APPLICATION AND COMPARISON OF SOY BASED BIODIESEL FUEL TO ULTRA LOW SULFUR DIESEL FUEL IN A HPCR DIESEL ENGINE - PART I: ENGINE PERFORMANCE PARAMETERS

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Abstract

In the US transporation sector uses two-thirds of the country's total oil consumption. In order to minimize the consumption in this sector there is a need to investigate alternate sources of energy. Biodiesel is a possible alternative to conventional diesel. Biodiesel has many characteristics similar to petroleum based diesel and can be blended with petroleum. However biodiesel's differences in fuel properties including viscosity, bulk modulus, density, and energy content can have significant impacts on engine performance parameters like BSFC and thermal efficiency. As the availability of biodiesel fuel increases, the need for engines capable of running on various mixtures of biodiesel fuel will be required. Similar to flex-fuel ethanol vehicles, control systems for the diesel engine and aftertreatment systems will need to detect and compensate for the fuel type.

In this work, a soy based B100 biodiesel fuel and an ultra low sulfur diesel fuel were tested in a high-speed direct-injection high pressure common rail four-cylinder 1.9 L diesel engine. An internally developed 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. Both the fuels were studied under varied injection timing (0°BTDC to 12°BTDC with increments of 3°) and EGR percentages of 0 and 10%. Analysis was performed to determine the Torque, BSFC and Brake thermal efficiency.

Keywords: *Biodiesel, EGR, injection timing, performance*

1. Introduction

The performance of biodiesel fuels in diesel engines is typically comparable to conventional diesel fuel in terms of thermal efficiency, BSFC, heat release and ignition delay (discussed in Part II) making it an alternative to conventional diesel fuel [1 - 4]. Wong et al. [8] discussed the effect of fuel properties and engine operating conditions on ignition delay. The effects of cetane number, engine torque, volatility, viscosity, intake air-pressure and temperature on ignition delay were examined. In their work, the results showed that the major factors affecting the ignition delay were the cetane number, engine load and intake air pressure. Volatility and viscosity affects were negligible.

Hribernik and Kegl [6] studied the effect of biodiesel fuel on the engine operation, combustion and emission formation in two different direct injection diesel engines. One a 7.1 L turbocharged and the other a 11.4 L naturally aspirated engine. The study was conducted to find the influence of a specific combustion process on the combustion and emissions results, to ascertain whether a generalization of the results is possible or whether they have to be interpreted as engine specific. They concluded that the operation parameters of both the engines were similar for biodiesel. With biodiesel fuel the maximum power and torque reduced by 5%, fuel consumption increased by 8%.

However there was a significant difference in the combustion processes of both the engines. Hence a generalization was not made by the authors. For the naturally aspirated engine, the rate of heat release was independent of fuel type (biodiesel and conventional diesel). For the turbocharged engine, biodiesel had a shorter ignition delay. The intensity of premixed combustion was reduced by 40% with the use of biodiesel but the combustion duration was unaltered.

These previous studies were compiled on a number of different engines and fuels, but none compared diesel fuel with biodiesel with similar levels of sulfur. In this study, a B100 fuel was compared with a ULS diesel fuel (less than 1 ppm sulfur) (see appendix for fuel details) with similar sulfur levels with sweeps of injection timing. In this part of the paper the perfromance parameters including Torque, BSFC and Brake thermal efficiency are discussed. Part II discusses the combustion and emissions trends.

2. Experimental setup

Testing was conducted on a 4 cylinder, 1.9 L HSDI diesel engine with common rail fuel system and variable geometry turbocharger typical of European and US diesel automotive engines. The engine was coupled to a 150 kW eddy current dynamometer. The engine specifications are listed in Tab. 1.

Table 1. Engine Specifications

Displacement (cm^3)	1896
Cylinder Arrangement	4-cylinder inline
Bore x Stroke (mm)	79.5 x 95.5
Compression Ratio	19.5:1
Max. Power (KW)	66 @ 3750 rpm
Max. Torque (Nm)	210 @ 1900 rpm
Engine Management	Turbo Direct Injection (TDI)
Fuel System	Bosch Common Rail
Engine Control Unit	Mototron/ Motorola MPC 555
Injector Driving Unit	Motorola HPCR IDU

The engine is controlled through the use of a Target-Based Rapid-Prototyping (TB-RP) control system developed by MotoTron [9]. This system utilizes a production engine control unit (ECU) with a production set of low level drivers with the ability to develop control strategies in MathWorks Simulink/Stateflow®. The fuel injection system is a high pressure common rail (HPCR) system where in the rail pressure can be varied between 350 bar to 1800 bar for engine operation. The HPCR injectors are driven by a separate Injector Driver Unit (IDU) that is operated in a slave mode and controlled by the ECU. The control system programming was done at Michigan Technological University using the TB-RP system [10, 11].

In-cylinder pressure was measured using 1 PCB (X175A01) and 3 Kistler (6123) pressure transducers. The pressure transducers were passage mounted in the cylinder head; further explanation is given in [20]. The engine position was measured by a 360 tooth optical encoder. Interpolation between the engine position and encoder teeth was done using software. This interpolation facilitates determination of the absolute engine position at each sample point in time.

Data on in-cylinder pressure and crank position were acquired using a National Instruments DAQ system. The DAQ system has eight differential analog inputs with an aggregate sampling rate of 1.25 Mega-samples per second. Data from the four pressure transducers and crankshaft encoder were captured at a sampling rate of 100 kilo samples per channel per second. In-cylinder pressure data was acquired for 50 engine cycles for each test condition. This data was then analyzed offline for further analysis.

EGR Percentage: Two NTK universal exhaust gas oxygen sensors model number LZA03-E1 were installed, with one in the intake manifold and another in the exhaust, downstream of the turbocharger and EGR valve. These sensors were used to calculate the EGR percentage. It is defined as a volumetric flow-rate of EGR gas divided by the total volumetric flow-rate at standard pressure and temperature conditions of the total charge gas into the cylinder as seen in Equation (1).

$$EGR\ \% = \frac{\dot{Q}_{EGR}}{\dot{Q}_{Air} + \dot{Q}_{EGR}}, \tag{1}$$

where $\dot{Q}_{EGR}$ = Volumetric flow-rate of EGR gas,

$\dot{Q}_{Air}$ = Volumetric flow-rate of fresh air.

The EGR percentage was characterized by the oxygen content of the EGR gas and the incoming air charge, as these two combine to lower the oxygen concentration in the intake manifold. The in-cylinder oxygen content can then be stated as Equation 2.

$$[O_2]_I = \frac{[O_2]_{Air} \cdot \dot{Q}_{Air} + [O_2]_{EGR} \cdot \dot{Q}_{EGR}}{\dot{Q}_{Air} + \dot{Q}_{EGR}}. \tag{2}$$

From Equations (1) and (2)

$$EGR\% = \frac{[O_2]_I - [O_2]_{Air}}{[O_2]_{EGR} - [O_2]_{Air}}, \tag{3}$$

where

$[O_2]_I$ is the measured oxygen concentration in the intake manifold,

$[O_2]_{Air}$ is the oxygen content of the atmosphere,

$[O_2]_{EGR}$ is the oxygen content of the exhaust stream.

3. Test Conditions

Two series of tests (test sets) were conducted to compare the two fuels. Test sets A and B were carried out maintaining an injection duration of 0.58 ms and a rail pressure of 700 bar. In test set A, the start of injection (SOI) timing was varied from 0°BTDC to 12°BTDC in increments of 3°with no EGR to measure and compare the rate of heat release, BMEP and ignition delay. Similarly in test set B, the SOI timing was varied but with the addition of 10% EGR to measure the combined effect of fuel and cooled EGR on heat release, BMEP, ignition delay.

4. Results & Discussion

Tests A and B were carried out at 25% load (52 N-m nominal torque). The engine speed was held constant at 1900 rpm, injection duration was fixed at 0.58 ms, rail pressure of 700 bar, and no EGR was used for test set A.

Fuel Mass and Energy Delivered: Table 2 shows the average mass of fuel per cylinder, the lower heating value (LHV) and energy delivered for both the fuels with the fixed injection duration of 0.58 ms.

Table 2. Average values mass and energy delivered for the two fuels

Fuel	Injection Duration (ms)	m_f (mg/cylinder)	LHV (MJ/Kg)	Energy in = $m_f \cdot$ LHV (J/cylinder)
ULS Diesel	0.58	14.3	43	618
Biodiesel	0.58	15.5	37	576
%Δ Biodiesel	0	7.2	-14.0	-6.7

The air to fuel ratio was calculated from the exhaust composition emitted by the engine. Then using the mass air flow value per cylinder and the calculated air to fuel ratio the mass of the fuel per cylinder was calculated. Multiplication of the mass of the fuel per cylinder value with the lower heating value of the fuel gave the energy delivered per cylinder per cycle.

As seen in Tab. 2, the mass of biodiesel fuel supplied per cylinder is 7.2% higher than ULS diesel fuel. The LHV of biodiesel is 14% less than ULS diesel and the energy delivered per cylinder for biodiesel fuel is 6.7% less than ULS diesel. The higher delivered mass of biodiesel fuel is a result of the higher density of biodiesel fuel which is 7% higher than ULS diesel. Even though more fuel is delivered in case of biodiesel, the energy delivered is less than ULS diesel because of a larger difference in the LHV.

Torque: Fig. 1 and 2 show the brake torque for the tested fuels while varying the main injection timing or the start of injection (SOI) for test sets A and B respectively. The torque produced for both the fuels ranged between 41 to 52 N-m for both test sets A and B. The average indicated mean effective pressure (IMEP) for ULS diesel and biodiesel were 545 kPa and 490 kPa respectively.

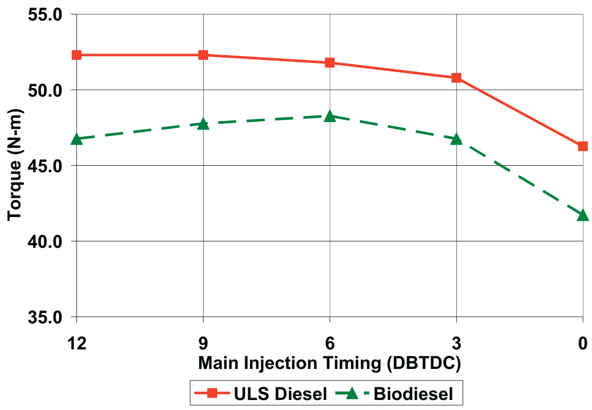

Fig. 1. Torque versus SOI timing (Test set A)

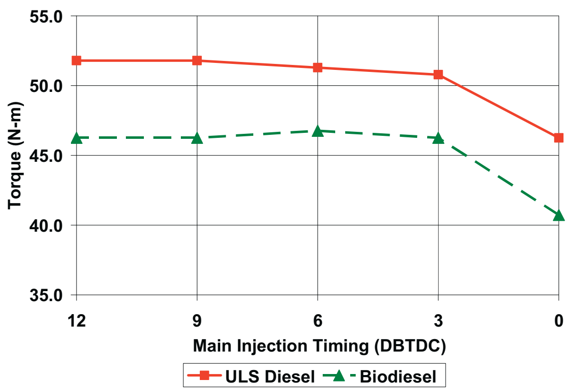

Fig. 2 Torque versus SOI timing (Test set B)

In this case, the torque (*T*) produced by a particular fuel can be explained by Equation (4)

$$T = N_{Cyl} \cdot \frac{\eta \cdot LHV \cdot m_f}{4\pi}, \tag{4}$$

where: N_{cyl} = Number of cylinders,
η = Thermal Efficiency,
LHV = Lower Heating Value (J/kg),
m_f = Fuel mass per cylinder (kg/cylinder).

Eq. (4) can also be written as:

$$T = \frac{\eta \cdot Energy\_in}{4\pi}, \tag{5}$$

where $Energy\_in = N_{Cyl} \cdot LHV \cdot m_f$.

From the above equations it can be seen that the torque produced by a particular fuel is directly proportional to its thermal efficiency and the energy delivered, which in turn is a function of the LHV and the fuel mass delivered per cylinder. As seen in Table 2 the energy delivered per cylinder in case of biodiesel is 6.7% less than ULS diesel, hence assuming that the engines are operating at similar efficiencies, the biodiesel will produce less torque than ULS diesel. The primary reason for this is the LHV of biodiesel which is 14% less than ULS diesel. In this study, for both test set A the torque produced by biodiesel with respect to the ULS diesel fuel is 8% less and for test set B it is 9% less. The torque produced in test set A is higher than test set B for but the difference is small.

Brake specific fuel consumption (BSFC): Fig. 3 - 4 show the brake specific fuel consumption with respect to th SOI timing from test sets A and B respectively. The BSFC for the biodiesel fuel is higher for all SOI timings.

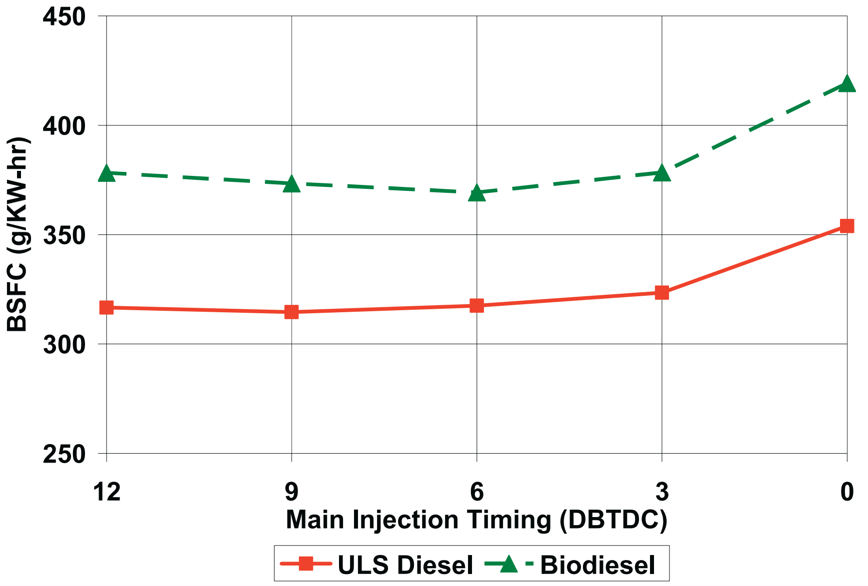

Fig. 3. BSFC versus SOI timing (Test set A)

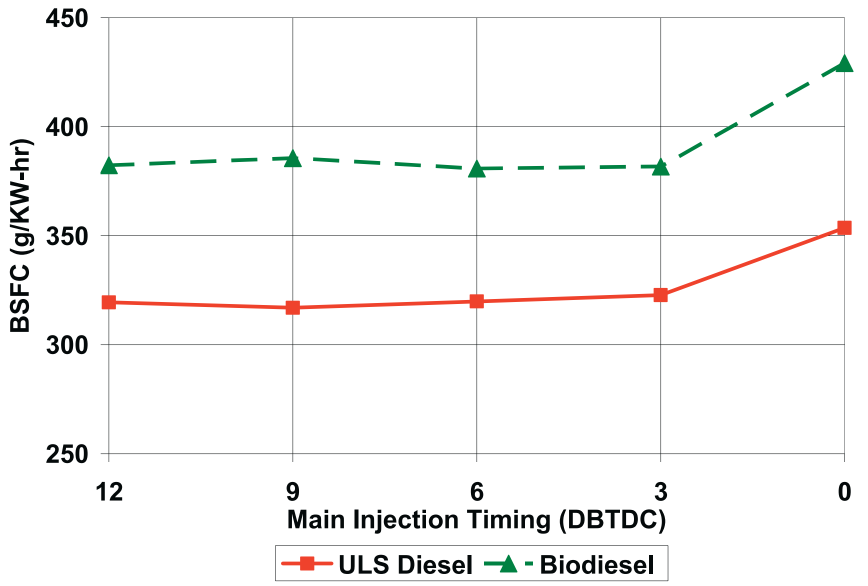

Fig. 4. BSFC versus SOI timing (Test set B)

The BSFC (g/KW-hr) in terms of torque and mass of fuel injection can be explained by Equation (6).

$$BSFC = \frac{3600 \cdot NCyl \cdot m_f}{4\pi \cdot T}, \tag{6}$$

where: m_f = Fuel mass per cylinder (mg/cylinder),

T = Torque (N-m).

From Equation 6 we can see that the BSFC is directly proportional to the mass of the fuel per cylinder and inversely proportional to the torque produced. As discussed earlier the torque produced by biodiesel fuel is an average 9% less than the ULS diesel primarily due to its LHV. As this value is lower for biodiesel it increases the BSFC value. Therefore, biodiesel has higher BSFC than ULS diesel. For test set A, the average percentage change is 18% higher and for test set B, it is 20%. The mass of biodiesel fuel injected per cycle having a smaller effect compared to the difference in torque.

Brake Thermal Efficiency: Figures 5 and 6 show the brake thermal efficiency versus SOI for test set A and B respectively. On an average for all SOI timings the brake thermal efficiency of biodiesel fuel is 2% lower than ULS diesel for test set A and 4% lower for test set B.

The thermal efficiency in terms of BSFC and LHV can be explained by Equation (7).

$$\eta = \frac{3600}{BSFC \cdot LHV}, \tag{7}$$

where:S³owa kluczowe: ograniczniki prêdkoœci the *BSFC* is in g/kW-hr, and *LHV* is in MJ/kg.

The thermal efficiency is inversely proportional to the product of BSFC and the LHV. The higher BSFC of biodiesel is directly rated to its lower fuel energy (LHV) and the lower BSFC of ULS diesel to its higher LHV. Thus both the fuels are seen to have similar thermal efficiencies.

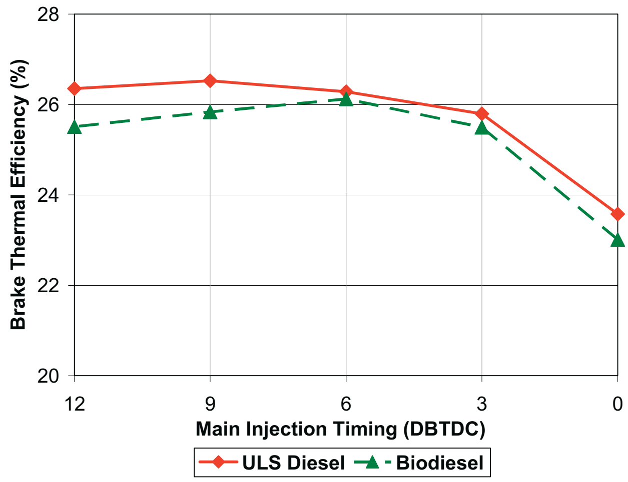

Fig. 5. Brake Thermal Efficiency versus SOI timing (Test set A)

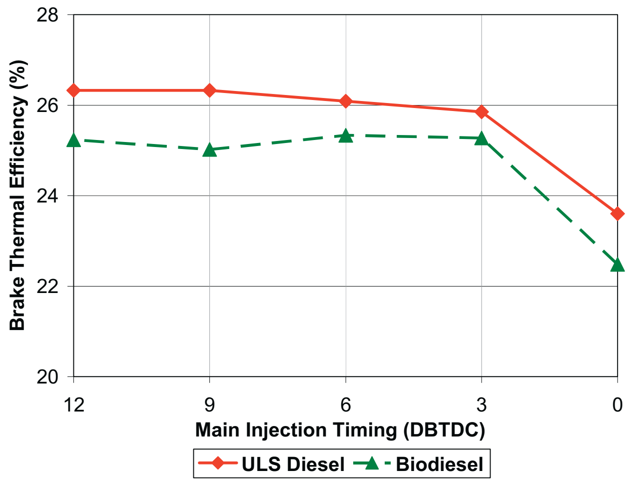

Fig. 6. Brake Thermal Efficiency versus SOI timing (Test set B)

5. Conclusions

The following conclusions are obtained:

- The measuremass of the fuel delivered per cylinder for biodiesel is 7.2% higher than ULS diesel. This is seen as primarily due to the 7% higher density of biodiesel fuel compared to ULS diesel.
- The energy delivered per cylinder for biodiesel is 6.7% lower than ULS diesel. This is attributed to the combination of the LHV of tested biodiesel fuel which is 14% lower than ULS diesel and the higher density of the biodiesel.
- The torque produced by the biodiesel with respect to the ULS diesel fuel is 8% less for test set A and 9% less for test set B at constant injection duration, rail pressure and speed. This is a result of the energy delivered per cylinder of biodiesel is less than ULS diesel which in turn is a resultant of the fuels LHV.
- The BSFC of biodiesel is 18% higher than ULS diesel for test set A and 20% higher for test set B. This is because biodiesel produced 10% less torque than ULS diesel which is again mostly a result of the LHV.
- The thermal efficiency of biodiesel is 2% lower than ULS diesel for test set A and 4% lower for test set B. The higher BSFC of biodiesel compensates for its lower fuel energy and the lower BSFC of ULS diesel compensates for its higher fuel energy. Thus both fuels are seen to have similar thermal efficiencies.

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Appendix - Table of Fuel Properties

Parameter	Units	ULS Diesel	Biodiesel	Test Standard
Carbon	% mass	86.48	77.22	ASTM D1319
Oxygen	% mass	-	10.94	ASTM D5622
Hydrogen	% mass	13.52	11.84	ASTM D5622
Sulfur	ppm wt.	0.7	1	ASTM D5453
Cetane No.	-	46	58	ASTM D613
Lower Heating Value (LHV)	MJ/kg	43.14	37.31	ASTM D240
Viscosity, Kinematic @ 40°C	mm^2/s	2.35	4.01	ASTM D445
Density @ 15.56°C	Kg/m^3	827	885	ASTM D4052
Distillation (T90)	°C	310	360*	ASTM D86/*D1160
H/C Atomic Ratio	-	1.86	1.83	SAE J1829
Stoic. A/F ratio	-	14.40	12.32	SAE J1829