Lh. 163

ANL/ES-64

ANL/ES-64

284
6-15-78

A PARAMETRIC STUDY OF TURBINE SYSTEMS FOR COMPRESSED AIR ENERGY STORAGE PLANTS

Final Report for FY 1977

by

Choong S. Kim and George T. Kartsounes

MASTER

ARGONNE NATIONAL LABORATORY

ANL

EES

ENERGY AND ENVIRONMENTAL SYSTEMS DIVISION

Prepared for the U. S. DEPARTMENT OF ENERGY
under Contract W-31-109-Eng-38

DISCLAIMER

Printed in the United States of America
Available from
National Technical Information Service
U. S. Department of Commerce
5285 Port Royal Road
Springfield, Virginia 22161
Price: Printed Copy $4.50; Microfiche $2.25

Distribution Categories:
Energy Storage--Thermal (UC-94a)
Energy Storage--Mechanical (UC-94b)

ANL/ES-64

ARGONNE NATIONAL LABORATORY
9700 South Cass Avenue
Argonne, Illinois 60439

A PARAMETRIC STUDY OF TURBINE SYSTEMS FOR COMPRESSED AIR ENERGY STORAGE PLANTS

Final Report for FY 1977

by

Choong S. Kim

and

George T. Kartsounes
Energy and Environmental Systems Division

April 1978

Prepared for the

Division of Energy Storage Systems
U. S. Department of Energy

THIS PAGE

WAS INTENTIONALLY

LEFT BLANK

TABLE OF CONTENTS

LIST OF FIGURES

LIST OF FIGURES (Cont'd)

LIST OF TABLES

THIS PAGE

WAS INTENTIONALLY

LEFT BLANK

THIS PAGE

WAS INTENTIONALLY

LEFT BLANK

NOMENCLATURE

NOMENCLATURE

a	$(k - 1)/k$
c_p	Specific heat at constant pressure
c_v	Specific heat at constant volume
$\dot{E}'_c$	Specific compression rate as defined in Eq. (C.11)
f_1	Fuel-air ratio for combustor 1
f_2	Fuel-air ratio for combustor 2
f_2'	$f_2/(1 + f_1 + r_{c_1})$
h	Enthalpy
k	Specific heat ratio, c_p/c_v
$\dot{m}$	Mass flow rate
$\dot{m}'_a$	Specific air flow rate
p	Pressure
p_{atm}	Atmospheric pressure
$\dot{Q}'$	Specific heat rate
R	Gas constant
r	Cooling air-turbine air ratio
T	Temperature
V'_s	Specific storage volume
$\dot{W}$	Power input or output
$\dot{W}_{comp}$	$\dot{W}_{BC} + \dot{W}_{LPC}$
$\dot{W}_{fuel}$	Fuel energy
$\dot{W}_{out}$	$\dot{W}_{LGT} + \dot{W}_{HGT}$
ε	Recuperator effectiveness
η	Efficiency
$\eta_{overall}$	Overall plant efficiency
ΔH_L	Lower heating value of fuel (21,500 Btu/lb)
ΔH_H	Off-peak heat rate (10,400 Btu/kWh)
Δp_f	Pressure loss due to friction
Δp_s	Pressure loss due to static head

<u>Subscripts</u>:

BC	booster compressor
C_1	combustor 1
C_2	combustor 2
c_1	cooling air for HGT
c_2	cooling air for LGT
ca	cavern
f_1	fuel to combustor 1
f_2	fuel to combustor 2
HGT	high-pressure gas turbine
LGT	low-pressure gas turbine
LPC	low-pressure compressor
s	isentropic process
1-7	correspond to Fig. 1.3
11-15	correspond to Fig. 1.2

A PARAMETRIC STUDY OF TURBINE SYSTEMS FOR COMPRESSED AIR ENERGY STORAGE PLANTS

Final Report for FY 1977

by

Choong S. Kim

and

George T. Kartsounes

ABSTRACT

A parametric study of possible turbine systems for compressed air energy storage (CAES) plants is made. The plant considered is divided into four subsystems: a turbine system, compressor system, reversible motor/generator, and an underground air storage reservoir. The turbine system comprises a high-pressure gas turbine, a low-pressure gas turbine, two combustors, and a recuperator. The compressor system consists of a low-pressure compressor, a booster compressor, an intercooler, and an aftercooler. A water-compensated mined cavern is the underground reservoir.

Thermodynamic analyses of subsystem components are made, and plant performance is evaluated. The results are given in terms of the five parameters: specific air flow rate, specific storage volume, specific heat rate, specific compression rate, and overall plant efficiency. The effects on plant performance of design parameters such as inlet gas temperature and pressure to each turbine were analyzed. Also considered is the effect of using cooling air for turbine blades and vanes.

SUMMARY

Effects of four turbine system design parameters on the performance of a CAES plant were determined: inlet pressures and inlet gas temperatures to the high-pressure gas turbine (HGT) and the low-pressure gas turbine (LGT). Also studied was the effect on plant performance of turbine cooling air, which is required for higher inlet gas temperature to the turbines. The results are given in terms of five performance parameters: specific air flow rate (lb/kWh), specific storage volume (ft^3/kWh), specific heat rate (Btu/kWh), specific compression rate (Btu/kWh), and overall plant efficiency. The four specific parameters are based on a kilowatt of power generated by the turbine

system. Overall plant efficiency is defined as the total power output of the turbines divided by the sum of the fuel energy input rate to the combustors and off-peak power input to the compressors.

For turbines that do not require cooling air, the following performance trends were observed in terms of the specific parameters and overall plant efficiency:

1. As the inlet gas temperatures increase, the heat rate and overall efficiency also increase, and the air flow rate, storage volume, and compression rate decrease.
2. The heat rate is influenced strongly by the inlet temperature to the low-pressure turbine and only weakly by the temperature to the high-pressure turbine.
3. As the inlet pressure to the high-pressure turbine increases, the compression rate increases slightly and the specific air flow and heat rates as well as the overall efficiency decrease slightly, whereas the storage volume decreases significantly.
4. The outlet pressure from the high-pressure turbine has a minor effect on performance.

Thus, the highest inlet gas temperatures possible, without requiring cooling air, should be used. The inlet pressure to the high-pressure turbine, which is determined by the storage pressure, should be as high as possible.

For turbines that require cooling air, the performance trends observed were as follows:

1. Uncooled high-pressure turbine and cooled low-pressure turbine:
   - As the flow rate of cooling air increases, the specific air flow rate, storage volume, and compression rate increase. These values are less than for equivalent uncooled turbines when the ratio of LGT cooling-air flow to main-air flow (r_{c_2}) is less than 0.2.
   - The specific heat rate is always greater than for the equivalent uncooled turbine.
   - The overall efficiency decreases with cooling-air flow. For r_{c_2} < ~0.1, it is greater than for uncooled turbines; but for r_{c_2} > ~0.1, it is less.
2. Both turbines air-cooled:
   - As the ratio of HGT cooling-air flow to main-air flow (r_{c_1}) increases, the specific air flow, storage volume, and compression rate decrease when the ratio of LGT cooling-air flow to main-air flow (r_{c_2}) is greater than ~0.2.

These values increase for $r_{c_2} < \sim 0.2$ but are less than for an equivalent uncooled turbine for most ratios considered.

- As r_{c_1} increases, the heat rate increases for $r_{c_2} < \sim 0.45$ but decreases for $r_{c_2} > \sim 0.45$. This rate is always greater than for the equivalent uncooled turbine.
- As r_{c_1} increases, the overall efficiency decreases for $r_{c_2} < \sim 0.3$ but increases for $r_{c_2} > \sim 0.3$. For most cooling-air ratios, the overall efficiency is greater than the equivalent uncooled turbine when $T_3 = T_5 = 2400°F$, but it is less when $T_3 = T_5 = 2000°F$.

The above trends indicate that the amount of cooling air has a major effect on the performance parameters. Cooling air can either increase or decrease the performance parameters as compared to equivalent uncooled turbines. However, using high inlet gas temperatures, which require cooling air, always increases the use of premium fuel.

1 BACKGROUND

Compressed air energy storage (CAES) is a near-term technology for the load leveling and peak shaving strategies being considered by electric utilities. Assessments of the technical and economic feasibility of this storage system indicate that it is economically competitive with conventional gas-turbine peaker units.

A CAES plant comprises four subsystems (see Fig. 1.1): a turbine system, compressor system, reversible motor/generator, and an underground air-storage reservoir. The CAES concept is based on a split Brayton cycle with an accompanying underground air reservoir. During periods of off-peak power demand, air is compressed with base-plant power and stored in the underground reservoir. The air is discharged, for power generation, through a combustion turbine during the peak demand period.

1.1 AIR STORAGE RESERVOIR

Because the storage reservoir is usually the most costly single component in a CAES plant, its volume is a sensitive design parameter. The volume required is affected by storage pressure and temperature, power level, and generation time; and (less obviously) by the reservoir type, air quantity required by the turbine system, and pressure ranges permitted by the turbomachinery (turbines and compressors).

Compressed air can be stored underground in caverns (natural or mined) or in the pore space of porous rock formations. Caverns may be mined by conventional mining, nuclear explosives, or solution-mining, as for salt structures. Many of the porous rock formations are aquifers (i.e., they contain water). To use an aquifer for gas storage, as has been done for many years in the natural gas industry, the water must be displaced.

1.2 COMPRESSOR SYSTEM

The compressor system comprises an axial compressor, centrifugal booster compressors, intercoolers, and an aftercooler (see Fig. 1.2). Intercooling is required to operate the compressors within temperature limits tolerable for standard materials (about 600°F maximum air temperature). An aftercooler is used to cool the air to avoid possible thermal-stress damage to the storage

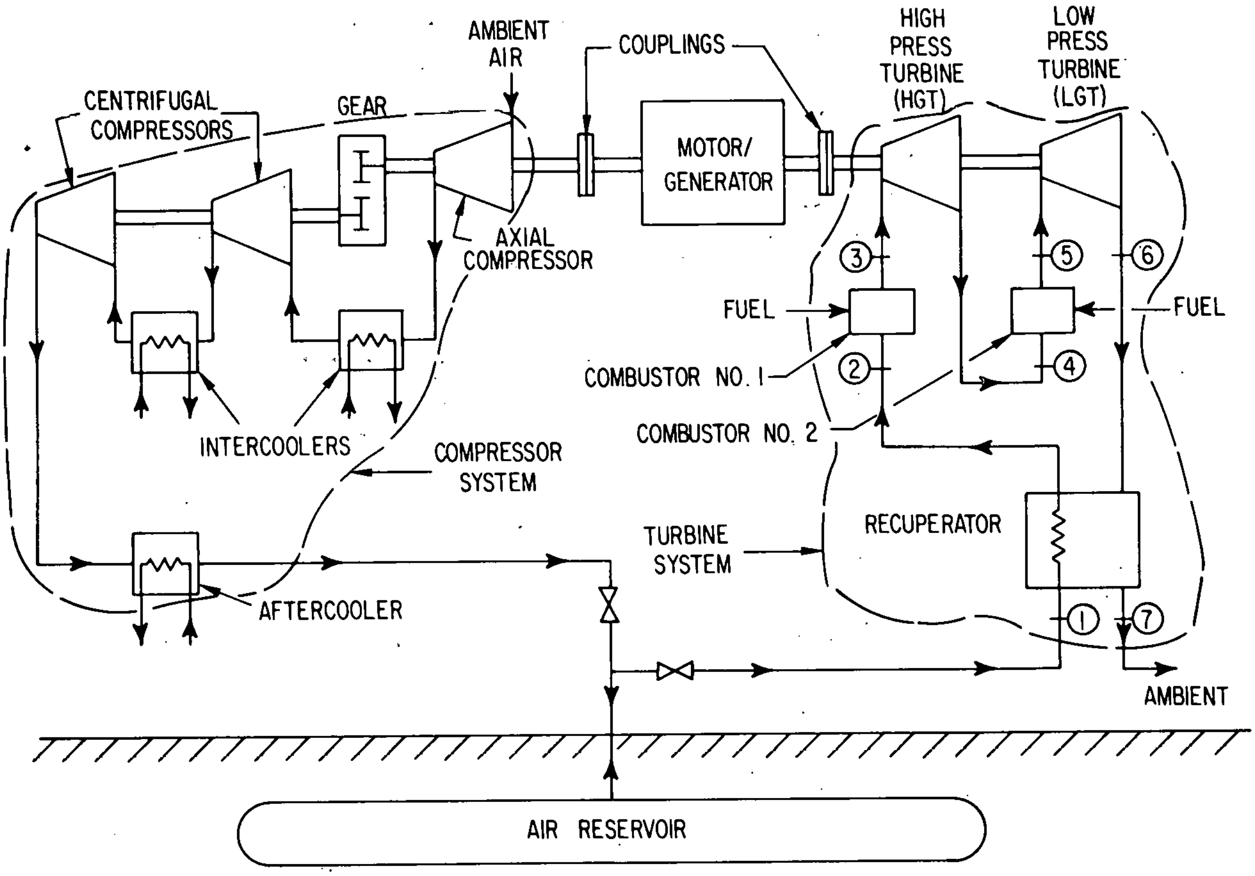


Fig. 1.1. Schematic Diagram of CAES Plant

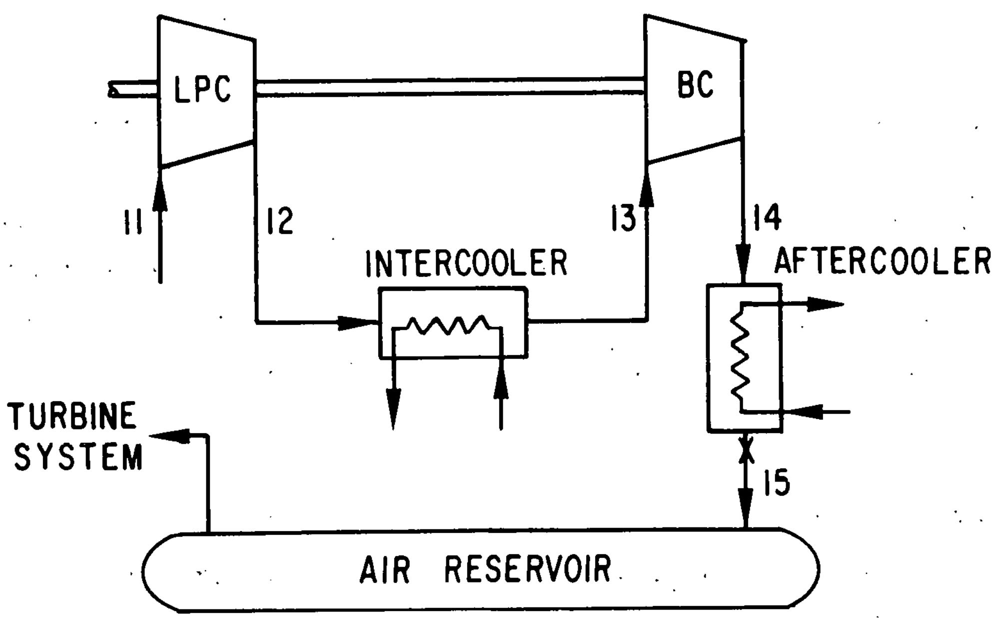


Fig. 1.2. Compressor System in a CAES Plant

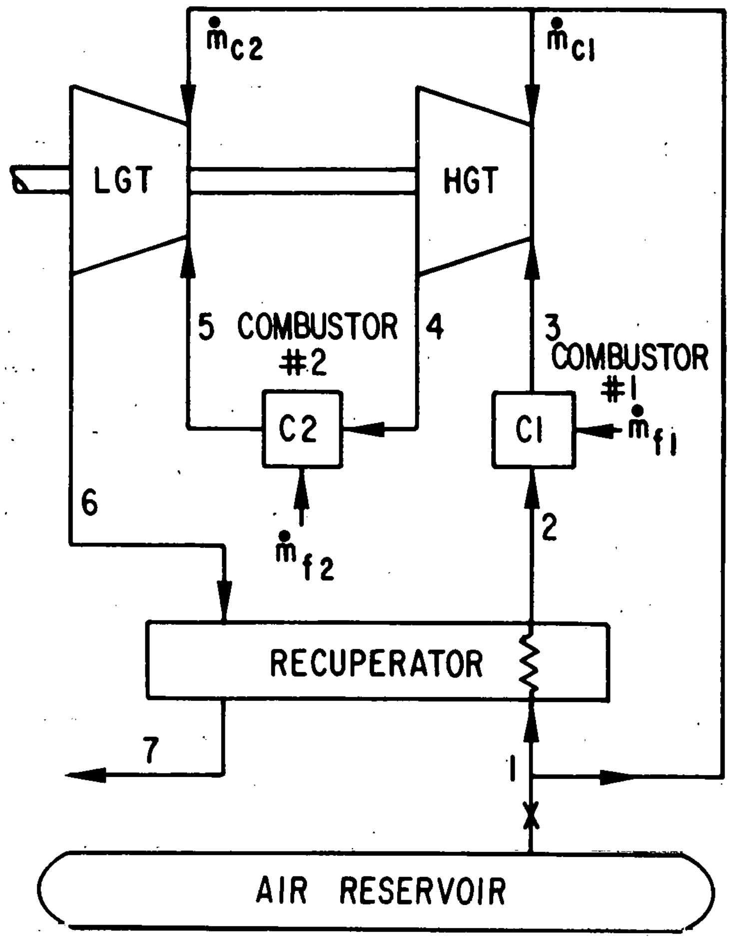


Fig. 1.3. Turbine System in a CAES Plant

reservoir. The axial compressors are units modified from conventional gas-turbine peakers. Several commercially available axial compressors have a nominal pressure ratio of 11:1, and a 16:1 unit is available that comprises two 4:1 compressors with intermediate intercooling. One or more centrifugal booster compressors, in series or in parallel, are used to raise the pressure to the desired storage level. These compressors are directly connected or geared to the main shaft of the axial compressor.

1.3 TURBINE SYSTEM

The turbine system consists of a low-pressure turbine (LGT), a high-pressure turbine (HGT), two combustors, and a recuperator (see Fig. 1.3). The LGT is a turbine modified from a conventional gas-turbine peaker unit.

Several commercially available turbines have a nominal pressure ratio of 11:1, and one manufacturer sells a 16:1 unit. For the LGT, two different designs are available: one, with internal air cooling, to operate at inlet gas temperatures of 1600°-2400°F; and the other is for a simpler, uncooled turbine, for gas temperatures below about 1600°F.

For proposed CAES plants, the HGT is a modified steam turbine operating at gas temperatures of about 1000°F. Optimized designs for compressed-air turbines, which operate at high temperatures, are being investigated. The required pressure ratio across the HGT is determined by the reservoir storage pressure and the inlet pressure to the LGT.

The combustors can be designs modified from conventional gas-turbine peaker units. Preliminary studies indicate that recuperators can be designed that are economically feasible for CAES application; they will differ from a conventional gas-turbine peaker unit because of the high-pressure air leaving the reservoir. Recuperators for conventional peakers are designed and built on a one-of-a-kind basis, and application of recuperators to large-scale peaking units (having the capacity of proposed CAES plants) is relatively recent. In the summer of 1974, Philadelphia Electric Company put into service the first large-scale installation of recuperators to be subjected to many starting cycles, as is expected in CAES plants.[1]

1.4 MOTOR/GENERATOR

The motor/generators in most CAES plants do not require special design features. Several U.S. and foreign manufacturers market synchronous units, operating at 3600 rpm, which could be used. For example, the Brown Boveri Corporation sells a nominal 200-MW motor/generator rated at 225 MVA, 17 kV, with hydrogen cooling and static excitation.

1.5 PERFORMANCE PARAMETERS

The performance of a CAES plant can be characterized in terms of five parameters: specific air flow, specific heat rate, specific storage volume, specific compression rate, and overall plant efficiency.

Specific air flow is the mass flow rate of air supplied to the turbine system per kilowatt power generated (lb of air/kWh). It is the major factor in determining the size of the turbines, compressors, and air-storage reservoir.

Specific heat rate is directly proportional to fuel consumption and is equal to the product of specific fuel consumption and the lower heating value of the fuel (Btu/kWh). It therefore affects the operating cost of the turbines.

Specific storage volume, the volume of reservoir required per kilowatt of power generated (ft^3/kWh), is dependent on the specific air flow rate and the temperature and pressure of stored air.

Specific compression rate is the energy equivalent of the power supplied to the compressors per kilowatt of power generated (Btu/kWh). This parameter is the amount of off-peak energy required to operate the compressors.

The overall plant efficiency is equal to the total energy output from the turbines divided by the sum of the energy input from the fuel and off-peak energy to the compressor system. This efficiency defines the ability of a CAES plant to store off-peak energy and return energy to the power grid during peak hours.

1.6 DESIGN PARAMETERS

A CAES plant can be designed with many combinations of surface equipment. The design of the compressor system using commercial equipment is straightforward and depends on the required reservoir pressure and flow rate needed for the turbine system for which a variety of equipment options are possible.

Since the objective of a CAES plant is to generate peak power, the turbine system represents the *heart* of the plant; and studies of CAES plants have focused on this system.[2] The performance of the turbine system, and of the complete plant, is a function of the inlet pressure to the HGT (the reservoir pressure), the intermediate pressure between the turbines, and the inlet gas temperature to each turbine. In addition, the selected inlet gas temperatures dictate whether internal air cooling is necessary.

1.7 SCOPE OF REPORT

The results of a parametric study of turbine systems for CAES plants are presented. The effect of the design parameters on overall plant performance were considered, with particular emphasis on the effect of turbine cooling air required for high inlet gas temperatures to the turbines on plant performance.

The report is divided into three chapters and an appendix. In Chapter 2, a thermodynamic analysis of each subsystem and the complete plant is presented. The appendix contains the detailed development of the equations. Chapter 3 presents and discusses the results of the study. Given first is the performance of the plant without the use of turbine cooling air and next the effect of turbine cooling air. The conclusions and recommendations resulting from this study are given in Chapter 4.

2 THERMODYNAMIC ANALYSIS

A thermodynamic analysis was carried out on each of the four systems of a CAES plant, and the results were combined to evaluate the performance of the plant. Parameters considered include: inlet gas temperature to the high-pressure gas turbine (HGT) and low-pressure gas turbine (LGT); inlet and outlet gas pressures of the HGT; and the amount of cooling air.

The following assumptions were made:

- The gas flow is steady and the state at each point in the control volume does not vary with time.
- Difference in kinetic energy and potential energy across each component are negligible.
- Heat loss to the ambient from each component is negligible (adiabatic control volume).
- The gas mixture behaves as a perfect gas.
- Natural gas is the fuel.

2.1 TURBINE SYSTEM

The turbine system chosen consists of two turbines (HGT and LGT), two combustors, and a recuperator (Fig. 1.3). The selection of the turbine system evolved from the results of a previous study,[2] which included comparison of several possible turbine systems for CAES.

In analyzing the performance of the turbine system, the following values of system parameters were considered:

Turbine efficiencies: $\eta_{LGT} = \eta_{HGT} = 0.85$,

Combustor efficiencies: $\eta_{C_1} = \eta_{C_2} = 1.0$,

Recuperator effectiveness: $\varepsilon = 0.8$,

Temperatures: $T_1 = 120°F$

$T_3 = 1000°, 1600°, 2000°, 2400°F$

$T_5 = 1600°, 2000°, 2400°F$

Pressures: $P_3 = 50, 70, 100$ atm,

$P_4 = 11, 16, 30$ atm, and

Power output of LGT: $\dot{W}_{LGT} = 200$ MW.

Subscripts given in the above parameters correspond to the components or stations in Fig. 1.3. The efficiencies of turbines and combustors are based on the state-of-the-art values of available equipment. Recuperator effectiveness is a function of the heat exchanger specifications. Because the temperature of the inlet gas to the turbines must be kept low enough to avoid thermal damage to the turbine blades and vanes, cooling air is required for higher inlet gas temperatures. Temperature T_1 is fixed by the condition of the underground air-storage cavern; and pressure p_3 depends on the pressure drops across the recuperator and the first combustor, and pressure of the cavern air. The thermodynamic analysis of the turbine system is given in detail in the Appendix.

Governing equations were written for each component. Mass-balance equations were formulated by considering addition of fuel to the combustors and cooling air to the turbines. Instead of momentum equations, equations that represent the pressure variations across each component were used. Energy-balance equations were written for the recuperator, the turbines, and the combustors. The definitions of recuperator effectiveness and thermal efficiency of turbines were also used.

The equations for mass, momentum, and energy balances were solved by use of a simulation computer program with which the following were calculated: turbine outlet temperatures, T_4 and T_6; recuperator outlet temperatures, T_2 and T_7; and fuel-air ratios for the combustors, f_1 and f_2. The rate of air flow from the underground storage reservoir is then obtained from the energy-balance equation for the LGT. From these results, the power output of HGT is calculated. Finally, the total power output of the turbine system, specific air flow rate, and specific heat rate are calculated.

2.2 UNDERGROUND AIR STORAGE SYSTEM

The underground air storage reservoir considered is a water-compensated cavern. Therefore, the pressure variation in the cavern during the operating cycle is negligible. Two factors influence the pressure variation in the air shaft: static head and friction. Variation in static-head pressure is a function of the air pressure in the cavern, and variation in frictional pressure is a function of air flow rate.

Air pressure in the storage cavern was calculated from the outlet air pressure of the cavern, with pressure variation due to the change in the static head and that due to friction being considered. The specific storage volume could then be obtained from the cavern pressure and the specific air flow rate. (See Appendix for the calculational procedures.)

2.3 COMPRESSOR SYSTEM

The study was extended to the compressor system in order to complete the analysis of the CAES plant. The compressor system selected is composed of two compressors (low-pressure, LPC, and booster, BC), and intercooler and an aftercooler, as shown in Fig. 1.2. The following parameters were assumed to be known or specified:

Adiabatic efficiency of compressors: $\eta_{LPC} = \eta_{BC} = 0.85$;
Temperatures: $T_{11} = 77°F$, $T_{13} = 100°F$, $T_{15} = 120°F$; and
Pressures: $p_{11} = P_{atm}$, $p_{12} = 16$ atm.

The required output includes the compressor outlet temperatures (T_{12} and T_{14}), from which the power input to the compressors may be obtained. (See Appendix for the detailed analysis.)

The rate of air flow into the compressor system was based on flow from the storage cavern into the turbine system, with the following being considered: loss of air in the cavern; pressure drops across the intercooler and aftercooler; and the frictional loss in the cavern shaft. The compressor outlet temperatures could then be calculated by use of the adiabatic efficiencies of the compressors. By using these results, the power inputs to the compressors could be calculated with the energy-balance equations. Specific compression energy was then evaluated from the input of the compressor system and the output of the turbine system.

2.4 COMPRESSED AIR ENERGY STORAGE PLANT

By using the results from the analysis of the turbine system, underground storage cavern, and compressor system, the overall performance of the CAES plant was evaluated. The overall plant efficiency ($\eta_{overall}$) was defined as,

$$\eta_{overall} = \frac{\dot{W}_{out}}{\dot{W}_{comp} + \dot{W}_{fuel}} , \tag{2.1}$$

where:

$\dot{W}_{out}$ = power output of the turbine system,
$\dot{W}_{comp}$ = power input to the compressor system,
$\dot{W}_{fuel}$ = rate of fuel supply to the combustors.

In the rate form of this equation, the charging time of the reservoir is assumed to equal the power-generation time of the turbine system.

3 RESULTS AND DISCUSSION

Results of this parametric study are presented in terms of the five performance parameters: specific air flow (lb/kWh), specific storage volume (ft^3/kWh), specific heat rate (Btu/kWh), specific compression rate (Btu/kWh), and overall plant efficiency; and are given as a function of turbine inlet temperatures (T_3 and T_5) and inlet and outlet pressures of HGT (p_3 and p_4). Given first are a set of results for which turbine-blade cooling is not considered. The effect of cooling air on results is discussed in Sec. 3.2.

3.1 PERFORMANCE WITHOUT CONSIDERING COOLING AIR

Specific air flow, the flow rate (lb/hr) of air coming out of the storage cavern per unit output of the system (kW), is directly proportional to the turbine and compressor sizes. Thus, it is important in determining the cost of the above-ground facility.

Specific storage volume, the required storage cavern volume per unit work output, is directly related to the cost of the underground facility for a CAES plant. This storage volume depends on the required specific air flow rate as well as on cavern conditions, such as pressure and temperature of stored air. Consequently, results for the storage volume show a trend similar to those for air flow. A plot of specific air flow and specific storage volume against HGT inlet temperatures at different LGT inlet temperatures (Fig. 3.1) shows that the air flow and volume decrease as inlet temperature of either HGT or LGT increases. Shown in Figs. 3.2 and 3.3 are the effects of HGT inlet pressure and outlet pressure, respectively, on the specific air flow and the specific storage volume. Smaller air flow and storage volume are required for a higher HGT inlet pressure, and the effect of HGT outlet pressure is almost negligible.

Specific heat rate is a measure of premium-fuel usage for the combustors per unit power output of the system. The effect of turbine inlet temperatures on the specific heat rate is given in Fig. 3.4: higher LGT inlet temperature requires more fuel and the heat rate decreases and then increases as the HGT inlet temperature increases. Figure 3.5 shows that the specific heat rate slowly decreases as the HGT inlet pressure increases. Figure 3.6

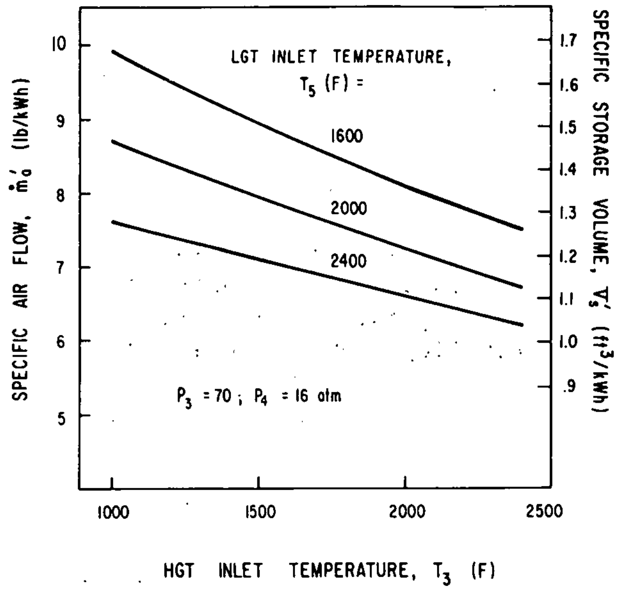


Fig. 3.1

Effect of Turbine Inlet Temperatures on Specific Air Flow and Specific Storage Volume (cooling air not considered)

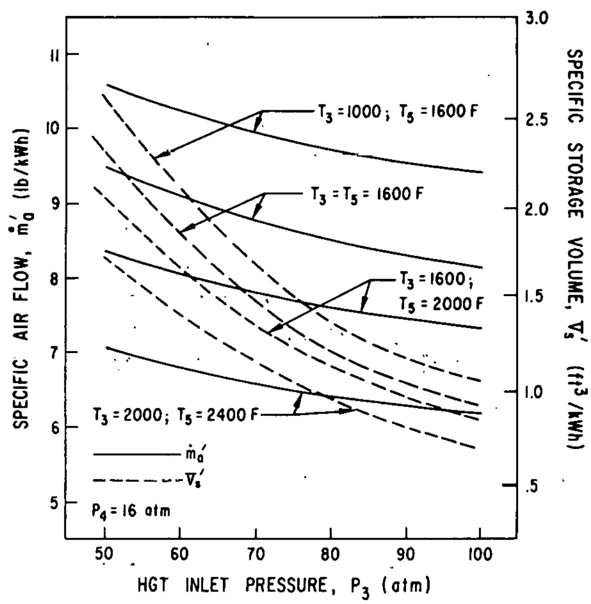


Fig. 3.2

Effect of HGT Inlet Pressure on Specific Air Flow and Specific Storage Volume (cooling air not considered)

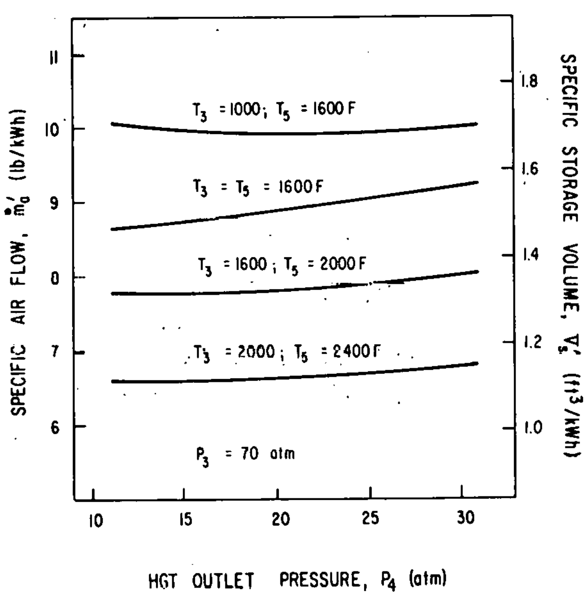


Fig. 3.3

Effect of HGT Outlet Pressure on Specific Air Flow and Specific Storage Volume (cooling air not considered)

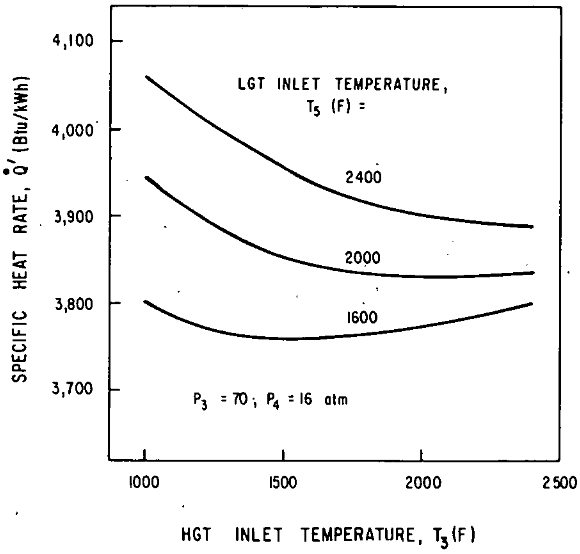


Fig. 3.4

Effect of Turbine Inlet Temperatures on Specific Heat Rate (cooling air not considered)

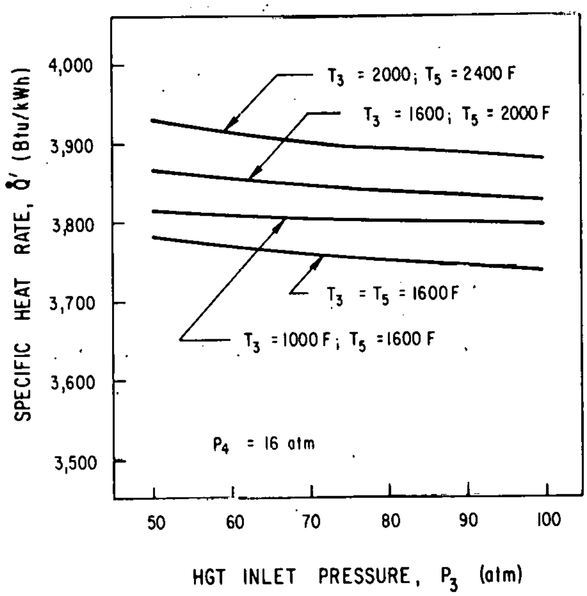


Fig. 3.5

Effect of HGT Inlet Pressure on Specific Heat Rate (cooling air not considered)

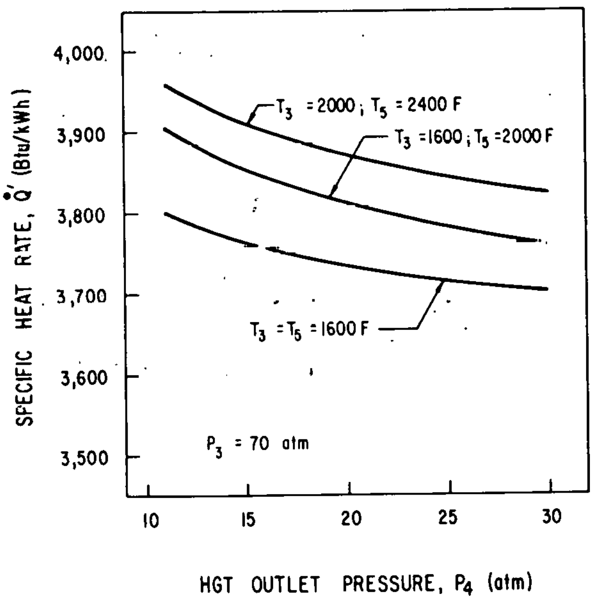


Fig. 3.6

Effect of HGT Outlet Pressure on Specific Heat Rate (cooling air not considered) shows that the specific heat rate decreases monotonically as HGT outlet pressure increases.

Specific compression rate, the fuel equivalent of the energy input to the compressors as defined in Eq. (C.11) of the Appendix, depends on the air flow and inlet and outlet temperatures of both compressors. Figure 3.7 shows that the compression rate decreases with increasing turbine inlet temperatures in a pattern similar to that for the air flow (Fig. 3.1). This similarity results because the compression rate is directly proportional to the air flow. A higher compression rate is required for higher HGT inlet pressure (Fig. 3.8). This trend is opposite to the relationship between specific air flow and HGT inlet pressure. The higher storage pressure induces higher compressor outlet temperatures, which require a higher compression rate. The HGT outlet pressure, as shown in Fig. 3.9, does not affect the specific compression rate appreciably.

The overall plant efficiency, the ratio of turbine power output to the sum of the power input to the compressors and the power equivalent of fuel energy, ranges from 0.48 to 0.58 for the conditions specified in this study. The effects on the overall plant efficiency are given in Figs. 3.10-3.12. Higher turbine inlet temperatures increase efficiencies (Fig. 3.10). However, for this set of results, the effect of cooling air, which will be required for high turbine inlet temperatures is neglected. The overall efficiency decreases with increasing HGT inlet pressure (Fig. 3.11), but the variation is considered insignificant. Figure 3.12 shows the effect of HGT outlet pressure on overall plant efficiency; the variation in plant efficiency differs from case to case according to turbine inlet temperatures, but the effect of HGT outlet pressure is insignificant.

3.2 EFFECT OF COOLING AIR

Turbine blades and vanes must be cooled to keep them within a temperature range to accommodate metallurgical limitations when high turbine inlet temperatures are to be used. Air cooling is the most common and practical method, although little detailed information on it appears in the open literature. Two data points were available. Giramonti et al.[3] considered a case for an inlet gas temperature to the turbine of 2000°F and a cooling-air turbine air ratio of 0.21 for the turbine pressure ratio of 14.2. Ayers and

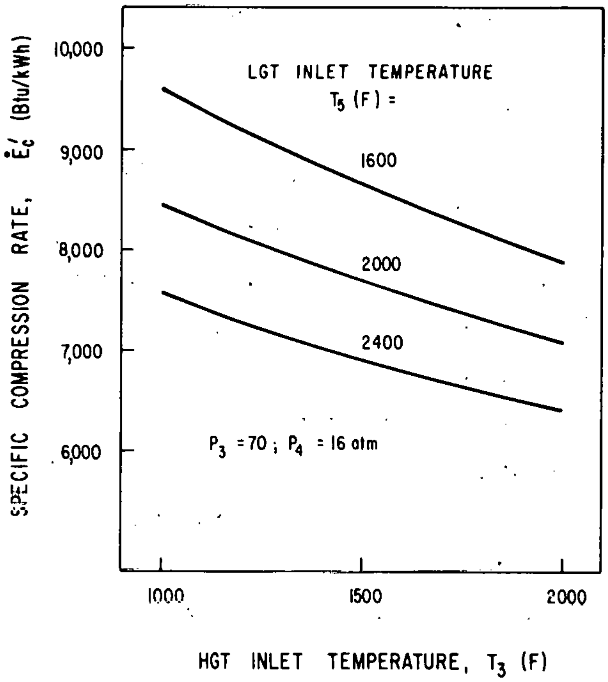


Fig. 3.7

Effect of Turbine
Inlet Temperatures on
Specific Compression Rate
(cooling air not considered)

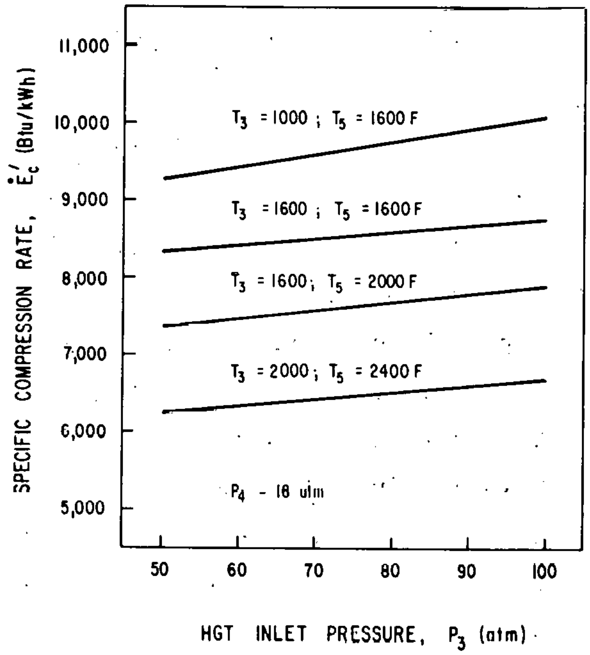


Fig. 3.8

Effect of HGT
Inlet Pressure on
Specific Compression Rate
(cooling air not considered)

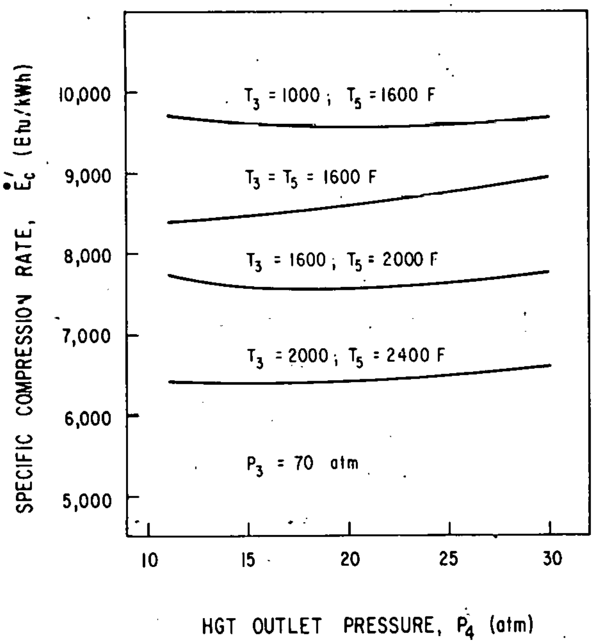


Fig. 3.9

Effect of HGT
Outlet Pressure on
Specific Compression Rate
(cooling air not considered)

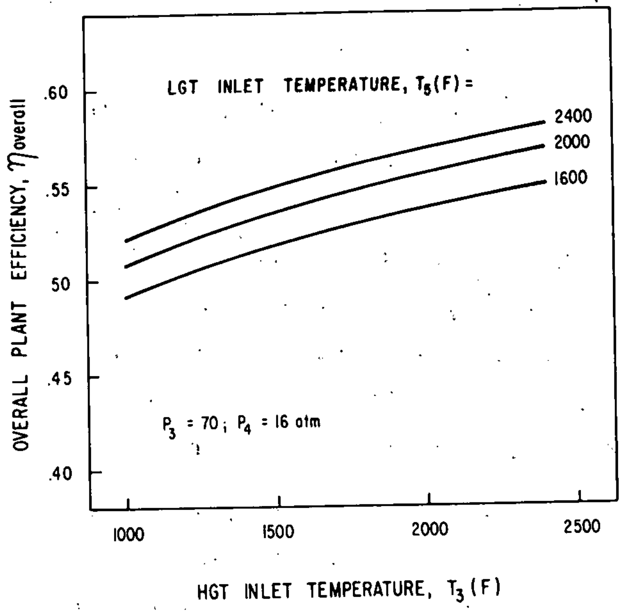


Fig. 3.10

Effect of Turbine
Inlet Temperatures on
Overall Plant Efficiency
(cooling air not considered)

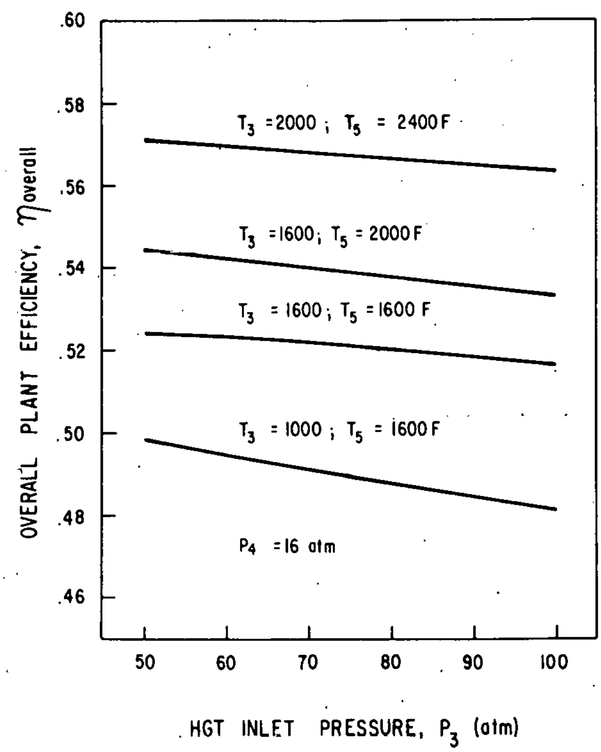


Fig. 3.11

Effect of HGT
Inlet Pressure on
Overall Plant Efficiency
(cooling air not considered)

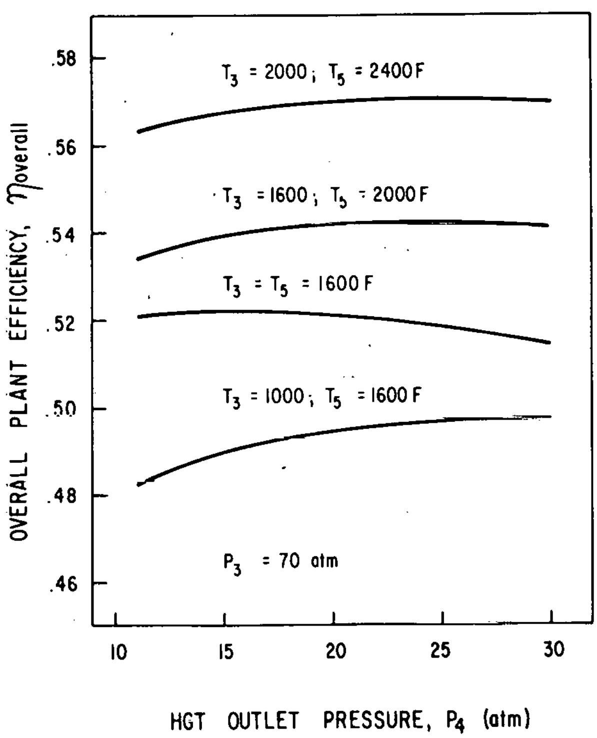


Fig. 3.12

Effect of HGT
Outlet Pressure on
Overall Plant Efficiency
(cooling air not considered)

Hoover[4] used a ratio of cooling air to turbine air of 0.085 for a turbine inlet gas temperature of 1850°F and a turbine pressure ratio of 10.

We assumed that cooling air is required for turbine inlet air temperatures above 1600°F and that mixing of cooling air with the main turbine flow is completed in the early stages. The second assumption may not be true in practical applications, since cooling air was found to be needed for more stages for such higher gas temperatures as 2400°F.[5] However, this simple cooling model predicts the performance of turbines with reasonable accuracy; e.g., it predicts a turbine outlet temperature of 821°F for a case in the UTRC report,[3] where the stage-by-stage model in the reference gave 774°F.

Before the study of effect of cooling air on the system performance, a case from Ref. 3 was examined to compare with results of the analysis in this report. The case was based on an input parameter different from the others cited in the report; power input to the compressor system was specified as 200 MW. In Table 3.1, in which results from the present analysis are compared with those in Ref. 3, specific turbine flow does not include the cooling air. The table shows that the present analysis predicts the performance of the CAES plant with acceptable accuracy, even though a very simple air-cooling model was used.

Table 3.1. Comparison of Results on a Case[a]

	Present Analysis	Ref. 3
Specific Turbine Flow (lb/kWh)	8.47	8.46
Specific Storage Volume (ft^3/kWh)	1.87	1.86
Specific Heat Rate (Btu/kWh)	4,040	4,130
Power Output (MW)	258	254
Overall Plant Efficiency	0.51	0.50

[a] Cavern pressure = 70 atm;
HGT inlet temperature = 1000°; LGT inlet temperature = 2000°F;
Cooling air-turbine air ratio (LGT) = 0.23; and
Power input to compressors = 200 MW.

Because the cooling air is supplied from the underground air-storage cavern, it increases the air flow rate and affects the results. The study includes the effect of cooling air on the following specific parameters: air flow, storage volume, heat rate, compression rate, and on overall plant efficiency. Four cases were examined:

Case	HGT Inlet Temperature, °F	LGT Inlet Temperature, °F
(a)	1000	2000
(b)	1600	2000
(c)	2000	2000
(d)	2400	2400

For cases (a) and (b), cooling air is needed for the LGT only, while both turbines require cooling in cases (c) and (d). Since there is little information on the amount of cooling air required as a function of turbine inlet temperature, a study was made by using the ratio of cooling-air flow to main-turbine flow as a parameter. Results were obtained for flow ratios of 0-0.5.

The effect of cooling air on the air flow and storage volume is given in Figs. 3.13a (cases a and b) and 3.13b (cases c and d). For all cases, air flow and storage volume increase by as much as 35% as the ratio of cooling air to turbine air increases from 0 to 0.5. Also shown in the figures are reference points representing the cases in which the highest allowable turbine inlet temperatures without blade cooling (1600°F) are used. Comparison with these reference data gives the limits on the ratio of cooling air to turbine air within which lower specific air flow and storage volume than the reference cases would be required: $r_{c_2} < 0.22$ for case (a), $r_{c_2} < 0.18$ for case (b), $r_{c_2} < 0.29 \sim 0.45$ depending on r_{c_1} for case (c), and $r_{c_2} \leq 0.6$ for all r_{c_1} for case (d).

The effects of cooling air on specific heat rate are given in Figs. 3.14a (cases a and b) and 3.14b (cases c and d): the specific heat rate increases for all cases regardless of the cooling air rate. The increases in specific heat rate are greater for the reference case by no more than 6, 14, and 17% for cases (a) and (b), (c), and (d), respectively.

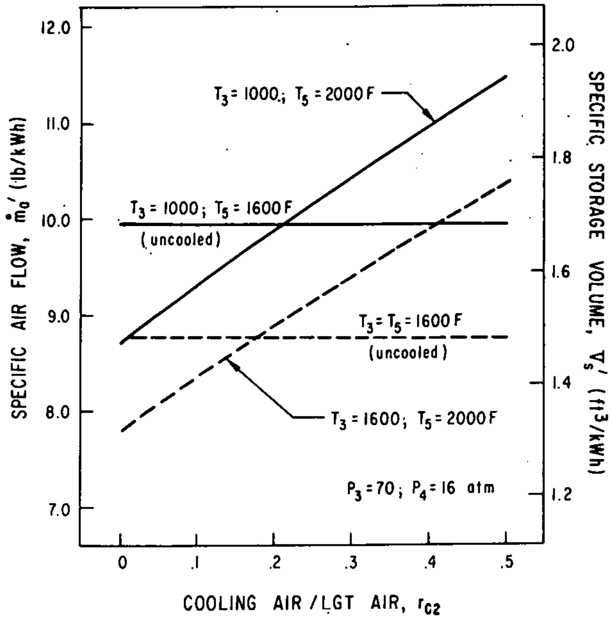


Fig. 3.13a

Effect of Cooling Air on Specific Air Flow and Specific Storage Volume (T_3=1000; T_5=2000°F and T_3=1600; T_5=2000°F)

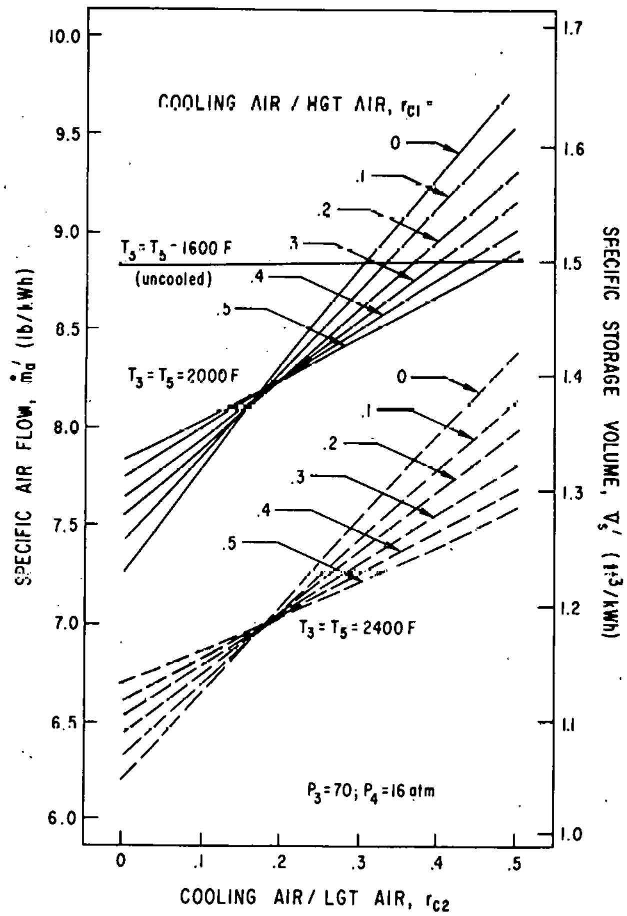
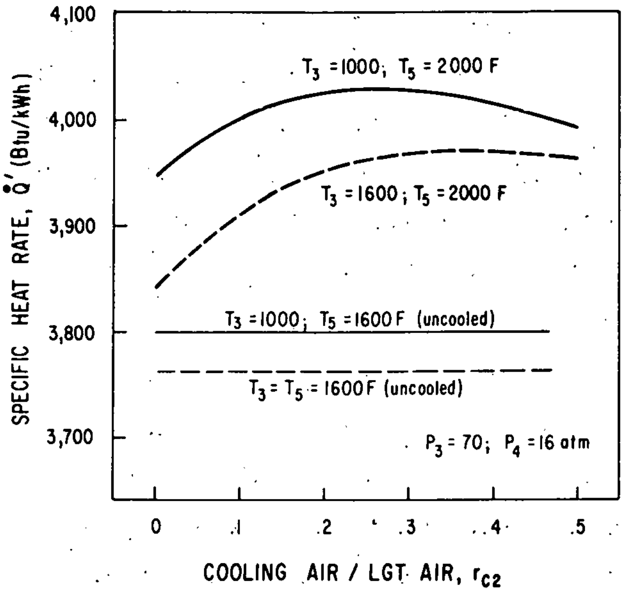


Fig. 3.13b

Effect of Cooling Air on Specific Air Flow and Specific Storage Volume (T_3=T_5=2000°F and T_3=T_5=2400°F)

Fig. 3.14a

Effect of Cooling Air on Specific Heat Rate (T_3=1000; T_5=2000°F and T_3=1600; T_5=2000°F)



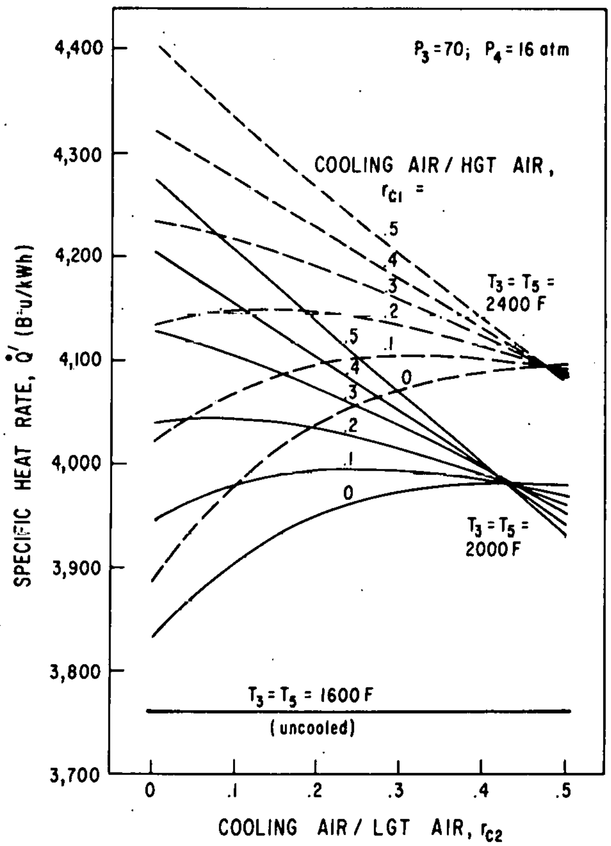


Fig. 3.14b

Effect of Cooling Air on Specific Heat Rate (T_3=T_5=2000°F and T_3=T_5=2400°F)

Effects of cooling air on specific compression rate (Figs. 3.15a and 3.15b) resemble those of specific air flow: compression energy increases monotonically as cooling air-turbine air ratio becomes larger. Comparison with the reference cases shows the limits on the cooling air-turbine air ratio within which less compression energy is required for higher turbine inlet temperatures. The limits are very close to those for air flow and storage volume.

Shown in Figs. 3.16a and 3.16b are the effects of cooling air on overall plant efficiency: in general, overall plant efficiency decreases with increasing ratio of cooling air to turbine air. Data for the reference cases are also shown for comparison. It is found that plant efficiency can be increased over that in the reference cases by using higher turbine inlet temperatures under the following conditions: $r_{c_2} < 0.1$ for cases (a) and (b); $r_{c_1} < 0.27$ and $r_{c_2} < 0.17$ for case (c); and $r_{c_1} < 0.48$ and $r_{c_2} < 0.34$ for case (d).

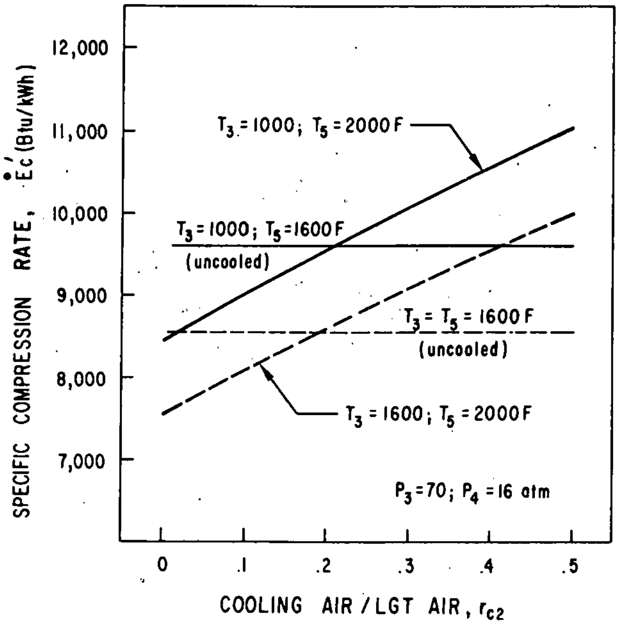


Fig. 3.15a

Effect of Cooling Air on Specific Compression Rate (T_3=1000; T_5=2000°F and T_3=1600; T_5=2000°F)

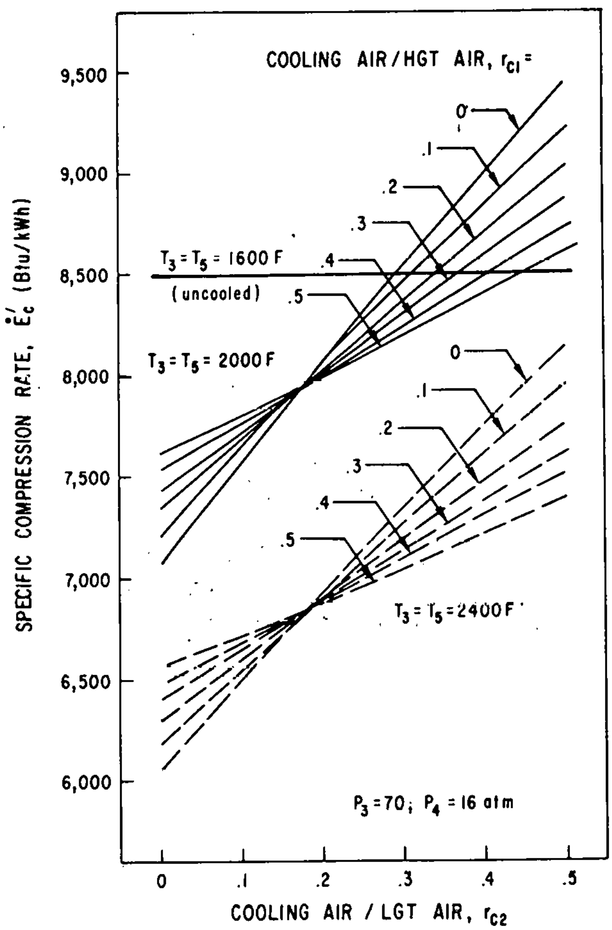
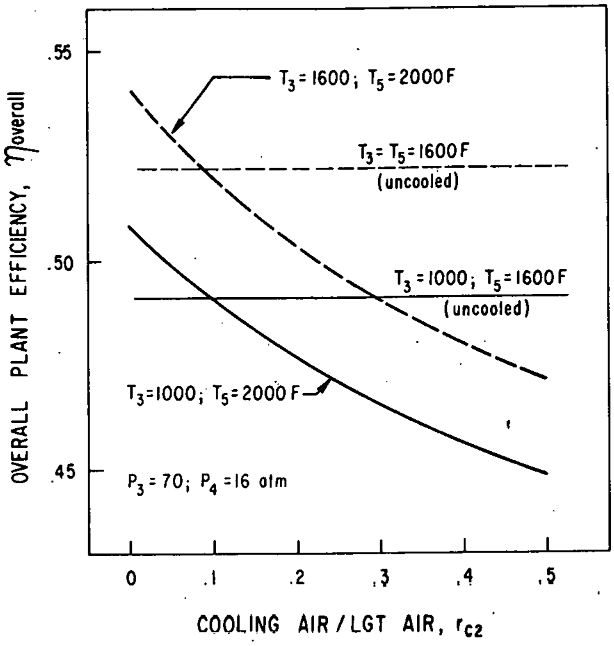


Fig. 3.15b

Effect of Cooling Air on Specific Compression Rate (T_3=T_5=2000°F and T_3=T_5=2400°F)

Fig. 3.16a

Effect of Cooling Air
on Overall Plant Efficiency
(T_3=1000; T_5=2000°F and
T_3=1600; T_5=2000°F)



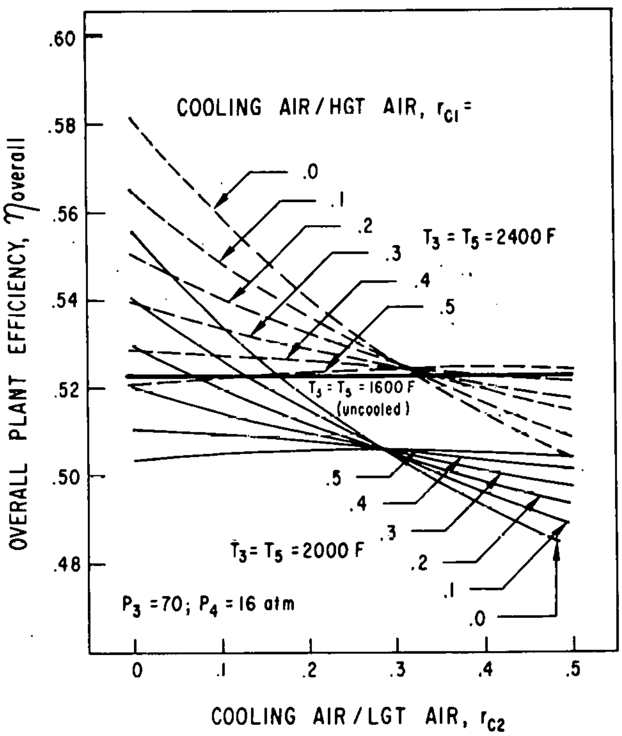


Fig. 3.16b

Effect of Cooling Air
on Overall Plant Efficiency
(T_3=T_5=2000°F and T_3=T_5=2400°F)

4 CONCLUSIONS AND RECOMMENDATIONS

For uncooled turbines, the following performance trends were observed:

1. As inlet gas temperatures to the turbine increase, specific air flow, storage volume, and compression rate decrease; specific heat rate and overall efficiency increase.
2. Specific heat rate is strongly influenced by inlet temperature to the low-pressure turbine (T_5 in Fig. 1.3), but only weakly affected by the inlet temperature to the high-pressure turbine.
3. As inlet pressure to the high-pressure turbine (p_3) increases, specific storage volume significantly decreases; specific air flow, heat rate, and overall efficiency slightly decrease; and specific compression rate slightly increases.
4. Outlet pressure from the high pressure turbine (p_4) affects performance only slightly.

Thus, it is recommended that the highest possible inlet gas temperatures without requiring cooling air should be used. The inlet pressure to the high-pressure turbine, which determines the storage pressure should be as high as possible. Since the outlet pressure from the high-pressure turbine has a minor effect on performance, conventional low-pressure turbines (from gas-turbine peaker units) having a nominal pressure ratio of 10-16:1 can be used.

For turbines that require cooling air, the following performance trends were observed:

1. Uncooled High-Pressure Turbine and Cooled Low-Pressure Turbine
   a) As r_{c_2} increases, specific air flow, storage volume, and compression rate increase. They are less than for equivalent uncooled turbines for $r_{c_2} < \sim 0.2$.
   b) The specific heat rate is always greater than that for equivalent uncooled turbines.
   c) The overall efficiency decreases with r_{c_2}. For $r_{c_2} < \sim 0.1$, the efficiency is greater than for uncooled turbines; but, for $r_{c_2} > \sim 0.1$, it is less.
2. Both Turbines Air-Cooled
   a) As r_{c_1} increases, specific air flow, storage volume, and compression rate decrease for $r_{c_2} > \sim 0.2$, but they increase for $r_{c_2} < \sim 0.2$. These values are less than for an equivalent uncooled turbine for most cooling air ratios considered.

b) There is an inflection point at $r_{c_2} \simeq 0.45$ for the specific heat rate: as r_{c_1} increases, the heat rate increases for $r_{c_2} < 0.45$ but decreases for $r_{c_2} > 0.45$. This rate is always greater than for the equivalent uncooled turbines.

c) There is an inflection point at $r_{c_2} \simeq 0.3$ for the overall efficiency: as r_{c_1} increases, the overall efficiency decreases for $r_{c_2} < 0.3$ but increases for $r_{c_2} > 0.3$. The overall efficiency is greater than for equivalent uncooled turbines for most cooling-air ratios when $T_3 = T_5 = 2400°F$, but it is less for most cooling-air ratios when $T_3 = T_5 = 2000°F$.

The above trends indicate that the amount of cooling air has a significant effect on the performance parameters; cooling air can either increase or decrease performance parameters compared with equivalent uncooled turbines. However, using high inlet gas temperatures, which require cooling air, always increases premium fuel usage. Detailed information on turbine cooling is therefore essential to accurately evaluate different turbine systems.

The United Technologies Research Center is completing a subcontract with ANL (the expected completion date is November 1, 1977) to evaluate the performance and cost of selected turbomachinery components for CAES plants. This work will provide the necessary data on turbine cooling. Completing the evaluation given here in this report is planned for FY 1978.

5 *APPENDIX*

A. *ANALYSIS OF TURBINE SYSTEM*

Governing equations were written for each component. Mass-balance equations were formulated first as follows:

$$\dot{m}_2 = \dot{m}_1 \tag{A.1}$$

$$\dot{m}_3 = \dot{m}_2 + \dot{m}_{f_1} \tag{A.2}$$

$$\dot{m}_4 = \dot{m}_3 + \dot{m}_{c_1} \tag{A.3}$$

$$\dot{m}_5 = \dot{m}_4 + \dot{m}_{f_2} \tag{A.4}$$

$$\dot{m}_5 = \dot{m}_5 + \dot{m}_{c_2} \tag{A.5}$$

$$m_7 = \dot{m}_6 \quad . \tag{A.6}$$

In the above equations, $\dot{m}$ is mass flow rate and subscripts 1-7 denote the stations shown in Fig. 1.3. The symbols $\dot{m}_{f_1}$ and $\dot{m}_{f_2}$ represent the rates of fuel gas supply to combustor 1 and combustor 2, respectively. Also, $\dot{m}_{c_1}$ and $\dot{m}_{c_2}$, respectively, denote the mass flow rates of cooling air into the HGT and LGT.

The fuel-air ratios are:

$$f_1 = \frac{\dot{m}_{f_1}}{\dot{m}_1} \quad \text{and} \tag{A.7}$$

$$f_2 = \frac{\dot{m}_{f_2}}{\dot{m}_1} \quad . \tag{A.8}$$

Ratios of cooling air to turbine air are defined as follows:

$$r_{c_1} = \frac{\dot{m}_{c_1}}{\dot{m}_1} \quad \text{and} \tag{A.9}$$

$$r_{c_2} = \frac{\dot{m}_{c_2}}{\dot{m}_1} \quad . \tag{A.10}$$

Then Eqs. (A.2)-(A.5) can be written as,

$$\dot{m}_3 = \dot{m}_2 + \dot{m}_1 f_1 \quad , \tag{A.2'}$$

$$\dot{m}_4 = \dot{m}_3 + \dot{m}_1 r_{c_1} \quad , \tag{A.3'}$$

$$\dot{m}_5 = \dot{m}_4 + \dot{m}_1 f_2 \quad , \text{ and} \tag{A.4'}$$

$$\dot{m}_6 = \dot{m}_5 + \dot{m}_1 r_{c_2} \quad . \tag{A.5'}$$

Pressure losses through the piping were not considered separately but were included in those across the components. The decrease in gas pressure was assumed to be 5% across the recuperator and 6% across the combustors. These values were based on data in Ref. 3. The following equations, which represent the pressure variations across each component, represent the momentum equations.

$$p_2 = 0.95\ p_1 \quad , \tag{A.11}$$

$$p_3 = 0.94\ p_2 \quad , \tag{A.12}$$

$$p_5 = 0.94\ p_4 \quad , \text{ and} \tag{A.13}$$

$$p_7 = 0.95\ p_6 \quad . \tag{A.14}$$

Equations for energy balance were then written for each component. From the definition of recuperator effectiveness,

$$\varepsilon = \frac{h_2 - h_1}{h_6 - h_1} \quad , \tag{A.15}$$

where ε is the recuperator effectiveness and h is the enthalpy at different states. Eq. (A.15) can be rewritten, with the use of the approximate method described in Ref. 2, as

$$\varepsilon = \frac{cp_2\ (T_2 - T_1)}{cp_6\ (T_6 - T_1)} \quad , \tag{A.16}$$

where c_p and T are the specific heats at constant pressure and temperature at different states. The energy balance is formulated for the recuperator as:

$$\dot{m}_1\ (h_2 - h_1) = \dot{m}_6\ (h_6 - h_7) \quad . \tag{A.17}$$

With use of the approximate method of analysis,[2] Eq. (A.17) becomes:

$$\dot{m}_1\ cp_2\ (T_2 - T_1) = \dot{m}_6\ cp_6\ (T_6 - T_7) \quad . \tag{A.18}$$

Thermal efficiencies of the turbines are defined as:

$$\eta_{HGT} = \frac{h_3 - h_4}{h_3 - h_{4s}} \quad ; \text{ and} \tag{A.19}$$

$$\eta_{LGT} = \frac{h_5 - h_6}{h_5 - h_{6s}} \quad . \tag{A.20}$$

In Eqs. (A.19) and (A.20), η_{HGT} and η_{LGT} are the thermal efficiencies of high-pressure and low-pressure gas turbines, respectively. Symbols h_{4s} and h_{6s} are the enthalpies at the turbine outlets when expansion through the turbines is isentropic. By using the approximate method, Eqs. (A.19) and (A.20) become:

$$\eta_{HGT} = \frac{T_3 - T_4}{T_3\left[1 - \left(\frac{p_4}{p_3}\right)^{a_3}\right]} \quad ; \text{ and} \tag{A.21}$$

$$\eta_{LGT} = \frac{T_5 - T_6}{T_5\left[1 - \left(\frac{p_6}{p_5}\right)^{a_5}\right]} \quad , \tag{A.22}$$

where $a = \frac{k - 1}{k}$ and

$$k = \frac{c_p}{c_v} \quad .$$

In the above equations, c_v is the specific heat at constant volume; the subscripts correspond to the stations illustrated in Fig. 1.3.

An energy-balance equation is written for HGT as follows:

$$\begin{aligned} \dot{W}_{HGT} &= \dot{m}_3\ (h_3 - h_4) + \dot{m}_{c_1}\ (h_c - h_4) \\ &\simeq \dot{m}_3\ c_{p_3}\ (T_3 - T_4) + \dot{m}_{c_1}\ c_{p_4}\ (T_c - T_4) \quad , \end{aligned} \tag{A.23}$$

where $\dot{W}_{HGT}$ is the power output of the HGT and subscript c represents the cooling air. Similarly, for the LGT:

$$\begin{aligned} \dot{W}_{LGT} &= \dot{m}_5\ (h_5 - h_6) + \dot{m}_{c_2}\ (h_c - h_6) \\ &\simeq \dot{m}_5\ c_{p_5}\ (T_5 - T_6) + \dot{m}_{c_2}\ c_{p_6}\ (T_c - T_6) \quad , \end{aligned} \tag{A.24}$$

where $\dot{W}_{LGT}$ is the power output of LGT.

The following equations represent energy balances based on the combustors. For combustor 1:

$$\dot{m}_2 h_2 + \dot{m}_{f_1}\ (h_{f_1} + \eta_{C_1}\Delta H_L) = \dot{m}_3 h_3 \quad , \tag{A.25}$$

where η_{C_1} is the efficiency of combustor 1 and ΔH_L is the lower heating value of fuel. Equation (A.25) can be reduced to:

$$c_{p_3}\ (T_3 - T_2) = f_1 \left[c_{p_3}\ (T_{f_1} - T_3) + \eta_{C_1} \Delta H_L\right] \quad . \tag{A.26}$$

Similarly, for combustor 2:

$$c_{p_5}\ (T_5 - T_4) = f_2{}' \left[c_{p_5}\ (T_{f_2} - T_5) + \eta_{C_2} \Delta H_L\right] \quad , \tag{A.27}$$

where $f_2{}' = \dfrac{f_2}{1 + f_1 + r_{c_1}}$.

The equations for mass, momentum, and energy balances were solved simultaneously with a simulation computer program written for this purpose. The following results were calculated: turbine outlet temperatures, T_4 and T_6; recuperator outlet temperatures, T_2 and T_7; and fuel-air ratios for the combustors, f_1 and f_2. The rate of air flow from the underground storage cavern is then obtained from:

$$\dot{m}_1 = \frac{\dot{W}_{LGT}}{(1 + f_1 + f_2 + r_{c_1})\ c_{p_5}\ (T_5 - T_6) + r_{c_2}\ c_{p_6}\ (Tc - T_6)} \quad . \tag{A.28}$$

From these results, the power output of HGT can be calculated as follows:

$$\dot{W}_{HGT} = \dot{m}_1 \left[(1 + f_1)\ c_{p_3}\ (T_3 - T_4) + r_{c_1}\ c_{p_4}\ (T_6 - T_4)\right] \quad . \tag{A.29}$$

The total power output of the turbine system is then:

$$\dot{W}_{out} = \dot{W}_{LGT} + \dot{W}_{HGT} \quad . \tag{A.30}$$

Specific air flow rate and heat rate are obtained from

$$\dot{m}_a' = \dot{m}_1\ (1 + r_{c_1} + r_{c_2})/\dot{W}_{out} \quad ; \text{ and} \tag{A.31}$$

$$\dot{Q}' = \dot{m}_1\ (f_1 + f_2)\ \Delta H_L/W_{out} \quad . \tag{A.32}$$

B. ANALYSIS OF UNDERGROUND AIR STORAGE SYSTEM

Pressure variation in the air shaft of an underground storage cavern is due to changes in static head and friction. The variation in static-head pressure is a function of the cavern pressure. From Ref. 3:

$$\Delta p_s = p_{ca} \left[1\text{-exp}\ (-0.00111\ p_{ca})\right] \quad , \tag{B.1}$$

where Δp_s and p_{ca}, respectively, denote variation in static-head pressure and the cavern pressure in atmospheres. The pressure loss due to friction is known as a function of air flow rate, but it is considered insignificant compared with the variation in static-head pressure. Variation in frictional pressure (Δp_f) is about 0.07 ~ 0.20 atm at air flows of 500 ~ 1000 lb/s. Based on this fact, a pressure drop of 0.15 atm was taken for the frictional loss in the present study. The air pressure in the cavern is related to that at the cavern outlet (p_1) as follows:

$$p_{ca} = p_1 - \Delta p_s + \Delta p_f \quad . \tag{B.2}$$

Specific storage volume (V'_s) is obtained from:

$$V'_s = \frac{\dot{m}'_a RT_{ca}}{p_{ca}} \quad , \tag{B.3}$$

where R is the gas constant and T_{ca} is the air temperature in the cavern.

C. ANALYSIS OF COMPRESSOR SYSTEM

The rate of air flow into the compressor system should match that from the storage cavern into the turbine system. A loss of 4% of the air flow in the cavern was assumed.[3] The rate of air flow into the compressor system ($\dot{m}_{11}$) can therefore be related to the turbine flow rate ($\dot{m}_1$) as:

$$\dot{m}_{11} = 1.042\ \dot{m}_1\ (1 + r_{c_1} + r_{c_2}) \quad . \tag{C.1}$$

If a negligible loss of air in the compressor system is assumed:

$$\dot{m}_{11} = \dot{m}_{12} = \dot{m}_{13} = \dot{m}_{14} = \dot{m}_{15} \quad . \tag{C.2}$$

The subscripts represent the stations given in Fig. 1.2.

The pressure drops across the intercooler and aftercooler were assumed to be 7.3% and 2%, respectively, based on Ref. 3. Therefore:

$$p_{13} = 0.927\ p_{12} \quad ; \text{ and} \tag{C.3}$$

$$p_{15} = 0.98\ p_{14} \quad . \tag{C.4}$$

Since the pressure of air entering the cavern is higher than that leaving by about 0.3 atm because of the frictional loss in the shaft, then:

$$p_{15} = p_1 + 0.3 \quad . \tag{C.5}$$

Adiabatic efficiencies of the compressors are defined as:

$$\eta_{LPC} = \frac{T_{11}\left[\left(\frac{p_{12}}{p_{11}}\right)^{a_{12}} - 1\right]}{T_{12} - T_{11}} \quad ; \text{ and} \tag{C.6}$$

$$\eta_{BC} = \frac{T_{13}\left[\left(\frac{p_{14}}{p_{13}}\right)^{a_{14}} - 1\right]}{T_{14} - T_{13}} \quad , \tag{C.7}$$

where $a = \frac{k - 1}{k}$, and

$$k = \frac{c_p}{c_v} \quad .$$

Figure 1.2 gives the subscripts in the above equations.

The compressor outlet temperatures, T_{12} and T_{14}, can be calculated from Eqs. C.6 and C.7. The power inputs into the compressors, $\dot{W}_{LPC}$ and $\dot{W}_{BC}$, can be obtained as follows:

$$\dot{W}_{LPC} = \dot{m}_{11}\, cp_{12}\, (T_{12} - T_{11}) \quad ; \text{ and} \tag{C.8}$$

$$\dot{W}_{BC} = \dot{m}_{11}\, cp_{14}\, (T_{14} - T_{13}) \quad . \tag{C.9}$$

The total power input to the compressors is then:

$$\dot{W}_{comp} = \dot{W}_{LPC} + \dot{W}_{BC} \quad . \tag{C.10}$$

Specific compression energy ($\dot{E}'_c$) is defined as the fuel energy equivalent of compressor input per unit total power output, i.e.,

$$\dot{E}'_c = \frac{\dot{W}_{comp} \cdot \Delta H_H}{\dot{W}_{out}} \quad , \tag{C.11}$$

where ΔH_H is the off-peak heat rate of the power plant, including electrical and mechanical losses.

6 REFERENCES

1. Jacob, R.O., R.W. Barta, and P.H. Huhtanen, *Utility Experience with Regenerative Cycle Combustion Turbines*, ASME Gas Turbine Conf., ASME Paper No. 76-GT-85 (1976).

2. Kartsounes, G.T., *Evaluation of Turbine Systems for Compressed Air Energy Storage Plants*, Argonne National Laboratory Report ANL/ES-59 (1976).

3. Giramonti, A.J., et al., *Preliminary Feasibility Evaluation of Compressed Air Storage Power Systems*, UTRC, R76-952161-5 (1976).

4. Ayers, D.L., and D.Q. Hoover, *Gas Turbine Systems Using Underground Compressed Air Storage*, Proc. Am. Power Conf., *36*:379-389 (1974).

5. Davidson, W.R., and R.D. Lessard, *Study of Selected Turbomachinery Components for Compressed Air Energy Storage Systems*, UTRC, R77-952923 (1977).

THIS PAGE

WAS INTENTIONALLY

LEFT BLANK

External:

U. Illinois, Chicago Circle:
  H.H. Chiu
  L.W. Rodgers
  A. Sharma
Public Service Indiana:
  T.W. McCafferty
  C.E. Foggatt
  J.M. Bumgarner
Patomic Electric Power Company:
  P.E. Schaub
  J.H. Rumbaugh
  J. Rasmussen
Middle South Services, New Orleans:
  L.A. Wilson (5)
  W.R. Barcelo
United Engineers and Constructors, Philadelphia:
  E.J. Sosnowicz (5)
  A.J. Karalis
West Virginia U., Morgantown:
  S.H. Schwartz
  D.J. Marinacci
  K. Mehta
A.J. Giramonti, United Technologies Research Center
W.C. Walke, Sargent & Lundy, Chicago
P.J. Womeldorff, Illinois Power, Decatur
M. Lovett, Technip, New York
A. Manaker, TVA, Chattanooga, TN
R.H. Shannon, Mined Storage Limited, Ontario
P.A. Berman, Westinghouse Electric, Philadelphia
K.G. Vosburgh, General Electric R&D Center, Schenectady
G.E. DeViney, Commonwealth Edison Co., Chicago

☆ U.S.GOVERNMENT PRINTING OFFICE: 1977 750-133/34