

## Article

# Characterization, at Partial Loads, of the Combustion and Emissions of A Dual-Fuel Engine Burning Diesel and A Lean Gas Surrogate.

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**Abstract:** For decentralized power generation in West Africa, gas from a small biomass gasification unit can be used as the main fuel in a dual-fuel engine with diesel as the pilot fuel. To study the combustion in this type of engine (Lister Petter), experiments were conducted with a surrogate gas composed of liquefied petroleum gas and nitrogen (LPGN2), the energy context of which is similar to that of syngas. The tests were conducted at different loads and for different diesel substitution rates. The combustion analysis showed that the LPGN2 mixture had an overall behaviour similar to neat diesel, while the pressure peaks were lower in dual-fuel mode. The results also indicated a longer ignition delay and a pronounced diffusive combustion phase leading to a lower indicated mean effective pressure with gas. The fuel efficiencies remained low in both mono- and dual-fuel operation. The relative instability of the combustion in dual-fuel mode gave rise to an increase in the coefficient of variation ( $COV_{IMEP}$ ). Compared to neat diesel, the engine running at low loads in dual-fuel mode showed higher emission levels of CO, a slight reduction of 2.5% of CO<sub>2</sub> and a substantial decrease of 73% for nitrogen oxides.

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**Keywords:** dual fuel; diesel engine; LPG; syngas; combustion

## 1. Introduction

Statistics on energy access worldwide indicate deep disparities. In 2018, nearly one billion people worldwide, including 600 million in Sub-Saharan Africa [1], had no access to electricity. In the same year, the national electrification rate in Burkina Faso was 21.44%, with only 3.16% in rural areas [2]. Consequently, the population in rural areas, which represented 77% of the total population, according to the report of the National Institute of Statistics and Demography (INSD) in 2018 [3], was deprived of services such as access to drinking water, health, education, and socio-economic activities. On the other hand, the high agricultural production of residues of cotton stalks, rice husks, and cashew shells is under-exploited. The estimated residues in 2018 in Burkina Faso are thought to be more than 8 million tons [4]. This biomass, converted into gas, which is called syngas [5,6], through the gasification process for sustainable bioenergy [7,8] and with additional treatment, could be utilized as an alternative fuel in a stationary engine for local electricity production and significantly contribute to socio-economic development in rural areas. Numerous works have reported on syngas as an essential fuel in several types of engines, including diesel engines in dual-fuel mode [9–20]. Despite its low energy content and the variability of its composition over time, especially for small installations, syngas has been found to be a good fuel for diesel engines operating in dual-fuel mode, since the engine

maintains its performance and efficiency at high loads and is environmentally friendly. However, very few points of data are available on the performance of diesel engines operating at part load in dual-fuel mode with syngas as the primary fuel. This situation is frequently encountered on rural installations, where the power delivered by the engine is very variable over time. To study the combustion of syngas in a dual-fuel diesel engine at low loads, a preliminary study was conducted on a diesel engine operating with a synthetic gas of stable composition and the same energy density as syngas. This synthetic gas is obtained from a combination of nitrogen and liquefied petroleum Gas (LPG) to obtain the same energy content as syngas. We have designated this gas as LPGN2. The unavailability of a local functioning gasifier and a syngas processing device for engine use led to the choice of this syngas surrogate, whose components are easily accessible in West Africa and particularly in Burkina Faso. In this work, we aim to evaluate the global performances of a diesel engine functioning with an LPGN2 of well-known composition and develop a model of optimal feeding of a dual-fuel engine. After a section devoted to the experimental setup and the fuels used, the results and their discussion are presented in Sections 3 and 4.

## 2. Materials and Methods

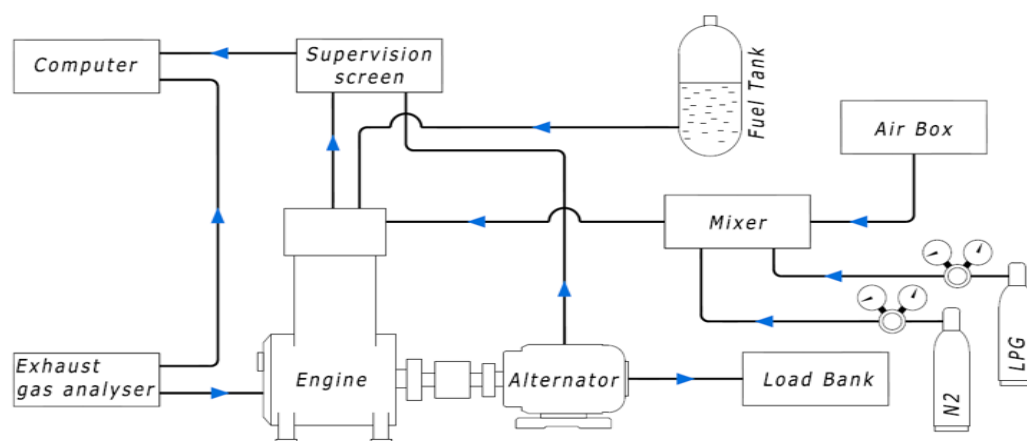
### 2.1. Experimental Device

The experimental set-up consisted of a diesel engine (with single-cylinder direct injection), an alternator, a load bank, and various data acquisition types of equipment. The choice of the diesel engine was based on the possibility of its implementation in a gasification facility in West Africa. Table 1 gives the specifications of the chosen diesel engine.

**Table 1.** Lister Petter engine specification.

| Model                           | Lister Petter (TR1)                                                                                     |
|---------------------------------|---------------------------------------------------------------------------------------------------------|
| Engine type                     | 4 stroke, direct injection (DI), compression ignition, single cylinder, naturally aspirated, air cooled |
| Displacement (cm <sup>3</sup> ) | 773                                                                                                     |
| Power (kW)                      | 4 kW at 1500 rpm                                                                                        |
| Bore/Stroke (mm)                | 98.42/101.6                                                                                             |
| Geometrical compression ratio   | 15.85:1                                                                                                 |

The test bench, built around a slightly modified generator set, is described in Figure 1. The intake manifold of the diesel engine was adapted to allow dual-fuel operation; the surrogate (i.e., combination of nitrogen and LPG) was mixed with air before being introduced into the diesel engine. Cylinder pressure was measured through an AVL GH15D pressure sensor, and the intake with a Cerabar PMC11 sensor. The diesel injection system remained unchanged, as the direct injection was suitable for dual-fuel operation. Moreover, the injection phasing was maintained at a crankshaft angle of 20° before the top dead centre (BTDC). The diesel fuel flow was measured using a graduated glass column and a stopwatch, whereas the LPG and nitrogen flow rates were estimated via a thermal mass flow controller of type F-201AV-50K-ABD-22-V and F-202AV-M10-ABD-55-V, respectively. The airflow measurement was carried out using an FCO332 pressure sensor. Other types of equipment, including Type K thermocouples, a charge amplifier (AVL FI PIEZO) and a gas analyzer (TESTO 350), were utilized to determine the temperature, heat release curves and exhaust emissions. All data was recorded by LabVIEW 2018 software on a computer via an acquisition rack.



**Figure 1.** Schematic of the experimental setup.

The accuracy and resolution of the main instruments are listed in Table 2

**Table 2.** Values of the uncertainties and resolutions of the measuring instruments used for parameter measurements.

| Instrument                     | Manufacturer—Model     | Resolution | Uncertainty |
|--------------------------------|------------------------|------------|-------------|
| Pressure sensor                | AVL/GH15D              | 0.01 MPa   | ±0.22%      |
| Angular encoder                | ROD426                 | 0.1 °V     | ±0.5%       |
| LPG flow meter                 | BRONKHORST/F-201AV-50K | 0.03 kg/h  | ±0.3%       |
| Nitrogen flow meter            | BRONKHORST/F-202AV-M10 | 0.01 kg/h  | ±0.3%       |
| Diesel flow measurement device | GRADUATED PIPETTE      | 0.1 ml     | ±10%        |
|                                | STOPWATCH              | 0.1 s      | ±1%         |

Table 3 gives the accuracy and resolution of the Testo 350 gas analyzer used for exhaust gas composition measurements.

**Table 3.** Accuracy and resolution of the gas analyzer Testo 350.

| Gas             | Range        | Accuracy                                                           | Resolution  |
|-----------------|--------------|--------------------------------------------------------------------|-------------|
| CO              | 0–10,000 ppm | ±5% of reading (200...2000 ppm)<br>±10% of reading (rest of range) | 1 ppm       |
| NO              | 0–4000 ppm   | ±5% of reading (100...1999.9 ppm)                                  | 1 ppm       |
| NO <sub>2</sub> | 0–500 ppm    | ±5 ppm (0...99.9 ppm)<br>±5% of reading (rest of range)            | 0.1 ppm     |
| CO <sub>2</sub> | 0–50% Vol. % | (0...25 Vol. %) ±0.5 Vol. %                                        | 0.01 Vol. % |

## 2.2. Fuels Used.

In this work, two operating modes are considered: (i) the conventional mode operation that uses neat diesel (D-P) and (ii) the dual-fuel mode in which the used fuels are diesel and liquefied petroleum gas (D-LPG) or diesel-surrogate (D-LPGN2) where diesel is the pilot or secondary fuel and LPG and LPGN2 are the dominant ones. The flow rates of each gas (LPG and N2) in the mixture are given by Equations (1) and (2):

$$Flow(LPG) = X \cdot Flow(gaz) \quad (1)$$

$$Flow(N_2) = \frac{(1 - X) \cdot Flow(LPG)}{X} \quad (2)$$

$$\text{With } X = \frac{LHV(\text{syngas})}{LHV(LPG)} \quad (3)$$

where the lower heating value (LHV) of syngas and LPG are about 4.42 MJ/kg and 45.75 MJ/kg, respectively and the calculated X value gives 0.097.

The combustion of the gaseous fuel [21] is initiated after introducing the surrogate-air mixture into the intake manifold and injecting the diesel fuel directly into the cylinder.

## 2.3. Test Procedure and Methods

In Table 4, the experimental test conditions presented are presented. The reference data used are those of the combustion and emissions of the engine operating on neat diesel fuel.

**Table 4.** Experimental design.

| Operating Mode      | Fuel                                  | Designation | Equivalence Ratio | Engine Settings                                     |
|---------------------|---------------------------------------|-------------|-------------------|-----------------------------------------------------|
| Conventional diesel | Neat diesel                           | D-P         | 0.28 to 0.57      | Speed: 1500 rpm                                     |
| Dual-fuel           | Pilot: Diesel                         |             |                   | Injection timing: 20 °CAD                           |
|                     | Primary fuel: Liquefied petroleum gas | D-LPG       | 0.35 to 0.59      | BTDC                                                |
| Dual-fuel           | Pilot: Diesel                         |             |                   | Load: 20%, 30%, 40%,                                |
|                     | Primary fuel: Gas surrogate           | D-LPGN2     | 0.25 to 0.52      | 50%, 60%, 70%<br>Diesel substitution rate: 0 to max |

The generator was loaded using a variable resistive load bank consisting of 300 W, 500 W and 1000 W resistors. Before testing, the engine was preheated and run with diesel fuel for 30 min at 1500 rpm without load until reaching a stable operating condition. At a defined engine load, the gas flow was slowly increased for a maximized substitution, while the speed controller automatically reduced that of the diesel. This procedure leads to the maximum gas flow approximation for proper engine operation in dual-fuel mode. The emission gas (CO, CO<sub>2</sub>, and NO<sub>x</sub>) was recorded using a gas analyser (Testo 350). The experimental measurements were repeated for loads ranging from 20% to 70% to ensure the viability of the results. The approximated rate of diesel substitution was obtained via the following relation:

$$SR(\%) = \frac{\dot{m}_{D/DP} - \dot{m}_{D/DF}}{\dot{m}_{D/DP}} \times 100 \quad (4)$$

where  $\dot{m}_{D/DP}$  is the diesel flow rate in neat diesel mode and  $\dot{m}_{D/DF}$  is the diesel flow rate in dual-fuel mode.

By applying the first principle of thermodynamics, the heat release was calculated according to Equation (5):

$$\frac{dQ_{gross}}{d\theta} = \frac{\gamma}{\gamma-1} P \left( \frac{dV}{d\theta} \right) + \frac{1}{\gamma-1} V \left( \frac{dP}{d\theta} \right) + Sh_c (T_{cyl} - T_{wall}) \quad (5)$$

where  $\frac{dQ_{gross}}{d\theta}$  is the gross heat release rate;  $\theta$  is the crank angle;  $P$  is the pressure in the cylinder;  $V$  is the working volume;  $\gamma$  is the ratio of the specific heats;  $S$  is the exchange surface; and  $T_{cyl}$ ,  $T_{wall}$  are temperatures in the cylinder and walls, respectively.

The brake-specific fuel consumption and thermal efficiency were calculated following Equations (6) and (7), respectively:

$$BSFC = \frac{\dot{m}_{D/DF} + \dot{m}_{gaz/DF}}{P} \quad (6)$$

$$\eta_{Overall} = \frac{3600}{BSFC \times LHV} \times 10^3 \quad (7)$$

BSFC represents the specific consumption in kg/kWh,  $P$  the efficient power of the engine in kW and  $\eta_{Overall}$  is the overall thermal efficiency of the engine, while LHV is the lower heating value of the fuel.

The indicated mean efficient pressure (IMEP) and its coefficient of variation  $COV_{IMEP}$ , which were used as combustion quality analysis parameters, were obtained by means of the cylinder pressure and Equation (8), respectively.

$$COV_{IMEP} = \frac{\sigma_{IMEP}}{\overline{IMEP}} \times 100 \quad (8)$$

$\sigma_{IMEP}$  is the standard deviation of the IMEP for 300 consecutive cycles and  $\overline{IMEP}$  is the average value of the IMEP for these 300 consecutive cycles.

### 3. Results and Discussion

#### 3.1. The Combustion Parameters

##### 3.1.1. Limit of Diesel Substitution

In dual-fuel mode, the appearance of unstable behaviour indicates the maximum allowable fraction of gas and the minimum amount of diesel required for proper engine operation. In practice, the limit is defined by the peak heat release in the pilot phase. The limit decreases with increasing gaseous fuel flow. This decrease follows a slope that changes at the substitution limit of the secondary fuel [22]. The substitution rates of pilot fuel for the gaseous fuels used are summarized in Table 5.

**Table 5.** Diesel substitution limits.

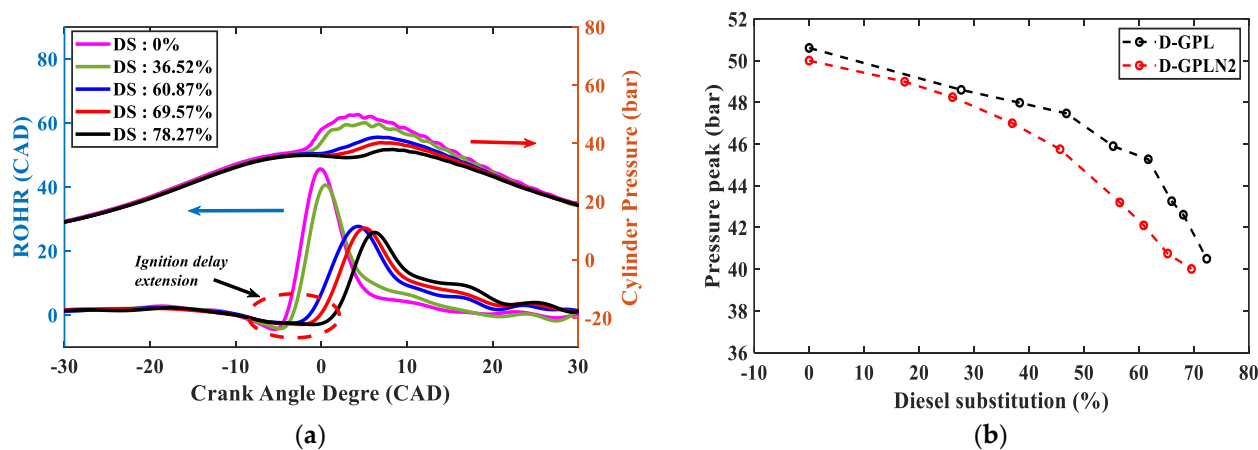
|                     | Load (%) | 20   | 30   | 40   | 50   | 60   | 70   |
|---------------------|----------|------|------|------|------|------|------|
| Diesel Substitution | D-LPG    | 61.7 | 58.9 | 60.3 | 59.6 | 58.8 | 57.9 |
| (%)                 | D-LPGN2  | 65.2 | 63.5 | 62.3 | 61.6 | 60.8 | 59.8 |

For all the gaseous fuels tested, the diesel substitution exceeded 50%. At D-LPGN2 operation, the maximum diesel substitution rate was systematically slightly higher than at D-LPG operation. In general, the substitution limits of the pilot fuel are lower than

those commonly encountered in dual-fuel mode with LPG or syngas as the dominant fuel [22,23], and are suggested to have originated from the smaller size and type of engine used in this work.

### 3.1.2. Cylinder Pressure and Heat Release

The heat release rate was evaluated from the average cylinder pressure over 300 consecutive cycles, in order to obtain representative results [15,24–27] by limiting the influence of combustion variation. Figure 2 shows the cylinder pressure and rate of heat release (ROHR) at 20% load in neat diesel and dual-fuel mode with LPGN2 as the essential fuel for different substitution rates (a) and compares the evolution of pressure peaks in dual-fuel operation (b).



**Figure 2.** Cylinder pressure and ROHR at D-LPGN2 (a) and evolution of the maximum cylinder pressure at 20% load (b).

In dual-fuel mode, the ignition delay is longer than with neat diesel only and generally increases as the gaseous fuel amount increases; see Figure 2a. This observation may be due to the presence of the primary fuel mixed with air, which alters the properties of the charge by the decrease in the oxygen concentration and the pre-ignition oxidation reactions during compression [28–30]. A large amount of gas extends the ignition time and spreads the combustion. Therefore, keeping the engine load constant, the maximum cylinder pressure decreases with increasing diesel substitution, as shown in Figure 2b. This reduction is about 9.51% with LPG and 15.81% with LPGN2 at 20% load. Compared to the LPG, this effect increases with LPGN2 because of its low LHV and the longer ignition delay, which extends the combustion phenomenon [20,28,31]. Figure 3 shows the evolution of the cylinder pressure and the heat release rate in dual-fuel mode at the maximum of the substituted diesel (Table 5) for the two gaseous fuels used at 20%, 40% and 70% engine load.

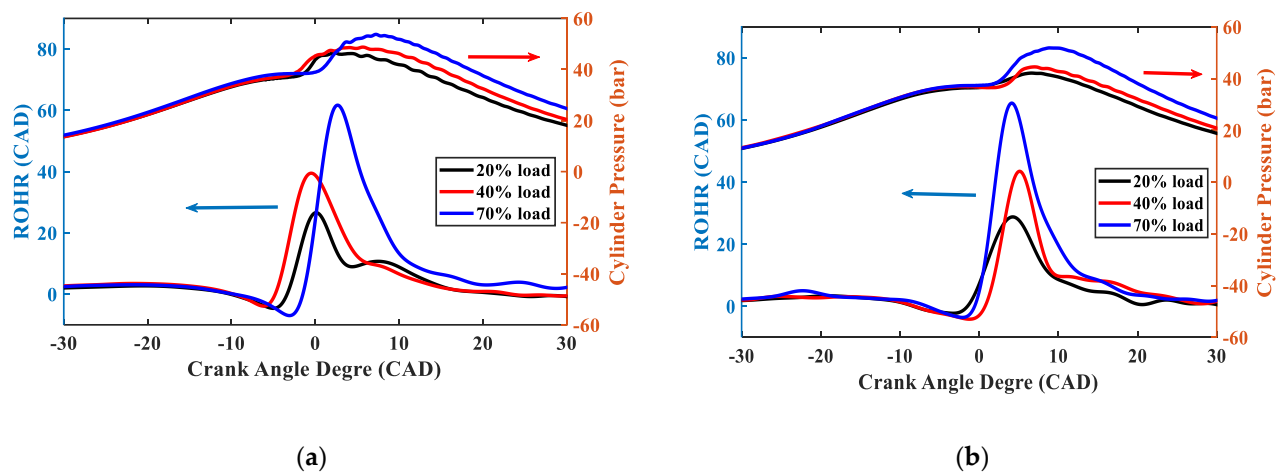


Figure 3. Cylinder pressure and ROHR for (a) D-LPG and (b) D-LPGN2.

In both operating modes, the peak cylinder pressure logically increases with engine load. At D-LPGN2 operation, this peak is lower and occurs later compared to the cylinder pressure with D-P. Mustafi and collaborators [32] made the same observation with a dual-fuel engine running on diesel and syngas. The peak of the heat release rate decreased when the amount of diesel injected decreased. Indeed, when the amount of pilot fuel injected decreased, this led to an increase in the ignition delay and in the combustion duration. Consequently, the pressure and temperature are lower at the end of the combustion phase of the pilot fuel. The presence of the gas reduces the amount of air in the cylinder, affecting the speed of gas combustion and leading to a decrease in the maximum heat release rate with the substitution of diesel [29,32–35]. The increase in the substitution rate delays the beginning of the combustion for all fuels. With D-LPGN2 operation, the ignition delay is longer for the same fuel load. The increase in the fuel quantity in the cylinder leads to a high combustion phase of the pilot premix at a high load.

### 3.1.3. The Mean Indicated Pressure and the Coefficient of Variation.

In the cyclic variations of an engine study, the most useful indicator is the coefficient of variation  $COV_{IMEP}$  [20,36]. The combustion is qualified to be unstable for a  $COV_{IMEP}$  exceeding 10% [36] and stable for a  $COV_{IMEP}$  value less than 5% [37,38]. In this study, the magnitudes of the cyclic variation are relatively higher. In the case of diesel only, the cyclic variation coefficient is 8.05% at 20% load, which decreases to 3.11% at 70% load, while D-LPGN2 presents a cyclic variation of 11.52% at 20% load and diminishes to 8.30% at 70% load. However, no failure is observed at  $COV_{IMEP}$  at slightly above 10%. Figure 4 shows the plot of  $COV_{IMEP}$  measurements. The gas injection increases the instability of the combustion. This result was also observed by Santoso et al. in their work [39].

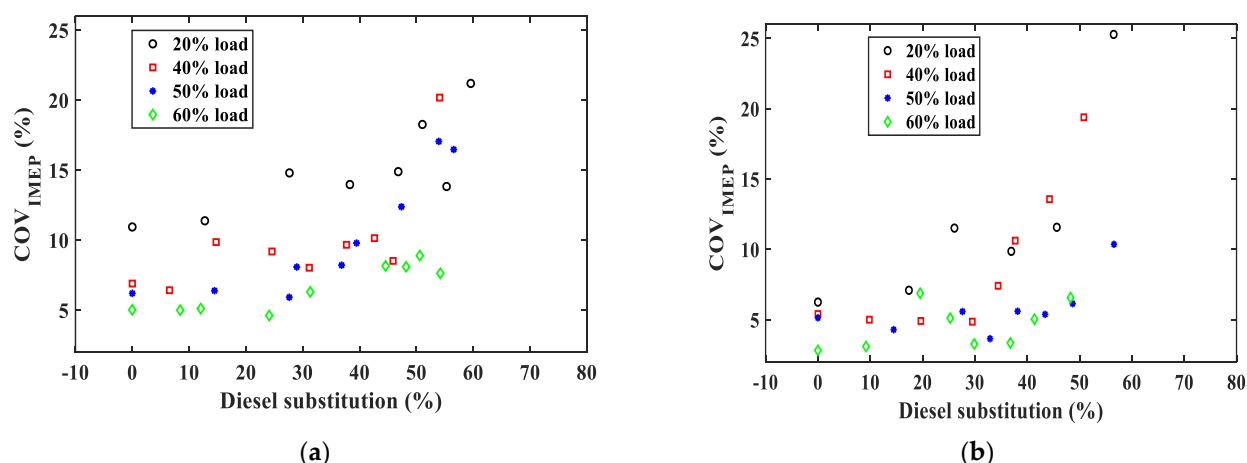


Figure 4. The COV of IMEP variation in dual-fuel mode: (a) D-LPG and (b) D-LPGN2.

The variation coefficients obtained are higher for both gaseous fuels. Tira et al. [34] and Goto et al. [40] have revealed that the low cetane number of LPG makes its auto-ignition difficult and leads to high cyclic variation during the combustion in diesel engines. This is a limitation of the studied surrogate compared to syngas, which has a more reactive composition, especially with heavier components facilitating self-ignition. At the maximum substitution in the dual-fuel mode, the COV<sub>IMEP</sub> value is about 11.52% at 20% load with LPGN2.

### 3.2. Performance Parameters

#### 3.2.1. Fuel Consumption

The results shown in Figure 5 indicate the reduction in diesel consumption in dual-fuel mode compared to pure diesel operation. Many studies have reported that dual-fuel technology saves 60–80% of diesel fuel [41–44], confirming the above results. Moreover, diesel consumption increases with load for all modes of operation, due to the increasing mass of injected fuel used to provide the required power.

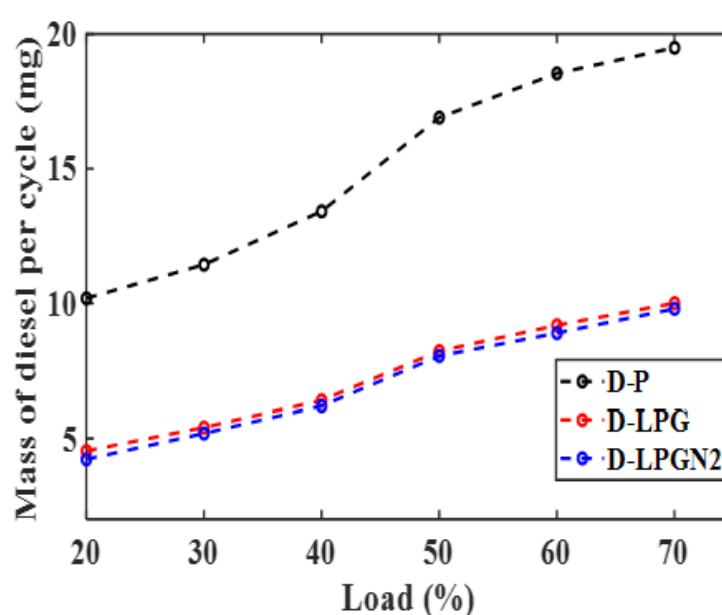


Figure 5. Diesel fuel consumption for the different operating modes.



The reduction in diesel consumption is uniform throughout the studied load range. The maximum value of diesel consumption reduction was observed at low load (20% of rated load) for a 61.70% reduction at D-LPG and 65.22% at D-LPGN2. Singh et al. [45] achieved a maximum diesel fuel reduction of 45.7% at low load with a dual-fuel engine fuelled by LPG and diesel. Malik and Mohapatra [46] have used a test bench, including an optimized diesel engine, to perform a maximum of 50% diesel fuel substitution. Yaliwal et al. [47] and Banapurmath et al. [41] have reported a maximum diesel reduction of 65% and 70.1%, respectively. At an engine load of 70%, the pilot fuel reduction reached for D-LPG is 57.92% and 59.77% for D-LPGN2. Sharma et al. [48] have used syngas in a dual-fuel engine mode with a compression ratio of 18 to obtain a diesel fuel economy of 58.18% at high loads. Singh et al. [45] have obtained a maximum reduction of 44.4% in diesel consumption at full load but with a relative decrease in indicated power of 3.49%. The addition of nitrogen resulted in a relative diesel savings of about 3.52% at a 20% load and 1.85% at a high load (70%), as compared to LPG. At high loads, the low LHV of the LPGN2 blend and its incomplete combustion reduce the pilot fuel substitution. Ramadhas et al. [49] have reported the same findings while operating a diesel engine in dual-fuel mode with syngas as the gaseous fuel. The gas consumption increases with increasing load and linearly with the substitution rate, as shown in Figure 6. As expected, the highest mass of gas consumed is observed at D-LPGN2, since the dilution of LPG by the nitrogen results in a gas mixture of low energy content. The linear variation with the fuel substitution indicates that the engine's performance is not much affected by substitution.

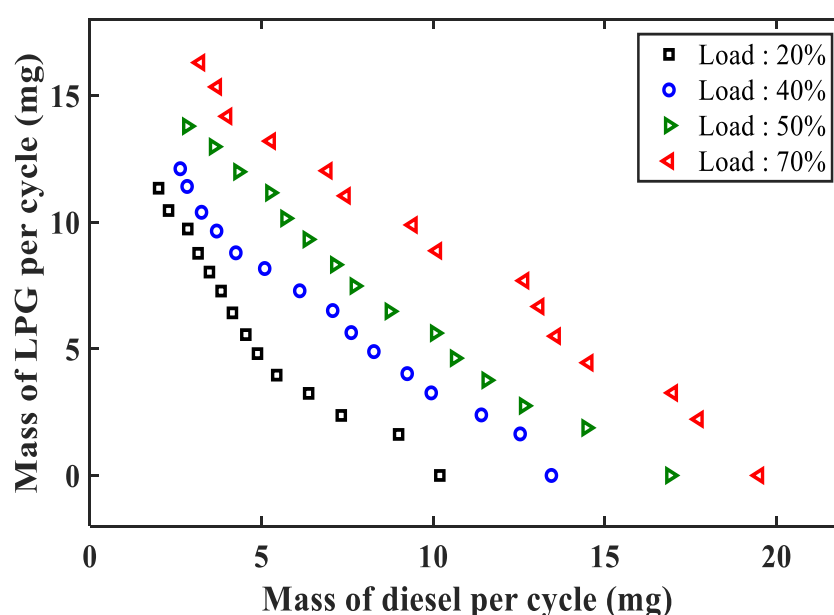
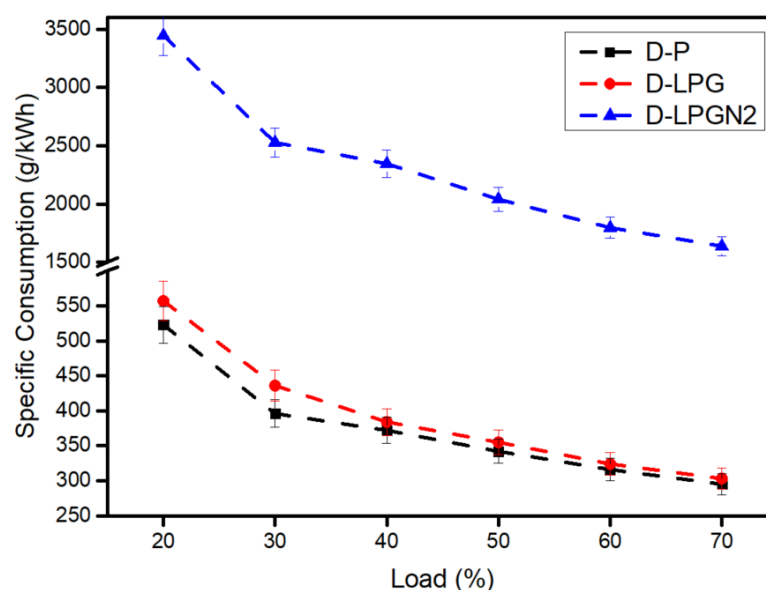


Figure 6. LPG consumption.

### 3.2.2. Brake-Specific Fuel Consumption (BSFC) and Thermal Efficiency ( $\eta_{th}$ )

The evolution of the specific fuel consumption of the engine is represented in Figure 7.



**Figure 7.** Brake-specific fuel consumption at different loads.

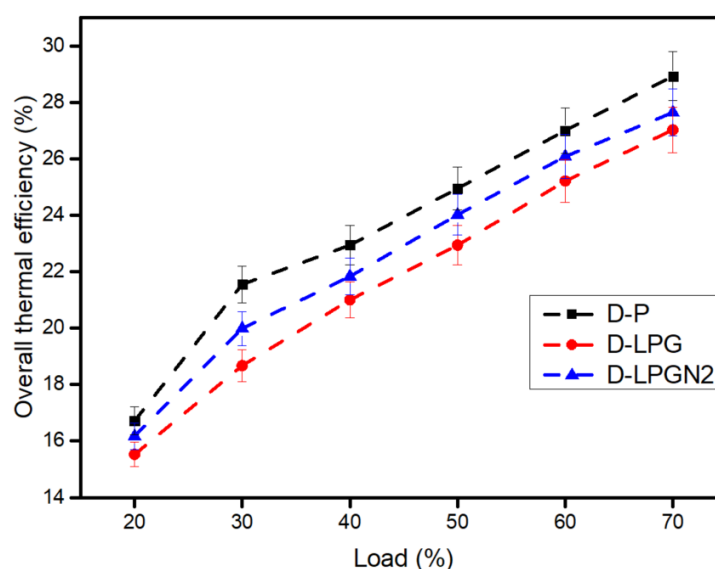
The brake-specific fuel consumption decreases as the load increases for all engine operating modes. As reported in the literature [33,42,44,49], increasing the load improves the efficiency of the engines, leading to a decrease in brake-specific fuel consumption. In dual-fuel operation, the consumption is slightly higher than that obtained with pure diesel due to the influence of the gaseous fuels composition on the combustion process. The brake-specific fuel consumption of the dual-fuel engine using LPGN2 is logically much higher than that of LPG. The energy density of LPGN2 (4.42 MJ/kg), relative to that of syngas, is lower than that of LPG (45.75 MJ/kg). Sahoo et al. [17] have observed the same trends while running a single-cylinder diesel engine in dual-fuel operation with methane (50 MJ/kg) and a mixture of natural gas (47.7 MJ/kg) and LPG (46.1 MJ/kg). Hagos et al. [11] also made the same discovery with a dual-fuel engine using natural gas or LPG as a gaseous fuel. In the present work, the brake-specific fuel consumptions measured at 20% load were 0.52 kg/kWh, 0.56 kg/kWh and 3.44 kg/kWh in diesel, D-LPG and D-LPGN2, respectively. At 70% load, the values are 0.295 kg/kWh, 0.30 kg/kWh and 1.64 kg/kWh. These results are in agreement with other works [33,42,44,49]. Table 6 summarises a few points of data on brake-specific fuel consumption, engine type and fuels used in dual-fuel operation mode.

**Table 6.** Brake-specific fuel consumption obtained by some researchers.

| Pilot Fuel | Primary Fuel | Engine Type                                           | Load (%) | Brake-Specific Fuel Consumption (kg/kWh) | Searcher |
|------------|--------------|-------------------------------------------------------|----------|------------------------------------------|----------|
| Diesel     |              | Single cylinder diesel engine, DI, 1500 rpm, CR: 17.6 | 20       | 0.35                                     | [33]     |
| Diesel     | Natural gas  | Single cylinder diesel engine, DI 1500 rpm CR: 17.6   | 20       | 0.67                                     | [33]     |
| Diesel     |              | Single cylinder diesel engine, DI 1500 rpm            | 60       | 0.4                                      | [33]     |

|          |              |                                                                                 |    |       |              |
|----------|--------------|---------------------------------------------------------------------------------|----|-------|--------------|
| CR: 17.6 |              |                                                                                 |    |       |              |
| Diesel   | Natural gas  | Single cylinder diesel engine, DI 1500 rpm<br>CR: 17.6                          | 60 | 0.63  | [33]         |
| Diesel   |              | Single cylinder, diesel engine, 1500 rpm<br>CR: 18                              | 80 | 0.282 | [49]         |
| Diesel   | Producer gas | Single cylinder, diesel engine, 1500 rpm<br>CR: 18                              | 80 | 0.34  | [49]         |
| Diesel   |              | Single cylinder diesel engine, DI 1500 rpm<br>CR: 18                            | 80 | 0.282 | [42]         |
| Diesel   | Syngas       | Single cylinder diesel engine, DI 1500 rpm<br>CR: 18                            | 80 | 0.34  | [42]         |
| Diesel   | Syngas       | Single cylinder variable compression ratio diesel engine, DI 1500 rpm<br>CR: 14 | 40 | 0.71  | [44]         |
| Diesel   | Syngas       | Single cylinder variable compression ratio diesel engine, DI 1500 rpm<br>CR: 14 | 80 | 0.59  | [44]         |
| Diesel   | LPGN2        | Single cylinder diesel engine, DI 1500 rpm<br>CR: 18.375                        | 20 | 3.44  | Present work |
| Diesel   | LPGN2        | Single cylinder diesel engine, DI 1500 rpm<br>CR: 18.375                        | 70 | 1.64  | Present work |

The overall thermal efficiency depends on the fuel consumption, the heating value of the fuel and the engine power, as shown in Figure 8.



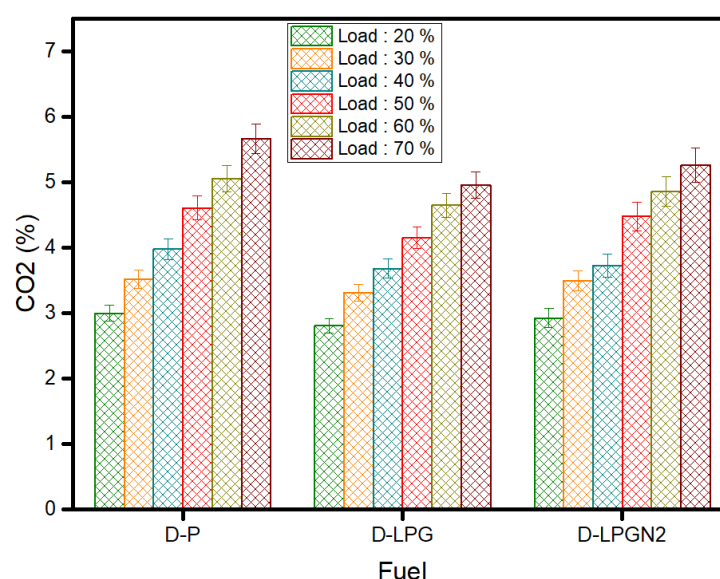
**Figure 8.** Overall thermal efficiency for the different fuels at different loads.

The overall thermal efficiency  $\eta_{th}$  of pure diesel is slightly higher than that of the dual-fuel mode. The decrease in dual-fuel mode efficiency is related to the ignition delay observed in this mode across a longer duration [17,48]. At low loads, the thermal efficiency is lower. At 20% of the rated engine load, the efficiency value reaches 16.71% in diesel and 16.17% in D-LPGN2. The value increases with the increase of the engine load. We recorded a maximum efficiency at 70% load of 28.93% and 27.64% in D-P and D-LPGN2, respectively. With a 4.4 kW single-cylinder engine at 1500 rpm and one injection, Shrivastava et al. [18] have obtained a maximum efficiency of 27.5% in diesel and 26% in dual-fuel with syngas. Sharma and Kaushal [48] also found 25.6% and 21.6% in conventional diesel mode and dual-fuel using syngas, with a single cylinder 3.5kW direct injection engine.

## 4. Emissions

### 4.1. CO<sub>2</sub> Emission

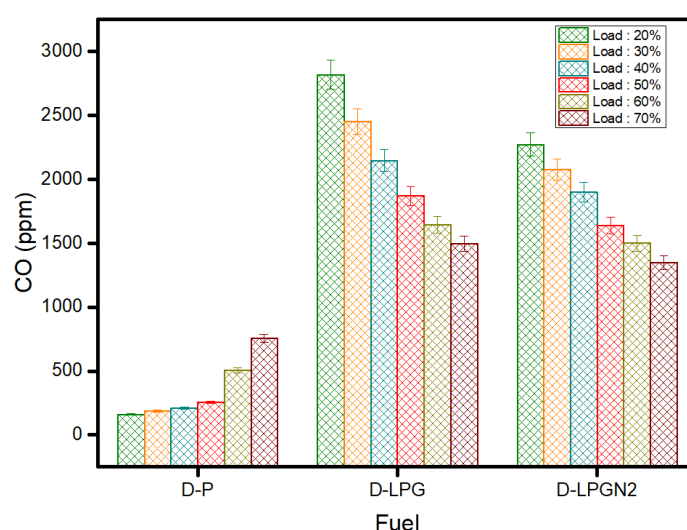
CO<sub>2</sub> emissions are related to the fuels used and their combustion quality. Since LPG is hydrogen-rich, there is a slight decrease in CO<sub>2</sub> emissions when operating in dual-fuel mode; see Figure 9. As the differences are small, the carbon dioxide concentrations are in the same order of magnitude for all engine-operating modes. These results are similar to results obtained by other authors [13,19,50]. However, for biomass fuels, CO<sub>2</sub> is considered part of the global carbon cycle, and its contribution to global warming is negligible [19]. The syngas used in the dual-fuel combination has no significant impact on CO<sub>2</sub> emissions.



**Figure 9.** CO<sub>2</sub> emission for the different fuels at different loads.

#### 4.2. CO Emissions

In pure diesel operation, carbon monoxide fraction increases with increasing load as the mixture equivalence ratio increases; see Figure 10. The formation of CO is usually known as a function of the air/fuel ratio, the homogeneity of the mixture and the temperature [29,32,36,51,52]. For all loads evaluated, the CO concentration in dual-fuel mode is significantly higher than in pure diesel operation. The highest emission levels in dual-fuel mode are observed at low loads. This can be explained by the fact that as the engine load increases, more fuel is required, and a fuel–air-rich mixture enters the cylinder, leading to more reactive mixtures at higher temperatures and thus less CO emissions [44].

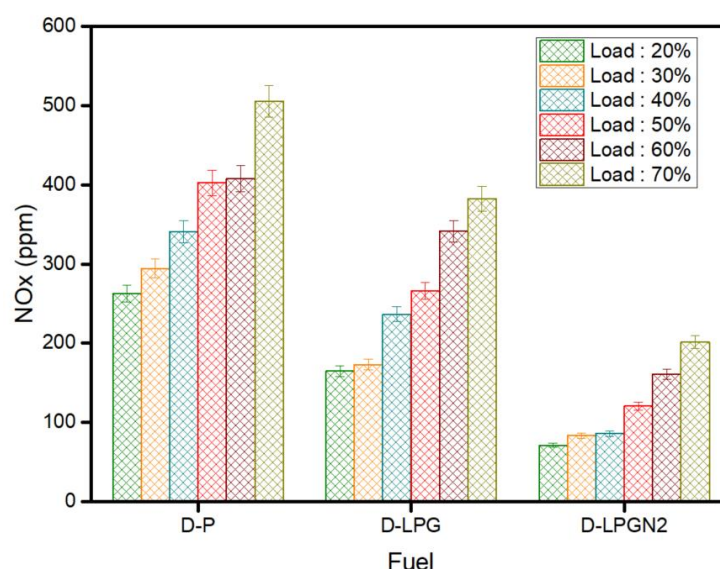


**Figure 10.** CO emission for the different fuels at different loads.

At 20% engine load, for D–LPG, there is an increase in CO concentration of about 2600 ppm compared to conventional diesel operation. With LPGN2 syngas, we observe a decrease in carbon monoxide emissions due to the lower quantity of carbon in the fuel admitted to the cylinder. The increase in load leads to an increase in value of the proportion between air and fuel admitted to the combustion chamber. This increase in richness improves combustion at high loads, hence the tendency to emit little at high loads [29,32].

### 4.3. NO<sub>x</sub> Emissions

Figure 11 presents the plotted data of the nitrogen oxide (NO<sub>x</sub>) emission measurements. The production of NO<sub>x</sub> is highly dependent on the maximum temperature of the burning gases, the oxygen content and the residence time available for the reactions to occur under these extreme conditions [53–55].



**Figure 11.** NO<sub>x</sub> emission for different fuels at different loads.

As expected, NO<sub>x</sub> production increases with increasing load for all engine-operating modes. The increase in load causes an increase in cylinder temperature. The formation of nitrogen oxides (NO<sub>x</sub>) is enhanced by high oxygen concentration and high load temperature [17,36,56–58]. Moreover, the NO<sub>x</sub> concentrations in engine exhaust are higher in diesel fuel compared to that measured in dual-fuel mode. Furthermore, we observe high combustion temperatures in diesel. As diesel substitution increases, the pressure and temperature in the cylinder decrease, resulting in the lowest NO<sub>x</sub> concentrations at the point of maximum diesel substitution in dual-fuel operation. The dilution of LPG with nitrogen accentuates this trend due to the reduction in combustion temperature in the cylinder [20] and the decrease in the amount of available oxygen. Thus, at a low load, the reduction in NO<sub>x</sub> emissions is 38% in D-LPG and 73% in D-LPGN2, as compared to the concentrations observed in diesel.

## 5. Conclusions

The objectives of the present work were to determine the overall performance and combustion parameters of a diesel engine operating at partial load in dual-fuel mode with a gas mixture (LPG + Nitrogen: LPGN2) for which the composition is well-known and of the same energy content as syngas from biomass gasification. The combustion characteristics, performance and emissions of the pure diesel and LPG engines were used as a reference. The phenomenology of LPGN2 combustion is similar to that of syngas combustion in the dual-fuel engine. The analysis of the heat release rate curves showed four combustion phases in the diesel engine in dual-fuel mode. The substitution levels for diesel with LPGN2 and the variations in cylinder pressure and heat release rate in the present work were comparable to those found in the literature. The slight difference observed in the combustion is related to the cetane number of the LPGN2 gas surrogate, which is different from that of syngas. The exhaust gas concentrations indicate a substantial reduction in pollutant emissions, such as NO<sub>x</sub> and CO<sub>2</sub>, as has been indicated by other experimental studies comparing dual-fuel operation with diesel operation. However, CO emissions are

higher in dual-fuel operation at low loads. The emission concentrations with the syngas surrogate (LPGN2) confirm the cleaner nature of dual-fuel combustion compared to conventional liquid fuels. The combustion parameters, namely, performance and emissions, obtained with D-LPGN2 are comparable to those observed with dual-fuel engines running on a syngas. Given the operation at partial load and the choice of a single-cylinder engine relevant for the applications in West Africa, the results obtained with LPGN2 give better indications for the study and optimization of the combustion of gasification gas in a diesel engine in dual-fuel mode at low load.

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