

Experimental Investigation of Different Blends of Diesel and Gasoline (Dieseline) in a CI Engine

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ABSTRACT

Combustion behaviour and emissions characteristics of different blending ratios of diesel and gasoline fuels (Dieseline) were investigated in a light-duty 4-cylinder compression-ignition (CI) engine operating on partially premixed compression ignition (PPCI) mode. Experiments show that increasing volatility and reducing cetane number of fuels can help promote PPCI and consequently reduce particulate matter (PM) emissions while oxides of nitrogen (NOx) emissions reduction depends on the engine load. Three different blends, 0% (G0), 20% (G20) and 50% (G50) of gasoline mixed with diesel by volume, were studied and results were compared to the diesel-baseline with the same combustion phasing for all experiments. Engine speed was fixed at 1800rpm, while the engine load was varied from 1.38 to 7.85 bar BMEP with the exhaust gas recirculation (EGR) application. Results show that, compared to the diesel baseline, the total particle number concentration of G50 was reduced by up to 50% and 90% and count median diameter (CMD) was reduced by 25% and 75% at medium and low loads respectively. The G50 blend had the lowest smoke emission level (0.5 FSN) for all tested load conditions. NOx emissions decreased by 50% at low load and increased by 20% at medium load for G50 compared with the diesel. Compared to diesel, Dieseline led to lower in-cylinder pressure rise and heat release rate peaks at low loads and it prolonged the combustion delay by up to 7 angles compared to the diesel-baseline condition. Dieseline proved to be an attractive fuel for improving the premixed combustion in CI engines.

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INTRODUCTION

Diesel engines have high thermal efficiency and they can provide high torque at low operating speeds; however, they suffer from high engine-out oxides of nitrogen (NOx) and soot emissions. Due to the mixing controlled nature of combustion in diesel engines, there is a trade-off between particulate and NOx formation inside the cylinder which are eventually emitted from the engine. Legislative controls on NOx and soot emissions are becoming increasingly stringent and diesel engines have to be equipped with high pressure common rail injection systems and costly after-treatment equipment to meet the exhaust legislation standard. On the other hand, harmful exhaust emissions from spark-ignition (SI) gasoline engines can be reduced by the usage of cost-effective three way catalyst; however, the thermal efficiency of gasoline engines is lower than diesel engines.

Extensive efforts have been focused on increasing the thermal efficiency of gasoline engines by using homogeneous charge compression ignition (HCCI) combustion [1, 2, 3]. The main challenges of the gasoline HCCI combustion are the ignition

timing control and its limited operation range (misfire at low loads and rapid combustion at high loads). Partially premixed compression ignition (PPCI) combustion mode has been applied to diesel engines to reduce harmful emissions, particularly NOx and particulate matter (PM) emissions [4, 5]. In PPCI engines, combustion takes place after the end of injection while the ignition is still controlled by the injection timing, just as in conventional diesel engines. Noehre et al. [6] showed that since the diesel fuel has low volatility and short ignition delay, special techniques are required to prolong the ignition delay and consequently improve the premixing process. Increasing cooled exhaust gas recirculation (EGR) and reducing compression ratio are the most commonly used techniques, which however reduce engine thermal efficiency [6]. Extending PPCI combustion load range to higher loads is another challenge, because increased EGR replaces some of the intake air which makes the overall air fuel mixture too rich. Currently, PPCI combustion mode is only possible at low loads with the conventional diesel fuel.

Implementation of PPCI combustion requires a suitable fuel with the right volatility and auto ignition characteristics. It should be noted that a proper EGR ratio is also necessary to reduce the combustion temperature and thus successfully achieving PPCI combustion in a CI engine. Pesant et al. [7] suggested that the diesel fuel is not favourable for premixed combustion due to its ease of ignition (high cetane number) and low volatility. Many researchers are investigating the possibility of applying gasoline-like fuels in PPCI engines. Kalghatgi et al. [8] successfully tested a gasoline fuelled PPCI engine and showed that even at high loads of 15.95 bar indicated mean effective pressure (IMEP), near-zero smoke emissions and indicated specific fuel consumption (ISFC) of 179 g/kW.h can be obtained. Ciatti et al. [9] developed low-octane gasoline fuel split injection strategies for low to high load PPCI combustion mode. Manente et al. [10] tested ethanol and gasoline in a highly boosted PPCI engine. Splitter et al. [11] applied the concept of dual fuel reactivity controlled compression ignition (RCCI) to the premixed combustion and an indicated thermal efficiency of 56% was demonstrated in a single cylinder heavy duty CI engine [11]. Curran et al. [12] implement the RCCI concept into a light duty multi-cylinder engine.

At the University of Birmingham, researchers investigated the idea of altering fuel characteristics through blending gasoline with diesel and the resulting fuel was named 'Dieseline' [13, 14, 15, 16]. Two of the publications from the present authors [14, 16] focus on the different fuel injection strategies, i.e. pilot-main and also split-injection, for the Dieseline fuel and have been compared with the conventional diesel. By studying advanced injection timings and high EGR ratios [15], it was found that the Dieseline fuel with 50% gasoline ratio (G50) facilitated the application of PPCI concept successfully. Other researchers, e.g. Weall and Collings [17] used Dieseline fuel to obtain low emissions from a PPCI engine operating up to 7 bar brake mean effective pressure (BMEP). Han et al. [18] found that the intake boost can shift the entire Dieseline PPCI operation to higher loads and the increase of EGR can extend it to lighter loads. The results show that the dependence of soot reduction on high injection pressure was reduced by using Dieseline fuel [19]. Muether et al. [20] observed that, for premixed combustion, the effect of fuel properties, particularly on thermal efficiency and noise, can be distinguished by maintaining a constant centre of combustion (AHR50) when testing different fuels.

In order to further extend the already existing knowledge on different mechanisms for achieving Dieseline PPCI combustion and also improve the limited knowledge of single-late-injection strategy, the present research investigates the effect of low and higher blending ratios of gasoline for the Dieseline fuel on the combustion and emission performance using conventional calibration strategies and fixing the combustion phase (AHR50). The calibration strategies are very close to what are used for the base diesel as oppose to what have been

previously tried by the authors with advanced injection strategies and varying AHR50. The engine load was varied from 1.38 bar to 7.85 bar BMEP. The AHR50 of each blend were adjusted to be the same as base diesel at each load, which was calibrated for a compromise of emission (Euro IV), efficiency and low pressure rise rate. It is expected that the findings from this investigation provide additional understanding of Dieseline fuel combustion and emission characteristics in a CI engine.

EXPERIMENTAL SYSTEM AND METHODS

Experimental System Setup

The engine used for the experiments was an in-line 4-cylinder Ford engine equipped with a modern common rail fuel injection system, a variable geometry turbocharger (VGT) and double overhead camshaft (DOHC). Its EGR cooler was modified to use conditioned water fixed at 20 °C. The bore, stroke and capacity of the engine are 86mm, 94.6mm and 2198cc, respectively and its compression ratio was maintained the same as the original production setup: 16.6. A schematic of the experimental system setup is shown in Figure 1.

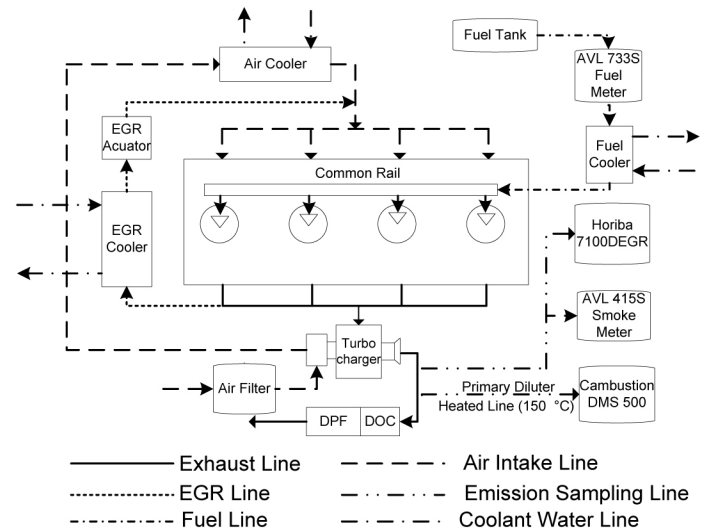


Figure 1. Schematic of the 4-cylinder Ford engine set-up and diagnostics

The in-cylinder pressure was measured with an air-cooled Kistler quartz pressure transducer and acquired by a National Instrument PCI6251 card at every one crank angle degree (CAD). All in-cylinder pressure plots were averaged for 50 complete engine cycles. The heat release rate was calculated from the in-cylinder pressure using the following equation [21]:

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

Where γ is the ratio of specific heats and its value was assumed constant at 1.32. V and p are the instantaneous cylinder volume and in-cylinder pressure respectively.

Several parameters, including start of combustion (SOC), combustion duration and combustion delay, were used to evaluate the combustion performance and their definitions are given as follows. The SOC was identified as the crank angle when the heat release rate curve went up from a negative value to a positive value [22]. The crank angle differences between SOC and 95% mass fraction burn (MFB 95) was used to determine combustion duration. Combustion delay was defined as the crank angle difference between SOI and AHR50.

It was shown by Kalghatgi et al. [23, 24] that combustion delay is a good parameter for understanding the effects of fuel on combustion performances of CI engines. Higher combustion delay means more premixed combustion.

The EGR ratio as well as NO_x, THC, CO and O₂ were measured using a Horiba MEXA-7100-DEGR exhaust gas analyser and the results were averaged over 60 measurements. According to the product's specifications, the errors are below $\pm 1\%$ full scale (FS). The smoke levels were measured using an AVL 415S smoke meter and the error is below 0.005 FSN plus 3% of measurement value. The particulate number and size distribution were measured using a Cambustion DMS500 equipped with a heated primary diluter. The gaseous emissions and particle total concentrations were normalized and are presented in terms of brake specific emissions. The fuel consumption was measured with an AVL 733s dynamic fuel meter equipped with an AVL 752-60 fuel cooler. Results were averaged for three measurements and every measurement was based on the fuel consumption for 2 minutes. The instrument error of the fuel meter was from 0.20% to 0.12% of measured values depending on the fuel consumption. The density and lower heating value differences of the tested fuels were considered for brake thermal efficiency calculations. All of the data were logged after the engine was stabilised and each test was repeated twice and the results presented here are the average of repeats to minimize uncertainty.

Studied Fuels and Methods

Three types of blends were used in the experiments, namely: neat diesel or 0% (G0), 20% (G20) and 50% (G50) gasoline blended with diesel by volume. The neat diesel was the European standard diesel (EN590) and the blended gasoline was 95 octane gasoline (ULG95) with sufficient lubricity additive. The cetane number of the blended fuels were taken as proportionate to their blending ratio and the cetane number of gasoline was assumed to be 15 [17]. Some of the fuel properties are listed in Table 1. The boiling point of diesel ranges from 160 to 340 °C, while the boiling point of gasoline ranges from 40 to 160 °C. Blended fuels are expected to have wide boiling ranges and the detailed data were presented by Weall and Collings [17].

Table 1. Fuel Properties (~ means estimated value)

Test Fuel	Diesel	G20	G50	Gasoline
Cetane Number	51	~45	~35	~15
Density (g/ml at 15 °C)	0.84	0.82	0.79	0.73
Lower Heating Value (MJ/kg)	42.5	~42.7	~43.2	44.0

Five engine operating conditions were selected from the original engine calibration map which was transformed from respective operation points of the new European driving cycle (NEDC) at the studied engine speed. The engine speed was fixed at 1800 rpm and the loads were 1.38, 2.95, 4.27, 5.2 and 7.85 bar BMEP. The fluctuations of the engine speed was below ± 5 rpm and it was ± 0.02 bar BMEP for the load. During the tests, the base diesel was firstly tested using the original engine strategy, with a quantity of 1.5 mm³/cycle pilot injection starting at around 20 CAD before top dead centre (BTDC). The EGR ratio of the base diesel was different at varied loads. It was 42%, 30%, 23%, 21%, 18% from the lowest to the highest loads. The single injection diesel, G20 and G50 tests were carried out with their EGR ratio and AHR50 adjusted to be the same as the base diesel (by changing the injection timing), which was optimised for the efficiency, noise and emissions by the engine manufacturer.

Table 2. Summary of Engine Operating Conditions

BMEP	Boost Pressure	Charge Temperature	Rail Pressure	AFR	EGR
Bar	kPa (abs)	DegC	Bar	Ratio	Ratio
1.38	108	65	550	35.6	42
2.95	112	61	710	28.4	27
4.27	123	59	760	25.2	23
5.2	130	61	840	23.7	18
7.85	150	69	1000	19.3	14

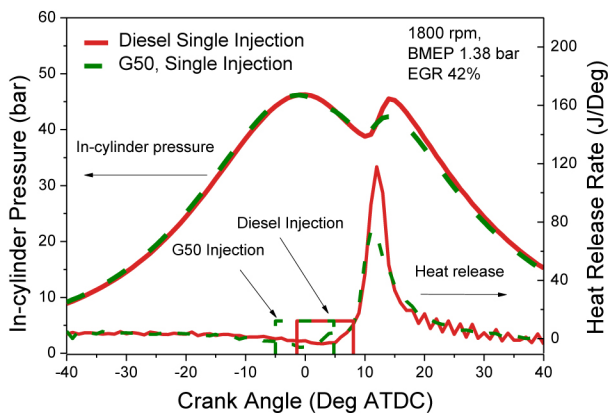
RESULTS AND DISCUSSION

Heat Release Rate Analysis

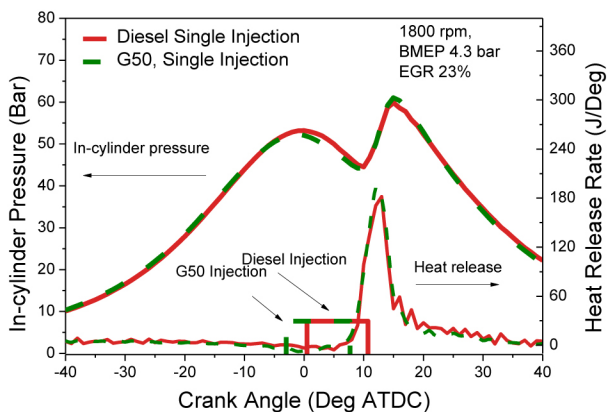
The in-cylinder pressure, heat release rate and injection signals (given by the engine control unit (ECU)) of the single injection diesel and G50 blend are plotted versus CAD for three engine loads in Figure 2. The injection duration was calculated from the injection pulse width and was compensated by the injector closing and opening delay, which was given by the ECU. The SOI has also been compensated by the injector lift opening delay. AHR50 was held constant for all fuels by choosing the appropriate SOI. At the low load of 1.38 bar BMEP, the SOC for both diesel and G50 blend were after the end of the injection event which indicates that the engine combustion was in PPCI mode. Two factors contributed to the separation between the fuel injection and the start of combustion at low load: usage of 40% EGR, and short injection duration. It can be seen in Figure 2 (a) that G50 blend had lower heat release rate peak compared to the diesel fuel. This might be due to the partially over-mixing of G50 with air. G50, with a longer combustion delay (shown in Figure 3) and higher volatility than diesel, was better premixed with air before combustion started. When the global mixture strength is lean,

as is the case at this low load condition (air excess ratio was 2.2), more locally burning mixture packets will be near the global mixture strength and hence over-lean. The partially over-lean mixing increased the combustion duration and reduced the pressure rise rate which will be shown later. The lower boiling range and cooling factor of gasoline in diesel also play a role in reducing peak heat release rate of the G50 at 1.38 bar BMEP engine condition (enthalpy of vaporization is 305 kJ/kg for gasoline and 270 kJ/kg for diesel).

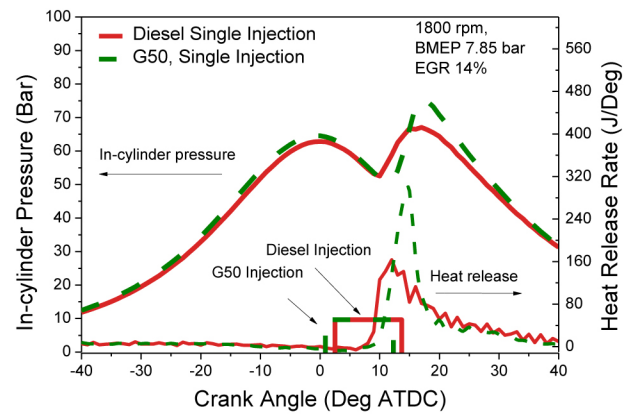
As shown in Figure 2 (b), the end of injection for diesel and G50 were closer to the start of combustion at the medium studied load of 4.27 bar BMEP compared to the 1.38 bar BMEP. The heat release rate curves for both diesel and G50 show a single peak and they nearly coincide. As the engine load increased to 7.85 bar BMEP, it is observed in Figure 2 (c) that diesel combustion process was partially overlapped with the fuel injection event, thus indicating long combustion duration and a low heat release peak. In contrast to diesel, injection event of G50 was separated from the combustion process and a relatively high heat release rate peak appeared. That was supposedly because a higher fraction of the local air/G50 mixture packets was close to the global equivalence ratio (λ 1.28), which was nearly stoichiometric, due to the improved mixing. Similar trend were also observed by Kalghatgi [23]. The heat release rate of G20 is similar to G50 at different loads, thus they are not presented.



(a)



(b)

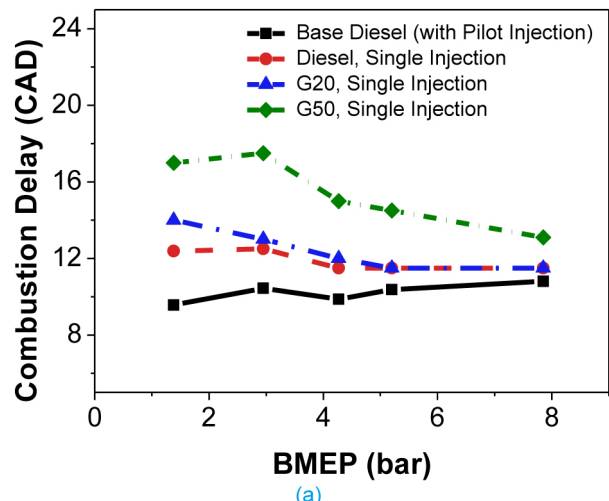


(c)

Figure 2. (cont.) The in-cylinder pressure and heat release rate of the single injection diesel and G50 versus crank angle: (a) 1.38 bar BMEP, (b) 4.27 bar BMEP, (c) 7.85 bar BMEP.

Combustion Performance

Combustion delay for different Dieseline blends at various engine loads are illustrated in Figure 3a. The figure shows that the combustion delay increased with higher gasoline ratio in the blend. At the lowest studied load, the combustion delay for G50 was 5 CA degrees longer than the single injection diesel fuel. The reason is that gasoline is much more resistant to auto-ignition and G50 has lower cetane number than diesel. As the load increased, the combustion delay of both the neat diesel and Dieseline decreased since the EGR ratio decreased and the global equivalence ratio increased. Another possible reason for that is the cylinder wall became much hotter with the increase of engine load and thus assisted the fuel auto-ignition. At the highest load tested, the difference of combustion delay between neat diesel and G50 became much smaller (by 2 CAD). For the diesel fuel with a pilot injection, the pilot combustion heated up the in-cylinder charge before the main injection occurred and combustion delay was lower than for the single injection case.



(a)

Figure 3. Combustion performances of the base diesel, single injection diesel, G20 and G50 versus BMEP: (a) Combustion delay, (b) coefficient of variation (COV) of IMEP.

Figure 2.