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EVALUATION OF TURBINE SYSTEMS FOR COMPRESSED AIR ENERGY STORAGE PLANTS

Final Report for FY 1976

by

George T. Kartsounes

ARGONNE NATIONAL LABORATORY

ENERGY AND ENVIRONMENTAL SYSTEMS DIVISION

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October 1976

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Division of Energy Storage Systems

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a = mass of air

A, B = variables defined in Eq. 10

A_c = cross-sectional area

c = $(k - 1)/k$

c_p = specific heat at constant pressure

c_v = specific heat at constant volume

f = mass of fuel

h = enthalpy

$\bar{h}$ = molal enthalpy

$\bar{h}_f^o$ = enthalpy of formation

k = specific heat ratio

LHV_o = lower heating value of fuel

$\dot{m}'_a$ = specific turbine flow rate

$\dot{m}'_o$ = specific turbine flow rate of Huntorf plant

$\dot{m}_f$ = mass flow rate of fuel

M = molecular weight

Ma = Mach number

n = moles

p = absolute pressure

p_r = relative pressure

P = turbine power output

P' = specific power output

$\dot{Q}'$ = heat rate

r_p = pressure ratio

R = gas constant of air

s = entropy

T = absolute temperature

V = axial velocity

x = mole fraction

y = r_p^c

Z = T_4/T_1 in Fig. 3.2

Greek Symbols

ε = recuperator effectiveness

η = efficiency

ρ = density

Subscripts

1,2,3,...., = state points in T-s diagrams

a = air

atm = atmospheric

C = compressor

comb = combustor

f = fuel

HGT = high pressure gas turbine

i = constituent or intermediate pressure

in = inlet

LGT = low pressure gas turbine

O = reference temperature for LHV_o or reference conditions based on Huntorf plant

P = products

R = reactants

s = isentropic

T = turbine

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ABSTRACT

Compressed air energy storage plants for electric utility peak-shaving applications comprise four subsystems: a turbine system, compressor system, an underground air storage reservoir, and a motor/generator. Proposed plant designs use turbines that are derived from available gas and steam turbines with proven reliability. The study examines proposed turbine systems and presents an evaluation of possible systems that may reduce capital cost and/or improve performance. Six new turbine systems are identified for further economic evaluation.

EXECUTIVE SUMMARY

Compressed air energy storage (CAES) plants are being considered by electric utilities for peak-shaving applications. They comprise a turbine system, compressor system, and an underground storage reservoir. Turbines for proposed CAES plants are derived from state-of-the-art gas turbines and steam turbines. The main criterion for these systems is to use presently available components with proven reliability. The question that must logically be asked is, Are there *better* turbine systems for CAES applications? Addressing this question and identifying turbine systems that may offer cost and/or performance incentives are the principal objectives of this study.

The study is focused on a two-turbine system composed of a high pressure gas turbine (HGT) and a low pressure gas turbine (LGT). The West German Huntorf plant, which is to be the world's first CAES plant, uses this type of arrangement.

For proposed CAES plants, the HGT is a slightly modified existing steam turbine design. The LGT is an existing gas turbine being used in conventional gas turbine peaking units. For the LGT, two different designs are available. One uses internal blade air-cooling to permit inlet gas temperatures of 1800-2000 F. The other uses simpler uncooled blading for gas temperatures below about 1600 F.

The use of a recuperator to reduce heat rate has been proposed for CAES turbine systems. Preliminary studies indicate that recuperators can be designed that are economically feasible for CAES applications. They will differ from designs for conventional gas turbine peaking units because of the high pressure air from the reservoir.

From a detailed analysis of possible turbine systems, several new system options, based on three different system configurations, have been identified. These options are outlined below. The report includes a discussion in detail of the advantages and disadvantages of each option.

1. System Composed of Two Turbines and Two Combustors

   Option A - High Turbine Inlet Temperatures
   Option B - Large Pressure Ratio LGT
   Option C - Low Inlet Temperature to LGT

2. System Composed of Two Turbines, Two Combustors, and a Recuperator

   Option D - Option A with a Recuperator
   Option E - Option B with a Recuperator

3. System Composed of a Turbine and Combustor Exhausting at a Low Temperature

   Option F - Turbine Composed of a HGT and LGT with the Exhaust Temperature Equal to the Inlet Temperature

High temperature turbines for conventional gas turbine peaking units are being extensively studied by turbine manufacturers. Designs for gas temperatures as high as 3000 F are being developed. These turbines could be used for the LGT in Options A and D. However, high temperature turbines for HGT applications have received little attention.

Option C, which uses a reduced inlet temperature to the LGT, results in decreased heat rate and permits the use of less expensive materials for the LGT. This option has application only to systems without a recuperator.

The use of a high pressure ratio LGT in Options B and E is very attractive due to a possible size reduction. Using this concept, the LGT would supply the majority of the power output and the HGT would be tailored to meet the required pressure ratio of the system.

Option F represents a significant departure from presently available large power output gas turbines. Because of the extremely low operating temperatures, the LGT resembles small industrial turbines. This option offers the potential for a significant cost reduction.

Two levels of cost information are required to assess the new options cited: (1) the development costs for the new turbines, combustors, and recuperators and (2) assuming large scale production, the capital costs of each system and of available equipment. With this information, an engineering economics evaluation of possible systems can be made.

The required cost information necessitates a comparative evaluation of turbines, combustors, and recuperators. The evaluation should be unbiased so that it does not merely reflect a particular manufacturer's design constraints.

It is therefore recommended that the study reported herein be extended to include an economic evaluation of possible turbine systems.

1 INTRODUCTION

The electric utility industry has as its main objective the supply of power at the lowest possible cost. This purpose has led to the development of large sophisticated nuclear and fossil-fuel-fired steam generating plants. For both technical and economical reasons, these plants should be operated at a steady load. However, to meet daily and seasonal fluctuations in power demand, the industry uses so called *peaker* units. The most common form of these units are gas turbine systems that use premium fuels such as natural gas and oil.

Because of the limited supply of oil and natural gas in this country and current problems in the supply of petroleum fuel from foreign sources, the price of premium fuel has become very expensive and the long-term supply is uncertain. Therefore, electric utilities have been exploring *better* ways of utilizing, or even eliminating, the use of premium fuels for peaker units and the possibility of operating their large power plants at steady or constant load. These considerations have led to the investigation of energy storage systems.

Suitable energy storage systems could store the excess power generated during off-peak hours, thereby allowing the power plant to operate continually at constant load and to return the stored energy as peak power when required. Candidate energy storage systems include the following: thermal storage in various materials; flywheels; electrolysis; batteries; pumped hydro (above or below ground); and compressed air energy storage (CAES). Studies conducted by electric utilities indicate that CAES power plants are attractive for consideration.

A CAES plant comprises a gas turbine-generator set, a combustor to preheat the air, a compressor system, and an underground air storage reservoir. In contrast to conventional peaker units, the turbine system and compressor system are uncoupled; each system operates independently. The purpose of this arrangement is to drive the compressor system with off-peak power from a main power plant and to generate peak power from the turbine system when needed.

The uncoupling of the turbine and compressor systems permits the utilization of the full power output of the turbine system to drive the generator. In a conventional gas turbine peaker unit, about one- to two-thirds of that output is used to operate the compressor. In a CAES plant, therefore, the required capacity (i.e., the gross power output) of the turbine system, as well as the quantity of fuel needed, will be reduced by the same fractional proportion. The capacity of the compressor system will also be reduced but the amount depends upon the charging and discharging time of the air reservoir.

Compressed air can be stored underground in caverns or in the pore space of porous rock formations. The caverns may be natural or mined. The latter may be constructed by conventional mining, nuclear explosives, or solution mining as in the case of salt structures. Because the porous rock formations generally contain water, they are called *aquifers*. To use an aquifer as a storage reservoir, as has been done for many years, the water in the rock must be displaced by air.

Various plant configurations are being evaluated for air storage pressures in the range of 10-80 atmospheres. For example, one possible configuration is schematically illustrated in Fig. 1.1. This plant is similar to the West German Huntorf plant[1] which is to be the world's first CAES plant and is scheduled to go into operation in mid-1977. The turbine system illustrated in Fig. 1.1 uses two turbines and two combustors. The combustor upstream of the high pressure turbine burns fuel with air supplied from the storage reservoir. The products of combustion are then expanded through the high pressure turbine and flow into the second combustor. The oxygen remaining in the products is used to burn fuel in the second combustor. The products leaving this combustor expand through the low pressure turbine, flow through the recuperator, and are then exhausted to the ambient. The recuperator, which is optional, preheats the air leaving the reservoir and thereby decreases the fuel required for reheat.

The overall performance of a turbine system is expressed in terms of two parameters: specific turbine flow rate and heat rate. Specific turbine flow rate is equal to the mass flow rate of air from the reservoir divided by the power output (i.e., lb of air/kWh). It is directly related to the size and, therefore, the cost of the turbines. Heat rate is directly proportional to fuel consumption and is equal to the product of specific fuel consumption (i.e., lb of fuel/kWh) and the lower heating value of the fuel. It is therefore related to the operating cost of the turbines.

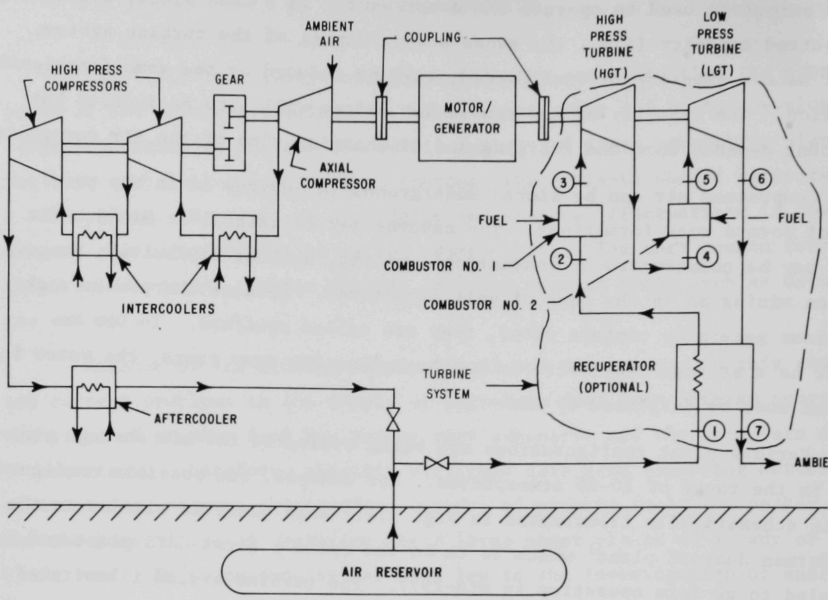

Fig. 1.1. Schematic Diagram of CAES Plant

The characteristics of combustion turbines which have been considered for CAES plants are presented in Table 1.1. Note that specific turbine flow rate varies from 11-14 lb air/kWh; the corresponding heat rates vary from 4000-6200 Btu/kWh. This wide range of heat rates is due to the use of a recuperator. A recuperator, which is illustrated in Fig. 1.1, is a heat exchanger that uses the high temperature turbine exhaust to preheat the air leaving the underground storage reservoir. As an example of its value, the heat rate of 5560 Btu/kWh (Ref. 8) is reduced to about 4100 Btu/kWh (Ref. 9) if a recuperator is used.

Total CAES plant costs are summarized in Table 1.2. Neglecting the storage cost presented in Ref. 11 and assuming that 5 hr/day power generation corresponds to the largest storage cost cited, then the capital cost of the reservoir is about 10-70% of the total plant cost. This large range is due to the type of reservoir employed. Reference 4 indicates that the majority of the surface plant cost is invested in the turbomachinery; the turbomachinery cost is estimated as $73/kW to $113/kW. In addition, more than half of the turbomachinery cost is due to the turbine system.

Table 1.1. Characteristics of Combustion Turbines for CAES Plants

Manufacturer	Ref.	Pressure Ratio	Inlet Temp. (°F)	Specific Turbine Flow (lb air/kWh)	Output (MW)	Heat Rate (Btu/kWh)
Westinghouse Electric	2	10:1	1850	14	168	6200
General Electric	3	11:1	-	-	169	4600
	4	10.3:1	1985	13.3	150	4300
United Tech. Corp.	5	40:1	2000	11	-	4000
Stal-Laval	6	43:1	1470	13	220	4770
Stal-Laval	7	25:1	1650	13	232	5370
Brown Boveri Corp.	8	4.5:1 10:1 } 45:1	1022 1517	11.4	290	5560

Table 1.2. Characteristics of CAES Plants[4]

Ref.	Year	Surface Plant ($/kW)	Storage ($/kWh)	Projected Life (yr)	Heat Rate (Btu/kWh)
10	1970	50	0.68	-	3960
11	1971	55	300-375	28-40	4770
12	1973	38	5.50	-	5830
13	1974	90	3-5	30	-
3	1974	85	-	-	4600
2	1974	56	7[a]	-	6200
5	1974	65	1-6[b]	-	4000
14	1974	92	2.80	-	3860
9	1975	96	15	-	5560
15	1975	102	13.80	-	5560

[a]Per kW for an aquifer.

[b]Corresponds to 10-$30/kW.

Thus, it can be concluded that the turbine system represents a significant portion of the total plant cost as well as the major operating cost. Turbine performance (e.g., air flow rate) will also affect reservoir size and cost and the required capacity of the compressor system. All of these factors imply that the *best* turbine system should be used to make CAES a viable energy storage scheme, the goal toward which this study is directed.

This report presents an evaluation of turbine systems for CAES plants. The main objectives of the study were to examine proposed turbine systems and to identify systems that offer cost incentives for future use. It was sponsored by the Energy Research and Development Administration. Work was initiated during March 1976 and this document presents the FY 76 effort.

The report is divided into three major parts. Section 2 presents a preliminary evaluation of possible CAES turbine systems whose purpose is to identify system configurations worthy of further investigation. It represents a prescreening of candidate systems.

Section 3 presents a detailed study of the turbine systems. Three general system configurations are considered and the performance trends of each are discussed, leading to the identification of possible system options. The advantages and disadvantages of each option are then summarized. Finally, in Section 4, overall conclusions are presented and recommendations are made for further work.

Appendix A offers analyses pertinent to the evaluation of possible CAES turbine systems. Given first is the analysis of conventional gas turbine systems that are being used as peaker units. This information is included as general background material because the turbines in these units are being considered for use in proposed CAES plants. Next, four different methods of analyzing the performance of CAES turbine systems are presented. Finally, a method is developed for estimating the relative size of new turbine design.

2 PRELIMINARY EVALUATION OF POSSIBLE TURBINE SYSTEMS

The purpose of this preliminary evaluation of possible turbine systems for CAES plants is to identify systems that offer possible cost and/or performance incentives. The turbine system of the Huntorf plant is used as a basis for comparison inasmuch as the candidate systems have the same inlet pressure and temperature. For simplicity, the systems are evaluated using the air-standard method of analysis (see Appendix).

2.1 HUNTORF SYSTEM

Fig. 2.1 presents a schematic diagram of the turbine system and corresponding temperature-entropy plot of the thermodynamic process for the Huntorf plant. The data labeled on the schematic diagram* were given by Zaugg[16] and Stys.[17]

From these data, the following results were calculated:
T_3 = 603 F and η_{HGT} = 0.839 and η_{LGT} = 0.1818.

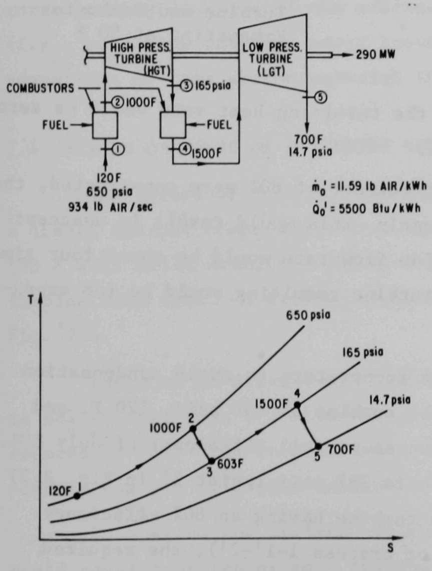

Fig. 2.1. Turbine System for Huntorf Plant

The present design of the Huntorf turbine system does not incorporate a recuperator. Luthi[9] estimated that using a recuperator would increase the capital cost by 17% while lowering the heat rate to 4100 Btu/kWh. This estimate was based on the work reported by Hartmann and Hoffman.[18] Recent studies by Brown Boveri Corp.[19] indicate that recuperators are economically feasible for CAES systems.

2.2 AIR TURBINE SYSTEM

Expanding the high pressure air through an air turbine would result in the simplest possible turbine system.

*The subscript o will be used to denote the Huntorf system performance parameters.

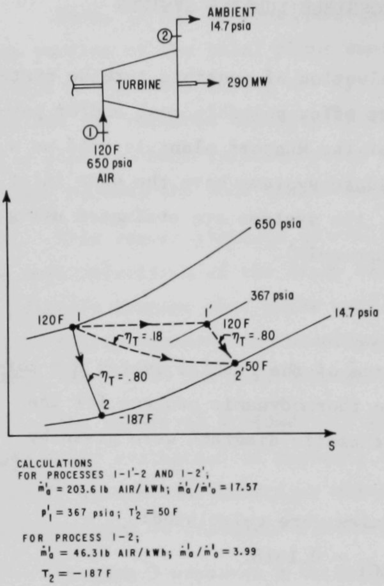

Fig. 2.2. Air Turbine System

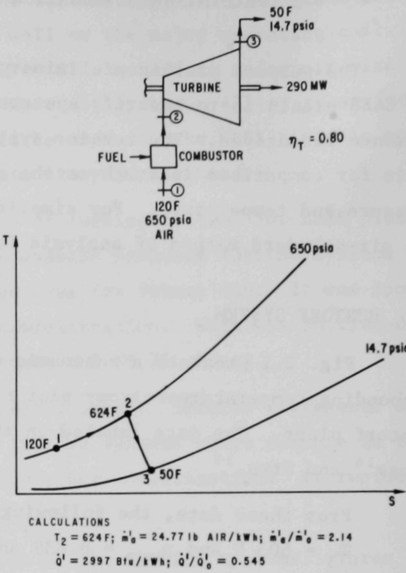

Fig. 2.3. System Composed of a Turbine and Combustor Exhausting at 50 F

No auxiliary fuel would be required and the resulting heat rate would be zero. This type of system is shown in Fig. 2.2.

If a turbine having an overall efficiency of 80% were constructed, the outlet temperature would be -187 F. Clearly, this would result in unacceptable icing problems. The specific turbine flow rate would be about four times that of the Huntorf system. The large turbine resulting would be too expensive for consideration.

The lowest practical turbine exit temperature to avoid condensation and/or icing is about 50 F. Entering the turbine at 650 psia, 120 F, and expanding to 50 F requires a turbine with an overall efficiency of only 18%. A more practical approach is to throttle to 367 psia (point 1' in Fig. 2.2) and expand to 50 F with, for example, a turbine having an 80% efficiency. For either process (i.e., process 1-2' or process 1-1'-2'), the required turbine would be unacceptable in size and cost.

Thus it can be concluded that an air turbine system should not be considered. In order to reduce the specific turbine flow rate and thereby decrease the physical size and cost of the system, the inlet temperature to the turbine must be increased. This increase suggests the use of a combustor and the supply of auxiliary fuel.

2.3 SYSTEM COMPOSED OF A TURBINE AND COMBUSTOR EXHAUSTING AT 50 F

The next level of complexity would be to add a combustor to the system and exhaust the products of combustion at 50 F as illustrated in Fig. 2.3.

An overall efficiency of 80% was assumed for the calculations. The specific turbine flow rate is about two times that of the Huntorf design and the heat rate is about 45% less. The inlet temperature to the turbine is only 624 F. At this greatly reduced temperature, less expensive materials could be used for the turbine components and reliability would be expected to improve. Note also that this system has a lower heat rate than the Huntorf design with a recuperator.

The doubling of the specific turbine flow rate, however, requires a corresponding doubling of the storage reservoir volume and of the capacity (i.e., mass flow rate and power input) of the compressors. Whether the advantages of this system outweigh the disadvantages is therefore doubtful.

2.4 SYSTEM COMPOSED OF A TURBINE AND COMBUSTOR EXHAUSTING AT 120 F

One method of further reducing specific turbine flow rate is to allow a higher exhaust temperature from the turbine. An exhaust temperature up to 120 F will still eliminate the need of a recuperator and decrease the specific turbine flow rate. A turbine system using a 120 F exhaust is illustrated in Fig. 2.4.

Based on an overall efficiency of 80%, the specific turbine flow rate is 88% greater than the flow rate for the Huntorf system and the heat rate is reduced by 38%. The turbine inlet temperature is only 772 F, which still permits the use of a less expensive material for construction.

To achieve the desired pressure ratio of 650/14.7 = 44.2, the turbine could consist of two or more turbines connected in series without reheat between turbines. For example, consider a high pressure turbine and a low

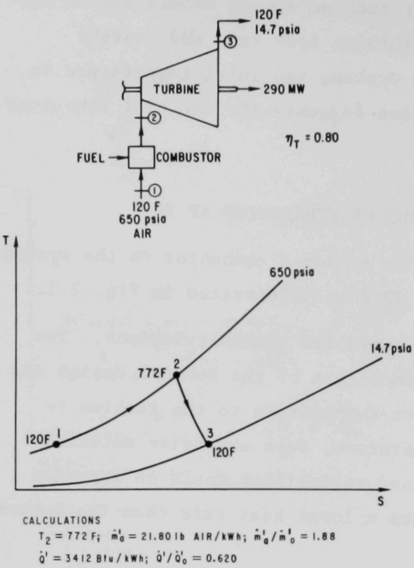

Fig. 2.4. System Composed of a Turbine and Combustor Exhausting at 120 F

pressure turbine with an intermediate pressure of 100 psia. If each turbine has an efficiency of 80%, analysis of the system will yield,

$$T_2 = 850 \text{ F}; \ \dot{m}'_a = 19.47 \text{ lb air/kWh};$$

$$\dot{m}'_a/\dot{m}'_o = 1.68; \text{ and}$$

$$\dot{Q}' = 3412 \text{ Btu/kWh}; \ \dot{Q}'/\dot{Q}'_o = 0.620.$$

Thus, the heat rate remains unchanged, but the specific turbine flow rate is now 68% greater than the flow rate of the Huntorf system. This reduction in flow rate is due to the so-called *reheat effect*, which states that the efficiency of a multistage turbine can be higher than the efficiency of any of its stages. This effect can be attributed to the divergence of constant pressure lines on a T-s diagram.

Thus, the 120 F exhaust temperature results in a reduction in the specific turbine flow rate. However, it remains questionable whether this system offers an economic advantage over the Huntorf system.

2.5 SYSTEM COMPOSED OF TWO TURBINES, TWO COMBUSTORS AND A RECUPERATOR

A system that has a high and a low pressure turbine, a combustor for each turbine, and a recuperator to recover heat from the hot exhaust gas, will be considered. It is similar to the Huntorf system with the addition of a recuperator. A recuperator will decrease the heat rate, but will not change the specific turbine flow rate for fixed turbine inlet temperatures.

One way to reduce the cost of the low pressure turbine is to reduce the inlet temperature below 1000 F, so that a less expensive material can be used for construction. This reduction will require an increase in specific turbine flow rate with subsequent increases in the size of the reservoir and the capacity of the compressors to maintain a fixed power output.

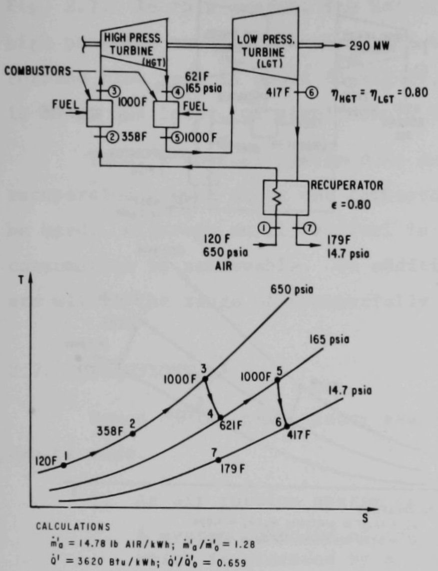

Fig. 2.5. System Composed of Two Turbines, Two Combustors, and a Recuperator

A turbine system using a 1000 F inlet temperature to both turbines is illustrated in Fig. 2.5. Analysis of this system indicates that the specific turbine flow rate is 28% greater than the flow rate for the Huntorf system, and the heat rate is about 34% less than the Huntorf system. Also note that the heat rate is 3620 Btu/kWh compared to 4100 Btu/kWh for the Huntorf system with a recuperator.

Another possible advantage of using a low inlet temperature to the low pressure turbine is mentioned. If the auxiliary fuel is corrosive, such as low Btu coal gas from a coal gasification process, the low gas temperatures in the turbine may reduce or even eliminate blade corrosion. Thus, this system offers a number of attractive features, and it should be economically evaluated to determine whether a cost advantage exists.

2.6 SYSTEM USING HOT AIR

Proposed designs of CAES systems rely on petroleum fuels to reheat the air. This use of petroleum fuels, although considerably less than conventional gas turbine peaking facilities, represents a weakness that may prove irreparable. The current world problems in the supply and demand of petroleum fuels make it open to debate whether a utility or other potential user of CAES technology would invest in a CAES system without the eventual possibility of eliminating the reliance on petroleum fuels.

One method of achieving this goal is to recover the heat produced during the compression process. Recovery can be accomplished either by storing the hot air from the compressor directly in the reservoir (this type of reservoir is referred to as an *adiabatic reservoir*) or by storing the heat

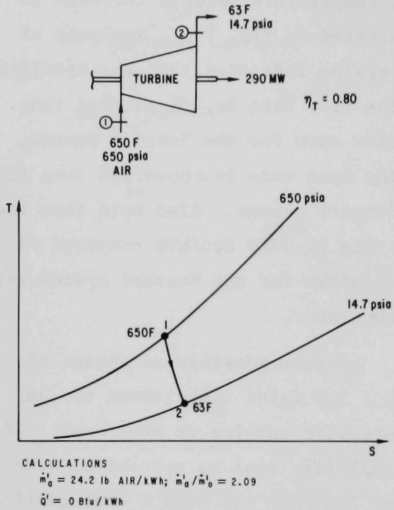

Fig. 2.6. "No Fuel" Turbine System

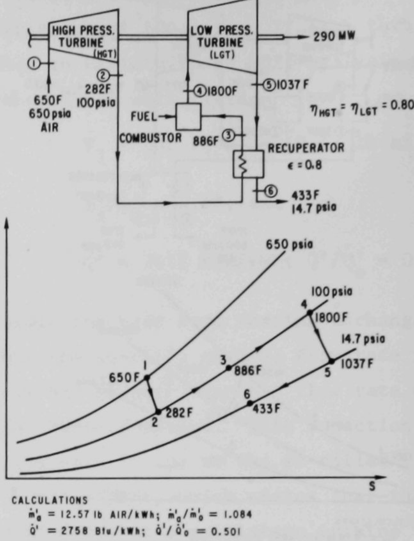

Fig. 2.7. Hot Air System with a Combustor

of compression in a regenerative heat storage system such as a pebble rock bed. If the hot air leaving the adiabatic reservoir or the regenerative heat storage system is expanded directly in a turbine without the addition of fuel, then this system is referred to as a *no-fuel* system.

A no-fuel turbine system is illustrated in Fig. 2.6. The turbine may be composed of a number of stages or two (or more) separate turbines. An inlet temperature of 650 F has been assumed, which represents a maximum in commercially available compressors. Analysis of this system indicates that the required specific turbine flow rate is about twice that of the Huntorf system. Although the heat rate is zero, the size and cost of the turbine would be large. The size of an adiabatic reservoir or a regenerative heat storage system would also be large due to the greater airflow rate and the lower density of the hot air. Correspondingly, the required capacity of the compressors would increase by a factor of about two.

One method of using the hot air while keeping the specific turbine flow rate in the range of commercially available equipment is illustrated in

Fig. 2.7. In this system, the hot air replaces the combustion chamber of the high pressure turbine. Analysis of this system indicates that the specific turbine flow rate is about 8% higher, or of little effect, and the heat rate is about 50% lower, or significant, than for the Huntorf system.

Thus, a no-fuel system does not appear to be economically feasible. A recuperated system using one combustor, such as illustrated in Fig. 2.7 should be used. Although auxiliary fuel is required, a significant reduction in fuel consumption is achievable. In addition, the turbines for this type of system are within the range of commercially available equipment.

2.7 CONCLUSIONS

Based on the preliminary evaluation given, the following conclusions can be made.

1. An air turbine system is not practical for a CAES plant.
2. A system composed of one or more turbines connected in a series, which is preceded by a combustor and exhausts at 50 F, requires too large a specific turbine flow rate for further consideration.*
3. A system composed of one or more turbines connected in a series, which is preceded by a combustor and exhausts at 120 F, requires further investigation to determine whether it is economically feasible.
4. A system comprising two turbines, two combustors, and a recuperator offers numerous advantages and should be further investigated. Lowering the inlet temperature to 1000 F permits the use of less expensive materials for the turbine components, reduces the heat rate, and causes an increase of less than 30% in the specific turbine flow rate.
5. A no-fuel system appears to be impractical.
6. The use of the hot air from either an adiabatic reservoir or a regenerative heat storage system to replace one combustor in a two-turbine, two-combustor system has little effect on specific turbine flow rate but significantly reduces heat rate.

---

*Recall that a 120 F reservoir has been assumed throughout the evaluation.

3 DETAILED STUDY OF TURBINE SYSTEMS

Systems that offer incentives for further investigation can be generalized into the following configurations:

1. System composed of two turbines and two combustors.
2. System composed of two turbines, two combustors and a recuperator.
3. System composed of a turbine and combustor exhausting at a low temperature.

The first system is essentially the same as the Huntorf system. The second has a recuperator added to reduce the heat rate, and the third is a new system that eliminates the need of a recuperator and has a low heat rate.

In this section, the performance of each system configuration is examined to identify designs that are economically attractive for CAES plants. Performance was evaluated using the approximate method of analysis discussed in Appendix A.2. The following parameters were fixed: η_{HGT}, η_{LGT}, T_3, T_5, and p_1 (see Figs. 1.1 and A.1). The values considered for these parameters are;

$\eta_{HGT} = \eta_{LGT} = 0.80, 0.85, 0.90,$
$T_3 = 500 - 1100$ F,
$T_5 = 900 - 2000$ F,
$p_1 = 400, 550, 650$ psia, and
$T_1 = 120$ F.

The method of analysis used was to select an intermediate pressure, p_i, in the range of $0 < p_i < p_1$ and to calculate the corresponding specific turbine flow rate and heat rate.

Equations are given that can be used to estimate the optimum intermediate pressure corresponding to minimum specific turbine flow rate and heat rate. These equations are based on an air-standard analysis (see Appendix A.2).

3.1 SYSTEM COMPOSED OF TWO TURBINES AND TWO COMBUSTORS

3.1.1 Optimum Intermediate Pressure

Using an air-standard analysis and considering a fixed overall pressure ratio (i.e., $r_p = p_1/p_{atm}$), fixed turbine inlet temperatures (T_3 and T_5), and fixed turbine efficiencies (η_{HGT} and η_{LGT}), it can be demonstrated that an optimum intermediate pressure, p_i, which minimizes specific turbine flow rate, $\dot{m}'_a$, exists. Following the method presented in the analysis of conventional gas turbine systems, the optimum p_i can be determined from,

$$\frac{p_i}{p_{atm}} = \left[\frac{\eta_{HGT}\ T_5}{\eta_{LGT}\ T_3}\right]^{1.75} \sqrt{r_p}\ , \tag{1}$$

where:

p_i/p_{atm} = the pressure ratio across the low pressure turbine.

To illustrate the use of this equation consider the performance parameters of the Huntorf system. Equation 1 indicates that the optimum intermediate pressure is p_i = 157 psia. This figure compares favorably with the pressure of 165 psia designated as the intermediate pressure.

It can be demonstrated that an optimum p_i that minimizes heat rate does not exist. The governing equation that relates the intermediate pressure to heat rate is,

$$\dot{Q}' = C_1 - C_2\, p_i^c, \tag{2}$$

where:

$$C_1 = c_{pa}\left[\eta_{HGT}\ T_3 + \left(T_5 - T_2\right)\right],$$

$$C_2 = \frac{c_{pa}\eta_{HGT}\ T_3}{p_1^c}, \text{ and}$$

$$c = (k - 1)/k.$$

This equation indicates that $\dot{Q}'$ decreases as p_i increases; the smallest $\dot{Q}'$ corresponds to $p_i = p_1$.

To obtain the lowest $\dot{m}'_a$ and $\dot{Q}'$ suggests using as high an intermediate pressure as possible. Using Eq. 1, this expedient would correspond to making T_5/T_3 as large as possible.

3.1.2 Performance Evaluation

The following performance trends were observed during the study.

1. For a fixed HGT* inlet temperature (T_3), as the LGT** inlet temperature (T_5) decreases, the specific turbine flow rate increases and the heat rate decreases, both linearly.
2. For a fixed LGT inlet temperature, as the HGT inlet temperature increases, the specific turbine flow rate decreases and the heat rate increases but at a slow rate.
3. At fixed LGT and HGT inlet temperatures, the intermediate pressure for the lowest specific turbine flow rate does not correspond to the minimum heat rate. The heat rate does not have a minimum and decreases as the intermediate pressure increases.
4. At low intermediate pressures, the heat rate decreases sharply and then gradually decreases as the intermediate pressure increases.
5. As the intermediate pressure decreases from the pressure corresponding to the minimum specific turbine flow rate down to ambient pressure, both the specific turbine flow rate and heat rate increase.
6. As the ratio of the LGT and HGT inlet temperatures (i.e., T_5/T_3) increases, specific turbine flow rate is less affected by intermediate pressure (i.e., the plot of $\dot{m}_a'$ vs. p_i becomes flatter).
7. As the HGT inlet temperature increases, the minimum specific turbine flow rate occurs at lower intermediate pressures.
8. The turbine efficiencies have a significant influence on both the heat rate and specific turbine flow rate. The affect on heat rate becomes more significant as the intermediate pressure increases.

Trends 4 and 5 suggest that the intermediate pressure should be greater than or equal to the value corresponding to minimum specific turbine flow rate. This selection tends to minimize both specific turbine flow rate and heat rate.

The main significance of trends 1 and 2 is that the turbine inlet temperatures control specific turbine flow rate. In addition, heat rate is not appreciably affected by the HGT inlet temperature. Therefore, a possible turbine system option is to use the largest possible inlet temperature to each turbine, an approach suggested by Giramonti and Lessard.[5] Considering Eq. A.59 (App. A) which relates the mean cross-sectional area and, hence, the size of the turbine to its performance, an increase in the inlet temperature will slightly offset the reduction in size due to the decrease in mass flow

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*HGT corresponds to high pressure gas turbine.

**LGT corresponds to low pressure gas turbine.

rate. This consequence results because the mean area is proportional to the square root of absolute temperature but directly proportional to mass flow rate. However, the net result will be a decrease in turbine size. Thus, the advantages of this option are the decreases in turbine size, airflow rate, reservoir size, and capacity of the compressors; the disadvantages are increases in heat rate and manufacturing cost of the blading due to the necessary internal cooling to achieve the elevated temperature.

Figures 3.1, 3.2, and 3.3 illustrate the performance trends for these case studies. In these figures, specific turbine flow rate and heat rate are plotted against intermediate pressure for fixed operating parameters. For comparison purposes, an inlet pressure of 650 psia was selected.

The turbine inlet temperatures and the inlet system pressure, as shown in Fig. 3.1, are the same as those in the Huntorf system. The performance of the Huntorf system is approximated by the curves for η = 80%. Considering Fig. 3.1, the optimum p_i for η = 80% ($\eta_{HGT} = \eta_{LGT} = \eta$) is about 175 psia. This pressure compares favorably with 164 psia calculated from Eq. 1. For p_i > 175 psia, it can be seen that as p_i increases specific turbine flow rate increases at a slow rate and heat rate decreases more rapidly. For example, for η = 80% and p_i = 450 psia, the specific turbine flow rate is 4.6% greater and the heat rate is 8.3% less than at the optimum pressure of 175 psia.

Using Eq. A.59 (App. A), the ratio of the mean cross-sectional area of the LGT for p_i = 450 psia, compared to 175 psia, is 0.41. This suggests that a less expensive LGT per unit of power generated would result using a 30:1 pressure ratio. The effect on the total turbine system can be seen in Fig. 3.4. Increasing the intermediate pressure from p_i to p_i' has the effect of moving part of the original HGT from 4'-4 to 5'-A and moving the LGT from 5-6 to A-6'. Considering Eq. A.59, a larger turbine results for the process 5'-A but a smaller turbine results for the process A-6'. In addition, the original HGT process 3-4' does not change. Since the size and power output of the LGT is much greater than the HGT (e.g., for the Huntorf turbine system, about 195 MW are generated by the LGT compared to 95 MW by the HGT), it can be concluded that a reduction in the size of the complete system can be expected. Thus, the advantages of this turbine system option are decreases in sizes of the turbines and in the heat rate; the disadvantages are a slight increase in airflow rate and, therefore, similar slight increases in the size of the reservoir and capacity of the compressors.

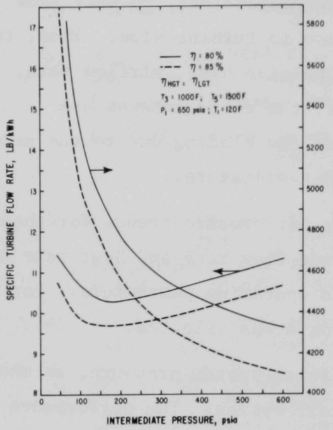

Fig. 3.1. System Performance for Configuration No. 1 with T_3 = 1000 F and T_5 = 1500 F

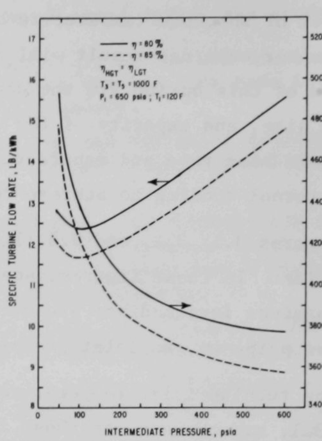

Fig. 3.2. System Performance for Configuration No. 1 with T_3 = T_5 = 1000 F

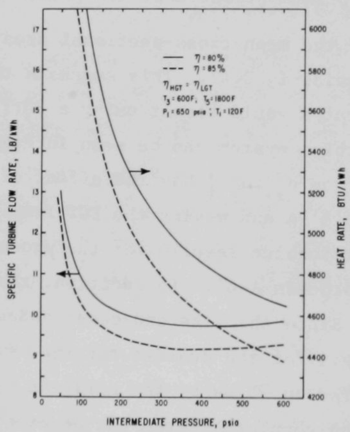

Fig. 3.3. System Performance for Configuration No. 1 with T_3 = 600 F and T_5 = 1800 F

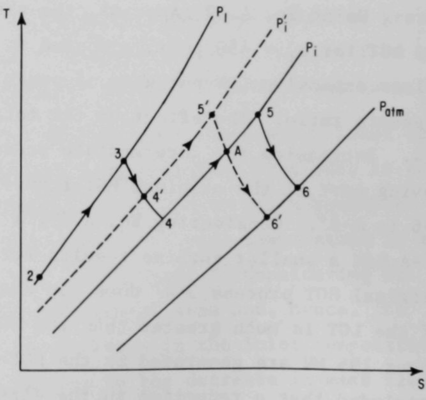

Fig. 3.4. Effect of Using a High Intermediate Pressure

As presented in the preliminary evaluation, reducing the LGT inlet temperature to 1000 F should be considered. The performance of the system for this condition is illustrated in Fig. 3.2. Comparing Figs. 3.1 and 3.2 for $\eta = 80\%$, reduction of the LGT inlet temperature to 1000 F reduces in turn the optimum pressure from 175 psia to 110 psia, increases the specific flow rate by about 20% and decreases the heat rate by about 11%. Calculating from Eq. A.59, the size will increase by only 4%. Thus, the advantages of this turbine system option are less expensive materials for turbine components and reduced heat rate; the disadvantages are a slight increase in turbine size and increased airflow rate resulting in increased reservoir size and increased capacity of the compressors.

Figure 3.3 shows the performance of a system that uses a large turbine inlet temperature ratio. As Eq. 1 suggests, the optimum intermediate pressure occurs at a high intermediate pressure (about 400 psia). Comparing the performance of this system with the system illustrated in Fig. 3.1 for $\eta = 80\%$, the optimum intermediate pressure is 400 psia compared to 175 psia, the specific turbine flow rate is about 5% less, and the heat rate is about the same. Thus, this system does not offer an advantage compared to a Huntorf-type system that uses commercially available equipment. Since new turbines would have to be developed, this system should not be further considered.

3.2 SYSTEM COMPOSED OF TWO TURBINES, TWO COMBUSTORS, AND A RECUPERATOR

The addition of a recuperator to a two-turbine, two-combustor system will significantly decrease heat rate but will have only a slight effect (in the order of a few percent) on specific turbine flow rate. Therefore, Eq. 1 can be used to predict optimum intermediate pressure, p_i, for minimum specific turbine flow rate. An optimum p_i exists for minimum heat rate, but the governing equation is complex and will not be given.

The following performance trends were observed during the study.

1. The performance trends for specific turbine flow rate do not change appreciably due to the addition of a recuperator.
2. Heat rate is considerably lowered with the addition of a recuperator.
3. When the two turbine inlet temperatures are close together (spread of around 200 F or less), the heat rate has a minimum; however, it is not a sharp one. At low intermediate pressures, heat rate increases as intermediate pressures decrease, but the rate of

increase is not as large as without a recuperator. As the intermediate pressure increases from low values, the heat rate remains fairly constant.

4. Increasing the spread of the turbine inlet temperatures, the heat rate increases more rapidly at low intermediate pressures. At higher intermediate pressures, the heat rate remains fairly constant and has a minimum close to the inlet pressure to the system (i.e., p_1).
5. The efficiency of the turbines has a lesser effect on heat rate than for a system without a recuperator.

Figures 3.5-3.7 illustrate the performance trends for a system with a recuperator. These figures suggest that intermediate pressure should be selected on the basis of minimum specific turbine flow rate. The addition of a recuperator tends to eliminate turbine options that are based on reducing heat rate. The high inlet temperature turbine option is more attractive with a recuperator because of the reduced heat rate. In contrast, the turbine option of using a reduced inlet temperature to the LGT (such as 1000 F or less) appears less attractive since the reduction in heat rate would be small. The high intermediate pressure option, such as using a 30:1 LGT, remains attractive since the main objective of the increased pressure is to reduce turbine size.

3.3 SYSTEM COMPOSED OF A TURBINE AND COMBUSTOR EXHAUSTING AT A LOW TEMPERATURE

For this system configuration, air from the reservoir is burned with fuel in a combustor and the products of combustion are expanded through a turbine and then exhausted at a low temperature. As discussed in Section 2, the exhaust temperature should be above about 50 F to avoid icing and/or condensation. The expansion process proceeds through the turbine without additional reheat so that a second combustor is eliminated. Due to the low exhaust temperature, a recuperator has no value and is not used.

The turbine for this system can consist of successive stages of blading enclosed by a housing or two or more separate turbines connected in a series by appropriate piping. One form of the latter is to use a high pressure gas turbine (HGT) followed by a low pressure gas turbine (LGT). This sequence permits the optimization of each turbine based on the desired performance and using presently available components. Another important feature is that

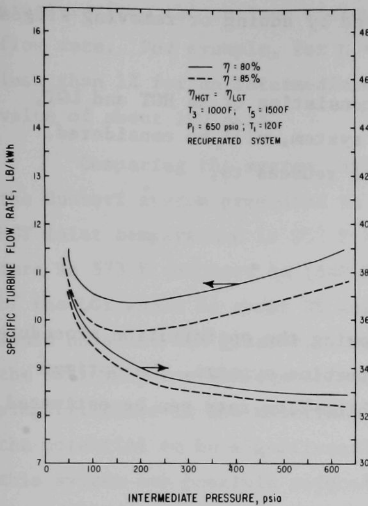

Fig. 3.5. System Performance for Configuration No. 2 with T_3 = 1000 F and T_5 = 1500 F

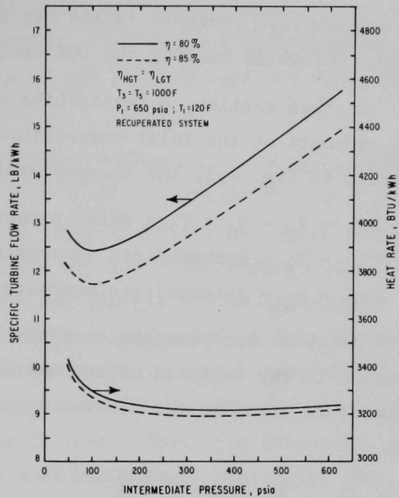

Fig. 3.6. System Performance for Configuration No. 2 with $T_3 = T_5$ = 1000 F

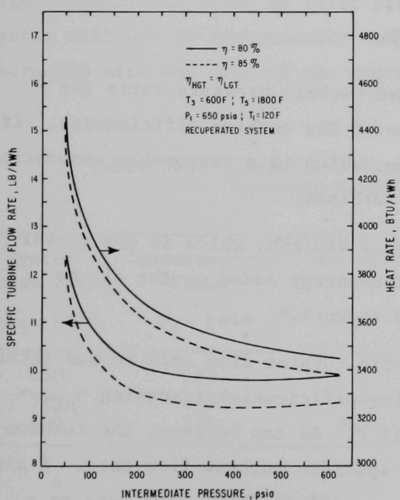

Fig. 3.7. System Performance for Configuration No. 2 with T_3 = 600 F and T_5 = 1800 F

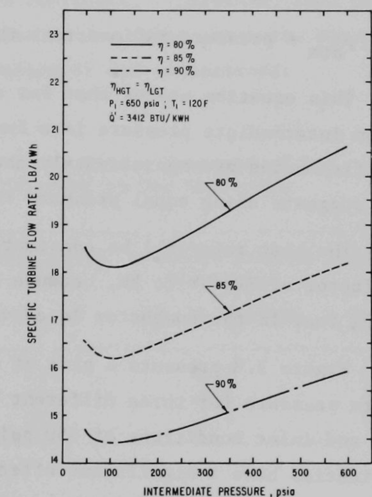

Fig. 3.8. System Performance for Configuration No. 3 with P_1 = 650 psia and T_1 = 120 F

different overall pressure ratios can be achieved by adding or removing stages from the HGT while keeping the LGT unchanged.

In this section, a two-turbine system, consisting of an HGT and LGT, which exhausts at the inlet temperature to the system, will be considered. Referring to Fig. A.1, the thermodynamic process reduces to,

$T_1 = T_2 = T_6 = T_7$ = inlet temperature,
$T_4 = T_5$ = intermediate temperature, and
$p_i = p_4$ = intermediate pressure.

Using an air-standard analysis and following the optimization procedure presented in the analysis of conventional gas turbine systems, the optimum intermediate pressure for minimum specific turbine-flow rate can be estimated from,

$$\frac{p_i}{p_{atm}} = \left[\frac{\eta_{LGT}\left(1 - \eta_{HGT}\right)}{\eta_{HGT}\left(1 - \eta_{LGT}\right)}\right]^{1.75} \sqrt{r_p} \tag{3}$$

where:

$r_p = p_1/p_{atm}$ = overall pressure ratio and

p_i/p_{atm} = pressure ratio across the LGT.

This equation states that for a fixed overall pressure ratio the optimum intermediate pressure is a function of the turbine efficiencies. If the efficiencies are approximately the same, which is a reasonable assumption, Eq. 3 suggests using equal pressure ratio turbines.

The heat rate will be constant at 3412 Btu/kWh, which is the conversion factor of Btu/hr to kW, because all the energy added to the air by burning fuel in the combustor is converted to work.*

Figure 3.8 presents a plot of specific turbine flow rate versus intermediate pressure for three different turbine efficiencies (assuming $\eta_{HGT} = \eta_{LGT}$) and inlet conditions of 650 psia, 120 F. As can be seen, the turbine efficiencies have a significant effect on specific turbine flow rate. A high efficiency system is needed to reduce the specific turbine flow rate to a reasonable level. Furthermore, it is possible to increase the intermediate

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*This theory is based on the assumption of negligible losses in the combustor, which has been used throughout this report.

pressure from the optimum without significantly affecting specific turbine flow rate. For example, for η = 90%, specific turbine flow rate increases by less than 1% for an intermediate pressure of 165 psia compared to the optimum value of about 100 psia.

Comparing the system performance for η = 90% and p_i = 165 psia with the Huntorf system presented in Section 2.1, $\dot{m}'_a/\dot{m}'_o = 1.25$, $\dot{Q}'/\dot{Q}'_o = 0.62$, the HGT inlet temperature is 957 F compared to 1000 F, and the LGT inlet temperature is 573 F compared to 1500 F. Using Eq. 62, the mean cross-sectional area of the LGT would be about 9% less than for the Huntorf system and the HGT would be about 23% greater. However, the main significance of this design is the extremely low temperatures in the LGT which will permit a substantial simplification in the materials for construction. Therefore, this system has the potential to be significantly reduced in cost. Thus, the advantages of this system are possible reduced turbine cost and reduced heat rate; the major disadvantage is increased airflow rate which results in increased reservoir size and increased capacity of the compressors.

Table 3.1 presents the performance of this type of system at three different inlet pressures. The data show that specific turbine flow rate significantly increases as inlet pressure decreases. Therefore, this type of system will not be economically competitive at low inlet pressures due to the increased size and cost of the storage reservoir and compressors.

Table 3.1. Overall Performance for a System Composed of Two Turbines in Series Preceded by One Combustor

System Inlet Press p_1, psia	Intermediate Press p_i, psia	HGT Inlet Temp. T_3, F	LGT Inlet Temp. T_5, F	Specific Turbine Flow Rate $\dot{m}'_a$, lb air/kWh	Optimum p_i[a] psia
650	165	957	573	14.5	98
550	165	906	573	15.6	90
400	165	813	573	17.9	77

$T_1 = T_6 = 120$ F

$\eta_{HGT} = \eta_{LGT} = 90\%$

$\dot{Q}' = 3412$ Btu/kWh

[a]Based on Eq. 3

4 CONCLUSIONS AND RECOMMENDATIONS

Turbine systems for proposed CAES plants are derived from state-of-the-art gas turbines and steam turbines. The main criteria for these systems is to use presently available components with proven reliability. The high pressure turbine (HGT) can consist of a slightly modified existing steam turbine design. The modification limits the maximum gas temperature to about 1000 to 1100 F. The low pressure turbine (LGT) can be an existing gas turbine that is being used in conventional gas turbine peaking units. Two different designs are available. One design uses internal blade air-cooling to permit inlet gas temperatures of 1800 to 2000 F. A second design, for gas temperatures below about 1600 F, uses simpler uncooled blading.

The use of a recuperator to reduce heat rate has been proposed for CAES turbine systems. Preliminary studies indicate that recuperators can be designed that are economically feasible for CAES application. The required recuperator design will be different from the design for a conventional gas turbine peaking 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. The 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 that were to be subjected to a large number of starting cycles, as anticipated in CAES plants.[20]

This study identifies several turbine systems that will require the development of new components. The systems can be categorized into three general configurations. The advantages and disadvantages of each system are summarized below. The turbine system for the Huntorf plant is used as a basis.

1. System Composed of Two Turbines and Two Combustors

<u>Option A -- High Turbine Inlet Temperatures</u>

Advantages: Decreased sizes of turbine and reservoir and decreased capacity of compressors.

Disadvantages: Increased heat rate, complexity of blading, and material cost for turbine components.

<u>Option B -- Large Pressure Ratio LGT</u>

Advantages: Decreased turbine size and heat rate.

Disadvantages: Slight increase in reservoir size and capacity of compressors.

Option C -- Low Inlet Temperature to LGT

Advantages: Less expensive materials for LGT; decreased heat rate.

Disadvantages: Slight increase in turbine size; increased reservoir size and capacity of compressors.

2. System Composed of Two Turbines, Two Combustors, and a Recuperator

Option D -- Option A with a Recuperator

Advantages: Decreased turbine and reservoir sizes; decreased heat rate; decreased capacity of compressors.

Disadvantages: Increased complexity of blading; capital cost of recuperator.

Option E -- Option B with a Recuperator

Advantages: Decreased turbine size; decreased heat rate (lower heat rate than Option B)

Disadvantages: Slight increase in reservoir size and capacity of compressors; capital cost of recuperator.

3. System Composed of a Turbine and Combustor Exhausting at a Low Temperature

Option F -- Turbine Composed of a HGT and LGT with the Exhuast Temperature Equal to the Inlet Temperature

Advantages: One combustor eliminated; decreased size of LGT; decreased heat rate (about the same as Options D and E); less expensive materials for LGT (less than Option C).

Disadvantages: Increased size of HGT; increased reservoir size; increased capacity of compressors; high efficiency blading necessary; applicable only to high pressure systems (e.g., $p_1 > 600$ psia).

High temperature turbines for conventional gas turbine peaking units are being extensively studied by turbine manufacturers. The Electric Power Research Institute (EPRI) is funding gas turbine research and development for two levels -- to improve the performance of available equipment and to develop high-temperature technology for future high efficiency designs.[21] The long-range goal is to develop reliable equipment that will operate at gas temperatures in the 2800 to 3000 F range, with 1986 as the target deadline. These high temperature turbines could be used for the LGT in Options A and D. However, high temperature turbines for HGT applications have received little attention.

Option C, which uses a reduced inlet temperature to the LGT, results in decreased heat rate and permits the use of less expensive materials for the LGT. However, if a recuperator is added, the resulting heat rate will be slightly lower than for a system using a standard inlet temperature to the LGT (e.g., 1500 F). But this reduction in heat rate does not offset the disadvantages of this option, for which reason a recuperator configuration was not considered.

The use of a high pressure ratio LGT in Options B and E is very attractive, and a cost reduction in the turbines appears possible. Using this concept, the LGT would supply the majority of the power output and the HGT would be tailored to the required overall pressure ratio of the system.

Option F represents a significant departure from the large power output gas turbines available. Due to the extremely low operating temperatures, the LGT resembles small industrial turbines. This option offers the potential for a significant cost reduction in the turbines.

Two levels of cost information are required to assess the options cited above. First, the development costs for the new turbines, combustors, and recuperators are needed. Second, assuming large-scale production, the capital costs of each system are required. In addition, capital cost information is needed for the equipment available. With this information, an engineering economics evaluation of possible systems can be made.

The required cost information necessitates a comparative evaluation of the turbines, combustors, and recuperators. This evaluation should be totally unbiased so that it does not merely reflect a particular manufacturer's design constraints. For example, some turbine manufacturers use internally cooled blading for the low pressure turbine and others do not.

It is therefore recommended that the study reported herein be extended to include an economic evaluation of turbine systems. This extension will require a coordinated effort involving input from equipment manufacturers. It is further recommended that a study be initiated to consider turbine systems for advanced concept CAES plants. One attractive candidate is a combined coal gasification-CAES system.

APPENDIX A ANALYSIS OF TURBINE SYSTEMS

This section presents analyses pertinent to the evaluation of possible CAES turbine systems. Given first is an analysis of conventional gas turbine systems being used as peaker units, because the turbines in these units are being considered for use as the low pressure turbines in proposed CAES plants. The method of determining the optimum pressure ratio for minimum specific turbine flow rate and heat rate is also described. This type of analysis will be used to determine the optimum intermediate pressure for two turbine CAES systems.

Next, four different methods of analysis of turbine systems for CAES plants are delineated. Three of the methods are conventional forms of analysis frequently used to evaluate thermodynamic systems. The fourth method, referred to as an *approximate analysis*, is a new method developed during the course of this study. If offers a simple but accurate computational procedure that can be used for turbine system evaluations. Through three case studies, the four methods of analysis are compared.

Finally, a method of estimating the relative size of turbines in proposed systems is developed to determine whether a cost advantage or penalty results from the use of a new turbine system.

A.1 CONVENTIONAL GAS TURBINE SYSTEMS

Conventional gas turbine systems for existing electric utility peaking applications are based on the open Brayton cycle. A schematic diagram of a peaking system using a recuperator is shown in Fig. A.1. The turbine and compressor are usually placed on the same shaft and the net power output is used to operate a generator. The optional recuperator is a heat exchanger used to preheat the air before it enters the combustion chamber, which results in reduced fuel consumption but an added system cost.

The thermodynamic cycle can be illustrated on a temperature, T, versus entropy, s, plot, where constant pressure, p, lines are drawn for reference. Figure A.2 illustrates such a plot for the system shown in Fig. A.1. This plot assumes a negligible pressure drop across the recuperator and the combustor. In a properly designed system, every effort is made to limit these pressure losses to less than a few psi.

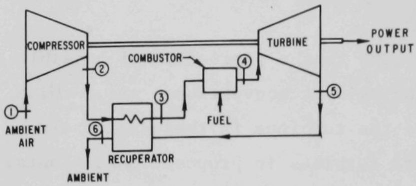

Fig. A.1. Schematic Diagram of a Conventional Gas Turbine System with a Recuperator

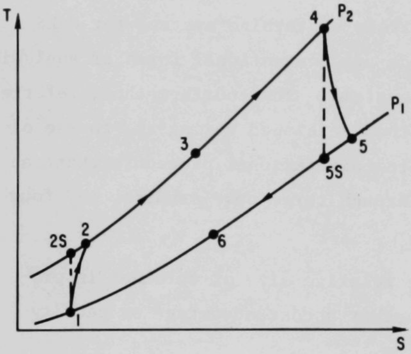

Fig. A.2. Temperature-Entropy Plot of a Conventional Gas Turbine System with a Recuperator

Detailed thermodynamic analysis of system performance must include the combustion process in the combustor and the expansion of the products of combustion through the turbine. A simplified air-standard analysis is frequently made based on the following assumptions.

1. Steady-state steady-flow.
2. Negligible change in kinetic energy and potential energy across each component.
3. Negligible heat transfer to the ambient from the compressor, turbine, combustor, recuperator, and piping.
4. Negligible pressure drop through the combustor, recuperator, and piping.
5. Working fluid is air that behaves as an ideal gas and is calorically perfect.

Assumption 5 seems justified since the fuel to air ratio is typically between 0.01-0.02 lb fuel/lb air for a conventional system.

Considering Fig. A.2 and using the assumptions above, the efficiency of the compressor and turbine can be formulated as follows:

η_C = compressor efficiency

$$= \frac{T_{2s} - T_1}{T_2 - T_1} = \frac{T_1\left\{r_p^c - 1\right\}}{T_2 - T_1}, \tag{A.1}$$

η_T = turbine efficiency

$$= \frac{T_4 - T_5}{T_4 - T_{5s}} = \frac{T_4 - T_5}{T_4\left(1 - r_p^{-c}\right)}, \tag{A.2}$$

where:

T = absolute temperature,

r_p = pressure ratio,
= p_2/p_1

c = $(k-1)/k$,

and k = specific heat ratio.

The effectiveness, ε, of the recuperator is defined as,

$$\varepsilon = \frac{T_3 - T_2}{T_5 - T_2}. \tag{A.3}$$

The thermal efficiency, η, of the system is defined as the ratio of the net work output to the heat input. In terms of temperatures,

$$\eta = \frac{\left(T_4 - T_5\right) - \left(T_2 - T_1\right)}{T_4 - T_3}. \tag{A.4}$$

Substituting Eqs. A.1, A.2, A.3 into Eq. A.4 and simplifying,

$$\eta = \frac{\eta_T \eta_C Z\left(1 - \frac{1}{y}\right) - (y - 1)}{\eta_C(1 - \varepsilon)(Z - 1) + \varepsilon Z \eta_C \eta_T\left(1 - \frac{1}{y}\right) - (1 - \varepsilon)(y - 1)}, \tag{A.5}$$

where:

$y = r_p^c$,

and $Z = T_4/T_1$,

If a recuperator is not used in the system, then $\varepsilon = 0$ and Eq. A.5 reduces to,

$$\eta = \frac{\eta_T \eta_C Z\left(1 - \frac{1}{y}\right) - (y - 1)}{\eta_C(Z - 1) - (y - 1)}. \tag{A.6}$$

The specific power output, P', of the system is equal to the net power output divided by the mass flow rate of air. In equation form,

$$P' = c_p\left[\left(T_4 - T_5\right) - \left(T_2 - T_1\right)\right]$$

$$= \frac{c_p T_1}{\eta_C}\left[\eta_T \eta_C Z\left(1 - \frac{1}{y}\right) - (y - 1)\right], \tag{A.7}$$

where:

c_p = specific heat at constant pressure.

This equation will be the same for a system with or without a recuperator.

From Eq. A.5 or A.6 it can be shown that for a fixed pressure ratio the thermal efficiency of the system increases as the turbine inlet temperature T_4 increases. This relationship can be most easily shown by considering a so-called *perfect* recuperator that corresponds to $\varepsilon = 1$. Equation A.5 reduces to,

$$\eta = 1 - \frac{y}{\eta_C \eta_T Z}. \tag{A.8}$$

The maximum allowable gas temperature T_4 is based on metallurgical limitations. Temperatures as high as 2000 F have been reported by turbine manufacturers, but reliable performance usually dictates about 1800 F as the upper limit.

Reported efficiencies of compressors and turbines are in the range of 80-90%. Equation A.8 illustrates the expected result that the higher the efficiency of either the turbine or compressor, the higher the overall system efficiency.

Since T_4 is based on metallurgical limitations and η_C and η_T are based on state-of-the-art equipment, the next question that must be asked is, What is the optimum pressure ratio? There are two possible choices to base the selection of the *best* pressure ratio. First of all, we may want to find the pressure ratio that will give the most work per lb of gas flowing, as this will tend to give the minimum size of turbine and compressor. This objective requires the determination of the value of y corresponding to the maximum P'. Second, and perhaps more commonly, we may want to find the ratio that will give the highest thermal efficiency; i.e., find the value of y at the maximum η.

Using ordinary calculus to maximize P' with respect to y in Eq. A.7, the resulting optimum pressure ratio is,

$$r_p \text{ for max } P' = (\eta_T \eta_C Z)^{\frac{k}{2k - 2}}. \tag{A.9}$$

Consider the following representative values for the parameters in the above equation.

$$\left.\begin{array}{l} T_4 = 1800 \text{ F} = 2260 \text{ R} \\ T_1 = 70 \text{ F} = 530 \text{ R} \end{array}\right\} Z = 4.26,$$

$$\eta_T = \eta_C = 0.85,$$

$$k = 1.4.$$

From Eq. A.9,

$$t_p \text{ max } P' = 7.15.$$

This optimum pressure ratio would apply to a system with or without a recuperator.

Consider now the thermal efficiency of a system with a recuperator. Using Eq. A.5 it can be shown that an optimum pressure ratio for maximum thermal efficiency does not exist; the thermal efficiency continually decreases as the pressure ratio increases. This opposition can be readily verified by examining Eq. A.8, which is for a perfect recuperator.

For a system without a recuperator Eq. A.6 must be used. Maximizing this function it follows that the optimum pressure ratio can be calculated from the equation,

$$r_p \text{ for max. } \eta \text{ (without a recuperator)} =$$

$$\left[\frac{A - \sqrt{A(A - B)(1 - B)}}{(A - B + 1)}\right]^{k/(k-1)}, \qquad (A.10)$$

where:

$$A = \eta_T \eta_C Z, \text{ and}$$

$$B = \eta_C (Z - 1) + 1.$$

Using the representative values of the parameters stated above,

$$r_p \text{ for max. } \eta \text{ (without a recuperator)} = 13.6.$$

Dusinberre and Lester[16] present a useful approximate equation for the optimum pressure ratio based on taking the heat supplied as $c_p(T_4 - T_{2s})$ instead of $c_p(T_4 - T_2)$. The resulting equation is,

r_p for max. η (without a recuperator) $\simeq$

$$\left[\frac{Z}{1+\sqrt{(Z-1)\left(\frac{1}{\eta_T\eta_C}-1\right)}}\right]^{k/(k-1)} \quad . \tag{A.11}$$

Evaluating this equation using the representative values presented above yields,

r_p for max. η (without a recuperator) $\simeq 11.49$

Thus, for the representative values of the parameters listed above, the pressure ratio for the minimum size (and cost) of the turbine and compressor is 7.15; whereas, for the maximum thermal efficiency the pressure ratio is 13.6. The most likely choice for the pressure ratio is somewhere between these extremes. Interestingly, the pressure ratio of gas turbines available for use in simple gas turbine peaking systems is about 10:1.

A.2 CAES TURBINE SYSTEMS

Four different methods of analysis that can be used to predict the performance of a turbine system in a CAES plant are given and evaluated in this section. These methods of analysis are referred to as an exact, air-table, air-standard, and approximate analyses. The exact analysis will be used as a reference analysis for comparison purposes. Both the air-table analysis and the air-standard analysis are conventional forms that are used to evaluate thermodynamic systems. The approximate analysis is a new method that is developed in this report. It offers a simplified calculational procedure compared to the more complicated exact analysis and air-table analysis without sacrificing computational accuracy.

The theoretical basis of each method, the governing equations, and the computational procedure are given. Three case studies are presented that are representative of possible CAES turbine systems. Using these case studies, the four methods of analysis are then compared.

A.2.1 Required Calculations

In analyzing the performance of a turbine system like the one illustrated in Fig. 2.1, the following parameters are considered as specified:

turbine efficiencies: η_{LGT}, η_{HGT},
recuperator effectiveness: ε,
temperatures: T_1, T_3, T_5,
pressures: p_1, p_4,
fuel composition, and
power output: P.

The turbine efficiencies are based on state-of-the-art values representative of available equipment. The turbine inlet gas temperatures usually represent the maximum allowable temperature with respect to metallurigical limitations. The pressure p_1 and the temperature T_1 are fixed by the reservoir storage conditions. The recuperator effectiveness is a function of the heat exchanger geometry.

The required outputs from such an analysis are the turbine outlet temperature (T_4 and T_6), the outlet temperatures from the recuperator (T_2 and T_7), the fuel consumption of each combustor, and the airflow rate from the reservoir. Basing the airflow rate and fuel consumption on one kilowatt (kW) of power generated leads to the use of the two overall system performance parameters -- specific turbine flow rate, $\dot{m}_a'$, and heat rate, $\dot{Q}'$.

A.2.2 Analysis Assumptions

In order to develop useful performance equations, the analysis will be restricted to the turbine system illustrated in Fig. 1.1. The analysis presented can be easily extended to a system composed of three turbines with two or three combustors, which is being considered for high reservoir pressures (e.g., see Ref. 22), or one turbine with one combustor, which has application to low reservoir pressures.

The following assumptions are made:

1. Steady-state steady-flow.
2. Negligible change in kinetic energy and potential energy across each component.
3. Negligible heat transfer to the ambient from the turbines, combustors, recuperator, and piping.
4. Negligible pressure drop through the combustors, recuperator, and piping.
5. Perfect gas mixture behavior.
6. Negligible dissociation in the combustors.

7. Complete combustion in the combustors.
8. Hydrocarbon fuel.

In addition to these general assumptions, it is also assumed that the pressure of the gas exiting the low pressure turbine is equal to atmospheric pressure with or without a recuperator in the system.

The various losses occurring in a combustor can be considered by using a combustion efficiency based on experimental data. One definition of this efficiency, suggested by Shepherd,[23] is to define efficiency as the ratio of the actual stagnation temperature rise of the gas to the theoretical rise determined by an energy balance. In equation form,

$$\eta_{comb} = \frac{\Delta T_{act}}{\Delta T_{ideal}} . \tag{A.12}$$

The use of this efficiency in an energy balance would take into account the losses neglected by assumptions 2 through 7 above. Dusinberre and Lester[16] illustrate the use of combustion efficiency in combustor analysis.

A.2.3 Methods of Analysis

Exact Analysis

In order to use available and proven turbine components, the design of the high pressure turbine is based on that of a steam turbine. The maximum inlet temperature (T_3) is usually selected in the range of 1000-1100 F. For temperatures in this range, the combustion process in combustor No. 1 requires over 500% theoretical air. In order to use the Gas Tables[24] to conduct a conventional analysis of this combustor, the large quantity of excess air necessitates a constituent chemical analysis of the combustion process. The combustion analysis of each combustor follows the treatment presented in most thermodynamics textbooks, e.g., see Van Wylen and Sonntag.[25]

To begin the analysis of combustor No. 1, the inlet temperature T_2 must be known. For a system with a recuperator, T_2 must be calculated from

$$\varepsilon = \frac{T_2 - T_1}{T_6 - T_1}. \tag{A.13}$$

The solution requires a knowledge of T_6, which cannot be determined until an energy balance is conducted on the low pressure turbine. A trial-and-error soluation must therefore be undertaken based on an initial estimate of T_6, using the air-standard analysis.

Selecting a hydrocarbon fuel and assuming complete combustion and negligible dissociation, the chemical equation for the combustion process can be written. The number of moles of air is an unknown in this equation. Since adiabatic combustion is assumed, an energy balance reduces to the fact that the enthalpy of the reactants at T_2 must equal the enthalpy of the products at T_3. Expressed in equation form,

$$\sum_R n_R \bar{h}_R = \sum_P n_p \bar{h}_p, \tag{A.14}$$

where:

$$h = \bar{h}_f^\circ + \Delta\bar{h}. \tag{A.15}$$

The value of each molal enthalpy, $\bar{h}$, can be determined using the Gas Tables and tables of the enthalpy of formation of different substances. Substituting into Eq. A.14 the number of moles of reactants and products from the chemical equation and the molal enthalpies, an equation results having one unknown -- the number of moles of air, n_a. From the calculated value of n_a, the fuel-air ratio (f_1/a) can be determined easily.

The enthalpy of the products, h_3, can be calculated from the equation,

$$h_3 = \sum_i x_{3i} \cdot \frac{\bar{h}_{3i}}{M_{3i}}. \tag{A.16}$$

The mole fractions, x_{3i}, can be calculated from the chemical equation.

The temperature T_4 is needed to analyze combustor No. 2. From the definition of turbine thermal efficiency,

$$\eta_{HGT} = \frac{h_3 - h_4}{h_3 - h_{4s}}. \tag{A.17}$$

Calculating h_4 from the above equation is necessary in order to determine T_4. The enthalpy h_{4s}, which is the enthalpy due to isentropic expansion to the intermediate pressure p_4 (see the T-s diagram of Fig. A.3), must be calculated first. This calculation requires the determination of T_{4s}. From the definition of relative pressure,

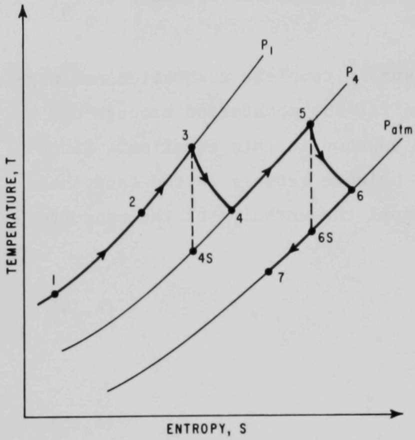

Fig. A.3. Temperature-Entropy Plot of a CAES Turbine System

$$p_{r4s} = p_{r3} \cdot \frac{p_{4s}}{p_3}. \qquad (A.18)$$

The value of p_{r3} can be determined from the equation,

$$\ln p_{r3} = \sum_i x_{3i} \ln p_{r3i}, \qquad (A.19)$$

where:

x_{3i} = constituent mole fractions of the products from combustor No. 1, and

p_{r3i} = corresponding relative pressure from the Gas Tables.

Similarly,

$$\ln p_{r4s} = \sum_i x_{4si} \ln p_{r4si}, \qquad (A.20)$$

where:

$x_{4si} = x_{3i}$ since the mole fractions of the gas do not change through the turbine.

Taking the natural logarithm of Eq. A.18 and substituting Eqs. A.19 and A.20 yields,

$$\sum_i x_{3i} \ln p_{r3i} + \ln \frac{p_{4s}}{p_3} = \sum_i x_{3i} \ln p_{r4si}. \qquad (A.21)$$

To evaluate the relative pressures p_{r4si}, the temperature T_{4s} must be known. Thus, by trial-and-error, the temperature T_{4s} which satisfies Eq. A.21 can be determined. The enthalpy h_{4s} can then be determined from the equation

$$h_{4s} = \sum_i x_{3i} \frac{\overline{h}_{4si}}{M_{3i}}. \qquad (A.22)$$

Using the calculated values of h_3 and h_{4s} in Eq. A.17, the enthalpy h_4 can be determined. By trial-and-error, T_4 can be determined from,

$$h_4 = \sum_i x_{3i} \frac{\bar{h}_{4i}}{M_{3i}}. \tag{A.23}$$

An evaluation of combustor No. 2 can now be made. The analysis of this combustor and the low pressure turbine follows the method presented for combustor No. 1 and the high pressure turbine.

To complete the analysis, the specific turbine-flow rate, $\dot{m}_a'$, and the heat rate, $\dot{Q}'$, are required. From an energy balance on the turbines,

$$\dot{m}_a' = \frac{1}{\left(1 + \frac{f_1}{a}\right)\left(h_3 - h_4\right) + \left(1 + \frac{f_1}{a} + \frac{f_2}{a}\right)\left(h_5 - h_6\right)}, \tag{A.24}$$

and

$$\dot{Q}' = \dot{m}_a'\left(\frac{f_1}{a} + \frac{f_2}{a}\right) \cdot LHV_o. \tag{A.25}$$

Air-Table Analysis

The air-table analysis assumes that the gas flowing through the turbine system is air. This assumption seems justified since the fuel-air ratio for the combustors in a plant such as presented in Fig. 1.1 is typically about 0.01 lb fuel/lb air. The variation of the specific heats with temperature are incorporated in the analysis by using thermodynamic data from Table 1 (air at low pressures) of the Gas Tables. Consistent with the assumption of air as the working fluid, the combustion process in each combustor is treated as a heat source that merely heats the air flowing through the combustor.

In order to begin the analysis of combustor No. 1, the inlet temperature T_2 must be known. As in the exact analysis method, if a recuperator is used in the system, a trial-and-error solution must be used based on an initial estimate of this temperature. This estimate can be made most easily using the air-standard method of analysis.

From an energy balance on each combustor, the fuel-air ratios can be calculated from,

$$\frac{f_1}{a} = \frac{h_3 - h_2}{LHV_o} \tag{A.26}$$

and

$$\frac{f_2}{a} = \frac{h_5 - h_4}{LHV_o}. \tag{A.27}$$

The enthalpies h_2, h_3 and h_5 can be obtained from air tables as a function of the known temperatures. The enthalpy h_4 and the corresponding temperature T_4 require an analysis of the high pressure turbine.

The enthalpy h_4 can be determined from Eq. A.17, which defines the efficiency of the high pressure turbine. Using Eq. A.18, the relative pressure p_{r4s} can be calculated where p_{r3} is determined from the air tables at the temperature T_3. Again, from the air tables, the enthalpy corresponding to the relative pressure p_{r4s} is h_{4s}. Similarly, entering the air tables with the calculated value of h_4, gives the corresponding temperature T_4.

A similar analysis of the low pressure turbine yields the enthalpy h_6 and the corresponding temperature T_6. The analysis is completed by calculating the specific turbine flow rate from Eq. A.24 and the heat rate from Eq. A.25.

Air-Standard Analysis

An air-standard analysis is commonly used to analyze turbine systems for CAES plants. This analysis assumes that the working fluid is air that behaves as an ideal gas and is calorically perfect. Li,[26] Glendenning,[27] and Hobson et al, [28] have presented evaluations of CAES plants based on an air-standard analysis. This type of analysis is simple and does not require thermodynamic property tables. Dusinberre and Lester[16] present an air-standard analysis of the basic gas turbine cycle that can be applied to turbine systems for CAES.

Using the air-standard analysis, the turbine outlet temperatures T_4 and T_6 can be determined directly from the equations,

$$\eta_{HGT} = \frac{T_3 - T_4}{T_3\left[1 - \left(\frac{p_4}{p_1}\right)^{(k-1)/k}\right]}, \tag{A.28}$$

and

$$\eta_{LGT} = \frac{T_5 - T_6}{T_5\left[1 - \left(\frac{p_{atm}}{p_4}\right)^{(k - 1)/k}\right]}, \tag{A.29}$$

where:

$k = 1.4.$

These temperatures are the same regardless of whether a recuperator is used in the system; thus, a trial-and-error solution is avoided.

The recuperator outlet temperatures T_2 and T_7 can be determined by combining the definition of effectiveness presented in Eq. A.13 with an energy balance of the recuperator. The air-standard analysis (and the air-table method of analysis) assumes that the gas flowing throughout the system is air. Thus it follows that the mass flow rate of air into the recuperator from the reservoir is equal to the gas (which is assumed to be air) flowing out of the low pressure turbine and into the recuperator. An energy balance of the recuperator reduces to

$$\left(T_6 - T_7\right) = \left(T_2 - T_1\right). \tag{A.30}$$

Combining Eqs. A.13 and A.30,

$$T_2 = (1 - \varepsilon)T_1 + \varepsilon T_6 \tag{A.31}$$

and

$$T_7 = (1 - \varepsilon)T_6 + \varepsilon T_1. \tag{A.32}$$

An energy balance of each combustor yields the following equations for the fuel-air ratios:

$$\frac{f_1}{a} = \frac{c_{pa}\left(T_3 - T_2\right)}{LHV_o}, \tag{A.33}$$

and

$$\frac{f_2}{a} = \frac{c_{pa}\left(T_5 - T_4\right)}{LHV_o} \tag{A.34}$$

where:

$$c_{pa} = 0.24 \text{ Btu}/(\text{lb}_m - \text{R}).$$

From an energy balance of the power turbines, the specific turbine flow rate can be determined from,

$$\dot{m}'_a = \frac{1}{c_{pa}\left(T_3 - T_4\right) + c_{pa}\left(T_5 - T_6\right)}. \quad (A.35)$$

The heat rate, $\dot{Q}'$, can be determined from Eq. A.25.

Approximate Analysis

The purpose of the approximate analysis is to provide a simple but accurate computational method that does not require the use of Gas Tables for solution. For simplicity, the analysis is modeled after the air-standard method.

As in the exact analysis and the air-table analysis, the use of a recuperator in the turbine system results in a trial-and-error analysis due to the unknown temperature T_2. An initial estimate of this temperature can be made using the air-standard analysis.

From an energy balance of combustor No. 1, the following equation results:

$$\dot{m}_a\left(h_2 - h'_2\right) + \dot{m}_{f1}\left(h_{f1} - h'_{f1}\right) + \dot{m}_{f1}\left(LHV_o\right) = \left(\dot{m}_a + \dot{m}_{f1}\right)\left(h_3 - h'_3\right), \quad (A.36)$$

where the primed terms refer to the reference temperature T_o for the lower heating value of the fuel. Dividing Eq. A.36 by $\dot{m}_a$ and solving for the fuel-air ratio (i.e., $f_1/a = \dot{m}_{f1}/\dot{m}_a$),

$$\frac{f_1}{a} = \frac{\left(h_3 - h_2\right) - \left(h'_3 - h'_2\right)}{LHV_o + \left(h_{f1} - h'_{f1}\right) - \left(h_3 - h'_3\right)}. \quad (A.37)$$

The enthalpy difference $(h_{f1} - h'_{f1})$ will be much less than LHV_o. In addition, the enthalpy difference $(h'_3 - h'_2)$ will be very small compared to $(h_3 - h_2)$. Thus Eq. A.37 reduces to,

$$\frac{f_1}{a} = \frac{h_3 - h_2}{LHV_o - \left(h_3 - h'_3\right)}. \quad (A.38)$$

The following approximations are now made:

$$h_3 - h_2 \simeq c_{pa3}\left(T_3 - T_2\right), \tag{A.39}$$

and

$$h_3 - h_3' \simeq c_{pa3}\left(T_3 - T_o\right), \tag{A.40}$$

where:

c_{pa3} = the specific heat at constant pressure of air at T_3.

The value of c_{pa3} can be determined from property tables or calculated using an empirical equation; e.g., Sweigert and Beardsley[29] suggest

$$c_{pa} = 0.32689 - \frac{119.78}{T} + \frac{40{,}041}{T^2}, \tag{A.41}$$

where:

c_{pa}[Btu/(lb$_m$ - R)] and T[R].

Substituting Eqs. A.39 and A.40 into Eq. A.38 yields,

$$\frac{f_1}{a} = \frac{c_{pa3}\left(T_3 - T_2\right)}{LHV_o - c_{pa3}\left(T_3 - T_o\right)}. \tag{A.42}$$

In order to determine the outlet temperature T_4, it would be desirable to use an equation similar to Eq. A.28 with the substitution of an apparent value of the specific heat ratio, k, which takes into account the changing temperature of the gas and its composition. It will be demonstrated that a suitable choice is to base k on the highest temperature T_3 so that,

$$\eta_{HGT} = \frac{T_3 - T_4}{T_3\left[1 - \left(\frac{P_4}{P_1}\right)^{(k_3 - 1)/k_3}\right]}. \tag{A.43}$$

The value of k_3 can be calculated from the equation,

$$k = \frac{c_{pa}}{c_{pa} - 0.06856} \tag{A.44}$$

where:

0.06856 [Btu/lb$_m$] = gas constant of air.

Once T_4 is calculated, combustor No. 2 can be analyzed. From an energy balance of combustor No. 2,

$$\left(\dot{m}_a + \dot{m}_{f1}\right)\left(h_4 - h_4'\right) + \dot{m}_{f2}\left(h_{f2} - h_{f2}'\right) + \dot{m}_{f2}\left(LHV_o\right) = \left(\dot{m}_a + \dot{m}_{f1} + \dot{m}_{f2}\right)\left(h_5 - h_5'\right) \tag{A.45}$$

Dividing this equation by $\dot{m}_a$ and simplifying,

$$\frac{f_2}{a} = \frac{\left(1 + \frac{f_1}{a}\right)\left[\left(h_5 - h_4\right) - \left(h_5' - h_4'\right)\right]}{LHV_o + \left(h_{f2} - h_{f2}'\right) - \left(h_5 - h_5'\right)}. \tag{A.46}$$

As for combustor No. 1,

$$\left(h_5' - h_4'\right) << \left(h_5 - h_4\right) \text{ and } \left(h_{f2} - h_{f2}'\right) << LHV_o.$$

Equation A.46 now becomes,

$$\frac{f_2}{a} = \frac{\left(1 + \frac{f_1}{a}\right)\left(h_5 - h_4\right)}{LHV_o - \left(h_5 - h_5'\right)}. \tag{A.47}$$

The enthalpy differences will be approximated as,

$$\left(h_5 - h_4\right) \simeq c_{pa5}\left(T_5 - T_4\right), \tag{A.48}$$

and

$$\left(h_5 - h_5'\right) \simeq c_{pa5}\left(T_5 - T_4\right). \tag{A.49}$$

Substituting Eqs. A.48 and A.49 into Eq. A.47,

$$\frac{f_2}{a} = \frac{\left(1 + \frac{f_1}{a}\right)c_{pa5}\left(T_5 - T_4\right)}{LHV_o - c_{pa5}\left(T_5 - T_o\right)}. \tag{A.50}$$

The outlet temperature T_6 can now be determined. Following the method for combustor No. 1, T_6 can be determined from the equation,

$$\eta_{LGT} = \frac{T_5 - T_6}{T_5\left[1 - \left(\frac{p_{atm}}{p_4}\right)^{\left(k_5 - 1\right)/k_5}\right]}, \tag{A.51}$$

where k_5 is calculated from Eq. A.44 at the temperature T_5.

To complete the analysis, the specific turbine flow rate and the heat rate must be determined. The specific turbine flow rate can be determined from Eq. A.24 once the enthalpies h_3, h_4, h_5 and h_6 are determined. It is demonstrated in the next section that these enthalpies can be closely approximated as,

$$h \simeq c_{pa} T, \tag{A.52}$$

where c_{pa} is determined at the temperature T.

The basis of this approximation is that the specific heat of the gas flowing through the turbine system is a function of both its temperature and chemical composition. As in the air tables, the enthalpy of air at a given temperature T is determined from,

$$h = \int_o^T c_{pa} dT \tag{A.53}$$

where:

c_{pa} = function of T.

This integration has the effect of averaging the influence of temperature on specific heat at constant pressure over the temperature range of 0 to T. Burning a hydrocarbon fuel in a combustor has the additional effect of increasing c_p due to the higher specific heats of the products.

Finally, once $\dot{m}'_a$ is determined, the heat rate, $\dot{Q}'$, can be calculated from Eq. A.25.

A.2.4 Comparison of Methods of Analysis

To compare the four methods of analysis presented, three case studies were conducted. The base data for each case study are given in Table A.1. Case A represents a turbine system similar to the Huntorf system. The Case B system has an 1800 F inlet temperature to the low pressure turbine. This temperature represents a realistic maximum gas temperature for available gas turbines. The Case C system represents a possible high temperature and high efficiency system at a pressure slightly reduced from those in Cases A and B. The inlet temperature T_2 = 800 F represents a temperature that could result from the use of a recuperator in the system.

Table A.1. Data for Case Studies

	Case A	Case B	Case C
T_2, F	100	200	800
T_3, F	1000	1100	1000
T_5, F	1540	1800	2000
p_1, psia	650	650	580
p_4, psia	165	165	185
T_f, F	77	77	77
Fuel	Methane	Methane	Methane
η_{HGT}, %	82	85	90
η_{LGT}, %	80	80	85

Note: Subscripts are relative to Fig. 1.1.

The accuracy of the enthalpy approximation used in the approximate method of analysis is illustrated in Table A.2 for the three case studies. In this table, the enthalpies h_4 and h_6 are based on the exact temperatures. The table indicates that the individual enthalpies are within a few percent of the exact values. However, the enthalpy differences, which are used in the calculation of $\dot{m}_a'$ and $\dot{Q}'$, are in greater error. The enthalpy difference $h_3 - h_4$ is 11-15% in error; whereas, the difference $h_5 - h_6$ is 3-6% in error.

Tables A.3, A.4 and A.5 compare the four methods of analysis for the three case studies. The % errors are based on the exact method of analysis. The air-table analysis and air-standard analysis both result in the same calculational trends. They underestimate the fuel-air ratio of both combustors, with the largest error resulting in the calculation for the second combustor. The outlet turbine temperatures are underestimated. The calculated specific turbine flow rates are greater than the exact values and the calculated heat rates are lower than the exact values.

The air-table method of analysis is in relatively close agreement with the exact analysis except in the calculation of the fuel-air ratio of the second combustor. The air-standard analysis results in larger errors than the air-table method. Considering the three case studies, the air-standard analysis underestimates the fuel-air ratio of the first combustor by 8-13%,

Table A.2. Comparison of Exact Enthalpies with Approximate Values

Ethalpies & Enthalpy Differences (Btu/lb_m)	Case A			Case B			Case C		
	Exact Value	Approx. Value	% Error	Exact Value	Approx. Value	% Error	Exact Value	Approx. Value	% Error
h_3	374.22	383.98	+2.6	402.20	415.90	+3.4	393.32	415.90	+5.7
h_4	277.23	276.32	-0.3	293.26	293.93	+0.2	297.69	305.86	+2.7
h_5	547.91	553.6	+1.0	634.59	636.64	+0.3	691.41	700.61	+1.3
h_6	337.94	329.9	-2.4	394.16	387.99	-1.6	404.43	403.44	-0.2
$h_3 - h_4$	96.99	107.66	+11.0	108.94	121.97	+12.0	95.63	110.04	+15.1
$h_5 - h_6$	209.97	223.1	+6.3	240.43	248.65	+3.4	286.98	297.17	+3.6

Notes: Subscripts are relative to Fig. 1.1

The approximate values for h_4 and h_6 are based on the exact temperature

Enthalpies are based on 0 R reference.

Table A.3. Solution Comparisons for Case A

	Exact Analysis		Air-Table Analysis		Air-Standard Analysis		Approximate Analysis	
	Value	% Error	Value	% Error	Value	% Error	Value	% Error
$\frac{f_1}{a}$, $\frac{\text{lb fuel}}{\text{lb air}}$	0.01087	0	0.01045	-3.9	0.01005	-7.5	0.0113	+2.4
$\frac{f_2}{a}$, $\frac{\text{lb fuel}}{\text{lb air}}$	0.01231	0	0.01119	-9.1	0.01015	-17.5	0.01194	-3.0
T_4, R	1100	0	1093	-0.6	1071	-2.6	1100	0
T_6, R	1285	0	1261	-1.9	1201	-7.0	1279	-0.5
$\dot{m}_a'$, $\frac{\text{lb air}}{\text{kWH}}$	10.91	0	11.46	+5.0	11.97	+9.7	10.10	-5.5
$\dot{Q}'$, $\frac{\text{Btu}}{\text{kWH}}$	5438	0	5310	-2.4	5199	-4.4	5001	-8.0

Note: Subscripts are relative to Fig. 1.1.

Table A.4. Solution Comparisons for Case B

	Exact Analysis		Air-Table Analysis		Air-Standard Analysis		Approximate Analysis	
	Value	% Error	Value	% Error	Value	% Error	Value	% Error
$\frac{f_1}{a}$, $\frac{\text{lb fuel}}{\text{lb air}}$	0.01104	0	0.01056	-4.3	0.01005	-9.0	0.01130	+2.4
$\frac{f_2}{a}$, $\frac{\text{lb fuel}}{\text{lb air}}$	0.01552	0	0.01384	-10.8	0.01240	-20.1	0.01491	-3.9
T_4, R	1160	0	1155	-0.4	1127	-2.8	1179	+1.6
T_6, R	1470	0	1437	-2.2	1357	-7.7	1458	-0.8
$\dot{m}'_a$, $\frac{\text{lb air}}{\text{kWH}}$	9.56	0	10.13	+6.0	10.64	+11.3	9.44	-1.3
$\dot{Q}'$, $\frac{\text{Btu}}{\text{kWH}}$	5460	0	5315	-2.7	5135	-6.0	5320	-2.6

Note: Subscripts are relative to Fig. 1.1.

Table A.5. Solution Comparisons for Case C

	Exact Analysis		Air-Table Analysis		Air-Standard Analysis		Approximate Analysis	
	Value	% Error	Value	% Error	Value	% Error	Value	% Error
$\frac{f_1}{a}$, $\frac{\text{lb fuel}}{\text{lb air}}$	0.00378	0	0.00365	-3.5	0.00330	-12.7	0.00377	-0.3
$\frac{f_2}{a}$, $\frac{\text{lb fuel}}{\text{lb air}}$	0.01968	0	0.01600	-18.7	0.01441	-26.8	0.01731	-12.0
T_4, R	1200	0	1195	-0.4	1169	-2.6	1191	-0.8
T_6, R	1520	0	1482	-2.5	1382	-7.8	1506	-0.9
$\dot{m}'$, $\frac{\text{lb air}}{\text{kWH}}$	8.76	0	9.21	+5.1	9.68	+10.5	8.10	-7.5
$\dot{Q}'$, $\frac{\text{Btu}}{\text{kWH}}$	4420	0	3891	-12.0	3684	-16.7	3672	-16.9

Note: Subscripts are relative to Fig. 1.1.

underestimates the outlet temperature from the low pressure turbine by 7-8%, and overestimates the specific turbine flow rate by 10-11%. The relatively close agreement of the specific turbine flow rates occurs because the specific heat at constant pressure is underestimated and the temperature difference across each turbine is overestimated so that the enthalpy differences are reasonably approximated.

Interestingly, the predicted heat rates using the air-standard analysis agree closely with the air-table analysis and the approximate analysis. This occurs because the heat rate is directly proportional to the product of the specific turbine flow rate and the sum of the two fuel-air ratios. The specific turbine-flow rate is overestimated and the fuel-air ratios are underestimated.

The approximate analysis more closely predicts the fuel-air ratios than the air-table analysis. The agreement for the second combustor is significantly improved. For example, for Cases A and B, the approximate analysis underestimates the fuel-air ratio of combustor No. 2 by 3-4%; whereas, the air-table analysis underestimates the fuel-air ratio by 9-11%. In addition, the outlet temperature T_6 is most closely predicted by the approximate analysis. Both the specific turbine flow rates and the heat rates are underestimated using the approximate analysis. In general, the predictive accuracy of the approximate analysis is about the same as for the air-table analysis.

The accuracy of the prediction of fuel-air ratios and, consequently, heat rates, can be improved for both the air-table analysis and the air-standard analysis by including in the combustor energy balances the enthalpy difference between the exiting air and the reference temperature T_o for the lower heating value of the fuel. This device amounts to subtracting the reference enthalpy difference from LHV_o in the equations for f_1/a and f_2/a. For example, Eq. A.27 for the air-table analysis becomes,

$$\frac{f_1}{a} = \frac{h_3 - h_2}{LHV_o - \left(h_3 - h_o\right)}. \qquad (A.54)$$

Table A.6 illustrates the results of including this reference enthalpy difference in the calculations for Case A.

Table A.6. Analysis of Case A Including Reference Enthalpy Difference in Combustor Energy Balance

	Air-Table Analysis		Air-Standard Analysis	
	% Error (from Table A.3)	% Error Incl. Ref. Enthalpy Difference	% Error (from Table A.3)	% Error Incl. Ref. Enthalpy Difference
$\frac{f_1}{a}$, $\frac{\text{lb fuel}}{\text{lb air}}$	-3.9	-2.8	-7.5	-6.6
$\frac{f_2}{a}$, $\frac{\text{lb fuel}}{\text{lb air}}$	-9.1	-7.5	-17.5	-14.4
$\dot{Q}'$, $\frac{\text{Btu}}{\text{kWH}}$	-2.4	-0.5	-4.4	-2.1

A.2.5 Conclusions

The exact method of analysis results in a tedious calculational procedure that requires the use of the Gas Tables and property tables to determine the thermodynamic properties of air and the constituent combustion gases. The analysis requires numerous trial-and-error calculations to predict the required performance parameters.

The air-table method of analysis is simpler than the exact analysis but still requires the use of air-tables for the thermodynamic properties. Interpolation of property values are required in the solution. The accuracy of the calculations are in relatively close agreement with the exact values.

The air-standard analysis treats air as an ideal gas that is calorically perfect. The analysis is extremely simple and thermodynamic property tables are not required. Using this method of analysis, the fuel-air ratio of each combustor and the turbine outlet temperatures are not accurately predicted (e.g., errors of 10-30%). This inaccuracy is particularly true of the calculation of the fuel-air ratio of the second combustor, where errors of 15-30% result. However, the overall system performance parameters -- specific turbine flow rate and heat rate -- are in closer agreement with the exact values. This agreement is due to the compensating errors of overestimation of the temperature decrease across each turbine and underestimation of the specific heats of the working gas. The general calculational accuracy of this method is less than that of the air-table analysis method.

The predictive accuracy of the approximate analysis is about the same as that of the air-table analysis. It also more closely predicts the fuel-air ratios of each combustor and the turbine outlet temperatures than both the air-table analysis and the air-standard analysis. The specific turbine flow rate and the heat rate are generally predicted with about the same accuracy as the air-table analysis.

The calculational procedure of the approximate analysis is slightly more complicated than the air-standard analysis. Additional terms are included in the equations to improve the predictive accuracy. This method requires the specific heat at constant pressure and the specific heat ratio of air as a function of temperature. These data can be obtained from simple property tables or empirical equations.

All four analytical methods lend themselves readily to rapid computer solution. The exact method will require a sophisticated computer procedure because of the complexity of programming the Gas Tables and the trial-and-error nature of the solution. Turbine manufacturers have already developed computer programs that can be used to analyze turbines for CAES plants. However, these proprietary programs are usually unavailable for general use.

For engineers in the electric utility industry who are investigating the feasibility and design of CAES plants, the approximate method may be more suitable. It lends itself to simple but relatively accurate hand calculations and can be easily incorporated into overall plant optimization studies.

A.3 ESTIMATING TURBINE SIZE

Specific turbine flow rate (or the inverse, specific power) is commonly used to relate overall turbine size to system performance; it is said to be directly proportional to size. Using an air-standard analysis for simplicity, specific turbine flow rate can be expressed as,

$$\dot{m}'_a = \frac{1}{T_{in}\eta_T c_{pa}\left(1 - r_p^{-c}\right)}$$

where:

T_{in} = inlet absolute temperature.

Thus, for a fixed pressure ratio, r_p, across the turbine, $\dot{m}'_a$ is inversely proportional to T_{in}. The term $(1 - r_p^{-c})$ increases with increasing r_p so that $\dot{m}'_a$ also varies inversely with r_p. Note that $\dot{m}'_a$ does not depend upon the inlet pressure (p_{in}).

The mass flow rate of air through the turbine can be expressed as,

$$\dot{m}_a = \rho V A_c,$$

$$= \frac{p_{in}}{RT_{in}} VA_c, \tag{A.56}$$

where:

V = axial velocity,

A_c = mean cross-sectional flow area (in a plane perpendicular to the axis), and

R = gas constant of air.

This equation is written for a typical or mean cross-sectional flow area that characterizes the size of the turbine.

Operating on Eq. A.56,

$$\dot{m}_a = \frac{p_{in}}{RT_{in}} \cdot \frac{V\sqrt{RT_{in}}\, A_c}{\sqrt{RT_{in}}},$$

$$= \frac{p_{in} Ma A_c}{\sqrt{RT_{in}}}, \tag{A.57}$$

where:

Ma = Mach number based on the axial velocity.

Solving for A_c,

$$A_c = \left(\frac{\sqrt{R}}{Ma}\right) \cdot \frac{\sqrt{T_{in}}\, \dot{m}_a}{p_{in}}. \tag{A.58}$$

The Mach number is approximately fixed (i.e., maintained within a narrow range of values) for proper aerodynamic blade design. Thus,

$$A_c \propto \frac{\sqrt{T_{in}}\ \dot{m}_a}{p_{in}}. \tag{A.59}$$

From this equation, the mean cross-sectional flow area, which is related to overall size, is directly proportional to both the square-root of the inlet temperature and the mass flow rate and inversely proportional to the inlet pressure.

The cross-sectional area, A_c, is also a function of r_p. This can be demonstrated by combining Eqs. A.55 and A.58 which gives,

$$A_c = \left[\frac{R}{Ma\eta_T c_{pa}}\right] \cdot \frac{P}{p_{in}\sqrt{T_{in}}\left(1 - r_p^{-c}\right)} \tag{A.60}$$

where:

P = power output.

Thus, for a fixed power output, A_c decreases if p_{in}, T_{in}, or r_p increases.

In a multiple turbine power system, as proposed for CAES plants, the use of $\dot{m}_a'$ based on the overall system does not indicate the relative size of each turbine. Application of Eq. A.59 is suggested to evaluate each turbine separately. This calculation requires the determination of $\dot{m}_a$ from the overall performance of the system.

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