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Experimental Assessment of the Performance and Fine Particulate Matter Emissions of a LPG-Diesel Dual-Fuel Compression Ignition Engine

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Abstract: The present work is focused on the assessment of the performance and fine particulate matter emissions ($PM_{2.5}$) of a turbocharged four-cylinder direct injection diesel engine operating under dual-fuel mode with Liquefied Petroleum Gas (LPG). For load levels of 30%, 60% and 100%, measurements were taken, keeping the engine speed constant at 2200, 2500 and 3200 rpm, while the engine knock detonation was detected through a non-invasive internal system. According to experimental measurements, the abnormal knock combustion occurred at full load operation with a maximum LPG energy fraction of ~60%. The brake fuel conversion efficiency increased by 2.6% with an LPG energy fraction of 10%, where a fuel saving of 11.9% was achieved with respect to the diesel-only operation. The reduction of diesel consumption was around 50% with respect to 100% diesel operation at full load operations, where the highest brake fuel conversion efficiency was achieved. The brake fuel conversion efficiency decreased as LPG addition increased for all the engine loads. Regarding emissions, $PM_{2.5}$ decreased with the addition of LPG. However, HC and CO emissions increased as LPG injection was higher. NO_x emissions and exhaust gas temperatures were reduced for operation with higher LPG fractions, except for full load levels at 2200 and 2500 rpm.

Keywords: LPG; diesel; dual fuel; combustion engines; knock detection

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1. Introduction

Today the highest percentage of worldwide energy consumption comes from fossil fuels. Contaminants such as particulate material (PM), nitrogen oxides (NOx), carbon monoxide (CO) and hydrocarbons (HC) are latent combustion emissions and harmful to the environment. In addition, fossil fuel supply sources are not renewable energy and are in depletion. Thus, given the growing demand for this type of energy, it is necessary to find alternatives to replace energy consumption sustainably. In the area of internal combustion engines (IC), one of the solutions is the use of LPG since it is accessible and abundant in quantity [1]. In the case of spark-ignition engines, the LPG can operate as an alternative fuel, that is, 100% LPG due to its high-octane rating and under dual operation, which is more recommended for wear issues in the engine. In engines ignited by compression (CI), due to its high compression ratio, LPG cannot work as an alternative fuel; however, a dual operation is possible.

In compression-ignited engines running in dual-fuel mode, the primary fuel is the one that delivers the greatest contribution to the engine brake power. This fuel becomes LPG; diesel is called a pilot fuel, used in smaller quantities. In other words, it is the first fuel to burn and is responsible for igniting the air-LPG mixture. The LPG is mixed with the air in the intake manifold. Once this fuel-air mixture is formed, it enters the combustion chamber through the intake duct. To ignite this mixture, diesel injection into the combustion chamber is necessary [2]. A dual-fuel compression ignition engine's performance and emissions depend on its operating conditions, specifically on the injection parameters. Operating conditions correspond to speed, load level, amount of pilot fuel and injection time. It has been shown that as the amount of LPG supplied to the engine increases, its brake fuel conversion efficiency tends to decrease. However, for certain loads and proper LPG replacement, engine performance may improve. Regarding PM emissions, they are significantly reduced, NO_x decreases slightly at partial loads, and HC and CO increase with respect to diesel-only operation [3–6].

Brake fuel conversion efficiency depends on the level of charge, and the amount of LPG substituted [2,7,8]. It has been studied that brake fuel conversion efficiency worsens for an operation at low loads. However, for high loads, improvements have been registered. Viyay et al. [9] recorded a 4% improvement over the diesel-only operation. G.A. Rao et al. [7] demonstrated in a four-stroke single-cylinder diesel engine that brake fuel conversion efficiency increases for a load of 80% and for an energy replacement of LPG with diesel of 10% and 20%. On the other hand, for a 20% load level, as the replacement of LPG increases, brake fuel conversion efficiency decreases progressively, a decrease from 14.4% (diesel-only operation) to 9.7% (50% LPG energy replacement) for that level of charge. The same author in another study [8] confirms that at low loads, brake fuel conversion efficiency decreases as the energy replacement of LPG increases, and for high loads, brake fuel conversion efficiency increases to a certain percentage of LPG replacement.

Volumetric efficiency decreases as the replacement of LPG by diesel increases for all loads [1,2,4,8]. The volume of air inside the combustion chamber is displaced by the same gas [2,7]. Vezir et al. [9] recorded that, for a single-cylinder, four-stroke, and direct injection diesel engine, operating at full load and replacement percentages of LPG 5, 10, 15 and 25% on a mass basis, volumetric efficiency decreases.

Specific fuel consumption decreases with a dual LPG operation for high loads, and this statement is supported by various studies [1,9,10]. On the other hand, for low loads, there has been an increase in specific fuel consumption [1,7,8], making the engine less efficient. Dong [1] indicates an average reduction of 8% operating at full load with a diesel engine in a dual mode with respect to its diesel-only operation. However, for a load of 25% and a 20% amount of LPG in operation, there is a slight increase in the specific fuel consumption compared to its normal operation.

The NO_x emissions in an engine are determined by the type of fuel and, therefore, by the combustion temperature [10]. Therefore, if there is no significant change in these factors with dual engine operation, there will not be much difference in these emissions compared to their conventional operation. Dong et al. [1] conducted tests according to the national standard procedure (GB 17691-1999 China) and demonstrated that regardless of the engine load level, NO_x emissions decrease slightly for most test modes, except for mode eight at full load. This is explained by an increase in the internal pressure in the combustion chamber that results in an increase in its temperature and, consequently, a higher level of these emissions. Several studies support that the NO_x emissions of the engine for partial loads decrease with respect to its diesel-only operation [1,7,11–13]. On the other hand, studies carried out at full load have registered that NO_x emissions tend to increase as the amount of LPG increases [1,9,13,14].

The emissions of HC with a dual operation increase with respect to emissions with a diesel-only operation [1,10–12]. Emissions increase at a higher level of LPG replacement and especially at low loads. D. H. Qi et al. [12] recorded an increase in these emissions for low loads as the mass percentage of LPG increases and similar to its diesel-only operation for high loads. Several authors explain that this behavior is due to incomplete combustion of LPG [1,10,11]; there have been problems such as ignition delays for low loads, which directly affects the quality of combustion [1] emitting unburned hydrocarbons present in the LPG [11]. The main reason for this emission is the gas's temperature in the combustion chamber that is lower at lower loads and because there is an increase in the amount of LPG. In addition, LPG contains aromatic hydrocarbons that are very stable to burn completely. Suitable spraying can reduce the mixed fuel near the cylinder walls, significantly reducing HC emissions [12].

CO emissions behavior is very similar to HC, except that at full load, these emissions have been practically the same when comparing a dual operation with LPG to the diesel-only operation and could even be reduced [1,10,12]. The main reason for producing this gas is the temperature of the mixture and the availability of unburned gases [11,12]. In addition, with high amounts of LPG, oxygen is reduced in the combustion chamber, and it is difficult for the conversion of CO to CO_2 to occur [11]; in other words, if the fuel-air ratio is reduced, there will be an increase in these emissions [14]. At high loads, due to the instantaneous boiling of the injection, there is an improvement in the efficiency of the fuel-air mixture, and it gives the mixture more opportunities to have contact with the air [12]; this explains a lower amount of CO emissions.

With respect to PM emissions, for all operating conditions, there is a decrease in its emissions when the engine is running in dual mode [1,15]. For example, Dong et al. [1] recorded that for high loads, there is a decrease greater than 50% of PM of the engine running under dual operation with respect to the baseline. The greater the amount of LPG used, the greater the decrease in these emissions.

The challenges for a compression-powered engine running dual are to improve its thermal performance and its HC and CO emissions, especially for an operation at low loads [16–18]. Another is to improve uncontrolled combustion for high loads [18].

LPG has a high tendency to auto-ignition since its auto-ignition temperature is low; for high loads, this produces an increase in the pressure rate during combustion, leading to knocking [19]. Studies confirm that the knocking phenomenon is mainly due to the autoignition of the mixture at an outdated time [18,20,21]. The EGR and the throttling of admission can eliminate the knock [22]. This is evident by increasing the addition of LPG as a result of the high burn rate [23]. In addition, the knock and ignition limits are strongly linked to the type of gas and its concentrations [24].

The present research used a turbocharged, four-cylinder direct injection diesel engine to:

- Determine the maximum diesel substitution by LPG until the knock phenomenon at different engine loads without any engine modification.
- Evaluate the impact of LPG substitution on emissions $PM_{2.5}$ and the performance of the engine.
- Measure the diesel saved when the engine runs in dual-fuel mode.

2. Experimental Setup

2.1. Test Cell

In this investigation, a Perkins Prima M80T Turbocharged marine diesel engine (Specifications in Table 1) was used to dual-operate with LPG-diesel. To regulate the load level of the engine, a 150-kW hydraulic dynamometer was used coupled to the engine, which can be seen in the experimental setup of Figure 1 and physically in Figure 2. The dynamometer has an analog indicator engine torque, which is regulated by a lever system. Regarding fuels, low-sulfur commercial diesel was used. The LPG was supplied by a 45 kg cylinder with an internal pressure of 100 psi. The way to control the flow of LPG was through a pressure regulator maintaining an outlet pressure between 0.1 and 2.0 bar. To measure the amount of LPG, an Aalborg GFC-57 mass flow controller was used, integrated into the injection line to the engine towards the air intake.

Table 1. Engine specifications (© SAE International).

Engine Model	Perkins Prima M80T Turbocharged
Cycle	4 stroke
Cylinders configuration	4 in line
Bore × Stroke	84.5 mm × 88.9 mm
Displacement	1994 cm^3

Table 1. *Cont.*

Engine Model	Perkins Prima M80T Turbocharged
Compression ratio	17.2:1
Power rating	58 kW at 4500 rpm
Torque rating	145 N-m at 2350 rpm
Injection system	Direct injection system, Bosch VE pump with cold start compensation and electric stop.
Injection timing	18° before TDC
Inlet valve opening	7° after TDC
Inlet valve closing	36° after TDC
Exhaust valve opening	36° before BDC
Exhaust valve closing	10° before TDC

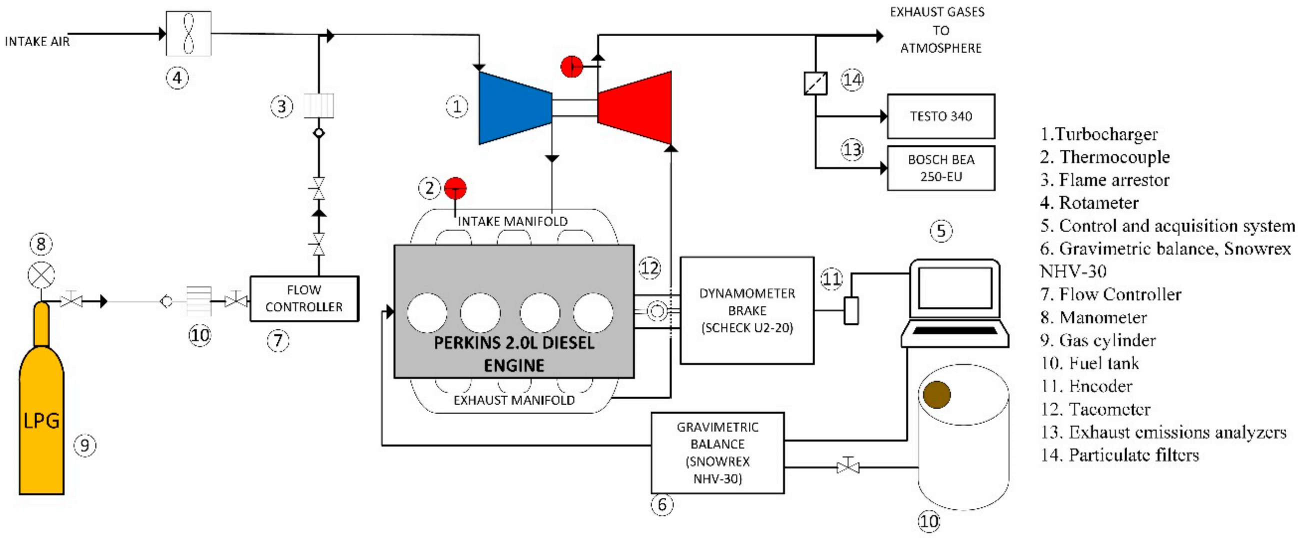
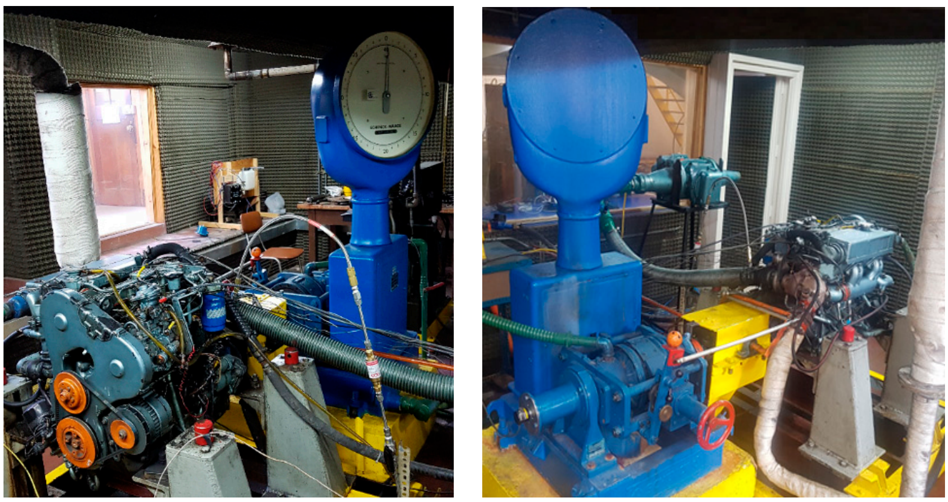


Figure 1. Experimental setup.



(**a**) (**b**)

Figure 2. UTFSM test engine facility: (**a**) Front view and (**b**) back view.

Three shut-off valves were installed in the injection line for safety and flow interruption in case of emergency. The engine had thermocouples that measured the temperature of the oil, exhaust gases, intake air, coolant, and cooling water under different operating points. A gravimetric balance Snowrex NHV-30 was used for fuel consumption, making measurements for every second of a diesel tank in which the fuel is extracted. Air consumption was measured using a rotameter model Aerzen D74, and the boost pressure was with a Hg-column manometer. The block vibration signal was measured with a commercial Bosch knock sensor mounted in the upper part of the engine block near cylinder 1. The knock sensor signal and the Omron E6B2-CWZ6C encoder were recorded with a DT9816-S data acquisition system using the QuickDAQ software version 1.6. The knock intensity was measured through a novel technique, which uses the Pondered Deviation from the Reference index defined as given in Equation (1).

$$PD_r = \frac{\sqrt{\frac{1}{M}\sum_{i=0}^{M-1}\left(KI_i - \overline{KI}_r\right)^2}}{\sigma_r} \tag{1}$$

KI_i is the knock intensity of the i-th combustion cycle, $\overline{KI}_r$ is the mean of reference knock intensity, σ_r is the standard deviation of reference knock intensity, and M is the number of KI values considered to compute the index. For details, see [25].

Two measurement phases were carried out, and two different types of equipment were used to measure polluting emissions. In the first phase of measurements, the Testo 340 device was used to measure O_2, CO_2, NO_x and CO, and the measurement of HC, Bosh device was used. On the other hand, for the second measurement phase, focused on the analysis of $PM_{2.5}$, the particulate material was measured by a gravimetric method using a non-motorized passive device called Cartridge, in which quartz filters are installed for the retention of $PM_{2.5}$. Table 2 summarizes the uncertainties of the measurements. Knock detection was performed using a non-invasive method based on the analysis of structural vibration.

Table 2. Uncertainties of measurements.

Item	Uncertainty
Torque	±0.7 N-M
Engine Speed	±1 Rpm
LPG Volume Flow Rate	±3 L/Min
CO	±1 Ppm Vol
NO_x	±1 Ppm Vol
HC	±1 Ppm Vol
CO_2	±1 Ppm Vol
O_2	±1 Ppm Vol
SNOWREX	±0.001 Gr

2.2. LPG Substitution

From the total input power, theoretical replacement percentages to be injected sequentially are calculated. This percentage of power translates into a volumetric flow rate of LPG, which is measured by the mass flow controller. The LPG energy fraction is defined as the ratio between the percentage of LPG energy that entered the engine over the total energy input. Mathematically, the energy fraction is defined as given in Equation (2). In addition, diesel fuel savings is calculated by Equation (3) as the brake-specific diesel fuel consumption (Bsfc).

$$\text{Fraction } (-) = \frac{LHV_{LPG} * \dot{m}_{LPG}}{LHV_{diesel} * \dot{m}_{diesel} + LHV_{LPG} * \dot{m}_{LPG}} \tag{2}$$

$$\text{Bsfc } (\text{g/kWh}) = \frac{\dot{m}_{diesel}}{P_{brake}} \tag{3}$$

2.3. Experimental Procedure

In the first place, a baseline of the engine was determined, operating only with diesel at certain operating points. To compare, under the same conditions, a dual operation with LPG. The operating points were at three different speeds (2200, 2500, and 3200 rpm) under three engine load levels at each speed (30%, 60%, and 100%). For each operating point, three measurements were carried out randomly for statistical purposes. In the second phase, the baseline considered only 2500 rpm engine speed; in this regime, LPG was injected in different percentages (0%, 10%, 20% and 30%), three replicas of each operation mode, recording the measurement of $PM_{2.5}$. Diesel fuel injection was carried out automatically by the fuel pump that was supplied depending on the operating point, either the engine running on only diesel or dual mode. The brake fuel conversion efficiency was calculated as follows:

$$Bfce = \frac{P_{brake}}{\dot{m}_{diesel,dual} LHV_{diesel} + \dot{m}_{LPG} LHV_{LPG}} \tag{4}$$

Regarding the supply of LPG, its flow was controlled and adjusted according to the partial energy requirement. The total energy input to the engine was defined in operation with only diesel, which corresponds to the baseline. From that energy, replacement energy percentages were calculated in multiples of 10. These percentages define the flow rate of the LPG required to replace the energy corresponding to diesel. In the dual operation process, the engine was adjusted with the same power output as in the baseline, and LPG was injected at different replacement percentages. For each percentage of substitution, three measurements were made.

3. Results and Discussion

3.1. Dual Operation Limits

The limits of operation under dual operation with LPG-diesel are predominantly twofold. The first one is the vaporization rate of the gas cylinder. This factor clearly depends on the ambient temperature and the amount of gas in the cylinder; the better the vaporization rate of the cylinder, the greater amount of LPG to be injected, especially at the full load where the torque is greater. In this work, there were no insufficiencies with the flow of LPG because the cylinder had an adequate vaporization ratio. The second reason is due to the vibrational level of the engine measured through the PDr index. The PDr as a function of the LPG energy fraction is shown in Figure 3 at different engine loads and speeds. The knock phenomenon occurs at full load, from an energy fraction of 50%, becoming severe at an energy fraction of 60%. The high knock tendency is caused by the low cetane number of the LPG (low ignitability) and a high compression ratio of the engine, which induces high temperature in the combustion chamber during the compression stage. According to thermocouple measurements, the highest temperatures are obtained at higher loads which promotes knock when the engine runs in dual-fuel mode. Specifically, the highest level of knock was reached at 2500 rpm, where the highest brake fuel conversion efficiency was obtained, implying a higher combustion temperature [2,14,16,18,19,24].

In contrast, at lower engine loads, a knock was not perceived since the brake fuel conversion efficiency decreases considerably, which implies that a fraction of the LPG may not burn totally [2,6,7,10–12]. In conclusion, the amount of LPG injection was limited, taking as a reference the engine performance and emissions' worsening.

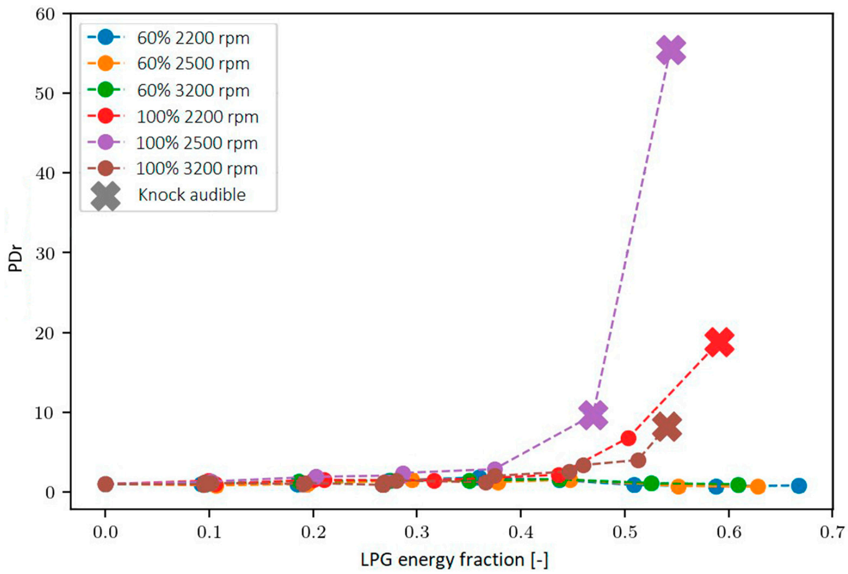


Figure 3. PDr vs. LPG energy fraction at different engine loads and speeds.

3.2. Engine Performance

According to Figure 4, the brake fuel conversion efficiency decreases for an operation at low loads. For both 30% and 60% engine load, there is a decrease of 16% in brake fuel conversion efficiency as the energy fraction of LPG is increased. This is because for larger fractions of LPG, the amount of pilot fuel decreases, and the greater amount of diesel, the better and stronger the ignition leading to better combustion [2,18–20]. For a full load operation, it is increased up to a certain fraction of LPG. At 2200 rpm, brake fuel conversion efficiency increases from 37.2% to 38.1% (which means a 2.6% increase) for a 20% substitution. Similarly, for a speed of 2500 rpm, there is an increase from 37.9% to 38.9% (which means an increase of 2.6%) for substitution of 10%, and finally, for a speed of 3200 rpm, there is an increase from 35.4% to 36.0%, (which means a 1.7% increase) for a 10% replacement of diesel with LPG. In addition, it should be noted that the highest brake fuel conversion efficiency was obtained for a speed of 2500 rpm for small and larger LPG fractions. For a 60% fraction of LPG, brake fuel conversion efficiency drops dramatically when knock occurs for an operation with a speed of 2200 and 3200 rpm, however, for a speed of 2500 rpm, there is a decrease in brake fuel conversion efficiency, but it is not as evident as the other cases. That is also explained by the instability during the engine operation [26].

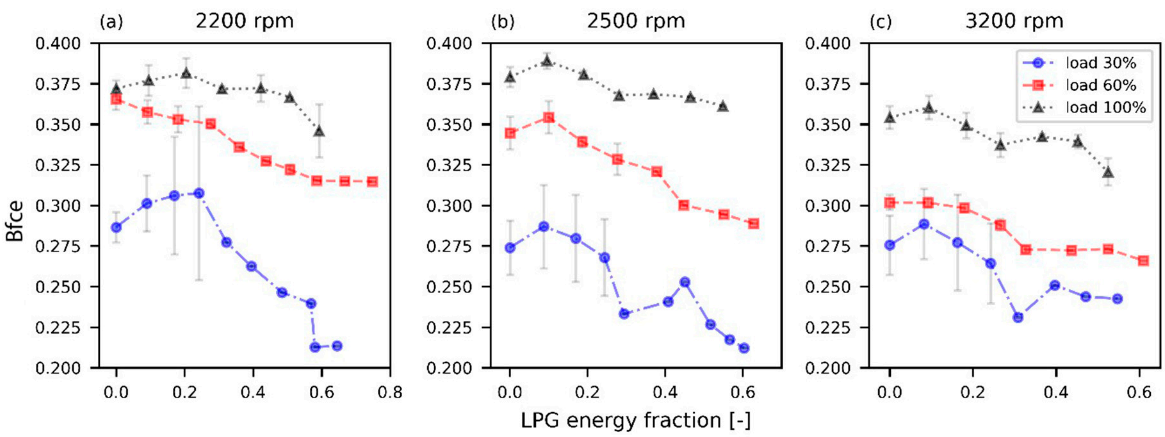


Figure 4. Brake fuel conversion efficiency v/s LPG energy fraction at (**a**) 2200 rpm, (**b**) 2500 rpm, and (**c**) 3200 rpm.

With respect to the saving fuel consumption, according to Figure 5, an effective saving of diesel fuel is clearly seen when the engine operates in dual-fuel mode. In the case of higher brake fuel conversion efficiency, at 2500 rpm, full load and 10% of diesel substitution, a saving of 11.9% were achieved compared to its normal operation with diesel fuel only. For the maximum substitution registered in dual operation, an average saving of 54.3% of diesel was achieved, independent of the level of load and engine speed. Diesel fuel savings are clearly explained by reducing the amount by adding LPG to the engine to maintain the total input power of a diesel-only operation. For reference, Figure 6 shows the equivalence ratio used in the experimental tests, which is higher (less air) the higher the load.

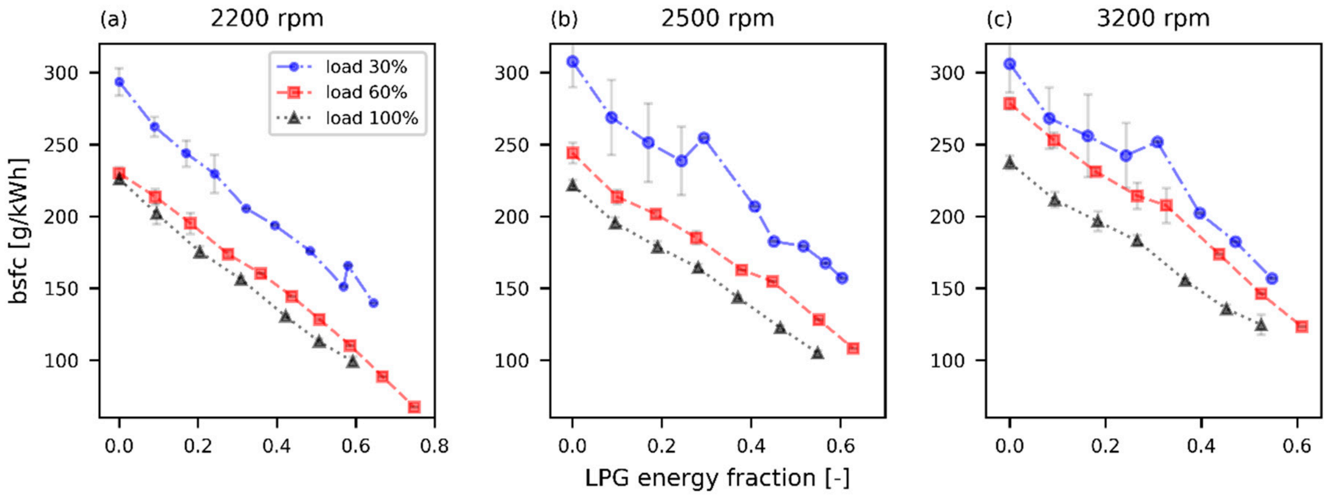


Figure 5. Brake-specific diesel fuel consumption (bsfc) v/s LPG energy fraction at (**a**) 2200 rpm, (**b**) 2500 rpm, and (**c**) 3200 rpm.

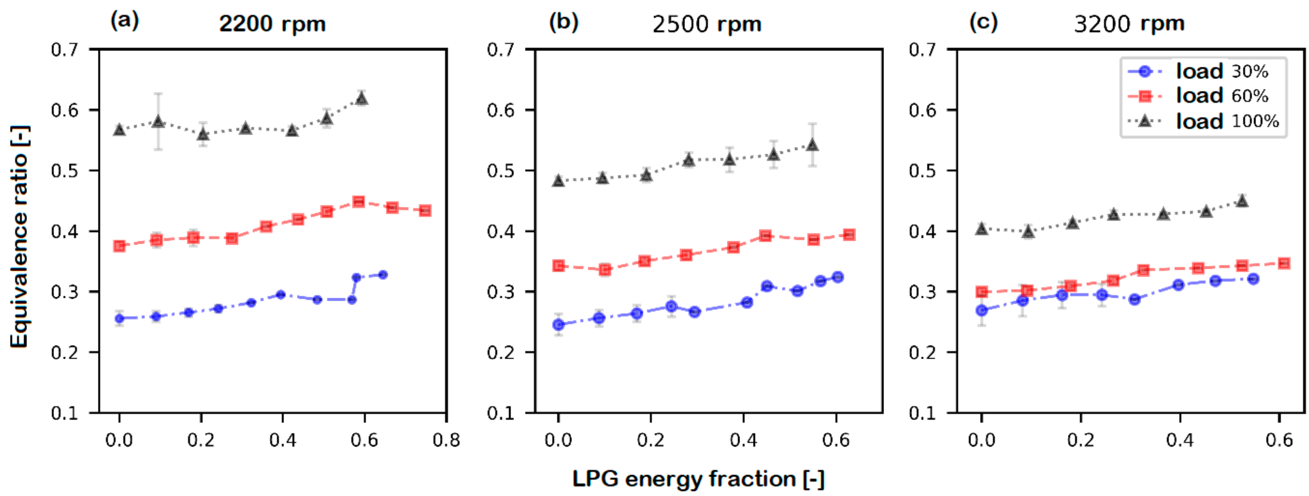


Figure 6. Equivalence ratio v/s LPG energy fraction at (**a**) 2200 rpm, (**b**) 2500 rpm, and (**c**) 3200 rpm.

Table 3 highlights the most important findings for PDr, Bfce and diesel fuel substitution for dual operation.

Table 3. Most important findings for PDr, Bfce and diesel reduction consumption for dual operation.

Speed (rpm)	Load Level (%)	LPG Energy Fraction (%)	PDr (-)	Bfce (%)	Diesel Reduction Consumption (%)
2200	30	20		38.1	16.9
		60			40.1
	60	60			44.1
	100	59	knock		
		60			56.1
2500	30	10		38.9	
		60			40.6
	60	60			47.5
	100	55	knock		
		60			52.6
3200	30	10		36.0	
		60			67.8
	60	60			47.5
	100	53	knock		
		60			47.5

3.3. Emissions

The behavior of HC emissions can be seen in Figure 7; as demonstrated, these emissions increase with the addition of LPG in the engine. The rate of variation of the increase depends on two operational factors: the speed and the level of engine load. There is a decrease in the rate of HC as the speed of rotation is greater and, in the same way, when operating with a higher level of load. HC emissions are higher at low loads due to incomplete combustion (low combustion temperature); also, an increase in the ignition delay occurs as the amount of LPG injected increases, which influences the combustion of the mixture [1,2,7,10,11,14].

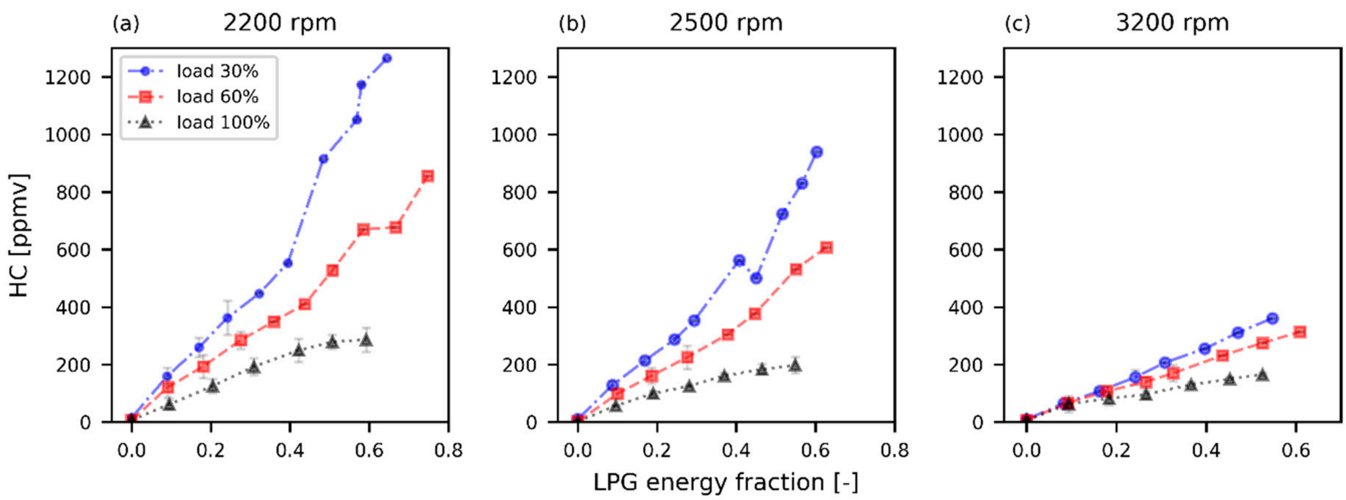


Figure 7. HC emissions v/s LPG energy fraction at (**a**) 2200 rpm, (**b**) 2500 rpm, and (**c**) 3200 rpm.

The CO emissions are shown in Figure 8. These emissions increase by adding LPG to the engine. These emissions are emitted in greater quantity at low engine loads. The rate of formation of these emissions can be reduced for a full load operation from a certain LPG fraction. CO emissions behavior is similar to that of HC. Due to this, the same reasons (incomplete combustion) that explain this behavior are shared. This is generally due to poor combustion quality for low loads to generate unburned emissions present in the LPG [1,2,7,10,11].

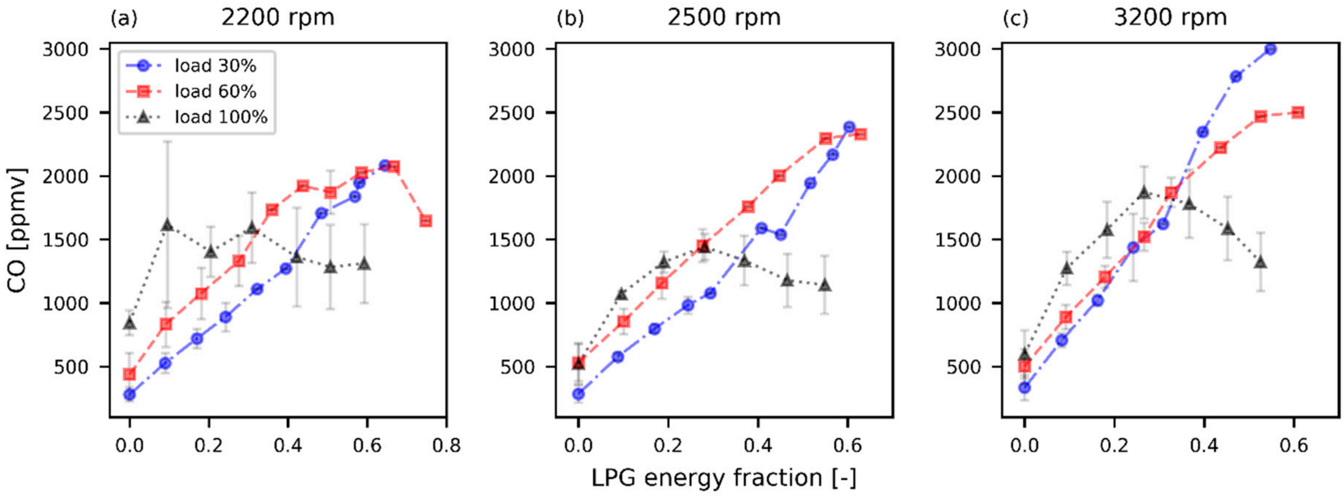


Figure 8. CO emissions v/s LPG energy fraction at (**a**) 2200 rpm, (**b**) 2500 rpm, and (**c**) 3200 rpm.

The NOx emissions, shown in Figure 9, show a decrease in these emissions for low loads as the percentage of LPG to the engine increases and the exhaust gas temperature decreases (Figure 10). This is due to the ignition delay that causes poor combustion and a low combustion temperature for those charge levels [1,2,12,16,18]. For maximum LPG substitutions, it is possible to reduce, on average, by 34%. However, for high loads, except for the graph of Figure 9c, these emissions increase. For maximum brake fuel conversion efficiency, a boost for NO_x of 1.7% is recorded for a speed of 2200 rpm and a decrease of 4.6% and 4.7% at 2500 rpm and 3200 rpm, respectively. At full load, NO_x emissions increases while exhaust gas temperature decreases for operation with higher LPG fractions at 2200 and 2500 rpm. That is explained by the equivalence ratio used, considering that it is greater than the 30% and 60% load (Figure 6).

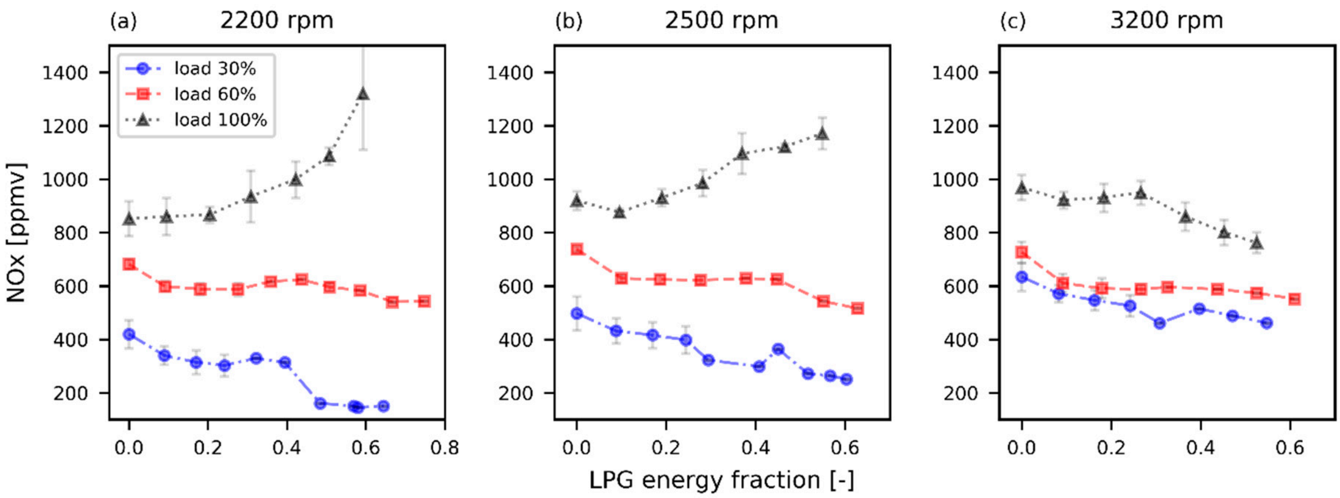


Figure 9. NOx emissions *v*/*s* LPG energy fraction at (**a**) 2200 rpm, (**b**) 2500 rpm, and (**c**) 3200 rpm.

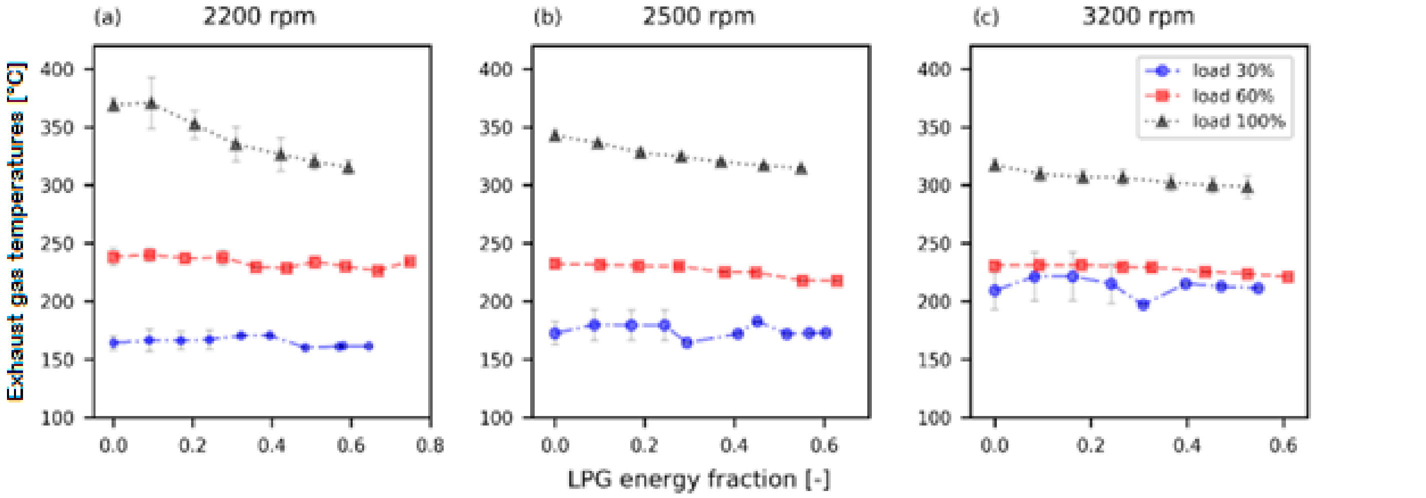


Figure 10. Exhaust gas temperatures v/s LPG energy fraction at (**a**) 2200 rpm, (**b**) 2500 rpm, and (**c**) 3200 rpm.

Table 4 highlights the most important findings for gaseous emissions in dual operation.

Table 4. Most important findings for gaseous emissions in dual operation.

Speed (rpm)	Load Level (%)	LPG Energy Fraction (%)	Equation Ratio (-)	CO (ppmv)	NO_x (ppmv)	HC (ppmv)	Exhaust Gas Temperature (°C)
2200	30	20	0.27	719	315	260	166.3
		60	0.18	1708	161	915	160.2
	60	60	0.26	1872	597	528	234.1
	100	59	knock				
		60	0.62	1310	1320	287	315.4
2500	30	10	0.26	577	432	128	179.8
		60	0.2	1539	364	501	182.9
	60	60	0.22	2297	534	532	218.0
	100	55	knock				
		60	0.54	1144	1171	199	314.8
3200	30	10	0.29	707	571	65	221.4
		60	0.2	2784	489	311	213.0
	60	60	0.2	2470	574	275	223.8
	100	53	knock				
		60	0.45	1324	762	165	298.6

Regarding the second phase, the LPG energy fraction, emission factors of $PM_{2.5}$, standard deviation (SD) and Pearson variation coefficient (VC) are found in Table 5. The VC indicates the variation of a sample independent of its magnitude. When this value is close to zero, it indicates a homogeneous sample, but if it is close to one, it indicates a heterogeneous sample. Given this definition, it can be concluded that although each percentage of GLP was repeated three times, the sample is homogeneous for all the LPG fractions. The emission factor data is plotted in Figure 11 for better visualization.

Table 5. $PM_{2.5}$ measurements in experiments.

Speed (rpm)	LPG Energy Fraction	Emission Factors (µg/kWh)	SD (-)	VC (-)	Average of Emission Factors (µg/kWh)
2500	0	119.0	0.1431	2.96×10^{-5}	89.5
	0	89.7	0.1427	2.96×10^{-5}	
	0	59.8	0.1441	2.96×10^{-5}	
	0.1	59.8	0.1445	2.99×10^{-5}	44.7
	0.1	0.0	0.1438	2.99×10^{-5}	
	0.1	29.6	0.1391	2.99×10^{-5}	
	0.2	59.5	0.1388	1.72×10^{-5}	39.6
	0.2	29.7	0.1424	1.72×10^{-5}	
	0.2	29.7	0.1369	1.72×10^{-5}	
	0.3	59.5	0.1436	8.66×10^{-5}	109.2
	0.3	209.0	0.1412	8.66×10^{-5}	
	0.3	59.1	0.1447	8.66×10^{-5}	

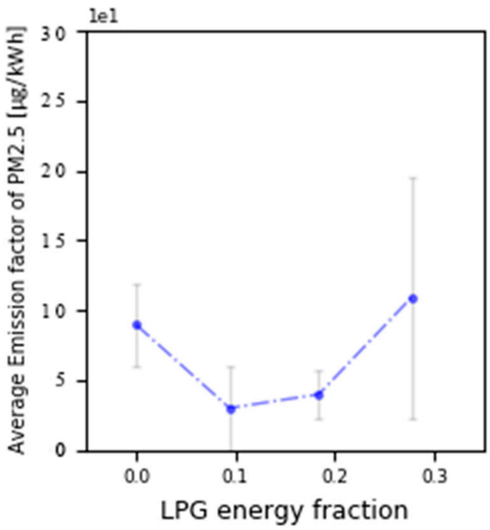


Figure 11. The average emission factor of $PM_{2.5}$ at 2500 rpm and full load conditions.

It can be concluded there is a decrease in specific $PM_{2.5}$ emissions by adding LPG to the engine in increasing amounts. For example, for a 10% replacement of diesel energy fraction by LPG, a decrease in $PM_{2.5}$ of around 50% was achieved. This is clearly due to the decrease in diesel use by increasing the addition of LPG, decreasing the emissions resulting from the combustion of diesel as $PM_{2.5}$ [1,7,8]. LPG fraction of 0.3 has the greatest experimental error due to the instability during the engine operation [26].

4. Conclusions

The combustion and emissions characteristics of a turbocharged four-cylinder direct injection diesel engine operating under dual-fuel mode with liquefied petroleum gas were investigated to determine the maximum diesel substitution until the knock phenomenon was detected.

The major findings can be summarized as follows:

- LPG diesel dual-fuel mode operation was possible without knocking in operations at low and medium loads. However, knock occurred at full load, both for a rotation regime of 2200 rpm, 2500 rpm and 3200 rpm, with an LPG energy fraction of 59%, 55% and 53%, respectively. The high knock tendency is caused by the low cetane number

of the LPG and a high compression ratio of the engine, which induce high temperature in the combustion chamber during the compression stage.
- A small increase in engine brake fuel conversion efficiency was obtained, especially at full load, but it decreases as the LPG energy fraction increases. This is explained by the fact that for larger fractions of LPG, the amount of pilot fuel decreases, and the greater the amount of diesel, the better and stronger the ignition, which leads to better combustion. If the fuel savings obtained for the maximum fractions of LPG are considered, a saving of 54.3% of specific fuel consumption is achieved.
- The emissions of HC and CO increased as a larger fraction of LPG was injected into the engine. The rate of formation of HC emissions increases while the load level is lower and similarly increases if the engine speed is lower. CO emissions have a dependence on the level of engine load; at full load, these emissions can decrease. These emissions are due to incomplete combustion (low combustion temperature). In addition, an increase in the ignition delay occurs as the amount of LPG injected increases, which influences the combustion of the mixture.
- NOx emissions can be reduced depending on the engine operating conditions. At low loads, these emissions decrease while the exhaust gas temperature decreases. However, at full load operation, they could increase while the exhaust gas temperature decreases. That is explained by the equivalence ratio used, considering that it is greater (less air) than the low and medium load.
- $PM_{2.5}$ tends to decrease as the fraction of LPG entering the engine increases.

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Nomenclature

Bsfc	Brake specific diesel fuel consumption
Bfce	Brake fuel conversion efficiency
CO	Carbon monoxide
CO_2	Carbon dioxide
CI	Compression ignition
EGR	Exhaust gas recirculation
HC	Hydrocarbon
IC	Internal combustion
LHV	Low heating value
LPG	Liquified petroleum gas
m	Mass flow rate
NO_x	Nitrogen oxide
P_{brake}	Power brake
PDr	Pondered deviation from reference index
$PM_{2.5}$	Fine particulate matter
rpm	Revolutions per minute
SD	Standard deviation
VC	Variation coefficient

References

1. Jian, D.; Xiaohong, G.; Gesheng, L.; Xintang, Z. Study on Diesel–LPG Dual-fuel Engines. *SAE* **2001**. [CrossRef]
2. Ashok, B.; Ashok, S.D.; Kumar, C.R. LPG diesel dual-fuel engine—A critical review. *Alex. Eng. J.* **2015**, *54*, 105–126. [CrossRef]
3. Waldie, A.; Jay, G.; Holland, D. The risk posed to vehicle occupants and rescue personnel by dual-fuelled vehicles fitted with liquid petroleum gas (LPG) tanks. *SAE* **2001**. [CrossRef]
4. Poonia, M.P.; Bhardwaj, A.; Jethoo, A.S.; Pandel, U. Experimental investigations on engine performance and exhaust emissions in an LPG diesel dual-fuel engine. *Int. J. Environ. Sci. Dev.* **2011**, *2*, 418–422. Available online: http://www.ijesd.org/papers/162-D557.pdf (accessed on 18 August 2022). [CrossRef]
5. Negurescu, N.; Pana, C.; Cernat, A. Theoretical and experimental investigations on the LPG fuelled diesel engine. In *Proceedings of FISITA 2012 World Automotive Congress*; Springer: Berlin/Heidelberg, Germany, 2013; Volume 191, pp. 37–49. [CrossRef]
6. Tiwari, D.R.; Sinha, G.P. Performance and emission study of LPG diesel dual-fuel engine. *Int. J. Eng. Adv. Technol.* **2014**, *3*, 2249–8958.
7. Rao, G.A.; Raju, A.V.S.; Rao, C.V.M.; Rajulu, K.G. Effect of LPG content on the performance and emissions of a diesel–LPG dual-fuel engine. *Bangladesh J. Sci. Ind. Res.* **2011**, *46*, 195–200. [CrossRef]
8. Rao, G.A.; Raju, A.V.S.; Rajulu, K.G.; Rao, C.V.M. Performance evaluation of a dual-fuel engine (diesel + LPG). *Indian J. Sci. Technol.* **2010**, *3*, 235–238. [CrossRef]
9. Ayhan, V.; Parlak, A.; Cesur, I.; Boru, B.; Kolip, A. Performance and exhaust emission characteristics of a diesel engine running with LPG. *Int. J. Phys. Sci.* **2011**, *6*, 1905–1914. [CrossRef]
10. Kumaraswamy, A.; Prasad, B.D. Use of LPG in A dual-fuel engine. *Int. J. Mod. Eng. Res.* **2012**, *2*. Available online: https://www.semanticscholar.org/paper/Use-of-LPG-in-A-Dual-Fuel-Engine-Kumaraswamy-Prasad/09087e2e262b058b9908a56dfa5c6da3a04cf346 (accessed on 18 August 2022).
11. Mirgal, N.; Kumbhar, S.; Ibrahim, M.M.; Chellapachetty, B. Experimental investigations on LPG–Diesel dual fuel engine. *J. Chem. Pharm. Sci.* **2017**, *10*, 211–214. Available online: https://www.semanticscholar.org/paper/Experimental-investigations-on-LPG-diesel-dual-fuel-Mirgal-Kumbhar/3c0541ec935d0706a09e3b7bd4f9cd519e867f1e (accessed on 18 August 2022).
12. Qi, D.H.; Bian, Y.Z.H.; Ma, Z.H.Y.; Zhang, C.H.H.; Liu, S.H.Q. Combustion and exhaust emission characteristics of a compression ignition engine using liquefied petroleum gas–diesel blended fuel. *Energy Convers. Manag.* **2007**, *48*, 500–509. [CrossRef]
13. Sudhir, C.V.; Desai, V.; Kumar, S.Y.; Mohanan, P. Performance and Emission Studies on the effect of Injection Timing and Diesel Replacement on a 4-S LPG–Diesel Dual-Fuel Engine. *SAE* **2003**. [CrossRef]
14. Le, T.A.; Nguyen, T.T. Experimental Study on Performance, Emissions and Combustion Characteristics of a Single Cylinder Dual-fuel LPG/Diesel Engine. *Appl. Energy* **2011**, *88*, 3959–3968. [CrossRef]
15. Van Dzung, D.; Tung, T.T.H. Study on the Effect of the Fuel Supply Mode in Dual-Fuel Engine (LPG-Diesel). Ph.D. Thesis, Da Nang University, Da Nang, Vietnam, 2018.
16. Poonia, M.P.; Ramesh, A.; Gaur, R.R. Experimental Investigation of the Factors Affecting the Performance of a LPG-Diesel Dual-fuel Engine. *SAE* **1999**, *108*, 499–508. [CrossRef]
17. Mohanan, P.; Kumar, Y.S. Effect of LPG Intake Temperature, Pilot Fuel and Injection Timing on the Combustion Characteristics & Emission of a LPG Diesel Dual-fuel Engine. *SAE* **2001**. [CrossRef]
18. Poonia, M.P.; Ramesh, A.; Gaur, R.R.; Joshi, A. Effect of pilot fuel quantity, injector needle lift pressure and load on combustion characteristics of a LPG diesel dual-fuel engine. *Int. J. Eng. Innov. Technol.* **2012**, *2*, 26–31. Available online: https://www.semanticscholar.org/paper/Effect-of-Pilot-Fuel-Quantity-%2C-Injector-Needle-and-Ramesh-Gaur/7771d67308d4478b5648aaef505ffefc92c47fe6 (accessed on 18 August 2022).
19. Selim, M.Y.E. Combustion Noise Measurements and Control from Small Diesel and Dual-fuel Engines. *SAE* **2004**. [CrossRef]
20. Selim, M.Y.E. Sensitivity of dual-fuel engine combustion and knocking limits to gaseous fuel composition. *Energy Convers. Manag.* **2004**, *45*, 411–425. [CrossRef]
21. Luf, S. Analysis of combustion process in a dual-fuel compression ignition engine fuelled with LPG in liquid phase. *J. KONES Power Train Transp.* **2007**, *14*, 355–362. Available online: https://www.semanticscholar.org/paper/Analysis-of-combustion-process-in-a-dual-fuel-with-Luft/08326b6a3e5151789985f16662e05be5223800c6 (accessed on 18 August 2022).
22. Ogawa, H.; Miyamoto, N.; Li, C.; Nakazawa, S.; Akao, K. Low Emission and Knock-Free Combustion with Rich and Lean Biform Mixture in a Dual-Fuel CI Engine with Induced LPG as the Main Fuel Low Emission and Knock-Free Combustion with Rich and Lean Biform Mixture in a Dual-Fuel CI Engine with Induced LPG as the Main Fuel. *SAE* **2001**. [CrossRef]
23. Ergenc, A.T.; Koca, D.O. PLC controlled single cylinder diesel–LPG engine. *Fuel* **2014**, *130*, 273–278. [CrossRef]
24. Elnajjar, E.; Selim, M.Y.E.; Hamdan, M.O. Experimental study of dual-fuel engine performance using variable LPG composition and engine parameters. *Energy Convers. Manag.* **2013**, *76*, 32–42. [CrossRef]
25. Rosas, M.; Amador, G. Knock Detection Method for Dual-Fuel Compression Ignition Engines Based on Block Vibration Analysis. *SAE Int. J. Engines* **2021**, *14*, 199–209. [CrossRef]
26. Castro, N.; Toledo, M.; Amador, G. An experimental investigation of the performance and emissions of a hydrogen-diesel dual fuel compression ignition internal combustion engine. *Appl. Therm. Eng.* **2019**, *156*, 660–667. [CrossRef]