JEE Journal of Ecological Engineering

Journal of Ecological Engineering 2023, 24(12), 293–303
https://doi.org/10.12911/22998993/173564
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Received: 2023.09.20
Accepted: 2023.10.21
Published: 2023.11.04

Modeling of Waste Vegetable Oil Biodiesel for Tractor Engine Utilization

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ABSTRACT

Biodiesel is regarded as a clean fuel alternative to fossil diesel fuel for fewer pollutant emissions of internal combustion engines. The biodiesel type can be made from waste frying oil, thus it has to be done right. Waste vegetable oil can be provided for free or at a low cost by restaurants and food processors that often use frying oils. Animal fat is also available for free or for a nominal fee from grocery stores, restaurants, and butchers who use lots of fats in their cooking. The methyl ester of oleic acid methyl ester (OAME) biodiesel was produced from used vegetable oil using the transesterification process in order to compare the performance of the 67 kW KUBOTA tractor diesel engine when utilizing OAME and fossil diesel. OAME biofuel was used without being mixed. The engine's reliability metrics and important indicators, including the brake torque, indicated power, brake-specific fuel consumption (BSFC) and burn duration, were identified. Optimal implementation was met by fossil diesel and the tested characteristics were very close. The OAME biofuel performs better in terms of volumetric efficiency and duration of combustion than the conventional diesel. The decision to choose a specific biofuel that is produced from a particular source so largely hinges on its availability and economic feasibility wherever it is used.

Keywords: waste oil, transesterification, fossil diesel, biodiesel, simulation model, engine

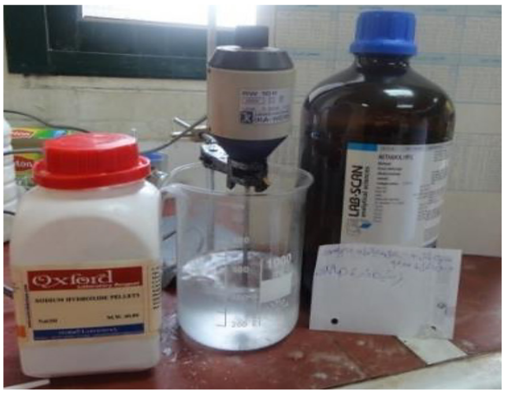
INTRODUCTION

For usage in diesel motors, biodiesel has emerged as a substitute for diesel fuel. Scientists are looking for renewable alternative fuels to oil, due to the near end of oil and the instability of its prices (Mahalingam et al., 2018). As compared to utilizing 100% biodiesel (MOBD100), the experimental study employing a blend of pentanol and biodiesel at all motor loads (MOBD90P10 and MOBD80P20) revealed a minor improvement in brake thermal efficiency (0.2–0.4%) and minimal brake-specific fuel usage (0.6–1.1%). Checks on the diesel motor demonstrate that adding Ag_2O nanoparticles to palm oil mill effluent (POME) improved the ignition characteristics during testing using a naturally aspirated, with the engine having two cylinders (Pandian et al. 2018). The trials involved the immediate injection of clear diesel and four different forms of biodiesel into four cylinders of a diesel engine. According to the test created by (Zhang et al. 2018), the biodiesel fuel's kinematic viscosity and ignition delay time (IDT) were significant determinants in the burning process. The performance and emissions indicators of a single-cylinder compression-ignition DI motor fed with mixed biodiesel at various motor speeds were examined in exploratory testing by (Norhafana et al. 2019). Results indicated that while diesel fuel had superior exhaust gas emissions than biodiesel, the latter had poorer performance metrics. A static inductive diesel engine was used to evaluate four fossil diesel fuels, biodiesel (cashew shell methyl ester), and a blend of 100% and 20% CBD by volume (CBD90DME10 and CBD80DME20). The CBD80DME20 should be utilized as an alternative fuel for diesel engines, according to (Devarajan et al. 2019). Tomato seed oil has been characterized as a potential source of biodiesel that contains about 24% oil and can make up to 72% of tomato waste by weight. Tomato seeds are one of the mission resources to use biodiesel fuel. Tomato seed oil (TSO) was used by (Karami, Rasul, and Khan 2020) since it is a cheap source that should not be eaten as an exporter of biodiesel. The composition of a novel biofuel made from a blend of rapeseed oil biodiesel and mahua biodiesel (RM) in the same amount. (Saravanan et al. 2020) carried out a test using a combination of biofuel and diesel in different quantities and they confirmed that the combination of the BL20 perform better.(Khan 2020) indicated the possibility of using biodiesel as an alternative fuel to run engines. This is because it is not used as food, it is rather to be used as a feedstock through the process of esterification with acids to convert Sterculia foetida kernel oil to methyl ester (Bhanu Teja et al. 2021). Biodiesel is manufactured from strange woody plant sources including a viscous brown liquid that is obtained and treated in wood pulping plants known as raw tall oil. With water emulsification of the biodiesel blended into a diesel engine, the research produced impressive results for producing exotic-source biodiesel that comply with the American Society for Testing and Materials (ASTM) standard for biodiesel and produces lower emissions of NOx as well as aldehydes than diesel fuel. Smoke emission and hydrocarbon have also been significantly decreased for blends compared to diesel. Burning experience tests confirmed that with the rising percentage of unsaturated fatty acids, the blend has a more burning time than diesel fuel. Fish oil and animal fats are used in the synthesis of biodiesel, according to (Shepel et al. 2021), and and therefore are considered raw materials. In terms of renewable energy, beef tallow biodiesel is the most sustainable biodiesel, because it reduces exhaust gases. In fact, it is comparable to fossil fuels in energy efficiency and it has a better value than the biodiesel understudy. Animal byproducts including pig fat and chicken skin were studied by (Srinivasan and Jambulingam 2018) in order to produce and characterize biodiesel. The created biodiesel is then mixed with petroleum fuel in varying volume amounts. In accordance with ASTM standards, several thermochemical characteristics were assessed. The results of the investigation demonstrated that the biodiesel blend's properties were mostly within the range permitted by ASTM regulations. The transesterification method was used by (Nguyen, Khoa, and Tuan 2021) to turn Egyptian sheep fat oil, which is thought to be bad for the environment and is being pushed out, into biodiesel. Diesel and lamb biodiesel are mixed in two different amounts, 10% and 20% respectively, and these amounts are denoted as B10 and B20. Benefit assessments of Cl motors running on clear diesel fuel, B10, and B20 mixtures are made while subjected to varying loads. According to (Emaish et al. 2021), the transesterification method was utilized to prepare a waste frying oil biodiesel sample that was mixed with diesel fuel to improve diesel engine efficiency at various fuel mixes and lessen the environmental effect of gas emissions. The experimental findings showed that fuel blend percentage and engine load percentage had a substantial impact on all analyzed aspects. The optimal engine loading range for maximizing engine efficiency and minimizing specific fuel consumption and gas emissions was between 25 and 75%. The interior works demonstrate the efforts done by researchers to modify biodiesel engines for higher performance, progressive combustion, and better emissions by straightening components and engine systems. This investigation's goal is to investigate whether using biofuels in a stock diesel engine could be an option. By conducting this study, a numerical simulation was performed to examine the performance and emissions parameters of biodiesel produced from frying oil (oleic acid methyl ester), which was transformed into biodiesel by implementing the transesterification process and using methanol alcohol and sodium hydroxide as reaction aids. At various speeds and loads, pure biodiesel and fossil diesel fuels are compared in terms of engine performance indicators, such as fuel consumption, torque, power, and exhaust gas emissions.

BIOFUEL PRODUCTION

Biodiesel is one of the most important types of fuel all of the world. It is created from discarded frying oil, algae, and many types of oil from vegetables, including edible and inedible ones, as well as animal and fish fats. Alternatively to being mixed with mineral diesel, biofuels may also be used directly in diesel engines. Though, some problems may be encountered, especially in winter, because the fuel becomes more viscous at low temperatures and according to the raw materials produced from it, it is used in Europe as a mixture of biodiesel and fossil diesel by 5%. Biofuel was used to reduce exhaust gas emissions very significantly because it contains high levels of hydrogen and oxygen compared to fossil diesel, which increases the rate of combustion and reduces greenhouse gases, knowing that biodiesel is considered oxidizing because it contains small amounts of carbon. The use of pure biodiesel fuel increases nitrogen oxide emissions. Biofuel production has evolved since its discovery and has passed through several generations: First-generation biofuels is a fuel produced from plants grown on arable land that convert the sugar, starch or oil content of the crop into biodiesel or ethanol by applying the transesterification process or yeast fermentation. The second biofuel generation is the fuel produced from cellulosic biomass or wood or agricultural waste. The raw materials used in this generation are by-products of the grown crop in marginal lands. The second-generation raw materials include bagasse, straw, grasses, waste frying oil, and solid waste. The third biofuel generation is algae biofuel, which is cultivated in different forms and harvested in several ways. Algae is growing in tanks or ponds and can be grown using saline and wastewater. The algae biofuel is biodegradable and has a high flash point. It is low harmless to the environment and degrades faster than other biodiesel types but becomes non-spillable in low temperatures.

In 2017, the efforts to produce biofuels from algae were changed due to advanced applications and economic considerations. This generation includes solar and electric fuels. Electric fuel is storing electrical energy in liquids and gases' chemical bonds. According to (Emaish et al. 2021), (Gomaa, Mohamed, and Al-Aseebee 2020), and (Gomaa, Mohamed, and Al-Aseebee 2020), biodiesel



(a) Mixing of catalyst and methanol

(b) Biodiesel (top) from glycerol (bottom) which is drained off

(c) Washing process

(d) Pure biodiesel from the transesterification process

Figure 1. Phases of the transesterification process

was produced in the present investigation using vegetable oil that is used as frying oil where the oil is heated to 80 °C and then it should be left to reach the feedback temperature (55 °C). 400 ml of methanol alcohol and 13 g of sodium hydroxide are created and thoroughly combined. It is measured and the combination is added when the oil reaches a temperature of 55 °C. The combination is added, the oil is stirred for an hour, and then it is put in a separating funnel for a further two hours. Washing the mixture with water that is 70 °C in temperature is required. During the washing procedure, which should be repeated several times until a pH of 6-7 is attained, water should be used in a ratio of 20% water to fuel. The leftover material must then be heated to a temperature of between 80 and 100 °C to evaporate the water.

ENGINE TEST ARRANGEMENT

The tractor utilized for the test, a Kubota M1-100S-DT 73.6 kW four-cylinder farm tractor, was operated in line with the tractor's specifications, which are shown in Table 1, and at varied engine speeds, loads, and indirect injection levels. Figure 2 illustrates how the engine and the dynamic exhaust from the gas test are directly related. The testing setup was set up with varying loads at rotational speeds of 297, 2234, and 2850 rpm that approximated the maximum fuel load and the average load capacity of the diesel engine. The fuel consumption, torque, power, and exhaust gases were all measured using an hydraulic brake stationary dynamometer and an exhaust gas analyzer meter.

Both petroleum diesel and OAME biofuel were tested under the same conditions where the same engine, measurements devices and operating conditions were used for both tests. This fact ensures that the comparison between the two fuels is done on the same basis.

COMPUTATIONAL METHOD

Numerical method

To simulate the tested engine in this work, a gas dynamics solver utilizing a staggered grid finite-volume discretization approach is used. Models of ducts, intake and exhaust valves, cylinders, and crankshafts are included. While the other engine components are based on analytical models with these components treated as boundary conditions, the model of the vents is based on the numerical finite volume approach. Previous research was done on the specifics of the engine simulation model (Al-Aseebee et al. 2023). In the implicit formulation of the flow conservation equations for the ducts model, continuity, momentum, and enthalpy are the variables that are resolved.

Performance parameters

Using a hydraulic brake stationary dynamometer, the torque and speeds of rotation were obtained in laboratory conditions for the tractor control take-off shaft computing. At diverse speeds, the power take-off shaft is calculated using the following equation:

$$P_{TO} = \frac{2\pi N\tau}{C} \quad (1)$$

where: N – the rotational speed in rpm, τ – the torque in N·m, C – a constant.

The power take-off shaft P_{TO} is about 88.9% of the engine brake power P_B, according to testing. For an utilized tractor, the current study making an

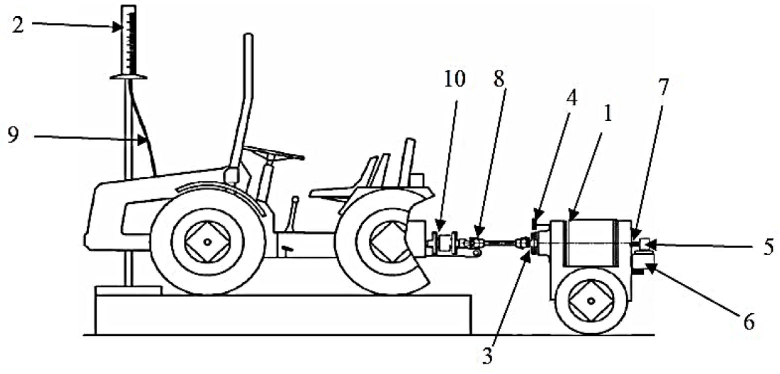


Figure 2. Plan of the tractor engine

Table 1. Tractor Kubota M1-100S-DT engine features

Model characteristic	Unit	Kubota M_1-100S-DT
E_S	RPM	2600
E_P	KW	73.6
B	mm	100
S	mm	120
N_C	Cylinder	4
C_R	–	21.8:1
T_E	–	liquid cooled diesel, turbocharged, 4 stroke, indirect injection
Technical details of stationary dynamometer		
M	–	NEB600
S_N	–	CD6190C5
R	KW	300 at 540rpm
	KW	600 at 1000 rpm
C_T	N.m	4338 N.m

Note: E_S, E_P, B, S, N_C, C_R, T_E, M, S_N, R, and C_T are rated engine speed, engine power, bore, stroke, number of cylinders, compression ratio, type of engine, model, serial number, range and capacity of torque, respectively.

allowance for the P_{TO} ability as 88% of the engine brake power P_B. The proportion between the engine and P_{TO} speed of shaft is equal to 4.08.

The brake mean effective pressure p_{BME} can be defined as the subtraction of the friction pressure of the engine from the net indicated pressure is related to the brake power P_B as follows:

$$p_{BME} = 120\frac{P_B}{LAN} \tag{2}$$

where: L – the piston displacement in m, A – the piston cross-sectional area in m^2, N – the engine rotation speed in rpm, n – the number of engine cylinders, 120 – the constant referring to the 4-stroke engine and the unit conversion.

The p_{BME} rise, which varies directly in relation to the load, boosts the motor's performance. The volumetric efficiency η_V, commonly abbreviated v, is the proportion of the real observed volume of intake air V_a in m^3 pulled into the cylinder/engine to the predicted volume of the engine/cylinder V_d in m^3, during the intake engine cycle:

$$\eta_v = \frac{V_a}{V_d} \tag{3}$$

The volumetric efficiency of an engine increases with its air intake capacity. Since the amount of fuel in indirect fuel injection engines, primarily those used in gasoline engines, is very tiny (1:14.7) in comparison to the amount of air, it is possible to ignore the fuel mass when calculating volumetric effectiveness. A function of air mass m_a in kg and air density ρ_a in kg.m^{-3} may be used to determine the actual itake air volume:

$$V_a = \frac{m_a}{\rho_a} \tag{4}$$

Thus, the volumetric efficiency can be written as follows:

$$\eta_v = \frac{m_a}{\rho_a \, V_d} \tag{5}$$

The engine performance stands typically measures the intake air mass flow rate in kg·s^{-1} rather than air mass in kg. For this reason, the volumetric efficiency calculation using the air mass flow rate may be expressed as follows:

$$\eta_v = \frac{m_a N 60}{n_r} \tag{6}$$

where: m_a – the air mass given in kg, N – the engine speed measured in rpm, n – the number of crankshaft spins needed for completing a 4-stroke engine cycle.

Thus, the volumetric efficiency is written as follows:

$$\eta_v = \frac{m_a n_r}{\rho_a V_d N_e} \tag{7}$$

When the intake manifold's temperature T_a and pressure p_a are determined, the intake air density may be calculated as follows:

$$\rho_a = \frac{p_a}{R_a \, T_a} \tag{8}$$

where: R_a is the gas constant of dry air equal to 286.9 J·kg^{-1}·K^{-1}.

The brake thermal efficiency is obtained by dividing the effective ability from the motor by the amount of energy given to the motor:

$$\eta = \frac{P_B 3600}{V_{FC} \rho_f H_V} \tag{9}$$

where: P_B – the brake power in kW, ρ_f – the density of the tested fuel in kg·L^{-1}, V_{FC} – the fuel consumption rate, given in L·h^{-1}, H_V – the heat value of the tested fuel in kJ·kg^{-1}.

The mass of air to the mass of fuel in the combustion process is known as the air fuel ratio, or R_{AF}. As a result, the subsequent actions needed to be decided:

- The mass flow rate of air,
- The system monitoring equipment fuel mass flow rate.

The R_{AF} is calculated as the product of steps 1 and 2. The volumetric effectiveness is defined as the ratio of the number of cylinders swept volume V_S to the air volume input V_a determined at the intake air conditions:

$$\eta_v = \frac{V_a}{V_s} \tag{10}$$

According to Figure 3, the volumetric effectiveness may be calculated at every operating position on the engine operating map.

For all tested fuels, the observed fuel consumption rate was divided by 4 cylinders to determine the fuel consumption rate per single cylinder C_{VF} in L·min^{-1}/cycle. The following formula is used to determine the mass flow rate of each fuel under consideration:

$$R_{AF} = \frac{m_a}{m_f} \tag{11}$$

where: m_a and m_f respectively are the air and fuel masses in kg.

A comparison of the ASTM D6751 fuel characteristics between fossil diesel and OAME can be seen in Table 2. The volumetric fuel consumption rate in L/h for every load is calculated using the formula below:

$$C_{VF} = 3.6 \frac{V}{t} \tag{12}$$

where: t – the test's running duration in seconds, V – the amount of fuel utilized in the glass bulb in cubic meters.

The brake power P_B in kW at the same loading circumstances, which are represented by the various levels of engine speed ranging from 408.33 rpm to 2858.31 rpm, is used to determine the fuel consumption rate C_{BSP} provided in L/h:

$$C_{BSP} = \frac{C_{VF}}{P_B} \tag{13}$$

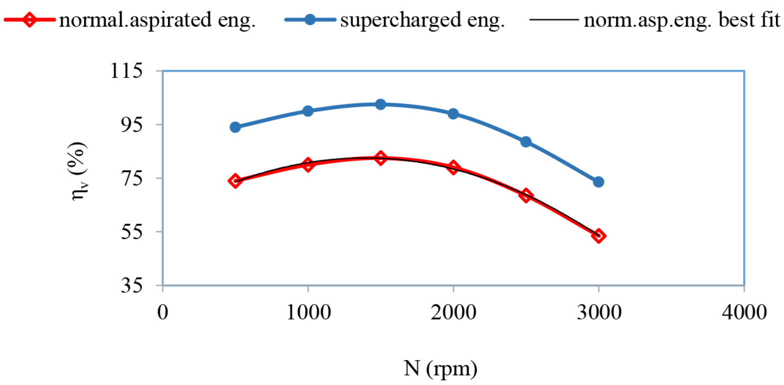


Figure 3. Variation of the volumetric efficiency (Gupta 2012)

Table 2. Results of operational characteristics of tested biofuels, according to ASTM D6751

Features	OAME	Diesel
Flash point (°C)	140	130 to 170
Pour point (°C)	- 6	10 to -15
Calorific value (MJ·kg^{-1})	42.3	40 to 43
Ash content (% by weight)	Nil	Nil
Density at 40 °C (g·ml^{-1})	0.835	0.845 to 0.820
Dynamic viscosity at 40 °C, (MPa·s)	8.966	2 to 2.5
Viscosity at 40 °C (mm^2·s^{-1})	10.6	5.5 to 24

Equation (14) can be used to calculate volumetric efficiency.

$$\eta_v = 100\frac{2m^0{}_{air}}{\rho_{air}V_s N} \quad (14)$$

where: η_v – the volumetric efficiency given in %, ρ_{air} – the density of air in kg·m^{-3}, N – the engine speed in rpm, m^0 – the mass flow rate of the fuel, kg/hr and V_s – the swept volume, m^3.

RESULTS AND DISCUSSION

Indicated and brake torque

Figure 4 compares the indicated torque distribution for the fossil diesel fuel and the oleic acid methyl ester biodiesel. According to these results, adding more biodiesel to the fuel mixture causes the reported torque values to slightly drop. The maximum indicated torque value of OAME fuel is 345.801 N.m at a speed of 1200 rpm. The maximum indicated torque value for fossil diesel is 387.736 N.m at a speed of 1600 rpm. The lowest indicated torque value for fossil diesel at a maximum speed engine is 199.061 N.m, whereas the lowest indicated torque value for OAME fuel is 154.768 N.m at a maximum speed engine. This drop can be explained given that, as compared to pure diesel fuel, the enthalpy of the fuel mixture reduces as the percentage of biodiesel increases. The increased lubrication and high oxygen content of biodiesel may lessen friction loss, improving effective brake torque while making up for the fuel's diminished heating value (Zweiri and Seneviratne 2007).

In Figure 5, it has been observed that the experimental and simulated results of the brake torque have the same direction trend for the two tested fuels. In fact, for both fuel cases, the brake torque reaches its maximum values at mid speed range about 1600 rpm. Then, it declines with the rise of the rotational speed. This observation can be explained by the reduction in the volumetric efficiency as less air amount can be swallowed by the engine regarding the lesser time available as the speed increases. The simulation results also show that the brake torque within the pure diesel case is slightly higher compared to the oleic acid methyl ester case.

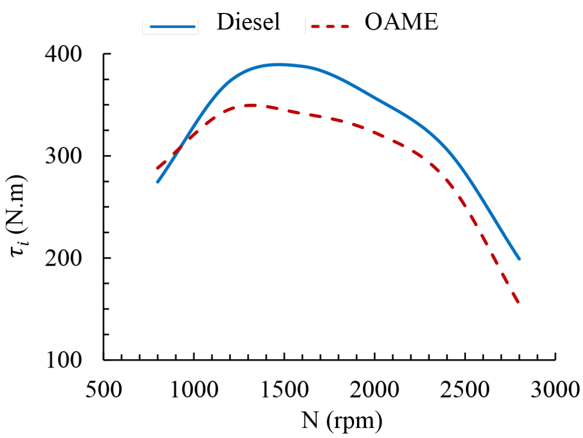


Figure 4. Cycle-average indicated torque distribution

Indicated and brake power

Figure 6 compares the indicated power between fossil diesel fuel and acid methyl ester fuels. From these tests, the indicated brake power values increase with increasing the engine speed. Globally speaking, fossil diesel fuel has greater indicated brake power ratings than OAME fuel. Since fossil fuel is utilized, the highest value of the suggested break power is 76.95 kW and is recorded at an engine speed of 2400 rpm, while the lowest value of 22.99 kW is recorded at an engine speed of 800 rpm. At an engine speed of 2400 rpm, the indicated break power for OAME fuels reaches its greatest value of 69.388 kW, while at 800 rpm, it reaches its lowest value of 24.13 kW. The actual and simulated brake power for two different fuel types – fossil diesel and OAME fuels are shown in Figure 7. These findings show that the experimental and simulated levels of brake power for the two studied fuels exhibit the same tendency (Geok et al. 2009). Furthermore, the simulation results show that the diesel brake

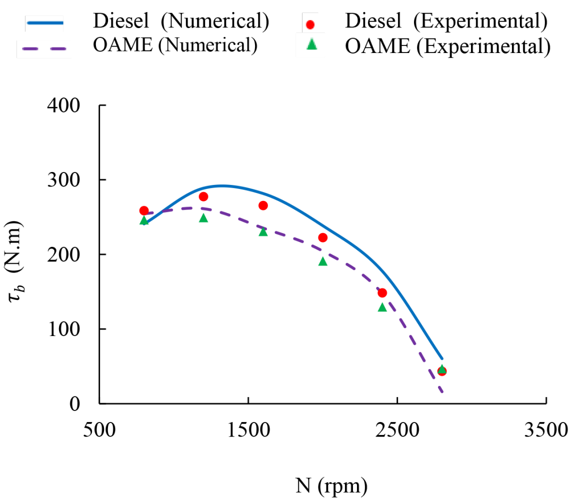


Figure 5. Cycle-average brake torque distribution

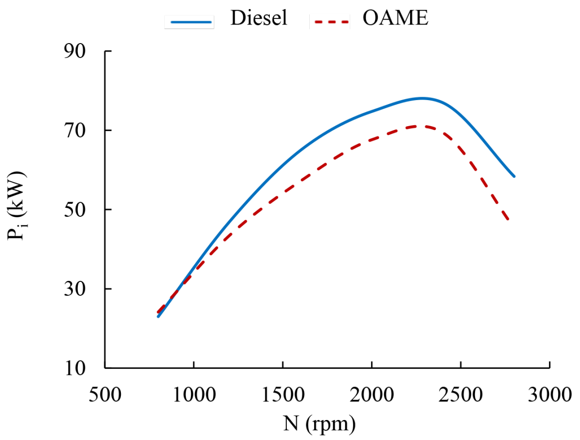


Figure 6. Cycle-average indicated power distribution

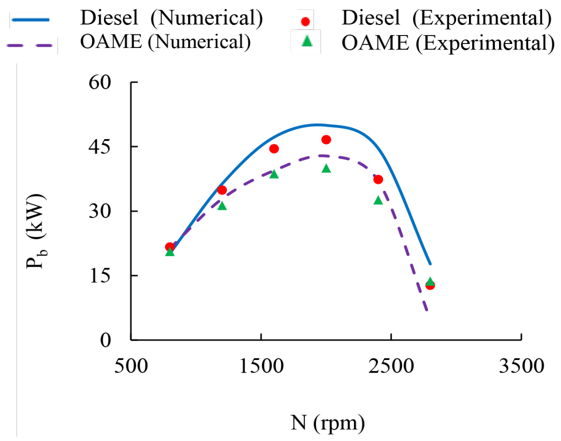


Figure 7. Cycle-average brake power distribution

power is slightly more than that of OAME. The possibility of using OAME fuel in place of diesel fuel is therefore suggested.

Fuel consumption of brake specific

Figure 8 shows the distribution of brake-specific fuel consumption (BSFC) for fossil diesel and OAME fuels at different speeds, ranging from 800 to 2400 rpm.

The maximum BSFC value OAME fuel is 757,833 $g.k.W.h^{-1}$ at a speed of 2400 rpm. The maximum BSFC value for fossil diesel is 549.061 $g.k.W.h^{-1}$ when tested at a speed of 800 rpm. The lowest value for fossil diesel at a speed engine of 1600 rpm is 409.374 $g.k.W.h^{-1}$, whereas the lowest value for OAME fuel is 530.006 $g.k.W.h^{-1}$ at a speed of 1200 rpm of the engine.

The outcomes show that, at a speed engine of 1600 rpm, the BSFC of the OAME fuel is higher than that of the fossil diesel fuel. Since it has a lower heating value per unit mass than fossil diesel fuel, when the engine starts to operate on OAME fuel, more fuel must be added to the fuel tank. (McCarthy, Rasul, and Moazzem 2011).

Volumetric efficiency

Figure 9 shows the variation of volumetric efficiency of fossil diesel fuel and OAME fuel mixtures versus engine speed. The volumetric efficiency depends on the cylinder temperature. The increased rate of heat release has an impact on the mass of incoming air and its temperature.

This is a result of the elevated cylinder temperature due to an increased NOx emissions when using biodiesel-diesel blends. This resulted in a reduction in the intake air temperature, subsequently decreasing the amount of air drawn into the engine, which in turn lowered the volumetric efficiency. The changes in latent heat of vaporization and thermal properties led to a decrease

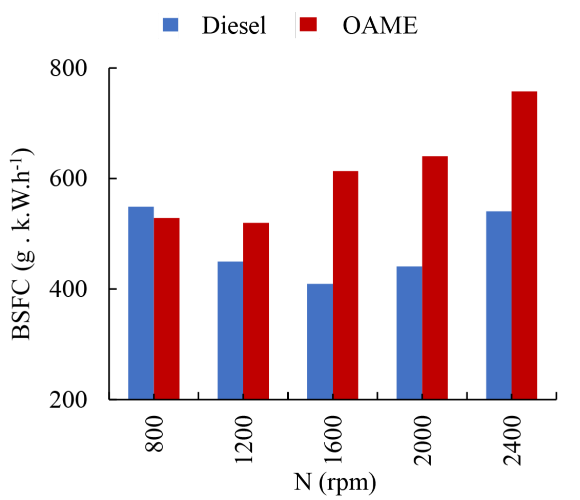


Figure 8. brake specific fuel consumption (BSFC) for varied rotational speeds

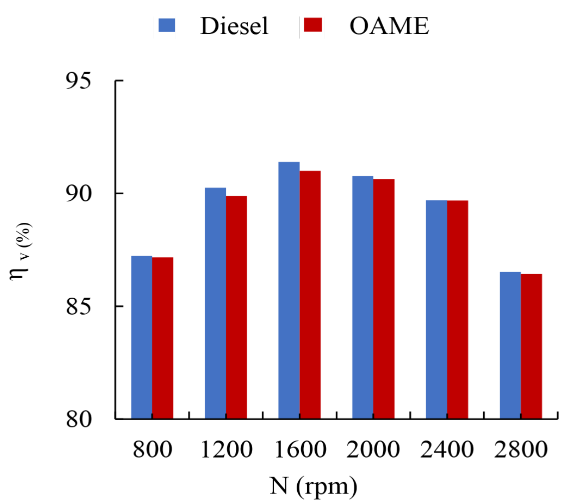


Figure 9. Volumetric efficiency for varied rotational speeds

in the inlet air temperature of biodiesel blends when compared to diesel fuel. The highest value registered of the volumetric efficiency was fossil diesel fuel is 91.40% and is recorded at an engine speed of 1600 rpm, while the lowest value was fossil diesel fuel at 86.52% recorded at an engine speed of 2800 rpm. The volumetric efficiency for OAME fuels reaches its greatest value of 90.99% at an engine speed of 1600 rpm, while at a speed of 2800 rpm, it reaches its lowest value of 86.43% (Gad, El-Shafay, and Hashish 2021).

Burn duration

Figure 10 shows the variation of burn duration 50% for the fossil diesel fuel and the OAME biofuel versus engine rotational speed. The burn duration 50% refers to the time it takes for the combustion process to reach 50% of the total combustion. This parameter can be important for understanding combustion efficiency and engine performance. The burn duration can be influenced by several factors in the context of internal combustion engines such as speed, load and fuel properties. High engine speeds often result in shorter burn durations, while low speeds can lead to longer burn durations. Also, high loads may result in longer burn durations as more fuel and air are combusted to produce power. The properties of the fuel such as its octane rating can impact burn duration. Fuels with different ignition characteristics can lead to variations in the burn duration. The fossil diesel fuel log with the highest value registered of the burn duration 50% of 35.804% recorded at an engine speed of 800 rpm. The lowest value of the burn duration 50% of 19.074% was recorded for the fossil diesel fuel under an engine speed of 2800 rpm. The burn duration 50% for the OAME biofuel reaches its greatest value of 31.109 % at an engine speed of 800 rpm, while at a speed of 2800 rpm, it reaches its lowest value of 19.074 %. Figure 11 depicts the variation of burn duration 90% of the fossil diesel fuel and the OAME biofuel versus the engine speed. As the burn duration 90% refers to the time it takes for the combustion process to reach 90% completion, this parameter is used to measure the timing and completeness of the combustion process in an internal combustion engine.

A shorter burn duration 90% generally indicates more efficient and controlled combustion, which is desirable for engine performance and emissions control. Factors that affect burn duration 90% are similar to those that affect burn duration 50%, as discussed earlier, and include engine speed, compression ratio, air-fuel mixture, ignition timing, and other engine parameters. The fossil diesel fuel log with the highest value registered of the burn duration 90% of 76.49% recorded at an engine speed of 800 rpm, while the lowest value was of 36.92% for the fossil diesel fuel recorded under an engine speed of 2800 rpm. The burn duration 90% for the OAME biofuel reaches its greatest value of 73.17 % under an engine speed of 800 rpm, while at a speed of 2800 rpm, it reaches its lowest value of 36.92 %. From these results and particularly at low engine speeds, the OAME biofuel performs better in terms of burn duration compared to the conventional petroleum diesel. This observation can be explained by the fact that the OAME biofuel contains more oxygen than the conventional petroleum diesel leading to a lower burn duration and more combustion completeness.

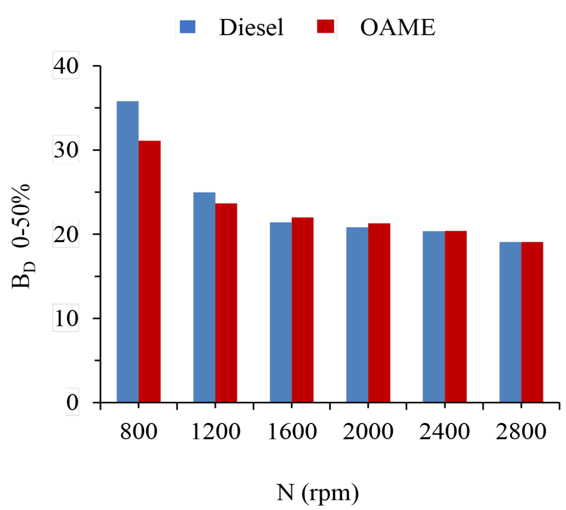


Figure 10. Cycle-average burn duration 50% for varied rotational speed

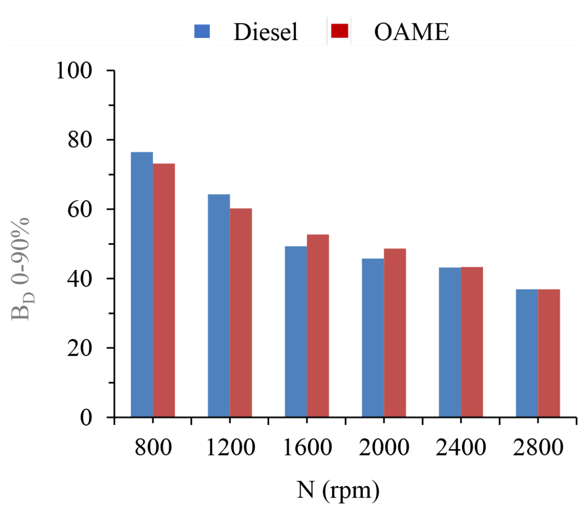


Figure 11. Cycle-average burn duration 90% for varied rotational speed

CONCLUSIONS

In the present paper, a comparison of the oleic acid methyl ester (OAME) biofuel based on waste oil to the conventional petroleum diesel in terms of engine performances was conducted through experimental and numerical studies. The experimental implementation makes use of a tractor diesel engine rated at 67 kW made by the KUBOTA manufacturer. Based on the obtained results, it has been found the following key points:

1. The volumetric efficiency of the engine is significantly decreased as the rotational speed increased for both the conventional petroleum diesel and the OAME biofuel. This fact is due to the less air amount that can be swallowed by the engine regarding the minor available time to swallow as the rotational speed increases.
2. For both the petroleum diesel and the OAME biofuel, the maximum brake torque is recorded at an engine rotational speed near to 1500 rpm. As the engine speed further increases, there is a decline in the brake torque followed by the significant reduction in the volumetric efficiency.
3. The maximum powers for both fuel cases are attained at engine speeds between 1900 rpm and 2000 rpm. The tested OAME fuel had a brake power level of 45 kW, whereas the tested petroleum diesel fuel had a maximum brake power of 53 kW.
4. Both petroleum diesel and OAME biofuel exhibit a similar trend of the brake-specific fuel consumption. The minimum brake fuel consumption was recorded at mid-range of the speed between 1300 rpm and 2000 rpm.
5. The findings also demonstrated that the petroleum diesel has a lower brake-specific fuel consumption than the OAME biofuel.
6. The burn duration is noticeably reduced with an increasing rotational speed for both the conventional petroleum diesel and the OAME fuels. In fact, this recorded decline in the burn duration is mainly lies on the reduction in the volumetric efficiency of the engine as less amount of oxygen will be available in the cylinder which influences the combustion completeness.
7. The burn duration of the OAME biofuel is smaller compared to the conventional petroleum diesel. The difference in the burn duration is more prominent at low engine speeds. This observation is due to the fact that the OAME biofuel molecule is richer in oxygen compared to the conventional diesel molecule and then, the OAME biofuel is expected to undergo more complete combustion than diesel which was confirmed by the current study.

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