Comparison of the Cyclic Variation of a Diesel-Ethanol Blend in a Diesel Engine

Comparison of the Cyclic Variation of a Diesel-Ethanol Blend in a Diesel Engine

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Abstract. Alcohols are renewable and sustainable second generation biofuels which are derived from various biomass feedstock sources. These fuels with similar properties to mineral diesel can be used as a blend or additive to improve the combustion characteristics and pollutant emissions in the automotive engines. However, different fuel properties characterize different combustion phasing parameters for the specific engine operation and test condition. This paper presents the preliminary results of coefficient of variations of IMEP (COVIMEP) and P_{max} ($COVP_{max}$) for a diesel engine fuelled with mineral diesel (B0) and DE10 blend at full load both engine speeds of 1100 rpm and 2300 rpm. The influence of ethanol content in a blend of diesel on the cyclic combustion variations is explained in the calculation values of the coefficient of cyclic variation (COV). The experimental results showed the DE10 fuelling exhibited larger cyclic variations than mineral diesel (B0) at the same test conditions, owing to the reduction of combustion temperature during combustion phasing and lower reactivity of ethanol.

1. Introduction

Alternative fuels have potential to go further to improve the compression ignition engines energetically and pollutant emissions. Alcohols including ethanol and butanol represent second generation biofuels which score high ratings in combustion properties, sustainable feedstock sources and possibly diminish the fossilized fuel consumption [1–3]. Therefore, ethanol is considered an alternative fuel for the automotive engines at present and could reduce the harmful emissions. A 40% - 60% reduction in the NO_x emission is found with the use of ethanol. Ethanol is considered an alternative sustainable fuel for automotive engines due to its several advantages including comparable properties with SI engines at specific operating conditions [4,5], produced from agricultural and waste products through chemical processes [6,7] and infrastructure for present fuel able to facilitate the distribution and storage possibilities [8]. Ethanol possesses better combustion properties comparative with the present fuels with higher octane number [9], more substantial oxygen content at the molecular level [8], greater autoignition temperature [10], greater laminar flame speed [11] and lower adiabatic flame temperature. There is no significant engine modification of its design needed when running with the ethanol in maintaining or increasing the engine performance energetically. Since ethanol has a higher heat of vaporization, an improved intake air efficient cooling effect is obtained which leads to a volumetric efficiency improvement and reduces the risk knock development. Also, pollutant emissions especially NOx is decreased owing to the lower in-cylinder temperature. Moreover, the use of diesel-

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ethanol blends leads to the increase of the maximum in-cylinder pressure and the pressure rise maximum rate due to the advanced combustion properties.
Cyclic variations determine the characterization of the engine when running with different fuel and test operating condition. Numbers of research work have focused the investigation on the cyclic variations, then proposed several techniques for measuring and analyzing the cyclic combustion variations in spark ignition (SI) engines [12]. Most of the studies are focused on reducing the SI engines unpredictability, particularly in engine knocking. However, combustion cyclic variation investigations recently have been conducted on standard diesel engines by other researchers operating with different fuels including alcohols [13]. The cyclic variations can be characterized by coefficients of variation (COV) in cylinder pressure. The intensity of the cyclic variation occurrence is defined by the coefficient of cycle variations for some specific cycles. The coefficient of cyclic change is described as a relative average deviation of maximum pressure values in the engine operation. For "n" consecutive cycles, if is considered a normal distribution of the deviation probabilities, the squared average standard deviation, σ can be calculated and the cycle variation coefficient (COV) is defined as:

$$COVa = \frac{SD(a)}{\bar{a}} \times 100\% \tag{1}$$

Where,
a is the parameter of which variability is studied and is defined as for indicated mean effective pressure (IMEP), maximum pressure (P_{max}) in the cycle number "i".

SD (a) is the standard deviation of the defined parameter
IMEP is the mean value of the defined parameter

$$SD(a) = \sqrt{\frac{\sum \left(a_i - \bar{a}\right)^2}{n_c}} \tag{2}$$

$$\bar{a} = \frac{1}{n_c} \sum_{i=1}^{N_c} a_i \tag{3}$$

Where,
a_i is the pressure at specific cycle (bar)
n_c is the number of cycles

In general, the way of cyclic variation evaluation for test condition closer to the value of maximum brake torque the COV of maximum pressure, P_{max} is suitable. The variation of the IMEP, translated by the coefficient of variation (COV) IMEP, is the most suitable instrument to characterize the engine respond to the differences in the combustion process. It is precisely shown that the limit value of COVIMEP defines strictly the limit of mixture leaning. Also, this coefficient of cyclic variation can also point toward the difference in flame development during the initial combustion phase. Hence, the fuel type used in the engine influences the cyclic variation as a result of its laminar flame velocity values. Therefore, the flame development of ethanol is much quicker, comparative to gasoline and diesel owing to the higher laminar combustion speed. This attribute reduces the influence of turbulence and the cyclic variations. However, the quality of the in-cylinder mixture influences the combustion process through chemical reaction speed, with a maximum is obtained in the area of rich region. From this point of view, the initial and final phases of the combustion process have a minimal duration in the region for which the chemical reaction speeds are maximum. At the mixture leaning the durations of those two phases increase and the total combustion duration also increases.

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2. Experimental Set up

In this research work, mineral diesel (B0) was purchased from a local supplier company, and ethanol was purchased from Merck through a local agent. Then, the fuel was blended at 10% by volume of ethanol for each 1 liter of mineral diesel (B0), which denoted as DE10 (90% mineral diesel+10% ethanol). The fuel properties of the blend are tested according to the ASTM standards for density, viscosity, cetane number, flash point and calorific value. Table 1 lists the test fuel properties in the study.

Table 1. Test fuel properties

Description	Testing Method (ASTM)	Mineral diesel (B0)	DE10
Density @ 20 ℃ g/cm^3	D287	0.8264	0.8226
Viscosity @40 ℃ mm^2/s	D445	5.144	3.674
Cetane number	D613	47.8	43.82
Flash Point (℃)	D93	60	55.2
Calorific value (MJ/kg)	D240	44.8	43

This research work was conducted on a Yanmar TF120-M four stroke, direct injection single cylinder diesel engine. It is a water-cooled, low-speed with a maximum power of 7.8 kW at 2400 rpm. A 15 kW eddy current, dump load dynamometer was coupled to the engine and been controlled with a universal controller model DC5-10KW for controlling the engine speed and torque. Two separate fuel tanks with thermocouples and a fuel valve system were used, one for mineral diesel, B0 and the other for the DE10 blend. In fuel delivery system, a burette was used to measure the fuel consumption of both fuels. Table 2 describes the specification of the test engine and Figure 1 illustrates the setup of the engine testing.

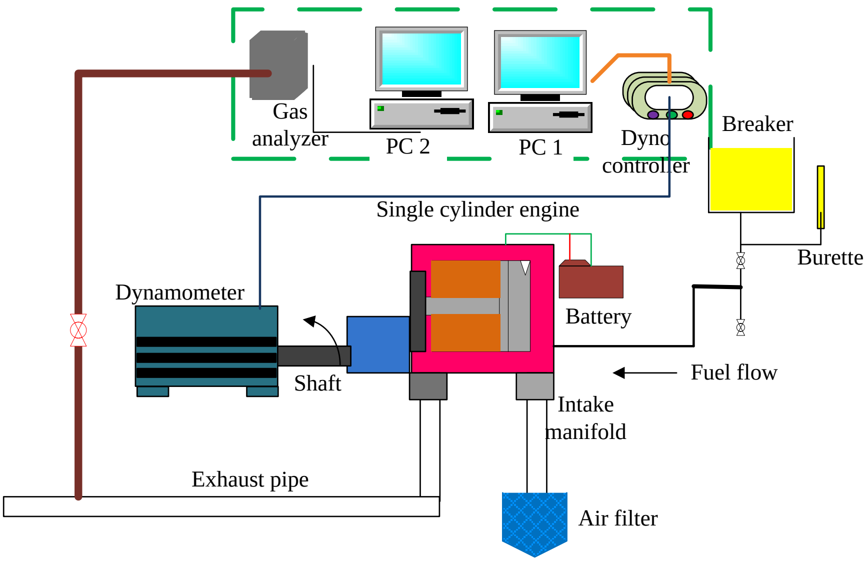

Figure 1. Engine testing set up

A preliminary comparative fuel study of cylinder pressure cyclic variations was developed for full load condition and both engine speeds of 1100 rpm and 2300 rpm. Therefore, the in-cylinder pressure was measured and recorded using an Optrand AutoPSI-S model C22294-Q pressure transducer with a measurement range from 0-5000 psi. The crank angle degree signal was obtained using a magnetic crank encoder. A TFX combustion analyzer was used to record and analyze the in-cylinder pressure and crank angle signal measurement at the specific test condition as listed in Table 3.

Table 2. Test engine specification

Description	Specification
Engine model	Yanmar TF120M
Number of cylinders	1
Combustion system	Direct injection
Total displacement (L)	0.638
Bore x stroke (mm)	92 x 96
Injection timing	18^o BTDC
Compression ratio	17.7:1
Continuous output (HP)	10.5 HP at 2400 rpm
Rated output (HP)	12 HP at 2400 rpm

Table 3. Test condition

Parameters	Test condition
Type of fuel	B0, DE10
Speed (rpm)	1100, 2300
Load (%)	100
Fuel temperature (℃)	27 ±1
Air temperature (℃)	30 ±1

These in-cylinder pressure data were selected to ensure the data consistency and were collected for 200 consecutive cycles for both B0 and DE10 fuelling. In this research work, a further calculation is needed to obtain the cyclic variation coefficients for indicated mean effective pressure (IMEP) and maximum pressure (P_{max}). To evaluate the way that the engine running is affected by the variability of the combustion process, these coefficients are calculated and presented in the following figures. Regarding cycle variability, the general tendency shows a significant decrease in this phenomenon when the DE10 fuel is used.

3. Results and Discussion

For all running test condition, full load and both engine speeds, the injection timing was maintained to ensure its reliability. At an operating test condition, defined by engine speed and full load, the fuel cycle dose was continued using the ball valve opening.

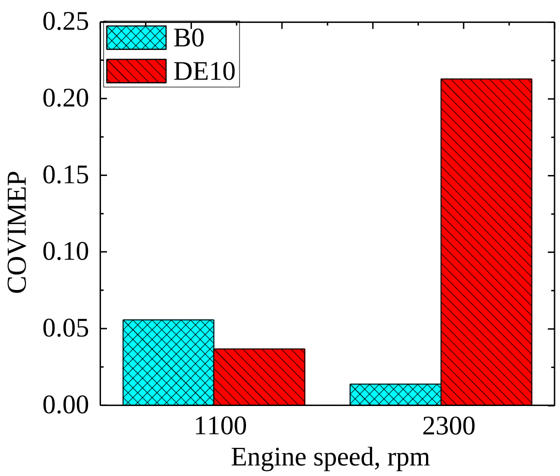

Figure 2. COV for IMEP (COVIMEP) versus full load at 1100 rpm and 2300 rpm

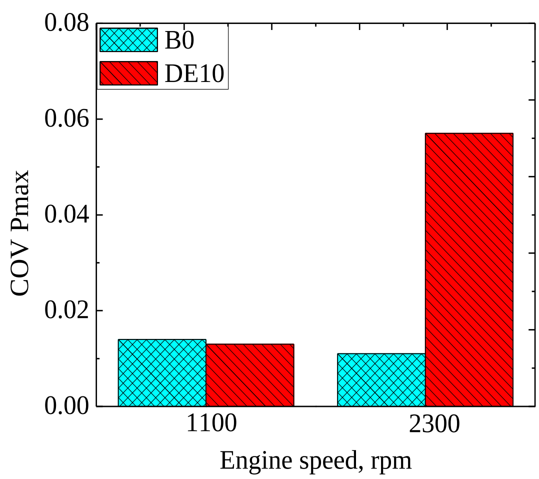

Figure 3. COV for maximum pressure, Pmax ($COVP_{max}$) versus full load at 1100 rpm and 2300 rpm

Comparing both two running conditions defined by 1100 rpm and 2300 rpm, at only DE10 fuelling, it was observed that at the speed of 2300 rpm, there was an increase of COVIMEP from 0.037% to

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0.213% for DE10, as Figure 2 shows. However, the lower tendency of cyclic variations was observed for B0 respectively from 1100 rpm to 2300 rpm. The decreasing effect also appears for the B0 fuelling when the COVIMEP decreases from 0.056% to 0.014%. The cycle variability coefficient of maximum pressure, COVPmax, for DE10 fuelling increases from 0.013% to 0.057% at full load condition when the speed rises from 1100 to 2300 rpm, as Figure 3 presents. The cyclic variation is deteriorated at speed increasing, increasing with 338.5% for the DE10 fuelling. On the other hand, the COVP$_{max}$ for the B0 fuelling tends to decrease with 21.4% from 0.014% to 0.011% as the speed increases.

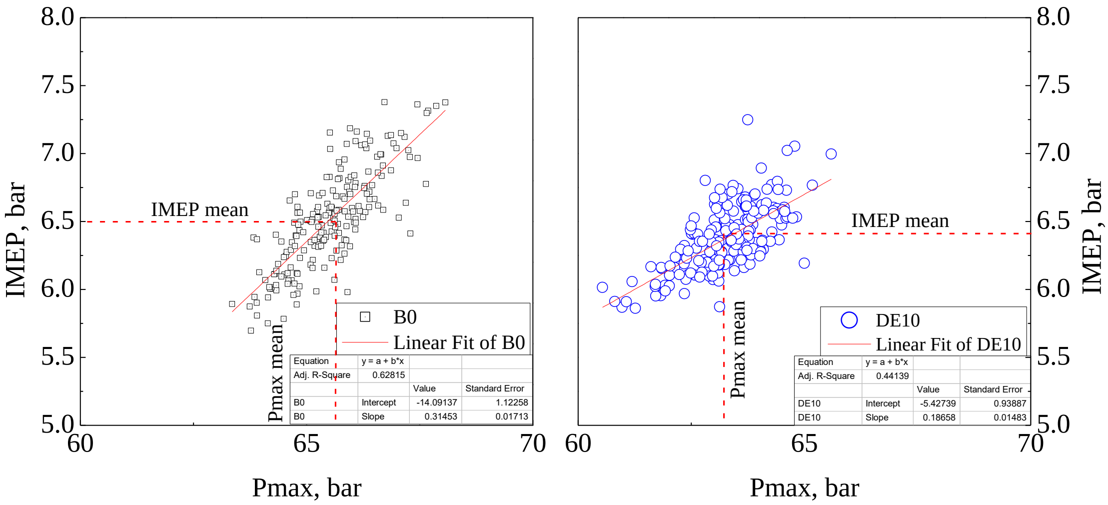

Figure 4. IMEP versus P$_{max}$ with full load at 1100 rpm

Comparison for both B0 and DE10 in IMEP versus P$_{max}$ in similar condition is illustrated in Figure 4. It was observed from the figure that maximum cylinder pressure traces for both B0 and DE10 vary for the 200 cycles. The higher intensity of cyclic variation was found for DE10, comparative to B0. The difference between the two pressure traces is further increases, not only due to the higher fuel mass at full load condition which is burned but also due to the faster combustion process at high engine load. Furthermore, combustion at high load with high engine speed starts earlier due to the more upper air and fuel masses, while remaining gas fraction that dilutes mixture is lower which benefits faster flame propagation.

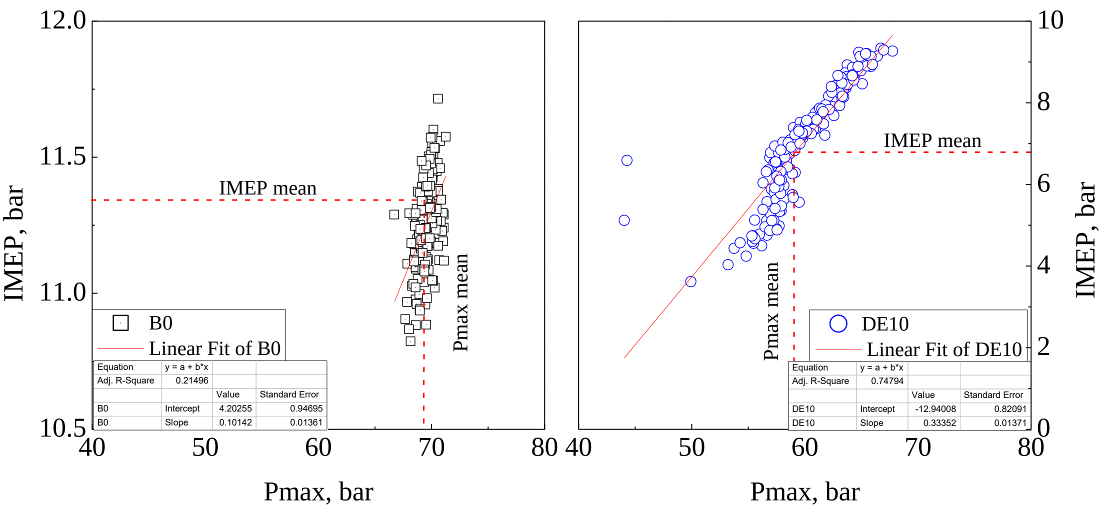

Figure 5. IMEP versus P$_{max}$ with full load at 2300 rpm

The IMEP versus maximum pressure, P_{max} values for both B0 and DE10 increase when engine speed increases, as shown in Figure 5. It was observed from the figure that B0 has higher intensity for maximum pressure, P_{max}, comparative to DE10. Also, the maximum pressure, P_{max} for B0 increases as the engine speed increases from 1100 rpm to 2300 rpm. From the point of view, high maximum cylinder pressure cycles are associated with the fast burn rates owing to the increase in piston speed movement. Also, earlier initial flame kernel development leads to timely combustion evolution before TDC, that resulted in higher cylinder pressure. However, lower maximum cylinder pressure for DE10 was observed at 2300 rpm with the DE10 fuelling produced higher cyclic variation over 200 consecutive cycles.

4. Conclusion

The conclusions of the research work are listed as follow:

i. For DE10 fuelling at both speed conditions, the values of the coefficient of variation (COV) for indicated mean effective pressure (IMEP), COVIMEP increases comparatively to the values for only B0 fuelling, a fact which shows the improvement of the general response of the engine at full load condition.

ii. The value of $COVP_{max}$ for DE10 fuelling increases of 338.5% at full load condition when the speed increases from 1100 rpm to 2300 rpm. Also, for all speed conditions, the DE10 fuelling leads to the rise of cyclic variations for P_{max} value comparative to B0 fuelling. The increasing tendency shown for $COVP_{max}$ is related to the variation of COVIMEP and shows the improvement of the combustion process at DE10 fuelling.

iii. For the full load condition, the cyclic variations for B0 are improved at speed increasing comparative to the DE10 fuelling, which indicates that B0 has better combustion stability.

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