DETERMINATION OF A SIMPLE AND ROBUST METHOD TO CALCULATE IGNITION DELAY IN COMPRESSION IGNITION ENGINES: APPLICATION TO DIESEL-BIODIESEL-ETHANOL BLENDS

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Received: Aug 13, 2022
Revised: Aug 17, 2022
Accepted: Aug 29, 2022

ABSTRACT

Today, the automobile industry is under increasing pressure from governments and environmental organizations to lower their carbon footprint. Car manufacturers release new models that follow modern and severe norms of pollutant emissions using eco-friendly technologies. Their strategy follows two main pathways: decarbonization and increase of efficiency. In order to achieve the aforementioned objective, the replacement of fossil fuels with renewable biofuels with lower carbon footprints, such as biodiesel and/or ethanol, contribute to the decarbonization of current internal combustion engines. However, changes in the fuel composition also imply some adjustments in the injection and combustion settings to optimize the engine operation. An important parameter to assess the combustion process in a compression ignition (CI) engine is the ignition delay (ID), defined as the duration between the start of fuel injection (SOI) and the start of combustion (SOC). Different direct and indirect approaches are reported in the literature to measure or calculate the ID. This study investigates two indirect methods based on the measurement of the indicated pressure inside the combustion chamber, throughout the displacement of the piston. These measurements are based on the position of the maximum of the third-order derivative of pressure as a function of the crank angle and the subjective shift from a polytropic compression process, respectively. The objective of the present work is to define an objective criterion to quantify this shift in order to allow a unique determination of the ID, independently of the adopted method. Thus, data of six different fuel blends of diesel-biodiesel-ethanol in a four-stroke CI engine from tests performed at PUC Rio's Vehicular Engineering Laboratory for different loads and engine speeds were analyzed using the software Matlab. As observed in previous literature, the results show that the addition of biodiesel and ethanol concentration in the blends slightly increases the ID. Moreover, the difference between the experimental pressure with combustion and the calculated pressure in the polytropic compression process at the SOC position is lower as engine load increases and/or engine speed decreases and seems independent of the fuel blend composition. Nonetheless, the variation of the pressure difference at the SOC stays moderate, allowing the determination of a unique value to directly read the ID from the indicated pressure profile, without requiring the smoothing process of the third-order derivative of the pressure.

Keywords: Combustion, DBE blends, Ignition Delay, Indicated Pressure, Polytropic Compression

NOMENCLATURE

CAD	crank angle degrees
CI	compression ignition
CO_2	carbon dioxide gas
DBE	diesel-biodiesel-ethanol blends
DIFF	parameter related to ID
GHG	greenhouse gas
ID	ignition delay
MWM	Brazilian manufacturer of engines in diesel
P	pressure, bar
Q_f	heat, Joule
ROHR	rate of heat releaser
RPM	revolutions per minute
SOC	start of combustion
SOI	start of fuel injection
TDC	top dead center
V	chamber volume, m^3

Greek symbols

φ	crank angle, º
κ	isentropic exponent, -
θ_{ID}	angle at start of ignition delay, º
θ_{SOC}	angle at start of combustion, º
θ_{Inj}	angle at start of injection, º

INTRODUCTION

Observing global energy consumption in the last 50 years, the pattern reveals that the world remains significantly dependent on fossil fuels (Rodrigue, 2020). Over the period 1965-2019, though global energy consumption rose, the percentage use of coal and oil steadily declined from 37 to 27% and 42 to 33% respectively. Natural gas was the only fossil fuel that increased in percentage use from 14 to 24%. Despite the fact that usage of renewable energies, globally, have steadily increased throughout the years, the total share still remains minimal, around 11% (Richie, 2021). Moreover, the per capita usage of energy consumption has also declined. Thus, these statistics indicate efforts in increasing energy conversion efficiency and using decarbonized energy sources. Focusing more specifically on energy consumption in Brazil for 2019, it can be seen in figure 1 that an important share (34.4%) comes from petroleum and other liquids, as the country is the ninth-largest producer of petroleum and other liquids (3.36 million barrels produced each day) and the eight largest energy consumers in the world.

Figure 1. Brazilian consumption energy mix by fuel type in 2019 (EPE, 2020).

It is important to also note that since 2010, Brazil's energy consumption has grown 28% due to its economic growth. Nonetheless, Brazil boasts a relieving 46.1% of energy consumption coming from renewable energy. This is largely due to Brazil's ethanol and biodiesel use in transports and numerous hydroelectric plants. Brazil is currently the second largest producer of ethanol in the world producing around 37,380,000 m^3 in 2019, which makes it an excellent candidate to tackle the greenhouse gas (GHG) emission problem. Brazil also produced 5,902,461 m^3 of biodiesel in 2019 with more than three quarters of it coming from soybean oil (ANP, 2020). For comparison purposes: France, a country thirteen times smaller than Brazil with a three times smaller population can only boast a meager 13.9% of energy consumption coming from renewable energy. Naturally, in 2017, Brazil's CO_2 emissions per capita was 2.23 metric tons, composing 1.29% of the world's carbon footprint whilst France's was 5.34 metric tons, composing 0.93% of the world's carbon footprint (Ritchie, 2018). It is essential that researchers find a way to tackle that number and find a solution to prevent the increase of global warming and health issues caused by pollution and reduce the dependency to fossil fuels. Due to the availability of renewable resources and its potential to have a sustainable energy matrix (high potential for wind and solar energy), Brazil can potentially be a leader worldwide.

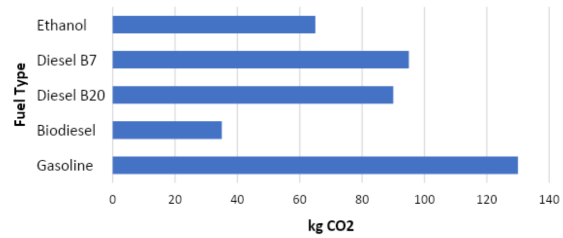

Figure 2. CO_2 emissions per 1000 km of different fuel types in a Brazilian 1.5 Sedan (Mecanicaonline, 2016).

Referring to figure 2, an ethanol run car (Sedan 1.5) runs 1000 km and produces around 65 kg of CO_2 which is around 30 kg less than that of a diesel B7 run car and half that of a gasoline run car. Biodiesel blends in figure 2 are denoted as B(n) where n is the volumetric percentage of biodiesel in a biodiesel-diesel blend. The best contesting fuel in the figure is the biodiesel, which produces 34 kg of CO_2 per 1000 km. This is why diesel-biodiesel-ethanol blends can potentially add up to significant differences in GHG emissions.

The aforementioned facts are why this research is focusing on evaluating ethanol and biodiesel additive fuels in diesel motors so that it may replace as much diesel as possible and keep a decent performance. It is important to also keep in mind that, though one desires to use greener energies, one still has to take the inferior impact biofuels have on vehicular performance, that will be discussed later in this paper, into account. On top of that, 100% replacement of diesel by biodiesel today is simply not viable as demand is high and there is not enough arable land to grow that much biomass (Pradelle, 2017). Moreover as seen on figure 2, biodiesel and ethanol consume more to go further and biofuels are more expensive which may give car buyers second thoughts. It is important to measure energy efficiency of engines and optimize the combustion (in particular, with the correct control of the ID) of these blends so one can determine the equilibrium between the best performing and greenest volumetric mixture of fuel to use.

The main objective of this study is to define, through experimental data extracted from the research done in the Vehicular Laboratory of PUC-Rio regarding the combustion of diesel-biodiesel-ethanol fuel blends, a robust and simple method to determine the ID based on the pressure profile inside the combustion chamber.

After calculating the ID of the different fuel blends by determining the start of combustion as the position of the maximum of the third order derivative of pressure, this study analyzes and discusses the impact of engine load (torque), speed and fuel composition. This study will also compare the parameter related to ID denoted as DIFF; which represents the difference between experimental logarithmic pressure and theoretical logarithmic pressure if the motor compression had been perfectly polytropic. The DIFF will inform us better on the accuracy of our ID results based on the indicated pressure profile. All in all, to avoid calculating the pressure derivatives, verifying whether there exists a simple equivalence related to the determination of ID and the diverging behavior of a polytropic compression can help evaluate the robustness of this pressure variation method.

LITERATURE REVIEW

Autoignition and Ignition Delay

In a CI engine, ignition delay is defined as the time span between the start of the fuel injection (SOI) by the fuel injector and the start of the combustion (SOC) of the air-fuel mixture (Tabaczynski, 1979). What makes ID so important is, since combustion should occur when, or shortly after, the piston head reaches top dead center (TDC), ID cannot be too long nor too short. Too early, the diesel ignites before the piston reaches its TDC, generating a force pushing the piston back the wrong way. Too late, the diesel combusts after the piston is already on its way down from the TDC and work is lost; the piston and the crankshaft rotate at a smaller velocity. Crankshaft engine speed is important because, in physical terms, the faster the crankshaft rotates, the higher the combustion temperature. By controlling ID, one can increase the engine efficiency too (Prabhahar, 2019).

ID is affected by multiple parameters such as compression ratio, inlet pressure, engine load, injection parameters and angles, and other fuel properties (Prabhahar, 2019). Regarding fuel properties, the cetane number is an indicator of the autoignitibility of the fuels and provides information on ID (Mabanaft, 2020). A higher cetane number means shorter ignition delay. In Brazil, vehicles running on diesel-biodiesel blends usually have a cetane number between 48-60 (Costa *et al.,* 2018). Typical values of ID in diesel engines vary between 0.5 – 2.0 ms (Stone, 1993).

EXPERIMENTS

The indirect pressure variation method

One way to deduce ID is with the help of pressure sensors located around the exposed interior cylindrical surface of the combustion chamber remaining when the piston reaches TDC (volume at TDC). These pressure sensors allow us to measure pressure inside the chamber as a function of the crank angle and, consequently, the piston displacement. A criterion used for the determination of the start of combustion (end of ID) is to consider the value of the crank angle where the maximum of the third order derivative of the cylinder pressure with respect to the crank angle is reached (Katrasnik *et al.,* 2004). Indeed, pressure variation is also directly proportional to the rate of heat release (ROHR) in the chamber.

$$\frac{dQ_f}{d\varphi} = \frac{1}{\kappa-1}\left(\kappa P\frac{dV}{d\varphi} + V\frac{dP}{d\varphi}\right), \tag{1}$$

$$\therefore \frac{dQ_f}{d\varphi} \propto \frac{dP}{d\varphi}, \tag{2}$$

Where Q_f, φ, κ, P, and V are the released heat [J], crank angle [°], isentropic exponent [-], gas pressure at TDC [bar], and chamber volume [m^3], respectively.

As a result, we are capable of obtaining graphs like such:

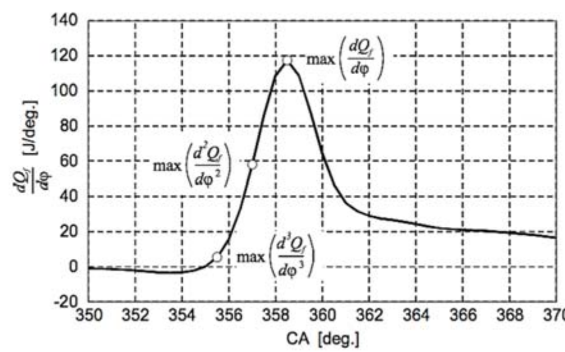

Figure 3. ROHR with indicated maximum values of its derivatives (Katrasnik et al., 2004).

In the figure above, the position of the maximum value of the first, second, and third order derivatives of heat release as a function of crank angle are displayed. Physically speaking, it is reasonable to deduce that the start of combustion occurs when $\frac{dQ_f}{d\varphi}$ significantly departs from 0. That corresponds to the location of $\left(\frac{d^3Q_f}{d\varphi^3}\right)$ (Katrasnik *et al.,* 2004). Then, by observing the crank angle degree correspondent to $\left(\frac{d^3Q_f}{d\varphi^3}\right)$, it is possible to deduce the crank angle at the start of combustion. As the crank angle degree at injection (start of ID) is known, the ID can be determined accurately.

Another method to determine ignition delay from the pressure data is using the logarithmic curve graph relating volume of the compressed chamber as a function of pressure throughout the piston's displacement. Before the start of combustion, there is a thermal exchange between the air and the surface of the chamber as the air is compressed by the piston that one can assume as polytropic (Heywood, 1988). A polytropic compression is a thermodynamic process that follows the reactions:

$$PV^n = constant \quad (3)$$

$$\therefore ln\, ln\,(P)\ + n \cdot ln\, ln\,(V)\ = p \quad (4)$$

Where P and n are the gas pressure in the chamber [bar] and polytropic exponent [-], respectively (p is a constant).

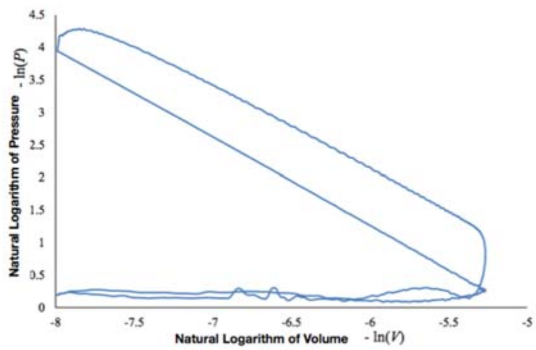

Figure 4. Template curve of a compression cycle from a diesel engine relating the natural logarithms of volume as a function of pressure (Pasqualette *et al.*, 2014).

As one can observe, the linear behavior during compression (and expansion, respectively) represents the phase where the process is correctly modeled as polytropic. As soon as the line stops its straight trajectory and curves, the polytropic phenomenon ends and that also marks the start of combustion (Pasqualette *et al.*, 2014). This method is therefore a tool to find the endpoint of an ID. Nevertheless, such lectures are quite subjective.

Experiment Overview

As stated in the abstract, this paper uses the data generated from a past experiment run at PUC Rio's Vehicular Engineering Laboratory in 2017. This research paper is derived from Pradelle's doctoral thesis (Pradelle, 2017). However, that data was used solely to analyze ignition delay.

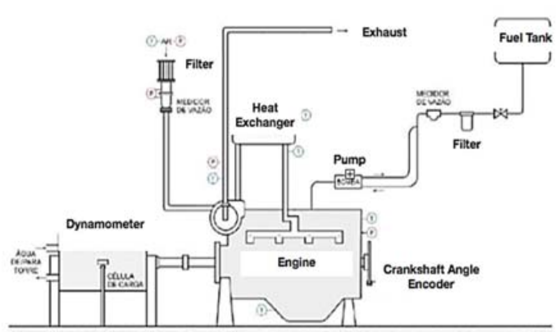

Figure 5. Scheme of the experimental bench used at PUC Rio's Vehicular Engineering Laboratory (Guedes, 2017).

The experiment was run using a four-stroke diesel compression motor with four aligned cylinders from the Brazilian brand *MWM*. More specifically, the motor is an MWM 4.10 TCA (Euro III) with a crank injection angle of 9°14' before TDC. The figure above represents a simplified scheme of the mechanism that was used to measure the engine's properties with different fuel blends used. In summary, this mechanism is essentially a functioning motor with various piezoelectric sensors and other instruments that measure or control torque, acceleration, water temperature, oil temperature, air temperature, engine speed, chamber pressure, humidity, and power. As aforementioned, to retrieve the pressure in the combustion chamber and the CAD as the diesel engine runs, a piezoelectric pressure transducer, model 6052CS31U20 from Kistler, capable of measuring dynamic pressures up to 300 bars was installed on the head of the cylinder number 4 along with a crank angle encoder, Kistler 2614A1, which has a resolution of 0.1 degree. Because these two instruments output an electric charge, they were connected to an amplifier signal converter INDIMETER 619 to convert the electric signals into readable data on the computer. To run this experiment, the researcher would essentially purge the motor, add the fuel studied, adjust the torque, engine speed, and engine load and run the mechanism. This research only used the collected data of two different properties: the pressure inside the combustion chamber as the piston is displaced in the diesel motor and the crank angle degrees (CAD) as the crankshaft rotates. With the help of these instruments, the fuels in table 1 were tested under 12 different combinations. The fuels were tested at 1500, 1800, 2100 rpm at 25, 50, 75, and 100% maximum engine torque three times each (Pradelle, 2017).

Table 1. Six different volumetric blends of diesel fuels studied (Pradelle, 2017).

Fuel Blend	% Diesel	% Biodiesel	% Ethanol	Additive
B7*	93.0%	7.0%	0.0%	No
B15	85.0%	15.0%	0.0%	Yes**
E5	80.7%	14.3%	5.0%	Yes**
E10	76.5%	13.5%	10.0%	Yes**
E15	72.2%	12.8%	15.0%	Yes**
E20	68.0%	12.0%	20.0%	Yes**
***(Commercial Diesel)**				
****The added additive represents a volumetric quantity of 1% of each fuel blend, guaranteeing the homogeneity and miscibility of these blends.**				

Matlab

First and foremost, it is important to reiterate the definition of ID which is the time span between fuel injection and start of combustion, represented by this mathematical equation:

$$\theta_{ID} = \theta_{SOC} - \theta_{Inj} \quad (5)$$

Where θ_{ID}, θ_{SOC}, and θ_{Inj} are the ID [°], angle at start of combustion [°], angle at injection [°], respectively.

As mentioned previously, θ_{Inj} of the MWM motor is 9°14' before the TDC. However, to compensate delay in instant of injection caused by the biodiesel blends in the diesel fuels tested, θ_{Inj} will be set at 10.5° before the TDC (Pradelle, 2017).

The first thing that will be done through *Matlab* is to find the second and third order derivatives of pressure using the fourth order centered formula:

$$\frac{df}{d\theta}\Big|_i = \frac{f_{i-2} - 8f_{i-1} + 8f_{i+1} - f_{i+2}}{12\Delta\theta} \quad (6)$$

This way, one will be able to calculate the position of the maximum of the third order derivative of pressure as a function of the crank angle and determine the angle at start of combustion as suggested by Kastranik et al. (2004):

$$\theta_{SOC} = \left(\frac{d^3P}{d\theta^3}\right) \quad (7)$$

One will also be able to use the thermodynamic principle of a polytropic transformation as introduced previously (equations 3 and 4) to obtain a line graph displaying the logarithmic pressure as a function of volume similar to that of figure 4.

To go even further, one can obtain a parameter denoted as log(DIFF) which represents the difference between experimental logarithmic pressure and theoretical logarithmic pressure if the compression had been perfectly polytropic. The equation is represented as such:

$$log(DIFF) = log\,log(P) - (n.log\,log(V) + p) \quad (8)$$

This will enable the plotting of a line graph of the log(DIFF) as a function of CAD to get a visual identification of the start of combustion and be able to determine its respective crank angle θ_{SOC} . It is important to note that to determine **n** and **p** in equation 8 and obtain the line graph, the crankshaft displacement range considered was solely between $-80 - 80$ CAD.

Lastly, because the tests were run multiple times, average ID with a standard deviation for each fuel under their respective testing conditions with a confidence interval of two times the mean squared can be pondered.

RESULTS AND DISCUSSION

Ignition Delay and Engine Load Result Analysis

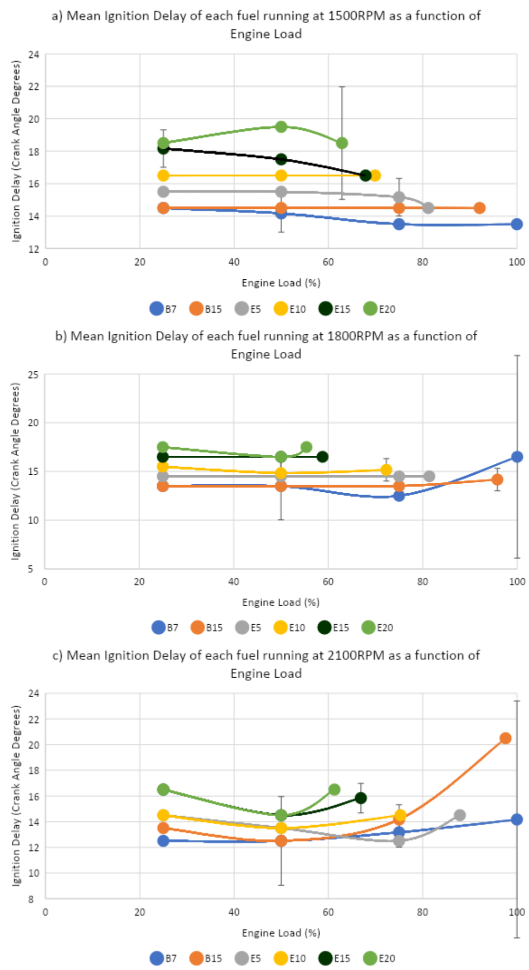

Figure 7. Mean Ignition Delay of each fuel running at 1500 RPM (top), 1800 RPM (center), and 2100 RPM (bottom) as a function of Engine Load.

Taking all the fuels into account, a recurring line tendency from 1500 RPM to 2100 RPM can be observed. Within the uncertainty range, all the investigated cases have constant behaviors. At 1500 RPM, the curves cover ignition delays ranging from 13 to 20 CAD. At 1800 RPM, the curves cover ignition delays ranging from 13 to 17 CAD. Then, at 2100 RPM, the curves cover ignition delays ranging from 12 to 21 CAD. Hence, results are quite repetitive. Moreover, an increase in engine load, regardless of the engine speed, seems to have very little effect on ignition delay. The only particular feature here is that at 2100 RPM, the ignition delay of B15 does seem to increase drastically as torque is increased. In general, it seems that fuel blends with ignition delays from shortest to longest are: B7, B15, E5, E10, E15, and E20. The more biodiesel and ethanol fuel blends contain, the longer their ignition delay. However, ignition delay differences between the shortest and longest vary no more than 8 CAD. Some displayed points do possess non zero standard deviations, however most are of up to 2 CAD. The fuel blend with the most important standard deviation is B7 that possesses points with errors over 8 CAD.

DIFF Result Analysis

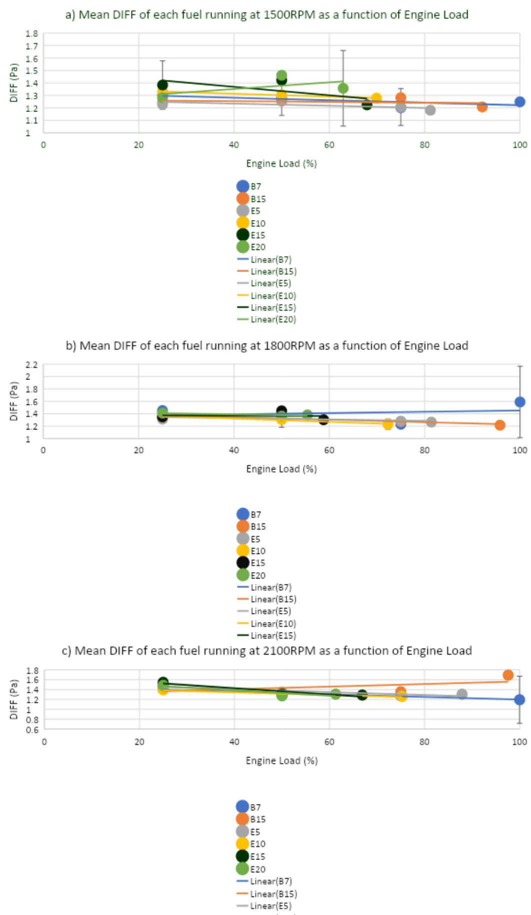

Figure 8. Mean DIFF of each fuel running at 1500 RPM (top), 1800 RPM (center), and 2100 RPM (bottom) as a function of Engine Load.

Looking at figure 8, most of the DIFF values belong to the range 1.2 to 1.6 Pa. Each time, only one trend line is not negative while the rest are always decreasing in DIFF as engine load increases. Because of its more important variance values, the DIFF graph at 2100 RPM may be the most accurate of all. As one looks for patterns in the results, the fuel blend E10 seems to generally have a lower DIFF value than its counterparts whilst E20 seems to have larger values. Nonetheless, these two statements are to be taken more as generalities as more defining patterns are unclear.

Discussion

This section focuses on three main factors: engine speed, engine load, and fuel blend to determine which parameter most affects the DIFF value. To avoid including data with important errors, points at 100% engine load have been excluded from this analysis.

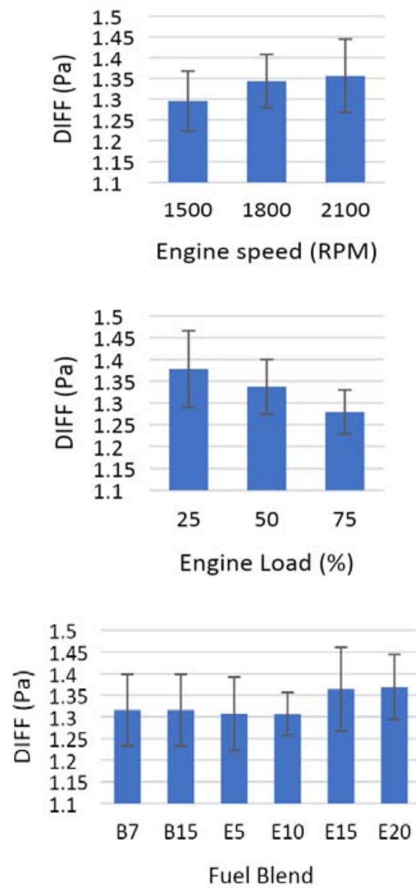

Figure 9. Mean DIFF of all fuel blends by Engine speed (left), Engine Load (center), and fuel blend (right).

After analyzing these three parameters in regards to DIFF values, the variation of DIFF values ranged up to 0.1 Pa. In more technical terms, one may aspire to possess a fuel blend that has a smaller DIFF value for sake of coherence where pressure variations respond fast to the piston compression. Nevertheless, a small DIFF can also complicate the experimental determination of itself compared to the theoretical one. This is simply to underline that a smaller DIFF is not always better. All in all, it seems that engine load had the most drastic effect on DIFF as it was the only one with varying values of more than 0.05 Pa at each category of engine load. To conclude, the mean DIFF value of all the parameters displayed is 1.33 Pa with a mean standard deviation of 0.07 Pa. Thus, as a general thumb rule, an adequate DIFF value to adopt would be between 1.26 – 1.40 Pa.

CONCLUSIONS

Two methods were investigated to determine the ignition delay of each fuel blend. With a given crank angle for start of injection, the crank angles differences between start of ignition and start of combustion would give ignition delay.

After obtaining the ignition delay for all fuel blends tested, one was able to conclude that the fuel blends B7, B15, E5, E10, E15, and E20 had the shortest ignition delays from shortest to longest. The engine speed and the engine load have mild to no impact on the lengths of ignition delays of these fuels. Between the fuel blends, the ignition delay varied no more than a mere 8 CAD. All in all, these 6 different types of fuel blends show mild differences in ignition delay and may potentially be a satisfactory replacement in the current diesel fuels fleet.

When comparing DIFF between all fuel blends, this research suggests that engine load is the factor that most affects its value. Large DIFF values mean that pressure variations from the piston polytropic compression must be significant, meaning that the given point of start of combustion given by the position of the maximum of the third order derivative of pressure could actually be earlier than it actually is. Thankfully, the DIFF values found between each fuel blend varied by a meager 0.1 Pa at best as almost all fuel blends had DIFF values of between 1.3 – 1.4 Pa, suggesting that start of combustion values used to determine ignition delay were close to reality. There was no obvious pattern that showed that certain fuel blends had consistently larger or smaller DIFF values, allowing the definition of a unique value for all blends. In the future, it would be interesting to analyze a fuel blend's lower calorific value and check if it has any impact on DIFF values and ignition delay. Moreover, engine performance and yield could also be studied in relation to the ignition delays of the different fuel blends.

RESPONSIBILITY NOTICE

The authors are the only responsible for the printed material included in this paper.

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