

Performance of Palm Oil-Based Biodiesel Fuels in a Single Cylinder Direct Injection Engine

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INTRODUCTION

Direct injection diesel engines are fuel-efficient power plants for automotive applications because of their superior fuel economy over spark ignition and indirect injection engines. However, the rising cost of diesel, stringent emission regulations and depletion of petroleum reserves are forcing us to search for new sources of energy. In the quest to reduce greenhouse gas emissions, biodiesel is a fuel to consider.

Biodiesel refers to methyl or ethyl esters derived from vegetable oils (Reed, 1993). It can also refer to pyrolysis products, diesel-vegetable oil blends, microemulsions of alcohols and water in vegetable oils and fermentation butanol. Vegetable oil was in fact the first diesel when Dr Rudolf Diesel ran his then newly developed engine on it in 1895. He successfully demonstrated his engine at the World Exhibition in Paris in 1900 using peanut oil.

Biodiesel is an alternative fuel that may help reduce the world's dependence on fossil fuels with very significant environmental benefits. It is an oxygenate, sulphur-free and biodegradable fuel. Its content of oxygen helps improve its combustion efficiency.

Biodiesel has gained much attention in recent years due to the increasing environmental awareness. It is produced from renewable resources and, more importantly, is a clean burning fuel that does not contribute to the net increase of carbon dioxide.

Malaysia has embarked on an extensive palm oil-based alternative fuel programme since 1982 (Choo *et al.*, 1995). The programme includes the development of production technology to convert palm oil to palm methyl esters (palm diesel), a pilot plant study of palm diesel production as well as exhaustive evaluation of using palm diesel in conventional diesel engines (both stationary and in vehicles).

A new phase of the programme started in 2002 to investigate blending palm oil with conventional diesel. Various blends incorporating different palm oil products, such as processed liquid palm oil (PLPO) and bleached and deodorized palm oil (RBDPO)

with petroleum diesel are being evaluated as possible diesel substitutes for industrial (Ahmad *et al.*, 2001) and vehicle fuels.

Blends of palm oil are promising renewable fuels for diesel engines and have been paid more attention in Malaysia. However, palm oil in diesel changes the fuel's physico-chemical properties, including viscosity, cloud point, cetane number, heat value and boiling point. For example, the viscosity is decreased causing the cetane number to fall. A high cetane number usually ensures good cold starting, reduced noise and long engine endurance. Therefore, the properties of blend must be maintained within certain limits.

Any alternative fuel for an internal combustion engine should satisfy certain criteria, such as requiring minimum engine modification, offering uncompromised engine life and being not hazardous to human health and the environment during production, transportation, storage and utilization. The fuel also must be low cost, renewable and in abundant and stable supply.

Ethanol is a possible alternative fuel for diesel engines. But, its cetane number is too low, which resulting in excessive ignition delay and knock. In addition, its

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production cost is high and specific energy (energy content per unit mass) low. Therefore, ethanol is unlikely to be an alternative fuel for diesel engines (Quick, 1980).

Previous researcher (McDonald, 1995) evaluated the emissions and engine performance by neat methyl soyate and its 30% blend with low sulphur No. 2 diesel were evaluated in a Caterpillar 3304 PCNA engine. The fuels were run through the ISO 8178-CI 8-mode off-highway test, and it was found that the peak brake power decreased 9% and the brake specific fuel consumption increased 13% to 14% for the neat biodiesel due to its lower energy content from No.2 diesel. The biodiesel and 30% blend increased the volatile organic fraction of the particulates compared with the reference fuel, but greatly decreased the solid carbon portion. The emissions of carbon monoxide (CO) and hydrocarbon (HC) were also reduced with biodiesel.

It has been reported by many researchers that biodiesel has lower exhaust emissions compared with petroleum-based diesel 20% less CO, 30% less HC, 50% less soot and 40% less particulate matter (McDonald *et al.*, 1995;

Schmidt and van Gerpen, 1996). Also, since biodiesel is a sulphur-free fuel, it has 99% less sulphur oxide (SO_x) emission than diesel fuel which SO_x is the main cause of acid rain. However, biodiesel produces 10% to 15% higher NO_x emissions.

BIODIESEL – BLENDED PALM OILS

In this study, the biodiesel fuels were provided by the Malaysian Palm Oil Board (MPOB). Four test fuels were used, including No. 2 diesel (D2) designated as the reference fuel. The other fuels were blends between PLPO and diesel at various volumetric proportions, B2, B5 and B10 (denoting 2%, 5% and 10% PLPO in petroleum diesel), respectively. Some selected properties of the biodiesels and reference fuel are given in *Table 1*.

The calorific values of the biodiesels were lower than that of the reference fuel. The higher the percentages of palm oil, the higher were the density. The pour point was 9.0°C, which would limit their use in the tropics. The distillation data indicated that the fuels were generally volatile with the dynamic viscosity increasing with the palm oil content. Traces

of sulphur were detected in these fuels. However, their tendency to form carbon during combustion will be similar to that of the reference fuel.

THE EXPERIMENTAL SET-UP

The experimental programme was carried out using a YANMAR-L70AE single cylinder engine, natural aspirated, direct injection diesel engine with a bore of 78 mm, stroke 62 mm and compression ratio of 19.5. It is air-cooled, low speed with maximum power output rated at 4.9 kW at 3400 rpm.

A precision ONOSOKKI volumetric fuel flow meter was used to determine the engine fuel consumption. Its sensor was mounted between the fuel tank and fuel pump with the return line rerouted back to the supply line just before the fuel injector. A 40.64 cm (19 inch) rack exhaust gas analyser TOCSIN IGD 300 system complete with 3 m sampling probe was used to measure CO and NO_x emissions. For smoke density, a dedicated Sampling Pump Type EFAW/65 BOSCH smoke meter was used. The darkening of the filter paper from smoke meter was assessed by an evaluating unit, type

TABLE 1. FUEL PROPERTIES

Property	Diesel	B2	B5	B10
Heat value (MJ kg ⁻¹)	45.91	45.165	45.135	44.78
Cloud point (°C)	-	13.4	13.7	13.6
Density @ 15°C (kg litre ⁻¹)	0.84	0.8441	0.8452	0.8497
Total sulphur (mass %)	-	0.2	0.194	0.178
Viscosity @ 40°C (cSt)	3.6	4.0	4.1	4.9
Carbon residue (wt. %)	<0.1	<0.1	<0.1	<0.1
Ash content (mass %)	0.001	0.003	0.004	0.004
Flash point (°C)	98.0	81.1	81.1	83.1
Pour point (°C)	15.0	9.0	9.0	9.0
Initial boiling point (°C)	228	197.3	197.3	200.0
10% vol. recovered (°C)	258	242.2	243.2	246.2
50% vol. recovered (°C)	298	290.5	293.1	298.7
70% vol. recovered (°C)	325	317.6	322.3	331.0
90% vol. recovered (°C)	376	360.8	364.3	356.8
Final boiling point (°C)	400	379.4	367.5	357.5
Final recovery (vol. %)	-	99.1	98.5	98.5
Residue (vol. %)	-	0.5	1.0	1.0
Loss (vol. %)	-	0.4	0.5	0.5

EFAW/68. The sampling probes of the smoke meter and gas analyser were mounted centrally at the end of the engine exhaust pipe.

The engine was directly coupled to an eddy-current brake MAGTROL dynamometer equipped with a load controller. The engine, dynamometer and other auxiliary items were mounted on a seismic steel bed (2 m x 4 m) to cushion any excessive engine vibration during the trial.

The intake engine airflow was measured by a sharp edged orifice mounted in the side of an air box, coupled to the engine inlet to dampen the pulsating air flow into the engine. A manometer tube was used to measure the pressure drop across the orifice.

The overall experimental set-up is shown in *Figure 1*. All the equipments were calibrated in accordance to the respective manufacturer's specifications, prior to conducting the tests.

A water-cooled piezoelectric pressure transducer (Kistler 6061B) was flush mounted with the cylinder head to measure the combustion pressure. The flush mounting was preferred to minimize the lag in the pressure signal and avoid pipe connecting passage resonance (Hua Zhao and Nicos, 2001). The combustion pressure data were averaged over 120 consecutive engine cycles with a crank angle encoder (Kistler 2613B) having resolution of 0.2° crank angle. The heat release rate was computed using the formula proposed by Heywood (Heywood, 1988; Heywood *et al.*, 1985).

In this study, the engine was run at full load using all the fuels to measure the maximum brake power and fuel consumption. Engine torques of 3, 5, 7 and 9 Nm were selected and each load was applied at engine speeds of 2000 rpm and 2300 rpm. The engine performance, in-cylinder data and emissions were recorded.

RESULTS AND DISCUSSION

Engine Performance

In this study, the brake specific fuel consumption (BSFC) was taken as a performance indicator to compare the fuels. *Figure 2* shows the BSFC plots with engine speed for the fuels. Diesel had the lowest BSFC throughout the test. The BSFC for B2, B5 and B10 increased from 0.4% to 3.7% at low and medium engine speeds, and from 4.14% to 13.35% at higher speed. The increase in BSFC was understandable as the biodiesels have approximately 1.6% less energy than the diesel. The higher the palm oil content in the biodiesels, the lower their heating values (*Table 1*), resulting in higher BSFC.

Figure 3 illustrates the engine outputs at full load. A close resemblance occurred at low speed indicating little difference in output between the fuels. However, at higher speed, a clear gap appeared between the alternatives and reference fuel. The maximum reduction brake power recorded was about 3.2% at the highest engine speed.

Figure 4 illustrates the BSFC curves at engine speeds of 2000 rpm and 2300 rpm. The biodiesels have lower heat value than the diesel. As the amount of PLPO in the blends increases, the heat value of the blends decreases. Therefore, to maintain the same brake mean effective pressure (BMEP), more fuel would have to be used increasing the BSFC. The maximum BSFC increment was at the highest engine load at 13% and 10% for 2000 and 2300 rpm, respectively.

The brake thermal efficiency characteristics of the engine on the different fuels are shown in *Figure 5*. It is a good measure in assessing the ability to convert the fuel energy into output. They generally show similar trends and closely resemble one another. The brake thermal efficiencies for B2,

B5 and B10 were lower over the medium to high load range compared to reference diesel. The maximum reductions were about 9% and 4% for the engine speeds of 2000 and 2300 rpm, respectively. This may have been due to lower power that can be generated from the lower calorific formulated fuels. At low engine load, the power outputs were similar for all the fuels.

The peak combustion pressures are summarized in *Figure 6*. The differences between the fuels were generally small. The peak pressures are in close agreement with the increasing NO_x emission (*Figure 8*) when the engine load is increased.

Figure 7 depicts the combustion pressure and heat release profiles. The combustion pressure profiles were smooth for all the fuels, revealing no knocking phenomenon. In the heat release profile, the diffusion combustion hump became more apparent as BMEP was increased for all the fuels.

The small differences in the premixed combustion peaks (in which peak was higher for D2 over B2, B5 and B10) are depicted in *Figure 8*. This may be due to the fact that D2 had a higher volatility. Lower viscosity and lower density of the fuel will result in higher volatility (Mustafa, 2003). Therefore, D2 was easier to vaporize, the better to convert its chemical to mechanical energy in a short time. In general, the heat release patterns for all fuels were similar. The combustion pressure also revealed similar patterns for all the fuels.

Ignition delay is defined as time between the fuel injection and its combustion. The combustion is when the heat release becomes zero (Richard, 1999). The fuel injection of this engine occurred at 14 bTDC °CA (Jiro, 2004). The ignition delay decreased as the blending percentage increased (*Figure 9*) showing that biodiesel was easier to ignite and that the ignition

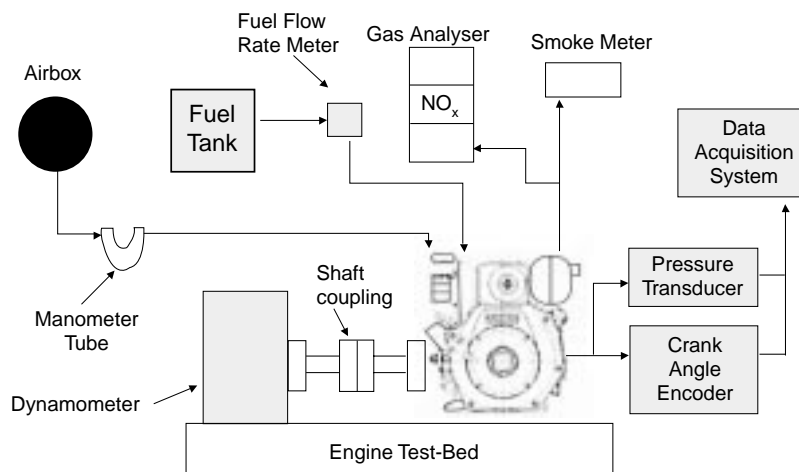


Figure 1. The overall experimental set-up.

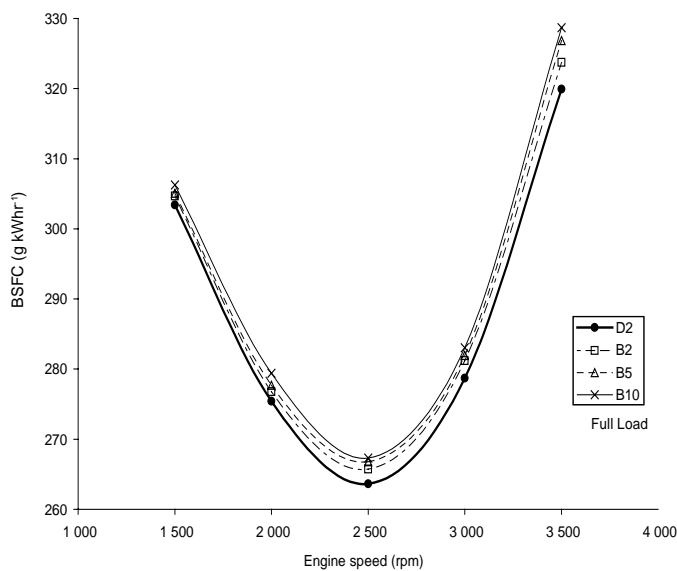


Figure 2. Effect of engine speed variation on brake specific fuel consumption (BSFC).

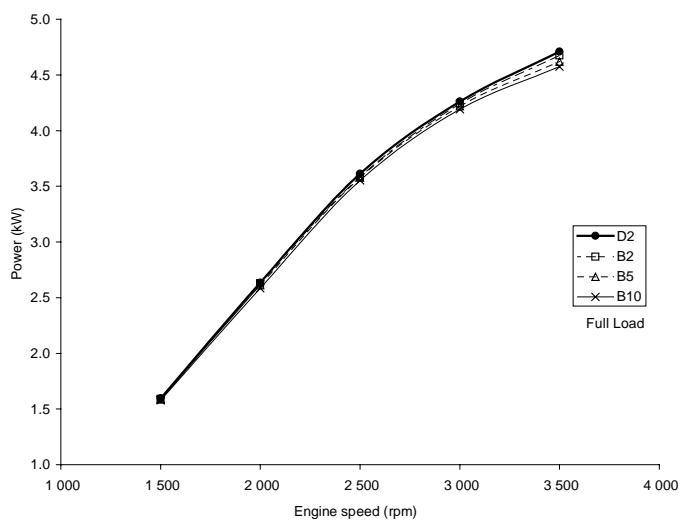


Figure 3. Effect of engine speed variation on brake power.

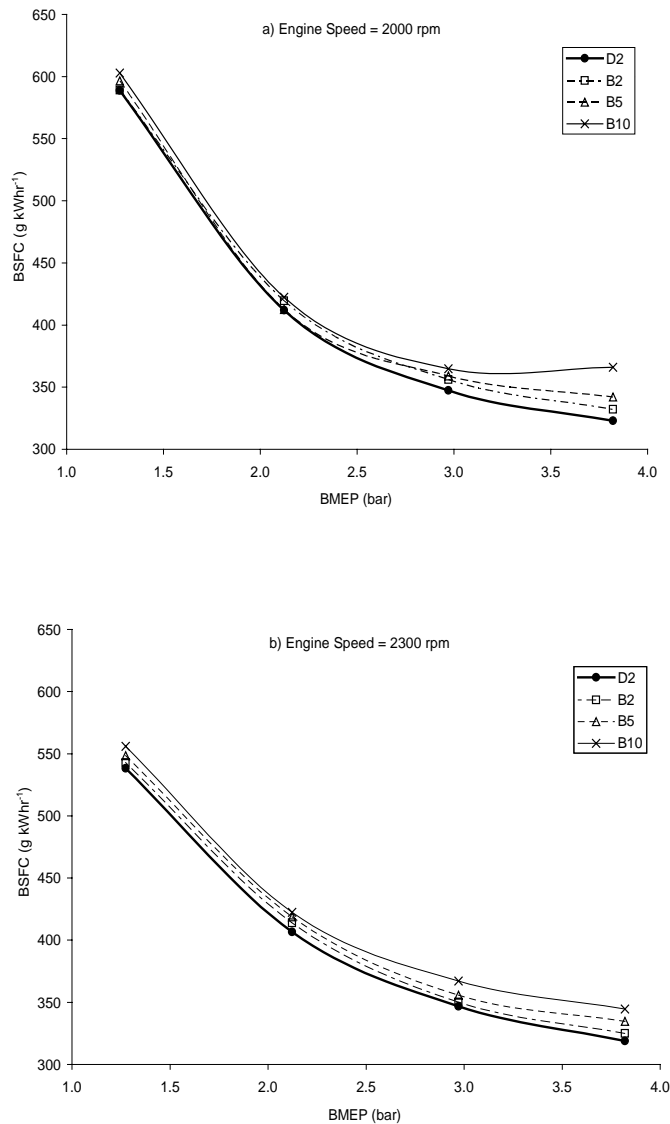


Figure 4. Effect of engine brake mean effective pressure (BMEP) on brake specific fuel consumption (BSFC).

occurred closer at TDC. This effect may be due to the biodiesels having higher oxygen contents.

Engine Exhaust Emission

The NO_x exhaust emissions are shown in Figure 10. For all of the biodiesel fuels, NO_x were higher over the entire load range than diesel fuel at both engine speeds. The highest increases of NO_x were about 16% and 14% for engine speeds of 2000 and 2003 rpm, respectively. The higher NO_x

emission may be due to the oxygen and nitrogen contents in palm oil as both may combine to form NO_x.

The results for smoke number are shown in Figure 11. All the biodiesel fuels had lower smoke number than the diesel throughout the entire load range. The smoke was reduced significantly at medium and high loads as biodiesel content in the blends increased. The maximum reductions were about 17% and 21% for the engine speeds of 2000 and 2300 rpm, respectively.

CONCLUSION

The palm oil diesel blends were successfully tested in a single cylinder unmodified diesel engine. The salient points derived on their use in a typical conventional diesel engine are:

- the biodiesel fuels provided similar brake thermal efficiency curves as diesel at low and medium engine loads. However, BSFC for the biodiesel fuels were slightly higher over the entire engine speed and load. The brake power was

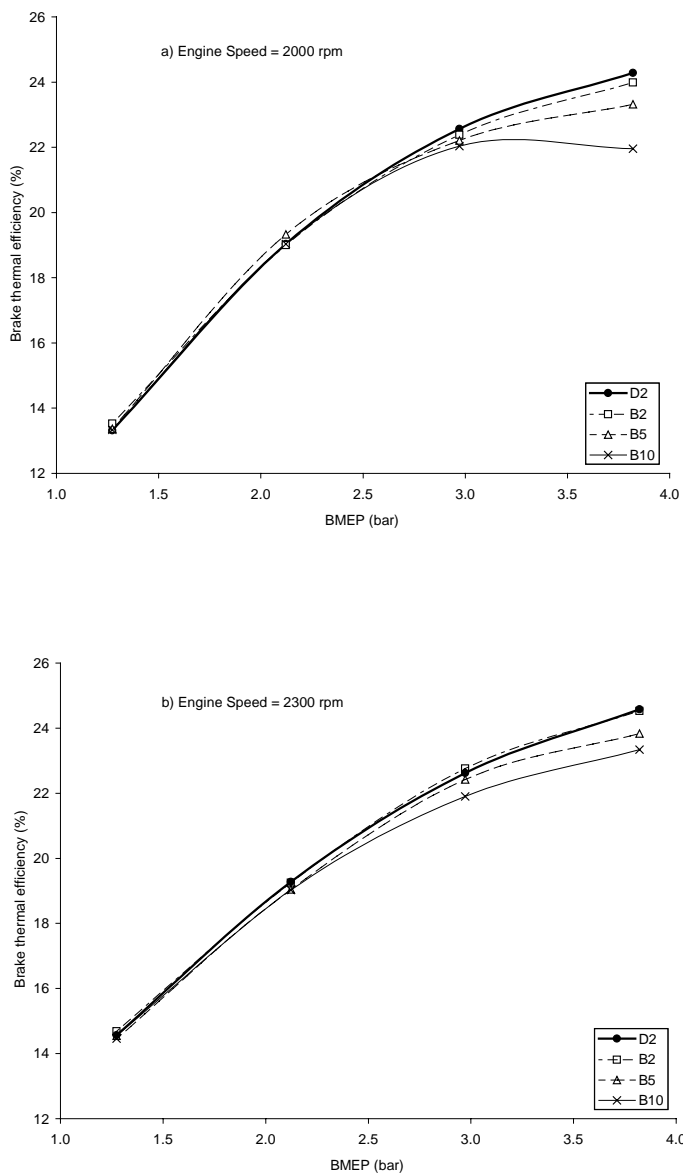


Figure 5. Effect of engine brake mean effective pressure (BMEP) variation on brake thermal efficiency.

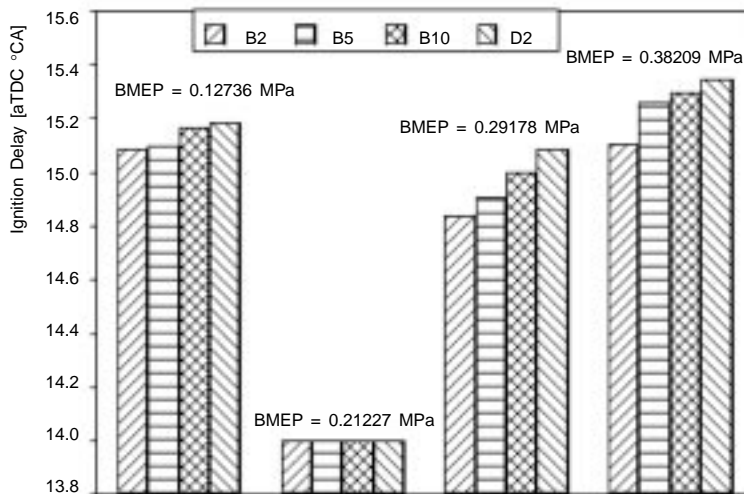


Figure 6. Effect of fuel on peak combustion pressure at 2300 rpm.

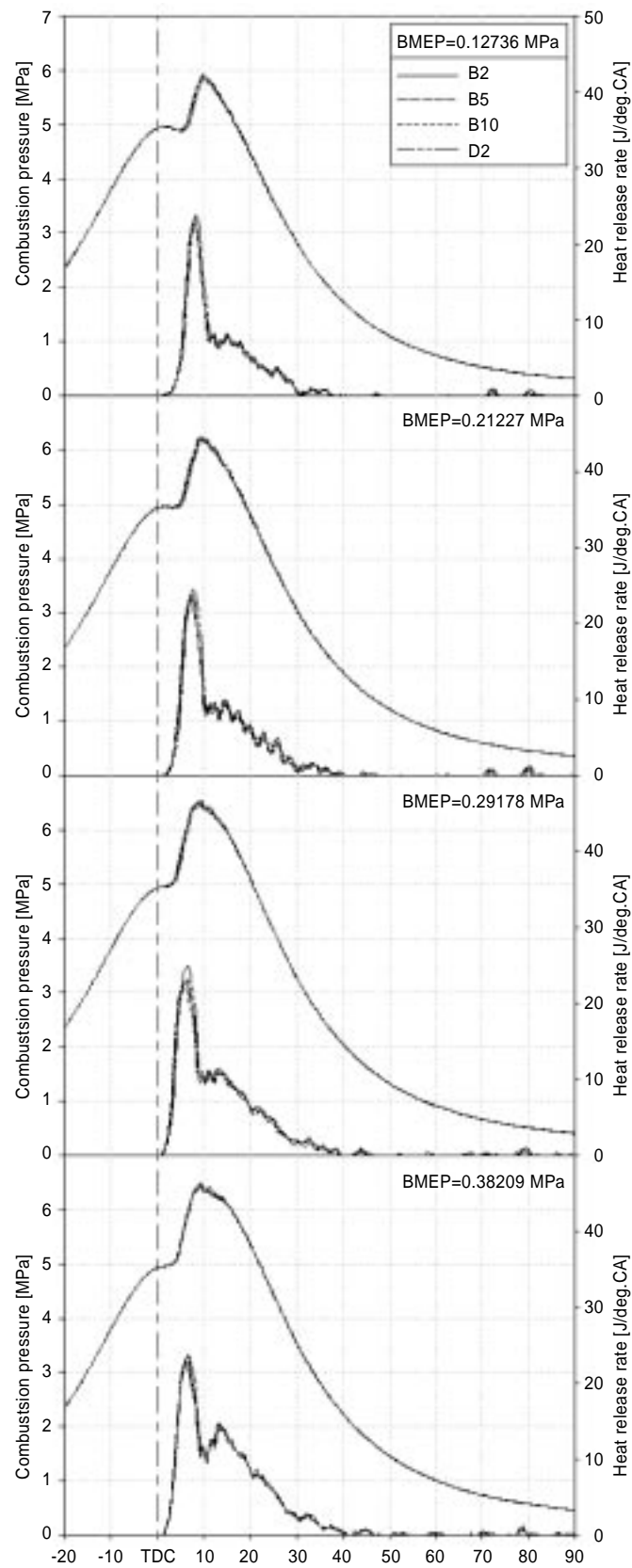


Figure 7. Effect of fuels on combustion pressure and heat release rate at 2300 rpm.

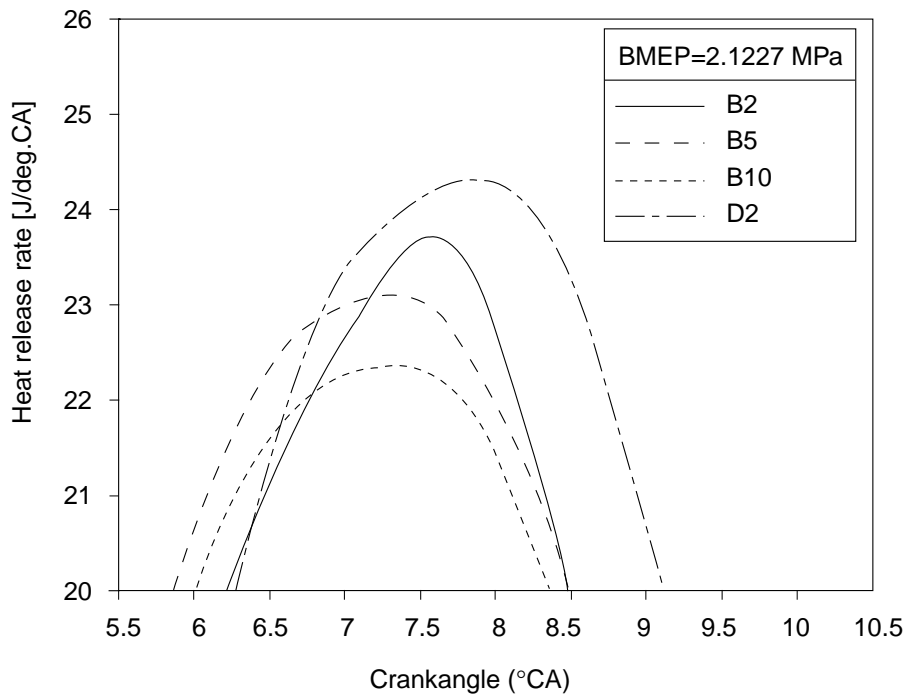


Figure 8. Pre-mixed heat release peaks at 2300 rpm.

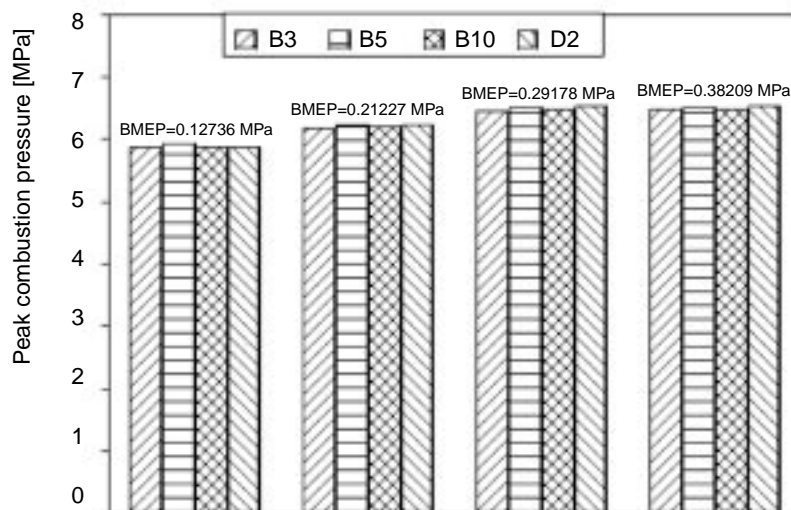


Figure 9. Effect of the fuels on ignition delay at 2300 rpm.

generally lower throughout the engine speed range due to the lower calorific values of the blended fuels.

- the biodiesel fuels produced less smoke than diesel under similar engine operating conditions, probably because palm oil contains oxygen which helps the combustion in the cylinder.
- the biodiesel produced more NO_x emissions, especially at the higher engine load. The

oxygen content in biodiesel may be the cause of this, as more oxygen during combustion will raise the combustion bulk temperature. It was, however, possible that the other properties of biodiesel or interaction with the fuel injection process and combustion chamber dynamics contributed towards the higher NO_x.

- the biodiesel and reference fuels provided similar combustion pressure patterns at low and medium engine loads, suggesting that the biodiesels had no adverse effect in terms of knocking.
- the biodiesel fuels lowered the premixed combustion of heat release because of the lower volatility.
- the ignition delay decreased as the palm oil content increased. This was

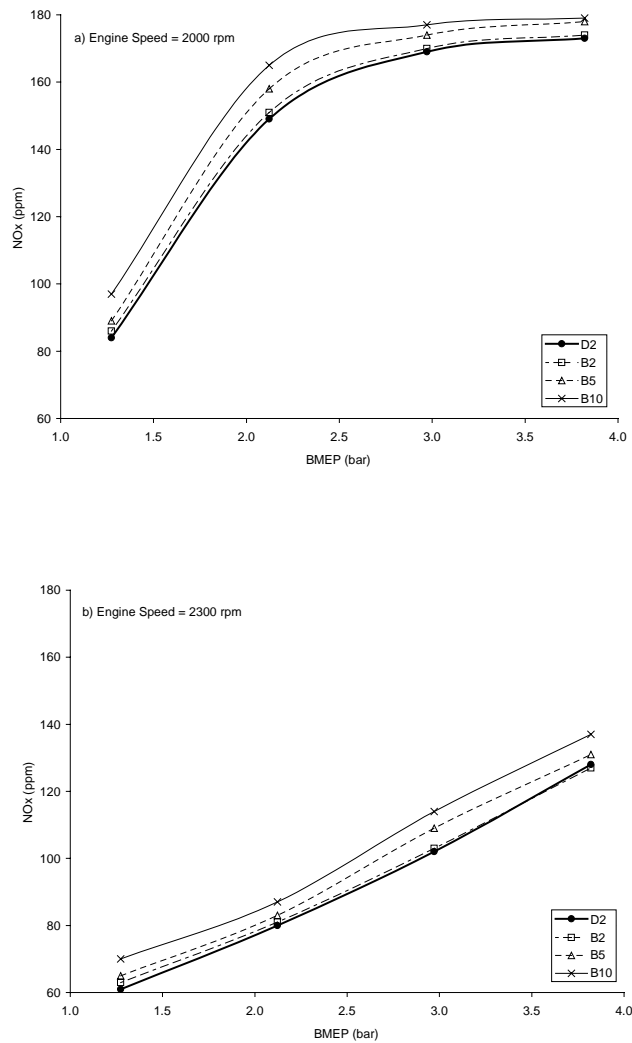


Figure 10. Effect of engine brake mean effective pressure (BMEP) variation on NOx.

suspected to be due to the higher oxygen content in the biodiesel.

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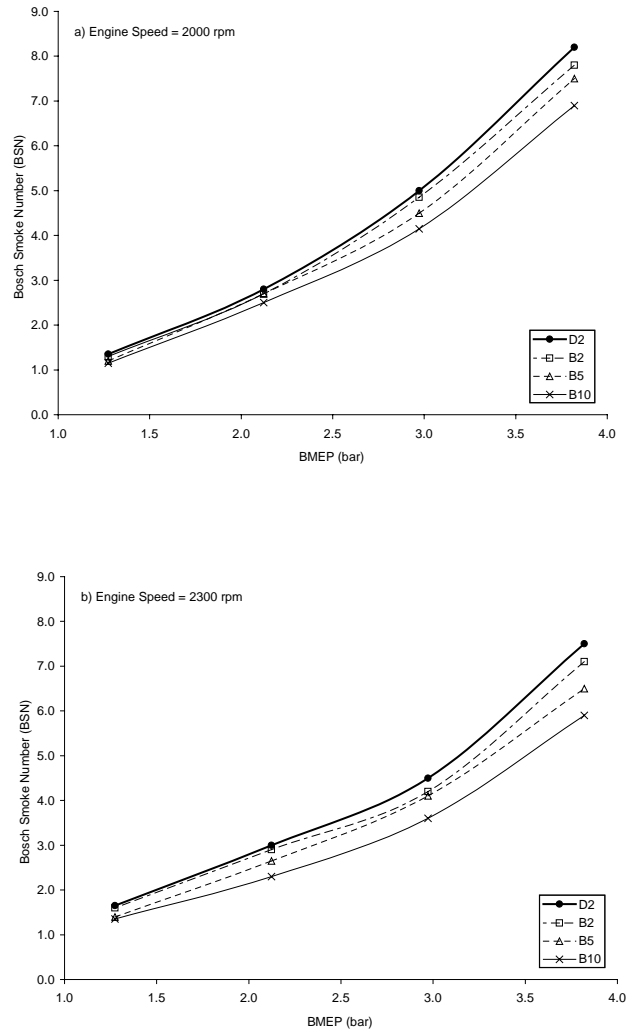


Figure 11. Effect of engine brake mean effective pressure (BMEP) variation on BSN.

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