EXPERIMENTAL EVALUATION ON LOW-HEATING VALUE FUEL ACCEPTABILITY OF MICRO GAS TURBINE SYSTEM OPERATION

Daisuke Yamashita*, Keisuke Kuwabara*,
Kazuya Tatsumi* and Kazuyoshi Nakabe*
*Dept. of Mech. Eng., Osaka Prefecture University
Corresponding author: *e-mail nakabe@me.osakafu-u.ac.jp,*
phone +81-72-254-9224, fax +81-72-254-9904

Keywords: *Micro gas turbine, Low-heating value fuel, System analysis, Thermal efficiency, Exhaust gas analysis*

Abstract

A micro gas turbine (MGT) system composed of two automobile turbochargers was operated using low-heating value fuels to examine to what extent the system can accept such unfavorable fuels. Gasified biomass and the vitiated gas of the exhausts from fuel cell or fuel reformer are possible examples of low-heating value fuels as the fuel alternatives for MGT. In the present experimental study, the heating value of the fuel was simulated by mixing nitrogen into liquefied petroleum gas (LPG). The turbine of the first turbocharger played a role as the gas generator turbine of the system, while the one of the second turbocharger as the power generator turbine. The load of the MGT system was changed using the outlet valve opening of the second turbocharger's compressor. The system thermal efficiency was decreased almost linearly with an increase in the mixing ratio of N_2 to LPG. A drastic change in the concentration of CO emission was obtained at the mixing ratio of around 0.4. NO_X concentration was contrary decreased with the mixing ratio. This trend could be related to the decrease of flame temperature in the combustion chamber, corresponding to the sensible heat of the mixed N_2. The present MGT system, however, was eventually successful in operating under each tested condition, where the minimum heating-value of the simulated fuel was approximately 0.43 of pure LPG.

1 Introduction

Compact heat engines of less than several-tens kW output can be a key device as well as fuel cells (FCs) for the development of distributed power supply systems [1]. A micro gas turbine (MGT), in particular, is one of the most promising heat engines for these systems since it has the advantages of relatively low NO_X emission, low mechanical vibration, small number of mechanical parts and easy maintenance, compared to reciprocating engines. If various types of fuel can be used for MGT operation, MGT may become more widely used for distributed electric power supply system and also make a contribution to the reduction of fossil fuel consumption. One of the possible reasons is that, for example, in local farms or mountain-girt areas, renewable energy resources of biomass like excrement of domestic animals or woody parts of tree-trimming can be used to yield carbon-neutral methane-based fuels. Such fuels are mostly low-heating value fuels, but they will have a potential to be alternate fuels for MGT. In such local areas, renewable energy from wind and/or solar power stations also can be utilized for directly generating electricity and reforming the biomass to hydrogen. If MGT system is linked with the concurrent use of the power stations, the fluctuation of the generated electric power from the renewable energy

depending on weather conditions is greatly improved [2].

FC is also well-known as a suitable and promising distributed electric power supply device, but low-temperature types of FC like PEFC (Polymer Electrolyte FC), PAFC (Phosphoric Acid FC) and DMFC (Direct Methanol FC) inevitably need catalysts including rare metals, like Platinum, for electrode reactions. The other type of FC like MCFC (Molten Carbonate FC) or SOFC (Solid Oxide FC) have a merit that the electrode reactions occur without Platinum catalyst, however, needs relatively high temperature atmosphere, 600 ~ 900ºC. For the latter-type of FC, an MGT can play an important role to keep a sufficient high temperature essential for the electrode reaction of the FC and to compose a so-called hybrid co-generation system, which is the most promising method to increase the overall thermal efficiency of the system [3, 4]. In this case, a part of the exhaust gas from FC can be effectively recycled and fed into MGT as a part of the fuels.

Following the above-mentioned aspects of various fuel sources for MGT operation, a wide acceptability of fuel variation is essential for MGT performance [5]. Large-scale gas turbine systems operated at central electric power stations require mostly LNG as their fuel. For MGT systems, on the other hand, higher acceptability of the fuel is a technological matter of the utmost concern to free ourselves from electric power supply systems based on fossil fuels. Thus, it is very important to examine to what extent an MGT system is applicable to various types of fuels, in particular, low-heating value fuels. The heating values of the gasified biomass or FC exhaust are less than 1/4 ~ 1/10 of that of pure LNG or LPG. In the present study, the acceptability of MGT for such low-heating value fuels is investigated by operating a laboratory-based MGT system using LPG diluted with nitrogen down to around a half of LPG's heating value.

2 Experimental Apparatus, Methods and Conditions
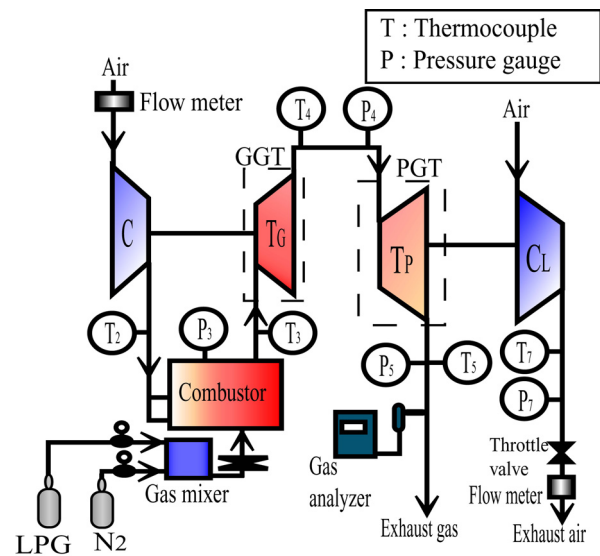

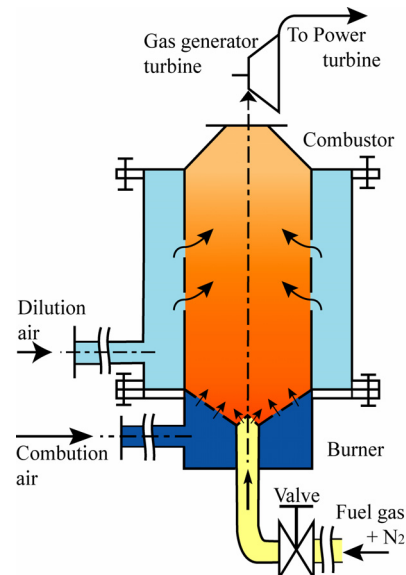Fig. 1   MGT system flowchart.

Fig. 2   MGT Combustor.

Figure 1 shows the system flowchart of the MGT system. The gas turbine system is a two-axial turbine system consisted of two conventional automobile turbochargers, where the first one plays a role as a gas compressor and a gas-generating turbine (referred to as GGT), and the other as a power-generating turbine (PGT) and a loading compressor (LC) [5, 6]. LC, which is a gas compressor of the second turbocharger shaft-coupled to PGT, was used for the load output instead of a shaft generator of electric power. The load was controlled by adjusting an opening amount of a butterfly valve mounted on the downstream side of LC in the

present study. The rotational frequencies of GGT and PGT were monitored using revolution indicators. Gas temperature and pressure were measured using sheath-type thermocouples and pressure gauges, respectively, in each monitoring location indicated in Fig. 1. Also, the concentrations of NO_X, CO, CO_2 and O_2 emissions in the exhaust gas from the MGT system were measured using a gas analyzer.

The low-heating value fuel was simulated by diluting LPG with N_2 using a gas mixer, and was then supplied to a combustion chamber without halting the operation of the MGT system. The combustor was shown in Fig. 2 which was equipped with a cone-type non-premixed burner. Primary combustion air was supplied just above the fuel nozzle tip to be mixed well with the fuel. Secondary dilution air was discharged into the second half of the combustor in order to suppress the generation of NO_X. The mass flow rate of the working gases, the mixing ratio of N_2 to LPG (R_N) and the load of LC were suitably adjusted to keep a stable self-running of the MGT system. The experiment was carried out firstly under the condition of feeding pure LPG to the combustor. The amount of LC valve opening (D_G) was set at three different levels, i.e. D_G = 100%, 66% and 50%. For each case, the GGT inlet temperature (TIT; T_3) was varied from 650ºC to 800ºC by controlling the mass flow rate of the feeding LPG.

For the case of N_2-diluted low-heating value fuel, the amount of the valve opening was fixed at D_G = 100%. In this case, the MGT system was started up by feeding pure LPG and was operated until T_3 reached a certain temperature (referred to as 'Starting temperature'), i.e. 700ºC, 750ºC or 800ºC. Then, N_2 was discharged and mixed with LPG in the gas mixer, adjusting R_N in the range of 0 ~ 1.27. Thus, the heating values of the fuels simulated using LPG-N_2 mixture were correspondingly changed in the range of 100% ~ 43% of pure LPG.

The thermal efficiency of the MGT system together with adiabatic efficiencies of compressor and turbine of each turbocharger was estimated thermodynamically by assuming a typical and simple Brayton cycle. Also, a virtual recuperator-type regenerative cycle applied to the present MGT system was numerically analyzed in order to compare with other commercially-available regenerative MGT systems or reciprocating engines, since the present MGT has no heat exchangers as a component of regenerative cycle. The flowchart of the analyzed regenerative gas turbine cycle is schematically illustrated in Fig. 3.
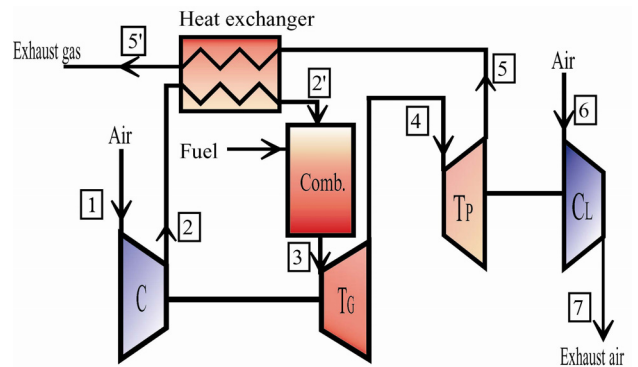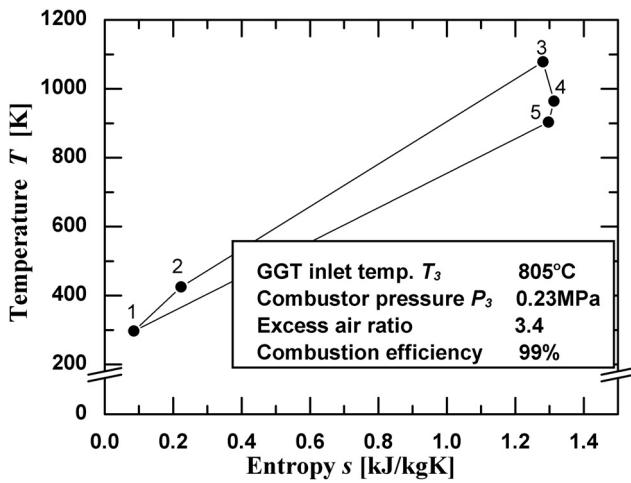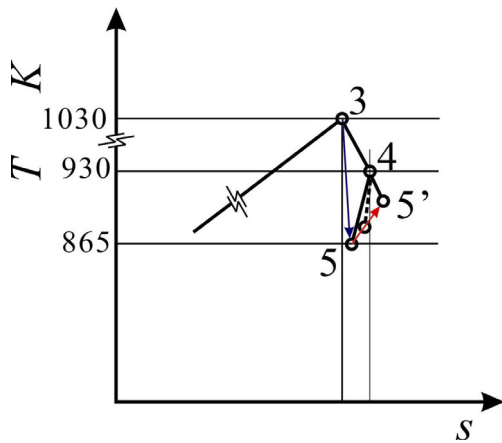

Fig. 3 Flowchart of MGT system with heat exchangers.

3 Results and Discussion

3.1 Performance of MGT using LPG only

The overall performance of the present MGT system is first discussed in this section under the operation of pure LPG fuel condition. Figure 4 (a) shows a T-s indicated diagram of the system. The index numbers, 1 ~ 5, in Figs. 4 (a) and (b) denote the thermodynamic states. In Fig. 4 (a), the process 1 → 2 represents the non-isentropic compression process of the gas compressor, which should be ideally an isentropic process of Brayton cycle. The process 2 → 3 represents constant pressure process at the combustor. The processes 3 → 4 and 4 → 5 represent the expansion processes at two-axial turbines, GGT and PGT, both of which were also non-isentropic. At the state 3, the pressure in the combustor and the turbine inlet temperature of GGT were 0.23MPa and 805ºC, respectively, in this operation. At the state 5, the working fluid was discharged to the surroundings with the ambient pressure, $P_5 = P_1$.

(a) *T-s* indicated diagram

GGT inlet temp. T_3	805ºC
Combustor pressure P_3	0.23MPa
Excess air ratio	3.4
Combustion efficiency	99%

(b) Expansion process of GGT ~ PGT.

Fig. 4 Brayton cycle for MGT system analysis.

The expansion processes 3 → 4 and 4 → 5 from the combustor to PGT through GGT are shown in Fig. 4 (b) in detail. In case that the distance from two turbochargers are considerably large and the exhaust manifolds of both turbochargers are exposed to the cool surroundings, the temperature of the working fluid in the system is remarkably decreased. Then, the process 3 → 4 → 5 becomes apparently isentropic, that is, unrealistically adiabatic with the constant ambient pressure of the state 5. If the system is sufficiently thermally-insulated, it can be predicted that the state 5 reaches the state 5' along the isobaric line, 1 - 5. The dashed line in the figure indicates one of the corrected data with taking a countermeasure for the heat leakage from the exhaust pipes.
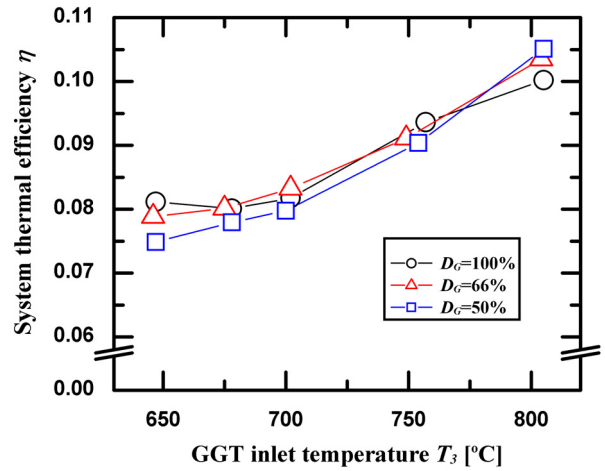

Fig. 5 System thermal efficiency vs. TIT (with LPG only).

Figure 5 shows the relation between the system thermal efficiency η and TIT (= T_3) for each valve-opening amount D_G. η was defined as the ratio of the PGT work output to the heat release quantity of the supplied fuel. Higher efficiency was obtained almost linearly as TIT was increased, rarely dependent on the load. The thermal efficiency of the present MGT system was 7.5% ~ 10.5% which is relatively small in comparison with other commercially-available MGTs since the present system was constructed as a simple Brayton cycle without installing any regenerative heat exchangers in the system itself. The output of the system was found to be ca. 10kW at the maximum.

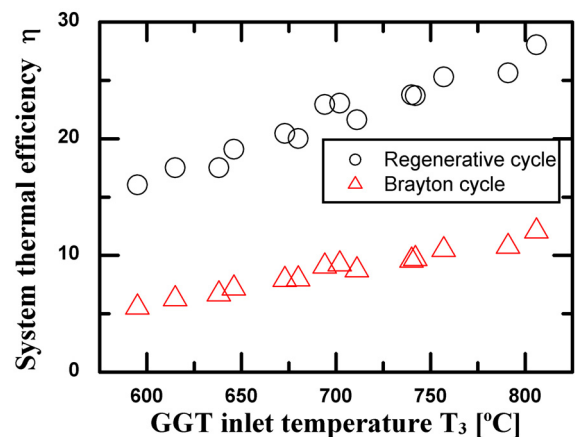The effect of regenerative cycle on the

Fig. 6 System thermal efficiency of MGT with regenerative heat exchangers.

thermal efficiency was numerically estimated using HYSYS (AspenTech Japan Co., Ltd.). A recuperator-type heat exchanger was installed between hot gas flows at PGT exhaust and cold gas flows at the gas compressor outlet, shown in Fig. 3 of the previous chapter. The heat exchange temperature effectiveness was assumed to be 80% with no pressure loss between the inlet and outlet. The obtained results of the thermal efficiency are shown in Fig. 6. Open symbols of triangle and circle, respectively, indicate the simple Brayton cycle and the regenerative one. In comparison with the former cycle, the latter has an almost twofold thermal efficiency, which was predicted to be 28% at the maximum, similar to the published data of other MGT makers.

Figures 7 and 8 show respectively CO and NO_X concentrations in the exhaust gas. CO emission was decreased with an increase in TIT,
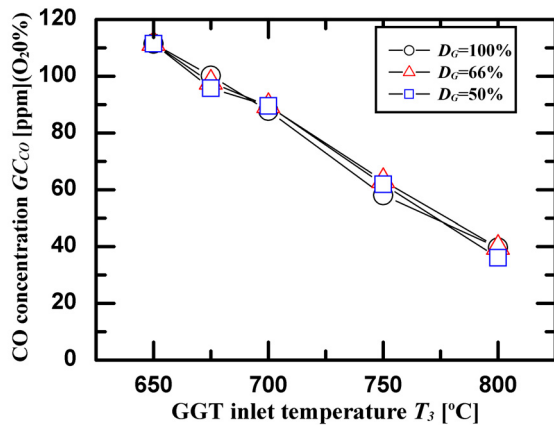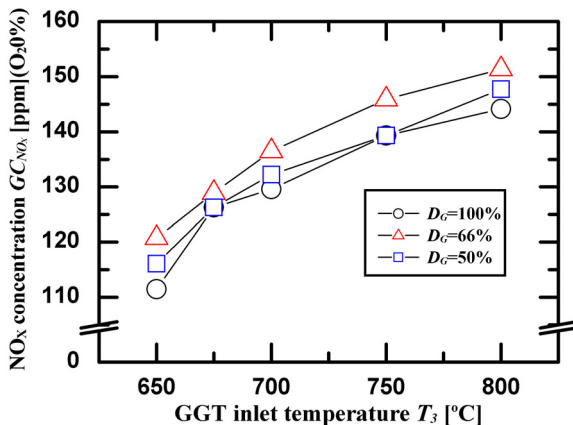
Fig. 7　CO concentrations vs. TIT.

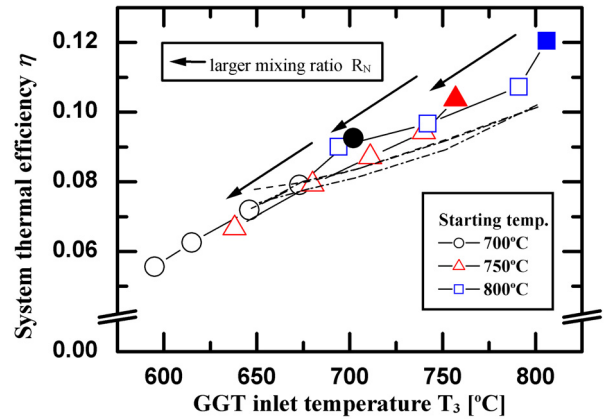Fig. 8　NO_X concentrations vs. TIT.

Fig. 9　System thermal efficiency vs. TIT (with LPG-N_2 mixture).

while NO_X emission was gradually increased. Both trends were not strongly dependent on the load.

3.2 Performance of MGT using LPG-N_2 mixture

Figure 9 shows the relationship between η and T_3 for the case of LPG-N_2 mixture. As mentioned in Chap. 2, the experiments using the low-heating value fuels were conducted after achieving the stable running at the fixed turbine inlet temperature of the GGT (referred to as Starting temperature) with the pure LPG feeding. The solid symbols in Fig. 9 indicate the points as the starting temperatures, from which each measurement with LPG-N_2 mixture was started. The arrow drawn in the same figure indicates the trajectory of the increase of the LPG-N_2 mixing ratio. When the mass flow rate of the supplied gaseous fuel (LPG+N_2) was fixed, TIT was decreased with an increase in the mixing ratio, R_N, which accompanies the decreases of the combustion pressure P_3 and also turbine revolution numbers. These eventually resulted in the deterioration of the overall thermal efficiency of the system.

Figures 10 and 11 show the concentration profiles of CO and NO_X in the exhaust gas emitted from PGT, respectively, against TIT. The solid symbols in these figures indicate the above-mentioned starting temperatures. As the mixing ratio, R_N, was increased, the TIT was correspondingly decreased. CO concentration
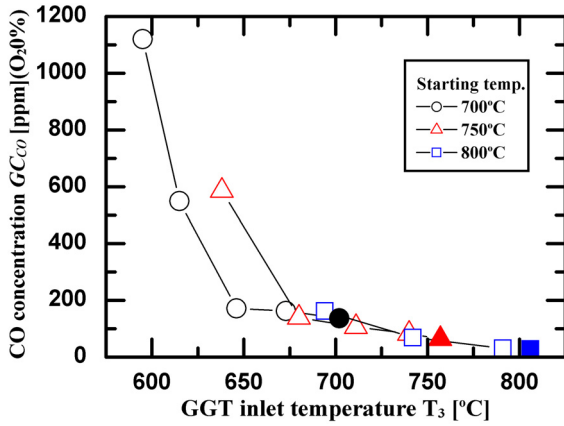
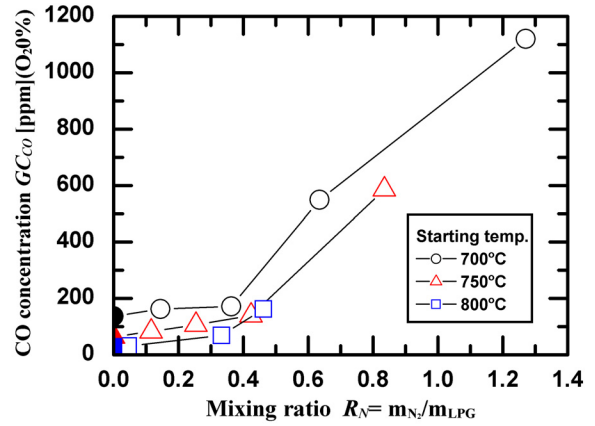Fig. 10 CO concentrations vs. TIT (with LPG-N_2 mixture).
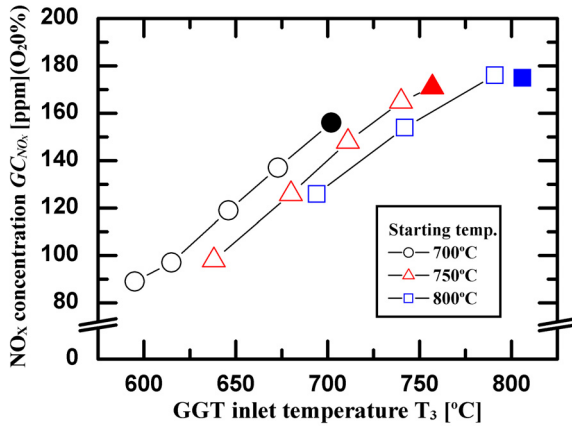

Fig. 12 CO concentrations vs. R_N (with LPG-N_2 mixture).

Fig. 11 NO_X concentrations vs. TIT (with LPG-N_2 mixture).
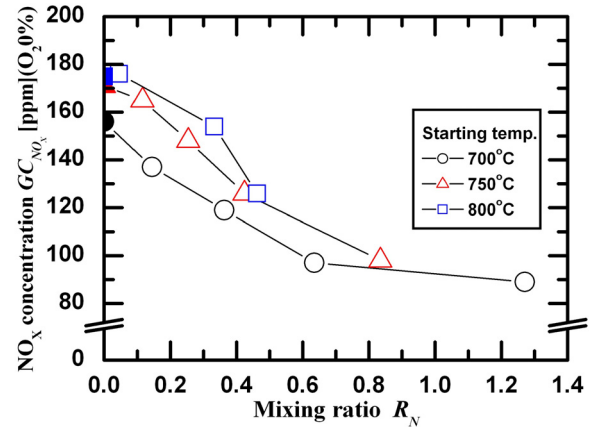

Fig. 13 NO_X concentrations vs. R_N (with LPG-N_2 mixture).

was correspondingly increased, while NO_X concentration was gradually decreased. The both concentration profiles were re-plotted against the LPG-N_2 mixing ratio, R_N, as abscissa, shown in Figs. 12 and 13.

As is clearly shown in Fig. 12, CO concentration suddenly jumps up when the LPG-N_2 mixing ratio exceeds 0.4 which indicates an incomplete combustion. Contrary to the CO concentration profile, NO_X concentration was gradually decreased with an increase in R_N, as clearly shown in Fig. 13.

These trends can be explained with the following reason. The increase of LPG-N_2 mixing ratio inevitably reduces the combustion heat release under the condition of constant mixture flow rate and requires larger sensible heat for temperature rise of the mixture including the inert component, N_2. Thus, the gas

temperature in the combustor should be decreased as R_N is increased. For example, under the condition of LPG-N_2 mixing ratio larger than 0.4 and the gas temperature of 1000K, the required heat for rising N_2 temperature from 300K to 1000K is more than 10% of the supplied LPG heating value. Consequently, this produces the decrease of combustion pressure and the suppression of local reactions. These trends should accompany the increase of CO emission and the reduction of thermal NO_X generation.

Nevertheless, the present MGT system was eventually successful in operating under each tested condition and durable even for using the simulated fuel of about a half heating value of pure LPG without any modification of the combustor design.

4 Conclusions

A laboratory-based MGT was operated in the present study with the experimental conditions using low-heating value fuels which were simulated in terms of the dilution of LPG with N_2. Efficiencies at each system component were calculated from the measured temperature and pressure. Also, gas components of the MGT exhaust were analyzed. The obtained results are summarized as follows:

- The experimentally-obtained overall thermal efficiency of the present MGT system was 7.5 ~ 10.5%. The validity of regenerative heat exchanger was numerically predicted to improve the thermal efficiency almost twofold.
- Under the condition of a fixed mass flow rate of the supplied fuel, TIT was decreased with an increase in the LPG-N_2 mixing ratio, which results in the deterioration of the overall system efficiency.
- The mixing of N_2 significantly affected the characteristic of the gas emission. As the LPG-N_2 mixing ratio was increased, NO_X emission was decreased while CO was increased. A remarkable increase of CO was observed when the mixing ratio exceeded 0.4, which is related to the incomplete combustion occurring at the combustor.
- The present MGT system was successful in operating at the fuel condition of heating value less than 0.5 of LPG, which proved the MGT acceptability to such low-heating value fuels without any modification of the combustor.

Acknowledgement

We'd like to express our gratitude to Prof. Y. Tsujimoto of Osaka University and Prof. K. Teshima of Okayama Science University for their helpful advices.

References

[1] Layne, A., Samuelsen, S., Williams, M. and Hoffman, P., Hybrid Heat Engines: The Power Generation Systems of the Future. *Proc. ASME Turbo Expo 2000*, 2000-GT-0549, pp. 1-6, 2000.

[2] Mie Prefecture, *Report on Research & Development of Combined Electric Power Generation Systems Using Renewable Energy*, JST report, 2003.

[3] Massardo, A. F., McDonald, C. F. and Korakianitis, T., Microturbine/Fuel-Cell Coupling for High-Efficiency Electrical-Power Generation. *Proc. ASME Turbo Expo 2000*, 2000-GT-0175, pp. 1-8, 2000.

[4] Buhre, B. J. and Andries, J., Biomass-based, Small-scale, Distributed Generation of Electricity and Heat Using Integrated Gas Turbine - Fuel Cell Systems. *ibid.*, 2000-GT-0022, pp. 1-7, 2000.

[5] Yamashita, D., Tatsumi, K. and Nakabe, K., Performance Evaluation of Micro Gas Turbine Operated Using Low-Heating Value Fuel, *Proc. 1st International Forum on Heat Transfer*, pp. 87-88, 2004.

[6] Mori, H., Tsujimoto, Y. and Yokota, K., Development of Turbo-Burner System and its Use in Education, *Proc. ISROMAC-8*, Vol. 1, pp. 364-370, 2000.