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Performance and Exhaust Emission of a Diesel Engine Using Crude Palm Oil as Fuel Extender

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Abstract

This paper investigates into the use of Crude Palm Oil (CPO) as a fuel extender for diesel engines. Three mixtures of CPO with ordinary diesel fuel (OD) were used in a direct-injection, stationary Perkins diesel engine. The three CPO-OD mixtures contained 25%, 50% and 75% of CPO by volume. In order to reduce the viscosity of the CPO-OD mixtures they were preheated to about 60^{o}C before injection. The paper compares the performance of these CPO-OD mixture, with that of ordinary diesel fuel, at a fixed throttle opening but variable engine speed of 1000-3000 rpm. The results obtained show two distinct trends when the speed is below or above an intermediate speed of about 2000 rpm. At speeds lower than 1800 rpm, all three CPO-OD mixtures gave higher torque and power output, but their brake specific fuel consumption (BSFC) was also higher compared to that of OD. The exhaust-gas analysis show that CPO-OD mixtures gave higher emission of CO, and lower emissions of NOx, compared to that of diesel fuel. The performance of CPO-OD mixtures became comparable to that of diesel fuel at speeds above 2000 rpm. Although the brake power of CPO-OD mixtures was less than that of OD in the high-speed range, their BSFC improves. These results indicate that when the engine is fuelled by a CPO-OD mixture, the fuel-pump and injection system tend to deliver more fuel than needed at low speeds, but less fuel at high speeds.

Introduction

Fossil fuels have been the main source of energy for various activities in our daily life for more than a hundred years. However, the natural reserves of these fuels are diminishing rapidly while the worldwide demand for energy is increasing. The need for new energy sources and the damage to the environment caused by the release of large quantities of green-house gases and pollutants into the air, waters and the ground, has encouraged intensified search for alternative sources of energy that are both renewable and environment-friendly. In this respect, vegetable oils have been identified as a potential source of energy that can substitute fossil fuels in certain applications.

Vegetable oils have low sulphur content, and fast and almost complete biological decomposition. Since plants consume CO_2 for photosynthesis, the combustion of vegetable oils leads to a near balanced CO_2 cycle and a favourable reduction in green-house effects. These factors make vegetable oils ideal alternative fuels, particularly for diesel engines that are widely used in the agriculture and transport sectors. The much higher flash-point of vegetable oils, as compared to liquefied fossil fuels, makes them safer to store and transport. However, apart from their higher price compared to ordinary diesel fuel, the use of vegetable oils to fuel modern diesel engines is hindered by a number of technical problems [1-4]. When crude vegetable oils are used as fuels, they form carbon deposit on the combustion chamber and cause injector choking and valve sticking. They also tend to dilute the engine's lubrication oil and usually have problems with cold-starting. These problems, which mainly arise because of the higher viscosity and lower volatility of vegetable oils compared to ordinary diesel fuel, and their tendency to polymerise at normal atmospheric condition, can be avoided in indirect-injection engines that have pre-combustion chambers [5]. However, modern diesel engines apply direct injection and, therefore, the problems have to be solved by a suitable modification either to the engine, or to the vegetable oil itself to make it acquire properties similar to those of ordinary diesel fuel. The problems associated with vegetable oils could be overcome by some diesel engines that have been designed specially for vegetable oils. One of such engines is the Elsbett engine [6] in which necessary modifications have been made to the combustion camber as well as the fuel's transport and injection systems. The viscosity of crude palm oil (CPO) could be reduced by heating the oil in the fuel transport system.

This paper investigates into the use of CPO as a fuel extender for diesel engines. Three mixtures of CPO with ordinary diesel fuel (OD) in different proportions of volume are used in this investigation on a direct-injection, stationary Perkins diesel engine. A comparison between CPO-OD mixture and only OD is made regarding the engine performance and exhaust.

Experimental Set Up and Testing Procedure

Experimental set-up

A stationary Perkins diesel engine model 4-108V was used in the experiment. The naturally aspirated, water cooled, direct injection engine has the following specifications: 4 cylinders, 4-stroke, bore 79.5 mm, stroke 88.9 mm, compression ratio 22, and 32 kW rated power output at 4000 rpm. Instruments were available to measure the engine's torque, fuel consumption, and speed. The load on the engine was varied via a Heenan Froude hydraulic dynamometer. A portable gas analyzer, Lancom 6500, was used to measure the exhaust-gas emissions. To form a CPO-diesel mixture, the two fuels were mixed manually in the required (volume) proportion in a special tank, which is heated using a water bath with an electric-resistance heater. A thermocouple and a simple electric circuit control the electric power to the heater. Proper mixing of the CPO-OD mixtures was achieved by using a small air blower placed into the mixing tank. A three-way valve switches the fuel supply from the diesel-fuel tank to the CPO-diesel mixing tank when required. The original copper pipe that carries the diesel fuel to the engine was replaced by a rubber hose with a slightly larger diameter to ease the flow of CPO mixtures and reduce their cooling due to heat loss to the surroundings. No additional heating was provided after the fuel filter. Fig. 1 shows a schematic diagram of the testing apparatus.

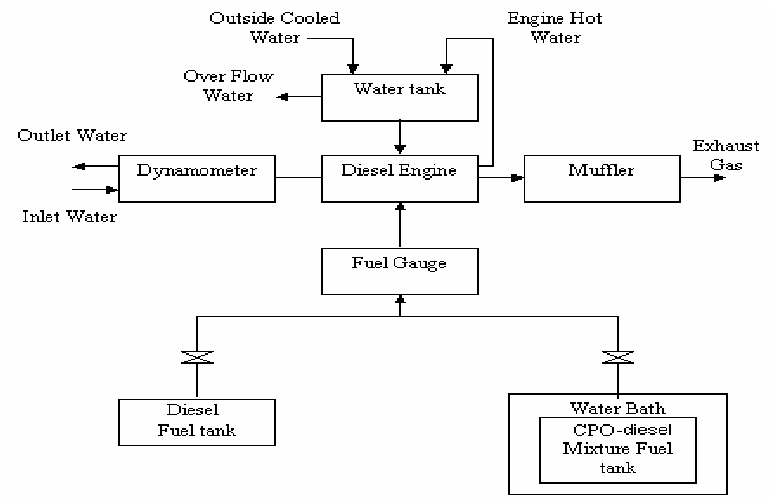


Fig. 1 Schematic layout of the test facilities

Testing procedure

The engine test procedure was repeated three times each test. The test was conducted under different operating conditions for both conventional and COP-diesel mixture fuel. The engine's performance at part-load was tested at speeds in the range of 1000-3000 rpm. At the start of each test, the throttle opening would be adjusted to give a speed of 3000 rpm at a minimum dynamometer-load. The load would then be increased gradually as the engine speed dropped, in steps of 200 rpm, down to 1000 rpm. For each speed, the torque applied and rate of fuel consumption were recorded. When testing CPO-OD mixtures, the engine would be started with diesel fuel and left to warm up for about 15 minutes before switching to the CPO mixture to be tested. Also, at the end of the test the engine would be left to run on diesel fuel for some time before been shut down in order to flush the fuelling system from any CPO residues. CPO-diesel mixtures with 25%, 50%, and 75% of CPO by volume were considered. Three series of complete tests were conducted for each fuel mixture and the mean of the three tests was taken. The CPO used in the tests was provided by the Malaysian Palm Oil Board (PORIM formerly) and the diesel fuel was obtained from commercial stations. The viscosity of CPO at normal atmospheric conditions is ten times higher than that of ordinary diesel fuel. According to de Almeida CPO has to be heated to a temperature of 100^{o}C in order to reduce its viscosity to a comparable value to that of OD [4]. Bari reported that a temperature of 60^{o}C ensured a smooth flow of CPO in the fuel-delivery system. In the present study the temperature was limited to about 60^{o}C to avoid flashing of the diesel fuel in the fuel mixture [3]. Moreover, the CPO-OD mixtures were heated in a water bath rather than directly by the heating element.

The Fuel's Power Performance

Fuel consumption and gross energy input

Fig. 2a shows the measured fuel consumption vs. engine speed for ordinary diesel and CPO-OD blends. The figure shows that the maximum fuel consumption for all fuels lied in the range of 1800-2000 rpm. Below this range, the consumption of CPO-OD mixtures was greater than that of pure diesel fuel and increased as the percentage of CPO in the mixture increased. However, for speeds above 2000 rpm the flow rate of CPO-OD mixtures was less than that of diesel fuel and decreased as the percentage of CPO in the mixture increased. Clearly, the rate of fuel delivery to the engine at a particular speed was affected by the density and viscosity of the fuel.

In order to compare the performance of the different fuels on the basis of their energy content rather than their volume or mass flow rate, the gross energy input should be used. The rate of gross energy input to the engine for each fuel could be estimated from the mass flow rate of the fuel (kg/s) and its calorific value (kJ/kg). To convert the measured volume flow rate of a fuel into mass flow rate, it was multiplied by the fuel's density. The specific gravity for ordinary diesel fuel and pure CPO were taken as 0.833 and 0.899, respectively [7]. The density (or specific gravity) of a CPO-OD mixture was found from those of pure CPO and pure diesel fuel in the appropriate ratio. Similarly, the calorific value of a CPO-OD blend was obtained from those of pure CPO and diesel fuel according to their mass ratios in the mixture. Fig. 2b shows the rate of gross energy input (kW) from the different fuels at all engine speeds considered. Because of their higher flow rate in the low-speed range, the input energy of CPO-OD mixtures could be greater than that of diesel fuel even though their calorific values were less. For the high-speed range, however, CPO-OD mixtures delivered less gross energy than that of pure diesel fuel since both their flow rates and calorific values were less than the corresponding values for diesel fuel.

Engine torque and brake power

Fig. 2c shows the indicated torque vs. engine speed for the different fuels used. Also here, the trend lines show maximum values in the range of 1800-2000 rpm. At speeds below 2000 rpm, the torque produced by CPO-OD mixtures was higher than that of pure diesel and increased with the percentage of CPO in the mixture. At speeds above 2000 rpm, the torque produced by the CPO-OD mixtures was less than, but comparable to, that of pure diesel fuel. The break power, or indicated power, was calculated from the measured torque and the respective engine speed. Fig. 2d shows the brake power vs. engine speed for the different fuels used. From the figure we can see that in the lower range of engine speed the brake power produced by CPO-OD mixtures was higher than that of pure diesel and the power increased as the percentage of CPO in the mixture increased. CPO-diesel mixtures could produce more power than diesel fuel in this low-speed range (despite the lower calorific value). In the high-speed range the flow rate and energy input of CPO-OD mixtures were actually less than those of pure diesel fuel. Nevertheless, the torque and brake power produced by the CPO-OD mixtures in this range were comparable to those of pure diesel fuel.

The difference in the gross energy input of CPO-OD mixtures and OD partially explains the difference observed between their respective torque and power output. Thus, it can be said that CPO-OD mixtures could produce a higher torque and engine power in the low-speed range because of their higher flow rate and energy input. However, the other factor that has to be taken into consideration is the efficiency of the combustion process since the fuel's energy is not completely converted by the engine into useful work. As will be shown later from the exhaust-gas analysis, the combustion process of CPO-OD mixtures in the low-speed range was incomplete and a large proportion of the fuel's energy was lost. It also appears from Fig 2c that the torque produced by CPO-OD mixtures was higher than that in the conventional diesel fuel.

Specific fuel consumption

The brake specific fuel consumption (BSFC), which is the fuel flow rate per hour (kg/h) divided by the engine's brake power (kW), is a measure of the engine efficiency. The BSFC vs. engine speed for all fuels is shown on Fig. 2e. Fig. 2f shows the ratio of the BSFC of CPO mixtures to that of pure diesel fuel. In general, the two figures show that, at low engine speeds, the BSFC of CPO-diesel mixtures was considerably higher than that of OD. However, the BSFC of a CPO-diesel mixture decreases as the speed increases and even become lower than that of OD at 2200-2500 rpm. The 25% CPO mixture has the closes BSFC to that of OD. The 50% and 75% CPO mixtures have close BSFCs which were about 10% higher than that of OD in the low-speed range. Their BSFCs become better at high engine speeds. These results indicate good combustion of CPO-diesel mixtures in the high-speed range, as we will seen later from the analysis of exhaust gases. The analysis also indicates that the fuel delivered to the engine at low-engine speeds was in excess when CPO-OD blends were used.

Exhaust-Gas Analysis

Carbon monoxide

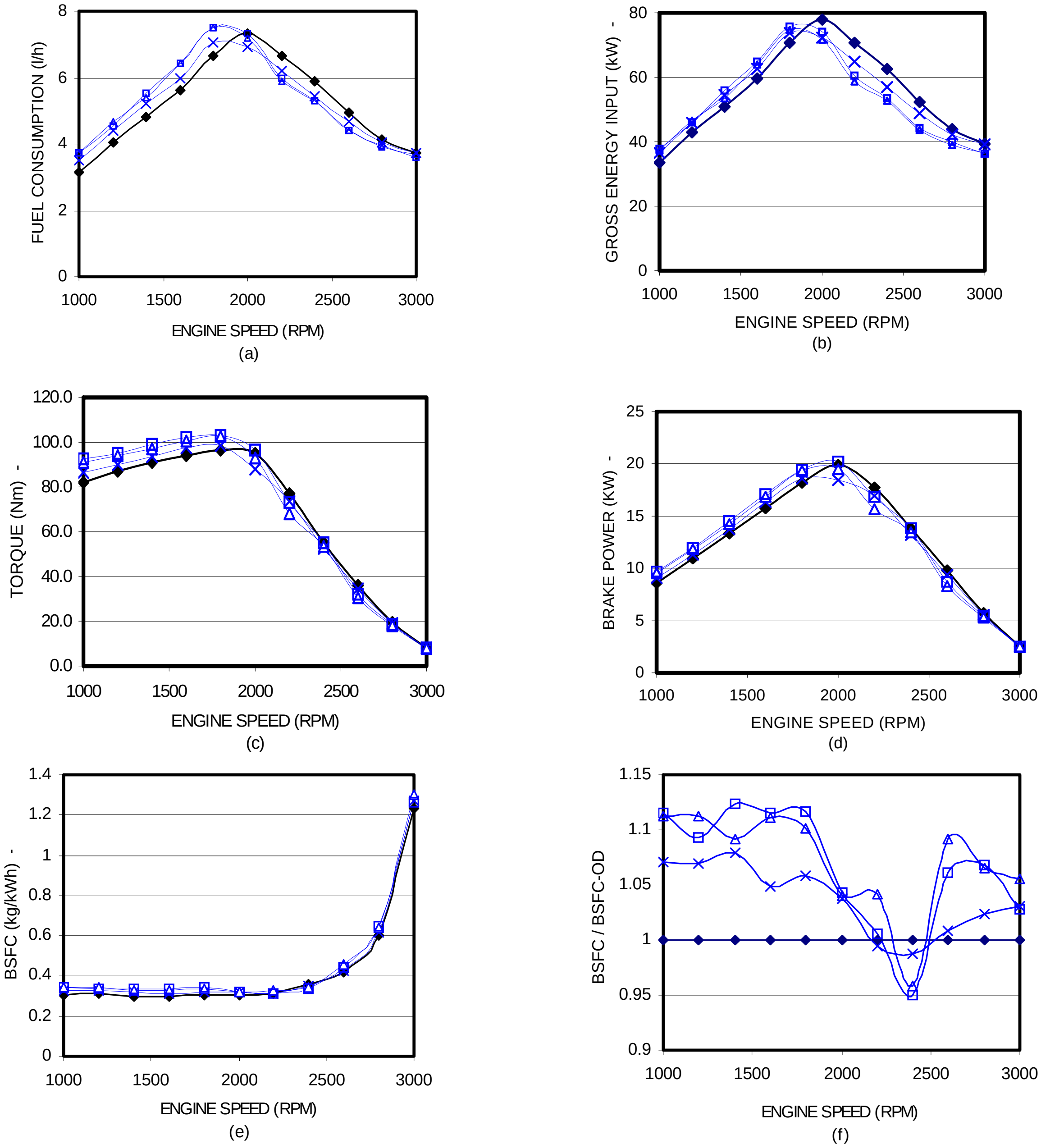
The presence of carbon monoxide (CO) in the exhaust indicates incomplete combustion of the fuel. We can see from Fig. 3a, which shows the fraction of carbon monoxide in the exhaust gas, that the maximum CO emission for pure diesel was around 2500 ppm, which occurred at a speed of 1800 rpm. For engine speeds below 2000 rpm, all CPO-diesel mixtures produced much higher percentages of CO in their exhaust gas than pure diesel. Since CPO-diesel mixtures also produced greater power than that of pure diesel fuel in the low-speed range (Fig. 2d), a reduction in the fuel flow rate will reduce the CO emission to a level that is comparable to that of pure diesel fuel. This will also improve the specific fuel consumption. At high engine speeds, Fig. 3a shows that the CO emissions of the CPO-diesel mixtures were comparable to, or even less than, that of pure diesel fuel. This could be attributed to the presence of oxygen in the CPO, which helped to achieve better combustion. The CO emission of the 75% CPO mixture was much higher than that of pure diesel over most of the speed-range except at speeds above 2600 rpm.

Oxygen

Contrary to the presence of CO in the exhaust gas, that of O_2 indicates complete combustion. However, due to the fact that air-fuel mixing cannot be perfect, there is always the possibility of some O_2 appearing in the exhaust gas even when the combustion is incomplete. From Fig. 3b, which shows the percentage of O_2 in the exhaust, we can see that for all the fuels considered the percentage of O_2 passes through its minimum value in the speed range 1800-2000 rpm, where the air-fuel ratio is closest to the stoichiometric value. As would be expected, in the low speed range all CPO-diesel blends showed lower O_2 percentage in their exhaust compared to pure diesel fuel. The Fig. also shows that the higher the concentration of CPO in the mixture, the lower the percentage of O_2 in the exhaust. These results clearly show that the engine was over-fuelled in the low speed region. At engine speeds above 2000 rpm, the 25% and 50% CPO mixtures showed slightly more O_2 in their exhaust than that of pure diesel fuel. Apparently, a good combustion of these two CPO mixtures in the high-speed range has made up for the low fuel consumption (see Fig. 2a) so that the fuel's brake power was comparable to that of ordinary diesel (Fig. 2d). The exhaust for the 75% CPO-mixture showed lower O_2 content compared to that of diesel fuel for all engine speeds.

Nitrogen oxides

The formation of nitrogen oxides (NO*x*) is governed by the availability of oxygen and, also, by other factors such as the combustion temperature and the delay period [12]. Fig. 3c shows the NO*x* emitted vs. engine speed. This figure also shows different trends in the low and high engine speeds. In the low-speed range all CPO-diesel mixtures emitted lower NOx levels compared to diesel fuel, with the 75% CPO mixture producing the lowest



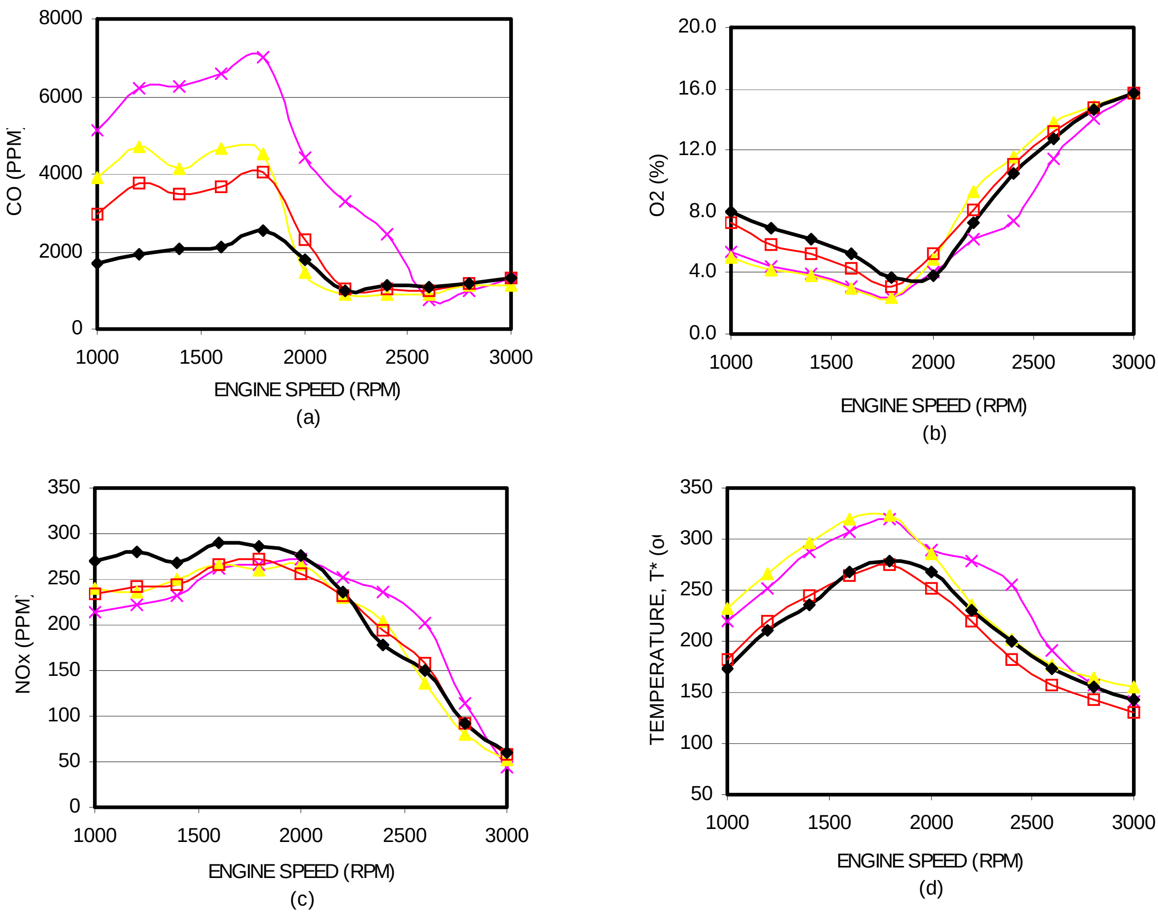
♦ Diesel fuel, × 25% CPO, Δ 50% CPO, □75% CPO

Fig. 2 Fuels power performance

NOx level. The low level of NOx emission in this range could be attributed mainly to the deficiency in oxygen (Fig. 3b). In the high-speed range, the NOx emissions of CPO-diesel mixtures were comparable to that of pure diesel fuel, except the 75% CPO mixture which produced a higher level of NOx emissions. The high level of NOx in the 75% CPO emissions could be attributed to a high combustion temperature and/or an extended delay period.

Exhaust-gas temperature

A high exhaust-gas temperature can be taken as an indicator of the combustion temperature. The gas-analyzer measures the difference (T^*) between the temperature of the exhaust gas (T_g) and the ambient atmospheric temperature (T_a), *i.e.* $T^* = (T_g - T_a)$. Referring to Fig. 3d, which shows T^* for the different fuels, it is clear that the temperatures for pure diesel fuel and all CPO-diesel mixtures reached their peak values at engine speeds lower than 1800 rpm. The CPO-diesel mixture with 75% CPO produced higher exhaust-gas temperatures than that of pure diesel fuel for all engine speeds. The 50% CPO mixture produced significantly higher exhaust-gas temperatures than that of pure diesel fuel in the low engine-speed range, but its exhaust temperature was comparable to that of ordinary diesel in the high-speed range. The exhaust-gas temperature of the 25% CPO mixture was comparable to that of OD in the low-speed range. In the high-speed range the exhaust-gas temperatures for the 25% CPO mixture was even lower than that of pure diesel fuel.



♦ Diesel fuel, × 25% CPO, Δ 50% CPO, □ 75% CPO

Fig. 3 Exhaust gas analysis

Conclusions

The experimental results show that the engine brake power, torque and brake specific fuel consumption when using CPO-OD mixture are very comparable with those when using ordinary diesel fuel under different operating conditions. At lower speeds such as 1800 rpm, the gross-energy input and the output power of CPO-diesel mixtures exceed that of ordinary diesel in spite of the lower heat content of CPO compared to ordinary diesel. Under light load operation, the CPO-diesel mixture suffered a loss of fuel efficiency and increased CO emissions relative to the diesel system. These results indicate that the fuel-metering and injection system tends to deliver more fuel at low speeds when the engine was fuelled by a CPO-diesel mixture, compared to pure diesel fuel. In the high-speed range, when using a CPO-diesel mixture, less fuel was injected into the engine compared to diesel fuel. The CPO mixtures showed comparable performance to that of diesel fuel for exhaust-gas composition and temperature, particularly the 25% and 50% CPO mixtures. The high exhaust temperatures and NOx emission shown by the 75% CPO mixture are possibly due to an extended ignition delay period. From these results we conclude that CPO-diesel mixtures, with up to 50% CPO by volume, can be used to fuel an unmodified diesel engine without affecting the engine's performance considerably. The type of results presented in this article can be used to produce an engine map that would be applied in the design of the CPO-Diesel Electronic Control Unit (ECU).

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