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Performance and Combustion Characteristics Analysis of Multi-Cylinder CI Engine Using Essential Oil Blends

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Abstract: Essential oils are derived from not-fatty parts of plants and are mostly used in aromatherapy, as well as cosmetics and perfume production. The essential oils market is growing rapidly due to their claimed health benefits. However, because only therapeutic grade oil is required in the medicinal sector, there is a substantial low-value waste stream of essential oils that can be used in the transportation and agricultural sectors. This study investigated the influence of orange, eucalyptus, and tea tree oil on engine performance and combustion characteristics of a multi-cylinder compression ignition engine. Orange, eucalyptus, and tea tree oil were blended with diesel at 10% by volume. For benchmarking, neat diesel and 10% waste cooking biodiesel-diesel blend were also tested. The selected fuels were used to conduct engine test runs with a constant engine speed (1500 RPM (revolutions per minute)) at four loads. As the load increased, frictional power losses decreased for all of the fuel samples and thus mechanical efficiency increased. At higher loads (75% and 100%), only orange oil-diesel blends produced comparable power to diesel and waste cooking biodiesel-diesel blends. Fuel consumption (brake and indicated) for the essential oil-diesel blends was higher when compared to base diesel and waste cooking biodiesel-diesel blends. Thermal efficiency for the essential oil-diesel blends was comparable to base diesel and waste cooking biodiesel-diesel blends. At higher loads, blow-by was lower for essential oil blends as compared to base diesel and waste cooking biodiesel-diesel blends. At 50% and 100% load, peak pressure was lower for all of the essential oil-diesel blends when compared to base diesel and waste cooking biodiesel-diesel blends. From the heat release rate curve, the essential oil-diesel blends ignition delay times were longer because the oils have lower cetane values. Overall, the low-value streams of these essential oils were found to be suitable for use in diesel engines at 10% blends by agricultural producers of these oils.

Keywords: essential oil; engine performance; combustion characteristics; compression ignition engine

1. Introduction

Essential oils are natural volatile oils responsible for many of the fragrances produced by plants, and they are easily obtained using extraction techniques, such as distillation or solvent extraction [1]. These oils are mainly used in the natural medicine sector, due to claimed health benefits, as well as the flavouring and fragrance sector. The market for essential oils has experienced rapid growth in recent years. The high quality required of these products leads to a significant low-value essential oil waste stream, which is available for use in the transportation and agricultural sectors. The use of essential oils in diesel engines has not been thoroughly researched in the past, with only a few studies being performed.

The advantage of using essential oils in a diesel engine is that their properties are similar to those of diesel fuel. For example, both pine oil and eucalyptus oil have similar density (875 and 890 kg/m^3) to diesel (822 kg/m^3), almost same heating value (42.8 and 43.2 MJ/kg) as compared to diesel (42.7 MJ/kg) [2]. However, one of the drawbacks is that these oils have quite low cetane number, which prohibits the use of pure essential oils. For example, pine oil and eucalyptus oil have cetane number of 10 and less than 15, respectively, as compared to 52 of neat diesel, reported by Vallinayagam et al. [2]. As a result, these oils must be blended with diesel to increase the cetane to an acceptable level.

There have been related studies that focused on using various essential oils in diesel engines. Some researchers used clove stem oil (CSO) by blending it with neat diesel at 25% and 50% blend ratio [3,4]. The aim of these studies was to investigate the potentiality of using CSO as an alternative fuel for diesel engines. As the CSO content in blended fuel increased, the brake thermal efficiency (BTE) increased, resulting in high energy output. This can be attributed to the high oxygen content of the CSO, which improved the combustion efficiency. However, as the CSO content in blended fuel increased, fuel consumption, and brake specific fuel consumption (BSFC) increased and brake specific energy consumption (BSEC) decreased due to the low energy heating value and high viscosity of the CSO [3,4]. Increasing the CSO content of the blended fuel decreased the cetane number, which resulted in an increase of ignition delay times.

Another essential oil, pine oil, has lower boiling point and viscosity when compared to diesel, which is deemed to enhance the fuel atomization and its mixing with air [2]. However, due to its lower cetane number, pine oil was blended with diesel at 10%, 20%, 30%, 40%, and 50% blend ratios. Increasing pine oil content in the blend increased the peak heat release rate. This increase is due to longer ignition delays, attributable to lower cetane numbers. This leads to an accumulation of combustible components in the combustion chamber, which in turn results in an increased heat release rate. Lower bulk modulus and compressibility results in the late onset of combustion [5]. Furthermore, enhanced vaporization and improved combustion, due to the lower viscosity and boiling point, resulted in higher peak combustion pressures for pine oil blends. Lower viscosity results in more complete combustion by causing fine dispersion of fuel droplets. Pine oil also contains oxygen, which promotes combustion. Thus, BTE of pine oil blends were higher when compared to neat diesel [2].

Lemongrass oil has low heating value (36 MJ/kg) [6] and slightly low cetane number (45, 38) [6,7], thus needed to be blended with diesel in order to use in diesel engine. When compared to base diesel, lemongrass oil-diesel blends increased BTE and BSFC of the engine [8]. The increase of BTE was attributed to more complete combustion due to better vaporization, and increased BSFC was attributed to lower calorific value.

Orange oil is extracted from *Citrus sinensis*, which is native to China, but it is now cultivated extensively in Australia [9]. Blending orange oil with diesel significantly reduced the viscosity and density and increased the heating value [10]. However, it also resulted in low flash point and fire point temperatures. Due to having low viscosity and higher calorific value, fuel blends with orange oil typically exhibit high brake thermal efficiency [10]. Neat orange oil blend (20% with diesel) resulted in higher peak cylinder pressure, which was credited to higher flame velocity ensuring more complete combustion, supported by Poola et al. [11]. Furthermore, in premixed combustion phase a longer ignition delay time for neat orange oil resulted in more fuel burning, which increased the peak pressure and the maximum rate of pressure rise, supported by Huang et al. [12]. The higher heat release rate for neat orange oil during the premixed combustion phase was attributed to the presence of oxygen in the oil [8]. Neat orange oil also exhibited better evaporation, which, along with longer ignition delay time, increased the maximum heat release rate. Another study reported higher BTE and lower BSEC for neat orange oil due to the improved evaporation and mixing resulting in more complete combustion [13].

Eucalyptus trees are native to Australia and are cultivated worldwide [14]. Some authors have reported the reduction of BSFC and BSEC, and higher BTE, when eucalyptus blends were used in a diesel engine. These results were attributed to the high calorific value and low density of the blends, which led to improved atomization [15,16]. However, in another study, BTE for neat eucalyptus oil was found to be load dependent: lower at lower loads and higher at higher loads when compared to base diesel [17]. At higher loads, BTE of neat eucalyptus was slightly higher than neat diesel due to the high heat content, high volatility, and low viscosity of the blends. The low viscosity of eucalyptus oil resulted in proper mixing and vaporization, as well as improved spray formation and air entrainment. Eucalyptus oil is highly volatile, less viscous, and has higher oxygen content than diesel. As a result, it burns quickly and releases heat in a shorter duration, resulting in a higher combustion temperature. This results in higher exhaust gas temperature (EGT)—especially at higher loads. Longer ignition delay times were as well reported for eucalyptus oil [17].

The oil of the leaves of Melaleuca alternifolia is commonly known as tea tree oil. This species is native to the Southeast of Queensland and the adjoining Northeast coast of New South Wales in Australia [18]. This oil has an oxygen content of around 5%. The main constituents of tea tree oil are terpinen-4-ol, γ-terpinene, and α-terpinene (shown in Figure 1). When compared to diesel, tea tree oil has a higher density, lower viscosity, slightly lower flash point, lower calorific value, significantly lower cetane number, and significantly lower induction time [19].

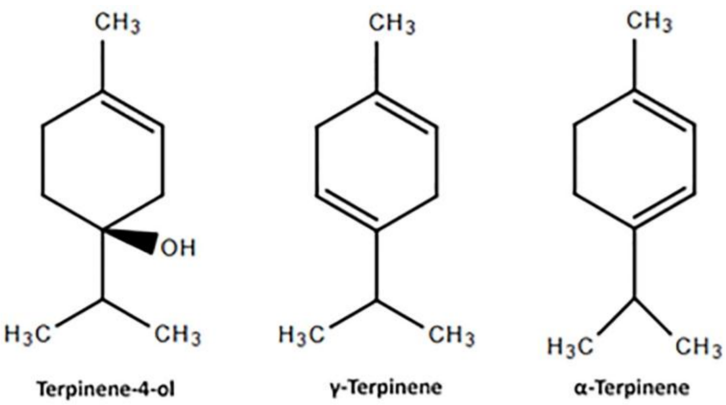


Figure 1. Major components of tea tree oil.

The essential oils tested in this study are: orange oil, eucalyptus oil, and tea tree oil, all of which are either native to or are being extensively cultivated in Australia. To date, no studies that investigate engine performance and combustion characteristics of a diesel engine operated with tea tree oil (neat/blended with diesel) have been published. The essential oils were blended with diesel at 10% blend ratio (by weight). For comparison, biodiesel produced from waste cooking oil was used in this study. Biodiesel was blended with diesel at 10% blend ratio (by weight). Finally, these blends were used to operate a multi cylinder diesel engine and engine performance and combustion parameters were recorded and evaluated.

2. Methodology

Essential oils selected for this study were: orange oil, eucalyptus oil, and tea tree oil. These oils, along with waste cooking biodiesel (WCB), were blended with diesel at 10% (by volume). The diesel fuel and WCB were supplied by Caltex (fuel certificate provided in Appendix A) and Ecotech Biodiesel (fuel certificate provided in Appendix B), respectively. The fuels used in the engine performance and combustion characteristics testing included: 100% diesel (100D), 10% orange oil blended with diesel (10O90D), 10% eucalyptus oil blended with diesel (10E90D), 10% tea tree oil blended with diesel (10T90D), and 10% WCB blended with diesel (10BD90D).

2.1. Fuel Property Analysis

Fuel properties analysis was carried out by Queensland University of Technology (QUT) and at the Engines and Energy Conversion Laboratory at Colorado State University (CSU). The details are provided in Rahman et al. (2017) [19]. The properties are listed in Tables 1 and 2.

Table 1. Properties of fuel blends used in the engine test measured at Queensland University of Technology (QUT) adopted from [19].

Property	Density@25 °C kg/m^3	Viscosity@40 °C m^2/s × 10^{-6}	Higher Heating Value kJ/g	Lower Heating Value [a] kJ/g	Surface Tension@25 °C mN/m
100D	837	3.12	45.92	43.95	26.09
10O90D	836	2.82	46.62	43.87	26.23
10E90D	839	2.9	45.10	42.35	27.30
10T90D	841	2.98	44.98	42.26	26.32
10BD90D	845	3.64	42.86	40.38	26.45

[a] derived.

Table 2. Properties of fuel blends used for Derived Cetane Number testing at Colorado State University (CSU) adopted from [19].

Blends	Density@15 °C kg/m^3	Viscosity@40 °C m^2/s × 10^{-6}	Flash Point °C	Oxidation Stability Induction Period [a] h	Heating Value KJ/g	Sound Speed m/s	Derived Cetane Number [b]
100D	853.11	2.75	60.7	14.22	45.585	1398	43.2
100O	849.12	0.89	55.0	0.30	45.115	1360	19.5
10O90D	852.64	2.37	57.5	7.88	45.375	1394	41.0
100E	913.99	1.66	55.0	0.63	40.648	1326	10.0
10E90D	858.66	2.60	57.5	18.53	45.144	1390	41.2
100T	900.31	1.84	53.5	0.22	41.899	1394	10.4
10T90D	857.01	2.53	56.5	0.04	44.982	1396	39.1

[a] EN14112; [b] ASTM D7170.

2.2. Engine Experiment Details

All of the experiments were conducted with a six-cylinder, turbocharged diesel engine. This engine was coupled with an electronically-controlled water brake dynamometer in order for the steady-state load to be controlled. A piezoelectric transducer (Kistler 6053CC60, Kistler, Winterthur, Switzerland) was used in order to collect in-cylinder data, along with a simultaneous analogue-to-digital converter (Data Translation DT9832). A crank angle encoder set (Kistler type 2614, Kistler, Winterthur, Switzerland) was used to collect the crank angle data, along with engine speed data. To eliminate the effect of the cycle-to-cycle variations and obtain an accurate in-cylinder pressure data, mean of 750 consecutive cycles was taken. Further information about the engine and experimental facilities is provided by Bodisco and Brown (2013) [20].

Running of the engine at 1500 RPM (i.e., max torque condition) was carried out at four different loads—25%, 50%, 75%, and 100%. At the beginning of each fuel cycle, the maximum load was therefore determined to be when the engine was at full throttle—1500 RPM (i.e., 100% load). After that, other loads were determined for that speed. The key engine specifications are given in Table 3, and a schematic diagram of the full engine test setup is provided in Figure 2.

Table 3. Test engine specification.

Model	Cummins ISBe220 31
Capacity	5.9 Litres
Maximum power	162 kW at 2000 RPM
Maximum torque	820 N.m at 1500 RPM
Number of cylinders	6
Number of valves per cylinder	4
Compression ratio	17.3:1
Bore × Stroke	102 × 120 (mm)
Dynamometer	Electronically-controlled water brake dynamometer
Injection system	Common-rail
Emission certification	Euro IIIA
Aspiration	Turbocharged

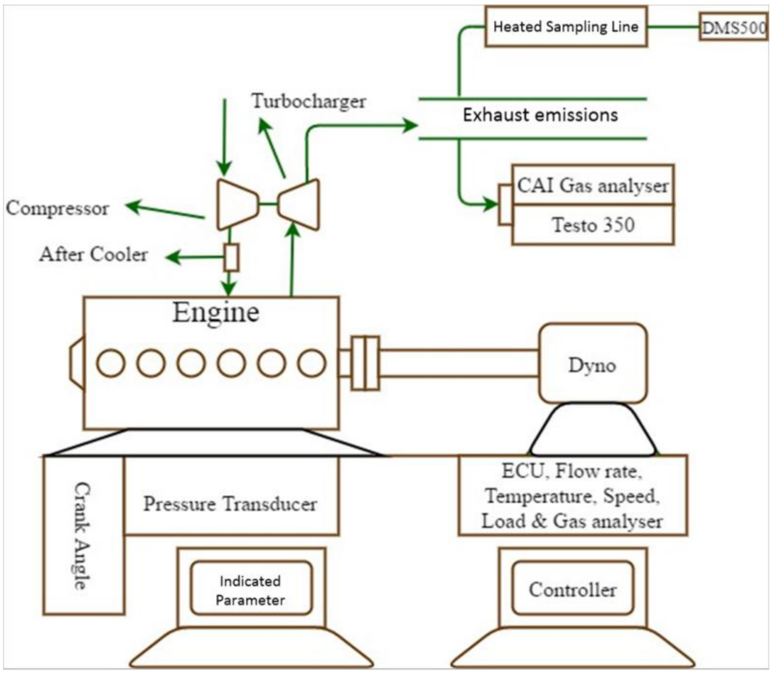


Figure 2. Schematic diagram of engine campaign setup.

The indicated power (IP) of an engine is the power produced by the combustion products on the piston in the cylinder. The brake power (BP) is the useful power available at the output shaft.

Brake thermal efficiency (BTE) is the measure of how effectively the engine is converting chemical energy into mechanical energy. The equation for BTE is as follows:

$$\text{BTE } (\%) = \frac{\text{BP} \times 100}{m_f \times \text{LHV}} \quad (1)$$

where, m_f = fuel flowrate (kg/s), LHV = lower heating value (MJ/kg), and BP = brake power (kW).

In contrast to BTE, BSFC is a measure of ratio of the fuel burned to produce a unit of power [21]. The equation for BSFC is:

$$\text{BSFC} \left(\frac{\text{g}}{\text{kWh}}\right) = \frac{3600 \times m_f \times 1000}{\text{BP}} \quad (2)$$

where, m_f = fuel flowrate (kg/s) and BP = brake power (kW).

Indicated thermal efficiency (ITE) and indicated specific fuel consumption (ISFC) are calculated using Equations (3) and (4).

$$\text{ITE}\ (\%) = \frac{\text{IP} \times 100}{m_f \times \text{LHV}} \tag{3}$$

$$\text{ISFC}\left(\frac{\text{g}}{\text{kWh}}\right) = \frac{3600 \times m_f \times 1000}{\text{IP}} \tag{4}$$

where, m_f = fuel flowrate (kg/s), LHV = lower heating value (MJ/kg), and IP = indicated power (kW).

Mechanical efficiency (ME) is the ratio between BP and IP. The equation is as follows:

$$\text{ME}\ (\%) = \frac{\text{BP}}{\text{IP}} \times 100 \tag{5}$$

where, IP = indicated power (kW) and BP = brake power (kW).

Friction mean effective pressure (FMEP) is an indicator of pressure loss to overcome friction. The equation is as follows:

$$\text{FMEP} = \text{IMEP} - \text{BMEP} \tag{6}$$

Engine blow-by is where pressure builds up and forces combustion gases to pass the piston rings and enter the crankcase as partly-combusted air-fuel. This phenomenon occurs mainly through the engine's power stroke, although it may possibly also take place during compression strokes [22]. This pressure is not utilised in rotating the crankshaft, and so, blow-by is treated as lost efficiency. Engine lubricants are also affected by this blow-by, due to partially un-combusted air-fuel mixture passing into the crankcase. This can cause a decrease in oil longevity and an increase in engine wear [23]. Adding soot to the lubricant will alter the oil's viscosity significantly, thus increasing the frictional losses in the engine. In turn, this leads to a further reduction of mechanical efficiency and increased fuel consumption [24]. The normalised blow-by equation is as follows:

$$\text{Blow-by}\left(\frac{\text{L}}{\text{kWh}}\right) = \frac{\text{Blow} - \text{by(Lpm)}}{\text{Power(kW)}} \times 60 \tag{7}$$

To study the combustion mechanism of the diesel engine, heat release rate (HRR) analysis was carried out using Equation (8), as derived by Heywood [25]. This analysis drew on the first law of thermodynamics, although heat lost through cylinder walls was not considered. Both the main combustion chamber and the pre-combustion chamber were combined to create a single-zone thermodynamic model. In between the two chambers, it is predicted that there will not be passage throttling losses. Fuel vaporisation, fuel mixing, gradients of temperature, and non-equilibrium conditions, along with pressure waves, can all be ignored. This method serves to simplify the identification of the SOC (start of combustion) timing, and variances in combustion rates from the diagram showing HRR versus crank angle. The HRR was calculated by using average in-cylinder data gleaned from 750 consecutive cycles.

$$\frac{dQ}{d\theta} = \frac{V\frac{dP}{d\theta} + \gamma P\frac{dV}{d\theta}}{\gamma - 1} \tag{8}$$

where, $\frac{dQ}{d\theta}$ = rate of heat release (J/°CA), V = instantaneous cylinder volume (m^3), θ = crank angle (°CA), P = instantaneous cylinder pressure (Pa), and γ = specific heat ratio. The input values are the pressure data and cylinder volume (with respect to crank angle).

3. Results and Discussion

Figure 3 represents the variation of BP and IP, respectively, of different fuels. Figure 4 represents FMEP of different fuels. It is evident that the difference between BP and IP is minimal (<3.0%) at higher loads. This indicates lower frictional loss at high loads when compared to low loads. At low loads, a difference between IP and BP of around 7 kW can be seen for comparable IMEP. This trend of falling frictional power with increasing IMEP can clearly be observed in Figure 4. From this figure, at 25% load the FMEP varies from 16.1% to 18.5% of IMEP and at 100% load, it varies from 0.4% to 2.2% of IMEP. The cetane number increase associated with biodiesel contributes to the earlier start of combustion when compared to diesel, which might lead to higher cylinder peak pressure and output power (see Figure 3) [26]. However, the cetane number of essential oil blends is quite low (Table 2) as compared to neat diesel and WCB. This results in lower BP for 10E90D (2.0% to 4.0% lower than neat diesel) and 10T90D (1.7% to 6.0% lower than neat diesel), which can be seen from Figure 3. However, orange oil has comparable calorific value to diesel, which results in similar BP (only 0.8% to 2.2% lower than neat diesel). From Figure 3, the maximum IP is achieved by 100D, followed closely by 10BD90D and 10O90D. At 100% load, IP of 10E90D and 10T90D is lower (4.0% and 6.0%, respectively) as compared to base diesel, which may be due to the lower LHV of the eucalyptus and tea tree oils.

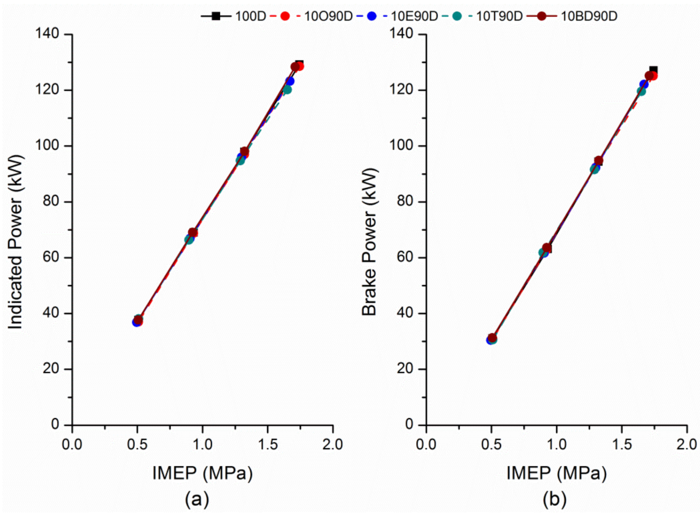


Figure 3. Power variation of tested fuel blends: (**a**) Indicated (**b**) Brake.

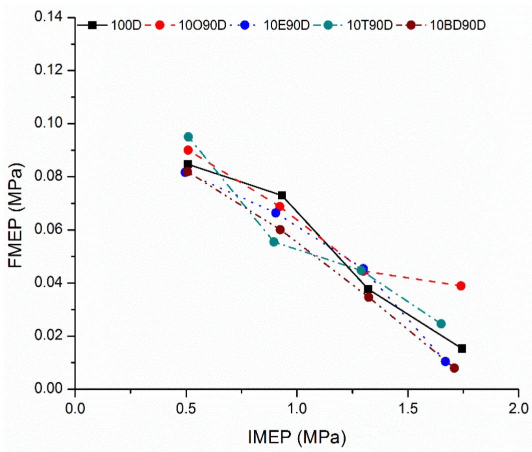
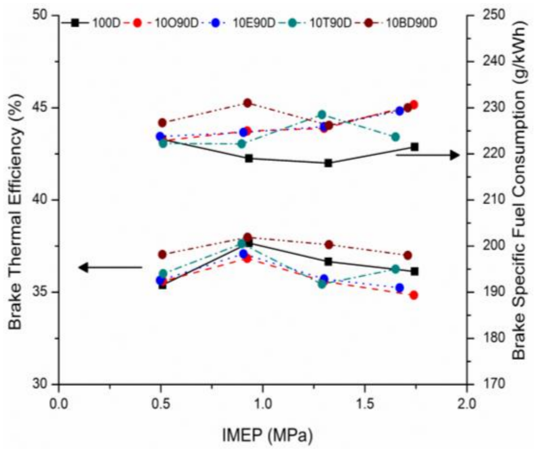


Figure 4. Frictional Mean Effective Pressure variation of tested fuel blends.

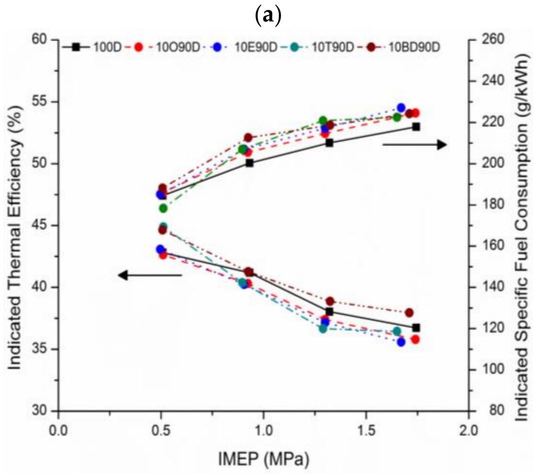
From Figure 5, all of the essential oil blends have lower brake thermal efficiencies when compared to base diesel. This may be due to the lower heating value of all essential oil blends as compared to the baseline diesel fuel [27,28]. However, BTE also depends on other factors such as cetane number, frictional loss, etc. As all of the essential oil blends have much lower cetane number when compared to base diesel, BTE is lower for these blends than for diesel. For all of the fuel blends, the BTE was the highest at 50% load. At this load, there is sufficient air available to promote combustion. At 25% load, higher frictional loss, overly lean mixture results in incomplete combustion. At 75% and 100% load, incomplete combustion occurs due to overly rich mixture and less time for combustion. Thus, at these loads, BTE for all of the fuels is lower.

As load increases, both the BP and fuel flow rate (m_f) increase. For all of the fuel blends, the increases in m_f and BP are almost proportional, which result in an insignificant change of BSFC with load variation. One of the properties that affect BSFC is cetane number. Increase in cetane number results in a reduction of ignition delay time, which in turn improves combustion and reduces fuel consumption [29]. All of the essential oils are inferior when compared to diesel for cetane value; hence, when essential oils are added to diesel, cetane value is reduced (as compared to base diesel) and BSFC increases [30].

From Figure 5, ITE decreases with increasing load for all fuel blends, which is consistent with other research [21,31]. ITE is the ratio of IP to fuel energy (the product of m_f and the fuel calorific value), and it decreases as the load increases. This is due to the rate of increase of IP being lower than the rate of increase for m_f. When compared to 100D, there was a reduction in ITE for all of the essential oil blends. At higher loads, BTE and ITE are similar, which is due to much lower friction loss at these loads compared to the lower loads. As load increased ISFC of all the blends increased. Compared to neat diesel, ISFC of all essential oil blends were higher. This could be due to essential oil blends lower heating value and higher surface tension and density [21].



(**a**)



(**b**)

Figure 5. Thermal efficiency and fuel consumption variation of tested fuel blends: (**a**) Brake; and, (**b**) Indicated.

With increasing load, frictional losses decrease and mechanical efficiencies increase (Figure 6). At full load, ME is highest for 10E90D and 10T90D and lowest for 10O90D. This means that the eucalyptus-diesel and tea tree-diesel blends convert most of the input energy to output energy.

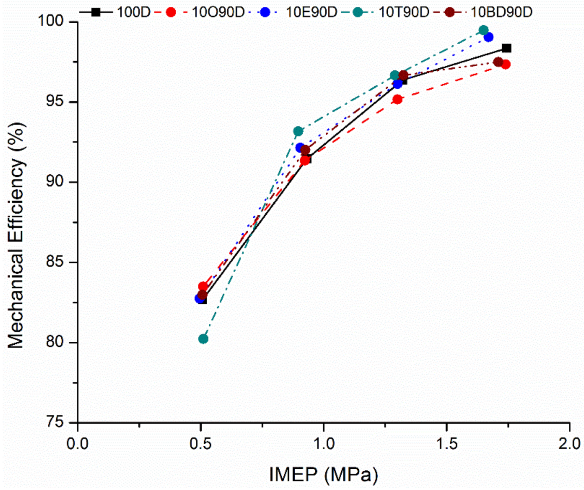


Figure 6. Mechanical Efficiency variation of tested fuel blends.

From Figure 7, maximum blow-by is observed for 100D and 10BD90D at 75% and 100% load. The essential oil-diesel blends reduce blow-by at these loads. Furthermore, with an increase in load, blow-by decreases for most of the fuel blends. These blow-by trends are quite similar to the FMEP variation seen in Figure 4.

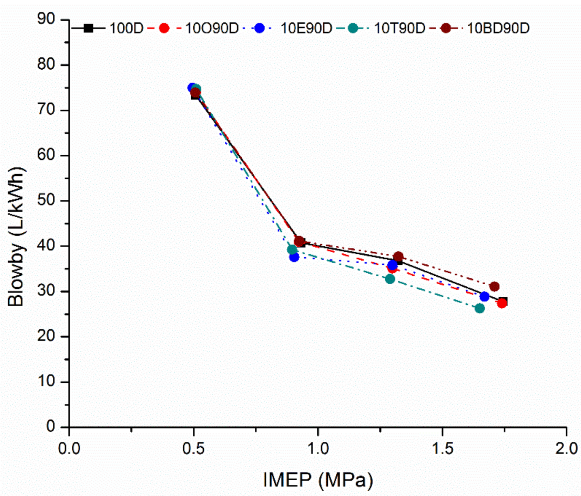
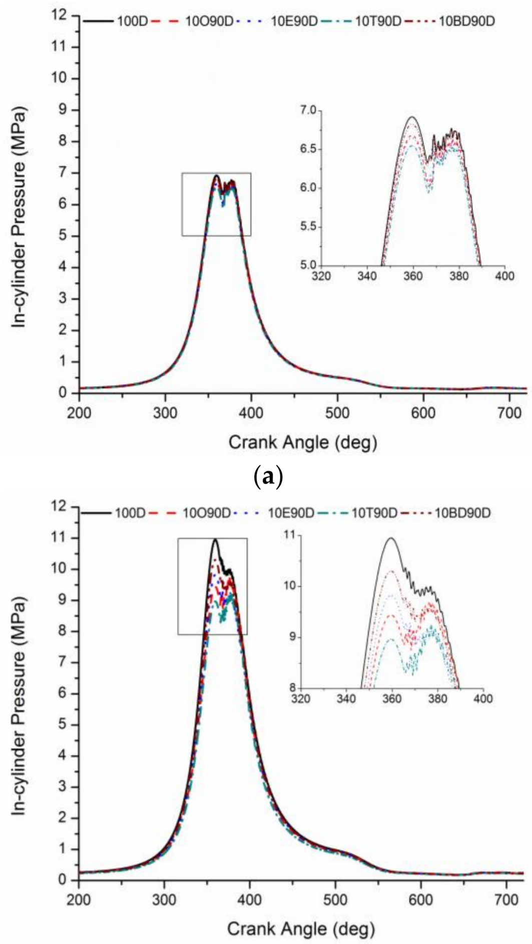


Figure 7. Blow-by variation of tested fuel blends.

From Figure 8, for 100% load, peak cylinder pressure was 10.95 MPa for reference diesel (100D), closely followed by 10.30 MPa for 10BD90D. For both of the loads, essential oil blends showed lower peak pressure when compared to base diesel (100D) and 10BD90D. A longer ignition delay time for the essential oil-diesel blends allowed for more air/fuel mixing, which is ready to auto-ignite and results in a lower premixed peak [32].

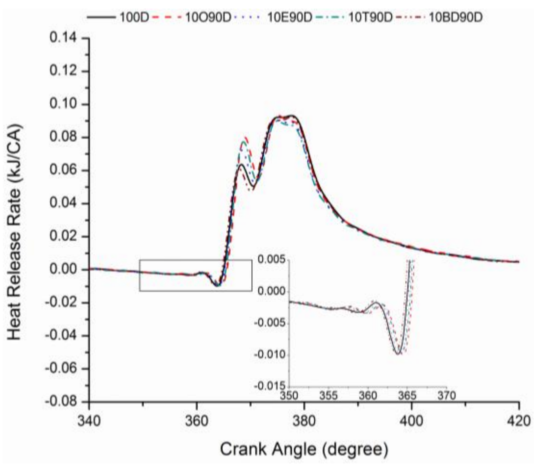
From Figure 9, it was quite evident that ignition starts later for all essential oil-diesel blends, which can be attributed to their lower cetane numbers and lower viscosities when compared to neat diesel. A longer ignition delay time will lead to the accumulation of combustible mixture inside the combustion chamber. Consequently, the peak heat release rate proved to be higher for essential oil blends when compared to neat diesel. Likewise, a number of researchers report a higher pre-mixed burning rate, and this is due to a longer ignition delay time, particularly when lower cetane fuels are used [2,33].



(**a**)

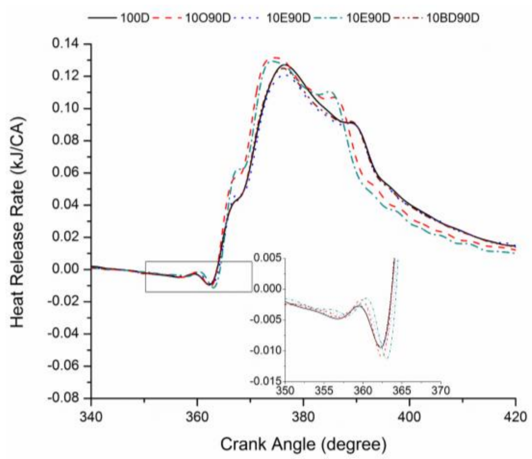
(**b**)

Figure 8. In-cylinder pressure variations of tested fuel blends @ (**a**) 50% and (**b**) 100% load.



(**a**)

Figure 9. *Cont.*



(**b**)

Figure 9. Heat Release Rate variations of tested fuel blends @ (**a**) 50% and (**b**) 100% load.

4. Conclusions

An inclusive investigation was performed to evaluate engine performance and combustion characteristics of a six-cylinder, 5.9 L, turbocharged diesel engine fuelled with orange, eucalyptus, and tea tree oil blends (10% blended with diesel). The results were compared to neat diesel (100D) and 10% waste cooking biodiesel-diesel blends. Engine test runs were conducted by using the selected fuels at four loads with constant engine speed (1500 RPM). Based on experimental observation, the following conclusions can be made:

Compared to the reference diesel:

- Most of the blends show higher mechanical efficiency.
- All of the blends indicate high BSFCs due to their low heating value.
- Thermal efficiencies with the blends were low. This was associated with the lower cetane number.
- All oil blends show low peak pressure due to low energy content.
- The blow-by emissions were lower with the blends.

No significant changes in engine performance were observed with the oil blends. When considering the fact of slightly low engine performance, further investigation is required by adding some cetane improver could improve the engine performance for the oil blends.

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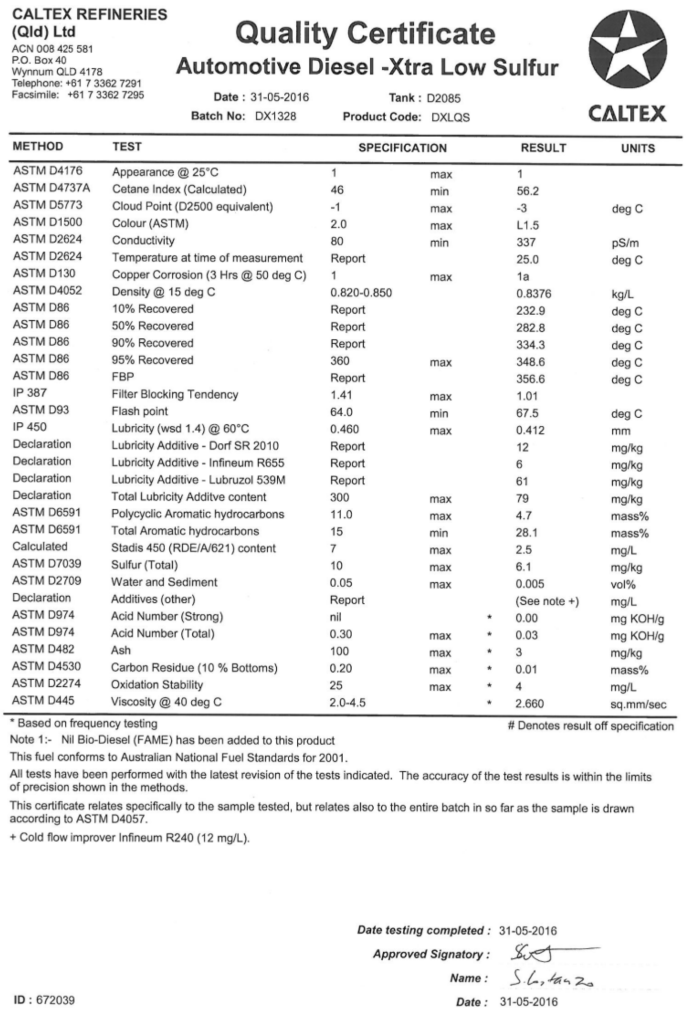
Acronyms

BMEP	Brake mean effective pressure
BP	Brake power
BSEC	Brake specific energy consumption

BSFC	Brake specific fuel consumption
BTE	Brake thermal efficiency
CSO	Clove stem oil
EGT	Exhaust gas temperature
FMEP	Friction mean effective pressure
HRR	Heat release rate
IMEP	Indicated mean effective pressure
IP	Indicated power
ISFC	Indicated specific fuel consumption
ITE	Indicated thermal efficiency
LHV	Lower heating value
ME	Mechanical efficiency
m_f	Fuel flow rate
RPM	Revolution per minute
SOC	Start of combustion
WCB	Waste cooking biodiesel

Appendix A

Quality certificate of Diesel (Figure A1):



CALTEX REFINERIES
(Qld) Ltd
ACN 008 425 581
P.O. Box 40
Wynnum QLD 4178
Telephone: +61 7 3362 7291
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Quality Certificate
Automotive Diesel -Xtra Low Sulfur

Date : 31-05-2016 Tank : D2085
Batch No: DX1328 Product Code: DXLQS

CALTEX

METHOD	TEST	SPECIFICATION			RESULT	UNITS
ASTM D4176	Appearance @ 25°C	1	max		1	
ASTM D4737A	Cetane Index (Calculated)	46	min		56.2	
ASTM D5773	Cloud Point (D2500 equivalent)	-1	max		-3	deg C
ASTM D1500	Colour (ASTM)	2.0	max		L1.5	
ASTM D2624	Conductivity	80	min		337	pS/m
ASTM D2624	Temperature at time of measurement	Report			25.0	deg C
ASTM D130	Copper Corrosion (3 Hrs @ 50 deg C)	1	max		1a	
ASTM D4052	Density @ 15 deg C	0.820-0.850			0.8376	kg/L
ASTM D86	10% Recovered	Report			232.9	deg C
ASTM D86	50% Recovered	Report			282.8	deg C
ASTM D86	90% Recovered	Report			334.3	deg C
ASTM D86	95% Recovered	360	max		348.6	deg C
ASTM D86	FBP	Report			356.6	deg C
IP 387	Filter Blocking Tendency	1.41	max		1.01	
ASTM D93	Flash point	64.0	min		67.5	deg C
IP 450	Lubricity (wsd 1.4) @ 60°C	0.460	max		0.412	mm
Declaration	Lubricity Additive - Dorf SR 2010	Report			12	mg/kg
Declaration	Lubricity Additive - Infineum R655	Report			6	mg/kg
Declaration	Lubricity Additive - Lubruzol 539M	Report			61	mg/kg
Declaration	Total Lubricity Additve content	300	max		79	mg/kg
ASTM D6591	Polycyclic Aromatic hydrocarbons	11.0	max		4.7	mass%
ASTM D6591	Total Aromatic hydrocarbons	15	min		28.1	mass%
Calculated	Stadis 450 (RDE/A/621) content	7	max		2.5	mg/L
ASTM D7039	Sulfur (Total)	10	max		6.1	mg/kg
ASTM D2709	Water and Sediment	0.05	max		0.005	vol%
Declaration	Additives (other)	Report			(See note +)	mg/L
ASTM D974	Acid Number (Strong)	nil		*	0.00	mg KOH/g
ASTM D974	Acid Number (Total)	0.30	max	*	0.03	mg KOH/g
ASTM D482	Ash	100	max	*	3	mg/kg
ASTM D4530	Carbon Residue (10 % Bottoms)	0.20	max	*	0.01	mass%
ASTM D2274	Oxidation Stability	25	max	*	4	mg/L
ASTM D445	Viscosity @ 40 deg C	2.0-4.5		*	2.660	sq.mm/sec

* Based on frequency testing # Denotes result off specification
Note 1:- Nil Bio-Diesel (FAME) has been added to this product
This fuel conforms to Australian National Fuel Standards for 2001.
All tests have been performed with the latest revision of the tests indicated. The accuracy of the test results is within the limits of precision shown in the methods.
This certificate relates specifically to the sample tested, but relates also to the entire batch in so far as the sample is drawn according to ASTM D4057.
+ Cold flow improver Infineum R240 (12 mg/L).

Date testing completed : 31-05-2016
Approved Signatory :
Name : S. Lo, tanzo
ID : 672039
Date : 31-05-2016

Figure A1. Quality certificate of Diesel.

Appendix B

Quality certificate of Biodiesel (Figure A2):

SpecCheck Laboratories

SPECCHECK LABORATORIES P/l
P.O.Box 835 Mittagong NSW 2575
Unit 4, 13 Lyell St
Mittagong NSW 2575
Phone/Fax: 4872 4590
Email: lab@speccheckIabs.com.au

REPORT NUMBER:	16E31053	DATE:	01-Jun-16
CUSTOMER NAME:	ECOTECH BIODIESEL	CONTACT:	Paul Hetherington
CUSTOMER CODE:	EB01	ADDRESS:	181 Potassium St
			Narangba Qld 4504
Ph: 07 3204 0467	Fax: 07 3204 0497	Email:	paulh@ecotechbiodiesel.com

SAMPLE ID:	Batch 180
DATE SAMPLED:	27-May-16
DATE RECEIVED:	30-May-16

TEST	RESULT	UNITS	METHOD	SPECIFICTION
Total contamination	2.0	mg/kg	EN12662	24 max
Free glycerol	0.009	%	ASTM D6584	0.020 max
Total glycerol	0.139	%	ASTM D6584	0.250 max
Monoglycerides	0.317	%	ASTM D6584	0.80 max
Ester content	96.9	%	EN14103	96.5 min
Viscosity @40C	4.730	mm^2/s	ASTM D445	3.5 – 5.0
Filter blocking tendency (B100)	1.05	-	ASTM D2068	2 max
Cold soak filterability	196	sec	ASTM D7501	360 max
Density	0.8804	kg/l	ASTM D1298	0.86 - 0.89
Moisture content	426	ppm	ASTM D6304	500
Total Acid Number	0.13	mgKOH/g	ASTM D664	0.80 max
Flash point	>130	^{o}C	ASTM D93	120 min
Alcohol content	0.02	%	EN14110	0.20max
Oxidation stability	>10	h	EN14112	10 min
Copper Corrosion	1A	-	ASTM D130	1 max
Carbon Residue (10% res)	0.117	%	ASTM D4350	0.3 max
Cold filter plugging point	6	^{o}C	ASTM D6371	-
Sulphated Ash	0.00	%	ASTM D874	0.02 max
Distillation temp @90% rec	310	^{o}C	ASTM D1160	360 max

Signed: S.Brennan

NATA
WORLD RECOGNISED ACCREDITATION

NATA Accredited Laboratory
Number: 14871
Accredited for compliance with ISO/IEC 17025

Sample tested as received

Figure A2. Quality certificate of Biodiesel.

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