

Article

Analysis of Cylinder Pressure and Heat Release Rate Variation in Diesel Engine Fueled with Croton Macrostachyus (CMS) Seed Oil Biodiesel as an Alternative Fuel

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Abstract: Despite its higher density, viscosity, and lower calorific value, biodiesel has been explored as an alternative energy source to diesel fuel. This study investigated biodiesels produced from croton macrostachyus (CMS) seed, a non-edible feedstock. The research aimed to experimentally analyze cylinder pressure, heat release rate, and ignition delay, as well as engine performance and emission characteristics, at a constant speed of 2700 rpm under varying loads (0–80%) using diesel, B10, B15, B20, and B25 blended fuels. Among the tested blends, B25 exhibited superior performance, achieving the highest peak cylinder pressure (CP) of 58.21 bar and a maximum heat release rate (HRR) of 543.9 J/CA at 80% engine load. Conversely, B20 at 60% engine load, followed by B25 and pure diesel at 80% engine load, demonstrated the shortest ignition delay (ID) and the most advanced start of combustion (SoC). Compared to the biodiesel blends, pure diesel showed: a 5.5–14% increase in brake thermal efficiency (BTE), a 17–26% decrease in brake-specific fuel consumption (BSFC), and a 7–12% reduction in exhaust gas temperature (EGT). Regarding emissions, carbon monoxide (CO) and hydrocarbon (HC) emissions were lower for pure diesel, while carbon dioxide (CO₂) and nitrogen oxide (NO_x) emissions were higher for biodiesel blends, attributed to their inherent oxygen content. In conclusion, CMS biodiesel displays promising characteristics, suggesting its potential suitability for use in internal combustion engines.

Keywords: croton macrostachyus; biodiesel; performance; emission; combustion; diesel–biodiesel blend



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1. Introduction

Energy is crucial to every aspect of human socioeconomic activity [1]. Diesel fuel is extensively used for power generation, mainly in the transportation industry for diesel engines [2]. Diesel engines are more applicable for heavy-duty vehicles because of their durability, higher torque output, and greater energy efficiency [3]. Therefore, diesel fuel consumption is increasing due to the ascension of the transportation industry, which is

responsible for environmental pollution and global warming [4]. Furthermore, the rise in diesel fuel consumption, which is produced from nonrenewable sources, posed an energy security issue due to the depletion of fossil fuel reserves [5,6]. Therefore, over the past ten years, there has been an increase in stricter environmental laws and the use of renewable and environmentally friendly energy sources [7]. Hence, researchers are looking for an alternate source of energy to fossil fuels. Biodiesels are renewable energy sources that can be utilized in similar diesel fuel applications because of the equivalent specification of diesel fuel [8]. Therefore, diesel engines can operate on a biodiesel blend instead of pure diesel fuel. Pure biodiesels alone are not compatible with the existing CI engine due to their higher viscosity and density.

Various edible and non-edible feedstocks are being used to produce biodiesel. However, due to the food competition, edible or first-generation feedstock, such as palm oil [9], sunflower [10], and soybeans [11,12], is not feasible for biodiesel production. Therefore, non-edible or second-generation feedstock, such as castor [13], jatropha [14], neem [15], juliflora [16], waste cooking [17,18], and others, are being explored by researchers as a source of biodiesel which cannot meet the global energy demand. There is also third-generation feedstock such as algal biomass [7,19] and fourth-generation feedstock such as genetically modified algae as a potential alternative to diesel [20,21]. Additionally, the drawbacks of using biodiesel include its reduced energy content, increased viscosity, and higher density when compared to conventional diesel. These characteristics negatively impact engine performance and lead to increased NO_x emissions [22]. The primary obstacles to using biodiesel in diesel engines include its inadequate cold flow characteristics and reduced resistance to oxidation. Biofuels' functional groups are also very critical in defining the combustion chemistry in an internal combustion engine which affects the performance and emission characteristics. According to a statement by Rotavera and Taatjes [23], continued integration of liquid biofuels for transportation energy purposes may not require complete displacement of petroleum-derived hydrocarbons and instead may utilize blending in small amounts.

Studies have been conducted to examine how biodiesel, derived from various feedstocks and mixed with diesel fuel, affects the performance characteristics of compression ignition (CI) engines. For instance, Owais et al. [24] compared the soapberry seed biodiesel diesel blend at different percentages with diesel fuel. The result reveals that the engine BTE of the diesel–biodiesel blend was less than that of the pure diesel. Furthermore, the resulting NO_x emission was also increased for the diesel–biodiesel blend compared to the diesel. The 30% soapberry seed biodiesel blend has shown HRR, CP, CO, HC, and smoke emissions, but elevated NO_x emission compared to pure diesel. Furthermore, Pracuch et al. [25] analyzed the physiochemical properties of waste frying oil biodiesel–diesel blend and found that raising the proportion of biodiesel increases the viscosity, cetane number (CN), and flash point, but reduces the calorific value, which is attributed to the oxygen molecule present in the vegetable oil. It has also been observed that the CN and flash point of the waste frying oil biodiesel have a negative linear correlation with the calorific value. In other research, Genet et al. [17] conducted the engine experimental response comparison of waste cooking oil biodiesel and diesel. The result reveals that the BTE and BSFC of the diesel–biodiesel blend were found to be 27.9% less and 4.8% higher than the diesel, respectively. Furthermore, the CO and HC emissions decreased while the increasing trend of CO₂ and NO_x emissions increased with the increasing biodiesel proportion and engine load.

Adib et al. [7] analyzed the engine experimental response of third-generation feedstock called microalgae and hydro-treated vegetable oil (renewable diesel) blended with diesel fuel. It was revealed that a 30% microalgae biodiesel blend has shown a relatively shorter

ID that causes a lower CP and HRR compared to pure diesel. It also showed a rise in BSFC, CO₂, and NO_x and a decline in BTE, smoke, and particulate matter emissions compared to diesel fuel. However, the addition of hydro-treated vegetable oil, known as renewable diesel, improved the engine performance and reduced the CO₂ and NO_x while increasing the particulate matter and smoke emissions. The same trend was also reported by Uyumaz [26] for the 10%, 20%, and 30% percentage of biodiesel produced from linseed oil. At higher engine load, the 30% linseed oil biodiesel has shown higher BSFC and reduced BTE with increased NO_x and reduced CO emission compared to diesel. The influence of boost pressure was also analyzed by Singh and Sandhu [27] for 20% argemone biodiesel blends. It is reported that lowering the boost pressure raises the ID that shifts the peak CP and HRR away from TDC. Furthermore, increasing the boost pressure lowers the ID and premixed combustion, thereby promoting the diffusion heat release phase. Rajak and Verma [28] compared biodiesel from different feedstock and found that biodiesels exhibited a lower ID and longer combustion durations (CD) with increasing engine load in contrast to diesel fuel. The effect of antioxidant additives in a microalgae biodiesel–diesel blend was also investigated by Saravanan et al. [29], who reported that biodiesel has reduced CO, HC, and smoke emissions comparatively.

This research utilizes Croton macrostachyus seed (CMS), a non-edible feedstock, for biodiesel production. CMS is a relatively new biodiesel source, with only one published study examining its engine performance and emission characteristics [30]. Existing literature primarily focuses on oil extraction and biodiesel production, highlighting the need for further investigation into its engine characteristics. The novelty of this study lies in its detailed analysis of the combustion characteristics of CMS biodiesel–diesel blends compared to pure diesel fuel. Specifically, ignition delay (ID) and start of combustion (SoC) were extracted from the heat release rate (HRR) plots and correlated with engine performance and emission outputs across varying engine loads for all tested fuels.

2. Experimental Methodologies

2.1. Preparation of Test Fuel Blends

This study uses Croton macrostachyus (CMS), biodiesel (B), and diesel fuel. The fuel samples were diesel (0 vol% B and 100 vol% diesel), B10 (10 vol% biodiesel and 90 vol% diesel), B15 (15 vol% biodiesel and 85 vol% diesel), B20 (20 vol% biodiesel and 80 vol% diesel), and B25 (25 vol% biodiesel and 75 vol% diesel). The blend was continuously stirred using a magnetic stirrer for 30 min to ensure the homogeneity and dispersion of the additive. Subsequently, enough time was allowed to see whether component separation started occurring or not. Only after confirmation of this were the blends employed for testing. As shown in Table 1, the CMS biodiesel properties were reported in [31], and the biodiesel blend's physiochemical properties are also calculated according to Kay's mixing rule [32–36] using the following equations:

$$\rho_b = \sum_{i=1}^n X_i \rho_i \quad (1)$$

$$V_b = \sum_{i=1}^n X_i V_i \quad (2)$$

$$CN_b = \sum_{i=1}^n X_i CN_i \quad (3)$$

$$LHV_b = \sum_{i=1}^n X_i LHV_i \quad (4)$$

where X_i is blending ratio, ρ_i , V_i , CN_i , and LHV_i are known parameters such as density, viscosity, cetane number, lower heating value of fuel, and ρ_b , V_b , CN_b , and LHV_b are their corresponding calculated parameters of fuel blends.

Table 1. Croton macrostachyus (CMS) biodiesel and its blend properties.

Properties	Diesel (B0)	B100	B10	B15	B20	B25
Lower heating value (MJ/kg)	43	40.04	42.70	42.55	42.408	42.26
Density at 15 °C (kg/cm ³)	830	846.3	831.63	832.44	833.26	834.07
Kinematic viscosity at 40 °C (mm ² /s)	2.73	4.43	2.9	2.98	3.07	3.15
Cetane number	48	50.89	48.289	48.43	48.57	48.72
Flashpoint °C	58	175	69.7	75.55	81.4	87.25
Pour point °C	−33	−15	−31.2	−30.3	−29.4	−28.5

2.2. Heat Release Rate (HRR) Calculation

The HRR referred here is the apparent heat release rate, i.e. the one without considering the heat loss. It was calculated based on the first law of thermodynamics without accounting for heat transfer to the cylinder wall or other losses using Equation (5). This is commonly derived from cylinder pressure data and volume changes during combustion. The instantaneous volume of the cylinder using Equation (6) and the rate of change in the cylinder volume with the crank angle using Equation (7) were also calculated to compute the heat release rate. The HRR was calculated from the cylinder pressure data using Equations (5)–(7), simultaneously employing finite difference techniques to estimate $dp/d\theta$ as $(P_n - P_{n-1})/(\theta_n - \theta_{n-1})$ and $dV/d\theta$ using Equations (6) and (7) at a particular θ . This calculation has been carried out by digitizing $P - \theta$ data at a constant speed of 2700 rpm under 0%, 20%, 40%, 60%, and 80% engine load for all test fuels.

$$\frac{dQ}{d\theta} = \left(\frac{\gamma}{\gamma - 1} * p * \frac{dv}{d\theta} \right) + \frac{1}{\gamma - 1} * V * \frac{dp}{d\theta} \quad (5)$$

$$V = V_c \left[1 + \frac{Rc - 1}{2} \left(R + 1 - \cos \theta - \sqrt{R^2 + \sin^2 \theta} \right) \right] \quad (6)$$

$$\frac{dV}{d\theta} = V_c \left[\frac{Rc - 1}{2} (\sin \theta) \left(\frac{1 - \cos \theta}{\sqrt{R^2 + \sin^2 \theta}} \right) \right] \quad (7)$$

where V is for instantaneous engine cylinder volume, V_c is the clearance volume, $dQ/d\theta$ is the heat release rate, $dp/d\theta$ is the pressure rise rate, $dV/d\theta$ is the change in volume with a change in crank angle, R is the ratio of the crank radius to the connecting rod length, which was adopted as 0.25 (as taken from cylinder pressure crank angle recording), γ is the specific ratio adopted as 1.35, and Rc is the compression ratio.

2.3. Calculation of the BTE and BSFC

The brake thermal efficiency (BTE) was calculated in Equation (8), based on the brake power (P_B) determined from the engine dynamometer setup using the direct measurement of applied torque and engine speed, lower heating value of the fuel (LHV), and mass flow rate ($\dot{m}_f$) of the fuel directly measured using a fuel flow meter, whereas brake-specific fuel consumption (BSFC) was calculated using brake power (P_B) and mass flow rate ($\dot{m}_f$) of the fuel.

$$BTE = \frac{P_B \cdot 3600}{\dot{m}_f \cdot LHV} \cdot 100 \quad (8)$$

$$BSFC = \frac{\dot{m}_f \cdot 1000}{P_B} \quad (9)$$

where the magnitudes of each parameter were expressed in such a way with BTE in %, BSFC in g/kWh, P_B in kW, $\dot{m}_f$ in kg/h, and LHV in kJ/kg.

2.4. Experimental Setup

As shown in Figure 1, the GUNT model CT110 of the engine dynamometer test setup was used to investigate the engine experimental response using a CT110.23 water-cooled and naturally aspirated diesel engine, which is equipped with the asynchronous motor for engine starting purposes. The dynamometer setup contains different sensors and actuators for measurement and control. The engine was allowed to work with pure diesel for 10 min before starting the experiment. The experiments were performed at a constant engine speed of 2700 RPM and variable engine load ranging from 0 to 80%. The engine was run on pure diesel for 10 min before initiating the experiment. Combustion parameters were determined through the analysis of pressure–crank angle diagrams, which were recorded using a data acquisition system. These diagrams were generated from pressure measurements via a flush-mounted piezoelectric pressure transducer in the cylinder head. Furthermore, HC and CO₂ emissions were measured by Infralyt smart (SAXON Junkalor GmbH., Dessau, Germany), whereas CO and NO_x were measured by a Seitron exhaust gas analyzer (Seitron Americas Inc., Trevose, PA, USA). The engine specification is shown in Table 2.

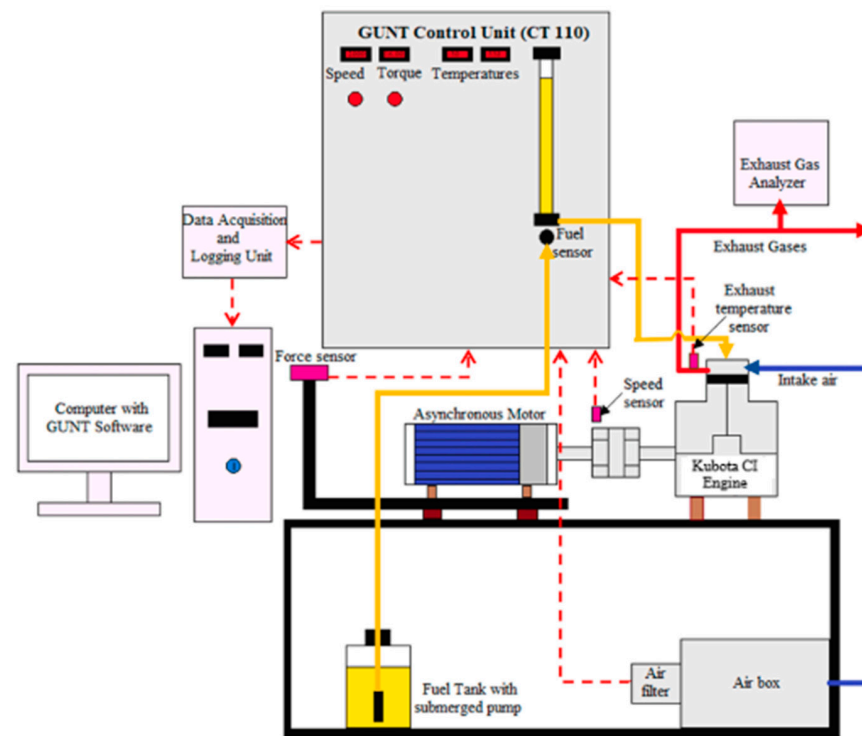


Figure 1. Schematic arrangement of the engine experimental setup.

Table 2. Test engine specifications.

Item	Specification
Company	Kubota (Osaka, Japan)
Type	EA300-E2-NB1
Engine type	Water-cooled, 4-stroke, single-cylinder compression ignition DI engine
Fuel	Diesel
Crank radius to connecting rod length ratio	0.25
Displacement	309 cm ³
Bore	75 mm
Stroke	70 mm
Output power	5.1 kW at 3000 rpm
Oil capacity	1.3 L
Noise level	95 dB(A)
Compression ratio	23:1
Injection timing	23 CA before TDC

2.5. Uncertainty Analysis

The measurement uncertainty of all parameters was calculated using Kline and McClintock's propagation of uncertainty approach [7]. All experiments were repeated five times. The uncertainties of all independent variables of n readings were calculated. The arithmetic mean X_m Equation (8), standard deviation σ Equation (9) and uncertainty U_i Equation (10), and the percentage of uncertainty U_i (%) of each independent parameter (CO, CO₂, HC, NO_x, load, speed, and torque) Equation (11) were determined. Furthermore, the uncertainty of each dependent parameter (BTE, BSFC, and HRR) U_R was also determined using Equation (12), where R stands for the dependent function of the independent variables. The overall percentage of uncertainty was found to be a maximum of 3.92%, and the respective uncertainties of each parameter are shown in Table 3.

$$X_m = \frac{\sum_{i=1}^n X_i}{n} \quad (10)$$

$$\sigma = \sqrt{\frac{\sum_{i=1}^n (X_i - X_m)^2}{n - 1}} \quad (11)$$

$$U_i = \frac{\sigma}{\sqrt{n}} \quad (12)$$

$$U_i(\%) = \frac{U_i}{X_m} * 100 \quad (13)$$

$$U_R = \left[\left(\frac{\partial R}{\partial x_i} U_i + \dots + \frac{\partial R}{\partial x_n} U_n \right)^2 \right]^{1/2} \quad (14)$$

Table 3. Type of instrument, range, and percentage of uncertainty.

Measured Parameters	Instrument	Measuring Range	Uncertainty
HC	Infralyt smart analyzer (SAXON Junkalor GmbH, Dessau, Germany)	0–2500 ppm	±1.67%
CO ₂	Infralyt smart analyzer (SAXON Junkalor GmbH, Dessau, Germany)	0–20.00% vol	±0.08%
CO	Seitron S500 analyzer (Seitron Americas Inc., Trevose, PA, USA)	0–4000 ppm	±0.58%
NO _x	Seitron S500 analyzer (Seitron Americas Inc., Trevose, PA, USA)	0–2000 ppm	±1.21%
Speed	CT 110	0–3000 rpm	±0.11%
Brake torque	CT 110	0–14 Nm	±1.39%
$\dot{m}_f$	CT 110	0–150 cm ³	±1.63%
CP	Pressure transducer CT 100.13	0–250 bar	±0.79%
EGT	Thermocouples	0–1300 K	±2.05%
	Computed parameters		
Brake power (BP)	Using engine torque and speed	-	±0.15%
BSFC	Using brake power and fuel flow rate	-	±0.79%
BTE	Using brake power and lower heating value	-	±0.78%
HRR	Using CP, V, and CA	-	±0.52%

3. Results and Discussion

This study investigated croton macrostachyus (CMS) biodiesel in a compression ignition (CI) engine at a constant speed of 2700 RPM and variable engine loads ranging from 0% to 80%, incremented in 20% steps. Performance parameters, including brake thermal efficiency (BTE), brake-specific fuel consumption (BSFC), exhaust gas temperature (EGT), and fuel mass flow rate ($\dot{m}_f$), were compared between the biodiesel blends and pure diesel. Emission characteristics, specifically carbon dioxide (CO₂), carbon monoxide

(CO), nitrogen oxides (NO_x), and hydrocarbons (HC), were also evaluated relative to pure diesel. Furthermore, combustion attributes such as cylinder pressure (CP), heat release rate (HRR), ignition delay (ID), and start of combustion (SoC) were examined in comparison to conventional diesel fuel. To ensure data reliability, each test was replicated five times.

3.1. Combustion Characteristics

3.1.1. Cylinder Pressure (CP)

Figure 2a–e depicts the cylinder pressure (CP) profiles as a function of crank angle, acquired at a constant engine speed of 2700 rpm and under varying load conditions of 0, 20, 40, 60, and 80%. Across all load scenarios, pure diesel consistently exhibited lower peak CP values compared to the biodiesel blends. A direct correlation was observed between engine load and maximum CP, attributed to the increased fuel injection required to meet higher load demands. The exothermic combustion process, resulting from fuel oxidation, directly contributes to the observed CP increase. At lower engine loads (0–40%), the B10 blend demonstrated the highest peak cylinder pressure, followed by B15, B25, B20, and pure diesel in descending order. Conversely, at higher engine loads (above 40%), B15 yielded the highest peak pressure, followed by diesel, B10, B20, and B25. This variation can be attributed to the relatively higher viscosity of biodiesel compared to diesel, as documented in [8]. Specifically, at lower load conditions, characterized by reduced fuel injection volumes and lower cylinder temperatures, the enhanced combustibility of B10 results in a higher pressure rise. However, as biodiesel blend percentages increase, a subsequent decrease in CP is observed, primarily attributed to the corresponding reduction in calorific value, as reported in [37]. Furthermore, at higher engine loads, the increased fuel injection rates significantly impact the ignition characteristics of biodiesel blends. A gradual decrease in peak pressure is observed with increasing biodiesel blend percentages up to B25. At 80% load, B10 and diesel exhibit comparable peak pressures to B15, while B25 shows the lowest peak pressure, followed by B20.

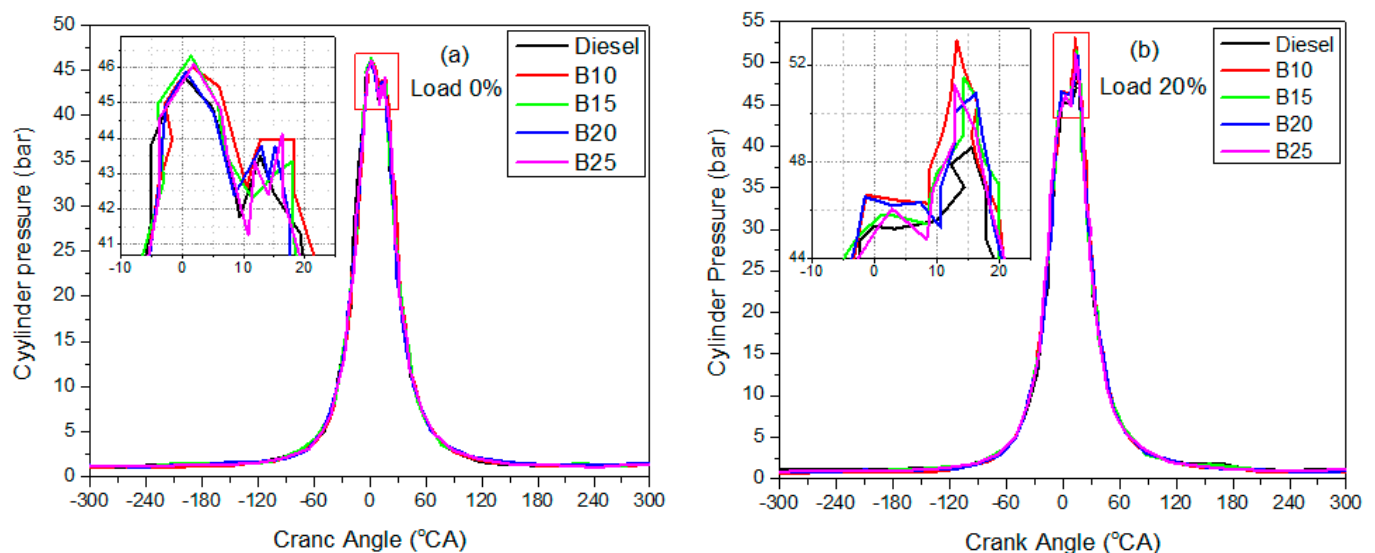


Figure 2. Cont.

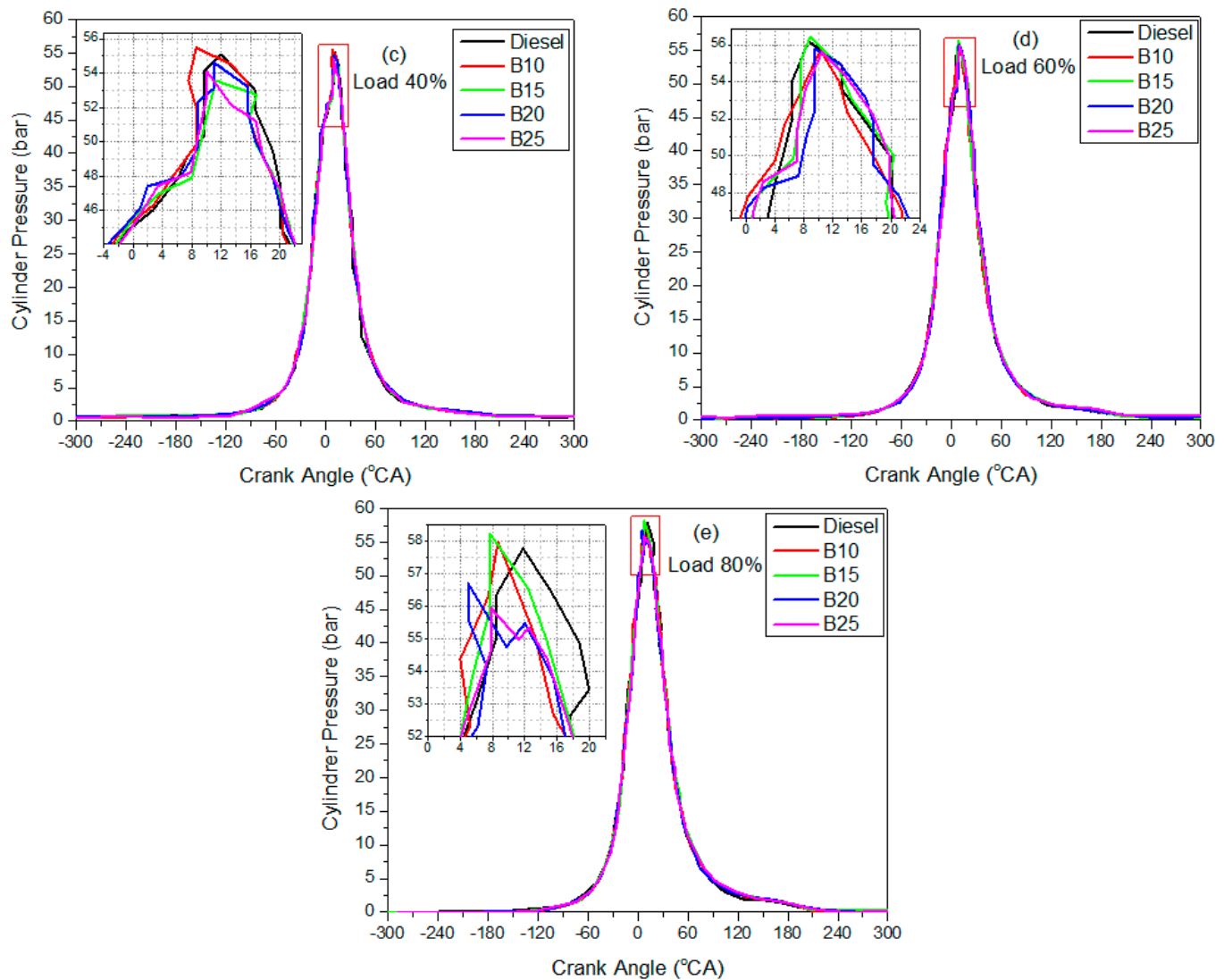


Figure 2. The CP of the test fuel based on the load variation: (a) no load, (b) 20% load, (c) 40% load, (d) 60% load, and (e) 80% load.

As depicted in Figure 3a, peak cylinder pressure (CP) increases with engine load. At lower load conditions, all fuel blends exhibit comparable pressures, with diesel showing the lowest values. At 20% load, diesel again demonstrates the lowest CP, while B10 records the highest, followed by B15, B25, and B20. However, at 40% load, diesel achieves the second highest CP after B10, followed by B20, B25, and B15. Furthermore, at 60% and 80% loads, B15 registers the maximum CP, whereas B20 and B25 yield lower values. These results suggest that peak CP generally increases with both biodiesel percentage and engine load. Biodiesel's lower energy density, higher viscosity, and inherent oxygen content may contribute to the observed CP increase, as supported by similar findings in [38]. Higher oxygen concentration enhances the oxidation reaction, leading to increased cylinder pressure (CP) and an earlier heat release rate (HRR). Density and viscosity are crucial parameters influencing fuel atomization, which, in turn, impacts injection characteristics. Higher density and viscosity result in larger droplet sizes, potentially increasing fuel consumption. The crank angle (CA) at which maximum CP is achieved was also recorded, as illustrated in Figure 3b. Overall, at 20% engine load, peak pressure occurred at a greater CA after top dead center (TDC). As the load increased, the CA of peak CP approached TDC. At no load, the maximum pressure was attained very close to TDC, specifically

2.06 °CA after TDC for B25. However, at 20% load, the maximum pressure was observed at 16.14 °CA after TDC for B20. Among all fuel blends, diesel exhibited peak pressure at a higher CA after TDC, particularly at higher engine loads. Furthermore, the location of peak pressure shifted away from TDC with increasing engine load for all tested fuels. This shift is attributed to the richer fuel–air mixture, which increases residence time and results in a retarded peak CP [39]. Furthermore, at zero load, the peak cylinder pressure (CP) for pure diesel occurred at 0.51 °CA after top dead center (TDC). For the biodiesel blends, the peak CP shifted from 1.66 °CA for B10 to 2.06 °CA after TDC for B25. Concurrently, the peak CP increased with increasing biodiesel percentage, from 45.6 bar for diesel to 46.2 bar for B15, which is attributed to the oxygen content in biodiesel enhancing combustion. However, as engine load increased, the magnitude of the peak CP continuously rose, and CA at which peak CP was achieved shifted to a maximum of 15 °CA after TDC at a 20% load. With further increases in engine load, the peak CP approached 9.03 °CA after TDC. Across all blends and loading conditions, the peak CP of biodiesel blends was achieved closer to TDC than that of pure diesel, with the exception of B20 at 16.14 °CA after TDC. This earlier occurrence of peak CP in biodiesel blends is attributed to faster combustion resulting from their inherent oxygen content, higher cetane number, and higher bulk modulus [40]. At 60% engine load operation, diesel, B10, B15, B20, and B25 achieved the peak CP 56.07 bar at 9.03 °CA, 55.64 bar at 10.23 °CA, 56.32 bar at 9.06 °CA, 55.79 bar at 9.86 °CA, and 55.15 bar at 10.66 °CA, respectively. Furthermore, at 80% engine load operation, diesel, B10, B15, B20, and B25 achieved the peak CP of 57.72 bar at 11.92 °CA, 57.9 bar at 8.77 °CA, 58.21 bar at 7.85 °CA, 56.64 bar at 5.01 °CA, and 55.99 bar at 7.84 °CA, respectively.

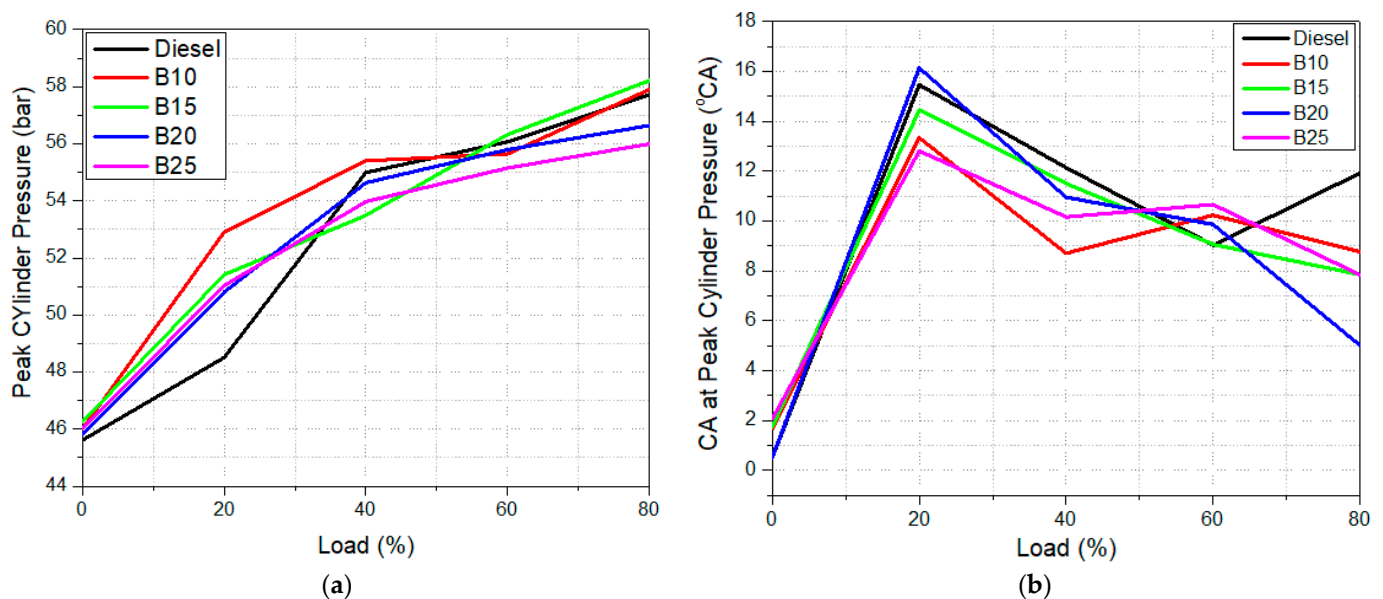


Figure 3. The peak CP and the corresponding CA variation for all test fuels with the engine load (a) maximum peak CP vs. engine load and (b) crank angle at the maximum peak CP vs. engine load.

3.1.2. Heat Release Rate (HRR)

The heat release rate (HRR) as a function of crank angle (CA) is presented in Figure 4a–e. HRR was calculated from cylinder pressure data using Equation (5) at a constant engine speed of 2700 rpm under engine loads of 0%, 20%, 40%, 60%, and 80% for all test fuels. In the calculation of HRR, heat loss correction was not applied. The results indicate that HRR increases with both biodiesel proportion and engine load. At 0% load, pure diesel exhibited the highest HRR of 379.36 J/CA, followed by B20 at 379.09 J/CA. Conversely, B15 achieved the minimum peak HRR of 349.85 J/CA, closely followed by B10, as shown

in Figure 4a. Notably, the highest peak HRR of 543.9 J/CA was achieved by B15 at 80% engine load at 20.9 °CA, while the lowest peak HRR of 349.85 J/CA was attained by B15 at 0% engine load at 30.35 °CA. Generally, diesel–biodiesel blends exhibited relatively higher HRR compared to pure diesel, which is attributed to the oxygen content in biodiesel. However, a slight reduction in HRR was observed with increasing biodiesel proportion at high loading conditions. This reduction is likely due to the richer mixture, which increases fuel viscosity, resulting in lower volatility and HRR. The decline in HRR with increased biodiesel at high engine loads is also attributed to shorter ignition delay (ID), which limits mixing before ignition [41].

As depicted in Figure 5a, the CA at which maximum heat release rate (HRR) occurs shifts further after top dead center (TDC) during the expansion stroke as engine load and biodiesel percentage increase. The earliest maximum HRR was observed at 17.83 °CA after TDC for B20 at 0% load, while the latest maximum HRR occurred at 30.53 °CA after TDC for B15 at 80% load. Notably, the majority of maximum HRR values fell between 20 °CA and 24 °CA after TDC. As illustrated in Figure 5b, a maximum peak HRR occurring significantly later after TDC suggests that a substantial portion of combustion energy is released after peak pressure. Compared to pure diesel, the peak HRR of biodiesel blends was achieved at a delayed CA after TDC, particularly at higher engine loads. This delayed burning of biodiesel may be attributed to the increased fuel quantity, which leads to higher viscosity [42]. This behavior is characteristic of diffusion-controlled combustion, where the burning rate is slower and more dependent on fuel–air mixing. Furthermore, the gradual combustion of the remaining fuel during the diffusion-controlled stage results in a delayed peak HRR. This delay in HRR indicates that combustion occurs during the cylinder's expansion, leading to lower pressure and temperature. This, in turn, may reduce NO_x emissions but potentially increase unburned hydrocarbon emissions. Fuels with higher cetane numbers exhibit slower burning characteristics, contributing to delayed HRR. The late peaks in HRR suggest that a significant portion of combustion energy is released after peak pressure, which may slightly reduce BTE while enhancing combustion stability and minimizing knock risks. These findings align with the results reported in [43].

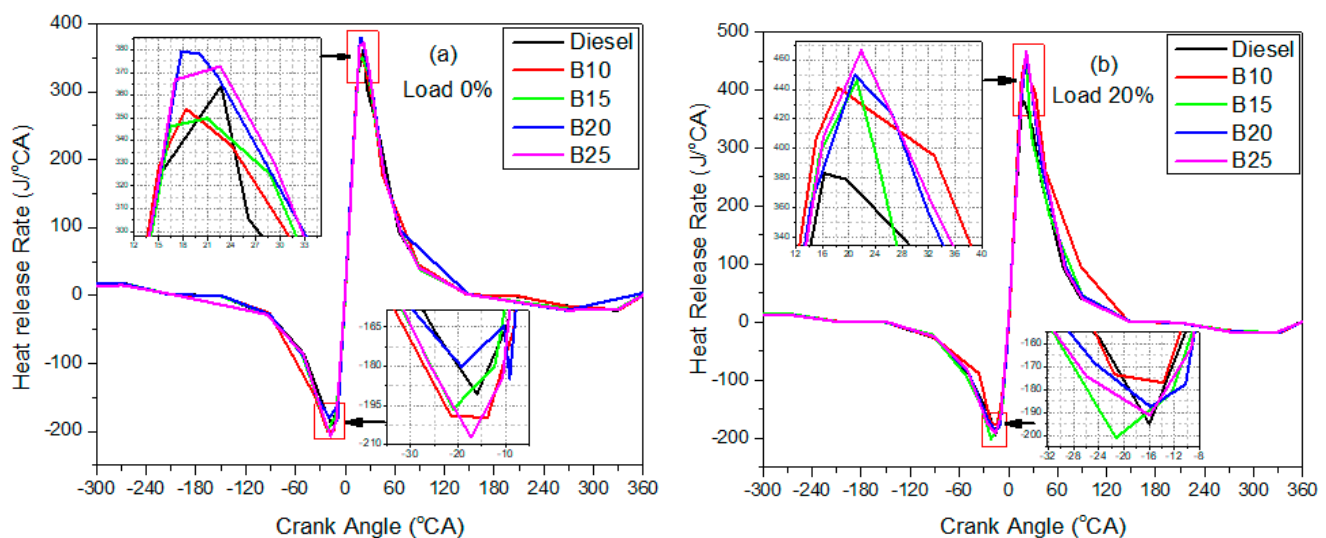


Figure 4. Cont.

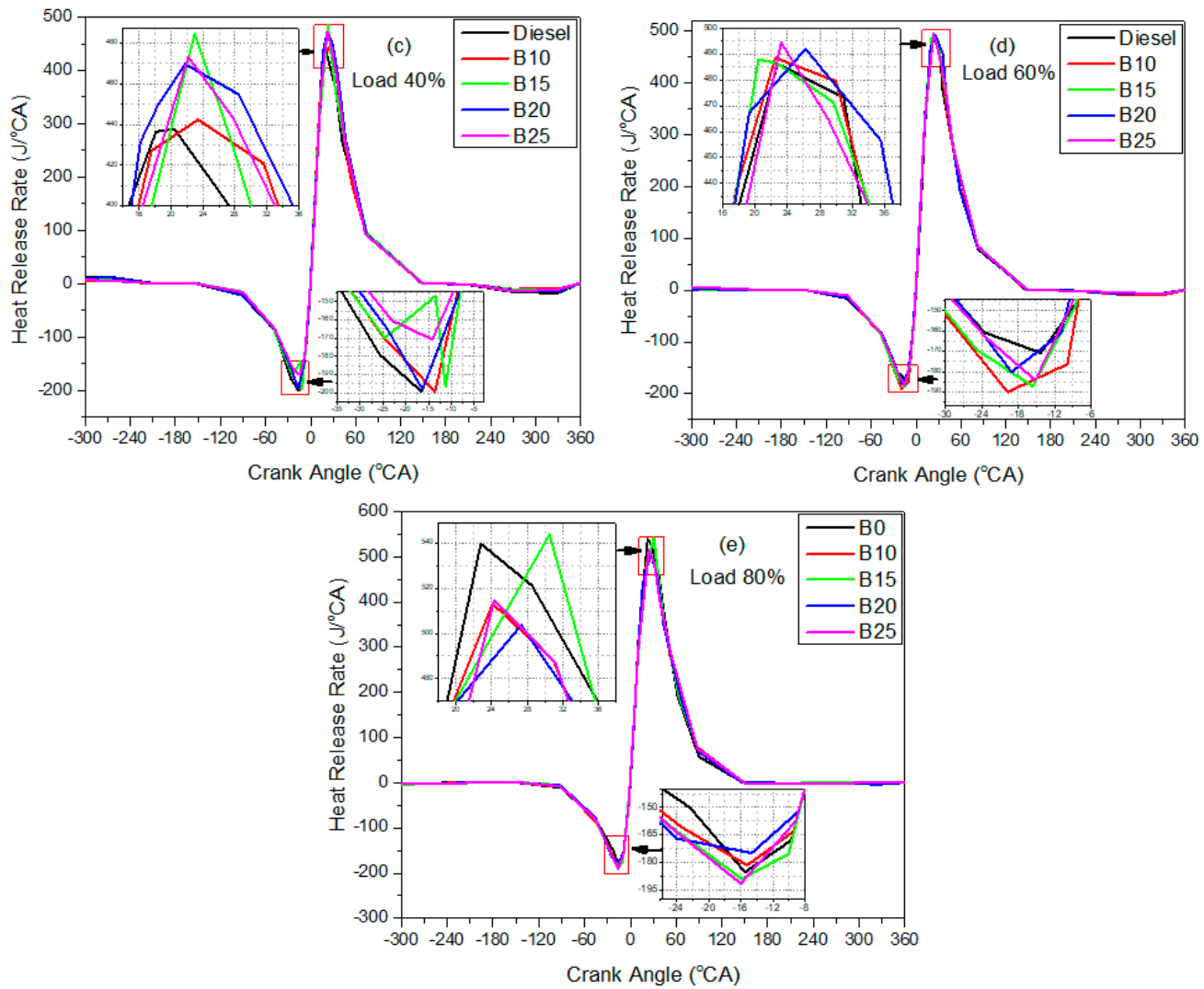


Figure 4. The influence of engine load variation on the HRR: (a) no load, (b) 20% load, (c) 40% load, (d) 60% load, and (e) 80% load.

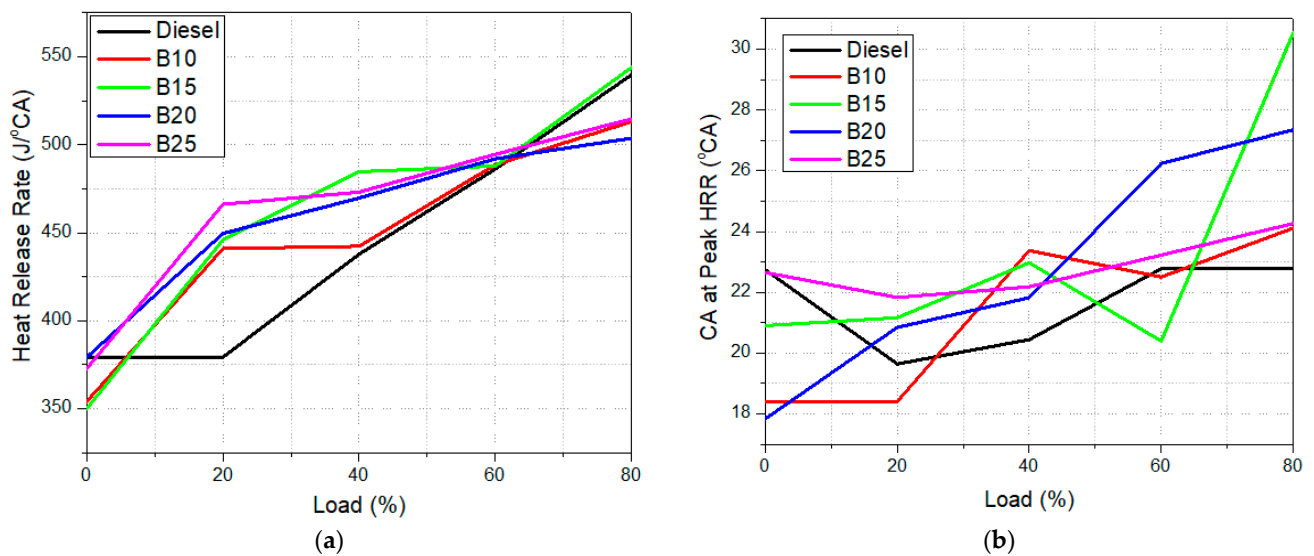


Figure 5. The variation in peak HRR and its crank angle for test fuels at different loads: (a) maximum peak HRR vs. load and (b) crank angle (CA) at peak HRR vs. load.

3.1.3. Start of Combustion (SoC)

Start of combustion (SoC) is defined as the point where the heat release rate (HRR) returns to zero, as depicted in Figures 6 and 7. The SoC for all test fuels was compared across various engine load conditions, as shown in Figures 6 and 7a–e. It was observed that, with the exception of pure diesel and B10 at 20% load, combustion initiated a few crank angles before top dead center (TDC). The behavior of B10 at 20% load may be attributed to its low biodiesel content, rendering it highly sensitive to diesel properties, particularly under the unstable combustion conditions associated with this load. Higher biodiesel blends, possessing improved cetane number and atomization characteristics, demonstrate more effective combustion under these conditions. Furthermore, under 20% load condition, the same trend was exhibited for pure diesel, and the SoC seems to occur after TDC. Specifically, biodiesel blends exhibited earlier SoC compared to pure diesel. At 0%, 20%, 40%, 60%, and 80% load, B25 achieved SoC at -0.1 , -0.01 , -0.137 , -0.131 , and -0.42 °CA bTDC, respectively, while pure diesel achieved SoC at -0.38 , 0.22 , 0.14 , 0.43 , and 0.45 °CA bTDC, respectively. Notably, SoC occurred earlier with increasing biodiesel blend proportion and engine load. This earlier SoC is primarily attributed to biodiesel's higher cetane number, lower volatility, and inherent oxygen content compared to pure diesel, facilitating easier combustion. Additionally, SoC can be influenced by increases in engine load and fuel quantity, which elevate cylinder temperature [44]. Consequently, at 80% load, SoC occurred earlier, with a greater number of crank angles before TDC, compared to no-load conditions. Specifically, the SoC of pure diesel at 80% load was 74% earlier, or advanced (-0.4 °CA before TDC), compared to 0% load (-0.1 °CA before TDC). Furthermore, B10 and B15 achieved SoC 96% earlier (-0.48 °CA and -0.28 °CA before TDC) at 80% load than at 0% load (0.03 °CA and 0.1 °CA after TDC), respectively. This advancement is attributed to the increased cylinder temperature resulting from the greater fuel quantities supplied. At 80% load, the SoC for pure diesel was comparable to that of the biodiesel blends. This is likely due to the increased viscosity of the biodiesel blends, which impedes fuel atomization. As illustrated in Figure 7a–e, combustion initiates when the HRR becomes positive, crossing the zero line, which is designated as the SoC line. The results demonstrate that biodiesel blends achieved an earlier SoC compared to diesel and that increasing the biodiesel percentage further advanced the SoC.

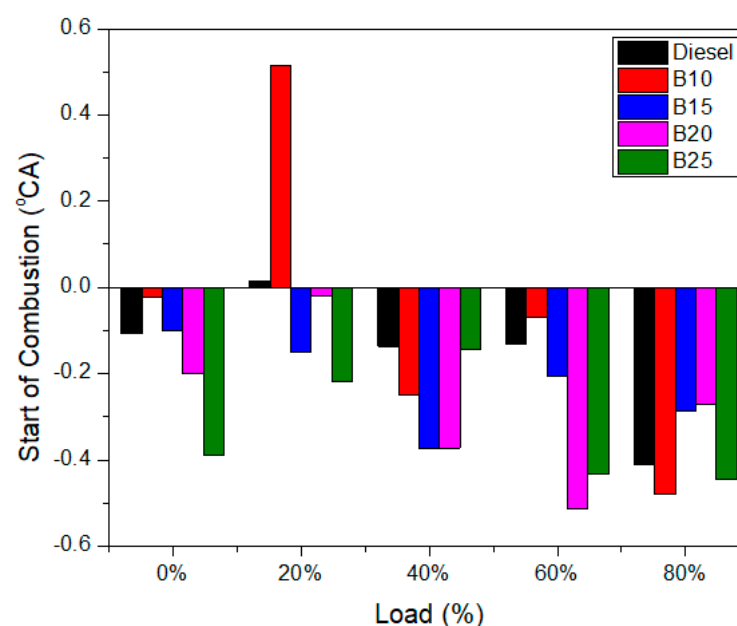


Figure 6. The variation in SoC with the engine load.

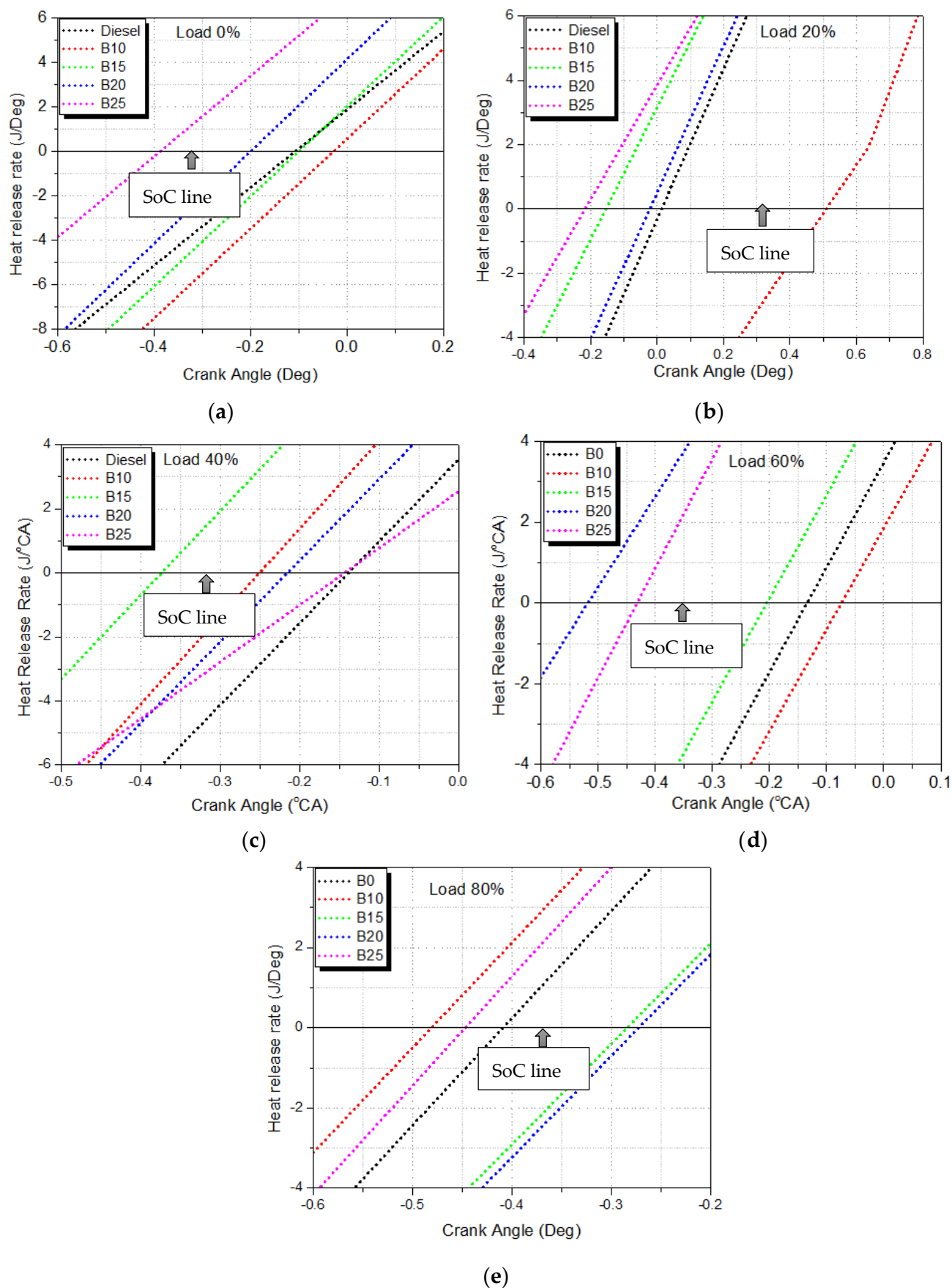


Figure 7. The HRR that shows SoC variation as a function of engine loading (a) load 0%, (b) load 20%, (c) load 40%, (d) load 60%, and (e) load 80%.

In general, biodiesel's phases of combustion deviate from diesel's due to its distinct chemical and physical properties. Its higher cetane number reduces ignition delay, leading to earlier combustion. The inherent oxygen content enhances combustion, potentially accelerating heat release. Viscosity influences fuel atomization, impacting mixture formation and consequently altering combustion timing. Biodiesel's lower heat of combustion affects the cylinder pressure rise, further influencing combustion phasing. These combined factors result in distinct combustion variations, which significantly impact engine performance and emissions.

3.1.4. Ignition Delay Period (ID)

ID is a crucial parameter for characterizing combustion behavior, defined as the CA interval between the start of injection and the onset of combustion. It comprises both physical and chemical delay periods. The physical delay encompasses fuel atomization, evaporation, and fuel–air mixture physical interaction, while the chemical delay involves the pre-combustion chemical reactions between the fuel and air mixture. ID is influenced by injection pressure and fuel properties. From the HRR diagrams in Figure 7a–e, the point where HRR becomes negative is attributed to the fuel's evaporative cooling effect, and the point where HRR becomes positive signifies the SoC. The ID period decreases with increasing engine load due to the elevated cylinder temperature, which enhances chemical reaction rates and thermal decomposition. As illustrated in Figure 8, the ID period decreases with increasing engine load. At 0% load, pure diesel and B10 exhibited higher ignition delays of 23.01 °CA and 23.5 °CA, respectively, compared to the other fuel samples. However, biodiesel blends consistently exhibited shorter ignition delays (ID) across all load conditions compared to diesel. Specifically, B20 achieved the shortest ID of 22.45 °CA, followed closely by B25 with 22.55 °CA. It is generally understood that biodiesel's higher density and viscosity tend to increase ID due to reduced atomization and vaporization rates [45]. Nevertheless, biodiesel possesses a relatively higher cetane number than diesel fuel, which counteracts this effect by shortening ID [46]. It was reported in [47] that increasing the biodiesel blend fraction leads to shorter ID due to the fact that the high reactivity of the fuel–air mixture controls the combustion process, decreases the premixed stage dimension, and thus alters the engine performance and emissions.

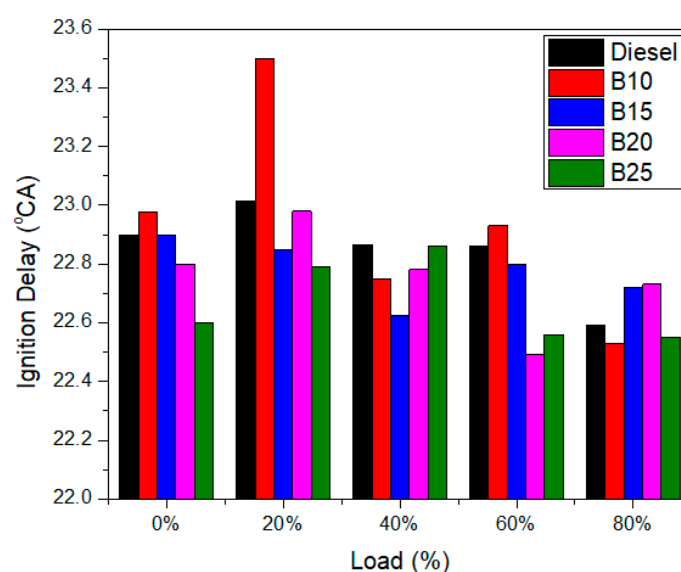


Figure 8. The ID variation vs. engine load for all test fuels.

During the ID period, all test fuels exhibited a minimum negative heat release rate (HRR). As shown in Figure 4a–e, biodiesel blends, particularly B25, achieved the lowest negative HRR of -207.3 J/CA at $-17.3 \text{ }^{\circ}\text{CA}$, compared to pure diesel's -170.71 J/CA at $-14.24 \text{ }^{\circ}\text{CA}$. This higher negative peak HRR in biodiesel is attributed to its lower latent heat of vaporization, resulting in a more pronounced evaporative cooling effect. Furthermore, biodiesel blends demonstrated an earlier start of combustion (SoC) and shorter ID compared to diesel, which can be attributed to their higher cetane number and inherent oxygen content [48].

3.2. Performance Analysis

3.2.1. Brake Thermal Efficiency (BTE)

Figure 9a illustrates the brake thermal efficiency (BTE) of each fuel blend as a function of engine load. BTE is a critical parameter indicating the effectiveness of fuel energy conversion into mechanical work. This study examined the impact of engine load and CMS biodiesel proportion on BTE. The results demonstrate that BTE increases with engine load but decreases with increasing biodiesel percentage, exhibiting a decline at maximum engine load. This decline is attributed to the lower energy density of B10, B15, B20, and B25 compared to pure diesel. The reduction in BTE at maximum load is likely due to the increased fuel quantity in the cylinder, resulting in partial combustion. Across all engine load conditions, pure diesel demonstrated the highest BTE, while B25 registered the lowest, primarily due to biodiesel's lower calorific value and higher viscosity [5]. Generally, B10 has shown a better efficiency closer to pure diesel fuel compared to other biodiesel blends. The maximum efficiency of 32.55% was obtained by diesel fuel at 60% loading condition, which was increased by 5.5%, 10.38%, 7.06%, and 14.74% compared to B10, B15, B20, and B25, respectively, and the same trend was reported by [41].

3.2.2. Brake-Specific Fuel Consumption (BSFC)

Figure 9a shows the relationship between engine loads and BSFC for all test fuel diesel–biodiesel blends. As the engine load rises, the BSFC declines up to some point, which is due to the rise in temperature in the cylinder. In all loading conditions, the BSFC of the biodiesel blends was increased by 17–26% compared to the diesel because of the higher energy density of the pure diesel [49]. The slight increment of BSFC at the maximum load is attributed to the incomplete combustion of the rich fuel–air mixture. From the diesel–biodiesel blend fuels, B10 has shown the lowest BSFC, and B25 shows the highest, which is in agreement with the result reported in [50].

3.2.3. Exhaust Gas Temperature (EGT)

The EGT indicates the combustion effectiveness and chemical energy conversion to mechanical work manifested as engine performance. As illustrated in Figure 9b, the EGT increases with the rise in engine load due to the higher fuel flow rate into the combustion chamber. The lowest EGT was recorded for the pure diesel, while B25 shows the highest temperature, and the same trend was reported by [51]. The reason for this is that biodiesel contains inherent oxygen, which increases the temperature of the cylinder [52]. Furthermore, the higher CN of the biodiesel causes an early start of ignition, leading to the increased residence time at higher temperatures, which increases the EGT.

3.2.4. Mass Flow Rate of the Fuel ($\dot{m}_f$)

As depicted in Figure 9c, the $\dot{m}_f$ was measured against the engine load. When the engine load increases, the fuel flow rate into the combustion chamber also rises to maintain the speed of the engine constant. The mass flow rate of biodiesel, especially for B25, increases by 16% and 25% at lower loading and higher loading operations, respectively,

compared to pure diesel. The mass flow rate of the fuel increases with the rise in biodiesel percentage, which is attributed to the lower energy density that requires more biodiesel to produce an equivalent power with the diesel [53].

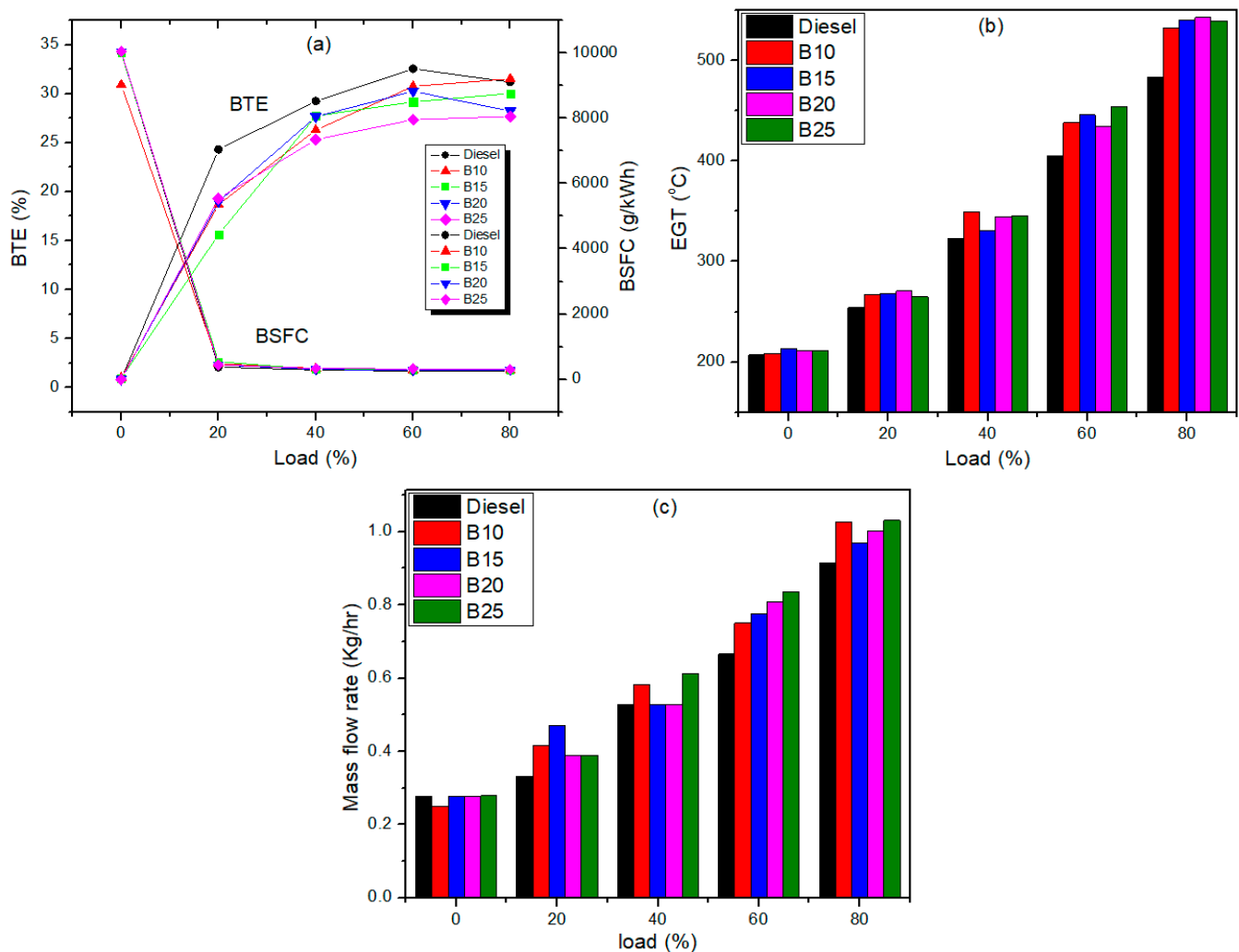


Figure 9. The performance characteristics of the CMS biodiesel/diesel blend (a) BTE and BSFC vs. load, (b) EGT vs. load, and (c) $\dot{m}_f$ vs. load.

3.3. Emission Characteristics

3.3.1. Carbon Monoxide (CO)

The effect of the CMS biodiesel percentage on the CO emission is depicted in Figure 10a. CO is an indicator of partially complete combustion due to various reasons in the combustion chamber. Initially, at no load conditions, the CO emission is higher and declines with the rise in engine load, up to a 40% load, and further increases due to incomplete combustion of rich fuel–air mixture. Among all test blends, B20 has shown the lowest CO emission in all loading conditions, followed by B15 and B10 due to the inherent oxygen in it, which helps the successive oxidation of CO to CO₂. B25 has shown a 34–84% increase in CO emission in the range of 0–60% engine load operation compared to pure diesel, but at 80% engine load operation, the CO emission of the pure diesel was 203 ppm increased by 34% compared to the B25 of 151 ppm, which is attributed to the increased viscosity with the engine load and the quantity of fuel supplied that leads to poor atomization and evaporative cooling. Moreover, the result shows that the CO emission reduces with the increase in biodiesel percentage.

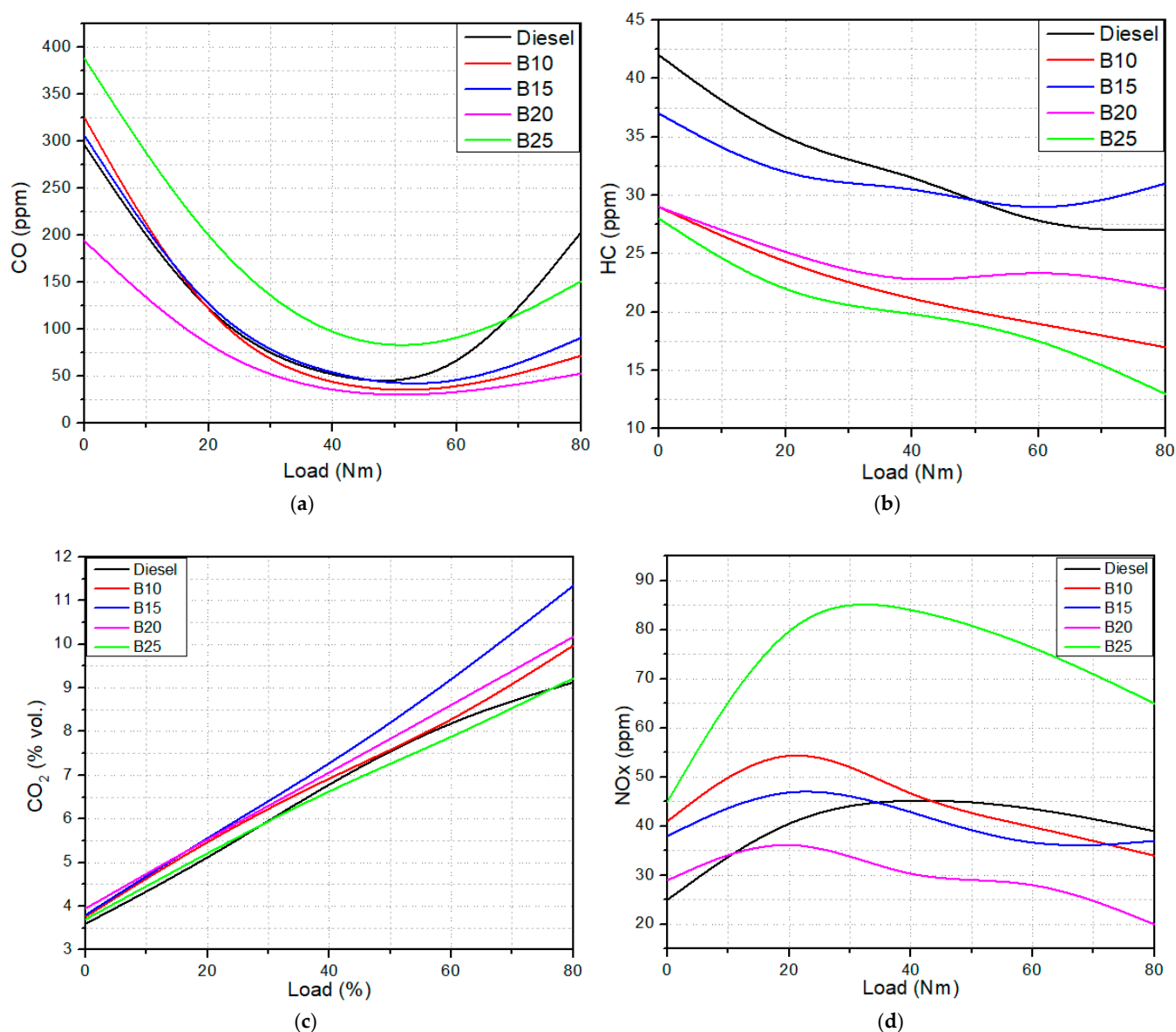


Figure 10. The variation in exhaust gas emission with load from the croton macrostachyus biodiesel blend. (a) Load vs. CO; (b) load vs. HC; (c) load vs. CO₂; and (d) load vs. NOx emission.

3.3.2. Hydrocarbon (HC)

In a diesel engine, unburned HC is formed due to lower cylinder temperature, incomplete flame propagation, and low injection pressure. As shown in Figure 10b, the HC emission reduces when the engine load increases, it was higher at no-load conditions, and started slightly declining as the load increased. The higher HC emission is registered by pure diesel, followed by B15, and the lowest HC is shown by B25, followed by B10 and B20. HC emissions of B25 were reduced by 33.3–51.8% compared to pure diesel caused by the increased cetane number and oxygen molecules in the biodiesel, which increases the conversion of hydrocarbons to CO₂ and H₂O, which is in agreement with the result reported in [54]. The higher HC emission of pure diesel is attributed to the longer ID compared to other blends. It is also correlated that the HC emission is directly proportional to the ID period, which reduces with the rise in engine load. Furthermore, biodiesel's fatty acid profiles are also an important factor, which shows that unburned HC emission and the unsaturation level of biodiesel are inversely proportional [55]. The higher unsaturation level in the biodiesel results in a lower HC emission, which is attributed to the low

volatility of saturated fatty acid that leads to incomplete combustion and thus increases the HC emissions.

3.3.3. Nitrogen Oxide (NO_x)

The main reasons for the formation of NO_x emission are high cylinder temperature, more oxygen concentration, and fuel residence time. The effect of CMS concentration on the NO_x emission with the engine load is illustrated in Figure 10d. The NO_x emission was less at no load conditions. However, at 20% loading, the NO_x emission was elevated to the maximum point and declined with a rise in engine load. At low loading conditions, less NO_x emission was obtained in pure biodiesel, but when the load increased, the NO_x emission of the diesel fuel increased more than B10, B15, and B20. The reason is that at low loading, the temperature of the engine is less due to the lean fuel–air ratio. In all loading conditions, B25 has shown a 67–102% rise in NO_x emission compared to pure diesel, which is caused by the oxygen molecule in the biodiesel that results in elevated temperature in the cylinder [18]. However, it was also revealed that B20 has shown a 48% decrease in NO_x emission compared to diesel fuel. The NO_x emission decreases with the rise in engine load, which is attributed to the incomplete combustion caused by a highly rich fuel–air ratio that increases the viscosity, which may reduce the cylinder temperature. It was also suggested that by delaying the fuel injection, the in-cylinder combustion temperatures can be reduced, which lowers the NO_x emissions [56–58].

3.3.4. Carbon Dioxide (CO₂)

CO₂ emission is always a byproduct of fuel and air when combustion takes place and increases with the quantity of fuel. The CO₂ emission variation with the engine load is shown in Figure 10c. It is linearly increasing with the increase in engine load due to the increase in mass flow rate of the fuel. In all loading conditions, the B25 has shown the lowest CO₂, followed by pure diesel, while the B15 has the highest CO₂ emission, followed by B20 and B10. The lowest CO₂ shown in B25 is attributed to incomplete combustion of the viscous rich mixture in the cylinder [48].

4. Comparative Analysis

The comparative analysis of the engine combustion, performance, and emission characteristics of croton macrostachyus (CMS) biodiesel was performed with another biodiesel produced from different feedstock. As illustrated in Table 4, the CMS biodiesel has a higher calorific value compared to the other mentioned biodiesels produced from various feedstocks. The engine performance results show that the BTE decreases and the BSFC and EGT rise with the rise in biodiesel blend. However, the increase in engine load results in a rise in BTE and a decrease in BSFC. Moreover, using biodiesel in the CI engines decreases the CO and HC emissions, whereas it increases the NO_x emission due to the inherent oxygen molecules compared to pure diesel. The effect of injection timings was also investigated in [45,49] for different biodiesel blends and it was obtained that advancing the injection timing decreases the CO, HC, smoke, and particulate matter but increases NO_x emission. The combustion analysis of the CP and HRR was also investigated by [24], and it was reported that pure diesel achieved a higher peak CP and HRR compared to biodiesel. However, in this study, an extensive experimental analysis was performed on the peak CP and peak HRR with their crank angle position for diesel and biodiesel blends, and the ID period and the CA where the combustion starts were also evaluated, which can be considered as the novelty of this work. Therefore, the result show that biodiesel blends have a higher peak CP and HRR than the pure diesel at low load operation. However, the biodiesel's CP and HRR become slightly lower than the pure diesel fuel at higher load operation, which is

attributed to the increase in the quantity of fuel and its viscosity with the rise in engine load that results in poor atomization. The position of the CA where the peak CP and peak HRR achieved were also advanced (earlier) and retarded (later) after TDC for biodiesel compared to the pure diesel is in agreement with a report in [40]. Furthermore, biodiesel also achieved a shorter ID and an earlier SoC. The heating value of the CMS biodiesel is also higher than the grape seed biodiesel, fusel oil, soapberry seed biodiesel, waste cooking oil biodiesel, roselle biodiesel, and rapeseed methyl ester by 9%, 25%, 5.5%, 4.3%, 3.2%, and 6.3%, respectively. The viscosity of CMS biodiesel is also less compared to the mentioned biodiesels, which makes it suitable for IC engine application without engine modification.

Table 4. Comparative analysis of CMS biodiesel with other biodiesels produced from different feedstock on the characteristics of CI engine.

Feedstock Type and Biodiesel Properties	Operating Conditions	Result	Author
Grape seed biodiesel: lower heating value of 36.45 kJ/kg, kinematic viscosity of 3.62 mm ² /s, density of 869 kg/m ³	Constant speed of 1500 rpm. Engine performance and emission characteristics were optimized at variable injection timing (6–30 bTDC), injection pressure (400–1000 bar), engine load (20–100%), and biodiesel blend (0–60%).	Achieved a BTE of 31.85%, mechanical efficiency of 64%, BSFC of 0.278 kg/kWh, CO of 0.127%, NOx of 357 ppm, and HC of 8 ppm at optimal independent variables of injection timing (6° bTDC), an engine load (82%), exhaust gas recirculation (6.7%) injection pressure of 1000 bar, and a grape biodiesel blend of 33%.	[59]
Waste cooking oil biodiesel: kinematic viscosity of 4.64 mm ² /s, lower heating value of 38.28 MJ/kg, density of 879 kg/m ³	At constant engine speed of 1500 rpm with the variation in engine load. The range of biodiesel blends from B5 to B40 wit 5% difference and B100.	Compared to pure diesel, biodiesel blends achieved lower BTE by 27%, lower brake power by 4.03%, lower torque by 16.76%, higher BSFC by 4.8%, lower CO and HC by 52.2% and 60%, and higher CO ₂ and NOx 28.1% and 45.4%, respectively.	[17]
Fusel oil: lower heating value of 30 MJ/kg, water content 15.5% density of 847 kg/m ³	Variable engine loads (0%–75%) with engine speeds of 1500 rpm and 2100 rpm. Sample fuel blend F20 of 20% vol fusel oil and 80% vol diesel	Compared to pure diesel, due to the higher water content in the fusel oil, NOx was reduced by up to 20% at 1500 rpm engine speed and 75% engine load. Both CO and HC emissions were increased and the BSFC increased.	[60]
Soapberry seed biodiesel: lower heating value of 37.82 MJ/kg, kinematic viscosity of 3.58 mm ² /s, density of 832 kg/m ³	At a constant engine speed of 1500 Variable loads of 25%, 50%, 75%, and 100% Fuel blends B10, B20, and B30	Compared to the biodiesel blends, pure diesel achieved a higher BTE, cylinder pressure, and HRR B30 has a lower CO and HC but higher NOx emission.	[24]
Roselle biodiesel: lower heating value of 38.74 MJ/kg, kinematic viscosity of 5.64 mm ² /s, density of 877 kg/m ³	Fuel samples of diesel, B20–B100 at 20% increment, injection timing of 19–27 bTDC, and engine load of 25, 50, 75, and 100% at a constant compression ratio of 17.5 and engine speed of 1500 RPM.	B20 showed greater EGT, ID, and maximum PRR. Biodiesel resulted in a lower NOx emission. B20 with varying the injection timing from 19 to 27 °CA bTDC, BSFC and EGT, CO ₂ , and NOx increased by 15.8%, 4.6%, 5.3%, and 12.9%, respectively, while BTE and smoke decreased by 4.4% and 18.8%, respectively.	[58]

Table 4. Cont.

Feedstock Type and Biodiesel Properties	Operating Conditions	Result	Author
Rapeseed methyl ester: lower heating value of 37.5 MJ/kg, kinematic viscosity of 4.48 mm ² /s, density of 882.4 kg/m ³	Fuel samples of diesel and B100, Advanced injection timing (AIT) of 6, 11, 16, and 21 °CA before TDC. Retarded injection timing (RIT) of 1, 5, 10, and 15 °CA after TDC. Engine load (in Brake mean effective pressure): 2.5 and 5 bar BMEP. At a constant engine speed of 1500 RPM.	Compared to diesel, For B100, HC and CO were reduced by 21% and 31% at a load bar of 5 BMEP, respectively. AIT reduced the CO and HC compared to the RIT, whereas the NOx decreased by 24% for RIT and increased by 7% for AIT, Smoke and particulate matter were reduced for AIT compared to RIT, especially for B100.	[52]
Fig seed oil methyl ester; lower heating value of 39.57 MJ/kg, kinematic viscosity of 4.78 mm ² /s, density of 919 kg/m ³	Fuel samples: diesel, B5, B10, and B20. At constant full load and engine speed of 1550, 1700, 1850, 2000, and 2150 RPM.	Biodiesel blends exhibited a higher CP and PRR and lower HRR compared to diesel. The CA at which the peak CP is attained is farther away from TDC.	[37]
Mustard oil biodiesel;	Fuel samples: diesel, B10, B20, B30, and B40 Engine load of 0, 25, 50, 75 and 100% at constant of 1500 rpm	Biodiesel blends have shown a higher CP. The CP rises with the rise in biodiesel blend percentage. The CA position of the peak CP for all biodiesel blends occurs slightly earlier than diesel. The peak HRR occurs at almost the same CA for all biodiesel blends and diesel at higher loading conditions, the peak HRR of biodiesel is closer to diesel.	[40]
Croton macrostachyus (CMS) seed biodiesel: lower heating value of 40.04 MJ/kg, kinematic viscosity of 4.43 mm ² /s, density of 846.3 kg/m ³	Fuel samples: diesel, B10, B15, B20, and B25 Engine load 0, 20, 40, 60, and 80% At a constant speed of 2700 RPM Injection timing of 23 °CA before TDC	Compared to the biodiesel blends, pure diesel achieved an increased BTE by 5.5–14%, and BSFC by 17–26% and decreased EGT by 7–12%. Compared to diesel, CO and HC are lower, while the carbon dioxide (CO ₂) and NOx are higher for biodiesel blends. Peak CP and HRR rises with the rise in engine load and biodiesel blend. The SoC advances CA before TDC, while the ID period decreases with the increase in engine load and biodiesel percentage.	Present study

5. Engine Durability and Implication of Using Biodiesel

Studies show biodiesel, derived from non-edible feedstocks, enhances engine component lubricity and reduces wear [61,62]. Furthermore, biodiesels are solvents that are capable of dissolving layers of foreign materials deposited onto surfaces of engine components [63,64]. Biodiesel's high FAME and monoglyceride content improves lubrication (due to higher viscosity), which is crucial for injection pumps to prevent filter clogging. However, this high viscosity leads to poor fuel atomization, injection system blockage, and increased carbon deposits, reducing engine durability [42,65–69]. Using biodiesel within standard viscosity ranges is essential for proper atomization and to prevent high pumping

pressure. Direct biodiesel use leads to injector coking, carbon buildup, and lubricant issues. Blending biodiesel with diesel or preheating vegetable oils effectively reduces viscosity, improving atomization and combustion [70]. In post-production, biodiesel properties can be improved by adding oxygenated additives, antioxidants, cetane improvers, lubricity enhancers, cold flow additives, and combustion enhancers to meet fuel standards and enhance quality [71,72]. Fuel aging, caused by chemical and physical degradation, alters fuel properties, reducing combustion efficiency, increasing emissions, and impairing engine performance. This necessitates strict adherence to fuel standards, particularly viscosity, for reliable industrial use [73]. Therefore, the viscosity of CMS biodiesel is also less and within the standard, which makes it suitable for IC engine application without engine modification.

In addition, oxidation stability is also the other challenge of utilizing biodiesel as a fuel in diesel engines. In an internal combustion engine, the fuel can be pressurized above 1000 bar, leading to a significant increase in temperature (above 100 °C), which can increase the thermal degradation of the biodiesel [74]. This leads to increased chances of gum and sediment formation, especially at the standard high temperatures at which a normal motor operates. These deposits are a well-known problem across the entire automotive industry as they can affect the injectors, leading to an early failure of the entire system [68]. The oxidation of biodiesel has a significant effect on the lubricating properties [75]. The oxidation stability of CMS biodiesel employed in the present study was tested using a professional Rancimat (892, Metrohm) device with and without tertbutyl hydroquinone (TBHQ) and reported in [31]. The result shows that the croton macrostachyus (CMS) biodiesel exhibited an induction time of 2.39 hours, which is highly susceptible to oxidation, and the shelf life is short. However, the induction time is enhanced to 16.17 hours due to the addition of 1000 ppm of TBHQ antioxidant. Therefore, the addition of an antioxidant to the biodiesel is crucial to improve oxidation stability. TBHQ is the most effective antioxidant to improve the oxidation stability of the biodiesel [76,77].

The cost of producing biodiesel varies based on the type of raw material used, the manufacturing methods employed, and the size of the production facility. Currently, it is typically more expensive than regular diesel because the technology is still relatively new and production is often smaller. However, as technology improves and production increases, along with supportive policies and better supply chains, the cost is expected to decrease [78]. Feedstock availability is crucial for economic viability, driving research into low-cost, year-round sources [79]. Along with feedstock costs, factors such as material loss, energy inefficiency, and reactor performance are also critical to the economic viability of biodiesel production [80]. While CMS biodiesel offers good engine performance, its high feedstock cost, poor atomization, low-temperature flow issues, high NO_x emissions, and fuel stability problems limit its widespread use [81]. Biodiesel's higher production cost makes it more expensive than diesel. Though it has lower energy content, increasing consumption, it enhances engine lubrication. Cold weather and filter clogging are potential issues that need to be under consideration. Therefore, CMS biodiesel blends (such as B20) offer economic viability through reduced engine wear and sustainability benefits.

6. Conclusions

This study experimentally investigated the effects of croton macrostachyus (CMS) biodiesel addition on the combustion, emissions, and performance of a compression ignition (CI) engine. As a novelty, the detailed combustion analysis CMS biodiesel–diesel blends were demonstrated in which the HRR plots were correlated with engine performance and emission outputs. Tests were conducted at a constant speed of 2700 rpm and variable engine loads of 0, 20, 40, 60, and 80%. The following conclusions were drawn from the results:

- The results indicate that at 20% load, peak cylinder pressure (CP) occurs after top dead center (TDC), and this peak shifts towards TDC as the load increases.
- At 80% engine load, B25 demonstrated the highest peak cylinder pressure (CP) of 58.21 bar and maximum heat release rate (HRR) of 543.9 J/CA compared to the other tested blends.
- Increasing engine load and biodiesel blend percentage resulted in the maximum heat release rate (HRR) occurring later in the expansion stroke further from top dead center (TDC). Specifically, the earliest maximum HRR was observed at 17.83 °CA after TDC for B20 at no load, while the latest occurred at 30.53 °CA after TDC for B15 at 80% load. Furthermore, ignition delay decreased with increasing engine load and biodiesel blend, and the combustion start advanced further before TDC for biodiesel blends compared to pure diesel.
- A maximum brake thermal efficiency (BTE) of 32.55% was recorded for pure diesel, which can be attributed to its superior energy density and lower viscosity relative to biodiesel blends. Consequently, pure diesel demonstrated the lowest brake-specific fuel consumption (BSFC) and exhaust gas temperature (EGT). B10 exhibited slightly higher values, while B25 showed the highest BSFC and EGT among the tested blends.
- Carbon monoxide (CO) emissions decreased with increasing biodiesel blend percentage up to B20, but increased beyond B20. Hydrocarbon (HC) emissions for B25 were reduced by 33.3–51.8% compared to diesel, which is attributed to the combined effect of cetane number and inherent oxygen content. Nitrogen oxide (NO_x) emissions for B25 increased by 67–102% compared to pure diesel. Carbon dioxide (CO₂) emissions peaked with B15, followed by B20 and B10. B25 exhibited the lowest CO₂ emissions, which is likely due to incomplete combustion from a richer mixture and increased fuel viscosity with higher biodiesel proportions.
- Overall, this experimental research demonstrated that croton macrostachyus (CMS) seed oil biodiesel exhibits promising characteristics for use in internal combustion engines. However, accelerated field testing and economies of large-scale CMS biodiesel production need to be assessed further.
- To further optimize the utilization of CMS biodiesel, future studies should investigate its combustion, performance, and emission characteristics in multi-cylinder compression ignition (CI) engines under real-time driving cycle conditions. Key areas of focus should include the influence of exhaust gas recirculation (EGR), the addition of nanoparticles, and the optimization of fuel injection pressure and timing apart from the engine wear test.

Author Contributions: Methodology, A.S.M.; investigation, A.S.M.; formal analysis, A.S.M.; writing—original draft preparation, A.S.M.; supervision, V.R.A., and S.M.A.; writing—review and editing, V.R.A., S.M.A., M.D. and R.B.; project administration, S.M.A.; funding acquisition, S.M.A., R.B. All authors have read and agreed to the published version of the manuscript.

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Conflicts of Interest: The authors declare no conflicts of interest in carrying out this work.

Abbreviations and Nomenclatures

The following abbreviations are used in this manuscript:

AIT	Advanced injection timing
aTDC	After top dead center
B10	10 vol% biodiesel and 90 vol% diesel
B15	15 vol% biodiesel and 85 vol% diesel
B20	20 vol% biodiesel and 80 vol% diesel
B25	25 vol% biodiesel and 75 vol% diesel
BMEP	Brake mean effective pressure
BSFC	Brake-specific fuel consumption
bTDC	Before top dead center
BTE	Brake thermal efficiency
CA	Crank angle
CD	Combustion duration
CI	Compression ignition
CMS	Croton macrostachyus
CN	Cetane number
CO	Carbon monoxide
CO ₂	Carbon dioxide
CP	Cylinder pressure
Diesel	0 vol% biodiesel and 100 vol% diesel
EGT	Exhaust gas temperature
HC	Hydrocarbon
HRR	Heat release rate
IC	Internal combustion
ID	Ignition delay
LHVi	Lower heating value
m _f	Mass flow rate of fuel
NO _x	Nitrogen oxide
P _B	Brake power
ppm	Parts per million
R	Crank radius to connecting rod length
R _c	Compression ratio
RIT	Retarded injection timing
SoC	Start of combustion
TDC	Top dead center
U _i	Uncertainty
U _i (%)	Percentage of uncertainty
V	instantaneous engine cylinder volume
V _c	clearance volume
ν _i	Viscosity
X _i	Blending ratio
X _m	Arithmetic mean
ρ _i	Density
γ	Specific heat ratio
θ	Crank angle
σ	Standard deviation

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