JOURNAL OF PROPULSION AND POWER
Vol. 18, No. 2, March–April 2002

Analysis of a High-Efficiency Low-Emissions "Chemical Gas Turbine" System

Takahisa Yamamoto,* Tomohiko Furuhata,† and Norio Arai‡
Nagoya University, Nagoya 464-8603, Japan
and
Noam Lior§
University of Pennsylvania, Philadelphia, Pennsylvania 19104-6315

This paper analyzes the energy, exergy, emissions, and heat exchanger sizes of a novel gas-turbine/steam-turbine combined cycle based on the chemical gas turbine system (ChGT). The system consists of a series arrangement of a fuel-rich and fuel-lean combustor with their gas turbines, recuperators, and a steam bottoming cycle. The fuel-rich combustion in the first stage creates a reducing atmosphere in the first turbine, which allows higher temperature operation (here 1773–2073 K) with carbon-fiber-reinforced-carbon (C/C) composites blades, and produces low NO_x emissions. The second-stage fuel-lean turbine operates at a lower temperature compatible with more conventional blade materials, and completes the fuel combustion. The results show that, compared with conventional combined cycles, the ChGT has distinct advantages, rising with the first-stage TIT: the efficiency is up to 5% higher, the exergy losses are up to 18% lower, the NO_x emissions are up to 40% lower, and the required heat exchanger area is up to 25% lower.

Nomenclature

A = surface area, m^2
C_p = specific heat at constant pressure, J/kg K
C_v = specific heat at constant volume, J/kg K
E = exergy, W
ER = equivalence ratio (actual mole ratio of fuel/air)/(fuel/air mole ratio at stoichiometric conditions), −
H = enthalpy, W
I = internal energy, W
m = mass flow rate, kg/s
P = pressure, Pa
Q = heat, W
S = entropy, W/K
S_{gen} = entropy generation, W/K
T = temperature, K
T_{lm} = log mean temperature, K
TIT1 = turbine inlet temperature of first combustor in the ChGT system, K
TIT2 = turbine inlet temperature of second combustor in the ChGT system, K
U = overall heat-transfer coefficient, W/m^2 K
V = volume, m^3
v = specific volume, m^3/kg
W = work, W
γ = specific heat ratio C_v/C_p, −
ΔP = pressure drop, Pa
ΔT_{lm} = log mean temperature difference, K
η = isentropic efficiency of each component, −
π = pressure ratio P_{out}/P_{in}, −

Subscripts

burn = burning state
CP = compressor
c = cold stream
GT = gas turbine
h = hot stream
in = inlet state
mix = mixing state
out = outlet state
P = pump
ST = steam turbine
0 = ambient state

Received 7 January 2001; revision received 1 July 2001; accepted for publication 17 July 2001.

*Ph.D. Student, Research Center for Advanced Energy Conversion, Furo-cho, Chikusa-ku.

†Assistant Professor, Research Center for Advanced Energy Conversion, Furo-cho, Chikusa-ku.

‡Professor, Research Center for Advanced Energy Conversion, Furo-cho, Chikusa-ku. Senior Member AIAA.

§Professor, Department of Mechanical Engineering and Applied Mechanics.

Introduction

THIS study contributes to the intense ongoing worldwide effort for increasing power generation efficiency and reducing the associated emissions of pollutants. Its main features are 1) use of a combined Brayton/Rankine-type cycle, 2) allowing increase of the turbine inlet temperature (TIT) by using novel turbine blade materials, and 3) fuel stoichiometry manipulation in the Brayton cycle combustors to reduce NO_x production and turbine blade damage.

An obvious method for increasing efficiency is to raise the TIT: a 100 K rise in the TIT of such a Brayton cycle results in increasing the thermal efficiency by about 1%. To increase the TIT, higher temperature endurance turbine blades are being developed in many countries by using increasingly advanced materials and cooling techniques.[1,2] For example, Mitsubishi Heavy Industries[3] has developed in recent years a high-temperature gas-turbine system, which realizes a TIT of 1773 K. This was made possible by using a steam-cooled blade system, single-crystal alloy blades, and thermal barrier coating techniques. The thermal efficiency with only a gas turbine reached 38% [low heating value (LHV) base], and its total combined-cycle thermal efficiency including a bottoming steam turbine reached 58%. Because cooling of turbine blades also involves efficiency losses, it is important to focus on the development of improved heat-resistant materials. A special effort in that direction is to use carbon-fiber-reinforced-carbon (C/C) composites[4,5] as there are currently no other materials that are tough enough for turbine blade use at temperatures above 2073 K. They are also used in the manufacture of the nose-cone leading edges of space shuttles and brakes of jet fighters. These are composed of carbon fibers as a stiffener and carbonized lamination of pitch or resin, or CVD carbon, as a matrix. Turbine blades, which are light, tough at high temperatures, and resistant to heat, have recently been indeed made from these materials, but are still in the development stage. Some of the major companies active in this area are SEP Industrie in France; and the Mechanical Engineering Laboratory of MITI, Mitsubishi Heavy Industries, Kawasaki Heavy Industries, Tokai Carbon, the Ultra-High Temperature Material Research Center at Ube, and TYK (Tokyo Yougyou Kougyu) in Japan. Extensive engineering information on forming C/C composites has been developed to improve the production of turbine made from such materials, and model turbines were fabricated for spin tests at normal temperatures, but much more research and design is required before practical C/C composites gas turbines become available.[6,7]

A key problem is that C/C composites age rapidly in high-temperature (>873 K) oxidizing atmospheres, greatly reducing their strength. Consequently, they are not yet directly applicable for use in gas turbines. To prevent this problem, studies are being carried out for the improvement of oxidation resistance by coating C/C composites with ceramics using CVD or chemical vapor infiltration processes.

The third author had a part in studying coating techniques of C/C composites, which reached 10 hours of successful operation under high-temperature oxidizing conditions.[8]

Chemical Gas-Turbine System

In response to these challenges for the development of high-efficiency low-emissions power generation systems, the Research Center for Advanced Energy Conversion, Nagoya University, has launched a research and design program, which resulted in creating the novel power system briefly outlined earlier.[9–11]

The schematic diagram of the proposed chemical gas-turbine (ChGT)/steam-turbine combined-cycle system is shown in Fig. 1. It consists of a fuel-rich combustor, a fuel-lean combustor, two sets of gas turbines, a steam turbine, a recuperator, and heat recovery steam generators (HRSG). An important feature of this system is the introduction of a fuel stoichiometry manipulation technique with fuel-rich combustion in the first combustor and fuel-lean combustion in the second.[12,13] Considering that the only promising materials for turbine blades operating at temperatures above 1773 K without internal cooling are C/C composites, fuel-rich (i.e., oxygen-lean) combustion, as well as coating with C/C composites, are employed to prevent their deterioration at these high temperatures. The second turbine is operated at temperatures of at most 1773 K, where conventional blades could thus tolerate fuel-lean conditions.

Firstly, compressed air and fuel are mixed and burned under fuel-rich (substoichiometric) conditions in the first combustor (CB-1). Exhaust gas from the first gas turbine (GT-1) would still contain chemical energy as possibly remaining methane, as well as hydrogen and carbon monoxide, which result from the fuel-rich combustion, and is mixed with compressed air before being burned under fuel-lean conditions in the second combustor (CB-2). Exhaust gas from the second combustor drives the second gas turbine (GT-2) operating at lower temperatures at which the construction materials are much less susceptible to oxidation. Heat from both the GT-1 and the GT-2 exhaust gases is used in the HSRG to generate the steam in the steam bottoming cycle. Although staged fuel-rich, fuel-lean combustion is well known, the system described here was named the ChGT because it has other distinctive features such as 1) the rich combustor here is also a chemical reactor, which produces fuel (hydrogen and CO) for the next combustor; 2) it incorporates important internal heat recovery; and 3) it has a bottoming cycle.

This paper presents a study and thermodynamic analysis of this ChGT. We have focused on the thermal efficiency, the exhaust gas emissions, exergy analysis, and the size of the required HRSG. The dependence of the thermal efficiency of this system on the equivalence ratio in CB-2 was examined. These aspects were also compared with conventional combined cycles.[14–16]

Simulation Model

The ChGT system was analyzed using the process simulator HYSYS.[17] The exergy analysis was performed using the HYSYS thermodynamic simulation results, including properties, concentrations, and mass flows, at each cycle point. Details of the exergy analysis are described in the next section. All of the ChGT results were compared with the conventional combined cycle (CCC) described in Fig. 2.

A summary of the assumptions for the ChGT system is given in Table 1. Figure 3 shows some detail of the CB-2 of the ChGT system and of the CB of the CCC. The equivalence ratio at CB-2 was varied to consider its influence on system performance. Once the equivalence ratio was fixed at CB-2, TIT2 was controlled by injecting colder air at into the GT2 inlet gas stream (point 11, Fig. 2). The turbine inlet temperature of the CCC was controlled in the same way.

The ambient air was assumed in the analysis to be dry, at 298 K, 0.1013 MPa, and the fuel was methane with a lower heating value of 50.01 kJ/kg, supplied at the required pressure from the gas mains at a mass flow rate of 383.45 kg/h, for a total heat input of 19.176 MJ/h.

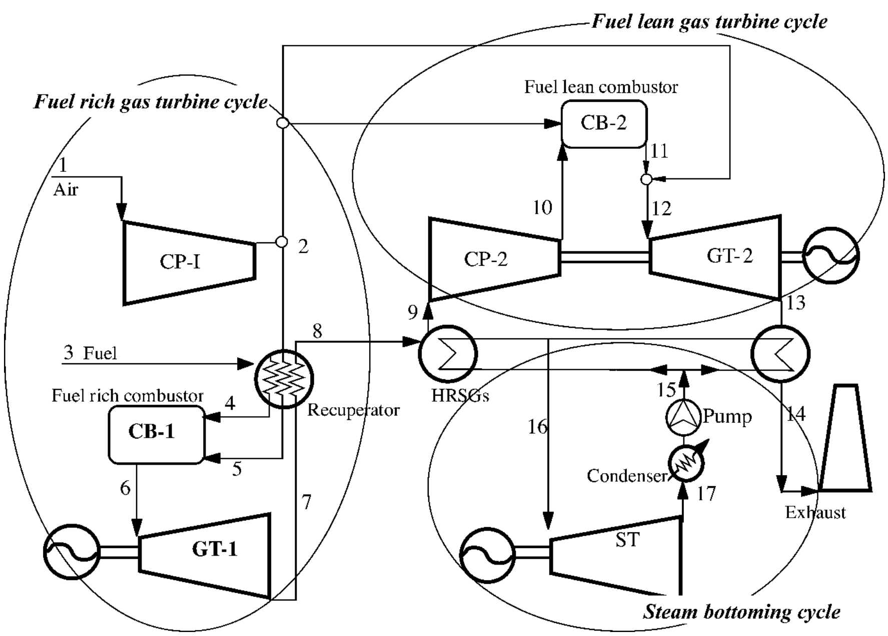


Fig. 1 Flow diagram of the combined cycle based on the ChGT.

Table 1 Summary of the main assumptions in the simulation of the ChGT and conventional combined cycle

Location and parameter	Chemical gas-turbine system	Conventional combined cycle
Gas-turbine cycle		
Fuel-rich stage		
Turbine efficiency, %	92	——
Compressor efficiency, %	91	——
Pressure ratio, –	20	——
Turbine inlet temperature, K	1773, 1873, 1973, 2073	——
Fuel-lean stage		
Turbine efficiency, %	92	92
Compressor efficiency, %	91	91
Pressure ratio, –	20	20
Turbine inlet temperature, K	1773	1773
Steam turbine		
Steam-turbine efficiency, %	90	90
Pump efficiency, %	92	92
Condenser pressure, MPa	0.05	0.05
Heat recovery steam generator		
Pressure drop, %	3	3
Steam pressure, MPa	14	14

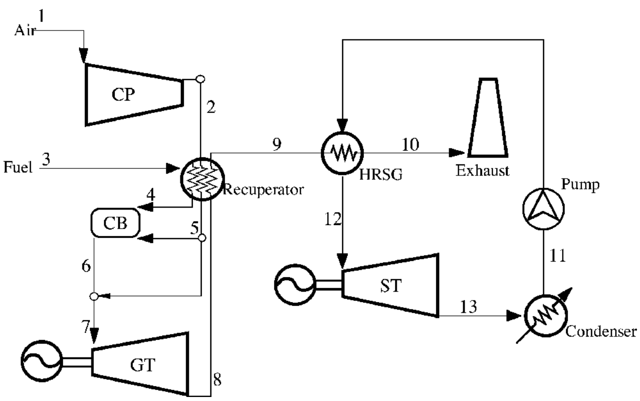


Fig. 2 Flow diagram of the conventional combined cycle used for comparisons.

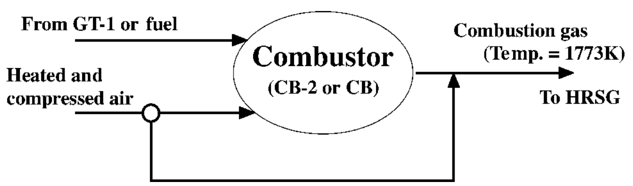


Fig. 3 Flow detail for the ChGT (Fig. 1) and for the CB of the conventional combined cycle (Fig. 2).

The steam bottoming cycle chosen for the analysis uses two steam generators supplying steam to the turbine at the same conditions of 823 K, 14.2 MPa. The minimal temperature approach between hot and cold streams in the HRSG was assumed to be[15,17,18] 15 K.

Chemical reaction equilibrium calculations (Gibbs free energy minimization) in both the fuel-rich (CB-1) and fuel-lean (CB-2) combustors were carried out. The chemical species considered include CH_4, H_2, N_2, O_2, CO, CO_2, and NO_x composed of NO, N_2O, and NO_2. Soot formation in the fuel-rich combustor is a possibility, but was not included in this study because its formation mechanism is still not completely clear and because its computation requires knowledge of the geometry of the combustor and of the flow field and the species' residence time in it. At this stage of the study, the detailed combustor configuration was not specified yet. There is also recent evidence (Ref. 19 and the authors' own work) that NO_x production is independent of, or may even be slightly reduced, in the presence of soot formation. Although equilibrium calculations are not strictly correct (for example, the predicted NO_x concentration would be higher than actual), they do reflect the trends of the actual exhaust gas compositions.

Gas-Turbine and Steam-Turbine Analysis

The power output of gas turbine and steam turbine is calculated as a function of the mass flow rate $\dot{m}$, the pressure ratio π, by

$$W_{\mathrm{ST,GT}} = \dot{m}C_p\eta_{\mathrm{ST,GT}}(T_{\mathrm{in}} - T_{\mathrm{out}}) = \dot{m}C_p\eta_{\mathrm{ST,GT}}\left[1 - \pi^{(1-\gamma)/\gamma}\right] \tag{1}$$

General approach for improving turbine power is increasing T_{in} in Eq. (1) by means of advanced blade material. The ChGT system is able to enhance TIT drastically using the C/C composites turbine blade and the fuel-rich combustion.

Heat Exchanger Analysis

The HRSG need to operate at pressures over 10 MPa and temperatures of about 900 K, are of significant size, and are thus economically important. We have hence also analyzed their total size as a function of the overall heat-transfer coefficient U and the heat-transfer area A as

$$UA = Q/\Delta T_{\mathrm{lm}} \tag{2}$$

where ΔT_{lm} is the overall log mean temperature difference (LMTD) defined as

$$\Delta T_{\mathrm{lm}} = \frac{(T_{\mathrm{hot,out}} - T_{\mathrm{cold,in}}) - (T_{\mathrm{hot,in}} - T_{\mathrm{cold,out}})}{\ell n\{(T_{\mathrm{hot,out}} - T_{\mathrm{cold,in}})/(T_{\mathrm{hot,in}} - T_{\mathrm{cold,out}})\}} \tag{3}$$

Exergy Analysis

An exergy analysis of the combined cycle based on the ChGT system and of the CCC was carried out, and the results were also compared with other published cycle analyses.[16,18,20,21] The flow exergy E is calculated using the equation

$$E = (H - H_0) - T_0(S - S_0) \tag{4}$$

The exergy loss of each process, particularly combustion, heat transfer, and the mechanical power components, was also calculated as follows.

Exergy Loss in the Combustion Process

The exergy loss in the combustion process is the exergy difference between the gases entering and exiting the combustor:

$$E_{\text{CB-loss}} = T_0[S_{\mathrm{burn}}(T_{\mathrm{burn}}) - S_{\mathrm{mix}}(T_{\mathrm{mix}})] \tag{5}$$

where S_{burn} and S_{mix} are the entropy of the combustion products and of the precombustion gas, respectively. Generally speaking, the irreversibility of a combustion process is caused by the diffusion of the chemical species, reaction and internal heat transfer, with the latter being the dominant.[22,23]

Exergy Loss in the Recuperator and Steam Generator

These exergy losses are caused by heat transfer and flow pressure drops. The entropy generation rate S_{gen} in these heat exchangers, caused by heat-transfer loss, is calculated by the equation

$$S_{\mathrm{gen}}|_Q = Q(1/T_{\mathrm{lm,c}} - 1/T_{\mathrm{lm,h}}) \tag{6}$$

where T_{lm} is the LMTD of the hot (subscript h) or cold (subscript c) streams

$$T_{\mathrm{lm}} = \frac{T_{\mathrm{out}} - T_{\mathrm{in}}}{\ell n(T_{\mathrm{out}}/T_{\mathrm{in}})} \tag{7}$$

The entropy generation rate caused by flow pressure drop loss is calculated by the equation

$$S_{\mathrm{gen}}|_{\Delta P} = \left(\frac{m v_{\mathrm{in}}\Delta P}{T_{\mathrm{in}}}\right)_h + \left(\frac{m v_{\mathrm{in}}\Delta P}{T_{\mathrm{in}}}\right)_c \tag{8}$$

The total entropy generation in the heat exchangers is thus

$$S_{\text{gen}} = S_{\text{gen}}|_Q + S_{\text{gen}}|_{\Delta P} = Q\left(\frac{1}{T_{\text{lm,c}}} - \frac{1}{T_{\text{lm,h}}}\right) + \left(\frac{m v_{\text{in}} \Delta P}{T_{\text{in}}}\right)_h - \left(\frac{m v_{\text{in}} \Delta P}{T_{\text{in}}}\right)_c \tag{9}$$

and the total exergy loss is thus

$$E_{\text{loss}} = T_0 \cdot S_{\text{gen}} = T_0\left[Q\left(\frac{1}{T_{\text{lm,c}}} - \frac{1}{T_{\text{lm,h}}}\right) + \left(\frac{m v_{\text{in}} \Delta P}{T_{\text{in}}}\right)_h - \left(\frac{m v_{\text{in}} \Delta P}{T_{\text{in}}}\right)_c\right] \tag{10}$$

The pressure drops ΔP incurred for each stream in a heat exchanger are assumed to be 3% of the inlet pressure for the stream.

Exergy Loss in the Mechanical Components

These components are the gas turbines, compressors, pumps and the steam turbine, in all of which the exergy losses are calculated by the equation

$$|E_{\text{in}} - E_{\text{out}}| = E_{\text{loss}} + W_{\text{GT,ST,CP,P}} \tag{11}$$

where E is the exergy calculated by Eq. (3) and $W_{\text{GT,ST,CP,P}}$ is the gas and steam turbine output power or compressor and pump input work, respectively.

Simulation Results and Discussion

In the chemical gas turbine system TIT1 (the TIT of GT-1) is controlled by prescribing the equivalence ratio at CB-1, and this relationship is shown in Table 2. Some of the parameters, such as maintaining the gas pressure in the HSRG at atmospheric pressure, have been chosen to increase system efficiency, but a formal optimization in which all relevant parameters are changed to maximize some objective function was not conducted. The equivalence ratio at CB-2 is then varied to examine its influence on the total thermal efficiency of the system, while TIT2 (the TIT of GT-2) is set at 1773 K by regulating, as just mentioned, the quantity of bypass airflow to GT2 at point 11 (Fig. 1). The results, in comparison with those for a CCC in which the equivalence ratio is varied in the same range, are shown in Fig. 4. Pressures, temperatures, and mass flow rates throughout the cycle for two representative cases for the ChGT are shown in Table 3 and for the CCC in Table 4.

Table 2 Equivalence ratios at CB-1, corresponding to the GT-1 turbine inlet temperatures considered in the simulation

Chemical gas-turbine system	
Turbine inlet temperature of GT-1, K	Equivalence ratio at CB-1, –
1773	2.831
1873	2.021
1973	1.963
2073	1.910

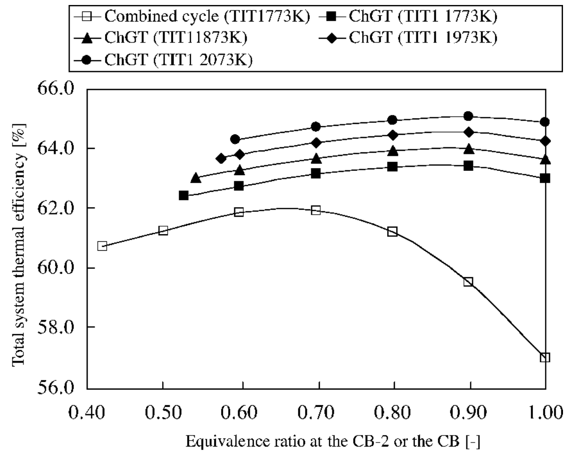


Fig. 4 Influence of the equivalence ratio at CB-2 on the efficiencies of the ChGT and of the conventional combined cycle.

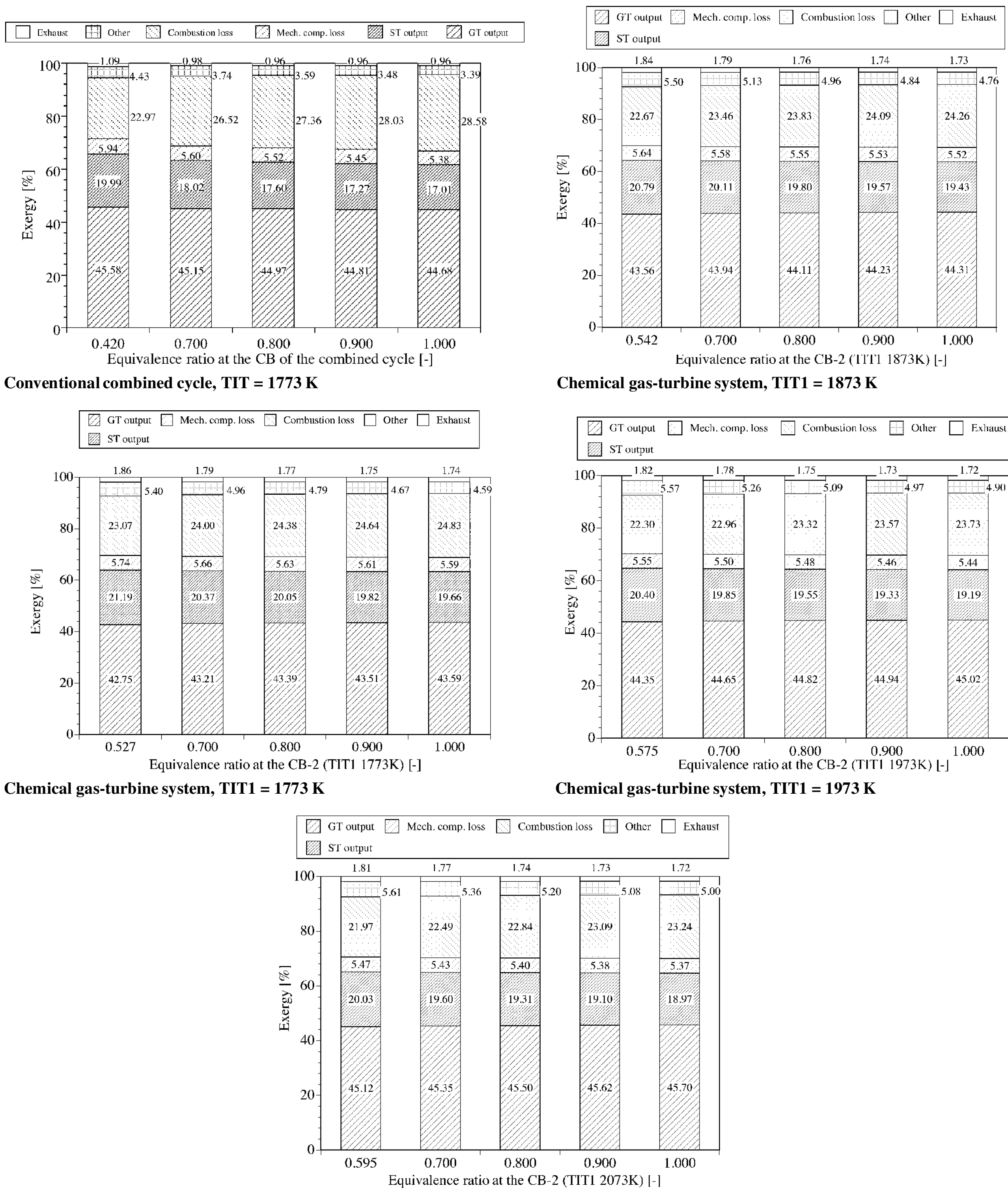
Table 4 CCC conditions for TIT = 1773 K, *ER* = 0.90

No.	Combined cycle, TIT = 1773 K, *ER*(CB) = 0.90 Pressure, MPa	Temperature, K	Mass flow rate, kg/h
1	0.101	298	13,227
2	2.087	724	13,227
3	2.087	298	383
4	2.026	716	383
5	2.026	724	13,227
6	2.026	2,476	7,680
7	2.026	1,773	13,611
8	0.107	968	13,611
9	0.104	872	13,611
10	0.101	373	13,611
11	14.182	306	2,575
12	13.757	823	2,575
13	0.05	306	2,575

Table 3 ChGT conditions for two representative cases

	Chemical gas-turbine system, TIT = 1773 K, *ER*(CB-2) = 0.90			Chemical gas-turbine system, TIT = 2073 K, *ER*(CB-2) = 0.90		
No.	Pressure, MPa	Temperature, K	Mass flow rate, kg/h	Pressure, MPa	Temperature, K	Mass flow rate, kg/h
1	0.101	298	10,252	0.101	298	8,690
2	2.087	724	10,252	2.087	724	8,690
3	2.087	298	383	2.087	298	383
4	2.026	481	383	2.026	778	383
5	2.026	724	3,113	2.026	778	3,821
6	2.026	1,773	3,113	2.026	2,073	3,821
7	0.107	940	3,113	0.107	1,095	3,821
8	0.104	905	3,113	0.104	925	3,821
9	0.101	373	3,113	0.101	373	3,821
10	2.026	876	3,113	2.026	872	3,821
11	2.026	2,107	7,333	2.026	2,045	6,950
12	2.026	1,773	10,635	2.026	1,773	9,074
13	0.104	943	10,635	0.104	978	9,074
14	0.101	373	10,635	0.101	373	9,074
15	14.182	306	2,956	14.182	306	2,838
16	13.757	823	2,956	0.06	823	2,838
17	0.05	306	2,956	0.05	306	2,838

The maximal efficiency of the CCC was found to be 62% (LHV base) and that of the ChGT 63.4% for the same turbine inlet temperature (1773 K) conditions. The system efficiency of the ChGT is found to continue rising with the equivalence ratio beyond the maximum attained by the CCC. Furthermore, it also rises with TIT1, temperatures which can be attained by using the C/C composites blades. The maximal efficiency of the ChGT was computed to be over 65% for TIT1 = 2073 K and $ER \approx 0.9$. The ChGT system with TIT1 = 2073 K thus has a thermal efficiency about 5% higher than that of the CCC and 2.5% higher than that of the ChGT cycle with TIT1 = 1773 K. The equivalence ratio at which the efficiency has a maximum increases with TIT1, but only slightly. The presence of a maximum in the efficiency is caused by the changes in compressor energy demand and turbine output as a result of the total enthalpy changes across these devices, created primarily by the variation in the overall gas mass flow rate, as follows. The amount of input fuel (methane) is kept constant in this analysis. Increase the equivalence ratio (ER) is obtained by lowering the airflow rate entering CB-2, which, in turn, raises the CB-2 exit temperature. To bring this temperature down to the desired TIT2 level, air, which has not passed



Conventional combined cycle, TIT = 1773 K

Chemical gas-turbine system, TIT1 = 1873 K

Chemical gas-turbine system, TIT1 = 1773 K

Chemical gas-turbine system, TIT1 = 1973 K

Chemical gas-turbine system, TIT1 = 2073 K

Fig. 5 Results of the exergy analysis of the conventional combined cycle and of the ChGT (the number in the bars are the exergy percentages, relative to the input fuel exergy, for each of the exergy losses and outputs).

Table 5 Effect of the ER in CB-2 on ChGT flow rates for TIT1 = 1773 K

Chemical gas-turbine system (TIT of GT-1 = 1773K)					
Eq. of CB2, –	Comb. gas from GT1, kg/h	Comp. air at the CB2 inlet, kg/h	Comb. gas temperature from CB2, K	Injection air for cooling, kg/h	Total mass flow of the exhaust, gas kg/h
0.527	3536	6318	1773	0	9854
0.600	3536	5692	1839	661	9888
0.700	3536	4879	1936	1511	9925
0.800	3536	4269	2020	2132	9936
0.900	3536	3795	2090	2583	9913
1.000	3536	3415	2140	2838	9789

through CB-2, must be injected into the CB-2 exhaust stream before it enters GT2. As shown in Table 5, the total amount of gas entering GT-2 and air entering CP-1 thus varies, going through a maximum at some value of ER. The enthalpy is affected primarily by the mass flow rate because the specific enthalpy change at state 12 (Fig. 1) is relatively small because it is, at the fixed TIT2, affected only by the gas composition there. A similar explanation is valid for the presence of the efficiency maximum in the conventional combined cycle.

Figure 5 shows the results of the exergy analysis of both the CCC and the ChGT at four values of TIT1. The numbers indicate exergy values relative to the fuel input exergy. As for the conventional combined cycle, increasing the ER decreases slightly the power output and increases the combustion exergy loss. Especially, the combustion loss is the highest in the system, amounting to about one-third of total.

As for the ChGT, the turbines' power again decreases as the ER at CB-2 increases, and the exergy loss causing the combustion process decreases. That decrease is caused by the smaller amounts of air introduced to CB-1 and thus reduced mixing and internal heat-transfer losses.[23] Although other exergy analyses of combined cycles[18,20,21] agree with ours that the combustion exergy loss is dominant, the losses in the ChGT are about 2–5% lower, probably because of the lower flow rates required for the same power output. Increasing the GT-1 turbine inlet temperature increases also its power output but decreases the GT-2 and steam turbine power output, resulting in an increase of the total gas turbine power output. When both systems are compared for the same GT and GT-1 TIT (1773 K), one can see that for the ChGT the steam turbine power is higher by 1 or 2%, whereas the overall gas turbine output is lower by almost the same amount. The combustion exergy loss in the ChGT is lower by about 11%, rising to 18% as TIT1 rises to 2073 K.

After generating some power in the first stage gas turbine (GT-1), the gas contains not only thermal but also chemical energy in the form of H_2 and CO (a syngas fuel mixture) generated in the fuel rich combustor CB-1. The thermal energy is recovered in the recuperator and HRSG. The chemical energy is passed as a fuel to the fuel-lean combustor powering the GT-2 turbine. Figure 6 shows the concentration of the main species emanating from CB-1, hydrogen, oxygen, carbon monoxide, carbon dioxide, and water. As just explained, TIT1 is an inverse function of ER and is also shown in Fig. 6.

As expected for fuel-rich combustion, H_2 and CO concentrations increase with the ER of CB-1 (and thus decrease with increasing TIT1), while H_2O decreases. The CO_2 and O_2 concentrations decrease slightly. The O_2 concentration is near zero because it was assumed that the reaction in CB-1 (fuel-rich) was completed to equilibrium, and thus the methane was also fully consumed.

Figure 7 shows the LHV of the syngas generated by fuel-rich stage in comparison with that of the methane fuel input. The difference between their LHV is the energy of fuel that was used in the fuel-rich stage, and the LHV of the syngas is that available for use in the fuel-lean stage.

One of the main objectives for developing the ChGT is the reduction of emission of nitrogen oxides. Figure 8 indicates the computed emissions on a power basis,[16] compared with those from the CCC, at the highest efficiency conditions. Noting that NO_x emissions from CB-1 are affected only slightly with the increase of TIT1 because of the fuel-rich conditions there, there are two obvious reasons why

Table 6 Efficiencies and NO_x emissions on power basis of gas-turbine systems[16,24]

System	Thermal efficiency, [%] (TIT = 1673 K)	NO_x emissions, (mg/s-kW)
Simple gas turbine	42.0	81.8
Recuperated gas turbine	48.0	80.0
Steam injection gas turbine	50.0	69.0
Intercooling/recuperated gas turbine	52.0	54.8
HAT (Humid Air Turbine)	56.0	55.5
ChGT system (TIT1 = 1773 K)	63.4	17.4

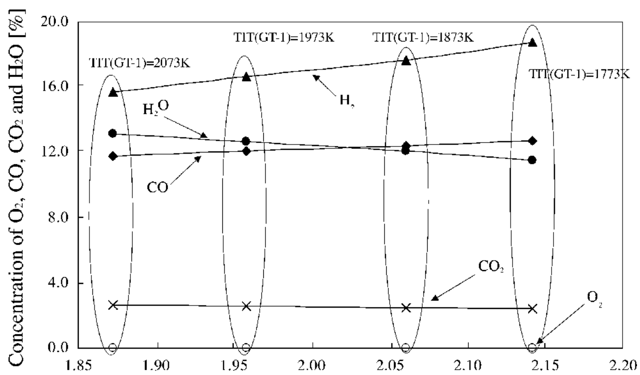


Fig. 6 Concentrations of major chemical species in the fuel-rich stage outflow.

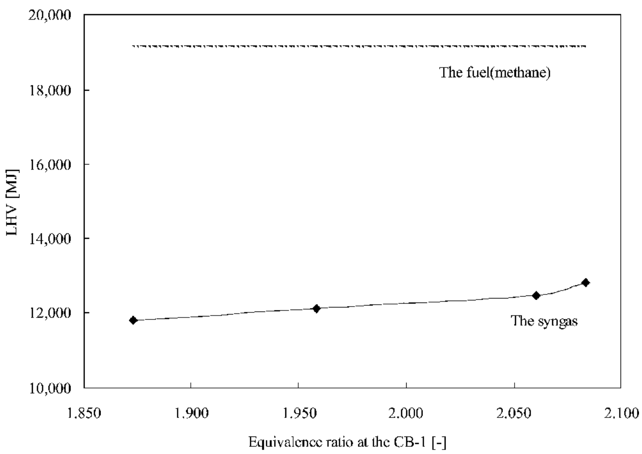


Fig. 7 Low heating value (LHV) of the syngas produced in the fuel rich stage, and of the methane.

the ChGT reduces overall power-based NO_x emissions: 1) as can be seen from Table 5, the exhaust mass flow decreases, and 2) the thermal efficiency increases. The CCC has the highest level of nitrogen oxides emissions. In the ChGT the emissions decrease as the turbine inlet temperature at GT-1 increases: at TIT1 = 2073 K the NO_x emissions are about 80% lower than those from the CCC, and 30% lower that those from the ChGT at TIT1 = 1773 K. As just shown, increasing TIT1 also increases the thermal efficiency, thus

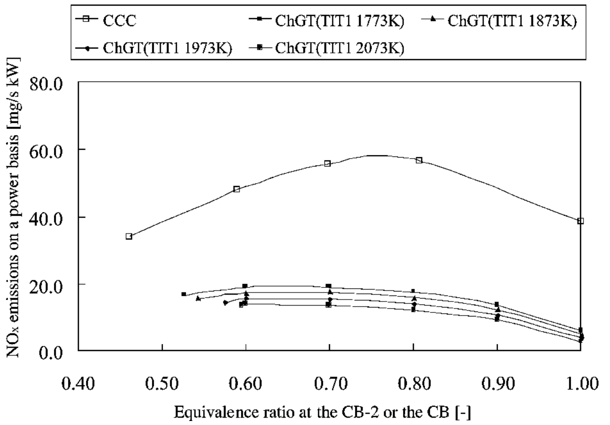


Fig. 8 NO_x emissions of the ChGT system in comparison with the CCC.

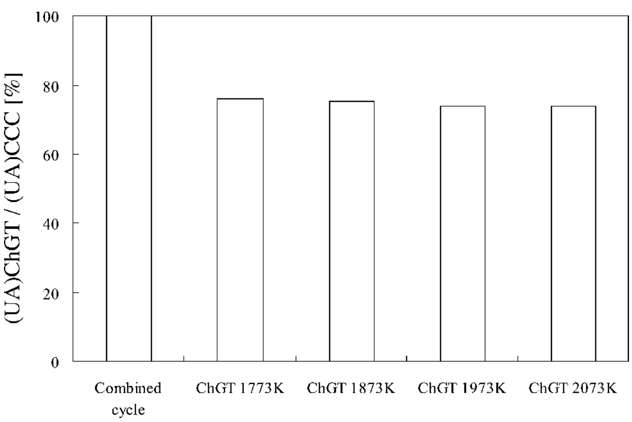


Fig. 9 UA ratio of the HRSGs: [UA for the ChGT]/[(UA for the conventional combined cycle (CCC)].

demonstrating the ability of the ChGT to decrease emissions and simultaneously increase efficiency.

A comparison of our results with those obtained for other gas turbine systems by Gallo[16] and by us[24] (for simple gas-turbine systems) is shown in Table 6. Gallo[16] analyzed the efficiencies of a simple gas-turbine, recuperated gas-turbine, steam-injection gas-turbine, intercooling, recuperated gas-turbine, and humid air-turbine (HAT) systems. The assumptions in these analyses are almost the same as those in the present study (e.g., pressure drop 3%, blade cooling losses). The predicted ChGT efficiency is 10–15% higher, and ChGT is also able to reduce the NO_x emissions per unit power produced, as compared with the HAT cycle. The main reasons for this advantage of the ChGT, in addition to the low amounts of NO_x produced in the fuel-rich CB1, are its higher efficiency and lower gas flow rates per unit power produced.

The computed sizes of the HRSG at the highest efficiency conditions, relative to the size required for the conventional combined cycle, are shown in Fig. 9, and they are seen to be lower than the latter by 25%, with the size slightly decreasing as TIT1 increases. The explanation for this reduced area requirement is that 1) because of the higher efficiency of the ChGT cycle, also increasing with TIT1, more of the fuel energy is used for power production and thus less is available for internal heat recovery; 2) relatively less of the power is produced in the steam cycle; and 3) the inlet temperature of the HRSG in the ChGT is about 50–100 K higher than that in the conventional combined cycle (Tables 3 and 4), and so is, therefore, the LMTD. The reduced size of the HSRG needed for the ChGT is thus yet another advantage of this system relative to conventional cycles.

Conclusions

An analysis of the novel combined gas-turbine/steam-turbine cycle system, the chemical gas turbine system, was carried out, in comparison with a conventional combined cycle. Energy, exergy, heat-exchanger size, and emissions concentrations were computed for a number of relevant conditions. The key advantage of this system is the combination of 1) fuel-rich combustion in the first-stage turbine combustor (CB-1), which minimizes NO_x production and also allows in that oxygen depleted atmosphere the use of C/C composites blades for turbine operation at temperatures higher than the state-of-the-art maximum of about 1773 K, to achieve higher efficiency, and 2) fuel-lean combustion in a second stage to complete the fuel use, at state-of-the-art temperatures, which a more conventional turbine can tolerate (1773 K in this analysis). The analysis confirms these key advantages of the ChGT:

1) The thermodynamics analysis shows that an overall thermal efficiency above 65% (LHV basis) might be possible. The efficiency increases with the turbine TIT, and a maximum is obtained for an equivalence ratio of 0.9 at the second-stage combustor (CB-2). This efficiency is higher than that of the analyzed CCC by 5%.

2) The exergy analysis confirms that the dominant exergy loss is in the combustion process. In the ChGT system this loss is 11–18% smaller than that in the conventional combined cycle, and it is reduced from a value of 24.6% to a value of 23.0% as TIT1 rises from 1773 to 2073 K.

3) The overall NO_x emissions from the ChGT are up to 40% lower than those from the CCC, the reduction increasing with TIT1.

4) The UA required for the HSRG of the ChGT was found to be about 25% lower than that required for the CCC, and it decreases with TIT1, both results favoring ChGT economics.

To bring the ChGT system to practice, more research is needed on the development and use of the C/C composite turbine blades and on high-pressure high-temperature fuel-rich combustion.

References

[1] St. Pierre, G. R., "Advanced Materials and Coatings for Energy Conversion Systems," *Energy Conversion and Management*, Vol. 38, Nos. 10–13, 1997, pp. 1035–1041.

[2] Oki, T., "Surface Modification Technique and its Trends," *Proceedings of the 5th International Symposium on Ultra-High Temperature Materials '95*, edited by R. Naslain, Japan Ultra High Temperature Material Inst., Tajimi, Japan, Vol. 1, 1995, p. 54.

[3] Ishii, J., "High Efficiency Combined Cycle Plant with 1500°C-Class Gas Turbine," *Journal of the Gas Turbine Society of Japan*, Vol. 27, No. 3, 1999, pp. 61–65.

[4] Kettunen, P., "Carbon-Carbon Composites," *Materials Science and Engineering*, Vol. 234-234, No. 1, 1997, pp. 223–225.

[5] Kinoshita, K., *Carbon*, Wiley-Interscience, New York, 1998, pp. 86–131.

[6] Wei, W.-C., "Oxidation of Carbon/Carbon Composite Coated with SiC-(Si/ZrSi$_2$)-ZrSi$_2$," *Carbon*, Vol. 32, No. 5, 1994, pp. 605–613.

[7] Fushitani, K., Kobayashi, N., and Arai, N., "Evaluation of High Temperature Oxidation Behavior of Carbon-Carbon Composites Exposed in a Field of Combustion by Using Raman Spectroscopy," *Journal of Chemical Engineering in Japan*, Vol. 30, No. 3, 1997, pp. 580–582.

[8] Kato, Y., Kakamu, K., Hironaka, Y., Arai, N., Kobayashi, N., and St. Pierre, G. R., "Improvement of High-Temperature Endurance of C/C Composites by Double Coating with SiC and Glass Materials," *Journal of Chemical Engineering in Japan*, Vol. 24, No. 4, 1996, pp. 669–674.

[9] Arai, N., and Kobayashi, N., "Challenges for Development of Highly Efficient Gas Turbine System: The Chemical Gas Turbine System," *American Society of Mechanical Engineers 1997 International Joint Power Generation Conference*, edited by A. Sanyal, ASME, NY, Vol. 1, 1997, pp. 423–429.

[10] Lior, N., and Arai, N., "Analysis of the Chemical Gas Turbine System, a Novel High-Efficiency Low-Emissions Power Cycle," *American Society of Mechanical Engineers 1997 International Joint Power Generation Conference*, EC Vol. 5, edited by A. Sanyal, ASME, NY, Vol. 1, 1997, pp. 431–437.

[11] Lior, N., and Arai, N., "Performance and Heat Transfer Considerations in the Novel Chemical Gas Turbine System," *AIAA/American Society of Mechanical Engineers Joint Thermophysics and Heat Transfer Conference*, edited by J. Velleux, ASME, NY, Vol. 4, 1998, pp. 33–39.

[12] Yamamoto, T., Kobayashi, N., Arai, N., and Tanaka, T., "Effect of Pressure on Fuel-Rich Combustion of Methane-Air Under High Pressure," *Journal of Energy Conversion and Management*, Vol. 38, Nos. 10–13, 1997, pp. 1093–1100.

[13] Kobayashi, N., Mano, T., and Arai, N., "Fuel-Rich Hydrogen-Air Combustion for a Gas Turbine System with no Emission of Carbon Dioxide," *Energy*, Vol. 22, No. 2, 1997, pp. 189–197.

[14]Lior, N., "Advanced Energy Conversion to Power," *Energy Conversion and Management*, Vol. 38, Nos. 10–13, 1997, pp. 941–955.

[15]Bolland, O., and Stadaas, J. F., "Comparative Evaluation of Combined Cycles and Gas Turbine Systems with Water Injection, Steam Injection and Recuperation," *Journal of Engineering for Gas Turbines and Power*, Vol. 117, No. 1, 1995, pp. 138–145.

[16]Gallo, W., "A Comparison Between the HAT Cycle and Other Gas Turbine Based Cycles: Efficiency, Specific Power and Water Consumption," *Energy Conversion and Management*, Vol. 38, Nos. 15–17, 1997, pp. 1595–1604.

[17]Hyprotech Co., HYSYS, Ver. 1.1 Reference, Calgary, Canada, Vols. 1–2, 1997.

[18]Oh, S.-D., Pang, H.-S., and Yong, K.-H., "Exergy Analysis of a Gas Turbine Cogeneration System," *Journal of Engineering for Gas Turbines and Power*, Vol. 118, No. 4, 1996, pp. 782–791.

[19]Beltrame, A., Porshnev, P., Merchan, W., Saveliev, A., Fridman, A., Kennedy, L. A., Petrova, O., Zhdanok, S., Amouri, F., and Charon, O., "Soot and NO Formation in Methane-Oxygen Enriched Diffusion Flames," *Combustion and Flame*, Vol. 124, No. 1, 2001, pp. 295–310.

[20]Kozu, M., and Tsuruno, S., "Exergy Analysis of Combined Cycle Using T-S Diagram," *Journal of the Gas Turbine Society of Japan*, Vol. 22, No. 87, 1994, pp. 64–72.

[21]Horlock, J. H., *Combined Power Plants*, Pergamon, Oxford, England, U.K., 1992, pp. 64–72.

[22]Kehohofer, R., *Combined Cycle Gas and Steam Turbine Power Plants*, Fairmont Press, Oxford, U.K., 1991, pp. 184–221.

[23]Dunbar, W. R., and Lior, N., "Sources of Combustion Irreversibility," *Combustion Science and Technology*, Vol. 103, 1994, pp. 41–61.

[24]Yamamoto, T., Furuhata, T., Arai, N., and Mori, K., "Thermodynamic Analysis and Optimization of New Concept Gas Turbine Cycle: Chemical Gas Turbine System," *Proceedings of the International Symposium on Efficiency, Costs, Optimization, and Environmental Aspects of Energy Systems (ECOS2000)*, edited by G. G. Hirs and R. V. Feboruk, Enchede, The Netherlands, July 2000, pp. 447–455.