Performance, combustion, and emissions in a diesel engine operated with fuel-in-water emulsions based on lignin

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HIGHLIGHTS

• Fuel-in-water emulsion was tested in a direct injection diesel engine.  
• Emulsion was stabilized by an industrial wood lignin with water as continuous phase.  
• Using effective fuel content, emulsion produces lower fuel consumption.  
• Pollutant emissions for emulsions are lower for most conditions than the base fuels.

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ABSTRACT

We report for the first time on the use of water-continuous emulsions stabilized by a bio-based macromolecule in a compression-ignition diesel engine and compare their performance, combustion and emissions against the base fuels (diesel, biodiesel, and jet fuel). For this purpose, high internal phase ratio emulsions (70:30 fuel-to-water) were produced by mechanical emulsification using carboxymethylated wood lignin as stabilizer. Combusting experiments were performed with the engine operating at 2000 rpm under three loads (0, 1.26 and 3.26 bar brake mean effective pressure, BMEP). Engine performance, in-cylinder combustion, and exhaust emissions were monitored and compared for the fuels tested. At no load condition and when compared to the respective base (single phase) fuels, an increase in the indicated work was observed for diesel and biodiesel emulsions. Compared to the base fuels, the emulsions resulted in higher engine mechanical efficiency at 1.26 and 3.26 bar BMEP except for jet fuel emulsion at 1.26 bar. Additionally, they displayed a lower brake specific fuel consumption (BSFC), if calculated on the basis of effective fuel content discounting emulsion water, and higher brake thermal efficiency. Compared to the base fuel, the respective emulsions generally presented lower peak in-cylinder pressure, lower heat release rates, and longer ignition delays at 1.26 bar and 3.26 bar BMEP; the opposite effect was observed at no-load conditions. Remarkably, a large reduction of nitrogen oxides (NOx) emissions was noted in the combustion of the fuel emulsions, which was accompanied with a relatively higher carbon monoxide (CO) release at 1.26 and 3.26 bar (at 0 bar BMEP, the emulsions produced less CO emissions). The effect of emulsions on hydrocarbon emissions and smoke opacity depended on the fuel type and the engine load. Overall, it is concluded that while reports on fuel emulsions involve oil-continuous systems, the proposed water-continuous alternative represents an opportunity for diesel engines, whereby the fuel is dispersed as micrometric droplets for improved combustion and reduced emissions. At the same time, the fuel emulsion formulation takes advantage of the surface activity and high calorific value of widely available, inexpensive lignin stabilizers, making the proposed system a viable option towards cleaner or fully bio-based fuels.

1. Introduction

The US Environmental Protection Agency (EPA) is set to introduce new motor vehicle emission standards starting in 2017. The program will seek to lower vehicle emissions and reduce the sulfur
content of gasoline. Moreover, it is estimated that by 2030 it will reduce vehicle nitrogen oxides (NOx) emissions by 25%, carbon monoxide (CO) emissions by 24% and direct particulate matter (PM) by 10% [1]. Engine manufacturers will need to adopt methodologies to meet the requirements of the new standard. Methods typically used to reduce emissions for diesel engines consider use of alternative fuels, introduction of water into the combustion chamber and application of measures to tackle individual controlled emissions such as particulate filters and oxidation catalysts. Although effective in reducing engine emissions, some of these systems are not cost effective, may require extensive modifications to the vehicle, and, in some cases, emissions are reduced at the cost of engine performance. After treatment systems in use particularly for diesel engines need to be well maintained for the engines to continue to meet emissions requirements. Biofuels as alternative fuels, on the other hand, can be very effective in reducing engine emissions, coupled with other benefits including energy security, lower toxicity, higher lubrication, local availability, and sustainability [2].

The use of emulsions of water and fuel can be an effective means by which regulated pollutant emissions can be reduced. The addition of water to fuel has the additional benefit of reducing the fuel consumption. An emulsion is a mixture of two or more liquids in which one (the dispersed phase) is present as droplets, of microscopic or ultramicroscopic size, stabilized and distributed throughout the other, continuous phase [3]. It is created by the agitation of two fluids in the presence of amphiphile molecules (surfactants, for example) that typically contain both polar and non-polar moieties so that they adsorb and stabilize the interface and reduce the interfacial tension, leading to kinetically-stable multi-phase systems [4]. Fuel emulsions are produced when fuel and water are mixed together in the presence of a surfactant with either water as the continuous phase and the fuel as the dispersed phase or vice versa. A number of reports exists on the use of emulsions containing waste cooking oil [5], animal fat [6], biofuel [7] and diesels [8–13]. Three – phase emulsions [14–16] and diesel-ethanol emulsions have been tested in diesel engines [17]. Investigations were also performed on the use of heavy oil – water emulsions on an industrial scale [18,19].

Kumar et al. performed experiments in a diesel engine using waste cooking oil (WCO)/water emulsions and observed improvements in the brake thermal efficiency (BTE) at high power outputs but low BTE at low power outputs when compared to the baseline fuel [5]. Lower smoke opacity than WCO but higher smoke opacity than diesel fuel was also observed. Hydrocarbon (HC) emissions varied with power output. At high power output, WCO emulsion had lower HC emissions than diesel but at low power output the opposite effect was observed. The same authors carried out additional experiments in a diesel engine using an emulsion of animal fat and water with ethanol [6]. They observed a decrease in smoke, NOx, HC and CO when compared to pure animal fat and neat diesel at high power outputs. Animal fat emulsions were observed to have higher HC and CO emissions at light engine loads. A comprehensive review on the use of biofuel emulsions in diesel engines was also published [7]. A leading conclusion in the related works is that biofuel emulsions improved the combustion efficiency and performance with reduced NOx and PM emissions. Research work in Refs. [8–13] showed a reduction in NOx, soot, HC and PM emissions, and an improvement of combustion efficiency for diesel-water emulsions. Alahmer et al. observed that as water content increased, BSFC increased at high engine speeds, BTE decreased, exhaust gas temperature (EGT) decreased, and NOx also decreased [10]. In the case of experiments by Zaid et al., an increase in water content resulted in an increase in BTE, an increase in BSFC (diesel and water considered as total fuel) and a decrease in EGT [12]. Chering-Yuan et al. [14–16] carried out a series of studies with three-phase diesel emulsions in a diesel engine and indicated an increase in BSFC and CO with a decrease in NOx, CO2 and smoke opacity. Varying concentrations of diesel-ethanol emulsions were also tested in a diesel engine [17]. The results showed that as the volume of ethanol was reduced an increase in BTE, a decrease in specific fuel consumption and a decrease in NOx occurred. The general consensus from experiments carried out so far with emulsions in a diesel engine is that they lead to reduced NOx and smoke emissions. Yang et al. [20] performed experiments in a diesel using an emulsion of diesel, water, organic oxygenated additives and NP-9 surfactant. They observed an improvement in BTE, NOx and noticed a longer ignition delay for the emulsions. Chang et al. [21] used emulsions made of water containing acetone – butanol – ethanol and diesel fuel in the absence of a surfactant at different concentration in a diesel engine. From their work, they observed an increase in BTE and a decrease in PM, NOx and PAHs. Additional experiments carried out in a natural gas fueled engine using water in biodiesel emulsion [22] resulted in increased HC and CO emissions with very subtle changes in NOx emissions. Research carried out to study the effect of water concentration on cylinder head deposits [23] showed significant decrease in carbon and hydrogen deposits as water content in emulsions is increased.

Micro-explosion and puffing are the two most important phenomena attributed to emulsion combustion in a diesel engine. Micro-explosion is the quick break down of droplets, which results in secondary droplet atomization [24]. It occurs because water and oil have different boiling temperatures, which results in water evaporating much faster than the oil in a hot combustion chamber. In water-in-oil emulsions, water droplets are the dispersed phase, surrounded by oil. In a hot combustion chamber, the water droplets will reach superheating much faster than the oil thus resulting in micro-explosion. In the case of puffing, water leaves the oil droplets as a fine mist [4,24]. Not all researchers have, however, agreed on the occurrence of micro-explosions in diesel engines as was discussed by Weibiao et al. [8].

Most of the experiments carried out in diesel engines using fuel-water emulsions have typically used water as the dispersed phase with the oil (fuel) as the continuous phase. In this work, the opposite emulsion type was used, namely, oil-in-water emulsions that were formulated by using diesel, jet fuel and biodiesel as base fuels (emulsion oil phase). It should be noted that this is the first demonstration of the utilization of oil-in-water fuel emulsions in a diesel engine. Importantly, the stability of the emulsions was attained by incorporation of a modified technical lignin (carboxymethylated Kraft lignin), which acted as amphiphile molecule or polymeric surfactant. Therefore, the main objective of this work is to study the feasibility of oil-in-water emulsions stabilized by an inexpensive and widely available biomolecule as fuels for diesel engines.

2. Experimental

2.1. Fuels and fuel emulsions

Ultra-low sulfur diesel fuel (No. 2 diesel) was used in the experiments. It was obtained by blending diesel fuels acquired from three different gas stations at the same ratio by volume. Biodiesel was obtained locally from a biodiesel plant and it was produced from waste cooking oil. The biodiesel had a cetane number of 55.4. Jet fuel was purchased locally from a civil airport.

Industrial Kraft lignin with low ash content (0.5%) was kindly donated by Domtar Inc. (Plymouth Pulp Mill, NC). The lignin molecular weight was ca. 6200 Dalton and its methoxy group content was measured to be 11.5% (NMR). Elemental analysis...
indicated a sulfur content of ~1.5 wt%. The lignin was carboxymethylated to facilitate the emulsification of the fuel in the water-continuous phase. The lignin was initially alkalinized with sodium hydroxide, forming ionized nucleophiles and then reacted with monochloroacetic acid to produce the carboxymethylated lignin used to stabilize the emulsions. The carboxymethylated lignin degree of substitution (35%) was measured by using a Bruker 300 MHz 31P NMR spectrometer. In order to produce and stabilize the emulsions, 0.5% of carboxymethylated lignin was dissolved in water before mixing with the fuel. The mixture was then mechanically agitated at room temperature with an Ultra-Turrax operating at 20,000 rpm. The produced emulsion was subject to a vacuum pump for de-aeration for 30 min. Fuel properties including, higher heating value (oxygen bomb calorimeter, Parr Instrument Company, Moline, IL) and dynamic viscosity (rheometer, AR-2000, TA Instruments, New Castle, DE), are given in Table 1. From this table it can be observed that the oil-in-water fuel emulsions have higher dynamic viscosities than the base fuels. There is a direct relationship between higher heating value (HHV) and lower heating value of a fuel. HHV was directly measured and thus listed in Table 1. The fuel emulsions were non-Newtonian, specifically, they displayed shear thinning behavior and therefore the dynamic viscosities are reported as apparent viscosities. Under the high shear rates typical of flowing systems in diesel engines, the viscosity is suitable for pumping, spraying and related operations. Fig. 1 includes images of the tested fuel and fuel emulsion samples.

In order to characterize the fuel components, samples of the three base fuels were measured in a gas chromatograph/mass spectrometer (GC/MS) system (Shimadzu). To perform the analysis, one micro liter of sample is injected into the GC at an injection temperature of 230 °C with the oven temperature set at 35 °C for 3 min before increasing to 270 °C at a rate of 10 °C/min for 60 mins. A Petrocol DH column was used for the analysis. A Total Ion Chromatogram (TIC) which is the sum of the intensities of all mass spectral peaks belonging to the same scan was then generated from the results of the analysis [25]. The GC/MS results for the three fuels are shown in Fig. 2a–c. From the GC/MS analysis, diesel is found to consist of C8 – C27 hydrocarbons. Biodiesel is made up of C15 – C21 hydrocarbons and the jet fuel used is made up of C8 to C16 hydrocarbons. The range of hydrocarbons for jet fuel indicates that it may have the lowest molecular weight. Due to its lower molecular weight, jet fuel is expected to have a lower boiling point and vaporize relatively easily. Considering its average hydrocarbon chain length, it is expected that jet fuel will have a reasonably high cetane number [26].

As indicated before, the emulsions used in the experiments contained water as the continuous phase and oil (fuel) as the dispersed phase. This is in contrast to the typical emulsions used in diesel engines, which contain water as the dispersed phase. The water-to-oil ratio used was fixed at 30:70 by mass, thus High Internal Phase Ratio (HIPR) emulsions were the subject of our work.

2.2. Engine setup

The combustion experiments were conducted in a 7.46 kW (10 hp) single-cylinder air-cooled compression-ignition engine (Model CS-186 from Carroll Stream Motor Company, Inc.). The engine had a bore of 86 mm, a stroke of 72 mm, a displacement of 418 cc, and a compression ratio of 19:1. The engine had an o-shape combustion chamber in the piston. The engine was coupled to a Go-Power water brake dynamometer. The engine had a jerk-type mechanical fuel injection system with an initial injection pressure of 19.6 MPa. The fuel injector had four orifices with a model number of DSLA150PN926. The engine specifications are listed in Table 2 and a schematic and a photo of the experimental setup are given in Figs. 3 and 4, respectively. In-cylinder combustion pressure was measured using a Kistler 6052A pressure sensor. A Hall-effect sensor and a shaft encoder (Hengstler 0521097) were used in combination to determine the engine’s top dead center (TDC) position as well as the engines crank angles (CADs). A M5100 series pressure transducer was used to measure load cell pressure to calculate the engine load. Air mass flow rate was measured with a Bosch air mass flow sensor. Fuel mass flow rate was obtained by measuring the mass of the fuel at certain time intervals during engine operation with an OHAUS GT2100 Scale. Atmospheric pressure was measured using a SSI tech pressure transducer. The intake and exhaust temperatures were measured with K-type thermocouples. Data from each sensor was sampled by a NI PCI-MIO-16E-4 data acquisition board controlled by a custom Labview program. Further analysis of data was done using Matlab and Microsoft Excel. Gaseous emissions were measured with an exhaust gas analyzer (FGA 4000XDS from infrared industries). The measurement principle and accuracy of gaseous emissions were summarized in our previous publication [27] and are not included here for brevity. Smoke opacity measurement was taken at the end of the exhaust tail pipe with a smoke meter (Wager 6500). The exhaust tail pipe ends with a T junction to allow for an increase in the light path length of the smoke meter, which helps with the measurement of low smoke under low load conditions. A schematic of the smoke meter setup and the sensor specifications can be found in [27]. Six data sets were taken at each steady state point for the five gaseous emissions measured. Four opacity readings were taken for each steady state point.

**Table 1**

<table>
<thead>
<tr>
<th></th>
<th>Diesel</th>
<th>Diesel emulsion</th>
<th>Biodiesel</th>
<th>Biodiesel emulsion</th>
<th>Jet fuel</th>
<th>Jet fuel emulsion</th>
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<tr>
<td>HHV (MJ/kg)</td>
<td>45.3</td>
<td>38.27</td>
<td>26.69</td>
<td>44.73</td>
<td>29.43</td>
<td></td>
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<tr>
<td>Dynamic viscosity @ 25 °C (mPa-s)</td>
<td>1.99</td>
<td>4.12</td>
<td>1.15</td>
<td>1.5</td>
<td>1.15</td>
<td></td>
</tr>
<tr>
<td>Cetane number</td>
<td>46</td>
<td>55.4</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td></td>
</tr>
<tr>
<td>Density (kg/l)</td>
<td>0.83</td>
<td>0.88</td>
<td>-</td>
<td>0.79</td>
<td>-</td>
<td></td>
</tr>
<tr>
<td>Surface tension (mN/m)</td>
<td>30.3</td>
<td>34.7</td>
<td>-</td>
<td>28.7</td>
<td>-</td>
<td></td>
</tr>
</tbody>
</table>

**Fig. 1.** From left to right, diesel, biodiesel, jet fuel, diesel emulsion, biodiesel emulsion, jet fuel emulsion.
Diesel, biodiesel, jet fuel, and their corresponding emulsions were tested under given conditions. The diesel engine was set at 2000 rpm for three different torque outputs of 0, 4.18 and 10.86 Nm corresponding to 0, 1.26 and 3.26 bar brake mean effective pressure (BMEP). Two fuel tanks containing a base fuel and the emulsified fuel were used. The two tanks each had a shut off valve and were connected with a T junction which fed into the engine’s fuel pump. The testing of the emulsified fuels began by starting the engine with the base fuel while keeping the valve from the tank containing the emulsion closed. The engine was left to run for 30 min to warm up. With the engine sufficiently warmed up, the base fuel valve was shut and the emulsion valve was opened. The engine was further left to run on the emulsion fuel for an extended period of time in order to ascertain that no base fuel was present in the fuel line before data acquisition began. It should be noted that for this experiment, the fuel supply timing for all fuels tested was kept constant at 17 degree before TDC as listed in Table 2. Data was collected after the engine had sufficiently warmed up and had become stable. Fuel mass flow rate was collected over an interval of 1 min for 10 min. Twenty-five firing cycles of in-cylinder pressure data were collected and smooth over

<table>
<thead>
<tr>
<th>Engine specifications.</th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Engine</td>
<td>Single cylinder, vertical, direct injection 4 stroke</td>
</tr>
<tr>
<td>Compression ratio</td>
<td>19:1</td>
</tr>
<tr>
<td>Bore × stroke</td>
<td>86 mm × 72 mm</td>
</tr>
<tr>
<td>Method of loading</td>
<td>Water brake dynamometer</td>
</tr>
<tr>
<td>Method of starting</td>
<td>Electric start</td>
</tr>
<tr>
<td>Method of cooling</td>
<td>Air cooled</td>
</tr>
<tr>
<td>Type of ignition</td>
<td>Compression ignition</td>
</tr>
<tr>
<td>Rated power</td>
<td>7.46 kW</td>
</tr>
<tr>
<td>Rated speed</td>
<td>3600 rpm</td>
</tr>
<tr>
<td>Initial injection</td>
<td>17.0 Crank angle degrees before TDC</td>
</tr>
<tr>
<td>Displacement</td>
<td>418 cc</td>
</tr>
<tr>
<td>Fuel consumption at rated power</td>
<td>340 g/kWh</td>
</tr>
</tbody>
</table>

Diesel, biodiesel, jet fuel, and their corresponding emulsions were tested under given conditions. The diesel engine was set at 2000 rpm for three different torque outputs of 0, 4.18 and 10.86 Nm corresponding to 0, 1.26 and 3.26 bar brake mean effective pressure (BMEP). Two fuel tanks containing a base fuel and the emulsified fuel were used. The two tanks each had a shut off valve and were connected with a T junction which fed into the engine’s fuel pump. The testing of the emulsified fuels began by starting the engine with the base fuel while keeping the valve from the tank containing the emulsion closed. The engine was left to run for 30 min to warm up. With the engine sufficiently warmed up, the base fuel valve was shut and the emulsion valve was opened. The engine was further left to run on the emulsion fuel for an extended period of time in order to ascertain that no base fuel was present in the fuel line before data acquisition began. It should be noted that for this experiment, the fuel supply timing for all fuels tested was kept constant at 17 degree before TDC as listed in Table 2. Data was collected after the engine had sufficiently warmed up and had become stable. Fuel mass flow rate was collected over an interval of 1 min for 10 min. Twenty-five firing cycles of in-cylinder pressure data were collected and smooth over

![Fig. 2. GC/MS results for (a) diesel (b) biodiesel (c) Jet fuel.](image-url)
a span of 20. It should be noted that in the heat release analysis the ratio of the constant pressure specific heat to the constant volume specific heat ($\gamma$), was kept constant at 1.35 for all the tested conditions [26,28]. Much more advanced means of determining heat release rate from in-cylinder pressure can be found in [29].

3. Results and discussion

3.1. Performance

Engine performance parameters for each tested fuel are included in Fig. 5. Fig. 5a shows the indicated work at the three different loads. As engine load is increased, the indicated work increases for all the fuels. The fuel emulsions have lower indicated work than the base fuels with the exception of the no load condition and jet fuel emulsion at 1.26 bar. Fig. 5b depicts the engine’s mechanical efficiency. Mechanical efficiency is observed to increase with increasing engine load for all tested fuels. Mechanical inefficiency is typically attributed to friction losses. Fuel emulsions result in higher mechanical efficiency. Jet fuel emulsion at 1.26 bar is the only exception with a lower mechanical efficiency than the base jet fuel. Fig. 5c shows the brake specific fuel consumption (BSFC) for different engine loads. For this figure, both water and oil contribute to the fuel consumption. BSFC determined in this manner decreases with increasing engine load and is higher for the emulsions when compared to their base fuels. The higher BSFC of emulsions is mainly due to their lower heating values due to the presence of water. Since water has no contribution to the heating values of the emulsions, it is reasonable to only consider the fuel used in the engine operation. Fig. 5d is a plot of BSFC determined by only considering the fuel in the emulsion. The percentage of fuel used in this experiment is 70% by mass and as such only 70% of the total fuel emulsion flow rate is used in determining the adjusted BSFC. By determining the BSFC in this manner, the fuel emulsions are observed to have noticeably lower BSFC values when compared to the base fuels.

The brake thermal efficiency (BTE) is a ratio of the brake work per cycle to the chemical energy released in the combustion process [28]. From Fig. 6a, BTE increases with increasing engine load for all fuels, it also increases for the fuel emulsions when compared to their base fuels, which is consistent with other findings [9,10 and 12]. This difference in BTE becomes more significant when only the actual base fuel is used in the calculation as is shown in Fig. 6b. In this figure, the brake thermal efficiency is adjusted to the quantity of base fuel used.

3.2. Combustion

Combustion pressure was obtained using a Kistler 6052A pressure sensor installed in the cylinder head of the engine. Fig. 7a
shows the in-cylinder pressure at no load conditions (0 bar BMEP). It is seen that the fuels and their emulsions have similar pressure profiles with the exception of the jet fuel and its emulsion. This figure suggests a longer ignition delay and slower rate of pressure rise for jet fuel emulsion, diesel and diesel emulsion compared to biodiesel, biodiesel emulsion and jet fuel. In Fig. 7b, a major difference in pressure profiles is observed for jet fuel and its emulsion compared to the other fuels. An initial decrease in cylinder pressure after fuel injection followed by a slight increase in the cylinder pressure upon ignition can be determined from this figure for jet fuel and its emulsion with the observation been more pronounced for jet fuel emulsion. It is also observed that fuel emulsions have lower in-cylinder pressures than their corresponding base fuels with biodiesel and its emulsions still having a faster rate of pressure rise than diesel, jet fuel and their emulsions. The initial decrease in pressure after the start of fuel injection for jet fuel and its emulsion may be attributed to a higher volatility and longer ignition delay. A long ignition delay means fuel would accumulate in the combustion chamber for a longer period, which would lead to significant cooling of the air-fuel mixture as a result of fuel evaporation.

At 3.26 bar, the effect of water evaporation are much more pronounced, as shown in Fig. 7c. Emulsions have much lower peak in-cylinder pressures than their base fuel counterparts and major differences between the pressure profiles of the base fuels and their emulsions can be observed. Unlike the lower load conditions, where biodiesel has the fastest rate of pressure increase, at 3.26 bar diesel has the fastest rate of pressure increase after the start of combustion. An additional observation is that while there is still a noticeable decrease in the combustion pressure for jet fuel, there is a significant increase in the in-cylinder pressure shortly after the start of combustion, possibly as a result of longer ignition delay and low volatility of jet fuel. However, jet fuel emulsions show a very long ignition delay compared with other base fuels and emulsions.

In Fig. 8a it is found that with the exception of jet fuel and its emulsion, at no-load condition there is no significant difference between the heat release rate profile of a pure fuel and its emulsion counterpart, which are shown to have completely different profiles. The heat release rate differences observed for jet fuel and its emulsions may be attributed to spray development and ignition quality resulting in significant differences in their ignition delays.
Diesel, its emulsion and jet fuel emulsion have a noticeably longer pre-mixed combustion phase when compared to the other fuels as a result of the longer ignition delay of these fuels. A longer ignition delay would result in more fuel present in the combustion chamber at the time when ignition finally takes place. The increased quantity of fuel in the premixed mixture would result in a stronger pre-mixed combustion phase with a higher peak heat release rate as is observed.

Fig. 8b shows a marked difference in the heat release profiles of all the fuels tested at 1.26 bar BMEP. Biodiesel and its emulsion are observed to have different mixing – controlled combustion phase when compared to the other fuels. The heat release rate between a fuel and its emulsion differs depending on the fuel type. While biodiesel and its emulsion have very similar heat release rate profiles, diesel, jet fuel and their emulsions have much larger values. These observed differences may again be attributed to differences in heating values, spray development, and ignition quality. In Fig. 8c, the engine load is increased to 3.26 bar BMEP. Jet fuel emulsion clearly stands out with a significantly longer ignition delay when compared to the other fuels. Biodiesel emulsion is noted to have a significantly longer pre-mixed combustion phase with a relatively lower heat release rate peak. Lower heating value, low ignition quality, and poor fuel atomization are believed to be the possible reasons accounting for the behavior of biodiesel emulsion in the pre-mixed combustion phase.

Fig. 9a depicts the cumulative heat release of the fuels at no load condition. It shows the same fuel grouping observed from the pressure and heat release rate plots. The profiles of the fuels are similar. Jet fuel and its emulsion experience overall lower cumulative heat release when compared to the other fuels. Fig. 9b shows cumulative heat release as engine load is increased to 1.26 bar. From this figure, we again notice a significant ‘scattering’ of the profiles. The high cumulative heat release associated with jet fuel and its emulsion can be attributed to their significantly longer ignition delay with more premixed combustion and probably less heat loss to the wall during combustion. In Fig. 9c, the cumulative heat releases of all tested fuels are similar regardless of the ignition delay under 3.26 bar BMEP. This is believed to be because of the fat that as engine load increases the effect of heating value and ignition quality are no longer as pronounced in a warmer engine cylinder.

Fig. 10a gives a quantitative comparison of the peak in-cylinder pressure for all the tested fuels. It is seen that a general decrease in peak in-cylinder pressure at 1.26 and 3.26 bar for emulsions compared to their base fuels. At no load condition, however, emulsions have a higher peak in-cylinder pressure except for jet fuel emulsion. The increase in pressure is believed to be as a result of enhanced fuel atomization caused by micro-explosion. In the case of jet fuel emulsion, the ignition delay is significantly longer than neat jet fuel leading to a lower peak pressure. At higher engine loads, the effects of a lower ignition quality, longer ignition delay and lower heating values surpass the effects of micro-explosions hence the lower peak cylinder pressure for fuel emulsions. In Fig. 10b, a similar trend as that shown in Fig. 10a is observed mostly for the peak heat release rate for the same reasons. The exceptions are jet fuel emulsion at no load condition and biodiesel emulsion at 1.26 bar. Jet fuel emulsion has a longer ignition delay. The long ignition delay results in more fuel in the chamber.
available for combustion, and as such there is a higher heat release rate upon ignition than its corresponding base fuel. The slightly longer ignition delay of biodiesel emulsion is responsible for its slightly higher peak heat release rate.

Ignition delay comparisons are shown in Fig. 10c. Emulsions are observed to have longer ignition delays than their base fuels for the most part. Ignition delay is closely related to the ignition quality of the fuel, and for this reason, the emulsions which are expected to have lower cetane numbers (ignition quality) will have longer ignition delays [26]. The only exception observed is biodiesel emulsion at 1.26 bar which has a shorter ignition delay of 0.1 deg than pure biodiesel. Biodiesel and its emulsion generally have shorter ignition delays than other fuels or emulsions. Although the heating values of biodiesel and its emulsion are lower than jet fuel and its emulsion, the higher bulk modulus of biodiesel and its emulsion would result in earlier fuel injection, which leads to a shorter ignition delay.

3.3. Emissions

Fig. 11a shows the HC emissions for the fuels. The general trend observed for most of the tested fuels is an increase in HC emissions between no load condition and 1.26 bar followed by a decrease upon increasing engine load to 3.26 bar. The decrease in HC emissions of the diesel and biodiesel emulsions, when compared to their based fuels at no-load conditions may be due to better atomization as a result of micro-explosion. Oil-in-water emulsions experience micro-explosion in the same manner as water-in-oil emulsions [30]. This is believed to be caused by a phase inversion from O/W (oil-in-water) to W/O (water-in-oil) emulsions followed by a phase separation as a result of an increase in droplet temperature. In addition, it has been suggested that as micro-droplet component, the base fuel vaporizes first for O/W emulsions. The effect of a longer ignition delay would seem to outweigh the micro-explosion phenomena in the case of jet fuel, hence its higher
HC emissions when compared to neat jet fuel. In general, the effect of ignition delay as well as cylinder temperature seems to be the major factor affecting HC emissions. Fuel emulsions generally have lower combustion temperatures, which in turn results in higher HC emissions. An increase in ignition delay results in an increase in HC emissions as a result of over mixing, which results in a very lean mixture that will not support autoignition with misfire under certain conditions [26].

CO and carbon dioxide ($CO_2$) emissions are typically opposite sides to a coin. With more complete combustion, $CO_2$ emission is prominent and CO emissions are low; the reverse is true with incomplete combustion. Fig. 11b and c are plots of CO and $CO_2$ emissions, respectively. For all the fuels tested, CO emissions decrease with increasing engine loads. The reasons for this are an increase in fuel consumption and a warmer cylinder under higher loads which would support more complete combustion. At loads of 1.26 bar and 3.26 bar BMEP, CO emissions for emulsions increase when compared to pure fuel mostly because of the effect of water in the chamber which lowers the flame temperature [30]. A significant increase in water vapor concentration lowers the cylinder temperature and hinders complete combustion. This is, however, not the case at no load conditions where with the exception of diesel emulsion where less CO emissions are observed for the emulsions. The reason for this is attributed to micro-explosion which allows for secondary atomization with better fuel–air mixing. From Fig. 11c, the general trend for all the fuels is an increase in $CO_2$ emissions as engine load increased. Emulsions also have a noticeable decrease in $CO_2$ emissions, as supported by Fig. 11b.

From Fig. 11d, NO$_x$ emissions increase for all fuels as engine load increased. Except at no load condition and biodiesel at 1.26 bar BMEP, we observe a decrease in NO$_x$ emissions for emulsions when compared to their base fuels. NO$_x$ in engine combustion chamber is typically formed as thermal NO$_x$ in internal combustion engines. The reduced in-cylinder temperature at 1.26 bar and 3.26 bar for emulsions helps the reduction of thermal NO$_x$. According to Javier et al. [18] NO$_x$ formation is reduced as a result of a lower flame temperature as well as in increase in OH radicals which would react with excess oxygen thus reducing the amount of NO$_x$ formed. The decrease in NO$_x$ for emulsions is supported by previous results [4,6,7,9,13–16,18]. There is a correlation between peak heat release rate and NO$_x$ for diesel, biodiesel and their emulsions as shown in Fig. 12. This correlation supports that faster fuel burning in the combustion chamber could generate high in-cylinder temperature to cause high NO$_x$ formation. Due to the NO frozen effects in engine cylinder, the formed NO during fast combustion period would go to the exhaust with part of the formed NO being oxidized to NO$_2$ [26].

Fig. 13 depicts the effect of engine loads on smoke opacity. At no load conditions, in general a decrease in smoke opacity is observed for fuel emulsions with the exception of biodiesel emulsion. An increase in engine loads to 1.26 bar results in the opposite effect as that observed at no load condition with biodiesel again the exception. As engine load is increased further to 3.26 bar, an increase in smoke opacity is observed for the emulsions compared to their base fuels with jet fuel been the exception in this case. The initial decrease in smoke opacity for the emulsions could be as a result of the overall decrease in hydrocarbon fuel quantity as compared to pure fuels. A decrease in pure fuel quantity would imply lower carbon content and as such less smoke formation.
As micro-explosion is influenced by combustion temperature, it is expected that as engine load and cylinder temperature increase, fuel emulsion should undergo better atomization from micro-explosion, which would facilitate better air-fuel mixing and more complete combustion. The improvement in combustion as a result of micro-explosion is outweighed by poor combustion due to a longer ignition delay for the emulsions with the exception of biodiesel and jet fuel emulsions at 1.26 bar at 3.26 bar, respectively. Opacity results obtained from this experiment seem to be quite scattered. This could be due to the fact that all the smoke opacity results have very low values, which are close to the detection limits of the smoke meter at very low smoke emissions.

The exhaust gas temperatures (EGT) are shown in Fig. 14. The exhaust temperature difference between a fuel and its emulsion seems to be dependent on the fuel type and engine load. At no-load conditions, emulsions have higher EGT than their base fuels. As engine load increases to 1.26 bar, emulsions have lower EGT except for diesel emulsion. At 3.26 bar, emulsions have lower EGT than their base fuels with the exception of jet fuel emulsion. In general, longer ignition delays result in higher exhaust temperature. However, the water content in the fuel can help reduce the combustion temperature and thus the exhaust temperature. The EGT results for fuel emulsions can be attributed to the combined effects of the two factors.

In the end, it should be noticed that the reported experiments with lignin-stabilized fuel-in-water emulsions in a single cylinder diesel are preliminary. Additional experiments would need to be carried out to help further explain the results obtained. It should also be noted that the experiments were limited to low load conditions due to injection limitations from the use of fuel emulsions.

4. Conclusion

The effect of different fuel-in-water emulsions stabilized with a Kraft lignin was studied in a compression-ignition diesel engine. Engine performance, fuel combustion and emissions were measured. Based on the results obtained, some important findings can be summarized in the following:

1. When compared to the base fuels, the fuel emulsions resulted in a general decrease in indicated work and an increase in mechanical efficiency. There is a noticeable increase in BSFC and BTE for the emulsions when compared to their base fuels. But adjusted BSFC is generally lower for the fuel emulsions.

2. Peak cylinder pressure, peak heat release rate and peak cumulative heat release are observed to be higher for the fuel emulsions when compared to their base fuels. Fuel emulsions have longer ignition delays.

3. Emissions results vary with fuel type and engine operating conditions. At high loads, for instance, NOx emissions are lower for all the emulsions but CO emissions are higher. At low engine loads, fuel emulsions have higher NOx but lower CO emissions. For most of the fuel emulsions, HC emissions increase as the engine load increases.

4. This work is a first demonstration of using fuel-in-water fuel emulsion in a real diesel engine. The results are preliminary and limited to three different loads and one engine speed in a single cylinder diesel engine. Results may differ for other engine speeds and loads in multi-cylinder diesel engines. More experiments are needed for a comprehensive understanding of the effects of the proposed fuel-in-water emulsions.

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