Eco-diesel engine fuelled with rapeseed oil methyl ester and ethanol. Part 1: efficiency and emission

A Kowalewicz
Technical University of Radom, ul. Chrobrego 45, 26-600, Radom, Poland. email: kowala@kiux.man.radom.pl

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Abstract: A novel concept of fuelling a diesel engine with biofuels – rapeseed oil methyl ester (RME) and ethanol was proposed, developed, and investigated. The idea of the concept depends on standard fuelling with RME and additional injection of ethanol into the inlet port during the inlet stroke. After ignition from burning RME droplets, the ethanol–air mixture burns very quickly and promotes combustion of the main fuel droplets. This results in a shorter combustion period. Experiments were carried out with the use of a one-cylinder direct injection diesel engine adapted to ethanol injection at three speeds and two loads: low and high and for three injection timings. At each experimental point the proportion of ethanol to RME was changed from zero to the value at which diesel knock occurred in such a way that the engine load was kept constant. This showed the influence of the ethanol energy to total fuel energy ratio ($\Omega_{E}$) on efficiency and emissions ($\Omega_{E}$ was a main independent variable parameter). A considerable decrease in CO$_2$ and the smoke level was obtained and, for high loads also CO and HC emissions. At low loads the NO$_x$ emissions were reduced. The optimum ratio of ethanol energy to total fuel energy, in terms of brake fuel conversion efficiency and benefit of emission decrease, was found to be about 25 per cent.

Keywords: rapeseed acid ethyl ester (RME), ethanol, emission, engine efficiency

1 INTRODUCTION

For the last two decades, a worldwide trend towards the application of biofuels in internal combustion engines has been observed. These fuels are mainly vegetable oils and alcohols. However, vegetable plant oils have very high viscosity in comparison with that of diesel fuel and therefore they are not applied as fuels. They are base materials for fabrication of fatty acid methyl esters (FAME), which have a much lower viscosity than crude vegetable oils, compared with that of diesel fuel [1], and/or added to diesel fuel as fractions.

As far as alcohols are concerned, methanol was first tested as a fuel for spark ignition (SI) engines and even for compression ignition (CI) engines [2, 3]. However, due to its corrosive features and non-miscibility with gasoline in the presence of water, as well as high costs of production, the work was moved to ethanol, which can be produced from biomass. The production of ethanol is cheaper than for alcohols.

Rapeseed oil methyl ester (RME) and ethanol are renewable biofuels that have been used for some time in Western Europe [4] and South America (Brazil). They will be used in Poland as components in diesel fuel and gasoline, respectively. The motivation for use of both these fuels is to decrease the greenhouse effect and the share of petroleum fuels on the fuel market.

In this work a novel concept of fuelling a diesel engine with both of these fuels, i.e. RME and ethanol, is presented. The concept, which was invented, developed, and investigated at Politechnika Radomska (Technical University of Radom), depends on dual fuelling: RME and ethanol are introduced separately. Base fuel, RME (instead of diesel fuel) was provided by a standard direct injection system of the engine and ethanol, an additional fuel, was injected during suction stroke to the inlet port with the use of a petrol injector, typical for SI engines. After injection, ethanol quickly evaporated in the cylinder and, together with air, formed an homogeneous gaseous mixture, which burned quickly after ignition from burning RME droplets, which had ignited earlier. RME has a shorter ignition delay than diesel fuel, but burns longer [5, 6]. The role of ethanol was to promote burning of RME droplets resulting in shorter
smokeless combustion. This concept of fuelling differs from fuelling the CI engine with an ethanol-diesel fuel mixture applied by the Scania Company [7], because the influence of ethanol on the combustion of the base fuel is higher, when ethanol forms a gaseous homogeneous mixture with air, but not with liquid fuel.

The application of RME and ethanol, which are biofuels, considerably decreases greenhouse gas emissions in comparison with fuelling with fossil fuels; the net production of CO2 is equal to zero (neglecting CO2 production during agricultural processes). Earlier experiments carried out by the present author and co-worker [8, 9] concerning fuelling with ethanol and diesel fuel showed that the combustion period of both these fuels was shorter, emissions of CO2 and smoke much lower, and combustion efficiency higher in comparison with DF fuelling (i.e. base diesel engine).

The main objectives of this paper are as follows:

(a) to investigate whether ethanol has any effect on combustion in a CI engine fuelled with RME as a base fuel;
(b) to measure emissions and efficiency as a function of the ratio of ethanol energy to total fuel energy for different engine operating conditions;
(c) to determine the optimum ratio of ethanol to RME from the point of view of emissions and engine efficiency.

This paper is to be supplemented by a paper in which combustion processes will be considered: the influence of the addition of ethanol on pressure in the cylinder, the rate of heat release, and the fraction of fuel burnt will be analysed. Also a paper detailing a comparison of the emissions and efficiency of the same modified engine, but fuelled with diesel fuel and RME as base fuels, has been submitted for publication.

2 ENGINE TEST STAND

For the experiment a test stand shown in Fig. 1 was prepared. The main part of it is a one-cylinder 1HC102 direct injection, naturally aspirated diesel engine. Engine data are shown in Table 1. Engine torque was measured by means of eddy-current dynamometer Vibrometer 3WB15. RME fuel consumption was measured with the use of an automatic fuel injection pump was replaced by another one, giving higher fuel delivery.

Table 1 Engine data

<table>
<thead>
<tr>
<th>Type of the engine</th>
<th>1HC102 (Polish production)</th>
</tr>
</thead>
<tbody>
<tr>
<td>No. of cylinders</td>
<td>1</td>
</tr>
<tr>
<td>Swept volume</td>
<td>980 cm³</td>
</tr>
<tr>
<td>Compression ratio</td>
<td>17</td>
</tr>
<tr>
<td>Bore/stroke</td>
<td>102/120 mm</td>
</tr>
<tr>
<td>Max. power</td>
<td>11 kW at 2200 r/min</td>
</tr>
<tr>
<td>Max. torque</td>
<td>55 N m at 1500 r/min</td>
</tr>
<tr>
<td>Injection pump</td>
<td>Plunger type</td>
</tr>
<tr>
<td>Orifice diameter</td>
<td>0.95 mm</td>
</tr>
<tr>
<td>Injection pressure</td>
<td>13.2–14.2 MPa</td>
</tr>
<tr>
<td>Injection pressure</td>
<td>0.30 MPa</td>
</tr>
</tbody>
</table>

Standard fuel injection pump was replaced by another one, giving higher fuel delivery.

![Fig. 1 Test stand](image-url)
dosemeter PG-80. The ethanol dose per cycle was measured indirectly by a volumetric method. Air flow was measured by a flowmeter installed on the air tank, which reduced pressure pulsation.

A pressure transducer AVL 8QP505 was inserted in the cylinder head to measure the pressure–time history in the cylinder with the use of a high-speed measurement system developed in the Department of Internal Combustion Engines and Automobiles [10]. For exhaust gas analysis, especially CO, CO₂, HC, and air excess ratio \( \lambda \), an AVL 465 DiGas analyser was used. NOx emissions were measured by a Beckman analyser model 951. Also HC was measured with Beckman analyser model 402. Properties of both fuels used are given in Table 2.

### 3 COURSE OF INVESTIGATION

An investigation was carried out into engine operating conditions, Table 3. Measurements were carried out in such a way, that at each speed and injection angle of RME the proportion of ethanol to RME was changed (from zero to the value at which diesel knock occurred), but the torque (load), \( T \), was kept constant. This procedure enabled a comparison to be made between emissions and efficiency under the same working conditions, but for different fractions of ethanol in the total fuel. The schematic of the engine used for this investigation is shown in Fig. 2.

#### 4 RESULTS AND DISCUSSION

4.1 Brake fuel conversion efficiency

Brake fuel conversion efficiency (b.f.c.e.) for each point of the experiment was computed with the measurement of fuel consumptions, engine speed, load and calorific values for both fuels. The b.f.c.e. depends strongly on load and speed; other parameters have a slight influence on it. Selected examples are shown in Figs 3 and 4.

1. In general the higher the load, the higher the efficiency, Fig. 3.
2. The b.f.c.e. depends also on engine speed: b.f.c.e. is at its maximum at middle speed and minimum at high speed, Fig. 3.
3. The influence of the addition of ethanol is not very strong, with an exception at high speed when it increases with the increase addition of ethanol, Figs 3 and 4.
4. The influence of the injection timing of RME is also slight with an exception for late injection, when it is low, especially at high load, and increases with the addition of ethanol, Fig. 4.

### Table 2 Physico-chemical properties of RME and ethanol

<table>
<thead>
<tr>
<th>Property</th>
<th>RME</th>
<th>Ethanol</th>
</tr>
</thead>
<tbody>
<tr>
<td>Chemical formula</td>
<td>-</td>
<td>C₂H₅OH</td>
</tr>
<tr>
<td>Molecular weight</td>
<td>-</td>
<td>46</td>
</tr>
<tr>
<td>Density @20 °C (kg/m³)</td>
<td>878</td>
<td>789</td>
</tr>
<tr>
<td>Calorific value (MJ/kg)</td>
<td>40.0</td>
<td>26.8</td>
</tr>
<tr>
<td>Calorific value of stoichiometric mixture (MJ/kg)</td>
<td>-</td>
<td>3.85</td>
</tr>
<tr>
<td>Heat of evaporation (kJ/kg)</td>
<td>250</td>
<td>840</td>
</tr>
<tr>
<td>Temperature of self-ignition (K)</td>
<td>~400</td>
<td>665</td>
</tr>
<tr>
<td>Stoichiometric air/fuel ratio (kg air/kg fuel)</td>
<td>13.6</td>
<td>9.06</td>
</tr>
<tr>
<td>Lower flammability ( \lambda_l )</td>
<td>-</td>
<td>2.06</td>
</tr>
<tr>
<td>Higher flammability ( \lambda_h )</td>
<td>-</td>
<td>0.30</td>
</tr>
<tr>
<td>Kinematic viscosity, mm²/s @40 °C</td>
<td>4.58</td>
<td>1.4</td>
</tr>
<tr>
<td>Octane number:</td>
<td></td>
<td></td>
</tr>
<tr>
<td>motored</td>
<td>94</td>
<td></td>
</tr>
<tr>
<td>research</td>
<td>11</td>
<td></td>
</tr>
<tr>
<td>Cetane number</td>
<td>60</td>
<td>8</td>
</tr>
<tr>
<td>Flame temperature (K)</td>
<td>-</td>
<td>2235</td>
</tr>
<tr>
<td>Molecular composition (by mass)</td>
<td></td>
<td></td>
</tr>
<tr>
<td>C</td>
<td>0.775</td>
<td>0.522</td>
</tr>
<tr>
<td>H</td>
<td>0.121</td>
<td>0.130</td>
</tr>
<tr>
<td>O</td>
<td>0.104</td>
<td>0.348</td>
</tr>
</tbody>
</table>

Ethanol contained water 8% (by vol.)

### Table 3 Conditions of investigation

| Engine speed \( n \) (r/min) | 1200 | 1800 | 2300 |
| Load \( T \) (N m)             | 20   | 20   | 20   |
| Angle of beginning of injection of RME, deg BTDC | 25, 30, 35 |
| Angle of beginning of injection of ethanol to inlet port, deg ATDC (during inlet stroke) | 60 |
| Ethanol energy to total fuel energy \( \Delta_{fu} \) | From 9 to about 50 per cent |
4.2 Emission

The best emission results were obtained for greenhouse gas and smoke. The emission of CO$_2$ (Figs 5 and 6):

(a) decreases at high loads with an increase of $\Omega_E$ for any injection timing and speed;
(b) is rather independent at low loads on $\Omega_E$, but shows decreasing tendency;
(c) is higher at high loads than at low loads for any ratio of ethanol energy to both fuels' energy ($\Omega_E$);
(d) at the same load the higher the speed, the higher the CO$_2$ emission for any $\Omega_E$.

These results can be explained as follows. A decrease in CO$_2$ with increasing $\Omega_E$ is a result of higher $\Omega_E$. Products of ethanol combustion contain less CO$_2$ and more H$_2$O. Higher CO$_2$ emissions at higher loads is a result of more fuel burnt at lower air excess. An increase of CO$_2$ at high speeds is a result of lower efficiency (Fig. 3) — more fuel is burnt at the same load.

4.3 Smoke emission

The influence of speed and injection timing of RME on smoke emissions as a function of ethanol energy to total fuel energy $\Omega_E$ at low and high loads is shown in Figs 7 and 8. It can be seen that smoke emissions:

(a) are higher at high loads than at low loads for any $\Omega_E$;
(b) decrease at high loads as $\Omega_E$ increases for any injection timing and at low loads smoke emissions are rather independent on $\Omega_E$ with a slight tendency to decrease with $\Omega_E$;
(c) are the highest at the highest engine speed and for fuelling with neat RME;
(d) the later the injection of RME, the higher the smoke emission.

These results can be explained as follows. At high loads, and especially with high speed and a late injection of RME, efficiency is low, which means that more fuel is provided to the engine. However, air excess coefficient $\lambda$ in these working conditions is low, Figs 9 and 10, and therefore in some regions of cylinder space there is not enough air for combustion. A decrease in the smoke level with increasing amounts of ethanol is a result of the fact that, when ethanol burns, no smoke is produced. High smoke emissions at late injection of RME are a result of the shorter time for combustion.

As far as NO$_x$, and CO, and HC emissions are concerned, the results are not as good as those shown above. For NO$_x$ the main influences are: load,
Fig. 5 Influence of speed and injection timing of RME on CO\textsubscript{2} emissions as a function of ethanol energy to total fuel energy, \(\Omega_E\) at low loads.

Fig. 6 Influence of speed and injection timing of RME on CO\textsubscript{2} emissions as a function of ethanol energy to total fuel energy, \(\Omega_E\) at high loads.

Fig. 7 Influence of speed and injection timing of RME on smoke emissions as a function of ethanol energy to total fuel energy, \(\Omega_E\) at low loads.
Fig. 8 Influence of speed and injection timing of RME on smoke emissions as a function of ethanol energy to total fuel energy $\Omega_E$ at high loads.

Fig. 9 Influence of speed and injection timing of RME on air excess coefficient $\lambda$ as a function of ethanol energy to total fuel energy $\Omega_E$ at low loads.

Fig. 10 Influence of speed and injection timing of RME on air excess coefficient $\lambda$ as a function of ethanol energy to total fuel energy $\Omega_E$ at high loads.

Fig. 11 Influence of speed and injection timing of RME on NO\textsubscript{x} emissions as a function of ethanol energy to total fuel energy $\Omega_E$ at low loads.

Injection timing of RME fuel, and the ratio of ethanol energy to total fuel energy ($\Omega_E$). The influences of load and injection timing are shown in Figs 11 and 12. They are:

(a) the higher the load, the higher the NO\textsubscript{x} emissions;
(b) the earlier the injection of RME fuel, the higher the NO\textsubscript{x} emissions.
The influence of the ratio of ethanol energy to total fuel energy is more complicated. NO\textsubscript{x} emissions at low load decrease as the amount of ethanol increases. This is true for all injection timings of RME and all speeds. At high loads the NO\textsubscript{x} emissions depend strongly on the engine speed and the injection timing of RME. For late injection and small amounts of ethanol, NO\textsubscript{x} levels may decrease while for larger amounts of ethanol they increase. In general, the influence of ethanol addition is unfavourable at high loads and is favourable at low loads. This can be explained as follows. At low loads the injection of ethanol, which requires a high heat for evaporation, results in a decrease in temperature of the charge. The temperature of the charge at the inlet port was measured for each experimental point listed in Table 2 and is shown in Figs 13 and 14.

It is evident that the higher the amount of ethanol injected, the greater the drop of inlet charge temperature, especially at high engine speeds and high loads. Due to the lower temperature at the beginning of compression, the temperature level at the end of compression and combustion is lower, which results in a lower b.f.c.e. (Fig. 3) at low loads. At high loads however, the increase in temperature due to greater amount of fuel per cycle compensates for the drop of charge temperature at the inlet, which results in higher NO\textsubscript{x} emissions (Fig. 14) and higher b.f.c.e. (Fig. 3). At high loads the combustion temperature is high and both fuels burn quickly because ethanol vapours promote burning.

Emissions of CO and HCs will be considered together because they are symptoms of incomplete combustion of the fuel. Selected examples of these emissions are shown in Figs 15–17. The influence of engine speed is predominant, so CO and HC emissions are shown together at three different engine speeds. At low speeds both CO and HC increase with increasing amounts of ethanol and HC emissions at low loads and high amounts of ethanol are very high (e.g. for Ω\textsubscript{E} ≈ 0.45), Fig. 15. At middle speeds CO emissions still increase with increasing amounts of
Fig. 15 CO and THCs (measured by Beckman Model 402) emissions as a function of the ratio of ethanol energy to total fuel energy, $\Omega_E$, for low speed ($n = 1200$ r/min), early injection of RME, and high and low loads.

These results can be explained as follows. At high loads, combustion is more efficient (Fig. 3) with increasing amounts of ethanol, resulting in a decrease of CO and HC emissions. At low load, the combustion temperature is lower (also due to the addition of ethanol and the decreasing charge temperature at the beginning of compression) and combustion is incomplete, resulting in a decrease in b.f.c.e.

5 CONCLUSIONS

From the experiments, the following conclusions can be drawn:

1. Dual fuelling with RME and ethanol in the manner proposed offers a drastic decrease in smoke and CO$_2$ emissions at a good b.f.c.e. in all working conditions and also NO$_x$ emissions at low loads and high speeds at low efficiency.
2. The ethanol fraction in total fuel energy $\Omega_E$ may reach 50 per cent at low loads and more than 30 per cent at high loads. The upper limit of the ethanol fraction is the occurrence of diesel knock.
3. Injecting ethanol into the inlet port improves b.f.c.e. at high loads.
4. From the point of view of efficiency, the optimum for smoke, CO, and NO$_x$ emissions at high loads is a ratio of ethanol energy to total fuel energy of about 25 per cent.
5. For the optimum fraction of ethanol, $\Omega_E = 0.25$ and angle at the beginning of RME injection $\alpha = 30$ CA °BTDC in the whole range of speed efficiency (b.f.c.e.) and emissions achieve the values given in Table 4.

6. Further work on measured indicator diagrams will shed more light on combustion processes, resulting in a better understanding of the influence of ethanol on the burning of RME fuel.

ACKNOWLEDGEMENT

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APPENDIX

Notation

- b.f.c.e. break fuel conversion efficiency
- BTDC before top dead centre
- CA crank angle
- CI compression ignition
- RME rapeseed oil methyl ester
- T torque (load) N m
- THC total hydrocarbons
- $\alpha$ crank angle at the beginning of injection of RME, deg BTDC