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PERFORMANCE EVALUATION OF MICRO GAS TURIBINE OPERATED USING LOW HEATING VALUE FUEL

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ABSTRACT

A practical micro gas turbine (MGT) has been operated under low heating value fuel conditions. The low heating value fuel was simulated by diluting the LPG by N_2 in the present study. Efficiencies at each component were calculated using the measured temperature and pressure. Exhaust gas analysis was also conducted. Under the condition of a fixed mass flow rate of the supplied fuel, the turbine inlet temperature (TIT) of the MGT decreases as the N_2-LPG mixing ratio increases, resulting in a deterioration of the overall system efficiency. This N_2-dilution also significantly affected the characteristic of the gas emission. These are believed to be related to the incomplete combustion occurring at the combustor. Nevertheless, the MGT was eventually successful in operating at the condition of heating value less than 1/2 of the LPG maintaining a stable combustion, demonstrating its applicability to such low heating value fuels.

1. INTRODUCTION

MGT is well known as one of the key technologies which frame the distributed power supply system. It is compact, provides less NO_x and CO_x emissions and is easy for maintenance compared to the reciprocated engines. On the other hand, considerable problems still exist. One of these is the applicability for various fuel sources. To address the global energy problems and the variety of the local area energy supplies, it is important to examine to what extent the current MGT system is applicable to various types of fuels. Especially, applicability for low heating value fuels is a matter of concern. For example, the heating values of the exhaust gas from the fuel cells or the gasified biomass are less than 1/4~1/10 of that of the pure propane. In the present study, the aforementioned applicability of the MGT for such fuels is investigated by operating a practical MGT system using low heating value fuels.

2. EXPERIMENTAL METHODS

2.1 Apparatus

Figure 1 shows the system flowchart of the MGT system. The gas turbine system corresponds to a two-axial Brayton cycle consisted of two commercial turbo-chargers where the first one operates as a gas-generating turbine (GGT) and the other as a power-generating turbine (PGT) (Mori, 2000). The low heating value fuel is simulated by diluting LPG with N_2 using the gas mixer and is then supplied to the combustor. In the present study, instead of a dynamo, the compressor of the second turbocharger plays a role of actual load. The load is controlled by changing the valve opening of the throttle valve mounted at the downstream side of the compressor. Temperature and pressure measurements are established by installing thermocouples and pressure gauges in each location shown in Fig. 1. CO and NOx in the exhaust gas from the PGT are measured by using an electrochemical sensor.

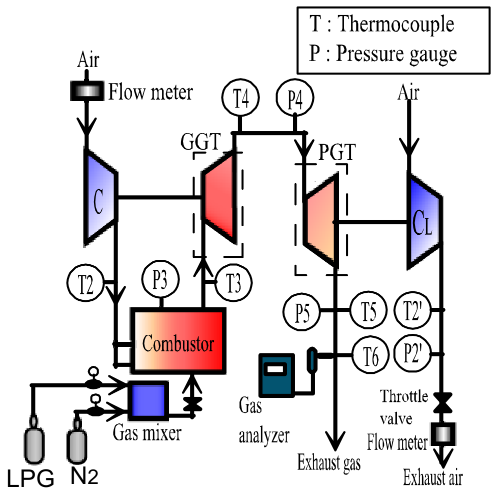

Fig. 1 System flowchart.

2.2 Conditions

Firstly, experiment was carried out under the condition of applying 100% LPG fuel to the system. The valve opening was set at three different levels, i.e. 100, 66 and 50%, and for each case, the TIT (=T_3) was varied from 650 to 800ºC by controlling the fuel supply rate.

For the diluted fuel case, the valve opening was fixed at 100%. In this case, the MGT was first started up by using 100% LPG fuel and was operated until T_3 reached a certain value (referred to as 'starting temperature'), i.e. 700, 750 and 800ºC. Then N_2 was injected and mixed with LPG, having the mixing ratio changed by several steps.

3. RESULTS AND DISCUSSION

The characteristics of the MGT operated under 100% LPG fuel condition are first discussed here. Figure 2 shows the relation between the system thermal efficiency η and T_3 for each valve-opening rate. η is defined by the ratio of the work output of the PGT to the heat supplied from the fuel. The values shown here are the averaged values of several measurements. Higher efficiency is obtained as the TIT increases and the load effect appears to be small. The thermal efficiency of the present system is 7~10% and is relatively small since no recuperator is installed for heat recovery.

In Fig. 3, the relation between η and T_3 is shown for the diluted fuel case. As mentioned in Section 2.2, experiment is conducted first starting with 100% LPG condition and then increasing the mixing ratio of N_2. The filled symbols in Fig. 3 indicate the point from which each measurement is started (starting temperature) and the arrows show the history when N_2 mixing ratio is increased. Figures 4 and 5 illustrate the concentration profiles of the CO and NO_x in the exhaust gas emitted from the PGT in relation to the N_2-LPG mixing ratio, respectively. In Fig. 3 we can see that increase of N_2 incurs a decrease of TIT which eventually accompanies deterioration of the system efficiency. It should be marked here that combustion pressure P_3 and GGT revolution number decreased also when N_2-LPG mixing ratio increased.

As is shown in Fig. 4, when the N_2-LPG mixing ratio exceeds 0.4, CO concentration increases rapidly, which indicates that a poor combustion is occurring. On the other hand, in Fig. 5, NO_x concentration decreases as N_2-LPG mixing ratio increases. The reason for this is considered as follows. As the mixing quantity of N_2 enlarges, larger sensible heat for temperature rise of N_2 is required, hence, the total combustion temperature decreases. For example, under the condition of N_2-LPG 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 combustion pressure decrease and disturbs the local reaction, which accompanies the increase of CO, and reduces the generation of the thermal NO_x.

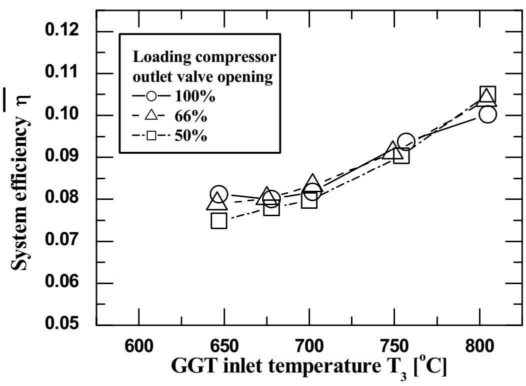

Fig. 2 System thermal efficiency (LPG 100% case).

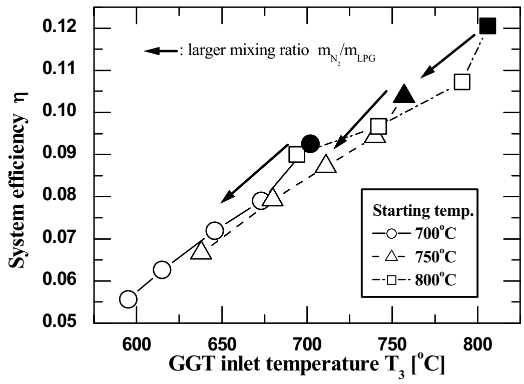

Fig. 3 System thermal efficiency (N_2 mixed case, valve opening 100%).

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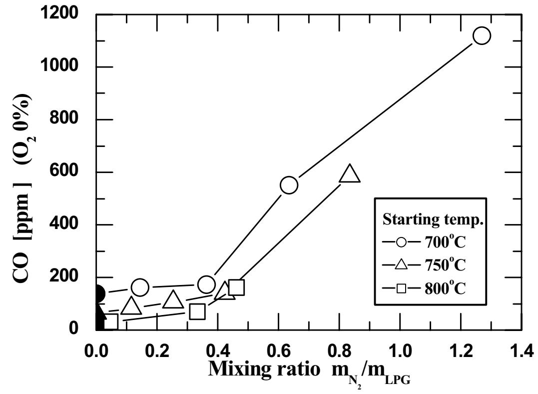

Fig. 4 CO concentrations.

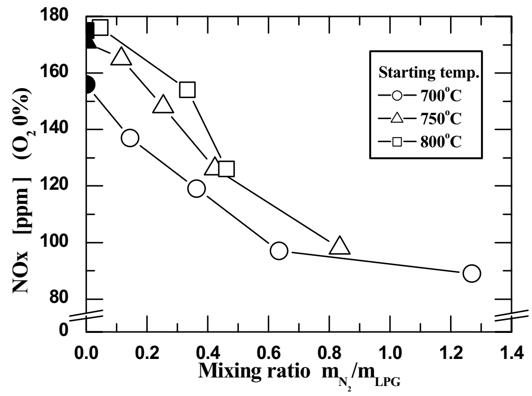

Fig. 5 NOx concentrations.

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