

Study on Combustion and Soot Emission of Ethanol or Butanol Blended with Gas Oil in a Direct Injection Diesel Engine

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ABSTRACT

In order to utilize bio-alcohols as the fuel for diesel engines, combustion characteristics of alcohol blended with gas oil were compared between ethanol and n-butanol in a direct injection diesel engine. In the case of the same cetane number between ethanol and butanol blends, the time-history of combustion, in other words, the ignition delay, the diffusion combustion and the combustion duration, coincided almost completely in both blend fuels. However, the smoke density of the butanol blend was smaller than that of the ethanol blend. This result must be caused by difference in soot formation process between ethanol and butanol blends. Thus, it is difficult to predict the trend of the soot emission in combustion of alcohol blends only by using the existing phenomenological model of the soot formation in the combustion of gas oil. In the present study, the concept of the threshold sooting index (TSI), which has been proposed as the sooting tendency in fuel flames, was incorporated into the soot formation model in KIVA-3V code. The soot emission trend obtained in the experiments was clearly shown in the present simulation by considering the characteristic TSI values for each alcohol and gas oil as well as the alcohol blend ratio.

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INTRODUCTION

Alcohols from renewable energy resources are carbon neutral fuel and the oxygenated fuel as well, then, it is a remarkable alternative fuel for not only spark ignition engine but also compression ignition engine. In order to ignite alcohol with low ignitability in compression ignition engines, researchers have proposed several methods, for example blends of alcohol and high ignitability fuel, ignition-assist by using glow plug and alcohol fumigation to ignite premixed air and alcohol by injecting gas oil directly into the combustion chamber. Among these methods, blends of alcohol is one of the most convenient methods of employing alcohol in the diesel engines. Miwa et. al. [1] has investigated the effect of surface activating agents on solubility of ethanol and gas oil, and they also conducted the combustion tests of gasoil with different amounts of ethanol from 0% to 100% by volume. They showed that the engine could operate smoothly without knock by using 60vol% ethanol blend fuel when the compression ratio was 21.2. Mohammadi et. al. [2] investigated the effect of the pilot injection quantity and

timing on the exhaust emissions in combustion of ethanol blended with gas oil by using the common rail injection system which has a lot of flexibility in fuel injection. They showed that by adapting appropriate pilot injection, unburned hydrocarbon emission at low engine load was reduced. Combustion tests on ethanol blended with biodiesels were done by Shudo et. al. [3]. They concluded that the soot reduction effects of ethanol were due to promotion of fuel-air premixing caused by long ignition delay period and the presence of oxygen included in ethanol. However, It has been unclear which factor is dominant. Saitoh et al. [4] focused on dominant factors of auto-ignition phenomenon in spray of normal-alcohols such as methanol, ethanol, propanol, butanol and ethanol/diethyl ether blends from the view point of fuel properties. Lately, combustion tests on butanol blend fuel have been performed by several researchers actively. Yao et al. [5] showed that the n-butanol addition could significantly improve soot and CO emissions without a serious impact on the brake specific fuel consumption and NOx by using diesel fuel with different amounts of butanol from 0%, 5%, 10% to

15% by volume. Rakopoulos et al. [6] have done combustion test in a heavy duty direct injection diesel engine fueled by diesel fuel blended with 8% and 16% butanol. They examined the effect of butanol blends on engine performance and exhaust emissions, and found that the butanol blends resulted in the larger ignition delay owing to the lower cetane number of butanol and the higher premixed combustion. A heat release analysis of the experimentally obtained cylinder pressure was also conducted.

Numerical simulations concerning combustion of alcohol blend fuel in direct injection diesel engines have also been extensively conducted. The combustion in a diesel engine fueled with ethanol blend fuel was simulated by Zhang et al. [7]. Their simulations captured adequately the in-cylinder pressure and emissions of soot and NO_x measured in the combustion tests. They also showed that the fuel mass impinged on combustion chamber wall and wall film mass decrease by increasing in ethanol blend ratio. Kokjohn et al. [8] simulated the combustion of Premixed Charge Compression Ignition (PCCI) fueled with ethanol blend fuel. The detailed chemical kinetic model was applied for modeling the combustion of fuel and they used the phenomenological soot formation and oxidation model for predicting soot emission. The hybrid model was proposed by Kong et al. [9]. It was clarified that the combustion characteristics of PCCI could be captured over the wide range of Low Temperature Combustion (LTC) operations by using these models. Two dimensional, multi-zone combustion model has been developed for direct injection diesel engines by Rakopoulos et al. [10]. The phenomenological soot formation and oxidation model were incorporated in the combustion model for simulating soot emissions. From view point of calculation cost, it is desirable that the characteristics of soot emission in alcohol blended with other fuel is accurately predicted by using phenomenological models.

In the present study, ethanol and n-butanol were blended with gas oil and the blend ratio was determined so that the both blend fuels have almost the same cetane number of 35 and 40 which are calculated by cetane number of each fuel and volumetric blend ratio. This aimed to equalize the fuel-air premixing time of ethanol blend fuel with that of butanol blend fuel. The effect of alcohol blended with gas oil on combustion characteristics and exhaust emission was compared experimentally between ethanol and butanol blends in view points of ignition delay, diffusion combustion, brake thermal efficiency and smoke density. In addition to combustion experiments, the combustion simulation of alcohol blend fuels was conducted. In order to predict precisely the soot emission in both alcohol blends, the concept of the threshold sooting index (TSI) [11], which has been proposed as the sooting tendency in fuel flames, was incorporated into the phenomenological soot formation model in KIVA-3V code with the aim to consider the difference in soot formation process between ethanol and butanol blends.

EXPERIMENTAL APPARATUS AND MEASUREMENTS

The test engine was a single cylinder high-speed naturally aspirated direct injection diesel engine, the type NFD 170-(E) manufactured by YANMAR Co. Ltd. Piston was a conventional one with toroidal cavity. Principal particulars of the test engine are shown in Table 1. The bore is 102 mm, the stroke is 105 mm, and the compression ratio is 17.8. Fuel injection pump is the conventional Bosch type PFR-1AW with 9 mm plunger dia. Figure 1 shows the section of test engine and Fig.2 shows the combustion test system. The suction air pressure at the intake manifold was always adjusted at 0.1013 MPa by using the electric blower. Intake charge temperature was kept constant at $T_{IN} = 40 \pm 0.5$ °C by using electric heater installed upstream of the mixing chamber. The cooled EGR system was installed between the exhaust pipe and the mixing chamber was located at 1,800 mm upstream of the intake manifold. However in the present experiment, the engine was operated without EGR. Air flow rate was measured by using the entrance nozzle installed in the plenum tank located upstream of surge tank. In order to calculate the fuel flow rate, a weight decrement of fuel tank was measured in the specified time interval of ten minutes by the electronic weighing instrument. The combustion tests were carried out at a constant engine speed of $1,200 \pm 5$ rpm, 50 % of the rated speed. Engine load was controlled by the electrical dynamometer measuring the shaft torque and the rotational speed, and it was changed from 20% load ($p_{me}=0.13$ MPa) to 80% load (0.51MPa), which correspond to equivalence ratios of 0.20 and 0.52. The fuel injection timing was set at 13.5 ± 0.5 deg.CA BTDC under engine running condition by adjusting the inclination of fuel injection pump relative to the engine. The test conditions are summarized in Table 2.

Table 1. Principal particulars of test engine

Model type	NFD-170 (E) , Single cylinder, Water-cooled, Four-stroke
Bore and stroke	102 and 105 mm
Compression ratio	17.8
Rated power	11.0 kW / 2400 rpm
Valve timing	Opening Closing
Intake	8 deg.BTDC 40 deg.ABDC
Exhaust	50 deg.BBDC 12 deg.ATDC
Fuel pump	Bosch PFR-1AW with 9 mm plunger
Injection nozzle	4 holes with 0.29 dia. mm
Opening pressure	19.6 MPa

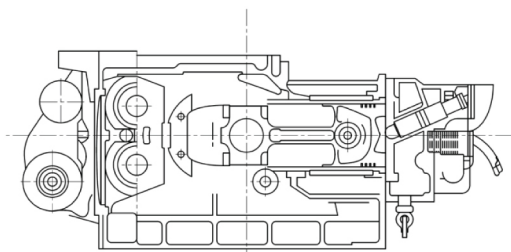


Figure 1. Section of test engine

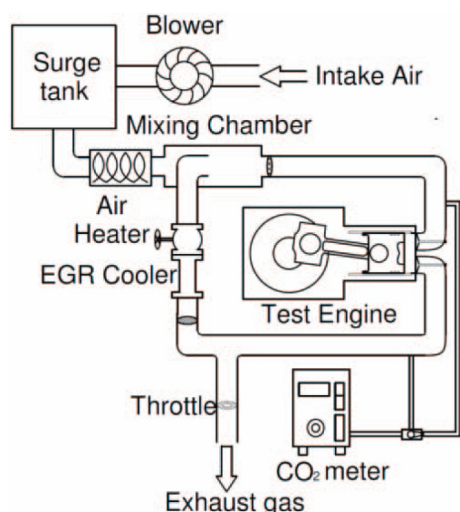


Figure 2. Combustion test system

Table 2. Combustion test condition

Engine speed	1200 rpm
Suction air pressure	0.1013 MPa
Intake air temperature	40 °C
Dynamic injection timing	13.5±0.5 deg.CA BTDC
Mean effective pressure	0.13(20%)-0.51(80%) MPa

Base fuels are low sulfur diesel fuel called gas oil JIS #2 in Japan, ethanol and n-butanol (abbreviated by butanol hereafter). Both alcohols are anhydrous and have the purity of 99% or more. Properties of base fuel are summarized in Table 3. The lower heating value of ethanol 26.8 MJ/kg is as small as 62 % of gas oil and that of butanol 33.1 MJ/kg is 77 % of gas oil. Ethanol has the lowest cetane number, the lowest evaporation temperature and the largest latent heat in the three base fuels. The properties of the tested blend fuels are shown in Table 4.

Table 3. Property of base fuels

Base fuels	Butanol	Ethanol	Gas oil
Cetane number	18	8	55
Lower heating value [MJ/kg]	33.1	26.8	43.1
Density [kg/m ³]	813	788	832
Boiling point [°C]	117	78	190-350
Latent heat of evaporation [kJ/kg]	585	840	250
Oxygen content [wt%]	21.6	34.8	0
Sulfur [wt%]	-	-	0.0008
Stoichiometric air-fuel ratio	11.1	9	14.2

Table 4. Property of blend fuels

Test fuels	Et30	Et40	Bu40	Bu55
Gas oil [vol%]	68	58	60	45
Alcohol [vol%]	29	39	40	55
Octanol [vol%]	3	3	0	0
Lower heating value [MJ/kg]	38.4	36.7	39.1	37.6
Cetane number	40	35	40.8	34.7
Oxygen content [wt%]	10.1	13.5	8.6	11.9

The blend fuel named Et30 consists of 29 % ethanol and 68 % gas oil including with 3 % octanol as a surface activating agent and Et40 consists of 39 % ethanol, 58 % gas oil and 3 % octanol. On the other hand, butanol could be blended with gas oil without surface-active agent. The blend fuel named Bu40 consists of 40 % butanol and 60 % gas oil, and Bu55 consists of 55 % butanol and 45 % gas oil. The cetane number of the blend fuel shown in Table 4 was calculated from the volumetric blend ratio between alcohol and gas oil. The exhaust gas temperature was measured by means of the E-type thermocouple at the exhaust manifold, and temperatures of fuel, air, cooling water and intake charge were measured by the T-type thermocouple. In the exhaust gas analysis, smoke density was measured. Table 5 shows summary of detection principle and accuracy of measurement in the respective equipment. The time-history of in-cylinder pressure and the needle valve lift were measured simultaneously by using the piezoelectric pressure transducer and the gap sensor, and the outputs from the sensors were sampled every one-fourth degree in crank angle by means of

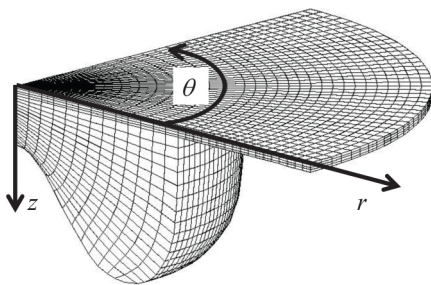
Table 5. Summary of equipments, detection principle and accuracy of measurement

Measuring item	Detection principle	Equipment (maker)	Scale range	Accuracy
Smoke	Bosch type	DSM-10 (BANZAI)	0-10 [Bosch]	±2% Linearity
In-cylinder pressure	Piezoelectric transducer/ charge amplifier	6125B/5011B (KISTLER)	0-250 [bar]	±0.5%FS
Needle valve lift	Gap sensor	PU-03A (AEC)	0-1[mm]	0.3[μm]
Crank angle	Encoder	PP-011A (Ono Sokki)	-	±0.25 deg
Torque	Torque detector/meter	DSTP-10/ DMT-408 (Ono Sokki)	0-100[Nm]	±0.5%FS
Temperature	Thermo-couple	E type/T type (YAMARI)	0-900/0-350[°C]	±1.7/±1.0[°C]
Fuel flow rate	Electronic weighing	EB-32KD (SHIMADZU)	0-32[kg]	±1[g]
Air flow rate	Inlet nozzle pressure drop	ISP-3-50 (SHIBATA)	0-50 [mmAq]	±1%FS

the 4 channel combustion analyzer CB-467 manufactured by Ono Sokki Co. Ltd. The time-history of combustion pressure was the ensemble average sampled over continuous 350 engine cycles. The data were transmitted to the personal computer and recorded on hard disks.

SIMULATION MODEL AND SETUP

3D-CFD code KIVA-3V [12] was modified to simulate the combustion of two component fuels. The evaporation model proposed by Struckmeier et. al. [13] was applied for modeling the evaporation of two component fuels with different evaporation temperature. The combustion was modeled using Characteristic Time Combustion (CTC) model [14]. The SHELL auto-ignition model [15] was used. In the ignition calculation, it was assumed that only the component with high ignitability reacted and it was based on the remark made by Wada et. al. [16]. Gas oil was represented by tetradecane. The Kelvin Helmholtz (KH) - Rayleigh Taylor (RT) breakup model [17] was used for modeling the breakup of droplet. In order to reduce the grid size dependency of the spray model, the Gasjet model [18] was used to model the relative velocity between the droplets and gas phase.

**Figure 3. Computational grid**

The soot formation model of Hiroyasu [19] and the soot oxidation model of Nagle-Strickland-Constable [20] were applied for predicting soot emission. These models were written in the Arrhenius single step form. The rate of change of soot mass during the period of combustion is determined from the soot formation rate and the soot oxidation rate.

$$dM_s/dt = dM_{sf}/dt - dM_{so}/dt.$$

(1)

The soot formation rate is given by

$$dM_{sf}/dt = A_f M_{fv} P^{0.5} \exp(-E_f/(RT)).$$

(2)

where $A_f = 350$, M_{fv} is the fuel vapor mass, P is the pressure, and $E_f = 12,500$ cal/mole. The soot oxidation rate is given by

$$dM_{so}/dt = 6 M_{wc} M_s R_{Total} / (\rho_s D_s).$$

(3)

where M_{wc} is the carbon molecular weight (12 g/mole), ρ_s is the soot density (2 g/cm³), D_s is the soot diameter (3×10^{-6} cm), M_s is the soot mass. R_{Total} is the net reaction rate [21].

In order to represent low sooting tendency of alcohol blend fuel, Equation (2) was modified as follow;

$$dM_{sf}/dt = K_{eff} A_f M_{fv} P^{0.5} \exp(-E_f/(RT)).$$

(4)

$$K_{eff} = TSI_{mix} / TSI_{go}$$

(5)

$$TSI_{mix} = TSI_{go} X_{tet} + TSI_{alc} X_{alc}$$

(6)

where TSI is the Threshold Sooting Index proposed by Calcote and Manos [11] and it correlates well with actual Particulate Matter (PM) emissions [22]. TSI expresses the sooting tendency of a fuel in diffusion flames. It is inversely proportional to the smoke point (SP), which is measured using a standard smoke point lamp for liquid fuels as specified by ASTM D1322-97. $TSI_{go} = 31.23$, $TSI_{eth} = 2.02$, $TSI_{but} = 2.66$. TSI_{go} , TSI_{eth} and TSI_{but} were used as TSI values for gas oil, ethanol and butanol respectively and these values were taken from the literature [22]. TSI_{mix} is for the blend fuel. X_{tet} and X_{alc} are the molecular fraction of tetradecane