COMBUSTION CHARACTERISTICS OF A PROPANE-FUELED COMBUSTOR FOR A SEVERAL HUNDRED W-CLASS MICRO GAS TURBINE

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Abstract: An annular-type propane-fueled micro combustor was developed for a several hundred W-class micro gas turbine. A lean-premixed combustion method was applied to realize low NOx emission. To prevent flash back when using premixture, the fuel was injected directly into the upstream region of the combustion chamber where the premixture formed. An orifice was located to separate the upstream region and main burning region of the combustion chamber for promoting the fuel/air mixing and the flame stability. The flame stability limit showed enough wide operation regions including design point. The CO concentrations were less than 70 ppm at the total equivalence ratios over 0.42. The NOx emission was around 20 ppm(@16%O_2) at the design point. The combustor attained the high space heating rate of 800 MW/(m^3・MPa) and the high combustion efficiency more than 99.5 %.

Key words: Micro Combustor, Micro Gas Turbine, Lean-Premixed Combustion, Propane Fuel

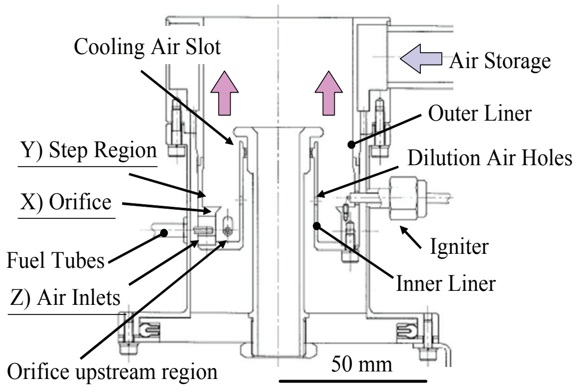
1. INTRODUCTION

Portable electronic devices or autonomous robots utilized to the human life support or rescue in disaster demand a power source satisfying a small size, light weight and long duration operation. To this demand batteries such as Li-ion and currently developed fuel cells have the weakness in weight and duration operability. In contrast, gas turbines are the power source satisfying both high power density and high energy density. Hence a several hundred W-class micro gas turbine is expected to be the demanded power source [1]. Also, micro gas turbines have the advantage of an easy and quick fuel supply over batteries. The gas turbines can continuously operate by supplying the fuel such as propane and kerosene. This is especially effective when the electrical power is shut down in disaster.

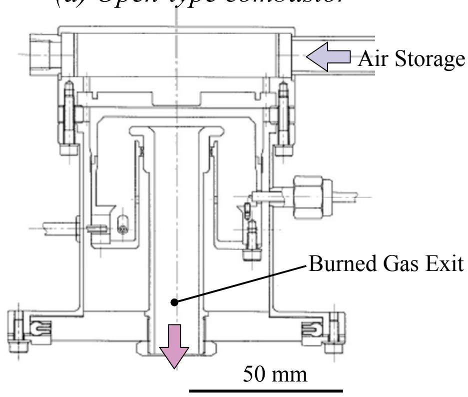
Based on this background, we have been developing micro combustors for the micro gas turbine [2-4]. The palmtop-sized gas turbine developed by IHI Corporation and Tohoku University in 2007 equipped the micro combustor developed by the authors [5]. This combustor required a small size and high combustion efficiency, and thus must have the high space heating rate. In addition, the combustor must satisfy low emissions. In this paper, the concept of the annular type propane-fueled micro combustor and its combustion characteristics are reported.

2. EXPERIMENTAL APPARATUS

To actualize a compact micro combustor, we designed the combustor with a combustion chamber of a volume of 50 cm^3, for obtaining the high space heating rate. The combustor is an annular type and uses propane as a fuel because propane is easy to liquefy and has a large vapor pressure of 0.8 MPa at room temperature. Figure 1 shows the schematic of the micro combustors used in this study. We made two types of combustors. An open-type combustor (Fig.1a) was used to observe the flame appearance in the combustion chamber and to evaluate the flame stability limit, while a closed-type combustor (Fig.1b) was used to examine the exhaust emissions at the exit of the combustor. The supplied air divided into the primary air for combustion, dilution of the burned gas, and liner cooling. In order to realize low NOx emission, a lean-premixed combustion method was applied. The primary air tangentially flowed into the combustion chamber in the orifice upstream region (Fig.1a-X) through four air slots (Fig.1a-Z). Propane was coaxially injected with primary air using two or four fuel tubes, and then formed premixture in the orifice upstream region. This method avoided the flash back of the premixed combustion. The feature of this micro combustor is an orifice locating in the chamber to separate the upstream region and main burning region of the combustion chamber for promoting the fuel/air mixing in the upstream region and forming the

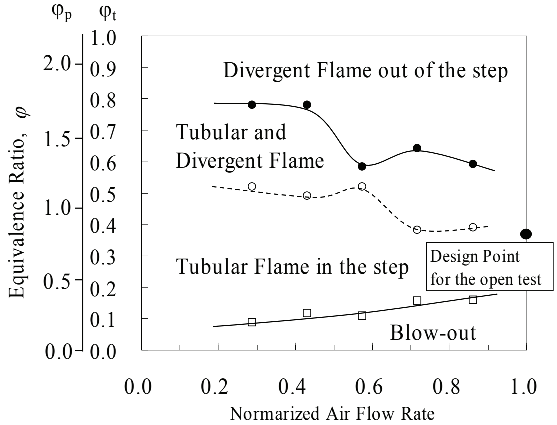


(a) Open-type combustor

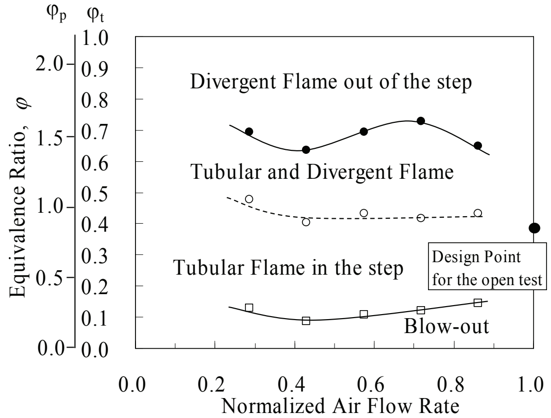


(b) Closed-type combustor

Fig.1: Schematic of the test combustor

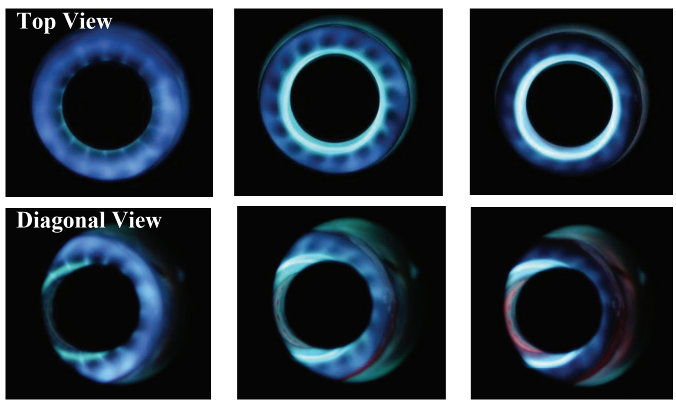


(a) 4 fuel tubes



(b) 2 fuel tubes

Fig.2: Flame stability limits of the combustor



(a) φ_t=0.58 *(b) φ_t=0.43* *(c) φ_t=0.36*

Fig.3: Typical flame appearances in the combustor (Normalized air flow rate: 0.86)

recirculation zone at the step (Fig.1a-Y) for flame holding. An orifice diameter was decided from the results of the previous study [2]. The tubular-shape flame formed from the orifice upstream region to the step region. In order to decrease the burned gas temperature and reduce NOx production, the dilution air was introduced from the inner liner at downstream from the orifice. The ignition was achieved using a spark-type igniter located at the step region.

The experiment was carried out under atmospheric pressure and temperature. The design operation point of this combustor at atmospheric condition is the air mass flow rate of 4.1 g/s and the total equivalence ratio (φ_t) of 0.38. Here the total equivalence ratio means the flow rate ratio of fuel to the total air which includes the primary, dilution and cooling air. On the other hand, the primary equivalence ratio (φ_p) is defined as the flow rate ratio of fuel to the primary air. Flame stability limits, the exhaust CO, total hydro carbon (THC) and NOx emissions were measured by varying the air mass flow rate and the equivalence ratio.

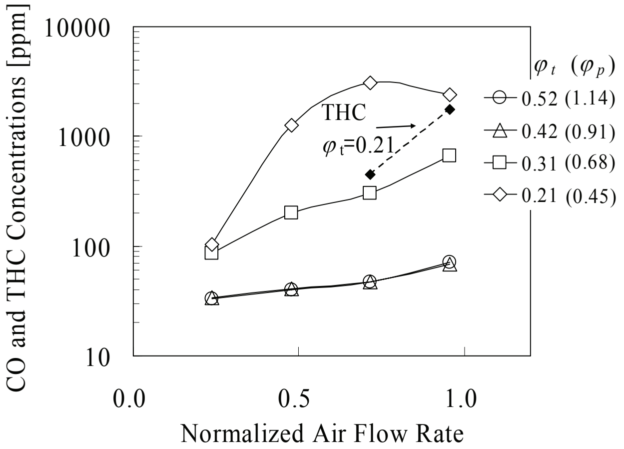
3. EXPERIMENTAL RESULTS AND DISCUSSION

3.1 Flame stability limits

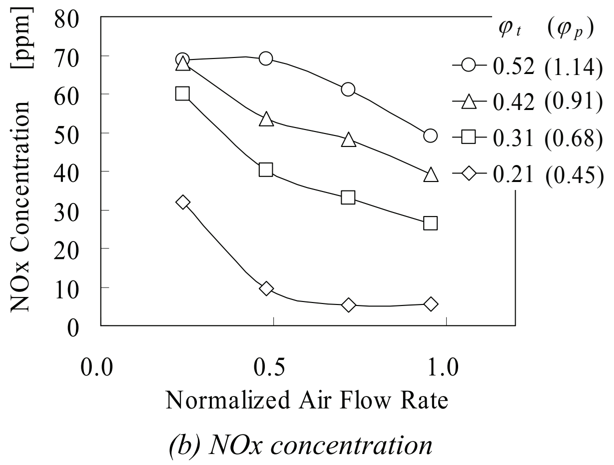
Figure 2 shows the flame stability limits of the combustor, and the typical flame appearances are shown in Fig.3. The x-axis shows the air flow rate normalized by the design operation condition. As the equivalence ratio decreased from the fuel rich condition, the flame shape changed from the divergent flame out of the step (Fig.3a) into the tubular and divergent flame (Fig.3b) and the tubular flame in the step (Fig.3c). Then, the flame finally blew off. These variations in flame shape were attributed to the local equivalence ratio in the orifice upstream region. If the local equivalence ratio was enough high, the burning velocity increased and the flame could propagate to the orifice upstream region, and then formed the tubular flame. As seen in φ_p, it was found that the region of the tubular flame existed around the stoichiometric condition of φ_p. Even though the flame shape was different, each flame was stable and the flame lengths were short. In Fig.3a and Fig.3b flame stripes were observed around dilution air holes as φ_t increased, but the flames spread out in circumference direction without large deviation. Therefore, the flame stable region was enough wide as shown in Fig.2 to satisfy the design operation point when the stability limits were extrapolated. Then maximum air mass flow rate was limited due to the capacity of the air supply system used in this study. However, as shown in Fig.3c, the red-heated wall appeared around the inner liner in the orifice upstream region. In this condition the tubular flame existed only in the orifice upstream region where the chemical reactions were almost completed and the flame temperature increased. Thus, the local heat flux from the flame became intensive. In order to avoid the liner melting, the cooling of the liner needs to be improved.

3.2 Exhaust emissions and combustion efficiency

CO, THC and NOx concentrations measured at the combustor exit by varying the equivalence ratio and the air mass flow rate are shown in Fig.4. In Fig.4a, when the equivalence ratio decreased under the constant air mass flow rate, the CO concentration increased. The CO variation at φ_t between 0.42 and 0.52 was quite small, but that at φ_t less than 0.31 was large. This showed that the chemical reactions were not completed enough at φ_t less than 0.31 due to the decrease of the flame temperature. When the air mass flow rate increased under the constant equivalence ratio, the CO concentration also increased. Only at φ_t of 0.21, the THC concentration was larger than several hundred ppm. This was because the residence time for chemical reactions decreased as the air mass flow rate increased, resulting in the incomplete chemical reactions. The NOx concentration showed the opposite tendency to CO with the equivalence ratio and the air mass flow rate. As the equivalence ratio increased or the air mass flow rate decreased, the NOx

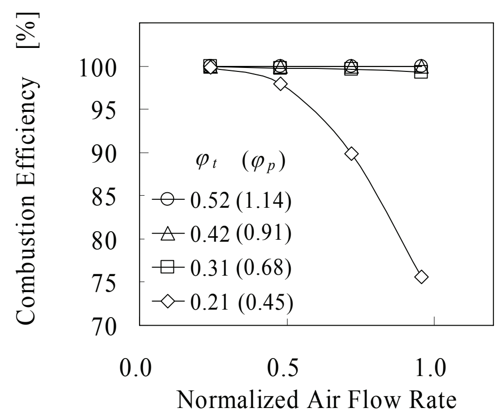


(a) CO and THC concentrations

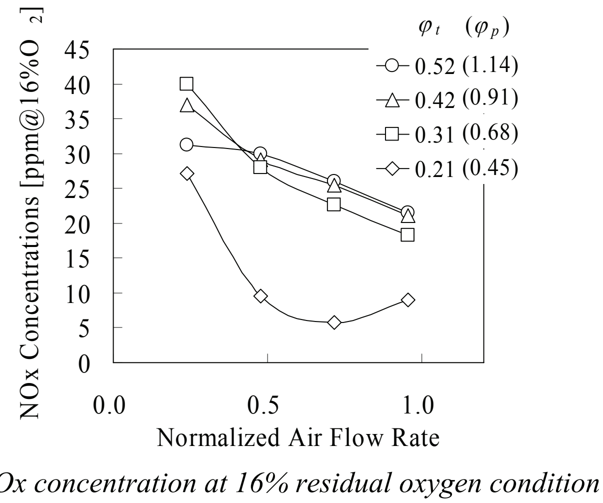


(b) NOx concentration

Fig.4: Variation of CO, THC and NOx concentrations (2 fuel tubes)



(a) CO and THC concentrations



(b) NOx concentration at 16% residual oxygen condition

Fig.5: Variation of combustion efficiency and NOx concentration at 16% residual oxygen (2 fuel tubes)

concentration increased. This is explained by the fact that the chemical reactions for NOx production become remarkable when the flame temperature is higher than 2000 K and the flow residence time becomes longer [6]. At φ_t of 0.21, the NOx concentrations were low because the reactions were not completed enough and the flame temperature was low.

Figure 5 shows the combustion efficiency and NOx concentration corrected at 16% residual oxygen. The combustion efficiency was calculated by the following equation;

$$\eta_b = \left\{ 1 - \left(\frac{\dot{m}_f + \dot{m}_a}{\dot{m}_f} \right) \left(\frac{X_{CO} \Delta H_{CO} M_{CO} + X_{THC} \Delta H_{THC} M_{THC}}{\Delta H_f M_b} \right) \right\} \times 100 \text{ [\%]}$$

Here, m_i is the mass flow rate, ΔH_i the heat of combustion, M_i the molecular weight, X_i the mole fraction in the burned gas. Subscript *i* shows *a*: air, *f*: fuel, *co*: CO, $_{THC}$: THC, *b*: burned gas. The combustion efficiencies were more than 99.5 % at the total equivalence ratios of 0.31 - 0.52 except for 0.21 at which the condition φ_p of 0.45 was under the lower flammable limit of propane. The space heating rate at the design operation point was 800 MW/(m^3 MPa) that was two times larger than conventional jet engine combustors having around 400 MW/(m^3 MPa). Thus, the developed micro combustor achieved both high combustion efficiency and high space heating rate. The NOx concentration corrected at 16% residual oxygen was generally used to compare the NOx emission with other gas turbine combustors operated at different conditions. The NOx emissions were low less than 40 ppm as a whole, and it was 20 ppm at the design operation point. For using a miniature power source, further NOx reduction is desired.

4. CONCLUSION

- The flame stability limit of the micro combustor was enough large for practical use.
- When the combustor operated at the total equivalence ratio over 0.42, the combustion efficiency achieved more than 99.5% and the CO concentrations were less than 70 ppm. The NOx emissions (@16%O_2) at this condition were less than 40 ppm and that was 20 ppm at the design operation condition.
- The developed micro combustor achieved quite high space heating rate of 800 MW/(m^3 MPa) with high combustion efficiency and low exhaust emissions.

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