The Effect of PME as the pilot injection in LPG-PME Dual Fuel in An IDI Compression Ignition Engine

การศึกษาผลของการใช้ปาล์มไบโอดีเซลช่วยจุดระเบิดในระบบเชื้อเพลิงร่วมที่ใช้ก๊าซหุงต้มในเครื่องยนต์ดีเซลชนิดห้องเผาไหม้ล่วงหน้า

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บทคัดย่อ

บทความวิจัยนี้มีวัตถุประสงค์ที่จะนำเสนอผลของการใช้ไบโอดีเซลเป็นเชื้อเพลิงช่วยจุดระเบิดในเครื่องยนต์ระบบเชื้อเพลิงร่วมในเครื่องยนต์ดีเซล 4 สูบ ชนิด IDI โดยแบ่งออกเป็นการนำเสนอผลของอิทธิพลของการใช้ก๊าซ LPG ซึ่งควบคุมการจ่ายโดยใช้วงจรอิเลคทรอนิกส์ต่อการทำงานของระบบเชื้อเพลิงคู่ LPG-ดีเซล ตามด้วยการนำเสนอผลการศึกษาผลต่อการฉีดและการเผาไหม้ของระบบเชื้อเพลิงคู่ LPG-ปาล์มไบโอดีเซล (PME) ซึ่งได้ดำเนินการบนแท่นทดสอบที่สภาวะคงตัว ตามจุดทดสอบที่เลือกจากวัฏจักร ECE15 + EUDC ในช่วงแรงบิด 10-70 N.m และความเร็วรอบเครื่องยนต์ 1250-2750 รอบต่อนาที โดยที่แต่ละจุดทดสอบได้นำข้อมูลปริมาณการใช้เชื้อเพลิง ความดันในท่อจ่ายเชื้อเพลิงและความดันในห้องเผาไหม้ มาวิเคราะห์เปรียบเทียบประสิทธิภาพการแปลงพลังงาน การสิ้นเปลืองเชื้อเพลิงรวม ปริมาณการทดแทนดีเซล การปลดปล่อยความร้อนสุทธิ พร้อมนำเสนอผลเปรียบเทียบไว้ในบทความนี้

Abstract

This investigation which aimed to identify the effect of biodiesel as the pilot injection in dual fuelled engine can be divided into 2 parts. Firstly, effects of liquefied petroleum gas (LPG) premixed charge-palm biodiesel (PME) dual fuelled on engine operation was studied. Next, the investigation continued 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 indirect injection (IDI) compression ignition (CI) engine, at selected high probability operating points corresponding to the ECE15+EUDC cycle, covering the range [10-70] Nm @ [1250-2750] rev/min. The engine ran at overall lean mixture. In this investigation, the LPG-air premixed mixture was maintained at four fixed values by an electronic controlling system. The acquired data included basic parameters, pressure history of fuel line and combustion chambers. The comparative analysis deal with: energy conversion efficiencies, specific total energy consumption, liquid fuel substitution, net heat release.

Keywords: Dual Fuel, LPG, Palm biodiesel, combustion

1. Introduction

Dual fuel (DF) engine is an engine that energy release in its operating cycle comes from two different fuels. The first fuel, having high cetane number, is called pilot which is acted as a numerous ignition sources to burn the mixture of air, residual gas, and a second fuel. The second fuel, with relatively high octane number, is called main fuel that be introduced to form homogeneously mixture during the intake process.

DF operation represents advantages compared to diesel counterparts and spark ignition (SI) engines: theoretically higher thermal efficiency resulted from faster burning, less toxic emission, high power density, strong ignition sources providing more reliable, less sensitive with respect to changes in the second fuel composition, lower fuel cost, ability to operate with many types of alternative fuel; gaseous fuels or alcohols, and ability to switch back as necessary.

In homogeneous DF 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. The dual fuel engine also operates at higher total fuel-air equivalent ratio since inducted air is partially replaced by the second fuel. A pilot-injection of oxygenated fuel such as palm biodiesel (PME) which can offset, to some extent, the prolonged ignition delay is expected to offer an improvement at the low end as well as having improved potential at the high end. PME is also expected to improve combustion process of richer fuel-air mixture of dual fuel engine in mid-load range as well.

To obtain a proper knowledge of PME in DF combustion phenomena of IDI combustion chambers, therefore, it has increasingly relied more on fundamental knowledge of quasi static (i.e., uniform in pressure and temperature) analysis requiring the normally used indicating methods.[1,2,3 and 4]

2. Objectives

The aims of this research are to investigate dual fuel combustion characteristic of liquefied petroleum gas (LPG) premixed charge dual fuelled engine with biodiesel (palm methyl ester, PME) as the pilot, to identify the effect of biodiesel injection timing on LPG-PME dual fuel combustion characteristic.

3. Experimental Apparatus and setup

3.1 Engine

The engine under study is a commercial IDI, water cooled four cylinders, in-line, natural aspirated engine. Engine combustion chamber system is of Ricardo Comet MK Vb type, with a downstream glow-plug. The following chart displays the main dimensions:

Engine type	WL 81
Pre-chamber	Swirl Pre-chamber
Displacement	2499 cm^3
Bore	93 mm
Stroke	92 mm
Compression ratio	21.6
Injection pump	Rotary distributor type
Injector starting pressure	11.4 – 12.1 MPa

The engine was connected to an AVL alpha-40 eddy-current dynamometer. The combustion characteristics were investigated by analyzing the cylinder pressure data (indirect analysis: heat release prediction; direct analysis: statistical analysis). Pressure history in pre-chamber, main chamber and liquid fuel line has been recorded by a high speed data acquisition system. The system includes:

- A DEWETRON acquisition system model 5000-CA-SE.
- Two AVL pressure transducers model AVL GU12P are used for combustion chamber pressure measurement.
- A Kistler high pressure transducer, model 607C1 is installed to the fuel injection line 15 mm upstream the fuel injector. Pressure history from this transducer is used to predict the start of injection (SOI).

Arrangement for these transducers is shown in Figure 1 and Figure 2.

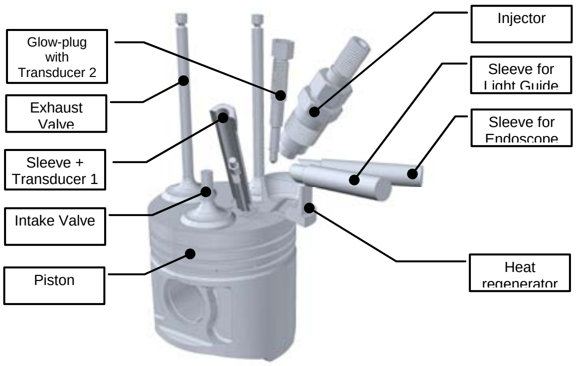

Figure 1 Setup of transducers for cylinder pressure measurement.

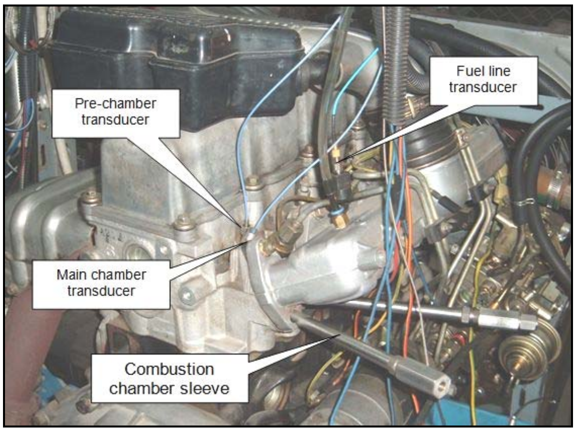

Figure 2 Setup for fuel line and combustion chamber pressure measurement, combustion chamber visualization.

The schematic arrangement of experimental set up is shown in Figure 3.

3.2 LPG supply system

LPG, in gaseous form, is supplied by an injector to a mixer installed upstream the manifold to ensure the well mixing of LPG with air before entering to the cylinder. The injectors and LPG regulator used in this study are from a conversion kit for SI engine of Zavoli Co., Italy. The injector’s duty cycle (LPG flow rate) is controlled by an in-house designed controller board. The injector’s duty cycle can be set manually from 7% to 62%

(resolution of 1%) and observed on the display of an oscilloscope (Tektronik TDS 210).

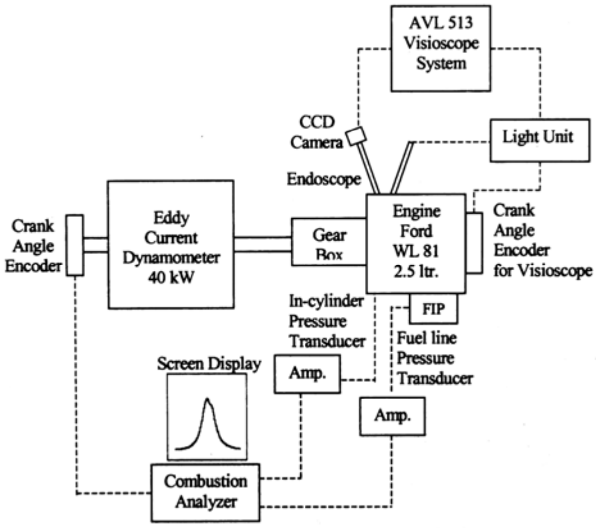

Figure 3 Schematic arrangement of experimental set up.

3.3 Measurement of liquid fuel consumption

The mass of liquid fuel consumed, M_f, was directly measured by a high precision balance, resolution of 2 grams, product of ACUWEIGH Co., Ltd. The duration, Δt, corresponding to the mass of liquid fuel consumed was measured by an ALBA stop-watch with a resolution of 1/100 second.

The air consumed by the engine was measured, by “air box” method, using an BS 1042 orifice. Differential pressure across the orifice was measured by an inclined manometer. Temperature of the air inside the tank was measured by thermocouple type K.

Exhaust gas, lube oil, intake air, and ambient temperatures are measured by thermocouples connected to a display unit.

3.4 Fuel

The general properties of the PME used in this investigation is shown in Table 1.

4. Experimental Procedure

The experiments were carried out at constant speed, steady state conditions at selected high probability operating points along ECE 15 + EUDC test cycle, as shown in Table 2. A comparison in energy consumption, conversion efficiency among these operations with different LPG energy fraction (L1, L2 L3 and L4) with diesel and palm biodiesel, respectively, will be performed.

In dual fuel modes four different LPG flow for different values of the premixed mixture strength (fuel-air equivalent ratio) are controlled by the injection system via adjusting the duty cycle of the LPG injector. The values are selected to achieve as closed as possible of the mixture strength among the considered speeds. Detail information is given in Table 3.

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	°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

Table 2 Test point along ECE 15+EUDC cycle.

Brake torque [Nm] / Speed [rev/min]	10	20	30	40	70
1250	X	X	X	X	
2000	X	X	X	X	X
2750	X	X	X	X	X

As the engine reaches steady state, speed, torque, air and fuel consumption, engine operating pressure and temperature for both fuels were recorded during each test. Then, specific fuel consumption, specific energy consumption, fuel-air equivalent ratio, ignition delay, net heat release rate and accumulative net heat release rate are then evaluated.

Table 3 Operation modes, LPG injection system setting and the LPG flow rate

LPG mode	Speed [rev/min]	Duty cycle [%]	LPG rate [g/s]	LPG rate [kg/h]
L1	1250	7	0.108568	0.390844
	2000	13	0.177428	0.638742
	2750	17	0.230780	0.830808
L2	1250	12	0.164563	0.592427
	2000	19	0.255608	0.920188
	2750	27	0.349435	1.257967
L3	1250	14	0.189597	0.682549
	2000	23	0.301438	1.085177
	2750	31	0.397902	1.432446
L4	1250	17	0.230780	0.830808
	2000	28	0.362473	1.304904
	2750	38	0.481863	1.734705

5. Experimental results and discussion [5]

5.1 Effect of the biodiesel on the LPG-PME operation

Engine operation with four fixed values of the mixture strength of the LPG-air premixed charge at different engine loads and speeds had been investigated. The engine could operate at DF mode with LPG at all tested matrix without end-gas knocking. The highest biodiesel substitutions at 1250, 2000, and 2750 rev/min, shown in Figure 4 to 6 were about 60%, 45%, and 38%, respectively. Figure 7 to 9 and Figure 10 to 12 depict the energy conversion efficiencies (ECE) and specific total energy consumption (STEC) in neat PME and LPG-PME modes at the three 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. There was deterioration trend in the energy conversion efficiency in neat PME as well as LPG-PME operation. The relative deterioration was lower as the engine load increased. At each fixed speed, the deterioration increased as the LPG ratio increased. As revealed, at 1250 rev/min the deterioration was about 8% to 1% in neat PME 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, and from 13.5% to 6% in LPG-PME modes, as the load increased from 10 to 70Nm. Those at 2750 rev/min were from 4.6% to 4% in neat PME and from 20.4% to 13.4% in LPG-PME modes.

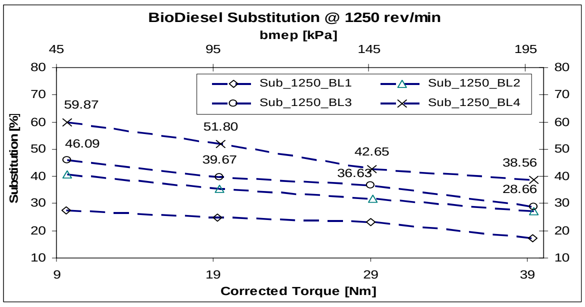

Figure 4 Substitution at 1250 rev/min, LPG- PME modes

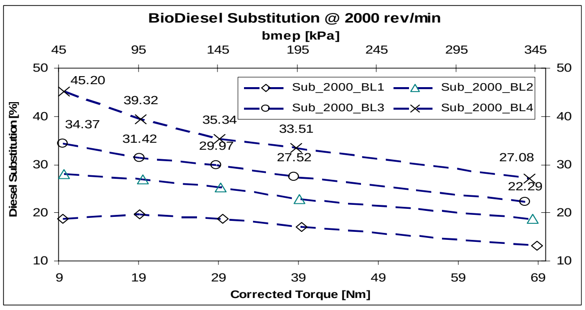

Figure 5 Substitution at 2000 rev/min LPG- PME modes

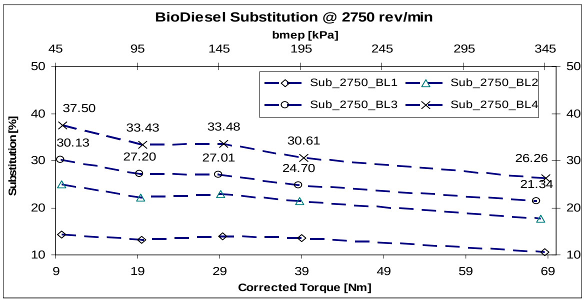

Figure 6 Substitution at 2750 rev/min, LPG- PME modes

It's also noted that LPG-PME dual fuel resulted in significant decreases in the exhaust gas temperature while the lube oil temperature, were almost unchanged.

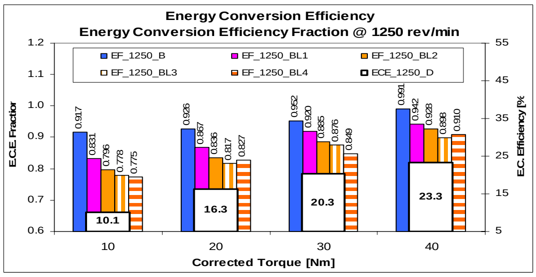

Figure 7 ECE at 1250 rev/min, PME & LPG- PME modes

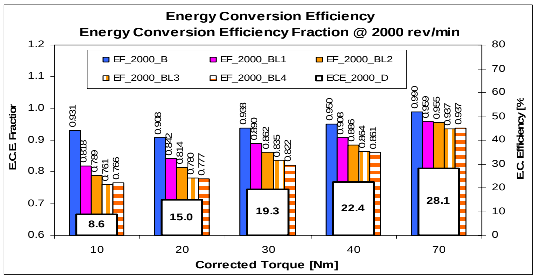

Figure 8 ECE at 2000 rev/min, PME & LPG- PME modes

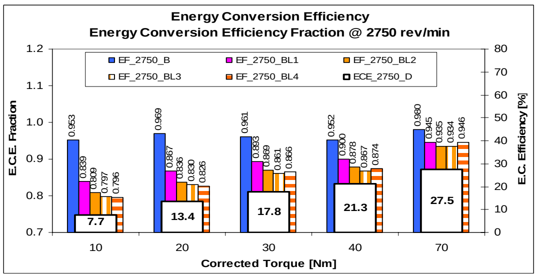

Figure 9 ECE at 2750 rev/min, PME & LPG- PME modes

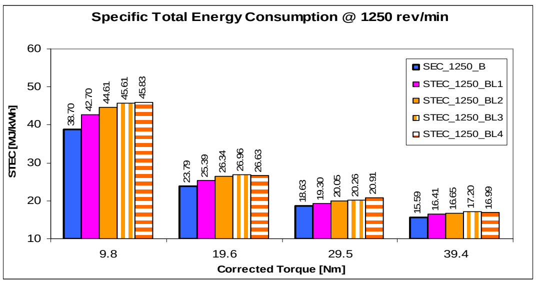

Figure 10 STEC at 1250 rev/min, PME & LPG- PME modes

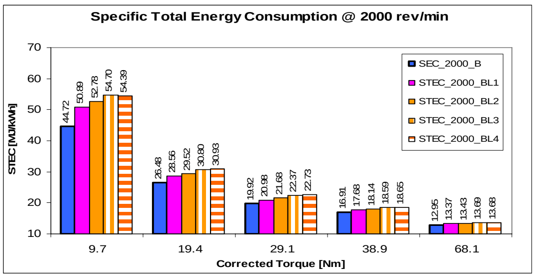

Figure 11 STEC at 2000 rev/min, PME & LPG- PME modes

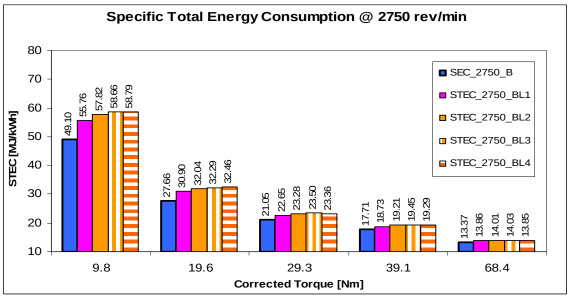

Figure 12 STEC at 2750 rev/min, PME & LPG- PME modes

5.2 Determination of liquid injection events

At the same condition of engine speed, torque, and fuel system setting, the injection timing in PME and LPG-PME modes was always earlier than that in diesel modes. This is due to the resultant effects of the PME properties. It has higher viscosity, higher adiabatic bulk modulus, higher density, and lower heating value compared to the diesel, as seen in Table1.

The start of injection (SOI) of liquid fuel into the combustion chamber is the instant at which the needle starts to lift out of its seat. The end of injection (EOI) is the instant at which it moves back and locates properly in the seat. Injection duration is the period between these two instants. In this study, as shown in Figure 13, the SOI determination is obtained from the crank angle of the appearance of the first local maximum fuel line pressure (before TDC, in the pumping stroke). This also agreed with the result from Reitz R.D. and Choi C.Y. [6].

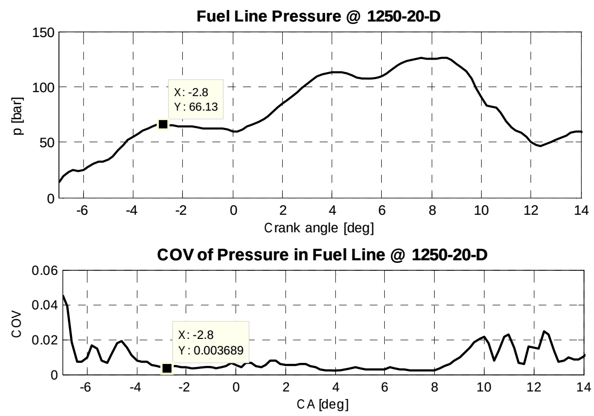

Figure 13 Average fuel line pressure and COV of fuel line pressure of 120 consecutive cycles, at engine speed of 1250 rev/min, 20 Nm, diesel fuel

5.3 Ignition delay, start of combustion and combustion duration

The result of detected start of injection (SOI), start of combustion (SOC) and ignition delays at different engine speeds, loads, and gas energy fraction are shown in Figure 14 to 16. At 1250 rev/min, the delay seemed unchanged at lowest load and speed. The maximum retardation in the delay was 1.2 degree at 30 Nm, with mode BL3. At 2000 rev/min, the LPG ratio seemed to have less effect to the delay, especially at higher load (70Nm). The maximum changes in the delay were about 0.4 degree, at lowest load, with mode BL1 and BL2. Almost the similar trend was observed at 2750 rev/min: the highest change was 0.4 degree at lowest load.

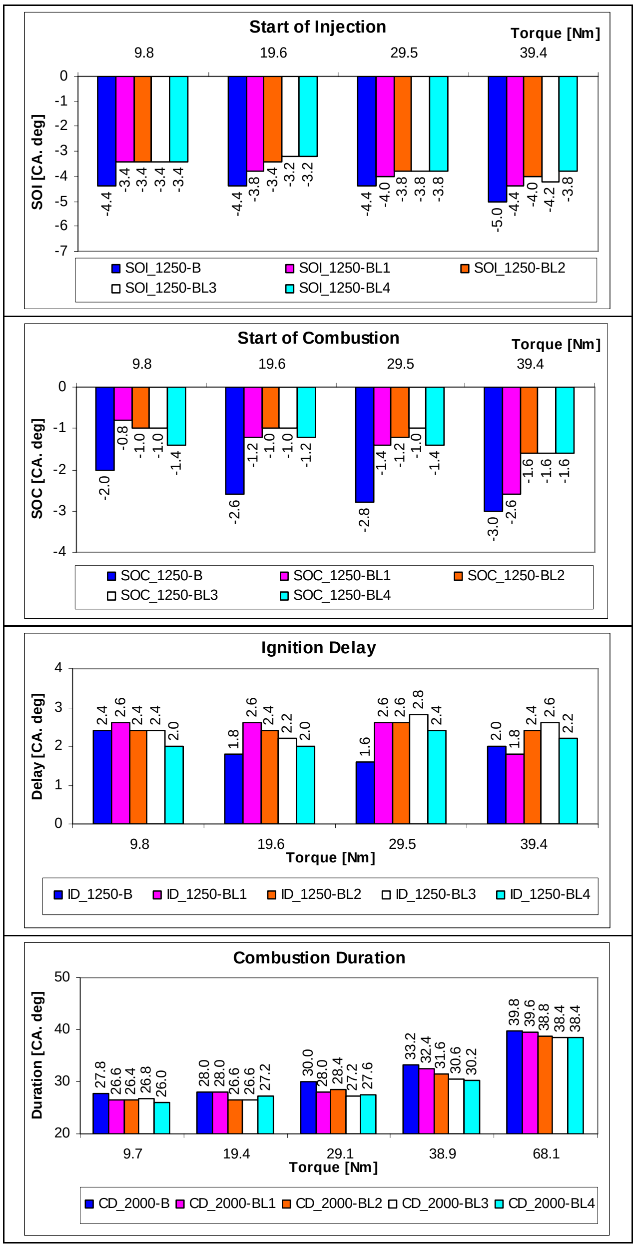

Figure 14 SOI, SOC, ignition delay and combustion duration at 1250 rev/min, PME and LPG-PME

Compared with neat diesel or LPG-diesel modes, at 1250 rev/min, 2000 rev/min, and 2750 rev/min, the corresponding with neat PME or LPG-PME produced shorter ignition delays. These differences may come from the combination effect of two factors: the earlier injection of PME has negative effect while the oxygen content in the PME has positive effect on the ignition delay. These shorter delays in cases of neat PME and LPG-PME ascertain that the effect of oxygen content in the PME is dominant over the effect of reduced pressure and temperature brought by the earlier injection.

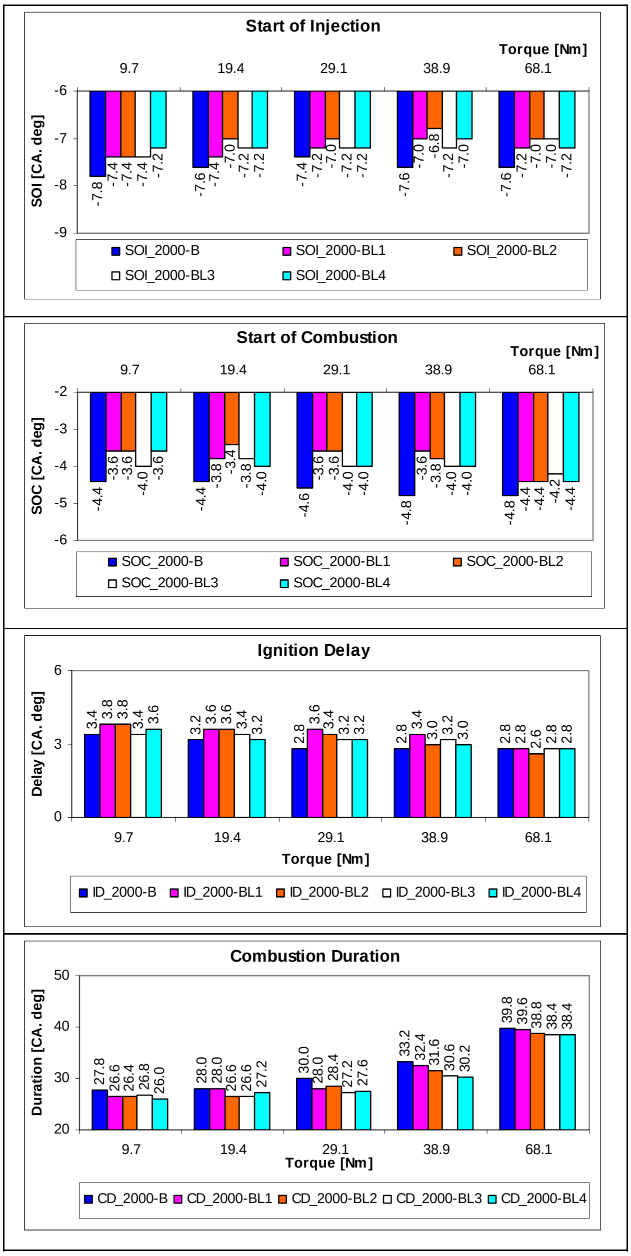

Figure 15 SOI, SOC, ignition delay and combustion duration at 2000 rev/min, PME and LPG-PME

The change in SOC was resulted from the change in the SOI and the delay. At low and medium speeds, the SOC in DF mode were later than that in neat PME mode but almost earlier than that in neat diesel or LPG-diesel modes. With constant engine torques at 1250 rev/min, compared to neat PME, the SOC starts later; in a range of [0.6-1.8]^{O}CA, with increased LPG except with mode BL4 (highest LPG ratio). At 2000 rev/min, the SOC were also later, in a range of [0.4-1.0] ^{O}CA, than that with neat diesel.

However, at this speed, when the load increased the retardation became smaller ([0.4-0.6] ^{O}CA) as the LPG increased. At 2750 rev/min, the SOC were almost not sensitive to loads and LPG ratio. Compare with neat PME, the combustion duration in LPG-PME DF mode were also shorter. Moreover, the combustion duration become smaller with increased LPG.

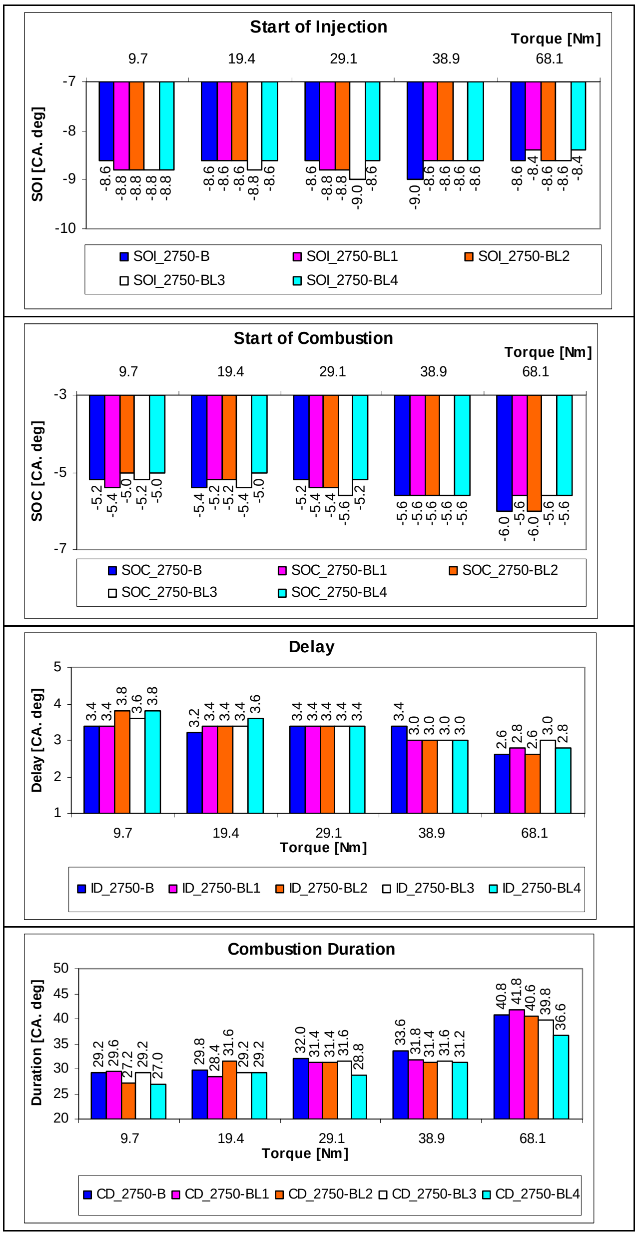

Figure 16 SOI, SOC, ignition delay and combustion duration at 2750 rev/min, PME and LPG-PME

5.4 The heat release rate, peak of pressure, pressure rise rate, and cyclic variation

The net HRR, PRR with neat diesel, LPG-diesel, PME, and LPG-PME modes are presented, in Figure 17 to 23. These parameters for modes D, DL1, and DL4 are presented by the solid-blue, solid-green, and solid-red lines, respectively. Those for neat modes B, BL1, and BL4 are presented by the dash-blue, dash-green, and dash-red lines, respectively. From these figures, these following features are revealed:

(a) The HRR profiles in LPG-PME modes were, also, almost similar to that in neat diesel modes. However, due to earlier injection compared to diesel, the HRR curve moved towards TDC (the left hand side) with increased LPG. The movement was about [1-2] degree at 1250 rev/min, [1.8-2.3] at 2000 rev/min, and [1.4-2.5] at 2750 rev/min. This causes either the positive effect as it is theoretically higher fuel conversion efficiency or the negative effect as it causes higher heat transfer to the chamber wall. The earlier SOC, hence earlier EOC, gives much more time for the products to expand, leading to significant decrease in exhaust gas temperatures. The movement in LPG-PME modes was higher compared to neat PME, and appeared proportional to the LPG ratio.

(b) The earlier combustion in PME/ LPG-PME modes caused the higher integrated HR than that of diesel/ LPG-diesel modes during the early and main stages but lower at the late stage of combustion. This maintained the high in-cylinder pressure for a period aTDC, or even lifted the pressure up about 10 bars during this period.

(c) At all speeds and loads, whereas the peaks of HRR in neat PME modes were lower than that in neat diesel modes, the peaks in LPG-PME modes might be higher than that in LPG-diesel modes (such as 1250-20-BL1, 2750-20-BL1, 2750-20-BL1 modes). The peaks of HRR in neat PME/ LPG-PME modes occurred approximate [2-3] degree earlier than the corresponding in neat diesel/ LPG-diesel modes.

(d) Although the LPG-PME dual fuel operation provided earlier heat release compared to the neat diesel and LPG-diesel, the total energy conversion efficiencies were lower. The following factors might contribute to this deterioration:

- The incomplete combustion of the LPG in the pre chamber.

- The higher combustion chamber heat loss.

The predicted Net HR results indicate that the reasonable modes were mostly DL4, as shown in Table 4. The reasonable substitution was in the ranges of about [34%-58%], [27%-40%], and [21%-40%] at 1250, 2000, and 2750 rev/min, respectively.

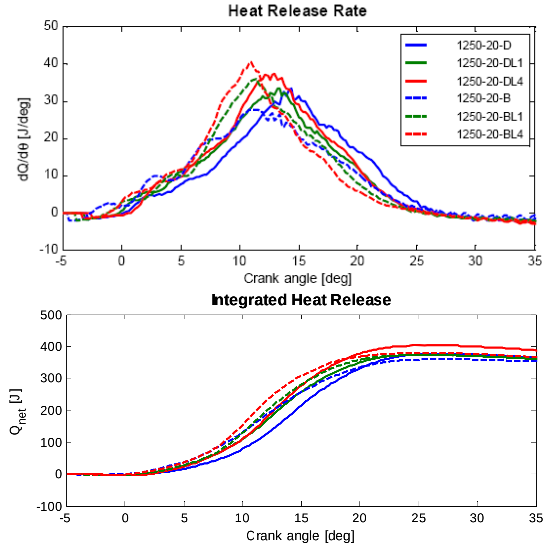

Figure 17 HRR at 1250 rev/min, 20Nm

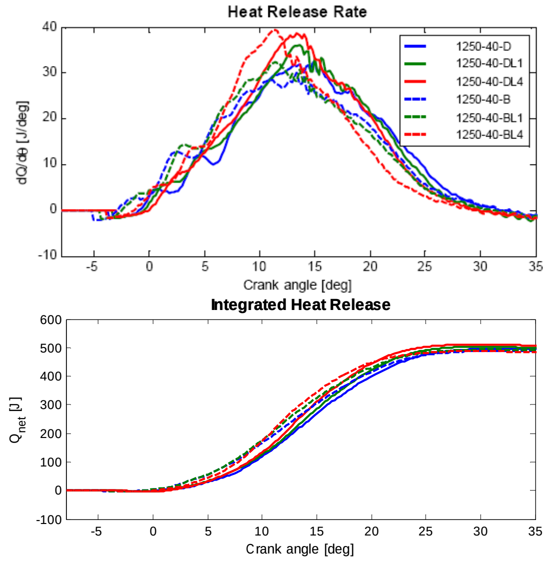

Figure 18 HRR at 1250 rev/min, 40 Nm

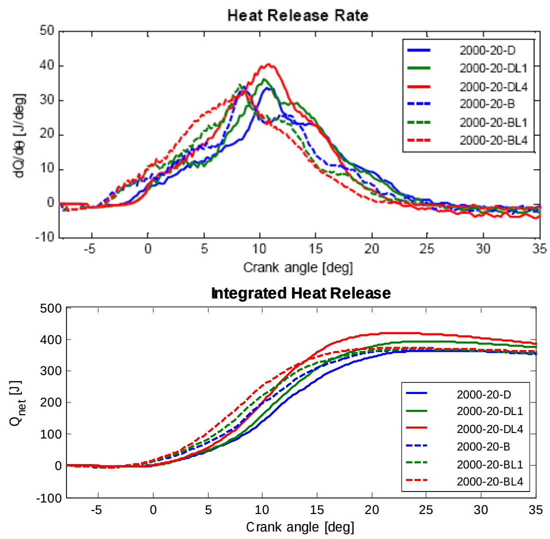

Figure 19 Net HRR at 2000 rev/min, 20 Nm

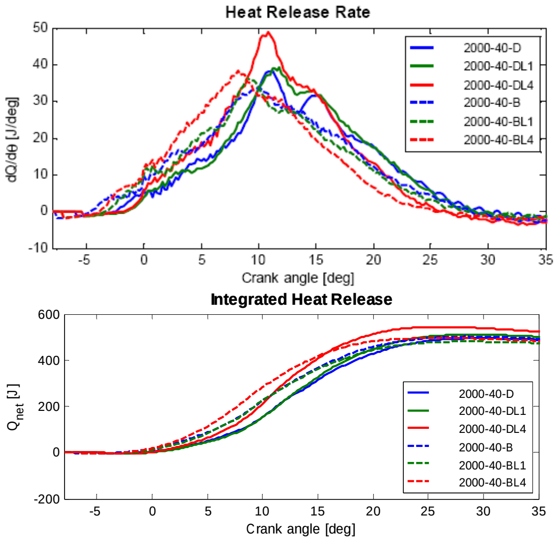

Figure 20 HRR at 2000 rev/min, 40 Nm

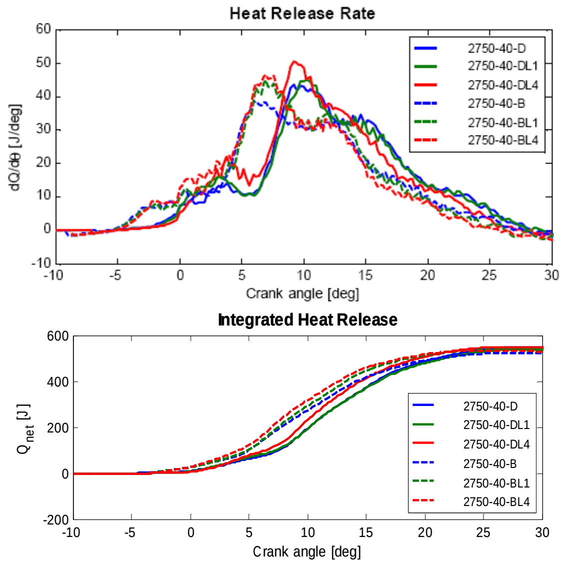

Figure 22 HRR at 2750 rev/min, 40 Nm

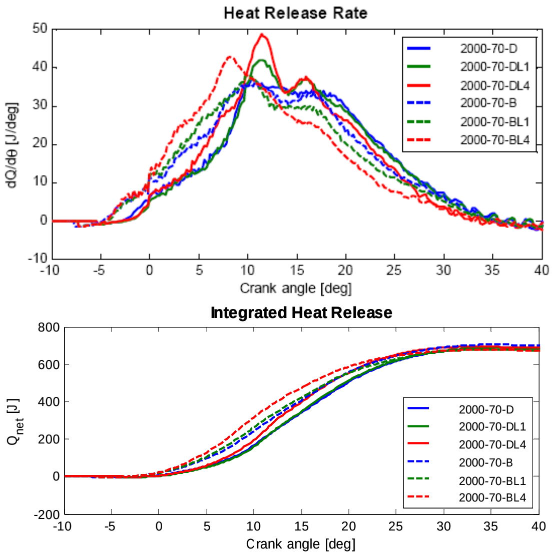

Figure 21 HRR at 2000 rev/min, 70 Nm

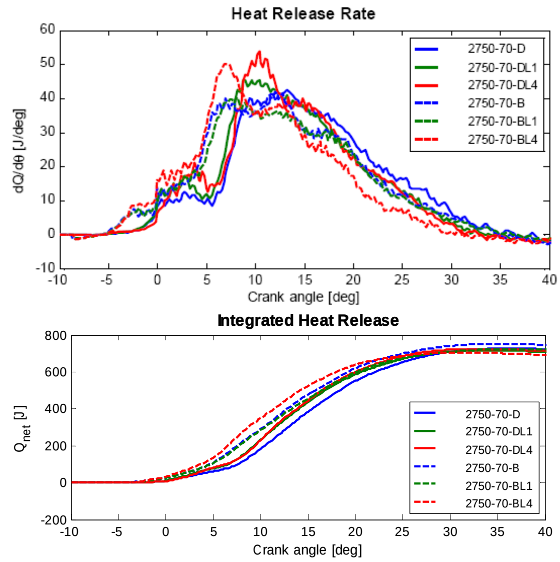

Figure 23 HRR at 2750 rev/min, 70 Nm

Table 4 Modes of LPG-PME operation that provide highest net heat release

Engine speed (Rev/min)	Brake Torque (Nm)	Operation mode	Fuel-Air equivalent ratio		Substitution Sub (%)
			ϕg	ϕt	
1250	40	BL4	0.13920	0.31806	38.56
	30	BL4	0.13721	0.28472	42.65
	20	BL4	0.13738	0.24857	51.8
	10	BL4	0.13694	0.20762	59.9
2000	70	BL4	0.14159	0.45769	27.1
	40	BL3	0.11738	0.35106	27.5
	30	BL4	0.14149	0.32574	35.3
	20	BL4	0.14119	0.29372	39.3
2750	70	BL3	0.11309	0.46692	21.3
	40	BL4	0.13954	0.38021	30.6
	30	BL4	0.13977	0.34599	33.5
	20	BL4	0.14001	0.32106	33.4

6. Conclusion

PME provided smooth, knock-free dual fuel operation, at all planned test points and LPG-air premixed charge, with the energy conversion efficiency deterioration increased at lower speeds and higher LPG ratios. Achieved substitution was about 26%-27% at high speed, load and 60%-65% at low speed, load. Ignition delays which retarded within 1.2^{O}CA with LPG-PME, especially at low speed, increased LPG. The start of combustion which was found to advance in LPG-PME due to their shorter ignition delays and advanced SOI caused by the higher bulk modulus and viscosity of the PME. LPG-PME produced faster combustion compared to the neat liquid fuel, leading to reduced exhaust gas temperatures and the centers of heat release area's moving of towards TDC. While the coefficient of variation of IMEP was comparable, the combustion noise of LPG-PME was slightly higher.

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8. References

[1] Ricart, L. M., et al "In-Cylinder Measurement and Modeling of Liquid Fuel Spray Penetration in a Heavy-Duty Diesel Engine", SAE 971591, 1997.

[2] Chmela, F. G., et al "Rate of Heat Release Prediction for Direct Injection Diesel Engines Based on Purely Mixing Controlled Combustion", SAE 1999-01-0186, 1999.

[3] Winklhofer, E., "Diesel Combustion – a Hierarchy of Simple Effects?", ERCOFTAC Bulletin No. 38, 1998.

[4] Zellat, M., et al "Three Dimensional Modeling of Combustion and Soot Formation in an Indirect Injection Diesel Engine", SAE 900254, 1990.

[5] Duc Phan Minh. A Study on the LPG Dual Fual combustion characteristics of an indirect injection compressions ignition engine. Ph.D. Thesis. Mechanincal Engineering Chulalongkorn University. 2006

[6] Choi CY, Bower GR, and Reitz RD. Effect of biodiesel blended fuels and multiple injections on DI Diesel engines. SAE Paper 970218, 1997.