

# Investigation into Light Duty Dieseline Fuelled Partially-Premixed Compression Ignition Engine

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## ABSTRACT

Conventional diesel fuelled Partially-Premixed Compression Ignition (PPCI) engines have been investigated by many researchers previously. However, the ease of ignition and difficulty of vaporization of diesel fuel make it imperfect for PPCI combustion.

In this study, Dieseline (blending of diesel and gasoline) was looked into as the Partially-Premixed Compression Ignition fuel for its combination of two fuel properties, ignition-delay-increasing characteristics and higher volatility, which make it more suitable for PPCI combustion compared to neat diesel. A series of tests were carried out on a Euro IV light-duty common-rail diesel engine, and different engine modes, from low speed/load to middle speed/load were all tested, under which fuel blend ratios, EGR rates, injection timings and quantities were varied. The emissions, fuel consumption and combustion stability of this dieseline-fuelled PPCI combustion were all investigated. The results showed that dieseline had great advantages as a PPCI combustion fuel in terms of emission reduction. This was particularly significant at high-speed engine mode. It was also found that with a blend of 50% gasoline in diesel, the particle numbers total concentration could be reduced by 90% while low NO<sub>x</sub> level and high brake fuel conversion efficiency (around 30%) were maintained at all the loads tested.

## INTRODUCTION

The diesel engine is widely used in commercial vehicles because of its high fuel efficiency and the high power it can offer. However, faced with stringent regulatory requirements for the reduction of NO<sub>x</sub>(NO+NO<sub>2</sub>) and particulate matter, it is critical for researchers and engineers to re-design the combustion mode of diesel engines to produce low emissions and maintain high efficiency. One potential way to reduce diesel engine emissions is to change the conventional diffusive combustion to highly premixed combustion or so called PCCI combustion [1].

Generally, there are two ways to achieve diesel HCCI combustion, either internal homogenous formation or external mixture preparation. For the former, early injection (usually more than 100 degree BTDC crank angle) and narrow injection angle which is used to avoid wall impingement are applied [2, 3]. In the latter, the fuel is injected into the intake system and intake air heating is usually required [4, 5]. However, for both of the two formation strategies, the combustion phase control and operating window widening are still challenges. To overcome the challenge, different solutions are used by researchers worldwide. Hasegawa and Yanagihara used two injections in one complete engine cycle, one injection at a very early stage to form a lean mixture and the other injection near TDC performing as a combustion trigger [6, 7]. More researchers looked into Partially Premixed Compression Ignition (PPCI) in which the Start of Combustion (SOC) is controlled by the

injection event. In PPCI combustion, high injection pressure is used to improve the mixing process, and high EGR ratio is used for reduction of  $\text{NO}_x$ . An increase in ignition delay and a moderate advance of the injection event (usually with SOI at 10 to 30 degree BTDC crank angle) are applied [8]. With the combination of these strategies, the combustion can be separated from the injection event while a link between them still exists.

A very important factor that affects the emissions and performance of PPCI engine is the characteristics of the fuel. It is suggested in [9] that low cetane number, highly volatility fuel is ideal for PPCI combustion. Kalghatgi investigated the possibility of gasoline fuelled PPCI engine in [10] and the results show that almost zero smoke emissions can be attained even at high loads of 15.95 bar IMEP. An investigation at University of Birmingham was based on the idea of designing the fuel characteristics through blending gasoline with diesel [11], and the techniques has been named 'Dieseline'. Adam Weall [12] examined a dieseline fuelled PPCI engine with high injection pressure.

Following the work of [11] and [12], this paper concentrates on the proper injection strategies for dieseline fuelled PPCI combustion under moderate injection pressure.

## EXPERIMENTAL SYSTEM AND SETUP

### TEST ENGINE

The engine used in this test is a Ford Euro IV standard 4-cylinder production engine equipped with common rail, variable nozzle turbocharger (VGT), intercooled EGR and dual camshaft. A general sketch map of the test engine is shown on [Figure 1](#) and the specification of the engine is shown in [Table 1](#).

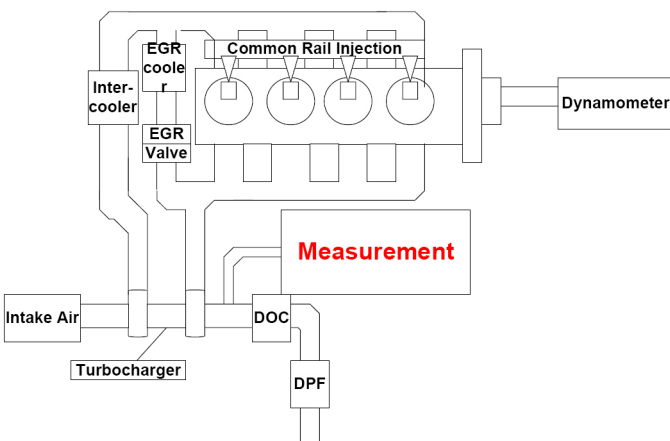


**Figure 1. Experimental Set-up**

**Table 1. Engine Specification**

|                   |                                     |
|-------------------|-------------------------------------|
| Bore              | 86 mm                               |
| Stroke            | 94.6 mm                             |
| Compression Ratio | 16.6                                |
| Engine Capacity   | 2198 cc                             |
| Max Power         | 96 KW ( $\pm 5\%$ )@3500 rpm        |
| Max Torque        | 310.0 NM( $\pm 5\%$ )@1600-2500 rpm |
| Injector type     | Common Rail, Direct Injection       |

In-cylinder pressure was measured with Kistler quartz pressure transducer and acquired by a National Instrument PCI6251 card every 1 crank angle degree. The heat release rate was calculated from the in-cylinder pressure based on the equation in [13]. All of the in-cylinder pressure plots shown on this paper are averaged for 50 complete engine cycles.

The EGR ratio as well as  $\text{NO}_x$ , THC, CO and  $\text{O}_2$  were measured by Horiba MEXA-7100. The particle number and size distribution was measured by Scanning Mobility Particle Sizer (SMPS, Model 3936) from TSI Inc. It consist of a Model 3080 Electrostatic Classifier (EC), a Model 3775 Condensation Particle Counter (CPC) and two Differential Mobility Analyzers (DMAs). The diluter used was a 379020A Model from TSI and the dilution ratio was set to 55:1 for all the tests.

### TEST FUELS

Three types of fuel are used in the test. They are European standard diesel (Specification EN590), two blends of standard diesel and 95 octane gasoline ULG95 with sufficient lubricant additive. The blending ratios are 70:30 and 50:50(diesel: gasoline) by volume and the blend fuel are referred to G30 and G50. The blend fuels are expected to possess lower cetane number and higher vitality. More detail about the blend fuel properties can be found in [12].

### TEST METHODS

The test is divided into two stages. In the first stage, neat diesel, blend fuel G30 and G50 are tested with different total injection quantity and different speed. The total injection quantity ranges from 13  $\text{mm}^3$  per engine cycle to 25  $\text{mm}^3$  per engine cycle. The engine speed is maintained at 1800 rpm and 2500 rpm. For the fuel neat diesel and G30, all of the injection parameters are the same. However, for the fuel G50, the combustion would become very unstable if the injection parameters are maintained. This is because the fuel G50 owns lower cetane number and the combustion phase would be very late and unstable when the engine is run with original engine calibration file. So the injection time of blend fuel G50 is a little advanced to keep the combustion stable and the rest of the injection parameters are the same as neat diesel and G30. All of the injection parameters and engine load for the three types of fuel are shown in [Appendix Table 2, 3, 4](#).

The second stage of the test focuses on investigating the combustion and emission of blend fuel G50 with different injection parameters. The injection parameters include start of main injection timing (SOI), EGR ratio and pilot injection quantity. At this stage, the engine speed was fixed at 1800 rpm and the total injection quantity was fixed at 18 mm<sup>3</sup> per engine cycle (the IMEP is about 4.2 bar for this fuel quantity).

It has to be mentioned that the pilot injection exists in all of the tests. If not specially emphasized, the pilot injection quantity is 1.5 mm<sup>3</sup>/engine cycle.

Fuel injection quantity control is according to ECU calibration software ATI. The accuracy of ECU reading was tested on [12]. The brake fuel conversion efficiency (defined as the output brake work over the input fuel energy) calculation is based on the equation on [13], the lower heating value and density difference between diesel and the blend fuel was seriously considered and the corrected brake fuel conversion efficiency are presented in this paper.

## RESULTS AND DISCUSSION

### IMEP SWEEP FOR DIESELINE

In this section, the total injection quantity per engine cycle ranges from 13 mm<sup>3</sup> to 25 mm<sup>3</sup>. The range of engine load is from 2.9 bar IMEP to 6.2 bar IMEP. Higher engine loads were not tested with the blend fuel for safety operation consideration, however this doesn't mean the maximum allowed engine load condition is 6.2 bar IMEP for the blend fuel. The operation window of the different blends will be investigated in the future.

Figure 2 shows the in-cylinder pressure and heat release rate for diesel, blend fuel G30 and G50 when the engine is operated at 1800 rpm and 18 mm<sup>3</sup>/cycle total fuel quantity (for diesel and G30, just the total fuel quantity is fixed and the other parameters are controlled by the ECU, for G50, the injection time is a little advanced to keep the combustion stable).

Concerning the diesel and G30, it is interesting to see that the pilot combustion of G30 is a little delayed while their main combustion phase stays virtually the same. The delayed pilot combustion of G30 is because it has a lower cetane number compared to diesel which has been proven by other researchers [12]. A probable reason for the same main combustion phase of diesel and G30 is that the delayed pilot combustion of G30 advances its main combustion which reduces the effect of its lower cetane number on longer ignition delay. This kind of phenomena is also observed for the other total fuel quantity and engine speed.

Focusing on the blend fuel G50, it is shown in Figure 2 that the heat release is very rapid and the combustion duration is much reduced compared to the diesel and G30. It is also noticed that no pilot combustion is observed for G50. The absence of pilot combustion would lengthen the main combustion ignition delay and change the diffusion combustion to partially premixed combustion. The reason for the absence of pilot combustion may be that the 1.5 mm<sup>3</sup>/cycle pilot injection quantity is too small to be compressed ignited for G50. Further investigation into the pilot injection of blend fuel G50 would be in later part of this paper.

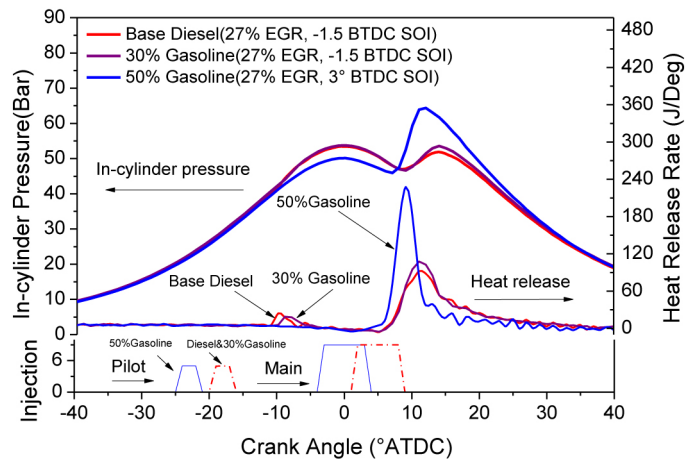


**Figure 2. Pressure and Heat release rate, diesel, 30% gasoline and 50% gasoline, fuel quantity 18 mm<sup>3</sup>/stroke, 1800 rpm, 600 bar rail pressure.**

Concerning the engine performance, it can be seen from Figure 3 that brake fuel conversion efficiency is higher for higher portion of gasoline when the engine speed is 1800 rpm and fuel quantity is no more than 25 mm<sup>3</sup>/cycle. For the 18 mm<sup>3</sup>/cycle fuel quantity (point A), Brake Fuel Conversion Efficiency is increased from 28.06% (diesel) to 31.90% (50% gasoline), which means almost 4% increase of brake fuel conversion efficiency is achieved at this point. There are two reasons for the increase; one is that the combustion of fuel G50 is much more rapid than the others (more rapid combustion means higher portion of constant-volume combustion which increases fuel conversion efficiency with a given compression ratio). The other one is that the combustion phase of G50 is slightly advanced. At higher load with 6.3 bar IMEP (fuel quantity 25 mm<sup>3</sup>/cycle) and higher speed, fuel conversion efficiency of the blend fuel is similar with diesel. That means the gasoline can be more efficiently used in CI engine rather than SI engine by blending it with diesel. The disadvantage of the dieseline fuelled engine is also very obvious: the max pressure rise rate is much higher than normal diesel, which means the engine would be noisier. The reason for this is the rapid partially premixed combustion as discussed in Figure 2.

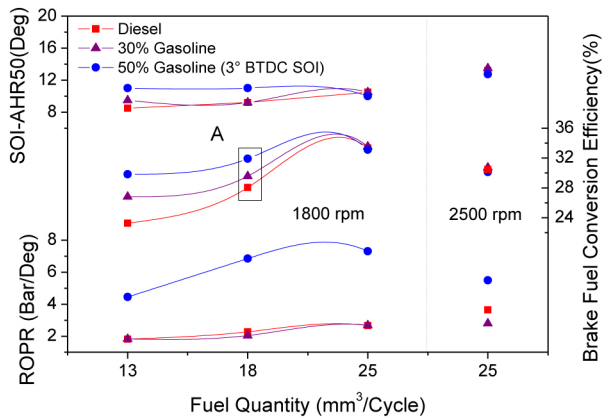


**Figure 3. ROPR, Brake Fuel Conversion Efficiency and Ignition delay (SOI-AHR50), diesel, 30% gasoline and 50% gasoline against different fuel quantity, IMEP ranges from 2.9 bar to 6.2 bar.**

Figure 4 shows the regular emissions of the three different kinds of fuel. Up to 25mm<sup>3</sup>/cycle fuel quantity, blend fuel G50 produces much more THC and CO, which is caused by the lower local temperature in much more premixed combustion. For higher fuel quantity, the difference is small. Clearly, NO<sub>x</sub> emissions depend on the combustion temperature and combustion phase. It is shown in Figure 4 that there is no difference between diesel and 30% gasoline on NO<sub>x</sub> emission while 50% gasoline exhibits higher NO<sub>x</sub> emission for all the different total fuel quantity. That can be explained by Figure 2, which shows that the combustion phase of diesel and 30% gasoline are the same while it is a little advanced for 50% gasoline. It has been discussed in [14] that a reduction of NO<sub>x</sub> emission can be seen in PPCI combustion when the ignition delay is sufficiently high (the local fuel/air mixture is sufficiently lean during combustion).

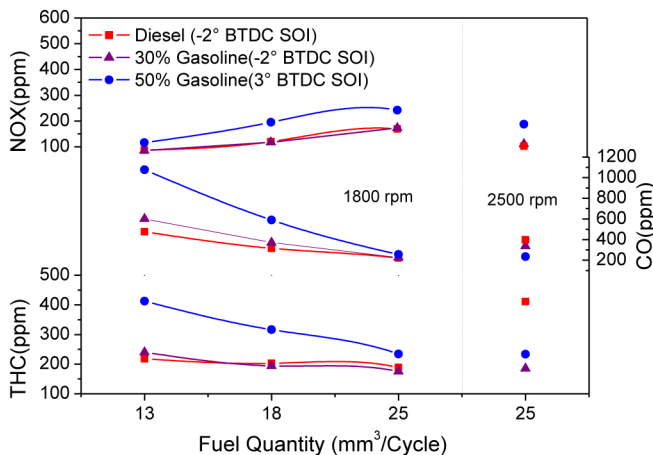
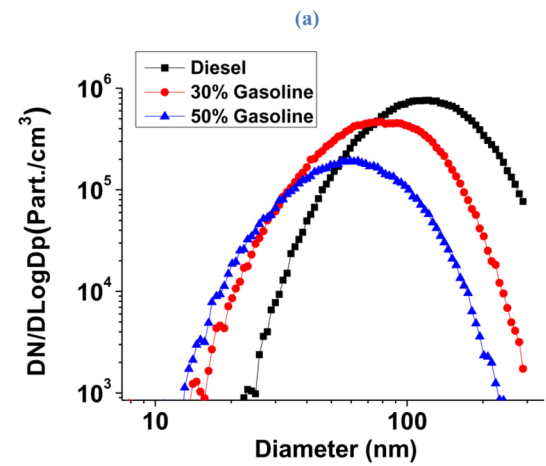
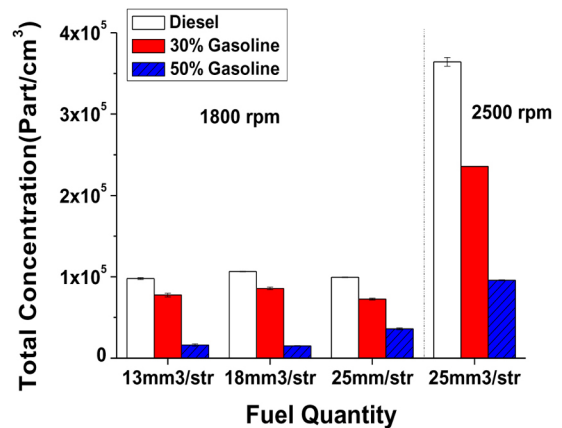


**Figure 4. THC, CO and NO<sub>x</sub> emissions, diesel, 30% gasoline and 50% gasoline.**

Concerning the future regulations for particle matter, the particle size and distribution have been investigated and the results are shown in Figure 5. As expected, due to the high volatility of the blend fuel G50, its particle number total concentration and the mean diameter are greatly reduced. At a high speed of 2500 rpm, the reduction is much clearer.

A short conclusion can be drawn on the effect of diesel fuel on the diesel engine on emissions and performance. The benefits of diesel fuel are the low particle number emissions for all loads tested and high fuel efficiency up to 5 bar IMEP. The disadvantage is the high pressure rise rate caused by the partially premixed combustion. Maximum emissions of THC and CO for the blend fuel are 400 ppm and 1000 ppm. However, they can be removed by a catalyst at lower cost. NO<sub>x</sub> emissions from partially premixed combustion of diesel are a challenging issue requiring special care as shown later. Further investigation below will focus on the 50% gasoline with variation of injection timing, EGR and pilot injection quantity.