



# Effects of Diesel Fuel Injection Timings on Combustion and Emissions of a Diesel-Propane Compression Ignition Engine

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## Abstract

## Original Research Article

A well optimised fuel injection timing enhances the combustion temperature and the rate of heat released in the cylinder. Fuel injections timing is sometimes used to optimised the combustion, performance and emissions characteristics of diesel engines. The study has experimentally investigated the effects of fuel injection timings on combustion, performance and emissions characteristics of a diesel engine running on propane and diesel fuels. In this research, both the pre- and main injections timings were advanced from 18° (before top dead centre (bTDC)) pre-injection and 8°bTDC (main injection) to 21°TDC (pre-) and 11°TDC (main) and these were investigated by displacing 15% and 25% of the diesel fuel energy with propane. The results revealed that with the adjustment of the diesel injection timing to D21-11, there was huge improvement in the brake thermal efficiency and rise in the in-cylinder temperature. As regards the emissions, rise in tailpipe emissions of Carbon monoxide (CO) and total unburnt hydrocarbon (THC) were attributed to unburnt propane. Although, the presence of propane fuel led to low emission of NO<sub>x</sub>, increase cylinder temperature promoted by D21-11 injection timing resulted to increase in both NO<sub>x</sub> and NO<sub>2</sub> emissions

**Keywords:** Dual Fuelling, Emissions, Combustion, Carbon monoxide, Nitrogen oxide, Propane.

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## 1. INTRODUCTION

Diesel engines basically emit more of nitrogen oxides (NO<sub>x</sub>) and particulate matter (PM) compared to gasoline engines [1]. This is as a result of the non-homogeneity of the air/fuel mixture. Meaning that even though the overall combustion is lean [2], there are locally fuel-rich zones. With the growing concern over health and environmental issues related to engine out pollutions [3], the emissions of PM and NO<sub>x</sub> from diesel exhaust are strictly limited by

regulations [4]. To achieve these, there are various techniques that are being explored. While some of these techniques are very well advanced, others are still at various stages of development. One of these is the integration of fuel injection timing in dual fuel engines [5].

A combination of dual fuel system and diesel fuel injection timings could be applied in regulating the charge conditions inside the engine cylinder (density, oxygen concentration and temperature) of a



compression ignition engine [6]. This method seeks to address the locally hot and fuel-rich regions by providing a homogeneous in-cylinder mixture through prolong residence time (ignition delay) for air-fuel mixing [5], [7], [8].

Dual fuel systems employ two fuels of different reactivity [9]. To achieve this, the incoming fresh air is premixed with a low reactivity, high octane fuel such as Liquefied Petroleum Gas in the intake manifold before being admitted into the engine cylinder where, the auto-ignition temperature of the compressed charge is enhanced by diesel-like fuel injected directly into the combustion chamber at timings earlier than conventional compression ignition engines [10], [11]. This gives adequate time for improved fuel to air mixing and supplies a near homogeneous mixture into the combustion chamber [12], [13].

LPG has long been used in spark ignition engines as auto-gas or auto-fuel [14] to run taxis and private cars [15]. This stems from the abundance of LPG in the world reserves, low cost and clean-burning nature compared to the conventional fuels [16]. The low emission benefit is attributable to the ability of gaseous fuel to form near homogeneous mixture [17] and supports better combustion process. In addition, high hydrogen to carbon ratio, absence of aromatic, low number of dissolved impurities (e.g. sulphur contents) compared to diesel and gasoline [18], [19] enhance the reduction of PM. Furthermore, whilst LPG exist in gaseous form at normal temperature and pressure, it transforms to a liquid form if it is under pressure or cooled [20], thus easing the transportation and storage in the liquid phase at typical pressures of 7-14 bar [1] onboard LPG carriers [20]. However, LPG is unsuitable as a sole fuel in diesel engines as a result of high ignition temperature and low cetane numbers. These drawbacks can be corrected when it is used in co-combustion with a fuel with high cetane number.

The combustion characteristics of engines running on a dual mode are influenced by the two (i.e. pilot and primary) fuels [21]. This is mainly due to the differences in reactivities of the two fuels in the combustion process. For instance, the prolonged ignition delay triggered by the LPG fuel provides more time for the two fuels to properly mix. This

creates a partially premixed charge that would lead to enhanced mixture homogeneity and therefore, promotes the real-time reduction of nitrogen oxide (Nox) and soot emissions [22]. Low NOx and soot formation in dual fuel engines is caused by reduced diesel fuel utilization with increase in LPG substitution [23]. LPG presence provides a lean in-cylinder mixture with higher heat capacity and therefore, reduction in combustion peak temperature. Furthermore, improved atomization, increased diffusion combustion velocity [21] and homogeneity in fuel-air mixture [24].

Although, at high load operations, dual fuelled engines have the advantage of providing more efficient, increased power output and reduced engine emissions in comparison to the corresponding diesel engines [25]. At low loads, as a result of the formation of lean mixture that is hard to ignite and completely burn [26], they suffer from low performance and high engine out HC and CO emissions [26], [27]. Several strategies have been investigated to address these drawbacks. One of these is the integration of multiple fuel injection timings in dual fuel engines [28].

Multiple injection is an effective strategy to achieve optimal combustion and emissions control with no additional cost and complex engine modifications [29], [30]. In a system with multiple injection, the amount of fuel delivered is divided over two or more injections for each burn cycle [31] to promote low temperature combustion [32]. Dividing the fuel injection in this manner, ensures that a little quantity of fuel is injected during the compression stroke before the main injection. This way shortened the main injection ignition delay due to the closeness of the injection interval between the pre- and main injection timings [22], [33], [34]. This in turn results to increase in the ignition area of the mixture in the combustion chamber and thus, increasing the pressure, temperature and the intensity of combustion [34]. It is therefore, believed that the prevailing low temperature, pressure and characteristics long ignition delay with dual fuelling combustion would benefit from the increased temperature resulting from the pre-injected fuel

This research is aimed at investigating the benefits in the application of split injection timing on the

performance and emissions characteristics of a dual fuelled engine running on propane and diesel fuels. In order to achieve these, experiments were conducted in a compression ignition engine (CIE) using propane and diesel fuels, with the diesel fuel injection divided into two: pre- and main injection timings while maintaining a constant dwell. The gaseous propane was introduced into the intake manifold to be premixed with the intake air before the combustion chamber where the premixed charged is further mixed with diesel fuel injected at pre- 18°bTDC, main 8°bTDC and pre- 21°TDC, main 11°TDC, respectively.

## 2. METHODOLOGY AND EXPERIMENTAL SETUP

### 2.1. Engine specification

A water-cooled single cylinder compression ignition engine (CIE) with engine characteristics detailed in Table 1 below, was used in carrying out the study. The schematic diagram of the experimental set-up represented in Figure 1, shows the engine with all necessary accessories and propane supply system. A high-pressure common rail fuel injection system equipped with the single cylinder engine which allows for the control of pre- and main injections (i.e. pressure, injection rate and fuel quantity). To control the engine load and speed the study used an electric dynamometer with a motor and load cell fixed to the engine.

**Table 1** Technical characteristics of test engine

Parameter	Units	Characteristics
Engine		Single-cylinder
Stroke type		4-stroke
Bore/stroke	mm	84.0 / 90.0
Connecting rod length	mm	160
Displacement	cm <sup>3</sup>	499
Compression ratio		16:1
Injection system		Common rail
IMEP range	bar	<7
Pressure range of fuel	bar	500-1500
Number of injections		3

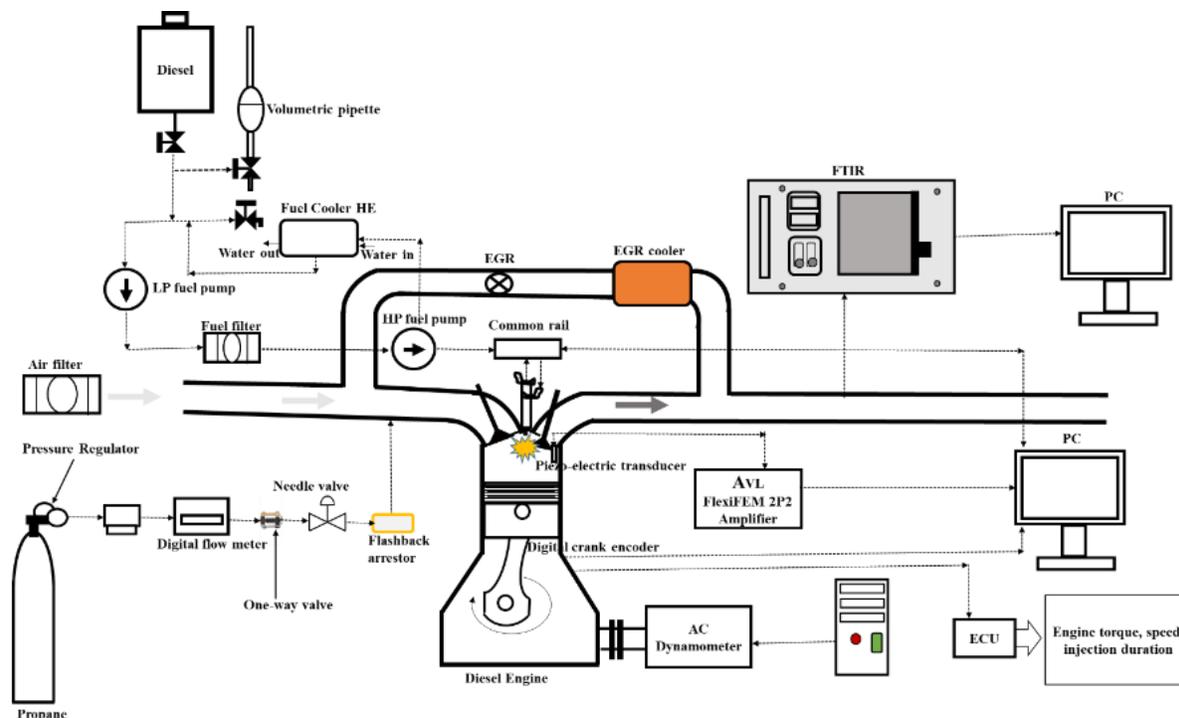


Figure 1: Outline of experimental apparatus

## 2.2. Fuel supply system

Ultra-low sulphur diesel (ULSD) fuel and propane gas were used for this research. The propane supplied in a cylinder was used to simulate LPG. The properties of both fuels sourced from [8, 33-36] are

represented in Table 2 below. The ULSD fuel flow rate was measured in volumetric basis. This was done using a stopwatch that monitors the time which it takes the engine to consume a specific quantity of ULSD fuel and a pipette.

**Table 2** Composition of test fuel

Properties	ULSD	Unit	Propane
Chemical formula	-		C <sub>3</sub> H <sub>8</sub>
Cetane number	43.4		5
Density (15°C)	826.9	(kg/m <sup>3</sup> )	1.861(17°C)
Energy content	43.11	(LHV, MJ/kg)	46.3
Sulphur	6.1	(mg/kg)	<1
Specific heat capacity	1.87	(kJ/kgK)	2.07
Air to fuel ratio (stoichiometric, wt.%)	14.6		15.6

Aromatic contents(wt.%)	24.4		0
Auto-ignition Temperature	180-220	(°C)	472

### 2.3. Instrumentation and data acquisition

To run the engine on dual fuelling mode, it was conveniently instrumented to allow propane injection into the intake manifold. The propane supply system comprises a cylinder bottle, pressure regulator, an electrical valve, a digital flow meter, one-way valve, needle valve and a flash-back arrestor. An AVL GH13P pressure transducer was used to determine the in-cylinder pressure. This was fixed to the top of the engine cylinder with signals amplified by an AVL Flexi FEM 2P2 Amplifier. To determine and record the instantaneous crankshaft position a digital shaft encoder which produces over 360 pulses per engine revolution was fitted to the engine. The in-cylinder pressure was determined using a specially designed Lab-View software for a minimum of 200 consecutive cycles of the engine at a revolution of 1° of crank angle degree (CAD). Standard instrumentations for engine test rig like the thermocouples and pressure gauges mounted at specific parts of the engine was used to monitor the oil and air temperatures, pressures and the inlet manifold. The engine oil, coolant temperatures, injection timings and durations, and engine cycles was adjusted and monitored from the electronic

control unit of the engine (ECU). The expected outputs of the analysis of successive engine cycles are the indicated mean effective pressure (IMEP), percentage coefficient of variation (%COV) of IMEP, rate of heat release (ROHR) and the engine cylinder peak pressure. Controlled exhaust emissions including CO, NO<sub>x</sub> (NO and NO<sub>2</sub>), Total Hydrocarbon was determined using a Multi-Gas 2030, Fourier Transform Infrared (FTIR) spectrometry-based analyser. The temperature of the FTIR was set at 191°C and emitted gases was sampled at specific point along the exhaust system using a heated sampling tube.

### 2.4. Experimental Procedure

Prior to testing, the engine was started and left to run on idle until the coolant and lubrication oil temperatures got to 33°C and 61°C respectively. This was done to reduce the emissions fluctuations caused by cold starts to the minimum. [37, 38]. The proportions of propane substituted for diesel, flow rates and injection timings for the experiments are represented below in Table 3.

**Table 3** Experimental test matrix

Engine conditions			Propane substitution Z (%)	Propane flow rates (l/min)	Diesel injection timing (°CA bTDC)	
Load (IMEP)	Speed (rpm)	Diesel injection pressure (bar)			Pre-injection	Main injection
2	1500	550	15	0.58	18	8
			25	0.96	18	8
			15	0.56	21	11
			25	0.94	21	11

The steady state condition of the engine was used for the experiments. This corresponds to a continuous speed of 1500 rpm, low load of 2 bar IMEP and

common rail fuel injection pressure of 550 bar. Same engine cycles and conditions were sustained for the experiments. Based on preliminary studies, two

diesel fuel injection timings were selected. These are: D18-8 (pre-18° crank angle (CA) before top dead centre (bTDC), and main 8°bTDC) and D21-11 (pre-21°TDC and main, 11°bTDC) well equal dwell angle (angle between the end of the main injection and the start of pre-injection) [38]. For the dual fuel mode, gaseous propane released from a cylinder bottle flows through a pressure regulator where the pressure was dropped before being sent to the engine to displace 15% and 25% of the ULSD fuel. Digital

flow meter was used to precisely direct the precise amount of propane to the inlet manifold. The premixed air-propane mixture was directed into the engine cylinder for compression. Towards the TDC, ULSD fuel was injected directly into the cylinder at pressure of 550 bar through the engine fuelling system. This ignites the compressed charge and keeps the engine running. Equation 1 was used to calculate the portion of chemical energy contribution from propane in the dual fuel mode[36].

$$\text{Propane substitution}(\%) = \frac{m_{\dot{C}_3H_8} \times LHV_{C_3H_8}}{m_D \times LHV_D + m_{\dot{C}_3H_8} \times LHV_{C_3H_8}} \times 100 \quad 1$$

Brake specific energy consumption (BSEC) was used to analyse the fuel energy consumption due to differences in energy values of the two fuels as shown in equation 2.

$$BSEC = \frac{m_D \times LHV_D + m_{\dot{C}_3H_8} \times LHV_{C_3H_8}}{P_B} \quad 2$$

Where:

$P_B$  brake power output (kW)

$m_D$  = mass flow rate of diesel (kg/s)

$m_{\dot{C}_3H_8}$  = mass flow rate of propane (kg/s),

$LHV_D$  = lower heating value of ULSD (MJ/kg)

$LHV_{C_3H_8}$  = lower heating value of propane (MJ/kg).

ROHR was determined from:

$$ROHR = \frac{1-\gamma}{\gamma} P \frac{dV}{d\theta} + \frac{1}{1-\gamma} V \frac{dP}{d\theta} \quad 3$$

ROHR is the difference between the gross heat release and heat transfer through the walls of the cylinder and is calculated using the first law of thermodynamics. In this study, the single zone model was considered as the in-cylinder content was assumed homogeneous.

$COV_{IMEP}$  was analysed for combustion stability and was calculated based on:

$$COV_{IMEP} = \frac{\sigma_{IMEP}}{\overline{IMEP}} \quad 4$$

Where  $\overline{IMEP}$  is the average of the IMEPS and n represents numbers sampled? The standard deviation of IMEP ( $\sigma_{IMEP}$ ) was calculated from:

$$\sigma_{IMEP} = \sqrt{\frac{\sum_{i=1}^n (IMEP_i - \overline{IMEP})^2}{n}}$$

### 3. RESULTS AND DISCUSSIONS

This section presents and discusses ULSD fuel effects on injection timings on combustion, performance and emissions of propane-diesel fuel engine. In-cylinder pressure data was used for the combustion characteristics analysis. Per cent mass fraction burnt (%MFB) gives an indication of the cumulative heat released [22]. The ignition delay, combustion phasing and end of combustion are respectively defined as the CAD positions

corresponding to 10%, 50% and 90% of the cumulative heat generated. Whilst the difference between 90%MFB and 10%MFB was used to define the duration of combustion.

#### 3.1. Combustion and performance characteristics

Shown in Figures 2(a) and (b) are the profiles of in-cylinder combustion pressures and ROHR for the D18-8 and D21-11 injection timings of the ULSD and dual fuel combustion.

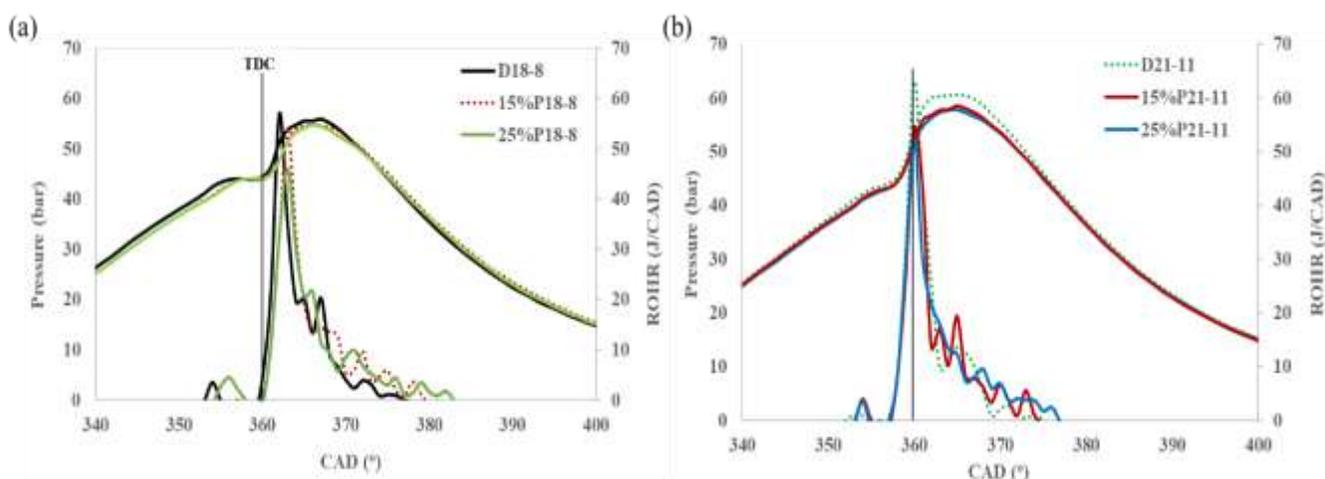


Figure 2: ROHR and in-cylinder profiles for injection timings of: (a) D18-8 (b) D 21-11 for ULSD and dual-fuel combustion with 15% and 25% propane substitution.

Figures 2(a) and (b) shows that the peak of the in-cylinder pressures and premixed combustion are higher and occurred earlier when engine was running on the ULSD fuel compared to when it is in dual fuelling mode. Also, with the advanced injection timing (Figure 2b), the delay in ignition became shorter as evidenced in the start of combustion (SOC). The SOC in this paper suggests the CAD position at which the ROHR curve first became positive after the start of fuel injection [39]. Xu et al. [32] and Cameretti et al. [40] have reported similar

results, where both authors in their separate work showed that earlier diesel injection timing resulted to an increase in the in-cylinder pressure. More of the fuel energy was released during the compression stroke could be attributed to early injection timing [41]. This causes the in-cylinder air temperature and density to increase and therefore, enhances the fuel evaporation and provides a more homogeneous air-fuel mixture [29]. The result is that both the physical preparation time and the chemical reaction time of the air-fuel mixture is reduced before the SOC [35].

All of these are believed to have contributed to reduced ignition delay and increased the peak of the premixed combustion phase and in-cylinder pressure.

Nonetheless, in the presence of propane, it was observed that the premixed combustion phase of the ROHR and the peak cylinder pressures are lower. Furthermore, the combustion phasing is slightly postponed with increasing propane substitution for both injection timings. With the presence of propane being the fuel with low cetane number (CN) and higher specific heat capacity, it is likely that whilst the low CN slowed down the chemical reaction rate [24] of the in-cylinder mixture, the adiabatic flame temperature was decreased due to its high specific heat capacity and thus, the peak pressure and ROHR were also decreased. Similar study was carried out by Leermakers et al. [42] and it was observed that advancing the fuel injection timing increases the residence time for mixing of air and the low reactive fuel and as a result, a leaner and less reactive local mixture is formed that led to a retarded combustion [43].

Moreover, comparing Figure 2(a) and Figure (b), it can be observed that the former exhibits lower in-cylinder peak pressure, lower ROHR that moved into the expansion stroke. This is could be due to D18-8 injection timing being closer to TDC than D21-11. As more fuel was injected in the main (8°bTDC) than the pre- (18°bTDC) as represented in Figure 2, the

period of ignition delay is believed to have extended into the expansion stroke. Consequently, lower temperature and pressure at this region had shortened the reaction time due to the late injection of fuel before the exhaust valve opening time [42].

Combustion durations due to injection timings and fuelling systems are shown in Figure 3. For injection timing of D18-8, the duration of combustion for the single diesel fuelling is similar to that of 15% C<sub>3</sub>H<sub>8</sub> but becomes shorter when propane content in the cylinder is increased to 25%. It is believed that as the propane fraction of the fuel increases, the in-cylinder mixture gets leaner and the consequence is, flame extinction (quenching) in the expansion stroke caused the rate of fuel combustion to decrease. However, it can be seen that as the injection timing was advanced to D21-11, the combustion duration increases. It should be noted that regardless of the quantity of diesel energy displaced, there is no significant difference in combustion duration under the D21-11 injection timing. It can therefore be concluded that, the increase in the combustion duration under this condition was solely caused by advance injection timing. Xu et al. [33], in their study of a natural gas-diesel dual fuelled combustion, reported that advanced injection timing prolongs the residence time for air-fuel mixture to produce a leaner in-cylinder mixture and as a consequence, slows the burning rate of the in-cylinder mixture and increase the combustion duration [32].

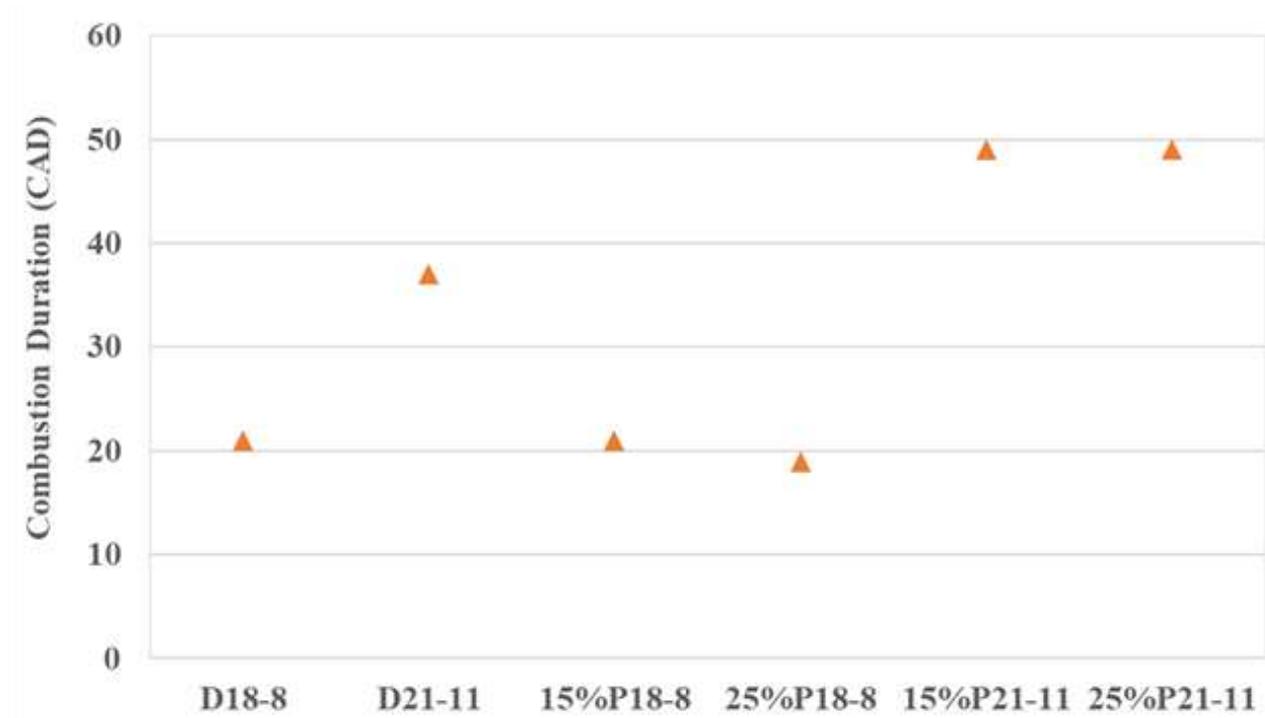


Figure 3: Combustion duration for D18 -8, D21-11 and 15% and 25% propane substitutions

The brake specific energy consumption (BSEC) for D18-8 and D21-11 for ULSD and 15% and 25% propane substitutions can be seen in Figure 4. BSEC was used to analyse the fuel energy consumption. The lower the BSEC the better, as it shows that less fuel energy was used to generate useful power. It can be observed that when only the diesel fuel was used to run the engine, the BSEC reduced conspicuously as the injection timing was advanced to D 21-11. It is likely that the higher in-cylinder pressure promoted the vaporization of the diesel fuel, better air-fuel mixing process and consequently, the fuel-to-work conversion efficiency was improved [44]. With the application of propane, the BSEC got higher as compared to running the engine with diesel

fuel. The reason is as propane was injected into the engine, part of the intake air was displaced causing the volumetric efficiency to be lower. Besides, as the peak of premixed combustion is usually lower for dual fuel combustion [45], it means that less heat energy was supplied to convert fuel and therefore, more fuel was consumed. Even though, advanced injection timing promotes the mixing of fuel and air to increase peak cylinder temperature and reduce the BSEC, increasing the propane fraction from 15% to 25% led to higher specific heat capacity of the in-cylinder charge and therefore, reduced the in-cylinder temperature and deteriorated the fuel conversion rate [45].

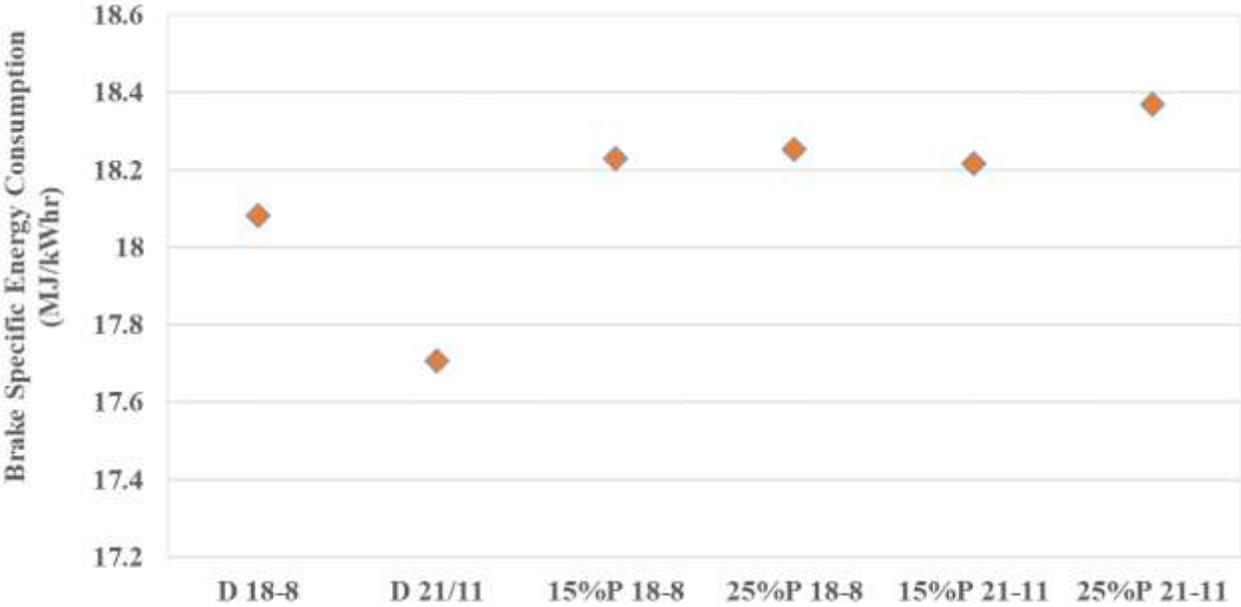


Figure 4: Brake specific energy consumption for D18 -8, D21-11 and 15% and 25% propane substitutions

Figure 5 shows the Brake thermal efficiencies (BTE) for D18-8, D21-11 for 15% and 25% propane substitutions. It can be seen that the BTE increases as the injection timing was advanced whilst running the engine on diesel fuel. Similar result was reported by [8] and the authors attributed this to better mixing due to prolonged ignition (physical) delay that resulted to more efficient combustion. Another

possibility is, with advanced fuel injection timing of D21-11, it is likely that more of the diesel fuel injected was burnt resulting to increase in peak combustion pressure as was shown in Figure 2(b) to produce more useful work. However, the BTE was lower with D18-8 injection timing. In addition, under the dual-fuel mode, the BTE was reduced further even with the D21-11 injection timing.

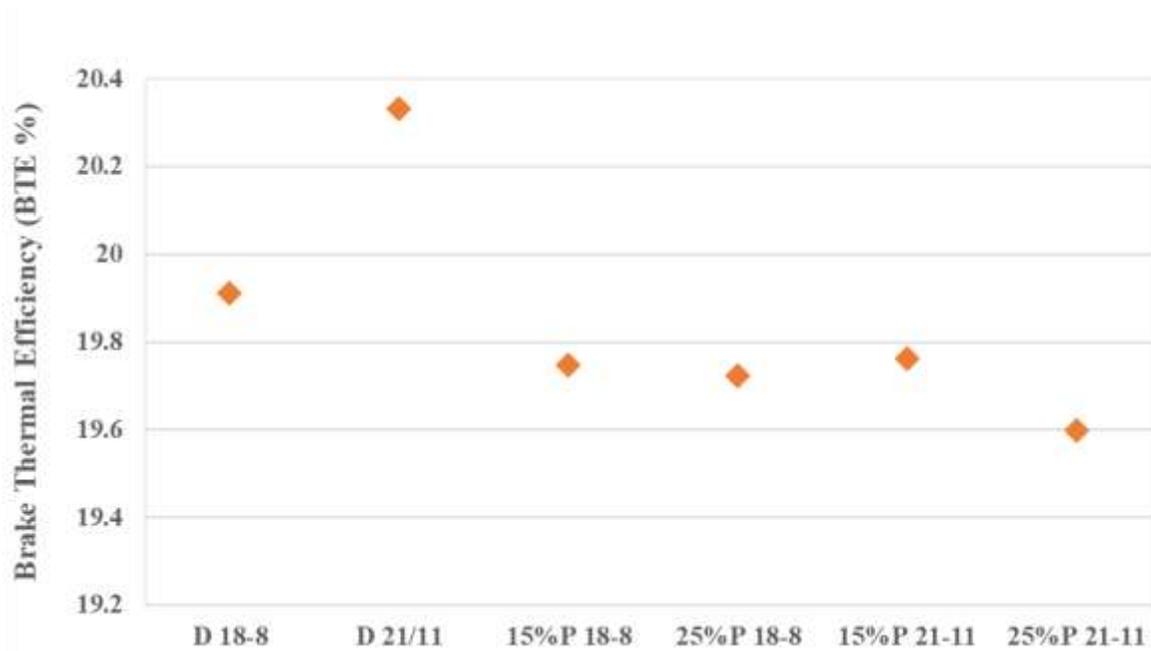


Figure 5: Brake thermal efficiency for D18 -8, D21-11 and 15% and 25% propane substitutions.

As it was observed for the ROHR profile in Figure 2(a), more fuel was combusted in the power stroke where the pressure and temperature are lower [8] for D18-8. Moreover, as the ignition delay becomes longer due as a consequence of the drop in volumetric efficiency and low CN value of propane, higher amount of unburnt fuel was trapped in the crevices volumes in the culinder [46], hence, leading to a deterioration in the fuel energy conversion, and consequently, reduced of BTE.

### 3.2. Emissions characteristics of injection timings and diesel-propane dual fuel combustion

The THC and CO emissions for D18-8 and D21-11 injection timings and 15% and 25% propane substitutions are shown in Figure 6. As expected of a typical dual fuel compression ignition engine operating on low load, higher amount of CO and THC emissions were recorded at the exhaust tailpipe. However, when the engine was running on diesel fuel mode and when injection timings of D21-11 and D18-8 were compared, slight increase in CO and decrease in THC emissions were observed as the

D21-11 injection timing. It is thought that although, higher temperature in the cylinder resulting from D21-11 injection timing assisted in further oxidizing the THC, the temperature was too low and therefore, could not completely convert the THC to CO<sub>2</sub> and therefore, resulting to the partially oxidised CO emissions. Similar trend was observed when propane was added for both D21-11 and D18-8. Since both CO and THC are products of improper mixing, wall quenching and incomplete combustion [47], they originate from areas where the air at excessively diluted levels and fuel mixture results in drop in temperature and accordingly, stops the beginning of the combustion process or from proceeding to completion [48]. It is thought that a greater proportion of the THC emission recorded came from the partially or unburnt propane that escaped into the crevice's volumes of the combustion chamber during compression stroke and released during the expansion stroke. Comparatively, because the oxidation of CO is several times slower than that of other species during combustion process[49], increasing the propane fraction provided conditions

in the cylinder that slowed down the oxidation of CO compared to THC.

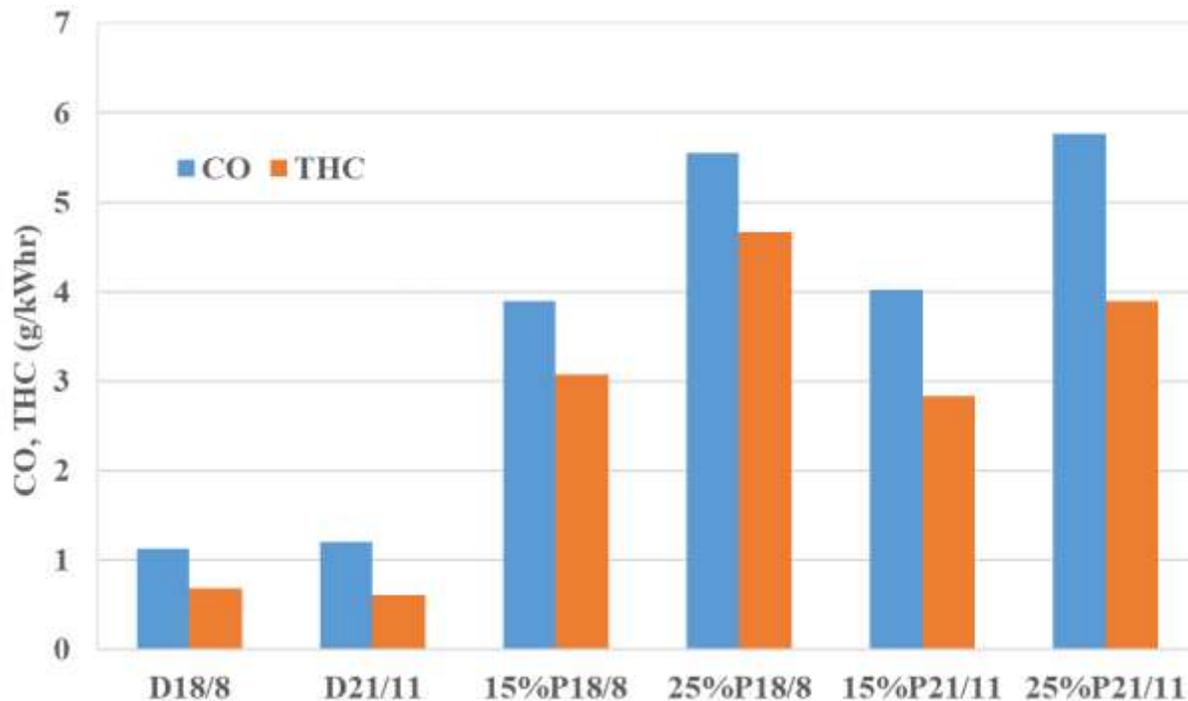


Figure 6: Effects of D18 -8, D21-11 and 15% and 25% on CO and THC emissions.

As the fraction of the propane was increased, the charge in the cylinder gets leaner and slow to vaporise and therefore, it becomes hard to entirely burn. Also, the high temperature prevailing in the cylinder caused by D21-11 injection timing to somewhat, favoured the oxidation of THC, consequently, lowering the formation of THC. However, the lower in-cylinder temperature and volumetric efficiency promoted by the presence of propane in the cylinder led to the reduction in the rate of conversion of CO to CO<sub>2</sub>.

Shown in Figure 7 are the NO<sub>x</sub> (nitrogen oxide) emission. The formation of nitrogen oxide is reliant on the reaction residence time, temperature, oxygen

content. Although, NO<sub>x</sub> is a generic term, this study analysis NO<sub>x</sub> and NO<sub>2</sub> (nitrogen dioxide). Both emissions are observed to be higher with D21-11. Effects of advanced injection timing on the emissions nitrogen oxide as reported by Shu et al. [43]. One explanation is that the earlier premixing resulting from advanced injection timing caused ignition and combustion to occur earlier and as a result, produced high peak in-cylinder pressure [8]. Another observation that may attributed to the increase in NO<sub>x</sub> is the increased BTE as shown in Figure 5 with D21-11. Similar result reported by Suh [48] found that increase in thermal efficiency due to early injection timing promoted the formation of NO<sub>x</sub>.

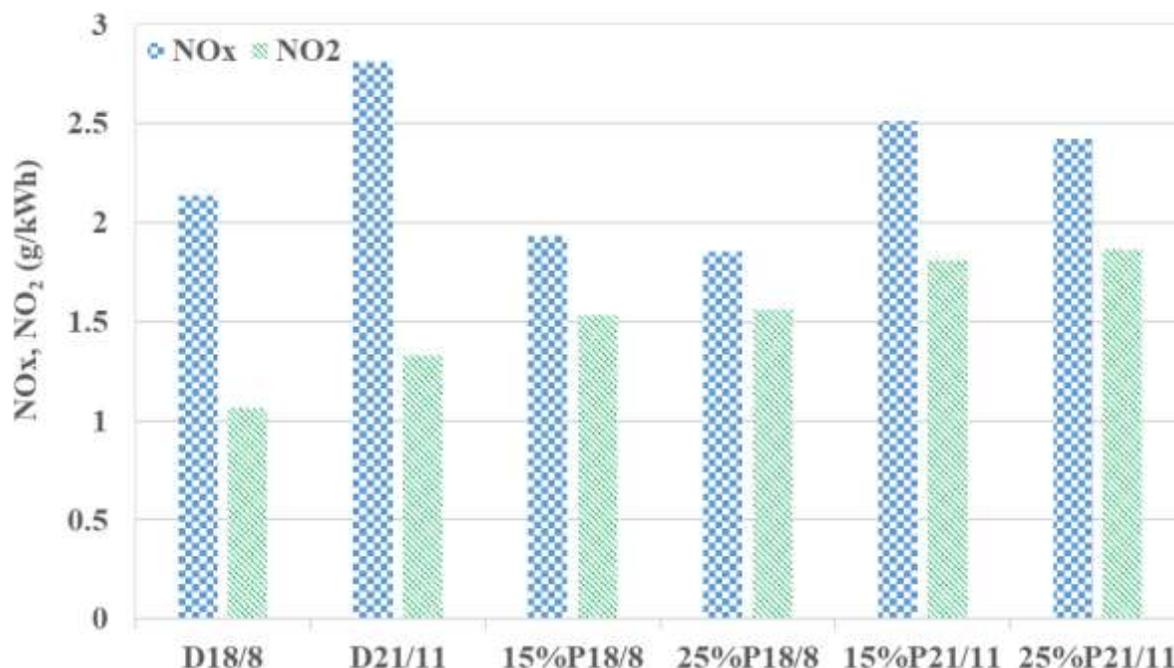


Figure 7: effects of advanced injections and propane substitution on NO and NO<sub>2</sub> emissions.

Although, the emissions of both NO<sub>x</sub> and NO<sub>2</sub> show some levels of reduction with the addition of propane, the more advanced injection timing seems to have more influence as the NO<sub>x</sub> level continues to increase even as the propane level was increased to 25%. The reduction in the NO<sub>x</sub> as propane was added could be attributed to: (1) the propane displacing part of the intake oxygen as it was inducted through the intake manifold, (2) the higher specific heat capacity of propane assisted in suppressing the in-cylinder temperature; and (3) the overall lean air-fuel mixture as propane was inducted leads to a decrease in the local high burnt gas temperature [22]. Furthermore, with the delay of the combustion phasing into the expansion stroke as propane was added, the residence time for NO<sub>x</sub> formation in the high temperature compression stroke was reduced [29]. However, the higher in-cylinder temperature from D21-11, offset the contribution from the propane addition and consequently, increased the NO<sub>x</sub> emissions. This is because the longer ignition delay due to thermal effect of propane as the injection was advanced is

believed to have enhanced the air-fuel mixing and therefore, increased the amount of combustible diesel at the point of ignition.

#### 4. CONCLUSION

Dual fuel engine combustion characteristics share the properties of the two fuels to achieve better combustion and emissions results. Nonetheless, when running on low temperatures and load conditions, dual fuel engines usually have the problem with poor engine performance and produce high level of CO and THC tailpipe emissions. In this study, experiments were carried out to investigate the impacts of injection timings on the combustion characteristics, performance and emissions of a dual fuelled engine running on propane and diesel fuels. It was shown that when the diesel fuel injection timing was advanced, improvement in combustion and performance of propane-diesel mode was achieved as the advanced injection timing caused increase in in-cylinder pressure and ROHR. It was observed that the high cylinder pressure and ROHR

promoted by advancing the injection timing to some extent, created a favourable in-cylinder environment that compensated the cooling effects from the presence of propane and therefore, both the BSEC and BTE were not obviously affected. Increase in emissions of THC and CO emissions was recorded when the engine was running on dual fuelling mode in comparison diesel fuel mode. THC and CO are products of unburnt fuel and with D21-11 and D18-8 injection timings, propane in the cylinder promoted the formation of both emissions. Furthermore, whilst decrease in NO<sub>x</sub> emission was observed, increase in NO<sub>2</sub> was recorded as the propane fraction was increased from 15% to 25%.

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