Turbocharged Engine Operations using Knock Resistant Fuel Blends for Engine Efficiency Improvements

by

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Abstract

Engine downsizing with a turbocharger has become popular these days in automotive industries. Downsizing the engine lets the engine operate in a more efficient region, and the engine boosting compensates for the power loss accompanied by downsizing. However, the use of high boost in a downsized engine is limited by knock. Changing operating parameters such as spark timing has shown to be effective in avoiding knock. However, those strategies usually deteriorate efficiency of the engine.

Another method to suppress knock without lowering efficiency is to use knock resistant fuels. Among them ethanol has gotten a large attention due to its renewable characteristics. About 13.3 billion gallons of ethanol were produced in 2012, and about 99 % of them are used as fuel added to gasoline. However, the optimal use of ethanol in a spark ignited engine as a knock suppressing additive is not well quantified. Also, operation limitations of a knock free engine are not well known.

The objective of this project was to determine the knock onset engine operating conditions and to explore the potential of a direct injection of ethanol enhanced fuels. An engine with a turbocharger was used to measure efficiencies of the engine over the wide range of operating points. Speed range was chosen from 1500 rpm to 3000 rpm in which vehicle is usually driven in the driving cycle. Then, knock onset of different ethanol-gasoline blends, from 0 % ethanol to 85 % ethanol contents with 91 RON gasoline, were determined.

Generated engine fuel consumption maps with knock onset limits were utilized in a vehicle driving simulation tool. In a simulation, the consumption of gasoline and knock suppressing fuels was determined in different driving cycles. Finally, effects of downsizig and spark retard on ethanol fraction in the fuel were determined.

Thesis Supervisor: John B. Heywood
Title: Sun Jae Professor of Mechanical Engineering

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Acknowledgements

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I am also grateful to staff in Sloan Automotive Laboratory and fellow students for helping me in many different ways. Also, I would like to thank my friends for making me laugh in hard times.

Finally, special thanks to my parents for their support. I couldn't have done this without their support.

Abbreviations

SI : Spark Ignited
DI : Direct Injection
EGR : Exhaust Gas Recirculation
WOT : Wide Open Throttle
HC : Hydrocarbon
MEP : Mean Effective Pressure
RON : Research Octane Number
ASTM : American Society for Testing and Materials
TDC : Top Dead Center
BDC : Bottom Dead Center
NIMEP : Net Indicated Mean Effective Pressure
COV : Coefficient of Variation
CAD : Crank Angle Degree
RPM : Revolution per Minute
MAP : Manifold Air Pressure
E10 : 10 % ethanol in a gasoline
E20 : 20 % ethanol in a gasoline
E85 : 85 % ethanol in a gasoline
EW15 : 15 % water in 85 % E85
LHV : Lower Heating Value
ECU : Engine Control Unit
MBT : Maximum Brake Torque
BMEP : Brake Mean Effective Pressure
EGT : Exhaust Gas Temperature
KLSA : Knock Limited Spark Advance
KOL : Knock Onset Line
MPG : Miles per gallon
ANL : Argonne National Laboratory
DOE : Department of Energy
PSAT : Powertrain System Analysis Toolkit
MATLAB : Matrix Laboratory
UDDS : Urban Dynamometer Driving Schedule
HWFET: Highway Fuel Economy Driving Schedule
US06 : High Acceleration Supplemental Cycle
MPG : Mile Per Gallon
EPA : Environmental Protection Agency

1. Introduction

1.1 Motivation

A dominant trend in the engine industry is to use a turbocharger with a downsized spark ignited (SI) engine. This lets the engine operate in a more efficient region of the operating map, and it allows more power from a given displacement volume of the engine. However, the engine knock is a limiting factor for using a turbocharger because it limits the amount of boost of the engine. Without knock, it is possible to raise the boost level of the engine while downsizing more for efficient engine operation.

There are several different strategies to avoid knock in the engine. Direct injection (DI), mixture enrichment, exhaust gas recirculation (EGR), and spark retard are frequently used methods to suppress knock in an engine. However, traditional methods of avoiding knock often lower the efficiency of an engine. Therefore, using knock resistant fuels such as ethanol and methanol is getting a large attention. Among knock resistant fuels, ethanol has gotten a significant attention because it is a sustainable bio-fuel, and it decreases U.S. dependence on oil imports.

1.2 Background

Turbocharging

Power of the spark ignition engine is mainly limited by the amount of air that the cylinder can take in. Therefore, increasing the number of cylinders, therefore increasing displacement volume of the engine, yields higher power. Other than increasing the size of the engine, it is possible to get higher power by using forced induction methods. Therefore, for engines with the same displacement volumes, an engine with forced air induction allows higher torque than a naturally aspirated engine (an engine not utilizing forced in function of air). Turbocharging is frequently used to enable the forced induction of the air to the engine. A turbocharger consists of a turbine in the exhaust side and a compressor in the air intake side which is driven by the turbine. Then, compressed air goes through the intercooler to maintain the low temperature. By treating air as an ideal gas, the ideal gas law states

$$P_{air}V_{air} = n\tilde{R}T_{air}$$

where P_{air}, V_{air}, T_{air} are pressure, volume, and temperature of the intake air going into the engine, and $\tilde{R}$ and n are the universal gas constant and number of air molecules respectively. This equation can be rearranged in terms of ρ as following,

$$\rho = \frac{P_{air}}{RT_{air}}$$

where R is the gas constant. For constant intake air temperature, density is directly proportional to the pressure of the air. With a given displacement volume, higher density allows for more airflow into the cylinder [1]. Therefore, turbocharging allows the engine to get higher power without increasing the size of the combustion chambers.

Other than increasing power, turbocharging can increase the efficiency of an engine. For a naturally aspirated engine, the maximum efficiency is achieved with wide open throttle (WOT) operation where throttling loss is minimized. For downsized engine with turbocharging, the throttle has to be opened more to yield the same amount of torque as the naturally aspirated engine. Therefore, forced induction of charge with downsizing lets the engine operate in more efficient regions on a fuel consumption map where pumping loss is less than that of the naturally aspirated engine.

<u>Engine knock</u>

Engine knock is an atypical combustion process of a spark ignited internal combustion engine. In the internal combustion engine, a normal combustion process is initiated by spark ignition followed by compression of the fuel-air charge. The flame front of normal combustion moves completely towards the wall of the combustion chamber. In contrast to the normal combustion of the charge, knock happens as a spontaneous combustion of the pre-mixed charge in a local region of the end gas. The spontaneous combustion of the pre-mixed charge is mainly caused by localized high temperature and pressure. This spontaneous release of chemical energy causes a sharpe increase in pressure and temperature of the gas, which produces a shock wave. The shock wave produced by the unwanted spontaneous combustion generates an oscillatory sound waves which we hear as knocking.

Knock not only causes audible noises but also can cause serious damages to a combustion chamber. There are some traditional methods to avoid knock in a spark ignited gasoline engine. Since the occurrence of the engine knock is function of temperature and pressure in the cylinders, it is necessary to reduce them to suppress knock. One way to suppress knock is to lower the combustion temperature of a fuel-air charge mass, and the other is to reduce the peak pressure of the combustion process. Direct injection of the fuel cools the fuel-air mixture going into the combustion chamber, therefore lowering the overall combustion temperature which reduces the likelihoods of knock. Another way to cool the charge is to enrich the air-fuel mixture. However enrichment of the mixture cannot be done for a long period of time since it results in significant increase of hydrocarbon (HC) emissions.

Since the spark timing is closely related to the peak pressure in the combustion chamber, knock can be avoided by retarding the spark which decreases peak pressures of the combustion process. Retarding the spark timing has its own drawbacks, since retarding means that the engine is not operating at the optimal spark timing. Also, spark retard results in higher exhaust gas temperature. Since combustion process at the late spark timing is not as efficient as that at the optimal spark timing, exhaust enthalpy increases with less extraction of work. At high load, exhaust gas temperature has to be lowered by fuel enrichment to protect component such as Catalyst or turbine in case of turbocharged engine. Therefore, spark retard not only lowers the efficiency of an engine but also can cause emissions problem with fuel enrichment.

Ethanol Fuel and Research Octane Number

As stated above, traditional ways of avoiding knock have some drawbacks. Other than changing the operating parameters of the engine, alternative way is to use knock resistant fuels with higher octane rating. Among knock resistant fuels, ethanol is getting the most attention because of its sustainability and contribution to lower the external energy dependency of U.S. Ethanol from corn is the most popular biofuel in U.S. About 13.3 billion gallons of ethanol were produced in 2012, and about 99 % of them are used as fuel added to gasoline. [2] Also, more than 96 % of gasoline in U.S. contains ethanol in it. [3]

Ethanol and gasoline fuel blends were shown to be especially effective with a turbocharged spark ignition engine. [4] A downsized engine with a turbocharger has more knock problems since mean effective pressure (MEP),work output normalized by the volume, has to be higher than that of naturally aspirated engine to yield the same amount of torque. Ethanol's high latent heat of vaporization results in a higher temperature drop of the charge mass, and this improves knock onset limit and therefore efficiency of the engine [5].

Research octane number (RON) indicates knock suppressing characteristic of the fuels. There is a standardized testing procedure developed by American Society for Testing and Materials (ASTM). The specified test can be conducted to quantitatively determine the RON of fuel for the spark ignition engine. [6] Higher research octane number implies that the fuel is not prone to knock. Regular gasolines in U.S. have RON of between 91-92, and premium gasoline is rated as 95-99 RON. Pure ethanol has a RON of 109.

Optimal Spark Timing

Retarding the spark timing results in a power loss. There is an optimal timing of a spark which results in the maximum brake torque for a given charge mass. Optimal spark timing depends on both engine design and operating condition, since it is closely related to flame development and propagation. [1] Optimal spark timing mainly depends on the in-cylinder volume at the event of spark. When it is too advanced with respect to top dead center (TDC) of the expansion stroke, compression stroke work done by piston to working fluids increases. Also, high in-cylinder temperature and pressure will result in higher heat transfer loss. However retarding the spark timing too much also reduces the torque output since it reduces the peak pressure magnitude. Therefore combustion temperature will be lower while exhaust temperature is higher, which indicates a reduction in useful work. It is possible to see the relationship between spark timing and useful work output by looking at a pressure-volume plot in which the enclosing area indicates the amount of work extracted from each cylinder per cycle. Figure 1 shows four different spark timing, and the amount of useful work changes along with them. Pressure data were collected at engine speed of 1500 rpm (revolution per minute) and 126 kPa MAP (manifold air pressure.)

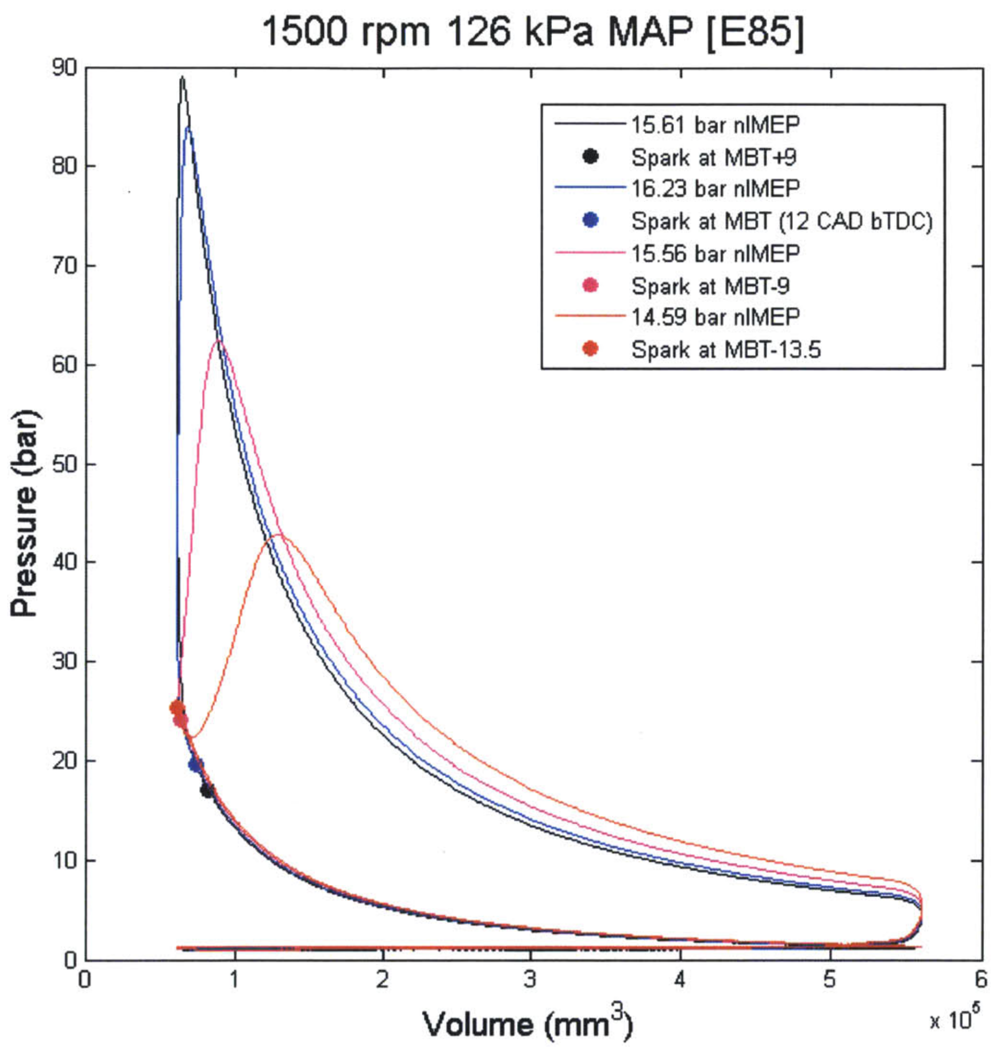

Figure 1. In Cylinder Pressure vs. Volume

The black line is the most advanced spark timing, and the red is the most retarded timing. By looking at the net indicated mean effective pressure (NIMEP), the effect of spark timing on work output is not negligible. Also, retarded spark timing can cause higher coefficient of variation (COV), which indicates the variability of combustion in cycle basis. Higher COV might cause the drivability problem.

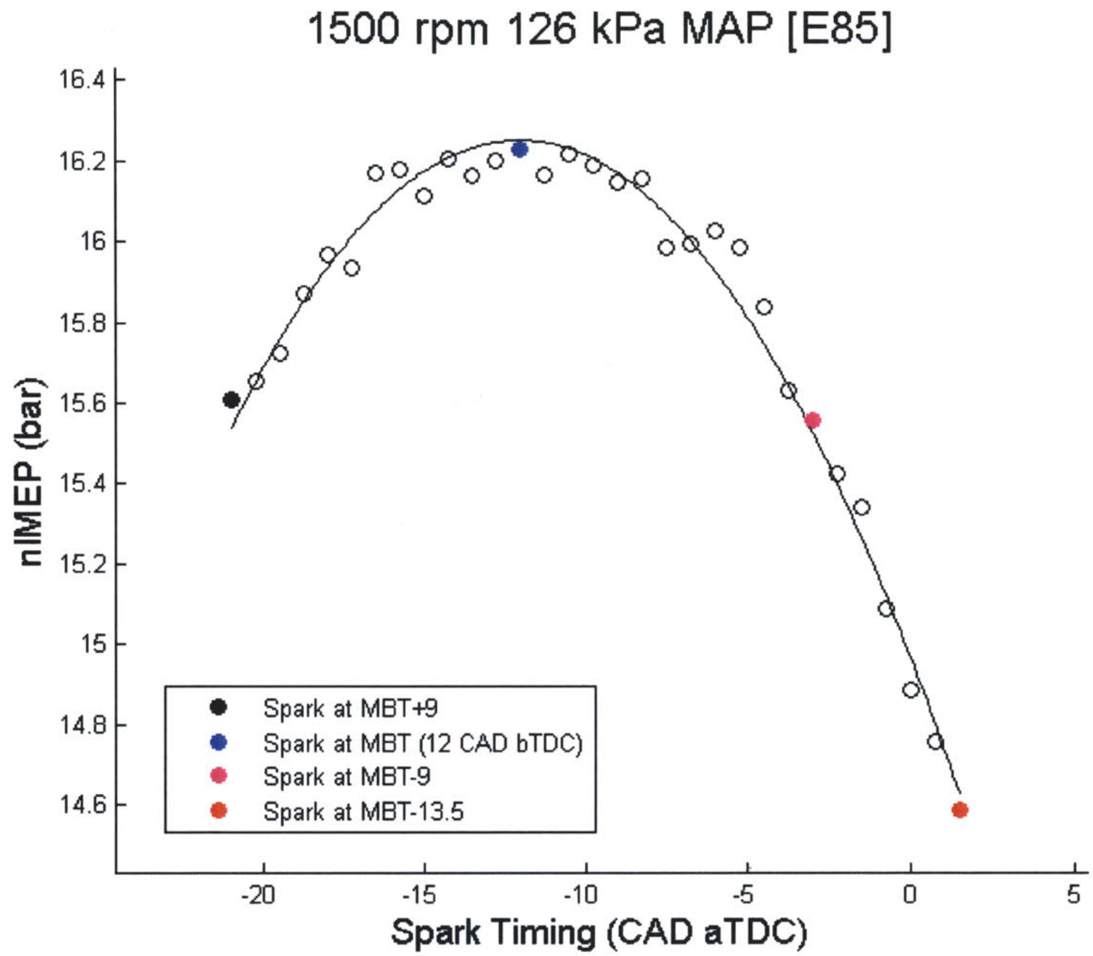

Figure 2. NIMEP vs. Spark Timing

2. Experimental Setup

2.1 Engine Specification

The engine used for the experiment is from second generation of Ecotec in GM family II engines. It was introduced in 2007 Pontiac Solstice GXP and Saturn Sky Red Line. Below is the performance and geometric specifications of the engine.

Table 1. Engine Specification

Geometry		
Cylinders	Inline 4 cylinders	
Bore	86 mm	
Stroke	86 mm	
Compression Ratio	9.2:1	
Displacement Volume	1998 cc	
Performance		
Maximum Torque	353 Nm	At engine speed range from 2500 rpm to 5250 rpm
Maximum Power	260 hp	At 5300 rpm
Boost Level	100 kPa	Additional boost to atmospheric pressure
Operating Characteristics		
Ignition	Direct Injected Spark Ignition	
Valve Timing	Dual Independent Cam Phasing	
Turbocharger	Twin scroll	
Fuel	Gasoline and blends up to E85	

2.2 Experimental setup Modifications

Engine maintenance

It was necessary to check and repair the existing 2-liter Ecotec turbocharged direct-injection engine. This engine test facility had been extensively used for several previous projects: it therefore required some repair and significant maintenance. Maintenance included repairing and rebuilding several components, replacing gaskets and certain bolts, checking the fluid circuits of the engine, and adding some parts to enable the desired experiments to be performed.

Two problems were discovered when the engine was first run. First, the pressure transducers were not giving reliable readings. As the source of this problem was unknown, the pressure transducers had to be removed, inspected and tested. It was necessary to remove the cylinder head to inspect the pressure transducers, so the engine was disassembled. After disassembling the engine, the water inside the engine was removed, and combustion deposits on the piston crowns and cylinder heads were scoured off. It was determined that the build-up of combustion deposits on thermal shields was the source of the problems with the pressure transducers. After being cleaned, they were put back into the cylinder head and the engine was reassembled. In the process of reassembling the engine, gaskets and head bolts were replaced.

The oil, water, and coolant loops were inspected and flushed out. Water that had gotten into the oil pan was removed by taking off the oil pan from the engine, and cleaning it and the oil was replaced. It was seen that the oil pan contained some water in it, which confirmed that the engine disassembly was necessary as there is a high chance that water leaked into the combustion chamber. Then, the coolant loop was filled with filtered coolant. After the engine was reassembled, the engine was hand-cranked and motored to ensure that the intake and exhaust camshafts were timed properly with the pistons. The intercooler was checked for any leakage, and the coolant heater and pump were checked to see if it properly heats up the coolants.

Since the engine's turbocharger is heated up to a high temperature, radiant and convective heat transfer were important things to consider before doing experiments in high loading conditions. To protect coolant tubes, electrical wirings, and data acquisition system close to the exhaust side of the engine, thermal shielding materials were surrounded around them. Also, some parts of the data acquisition system such as pressure amplifier were moved to the new electrical rack which was built on the cool side of the test bed. Picture 1 shows the new electric rack.

Intercooler modification

It was discovered that the air intercooler used by researchers in previous experiments was leaking, so it had to be replaced. The intercooler was originally air to air intercooler modified to accommodate air to water intercooling. It was suspected that the modified parts caused leakages or that it could not withstand the high pressure of water since it was designed for air flow. The intercooler was replaced with an air-to-water heat exchanger, which is specifically designed for air-to-water heat transfer. Then, intercooler had to be modified again to accommodate significantly increased flow rates for high load experimental conditions and high water temperatures in summer. The most recent intercooler has the core size three times that of the previous one, so it requires less water flow rates while having a higher cooling capacity. It was verified that it is capable to cool the air close to the city water temperature even at high load condition.

Fuel Tank modification

Originally, there were two 5 gallon fuel tanks installed on the test bed. One was for ethanol-gasoline blends and the other one was for pure gasoline. Since the engine was running at high load, and all four cylinders were firing, It was required to upgrade the size of the fuel tanks. Common fuel lines of two tanks that connect to the high pressure fuel pump on the engine had to be reduced in length to minimize different fuels mixing.

2.3 Measurements and data acquisition

Engine operating parameters

Engine speed was controlled by a Dynesystems Dyn-Loc IV controller which controls the eddy current dynamometer speed. Then changing the throttle and wastegate position of the engine changed the amount of the torque absorbed by dynamometer. The engine operating parameters were controlled by the ECU (Engine Control Unit) communication program called INCA. Below are the list of variables controlled by INCA

Table 2. Engine Control Variables

Control Variable	Resolution and units	Comments
Spark Timing	CAD (Crank Angle Degree) increments in 0.75	With respect to TDC compression
Intake valve opening	CAD	With respect to TDC gas exchange
Exhaust valve closing	CAD	With respect to TDC gas exchange
Throttle position	% open	
Wastegate solenoid valve actuator	Pulsewidth (%)	
Lambda set point		
Knock sensor	On/Off	
Injection timing	CAD	
Fuel rail pressure	MPa	
Fuel injection pulsewidth	ms	

The engine calibration maps and values generated by GM for my specific engine were given as reference values. Also, calibration maps for E85 were provided as well.

<u>Experimental setup controls</u>

Air is drawn through an air-filter and then goes through the damper. The damper reduces the oscillation of the intake air drawn to the engine. Then it goes through the water-air intercooler after going through the compressor to reduce the temperature of the air entering air heater. Then, air heater controls the intake air temperature before the throttle. Coolant temperature is controlled by the electric heater in the coolant tank and solenoid valve which opens and closes to control the amount of city water flowing through the heat exchanger. ETAS lambda meter was used to measure the lambda value. H/C, O/C, and air to fuel ratio of the fuel blends were calculated to put into the meter.

<u>Pressure and temperature data</u>

There are piezo resistive pressure transducers (Omega PX 219) for measuring Manifold Air Pressure (MAP), exhaust MAP, post compressor pressure, etc. Cylinder head of the engine was modified so that piezoelectric pressure sensors (Kistler 6125 A) can be fitted with flame arrestors in front. Two pressure transducers were fitted on cylinder 1 and cylinder 3. Pressure trace from cylinder 1 was sampled every quarter of the CAD, and pressure trace from cylinder 3 was sampled at 100 kHz. This was assuming that cylinder 3 knocks first since it is in the middle and hotter.

3. Fuel Test Matrix and Properties

3.1 Fuel Properties

Two certified gasolines for research were used to give two different variables of RON. RON 91 gasoline was from Gage Products, and RON 96 gasoline was from Haltermann Solutions. 200 proof anhydrous ethanol was obtained from KOPTEC, and ethanol-gasoline blend fuels were made by mixing RON 91 gasoline and 100 % ethanol by volume. E10, E20, and E85 denote 10 %, 20 %, and 85 % of ethanol by volume mixed with RON91 gasoline. EW15 denotes 15 % of distilled water mixed with 85 % of E85.

Table 3. Fuel Property Matrix

	Gasoline (RON96)	Gasoline (RON91)	E10	E20	E25	E50	E85	Ethanol	EW15
LHV (MJ/kg fuel)	43	43.27	41.76	40.27	39.53	35.91	31.03	29	27.27
Air to Fuel Ratio	14.6	14.65	14.06	13.47	13.17	11.74	9.80	9	8.61
LHV (MJ/kg mixture)	2.76	2.76	2.77	2.78	2.79	2.82	2.87	2.9	2.83
Research Octane Number [7]	96.4	90.5	95.05	98.84	100.4	105.2	108.8	109	N/A

Lower heating value (LHV_{blend}) of fuel blends can be determined by taking a weighted average of LHV of each fuel using density of gasoline and ethanol as

$$LHV_{blend} = \frac{(LHV_g * (1 - Ef) * \rho_g + LHV_e * Ef * \rho_e)}{((1 - Ef) * \rho_g + Ef * \rho_e)}$$

where LHV_g, LHV_e, ρ_g, ρ_e, and Ef indicate LHV of gasoline, ethanol, density of gasoline, ethanol, and ethanol fraction by volume. Since air to fuel ratios of gasoline and ethanol are different, air and fuel mass does not increase proportionally with increasing fuel mass. LHV_{mixrue} can be defined as LHV of fuel per unit mass of mixture as

$$LHV_{mixrue} * \left(m_{fuel} + m_{air}\right) = LHV_{fuel} * m_{fuel}$$

$$LHV_{mixrue} = \frac{LHV_{fuel}}{(1 + Air\ to\ Fuel\ Ratio)}$$

where LHV_{mixrue} of gasoline is around 2.76 MJ/kg, and $LHV_{mixrue}$of pure ethanol is 2.9 MJ/kg. Therefore, any ethanol blends should be in between those two extremes.

3.2 Research Octane Number

Properties of different gasoline are listed in a certificate of analysis of gasoline. They differ from drum to drum, but variation in property is insignificant. One of the important properties listed on the certificate of analysis is research octane number of the fuel. Specification for RON 91 and RON 96 gasoline have error bounds of ± 0.5 RON, and every drum satisfies this specification. The most representative RON of two different fuels that were used for experiments were 90.5 and 96.4 for RON91 and RON96 standard research fuels.

Several studies were conducted to determine the research octane number of different fuel blends. A paper by A. Stein shows experiments on measuring RON of different ethanol blends using standard testing procedure for RON measurement. [7] Standard procedure ASTM D2699 is specified by American Society for Testing and Materials, and it correlates antiknock properties of fuel in a SI engine.

4. Engine Performance Map

4.1 Test Matrix

An initial set of tests was designed to confirm that the engine is operating properly and the data acquisition system is reliable. Pressure data was measured at different speed and load to see if NIMEP calculated from the data is reasonable. One of the important measurements to check was fuel flow rate from engine ECU since thermal efficiency is sensitive to the fuel flow rate. Fuel flow rate was calculated by the injector calibration map provided by General Motors for previous research project on this engine. Average values of injection pulse-widths and fuel rail pressures for four cylinders were needed to interpolate them using GM's map. Finally, fuel flow rate from ECU, air flow rate measured by rotary flow meter, and lambda from oxygen sensor were compared at different loads at engine speed of 1500 rpm. Lambda calculated from airflow rate and fuel flow rate were compared to lambda measured by oxygen sensor as seen in Figure 3.

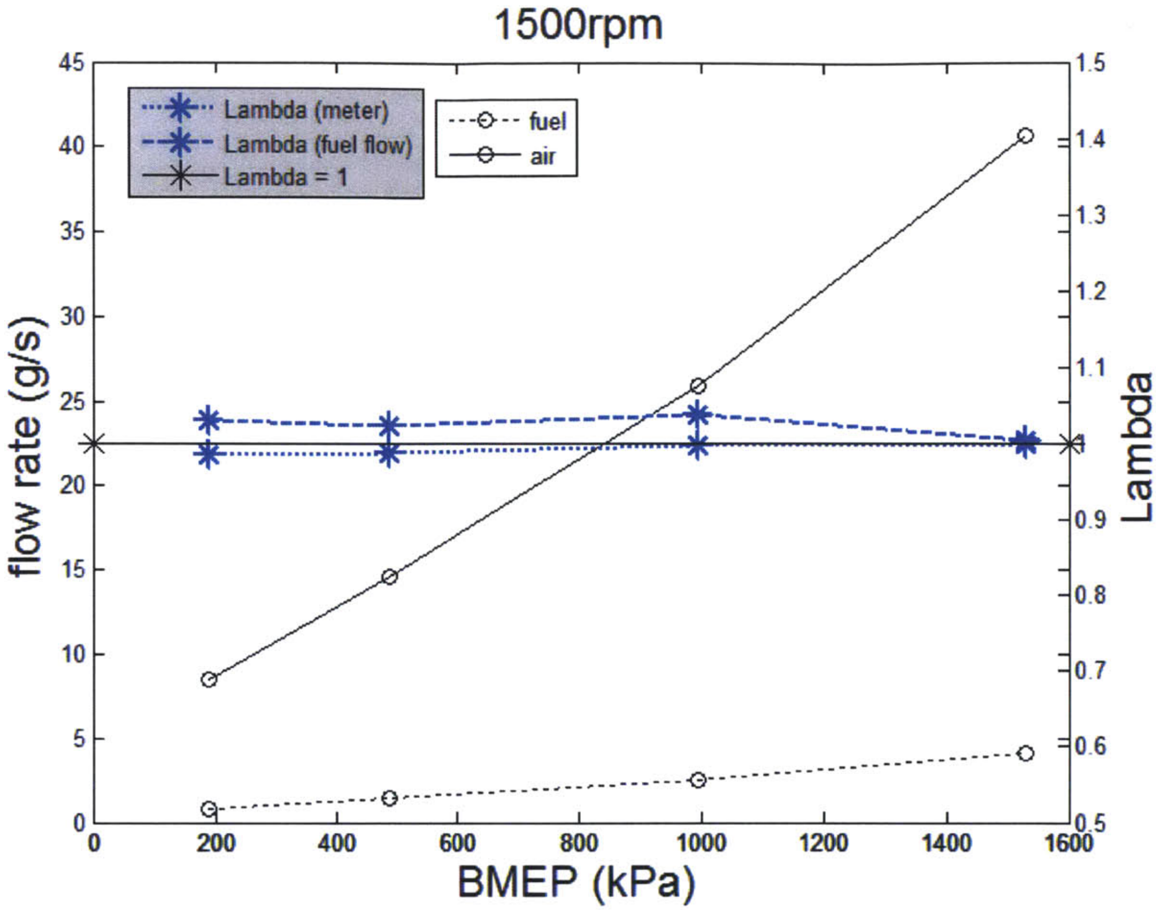

Figure 3. Fuel Flow Rate, Air Flow Rate, and Lambda

It can be seen that error between two lambda is around 3~4 %. Air flow rate measured by the rotary flow meter is in volumetric flow rate, so it had to be converted to mass flow rate by assuming a fixed value. This is a possible source of error since temperature of air changes when it flows from rotary flow meter to the intake of the cylinder. Even though there were some errors, it could be confirmed that fuel flow rate and lambda measured from oxygen sensor are well behaving. After a set of initial engine tests, the next set of tests for performance map generation were planned.

Table 4. Engine Test Matrix for Performance Map

Operating Parameter	Values
Fuel Type	E85 (RON91 base) and RON96 gasoline
Speed set points (rpm)	1500, 2000, 2500, 3000
Load in MAP (bar)	0.4, 0.7, 1.0, 1.3, 1.6, 1.8, WOT
Valve Timing	Valve Timing Map in ETAS
Lambda Set point	1.0 (Stoichiometric)
Experiments Type	Spark Sweep

A major goal for the first set of experiments was to generate the engine operating data, with E85 as fuel, over the full range of speed and loads required to produce an engine performance map. Testing points were chosen such that manifold air pressure is fixed at low, medium, and high levels, at different engine speeds levels. This effectively provides data at close to constant BMEP (Brake Mean Effective Pressure) levels, which scale closely with intake manifold air pressure as shown in Figure 4. At each testing point, maximum brake torque spark timing was determined by spark sweeps over a certain range of spark timings. However, maximum brake torque timing could not be reached at some testing points since that degree of spark advance results in higher maximum pressure than the 100 bar which is effectively our engine's upper limit. In this situation, brake torque was measured at peak pressure limit. Engine limitations will be discussed further in the next chapter.

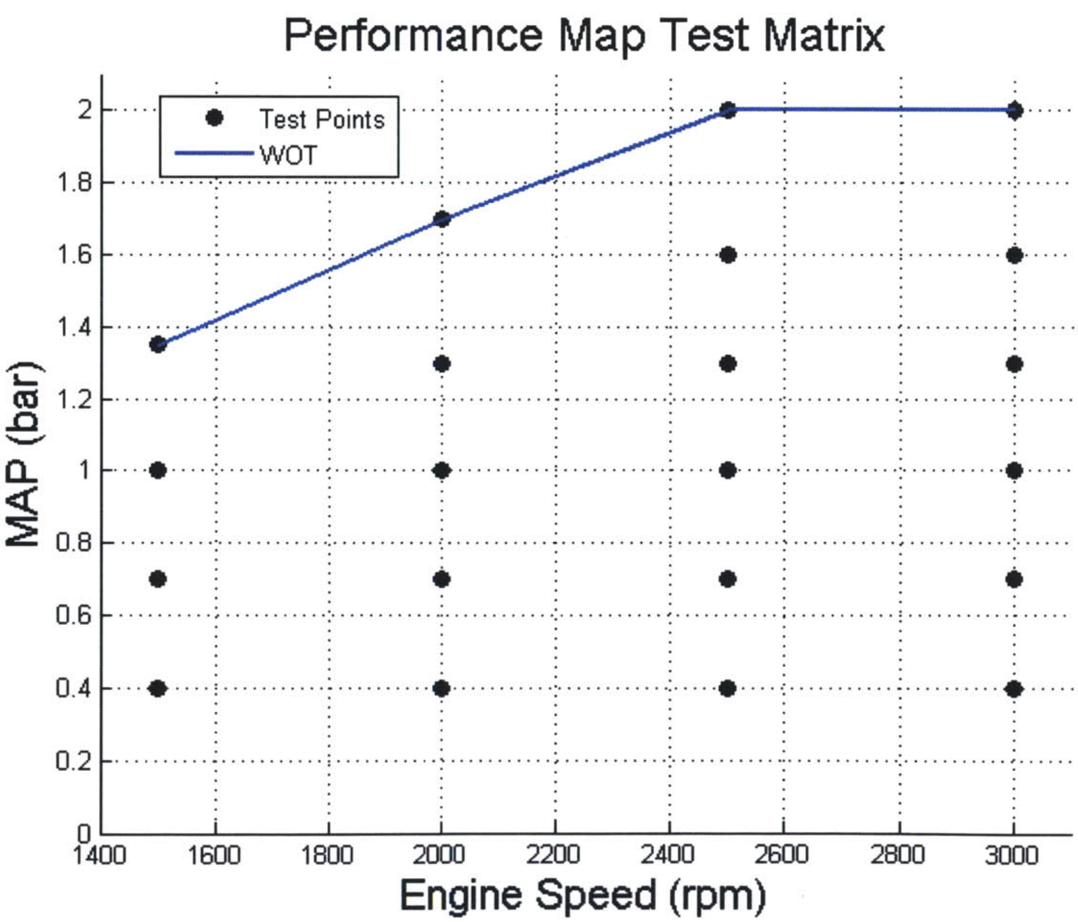

Figure 4. Performance Map Test Matrix

With maximum brake-torque timing (or maximum-pressure-limited spark timing), engine torque was measured. Then, measured fuel flow rates were used to calculate the specific fuel consumption and fuel conversion efficiency of the engine at each operating point. However, wide-open-throttle operation of the engine had to be defined since a turbocharged engine has not only a throttle but also a wastegate, which controls the boost level.

4.2 WOT line definition for turbocharged engine

There were several variables that had to be chosen while determining WOT operation of the engine at different engine speeds. With gasoline fuel, combination of spark retard and enrichment of mixture is required for component protection and knock suppression at high load operations. However, it was discovered that E85 is not limited by knock in the current compression ratio of 9.2. However, spark advance was limited by peak pressure limit of the engine. Peak pressure limit estimated by GM was that the average peak pressure added by 2 standard deviation does not exceed 110 bar. However, the limit was taken to be 100 bar for our engine to ensure it does not break. Turbine inlet exhaust gas temperature was monitored so that it does not exceed 980°C which is the physical limit for the turbine. Then, the engine was run at speeds from 1500 rpm to 3000 rpm with 100 % open throttle and 100 % close wastegate.

At each speed, throttle was gradually opened until it was wide open position. Then wastegate opens to bypass exhaust as much as possible to control the amount of boost. This engine geometry results in manifold air pressure around 130 kPa, and this configuration is called a baseboost condition. From that point, wastegate actuator is controlled by ECU to generate a higher boost. It was possible to reach manifold air pressure of 200 kPa, which is the highest boost level specified by GM. There were two important physical limitations of the engine to be considered when spark sweeps were conducted. Peak cylinder pressure pressure had to be monitored carefully when advancing the spark timing, and turbine inlet temperature had to be monitored carefully when retarding the spark timing.

4.2 Performance Map

Fuel Conversion Efficiency

At each testing point, fuel flow rate and torque output were averaged over 100 engine cycles to give a repeatable results. To interpolate the fuel consumption of the engine at different speeds and loads, specific fuel consumption and fuel conversion efficiency were calculated at each testing point. Brake specific fuel consumption is the fuel flow rate per unit power output based on the torque measurement from a dynamometer. [1] Specific fuel consumption is defined as

$$sfc = \frac{\dot{m}_{fuel}}{Power}$$

where power output is

$$Power = Torque * Engine\ Speed$$

Also, fuel conversion efficiency is

$$\eta_f = \frac{Power}{\dot{m}_{fuel} * LHV_f}$$

where LHV_f is lower heating value of the fuel. Depending on how power is calculated, fuel conversion efficiency may include friction or pumping work. If power is calculated by using torque data from the dynamometer, fuel conversion efficiency includes the losses from pumping and friction. Since efficiency and fuel consumption measures are required for driving cycle simulation, brake fuel conversion efficiency had to be used.

To obtain contours of constant fuel consumption and engine efficiency over a performance map, brake specific fuel consumption and brake fuel conversion efficiency were interpolated over the tested ranges of the BMEP and engine speeds. Mesh grids were created, and two-dimensional cubic interpolations were done to generate interpolated data sets in three-dimensions. For generation of smooth contours, the measured fuel flow rate was corrected by setting trend lines using the least squares method over the range of manifold air pressures tested. The upper boundary is set as the wide-open throttle curve: the lower boundary is engine idle.

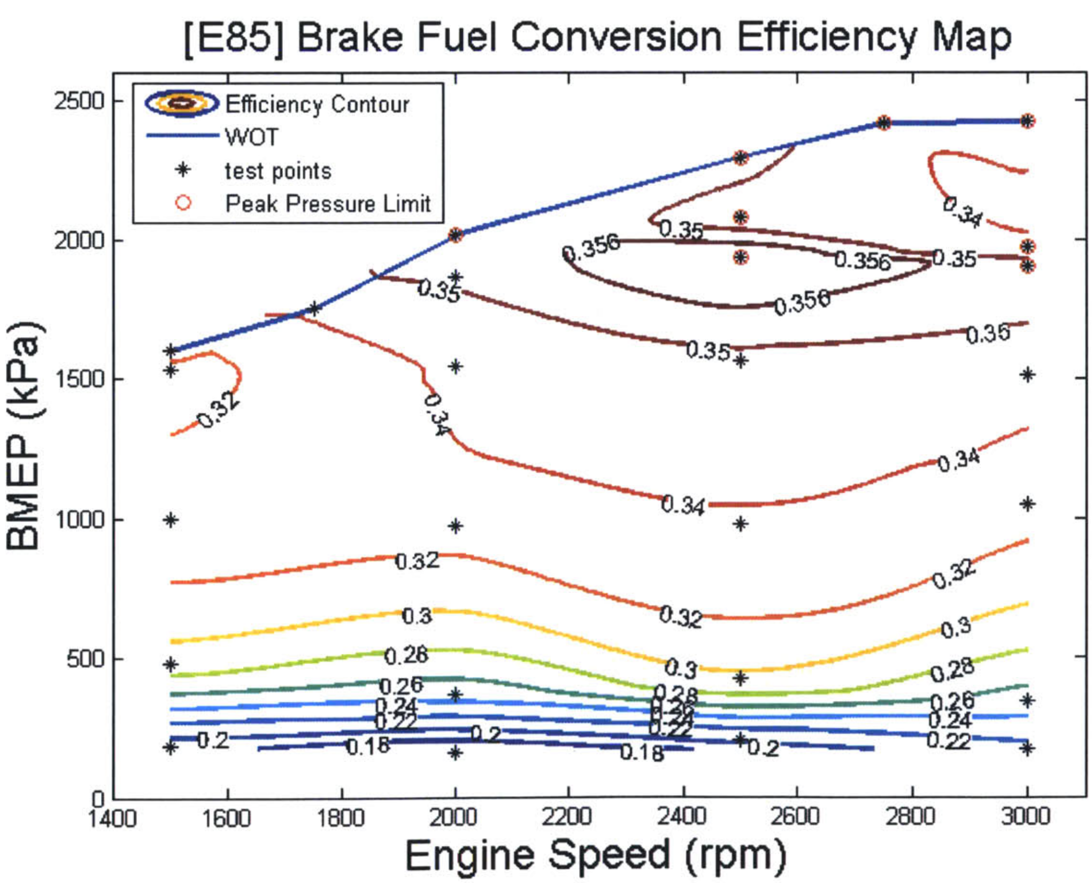

Figure 5. Brake Fuel Conversion Efficiency Contour

Testing points where spark timing was limited by maximum pressure are marked by red circles on the above graph. These conditions limited the engine reaching MBT (Maximum Brake Torque) timing at 2000 rpm and 3000 rpm at high loads; those testing points are marked with red circles on a performance map. Slightly less work output due to spark retard explains the slightly lower efficiency values right below the wide open throttle line. Decrease in efficiency with spark retard of turbocharged engine at high load will be explored further in the later section. None of the tests above exceeded turbine inlet temperature of 980 C. Figure 6 shows brake specific fuel consumption contours.

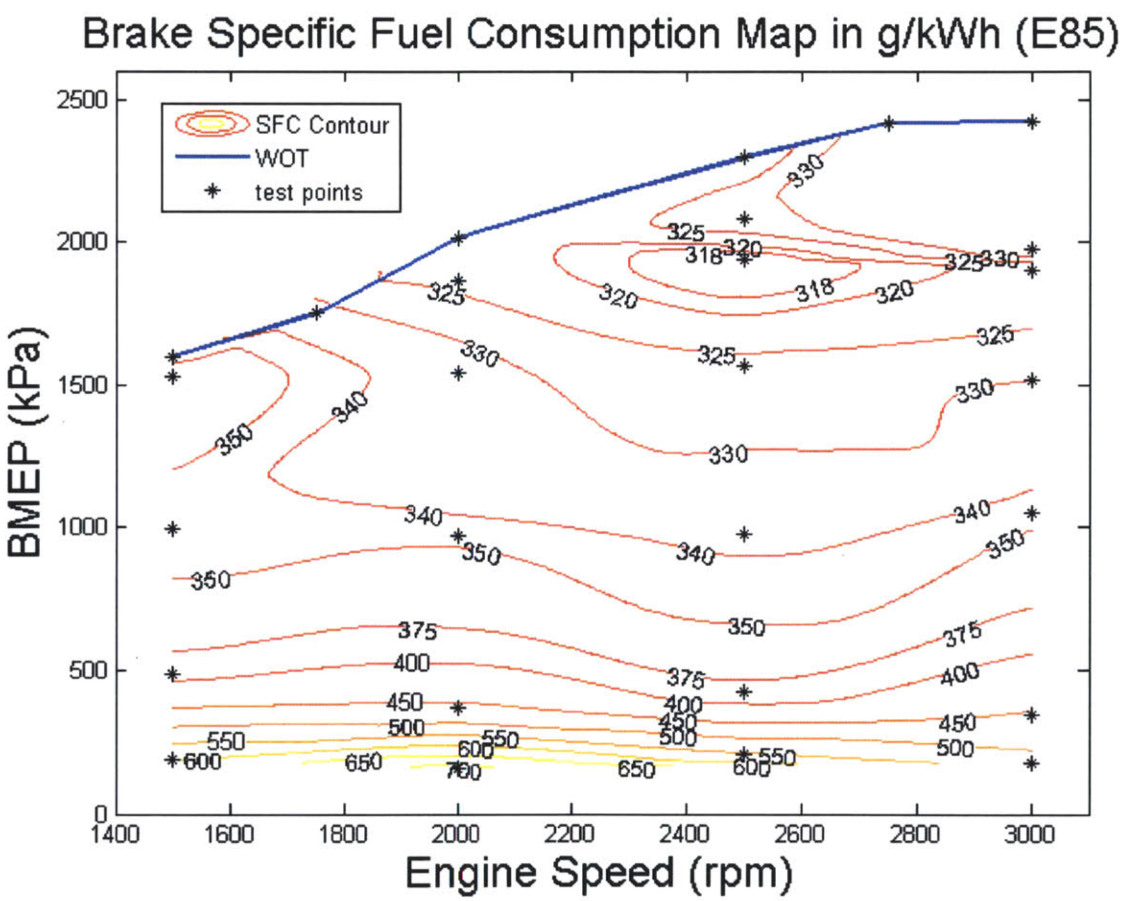

Figure 6. Brake Specific Fuel Consumption Map

Brake specific fuel consumption and fuel conversion efficiency show that the engine is the most efficient around 2500 rpm. General trend is that the engine gets more efficient as the load increases. There is a diminishing return as the boost increases, which corresponds to BMEP increase. Around speed of 3000 rpm, efficiency contour lines are curving upwards, which represents decreasing efficiency compared to same load at slower speeds. This trend can be explained by the increase in mechanical friction and pumping at higher speeds. Since the engine experimental setup was not capable of high speed experiments, simulation results were used for speeds higher than 3000 rpm.

The best value of brake specific fuel consumption was around 318 g/kWh, which is significantly higher than normal engines. This is because lower heating value of the ethanol is significantly lower than gasoline, so much more fuel is required to result in the same output of power. The lower heating value of gasoline is 43MJ/kg and that of E85 is 31MJ/kg, so 318 g/kWh of E85 results in equivalent power as 230g/kWh of gasoline. This vale corresponds to the best brake specific fuel consumption value of modern gasoline engine.

4.3 Universal efficiency

Thermal Efficiency

To compare the fuel conversion efficiencies of E85 and RON96, the performance map efficiency contours for RON96 have been generated. The knock onset constraints limited engine load to 120 kPa MAP. Wide-open-throttle operation with RON96 gasoline at speeds higher than 2500 rpm involves some engine operating issues. As spark timing is retarded significantly to avoid knock, the exhaust gas temperature (EGT) increases. It is necessary to monitor the temperature of the exhaust gas so it does not exceed the turbine inlet temperature limit. In current engine operation, the engine may be run rich to cool down the EGT. To run the engine stoichiometric, temperature sensors are being checked to ensure they provide reliable temperature reading.

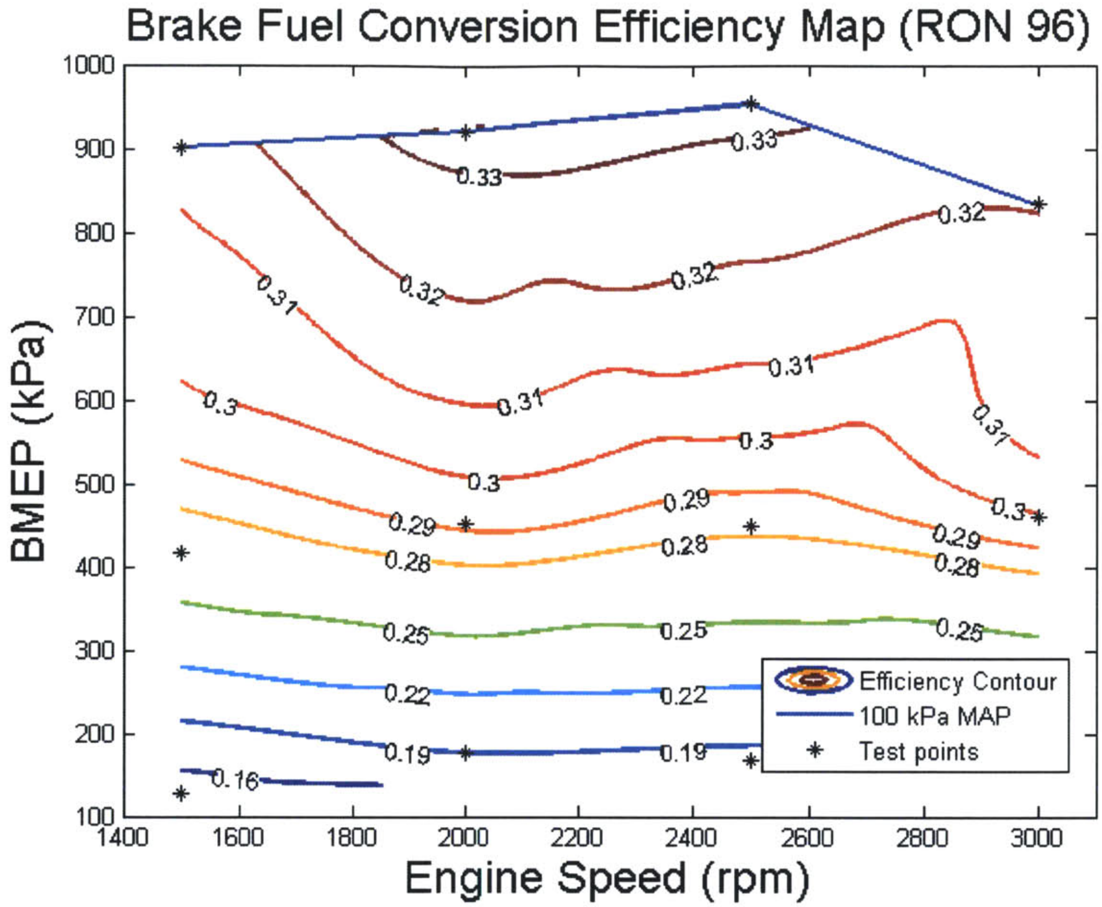

Figure 7. Brake Fuel Conversion Efficiency of RON 96

Figure 7 is a performance map with brake efficiency contours for RON96 gasoline fuel. To compare efficiency at MBT timing, only part load data were taken where knock was not a constraint. It could be seen that general trend of increasing efficiency with higher load was similar to that of E85. In Figs. 8 to 11 below, to make a comparison easier, brake fuel conversion efficiencies for RON96 gasoline and E85 were plotted at fixed speeds.

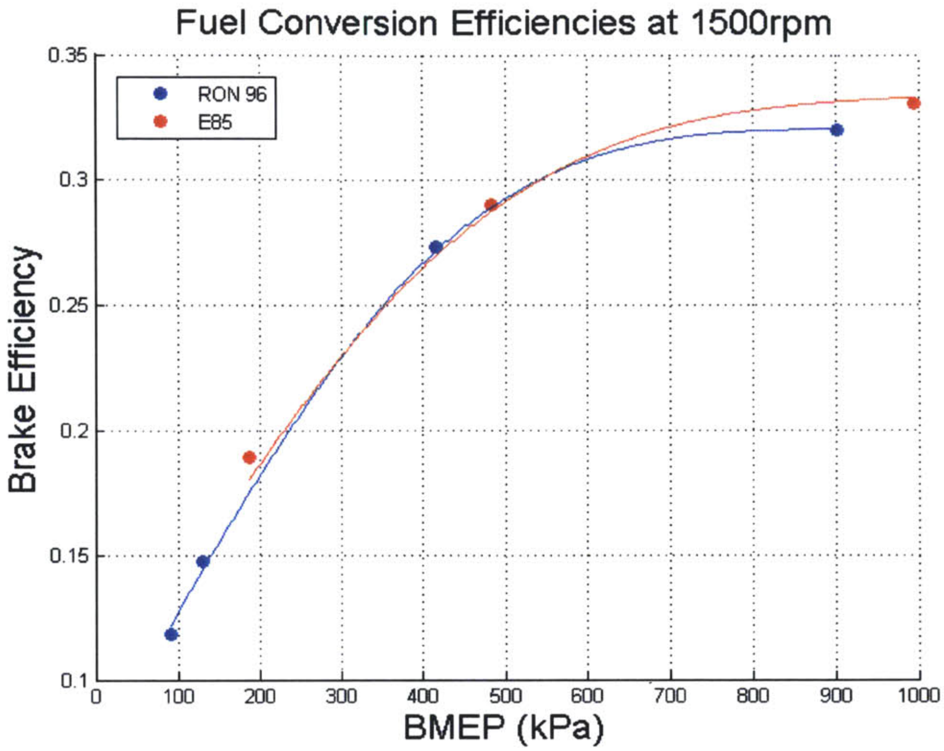

Figure 8. Fuel Conversion Efficiency at 1500 rpm

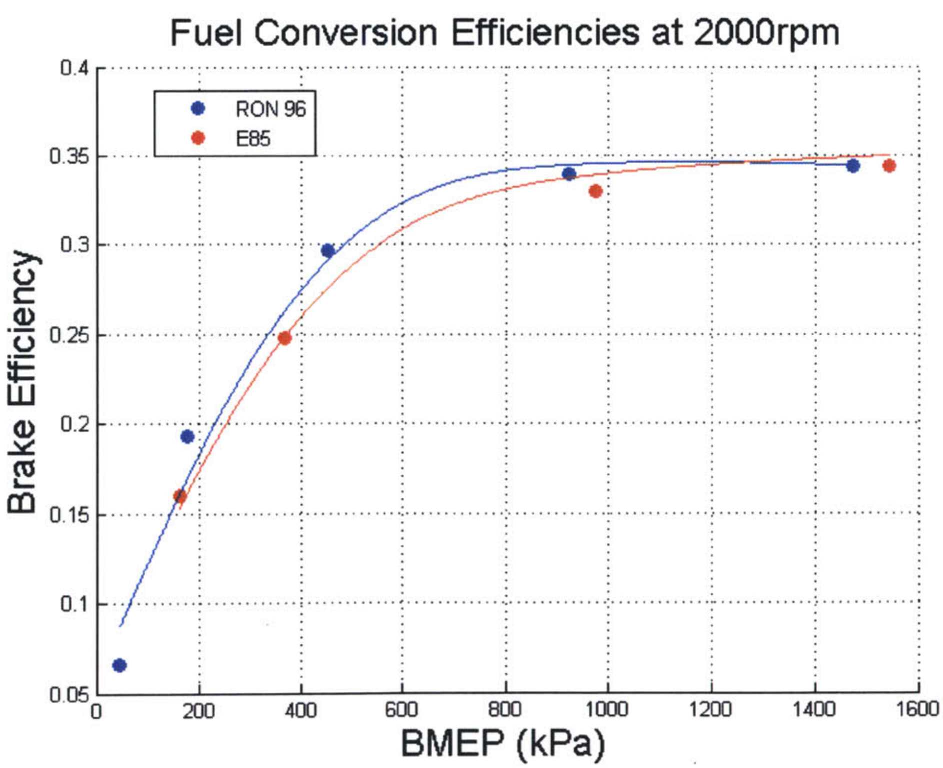

Figure 9. Fuel Conversion Efficiency at 2000 rpm

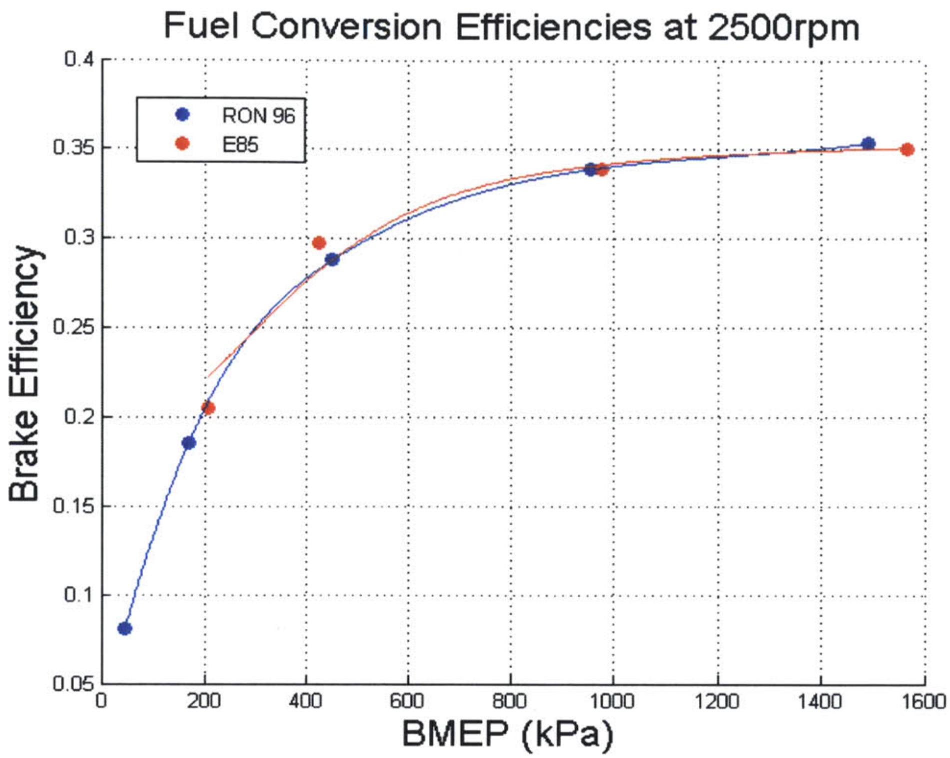

Figure 10. Fuel Conversion Efficiency at 2500 rpm

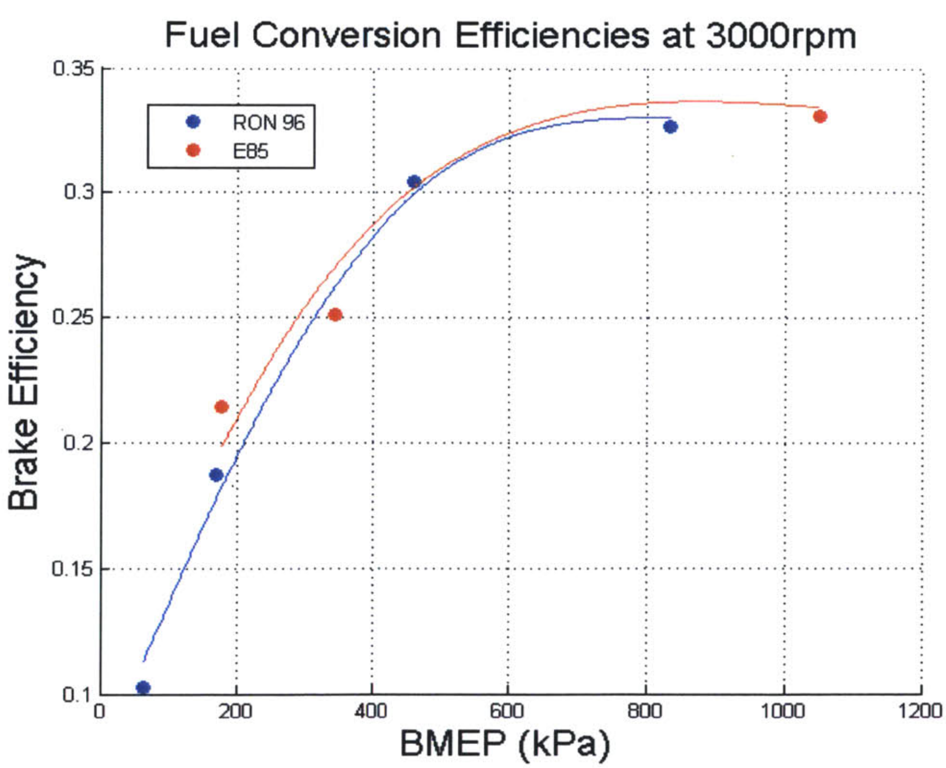

Figure 11. Fuel Conversion Efficiency at 3000 rpm

At each speed, each fuels' data points were fitted by a smooth cubic spline. Where the curves appear different, focusing on the data points directly shows that both the RON96 gasoline and E85 data follow an almost identical trend. Efficiency values at a given load do not differ significantly. We conclude that the brake efficiencies of two fuels as a function of speed and load are essentially the same. Thus it is appropriate to generate a universal performance map for each engine we study.

Pumping Work

With the brake efficiency question resolved, it is useful to break out its components to see how pumping work and mechanical friction work compare for the two fuels. Increasing fuel conversion efficiencies at high loads indicate less pumping work where the difference between intake MAP and exhaust MAP is small. The observed trends are essentially the same for the high octane gasoline and E85 fuels. Figure 12 compares pumping work for the two fuels at different speeds and loads.

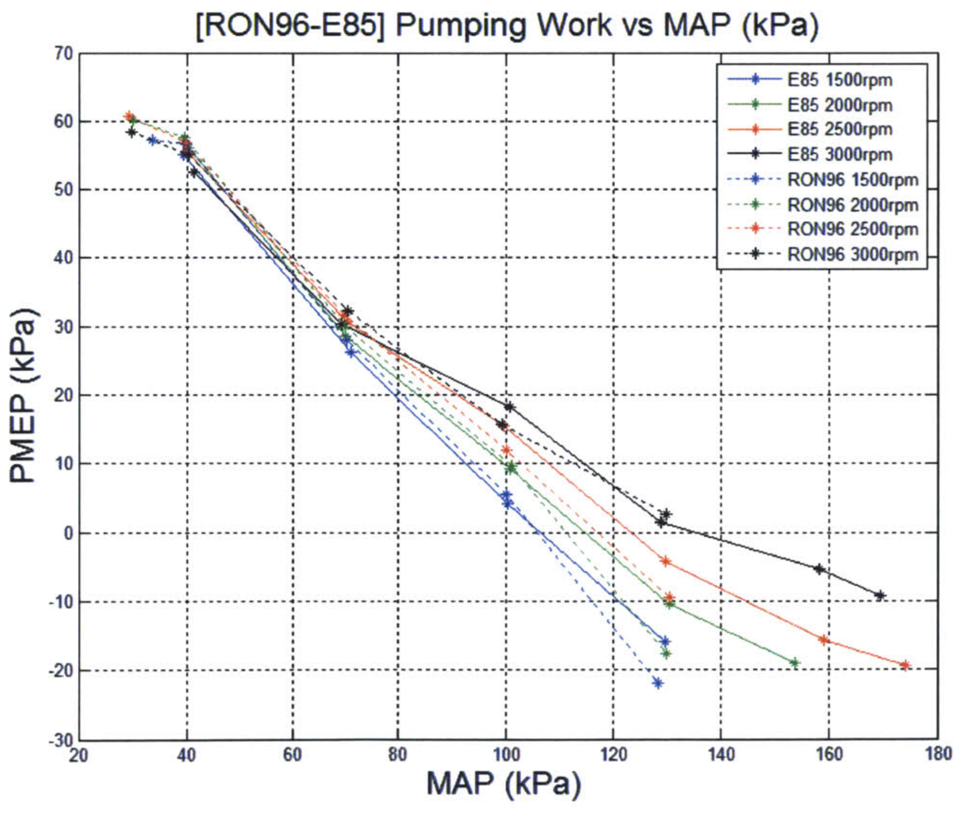

Figure 12. Pumping Work of RON 96 Gasoline and E85 at wide range of MAP

Pumping mean effective pressure (PMEP) is the net work done to the piston over the exhaust and intake strokes divided by the cylinder volume. In the above case, PMEP is simply defined as

$$PMEP = P\_exhaust - P\_intake$$

where P_exhaust and P_intake denote exhaust manifold pressure and intake manifold pressure respectively. Therefore, positive PMEP indicates that work is done by piston to pump the air from intake to exhaust. PMEP as defined above, is positive in naturally aspirated engines (i.e., is a loss). For our turbocharged engine, the graph above shows negative pumping work at higher loads which means that work is done to the piston. Comparable PMEP between two different fuels further supports our universal performance map assumption.

Mechanical Efficiency

Other than pumping work, part of gross indicated work is used to overcome friction in various parts in the engine. Ratio of useful power delivered by the crankshaft of the engine divided by the gross indicated power from the cylinder is defined as the mechanical efficiency. Figure 13 shows the mechanical efficiency trends for the two fuels.

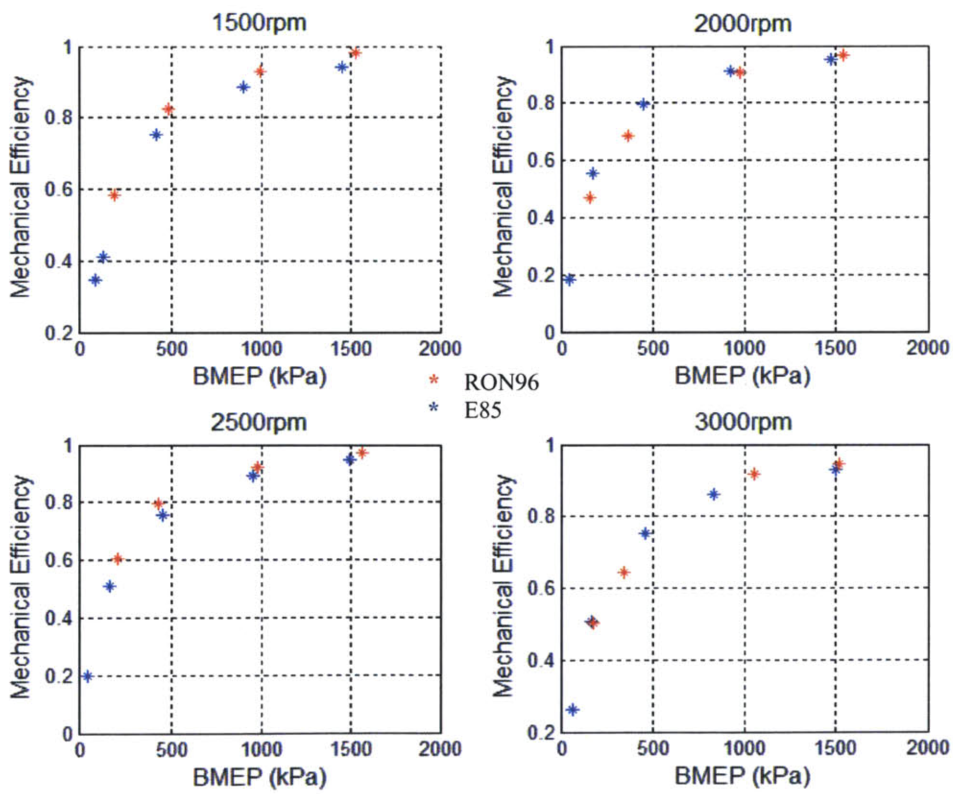

Figure 13. Mechanical Efficiency of RON 96 Gasoline and E85 at Different Loads

Mechanical efficiency is the ratio of brake output (usable output) to indicate output (transferred to the pistons). As load (BMEP) increases, mechanical efficiency increases, but at a decreasing rate indication that the negative impact of engine friction is decreasing. Note that at high loads, mechanical efficiency almost reaches 1.0. This is due to the engine's pumping work at these conditions transitioning from being a loss (and thus a component of friction) to a gain as the engine intake pressure exceeds the exhaust pressure.

5. Knock Onset

5.1 Test Matrix

Table 5. Test Matrix for Knock Onset Experiments

Operating Parameter	Values
Fuel Type	RON 91 gasoline, RON96 gasoline, E10, E20, E25, E50, E85, EW15
Speed set points (rpm)	1500, 1750, 2000, 2250, 2500, 2750, 3000
Load in MAP (bar)	1.0, 1.1, 1.2, ... WOT
Valve Timing	Valve Timing Map in ETAS
Lambda Set point	1.0 (Stoichiometric)
Experiments type	Spark Sweep

5.2 Knock Onset Determination

Before investigating knock-limited maximum brake torque spark timing, it is necessary to define the knock onset in the spark-ignition engine. Knock onset is when maximum knock intensity is larger than 200 kPa and knocking frequency is larger than 10 %.

At a given speed and load, the spark timing was advanced until the engine knocks. Since engine control unit (ECU) avoids knocking by retarding the spark timing when knock sensor senses knock, it was necessary to disable the knock sensor. After collecting knock data, a spark sweep was conducted to see whether MBT timing could be reached. As you can see in a Figure 14 below, at 2000 rpm and 110 kPa MAP, knock onset was at 25.5 CAD bTDC. MBT timing is around 20 CAD bTDC, so MBT timing was not knock limited.

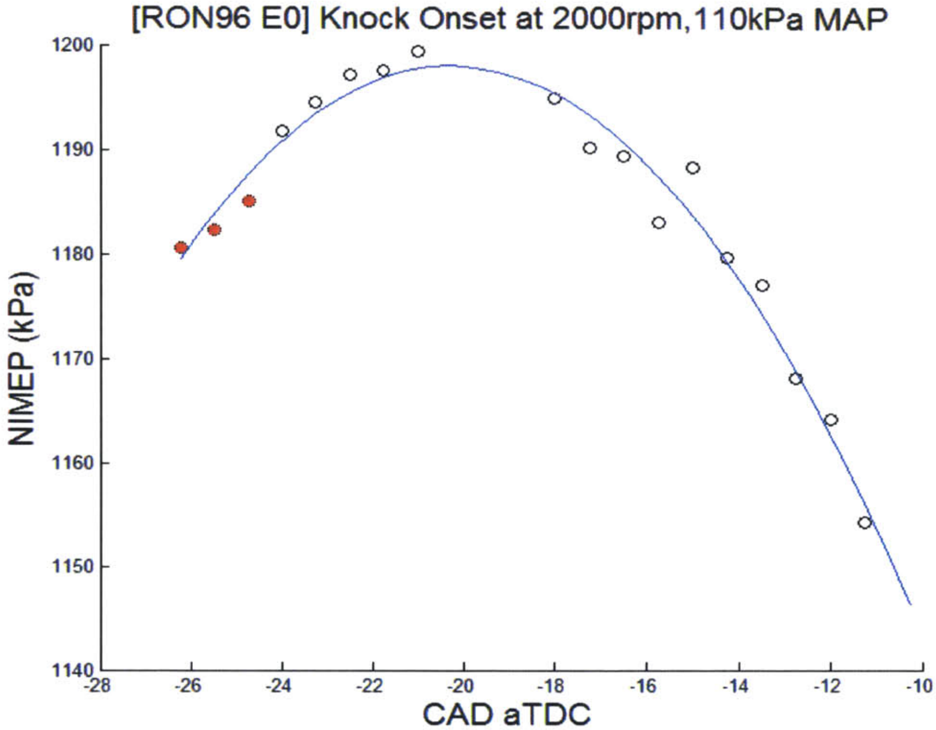

Figure 14. Knock Onset at 2000 rpm, 100 kPa MAP of RON 96 Gasoline

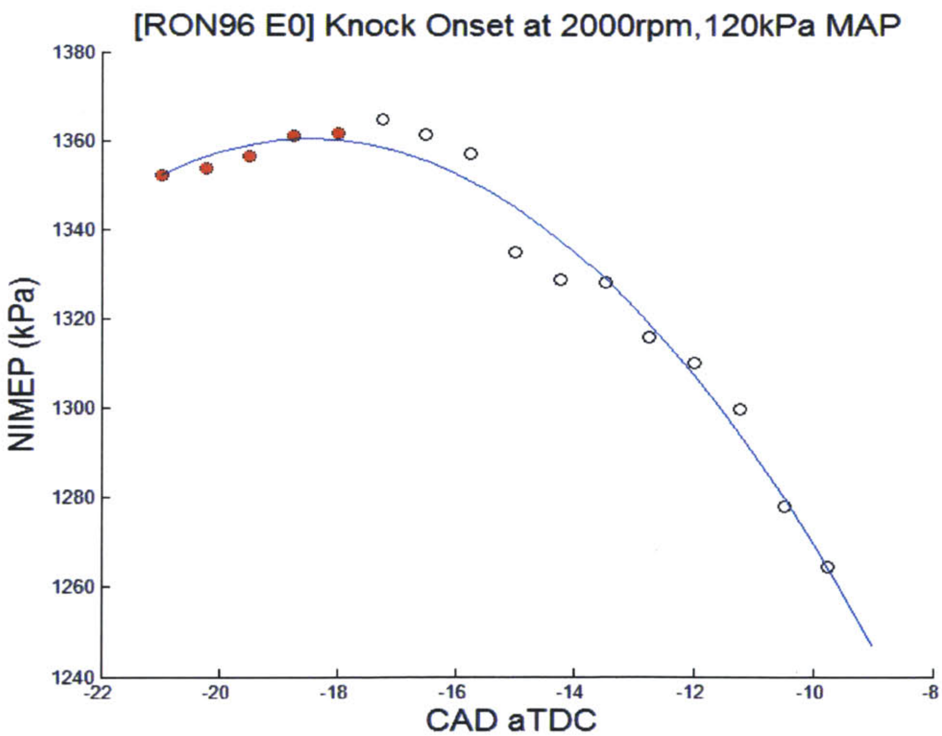

Figure 15. Knock Onset at 2000 rpm, 120 kPa MAP of Ron 96 Gasoline

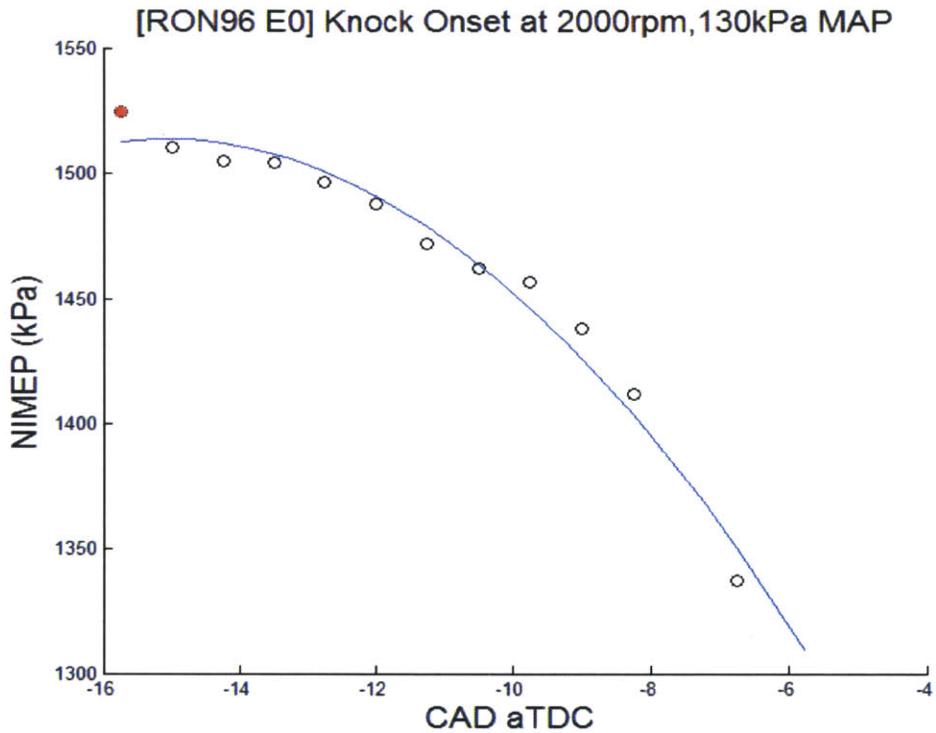

Figure 16. Knock Onset of RON 96 Gasoline at 2000 rpm, 130 kPa MAP

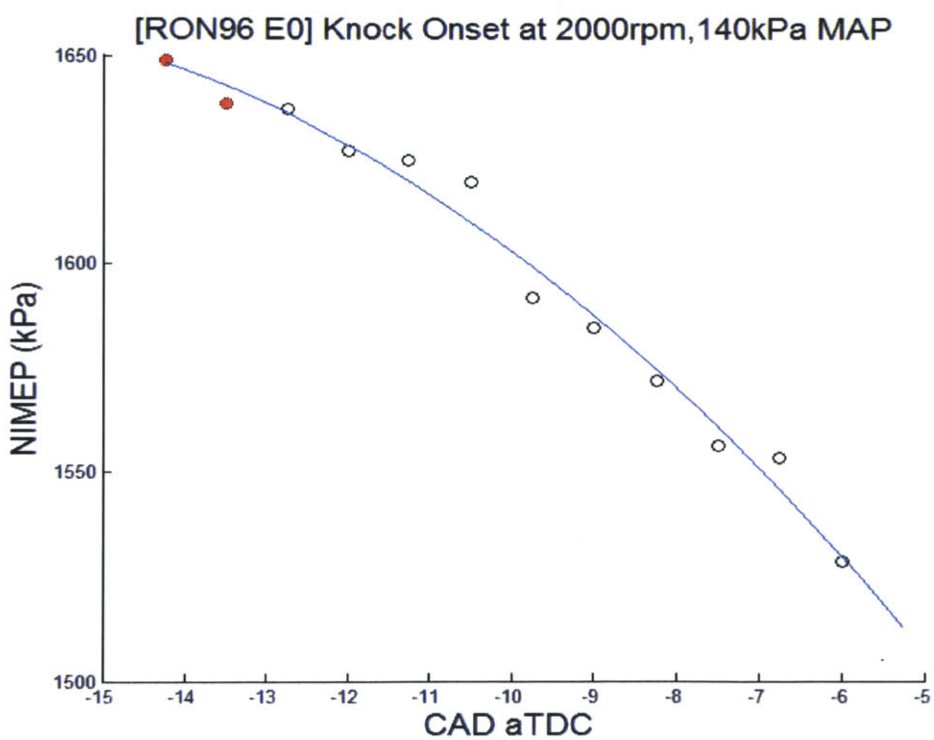

Figure 17. Knock Onset of RON 96 Gasoline at 2000 rpm, 140 kPa MAP

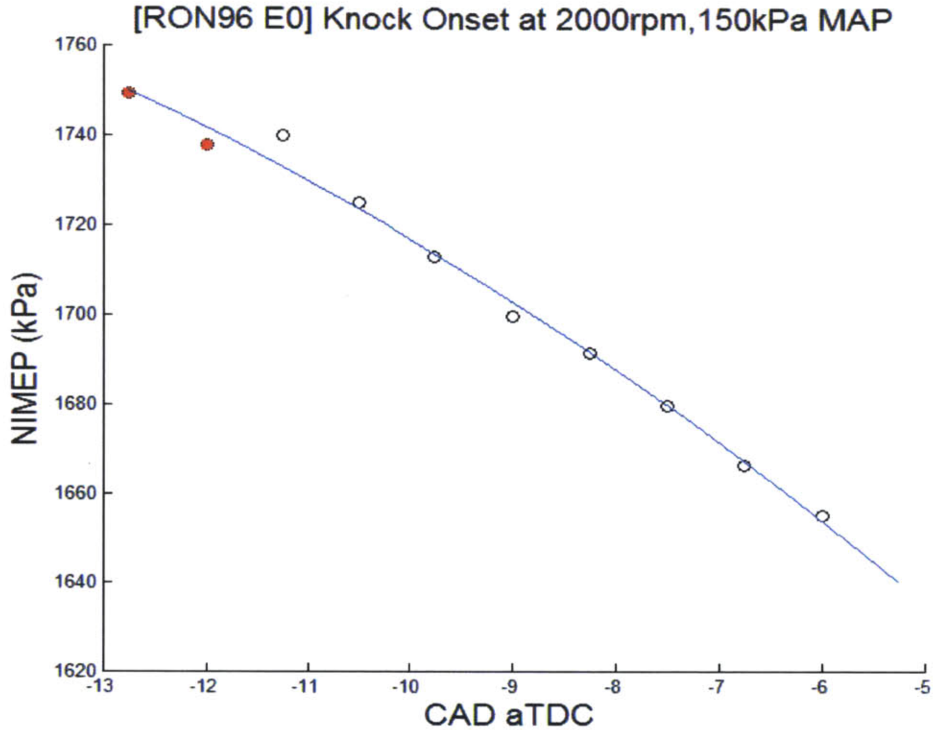

Figure 18. Knock Onset of RON 96 Gasoline at 2000 rpm, 150 kPa MAP

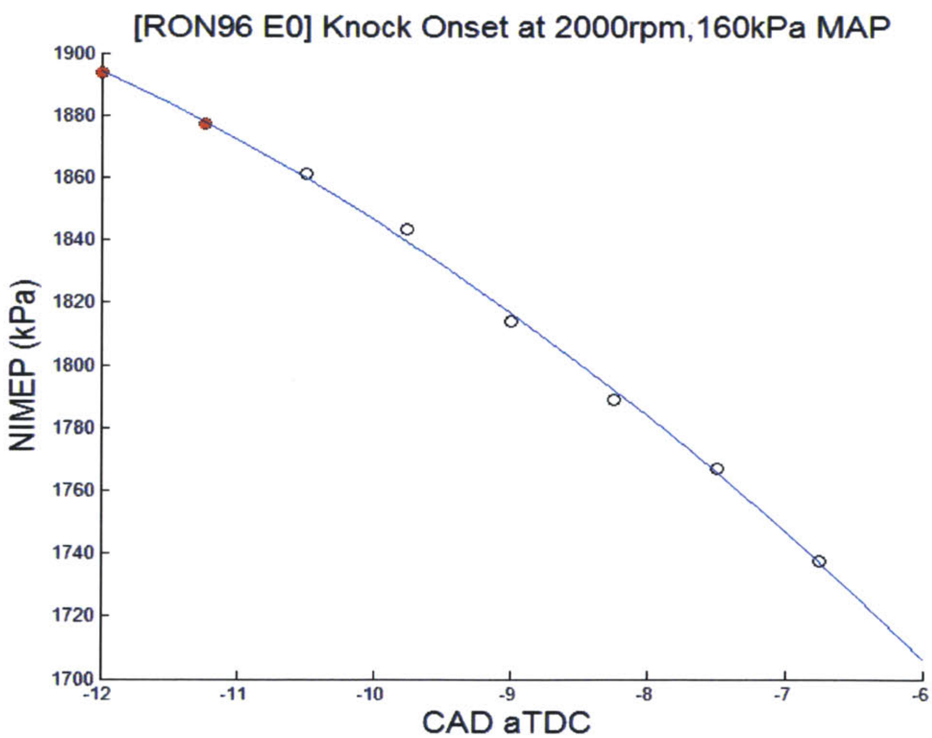

Figure 19. Knock Onset of RON 96 Gasoline at 2000 rpm, 160 kPa MAP

When manifold air pressure is increased above 130 kPa, it can be seen that MBT timing was not reached. At this point, a spark retard must be used to avoid knock, but the spark retard due to knock causes some efficiency loss as actual spark timing gets farther from MBT timing. This spark retard from MBT timing causes an efficiency degradation. At higher loads where the spark timing is increasingly retarded to avoid knock, the knock-suppressing effect of ethanol will improve the efficiency of the engine significantly.

The same experiments were done with different fuels at different range of speeds as in the test matrix. Figure 20 includes three sets of spark timing sweeps conducted at intake manifold pressures of 100 kPa, 110 kPa and 120 kPa with the engine speed fixed at 2500 rpm. Knocking conditions are indicated by filled markers. Conditions at knock, KLSA (Knock Limited Spark Advance) were determined by the definition of knock onset. This indicates the operating point where knock intensity is larger than 2 kPa and knock frequency is more than 10 %. The solid red line connects knock onset data points using a fitted quadratic curve. By interpolating the knock onset data points, it is possible to divide the engine's operating region into two. The engine can operate without knocking under the red line with MBT (optimum) park timing, but there are strategies (such as spark retard, ethanol injection, etc.) to avoid knock above the red line.

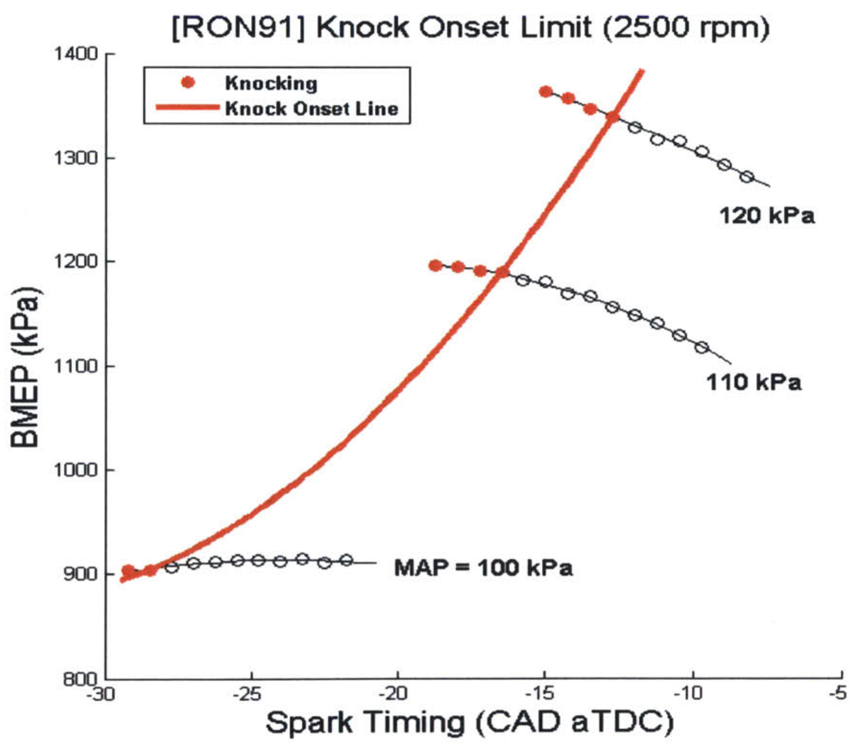

Figure 20. Knock Onset Limit of RON 91 Gasoline at 2500 rpm

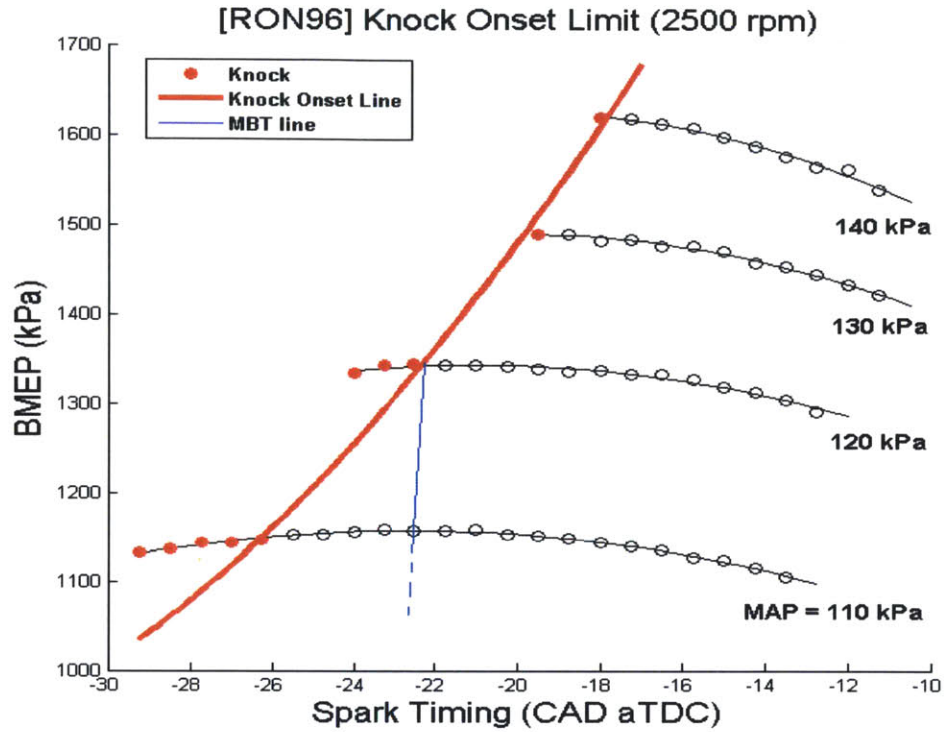

Figure 21. Knock Onset Limit of RON 96 Gasoline at 2500 rpm

Figure 21 shows the knock onset line interpolated by knock onset operating points for the higher octane gasoline tested. At spark timing of -25 crank angle degree (CAD) after top dead center (TDC), RON91 gasoline knocks at 950 kPa BMEP, but RON96 gasoline knocks at 1200 kPa BMEP. This demonstrates how higher octane gasoline enlarges the knock free region of the engine operating map.

To determine how the maximum brake torque (MBT) ignition timing is limited by the knock onset, the MBT spark-timing line was drawn. MBT timing is spark ignition angle where brake torque is maximum. Then brake torque at +/- 5 CAD were checked to see if they are about 1 % lower than the MBT. The point where the MBT line and knock onset line meet is a useful reference knock onset limit at a given speed. In this case, it is 1340kPa BMEP. Figure 22 below indicates the MBT line and KOL (Knock Onset Line) of RON91 gasoline mixed with 10 percent ethanol.

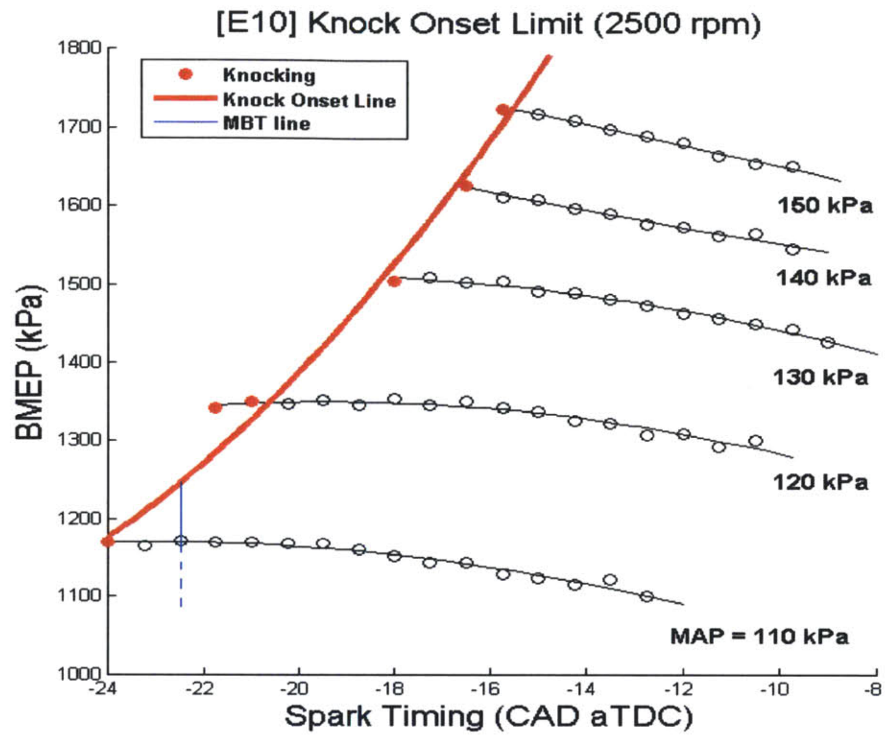

Figure 22. Knock Onset Limit of E10 at 2500 rpm

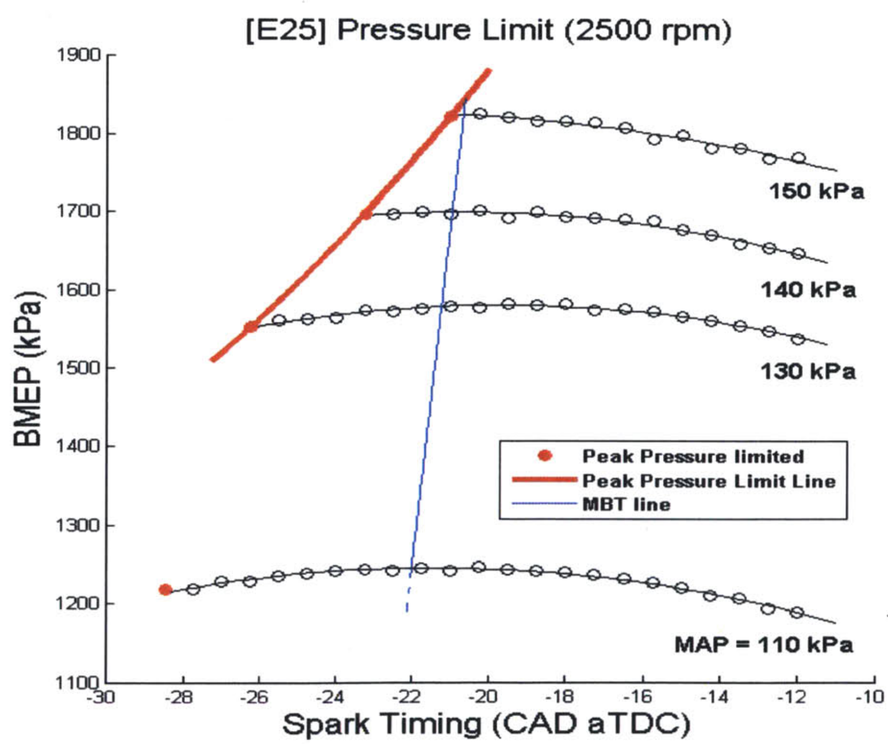

Figure 23. Knock Onset Limit of E25 at 2500 rpm

Figure 23 shows the spark sweep for E25 fuel: the MBT spark timing for E25 was not limited by knock. However, it was limited by in-cylinder maximum peak pressure. The GM LNF Ecotec engine was designed to withstand maximum pressure of 110 bar mean plus 2 sigma, and the highest mean peak pressure tested at GM was 97 bar at 5300rpm. In a paper by Stein, et al. the physical limitation for a similar engine was 100 bar mean + 3 sigma [SAE paper 2009-01-1490]. To protect the engine, peak pressure was limited by 100 bar mean + 2 sigma. At 2500 rpm, E25 was not limited by knocking but was limited by peak pressure limit.

5.3 Knock Onset Limit on a Performance Map

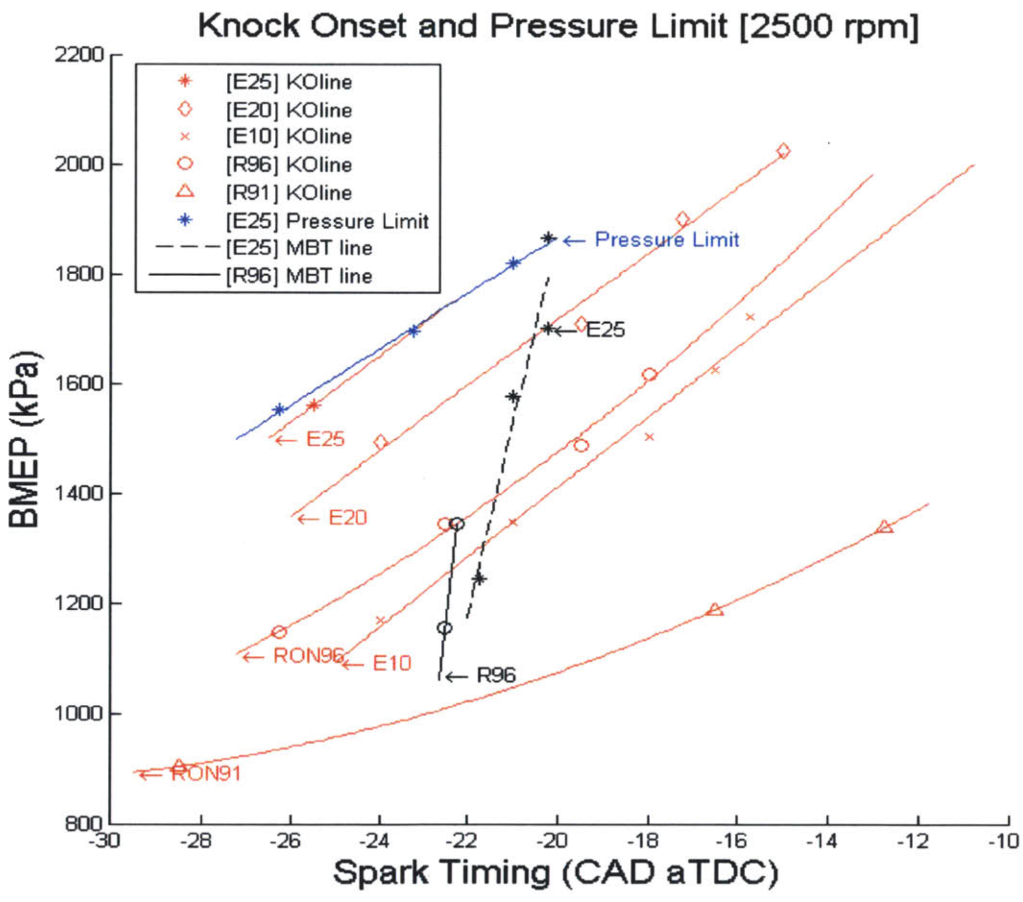

Figure 24. Knock Onset Limit of different fuels at 2500 rpm

Figure 24 includes knock onset limits, the maximum pressure limit line, and MBT timing line for different fuels at 2500 rpm. By comparing the four fuels, it can be seen which fuel has the larger operating region before reaching knock onset. E25 was primarily limited by peak pressure rather than by knock limit. It did knock when spark timing was advanced far enough, but MBT timing was not limited by knocking. RON 96 gasoline was better than E10 or RON91, but it was still knock limited at 1346 kPa BMEP. Also, it is interesting to note that knock onset for E10 was slightly lower than RON96 gasoline. The gasoline from Haltermann actually had RON of 97.1, and an E10 blend with base gasoline of RON 92 gives RON number of 96 [7]. The same analysis can be applied at 2250rpm and 2750 rpm as shown in the Figure 25 and Figure 26 below.

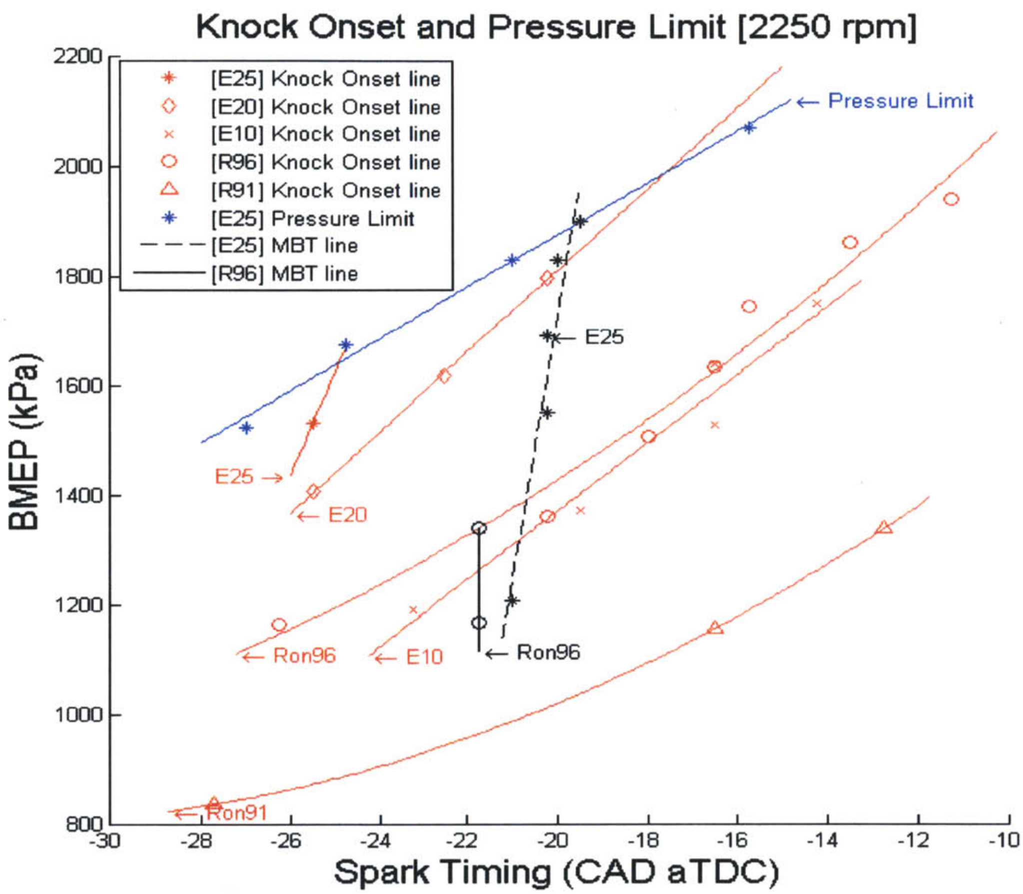

Figure 25. Knock Onset Limit of different fuels at 2250 rpm

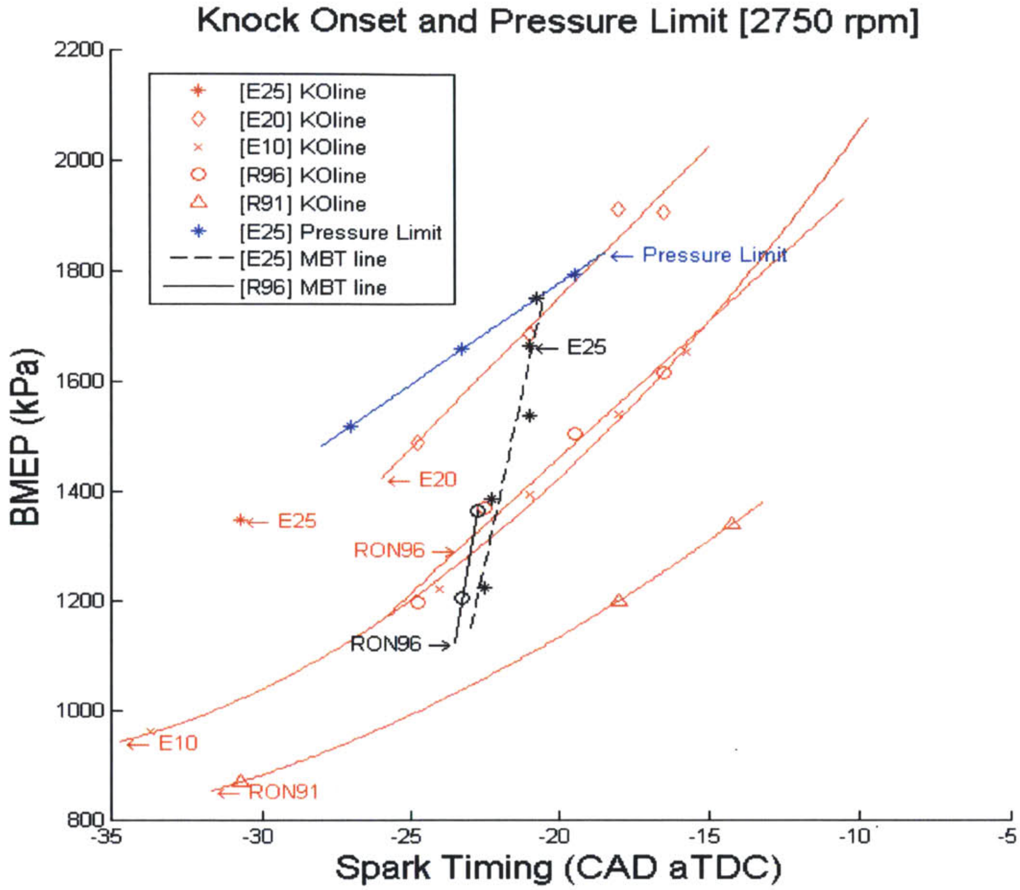

Figure 26. Knock Onset Limit of different fuels at 2750 rpm

From these spark sweeps at a fixed speed, the knock onset limit and peak pressure limit for different fuels has been determined. This information can then be inserted onto a universal version of our engine's performance map as shown in Figure 27.

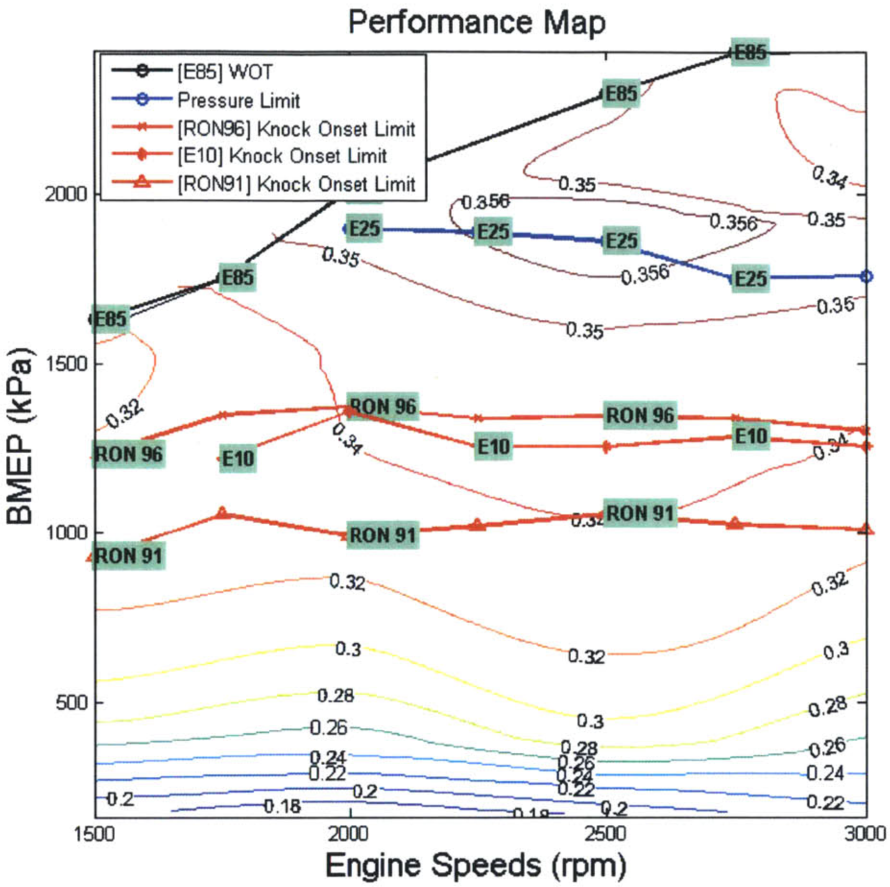

Figure 27. Performance Map of the Engine with Knock Onset Limit Lines

Efficiency contours on the performance map in Figure 827 are based on E85 fuel. Knock limits and pressure limits of different fuels were overlaid on E85 efficiency contours because we determined that fuel conversion efficiencies are not dependent on fuels in the region where the engine is not knock limited.

Naturally-aspirated engines would give maximum BMEP of around 1000 kPa, where RON 91 gasoline at MBT spark timing is limited by knock. Thus it is necessary to enrich the mixture and retard the spark timing to avoid knock to attain maximum BMEP levels of 1200 kPa. In a turbocharged engine, the BMEP level is almost twice as large as a naturally aspirated engine's BMEP level. As seen in Figure 27, both gasoline fuels (RON 96 and RON 91) are knock limited at well below the WOT line. Substantial amounts of spark retard are needed to avoid the knock, but enrichment of mixture is also required to keep turbine inlet temperature under its limit. Enrichment of mixture and spark retard negatively effect the efficiency of the engine. However, using ethanol will allow knock to be suppressed with optimum spark timing, and the efficiency of the engine to be improved.

It was anticipated that as engine speed increases, the knock margin would increase. This is partly true because spark timing with respect to TDC could be more advanced. However, MBT timing also gets more advanced with respect to TDC as the engine speed increases. Therefore, the knock margin was not greatly increased as engine speed increase. However, WOT brake torque does increase substantially as engine speed increases. Thus making use of ethanol in the mid-speed range is important since using ethanol to suppress knock enlarges the knock-free operating region greatly.

6. Boost Level, Spark Retard, and RON Requirements

Standard knock suppression strategies include retarding the spark timing. The spark retard results in a loss in torque as combustion occurs later than is optimum in the compression stroke. The torque loss is not significant with less than 5 CAD retard in spark. However, torque loss increases to 5~6 % with 10 CAD retard depending on the operating condition. [11] To estimate the decrease in ethanol consumption that is achieved with retarded spark timing, it was necessary to develop a methodology that incorporated spark retard into the driving cycle simulation. First, RON requirements at high boost conditions with different spark retard levels were estimated.

6.1 RON vs Boost and Spark Retard

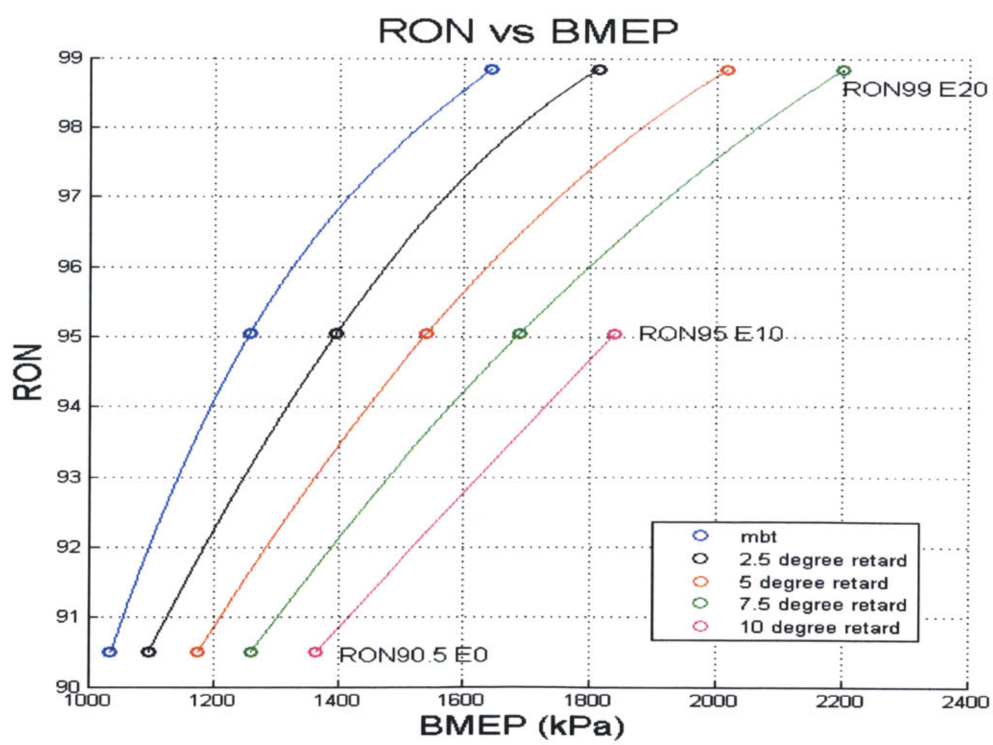

Figure 28. RON vs. BMEP for various spark retard

Figure 28 shows how spark retard increases the knock free margin of the operating regime. The blue line indicates the knock limit of RON 91 gasoline, E10, and E20 gasolines with no spark retard. The black line is the case with retarding the timing 2.5 CAD to increase the knock margin, etc.

6.2 Spark Retard in a Turbocharged Engine

Torque decreases with spark retard in a turbocharged engine, but the decrease is not as large as that of naturally aspirated engine. It is because increase in exhaust gas enthalpy drives the turbocharger faster, therefore increasing the intake manifold pressure. This effect becomes more and more significant as the boost level increases. The trend of increasing manifold air pressure and decreasing torque loss is shown in the Figure 29 and 30.

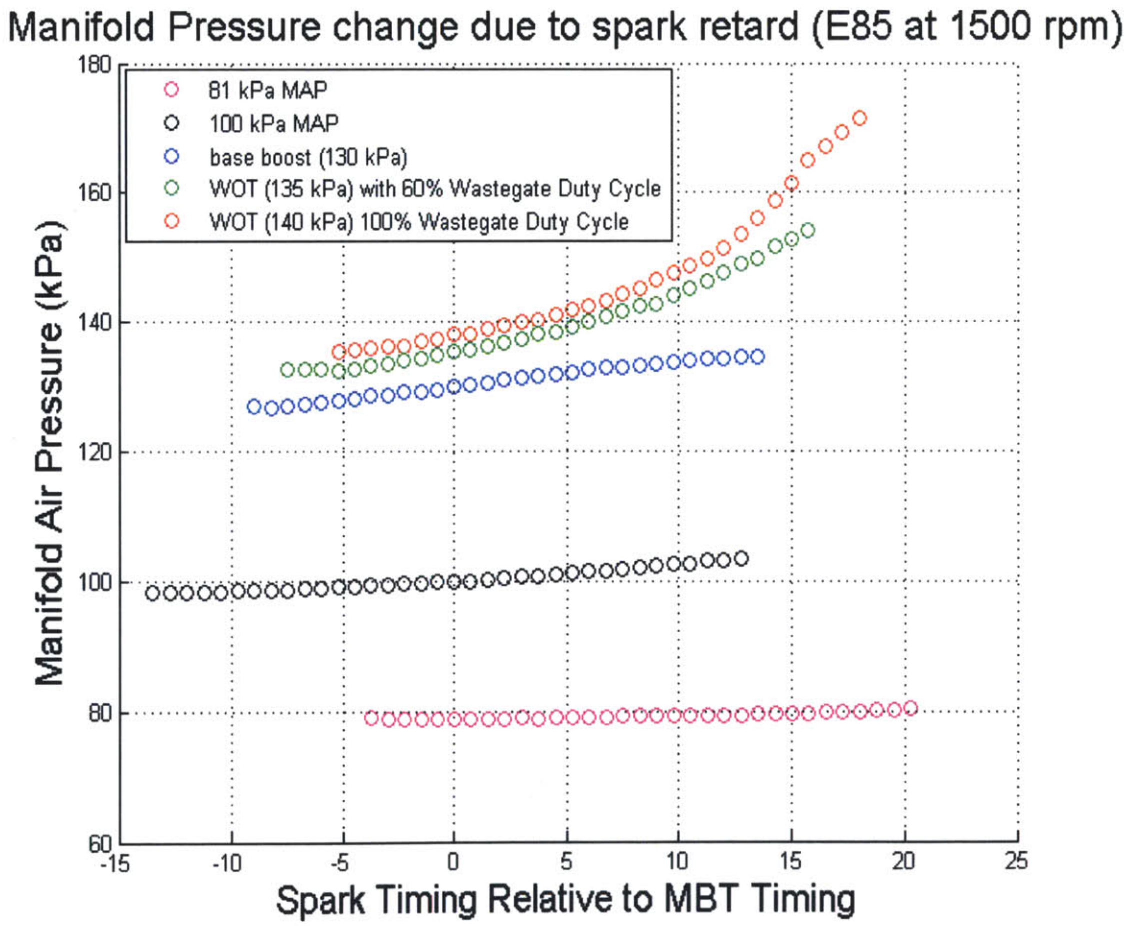

Figure 29. Manifold air pressure increase with spark retard

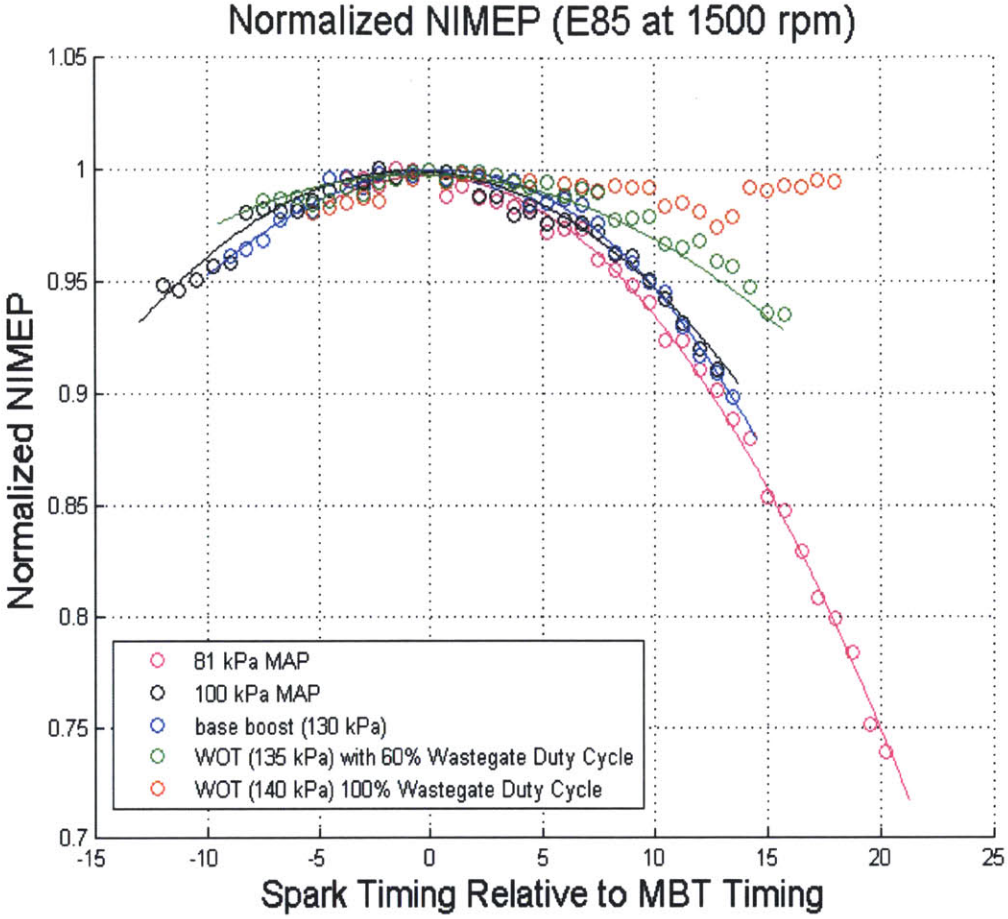

Figure 30. Normalized NIMEP vs. spark timing relative to MBT

In general, 5 CAD retard and 10 CAD retard in spark timing from MBT results in 2 % and 5 % loss in torque for naturally aspirated engines. A turbocharged engine generally follows this rule until the turbocharger starts creating boost. With significant boost, 5 CAD retard results in about 2 % loss in torque, and 10 CAD retard results in 6 % loss in torque. As boost further increases, turbocharger generates more boost, and torque loss is less than 4 % with 10 CAD retard.

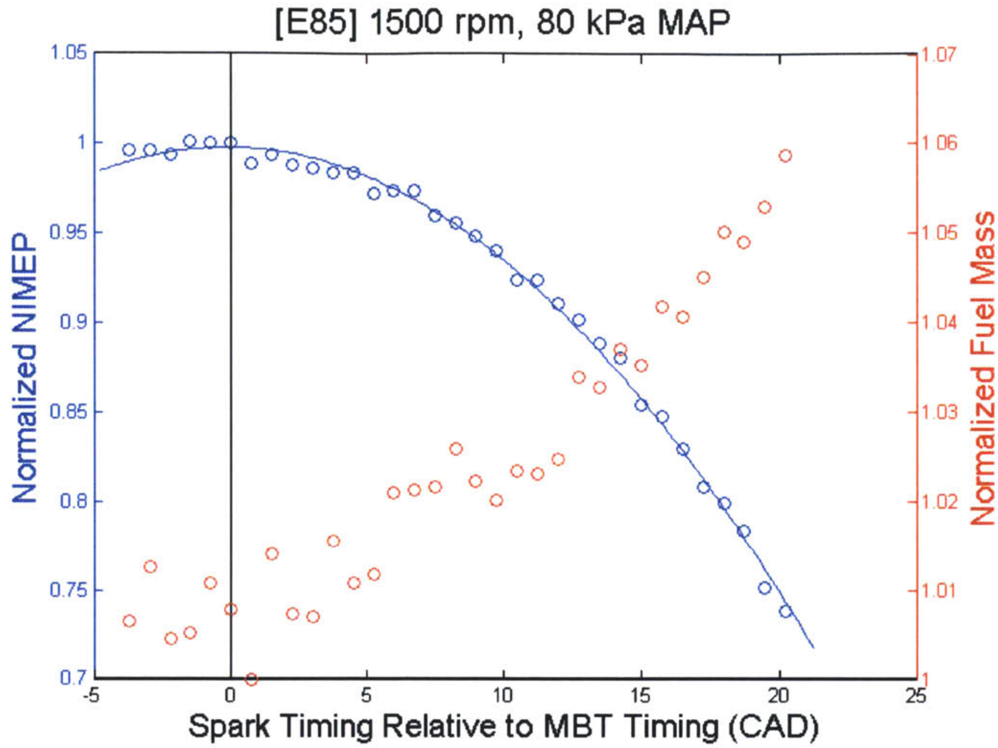

Figure 31. Spark retard and fuel flowrate increase at low load

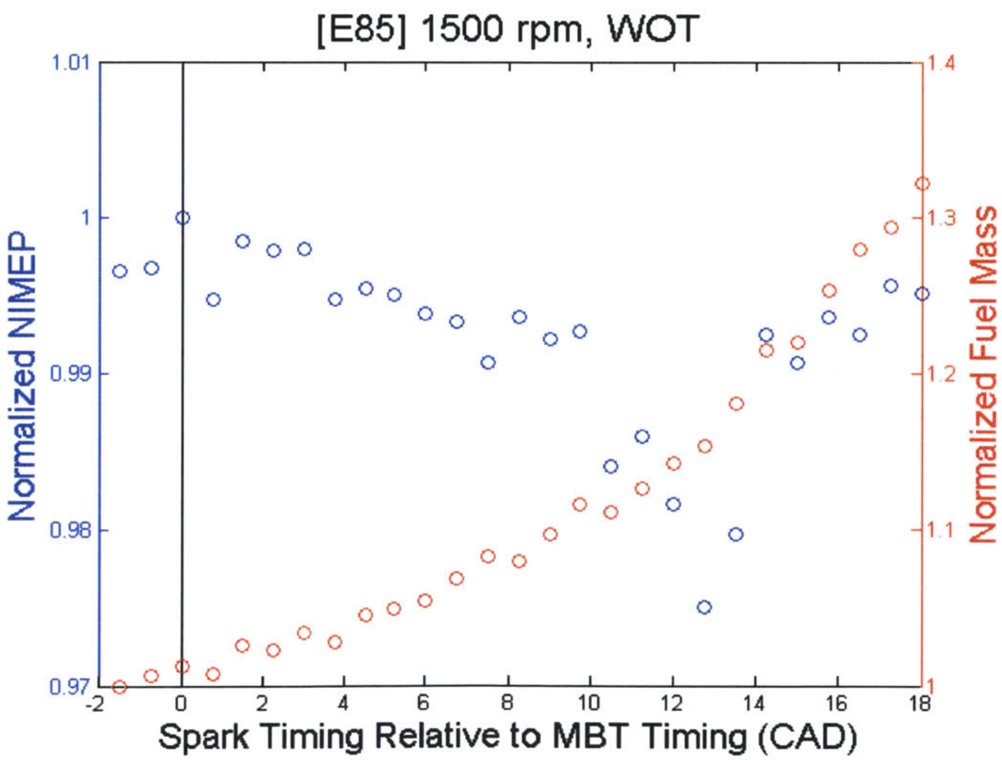

Figure 32. Spark retard and fuel flowrate increase at high load

Figures 31~32 compare how increase in fuel flow rate is different at low load and high load. At low load, fuel flow rate only increased about 2 % with 10 CAD spark retard, but fuel flow rate increased about 10 % at WOT. The increase in the fuel mass flow rate, even with not much loss in torque, deteriorates efficiency significantly as shown in the Figure 33. Because of fuel flow rate increase with spark retard, MBT timing was no longer the maximum efficiency timing.

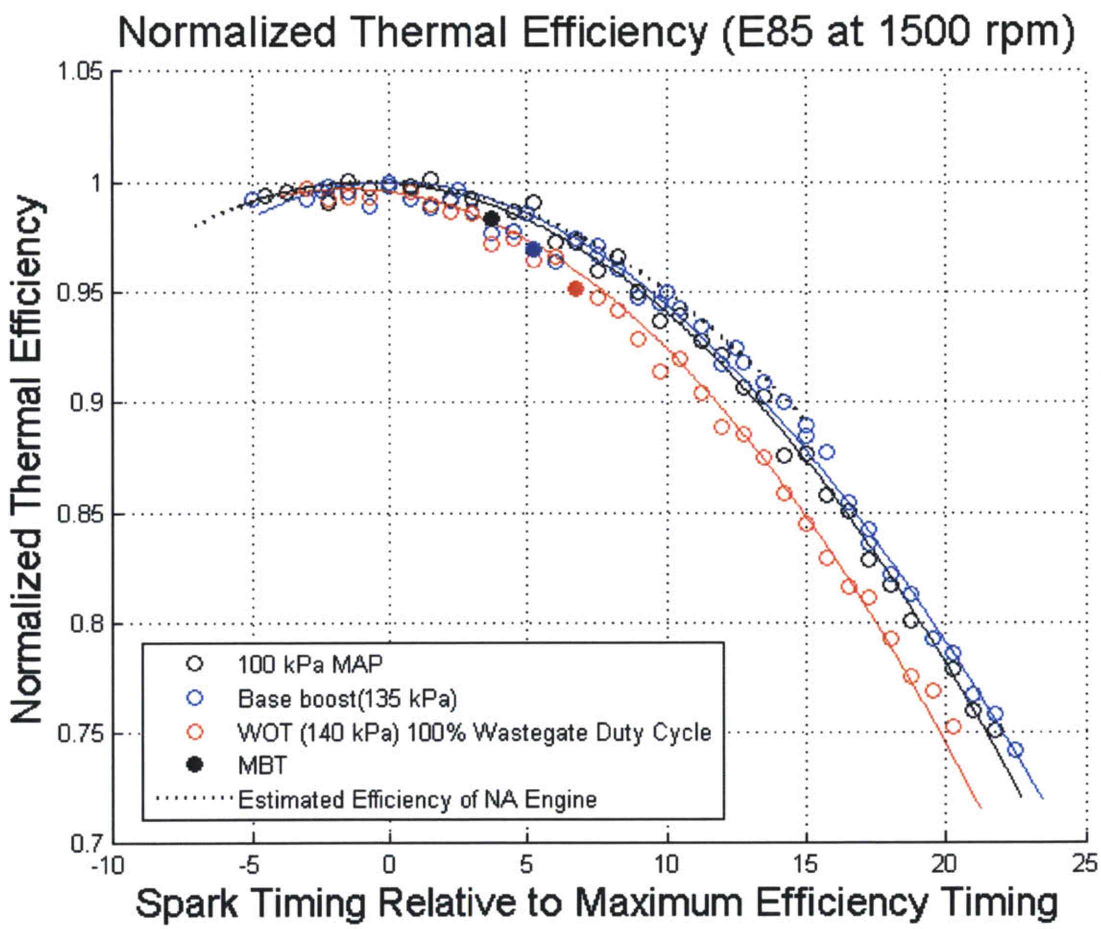

Figure 33. Efficiency decrease with respect to maximum efficiency timing

7. Engine in Vehicle Simulation

Driving cycle simulation is used extensively in the automotive industry to estimate fuel consumption. It is also a good option for evaluating relatively new engine technologies. Therefore, driving cycle simulation was chosen as our primary tool to quantify the ethanol consumption and efficiency gain of our dual fuel concept.

A software called Autonomie, developed by the Argonne National Laboratory (ANL) and the Department of Energy (DOE), was chosen for the simulation. It is a newer version of the

Powertrain System Analysis Toolkit (PSAT) which was widely used in the industry and universities for research purposes. Autonomie includes more recent technologies and complex processes for evaluating each component of the vehicle such as engines, batteries, motors, transmissions, and chassis. It is a MATLAB (Matrix Laboratory) and Simulink based software, with its own graphic user interfaces that allow the user to change the environment, vehicle propulsion, and driving cycle settings.

7.1 Vehicle Setup

To incorporate and evaluate our engine operation maps, a 2013 Toyota Camry was chosen as our vehicle frame. Specifications were taken from Toyota's website [8].

Table 6. Vehicle Parameter setup for simulation

Component	Parameter	Value	Comments
Chassis	Chassis mass	990 kg	Without powertrain
	Overall mass	-	Mass overriding
	Coefficient of drag	0.28	
	Wheel base	2.6695m	From website
	Vehicle height	1.4503m	From website
	Vehicle track width	1.5519m	From website
	Frontal area	2.2508 m^2	Height x Width
	Weight distribution	64 %	From website
Torque Converter	Size	Predefined Map	Depends on the engine torque output
Electric Accessory	Electrical accessory power loss	200 W	Power loss depends on the driving cycle
Energy Storage	State of charge	(70 %,100 %,30 %) for (initial, max, min)	12V Lead battery for conventional light, heavy duty
Final Drive	Final drive gear ratio	4.438	
Gearbox	Gear ratios at different gears	[2.563, 1.552, 1.022, 0.727, 0.52]	5 speed gear ratio from Accord V6
	Gear efficiencies at different gears	Pre-defined map	
Generator	Efficiency	Pre-defined map	
Mechanical Accessory	Power loss	0W	
Starter	Starter pinion and engine ring gear ratio	10:1	
Torque Coupling	Efficiency	Pre-defined map	
Wheels	Wheel radius	0.317 m	Wheel model for Toyota Camry

7.2 Engine Setup

Table 7. Simulation parameter setup for engine

Engine	Fuel mass	42 kg
	Operating temperature	30 C
	Initial response time	0.3 s
	Maximum power scaling	No scaling used
	Engine operating Map	X axis: engine speed in rad/s Y axis: Torque Z axis: fuel flow rate

Engine Map Expansion

Since some of the driving cycles required engine speeds higher than 3000 rpm, it was necessary to expand the performance maps for use in Autonomie. Efficiency values at speeds lower than 1000 rpm and higher than 3500 rpm were generated through use of the GT power simulation. Since driving cycles can require engine speeds up to 6000 rpm, efficiency values from 4500 rpm to 6000 rpm were chosen which give the appropriate shape to the performance map. After running Autonomie, it was confirmed that the engine speed exceeds 4500 only for 6.3 seconds a 1.2L turbocharged engine. Energy output by the engine in those simulation points were about 5 % of the total energy output.

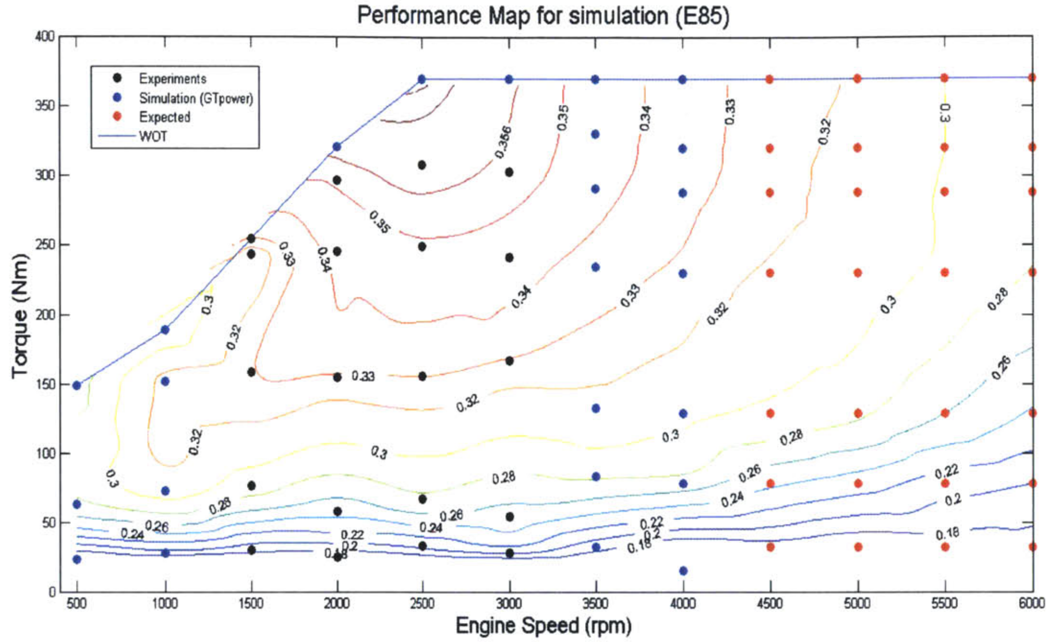

Figure 34. Extended performance map

In Figure 34, the black points are data collected from engine experiments, blue points are data generated by simulation, and red points are values assumed. Expected efficiency values at higher engine speeds were chosen so that they give appropriate decreases in efficiency as the engine speed increases. Mechanical friction, pumping, and accessory losses all increase with higher engine speed.

<u>Engine Scaling</u>

To define downsized engine maps, the original engine map was scaled for use in Autonomie. The engine map based on BMEP was used as the reference engine, since it can be scaled by simple relationship:

$$\mathrm{BMEP} = \frac{(\mathrm{Torque} * 2 * \mathrm{pi} * 2)}{V_d},$$

where V_d is a displaced volume of the engine in dm^3. 300 cc and 400 cc per cylinder, the two typical sizes of a small engine, were chosen so that total displaced volumes of the scaled engines were 1.2L and 1.6L respectively.

7.3 Simulation Results

<u>Ethanol Fraction Calculations</u>

To determine the amount of energy required to drive the vehicle in a particular driving cycle, Autonomie calculates the required engine speed and torque output at each time step. Therefore, simulation results from Autonomie incorporate the fuel consumption rate of the engine at specific points on the engine performance map. Using this information, we determine the amount of ethanol fuel required to avoid knock in a driving cycle simulation.

Required ethanol fraction for knock suppression was interpolated and extrapolated from the knock onset line of various fuels determined from our previous experiments. A MATLAB code was written so that it takes engine speed and torque as input and generates ethanol fraction as output. Once ethanol fraction is known at a specific point, the simulated result of gasoline consumption was divided into two parts by the calculation below. Since at a given simulated time interval of 0.1s, energy released by gasoline-ethanol blend should be equivalent to that of gasoline,

$$\dot{m}_{sim} * 0.1s * LHV_{gasoline} = V_g * \rho_g * LHV_{gasoline} + V_e * \rho_e * LHV_{ethanol},$$

where $\dot{m}_{sim}$ is gasoline consumption rate of the engine at a given time interval, $LHV_{gasoline}$ and $LHV_{ethanol}$ are lower heating values of gasoline and ethanol fuel, $V_g * \rho_g$ and $V_e * \rho_e$ represent the mass of gasoline and ethanol. Ethanol volume can be written as

$$V_e = \frac{E_f}{(1 - E_f)} * V_g$$

where E_f is ethanol fraction required to suppress knock. Then, V_e can be substituted to the equation (i) and then the expression for V_g is

$$V_g = \frac{\dot{m}_{sim} * 0.1s * LHV_{gasoline}}{\rho_g * LHV_{gasoline} + \frac{E_f}{(1 - E_f)} * \rho_e * LHV_{ethanol}}$$

This process can be run for the complete time frame of a specified driving cycle, and the volume of ethanol and gasoline required in a driving cycle can be determined.

Driving cycles

Driving cycles such as UDDS (Urban Dynamometer Driving Schedule), HWFET (Highway Fuel Economy Driving Schedule), and US06 (Supplemental Cycle) were chosen as the cycles to be simulated in Autonomie. They are standard driving cycles used by EPA (Environmental Protection Agency) for fuel mileage and emission testing. UDDS represents city and suburban driving conditions for light-duty vehicles, and HWFET represents highway driving conditions under 60 mph. Results from those two cycles are used for calculating MPG (Mile Per Gallon) values. US06 was used to test engines in a more aggressive acceleration and higher speed driving cycle. Figures 35~40 show vehicle speeds for each cycle and operating points on the performance maps.

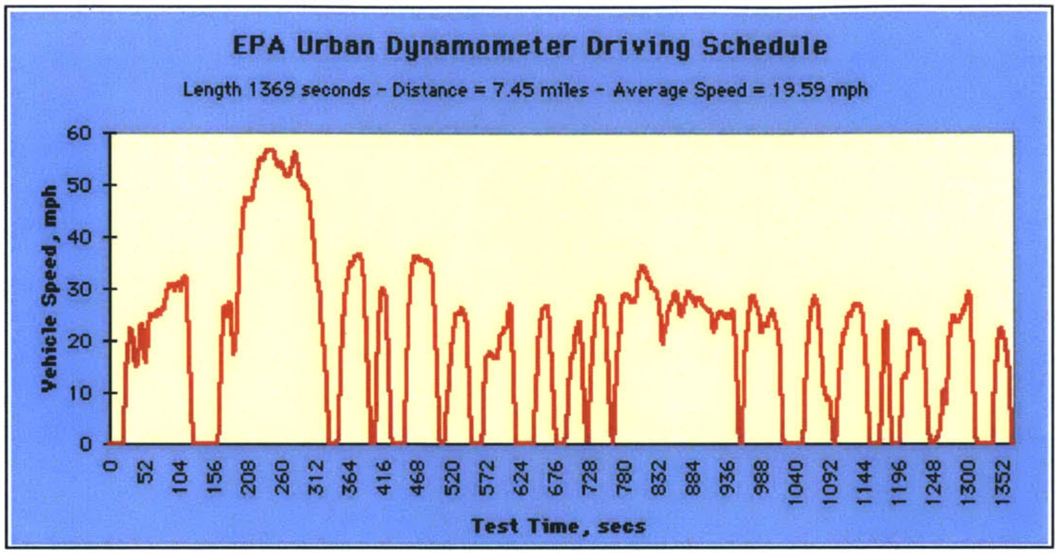

Figure 35. UDDS (Urban Dynamometer Driving Schedule) speed vs test time

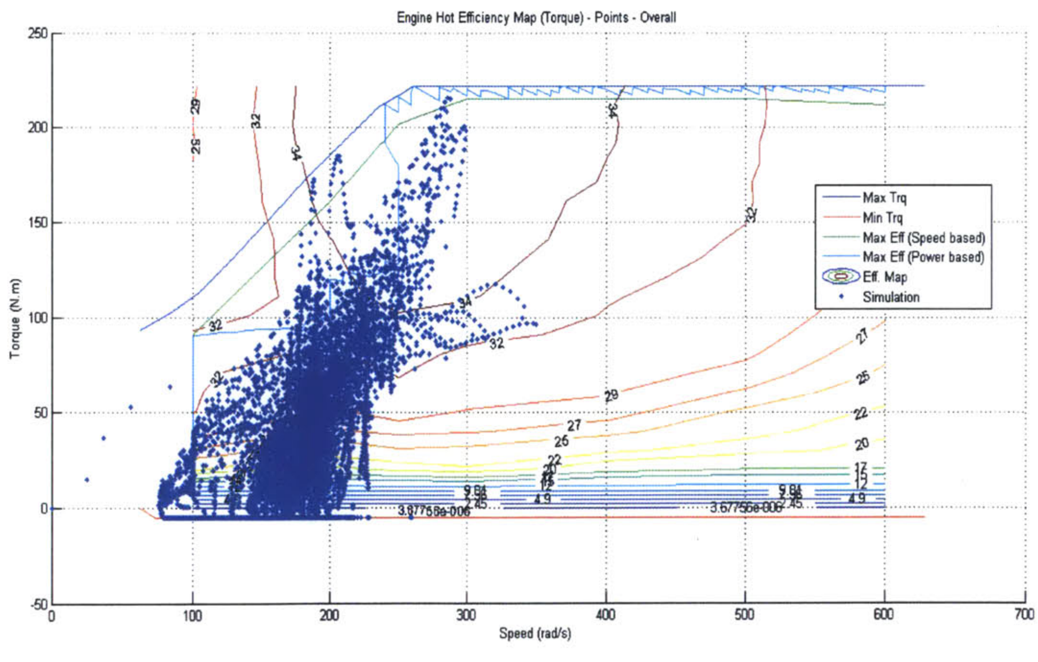

Figure 36. UDDS operating points on the downsized (1.2 L) engine map

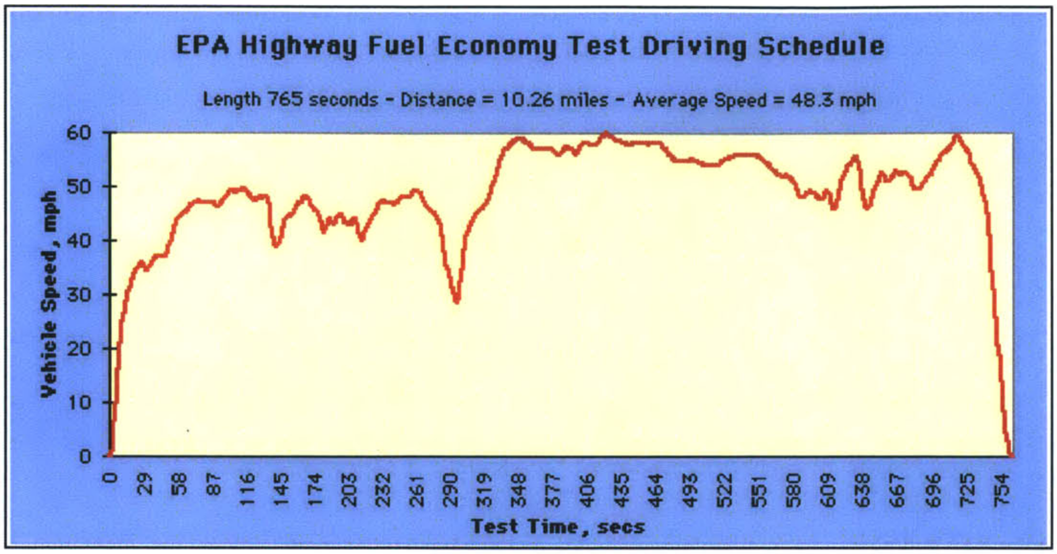

Figure 37. HWFET (Highway Fuel Economy Test) speed vs test time

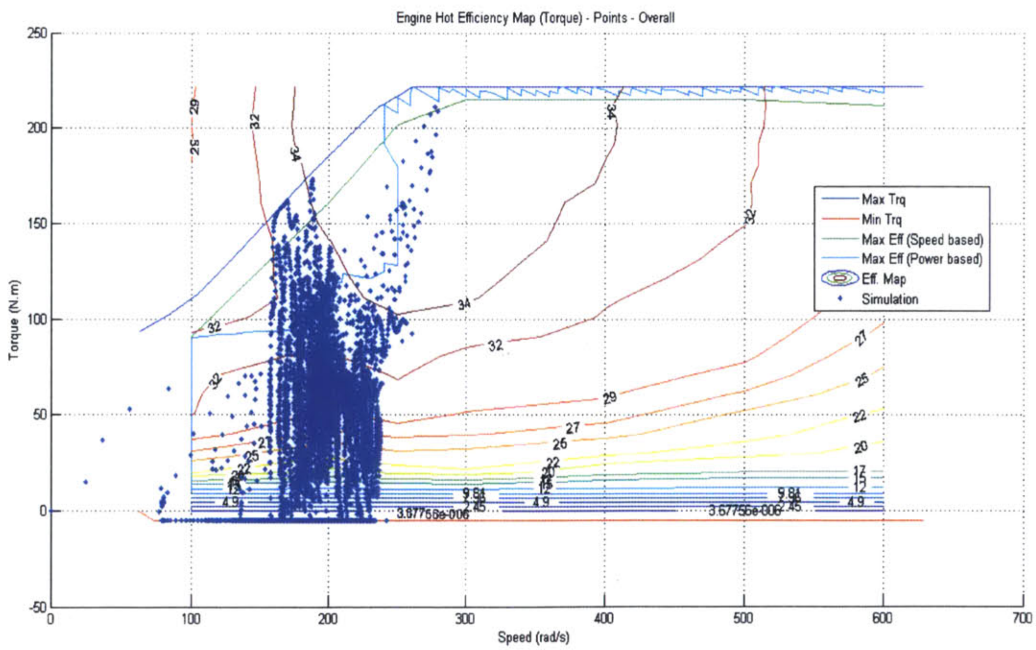

Figure 38. HWFET operating points on the downsized (1.2 L) engine map

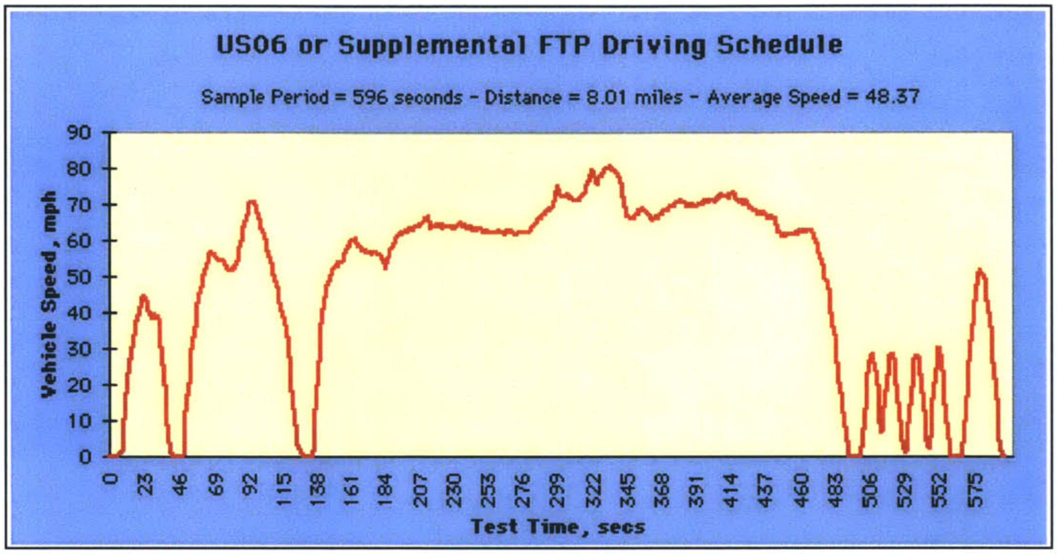

Figure 39. US06 (Supplemental FTP Driving Schedule) speed vs test time

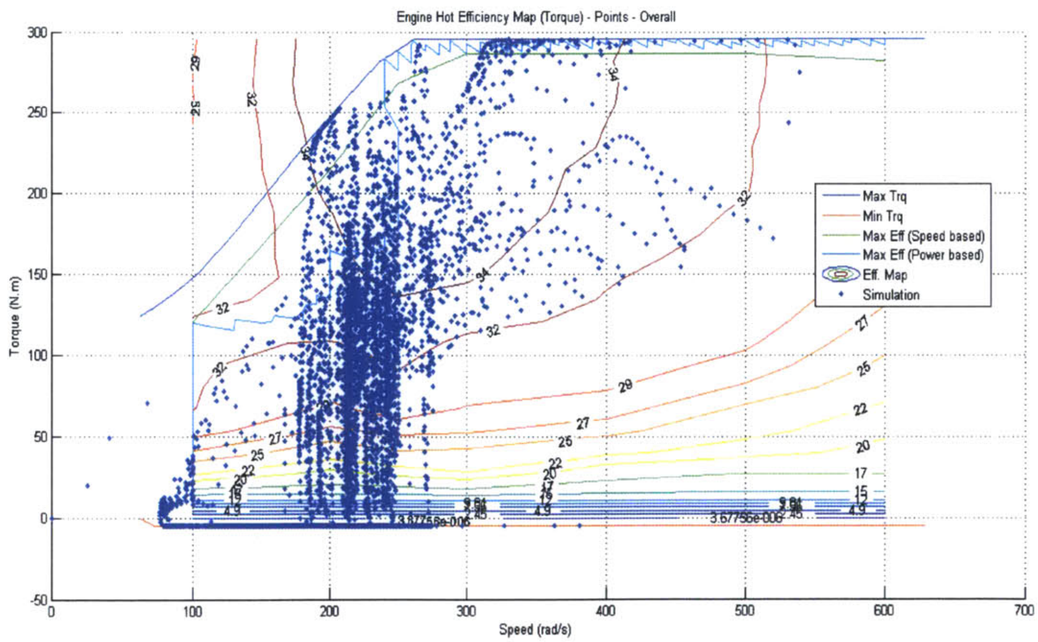

Figure 40. US06 operating points on the downsized (1.2 L) engine map

Engine Downsizing

To explore the effect of engine downsizing on the ethanol fraction required in a given driving cycle in these initial simulations, two downsized turbocharged engines were run in the Autonomie. As seen in the Table 8, maximum power output of the downsized 1.2L engine were similar to the maximum power of the naturally aspirated 2.5 L engine in Toyota Camry.

Table 8. Results: Engine Downsizing

Downsized engines				
	Toyota Camry	Downsized 1.2L	Downsized 1.6L	2.0L Reference
Max Power (kW)	132.7	139.3	185.8	232.2
Max Torque (Nm)	230	221.7	295.6	369.5
Driving cycle simulation Results				
	MPG	Total Fuel Consumption for driving 1000 miles (in gallons)	Ethanol Consumption for driving 1000 miles (in gallons)	Ethanol Fraction in Volume
1.2 L Engine				
UDDS	30.2	33.1	1.05	3.2 %
HWFET	44.6	22.4	0.46	2 %
US06	25.7	38.9	6.44	16.6 %
Adjusted Total Mileage	28.3			
1.6 L Engine				
UDDS	28.3	35.3	0.20	1 %
HWFET	41.4	24.2	0.24	0.5 %
US06	25.7	38.9	4.3	11.1 %
Adjusted Total Mileage	26.4			
2.0 L Engine				
UDDS	26.8	37.3	0.07	0.35 %
HWFET	38.8	25.8	0.03	0.07 %
US06	25.1	39.8	2.93	7.4 %
Adjusted Total Mileage	24.9			

In the table 8, mileage, total volume of fuel consumed for driving 1000 miles, and ethanol fraction required to operate the engine without knock were calculated. Then, MPG values from the UDDS and HWFET cycles were used to get a combined MPG:

$$MPG_{combined} = \frac{1}{\frac{0.55}{MPG_{UDDS}} + \frac{0.45}{MPG_{HWFET}}} \text{ [9].}$$

An adjustment factor of close to 0.8 multiplies the combined mileage value to obtain the "adjusted" fuel economy used on the official vehicle label. The official adjusted value for the 2013 Toyota Camry is 28.4 MPG [10]. It is only 0.5 % higher than the adjusted combined MPG from the simulated 1.2 L downsized engine.

Using the 2.0 L turbocharged engine, the amount of ethanol required to operate the engine without knock in the UDDS and HWFET cycles was less than 0.5 %. It was 7.4 % for US06 cycle, which has the highest ethanol use of all cycles. As the engine is downsized, the ethanol fraction required for each cycle increased. The amount of ethanol required to run US06 cycle increased from 7.4 % to 16.6 % with turbocharged engine downsizing from 2.0 L to 1.2 L. However, the adjusted total MPG increased from 24.9 to 28.3, about a 14 % increase. This is because the downsized engine could be operated in a more efficient region of its performance map.

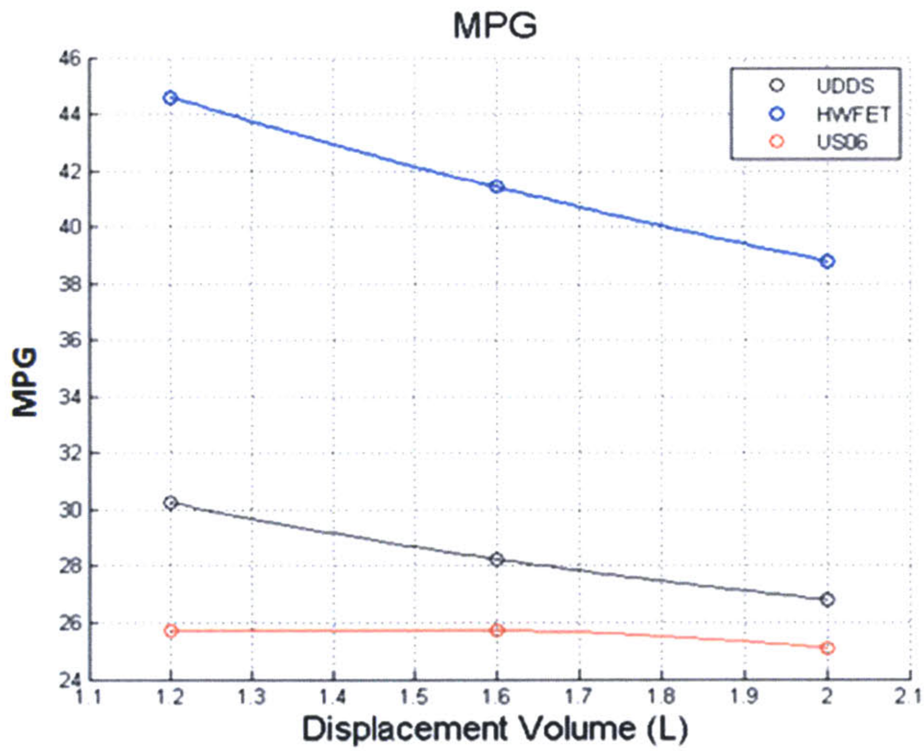

Figure 41. Mileage increase with downsizing

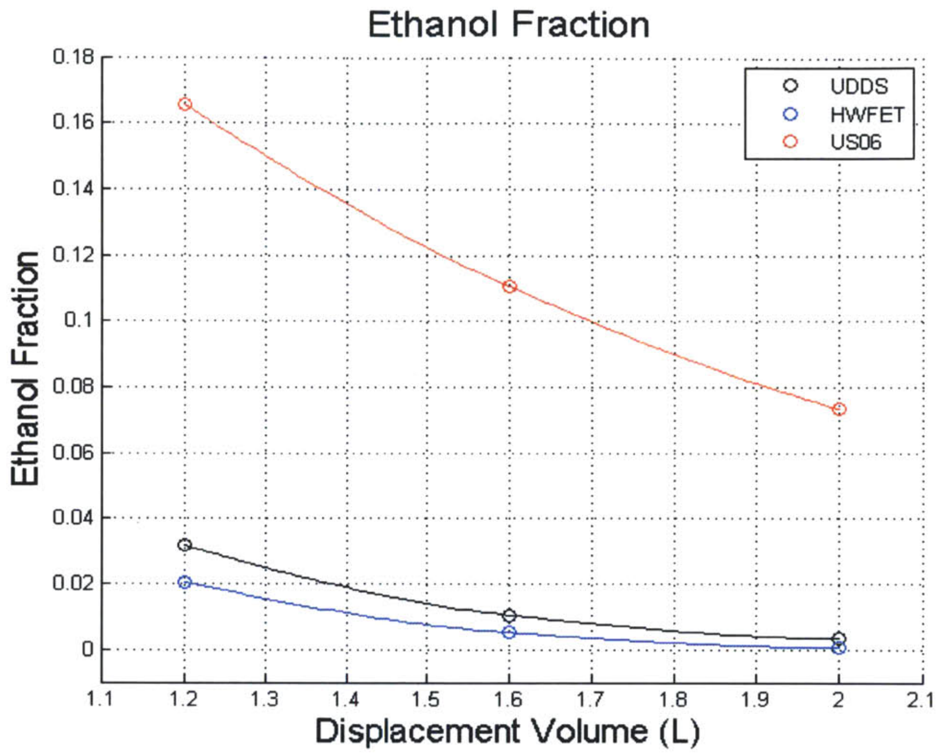

Figure 42. Change in ethanol fraction required to suppress knock with downsizing

<u>Spark Retard</u>

To run the driving cycle simulation with spark retard, the RON in Figure 28 had to be converted to the ethanol fraction. The relationship between RON and ethanol fraction could be found in the SAE paper by Stein. [7] In the paper, different base gasolines were tested for different ethanol fractions, so RON for our blendstock gasoline had to be interpolated. Base gasolines of RON 88 and RON 92 data were used to interpolate RON 90.5 case, which is the RON of our base gasoline. The red line in the Figure 43 shows interpolated RON vs. ethanol fraction curve.

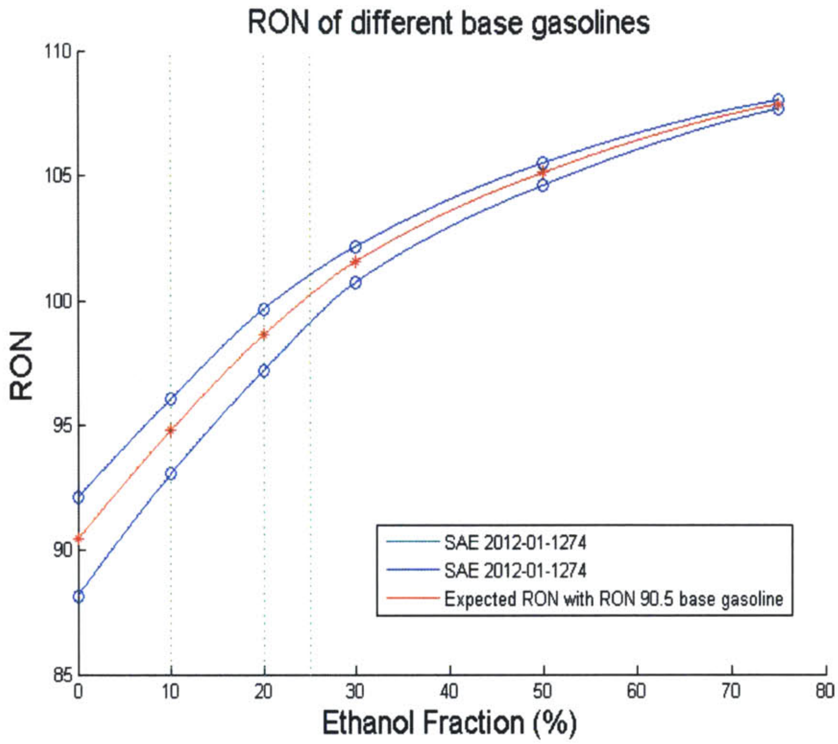

Figure 43. RON vs Ethanol fraction

Then, from the information in the Figure 43, ethanol fraction required for different BMEP level at a given spark retard could be calculated. Figures 39 and 40 show ethanol fraction required with 2.5 and 5 CAD spark retard respectively. Above 1900 kPa BMEP, experiments could not be done because of the engine's peak pressure limitation. Therefore, the graph was extrapolated to estimate the ethanol fraction required at higher loads.

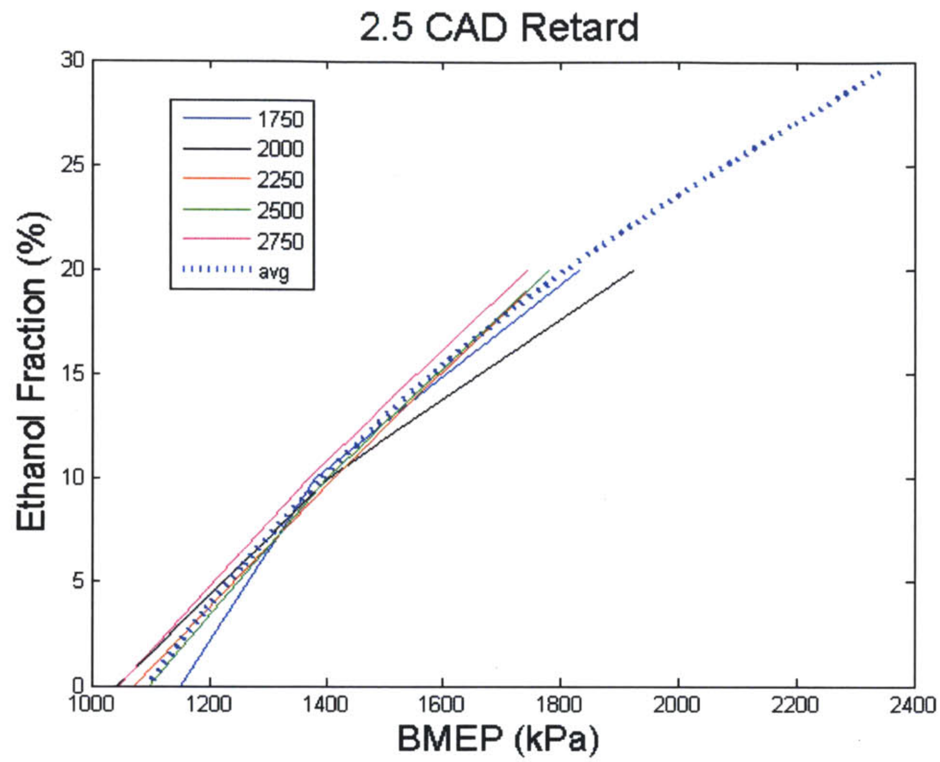

Figure 44. Ethanol fraction required vs. BMEP at 2.5 CAD retard

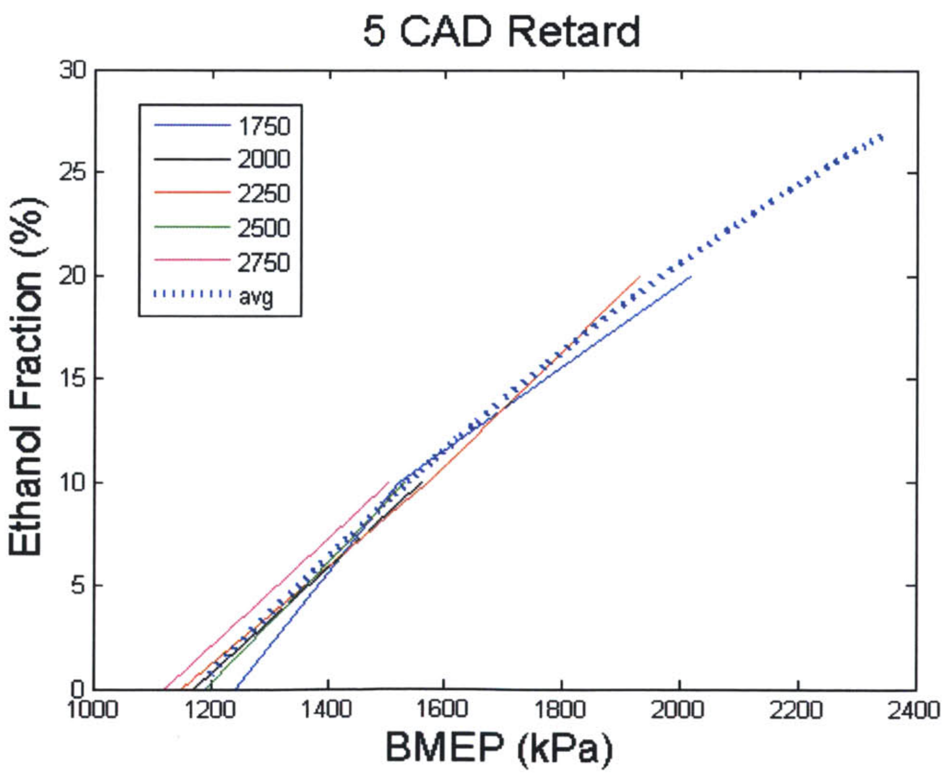

Figure 45. Ethanol fraction required vs. BMEP at 5 CAD retard

It was necessary to add a degradation factor to the performance map to replicate the efficiency loss with spark retard. Graph 14 shows how efficiency decreases with retarding the spark timing. Table 9 numerates the relative efficiency drop in percent with spark retard from 2.5 to 10 CAD. The efficiency drop with spark retard in a boosted turbocharged engine will be discussed in the later section.

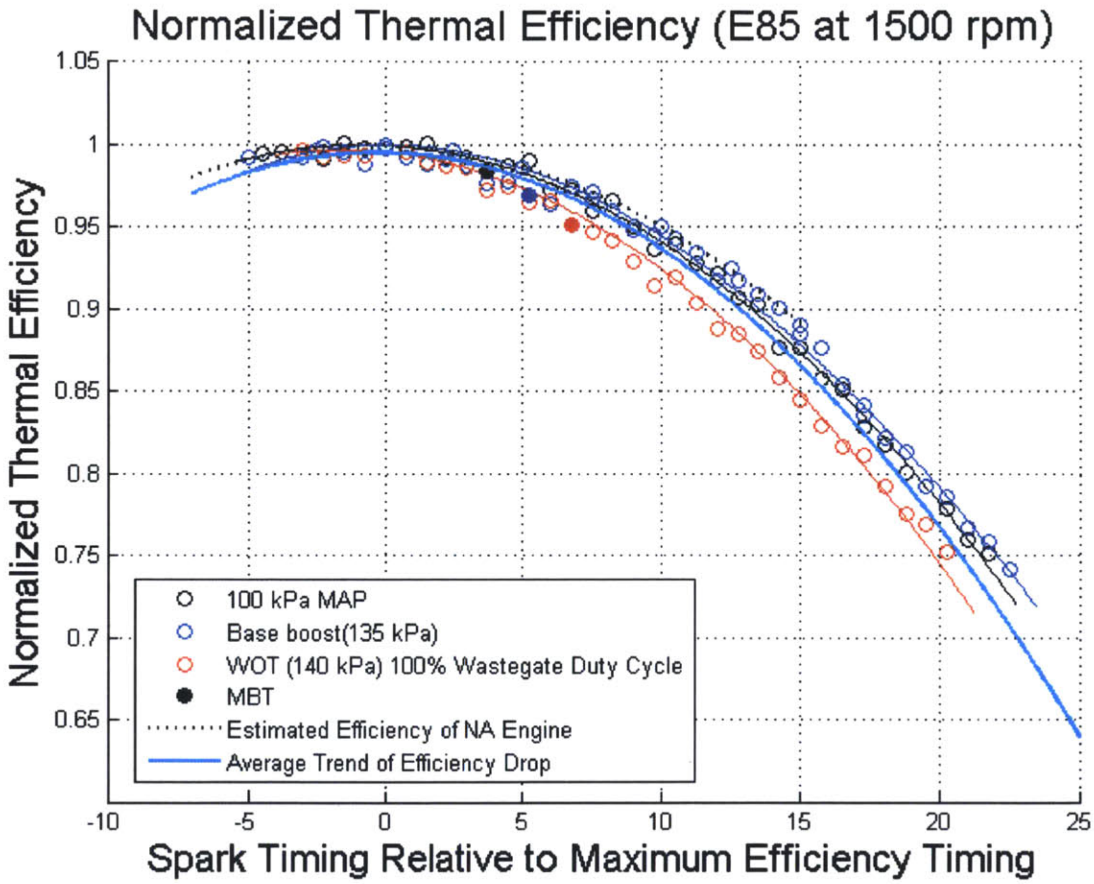

Figure 46. Efficiency drop of a turbocharged engine with spark retard

Table 9. Efficiency Drop with Spark Retard

Retard (CAD)	2.5	5	7.5	10
Efficiency Drop (%)	0.43	1.55	3.37	5.88

These degradation factors were incorporated into the engine performance map of the downsized 1.2 L engine we are using in the vehicle simulation. For example, if 2.5 CAD of retard was allowed, the degradation factor was applied to that part of engine efficiency map where low grade gasoline (RON 90.5) is knock limited. The same was done for 5, 7.5, and 10 CAD retard cases. Finally, ethanol fraction and miles per gallon changes with spark retard were determined. Figures 47 and 48 show how ethanol fraction and spark retard influences mileage.

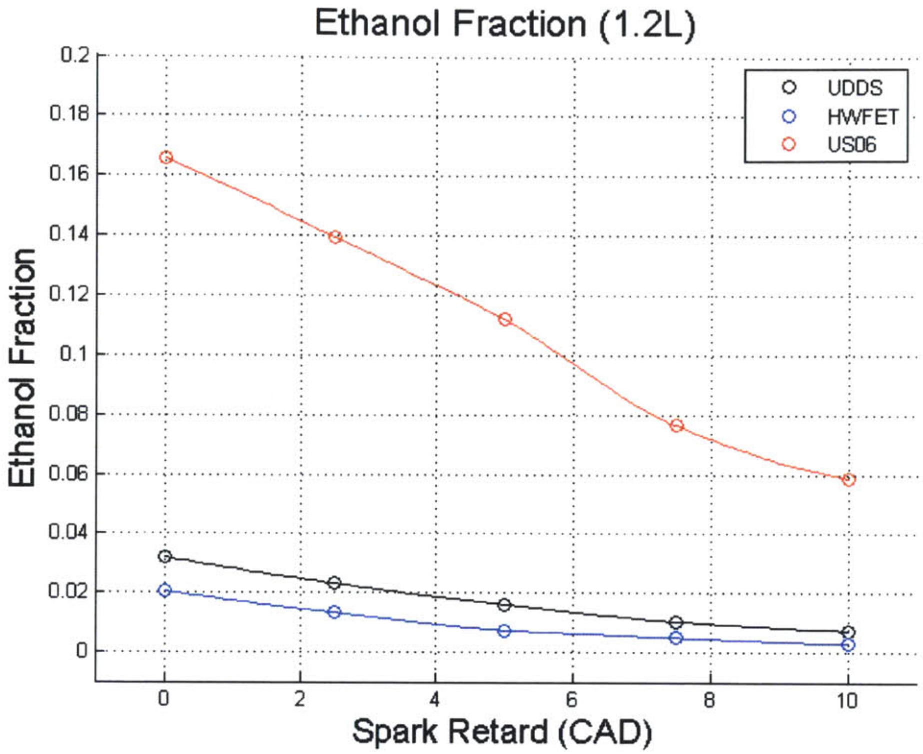

Figure 28. Ethanol fraction change with spark retard

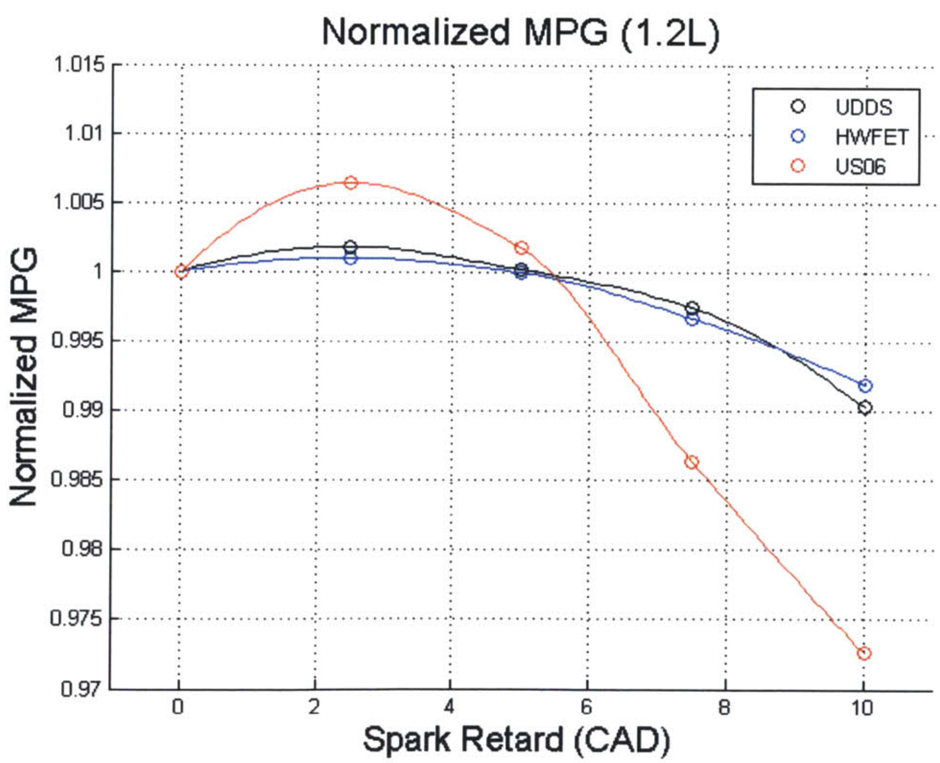

Figure 29. Mileage change with spark retard

As expected, spark retard allowed the engine to operate without knock but (at these conditions) with some loss in efficiency. The ethanol fraction could be decreased from 16 % to 6 % with 10 CAD retard the US06 driving cycle. In Figure 41, it can be seen that moderate retard of spark (2.5~5 CAD retard) results in an increase in MPG. This is because the ethanol fraction decreases, and the energy content of the fuel is larger as the gasoline replaces ethanol: the efficiency loss is not significant with moderate retard in spark timing. So, MPG measured by volume of the ethanol-gasoline mixture is larger.

8. Conclusions and Future Work

8.1 Conclusions

Knock Onset and Efficiency of a Turbocharged Engine

Thermal efficiencies of the turbocharged engine from part load to WOT condition at speeds from 1500 rpm to 3000 rpm were determined with both RON 96 gasoline and E85 ethanol blends. At a fixed speed, efficiencies of the engine using two fuels were compared. Efficiency of the engine was "universal" in the performance map as long as the engine was not knock limited.

Knock limits of two gasoline blends and ethanol blends with different ethanol contents were tested. The knock limit was defined as a load level where the engine could not be operated at maximum brake torque spark timing. RON 91 gasoline was knock limited at 10 bar BMEP, and RON 96 gasoline was knock limited at 13 bar BMEP. Ethanol has to be added to operate the engine at MBT timing above 13 bar BMEP.

Ethanol Fraction

The amount of ethanol required to suppress knock at engine operating conditions where, with gasoline, knock would occur were determined. At a compression ratio of 9.2 with 1 bar maximum boost, 20~35 % of ethanol enables knock free operation of a turbocharged engine. The amount of ethanol required to suppress knock closely depends on the speed and boost level. This is because the amount of boost created by a turbocharger depends on the speed, and higher boost requires more ethanol to suppress knock.

The knock limit of the engine can be further pushed by retarding the ignition timing. With 20 % ethanol content in RON 91 gasoline, 5 degree retard in a spark timing increased the knock limit from 16 bar BMEP to near 20 bar BMEP. However, the efficiency of the engine decreases with retarding the spark timing. About 5 CAD retard in a spark timing results in an efficiency loss of 1.6 %.

Engine in vehicle simulation was used to estimate the amount of ethanol required to operate the engine without knock in various driving cycles. 2.0 L engine with 1 bar maximum boost required about 7.4 % of ethanol fraction in volume for US06 driving cycle. Less than 1 % of ethanol was required for city and highway driving cycles. US06 represents more aggressive driving style with high acceleration, so the engine operates in a knock limited region more often, which results in a high ethanol fraction.

As the engine size was reduced from 2.0 L to 1.2 L, the fraction of ethanol required increased about twice for US06 cycle. Other cycles showed a significant increase in the ethanol fraction too. Even though the engine operates in a more efficient region with downsizing, more ethanol has to be injected to suppress knock.

Different degrees of spark retard were applied to reduce the amount of ethanol required. Spark retard up to 5 CAD increases mile per gallon of the vehicle. This is because efficiency loss is insignificant with 5 CAD retard, but ethanol fraction decreases significantly therefore increasing the energy content of the fuel per volume. This suggests that the spark retard and ethanol injection can be incorporated together to optimize the efficiency of the engine.

Turbocharged Engine Operation

Not only knock but also peak cylinder pressure limits the operation of a turbocharged engine at maximum efficiency. Above 18~19 bar BMEP, spark timing has to be retarded to lower the peak cylinder pressure. The spark retard due to peak pressure limit slightly decreases the efficiency of the engine above the peak pressure limit line in the performance map.

Efficiency degradation due to spark retard is higher with a high boosting condition. This is because spark retard increases exhaust enthalpy, which drives the turbine to create more boost. Therefore, intake manifold air pressure increases and more fuel is consumed. Due to change of manifold air pressure with spark timing, maximum brake torque timing is not necessarily the maximum efficiency point for a turbocharged engine. This feedback effect of spark retard was higher when the wastegate actuator was manipulated to generate more boost.

8.1 Future Work

Expending the Test Variables

The current engine has a fixed boost level and a compression ratio. Higher compression ratio and higher boost pressure will allow the engine to operate in a more efficient region. However, higher compression ratio and higher boost pressure will make the engine more prone to knock. Therefore quantifying ethanol requirements and efficiency gain at higher compression ratio and higher boost should be done.

Methanol, like ethanol, is one possible additive to gasoline to suppress knock in a spark ignited engine. The methodology used with ethanol can be applied with methanol to quantify knock suppressing and operating characteristics of methanol in a spark ignited turbocharged engine. Also, water injection with high ethanol content is another potential for reducing knock.

Technology Development and Cost Analysis

Direct injection of ethanol can be incorporated to port fuel injected or direct injected spark ignited gasoline engine. To incorporate the technology, it is necessary to have a separate fueling system including ethanol fuel tank and fuel line. The common fuel rail can be used to prepare a pre-mixed fuel mixture with desired ethanol fraction. This concept can be further developed in a near future and tested. Also, cost analysis should be done on the technology.

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