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Parameter Optimization of a Turbo Charged Direct Injection Flex Fuel SI Engine

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ABSTRACT

With the increased interest in the use of ethanol as an alternative fuel to gasoline, Original Equipment Manufacturers (OEMs) have responded by adapting their current range of vehicles to be able to run on gasoline/ethanol blends. Flex fuel vehicles are defined are defined as those that are capable of running gasoline up to 100% ethanol. Other than changes to materials compatibility, to enable the required durability targets to be met when running on ethanol, very little in the way of changes are performed to take advantage of the properties of ethanol. Calibration changes are typically limited to changes in fueling requirements and ignition timing.

The physical and chemical properties of ethanol/gasoline blends offer a mixture of advantages and disadvantages. Lower energy density in the form of lower heating value reduces vehicle range whilst higher octane ratings make these excellent fuels for boosted operation.

Further enhancements can be made to flex fuel vehicles that better exploit the properties of ethanol by modifications to both the base engine and calibration. This paper describes an investigation into the potential benefits of an enhanced calibration strategy. Through the use of such strategies, it is possible to improve efficiency when running on ethanol. Design of experiments (DoE) methods have been used to investigate the behavior of a turbocharged direct injected spark-ignited engine on various blends of ethanol ranging from non-oxygenated gasoline (E0) to 85% ethanol (E85).

Enhanced wide-open throttle performance can also be realized when ethanol is combined with boosting and direct injection. Behavioral differences between gasoline and ethanol have been studied and the trade offs identified.

OVERVIEW

With desire to reduce the dependency on imported crude oil, ethanol has become a key alternative fuel particularly in the North American market. Some countries such as Brazil have adapted to running on E100 or 100% ethanol refined from sugar beets. In North America, ethanol is sold in blends up to 85% Ethanol, 15% Gasoline.

From an engineering perspective, the adaptation of an engine to run on ethanol requires design and development efforts to achieve satisfactory durability and a robust calibration. Table 1 compares the physical properties of E85 ethanol when compared to gasoline.

Fuel	Gasoline	E85
Octane RON	91-98	105
Heating Value (kJ/kg)	43105	29125
Density (kg/m^3)	598.7	648.6
Stoichiometric air/fuel	14.6	9.8
Volume % fuel in stoichiometric mixture	2	6.5
(kJ/kg) of air for stoichiometric mixture	23.3	102.5
Autoignition Temp C	257	423
Surface Ignition Temp C	899+	849
Freezing point C	-40	-114

Table 1 Physical Properties of Gasoline and E85 Ethanol

A key consideration when using ethanol blends is fuel volatility. Figure 1 shows the differences in Reid Vapor Pressure (RVP) for different ethanol content. As the percentage of ethanol increases the RVP drops, which becomes important when developing cold start calibrations. In North America E85 blends vary during the season to help with change of temperature. Winter blends are typically only 70% ethanol while summer blends are typically 80 to 83%. Discussions as to how to increase ethanol market penetration have included increasing the percentage of ethanol in standard pump gasoline to 20%. Cold start of ethanol blends remains one of the greater calibration tasks particularly with port fuel injected engines and direct injected engines at extreme cold temperatures

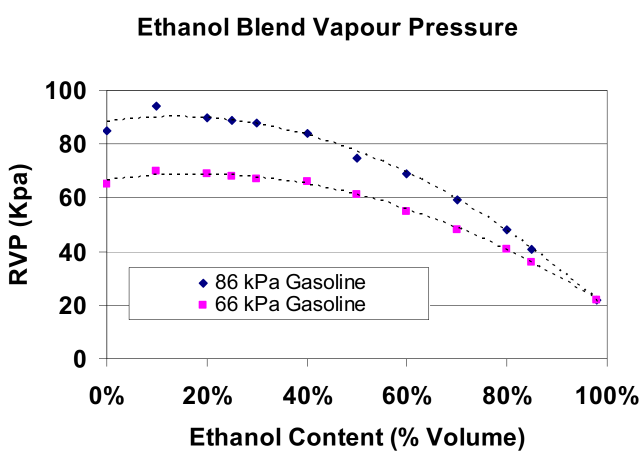

Figure 1 RVP for Ethanol/Gasoline Blends

E85 has similar density to gasoline, but a heating value that is 25 to 30% lower, which results in a significant reduction in vehicle range.

E85 is attractive as its higher octane number (RON), which whilst improving naturally aspirated engines Wide Open Throttle (WOT) performance, works exceedingly well with boosted engines. Boosted DI engines have already demonstrated great potential for improved efficiency whilst offering comparable performance to a larger naturally aspirated engine [1, 2, 3]. A boosted E85 engine has the potential to have greatly improved specific power output. With a penalty in heating value to overcome, the improved performance when the engine is run on E85 allows the possibility of further downsizing. A smaller displacement boosted E85 engine can produce the same power and torque as a larger boosted gasoline engine with the potential to reduce the difference in cycle fuel consumption as a result of the efficiency gain. Improved knock resistance also allows the potential to increase the compression ratio, helping to improve the thermodynamic efficiency. The challenge for the powertrain engine is how to achieve a balance in these attributes for an engine that has to be capable of running on both gasoline and ethanol (flex fuel capable)

Alcohol fuels require different materials to be used in several key subsystems. Lower lubricity, high oil dilution and higher cylinder pressures all influence the base engine specification. Typically to enable an engine to run on ethanol with acceptable durability the following subsystems are modified:

- Valve and seat interface
- Bore and piston rings
- Piston and cylinder head gasket
- Bearings
- Timing drive
- Seal materials
- Fuel system materials

When calibrating the current generation of FFV, the following functions are most heavily affected by having to run E85:

- Fuel Calibration
- Spark Calibration
- Torque Structure
- Ethanol Detection*

*Ethanol detection is a unique function for FFV applications.

Fueling levels change as stoichiometric ratio changes as a function of the alcohol concentration. Spark timing requirement changes due to the difference in flame speed with alcohol concentration. There are additional issues to consider with spark timing. With E85 there is a much greater propensity for pre-ignition to occur due to the lower surface ignition temperature. In some cases spark advance has to be limited even though the spark advance is not limited by knock. Spark plug fouling during cold start is also an issue on E85 often requiring compromise between spark plug heat range selection for cold start and that required to avoid pre-ignition.

The percentage of ethanol is typically calculated using one of two methods; either directly with a sensor in the vehicle fuel line that detects the change in conductivity or an indirect method that uses the oxygen sensor. The oxygen sensor method relies on the change in fuel trim that occurs immediately after the vehicle has been refueled.

E85 burns with a faster flame speed than gasoline, which should directly enhance the engines EGR tolerance at part load. Lower percentage trapped residuals result for a given overlap area when E85 is burned compared to gasoline as the C:H ratio of E85 is much lower than gasoline. Therefore it has been speculated that greater EGR rates can be tolerated on E85 than on gasoline. Most current FFV calibrations do not change the VVT or EGR calibration as a function of percentage of ethanol. One of the objectives of work reported in this paper is to quantify this effect.

OBJECTIVES

The main objective of this work was to study the combustion characteristics of a turbocharged direct injection engine running a variety of blends of ethanol and gasoline from E0 to E85. From the results obtained it is intended to quantify the efficiency gains possible and the potential calibration strategies needed.

EXPERIMENTAL DETAILS

ENGINE CONFIGURATION

The engine selected for the experimental investigation is a 2.0L LNF L850 first introduced by General Motors in 2006 in the Pontiac Solstice [4].

Type:	**Twin Turbocharged I4**
Bore X Stroke:	**85 X 86 mm**
Displacement:	**1998 cc**
Boosting System	**Twin Scroll Turbocharger**
Compression Ratio:	**9.5:1**
Firing Order:	**1-3-4-2**
Valve train:	**DOHC, 4-Valve, VVT**
Combustion System:	**Direct Injection Spark Ignited**
Power	**190 kW @ 5800 rpm**
Torque	**353 Nm @ 2000 – 5000 rpm**

The engine used for the experimental investigation represents what can be considered as a good example of a current production DI turbocharged engine. Figure 2 shows the LNF engine.

Figure 2 General Motors LNF Engine

The engine uses a wall guided combustion system with a side mounted fuel injector and camshaft driven on demand fuel pump. The solenoid type injector is a free orientated spray beam multi-hole injector. The piston crown has a feature developed to help enhance cold start performance when using split injection. Figure 3 shows the injector and piston layout.

Figure 3 Piston and Injector

Cam phasers are present on both the intake and exhaust camshafts providing 50 crank degrees of authority for each camshaft.

The turbocharger is a twin scroll unit with a maximum turbine inlet temperature of 950 °C. The engine uses an air to air charge air cooling system (Fig 4.)

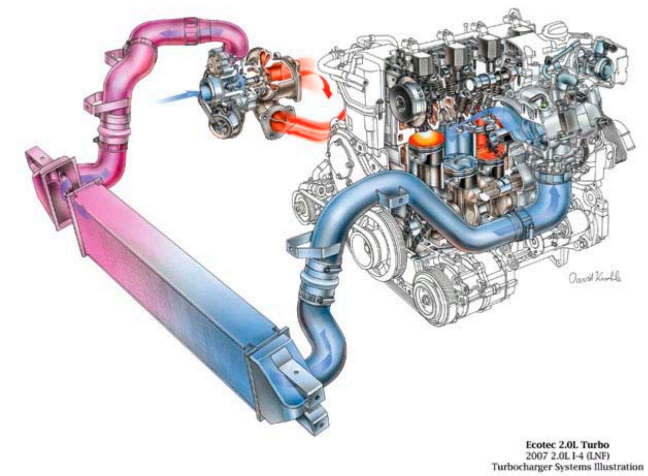

Figure 4 Inlet System Layout

The engine management system (EMS) used for this project is the Bosch DI Motronic MED9 engine control unit. It contains a 32 bit microprocessor that communicates via LAN to the vehicle.

The EMS employs a torque-based system that controls the positions of the intake and exhaust cams, throttle, and waste gate positions based on inputs from various sensors and the pedal demand of the driver. Air fuel ratio is closed-loop controlled utilizing signals from a mass-air-flow meter and a wideband lambda sensor positioned in front of the close-mounted catalyst. The EMS controls injection duration, injection timing, fuel pump delivery, fuel-rail pressure and ignition timing. A knock sensor positioned one on the side of the block is used to control knock.

In addition to the usual temperatures and pressures, the engine was instrumented for the following parameters:

- Cylinder pressure
- Engine out raw emissions
- Smoke
- Turbo speed

EMS parameters were also recorded via ASAP3 protocols.

EXPERIMENTAL APPROACH

Four fuels were specially blended for the experiment. Table 2 shows the properties of the four fuels.

	E0	E24	E55	E85
Ethanol %	0	23.17	56.44	82.8
Stoichiometric Air Fuel Ratio	14.56	13.2	11.26	10.1
Net Heating Value (MJ/kg)	43.2	38.11	34.02	29.22
Octane RON	97.2	99.8	102.1	103

Table 2 Characteristics of Test Fuels

Due to the large number of variables involved, a Design of Experiments (DoE) approach was adopted. DoE methods are effective in isolating the influence of each variable under consideration. Traditional mapping methods are not feasible given the large number of variables. Of the available advanced modeling methods, Stochastic Process Models (SPM) have been shown to be superior to the main alternatives of neural networks and radial basis functions [5]. Stochastic process models are a development of the statistical method known as Kriging. An SPM response interpolates between data points after adjustments for noise on the data. The DoE used was a global model (speed and load) in the stoichiometric region of operation with each fuel. For each fuel blend the design was an optimal Latin Hypercube design with 480 test points. The variables included in the DoE were:

- ***Engine Speed***
- ***Mass air flow***
- ***Intake cam timing (IVT)***
- ***Exhaust cam timing (EVT)***
- ***Fuel Pressure***
- ***Injection Timing***

Spark was not included as a DoE variable because a response of optimum spark was to be created during the experiment

Fully automated mapping routines were used to enhance data collection speed and improve test quality metrics

For each experiment, models of the following responses were created:

- ***BSFC***
- ***CoV of IMEP***
- ***HC***
- ***NOx***
- ***Smoke***
- ***Optimum spark***

In an additional step, a master global model was produced that combined the global models from all four fuel blends. This enabled responses to be generated as a function of ethanol percentage as well as speed and load.

The models were used to determine the optimum settings of IVT, EVT, injection timing and rail pressure for each stoichiometric ECU airflow based break point. The Bosch ECU uses Relative Load (RL) as its load parameter. RL is essentially a volumetric efficiency term derived from measured mass air flow. For each ECU map site for speed, MAF and percentage ethanol the optimization objective was minimizing fuel consumption with the following constraints:

- ***CoV of IMEP < 3%***
- ***Smoke < 0.1 FSN***

Figure 5 shows a comparison of actual test points compared to design points within the design space

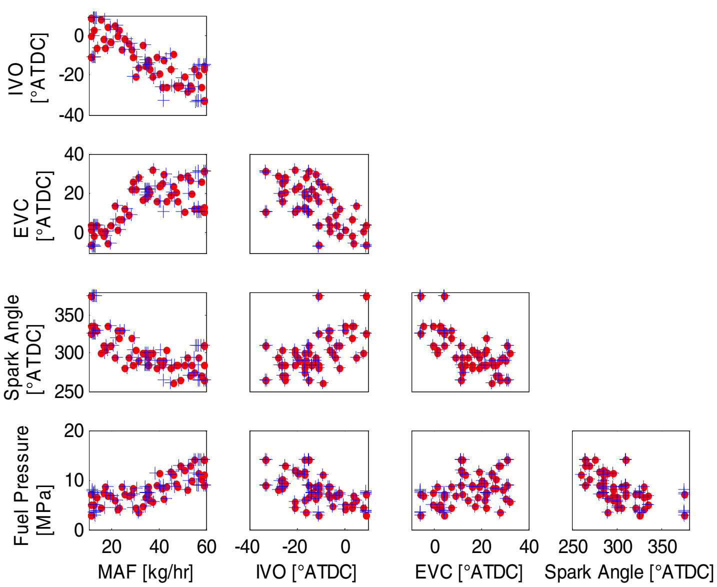

Figure 5 Test Points within Design Space at 2000 rpm

An example of the responses from the global model can be seen in the Appendix of this paper

RESULTS

PART LOAD

Response models were used to produce optimized calibration settings for E0 Gasoline. At these optimized settings, a fuel consumption map was produced, and this formed the baseline for comparison to other ethanol blends. Optimized calibration settings were established for intake cam position, exhaust position, fuel pressure, injection timing and ignition timing. Figure 6 shows the fuel consumption map.

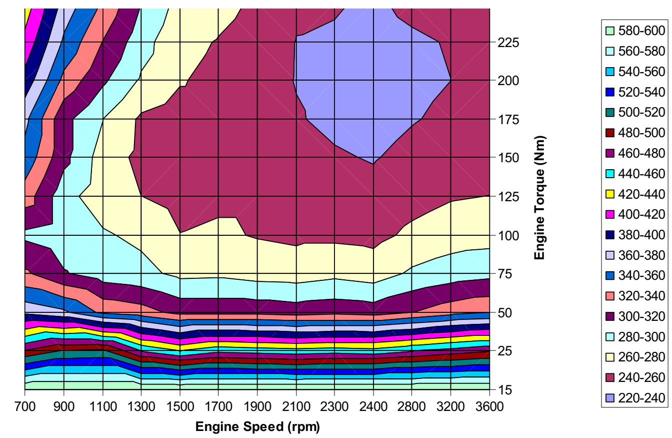

Figure 6 E0 Fuel consumption (BSFC) map

Similarly response models for E85 were used to predict fuel consumption assuming two possible calibration strategies in the stoichiometric region:

1. Using the same intake exhaust cam positions and fuel pressure as E0. Ignition and injection timing were optimized for E85.

2. Intake and exhaust cam positions, fuel pressure, injection and ignition timing optimized for E85.

Figures 6 through 13 show the calibration settings and responses at 1000 rpm in the stoichiometric region.

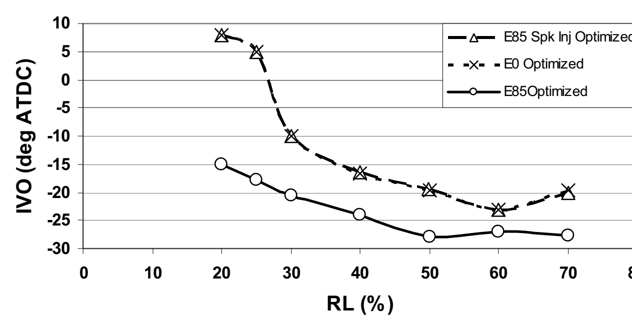

Figure 6 Intake Valve Opening Responses

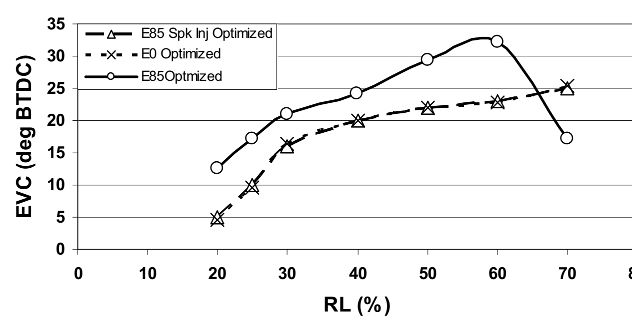

Figure 7 Exhaust Valve Closing Responses

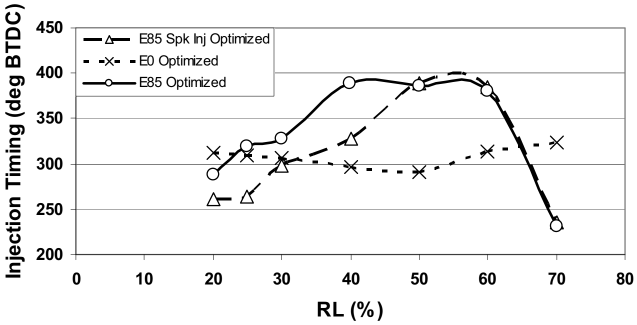

Figure 8 Injection Timing Responses

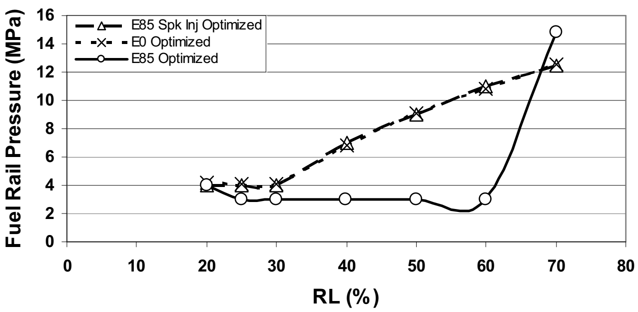

Figure 9 Fuel Rail Pressure Responses

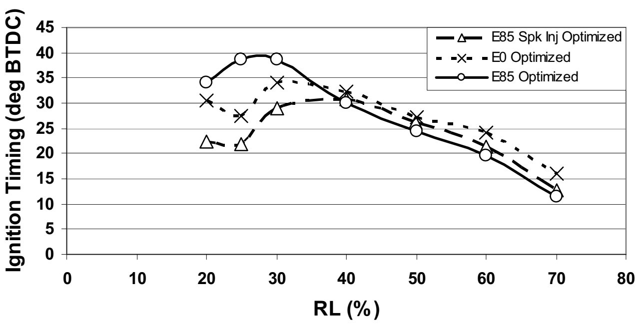

Figure 10 Ignition Timing Responses

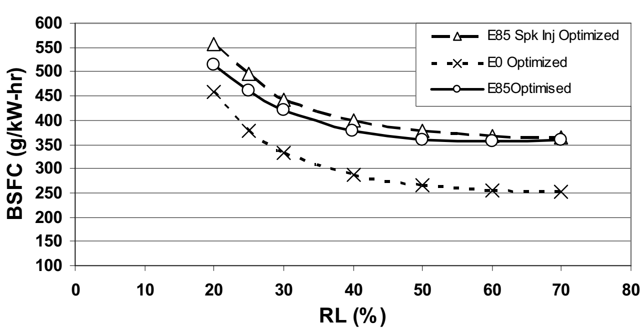

Figure 11 Brake Specific Fuel Consumption Responses

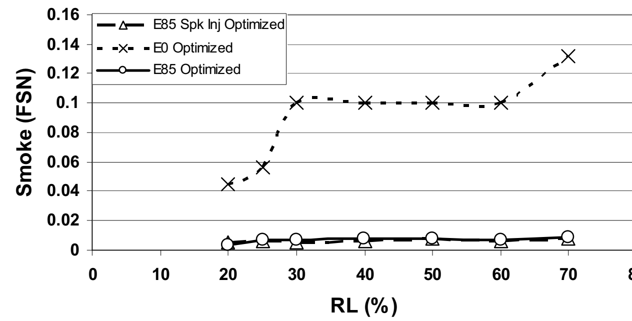

Figure 12 Smoke Responses

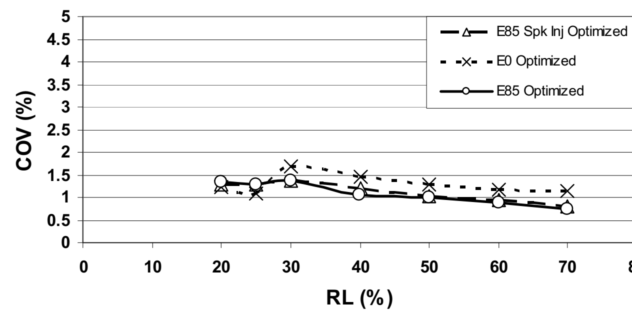

Figure 13 CoV of IMEP Responses

Through optimization of the calibration parameters on E85, improvements in fuel consumption can be realized without penalties in emissions or combustion stability. The cam timing responses in figure 6 and 7 show that the engine will tolerate much larger amounts of overlap than on E0 without degradation of combustion stability. Earlier injection timing is possible with E85 as the normal smoke limitation present when running on E0 is not present to the same degree. Lower fuel rail pressure set points when running on E85 are consistently observed at all speeds. It can be theorized that this is due to a beneficial trade off between spray penetration and earlier injection timing. The reduced smoke levels noted previously when running on ethanol is also confirmed.

Improvements noticed were largely limited at lower speeds (less than 2000 rpm) to relative loads less than 50%. As the speed increased above 2000 rpm, the brake specific fuel consumption improvement observed was negligible. After review of the response models it was observed that the ability to increase the overlap through increased intake advance was limited by the physical range of operation of the cam phasers. Ultimately intake advance and exhaust retard are limited by piston to valve clearance. Figures 14 and 15 shows the optimum settings for intake and exhaust cam positions at 3500 rpm. Figure 16 shows the fuel consumption response.

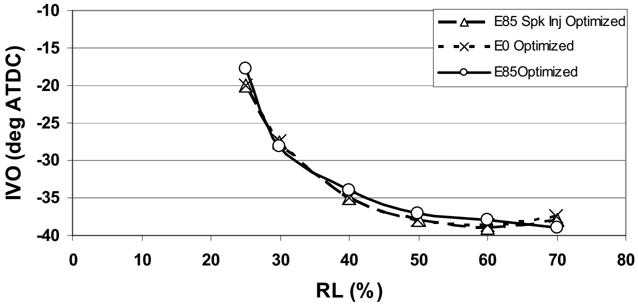

Figure 14 Intake Valve Timing Response at 3500 rpm

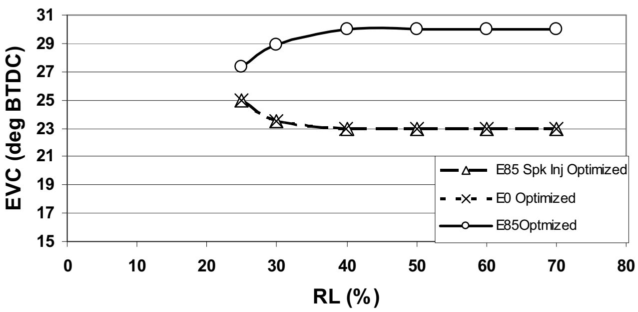

Figure 15 Exhaust Valve Timing Responses at 3500 rpm

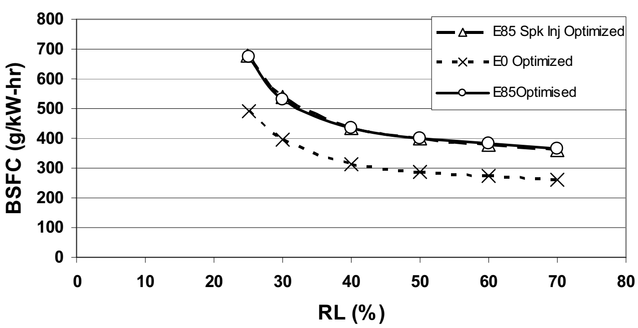

Figure 16 Brake Specific Fuel Consumption Responses at 3500rpm

One key conclusion from the response modeling is that consideration for the maximum range of cam phaser authority should be made when specifying a flex fuel engine of this type in order to exploit the potential to run with greater overlap. In many cases this may not be possible due to architectural limitations of the engine.

With fuel consumption maps developed for both potential E85 calibrations; 1. calibrations with cam positions remain the same as the E0 settings and 2. with cam positions optimized, fuel consumption maps for each calibration were developed.

By using an EASY 5 based vehicle simulation model, a comparison of both calibrations was made. With the existing hardware, results suggest that combined fuel economy figures could be improved by a further 2 to 3% over the basic E85 calibration (E0 cam positions), by using the fully optimized E85 calibration strategy (E85 cam positions)

ENGINE PERFORMANCE

Studies were performed on the test engine at WOT on different blends of ethanol from E0 to E85. The torque curve shown in figure 17 shows the torque curve for the test engine on E0 in standard production. Each subsequent fuel blend was tested at the same torque level.

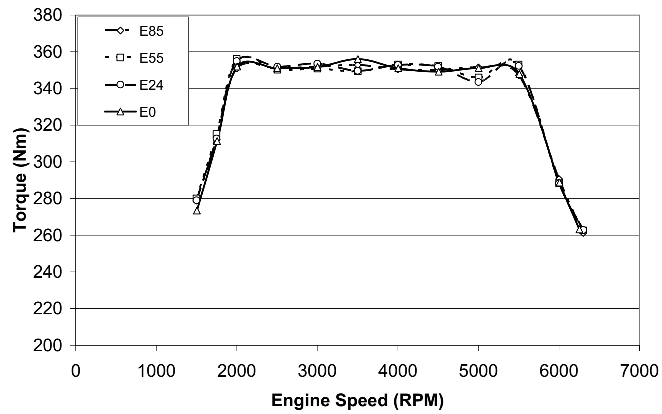

Figure 17 WOT torque curve on E0, E24, E55 and E85.

For each fuel blend, the engine was optimized such that lambda 1 was maintained until exhaust temperatures limits were exceeded. Ignition timing was set to MBT/DBL.

Figure 18 shows the lambda values that were obtained for each fuel blend. On E85, it was possible to maintain lambda one through most of the speed range. Where as on gasoline, it was only possible to run at lambda one at speeds up to 3000 rpm.

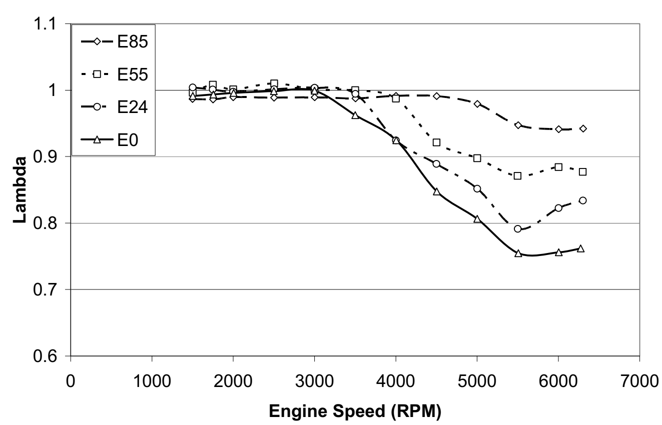

Figure 18 Lambda value for E0, E24, E55 and E85

E85 was not able to run at lambda one at higher rpm as the maximum turbine inlet temperature (Fig 19) was reached and therefore lambda had to be decreased to maintain safe operating temperatures. Higher specification turbine material and cooled exhaust manifolds would be enablers for lambda one operation over the entire engine speed range.

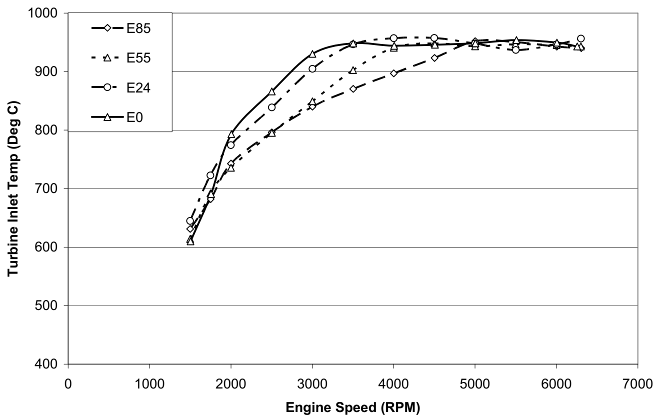

Figure 19 Turbine Inlet Temperatures

The impact of higher octane number of E85 compared to E0 can be clearly seen when comparing combustion phasing (Crank Angle of 50% Mass Fraction Burn) in figure 20. When running on E85, it is possible to maintain MBT throughout the entire speed range; whilst on E0 the engine is knock limited at all speeds.

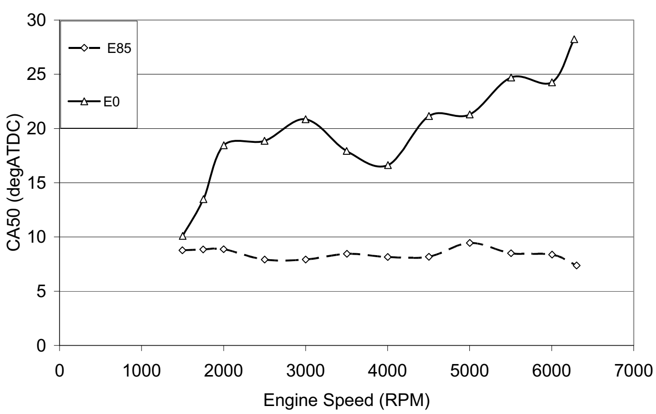

Figure 20 CA 50% Mass Fraction Burn E85 and E0.

When running under boosted conditions on E85, the resulting higher spark efficiency and faster burn rate leads to high rates of pressure rise (Fig 21) and increased maximum cylinder pressures (Fig 22). When running on E85 it was possible to reach the 12000 kPa limit set for the test program. One concern from the observed behavior is the impact on powertrain NVH that the high rate of pressure rise and maximum cylinder pressure would have. Typically rates of pressure rise would be limited to 400 kPa/CAdeg. Cylinder pressures in excess of 10000 kPa would also lead to potentially worse noise characteristics.

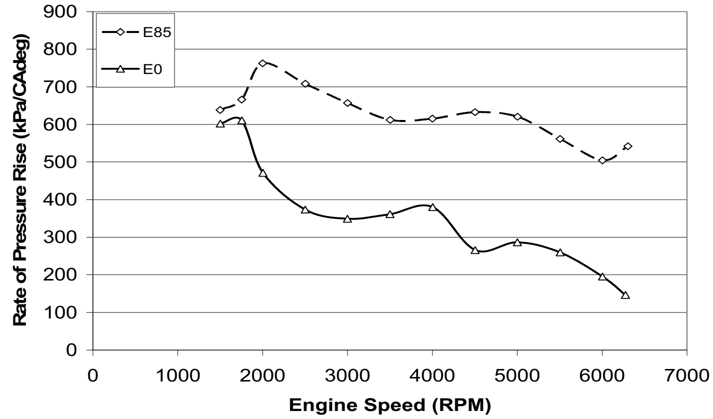

Figure 21 Maximum Rate of Pressure Rise E85 and E0

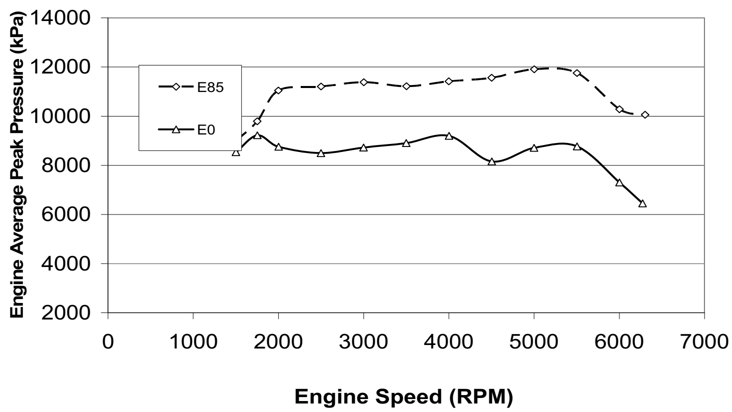

Figure 22 Maximum Peak Cylinder Pressure E85 and E0

With higher spark efficiency the engine requires less boost pressure for the same torque (Fig 23).

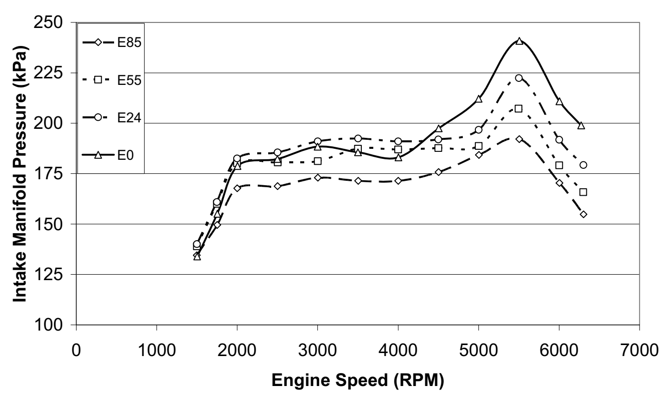

Figure 23 Manifold Pressure E85, E55, E24, E0

Figure 24 shows the difference in turbo speed that result from requiring reduced boost on the different blends from E0 through to E85.

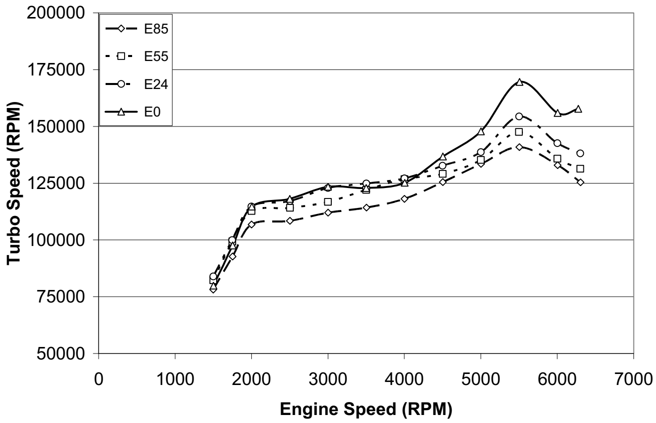

Figure 24Turbo Speed E85, E55, E24, E0

One of the advantages of using E85 is that it is possible to extend the specific rating of a boosted DI engine. The data collected during this work has highlighted the need for the base engine structure to be capable of operating at higher cylinder pressures if this capability is to be realized. As the specific rating increases, it becomes possible to increase the downsizing factor, which in turn would lead to improved efficiency.

Whilst the test engine used was limited to 12000 kPa engine simulation was used to estimate the likely level of performance that could be expected if the base engine structure was capable of higher performance.

During testing it was possible to achieve over 500 Nm with ethanol blends, but due to the base engine limitations, this was only possible with retarded spark and one air fuel ratio richer than one

Figure 25 shows simulation results of predicted performance levels on E85 at higher cylinder pressure levels. 500 Nm of torque could be expected if the base engine were capable of with standing 14000 kPa levels of cylinder pressure.

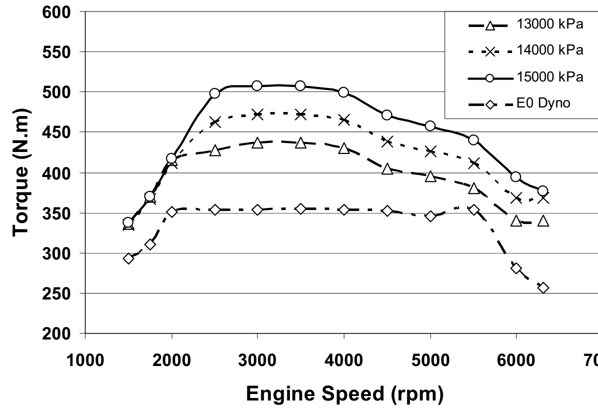

Figure 25 Predicted Performance Levels on E85

CONCLUSIONS

This study has shown that with more sophisticated calibration strategies it is possible to improve fuel consumption of a flex fuel turbo charged DI engine equipped with dual independent cam phasing by 2 to 3% over the combined drive cycle. The engine had improved dilution tolerance when running on E85 which in turn allowed for more valve overlap to be used. Physical limitations of the test hardware prevented further exploration of the ultimate limit of the dilution tolerance.

Significant improvements in WOT performance can be achieved when running on Ethanol. BMEP levels of over 30 bar can be realized if the base engine structure is designed to withstand the resulting higher cylinder pressure. Designing the base engine structure for 140 bar is recommended based on these study results. With higher specific ratings increased downsizing can be used further reducing the efficiency loss between ethanol and gasoline. Higher rates of pressure rise and higher maximum cylinder pressures could be potential NVH concerns.

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DEFINITIONS, ACRONYMS, ABBREVIATIONS

BTDC: Before top dead centre

VCT: Variable Cam Timing

WOT: Wide open throttle

DI: Direct Injection

GDI: Gasoline direct injection

IMEP: Indicated Mean Effective Pressure

BMEP: Brake Mean Effective Pressure

HP: Brake Horsepower

EMS: Engine Management System

MAF: Mass Airflow Meter

SPM: Stochastic Process Models

BSFC: Brake Specific Fuel Consumption

FSN: Filter Smoke Number

APPENDIX

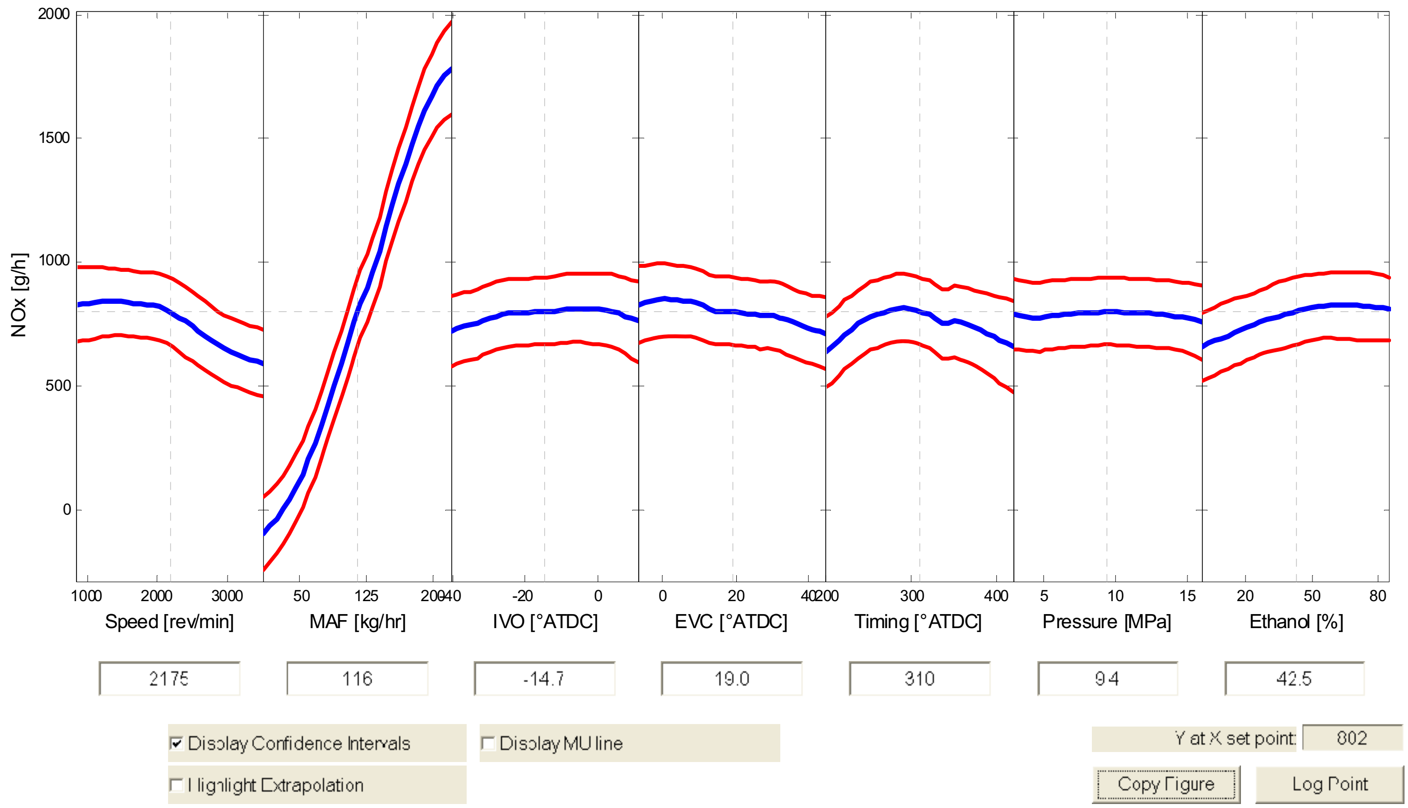

Figure 26 Example Responses from the Global DoE Model