

Article

Combustion and Emission Characteristics of a Diesel Engine Fueled with Crude Palm Oil Blends at Various Idling Speeds

Jun Cong Ge ^{1,*}, Sam Ki Yoon ^{2,*}  and Jun Hee Song ^{2,3,*} 

¹ Division of Mechanical Design Engineering, Jeonbuk National University, 567 Baekje-daero, Deokjin-gu, Jeonju-si 54896, Jeollabuk-do, Korea

² Department of Convergence Technology Engineering, Jeonbuk National University, 567 Baekje-daero, Deokjin-gu, Jeonju-si 54896, Jeollabuk-do, Korea

³ Department of Energy Storage/Conversion Engineering of Graduate School, Jeonbuk National University, 567 Baekje-daero, Deokjin-gu, Jeonju-si 54896, Jeollabuk-do, Korea

* Correspondence: jcge@jbnu.ac.kr (J.C.G.); sky596072@hanmail.net (S.K.Y.); sjhee@jbnu.ac.kr (J.H.S.)

Abstract: At present, the preparation cost of biodiesel is still higher than that of diesel. Thus, directly using vegetable oil before transesterification can greatly reduce the cost. This study investigated the application characteristics of the direct use of crude palm oil (CPO) in a common rail direct injection (CRDI) diesel engine. In this study, the combustion characteristics, engine performance, and emission characteristics of the CRDI diesel engine operated with CPO0 (neat diesel fuel), CPO10 (10% CPO blended with 90% diesel fuel by volume), CPO30 (30% CPO blended with 70% diesel fuel by volume), and CPO50 (50% CPO blended with 50% diesel fuel by volume) at three idling speeds (750 rpm, 1500 rpm, and 2250 rpm) were evaluated. The results obtained from the experiment elucidate that combustion starts earlier by increasing the idling speed. The addition of CPO to diesel fuel resulted in a decrease in the peak in-cylinder pressure at all idle speeds and a decrease in the maximum heat release rate (HRR_{max}) at 750 and 1500 rpm, but an increase in HRR_{max} at 2250 rpm. On the other hand, increasing idle speed is beneficial for reducing carbon monoxide (CO), hydrocarbons (HC), and nitrogen oxides (NO_x), but it increases smoke emissions. In addition, adding 10% and 30% CPO to diesel fuel can reduce both HC, NO_x, and smoke emissions simultaneously.

Keywords: crude palm oil; diesel engine; combustion characteristics; engine performance; idling emissions



Citation: Ge, J.C.; Yoon, S.K.; Song, J.H. Combustion and Emission Characteristics of a Diesel Engine Fueled with Crude Palm Oil Blends at Various Idling Speeds. *Appl. Sci.* **2022**, *12*, 6294. <https://doi.org/10.3390/app12136294>

Academic Editor:
Georgios Karavalakis

Received: 18 May 2022

Accepted: 18 June 2022

Published: 21 June 2022

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1. Introduction

As clean and renewable energy, biodiesel has been proved to be an important alternative fuel for diesel engines, and has been widely used worldwide. This is because biodiesel has the following advantages: (1) a generally higher oxygen content, as biodiesel fuels obtained by transesterification contain about 11–15% oxygen by weight from the carboxyl group in their structure [1]. The oxygen carried by biodiesel improves the problem of local oxygen deficiency caused by an uneven distribution of small portions of fuel during diesel combustion, which leads to complete combustion to reduce the emissions of carbon monoxide (CO), hydrocarbons (HC), and smoke. In addition, the high oxygen content in biodiesel is beneficial to increasing combustion efficiency and reaction temperature, thereby shortening ignition delay and total combustion duration [2,3]; (2) the cetane number of most biodiesel is between 45 and 65, which is higher than that of diesel (40–55). The cetane number is an important index for measuring the ignition performance of diesel engines. The higher the cetane number, the better the ignition performance, the shorter the ignition delay period, and the more uniform and stable combustion. In general, burning fuel with a high cetane number beneficially increases power output, reduces exhaust smoke and engine noise, and improves cold start characteristics. In contrast, low cetane fuel leads to difficult ignition and combustion, even misfire, a long ignition delay period, incomplete

combustion, large heat loss, and more harmful exhaust emissions [1,4]; (3) since biodiesel does not contain sulfur and aromatic compounds [5], adding biodiesel to diesel can greatly reduce the emissions of sulfur and aromatic compounds from diesel engines [6]; (4) superior flash point, better lubricity, and biodegradability [5,7,8]; although the flash point has no impact on combustion and emission characteristics, it can ensure the safety of storage and transportation. Moreover, the better lubrication properties can avoid excessive friction between moving parts, and the biodegradable properties provide numerous positive contributions to the ecological environment.

Currently, most of the research on biodiesel in a diesel engine is focused on adding different biodiesel with different mixing proportions to diesel and then studying the impact of different blended fuels on the combustion, engine performance, and emissions of a single-cylinder diesel engine under different engine load conditions. Uyumaz [9] investigated the effects of mustard oil biodiesel fuel blends on the combustion, performance, and emissions of a single-cylinder direct injection (DI) diesel engine at different engine loads. Test results showed that the mustard oil biodiesel fuel blends could be used effectively in the DI diesel engine without modifications, and the CO and smoke emissions are significantly reduced with mustard oil biodiesel-diesel fuel blends compared to standard diesel fuel. Moreover, there is no significant difference in in-cylinder pressure between mustard oil biodiesel blends and diesel fuel. Gonca et al. [10] evaluated the effects of engine load and soybean biodiesel percentage on the performance of a single-cylinder DI diesel engine through experiments and a theoretical model. They showed that the theoretical results agree with the experimental data. And at a constant biodiesel percentage and compression ratio, the effective power increases continuously, the effective efficiency increases to the specified value, and then begins to decrease with the engine load increase. Uyumaz et al. [11] studied the effects of poppy oil biodiesel-diesel fuel blends on the combustion, performance, and emissions of a single-cylinder, four-stroke, naturally aspirated, direct injection diesel engine with a compression ratio of 18:1 under different engine loads. It was reported that the poppy oil biodiesel fuel could be used in a single diesel engine without any modification. Gharehghani et al. [12] performed experimental studies to investigate the influence of waste fish oil biodiesel on combustion and emission characteristics of a single-cylinder variable compression ratio (VCR) Ricardo E6 diesel engine. They concluded that waste fish oil biodiesel leads to higher in-cylinder pressure, shorter heat release rate duration, and lower CO and HC emissions than common diesel fuel. In addition, more stable combustion without large cycle-to-cycle variations could be reached by using the waste fish oil biodiesel blends. Nalgundwar et al. [13] compared the effects of different types of blended fuels with diesel, palm, and jatropha biodiesel on the performance and emissions characteristics of a single cylinder DI diesel engine with varying loads. They observed lower biodiesel blends help reduce CO emissions, while higher biodiesel blends increase CO emissions due to a higher viscosity and density than diesel fuel.

Although most researchers have reported that a variety of biodiesel can be well used in diesel engines without any modification, most commercial biodiesel is produced by transesterification. In the transesterification process, a large amount of methanol and catalyst must be consumed and washed continuously. This not only increases costs, but also produces water pollution and so on. Gebremariam et al. [14] also indicated that the high production cost is the major barrier to extensively using biodiesel as an alternative fuel to diesel engines. Therefore, directly using biodiesel raw materials such as vegetable oils blended with diesel fuel in the diesel engine is the simplest and most effective way to reduce production costs. Prabu et al. [15] investigated the effects of palm oil/diesel blends with antioxidants on the performance, combustion, and emission characteristics of a single-cylinder diesel engine at a constant speed of 1500 rpm for different load conditions. They reported the feasibility of using the palm oil/diesel blends with additives as an alternative fuel in diesel engines. By reviewing a large amount of literature on the direct use of vegetable oil in diesel engines, Capuano et al. [16] showed that the direct use of waste vegetable oil (WVO) as fuel is very attractive for reducing fossil fuels utilization and

increasing sustainability. Moreover, the operation, when combined with high injection pressure, can improve the atomization of high viscosity fuel and make the mixture more completely combustible to reduce CO, HC, and soot emissions. On the other hand, with the continuous renewal of diesel engine fuel injection technology, high-pressure common rail system has been widely used in diesel engines. The use of high-pressure injection technology can overcome the negative effects (e.g., illiquid and atomizing difficulty) caused by the high viscosity of vegetable oil.

Based on how few studies on the direct use of vegetable oil on four-cylinder common-rail diesel engines exist, this study mainly investigates the effects of crude palm oil (CPO)-diesel blended fuels on combustion, engine performance, and emission characteristics in a common rail direct injection (CRDI) diesel engine. Furthermore, the influences of various idling speeds, including 750 rpm, 1500 rpm, and 2250 rpm, on combustion and emission characteristics are also investigated. Here, the idling operational condition is defined as a condition in which the engine runs at a low load and a low-rated speed [17,18]. This engine idling is necessary for most drivers. While resting in the cab, they keep the engine idling to maintain cab comfort and provide power to loads such as air conditioning, heating, and refrigerators [19]. It is reported that most long-haul trucks need to idle for 6 to 16 h a day [20]. When the engine is idling, the harmful exhaust emissions increase due to the low combustion temperature and pressure in the cylinder. Engine idling pollution has become one of the main factors of urban air pollution [17]. Therefore, this study of the effects of CPO blends and idling speeds on combustion and emissions of a CRDI diesel engine has considerable originality and novelty.

2. Materials and Methods

2.1. Test Fuels

In this study, crude palm oil (CPO) was obtained from GS Bio Co., Ltd. in South Korea. The CPO was heated to 40 °C and blended with diesel fuel at the volume ratio of 0:100, 10:90, 30:70, and 50:50 to obtain CPO0, CPO10, CPO30, and CPO50 blended fuels. The main properties of the tested fuels, including viscosity, density, calorific value, and oxygen content, are presented in Table 1. As shown in Table 1, the density and viscosity of CPO are higher than those of diesel, while the calorific value is lower than that of diesel. Therefore, more fuel would be required to be injected into the combustion chamber while using CPO with a lower calorific value to obtain the same power output as diesel fuel with a higher calorific value.

Table 1. Properties of diesel fuel and crude palm oil.

Properties (Units)	Diesel Fuel	Crude Palm Oil
Density (kg/m ³ at 15 °C)	836.8	903.8
Viscosity (mm ² /s at 40 °C)	2.719	42.21
Calorific value (MJ/kg)	43.96	39.34
Cetane index	55.8	49
Flash point (°C)	55	260
Oxygen content (%)	0	11.4

2.2. Engine Test

The experiments were conducted on a 4-cylinder, common rail direct injection (CRDI) turbocharged diesel engine with a displacement of 1991cc. Figure 1 shows the schematic of the test engine setup, and Table 2 gives the engine specifications. The CRDI diesel engine was coupled with an eddy current (EC) type water-cooled dynamometer (DYTEK230, Hwanwoong Mechatronics Co., Ltd., Gyeongnam, Korea) with a maximum load of 230 kW. The engine speed and engine load were controlled by the dynamometer through a controller. The experiments were conducted at a constant engine load of 30 Nm with three idling speeds of 750 rpm, 1500 rpm, and 2250 rpm. A precise electronic balance with 1 g precision (GP-100K, A&D Co., Ltd., Tokyo, Japan) was used to measure fuel consumption.

A piezoelectric pressure sensor (type 6056A, Kistler Korea Co., Ltd., Seongnam-si, Korea) was employed to measure the in-cylinder pressure. A charge amplifier (type 5011B, Kistler Korea Co., Ltd., Seongnam-si, Korea) was utilized to amplify the recorded in-cylinder pressure signal. The in-cylinder pressure value with a crank angle was recorded from 200 engine cycles to calculate the heat release rate (HRR) for analyzing combustion characteristics. All combustion data were acquired using a National Instruments PCI-6040E (National Instruments, Austin, TX, USA) data acquisition (DAQ) board and stored on a desktop. The CO, HC, and NO_x emissions were measured by an MK2 (GreenLine MK2, Eurotron (Korea) Ltd., Seoul, Korea) and HPC-501 (Nantong Huapeng Electronics Co., Ltd., Jiangsu, China) multi-gas analyzer. The smoke opacity was measured with an OPA-102 (QROTECH Co., Ltd., Bucheon-si, Korea) smoke meter. The specifications of the exhaust emission device and measurement systems are given in Table 3. Three engine speeds, 750 rpm, 1500 rpm, and 2250 rpm, were selected as the main engine operating variables to test the blended fuels. The engine load and the pilot and main injection timings were fixed at 30 Nm, 22° before top dead center (BTDC), and 7° BTDC, respectively. Detailed operating conditions are listed in Table 4. Before recording data, the engine was first warmed up with diesel under an idling speed of 750 rpm without an engine load for 30 min until the cooling water temperature reached 85 °C. The experiments were conducted firstly for neat diesel fuel (CPO0), and thereafter for the diesel-CPO blended fuels, CPO10, CPO30, and CPO50. After the testing of one kind of fuel, the fuel return valve was opened, all the fuel inside the fuel line, injection pump, and injector were discharged by using a fuel pump, and finally, the fuel supply and return lines were washed with the next test fuel 3 times. After the fuel change was completed, the engine was allowed to run for about 10 min to attain a steady-state condition for each new test fuel. After all tests were finished, the engine was switched back to diesel fuel until the blended fuel was removed from the fuel line, injection pump, and injector, and then the engine was stopped.

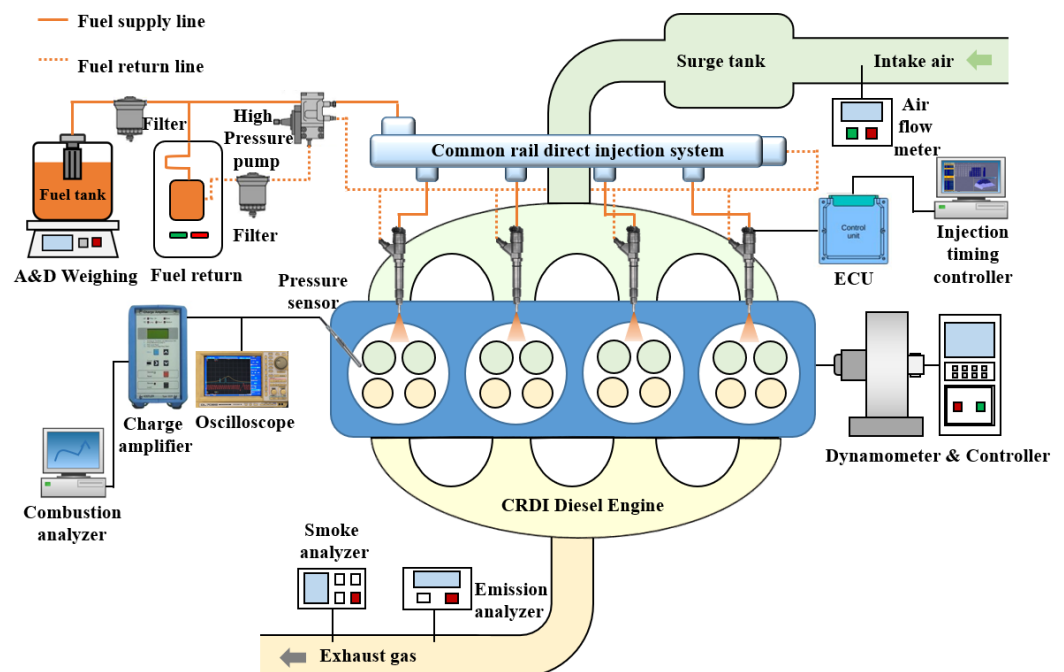


Figure 1. Schematic diagram of the engine test bed.

Table 2. Specifications of the test engine.

Engine Parameter	Units	Specifications
Type	-	turbocharged CRDI diesel engine
Number of cylinders	-	4
Bore × stroke	mm	83 × 92
Injector hole diameter	mm	0.17
Compression ratio	-	17.7:1
Max. power	kW/rpm	82/4000

Table 3. Specifications of exhaust emission device and measurement systems.

Parameter	Accuracy
CO (ppm)	±0.62%
NOx (ppm)	±0.25%
HC (ppm)	±5%
Smoke opacity (%)	±1%
Load monitoring (Nm)	±0.2%
Speed measuring (rpm)	±5
Fuel consumption (g)	±2
Fuel injection pressure (bar)	±1
Intake air temperature (°C)	±3
Cooling water temperature (°C)	±3

Table 4. Experimental and operating conditions.

Item	Conditions
Test fuels	CPO0, CPO10, CPO30, CPO50
Engine load	30 Nm
Engine speed	750, 1500, 2250 rpm
Fuel injection pressure	45 MPa
Pilot injection timing	22° BTDC
Main injection timing	7° BTDC
Intake air temperature	25 ± 3 °C
Cooling water temperature	85 ± 3 °C

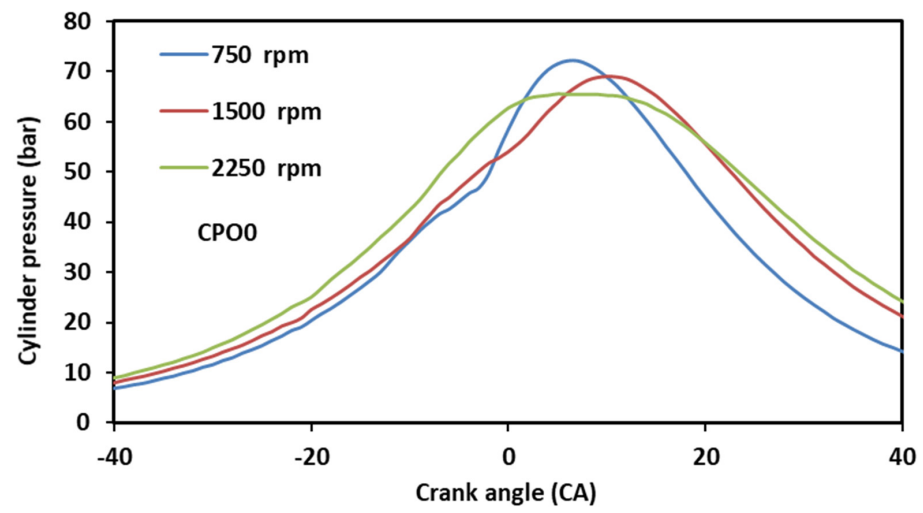
3. Results and Discussion

3.1. Combustion Characteristics

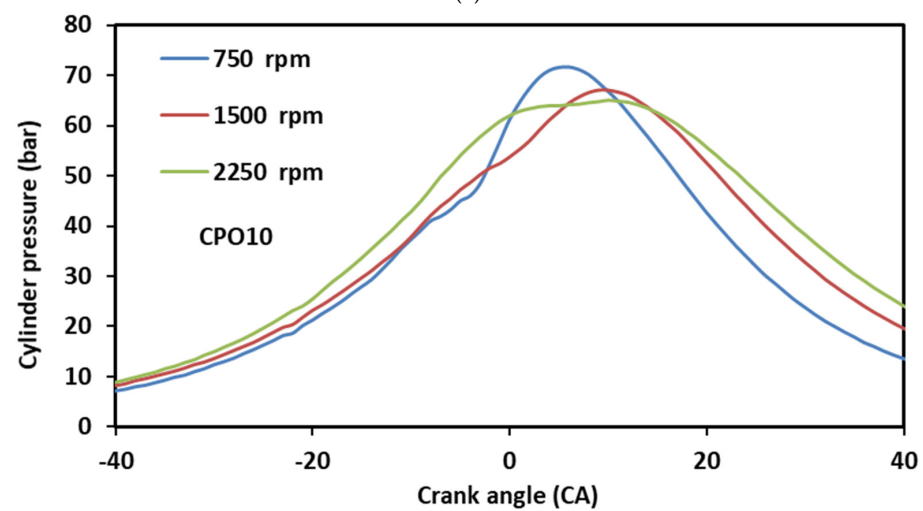
3.1.1. In-Cylinder Pressure

Figure 2 indicates the variation in the in-cylinder pressure of all test fuels with the crank angle according to different engine speeds. It can be clearly observed from Figure 2 that all crude palm oil (CPO)-diesel blends have no trace of pressure waves, and the in-cylinder pressure varies smoothly at the idling modes. The influence of the difference in engine speeds on the variation in in-cylinder pressure is much greater than that of the test fuels. For each test fuel, the maximum in-cylinder pressure is the highest at 750 rpm, followed by 1500 rpm and 2250 rpm. In addition, the start of combustion (SOC) gradually advanced with the increase of engine speed. The main reason may be that the increase in engine speed improves the combustion conditions of the mixture in the cylinder, such as increasing the temperature in the cylinder, especially the cylinder wall temperature. In addition, it may also be related to the increase of residual gas temperature and wall temperature, which leads to an increased charging temperature during injection and the shortening of ignition delay. Moreover, increasing the engine speed improves the air-fuel ratio and turbulence intensity [21]. The combustion process of all test fuels is almost the same, which includes two combustion phases: the premixed combustion phase followed by the diffusion combustion phase. Through the span of crankshaft angle, it can be seen that the total combustion duration gradually increases with the increase of engine speed,

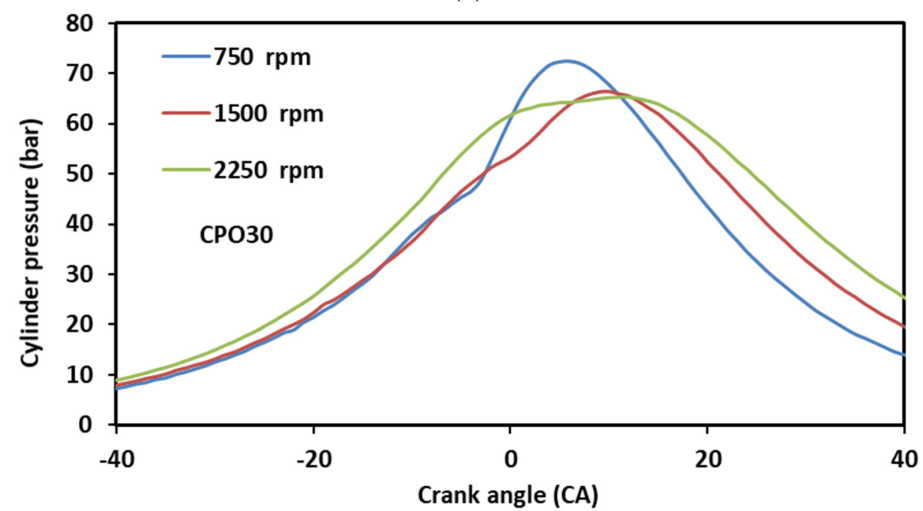
which is due to the increase of engine power output caused by the increase of engine speed, resulting in more fuel injected into the cylinder. Similar results can be found in other studies [22,23].



(a)



(b)



(c)

Figure 2. Cont.

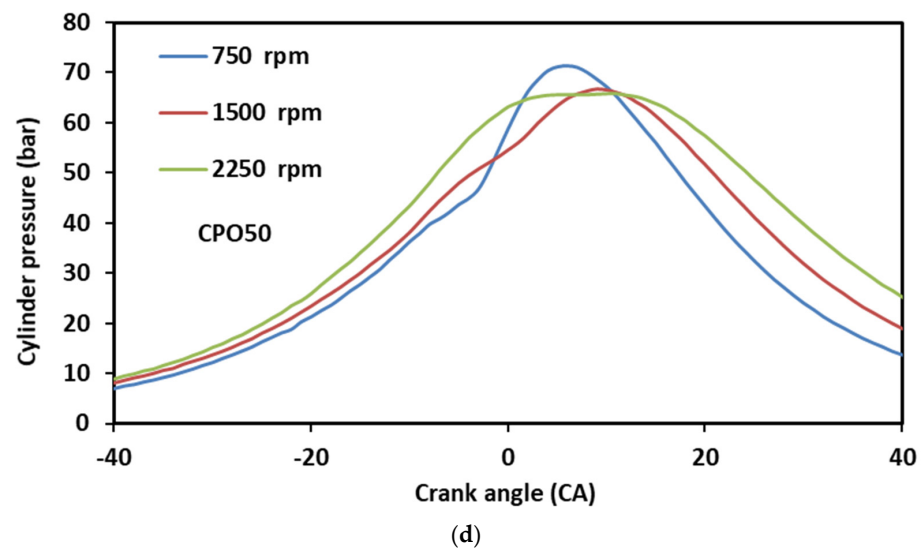


Figure 2. In-cylinder pressure of (a) CPO0, (b) CPO10, (c) CPO30 and (d) CPO50 according to various engine speeds.

Figure 3 shows the variation in maximum in-cylinder pressure for all test fuels according to various engine loads at an engine load of 30 Nm. It can be seen from Figure 3 that the maximum in-cylinder pressure gradually decreases as the engine speed increases from 750 rpm to 2250 rpm. Overall, the maximum in-cylinder pressure of all test fuels is reduced by 6.40% and 8.97% at 1500 rpm and 2250 rpm compared with 750 rpm, respectively. The maximum in-cylinder pressure generated by the engine at the lowest speed (750 rpm) is significantly higher than at the engine speed of 1500 rpm and 2250 rpm. The main reason may be that the combustion conditions in the cylinder at very low speeds, including temperature and pressure, are normally rather low, resulting in the need to consume more fuel to maintain the stable start of the engine. On the other hand, the maximum in-cylinder pressure of most crude palm oil (CPO)-diesel blends is lower than that of neat diesel fuel. At 1500 rpm, the maximum in-cylinder pressure of CPO10, CPO30, and CPO50 is reduced by 2.90%, 3.62%, and 3.33% compared with neat diesel fuel, respectively. Although palm oil has high oxygen content, which is beneficial to promote full combustion, its high viscosity leads to poor atomization effect, low cetane number further delays ignition delay, and finally leads to decreased in-cylinder pressure. Similar results can be found in other studies [24]. Dhar et al. [25] also pointed out that the fuel with a low cetane number is one of the main reasons for the decrease in in-cylinder pressure.

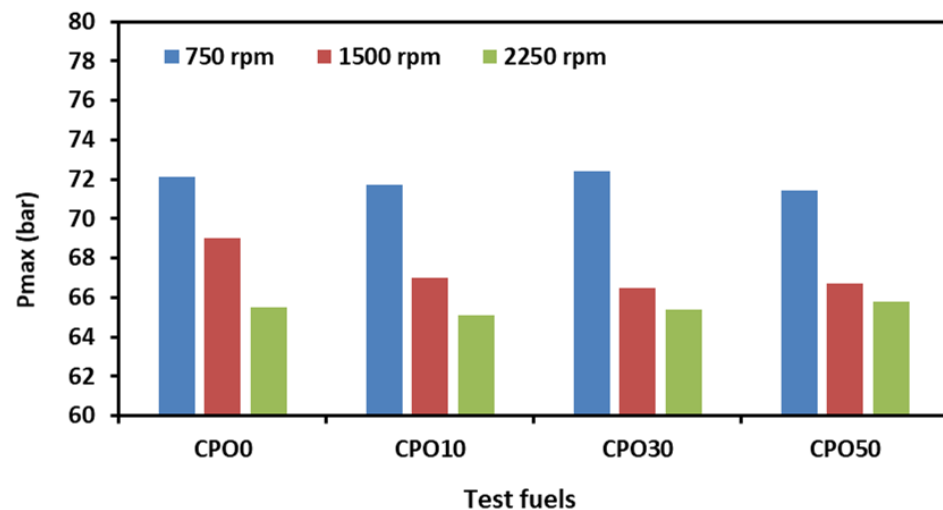


Figure 3. Maximum in-cylinder pressure of all test fuels according to various engine speeds.

3.1.2. Heat Release Rate

Figure 4 demonstrates the heat release rate (HRR) of all test fuels according to various engine speeds. The energy released by the fuel at each crankshaft angle can be obtained by analyzing HRR, which is one of the important combustion parameters [26]. It can be observed from Figure 4 that the maximum value of HRR (HRRmax) for each test fuel gradually decreases with the increase of engine speed, and the location of the HRRmax occurs away from the top dead center (TDC). As the engine speed increases from 750 rpm to 1500 rpm and then to 2250 rpm, for CPO0, the location of HRRmax occurs at 4 crank angle degree ($^{\circ}$ /CAD) after top dead center (ATDC), 10 $^{\circ}$ ATDC, and 14 $^{\circ}$ ATDC, respectively; for CPO10, the location of HRRmax occurs at 3 $^{\circ}$ ATDC, 9 $^{\circ}$ ATDC, and 14 $^{\circ}$ ATDC, respectively; for CPO30, the location of HRRmax occurs at 4 $^{\circ}$ ATDC, 10 $^{\circ}$ ATDC, and 15 $^{\circ}$ ATDC, respectively; for CPO50, the location of HRRmax occurs at 4 $^{\circ}$ ATDC, 9 $^{\circ}$ ATDC, and 16 $^{\circ}$ ATDC, respectively. In addition, for each test fuel, the amplitude of the crankshaft angle corresponding to HRR increases gradually with the increase of engine speed. Moreover, as seen in Figure 4, the start of combustion (SOC) timing for all test fuels is advanced near 8 $^{\circ}$ BTDC with increased engine speed. As mentioned above, for analyzing the change of in-cylinder pressure, the combustion environment of the mixture in the cylinder is improved, and the ignition delay is shortened with the increase of engine speed.

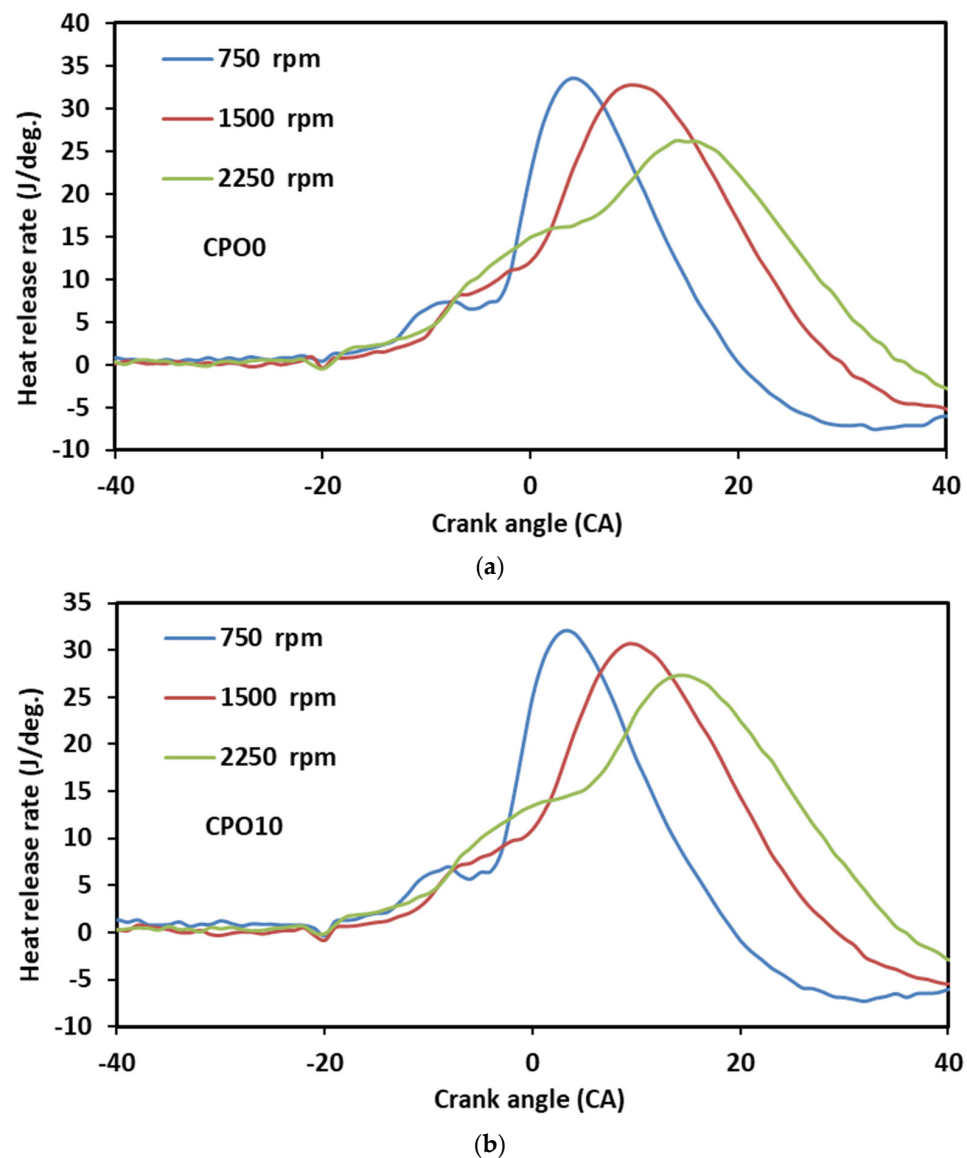


Figure 4. Cont.

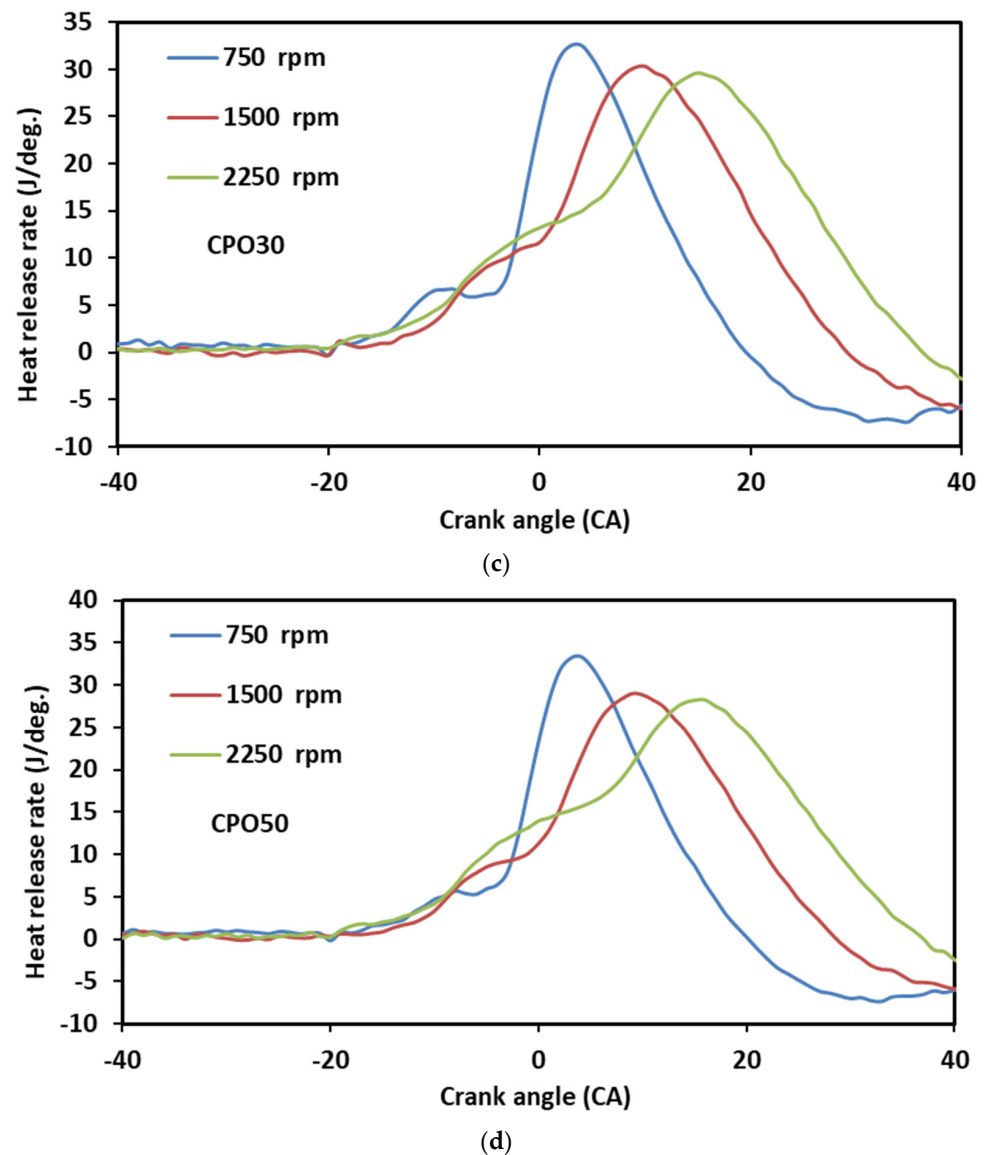


Figure 4. Heat release rate of (a) CPO0, (b) CPO10, (c) CPO30 and (d) CPO50 according to various engine speeds.

Figure 5 shows the HRRmax of all test fuels according to various engine speeds. As shown in Figure 5, it is obvious that HRRmax shows a decreasing trend with the increase of engine speed from 750 rpm to 2250 rpm. Compared with the HRRmax at 750 rpm, the HRRmax of CPO0 is reduced by 2.54% at 1500 rpm and 22.13% at 2250 rpm, respectively; the HRRmax of CPO10 is reduced by 4.36% at 1500 rpm and 14.87% at 2250 rpm, respectively; the HRRmax of CPO30 is reduced by 6.72% at 1500 rpm and 9.04% at 2250 rpm, respectively; the HRRmax of CPO50 is reduced by 12.99% at 1500 rpm and 15.36% at 2250 rpm, respectively. On the other hand, adding CPO to diesel fuel resulted in a decrease in HRRmax for all test fuels at 750 rpm and 1500 rpm; at 750 rpm, the HRRmax of CPO10, CPO30, and CPO50 are respectively reduced by 4.71%, 3.11%, and 0.78% compared with that of CPO; at 1500 rpm, the HRRmax of CPO10, CPO30 and CPO50 are respectively reduced by 6.49%, 7.26%, and 11.43% compared with that of CPO. However, at 2250 rpm, the HRRmax of CPO10, CPO30, and CPO50 increased by 4.18%, 13.18%, and 7.85%, respectively, compared with that of CPO. The reason may be that the high viscosity and density of CPO lead to poor fuel atomization and curb the release of heat at low idle speed. At high speed, the temperature and mixture in the cylinder are improved, and the negative impact of the high viscosity of CPO is weakened [27].

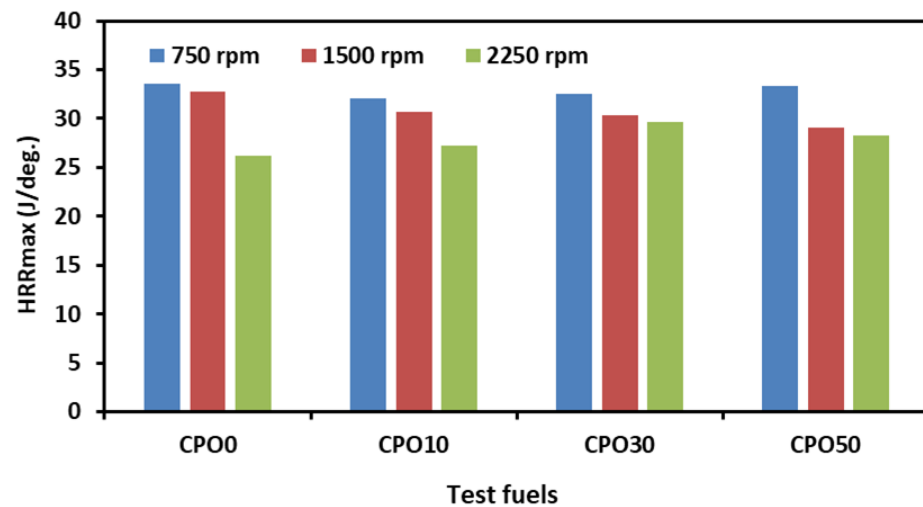


Figure 5. Maximum heat release rate of all test fuels according to various engine speeds.

3.2. Engine Performance

3.2.1. Brake Specific Fuel Consumption

Figure 6 displays the brake-specific fuel consumption (BSFC) variation of the test engine while running with CPO, CPO10, CPO30, and CPO50 according to various idling engine speeds. Overall, as the engine speed increases from 750 rpm to 1500 rpm and then to 2250 rpm, the BSFC of all test fuels first decreases and then increases slightly. And the BSFC of all fuels seems to be the highest at the lowest engine speed of 750 rpm. The average BSFC of all test fuels is reduced by 10.90% and 3.89% at 1500 rpm and 2250 rpm, respectively, compared with that at 750 rpm. This may be attributed to the increase in engine speed, as more air is introduced into the engine, and air turbulence improves. Thus, the decrease in BSFC values with increasing engine speed may be explained by the increase in engine excess effective power. Similar results are noted for different biodiesel blends with various engine speeds [28]. On the other hand, the BSFC of all blended fuels is higher than that of diesel at all engine speeds. At 750 rpm, the BSFC of CPO10, CPO30, and CPO50 increased by 11.40%, 20.96%, and 24.63% compared with diesel. At 1500 rpm, the BSFC of CPO10, CPO30, and CPO50 increased by 9.16%, 8.38%, and 14.23% compared with diesel. At 2250 rpm, the BSFC of CPO10, CPO30, and CPO50 increased by 4.73%, 10.52%, and 8.39% compared with diesel. The researchers suggest that the increase of BSFC is mainly related to the low calorific value and high viscosity of vegetable oil, which requires more fuel consumption to obtain the same engine performance as diesel fuel [29].

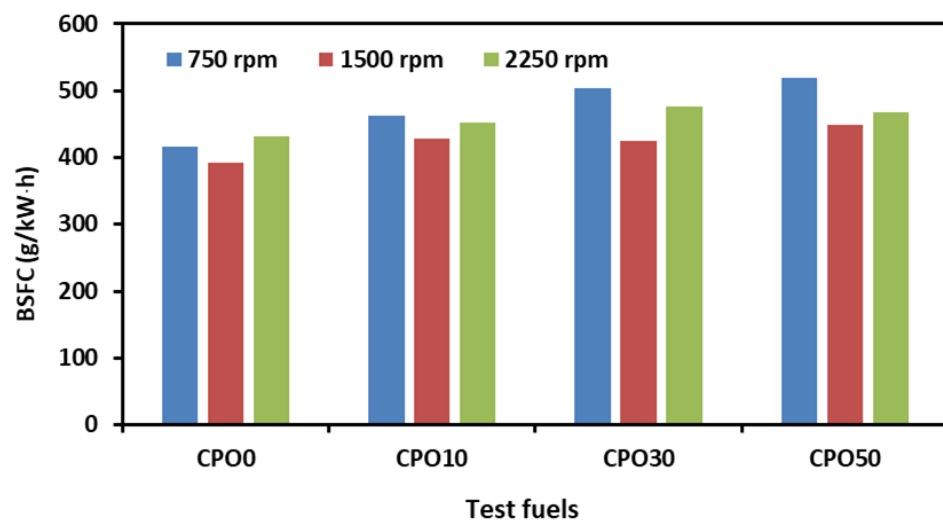


Figure 6. Brake-specific fuel consumption of all test fuels according to various engine speeds.

3.2.2. COVimep

Figure 7 demonstrates the coefficient of variation of the indicated mean effective pressure (COVimep) of all test fuels for 200 working cycles according to various idling engine speeds. COVimep is considered an important parameter for checking the combustion stability (cycle to cycle variation) of different fuels. The coefficient is defined as the ratio between the standard deviation of the indicated mean effective pressure (IMEP) over many consecutive cycles divided by its mean value [30]. During combustion of the internal combustion engine, due to the charge mixture composition (fuel and air), the change of pressure and temperature in-cylinder, and other factors, the change of COVimep is different. Generally, the higher the COVimep, the more unstable the engine is. Many researchers have pointed out that the COVimep of an internal combustion engine should not exceed 10% for stable combustion and operation [31,32]. As shown in Figure 7, the COVimep of most blended fuels is higher than that of diesel fuel. On the whole, the average COVimep of CPO10, CPO30, and CPO50, respectively, increased by 60.87%, 8.99%, and 13.38%, compared with that of diesel. This may be due to the higher viscosity and density and the low calorific value of palm oil, resulting in slightly unstable engine combustion compared with diesel. However, it is worth noting that the COVimep of all blended fuels is lower than 3%, indicating that this CRDI diesel engine can burn palm oil blends well and without large cyclic variation under these three idling conditions.

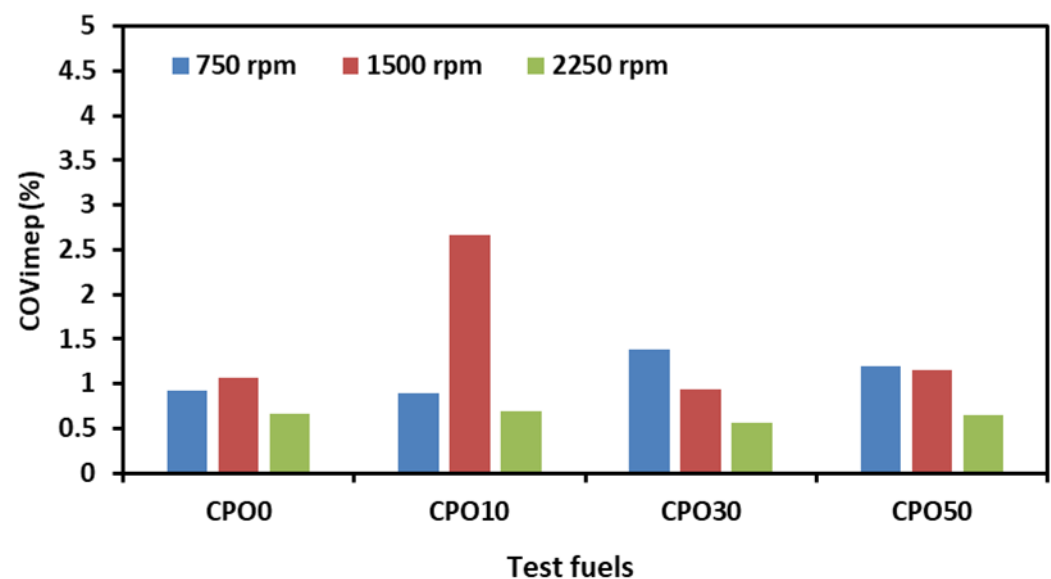


Figure 7. COVimep of all test fuels according to various engine speeds.

3.3. Emission Characteristics

3.3.1. CO Emissions

Figure 8 represents the CO emissions of the CRDI diesel engine fuels various palm oil blends at different idling speeds. As shown in Figure 8, it can be observed that the CO emission first increases and then decreases as the engine speed increases from 750 rpm to 1500 rpm and finally to 2250 rpm. Other researchers also reported that the CO emissions first increased (1000 rpm to 1500 rpm) and then decreased (1500 rpm to 2000 rpm) with the increase in engine speed [33]. The main reason why CO emission produces the least at the higher speed of 2250 rpm may be attributed to the optimal conditions such as temperature and pressure in the cylinder, especially the increase of temperature provides favorable conditions for the oxidation of CO to carbon dioxide (CO₂). On the other hand, the CO emission of all blended fuels is higher than that of neat diesel at each engine speed. At 750 rpm, the CO emissions of COP10, COP30, and COP50 are increased by 8.37%, 22.17%, and 46.79% compared with CPO0. At 1500 rpm, the CO emissions of COP10, COP30, and COP50 are increased by 7.04%, 31.69%, and 36.62% compared with CPO0. At 2250 rpm, the

CO emissions of COP10, COP30, and COP50 are increased by 8.59%, 12.27%, and 20.25% compared with CPO0. The formation of CO is mainly caused by the incomplete combustion of fuel due to insufficient oxygen. The high viscosity and density of palm oil mean that good atomization during fuel injection is not obtained to reduce the uniformity of fuel and air mixing and form more oil-rich areas. The main observations from CO emission results are as follows: (1) the higher the engine speed, the lower the CO emissions for all test fuels, due to a better mixing of air and fuels at the highest speed for higher turbulence; (2) the higher the CPO percentage in palm oil–diesel blends, the higher the CO emissions. This could be due to the high viscosity and density of palm oil, which worsens the atomization effect. Other researchers have also reported that the formation of CO emission in diesel engines strongly depends on the temperature and air equivalence ratio in the cylinder, and it increases with the increase of incomplete combustion [34].

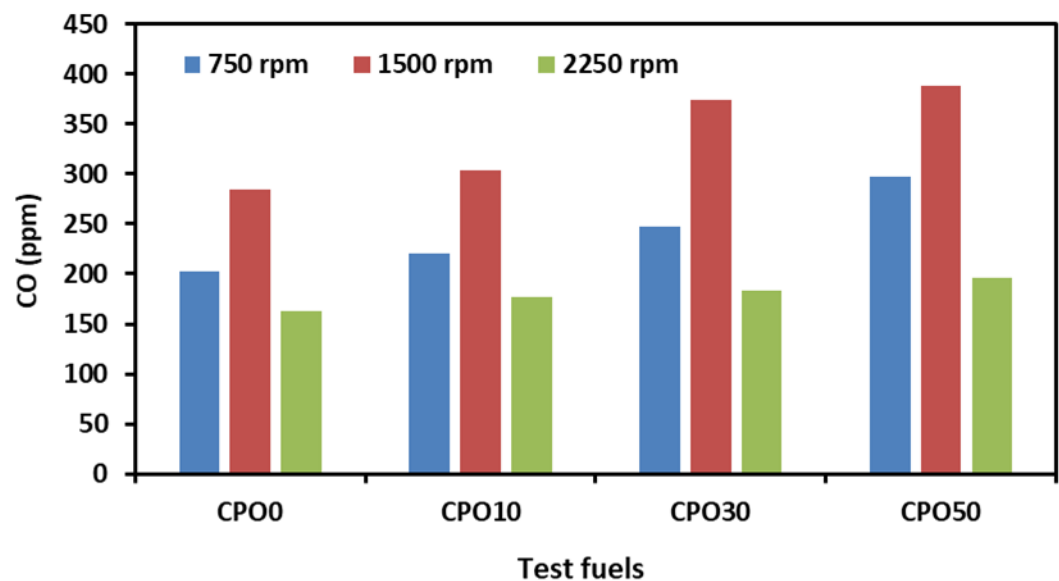


Figure 8. CO emissions of all test fuels according to various engine speeds.

3.3.2. HC Emissions

Figure 9 demonstrates the variations of HC emission for four test fuels according to various engine speeds. As shown in Figure 9, the HC emissions of all test fuels show a downward trend with increased engine speed. The reason for the decrease in HC emissions may be attributed to the increase in engine speed, which increases the temperature in the cylinder, especially the cylinder wall temperature, and reduces the misfire phenomenon of the mixture contacting the cold wall at idling conditions. In addition, the increase in engine speed ensures better mixing of air and fuel [34,35]. The running state of the engine and the fuel properties directly affect the HC emission. Generally speaking, HC emission is one of the main emission pollutants of the gasoline engine. However, under idle conditions, various conditions such as fuel injection pressure, temperature and pressure in the cylinder, turbulence intensity, and so on are not optimal, which will lead to a large amount of HC emission from diesel engines. Therefore, HC emission has become a serious problem for diesel engines under idling conditions [17]. On the other hand, adding palm oil to diesel reduces the HC emission of diesel engines, which may be related to the oxygen contained in palm oil, and promotes the oxidation of HC.

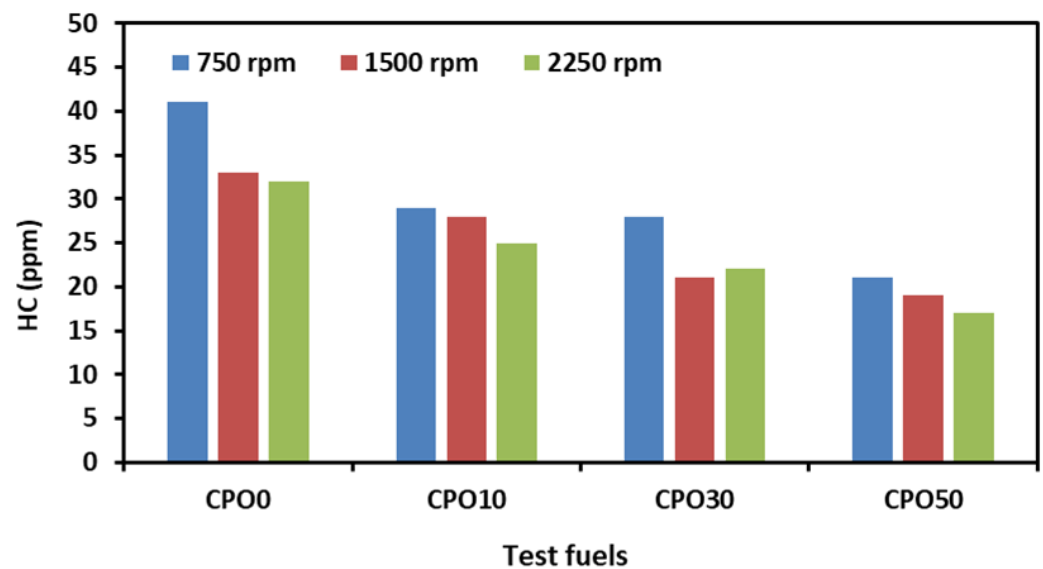


Figure 9. HC emissions of all test fuels according to various engine speeds.

3.3.3. NO_x Emissions

Figure 10 represents the NO_x emissions at different idling conditions for diesel, palm oil, and diesel fuel blends. As shown in Figure 10, it can be seen that the NO_x emissions have a significant decreasing trend for each test fuel with the increase in engine speed. At 1500 rpm, the NO_x emissions of CPO0, CPO10, CPO30, and CPO50 are respectively reduced by 64.28%, 64.26%, 65.24%, and 62.68% compared with at 750 rpm. At 2250 rpm, the NO_x emissions of CPO0, CPO10, CPO30, and CPO50 are respectively reduced by 77.21%, 78.69%, 79.83%, and 82.28% compared with at 750 rpm. The decrease of NO_x emissions with the increase of engine speed occurs because increasing engine speed improves the volumetric efficiency and gas flow motion within the combustion cylinders, ensures better mixing of air and fuel, and shortens the ignition delay, which allows less time for NO_x formation [19]. This is consistent with the research results from Tesfa et al. [23]. They also pointed out that the main mechanism of NO_x generation is related to the shortening of ignition delay, which leads to insufficient time for the chemical reaction of free oxygen and nitrogen in the cylinder. On the other hand, the NO_x emissions of all blended fuels are lower than that of neat diesel at each engine speed. At 750 rpm, the NO_x emissions of COP10, COP30, and COP50 are reduced by 2.48%, 7.64%, and 17.56% compared with CPO0. At 1500 rpm, the NO_x emissions of COP10, COP30, and COP50 are reduced by 2.44%, 10.13%, and 13.88% compared with CPO0. At 2250 rpm, the NO_x emissions of COP10, COP30, and COP50 are reduced by 8.82%, 18.24%, and 35.88% compared with CPO0. In general, because biodiesel is an oxygenated fuel with a high cetane number, the NO_x emission of most diesel engines fueled with biodiesel is higher than that of neat diesel fuel [36]. However, high viscosity fuel leads to improper atomization and poor mixing, as well as a decrease in temperature in the cylinder, which ultimately reduces the generation of NO_x. In this study, the high viscosity and low cetane number (see Table 1) of palm oil are the reasons for the reduction of NO_x.

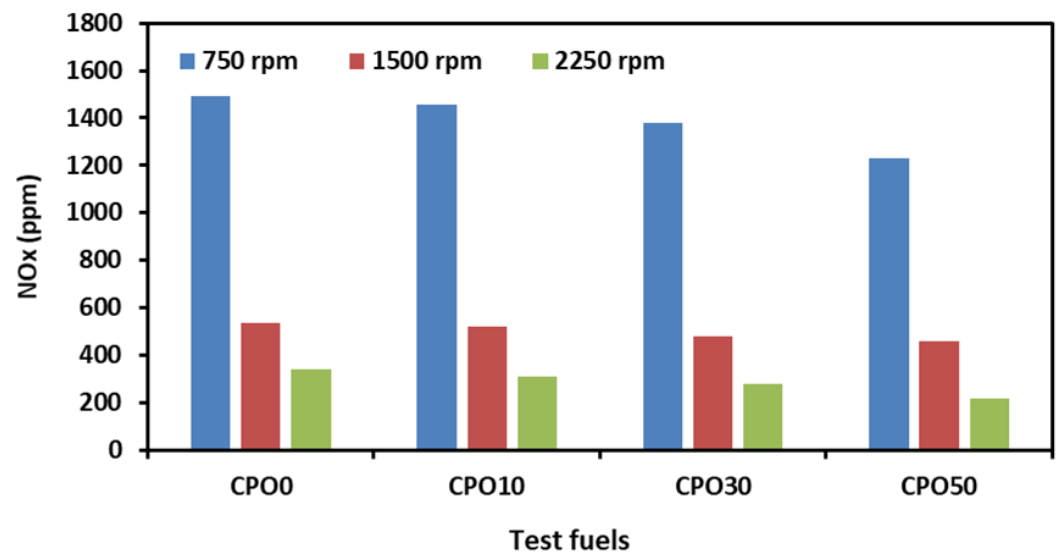


Figure 10. NOx emissions of all test fuels according to various engine speeds.

3.3.4. Smoke Emissions

Figure 11 depicts the variation of smoke emissions for all test fuels according to different idling speeds. As shown in Fig. 11, it can be seen that the smoke opacity of all test fuels increases with the increase of engine speed from 750 rpm to 2250 rpm. The smoke opacity of CPO0, CPO10, CPO30, and CPO50 is respectively increased by 23.08%, 29.17%, 25.00%, and 8.57% at 1500 rpm, compared with 750 rpm. The smoke opacity of CPO0, CPO10, CPO30, and CPO50 is respectively increased by 65.38%, 58.33%, 41.67%, and 14.29% at 2250 rpm, compared with 750 rpm. It can be attributed to the fact that the increase in engine speed causes more fuel to be injected into the combustion chamber, which reduces the air-fuel ratio and increases the fuel-rich area. In addition, under low load conditions, the temperature in the cylinder and the flow intensity of the mixture are weak, and the mixture is uneven, resulting in more smoke emissions generation. Ghadikolaei et al. [34] also found the increase in engine speed causes an increase in smoke emissions. They indicated that the increase in engine speed causes the decrease in volumetric efficiency and the increase in the fuel-air ratio, which leads to the decrease of oxygen availability for soot oxidation and the increase of carbon availability of soot formation, respectively. Moreover, the incomplete combustion due to insufficient time for fuel and air mixing causes an increase in both PM mass and number at high engine speeds. On the other hand, adding an appropriate proportion (e.g., 10 vol.%, 30 vol.%) of palm oil to diesel resulted in a lower smoke exhaust than diesel. At 750 rpm, the smoke opacity of CPO10 and CPO30 is reduced by 7.69% and 7.69% compared with CPO0. At 1500 rpm, the smoke opacity of CPO10 and CPO30 is respectively reduced by 3.13% and 6.25% compared with CPO0. At 2250 rpm, the smoke opacity of CPO10 and CPO30 is reduced by 11.63% and 20.93% compared with CPO0. However, when excess palm oil such as 50 vol.% is added to diesel fuel, the smoke opacity of CPO50 increases by 34.62% and 18.75%, respectively, at 750 rpm and 1500 rpm, compared with diesel. This is because palm oil is an oxygenated fuel. As shown in Table 1, the oxygen content of palm oil is as high as 11.4%. The oxygen in the fuel improves the air-fuel mixing rate, reduces the formation of fuel-rich areas, promotes complete combustion, and thus reduces smoke emissions [37]. On the other hand, the increase in smoke emissions caused by excessive palm oil added to diesel fuel is related to the high viscosity and density of palm oil, which leads to poorer volatility and reduces premixed combustion.

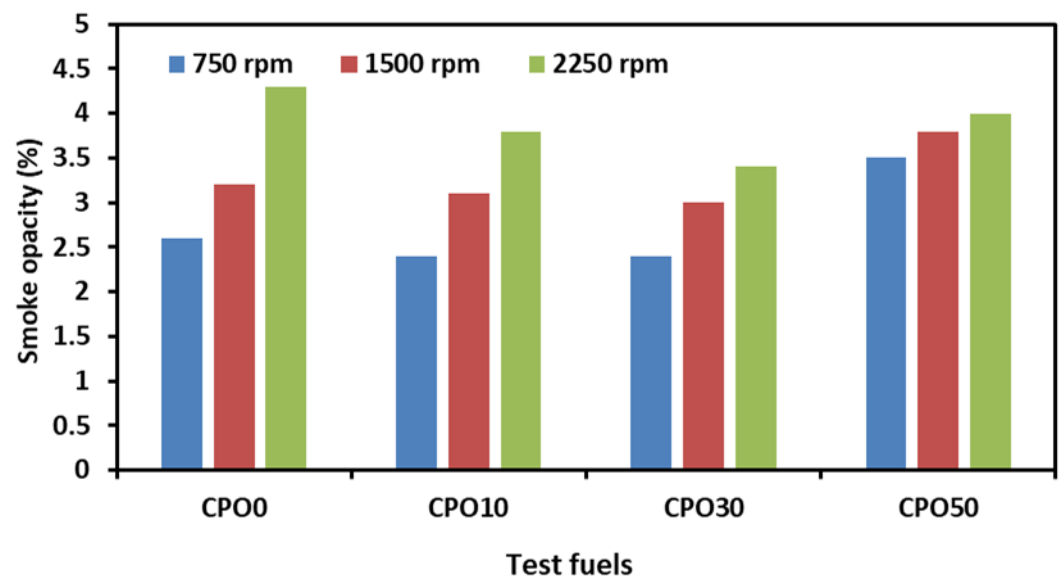


Figure 11. Smoke emissions of all test fuels according to various engine speeds.

4. Conclusions

In the present study, an experimental investigation has been carried out on the combustion characteristics, engine performance, and emission characteristics of a common rail direct injection (CRDI) diesel engine fueled with different crude palm oil (CPO)-diesel blends at various idling operations. Based on the experimental results, the main findings are summarized as follows:

1. For the combustion characteristics, it is found that the increase in engine speed causes a reduction in the peak in-cylinder pressure and the peak heat release rate. Adding CPO to diesel fuel resulted in a decrease in peak in-cylinder pressure and the peak heat release rate for all test fuels at 750 rpm and 1500 rpm.
2. For engine performance, the brake-specific fuel consumption (BSFC) of all tested fuels at 1500 rpm is the lowest, compared with that at 750 rpm and 2250 rpm. In addition, the BSFC of all blended fuels is higher than that of diesel at all idling speeds. The coefficient of variation of the indicated mean effective pressure (COVimep) of all tested fuels is the lowest at 2250 rpm, compared with that at 750 rpm and 1500 rpm. Moreover, the COVimep of all tested fuels at all idling speeds is much lower than 10%, which indicates that this diesel engine does not have the problem of unstable operation even if running with the CPO blended fuel with a mixing ratio of up to 50%.
3. For the emission characteristics, the carbon monoxide (CO), hydrocarbons (HC), and nitrogen oxides (NO_x) of all tested fuels at 2250 rpm is the lowest compared with that at 750 rpm and 1500 rpm. The addition of CPO to diesel fuel leads to an increase in CO emission, but it is beneficial to the reduction of HC and NO_x emissions. In addition, the smoke emission increases with the increase of engine speed. And adding an appropriate proportion of CPO, such as 10% and 30%, is beneficial to reducing the smoke emission.

Through the above experiments, we concluded that the CPO can be directly mixed with diesel and used in the CRDI diesel engine. In addition, it can be seen from the economic analysis that the direct use of palm oil reduces the production cost of biodiesel and has great economic value. In the future, we plan to investigate the application characteristics (e.g., engine performance, combustion and emission characteristics, particle morphology, and particle size distribution) of various vegetable oils running directly in diesel engines, according to various engine operating conditions such as engine speed and load, injection pressure, main injection, and pilot injection timing.

Author Contributions: Conceptualization, J.C.G.; methodology, J.C.G.; software, J.C.G.; validation, J.C.G., S.K.Y. and J.H.S.; formal analysis, J.C.G., S.K.Y. and J.H.S.; investigation, J.C.G., S.K.Y. and J.H.S.; resources, J.C.G., S.K.Y. and J.H.S.; data curation, J.C.G.; writing—original draft preparation, J.C.G.; writing—review and editing, J.C.G., S.K.Y. and J.H.S.; supervision, J.C.G.; project administration, J.C.G.; funding acquisition, J.C.G. All authors have read and agreed to the published version of the manuscript.

Funding: This research was supported by the Basic Science Research Program through the National Research Foundation of Korea (NRF), funded by the Ministry of Education (No. 2019R111A1A01057727).

Institutional Review Board Statement: Not applicable.

Informed Consent Statement: Not applicable.

Data Availability Statement: Not applicable.

Conflicts of Interest: The authors declare no conflict of interest.

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