

APPLICATION AND COMPARISON OF SOY BASED BIODIESEL FUEL TO ULTRA LOW SULFUR DIESEL FUEL IN A HPCR DIESEL ENGINE - PART II: COMBUSTION AND EMISSIONS

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Abstract

Biofuels have the potential to diversify transportation energy sources and reduce dependence on petroleum based fuels. Of these biofuels, Methyl-ester biodiesel holds significant potential as it has many characteristics similar to petroleum based diesel and can be blended with petroleum. However biodiesel's differences in viscosity, specific energy, oxygen content, and cetane number can cause significant changes in engine performance and emissions. Therefore it is of prime interest to understand the combustion behaviour of biodiesel and identify key factors that contribute changes in engine performance and emissions.

In this study, a 100% biodiesel fuel derived from soy and an ultra low sulfur diesel fuel were tested in a high-speed direct-injection high pressure common rail four-cylinder 1.9L diesel engine. The engine control strategy allowed real-time calibration and testing of independent control parameters including start of injection, injection duration, injection pressure, and exhaust gas recirculation (EGR) level. The engine was equipped with in-cylinder pressure transducers for combustion analysis. Instrumentation for gaseous emissions detection and carbaceous particulate matter (PM) sampling was also utilized. Both the fuels were studied under varied injection timing of 0°BTDC to 12°BTDC in increments of 3°, EGR percentages of 0 and 10%, and injection pressures of 400 to 900 bar. Analysis was performed to determine the rate of heat release, ignition delay, NO_x and PM emissions.

Keywords: Biodiesel, Combustion, EGR, Injection Timing, Emissions

1. Introduction

One key characteristic of biodiesel is shorter ignition delay [1, 5, 6]. The ignition delay is dependent upon a number of fuel and engine characteristics most notably the cetane number of the fuel. A shorter ignition delay reduces the amount of fuel that is sprayed into the combustion chamber before ignition occurs, resulting in a reduction of the amount of fuel burned in the premixed portion of the diesel combustion. As a result, the rate of pressure rise in the cylinder is typically reduced with higher cetane number fuels [6, 7].

Another key characteristic of biodiesel is the presence of oxygen in the hydrocarbon chains. Senatore and coworkers [3] studied the performance and emissions using biodiesel (B100 and B25 blend) in comparison to conventional diesel fuel in a 1.9 L common rail diesel engine. The authors found higher NO_x emissions for biodiesel and attributed this to the chemical characteristics mainly the oxygen content (11% by wt.). In their studies the engine calibration was modified by varying the injection start advance and EGR rate to improve NO_x emissions. Schmidt and Gerpen [5] conducted a study to assess the impact of fuel oxygen content and cetane improvers on particulates and other emissions. It was found that the particulates decreased by 33% and NO_x emissions increased by 25% as the oxygen content of the intake air was increased from 20.5% to 22%.

A technical review [1] by the EPA analyzed existing data from various emissions tests to investigate the effects of biodiesel on exhaust emissions from diesel-powered vehicles.

The database contained no engines equipped with exhaust gas recirculation, NO_x adsorbers, or PM traps. The correlations between biodiesel concentration and emissions for on-road engines were compared to data collected on nonroad engines and light-duty vehicles. The report stated that biodiesel impacts on emissions varied depending on the type of biodiesel (soybean, rapeseed, or animal fats) and on the type of conventional diesel fuel in which the biodiesel was blended. For heavy duty on-road applications, the results showed a 48% reduction in CO, 47% in PM, and a 10% increase in NO_x. There were limited results for automotive diesels, and then only results for B20 (20% blend of biodiesel).

Graboski et al. [10] reviewed biodiesel fuels derived from fat and oils and compared them to D2 diesel fuel to study fuel properties, engine performance and emissions production. Their results show that use of biodiesel in unblended or blended form has no effect on energy based engine fuel economy. The results also showed that the lubricity of the biodiesels is superior to D2 diesel and this property is imparted to the blended biodiesel fuels. The review also indicated that the PM concentrations decreased for all biodiesels when compared to D2 diesel (500 ppm sulfur max). NO_x production increased in 2 and 4 stroke diesel engines for both the unblended and blended biodiesel fuels. Efforts to reduce NO_x production by varying injection timing and the addition of a Soluble Organic Fraction (SOF) oxidation catalyst were successful for biodiesels, blended diesels, and D2 diesel.

In conclusion, earlier research shows the use of biodiesel at any blended ratio typically decreases CO, HC and particulate emissions, but increases NO_x emissions when compared to conventional diesel. Biodiesel fuels have a shorter ignition delay which is attributed to its higher cetane number compared to US diesel fuels. For better understanding of the combustion differences between the two fuels, analysis was carried out to calculate the rate of heat release and the cumulative heat release. Through the use of this heat release analysis, the start of combustion (SOC) and ignition delay were computed. The effect of rail pressure on NO_x and PM emissions with the application of EGR was examined as well.

2. Experimental Setup

Testing was conducted on a 4 cylinder, 1.9 L high-speed direct-injection diesel engine with common rail fuel system and variable geometry turbocharger. Complete specifications of the engine and further details of the setup are described in Part I.

Emissions Measurement: The raw gas analysis was performed using a Vetronix Corporation PXA-1100 gas analyzer mounted downstream of the turbine outlet. The gas analyzer provided dry concentrations of O₂, CO, CO₂ and NO_x. The NO_x sensor is an electro-chemical sensor having a range of 0 - 4000 ppm and a resolution of 1 ppm. NO_x levels measured during this study were in the range of 50 - 900 ppm. Based on the exhaust gas composition, the air-to-fuel (a/f) ratio was calculated. Using this calculated a/f ratio and the amount of air entering the engine the fuel consumption was determined.

To ensure the accuracy of the analyzer, it was calibrated using set of calibration gases as a reference. The calibration gas was passed through the gas analyzer and the gas composition values for CO, CO₂, NO_x and HC were verified on the analyzer during regular intervals of the testing.

Ignition Delay Detection: In-cylinder pressure data was used to study the combustion behavior and determine the start of combustion (SOC) and ignition delay. The cylinder pressure, engine position data and Equation (1) were utilized to determine the apparent rate of heat release $\frac{dQ}{d\theta}$ [9],

$$\frac{dQ}{d\theta} = \left(\frac{\gamma}{\gamma-1} \right) P \left(\frac{dV}{d\theta} \right) + \left(\frac{1}{\gamma-1} \right) V \left(\frac{dP}{d\theta} \right), \quad (1)$$

where $\frac{dQ}{d\theta}$ = Rate of Heat Release (RoHR),

P = Cylinder Pressure (Pa),
 V = Cylinder Volume (m^3).

The γ (gamma) value was calculated based on the following correlation which defines gamma as a function of temperature discussed by Chun et al. in [11]

$$\gamma = 1.321 - 3.053 \cdot 10^{-5} \cdot T, \quad (2)$$

where T = Temperature (K)

Based upon the assumption that temperature and pressure in the cylinder are spatially homogeneous the average temperature in the cylinder was calculated from the measured Pressure-Volume relationship using the ideal gas law.

By choosing a suitable crank angle window in which combustion occurs, the cumulative heat release or net heat release for each cylinder was obtained from the apparent rate of heat release as follows:

$$Q_{net} = \int_{\theta_{start}}^{\theta_{end}} \frac{dQ}{d\theta} d\theta, \quad (3)$$

where θ_{start} = Start of crank angle window, θ_{end} = End of crank angle window.

To determine the SOC, a threshold value of 100 J was applied to the cumulative heat release. When the amount of net heat release surpassed this threshold value, the corresponding crank angle is defined as the SOC. Ignition delay was then determined as the time or crank angle interval difference between SOI and SOC.

Test Conditions: In all, three series of tests (test sets) were conducted to compare the two fuels. Test sets A and B are discussed in Part I. Test set C was conducted in order to study the effect of changes in rail pressure on NO_x and PM emissions. In test set C, the SOI was held constant at 9°BTDC while the rail pressure was varied from 400 - 900 bar.

3. Results & Discussion

Heat Release: Fig. 1 shows the rate of heat release from test set A with a SOI at 12°BTDC. The figure shows that when the fuels were injected at 12°BTDC, the peak rate of heat release for the ULS diesel was higher than the biodiesel fuel. Also, for this condition the SOC for ULS diesel occurs at 1.5°BTDC and the ignition delay was 0.92 ms whereas for biodiesel the SOC occurs at 2.8°BTDC and the ignition delay is 0.81 ms. There was an advance of about 1.3 degrees. Therefore it is seen that biodiesel has a shorter ignition delay compared to ULS diesel under these conditions.

Fig. 2 shows the rate of heat release from test set B with a SOI at 12°BTDC. Again the peak rate heat release for ULS diesel is greater than biodiesel. However the SOC, for ULS diesel occurs at 0.5°BTDC and the ignition delay was 1.01 ms whereas for biodiesel the SOC occurs at 2.4°BTDC and the ignition delay is 0.84 ms. The ignition delay increases noticeably with the addition of EGR in the case of the ULS diesel, whereas for the biodiesel it remains nearly the same, independent of the addition of EGR here.

Heat Release SOI Sweep: Figs 3 and 4 show the rate of heat release versus crank position for SOI timings of 0°BTDC to 12°BTDC with ULS diesel and biodiesel respectively for test set A.

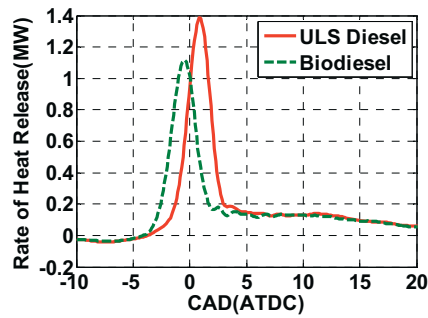


Fig. 1. RoHR (Set A, Cylinder 3, SOI 12°BTDC, no EGR)

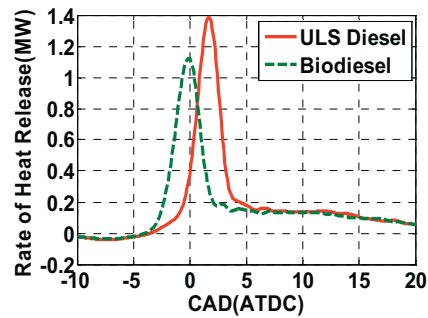


Fig. 2. RoHR (Set B, Cylinder 3, SOI 12°BTDC, 10% EGR)

Upon close examination of both the Figs, one can see the effect of shorter ignition delay of the biodiesel on combustion. For example, in Fig. 3 the SOC for SOI 3°BTDC occurs at 9.5°ATDC whereas for the same condition for biodiesel in Fig. 4 the SOC occurs at 3.1°ATDC. It is clear from these Figs that the tested biodiesel fuel has a shorter ignition delay in all cases which can be correlated to its higher cetane number. Except for SOI at 0°BTDC, the peak rate of heat release for ULS diesel is higher than biodiesel. The peak rate of heat release of ULS diesel is higher than biodiesel due to both the higher fuel energy (see details in Part I) and increased mixing of the fuel with the longer ignition delay.

Fig. 5-6 show the SOI sweep of the rate of heat release with the 10% EGR. The peak rate of heat release does not change significantly for both fuels except for the SOI at 0°BTDC with the application of EGR. The peak rate of heat release for biodiesel is again less than ULS diesel. In addition, when 10% EGR was used the ignition delay increased by an average 1 degree. This is because as the oxygen concentration in the gas mixture decreases (with EGR) which increases ignition delay.

The peak rate of heat release for ULS diesel is lower than biodiesel in test set A, for SOI at 0°BTDC and in test set B for SOI at 0°BTDC and 3°BTDC. This reverse trend is seen at these injection timings due to over mixing of the fuel with extended ignition delay.

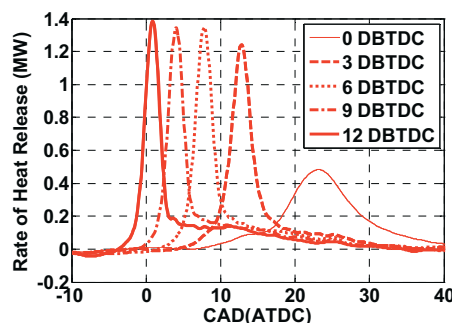


Fig. 3. ULS Diesel SOI sweep (Test set A, Cylinder 3, no EGR)

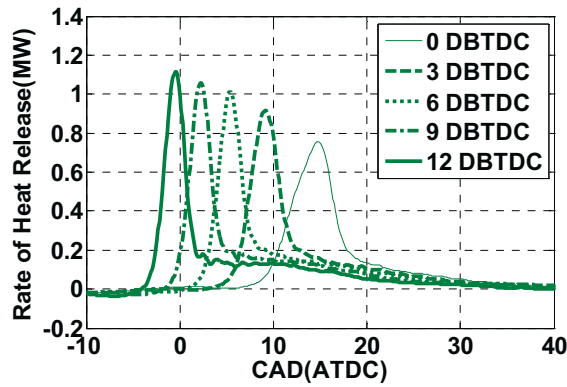


Fig. 4. Biodiesel SOI sweep (Test set A, Cylinder 3, no EGR)

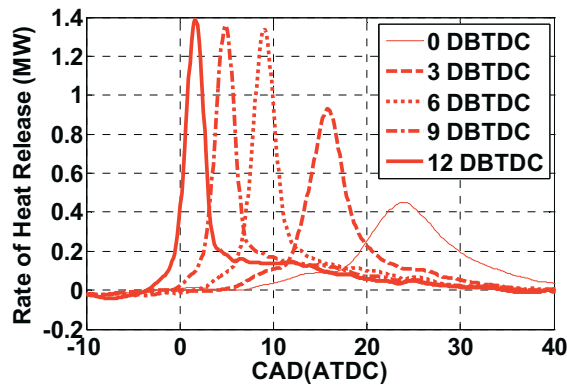


Fig. 5. ULS Diesel SOI sweep (Test set B, Cylinder 3, 10% EGR)

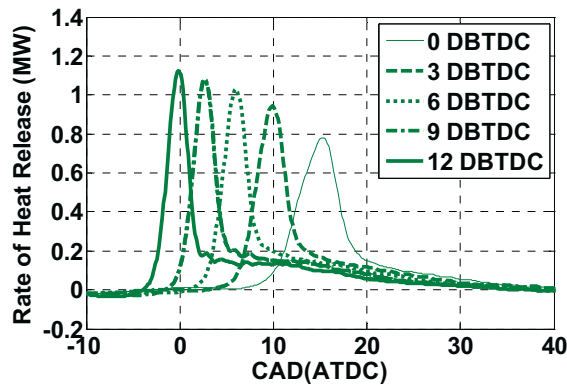


Fig. 6. Biodiesel SOI sweep (Test set B, Cylinder 3, 10% EGR)

Ignition Delay: Figs 7 and 8 show the ignition delay for the two tested fuels with respect to main injection timing and EGR concentration. The ignition delay of test set A is depicted in Fig. 7. At earlier SOI (i.e. for 6, 9, 12°BTDC) the ignition delay of biodiesel was 17% shorter than the ULS diesel. However, for 0 and 3°BTDC it is 33% shorter. In test set B with 10% EGR, depicted in Fig. 8, biodiesel averaged 19% shorter ignition delay than the ULS diesel for 6, 9 and 12°BTDC but, for 0 and 3°BTDC it is 39% shorter. Ignition delay increased for both fuels with the addition of EGR (an average 0.07 ms for ULS diesel and 0.04 ms for biodiesel).

Ignition delay of biodiesel is on average 0.2 ms shorter than that for ULS diesel. The ignition delay decreased significantly when the injection timing was advanced in case of ULS diesel. However, the SOI had almost no impact on ignition delay of biodiesel fuel.

Fig. 9 shows the ignition delay versus the peak rate of heat release for both fuels. For the ULS diesel fuel the peak rate of heat release remains nearly the unchanged until an ignition delay

of 1.1 ms. At this point the peak rate of heat release drops off dramatically with a small change in the ignition delay. In this case for the ULS diesel, the fuel continues to mix with increased ignition delay until ultimately resulting in over mixing with the extended delay. Hence a lower peak. As the ignition delay for the biodiesel is consistently shorter, the overmixing does not occur, and the dramatic decline in the peak rate of heat release is thus not observed.

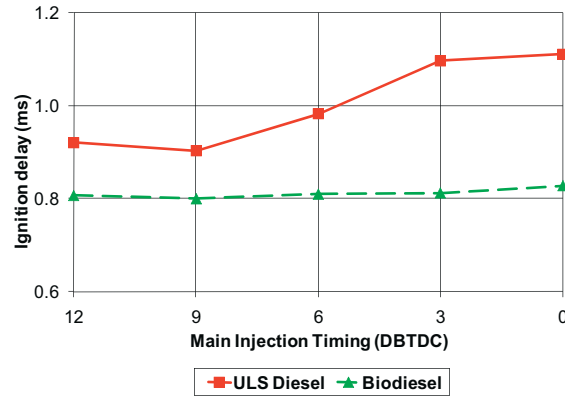


Fig. 7. Ignition Delay versus SOI timing (test set A, Cylinder 3, no EGR)

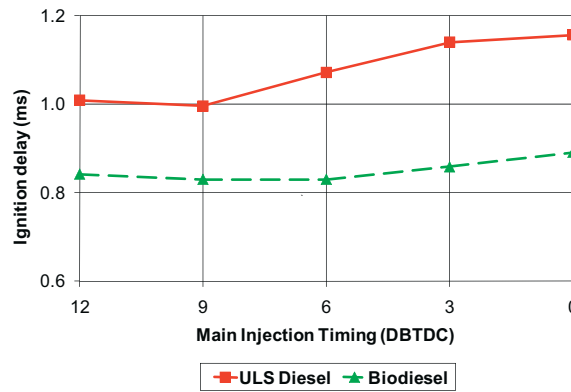


Fig. 8. Ignition Delay versus SOI timing (test set B, Cylinder 3, 10% EGR)

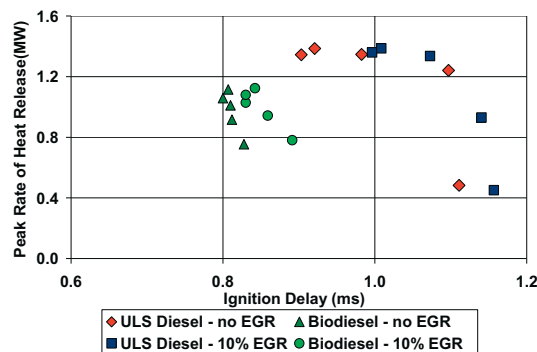


Fig. 9. Peak Rate of Heat Release versus Ignition Delay (For Test set A and B)

NO_x Emissions: Fig. 10 shows the nitrogen oxides emissions for both the fuels with varying injection timing for test set A. As the injection timing is retarded the NO_x emissions decrease as combustion phasing is retarded. With SOI retarded, the peak temperatures and pressures are lower, hence the NO_x formation is less at 0°BTDC compared to 12°BTDC. For 0% EGR both the fuels had similar NO_x emissions at SOIs of 6, 9, 12°BTDC. At SOIs of 0 and 3°BTDC, the biodiesel NO_x emissions were 28% higher than ULS diesel.

Fig. 11 shows the NO_x emissions for test set B with the addition of 10% EGR. Again it is seen that the NO_x emissions decreased when the injection timing was retarded. In comparing Figs 10 and

11 it is seen that the NO_x emissions decreased with application of 10% EGR. On an average for test set B biodiesel had 24% higher NO_x emissions compared to the ULS diesel.

Fig. 12 shows the change in NO_x emissions with the SOC for both test sets A and B. In case of test set A, the NO_x values are impacted by the difference in the fuel, but in case of test set B, the NO_x values are not impacted by the fuel, instead they are related strictly to the combustion timing.

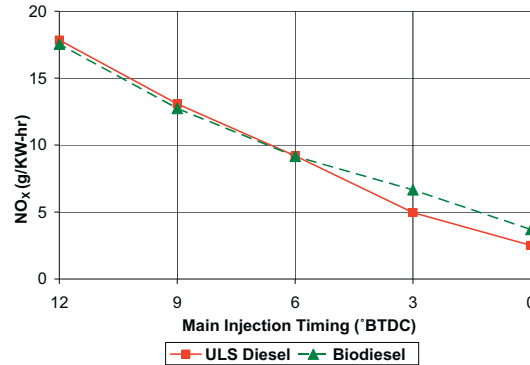


Fig. 10. NO_x emissions versus SOI timing for test set A

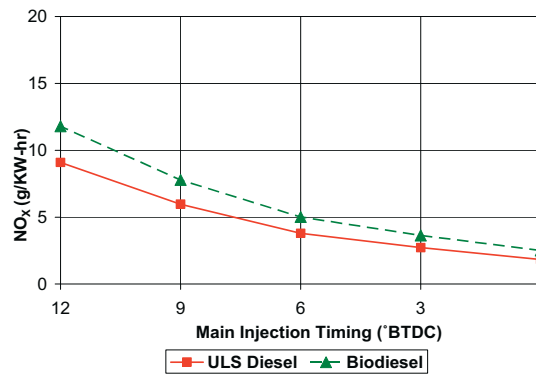


Fig. 11. NO_x emissions versus SOI timing for test set B

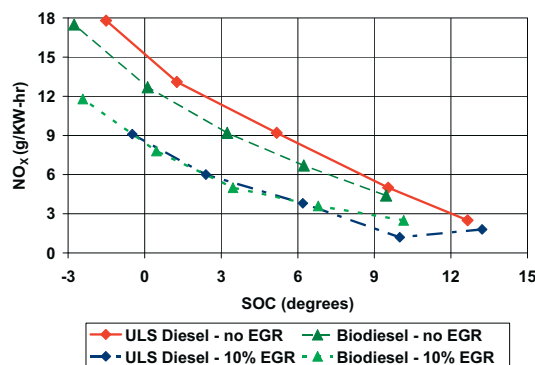


Fig. 12. NO_x emissions versus SOC (For Test set A and B)

Particulate Matter (PM) Emissions for Rail Pressure Sweep: Fig. 13 shows the effect of rail pressure on the particulate matter emissions. This test was carried out at 25% load at a constant speed of 1600 rpm and the main injection timing was kept constant at 9°BTDC. A decreasing trend in PM emissions is seen with the increase in the rail pressure for both the fuels. On an average the PM emissions of biodiesel were 4% lower than ULS diesel at rail pressures of 500, 700, and 900 bar.

However, at lowest rail pressure of 400 bar, the PM emissions of biodiesel were higher, this

might be due to poorer atomization of the biodiesel fuel at this low rail pressures as a resultant of its high viscosity.

Fig. 14 shows the variation of NO_x emissions with change in rail pressure. For all cases the NO_x emissions for biodiesel were approximately 4% higher than ULS diesel. Fig. 15 shows the NO_x and PM trade-off as the rail pressure was changed. As the NO_x emissions is increasing the PM emissions is decreasing. The comparatively lower PM emissions of biodiesel can be attributed to the oxygen content present in the fuel. The oxygen present in the fuel contributes to an increase in the NO_x emissions. However, the differences between both the fuels are not significant.

4. Conclusions

The following conclusions are obtained:

- Heat release analysis shows that biodiesel has a shorter ignition delay (an average 0.2 ms) than ULS diesel for both no EGR and 10% EGR.
- The peak rate of heat release is not significantly unaffected by application of 10% EGR but the ignition delay increased by an average 0.08 ms for ULS diesel and 0.04 ms for biodiesel when EGR was used.
- The ignition delay decreased significantly when the injection timing is advanced in case of ULS diesel. SOI had almost no impact on ignition delay of the biodiesel fuel.
- As the injection timing is retarded, the NO_x emissions decreased for both fuels. This is due to retarding of combustion phasing which results in lower temperature and pressure, hence lower NO_x.
- For test set A both the fuels had similar NO_x emissions for 6, 9 and 12°BTDC but for 0 and 3°BTDC biodiesel had 28% higher NO_x emissions than ULS diesel. For test set B on an average biodiesel had 24% higher NO_x emissions compared to the ULS diesel. The NO_x values for test set B are not impacted by the fuel. Instead they are related to the difference in the combustion timing.
- A rail pressure sweep was performed to see the effect of rail pressure on NO_x and PM emissions.
 - On an average the PM emissions of biodiesel were 4% lower than ULS diesel at rail pressures of 500, 700, and 900 bar.
 - The NO_x emissions for biodiesel were approximately 3% higher than ULS diesel.

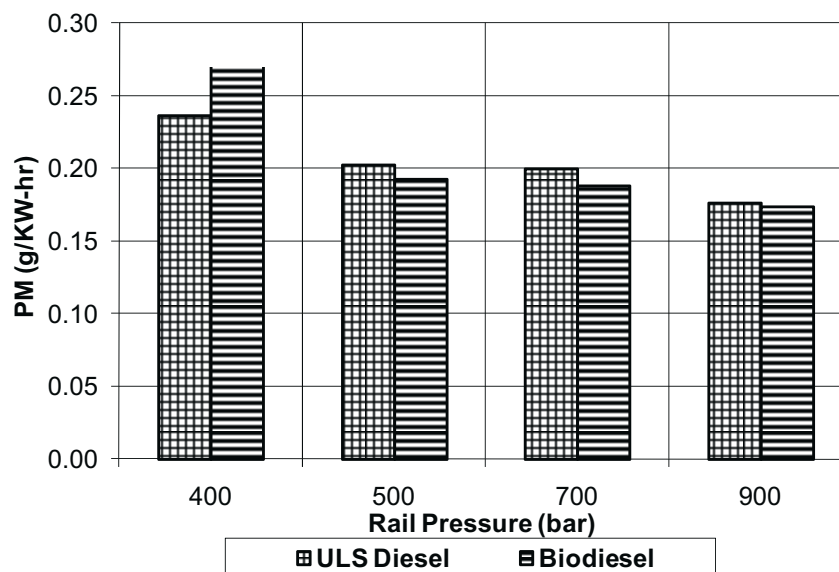


Fig. 13. PM variation with Rail Pressure (test set C)

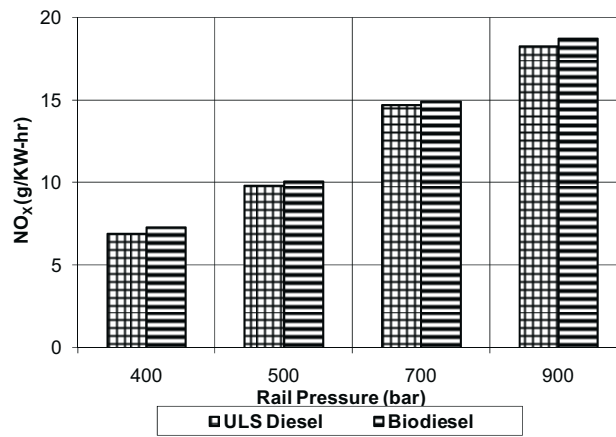


Fig. 14. NO_x Variation with Rail Pressure (test set C)

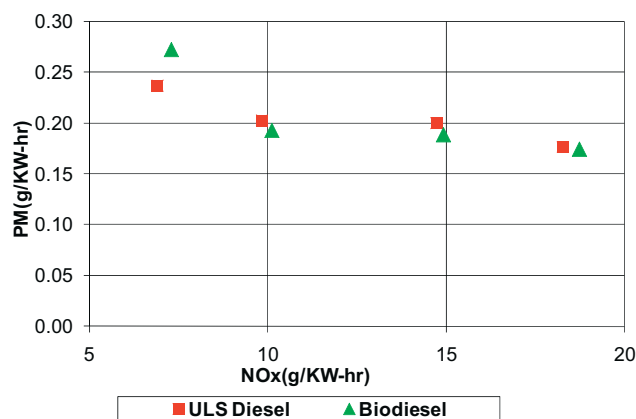


Fig. 15. NO_x-PM trade-off (test set C)

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