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Alexandrian Laurel for Biodiesel Production and its Biodiesel Blends on Performance, Emission and Combustion Characteristics in Common-Rail Diesel Engine

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Abstract: A two-step transesterification process was employed in the biodiesel production from non-edible *Alexandrian Laurel*. The key physicochemical properties of the *Alexandrian Laurel* biodiesel (ALB), diesel and blends of both fuels were compared and analyzed. The effects of blending biodiesel (ALB) and petroleum diesel on engine performance, combustion and exhaust emissions were investigated in a turbocharged, high-pressure common-rail diesel engine under six different speed operations and at full load conditions. The test fuels comprised a conventional diesel fuel and four different fuel blends of ALB. The results showed relatively close physicochemical properties of ALB and its blends when compared with petroleum diesel. However, the use of ALB-blended fuel resulted in penalties engine brake power, brake specific fuel consumption (BSFC) despite slightly improved brake thermal efficiency (BTE). Brake specific nitrogen oxide (BSNOx) was found worsened with higher ALB content in the blends. Nonetheless, consistent improvements in brake specific carbon monoxide (BSCO), brake specific carbon dioxide (BSCO$_2$), and smoke were noticed when ALB blends were used. Additionally, ALB blends contributed to reduction in peak combustion pressure, peak heat release rate (HRR) and combustion duration. In general, the findings suggest satisfactory operation with ALB biodiesel-diesel blends in an unmodified diesel engine.

Keywords: non-edible; combustion; *Alexandrian Laurel* oil; biodiesel; common-rail; emissions

1. Introduction

Nowadays, people all around the world are forming a single community due to their shared strong demand for energy to fulfill their daily needs [1]. Presently, the transportation sector heavily depends on petroleum fossil oil and its demand is growing day by day [2,3]. On the other hand, the pitfalls of fossil fuel consumptions have been brought to light. It has been claimed that petroleum diesel fuel takes the major contribution to environmental pollution besides creating adverse health issues for living things [4,5]. As reported by the World Health Organization, roughly 360,000 premature deaths are happening every year in the region of Asia as a result of the air pollution in the main cities [6]. In addition, decline of the stock of fossil fuels were also reported. This has stimulated the use of renewable and environmentally friendly biofuels as alternatives to conventional fossil fuels [7,8].

A number of attempts have been taken in order to reduce the reliance on petroleum fuels, including the development of renewable energy sources [9–11]. Renewable and non-toxic biodiesel, usually derived from plants and animal fats as a substitute of petroleum diesel is claimed to be the most promising solution to limit the conventional diesel fuel, particularly in transportation [12,13]. In this quest, the most successful method found is to replace a part of the fossil diesel with the biodiesel through blending of both fuels [14,15]. In fact, one of the major advantages of using biodiesel is it requires minor or no modification to the traditional diesel engines [16,17].

The use of biodiesel also creates positive impacts on the environment when compared to conventional fossil diesel. As indicated in studies by many researchers, diesel engines that run on biodiesel or its blends were found to emit less carbon monoxide (CO), unburned hydrocarbon (HC) and particulate matter (PM) [18–20]. However, the variations in nitrogen oxides (NO_x) emission from biodiesel-fueled engines showed no unified trends, but it is mostly recorded with an inclination towards increased NO_x emission from the engines [21–23]. This increment is referred to as the "biodiesel NO_x effect" [24,25]. It is also claimed that the rise in NO_x emission is not only dominated by a single factor, but may be due to combined effects of multiple factors that rely on each other, especially the complex physicochemical properties of the fuel [26].

Generally, biodiesel fuel properties depend on its raw material [27]. Biodiesel possesses some variations in its physicochemical properties with respect to conventional diesel fuel. These properties include calorific value, cetane number, viscosity, cloud and pour points as well as the oxygen content [28–30]. Therefore, variations in engine performance and emissions are usually attained when using biodiesel since the combustion characteristics such as ignition delay, combustion duration, heat release rate, peak in-cylinder pressure and power are a strong function of the physicochemical properties of the fuel [31].

Ng et al. studied the impact of blend fraction and biodiesel fuel with a wide spectrum of level of saturation and fatty acid composition on light-duty diesel engine exhaust emissions. The experiment involved different types of biodiesel methyl ester, including those derived from coconut oil, palm oil and soybean oil. It was concluded that all the biodiesel fuels indicated a reduction in the HC emission. For the NO emission, soybean biodiesel reported an increment of 8.4%, but coconut and palm oil biodiesel showed a decrement of 5.4%, as compared to the diesel fuel. The higher number of double bonds in the fatty acid composition of the soybean biodiesel resulted in the higher emission of NO, as discussed by the researcher [32].

A biodiesel is classified as second generation feedstock biodiesel if it is produced from non-edible harvests, such as Jatropha curcas, Ceiba pentandra, Karanja, Neem, Jojoba, and rubber seed [33]. In this work, the use of a second generation biodiesel derived from *Alexandrian Laurel* in a diesel engine was investigated. *Alexandrian Laurel* or Calophyllum inophyllum, commonly known as Penaga Laut or Bintangor in Malaysia is a member of *Guttiferae Juss* family that produces non-edible oilseeds. In Greek word, the scientific name of "*Calophyllum*" means "*beautiful leaf*". The use of non-edible feedstock has the advantage over the first generation edible ones in biodiesel cost due to lower feedstock price since it does not interfere food commodity supplies. Besides, non-edible feedstocks like *Alexandrian Laurel* can be grown on degraded lands that may be unsuitable for edible feedstocks, thus avoiding issues such as deforestation, crucial soil resources destruction, and exploitation of the available arable land [34]. Potentials of *Alexandrian Laurel* as a biodiesel can be seen from its high survival potency and wide availability in South East Asia, India and Australia [35]. Furthermore, it also exhibits relatively higher oil yield compared to other non-edible feedstocks [36].

Purpose of Study

Most of the studies in the past concentrated on the impact of edible biodiesel on the engine performance and exhaust emissions, but there is an increasing trend of studies on non-edible biodiesels recently [37]. As prolonged reliance on edible feedstocks for biodiesel production will threaten the food industry and raise environmental issues, there is an urge for fellow researchers to come up with suitable biodiesel based on non-edible feedstock. Notably, this work investigated a new alternative biodiesel feedstock as a replacement for conventional diesel fuel. However, in the present, there are only few studies performed to date on combustion characteristics using *Alexandrian Laurel* biodiesel (ALB) blends, results on important parameters such as combustion duration are also not yet widely accessible.

To contribute as one of the many efforts toward sustainable solutions, this work was intended to provide insights on the effects of using blends of diesel with *Alexandrian Laurel* biodiesel (ALB) in a turbocharged common-rail direct-injection diesel engine. Engine-out characterizations including engine performance, emissions and combustion characteristics tests with four levels of blends were conducted and compared with conventional diesel to give a better picture of the combustions occurred to reduce the existing research gap. Additionally, in this study, quantification of parameters with units that provide more direct implication was made for the ease of comparison. For example, the unit of g/kWhr was used for engine emissions instead of ppm.

2. Methodology

2.1. Materials

In this work, commercial Grade D2 diesel fuel was obtained from a local gas station in Malaysia, while the biodiesel was derived from *Alexandrian Laurel* oil. The crude *Alexandrian Laurel* oil used in this study was obtained from Central Java, Indonesia. In current study, the ALB fuel was derived from the crude *Alexandrian Laurel* oil by using transesterification process with associated pre-treatments (esterification and neutralization) and post-treatments (purification and drying).

2.2. Fuel Properties Test and Analysis

The result of fatty acid composition of the crude *Alexandrian Laurel* oil is presented in Table 1. It was found that this oil contained higher levels of unsaturated fatty acids (46.1% oleic acid and 24.7% linoleic acid) than saturated ones (14.5% palmitic acid and 13.2% stearic acid). A similar result was also reported by Atabani et al. [38] on fatty acid compositions of crude *Alexandrian Laurel* oil. Essentially, the distribution of fatty acid composition has direct impact on physicochemical properties of biodiesel. Most non-edible oil, including *Alexandrian Laurel*, contain higher level of double carbon chain that will affect the key properties of biodiesel such as kinematic viscosity, cetane number, oxidation stability and calorific value [39]. In fact, long carbon chain length increases the calorific value and significantly alters biodiesel properties under cold conditions [40]. Particularly, a higher degree of unsaturated fatty acids can substantially enhance cold filter plugging points and cloud point of biodiesel [41].

In Table 2, the physicochemical properties of crude oil and neat ALB as determined according to American Society for Testing and Materials (ASTM) and European (EN) standards are shown. It is well known that kinematic viscosity of a fuel has a dominant effect on its operation in diesel engine due to highly sensitive fuel injection system. Employing high viscosity biofuel, such as straight vegetable oil in diesel engines, will inevitably cause issues like poor fuel atomization, combustion inefficiency and engine deposits. From the fuel test results, it is evident that the viscosity of crude *Alexandrian Laurel* oil was reduced by a factor of 12 after its conversion into biodiesel via transesterification process. Other key fuel properties such as density, calorific value and acidity were also simultaneously enhanced via this process. This implies the effectiveness of transesterification process used in this study in resolving the unfavorable viscosity issue in biofuels. Besides, based on the results of comprehensively measured ALB properties, the produced biodiesel apparently satisfies biodiesel standards as in ASTMD6751 and EN14214.

Table 1. Fatty acid composition of the biodiesel.

No.	Fatty Acid Name (Systematic)	Structure	Formula	Molecular Mass	Mass Fraction (wt.%)
1	Lauric (Dodecanoic)	12:0	$C_{12}H_{24}O_2$	200	0.1
2	Myristic (Tetradecanoic)	14:0	$C_{14}H_{28}O_2$	228	0.1
3	Palmitic (Hexadecanoic)	16:0	$C_{16}H_{32}O_2$	256	14.5
4	Stearic (Octadecanoic)	18:0	$C_{18}H_{36}O_2$	284	13.2
5	Arachidic (Eicosanoic)	20:0	$C_{20}H_{40}O_2$	312	0.8
6	Palmitoleic (Hexadec-9-enoic)	16:1	$C_{16}H_{30}O_2$	254	0.3
7	Oleic (Cis-9-Octadecanoic)	18:1	$C_{18}H_{34}O_2$	282	46.1
8	Linoleic (Cis-9-cis-12 Octadecanoic)	18:2	$C_{18}H_{32}O_2$	280	24.7
9	Linolenic (Cis-9-cis-12)	18:3	$C_{18}H_{30}O_2$	278	0.2
				Saturated fatty acid	28.7
				Unsaturated fatty acid	71.3
				Total	100

Table 2. The fuel properties of crude oil and neat *Alexandrian Laurel* biodiesel (ALB) in comparison with ASTM and EN standards.

Properties	Unit	Crude *Alexandrian Laurel* Oil	Limit (ASTM D6751)	Limit (EN 14214)	*Alexandrian Laurel* Biodiesel	Test Method
Oil content	%	75	-	-	-	-
Free fatty acid	%	29.66	-	-	-	-
Kinematic viscosity @ 40 °C	mm^2/s	53.17	1.9–6.0	3.5–5.0	4.27	D445
Density @ 15 °C	kg/m^3	951.1	880	860–900	878.5	D127
Acid number	mg KOH/g	59.33	0.5 max	0.5 max	0.45	D664
Calorific value	MJ/kg	38.51	-	35	40.1	D240
Flash point	°C	195.5	130 min	120 min	168.5	D93
Pour point	°C	-	-	-	2	D2500
Cloud point	°C	-	report	-	2	D2500
Cold filter plugging point	°C	-	-	-	1	D6371
Oxidation stability @ 100 °C	hours	-	3 min	6 min	13.08	EN14112
Cetane number	-	-	47 min	51 min	59.6	D6890
Carbon	wt.%	-	77	-	75.8	D5291
Hydrogen	wt.%	-	12	-	12.5	D5291
Oxygen	wt.%	-	11	-	11.72	D5291

Generally, diesel engines can be run with biodiesel that is either in its pure form or as a blend with baseline diesel. Low-biodiesel blend, which usually consists of up to 30% of biodiesel content is the most common biodiesel blends used nowadays as it provides good compensations in terms of engine emissions and cold-weather performance. Less biodiesel content in blends also offers benefits in production cost and better material compatibility than high-level blends. Moreover, operation cost can be significantly reduced with lower-level blends because they require no or only little engine modifications.

In this study, four biodiesel blends, namely ALB10 (10% biodiesel, 90% petroleum diesel), ALB20 (20% biodiesel, 80% petroleum diesel), ALB30 (30% biodiesel, 70% petroleum diesel) and ALB50 (50% biodiesel, 50% petroleum diesel) were prepared on volume basis and tested. The physicochemical properties of the blends are listed in Table 3. It is evident that the physicochemical properties of all the biodiesel blends satisfy the ASTM D7467 biodiesel blend standards. Moreover, it appears that blending ALB with petroleum diesel substantially enhanced the resultant biodiesel blend properties. For instance, reduction in kinematic viscosity of the blended fuel with higher petroleum diesel content in the blends. Additionally, flash points for all the biodiesel blends were found relatively higher than that of baseline diesel as a result of blending. This indicates that ALB–diesel blends are much more suitable for application as a transportation fuel [42]. In addition, cetane number, which is influential on engine performance, emissions, and combustion characteristics, was found to be relatively higher for all biodiesel blends. Nonetheless, disadvantageous effects of blending can also be noticed as biodiesel blends were tested with lower calorific values than unadulterated diesel.

Table 3. The fuel properties of diesel fuel and its blends with ALB.

Properties	Unit	Diesel Fuel	Biodiesel Blends		ALB10	ALB20	ALB30	ALB50
			Limit (ASTM D7467)	Test Method				
Kinematic viscosity @ 40 °C	mm^2/s	3.34	1.9–4.1	D445	3.55	3.61	3.98	4.25
Density @ 15 °C	kg/m^3	839.0	858 max	D127	851.0	855.1	857.9	867.9
Acid number	mg KOH/g	0.12	0.3 max	D664	0.17	0.19	0.22	0.28
Calorific value	MJ/kg	45.31	35	D240	44.80	44.17	43.58	42.34
Flash point	°C	71.5	52	D93	77.5	79.5	82.5	83.5
Pour point	°C	1	Not specified	D2500	0	2	3	4
Cloud point	°C	8	Not specified	D2500	4	4	5	4
Oxidation stability @ 100 °C	hours	>30	6	EN14112	25.08	24.08	20.08	19.26
Cetane number	-	52	47 min	D6890	52.4	53.9	55.8	56.7

2.3. Engine Setup and Instrumentation

In this work, the effect of the variation in two parameters, the concentration of ALB in the blend (i.e., 10%, 20%, 30% and 50% by volume basis) and engine speed (from 1500 to 4000 rpm with an increment of 500 rpm between each) on engine-out responses were investigated under full load conditions. In this study, the performance settings of the engine were tested using the full-load curve. The engine is operated at full load condition to provide a clear analysis of the effect of various fuel blends on engine operation parameters. This is due to the fact that at full load (maximum power), engines demand maximum fuel rates. Additionally, the engine speed is varied from 1500 to 4000 rpm with increments of 500 rpm. These six levels of engine speeds under full load condition were selected as the most representative of a wide variety of engine operating ranges.

The test engine employed in this work is a four-cylinder, turbocharged diesel engine with common-rail injection system. The engine specifications are shown in Table 4. The test engine load and speed were monitored by a 150-kW eddy current dynamometer. A Bosch air mass sensor was installed upstream of the intake manifold to measure the flow rate of intake air. On the other hand, to assess the engine fuel consumption during the experiment, a fuel flow meter was utilized. Besides, several K-type thermocouples were also employed to quantify temperatures of intake air, exhaust gas, lubricant oil and cooling water. The set-up of the apparatus in this study is illustrated in Figure 1.

A Kistler 6058A piezoelectric sensor was employed to measure the in-cylinder pressure for combustion characteristics analysis in this study. The pressure sensor was fixed on a glow plug adapter and installed onto the first engine cylinder. The instantaneous signal from the pressure sensor was transmitted to a charge amplifier before a high-speed data acquisition system (DAQ) to enable data processing for heat release rate and combustion duration. An incremental encoder with 0.125°CA resolution was used to monitor the engine crankshaft rotational displacement. The data from the encoder were also fed into the DAQ for synchronized with cylinder pressure data. In each test, cylinder pressure was continuously measured for 100 cycles, the data obtained were then recorded and averaged. Tests on exhaust gas emissions and smoke opacity were both sampled using an AVL DiCom 4000 emission tester. This instrument has the functions of DiGas and DiSmoke for 5-gas and smoke opacity measurement, respectively. In this study, the exhaust components of CO, CO_2 and NO_x have been converted from concentrations (% vol/ppm) to g/kWhr because it is more ecologically representative of estimating the amount of exhaust emissions released into environment. The CO, CO_2 and NO_x emissions were converted into brake specific emissions by using the following equations according to SAE J177:

$$BSCO_2\ (g/kWhr) = 10 \times CO_2\ (\%\ vol) \times \text{Exhaust mass flow rate (kg/min)} \times 60/\text{Brake power (kW)} \quad (1)$$

$$BSCO\ (g/kWhr) = 0.0580 \times CO\ (ppm) \times \text{Exhaust mass flow rate (kg/min)}/\text{Brake power (kW)} \quad (2)$$

$$BSNO_x\ (g/kWhr) = 0.0952 \times NO_x\ (ppm) \times \text{Exhaust mass flow rate (kg/min)}/\text{Brake power (kW)} \quad (3)$$

The fuel injection timing in this work was determined from the injector voltage monitored by the DAQ. In addition, the Electronic Control Module (ECM) data including the fuel rail pressure, engine load, manifold pressure and manifold air temperature were all recorded by using an On-Board Diagnostic (OBD) connection.

Table 4. Specifications of the test engine.

Engine Type	Diesel, four strokes, turbocharged, DI
Fuel system	High-pressure common-rail (up to 140 MPa)
Number of cylinders	4
Number of valves per cylinder	2
Bore	76.0 mm
Stroke	80.5 mm
Displacement	1461 cm^3
Compression Ratio	18.25:1
Maximum power	48 kW @ 4000 rpm
Maximum torque	160 Nm @ 2000 rpm

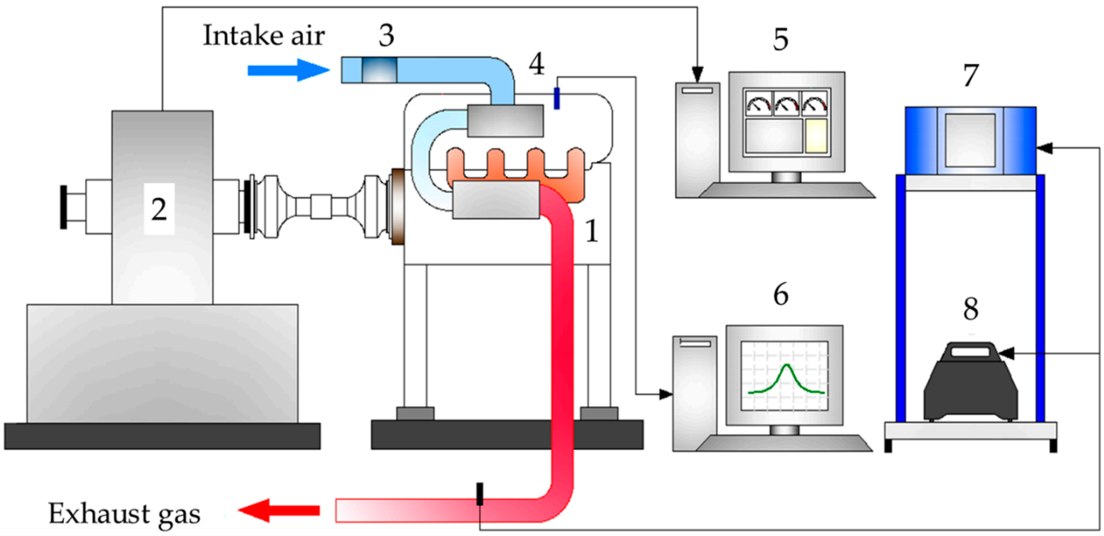

Figure 1. Schematic diagram of the experiment set up [43]. 1. Test engine, 2. eddy current dynamometer, 3. air mass sensor, 4. pressure sensor, 5. dynamometer controller, 6. data acquisition system, 7. AVL gas analyzer, 8. smoke opacity meter.

2.4. Heat Release Rate (HRR) Analysis

As a major part of the combustion characteristics analysis, HRR analysis provides insights into the fuel combustion process and facilitates the assessment of combustion-related parameters such as combustion rate, ignition delay time and combustion duration; HRR was calculated based on the in-cylinder pressure and cylinder volume data. Based on the First Law of Thermodynamics, the HRR formula used is as specified in Equation (4).

$$\frac{dQ}{d\theta} = \frac{\gamma}{\gamma - 1} P \frac{dV}{d\theta} + \frac{1}{\gamma - 1} V \frac{dP}{d\theta} \tag{4}$$

where the variables are: HRR per crank angle ($\frac{dQ}{d\theta}$), crank angle (θ), pressure (P), cylinder volume (V) and specific heat ratio (γ).

2.5. Statistical and Equipment Uncertainty Analysis

Generally, all experimental measurement is subject to some uncertainties or errors. The uncertainty in an experimental result can arise from sensor selection, condition, calibration, observation, and test procedure. The summary of the equipment used in this study including the measurement range and accuracy of the instruments is given in Table 5. Uncertainty analysis is essential to verify the accuracy of the experiments. Hence, the percentage uncertainties of various parameters such as brake thermal efficiency (BTE) and brake specific fuel consumption (BSFC) were determined based on the percentage uncertainties of the instruments used in the experiments. The overall experimental uncertainty was determined using the following equation:

Overall experimental uncertainty = square root of ((uncertainty of fuel flow rate)^2 + (uncertainty of BSFC)^2 + (uncertainty of BTE)^2 + (uncertainty of NOx)^2 + (uncertainty of smoke)^2 + (uncertainty of pressure sensor)^2 + (uncertainty of crank angle encoder)^2) = square root of ((0.5)^2 + (1.5)^2 + (1.7)^2 + (1.3)^2 + (1)^2 + (0.5)^2 + (0.03)^2) = ± 2.9%

Table 5. List of measurement range, accuracy, and percentage uncertainties.

Measurement	Measurement Range	Accuracy	Measurement Techniques	% Uncertainty
Load	±600 Nm	±0.1 Nm	Strain gauge type load cell	±0.25
Speed	0–10,000 rpm	±1 rpm	Magnetic pick up type	±0.1
Time	-	±0.1 s	-	±0.2
Fuel flow measurement	0.5–36 L/hr	±0.04 L/hr	Positive displacement gear wheel flow meter	±0.5
NO_x	0–5000 ppm	±1 ppm	Electrochemical	±1.3
Smoke	0–100%	±0.1%	Photodiode detector	±1
Pressure sensor	0–25,000 kPa	±10 kPa	Piezoelectric crystal type	±0.5
Crank angle encoder	0–12,000 rpm	±0.125°	Incremental optical encoder	±0.03
Computed				
Brake specific fuel consumption (BSFC)	-	±5 g/kWhr	-	±1.5
Brake thermal efficiency (BTE)	-	±0.5%	-	±1.7

3. Results and Discussion

3.1. Engine Performance

3.1.1. Brake Power

Figure 2a illustrates the engine brake power of the fuel samples tested with engine speed variation. Generally, it is evident that the brake power gradually increased with increasing engine speed for all fuel types. However, brake powers for all fuels at 1500 rpm was relatively lower compared to when the engine was run at other engine speeds. At the given operating speeds, it can be observed that baseline diesel developed the highest brake power consistently across engine speeds, followed by ALB10, ALB20, ALB30 and lastly ALB50. For instance, the ALB50, ALB30, ALB10 and ALB20 fuels developed brake powers less than baseline diesel by 6.9%, 2.5%, 0.8%, and 0.7%, respectively, at the engine speed of 4000 rpm. The reduction in brake power with ALB–diesel-blended fuel is in congruent with Sahoo et al.'s results [44]. The low calorific value of ALB50 may be the cause of its great reduction in engine power and performance. This result is also in good agreement with Debnath et al.'s study which reported a lower calorific value of palm methyl ester than baseline diesel and observed significant power output reduction for the biodiesel fuel [45].

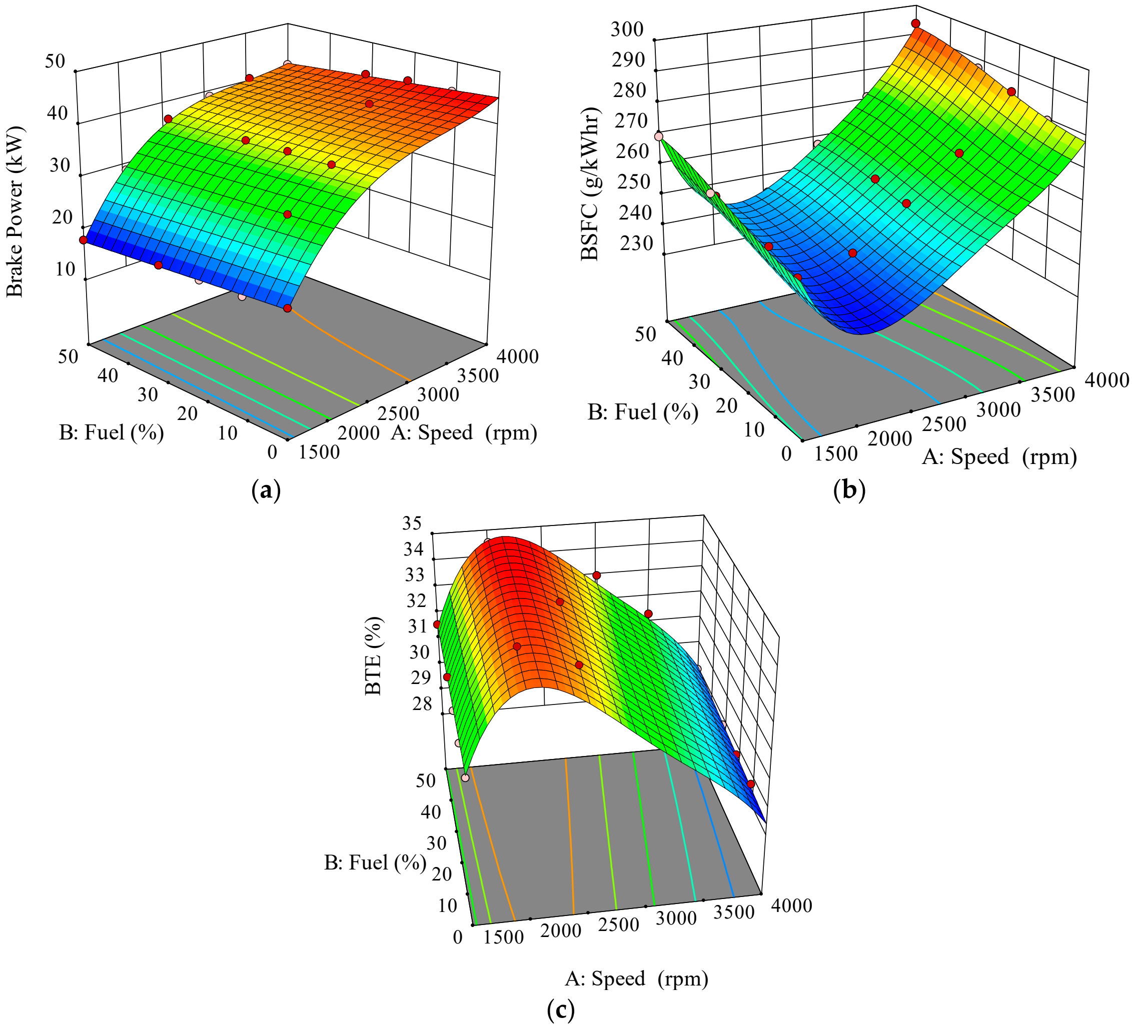

Figure 2. Interactive effect of fuel blends on (**a**) brake power, (**b**) BSFC and (**c**) BTE at varying speeds.

3.1.2. Brake Specific Fuel Consumption

Figure 2b presents the brake specific fuel consumption (BSFC) of all the fuel when operated with a range of engine speed. BSFC is defined as the ratio of fuel consumption rate to the output brake power, higher BSFC corresponds to more consumed fuel with less power developed. There was a decrement in BSFC when the engine speed went up from 1500 to 2000 rpm, reflecting the relatively lower brake power generated by the engine at 1500 rpm before steady BSFC increments for further increased engine speeds. This phenomenon may also be due to the boost threshold of the turbocharger employed in this work to produce significant boost for combustion efficiency improvements. As the boost pressure is a function of intake air flow rate, thus this phenomenon can be clearly seen with the increasing effect in intake air flow rate, as demonstrated in Figure 3. This result is in good agreement with the variations of BSFC in Karabektas's research [46] and Palash et al.'s research [47] that employed rapeseed oil methyl ester and Aphanamixis polystachya oil methyl ester, respectively, in a turbocharged diesel engine.

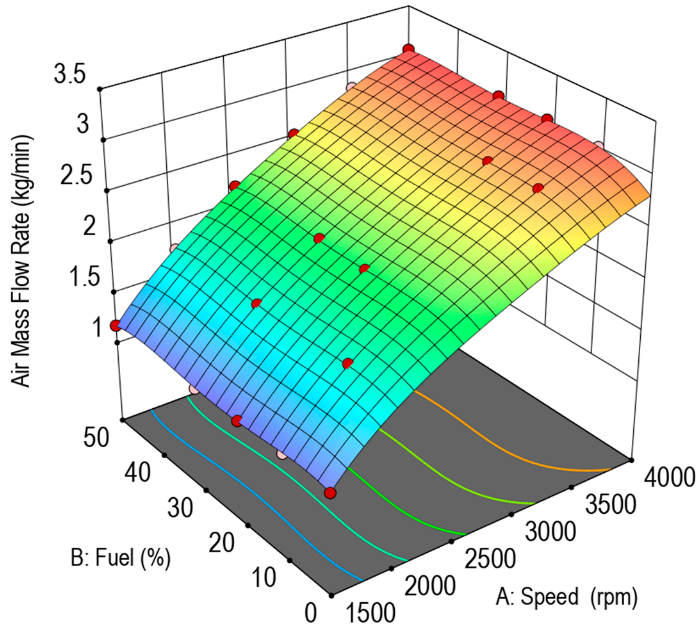

Figure 3. Interactive effect of fuel blends on intake air mass flow rate at varying speeds.

From the illustration, it is noticeable that BSFC for all ALB–diesel blends were greater than unadulterated diesel. There was also an increment trend across all engine speeds when the concentration of ALB in the blends increased. For example, at the engine speed of 2000 rpm, it was observed that the BSFC was lowest for baseline diesel (233.1 g/kWhr). The BSFC increased in ascending order for ALB10, ALB20, ALB30 and ALB50 with 233.6 g/kWhr, 238.1 g/kWhr, 241.7 g/kWhr and 246.2 g/kWhr, respectively. This may be attributed to the relatively lower calorific value of ALB compared to diesel which was approximately 11.5% less than that of diesel fuel as shown in Tables 3 and 4. Since ALB had lower energy content in it, higher amount of fuel was needed for the same engine load. These results are in agreement with those found by Kivevele et al. [48].

3.1.3. Brake Thermal Efficiency

The brake thermal efficiency (BTE) for all fuel samples tested is shown in Figure 2c. Generally, the BTE decreased when higher engine speed was employed except from 1500 to 2000 rpm. For example, BTE for diesel fuel recorded 31.1%, 34.1%, 33.5%, 31.9%, 30.4% and 28.6% at 1500 rpm, 2000 rpm, 2500 rpm, 3000 rpm, 3500 rpm and 4000 rpm, respectively. The decline in BTE may be attributed to the rise in BSFC and higher engine friction losses at higher engine speed. However, the lower BTE at low engine speed of 1500 rpm may also be explained by reduced output power due to the turbocharger boost threshold.

In addition, it can be observed that BTEs of the ALB–diesel blends were slightly superior compared to that of diesel when operated at any speeds. This reflects the shorter combustion durations of the blended fuels compared to diesel. Additionally, BTE was found to be greater when higher ALB content was present in the blends. The maximum difference in BTE was observed between diesel and ALB50 fuels at the engine speed of 3000 rpm, there was an increment of 1.1% BTE when the half of the diesel fuel content was replaced by ALB. The improvements in BTE when using ALB–diesel blends compared to diesel may be attributable to superior lubricity, friction and wear characteristics of the blends [49,50]. The increase in BTE with the addition of ALB in the blends may be credited to the early initiation of combustion and faster combustion process due to the higher cetane number of ALB (14.6% higher than diesel), and thus caused a reduction in the wall heat loss that contributes to combustion inefficiency. Another possible explanation is the increased availability of fuel bound oxygen content in the biodiesel fuels; this improves the efficiency of combustion and causes larger BTE when compared with baseline diesel [51].

3.2. Exhaust Emissions

3.2.1. Brake Specific Nitrogen Oxide

Figure 4a indicates the brake specific nitrogen oxides ($BSNO_x$) emission of the tested fuels obtained at various speed conditions. Overall, it is noticeable that as the engine speed increased, higher NO_x emission was recorded for all the fuel samples. This may be due to higher in-cylinder temperature caused by friction-generated heat when the engine was run at a higher speed that favors NO_x formation through the Zeldovich mechanism. This result is similar with the variations of NO_x in Ruhul et al.'s study [52]. Then, ALB fuel blends also revealed higher NO_x emission compared to the baseline diesel under all tested engine speeds. Similar results were also reported in several studies, including which by Argawal [53], Devan et al. [54], Sahoo et al. [55] and Kumar et al. [56]. This phenomenon may be attributable to the intrinsic greater oxygen content in ALB; about 11.7%wt of the ALB biodiesel was reported to be oxygen as shown in Table 2. This abundance of oxygen in the fuel might promotes oxidation of nitrogen that leads to NO_x formation [57]. Besides, the fuel blend that contains higher biodiesel concentration, such as ALB50 had consistently exhibited higher NO_x emissions regardless of the engine speed. For instance, the ALB50 exhibited the highest NO_x emission of 10.70 g/kWhr followed by ALB30 (10.15 g/kWhr), ALB20 (9.94 g/kWhr), ALB10 (9.58 g/kWhr) and the lowest by diesel (9.35 g/kWhr) at 4000 rpm. A similar trend was also observed by Silitonga et al. [58]. The higher level of unsaturated fatty acids composition in ALB, which accounts for approximately 71.3% of its structure might contribute to higher adiabatic temperatures during combustion that ease the thermal NO_x formation pathway [59].

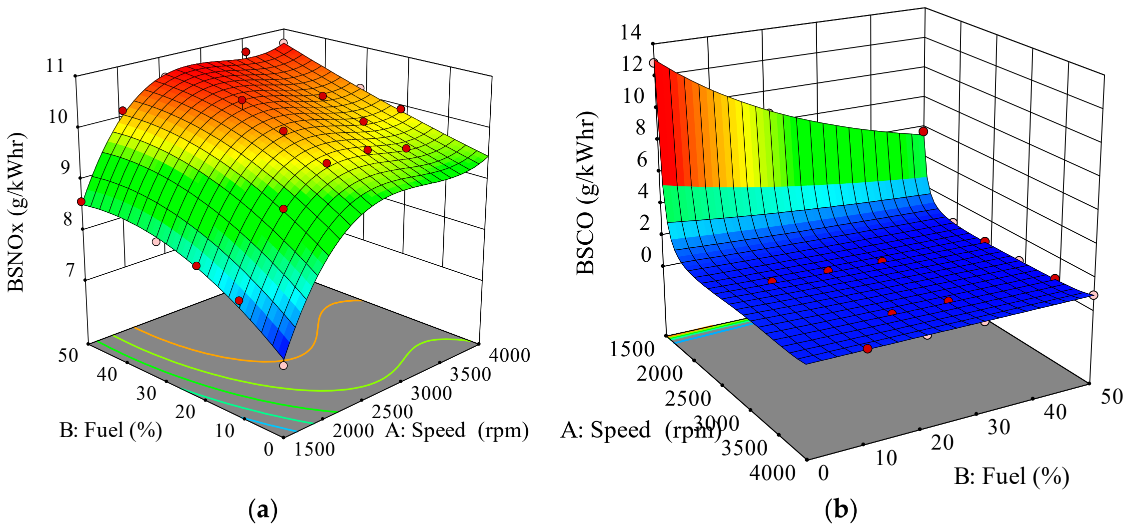

Figure 4. Interactive effect of fuel blends on (**a**) brake specific nitrogen oxide (BSNOx) and (**b**) brake specific carbon monoxide (BSCO) at varying speeds.

3.2.2. Brake Specific Carbon Monoxide

The brake specific carbon monoxide (BSCO) emission traces with increasing engine speed is depicted in Figure 4b. It is clearly shown that CO emission recorded was relatively higher at low engine speed of 1500 rpm than at higher engine speeds. For instance, BSCO for diesel was 12.838 g/kWhr at 1500 rpm compared to 1.322 g/kWhr at 2000 rpm. As explained in previous sections, this phenomenon may be due to the turbocharger boost threshold in this experiment.

CO emissions for engine speeds of 2000 to 4000 rpm were found comparable, despite a slight decline of CO emission at 3000 rpm. CO emission of the fuels also consistently reduced with higher ALB content in them, the lowest CO emission was observed for ALB50 (0.481 g/kWhr) at 3000 rpm. On the other hand, significant improvements in CO emission by using ALB–diesel blends instead of baseline diesel were especially apparent at 1500 rpm. There was a 55.4% reduction of CO emission (12.838 g/kWhr to 5.726 g/kWhr) when half of the diesel was replaced by ALB. These findings are well aligned with results in Ashok et al.'s study that utilized calophyllum inophyllum methyl ester in a diesel engine operated at constant 1500 rpm [60].

In fact, CO emission in engines is attributed to partial oxidation of carbon atoms due to lack of sufficient oxygen to fully oxidize them to carbon dioxide, especially in fuel-rich regions during combustions. The use of oxygenated fuel like ALB is expected to support better combustion quality and hence reduce CO emissions [61,62]. From the results obtained in this work, this theory is applicable to explain the decreasing trend in CO emission with higher level ALB–diesel blends that contains higher biodiesel concentration, such as ALB50 has consistently exhibited lower CO emissions over the entire range of engine speeds.

3.2.3. Brake Specific Carbon Dioxide

Figure 5a represents the variation of brake specific carbon dioxide ($BSCO_2$) for all of the fuel samples at different engine speeds. Generally, it can be observed that the CO_2 emissions for all fuel blends increased with higher engine speed except the rather apparent decrement from 1500 to 2000 rpm, which might be a consequence of turbocharger boost threshold. In addition, all the ALB fuel blends emitted lower CO_2 compared to the baseline diesel under all tested engine speeds. This finding is in good agreement with the CO_2 results obtained in a study by Ong et al. [63]. It was also found that when the engine was operated at 2000 rpm, the biodiesel fuel blends ALB50, ALB30, ALB20 and ALB10 produced 6.58%, 4.75%, 3.98% and 1.82% less CO_2 emissions than the baseline diesel, respectively. This may be attributed to the lower carbon to hydrogen ratio of the biodiesel fuel compared to the diesel fuel [49,64]. Although the combustion of biofuel produces CO_2, biofuel has advantages over conventional diesel in life cycle CO_2 emission. A study stated that reduction of net CO_2 emission by 78.45% could be achieved by using biofuels as a replacement for diesel [65].

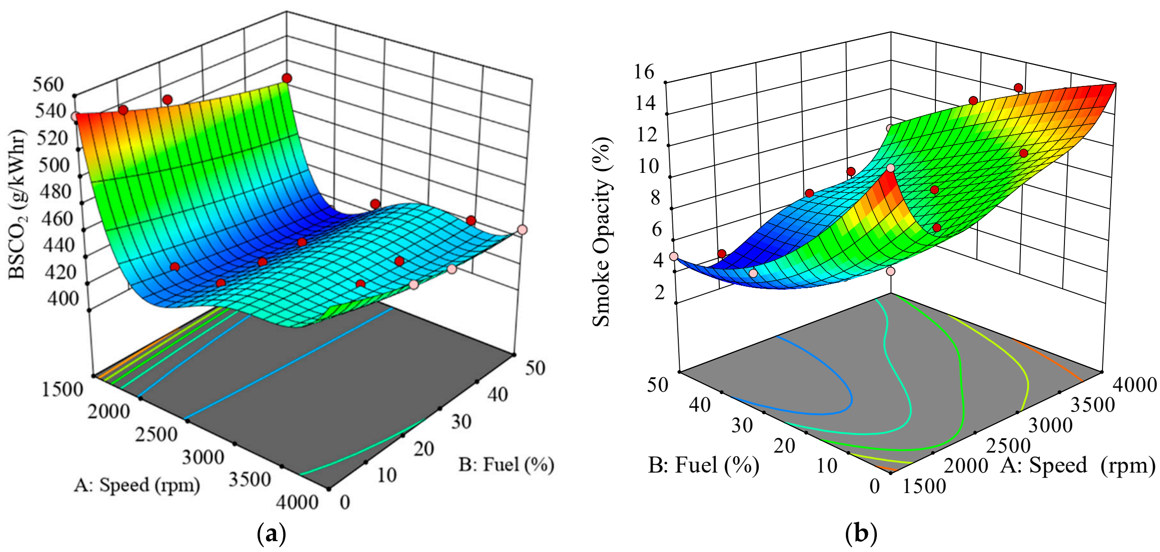

Figure 5. Interactive effect of fuel blends on (**a**) brake specific carbon dioxide ($BSCO_2$) and (**b**) smoke opacity at varying speeds.

3.2.4. Smoke Opacity

The variation of smoke opacity for all the tested fuels at a range of engine speeds is shown in Figure 5b. In general, it is found that the smoke emissions reduced from engine speed of 1500 to 2000 rpm and then increased with further rise in speed. The reduction at low engine speed may be a result of the use of turbocharger in this work, which might only be in optimum operation when engine speed exceeds 1500 rpm. The lowest smoke opacity for all fuels was observed at 2000 rpm; the speeds were 4.2%, 4.6%, 5.8%, 7.2% and 8.8% for ALB50, ALB30, ALB29, ALB10 and diesel, respectively.

All ALB–diesel blends recorded improved smoke opacity than that of diesel. Moreover, higher concentration of ALB in the fuel blends had a tendency to decrease the opacity of smoke emitted. Similar improvements in smoke emission when using *Alexandrian Laurel* biodiesel in CI engine were reported by Ong et al. [63], Fattah et al. [66] and also Sahoo et al. [44]. A research by Mishra et al. also reported significant decline in smoke opacity when *Calophyllum* Methyl Ester was used at high loads [67].

Furthermore, it can be seen that with the ALB fuel blends, the reduction in smoke opacity for ALB–diesel blends was more prominent at higher engine speed. It is evident that the maximum 56.8% reduction of smoke opacity occurred between baseline diesel and ALB50 fuels at 4000 rpm. This may be a result of the oxygenated nature of ALB blends, which results in consistent complete combustion [53]. Significant reduction in smoke opacity may also be attributed to the lower carbon residue of ALB, which reduces soot formation in the engine [68]. Higher oxygen content in ALB also may reduce the fuel-rich zone during combustion and leads to soot nuclei oxidation and hence resulted in cleaner exhaust [69].

3.3. Combustion Characteristics

3.3.1. Cylinder Combustion Pressure

The plot of cylinder pressure at various crank angles for all the tested fuels is shown in Figure 6a. Specifically, the subplots represent the plot of cylinder pressure at various engine speeds of 2000 rpm, 3000 rpm and 4000 rpm, respectively. Generally, the rise of the ALB fraction in the blends resulted in the deterioration of the cylinder pressure for all the engine speeds. Based on Figure 6a, it was observed that the cylinder pressure of ALB-blended fuels was lower than that of the neat diesel fuel at 2000 rpm. For instance, the baseline diesel recorded the highest peak cylinder pressure of 107.2 bar whereas the ALB50 marked the lowest peak cylinder pressure of 102.9 bar, both at the crank angle of 14° ATDC. With the rise of engine speed to 3000 rpm, the ALB20 indicated the highest peak cylinder pressure of 107.2 bar, as shown in Figure 6b. Meanwhile, the ALB50 showed the lowest peak cylinder pressure again, with the pressure of 106.0 bar. The highest and the lowest cylinder pressures were at the crank angle of 10° ATDC. Furthermore, the situation is similar to that at 3000 rpm when the engine speed comes to 4000 rpm, with the largest and lowest peak cylinder pressure of 112.1 bar and 110.5 bar accordingly, both at 8° ATDC indicated by the ALB20 and ALB50, respectively, as shown in Figure 6c.

Similar decrements in in-cylinder pressure were also reported by Mishra et al. when the authors examined in-cylinder combustion with *Calophyllum* methyl ester blends over 91 consecutive cycles at engine full load [67]. A research study carried out by S. Gnanasekaran et al. [70] on the combustion behavior of a compression-ignition engine also marked a similar observation. The fish oil biodiesel blends exhibited relatively lower cylinder pressure with respect to the neat diesel fuel, as a consequence of the shorter ignition delay. To summarize, the decrement of peak cylinder pressure with the increment of ALB content in the blends was probably due to the significantly lower calorific value of ALB with respect to that of the neat diesel fuel. In addition, the higher viscosity property of the ALB resulted in the slower vaporization process, hence lower combustion rate and lower peak cylinder pressure produced [71,72].

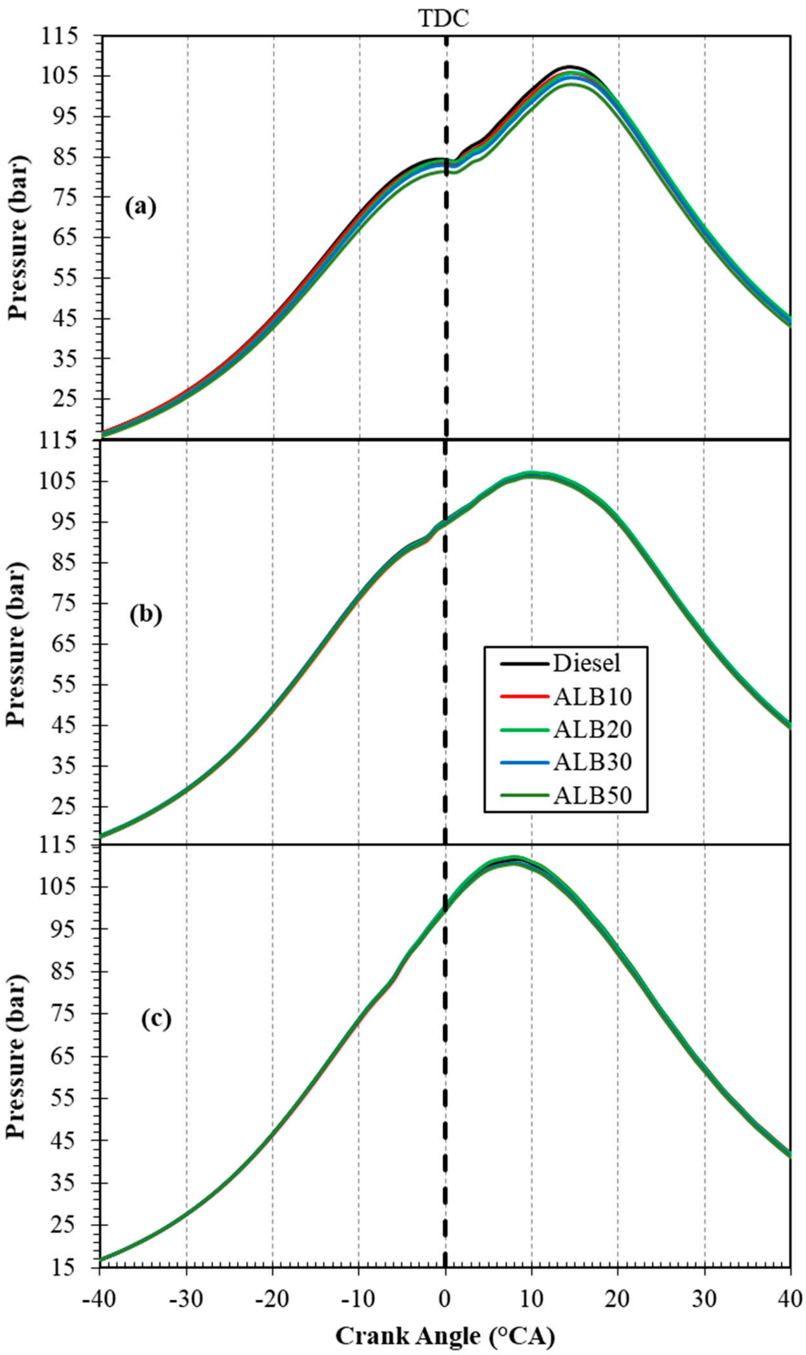

Figure 6. Cylinder pressure versus crank angle degree for various ALB and diesel fuels at. (**a**) 2000 rpm, (**b**) 3000 rpm and (**c**) 4000 rpm.

3.3.2. Heat Release Rate

The graph of the heat release rate (HRR) versus the crank angle for all the ALB-blended fuels and the neat diesel fuel is presented in Figure 7. Generally, the ALB-blended fuels indicated lower peak HRR during the premixed combustion stage compared to the baseline diesel. In fact, there was a slight decrement in the peak HRR with the increment of the ALB blend ratio. Based on the figure, the ALB50 indicated the peak HRR reduction with respect to the baseline diesel at all engine speeds. For instance, the percentage reduction marked 1.9%, 1.9% and 2.6% with the rise of the engine speed in the increasing order.

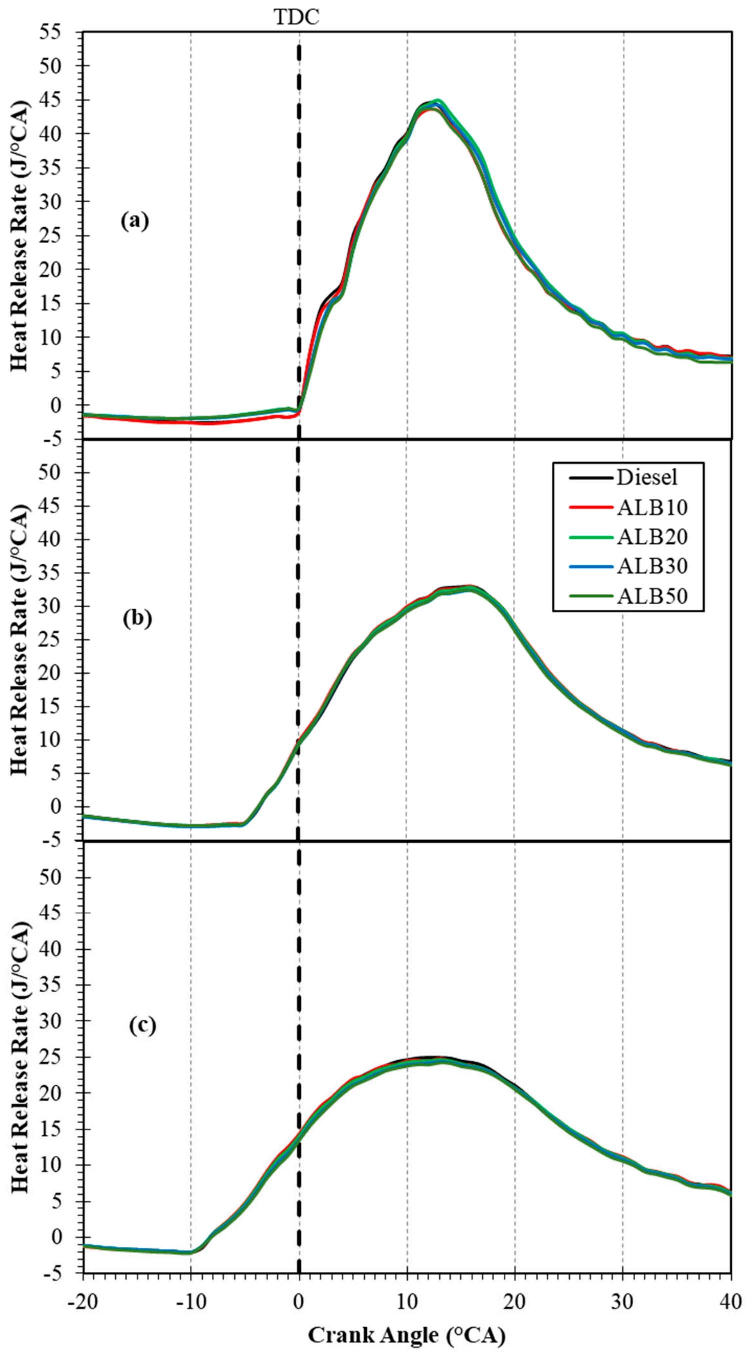

Figure 7. Heat release rate versus crank angle degree for various ALB and diesel fuels at (**a**) 2000 rpm, (**b**) 3000 rpm and (**c**) 4000 rpm.

This situation can be attributed to the physicochemical properties of the ALB-blended fuels, which have a lower calorific value as compared to the neat diesel fuel [73]. This result aligns with the observation reported by Pankaj S. Shelke et al. in a study on the combustion behavior for the cottonseed biodiesel. They found that the biodiesel resulted in lower heat release rate compared to the diesel fuel due to smaller heating value and BTE [74]. Besides, as previously explained, larger viscosity of the ALB blends may be another reason to reduced peak HRR, as it will slow down the vaporization process of the ALB and hence less premixed combustion occurred.

3.3.3. Fuel Combustion Duration

Notably, the combustion duration is also one of the key indicators of the peak HRR. Combustion duration is defined as the period between 10% and 90% mass burnt. Based on Figure 8, ALB blends marked shorter combustion durations as compared to the baseline diesel regardless the engine speed. For instance, the shortest combustion duration was attained by the ALB50, with percentage reduction of 4.84%, 2.76% and 2.69% at engine speeds of 2000 rpm, 3000 rpm and 4000 rpm, respectively, as compared to that of baseline diesel.

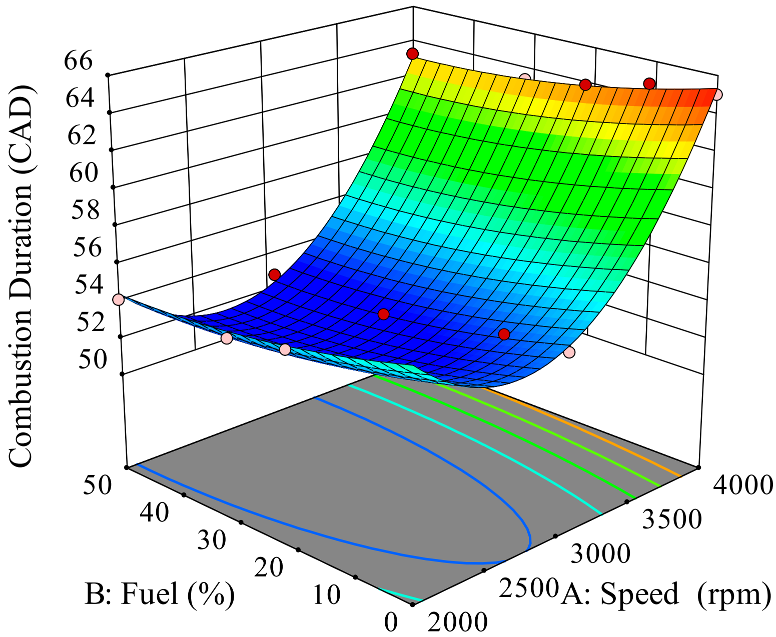

Figure 8. Interactive effect of fuel blends on combustion duration at varying speeds.

This result agrees with the result obtained in several studies conducted using *Alexandrian Laurel* biodiesel [75], Karanja biodiesel [76] and cotton seed oil biodiesel [77]. The researchers deduced shorter combustion time for the biodiesel blends as compared to the baseline diesel. In addition, An et al. [31] also showed similar results, where the shorter combustion duration was also noticed for the biodiesel. With the decrement of the combustion duration, this indicates that ALB burns with higher rate at the stage of diffusive combustion. This may be ascribed to the fuel properties of the ALB, where it has superior flammability due to its higher cetane number, richer oxygen content and ability to exhibit rapid chemical reactions, thus causing an earlier start of combustion and shortening the mixing duration necessary for the diffusive burning [78,79].

4. Conclusions

The feasibility of *Alexandrian Laurel* oil as a non-edible feedstock for biofuel production was confirmed to be viable. In this study, the biodiesel was produced by using two-step transesterification process, with the presences of acid and base catalysts. The physicochemical properties of the ALB blends were comparable to the neat diesel fuel. All properties met the international biodiesel standards based on ASTM and EN. Besides, increase in biodiesel ratio showed advantages in blended fuel properties such as kinematic viscosity, density, cetane number and flash point. Furthermore, the impacts of the ALB blends and neat diesel fuel on the engine performance, exhaust emissions and the combustion characteristics were determined at full load with an engine speed range from 1500 to 4000 rpm. From the results, the following main conclusions are summarized:

1. A prominent decline in engine brake power was found across all engine speeds due to the smaller calorific value of ALB. The highest reduction of 3.17 kW was obtained for ALB50 at the engine speed of 4000 rpm compared to baseline diesel. Besides, the use of ALB elevated the BSFC with respect to the conventional diesel fuel. Furthermore, the BTE of ALB50 is consistently higher than other fuels for all engine speeds.
2. In terms of engine-out emissions, the $BSNO_x$ emission increased for all the ALB-blended fuels as compared to that for the baseline diesel under all tested engine speeds. The largest increment in $BSNO_x$ recorded was approximately 1.56 g/kWhr for the ALB50 at 1500 rpm. On the other hand, the exhaust emissions also showed enhancement with reduced BSCO, $BSCO_2$ and smoke emissions by using ALB-blended fuels across all engine speeds.
3. On the combustion aspects, the ALB-blended fuels indicated the deteriorations in the peak combustion pressure and peak HRR during the premixed combustion stage, probably as a result of lower calorific value of the ALB blends as compared to the conventional diesel fuel. On the other hand, shorter combustion durations were also observed for all the ALB–diesel blends. With an increasing portion of ALB in blends, more rapid combustion occurred for the fuel.

Overall, ALB is considered a suitable and practicable biodiesel fuel. As shown with the results in this study, ALB–diesel blends can be employed satisfactorily in the modern high-pressure common-rail diesel engine without modifications. However, more efforts are needed in order to investigate material compatibility, fuel economy and engine tribology with ALB in diesel engines before it can be fully utilized.

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