

RESEARCH PAPER

Investigation of Effects of Kerosene Variation on a Tertiary Blended Biodiesel Powered Internal Combustion Engine Performance and Emission

Elijah Eferoghene Onojowho^{1,*} and Godwin E. Sadjere²

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¹ Department of Mechanical Engineering, Obafemi Awolowo University, Ile-Ife, Nigeria

² Department of Mechanical Engineering, University of Benin, Benin City, Nigeria

*Correspondence: Department of Mechanical Engineering, University of Nigeria, Nsukka, Nigeria.

E-mail: efe.elijah@gmail.com

Abstract

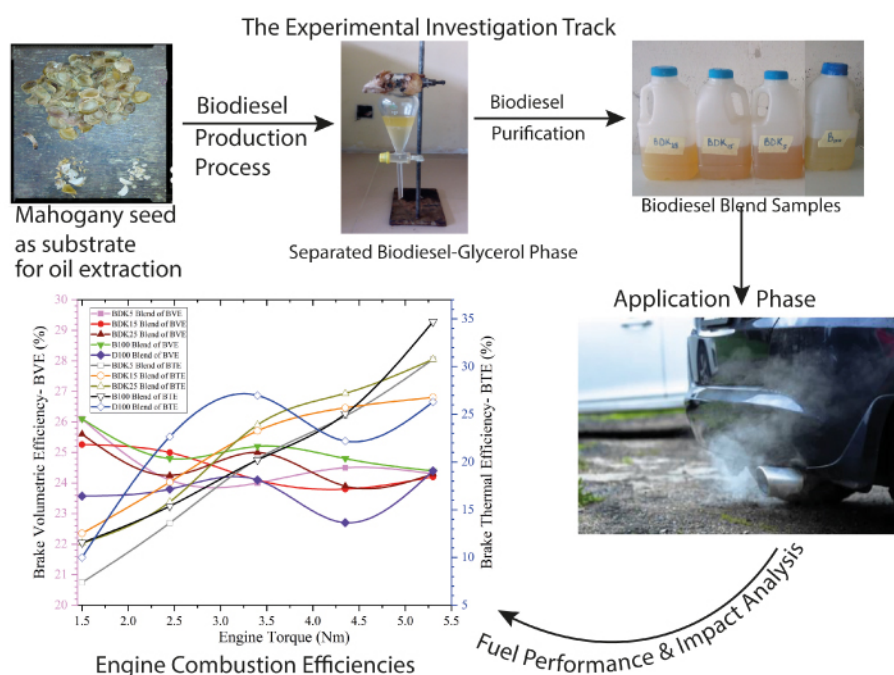
Blended fuel performance and emissions have been suggested as a surrogate for pure conventional diesel. Few countries have adopted 15% and lower biodiesel blending. Yet, lower emission levels than at present remains elusive. This study investigated the tertiary blends of *Khaya senegalensis* (African Mahogany) biodiesel and conventional diesel with varied kerosene proportion in a direct injection compression ignition engine to improve engine performance and reduce emissions. It is an experimental-based methodology process involving ASTM standard characterizations for 5% kerosene to biodiesel-diesel (BDK₅), 15% kerosene to biodiesel-diesel (BDK₁₅), 25% kerosene to biodiesel-diesel (BDK₂₅), pure diesel (D₁₀₀), pure biodiesel (B₁₀₀) blends at constant 10% biodiesel proportion in each tertiary blend. Results showed significant decrease in viscosity and density leading to good atomization of the tertiary blends. Furthermore, the rich mixture combustion of blends indicated BDK₁₅ and BDK₅ to be comparatively better than D₁₀₀ in air-fuel ratio with 12.28, 10.3 and 8.99 (BDK₁₅); 11.32, 11.49 and 10.6 (BDK₅) as against 14.35, 9.81 and 8.39 (D₁₀₀). The brake mean effective pressure effects were 2.117 bar, 2.752 bar and 3.37 bar (BDK₁₅); 2.122 bar, 2.527 bar, and 3.255 bar (BDK₅); 2.058 bar, 2.377 bar and 3.355 bar (D₁₀₀) at 3.4 N m, 4.35 N m and 5.3 N m,



respectively. Similarly, brake thermal efficiency significantly improved with BDK₁₅ and BDK₅ over D₁₀₀ on progressive torque increments whereas the energy liberated performance of BDK₁₅ was comparatively better. All tertiary blends emitted lower CO₂ than D₁₀₀. However, D₁₀₀ had the lowest exhaust gas temperature. There is a significant kerosene blended fuel effect on compression ignition engine performance and emissions.

Keywords: tertiary blend performance, kerosene blended fuel combustion, air-fuel ratio, *Khaya senegalensis* biodiesel emission, mass flow rate

Graphical Abstract



1. Introduction

Heat energy liberation from fuel demands combustion processes in an internal combustion engine to enhance energy conversion. This activity is usually associated with emission of byproducts that need to be addressed. Micro-level emission reduction study approaches considered so far are based on these variables; specific vehicle brands, location based, conventional fuel based, data based, and revised or localized emission models [1–5]. Quite a few studies are also simulation dependent. Fuel blending has been a major technique to reduce emission of byproducts during combustion and improving or maintaining engine performance simultaneously.

Blending originated with a secondary or binary combination state with a parent fuel and an additive in minor proportions. Biodiesel-diesel binary blending has been investigated for engine and auxiliary components' performance enhancement, combustion and emissions reduction from waste oil, edible and non-edible seed based biodiesel substrates [6–11]. These performance and emission studies are now exposed to artificial intelligence optimization [12] of the engine and its components in determining factors at various blending levels. Computational fluid dynamics has also gained a lot of relevance in engine performance and emissions with biodiesel blending [13, 14].

Still, in a quest for better outcomes, tertiary diesel blending was explored with biodiesel, alcohol, compressed natural gas, kerosene, oil and water [15–20]. Alcohol sourced from biomass [21, 22] or synthesized are the major fuels explored so far on biodiesel-diesel blending despite solubility challenges. Kwanchareon *et al.* [23] and He *et al.* [24] focused on ethanol solubility and homogeneous blending techniques to determine the best blending ratio of 10% to 30% ethanol volume. Building on this, Xiaoyan *et al.* [25] expanded the base to ethanol-selective catalytic reduction through Ag/Al₂O₃ introduction as a catalyst to improve emission reduction and an extension of alumina, zinc and other nanoparticle technologies were also employed [26–30]. With compressed natural gas inclusion, the engine is operated as a dual-fuel injection mode, occasionally in a compressed natural gas flow rate variation [19] mode.

A post-tertiary blending level report has also been published where vegetable oil is added to the ethanol-biodiesel-diesel blend [31]. Acetone-butanol-ethanol containing water [32] and nanoparticles added to ethanol-isopropanol-diesel-biodiesel [26] has been tested for better engine performance and reduced emissions.

1.1. Blended Kerosene Combustion

Kerosene combustion has gained more ground in aviation jet and aircraft engines in contrails and cruise states [33–37] than in internal combustion engine applications. Even in the pursuit of alternative fuels towards emission reduction in industrial gas turbines and the aforementioned engines, kerosene blending has received considerable attention. Physico-chemical properties contrast kerosene, its blends and diesel [38] stimulated hydrocarbon compounds which are studied using chromatography and magnetic resonance approaches to determine the abundance of n-alkanes and aromatics in the fuels [39]. Combustion studies involving kerosene-diesel and alcohol-diesel blends in flow combustors have shown that the hydrogen-carbon ratio influences the combustion process, emissions and efficiencies through blending, although kerosene blending increases heat transfer rate [40]. Ignition, oxidation, kinetics and detailed review of kerosene



combustion [41] are the predominant focal points of these studies. The combustor kerosene-diesel study based on 25% diesel blend of kerosene showed improved combustion efficiency and better fuel consumption than K_{100} or D_{100} due to high volatility of K_{100} [42, 43] predominantly under lean mixture conditions. Effects of K_{100} concentration on flame temperature and velocity of a turbulent premixed mixture [44] revealed that K_{100} blending for lean and rich combustion flame does not have linear dependent relations with the abovementioned characteristics.

With compression ignition engine, pure mustard oil-kerosene [10] and diesel-kerosene [45] binary blends were tested while tertiary blending with biodiesel variation [46] was conducted, in the interest of determining better engine performance with lower emissions.

In this experimental study, the effect of varying kerosene concentrations in a tertiary biodiesel-diesel blended fuel on engine performance and emissions was investigated. The study focuses on reducing emissions during the operation of a high-speed fuelled-engine. In this regard, the K_{100} variations with blends between 5% to 25% at constant volume concentration of B_{100} and D_{100} as well as in parent fuel under engine load or torque variable during combustion were developed. This K_{100} variation effect investigation was the gap-bridging goal proving this study's originality.

2. Materials and methods

Samples of conventional diesel and kerosene were obtained from a local filling station in Ile-Ife, Osun state, Nigeria. Biodiesel produced from *Khaya senegalensis* oil seeds of 2.2 kg net weight was obtained from the Forest Research Institute in Ibadan, Nigeria. Plantain peels were used to produce a catalyst for transesterification. The specifications and setup of the test rig fitted with hydraulic dynamometer are shown in Table 1 and Figure 1, respectively. The gas analyser specifications for emission quantification are listed in Table 2. Soxhlet extraction was used to obtain oil extracts using n-hexane (BDH Chemicals Ltd, England) as an organic and cost-effective extracting solvent.

After the oil extracts were obtained by Soxhlet method, biodiesel production was optimized using a central composite design experiment (Table 3) using Design-Expert (Stat-Ease Inc., Minneapolis, MN, USA) as detailed in the work by Onojowho *et al.* [47]. Analytical grade reagents used in the experiments were obtained from BDH Chemicals Ltd., Poole, England, and GFS Chemicals, Inc., Columbus, Ohio. The blending and characterization of samples were performed using the ASTM and AOAC standard testing methods, as presented in Table 4. Mixing proportions of 10% constant B_{100} , 65%, 75%, and 85% of D_{100} to 5%, 15%, and 25% K_{100} by volume and IKA C-MAG HS 4 magnetic stirrer were used to



Table 1. Engine test rig specification.

Parameters		Values
1.	Dynamometer const. head	1 bar @ 5 L/min (min.)
2.	Dynamometer max. power and speed rating	7.5 kW and 7000 rpm
3.	Engine cylinder, capacity and stroke	Single, 0.232 L and 4
4.	Engine max. rating	3.5 kW @ 3600 rpm
5.	Bore, stroke and crank radius	69 /62 /31 mm
6.	Connecting rod length	104 mm
7.	Compression ratio	22:1
8.	Thermocouple	Type-k
9.	Engine model	TD212, TQ182785-002

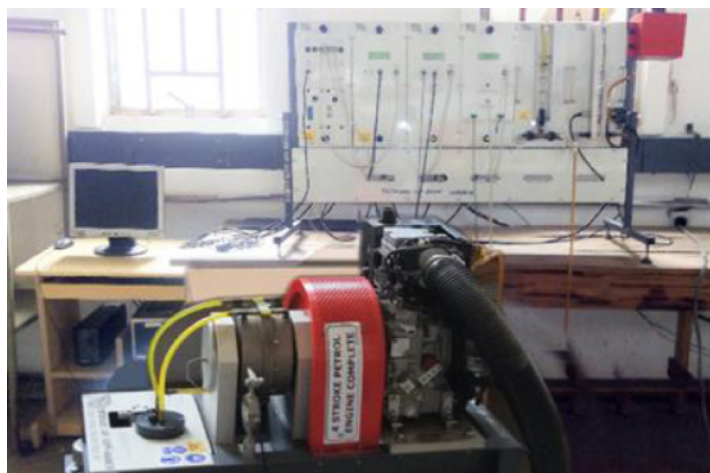


Figure 1. Engine test rig experimental setup.

prepare three homogenous blend samples in addition to B₁₀₀ and D₁₀₀. During combustion, the gas analyser determined the combustion efficiency of all five blends from the captured flue gas properties. The engine operating torque variables for this study are 1.5 N m, 2.45 N m, 3.4 N m, 4.35 N m, and 5.3 N m.

2.1. Theory/calculations of engine performance parameters

The complexity of combustion involves turbulence, species transportation, and ignition phenomena, among others. However, a few fundamentals are mentioned, in this study such as touching the CI engine performance quantification. Equation (1) represents the basic stoichiometric combustion with 100% complete combustion. For rich or excess air combustion with 100(x - 1) % at an equivalent ratio $x \geq 1$, see Equation (2). The mass flow rates of air and fuel are expressed in Equations (3) and

Table 2. Testo 330-2LL gas analyser specification.

S/N	Parameters	Measuring range	Accuracy	Resolution
1.	Combustion Efficiency	0–120%	–	0.1%
2.	Flue gas loss	–20–99.9	–	0.1%
3.	Temperature	–40–1200 °C	±0.5 °C @ 0–100 °C ±0.5% of range reading	0.1 °C @ –40–999 °C 1 °C @ 1000–1600 °C
4.	Oxygen	0–21 vol. %	±0.2 vol. %	0.1 vol. %
5.	CO ₂	0–10000 ppm	±5% of reading @ 40.1–300 ppm ±50 ppm +2% @ 0–5000 ppm ±100 ppm +3% of reading @ 5001–10000 ppm	1 ppm

Table 3. Biodiesel optimized production plan.

Factor	Unit	Coded factor levels				
		–α	–1	0	+1	+α
Methanol: Oil (X ₁)	vol./vol.	4.7574	6	9	12	13.2426
Catalyst loading (X ₂)	wt%	1.0858	1.5	2.5	3.5	3.9142
Reaction time (X ₃)	min	11.7157	20	40	60	68.2843

(4), respectively. Furthermore, Equation (5) calculates the brake power, and the brake mean effective pressure (BMEP) is given in Equation (6), as determined by the built-in pressure pickup to TecQuipment software.

The engine fuel consumption rate, brake thermal efficiency, brake volumetric efficiency, heat of combustion, and combustion efficiency were determined using Equations (7)–(11), respectively: The land size cultivation requirement for *Khaya senegalensis* seed biodiesel production in hectares was obtained using Equation (12). The overall uncertainty of all parameters was estimated from the uncertainty of each parameter at the measurement determination, as presented in Equation (13) using the root sum squared approach [48].

$$C_n H_m O_c + (n + m/4)(O_2 + 3.76N_2) \rightarrow nCO_2 + m/2H_2O + (n + m/4)3.76N_2 \quad (1)$$

$$C_n H_m O_c + x(n + m/4)(O_2 + 3.76N_2) \rightarrow nCO_2 + m/2H_2O + (x - 1)(n + m/4)O_2 + 3.76x(n + m/4)N_2 \quad (2)$$

$$m_a = \pi d^2 / 4 C_d \sqrt{2P_a \Delta P / RT_a} \quad (3)$$

$$m_f = \rho_f C_n A_n (\Delta \theta / 360 N) \sqrt{2\rho_f \Delta P / 1000} \quad (4)$$

$$P = 2\pi N \tau / 60 \quad (5)$$

$$BMEP = 60P(L/2)/0.1NV_c \quad (6)$$



$$\text{BSFC} = 3600 \times 60m_f / 2\pi N\tau \quad (7)$$

$$\text{BTE} = 100P/H_f \quad (8)$$

$$\text{BVE} = 30m_a T_a R / P_a V_c N \quad (9)$$

$$H_f = m_f \text{LHV} \times 10^6 \quad (10)$$

$$\eta = 100 - [(T_f - T_a)(A/O_{21} - O_m) + B] - K \quad (11)$$

$$\text{Land size} = \frac{\text{Total seed quantity produced (tonnes)}}{\text{Yield per hectare (tonnes)}} \quad (12)$$

$$U_r = \left[\sum_{i=1}^1 (u_{\bar{x}_i} \partial r / \partial x_{i_{x=\bar{x}}})^2 \right]^{1/2} \quad (13)$$

$$U_r = [\text{BMEP}^2 + \text{BSFC}^2 + \text{Power}^2 + \text{BVE}^2 + \text{BTE}^2 + A - F^2 + H_f^2 + \text{CO}_2^2 + m_f^2 + \eta^2 + \text{O}_2^2 + \text{EQT}^2]^{1/2} = \pm 491.32. \quad (14)$$

3. Results and discussion

3.1. Engine performance parameters

3.1.1. Fuel mass flow rate

As shown in Table 4, the density and viscosity of the BDK blend decreases with increasing kerosene content. This improves the low-temperature fluidity of the fuel blends. This kinematic viscosity decrease with respect to the kerosene proportion has been reported in literature [49]. This implies that BDK blends provide smaller droplets at atomization than the B₁₀₀ and D₁₀₀ blends. These influencing properties of kerosene were significant for the blend flow rate, as shown in Figure 2. A quasi-steady flow rate was almost maintained with BDK₅ up to 75% loading condition of the engine, whereas BDK₁₅ maintained a good fuel flow rate that could even replace D₁₀₀ in performance. This presents a quasilinear response and presents a potentially significant effect on blend injector atomization for proper combustion.

3.1.2. Air-fuel ratio

Diesel engines have typical air-fuel ratios ranging between 18–70 for combustion mixtures [50]. Mixtures below ratios of 18 are too rich, while above 70 are too lean. A general trend of an overly rich air-fuel mixture was observed for all blends. That is, all fuel blends were mixed with excess air as detailed in literature [46]. These performances were good translations of the corresponding fuel flow rates. The higher the torque or load, the lower is the mixture ratio, as shown in Figure 3, which implies that the engine has a high capacity to ensure complete combustion. BDK₅ was maintained next to the richest mixture from the idling position until



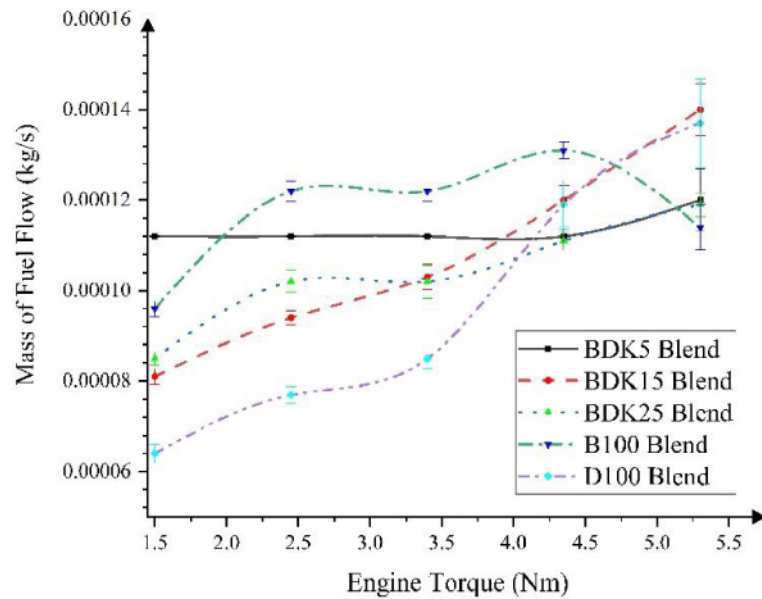


Figure 2. Fuel flow rating of blends.

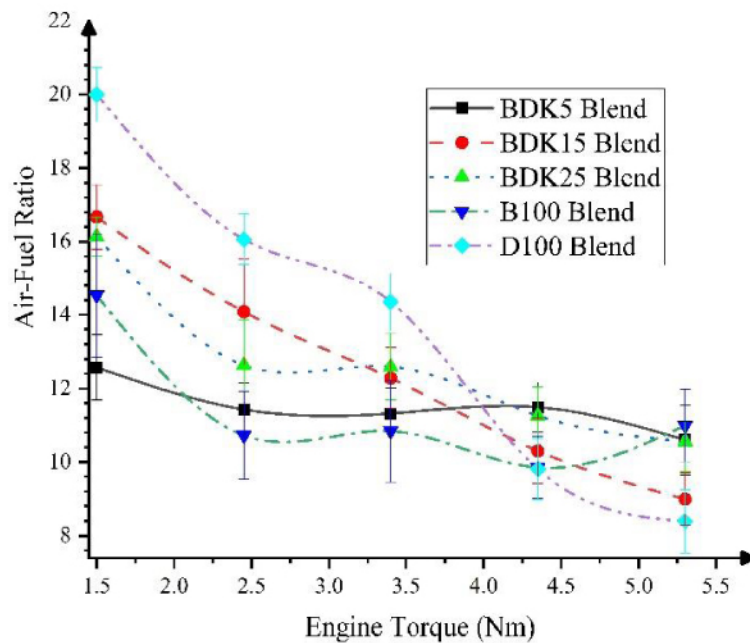


Figure 3. Combustion mixture quality of fuel blends.

approximately 50% loading after B_{100} . A linear relation response is presented by the BDK_{15} blend, which yields the best blending mixture above 75% engine load.

3.1.3. Brake mean effective pressure (BMEP)

There was a progressive increase in the in-cylinder combustion pressure with engine torque for all blends, as shown in Figure 4. The experiment shows that BDK_{15} is



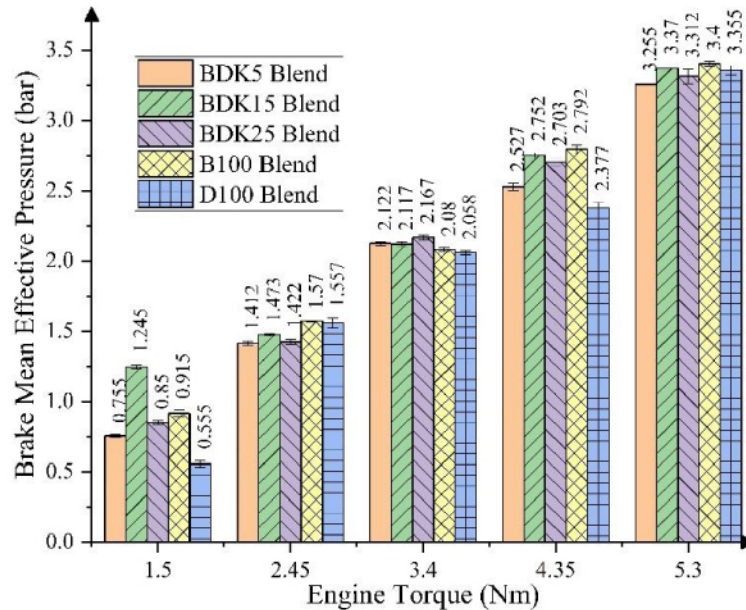


Figure 4. Engine combustion pressure of blends.

generally stable and comparable to D_{100} . Engine performances with blends are closely related to each other, similar to the diesel-biodiesel-alcohol-nanoparticle trend result [26], and with 70 to 100% load performance of Jamrozik *et al.* [51].

3.1.4. Brake specific fuel consumption (BSFC)

Kerosene increments in blends significantly reduce the fuel consumption rate lower than B_{100} and is similar to D_{100} in this study. The results obtained by Kumar *et al.* [46] showed similarities with results presented in Figure 5, however, the variation of diesel and constant biodiesel percentage proportions could be responsible for the significant reduction in consumption than D_{100} at 50% to 75% load (5.3 N m) of the engine performance. This can be maintained even at full load.

3.1.5. Power development

The brake power developed from D_{100} exhibited the lowest performance. The B_{100} blend is the highest, whereas BDK_{15} is steadily reliable owing to its quasi-linear nature (Figure 6). This performance can easily be deduced from the fact that D_{100} possesses the lowest caloric value. The tertiary blend of water in diesel-alcohol in a CI engine also exhibited an increasing power trend, but with engine speed [16] while a decreasing power trend was presented by Akar [54].



Table 4. Properties of fuel blends.

Properties	Samples					ASTM				
	BDK ₅	BDK ₁₅	BDK ₂₅	B ₁₀₀	D ₁₀₀	K ₁₀₀	Method	B _{6–20} (D 7467)	B ₁₀₀ (D 6751)	K ₁₀₀
Grades	B ₁₀ D ₈₅ K ₅	B ₁₀ D ₇₅ K ₁₅	B ₁₀ D ₆₅ K ₂₅	B ₁₀₀	D ₁₀₀	1-k*	–	–	–	–
Density @ 15 °C (kg/m ²)	860.2	857.1	852.6	874	852.8	790	AOAC ⁺	–	–	780–810
Kinematic viscosity @ 40° (mm ² /s)	4.417	3.920	3.683	5.862	4.635	1.612	D 445	1.9–4.1	1.9–6	–
Saponification value (mg KOH/g)	29.73	65.08	60.59	121.74	108.78	90.13	AOAC ⁺	–	–	–
Iodine value (g iodine/100 g)	181.21	171.06	179.5	62.78	182.11	265.14	AOAC ⁺	–	–	–
Cetane number	189.114	91.68	95.99	77.01	55.5	47	D 613	40 min	47 min	–
Sulphur	–	–	–	–	0.082‡	–	D 5453	0.0015 max. (% mass)	0.0015 max. (% mass)	0.04%w
Cloud point (°C)	–0.3	–0.5	–0.8	8.3	0.2	< –1.5	D 2500	Report	Report	–40
Pour point (°C)	< –1.5	< –1.5	< –1.5	2	< –1.5	< –1.5	D 97–96a	–	–	–47
Smoke point (°C)	62	60	56	89.3	67	35	D 93	–	–	–
Flash point (°C)	85	84	82	124	83	50	D 93	125 min	130 min	37–65
Calorific value- LHV (MJ/kg)	39.99	40.32	40.65	37.44	35.65	41.25	AOAC ⁺	–	–	43.25

* Grade is based on Sulphur proportion present in K₁₀₀ Grade 2-k is 0.3 wt%. While grade S₁₅ (0.0015% mass) and S₅₀₀ (0.05% mass) are for biodiesel blends and D₁₀₀. ⁺ AOAC [52] (Association of Official Analytical Chemists). [‡] See reference [53].



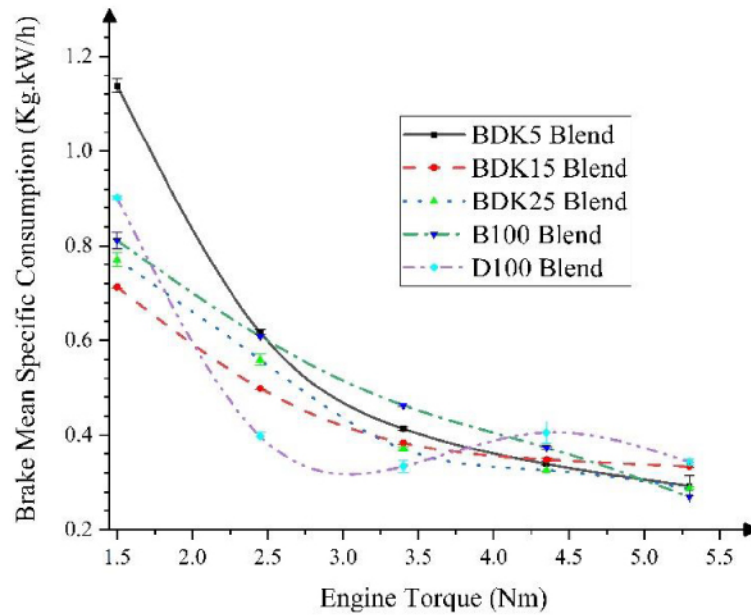


Figure 5. Engine fuel consumption index.

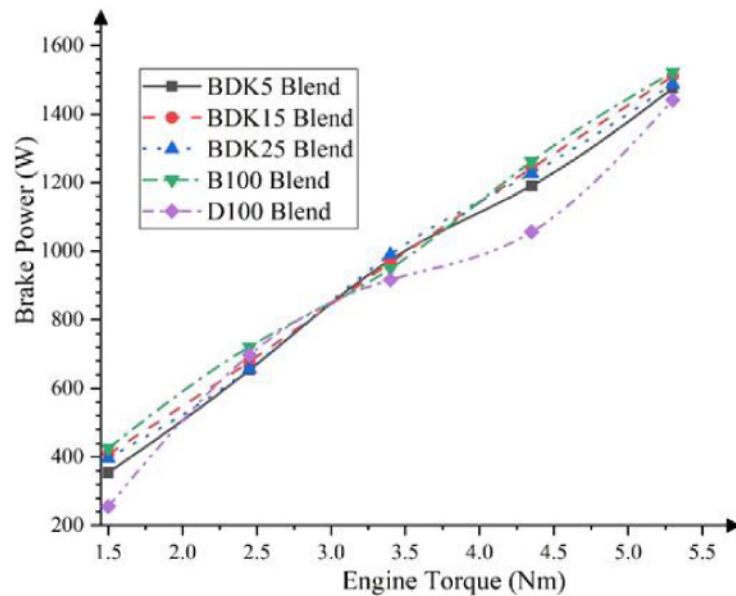


Figure 6. Power developed from blended mixtures.

3.1.6. Efficiencies: brake volumetric efficiency (BVE), brake thermal efficiency (BTE)

The actual air intake to stoichiometric air volume ratio describes the volumetric efficiency of an engine in mixture quality indexing. The experiment showed that the brake volumetric efficiency had an inverse relationship with engine torque. That is, the BVE decreased in response to increasing torque. The engine air-fuel mixing



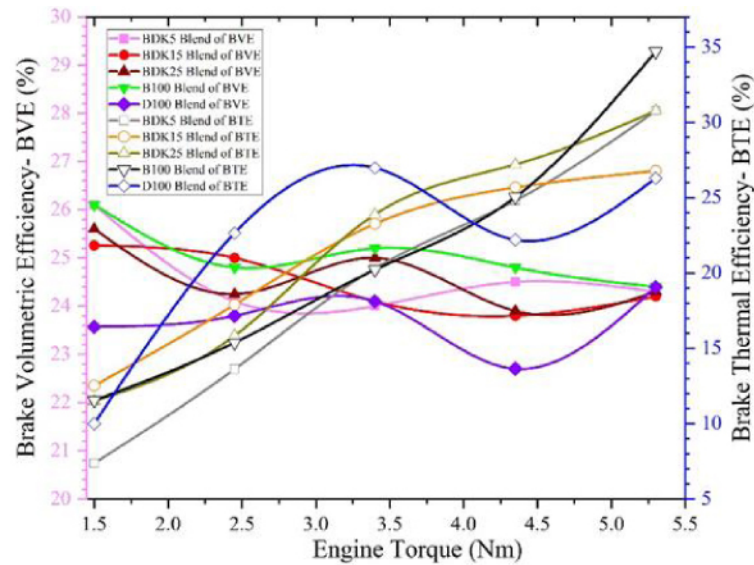


Figure 7. Engine efficiencies at loading capacities.

ability is represented as BVE in Figure 7, where B_{100} gives the best response with BDK_{25} as runner-up in volumetric efficiency. Blends BDK_5 and BDK_{15} performed better than blend D_{100} . Hence, introducing 5–25% of K_{100} is significant in the rich air-fuel mixture, enhancing better combustion with reference to D_{100} . BTE is another level of engine performance efficiency that indexes engine energy conversion strength. Thermal energy liberated from fuel is usually converted into mechanical energy. Figure 7 shows the positive BTE presentation of the rising torque, i.e., the engine converts more thermal energy in response to increasing load demand. Lower heat of combustion is expected to promote a high BTE for higher mechanical energy harvesting. From idling, D_{100} appears to perform better; however, the K_{100} tertiary blending effect reveals the potential of BDK_5 , BDK_{15} , and BDK_{25} to perform better than D_{100} from 4.35 N m (50% loading) and above. Conversion from BDK_{25} offers the highest BTE with BDK_5 following a contributive effect of K_{100} proportion. The BVE and BTE results conform to those of existing works [16, 46, 51].

3.2. Combustion performance

3.2.1. Heat of combustion

The fuel flow rate and calorific value are the determining variables. All blends followed an increasing trend of liberated heat at torque increase during combustion. However, BDK_5 was an exception to steady heat liberation irrespective of torque or load variation owing to the constant fuel flow rate. Furthermore, as shown in Figure 8, BDK_{15} had the most stable and highest heat liberation, although this was not visible in the energy conversion process. The lowest value was observed for D_{100} , which has risen from its low calorific value. It can be observed that H_f and power



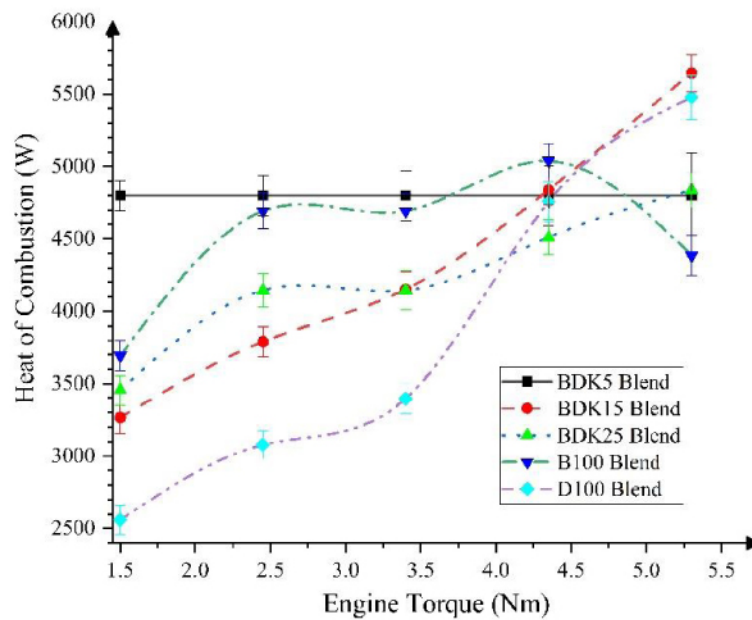


Figure 8. Energy liberation profile of fuel blends.

uncertainties are high, as shown in Table 5, owing to the Watt unit presentation. The individual uncertainties obtained in the measurement and instrumentation process for each parameter indicate a high accuracy level to meet the design accuracy for each parameter. The plots indicate the standard error of each measurement.

Table 5. Uncertainty and sensitivity of equipment's and measurements.

S/N	Parameters	Accuracy	Uncertainty
1.	BMEP	—	± 0.0377 bar
2.	BSFC	—	± 0.002546 kg·kW/h
3.	Power	$\pm 3\%$	± 21.9151 W
4.	BVE	—	$\pm 0.8793\%$
5.	BTE	—	$\pm 0.4592\%$
6.	A-F	± 0.5	± 0.4548
7.	m_f	—	$\pm 2.1915 \times 10^{-5}$ kg/s
8.	H_f	—	± 490.8238 W
9.	η	—	$\pm 1.489\%$
10.	CO ₂	$+3\%$	$\pm 0.2438\%$
11.	O ₂	± 0.2 vol. %	$\pm 0.1223\%$
12.	EQT	$\pm 0.5\%$	± 0.7971 °C



3.2.2. Combustion efficiency

Flue gas analysis of the engine combustion process of Figure 9 defines how well fuel burns. It can be seen that all fuel blends burned well above 70% and were higher than diesel-kerosene blends [42]. The increasing effect of K_{100} proportion in blends translated into the same order of increasing efficiency well above B_{100} as engine torque rises. D_{100} had the best combustion efficiency than others. Perhaps the air-fuel mixture of other blends was excessively richer based on the ratio presented above.

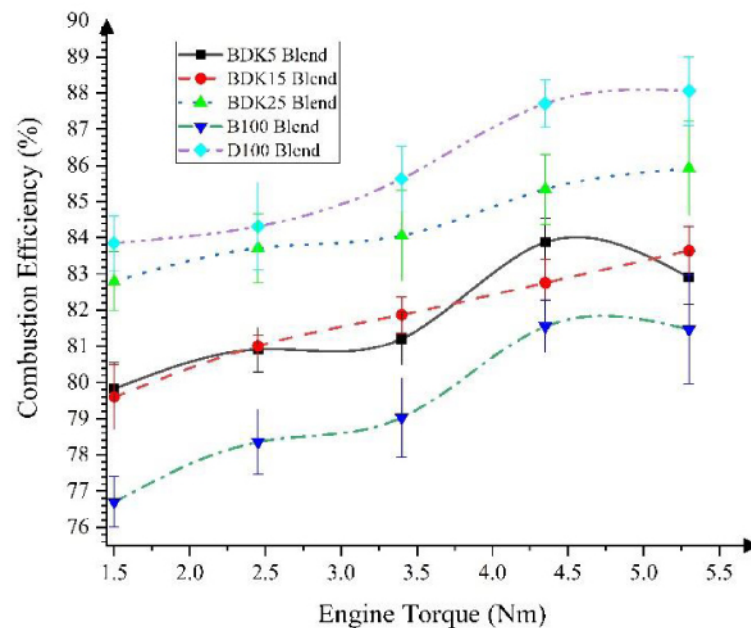


Figure 9. Engine burning efficiency of blended mixtures.

3.3. Emission

3.3.1. Carbon (IV) oxide (CO_2)

Figure 10(a) depicts that the percentage of CO_2 emissions increases as engine torque increases, as previously stated [16, 26, 54], including the relationship with speed increase. Emissions from B_{100} from no-load state to 75% load maintained emissions within a minimum range of 6.06 to 12.44% lower than D_{100} . Tertiary blends vividly followed suit, with BDK_5 at 22.882%, BDK_{15} at 26.375%, and BDK_{25} at 15.375% lower than D_{100} . Therefore, the BDK_{15} profile shows the lowest CO_2 emissions. These effects on emissions can be traced back to lower carbon atoms existing in K_{100} [42, 44].



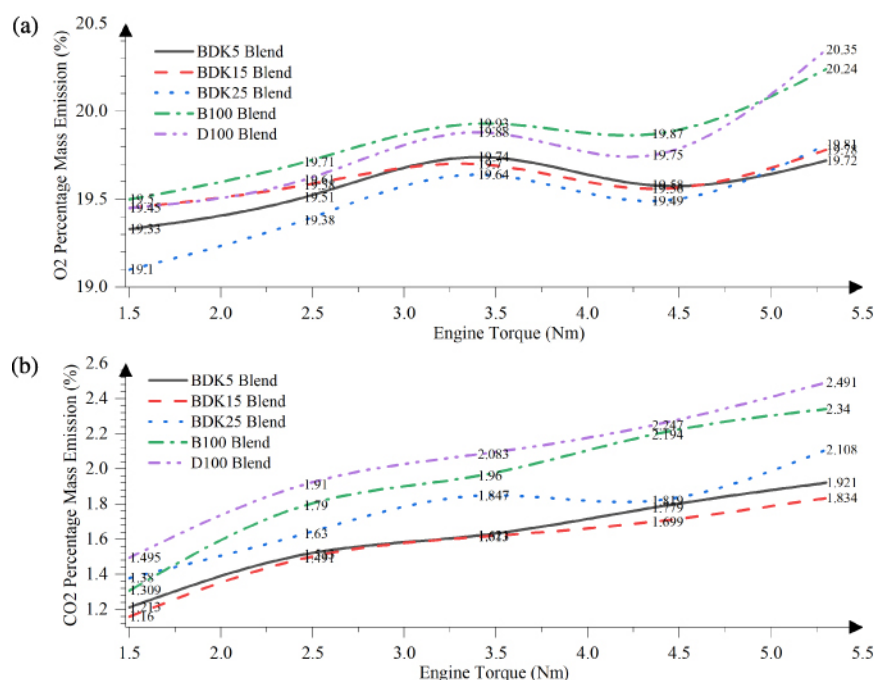


Figure 10. Engine fuel combustion gas emissions.

3.3.2. Oxygen (O₂)

The oxygen emissions shown in Figure 10(b) indicate that the experiment validated the excessively rich stoichiometric combustion with all blends resulting in complete combustion. The oxygen content in flue gas increased with increasing torque or load, in contrast to ethanol tertiary blend [49] and waste oil binary blend [55] which showed incomplete combustion. The oxygen constituent of B₁₀₀ favored its emission.

3.3.3. Exhaust gas temperature

Flue gas temperature loss in this study points out that BDK₁₅ blends to be the maximum among BDK₅, BDK₂₅ and B₁₀₀ between closer range temperature losses, whereas D₁₀₀ has the least. This energy loss from the heat of combustion with blends in Figure 11 is undesirable, even though it has similarities with others [46], including tertiary blend studies with ethanol [18]. Systems with exhaust recirculation [56] may help minimize this loss in utilization.

3.4. Land use and resource depletion footprint

Assuming maturity, spacing, climate and soil quality factors are favourable for the 50.15% oil yield and 75.6–97.93% biodiesel yield reported in Onojowho *et al.* [47], it may be necessary to mass produce biodiesel, since it is a non-edible substrate. The land use requirement for this is estimated by Equation (12) to be 10 hectares for 10



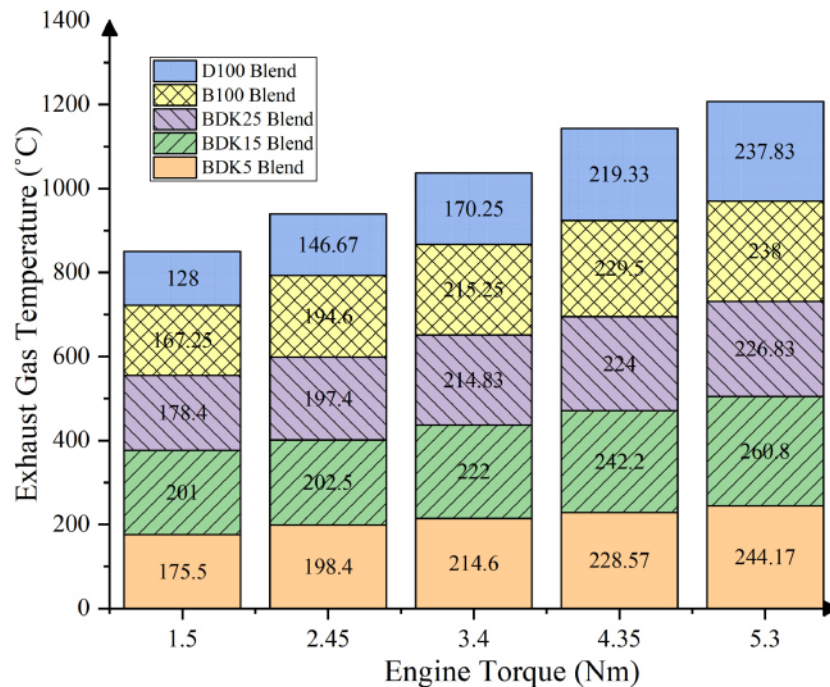


Figure 11. Engine energy conversion losses potential from fuel combustion emissions.

tonnes of seeds. The impact of converting a natural land habitat into a mahogany plantation is socioeconomically profitable. Food production can even gain promotion simultaneously if appropriate mixed cropping is practised with the plantation. The risk of water and energy consumption at cultivation and processing, together with chemical usage could be well managed for minimal environmental impact.

4. Conclusions

The study results revealed the significant effect of the varying proportion of kerosene on biodiesel-diesel blending for compression ignition engine. Significant levels are observed vividly in the K_{100} , increasing proportion of engine performance, combustion and emissions powered by blends.

- The A–F ratio decreased with increasing torque and increasing K_{100} blend proportions. The BDK₁₅ blend presented the best mixture ratio compared to others, including D₁₀₀. It also appears to be the most economical blend for fuel consumption, harmonizing with BSFC results as torque increases.
- The average in-cylinder pressure required to keep maximum and stable power output performance of the engine is obtained with BDK₁₅ than D₁₀₀ and other blends.



- Fuel heat energy conversion to mechanical energy usually indexed with BTE is directly and indirectly proportional to power and combustion heat released respectively. BDK₁₅ and BDK₅ blends are readily converted compared with D₁₀₀. Power developed with all blends was increasing with corresponding increasing load.
- Observations on all blends reveal that increasing load translated into combustion efficiency improvement. However, the efficiency of D₁₀₀ is better than all other blends. It has a 2.14% difference between D₁₀₀ and the highest efficiency of the runner-up blend. CO₂ emission of BDK₁₅ is least compared to all other blends, with an advantage of at least 0.087% difference at 5.3 N m. Further studies could be based on reducing the mature age of mahogany seeds and also on the fuel stability of a tertiary blend with nanoparticle effect.

Author's contribution

Onojowho, Elijah: Conceptualization, Writing- original draft, **Sadjere, Godwin:** Writing- review and editing.

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Ethical statement

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Data availability statement

Source data is not available for this article.

Conflict of interest

The authors declare no conflict of interest.

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Nomenclature

BMEP	Brake mean effective pressure (bar)	A	Constant 0.68	m_f	Fuel flow rate (kg/s)
BSFC	Brake specific fuel consumption (Kg·kW/h)	B	Constant 0.007	m_a	Air flow rate (kg/s)
BTE	Brake thermal efficiency (%)	η	Combustion efficiency (%)	ΔP	Pressure differential
BVE	Brake volumetric efficiency (%)	τ	Engine torque (N m)	$\Delta \theta$	Injection crank angle
LHV	Lower heating value (MJ/kg)	N	Engine speed (rpm)	A_n	Orifice area (m^2)
ICE	Internal combustion engine	R	Gas constant (287 J·K/kg)	ρ_f	Fuel density (kg/m ³)
CI	Compression ignition	P	Power (W)	C_d	Flow factor
BDK ₅	5% volume kerosene blend sample	K	Dew point heat condensate	d	Orifice diameter (m)
BDK ₁₅	15% volume kerosene blend sample	T_a	Ambient temperature (°C)	π	3.142 constant
BDK ₂₅	25% volume kerosene blend sample	P_a	Ambient pressure (bar)	L	Piston stroke (mm)
B_{100}	Pure biodiesel sample	V_c	Engine capacity (cc)	O_m	Stack flue measure O ₂
D_{100}	Pure diesel sample	T_f	Stack flue temperature (°C)	H_f	Heat of combustion (W)
K_{100}	Pure kerosene sample	O_{21}	21% atmospheric O ₂ value	A–F	Air to fuel ratio
EQT	Exhaust gas temperature	U_r	Overall Uncertainty		

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