18th Australasian Fluid Mechanics Conference
Launceston, Australia
3-7 December 2012

Performance Improvement of a Compression Ignition Engine by Ethanol and Diesel Dual-Fuelling

S. Padala[1], C. Woo[1], S. Kook[1], E.R. Hawkes[1,2]

[1]School of Mechanical and Manufacturing Engineering
The University of New South Wales, Sydney, NSW 2052, Australia

[2]School of Photovoltaic and Renewable Energy Engineering
The University of New South Wales, Sydney, NSW 2052, Australia

Abstract

Renewable feed stock and higher octane rating make ethanol as a promising alternative fuel. In contrast to the conventional approach of applying ethanol-petrol blends in spark-ignition engines, this study investigates the potential of ethanol fuelling in a compression ignition engine to achieve higher efficiency. Experiments are performed using a single-cylinder version of a common-rail diesel engine that is widely used in passenger cars. A dual-fuelling technology is implemented such that ethanol is introduced into the intake manifold using a port fuel injector while diesel is injected directly into the cylinder. The main focus is the effect of ratio of ethanol fuel energy to the total energy on engine efficiency and nitrogen oxides emissions at fixed total energy of ethanol and diesel. From the measurements, it is found that the increased ethanol energy fraction can increase the engine efficiency up to the point where the operation is limited by misfiring. The results are compared to the diesel-only operation with varying injection timings, which can explain whether the increased efficiency is due to the combustion phasing or improved combustion associated with fast burning of ethanol. Detailed analysis of the results revealed that the latter was the primary cause for the efficiency gain.

Introduction

Ethanol blends in gasoline are widely used in spark ignition (SI) engines. However, the current approach, *i.e.* ethanol blended with gasoline and burned in conventional SI engines is not optimal with the following reasons: First, ethanol has a much higher resistance to knock, and hence could be burned in higher compression ratio engines resulting in higher efficiency; Second, to avoid phase separation, the ethanol must not contain water [5], which costs a lot more to produce than ethanol containing small amounts of water; Third, ethanol has a high heat of vaporisation and a low vapour pressure compared with gasoline, leading to poor vaporisation and hence ignition problems especially during cold start [4].

These issues can be resolved by using the ethanol in a dual-fuelling diesel engine where ethanol is delivered in the intake manifold and diesel is directly injected into the cylinder using two separate fuel injection systems [6, 7, 11, 12]. In this engine, a premixed ethanol-air mixture is ignited by in-cylinder injection of diesel and therefore combustion phasing is controlled by the diesel injection timing. The higher compression ratio of a diesel engine can effectively take advantage of the high octane number of ethanol with suppressed knocking. Since separate fuel supply systems are used, water in ethanol would not be an issue. The cold start problem can also be addressed by using diesel-only combustion during the engine warm-up. In addition, ethanol is an oxygenated fuel that can help reduce harmful soot emissions [3].

In a dual fuelling scenario, the thermodynamic efficiency of the engine can either be increased or decreased relative to a diesel-only baseline. Assuming relatively good combustion efficiency and roughly the same heat loss in both scenarios, this mainly depends on the timing and duration of the combustion event. Timing is controllable by injection but duration depends on the prevailing combustion mode and may be shorter or longer than the diesel-only combustion duration.

To investigate the potential of ethanol dual fuelling and understand the details of efficiency variations when the ethanol energy fraction increases, we performed dual-fuelling experiments in a single-cylinder automotive-size diesel engine. Ethanol was supplied into the intake port using a conventional port-fuel injector (PFI) and the diesel direct-injection was conducted using a common-rail injection system. In-cylinder pressure and engine-out nitrogen oxide (NO_x) emissions were measured for various ethanol energy fractions at a fixed total energy of ethanol and diesel per engine cycle. In-cylinder pressure traces were further analysed to determine the heat release rates, combustion phasing, and burn duration that help explain the causes of observed efficiency trends.

Experiments

The schematic of the experimental setup is shown in figure 1 and the engine specifications are listed in table 1. Experiments were conducted in a single-cylinder automotive-size, naturally aspirated research engine. The engine is coupled to an eddy current dynamometer (Froude Hoffmann, AG-30HS) capable of a constant-speed operation. Since a single-cylinder engine was used, pressure fluctuations in the intake and exhaust pipes were identified as a potential issue. To minimise them, large-volume

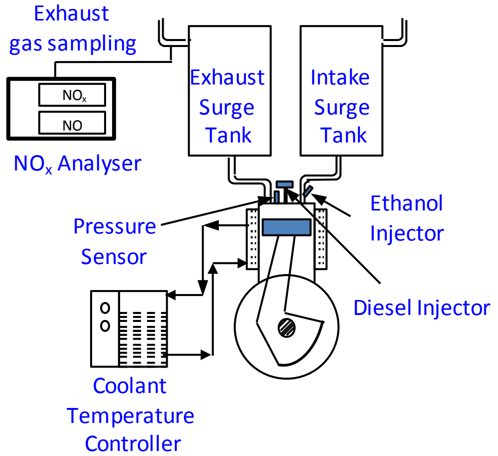

Figure 1. Schematic diagram of experimental facility

(60 times the displacement volume of the engine) surge tanks were placed in both intake and exhaust sides. A water circulator and temperature controller was used to fix the temperature of the engine head and cylinder liner at 90°C. In-cylinder pressure measurements were carried out using a piezo-electric pressure transducer (Kistler 6056A). The exhaust gases were sampled downstream of the exhaust surge tank for a chemiluminescence-type NO_x analyser (Ecotech, 9841 AS).

All experiments were conducted at fixed conditions as listed in table 1. The engine was started in diesel-only mode during the engine warm-up period. Then the PFI injector was turned on to inject the ethanol into the intake port. The total energy supplied to the engine is kept constant at 1098 J by reducing the diesel injection mass. This fuel energy corresponds to a brake mean effective pressure of around 450 kPa, namely, a medium load condition of the used engine.

Ethanol energy fraction E_x was calculated by the following equation:

$$\text{Ethanol energy fraction } E_x = \frac{\dot{m}_{Ethanol} * CV_{Ethanol}}{\dot{m}_{Ethanol} * CV_{Ethanol} + \dot{m}_{Diesel} * CV_{Diesel}} \quad .....(1)$$

where $\dot{m}$ is a flow rate, CV stands for the calorific value (heat of combustion). In the present work, up to 50% ethanol (E_x) was tested.

Results and Discussions

In-cylinder pressure traces and apparent heat release rates for various ethanol energy fractions are plotted against crank angle degrees (°CA) as shown in figure 2. The diesel start of injection (SOI) was set at -3°CA after top dead centre (aTDC). In-cylinder pressure was measured for 100 injection (firing) cycles and ensemble-averaged for each ethanol energy fraction. Figure 2 shows that an increase in the ethanol fraction leads to a decrease in the in-cylinder pressures at TDC. This happens because of the evaporation cooling of the ethanol [13]. Also, higher ethanol fraction leads to a decreased specific heat ratio (γ) than that of lower or zero ethanol fractions. Both cause decreased TDC pressure and temperature. It is clearly seen from the results that the ignition delay period (*i.e.* time between the diesel start of injection and start of pressure rise due to combustion) increases with increasing ethanol fraction because of these cooling effects. Using these measured in-cylinder pressure traces, the apparent heat release rates are calculated following the first law of thermodynamics for a closed volume system as shown at the bottom of figure 2. An expected trend is observed that the peak apparent heat release rate increases with increasing ethanol fraction. In addition to the increased ignition delay due to the cooling effects, the ambient gas entraining into the diesel jet is ethanol-air mixture and therefore oxygen concentration would be lower than the compressed air. This ambient gas dilution (lower oxygen concentration) further extends the ignition delay [9] resulting in increased pre-combustion mixing and thereby higher peak apparent heat release rate. An exception was for E_{50} where the longest ignition delay is measured but the peak apparent heat release rate appears to be the lowest. It is noticed that the combustion phasing is over-retarded for E_{50} leading to misfiring condition as the combustion occurs very late in the expansion stroke. This misfiring limits the maximum ethanol fraction for a given engine operating condition.

How this combustion translates to the power output is quantified by the indicated mean effective pressure (IMEP) plot as shown in figure 3 (top). The figure shows that IMEP increases with increasing ethanol fraction of up to 40% and then decreases suddenly when the misfiring occurs at 50% ethanol fraction. The

Engine Hardware	
Displacement	497.8 cc
Bore	83 mm
Stroke	92 mm
Compression ratio	17.7 : 1
Number of valves	2 intake and 2 exhaust
Piston	Cylindrical bowl
Diesel injector (In-cylinder direct injection)	Bosch common-rail 7 holes (0.134 mm in diameter) HFR 400 cc/30s, K1.5/86 Included angle 150° 130 MPa injection
Ethanol injector (Port fuel injection)	Bosch EV6, 6 holes (261 g/min @ 3 bar) 300 kPa injection at -359 °CA aTDC
Operating Conditions	
Engine speed	2000 rpm
Engine IMEP	~900 kPa
Engine BMEP	~450 kPa
Intake air temperature	~ 28° C
Total energy per cycle	1098 J
Ethanol energy fraction	0 ~50%

Table 1. Engine specifications and operating conditions

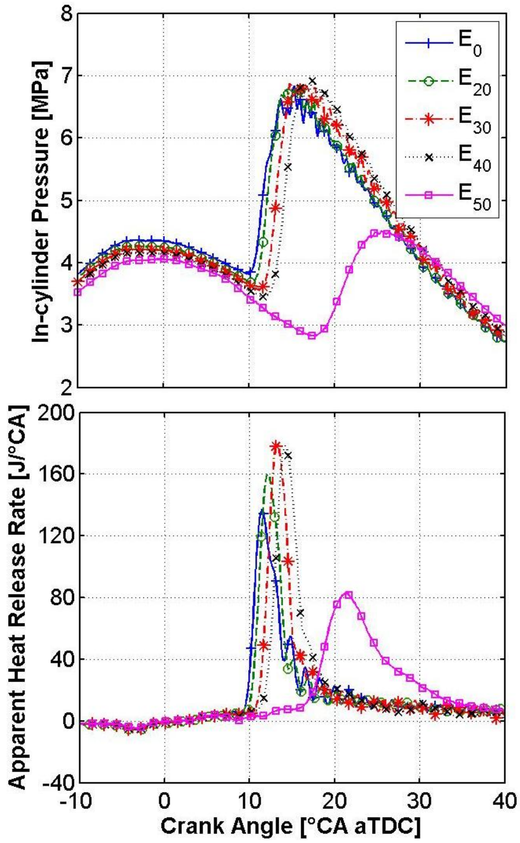

Figure 2. In-cylinder pressure (top) and apparent heat release rate (bottom) for different ethanol energy fractions (diesel injection at -3° CA aTDC)

coefficient of variation (CoV) of IMEP further confirms the misfiring. The engine appears to be running stable at up to 40% ethanol fraction with less than 5% of CoV of IMEP [14]. For E_{50} condition, an unacceptable CoV of IMEP of 17% was measured. Also shown in the figure 3 is brake thermal efficiency calculated from measured brake torque and fuelling rate. Brake thermal efficiency is increasing with increase in ethanol fraction except for misfiring cases. This is expected as the engine is running at constant speed which expects to have constant frictional power.

An interesting finding from figure 3 is that ethanol and diesel dual-fuelling can achieve higher IMEP than that of the diesel-only operation of the same fuel energy and thereby higher efficiency. One might simply think that increasing apparent heat release rate seen in figure 2 explains this efficiency gain. However, it is well known that IMEP depends not only on how much of heat energy is released from the combustion but also at what timing in respect to the crank angle position the heat energy is released [2]. Higher heat release rate would certainly increase the power output but if the combustion occurs during the later part of expansion stroke, the resulting power might not be as high as lower heat release rate occurs near TDC. This timing factor is termed as a combustion phasing. Since figure 2 shows an

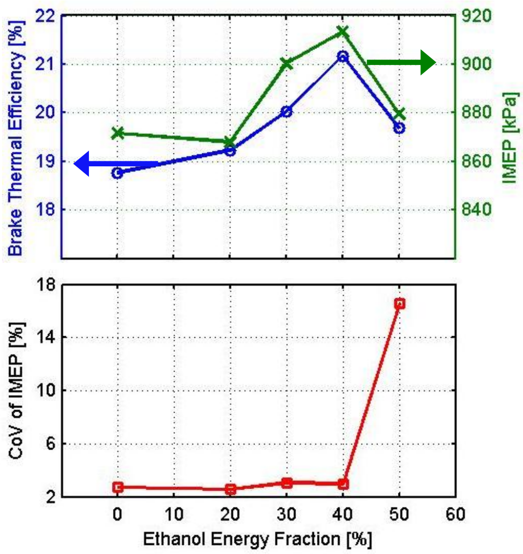

Figure 3. Indicated mean effective pressure (IMEP) and coefficient of variation of IMEP for different ethanol energy fractions (diesel injection at -3° CA aTDC)

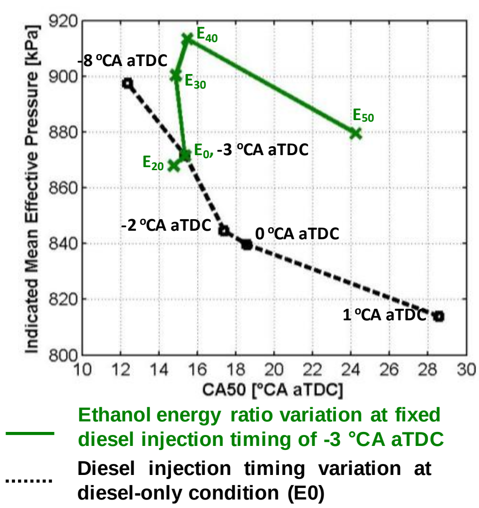

Figure 4. Indicated mean effective pressure (IMEP) against CA50 for dual-fuelling and diesel only operation

increasing trend of the apparent heat release rate but retarding combustion phasing, it is unclear whether the increased IMEP was a result of the increased heat release rate or properly positioned combustion phasing.

To address this question, we performed diesel-only operation (E_0) with various injection timings that can simulate the combustion phasings of the dual-fuelling cases. The combustion phasing is best characterised by measuring a crank angle position of 50% of total heat release (*i.e.* CA50) [8]. Therefore, we calculated CA50 for both dual-fuelling and diesel-only operations as shown in figure 4. In the figure, the injection timings and ethanol energy fractions are annotated for each operating condition. The results suggest that the combustion phasing was not a cause for the efficiency gain of the ethanol and diesel dual-fuelling. For example, the dual-fuelling delivers higher IMEP than diesel-only operation for the same combustion phasing of 15°CA aTDC. In fact, the combustion phasing does not vary much for the dual-fuelling cases except the misfiring E_{50}. This analysis suggests that replacing a fraction of diesel fuel with ethanol can increase the amount of heat release, compared to the diesel-only operation of the same combustion phasing.

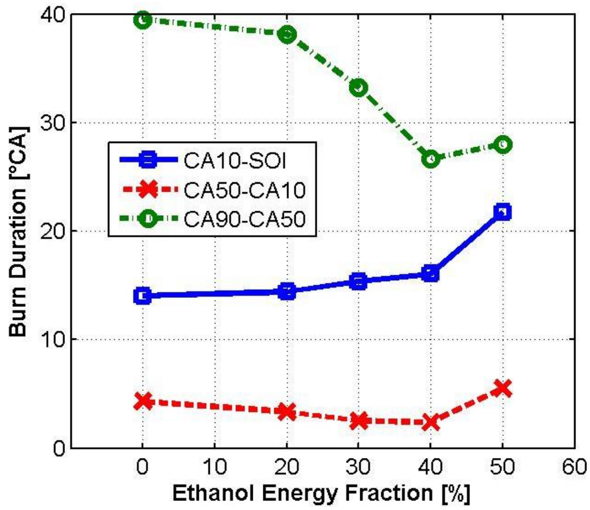

Figure 5. Burn duration for various stages of combustion: ignition delay (CA10-SOI), initial burning (CA50-CA10), late cycle burning (CA90-CA50) at different ethanol energy fractions (diesel injection at -3° CA aTDC)

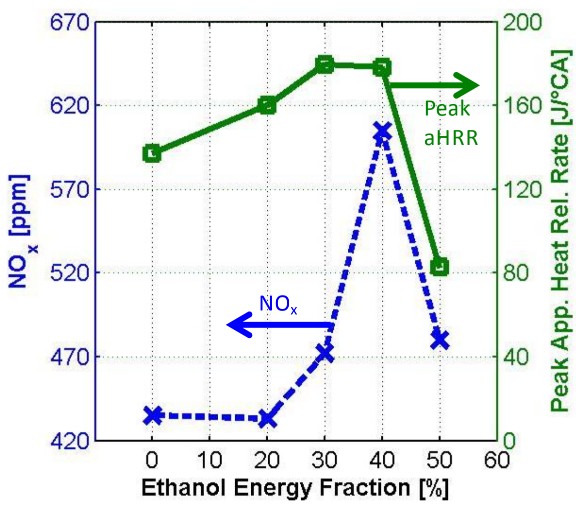

Figure. 6. NO_x emission and peak apparent heat release rate for various ethanol energy fractions (diesel injection at -3° CA aTDC)

Another parameter that can help understand the efficiency gain from the dual-fuelling combustion is burn duration. Typically, faster burning can increase the amount of heat release for the same fuel energy so that higher IMEP is measured [1, 10]. Figure 5 shows various burn durations calculated from heat release rates of all ethanol energy fractions tested in this study. The CA10-SOI corresponds to the ignition delay that increases with increasing ethanol fraction as was discussed previously. Of high interest is the CA50-CA10, the initial burn duration where the heat release is driven mostly by the combustion of pre-mixed charge of ethanol and air as well as diesel. It is noticeable that this premixed burn duration decreases with increasing ethanol fraction, inversely proportional to the ignition delay trend. The late-cycle combustion (*i.e.* CA90-CA50) is also consistent with the premixed burn duration, showing the same decreasing trend with increasing ethanol fraction. This analysis suggests that premixing of ethanol and air via port fuel injection in the intake manifold can have a great potential to increase the power and efficiency of a diesel engine.

This efficiency gain, however, does not come without a price. Figure 6 shows how the ethanol fraction affects the exhaust NO_x emissions, a strictly regulated gas due to its negative impact on environments and human health. In the figure, the peak apparent heat release rate is also plotted as there is a well-known correspondence between the NO_x and peak apparent heat release rate. This is because nitrogen oxide formation in an engine is primarily governed by thermal Zel'dovich mechanism and generally the peak apparent heat release rate is proportional to the flame temperature. Figure 6 shows that with increasing ethanol fraction, the peak apparent heat release rate increases and as a result, the NO_x emissions increases. As expected, misfiring E_{50} case shows a low peak apparent heat release rate and thereby low NO_x emissions. The results suggest that the NO_x emissions must be closely monitored while the dual-fuelling of ethanol and diesel is implemented.

Conclusions

Dual-fuelling of ethanol port injection and diesel in-cylinder direct injection was carried out in a single-cylinder automotive-size compression-ignition engine. The effect of ethanol energy fraction on the engine efficiency was discussed. The major findings from this study are:

- With increasing ethanol supply via port injection, the engine power output (IMEP) and efficiency increase until they are limited by over-retarded combustion phasing and thereby misfiring.
- The efficiency gain from the dual-fuelling of ethanol and diesel is not a result of the combustion phasing but increased rate of heat release, *i.e.* better burning of ethanol than diesel.
- Detailed analysis of burn duration for various combustion stages suggest that the increased heat release is likely caused by fast burning of the premixed ethanol-air-diesel mixture as well as fast late-cycle burning of the remaining charge.
- Care should be taken to make a use of the efficiency gain of the ethanol and diesel dual-fuelling because NO_x emissions might increase with increasing ethanol energy fraction.

Acknowledgements

Support for this research was provided by Australian Research Council under grant number LP110100595.

References

[1] Aceves, S.M., Flowers, D.L., Martinez-Frias, J., Ray, J.S., Dibble, R., Au, M. & Girard, J., HCCI Combustion: Analysis and Experiments, *SAE Technical Paper* 2001-01-2077, 2001.

[2] Bittle, J., Zheng, J., Xue, X. & Jacobs, T., Cylinder-To-Cylinder Variation Sources in Diesel Low Temperature Combustion and the Influence They Have on Emissions, *Spring Technical Meeting of the Central States Section of the Combustion Institute*, Dayton, Ohio, USA, April 22-24, 2012.

[3] Chen, H., Shi-Jin, S. & Jian-Xin, W., Study on Combustion Characteristics and PM Emission of Diesel Engines Using Ester-Ethanol-Diesel Blended Fuels, *Proc. Combust. Inst.*,**31**, 2007, 2981-2989.

[4] Chen, R.H., Chiang, L.B., Chen, C.N. & Lin, T.H., Cold-Start Emissions of an SI Engine Using Ethanol Gasoline Blended Fuel, *Appl. Therm. Eng.*,**31**, 2011, 1463-1467.

[5] Demirbas, A. Conversion of Biomass using Glycerin to Liquid Fuel for Blending Gasoline as Alternative Engine Fuel, *Energ. Convers. Manage.*,**41**, 2000, 1741-1748.

[6] Ekholm, K., Karlsson, M., Tunestål, P., Johansson, R., Johansson, B. & Strandh, P., Ethanol-Diesel Fumigation in a Multi-Cylinder Engine, *SAE Technical Paper* 2008-01-0033.

[7] Ogawa, H., Setiapraja, H. & Nakamura, T., Improvements to Premixed Diesel Combustion with Ignition Inhibitor Effects of Premixed Ethanol by Intake Port Injection, *SAE Technical Paper* 2010-01-0866, 2010.

[8] Olsson, J., Tunestål, P. & Johansson, B., Closed-Loop Control of an HCCI Engine, *SAE Technical Paper* 2001-01-1031, 2011.

[9] Pickett, L.M., Siebers, D.L. & Cherian, I., Relationship Between Ignition Processes and the Lift-Off Length of Diesel Fuel Jets, *SAE Technical Paper* 2005-01-3843, 2005.

[10] Senthil, K.M., Ramesh, A. & Nagalingam, B., An Experimental Comparison of Methods to use Methanol and Jatropha Oil in a Compression Ignition Engine, *Biomass. Bioenerg.*,**25**, 2003, 309-318.

[11] Situ, R., Brown, R.J., Zoran. R., Uli, K. & Douglas, H.J., Analysis of Dual Fuel Compression Ignition (Diesel) Engine, *The Seventh International Conference on Modeling and Diagnostics for Advanced Engine Systems*, Sapporo, Japan, July 28-31, 2008.

[12] Splitter, D., Hanson, R., Kokjohn, S. & Reitz, R., Reactivity Controlled Compression Ignition (RCCI) Heavy-Duty Engine Operation at Mid and High-loads with Conventional and Alternative Fuels, *SAE Technical Paper* 2011-01-0363, 2011.

[13] Tsang, K.S., Zhang, Z.H., Cheung, C.S. & Chan, T.L., Reducing Emissions of a Diesel Engine Using Fumigation Ethanol and a Diesel Oxidation Catalyst, *Energ. Fuel*,**24**, 2010, 6156-6165.

[14] Weall, A., Szybist, J.P. & Edwards, K.D., HCCI Load Expansion Opportunities Using a Fully Variable HVA Research Engine to Guide Development of a Production Intent Cam-Based VVA Engine: The Low Load Limit, *SAE Technical Paper* 2012-01-1134, 2012.