Conf-780599--3

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7-25-78

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A PARAMETRIC ANALYSIS OF TURBOMACHINERY OPTIONS FOR COMPRESSED AIR ENERGY STORAGE PLANTS

C. S. Kim, G. T. Kartsounes

prepared for

CAES Technology Symposium 1978
May 15-17, 1978
Pacific Grove, CA

U of C-AUA-USDOE

ARGONNE NATIONAL LABORATORY, ARGONNE, ILLINOIS

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

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A PARAMETRIC ANALYSIS OF TURBOMACHINERY OPTIONS FOR COMPRESSED AIR ENERGY STORAGE PLANTS

C.S. Kim*, G.T. Kartsounes**

This paper presents a parametric study of possible turbomachinery options for compressed air energy storage plants. The plant is divided into the four subsystems: a turbine system, compressor system, motor/generator, and an underground air storage reservoir. The turbine system comprises a high-pressure turbine, a low-pressure turbine, two combustors, and a recuperator. The compressor system comprises a low-pressure compressor, high-pressure compressor, booster compressor, intercoolers, and an aftercooler. A water-compensated mined cavern constitutes the underground air-storage reservoir.

Plant performance is presented in terms of five parameters: specific air flow rate, specific heat rate, specific storage volume, specific compression rate, and overall plant efficiency.

The capital and operating costs of the plant as a function of the turbomachinery options are presented. Design variables of the turbomachinery are the reservoir pressure and inlet gas temperatures to the turbines.

Introduction

Compressed air energy storage (CAES) is a near-term technology for the load leveling and peak shaving strategies being considered by electric utili-

*Asst. Chemical Engineer, Energy and Environmental Systems Division, Argonne National Laboratory, Argonne, Illinois 60439.
**Mechanical Engineer, Energy and Environmental Systems Division, Argonne National Laboratory, Argonne, Illinois 60439.

ties. Assessments of the technical and economic feasibility of this storage system indicate that it is economically competitive with conventional gas-turbine peaker units. A CAES plant has four subsystems (see Fig. 1): a turbine system, compressor system, motor/generator, and an underground air-storage reservoir. The CAES concept is based on a split Brayton cycle with an accompanying underground air reservoir. During periods of off-peak power demand, air is compressed with base plant power and stored in the underground reservoir. For power generation, the air is discharged through a combustion turbine during the peak demand period.

Because the storage reservoir is usually the most costly single component in a CAES plant, its volume is a sensitive design parameter. The volume required is affected by storage pressure and temperature, power level, and generation time; and by the reservoir types, air quantity required by the turbine system, and pressure ranges permitted by the turbomachinery (turbines and compressors). Compressed air can be stored underground in caverns or in the pore space of porous rock formations.

The components of the subsystems of a CAES plant are delineated here from precision of reference in the evaluations presented in this report. The turbine system consists of a low-pressure gas turbine (LGT), a high-pressure gas turbine (HGT), two combustors, and a recuperator (see Fig. 1). The LGT is a turbine modified from a conventional gas-turbine peaker unit. For proposed CAES plants, the HGT is a modified steam turbine operating at gas temperatures of about 1000°F. Optimized designs for compressed-air turbines that operate at high temperatures, are being investigated. The combustors can be designs modified from conventional gas-turbine peaker units. Preliminary studies indicate that recuperators can be designed that are economically feasible for CAES application. These differ from conventional gas-turbine peaker units because of the high-pressure air leaving the reservoir.

The compressor system contains a low-pressure, high-pressure, and booster compressor, intercoolers, and an aftercooler (see Fig. 1). Intercooling is required to operate the compressors within temperature limits tolerable for standard materials. An aftercooler is used to cool the air to avoid possible thermal-stress damage to the storage reservoir.

The performance of a CAES plant can be characterized in terms of four specific parameters and overall plant efficiency:

- Specific air flow is the mass flow rate of air supplied to the turbine system per kilowatt power generated. It is the major factor in determining the size of the turbines, compressors, and air-storage reservoir.
- Specific heat rate is directly proportional to fuel consumption and is equal to the product of specific fuel consumption and the lower heating value of the fuel. It therefore affects the operating cost of the turbines.
- Specific storage volume, the volume of reservoir required per kilowatt of power generated, is dependent on the specific air flow rate and the temperature of stored air.
- Specific compression rate is the energy equivalent of the power supplied to the compressors per kilowatt of power generated. This parameter is the amount of off-peak energy required to operate the compressors.
- Overall plant efficiency is equal to the total energy output from the turbines divided by the sum of the energy input from the fuel and off-peak energy to the compressor system.

The cost of a CAES plant can be characterized in terms of capital cost and operating cost. Capital cost includes the direct cost of the air storage facility, the turbomachinery, and balance of plant, and the indirect cost due to a contingency allowance, engineering and administration, and escalation and interest during construction. The operating cost of the plant includes the capital charge, cost of fuel to the combustors, off-peak electricity to the compressors, and operation and maintenance costs.

This paper presents a parametric study of turbine systems for CAES plants. The performance and cost of the complete plant resulting from different turbomachinery options are presented. The turbine system design parameters considered are the reservoir storage pressure and the inlet gas temperatures to the LGT and HGT. The LGT was based on a 200-MW power output and a nominal pressure ratio of 16:1. A water-compensated mined cavern was chosen as the compressed air storage reservoir.

Performance Evaluation

Thermodynamic Analysis

A thermodynamic analysis was carried out on each subsystem of a CAES plant, and the results were combined to evaluate overall plant performance. Design parameters considered in the analysis include: air storage pressure and inlet gas temperatures to the high-pressure gas turbine (HGT) and low-pressure gas turbine (LGT).

The following assumptions were made.

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

Underground Air Storage System. The underground air storage reservoir considered is a water-compensated cavern. Therefore, the pressure variation in the cavern during the operating cycle is negligible. Two factors influence the pressure variation in the air shaft: static head and friction. Variation in static-head pressure is a function of the air pressure in the cavern. The pressure loss due to friction is known as a function of air flow rate but is considered insignificant as compared with the variation in static-head pressure. (See Appendix for details.) The air temperature of the storage cavern (T_o) was assumed as 120°F (322°K) and four different air storage pressures (p_o) were considered in the analysis: 30, 50, 70, and 100 atm (3×10^6, 5×10^6, 7×10^6, and 1×10^7 Pa).

Turbine System. The turbine system chosen consists of two turbines (HGT and LGT), two combustors, and a recuperator (Fig. 1). The selection of the turbine system evolved from the results of a previous study [1]. The following values of system parameters were considered:

Turbine efficiencies: $\eta_{LGT} = \eta_{HGT} = 0.90$,

Recuperator effectiveness: $\varepsilon = 0.8$,

Temperatures: T_3 = 1000°, 1600°, 2000°, 2400°F (811°, 1144°, 1366°, 1589°K),

T_5 = 1600°, 2000°, 2400°F (1144°, 1366°, 1589°K),

Pressures: P_5 = 16 atm (1.6×10^6 Pa), and

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

Subscripts given in the above parameters correspond to the components or stations in Fig. 1. The efficiencies of turbines and combustors are based on state-of-the-art values of available equipment [2]. Recuperator effectiveness is a function of the heat exchanger specifications. Because the temperature of the inlet gas to the turbines must be kept low enough to avoid thermal damage of the turbine blades and vanes, cooling air is required for higher inlet gas temperatures. The amount of cooling air required was determined from data presented in Ref. 2.

The thermodynamic analysis of the turbine system is given in detail in the Appendix. Governing equations were written for each component. Mass-balance equations were formulated by considering addition of fuel to the combustors and cooling air to the turbines. Instead of momentum equations, equations that represent the pressure variations across each component were used. Energy-balance equations were written for the recuperator, the turbines, and the combustors. The definitions of recuperator effectiveness and thermal efficiency of turbines were also used to solve the equations for energy balance.

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

Compressor System. The study was extended to the compressor system in order to complete the analysis of the CAES plant. The compressor system selected comprises three compressors (low-pressure, LC, high-pressure, HC, and booster, BC), three intercoolers, and an aftercooler as shown in Fig. 1. The following parameters were assumed to be known or specified.

Adiabatic efficiency of compressors: $\eta_{HC} = \eta_{LC} = \eta_{BC} = 0.90$;

Temperatures: $T_{11} = 77°F$, $T_{13} = T_{15} = T_{17} = 200°F$, $T_{19} = 120°F$; and

Pressures: $p_{11} = 1$ atm, $p_{14} = 16$ atm.

The required output includes the compressor outlet temperatures (T_{12}, T_{14}, and T_{18}), from which the power input to the compressors may be obtained. (See Appendix for the detailed analysis.)

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

<u>Compressed Air Energy Storage Plant</u>. By using the results from the analyses of the turbine system, underground storage cavern, and compressor system, the overall performance of the CAES plant was evaluated.

Results and Discussion

Results of the parametric study are presented in terms of the five performance parameters: specific air flow rate, specific storage volume, specific heat rate, specific compression rate, and overall plant efficiency. These values are given as a function of air storage pressure and inlet gas temperatures to the HGT and LGT.

Specific air flow rate is the flow rate of air coming out of the cavern per unit output of the turbine system. It is directly proportional to the turbine and compressor sizes, and, thus, is an important factor in determining the cost of the above-ground facility. A plot of specific air flow rate against the air storage pressure at different turbine inlet gas temperatures (Fig. 2) shows that the air flow rate ranges from 6.6-12.0 lb/kWh (3.0-5.4

kg/kWh) for the conditions specified in this study, and it decreases as air storage pressure increases. Shown in Fig. 3 are the effects of turbine inlet gas temperatures on the air flow rate at the air storage pressure of 70 atm (7×10^6 Pa). It can be seen that higher turbine inlet gas temperatures result in smaller air flow rate, even though cooling air is required.

Specific storage volume, the required storage cavern volume per unit work output, is directly related to the cost of the underground facility for a CAES plant. This storage volume depends on the required specific air flow rate as well as on cavern conditions, such as pressure and temperature of stored air. Consequently, results for the specific storage volume show a trend similar to that for the specific air flow. Figures 4 and 5, respectively, show the effects of air storage pressure and turbine inlet gas temperatures on the storage volume. It is seen that smaller storage volume results from higher air storage pressure or higher turbine inlet gas temperatures. Specific storage volume in this study ranges from 0.96 ft^3/kWh (0.027 m^3/kWh) to 5.84 ft^3/kWh (0.162 m^3/kWh).

Specific heat rate is a measure of premium-fuel usage for the combustors per unit power output of the system. It varied in this study from 3770 Btu/kWh (3.98×10^6 J/kWh) to 4280 Btu/kWh (4.52×10^6 J/kWh). The effect of storage pressure on the heat rate is given at different turbine inlet gas temperatures in Fig. 6: higher storage pressure results in lower heat rate. Figure 7 shows that heat rate increases as the LGT inlet gas temperature increases and that the HGT inlet gas temperature has a minor effect on the heat rate.

Specific compression rate is the fuel equivalent of the off-peak electrical energy input to the compressor system per unit power output of turbine system as defined in Eq. (D.4) in Appendix. For the conditions of this study, the rate ranges from 5280 But/kWh (5.57×10^6 J/kWh) to 7790 Btu/kWh (8.22×10^6 J/kWh). Figure 8 shows that, in general, compression rate increases slowly with increasing storage pressure. Shown in Fig. 9 are the effects of the turbine inlet gas temperatures on the compression rate at p_o = 70 atm (7×10^6 Pa): smaller compression rate is required by higher turbine inlet gas temperatures.

The overall plant efficiency, the ratio of turbine power output to the sum of the power input to the compressors and the power equivalent of fuel energy, ranges from 0.538–0.581 for the conditions specified in this study. The effects on the overall plant efficiency are given in Figs. 10 and 11.

Figure 10 shows the effects of storage pressures on plant efficiency: (a) for $T_3 = T_5$ = 2400°F (1589°K) or $T_3 = T_5$ = 2000°F (1366°K), plant efficiency increases with the storage pressure; (b) for $T_3 = T_5$ = 1600°F (1144°K), plant efficiency increases up to 70 atm (7×10^6 Pa) and then decreases as the storage pressure further increases; and (c) for T_3 = 1000°F (811°K); T_5 = 1600°F (1144°K), plant efficiency decreases monotonically with storage pressure.

The effects of turbine inlet gas temperatures on plant efficiency are given in Fig. 11. It shows that higher plant efficiency is obtainable with higher HGT inlet gas temperature. It also shows that efficiency increases with the LGT inlet gas temperature for T_3 = 2000°F (1366°K) or 2400°F (1589°K), and it has a minimum at about T_5 = 2000°F (1366°K) for T_3 = 1000°F (811°K) or 1600°F (1144°K).

Economic Analysis

An economic analysis of the CAES plant was made to show the effects of the parameters on capital and operating costs. The analysis was based on the results of the performance evaluation described in the section Results and Discussion. In order to provide a reasonable basis for the economic analysis, the following operating cycle was chosen: 20-hr nominal cavern storage capacity and 2190-hr/yr generation time.

Capital Cost

Direct capital cost of the CAES plant was divided into the following: cost of underground air storage cavern and water-compensated reservoir, cost of turbomachinery equipment, and *balance of the plant*.

The storage cavern cost included the cost of the air and water shafts, cavity, development and mobilization, and completion. The cost of the air and water shafts was estimated based on the cavern depth which was determined by the air storage pressure. The cost of the cavity was estimated based on specific storage volume with a 10% capacity margin. Since the storage cavern considered in the analysis is water-compensated, the cost of the water reservoir was also included. The storage-related costs were based on Ref. 3.

Estimation of the turbomachinery cost was based on Ref. 2. The cost of the low-pressure gas turbine with a cycle-pressure ratio of 16:1 was determined by the inlet gas temperature and the cost of the high-pressure gas turbine was determined by both the inlet gas temperature and air storage pressure. Costs of the LC and HC with the overall compression ratio of 1:16 were estimated from the air flow rate, and the cost of BC was determined by the air flow rate and air storage pressure. A 25% allowance was given for the ducting and installation of the turbomachinery equipment.

The remainder of the plant equipment, which includes the clutches, motor/generator, recuperator, combustors, fuel storage, coolers, electrical power system, land, and plant structure was denoted as the *balance of plant*. This equipment is relatively insensitive to CAES design parameters and a fixed cost of \$80/kW was used for the balance of plant for all cases of this study.

Total capital cost of the plant was estimated from the direct capital cost considering the following allowances: 15% for contingency, 10% for engineering and administration, and 30% for escalation and interest during the construction period.

Operating Cost

Operating cost of the CAES plant consists mainly of capital charge, cost of fuel to the combustors, off-peak electricity to the compressors, and operation and maintenance. Annual capital charge was estimated from the total capital cost based on the fixed capital charge rate of 18% a year. Estimation of the cost of premium fuel was made by multiplying the specific heat rate by the cost of No. 6 oil ($2.50/$10^6$ Btu). Cost of the off-peak electricity to the compressors was estimated from the specific compression rate and the electricity cost from the base plant (15 mills/kWh). A value of 2 mills/kWh was used as the cost of operating and maintenance for all cases.

Results and Discussion

Results of the economic study are given in terms of the two specific costs: capital cost ($/kW) and operating cost (mills/kWh). The values are presented as a function of the storage pressure (p_o) and turbine inlet temperatures (T_3 and T_5).

Capital cost of a CAES plant varied from $285/kW to $406/kW for the range of design parameters specified in the study. The cost of the underground storage cavern was found to be the highest component cost for most cases ranging from 26–46% of the total capital cost and the cost of the turbomachinery equipment varied from 16–31% of the total direct capital cost. Figure 12 shows that the cost of the storage cavern sharply decreases as the storage pressure increases from 30 atm (3×10^6 Pa) to 70 atm (7×10^6 Pa) and then slowly decreases thereafter, and that higher turbine inlet gas temperatures result in lower storage cost. In addition, this figure illustrates that the cost of turbomachinery is weakly affected by storage pressure for the two cases: $T_3 = 1000°F$ (811°K); $T_5 = 1600°F$ (1144°K) and $T_3 = T_5 = 1600°F$ (1144°K). For the other cases where $T_3 = T_5 = 2000°F$ (1366°K) or $T_3 = T_5 = 2400°F$ (1589°K), the cost of turbomachinery increases with increasing storage pressure. Also shown is that, in general, higher turbine inlet temperatures result in higher turbomachinery cost.

Total capital cost is given in Fig. 13 as a function of storage pressure for the four different combinations of inlet gas temperatures to the HGT and LGT. Capital cost sharply decreases with increasing storage pressure for all the cases up to 70 atm (7×10^6 Pa) and either slowly decreases or increases thereafter. Higher turbine inlet temperatures result in lower capital cost at low storage pressures, for example 30 atm (3×10^6 Pa). However, at storage pressures greater than 70 atm (7×10^6 Pa), higher turbine inlet temperatures result in higher capital cost. Among the cases considered in the study, the design parameters that result in the lowest capital cost are those when $T_3 = T_5 = 1600°F$ (1144°K) and $p_o = 100$ atm (1×10^7 Pa).

Operating cost of the CAES plant is given in Fig. 14 as a function of the design parameters. It ranges from 44.8-55.5 mills/kWh for the specified design parameters. The capital charge was found to be much higher than the cost of fuel or electricity; it amounts to 52-60% of the total operating cost. Consequently, the operating cost in Fig. 14 shows a similar trend to that of the capital cost. The figure shows that the operating cost decreases with increasing air storage pressure for all the cases but $T_3 = T_5 = 2400°F$ (1589°K), which has a minimum at about 70 atm (7×10^6). It also shows that, among the cases studied, the lowest operating cost results when $T_3 = T_5 = 1600°F$ (1144°K) for $p_o \geq 58$ atm (5.8×10^6 Pa) and $T_3 = T_5 = 2400°F$ (1589°K) for $p_o < 58$ atm (5.8×10^6 Pa).

Figure 14 also illustrates that the operating cost for T_3 = 1000°F (811°K) and T_5 = 1600°F (1144°K) is greater at any storage pressure than the other three cases considered. This cost penalty results from the use of a 1000°F (811°K) inlet temperature to the HGT, which is similar to that of a steam turbine.

Conclusions

A parametric study of turbine systems for CAES plants was carried out in this study. It considered the effects of different turbine system design parameters on performance and capital and operating costs of the plant.

The following performance trends were observed:

1. Specific air flow rate and storage volume decrease as p_o, T_3, or T_5 increases.
2. Specific heat rate decreases as p_o or T_5 increases; but is relatively insensitive to T_3.
3. Specific compression rate, in general, slightly increases as p_o increases; it decreases with increasing T_3 or T_5.
4. In general, overall plant efficiency increases as T_3 increases; is only weakly affected by p_o or T_5.

The results on the economic study of the CAES plant are summarized as follows:

1. (a) Capital cost sharply decreases as p_o increases for $30 \leq p_o < 70$ atm; it is weakly affected by p_o for $70 \leq p_o \leq 100$ atm.

   (b) Higher T_3 or T_5 results in slightly lower capital cost at lower p_o, but in higher capital cost at higher p_o.
2. (a) Operating cost decreases with increasing p_o for

all the cases but $T_3 = T_5 = 2400°F$ (1589°K), which has a minimum at $p_o \simeq 70$ atm.

(b) The turbine system with $T_3 = T_5 = 2400°F$ results in the lowest operating cost for $30 \leq p_o < 58$ atm; $T_3 = T_5 = 1600°F$ (1144°K) results in the lowest for $58 \leq p_o \leq 100$ atm.

3. The turbine system with $T_3 = T_5 = 1600°F$ (1144°K) and p_o = 100 atm results in the lowest capital and operating cost.

Acknowledgments

The research activities in compressed air energy storage, which formed the basis of this paper, were funded by the Division of Energy Storage Systems, Office of Conservation, U.S. Department of Energy.

Nomenclature

$\dot{E}'_c$	Specific compression rate as defined in Eq. (D.4)
f_1	Fuel-air ratio for combustor 1
f_2	Fuel-air ratio for combustor 2
h	Enthalpy
$\dot{m}$	Mass flow rate
$\dot{m}'_a$	Specific air flow rate
p	Pressure
$\dot{Q}'$	Specific heat rate
R	Gas constant
r	Cooling air-turbine air ratio
T	Temperature
V'_s	Specific storage volume
$\dot{W}$	Power input or output

$\dot{W}_{comp}$	$\dot{W}_{BC} + \dot{W}_{LC} + \dot{W}_{HC}$
$\dot{W}_{fuel}$	Fuel energy
$\dot{W}_{out}$	$\dot{W}_{LGT} + \dot{W}_{HGT}$
ε	Recuperator effectiveness
η	Efficiency
$\eta_{overall}$	Overall plant efficiency
ΔH_L	Lower heating value of fuel, 21,500 Btu/lb (5.001×10^7 J/kg)
ΔH_H	Off-peak heat rate, 10,400 Btu/kWh (1.097×10^7 J/kWh)
Δp_f	Pressure loss due to friction
Δp_s	Pressure loss due to static-head

Subscripts

BC	Booster compressor
C_1	Combustor 1
C_2	Combustor 2
c_1	Cooling air for HGT
c_2	Cooling air for LGT
f_1	Fuel to combustor 1
f_2	Fuel to combustor 2
HGT	High-pressure gas turbine
LGT	Low-pressure gas turbine
HC	High-pressure compressor
LC	Low-pressure compressor
s	Isentropic process
0-19	Correspond to Fig. 1

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Appendix

A. Analysis of Air Storage Cavern

Pressure variation in the air shaft was considered to be influenced by the two factors: static-head and friction. Variation in static-head pressure is a function of the air storage pressure [3]:

$$\Delta p_s = p_o\,[1 - \exp(-0.00111\ p_o)], \tag{A.1}$$

where Δp_s and p_o, respectively, denote the variation in static-head pressure and the air storage pressure in atmospheres. Pressure loss due to friction, Δp_f, is a function of the air flow rate, but it was about 0.15 atm for the conditions specified in this study. Therefore,

$$\Delta p_f = 0.15 \quad . \tag{A.2}$$

Air pressure at the cavern outlet, p_1, is related to the air storage pressure as follows:

$$p_1 = p_o + \Delta p_s - \Delta p_f \quad . \tag{A.3}$$

B. Analysis of Turbine System

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

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

$$\dot{m}_3 = \dot{m}_2 + \dot{m}_1 f_1 \tag{B.2}$$

$$\dot{m}_4 = \dot{m}_3 + \dot{m}_1 r_{c1} \tag{B.3}$$

$$\dot{m}_5 = \dot{m}_4 + \dot{m}_1 f_2 \tag{B.4}$$

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

$$\dot{m}_7 = \dot{m}_6 \ . \tag{B.6}$$

In the above equations, $\dot{m}$ is mass flow rate and subscripts 1-7 denote the stations as shown in Fig. 1. The symbols f_1 and f_2 are the fuel-air ratios for Combustor 1 and Combustor 2, respectively; and r_{c_1} and r_{c_2} are the ratios of cooling air to turbine air for HGT and LGT, respectively.

Pressure losses through the piping were not considered separately but were included in those across the components. The decrease in gas pressure was assumed to be 5% across the recuperator and 6% across the combustors [3]. Thus,

$$p_2 = 0.95\ p_1 \ , \tag{B.7}$$

$$p_3 = 0.94\ p_2 \ , \tag{B.8}$$

$$p_5 = 0.94\ p_4 \ , \text{ and} \tag{B.9}$$

$$p_7 = 0.95\ p_6 \ . \tag{B.10}$$

Energy balance equations were then written for each component. For HGT and LGT:

$$\dot{W}_{HGT} = \dot{m}_3(h_3 - h_4) + \dot{m}_{c_1}(h_c - h_4) \ , \tag{B.11}$$

$$\dot{W}_{LGT} = \dot{m}_5(h_5 - h_6) + \dot{m}_{c_2}(h_c - h_6) \ , \tag{B.12}$$

where $\dot{W}_{HGT}$ and $\dot{W}_{LGT}$ are the power outputs of the HGT and LGT, and subscript c represents the cooling air. Symbol h denotes the enthalpy at different states. For Combustor 1:

$$\dot{m}_2 h_2 + \dot{m}_{f_1}(h_{f_1} + \Delta H_L) = \dot{m}_3 h_3 \ , \tag{B.13}$$

For Combustor 2:

$$\dot{m}_4 h_4 + \dot{m}_{f_2}(h_{f_2} + \Delta H_L) = \dot{m}_5 h_5 \ , \tag{B.14}$$

where ΔH_L is the lower heating value of the premium fuel.

For the recuperator:

$$\dot{m}_1(h_2 - h_1) = \dot{m}_6(h_6 - h_7) \ . \tag{B.15}$$

The following definitions were also used to solve the energy-balance equations:

Recuperator effectiveness:

$$\varepsilon = \frac{h_2 - h_1}{h_6 - h_1} . \tag{B.16}$$

Thermal efficiencies of the turbines:

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

$$\eta_{LGT} = \frac{h_5 - h_6}{h_5 - h_{6s}} . \tag{B.18}$$

In Eqs. (B.17) and (B.18), η_{HGT} and η_{LGT} are the thermal efficiencies of the HGT and LGT, respectively. Symbols h_{4s} and h_{6s} are the enthalpies of the turbine outlet gas when the expansion through the turbine is isentropic.

The mass, momentum, and energy balance equations were solved simultaneously by using an approximate method developed in Ref. 1. The following results were calculated: turbine outlet temperatures, T_4 and T_6; recuperator outlet temperatures, T_2 and T_7; and fuel-air ratios for the combustors, f_1 and f_2. The rate of air flow from the underground storage cavern is then obtained from:

$$\dot{m}_1 = \frac{\dot{W}_{LGT}}{(1 + f_1 + f_2 + r_{c1})(h_5 - h_6) + r_{c2}(h_c - h_6)} . \tag{B.19}$$

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

$$\dot{W}_{HGT} = \dot{m}_1 \left[(1 + f_1)(h_3 - h_4) + r_{c1}(h_6 - h_4)\right] . \tag{B.20}$$

The total power output of the turbine system is then:

$$\dot{W}_{out} = \dot{W}_{LGT} + \dot{W}_{HGT} . \tag{B.21}$$

C. Analysis of Compressor System

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

$$\dot{m}_{11} = 1.042\,\dot{m}_1\,(1 + r_{c1} + r_{c2}) \quad . \tag{C.1}$$

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

$$\dot{m}_{11} = \dot{m}_{12} = \dot{m}_{13} = \dot{m}_{14} = \dot{m}_{15} = \dot{m}_{16} = \dot{m}_{17} = \dot{m}_{18} = \dot{m}_{19} \quad . \tag{C.2}$$

The subscripts represent the stations given in Fig. 1.

The pressure drop across the intercooler or aftercooler was assumed to be 2%. Therefore:

$$p_{13} = 0.98\,p_{12} \quad , \tag{C.3}$$

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

$$p_{17} = 0.98\,p_{16} \quad , \text{ and} \tag{C.5}$$

$$p_{19} = 0.98\,p_{18} \quad . \tag{C.6}$$

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

$$p_{19} = p_1 + 0.3 \quad . \tag{C.7}$$

Adiabatic efficiencies of the compressors are defined as:

$$\eta_{LC} = \frac{h_{12s} - h_{11}}{h_{12} - h_{11}} \quad , \tag{C.8}$$

$$\eta_{HC} = \frac{h_{14s} - h_{13}}{h_{14} - h_{13}} \quad , \tag{C.9}$$

$$\eta_{BC1} = \frac{h_{16s} - h_{15}}{h_{16} - h_{15}} \quad , \text{ and} \tag{C.10}$$

$$\eta_{BC2} = \frac{h_{18s} - h_{17}}{h_{18} - h_{17}} \quad . \tag{C.11}$$

The compressor outlet temperatures, T_{12}, T_{14}, T_{16}, and T_{18} were calculated from Eqs. C.8 through C.11. The power inputs into the compressors, $\dot{W}_{LC}$, $\dot{W}_{HC}$, $\dot{W}_{BC}$ were then obtained as follows:

$$\dot{W}_{LC} = \dot{m}_{11}(h_{12} - h_{11}) \quad , \tag{C.12}$$

$$\dot{W}_{HC} = \dot{m}_{11}(h_{14} - h_{13}) \quad , \tag{C.13}$$

$$\dot{W}_{BC} = \dot{W}_{BC1} + \dot{W}_{BC2} = \dot{m}_{11}(h_{16} - h_{15} + h_{18} - h_{17}) \quad . \tag{C.14}$$

The total power input to the compressor system is then:

$$\dot{W}_{comp} = \dot{W}_{LC} + \dot{W}_{HC} + \dot{W}_{BC} \quad . \tag{C.15}$$

D. Performance Parameters

Specific air flow rate, storage volume, heat rate, and compression rate were obtained from the following equations:

$$\dot{m}'_a = \dot{m}_1(1 + r_{c_1} + r_{c_2})/\dot{W}_{out} \quad , \tag{D.1}$$

$$V'_s = \dot{m}'_a\, RT_o/p_o \quad , \tag{D.2}$$

$$\dot{Q}' = \dot{m}_1(f_1 + f_2)\Delta H_L/\dot{W}_{out} \quad , \text{ and} \tag{D.3}$$

$$\dot{E}'_c = \dot{W}_{comp}\; \Delta H_H/\dot{W}_{out} \quad . \tag{D.4}$$

where R is gas constant, ΔH_L is lower heating value of premium fuel, and ΔH_H is the off-peak heat rate of the base plant including electrical and mechanical losses.

The overall plant efficiency ($\eta_{overall}$) was evaluated as,

$$\eta_{overall} = \dot{W}_{out}/(\dot{W}_{comp} + \dot{W}_{fuel}) \quad . \tag{D.5}$$

where $\dot{W}_{fuel}$ is the rate of fuel supply to the combustors, and the charging time of the reservoir was assumed to equal the power-generation time of the turbine system.

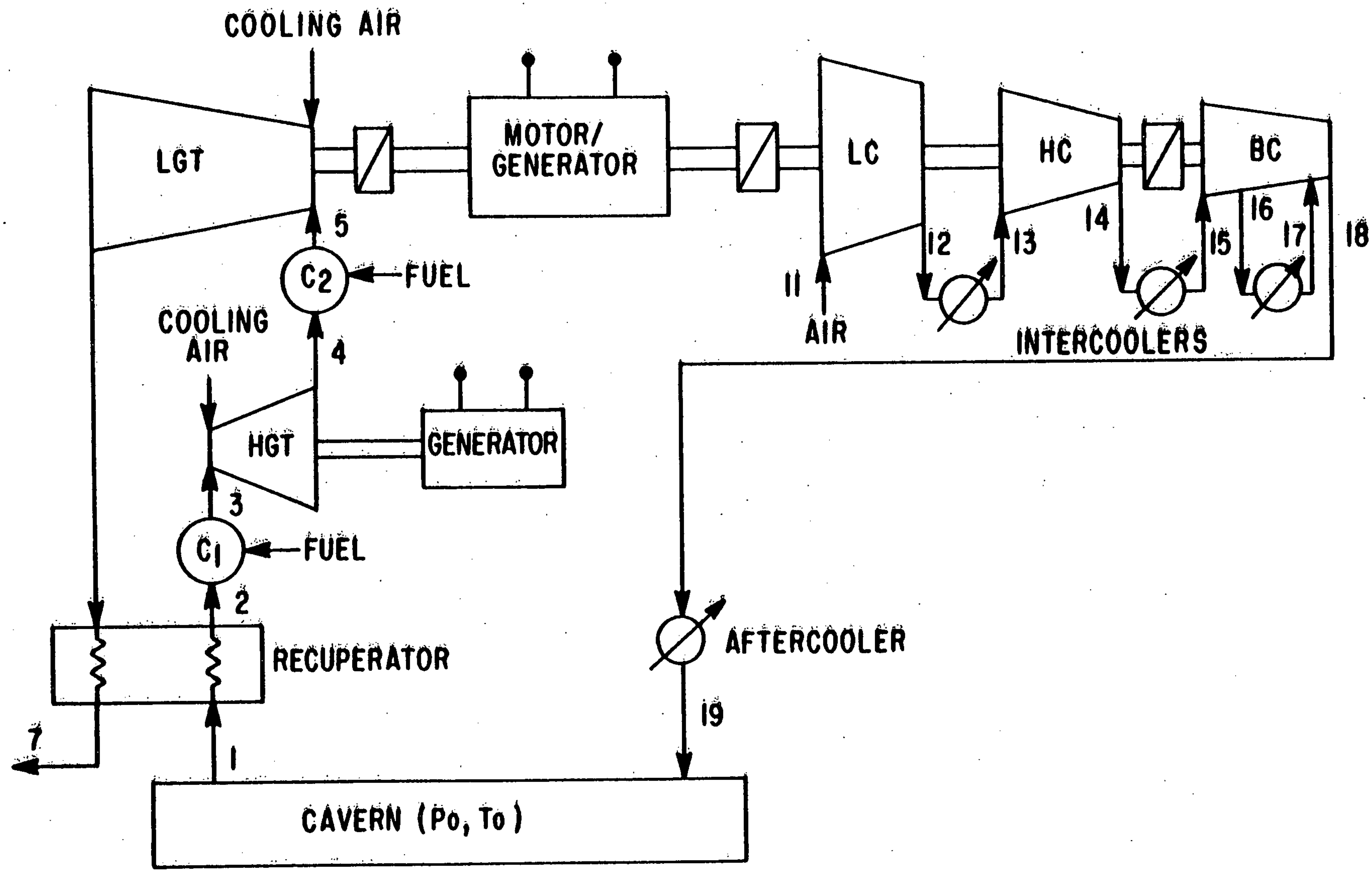

Fig. 1. Schematic Diagram of CAES Plant

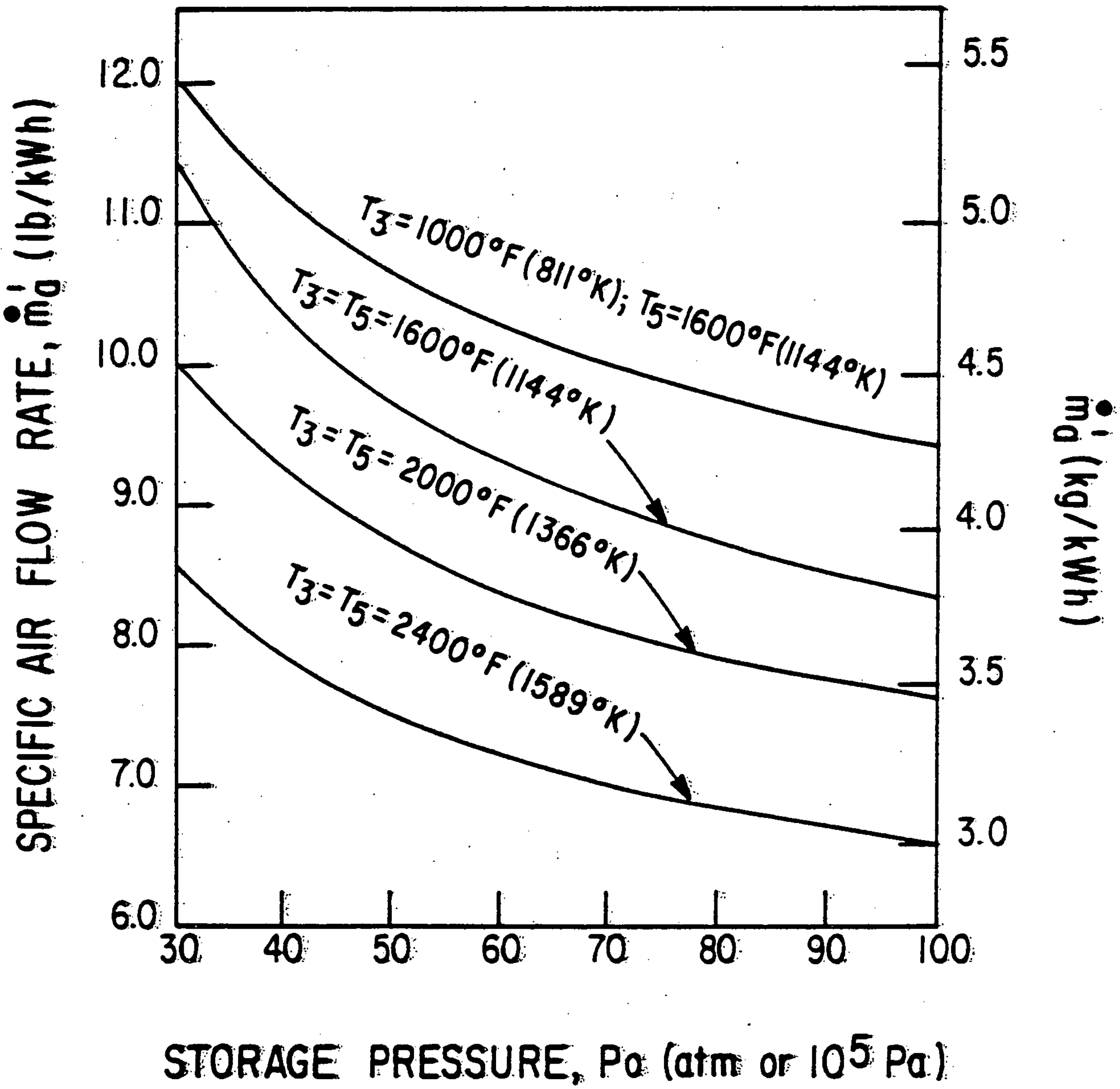

FIG. 2. EFFECT OF STORAGE PRESSURE ON SPECIFIC AIR FLOW RATE

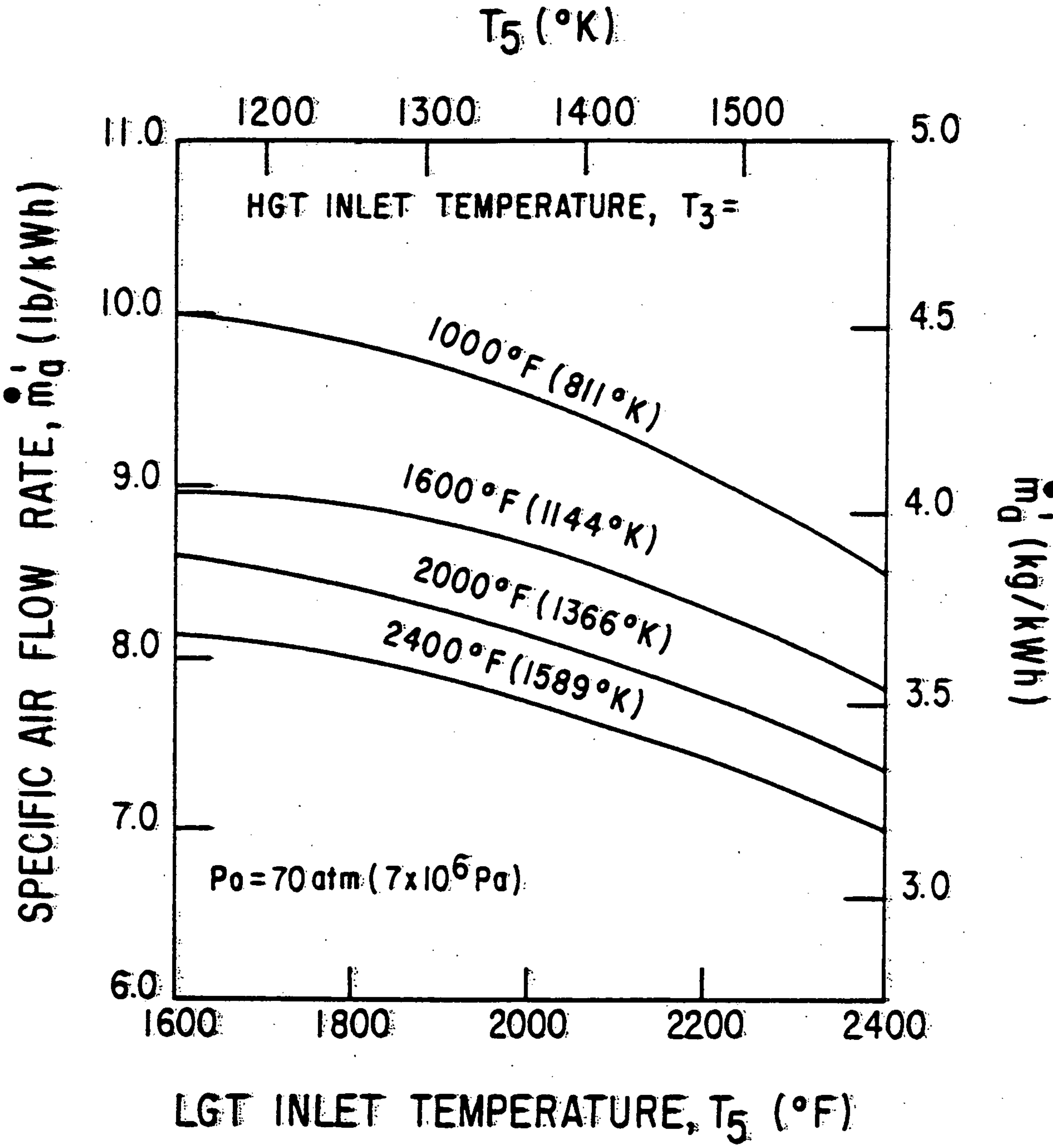

FIG. 3. EFFECT OF TURBINE INLET TEMPERATURES ON SPECIFIC AIR FLOW RATE

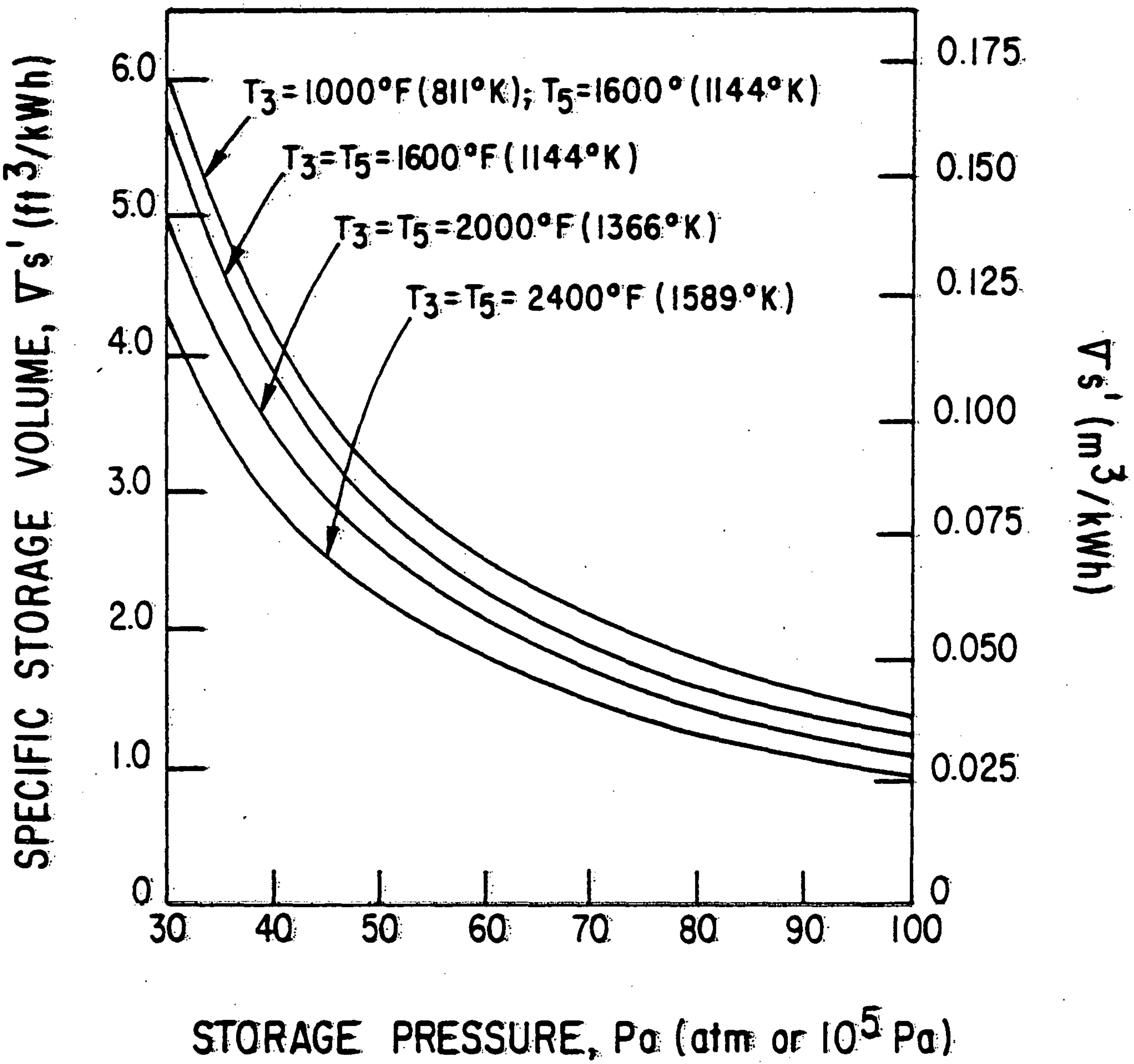

FIG. 4. EFFECT OF STORAGE PRESSURE ON SPECIFIC STORAGE VOLUME

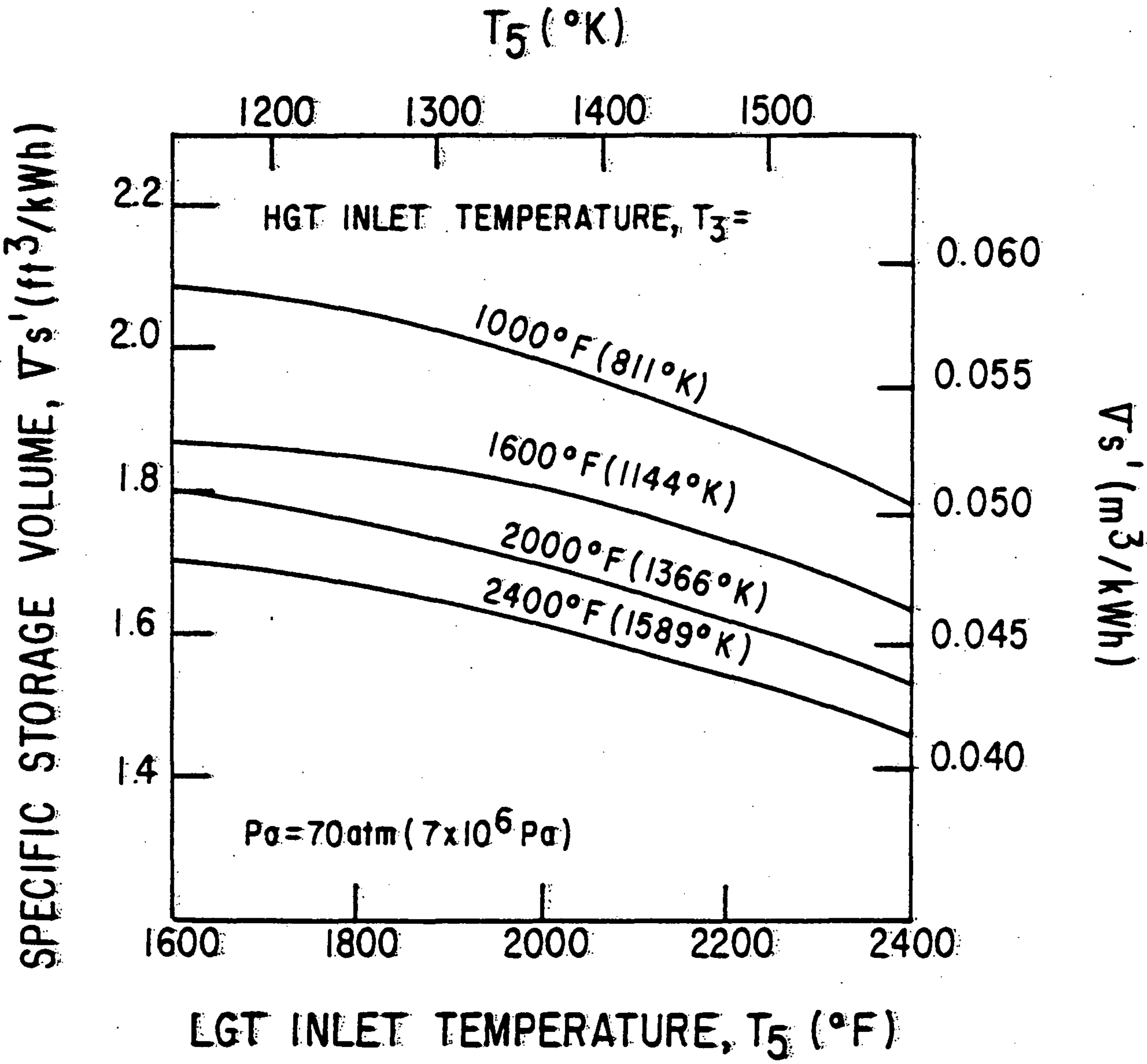

FIG. 5. EFFECT OF TURBINE INLET TEMPERATURES ON SPECIFIC STORAGE VOLUME

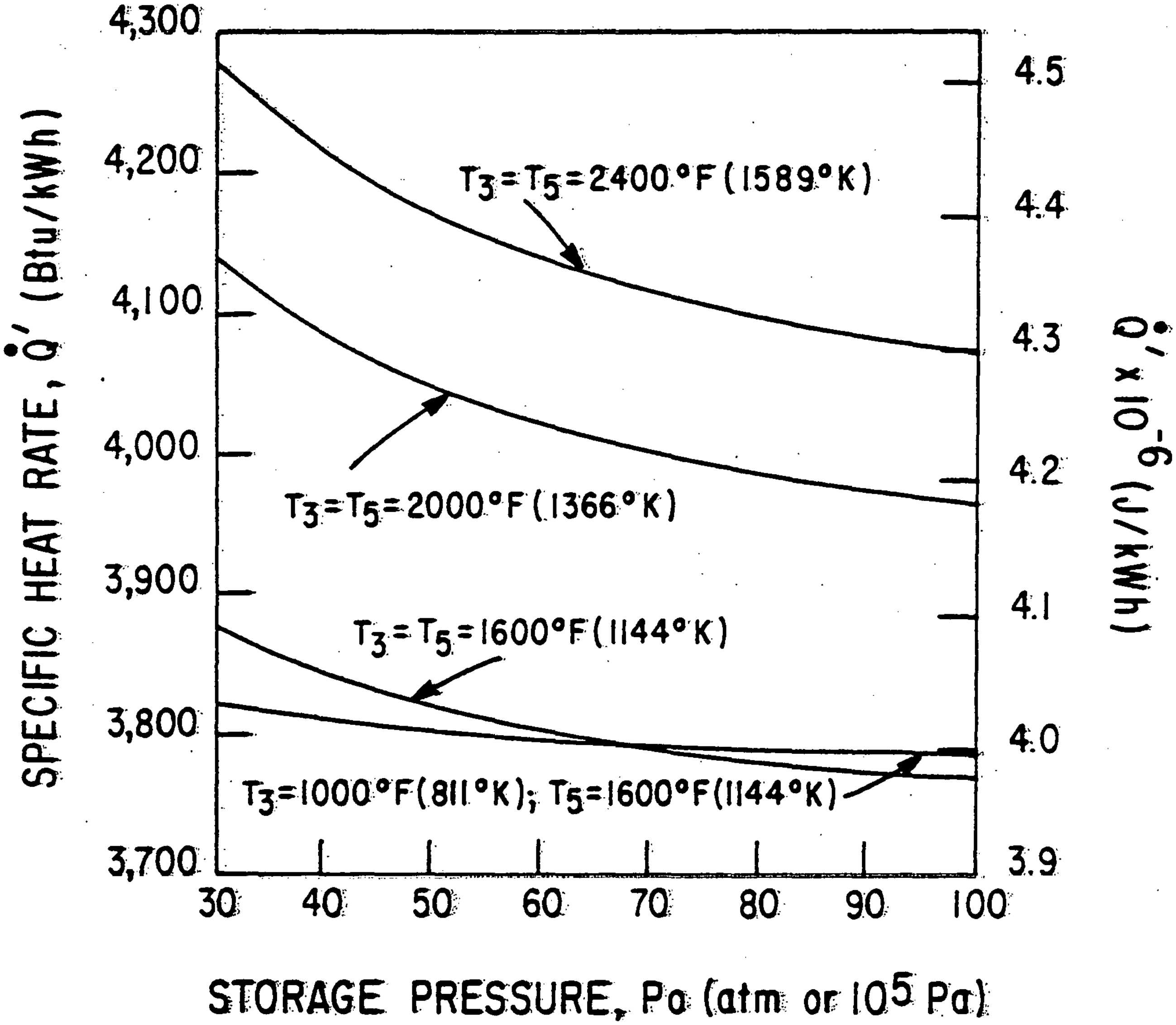

FIG. 6. EFFECT OF STORAGE PRESSURE ON SPECIFIC HEAT RATE

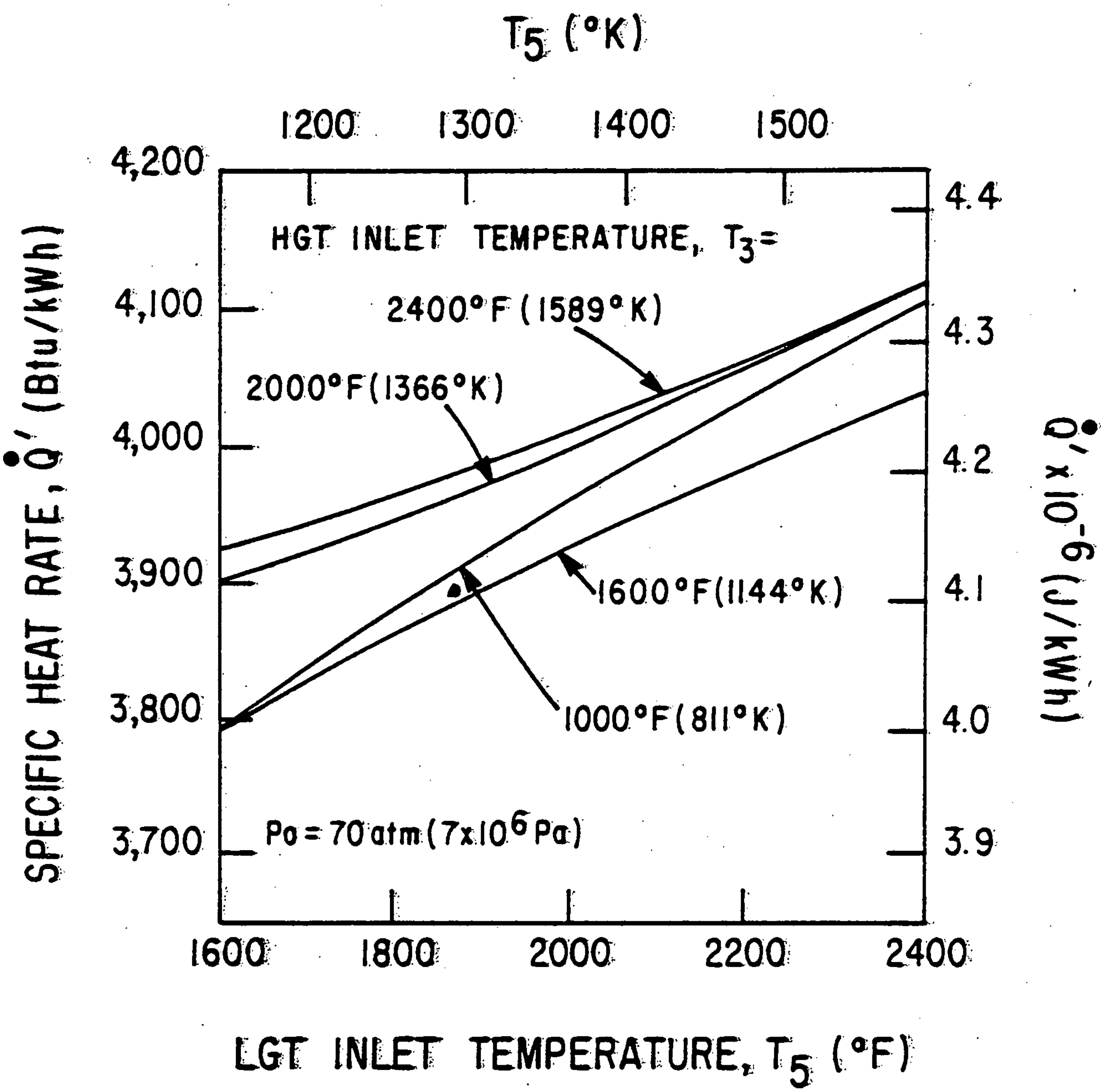

FIG. 7. EFFECT OF TURBINE INLET TEMPERATURES ON SPECIFIC HEAT RATE

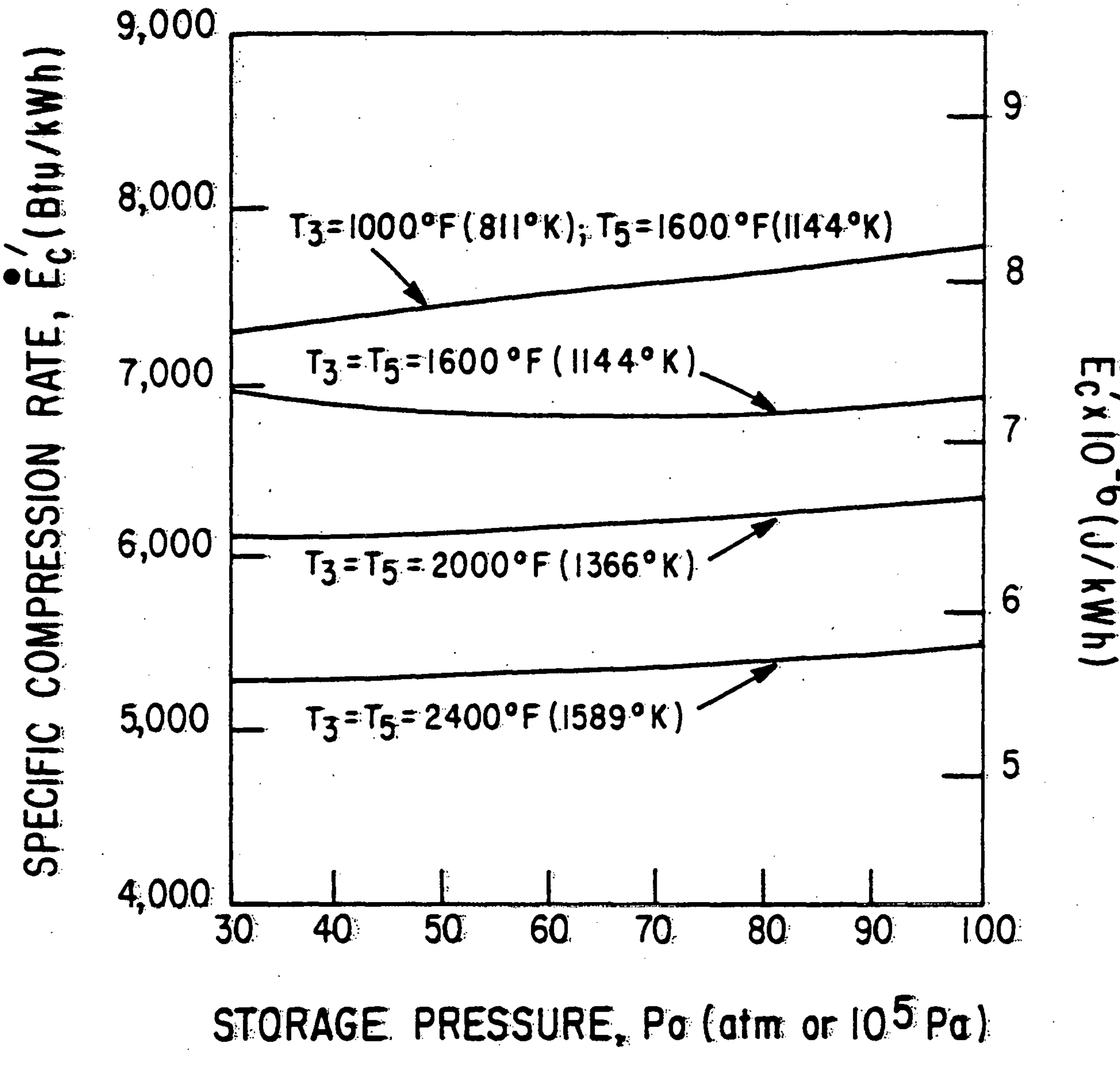

FIG. 8. EFFECT OF STORAGE PRESSURE ON SPECIFIC COMPRESSION RATE

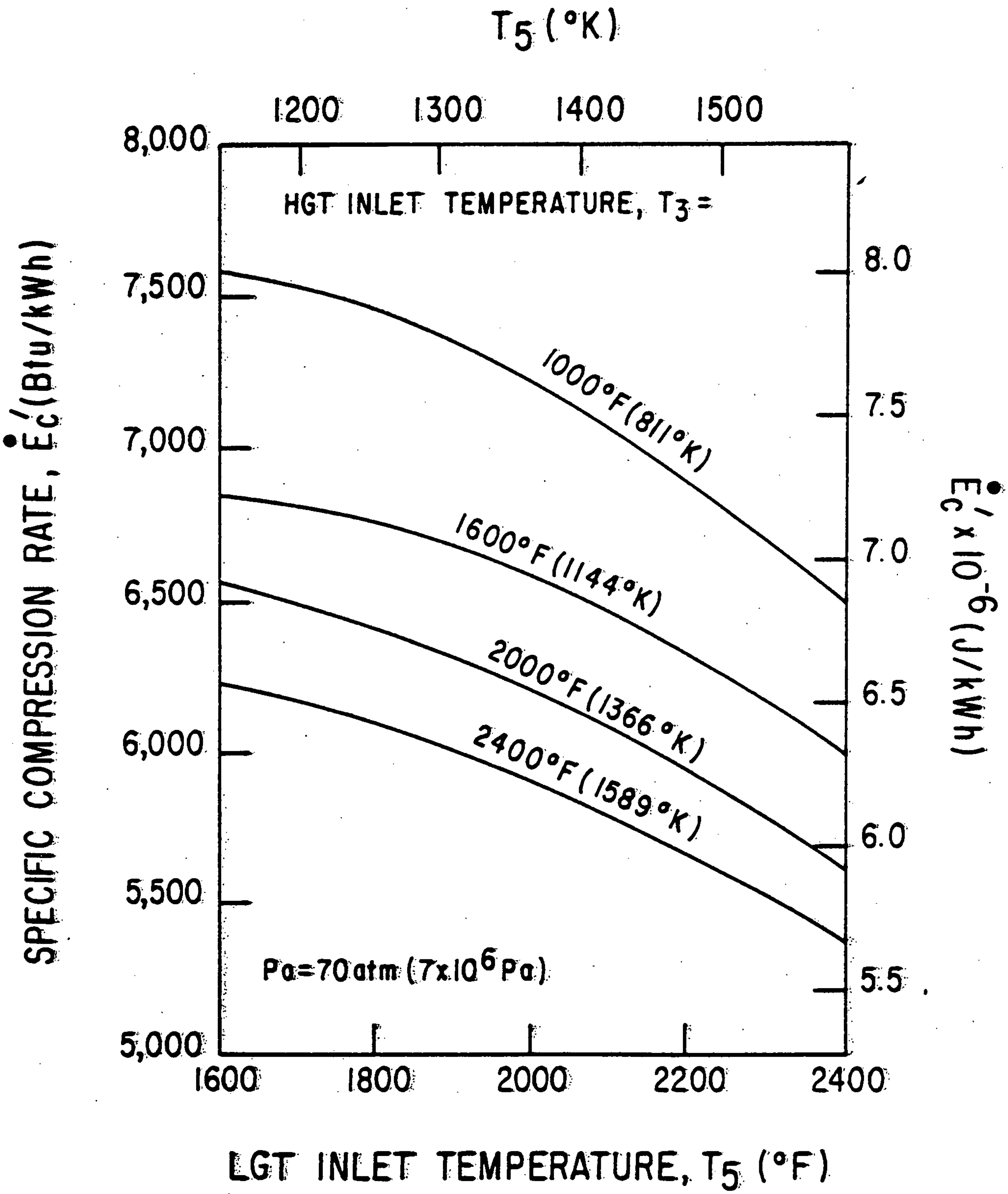

FIG. 9. EFFECT OF TURBINE INLET TEMPERATURES ON SPECIFIC COMPRESSION RATE

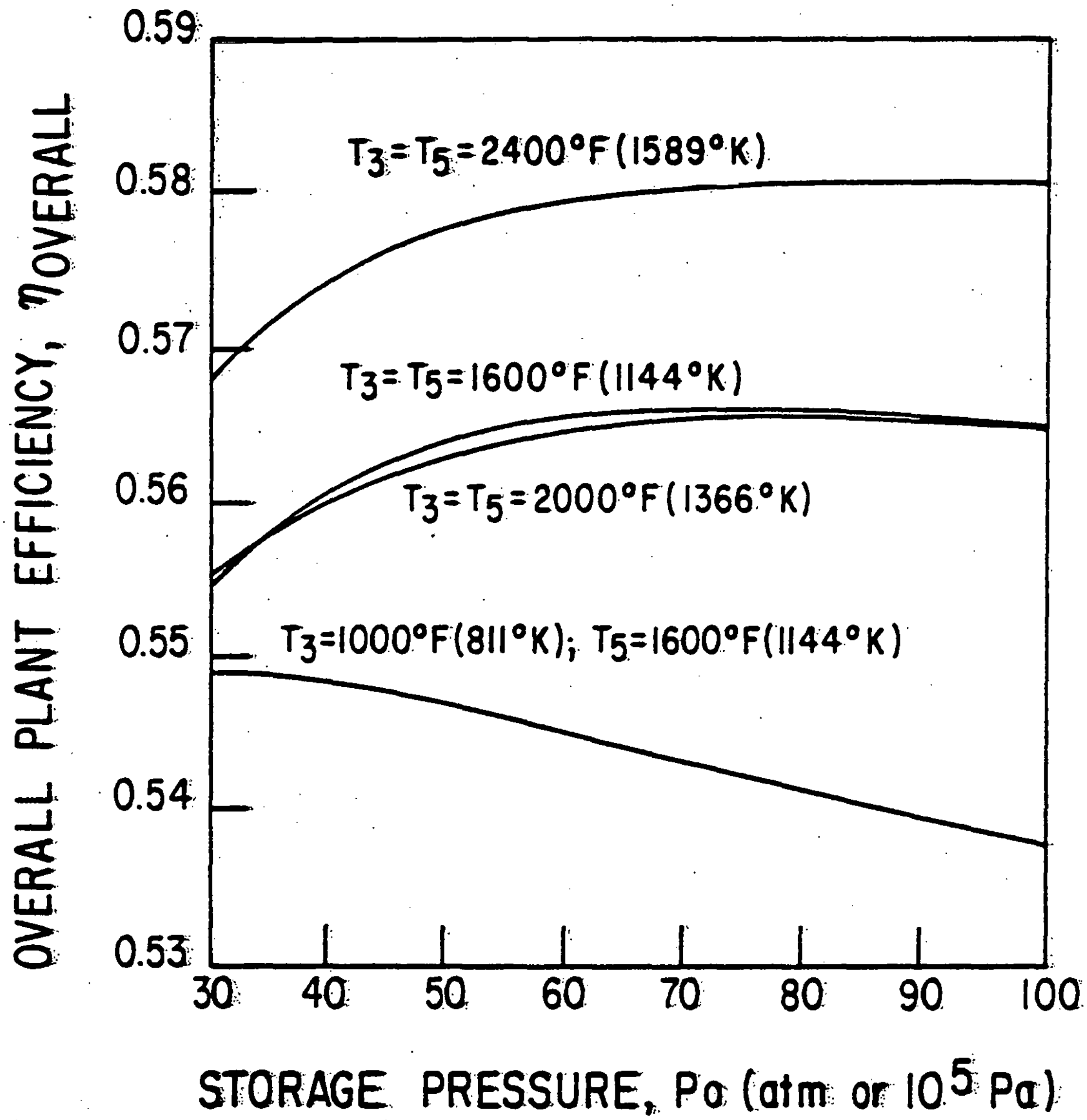

FIG. 10. EFFECT OF STORAGE PRESSURE ON OVERALL PLANT EFFICIENCY

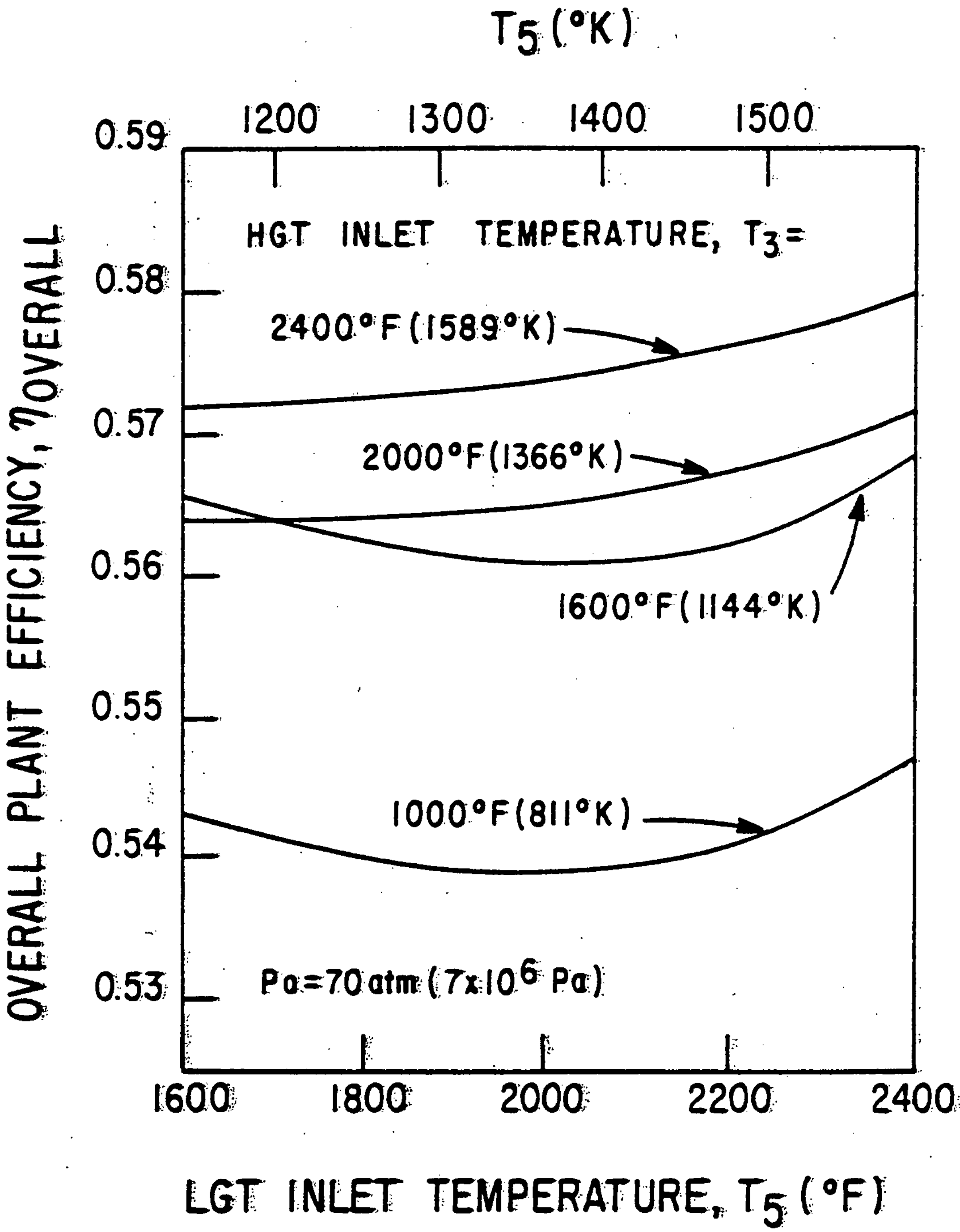

FIG. II. EFFECT OF TURBINE INLET TEMPERATURES ON OVERALL PLANT EFFICIENCY

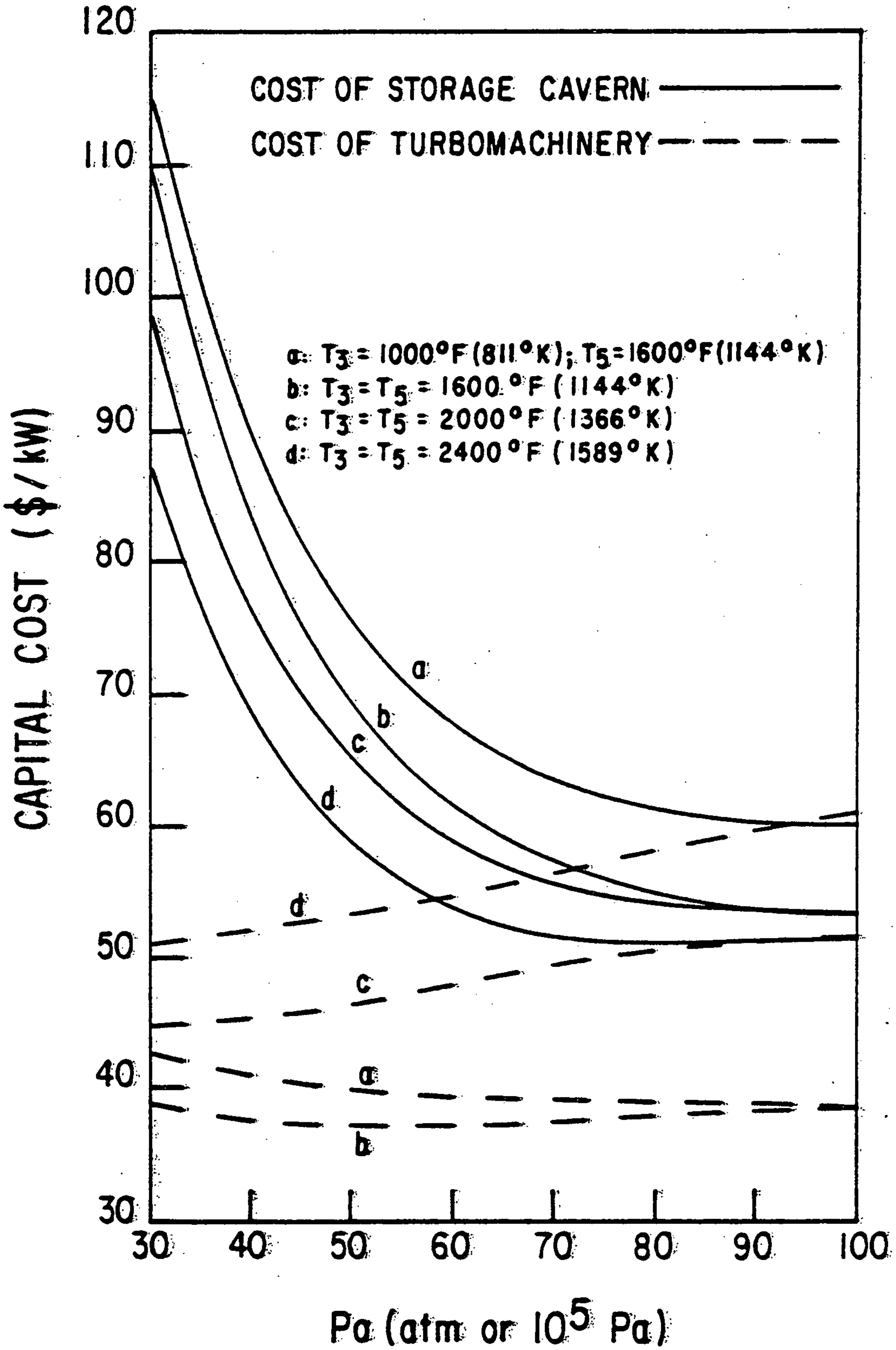

FIG. 12. COSTS OF STORAGE CAVERN AND TURBOMACHINERY

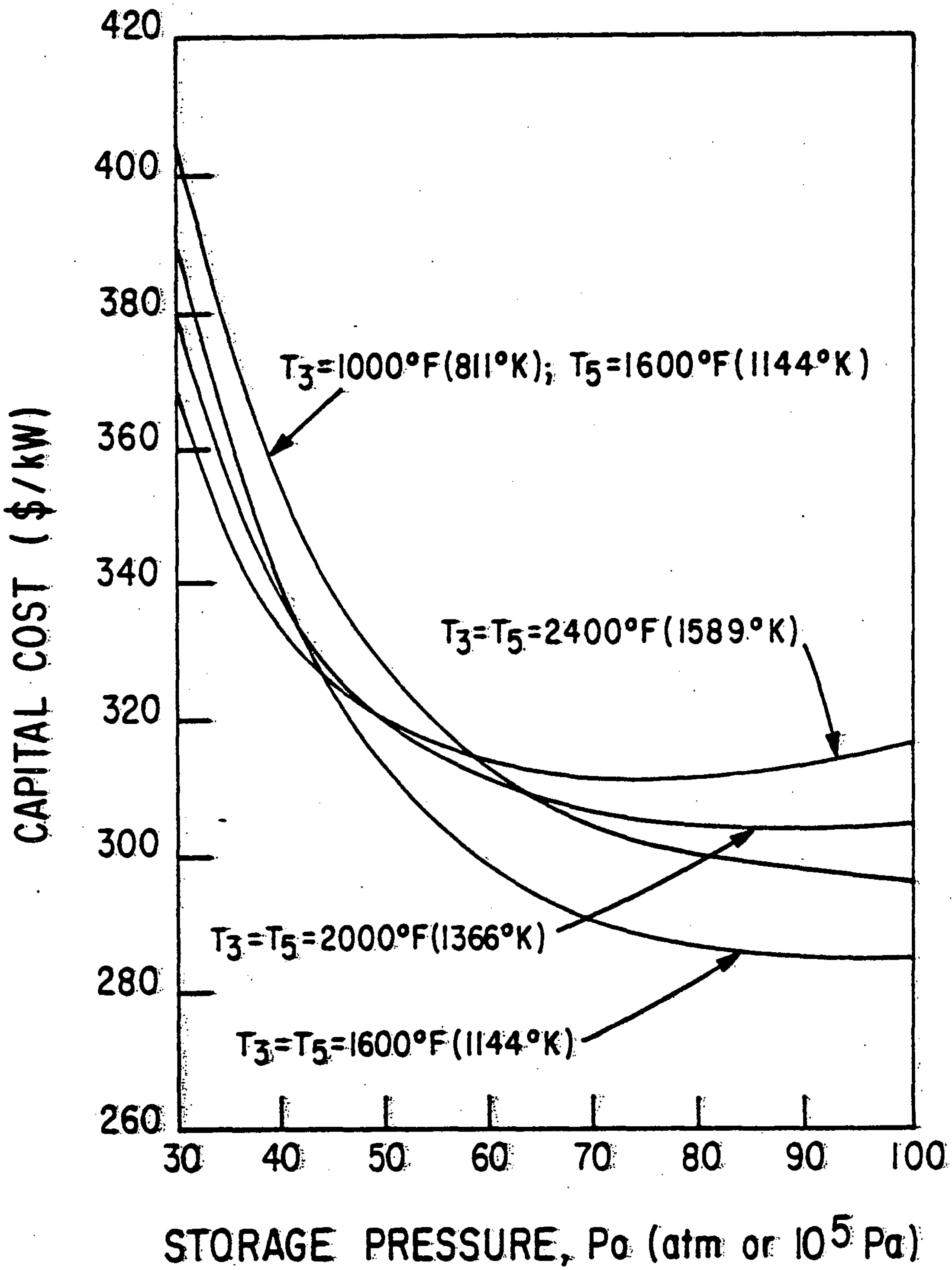

FIG. 13. EFFECT OF TURBINE OPTIONS ON CAPITAL COST

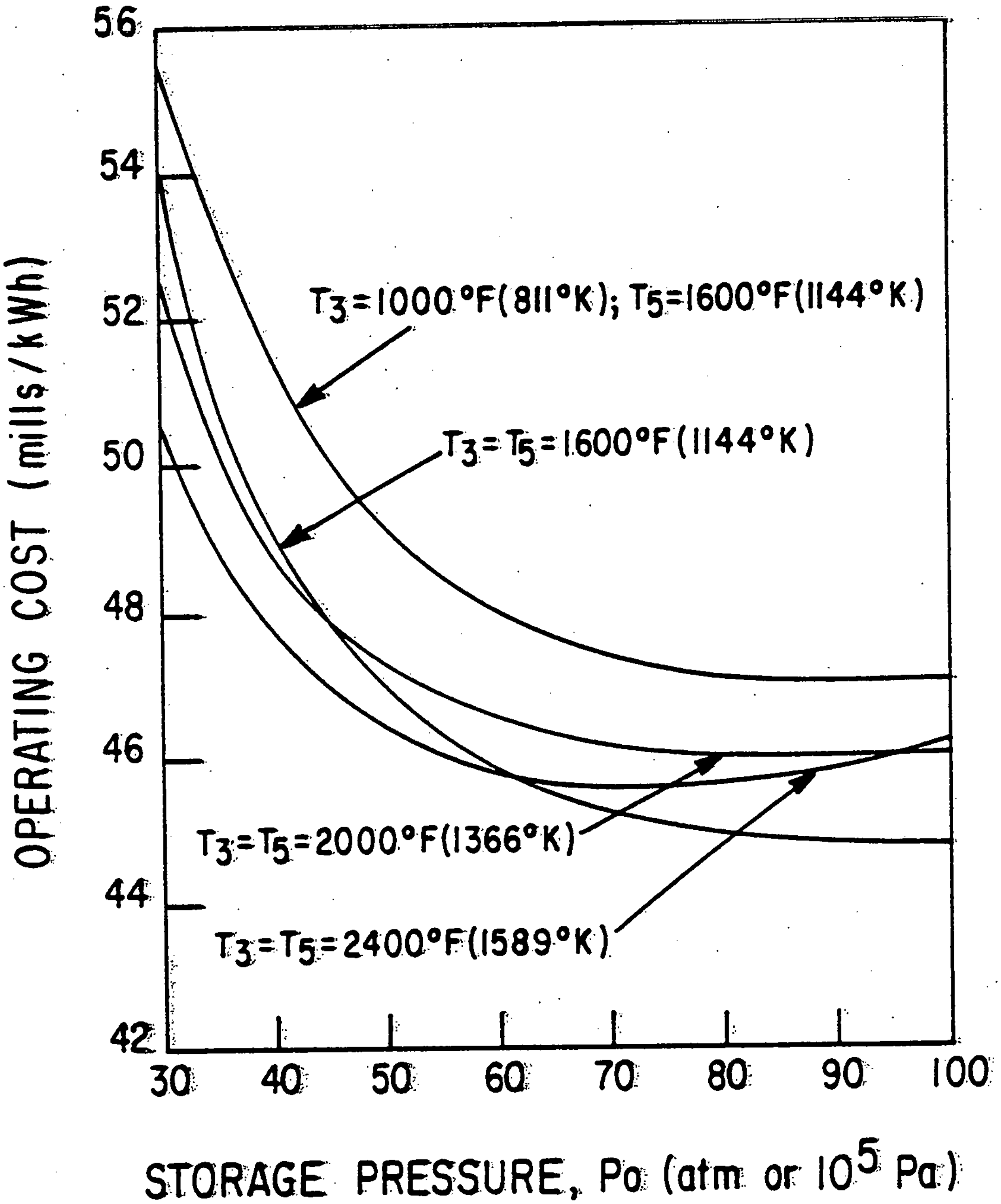

FIG. 14. EFFECT OF TURBINE OPTIONS ON OPERATING COST