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Performances of the Chemical Gas Turbine System and Comparison with Other Gas Turbine Based Cycles*

Takahisa YAMAMOTO#, Tomohiko FURUHATA, Norio ARAI
Research Center for Advanced Energy Conversion, Nagoya University
Furo-cho, Chikusa-ku, Nagoya, 464-8603 - Japan
Tel. +81-52-789-3914, Fax +81-52-789-3910
E-mail: h991306d@mbox.media.nagoya-u.ac.jp

Koichi MORI
Toyo Tekunika Co.
Sakae, Naka-ku, Nagoya, 460-0008 - Japan

Abstract

This paper describes a novel combined cycle based on a "Chemical Gas Turbine" system. The system consists of fuel-rich and fuel-lean combustors with their gas turbines, recuperators, and a steam bottoming cycle. Important features of this system are the gas turbine with C/C composites blades and the fuel-rich combustion techniques. These techniques result in no cooling of turbine blades and much higher turbine inlet temperature, therefore, much higher thermal efficiency. This paper analyzes the energy, exergy, and heat exchanger sizes of the proposed system. Furthermore, optimizations from pressure ratio aspects are discussed. All results are compared with a simple gas turbine system and a conventional combined cycle. The following results were obtained: the chemical gas turbine system achieves a thermal efficiency of 64%, and low exergy loss in the combustion processes. In addition, characteristics of the system are similar to the simple gas turbine system.

Key words: fuel-rich combustion, exergy analysis, heat recovery steam generator, high thermal efficiency

1. Chemical Gas Turbine System

Authors have proposed a new-concept gas turbine system, the "Chemical Gas Turbine (ChGT)" system that is based on promising developments of advanced fuel-rich combustion techniques and carbon/carbon (C/C) composites materials for the turbine blades (Fushitani et al., 1997; Kato et al., 1996; Arai and Kobayashi, 1997; Lior and Arai, 1998).

A schematic diagram of the proposed system is shown in *Figure 1*. This new system consists of a fuel-rich combustor, a fuel-lean combustor, two sets of gas turbines, a steam turbine, recuperators and heat recovery steam generators (HRSGs). 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 (Yamamoto, 1997). We consider that C/C composites are the only promising materials for turbine blades operating at temperature above 1500°C without internal cooling. Such materials, however, are sensitive to high temperature oxidation. From this point of view, fuel-rich combustion, which results in significantly low concentration of oxygen in the exhaust stream, and coating techniques with the C/C composites are very well suited for this application. The second gas turbine is operated at temperature of almost 1450°C, where conventional gas turbine blade materials could be thus available.

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# Author to whom all correspondence should be addressed.

Firstly, compressed air and fuel is mixed and burned with the fuel-rich condition in the first combustor (CB-1). Exhaust gas from the first gas turbine (GT-1) still contains 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 fuel-lean in the second combustor (CB-2). Exhaust gas from the second combustor drives the second gas turbine (GT-2). Thermal energy of the GT-2 exhaust gas is used to generate the steam in the steam bottoming cycle.

2. Simulation Conditions

A summary of assumptions for the ChGT system is shown in Table.1. Isentropic efficiencies of compressors, gas turbines, steam turbine and pump are set at 91%, 92%, 90% and 93%, respectively. No cooling is assumed for the fuel-rich gas turbine cycle with the C/C composite blades. The turbine inlet temperature (TIT1), assumed as 1450^{o}C, 1600^{o}C and 1800^{o}C, is controlled by equivalence ratio at the CB-1 only. The fuel-lean gas turbine cycle uses the air-cooling technique and the turbine inlet temperature (TIT2) is assumed as 1450^{o}C. The standard pressure ratios at the GT-1 and the GT-2 are fixed at 20.

A single pressure scheme with reheat is chosen for the steam bottoming cycle. Both steam generators produce superheated steam of 530^{o}C and 14.2MPa, which is fed into a common steam turbine. The minimum temperature approach between hot and cold streams in the HRSGs is assumed as 15^{o}C. The following ambient conditions are considered: air temperature of 25^{o}C and atmospheric pressure of 0.1013MPa; relative humidity is not taken into account. It is assumed that the fuel (methane with a lower heating value of 50.01kJ/kg) is supplied at the required pressure from the gas pipeline, and the amount of input fuel is kept constant at 383kg/h (19MJ/h, based on LHV).

The conventional combined cycle (CCC) and the simple gas turbine cycle (SGT), with which the ChGT is compared, are described in *Figure 2* and *Figure 3* respectively.

TABLE I. MAIN ASSUMPTIONS OF THE CHGT SYSTEM AND THE OTHER GAS TURBINE BASED SYSTEMS

Gas turbine cycle		**Chemical gas turbine system**	**Simple Gas Turbine System**	**Conventional combined cycle**
Fuel rich stage				
Turbine efficiency	[%]	92.0	-	-
Compressor efficiency	[%]	91.0	-	-
Pressure ratio	[-]	20.0	-	-
Turbine inlet temperature	[^{o}C]	1450, 1600, 1800	-	-
Fuel lean stage				
Turbine efficiency	[%]	92.0	92.0	92.0
Compressor efficiency	[%]	91.0	91.0	91.0
Pressure ratio	[-]	20.0	20.0	20.0
Turbine inlet temperature	[^{o}C]	1450	1450	1450
Steam turbine cycle				
Steam turbine efficiency	[%]	90.0	-	90.0
Pump efficiency	[%]	92.0	-	92.0
Condenser pressure	[MPa]	0.05	-	0.05
HRSG				
Pressure drop	[%]	3.0	-	3.0
Steam pressure	[MPa]	14.0	-	14.0

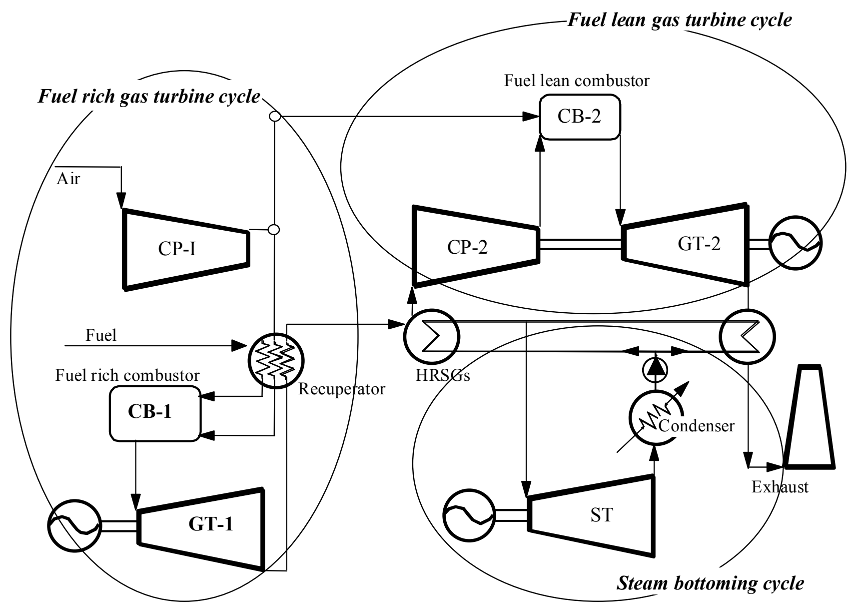

Figure 1 Flow diagram of combined cycle based on the chemical gas turbine system

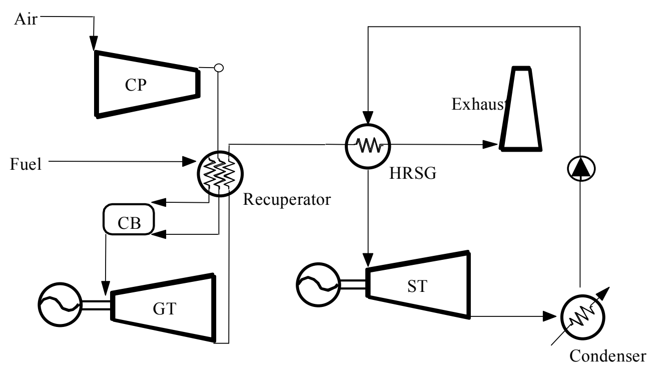

Figure 2. Flow diagram of conventional combined cycle

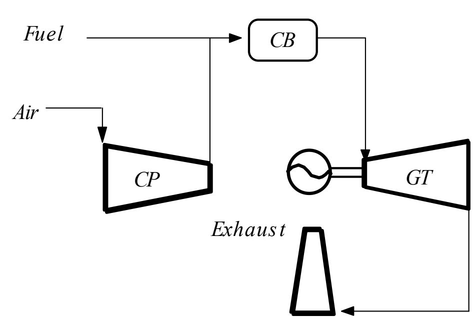

Figure 3. Flow diagram of simple gas turbine cycle

3. Exergy Analysis of the Chemical Gas Turbine System and Other Gas Turbine Based Cycles

In this study the energy and exergy analysis of the CCC and ChGT systems at full (100%) fuel input has been carried out (Waldyr, 1997; Oh et al., 1996; Kehohofer, 1991; Kozu and Tsuruno, 1994).

Figure 4 presents the results of the energy analysis as a relationship between turbine inlet temperature and thermal efficiency. The maximum efficiency of the CCC system is 61.9% (LHV base) and of SGC system is 42.3% for TIT=1450^{o}C. On the other hand, the thermal efficiency of the ChGT system is 62.2% for the same turbine inlet temperature (TIT1=1450^{o}C). The system TIT1 can be raised using the C/C composites blades, and maximum efficiency will reach 65%, when the TIT1 is 1800^{o}C.

Figure 5 shows results of the exergy analysis of the SGC, CCC, and ChGT systems. Generally speaking, the exergy loss in the combustion process is dominant in gas turbine plants. For the proposed system, the net power of the gas turbine cycle increases and that of the steam cycle decreases as the TIT1 rises. The CCC and the ChGT systems are compared with each other; the exergy loss of the ChGT system with two stage combustion process is suppressed by 1 to 3%. Thusly, the net power of the ChGT cycle is larger than the conventional combined cycle by 1 to 2%. On the other hand, the power of the steam cycle is smaller than that of the conventional combined cycle by 1%, approximately.

The reason of reduction of the exergy loss in the combustion process of ChGT cycle is explained by *Figure 6*. It shows a typical relationship between the exergy loss and the equivalence ratio in the combustion process for inlet temperature 25^{o}C and atmospheric pressure of fuel and air. The combustion exergy loss is expressed as a percentage of its value for equivalence ratio (ER) 1.0 (stoichiometric condition). It shows an important characteristics that the combustion exergy loss decreases as the equivalence ratio rises. Therefore, the combustion exergy loss in the ChGT system is reduced.

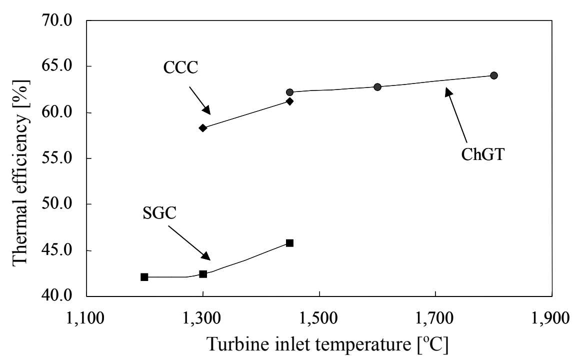

Figure 4. Comparison between thermal efficiency of the ChGT system and that of the other gas turbine based systems

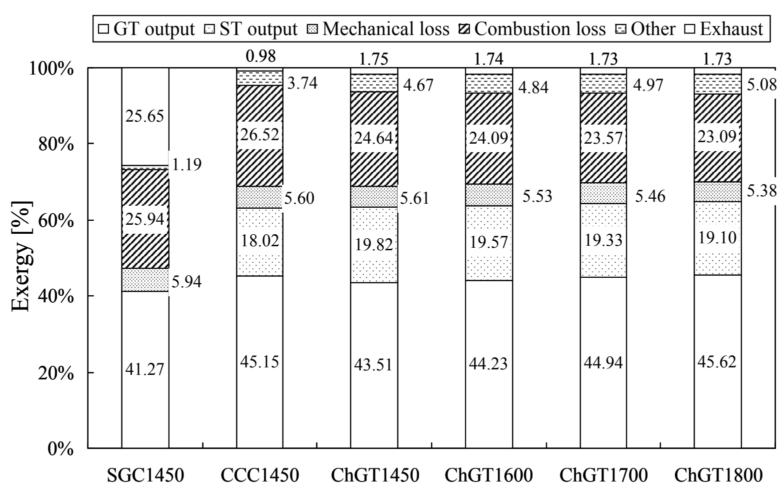

Figure 5. Exergy analysis of the ChGT system and the other gas turbine based systems

4. Efficiency and Specific Power of the Chemical Gas Turbine System

This study discusses the performance of the ChGT system and compares it with that of the SGC and the CCC systems in order to optimize (Waldyr, 1997).

Figure 7 presents the performances of the SGC system. The best efficiency is TIT=1450°C and a pressure ratio 60, and shifts to higher pressure ratio as TIT rises. For the same TIT, the specific power peaks at a pressure ratio between 20 and 30. This cycle will be the base case for comparisons with all other analyzed systems.

Figure 8 presents the performances of the CCC system. The rising TIT increases the thermal efficiency and the specific power for a given pressure ratio. Also both the specific power and the thermal efficiency increase as the pressure ratio rises. The best performance is at a TIT=1450°C and a pressure ratio 30.

Figure 9 presents the performances of the ChGT system. Both the specific power and the efficiency are the highest compared with other analyzed gas turbine cycles. The best efficiency is at TIT1=1800°C and pressure ratio 30. For the same TIT1, the specific power peaks at a pressure ratio between 15 and 20. Maximum specific power of ChGT is same as that of the SGC system.

The results from the above analysis have shown that tendencies of the CCC system characteristics are different from the other two gas turbine systems. The reason for this is not hard to understand. It is explained as follows: The compressor work increases as pressure ratio rises. Then the compressor performance affects the total cycle performance significantly. Since there is no heat recovery in the SGC system and there are two compressors in the ChGT system, the compressor plays an important role in both systems. Therefore the ChGT system shows the same tendency as that of SGC system.

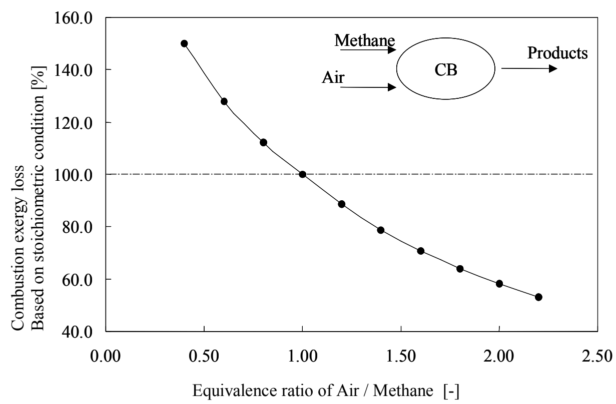

Figure 6. Typical relationship between the exergy loss and the equivalence ratio in methane/air combustion

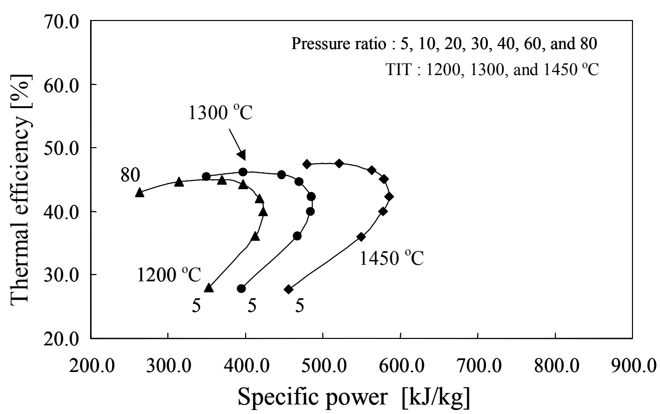

Figure 7. Performances of the SGC system

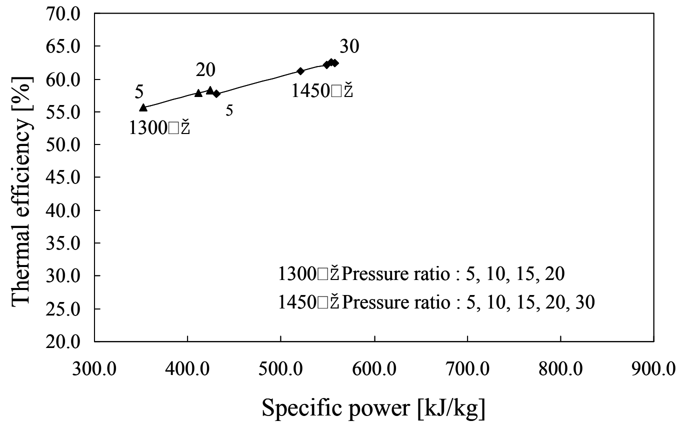

Figure 8. Performance of the CCC system

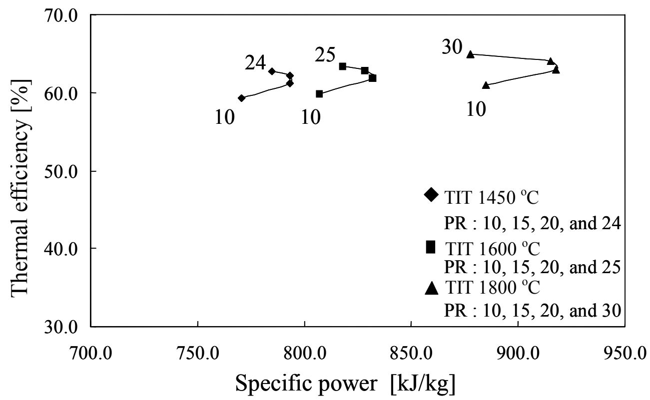

Figure 9. Performance of the ChGT system

TABLE II. THE REQUIRED (UA) OF THE HRSGS

		(UA) of HRSGs [MJ/K-h]	Percentages base on the CCC [%]
CCC system		202.30	100.00
ChGT system			
TIT1	1450°C	154.45	76.35
	1600°C	152.95	75.61
	1800°C	149.78	74.04

5. HRSGs Size Consideration

HRSGs are located at bottoms of the fuel-rich and the fuel-lean stages in the ChGT system. Their sizes are the largest in the gas turbine system because they have high-pressure steam (over 10MPa) and high temperature gas (about 1000°C). The total size of the HRSGs thus is an important factor economically in power plant construction. In this analysis also the total size of the HRSGs as a product of overall heat transfer coefficient 'U' and surface area available for heat transfer 'A' has been considered.

The results of the analysis on HRSGs size of ChGT and CCC are shown in Table.2. The former are seen to be lower than the latter by around 25%, with the size slightly decreasing as TIT1 increases. The explanation for this reduced area requirement is that (1) due to the higher efficiency of the ChGT system, 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 power is produced in the steam cycle. The reduced size of the HRSG needed for the ChGT system is yet another advantage of this system relative to CCC system.

6. Conclusion

This paper proposes a "chemical gas turbine system" which is a novel gas turbine/steam turbine combined cycle system. Important features of this system are the combination of chemical combustion (fuel-rich) and the fuel-lean combustion techniques and high turbine inlet temperature. Turbine blades with C/C composites which are sensitive to high temperature oxidation can resist high temperature because of significant reduction of oxygen concentration by the chemical combustion.

The thermodynamics energy analysis of this ChGT system shows that thermal efficiency based on LHV can be as high as 64%. This value is higher than that of the CCC system by 2-4%.

As for exergy analysis, the exergy loss of the combustion process is dominant in each gas turbine systems. However, in the ChGT system, the combustion exergy loss is less than in the CCC system by 2-4%. This reduction is explained by the typical relationship between the exergy loss and the equivalence ratio in the methane/air combustion.

From the performances of the gas turbine systems considered in this study it is found that the ChGT system has the same tendency as the characteristics of the SGC system because the compressor performance is dominant in both systems.

This study also analyzed the product of overall heat transfer coefficient and surface area available for heat transfer 'UA' as the required size of HRSGs. As a result, the required size of HRSG of the ChGT system was found to be 25% less than that of the CCC system, and it decreases as TIT1 rises. Both results are desirable for the ChGT system economics.

These results lead to the conclusion that the proposed combined cycle based on the chemical gas turbine system has higher performance than the other gas turbine based systems. However, the proposed system uses fuel-rich combustion techniques and turbine blades with the C/C composites which are not common. These need further discussion and research.

Acknowledgement

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Nomenclature

A	surface area [m^2]
TIT	turbine inlet temperature [^oC]
TIT1	inlet temperature of the 1st turbine in the ChGT system [^oC]
TIT2	turbine inlet temperature of 2nd turbine in the ChGT system [^oC]
U	overall heat transfer coefficient [W/m^2K]

Abbreviations

CB	the combustor of the conventional combined cycle
CB-1	1st combustor
CB-2	2nd combustor
CCC	the conventional combined cycle
ChGT	the chemical gas turbine system
CP	the compressor of the conventional combined cycle
CP-1	1st compressor
CP-2	2nd compressor
GT	the gas turbine of the conventional combined cycle
GT-1	1st gas turbine with the C/C composites blades
GT-2	2nd gas turbine
HRSG	heat recovery steam generator
SGC	the simple gas turbine cycle
ST	steam turbine

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