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ABSTRACT

Ethanol shows promise as a renewable automotive fuel. However the widespread adoption of ethanol as a fuel rather than gasoline diluent is hindered by several issues including cost. The current study evaluates the comparative performance for ethanol and gasoline fuels in a multi-cylinder turbo-charged direct injection SI engine. In particular the study investigates the potential for high specific output with ethanol, an enabler for highly efficient and more market tenable powertrain solutions.

Data indicate that the operation of a turbo-charged spray guide DI engine on pure ethanol can efficiently achieve very high specific output with some update to engine design. The ethanol direct injection or EDI approach shows overall significant potential for aggressive engine downsizing for a dedicated or dual-fuel solution.

INTRODUCTION

The global vehicle park is expanding at an increasing rate and has been projected to exceed 1 billion vehicles in the 2020 to 2025 timeframe [1, 2]. It is also indicated that within this timeframe the internal combustion engine will remain the dominant vehicle powerplant [3]. At present the ICE is largely dependent on liquid fuels derived from crude oil, and the projected rate of consumption presents key issues for the environment, and longevity and security of energy supply. It is therefore imperative to both improve the efficiency of the ICE and to enable operation with alternative fuels, especially those that may be considered renewable.

Ethanol has been considered as an alternative and renewable automotive fuel for at least 100 years [4], and for some considerable time as an extender for compression ratio and output in dual-fuel operation with fuels of lower octane [5, 6]. Interest in ethanol has continued through to the present, and the renewed urgency to find sustainable energy vectors coupled with advances in engine technology has driven a new wave of worldwide research into ethanol as a fuel rather than a gasoline diluent [7-13]. Notwithstanding the above, Brazil has singularly demonstrated the widespread production and utilization of ethanol as an automotive fuel in pure azeotropic form [14].

There does however remain considerable debate on the viability of ethanol as an alternative to fossil fuels. For fuels of high ethanol content the net environmental advantage as measured by greenhouse gas CO_2 equivalent may be positive with regard to Tank to Wheel emissions [15]. However for entire lifecycle Well to Wheel GHG emissions which include the supply side contribution, benefit is highly dependent on the method of production [16, 17]. Moreover, supply side infrastructure at present limits supply and offers a cost barrier to the ready adoption of ethanol [18]. The low volumetric heating value of ethanol compounds this latter issue, and in addition presents a fuel economy disadvantage.

To offset the impact of supply side issues, research activity on the demand side is focusing on approaches that leverage the particular properties of ethanol to achieve a highly efficient ICE. These approaches may be viewed as those of dedicated fuelling [7, 8, 12], and those utilising a dual or multiple fuel strategy [9, 11] in which ethanol is applied in areas of knock constraint. In either case success is contingent upon achieving a high specific output, ideally with high compression ratio.

The concept of enabling higher gasoline ICE operating efficiency through higher specific output and down-sizing is well reported in the literature [19-22]. It is typical that turbo-charging is utilised to efficiently achieve high engine airflow across the engine speed range, and is coupled with direct injection to facilitate low end performance and increased compression ratio. Nonetheless specific output is still constrained by the combustion characteristics of gasoline, and whilst this may be partially offset by dilution strategies [22-24] it remains the case that compression ratio and output are both limited. It is anticipated that the operation of a turbo-charged DI engine at high load on pure ethanol may resolve these issues and deliver exceptional output, thereby enabling a more effective use of ethanol as a renewable fuel.

This study is directed at the experimental evaluation of SI ICE specific output when fuelled with pure ethanol, and is intended to serve both dedicated and multiple fuelling strategies. In particular the multi-cylinder test engine is one incorporating both turbo-charging and centrally mounted spray-guide direct injection, a configuration for which the literature is not extensive in regard to test data for ethanol. Also, for dedicated fuelling applications, it is anticipated that the direct injection system employed may offer an advantage in cold start performance which is broadly acknowledged to be a major technical challenge [7, 8, 10, 12, 13].

PROPERTIES OF ETHANOL

The properties of anhydrous ethanol are presented for reference in Table 1, with values for gasoline as a basis for comparison [8-10, 12, 13, 28, 29].

Property	Gasoline	Ethanol
Molecular Formula	~$CH_{1.85}$	C_2H_5OH
C (% mass)	~86.6	52.1
O (% mass)	0	34.7
Density at 20°C (kg/l)	~0.74	0.79
Stoichiometric AFR (:1)	~14.6	9.0
Lower Heating Value (MJ/kg)	~43.5	26.8
Lower Heating Value (MJ/l)	~32.2	21.2
Lower Heating Value (MJ/kg/AFR_s)	~2.90	2.98
CO_2 (g/MJ)	~72.9	71.3
Boiling Point (°C)	25~200	78.4
Latent Heat of Vaporisation (kJ/kg)	~300	855
Research Octane Number	91~98	~110
Ignition Temperature (°C)	~300	420

Table 1. Properties of Ethanol

The higher oxygen content of ethanol yields a lower stoichiometric AFR and heating value. For a DI engine, comparison between fuels may be made against a given mass of air, as fuel and air delivery occur somewhat independently and performance may be considered at first as a function of airflow. On this basis and for a given mass of air, it is seen that ethanol gives a nominally greater energy input although at a significantly higher flow rate. Irrespective of other factors it may therefore be expected that ethanol will deliver a similar engine output in relation to gasoline. In addition, for the same energy flow rate ethanol will produce lower emissions of CO_2, where other carbon containing exhaust emissions are assumed to be nominal.

The challenge for cold start of ethanol is apparent from the singular boiling point and relatively high heat of vaporisation. The former indicates a far lower vapour pressure at typical ambient conditions, and the latter a greater heat input to transition from the liquid to vapour phase (compounded by the low stoichiometric AFR).

The opportunity for extension of specific output and compression ratio is apparent from the reported Research Octane Number and relatively high heat of vaporisation. Tendency to knock is by definition less, and in particular the charge cooling on transition to vapour phase will be substantially higher for ethanol due to both low AFR and high heat requirement per unit mass. The potential for maximum charge cooling exists for a spray-guide DI fuelling system as on intake stroke injection there is anticipated to be minimal contact of the spray with chamber surfaces.

Figure 1 illustrates simply the relative reductions in charge temperature for gasoline and ethanol, displayed in terms of timing for liquid to vapour phase transition, and assuming all heat is taken from the charge. For vaporisation before intake valve closure, the transition is assumed to occur at constant pressure. For vaporisation after intake valve closure, the transition is assumed to occur at constant volume. It may be observed that direct injection of ethanol will result in an initial charge temperature reduction significantly exceeding that of gasoline.

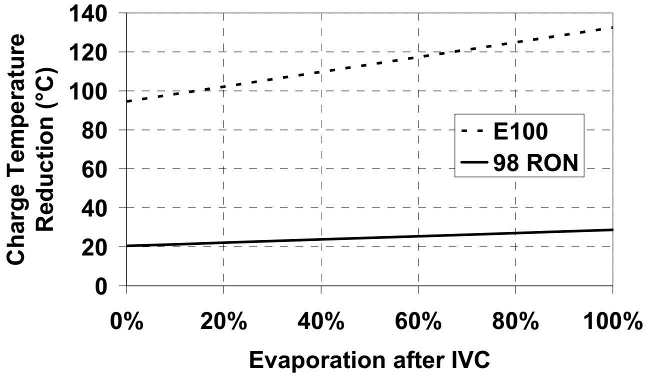

Figure 1. Charge Temperature Reduction

The ignition temperature of ethanol is somewhat higher than gasoline, suggesting that auto-ignition may also present less of a constraint to higher specific output.

EXPERIMENTAL SET UP

DIRECT INJECTION

The direct injector used is a type known in the market, and is alternately described as dual-fluid or air-assisted. With reference to Figure 2, fuel is first metered into a pre-chamber through a conventional automotive port injector, and then delivered into the combustion chamber with the assistance of air at pressure. The air-assist injector is known to deliver fuel with SMD in the order of 10 μm and less [26, 27].

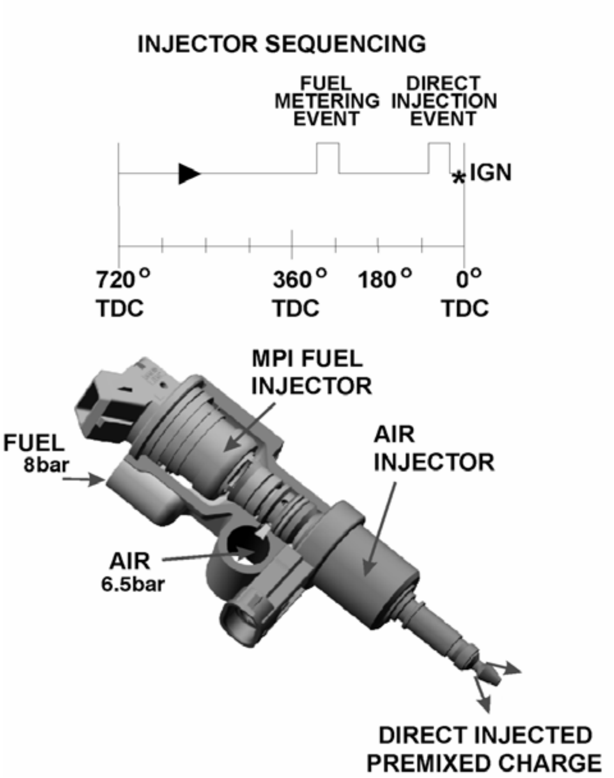

Figure 2. Schematic of Air-assist Direct Injector

Figure 3. Production 4 Stroke Direct Injector

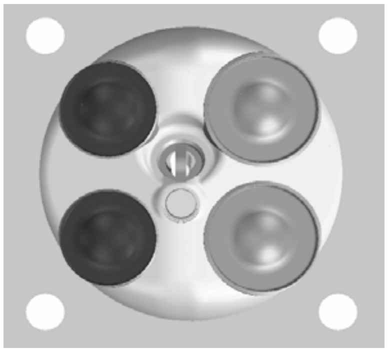

Figure 4. Combustion Chamber Layout

In this instance a high flow variant from a production 4 stroke application was utilised (Figure 3), and can be distinguished by the absence of a flow-profiling projection which typically identifies components used in 2 stroke applications. The injector is centrally mounted in close proximity to the spark plug, as illustrated in Figure 4, and the system is of spray-guide type and so not primarily reliant on chamber wall or charge motion to prepare the fuel.

The air-assist injector decouples the fuel metering and delivery events, thereby assisting the dynamic range of the injector and making it suited to boosted applications [25], particularly for ethanol which requires significantly higher fuel flow rates than gasoline. The low SMD is understood to offer advantage in low temperature starting for low volatility fuels [26], and is therefore proposed to offer benefit for low temperature starting of ethanol.

ENGINE

The engine selected for the study is an in-line 4 cylinder turbo-charged unit with dual overhead camshafts. Further details are shown in Table 2. Whilst the engine is intended for boosted operation on gasoline, it was not designed for unusual specific output and therefore load was limited under certain conditions.

Engine	In-line 4 Cylinder	
Bore	86 mm	
Stroke	86 mm	
Displacement	500 cc	
Number of valves	4	
Compression ratio	10.4:1	
Fuel System	Air-assist Direct Injection	
Pressure Charging	Garrett M53	
Valve Events (ATDC$_f$)	Low Overlap	High Overlap
EVO	139	159
EVC	375	395
IVO	351	331
IVC	582	562

Table 2. Test Engine Configuration

TEST SCHEDULE

The test program and reporting of results has been divided into three main sections, each involving an experimental comparison of gasoline and ethanol fuelled engine operation as follows.

1. Baseline Performance Evaluation:
   comparison at typical boosted gasoline engine output levels
2. Extended Performance Evaluation:
   comparison of output potential at low and higher engine speeds
3. Cold Start and Light-Off Evaluation:
   initial comparison at 25°C

Gasoline of 98 RON premium grade and anhydrous ethanol were used in these comparative tests.

RESULTS

BASELINE PERFORMANCE EVALUATION

Ignition Timing & Boost Pressure

Comparative data are presented at lambda 1, moderate boost pressure and a speed of 2000 rpm to illustrate the typical trends observed for ethanol and gasoline. At this and all other higher load conditions the throttle was held fully open and engine airflow controlled by wastegate duty.

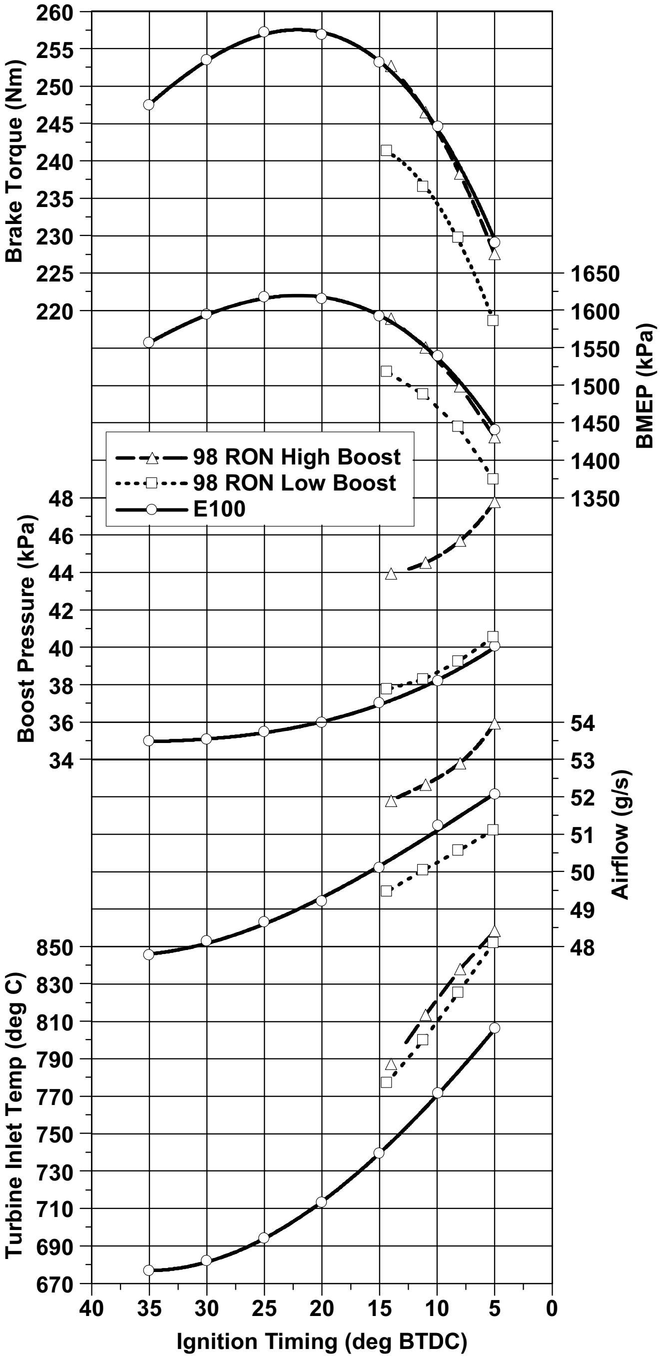

Figure 5. Engine Output, Boost Pressure, Airflow and Exhaust Temperature at 2000 rpm with Moderate Boost

Figure 5 illustrates the effect of variation in ignition timing for ethanol and gasoline with the latter at two boost pressure levels, one as for ethanol and the other set to achieve similar engine output. Ignition timing for gasoline is shown at DBL-1°CA at low boost pressure, and DBL at higher boost. The absence of knock restriction is evident for ethanol and no knock was observed for ethanol at any speed or load, hence MBT ignition timing was employed throughout.

A similar boost pressure and ignition timing yields higher output for ethanol, in the order of 5% at this condition. This may be in part attributed to an air mass flow improvement achieved through increased charge cooling for ethanol during the intake stroke. Exhaust gas temperature at the turbine inlet is observed to be approximately 50°C lower for ethanol, due in part to reduced initial charge temperature and the higher heat capacity of ethanol combustion products. Lower exhaust temperatures promote reduced heat loss, thereby increasing efficiency and contributing to the higher output.

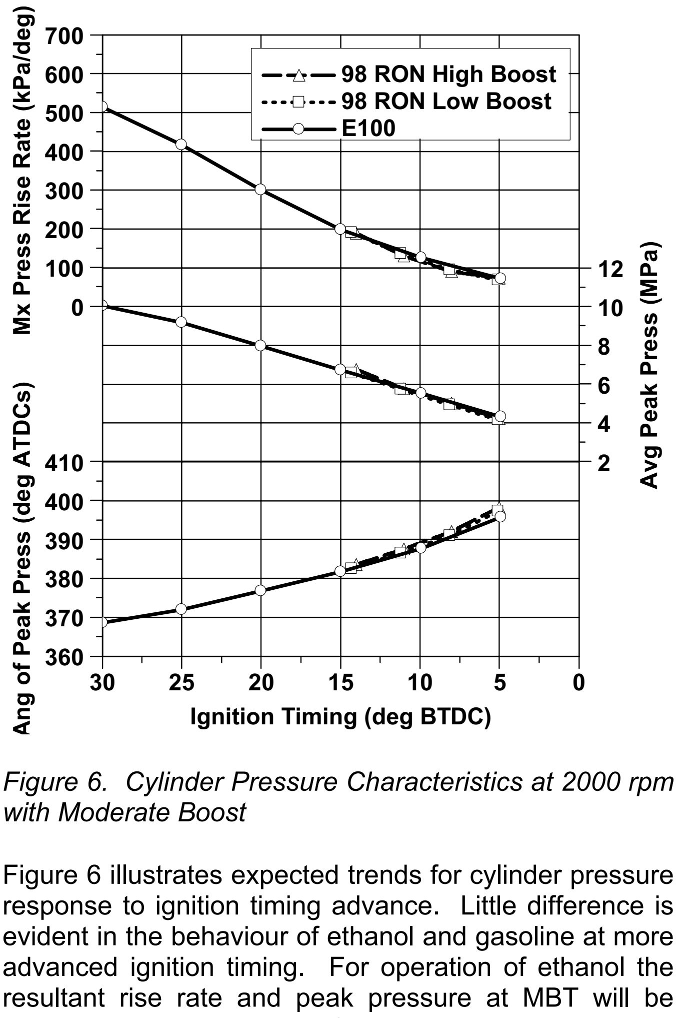

Figure 6. Cylinder Pressure Characteristics at 2000 rpm with Moderate Boost

Figure 6 illustrates expected trends for cylinder pressure response to ignition timing advance. Little difference is evident in the behaviour of ethanol and gasoline at more advanced ignition timing. For operation of ethanol the resultant rise rate and peak pressure at MBT will be higher than that observed for gasoline at DBL or later ignition timings.

Figure 7 shows further detail of combustion behaviour. At similar engine output and ignition timing the burn rates of gasoline and ethanol from 10 to 90% MFB are similar, although rate quickens for gasoline at the onset of knock. Combustion phasing is however more advanced for ethanol at the same ignition timing at 10 and 50% MFB, indicating that ignition delay is reduced relative to gasoline. This advance in combustion phase contributes to the increase in efficiency, the reduction of exhaust temperature and higher output at the same ignition timing.

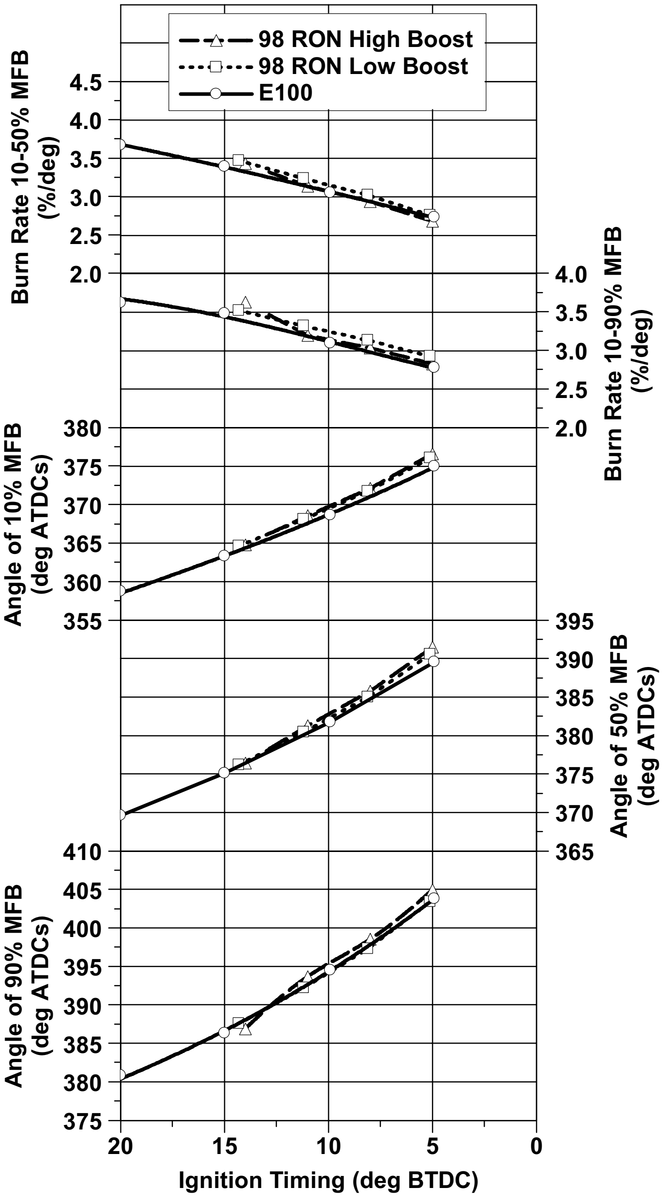

Figure 7. Burn Data at 2000 rpm with Moderate Boost

Figure 8 illustrates the trends observed for fuel consumption and associated emissions of CO_2 as determined from carbon flow rate of the fuel. Clearly the fuel flow rate for ethanol is significantly higher due to lower energy density. When expressed in terms of fuel energy flow it is apparent that ethanol requires a lower input to achieve the same engine output, which is reflected in the higher brake efficiency. In consequence the level of CO_2 emitted at a given condition is seen to fall by approximately 4%. For operation of ethanol the resultant efficiency and CO_2 emissions at MBT will be more favourable than those observed for gasoline at DBL or later ignition timings.

From trends illustrated at this single point, it may be expected that at a given engine output ethanol may demonstrate:

- more advanced ignition timing
- reduced boost and airflow requirement
- reduced exhaust gas temperature
- higher efficiency and lower emissions of CO_2

This will be evaluated in the following section.

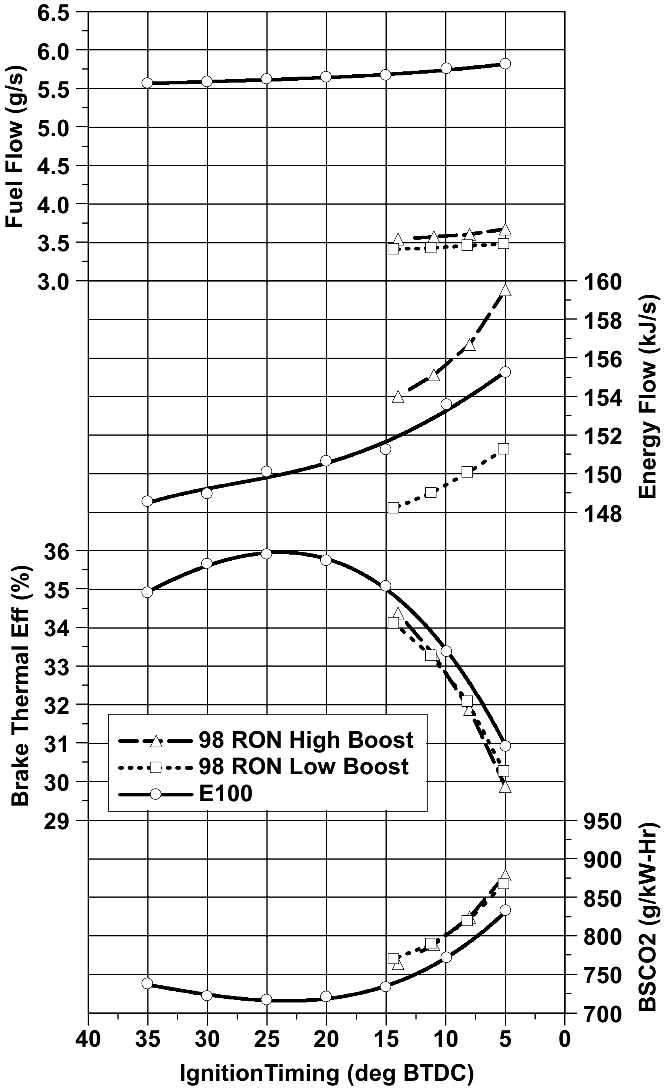

Figure 8. Efficiencies at 2000 rpm with Moderate Boost

Comparison at Target Torque

The target full load torque curve for the base turbocharged gasoline DI engine is used as a datum to establish comparative performance of the fuels. Data are presented for ethanol and gasoline with calibration optimized to achieve the target torque:

- throttle wide open
- airflow determined by wastegate duty
- lambda 1 unless enrichment required to limit turbine inlet temperature to 980°C
- ignition at MBT, or at DBL -1°CA
- start of injection to deliver maximum torque, and typically between 260 and 360°CA BTDC$_f$

Valve timing is set to either a low or high overlap condition. At higher engine speeds where exhaust pressure is higher than inlet pressure, a low overlap condition may be used to minimize residual trapping. At lower engine speeds where inlet pressure is higher than exhaust, a higher overlap condition may be used to increase engine output. In this instance high overlap was employed up to 2000rpm, and low overlap from 2000 rpm.

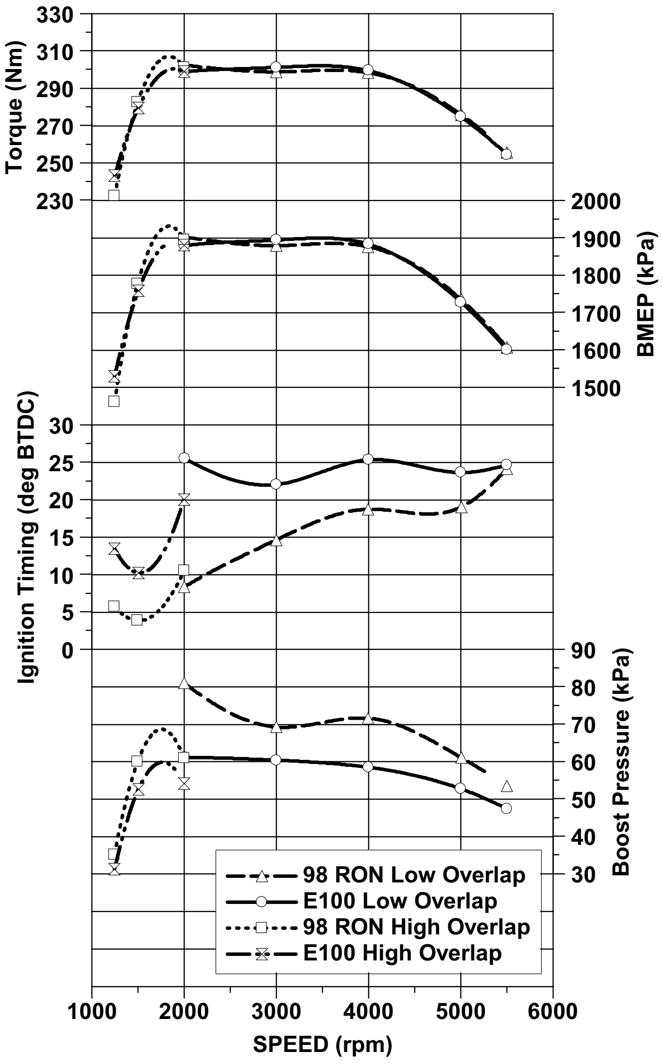

Figure 9. Engine Output, Ignition Timing and Boost Pressure at Target Torque

Figure 9 shows basic parameters at the target torque level, and shows the same torque and BMEP achieved with ethanol and gasoline. Ignition timing is held at MBT for ethanol throughout the speed range, whilst for gasoline the DBL -1°CA setting is somewhat more retarded at lower engine speed. The required boost pressure is lower for ethanol due to charge cooling and improved mass flow, and also increased brake efficiency requiring a lower rate of air flow for the same output.

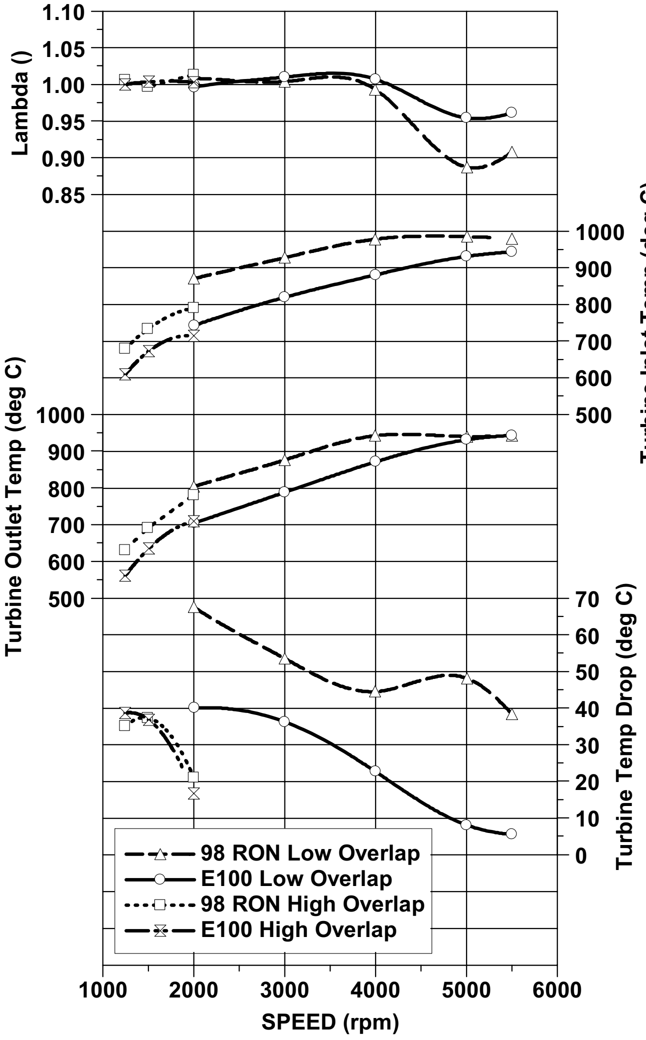

Figure 10. Lambda and EGT at Target Torque

Figure 10 illustrates lambda requirement and EGT level. At lower and medium engine speed lambda is nominally held at 1 and exhaust gas temperature for ethanol is typically 70 to 130°C lower than for gasoline, due to higher efficiencies and other factors previously discussed. At higher speed enrichment in the order of 10% is required for gasoline to limit turbine inlet temperature. For ethanol enrichment was applied to limit turbine outlet temperature, a trend not observed with gasoline. With reference to the relative temperature differential across the turbine, the lower rate of heat loss might in part be attributed to the lower EGT, but may also be indicative of a higher exotherm at the turbine. This phenomenon warrants further investigation.

Figure 11 shows trends in cylinder pressure response, which may be largely attributed to the more advanced ignition timing possible with ethanol. Clearly ethanol combustion is initiated earlier resulting in higher rate of heat release and higher peak pressures which occur earlier in the cycle. The peak pressures observed under these conditions are within typical limits for a boosted SI engine. Rates of pressure rise are somewhat higher than typical SI engine values and may present a development task for NVH and possibly component durability.

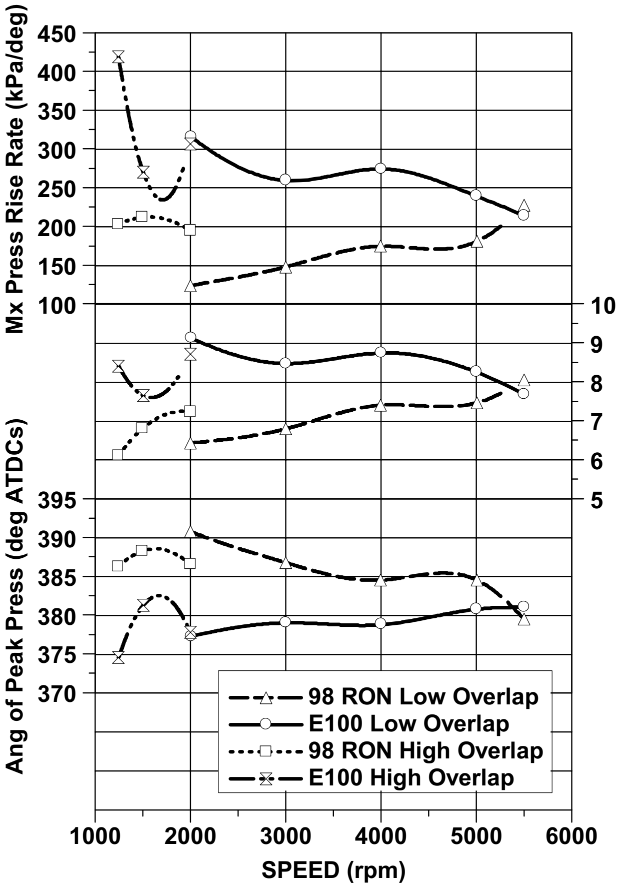

Figure 11. Cylinder Pressure Characteristics at Target Torque

Figure 12 shows resultant brake efficiency and specific CO_2 emissions for the fuels at target torque. BTE is considerably improved for ethanol by between 7 and 13%. This may be attributed to optimal phasing of combustion, reduced heat loss by virtue of lower combustion and exhaust temperature, and at higher speeds a reduced requirement for component protection enrichment. Emissions of CO_2 are commensurately reduced by between 7 and 13%.

From a comparison of the fuels at a target maximum torque representative of a boosted gasoline engine, it is apparent that ethanol delivers the same output. This is achieved at lower boost pressure and optimized ignition timing, with lower exhaust temperature, higher efficiency, and lower emissions of CO_2. Also resulting are an increase in cylinder peak pressure and rate of pressure rise.

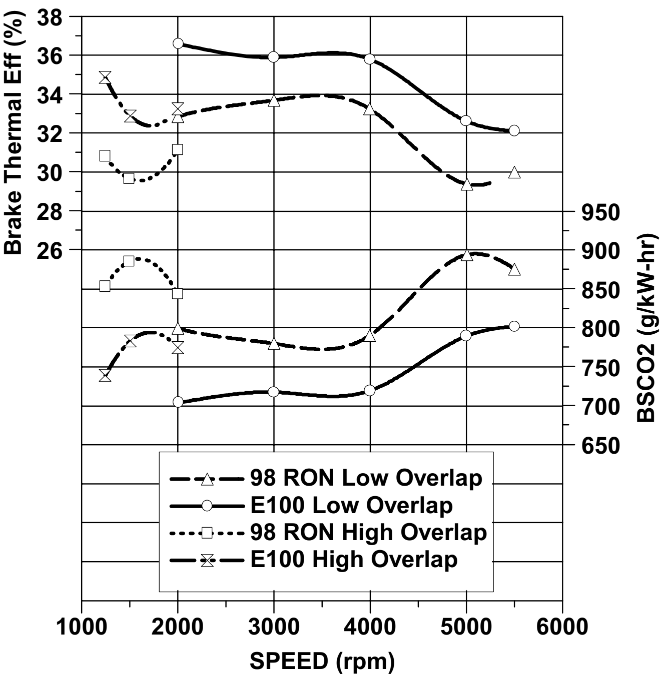

Figure 12. BTE and $BSCO_2$ at Target Torque

Summary

Comparatively high ethanol fuel consumption is mitigated to some extent at high output by the improvement in efficiency. However there remains a significant differential, and there will be a reduced improvement at part load where gasoline combustion is less knock limited. Further improvement in total engine operating efficiency may be achieved with increase of compression ratio and some degree of down-sizing enabled through increased specific output. The trends observed with ethanol indicate that both increased CR and specific output are quite feasible, and whilst the literature is clear with regard to the potential of CR [7, 8, 10, 11], the specific output achievable with a turbo-charged and direct injected ethanol engine is less understood. An investigation into the potential for specific output therefore follows.

EXTENDED PERFORMANCE EVALUATION

For evaluation at target torque it was observed that ethanol exhibited a higher tendency for pre-ignition than gasoline. To address this for evaluation at higher load some detailing of the combustion chamber was undertaken. Whilst ethanol exhibits a nominally higher auto-ignition temperature than gasoline, it is suggested that differences in fuel preparation and flammability in conjunction with chamber surface temperatures contribute to this phenomenon.

The evaluation of specific output focuses initially on the potential at lower engine speeds below 2000 rpm, as this area presents the greatest challenge to an engine downsizing strategy.

Low Speed Valve Timing Optimisation

Inlet and exhaust valve timing scans were conducted to establish maximum output with gasoline at wide open throttle, and with wastegate closed. Ignition timing was held at DBL -1°CA, and injection timing to deliver maximum torque. As valve overlap increases the degree of scavenging increases and consequently the relationship between measured and actual lambda becomes variable. Determination of required fuel mass may be made in terms of predicted air trapping, and also with regard to variation in output, efficiency, stability and emission levels. In this instance fuel mass was specified at a point of leanest for best torque (LBT).

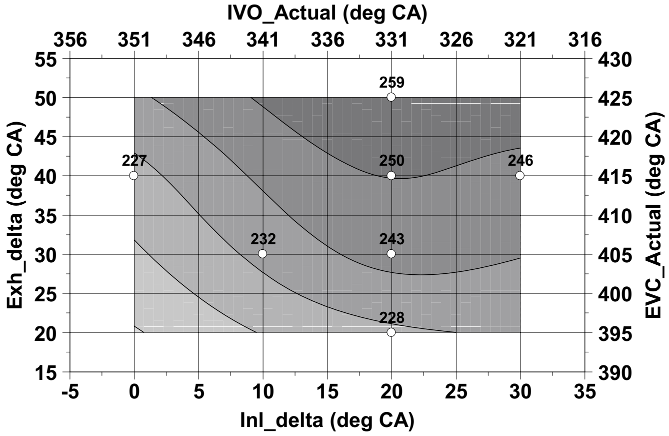

Figure 13a. Torque at 1250 rpm

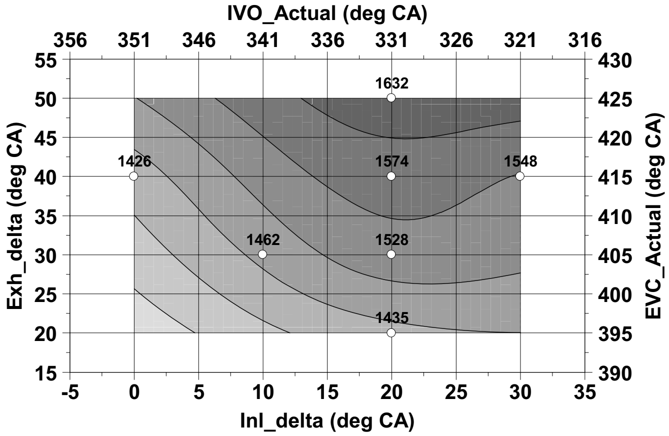

Figure 13b. BMEP at 1250 rpm

Figures 13a and 13b illustrate the rise in output with increasing overlap. At 1250 rpm maximum output of 259 Nm or 1632 kPa BMEP was developed with IVO 29° BTDC and EVC 65° ATDC.

Figures 14a and 14b show at 1500 rpm a maximum output of 354 Nm or 2224 kPa BMEP developed with IVO 29° $BTDC_{nf}$ and EVC 55° $ATDC_{nf}$.

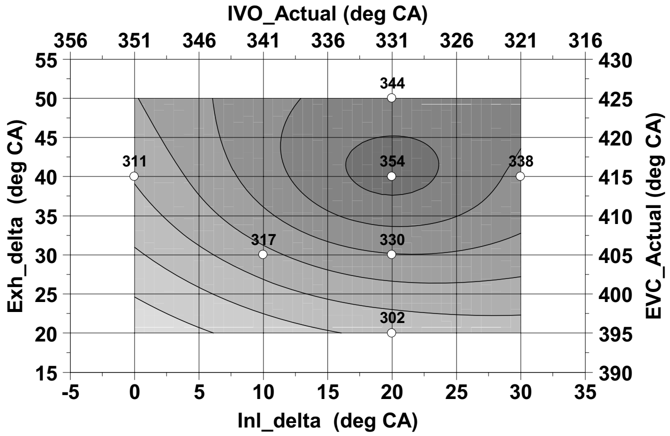

Figure 14a. Torque at 1500 rpm

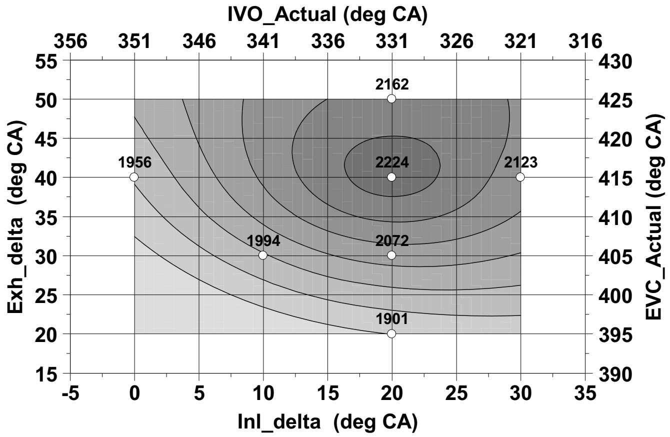

Figure 14b. BMEP at 1500 rpm

Low Speed Evaluation

Performance of ethanol was evaluated against that of gasoline at valve timing of IVO 29° $BTDC_{nf}$ and EVC 55° $ATDC_{nf}$. Ignition timing was held at MBT, and injection parameters were optimized for maximum output.

Figure 15 illustrates the comparative performance of ethanol and gasoline at 1250 and 1500 rpm. Ethanol develops in excess of 300 Nm (~1900 kPa BMEP) at 1250 rpm, and in excess of 400 Nm (~2550 kPa BMEP) at 1500 rpm. Ethanol therefore shows higher output than gasoline in the order of 50 Nm or 300 kPa BMEP. The increase in performance may be attributed to the higher boost pressure and higher efficiency observed with ethanol.

It was noted at this low speed and high boost condition that the burn rate of ethanol was reduced, resulting in combustion instability and reduced output. Other workers have identified low burn rate as an issue in highly boosted engines at low speed, and have suggested a need to actively promote charge motion in these typically quiescent chambers in such conditions [21]. In this instance an optimisation of the dual-fluid injector parameters served to elevate burn rate and achieve the demonstrated high specific output.

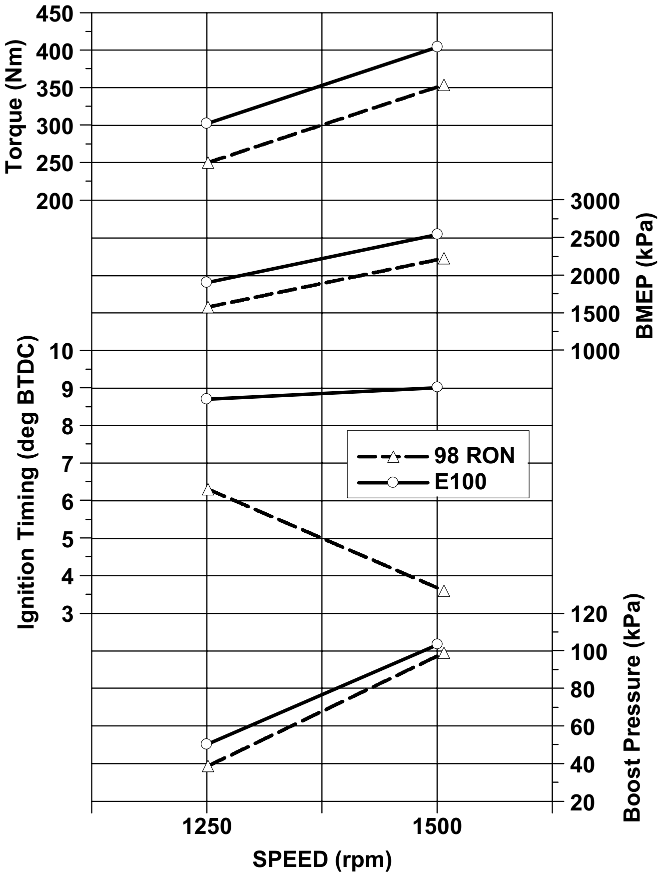

Figure 15. Optimised Performance at Low Speed

Higher Speed Evaluation

The comparative performance of ethanol and gasoline at moderate and higher engine speed is illustrated by way of data presented at 4000 rpm. In this instance target output was limited to 2300 kPa BMEP to ensure integrity of engine hardware.

Figure 16 illustrates that similar boost pressure levels yield significantly higher output for ethanol, which is reflected in values for brake specific air consumption. Also as load increases a rapid increase in knock restriction is evident for gasoline in ignition timing, resulting in only a moderate increase of cylinder peak pressure. For ethanol, ignition timing was maintained at MBT, although at highest load some retard was necessary to constrain cylinder peak pressure.

Figure 17 illustrates the limitation in exhaust gas temperature and corresponding lambda requirement. For gasoline, enrichment to lambda 0.92 is required at 300 Nm which at 340 Nm decreases to 0.82. This is necessitated by the inherently high exhaust gas temperatures of gasoline and substantially compounded by the degree of ignition retard needed at these conditions. In contrast ethanol requires no enrichment at an output of 340 Nm and light enrichment to lambda 0.98 at 360 Nm, the latter arising from ignition retard for cylinder peak pressure.

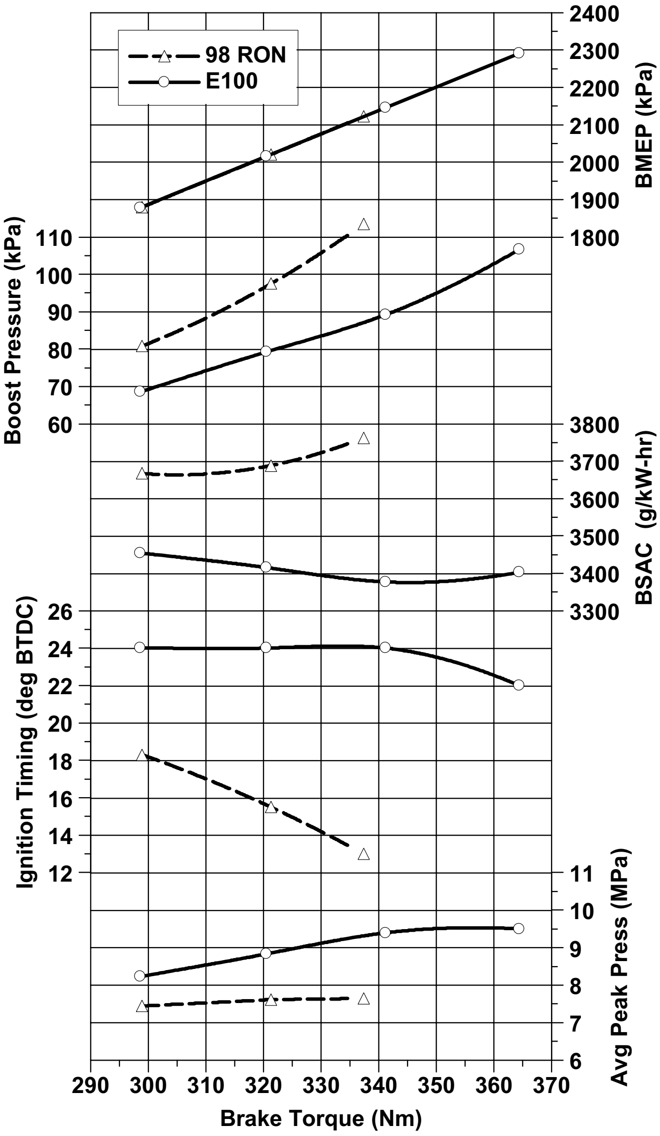

Figure 16. Performance at Higher Speed

Figure 17 also shows resultant brake thermal efficiency. For ethanol this increases with engine load from 35% to 36% until cylinder pressure restriction dictates retard of ignition and enrichment, at which point it falls back to 35%. For gasoline, efficiency decreases rapidly with engine load, falling from 30.5% to 26.5%. This may be attributed to the retard and enrichment levels required at higher load, and it should be noted that specific output is limited by this trend.

Summary

With regard to specific output, ethanol shows potential to efficiently achieve BMEP in excess of 2500 kPa BMEP across much of the engine speed range, and in particular at speeds of 1500 rpm and below. The possibility of an aggressive engine downsizing strategy is therefore apparent, with either a dual or dedicated fuel approach. In the latter case cold start of ethanol may present issues and an initial investigation of this area follows.

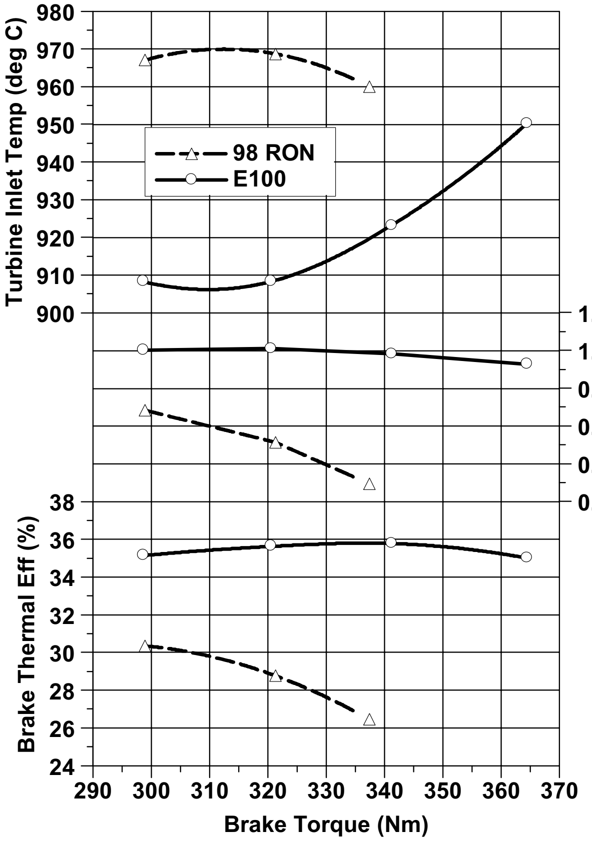

Figure 17. EGT, Lambda and BTE at Higher Speed

COLD START AND LIGHT-OFF EVALUATION

Evaluations of cold start and catalyst light-off performance for gasoline and ethanol were undertaken at ambient temperature. The calibration for gasoline was carried over from a previous program. As a starting point for the ethanol calibration, fuel delivery for gasoline was simply factored by the ratio of respective AFRs.

Starting Performance

Figure 18 illustrates a typical engine speed trace during cranking, start, flare and return to idle for gasoline at a start temperature of 25°C. Also shown are several traces recorded for ethanol with initial and subsequent calibrations. Note that these traces do not include a catalyst light-off phase with the associated higher idle speed.

It is apparent that time to first fire and engine speed rise through flare are comparable between ethanol and gasoline. Ethanol exhibited some difference at flare exit and return to idle which could be accommodated through small changes in calibration. Having established a robust start calibration a comparison of catalyst light-off capability was undertaken.

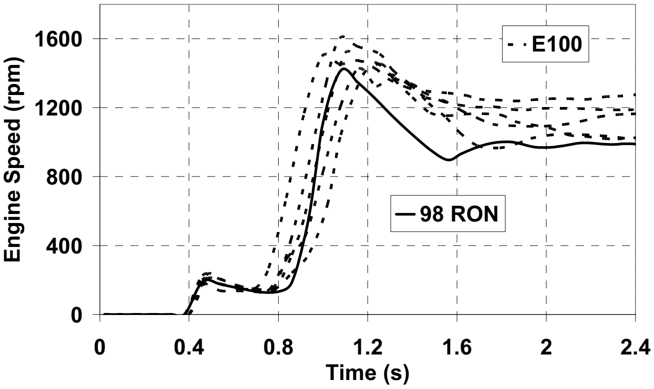

Figure 18. Engine Speed at 25°C Start

Light-Off Performance

Figure 19 illustrates exhaust gas temperatures at exhaust manifold exit (turbine inlet) following a start at 25°C and with a catalyst light-off strategy active. The strategy includes an elevated idle speed and airflow rate, coupled with significantly retarded ignition and a lean bias to fuelling levels. The calibration for ethanol was adjusted to match the performance of gasoline. Exit from the strategy may be identified by the peak and decline in exhaust temperature.

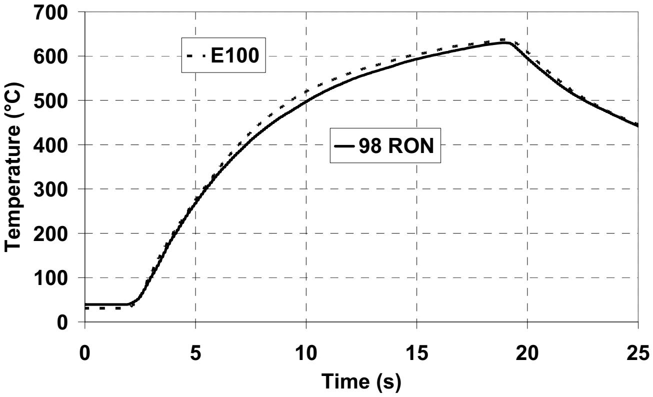

Figure 19. Turbine Inlet Temperature at 25°C Start

It is observed that exhaust temperatures for ethanol may match those of gasoline during a catalyst light-off phase. In considering that the exhaust mass flow may be nominally higher for ethanol, and that the exhaust heat capacity is higher, a similar temperature will result in a greater heating effect.

Summary

A summary evaluation of cold start and light-off performance at higher ambient temperature shows no immediate difference between ethanol and gasoline. Clearly the starting performance at lower temperatures is a requirement in the market, and further detailed work will confirm the potential in this area.

DISCUSSION

Comparative testing of ethanol and gasoline fuels has been completed on a spray-guide direct injection multi-cylinder turbo-charged engine at a compression ratio of 10.4:1. Evaluation was conducted under high load at output typical of a boosted gasoline engine, then at extended output levels, and finally for start and light-off performance at a temperature of 25°C. The key findings of the work may be summarized as follows.

1. At the same ignition timing and intake manifold pressure, ethanol demonstrated higher output which is attributed to higher airflow, more advantageous combustion phasing and lower heat loss.

2. At the same ignition timing and output, ethanol required lower airflow and boost pressure, and delivered lower exhaust temperature, higher brake efficiency and lower emissions of CO_2.

3. At all speeds and loads tested, ethanol exhibited no tendency to knock and could be operated with MBT ignition timing (unless constrained by cylinder peak pressure).

4. At target torque levels representative of a boosted gasoline engine, ethanol operated with lower boost pressure and MBT ignition timing, with lower exhaust temperature, higher efficiency, and lower emissions of CO_2. Also resulting were an increase in cylinder peak pressure and rate of pressure rise.

5. Engine output at speeds below 2000 rpm was significantly improved through optimisation of inlet and exhaust valve timing.

6. At speeds of 1250 and 1500 rpm, ethanol exhibited output 300 kPa BMEP higher than that of gasoline.

7. For higher engine speeds typified at 4000 rpm, ethanol may operate at increasing load with minimal requirement for fuelling enrichment, and with ignition timing only constrained by cylinder peak pressure. On the other hand, gasoline operates at a high level of retard and requires significant enrichment to limit exhaust gas temperature, causing a significant loss of efficiency and constraining output.

8. At a start temperature of 25°C, ethanol and gasoline show no appreciable difference in starting or light-off performance as indicated by exhaust gas temperature.

9. The design and development of an engine intended for high output operation with ethanol should address requirements for higher compression ratio, higher peak cylinder pressure, faster rate of pressure rise, reduced incidence of pre-ignition and an injection system capable of high fuel flow rate.

CONCLUSIONS

The operation of a turbo-charged spray guide DI engine on pure ethanol may achieve very high specific output with some update to engine design. The ethanol direct injection or EDI approach shows overall significant potential for aggressive engine downsizing for a dedicated or dual-fuel solution.

Further work in support of this approach is on-going:

1. The evaluation of ethanol water content on high load performance
2. Development of low temperature starting capability
3. Turbo-charger application development for transient performance and high specific output

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REFERENCES

1. Hagino, M., “Global Role of Automotive Engineers”. Planet & Society Forum, FISITA World Automotive Congress 2004.
2. United Nations (U.N.) Population Division, World Population Prospects 1950-2050: The 1996 Revision, Annex 1: Demographic Indicators (U.N., New York, 1997), pp. 11-45.
3. Chernoby, M., Hermance, D.W., Moulton, J., Oge, M.T., Smyth, J.G., Botti, J.J.M., “Hybrids vs. Diesel vs. Modified Gasoline, The Near Term - Is It Time for a Reality Check?”. SAE World Congress 2005.
4. White, T.L., “Alcohol as a Fuel for the Automobile Motor”. SAE paper number 070002.
5. Colwell, A.T., Cummings, R.E., Anderson, D.E., “Alcohol-Water Injection”. SAE paper number 450196.
6. Porter, J.C., “Alcohol as a High Octane Fuel”. SAE paper number 510086.
7. Brusstar, M.J., Mark Stuhldreher, M., David Swain, D., William M. Pidgeon, W.M., “High Efficiency and Low Emissions From a Port-Injected Engine With Neat Alcohol Fuels”. SAE paper number 2002-01-2743.
8. Jeuland, N., Montagne, X., Gautrot, X., “Potential of Ethanol as a Fuel for Dedicated Engine”. IFP International Conference proceedings 2004, pp. 7-19. ISBN 2-7108-0851-X.
9. Bromberg, L., Cohn, D.R., Heywood, J.B., “Calculations of Knock Suppression in Highly Turbocharged Gasoline/Ethanol Engines Using Direct Ethanol Injection”. Massachusetts Institute of Technology, 2006.
10. Nakata, K., Utsumi, S., “The Effect of Ethanol Fuel on a Spark Ignition Engine”. SAE paper number 2006-01-3380.
11. Baeta, J.G.C., Valle, R.M., Barros, J.E.M., de Paula, E.A.M., “Optimization Performance of Multi-Fuel Spark Ignition Engine Using a Turbocharging System”. SAE paper number 2006-01-2641.
12. Turner, J. W. G., Pearson, R. J., Holland, B., Peck, R., “Alcohol-Based Fuels in High Performance Engines”. SAE paper number 2007-01-0056.
13. Kapus, P.E., Fuerhapter, A., Fuchs, H., Fraidl, G.K., “Ethanol Direct Injection on Turbocharged SI Engines - Potential and Challenges”. SAE paper number 2007-01-1408.
14. Kremer, F., Fontes, S., “Alcohol as an Automotive Fuel – Updated Vision of the Brazilian Experience”. FISITA paper number F2004V175.
15. Papasavva, S., Weber, T., Cadle, S.H., “Tank-to-Wheels Preliminary Assessment of Advanced Powertrain and Alternative Fuel Vehicles for China”. SAE paper number 2007-01-1609.
16. Rouveirolles, P., “Well to Wheel Analysis of Future Automotive Fuels and Powertrains in the European Context : A General Overview”. FISITA paper number F2006P096.
17. Almeida, M.C., Moreira, J.R., “Life Cycle Inventory Analysis of Sugarcane Ethanol: Investigating Renewable Fuels Environmental Sustainability in Brazil”. SAE paper number 2005-01-3981.
18. Walter, S.R., “Corn Ethanol as an Alternative Fuel: Technical and Economic Issues”. SAE paper number 2006-01-0179.
19. Lecointe, B., Monnier, G., “Downsizing a Gasoline Engine Using Turbocharging With Direct Injection”. SAE paper number 2003-01-0542.
20. Kleeberg, H., Tomazic, D., Lang, O., Habermann, K., ”Future Potential and Development Methods for High Output Turbocharged Direct Injected Gasoline Engines”. SAE paper number 2006-01-0046.
21. Bandel, W., Fraidl, G.K., Kapus, P.E., Sikinger, H., Cowland, C.N., “The Turbocharged GDI Engine: Boosted Synergies for High Fuel Economy Plus Ultra-Low Emissions”. SAE paper number 2006-01-1266.
22. Lake, T., Stokes, J., Murphy, R., Osborne, R., Schamel, A., “Turbocharging Concepts for Downsized Di Gasoline Engines”. SAE paper number 2004-01-0036.
23. Duchaussoy, Y., Lefebvre, A., Bonetto, R., “Dilution Interest on Turbocharged SI Engine Combustion”. SAE paper number 2003-01-0629.
24. Cairns, A., Blaxill, H., Irlam, G.A., “Exhaust Gas Recirculation for Improved Part and Full Load Fuel Economy in a Turbocharged Gasoline Engine”. SAE paper number 2006-01-0047.
25. Cathcart, G.P, Tubb, T., Railton, D., Carlisle, H.W., “Application of Air-Assisted Direct Fuel Injection to Pressure-Charged Gasoline Engines”. SAE paper number 2002-01-0705.
26. Cathcart, G., Dickson, G., Ahern, S., “The Application of Air-Assist Direct Injection for Spark-Ignited Heavy Fuel 2-Stroke and 4-Stroke Engines”. SETC paper number 2005-32-0065.
27. Boretti, A. A., Jin, S. H., Zakis, G., Brear, M. J., Attard, W., Watson, H. C., Carlisle, H., Bryce, W., “Experimental and Theoretical Study of an Air Assisted Fuel Injector for a D.I.S.I. Engine”. SAE paper number 2007-01-1415.
28. Bosch Automotive Handbook, 6th Edition, 2004. ISBN 0-7680-1513-8.
29. Heywood, J.B., “Internal Combustion Engine Fundamentals”. McGraw-Hill Book Company, 1988. ISBN 0-07-100499-8.

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ACRONYMS & ABBREVIATIONS

98 RON: gasoline of 98 Research Octane Number

AFR: Air Fuel Ratio

AFR$_s$: Air Fuel Ratio at stoichiometry

Ang: Angle

ATDC$_f$: After Top Dead Centre (firing)

ATDC$_{nf}$: After Top Dead Centre (non-firing)

ATDC$_s$: After Top Dead Centre (scavenging = non-firing)

Avg: Average

BMEP: Brake Mean Effective Pressure

BSAC: Brake Specific Air Consumption

BSCO$_2$: Brake Specific CO_2

BTDC$_f$: Before Top Dead Centre (firing)

BTDC$_{nf}$: Before Top Dead Centre (non-firing)

BTE: Brake Thermal Efficiency

CA: Crank Angle

CO$_2$: Carbon Dioxide

CO$_2$e: Carbon Dioxide equivalent

CR: Compression Ratio

DBL: Detonation Border Line

deg: Degree

DI: Direct Injection

E100: Ethanol at 100% concentration

EDI: Ethanol Direct Injection

Eff: Efficiency

EGT: Exhaust Gas Temperature

EVC: Exhaust Valve Closing

Exh: Exhaust

GHG: Greenhouse Gas

ICE: Internal Combustion Engine

Inl: Inlet

IVO: Intake Valve Opening

LBT: Leanest for Best Torque

LHV: Lower Heating Value

MBT: Minimum advance for Best Torque

MFB: Mass Fraction Burned

Mx: Maximum

NVH: Noise, Vibration & Harshness

Press: Pressure

RON: Research Octane Number

rpm: revolutions per minute

SI: Spark Ignition

SMD: Sauter Mean Diameter

Temp: Temperature

WOT: Wide Open Throttle