On the possibilities to reduce compression ignition engine emissions by controlling the reactivity of diesel-biofuel mixtures

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Abstract. In most European countries the use of different biodiesel fuels has been proved to be a sustainable method to reduce the pollutant emissions of the operating Diesel engines. An in-debt analysis of combustion characteristics is necessary to give a detailed perspective on the advantages of biodiesel use in compression ignition engines. Among all the essential parameters characterizing the compression ignition engine operation, the ignition delay is the most difficult to be estimated. This parameter offers the possibility to have more control over the combustion process inside the cylinder. This gives the ability to better quantify and ultimately to decrease the pollutant emission for this type of engine. An experimental study has been conducted on a tractor Diesel engine using different biofuel mixtures with 7% and 20% volumetric fractions of fatty acid methyl esters in Diesel fuel. No modifications were made to the fuel injection system in order to have a better image of the direct impact of biodiesel fuel on performance and emissions of the engine. The obtained results showed some reductions of total unburned hydrocarbons and nitrogen oxide emissions over the entire engine speed domain for both tested biodiesel fuels. The estimation of the ignition delay was experimentally carried out and correlated with the properties of the biodiesel fuel; a proper control of the combustion process by fuel reactivity ensured thus the mitigation of carbon monoxide, carbon dioxide and smoke. The link between the experimentally determined ignition delay, mainly it’s reduction, and fuel reactivity is what the present work tries to analyse. A better evaluation of the ignition delay parameter, shorter when using biodiesel mixtures, should lead to better control of combustion and further lowering the exhaust pollutant emissions, without major impact on engine performance. An in-debt analysis of the combustion process starting from the injection characteristics correlated with the fuel reactivity is necessary for shortening the ignition delay and reducing carbon dioxide emissions.

Keywords: Biofuels, Reactivity, Diesel engine, Pollutant emissions.

1. Introduction
Due to the scarcity of fossil fuels the need for alternative fuel utilisation has become a global priority. The estimated oil reserves at the end of 2018 were 1730 billion barrels which accounted for 50 years of current production [1]. Also, a major problem that the world is facing is pollution and global warming. The transportation sector represents, next to the industrial sector, one of the great pollutant emissions generators.
The restrictive norms adopted by governing bodies do not have a broad perspective concerning the long-term effect of these restrictions. For example, the shift from Diesel to Gasoline engines has led to the highest average rise in carbon dioxide (CO₂) emissions since 2014 [2]. Two essential properties of the fuel used in compression ignition engines are the cetane number and viscosity. The cetane number represents the ease of a fuel to auto ignite and a higher value implicates a better combustion which in turn leads to a better functioning of the engine, better cold start performance, low noise and a global reduction in pollutant emissions and carbon deposits. In the case of viscosity, when there is a high value of this parameter, an insufficient fuel jet atomisation occurs, which can lead to engine deposits, especially on the piston rings. The compression ignition engine works mainly with lean fuel mixtures leading to lower carbon monoxide (CO) and unburnt hydrocarbon (THC) emissions. There exists a need for an alternative fuel for use in compression ignition engines, which has performance characteristics like pure diesel while showing a decrease in pollutant emissions. The global production of vegetable oils in 2018 was 601.4 million tons [3]. Biodiesel represents one of the most utilised alternative fuels on the market. Numerous studies [4]-[6] have consensually agreed that the performances of biodiesel are similar with those of pure Diesel fuel. Biodiesel is obtained through the process of transesterification (methyl or ethyl esters of fatty acids obtained from animal fats and vegetable oils). It is renewable, biodegradable, contains oxygen and is commonly used as a blend element with pure Diesel having the aim of passing certain pollutant emissions norms. Studies have shown that the characteristics of this alternative fuel contribute to reduce the pollutant emissions levels (CO, CO₂, THC and soot) [7]. On a world scale, an important step concerning the use of Diesel – Biodiesel blends has been achieved by crossing the 6 billion litre of Biodiesel production threshold, the most common fuel blend present in everyday use being B5 (5% biofuel and 95% pure Diesel in volumetric fractions) [7]. The main problem with biodiesel usage today is caused by its stability, meaning the change in physio-chemical properties after a certain time interval, when stored; Table 1 below displays some physio-chemical properties of fuels tested in the present study.

**Table 1. Physio-chemical properties of the tested fuels**

| Properties/Fuel type                  | Diesel | B7   | B20  | B100  | Method of testing     |
|--------------------------------------|--------|------|------|-------|-----------------------|
| Density (g/cm³) @20°C                | 0.82   | 0.8467 | 0.8565 | 0.8864 | SR EN ISO 3675         |
| Cold filter plugging point (°C)      | -24    | -24  | -24  | -2    | SR EN 116 2016        |
| Viscosity (cst) @20°C                | 2.53   | 2.6  | 5.12 | 8.06  | SR EN ISO 3104        |
| Pour point (°C)                      | -24    | -20  | -18  | -7    | SR 13552              |
| Flash point (°C)                     | 58.5   | 66.5 | 85   | 184   | SR 5489               |
| Lower heating value (MJ/kg)          | 41.87  | 41.98 | 40.59 | 37.34 | ASTM D240             |
| Cetane number                        | 51.1   | 51.2 | 52.5 | 53.5  | EN ISO 516598         |
| Cloud point (°C)                     | -16    | -12  | -10  | -4    | SR EN 23015           |

The physio-chemical properties of the fuel used in compression ignition engines are essential in determining a fundamental parameter, the ignition delay (ID). The ignition delay represents the time interval between the start of fuel injection inside the cylinder (SOI) and the start of the combustion process of the air-fuel mixture (SOC). Its identification has a high degree of difficulty, problems often occur in its experimental determination. A recently studied method of improving the thermal efficiency and lowering pollutant emissions of the compression ignition engines is by using different combustion strategies. The aim of using these strategies is to operate the engine in a low temperature combustion mode that has been shown to have an improved thermal efficiency and a high global reduction in NOₓ and soot emissions. These reductions in emissions lead, to a decrease in aftertreatment device dependency. The low temperature combustion strategies are categorised as homogeneous charge compression ignition (HCCI), partially stratified charge compression ignition (SCCI), partially premixed compression ignition (PPCI) and reactivity-controlled compression ignition (RCCI). The latter method applied in Diesel engines operation is used in order to ensure high mechanical performance, high efficiency and low emissions. The RCCI method uses two fueling agents with different reactivities, so that the combined combustion and the heat release inside the engine cylinder could efficiently provide the maintaining of low temperature levels. The present study is dedicated to the understanding of the complex link existing between the fuel-air mixture reactivity, fuel properties, the ignition delay and the engine operating conditions.
for different individual biofuels before they could be used in blends with other fuels in compression ignited engines operating on such new technologies as HCCI.

2. Experimental setup
An in-line, four-stroke, naturally aspirated diesel engine with four-cylinder and direct injection was tested. The engine specifications are listed in Table 2. An injection system made by Delphi (CV34 6AG, Delphi Diesel Systems Ltd, England) consisting of a rotary pump (DP210), pressure lines, three fuel injectors having five 0.24 mm diameter holes, with a 330-bar opening pressure and one Perkins-Lucas injector modified with a Wolf needle lift sensor. Two pressure transducers (AVL GM 12 D) with a 15.76 pC/bar sensitivity were used for measuring the in-cylinder pressure. An AVL IndiSet 621 data acquisition system was used for registering high-speed variation parameters. An eddy current dynamometer (AVL Alpha 160) was coupled to the engine. A system controller (AVL Emcon 400) was used for controlling the test bed. A Gas Analyzer (HORIBA Mexa 7170D) has been used for measuring emissions. A test bed computer running the AVL PUMA v1.4 software has been used to register low speed variation parameters.

| Table 2. The main engine specifications |
|----------------------------------------|
| Bore x stroke (mm)                     | 102 x 115 |
| Rated power (kW) at 2400 rpm           | 50        |
| Maximum brake torque (Nm) at 1400 rpm  | 228       |
| Fueling system                         | Direct injection (DI) |
| Combustion chamber shape               | Bowl in piston |
| Compression ratio                      | 17.5:1    |

The experimental determinations have been done by injecting different fuels, alternatively, inside the cylinders. Pure Diesel fuel was used as a reference base and data was acquired for all investigated engine operating conditions (full load and several speed points, every 200-rpm in between 1200 and 2400 rpm). Afterwards the engine was fuelled with B7 and B20 biodiesel blends. After every testing session the engine was fuelled with pure Diesel with the purpose of purging the fuel lines and injection pump of possible gum deposits. For each investigated regime, two hundred consecutive cycles were acquired in a repetitive procedure at least by ten times, then adjusted in relationship with the ambient parameters, reported and averaged.

The ignition delay \( \tau_a \) was determined experimentally based on the averaged needle lift and on the rate of heat release curves. This was done by computing the time interval in CAD between the start of injection SOI moment when the needle lift jumps over 5 \( \mu \)m and the first positive value of the rate of heat release (RHOR) meaning the start of combustion (SOC) event (Figure 1). These experimental values were compared to those obtained by calculation using Arrhenius type equations developed by Hardenberg and Hase [8] Assanis [9] and El Kasaby [10], the temperature intervals used for these equations being computed using a simulation model of the engine developed with AVL Boost software [11][12] for every analysed engine condition.
3. Results

The experimentally determined needle lift shown in Figure 2. and rate of heat release shown in Figure 3., for pure Diesel fuel, were determined as a mean of 200 cycles for all analysed engine speeds. Similar data values as for pure Diesel fuel were obtained for B7 and B20 and have been used to establish the ignition delay values for the tested fuels and engine operating conditions.

The following results were obtained concerning performance characteristics and pollutant emissions:

Figures 4 and 5 show the evolution of brake power and brake specific fuel consumption relative to engine speed at full load condition.

It can be noticed that the use of B7 and B20 offer similar engine performance as for Diesel fuel. A slight degradation in the engine efficiency with an average value of 2% for B7 and of 5% for B20 is registered in the BSFC variation. This behaviour can be explained by greater viscosity and reduced lower hating value of
biodiesel blends which increase fuel consumption despite increased cetane number which theoretically would enhance fuel reactivity.

Figures 6 and 7 show the reported CO and CO₂ emissions relative to the same tested conditions.

The carbon monoxide emissions CO, as well as carbon dioxide CO₂, register a slight increase in all the investigated engine speeds compared to pure Diesel, an average increase of 7% for CO and 3% for CO₂ for B7 biodiesel and an average increase of 23% in CO emissions and 6% in CO₂ emissions for biodiesel B20 are to be noticed.

Figures 8 and 9 show the reported THC and NOₓ emissions relative to investigated engine conditions. A decrease in THC emissions could be observed with an average of 24% for B7 biodiesel and 5% for B20 biodiesel compared to pure Diesel fuel. Reduced values of NOₓ emissions with an average of 10% for B7 biodiesel and 5.5% for B20 biodiesel were observed when comparing to pure Diesel fuel.

Figures 10 shows the variation by speed of reported soot emission at full load and Figure 11 shows the variation of equivalence ratio by speed for the tested fuels.

An increase of soot with an average of 6.7% for B7 biodiesel and by 25% for B20 biodiesel could be observed when compared to pure Diesel fuel.

A theoretical analysis was also developed in order to reach a better understanding of the complex link existing between the fuel-air mixture reactivity, fuel properties, the ignition delay and the engine operating condition. In this sense, 3 Arrhenius type correlations were used to assess ignition delay:

Hardenberg and Hase correlation:

\[ t_x = 0.006n(0.36 + 0.22S_p) \left[ E_a \left( \frac{1}{RT} \right) - \frac{1}{17190} \right] + \left( \frac{212}{p - 12.4} \right)^{0.63} \]

\[ E_a = \frac{618840}{CN + 25} \]

\( t_x \) - average speed of the piston
\( n \) - engine speed
\( E_a \) – activation energy
\( R \) - universal gas constant
\( CN \) – Cetane number

(1)
Assanis correlation:

\[
\tau_a = 2.45p^{-1.02}\exp\left(\frac{2100}{T}\right)
\]

El-Kasaby correlation:

\[
\tau_{\text{ad}} = 26.06p^{-1.21}\phi^{-1.36}\exp\left(\frac{1038}{T}\right)
\]

\[
\tau_{\text{ad}10} = 79.51p^{-1.45}\phi^{-0.81}\exp\left(\frac{1028}{T}\right)
\]

\[
\tau_{\text{ad}20} = 74.32p^{-1.29}\phi^{-1.39}\exp\left(\frac{1022}{T}\right)
\]

Figure 12 shows the experimentally determined ignition delay in CAD for the before mentioned engine operating conditions.

It characterises the fuel-air mixture reactivity before the start of combustion. The variation of ignition delay with engine speed for all tested fuels shows a good correlation with the rise in BSFC and CO2 emissions respectively a decrease in NOx emissions. This behaviour could be explained by lower reactivity of the biofuel mixture B20. The lower errors of the Hardenberg and Hase correlation compared with the experimentaly determined values of the ignition delay could be because the cetane number of the fuel is considered when determining the activation energy, thus considering the fuel reactivity and also the engine functional regime by utilising the piston speed, pressure and temperature.

Results for the calculated ignition delays and the corresponding errors are displayed below in Table 3:

| Engine Speed (rpm) | Experimental (ms) | Hardenberg & Hase (ms) | Assanis (ms) | El-Kasaby (ms) | \( t \) Hardenberg & Hase (%) | \( t \) Assanis (%) | \( t \) El-Kasaby (%) |
|-------------------|------------------|----------------------|-------------|----------------|-----------------|----------------|----------------|
| 1200              | 0.49             | 0.42                 | 0.35        | 0.49           | 0.45            | 0.45           | 0.43           |
| 1400              | 0.48             | 0.42                 | 0.39        | 0.48           | 0.45            | 0.45           | 0.43           |
| 1600              | 0.36             | 0.21                 | 0.31        | 0.36           | 0.33            | 0.33           | 0.33           |
| 1800              | 0.37             | 0.32                 | 0.31        | 0.37           | 0.32            | 0.32           | 0.32           |
| 2000              | 0.38             | 0.38                 | 0.29        | 0.37           | 0.32            | 0.32           | 0.32           |
| 2200              | 0.34             | 0.34                 | 0.27        | 0.36           | 0.27            | 0.27           | 0.27           |
| 2400              | 0.35             | 0.38                 | 0.28        | 0.35           | 0.34            | 0.34           | 0.34           |

A good agreement was registered between the experimentally determined ignition delays and the values obtained by the Hardenberg and Hase and Assanis’s correlations for the tested fuels; the El-Kasaby correlation needs changes for a good fitting to our experimental data and this will represent an objective of a future work.
4. Conclusions

Minor deviations of the performance and efficiency parameters have been observed, these may be due to the low differences in the mixture formation and in the global evolution of combustion process.

Compared to pure Diesel, carbon monoxide and carbon dioxide emissions register increases under all investigated engine conditions with a maximum of 9% for carbon monoxide and 5% for carbon dioxide in the case of 7% volumetric fraction of biodiesel mixed with pure Diesel and 25% carbon monoxide, respectively 6% carbon dioxide for 20% volumetric fraction of biodiesel mixed with pure Diesel.

Unburnt hydrocarbon emissions were significantly lower under all tested engine conditions for 7% volumetric fraction of biodiesel mixed with pure Diesel, with a maximum reduction of 34% and with 14.9% for 20% volumetric fraction of biodiesel mixed with pure Diesel, but for some engine speeds (1800-2000) unburnt hydrocarbon emissions were higher.

The nitrous oxide emissions were reduced with a maximum drop of 15% for 7% volumetric fraction of biodiesel mixed with pure Diesel and with 14% for 20% volumetric fraction of biodiesel mixed with pure Diesel.

Experimentally determined ignition delay diminishes with the risen fraction of fatty acid methyl esters in the biodiesel fuel; the shortest ignition delays of all the tested fuels being that of 20% volumetric fraction of biodiesel mixed with pure Diesel. This behavior could be due to the later mixture’s higher oxygen content and greater cetane number. The shorter ignition delay involving higher reactivity of the fuel-air mixture clearly controls the combustion process decreasing the dimension of the premixed stage.

Such simple Arrhenius type correlations prove to be useful tools on the assessment of the fuel-air mixture reactivity.

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