Alcohol-Based Fuels in High Performance Engines

J. W. G. Turner, R. J. Pearson, B. Holland and R. Peck
Lotus Engineering
ABSTRACT

The paper discusses the use of alcohol fuels in high performance pressure-charged engines such as are typical of the type being developed under the ‘downsizing’ banner. To illustrate this it reports modifications to a supercharged high-speed sports car engine to run on an ethanol-based fuel (ethanol containing 15% gasoline by volume, or ‘E85’). The ability for engines to be able to run on alcohol fuels may become very important in the future from both a global warming viewpoint and that of security of energy supply. Additionally, low-carbon-number alcohol fuels such as ethanol and methanol are attractive alternative fuels because, unlike gaseous fuels, they can be stored relatively easily and the amount of energy that can be contained in the vehicle fuel tank is relatively high (although still less than when using gasoline). These fuels also have a much higher octane rating than gasoline which makes them attractive as a fuel for pressure-charged engines, which are frequently knock-limited in their ignition advance curves.

The modifications made to the engine include the reconfiguration of the fuel system to investigate making best use of the high latent heat of vaporisation of alcohol by injection of a proportion of the fuel mass before the supercharger – the so-called ‘wet compressor’ technique. With this configuration, engine performance data on E85 with optimised engine management settings are presented and compared to the original gasoline-fuelled performance. Discussion is made of the nature of the evaporative effect versus oxygen displacement, its effect on supercharger drive power and of the improvement of the spark advance curve as a result of the increase in octane rating of the fuel. To illustrate this response, curves when varying the percentage of fuel delivered upstream of the supercharger are presented. Some discussion of issues to be addressed on the fitment of such an engine to a vehicle is also made, together with how the wet compressor technique might be adopted with minimal impact on evaporative emissions. Conclusions as to the attractiveness of this approach in maximising ethanol-fuelled engine performance are drawn based on the results presented and the vehicle issues listed. There is also some discussion as to how to best configure an ‘omnivorous’ engine to make best use of a range of liquid fuels as they are introduced to the marketplace, with the best efficiency on each.

ALTERNATIVE FUELS FOR PASSENGER CARS

Ethanol is an attractive fuel for passenger cars because it is possible to offset its CO₂ impact when burnt by using biomass as the main feedstock in its production. This leads to a partially closed CO₂ ‘cycle’. Methanol, meanwhile, is seen by some academics as a potential energy carrier in the long-term future for society because it can be synthesized directly from CO₂ which has already been released into the atmosphere in a shorter CO₂ cycle than that for ethanol derived from biological sources [1]. Thus, with either of these approaches, the rate of release of CO₂ into the atmosphere from transport can be reduced. Indeed, theoretically, further release of CO₂ into the atmosphere can be completely eliminated if efficiency in manufacture can be achieved such that there is no net release of the gas.

Additionally, and unusually for ‘alternative’ fuels, alcohols such as ethanol and methanol have the potential to increase engine performance over that achievable with gasoline due to a combination of factors [2]. These include a lower value of stochiometric air-fuel ratio (AFR) (which maximizes the mass of fuel that can be burnt), a higher mole ratio of reactants to products, high octane rating and, if the benefits can be realized, much higher latent heat of vaporisation. A comparison of some of the pertinent properties of isooctane, gasoline, ethanol, methanol, natural gas and hydrogen is given in Table 1.

Clearly there are significant safety issues to be considered with any fuel. A major concern with both ethanol and methanol is that they burn with an invisible flame, conversely their miscibility in water means that they can be extinguished simply without the need for more complex chemicals. This latter fact has meant that methanol, in particular, has become a mandated fuel in some forms of racing. Due to the interest in ethanol as a transport fuel, it is replacing methanol in some US racing formulae, and is in use by one team in the British Touring Car Championship. This reflects legislators’ desire to promote biofuels in racing as a means of reducing environmental criticism.
Table 1: Properties of several hydrocarbon and alcohol fuels and hydrogen

<table>
<thead>
<tr>
<th>Property</th>
<th>Isooctane</th>
<th>Gasoline (Typical)</th>
<th>Ethanol</th>
<th>Methanol</th>
<th>Natural Gas (Typical)</th>
<th>Hydrogen</th>
</tr>
</thead>
<tbody>
<tr>
<td>Chemical Formula</td>
<td>C₈H₁₈</td>
<td>Various</td>
<td>C₂H₅OH</td>
<td>CH₃OH</td>
<td>CH₄, C₂H₆…, H₂</td>
<td></td>
</tr>
<tr>
<td>Density at Atmospheric Pressure and Temperature (kg/l)</td>
<td>0.69</td>
<td>0.74</td>
<td>0.79</td>
<td>0.79</td>
<td>0.00083</td>
<td>0.00009</td>
</tr>
<tr>
<td>Lower Heating Value (MJ/kg)</td>
<td>44.3</td>
<td>42.7</td>
<td>26.8</td>
<td>19.9</td>
<td>47</td>
<td>120</td>
</tr>
<tr>
<td>Stochiometric AFR</td>
<td>15.1</td>
<td>14.7</td>
<td>9</td>
<td>6.5</td>
<td>17</td>
<td>33.4</td>
</tr>
<tr>
<td>Specific Energy (MJ/kg mixture)</td>
<td>2.934</td>
<td>2.905</td>
<td>2.978</td>
<td>3.062</td>
<td>2.765</td>
<td>3.593</td>
</tr>
<tr>
<td>Specific Energy Ratio*</td>
<td>1.000</td>
<td>0.990</td>
<td>1.015</td>
<td>1.043</td>
<td>0.942</td>
<td>1.225</td>
</tr>
<tr>
<td>Volumetric Energy Content (MJ/litre)</td>
<td>30.6</td>
<td>31.6</td>
<td>21.2</td>
<td>15.7</td>
<td>0.039</td>
<td>0.012</td>
</tr>
<tr>
<td>Research Octane Number</td>
<td>100</td>
<td>95</td>
<td>109</td>
<td>106</td>
<td>120⁸</td>
<td>130⁶</td>
</tr>
<tr>
<td>Motor Octane Number</td>
<td>100</td>
<td>85</td>
<td>98</td>
<td>92</td>
<td>120⁹</td>
<td>-</td>
</tr>
<tr>
<td>Sensitivity**</td>
<td>0</td>
<td>10</td>
<td>11</td>
<td>14</td>
<td>0</td>
<td>-</td>
</tr>
<tr>
<td>Latent Heat of Vaporisation (kJ/kg)</td>
<td>270</td>
<td>180</td>
<td>930</td>
<td>1170</td>
<td>NA</td>
<td>NA</td>
</tr>
<tr>
<td>Mole Ratio of Products to Reactants***</td>
<td>1.058</td>
<td>0.937</td>
<td>1.061</td>
<td>1.065</td>
<td>1 (CH₄)</td>
<td>0.852</td>
</tr>
<tr>
<td>Oxygen Content by weight (%)</td>
<td>0</td>
<td>0</td>
<td>34.8</td>
<td>50</td>
<td>0</td>
<td>0</td>
</tr>
</tbody>
</table>

* Relative to isoctane ** Defined as RON minus MON *** Including atmospheric nitrogen § Methane §§ At φ = 0.4 (λ = 2.5)
†We have not been able to find a reliable published source for this data

From Table 1 it is readily apparent that alcohol fuels possess some significant advantages over gasoline when engine performance is considered; indeed, methanol has for some time been accepted as a good fuel for turbocharged engines [3]. To this must be added that at the mixture strengths typical of those used at maximum power, alcohols such as methanol generally have significantly faster burning velocities than paraffins [4]. This can give a further benefit in reduced negative work in the cycle (i.e. less ignition advance for the same angle of maximum pressure). However, while the combustion performance of alcohol fuels is undoubtedly attractive, the low volumetric energy content caused by their partially-oxidized state has always been a major disadvantage in aviation, the other major area where engine performance is paramount. This is because refuelling during a journey, while possible, is not desirable (at least for civil operation). The subject of this low volumetric energy content will be returned to later. The high octane values of these fuels means that they are often used as blend agents in gasoline, and historically have been considered as a major potential constituent of gasoline fuels, until tetraethyl lead (TEL) was developed as a means of reducing fuel consumption of the vehicle fleet [5].

The data in Table 1 is taken from a variety of sources [2,6,7]. It shows that while some gaseous fuels possess advantageous characteristics compared with gasoline in some respects, they do displace oxygen to a much greater extent, which is extremely disadvantageous at full load unless the fuel can be admitted directly into the cylinder after the intake valve is closed (in which case a form of supercharging actually occurs). Notably, hydrogen possesses a specific energy ratio at stochiometric conditions approximately 20% greater than isoctane, and, with its extremely wide flammability limit of 4-75%, has the potential to produce much more power than gasoline. It also possesses an extremely high octane rating (at least when operated at near to stochiometric conditions [8]). Furthermore, the lighter the gas, the more its propensity to displace oxygen can be used as an advantage at part load in a 4-stroke engine, because this can be used to offset that most unfortunate of 4-stroke cycle disadvantages, throttling loss at part load [9,10].

A practical problem for the end-user of moving to alcohol-based fuels is that the volumetric energy content of these fuels is significantly lower than that of gasoline, as is shown in Table 1. This is perhaps one of gasoline’s greatest advantages. Figure 1 shows a plot of volumetric energy content for ethanol-gasoline blends. The lower volumetric energy content results from the lower calorific (heating) value of alcohols which is due to their partially-oxidized state. However, against the background of the problems of storing light gases on board a vehicle (whether cryogenically or under high pressure), and consequently the necessarily low installed energy, storage and use of alcohol fuels is perhaps a simpler problem. Both ethanol and methanol can corrode light metals and so the fuel system needs to be modified to suit, though these challenges are well understood. They can also be distributed and stored with modifications to the existing infrastructure, which is a significant advantage in their acceptance over hydrogen, for instance. Indeed, distribution and storage of hydrogen is a significant problem, and it is yet to be shown that this can be practically achieved in mass production in particular on board a vehicle, despite some recent success in this area [11].

‘Startability’ can be problematic with alcohol fuels, due to their lower volatility, itself a function of hydrogen bonding. The volatility of gasoline is a consequence of it being a cocktail of different hydrocarbons with a wide range of boiling points. It is for this reason that 15% gasoline is blended into ethanol to form E85 for use in port-fuel-injected engines. This gasoline would not be required when engine performance is considered; indeed, the low volumetric energy content caused by their partially-oxidized state has always been a major disadvantage in aviation, the other major area where engine performance is paramount. This is because refuelling during a journey, while possible, is not desirable (at least for civil operation). The subject of this low volumetric energy content will be returned to later. The high octane values of these fuels means that they are often used as blend agents in gasoline, and historically have been considered as a major potential constituent of gasoline fuels, until tetraethyl lead (TEL) was developed as a means of reducing fuel consumption of the vehicle fleet [5].
A relatively small amount of ethanol or methanol blended in gasoline can have a significant effect [12].

![Figure 1: Volumetric energy content of ethanol-gasoline blends](image1)

In pressure-charged engines, as an alternative to direct injection, introducing the fuel before the compressor can influence performance through reducing compressor work and increasing the charge-air mass flow rate due to the reduced air temperature at the intake. This ‘wet compressor’ strategy was used in aircraft engines for many years as an additional boost in take-off power through injection of alcohols and of water/alcohol mixtures. Indeed, pre-compressor introduction of all fuel to the air stream was the normal means of fuelling an engine for many aero-engine manufacturers, since even gasoline has a beneficial effect when introduced at the inlet to the compressor [13]. The technique has also been shown to be of benefit in gasoline supercharged engines in which it has been shown that in excess of 108 bhp/litre can be achieved without an intercooler [14]. It has also been investigated as a fuel consumption aid [15].

In the test results presented in this work, an ethanol-based fuel containing 15% by volume gasoline was used to fuel what is already a high performance port-fuel-injected gasoline engine. This ‘E85’ fuel is in common use in parts of the world including Sweden and the US and has just been introduced to the market in the UK. There is currently some debate as to how best to introduce ethanol into the spark-ignition fuels market, and the reason for this research was to investigate the magnitude of performance benefit that could be achieved in a sports car vehicle so that ethanol use could perhaps be seen as attractive to the end user. This is the approach employed by Saab in their ‘Biopower’ vehicles, where fuelling the car on E85 enables the customer to increase the power output of the vehicle by 20%, due to the synergy between the high octane value of E85 and pressure-charged engines [16].

The approach could be attractive to some premium OEMs since it may cause ethanol fuelling to be seen as a desirable attribute by the market place and hence accelerate its uptake, rather than its being introduced piecemeal into the gasoline fuel pool. Brazil, however, has introduced large amounts of ethanol into gasoline for all SI-engine vehicles (to create ‘gasohol’), though a practical limit of 24% by volume has previously been reported without resorting to major engine modifications [17]. Using a high percentage ethanol blend, however, would become an attractive proposition to premium OEMs if a CO₂ offset factor can be agreed for ethanol-fuelled vehicles due to the CO₂-regenerative nature of the production process. In order for this to be a benefit, of course, all ethanol would have to be mandated to come from renewable sources. The contract with the vehicle customer will then take the form of balancing increased performance with higher volumetric fuel consumption, and (it could be argued) the high performance, premium vehicle purchaser will be less affected by the higher running costs due to increased fuel consumption.

A final consideration is that, due to the finite nature of fossil fuel reserves, the need to find a viable alternative is gaining in importance. Furthermore, aviation fuel stocks will need to be protected for a longer period of time than automotive fuel reserves. The former require the high energy density that is most easily obtained from the extremely long chain hydrocarbons found in crude oil. Thus, gasoline from fossil sources may have to be replaced first - there is no guarantee that gasoline will be available for as long as the fossil stocks are. The authors believe that alcohol fuels offer a practical solution for passenger vehicle use, offering as they do improved performance and the freedom from fossil-based sources which will become important in the future [1].

The base engine employed for this work, fuelling philosophy adopted, dynamometer results gathered and some of the practical considerations in applying the approach to a vehicle are discussed in the following sections. Some future considerations with regard to future flex-fuel engines are also included.

**BASE ENGINE**

The base engine used for this work was a Toyota 2ZZ-GE engine, supercharged by Lotus for sale in the ‘Exige S’ vehicle, as shown in Figure 2.

![Figure 2: Lotus Exige S vehicle. Note roof scoop for conducting cooling air to the charge air cooler](image2)
The initial work in supercharging this unit for the vehicle application has been reported elsewhere [18]. The charging system employed comprises an Eaton M45 Roots-type blower with air-to-air charge cooler fed by cooling air ducted from an intake pressurized by forward motion of the vehicle. Figure 3 is a photograph of the production engine. Table 2 lists key data of the engine when operated on 95 RON gasoline. The engine management system is Lotus’ T4e production system with electronic throttle actuation.

Table 2: Base engine data of supercharged Toyota 2ZZ-GE engine as fitted to Lotus Exige S

<table>
<thead>
<tr>
<th>General Architecture</th>
<th>In-Line 4-Cylinder with Chain Driven DOHC and 4 Valves per Cylinder with VVTL-i Cam Profile Switching mechanism</th>
</tr>
</thead>
<tbody>
<tr>
<td>Material</td>
<td>All Aluminium with MMC Cylinder Block</td>
</tr>
<tr>
<td>Bore x Stroke</td>
<td>82.0 mm x 85.0 mm</td>
</tr>
<tr>
<td>Displacement</td>
<td>1796 cm³</td>
</tr>
<tr>
<td>Compression Ratio (CR)</td>
<td>11.5:1</td>
</tr>
<tr>
<td>Maximum Power (DIN 70020)</td>
<td>162.5 kW (218 bhp) at 7800 rpm</td>
</tr>
<tr>
<td>Maximum Torque (DIN 70020)</td>
<td>215 Nm (158.6 lbft) at 5500 rpm</td>
</tr>
<tr>
<td>Charging System</td>
<td>Eaton M45 Roots-type Blower and air-to-air charge cooler with force-fed cooling air</td>
</tr>
<tr>
<td>Intake Cam Timing Variation</td>
<td>43°C</td>
</tr>
<tr>
<td>Cam Profile Switch Point at Full Load</td>
<td>4500 rpm</td>
</tr>
<tr>
<td>Maximum Engine Speed</td>
<td>8500 rpm (Intermittent) 8000 rpm (Continuous)</td>
</tr>
<tr>
<td>Minimum Octane Requirement</td>
<td>95 RON</td>
</tr>
<tr>
<td>Engine Management System</td>
<td>Lotus T4e</td>
</tr>
<tr>
<td>CO₂ Output in Vehicle</td>
<td>216 g/km</td>
</tr>
</tbody>
</table>

From the initial research work into supercharging the 2ZZ-GE for application to this vehicle, it was known that the base engine is very knock resistant, as is illustrated by its ability to tolerate a compression ratio of 11.5:1 when supercharged and operating on 95 RON gasoline. This can be seen by the spark advance curves for the naturally aspirated and supercharged engines shown in Figure 4. The NA engine is not very knock limited; the supercharged engine, while becoming knock limited at full load, shows a maximum reduction in spark advance in the mid range of 10° crank angle. The use of ethanol fuel was thus expected to allow the engine to operate substantially knock-free through most of its speed range.

FUELLING PHILOSOPHY ADOPTED

Since it was known that the fuelling rates would be higher on E85 than 95 RON gasoline, the port fuel injectors were changed to a higher-flowrate specification. These uprated injectors had a static flow rate at 3 bar pressure of 425 cc/min versus 300 cc/min for the standard 2ZZ-GE injectors, and are essentially an upgraded version of those fitted to the standard Exige S. Even with the high-flow specification chosen, it was calculated that there would be insufficient fuelling to run the engine entirely on port injection above 6500 rpm only, and thus two additional injectors were added. This is discussed in more detail later. These extra injectors were positioned before the supercharger so that an investigation into the effect of deliberate excess pre-supercharger fuelling could be made when operating on E85. The intention here was to investigate the value of attempting to realise the greatest benefit of the high latent heat of vaporisation of the E85 fuel by injecting a high proportion of the fuel mass before the supercharger.

The engine management system, Lotus’ T4e system, was modified to control the two additional pre-supercharger (PC) injectors¹, which were themselves of the same specification as the upgraded port fuel injectors. They were positioned in the ‘swan neck’ between the throttle body and the supercharger entry. A photograph of the PC injectors in the swan neck is shown in Figure 5.

¹ In this work the term ‘pre-supercharger’ has been abbreviated to ‘PC’ to be consistent with the previous work by the present authors described in reference [14]. However, the written text here will continue to use the term ‘pre-supercharger’ in recognition of the fact that, because it lacks internal compression, a Roots blower is not a true compressor.
The two extra PC injectors were positioned so that the path of fuel into the supercharger entry corresponded with the maximum air velocity at the exit bend in the swan neck. An additional fuel rail was fabricated and the wiring loom modified to allow the engine control unit (ECU) to drive the extra injectors. Because of the high heat of vaporisation of the fuel and the fixed pulley ratio of the positive displacement supercharger, one of the primary mechanisms for increasing the power of the engine is to increase air mass flow through the supercharger by reducing the intake temperature to the rotors. This is discussed below.

A special software program and calibration of the T4e engine management system was created to drive the PC injectors independently of the port injectors; they were timed such that one fired at the same time as the port injector of cylinder 1 and the other with that of cylinder 4, i.e. 360°CA apart. Thus, when operating on long pulse widths, there would be a significant overlap of flow from these injectors. With these modifications, tests to gauge the benefit of attempting to evaporate the fuel before the supercharger could be conducted.

An operational difference between the injectors was that the PC injectors were subject to a greater pressure differential across them versus those in the intake ports (due to the lack of boost pressure on their outlet side). This allowed high rates of fuel load to be applied pre-supercharger despite there being half the number of injectors as at the intake ports.

Table 1 shows that the specific energy content (the lower heating value divided by the stoichiometric AFR) of ethanol is about 2.5 per cent greater than gasoline. For E85 the advantage is still over 2 per cent. The greater molar ratio of products to reactants, and the faster flame speed of ethanol both provide possible further performance benefits from the E85 fuel. It is, however, the significantly higher octane index and latent heat of vaporization which offer the greatest potential for performance increase.

The addition of some portion of the fuel upstream of the supercharger is aimed at exploiting the latter characteristic to some extent. With a fixed drive ratio to the supercharger the only way to increase the mass flow rate is to increase the air density upstream of it. With the arrangement shown in Figure 5 the close proximity of the fuel injectors to the supercharger will cause a significant amount of the latent heat of vaporization of the fuel to be extracted from the metal of the rotors. It is also likely that there will be a gradual evaporation of the fuel as it passes through the supercharger. This makes it very difficult to make a simplistic prediction of the magnitude cooling effect of the fuel on the air charge. The other factors which impact on the engine performance through adding the fuel upstream of the supercharger relate to the effect on the ‘compressor’ work. Increasing the mass flow rate of fuel through the supercharger increases the compression work whilst reducing the temperature rise across serves to reduce the compressor work level. Adding fuel to the air changes its properties by increasing the specific heat capacity and lowering the value of the ratio specific heats – the former effect increases the compression work and the latter decreases it. It was shown in [14] that the net effect of all these phenomena is a decrease in the amount of work performed on the air by the supercharger.

**DYNAMOMETER TESTING AND RESULTS**

The engine was coupled to the dynamometer and instrumented to record temperatures and pressures in the intake and exhaust system. It was also fitted with a water-cooling system for the standard chargecooler matrix for this work, to accurately control outlet temperature to the plenum chamber. The air side of the chargecooler was kept standard. Figure 6 is a photograph of the engine installed in the test cell.

Since this was an engine taken from a vehicle and thus had not had a controlled run-in performed on it, it was decided to perform a baseline test on the engine with a vehicle calibration suitable for the upgraded port injectors used. This test was carried out without any gasoline being introduced via the PC injectors as it was not considered advantageous when running on gasoline with a chargecooler in use [14]. This effectively created a specification expected to deliver approximately 235bhp with the opening angle of the electronic throttle adjusted to 74%, chosen to avoid a 100% duty cycle on the injectors. Power, torque and brake specific fuel consumption (BSFC) results of this baseline test are shown in Figure 7. In order to minimize variables, this was performed on an auto calibration suitable for this injector specification; base fuelling was leanest fuel for best torque (LBT) up to the cam switch point of 4500 rpm and essentially an air-fuel ratio (AFR) of 12.7:1 above that speed. Spark advance was minimum advance for...
Best torque (MBT) or borderline knock (BLD) minus 2°, whichever was reached first; this was the case for most of the speed range.

Maximum power for this specification was 172.4 kW (231 bhp) at 7800 rpm and maximum torque was 223 Nm at 6000 rpm, representing a brake mean effective pressure (BMEP) of 15.6 bar. These results were as expected from the above discussion of injector and calibration changes and so the engine was passed off for the E85 development work.

For these tests, the plenum temperature was held constant at 30 and 41°C for the speeds of 2000 and 5500 rpm, respectively. These are the standard temperatures for the engine when operating on gasoline, and were adopted to minimize variables. It should be noted that for both engine speeds presented, the compressor outlet temperature became less than the controlled plenum temperature at 50% PCI fuelling rate. Therefore, the increase in torque above this rate for both speeds in Figure 8 is achieved despite the water-cooled chargecooler actually heating the charge up. The magnitude of the improvement due to PCI fuelling is such that a large proportion of it is likely to be due to compressor work reduction. This will be the subject of a later publication.

It should be mentioned that the percentage of fuel delivered by each set of injectors was not independently measured during this work. Instead the figure quoted is a calculation performed in the T4e ECU based upon various parameters, including static flow for each set of injectors. The differential pressure for each set of injectors is not taken into account, and so for each of the percentage fuel loadings shown for the pre-supercharger injectors, slightly increased flow rate may be expected at this position.

Results are not shown at 8000 rpm as the response was seen to be flat – less than 0.7%. Part of the reason for this is that, as mentioned above, the port injectors cannot supply all of the fuel required to run the engine at this speed.
point. This is illustrated in Figure 9, which shows the base fuel delivery required to run the engine at full load, the total that could be supplied by the port injectors, and the point at which the PC injectors are brought into action because the port injectors are at their flow limit. These calculations take into account the static flow rate of the injectors together with a closed period for the injectors of 500 μs per cycle. PC fuelling is not a necessity until the total fuel load required exceeds that which can be supplied by the port injectors alone, which occurs at above 6500 rpm. It should also be pointed out that these tests were reported at 74% throttle, for reasons discussed in the following section.

It can be seen in Figure 9 that at 8000 rpm the PC injectors have to supply 15.6% of the total fuel load. Other reasons for the flat response above 20% PC fuelling rate during the 8000 rpm test are unclear, but could be due either to the throttle becoming an air restriction at this speed, or the fixed speed nature of the supercharger drive. On a turbocharged engine, with its variable ratio supercharger drive system (i.e. the turbine and wastegate), it is possible to make more benefit of the higher knock resistance of the ethanol-based fuel, since the boost can be increased easily as well as the ignition advanced. This cannot be achieved with a simple Roots blower/fixed ratio drive system. Further evidence of this effect could perhaps be seen in the fact that between 2000 to 5500 rpm the benefit of introducing fuel pre-supercharger is approximately halved, from 8 to 3.5%, as is shown in Figure 8. This is also in line with the reduced improvement in spark advance at 5500 rpm (see later).

With 95 RON gasoline, the torque generated at these speeds was 168, 221 and 204 Nm, respectively. Performance of the engine at these speeds with the different fuels and the pre-supercharger fuelling philosophy adopted is summarized in Table 3.

Figure 10 shows that the air is fully saturated by the E85 as the fuelling rate to the PC injectors reaches 40% for both the 2000 and 5500 rpm cases. Indeed, the implication of this is that, since the temperature increase across the supercharger becomes zero from this fuelling rate onwards, it would be possible to remove the intercooler with no reduction in power for this condition.

Hence, for high performance engines running on low-carbon-number alcohol fuels with PCI, saturation of the air can be achieved at moderate boost pressures such that there is no net increase in charge temperature across the supercharger. Therefore, splitting the fuelling rate between the port and the pre-supercharger location allows the maximum performance to be obtained through optimised charge cooling and minimum air displacement via precise saturation of the air by the fuel.

ENGINE PERFORMANCE ON E85 AND COMPARISON WITH 95 RON GASOLINE

A torque curve while operating on E85 was obtained at a fixed PC fuelling rate of 30%. This figure was decided upon as a good compromise with respect to performance while avoiding creating an ignitable mixture in the intake system. It also avoided operating the injectors on a 100% duty cycle (because of the lower volumetric energy content of E85 versus gasoline), and, as discussed above, it was not possible to operate above 6500 rpm solely on the uprated port injectors anyway.

As before, the throttle angle was held at the same 74%
to ensure a true comparison to the gasoline baseline based on atmospheric oxygen flow rate. On E85, fuelling was set to 9.1:1 at 1500 rpm and 8.7:1 AFR at all points above this and, similar to the 95 RON tests, spark advance was set to MBT; this was possible because the performance was not limited by the onset of knock throughout the speed range. This approach resulted in an increase in engine power of 33 bhp, the engine now producing 197.1 kW (264 bhp) at 8000 rpm (+14%) and 246 Nm at 5500 rpm, representing a BMEP of 17.2 bar and an increase in torque of 23 Nm (+10%). Figure 11 shows the performance of the engine when operating in this manner on E85 fuel.

The major proportion of the improvement must be due to the extra spark advance made possible by the higher octane value of ethanol. Figure 12 shows a comparison plot of torque and power for these fixed maximum throttle position tests on the two test fuels, while Figure 13 compares the spark advance for the two supercharged conditions with the naturally-aspirated spark advance curve reproduced from Figure 4.

The increase in spark advance that can be sustained on E85 is striking; at 2000 rpm on 95 RON the maximum spark advance (limited by knock) was 2.5° btdc, while on E85 the maximum spark advance (at MBT) was 16.5° btdc, an increase of 14° Crank Angle (CA). This increase is typical of that obtained at the lower end of the curve, until the switch point from the low-lift to the high-lift cam profiles at 4500rpm. After this speed the reduced effective compression ratio due to the later intake valve closing timing and improved scavenging of residuals due to the increased angle-area presumably helps with the knock limit when operating on gasoline. The improvement in spark advance due to operation on E85 is still in the region of 5-7° CA.

Figure 13 shows that this test engine with its slightly different injectors can support approximately 5° greater spark advance at 4500 rpm (the cam profile switchover point). Furthermore, and more pertinently, as shown on Figure 13, the supercharged engine when operating on E85 relative to the NA engine on 95 RON gasoline, can still support 4-5° more spark advance through most of the speed range, and 8° more at 1500 rpm. These facts would tend to suggest an improvement in thermal efficiency, which will be returned to later.

Figure 14 shows that operation on E85 is lowering the outlet temperature by almost 40° C at 4500rpm relative to gasoline. The slight increase in plenum pressure is due to the increase in overall mass flow rate due to the lower stochiometric AFR value of ethanol.

The plenum temperature was held to a fixed relationship with respect to engine speed for all of these tests by altering the flow rate of cooling water through the chargecooler. This curve was the same as that for homologation of the Exige S vehicle and was adopted in the interest of reducing the number of test variables. A combination of the lower adiabatic flame temperature of E85, the significantly increased spark advance its higher octane rating permits, and the higher latent heat of vaporisation of any fuel passing into the exhaust during the overlap phase would therefore be expected to yield markedly reduced exhaust temperatures. This is shown to be the case in Figure 15, a plot of the temperature in
the runner of cylinder number 3, which was the hottest for both tests.

The difference in exhaust temperatures between 2000 and 4000 rpm due to operation on E85 is in the region of 100 to 150°C. After the cam profiles switch at 4500 rpm and the difference in spark advance is reduced, this difference similarly reduces to a fairly constant 50°C. Figure 16 shows a comparison of exhaust back pressure (EBP) for operation on the two fuels. The EBP for E85 is higher, as may be expected from the increased mass flow (inferred from higher plenum pressure and higher AFR), though the point at which the cam profiles switch is clearly discernible as well.

**DISCUSSION**

The response of the engine to being fuelled with E85 is in line with that expected from the fuel characteristics, since the 14% improvement in maximum power bears comparison with the 20% improvement claimed by Saab for their turbocharged ‘Biopower’ engines [16], in which the boost pressure is presumably increased and which is not possible with this configuration of positive displacement supercharger and fixed-ratio drive. Variation of the amount of fuel introduced before the supercharger also showed the expected response at low speed, insofar as a performance benefit was seen in the tests conducted at 2000 and 5500 rpm (Figure 8).

However, the pre-supercharger fuelling tests did suggest that something was restricting performance of the engine at 8000 rpm. In order to investigate the notion that the throttle angle was becoming a restriction at 8000 rpm, a power curve was conducted with it fully open using E85 with 30% PC fuelling rate. This resulted in a maximum power of 199.9 kW (268 bhp) and maximum torque of 249 Nm. From this it was deduced that the throttle angle was not a serious restriction on performance, and that the swallowing capacity of the supercharger was the limit. Due to the knock-free operation on E85 it was therefore concluded that the fixed drive ratio of the positive displacement blower should be altered for any future testing, in order to generate more boost and exploit the knock limit benefit of E85 further. This has not been carried out in this test programme, but suggests a worthwhile future direction; operationally, this is of course one major advantage a turbocharged engine possesses over a supercharged one. Nonetheless, the fixed swallowing capacity of the engine as determined by its charging system configuration has allowed combustion effects to be divorced from charging system effects to an extent that operation with a turbocharger may preclude. The benefit of this approach has been shown in the results given in the PCI tests.

Investigation of the % PC fuelling tests in Figure 8 shows that further optimisation of PC fuelling would only increase maximum torque over the 74% throttle angle E85 power curve conducted at 30% PC fuelling rate by approximately 2 Nm (see Table 3). However, at 2000 rpm a further increase of 12 Nm (to 224 Nm) could be realized through detailed optimisation of pre-supercharger injection (PCI). When compared to the 168 Nm that the engine delivered on gasoline fuelling at this speed, this would be an improvement of 56 Nm, or 33%.

Clearly the extension of the knock limit afforded by using

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![Fig. 14: Comparison of supercharger outlet temperature and plenum pressure for test engine operating on E85 and 95 RON Gasoline](image1)

![Fig. 15: Comparison of exhaust runner number 3 temperature for test engine operating on E85 and 95 RON Gasoline](image2)

![Fig. 16: Comparison of exhaust back pressure for test engine operating on E85 and 95 RON Gasoline](image3)
alcohol-based fuel can be exploited to a significant degree in pressure-charged engines by employing alcohol-based fuels, and the subject of how to use this to improve the thermal efficiency of such engines is returned to in a later section.

BSFC when operating on alcohol-based fuels is thrown into sharp relief by this series of tests. Figure 17 shows the BSFCs when operating on the two fuels, together with the ratio of the two.

The graph in Figure 17 underlines a fundamental point when comparing the specific fuel consumption of paraffinic and alcohol-based fuels: that the disadvantage of the energy content of the alcohol fuel in volumetric terms (as indicated in Figure 1) can be minimized if the engine can operate in a knock-free manner at a high compression ratio. For example, the calorific content of 95 RON gasoline is approximately 42.7 MJ/kg and that for E85 is 29.1 MJ/kg. The ratio of these values is 1.47, so that for the same combustion efficiency one might expect the BSFC of E85 to be 1.47 times higher than that on 95 RON. Clearly Figure 17 shows this not to be the case, and this is due to the fact that one of the fuels was knock limited under these conditions and the other could operate at MBT.

A far better comparison of these fuels is that for thermal efficiency [19]. From this, the degradation in efficiency due to operation on gasoline versus the alcohol-based fuel can be calculated quite simply. These results are shown in Figure 18, in which the peak efficiency for E85 is shown to be 33.8% at 2500 rpm, while that for 95 RON is 30.7% at 1500 rpm, i.e. gasoline is 9% worse in terms of peak thermal efficiency. More tellingly, at maximum power and maximum torque, the knock-limited nature of gasoline degrades the thermal efficiency of the engine by 12.5 and 16% respectively, though it is accepted that some of the change in performance is related to the use of the wet-compressor technique when operating on E85. With charge-air cooling, the wet compressor technique is not believed to be especially beneficial when operating on gasoline, due to the significantly lower latent heat of vaporization (see Table 1 and reference [14]).

As an aside, using PCI with E85 also has an effect on thermal efficiency and can improve it further, as is shown in Figure 19, which is data for the loops at 2000 and 5500 rpm. At the 5500 rpm maximum torque point the improvement is presumably due to the reduction in supercharger drive power as discussed above. At 2000 rpm, there was a lower change in thermal efficiency during the loop, however.

The type of engine used in this series of tests is unusual insofar as it can be operated on gasoline with a combination of a high CR and pressure charging, which allows a better comparison of the efficiency benefit when operated on alcohol-based fuels. This was illustrated by the data in Figure 18, with the percentage degradation in efficiency when operated on gasoline. Nevertheless, this degradation is primarily a result of the combination of characteristics chosen for the engine which preclude it being operated at MBT everywhere when operating on gasoline. Nevertheless, this degradation is primarily a result of the combination of characteristics chosen for the engine which preclude it being operated at MBT everywhere when operating on gasoline. Nevertheless, this degradation is primarily a result of the combination of characteristics chosen for the engine which preclude it being operated at MBT everywhere when operating on gasoline. Nevertheless, this degradation is primarily a result of the combination of characteristics chosen for the engine which preclude it being operated at MBT everywhere when operating on gasoline. Nevertheless, this degradation is primarily a result of the combination of characteristics chosen for the engine which preclude it being operated at MBT everywhere when operating on gasoline. Nevertheless, this degradation is primarily a result of the combination of characteristics chosen for the engine which preclude it being operated at MBT everywhere when operating on gasoline.
fuel ratio, has been known for many years [19].

CONSIDERATIONS IN APPLYING A PRE-SUPERCHARGER INJECTION APPROACH WITH E85

The basic problems of operating with a high percentage of alcohol in gasoline are well known, and have been addressed for some years in Brazil [17]. Employing a pre-supercharger injection approach for some of the fuel loading is not without further challenges, however. Chief amongst these are probably evaporative emissions and the effect on the supercharger of ethanol being carried in the air stream.

With regard to evaporative emissions, it is known that E85 is significantly better in this respect than gasoline [20]. Thus, a fuelling strategy which minimizes the amount of gasoline contained in the fuel rail spur to the PC injectors by only permitting PCI when the ethanol content in the fuel line from the tank is high would be expected to similarly minimize evaporative emissions at this point in the engine intake system. This approach would need to be studied in greater detail, and may require an ethanol sensor in the line rather than a reactive sensing approach based on a fast-acting exhaust gas oxygen sensor. With control of the electronic throttle as well as fuelling and ignition, this would be the prime route to compliance in this area. Minimizing the amount of fuel sent to the PC injectors based upon a map of air flow against PC fuelling rate would also be of benefit in this respect.

Ethanol in the air stream may create a harder challenge, however. The engine operated for only a few hours on E85 with PC fuelling, but the rotors showed signs of removal of the abradable coating which they carry. This is shown in Figure 20. However, it should be borne in mind that this supercharger was not new when subjected to E85 operation, so the exact amount of damage due to the abrasive nature of ethanol, if any, cannot be fully quantified. It may be that this damage was due to droplet impact and may not be primarily due to the abrasiveness of the liquid; the exact reason for the damage is unknown at present.

Fig. 20: Removal of supercharger rotor coating

Coatings are typically damaged by alcohols, and the underlying aluminium of the rotors shown in Figure 20 was not harmed. If a non-coated turbocharger compressor were used instead, this may not suffer the same amount of damage. Since introduction of the fuel before the compressor has been used for Indianapolis-type racing [21], as well as to increase the take-off performance of aircraft engines, it is presumed that this challenge is not an insurmountable one, given that the improvements in specific output and thermal efficiency are deemed attractive enough to warrant the approach. Since this would enable further downsizing of an engine and greater reduction in throttling loss, there is perhaps some merit in investigating the concept further, especially since, in a turbocharged engine, this would result in lower pumping work through the reduced turbine expansion ratio required to drive the compressor.

If these problems were viewed as too difficult to attempt to surmount, an alternative is to introduce the extra fuel via extra injectors positioned after the pressure-charging device, which has been arranged previously in gasoline engines where secondary injectors were necessary because of flow-range limitations of the port fuel injectors [22]. Introducing alcohol-based fuel at this point in the intake system will not have the same benefit in reducing compressor power, but could still allow a significant reduction in charge air temperature due to the high latent heat of vaporisation of the alcohol. This could be more beneficial if a variable-speed drive was employed, as is the case in a turbocharged engine. Positioning the extra injectors here would also be expected to have a beneficial impact on evaporative emissions in the case of an engine supercharged by a positive-displacement device, since this would form a gate to the fuel vapour. It is unlikely that the engine could be operated without a charge cooler with this approach, however, the potential of which will be discussed in the next section.

CONCEPTS TO IMPROVE FUEL EFFICIENCY IN FLEX-FUEL VEHICLES

This series of tests shows that alcohol-based fuels, with their high octane rating, can permit such an extension of the knock limit that within the constraints of conventional engine architectures there is a significant compromise to be accepted if the engine is also required to operate on gasoline. Alcohol-based fuels can be used to permit high compression ratio and high boost pressures to permit downsizing such as is currently being pursued for gasoline engines. Thus if the engine is developed primarily for alcohol-based fuels, part load thermal efficiency could exceed gasoline operation. This would be expected to result from a combination of throttling loss reduction due to downsizing without the attendant decrease in thermal efficiency a lower CR enforces, despite the fact that a higher CR works against the reduction of throttling loss. This is effectively what the test engine used here represents, though its high-speed nature is another factor to be considered in comparing it to engines of similar power output [18].
The observations in the section on PCI fuelling tests with regard to the reduction in charge temperature increase across the compressor as the PCI fuelling rate is increased show that, for this boost level, operation without an intercooler on alcohol-based fuels would be possible; a zero-degree temperature rise implies an intercooler would be redundant. Removal of this system could represent a significant reduction in bill-of-material (BOM) and aerodynamic drag in-vehicle. Any flex-fuel vehicle with a bias towards operation on alcohol-based fuels could thus beneficially exploit this approach.

TRANSMISSION EFFECTS

The amount of degradation in thermal efficiency caused by operating on 95 RON is very significant. This clearly shows that if alcohol-based fuels are to be introduced rapidly (in terms of proportion of vehicle miles fuelled on them) a new approach should be adopted to minimize the disadvantage of requiring the engine to be capable of flexible fuel operation. This will encourage greater uptake of a fuel the manufacture of which captures CO₂ from the atmosphere, and will drive the development of the infrastructure necessary to manufacture and distribute it.

One approach would be to configure an engine with a high compression ratio which is preferentially specified to operate most efficiently on alcohol fuels – say, with 12 or 13:1 CR – and be made capable of using gasoline in a lower power mode only. However, with a manual transmission, this approach would require that the end user accept the compromise in vehicle performance brought about by the inefficiency of gasoline at the times when that fuel is used.

At full load, an engine gives its highest efficiency when it operates at the highest knock-free CR on optimum ignition timing and stoichiometric AFR. Therefore, employing a full or series hybrid transmission in such a vehicle could permit the engine to be configured with a combination of CR and pressure charging such that the most efficient performance on each fuel could be realized at all times. Here the hybrid system would allow the load levelling necessary to conceal from the driver the difference in performance dictated by the knock limit of each fuel.

COMPLIMENTARY TECHNOLOGIES FOR MAXIMISING EFFICIENCY IN ENGINES EXPECTED TO BE CAPABLE OF OPERATING ON BOTH GASOLINE AND ALCOHOL-BASED FUELS

From all of the foregoing, it is to be expected that variable compression ratio (VCR) is the one key technology that would permit optimised operation on whatever proportion of alcohol and gasoline are used to fuel an engine. With such a system the optimum combination of boost and CR can be used for all operating conditions, with power output equilibrated if desired. The part-load benefit of VCR is well known when an engine is fuelled on gasoline [23]. Moving to a wider-range system (to cover a low setting for full-load gasoline operation to a high CR for part-load operation on ethanol or methanol) would permit far higher efficiency on the alcohol fuel, and hence better volumetric fuel consumption.

The benefit of direct injection of alcohol fuels can be exploited to increase volumetric efficiency [12], and hence would be a good complimentary technology to a VCR system for such a ‘no-compromise’ flexible-fuel engine. Similarly, both ethanol and methanol possess desirable attributes for homogeneous charge compression ignition (HCCI) operation, since they have a high RON coupled with a high sensitivity (S), where

\[ S = RON - MON \]

They should thus be very good fuels for this combustion process [24]. VCR would be a very complimentary technology to this approach, too, giving the ability to vary compression heat independently of retained heat from trapped EGR.

As implied above, a disadvantage of increasing the CR of 4-stroke engines at part load is that throttling losses increase as a result. Furthermore, these losses will also be slightly worse when operating on alcohol fuels because a proportion of the oxygen required for combustion is bound in the fuel itself, and hence is not inducted in gaseous form. Therefore, if a technology existed that minimized throttling loss, permitted simple wide-range variation of CR, operated well on HCCI and provided a suitably flexible architecture (so that DI could be incorporated easily), it should present a compelling case for adoption in a flexible fuel engine.

Such a concept could, in fact, be realized easily in the form of a VCR 2-stroke engine. The advantages of 2-stroke engines in this form can be summarized as follows:

1. They do significantly less work in performing gas exchange at part load than 4-stroke engines, minimizing the problem of throttling losses at high CR;
2. They are known to operate better in HCCI because (a) they inherently trap large amounts of hot EGR due to the operating cycle and (b) there is less time for heat to be lost from the combustion chamber because there are no pumping strokes during the cycle;
3. For any practical automotive application they already require DI to control unburnt HC emissions; and
4. They are, in loop-scavenged form, so architecturally straightforward that a wide-range VCR system is simple to arrange.

A 2-stroke VCR system permitting variation from (say) 8:1 to 40:1 has been schemed and is shown in Figure 21.
in what has been termed the ‘OMNIVORE’ engine due to its inherent flex-fuel potential. Clearly the approach to VCR shown in Figure 21 cannot easily be adopted in a poppet valve engine without significant adverse impact on valve sizes and combustion chamber layout. It is thought that operating at geometric CRs in excess of 20:1 may allow operation in HCCI at idle, something difficult to achieve in a practical SI-based 4-stroke HCCI engine. In fact, the only way to achieve such a simply-implemented VCR mechanism in a 4-stroke engine may be by adopting a sleeve-valve layout [25], which in itself would require some development in tribology and oil control.

Therefore, with an oscillating charge trapping valve (CTV) providing cyclically-variable asymmetrical timing of the exhaust port, such as was employed on the ELEVATE 2-stroke engine [26] and is illustrated in both Figures 21 and 22, such an engine would have two separate systems giving completely independent control of retained heat and compression heat. It would therefore possess controls on the HCCI process which are not even available to well-optimised 4-stroke engines with fully variable valve train (because valve timing influences EGR rate and CR simultaneously [27]). Such an omnivorous 2-stroke VCR engine could help provide a more efficient route to a future in which high-alcohol-content biofuels have a significant degree of penetration in the market place.

CONCLUSIONS

The conclusions from this work are that:

1. The use of alcohol fuels can afford a reduction in CO₂ released into the atmosphere by forming a loop in which CO₂ is trapped.

2. The characteristics of the low-carbon-number alcohol fuels methanol and ethanol are ideally suited to use in high compression ratio pressure charged spark ignition engines.

3. Storage of these alcohol fuels aboard passenger vehicles is significantly less problematic than storage of light gases such as natural gas or, in particular, hydrogen.

4. The high octane rating of an ethanol-gasoline blend, E85, has been shown significantly to extend the knock limit of a high performance engine which, while knock-limited on gasoline, is believed to be already good in these terms.

5. Thermal efficiency of this high compression ratio engine when operating on the ethanol blend was significantly better than for gasoline, primarily because of the extent to which the knock limit was increased. In this respect, it is suggested that it is better to think in terms of how much gasoline compromises the performance of an SI engine due to its lower octane rating.

6. The high latent heat of vaporisation of ethanol has been shown to be exploitable in terms of extra performance through introduction into the air stream before a supercharger. Optimisation of the proportion of total fuel load provided at this point has been shown to increase both BMEP and thermal efficiency.
7. The injection of approximately 40% of the total fuel load at the supercharger entry resulted in no increase in temperature across the supercharger relative to atmospheric conditions. This was at a boost pressure of approximately 40-50 kPa.

8. Ethanol and methanol are both known to be corrosive and abrasive to certain materials in comparison to gasoline. This has possibly been borne out by damage to the coating of the supercharger used. A different strategy for any production application may be needed, but an uncoated turbocharger impeller may well be unaffected.

9. Future engines can be optimised to greater thermal efficiency on alcohol fuels by increasing their compression ratio. Strategies to allow the engine to operate on gasoline – albeit at lower performance – would then have to be developed.

10. Variable compression ratio would be a key technology to obtain the best performance from an SI engine expected to operate flexibly at the best possible efficiency on different hydrocarbon and alcohol fuels.

11. Due to the high RON and sensitivity of ethanol and methanol, a concept to enable HCCI (with minimum pumping work and independent control of trapped and compression heat) could be provided by a variable compression ratio 2-stroke engine. This could provide extremely high thermal efficiency in a cost-effective engine with omnivorous fuelling capability.

12. Optimizing engines to run at high efficiency on alcohol fuels would permit more rapid introduction and penetration of these fuels into the market for transport fuels.

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CONTACT

James Turner
Powertrain Research
Lotus Engineering
Norwich
Norfolk NR14 8EZ
United Kingdom

Email: jturner@lotuscars.co.uk

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Abbreviations:

BLD Borderline knock (previously ‘detonation’)
BMEP Brake mean effective pressure
BOM Bill-of-material
BSFC Brake specific fuel consumption
btdc Before top dead centre
CA Crank angle
CR Compression ratio
EBP Exhaust back pressure
ECU Engine control unit
LBT Leanest mixture for best torque
MBT Minimum advance for best torque
NA Naturally aspirated
OEM Original equipment manufacturer
PC Pre-supercharger
PCI Pre-supercharger injection
SI Spark ignition
TEL Tetraethyl lead
VCR Variable compression ratio