

Research of Natural Gas Ultra Low Emission Combustor

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ABSTRACT

A lean burn combustor, which provides homogeneous mixing of air and natural gas and prevents autoignition, was developed by adopting a dispersely arranged multi-venturi design. By taking advantage of the characteristics of the venturi mixers, a highly accurate determination of the state of combustion was enabled and an expansion of the premixed combustion range was achieved. The combustor system above was installed on a micro-gasturbine, achieving an ultra low emission level of 3 ppm (at 15 percent oxygen equivalent) NO_x, CO and THC at a 50 to 100 percent load.

1. Introduction

Micro-gasturbines are receiving attention as a distributed energy source, but low emissions performance equivalent to a large capacity power plant is still demanded, for environmental reasons. The lean premix combustion method, which combusts a homogenous mixture at relatively low temperatures, is an effective method for reducing gas turbine emissions. However, it is difficult to generate a homogenous mixture and at the same time prevent autoignition during the mixing process. Also, a lean mixture can easily lower the combustion efficiency and produce unstable combustion, such as combustion oscillation.

Furthermore, to prevent flameout and to ensure a wide, low-emissions operating range when applying this method, especially to a micro-gasturbine, it is necessary to establish a simple method for accurately acquiring the combustion state, which changes according to the operating conditions, and for switching between diffusion combustion and premix combustion as appropriate^{(2), (3)}. This research was intended to develop an environmentally-friendlier micro-gasturbine with low emissions. The goal was to produce a practical lean premix combustor, which was previously considered to be difficult^{(1), (2), (3), (4)}.

2. Premix Combustor

2.1. Emission Targets

This combustor is intended for application on a micro-gasturbine, so the NO_x emission target was set at 3 ppm or less (at 15% oxygen equivalent), which is below the average of all electric power plants including nuclear energy as shown in Fig. 1. In addition, the CO and THC emissions targets were set at the same level as NO_x.

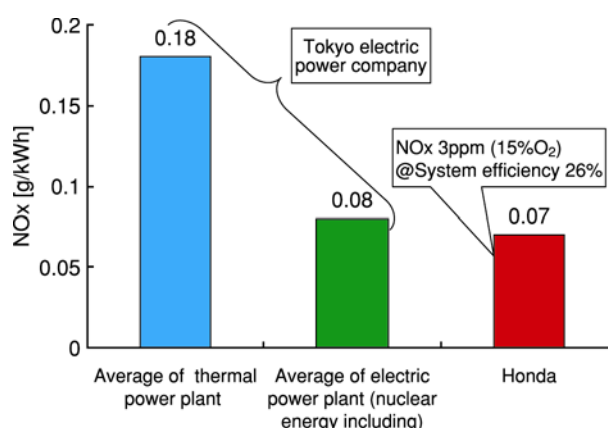
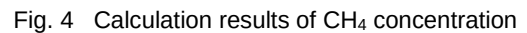
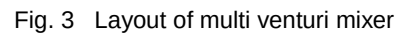
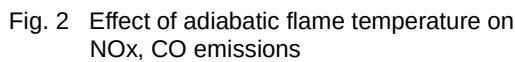


Fig. 1 NO_x emission of electric power plant ⁽⁵⁾

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- (1) Degree of mixing for the premixed air-fuel
Nonuniformity ratio: $\pm 10\%$ or less
(Low-emission combustion Tad bandwidth: $200\text{ }^{\circ}\text{C}$)
- (2) Combustion state estimation accuracy
Tad calculated estimation error: $\pm 25\text{ }^{\circ}\text{C}$ or less
(Combustion air mass flow estimation error: equivalent to $\pm 5\%$ or less)



	Velocity [m/s]	Temperature [K]	Pressure [kPaA]
Air in	100	873	–
CH ₄ in	2.2	473	–
Outlet	–	873	400

– 86 –

3.2. Single Venturi Mixer Unit Test

A single venturi mixer was fabricated based on the CFD calculation results, to analyze the mixing state and optimize the fuel injection method.

3.2.1. Unit test rig

Figure 5 shows the single venturi mixer unit test rig. The mass flow of the air and fuel is arbitrarily adjusted and then supplied to the test rig. The mixture is sampled and the CH_4 concentration is measured using a sampling probe installed downstream from the diffuser.

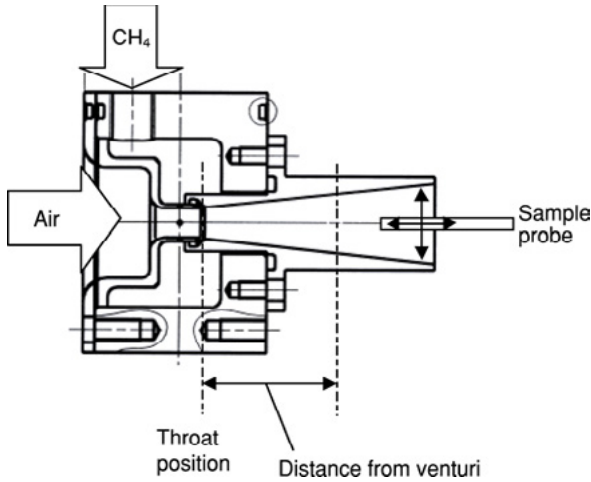


Fig. 5 Cross-sectional view of single venturi mixer

3.2.2. Effect of diffuser expansion angle and flow velocity through the throat on nonuniformity ratio

Figure 6 shows results of the measured nonuniformity ratio of a cross section at an arbitrary distance from the throat. The nonuniformity ratio is defined as the value obtained by dividing the standard deviation of the local concentration by the average concentration. In accordance with the CFD results, the mixing length becomes shorter as the diffuser expansion angle (deceleration rate) increases, and the target nonuniformity ratio at an expansion angle of 12° (TYPE-B) was obtained at the distance of 28 mm from the venturi throat (mixing length/throat diameter: $L/D = 7$).

In addition, Fig. 7 shows the nonuniformity ratio for different air flow velocities through the venturi mixer, and shows that the difference in the flow velocity has little effect on the mixing length.

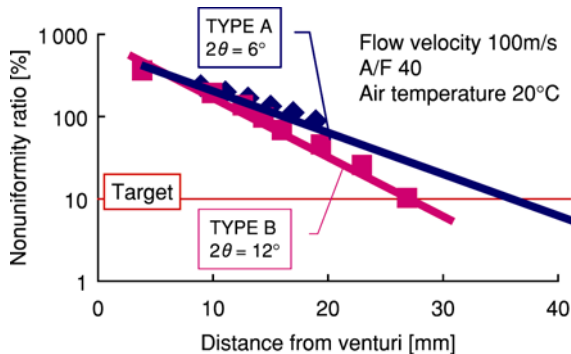


Fig. 6 Effect of diffuser angle on nonuniformity ratio

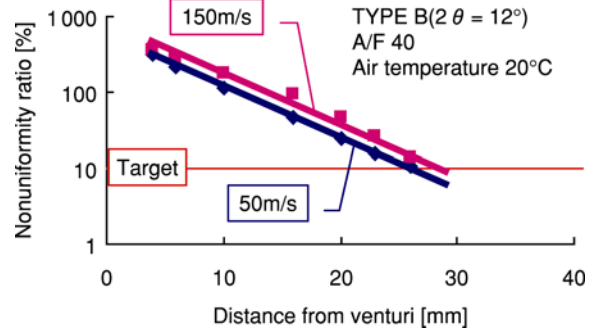


Fig. 7 Effect of velocity on nonuniformity ratio

3.2.3. Effect of fuel injection method on nonuniformity ratio

In order to investigate the effect of the fuel injection method on the nonuniformity ratio, the mixing state downstream from the throat was visualized and the nonuniformity ratio was measured and optimized for the three fuel injection methods shown in Fig. 8. Figure 9 shows the mixing states immediately after the venturi throat and 20 mm downstream from the throat. The mixing state was visualized by supplying oil smoke as a seeding to the fuel side and filming with a laser sheet beam as a light source. In the case of the base type which injects the swirling fuel to the air flow from slits in the throat, the fuel does not disperse over the entire diffuser. This is thought to be because the fuel sticks to the outer walls due to centrifugal force. For the two holes type which injects the fuel toward the center of the throat, the fuel penetrates the air flow appropriately without sticking to the walls, and mixing is promoted.

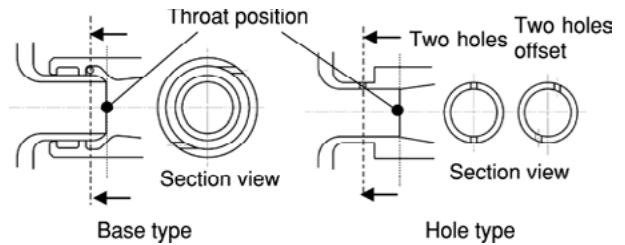


Fig. 8 Cross-sectional view of fuel supply shape

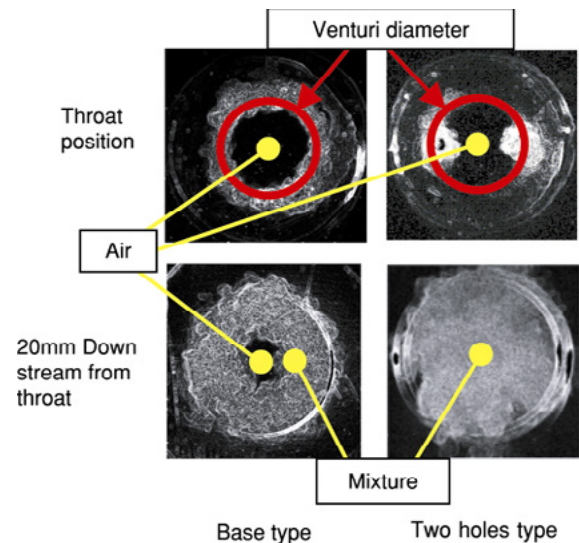


Fig. 9 Comparison of mixture distribution for two different fuel supply shapes

Figure 10 shows the nonuniformity ratio at the diffuser outlet. Homogeneous mixing can be improved by offsetting the injection holes. This is thought to be because the offset causes the fuel to disperse in the perpendicular direction near the throat walls. Therefore, the two holes + offset type is effective for mixing with in a shorter length.

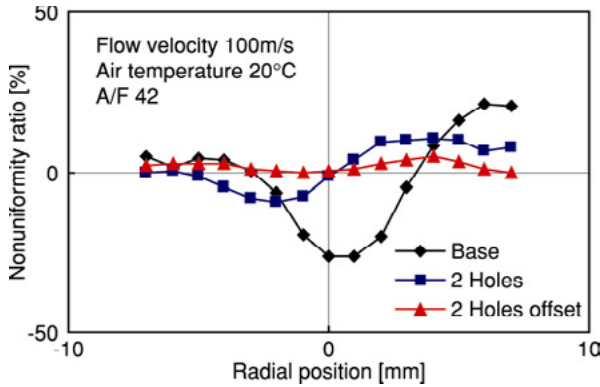


Fig. 10 Effect of fuel supply shape on nonuniformity ratio

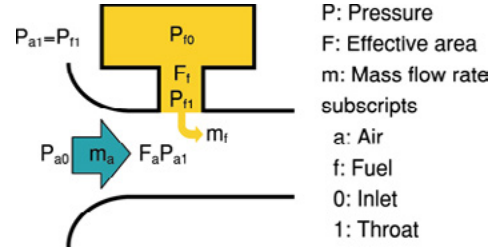


Fig. 11 Model of venturi mixer

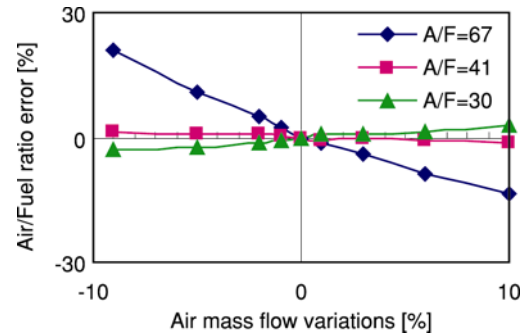


Fig. 12 Effect of air mass flow variations on air/fuel ratio error

3.3. Multi Venturi Mixer

3.3.1. Synchronization of mixture ratio between venturi mixers

The multi-venturi mixer for use in an actual combustor shown in Fig. 3 may experience variation in the air mass flow passing through each venturi mixer due to the pressure distribution inside the combustion cylinder during combustion, diffuser shape error, etc. This causes variation in the mixture ratio of each venturi mixer, and consequently variation in the distribution of the mixture concentration within the combustion cylinder. Therefore, the following method was used to synchronize the mixture ratio of each venturi mixer.

Assuming the fuel inlet total pressure, P_{f0} , and the air inlet total pressure, P_{a0} , in Fig. 11 to be equal (fuel throat static pressure, P_{f1} , equals air throat static pressure, P_{a1}), the flow velocities through the fuel effective area, F_f , and the air effective area, F_a , are equal, so the effective area ratio, F_a/F_f , equals the mass flow ratio, m_a/m_f .

Since the physical properties (density, specific heat ratio) of fuel and air are similar under actual operating conditions, it can be said that when P_{f0} and P_{a0} are equal, F_a/F_f and m_a/m_f are approximately equal. This indicates that even if the air mass flow changes due to the effects of the venturi downstream, m_a/m_f can be kept constant by maintaining a constant P_{f0} and P_{a0} . Application of this principle made it possible to reduce the air/fuel ratio variation between each mixer by setting the F_a/F_f of the multi-venturi mixer near the required air/fuel ratio.

Figure 12 shows the error relative to the set air/fuel ratio when the fuel and air inlet pressures are kept constant and the air mass flow passing through the venturi mixer is varied by changing the outlet pressure on a venturi mixer with the effective area ratio set to 40. This shows that the air/fuel ratio error relative to changes in the air mass flow becomes

smaller at an air/fuel ratio near the set effective area. This indicates that the air/fuel ratio variation between each venturi mixer can be reduced even when the air mass flow passing through each venturi mixer in a multi-venturi mixer changes.

3.3.2. Nonuniformity ratio of the multi-venturi mixer

Figure 13 shows results of the measured nonuniformity ratio at the mixer outlet cross section (combustion cylinder inlet cross section) of a multi-venturi mixer for use in an actual combustor in order to confirm the nonuniformity ratio of the mixture. The nonuniformity ratio was within $\pm 5\%$ for over 95% of the mixer outlet cross section, which sufficiently achieved the target of $\pm 10\%$ or less. This confirmed that a homogenous mixture could be formed within the combustion cylinder even with a multi-venturi mixer arrangement by synchronizing the mixture ratio between each venturi mixer.

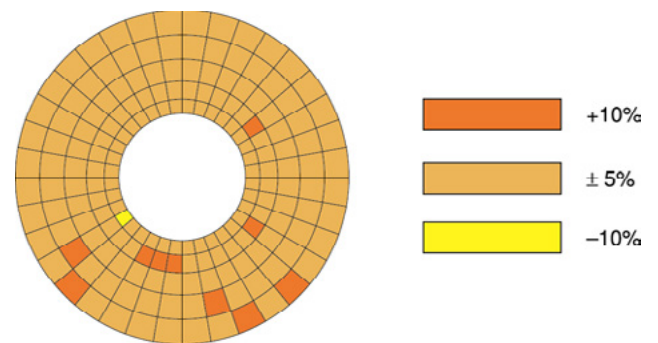


Fig. 13 Nonuniformity ratio at the outlet of multi venturi mixer

4. State Detection

4.1. Combustion State Detection Method

In order to ensure the low emission performance of 3 ppm or less at a micro-gasturbine load of 50% to 100% and to realize combustion switching without flameout, the T_{ad} inside the combustor must be acquired in real time. Figure 14 shows the parameters needed to estimate T_{ad} .

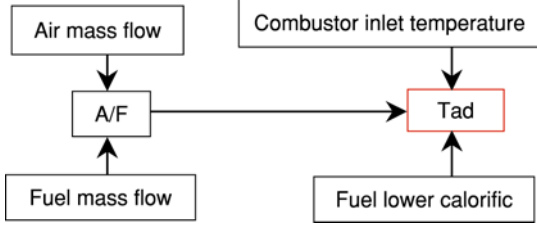


Fig. 14 Parameters for estimation of T_{ad}

The air passing through this multi-venturi mixer becomes the combustion air, so the air mass flow was estimated using the venturi characteristics of the venturi mixers. When the venturi equation is transformed as shown in equation (1), it can be broadly divided into two parts, the inlet and throat pressure term and a flow characteristic term containing the effective area, which expresses the venturi mass flow characteristics. The mass flow characteristics (effective area) change due to the effects of the flow velocity and the viscosity, thus investigations were conducted to obtain the air mass flow from the measured venturi inlet conditions, fuel mass flow and fuel pressure from a map expressing the air mass flow with the fuel mass flow and pressure provided to the venturi mixer.

$$Ga^* = F \sqrt{\frac{2\kappa}{\kappa-1} \frac{1}{\sqrt{R}}} \sqrt{\left(\frac{P_s}{P_{in}}\right)^{\frac{2}{\kappa}} - \left(\frac{P_s}{P_{in}}\right)^{\frac{\kappa+1}{\kappa}}} \quad (1)$$

Flow characteristic term
Pressure term

Ga^* : Corrected Air mass flow, F : Effective area,
 P : Pressure

κ : Specific heat ratio, T : Temperature, R : gas constant
 Subscripts in: Inlet, s: Throat

Based on the corrected test results of the venturi mixer, when the mass flow that includes varying mass flow characteristics is converted to the standard conditions (15 °C at standard atmospheric pressure) and expressed as the corrected air mass flow, Ga^* corrected, the mass flow characteristics remain fairly constant and do not change relative to the pressure term value even when the inlet conditions differ as shown in Fig. 15. This means that it is possible to obtain the air mass flow based on the inlet pressure and the throat pressure. However, measuring the throat pressure is difficult in an actual combustor, so the fuel mass flow and the fuel pressure were corrected to the standard state and expressed as the corrected fuel mass flow, Gf^* , and the corrected fuel pressure, Pf^* . In this case as well, the Ga^* at different inlet conditions exhibited approximately the same gradient (Fig. 16).

The air mass flow at different inlet conditions can be obtained by compiling the Gf^* , Pf^* and Ga^* obtained from the above corrected tests into a three-dimensional map (Fig. 17). The feature of this map is that when the air mass

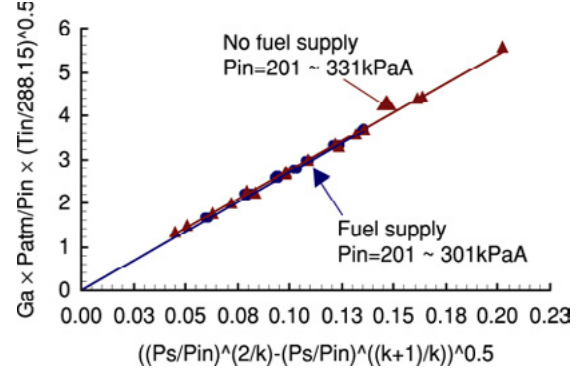


Fig. 15 Venturi flow characteristic

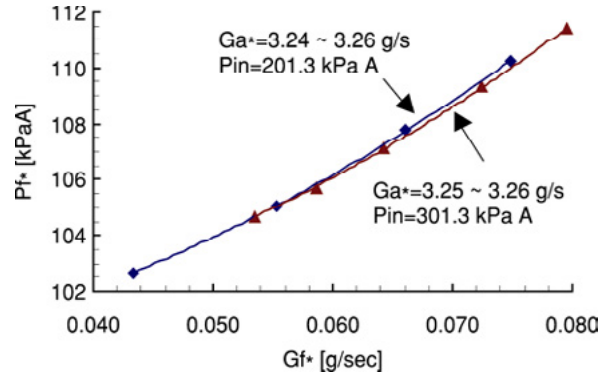


Fig. 16 Venturi flow characteristic

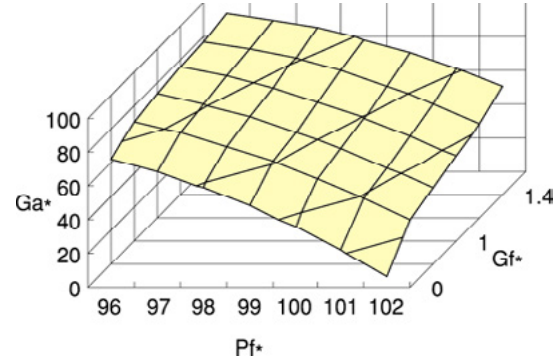


Fig. 17 Air mass flow map

flow is reduced at a constant fuel mass flow, the venturi characteristics cause the throat static pressure to rise, which causes the fuel supply pressure to also rise. Thus, the map takes the general form of a graph sloping downward to the right.

4.2. State Detection Accuracy

4.2.1. Combustion air mass flow calculation accuracy

Figure 18 shows the results of combustion tests conducted in the condition where the entire mass of the supplied air was used as combustion air in order to confirm the accuracy of the air mass flow calculations. The calculation error was approximately $\pm 5\%$ or less in all regions except for the low mass flow region of diffusion combustion, which confirmed that there are no problems for practical application. The factor lowering the measurement accuracy in the low mass flow region is thought to be the accuracy of the pressure and temperature sensors.

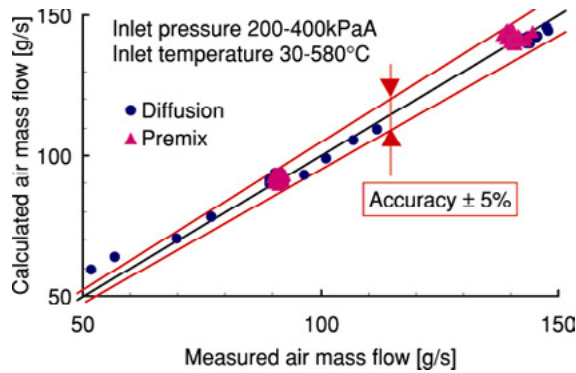
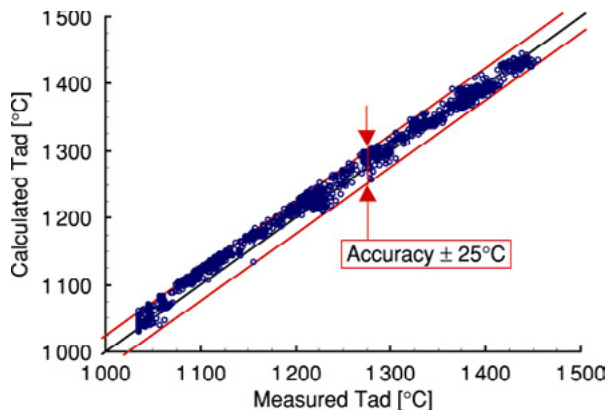


Fig. 18 Accuracy of calculated air mass flow

4.2.2 Adiabatic flame temperature calculation accuracy

The adiabatic flame temperature (T_{ad}) is calculated using the air/fuel ratio, combustor inlet air temperature, fuel temperature, fuel lower calorific value, and the gas enthalpy before and after combustion. However, regressive calculation should be avoided in actual combustor control calculations. In addition, calculation was performed using a map in order to simplify the calculations. Figure 19 shows the T_{ad} calculation results based on tests to confirm the fuel and air mass flow calculation accuracy. This confirmed that the calculation error does not worsen when using a map, and that the error was within the target range of $\pm 25^\circ\text{C}$.

Fig. 19 Accuracy of calculated T_{ad}

5. Combustion Tests

Methods for generating a homogenous mixture using a multi-venturi mixer and for detecting the combustion state by T_{ad} estimation were established, so the combustion and practical performance were evaluated using a combustion test rig.

5.1. Combustion Test Rig

Figure 20 shows the test rig used for the combustor combustion tests. The air is pressurized by a compressor, adjusted to the prescribed mass flow and temperature by a flow adjusting valve and a heater, and then supplied to the combustion test rig. In addition, the pressure is adjusted by a pressure regulation valve located downstream from the combustion test rig. The fuel (natural gas, equivalent to 13A: $\text{CH}_4 = 88.5\%$, $\text{C}_2\text{H}_6 = 4.6\%$, $\text{C}_3\text{H}_8 = 5.4\%$, $\text{C}_4\text{H}_{10} = 1.5\%$) is pressurized by a fuel compressor, then the mass flow is

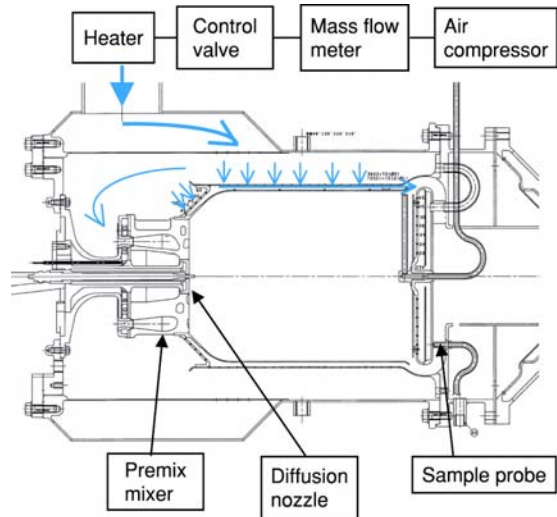
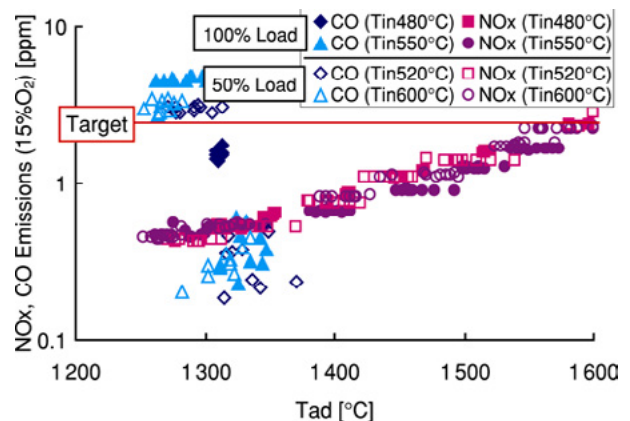


Fig. 20 Cross-sectional view of combustor rig

controlled by independent diffusion and premix fuel control valves, and the fuel is supplied to the combustion test rig. The combustion gas mixes with the diluted air that has cooled the combustion cylinder, and the mixture is then sampled by a sampling probe installed in the exhaust passage. Note that analysis was performed using two analysis meters with different resolutions in accordance with the concentration of the combustion gas components.

5.2. Premix Combustion Performance

Figure 21 shows the results of combustion tests performed at air mass flow and pressure conditions equivalent to the micro-gasturbine load conditions of 50% and 100% with the combustor inlet air temperature, T_{in} and T_{ad} , as parameters. Note that the tests were performed with T_{in} set from 420°C to 600°C which corresponds to actual combustor conditions. The results showed that the low-emission combustion T_{ad} range during premix combustion was 250°C , which satisfied the target of 200°C . This is thought to be because the nonuniformity ratio was $\pm 5\%$ or less for over 95% of the cross section area at the mixer outlet as shown in Fig. 13. Thus, an even temperature distribution was achieved over the entire combustion flame. In addition, there was no combustion oscillation or autoignition which had been a concern for premix combustion. Stable combustion was obtained.

Fig. 21 Effect of T_{ad} and inlet temperature on NOx, CO emissions

5.3. Switching between Diffusion Combustion and Premix Combustion

The results in Fig. 21 show that the point at which the CO concentration exceeds 3 ppm is around $T_{ad} = 1300\text{ }^{\circ}\text{C}$, although this varies slightly according to T_{in} . Therefore, the switching test conditions were set as shown in Table 2. The diffusion $\rightarrow$ premix and premix $\rightarrow$ diffusion conditions differ for the following two reasons.

- There is a T_{ad} calculation error of $\pm 25\text{ }^{\circ}\text{C}$
- To prevent unnecessary switching due to temperature variation

The combustion state is switched when the T_{ad} calculation value and T_{in} exceed the threshold values given in Table 2. However, the fuel passage differs for diffusion combustion and premix combustion, so a time lag occurs until the fuel pressure and temperature reach a static state depending on the piping and fuel passage volume, etc. This causes combustion to become unstable, and under some conditions can result in flameout. To prevent this, a time interval was established during which the fuel is supplied to both fuel passages, and switching was performed in steps. That is, diffusion combustion is possible even at 0.3 g/sec fuel mass flow (equivalent to an air/fuel ratio of 400), so when switching, the total fuel supply was maintained and a distribution ratio was used whereby the diffusion fuel mass flow was set to 0.3 g/sec and the remainder was provided as premix fuel. Figure 22 shows the switching test results. In both cases (diffusion $\rightarrow$ premix, premix $\rightarrow$ diffusion), stable combustion switching was confirmed with no flameout. These results provided prospects for stable operation of an actual micro-gasturbine.

Table 2 Set value for switching conditions

	Diffusion $\rightarrow$ Premix	Premix $\rightarrow$ Diffusion
$T_{in}\text{ [}^{\circ}\text{C]}$	520	480
$T_{ad}\text{ [}^{\circ}\text{C]}$	1350	1300

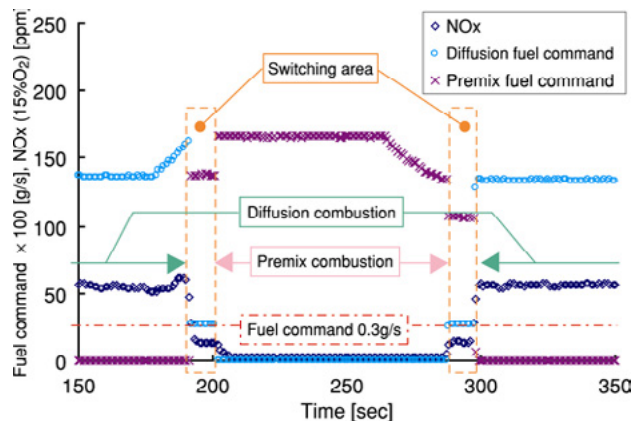


Fig. 22 Switching test results

5.4. Observation of the Combustion Flame

Figure 23 shows photographs of the combustion flame before and after diffusion and premix combustion switching as observed downstream from the combustion test rig. The flames (luminous flame, blue flame, etc.) evident during normal combustion cannot be observed at all during premix combustion, indicating that an ideal lean premix combustion was realized.

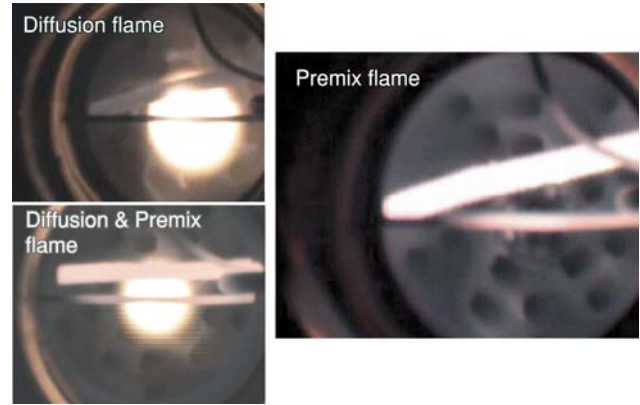


Fig. 23 Combustion flame pictures

6. Actual Micro-gasturbine Tests

Figure 24 shows the operation results for the combustor installed in a micro-gasturbine when the load is varied incrementally from ignition to a 100% load. This shows that operation switches from diffusion combustion to premix combustion and the NOx level drops sharply when the engine load rises to around a 50% load (22.5 kW) which is the combustion switching condition. At this time the engine revolution speed fluctuation was 1% or less, and operation was stable. In addition, operation at the emission (NOx, CO, THC) level target of 3 ppm or less (at 15% oxygen equivalent) was confirmed within the operation range of a 50% to 100% engine load as shown in Fig. 25^{(6), (7)}.

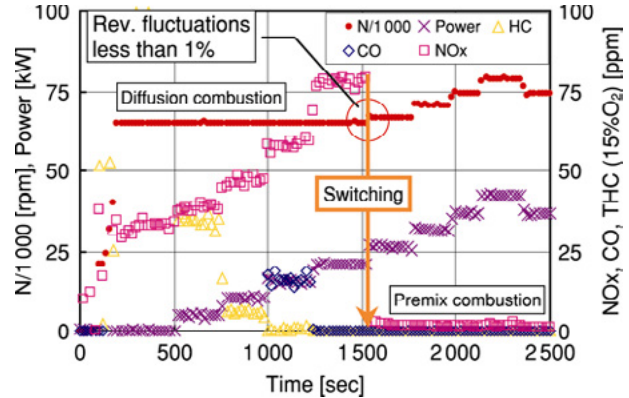


Fig. 24 Engine operation chart

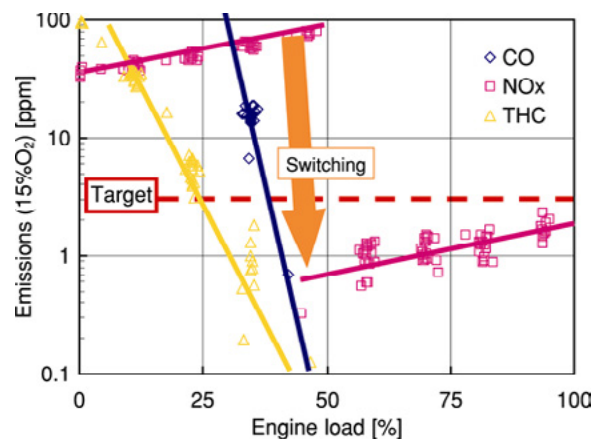


Fig. 25 Emissions characteristics of engine test at engine inlet temperature 10 °C

7. Conclusions

A multi-venturi mixer was adopted as a natural gas lean premix low emission combustor for micro-gasturbine, and the following knowledge was gained as a result of unit tests and actual engine tests.

- (1) Venturi mixers were applied as the premix mixer, and the mixer shape and fuel injection method were optimized to create a compact premix mixer that can generate a homogenous mixture. This resulted in a natural gas lean premix low emission combustor that simultaneously achieves wide low emission combustion range and autoignition immunity.
- (2) An accurate fuel control method that does not require complex devices was constructed by utilizing the venturi characteristics of the premix mixer.
- (3) This combustor was mounted in a micro-gasturbine, and an emission value of 3 ppm or less (NO_x, CO, THC at 15% oxygen equivalent) was verified within the normal operating range of a 50% to 100% load.
- (4) These research results provide prospects for the practical application of a low emission micro-gasturbine.

References

- (1) Tou, H., Watanabe, T., Hosoi, J.: Research and Development of Gas Turbine Combustors, IHI ENGINEERING REVIEW, Vol. 34, No. 3, p. 181-185 (1994)
- (2) Kajita, S., Ohga, S., Ogata, M., Itaka, S., Kitajima, J., Kimura, T., Okuto, A.: Development of a Dry, low-NO_x Combustor for 1 500-Class Gas Turbine, Kawasaki Technical Review, No. 128, p. 25-30 (1996)
- (3) Fujimori, T., Saito, T., Mizutani, T., Tou, H.: Research on Low NO_x Combustor Using Lean Premixed Pre-Vaporized Combustion in High Temperature and High Pressure Gas Turbine, IHI Engineering Review 2000, Vol. 40, No. 3, p. 83-87 (2000)
- (4) Manda, S., Inada, M., Fukue, I., Tanimura, S.: Research on Low NO_x Combustor for High-Temperature Gas Turbine, Mitsubishi Heavy Industries Technical Review, Vol. 31, No. 4, p. 243-246 (1994)
- (5) Tokyo Electric Power Company: TEPCO Environmental Action Report 2002, p. 113 (2002)
- (6) Shinmura, K., Yamaguchi, K., Tsuzuki, S., Kimura, H., Kato, T.: Research of Micro-Gasturbine Generator, HONDA R&D Technical Review, Vol. 15, No. 2, p. 7-16 (2003)
- (7) Shinmura, K.: Microturbine Research at HONDA, Presented in 2003 ASME TURBO EXPO Panel session (2003)

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