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Thermodynamic Study of the Overall Performance of an Air-Turbine Steam-Turbine Combined-Cycle Power Plant Based on First-Law as well as Second-Law Analysis

FRANCIS F. HUANG
Professor
Department of Mechanical Engineering
San Jose State University
San Jose, California

ABSTRACT

The overall performance of an air-turbine steam-turbine combined-cycle power plant that is similar to a recent invention has been studied based on first-law as well as second-law analysis. It has been shown that the prerequisite for such a power plant to achieve a high overall performance is an air-turbine air-heater system having a high second-law efficiency which is not sensitive to the compressor compression ratio of the air-turbine cycle. For optimum performance, a moderate cycle pressure ratio (say 8 to 14) may be used for TIT of 1700°F (926.7°C).

NOMENCLATURE

$\dot{B}_f$ = exergy of fuel input
$\dot{B}_{HG}$ = exergy of hot gas
b_{HG} = specific exergy of hot gas
b_{HG}^{th} = thermomechanical component of specific exergy b_{HG}
b_{HG}^{ch} = chemical component of specific exergy b_{HG}
c_p = constant pressure specific heat of hot gas
$\dot{E}_f$ = energy of fuel input
F/A = mass fuel-air ratio used in combustion process
h = enthalpy
h_C = enthalpy of condensate
h_f = enthalpy of saturated water at steam pressure
h_g = enthalpy of saturated vapor at steam pressure
h_O = enthalpy of hot gas at temperature of environment
h_{pp} = enthalpy of hot gas at pinch point temperature
h_{THG} = enthalpy of hot gas at hot gas temperature
HV = heating value of fuel, kJ/kg
M = molecular weight of hot gas
$\dot{m}_a$ = mass flow rate of air in air-turbine cycle
m_s = mass ratio of steam flow to hot gas flow
p = absolute pressure
p_O = pressure of environment
$\dot{Q}_{SP}$ = amount of heat input for steam power plant
R = gas constant of hot gas
s = entropy
T = temperature
T_C = temperature of condensate
T_O = temperature of environment
T_p = saturation temperature at pressure of steam
T_{pp} = temperature of hot gas at pinch point
T_{SG} = temperature of stack gas
T_{ss} = temperature of superheated steam
THG = temperature of hot gas
TIT = air-turbine inlet temperature
x_k = mole fraction of species k in hot gas
x_k^O = mole fraction of species k in environment
$(\dot{W}_{el})_{AP}$ = power output of air-turbine plant
$(\dot{W}_{el})_{CP}$ = power output of combined-cycle power plant
$(\dot{W}_{el})_{SP}$ = power output of steam power plant
$(\dot{W}_{el}/\dot{m}_a)_{AP}$ = specific work of air-turbine plant
$(\dot{W}_{el})_{AP}/(\dot{W}_{el})_{CP}$ = ratio of power output of air-turbine plant to power output of combined-cycle power plant
$(w_{el})_{SP}$ = electrical energy recovered in steam power plant per unit mass of hot gas
$(w_{net})_{SP}$ = net work of steam power plant per unit mass of steam flow
$(w_p)_s$ = isentropic pump work per unit mass of steam flow
$(w_T)_s$ = isentropic turbine work per unit mass of steam flow
$(w_{AC})_s$ = isentropic work of air compressor per unit mass of air flow

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$(w_{AT})_s$ = isentropic work of air turbine per unit mass of air flow

y = percent of theoretical air used in combustion process

β = pressure drop factor

ε_f = exergy factor of fuel input, defined by Eq.(7)

η_g = generator efficiency

η_p = adiabatic pump efficiency

η_T = adiabatic steam turbine efficiency

η_{AC} = adiabatic air compressor efficiency

η_{AT} = adiabatic air turbine efficiency

$(\eta_I)_{AP}$ = first-law efficiency of air-turbine air-heater plant, defined by Eq. (5)

$(\eta_I)_{CP}$ = first-law efficiency of combined-cycle plant, defined by Eq. (1)

$(\eta_{II})_{AP}$ = second-law efficiency of air-turbine air-heater plant, defined by Eq. (11)

$(\eta_{II})_{CP}$ = second-law efficiency of combined-cycle power plant, defined by Eq. (6)

Π_C = compressor compression ratio

Π_T = turbine expansion ratio

INTRODUCTION

The worldwide demand for combined-cycle power plants using a combustion gas turbine as the topping unit is growing dramatically. High efficiency, low capital cost, short construction time as well as low air emission and water consumption are some of the reasons underlying the rapid growth in demand of such systems.

Combustion gas turbines, however, must burn premium fuels (oil and natural gas). On the other hand, an air-turbine system could burn coal or other low cost fuels providing the combustion process can be carried out in an efficient and environmentally acceptable manner. With increase interest and advancement in fluidized bed combustion technology, there is now interest in indirect-fired air-turbine combined-cycle plants (Foster-Pegg, 1990, LaHaye and Zabolotny, 1989, EXXON Corp., 1984, Strickland, et al., 1981). In the recent invention of EXXON, an air-turbine cycle and a conventional steam-turbine cycle are integrated. The combustion air is compressed, heated by flue gas in the furnace (to 1700°F), then expanded in an air turbine before it is used as combustion air. A preliminary assessment of this invention has been made (Rahbany, 1989) in a study funded by the Electric Power Research Institute.

This paper deals with the thermodynamic aspects of a system that is slightly different from the EXXON system having an air-turbine air-heater system coupled to a steam-turbine system. Thus it is very similar to a gas-steam combined-cycle plant. The major objective of this work is to make use of a recently proposed methodology for the study of combined-cycle gas-steam power plants (Huang, 1989, Huang and Naumowicz, 1990) to identify the major thermodynamic parameters of the system examined in this paper.

CONCEPT DESCRIPTION

An indirect-fired air-turbine combined-cycle power plant can be configured in different ways. This study is concerned with a system as shown schematically in Fig. 1 with the corresponding enthalpy-entropy diagram of the air-turbine cycle shown in Fig. 2.

Air is first compressed in the compressor. After being heated to a desirable temperature, air expands in the air turbine to produce part of the power ouput of the combined-cycle plant. Air leaving the air turbine is then used as the combustion air in the air-heater system. The hot gas from the air-heater system is the heat source for the steam power plant. Temperature of the hot gas which plays an important role in the amount of power output obtainable from the bottoming cycle, will be controlled by varying the amount of fuel input to the air-heater system. The air-heater system could be just a fludized bed combustor. It could also be a combination of a regular combustor together with a heat exchanger and an after burner. Essentially only proven technologies are needed for the successful applications of this concept. Consequently this concept should be a viable one for energy conservation. It is particularly attractive for countries without an abundant supply of premium fuels.

The TIT used in this study is 1700°F (926.7°C).

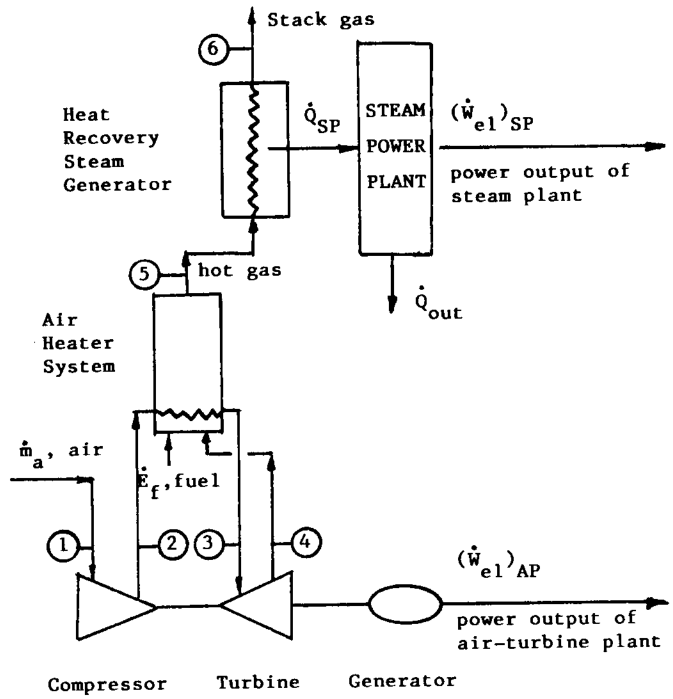

Fig. 1 Schematic diagram of an air-turbine steam-turbine combined-cycle power plant

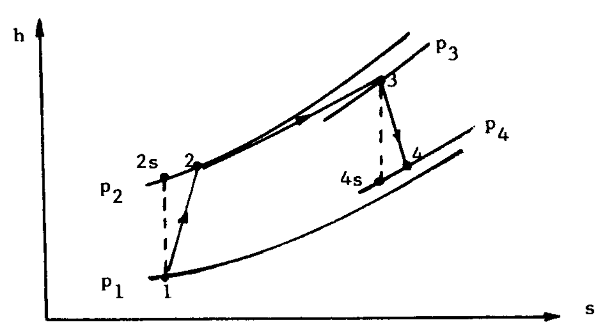

Fig. 2 Enthalpy-entropy diagram of air-turbine cycle

ENERGY RECOVERY SYSTEM

The purpose of the energy recovery system is to produce additional power for the combined-cycle plant. The bottoming cycle can be configured in different ways. It is well known that it is more effective to use a double-pressure steam system than a single-pressure steam system. But the gain in efficiency is obtained at the expense of complexity. For calculations made in this work, a single-pressure steam system, shown in Fig. 3, will be used. The corresponding temperature profile in the heat recovery steam generator is shown in Fig. 4. Since all fuel input to the combined-cycle plant in this study occurs at the air-heater system, the bottoming cycle in this study is similar to that used for a combustion gas turbine combined-cycle plant. It has been shown (Huang, 1989, Huang and Naumowicz, 1990) that the amount of electrical energy that could be recovered from each unit mass of hot gas will depend on the temeprature of the hot gas, the pinch point used in the design of the heat recovery steam generator and the parameters used in the design of steam power plant.

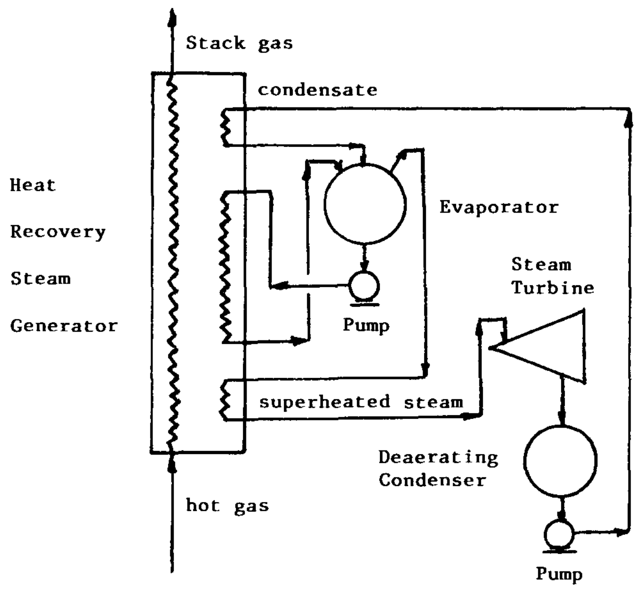

Fig. 3 Schematic diagram of a steam power cycle

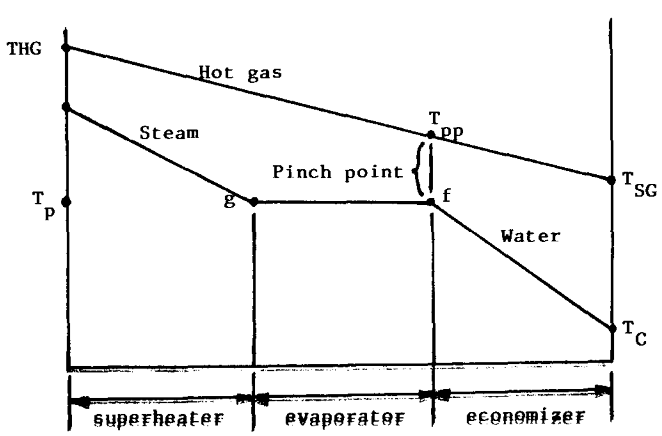

Fig. 4 Temperature profile in heat recovery steam generator

EXPRESSIONS FOR PERFORMANCE CALCULATIONS

1. First-Law Efficiency of Combined-Cycle Plant

The first-law efficiency, $(\eta_I)_{CP}$, also known as the thermal efficiency, of a combined-cycle power plant is simply the ratio of the total power output to the energy of fuel input, $\dot{E}_f$, of the combined-cycle plant. By definition,

$$(\eta_I)_{CP} = \left[(\dot{W}_{el})_{AP} + (\dot{W}_{el})_{SP}\right]/\dot{E}_f \tag{1}$$

Since $(\dot{W}_{el})_{SP}$ and $\dot{E}_f$ may be written as

$$(\dot{W}_{el})_{SP} = \dot{m}_a(1+F/A)(w_{el})_{SP} \tag{2}$$

$$\dot{E}_f = \dot{m}_a(F/A)(HV) \tag{3}$$

Eq. (1) may be written as

$$(\eta_I)_{CP} = (\eta_I)_{AP} + (1+F/A)(w_{el})_{SP}/(F/A)(HV) \tag{4}$$

where

$$(\eta_I)_{AP} = (\dot{W}_{el})_{AP}/\dot{E}_f = (\dot{W}_{el}/\dot{m}_a)_{AP}/(F/A)(HV) \tag{5}$$

may be called the first-law efficiency of the air-turbine air-heater system.

The quantity $(w_{el})_{SP}$ in Eq. (2) is the electrical energy that is recovered in the steam power plant per unit mass of hot gas. It will depend on how effectively the bottoming unit is coupled to the air-turbine air-heater system.

In Eq. (4), the first term on the right-hand side of the equation will decrease with an increase in F/A (that is, with an increase in the temperature of hot gas). The quantity (1+F/A)/(F/A) in the second term on the right-hand side of the equation will also decrease with an increase in F/A. However, the quantity $(w_{el})_{SP}$ will increase with an increase in F/A. Thus according to the first-law efficiency of the combined-cycle plant, it is not obvious whether the system be desighed with high or low temperature for the hot gas. But Eq. (4) does indicate that for a given design of the air-turbine cycle, we can determine $(\eta_I)_{CP}$ if we can determine F/A and $(w_{el})_{SP}$.

2. Second-Law Efficiency of Combined-Cycle Plant

According to the first law of thermodynamics, energy is consumed in the production of any useful product. But it is exergy (also known as availability, available energy or available work) that is really consumed according to the second law of thermodynamics (Gaggioli, 1980). Thus the true thermodynamic efficiency of a combined-cycle plant is really the second-law efficiency, $(\eta_{II})_{CP}$, which is simply the ratio of the total power output of the plant to the exergy content of fuel input, $\dot{B}_f$. By definition,

$$(\eta_{II})_{CP} = \left[(\dot{W}_{el})_{AP} + (\dot{W}_{el})_{SP}\right]/\dot{B}_f \tag{6}$$

Introducing the exergy factor of fuel input as

$$\mathcal{E}_f = \dot{B}_f/\dot{E}_f \tag{7}$$

we have $(\eta_{II})_{CP} = (\eta_I)_{CP}/\mathcal{E}_f$ (8)

Equation (8) shows that the second-law efficiency of the combined-cycle plant will be approximately equal to the first-law efficiency of the combined-cycle plant if the exergy factor of fuel input is close to unity such as in the case of many hydrocarbon fuels.

However, the second-law efficiency of the combined-cycle plant may be given in a different manner by introducing the exergy content of the hit gas, $\dot{B}_{HG}$, into our definition. Thus

$$(\eta_{II})_{CP} = \frac{(\dot{W}_{el})_{AP} + \dot{B}_{HG} + (\dot{W}_{el})_{SP} - \dot{B}_{HG}}{\dot{B}_f} \tag{9}$$

which may be written as

$$(\eta_{II})_{CP} = (\eta_{II})_{AP} - \frac{(1+F/A)\left[b_{HG} - (w_{el})_{SP}\right]}{\varepsilon_f (F/A)(HV)} \tag{10}$$

where

$$(\eta_{II})_{AP} = \frac{(\dot{W}_{el}/\dot{m}_a)_{AP} + (1+F/A)(b_{HG})}{\varepsilon_f (F/A)(HV)} \tag{11}$$

is the second-law efficiency of the air-turbine air-heater system and

$$b_{HG} = \dot{B}_{HG}/(\dot{m}_a)(1+F/A) \tag{12}$$

is the specific exergy of the hot gas entering the heat recovery steam generator.

Equation (10) indicates that for a given design of the air-turbine cycle, we can determine $(\eta_{II})_{CP}$ if we can determine F/A, b_{HG} and $(w_{el})_{SP}$.

It is of interest to rewrite the second-law efficiency of the combined-cycle plant as

$$(\eta_{II})_{CP} = 1 - \left[1 - (\eta_{II})_{AP}\right] - \frac{(1+F/A)\left[b_{HG} - (w_{el})_{SP}\right]}{\varepsilon_f (F/A)(HV)} \tag{13}$$

Making use of Eq. (13), we may interpret the second term on the right-hand side of this equation as the fraction of exergy in fuel input that is destroyed in the air-turbine air-heater system while the last term on the right-hand side is the fraction of the exergy in fuel input that is destroyed in the energy recovery system.

It has been shown (Huang, 1989, Huang and Naumowicz, 1990) that in the case of combustion gas turbine combined-cycle power plants with no supplmentary firing, the fraction of exergy in fuel input that is destroyed in the energy recovery system is much smaller than the fraction of exergy in fuel input that is destroyed in the combustion gas-turbine engine. This is because the highly irreversible combustion process takes place in the combustion gas-turbine engine. We have a similar situation in our system with the combustion process taking place in the air-heater system. Consequently, Eq. (13) would imply that $(\eta_{II})_{CP}$, thus $(\eta_I)_{CP}$, depends strongly on the second-law efficiency of the air-turbine air-heater system involved.

3. Specific work of air-turbine plant

The specific work of the air-turbine plant may be given as

$$(\dot{W}_{el}/\dot{m}_a)_{AP} = \eta_g \left[\eta_{AT}(w_{AT})_s - (w_{AC})_s/\eta_C\right] \tag{14}$$

where $(w_{AT})_s$ is the isentropic work of the air turbine and $(w_{AC})_s$ is the isentropic work of the air compressor.

From the h-s diagram for the air-turbine cycle, we have

$$(w_{AC})_s = h_{2s} - h_1$$

and $(w_{AT})_s = h_3 - h_{4s}$

The compressor work will depend on inlet temperature and the compression ratio Π_C. The turbine work will depend on the turbine inlet temperature and the turbine expansion ratio Π_T.

The air-turbine expansion ratio may be expressed in terms of the compressor compression ratio and the pressure drop to be used in each of air flow circuits involved. Making use of the concept of pressure drop factor (Huang and Wang, 1987), the turbine expansion ratio may be given as

$$\Pi_T = \beta_{23}\,\beta_{45}\,\beta_{56}\,\Pi_C = \beta_{TOT}\,\Pi_C \tag{15}$$

where $\beta_{23} = 1 - (p_2 - p_3)/p_2$

is the pressure drop factor for the cold side of the air heater,

$$\beta_{45} = 1 - (p_4 - p_5)/p_4$$

is the pressure drop factor for the hot side of the air heater,

$$\beta_{56} = 1 - (p_5 - p_6)/p_5$$

is the pressure drop factor for the heat recovery steam generator, and

$$\beta_{TOT} = \beta_{23}\,\beta_{45}\,\beta_{56}$$

is the total pressure drop factor.

4. Specific exergy of hot gas

The specific exergy of hot gas may be given as (Moran, 1982),

$$b_{HG} = b^{th}_{HG} + b^{ch}_{HG} \tag{16}$$

where $b^{th}_{HG} = h - h_0 - T_0(s - s_0)$ is the thermomechanical component and b^{ch}_{HG} is the chemical component of the specific exergy b_{HG}.

On the basis of the ideal-gas model, the specific exergy of hot gas may be given as (Huang and Naumowicz, 1990),

$$b_{HG} = \int_{T_0}^{T} c_p\,dT - T_0\left[\int_{T_0}^{T} \frac{c_p\,dT}{T} - R\ln(p/p_0)\right] + RT_0 \sum x_k \ln(x_k/x^0_k) \tag{17}$$

where T_0 and p_0 are temperature and pressure of the environment respectively, x_k is the mole fraction of species k in the hot gas while x^0_k is the mole fraction of species k in the environment. The pressure p of the hot gas is not too different from p_0. A good approximation is that this pressure be made equal to $1.02p_0$. Thus we essentially only need data on c_p and the composition of hot gas to determine the specific exergy of hot gas according to Eq. (17). The thermomechanical component of specific exergy of hot gas is given by the first two terms on the right-hand side of Eq. (17) while the chemical component is given by the last term on the right-hand side of the same equation.

5. Composition of hot gas

Since the hot gas is dominated by nitrogen in the burning of hydrocarbon fuels, the effect of different fuels on the specific heat of hot gas will not be significant. Thus the amount of energy recovered in the steam power plant is not too sensitive to the kind of fuel burned. For simplicity, we will make calculations based on pure methane gas. The combustion equation, on the basis of complete combustion, is then given by

$$CH_4 + (2y)\,O_2 + (2y)(3.76)\,N_2 \longrightarrow CO_2 + 2\,H_2O + (2y-2)\,O_2 + (2y)(3.76)\,N_2 \quad (18)$$

where y gives the percent of theoretical air used.

To determine y, we must make an energy balance around the air-heater system. Knowing y, the determination of M, F/A and the composition of hot gas may be made readily using Eq. (18). Then the determination of enthalpy change and specific exergy of hot gas may be made if we have the appropriate specific heat data of the different species involved.

6. Power Output of Steam-Turbine Plant

The power output of the steam-turbine plant is given by Eq. (2)

$$(\dot{W}_{el})_{SP} = \dot{m}_a(1+F/A)(w_{el})_{SP} \quad (2)$$

in which the quantity $(w_{el})_{SP}$ will depend on the kind of bottoming unit we use. For the single-pressure steam system shown in Fig. 3, it has been shown (Huang, 1989, Huang and Naumowicz, 1990) that the electrical energy that could be recovered per unit mass of hot gas may be given as

$$(w_{el})_{SP} = m_s(w_{net})_{SP} \quad (19)$$

where $$m_s = (h_{THG} - h_{pp})/(h_{ss} - h_f) \quad (20)$$

and $$(w_{net})_{SP} = \eta_g\left[\eta_T(w_T)_s - (w_p)_s/\eta_p\right] \quad (21)$$

Equation (20) indicates that the quantity m_s depends on the temperature and composition of hot gas, pinch point and steam conditions used in the design of the heat recovery steam generator. Equation (21) indicates that $(w_{net})_{SP}$, the net work output of the steam-turbine plant, depends only on the design oarameters of the steam power plant. The quantity m_s may thus be looked upon as the coupling factor between the air-turbine air-heater system and the steam power plant.

INPUT DATA FOR PERFORMANCE CALCULATIONS

The primary objective of the calculations performed in this work is to gain insight into the behavior of our system. To study the effects of compressor compression ratio and hot gas temperature on system performance,the following common characteristics were chosen for the design of the air-turbine air-heater system:

Fuel	Methane
ε_f, exergy factor of fuel input	1.0347
Temperature of air at compressor inlet	15°C
Temperature of air at turbine inlet(TIT)	1700°F (926.7°C)
Π_T, air-turbine expansion ratio	0.9 Π_C
η_g, generator efficiency	0.98
η_{AC}, adiabatic air compressor efficiency	0.9
η_{AT}, adiabatic air turbine efficiency	0.9

Conditions of environment:

T_0, temperature	25°C
P_0, pressure	101.325 kPa
$x^0_{CO_2}$, mole fraction of carbon dioxide	0.0003
$x^0_{H_2O}$, mole fraction of water vapor	0.0303
$x^0_{O_2}$, mole fraction of oxygen	0.2035
$x^0_{N_2}$, mole fraction of nitrogen	0.7567

In addition, the following common characteristics were chosen for the design of the energy recovery system:

pp, pinch point	20°C
P_{cond}, back pressure of steam turbine	7.0 kPa
T_{ss}, temperature of superheated steam	50°C below THG
η_g, generator efficiency	0.98
η_p, adiabatic pump efficiency	0.75
η_T, adiabatic steam turbine efficiency	0.85

RESULTS

1. Effects of compressor compression ratio and hot gas temperature on mass fuel-air ratio

The effects of compressor compression ratio and hot gas temperature on the mass fuel-air ratio are shown graphically in Fig. 5. At a given hot gas temperature, the mass fuel-air ratio is not sensitive to change in the compressor compression ratio. However, the fuel-air ratio increases with an increase in the hot gas temperature which is as expected. The peak value of this ratio occurs at a compressor compression ratio of about ten for every hot gas temperature.

2. Effects of compressor compression ratio and hot gas temperature on specific exergy of hot gas

The effects of compressor compression ratio and hot gas temperature on the specific exergy of hot gas are also shown in Fig. 5. We note that the specific exergy of hot gas is essentially a constant at a given hot gas temperature. This is in agreement with previous results on combustion products (Huang, 1989, Huang and Naumowicz, 1990) that the specific exergy of such gas mixtures is not sensitive to its composition but strongly depending on its temperature.

3. Effects of compressor compression ratio and hot gas temperature on first-law efficiency on air-turbine plant

The effects of compressor compression ratio and hot gas temperature on $(\eta_I)_{AP}$, the first-law efficiency of the air-turbine air-heater unit, are shown graphically in Fig. 6. For each hot gas temperature, this first-law efficiency increases slowly from a small compressor compression ratio to a peak value at a compressor compression ratio of about ten, then decreases slowly with an increase in the compressor compression ratio. This first-law efficiency also decreases with an increase in the hot gas temperature which is as expected because of the manner in which this quantity is defined. With information on $(\eta_I)_{AP}$ and F/A, we would be able to determine $(\eta_I)_{CP}$, the first-law efficiency of the combined-cycle plant, if we can determine $(w_{el})_{SP}$, the electrical energy that could be recovered from the steam plant per unit mass of hot gas.

4. Effects of compressor compression ratio and hot gas temperature on second-law efficiency of air-turbine plant

The effects of compressor compression ratio and hot gas temperature on $(\eta_{II})_{AP}$, the second-law efficiency of the air-turbine air-heater unit, are shown graphically in Fig. 7. We note that this second-law efficiency is not very sensitive to the compressor compression ratio at a given hot gas temperature but increases with an increase in the hot gas temperature. The peak value of this efficiency occurs at a compressor compression ratio of about ten for each hot gas temperature although the variation of this value between the compressor compression ratio of eight and fourteen is very little. With information on $(\eta_{II})_{AP}$, F/A and b_{HG}, we may determine $(\eta_{II})_{CP}$, the second-law efficiency of the combined-cycle plant, if we can determine $(w_{el})_{SP}$.

5. Overall performance of combined-cycle plant

Results on b_{HG}, F/A and $(\eta_{II})_{AP}$ given in Fig. 5 and Fig. 7 clearly indicates that good overall performance for the combined-cycle plant may be achieved by using a compressor compression ratio of ten for the air-turbine cycle. The overall performance of a combined-cycle plant using a compressor compression ratio of ten for several hot gas temperatures are given in Table 1. From these results, we may make the following observations:

(a) The highest overall efficiency corresponds to the system having the highest second-law efficiency for the air-turbine air-heater unit.

(b) Electrical energy is recovered more effectively at a higher hot gas temperature.

(c) The ratio of power output of the air-turbine plant to power output of the combined-cycle plant decreases with an increase in the hot gas temperature. This is as expected.

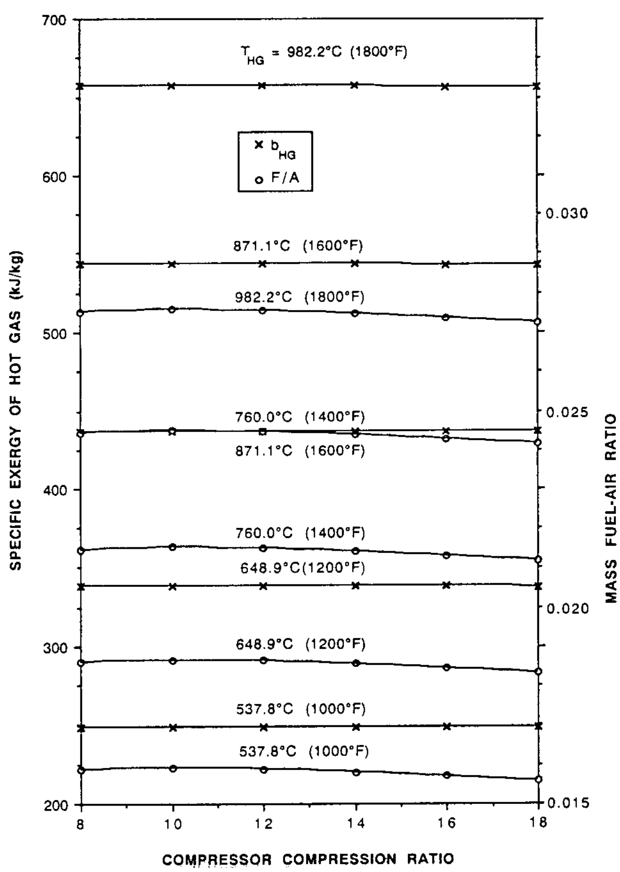

Fig. 5 Effects of compressor compression ratio and hot gas temperature on mass fuel-air ratio and specific exergy of hot gas

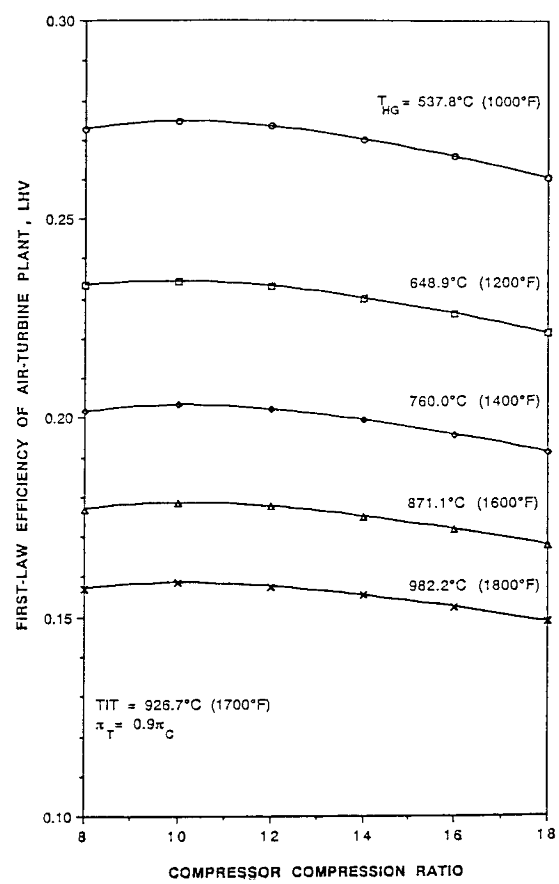

Fig. 6 Effects of compressor compression ratio and hot gas temperature on first-law efficiency of air-turbine air-heater system

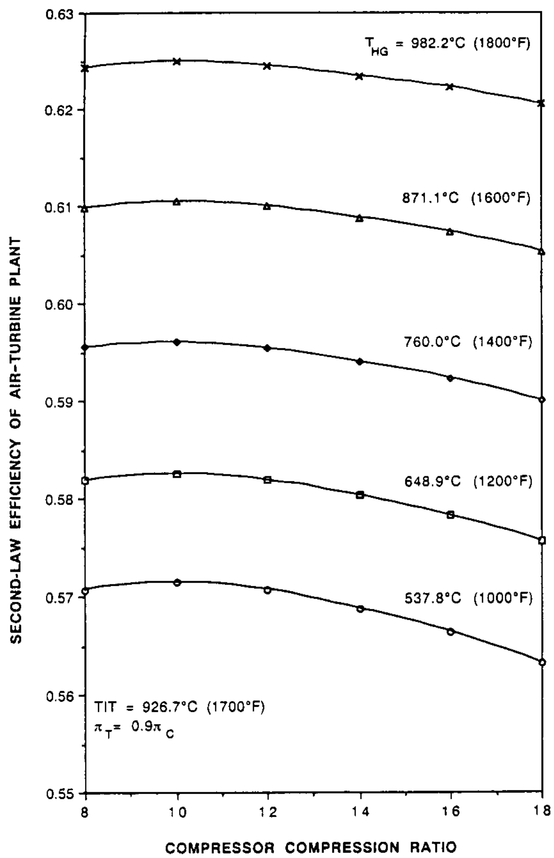

Fig. 7 Effects of compressor compression ratio and hot gas temperature on second-law efficiency of air-turbine air-heater system

CLOSURE

The concept of an air-turbine steam-turbine combined-cycle power plant has been examined. Expressions involving relevant variables for the first-law and second-law efficiencies of the air-turbine air-heater system as well as the first-law and second-law efficiencies of the combined-cycle power plant have been developed. Performance data based on the burning of methane gas have been generated. Important conclusions are as follows:

1. The prerequisite to a high overall efficiency is a high second-law efficiency for the air-turbine air-heater system.

2. At any hot gas temperature, the second-law efficiency of the air-turbine air-heater system is not sensitive to the compressor compression ratio of the air-turbine cycle.

3. Optimum performance may be achieved by using a moderate cycle pressure ratio (say 8 to 14) for the air-turbine cycle with a TIT of 1700°F (926.7°C).

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Table 1. Performance of air-turbine steam-turbine combined-cycle plant (TIT=1700°F; π_C= 10; π_T= 9)

	Hot gas temperature, °F (°C)			
	1000 (537.8)	1100 (593.3)	1200 (648.9)	1300 (704.4)
1. Specific work of air-turbine plant, $(\dot{W}_{el}/\dot{m}_a)_{AP}$, kJ/kg	219.4	219.4	219.4	219.4
2. Specific exergy of hot gas, b_{HG}, kJ/kg	249.0	292.6	338.7	386.8
3. First-law efficiency of air-turbine plant, $(\eta_I)_{AP}$, percent, LHV	27.5	25.3	23.4	21.8
4. Second-law efficiency of air-turbine plant, $(\eta_{II})_{AP}$, percent	57.2	57.7	58.3	58.9
5. First-law efficiency of combined-cycle plant, $(\eta_I)_{CP}$, percent, LHV	44.1	44.6	45.6	47.6
6. Second-law efficiency of combined-cycle plant, $(\eta_{II})_{CP}$, percent, LHV	42.6	43.1	44.1	46.0
7. Electrical energy recovered, $(w_{el})_{SP}$, kJ/kg of hot gas	130.6	163.8	203.4	254.9
8. Steam conditions				
pressure of superheated steam, MPa	5.4	8.2	12.4	18.0
quality of exhaust	0.9	0.9	0.9	0.9
9. Exergy in fuel destroyed in energy recovery system, percent	14.6	14.6	14.2	12.9
10. Ratio of power output of air-turbine plant to power output of combined-cycle plant, percent	62.3	56.8	51.4	45.8

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