Influence of Type of Burner on NOx Emission Characteristics from Combustion of Palm Methyl Ester

Nozomu Hashimoto, Hiroyuki Nishida, Yasushi Ozawa, Tetsushiro Iwatsubo and Jun Inumaru

Abstract—Palm methyl ester (PME) is one of the alternative biomass fuels to liquid fossil fuels. To investigate the combustion characteristics of PME as an alternative fuel for gas turbines, combustion experiments using two types of burners under atmospheric pressure were performed. One of the burners has a configuration making strong non-premixed flame, whereas the other has a configuration promoting prevaporization of fuel droplets. The results show that the NOx emissions can be reduced by employing the latter burner without accumulation of soot when PME is used as a fuel. A burner configuration promoting prevaporzation of fuel droplets is recommended for PME.

Keywords—Palm methyl ester (PME), biodiesel fuel, gas turbine, spray combustion, NOx emission.

I. INTRODUCTION

IN recent years, a great deal of attention has been paid to biomass fuels as renewable energy resources from the viewpoint of environmental concerns and the rising price of fossil fuels. Palm oil is one of the most promising alternative fuels to liquid fossil fuels because palm trees have a very high oil yield compared to other crops [1], and recently, its production has been increasing rapidly [2]. Palm methyl ester (PME) derived from palm oil is the most promising biomass fuel as alternative diesel fuel because its property is mostly similar to conventional diesel fuel. Some researchers have conducted combustion experiment on PME for diesel engine and have found good combustion performance [3-8]. However, the biggest weak point of PME as a fuel for automobile is its high pour point [9]. On the other hand, PME may be used as an alternative fuel for combined cycle power plants with a gas turbine, which have high thermal efficiency. In contrast to that for automobiles, fuel temperature control for power plants is easier.

Authors have conducted the PME combustion experiment

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employing a burner promoting the prevaporization of fuel droplets and found that NOx emission from PME combustion is lower than that from diesel fuel combustion [10]. In the actual gas turbine combustor, however, burners with a configuration having air inlets near fuel nozzle for preventing the accumulation of soot are commonly employed.

In this study, the influence of type of burner on NOx emission and soot accumulation from combustion of PME was investigated. Combustion experiments on PME and No. 2 diesel fuel at atmospheric pressure using two types of burners were conducted.

II. FUEL PROPERTIES

PME is produced by the esterization of palm oil using methanol. Three moles of methanol is consumed for one mole of palm oil through esterization. Unlike diesel fuel, PME has oxygen and no aromatic ring in the molecule. The fuel properties of the PME used in this study are listed in Table I. The properties of typical diesel fuel are also listed for comparison. The lower heating value (LHV) of PME is about 10% lower than that of diesel fuel. The flash point, pour point and viscosity of PME are higher than those of diesel fuel. It should be noted that because of the high pour point of PME, a heat trace system for a fuel supply line is required to use PME in winter in Japan.

Distillation curves for PME and diesel fuel are shown in Fig. 1. From Fig. 1, it is expected that the vaporization rate for PME droplets is lower than that for diesel fuel. It should be noted that the temperature extent of the boiling points of substances constituting PME are narrower than that constituting diesel fuel.

TABLE I
FUEL PROPERTIES

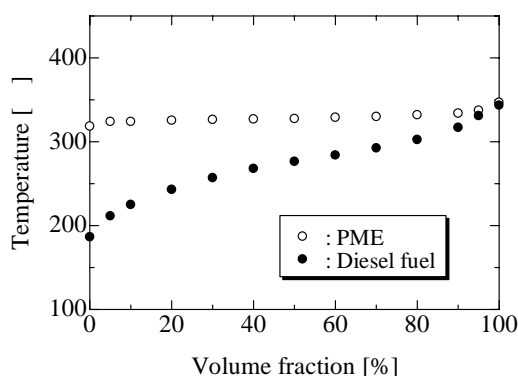|	PME	No. 2 Diesel Fuel
Density (g/cm^3) at 303 K	0.8636	0.8252
Flash point (K)	447	336
Pour point (K)	289	251
Surface tension (dyn/cm) at 300 K	30.6	28.0
Kinematic viscosity (mm^2/s) at 303 K	5.627	3.292
Carbon content (wt%)	76.4	86.4
Hydrogen content (wt%)	12.4	13.5
Oxygen content (wt%)	11.2	< 0.1
Carbon residue content (wt%)	< 0.01	0.01
Lower heating value (J/g)	36,770	43,100

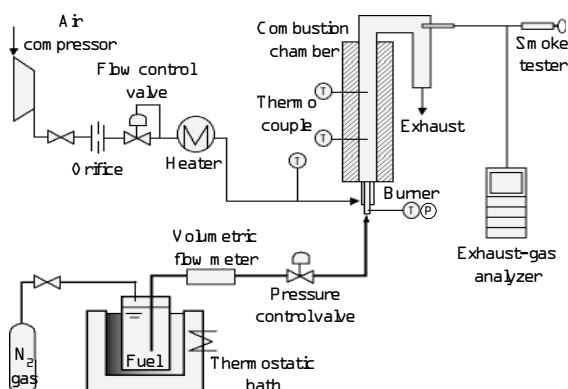Fig. 1 Distillation curves for PME and diesel fuel

Fig. 2 Experimental apparatus

III. EXPERIMENTAL

Fig. 2 shows a schematic of the experimental apparatus. The apparatus consisted of a combustion chamber, an air supply line, a fuel supply line, and an exhaust gas analyzer. The combustion experiments were carried out under atmospheric pressure. Inner diameter of the exhaust line is large enough to maintain the combustion chamber pressure at atmospheric pressure. To take into account the high temperature of compressed air in the actual gas turbine system, the air supplied to the combustion chamber was heated to 673 K by an electric heater in all cases. All of air was supplied from a burner placed at the bottom of the combustion chamber. The combustion chamber was cylindrical shape with inner diameter of 150 mm and an axial length of 910 mm, and was made of an insulation material covered with a metal jacket. The outside of the metal jacket was cooled by a water-cooled tube. A pressure-swirl-type atomizer was used for this research. The fuel was pressurized by supplying N_2 gas to the fuel chamber. The pressure the nozzle was measured by a pressure sensor. The fuel flow rate was measured using a volumetric flowmeter placed in the fuel supply line. The entire fuel supply line including the fuel chamber was heated by

electric heaters so that the temperature just upstream of the nozzle tip could be controlled. The temperatures of PME and diesel fuel were set at 325 K and 303 K, respectively (kinematic viscosity of 3.3 (mm^2/s) for both fuels). The concentrations of CO, NOx, O_2, and total hydro carbons (THC) in the exhaust gas from the combustion chamber were measured by a gas analyzer (CO, non-dispersive infrared method; NOx, chemi-luminescence method; O_2, paramagnetic method; THC, frame ionization method). The soot in the exhaust gas was also analyzed by a smoke tester and was evaluated Bosch smoke number (BSN). Ignition was carried out by electric discharge from the spark igniter.
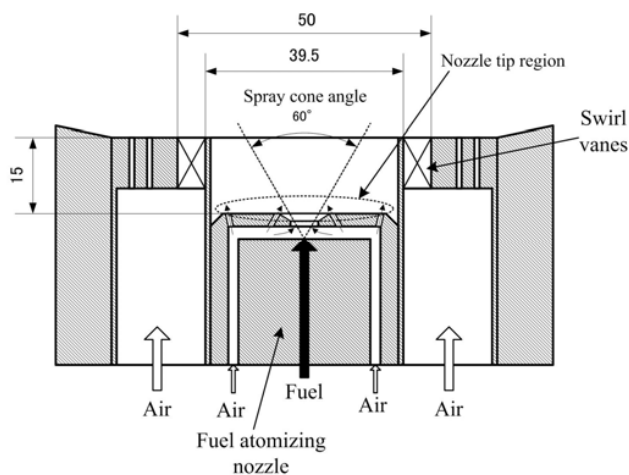

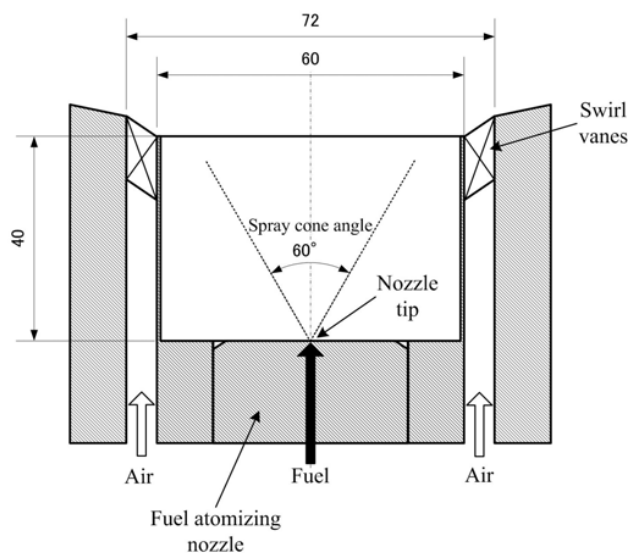
Fig. 3 Schematic of type A burner

Fig. 4 Schematic of type B burner

In this study, two types of burner were used. Fig. 3 shows schematic of type A burner. Fuel atomizing nozzle is placed at the center of the burner. The spray cone angle of the nozzle is

60 degree for both burners. Detailed structure of the atomizing nozzle is available in [10]. For the type A burner, a small amount of air (9% of total air) is supplied to the nozzle tip region for preventing the accumulation of soot. Consequently, it is considered that the flame is formed in the region near the atomizing nozzle tip.

Fig. 4 shows schematic of type B burner. The largest difference from the type A burner is that there is no air outlet near the atomizing nozzle tip for the type B burner. All of air is supplied to the swirler. Therefore, it is considered that the flame is formed at the region distant from the atomizing nozzle tip. Consequently, the prevaporization of fuel droplets is promoted.

IV. RESULTS AND DISCUSSION

Fig. 5 shows photographs of the atomizing nozzle after the combustion experiment for type B burner. It is found that heavy accumulation of soot on the nozzle is observed for diesel fuel combustion. For PME combustion, on the other hand, there is little accumulation of soot on the nozzle. This difference in the tendency of soot accumulation is considered to be caused by the difference in chemical structure between fuels. As mentioned in section 2, PME has oxygen and no aromatic ring in the molecule. Wang et al. [11] found that the sooting tendency of the fuel containing aromatic ring in the molecule is higher than that of the fuel containing no aromatic ring. The same trend was also seen in the research by Cignoli et al. [12] who observed the sooting tendency of various diesel fuels containing different amounts of aromatics. They also found the reduction in soot formation with oxygen addition to the fuel. From previous research results cited above, it is considered that PME has less tendency to produce soot than diesel fuel. This is the reason why there is little accumulation of soot for PME combustion whereas the heavy accumulation of soot is observed for diesel fuel combustion. Employing the type B burner for diesel fuel may cause the problem because of the soot accumulation.

No soot accumulation on the nozzle was observed for the type A burner for both fuels. This indicates that air outlets near the nozzle tip shown in Fig. 3 are working well.

Although soot accumulation is observed around the type B burner in for diesel fuel, no measurable soot was detected by smoke tester in the exhaust gas sampled from the sampling port for all cases. This is considered to be due to the fact that all the soot produced in the burner region was completely consumed before the combustion gas arrived at the sampling port.

Fig. 6 shows NOx emissions as functions of adiabatic flame temperature. In this study, NOx emissions are evaluated in volumetric concentration in the exhaust gas corrected to 16% O_2. The word "adiabatic flame temperature" in this study is defined as the gaseous temperature if all of the air and the fuel supplied into the combustion chamber react in the adiabatic constant-pressure condition, and was calculated by the Chemical Equilibrium with Applications (CEA) code [13]. The values of LHV and fractions of carbon, hydrogen and oxygen listed in Table I were used to input data for CEA code.

(a) After diesel fuel combustion

(b) After PME combustion

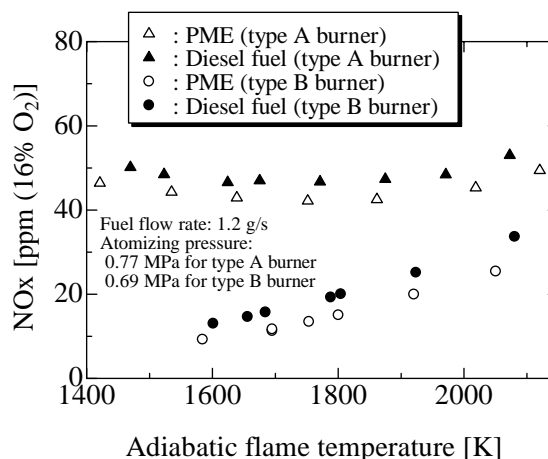Fig. 5 Photographs of the atomizing nozzle after combustion experiment for type B burner

Fig. 6 NOx emissions as functions of adiabatic flame temperature.

In Fig. 6, air supplying rate is changed at a constant fuel supplying rate. It is found that NOx emissions for the type B

burner are lower than that for the type A burner over the whole range of adiabatic flame temperature. In addition to that, it is also found that effects of the adiabatic flame temperature on NOx emission between burners are significantly different. For the type B burner, NOx emission decreases with the decrease of the adiabatic flame temperature. For the type A burner, on the other hand, there is no significant change in the NOx emission regardless of the adiabatic flame temperature. These differences are considered to be caused by the difference in the dominant combustion process, i.e., the dominant combustion process for the type A burner is in non-premixed mode, whereas that for the type B burner is in premixed mode. For the combustion process in non-premixed mode, the combustion reaction is considered to occur under near-stoichiometric air/fuel ratio, in which local flame temperature is 2400-2430 K in the case of this study and is almost unaffected by the adiabatic flame temperature. For the combustion process in premixed mode, on the other hand, the local flame temperature is affected by local air/fuel ratio of premixed gas, which is considered to be decreased with the decrease of adiabatic flame temperature. Because the type B burner has the configuration promoting prevaporization, some proportion of the fuel is prevaporized and premixed with the air in the upstream region of combustion zone. Decrease in the adiabatic flame temperature by increasing of air flow rate causes the increase in the air/fuel ratio of premixed gas, and consequently causes the decrease in local flame temperature. Therefore, decrease in the adiabatic flame temperature causes decrease in the local flame temperature for the type B burner. Consequently, the formation of thermal NOx, which is very sensitive to the local flame temperature, is decreased with decrease of the adiabatic flame temperature. For the type A burner, on the other hand, the formation of thermal NOx is less sensitive to the adiabatic flame temperature than that for the type B burner, because the local flame temperature is almost constant in the non-premixed mode. Above discussion is also supported by Fig. 7 and Fig. 8.

Fig. 7 and Fig. 8 show NOx emissions as functions of the fuel kinematic viscosity for the type A burner and for the type B burner, respectively. It should be noted that only the fuel kinematic viscosity is changed in the graphs, i.e., the fuel flow rate and air flow rate are fixed in the graphs. The fuel kinematic viscosity is changed by the changing the fuel temperature. It is considered that the mean diameter of fuel droplets is decreased with the decrease of the viscosity. For the type A burner (Fig. 7), NOx emission is almost constant regardless of the fuel kinematic viscosity. For the type B burner (Fig. 8), on the other hand, NOx emission is increased with the increase of the fuel kinematic viscosity. This difference is considered to be caused by the difference in the dominant combustion process between burner types, as discussed previously. For the type B burner, the ratio of the combustion process in non-premixed mode to the combustion process in premixed mode increases as the prevaporization is suppressed by the increase in the fuel droplet diameters due to the increase in the fuel kinematic viscosity. This is the reason why the NOx emission is increased with the increase of the fuel kinematic viscosity for the type B burner.

It is considered that the ratio of combustion process in premixed mode for the type A burner is low regardless of the mean diameter of fuel droplets because of the burner configuration. Therefore, the fuel kinematic viscosity does not affect the ratio of the combustion process in non-premixed mode to the combustion process in premixed mode for the type A burner. Consequently, NOx emission is almost constant regardless of the fuel kinematic viscosity.
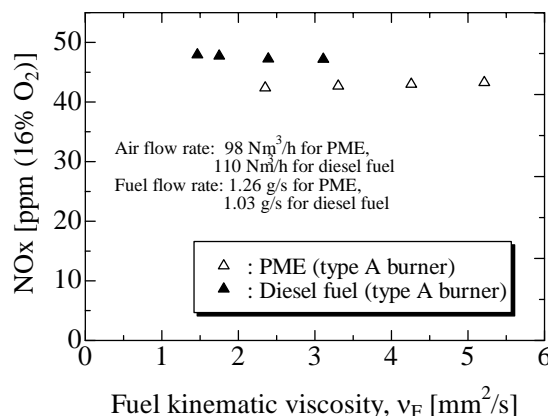

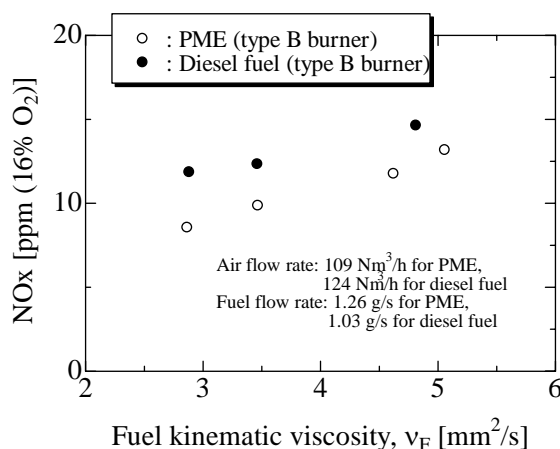Fig. 7 NOx emissions as functions of fuel kinematic viscosity for the type A burner

Fig. 8 NOx emissions as functions of fuel kinematic viscosity for the type B burner

From Fig. 6, it can be said that when PME is used as fuel, the NOx emission can be reduced by employing a burner configuration promoting prevaporzation like the type B burner in this study, although the burner cannot be employed for diesel fuel because of the soot accumulation. In addition to that, further reduction of NOx emission is possible by decreasing fuel kinematic viscosity by preheating of the fuel.

Fig. 9 shows CO emissions as functions of the adiabatic flame

temperature. It is found that CO emissions for the type A burner are higher than that for the type B burner over the whole range of adiabatic flame temperature. For the type A burner, CO emission increases with the decrease of the adiabatic flame temperature. This is expected to be due to the instability of the flame under the low adiabatic flame temperature condition.

Fig. 10 shows THC emissions as functions of the adiabatic flame temperature. It is found that THC emissions are very low level for both types of burners over the whole range of the adiabatic flame temperature in this study.
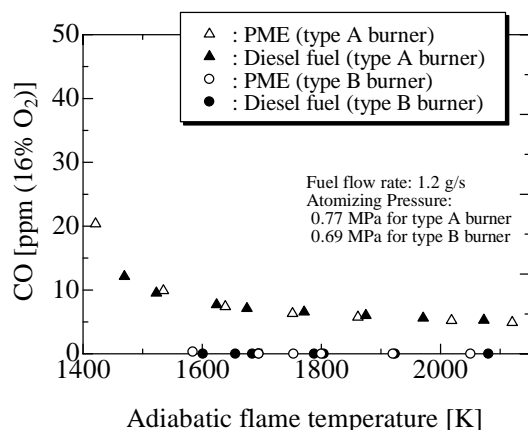

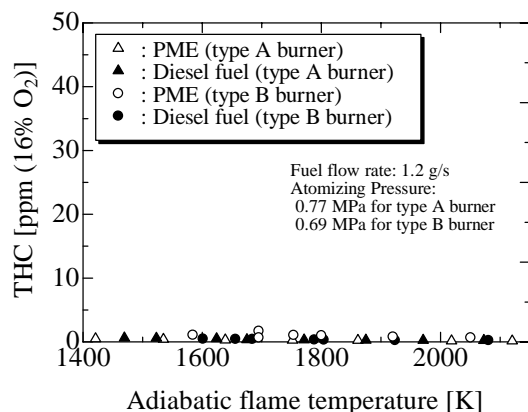Fig. 9 CO emissions as functions of adiabatic flame temperature

Fig. 10 THC emissions as functions of adiabatic flame temperature

V. CONCLUSIONS

In this study, combustion experiments using PME and diesel fuel employing two types of burners at atmospheric pressure were performed. As a result, the employing a burner configuration promoting prevaporzation is promising for PME to reduce NOx emission without soot accumulation, although a caution for the soot accumulation is necessary for diesel fuel. Combustion experiments under the pressurized condition are our future work to examine the effectiveness such a burner for actual gas turbines.

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