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Combustion Parameters of A Dual-Fuel Diesel-LPG Engine Operating at Partial Loads

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ABSTRACT

Burkina Faso has a low rate of access to electricity, characterized by disparities between residential areas. In 2024, the electrification rate stood at 34.2%, with only 10.2% in rural areas. To address this shortfall, electricity generation using stationary diesel engines is a promising solution. However, the cost of diesel limits the use of these engines in conventional diesel mode. Thus, the use of alternative gaseous fuels such as liquefied petroleum gas (LPG) appears to be a viable solution. The engine operates in dual fuel mode, with LPG as the primary fuel. This study examines the combustion characteristics of a Lister Petter diesel engine using LPG as an alternative gaseous fuel. The objective is to compare the engine's combustion parameters in both operating modes (pure diesel and dual fuel). An analysis of cylinder pressure curves and heat release rates shows combustion parameters comparable to those of pure diesel. However, dual-fuel mode results in lower pressure peaks, particularly at low loads. An 8.5% reduction in maximum pressure in the diesel-LPG mode at 20% load. The introduction of gas prolongs the ignition delay, enhances the diffusive combustion phase, and extends the duration of combustion. Furthermore, the relative instability of combustion in dual-fuel mode results in an increase in the coefficient of variation of the indicated mean pressure (COVIMEP). At 20% load, the COVIMEP reaches 11.5% in diesel-LPG mode while maintaining stable engine operation. These results point toward the future use of syngas on a test bench built around the Lister Petter engine.

Keywords: Dual fuel ; diesel engine; LPG ; combustion characteristics.

1. Introduction

According to the International Energy Agency's World Energy Outlook (WEO) 2025 report, around 630 million people worldwide were living without electricity in 2024 (Agency International Energy, 2025). 80% of them are from Sub-Saharan Africa, which has electricity access rate of 34%. This rate is characterized by significant disparities between different residential areas. Around 80% of the rural population in sub-Saharan African countries have no access to electricity (Agency International Energy, 2025). The lack of modern energy services deprives them of basic needs such as access to clean water, healthcare, quality education, daily comfort, safety, income-generating activities. The lack of electricity is therefore a real obstacle to the socio-economic development of these populations. Burkina Faso, a sub-Saharan country, had an electrification rate of 34.2% in the same year, with only 10.2% in rural areas (MINISTERE DE L ' ENERGIE DES MINES ET DES CARRIERES, 2025). To expand access to energy services, sub-Saharan countries such as Togo, Mali, Ghana, Senegal, and Burkina Faso have launched several rural electrification programs and projects, including the Multifunctional Platforms Program (PTFM) (Nygaard, 2010). These projects, supported by the United Nations Development Program (UNDP), aim to improve living conditions in rural areas. The multifunctional platform is a decentralized energy infrastructure in rural areas. Built around a diesel engine, this system integrates several mechanical and electrical applications: agri-food processing (mills, huskers), electricity generation (alternator, battery charger), water pumping and craft activities (welding, carpentry). Designed to address the lack of access to modern energy services, the technology aims to replace human labor with motorized mechanical power, thereby reducing the physical strain of work for communities at the bottom of the energy ladder. The cost of diesel accounts for approximately 70% of the multifunctional platform's operating expenses (Vaitilingom, 2006). The high proportion of diesel costs in the operating expenses of these applications, the volatility of oil prices, uncertainty regarding security of supply, and environmental concerns are driving the search for alternative fuels that can reduce the operating costs of diesel-powered applications (Vaitilingom, 2006). The use of alternative gaseous fuels such as natural gas and liquefied petroleum gas in the Lister Petter diesel engine, in addition to diesel, is one of the most promising solutions. This approach reduces dependence on liquid fuels.

Produced through oil refining, liquefied petroleum gas (LPG) is a viable alternative gaseous fuel with a higher lower heating value than diesel. Although its cetane number is low, its high-octane rating makes it suitable for direct use in spark-ignition engines (Ashok et al., 2015; Oguma et al., 2003; Tira et al., 2012). The strategy for using LPG in a compression-ignition engine involves operating the engine in dual-fuel mode. LPG is the primary fuel and the engine's main source of energy; it is either injected directly into the cylinder or mixed with air before being fed into the intake manifold designed for this purpose. Diesel, the secondary fuel also known as pilot fuel, is injected into small quantities in the manner of a conventional diesel engine towards the end of the compression of the air-LPG mixture. The spontaneous ignition of the pilot diesel initiates the combustion of the gaseous mixture. The performance of the dual-fuel diesel engine is impressive, with low particulate and smoke emissions (Ashok et al., 2015; Iswantoro et al., 2022). This mode of operation of the diesel engine has been the subject of several studies. However, few data are available on the combustion parameters of the Lister Petter-type diesel engine operating in dual-fuel diesel-LPG mode.

The aim of this study is to assess the combustion parameters of the Lister Petter dual-fuel diesel-LPG engine operating at partial loads. Following a section on the experimental setup, the results and their discussion are presented in Section 3. The conclusion is presented in Section 4.

2. Materials and Methods

2.1 Experimental setup

The diesel engine used is a single-cylinder, four-stroke, direct-injection, naturally aspirated, air-cooled engine. An alternator is coupled to the engine. Table 1 lists the technical specifications of the engine and the alternator.

Table 1 *:* Lister Petter engine specifications

Engine specifications		Alternator specifications	
Model	**Lister Petter (TR1)**	Alternator type	IP23
Engine type	Four strokes, Direct Injection (DI)		
displacement (cm³)	773	Max power	6 kW at 1500 rpm
Power (kW)	4 kW at 1500 rpm		
Bore/Stroke (mm)	98.42/101.6	Cos φ	0,8
Geometrical compression ratio	15.85 :1	RPM	1500

The engine's intake manifold has been modified to operate in dual-fuel mode. The engine is equipped with pressure and temperature sensors, flow meters, and a Testo 350 portable exhaust gas analyzer. The cylinder pressure measurement system consists of an AVL GH15D piezoelectric pressure sensor mounted in the cylinder head and an AVL FI PIEZO charge amplifier. The angular position of the piston is determined by a Heidenhain ROD 426 rotary encoder mounted on the engine's camshaft. The engine is loaded using a resistive load bank consisting of resistors with power ratings of 250 W, 500 W, and 1000 W. A 50 W fan is used to cool the resistors. The experimental setup allows for flexible adjustment of the engine load to simulate partial-load operations. Table 2 provides information on the electrical powers assessed and their correspondence with the load applied to the engine.

Table 2 *:* Power and load applied to the engine

Electrical power (W)	Motor load (%)
0	0
800	20
1300	32,5
1500	37,5
2050	51,25
2500	62,5
2800	70

Component and exhaust temperatures were monitored using Type-K thermocouples. All sensors are connected to a data acquisition rack equipped with National Instruments (NI) modules. The rack is connected to a computer, where data is collected using LabVIEW 2018 software. Figure 1 illustrates the key components of the test engine bench.

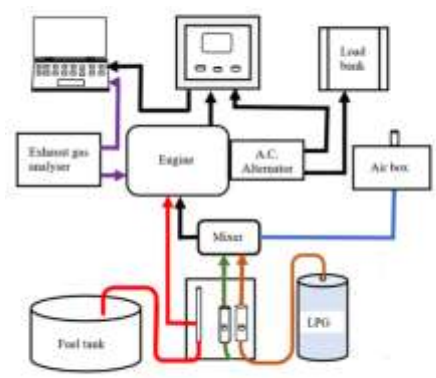

Figure 1 : Experimental setup

2.2 Engine Fuel Supply

The engine operated in dual-fuel mode. Liquefied petroleum gas (LPG) served as the primary fuel, and diesel as the pilot fuel. A Bronkhorst mass flow controller connected to a high-pressure cylinder was used to control the

LPG flow rate. The diesel flow rate is measured by a device consisting of a graduated glass column and a T-valve placed between the tank and the glass column, which opens or closes the pilot fuel flow path. Figure 2 illustrates the diesel fuel flow measurement device.

1: Graduated glass column

2: T-valve

3: Connection between the valve and the pump

4: Overflow return from the injection system

5: Connection between the valve and the tank

6: Overflow return from the column

Figure 2 : Diesel flow meter

The diesel fuel substitution rate, which quantifies the energetic contribution of the gaseous fuel relative to total energy input, is defined by Equation 1.

$$DS = \frac{Q_{diesel} - Q_{dual-fuel}}{Q_{diesel}} \times 100 \qquad (1)$$

Q_{diesel} represents the energy contribution from diesel when the engine is running without any additional fuel.

$Q_{duel\ fuel}$ the energy contribution of diesel during dual-fuel operation with the addition of gas. Specific fuel consumption and overall efficiency were calculated using the equations.

The air flow rate is measured by a pressure-reducing device consisting of a diaphragm fitted to the housing of a differential pressure sensor.

The engine starts with diesel and runs unloaded for a sufficient period to reach a stable operating state. It is then loaded. Under stable operating conditions, at a defined engine load, LPG gradually replaces the diesel. When the maximum diesel substitution rate is achieved, the engine approaches its operational limit and begins to exhibit combustion instability.

The average pressure in the cylinder was calculated to be over three hundred consecutive cycles. The heat release rate was computed using equation 2:

$$\frac{dQ}{d\theta} = \frac{1}{\gamma - 1} V \frac{dP}{d\theta} + \frac{\gamma}{\gamma - 1} P \frac{dV}{d\theta} \qquad (2)$$

$\frac{dQ}{d\theta}$ is the gross heat release rate; θ is the crank angle; V is the working volume; P the cylinder pressure; γ is the ratio of specific heats.

The coefficient of variation (COV) was derived from the indicated mean effective pressure (IMEP) using the relationship expressed by equation 3 :

$$COV_{IMEP} = \frac{\sigma_{IMEP}}{\overline{IMEP}} \times 100 \qquad (3)$$

σ_{IMEP} is the standard deviation of the IMEP for three hundred consecutive cycles and $\overline{IMEP}$ is the average value of the IMEP for these three hundred consecutive cycles.

2.3 Determination of the substitution limit

In dual-fuel mode, diesel fuel is gradually substituted by gaseous fuel. The onset of signs of engine malfunction indicates the maximum proportion of gas allowed into the cylinder and the minimum amount of diesel required for the engine to operate correctly. This point, known as the diesel substitution limit, is identified as shown in Figure 3. The peak heat release rate from the combustion of the pilot fuel decreases as the flow rate of the gaseous fuel increases. This decrease is linear and changes slope at the diesel substitution limit (Bilcan et al., 2001).

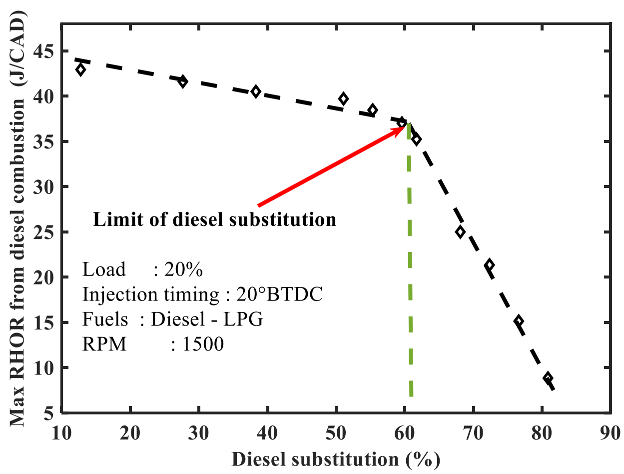

Figure 3: Determination of the diesel substitution limit at 20% load on D-LPG

3. Results and discussion

3.1 Pilot fuel substitution ratio

Figure 4 shows the substitution limits for diesel fuel when the engine is operating in dual-fuel mode with LPG as the gaseous fuel.

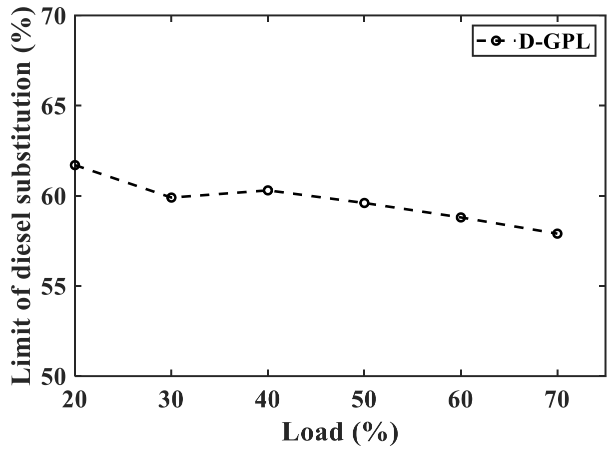

Figure 4 : Diesel substitution limit for different engine loads when running on Diesel-LPG

The highest diesel fuel substitution rates are observed at low loads and decrease linearly as the load increases due to the higher pilot fuel consumption required by increased engine load. For all loads examined, more than 50% of the diesel is substituted. The pilot fuel substitution limits obtained are lower than those commonly encountered in dual-fuel mode in LPG-diesel engines (Bilcan et al., 2001; Garnier et al., 2005). This could be explained by the small size and low fill ratio of the engine used in this study. Furthermore, the low cetane number of LPG causes significant cyclic variation, limiting its use in large proportions in compression-ignition engines (Ashok et al., 2015; Oguma et al., 2003; Tira et al., 2012).

3.2 Cylinder Pressure

Figure 5 illustrates the variation in cylinder pressure as a function of engine load at maximum power for the modified diesel engine.

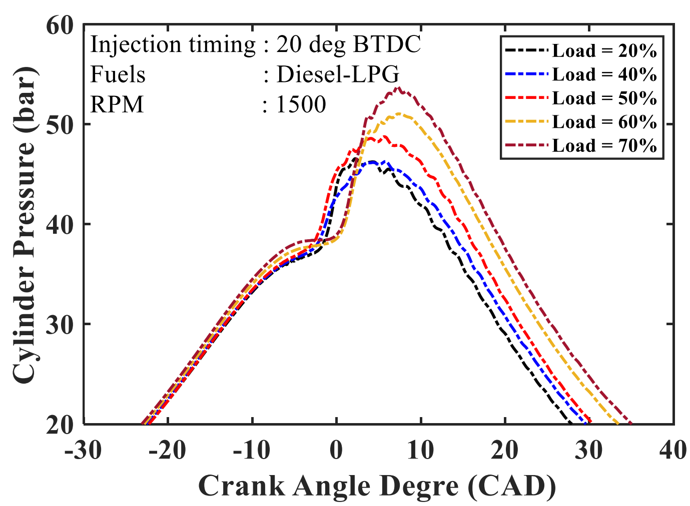

Figure 5 : Cylinder pressure for different loads at the Diesel substitution limit

The maximum cylinder pressure increases logically with load. An increase in load improves temperature conditions in the combustion chamber, leading to a rise in cylinder pressure across the various operating modes.

In diesel-LPG mode, maximum cylinder pressure values are lower than those in pure diesel mode. At a constant load, increasing the amount of gas introduced into the cylinder causes a proportional decrease in cylinder pressure. Figure 6 illustrates the evolution of cylinder pressure for different diesel substitution rates during dual-fuel engine operation at partial loads of 20%, 40%, 50%, and 70%.

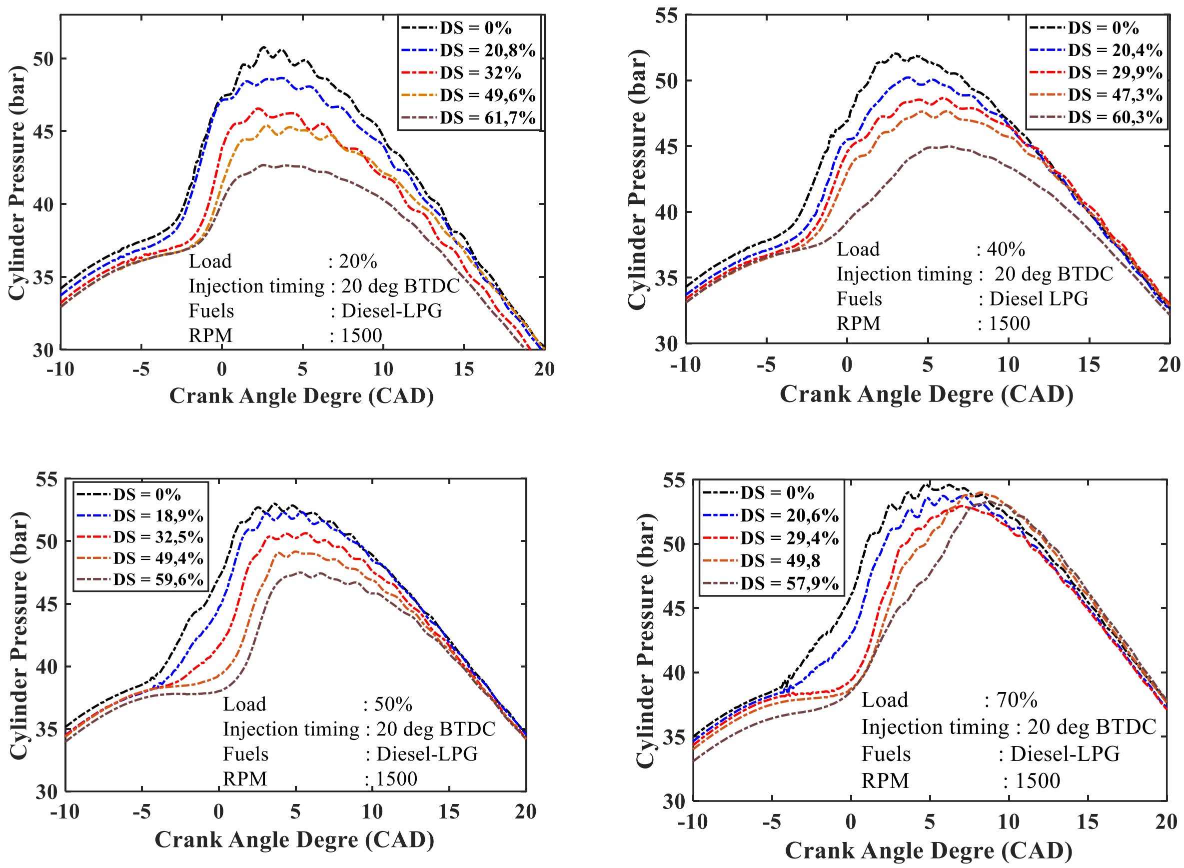

Figure 6 : Cylinder pressure variation at low and medium loads

A reduction in cylinder pressure is observed for all loads examined. The largest deviations are observed at lower loads. At 20% load, the decrease in peak cylinder pressure is 8.5% compared to conventional diesel engine operation. Increasing the load reduces this deviation, which reaches 3% when the engine load is 70%. At a constant load, an increase in the gaseous fuel fraction induces a progressive reduction in peak cylinder pressure, resulting from the lower thermal energy release characteristic of dual-fuel combustion.

Furthermore, under partial-load operating conditions in dual-fuel mode, the peak cylinder pressure is primarily controlled by the combustion of the pilot fuel (Lata et al., 2011). As the amount of pilot fuel decreases with an increase in the amount of gaseous fuel, the amount of fuel available for the first stage of combustion is reduced. This results in a decrease in the peak cylinder pressure.

Moreover, the pressure drop in the combustion chamber during dual-fuel operation is primarily driven by an extended ignition delay, which shifts the combustion process further into the expansion stroke and subsequently prolongs the combustion duration (Ali et al., 2023; Guan et al., 2017; Sadig et al., 2017; Surawski et al., 2014).

Replacing diesel with the gas mixture results in a decrease in combustion rate due to the increased ignition delay and combustion duration, leading to low cylinder pressure levels.

3.3 Heat release rate

The combustion behavior in the engine operating on dual fuel with LPG at 40% of the rated load is shown in Figure 7.

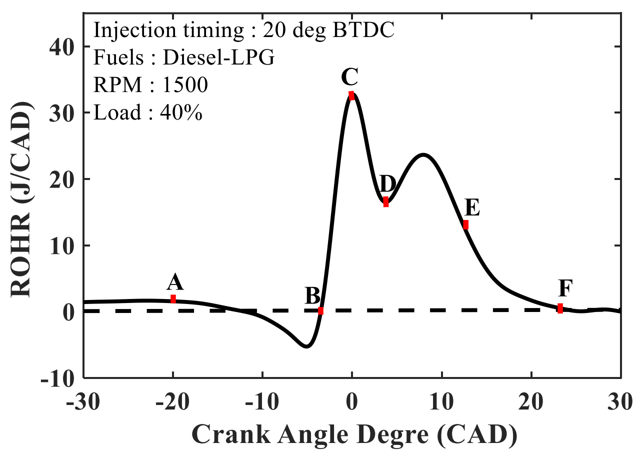

Figure 7 : The stages of dual-fuel Diesel-LPG combustion at 40% load

Analysis of the heat release rate diagram indicates that the combustion mechanism in dual-fuel diesel-LPG mode consists of four phases.

The first phase (AB) corresponds to the ignition delay of the pilot fuel (conventional diesel fuel). It refers to the period between the injection of the pilot fuel into the cylinder, during the compression stroke, and the onset of combustion. This phase is longer in dual-fuel mode than in pure diesel operation due to the reduced oxygen concentration resulting from the partial replacement of air by the gaseous fuel. The subsequent phase (BC) represents the rapid combustion of the premixed pilot fuel. It typically releases less heat than pure diesel operation due to the smaller quantity of diesel injected during dual-fuel mode. The phase (CD) corresponds to the ignition delay of the air-LPG gas mixture. There is a decrease in heat release until the start of combustion of the gas mixture. The gaseous premix combustion phase (DE) begins with flame propagation initiated by the spontaneous ignition of the pilot fuel. The diffusive combustion phase (EF) is characterized by the combustion of residual pilot fuel and LPG gas (Caligiuri & Renzi, 2017; Das et al., 2020; Sahoo et al., 2009).

Figure 8 shows the heat release rates derived from the cylinder pressure curves for different diesel fuel substitution rates under engine operating conditions at low (20% and 30%) and medium loads (40%, and 50%) using a diesel-LPG mixture.

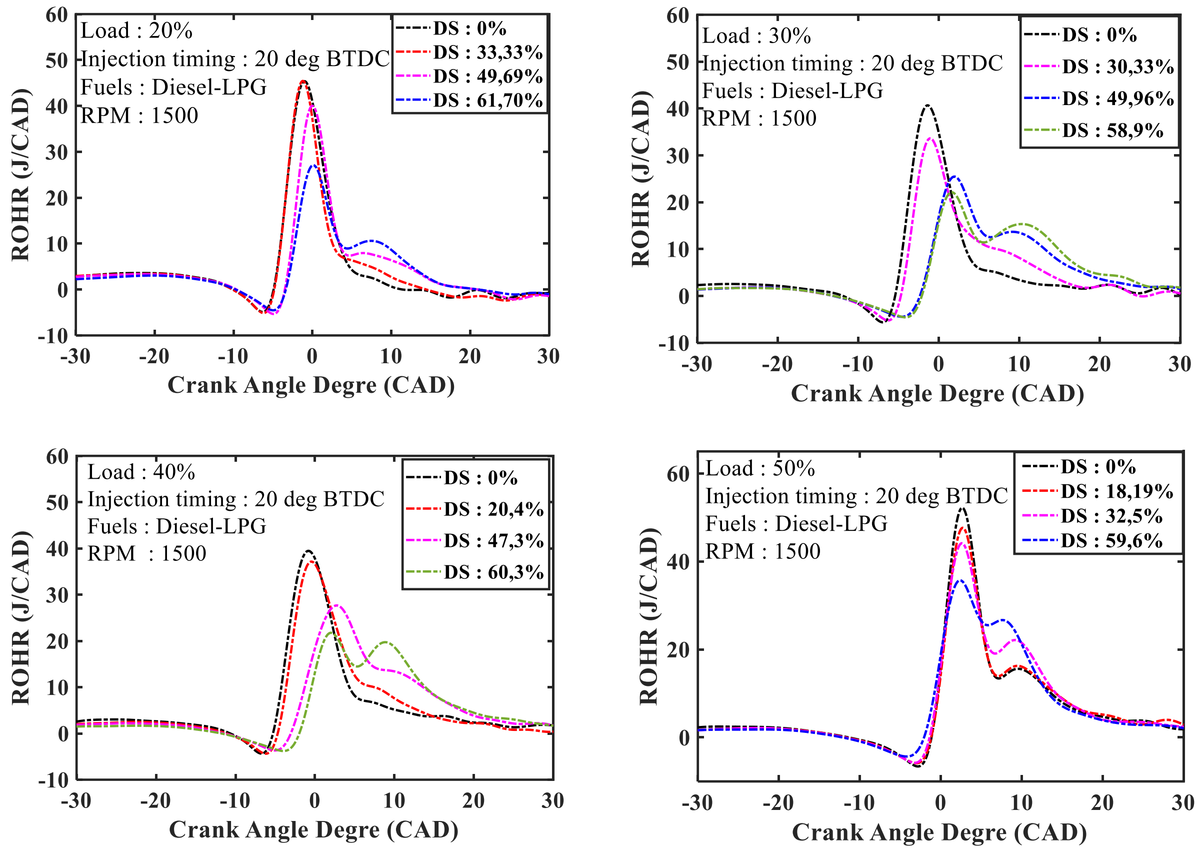

Figure 8 : Rate of heat release curves at low and medium loads for Diesel-LPG

With the pure diesel heat release rate curve serving as a baseline reference for each load condition, the corresponding diesel-LPG curves clearly delineate the three distinct phases of dual-fuel combustion across all examined loads.

A decrease in the peak heat release rate of the first stage of combustion is observed as the amount of gaseous fuel increases for all loads. This reduction is attributed to the lower mass of pilot diesel injected as the intake charge shifts toward a higher concentration of the air-LPG mixture. Furthermore, when the amount of gaseous fuel introduced into the cylinder becomes significant, the temperature and pressure decrease in the combustion chamber. This affects the flame propagation speed within the fuel. Consequently, the combustion duration is prolonged, and the combustion rate slows down, culminating in an overall decrease in the peak heat release rate. A trend that closely aligns with findings reported in similar dual-fuel studies (Mustafi et al., 2013; Papagiannakis & Hountalas, 2004; Saleh, 2008; Tira et al., 2012).

The peak heat release rate of pilot premix combustion tends to occur later as the amount of gaseous fuel introduced into the cylinder increases. This delay is significantly reduced at medium load. This is due to the longer ignition delay when the amount of gaseous fuel becomes substantial. The result is a widening of the reaction zone. As the load increases, temperature and pressure conditions improve in the combustion chamber, thereby reducing the delay in the onset of the heat release rate from the combustion of the diesel premix. The gaseous fuel burns primarily during the second stage of combustion. At the diesel fuel substitution limit corresponding to the maximum amount of gaseous mixture admitted into the cylinder, the richness of the gaseous premix promotes more efficient combustion, thereby leading to greater heat production during the second phase of combustion.

Under light-load operating conditions with a lean mixture, a substantial portion of the injected pilot fuel, as well as a certain amount of the gaseous fuel, may burn during the first stage of combustion.

The gaseous premix combustion phase is less intense for all loads studied. These observations can be attributed to diesel fuel substitution rates that are lower than those reported in the literature. Indeed, several authors have reported diesel fuel substitution rates of 70 to 90% in diesel engines operating in dual-fuel mode with renewable gaseous fuels such as LPG, natural gas, hydrogen, alcohols, and gasifier gas (Banapurmath et al., 2008; Bhattacharya et al., 2001; Bilcan et al., 2001; Bridgwater et al., 2002; Garnier et al., 2005; Mohammadi et al., 2006; Parikh et al., 1989; Ramadhas et al., 2006; Senthil Kumar et al., 2003; Singh et al., 2007; Sridhar et al., 2005; Uma et al., 2004). Furthermore, LPG's low cetane number makes it difficult to self-ignite, leading to significant cyclic variations and thus limiting its use in large quantities.

However, for all the mixtures examined, the diffusion combustion phase is more pronounced. In the diesel-LPG mixture, the longer ignition delay caused by the reduction in oxygen concentration in the cylinder as the gas content increases slows down the combustion rate. This results in a longer diffusion combustion phase.

3.4 Cyclic Dispersion

Figure 9 shows the coefficients of cyclic variation (COV) at the diesel substitution limit for the various feedstocks studied.

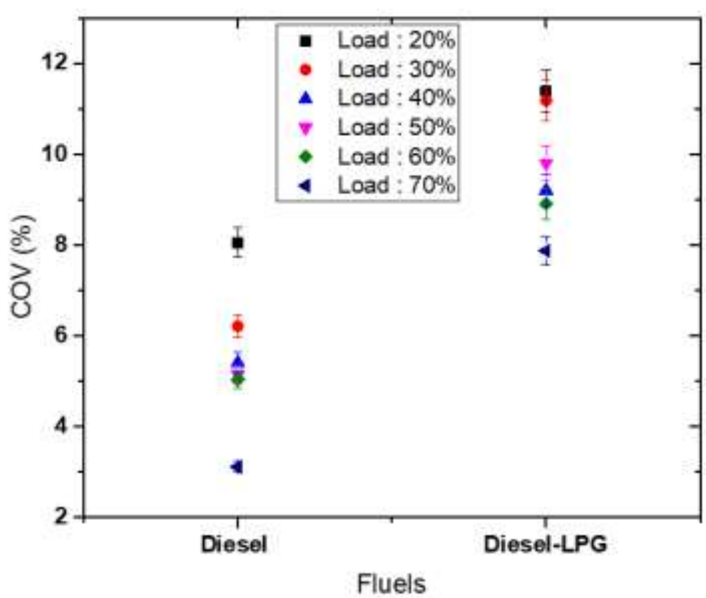

Figure 9 : The COV of IMEP variation in pure Diesel and dual-fuel mode at maximum Diesel substitution for different loads

The highest values of the coefficient variation of indicated average pressure (COVIMEP) are observed at lower loads for all engine operating modes. At lower loads, combustion irregularity stems from insufficient temperature and pressure levels in the cylinder. Self-ignition is more difficult, leading to longer ignition delays and making combustion unstable, as noted by several authors (Liu et al., 2022; Wang et al., 2016). Cyclic variability is more pronounced when the engine runs in diesel-LPG mode, sometimes exceeding the 10% threshold value (Heywood, 1988), beyond which combustion is considered unstable.". Combustion irregularity increases with the addition of gas (Santoso et al., 2012). Thus, at lower load (20%), the COVIMEP for diesel-LPG is 11.5% and decreases as the load increases, reaching 8.9% at medium load. These high COVIMEP values are linked to liquefied petroleum gas. It is established that LPG's low cetane number makes self-ignition difficult and leads to significant cyclic variations during its combustion in copious quantities in diesel engines (Peng et al., 2008; Qi et al., 2007; Santoso et al., 2012).

3.5 Ignition delay and combustion duration

Figure 10 shows the variation in ignition delay as a function of the substitution rate of diesel with liquefied petroleum gas at low load (30% load). Operating the engine in dual-fuel mode inherently prolongs the ignition delay compared to conventional diesel mode. As the amount of gas admitted into the cylinder increases, the delay becomes longer.

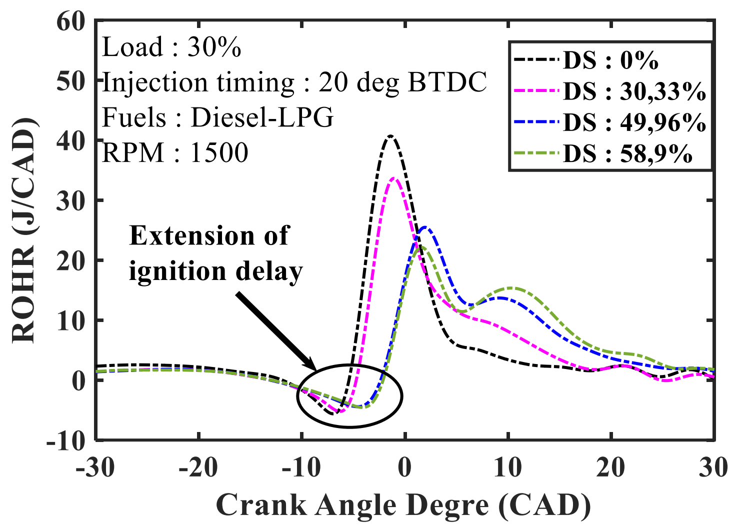

Figure 10 : Illustration of the extended ignition delay on the engine heat release rate curves for the D-GPLN2 at 30% load

Increasing the amount of gaseous fuel in the air-fuel mixture alters the properties of the charge admitted into the cylinder by reducing the oxygen concentration, which affects pre-ignition reactions during compression. This results in a longer ignition delay. Several authors (Poonia et al., 1998; Saleh, 2008; Surawski et al., 2014) have reached the same conclusions. An increase in the amount of gas injected into the cylinder lengthens the ignition duration and spreads out combustion.

For the same charge, the ignition delay is longer in dual-fuel mode compared to diesel fuel. The presence of gaseous fuel and the increase in its quantity with the charge cause a reduction in the oxygen concentration in the cylinder. This phenomenon leads to a deterioration of the reaction between diesel and air as well as those within the gaseous mixture (Dhole et al., 2016; Mustafi et al., 2013).

Figure 11(a) shows the evolution of the maximum combustion duration for diesel fuel substituted with LPG-diesel and conventional diesel as a function of load, and Figure 11(b) illustrates the combustion duration in dual-fuel mode at a 40% diesel substitution rate.

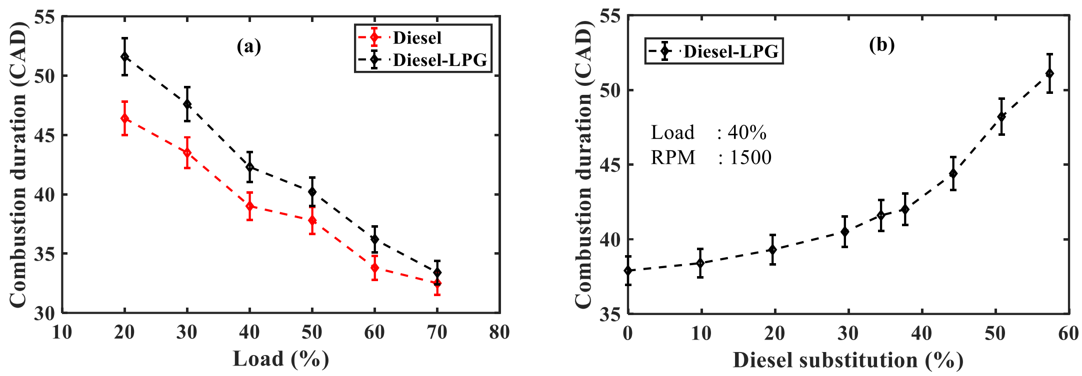

Figure 11 : Combustion duration: **(a)** as a function of load, with maximum Diesel substitution in dual-fuel mode; **(b** as a function of the Diesel substitution rate at 40% load in dual-fuel mode

The combustion duration is longer at low load but decreases as the load increases for all engine operating modes. This is due to the low temperatures and air-fuel ratios in the combustion chamber when the engine is operating at low load. In dual-fuel mode, the combustion duration is longer than that achieved with conventional diesel fuel. At 20% load, the combustion duration for diesel-LPG is 5.2°V longer than that of pure diesel fuel. This difference is reduced to 0.9°V when the engine load is 70%. These variations in combustion duration are related to the temperature and pressure conditions in the cylinder. The presence of gaseous fuel lowers the temperature and pressure in the cylinder, which leads to a reduction in the combustion rate. The low cetane number of LPG prolongs the ignition delay, leading to longer combustion durations. At constant load, the combustion duration increases with the increase in the amount of gaseous fuel (Figure 11 (b)).

When the engine is operating at 40% of its rated load, the combustion duration for diesel-LPG is 4°V higher than that for pure diesel under conditions where the diesel substitution rate is 38%. This difference increases by 1.5°V when the diesel substitution rate reaches 55%. Increasing the amount of LPG mixed with the intake air reduces the oxygen concentration of the fuel charge, thereby degrading its properties. This lengthens the ignition delay and slows the flame propagation in the lean air-fuel mixture. This effect is more pronounced under low-load operating conditions where temperature and pressure are lower. Dhole et al (Dhole et al., 2016) obtained similar results.

4. Conclusion

The main objective of this study was to evaluate the combustion parameters of a Lister Petter diesel engine operating in dual-fuel (diesel-LPG) mode, with a view to the future use of synthesis gas as the primary fuel for electricity generation in rural areas of Burkina Faso. The results indicate a diesel substitution rate of over 50% across all examined loads, with the highest values obtained at partial loads. . Furthermore, gas intake significantly affected pressure levels.in the combustion chamber. Maximum pressures at the diesel substitution limit are lower compared to those observed when the engine operates on conventional diesel. Similarly, exhaust gas temperatures are lower in diesel-LPG mode. The measured cylinder pressure data was used to develop heat release profiles. The heat release rate curves highlighted the three main phases of combustion in dual-fuel mode. Combustion is significantly affected by the low cetane number of liquefied petroleum gas. Ignition delays are longer, leading to irregular combustion cycles. Consequently, the values of cyclic dispersion are high and even exceed the threshold value of 10% at the maximum diesel substitution rate under light loads. Optimizing the performance of an internal combustion engine requires a thorough understanding of combustion parameters to minimize losses and reduce emissions. These findings provide a robust foundation for future research on this specific engine class, which is widely deployed across rural Burkina Faso. Ultimately, this work supports the optimization of decentralized power generation systems transitioning to low-energy-content gaseous fuels, such as syngas.

Conflicts of Interest: There are no conflicts to declare.

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