

Performance, Combustion and Emission Characteristics of Polyoxymethylene Dimethyl Ethers (PODE₃₋₄)/ Wide Distillation Fuel (WDF) Blends in Premixed Low Temperature Combustion (LTC)

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ABSTRACT

Wide Distillation Fuel (WDF) refers to the fuels with a distillation range from Initial Boiling Point (IBP) of gasoline to Final Boiling Point (FBP) of diesel. Polyoxymethylene Dimethyl Ethers (PODEn) have high oxygen content and cetane number, are promising green additive to diesel fuel. In this paper, WDF was prepared by blending diesel and gasoline at ratio of 1:1, by volume; the mass distribution of oligomers in the PODE₃₋₄ product was 88.9% of PODE₃ and 8.46% of PODE₄. Diesel fuel (Diesel), WDF (G50D50) and WDF (80%)-PODE₃₋₄ (20%) (G40D40P20) were tested in a light-duty single-cylinder diesel engine, combustion characteristic, fuel consumption and exhaust emissions were measured. The results showed that: at idling condition, G40D40P20 has better combustion stability, higher heat release rate, higher thermal efficiency compared with G50D50. At medium load, G40D40P20 can be run with higher EGR to get much lower NOx emissions compared with G50D50; both G40D40P20 and G50D50 maintain very low soot emissions, while diesel has a soot bump during the EGR sweep; G40D40P20 leads to highest thermal efficiency during the EGR sweep.

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INTRODUCTION

Diesel engines, due to the high thermal efficiency and power performance, are widely used in vehicles and engineering machinery. However, with the increasingly strict emission regulation, it's a big challenge for conventional diesel engines to reduce NOx and PM emissions simultaneously to achieve the emission target. To defeat the trade-off between NOx and soot emissions, Low Temperature Combustion (LTC) was proposed [1] in 2001. Due to the high ignitability, low volatility and oxygen-free compositions of diesel fuel, large amount of cold Exhaust Gas Recirculation (EGR) has to be used in LTC to reduce combustion temperature and the combustion occurred in the lower temperature side of the soot formation peninsula in the ϕ -T map. Therefore, Han *et al.* pointed out that EGR has to be further increased to reduce soot emissions although NOx emissions have already been reduced to an acceptable extent [2] in 2011. Significant reduction in combustion efficiency was observed due to the lack of oxygen, while load range was limited to 5-6 bar Indicated Mean Effective Pressure (IMEP).

In order to improve combustion efficiency and extend the load range of LTC, fuel concept called Wide Distillation Fuel (WDF) was proposed in recent years. WDF refers to fuels with a wide distillation range from initial Boiling Point (IBP) of gasoline to Final Boiling Point (FBP) of diesel. WDF contains high volatile components which

are beneficial for atomization and evaporation. Meanwhile, WDF usually has lower ignitability than diesel, so WDF has longer ignition time allowing better air-fuel mixture before ignition. Therefore, WDF has more premixed charge than diesel under the same EGR level. If equivalence ratio is lower than 2, soot formation is minimized regardless the combustion temperature. Low NOx and soot emissions can be achieved by WDF under moderate EGR level. In this combustion mode, emission control does not mainly rely on temperature reduction, air-fuel premixing also plays an essential role. Therefore, it is called premixed-LTC [2,3,4].

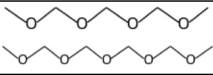
Direct blending of commercial gasoline and diesel is the most convenient method to produce WDF. Until now, some studies have been carried out using commercial gasoline and diesel blends [5,6,7,8,9]. The results have proved that soot emissions of WDF are much lower than those of diesel fuel and not sensitive to EGR variation. NOx and soot emissions can be reduced to low level at moderate EGR. However, the low-emission performance relies on high gasoline proportion, and there were still some limitations in using WDF with high gasoline proportion: Well and Collings [8] reported combustion deterioration at low load when using 50% gasoline/diesel blend fuel in 2007; at medium load, high gasoline addition limits the usage of high EGR because of the combustion stability and NOx emissions become issues [9].

In this paper, PODEn are added into WDF for the first time and expected to improve the shortcoming above and maintain advantages in reducing emissions. PODEn stand for the mixtures of ethers with the formula of $\text{CH}_3\text{O}(\text{CH}_2\text{O})_n\text{CH}_3$ ($n \geq 2$). PODEn have nearly 50% oxygen content and significantly high cetane number, are promising green fuel additives. Among PODEn, PODE₂ does not fulfill the safety concern due to its low flash point [10], and if 'n' is higher than 5, melting points will be too high, and there will be precipitate in diesel/PODEn blend fuel at low temperature. Therefore, only PODE₃, PODE₄, PODE₅ or their mixtures are ideal fuel additives. From 2012 to 2013, studies of using PODE₃₋₅ (mixtures of PODE₃, PODE₄ and PODE₅) as diesel engine fuels or fuel additives have been conducted before, and the results showed that PODE₃₋₅ can reduce PM and soot emissions significantly [11-12]. In this paper, comparative study of WDF and WDF/PODE₃₋₄ blend was made at both idling and medium load. Diesel fuel was also tested as reference fuel. All the experiments in this research were done with a single injection to show the sole effects of fuel properties.

TEST FUELS

Beijing market diesel, gasoline and PODE₃₋₄ were used as the base fuel for fuel blending. The PODE₃₋₄ was synthesized and separated by Yuhuang Company using the technique provided by [13,14,15], which has a mass distribution of PODE₂:PODE₃:PODE₄ = 2.553%:88.9%:8.48%. Three fuels were tested in this study: gasoline and diesel fuel were blended with the ratio of 1:1 by volume as WDF (G50D50), the reason for this blend ratio had been discussed in [4]; PODE₃₋₄ were blended with WDF at blending ratios of 20/80 (G40D40P20) by volume; pure diesel fuel (Diesel) was also tested as reference. Table 1 shows the properties of diesel and gasoline and PODE₃₋₄ used in this study. The boiling point of PODE₃ and PODE₄ is 156 and 202 °C, respectively. Therefore, the volatilities for the three fuels from high to low are gasoline, PODE₃₋₄ and diesel.

Table 1. Properties of diesel, gasoline and PODE₃₋₄

Item	Diesel	Gasoline	PODE ₃₋₄
Chemical formula	C16-C23	C5-C11	
Density [g/mm ³]	0.830	0.753	1.019
Cetane number	56.5	15.1 ^a	78.6 ^a
Lower heating value [MJ/kg]	42.68	42.84	19.05 ^b
Boiling point [°C]	176.0-333.0	35.8-203.1	156.0/202.0
Aromatics [% (m/m)]	28.7	33.2	0.0
C [% (m/m)]	86.45	86.56	43.97
H [% (m/m)]	13.49	13.01	8.78
O [% (m/m)]	0.05	0.43	46.98
S [ppm]	4.3	21.0	0

^a estimated by [16]

^b calculated by adding the heating value of each component

EXPERIMENTAL SETUP AND METHOD

The experiments were performed using a single-cylinder research engine retrofitted from a four-cylinder common-rail compression ignition engine. The engine specifications are listed in Table 2. The engine was equipped with a Delphi seven-hole injector with the cone angle of 156°. Turbocharger was removed from the engine and an external air compressor was used to supply intake air. The engine was controlled by the Electronic Control Unit (ECU) which was transformed to a research module, and the injection parameters such as injection pressure, number of injections and injection timing were set by PC flexibly. Fig. 1 illustrates the schematic of the engine testing system. In-cylinder pressure was measured with a pressure transducer (AVL GH14P) and combustion pressure data were recorded with the resolution of 0.2 °Crank Angle (CA). Combustion analysis was performed based on the average cylinder pressure of 250 consecutive cycles. Heat Release Rate (HRR) was calculated using the following equation:

$$\frac{dQ}{d\theta} = \frac{\gamma}{\gamma-1} p \frac{dV}{d\theta} + \frac{1}{\gamma-1} V \frac{dp}{d\theta}$$

where, γ is the specific heat ratio, V is the instantaneous cylinder volume, and p is the in-cylinder pressure. CA10, CA50, and CA90 are defined as the crank angles at which 10%, 50%, and 90% fuel mass fraction was consumed, respectively. Ignition delay is defined as the crank angle interval between Start of Injection (SOI) and CA10. SOI is the CA position of the electric signal that marks when the injection starts. Combustion duration is defined as the crank angle interval between CA90 and CA10. Gaseous emissions including CO, HC and NOx were measured using an AVL CEBII pollutants analyzer, while soot emissions were measured by AVL 439 Opacimeter.

The Indicated Thermal Efficiency (ITE) was calculated according to the indicated work and the measured fuel flow rate:

$$\eta_i = W_i / (m_f \times Hu_f)$$

where, W_i is the indicated work, m_f is the fuel consumed per cycle, and Hu_f is the fuel lower heating value.

The combustion efficiency was calculated according to the heating value of the CO and HC emissions, and assuming the heating value of HC is equal to the fuel.

$$\eta_c = 1 - (m_{\text{CO}} \times Hu_{\text{CO}} + m_{\text{HC}} \times Hu_{\text{HC}}) / (m_f \times Hu_f)$$

where, m_{CO} and m_{HC} are the mass of the CO and HC, Hu_{CO} and Hu_{HC} are the lower heating value of CO and HC.

Table 2. Engine Specifications

Compression ratio	16.7
Displacement	0.5 L
Bore	83.1 mm
Stroke	92 mm
Connecting rod length	145.8 mm
Swirl ratio	1.7
Injection pressure	40~180 MPa

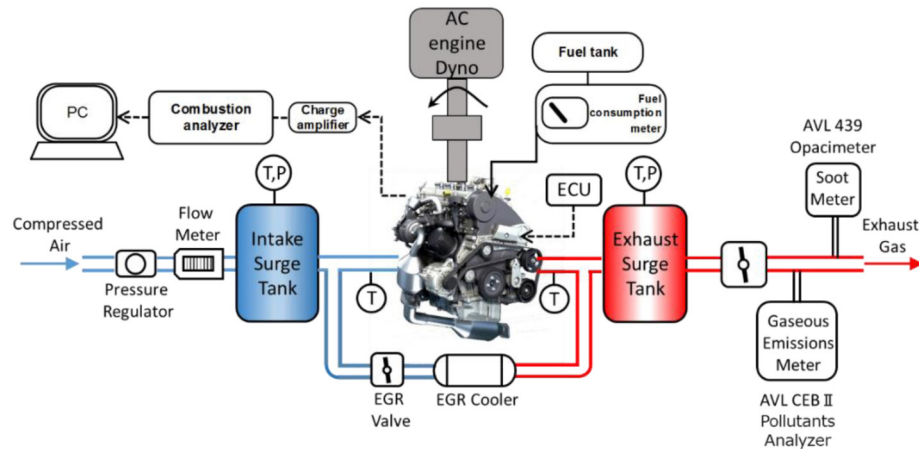


Figure 1. Engine testing system

RESULTS AND DISCUSSION

Injection Timing Effects on Combustion and Emissions at 1200 rpm and 0.22 MPa IMEP, no EGR

The first experiment was conducted at engine idling condition without EGR. The intake pressure was set at 1.0 bar absolute, the injection pressure was 40 MPa and single injection was used. For each fuel, the fuelling rate was fixed to get an IMEP of around 0.22 MPa IMEP at the CA50 of -8°CA BTDC and then the injection timing was swept.

Figure 2a and 2b show ignition delay and CA50 plotted against SOI for the three fuels at 1200 rpm and 0.22 MPa IMEP, respectively. G50D50 has the longest ignition delay, the ignition delay of G40D40P20 is about 5 degrees shorter and diesel fuel has the shortest ignition delay. According to the formula 7 in [16], the estimated cetane numbers of G50D50 and G40D40P20 are 35.8 and 44.4, respectively. For diesel, the cetane number is 56.5. The different cetane numbers mainly cause the different ignition delays. Besides, gasoline in G50D50 and G40D40P20 has high vaporization latent heat, this reduces cylinder gas temperature after fuel injection and further extends the ignition delay. Early and late injection both increase ignition delay at this test condition. At early injection point, cylinder pressure and temperature are too low to support fuel ignition when injection occurs. At late injection point, cylinder temperature decreases rapidly during mixing and chemical reaction processes [17]. For diesel fuel, the ignition delay curve increases slowly when SOI is advanced. Therefore, as SOI is advanced, CA50 of diesel fuel keeps getting earlier, and SOI is stopped being advanced when CA50 arrives at $1.2^{\circ}\text{CA BTDC}$. For G50D50 and G40D40P20 the ignition delay increases rapidly as SOI is advanced at early SOI, so CA50 turns back at SOI of around 21°CA BTDC , as a results, the earliest CA50s which G40D40P20 and G50D50 can reach are -2.5 and $-7.0^{\circ}\text{CA BTDC}$, respectively.

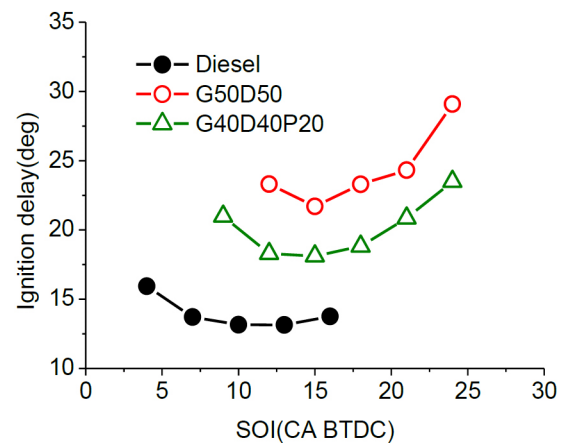


Figure 2a. Ignition delay vs. SOI, three fuels, 1200 rpm and 0.22 MPa IMEP, 1.0 bar absolute intake pressure and 40 MPa injection pressure

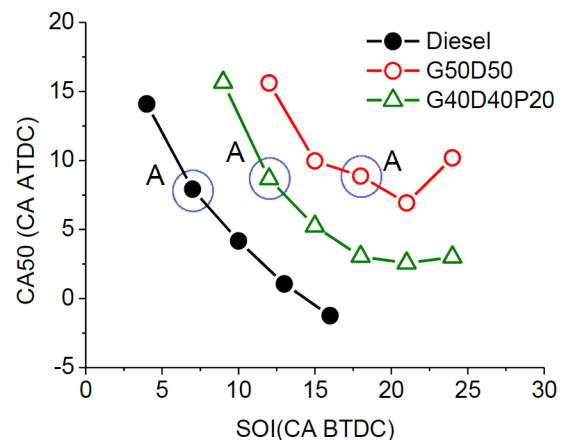


Figure 2b. CA50 vs. SOI, three fuels, 1200 rpm and 0.22 MPa IMEP, 1.0 bar absolute intake pressure and 40 MPa injection pressure

Figure 3 shows the cylinder pressure and HRR at 1200 rpm and 0.22 MPa IMEP, point A in Figure 2b. At point A, all fuels get the same CA50 of -8°CA BTDC . For G50D50 and G40D40P20, cylinder pressure is lower compared to diesel fuel after the injection due to the high latent heat of gasoline vaporization. G50D50 has much lower