Title: Performance And Heat Release Rate Of A Diesel Engine Using Sunflower Methyl Esters And Diesel Fuel Blends In Experimental Comparison

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Performance and Heat Release Rate Comparison in a Diesel Engine fueled with Sunflower Methyl Esters/Diesel Fuel Blend

Aykut Safa*1

Abstract

Biodiesel fuels are among the leading renewable alternative fuels. And, using biodiesel fuels is available and easy to implement on diesel engines. Biodiesel fuels, methyl/ethyl esters of various oils extracted from vegetables or animals, are being used by many researchers on various types of engines and they were tested at various conditions. Since chemical properties very close to diesel fuel, similar test results for nominal engine performance data with diesel can be obtained by using biodiesel fuels on engines. In this study, effect of sunflower oil methyl esters on engine performance, heat release rate and indicated engine parameters are investigated experimentally on a single cylinder, DI Diesel engine. Tests are performed with diesel fuel and B50 blend fuel, consisting of 50% diesel and 50% biodiesel fuels, at full load condition and at different speeds. Engine brake torque, brake power, brake specific fuel consumption, brake thermal efficiency, heat release rate, and indicated engine parameters are calculated from the test data, and the results obtained by using diesel fuel and B50 blend fuel are compared. Although some worsening on performance data and heat release rate on B50 side, results are very close to each other.

Keywords: Diesel Engine performance, Sunflower Biodiesel Fuel Blends, Heat Release Rate analysis

1. INTRODUCTION

In literature, studies on various biodiesels including Sunflower Methyl Ester (SFME) are present [1-4]. Biodiesel properties and effect on engine performance are investigated in various aspects [5-9]. And studies on heat release data calculated by pressure data can be found in literature [15-18]. In this study, experimental results of a DI CI engine using diesel and biodiesel of SFME and blend fuels are investigated in comparison. Engine performance data, power, torque, and BSFC are revealed and heat release rates obtained from pressure data and engine data are calculated for diesel fuel and B50 blend fuels. The results found close to each other.

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Biodiesel fuel is a renewable and it can be obtained from waste/raw vegetable oils or animal fats, etc. Since biodiesels are produced by transesterification of fatty acids, they offer high energy density and serve for sustainability through conversion of fatty acids from bio-resources. Thus, the studies on biofuels are increasing in number due to these facts. For widespread usage of biodiesels, standardization is required. However, deviations occur in biodiesel characteristics depending on fuel content [3]. Therefore, for improved insight for correlating biodiesel properties and engine performance and emissions, the corresponding relations should be investigated. Another advantage of biodiesels is the opportunity to be used in a conventional diesel engine easily [9]. [12] Alptekin et al, 2008 reported densities and viscosities of blend fuels, blend of diesel and biodiesel. Experiments on blend fractions, B2-B75, were reported for biodiesel fuels, from sunflower, canola, soybean, cottonseed, corn oils and waste palm oils, according to ASTM test methods. It is reported that, densities and viscosities of the blend fuels increase with the increase of biodiesel fraction in the fuel. [8] Demirbas, 2008, studied viscosity, density, flash point and HHV properties of various biodiesels and concluded aspects to use biodiesels in comparison to conventional fuels of higher viscosity and lower HHV. And also, it is concluded that, viscosity and HHV were highly correlated for vegetable oil methyl esters. [6] Sayin et al, 2009, tested diesel and methanol blends with various fuel injection timings, to improve engine emissions, using a single cylinder CI test engine. As reported, increase in methanol fraction causes decrease in smoke opacity, CO and UHC emissions, but increase in NOx emissions, and advance in injection timing cause reductions in smoke opacity, CO and UHC, but increase in NOx and CO2 emissions. [10] Ozsezen et al, 2011 tested COME and WPOME fuel on a DI diesel engine. Engine with compression ratio, 15.9, and fuel injection of pressure, 197 bar, are used at tests. Heat release analysis is also utilized. Reduction in engine brake power, CO, HC, CO2, smoke and increase in BSFC and in NOx emissions are reported compared to diesel fuel results under full load at constant engine speeds [5] Hoekman, 2012 studied 12 common biodiesel feedstocks and presented a detailed report on several properties of biodiesels. [2] Santos et al., 2013 showed the effect of engine parameters and biodiesel fractions on performance and emissions using SFME, SME and neat diesel on two diesel engines. [13] Lahane, 2015 investigated fuel injection, performance, and emissions characteristics of a DI diesel engine at 1500 rpm, using diesel and biodiesel of karanja blend fuels. As reported, ignition delay and pressure rate are lowered for biodiesel fractions. Since biodiesel has a greater cetane number than diesel, CO, HC and smoke emissions are reduced, but NOx emissions are increased. [4] Pearson et al, 2015 investigated iso-stoichiometric fuel blends and information on chemical properties of hydrous blends, containing water, gasoline, ethanol and methanol is given. [7] Prashant et al, 2016 using diesel and methanol blend fuel studied ignition delay, pressure rate and heat release rate on a dual fuel CI engine. As the methanol fraction is increased, ignition delay, and the angle corresponding to the maximum heat release is increased. [9] Prajapati et al, 2016 studied diesel and biodiesel of sunflower, blend fuels on a single cylinder, DI CI engine at 1500 rpm at various loads. And, for 25% fraction of biodiesel in diesel is concluded to have very close results specific fuel consumption and brake thermal efficiency, compared to neat diesel fuel. [1] Ayhan et al, 2018 proposed conditional blend fuel fractions containing diesel fuel, DEE, biodiesel (SFME) and water. The experimental study done at engine speeds, 1000-2200 rpm, and full load on DI diesel engine is used. The reason of DEE addition is reported to increase blend fuel cetane number and the surfactant addition to obtain a stabilized fuel with water addition. At the maximum torque condition, reductions in engine brake power, brake torque and in CO, NO and smoke levels but increase in brake efficiency and in HC are reported. Fuel injection pressure was 175 bar. [11] Kirankumar et al, 2018 was run a single cylinder diesel engine on neat diesel, B25, B50, B75, and B100 blend fuels of SFME at 1500 rpm. Engine of compression ratio with 16.5 is used in tests. And specific fuel consumption, brake thermal efficiency, exhaust gas temperature increase with brake power has been reported. While, at full load condition using diesel fuel causes lower CO, CO2,
NO\textsubscript{x} and smoke emissions, but using B100 fuel causes the lowest HC emission. In oxygenated fuel blends, complete combustion takes place resulting in high NO\textsubscript{x} has also been reported. Because high combustion temperatures occur during the complete combustion, the resulting NO\textsubscript{x} formation is high. And also presented that, decrease in the maximum in-cylinder pressure lowers as the blend fraction increases. And it was concluded that the maximum heat release rate for diesel fuel is twice the B100.

Heat release during combustion occur in cylinder is affected by various parameters such as mixture formation, heat loss, engine operational parameters, etc. Therefore, using experimental data, apparent heat release rate can be estimated. [14] Gogoi and Baruah, 2010 studied the effect of biodiesel, blends of karanja oil methyl ester and diesel fuel, on performance through a zero-dimensional cycle simulation model and experimental data. And an increase in brake power for B40 and B60 blend fuels, but a decrease for B20, and also, increase in thermal efficiency for all the biodiesel blends in comparison to neat diesel oil is reported. [15] Asad and Zheng, 2008 investigated heat release mechanisms in compression ignition engines. And they suggest importance of determining combustion parameters based on obtained experimental data. [16] Abbaszadehmosayebi and Ganippa, 2014 calculated the heat release using combustion burnt factor. They concluded, using combustion burnt factor, in comparison to Wiebe function, gives better results. [17] Vipavanich et al., 2018, conducted tests on a CI engine running on gasoline, injected into port, and diesel fuel. Thermal efficiency was evaluated through heat release analysis and combustion parameters are investigated.

This study presents the effect of using biodiesel from sunflower oil on engine performance parameters, brake torque, brake power, brake specific fuel consumption, cumulative heat release and heat release rate. For this reason, test results for fuel blend of diesel fuel and SFME fuel at 50% fraction and neat diesel fuel are given for comparison. And heat release data calculated from engine pressure data are investigated to understand the effect of biodiesel blend fuel better.

2. EXPERIMENTAL SETUP

2.1. Production of Sunflower Oil Methyl Ester

Biodiesel from corn oil is produced by transesterification method. Methanol, with 99% pure alcohol, and KOH, catalyst, are used. Alcohol and catalyst are weighed, and placed in a glass container to dissolve completely. Later, corn oil is heated to the required temperature, and then alcohol and catalyst preparation is added in. The preparation is mixed with a mechanical mixer at a constant temperature for the transesterification reactions to take place. And, glycerin is separated from the ester. Extracted ester is washed with pure water and then taken to drying process. Fuels properties used in the study are given in Table 1.

Table 1 Properties of fuels used in tests

<table>
<thead>
<tr>
<th>Fuel property</th>
<th>Diesel C\textsubscript{15}H\textsubscript{25.5}</th>
<th>Biodiesel C\textsubscript{19}H\textsubscript{35}O\textsubscript{2}</th>
<th>B50</th>
</tr>
</thead>
<tbody>
<tr>
<td>Density [kg/m\textsuperscript{3}]</td>
<td>832</td>
<td>880</td>
<td>862.5</td>
</tr>
<tr>
<td>Cetane number</td>
<td>57</td>
<td>46.8</td>
<td>51.9</td>
</tr>
<tr>
<td>Lower heating value [MJ/kg]</td>
<td>42.6</td>
<td>39.5</td>
<td>41.05</td>
</tr>
<tr>
<td>Kinetic viscosity at 40 °C [mm\textsuperscript{2}/s]</td>
<td>4.5</td>
<td>4.7</td>
<td>4.6</td>
</tr>
<tr>
<td>Self-ignition temperature [°C]</td>
<td>250</td>
<td>125</td>
<td>187.5</td>
</tr>
</tbody>
</table>

As given in Tab.1 mass fractions are compared. Carbon content in biodiesel is lower compared to diesel fuel, 85.4%, and 87.6% respectively. Therefore, carbonaceous pollutants from biodiesel fueled diesel engines will be less than neat diesel. The physicochemical properties given in Tab1. is close to the data given by Richard J Pearson [4] et al. And intermediate data [4] are calculated by mass ratios.

2.2. Engine Tests

A single cylinder, 4-stroke, water cooled, direct injection Superstar make test engine is used in the experiments, Table 2.
Table 2 Test Engine specifications

<table>
<thead>
<tr>
<th>Engine</th>
<th>Super Star 4 stroke CI engine</th>
</tr>
</thead>
<tbody>
<tr>
<td>Aspiration</td>
<td>Atmospheric</td>
</tr>
<tr>
<td>Cooling</td>
<td>Water circulation</td>
</tr>
<tr>
<td>Injection</td>
<td>DI</td>
</tr>
<tr>
<td>Bore [mm]</td>
<td>108</td>
</tr>
<tr>
<td>Stroke [mm]</td>
<td>100</td>
</tr>
<tr>
<td>Displacement volume [dm^3]</td>
<td>0.92</td>
</tr>
<tr>
<td>Compression ratio</td>
<td>17</td>
</tr>
<tr>
<td>Rated power @2200 rpm, [kW]</td>
<td>13</td>
</tr>
<tr>
<td>Injection pressure, [bar]</td>
<td>225</td>
</tr>
<tr>
<td>Injection timing, [°CA bTDC]</td>
<td>29</td>
</tr>
<tr>
<td>Piston type</td>
<td>Bowl-in-piston</td>
</tr>
</tbody>
</table>

Dynamometer is loaded by a Type, “S” load cell of 0.1 accuracy, calibrated before the tests, precisely. Test engine specifications are given in Tab.2. Tests are performed at full load condition and at 1200, 1400, 1600, 1800, 2000 and 2200 rpm engine speeds. The optimum static injection advance angle is obtained 29°CA bTDC, as a result of conducting several tests. And the obtained optimum static injection advance angle is used at tests. In Fig. 1, test bed is sketched.

For comparison, modified Wiebe function given in Eq.2, can be used in engine heat release analyses for improved results [16], providing parameter, θ50 for burnt fuel fraction at 50% [18]. Instead of form factor, a, θ50 is used to obtain closer results.

\[
MFB(\theta) = 1 - e^{-a\left(\frac{\theta-\theta_b}{\theta_0-\theta_b}\right)^{m+1}} 
\]

where, \(MFB\) is fraction of burnt fuel, \(a\) is form factor, \(\theta\) is instantaneous crank angle, \(\theta_0\) is crank angle for start of combustion, \(\theta_b\) is engine crank angle for burn duration and \(m\) is efficiency factor.

For heat release rate calculations, the first law of thermodynamics is utilized as given in literature [10]. Heat release rate data are obtained using in-cylinder pressure data and engine data as given in

2.3. Heat Release Analysis

Wiebe function, given in Eq.1, is frequently used in engine heat release analyses [16-17].

\[
MFB(\theta) = 1 - e^{-a\left(\frac{\theta-\theta_0}{\theta_50-\theta_0}\right)^{m+1}} 
\]

Figure 1 Engine test setup

During experiments, test engine is loaded by a 20 kW capacity electric dynamometer. Before testing, the oil temperature was measured 60 ±2°C. The Heat addition was kept constant, 17.3 mg fuel per cycle at 1100 rpm. Volumetric metering is used for fuel consumption measurements. Prior to experiments, test engine is run sufficient to maintain steady operating temperatures. In the closed circulation cooling system, cooling water outlet temperature was 85°C. Orifice plate – surge tank system is used for air consumption measurements. Kistler 6061B type water cooled piezoelectric sensor, using a Kistler 5011B charge amplifier and AVL make signal conditioner, software for pressure measurements are used for in cylinder pressure measurements, and encoder on the engine crankshaft for angular position detection. Pressure sensor is placed in a bored slot on cylinder head opening to combustion chamber. Pressure sensor technical specifications are given in Tab. 3. For the cylinder pressure and top dead center (TDC) data acquisition, National Instrument PCIe 6251 fast data acquisition card is used for the signals from charge amplifier and the magnetic pick-up.

Table 3 Pressure sensor technical specifications

<table>
<thead>
<tr>
<th>In-cylinder pressure sensor (6061B, water cooled, piezoelectric type)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Pressure measuring range [bar]</td>
</tr>
<tr>
<td>Sensitivity</td>
</tr>
<tr>
<td>Stable operating range [°C]</td>
</tr>
<tr>
<td>Natural frequency [kHz]</td>
</tr>
<tr>
<td>Overload capacity [bar]</td>
</tr>
</tbody>
</table>

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Eq. 3. Ratio of specific heats, $\gamma$, is considered 1.32 [17].

$$\frac{dQ}{d\theta} = \frac{\gamma}{\gamma - 1} P \frac{dV}{d\theta} + \frac{1}{\gamma - 1} V \frac{dP}{d\theta}$$

(3)

where, instantaneous cylinder volume,

$$V(\theta) = V_c + V_d(\theta)$$

(4)

and in Eq. 4, clearance volume, $V_c$ is measured from the engine cylinder. Instantaneous cylinder displacement volume [19],

$$V_d(\theta) = \frac{\pi B^2}{4} S(\theta)$$

(5)

Instantaneous piston stroke,

$$S(\theta) = R \left( (1 - \cos(\theta)) + \frac{L}{R} \left( 1 - \sqrt{1 - \left( \frac{R}{L} \right)^2 \sin^2(\theta)} \right) \right)$$

(6)

Derivative of cylinder volume,

$$\frac{dV(\theta)}{d\theta} = \frac{\pi B^2}{4} \frac{dS(\theta)}{d\theta}$$

(7)

Derivative of piston stroke,

$$\frac{dS(\theta)}{d\theta} = R \left( \sin(\theta) + \frac{R}{2L} \frac{\sin(2\theta)}{\sqrt{1 - \left( \frac{R}{L} \right)^2 \sin^2(\theta)}} \right)$$

(8)

3. RESULTS

In this section, engine performance parameters are investigated at engine speeds of 1200-2400 rpm range. Throughout the experiments, while engine speed increases, fuel mass flow rates decrease, but injection timings are kept constant at 29°bTDC. Experimental results are given in the Figs.2-7.

In Fig. 2, engine torques, force times the cantilever arm length from measured on loadcell are given for using diesel and B50 blend fuels at full load condition at testing speeds.

(a)

(b)
During engine tests, the Maximum Brake Torque (MBT) conditions are sustained. And the results are shown below. At low speeds, engine torque increases, and after reaching a peak value, it starts decreasing. Compared to B50, obtained torques are higher at low speeds for neat diesel fuel and after the maximum value, obtained torques are very close to neat diesel fuel torques. Maximum torque is obtained at 1600 rpm, as given in Fig.2.a. The reasons of decrease in engine torque at low loads are higher heat losses, and degradation in mixture formation and volumetric efficiency. The speed at maximum torque condition also corresponds to the speed at the maximum fuel conversion efficiency condition. Above this speed, engine torque decreases, because of the decrease in volumetric efficiency, increase in mechanical friction loss and insufficient time for combustion to complete. As seen in Fig.2.b, using B50 fuel decreases the torque up to 3%, compared to neat diesel fuel. Highest torque obtained is 49.91 Nm for neat diesel fuel.

As seen in Fig. 3.a, engine power increases with speed, and reaching a peak, then starts decreasing. Slightly higher powers are obtained by using neat diesel fuel, compared to B50 fuel. And, lowest reduction in power is 3% as given in Fig.3.b. Highest power obtained is 10.54 kW for neat diesel fuel.

Figure 3 (a) Engine brake power vs engine speed, (b) % Relative difference vs engine speed
Obtained BSFCs are shown in Fig. 4.a. For B50 fuel case, more fuel is consumed compared to neat diesel fuel, up to 6.4%. Lowest BSFC obtained is 254.58 g/kWh for neat diesel fuel.

![Figure 4](image_url)

Figure 4 (a) Engine brake torque vs engine speed for the fuels, (b) % Relative difference vs engine speed

In-cylinder pressure data is used in engine performance analyses. In Fig.5a in-cylinder pressure data versus crank angle is shown. When compared, pressure histories are close and similar. And, maximum pressures at MBT condition are depicted in Fig.5.b. Peak values are obtained around 1600 rpm for both fuels, and 79.1 bar for neat diesel fuel.

![Figure 5](image_url)

Figure 5 (a) In-cylinder pressure vs crank angle for the fuels, (b) Maximum pressures vs engine speed

In Fig.6, experimental heat release data are presented. Heat release rate curves start from the beginning of ignition and end with combustion. When compared, heat release at premixed combustion stage for neat diesel is less than B50.
fuel. Maximum heat release rate is obtained earlier for B50 fuel. As seen Fig. 7, the angles for MFB50 are pretty close. However, diesel fuel reaches MFB50 slightly before B50 fuel during combustion at 1600 rpm engine speed. In the evaluation of start and end angles for MFB, \( \theta_{5} \) and \( \theta_{95} \) are considered, respectively [18]. Cumulative heat release curves are calculated by using equations (3-8).

![Figure 6 Heat release rate at 1600 rpm vs crank angle for diesel (solid) and B50 (dashed) fuels](image)

Figure 6 Heat release rate at 1600 rpm vs crank angle for diesel (solid) and B50 (dashed) fuels

![Figure 7 Burned fuel mass vs crank angle for diesel (solid) and B50 (dashed) fuels](image)

Figure 7 Burned fuel mass vs crank angle for diesel (solid) and B50 (dashed) fuels

4. CONCLUSION

Biodiesels have very high potential to be blended or used as a substitute of diesel fuel. Today, most of the classification societies and legal authorities have issued standards for biodiesels, ASTM, EN etc. Therefore, a gate for a widespread of biodiesels is available.

In this study, biodiesel of sunflower oil, a common vegetable oil, is used to investigate the performance in comparison to neat diesel fuel. It is concluded that, torque, brake power and in-cylinder maximum pressure decrease by using biodiesel blend fuel, B50, and, BSFC increases with biodiesel, since SFME has lower LHV than diesel fuel. However, bearing in mind that, this discrepancy can be overcome by using cheaper biodiesel fuel production methods. Also, using biodiesel fuels is sustainable for being renewable.

Heat release rate data obtained is also presented. And, it is concluded that, for B50 fuel combustion premixed combustion stage is significant compared to neat diesel fuel. Premixed combustion stage is longer in B50 fuel than neat diesel fuel, while, diffusive combustion stage is longer for neat diesel fuel.

As a result, there is a difference in performance parameters compared to diesel fuel. However, this situation is not important considering the economic and diffusive potential of biodiesel fuel.

5. NOMENCLATURE

<table>
<thead>
<tr>
<th>Acronym</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>ASTM</td>
<td>American Society for Testing and Materials</td>
</tr>
<tr>
<td>B</td>
<td>Cylinder bore</td>
</tr>
<tr>
<td>B10</td>
<td>Blend fuel of 90% Diesel and 10% Biodiesel (by mass)</td>
</tr>
<tr>
<td>B20</td>
<td>Blend fuel of 80% Diesel and 20% Biodiesel (by mass)</td>
</tr>
<tr>
<td>B5</td>
<td>Blend fuel of 95% Diesel and 5% Biodiesel (by mass)</td>
</tr>
<tr>
<td>B50</td>
<td>Blend fuel of 50% Diesel and 50% Biodiesel (by mass)</td>
</tr>
<tr>
<td>B75</td>
<td>Blend fuel of 25% Diesel and 75% Biodiesel (by mass)</td>
</tr>
<tr>
<td>BSFC</td>
<td>Brake Specific Fuel Consumption</td>
</tr>
<tr>
<td>CI</td>
<td>Compression ignition</td>
</tr>
<tr>
<td>CO</td>
<td>Carbon monoxide</td>
</tr>
<tr>
<td>CO(_2)</td>
<td>Carbon dioxide</td>
</tr>
<tr>
<td>COME</td>
<td>Canola oil methyl ester</td>
</tr>
<tr>
<td>DEE</td>
<td>Diethyl ether</td>
</tr>
<tr>
<td>DI</td>
<td>Direct injection</td>
</tr>
<tr>
<td>EN</td>
<td>European Norm</td>
</tr>
<tr>
<td>HC</td>
<td>Hydrocarbons</td>
</tr>
<tr>
<td>HHV</td>
<td>Higher heating Value</td>
</tr>
<tr>
<td>L</td>
<td>Connecting rod length</td>
</tr>
</tbody>
</table>
6. REFERENCES


