Fuel Processing Technology xxx (2016) xxx–xxx

Contents lists available at ScienceDirect

Fuel Processing Technology

journal homepage: www.elsevier.com/locate/fuproc

Research article

Experimental study on spray, combustion and emission characteristics of pine oil/diesel blends in a multi-cylinder diesel engine

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ARTICLE INFO

Article history:
Received 1 June 2016
Received in revised form 14 July 2016
Accepted 18 July 2016
Available online xxxx

Keywords:
Pine oil/diesel blends
Constant volume chamber
Engine
Spray characteristics
Combustion
Emission

ABSTRACT

Pine oil has a high calorific value, which makes it an appropriate biofuel for diesel engines. In this paper, spray characteristics such as spray morphology, spray penetration and spray cone angle of pine oil-diesel blends were investigated under various injection pressures using a constant volume spray chamber. The effects of the injection pressure on the combustion and emission characteristics of pine oil-diesel blends were experimentally investigated in a four-cylinder diesel engine under medium EGR (24.6%). Four different fuels including pure diesel (P0), three blends of pine oil and diesel fuel denoted as P20 (20% pine oil and 80% diesel in volume), P40 (40% pine oil and 60% diesel in volume), and P50 (50% pine oil and 50% diesel in volume) were tested. Our results indicate that, as the injection pressure increases, the spray penetration of blended fuels increase, while the spray cone angle shows slightly change. And with the increasing of injection pressure, the peak values of heat release rate and in-cylinder pressure during the combustion of the four blended fuels increase, the BSFC slightly increase, the emissions of soot, CO and THC decrease, however NOx emissions increase. The effects of the increase of injection pressure on soot emissions of pure diesel are greater than that of blended fuels. When the mixing ratio of pine oil exceeds 40%, the beneficial effects of injection pressure on soot emissions from the combustion of blended fuels are weakened. At the same injection pressure, the BSFC of P20 almost equals the value of P0. As the mixing ratio of pine oil increases, the spray penetration and the spray cone angle of blended fuels increase, which enhanced the atomization process and fuel evaporation. With the increase ratio of pine oil, the peak values of in-cylinder pressure and heat release rate increase, the emissions of NOx, CO and THC increase and soot emissions decrease dramatically. For the combustion of the tested fuels, the number concentration and mass concentration of total PMs can be reduced by increasing the injection pressure or the amount of pine oil in the blends. At an injection pressure of 100 MPa, the total PM number concentration and mass concentration of P50 are respectively lower 86.30% and 96.55% than the values for pure diesel.

1. Introduction

On the concerning of internal engine clean and high efficient combustion, HCCI [1] and LTC show great potential [2]. LTC relies on large amount of exhaust gas [3–5]. The application of LTC could decrease soot and NOx emissions emitted from diesel engines. The results show that with the increasing of EGR, the soot emitted from engine increases at first and then decreases, at medium or high EGR, creating the soot-bump region is created, which is difficult to eliminate [6]. Although further increase the rate of EGR can reduce the smoke emissions, the fuel consumption is deteriorated. Changing the physical and chemical properties by the mixing of the oxygenated fuel and the diesel is an important technical approach to improving the performance and reducing the harmful substance emission from diesel engine with LTC [7].

Aiming at the investigation of oxygenated fuels applied to diesel engine, Rakopoulos et al. [8–10] have investigated the influence on emission and performance of ethanol and n-butanol in a diesel engine. The results show that the diesel blending with oxygenated fuels in diesel can decrease the soot emission significantly, and with oxygenated fuels proportion increase, the soot emission further decrease. Nabi et al. [11] have found that the soot emissions emitted from diesel engine decreased linearly with the increase of fuel oxygen content, and when the oxygen content of >38%, can completely eliminate soot emissions. As an oxygenated fuel, compared with methanol and ethanol, n-butanol has better mutual solubility with diesel, and higher heating value, better lubrication performance. In recent years, many researchers spared no efforts to concentrate on investigating butanol and found that butanol is a promising additive for diesel [12–17]. The n-butanol additive led to a rapid decrease of soot emissions [18,19]. Zunqing Zheng et al. found that diesel blending n-butanol improved soot emission, nevertheless, the maximum pressure rise rate increase [6]. A higher fraction of n-butanol led to the cetane number decrease, the oxygen content

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http://dx.doi.org/10.1016/j.fuproc.2016.07.016
0378-3820/

and the ignition delay increase, which improves the soot and NOx emissions [20]. Additionally, increasing n-butanol content in the fuel reduced the CO emissions, whereas it deteriorated the HC emissions [21]. However, incorporation of n-butanol led to sacrificing some combustion characteristics, due to its low cetane number and heat value. An indication higher specific fuel consumption and lower thermal efficiency have to be compromised with an increase in n-butanol [22].

Unlike other alcohol-based fuels, pine oil a oxygenated fuel, which is obtained from the resin of pine trees, is the renewable fuel, has a high heating value. And the viscosity and boiling point of pine oil is lower comparable to diesel, which can improve the fuel vaporization and atomization after injection. These characteristics make pine oil appropriate for the using in diesel engines [23]. And some reports are available concerning the application of pine oil in different diesel engines [24–27]. R. Vallinayagam et al. [28] have studied the influences of different ratio pine oil/diesel blends on combustion and harmful substance emitted from a single-cylinder, DI diesel engine under different loads. The results show that pine oil significantly decreases the CO, HC, and soot emissions at the condition of full load, and increases the brake thermal efficiency. Nevertheless, the NOx emission increases. However, limited reports are available concerning the influence on the performance and harmful substance emitted from a multi-cylinder diesel engine fuelled with pine oil-diesel blends.

In order to improve fuel evaporation and increase spray penetration to obtain an effective mixing with air, thus enhancing the combustion process, it is very important to control the injection pressure for direct injection diesel engine. With the increasing of injection pressure, have smaller droplets of fuel, and have better atomization, which improving completed combustion and increasing the output power [29–33]. Increase the injection pressure, increase the local Air/Fuel ratio, the in-cylinder peak pressure and NOx emissions [34]. For purpose of improving the emission and performance, the investigation of injection pressure become a hot topic recent years [35]. Whereas the amount of research on pine-oil diesel blends is very limited to our knowledge.

However, the effects on the common emission characteristics and combustion performance of a four-cylinder diesel engine fuelled with pine-oil/diesel with LTC under various injection pressures still have not investigated deeply. Only a few papers have investigated combustion performance of pine-oil/diesel, there are very limited studies on the emission characteristics including the particle emission characteristics of pine-oil/diesel blends, and there are few focuses on the spray characteristics of pine-oil/diesel blends under various injection pressures. The objective of the present work is to study the spray characteristics (spray morphology, spray penetration and spray cone angle) of pine-oil/diesel blends under various injection pressures, and the performance of four blended fuels (blending 0%, 20%, 40%, and 50% by volume fraction of pine oil with pure diesel) in terms of combustion and emission characteristics (including PM size distribution, PM number concentration, and PM mass concentration) in a four-cylinder diesel engine with LTC under different injection pressures.

2. Experimental apparatus and procedures

2.1. Test engine and apparatus

The test was conducted on a four-cylinder diesel engine. The major parameters of the engine are given in Table 1 and the test system is displayed in Fig. 1.

As displayed in Fig. 1. The engine speed was maintained at 1800 rpm (corresponding to the maximum brake torque conditions) in this test. A pressure sensor (Kistler 6052CU20) was used to measure the cylinder pressure. The pressure was recorded with a step of 1 crank angle increment, and 200 consecutive pressure cycles were measured and stored at each operating point, with the recorded values being the mean one. The coefficient of variance (standard deviation divided by the mean value) for all reported quantities was determined <3% [36]. The testing engine was equipped with a common rail fuel injection system with an operating range up to 160 MPa. A measurement, calibration and diagnostic software (INCA, by ETAS) and a Bosch open-type ECU (Engine Control Unit) controlled the common rail fuel injection system. Through ECU, users accommodated parameters like injection fuel mass, injection timing, common-rail pressure. The T/C (turbocharger) engine's intake pressure was controlled at 0.15 MPa and the intake temperature was (30 ± 2) °C. The Exhaust Gas Recirculation rate was controlled by the EGR value, the EGR rate and exhaust gas emission (NOx, CO, HC) were measured by Horiba MEXA 7100DEGR, soot emission was measured using the AVL 415S and the particle emission were measured by the Cambustion DMS500. The uncertainties of the apparatus were displayed in Table 2. The rates of heat release and pressure rise were calculated using the data obtained from the piezo-transducer. The heat release rate was calculated using the following equation [37]:

Table 1
Technical specification of test engine.

Model	Specification
Number of cylinders	4
Cylinder diameter (mm)	85
Number of valves per cylinder	4
Stroke (mm)	88.1
Number of injector nozzle holes	8
Diameter of injector nozzle holes (mm)	0.17
Shape of combustion chamber	ω
Total displacement (L)	1.99
Maximum torque (Nm)	286
Compression ratio	16.5
Rated power (kW)/speed (r/min)	100/4000

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

where γ is the specific heat ratio, V is the instantaneous cylinder volume and p is the in-cylinder pressure.

2.2. Test fuels

Pine oil was blended with diesel, and four blends of 0%, 20%, 40%, and 50% by volume fraction of pine oil were tested, and will be denoted by P0, P20, P40, and P50 according to R. Vallinayagam [26]. The EGR ratio was kept at about 24.6%, based on our previous studies, where we demonstrated that this EGR value was a turning point before a drastic

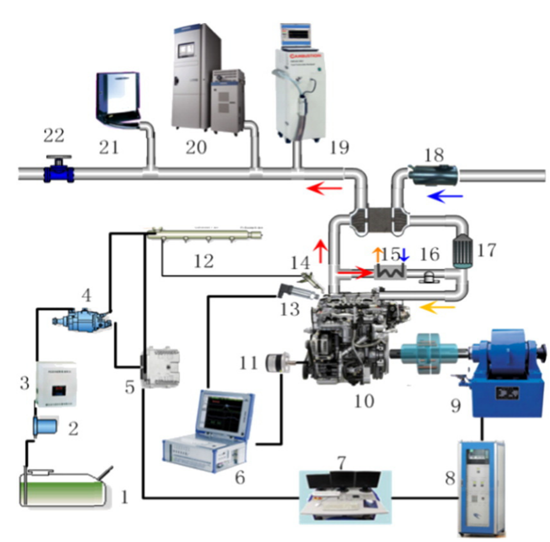

Fig. 1. Schematic diagram of the experimental system.

Please cite this article as: H. Huang, et al., Experimental study on spray, combustion and emission characteristics of pine oil/diesel blends in a multi-cylinder diesel engine, Fuel Processing Technology (2016), http://dx.doi.org/10.1016/j.fuproc.2016.07.016