Visualization of LPG-PME Dual Fuel Combustion in an IDI CI Engine

Kanit Wattanavichien

Internal Combustion Engine Research Laboratory,
Department of Mechanical Engineering, Faculty of Engineering,
Chulalongkorn University, Bangkok 10330, Thailand.
Author's correspondence e-mail: wkanit@chula.ac.th **Tel: 662 218 6627 Fax 662 252 2889**

Abstract

This study is aimed to investigate and to identify the effect of biodiesel as the pilot injection in dual fuelled engine. Firstly, images of spray and combustion characteristics of liquefied petroleum gas (LPG) premixed charge-diesel dual fuelled engine was studied. Next, the investigation continued with the pilot injection changed to palm biodiesel (PME). Lastly, it finished with varied injection timing for neat liquid as well as dual fuelling to fulfill a comparison. Test bench experiments (steady state) were conducted with a 4-cylinder IDI CI engine, at selected fixed load, high probability operating points corresponding to the ECE15+EUDC cycle. The engine ran as LPG-air premixed mixture was maintained at four fixed values by an electronic controlling system. The acquired data included basic parameters and accessed combustion chamber visualization. The comparative analysis deal with: energy efficiencies, liquid fuel substitution, combustion chamber phenomena including spray, combustion, flame probability distribution, flame temperature, and soot concentration (two color method).

With LPG-PME, the flame probability distribution and the area of high flame temperature was smaller, due to the PME properties: lower heating value, lower adiabatic flame temperature, and heavier. This was also thought due to the limit of the two color method when applied for gaseous and oxygenated fuel combustion. Concentration of soot in flame was observed to be lower with higher LPG and was much lower in LPG-PME cases. The 1.2-degree-advanced injection timing gave better LPG-diesel combustion while the OEM setting was suitable for LPG-biodiesel combustion.

Keywords : Dual Fuel, LPG, biodiesel, combustion, visualization

1. Introduction

Visualization of LPG-PME Dual Fuel combustion reveals the effects of biodiesel as the pilot injection in dual fuel engine. Dual fuel (DF) engine is an engine that energy release in its operating cycle comes from two different fuels. The first fuel, is called ***pilot***, having high cetane number. The second fuel, is called ***main fuel***, having relatively high octane number. This fuel is introduced to form homogeneously mixture during the intake process. After the auto-ignition of pilot, the main fuel nearby the pilot spray combustion region will be ignited and continued, as flame propagation process throughout the combustion chamber. Then, combustion of the remaining pilot fuel will also be continued in a manner of mixing control process.

DF operation has strong ignition sources providing more reliable and less sensitive with respect to changes in the composition of the main fuel as well as ability to switch back to diesel as necessary. In diesel dual fuel (DDF) engines, good diesel substitution levels are only obtained at mid-load range; at low load the pilot diesel injectors still require a substantial fuel delivery while at high load the prolonged ignition delay increases the tendency for diesel knock as well as end-gas knock. Moreover, information on DF combustion with oxygenated liquid fuel has not been found. A pilot-injection of oxygenated fuel, such as Palm methyl ester, PME, can offset the prolonged ignition delay. It is expected to offer an improvement at the low end as well as having improved potential at the high end. In addition, oxygenated fuel is also expected to improve combustion process of richer fuel-air mixture of dual fuel engine in mid-load range as well. However, in order to gain a proper utilize of PME in a DF combustion chambers, it has increasingly relied more on fundamental knowledge, to understand the effects of this oxygenated fuel on engine combustion process, requiring a direct optical observation. This allows a precise investigation of the problem as it provides all variables at all points of the geometry [1, 2, 3, 4, 5, 6 and 7]

2. Delay Characteristics Of DF

Ignition delay affects the whole combustion process of conventional diesel as well as DF engines. The physical/chemical properties of the compressed charge results in change in ignition delay of the pilot by means of several factors as:

First, relatively lower polytropic index compared with air, causes decrease in maximum charge temperature.

Second, volumetric efficiency reduction either caused by the replacement of gaseous/vapor fuel or by higher flow loss from a gas mixer leads to relatively lower oxygen concentration of the charge.

Third, undergo chemical pre-ignition reactions of gaseous fuels prior to pilot injection will actively participate to the pre-ignition chemical processes of the pilot. Pre-ignition reaction of the premixed charge would cause its temperature to increase.

The prolonged DF ignition delay, that is the net effect of above factors, would lead to the shift of DF combustion process some CA degrees towards BDC. In Figure 2, comparison in the ignition delay of a DF engine operating with different gaseous fuels shows that hydrogen produced least prolonged ignition delay, followed by methane, ethylene and propane.

A predicted ignition delay for DF engine, proposed by Prakash G. et al [5], is based on the modification of Hardenberg and Hase correlation for the delay in diesel engine.

From literature reviews, a number of feasible solutions to improve DF light-load operation have been identified and analyzed by G.A. Karim [8] and others [9, 10, 11, 12, 13, 14]. It is believed that DF combustion can be improved if the pilot ignition delay is reduced to the level of diesel operation. Use of other fuels as pilot have less investigated satisfactory results. In this study, pilot of oxygenated fuel as well as adjusting pilot injection timing are expected to improve DF combustion delay process.

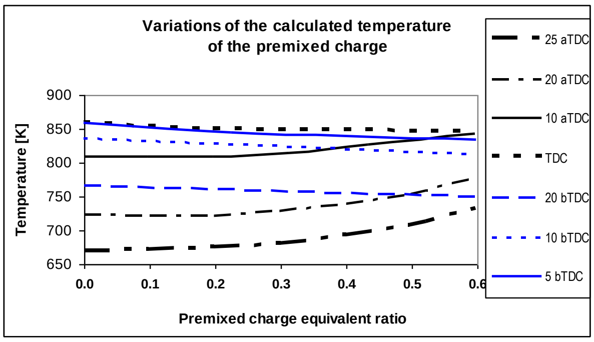
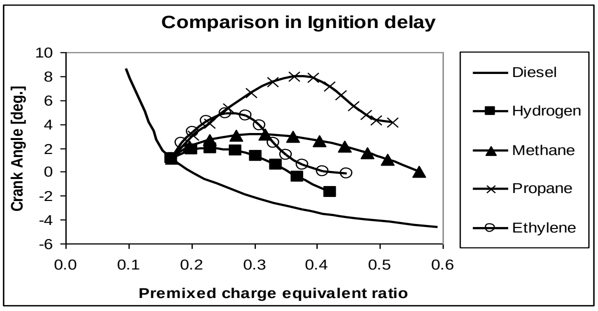


Figure 1 Variations of calculated CH_4-air charge temperature in compression and expansion strokes of a motored engine without pilot injection, *(reproduced from*[6]*)*



Intake charge temperature: T_{int} = 0^{O}C; Pilot injection: 4kg/h;

Injection timing $\theta = -20^{O} CA$

Figure 2 Ignition delay of a diesel-gaseous DF engine, (reproduced from [7])

3. Objectives

The main purposes of this research are to investigate and to identify, by means of combustion visualization, the effects of biodiesel (PME) as the pilot injection, and the effects of pilot injection timing, in liquefied petroleum gas (LPG) premixed charge dual fuelled engine.

4. Research Equipments

The engine under this study is an IDI Ford WL 81, water cooled four cylinders, in-line, natural aspirated engine. The engine was connected to an AVL alpha-40 eddy-current dynamometer. Direct photography was taken with an AVL Engine Visioscope, consists of a PixelFly VGA Color CCD camera (resolution 640x480 pixel), a crank angle encoder, optical linkage to the camera and the endoscope viewing angle of 30° forwarded view. To capture the spray images, the light source unit with fibre optic (40 mJ/flash with 20 μs duration at frequency of 10 Hz) was used.

The combustion heat released rate (HRR) characteristics were investigated by analyzing the cylinder pressure data. Pressure history in pre-chamber, main chamber and liquid fuel line has been recorded by a high speed DEWETRON acquisition system model 5000-CA-SE. Arrangement for these transducers is shown in Figure 3. The schematic arrangement of experimental set up is shown in Figure 4. LPG, in gaseous form, is supplied by an injector to a mixer, shown in Figure 5. The arrangement of LPG injection system is shown in Figure 6. The mass of liquid fuel consumed, M_f,, was directly measured by balance (resolution of 2 gms). The duration, Δt, corresponding to the mass of liquid fuel consumed was measured by a stop-watch with a resolution of 1/100 second.

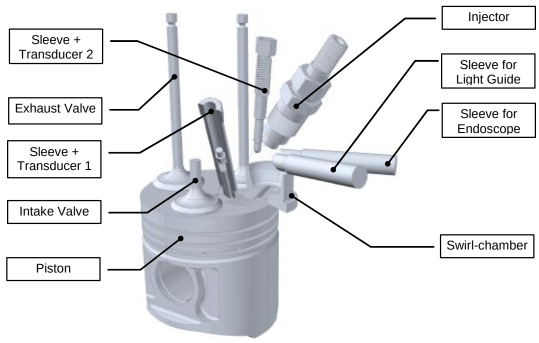


Figure 3 Setup of transducers for cylinder pressure measurement.

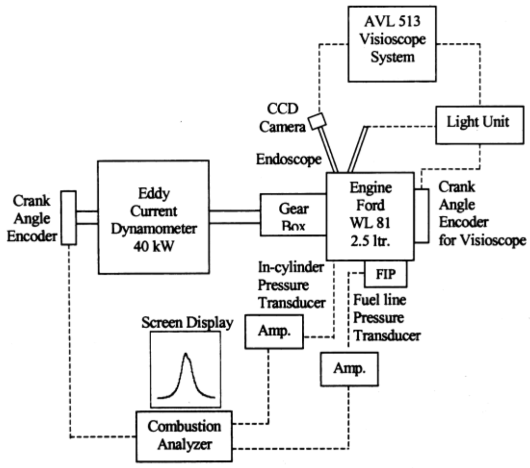


Figure 4 Schematic arrangement of experimental set up

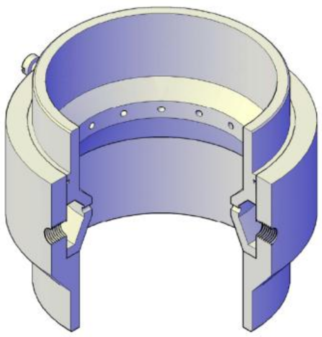


Figure 5 Gas-mixer

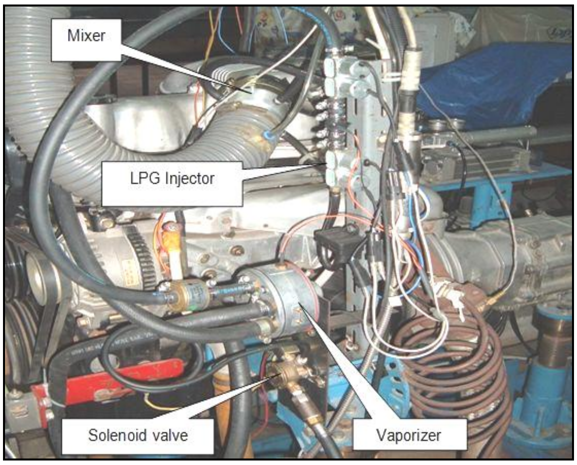


Figure 6 Arrangement of LPG injection system

Inlet air, exhaust gas, lube oil, intake air, and ambient temperatures are measured by thermocouples type K and a display unit.

The physical properties of the PME that used in this investigation have shown in Table 1.

Table 1 Properties of PME

Properties	Unit	Test method (ASTM)	Value
Methyl Ester content	% mass		99.85
Specific gravity	-	D 1298	0.877
Cetane number	-	D 613	61
Flash point	^{0}C	D 92	166
Viscosity at 40^{0}C	Mm^{2}/s	D 277	4.63
Specific gravity	Kg/m^{3}	D 1298	880
Heating value, LHV	MJ/kg	D 611	37.215

5. Experimental Procedure

The experiments were first carried out with fixed OEM injection timing at twelve mode, that are high probability operating points along ECE 15 + EUDC test cycle, of constant speed steady state conditions at selected 20 Nm operating points of 1250 and 2000 rpm. In this test, firstly, the engine is test with neat diesel and neat biodiesel. The dual-fuel operation (LPG-diesel and LPG-biodiesel) for comparison in actual injection events and spray combustion among these operations with different values of the LPG premixed mixture strength (fuel-air equivalent ratio) of L1, L2 L3 and L4 will be investigated. They include six modes at 1250 rev/min (1250-20-D, 1250-20-DL1, 1250-20-DL4, 1250-20-B, 1250-20-BL1, and 1250-20-BL4) and six modes at 2000 rev/min (2000-20-D, 2000-20-DL1, 2000-20-DL4, 2000-20-B, 2000-20-BL1, and 2000-20-BL4) The fuel-air equivalent-ratio of homogeneous premixed mixture and the total fuel-air equivalent ratio are shown in Figure 7.

Then, three values of injection timing setting were chosen and set for the test engine. The first value (1.22-degree-advanced, designated by i3) was applied for neat diesel, PME, LPG-diesel, and LPG-PME. The second (1.0-degree-retarded, designated by i2) and the third (1.8-degree-retarded, designated by i1) were applied for neat PME/ LPG-PME modes.

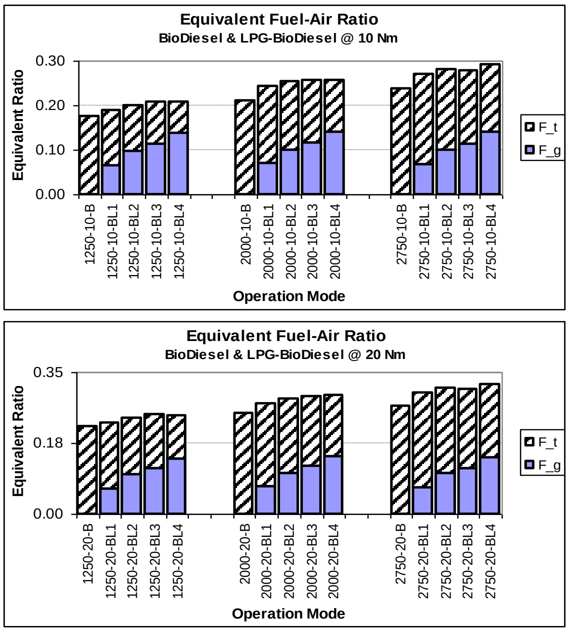


Figure 7 ϕ of the premixed charge and total equivalence ratio

As the engine reaches each steady state test condition, speed, torque, air and fuel consumption, engine operating pressure and temperature for both fuels, in-cylinder pressure and fuel line pressure as well as visualized image in engine pre-chamber were recorded.

Visualization of the spray and spray combustion was conducted during the range between –10 ºCA to 40 ºCA. Illuminating the spray through the light source window was performed for the spray investigation. For spray combustion process, the images were taken without illuminating source. Measured data for each test point is organized as a matrix of 707 images (RGB 640 x 480 pixel): 7 repetitions; from -10 to 40^{o}CA with an increment of 0.5 deg.

Throughout this work, "ignition" will refer to the first appearance of luminous combustion, and "flame" will denote the luminous combustion regions. Cycle-to-cycle variability will be taken in consideration with from 7 repetitions of each image out of a total of 707 cycles. Combustion analyses are then investigated, using two-color-method, for spray, inflammation and combustion processes in the combustion chamber.

6. Results and Analysis

6.1 Effect of the LPG on the combustion characteristic of DF engine operation

Engine operation with four fixed mixture strength of the LPG-air at different engine loads and speeds had been investigated. The highest diesel substitutions at 1250 and 2000 rev/min, shown in Figure 8 and 9 were about 65 % and 47%, while the highest biodiesel substitutions at 1250 and 2000 rev/min, shown in Figure 10 and 11 were about 60% and 45%, respectively.

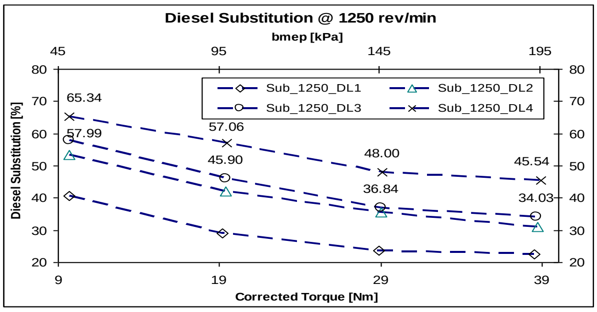


Figure 8 Diesel substitution # 1250 rev/min, LPG-diesel modes

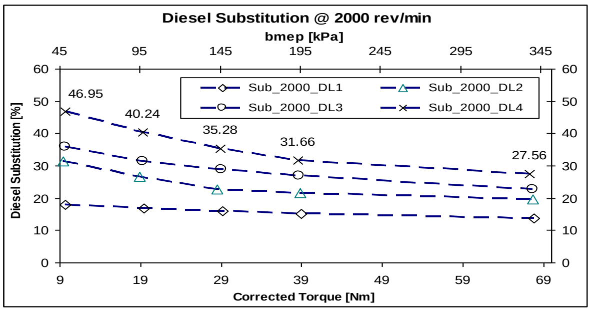


Figure 9 Diesel substitution # 2000 rev/min, LPG-diesel modes

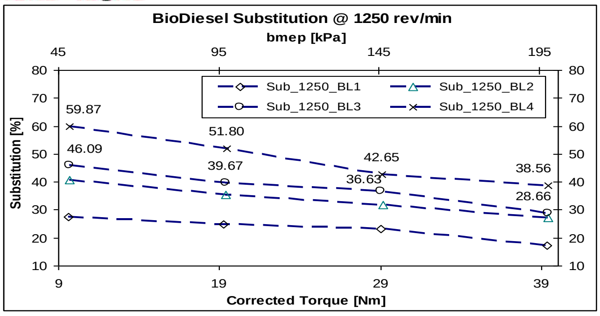


Figure 10 Substitution @ 1250 rev/min, LPG-PME modes

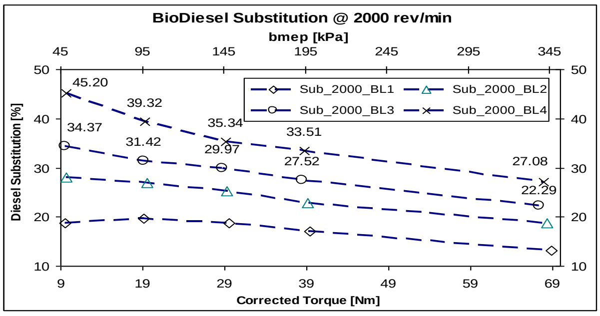


Figure 11 Substitution @ 2000 rev/min LPG-PME modes

Figure 12, 13, 14 and 15 depict the energy conversion efficiencies (ECE) in neat Diesel, LPG-Diesel, PME and LPG-PME modes at the two engine speeds, respectively. The energy conversion efficiency fractions presented here are the ratios between energy conversion efficiencies in the considered modes and that in the corresponding neat diesel modes. The deterioration trend in the energy conversion efficiency in neat PME as well as LPG-Diesel and LPG-PME operation was lower as the engine load increased. At each fixed speed, the deterioration increased as the LPG ratio increased. At 1250 rev/min the deterioration was about 8% to 1% in neat PME modes, from 20% to 6% in LPG-Diesel modes and from 22.5% to 9% as the load increased from 10 to 40Nm in LPG-PME modes. Similarly, the decrease at 2000 rev/min was from 7% to 1% in neat PME modes, from 20% to 5% in LPG-Diesel modes and from 23.5% to 6% in LPG-PME modes, as the load increased from 10 to 70Nm.

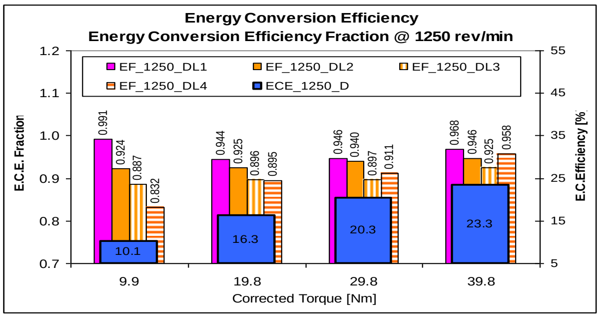


Figure 12 ECE # 1250 rev/min, diesel and LPG-diesel modes

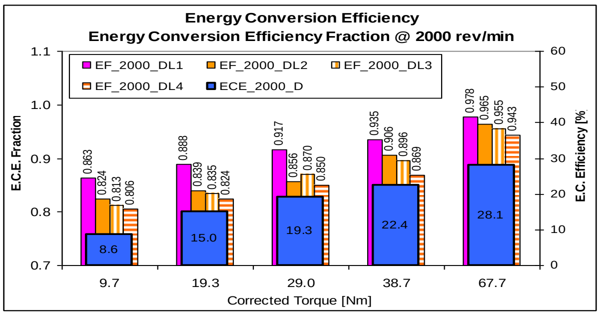


Figure 13 ECE # 2000 rev/min, diesel and LPG-diesel modes

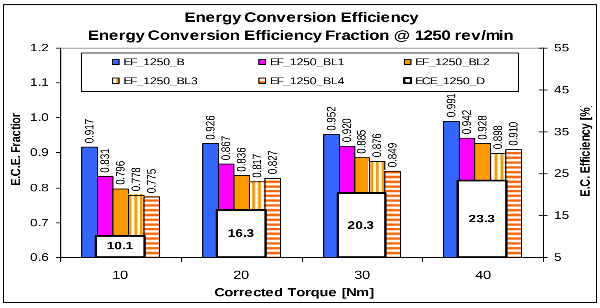


Figure 14 ECE @ 1250 rev/min, PME & LPG-PME modes

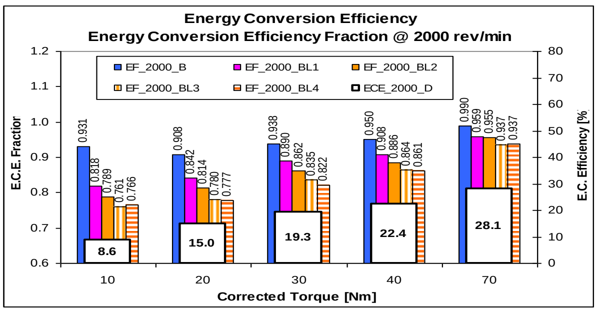


Figure 15 ECE @ 2000 rev/min, PME & LPG-PME modes

There was increasing trend in the specific total energy consumption (STEC), compared with diesel, in neat PME as well as LPD-Diesel and LPG-PME operation, respectively. The STEC, at each fixed speed, was lower as the engine load increased. At each fixed speed, the energy demand increased as the LPG ratio increased.

6.2 Spray Combustion Characteristics

6.2.1 The difference between neat diesel and neat PME operation

The spray, in Figure 16, entered the pre-chamber at the top of it and continues a tangential motion, with clockwise direction, with the chamber wall. Consider the diesel case 1250-20-D in Figure 17, it is seen that the start of luminescence (SOL) appeared at -1.0^{O}CA (1.5^{O}CA after the first appearance of the spray, AOS). The ignition started at a small space around the tip of the spray and then was expelled far from that due to the high swirl intensity in the chamber. While the injection was continued the spray quickly became a torch curved with the swirl. At 4^{O}CA, the torch enveloped approximate half of the viewed area and then almost all the area at 10^{O}CA. From the Figure 18, it can also be recognized that during the whole process the forth quadrant of the area, was the lowest probability of flame distribution whereas the second was the highest. Toward border of the area, this probability reduced due to the effect of heat transfer to the wall. The combustion was continued but the temperature in the chamber was difficult to increase since the total chamber volume increased increasingly as the piston moved down whereas the luminosity of the direct images tended to reduce. Figure 19, it can be recognized that high flame temperature appeared at the top of the torch during the first-three stages of combustion whereas at area around the connecting passage in the last stage. The flame temperature was lower at upstream of the nozzle as well as in the spray core area due to the cooling effect of fuel heating and vaporization. In the late stage, the high flame temperature appeared at the connecting area due to the heat regenerative effect of the passage and the contribution of the flame in the main chamber. In Figure 20, the high soot concentration area appeared along the spray core which was the rich fuel-air equivalent ratio.

At 2000 rev/min, larger amount of diesel injection resulted in much more luminous of the direct images, larger maximum area of the high flame probability distribution, higher flame temperature distribution. However, the larger amount of diesel injected caused longer injection period and longer spray penetration, leading to larger area of the “dense-soot-cloud”.

The PME auto-ignites and burnt few degrees just after its injection without the aid of the glow-plug. The first SOL appeared 2.0^{O}CA after the first AOS. The PME was injected earlier, longer penetration and smaller angle than the diesel spray, and combust earlier than the diesel.

It is recognized that the PME flame is smaller and less luminous than the diesel. The flame luminosity was then faded significantly but it seemed to continue longer in comparison to the diesel flame.

The prediction result of flame probability distribution (Figure 18) and area of the 100% of flame probability distribution (Figure 21) show that, although PME injection starts earlier, the PME combustion seemed less luminous than the diesel. The high probability in images of PME flame revealed smaller size. Towards the end of combustion (from 20^{O}CA), the PME flame was wider but less luminous and lower flame temperature than the diesel.

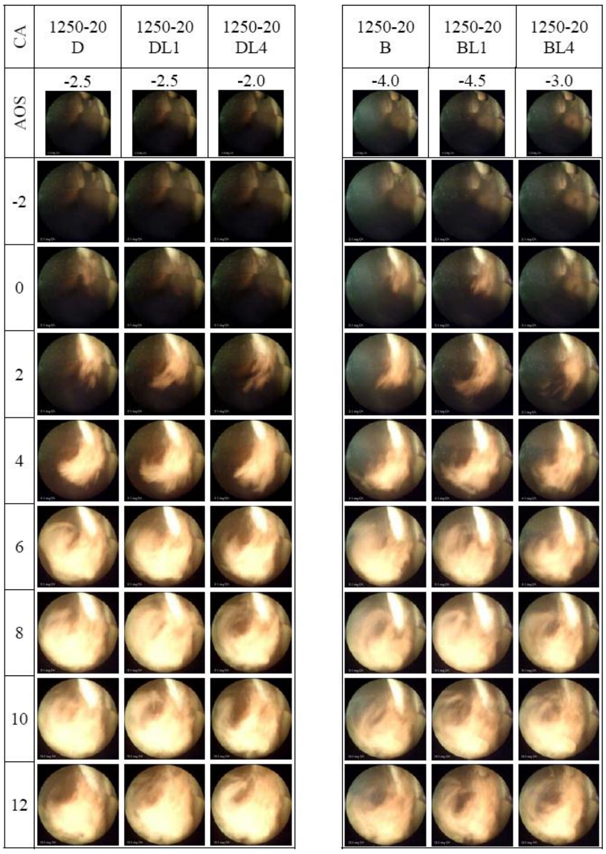


Figure 16 Sequential images of combustion process with cool light at 1250 rev/min, 20 Nm (modes: D, DL1, DL4, B, BL1, and BL4) [15]

Images of the soot concentration (Figure 20) revealed lower soot concentration in the PME flame. The factors that may contribute to the above addressed features of the PME flame are:

a. The higher viscosity may result in larger size of its droplet;
b. The higher density of PME results in longer but narrower spray penetration;

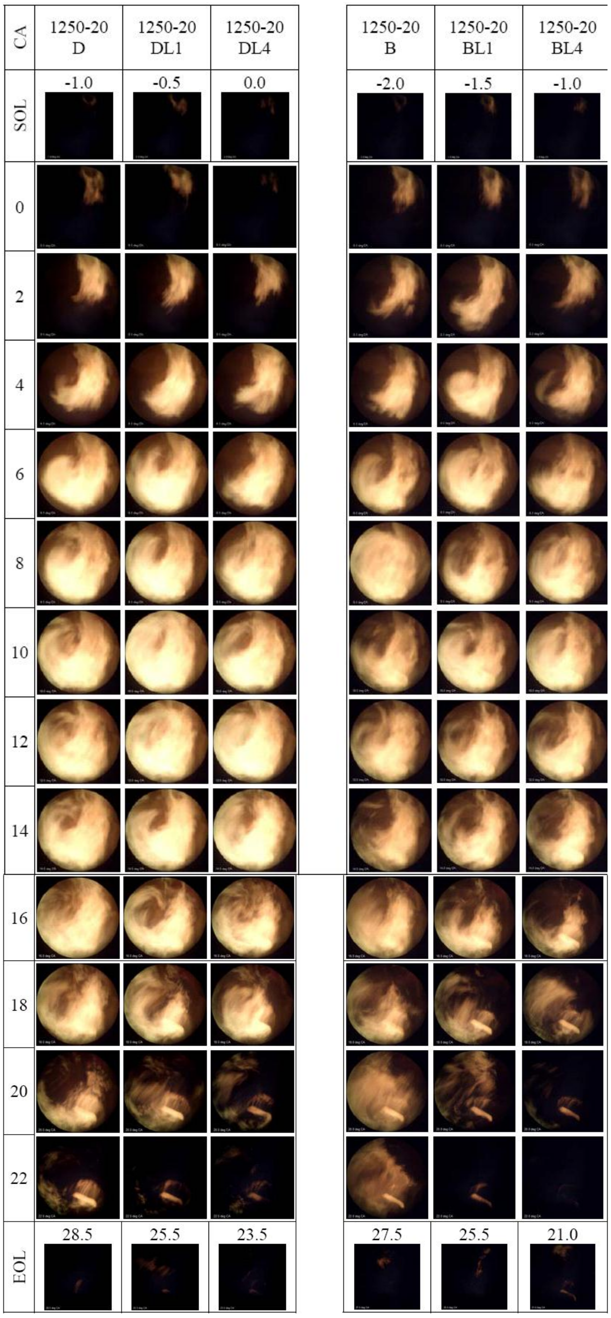


Figure 17 Sequential images of combustion process at 1250 rev/min, 20 Nm (modes: D, DL1, DL4, B, BL1, and BL4) [15]

c. The lower heating value requires larger amount of mass injected per cycle (and hence higher energy for vaporization is required);
d. Its inherent lower volatility contribute to the worse vaporization and mixing with the air within available time during combustion

process, leads to lower PME flame temperature.

e. The lower adiabatic flame temperature of PME compared to that of diesel.

At 2000 rev/min, the difference between diesel and PME spray and combustion at this speed is similar to that at 1250 rev/min. However, these differences between PME and diesel seemed much higher.

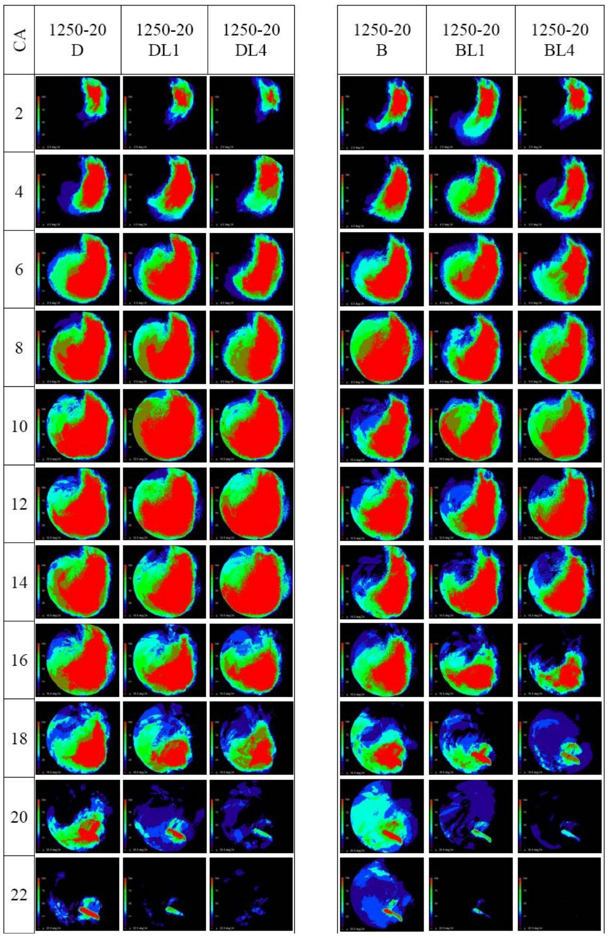


Figure 18 Images of flame probability distribution at 1250 rev/min, 20 Nm (modes: D, DL1, DL4, B, BL1, and BL4) [15]

The PME spray was longer and narrower. Once the combustion took place, the diesel spray rapidly spread out but with larger flow momentum the PME spray seemed to penetrate longer and impinge to the chamber wall. Although the PME injection was earlier and it was also combusted earlier than the diesel, its combustion evolution was weaker than diesel's combustion. The percentage area of 100% flame probability and area of flame temperature of PME were much lower than that of diesel; about [10%-15%].

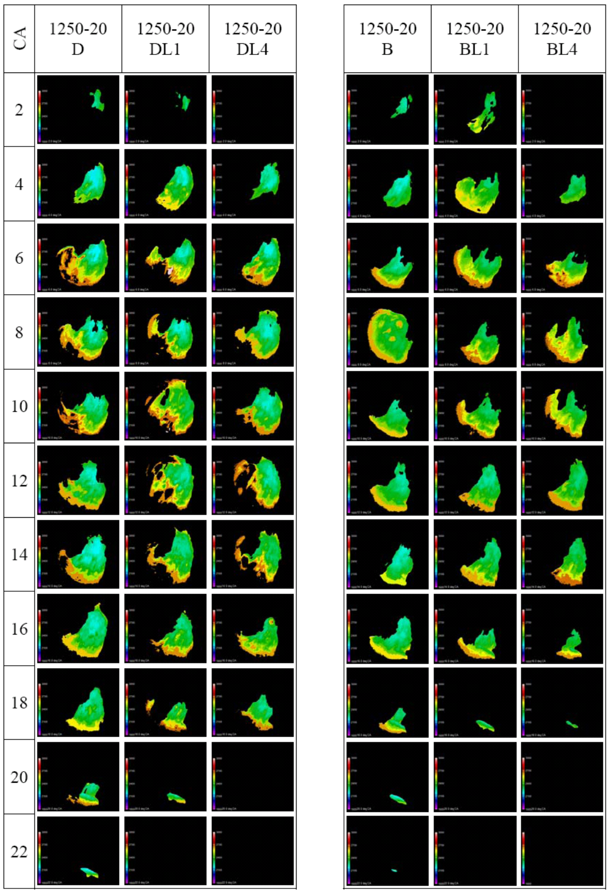


Figure 19 Images of flame temperature distribution at 1250 rev/min, 20 Nm (modes: D, DL1, DL4, B, BL1, and BL4) [15]

6.2.2 The development of the process with dual fuelling

The higher LPG ratio in dual fuel modes leads to two effects. First, premixed combustion and the speed of flame propagation increases but the mixing-controlled combustion for the liquid fuel reduces. Second, the reduced amount of pilot injection, smaller size of the ignition sources, therefore increase the path that the flame needs to propagate to consume all the premixed mixture in the chamber.

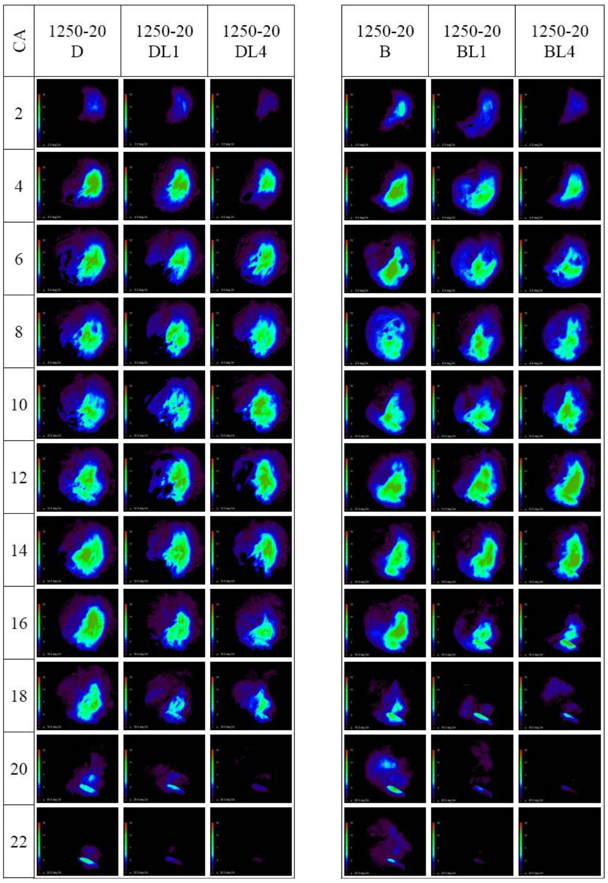


Figure 20 Images of soot concentration distribution at 1250 rev/min, 20 Nm (modes: D, DL1, DL4, B, BL1, and BL4). [15]

Firstly, consider the diesel dual in mode 1250-20-DL4, the sequential images appeared with lesser luminous and the flame temperature seemed lower and the flame area seemed smaller than that in 1250-20-D and 1250-20-DL1. In the mode 1250-20-DL1 although the combustion started later, it was enhanced by the premixed mixture and became faster just after auto-ignition of the pilot injection. The percentage of 100% probability area was slightly larger than that in diesel mode. Accompanied with wider flame, the flame temperature was also higher and initially the area of high flame temperature, above 2300 and 2600K, was wider compared to that in neat diesel mode.

With PME dual fuelling, it can also be recognized that the flame, in mode 1250-20-BL1, is even initially weaker than that in mode 1250-20-B, it spreads faster and envelops almost all the area. In contrast, in mode 1250-20-BL4, the sequential images appear with less luminous and the flame temperature, soot concentration seems lower than that in 1250-20-B and 1250-20-BL1.

Due to the benefit of oxygen content in the PME accompanied with the faster combustion of the LPG in the vicinity of the initial PME flame, the combustion of all the 3 modes then ended with smaller area of high flame probability distribution than that with diesel mode.

The development of the combustion process with higher speed, 2000 rev/min was enhanced by the premixed mixture and became faster just after auto-ignition of the pilot injection. The percentage of 100% probability area was only slightly smaller than that in diesel mode during the early combustion stage then enhanced and its flame became wider as well as combustion was shorter than the diesel.

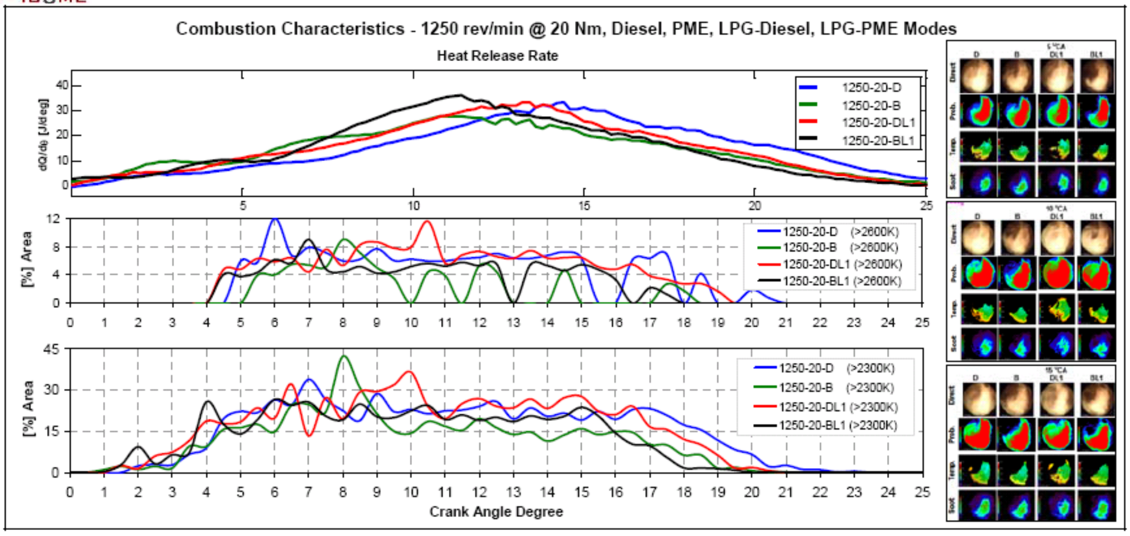


Figure 21 HRR and area of flame temperature above 2300 and 2600K at 1250 rev/min, 20Nm, (mode D, DL1, B, and BL1) [15]

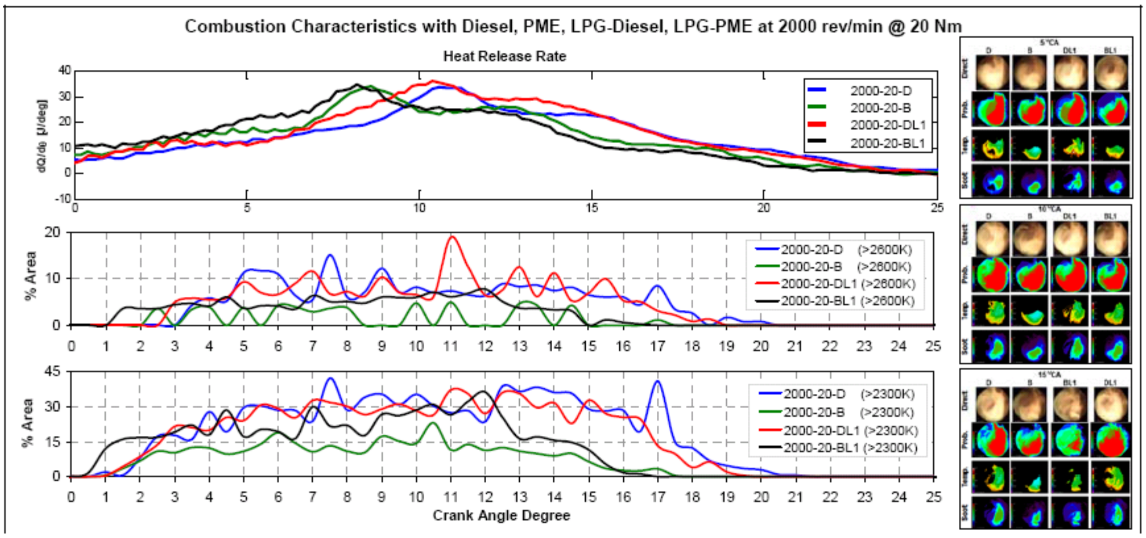


Figure 22 HRR and area of flame temperature above 2300 and 2600K at 2000 rev/min, 20Nm, (mode D, DL1, B, and BL1) [15]

The development of the combustion process with higher speed, 2000 rev/min was enhanced by the premixed mixture and became faster just after auto-ignition of the pilot injection. The percentage of 100% probability area was only slightly smaller than that in diesel mode during the early combustion stage then enhanced and its flame became wider as well as combustion was shorter than the diesel.

In summary, at these two speeds and low load (20Nm), all operation modes with neat PME, LPG-PME appeared far to reach the result as good as diesel combustion. Comparison in the soot concentration, for 1250 and 2000 rev/min, it can be seen that the soot concentration with all modes was very low.

6.2.3 The effect of injection timing

Three values of injection timing were evaluated. The total brake energy conversion efficiencies corresponding to these modes of operation are shown in Figure 23 and 24. The exhaust gas temperatures are shown in Figure 25 and 26. The following remarks can be recognized.

In LPG-diesel modes, the advanced injection timing i3 resulted in better total energy conversion efficiencies. These relative increases, in mode DL1 and DL4, were: 4% and 2.6%, 4.7% and 4%, 2.7% and 3.6% at the engine working point (1250@20), (2000@40), and (2750@70), respectively. The operation even was observed to be better (ECE fraction of 1.005) than the diesel baseline at mode 2000-70-DL1-i3.

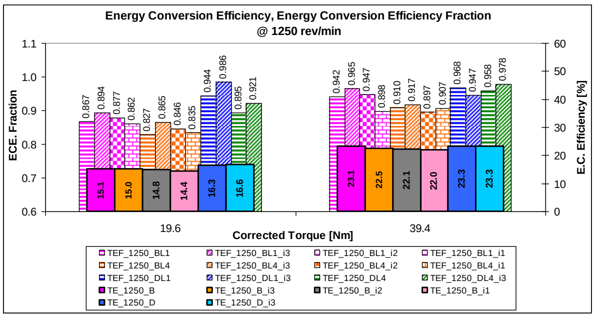


Figure 23 ECE and ECE fraction with different modes of operation and injection timing, at 1250 rev/min. [15]

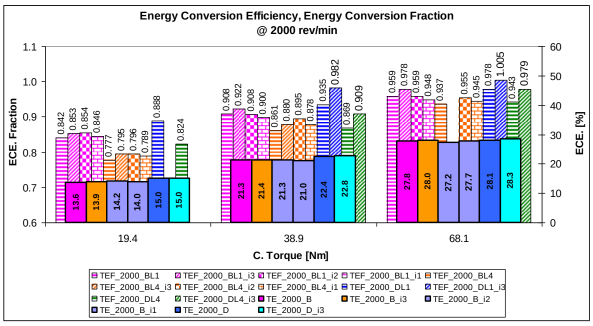


Figure 24 ECE and ECE fraction with different modes of operation and injection timing, at 2000 rev/min. [15]

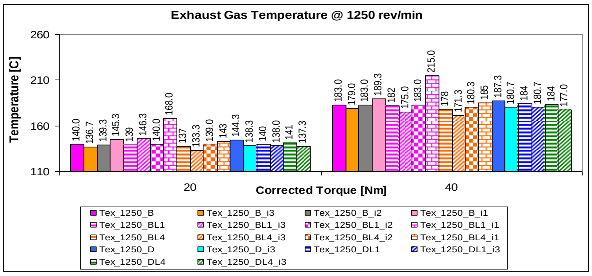


Figure 25 Exhaust gas temperature with different modes of operation and injection timing, at 1250 rev/min. [15]

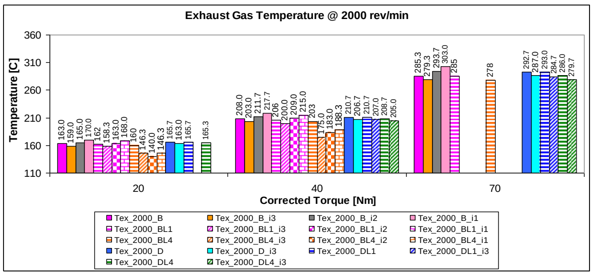


Figure 26 Exhaust gas temperature with different modes of operation and injection timing, at 2000 rev/min. [15]

In LPG-PME modes, the injection timing i3 also resulted in higher total energy conversion efficiencies compared to that with the OEM setting. However, these values were still less than that in the corresponding LPG-diesel modes. In mode BL1, the corresponding relative increases were about 2.7%, 1.0%, 1.0%, and 1.9% at the working point (1250 @ 20), (2000 @ 20) (2000 @ 40), and (2750 @ 70), respectively. Those were about 4% and 0.7% at 1250 rev/min @ 20 and 40Nm, and 1.8%, 1.9%, at 2000 rev/min @ 20 and 40 Nm in mode BL4 of LPG supply. The injection timing i2 seemed not to result significant change in the efficiency in dual fuel whereas the injection timing i1 caused reduction since the combustion took place later, leading to higher energy loss brought by the exhaust gas. With this injection timing (i1), the exhaust gas temperatures in neat PME and LPG-PME modes increased significantly.

7. Conclusions and Further Works

The engine could be able to operate with LPG-PME at the planned condition similar as that with LPG-diesel without end gas knock. The substitution appeared lower at low speed and load but comparable at 2000 rev/in. As fuelled with LPG-PME there was also deterioration in the total energy conversion efficiency and the trend was similar to that with LPG-diesel; the deterioration increased with increased LPG ratio or decreased engine loads. The levels of deterioration appeared higher at low speed and load due to the long injection period accompanied with the less turbulent in the pre chamber. The deterioration was higher than that in the corresponding LPG-diesel operation.

The injection timing was earlier when the engine was fuelled with the PME. This is thought due to the higher adiabatic bulk modulus of the PME and the "pump effect" resulted from the higher viscosity, larger volume per cycle of the PME fuel.

The ignition delay in PME/LPG-PME operation was shorter than that in the corresponding diesel/LPG-diesel operation. This is thought due to the total effect of two factors: the earlier injection of PME has negative effect whereas the oxygen content in PME has positive effect on the ignition delay. These shorter delays in case of neat PME and LPG-PME dual fuel ascertain that the effect of oxygen content of the PME on the delay is dominant over the effect of reduced pressure and temperature brought by earlier injection.

At all modes of PME/LPG-PME, the late stage of combustion in neat PME/ LPG-PME modes seemed to be longer than that in neat diesel/ LPG-diesel modes. This might be resulted from the longer injection duration and lower volatility of the PME compared with the diesel. With LPG-PME dual fuelling, the reasonable LPG ratios were L4 at almost all operating points.

Although the dual fuelling with premixed LPG-air and either diesel or PME pilot injection seemed to provide better combustion recognized by the shorter combustion duration and the movement of the center of HRR area towards TDC, the DF total energy conversion efficiencies were worse than that in the neat liquid fuel. The possible reasons are the energy loss by blow-by and the incomplete combustion in the main chamber because of its geometry which has been optimized for diesel operation.

Based on the observation of phenomena in the pre-chamber with two color method, it is revealed that the LPG-PME dual fuel seemed to result in worse combustion than the LPG-diesel dual fuel; appearing far to reach the result as good as diesel combustion.

Further works that shall be considered are listed. First, the reason of high COV_{IMEP} in dual fuel operation regardless liquid fuel type at 2000 rev/min remains unclear. Next, the investigation on exhaust emissions of LPG-PME dual fuel has also not been considered. In addition, finally, the negative effect of the combustion chamber geometry has been mentioned but not quantified. Further investigation is needed to fill these gaps.

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