

# Premixed Low Temperature Combustion of Biodiesel and Blends in a High Speed Compression Ignition Engine

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## ABSTRACT

The effects of combining premixed, low temperature combustion (LTC) with biodiesel are relatively unknown to this point. This mode allows simultaneously low soot and NO<sub>x</sub> emissions by using high rates of EGR and increasing ignition delay. This paper compares engine performance and emissions of neat, soy-based methyl ester biodiesel (B100), B20, B50, pure ultra low sulfur diesel (ULSD) and a Swedish, low aromatic diesel in a multi-cylinder diesel engine operating in a late-injection premixed LTC mode.

Using heat release analysis, the progression of LTC combustion was explored by comparing fuel mass fraction burned. B100 had a comparatively long ignition delay compared with Swedish diesel when measured by start of ignition (SOI) to 10% fuel mass fraction burned (CA10). Differences were not as apparent when measured by SOI to start of combustion (SOC) even though their cetane numbers are comparable. Overall, cetane number was a better indicator of combustion phasing as measured by start of injection to 50% mass fraction burned (CA50) than ignition delay when comparing biodiesel with petroleum diesel in premixed LTC. To normalize for combustion phasing, injection timing was adjusted to achieve the same CA50 point for a baseline condition with each fuel. NO<sub>x</sub> emissions were the same among fuels for a given CA50 whether combustion phasing was adjusted by changing injection timing or injection pressure. Hydrocarbon (HC) emissions for B100 were lower than for ULSD and the same as for Swedish diesel. The filter smoke number (FSN) decreased as the concentration of biodiesel was

increased although the particle size distribution increased indicating higher levels of soluble hydrocarbon species were present.

## INTRODUCTION

Biodiesel is the general term used to describe alkyl-esters derived from triglycerides through the process of transesterification. Performance and emissions from diesel engine operation on various types of biodiesel is well known [1-3]. General consensus from literature is that when compared to petroleum-based diesel fuels, biodiesel has little effect on fuel conversion efficiency and decreases CO and gaseous HC when run in conventional diesel combustion modes. For oxygen-containing fuels like biodiesel, soot formation is generally lower due to lower formation during combustion [4] and better oxidation during the mixing controlled phase of combustion. Particle size distribution has been reported to trend smaller due to the lower formation of carbonaceous species leading to less agglomeration of particles in the expansion stroke and exhaust processes [5].

Although some report the opposite trend [6], NO<sub>x</sub> emissions are generally reported to increase for biodiesel. Various reasons are cited for the increase including changing start of combustion, higher premixed burn fraction, higher diffusion burn temperature and different combustion chemistry [7]. In some cases, NO<sub>x</sub> emissions have been shown to be primarily a function of combustion phasing [8] and do not show significant differences between biodiesel and petroleum diesel

when plotted versus crank angle of maximum heat release for example.

One classical feature of diesel combustion is the soot- $\text{NO}_x$  tradeoff. Both pollutants arise primarily during the mixing-controlled portion of combustion in the fuel-rich zones at the periphery of the fuel spray and the adjacent high temperature lean regions. In conventional combustion, it is generally assumed that any decrease in  $\text{NO}_x$  emissions by introducing higher EGR or lowering injection pressure leads to an increase in soot emissions and vice versa. To overcome this tradeoff, premixed low temperature diesel combustion (LTC) has been developed for limited operation over a range of speeds and loads. By using high EGR rates, high injection pressures and normal injection timing near TDC, it has been shown that ignition delay increases thus increasing the premixed portion of combustion while simultaneously reducing peak combustion temperatures which led to thermal  $\text{NO}_x$  formation [9,10]. This strategy, referred to as “late” LTC is effective in a mid-load range for common rail DI engines. An “early” LTC mode where injection is advanced into the compression stroke, similar to HCCI combustion, has been demonstrated at lower engine speeds and loads [11] and exhibits excellent BSFC while maintaining  $\text{NO}_x$  emissions below 1.5 g/kgfuel and smoke numbers below 2.0. One disadvantage of all LTC operation is higher CO and HC emissions primarily due to partially combusted fuel escaping the premixed burn process from over-lean pockets in the combustion chamber. Jacobs et al. [12] reported about two times the HC emissions and 20 times the amount of CO for an LTC condition compared with a best BFSC case with no EGR at the same fueling rate in a 4-cylinder DI diesel engine operating at 1500 rpm.

More recent work has explored the combination of using biodiesel with LTC. Zheng et al. [13] experimented with running biodiesel derived from various sources in a 4-cylinder direct-injected diesel engine with a compression ratio of 18.2:1 at a speed of 1500 rpm and 8 bar indicated mean effective pressure (IMEP). SOI was varied over a 20 degree range and EGR rates ranged from 0% to 70%. In their work, they found that compared with conventional diesel fuel, biodiesel had comparable  $\text{NO}_x$  emissions and that CO and HC emissions were lower. They also found that biodiesel was able to better sustain high EGR rates in LTC combustion due to its higher cetane number than the reference diesel and available fuel oxygen. Confirming these findings, Cheng et al. [14] found that oxygenated fuels are capable of sustaining more dilute combustion.

Fang et al. [15] also experimented with both early and late LTC strategies with biodiesel and petroleum diesel in an optical single cylinder direct-injected engine at 1500 rpm and 2 bar IMEP. They found that soy-based methyl ester (SME) had a longer ignition delay than a

reference diesel which was indicative of the cetane numbers of the fuels tested. Some European diesel fuel has very low aromatic content and therefore can have a higher cetane number than biofuels. In the LTC regime, Fang et al. also found similar  $\text{NO}_x$  emissions for SME compared with diesel fuel. In the case of LTC, since thermal  $\text{NO}_x$  formation is usually avoided due to lower combustion temperatures, it may be assumed that any formation of  $\text{NO}_x$  is then by the prompt mechanism [16] in which the initial fuel composition may have little effect.  $\text{NO}_x$  is formed at lower temperatures via this mechanism due to radical hydrocarbon species from combustion combining with nitrogen.

LTC is an emerging strategy for the reduction of emissions in production diesel engine-powered vehicles. If biofuels are to be further introduced into the fueling infrastructure, gathering more information about how they perform in different combustion regimes is important. The objective of this work is to add to the limited body of knowledge of biodiesel operation in a diesel engine running in a late premixed LTC mode. A production-type multi-cylinder diesel engine operating at a fixed speed and load was used to examine neat biodiesel compared with two petroleum-based diesel fuels. Combustion in the LTC regime was explored via heat release analysis to differentiate between cetane number and fuel chemistry effects for biodiesel and petroleum-based diesel.

Emissions were examined by normalizing for both combustion phasing and lower heating value to isolate chemical differences in combustion between fuels. Gaseous HC and CO are reported to be lower for biodiesel in both the LTC and conventional modes but since exhaust HC concentrations are considerably higher for LTC in general, it stands to reason that a large amount could be on the particulate matter in the exhaust especially since biodiesel has lower volatility. Particle size distribution was measured to test this hypothesis.

## EXPERIMENTAL

For exploring the effects of biodiesel on premixed LTC operation, a GM 4-cylinder common rail diesel engine was used. It was equipped with a variable geometry turbocharger (VGT), EGR valve and oversized EGR cooler for more precise control of LTC modes utilizing higher than normal EGR rates. Specifications of this engine are shown in Table 1 and the experimental setup is more thoroughly explained in [12].

**Table 1:** Specifications of the engine used in the experimental study.

| Type              | DI Diesel Engine  |
|-------------------|-------------------|
| Manufacturer      | General Motors    |
| Cylinders         | 4                 |
| Displacement (L)  | 1.7               |
| Bore (m)          | 0.079             |
| Stroke (m)        | 0.086             |
| Compression Ratio | 16.0              |
| Piston Geometry   | Bowl-in-Piston    |
| Valves/Cylinder   | 4                 |
| Injection System  | Common Rail       |
| Injector Location | Centrally Mounted |

Crank angle-resolved cylinder pressure data was obtained by water cooled Kistler 6041A piezo-electric pressure transducers and a Redline CAS II data acquisition system manufactured by A&D Technology Inc. Heat release curves were derived from the in-cylinder pressure data from cylinder number 1. Constant mass during the cycle was assumed for the analysis neglecting changes due to fuel injection, blow-by or crevice flows. Thermodynamic properties were calculated using the correlations of Krieger and Borman [17]. Hohenberg's correlation [18] for heat transfer was also used.

An AVL CEB II emissions bench was used to measure CO, CO<sub>2</sub>, O<sub>2</sub>, HC and NO<sub>x</sub> emissions and an AVL 415S variable sampling smoke meter was used to determine soot emissions from the engine. Combustion noise of the engine was estimated by using the cylinder number 1 pressure signal as the input to an AVL 450 Combustion Noise Meter. For particle size measurement, a DMS500 fast particulate spectrometer from Cambustion Inc. was used which utilized an internal dilution ratio of 10:1.

Fuels chosen for study included a soy-based methyl-ester (SME) biodiesel purchased from Peter-Cremer Inc. The biodiesel met the ASTM 6751 specifications at the time of testing and the fuel was used within 4 months of transesterification. Primary fatty acids found in the SME are shown in Table 2. A US-specification ultra-low sulfur diesel (ULSD) and low aromatic Swedish diesel were chosen as petroleum diesel examples. Two blends, 20% (B20) and 50% (B50) by volume biodiesel blended with ULSD were also tested for comparison. The properties of the fuels used are shown in Table 3.

**Table 2:** Fatty Acid Profile of fuels used in the study.

| Fatty Acid              | Concentration |
|-------------------------|---------------|
| Linoleic (C18:2)        | 52.3%         |
| Oleic (C18:1)           | 22.4%         |
| Palmitic (C16:0)        | 11.2%         |
| Alpha Linolenic (C18:3) | 8.28%         |
| Stearic (C18:0)         | 4.53%         |
| Arachidic (C20:0)       | 0.34%         |

**Table 3:** Properties of fuels used in the study. SWE is Swedish, low aromatic diesel fuel.

| Fuel                       | ULSD  | B20  | B50  | B100  | SWE   |
|----------------------------|-------|------|------|-------|-------|
| Cetane                     | 45    | 45   | 47   | 48    | 50    |
| LHV (MJ/kg)                | 43.1  | 41.8 | 40.2 | 37.5  | 43.6  |
| Visc. (mm <sup>2</sup> /s) | 2.38  | 2.58 | 3.02 | 3.93  | 1.85  |
| Oxygen (Wt %)              | <0.05 | 2.28 | 5.78 | 11.04 | <0.05 |
| Sulfur (Wt ppm)            | 12.0  | 7.0  | 0.8  | <0.05 | 8.0   |
| H/C ratio                  | 1.84  | 1.83 | 1.83 | 1.82  | 1.97  |

As a measure of the fuel volatility, the distillation properties are shown in Table 4. Since SME is virtually a single-component fuel, it exhibited a boiling point instead of a distillation curve.

**Table 4:** Distillation curve of fuels used in study. Values in degrees Celsius, IBP- Initial boiling point, FBP- final boiling point.

| Distillation | ULSD | SWE | SME     |
|--------------|------|-----|---------|
| IBP          | 183  | 181 | ≈ 330 * |
| 10%          | 208  | 216 |         |
| 50%          | 242  | 262 |         |
| 90%          | 309  | 308 |         |
| FBP          | 332  | 333 | 340     |

\*IBP not measured for SME

A single primary engine condition for the study was selected with ULSD as the reference fuel. Around this baseline point for each fuel, injection timing and injection pressure were varied while keeping EGR, load and speed constant at the values shown in Table 5. The baseline timing and pressure are shown in bold in the table. This point was chosen as a representative medium speed and load of steady-state vehicle operation. High levels of EGR were generated by using the VGT to increase exhaust manifold pressure as opposed to lowering the pressure of the intake manifold with the throttle plate. Jacobs and co-workers [19] indicate that CO and HC emissions are lower with a boosted LTC condition compared to a throttled condition.

**Table 5:** Engine conditions for baseline LTC condition and variation for injection pressure and timing with ULSD reference fuel.

| Condition                        | Baseline LTC  |
|----------------------------------|---------------|
| Engine Speed (rpm)               | 1500          |
| BMEP (kPa)                       | 400           |
| EGR Rate (%)                     | 50            |
| Injection Pressure (bar)         | 800,1000,1200 |
| Injection Timing (°BTDC)         | 5,7,9         |
| Air/Fuel Ratio                   | 20.8          |
| Intake Manifold Pressure (kPa)   | 107           |
| Intake Manifold Temperature (°C) | 47            |
| Coolant Temperature (°C)         | 85            |

For comparison of emissions, all concentrations were converted to a wet basis upon post-processing. Once wet emissions were calculated, the emissions index (EI) in grams per kilogram of fuel input to the engine was found using the following equation for each species,  $j$  [20].

$$EI_j = X_j \frac{MW_j}{MW_f [X_{CO} + X_{CO_2} + X_{HC_1}]} \quad (1)$$

In Equation 1,  $MW_j$  is the molecular weight, and  $X_j$  is the mole fraction of a species  $j$ . This near-dimensionless metric is commonly used to show the mass flow of a species in the exhaust with respect to the mass flow of fuel into the engine. Since different fuels with varying energy contents were compared in this study, it was necessary to normalize EI emissions with respect to the baseline fuel to remove the effect of differing lower heating value (LHV) on the results. The following formula was therefore applied to obtain the normalized EI ( $nEI_j$ ) emissions of a species  $j$ .

$$nEI_j = EI_j \frac{LHV_{ULSD}}{LHV_{fuel}} \quad (2)$$

Error of experimental data was estimated by combining the systemic uncertainty associated with design-stage errors and the repeatability uncertainty for each data point. These uncertainties were combined using the root-mean squares method assuming that variations of an error over repeated measurements tend to follow a Gaussian distribution. The method of sequential perturbation [21] was used to propagate error through calculations for variables consisting of multiple measured parameters. All errors were calculated to a 95% confidence interval. For a given sample size this is equivalent to  $\pm 2$  standard deviations ( $\sigma$ ).

Plots in this paper show error bars associated with this uncertainty calculation for individual data points. They assist in determining whether seemingly similar magnitudes of measured parameters actually indicated statistically significant differences. Linear fit lines are also shown on the plots which are solely for illustrative purposes and are not meant to indicate a linear trend for any of the data shown.

## RESULTS AND DISCUSSION

**BASELINE CONDITION** – For the baseline condition, identification of a premixed LTC mode of operation was based on the following chosen criteria:

1. Injection duration shorter than ignition delay based on start of injection (SOI) to start of combustion (SOC) to ensure adequate mixing of fuel and air
2.  $nEI\text{-NO}_x$  less than 1.5 g/kgULSD ( $\approx 45$  ppm)
3. FSN less than 0.5
4. Combustion noise less than 90 dB for practical vehicle considerations

In Figure 1, it is visually apparent that the first criterion was met by observing the injector current sensor with respect to the apparent rate of heat release (RoHR) curve.

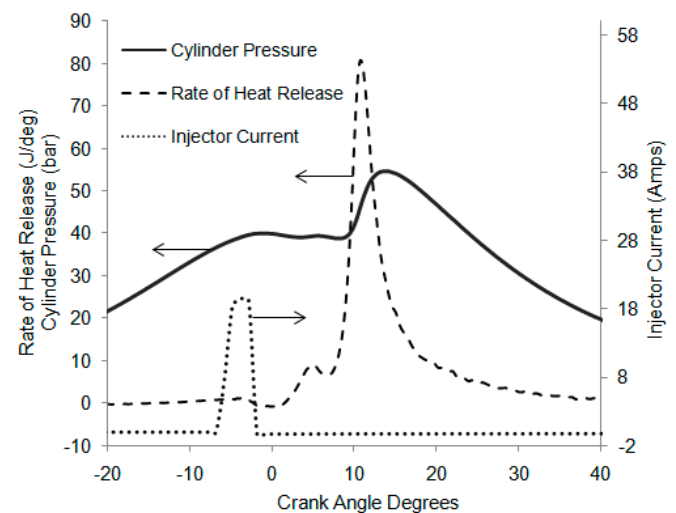


**Figure 1:** Cylinder 1 pressure, rate of heat release and injector current versus crank angle for baseline ULSD LTC condition.

From the baseline condition, injection timing and injection pressure were varied for each fuel while keeping speed and load constant at 1500 rpm and 400 kPa respectively. Between the five fuels tested, the 50% fuel mass fraction burned point (CA50) was kept constant by adjusting injection timing to establish the same baseline point. This enabled fuel comparisons to be made independent of combustion phasing. The