 7-5. The nominal power ratings shown on these figures reflect the power requirements of the equipment items shown on the optimized heat and material balances in Section 6 of this report.

Three configurations for the 200 psi system are presented in Figure 7-1. The first configuration A-1, a single shaft layout, is the least complicated.

Since there is no reheat and the pressure ratio of the turbine is modest, 10.7:1 for the 200 psi system, the turbine train would consist of one element. This combination of the HP and LP turbine into one casing limits the number of possible 200 psi system machinery configuration options. Configurations B-1 and B-2 are two shaft arrangements with sections of the compressor train driven by an independent motor.

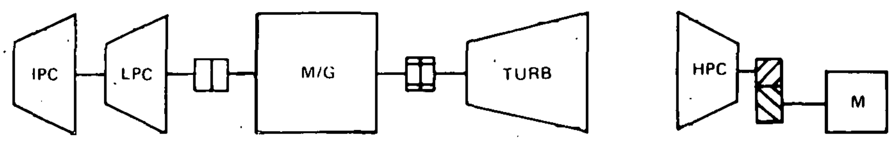
In Figures 7-2 and 7-3, seven machinery configurations for the 600 psi system are presented. There are more possible configurations for this storage pressure because of the reheat and the greater overall turbine pressure ratio of approximately 30:1. Machinery configuration A-1, the single shaft layout, is again felt to be the least complex. In configurations B-1 and B-2, the HP compressor is driven by a separate motor. The difference between configurations B-1 and B-2 is the speed of the IP compressor. In configuration B-1 a high speed IP compressor and a speed step up gear is used as opposed to the 3600 rpm IP compressor shown in B-2. Configurations B-3 and B-4 differ from B-1 and B-2 respectively by the HP turbines which drive separate generators. Configurations C-1 and C-2 are like configurations B-3 and B-4, except that the separate motors and generators were combined into motor/generator units.

Seven candidate 1000 psi system machinery configurations are presented in Figures 7-4 and 7-5. The candidate arrangements for this storage pressure are the same as for the 600 psi system.



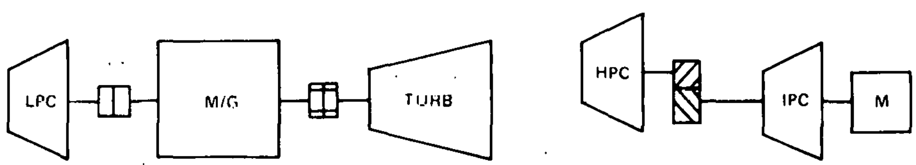
MACHINERY CONFIGURATION A-1

MW (NOMINAL)	36	36	37		49	122	122 / 172	172	172
RPM	5000	5000 / 3600	3600		3600	3600	3600	3600	3600



MACHINERY CONFIGURATION B-1

MW (NOMINAL)	37		49	86	86 / 172	172	172		36	36	36
RPM	3600		3600	3600	3600	3600	3600		5000	5000 / 3600	



MACHINERY CONFIGURATION B-2

MW (NOMINAL)	49	49	49 / 172	172	172		36	36	37		73
RPM	3600	3600	3600	3600	3600		5000	5000 / 3600	3600		3600

(W 4342)

Coupling

Clutch

250 PSIA CHARGING PRESSURE
1500/UF°F TURBINE INLET TEMPERATURE

Figure 7-1. 200 PSI System Machinery Configurations

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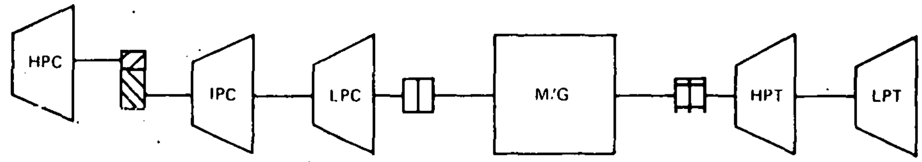
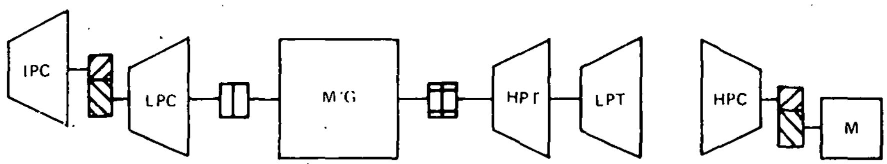


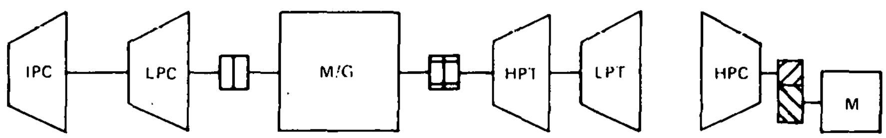
Machinery Configuration A-1

MW (NOMINAL)	53	53	53		65	171	171 / 261	261	82		179
RPM	7500	7500 / 3600	3600		3600	3600	3600	3600	3600		3600



Machinery Configuration B-1

MW (NOMINAL)	53	53	65	118	118 / 261	261	82		179		53	53	53
RPM	7500	7500 / 3600	3600	3600	3600	3600	3600		3600		7500	7500 / 3600	3600



Machinery Configuration B-2

MW (NOMINAL)	53		65	118	118 / 261	261	82		179		53	53	53
RPM	3600		3600	3600	3800	3600	3600		3600		7600	7500 / 3600	3600

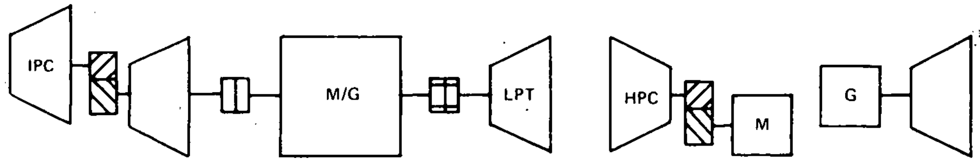
(W4347)

Coupling

Clutch

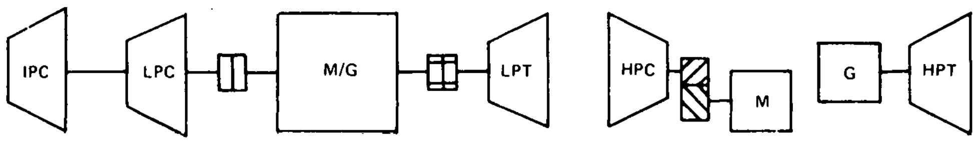
700 PSIA CHARGING PRESSURE
1500/1500°F TURBINE INLET TEMPERATURE

Figure 7-2. 600 PSI System Machinery Configurations



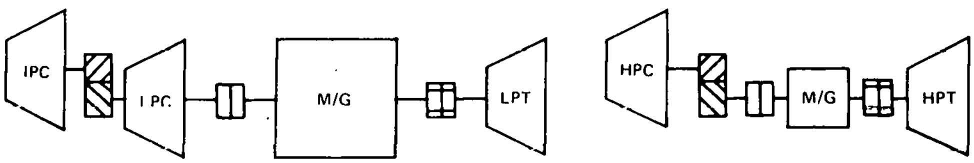
Machinery Configuration D-3

MW (NOMINAL)	53	53	65	118	118 / 179	179	179		53	53	53		82	82
RPM	7500	7500 / 3600	3600	3600	3600	3600	3600		7500	7500 / 3600	3600		3600	3600



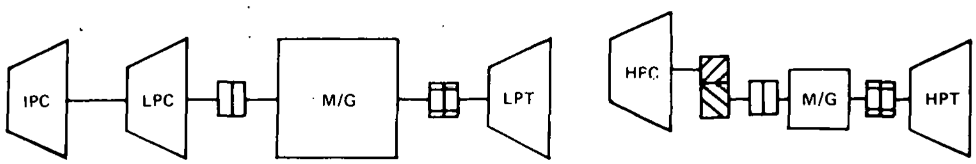
Machinery Configuration B-4

MW (NOMINAL)	53		65	118	179	179	179		53	53	53		82	82
RPM	3600		3600	3600	3600	3600	3600		7500	7500 / 3600	3600		3600	3600



Machinery Configuration C-1

MW (NOMINAL)	53	53	65	118	118 / 179	179	179		53	53	53	53 / 82	82	82
RPM	7500	7500 / 3600	3600	3600	3600	3600	3600		7500	7500 / 3600	3600	3600	3600	3600



Machinery Configuration C-2

MW (NOMINAL)	53		65	118	118 / 179	179	179		53	53	53	53 / 82	82	82
RPM	3600			3600	3600	3600	3600		7500	7500 / 3600	3600	3600	3600	3600

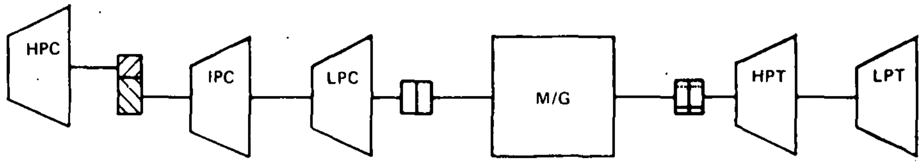
(W4346)

Coupling

Clutch

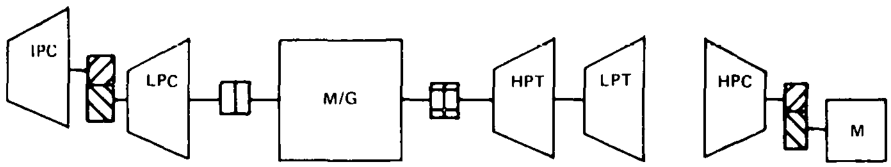
700 PSIA CHARGING PRESSURE
1500/1500°F TURBINE INLET TEMPERATURE

Figure 7-3. 600 PSI System Machinery Configurations



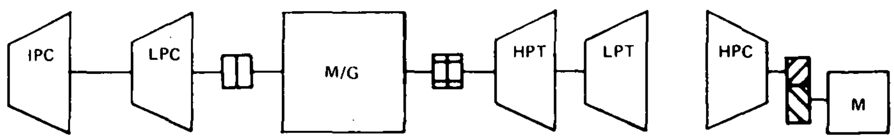
Machinery Configuration A-1

MW (NOMINAL)	61	61	62		72	195	195 / 296	296	117		179
RPM	9500	9500 / 3600	3600		3600	3600	3600	3600	3600		3600



Machinery Configuration B-1

MW (NOMINAL)	62	62	72	134	134 / 296	296	117		179		61	61	61
RPM	9500	9500 / 3600	3600	3600	3600	3600	3600		3600		9500	9500 / 3600	3600



Machinery Configuration B-2

MW (NOMINAL)	62		72	134	134 / 296	296	117		179		61	61	61
RPM	3600		3600	3600	3600	3600	3600		3600		9500	9500 / 3600	3600

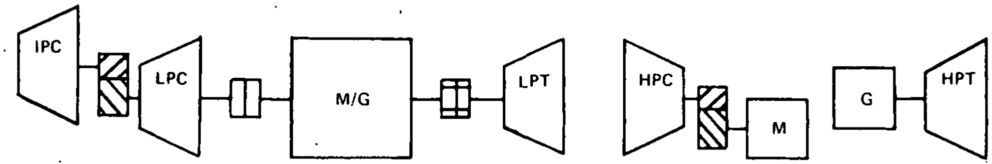
(W 4348)

Coupling

Clutch

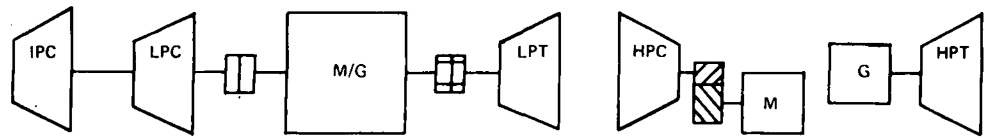
1150 PSIA CHARGING PRESSURE
1500/1500°F TURBINE INLET TEMPERATURE

Figure 7-4. 1000 PSI System Machinery Configurations



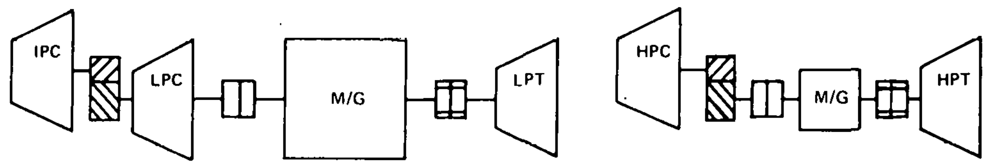
Machinery Configuration B-3

MW (NOMINAL)	62	62	72	134	134 / 179	179	179		61	61	61		117	117
RPM	9500	9500 / 3600	3600	3600	3600	3600	3600		9500	9500 / 3600	3600		3600	3600



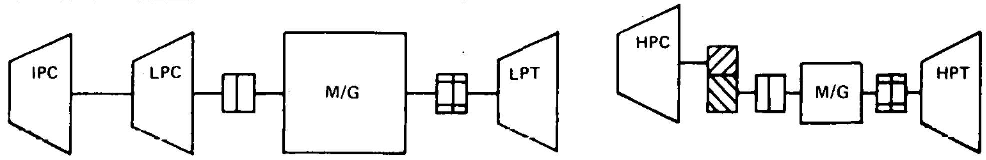
Machinery Configuration B-4

MW (NOMINAL)	62		72	134	134 / 179	179	179		61	61	61		117	117
RPM	3600		3600	3600	3600	3600	3600		9500	9500 / 3600	3600		3600	3600



Machinery Configuration C-1

MW (NOMINAL)	62	62	72	134	134 / 179	179	179		61	61	61	61 / 117	117	117
RPM	9500	9500 / 3600	3600	3600	3600	3600	3600		9500	9500 / 3600	3600	3600	3600	3600



Machinery Configuration C-2

MW (NOMINAL)	62		72	134	134 / 179	179	179		61	61	61	61 / 117	117	117
RPM	3600		3600	3600	3600	3600	3600		9500	9500 / 3600	3600	3600	3600	3600

(W4349)

Coupling

Clutch

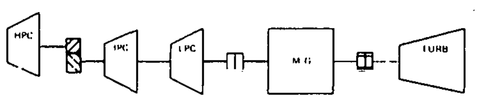
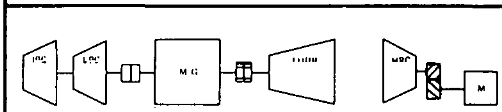
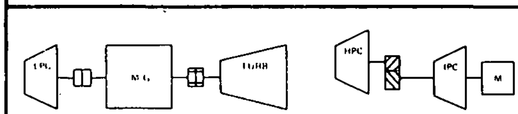
1150 PSIA CHARGING PRESSURE
1500/1500°F TURBINE INLET TEMPERATURE

Figure 7-5. 1000 PSI System Machinery Configurations

MACHINERY CONFIGURATION COMPARISON

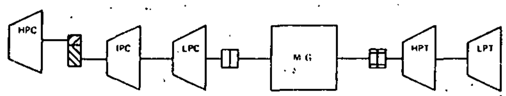
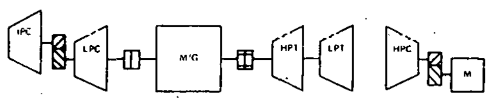
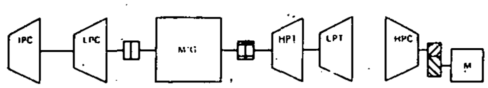
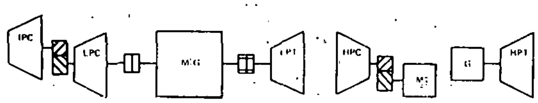
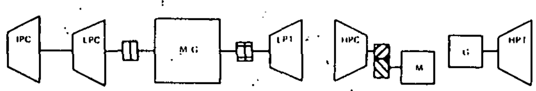
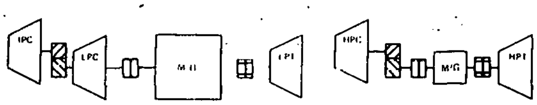
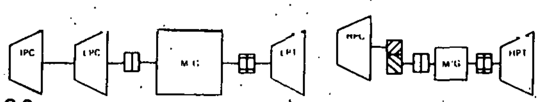
A comparison of the relative capital price of the major equipment items, the number of rotating components, the relative control complexity and the number of auxiliary equipment packages required (when they impact the comparison) is shown on Figures 7-6, 7-7 and 7-8 for the 200 psi, 600 psi and 1000 psi system pressure levels.

Budgetary pricing of the turbines and compressors for this comparison was based on the preliminary sizing of these units in terms of RPM, number of stages, and first and last stage hub and blade tip diameters. Motor, generator, and motor/generator costs were based on the required power of these units determined from the heat and material balances. Budgetary price estimates for the gearboxes, clutches/couplings, coolers and regenerators were obtained from representative equipment vendors.

200 PSI SYSTEM 250 PSIA CHARGING PRESSURE	RELATIVE MAJOR EQUIPMENT CAPITAL PRICE	NUMBER OF ROTATING COMPONENTS	NUMBER OF AUXILIARY PACKAGES	CONTROL COMPLEXITY FACTOR	OVERALL RANKING
 A-1	1.000	8	3	1.0	1
 B-1	1.126	9	4	1.0	2
 B-2	1.171	9	4	1.0	3

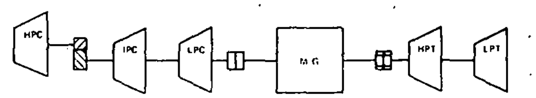
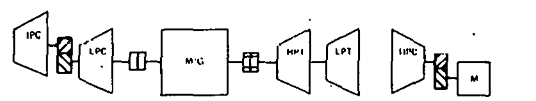
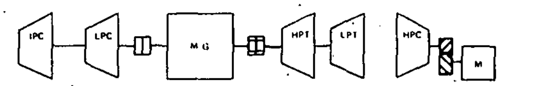
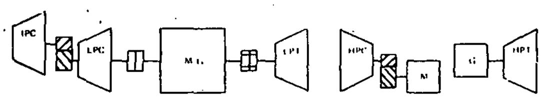
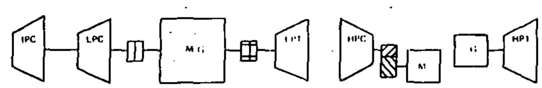
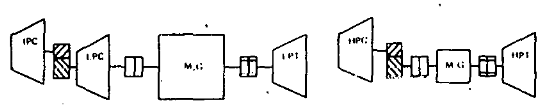
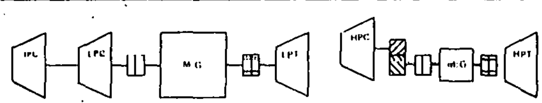
(W 4993)

Figure 7-6. 200 PSI System Machinery Configuration Comparison

600 PSI SYSTEM 700 PSIA CHARGING PRESSURE	RELATIVE MAJOR EQUIPMENT CAPITAL PRICE	NUMBER OF ROTATING COMPONENTS	NUMBER OF AUXILIARY PACKAGES	CONTROL COMPLEXITY FACTOR	OVERALL RANKING
 A-1	1.000	9	4	1.0	1
 B-1	1.110	11	6	1.0	3
 B-2	1.121	10	6	1.0	2
 B-3	1.216	12	7	1.2	7
 B-4	1.227	11	7	1.2	6
 C-1	1.128	13	6	1.2	5
 C-2	1.140	12	6	1.2	4

(W4994)

Figure 7-7. 600 PSI System Machinery Configuration Comparison

1000 PSI SYSTEM 1150 PSIA CHARGING PRESSURE	RELATIVE MAJOR EQUIPMENT CAPITAL PRICE	NUMBER OF ROTATING COMPONENTS	NUMBER OF AUXILIARY PACKAGES	CONTROL COMPLEXITY FACTOR	OVERALL RANKING
 A-1	1.000	9	4	1.0	1
 B-1	1.111	11	6	1.0	3
 B-2	1.124	10	6	1.0	2
 B-3	1.207	12	7	1.2	7
 B-4	1.222	11	7	1.2	6
 C-1	1.127	13	6	1.2	5
 C-2	1.141	12	6	1.2	4

(W4995)

Figure 7-8. 1000 PSI System Machinery Configuration Comparison

The following items were included in the price comparison:

- Compressors
- Turbines
- Motors
- Generators
- Motor/Generator
- Gear Boxes
- Clutches/Disconnect Couplings
- Coolers
- Regenerators
- Basic Control System
- Auxiliary Packages

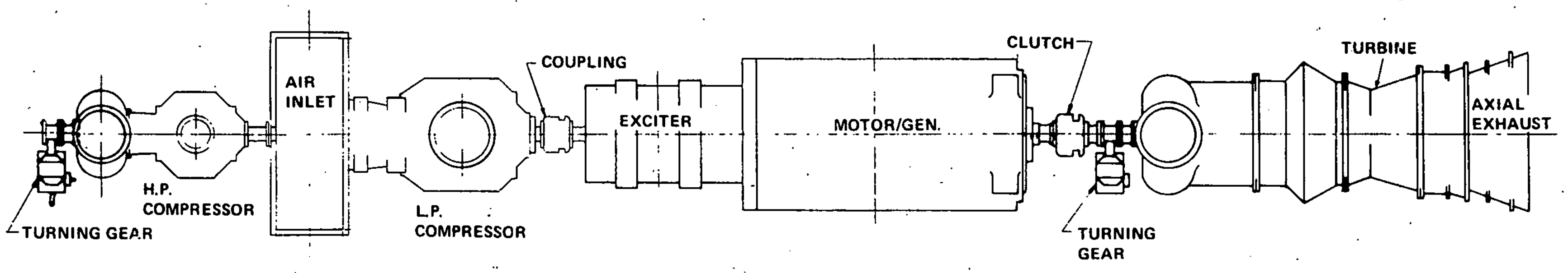
Among the price items that were not included in the comparison since they were considered to be common to all configurations to the degree that they did not impact the evaluation:

- Main Air Piping and Valves
- Electric Power Distribution and Control Equipment
- Permanent Couplings
- Inlet and Exhaust Ducting
- Stacks and Sound Treatment
- Installation Cost
- Development Cost

SELECTED MACHINERY CONFIGURATIONS

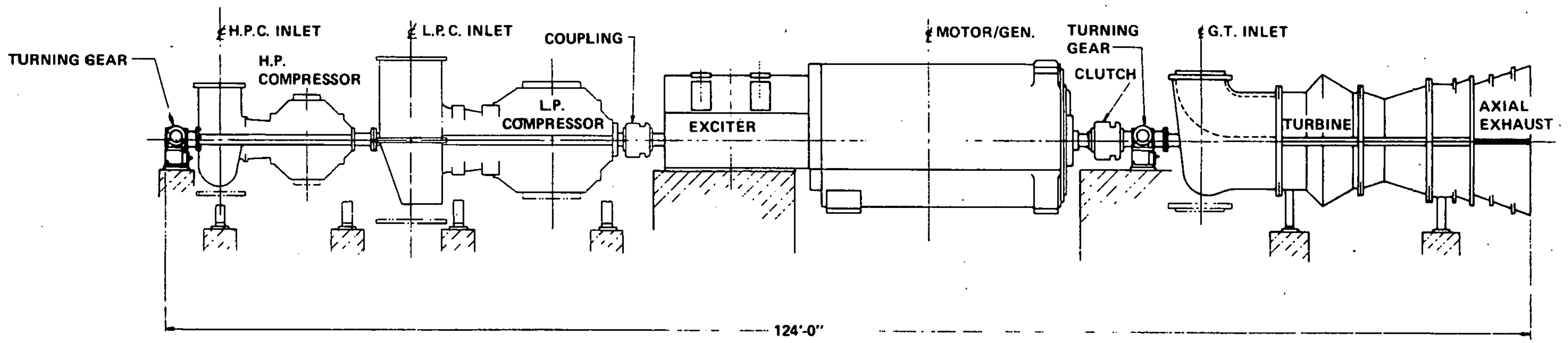
When all of the candidate configurations were reviewed, the single shaft concept was selected as the basis for the layout of the Task 1 equipment trains for each of the three pressure levels being studied. In addition to the apparent equipment capital price advantage, the single shaft configuration is also judged to be the most desirable from an operational viewpoint with respect to start up and synchronization.

Machinery outline drawings and estimates of the weight of the major components were generated for use in the preparation of plant layout drawings. These outline drawings and estimated weights, shown on Figures 7-9, 7-10 and 7-11, utilize the standardized, as opposed to the optimized, components developed in Section 6 of this report.



ESTIMATED CAES ROTATING EQUIPMENT WEIGHTS (LBS)

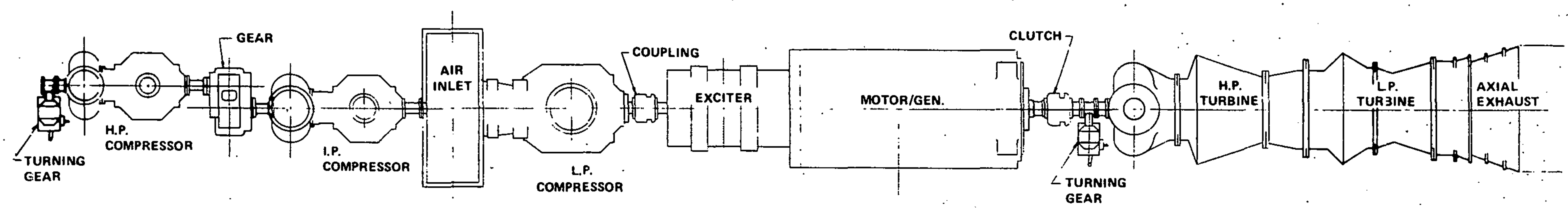
SYSTEM NOMINAL PRESSURE	HIGH PRESS. COMPRESSOR, H.P.C.	LOW PRESS. COMPRESSOR, L.P.C.	COUPLING, CO	MOTOR/ GEN., M/G	CLUTCH, CL	TURBINE, GT
200 PSI	155,000	150,000	2,500	543,900	4,250	270,000



(W 1522)

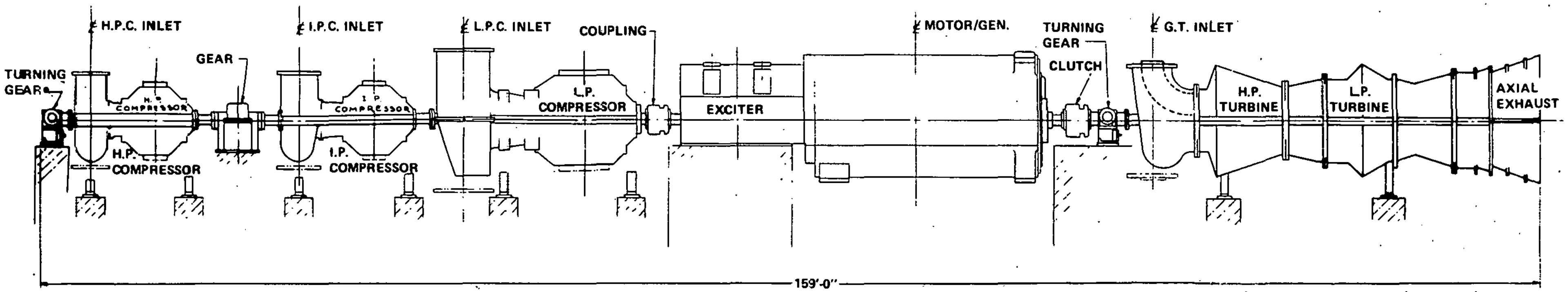
Figure 7-9. 200 PSI System Machinery Outline

PUBLIC SERVICE INDIANA
CAES IN AQUIFER
DOE NO. ET-78-C-01-2159



ESTIMATED CAES ROTATING EQUIPMENT WEIGHTS (LBS)

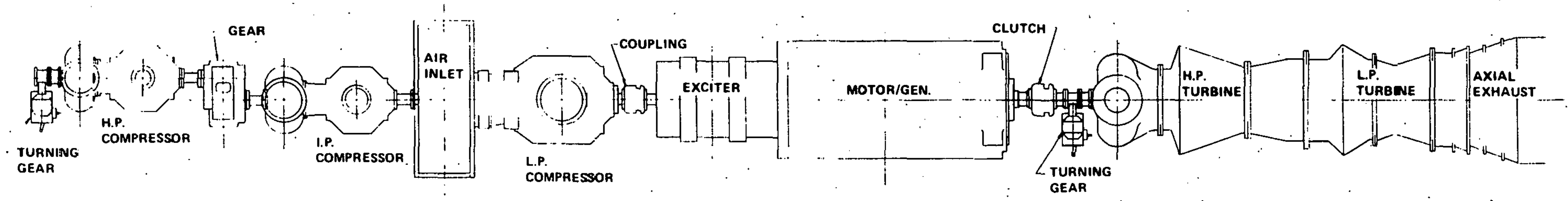
SYSTEM NOMINAL PRESSURE	HIGH PRESS. COMPRESSOR, H.P.C.	GEAR BOX, G.B.	INTER. PRESS. COMPRESSOR, I.P.C.	LOW PRESS. COMPRESSOR, L.P.C.	COUPLING, CO	MOTOR/ GEN., M/G	CLUTCH, CL	TURBINE, GT
600 PSI	135,000	33,000	155,000	150,000	4,250	639,700	6,000	370,000



(W 1553)

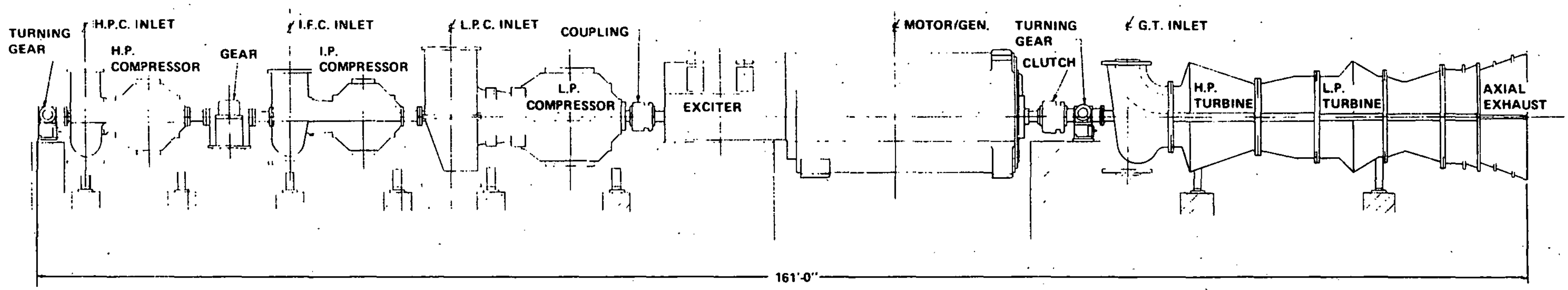
Figure 7-10. 600 PSI System Machinery Outline

PUBLIC SERVICE INDIANA
CAES IN AQUIFER
DOE NO. ET-78-C-01-2159



ESTIMATED CAES ROTATING EQUIPMENT WEIGHTS (LBS)

SYSTEM NOMINAL PRESSURE	HIGH PRESS. COMPRESSOR, H.P.C.	GEAR BOX G.B.	INTER. PRESS. COMPRESSOR, I.P.C.	LOW PRESS. COMPRESSOR, L.P.C.	COUPLING, CO	MOTOR/ GEN., M/G	CLUTCH, CL	TURBINE, GT
1000 PSI	135,000	33,000	155,000	150,000	4,250	639,700	6,000	380,000



(W 1552)

Figure 7-11. 1000 PSI System Machinery Outline

PUBLIC SERVICE INDIANA
CAES IN AQUIFER
DOE NO. ET-78-C-01-2159

Section 8

ELECTRICAL ROTATING EQUIPMENT APPLICATION

GENERAL

It is the purpose of this section to review the electrical rotating machinery requirements for the plant configurations chosen in Section 7 and to provide preliminary recommendations concerning the feasibility and relative advantages of these configurations from an electric machinery standpoint.

The Compressed Air Energy Storage (CAES) plant, Figure 8-1, will function as a basic combustion turbine-generator with pumped storage features. The traditional combustion turbine-generator plant consists of a combustion turbine with self-contained compressor which drives an electric generator. During normal operation, approximately 2/3 of the mechanical output of the turbine is consumed in driving its own compressor. This leaves 1/3 of the turbine energy available as shaft power to run the generator. A typical gas turbine plant might have a 300 MW combustion turbine which would drive an integral 200 MW compressor and produce 100 MW net mechanical output. This mechanical output would run a 100 MW synchronous electrical generator. The net electrical output of the plant would be about 100 MW (less auxiliary loads) of electrical power.

The CAES plant has altered this equipment configuration to allow increased electrical output. In the CAES plant, the combustion turbine with its integral compressor is separated into its two distinct pieces of equipment. The compressor or turbine can be selectively removed from the drive train, and a compressed air storage reservoir is added. The plant can then be operated in the following manner.

With the turbine decoupled from the system, the synchronous generator is operated as a synchronous motor driving the compressor which fills the compressed air storage reservoir. In this manner energy is stored during periods of low system power demand.

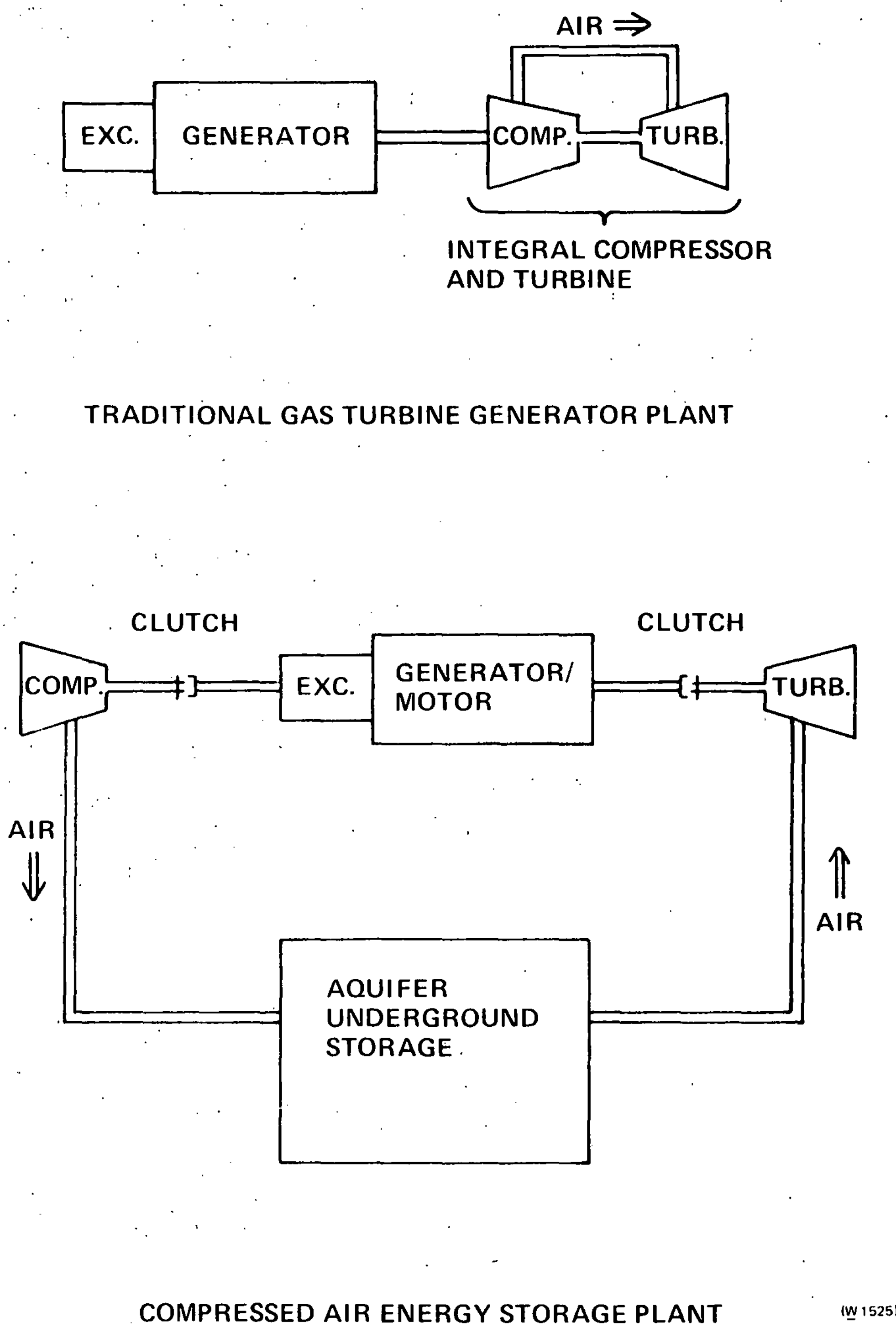


Figure 8-1. CAES Equipment Arrangement

When electric power is required during periods of high power demand, the compressor is decoupled from the power plant. The turbine then draws its air from the storage reservoir, adds fuel and burns this mixture in a conventional manner. Turbine output power is used to drive a synchronous generator and deliver electrical power to the power system. Using this CAES equipment arrangement, compressor work is obtained from relatively inexpensive power available during light power system loads. During peak load hours, the turbine is operated without the associated compressor load and, the more expensive gas turbine fuel can be used to produce electrical energy and not used to power the compressor.

An additional advantage lies in the ratings of the various pieces of equipment. In a conventional combustion turbine-generator plant a 300 MW turbine drives a 200 MW compressor and a 100 MW generator. Using the CAES plant concept, the same 300 MW turbine, free from the normal compressor load, would drive a 300 MW generator in the generating mode. The turbine can thus drive a generator of approximately three times its standard generator rating. In the pumping mode, the 300 MW generator/motor is used as a motor to drive a 200 MW compressor which recharges the storage reservoir.

ELECTRICAL FEASIBILITY OF PROPOSED MACHINERY CONFIGURATIONS

The three machinery configurations shown on Figure 8-2 were investigated.

Preliminary screening of the three configurations concluded that all three are feasible from an electrical viewpoint and presented no overwhelming electrical design problems. Present technology will allow the design and operation of the electrical equipment in each configuration. Some plant designs were preferred over others. System A is the least complex and expensive system from an electrical equipment standpoint.

Again from an electrical standpoint, System B is the least economical in that the high pressure generator and high pressure motor would be essentially identical in design and could, therefore, be combined into the single unit shown in System C. The two most viable systems from an electrical design viewpoint were, therefore, Systems A and C.

Specific comments regarding the feasibility of each system are now presented. The assumed plant rating was 300 MW, which approximates the turbine power output

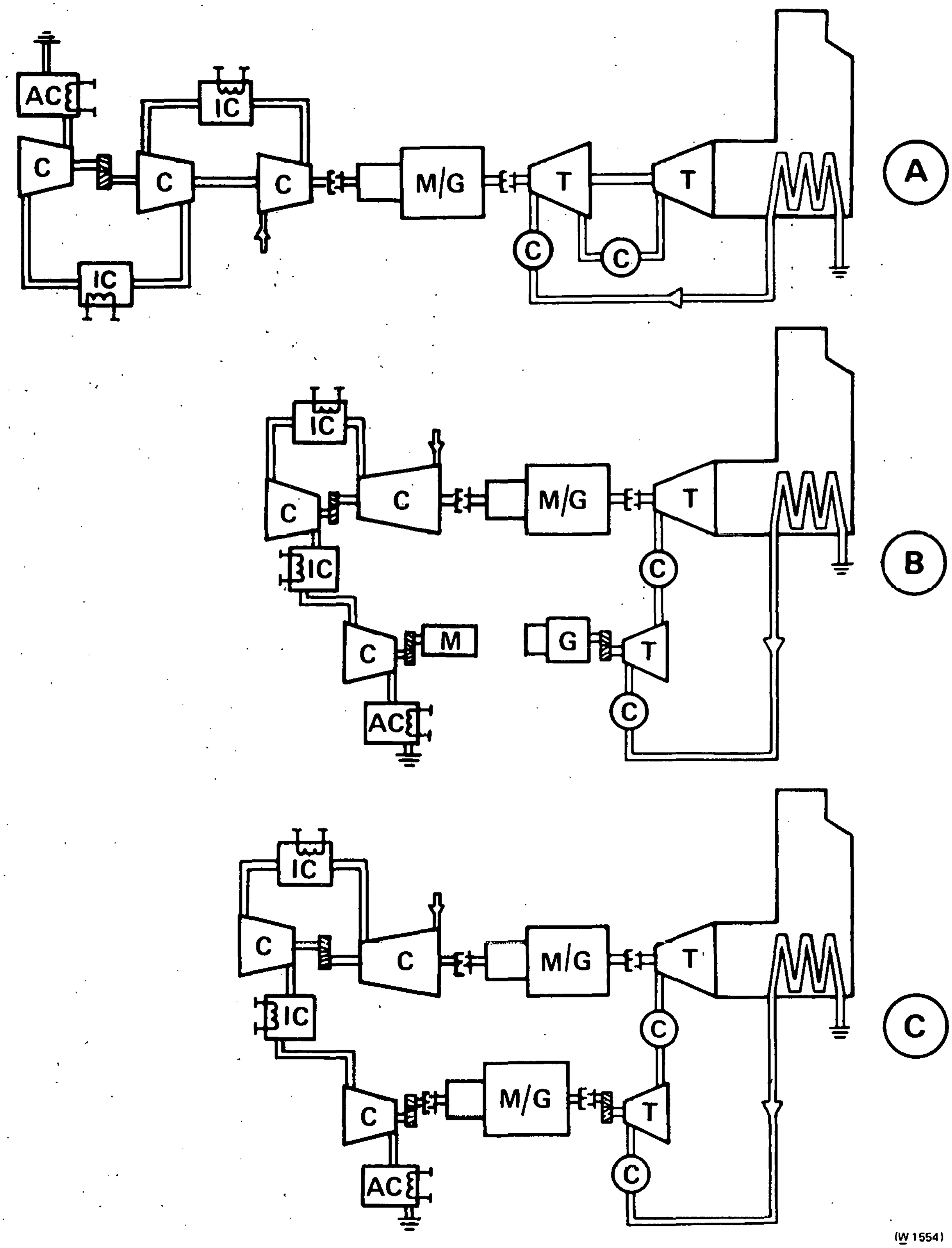


Figure 8-2. Basic Machinery Configurations

in the 1000 psi storage system. Although other ratings are being considered to accommodate the characteristics of the storage reservoir, comments made concerning the 300 MW system are applicable to these other plant ratings.

SINGLE SHAFT CONFIGURATION - SYSTEM A

System A is the most desirable system from the electrical standpoint in that it contains the fewest number of components. A hydrogen cooled generator rated at 300 MW can be built with present technology and can also serve as a synchronous motor with a maximum rating of approximately 300 MW. The generator/motor will not be self-starting since the rotor damper circuits could not withstand the severe eddy current heating induced during such a startup. In all three systems, a separate starting motor would be required unless the turbine were used to start the unit from standstill. System A is capable of operating independently of the underground storage, at the reduced electrical power output of approximately 100 MW. This system is the least expensive of the three as well as requiring less instrumentation than the other two more complex systems.

THREE SHAFT CONFIGURATION - SYSTEM B

System B is the least desirable of the three systems from an electrical equipment standpoint. With the high pressure equipment in the range of 30 MW or above, startup will be difficult. Self-starting motors have been built in the range of 60,000 HP, but these are slow speed salient pole devices which would greatly complicate the gearbox requirements for the compressor drive. System B would thus require an expensive starting package.

As a practical matter, the high pressure motor and generator would be designed as synchronous devices, virtually identical in construction. Considerable savings would be achieved by combining these two devices into a single machine as in System C. System B with its separate motor and generator is the most expensive and complex of the three systems.

TWO SHAFT CONFIGURATION - SYSTEM C

System C is designed to provide separation of low pressure and high pressure equipment. Since the generator cannot operate at speeds greater than 3600 RPM, a speed increasing gear is required as shown. System C, from an electrical equipment viewpoint, is however more expensive than System A.

DUTY CYCLES

As a Task 1 assumption the CAES plant will be dispatched in a manner similar to that of a conventional combustion gas turbine generator (full-load whenever generating). The units would normally operate at rated capability approximately 20 hours a day, 6 days a week. Under these assumptions the electric machinery in a CAES plant would undergo as many as 500 to 600 starts per year (250 to 300 starts as a generator and an equal number of starts as a motor to recharge the aquifer). The effects of this thermal cycling on the electrical equipment must be carefully considered in order to assure that the machinery has an acceptable operating life. At present, gas turbine generators are designed to withstand 500 starts per year for 30 to 40 years. This duty cycle is quite comparable to the duty cycle expected for the CAES plant. It is expected that electrical equipment in the CAES plant can be successfully designed to meet this cyclic duty. An accurate determination of the CAES dispatch schedule is, however, necessary if the electrical equipment is to be designed to meet the realistic thermal cycling duty. If the plant is started several times a day, to trim multi-peak load curves, the number of thermal cycles could well exceed the 500 to 600 cycles per year presently assumed. This increased cycling must be accounted for in the design of the equipment. In addition, a detailed knowledge of the dispatch schedule will be needed to determine the true economic value of the CAES plant. This detailed data will be supplied by the Sponsoring Utilities and will be incorporated into the detailed design of the electrical equipment.

PART LOAD OPERATION

The possibility of varied compressed air conditions in the storage reservoir (low reservoir pressure for example) may add special constraints to the design of the turbines if they are to operate under these conditions. These special considerations do not carry over into the design of the electric machinery. A generator can easily operate over a continuous range of MW loads ranging from zero load through full load. Therefore, although "off load" operation may be of concern to the turbine designer, it presents no special problems to the generator/motor designer. It is anticipated that the generator/motor designs for the CAES plant will have traditional characteristics with regard to MW and MVAR capability. As with gas turbine generators, maximum unit rating will be affected by the ambient temperatures of the generator cooling fluid (air or water).

One special case of "off load" capability for the electrical equipment is the ability of the generator to function as a synchronous condenser, supplying only reactive power to the system. In this mode the generator is brought to synchronous speed and declutched from the turbine. The desirability of incorporating this function will depend on the utilities system requirements.

STABILITY

Since the generators selected for the CAES plant will employ relatively standard designs no unusual stability problems are anticipated due to generator reactances. No unusual problems are presented by the design of a two generator plant such as shown in configurations B and C of Figure 8-2. The unusual equipment configuration will, however, require a close look at system stability in all three CAES plant configurations. In the generator mode, the compressor is decoupled from the turbine-generator combination and the total inertia of the rotating equipment is therefore less than in a conventional turbine generator drive train. The situation will be studied in detail in Tasks 3 and 5 of this program. Once the site selection is made and power system characteristics are known, a transient and dynamic stability analysis will be conducted to ensure the satisfactory performance of the system.

STARTUP

The question of how the rotating equipment is started was considered. Under present assumptions the plant will be brought to standstill twice a day for changeover to the alternate operational mode. When the plant is started a method must be found by which the rotating equipment can be brought to synchronous speed. Several methods were considered:

1. Use of generator/motor as a starting motor.
2. Use of generator/motor as a starting motor with other power plants (synchronous starts).
3. Use of auxiliary starting motors.
4. Use of turbine with aquifer air to start plant.
5. Use of exciter as starting motor.

At present, Method 4, use of turbine with aquifer air to start plant, appears the most inexpensive and practical. The present assumption of this study is that this method will be employed.

SUMMARY OF PLANT CONFIGURATION STUDY

Given the availability of clutches and gearboxes of appropriate ratings, all three proposed systems are feasible from an electrical standpoint. The three systems have certain advantages and disadvantages relative to each other.

System A (single shaft system) is the least complex and expensive. System B is the most expensive and contains expensive duplication of equipment since the high pressure generator and motor would be almost identical in construction and would not operate at the same time. Thus, System B would most likely be discarded in favor of System C. System C, however, is more complex and expensive than System A.

Table 8-1 shows the relative ranking of the three proposed plants with respect to the application of the electrical machinery.

Table 8-1

COMPARISON OF ELECTRICAL EQUIPMENT IN THREE CAES SYSTEM CONFIGURATIONS

RATINGS: +, 0, - SHOW RELATIVE ADVANTAGES

PARAMETER	SYSTEM A	SYSTEM B	SYSTEM C
COST	+	-	0
SIMPLICITY	+	-	0
OFF LOAD OPERATION (ELECTRICAL)	+	+	+
STABILITY	+	0	0
OPERATION INDEPENDENT OF CAVERN	+	0	0
STARTUP SYSTEM COST	+	-	0

PRELIMINARY ELECTRIC MACHINERY DATA

Studies conducted in Section 7 have selected the single shaft plant configuration for further development. The rotating electric machinery consists of a single synchronous machine that functions as both a generator and motor. This section of the report explores in some detail the design and application requirements of this synchronous machine. As will be concluded, a relatively standard synchronous generator can be used in the CAES plant. Areas where modifications may be required are identified and will be studied in detail in Task 5. It is felt that the few required modifications can be readily accomplished using existing technology.

Although a single plant configuration has been selected for study, some uncertainty exists in the characteristics of the underground air storage site. Accordingly, three plant ratings were developed based on nominal storage pressures of:

200 PSI

600 PSI

1000 PSI

Plants designed for these different pressures have different output ratings and equipment design. Choosing the optimum equipment design was the subject of a comprehensive study performed by the Westinghouse Combustion Turbine Systems Division. The impact of this study on the electric machinery design lies chiefly in the different generator/motor ratings required for the three different plants as shown in Table 8-2.

The rotating machinery studied in this task is limited to the synchronous generator/motor. It is assumed at this time that the turbine will be used to accelerate the electric machinery to synchronous speed in all plant operating modes and that no large starting motors will be required.

Table 8-2

GENERATOR/MOTOR RATING REQUIREMENTS FOR CAES SINGLE SHAFT SYSTEMS

Electric Machinery Rating (MW)	Nominal Storage Pressure		
	200 PSI	600 PSI	1000 PSI
Generator/Motor	178.5 Gen. 130.7 Motor	256.6 Gen. 178.5 Motor	291.1 Gen. 204.9 Motor

APPLICATION OF SYNCHRONOUS GENERATOR/MOTOR TO CAES SYSTEM

The CAES plant is a complex mechanical and thermodynamic system. Design of turbines, compressors, regenerators, and intercoolers must be carefully done to assure high plant efficiency and reliability. The design of the rotating electric machinery is much less complex. In effect, it doesn't matter what system of machinery drives the generator as long as mechanical power is provided to its input shaft at an appropriate RPM. Likewise, the synchronous machine can operate as a motor to drive any reasonable mechanical load once it has been accelerated to its nominal speed and coupled to this load.

As previously stated, the synchronous generator can function well as a synchronous motor with only a slight decline in its maximum rating. Fortunately, the relative ratings of the turbine and compressors in the CAES plant require the motor load (Compressors) to be only 70% of the generator output rating. A single synchronous generator/motor with sufficient generator capability will readily serve as a synchronous motor at 70% load.

In selecting synchronous generators for the three plant ratings, an attempt was made to use standard "off the shelf" generator designs when possible to minimize uncertainty in the plant's performance characteristics and to insure high plant reliability.

Given the generator output and motor output requirements of the three different storage pressure plant designs, synchronous generator/motors have been chosen as described in Table 8-3. The generator/motors chosen represent designs that are economical and efficient at the given specified ratings. Voltages are the standard voltage for each rating. A step up transformer will be required to link the generator with the transmission system. At present, no definite plant site or transmission voltage has been selected. It is assumed that a standard generator voltage will be acceptable. If compelling reasons emerge, non-standard generator voltages can be accommodated.

Generator reactances, time constants, and output capability characteristics of the generators are typical of generators with these ratings and no unusual requirements are expected. If plant siting considerations should require special generator characteristics, these can be incorporated as they would be for conventional generating plants. Table 8-4 and Figure 8-3 display the stability data and generator capability curve for the 329 MVA generator/motor (1000 psi system). Data

Table 8-3

CANDIDATE GENERATORS FOR CAES SYSTEMS

System	Generator Description		
200 PSI	193 MVA	18 K.V.	Hydrogen - Indirectly Cooled
- Single Shaft	.90 P.F.	3600 RPM	Cooled Stator & Rotor
		or	
Generator = 173 MW	193 MVA	20 K.V.	Hydrogen Inner Cooled Stator
Motor = 125 MW	.90 P.F.	3600 RPM	Hydrogen Inner Cooled Rotor
600 PSI	290 MVA	22 K.V.	Hydrogen Inner Cooled Stator
- Single Shaft	.90 P.F.	3600 RPM	Hydrogen Inner Cooled Rotor
Generator = 261 MW			
Motor = 171 MW			
1000 PSI	329 MVA	22 K.V.	Hydrogen Inner Cooled Stator
- Single Shaft	.90 P.F.	3600 RPM	Hydrogen Inner Cooled Rotor
Generator = 296 MW			
Motor = 196 MW			

(W 1595)

for the other two generators is also conventional and is not included in this report. Basic footprint sketches for the generator/exciter combinations have been identified. These footprint drawings are included as Figures 8-4 and 8-5. Generator footprints for the 600 psi and 1000 psi systems are the same.

Two possible generators have been shown for the 200 psi system. While both are hydrogen cooled, one is indirectly cooled and the other is inner cooled. A final selection will be made in Task 3 if the 200 psi system is selected as the demonstration plant.

Table 8-4

GENERATOR/MOTOR STABILITY DATA - 1000 PSI CAES SYSTEM

Gen. Rating:	KVA	329,000	P.F.	0.90	P.S.I.G.	63	
KV	22	Hertz	60	RPM	3600	S.C.R.	0.58

Full-Load Field Amperes at Rated Load 3062

Field Amperes at Rated Voltage - Air Gap Line 1028

Field Voltage at Rated Load 399

Gen. + Exc. Inertia WR^2 81000 $LB\text{-}FT^2$

Field Res. (75°C)	.1243	Exciter Rated KW	1430
Saturation Curve No.	662755-A	Exciter Rated Volt	425
*X_d	174.2%	X''_{qi}	26.4%
*X_q	171.5%	X_2	24.3%
*X'_{dv}	28.8%	X_o	12.9%
X'_{di}	32.7%	*X_ℓ	20.1%
*X'_{qv}	46.8%	*r_a	.143%
X'_{qi}	53.2%	*T_{do}'	5.016 Sec
*X''_{dv}	24.4%	*T_{qo}'	.557 Sec
X''_{di}	26.6%	*T_{do}''	.041 Sec
*X''_{qv}	24.2%	*T_{qo}''	.068 Sec

*Data of particular importance in stability studies.

DESIGN CONSIDERATIONS UNIQUE TO ELECTRICAL MACHINERY FOR THE CAES PLANT

Three relatively standard generators have been identified for use in the CAES plants. While the CAES application presents no major problems for the rotating electric machinery, there are several design considerations which present unique requirements. While each area can be solved with existing technology, special considerations will be given to these areas as described below.

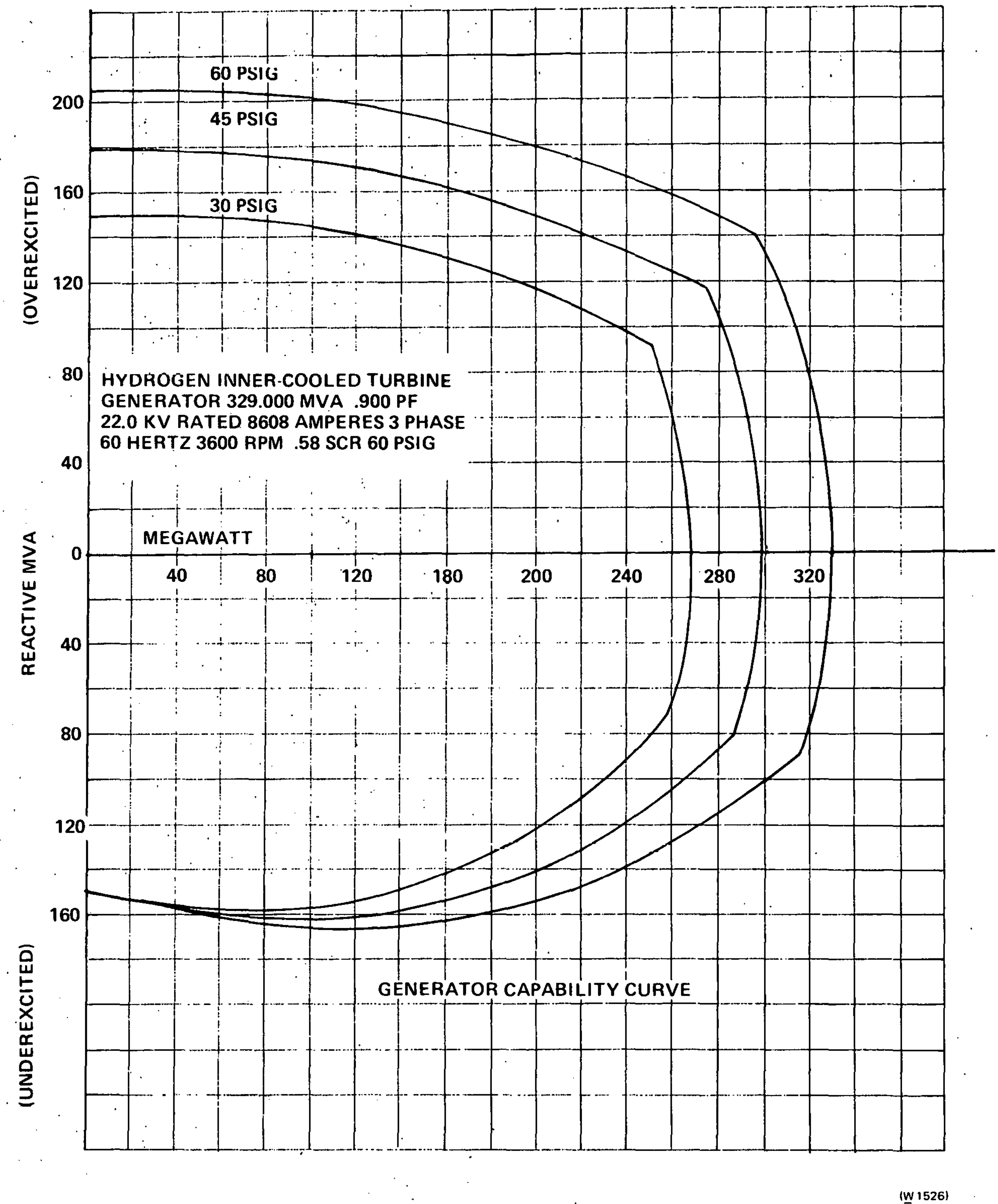


Figure 8-3. 1000 PSI CAES System

REGULATOR FOR GENERATOR/MOTOR

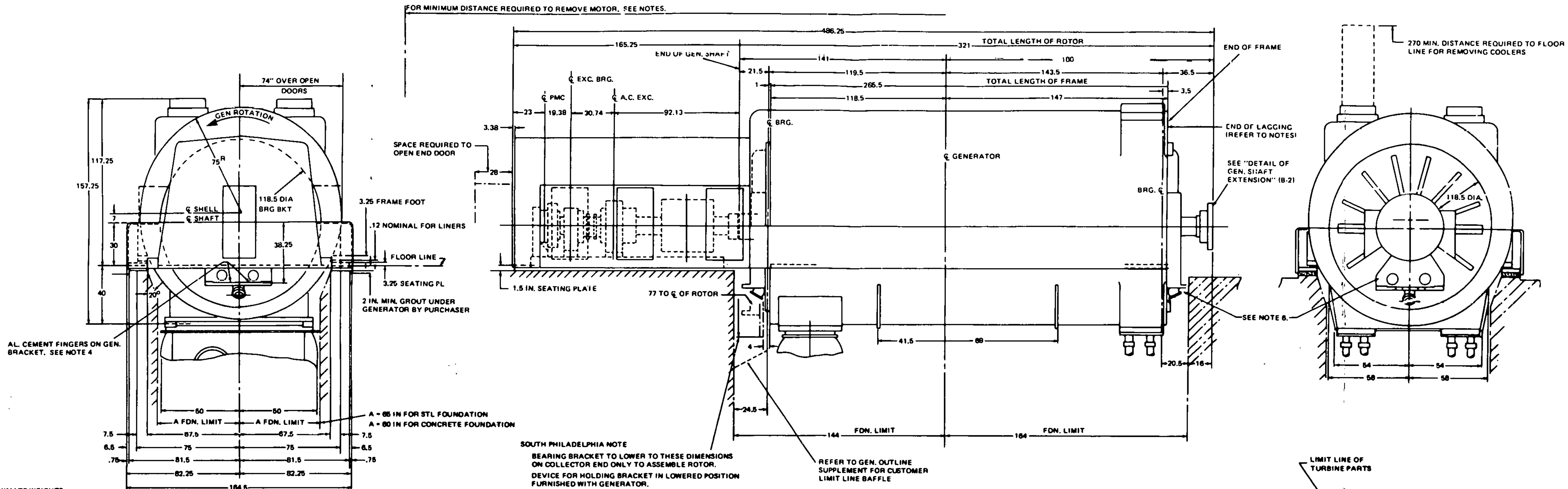
As discussed, the operation of a synchronous generator as a motor presents no problems to the generator itself. Minor modifications to the voltage regulator are required to allow it to control the generator in its motor mode. When the plant is operated in the motor/compressor mode it might be desirable, depending on the plant site and power system, to control the plant power factor. If this capability is desired, a power factor regulator feature must be added to the voltage regulator. This is a minor addition. It may be desirable to operate the synchronous machine as a synchronous condenser (neither a generator nor a motor but as a source of reactive power). If this is desired, the regulator will have to be modified slightly to include this feature. When final decisions are made concerning the modes in which the generator/motor will be operated, the standard voltage regulator design can be easily altered to include the desired modes of operation.

EXCITER

Field current will be supplied to the generator/motor by a standard rotating brushless exciter. Static excitation could be provided if requested. The unique arrangement of rotating machinery in the CAES plant will require mechanical modifications of the standard exciter design. The final analysis will be performed during Task 3.

In a standard equipment arrangement, Figure 8-6, the 300 MW turbine is connected directly to the 200 MW integral compressor and the shaft between the two is designed to transmit 300 MW. The net torque at the generator shaft input is approximately 101.5 MW. The shaft between the generator and the exciter must carry 1.5 MW (the exciter load) and the exciter shaft and rotor body must be designed to carry only the exciter power of approximately 1.5 MW.

The exciter rotor in the CAES arrangement must carry much more mechanical power than it carries in a conventional plant design. As shown in Figure 8-6, the exciter rotor must carry the full load of the compressor (approximately 200 MW for the 1000 psi system). The CAES exciter rotor must, therefore, be redesigned to carry this increased load.



APPROXIMATE WEIGHTS:

TOTAL GENERATOR WITH EXCITER	543,900
GENERATOR ASSEMBLED	500,700
EXCITER ASSEMBLED	43,200
HEAVIEST PIECE (WOUND STATOR, LESS COOLERS, BRG BRKTS, ETC.)	349,000
GENERATOR ROTOR	65,100

DISTANCE FROM ℄ GENERATOR REQUIRED FOR FOLLOWING OPERATIONS:

TO REMOVE ROTOR STRAIGHT OUT	444"
TO REMOVE ROTOR BY SKEWING	370" x 196"

SEE GENERATOR OUTLINE SUPPLEMENT FOR THE FOLLOWING INFORMATION –

(1) GAS CONNECTIONS
(2) ELECTRICAL CONNECTIONS
(3) WATER CONNECTIONS
(4) OIL CONNECTIONS
(5) EXCITER AIR FLOW
(6) OPENINGS IN FOUNDATION FOR EXCITER & COLLECTOR & ELECTRICAL LEADS.

SOUTH PHILADELPHIA NOTE:

ALL INSULATION FOR TE & EE BEARINGS IS PROVIDED WITH GENERATOR. THERMOMETERS FOR BEARING OIL DRAIN LINE SUPPLIED BY SO. PHILA. WORKS WHEN REQUIRED BY CONTRACT. SEE GEN. OUTLINE SUPPLEMENT FOR THERMOMETER DETAIL.

NO SEAL OIL PRESSURE GAUGE LINES ARE REQUIRED BETWEEN GENERATOR AND OTHER EQUIPMENT.

NOTES:

1. REFER TO EAST PGH DWG 620J157 FOR METHOD OF MOUNTING GENERATOR ON FOUNDATION FOR SECTIONS L-L, O-O, D-D, R-R, V-V, G-G, J-J, W-W, A-A & AMS. "Z", "X-X".
2. SEE METHOD OF SHIPPING AND HANDLING FOR SHIPPING WEIGHTS, SHIPPING CLEARANCES AND METHOD OF HANDLING.
3. REFER TO EAST PGH. DWG. 714C890 FOR JOINT BETWEEN GENERATOR AND TURBINE LAGGING.
4. REFER TO EAST PGH. DWG. 215C591 FOR TRANSVERSE ANCHOR DETAILS.
5. APPROVAL
   SOUTH PHILADELPHIA WORKS TO APPROVE OUTLINE AND ASSEMBLIES
   ASSEMBLY "A" – GENERATOR WITHOUT VIBROMETER
   ASSEMBLY "B" – GENERATOR WITH WESTINGHOUSE VIBROMETER WITH SHAFT RIDER. (REFER TO EAST PGH. DWG. 520D782 LINE 3 FOR VIBROMETER PICKUP INFORMATION.)
   ASSEMBLY "C" – COUPLING DWG. 393C310..
   ASSEMBLY "D" – COUPLING DWG. 393C360.
   ASSEMBLY "E" – COUPLING DWG. 393C361.
   ASSEMBLY "F" – EXCITER WITH WESTINGHOUSE VIBROMETER. WITH SHAFT RIDER. (REFER TO E. PGH., DWG. 897D593 LINE NO. 1 FOR VIBROMETER PICKUP INFORMATION.)
6. REFER TO BRG. BRKT OUTLINE DWG. 215C542.

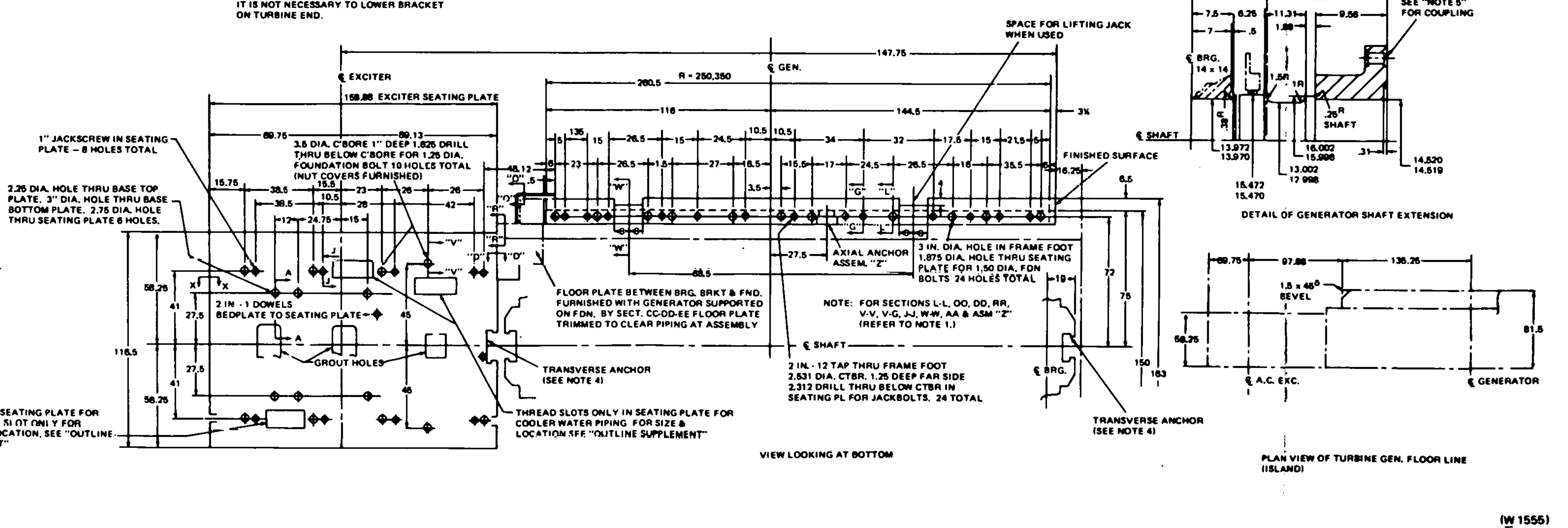


Figure 8-4. Generator and Exciter Footprint 200 PSI CAES Plant

PUBLIC SERVICE INDIANA
CAES IN AQUIFER
DOE NO. ET-78-C-01-2159

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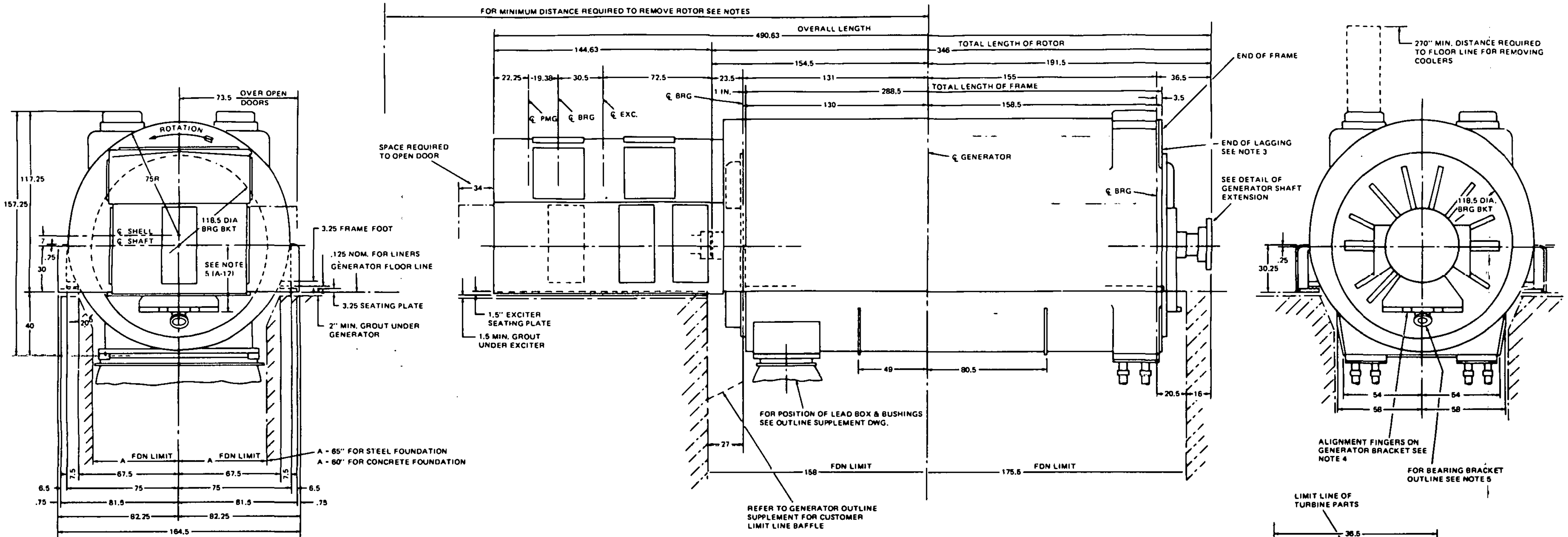

APPROXIMATE WEIGHTS

TOTAL GENERATOR WITH EXCITER	639,700 LB.
GENERATOR ASSEMBLED	593,600 LB.
EXCITER ASSEMBLED	46,500 LB.
HEAVIEST PIECE(WOUND STATOR LESS COOLERS, BRG, BKTS, ETC.)	435,000 LB.
GENERATOR ROTOR	74,300 LB.

DISTANCE FROM ℄ GENERATOR REQUIRED FOR FOLLOWING OPERATIONS

TO REMOVE ROTOR STRAIGHT OUT	480 IN.
TO REMOVE ROTOR BY SKEWING	440 IN. x 185 IN.

SEE GENERATOR OUTLINE SUPPLEMENT FOR THE FOLLOWING INFORMATION

GAS CONNECTIONS	OIL CONNECTIONS
ELECTRICAL CONNECTIONS	OPENING IN EXCITER SEATING PLATE FOR ELECTRICAL
WATER CONNECTIONS	CONNECTIONS AND WATER PIPING
EXCITER AIR FLOW	

SOUTH PHILADELPHIA NOTE.

ALL INSULATION FOR I.B. & O.B. BEARINGS IS PROVIDED WITH GENERATOR. THERMOMETERS FOR BEARING OIL DRAIN LINE SUPPLIED BY SO. PHILA. WORKS WHEN REQUIRED BY CONTRACT.

SEE GENERATOR OUTLINE SUPPLEMENT FOR THERMOMETER DETAILS.

NO SEAL OIL PRESSURE GAUGE LINES ARE REQUIRED BETWEEN GENERATOR AND OTHER EQUIPMENT.

NOTES:

1. REFER TO EAST PGH. DRAWING 620J157 FOR METHOD OF MOUNTING GENERATOR ON FOUNDATION.
2. SEE METHOD OF SHIPPING & HANDLING FOR SHIPPING WEIGHTS, SHIPPING CLEARANCES, AND METHOD OF HANDLING.
3. REFER TO EAST PGH. DRAWING 1250J44 FIGURE I FOR JOINT BETWEEN GENERATOR AND TURBINE LAGGING.
4. REFER TO EAST PGH. DRAWING 215C591 FOR TRANSVERSE ANCHOR DETAIL.
5. REFER TO EAST PGH. DRAWING 1655F11 FOR BEARING BRACKET OUTLINE.
6. FOR LOCATION OF LEVELING PLATES FOR JACKING BOLTS, REFER TO EAST PGH. SEATING PLATE DRAWINGS FOR GENERATOR AND EXCITER HOUSING.
7. APPROVAL:

   SOUTH PHILADELPHIA TO APPROVE OUTLINE AND ASSEMBLIES

   ASSEMBLY "B" – GENERATOR WITH VIBROMETER PICKUP – DWG. 4732C24 FIG. 1

   ASSEMBLY "C" – COUPLING DWG. 4748C90

   ASSEMBLY "D" – EXCITER WITH VIBROMETER PICKUP – MODEL 1249J52 GR 4

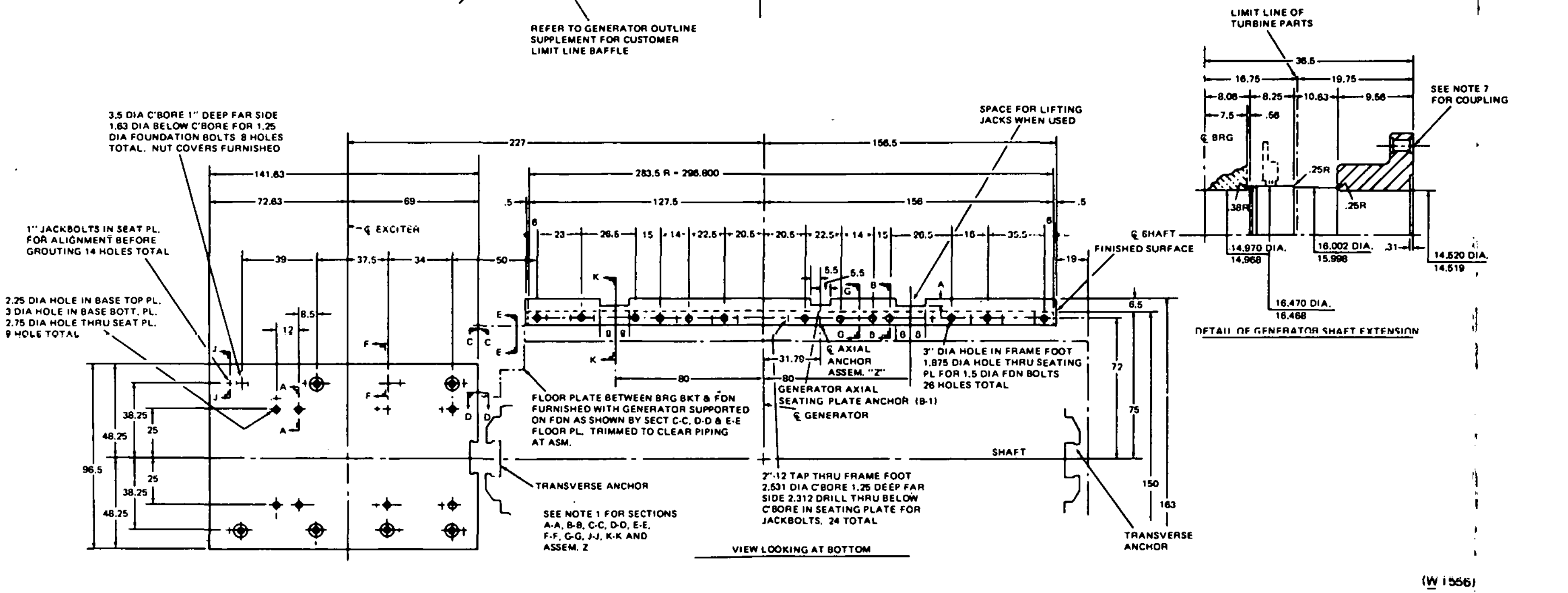

Figure 8-5. Generator and Exciter Footprint
600 PSI and 1000 PSI CAES Plants

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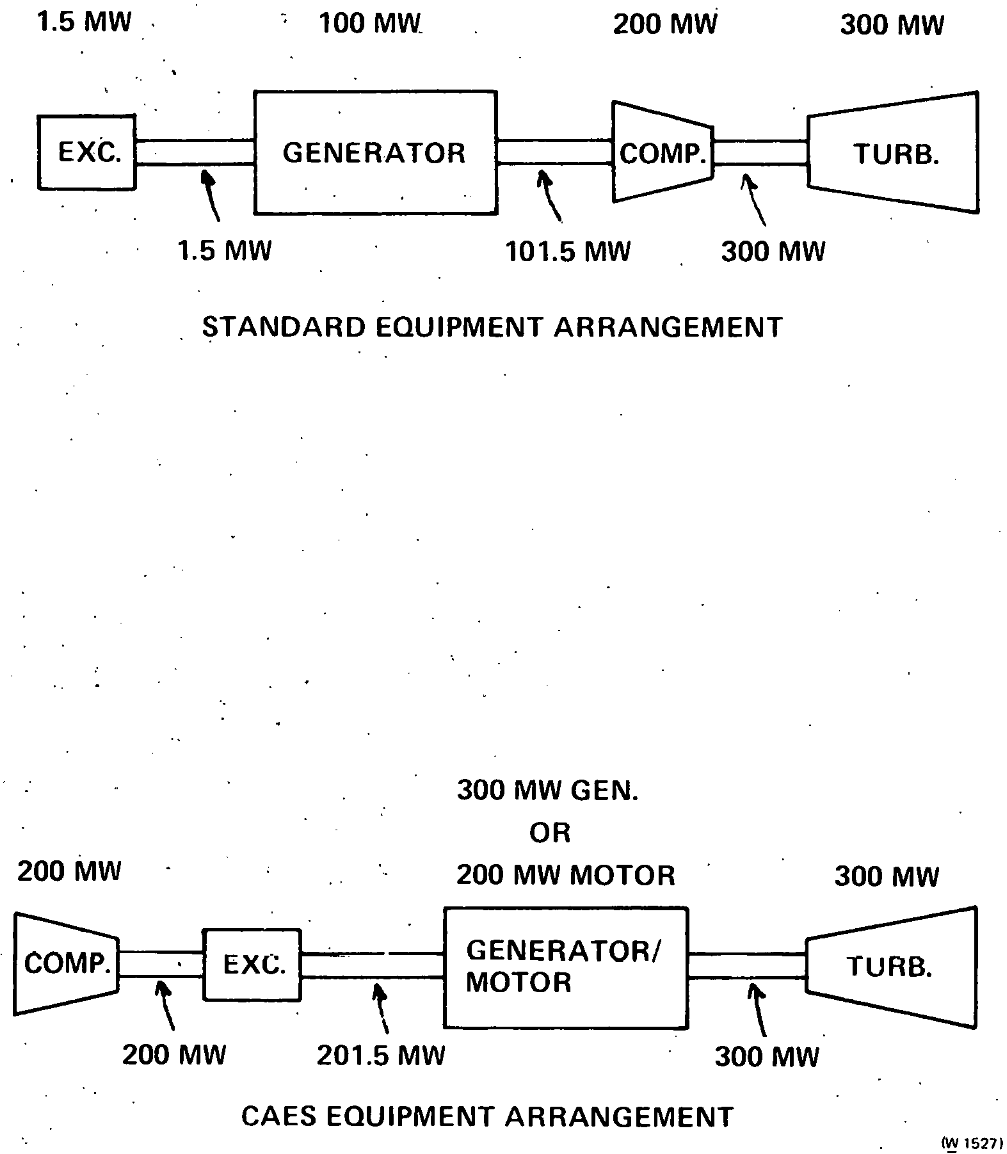


Figure 8-6. Mechanical Design of CAES Exciter

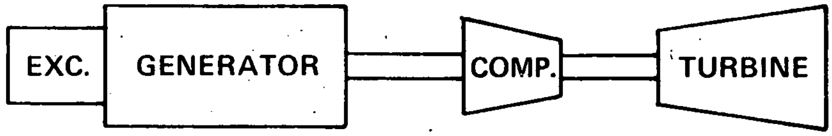
GENERATOR THRUST BEARINGS

The traditional gas turbine generator power plant incorporates a single thrust bearing in the turbine to counteract the axial forces placed on the rotating machinery by the generator and turbine. Use of a single thrust bearing in the CAES plant will not be sufficient since in the motor mode the turbine will be decoupled from the generator. A thrust bearing will be incorporated in the compressor, but axial play in both the special coupling between the generator and compressor, and in the clutch between the generator and turbine will require that a thrust bearing be installed in the generator. While this is not generally done in a generator, it is felt that solution of this problem is well within the realm of present technology.

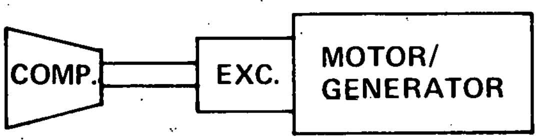
TORSIONAL ANALYSIS

When designing large complex systems of rotating machinery, care is taken to ensure that no problems are encountered with torsional resonance of either the individual components or the systems as a whole. Individual components (such as the generator, turbine or couplings) are designed to tune the system to natural frequencies which are far removed from natural driving frequencies. This ensures that the system will be free of oscillations and large mechanical stresses.

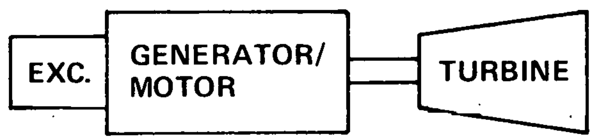
The design of the CAES plant will require a careful torsional analysis because of the many different configurations in which the plant will be operated. The unique combination of components which form the CAES plant will be described using the 300 MW, 1000 PSI system as an example. In a normal gas turbine plant, 300 MW gas turbine would be used to drive an integral 200 MW compressor. The net power (100 MW) would be used to drive a 100 MW generator. In the CAES system, the basic turbine and compressor are used with a 300 MW generator which was originally designed to operate with a steam turbine. In addition, the compressor or turbine will be decoupled from the generator/motor depending on the desired mode of plant operation. It may also be desirable to operate the plant independent of the underground storage reservoir. In this mode, the turbine drives both the compressor and generator and the net electrical output of the plant is 100 MW. The rotating equipment of the CAES plant will be operated in three different modes as shown on Figure 8-7. The equipment must be carefully designed to ensure that satisfactory torsional performance is achieved in each mode. Critical frequencies for each mode will be considered in developing the startup procedures for each mode.



CONVENTIONAL COMBUSTION TURBINE PLANT – ONE OPERATIONAL MODE



CHARGING MODE



GENERATING MODE

COMBINED MODE

CAES PLANT – THREE OPERATIONAL MODES

(W 1528)

Figure 8-7. Torsional Analysis of CAES Plant

CYCLIC DUTY

It is presently an assumption of this study that the CAES plant will operate on a daily cycle consisting of 10 hours charging (maximum charging) 2 hours off, 10 hours generating (full load), and 2 hours off. Generators with ratings of 300 MW are not typically subjected to such frequent no load to full load cycles. The high cyclic duty proposed for the generator/motor in the CAES plant will require careful mechanical design to ensure an adequate unit lifetime (2 cycles/day - 6 days/week - 30 years = 18,000 cycles).

Major factors to be considered are:

(1) Mechanical Cycling

(a) Rotating Members

Mechanical Fracture Analysis

(b) Material Selection for Rotating Parts

(2) Thermal Cycling

(a) Rotor Coil Slip Layers

(b) Matching Radial and Axial Expansion of Stator Winding and Bracing Components.

Westinghouse has designed synchronous generators with ratings of 300 to 400 MW for peaking duty. Development of generators for the CAES plant is well within the capability of existing technology and experience and should present no serious problems.

GENERATOR STABILITY

When operated in the generating mode, the CAES gas turbine will drive the generator and the compressors will be decoupled from the system. The inertia of the compressors as well as their normal load will thus be absent from the mechanical system. This will result in a lower than normal system inertia constant (H). This constant is influential in determining the electrical stability of the generator during a system disturbance. During an electrical system fault, circuit breakers disconnect the generator from the transmission system. During this time the turbine continues to produce rated mechanical power while the generator load is reduced to zero. The turbine power, therefore, is consumed accelerating the mass of the rotating equipment. This acceleration must be kept to a minimum since the ability of the generator to remain stably connected to the system when the circuit breakers reclose depends on the generator remaining close to synchronous speed. The more inertia there is in the system, the less acceleration will occur during the fault and the higher are the chances of the generator returning to the system and maintaining its electrical stability. The removal of the compressor load and inertia from the normal plant configuration will reduce the electrical stability of the CAES plant. The exact extent of this reduction will be studied in detail in Task 5 of the CAES program. Determination of the CAES stability limits will be made when the plant site is identified and characteristics of the transmission system to which it will be connected are known. At that time the critical fault clearing time for the CAES plant can be determined and the need for special provisions required to produce acceptable stability performance can be determined.

SYNCHRONIZING

It is projected that the CAES plant will operate 10 hours per day generating and 10 hours per day compressing. To accomplish this changeover, the plant will be disconnected from the power system, reconfigured, and resynchronized with the power system. Two generator/motor synchronizations per day will occur and must be cautiously done since synchronizations can result in electromechanical shocks to the generator and unit step up transformer. In order to minimize long term damage to the rotating machinery and other electrical equipment, the synchronization must be done as precisely as possible. It is recommended that automatic synchronizers which are readily available be used for CAES plant. Plant operators generally prefer to synchronize manually; however, this may lead to a large number of severe shocks to the system over the life of the plant. It is suggested that the use of automatic synchronizers be considered for this new type of plant.

UNIT TRANSFORMER DESIGN

It is anticipated the CAES generator/motor will be connected to the power system through a conventional unit step up transformer. The biggest concern in the application of this transformer is the electromechanical shocks it will be exposed to through improper synchronizations. Use of the automatic synchronizer will minimize this problem. Special design methods and materials have been developed by transformer manufacturers to deal with application problems of this matter. Transformers have been designed with extra bracing and provisions for periodic retightening of the support structures.

The use of automatic synchronizers and special transformer designs should result in the successful operation of the unit transformer in the CAES plant.

SUMMARY OF ELECTRICAL MACHINERY APPLICATION

Preliminary results of Task 1 concluded that the single shaft plant configuration was the most desirable. Three plant designs were developed using different storage pressures which covered the practical range of pressures likely to be used. This section of the report has identified specific synchronous generator/motor designs for each of the three plant designs. The designs are largely "off the shelf" since the electrical machinery requirements of the unique CAES plant are relatively conventional.

Some application requirements imposed on the electric machinery are, however, worthy of special attention. These areas have been identified and discussed. It is felt that all areas are solvable using existing technology and relatively minor modifications to the existing equipment and systems. Detailed development of these areas will be undertaken in Tasks 3 and 5, when more detailed information is available concerning the plant design, plant site location, and system interaction constraints.

ELECTRIC MACHINERY CONTROL/INSTRUMENTATION

The selection of the single shaft CAES plant configuration has greatly simplified the control and instrumentation requirements for the electric machinery. Several plant configurations originally proposed utilized multiple generators and motors. Control of these multiple units would have required a complex system to control the startup and shutdown sequences, and to coordinate the operation of the multiple units in a way which ensured operation consistent with the thermodynamic and storage requirements of the plant.

The selection of a single shaft system (one synchronous generator/motor) and the decision to start this system using the gas turbine and aquifer air has minimized the problem of designing the control and instrumentation system. The electric machinery control system will now be very similar to that in a conventional gas turbine plant.

Although the control system for the rotating electric machinery will be controlled largely as a conventional power plant, changes previously noted in this report may be desirable. Small changes to the voltage regulator which will provide several plant operating features such as power factor regulation in the motor mode will be considered. Automatic synchronizers are recommended as standard equipment. A standard package of Westinghouse protective relays is recommended in Table 8-5.

In summary, the control and instrumentation for the single shaft CAES plant will closely resemble the procedures and equipment used in a conventional gas turbine power plant.

Table 8-5

RECOMMENDED PROTECTIVE RELAYING FOR CAES TURBINE GENERATOR

Function	Westinghouse Relay Type
Generator Differential	SA-1
Step-up Transformer Differential	HU or HU-1
Unit Auxiliary Transformer Differential	HU or HU-1
Unit Overall Differential	HU-1
Negative Sequence	COG (Future - SOQ)
Generator Stator Ground	CV-8
Generator Field Ground - Nonbrushless System	DGF
Generator Loss-of-Field	KLF or KLF-1
Excessive Volts/Hertz	SV plus Timer
Loss of Synchronism	SDBU-1 or SDBU-2
Underfrequency (if used)	KF or SDF-1
Unit Back-up	KD-11 or COV

Protective Features in Excitation Switchgear

1. Minimum Excitation Limiter - MEL
2. Maximum Excitation Limiter - MXL
3. Overexcitation Protection - OXP
4. Volts/Hertz Limiter - HXL
5. Generator Field Ground Detection on Brushless System - Sequencing Control and Indication

Other

1. Blown Diode Fuse Indicators
2. Exciter Field Ground Detection
3. Exciter and Generator Coolant Temperature Monitoring and Alarms
4. Bearing and Seal Oil Temperature, Flows, etc.
5. Hydrogen Pressure, Purity, etc.

(W 1596)

Section 9

CAES PART LOAD PERFORMANCE

COMPRESSOR TRAIN CHARACTERISTIC CURVES

Flow and discharge pressure characteristic curves for the nominal 200, 600 and 1000 PSI storage systems optimized compressor trains utilizing two intercoolers were prepared for use by Sargent & Lundy to determine the number of wells required for each candidate aquifer site. The characteristic curves were based upon dry air at the conditions shown in Figure 9-1. It was determined by the Westinghouse, CTSD turbomachinery design group, that the part load performance of the standardized compressor trains would be essentially the same as the optimized trains.

The design point air flow for all compressor trains was 770 lb. of dry air per second. The characteristic curves are shown on Figures 9-2, 9-3 and 9-4.

CAES TURBINE TRAIN CHARACTERISTIC CURVES

Off-design flow and pressure characteristic curves were prepared for the 200, 600 and 1000 PSI nominal storage pressure optimized turbine trains. These curves are shown on Figures 9-5- 9-6 and 9-7. The design point regenerator mass flow rates are equal to the aftercooler compressed air discharge design point mass flow rates less the turbine cooling air flows. The regenerator inlet pressure is based upon

DESIGN POINT PRESSURE RATIOS				
P_S PSI	P_{LPC}	P_{IPC}	P_{HPC}	P_{2AC} PSIA
200	3.38	2.40	2.40	250
600	4.77	3.39	3.39	700
1000	5.47	4.05	4.05	1150

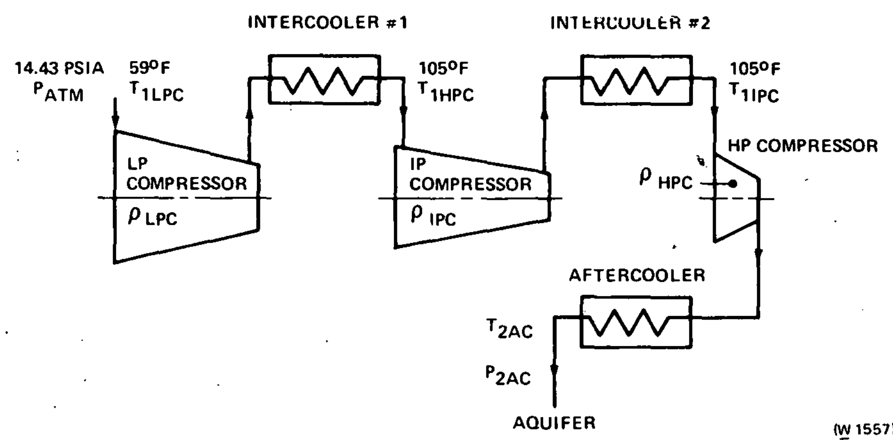


Figure 9-1. CAES Optimized Compressor Trains

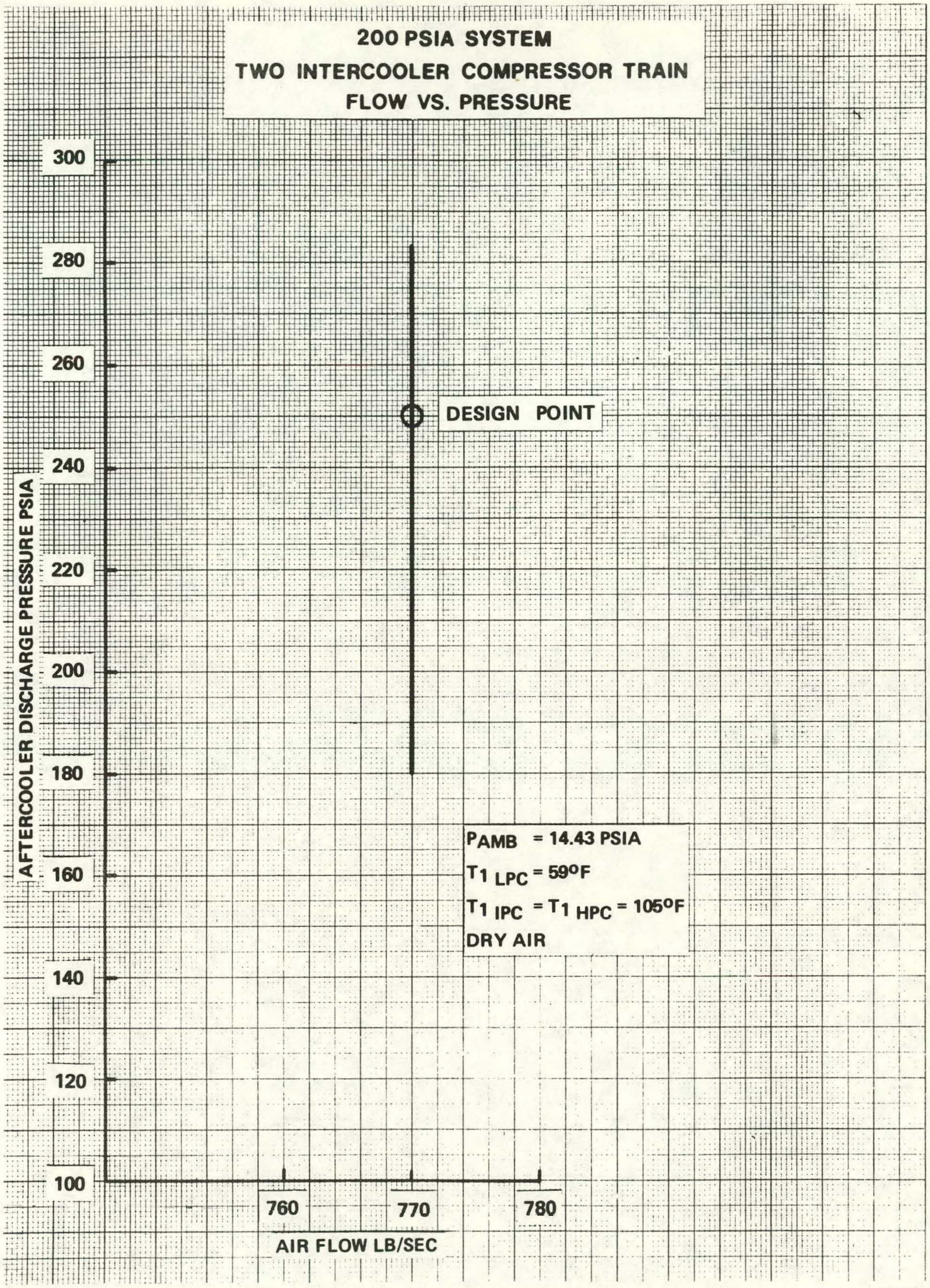


Figure 9-2. Optimized Compressor Train - 200 PSIA System

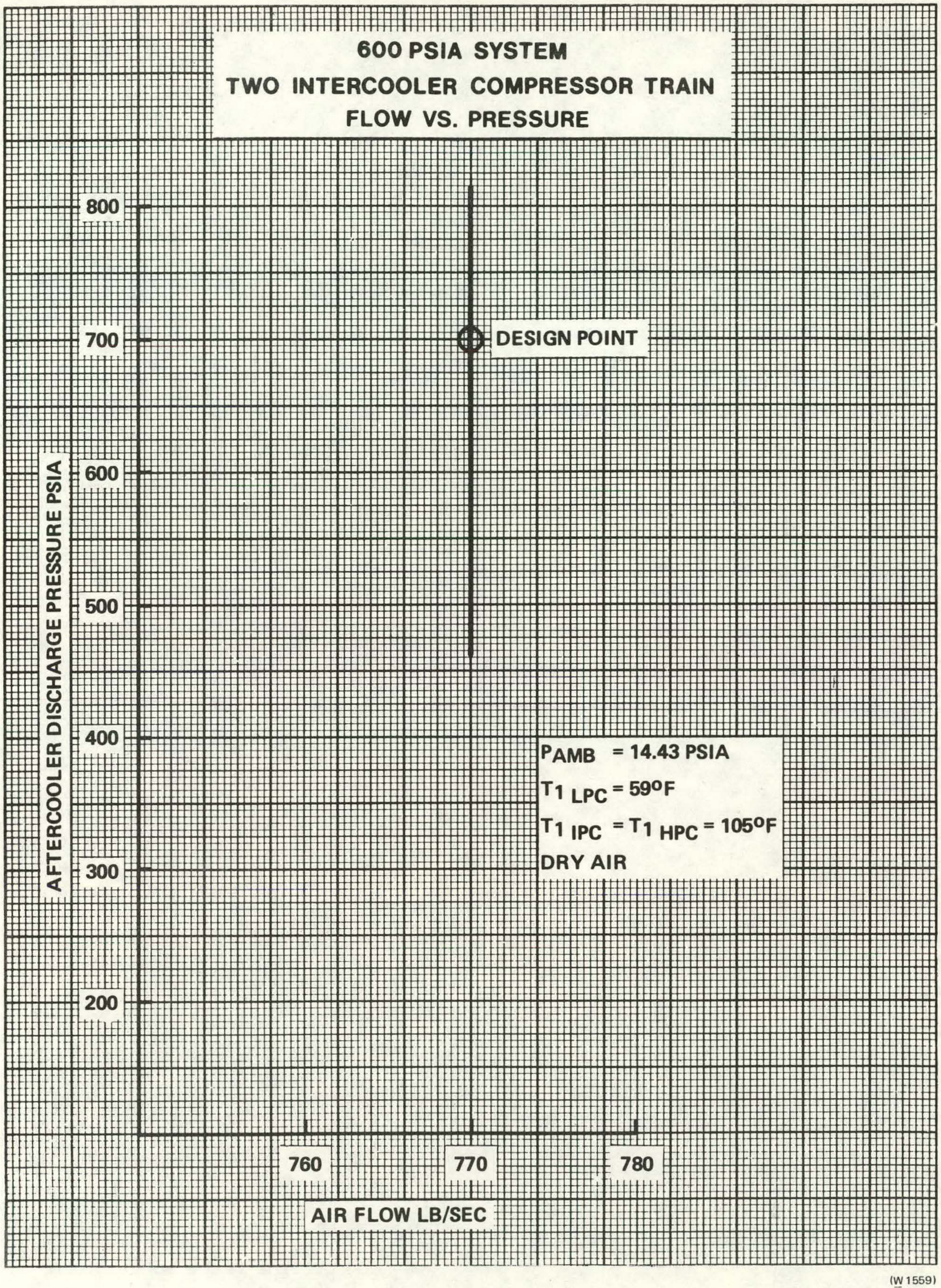


Figure 9-3. Optimized Compressor Train - 600 PSIA System

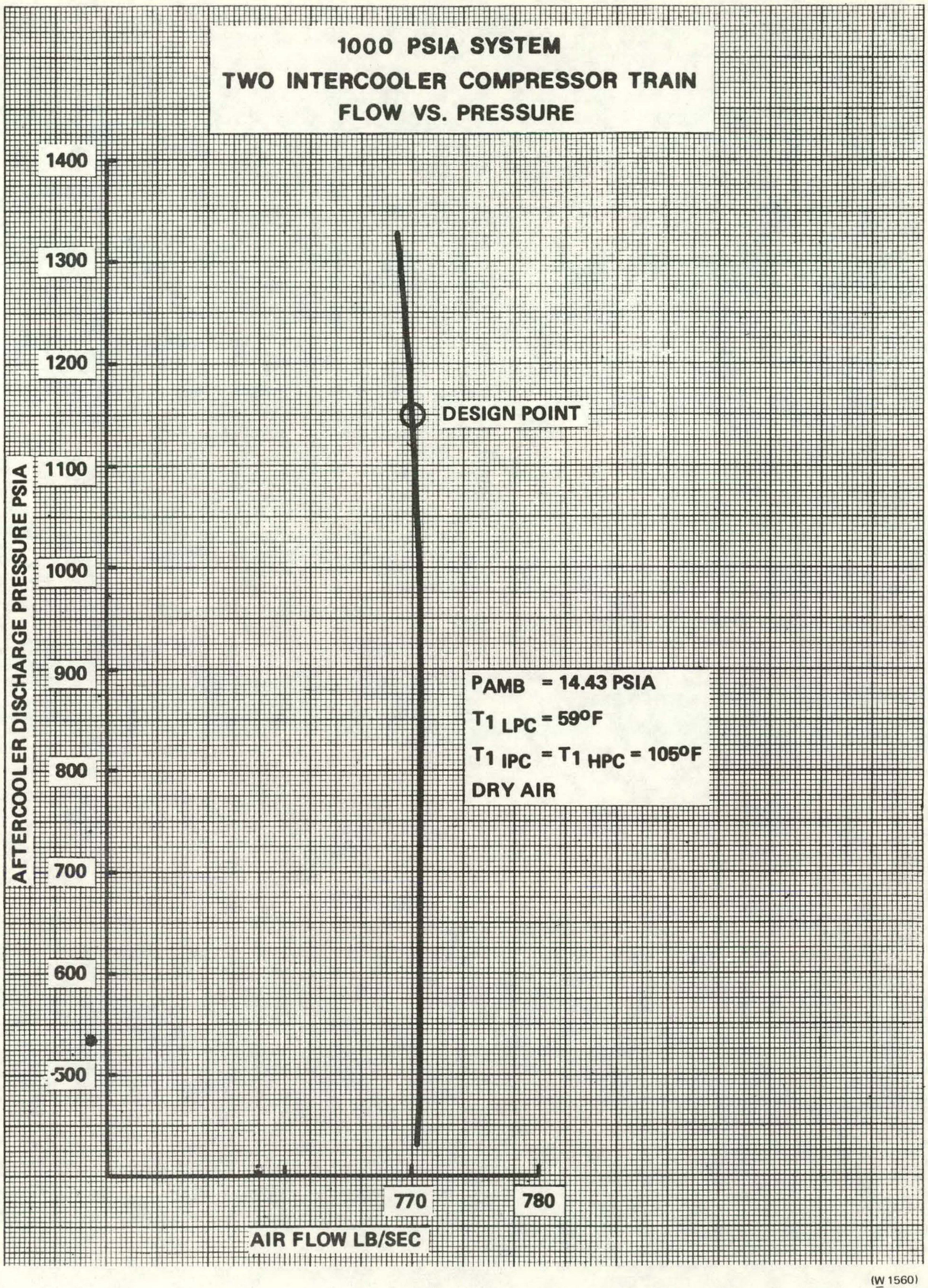


Figure 9-4. Optimized Compressor Train - 1000 PSIA System

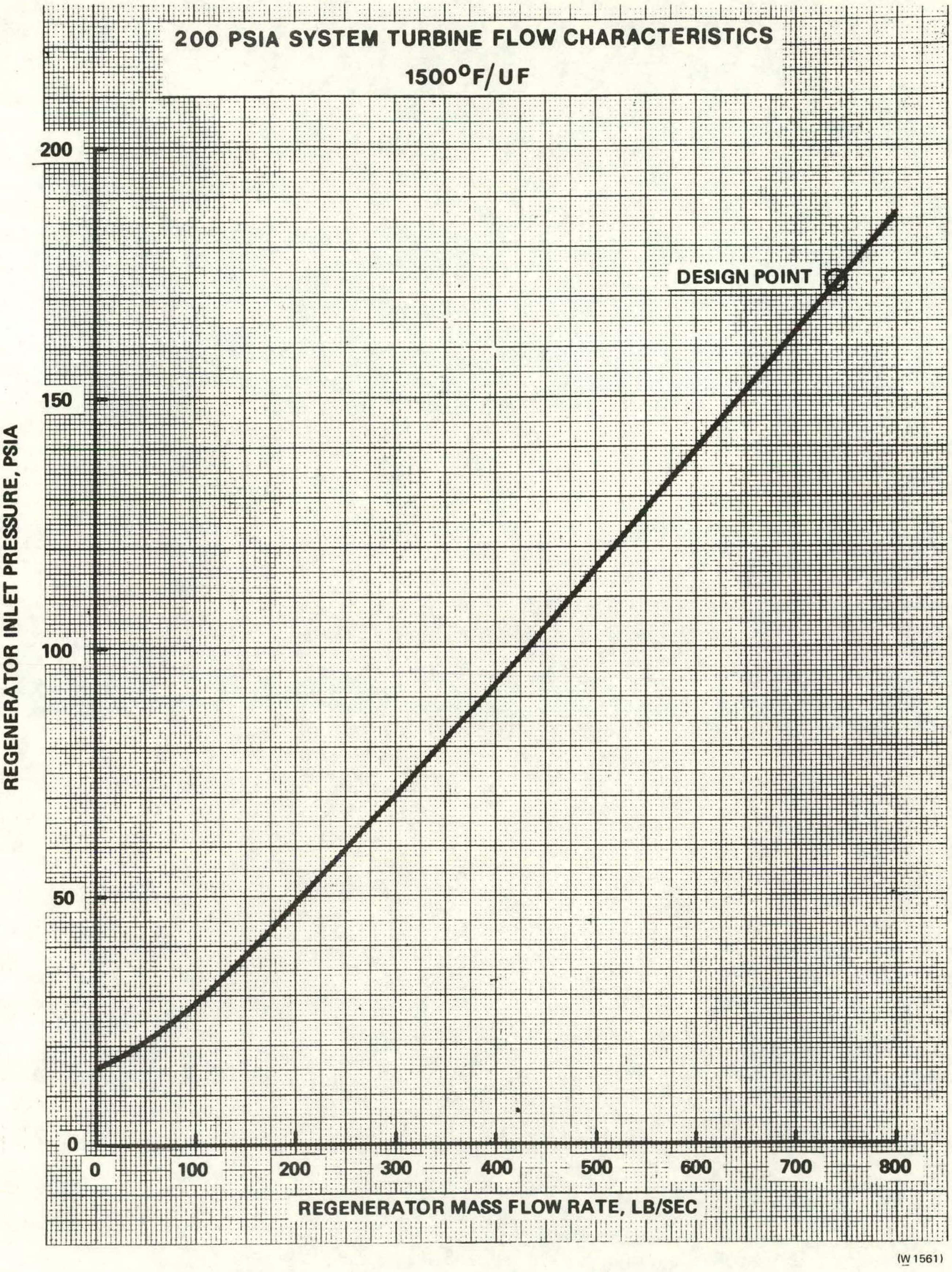


Figure 9-5. 200 PSIA System Off-Design Turbine Flow

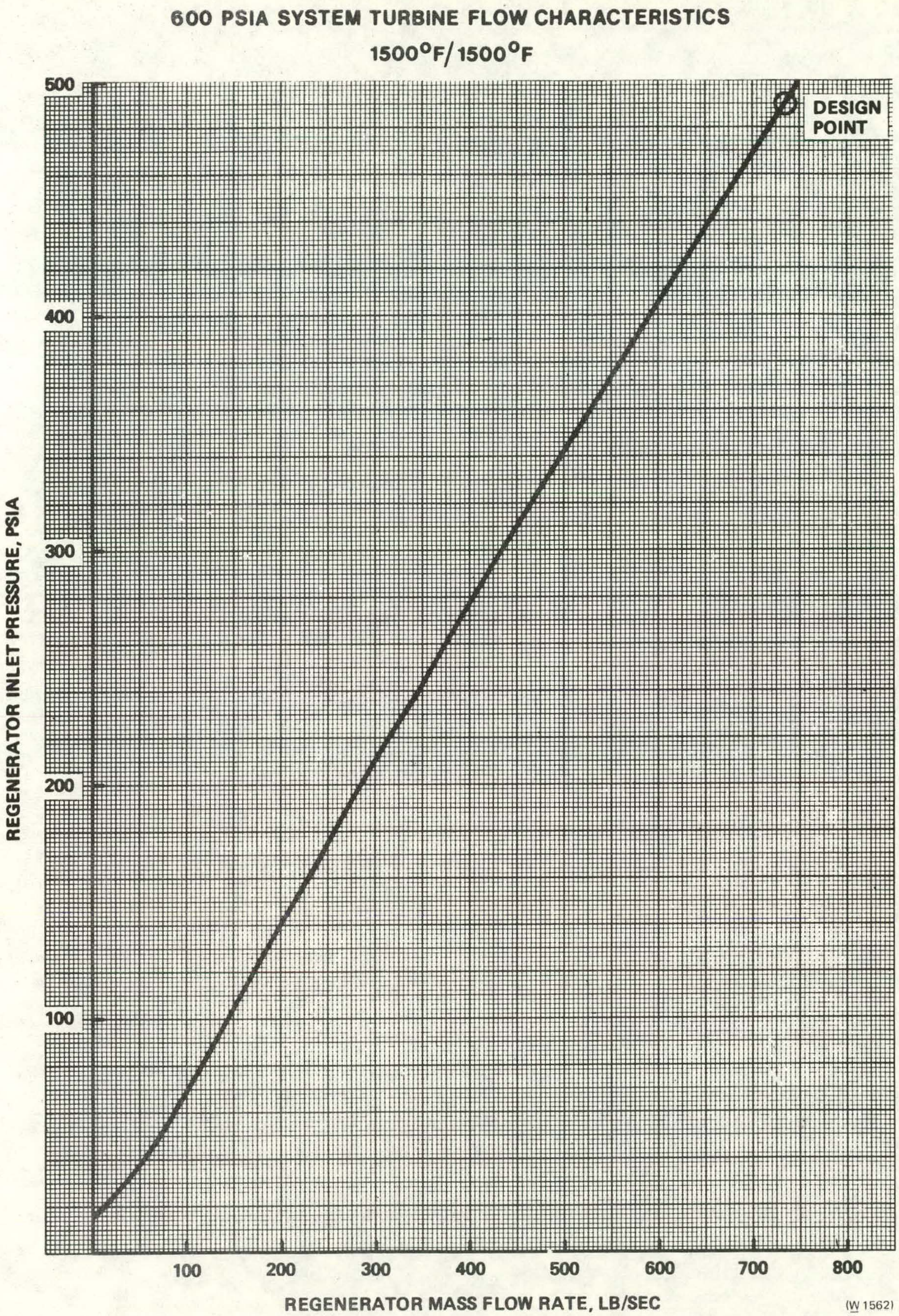


Figure 9-6. 600 PSIA System Off-Design Turbine Flow

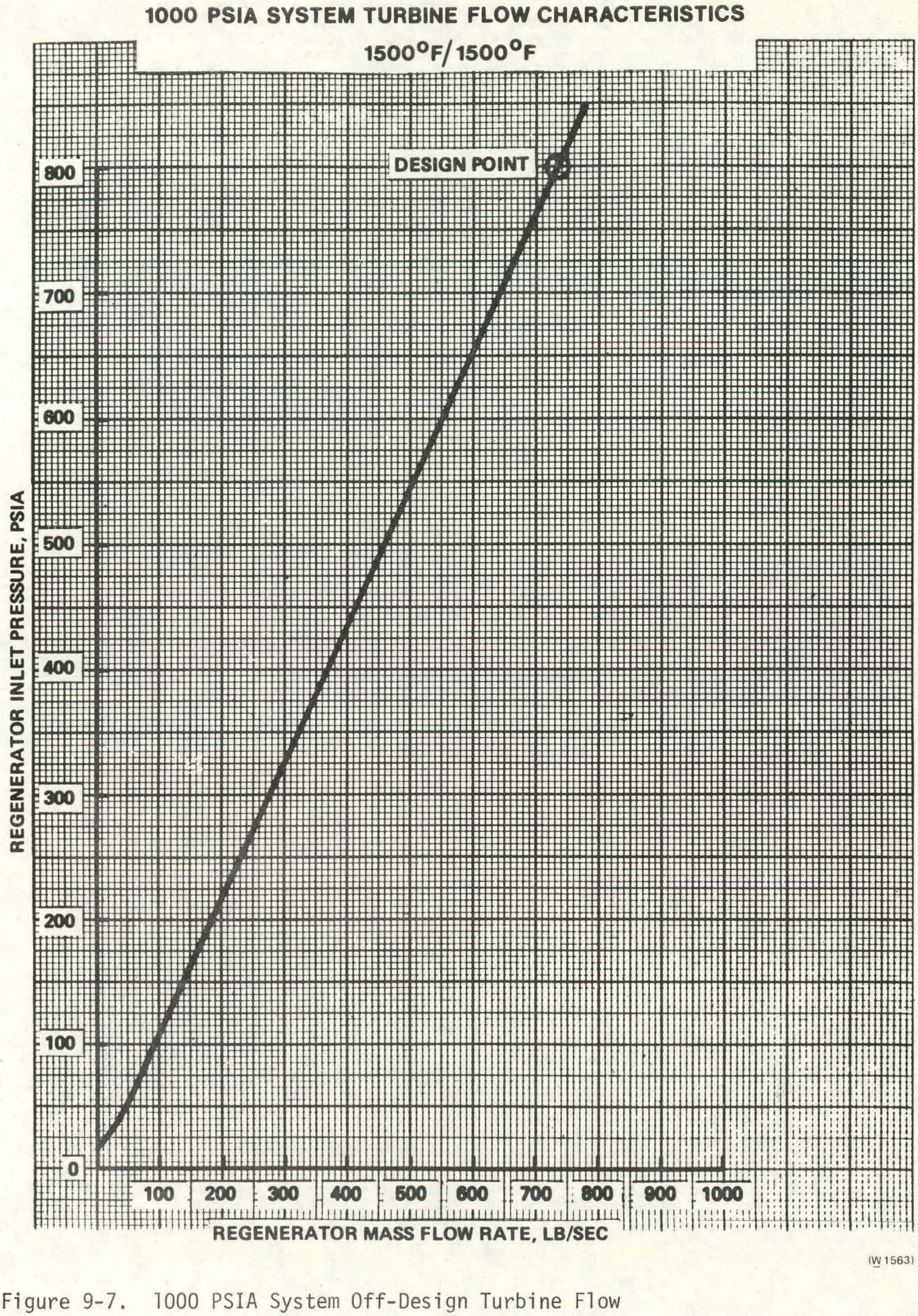


Figure 9-7. 1000 PSIA System Off-Design Turbine Flow

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CAES IN AQUIFER
DOE NO. ET-78-C-01-2159

an external plant air pressure loss ratio of 1.4 from the aftercooler discharge to the regenerator inlet. Since the refined standardized turbine trains are almost identical to the optimized turbine trains, the above curves are appropriate for either.

CAES TURBINE TRAIN PART LOAD PERFORMANCE

The part load performance of the 200, 600 and 1000 PSI nominal storage pressure CAES turbine trains is illustrated on Figures 9-8, 9-9 and 9-10. Net turbine outputs, net LHV heat rates and regenerator inlet pressures are plotted as a function of the turbine train regenerator mass flow rate. Selected off-design data points are presented in Table 9-1.

The CAES turbine train off-design performance was calculated in the following manner. The turbine inlet temperatures were maintained and turbine element efficiencies were initially held constant. The HP turbine inlet pressure was decreased to a preselected value while the HP and LP turbine pressure ratios and discharge mass flow rates were varied until the off-design Stodola Flow Coefficient matched the design point Stodola number. The turbine element efficiencies were then adjusted to account for the off-design conditions. The above matching procedure was then repeated with the adjusted turbine efficiencies.

As can be seen from the data presented above, there is a small increase in the heat rate as the turbine output power is reduced below the 100% design point. This is the result of reducing the discharge mass flow rate instead of turbine inlet temperatures.

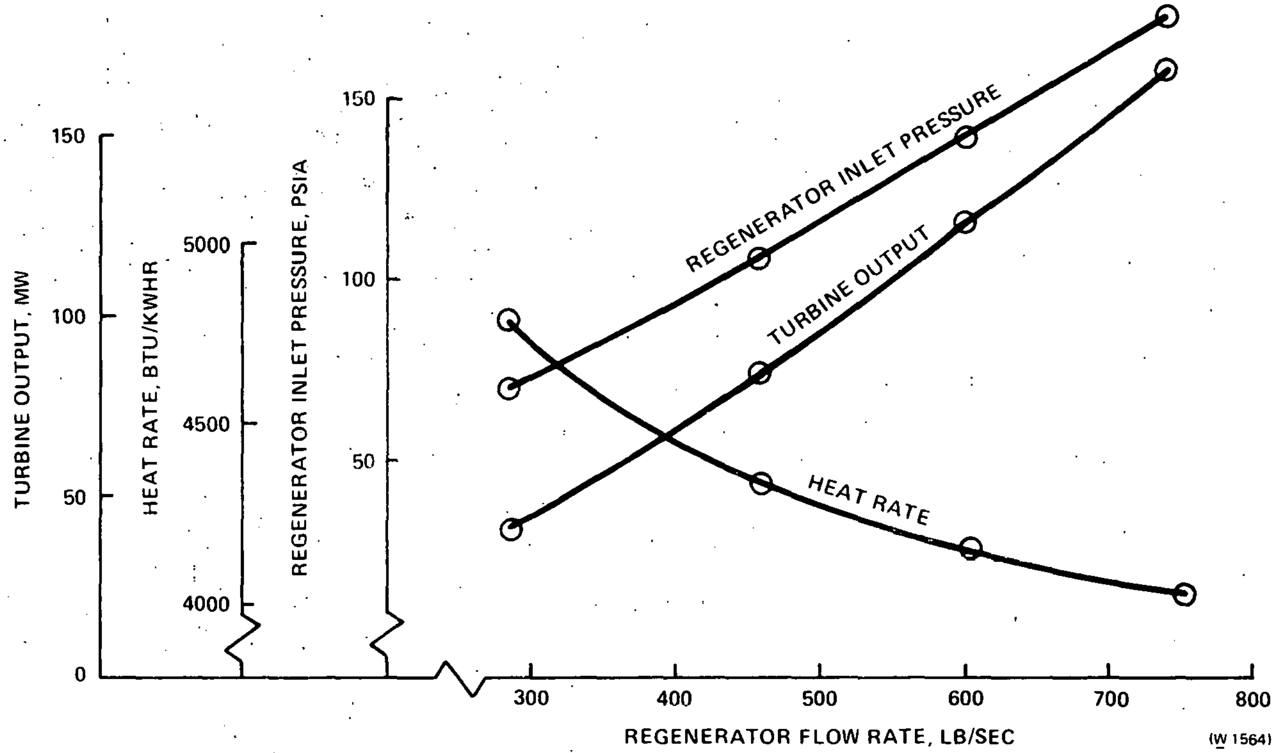


Figure 9-8. 200 PSIA System Part Load Performance

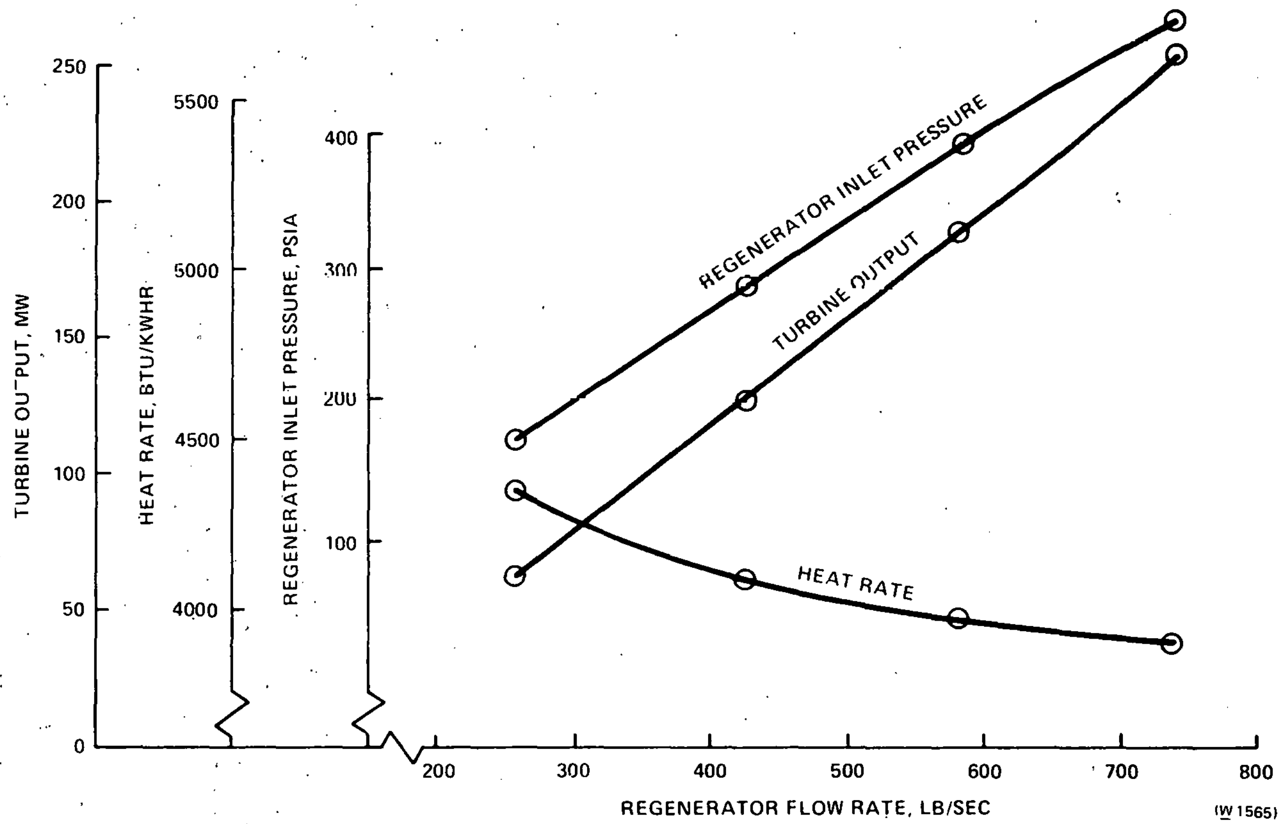


Figure 9-9. 600 PSIA System Part Load Performance
The above performance values are as calculated and do not include margins.

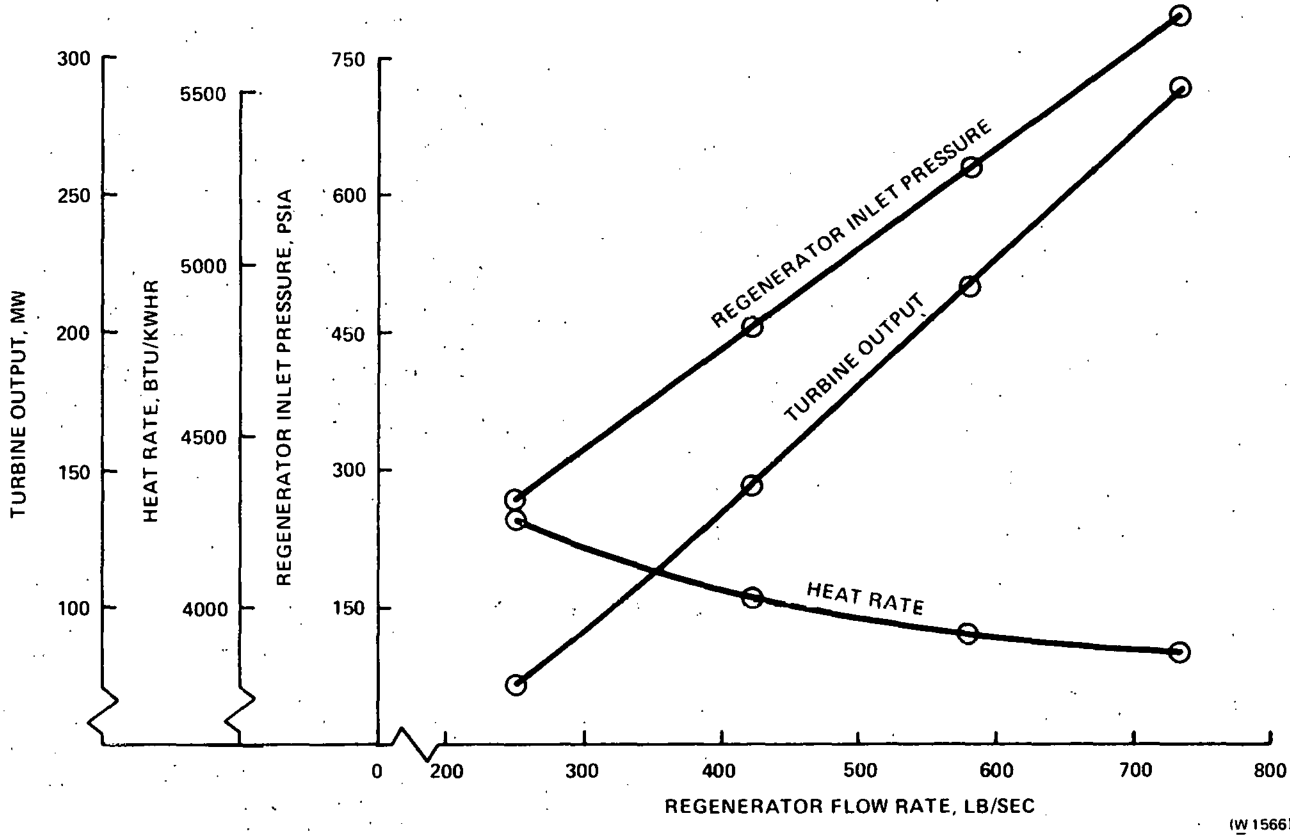


Figure 9-10. 1000 PSIA System Part Load Performance

Table 9-1

PART LOAD PERFORMANCE OF THE OPTIMIZED TURBINE TRAINS

	% Design Load	Regenerator Inlet Pressure, PSIA	Heat Rate, BTU/KWHR	Net Power Output for Machinery Set, KW
200 PSI System	100	*173.2	*4075	*168,220
	75	140	4160	126,460
	50	106	4320	84,310
	25	70	4780	42,155
600 PSI System	100	*487.6	*3915	*254,930
	75	395	3980	191,115
	50	290	4100	127,410
	25	175	4350	63,705
1000 PSI System	100	*796.1	*3880	*289,250
	75	630	3920	216,938
	50	460	4020	144,625
	25	265	4260	72,315

*Indicates turbine train design point.
The above performance values are as calculated and do not include margins.

Section 10

CAES AQUIFER INDEPENDENT GENERATING OPERATION

The CAES systems were analyzed as aquifer independent generators (see Figure 10-1) in the following manner. Since the turbine inlet temperatures will be maintained at their 1500°F design point conditions and the axial compressor trains maintain constant mass flow rate as the discharge pressure is varied, the turbine inlet pressure will, due to Stodola law, remain at the design point conditions. However, since the system storage pressure loss ratio will be reduced from 44% to 5% and the 2% pressure loss associated with the by-passed aftercooler will be eliminated, the required total compressor train pressure ratio will be substantially lower than the design point pressure ratio unless a throttle valve is employed. It was determined that the use of a throttle to maintain the design point compressor discharge pressure would be very inefficient. During the compressor train part load performance noted above, only the HP compressor would be affected. It was determined that the HP compressors would choke at approximately 80% of their design point discharge pressure. As a result, some throttling was assumed to be performed in the tie over pipe to maintain 80% of the design point discharge pressure and to aid in controlling the unit during aquifer independent generators operation. For reasons stated above, the LP and IP compressors efficiencies remain at their design points. The following HP compressors 80% part load efficiencies were determined.

Nominal Storage Pressure	Off Design Point Efficiency
200	$.950\ \eta_{DP} = .950\ (.875) = 83.1\%$
600	$.925\ \eta_{DP} = .925\ (.870) = 80.5\%$
1000	$.940\ \eta_{DP} = .940\ (.880) = 82.7\%$

The heat and material balances for the 200, 600 and 1000 PSI refined standardized CAES systems operating as intercooled, regenerative, reheated aquifer independent generators are presented in Figures 10-2, 10-3 and 10-4. Thus, the CAES turbomachinery could be utilized as an efficient generating unit prior to the completion

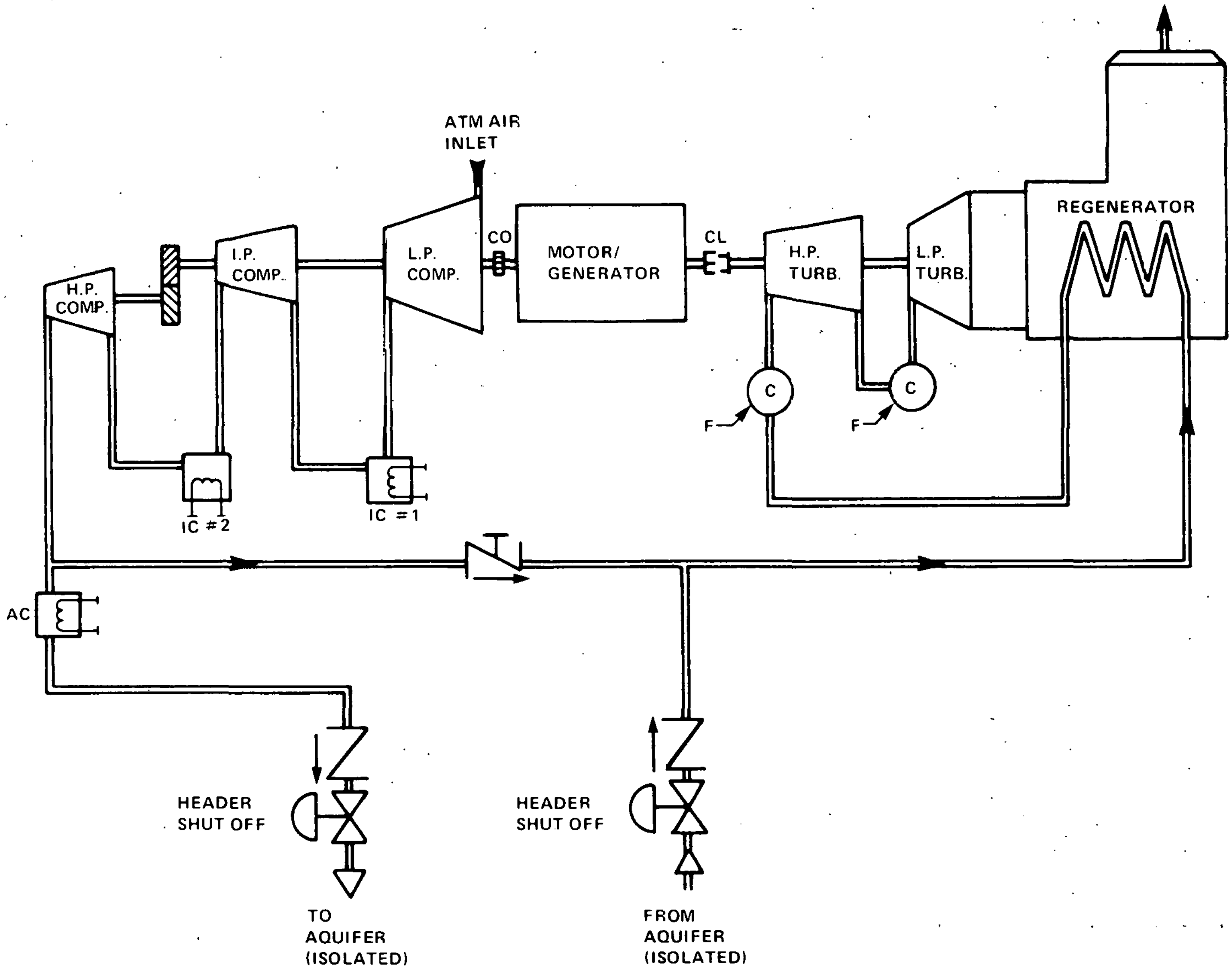
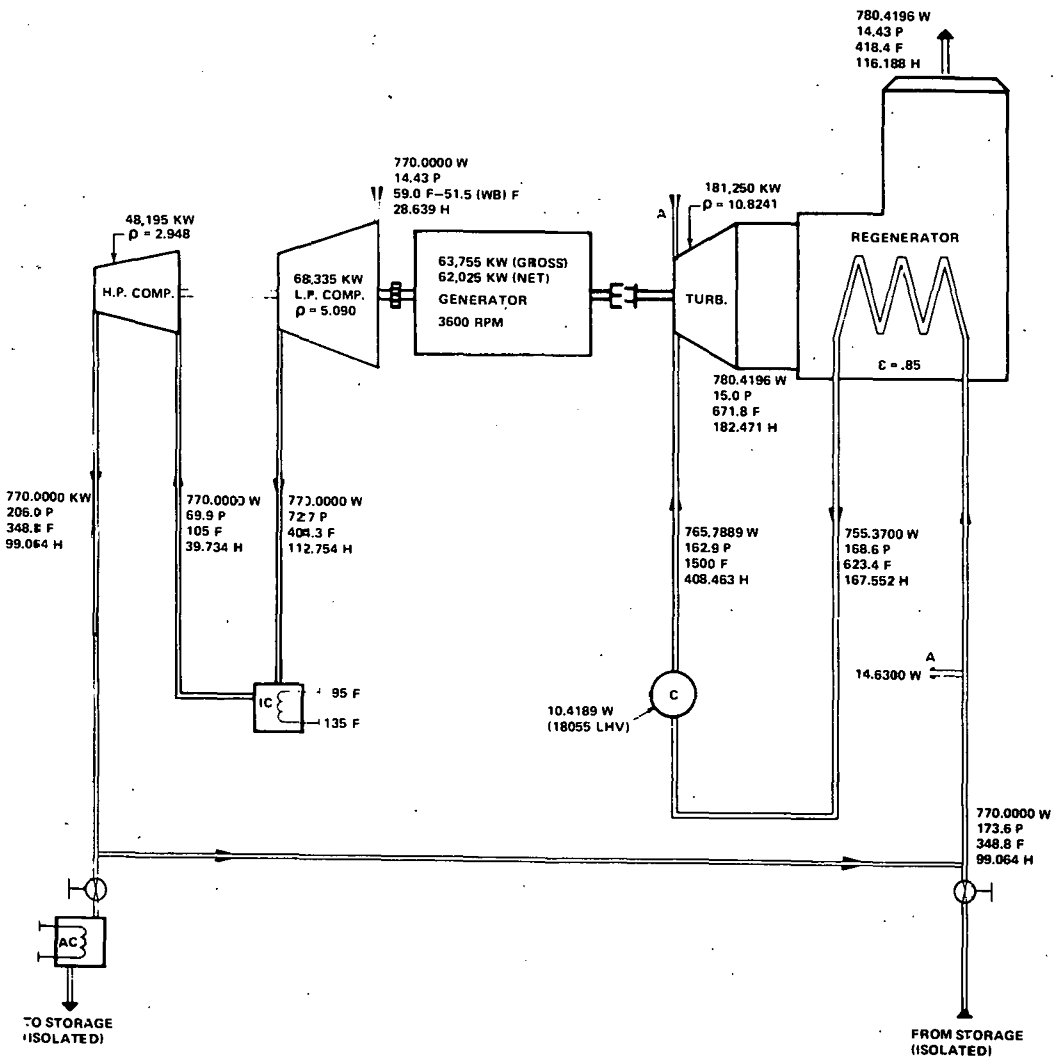


Figure 10-1. Schematic of a CAES System Operating as a Aquifer Independent Combustion Turbine Generator

250 PSIA CHARGING PRESSURE CAES UNIT AS AN AQUIFER INDEPENDENT GENERATOR USING OFF DESIGN HP COMPRESSOR. 1500°F TURBINE INLET TEMPERATURE UPDATED STANDARDIZED LP SET.



COMPRESSED AIR ENERGY STORAGE (CAES) POWER PLANT OPERATING AS AN AQUIFER INDEPENDENT GENERATOR ENERGY BALANCE

INPUT	BTU/HR
1. ATMOSPHERIC AIR:	79,387,310
2. HP COMBUSTOR FUEL (LHV = 18,055 BTU/LB):	677,207,660
3. LP COMBUSTOR FUEL (LHV = 18,055 BTU/LB):	0
TOTAL INPUT	7.566×10^8

OUTPUT	BTU/HR
1. HP COMPRESSOR GEAR LOSS:	0
2. INTERCOOLER #1 REJECTION HEAT LOSS:	202,411,440
3. INTERCOOLER #2 REJECTION HEAT LOSS:	0
4. AIR COOLER DRAIN LOSS:	0
5. HP COMBUSTOR LOSS:	6,772,075
6. LP COMBUSTOR LOSS:	0
7. TOTAL TURBINE POWER (@ GENERATOR TERMINALS):	217,541,165
8. TOTAL TURBINE ELECTRICAL LOSS:	3,312,810
9. EXHAUST GAS LOSS:	326,431,415
TOTAL OUTPUT	7.565×10^8

PLANT PERFORMANCE

TOTAL OUTPUT (@ GENERATOR TERMINALS):	63,755 KW
AUX. LOAD:	1,730 KW
NET TURBINE OUTPUT:	62,025 KW
NET LHV HEAT RATE:	10,920 BTU/KWHR

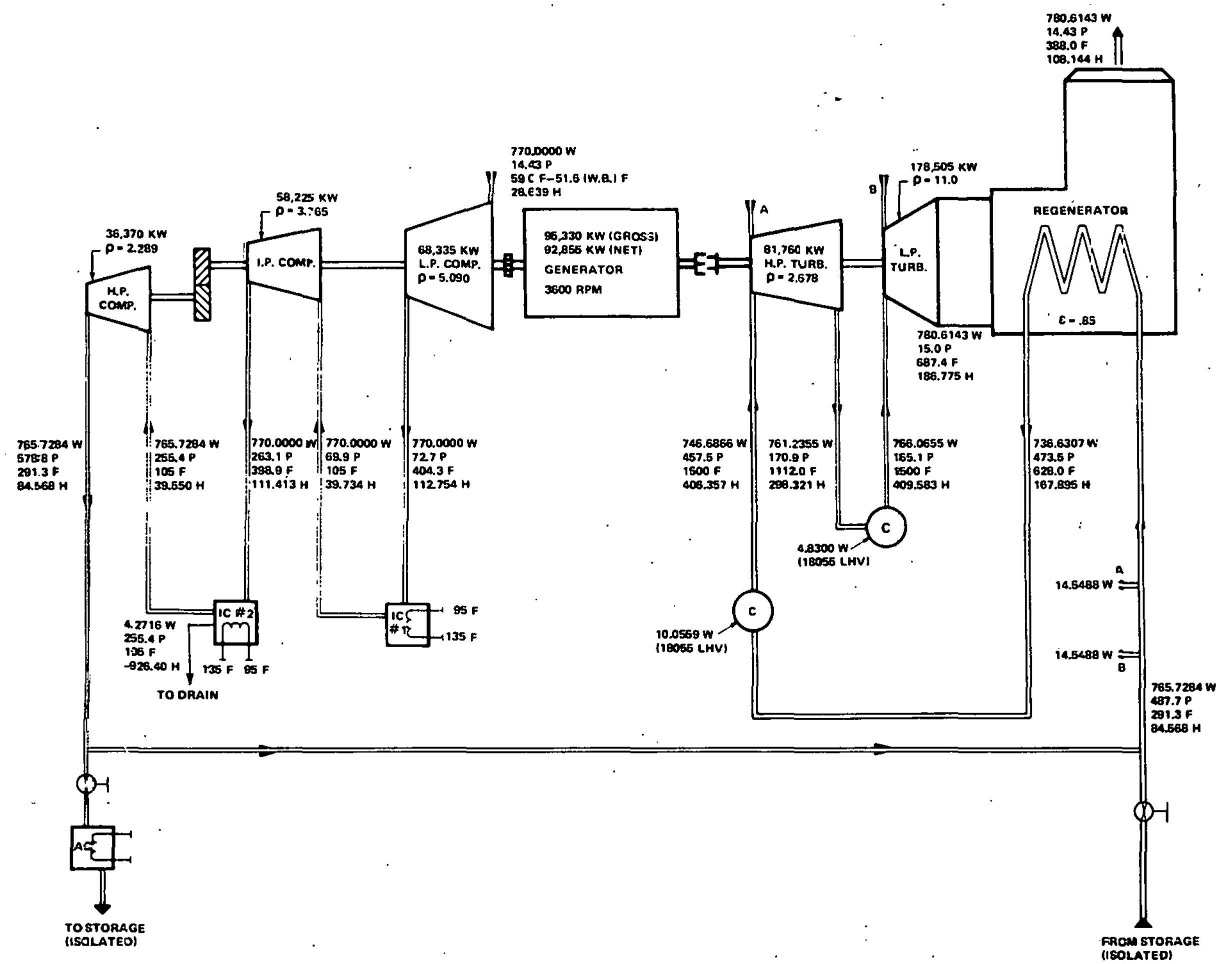
NOTES:

1. THE ABOVE DATA REFLECTS AS CALCULATED PERFORMANCE. GUARANTEED PERFORMANCE WOULD INCLUDE MARGINS (NOT NECESSARILY THE SAME) FOR THE POWER OUTPUT, AND HEAT RATE.
2. THE ABOVE REFLECTS AN ENTHALPY BASE WHERE THE ENTHALPY OF -60°F GAS EQUALS 0.0.

(W 1567)

Figure 10-2. 200 PSI Nominal Refined CAES System Utilized As An Aquifer Independent Generator Heat And Material Balances

700 PSIA CHARGING PRESSURE CAES UNIT AS AN
AQUIFER INDEPENDENT GENERATOR USING OFF DESIGN
HP COMPRESSOR. 1500-1500°F TURBINE INLET TEMPERATURE.
UPDATED STANDARDIZED LP SET.



COMPRESSED AIR ENERGY STORAGE (CAES) POWER PLANT OPERATING AS AN AQUIFER INDEPENDENT GENERATOR ENERGY BALANCE

INPUT	BTU/HR
1. ATMOSPHERIC AIR:	79,387,310
2. HP COMBUSTOR FUEL (LHV = 18,055 BTU/LB):	653,361,390
3. LP COMBUSTOR FUEL (LHV = 18,055 BTU/LB):	313,940,340
TOTAL INPUT	1.047×10^9

OUTPUT	BTU/HR
1. HP COMPRESSOR GEAR LOSS:	1,889,840
2. INTERCOOLER #1 REJECTION HEAT LOSS:	202,411,440
3. INTERCOOLER #2 REJECTION HEAT LOSS:	214,611,985
4. AIR COOLER DRAIN LOSS:	-14,245,955
5. HP COMBUSTOR LOSS:	6,533,615
6. LP COMBUSTOR LOSS:	3,139,405
7. TOTAL TURBINE POWER (@ GENERATOR TERMINALS):	325,279,575
8. TOTAL TURBINE ELECTRICAL LOSS:	4,953,495
9. EXHAUST GAS LOSS:	303,907,510
TOTAL OUTPUT	1.048×10^9

PLANT PERFORMANCE

TOTAL OUTPUT (@ GENERATOR TERMINALS):	95,330 KW
AUX. LOAD:	2,475 KW
NET TURBINE OUTPUT:	92,855 KW
NET LHV HEAT RATE:	10,420 BTU/KWHR

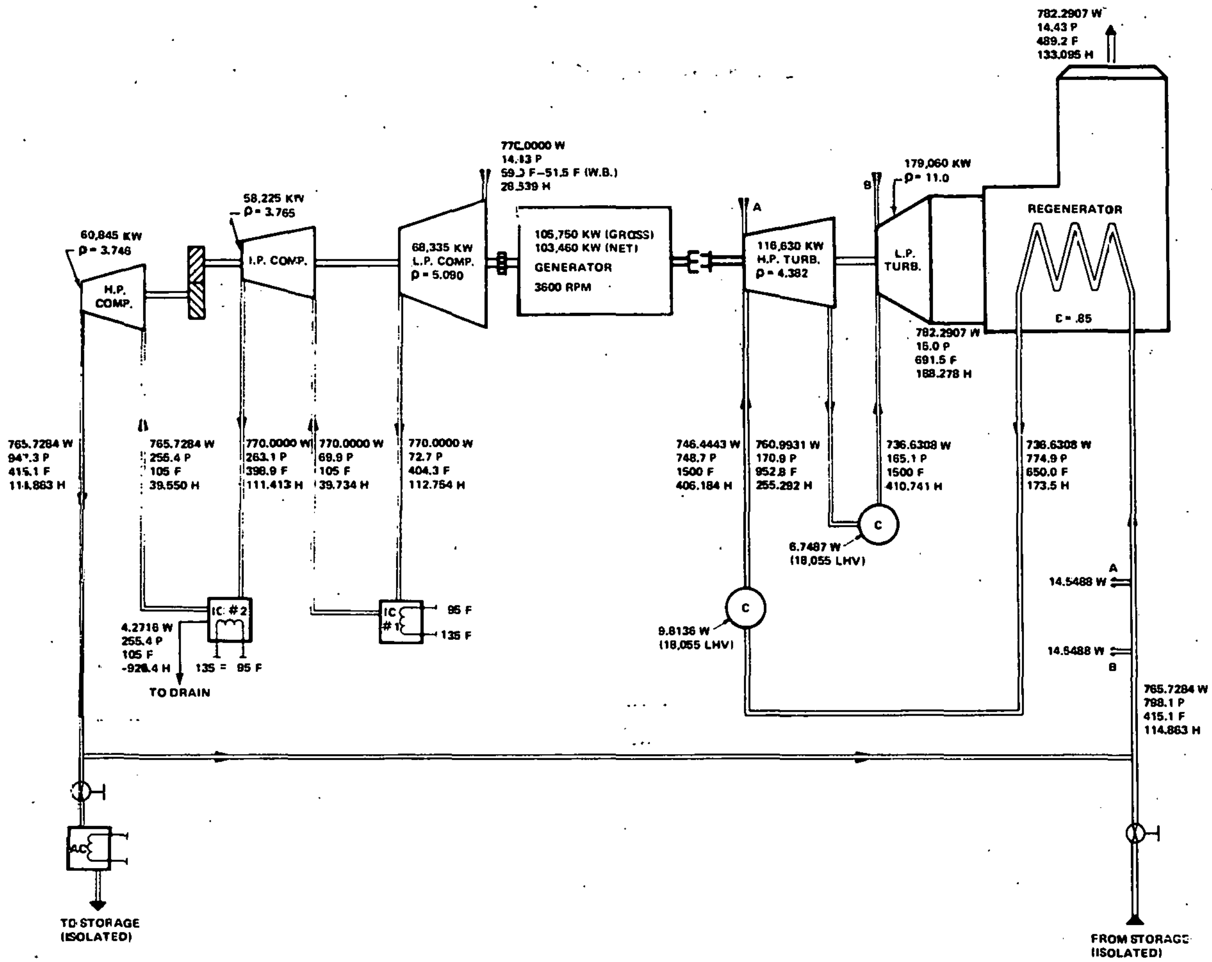
NOTES:

1. THE ABOVE DATA REFLECTS AS CALCULATED PERFORMANCE. GUARANTEED PERFORMANCE WOULD INCLUDE MARGINS (NOT NECESSARILY THE SAME) FOR THE POWER OUTPUT, AND HEAT RATE.
2. THE ABOVE REFLECTS AN ENTHALPY BASE WHERE THE ENTHALPY OF -60°F GAS EQUALS 0.0.

(W 1568)

Figure 10-3. 600 PSI Nominal Refined CAES System Utilized As An Aquifer Independent Generator Heat And Material Balances

1150 PSIA CHARGING PRESSURE CAES UNIT AS AN
AQUIFER INDEPENDENT GENERATOR USING OFF DESIGN
HP COMPRESSOR. 1500-1500°F TURBINE INLET TEMPERATURE.
UPDATED STANDARDIZED LP SET.



COMPRESSED AIR ENERGY STORAGE (CAES) POWER PLANT OPERATING AS AN AQUIFER INDEPENDENT GENERATOR ENERGY BALANCE

INPUT	BTU/HR
1. ATMOSPHERIC AIR:	79,387,310
2. HP COMBUSTOR FUEL (LHV = 18,055 BTU/LB):	637,864,375
3. LP COMBUSTOR FUEL (LHV = 18,055 BTU/LB):	438,652,005
TOTAL INPUT	1.156×10^9

OUTPUT	BTU/HR
1. HP COMPRESSOR GEAR LOSS:	3,161,550
2. INTERCOOLER #1 REJECTION HEAT LOSS:	202,411,440
3. INTERCOOLER #2 REJECTION HEAT LOSS:	214,611,985
4. AIR COOLER DRAIN LOSS:	-14,245,955
5. HP COMBUSTOR LOSS:	6,378,645
6. LP COMBUSTOR LOSS:	4,386,520
7. TOTAL TURBINE POWER (@ GENERATOR TERMINALS):	360,834,100
8. TOTAL TURBINE ELECTRICAL LOSS:	5,494,935
9. EXHAUST GAS LOSS:	374,828,330
TOTAL OUTPUT	1.157×10^9

PLANT PERFORMANCE

TOTAL OUTPUT (@ GENERATOR TERMINALS):	105,750 KW
AUX. LOAD:	2,290 KW
NET TURBINE OUTPUT:	103,460 KW
NET LHV HEAT RATE:	10,405 BTU/KWHR

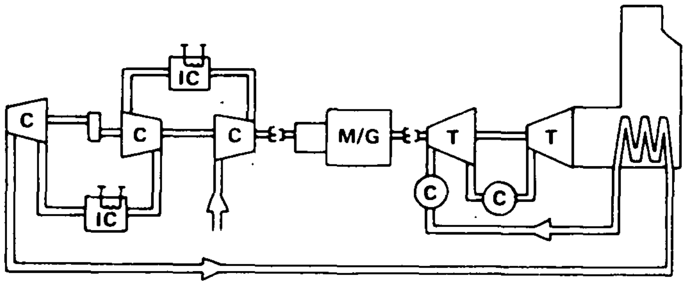
NOTES:

1. THE ABOVE DATA REFLECTS AS CALCULATED PERFORMANCE. GUARANTEED PERFORMANCE WOULD INCLUDE MARGINS (NOT NECESSARILY THE SAME) FOR THE POWER OUTPUT, AND HEAT RATE.
2. THE ABOVE REFLECTS AN ENTHALPY BASE WHERE THE ENTHALPY OF -60°F GAS EQUALS 0.0.

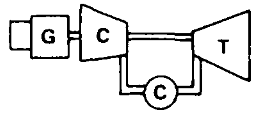
(W 1569)

Figure 10-4. 1000 PSI Nominal Refined CAES System Utilized As An Aquifer Independent Generator Heat And Material Balances

of the air storage system or whenever the storage system was not operational. Figure 10-5 presents a comparison of the CAES turbomachinery operating as an aquifer independent combustion turbine generator and a conventional W501 combustion turbine. As can be observed, the 1500° inlet temperature, 1000 psi system performance is comparable with a much higher firing temperature conventional combustion turbine. Since no input electrical energy is utilized, the unit with the lowest heat rate (the reciprocal of the thermal efficiency after appropriate unit changes) will have the lowest total power production energy cost.



CAES AQUIFER INDEPENDENT GENERATOR



CONVENTIONAL
COMBUSTION TURBINE

	CAES SYSTEM AQUIFER INDEPENDENT OPERATION			CONVENTIONAL COMBUSTION TURBINE W501 OPERATION
	200 PSI 1500/UF °F	600 PSI 1500/1500°F	1000 PSI 1500/1500°F	APPROXIMATELY 2000°F
NET TURBINE OUTPUT, KW	59,070	88,435	98,535	94,715
NET LHV HEAT RATE BTU/KWHR	11,465	10,940	10,925	10,660

NOTES:

1. THE CAES SYSTEM AQUIFER INDEPENDENT PLANT PERFORMANCE INCLUDES MARGINS TO ALLOW COMPARISON WITH TYPICAL PUBLISHED PERFORMANCE DATA FOR A W501 COMBUSTION TURBINE.
2. AN ALLOWANCE HAS BEEN MADE IN THE CAES AQUIFER INDEPENDENT PLANT PERFORMANCE (POWER OUTPUT AND HEAT RATE) FOR THE COOLING TOWER FANS, LOW PRESSURE SERVICE WATER AND WATER PUMPING ELECTRICAL LOADS.
3. RATINGS ARE AT BASE LOAD OPERATION, 14.43 PSIA AND 59°F.

(W 1531)

Figure 10-5. CAES Aquifer Independent and Conventional Combustion Turbine Operation

Section 11

CONTROL SYSTEM PHILOSOPHY

GENERAL

The Compressed Air Energy Storage power plant includes control system requirements like those of both a conventional reheat steam plant and a regenerative open cycle combustion turbine generator. These requirements have been evaluated using known and assumed equipment characteristics. The primary result of this study is a preliminary description of control system operation for various plant operating modes. The secondary result is a list of problem areas which may require special attention during later phases of plant development.

DESCRIPTION OF CONTROL SYSTEM OPERATION

The three modes of operation are:

- Aquifer Dependent Power Production (Mode A)
- Aquifer Charging (Mode B)
- Aquifer Independent Power Production (Mode C)

Each mode will be discussed as it affects control system design. In discussion of all modes, reference is made to Figure 11-1, the MAIN AIR CONTROL SCHEMATIC, which shows the air system values. In discussion of the power making modes, reference is made to Figure 11-2, the TURBINE CONTROL SCHEMATIC.

Aquifer Dependent Power Production

The CAES turbine with its HP and LP combustion systems will normally be operated very much like a reheat steam turbine. Air supplied to the HP turbine comes from the aquifer header system which has its pressure held fairly constant by valves and controls provided separately from the combustion turbine's control system. This header, similar to a pressure controlled steam header, provides air to the HP turbine via the regenerator. The temperature of the air leaving the regenerator

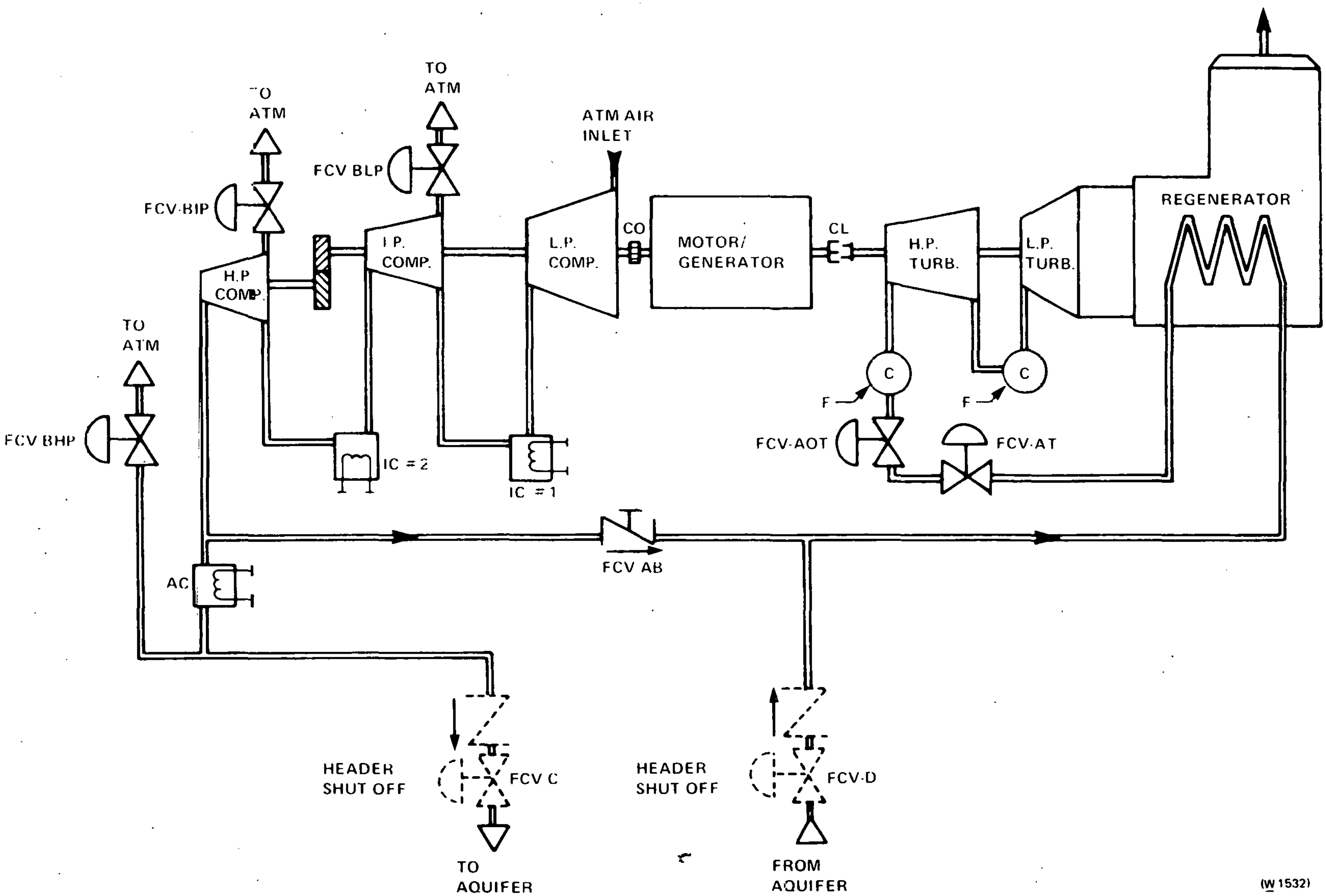


Figure 11-1. Main Air Control Schematic

is uncontrolled, with a maximum of approximately 610°F expected. The three possible aquifer pressure levels being considered would result in regenerator inlet pressures of about 175, 500 or 800 PSIA.

The inlet stop (overspeed trip) valve, FCV-AOT, and the control (throttle) valve, FCV-AT, are located downstream of the large volume regenerator to provide overspeed protection after electrical load dumps. These valves will be sized with some margin to pass 770 pounds/sec. of air at 610°F and at the selected supply pressure. Both valve actuators will be designed having normal steam turbine valve response: trip closing in 0.15 seconds. The control valve will be designed to open in 2.5 to 5 seconds and the stop valve in about 20 seconds.

The turbine air throttle valve is under control of a speed/load controller which provides closed loop megawatt/frequency droop governor action. This controller provides speed control during startup and load dumps. On-line, it causes the turbine output power to respond linearly to line frequency error.

For each combustor system (HP and LP), fuel flow is set by the low-selected output of two temperature controllers: turbine inlet and turbine exhaust. Both HP and LP inlet temperatures will be measured using dual element thermocouples in each combustor. Past experience indicates that Type K thermocouples can be developed to provide reliable operation at 1500°F. Multiple dual thermocouples will also be used to measure exhaust temperatures. The inlet temperature controllers provide the same function as do throttle or reheat steam temperature controllers with a reheat steam turbine. Both inlet controllers' normal set-points are 1500°F, but for startup, their set-points will initially be lower and then will be ramped up at a rate compatible with transient thermal stress limitations of the turbine. For each of the HP and LP turbines, the exhaust temperature controller's output may be low-selected and control fuel flow in the lower load range. This becomes necessary when the turbine's pressure ratio is too small to prevent excessive back-end (blading) temperature with rising inlet temperature. Both HP and LP exhaust temperature controllers will have fixed set-points at about 1100°F.

Variations of air flow and fuel flow, in response to the speed/load controller and the HP and LP temperature controllers, will cause measured variable interactions. These undesirable effects will be minimized by using the speed/load controller signal output, SLCSO, as the input to multipliers which will automatically vary the gain of HP and LP fuel control loops in proportion to air flow.

The operation and sequencing required for starting up, loading, and shutting down the turbine/generator in this mode: Aquifer Dependent Power Production, will now be discussed.

Prestart. Coupling CO is disconnected and the compressors are placed on turning gear. Clutch CL is set to transmit positive and negative torque. Header stop valve, FCV-D, is opened, and header stop valve, FCV-C, and simple cycle aquifer bypass valve, FCV-AB, both remain closed.

Startup to Self-sustaining. The "Start" switch is pushed and the automatic sequence control opens the air overspeed trip valve, FCV-AOT, and places the air throttle valve, FCV-AT, under speed/load control. The speed reference is initially ramped as a function of time. The speed/load controller's signal output, SLCSO, rises from zero to control the opening of the air throttle valve and accelerate the turbine/generator as scheduled. During the acceleration when the SLCSO reaches a preset value corresponding to a desired ignition air flow, the HP turbine's combustors are fired. This is accomplished by the following sequencer operations:

- Open the HP fuel overspeed trip valve.
- Open the HP fuel isolation valve.
- For liquid fuel, start the main fuel pump and activate the pressure controlled bypass valve. For either liquid or gas fuel, the HP throttle valve is initially held at its minimum lift position and acts as a fixed orifice to set ignition fuel flow for existing fuel supply pressure.
- Change the speed/load controller's reference from a scheduled ramp to tracking turbine speed. This zeroes the controller's input error and prevents any further rise in controller output until after flame is sensed.
- Change the speed/load controller "high select" input B from zero to "ignition air." This prevents reduction in air flow to less than the ignition value unless the turbine is tripped.
- Change the HP temperature controller "high select" input B from zero to "minimum" HP fuel setting.

The HP ignition trial period will be about thirty seconds. As soon as flame is sensed, the HP fuel flow will be brought under closed loop control at its minimum setting, slightly greater than that at ignition. The flow rate and temperature of the HP combustor's output will be such as to cause the turbine to accelerate along an uncontrolled speed-versus-time curve involving two time constants. The first time constant involves rotor inertia and turbine/generator net torque versus speed (damping). The second time constant involves the heat capacitance and heat transfer coefficient of the regenerator, and slows the rise in temperature of the air entering the combustor. The fuel and air flow rates selected for HP combustor ignition will be as small as possible for reliable ignition. They must be small enough that, after ignition and initiation of closed loop fuel control, the uncontrolled speed will rise to a maximum that is less than 3600 RPM. If these two requirements (reliable ignition and maximum speed) are mutually exclusive, it will be necessary to increase the fuel flow until flame is sensed and then cut it back to the minimum for reliable combustion.

During the uncontrolled speed rise, the acceleration rate will be measured, and when it drops to a preset rate, the speed reference will revert from tracking the turbine speed to following the scheduled ramp. This will provide bumpless transfer to speed/load control, and the air throttle valve will resume its controlled opening to accelerate the unit to 3600 RPM as scheduled.

From the push of the Start switch, the HP combustor outlet thermocouples are monitored and used for flame detection. The thermocouples are also used to determine the average inlet temperature to the HP turbine. Initially, however, the HP turbine's inlet temperature controller has its reference tracking a few degrees below the uncontrolled HP turbine inlet temperature and this causes the controller output to remain out of control at its lower limit. When the speed/load controller resumes control as described above, the temperature controller's reference is changed from its tracking mode to ramping up to the rated temperature, 1500°F. During startup and lower load operation, a "hold" of the HP inlet temperature reference ramping will occur if the HP exhaust temperature rises to 1100°F and causes the exhaust temperature controller to assume control.

The turbine/generator, with no compressor load, will normally reach synchronous speed in about five minutes. In addition to the analog speed/load and temperature controls described above, there will be alarm and trip functions to protect against overtemperature, gas/metal temperature mismatch, vibration, and such other contingencies requiring monitoring and control. It is possible that there will be mechanical requirements for rotor "soaking" which will extend the cold startup time.

Synchronizing and Loading. The turbine/generator is placed "on-line" at minimum load by an automatic synchronizer or manually. Once "on-line", the sequence logic will normally initiate firing of the LP combustors unless the operator chooses manual LP combustor control. In either case, the following sequence of operations is accomplished automatically to fire the LP combustors:

- Open the LP fuel overspeed trip valve.
- Open the LP fuel isolation valve.

  (For either liquid or gas fuel, the LP throttle valve is initially held at its closed position.)
- Change the LP temperature controller "high select" input B from zero to "minimum" LP fuel setting.
- As soon as LP fuel flow is sensed, it is brought under closed loop control at the "minimum" setting.
- The LP turbine's inlet temperature controller has its reference tracking a few degrees below the LP turbine inlet temperature, and this causes the controller output to remain out of control at its lower limit.
- Fuel flow set-point is the high selected signal: "minimum" fuel at input B.

  The LP turbine's exhaust temperature controller has its reference fixed at 1100°F, and this causes the controller's output to remain out of control at its upper limit.

The LP ignition trial period will be about thirty seconds. The fuel flow, during the ignition period until flame is sensed, increases from zero to the controlled flow rate determined by the LP minimum fuel setting multiplied by the air flow demand signal, SLCSO. Thus, the ignition fuel flow for the LP turbine is proportional to the LP turbine's air flow. Reliable ignition should be possible over a wide range of starting and loading conditions.

As soon as flame is sensed, the LP inlet temperature controller has its reference changed from its tracking mode to ramping up to the rated temperature, 1500°F. During operation in the lower load range, a "hold" of the LP inlet temperature reference ramping will occur if the LP exhaust temperature rises to 1100°F and causes the exhaust temperature controller to assume control.

The turbine/generator is loaded and unloaded by manual or automatic control of the speed/load controller's reference. For constant line frequency, closed loop megawatt control makes the generator output linearly proportional to the reference setting and independent of the regenerator's outlet pressure and temperature and of HP and LP combustors' outlet temperatures.

Load Drops. Rapid loss of generator load is caused by a:

- Turbine trip
- Generator trip
- Electrical line frequency rise
- Electrical line short circuit fault.

While it is beyond the scope of this work to design all the control system's functions for the various situations listed, the following ideas are pertinent:

- Normal response of the speed/load controller to load loss will be limited to reducing air flow to the "ignition air" setting of the high select's input B.
- A generator trip will necessitate a shut-off of LP fuel.
- A turbine trip may be necessitated by a generator trip if air mass storage and turbine speed are both excessive.
- A turbine trip causes shutoff of all fuel and closing of FCV-AOT and FCV-AT. The turbine will be motorized if the generator is not tripped.
- Turbine cooling air flow may be required when the turbine is being motorized.
- HP combustors can be refired at any turbine speed and whether or not the generator is tripped.
- Clutch CL is held engaged at all times for the power making mode, thereby utilizing turbine windage to decelerate the motor/generator after any overspeed transient.

Aquifer Charging and Aquifer Independent Power Production

The motor/generator, acting as a motor, uses off-peak electrical power to drive the compressors. However, the motor depends on the combustion turbine to bring it to synchronous speed and the aquifer must supply starting air until the unit becomes self-sustaining both for the aquifer charging and aquifer independent power production modes. It is expected that it is necessary to fire the LP combustors in order to reach synchronous speed. The control requirements for these modes -- Aquifer Charging and Aquifer Independent Power Production -- will now be discussed.

Prestart. Coupling CO is connected and Clutch CL is set to transmit positive and negative torques. Header stop valves FCV-C and FCV-D are opened along with simple cycle aquifer bypass valve, FCV-AB. Compressor bleed valves FCV-BLP, FCV-BIP, and FCV-BHP are opened to the atmosphere under automatic sequencer control. The LP compressor's inlet guide vanes are in a scheduled position to prevent compressor surge during startup. The cooling water control valves for air coolers, IC1, IC2, and AC are under temperature control ready to open automatically to maintain cooler air discharge temperatures at their required values.

Startup to Synchronous Speed. The "Start" switch is pushed and the automatic sequence control opens the air overspeed valve, FCV-AOT, and places the air throttle valve, FCV-AT, under speed/load control. The speed reference is initially ramped as a function of time. The speed/load controller's signal output, SLCSO, rises from zero to control the opening of the turbine's air throttle valve, FCV-AT, and accelerates the turbine/generator as scheduled.

During the acceleration when the SLCSO reaches a preset value corresponding to the desired ignition air flow, the HP turbine's combustors are fired. The necessary sequencer operations are the same as for the aquifer dependent mode, but the ignition speed will be lower because the compressors will absorb considerable power even before ignition. The LP combustors will be fired automatically as soon as HP combustor "flame-on" is verified and without waiting until after the unit is put on-line. Compressor power absorption will quickly reduce the uncontrolled acceleration rate to the scheduled rate and thereby initiate the further increase of air and fuel flows as described for the aquifer dependent mode.

As the unit nears synchronous speed, the sequencer will automatically initiate any required repositioning of the inlet guide vanes and close the three bleed valves. Bleed valve sequencing will involve first closing FCV-BLP and verifying its closure, then closing FCV-BIP and verifying its closure, and finally closing FCV-BHP and verifying its closure. As these valves are closed and the inlet guide vanes move to their proper position, the HP compressor's discharge pressure will rise to the pressure of the aquifer air header, and air will begin to flow through FCV-C and FCV-AB. The unit is now self-sustaining, and from this point on, the control sequence depends on whether the operator has chosen to start for Aquifer Charging or for Aquifer Independent Power Production.

Self-sustaining to Aquifer Charging. The transition from turbine drive to motor drive of the compressors requires that the motor/generator be placed on-line at near zero load. Then the turbine power output is gradually reduced until the full 185 megawatts of compressor power is being supplied from the electrical line. The following control actions are required:

- Clutch CL is reset to transmit only positive torques.
- The speed/load controller's reference is ramped down, decreasing the air flow. When the net power output of the turbine reaches zero, clutch CL automatically disconnects the turbine from the motor/generator and the turbine begins to slow below synchronous speed. The air flow and fuel flow at the point of turbine disconnect are not yet known, but air flow will not be reduced below the minimum for stable combustion. If this minimum is reached before the clutch disconnects, the LP combustor fuel will be tripped off and then, if necessary, the HP inlet temperature controller will have its reference ramped down.
- When the turbine speed decreases to an underfrequency set-point, the turbine is automatically tripped. Both air and fuel flows are shut off by step closing of the overspeed trip, isolation, and throttle valves. The turbine decelerates to turning gear speed without the blading being cooled as it would in a simple cycle unit. If regenerator depressurization is desired, valves FCV-AB and FCV-D can be closed by the operator.

Self-sustaining to Aquifer Independent Power Production. The unit is self-sustaining as soon as the HP compressor's discharge air flow matches the flow through air throttle valve, FCV-AT. While this may occur before the compressor bleed valves are closed, it does not seem necessary to isolate the unit from the aquifer before reaching synchronous speed. Therefore, to reduce control system complexity, the unit will be synchronized and partially loaded under speed/load control of the air throttle valve. The only difference in operation from the Aquifer Dependent Power Production mode is that the compressor will be absorbing considerable power and producing considerable air. The operator can choose to make the transition to Aquifer Independent Power Production at any load. The following control actions are required:

- Valves FCV-C and FCV-D are closed to isolate the unit from the aquifer and make it a simple cycle unit.
- The air throttle valve, FCV-AT is removed from speed/load control and ramped to its full open position. This reduces the regenerator and HP compressor discharge pressures, and increases the thermodynamic cycle efficiency. In this simple cycle mode, the throttle valve no longer can be used to control the air flow which instead

is determined by the compressor train's characteristics including the effects of LP ambient pressure and temperature, inlet guide vane position and intercooler outlet air temperatures and pressures.

- Control of the HP and LP fuel flows is transferred from variable gain temperature control to speed/load control. The speed/load controller's output is low-selected with the outputs of the HP and LP inlet and exhaust temperature controllers which provide temperature high limit control. The MODE SEL. blocks, shown on Figure 11-2, the TURBINE CONTROL SCHEMATIC, includes the logical controls to transfer control from the MODE A inputs to the MODE C inputs bumplessly.

Completion of the transition to MODE C control results in the regenerator's mass and energy storages being included in the speed/load control loop dynamics. The effect of the regenerator is to delay the rise in HP turbine inlet pressure and flow following an increase in HP fuel flow. The speed/load controller will tend to offset this delay by transiently increasing fuel flow so that the delay in increase of regenerator outlet temperature is compensated by extra HP combustor fuel, and the delay in increase of HP turbine flow (and to a lesser extent power) is compensated by extra flow into, and power from the LP turbine.

POSSIBLE PROBLEM AREAS:

The following initial list of control problem areas which may require special attention during later phases of plant development is offered to stimulate thinking. Obviously, as the design of the plant proceeds, many problems will be perceived and solved in the ordinary course of control system design. Therefore, the "problem areas" listed are really suggestions as to future tasks.

- <u>Control Hardware.</u> The signal processing functions shown on Figure 11-2, the TURBINE CONTROL SCHEMATIC, can be performed by various combinations of solid state logic and analog components, relay sequencers, and microprocessor based control and information systems. While the existing Westinghouse POWERLOGIC turbine control system could be extended to include the additional functions required by the more complex CAES system, there would seem to be good reasons to consider the application of redundant digital control.

- <u>Final Control Elements.</u> The size and transient response requirements of the air control valves points to the application of electro-hydraulic actuators operating with 2000 psig supply systems. This technology, already in common use with steam turbines, might also need to be extended to the fuel control systems to achieve compatible control responses.

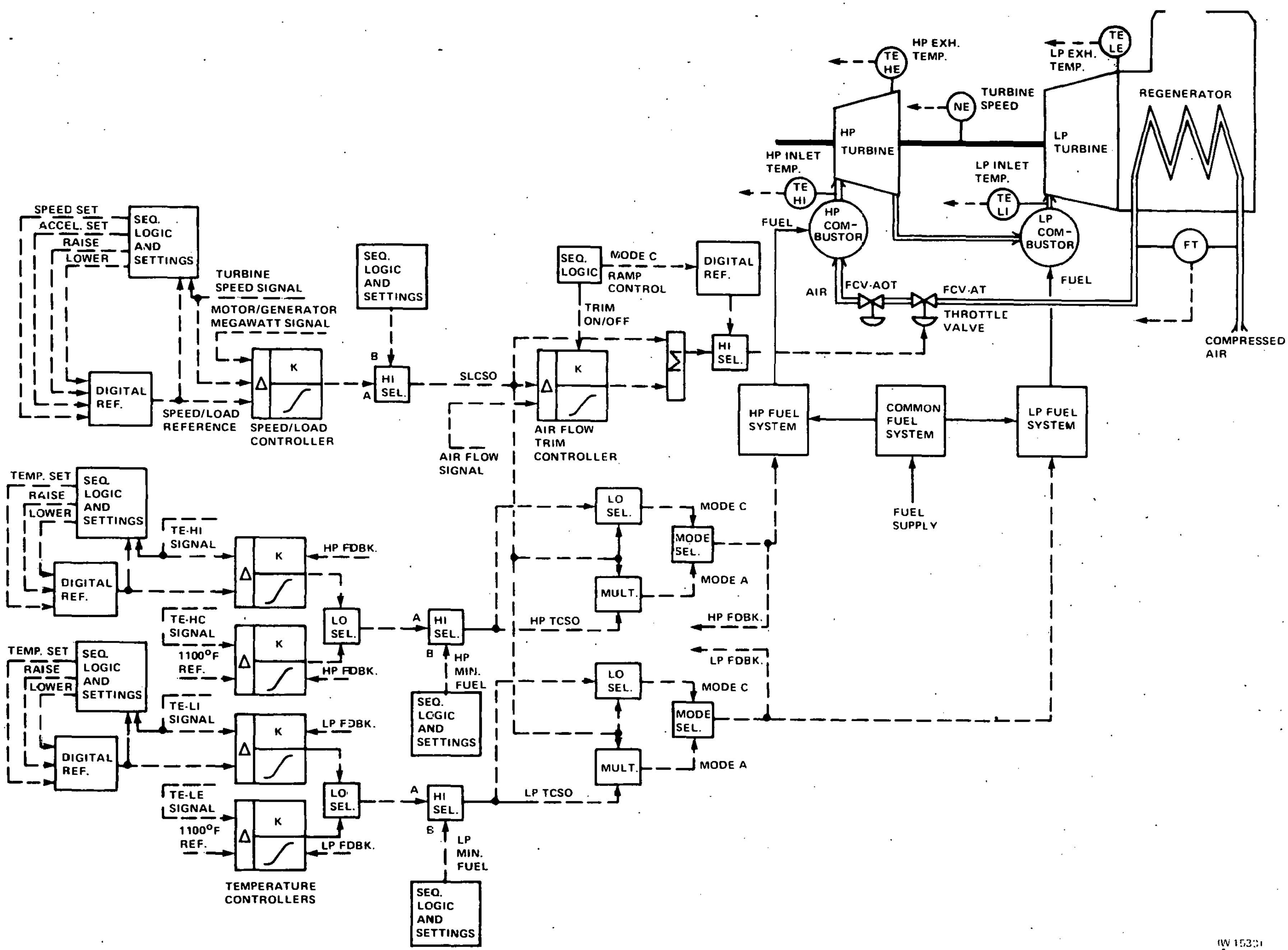


Figure 11-2. Turbine Control Schematic

- Combustors. Combustors, designed to operate with large air and fuel turndown ratios, must be mated with suitable reliable temperature and fuel-flow sensors and such purge and atomizing systems as are required to achieve very high levels of starting reliability.

- Plant Dynamics. The complex nature of the plant with its multiple compressors, turbines, combustors and fuel system, regenerator, and motor/generator makes desirable the development of a computer model for dynamic simulation of the plant with its controls. This simulation would include the steady state and transient characteristics of the compressors, air coolers, air bleed valves, air throttle valve, combustors, turbines, and regenerator as they are determined by the equipment designers. It would provide the means for evaluating startup and loading requirements of equipment and controls.

- Electrical Network and Mechanical Equipment Dynamics. The characteristics of the electrical network and the system load curve will affect the design of the CAES plants' controls as regards the needs for fast valving and response to any Area Generation Controller. Also, the rate of plant loading and unloading and the operating cycle for aquifer charging and aquifer dependent power production will affect the needs for hot standby control and equipment condition monitoring. Included here are the possible needs for hot standby heating of turbine casings, and for monitoring of differential expansion, eccentricity, and various metal temperatures.

Section 12

EQUIPMENT PRICE ESTIMATES

Budgetary price estimates which included the projected price for the LP, IP and HP compressors, gearbox, couplings and turning gear required for the compressor train; the motor/generator, exciter, voltage regulator and protective relays required for the electric dynamo; the high and low pressure turbine, clutch and turning gear required for the turbine train; the control panel, sequencer, sensors, throttle and overspeed trip valves, compressor surge valves and the simple cycle shut-off valve required for the control system; the mechanical skids, fuel packages, compressor, generator and turbine piping skid, and the NO_x water injection equipment required for the Mechanical Support Equipment; the intercoolers; the aftercoolers; the inlet filters and silencers; the exhaust silencers and stacks; and the regenerators were prepared in Task 1 to support the aquifer site evaluation performed in Task 2, "Characterize and Explore Potential Sites and Prepare Site Research and Development Plan." Early in the program, prior to the selection of the preferred heat cycles, it was necessary that a gross estimate be made as an input to the site ranking and evaluation being performed concurrently in Task 2. These estimates, for each of the nominal pressure levels, were:

	200 PSI System	600 PSI System	1000 PSI System
Dollars per Kilowatt ($ 1979)	190	150	165

These price estimates were revised, based upon the selected heat cycles and machinery configurations, and the use of standardized components in the low and intermediate pressure sections of each system. In the initial estimate the effect of the higher pressure, in the 1000 PSI system, on the cost of the equipment was overemphasized. This is apparent by noting that the above price trend changed and by noticing the much lower price of the 1000 PSI machinery in the revised estimate presented below.

	200 PSI System	600 PSI System	1000 PSI System
Dollars per Kilowatt ($ 1979)	186	151	134

In determining the price per kilowatt, a 5% reduction or margin allowance was applied to the net turbine output calculated from the heat and material balances. No allowance was taken for the balance of plant power requirements.

These estimates do not include installation and field support costs, interconnecting main air and water piping and ducting, or the Phase III design and development costs. All estimated prices are in 1979 dollars. The above prices do not include transportation, or, taxes normally imposed upon the sale of this type of equipment. The standard warranty on material applies.

These prices are preliminary since they are based upon the design criteria generated in Task 1, and as such they are subject to refinement as the physical definition of the machinery is undertaken in subsequent tasks in this program. This should be taken into consideration when utilizing these prices in the Task 1 utility benefits pricing runs to project plant economics. If possible, the initial pricing runs should also be used to identify target prices which could then be guides in developing the preliminary design of the machinery.

After the identification of the aquifer characteristics in Task 2, the final machinery configuration will be selected in Task 3 for the remaining work to be performed in this Phase of the program.

In Task 5 the preliminary design of the Task 3 configuration will be performed and a final price estimate will be prepared for this Phase (II) of the program.

Appendix A

THE OPTIMUM COMPRESSOR TRAIN PRESSURE RATIO SPLIT

In order to minimize the total compressor work (and therefore charging cost) required to charge the air storage aquifer of a CAES system, the compressor train intercooler(s) should be placed at an optimum location(s). The equations resulting from this analysis allow the direct calculation of the pressure ratio required by each compressor for the most efficient charging process.

The following assumptions were made:

(1) Pressure ratios that are constant with respect to time

(2) Mass flow rates that are constant with respect to time

(3) Intercooler(s) exit temperature(s) that is constant with respect to time

(4) Constant compressor efficiencies

(5) Frictionless flow

(6) Dry, ideal air

The following analysis is broken up into two parts. The first section deals with a compressor train, using one intercooler while the second part handles a compressor train utilizing two intercoolers.

THE OPTIMUM PRESSURE RATIO SPLIT FOR A COMPRESSOR TRAIN UTILIZING ONE INTERCOOLER

This configuration (see Figure A-1) consists of a LP compressor followed by an intercooler then the HP compressor. The total work for this arrangement can be expressed as:

$$W_{\text{Compressor TOTAL}} = W_{CT} = W_{LPC} + W_{HPC}$$

$$= \frac{m_1 cp_1\ T_1\ (\rho_1^{\left(\frac{K-1}{K}\right)} - 1)}{\eta_1} + \frac{m_2 cp_2\ T_2\ (\rho_2^{\left(\frac{K-1}{K}\right)} - 1)}{\eta_2}$$

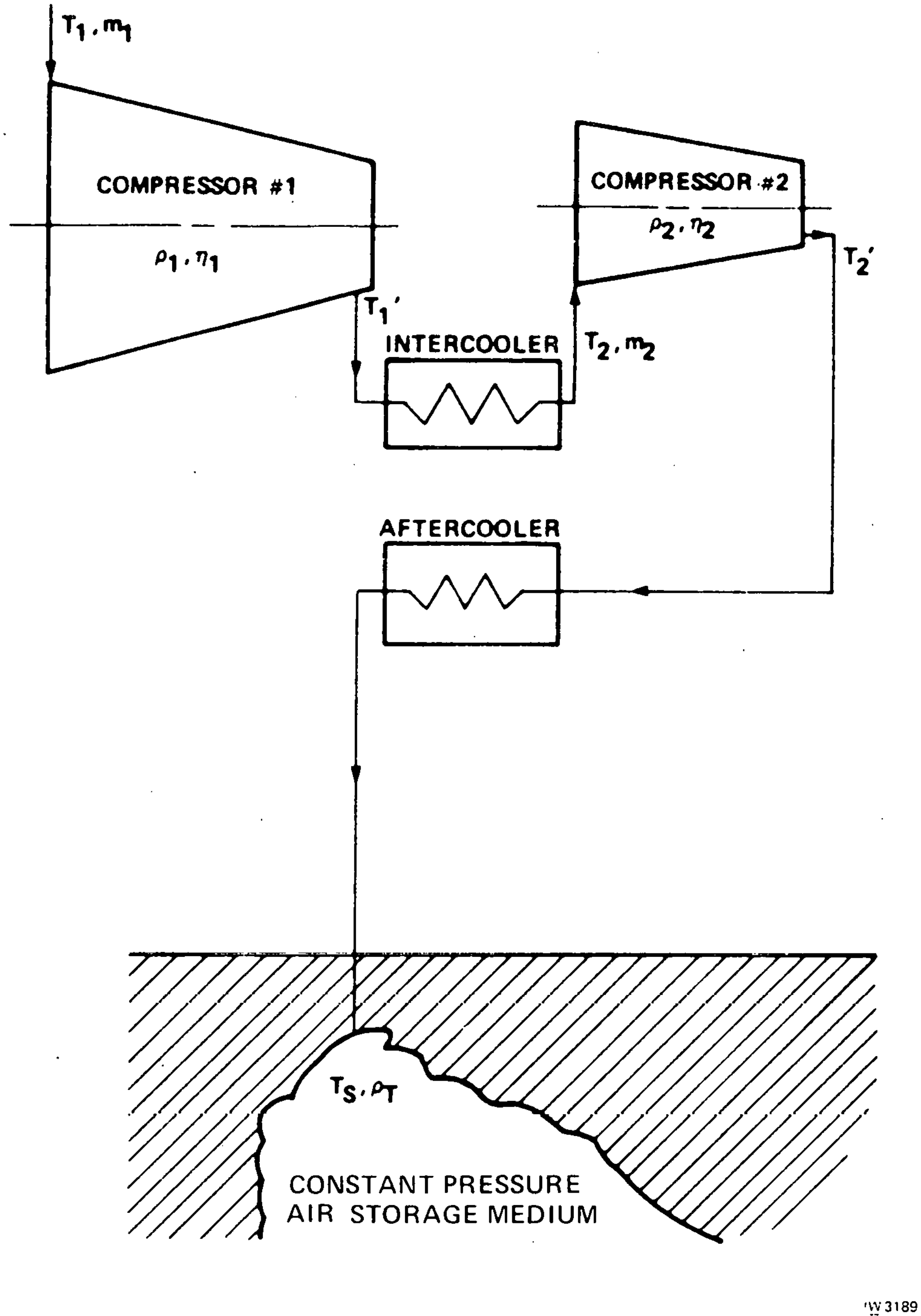


Figure A-1. Schematic of a CAES Compressor Train Utilizing One Intercooler by substituting $\rho_2 = \frac{\rho_T}{\rho_1}$ into the above equation one obtains:

$$W_{CT} = \frac{m_1 cp_1 T_1 \left(\rho_1^{\left(\frac{K-1}{K}\right)} - 1\right)}{\eta_1} + \frac{m_2 cp_2 T_2 \left[\left(\frac{\rho_T}{\rho_1}\right)^{\left(\frac{K-1}{K}\right)} - 1\right]}{\eta_2}$$

The partial derivative of the above expression with respect to ρ_1 is equal to:

$$\frac{\partial W_{CT}}{\partial \rho_1} = \frac{m_1 cp_1 T_1 \left(\frac{K-1}{K}\right) \rho_1^{\left(\frac{K-1}{K}\right) - 1}}{\eta_1} + \frac{m_2 cp_2 T_2 \rho_T^{\left(\frac{K-1}{K}\right)} - \left(\frac{K-1}{K}\right) \rho_1^{\left(\frac{K-1}{K}\right) - 1}}{\eta_2}$$

To find the value of ρ_1 required to permit the minimum compressor train work, the above expression is set equal to zero. After a few algebraic simplification steps, the resulting expression is:

$$\rho_1 = \left(\frac{m_2 cp_2 T_2 \eta_1}{m_1 cp_1 T_1 \eta_2}\right)^{\left(\frac{K}{2(K-1)}\right)} (\rho_T)^{1/2}$$

and

$$\rho_2 = \frac{\rho_T}{\rho_1}$$

where:

- m = the charging mass flow rate
- cp = the constant pressure specific heat of air
- η = compressor adiabatic efficiency
- ρ = compressor pressure ratio
- ρ_T = total compressor train pressure ratio
- T = compressor inlet temperature (^{o}R)
- K = specific heat ratio of air

and the subscripts 1 and 2 refer to the LP and HP compressor, respectively.

THE OPTIMUM PRESSURE RATIO SPLIT FOR A COMPRESSOR TRAIN UTILIZING TWO INTERCOOLERS

This configuration consists of three compressor stages and two intercoolers (see Figure A-2). The total work for this arrangement can be mathematically expressed as:

$$W_{\text{Compressor TOTAL}} = W_{CT} = W_{LPC} + W_{IPC} + W_{HPC}$$

$$= \frac{m_1 cp_1 T_1 (\rho_1^{\left(\frac{K-1}{K}\right)} - 1)}{\eta_1} + \frac{m_2 cp_2 T_2 (\rho_2^{\left(\frac{K-1}{K}\right)} - 1)}{\eta_2} + \frac{m_3 cp_3 T_3 (\rho_3^{\left(\frac{K-1}{K}\right)} - 1)}{\eta_3}$$

by letting

$$B = \left(\frac{K-1}{K}\right) \quad \text{and} \quad C_1 = \frac{m_1 cp_1 T_1}{\eta_1}; \; C_2 = \frac{m_2 cp_2 T_2}{\eta_2}; \; C_3 = \frac{m_3 cp_3 T_3}{\eta_3}$$

one obtains:

$$W_{CT} = C_1 \rho_1^B - C_1 + C_2 \rho_2^B - C_2 + C_3 \rho_3^B - C_3$$

Using the method of Lagrance Multipliers[1] one can let f be equal to W_{CT} and g be equal to the following equation:

$$\rho_1 \rho_2 \rho_3 - \rho_T = 0.$$

Next the following partial derivatives are found.

$$\frac{\partial f}{\partial \rho_1} + \lambda \frac{\partial g}{\partial \rho_1} = 0 \quad \Longrightarrow \quad B\, C_1 \rho_1^{B-1} + \lambda \rho_2 \rho_3 = 0 \tag{1}$$

$$\frac{\partial f}{\partial \rho_2} + \lambda \frac{\partial g}{\partial \rho_2} = 0 \quad \Longrightarrow \quad B\, C_2 \rho_2^{B-1} + \lambda \rho_1 \rho_3 = 0 \tag{2}$$

$$\frac{\partial f}{\partial \rho_3} + \lambda \frac{\partial g}{\partial \rho_3} = 0 \quad \Longrightarrow \quad B\, C_3 \rho_3^{B-1} + \lambda \rho_1 \rho_2 = 0 \tag{3}$$

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[1] Kaplan, Wilfred, "Advanced Calculus," 2nd Ed., Reading, Massachusetts, Addision - Wesley, 1973.

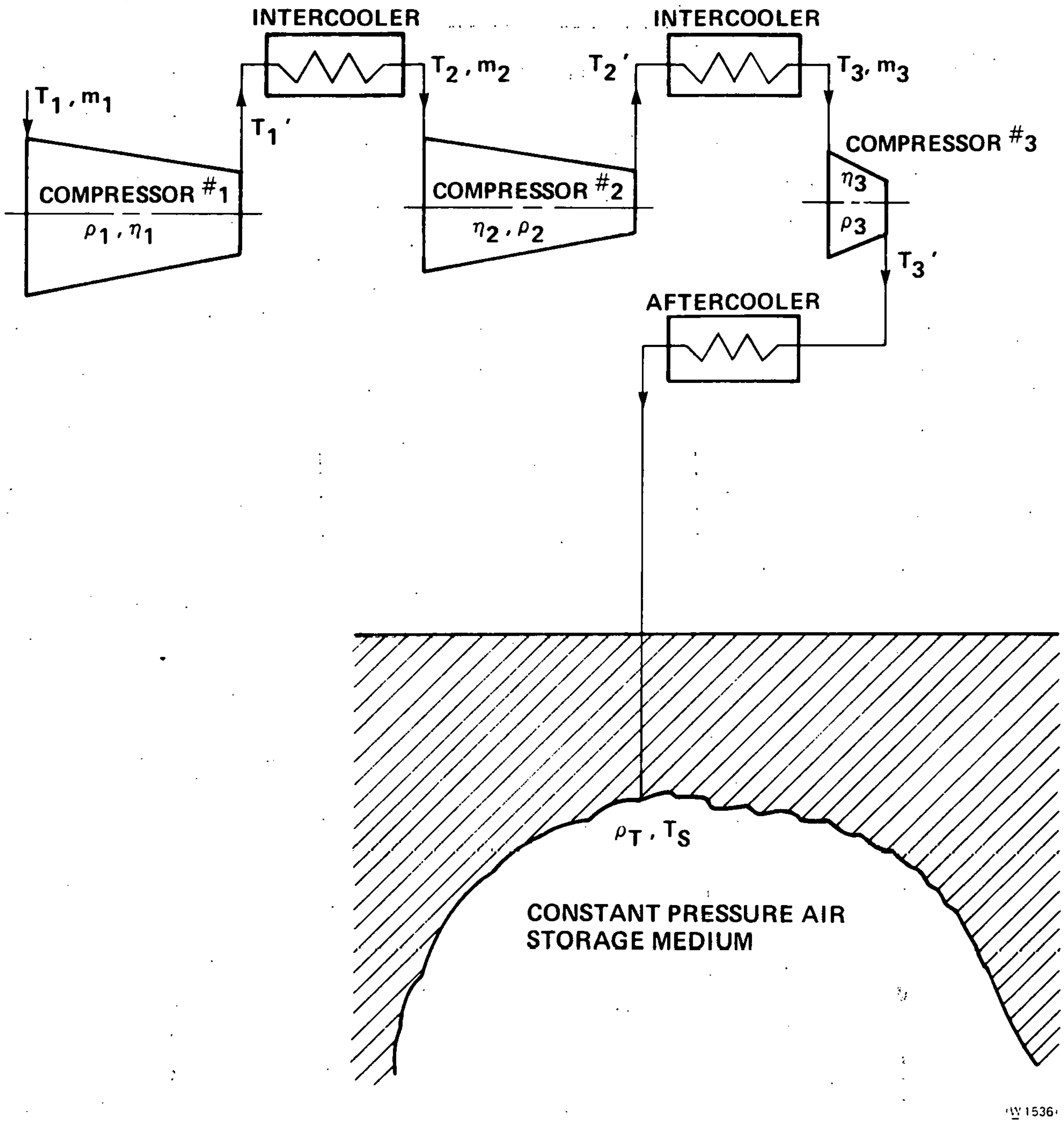

Figure A-2. Schematic of a CAES Compressor Train Utilizing Two Intercoolers

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and the following equation, g, is used:

$$\rho_1 \rho_2 \rho_3 = \rho_T \tag{4}$$

The above four equations contain four unknowns:

ρ_1, ρ_2, ρ_3 and λ .

Solving for ρ_1 one obtains:

$$\rho_1 = \left(\frac{C_2 C_3}{C_1 C_1}\right)^{1/3B} (\rho_T)^{1/3}$$

solving for ρ_2 one obtains:

$$\rho_2 = \left(\frac{C_1}{C_2}\right)^{1/B} \rho_1$$

and solving for ρ_3 one obtains:

$$\rho_3 = \frac{\rho_T}{\rho_1 \rho_2}$$

replacing the constant C_1, C_2, C_3 and B in the above equation one obtains:

$$\rho_1 = \left(\frac{m_2 cp_2 T_2 \; {}_1 m_3 cp_3 T_3 \eta_1}{m_1 cp_1 T_1 \; {}_2 m_1 cp_1 T_1 \eta_3}\right)^{\frac{K}{3(K-1)}} (\rho_T)^{1/3}$$

$$\rho_2 = \left(\frac{m_1 cp_1 T_1 \eta_2}{m_2 cp_2 T_2 \eta_1}\right)^{\frac{K}{K-1}} \rho_1$$

$$\rho_3 = \frac{\rho_T}{\rho_1 \rho_2}$$

where:

m = the charging mass flow rate

cp = the constant pressure specific heat of air

η = compressor adiabatic efficiency

ρ = compressor pressure ratio

ρ_T = compressor train total pressure ratio

$K = \frac{cp}{cv}$, the ratio of specific heat of air

T = compressor inlet temperature, (^oR)

and the subscripts 1, 2 and 3 refer to the LP, IP and HP compressor, respectively.

Appendix B

THE PRESSURE RATIO SPLIT REQUIRED BY A REHEATED CAES COMBUSTION TURBINE TO PRODUCE MAXIMUM WORK

In order to maximize the total turbine work produced by a CAES system, given fixed turbine efficiencies, firing temperatures, mass flow rates and total pressure ratio, the reheat has an optimum location. The equation resulting from this analysis allows the direct calculation of the pressure ratio required by each turbine to produce the maximum total turbine work possible under the above restrictions.

The following assumptions were made:

(1) Constant with respect to time pressure ratios

(2) Constant with respect to time mass flow rates

(3) Fuel mass not automatically added to the total mass flow rate

(4) Constant with respect to time combustor exit temperatures

(5) Constant turbine efficiencies

(6) Frictionless flow

(7) Dry, ideal air

The turbine configuration analyzed appears in Figure B-1. The total work for this arrangement can be expressed as:

$$W_{\text{Turbine Total}} = W_{TT} = W_{T1} + W_{T2}$$

$$= m_1 cp_1 T_1 \eta_1 (1 - \rho_1^{-B}) + m_2 cp_2 T_2 \eta_2 (1 - \rho_2^{-B})$$

where:

$$B = \frac{K-1}{K}$$

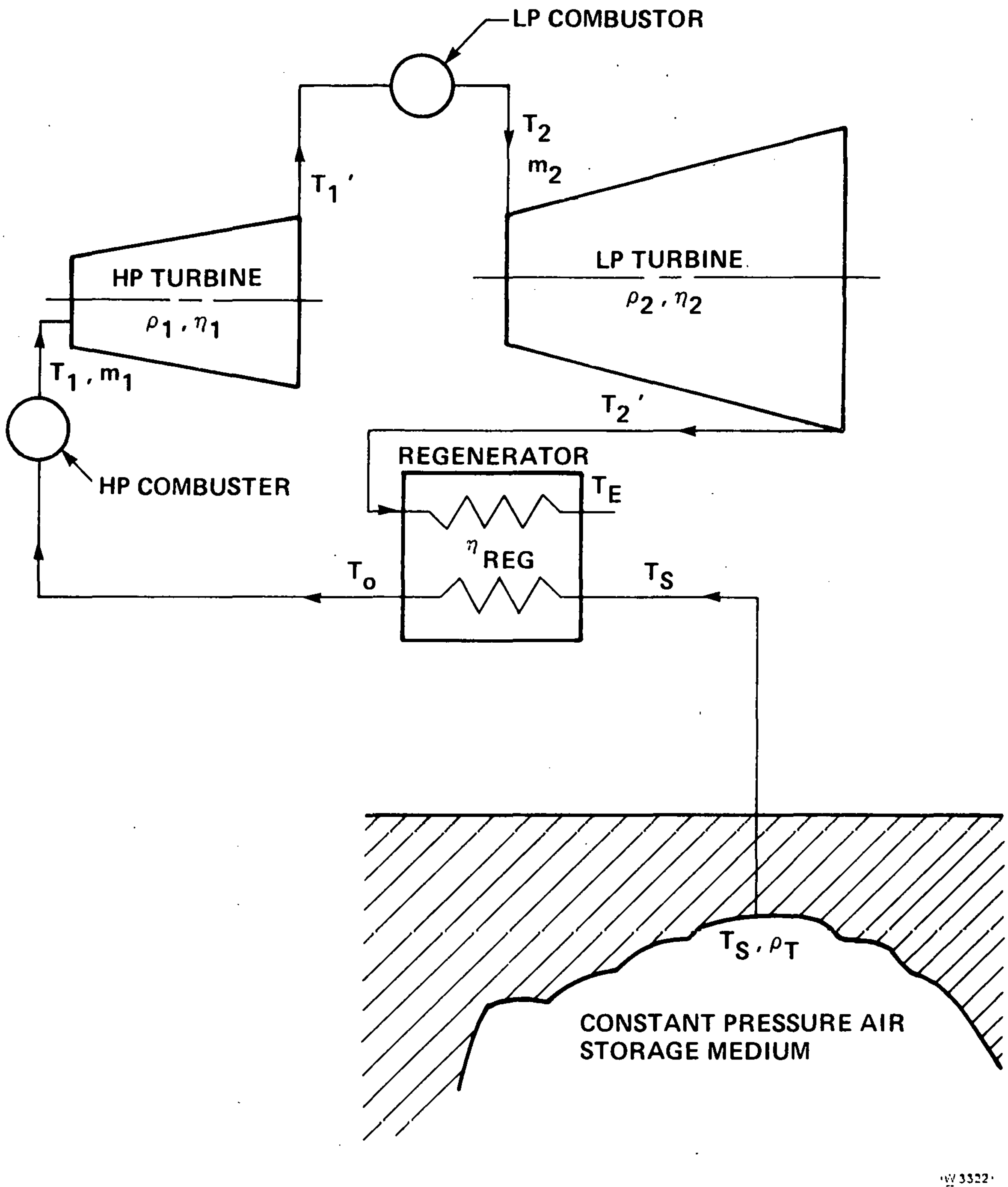

Figure B-1. Schematic of a CAES Turbine Train with Reheat

Since $\rho_1 \rho_2 = \rho_T$, the above expression can be expressed as:

$$W_{TT} = m_1 cp_1 T_1 \eta_1 \left(1 - (\rho_1)^{-B}\right) + m_2 cp_2 T_2 \eta_2 \left(1 - \left(\frac{\rho_T}{\rho_1}\right)^{-B}\right)$$

The partial derivative of the above expression with respect to ρ_1 is equal to:

$$\frac{\partial W_{TT}}{\partial \rho_1} = m_1 cp_1 T_1 \eta_1 B (\rho_1)^{-B-1} - m_2 cp_2 T_2 \eta_2 (\rho_T)^{-B} B (\rho_1)^{B-1}$$

To find the value of ρ_1 required to produce the maximum turbine work, the above expression is set equal to zero. After a few algebraic simplification steps, the resulting expression for ρ_1 is:

$$\rho_1 = \left(\frac{m_1 cp_1 T_1 \eta_1}{m_2 cp_2 T_2 \eta_2}\right)^{\frac{K}{2(K-1)}} (\rho_T)^{1/2}$$

where:

- m = the discharging mass flow rate
- cp = the constant pressure specific heat of air
- η = turbine adiabatic efficiency
- ρ_1 = HP turbine pressure ratio
- ρ_T = the total turbine pressure ratio
- $K = \frac{cp}{cv}$, the ratio of the specific heats of air
- T = the inlet temperature to a turbine, (^{o}R)

and the subscripts 1 and 2 refer to the HP and LP turbine, respectively.

The above equation is also valid for a turbine configuration utilizing a heat storage device instead of a recuperator, or a two stage turbine not utilizing either.

An equation for a special regenerator configuration — one not utilizing a HP combustor — was also developed. In such a configuration, T_1 will be equal to T_o. T_o can be expressed as:

$$T_o = T_1 = (\eta_{reg} [\{T_2 + \eta_2 T_2 (\rho_2^{\frac{K-1}{K}} - 1)\} - T_s]) + T_s$$

where:

η_{reg} = the recuperator effectiveness

T_o = the recuperator compressed air exit temperature, (^oR)

T_1 = the HP turbine inlet temperature, (^oR)

T_2 = the LP turbine inlet temperature, (^oR)

T_s = the air storage temperature, (^oR)

η_2 = the LP turbine efficiency

ρ_2 = the LP turbine pressure ratio

K = $\frac{cp}{cv}$, the specific heat ratio of air

Substituting the above expression for T_1 into the above maximum turbine work equation results in:

$$\rho_1 = (\rho_T)^{1/2} \left(\frac{m_1 cp_1 \eta_1 \, [(\eta_{reg} [\{ T_2 + \eta_2 T_2 (\rho_2^{\frac{K-1}{K}} - 1)\} - T_s]) + T_s]}{m_2 cp_2 \eta_2 T_2} \right)^{\frac{K}{2(K-1)}}$$

Appendix C

CAES POWER PLANT TOTAL POWER PRODUCTION ENERGY COST

The total power production energy cost consists of the sum of the "off-peak" electric power charging cost and the turbine fuel cost. Therefore, the optimum (lowest total power production energy cost) CAES cycle is dependent on the turbine heat rate, the work of compression and the relative cost of the turbine fuel with respect to the off-peak electrical power used to charge the aquifer.

The data included in Appendix C was generated from analyses using optimum (minimum work) axial compressor trains, utilizing one and two intercoolers, and regenerative and regenerative reheat turbine cycles. During the initial investigations the reheat turbine trains were analyzed with the reheat combustor placed to produce maximum turbine work which from earlier studies was known to be near the optimum reheater location.

The total power production energy costs presented were calculated at various turbine inlet temperature combinations, reheater locations, regenerator effectiveness, storage pressure loss ratios and fuel/off-peak electrical energy cost ratios.

Appendix C Table Legend:

CASF = Identifies sequential runs, except on the 1500/1500^{o}F runs, which were performed to find the optimum reheater location, where the case number reflects the LP turbine pressure ratio.

T1HPT = High pressure turbine inlet temperature in ^{o}F.

T2HPT = High pressure turbine exhaust temperature in ^{o}F.

T1LPT = Low pressure turbine inlet temperature in ^{o}F.

CAV LOS = Storage pressure loss ratio from the aftercooler exit to the aquifer inlet or from the aquifer exit to the regenerator inlet. Therefore, a CAV LOS of 1.20 represents a system storage pressure loss ratio of 1.44 (1.20 x 1.20).

REG EFT = Regenerator Effectiveness

COMP KW = Work of compression in KW/LB/SEC.

TURB KW = High Pressure + Low Pressure Turbine Net Work Output in KW/LB/SEC.

H.R. = Heat rate in BTU/KWHR based on a fuel oil #2 lower heating value of 18,055 BTU/LB.

The cost calculations are arranged in order of increasing total power production energy costs.

The term FUEL COST represents the cost of the fuel oil burned in the combustors in units of dollars per million BTU. The term ELECT COST represents the cost of the surplus electrical power utilized to charge the aquifer in units of mils per kilowatt hour. The term ENERGY COST RATIO is equal to the FUEL COST divided by the ELECT COST.

Total power production energy costs are calculated for eight energy cost combinations. The first line of calculations for each energy cost group is the turbine fuel costs in MILS per generated KWHR, and the second line of calculations is the compressor charging power cost in MILS per generated KWHR. The third line of calculations in the total power production energy cost which is the sum of the above two values and of course has the units of MILS per generated KWHR.

It should be noted that the CAV LOS (storage pressure loss ratio) and the REG EFT (regenerator effectiveness) are varied in the calculations, and must be considered when comparing the total power production cost at the various turbine firing temperatures.

The following is an index for the tables included in this Appendix.

DATA INDEX CHART

Aftercooled Discharge Pressure, PSIA	Number of Intercoolers	Cooled or Uncooled Disks	Remarks	Table No.
250	1	Cooled		C-1
250	2	Cooled		C-2
700	1	Cooled		C-3
700	1	Cooled	1500/1500°F at Various LPT Pressure Ratios	C-4
700	2	Cooled		C-5
700	2	Cooled	1500/1500°F at Various LPT Pressure Ratios	C-6
1150	1	Cooled		C-7
1150	1	Cooled	1500/1500°F at Various LPT Pressure Ratios	C-8
1150	2	Cooled		C-9
1150	2	Cooled	1500/1500°F at Various LPT Pressure Ratios	C-10
250	1	Uncooled		C-11
250	2	Uncooled		C-12
700	1	Uncooled		C-13
700	2	Uncooled		C-14
1150	1	Uncooled		C-15
1150	1	Uncooled	1500/1500°F at Various LPT Pressure Ratios	C-16
1150	2	Uncooled		C-17
1150	2	Uncooled	1500/1500°F at Various LPT Pressure Ratios	C-18
1150	1 and 2	Uncooled	Effect of High Pressure Turbine Firing Temperature	C-19

(W 1570)

Table C-1

250 PSIA CHARGING PRESSURE, COOLED LP TURBINE DISKS , 1 INTERCOOLER,A=A
DANIEL J. MARINACCI, LONG RANGE DEVEOPMENT, A=703 , (215) 595=3283

CASE	1.10	1.20	1.30	1.50	1.60	1.70	1.80	1.90	1.10	1.11	1.12	1.13
T1HPT	1500.	1500.	1300.	570.	2150.	1500.	1500.	1500.	1500.	1500.	1500.	1500.
T2HPT	1110.	1095.	1009.	464.	1277.	1044.	1095.	1044.	1094.	1043.	1057.	1147.
T1LPT	2150.	1500.	1500.	1500.	1277.	1044.	1500.	1044.	1500.	1043.	1500.	1500.
CAV LOS	1.20	1.20	1.20	1.20	1.20	1.20	1.20	1.20	1.20	1.20	1.10	1.50
REG EFT	.70	.70	.70	.70	.70	.70	.80	.80	.90	.90	.80	.80
COMP KW	171.5	171.5	171.5	171.5	171.5	171.5	171.5	171.5	171.5	171.5	171.5	171.5
TURB KW	260.6	231.6	220.6	203.8	239.3	204.2	231.3	204.0	231.0	203.8	245.9	189.6
H.R.	5641.	4928.	4849.	4728.	5193.	4501.	4633.	4296.	4336.	4090.	4551.	5027.

FUEL COST = $2.50 PER MILLION BTU
ELECT COST = 10.00 MILS PER KW=HR   ENERGY COST RATIO = .2500

14.10	12.32	12.12	11.82	12.98	11.25	11.58	10.74	10.84	10.23	11.38	12.57
6.58	7.40	7.77	8.41	7.17	8.40	7.41	8.40	7.42	8.41	6.97	9.04

TOTAL POWER PRODUCTION COSTS IN MILS PER KW=HR

20.68	19.72	19.89	20.23	20.15	19.65	19.00	19.14	18.26	18.64	18.35	21.61

FUEL COST = $2.50 PER MILLION BTU
ELECT COST = 20.00 MILS PER KW=HR   ENERGY COST RATIO = .1250

14.10	12.32	12.12	11.82	12.98	11.25	11.58	10.74	10.84	10.23	11.38	12.57
13.16	14.81	15.54	16.83	14.33	16.79	14.83	16.81	14.85	16.83	13.95	18.08

TOTAL POWER PRODUCTION COSTS IN MILS PER KW=HR

27.26	27.13	27.67	28.65	27.31	28.04	26.41	27.55	25.69	27.05	25.32	30.65

FUEL COST = $4.25 PER MILLION BTU
ELECT COST = 15.10 MILS PER KW=HR   ENERGY COST RATIO = .2815

23.98	20.94	20.61	20.09	22.07	19.13	19.69	18.26	18.43	17.38	19.34	21.36
9.93	11.18	11.74	12.70	10.82	12.68	11.20	12.69	11.21	12.70	10.53	13.65

TOTAL POWER PRODUCTION COSTS IN MILS PER KW=HR

33.91	32.12	32.34	32.80	32.89	31.80	30.89	30.95	29.64	30.09	29.87	35.02

FUEL COST = $5.00 PER MILLION BTU
ELECT COST = 17.06 MILS PER KW=HR   ENERGY COST RATIO = .2931

28.21	24.64	24.24	23.64	25.96	22.50	23.17	21.48	21.68	20.45	22.75	25.13
11.22	12.63	13.26	14.35	12.23	14.32	12.65	14.34	12.67	14.35	11.90	15.43

TOTAL POWER PRODUCTION COSTS IN MILS PER KW=HR

39.43	37.27	37.50	37.99	38.19	36.83	35.82	35.82	34.35	34.80	34.65	40.56

FUEL COST = $5.00 PER MILLION BTU
ELECT COST = 20.00 MILS PER KW=HR   ENERGY COST RATIO = .2500

28.21	24.64	24.24	23.64	25.96	22.50	23.17	21.48	21.68	20.45	22.75	25.13
13.16	14.81	15.54	16.83	14.33	16.79	14.83	16.81	14.85	16.83	13.95	18.08

TOTAL POWER PRODUCTION COSTS IN MILS PER KW=HR

41.37	39.44	39.79	40.47	40.30	39.29	37.99	38.29	36.53	37.28	36.70	43.22

FUEL COST = $7.50 PER MILLION BTU
ELECT COST = 10.00 MILS PER KW=HR   ENERGY COST RATIO = .7500

42.31	36.96	36.37	35.46	38.94	33.75	34.75	32.22	32.52	30.68	34.13	37.70
6.58	7.40	7.77	8.41	7.17	8.40	7.41	8.40	7.42	8.41	6.97	9.04

TOTAL POWER PRODUCTION COSTS IN MILS PER KW=HR

48.89	44.36	44.14	43.87	46.11	42.15	42.16	40.62	39.95	39.09	41.10	46.74

FUEL COST = $7.50 PER MILLION BTU
ELECT COST = 20.00 MILS PER KW=HR   ENERGY COST RATIO = .3750

42.31	36.96	36.37	35.46	38.94	33.75	34.75	32.22	32.52	30.68	34.13	37.70
13.16	14.81	15.54	16.83	14.33	16.79	14.83	16.81	14.85	16.83	13.95	18.08

TOTAL POWER PRODUCTION COSTS IN MILS PER KW=HR

55.47	51.76	51.91	52.29	53.28	50.55	49.58	49.03	47.37	47.50	48.07	55.78

FUEL COST = $7.75 PER MILLION BTU
ELECT COST = 60.00 MILS PER KW=HR   ENERGY COST RATIO = .1292

43.72	38.19	37.58	36.64	40.24	34.88	35.91	33.29	33.60	31.70	35.27	38.96
39.47	44.42	46.63	50.48	43.00	50.37	44.48	50.43	44.55	50.48	41.84	54.25

TOTAL POWER PRODUCTION COSTS IN MILS PER KW=HR

83.20	82.61	84.21	87.12	83.24	85.25	80.39	83.72	78.15	82.18	77.10	93.21

Table C-2

250 PSIA CHARGING PRESSURE, COOLED LP TURBINE DISKS , 2 INTERCOOLERS
DANIEL J. MARINACCI, LONG RANGE DEVEOPMENT, A-703 , (215) 595-3283

CASE	1.10	1.20	1.30	1.50	1.60	1.70	1.80	1.90	1.10	1.11	1.12	1.13
T1HPT	1500.	1500.	1300.	570.	2150.	1500.	1500.	1500.	1500.	1500.	1500.	1500.
T2HPT	1110.	1095.	1009.	464.	1277.	1044.	1095.	1044.	1094.	1043.	1057.	1147.
T1LPT	2150.	1500.	1500.	1500.	1277.	1044.	1500.	1044.	1500.	1043.	1500.	1500.
CAV LOS	1.20	1.20	1.20	1.20	1.20	1.20	1.20	1.20	1.20	1.20	1.10	1.50
REG EFT	.70	.70	.70	.70	.70	.70	.80	.80	.90	.90	.80	.80
COMP KW	166.2	166.2	166.2	166.2	166.2	166.2	166.2	166.2	166.2	166.2	166.2	166.2
TURB KW	260.6	231.6	220.6	203.8	239.3	204.2	231.3	204.0	231.0	203.8	245.9	189.6
H.R.	5641.	4928.	4849.	4728.	5193.	4501.	4633.	4296.	4336.	4090.	4551.	5027.

FUEL COST = $2.50 PER MILLION BTU
ELECT COST = 10.00 MILS PER KW-HR ENERGY COST RATIO = .2500

14.10	12.32	12.12	11.82	12.98	11.25	11.58	10.74	10.84	10.23	11.38	12.57
6.38	7.18	7.53	8.15	6.95	8.14	7.19	8.15	7.20	8.15	6.76	8.76

TOTAL POWER PRODUCTION COSTS IN MILS PER KW-HR

20.48	19.50	19.66	19.98	19.93	19.39	18.77	18.89	18.04	18.38	18.14	21.33

FUEL COST = $2.50 PER MILLION BTU
ELECT COST = 20.00 MILS PER KW-HR ENERGY COST RATIO = .1250

14.10	12.32	12.12	11.82	12.98	11.25	11.58	10.74	10.84	10.23	11.38	12.57
12.75	14.35	15.07	16.31	13.89	16.28	14.37	16.29	14.39	16.31	13.52	17.53

TOTAL POWER PRODUCTION COSTS IN MILS PER KW-HR

26.86	26.67	27.19	28.13	26.87	27.53	25.96	27.03	25.23	26.53	24.89	30.10

FUEL COST = $4.25 PER MILLION BTU
ELECT COST = 15.10 MILS PER KW-HR ENERGY COST RATIO = .2815

23.98	20.94	20.61	20.09	22.07	19.13	19.69	18.26	18.43	17.38	19.34	21.36
9.63	10.84	11.38	12.31	10.49	12.29	10.85	12.30	10.87	12.31	10.21	13.23

TOTAL POWER PRODUCTION COSTS IN MILS PER KW-HR

33.61	31.78	31.98	32.41	32.56	31.42	30.54	30.56	29.30	29.70	29.55	34.60

FUEL COST = $5.00 PER MILLION BTU
ELECT COST = 17.06 MILS PER KW-HR ENERGY COST RATIO = .2931

28.21	24.64	24.24	23.64	25.96	22.50	23.17	21.48	21.68	20.45	22.75	25.13
10.88	12.24	12.85	13.91	11.85	13.88	12.26	13.90	12.28	13.91	11.53	14.95

TOTAL POWER PRODUCTION COSTS IN MILS PER KW-HR

39.09	36.88	37.10	37.55	37.81	36.39	35.43	35.38	33.96	34.36	34.28	40.09

FUEL COST = $5.00 PER MILLION BTU
ELECT COST = 20.00 MILS PER KW-HR ENERGY COST RATIO = .2500

28.21	24.64	24.24	23.64	25.96	22.50	23.17	21.48	21.68	20.45	22.75	25.13
12.75	14.35	15.07	16.31	13.89	16.28	14.37	16.29	14.39	16.31	13.52	17.53

TOTAL POWER PRODUCTION COSTS IN MILS PER KW-HR

40.96	38.99	39.31	39.95	39.86	38.78	37.54	37.77	36.07	36.76	36.27	42.66

FUEL COST = $7.50 PER MILLION BTU
ELECT COST = 10.00 MILS PER KW-HR ENERGY COST RATIO = .7500

42.31	36.96	36.37	35.46	38.94	33.75	34.75	32.22	32.52	30.68	34.13	37.70
6.38	7.18	7.53	8.15	6.95	8.14	7.19	8.15	7.20	8.15	6.76	8.76

TOTAL POWER PRODUCTION COSTS IN MILS PER KW-HR

48.69	44.13	43.90	43.62	45.89	41.89	41.94	40.37	39.72	38.83	40.89	46.46

FUEL COST = $7.50 PER MILLION BTU
ELECT COST = 20.00 MILS PER KW-HR ENERGY COST RATIO = .3750

42.31	36.96	36.37	35.46	38.94	33.75	34.75	32.22	32.52	30.68	34.13	37.70
12.75	14.35	15.07	16.31	13.89	16.28	14.37	16.29	14.39	16.31	13.52	17.53

TOTAL POWER PRODUCTION COSTS IN MILS PER KW-HR

55.07	51.31	51.43	51.77	52.84	50.03	49.12	48.51	46.91	46.99	47.65	55.23

FUEL COST = $7.75 PER MILLION BTU
ELECT COST = 60.00 MILS PER KW-HR ENERGY COST RATIO = .1292

43.72	38.19	37.58	36.64	40.24	34.88	35.91	33.29	33.60	31.70	35.27	38.96
38.26	43.05	45.20	48.93	41.68	48.83	43.12	48.88	43.18	48.93	40.55	52.59

TOTAL POWER PRODUCTION COSTS IN MILS PER KW-HR

81.98	81.24	82.78	85.57	81.92	83.71	79.03	82.17	76.78	80.63	75.82	91.54

Table C-3

700 PSIA CHARGING PRESSURE, COOLED LP TURBINE DISKS , 1 INTERCOOLER
DANIEL J, MARINACCI, LONG RANGE DEVEOPMENT, A-703 , (215) 595-3283

CASE	2,10	2,11	2,12	2,13	2,14	2,15	2,16	2,20	2,30	2,40	2,50	2,60	2,70
T1HPT	1500,	1500,	2150,	622,	863,	2150,	2150,	1500,	1500,	1500,	1400,	1500,	1300,
T2HPT	1109,	1105,	1096,	252,	414,	1095,	1072,	874,	1052,	1264,	802,	816,	832,
T1LPT	1500,	2150,	1096,	1500,	2150,	2150,	1500,	874,	1500,	1500,	802,	1300,	1500,
CAV LOS	1,20	1,20	1,20	1,20	1,20	1,20	1,20	1,20	1,10	1,50	1,20	1,20	1,20
REG EFT	,70	,70	,70	,70	,70	,70	,70	,70	,80	,80	,70	,70	,70
COMP KW	249,8	249,8	249,8	249,8	249,8	249,8	249,8	249,8	249,8	249,8	249,8	249,8	249,8
TURB KW	308,6	361,9	311,5	243,1	298,4	372,0	344,9	264,6	323,2	267,1	249,7	298,4	300,8
H,R,	4194,	4580,	4366,	4672,	5185,	5265,	4738,	3952,	4040,	4128,	3901,	4367,	4332,

FUEL COST = $2,50 PER MILLION BTU
ELECT COST = 10,00 MILS PER KW-HR ENERGY COST RATIO = ,2500

10,48	11,45	10,92	11,68	12,96	13,16	11,84	9,88	10,10	10,32	9,75	10,92	10,83
8,09	6,90	8,02	10,27	8,37	6,71	7,24	9,44	7,73	9,35	10,00	8,37	8,30

TOTAL POWER PRODUCTION COSTS IN MILS PER KW-HR

18,58	18,35	18,93	21,95	21,33	19,88	19,09	19,32	17,83	19,67	19,76	19,29	19,13

FUEL COST = $2,50 PER MILLION BTU
ELECT COST = 20,00 MILS PER KW-HR ENERGY COST RATIO = ,1250

10,48	11,45	10,92	11,68	12,96	13,16	11,84	9,88	10,10	10,32	9,75	10,92	10,83
16,18	13,80	16,04	20,55	16,74	13,43	14,48	18,88	15,46	18,70	20,00	16,74	16,61

TOTAL POWER PRODUCTION COSTS IN MILS PER KW-HR

26,67	25,25	26,95	32,23	29,70	26,59	26,33	28,76	25,55	29,02	29,76	27,66	27,44

FUEL COST = $4,25 PER MILLION BTU
ELECT COST = 15,10 MILS PER KW-HR ENERGY COST RATIO = ,2815

17,82	19,46	18,56	19,86	22,04	22,37	20,14	16,80	17,17	17,54	16,58	18,56	18,41
12,22	10,42	12,11	15,51	12,64	10,14	10,93	14,26	11,67	14,12	15,10	12,64	12,54

TOTAL POWER PRODUCTION COSTS IN MILS PER KW-HR

30,04	29,89	30,67	35,37	34,67	32,51	31,07	31,05	28,84	31,67	31,68	31,20	30,95

FUEL COST = $5,00 PER MILLION BTU
ELECT COST = 17,06 MILS PER KW-HR ENERGY COST RATIO = ,2931

20,97	22,90	21,83	23,36	25,92	26,32	23,69	19,76	20,20	20,64	19,51	21,83	21,66
13,81	11,77	13,68	17,53	14,28	11,46	12,35	16,11	13,18	15,95	17,06	14,28	14,17

TOTAL POWER PRODUCTION COSTS IN MILS PER KW-HR

34,77	34,67	35,51	40,89	40,20	37,78	36,04	35,87	33,38	36,60	36,57	36,11	35,83

FUEL COST = $5,00 PER MILLION BTU
ELECT COST = 20,00 MILS PER KW-HR ENERGY COST RATIO = ,2500

20,97	22,90	21,83	23,36	25,92	26,32	23,69	19,76	20,20	20,64	19,51	21,83	21,66
16,18	13,80	16,04	20,55	16,74	13,43	14,48	18,88	15,46	18,70	20,00	16,74	16,61

TOTAL POWER PRODUCTION COSTS IN MILS PER KW-HR

37,15	36,70	37,87	43,91	42,66	39,75	38,17	38,64	35,65	39,34	39,51	38,57	38,27

FUEL COST = $7,50 PER MILLION BTU
ELECT COST = 10,00 MILS PER KW-HR ENERGY COST RATIO = ,7500

31,45	34,35	32,75	35,04	38,89	39,48	35,53	29,64	30,30	30,96	29,26	32,75	32,49
8,09	6,90	8,02	10,27	8,37	6,71	7,24	9,44	7,73	9,35	10,00	8,37	8,30

TOTAL POWER PRODUCTION COSTS IN MILS PER KW-HR

39,54	41,25	40,77	45,31	47,25	46,20	42,77	39,08	38,03	40,31	39,26	41,12	40,80

FUEL COST = $7,50 PER MILLION BTU
ELECT COST = 20,00 MILS PER KW-HR ENERGY COST RATIO = ,3750

31,45	34,35	32,75	35,04	38,89	39,48	35,53	29,64	30,30	30,96	29,26	32,75	32,49
16,18	13,80	16,04	20,55	16,74	13,43	14,48	18,88	15,46	18,70	20,00	16,74	16,61

TOTAL POWER PRODUCTION COSTS IN MILS PER KW-HR

47,64	48,15	48,79	55,58	55,62	52,91	50,02	48,52	45,75	49,66	49,26	49,49	49,10

FUEL COST = $7,75 PER MILLION BTU
ELECT COST = 60,00 MILS PER KW-HR ENERGY COST RATIO = ,1292

32,50	35,49	33,84	36,21	40,18	40,80	36,72	30,63	31,31	31,99	30,24	33,84	33,58
48,55	41,41	48,11	61,64	50,21	40,29	43,44	56,64	46,37	56,11	60,01	50,22	49,82

TOTAL POWER PRODUCTION COSTS IN MILS PER KW-HR

81,05	76,90	81,95	97,85	90,40	81,09	80,16	87,27	77,67	88,10	90,25	84,07	83,39

Table C-4

700 PSIA , 1500-1500 F , COOLED LPT DISKS, 1 INTERCOOLER, (A-A)
DANIEL J, MARINACCI, LONG RANGE DEVEOPMENT, A-703 , (215) 595-3283

CASE	5,00	6,00	7,00	8,00	9,00	10,00	11,00
T1HPT	1500,	1500,	1500,	1500,	1500,	1500,	1500,
T2HPT	854,	910,	960,	1002,	1044,	1078,	1109,
T1LPT	1500,	1500,	1500,	1500,	1500,	1500,	1500,
CAV LOS	1,20	1,20	1,20	1,20	1,20	1,20	1,20
REG EFT	,70	,70	,70	,70	,70	,70	,70
COMP KW	249,8	249,8	249,8	249,8	249,8	249,8	249,8
TURB KW	315,0	315,6	315,0	314,4	312,2	310,6	308,6
H,R,	4468,	4394,	4337,	4289,	4252,	4220,	4194,

FUEL COST = $2,50 PER MILLION BTU
ELECT COST = 10,00 MILS PER KW-HR ENERGY COST RATIO = ,2500

11,17	10,98	10,84	10,72	10,63	10,55	10,48
7,93	7,91	7,93	7,94	8,00	8,04	8,09

TOTAL POWER PRODUCTION COSTS IN MILS PER KW-HR

19,10	18,90	18,77	18,67	18,63	18,59	18,58

FUEL COST = $2,50 PER MILLION BTU
ELECT COST = 20,00 MILS PER KW-HR ENERGY COST RATIO = ,1250

11,17	10,98	10,84	10,72	10,63	10,55	10,48
15,86	15,83	15,86	15,89	16,00	16,08	16,18

TOTAL POWER PRODUCTION COSTS IN MILS PER KW-HR

27,03	26,81	26,70	26,61	26,63	26,63	26,67

FUEL COST = $4,25 PER MILLION BTU
ELECT COST = 15,10 MILS PER KW-HR ENERGY COST RATIO = ,2815

18,99	18,67	18,43	18,23	18,07	17,94	17,82
11,97	11,95	11,97	12,00	12,08	12,14	12,22

TOTAL POWER PRODUCTION COSTS IN MILS PER KW-HR

30,96	30,62	30,40	30,23	30,15	30,08	30,04

FUEL COST = $5,00 PER MILLION BTU
ELECT COST = 17,06 MILS PER KW-HR ENERGY COST RATIO = ,2931

22,34	21,97	21,68	21,45	21,26	21,10	20,97
13,53	13,50	13,53	13,55	13,65	13,72	13,81

TOTAL POWER PRODUCTION COSTS IN MILS PER KW-HR

35,87	35,47	35,21	35,00	34,91	34,82	34,77

FUEL COST = $5,00 PER MILLION BTU
ELECT COST = 20,00 MILS PER KW-HR ENERGY COST RATIO = ,2500

22,34	21,97	21,68	21,45	21,26	21,10	20,97
15,86	15,83	15,86	15,89	16,00	16,08	16,18

TOTAL POWER PRODUCTION COSTS IN MILS PER KW-HR

38,20	37,80	37,54	37,33	37,26	37,18	37,15

FUEL COST = $7,50 PER MILLION BTU
ELECT COST = 10,00 MILS PER KW-HR ENERGY COST RATIO = ,7500

33,51	32,95	32,52	32,17	31,89	31,65	31,45
7,93	7,91	7,93	7,94	8,00	8,04	8,09

TOTAL POWER PRODUCTION COSTS IN MILS PER KW-HR

41,44	40,87	40,45	40,11	39,89	39,69	39,54

FUEL COST = $7,50 PER MILLION BTU
ELECT COST = 20,00 MILS PER KW-HR ENERGY COST RATIO = ,3750

33,51	32,95	32,52	32,17	31,89	31,65	31,45
15,86	15,83	15,86	15,89	16,00	16,08	16,18

TOTAL POWER PRODUCTION COSTS IN MILS PER KW-HR

49,37	48,78	48,38	48,06	47,89	47,74	47,64

FUEL COST = $7,75 PER MILLION BTU
ELECT COST = 60,00 MILS PER KW-HR ENERGY COST RATIO = ,1292

34,62	34,05	33,61	33,24	32,95	32,71	32,50
47,58	47,48	47,57	47,66	48,00	48,25	48,55

TOTAL POWER PRODUCTION COSTS IN MILS PER KW-HR

82,20	81,53	81,18	80,91	80,95	80,96	81,05

(W 1574)

Table C-5

700 PSIA CHARGING PRESSURE, COOLED LP TURBINE DISKS , 2 INTERCOOLERS
DANIEL J. MARINACCI, LONG RANGE DEVEOPMENT, A-703 , (215) 595-3283

CASE	2.10	2.11	2.12	2.13	2.14	2.15	2.16	2.20	2.30	2.40	2.50	2.60	2.70
T1HPT	1500.	1500.	2150.	622.	863.	2150.	2150.	1500.	1500.	1500.	1400.	1500.	1300.
T2HPT	1109.	1105.	1096.	252.	414.	1095.	1072.	874.	1052.	1264.	802.	816.	832.
T1LPT	1500.	2150.	1096.	1500.	2150.	2150.	1500.	874.	1500.	1500.	802.	1300.	1500.
CAV LOS	1.20	1.20	1.20	1.20	1.20	1.20	1.20	1.20	1.10	1.50	1.20	1.20	1.20
REG EFT	.70	.70	.70	.70	.70	.70	.70	.70	.80	.80	.70	.70	.70
COMP KW	233.4	233.4	233.4	233.4	233.4	233.4	233.4	233.4	233.4	233.4	233.4	233.4	233.4
TURB KW	308.6	361.9	311.5	243.1	298.4	372.0	344.9	264.6	323.2	267.1	249.7	298.4	300.8
H.R.	4194.	4580.	4366.	4672.	5185.	5265.	4738.	3952.	4040.	4128.	3901.	4367.	4332.

FUEL COST = $2.50 PER MILLION BTU
ELECT COST = 10.00 MILS PER KW-HR ENERGY COST RATIO = .2500

10.48	11.45	10.92	11.68	12.96	13.16	11.84	9.88	10.10	10.32	9.75	10.92	10.83
7.56	6.45	7.49	9.60	7.82	6.27	6.77	8.82	7.22	8.74	9.35	7.82	7.76

TOTAL POWER PRODUCTION COSTS IN MILS PER KW-HR

18.05	17.90	18.41	21.28	20.78	19.44	18.61	18.70	17.32	19.06	19.10	18.74	18.59

FUEL COST = $2.50 PER MILLION BTU
ELECT COST = 20.00 MILS PER KW-HR ENERGY COST RATIO = .1250

10.48	11.45	10.92	11.68	12.96	13.16	11.84	9.88	10.10	10.32	9.75	10.92	10.83
15.12	12.90	14.98	19.20	15.64	12.55	13.53	17.64	14.44	17.48	18.69	15.64	15.52

TOTAL POWER PRODUCTION COSTS IN MILS PER KW-HR

25.61	24.35	25.90	30.88	28.60	25.71	25.38	27.52	24.54	27.80	28.44	26.56	26.35

FUEL COST = $4.25 PER MILLION BTU
ELECT COST = 15.10 MILS PER KW-HR ENERGY COST RATIO = .2815

17.82	19.46	18.56	19.86	22.04	22.37	20.14	16.80	17.17	17.54	16.58	18.56	18.41
11.42	9.74	11.31	14.49	11.81	9.47	10.22	13.32	10.90	13.19	14.11	11.81	11.71

TOTAL POWER PRODUCTION COSTS IN MILS PER KW-HR

29.24	29.20	29.87	34.35	33.84	31.85	30.35	30.12	28.07	30.74	30.69	30.37	30.13

FUEL COST = $5.00 PER MILLION BTU
ELECT COST = 17.06 MILS PER KW-HR ENERGY COST RATIO = .2931

20.97	22.90	21.83	23.36	25.92	26.32	23.69	19.76	20.20	20.64	19.51	21.83	21.66
12.90	11.00	12.78	16.38	13.34	10.70	11.54	15.05	12.32	14.91	15.94	13.34	13.24

TOTAL POWER PRODUCTION COSTS IN MILS PER KW-HR

33.87	33.90	34.61	39.74	39.26	37.03	35.23	34.81	32.52	35.55	35.45	35.18	34.90

FUEL COST = $5.00 PER MILLION BTU
ELECT COST = 20.00 MILS PER KW-HR ENERGY COST RATIO = .2500

20.97	22.90	21.83	23.36	25.92	26.32	23.69	19.76	20.20	20.64	19.51	21.83	21.66
15.12	12.90	14.98	19.20	15.64	12.55	13.53	17.64	14.44	17.48	18.69	15.64	15.52

TOTAL POWER PRODUCTION COSTS IN MILS PER KW-HR

36.09	35.80	36.82	42.56	41.56	38.87	37.22	37.40	34.64	38.12	38.20	37.48	37.18

FUEL COST = $7.50 PER MILLION BTU
ELECT COST = 10.00 MILS PER KW-HR ENERGY COST RATIO = .7500

31.45	34.35	32.75	35.04	38.89	39.48	35.53	29.64	30.30	30.96	29.26	32.75	32.49
7.56	6.45	7.49	9.60	7.82	6.27	6.77	8.82	7.22	8.74	9.35	7.82	7.76

TOTAL POWER PRODUCTION COSTS IN MILS PER KW-HR

39.01	40.80	40.24	44.64	46.71	45.76	42.30	38.46	37.52	39.70	38.61	40.57	40.25

FUEL COST = $7.50 PER MILLION BTU
ELECT COST = 20.00 MILS PER KW-HR ENERGY COST RATIO = .3750

31.45	34.35	32.75	35.04	38.89	39.48	35.53	29.64	30.30	30.96	29.26	32.75	32.49
15.12	12.90	14.98	19.20	15.64	12.55	13.53	17.64	14.44	17.48	18.69	15.64	15.52

TOTAL POWER PRODUCTION COSTS IN MILS PER KW-HR

46.57	47.25	47.73	54.24	54.52	52.03	49.07	47.28	44.74	48.44	47.95	48.39	48.01

FUEL COST = $7.75 PER MILLION BTU
ELECT COST = 60.00 MILS PER KW-HR ENERGY COST RATIO = .1292

32.50	35.49	33.84	36.21	40.18	40.80	36.72	30.63	31.31	31.99	30.24	33.84	33.58
45.37	38.69	44.95	57.59	46.92	37.64	40.59	52.93	43.32	52.43	56.07	46.93	46.55

TOTAL POWER PRODUCTION COSTS IN MILS PER KW-HR

77.87	74.19	78.79	93.80	87.10	78.44	77.31	83.55	74.63	84.42	86.31	80.77	80.12

(W 1575)

Table C-6

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700 PSIA , 1500-1500 F , COOLED LPT DISKS,  2 INTERCOOLERS, (A-A-A)
DANIEL J. MARINACCI, LONG RANGE DEVEOPMENT, A-703  , (215) 595-3283

CASE       5.00   6.00   7.00   8.00   9.00  10.00  11.00
T1HPT     1500.  1500.  1500.  1500.  1500.  1500.  1500.
T2HPT      854.   910.   960.  1002.  1044.  1078.  1109.
T1LPT     1500.  1500.  1500.  1500.  1500.  1500.  1500.
CAV LOS    1.20   1.20   1.20   1.20   1.20   1.20   1.20
REG EFT     .70    .70    .70    .70    .70    .70    .70
COMP KW   233.4  233.4  233.4  233.4  233.4  233.4  233.4
TURB KW   315.0  315.6  315.0  314.4  312.2  310.6  308.6
H.R.      4468.  4394.  4337.  4289.  4252.  4220.  4194.


          FUEL  COST = $2.50 PER MILLION BTU
          ELECT COST = 10.00 MILS PER KW-HR    ENERGY COST RATIO = .2500
          11.17  10.98  10.84  10.72  10.63  10.55  10.48
           7.41   7.39   7.41   7.42   7.47   7.51   7.56
          TOTAL POWER PRODUCTION COSTS IN MILS PER KW-HR
          18.58  18.38  18.25  18.15  18.10  18.06  18.05


          FUEL  COST = $2.50 PER MILLION BTU
          ELECT COST = 20.00 MILS PER KW-HR    ENERGY COST RATIO = .1250
          11.17  10.98  10.84  10.72  10.63  10.55  10.48
          14.82  14.79  14.82  14.85  14.95  15.03  15.12
          TOTAL POWER PRODUCTION COSTS IN MILS PER KW-HR
          25.99  25.77  25.66  25.57  25.58  25.58  25.61


          FUEL  COST = $4.25 PER MILLION BTU
          ELECT COST = 15.10 MILS PER KW-HR    ENERGY COST RATIO = .2815
          18.99  18.67  18.43  18.23  18.07  17.94  17.82
          11.19  11.16  11.19  11.21  11.29  11.35  11.42
          TOTAL POWER PRODUCTION COSTS IN MILS PER KW-HR
          30.18  29.84  29.62  29.44  29.36  29.28  29.24


          FUEL  COST = $5.00 PER MILLION BTU
          ELECT COST = 17.06 MILS PER KW-HR    ENERGY COST RATIO = .2931
          22.34  21.97  21.68  21.45  21.26  21.10  20.97
          12.64  12.61  12.64  12.66  12.75  12.82  12.90
          TOTAL POWER PRODUCTION COSTS IN MILS PER KW-HR
          34.98  34.58  34.32  34.11  34.01  33.92  33.87


          FUEL  COST = $5.00 PER MILLION BTU
          ELECT COST = 20.00 MILS PER KW-HR    ENERGY COST RATIO = .2500
          22.34  21.97  21.68  21.45  21.26  21.10  20.97
          14.82  14.79  14.82  14.85  14.95  15.03  15.12
          TOTAL POWER PRODUCTION COSTS IN MILS PER KW-HR
          37.16  36.76  36.50  36.29  36.21  36.13  36.09


          FUEL  COST = $7.50 PER MILLION BTU
          ELECT COST = 10.00 MILS PER KW-HR    ENERGY COST RATIO = .7500
          33.51  32.95  32.52  32.17  31.89  31.65  31.45
           7.41   7.39   7.41   7.42   7.47   7.51   7.56
          TOTAL POWER PRODUCTION COSTS IN MILS PER KW-HR
          40.92  40.35  39.93  39.59  39.37  39.17  39.01


          FUEL  COST = $7.50 PER MILLION BTU
          ELECT COST = 20.00 MILS PER KW-HR    ENERGY COST RATIO = .3750
          33.51  32.95  32.52  32.17  31.89  31.65  31.45
          14.82  14.79  14.82  14.85  14.95  15.03  15.12
          TOTAL POWER PRODUCTION COSTS IN MILS PER KW-HR
          48.33  47.74  47.34  47.02  46.84  46.68  46.57


          FUEL  COST = $7.75 PER MILLION BTU
          ELECT COST = 60.00 MILS PER KW-HR    ENERGY COST RATIO = .1292
          34.62  34.05  33.61  33.24  32.95  32.71  32.50
          44.46  44.36  44.45  44.54  44.85  45.08  45.37
          TOTAL POWER PRODUCTION COSTS IN MILS PER KW-HR
          79.08  78.42  78.06  77.78  77.80  77.79  77.87
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(W 1576)

Table C-7

1150 CHARGING PRESSURE, COOLED LPT DISKS, 1 INTERCOOLER, (A=A)
DANIEL J, MARINACCI, LONG RANGE DEVEOPMENT, A=703 , (215) 595=3283

CASE	3,01	3,02	3,10	3,20	3,30	3,40	3,50	3,60	3,70	3,80	3,90	3,10	3,12	3,13
T1HPT	2150,	2150,	1500,	1500,	1300,	675,	473,	2150,	1500,	1500,	1500,	1500,	1500,	1500,
T2HPT	1102,	1105,	1105,	830,	759,	347,	202,	1016,	802,	829,	802,	828,	883,	981,
T1LPT	2150,	1500,	2150,	1500,	1500,	2150,	1500,	1016,	802,	1500,	802,	1500,	1500,	1500,
,AV LOS	1,20	1,20	1,20	1,20	1,20	1,20	1,20	1,20	1,20	1,20	1,20	1,20	1,10	1,50
REG EFT	,70	,70	,70	,70	,70	,70	,70	,70	,70	,80	,80	,90	,80	,80
COMP KW	291,7	291,7	291,7	291,7	291,7	291,7	291,7	291,7	291,7	291,7	291,7	291,7	291,7	291,7
TURB KW	420,6	381,4	401,9	353,2	335,9	350,0	278,4	341,5	288,8	352,8	288,7	352,3	364,2	317,7
H,R,	4822,	4408,	4310,	4236,	4179,	4566,	4221,	4130,	3799,	4096,	3750,	3954,	3999,	4133,

FUEL COST = $2,50 PER MILLION BTU
ELECT COST = 10,00 MILS PER KW=HR ENERGY COST RATIO = ,2500

12,06	11,02	10,77	10,59	10,45	11,41	10,55	10,33	9,50	10,24	9,37	9,89	10,00	10,33
6,94	7,65	7,26	8,26	8,68	8,33	10,48	8,54	10,10	8,27	10,11	8,28	8,01	9,18

TOTAL POWER PRODUCTION COSTS IN MILS PER KW=HR

18,99	18,67	18,03	18,85	19,13	19,75	21,03	18,87	19,60	18,51	19,48	18,16	18,01	19,52

FUEL COST = $2,50 PER MILLION BTU
ELECT COST = 20,00 MILS PER KW=HR ENERGY COST RATIO = ,1250

12,06	11,02	10,77	10,59	10,45	11,41	10,55	10,33	9,50	10,24	9,37	9,89	10,00	10,33
13,87	15,30	14,52	16,52	17,37	16,67	20,95	17,08	20,20	16,54	20,21	16,56	16,02	18,37

TOTAL POWER PRODUCTION COSTS IN MILS PER KW=HR

25,93	26,32	25,29	27,11	27,82	28,08	31,51	27,41	29,70	26,78	29,58	26,44	26,02	28,70

FUEL COST = $4,25 PER MILLION BTU
ELECT COST = 15,10 MILS PER KW=HR ENERGY COST RATIO = ,2815

20,49	18,73	18,32	18,00	17,76	19,40	17,94	17,55	16,15	17,41	15,94	16,81	16,99	17,56
10,47	11,55	10,96	12,47	13,11	12,58	15,82	12,90	15,25	12,49	15,26	12,50	12,10	13,87

TOTAL POWER PRODUCTION COSTS IN MILS PER KW=HR

30,97	30,28	29,28	30,47	30,87	31,99	33,76	30,45	31,40	29,89	31,20	29,31	29,09	31,43

FUEL COST = $5,00 PER MILLION BTU
ELECT COST = 17,06 MILS PER KW=HR ENERGY COST RATIO = ,2931

24,11	22,04	21,55	21,18	20,90	22,83	21,11	20,65	19,00	20,48	18,75	19,77	19,99	20,66
11,83	13,05	12,38	14,09	14,82	14,22	17,88	14,57	17,23	14,11	17,24	14,12	13,67	15,67

TOTAL POWER PRODUCTION COSTS IN MILS PER KW=HR

35,94	35,09	33,93	35,27	35,71	37,05	38,98	35,22	36,23	34,59	35,99	33,90	33,66	36,33

FUEL COST = $5,00 PER MILLION BTU
ELECT COST = 20,00 MILS PER KW=HR ENERGY COST RATIO = ,2500

24,11	22,04	21,55	21,18	20,90	22,83	21,11	20,65	19,00	20,48	18,75	19,77	19,99	20,66
13,87	15,30	14,52	16,52	17,37	16,67	20,95	17,08	20,20	16,54	20,21	16,56	16,02	18,37

TOTAL POWER PRODUCTION COSTS IN MILS PER KW=HR

37,98	37,34	36,06	37,70	38,26	39,50	42,06	37,73	39,20	37,02	38,96	36,33	36,01	39,03

FUEL COST = $7,50 PER MILLION BTU
ELECT COST = 10,00 MILS PER KW=HR ENERGY COST RATIO = ,7500

36,17	33,06	32,32	31,77	31,34	34,24	31,66	30,98	28,49	30,72	28,12	29,66	29,99	31,00
6,94	7,65	7,26	8,26	8,68	8,33	10,48	8,54	10,10	8,27	10,11	8,28	8,01	9,18

TOTAL POWER PRODUCTION COSTS IN MILS PER KW=HR

43,10	40,71	39,58	40,03	40,03	42,58	42,14	39,52	38,59	38,99	38,23	37,94	38,00	40,18

FUEL COST = $7,50 PER MILLION BTU
ELECT COST = 20,00 MILS PER KW=HR ENERGY COST RATIO = ,3750

36,17	33,06	32,32	31,77	31,34	34,24	31,66	30,98	28,49	30,72	28,12	29,66	29,99	31,00
13,87	15,30	14,52	16,52	17,37	16,67	20,95	17,08	20,20	16,54	20,21	16,56	16,02	18,37

TOTAL POWER PRODUCTION COSTS IN MILS PER KW=HR

50,04	48,36	46,84	48,29	48,71	50,91	52,61	48,06	48,69	47,26	48,33	46,21	46,01	49,36

FUEL COST = $7,75 PER MILLION BTU
ELECT COST = 60,00 MILS PER KW=HR ENERGY COST RATIO = ,1292

37,37	34,16	33,40	32,83	32,39	35,39	32,71	32,01	29,44	31,74	29,06	30,65	30,99	32,03
41,62	45,89	43,55	49,56	52,11	50,00	62,86	51,25	60,61	49,62	60,63	49,67	48,06	55,10

TOTAL POWER PRODUCTION COSTS IN MILS PER KW=HR

78,99	80,05	76,95	82,39	84,49	85,39	95,58	83,26	90,05	81,36	89,69	80,32	79,05	87,13

Table C-8

1150 PSIA, 1500-1500 F, COOLED LPT DISKS, 1 INTERCOOLER, (A-A)
DANIEL J. MARINACCI, LONG RANGE DEVEOPMENT, A-703 , (215) 595-3283

CASE	4.00	5.00	6.00	7.00	7.51	8.00	9.00	10.00	11.00	12.00	14.00	15.00	16.00	17.00
T1HPT	1500.	1500.	1500.	1500.	1500.	1500.	1500.	1500.	1500.	1500.	1500.	1500.	1500.	1500.
'2HPT	653.	712.	763.	808.	830.	848.	884.	915.	947.	974.	1027.	1049.	1070.	1089.
'1LPT	1500.	1500.	1500.	1500.	1500.	1500.	1500.	1500.	1500.	1500.	1500.	1500.	1500.	1500.
CAV LOS	1.20	1.20	1.20	1.20	1.20	1.20	1.20	1.20	1.20	1.20	1.20	1.20	1.20	1.20
REG EFT	.70	.70	.70	.70	.70	.70	.70	.70	.70	.70	.70	.70	.70	.70
COMP KW	291.7	291.7	291.7	291.7	291.7	291.7	291.7	291.7	291.7	291.7	291.7	291.7	291.7	291.7
TURB KW	345.3	349.9	352.0	353.2	353.2	353.2	352.6	352.0	350.1	348.9	344.9	343.4	341.8	340.1
H.R.	4481.	4385.	4315.	4260.	4236.	4216.	4180.	4149.	4125.	4102.	4066.	4050.	4035.	4022.

FUEL COST = $2.50 PER MILLION BTU
ELECT COST = 10.00 MILS PER KW-HR ENERGY COST RATIO = .2500

11.20	10.96	10.79	10.65	10.59	10.54	10.45	10.37	10.31	10.26	10.17	10.13	10.09	10.06
8.45	8.34	8.29	8.26	8.26	8.26	8.27	8.29	8.33	8.36	8.46	8.50	8.54	8.58

TOTAL POWER PRODUCTION COSTS IN MILS PER KW-HR

19.65	19.30	19.08	18.91	18.85	18.80	18.72	18.66	18.64	18.62	18.62	18.62	18.62	18.63

FUEL COST = $2.50 PER MILLION BTU
ELECT COST = 20.00 MILS PER KW-HR ENERGY COST RATIO = .1250

11.20	10.96	10.79	10.65	10.59	10.54	10.45	10.37	10.31	10.26	10.17	10.13	10.09	10.06
16.90	16.67	16.57	16.52	16.52	16.52	16.54	16.57	16.66	16.72	16.92	16.99	17.07	17.15

TOTAL POWER PRODUCTION COSTS IN MILS PER KW-HR

28.10	27.64	27.36	27.17	27.11	27.06	26.99	26.95	26.97	26.98	27.08	27.12	27.16	27.21

FUEL COST = $4.25 PER MILLION BTU
ELECT COST = 15.10 MILS PER KW-HR ENERGY COST RATIO = .2815

19.05	18.64	18.34	18.11	18.00	17.92	17.76	17.63	17.53	17.44	17.28	17.21	17.15	17.09
12.76	12.59	12.51	12.47	12.47	12.47	12.49	12.51	12.58	12.62	12.77	12.83	12.89	12.95

TOTAL POWER PRODUCTION COSTS IN MILS PER KW-HR

31.80	31.22	30.85	30.58	30.47	30.39	30.25	30.15	30.11	30.06	30.05	30.04	30.04	30.04

FUEL COST = $5.00 PER MILLION BTU
ELECT COST = 17.06 MILS PER KW-HR ENERGY COST RATIO = .2931

22.41	21.93	21.58	21.30	21.18	21.08	20.90	20.75	20.62	20.51	20.33	20.25	20.18	20.11
14.41	14.22	14.14	14.09	14.09	14.09	14.11	14.14	14.21	14.26	14.43	14.49	14.56	14.63

TOTAL POWER PRODUCTION COSTS IN MILS PER KW-HR

36.82	36.15	35.72	35.39	35.27	35.17	35.01	34.89	34.84	34.78	34.76	34.74	34.74	34.74

FUEL COST = $5.00 PER MILLION BTU
ELECT COST = 20.00 MILS PER KW-HR ENERGY COST RATIO = .2500

22.41	21.93	21.58	21.30	21.18	21.08	20.90	20.75	20.62	20.51	20.33	20.25	20.18	20.11
16.90	16.67	16.57	16.52	16.52	16.52	16.54	16.57	16.66	16.72	16.92	16.99	17.07	17.15

TOTAL POWER PRODUCTION COSTS IN MILS PER KW-HR

39.30	38.60	38.15	37.82	37.70	37.60	37.44	37.32	37.29	37.23	37.25	37.24	37.25	37.26

FUEL COST = $7.50 PER MILLION BTU
ELECT COST = 10.00 MILS PER KW-HR ENERGY COST RATIO = .7500

33.61	32.89	32.37	31.95	31.77	31.62	31.35	31.12	30.94	30.77	30.50	30.38	30.27	30.17
8.45	8.34	8.29	8.26	8.26	8.26	8.27	8.29	8.33	8.36	8.46	8.50	8.54	8.58

TOTAL POWER PRODUCTION COSTS IN MILS PER KW-HR

42.06	41.22	40.65	40.21	40.03	39.88	39.62	39.41	39.27	39.13	38.95	38.87	38.80	38.74

FUEL COST = $7.50 PER MILLION BTU
ELECT COST = 20.00 MILS PER KW-HR ENERGY COST RATIO = .3750

33.61	32.89	32.37	31.95	31.77	31.62	31.35	31.12	30.94	30.77	30.50	30.38	30.27	30.17
16.90	16.67	16.57	16.52	16.52	16.52	16.54	16.57	16.66	16.72	16.92	16.99	17.07	17.15

TOTAL POWER PRODUCTION COSTS IN MILS PER KW-HR

50.51	49.56	48.94	48.47	48.29	48.14	47.89	47.69	47.60	47.49	47.41	47.37	47.34	47.32

FUEL COST = $7.75 PER MILLION BTU
ELECT COST = 60.00 MILS PER KW-HR ENERGY COST RATIO = .1292

34.73	33.98	33.44	33.02	32.83	32.68	32.39	32.16	31.97	31.79	31.51	31.39	31.27	31.17
50.69	50.02	49.72	49.56	49.55	49.56	49.63	49.72	49.99	50.16	50.75	50.97	51.21	51.46

TOTAL POWER PRODUCTION COSTS IN MILS PER KW-HR

85.42	84.00	83.17	82.58	82.38	82.23	82.02	81.88	81.95	81.96	82.26	82.36	82.48	82.63

Table C-9

1150 CHARGING PRESSURE, COOLED LPT DISKS, 2 INTERCOOLERS, (A=A=A)
DANIEL J, MARINACCI, LONG RANGE DEVEOPMENT, A=703 , (215) 595=3283

CASE	3,01	3,02	3,10	3,20	3,30	3,40	3,50	3,60	3,70	3,80	3,90	3,10	3,12	3,13
T1HPT	2150,	2150,	1500,	1500,	1300,	675,	473,	2150,	1500,	1500,	1500,	1500,	1500,	1500,
""HPT	1102,	1105,	1105,	830,	759,	347,	202,	1016,	802,	829,	802,	828,	883,	981,
LPT	2150,	1500,	2150,	1500,	1500,	2150,	1500,	1016,	802,	1500,	802,	1500,	1500,	1500,
V LOS	1,20	1,20	1,20	1,20	1,20	1,20	1,20	1,20	1,20	1,20	1,20	1,20	1,10	1,50
REG EFT	,70	,70	,70	,70	,70	,70	,70	,70	,70	,80	,80	,90	,80	,80
COMP KW	266,7	266,7	266,7	266,7	266,7	266,7	266,7	266,7	266,7	266,7	266,7	266,7	266,7	266,7
TURB KW	420,6	381,4	401,9	353,2	335,9	350,0	278,4	341,5	288,8	352,8	288,7	352,3	364,2	317,7
H,R,	4822,	4408,	4310,	4236,	4179,	4566,	4221,	4130,	3799,	4096,	3750,	3954,	3999,	4133,

FUEL COST = $2,50 PER MILLION BTU
ELECT COST = 10,00 MILS PER KW=HR ENERGY COST RATIO = ,2500

12,06	11,02	10,77	10,59	10,45	11,41	10,55	10,33	9,50	10,24	9,37	9,89	10,00	10,33
6,34	6,99	6,64	7,55	7,94	7,62	9,58	7,81	9,24	7,56	9,24	7,57	7,33	8,40

TOTAL POWER PRODUCTION COSTS IN MILS PER KW=HR

18,40	18,01	17,41	18,14	18,39	19,04	20,13	18,14	18,73	17,80	18,61	17,46	17,32	18,73

FUEL COST = $2,50 PER MILLION BTU
ELECT COST = 20,00 MILS PER KW=HR ENERGY COST RATIO = ,1250

12,06	11,02	10,77	10,59	10,45	11,41	10,55	10,33	9,50	10,24	9,37	9,89	10,00	10,33
12,68	13,99	13,27	15,11	15,88	15,24	19,16	15,62	18,47	15,12	18,48	15,14	14,65	16,79

TOTAL POWER PRODUCTION COSTS IN MILS PER KW=HR

24,74	25,01	24,05	25,70	26,33	26,66	29,71	25,95	27,97	25,36	27,86	25,03	24,65	27,13

FUEL COST = $4,25 PER MILLION BTU
ELECT COST = 15,10 MILS PER KW=HR ENERGY COST RATIO = ,2815

20,49	18,73	18,32	18,00	17,76	19,40	17,94	17,55	16,15	17,41	15,94	16,81	16,99	17,56
9,58	10,56	10,02	11,40	11,99	11,51	14,47	11,79	13,95	11,42	13,95	11,43	11,06	12,68

TOTAL POWER PRODUCTION COSTS IN MILS PER KW=HR

30,07	29,29	28,34	29,41	29,75	30,91	32,41	29,35	30,09	28,82	29,89	28,24	28,05	30,24

FUEL COST = $5,00 PER MILLION BTU
ELECT COST = 17,06 MILS PER KW=HR ENERGY COST RATIO = ,2931

24,11	22,04	21,55	21,18	20,90	22,83	21,11	20,65	19,00	20,48	18,75	19,77	19,99	20,66
10,82	11,93	11,32	12,89	13,55	13,00	16,35	13,32	15,76	12,90	15,76	12,92	12,50	14,33

TOTAL POWER PRODUCTION COSTS IN MILS PER KW=HR

34,93	33,97	32,87	34,06	34,44	35,83	37,45	33,98	34,75	33,38	34,51	32,69	32,49	34,99

FUEL COST = $5,00 PER MILLION BTU
ELECT COST = 20,00 MILS PER KW=HR ENERGY COST RATIO = ,2500

24,11	22,04	21,55	21,18	20,90	22,83	21,11	20,65	19,00	20,48	18,75	19,77	19,99	20,66
12,68	13,99	13,27	15,11	15,88	15,24	19,16	15,62	18,47	15,12	18,48	15,14	14,65	16,79

TOTAL POWER PRODUCTION COSTS IN MILS PER KW=HR

36,80	36,03	34,82	36,28	36,78	38,07	40,27	36,27	37,47	35,60	37,23	34,91	34,64	37,46

FUEL COST = $7,50 PER MILLION BTU
ELECT COST = 10,00 MILS PER KW=HR ENERGY COST RATIO = ,7500

36,17	33,06	32,32	31,77	31,34	34,24	31,66	30,98	28,49	30,72	28,12	29,66	29,99	31,00
6,34	6,99	6,64	7,55	7,94	7,62	9,58	7,81	9,24	7,56	9,24	7,57	7,33	8,40

TOTAL POWER PRODUCTION COSTS IN MILS PER KW=HR

42,51	40,05	38,96	39,32	39,28	41,86	41,24	38,79	37,73	38,28	37,36	37,23	37,31	39,39

FUEL COST = $7,50 PER MILLION BTU
ELECT COST = 20,00 MILS PER KW=HR ENERGY COST RATIO = ,3750

36,17	33,06	32,32	31,77	31,34	34,24	31,66	30,98	28,49	30,72	28,12	29,66	29,99	31,00
12,68	13,99	13,27	15,11	15,88	15,24	19,16	15,62	18,47	15,12	18,48	15,14	14,65	16,79

TOTAL POWER PRODUCTION COSTS IN MILS PER KW=HR

48,85	47,05	45,60	46,87	47,22	49,49	50,82	46,60	46,97	45,84	46,60	44,80	44,64	47,79

FUEL COST = $7,75 PER MILLION BTU
ELECT COST = 60,00 MILS PER KW=HR ENERGY COST RATIO = ,1292

37,37	34,16	33,40	32,83	32,39	35,39	32,71	32,01	29,44	31,74	29,06	30,65	30,99	32,03
38,05	41,96	39,82	45,32	47,65	45,73	57,49	46,86	55,42	45,37	55,44	45,42	43,95	50,38

TOTAL POWER PRODUCTION COSTS IN MILS PER KW=HR

75,43	76,12	73,22	78,14	80,03	81,11	90,20	78,87	84,86	77,11	84,50	76,07	74,94	82,41

Table C-10

1150 PSIA, 1500-1500 F, COOLED LPT DISKS, 2 INTERCOOLERS, (A-A-A)
DANIEL J, MARINACCI, LONG RANGE DEVEOPMENT, A-703 , (215) 595-3283

CASE	4,00	5,00	6,00	7,00	7,51	8,00	9,00	10,00	11,00	12,00	14,00	15,00	16,00	17,00
T1HPT	1500,	1500,	1500,	1500,	1500,	1500,	1500,	1500,	1500,	1500,	1500,	1500,	1500,	1500,
T2HPT	653,	712,	763,	808,	830,	848,	884,	915,	947,	974,	1027,	1049,	1070,	1089,
1LPT	1500,	1500,	1500,	1500,	1500,	1500,	1500,	1500,	1500,	1500,	1500,	1500,	1500,	1500,
AV LOS	1,20	1,20	1,20	1,20	1,20	1,20	1,20	1,20	1,20	1,20	1,20	1,20	1,20	1,20
REG EFT	,70	,70	,70	,70	,70	,70	,70	,70	,70	,70	,70	,70	,70	,70
COMP KW	266,7	266,7	266,7	266,7	266,7	266,7	266,7	266,7	266,7	266,7	266,7	266,7	266,7	266,7
TURB KW	345,3	349,9	352,0	353,2	353,2	353,2	352,6	352,0	350,1	348,9	344,9	343,4	341,8	340,1
H,R,	4481,	4385,	4315,	4260,	4236,	4216,	4180,	4149,	4125,	4102,	4066,	4050,	4035,	4022,

FUEL COST = $2,50 PER MILLION BTU
ELECT COST = 10,00 MILS PER KW-HR ENERGY COST RATIO = ,2500

11,20	10,96	10,79	10,65	10,59	10,54	10,45	10,37	10,31	10,26	10,17	10,13	10,09	10,06
7,73	7,62	7,58	7,55	7,55	7,55	7,56	7,58	7,62	7,65	7,73	7,77	7,80	7,84

TOTAL POWER PRODUCTION COSTS IN MILS PER KW-HR

18,93	18,59	18,37	18,20	18,14	18,09	18,01	17,95	17,93	17,90	17,90	17,89	17,89	17,90

FUEL COST = $2,50 PER MILLION BTU
ELECT COST = 20,00 MILS PER KW-HR ENERGY COST RATIO = ,1250

11,20	10,96	10,79	10,65	10,59	10,54	10,45	10,37	10,31	10,26	10,17	10,13	10,09	10,06
15,45	15,25	15,16	15,11	15,10	15,11	15,13	15,16	15,24	15,29	15,47	15,54	15,61	15,69

TOTAL POWER PRODUCTION COSTS IN MILS PER KW-HR

26,65	26,21	25,94	25,76	25,69	25,65	25,58	25,53	25,55	25,55	25,63	25,66	25,70	25,74

FUEL COST = $4,25 PER MILLION BTU
ELECT COST = 15,10 MILS PER KW-HR ENERGY COST RATIO = ,2815

19,05	18,64	18,34	18,11	18,00	17,92	17,76	17,63	17,53	17,44	17,28	17,21	17,15	17,09
11,66	11,51	11,44	11,41	11,40	11,40	11,42	11,44	11,50	11,54	11,68	11,73	11,79	11,84

TOTAL POWER PRODUCTION COSTS IN MILS PER KW-HR

30,71	30,15	29,78	29,51	29,41	29,32	29,19	29,08	29,03	28,98	28,96	28,94	28,94	28,94

FUEL COST = $5,00 PER MILLION BTU
ELECT COST = 17,06 MILS PER KW-HR ENERGY COST RATIO = ,2931

22,41	21,93	21,58	21,30	21,18	21,08	20,90	20,75	20,62	20,51	20,33	20,25	20,18	20,11
13,18	13,01	12,93	12,89	12,88	12,89	12,90	12,93	13,00	13,04	13,19	13,25	13,32	13,38

TOTAL POWER PRODUCTION COSTS IN MILS PER KW-HR

35,59	34,93	34,51	34,19	34,06	33,97	33,80	33,68	33,62	33,56	33,53	33,50	33,49	33,49

FUEL COST = $5,00 PER MILLION BTU
ELECT COST = 20,00 MILS PER KW-HR ENERGY COST RATIO = ,2500

22,41	21,93	21,58	21,30	21,18	21,08	20,90	20,75	20,62	20,51	20,33	20,25	20,18	20,11
15,45	15,25	15,16	15,11	15,10	15,11	15,13	15,16	15,24	15,29	15,47	15,54	15,61	15,69

TOTAL POWER PRODUCTION COSTS IN MILS PER KW-HR

37,86	37,17	36,73	36,41	36,28	36,19	36,03	35,90	35,86	35,80	35,80	35,79	35,79	35,80

FUEL COST = $7,50 PER MILLION BTU
ELECT COST = 10,00 MILS PER KW-HR ENERGY COST RATIO = ,7500

33,61	32,89	32,37	31,95	31,77	31,62	31,35	31,12	30,94	30,77	30,50	30,38	30,27	30,17
7,73	7,62	7,58	7,55	7,55	7,55	7,56	7,58	7,62	7,65	7,73	7,77	7,80	7,84

TOTAL POWER PRODUCTION COSTS IN MILS PER KW-HR

41,33	40,51	39,94	39,50	39,32	39,17	38,91	38,70	38,55	38,41	38,23	38,14	38,07	38,01

FUEL COST = $7,50 PER MILLION BTU
ELECT COST = 20,00 MILS PER KW-HR ENERGY COST RATIO = ,3750

33,61	32,89	32,37	31,95	31,77	31,62	31,35	31,12	30,94	30,77	30,50	30,38	30,27	30,17
15,45	15,25	15,16	15,11	15,10	15,11	15,13	15,16	15,24	15,29	15,47	15,54	15,61	15,69

TOTAL POWER PRODUCTION COSTS IN MILS PER KW-HR

49,06	48,13	47,52	47,06	46,87	46,73	46,48	46,28	46,17	46,06	45,96	45,91	45,88	45,85

FUEL COST = $7,75 PER MILLION BTU
ELECT COST = 60,00 MILS PER KW-HR ENERGY COST RATIO = ,1292

34,73	33,98	33,44	33,02	32,83	32,68	32,39	32,16	31,97	31,79	31,51	31,39	31,27	31,17
46,35	45,74	45,47	45,32	45,31	45,32	45,38	45,47	45,71	45,87	46,40	46,61	46,83	47,06

TOTAL POWER PRODUCTION COSTS IN MILS PER KW-HR

81,08	79,72	78,91	78,34	78,14	77,99	77,78	77,63	77,68	77,66	77,92	78,00	78,10	78,23

Table C-11

250 PSIA CHARGING PRESSURE, UNCOOLED TURBINE DISKS, 1 INTERCOOLER, A=A
DANIEL J, MARINACCI, LONG RANGE DEVEOPMENT, A=703 , (215) 595=3283

CASE	11,10	11,20	11,30	11,40	11,50	11,60	11,70	11,80	11,90	11,10
T1HPT	1500,	1300,	607,	1500,	1500,	1500,	1500,	1500,	1500,	1500,
T2HPT	1096,	1010,	497,	1045,	1096,	1045,	1095,	1044,	1058,	1148,
T1LPT	1500,	1500,	1500,	1045,	1500,	1045,	1500,	1044,	1500,	1500,
CAV LOS	1,20	1,20	1,20	1,20	1,20	1,20	1,20	1,20	1,10	1,50
REG EFT	,70	,70	,70	,70	,80	,80	,90	,90	,80	,80
COMP KW	171,5	171,5	171,5	171,5	171,5	171,5	171,5	171,5	171,5	171,5
TURB KW	248,6	236,5	218,9	219,7	248,3	219,5	247,9	219,2	264,0	203,6
H,R,	4788,	4719,	4610,	4401,	4466,	4177,	4140,	3951,	4397,	4791,

FUEL COST = $2,50 PER MILLION BTU
ELECT COST = 10,00 MILS PER KW=HR ENERGY COST RATIO = ,2500

11,97	11,80	11,53	11,00	11,16	10,44	10,35	9,88	10,99	11,98
6,90	7,25	7,84	7,80	6,91	7,81	6,92	7,82	6,50	8,42

TOTAL POWER PRODUCTION COSTS IN MILS PER KW=HR

18,87	19,05	19,36	18,81	18,07	18,25	17,27	17,70	17,49	20,40

FUEL COST = $2,50 PER MILLION BTU
ELECT COST = 20,00 MILS PER KW=HR ENERGY COST RATIO = ,1250

11,97	11,80	11,53	11,00	11,16	10,44	10,35	9,88	10,99	11,98
13,79	14,50	15,67	15,61	13,81	15,62	13,84	15,64	12,99	16,84

TOTAL POWER PRODUCTION COSTS IN MILS PER KW=HR

25,76	26,30	27,20	26,61	24,98	26,07	24,19	25,52	23,98	28,82

FUEL COST = $4,25 PER MILLION BTU
ELECT COST = 15,10 MILS PER KW=HR ENERGY COST RATIO = ,2815

20,35	20,06	19,59	18,70	18,98	17,75	17,59	16,79	18,69	20,36
10,41	10,95	11,83	11,78	10,43	11,80	10,45	11,81	9,81	12,72

TOTAL POWER PRODUCTION COSTS IN MILS PER KW=HR

30,76	31,00	31,42	30,49	29,41	29,55	28,04	28,60	28,50	33,08

FUEL COST = $5,00 PER MILLION BTU
ELECT COST = 17,06 MILS PER KW=HR ENERGY COST RATIO = ,2931

23,94	23,60	23,05	22,00	22,33	20,88	20,70	19,76	21,99	23,96
11,77	12,37	13,37	13,31	11,78	13,33	11,80	13,34	11,08	14,37

TOTAL POWER PRODUCTION COSTS IN MILS PER KW=HR

35,71	35,97	36,42	35,32	34,11	34,21	32,50	33,10	33,07	38,32

FUEL COST = $5,00 PER MILLION BTU
ELECT COST = 20,00 MILS PER KW=HR ENERGY COST RATIO = ,2500

23,94	23,60	23,05	22,00	22,33	20,88	20,70	19,76	21,99	23,96
13,79	14,50	15,67	15,61	13,81	15,62	13,84	15,64	12,99	16,84

TOTAL POWER PRODUCTION COSTS IN MILS PER KW=HR

37,73	38,10	38,72	37,61	36,14	36,51	34,53	35,40	34,98	40,80

FUEL COST = $7,50 PER MILLION BTU
ELECT COST = 10,00 MILS PER KW=HR ENERGY COST RATIO = ,7500

35,91	35,39	34,58	33,01	33,49	31,32	31,05	29,63	32,98	35,93
6,90	7,25	7,84	7,80	6,91	7,81	6,92	7,82	6,50	8,42

TOTAL POWER PRODUCTION COSTS IN MILS PER KW=HR

42,81	42,64	42,41	40,81	40,40	39,14	37,97	37,46	39,47	44,36

FUEL COST = $7,50 PER MILLION BTU
ELECT COST = 20,00 MILS PER KW=HR ENERGY COST RATIO = ,3750

35,91	35,39	34,58	33,01	33,49	31,32	31,05	29,63	32,98	35,93
13,79	14,50	15,67	15,61	13,81	15,62	13,84	15,64	12,99	16,84

TOTAL POWER PRODUCTION COSTS IN MILS PER KW=HR

49,70	49,89	50,25	48,61	47,31	46,95	44,88	45,28	45,97	52,78

FUEL COST = $7,75 PER MILLION BTU
ELECT COST = 60,00 MILS PER KW=HR ENERGY COST RATIO = ,1292

37,11	36,57	35,73	34,11	34,61	32,37	32,08	30,62	34,08	37,13
41,38	43,50	47,01	46,82	41,44	46,87	41,51	46,93	38,97	50,53

TOTAL POWER PRODUCTION COSTS IN MILS PER KW=HR

78,49	80,07	82,74	80,93	76,05	79,24	73,59	77,55	73,05	87,66

PUBLIC SERVICE INDIANA
CAES IN AQUIFER
DOE NO. ET-78-C-01-2159

(W 1581)

Table C-12

250 PSIA CHARGING PRESSURE, UNCOOLED TURBINE DISKS, 2 INTERCOOLERS
DANIEL J. MARINACCI, LONG RANGE DEVEOPMENT, A-703 , (215) 595-3283

CASE	11.10	11.20	11.30	11.40	11.50	11.60	11.70	11.80	11.90	11.10
T1HPT	1500.	1300.	607.	1500.	1500.	1500.	1500.	1500.	1500.	1500.
T2HPT	1096.	1010.	497.	1045.	1096.	1045.	1095.	1044.	1058.	1148.
T1LPT	1500.	1500.	1500.	1045.	1500.	1045.	1500.	1044.	1500.	1500.
CAV LOS	1.20	1.20	1.20	1.20	1.20	1.20	1.20	1.20	1.10	1.50
REG EFT	.70	.70	.70	.70	.80	.80	.90	.90	.80	.80
COMP KW	166.2	166.2	166.2	166.2	166.2	166.2	166.2	166.2	166.2	166.2
TURB KW	248.6	236.5	218.9	219.7	248.3	219.5	247.9	219.2	264.0	203.6
H.R.	4788.	4719.	4610.	4401.	4466.	4177.	4140.	3951.	4397.	4791.

FUEL COST = $2.50 PER MILLION BTU
ELECT COST = 10.00 MILS PER KW-HR ENERGY COST RATIO = .2500

11.97	11.80	11.53	11.00	11.16	10.44	10.35	9.88	10.99	11.98
6.68	7.03	7.59	7.56	6.70	7.57	6.71	7.58	6.30	8.16

TOTAL POWER PRODUCTION COSTS IN MILS PER KW-HR

18.66	18.83	19.12	18.57	17.86	18.01	17.05	17.46	17.29	20.14

FUEL COST = $2.50 PER MILLION BTU
ELECT COST = 20.00 MILS PER KW-HR ENERGY COST RATIO = .1250

11.97	11.80	11.53	11.00	11.16	10.44	10.35	9.88	10.99	11.98
13.37	14.05	15.19	15.13	13.39	15.14	13.41	15.16	12.59	16.33

TOTAL POWER PRODUCTION COSTS IN MILS PER KW-HR

25.34	25.85	26.71	26.13	24.55	25.59	23.76	25.04	23.58	28.30

FUEL COST = $4.25 PER MILLION BTU
ELECT COST = 15.10 MILS PER KW-HR ENERGY COST RATIO = .2815

20.35	20.06	19.59	18.70	18.98	17.75	17.59	16.79	18.69	20.36
10.09	10.61	11.47	11.42	10.11	11.43	10.13	11.45	9.51	12.33

TOTAL POWER PRODUCTION COSTS IN MILS PER KW-HR

30.44	30.67	31.06	30.12	29.09	29.18	27.72	28.24	28.19	32.69

FUEL COST = $5.00 PER MILLION BTU
ELECT COST = 17.06 MILS PER KW-HR ENERGY COST RATIO = .2931

23.94	23.60	23.05	22.00	22.33	20.88	20.70	19.76	21.99	23.96
11.40	11.99	12.96	12.90	11.42	12.92	11.44	12.93	10.74	13.93

TOTAL POWER PRODUCTION COSTS IN MILS PER KW-HR

35.35	35.59	36.01	34.91	33.75	33.80	32.14	32.69	32.73	37.88

FUEL COST = $5.00 PER MILLION BTU
ELECT COST = 20.00 MILS PER KW-HR ENERGY COST RATIO = .2500

23.94	23.60	23.05	22.00	22.33	20.88	20.70	19.76	21.99	23.96
13.37	14.05	15.19	15.13	13.39	15.14	13.41	15.16	12.59	16.33

TOTAL POWER PRODUCTION COSTS IN MILS PER KW-HR

37.31	37.65	38.24	37.13	35.72	36.03	34.11	34.92	34.58	40.28

FUEL COST = $7.50 PER MILLION BTU
ELECT COST = 10.00 MILS PER KW-HR ENERGY COST RATIO = .7500

35.91	35.39	34.58	33.01	33.49	31.32	31.05	29.63	32.98	35.93
6.68	7.03	7.59	7.56	6.70	7.57	6.71	7.58	6.30	8.16

TOTAL POWER PRODUCTION COSTS IN MILS PER KW-HR

42.60	42.42	42.17	40.57	40.19	38.90	37.75	37.22	39.28	44.10

FUEL COST = $7.50 PER MILLION BTU
ELECT COST = 20.00 MILS PER KW-HR ENERGY COST RATIO = .3750

35.91	35.39	34.58	33.01	33.49	31.32	31.05	29.63	32.98	35.93
13.37	14.05	15.19	15.13	13.39	15.14	13.41	15.16	12.59	16.33

TOTAL POWER PRODUCTION COSTS IN MILS PER KW-HR

49.28	49.45	49.76	48.13	46.88	46.47	44.46	44.80	45.57	52.26

FUEL COST = $7.75 PER MILLION BTU
ELECT COST = 60.00 MILS PER KW-HR ENERGY COST RATIO = .1292

37.11	36.57	35.73	34.11	34.61	32.37	32.08	30.62	34.08	37.13
40.11	42.16	45.57	45.38	40.17	45.43	40.23	45.49	37.77	48.98

TOTAL POWER PRODUCTION COSTS IN MILS PER KW-HR

77.22	78.74	81.30	79.49	74.78	77.80	72.32	76.11	71.85	86.11

Table C-13

700 PSIA CHARGING PRESSURE, UNCOOLED TURBINE DISKS, 1 INTERCOOLER, A-A
DANIEL J. MARINACCI, LONG RANGE DEVEOPMENT, A-703 , (215) 595-3283

CASE	12.10	12.20	12.30	12.40	12.50	12.60	12.70	12.80
T1HPT	1500.	662.	1500.	1500.	1500.	1400.	1500.	1300.
T2HPT	1110.	279.	875.	1053.	1266.	803.	816.	833.
T1LPT	1500.	1500.	875.	1500.	1500.	803.	1300.	1500.
CAV LOS	1.20	1.20	1.20	1.10	1.50	1.20	1.20	1.20
REG EFT	.70	.70	.70	.80	.80	.70	.70	.70
COMP KW	249.8	249.8	249.8	249.9	249.8	249.8	249.8	249.8
TURB KW	331.5	264.2	284.8	347.3	286.6	268.8	320.9	322.2
H.R.	4132.	4550.	3914.	3969.	4042.	3869.	4286.	4262.

FUEL COST = $2.50 PER MILLION BTU
ELECT COST = 10.00 MILS PER KW-HR ENERGY COST RATIO = .2500

10.33	11.37	9.79	9.92	10.10	9.67	10.71	10.66
7.53	9.45	8.77	7.20	8.71	9.29	7.78	7.75

TOTAL POWER PRODUCTION COSTS IN MILS PER KW-HR

17.87	20.83	18.56	17.12	18.82	18.96	18.50	18.41

FUEL COST = $2.50 PER MILLION BTU
ELECT COST = 20.00 MILS PER KW-HR ENERGY COST RATIO = .1250

10.33	11.37	9.79	9.92	10.10	9.67	10.71	10.66
15.07	18.91	17.54	14.39	17.43	18.58	15.57	15.50

TOTAL POWER PRODUCTION COSTS IN MILS PER KW-HR

25.40	30.28	27.33	24.32	27.53	28.25	26.28	26.16

FUEL COST = $4.25 PER MILLION BTU
ELECT COST = 15.10 MILS PER KW-HR ENERGY COST RATIO = .2815

17.56	19.34	16.63	16.87	17.18	16.44	18.21	18.12
11.38	14.27	13.24	10.87	13.16	14.03	11.75	11.70

TOTAL POWER PRODUCTION COSTS IN MILS PER KW-HR

28.94	33.61	29.88	27.73	30.34	30.47	29.97	29.82

FUEL COST = $5.00 PER MILLION BTU
ELECT COST = 17.06 MILS PER KW-HR ENERGY COST RATIO = .2931

20.66	22.75	19.57	19.84	20.21	19.34	21.43	21.31
12.85	16.13	14.96	12.28	14.87	15.85	13.28	13.22

TOTAL POWER PRODUCTION COSTS IN MILS PER KW-HR

33.52	38.88	34.53	32.12	35.08	35.20	34.71	34.54

FUEL COST = $5.00 PER MILLION BTU
ELECT COST = 20.00 MILS PER KW-HR ENERGY COST RATIO = .2500

20.66	22.75	19.57	19.84	20.21	19.34	21.43	21.31
15.07	18.91	17.54	14.39	17.43	18.58	15.57	15.50

TOTAL POWER PRODUCTION COSTS IN MILS PER KW-HR

35.73	41.66	37.11	34.24	37.64	37.93	36.99	36.81

FUEL COST = $7.50 PER MILLION BTU
ELECT COST = 10.00 MILS PER KW-HR ENERGY COST RATIO = .7500

30.99	34.12	29.36	29.76	30.31	29.02	32.14	31.97
7.53	9.45	8.77	7.20	8.71	9.29	7.78	7.75

TOTAL POWER PRODUCTION COSTS IN MILS PER KW-HR

38.53	43.58	38.13	36.96	39.03	38.31	39.93	39.72

FUEL COST = $7.50 PER MILLION BTU
ELECT COST = 20.00 MILS PER KW-HR ENERGY COST RATIO = .3750

30.99	34.12	29.36	29.76	30.31	29.02	32.14	31.97
15.07	18.91	17.54	14.39	17.43	18.58	15.57	15.50

TOTAL POWER PRODUCTION COSTS IN MILS PER KW-HR

46.06	53.03	46.90	44.16	47.74	47.60	47.71	47.47

FUEL COST = $7.75 PER MILLION BTU
ELECT COST = 60.00 MILS PER KW-HR ENERGY COST RATIO = .1292

32.03	35.26	30.33	30.76	31.32	29.98	33.21	33.03
45.20	56.72	52.62	43.18	52.29	55.75	46.70	46.51

TOTAL POWER PRODUCTION COSTS IN MILS PER KW-HR

77.23	91.98	82.95	73.94	83.61	85.73	79.91	79.54

Table C-14

700 PSIA CHARGING PRESSURE, UNCOOLED TURBINE DISKS, 2 INTERCOOLERS, A-A
DANIEL J. MARINACCI, LONG RANGE DEVEOPMENT, A-703 , (215) 595-3283

CASE	12,10	12,20	12,30	12,40	12,50	12,60	12,70	12,80
T1HPT	1500.	662.	1500.	1500.	1500.	1400.	1500.	1300.
T2HPT	1110.	279.	875.	1053.	1266.	803.	816.	833.
T1LPT	1500.	1500.	875.	1500.	1500.	803.	1300.	1500.
CAV LOS	1,20	1,20	1,20	1,10	1,50	1,20	1,20	1,20
REG EFT	,70	,70	,70	,80	,80	,70	,70	,70
COMP KW	233,4	233,4	233,4	233,4	233,4	233,4	233,4	233,4
TURB KW	331,5	264,2	284,8	347,3	286,6	268,8	320,9	322,2
H.R.	4132.	4550.	3914.	3969.	4042.	3869.	4286.	4262.

FUEL COST = $2,50 PER MILLION BTU
ELECT COST = 10,00 MILS PER KW-HR ENERGY COST RATIO = ,2500

10,33	11,37	9,79	9,92	10,10	9,67	10,71	10,66
7,04	8,83	8,19	6,72	8,14	8,68	7,27	7,24

TOTAL POWER PRODUCTION COSTS IN MILS PER KW-HR

17,37	20,21	17,98	16,64	18,25	18,35	17,99	17,90

FUEL COST = $2,50 PER MILLION BTU
ELECT COST = 20,00 MILS PER KW-HR ENERGY COST RATIO = ,1250

10,33	11,37	9,79	9,92	10,10	9,67	10,71	10,66
14,08	17,67	16,39	13,44	16,28	17,36	14,55	14,48

TOTAL POWER PRODUCTION COSTS IN MILS PER KW-HR

24,41	29,04	26,17	23,36	26,39	27,03	25,26	25,14

FUEL COST = $4,25 PER MILLION BTU
ELECT COST = 15,10 MILS PER KW-HR ENERGY COST RATIO = ,2815

17,56	19,34	16,63	16,87	17,18	16,44	18,21	18,12
10,63	13,34	12,37	10,15	12,29	13,11	10,98	10,94

TOTAL POWER PRODUCTION COSTS IN MILS PER KW-HR

28,19	32,67	29,01	27,01	29,47	29,55	29,20	29,05

FUEL COST = $5,00 PER MILLION BTU
ELECT COST = 17,06 MILS PER KW-HR ENERGY COST RATIO = ,2931

20,66	22,75	19,57	19,84	20,21	19,34	21,43	21,31
12,01	15,07	13,98	11,47	13,89	14,81	12,41	12,36

TOTAL POWER PRODUCTION COSTS IN MILS PER KW-HR

32,67	37,82	33,55	31,31	34,10	34,16	33,84	33,67

FUEL COST = $5,00 PER MILLION BTU
ELECT COST = 20,00 MILS PER KW-HR ENERGY COST RATIO = ,2500

20,66	22,75	19,57	19,84	20,21	19,34	21,43	21,31
14,08	17,67	16,39	13,44	16,28	17,36	14,55	14,48

TOTAL POWER PRODUCTION COSTS IN MILS PER KW-HR

34,74	40,41	35,96	33,28	36,49	36,71	35,97	35,80

FUEL COST = $7,50 PER MILLION BTU
ELECT COST = 10,00 MILS PER KW-HR ENERGY COST RATIO = ,7500

30,99	34,12	29,36	29,76	30,31	29,02	32,14	31,97
7,04	8,83	8,19	6,72	8,14	8,68	7,27	7,24

TOTAL POWER PRODUCTION COSTS IN MILS PER KW-HR

38,03	42,96	37,55	36,48	38,45	37,70	39,41	39,21

FUEL COST = $7,50 PER MILLION BTU
ELECT COST = 20,00 MILS PER KW-HR ENERGY COST RATIO = ,3750

30,99	34,12	29,36	29,76	30,31	29,02	32,14	31,97
14,08	17,67	16,39	13,44	16,28	17,36	14,55	14,48

TOTAL POWER PRODUCTION COSTS IN MILS PER KW-HR

45,07	51,79	45,74	43,21	46,60	46,38	46,69	46,45

FUEL COST = $7,75 PER MILLION BTU
ELECT COST = 60,00 MILS PER KW-HR ENERGY COST RATIO = ,1292

32,03	35,26	30,33	30,76	31,32	29,98	33,21	33,03
42,24	53,00	49,17	40,32	48,85	52,09	43,64	43,45

TOTAL POWER PRODUCTION COSTS IN MILS PER KW-HR

74,26	88,26	79,50	71,08	80,18	82,07	76,85	76,49

Table C-15

1150 PSIA CHARGING PRESSURE, UNCOOLED TURBINE DISKS, 1 INTERCOOLER, A=A
DANIEL J. MARINACCI, LONG RANGE DEVEOPMENT, A=703 , (215) 595=3283

CASE	13.10	13.20	13.30	13.40	13.50	13.60	13.70	13.80	13.90
T1HPT	1500.	501.	1500.	1500.	1500.	1500.	1500.	1500.	1500.
T2HPT	830.	222.	803.	830.	803.	829.	803.	884.	982.
T1LPT	1500.	1500.	803.	1500.	803.	1500.	803.	1500.	1500.
CAV LOS	1.20	1.20	1.20	1.20	1.20	1.20	1.20	1.10	1.50
REG EFT	.70	.70	.70	.80	.80	.90	.90	.80	.80
COMP KW	291.7	291.7	291.7	291.7	291.7	291.7	291.7	291.7	291.7
TURB KW	379.3	301.3	310.9	378.9	310.8	378.4	310.7	391.6	340.9
H.R.	4173.	4152.	3779.	4019.	3724.	3863.	3669.	3935.	4049.

FUEL COST = $2.50 PER MILLION BTU
ELECT COST = 10.00 MILS PER KW=HR ENERGY COST RATIO = .2500

10.43	10.38	9.45	10.05	9.31	9.66	9.17	9.84	10.12
7.69	9.68	9.38	7.70	9.39	7.71	9.39	7.45	8.56
TOTAL POWER PRODUCTION COSTS IN MILS PER KW=HR								
18.12	20.06	18.83	17.75	18.69	17.37	18.56	17.29	18.68

FUEL COST = $2.50 PER MILLION BTU
ELECT COST = 20.00 MILS PER KW=HR ENERGY COST RATIO = .1250

10.43	10.38	9.45	10.05	9.31	9.66	9.17	9.84	10.12
15.38	19.37	18.76	15.40	18.77	15.42	18.78	14.90	17.11
TOTAL POWER PRODUCTION COSTS IN MILS PER KW=HR								
25.81	29.74	28.21	25.45	28.08	25.08	27.95	24.74	27.24

FUEL COST = $4.25 PER MILLION BTU
ELECT COST = 15.10 MILS PER KW=HR ENERGY COST RATIO = .2815

17.73	17.64	16.06	17.08	15.83	16.42	15.59	16.72	17.21
11.61	14.62	14.17	11.63	14.17	11.64	14.18	11.25	12.92
TOTAL POWER PRODUCTION COSTS IN MILS PER KW=HR								
29.35	32.27	30.23	28.71	30.00	28.06	29.77	27.97	30.13

FUEL COST = $5.00 PER MILLION BTU
ELECT COST = 17.06 MILS PER KW=HR ENERGY COST RATIO = .2931

20.86	20.76	18.89	20.09	18.62	19.32	18.34	19.67	20.25
13.12	16.52	16.01	13.14	16.01	13.15	16.02	12.71	14.60
TOTAL POWER PRODUCTION COSTS IN MILS PER KW=HR								
33.98	37.28	34.90	33.23	34.63	32.47	34.36	32.38	34.85

FUEL COST = $5.00 PER MILLION BTU
ELECT COST = 20.00 MILS PER KW=HR ENERGY COST RATIO = .2500

20.86	20.76	18.89	20.09	18.62	19.32	18.34	19.67	20.25
15.38	19.37	18.76	15.40	18.77	15.42	18.78	14.90	17.11
TOTAL POWER PRODUCTION COSTS IN MILS PER KW=HR								
36.24	40.12	37.66	35.49	37.39	34.74	37.12	34.57	37.36

FUEL COST = $7.50 PER MILLION BTU
ELECT COST = 10.00 MILS PER KW=HR ENERGY COST RATIO = .7500

31.30	31.14	28.34	30.14	27.93	28.97	27.52	29.51	30.37
7.69	9.68	9.38	7.70	9.39	7.71	9.39	7.45	8.56
TOTAL POWER PRODUCTION COSTS IN MILS PER KW=HR								
38.99	40.82	37.72	37.84	37.31	36.68	36.91	36.96	38.93

FUEL COST = $7.50 PER MILLION BTU
ELECT COST = 20.00 MILS PER KW=HR ENERGY COST RATIO = .3750

31.30	31.14	28.34	30.14	27.93	28.97	27.52	29.51	30.37
15.38	19.37	18.76	15.40	18.77	15.42	18.78	14.90	17.11
TOTAL POWER PRODUCTION COSTS IN MILS PER KW=HR								
46.68	50.50	47.10	45.54	46.70	44.39	46.30	44.41	47.48

FUEL COST = $7.75 PER MILLION BTU
ELECT COST = 60.00 MILS PER KW=HR ENERGY COST RATIO = .1292

32.34	32.17	29.28	31.14	28.86	29.94	28.43	30.49	31.38
46.14	58.10	56.29	46.20	56.31	46.26	56.34	44.70	51.34
TOTAL POWER PRODUCTION COSTS IN MILS PER KW=HR								
78.48	90.27	85.57	77.34	85.17	76.20	84.77	75.19	82.72

(W 1585)

Table C-16

1150 PSIA CHARGING PRESSURE, UNCOOLED TURBINE DISKS, 1500-1500, 1 INTERR

DANIEL J. MARINACCI, LONG RANGE DEVEOPMENT, A-703 , (215) 595-3283

CASE	5.00	6.00	7.00	7.51	8.00	9.00	10.00	11.00	12.00	14.00	15.00	16.00	17.00	18.00
T1HPT	1500.	1500.	1500.	1500.	1500.	1500.	1500.	1500.	1500.	1500.	1500.	1500.	1500.	1500.
T2HPT	712.	764.	808.	830.	849.	885.	916.	948.	975.	1028.	1050.	1071.	1091.	1110.
T1LPT	1500.	1500.	1500.	1500.	1500.	1500.	1500.	1500.	1500.	1500.	1500.	1500.	1500.	1500.
CAV LOS	1.20	1.20	1.20	1.20	1.20	1.20	1.20	1.20	1.20	1.20	1.20	1.20	1.20	1.20
REG EFT	.70	.70	.70	.70	.70	.70	.70	.70	.70	.70	.70	.70	.70	.70
COMP KW	291.7	291.7	291.7	291.7	291.7	291.7	291.7	291.7	291.7	291.7	291.7	291.7	291.7	291.7
TURB KW	376.2	378.3	379.4	379.3	379.3	378.6	378.1	376.4	375.4	371.3	369.6	367.8	366.0	364.2
H.R.	4306.	4244.	4195.	4173.	4155.	4123.	4094.	4071.	4049.	4016.	4001.	3988.	3976.	3965.

FUEL COST = $2.50 PER MILLION BTU
ELECT COST = 10.00 MILS PER KW-HR   ENERGY COST RATIO = .2500

10.76	10.61	10.49	10.43	10.39	10.31	10.24	10.18	10.12	10.04	10.00	9.97	9.94	9.91
7.75	7.71	7.69	7.69	7.69	7.70	7.71	7.75	7.77	7.86	7.89	7.93	7.97	8.01
TOTAL POWER PRODUCTION COSTS IN MILS PER KW-HR													
18.52	18.32	18.18	18.12	18.08	18.01	17.95	17.93	17.89	17.90	17.90	17.90	17.91	17.92

FUEL COST = $2.50 PER MILLION BTU
ELECT COST = 20.00 MILS PER KW-HR   ENERGY COST RATIO = .1250

10.76	10.61	10.49	10.43	10.39	10.31	10.24	10.18	10.12	10.04	10.00	9.97	9.94	9.91
15.51	15.42	15.38	15.38	15.38	15.41	15.43	15.50	15.54	15.71	15.79	15.86	15.94	16.02
TOTAL POWER PRODUCTION COSTS IN MILS PER KW-HR													
26.27	26.03	25.86	25.81	25.77	25.72	25.66	25.68	25.67	25.75	25.79	25.83	25.88	25.93

FUEL COST = $4.25 PER MILLION BTU
ELECT COST = 15.10 MILS PER KW-HR   ENERGY COST RATIO = .2815

18.30	18.04	17.83	17.73	17.66	17.52	17.40	17.30	17.21	17.07	17.01	16.95	16.90	16.85
11.71	11.64	11.61	11.61	11.61	11.63	11.65	11.70	11.73	11.86	11.92	11.97	12.03	12.09
TOTAL POWER PRODUCTION COSTS IN MILS PER KW-HR													
30.01	29.68	29.44	29.35	29.27	29.16	29.05	29.00	28.94	28.93	28.92	28.92	28.93	28.95

FUEL COST = $5.00 PER MILLION BTU
ELECT COST = 17.06 MILS PER KW-HR   ENERGY COST RATIO = .2931

21.53	21.22	20.97	20.86	20.78	20.61	20.47	20.35	20.25	20.08	20.01	19.94	19.88	19.83
13.23	13.16	13.12	13.12	13.12	13.14	13.16	13.22	13.26	13.40	13.47	13.53	13.60	13.66
TOTAL POWER PRODUCTION COSTS IN MILS PER KW-HR													
34.76	34.38	34.09	33.98	33.90	33.76	33.63	33.58	33.51	33.48	33.47	33.47	33.48	33.49

FUEL COST = $5.00 PER MILLION BTU
ELECT COST = 20.00 MILS PER KW-HR   ENERGY COST RATIO = .2500

21.53	21.22	20.97	20.86	20.78	20.61	20.47	20.35	20.25	20.08	20.01	19.94	19.88	19.83
15.51	15.42	15.38	15.38	15.38	15.41	15.43	15.50	15.54	15.71	15.79	15.86	15.94	16.02
TOTAL POWER PRODUCTION COSTS IN MILS PER KW-HR													
37.04	36.64	36.35	36.24	36.16	36.02	35.90	35.85	35.79	35.79	35.79	35.80	35.82	35.84

FUEL COST = $7.50 PER MILLION BTU
ELECT COST = 10.00 MILS PER KW-HR   ENERGY COST RATIO = .7500

32.29	31.83	31.46	31.30	31.17	30.92	30.71	30.53	30.37	30.12	30.01	29.91	29.82	29.74
7.75	7.71	7.69	7.69	7.69	7.70	7.71	7.75	7.77	7.86	7.89	7.93	7.97	8.01
TOTAL POWER PRODUCTION COSTS IN MILS PER KW-HR													
40.05	39.54	39.15	38.99	38.86	38.63	38.42	38.28	38.14	37.97	37.90	37.84	37.79	37.75

FUEL COST = $7.50 PER MILLION BTU
ELECT COST = 20.00 MILS PER KW-HR   ENERGY COST RATIO = .3750

32.29	31.83	31.46	31.30	31.17	30.92	30.71	30.53	30.37	30.12	30.01	29.91	29.82	29.74
15.51	15.42	15.38	15.38	15.38	15.41	15.43	15.50	15.54	15.71	15.79	15.86	15.94	16.02
TOTAL POWER PRODUCTION COSTS IN MILS PER KW-HR													
47.80	47.25	46.84	46.68	46.55	46.33	46.14	46.03	45.91	45.83	45.79	45.77	45.76	45.76

FUEL COST = $7.75 PER MILLION BTU
ELECT COST = 60.00 MILS PER KW-HR   ENERGY COST RATIO = .1292

33.37	32.89	32.51	32.34	32.20	31.95	31.73	31.55	31.38	31.12	31.01	30.91	30.82	30.73
46.52	46.27	46.13	46.14	46.14	46.23	46.29	46.50	46.63	47.14	47.36	47.58	47.81	48.06
TOTAL POWER PRODUCTION COSTS IN MILS PER KW-HR													
79.90	79.16	78.64	78.48	78.35	78.18	78.02	78.05	78.01	78.26	78.36	78.49	78.63	78.79

Table C-17

1150 PSIA CHARGING PRESSURE, UNCOOLED TURBINE DISKS, 2 INTERCOOLERS,AAA
DANIEL J. MARINACCI, LONG RANGE DEVEOPMENT, A-703 , (215) 595-3283

CASE	13,10	13,20	13,30	13,40	13,50	13,60	13,70	13,80	13,90
T1HPT	1500.	501.	1500.	1500.	1500.	1500.	1500.	1500.	1500.
T2HPT	830.	222.	803.	830.	803.	829.	803.	884.	982.
T1LPT	1500.	1500.	803.	1500.	803.	1500.	803.	1500.	1500.
CAV LOS	1,20	1,20	1,20	1,20	1,20	1,20	1,20	1,10	1,50
REG EFT	,70	,70	,70	,80	,80	,90	,90	,80	,80
COMP KW	266.7	266.7	266.7	266.7	266.7	266.7	266.7	266.7	266.7
TURB KW	379,3	301,3	310,9	378,9	310,8	378,4	310,7	391,6	340,9
H.R.	4173.	4152.	3779.	4019.	3724.	3863.	3669.	3935.	4049.

FUEL COST = $2,50 PER MILLION BTU
ELECT COST = 10,00 MILS PER KW-HR   ENERGY COST RATIO = ,2500

10,43	10,38	9,45	10,05	9,31	9,66	9,17	9,84	10,12
7,03	8,85	8,58	7,04	8,58	7,05	8,59	6,81	7,82

TOTAL POWER PRODUCTION COSTS IN MILS PER KW-HR

17,46	19,23	18,03	17,09	17,89	16,71	17,76	16,65	17,95

FUEL COST = $2,50 PER MILLION BTU
ELECT COST = 20,00 MILS PER KW-HR   ENERGY COST RATIO = ,1250

10,43	10,38	9,45	10,05	9,31	9,66	9,17	9,84	10,12
14,06	17,71	17,16	14,08	17,16	14,10	17,17	13,62	15,65

TOTAL POWER PRODUCTION COSTS IN MILS PER KW-HR

24,50	28,09	26,60	24,13	26,47	23,76	26,34	23,46	25,77

FUEL COST = $4,25 PER MILLION BTU
ELECT COST = 15,10 MILS PER KW-HR   ENERGY COST RATIO = ,2815

17,73	17,64	16,06	17,08	15,83	16,42	15,59	16,72	17,21
10,62	13,37	12,95	10,63	12,96	10,65	12,97	10,29	11,82

TOTAL POWER PRODUCTION COSTS IN MILS PER KW-HR

28,35	31,01	29,01	27,71	28,79	27,06	28,56	27,01	29,03

FUEL COST = $5,00 PER MILLION BTU
ELECT COST = 17,06 MILS PER KW-HR   ENERGY COST RATIO = ,2931

20,86	20,76	18,89	20,09	18,62	19,32	18,34	19,67	20,25
12,00	15,11	14,64	12,01	14,64	12,03	14,65	11,62	13,35

TOTAL POWER PRODUCTION COSTS IN MILS PER KW-HR

32,86	35,86	33,53	32,11	33,26	31,34	32,99	31,30	33,60

FUEL COST = $5,00 PER MILLION BTU
ELECT COST = 20,00 MILS PER KW-HR   ENERGY COST RATIO = ,2500

20,86	20,76	18,89	20,09	18,62	19,32	18,34	19,67	20,25
14,06	17,71	17,16	14,08	17,16	14,10	17,17	13,62	15,65

TOTAL POWER PRODUCTION COSTS IN MILS PER KW-HR

34,93	38,47	36,05	34,17	35,78	33,42	35,52	33,30	35,90

FUEL COST = $7,50 PER MILLION BTU
ELECT COST = 10,00 MILS PER KW-HR   ENERGY COST RATIO = ,7500

31,30	31,14	28,34	30,14	27,93	28,97	27,52	29,51	30,37
7,03	8,85	8,58	7,04	8,58	7,05	8,59	6,81	7,82

TOTAL POWER PRODUCTION COSTS IN MILS PER KW-HR

38,33	39,99	36,92	37,18	36,51	36,02	36,10	36,32	38,19

FUEL COST = $7,50 PER MILLION BTU
ELECT COST = 20,00 MILS PER KW-HR   ENERGY COST RATIO = ,3750

31,30	31,14	28,34	30,14	27,93	28,97	27,52	29,51	30,37
14,06	17,71	17,16	14,08	17,16	14,10	17,17	13,62	15,65

TOTAL POWER PRODUCTION COSTS IN MILS PER KW-HR

45,36	48,85	45,50	44,22	45,09	43,07	44,69	43,14	46,02

FUEL COST = $7,75 PER MILLION BTU
ELECT COST = =0,00 MILS PER KW-HR   ENERGY COST RATIO = ,1292

32,34	32,17	29,28	31,14	28,86	29,94	28,43	30,49	31,38
42,19	53,12	51,47	42,24	51,49	42,30	51,52	40,87	46,95

TOTAL POWER PRODUCTION COSTS IN MILS PER KW-HR

74,53	85,30	80,76	73,39	80,35	72,24	79,95	71,37	78,33

(W 1587)

Table C-18

1150 PSIA CHARGING PRESSURE,UNCOOLED TURBINE DISKS,1500-1500, 2 INTERR
DANIEL J. MARINACCI, LONG RANGE DEVEOPMENT, A-703 , (215) 595-3283

CASE	5.00	6.00	7.51	7.00	8.00	9.00	10.00	11.00	12.00	14.00	15.00	16.00	17.00	18.00
T1HPT	1500.	1500.	1500.	1500.	1500.	1500.	1500.	1500.	1500.	1500.	1500.	1500.	1500.	1500.
HPT	712.	764.	830.	808.	849.	885.	916.	948.	975.	1028.	1050.	1071.	1091.	1110.
LPT	1500.	1500.	1500.	1500.	1500.	1500.	1500.	1500.	1500.	1500.	1500.	1500.	1500.	1500.
CAV LOS	1.20	1.20	1.20	1.20	1.20	1.20	1.20	1.20	1.20	1.20	1.20	1.20	1.20	1.20
REG EFT	.70	.70	.70	.70	.70	.70	.70	.70	.70	.70	.70	.70	.70	.70
COMP KW	266.7	266.7	266.7	266.7	266.7	266.7	266.7	266.7	266.7	266.7	266.7	266.7	266.7	266.7
TURB KW	376.2	378.3	379.3	379.4	379.3	378.6	378.1	376.4	375.4	371.3	369.6	367.8	366.0	364.2
H.R.	4306.	4244.	4173.	4195.	4155.	4123.	4094.	4071.	4049.	4016.	4001.	3988.	3976.	3965.

FUEL COST = $2.50 PER MILLION BTU
ELECT COST = 10.00 MILS PER KW-HR ENERGY COST RATIO = .2500

10.76	10.61	10.43	10.49	10.39	10.31	10.24	10.18	10.12	10.04	10.00	9.97	9.94	9.91
7.09	7.05	7.03	7.03	7.03	7.05	7.05	7.09	7.11	7.18	7.22	7.25	7.29	7.32

TOTAL POWER PRODUCTION COSTS IN MILS PER KW-HR

17.86	17.66	17.46	17.52	17.42	17.35	17.29	17.26	17.23	17.22	17.22	17.22	17.23	17.24

FUEL COST = $2.50 PER MILLION BTU
ELECT COST = 20.00 MILS PER KW-HR ENERGY COST RATIO = .1250

10.76	10.61	10.43	10.49	10.39	10.31	10.24	10.18	10.12	10.04	10.00	9.97	9.94	9.91
14.18	14.10	14.06	14.06	14.07	14.09	14.11	14.17	14.21	14.37	14.43	14.50	14.57	14.65

TOTAL POWER PRODUCTION COSTS IN MILS PER KW-HR

24.95	24.71	24.49	24.55	24.45	24.40	24.34	24.35	24.34	24.41	24.44	24.47	24.51	24.56

FUEL COST = $4.25 PER MILLION BTU
ELECT COST = 15.10 MILS PER KW-HR ENERGY COST RATIO = .2815

18.30	18.04	17.73	17.83	17.66	17.52	17.40	17.30	17.21	17.07	17.01	16.95	16.90	16.85
10.71	10.65	10.62	10.62	10.62	10.64	10.65	10.70	10.73	10.85	10.90	10.95	11.00	11.06

TOTAL POWER PRODUCTION COSTS IN MILS PER KW-HR

29.01	28.68	28.35	28.44	28.28	28.16	28.05	28.00	27.94	27.91	27.90	27.90	27.90	27.91

FUEL COST = $5.00 PER MILLION BTU
ELECT COST = 17.06 MILS PER KW-HR ENERGY COST RATIO = .2931

21.53	21.22	20.86	20.97	20.78	20.61	20.47	20.35	20.25	20.08	20.01	19.94	19.88	19.83
12.10	12.03	11.99	12.00	12.00	12.02	12.03	12.09	12.12	12.26	12.31	12.37	12.43	12.49

TOTAL POWER PRODUCTION COSTS IN MILS PER KW-HR

33.63	33.25	32.86	32.97	32.78	32.63	32.51	32.44	32.37	32.34	32.32	32.31	32.31	32.32

FUEL COST = $5.00 PER MILLION BTU
ELECT COST = 20.00 MILS PER KW-HR ENERGY COST RATIO = .2500

21.53	21.22	20.86	20.97	20.78	20.61	20.47	20.35	20.25	20.08	20.01	19.94	19.88	19.83
14.18	14.10	14.06	14.06	14.07	14.09	14.11	14.17	14.21	14.37	14.43	14.50	14.57	14.65

TOTAL POWER PRODUCTION COSTS IN MILS PER KW-HR

35.71	35.32	34.93	35.03	34.84	34.70	34.58	34.53	34.46	34.45	34.44	34.44	34.46	34.47

FUEL COST = $7.50 PER MILLION BTU
ELECT COST = 10.00 MILS PER KW-HR ENERGY COST RATIO = .7500

32.29	31.83	31.30	31.46	31.17	30.92	30.71	30.53	30.37	30.12	30.01	29.91	29.82	29.74
7.09	7.05	7.03	7.03	7.03	7.05	7.05	7.09	7.11	7.18	7.22	7.25	7.29	7.32

TOTAL POWER PRODUCTION COSTS IN MILS PER KW-HR

39.38	38.88	38.33	38.49	38.20	37.97	37.76	37.62	37.48	37.30	37.23	37.16	37.11	37.06

FUEL COST = $7.50 PER MILLION BTU
ELECT COST = 20.00 MILS PER KW-HR ENERGY COST RATIO = .3750

32.29	31.83	31.30	31.46	31.17	30.92	30.71	30.53	30.37	30.12	30.01	29.91	29.82	29.74
14.18	14.10	14.06	14.06	14.07	14.09	14.11	14.17	14.21	14.37	14.43	14.50	14.57	14.65

TOTAL POWER PRODUCTION COSTS IN MILS PER KW-HR

46.47	45.93	45.36	45.52	45.23	45.01	44.82	44.70	44.58	44.49	44.44	44.41	44.40	44.39

FUEL COST = $7.75 PER MILLION BTU
ELECT COST = 60.00 MILS PER KW-HR ENERGY COST RATIO = .1292

33.37	32.89	32.34	32.51	32.20	31.95	31.73	31.55	31.38	31.12	31.01	30.91	30.82	30.73
42.54	42.31	42.18	42.19	42.20	42.27	42.33	42.52	42.64	43.11	43.30	43.51	43.72	43.94

TOTAL POWER PRODUCTION COSTS IN MILS PER KW-HR

75.91	75.20	74.52	74.69	74.40	74.22	74.06	74.07	74.02	74.23	74.31	74.42	74.54	74.67

Table C-19

1150 CHARGING PRESSURE, EFFECT OF HP TURBINE FIRING TEMP UNCOOLED DISKS
DANIEL J. MARINACCI, LONG RANGE DEVEOPMENT, A-703 , (215) 595-3283

CASE	.11	.12	.21	.22	.31	.32	.41	.42
T1HPT	1500.	1500.	1300.	1300.	1000.	1000.	465.	465.
T2HPT	830.	830.	748.	748.	652.	652.	232.	232.
T1LPT	1500.	1500.	1500.	1500.	1500.	1500.	1500.	1500.
CAV LOS	1.20	1.20	1.20	1.20	1.20	1.20	1.20	1.20
REG EFT	.70	.70	.70	.70	.70	.70	.70	.70
COMP KW	291.7	266.7	291.7	266.7	291.7	266.7	291.7	266.7
TURB KW	379.3	379.3	360.4	360.4	336.7	336.7	303.9	303.9
H.R.	4173.	4173.	4135.	4135.	4029.	4029.	4085.	4085.

FUEL COST = $2.50 PER MILLION BTU
ELECT COST = 10.00 MILS PER KW-HR ENERGY COST RATIO = .2500

10.43	10.43	10.34	10.34	10.07	10.07	10.21	10.21
7.69	7.03	8.09	7.40	8.66	7.92	9.60	8.78

TOTAL POWER PRODUCTION COSTS IN MILS PER KW-HR

18.12	17.46	18.43	17.74	18.74	18.00	19.81	18.99

FUEL COST = $2.50 PER MILLION BTU
ELECT COST = 20.00 MILS PER KW-HR ENERGY COST RATIO = .1250

10.43	10.43	10.34	10.34	10.07	10.07	10.21	10.21
15.38	14.06	16.19	14.80	17.33	15.85	19.20	17.55

TOTAL POWER PRODUCTION COSTS IN MILS PER KW-HR

25.81	24.50	26.52	25.14	27.40	25.92	29.41	27.77

FUEL COST = $4.25 PER MILLION BTU
ELECT COST = 15.10 MILS PER KW-HR ENERGY COST RATIO = .2815

17.73	17.73	17.57	17.57	17.12	17.12	17.36	17.36
11.61	10.62	12.22	11.18	13.08	11.96	14.49	13.25

TOTAL POWER PRODUCTION COSTS IN MILS PER KW-HR

29.35	28.35	29.80	28.75	30.21	29.09	31.86	30.62

FUEL COST = $5.00 PER MILLION BTU
ELECT COST = 17.06 MILS PER KW-HR ENERGY COST RATIO = .2931

20.86	20.86	20.68	20.68	20.15	20.15	20.43	20.43
13.12	12.00	13.81	12.63	14.78	13.52	16.38	14.97

TOTAL POWER PRODUCTION COSTS IN MILS PER KW-HR

33.98	32.86	34.48	33.30	34.93	33.66	36.80	35.40

FUEL COST = $5.00 PER MILLION BTU
ELECT COST = 20.00 MILS PER KW-HR ENERGY COST RATIO = .2500

20.86	20.86	20.68	20.68	20.15	20.15	20.43	20.43
15.38	14.06	16.19	14.80	17.33	15.85	19.20	17.55

TOTAL POWER PRODUCTION COSTS IN MILS PER KW-HR

36.24	34.93	36.86	35.48	37.48	35.99	39.62	37.98

FUEL COST = $7.50 PER MILLION BTU
ELECT COST = 10.00 MILS PER KW-HR ENERGY COST RATIO = .7500

31.30	31.30	31.01	31.01	30.22	30.22	30.64	30.64
7.69	7.03	8.09	7.40	8.66	7.92	9.60	8.78

TOTAL POWER PRODUCTION COSTS IN MILS PER KW-HR

38.99	38.33	39.11	38.41	38.89	38.14	40.24	39.42

FUEL COST = $7.50 PER MILLION BTU
ELECT COST = 20.00 MILS PER KW-HR ENERGY COST RATIO = .3750

31.30	31.30	31.01	31.01	30.22	30.22	30.64	30.64
15.38	14.06	16.19	14.80	17.33	15.85	19.20	17.55

TOTAL POWER PRODUCTION COSTS IN MILS PER KW-HR

46.68	45.36	47.20	45.82	47.55	46.07	49.84	48.19

FUEL COST = $7.75 PER MILLION BTU
ELECT COST = 60.00 MILS PER KW-HR ENERGY COST RATIO = .1292

32.34	32.34	32.05	32.05	31.23	31.23	31.66	31.66
46.14	42.19	48.56	44.40	51.99	47.54	57.59	52.66

TOTAL POWER PRODUCTION COSTS IN MILS PER KW-HR

78.48	74.53	80.61	76.45	83.22	78.77	89.25	84.32

(W 1589)

3

COMPRESSED AIR ENERGY STORAGE
PRELIMINARY DESIGN AND SITE DEVELOPMENT PROGRAM IN AN AQUIFER
DOE CONTRACT NO. ET-78-C-01-2159

FINAL DRAFT

Task 1 - Volume 3

Establish Facility Design Criteria and Utility Benefits

October, 1980

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Interim Report

Economic Analysis of CAES in Aquifers

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Interim Report

Economic Analysis of CAES in Aquifers

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Exhibits

Interim Report

Economic Analysis of CAES in Aquifers

Introduction

This study examines the comparative economics of Compressed Air Energy Storage (CAES) and intermediate load coal-fired power plants during the 1980's and 1990's. The study is of a hypothetical system generally consisting of the combined sponsoring utilities' systems. For this report, two scenarios are investigated; both scenarios have a target generation mix of 65% base, 25% intermediate, and 10% peaking capacity. The first scenario includes installing new capacity at a ratio of 50% nuclear and 50% coal-fired to the base load portion of the target generation mix. CAES is evaluated by replacing an intermediate load coal-fired unit with the CAES unit. The second scenario examines an expansion program under which all new base load generation is nuclear capacity. Again, CAES is evaluated by replacing an intermediate load coal unit with the CAES unit. For each scenario, CAES is evaluated as a 290 MW demonstration unit installed in 1985 with the site expanded to 1160 MW with the addition of 870 MW in 1990.

Conclusions

The conclusions that are presented below are based on conceptual data that has been supplied to the System Studies Group.

Future CAES studies might utilize more definitive data resulting in slightly different conclusions.

1. The CAES alternative is more economical when compared to a fossil unit for a system in which:

   a. All future base load capacity is nuclear and,

   b. The distillate fuel escalation rate is approximately the same as coal.

2. The CAES alternative is marginal when compared to a fossil unit for the hypothetical system in which the escalation rate for oil is three percentage points higher than the coal escalation rate.

3. The CAES facility for the hypothetical system becomes more attractive as the amount of nuclear capacity included in the expansion program is increased.

<u>Basis of Analysis</u>

<u>Future Unit Parameters</u>

The future unit parameters selected for use in this analysis are included in Exhibit 1. Generating unit types used in the study are consistent with unit sizes currently being projected by the utility sponsors. Smaller fossil units will be considered as participatory units of a larger size.

The full and partial forced outage rates were determined by surveying the sponsors. Since flue gas desulfurization (FGD) systems are relatively new, there is not sufficient operating data available to use as a basis for future projections. Therefore, three assumptions were made:

1. No "bypasses" would be operated,

2. Redundant SO_2 removal equipment would be available, and

3. SO_2 removal technology will advance with time.

The latter two assumptions greatly reduce the impact of flue gas desulfurization equipment related forced outages. The full forced outage rates for nuclear, peaking, and non-scrubber fossil units are consistent with values used by the sponsors for existing units.

Heat rate data was developed from data obtained from the utility sponsors. Nuclear unit heat rate data was provided by Commonwealth Edison due to its extensive experience in operating nuclear plants. This data is representative of similar type units being projected throughout the country.

All CAES unit specific data was supplied by Sargent and Lundy Engineers and Westinghouse Electric Corporation in partial fulfillment of their involvement in the work effort for this CAES project.

Capacity Addition Schedule

The Capacity Addition Schedule shown in Exhibit 2 was developed by the CAES Systems Studies Group. This schedule is based upon the capacity addition plan of a hypothetical system supplying the combined projected load of the utility sponsors. The existing generating equipment for this hypothetical system consists of the equipment currently owned and operated by the utility sponsors. The future additions to the system are based on meeting a targeted reserve margin of 18% and a generation mix (Exhibit 3) of 65% base capacity, 25% intermediate capacity, and 10% peaking capacity. The generation mix used for the theoretical system is consistent with the mix of equipment currently in use and projected by the utility sponsors.

Fuel Cost Projections

Each sponsor was surveyed to determine a range of expected fuel costs and escalation rates during the study period. Based upon this information, estimates of price for the various fuel types and the associated escalation rates were made. Two distillate oil escalation rates are used as a sensitivity analysis beginning with the 1985 cost. The fuel costs and the escalation rates are provided in Exhibit 4.

Capital Cost

The capital costs utilized in the CAES system studies are presented in Exhibit 5. In the development of the capital costs, a literature search was made and a survey of the utility sponsors was conducted. Capital cost estimates were prepared through the use of the CONCEPT V computer code developed by Oak Ridge National Laboratory. In comparing the CONCEPT estimates to the survey results, it was observed that CONCEPT compared favorably with the survey results for the two-unit plant average for typical Midwestern locations of the 575 MW and 450 MW fossil units. However, the CONCEPT code consistently understated the two-unit plant average of the 1150 MW PWR nuclear units. Therefore, the CONCEPT estimates for the two fossil unit sizes were used as calculated while the nuclear plant costs are based on 110% of the CONCEPT estimate. The combustion turbine costs were developed from current estimates by the CAES utility sponsors.

The cost of the 290 MW pilot CAES plant and the 1160 MW production CAES plant were developed by Sargent and Lundy Engineers. The estimates were not made for any specific site, but were representative of many sites presently being investigated.

A capital escalation rate of 7.5% per year and an Allowance for Funds Used During Construction (AFDC) rate of

9% per year (compounded annually) were selected by the Study Group.

Operation and Maintenance

The operation and maintenance (O&M) costs of future units are shown in Exhibit 6. The O&M costs were prepared by surveying the utility sponsors and comparing the results to estimates determined from a literature search. The fossil and nuclear estimates used in the CAES system studies were intended to be representative of the O&M cost projected for new units in the Midwest. These costs are expressed as both fixed ($/MW/Week) and variable ($/MWH) quantities. The O&M cost for the CAES unit was provided by Sargent and Lundy Engineers and Westinghouse Combustion Turbine Division. A seven percent annual escalation rate on variable and fixed O&M costs was selected by the Systems Studies Group.

Financial Parameters

The financial parameters used in the study analysis were developed by the CAES Systems Studies Group to be used in the comparison of revenue requirements of each alternative during the study period. "Revenue requirements include fixed costs such as (a minimum) return on investment, depreciation, tax, etc., which result from having made an investment, and operating costs, such as fuel cost, operation and maintenance cost, etc., which result from the use of the investment" (EPRI Technical Assessment Guide, June 1978, PS866-RS). The minimum acceptable return (MAR) is the lowest amount that investors will accept to provide the funds needed to make an investment. The MAR is equal to the weighted cost of capital. For the purpose of this study, the economic comparison is the difference in revenue requirements of the two alternatives.

Exhibit 7 presents the financial parameters of a hypothetical system developed by the CAES sponsors. The values assumed for the development of the parameters used in the financial calculations are shown in the same exhibit. The utilization of a fixed charge rate that includes tax preferences is consistent with the methods used by the majority of the CAES sponsors. The discount rate used is different from the MAR since tax effects are included in the discount rate. The value for the fixed charge rate in Exhibit 7 is consistent with the levelized annual fixed charge rate with tax preference as cited in EPRI Technical Assessment Guide, June, 1978 (PS866-RS).

Energy Storage Sizing

CAES installations must provide a system with dependable capacity in addition to the energy they supply. The dependability of any storage capacity is a function of the storage capability and the load characteristics of the particular utility system. Proper storage sizing is essential to the dependability of CAES capacity and to the effective utilization of this capacity for supplying economical energy. Consequently, storage size has a direct bearing on the capacity and energy value of a CAES installation. The storage size of the CAES facility is a function of the output energy provided by supplemental fuel consumption and may, therefore, be somewhat less than normally expected for other types of energy storage facilities with the same megawatt capacity.

The storage sizing methodology developed by the Systems Studies Group is applied to a 1160 MW CAES project which is installed on the hypothetical system described earlier in this report. This methodology is described in the Appendix. The results of the evaluation using this methodology indicate that the operational uncertainties (e.g. weather patterns) and the energy demands on the CAES facility are the critical storage sizing criteria. Whether the uncertainty criterion or the energy demand requirements dominate depends on the variables that are selected to simulate the uncertainties associated with operating the CAES facility.

The results of the CAES storage sizing evaluation indicate that a range of 26 to 36 hours is adequate to support the operation of the CAES facility. The risk of occasional inoperability due to storage limitations will increase with a smaller storage size.

System Production Cost Modeling

The PROMOD production cost program developed by Energy Management Associates (EMA), Atlanta, Georgia, was utilized for all production cost analyses. Modification of EMA's Pumped Storage Module to model compressed air systems was performed by EMA under a previous contract with EPRI.

Each sponsoring company provided the input for modeling its own system. One year validation cases were run to verify that each system was being correctly modeled. These individual cases were then merged with the two generation expansion programs (Exhibit 2) to produce the base cases.

A composite load profile was developed from each sponsor's typical year. Data was provided in the form of 8760 hourly loads in EEI format. Monthly demand and energy estimates were provided by each member and were integrated to develop a composite demand and energy profile for the hypothetical system.

Sixteen years (1980-1995) of production cost estimates were obtained using PROMOD. Since the hypothetical system achieved its reserve criteria and targeted generation mix by 1990, an analysis was made of the production cost trends in the six year period (1990-1995). These trends were then used to develop long term production costs through the year 2020. Using trended production costs resulted in savings in computer time while allowing for comparison of operating costs over a thirty year book life for the CAES facility.

Cycle Hours

The operating cycle of the CAES unit is fixed by the definition of the weekday, weeknight, and weekend hours in PROMOD. The CAES unit is modeled to generate during the weekday period and charge during the weeknight and weekend periods.

Monthly sensitivities of start times were studied for every hour from 6:00 a.m. through 10:00 a.m. in order to determine an economical time for dispatch of the facility. For each starting hour, the generating mode time was varied from 12 to 16 hours. A six day CAES generation week and a weekly refill cycle was assumed. Exhibit 8 is a typical graph/or a typical month of the results of these runs. The dispatch period which resulted in the lowest total production cost was chosen to define each monthly operating cycle. The start times ranged from 6:00 a.m. to 9:00 a.m. while the duration was either 15 or 16 hours, leaving eight to nine hours for charging.

Total Generation Cost Versus Capacity Factor

To gain an insight into the attractiveness of the different forms of generation alternatives and, in particular, how CAES compares with other types of conventional generating units, curves which show total generation cost as a function of capacity factor were developed. Commonly known as screening curves, they are meaningful for one point in time only.

Screening curves are useful as a simple method for eliminating from further consideration those alternatives which are significantly less economical. They cannot be used as a substitute for a detailed revenue requirement analysis.

The screening curves presented in Exhibit 9 are based on generic plant costs. The exhibit illustrates the total annual production and owning cost of five different types of generation alternatives for the year 1985. As can be seen from the exhibit, the CAES plant would be economically competitive with an intermediate fossil unit if its capacity factor is below 39%. Likewise, it would be more economical than a combustion turbine plant at a capacity factor above 11%.

Results

CAES Fuel Analysis

The substitute of a CAES facility for an intermediate load coal fired plant results in a decrease in coal consumption and an increase in the oil burned. A summary of the amount of fuel consumed during the period of 1990-1995 is shown in Exhibit 10.

The increase in oil consumption is the result of the intermediate load coal fired plant which operates at approximately a 49% capacity factor being replaced by the CAES facility which operates near 20% capacity factor.

Revenue Requirement Results

Exhibits 11 and 12 present the results of the revenue requirement analysis for the period 1980 through 2020 for both expansion scenarios and both distillate fuel escalation sensitivities. Exhibit 11, the all nuclear base load capacity expansion scenario, indicates that for a 9% distillate escalation rate beginning in 1985, the CAES expansion alternative is more economical than the fossil alternative. The difference in total revenue requirement represents a 0.42% savings while the difference in production cost amounts to a 0.15% savings. For a 12% distillate escalation rate, there is an approximate breakeven in the total revenue requirements.

In the 50% nuclear - 50% coal base load scenario, CAES does not appear to show any advantage over the competitive fossil unit. For both 9 and 12% distillate fuel escalation rates, production costs are higher than the base costs and the capital cost benefits for CAES are not sufficient to show an overall revenue requirement savings for the CAES alternative.

System Studies Group

JCB:djf

Exhibit 1

Future Unit Parameters

Unit Type	Forced Outage Rate (%)	Capacity State Intermediate (MW)	Capacity State Base (MW)	State Availabilities (%)	Average Heat Rate (Btu/kWh)
Fossil-Coal	13	450	575	31	10762
		405	518	22	10820
		360	460	13	11000
		306	390	21	11380
Nuclear	8	1150		54	10518
		950		16	10722
		695		12	11213
		437		7	12437
		330		3	13505
Peaking	15	50		85	13500

CAES Parameters

	Demonstration Unit	Full Size Plant[1]
Size	290 MW	1160 MW
Module Compressor Size	205 MW	205 MW
Module Generator Size	290 MW	290 MW
Overall Cycle Efficiency	141%	141%
Average Heat Rate	3880 Btu/kWh	3880 Btu/kWh
Forced Outage Rate	5%	19%
Planned Maintenance Per Module	6 Week/Year	6 Week/Year

(1) Including demonstration plant.

Exhibit 2

CAPACITY ADDITION SCHEDULE
ALL NUCLEAR BASE SCENARIO
CAES Theoretical System
1979-1995

Year	Capacity Addition (Reduction) MW		Adjusted Capacity (MW)	Adjusted Demand (MW)	% Reserve
1979	-	-	36,336	28,792	26.2
1980	102 P	(19) BF	36,419	30,143	20.8
1981	1048 BN	(21) P	37,186	31,306	18.8
1982	1048 BN 720 BF		38,714	32,230	20.1
1983	1150 BN 650 I		40,514	33,742	20.0
1984	1350 I	(15) BF (271) P	41,578	35,225	18.0
1985	1350 I 400 P		43,328	36,724	18.0
1986	1150 BN 450 I 550 P		45,478	38,550	18.0
1987	1150 BN 300 P		46,928	39,781	18.0
1988	2300 BN		49,228	41,595	18.4
1989	1150 BN 450 I 300 P		51,128	43,329	18.0
1990	1150 BN 900 I 100 P		53,278	45,150	18.0
1991	2300 BN 450 I	(520) I	55,508	47,029	18.0
1992	1150 BN 900 I 250 P		57,808	49,010	18.0
1993	1150 BN 900 I 500 P	(127) P	60,231	51,056	18.0
1994	1150 BN 900 I 500 P		62,781	53,190	18.0
1995	2300 BN 450 I		65,531	55,404	18.3

BN - Base Nuclear
BF - Base Fossil
I - Intermediate
P - Peaking Oil

Exhibit 2 (cont'd)

CAPACITY ADDITION SCHEDULE
50% COAL/50% NUCLEAR BASE SCENARIO
CAES Theoretical System
1979-1995

Year	Capacity Addition (Reduction) MW		Adjusted Capacity (MW)	Adjusted Demand (MW)	% Reserve
1979	-	-	36,336	28,792	26.2
1980	102 P	(19) B	36,419	30,143	20.8
1981	1048 BN	(21) P	37,186	31,306	18.8
1982	1048 BN 720 BF		38,714	32,230	20.1
1983	1150 BN 650 I		40,514	33,742	20.0
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1988	1150 BF 1150 BN		49,228	41,595	18.4
1989	1150 BF 450 I 300 P		51,128	43,329	18.0
1990	1150 BN 900 I 100 P		53,278	45,150	18.0
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1992	1150 BF 900 I 250 P		57,808	49,010	18.0
1993	1150 BN 900 I 500 P	(127) P	60,231	51,056	18.0
1994	1150 BF 900 I 500 P		62,781	53,190	18.0
1995	1150 BN 1150 BF 450 I		65,531	55,404	18.3

BN - Base Nuclear　　I - Intermediate
BF - Base Fossil　　P - Peaking

Exhibit 3

PROJECTED GENERATION MIX
CAES Theoretical System
1979-1995

Year	Base	Intermediate	Peak	Total
1979	23,749 (66%)	8,058 (22%)	4,029 (11%)	35,836
1980	(19) 23,730 (66%)	8,058 (22%)	102 4,131 (12%)	35,919
1981	1,048 24,778 (67%)	8,058 (22%)	(21) 4,110 (11%)	36,946
1982	1,768 26,546 (69%)	8,058 (21%)	4,110 (11%)	38,714
1983	1,150 27,696 (68%)	650 8,708 (21%)	4,110 (10%)	40,514
1984	(15) 27,681 (67%)	1,350 10,058 (24%)	(271) 3,839 (9%)	41,578
1985	27,681 (64%)	1,350 11,408 (26%)	400 4,239 (10%)	43,328
1986	1,150 28,831 (63%)	450 11,858 (26%)	550 4,789 (11%)	45,478
1987	1,150 29,981 (64%)	11,858 (25%)	300 5,089 (11%)	46,928
1988	2,300 32,281 (66%)	11,858 (24%)	5,089 (10%)	49,228
1989	1,150 33,431 (65%)	450 12,308 (24%)	300 5,389 (11%)	51,128
1990	1,150 34,581 (65%)	900 13,208 (25%)	100 5,489 (10%)	53,278
1991	2,300 36,881 (66%)	(70) 13,138 (24%)	5,489 (10%)	55,508
1992	1,150 38,031 (66%)	900 14,038 (24%)	250 5,739 (10%)	57,808
1993	1,150 39,181 (65%)	900 14,938 (25%)	373 6,112 (10%)	60,231
1994	1,150 40,331 (64%)	900 15,838 (25%)	500 6,612 (11%)	62,781
1995	2,300 42,631 (65%)	450 16,288 (25%)	6,612 (10%)	65,531

Exhibit 4

Fuel Cost Projection

Fuel Type	Cost (January, 1985) (¢/MBtu)	Escalation (%)
Nuclear	94	7
High Sulfur Coal	200	9
Low Sulfur Coal	300	9
Distillate Oil	790	9,12

NOTE: For the preliminary report, synthetic fuels and a corresponding escalation rate were not considered.

Exhibit 5

CAPITAL COSTS

Generating Unit Type	Installed Cost ($/kW)
1150 MW PWR (Nuclear)	1117
575 MW Base Fossil	1012
450 MW Intermediate Fossil	1103
50 MW Peaking	262
290 MW CAES	882
870 MW CAES	549

Assumptions:

1) Costs for nuclear and fossil units are based on a two unit average
2) All costs are in 1985 dollars
3) An escalation rate of 7.5%/Year is used
4) An AFDC rate of 9% compounded annually is used
5) The 870 MW CAES Facility is the add on to the 290 Demonstration facility

Exhibit 6

OPERATION AND MAINTENANCE COSTS

Generating Unit Type	Fixed O/M ($/MW/WK)	Variable O/M ($/MWH)
1150 MW PWR (Nuclear)	295	.48
575 MW Base Fossil	132	5.30
450 MW Intermediate Fossil	170	5.30
50 MW Peaking	0	8.65
290 MW CAES	15	2.95
1160 MW CAES	15	2.95

Assumptions:

1) Escalation Rate of 7.0%/Year

2) All costs are in 1985 dollars

Exhibit 7

LEVELIZED CARRYING CHARGE RATE

Goal Capital Structure

	Ratio	Cost					
Debt	50%	9.6%	4.80%	-	2.30%	=	2.50%
Preferred	10%	10%	1.00%				1.00%
Common	40%	15%	6.00%				6.00%
		MAR =	11.80%				9.50% = Discount rate

Assumptions

	Fossil	Nuclear	Comb. Turbine	Comp. Air Storage
Book Life (Yrs.)	30	30	30	30
Tax Guideline Life (Yrs.)	22.5	16	16	16
Composite Income Tax (%)	48	48	48	48
Investment Tax Credit (%)	10	10	10	10
Financing Costs (%)	12.6	11.8	11.8	11.8
Other Taxes & Insurance (%)	2.9	2.9	2.9	2.9
Decommissioning Allocation	-	0.3	-	-
Levelized Fixed Charge Rate (%)	15.5	15.0	14.7	14.7

Exhibit 8

CAES Operating Hours

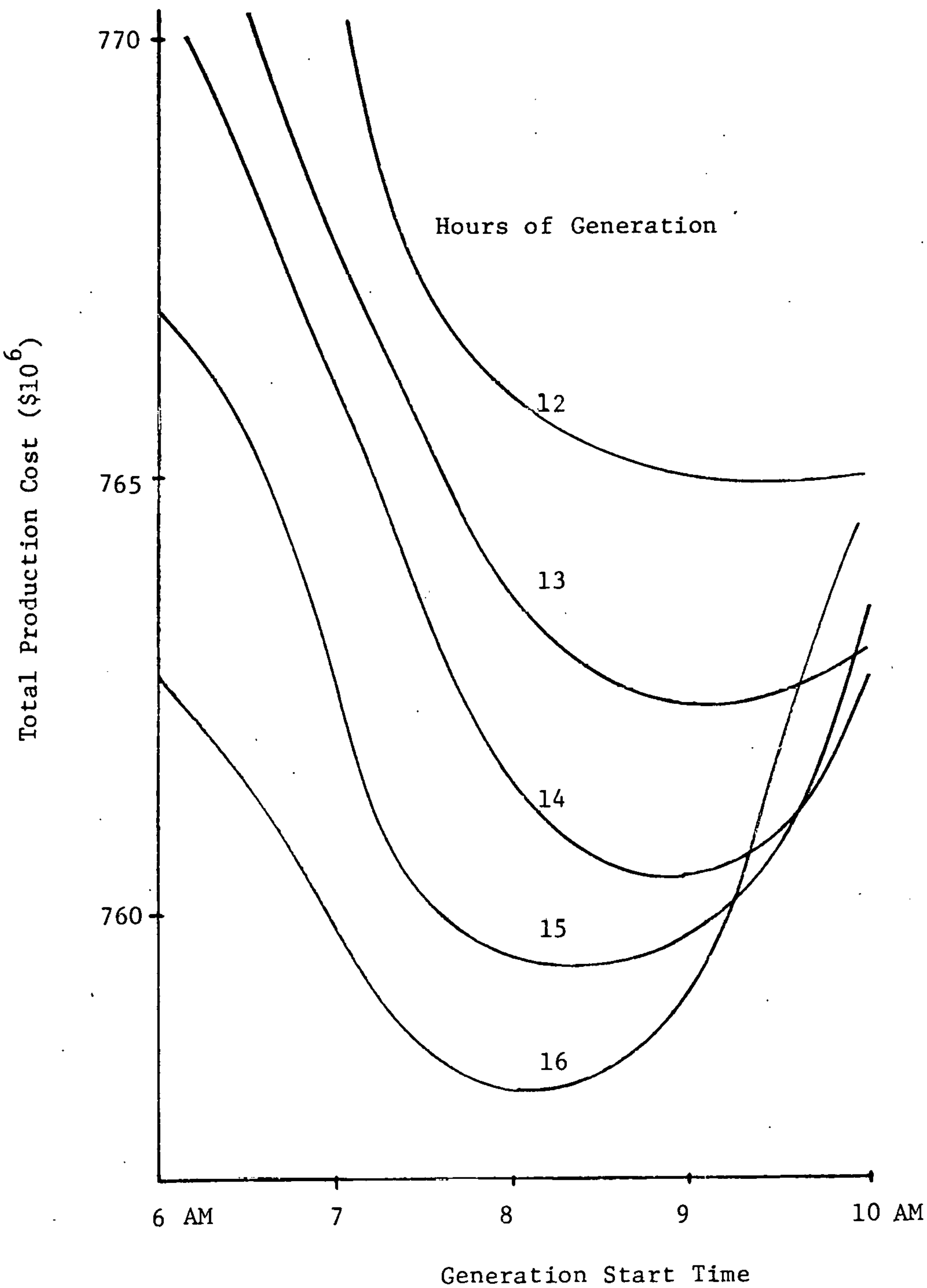


Exhibit 9

Total Generating Costs Versus Capacity Factor
(Screening Curves)
1985

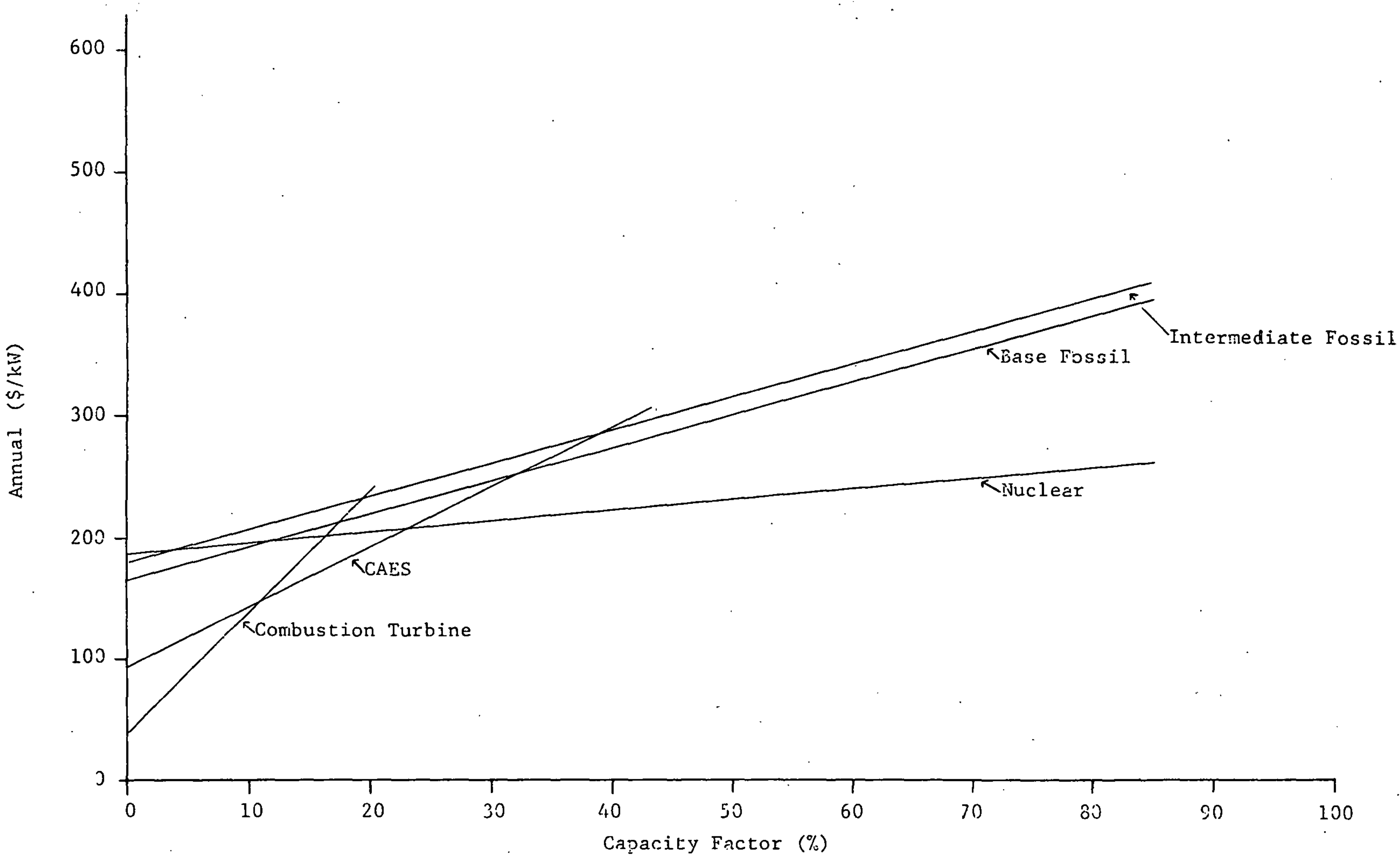


Exhibit 10

Units of Fuel Consumed 1990-1995
(Value x 10^6)

	Base Case - Total				CAES Case - Increase			
	MBtu (Nuclear)	Tons (Coal)	BBLS (#6 Fuel Oil)	BBLS (Distillates)	MBtu (Nuclear)	Tons (Coal)	BBLS (#6 Fuel Oil)	BBLS (Distillates)
12% Escalation								
All Nuclear Base Load	7416	409	79	68	17	-7	5	8
Half Nuclear/Fossil Base Load	5243	526	80	69	2	-5	3	8
9% Escalation								
All Nuclear Base Load	7418	409	79	67	16	-7	5	9
Half Nuclear/Fossil Base Load	5243	525	80	69	2	-4	3	9

Exhibit 11

1980-2020

Revenue Requirement Summary
All Nuclear Base Scenario
(1980 Present Worth Dollars x 10^3)

	9% Distillate Escalation Total Present Value
Base Production Costs	183,933,800
Capital Costs	1,056,800
Total	184,990,600
CAES Production Costs	183,661,400
Capital Costs	579,800
Total	184,241,200
Savings	749,400
Percentage	0.41%

	12% Distillate Escalation
Base Production Costs	214,200,500
Capital Costs	1,056,800
Total	215,257,300
CAES Production Costs	214,558,100
Capital Costs	579,800
Total	215,137,900
Savings	119,400
Percentage	0.06%

Exhibit 12

1980-2020

Revenue Requirement Summary
50% Coal/50% Nuclear Base Scenario
(1980 Present Worth Dollars x 10^3)

	9% Distillate Escalation Total Present Value
Base Production Costs	227,268,600
Capital Costs	1,056,800
Total	228,325,400
CAES Production Costs	229,002,500
Capital Costs	579,800
Total	229,582,200
Savings	(1,256,800)
Percentage	(.55%)

	12% Distillate Escalation
Base Production Costs	265,668,600
Capital Costs	1,056,800
Total	266,725,400
CAES Production Costs	268,269,000
Capital Costs	579,800
Total	268,848,800
Savings	(2,123,400)
Percentage	(.80%)

Appendix

Storage Requirements CAES Installations

The storage requirements for a Compressed Air Energy Storage (CAES) installation (defined as the number of hours of generation at rated capacity) is dependent on one or more of the following criteria:

1. Storage required to assure the installation is considered dependable at its design capacity rating.

2. Storage required to realize the economic benefits derived from shifting low cost "off peak" energy to "on peak" use.

3. Storage required to provide sufficient operating flexibility to allow dispatch of the installation "today" when confronted with the uncertainty of the system need for CAES generation "tomorrow".

The number of hours of "storage required" is a function of the CAES generate-to-recharge ratio, the particular utility system load shape and the characteristics of the generation system (generation mix, fuel type, etc.).

The storage sizing methodology developed for this project is designed to properly size the storage of a CAES project so that the sequential daily energy requirement is supported and the project capacity is dependable. The methodology is applied to a 1160 MW (four

290 MW units) CAES project which is installed on the hypothetical system described in the report. The existing composite system chronological load data used in this analysis was modified to reflect projected monthly peaks and energies for the years 1990 and 1995.

Storage Sizing Methodology

A methodology to determine the adequate storage for a CAES installation is presented. This methodology is comprised of the following independent evaluations:

1. Dependable Capacity Evaluation
2. Energy Shifting Evaluation
3. Uncertainty Evaluation

From the results of these evaluations, the CAES storage requirement necessary to assure the proper integration of a CAES installation into a given utility system is determined.

Dependable Capacity Evaluation - The amount of storage required to provide dependable facility capacity is analyzed with a technique referred to as the "weekly cutoff level" method. This technique, as illustrated in Figure 1, assumes that the weekly generation requirements for all capacity sources available to a system are a function of the weekly peak load. The technique further assumes that the only constraints to recharging the storage of a CAES installation are the physical equipment limitations and the availability of recharging energy. In situations where alternative sources of generation are not available, economic dispatch is not considered a constraint since capacity needs are independent of economics. CAES is assumed to provide rated capacity for each week examined. Additional aspects of the technique that are applicable to each week examined are outlined below:

- Aquifier storage will be fully charged by the time generation is required on Monday.

- Existing energy limited facilities have priority with regard to recharging energy and peak load generation use.

- The peak day of the week is used to establish the weekly capacity level that separates the energy supplied by limited energy facilities from the energy supplied from the unlimited energy plants, thus determining the daily generation requirements for the two types of facilities. The CAES capacity is always utilized at full-rated capacity on the peak day of the week. On other days it may not be fully utilized.

- Capacity which is not energy limited is assumed to be available for the entire week at the level established on the peak day.

For each week evaluated, weekly cutoff levels establish the daily generation requirements and the amount of total recharging energy available. Furthermore, by assigning priority to existing limited energy facilities this technique assures that existing limited energy capacity is utilized as designed, thus preventing the CAES capacity from being designed to serve the same system load characteristics that prior limited energy capacity was installed to serve. This technique evaluates the storage requirements necessary to support the sequential daily CAES generation requirements (allowing for corresponding daily recharging) as part of the overall weekly evaluation thus assuring that the resulting storage requirement will support the CAES generation needed for each day of the week.

Energy-Shifting Evaluation

A major function of the CAES installation is the shifting of lower cost "off peak" generated energy to serve "on peak" load. Although the economics of the system will determine the actual daily use of the CAES installation, the storage must be sized to accommodate reasonable expected daily duty cycles. These daily duty cycles are expressed in hours per day the CAES installation is assumed to serve load. Although the amount of energy generated by the installation will vary from day to day depending on the system load shape, the number of hours the installation provides energy to serve load is constant (i.e., the duty cycle). Likewise the energy available for recharging will vary from day to day. Even though the CAES generation and recharging values are determined on a daily basis, the evaluation assumes that a weekly operating cycle will be utilized when load conditions dictate.

For the evaluation, the CAES installation is assigned a daily duty cycle and a period of weekly operation (i.e., Monday through Friday). The storage requirement necessary to support each week's CAES generation at the assigned duty cycle and period of operation is determined. From the results of this determination, the storage requirement adequate to support the expected annual use of the CAES installation can be selected.

Operating Uncertainty Evaluation

The capacity and energy evaluations are based on an assumed perfect knowledge of future events. Furthermore, these evaluations are applied to "typical" weekly profiles which do not simulate abnormal and unusual situations that do occur in actual operating circumstances. Because the operator of a CAES installation does not have perfect knowledge of future events and since unusual situations do occur, the sizing of the CAES storage must accommodate uncertainties associated with "real world" operation. A formula to assist in quantifying the size of the CAES storage that will make allowances for the operational uncertainties is presented below:

FLD = Days of full load generation

FLH = Hours per day of full load generation

RCD = Days of recharging is available

RCH = Hours per recharged day that recharging energy is available

K = Recharge/generation ratio

Storage = (FLD x FLH) - (RCD x RCH) x K

Where: RCD = FLD -1

The selection of values for the variables in this test must be based on operating experience and knowledge of system conditions that create such operating uncertainties.

Storage Sizing for 1160 MW CAES Project

Assumptions

The following assumptions are used to determine the storage size of a 1160 MW CAES project:

1. CAES rated capacity is 1160 MW. This capacity is comprised of four 290 MW modular units.

2. The total electrical power required to obtain maximum hourly compression is equal to 205 MW per module, which in turn will support 290 MW per module (i.e., a 290 MW CAES unit will require the quantity of air equal to the electrical equivalent of one hour's compression at 205 MW of electrical input).

3. Output to input ratio is 1.41 (i.e., for every unit of electrical input energy, the CAES installation can, with the aid of oil or synthetic fuel, output 1.41 units of electical energy.)

4. Storage is expressed in hours of installed capacity at full load generation.

5. The load curve for the hypothetical system in the evaluation is modified to assure that monthly peak loads and energies are consistent with projected values.

6. The evaluation is for years 1990 and 1995.

7. Weekly cycle operation is assumed possible if needed so that weekend recharging may be utilized.

8. Recharge/generate ratio (K) equals 1.0.

Capacity Evaluation

The storage required to underwrite the rated capacity of a CAES installation is influenced by many factors not the least of which is the percentage of the total installed system capacity the installation represents. Since the 1160 MW of CAES capacity is less than 3% of the total installed capacity on the hypothetical system (and decreasing each year), the storage requirements resulting from applying the weekly cutoff level technique to each week in the study period (See Graph I) are not surprising. These results indicate that rated capacity is assured with 100% confidence with storage of 9 hours, however, the ultimate storage size is dependent on more than one criteria as the following evaluation will demonstrate.

Energy Evaluation

The storage to support the weekly CAES generation associated with duty cycles that range from 14 to 16 hours per day is determined. The period of generation for these daily cycles is defined as Monday through Friday. Results of the determination are illustrated by Graph II (1990) and Graph III (1995). The PROMOD production costing analysis indicates that a 20% annual capacity factor is expected for CAES operation in the early to mid-1990's. Operating experience of intermediate load plants indicates that a daily duty cycle of 15 hours per day is a reasonable expectation. On this basis, a storage size of about 26 hours is indicated.

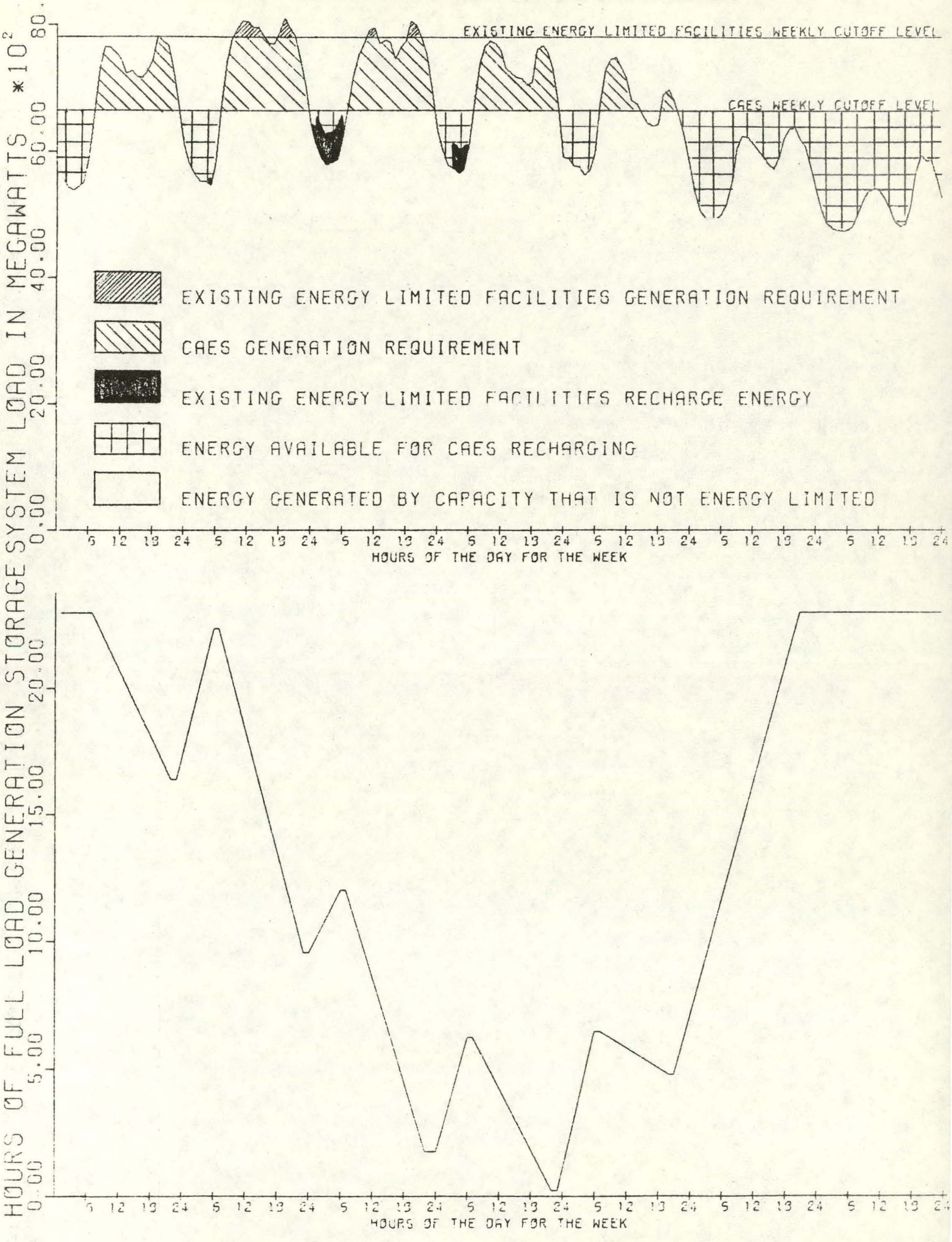
Uncertainty Evaluation

The results of the application of the formula are tabulated in Table 1. These results indicate that a storage size of 36 hours of generation at rated capacity will provide adequate flexibility to assure continued operation for a wide range of variable combinations.

Conclusion

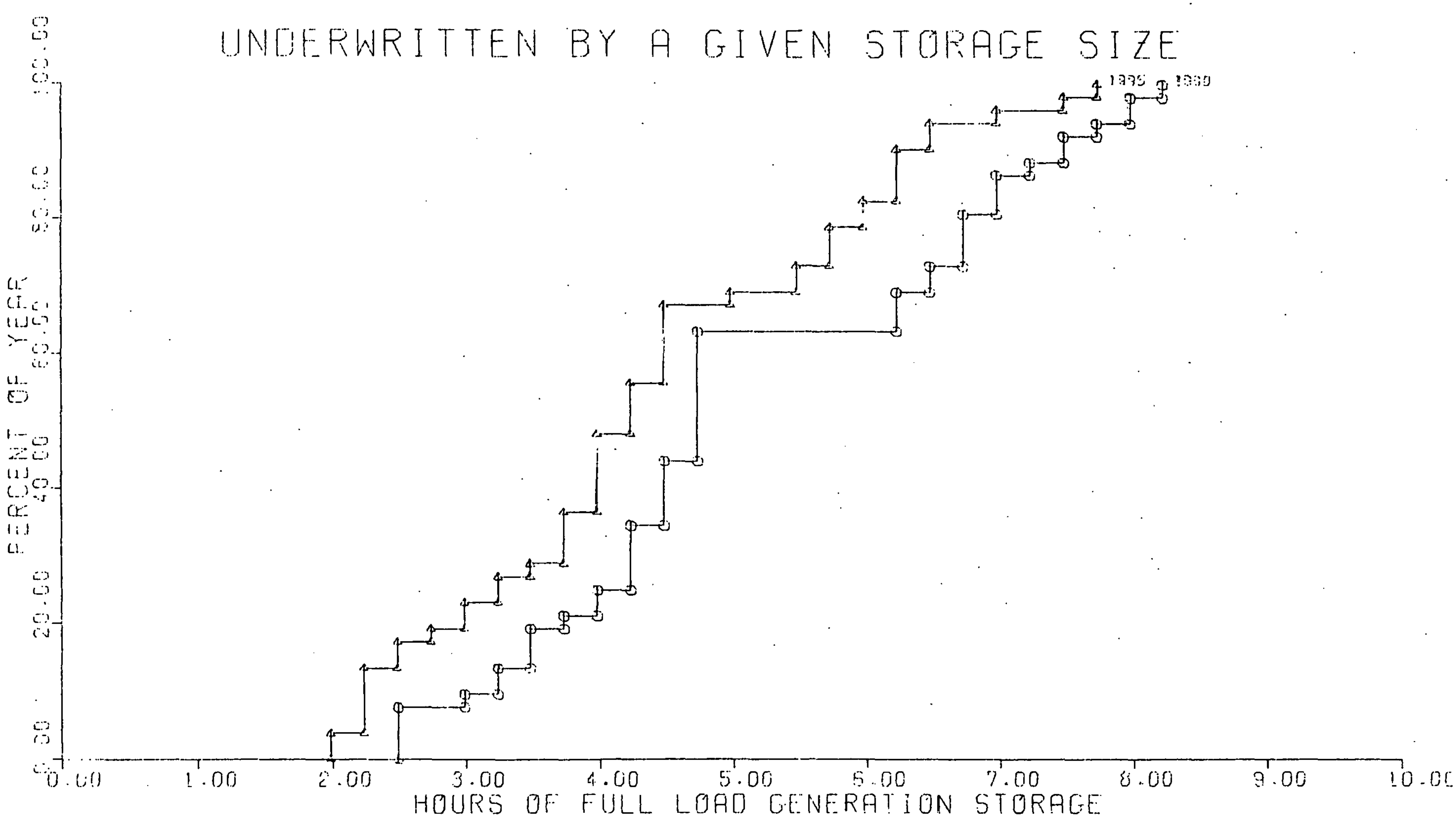
Based on this analysis the storage size of the 1160 MW CAES project should be between 26 to 36 hours of rated capacity generation. This is based on the results of the three independent evaluations that comprise the storage sizing methodology. A storage capability in this range will provide adequate storage energy to satisfy the operational requirements of this CAES installation.

Illustration of Weekly Cutoff Level Methodology



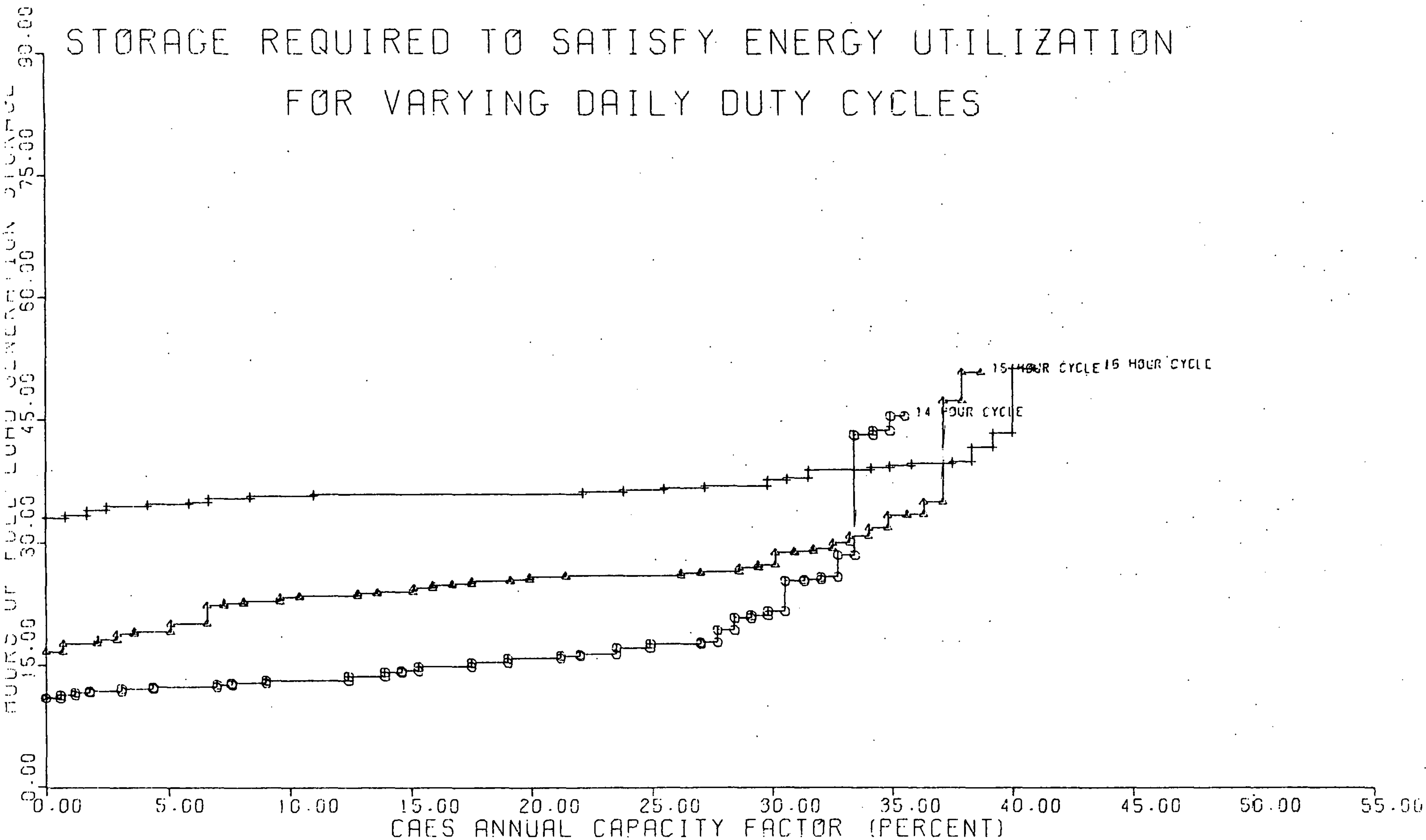
F I G U R E I.

RESULTS OF THE CAPACITY EVALUATION
ILLUSTRATING THE PERCENT OF THE YEAR
THE RATED CAES CAPACITY IS DEPENDABLY
UNDERWRITTEN BY A GIVEN STORAGE SIZE



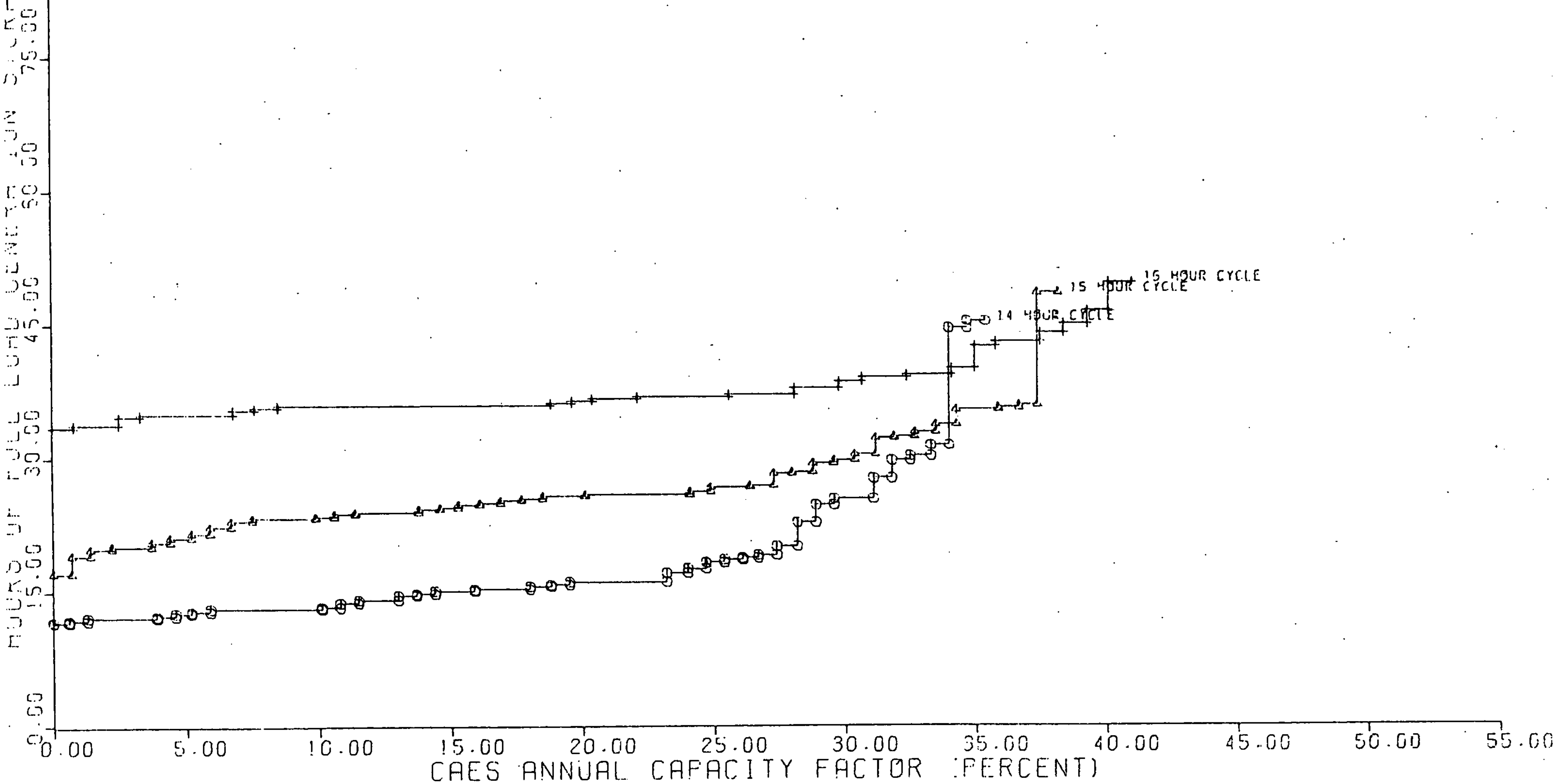
GRAPH I.

RESULTS OF THE 1990 EVALUATION TO DETERMINE
STORAGE REQUIRED TO SATISFY ENERGY UTILIZATION
FOR VARYING DAILY DUTY CYCLES



GRAPH II.

RESULTS OF THE 1995 EVALUATION TO DETERMINE STORAGE REQUIRED TO SATISFY ENERGY UTILIZATION FOR VARYING DAILY DUTY CYCLES



G R A P H III.

Uncertainty Evaluation

Hours-Daily Generation	4 Days Uncertainty Hours-Daily Recharge 9	8	7	3 Days Uncertainty Hours-Daily Recharge 9	8	7
16	-	40.0	43.0	-	32.0	34.0
15	33.0	36.0	39.0	27.0	29.0	31.0
14	29.0	32.0	35.0	24.0	26.0	28.0
13	25.0	28.0	31.0	21.0	23.0	25.0

Total Hours of Storage
(In Hours of Generation at Rate Capacity)

Table I