A Comparative Analysis of Engine's Combustion and Performance of an OEM I.D.I. Diesel engine fuelled with diesohol and diesel fuel

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ABSTRACT

The use of ethanol blend with diesel fuel in a compression ignition engine has some potential on exhaust black smoke reduction. However, lower energy content of ethanol and lower cetane rating of the blend tend to increase ignition delay and reduce engine efficiency. It is believed that the engine combustion process plays an important role in performance improvement. The combustion process can be improved only after it has been properly understood. In this study, this investigation had the object, to get more knowledge on the characteristics of an OEM high-speed Light Duty IDI Diesel engine operating with diesohol. Comparative analysis of engine performance and fuel consumption characteristics, such as torque and power output, fuel and energy consumption, operating pressures and temperatures, combustion chamber pressures between engine fuelled with diesohol and diesel are reported in this paper. Measurements of the cylinder pressures were taken in the engine operating points according to ECE R49 (13 mode test), all other measurements were carried out in the whole engine operating range. In particular, the experimental investigation has performed a careful analysis of heat release, which has made it possible to give more precise comparative information about the combustion process.

1. Introduction

Nowadays, there are many of attempts using alternative fuels in CI engine. Some alternative fuels cause poor exhaust emissions and some require engine modification that usually is expensive. Ethanol has been used to fuel engines since the birth of the auto industry. Henry Ford powered one of his first cars, the quadricycle, with ethanol. He believed that a renewable fuel would better serve the needs of automobile drivers of the future and the local economy [1]. Some past studies have considered numerous methods of introducing ethanol into compression ignition (CI) engines, some works focused on the development of blends of diesel fuel and ethanol [2,3].

Due to the increasing of diesel fuel price, the blend between diesel and ethanol (is called "diesohol") has been considered by the Thai government as one of a candidate alternative fuel for diesel substitution. The use of ethanol blend with diesel fuel in a compression ignition engine has some potential on exhaust black smoke reduction. However, lower energy content of ethanol with higher heat of vaporization and lower cetane rating of the blend tend to increase ignition delay and reduce engine efficiency. It is believed that the engine combustion process plays an important role in performance improvement. In order to carry out a proper use of this fuel in CI engine, it is necessary to understand the effects of this fuel on engine combustion process.

To obtain a proper knowledge of diesohol combustion phenomena in an IDI combustion chambers, the interactions between different phenomena (e.g. turbulent flow, spray, combustion) and naturally the geometry of the combustion chamber must be taken into account. 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 together with a direct optical observation. This allows a precise investigation of the problem as it provides all variables at all points of the geometry.[4,5,6,7,8,9,10]

The outcome presented in this paper is aimed to compare indicating measurement of the engine's performance and combustion of an indirect injection diesel engine fuelled with diesohol (10% ethanol, 89% Diesel fuel and 1% additive) and diesel fuel.

2 Diesohol

The ethanol (anhydrous alcohol 99.5%) 10% by volume was blended with 89% reference diesel and 1% additive for this study. The emulsifier was added to this diesohol to prevent phase separation. The specification of test fuels is shown in table 1.

Table 1 Fuel properties for standard diesel and diesohol.

Properties	Unit	Test Method	Reference Diesel	Diesohol
Cetane number		ASTM D613	57.8	49.7
Cetane Index		ASTM D976	54.8	53.0
Distillation		ASTM D86		
IBP	^{o}C		174.4	-
10 % recovered	^{o}C		216.5	-
50 % recovered	^{o}C		285.6	-
90 % recovered	^{o}C		351.6	335.2
End point	^{o}C		373.4	-
Specific Gravity @ 15.6/15.6 ^{o}C		ASTM D1298	0.8378	0.8233
API gravity @ 15.6/15.6 ^{o}C		ASTM D4052	37.4	-
Viscosity @ 40 ^{o}C	CST	ASTM D445	3.227	2.574
Lubricity by HFRR	μm	CEC F-06-A-96	545	426
Pour point	^{o}C	ASTM D97	-3	-6
Cloud point	^{o}C	ASTM D2599	3.6	-
Oxidation stability	mg/100 ml.	ASTM D2274	0.63	-
Sulfer content	%wt.	ASTM D4294	0.042	-
Flash point	^{o}C	ASTM D93	71	12
Copper strip corrosion number	Number	ASTM D130	1a	1a
carbon residue	%wt.	ASTM D4530	< 0.001	0.001
Ash, %wt.		ASTM D482	< 0.001	0.001
Total Acid Number		ASTM D974	0.02	-
Water content	%wt.	ASTM D4928	0.0074	-
Lower heating value	J/g		45,920	44,202

Many properties of the diesohol fuel can be attributed directly to the dilution effect of the ethanol on the diesel fuel. The lower density and lower viscosity of ethanol compared to diesel fuel resulted in slight reductions of these properties in the resulting diesohol blends. Similarly, ethanol has a lower energy content than diesel fuel and the resulting diesohol blends have roughly 5% less energy per volume than diesel fuel. Ethanol does not contain aromatics, thus the diesohol blend contains roughly 10 % less aromatics simply by dilution. The diesohol shows the reductions in T_{90} point that may effect the poor long-trip economy. While ethanol has very high-octane value for spark ignition engines, it has correspondingly poor cetane value for compression ignition engines. It is widely known that the addition of ethanol to diesel fuel will degrade the cetane number of the resulting diesohol blend. The flash point of diesohol is controlled by low flash point of the ethanol. The flashpoint of diesohol is lower than that of diesel fuel and lower than the minimum flash point accepted by the Thailand diesel fuel specification. The lower flash point of the diesohol affects the shipping and storage classification. The precautions should be used in handling and transporting the fuel. [11]

3. Analysis of Cylinder Pressure Data [12,13,14]

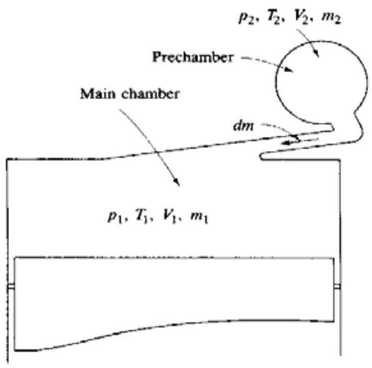

Figure 1 Schematic defining variables in main chamber (subscript 1) and prechamber (subscript 2)

Cylinder pressure versus crank angle data over the compression and expansion strokes of the engine operating cycle can be used to obtain quantitative information on the progress of combustion. The rate the fuel's heat release, or rate of fuel burning, through the diesel engine combustion process can be described by the methods of quasi static (i.e., uniform in pressure and temperature) analysis start with the first law of thermodynamics for an open system. The first law for such a system (see Figure 1) is

$$\frac{dQ}{dt} - p\frac{dV}{dt} + \sum_i \dot{m}_i h_i = \frac{dU}{dt} \qquad (1)$$

Where dQ/dt is the heat-transfer rate across the system boundary into the system, P dV/dt is the rate of work transfer done by the system due to system boundary displacement, $\dot{m}_i$ is the mass flow rate into the system across the system boundary at location i, h_i is the enthalpy of flux i entering the system, U is the energy of the material contained inside the system boundary.

In IDI diesel engines, the pressures in each of the two chambers, main and auxiliary, are not the same during the combustion process. Since combustion starts in the prechamber, the fuel energy release in the prechamber causes the pressure there to rise above the main chamber pressure. Depending on combustion chamber design and operating conditions, the prechamber pressure rises to be 0.5 to 5 atm. above that in the main chamber. This pressure difference causes a flow of fuel, air, and burned gases into the main chamber, where additional energy release now occurs.

In practice, it is difficult to get experimental data for both the main and auxiliary chamber pressures throughout the combustion process. Access for two pressure transducers through the cylinder head is not often available even when access can be achieved, the task of obtaining pressure data from two different transducers under the demanding thermal loading conditions found in IDI diesels, of sufficient accuracy such that the difference between the pressures (of order 0.5 to 5 atm.) at pressure levels of 60 to 80 atm. can be interpreted, requires extreme diligence in technique.

Assuming $p_2 = p_1$, and using either main chamber or auxiliary chamber pressure alone. The error associated with this approximation can be estimated as follows. If we write $p_2 = p_1 + \Delta p$ then the net heat release rate is

$$\frac{dQ}{d\theta} = \frac{\gamma}{\gamma - 1} p_1 \frac{dV_1}{d\theta} + \frac{V_1 + V_2}{\gamma - 1}\frac{dp_1}{d\theta} + \frac{V_2}{\gamma - 1}\frac{d(\Delta p)}{d\theta} \qquad (2)$$

Where γ is specific heat ratio. If the last term is omitted, Eq.(2) is identical derived for the DI diesel engine.

$$\frac{dQ}{dt} = \frac{\gamma}{\gamma - 1} p \frac{dV}{d\theta} + \frac{V}{\gamma - 1}\frac{dp}{d\theta} \qquad (3)$$

The net heat release is calculated from integrate dQ/dt.

$$Q = \int \frac{dQ}{d\theta} d\theta \qquad (4)$$

And mass fraction burned, which relate with heat release give information about how much fuel was burned at any point of the combustion cycle.

4. Experimental Setup

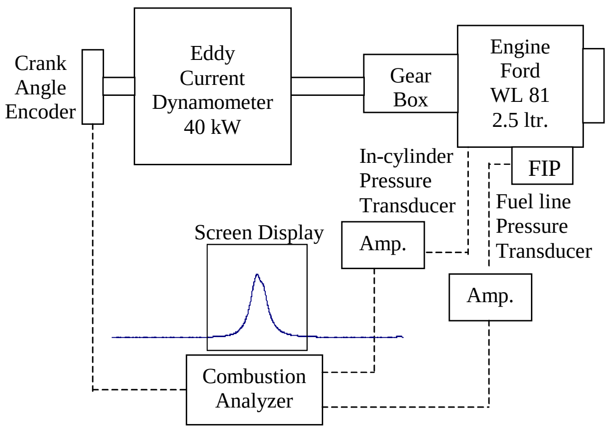

Figure 2 schematic arrangement of measuring system

This experiment was performed with an OEM high-speed light duty IDI diesel engine. The engine specification is shown below:

Engine model	WL81
Engine type	In-line, water cooled, four cylinders
Induction system	Natural aspirated
Bore	93 mm.
Stroke	92 mm.
Displacement	2.499 liters
Combustion system	Pre-chamber
Compression ratio	21.6:1

The engine was connected to MEIDEN EC-80 eddy current dynamometer.

The pressure data was measured by AVL GU12P piezoelectric pressure transducer that was installed in a glow plug adapter in the swirl chamber of the forth cylinder. The fuel line's pressure was measured by KISTLER 607C1 pressure transducer. The engine crank angle is measured by the Cussons P4503 shaft encoder. The Cussons P4500 (Autoscan) was employed for data acquisition. These indicating data were collected at every 0.1 degree crank angle for 7 executive cycles. The schematic arrangement of experimental set up is shown in figure 2.

The experiment of performance is carried out at constant speed, steady state conditions to compare the full load performance, fuel consumption and etc. For the combustion analysis, measurements of in-cylinder pressures were taken at each engine steady state operating points according to ECE R49 (13 mode test) and full load. Speed, torque, fuel consumption, operating pressure and temperature for both fuels were recorded during each test.

5. Results and Discussion

5.1 Performance Comparison

The corrected full load brake torque and power of engine fuelled with reference diesel and diesohol were compared. The comparison of maximum brake power and torque of engine with both fuels is shown in figure 3. The engine maximum brake torque for both fuels occurred at the same engine speed of 2250 rev/min. The results show that the maximum power and brake torque at each speed of the engine fuelled with diesohol is lower than fuelled with reference diesel. The progressive worse at lower engine speed were observed.

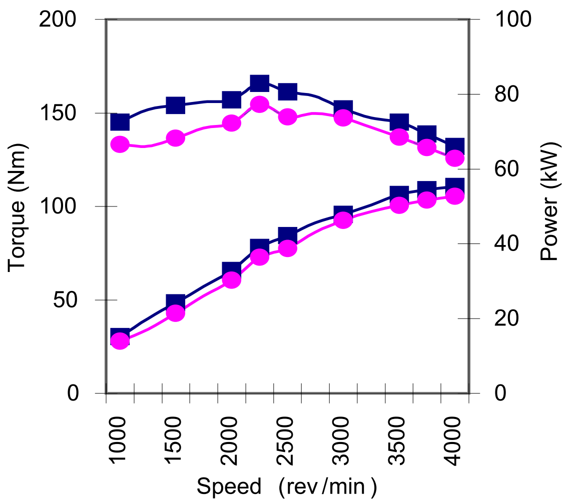

Figure 3 Maximum brake torque and power of engine.

Diesel and diesohol

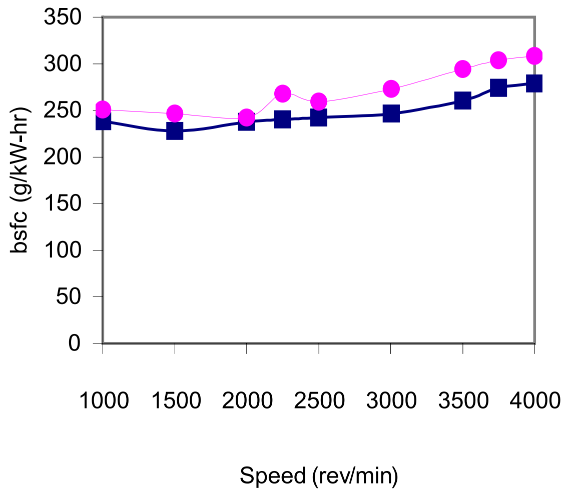

Figure 4 Brake specific fuel consumption at full load.

diesel and diesohol

Figure 4 shows the trend of brake specific fuel consumption at full load for both fuels. Brake specific fuel consumption of diesohol engine is higher than diesel engine and the difference are progressive worse at higher engine speed.

The fuel consumption of diesohol engine is higher than diesel engine because of its lower volumetric energy. The brake specific fuel consumption of engine with diesohol is higher than fuelled with diesel in a range between 2% to 19%, depends on engine's operating condition. The comparative results of break specific consumption (bsfc) at each steady state operating point according to ECCR49 and full load are shown in Table 2.

5.2 Combustion Analysis

The combustion analyses results at selected operating points of reference diesel and diesohol are summarized in Figure 5 (a) to (f). The in-cylinder pressure achieved using diesel and diesohol is shown in Figure 5(a). It can be seen that diesel has higher combustion pressure than diesohol. Figure 5(b) and (c) show that diesohol has approximately 1° of injection timing delay compared with diesel. The injection timing delay is probably due to the lower isentropic bulk modulus and lower viscosity of ethanol compared to diesel resulting in slightly reductions of these properties in the resulting blends [15].

Due to the late injection timing, the higher heat of vaporization of ethanol and the lower cetane, the start of combustion of diesohol tends to retard from diesel combustion, as shown in Figure 5(d). Thus, the lower combustion pressure, compared with diesel, is obtained. However, the late diesohol injection occurred during the higher ambient pressure and temperature, difference in ignition delay between diesel and

Table 2 bsfc comparison of diesel and diesohol.

Speed (rev/min)	Torque (Nm)	Diesel bsfc (g/kW.h)	Diesohol bsfc (g/kW.h)	Different of bsfc	% of Different
1000	20	3612.7	3770.7	-158.1	-4.4
1000	30	480.5	506.8	-26.2	-5.5
1000	145[1]/133[2]	375.7	408.7	-33.0	-8.8
1500	154[1]/137[2]	238.3	250.8	-12.4	-5.2
2000	30	228.0	246.5	-18.4	-8.1
2000	50	392.7	399.4	-6.7	-1.7
2000	157[1]/144[2]	300.5	312.9	-12.4	-4.1
2250	10	237.5	242.3	-4.7	-2.0
2250	20	897.5	971.0	-73.5	-8.2
2250	40	506.5	566.6	-60.1	-11.9
2250	61	335.8	371.7	-35.9	-10.7
2250	80	275.7	310.7	-34.9	-12.7
2250	130	253.8	284.7	-30.9	-12.2
2250	166[1]/155[2]	230.7	260.0	-29.3	-12.7
2500	40	240.4	267.8	-27.5	-11.4
2500	90	348.5	359.2	-10.7	-3.1
2500	161[1]/148[2]	252.8	271.4	-18.6	-7.3
2750	20	242.4	259.3	-16.8	-6.9
2750	40	553.5	578.2	-24.7	-4.5
2750	70	353.2	373.5	-20.3	-5.7
2750	111	274.0	300.9	-26.9	-9.8
3000	82	239.8	260.2	-20.4	-8.5
3000	120	254.8	303.2	-48.3	-19.0
3000	152[1]/147[2]	233.9	274.3	-40.3	-17.2
3500	90	246.4	273.1	-26.7	-10.8
3500	145[1]/137[2]	266.9	302.9	-36.0	-13.5
3750	100	260.7	294.5	-33.8	-13.0
3750	139[1]/132[2]	268.9	298.5	-29.7	-11.0
4000	132[1]/126[2]	274.0	303.9	-29.8	-10.9

diesohol could not significantly be observed. It can be seen from Figure 5(e) and (f) that diesohol has better late combustion phase compared to diesel. This may come from the benefit of oxygen content of ethanol in the diesohol. Without any engine modification, therefore, combustion duration of diesohol is only slightly longer than diesel.

It was found that the maximum in-cylinder pressure increases with engine load and speed as shown in figure 6(a) and (b) respectively. The maximum in-cylinder pressure of diesel combustion is slightly higher than diesohol combustion because the diesohol has either lower heating value or higher heat losses due to the higher heat of vaporization of ethanol in the blend.

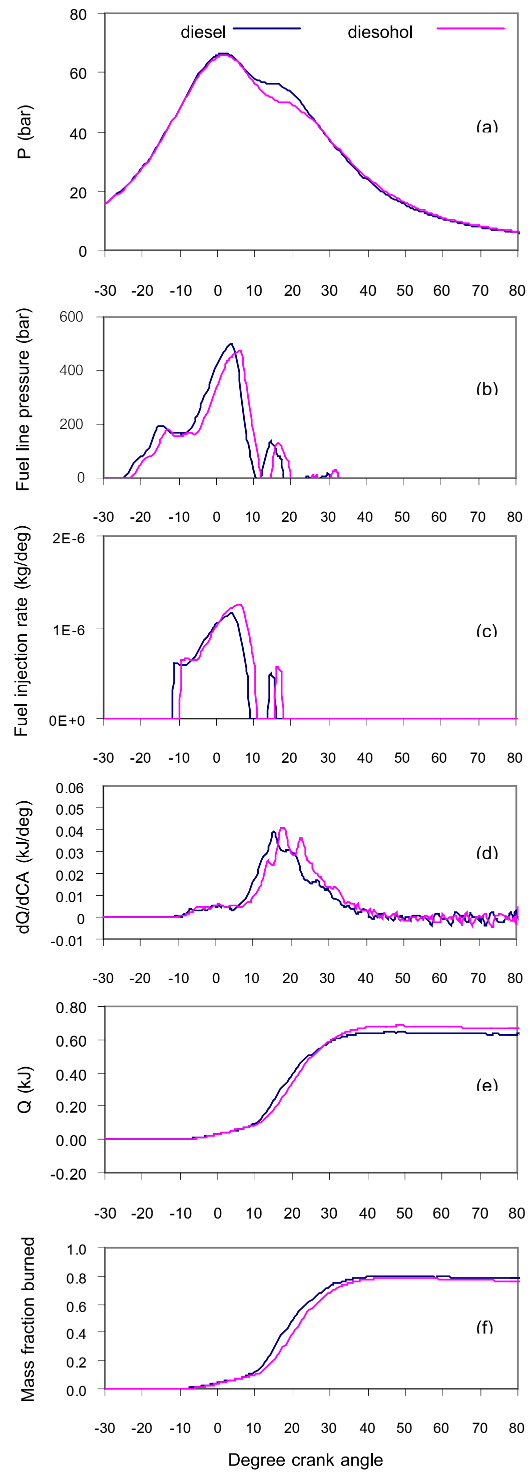

Figure 5 (a) In cylinder Pressure data, (b) Fuel line pressure, (c) Fuel injection rate (d) Heat release rate, (e) Net heat release, and (f) Mass fraction burned, at 2250 rev/min, 80 Nm.

The lower in a peak of net heat release was found with diesohol during the full load test, except at 4000 rev/min. These may due to the combination effects of the lower cetane number as well as the higher heat of vaporization. It was also found that the starting point of diesohol combustion tends to have some slightly delay from the point of diesel combustion.

Net heat release from engine combustion increases with increasing engine speed, as shown in figure 7 (a). Diesel engine has greater net heat release than diesohol engine except at 4000 rev/min.

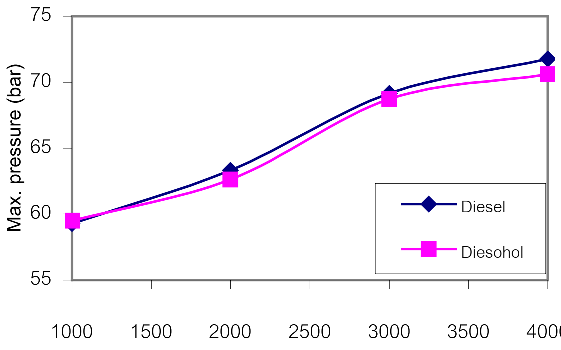

Figure 6 (a) Peak pressure at full load

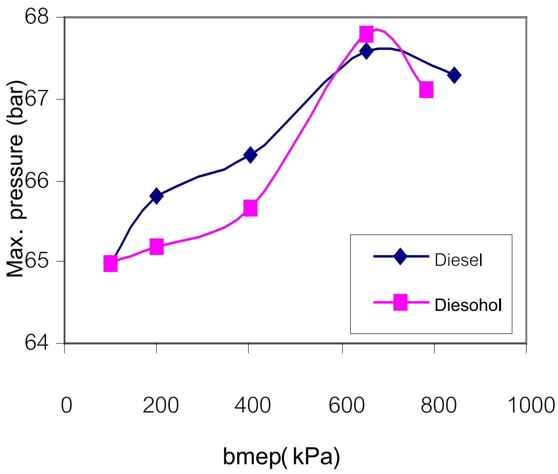

Figure 6 (b) Peak pressure at 2250 rev/min.

The constant speed of 2250 rev/min that contains a number of engine loads along the ECE R49 operating points is selected as the engine part-load representation. As shown in figure 7 (b), the net heat release of both fuels tends to increase with increasing load. Due to the higher heating value of diesel, the higher net heat release of the diesel fuel than of diesohol at the same test point are expected. However, this tend is compensated by the higher fuel injection rate per cycle of diesohol than the diesel engine.

From the indicating information for all experimental points, the similar trend can also be observed. The comparative results of injection and combustion characteristics between reference diesel and diesohol are shown in table 3.

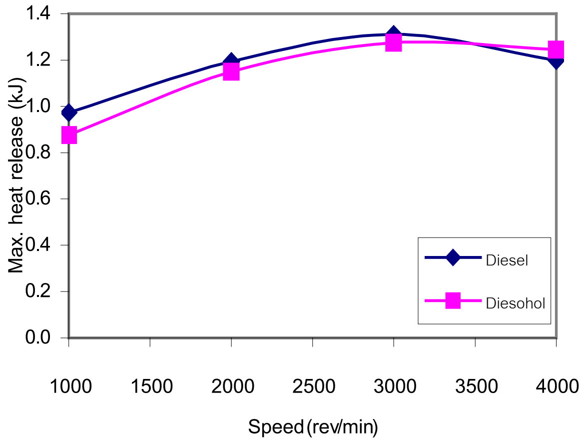

Figure 7 (a) Maximum heat release at full load.

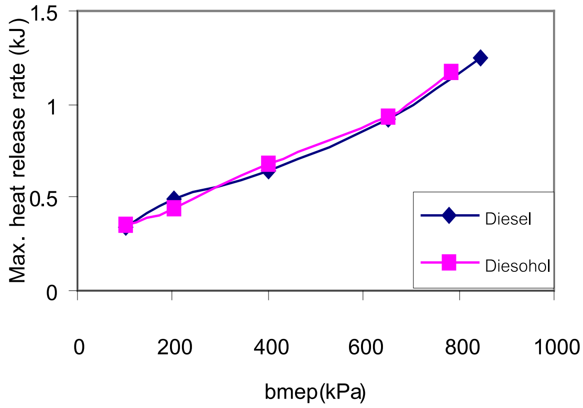

Figure 7 (b) Maximum heat release at 2250 rev/min.

6. Conclusion

The results from these studies show that diesohol with 10 % ethanol by volume can be used in an IDI engine with some power drop. The lower heating value compare with diesel, the diesohol engine full load power and torque are lower and the difference has shown progressive worse at lower speed. The Brake specific fuel consumption of diesohol engine is higher than diesel engine and the difference has shown progressive worse at higher speed.

In-cylinder pressure, fuel line pressure, and crank angle were measured. Then, fuel injection rate, heat release rate, net heat release and mass fraction burn were calculated. Results of engine fuelled with reference diesel and diesohol were compared. The injection timing of diesohol is approximately 1 degree delay compared with the injection timing of reference diesel. The injection timing delay is probably due to the lower isentropic bulk modulus and lower viscosity of ethanol compared to diesel resulting in slightly reductions of these properties in the diesohol. The in-cylinder combustion pressure of diesohol combustion is lower than diesel combustion because of combination effects from (a) the delay of injection timing of diesohol, (b) higher heat

Table 3 Injection timing, ignition delay and burn duration (ms) at selected test conditions for reference diesel and diesohol

Speed (rev/min)	Torque (N.m)	Injection Timing (^{o}CA ATDC)		Ignition Delay (ms)		Burn Duration (S.O.C to 95% Max. Net heat release, ms)	
		Diesel	Diesohol	Diesel	Diesohol	Diesel	Diesohol
750	Idle	-4	-4	2.08	1.78	3.72	4.00
1000	20	-11.5	-11.0	3.25	3.33	2.33	2.25
2000	30	-11.0	-9.5	1.5	1.25	1.42	1.79
2250	10	-9.0	-8.0	0.22	0.26	2.26	2.11
2250	20	-9.5	-8.5	0.30	0.22	2.33	2.37
2500	40	-11.0	-10.0	0.20	0.27	2.37	2.50
2750	20	-9.0	-8.5	0.15	0.15	2.00	2.12
2750	40	-10.5	-9.0	0.15	0.15	2.36	2.42

losses due to the higher heat of vaporization of ethanol in the blend and (c) the lower cetane of diesohol. From heat release rate diagram, it was found that the starting point of diesohol combustion tends to have some slightly delay from the point of diesel combustion. The combustion for both fuels tends to start faster with increasing speed. The net heat release of both fuels tends to increase with increasing load. As a result of its higher fuel injection rate at the same load and speed as well as its high heat of vaporization, diesohol combustion duration tends to have a slightly longer period than diesel. Nevertheless, as engine speed is increased diesohol combustion has similar ignition delay to diesel. This may due to the benefit of oxygen content in the fuel.

However, the optimised use of diesohol as an alternative fuel in a diesel engine requires some further investigations. These include either improvement of diesohol properties, such as ignition quality, etc., or engine calibration, such as injection timing optimization, etc.

7 Acknowledgments

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