CRC Project No. CM-137-11-1b

REVIEW TO DETERMINE THE BENEFITS OF INCREASING OCTANE NUMBER ON GASOLINE ENGINE EFFICIENCY: ANALYSIS AND RECOMMENDATIONS – Tasks 2-5

September 2012

COORDINATING RESEARCH COUNCIL, INC.

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Final Report

Review to Determine the Benefits of Increasing Octane Number on Gasoline Engine Efficiency: Analysis and Recommendations – Tasks 2-5

CRC Project No. CM-137-11-1b

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EXECUTIVE SUMMARY

E.1 OVERVIEW

The overall objectives of Project CM-137-11-1 are to establish the relationship between engine (and vehicle) energy efficiency for a spark ignition (SI) engine and the octane number and composition of the fuel. This project is sponsored by the Coordinating Research Council (CRC) and is divided into several tasks. The detailed literature review conducted under Task 1 of this project[1] revealed 45 technical papers that covered various aspects of the relationship between engine/vehicle efficiency and fuel octane number and composition. The papers covered a wide range of engine types, fuel formulations, engine operating conditions and testing constraints. Results were also presented using various metrics, e.g., indicated and brake mean effective pressure, indicated and brake specific fuel consumption, relative and absolute efficiency.

The Task 1 review found that the relationship between octane number and efficiency was influenced by a number of intermediate variables covering engine type, operating condition, and fuel formulation. Hence, it was not possible to simply collate the data from the papers and develop an average relationship; considerable effort was required to organize the reported data into subsets of similar engine types and operating conditions, and to convert the different data metrics into a comparable set of metrics.

The analysis was also assisted by inputs from technical experts at three auto-manufacturers (GM, Ford and Toyota) obtained from interviews conducted by H-D Systems staff. All three auto-manufacturers interviewed suggested that this analysis for CRC use only the results from the engine studies because they anticipated that in vehicle studies, many variables were vehicle calibration dependent and therefore, largely uncontrolled in these studies. As a result, this study focuses on engine data where all or most of the data on variables relevant to the study was controlled and/or documented.

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[1] The literature review is available at the CRC website as Review to Determine the Benefits of Increasing Octane Number on Gasoline Engine Efficiency (CM-137-11-1) Task 1 report.

E.2 IDEAL ENGINE EFFICIENCY AND RELATIONSHIP TO FUEL OCTANE NUMBER

The thermal efficiency of an engine, using an ideal model of the SI engine operating on the Otto cycle, is a function of the compression ratio and is given by the relationship:

$$\eta_t = 1 - (1/CR)^{\gamma-1} \qquad (1)$$

where η_t is the thermal efficiency, CR the compression ratio, and γ the ratio of specific heat at constant volume to the specific heat at constant pressure for the combustion gas.[2] In equation (1), the efficiency is referred to as 'indicated efficiency' since it does not account for the effects of friction and pumping losses. This equation is true for naturally aspirated SI engines, however boosted engines require more complex equations relating the efficiencies of the compressor and turbine or supercharger to engine speed and load to describe the net thermal efficiency of the reciprocating engine and turbo/super-charger combination.

The ideal model of the Otto cycle assumes instantaneous combustion of the air-fuel mixture, but actual combustion occurs over finite time and the spark timing affects efficiency. The highest torque (and hence, the highest efficiency) at a given engine speed in revolutions per minute (RPM) and at wide open throttle (WOT) occurs at a spark timing advance level from piston top dead center (TDC) termed as 'minimum for maximum brake torque'(MBT) timing. Higher and lower levels of spark advance away from the optimum result in reduction in torque and efficiency. However, the maximum level of spark advance possible without causing abnormal combustion, or knock, in an engine with a specific configuration is related to the fuel octane number. Hence, there is a positive correlation between fuel octane number and efficiency as long as the knock limited spark advance (KLSA) is less than MBT spark advance. The fuel octane number when the KLSA is equal to the MBT at WOT conditions is termed the octane number requirement (ONR) of the engine.

Although the ideal SI engine's efficiency relationship is specified by equation (1), the value of γ to be used depends on the fuel composition and air-fuel ratio of the engine. Air has a γ of 1.4; but for a stoichiometric mixture, the γ values are about 1.3 for the compression stroke and about 1.2

[2]In the case of other SI engine cycles like the Atkinson cycle or Miller cycle, the relationship is somewhat different and involves both the compression and expansion ratio, but the trend with compression ratio is similar to that of the Otto Cycle.

for the expansion stroke. Fitting actual pressure-volume data from engines has typically provided average γ values of 1.25 to 1.3 and values such as 1.28 are commonly used in the literature. However, data from a very detailed modeling analysis by Nissan provided an indicated efficiency curve with CR that we were able to replicate using a γ value of 1.265.

The relationship between engine ONR and CR is defined by a number of engine design and operating variables including combustion chamber shape and size, spark plug location, cylinder head and wall temperatures, intake air temperature, etc. (The list of variables is intended to be illustrative, not comprehensive). The relationship of engine ONR to these variables has been established empirically for specific engine designs, but no simple theoretical relationship has been determined. However, a key mechanism used to control knock in SI engines is the spark timing, which impacts the peak and average cycle pressure, and hence, the torque output of the engine as well as its efficiency. Maximum torque at any operating point (and hence, maximum efficiency) is obtained at MBT spark timing, and any spark timing retard from this MBT timing, normally measured in crank angle degrees before TDC, causes a reduction in torque as well as a reduction in the engine octane number limit that we have termed as ONL for convenience. The change in mean effective pressure as a function of spark timing is well understood, as is the nature of the relationship.

A study by MIT (Ref. 3) provided a robust relationship between spark timing as measured in degrees retard from MBT timing to Net Indicated Mean Effective Pressure (NIMEP). NIMEP is the term used to describe average cylinder pressure over the entire two revolutions (720°) of the Otto Cycle. Based on the data presented, we derived the following equation for wide open throttle:

$$\eta(\theta)/\eta(MBT) \approx NIMEP(\theta)/NIMEP(MBT) = 1 - 0.5125 * 10^{-3} * (\theta_s - \theta_{s,\,mbt})^2 \quad (2)$$

where θ_s is the spark timing and $\theta_{s,\,mbt}$ is the MBT spark timing. This equation assumed that at WOT, NIMEP is approximately equal to IMEP since pumping losses are quite small, typically less than 2% of IMEP. Equation (2) shows that a 10° spark retard from MBT timing results in a 5.1% relative loss in efficiency and a 20 degree retard reduces relative efficiency by 20.5%. Hence, small changes in spark timing from MBT have very small efficiency effects but the non-linear nature of the relationship shows that retard over 10° can cause significant losses in efficiency.

E.3 OCTANE NUMBER MEASUREMENT AND FUEL OCTANE NUMBER

A fuel's propensity to auto-ignite and cause "knock" is quantified in terms of the fuel's octane number (ON). The ON is measured on a standard engine called the CFR (for Cooperative Fuel Research) engine, where the engine is set to run at fixed intake conditions and RPM. Modern engines do not resemble the CFR engine and the measurement conditions for Research and Motor Octane Number (RON and MON) do not represent typical engine intake air temperatures, cylinder pressures, or operating RPM. Hence the relationship between CR and fuel octane number could be substantially different for modern engines.

The relevance of RON and MON ratings to modern engines has been examined at the research engine, production engine and vehicle levels and documented in a number of papers, and a new rating termed octane index (OI) has been derived. The basic principle is that the OI of a fuel for modern engines can be expressed as a linear combination of the RON and MON values with K as the weighting factor such that:

$$\begin{aligned} OI &= RON - K*(RON - MON) \\ &= RON - K * S \qquad (3) \end{aligned}$$

The term (RON – MON) in Equation 3 is called the Sensitivity (S) of the fuel. In a study by Shell (Ref.4), 21 fuels with a wide range of RON and MON were tested in two single-cylinder engines. The study found that the engine torque increased with spark advance, flattening out at a maximum value at the MBT spark level and declining with further increases in spark advance, as predicted by Equation (2). Many fuels had too high a knock intensity at levels of spark advance much less than the MBT level so that they were tested only to the knock limit.K, as defined in the Equation (3), was determined for each RPM/load test condition and was found to be negative at low RPM (<2000) and high load. K increased with RPM, but decreased with increased CR and load. The K value at any speed/load condition appeared to be largely independent of fuel type or RON. The result of a negative K value is that for a fuel with Sensitivity (S)>0,the OI is **greater** than either the RON or MON of the fuel. In general, K values that are small or zero imply that fuel RON is much more important than MON for engine knock suppression, and a negative K indicates that lower values of MON are preferred.

Analysis conducted at MIT (Ref. 6) and at other research facilities confirmed the finding that K is negative for modern vehicles. The MIT analysis also found that K was:

- Only weakly dependent on spark plug location and CR (varying from 9.8 to 13.4).
- Non-linearly dependent on air-fuel ratio, being highest at lambda = 1 but decreasing at both richer and leaner air-fuel ratio.
- Increased linearly with intake air temperature
- Decreased almost linearly with increased intake air pressure (boost).

The finding that K decreases with intake boost is particularly interesting, since it implies that more sensitive fuels are well suited to turbocharged engines as their OI is significantly higher than the RON of the fuel. The MIT paper indicated that at a boost level of 1.4 bar, the K value was -0.65. For a typical retail gasoline with a Sensitivity of about 10 octane numbers, the -0.65 estimate indicates that the OI is 6.5 numbers above the RON at this boost level.

However, a recent study by Toyota found that hot spot related pre-ignition occurs at much lower hot spot temperatures in a high CR engine than in a low CR engine, and that the pre-ignition temperatures for this engine with the different fuels showed MON to have a stronger relationship than RON to pre-ignition temperature. The study also found that increasing MON by 16 ,(from 80 to 96) resulted in pre-ignition temperature increasing by about 200° C. Hence, the study concluded that a higher MON was useful in suppressing hot spot related pre-ignition.

E.4 EFFECTS OF FUEL COMPOSITION

Fuel composition affects the fuel octane index and latent heat of vaporization, and can also directly influence engine efficiency by changes in the mixture burning velocity. Researchers (Ref. 10) find that paraffin compounds have high auto-ignition resistance at higher temperatures where the kinetics of paraffin decomposition have a negative temperature coefficient. Hence, the reference fuels which are a mixture of paraffin compounds are more resistant to knock at higher temperatures similar to those encountered under the MON test conditions than olefin and aromatic compounds. At conditions closer to those of the RON test, the olefin and aromatic compounds have better knock resistance than paraffin compounds, and mixtures containing higher olefin and aromatic compounds can have an octane number higher than the octane number of the reference fuel, leading to a negative K value.

In an analysis of fuel effects over a range of air-fuel ratios, an ExxonMobil/Toyota paper (Ref. 11) examined the burning velocities of fuels with different mixes of aromatic, olefin and paraffin compounds at λ values ranging from 1 to 1.7. The paper stated that laminar burning velocities varied by more than 11% between the slowest and fastest fuels, and that burning velocities correlate well with olefin content and higher fuel volatility. Tests on a variety of fuel formulations on an engine operating at WOT, 2400 RPM and $\lambda = 1$ showed that fuels with high burning velocities gave up to 3% higher torque at the same spark advance. The faster burning fuels also allowed higher spark advance at the knock limit.

In port fuel injected (PFI) engines, much of the fuel is inducted into the cylinder after it has vaporized due to heat transfer from the engine. In direct injection (DI) engines, the fuel is introduced into the cylinder in liquid form, and the vaporization of fuel causes a reduction in the air-fuel mixture temperature relative to mixture for a PFI engine, thereby reducing the octane number requirement of the engine. The latent heat of vaporization for gasoline is estimated at 420 kJ/kg and varies somewhat depending on gasoline composition. However, the introduction of ethanol has a particularly large effect on temperature since it has a latent heat of vaporization (LHV) which is approximately double that of gasoline. In addition, the stoichiometric air-fuel ratio for ethanol is 9.0, compared to 14.5 to 14.7 for gasoline, so that a larger quantity of ethanol is required at stoichiometric conditions for a given volume of air. Analysis shows that if all the heat of vaporization is taken from the air, the temperature drop with gasoline is about 20°C while the drop with ethanol is about 90°C. The lower temperature causes a significant reduction in the ONR of the engine.

E.5 MAJOR FINDINGS OF THE ANALYSIS OF ACTUAL ENGINE EFFICIENCY

The effect of increasing fuel octane number on engine and vehicle energy efficiency cannot be quantified in a simple way but requires a complex, nuanced answer. The following findings from the literature review and analysis are the basis for the conclusions of this report.

Relationship between Actual Engine Efficiency and Fuel Octane Number

1) The thermal efficiency of naturally aspirated engines primarily depends on the engine's compression ratio, but is also adversely affected by increasing surface to volume ratio of the combustion chamber which depends directly on the CR and inversely on the cylinder

displacement (and typically, the bore size). As a result, actual maximum indicated thermal efficiency is almost constant beyond CR values of 14 for small bore engines (<70mm) and almost constant beyond a CR of 16 for bore size over 85mm. The marginal thermal efficiency benefit of increasing CR declines continuously with increasing CR.

2) The engine octane number requirement (ONR) is primarily a function of the CR and to a slightly lesser extent, operating conditions of speed and air-fuel ratio. It is also a function of engine design variables such as the shape of the combustion chamber, efficacy of cooling, and bore diameter[3]. The highest octane number requirement occurs at low RPM (<1500) and WOT for naturally aspirated engines.

Naturally Aspirated Engines

3) The typical ONR for a port fuel injected naturally aspirated engine with a CR of 10 and a bore diameter of about 85mm operating at λ= 1, 1500 RPM is 100 to 102 RON at WOT. Smaller bore engines have a lower octane number requirement while larger bore engines have a higher octane number requirement. The relationship between ONR and bore diameter is not well quantified in the literature but may be around 2 to 3 octane numbers per 10mm, based on the data collected for this study. Hence a small bore engine (70mm) at the same conditions can have an ONR of 96 to 99 while a large bore engine (100mm) can have an ONR of 103 to 106.

4) The ONR increases with engine CR by about 4 to 5 octane numbers per unit CR increase. (It is not clear if this relationship holds at higher CR values over 13.) The ONR decreases with RPM and the ONR dependence appears to be approximately linear with time per revolution (the inverse of RPM) to about 40 ms (1500 RPM), but not all studies confirm this. The ONR declines by approximately 2 octane numbers per 10ms decline in the time per revolution. Hence the ONR declines by about 4 numbers between 1500 RPM and 3000 RPM (40ms to 20ms per revolution).

5) The engine actual octane number limit (ONL) is sensitive to spark timing advance and air-fuel ratio, and these are the two primary control levers used to prevent knock in production naturally aspirated engines. The ONL is reduced with spark retard from MBT values and this relationship is approximately linear, with the OR being reduced by 1

[3] The variables listed are illustrative but this not a comprehensive list.

octane point for every 1.5 ± 0.5 degrees spark retard. The relationship becomes more non-linear at retarded timing very close to MBT and at timing advance beyond MBT. The ONL at WOT is highest at λ of 0.95 to 1 and every 0.1λ richer or leaner than this level leads to about a 3 point reduction in ONL.

6) The influence of spark timing on torque and engine efficiency is very nonlinear with the first 5 crank angle degrees (CAD) retard from MBT reducing torque and relative efficiency by only about 1.3%, while a 10 CAD retard causes a 5.1% reduction. Hence, spark retard by 5° from MBT can reduce the engine ONL by 3 to 4 numbers while having only modest negative effects of about 1.3% on available torque and relative thermal efficiency.
7) With direct injection (DI) of the fuel into the cylinder, the latent heat of evaporation of the fuel cools the air-fuel mixture. The cooling power of gasoline for a stoichiometric mixture of air and fuel is about 24 kJ/kg air. The reduced temperature reduces the octane number requirement by one octane number for about 4.5 to 6 kJ/kg of cooling power. Hence direct injection of gasoline reduces the ONR by about 4 to 5 octane numbers. Ethanol has a cooling power that is about 4 times as high as gasoline, suggesting a potential ONL reduction of 16 to 20 octane numbers when a DI engine is fueled with neat ethanol.

Turbocharged Engines

8) The relationship between engine CR, spark timing, and engine ONL is more complex in a turbocharged engine since the reciprocating engine and turbocharger act as system. If the engine is not boost limited, the optimal combustion phasing for best torque is delayed relative to that for a naturally aspirated engine since there is a compromise between enthalpy flow to the turbine and the best torque achieved by the engine. At high RPM, the boost, air-fuel ratio, and spark timing at WOT have to be limited not only to avoid knock but also to limit turbine inlet temperature (which limits the employment of spark retard) and engine peak pressure, so that fuel octane number effects are very varied across different operating conditions.
9) The available data show that increasing fuel octane number increases mid-range torque (~ 3000 to 4000 RPM) by 2% to 3% per octane number in RON for a turbocharged PFI

engine, while increasing engine thermal efficiency by about 0.7% per octane point. Low RPM and high RPM torque are usually limited by other factors as well, so that the effect of octane number increase at these RPM levels may be engine design specific.

10) Increasing octane number can have a very significant effect on mid-range torque for turbocharged DI engines that are not boost limited. Limited data on DI engines show that an 8 increase in fuel RON can increase 3000 RPM torque by about 50%, or about 5% per octane number. Torque benefits at high and low RPM may be much smaller as they are subject to the same constraints as for PFI engines.

11) The use of high ethanol concentration blends (E30 and higher) with gasoline may be particularly beneficial to turbocharged DI engines for two reasons – the high RON and the high cooling power. Research shows that with high concentration ethanol blends, the combination of high boost and retarded timing allows knock free operation at very high brake mean effective pressure (BMEP) levels. Recent experimental work has shown the capability of an E50 blend to allow engine operation at 38 bar to 40 bar indicated mean effective pressure (IMEP), which is about twice the IMEP level of current gasoline turbocharged DI engines. These high IMEP values will not likely be reproduced in production engines, as ethanol has other limitations such as reduced hot spot pre-ignition temperature relative to hydrocarbon gasoline.

Special Conditions

12) The use of lean burn offers a substantial fuel economy opportunity especially at part load conditions, but is not used in the US due to difficulties in meeting emission standards. Under lean conditions with a homogeneous mixture, lower RON fuels have been found to give better engine efficiency because the lower octane number leads to a combustion mode described as spark initiated compression ignition of end gases that are not fully combusted with high octane number fuels. If lean combustion becomes viable, lower RON fuels may be more desirable. However, the limiting condition for engine ONR may still be at full load where stoichiometric or rich operation is desirable for maximum power, so that the fuel octane number will have to be tailored to this condition in actual practice.

E.5 CONCLUSIONS

One objective of the project was to assess the impact of increasing fuel octane number on engine and vehicle efficiency. The conclusions for a **4 to 5 octane number** increase are as follows:

1) In PFI naturally aspirated engines with mid-size (~85mm) and larger bore sizes and a compression ratio (CR) of 10, this octane number increase will facilitate a 1 point increase in CR, which can provide a 2% relative improvement in engine efficiency. For small bore sizes (~70mm), the increase in CR will be similar but the improvement in engine efficiency will be smaller at 1.3% to 1.4%.
2) For midsize and large bore naturally aspirated DI engines that have a CR of 11, this octane number increase will facilitate a 1 point increase in CR and allow engine efficiency to increase by 1.5% to 1.6%. Small bore engines will have an improvement of 0.8% to 0.85%.
3) Vehicle efficiency gains for vehicles with naturally aspirated engines can be somewhat larger than engine efficiency gains due to increased engine torque when CR is increased, permitting engine down-speeding. This can provide up to a 1% to 2% fuel economy gain over and above engine efficiency gains.
4) In turbocharged engines, the effect of this improved octane number could be through the increase in engine peak torque if the engine is not boost limited. In PFI engines, the 4 to 5 point octane number increase can provide 10% to 15% more torque. In a DI engine, this increase in octane number can translate into a 20% to 25% increase in torque. This torque increase is for operation at 2000 to 4000 RPM; at engine speeds below 1500 RPM and above 4000 RPM other factors control available torque increases so that no general conclusion is possible.
5) The torque increases with turbocharged engines could allow significant downsizing and down-speeding of the engine but the extent of downsizing and down-speeding is dependent on engine low RPM performance. As an example of the size of the potential benefit, a 5% to 7% fuel economy improvement is possible if the engine can be downsized by 20% to 25% to keep absolute mid-range torque constant.
6) If the octane number increase were to be derived by addition of ethanol to gasoline, larger torque increases may be possible in turbocharged DI engines due to the high latent heat of vaporization of ethanol. An E30 blend with 91 RON gasoline blend-stock could

potentially improve torque by 50%, but this level of increase is not proven in any vehicle application and could be limited by ethanol's lower resistance to hot spot pre-ignition.

7) Developments in engine technology will allow CR to increase in the future with no change to fuel RON and MON. The increased use of DI technology will allow CR increases of 1 unit. Engine downsizing, with resulting smaller bore sizes, as well as the use of cooled exhaust gas recirculation (EGR) and improved exhaust scavenging, may allow continuing increases in CR over the next decade with unchanged fuel octane number.

E.6 DATA GAPS AND RECOMMENDATIONS FOR FUTURE RESEARCH

Two types of data gaps have been identified. The first type is where the data exists but are not public. For example, the data set is on actual engine efficiency and octane number requirement is very limited across engine sizes and across the RPM range. Other key variables with limited data include the effect of spark timing as well as the effect of air-fuel ratio on fuel octane number requirements across a range of engine bore sizes and RPM. These data clearly exist for a wide variety of production engines; if manufacturers participate in sharing the data (the origin of the data can be obscured to maintain confidentiality), the database could pave the way for developing more robust relationships between efficiency and fuel octane number for different bore sizes and operating conditions.

The second type of gap is where the data available are too limited to provide specific conclusions. Data on turbocharged DI engines, especially those with small displacement, and their response to fuel octane number and heat of vaporization in terms of efficiency and torque improvement across a range of engine speeds (especially at low RPM),are very thin and often derived from experimental engines. Octane number improvements and improvements from ethanol blends appear to hold significant promise, but the available data are inadequate to estimate the benefits in a real vehicle application. Newer technologies are also not represented in existing data. The latest DI engines in Europe have sequential turbocharging with two stage cooling; experiments have been conducted with cooled EGR at all loads and speeds, but little information exists on the interaction with fuel octane number.

The testing of turbocharged DI engines of different sizes, and those equipped with newer technology such as sequential turbocharging, as well cooled EGR, can permit a look forward into the future interaction of fuel octane number and engine and vehicle efficiency. Future research should focus on these newest technologies for the greatest relevance to the future.

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1. INTRODUCTION

1.1 ANALYSIS OBJECTIVES

The overall project objectives are to establish the relationship between engine (and vehicle) energy efficiency for a SI engine and the octane number and composition of the fuel. This project is sponsored by the Coordinating Research Council (CRC) and is divided into several tasks. The detailed literature review[4] conducted under Task 1 of this project revealed 45 technical papers that covered various aspects of the relationship between engine/ vehicle efficiency and fuel octane number and composition. The papers by research topic are as follows:

- 10 papers on single cylinder research engines
- 24 papers on prototype or production engines
- 4 papers on vehicle tests
- 6 papers that use simulation models, and
- 1 review paper.

It should be noted that the research focus of these papers was not uniformly on the relationship between efficiency and fuel octane, and in many of the papers, the relationship was only incidental to the main focus of that study. The papers covered a wide range of engine types, fuel formulations, engine operating conditions and testing constraints. Results were also presented using various metrics, e.g., indicated and brake mean effective pressure, indicated and brake specific fuel consumption, relative and absolute efficiency.

The Task 1 review found that the relationship between octane number and efficiency was influenced by a number of intermediate variables covering engine type, operating condition and fuel formulation. Hence, it was not possible to simply collate the data from the papers and develop an average relationship; considerable effort was required to organize the reported data into subsets that had data for similar engine types and operating conditions, and to convert the different data metrics into a comparable set of metrics.

[4] The literature review is available at the CRC website as a separate Task 1 report.

The analysis was also assisted by interview inputs from technical experts at three auto-manufacturers (GM, Ford and Toyota) obtained from interviews conducted by H-D Systems staff. The interviews provided qualitative inputs useful to the study and manufacturers also provided two additional papers (References 32 and 37)released after completion of the literature survey in late 2011. No confidential data directly relevant to the analysis were used in this study so that all of the results of this study are based on literature in the public domain.

All three auto-manufacturers interviewed suggested that this analysis for CRC use only the results from the engine studies. The manufacturers anticipated that in vehicle studies, many variables such as spark timing and fuel enrichment, transmission shift points, and others were calibration-dependent and therefore largely uncontrolled in these studies. Hence, the relationships between vehicle efficiency (fuel economy) and octane number could not be derived in any consistent manner from these vehicle studies. As a result, this study focuses on engine data where all or most of the data on variables relevant to the study was controlled and/or documented. Results from simulation studies are used to develop the correct form of the theoretical relationships between variables used in the analysis, and are compared to results from tests in some cases.

1.2 FOCUS OF ANALYSIS

Task 1 of this study revealed that the engine operating points where octane number constraints are typically critical are at WOT and low RPM (<2000) for naturally aspirated engines, and at high boost and low RPM for turbocharged engines. The combination of stoichiometric operation at low RPM and wide open throttle is considered in some detail in this analysis since much of the literature on octane Sensitivity and knock is focused on these conditions. While the focus of this analysis is on WOT and low RPM conditions (1200 to 2000 RPM), efficiency implications at part load conditions and at high RPM conditions are also discussed in the following sections.

As noted, the issues related to knock and octane number are governed by many variables but the focus of this analysis is more limited. Since most SI engines today use 4-valve heads with a central spark plug, only limited data are available from other combustion chamber designs with a spark plug at one side, and no specific conclusions regarding spark plug location and octane number were possible. In addition, issues regarding coolant temperature and intake air temperature are not examined in any detail since, for a fully warmed up engine, these temperatures do not vary much across the papers compiled for this study. The issues of octane number effects at part load and at non-stoichiometric air-fuel ratios are very relevant for vehicle operation and are considered to the extent documented in these data.

1.3 ORGANIZATION OF THIS REPORT

Section 2 of the report is a review of the engineering principles that govern the relationship between engine efficiency and engine design and operating variables, as well as a discussion of the determination of a fuel's octane number and its relationship to engine design and operating variables. The relationship between octane number and efficiency is then derived in a two step process with these engine design and operating variables as the intermediate variables. Section 3 of this report examines the findings of the papers analyzed with respect to naturally aspirated engines, while Section 4 examined them with respect to turbocharged engines. Section 5 presents the major findings of the analysis in Sections 2, 3 and 4 and examines the relationships between CR and recommended fuel octane number for the current U.S. new vehicle fleet in the context of the findings. The CRC had also requested specific conclusions on the efficiency benefit associated with a 4 to 5 octane number increase in US fuel specifications, and these are provided by engine type at the end of Section 5. Section 6 identifies the data gaps in the collected literature, and provides some recommendations for future research directions.

2. IDEAL MODELS OF ENGINE EFFICIENCY AND RELATIONSHIP TO FUEL PROPERTIES

2.1 BACKGROUND

The thermal efficiency of an engine, using an ideal model of the SI engine operating on the Otto Cycle, is a function only of the compression ratio and is given by the relationship:

$$\eta_t = 1 - (1/CR)^{\gamma-1} \qquad (1)$$

where η_t is the thermal efficiency, CR the compression ratio, and γ the ratio of specific heat at constant volume to the specific heat at constant pressure. In the case of other SI engine cycles like the Atkinson Cycle or Miller Cycle, the relationship is somewhat different and involves both the compression and expansion ratio, but the trend with compression ratio is similar to that of the Otto Cycle. In equation (1), the efficiency is referred to as 'indicated efficiency' since it does not account for the effects of friction and pumping losses. This equation is true for naturally aspirated SI engines but boosted engines require more complex equations relating the efficiencies of the compressor and turbine or supercharger to engine speed and load to describe the net thermal efficiency of the reciprocating engine and turbo/super-charger combination.

Efficiency increases non-linearly with increasing CR according to equation (1). The ideal model of the Otto Cycle assumes instantaneous combustion of the air-fuel mixture, but actual combustion occurs over finite time and the spark timing affects efficiency. The highest torque (and hence, the highest efficiency) at a given RPM and WOT occurs at a spark timing advance level from piston top dead center (TDC) termed as 'minimum for maximum brake torque,' or MBT timing. Higher and lower levels of spark advance away from the optimum result in reduction in torque and efficiency, and the shape of the relationship of torque to spark advance is given by an inverted U-shaped curve. However, the maximum level of spark advance possible without causing abnormal combustion, or knock, in an engine with a specific configuration is related to the fuel octane number. Hence, there is a positive correlation between fuel octane number and efficiency as long as the knock limited spark advance (KLSA) is less than MBT spark advance. The fuel octane number when the KLSA is equal to the MBT at WOT conditions is termed the octane number requirement (ONR) of the engine, and the relationship between fuel octane number and efficiency only holds for fuel octane numbers below the ONR for a specific engine.

However, even within this construct of relating fuel octane number to engine thermal efficiency, its application to the real world requires the detailed understanding of the impact of a number of additional variables. The peak indicated engine efficiency of naturally aspirated engines is a function not only of compression ratio but also a function of the operating conditions such as air-fuel ratio, engine RPM, and spark timing (EGR is currently not used at WOT but could be used in the future). The absolute octane number requirement of an engine is also a function not only of CR and operating conditions but also of other engine design variables such as combustion chamber size and shape and engine cooling, as well as ambient conditions of intake air temperature and humidity. The variables described are not intended to be comprehensive but are illustrative. The relationships of some of these independent variables to both the ONR and the efficiency is non linear and can even be potentially decoupled from one another.

The analysis in this report uses the data and findings from the literature search in Task 1 to provide a more comprehensive picture of the most significant engine parameters affecting engine efficiency and, independently, ONR. The second part of the analysis links the two to provide an understanding of the complex relationship between efficiency and octane number.

2.2 IDEAL CYCLE ENGINE EFFICIENCY

Although the ideal SI engine's efficiency relationship is specified by Equation (1), the value of γ to be used depends on the fuel composition and air-fuel ratio of the engine. Air has a γ of 1.4, but for a stoichiometric mixture, the γ values are about 1.3 for the compression stroke and about 1.2 for the expansion stroke (Heywood, Section 4.4, Ref. 1). Fitting actual pressure-volume data from engines has typically provided average γ values of 1.25 to 1.3 and values such as 1.28 are commonly used in the literature. However, data from a very detailed modeling analysis by Nissan (Muranaka, Takagi and Ishida, Ref. 2) provided an indicated efficiency curve with CR that we were able to replicate using a γ value of 1.265. As shown in Figure 2-1, reducing the γ value from 1.4 for an air cycle (where the gas in the cylinder contains only air) to 1.265 for a stoichiometric air-fuel mixture reduces the indicated efficiency by 14% to 15% absolute across the range of CR values from 8 to 16.

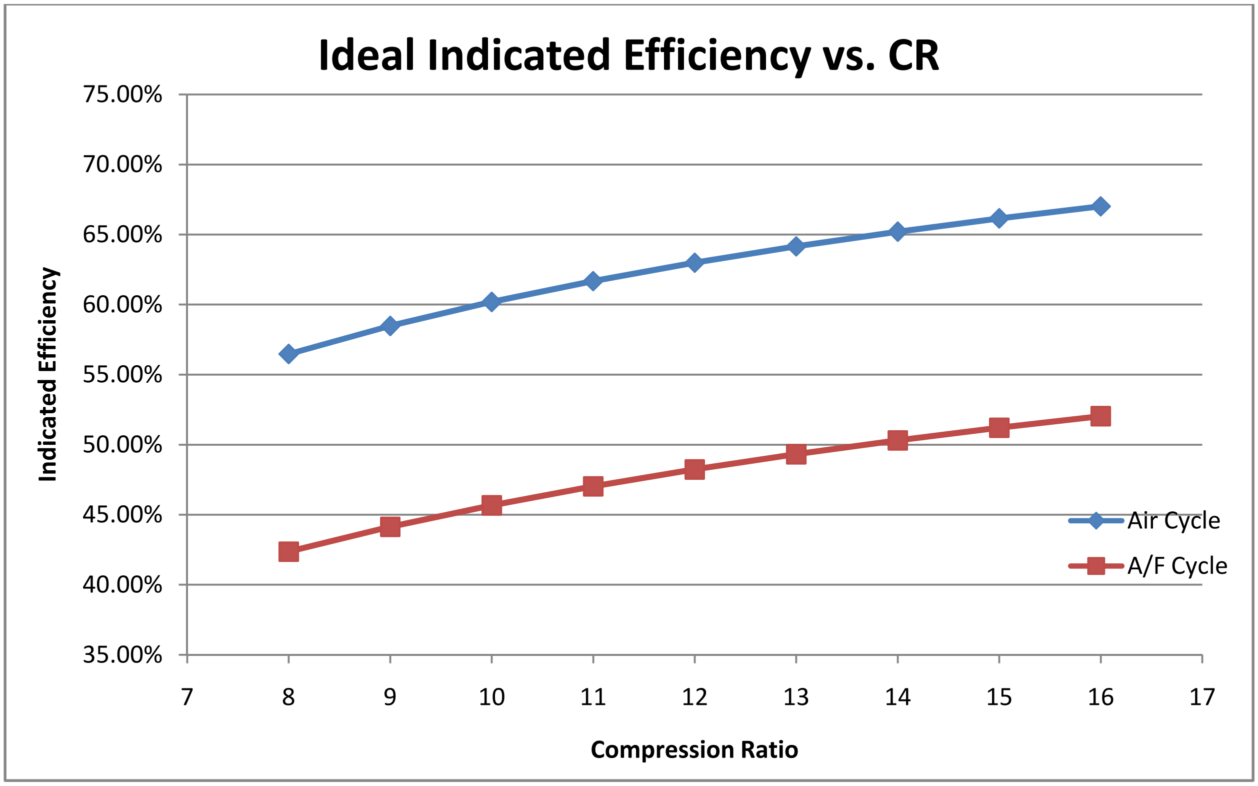

Figure 2-1

However, as pointed out by Muranaka, et al. (Ref. 2), the percentage effect of a unit change in CR on thermal efficiency is always higher on the air-fuel cycle than on the air cycle since it starts at a much lower value of efficiency. This is shown in Figure 2-2; increasing the CR by 1 unit from 9 improves indicated thermal efficiency by 3.73 relative percent[5] and from a base of 11 CR, indicated thermal efficiency increases by 2.7 relative percent, using idealized equations for an air-fuel cycle. However, consideration of all of the losses in real engines leads to a somewhat different relationship between indicated efficiency and CR.

No ideal equation exists to relate the engine ONR to the compression ratio as this relationship is defined by a number of engine design and operating variables including combustion chamber shape and size, spark plug location, cylinder head and wall temperatures, intake air temperature, etc. (As before, the list is intended to be illustrative, not comprehensive.) The relationship of engine ONR to these variables has been established empirically for specific engine designs, but no simple theoretical relationship has been determined. However, a key mechanism used to

[5] The term "relative percent" is used to denote the percent change in percent efficiency. If absolute efficiency changes from 40% to 44%, the 4% absolute change would equate to a 10 % relative change.

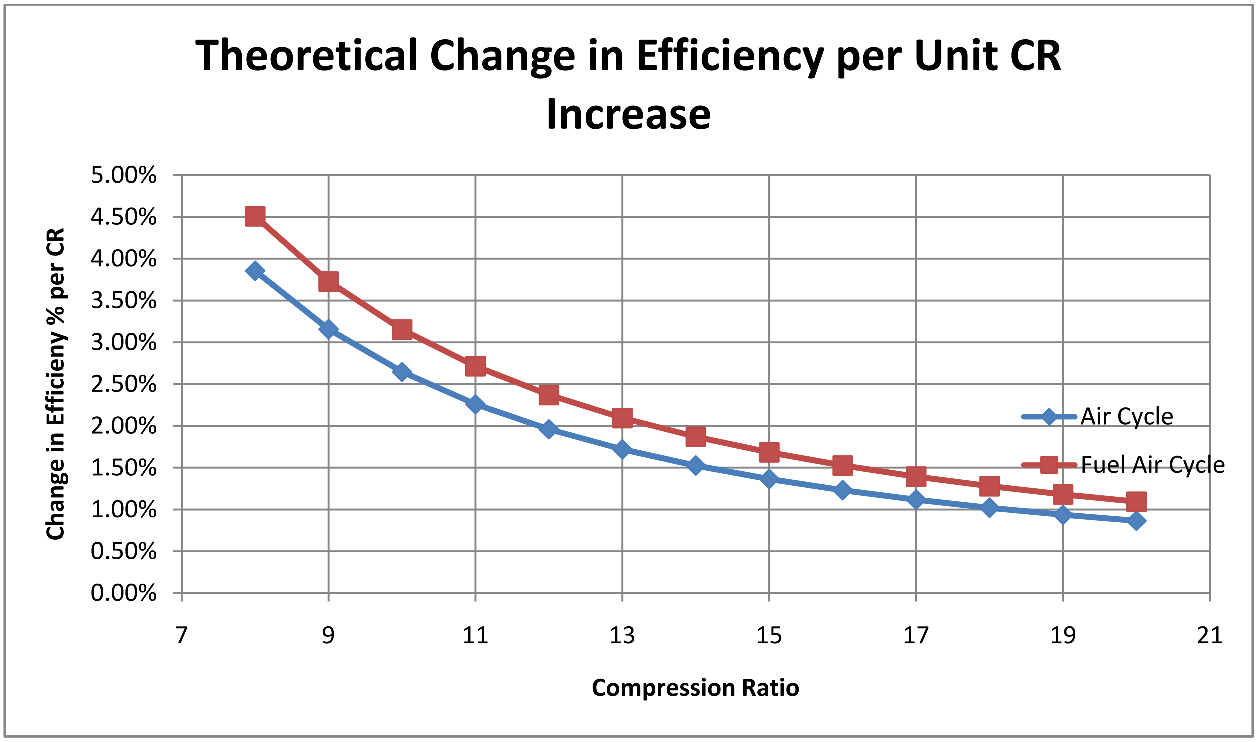

Figure 2-2

control knock in SI engines is the spark timing, which impacts the peak and average cycle pressure, and hence, the torque output of the engine as well as its efficiency. Maximum torque at any operating point (and hence, maximum efficiency) is obtained at MBT spark timing, and any spark timing retard from this MBT timing, normally measured in crank angle degrees before TDC, causes a reduction in torque as well as a reduction in the engine octane number limit that we have termed as ONL for convenience.

It should be noted that as spark timing changes at constant RPM and wide open throttle, total air flow may be affected in a minor way due to changes in residual gas temperatures, but the following discussion assumes that the changes in volumetric efficiency and heat transfer with spark timing can be neglected to provide the following approximation at WOT for any spark timing, θ, relative to MBT conditions:

$$IMEP(\theta)/IMEP\ (MBT) \approx \eta(\theta)/\eta(MBT)$$

whereη is the indicated efficiency. The IMEP is the indicated mean effective pressure, or the cylinder pressure averaged over the compression and expansion stroke which is one revolution of the engine (360°).

The change in mean effective pressure as a function of spark timing is well understood, as is the nature of the relationship (Heywood, Section 9.1, Ref.1). A recent study at MIT (Ayala, Gerty and Heywood, Ref.3) provided a robust relationship between spark timing as measured in degrees retard from MBT timing to Net Indicated Mean Effective Pressure (NIMEP). NIMEP is the term used to describe average cylinder pressure over the entire two revolutions (720°) of the Otto cycle and is equal to IMEP – PMEP, where PMEP is the pumping mean effective pressure. The authors developed a new combustion parameter that related the NIMEP at any spark timing $\theta$to the NIMEP at MBT spark timing with the formula:

$$1 - NIMEP(\theta)/NIMEP(MBT) = 0.168 * ((1 + 0.004443*(\theta_{50} - \theta_{50,\,mbt})^2)^{0.5} - 1)$$

Where θ_{50} is the crank angle at 50% mass burn fraction and $\theta_{50,\,mbt}$ is the crank angle of the 50% mass burn fraction at MBT spark timing, with the difference between the two variables termed combustion retard. The correlation was found to hold across a wide range of loads, speeds and air-fuel ratios. This equation is difficult to use in practice since the 50% burn durations are not known a priori. Considering only stoichiometric air-fuel ratios and WOT operation, the combustion retard is closely correlated with spark retard from MBT and the data in Figure 2 of the MIT paper for this condition were fitted to a quadratic with the resulting fit:

$$\eta(\theta)/\eta(MBT) = IMEP(\theta)/IMEP(MBT) = 1 - 0.5125*10^{-3} * (\theta_s - \theta_{s,\,mbt})^2 \qquad (2)$$

whereθ_s is the spark timing and $\theta_{s,\,mbt}$ is the MBT spark timing. This equation assumed that at WOT, NIMEP is approximately equal to IMEP since pumping losses are quite small, typically less than 2% of IMEP. The curve fit in Figure 2-3 shows that a 10°spark retard from MBT timing results in a 5.1% relative loss in efficiency and a 20 degree retard reduces relative efficiency by 20.5%. Hence, small changes in spark timing from MBT have very small efficiency effects but the non-linear nature of the relationship shows that retard over 10° can cause significant losses in efficiency.

The following sections of this report utilize the relationship (Heywood, Section 13.2, Ref. 1):

$$IMEP = BMEP + FMEP + PMEP \qquad (3)$$

where MEP is the mean effective pressure and I, B, F, and P stand for Indicated, Brake, Friction and Pumping, respectively. Data presented in Ref.1 shows that the IMEP of most modern naturally aspirated engines at wide open throttle stoichiometric operation at low RPM

(~1500RPM) ranges from about 11 to 12.5 bar while FMEP is around 0.8 bar to 0.9 bar and PMEP at around 0.1 to 0.2 bar (due to pressure loss across the valves and the exhaust system). Hence BMEP is 90% to 93% of IMEP. Typically, changes in IMEP at constant RPM do not change FMEP or PMEP significantly, so that the absolute IMEP change is equal to the absolute BMEP change. Hence, a 10% change in IMEP results in a larger percentage change in BMEP of 10/0.92 or 10.9% at WOT.

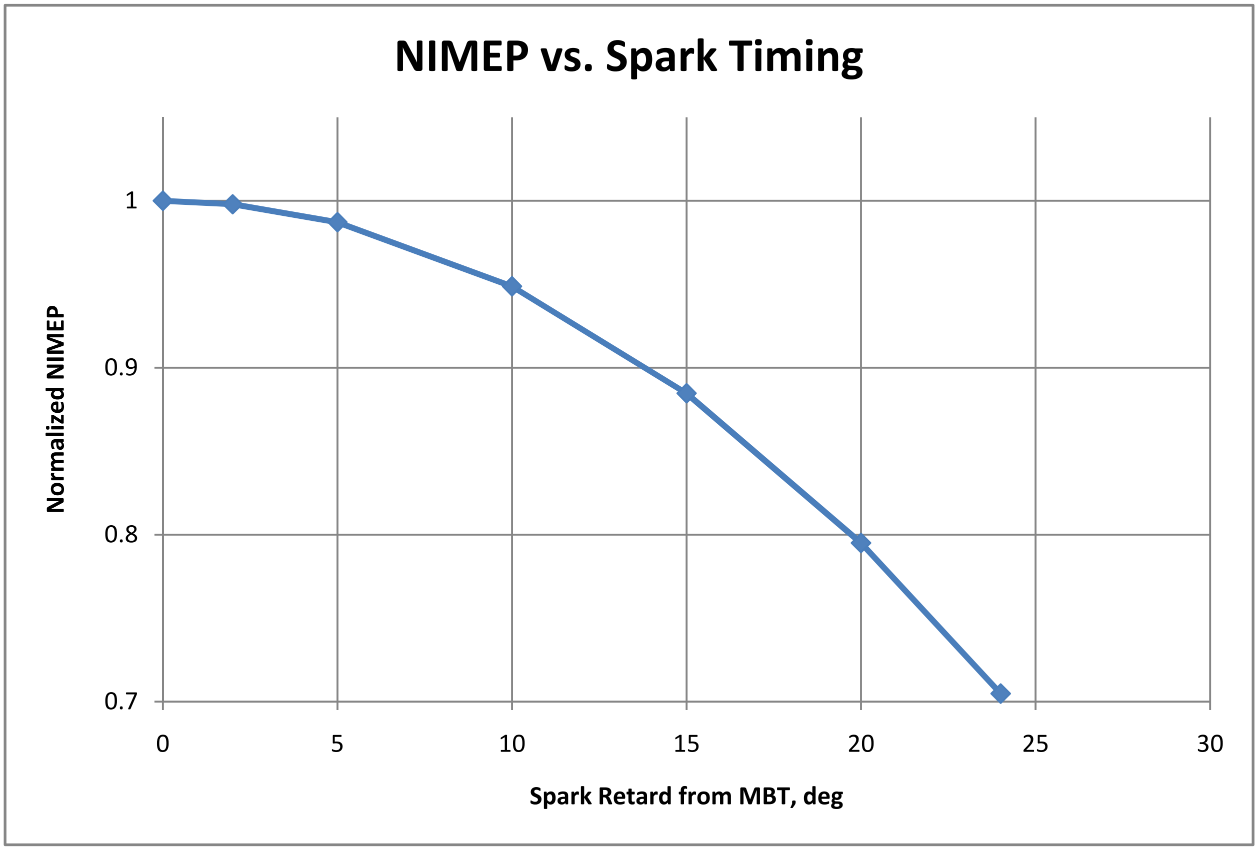

Figure 2-3

2.3 OCTANE NUMBER MEASUREMENT AND FUEL OCTANE NUMBER

A fuel's propensity to auto-ignite and cause "knock" is quantified in terms of the fuel's octane number. The octane number is measured on a standard engine called the CFR (for Cooperative Fuel Research) engine, where the engine is set to run at fixed intake conditions and RPM. The engine CR is increased until the engine knocks at a specified intensity as measured by a pressure transducer, while the air-fuel ratio is set to maximize knock. The procedure is then repeated using a blend of iso-octane (which defines 100 octane) and n-heptane (which defines 0 octane) and the blend is called a primary reference fuel (PRF). The volumetric percentage of iso-octane is adjusted until a PRF is found that knocks at the same intensity at the same CR as the fuel whose octane number is being measured. The octane number of the fuel is equal to the percentage of iso-octane of the PRF that has the same knocking tendency. The PRF is a combination of paraffins, but the octane number of typical hydrocarbon fuels depends on the test conditions since olefin and aromatic hydrocarbons found in commercial gasoline do not have the same temperature dependence to auto-ignition as paraffins. Hence, the octane number is measured at two test conditions; one termed “Research Octane Number” (RON) test which specifies a 52° C intake air temperature and an engine speed of 600 RPM, with a fixed spark timing of 13 degrees before top dead center. The second test condition is termed “Motor Octane Number” (MON) test which specifies a 149° C intake air temperature and an engine speed of 900 RPM, with spark timing varied set as a function of CR. Under both test conditions, air-fuel ratio is varied to maximize knock. By definition, the RON and MON of a PRF are numerically the same. Octane numbers higher than 100 are measured using a mix of iso-octane and tetra-ethyl lead (TEL), and a mixture containing 6 ml/ gallon TEL has an octane number of 120.3.

Modern engines do not resemble the CFR engine and the measurement conditions do not represent typical engine intake air temperatures, cylinder pressures, or operating RPM. These conditions of RPM, pressures and intake air temperature in modern engines are very different. Most modern SI engines operate between 1000 to 5000 RPM during normal driving, and intake air temperatures are close to ambient. In addition, the combustion chamber shape and air turbulence in the chamber in modern engines promote substantially faster burn rates than in the CFR engine. Hence, the relationship between CR and fuel octane number could be substantially different for modern engines.

The relevance of RON and MON ratings to modern engines have been examined at the research engine, production engine, and vehicle levels by a number of papers, and a new rating termed octane index has been derived. The basic principle is that the octane index (OI) of a fuel for modern engines can be expressed as a linear combination of the RON and MON values with K as the weighting factor such that:

$$OI = (1-K)*RON + K* MON \qquad (4)$$

Equation 4 can be rearranged as

$$\begin{aligned} OI &= RON - K*(RON - MON) \\ &= RON - K * S \end{aligned} \quad (5)$$

The term (RON – MON) in Equation 5 is called the Sensitivity[6], S, of the fuel. In a study by Shell (Kalghatgi, 2001, Ref. 4), 21 fuels with a wide range of RON and MON were tested on two single-cylinder engines with PFI and a more modern combustion chamber configuration, and a CR of 10.5.(CR was reduced to 8 for some tests in one engine.) The spark timing was advanced until a limiting level of knock was reached with each fuel, and the engine brake torque and knock intensity were measured. The study found that the engine torque increased with spark advance, flattening out at a maximum value at the MBT spark level and declining with further increases in spark advance, as predicted by Equation (2). Many fuels had too high a knock intensity at levels of spark advance much less than the MBT level so that they were tested only to the knock limit.

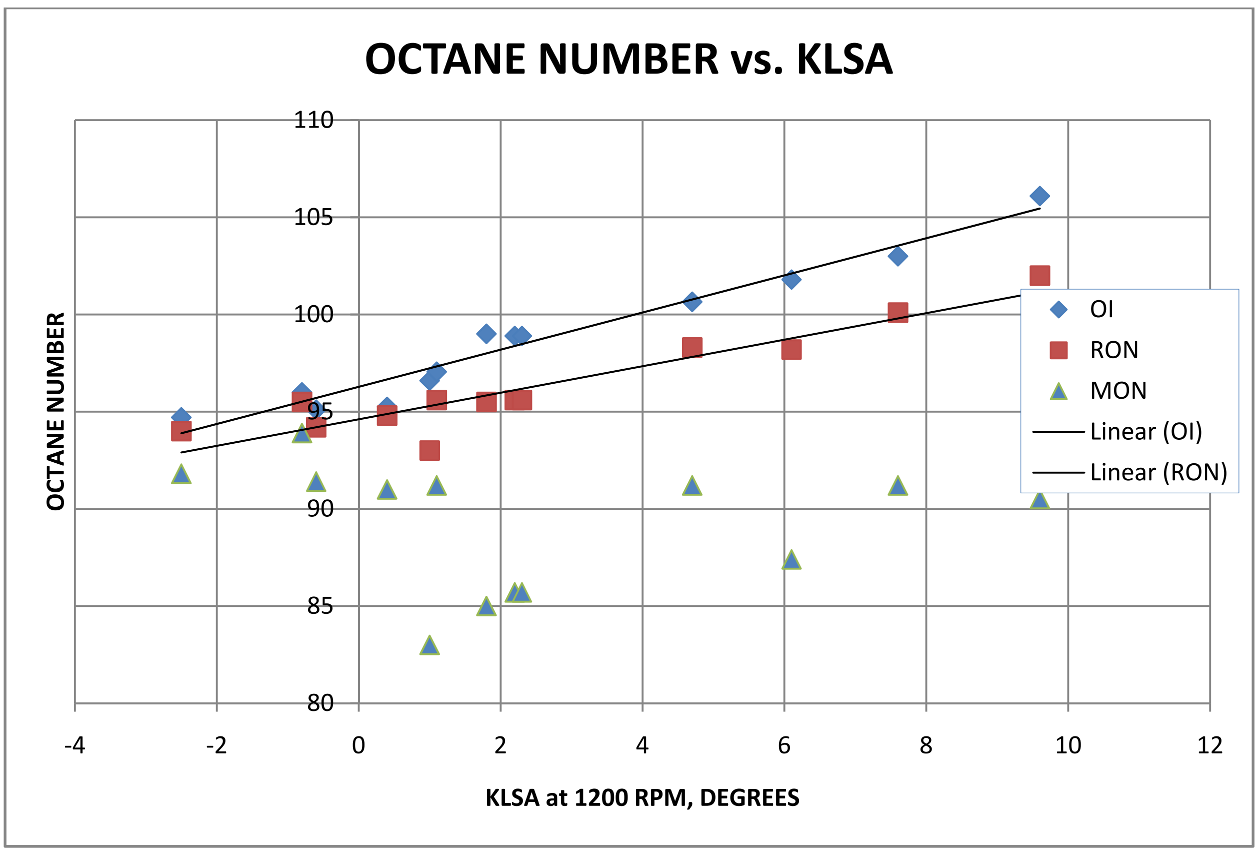

Figure 2-4

[6] The capitalized word "Sensitivity" in this report is used to denote the difference between RON and MON.

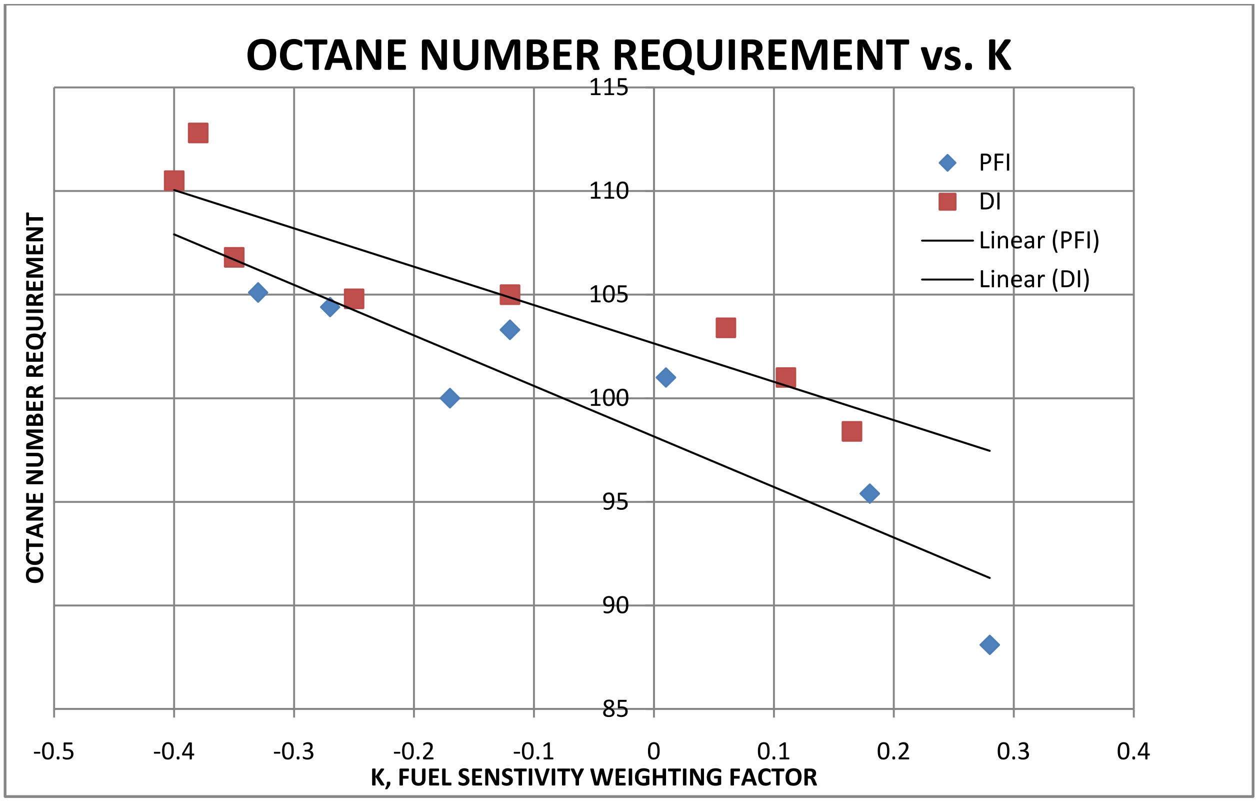

Figure 2-5

The analysis found that the knock limited spark advance was poorly correlated with a fuel's MON and better correlated with RON, but the best correlation was with the fuel OI. Figure 2-4 shows the correlation of the different octane ratings with KLSA for one engine at 1200 RPM. K, as defined in the equation above, was determined for each RPM/load test condition and was found to be negative at low RPM (<2000) and high load. K increased with RPM, but decreased with increased CR and load. The K value at any speed/load condition appeared to be largely independent of fuel type or RON. The result of a negative K value is that for a fuel with Sensitivity (S)>0, the OI is **greater** than either the RON or MON of the fuel.

K was found to decrease with increasing ONR of the engine at the different speed/load and CR values evaluated, so that as the ONR of the engine increased, K became more negative. In a follow on study by Shell and Toyota (Kalghatgi, Nakata and Mogi, Ref. 5) with DI engines operating at a CR of 11 and 12.5, the improved relationship of KLSA to the OI of the fuel relative to the relationship with either RON or MON was confirmed although the paper reported a quadratic relationship between KLSA and OI. The fit may have been due to some high octane number fuels used in this study that permitted the OI to meet or exceed the octane number requirement of the high CR engine. Figure 2-5 shows the relationship between K and the engine ONR, and there is a general trend for K to be more negative with increasing ONR, but the data scatter on K is apparent suggesting that K is influenced by other factors such as engine combustion chamber design, as well.

Figure 2-6 shows the KLSA as a function of the OI of the fuel for a DI engine with 12.5 CR (from Ref. 5) and the departure from linearity is obvious for the high OI fuels, while the linear fit

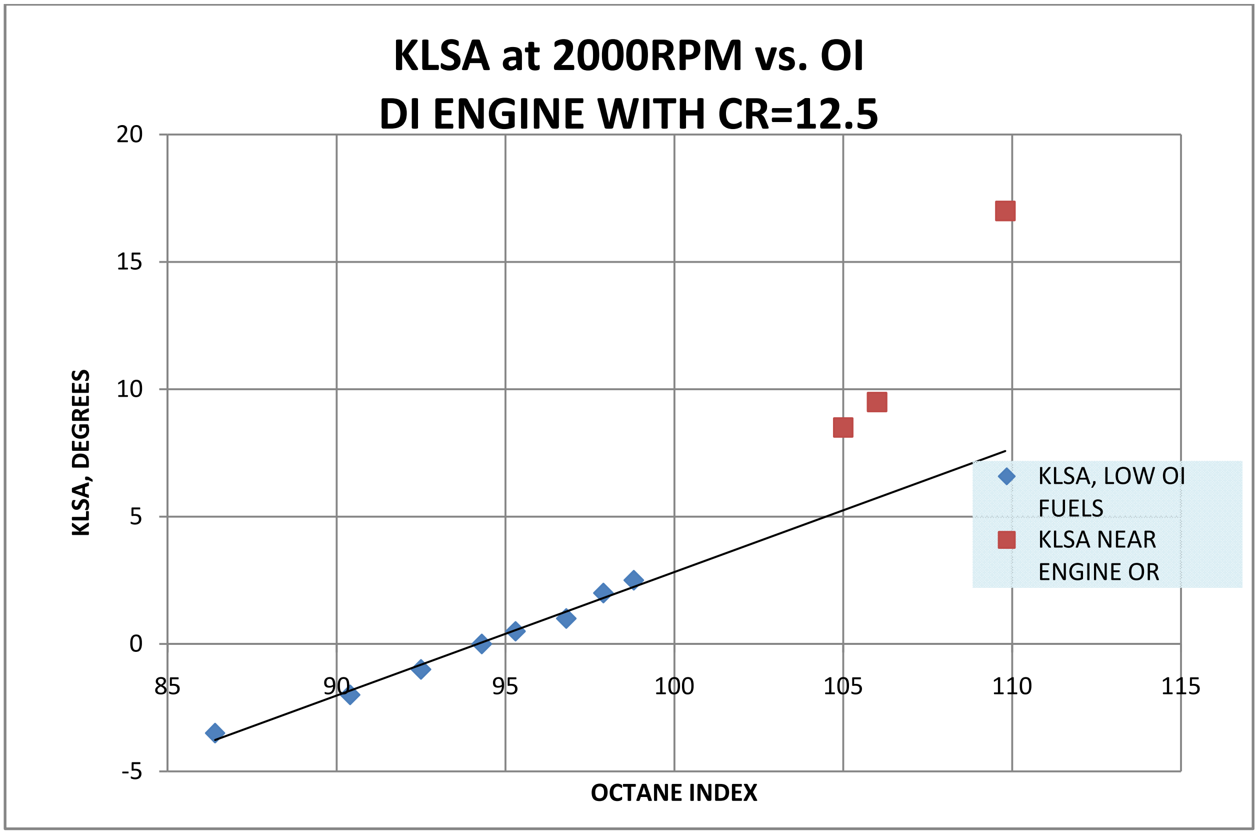

Figure 2-6

is quite good for the low OI fuels. As the fuel's OI approaches engine ONR, it appears that knock limited spark advance increases rapidly and the quadratic relationship is similar to that observed between engine torque; and spark advance, i.e., both engine torque and knock become less sensitive to spark advance when the fuel's octane index is close to the engine ONR and timing is close to MBT.

The finding that K was negative for modern engines was validated by tests at MIT (Mittal and Heywood, 2008, Ref. 6), conducted on a single cylinder engine with a CR of 9.8 at 1500 RPM, WOT and a λ of 1. Tests were conducted on fuels with almost similar RON of about 96, but with MON values ranging from 87.5 to 96. The KLSA was found to increase linearly with increasing fuel Sensitivity (decreasing MON). K was also found to increase with RPM, similar to the finding by Kalghatgi (Ref. 4). When we plotted K against 1/RPM (or time per revolution) for both the MIT and Shell data sets, a linear dependence of K was observed but the dependence with RPM was different for the two engines, as shown in Figure 2-7. In this case, 20ms corresponds to 3000 RPM, while 50ms corresponds to 1200 RPM.

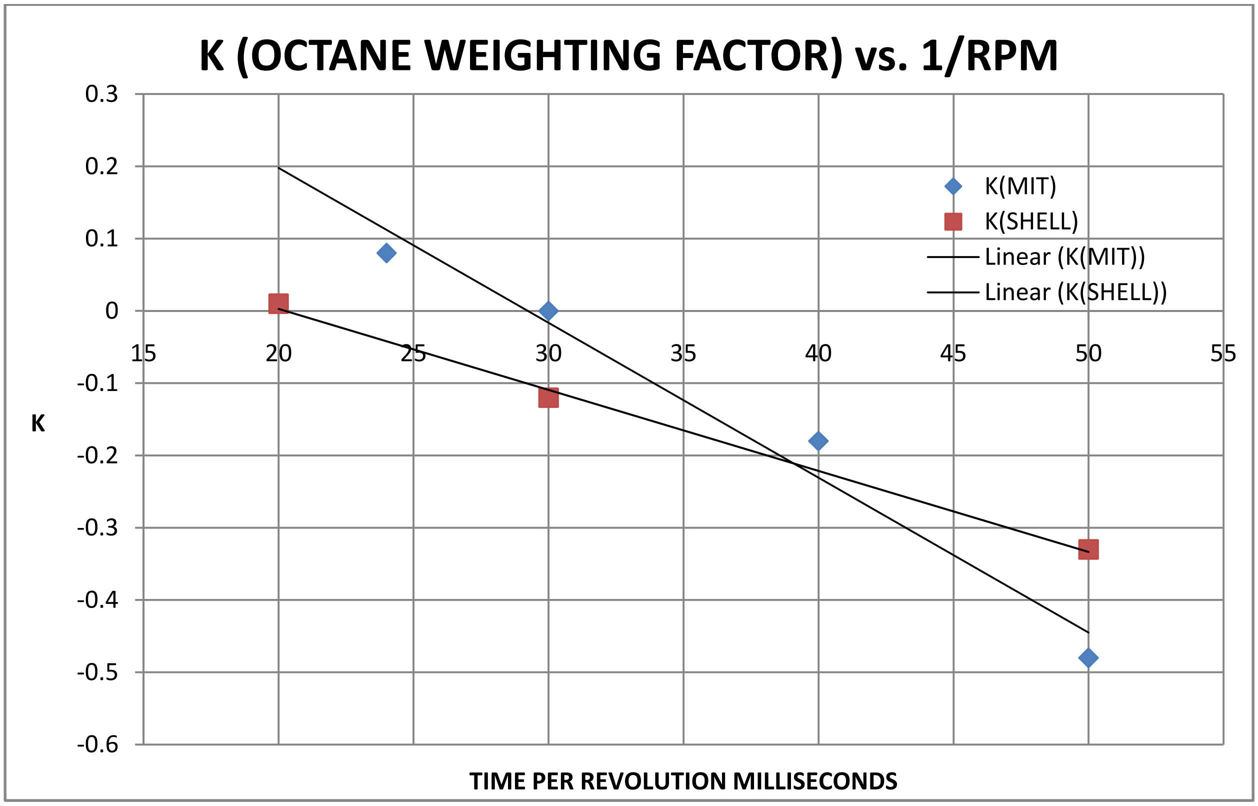

Figure 2-7

The reduction in K implies that the RON rating is most important at lower RPM while the MON becomes a factor in knock resistance at high RPM. The MIT analysis also found that K was:

- Only weakly dependent on spark plug location and CR (varying from 9.8 to 13.4).
- Non-linearly dependent on air-fuel ratio, being highest at lambda = 1 but decreasing at both richer and leaner air-fuel ratio.
- Increased linearly with intake air temperature.
- Decreased almost linearly with increased intake air pressure (boost).

The finding that K decreases with intake boost is particularly interesting, since it implies that sensitive fuels are well suited to turbocharged engines as their OI is significantly higher than the RON of the fuel. The MIT paper indicated that at a boost level of 1.4 bar, the K value was -0.65. For a typical retail gasoline with a Sensitivity of about 10 octane numbers, the MIT estimate indicates that the OI is 6.5 numbers above the RON at this boost level.

In a subsequent paper, Mittal and Heywood (2009, Ref. 7) examined the historical relevance of RON and MON based on the results from CRC octane number surveys from 1950 to 1990 where identical tests had been performed to determine vehicle OI with primary reference fuels and full boiling range fuels. K was determined by the equation:

$$K = (RON - ON)/ S \qquad (6)$$

Where RON is the research octane number for the full boiling range fuel and S its Sensitivity, and ON is the octane number of the primary fuel. The computations showed that K values had decreased over time with average K values declining from 0.28 in 1951 to just under 0.1 in 1991. This was in spite of the fact that the average knock-limited engine speed increased from 1500 RPM in 1951 to about 2400 in 1991, as K increases with increasing RPM. The paper suggested that increasing CR, the reduction of intake air temperature by eliminating intake air pre-heat, improved volumetric efficiency, improved engine cooling and decreased bore size were the contributing factors. Simulation modeling of typical engine designs for a 1951 engine and a modern turbocharged engine supported the calculated values of K. The authors suggested that with the use of DI and turbo-charging, K would decline further. In an earlier review paper, Kalghatgi (2005, Ref.9) came to a similar conclusion, suggesting that future engines would avoid knock using fuels with a lower MON and higher RON.

A 2011 paper by Toyota (Sasaki, et al., Ref.37) supports the view that higher MON will continue to be required in the future to avoid other forms of abnormal combustion. The study investigated pre-ignition due to hot spots by controlling the spark plug center electrode temperature. Two engines were used in the study, one a DI engine with a CR of 11.5 and the second a PFI engine with a CR of 15. The second engine was used to simulate the cylinder pressures under boosted conditions for a turbocharged engine; the high CR was chosen to obtain better control of cylinder pressure, as in a turbocharged engine, the boost pressure varies with spark timing. Three gasolines with different RON and Sensitivity and 15 specific hydrocarbon compounds covering a range of paraffin, olefin, and aromatic compounds were tested. The study confirmed that near linear relationship of engine torque with fuel RON at 4400 RPM, WOT. However, the study found that pre-ignition occurs at much lower hot spot temperatures in the high CR engine and that the pre-ignition temperatures for this engine with the different fuels showed MON to have a stronger relationship than RON to pre-ignition temperature. The study found that increasing MON by 16,(i.e., from 80 to 96), resulted in pre-ignition temperature increasing by about 200° C. The coefficient of determination (R^2) of pre-ignition temperature with MON was 0.93 while it was only 0.66 with RON. Hence, the study concluded that a higher MON was useful in suppressing hot spot related pre-ignition.

2.4 FUEL COMPOSITION EFFECTS ON OCTANE NUMBER AND EFFICIENCY

Retail fuels for SI engines are a mixture of aromatic, paraffin and olefin compounds, and can also include oxygenates like ethers and alcohols. The compounds can influence octane number and fuel efficiency of engines differently since they have different combustion kinetics and different flame speeds in mixtures with air.

A chemical kinetic modeling study by Mittal, Heywood and Green (Ref.8) concluded that the auto-ignition chemistry for modern engines occurs in the transition between high temperature and low temperature combustion regimes corresponding to end-gas temperatures between 500°C and 625°C. Fuels with higher octane Sensitivity have a stronger temperature dependence of the auto-ignition delay time which results in slower low temperature combustion and faster high temperature combustion. Modern engines operate with end gas temperatures well below the high temperature regime where sensitive fuels are less prone to auto-ignite. Hence, fuels with higher Sensitivity will be less prone to auto-ignite (i.e., will show higher octane numbers) when this is the case.

In a review paper, Kalghatgi (Ref. 9) speculates that the RPM dependence of K could be the result of a reduction in heat transfer due to less time available, and hence, resulting in an increase in mixture temperature and an increase in the pressure rise rate. As the end-gas temperature increases, the conditions become closer to those for the MON test and the importance of the MON increases making K more positive. Akihama, et al., (Ref.10) find that paraffin compounds have high auto-ignition resistance at higher temperatures where the kinetics of paraffin decomposition have a negative temperature coefficient. Hence, the reference fuels which are a mixture of paraffin compounds are more resistant to knock at MON conditions than fuels containing olefin and aromatic compounds. At conditions closer to those of the RON test, the olefin and aromatic compounds have better knock resistance, and mixtures containing higher olefin and aromatic compounds can have an octane number higher than the octane number of the reference fuel, leading to a negative K.

In an analysis of fuel effects over a range of air-fuel ratios, an ExxonMobil/Toyota paper (Farrell, et al., Ref. 11) examined the burning velocities of fuels with different mixtures of aromatic, olefin, and paraffin compounds at λ values ranging from 1 to 1.7. The paper reported that laminar burning velocities varied by more than 11% between the slowest and fastest fuels, and that burning velocities correlate well with olefin content and higher fuel volatility.

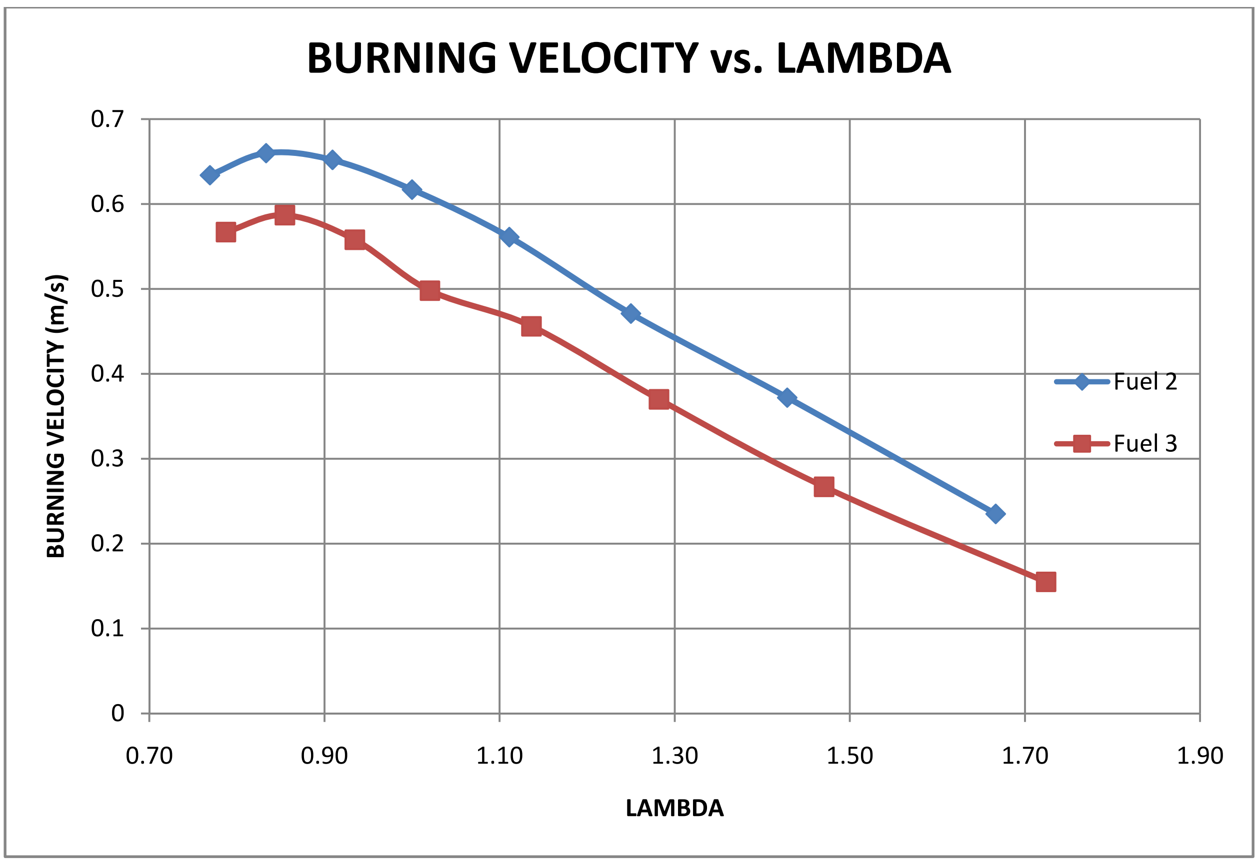

Figure 2-8

Figure 2-8 shows the burning velocity of two fuels as a function of air-fuel ratio and that the burning velocity increases by a factor of 3 between λ at 0.9 to 1.6. Fuel 2 had a 35% aromatic and 25% olefin content, while Fuel 3 had a 35% aromatic and a 8% olefin content. Tests of these fuels on an engine operating at WOT, 2400 RPM and $\lambda = 1$ showed that fuels with high burning velocities gave up to 3% higher torque at the same spark advance. The faster burning fuel also allowed higher spark advance at the knock limit.

2.5 LATENT HEAT OF VAPORIZATION

In PFI engines, much of the fuel is inducted into the cylinder after it has vaporized due to heat transfer from the engine. In DI engines, the fuel is introduced into the cylinder in liquid form; vaporization of the fuel causes a reduction in the air-fuel mixture temperature relative to mixture for a PFI engine, thereby reducing the octane number requirement of the engine. The latent heat of vaporization for gasoline is estimated at 420 kJ/kg and varies somewhat depending on gasoline composition. However, the introduction of ethanol has a particularly large effect on temperature, as it has a latent heat of vaporization(LHV)which is approximately double that of gasoline per unit weight, at 845 kJ/kg, as reported by Toyota (Nakata, et al 2006, Ref. 12). In addition, the stoichiometric air-fuel ratio for ethanol is 9.0 compared to 14.5 to 14.7 for gasoline so that a larger quantity of ethanol is required at stoichiometric conditions for a given volume of air. Ethanol also has a RON of 111, giving it high knock resistance. Others have quoted much higher numbers, and an MIT paper (Bromberg and Cohn, Ref. 13) cited the RON as 129 and the MON as 102 (which are blending octane numbers) and list the latent heat of vaporization as 910 kJ/kg. However, ethanol air mixtures have lower flame speeds than gasoline air mixtures.

In a PFI engine, the vaporized ethanol volume (to the extent that the ethanol is vaporized before induction into the cylinder) displaces air and reduces the air volume relative to the air volume with gasoline, so that the engine volumetric efficiency is reduced. When used in a DI engine, there is virtually no air displacement effect, and the charge cooling due to ethanol's higher LHV results in volumetric efficiency improvements. In the study of knock behavior in a turbocharged DI engine. Milpied, et al., (Ref. 31) compute a "cooling power" value defined as latent heat of vaporization / air-fuel ratio at stoichiometric conditions. They used latent heat of vaporization values of 335 kJ/kg for gasoline and 910 kJ/kg for ethanol, so the cooling power for ethanol is 4.4 times that of gasoline, while the 845 kJ/kg value cited above suggests a cooling power improvement by a factor of 3.1.A paper by Brewster (Ref. 33) shows that if all the heat of vaporization is taken from the air, the temperature drop with gasoline is about 20°C while the drop with ethanol is about 90°C (using latent heat values closer to those of Ref. 31).

While ethanol has a lower energy density relative to gasoline, the larger quantity of ethanol required for a stoichiometric mixture with air results in the energy density of the ethanol -air mixture being only 1% to 2% lower than a stoichiometric gasoline air mixture. Hence, the use of ethanol can create some power loss in a PFI engine if most of the fuel enters the cylinder after vaporization, but can result in a power gain in a DI engine due to the colder mixture with ethanol fuel. The power loss or gain does not directly impact engine efficiency but can have secondary impacts on vehicle efficiency if the power changes are accounted for by changes in engine size or gear ratio.

3. NATURALLY ASPIRATED ENGINES

3.1 ACTUAL ENGINE EFFICIENCY

While the idealized relationship between efficiency and CR was developed in the previous section, real engines have a number of losses that result in the actual indicated efficiency being much lower than predicted by the ideal cycle relationship. Muranaka, et al., (Ref. 2) at Nissan have estimated the factors that reduce indicated efficiency as follows:

- Cooling losses which occur due to heat transfer from the combustion chamber to the coolant.
- Time loss which accounts for the fact that combustion is not instantaneous but occurs as the piston is moving.
- Unburned fuel loss which accounts for portion of fuel that is exhausted as a hydrocarbon or a product of partial combustion.

The analysis developed from detailed simulations in Ref. 2 shows that as cylinder volume is reduced, the indicated efficiency decreases. At a compression ratio of 9, indicated efficiency decreases from almost 40% at 2000 cc swept volume to about 36% at 200cc, which is a 10% relative loss in indicated efficiency (4/40). The analysis also found that heat transfer losses and indicated efficiency reductions were linearly proportional to the combustion chamber surface to volume ratio.

The effect of combustion occurring over a finite time period was also evaluated in Ref. 2 and the authors found, unlike historical assumptions that instantaneous combustion would result in the highest efficiency, a combustion duration of 30 crank angle degrees (CAD)was optimal; at shorter durations, the higher pressure and temperature resulted in higher heat loss, offsetting the improvement from reduced time loss. However, the shape of the efficiency curve as a function of combustion duration shows a broad and relatively flat peak over the 0 to 40 CAD range, and the shape is relatively independent of CR changes over this CAD range.

The effect of unburned fuel was also estimated as a function of engine out HC emissions, and an engine out emissions of 3000 ppm HC was equivalent to 2% unburned fuel at $\lambda = 1$. HC concentration increases with decreasing bore size and increasing CR, but no specific relationships were given. Estimates of unburned fuel range from 2.5% to 5% for current engines, but this may be larger for high CR engines, and especially for the combination of high CR and small cylinder size. In general, available literature treats this effect empirically by multiplying the engine efficiency, computed assuming complete combustion by 0.96 or 0.97, as a combustion efficiency parameter, η_c.A smaller value may be required for small bore and high compression engines.

In an attempt to provide a unified estimate of actual efficiency, we used the air-fuel cycle efficiency as a starting point and examined the cooling loss effect for the simplified example of a disc shaped combustion chamber. The surface to volume ratio (S/V) of a disc shaped combustion chamber of height Land bore size, b, is given by:

$$S/V = (\pi b^2/2 + \pi bL) / (\pi b^2 L/4) \qquad (7)$$

The combustion chamber height L can be related to the stroke, s, and compression ratio, CR, by:

$$CR = L+s / L \text{ or } L = s/ (CR-1)$$

Substituting the above expression for L in equation(1) for S/V gives:

$$S/V = (2(CR-1)/s + 4/b), \text{ or}$$

$$S/V = 2/b *((CR-1)*b/s + 2) \qquad (8)$$

In many modern engines, the bore and stroke are very nearly equal (sometimes referred to as a "square" engine) and for $b/s \approx 1$, equation (2) simplifies to:

$$S/V = 2*(CR+1)/b \qquad (9)$$

In equation 9, any decrease in bore size also results in displacement reduction due to the assumption that b = s. Since the simulation models show that cooling loss is linearly proportional to the S/V ratio, we utilized the following model to fit the actual measured data:

$$\eta_{HT} = (1- A*(CR + 1)/b) \qquad (10)$$

$$\eta_I = \eta_t * (1- A*(CR + 1)/b) * \eta_c \qquad (11)$$

Where $\eta_i$the indicated efficiency of a real engine, η_t the ideal cycle efficiency for $\gamma = 1.265$ as described in Section 2, $\eta_{HT}$the heattransfer related factor, A, a constant and η_c the combustion efficiency parameter. In Reference 2, Nissan reported data on two engines, one with 500cc swept volume and the other with 250cc swept volume The bore and stroke were not reported in the paper and b was computed assuming the bore and stroke were equal, so that bore sizes were 86mm for the 500cc engine and 68.25mm for the 250cc engine.

The fit of this equation to the actual data at 1500 RPM proved to be excellent for constant A = 1.13mm, and η_c = 0.96 for the 500cc engine and 0.95 for the 250cc engine (which is consistent with the concept that smaller bore engines have higher unburned fuel loss). The same value of η_c was used for all compression ratios, and we assume that the CR dependence of η_c has been subsumed into the heat transfer loss constant k. The table below shows the computed efficiency from the equation to the actual data points (as estimated from the figure) reported by Nissan.

Compression Ratio	**Computed η 500cc engine**	**Reported η 500cc engine**	**Computed η 250cc engine**	**Reported η 250cc engine**
8	35.79	35.8	34.25	34.2
9	36.73	36.75	35.00	35.0
10	37.43	37.4	35.49	35.6
12	38.32	38.3	35.96	35.9
14	38.70	38.7	35.92	Not reported

The relationship of indicated efficiency to CR for the two engines derived above is shown in Figure 3-1, and the impact of bore size reduction is significant – at a CR of 13, the smaller engine has an efficiency that is 6.7% (relative) less than that of the larger engine (36% vs. 38.6% absolute). The marginal benefit of a unit increase in CR from 9 to 10 is only 1.5%, and the equation projects that for engine sizes of 250cc/ cylinder, increasing CR beyond 13 has no benefit to efficiency. The three auto-manufacturers interviewed for this project stated that their own experience with a 75mm bore engine showed an efficiency maximum at a CR of about 13.5 to 14, somewhat higher than predicted by this equation.

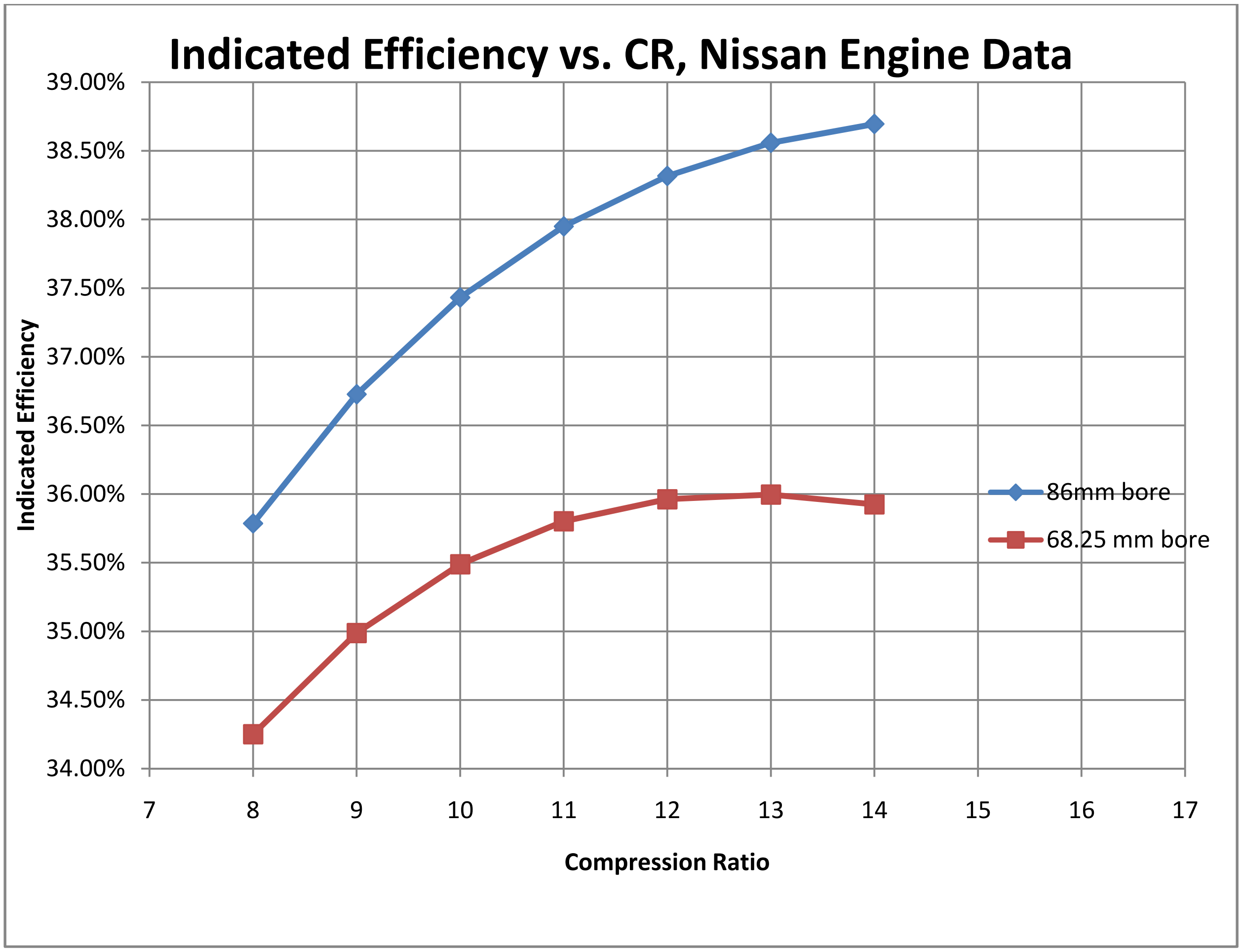

Figure 3-1

Ayala, Gerty, and Heywood from MIT (Ref. 3) also plot the increase in NIMEP (≈IMEP at WOT) with CR for an engine with an 83mm bore. Their paper reports a quadratic curve fit as :

$$NIMEP/(NIMEP)_{9.8} = 0.126 + 0.137*CR - 0.00487*(CR)^2 \qquad (12)$$

The fit appears to be based on only three data points at 9.8, 11.6, and 13.4 CR; the reported baseline gross efficiency of 39.0% at 9.8 CR closely matches the estimate from the equation derived above <u>without the combustion efficiency</u> parameter, and this parameter is not referenced in the text of the MIT paper. However, the rate of change of efficiency implied by the quadratic fit in Equation 12from a CR of 10 to 11 is 3.45% relative, which is much higher than estimated by Equation 11, and the quadratic estimates a sharp efficiency peak at a CR of 14.07. The high rate of change and peak at 14.07 CR implied by the quadratic are not supported by data from other papers. However, the consensus view of the three auto-manufacturers consulted was that increasing the CR from 10 to 11 would yield about 2% relative efficiency increase. Even the simulations in Reference 2 show significantly higher improvement in thermal efficiency with the 500cc engine showing a 12% relative increase in efficiency with CR increasing from 9 to 16 compared to about 8% from the fit to the measured data points.

While there is limited data on absolute efficiency changes with CR from modern engines at WOT, there is more data at part load points. Both the Nissan and MIT papers show that the absolute value of indicated efficiency decreases with decreasing load. For example, the MIT paper projects that indicated efficiency declines from about 35% absolute at 8bar/1500RPM to about 33% absolute at 2bar/1500 RPM, at a lambda of 1. However, the relative changes in indicated efficiency with load were found to be very similar over a range of CR values so that the normalized changes should be indicative of the change in efficiency at WOT.

A comprehensive analysis of the impact of RON and engine thermal efficiency was conducted by Nakata, et al., (Ref. 15) on naturally aspirated PFI engines, one with an Atkinson Cycle and the second with a normal Otto Cycle. Both engines were dimensionally identical with a 75 mm bore and 84.7 mm stroke, and had a CR of 13, but the Otto cycle engine was also tested at a CR of 10. Tests on two gasolines with a RON of 91.5 and 99.6, respectively, were conducted at a range of loads at 2000 RPM. As shown in Figure 3-3 below, the higher CR on the Otto Cycle results in a significant increase in part load efficiency even with the 92 RON gasoline, but spark retard at full load on the 13 CR engine to prevent knock while operating on the 91.5 RON gasoline results in brake efficiency levels below those of the 10 CR engine. With the 99.6 RON gasoline, the 13 CR engine was knock limited only at full load but allowed peak thermal efficiency to improve by 15% relative to the peak efficiency of the 10 CR engine. The benefits of higher RON gasoline to efficiency with the Atkinson Cycle engine (not shown in the figure below) were much smaller since the effective CR is much lower than the geometric CR of 13.

A more recent paper by Nakata, et al., of Toyota (Ref. 15) showed a 9% relative increase in brake thermal efficiency when CR was increased from 9.8 to 13 in a 88.5mm bore size engine or a 2.8% increase per CR, at a BMEP of 0.2 MPa. Even allowing for the fact that brake thermal efficiency increase can be higher than indicated efficiency increase, the difference cited in both Toyota papers is larger than that shown by Nissan engine data.

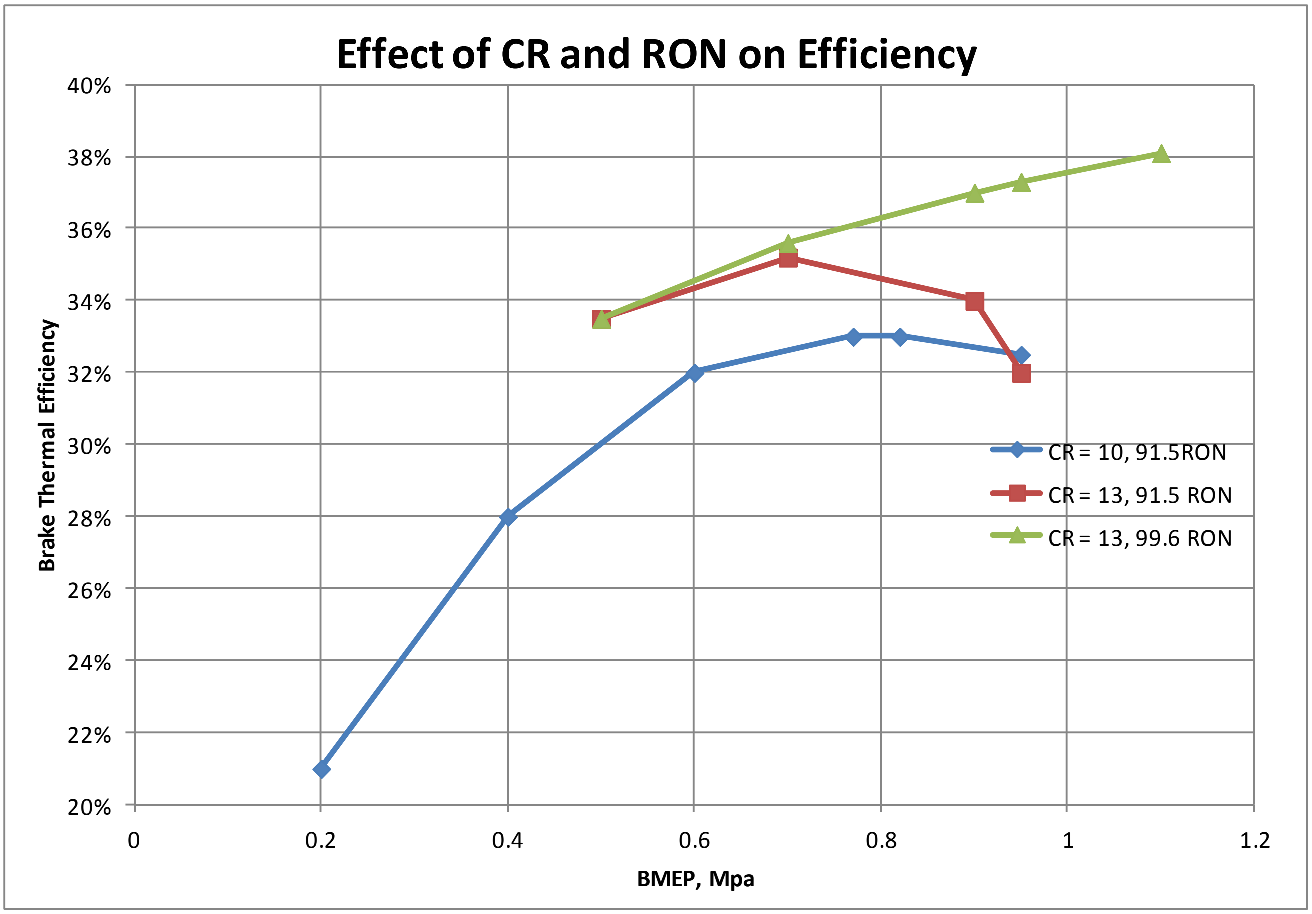

Figure 3-2

Similarly, a paper by Munoz at al. of Ford (Ref. 14) showed indicated efficiency at 3 part load points for a PFI engine with a bore size of 89mm from a CR of 8.5 to a CR of 11, and indicated efficiency rises by 9% to 10%(relative) over the range (See Figure 3-3).The trend line shows a 0.9% to1% increase in absolute efficiency per unit CR increase or about a 3% increase in relative efficiency per unit CR increase.

Based on the higher sensitivity of efficiency to CR reported by the Toyota and Ford papers, we modified the heat transfer loss η_{HT} to fit the simulated efficiency values reported in Reference 2 without the combustion efficiency factor, so that the equation was modified to:

$$\eta_{HT} = 0.96^* (1- A^*(CR + 1)/b) \qquad (12)$$

where the 0.96 factor was incorporated to model heat transfer loss independent of combustion surface-to-volume ratio, and constant, A, was estimated to be 0.8mm.

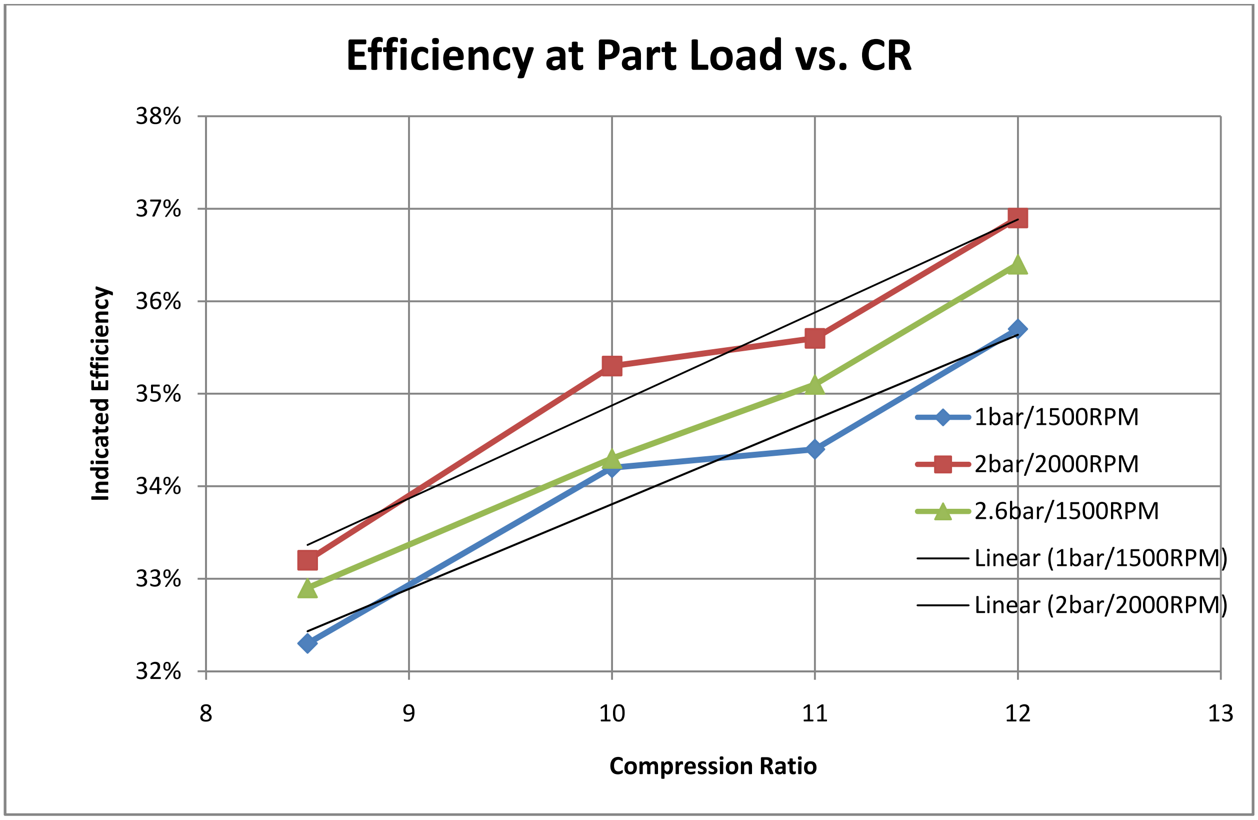

Figure 3-3

The equation yielded a good fit to the simulated results in Reference 2 over the CR range from 9 to 14 at 1500 RPM but appeared to underestimate the benefit of CR at values over 14. The computed efficiency versus CR and the marginal benefit of CR increases computed using this equation for heat transfer in Equation (4) are shown in Figures 3-4 and 3-5, respectively. The marginal benefit is of particular interest to the study, and the projected benefit increasing CR from 10 to 11 is around 2% (relative)for the larger bore engines, consistent with auto-manufacturer inputs on this issue. In addition, the benefits of increasing CR beyond 14 are less than 0.5% per unit increase for small bore engines and small increases in friction with increasing CR could eliminate this benefit in indicated efficiency. The marginal benefits should be applicable over a range of loads at 1200 to 2000 RPM, indicated efficiency increases at higher RPM, especially at higher compression ratio. Reference 2 reported that indicated efficiency increased by 3% (relative) at 3000 RPM and 5% (relative) at 6000 RPM, relative to the value at 1500RPM at a CR of 14. However, at a CR of 9, the increases at higher RPM were negligible. The higher engine speeds affect the cooling loss due to reduced time for heat transfer and become significant when cooling loss ratios are high; i.e., for higher S/V ratios.

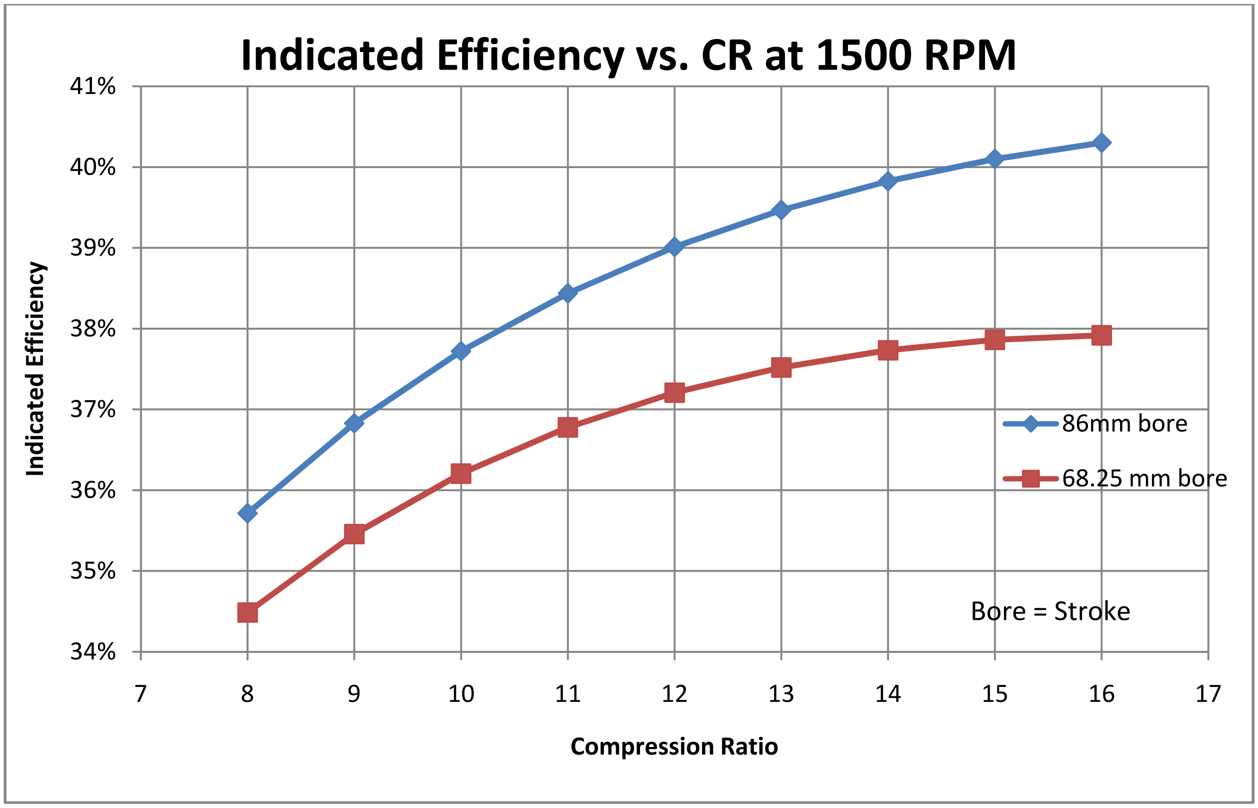

Figure 3-4

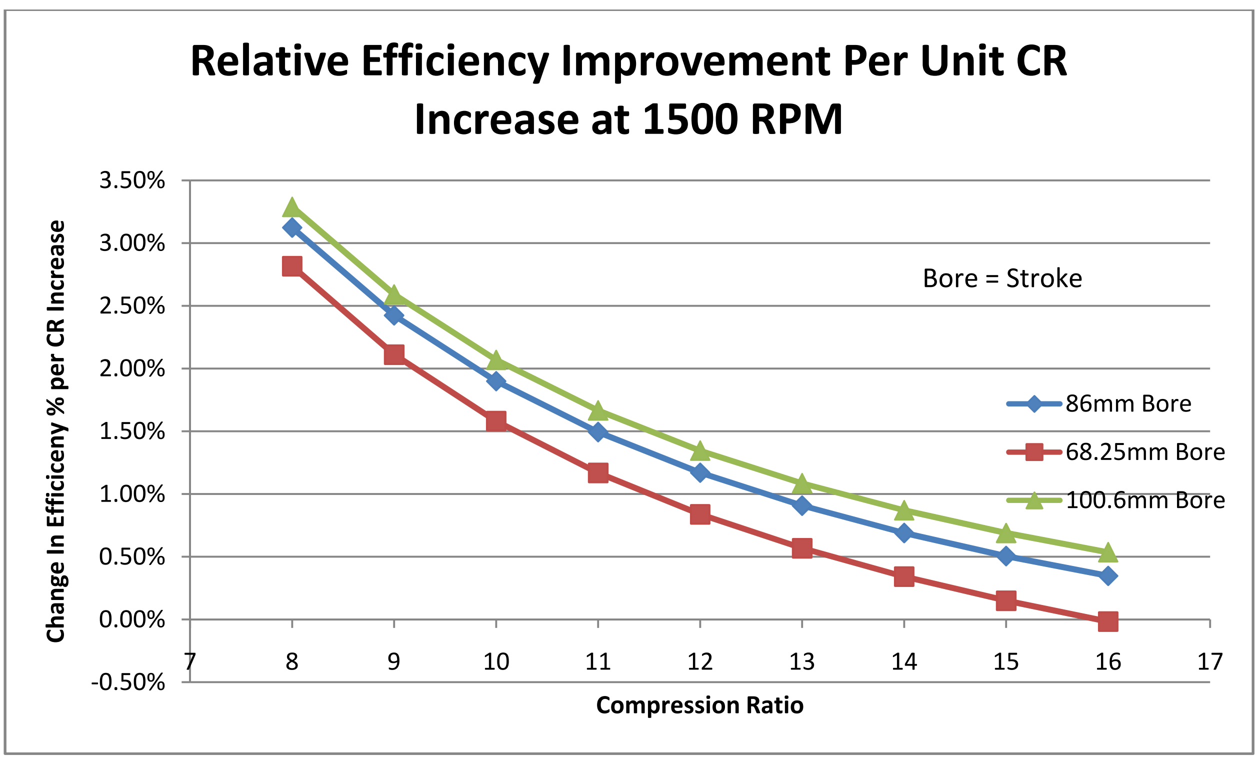

Figure 3-5

It should also be noted that none of the papers reviewed suggest that the method of fuel induction affects the indicated efficiency; i.e., port fuel injection or direct injection should have the same indicated efficiency if all other factors such as bore size, CR, and RPM are held constant.

3.2 ENGINE OCTANE NUMBER LIMIT AND SPARK ADVANCE

It is well known that the actual engine ONR is a function of many engine design variables but is primarily a function of the engine CR, spark timing, and operating point (speed and load) and, to a somewhat lesser degree, a function of the engine bore size and air-fuel ratio. In general, the engine ONR is defined as the octane number of the PRFthat allows the knock limited spark advance (KLSA) timing to equal the MBT spark timing at wide open throttle, and is a function of the RPM for a given engine. Because of the multivariate nature of the relationship, and the fact that spark timing is the primary control variable used to tune the engine for a particular octane number fuel, the relationship between fuel octane number and spark advance is first detailed.

Spark retard from MBT timing reduces the tendency to knock at a given fuel octane number, and the relationship between the level of timing retard and fuel octane number has been extensively studied by Shell (Kalghatgi, Ref. 4). Tests conducted on a single cylinder engine with a 4-valve pent roof combustion chamber (fuel system not documented but appears to be port fuel injected) and a CR of 10.5 with a variety of fuels provide the relationship between KLSA and fuel octane index (defined in Section 2.3) as shown in Figure 3-4.The figure shows a very linear relationship between fuel octane number and spark advance, with the last data point close to spark advance at MBT. The MBT level was indicated as 9° at 1200 RPM with an engine ONR of 105.1, and 12.5° at 3000 RPM with an engine ONR of 103.3, so that both the KLSA and ONR of the engine are strong functions of RPM. The 1200 RPM data provides a relationship of spark advance to OI of about 1 degree per octane number, while the 3000 RPM data provides a relationship of 1 degree per 3 octane numbers. The same paper also provided data on an engine with a smaller bore size of 75mm, and the data at 1200 RPM is shown in Figure 3-5 below. While the ONR of this engine at 1200 RPM is given in the paper as 102 at a MBT timing of 14 degrees, one data point at an OI of 105.3 and a KLSA of 20.5 is provided suggesting further increases in spark timing beyond ONR, and is plotted separately.

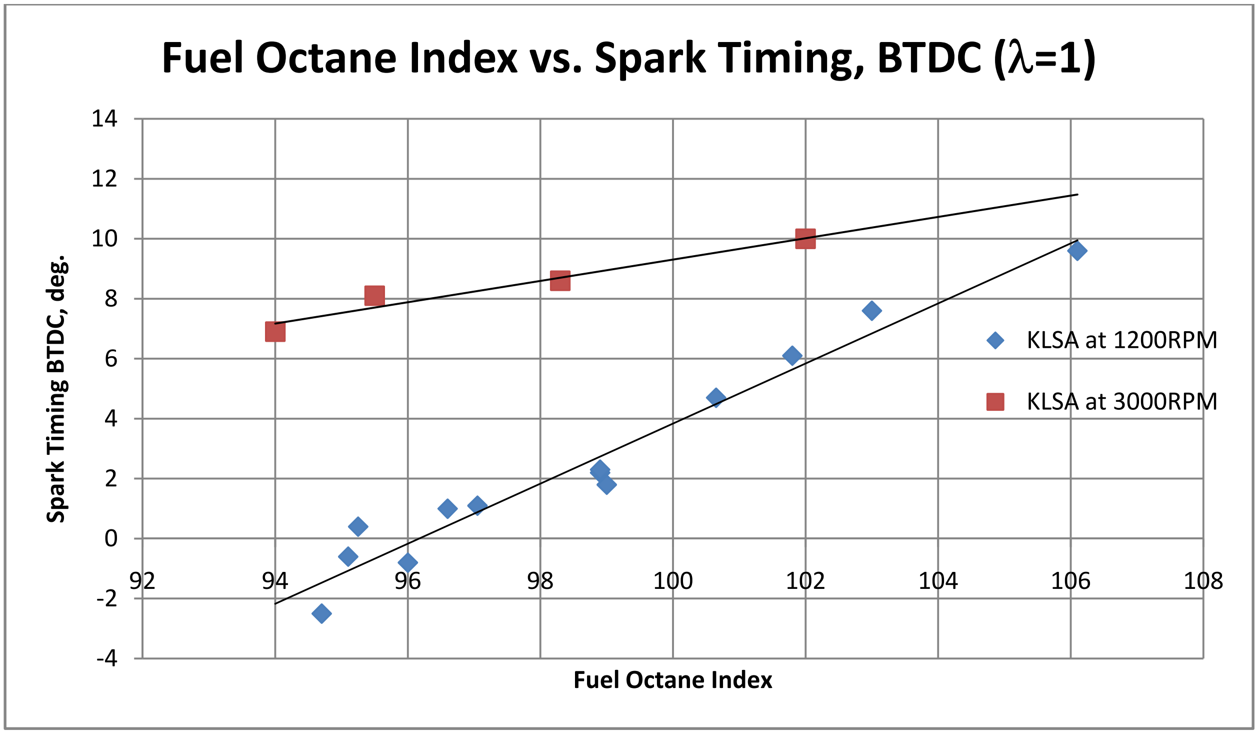

Figure 3-4

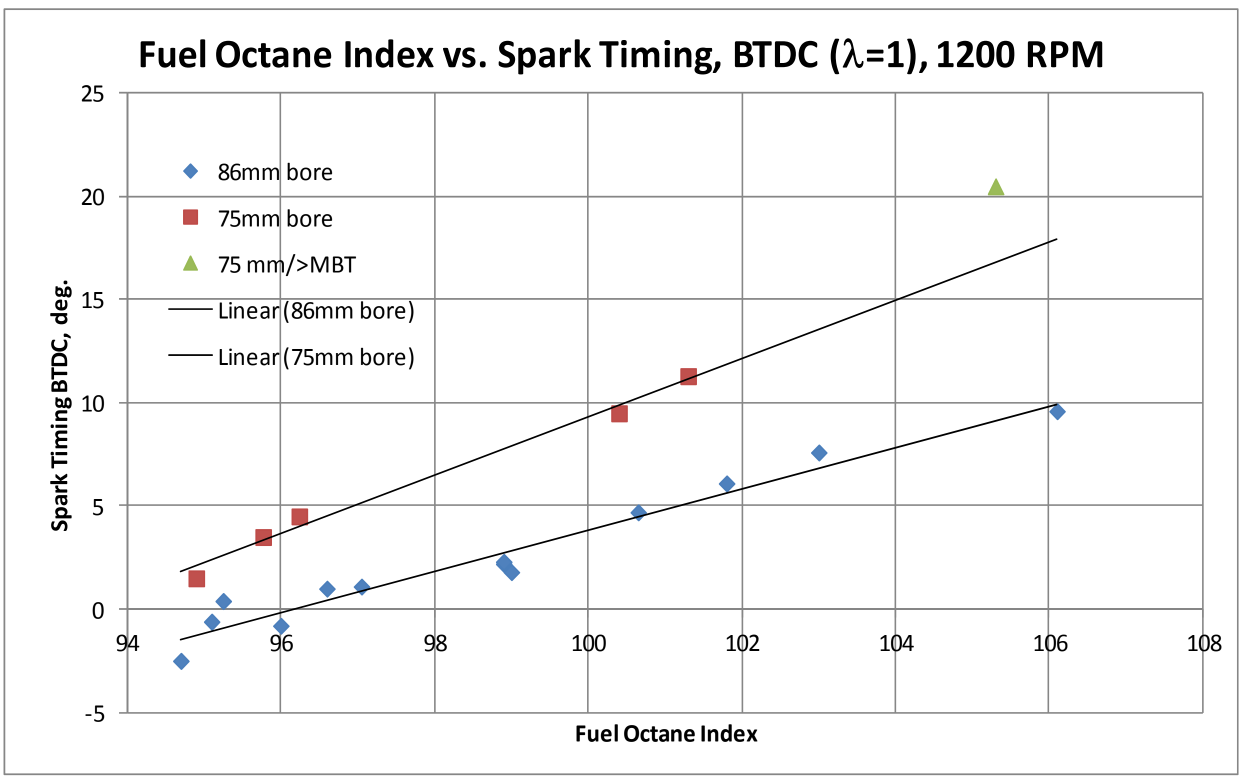

Figure 3-5

These data suggest the KLSA vs. OI relationship extends beyond the MBT spark level, but the data presented in Section 2 and Figure 3-5 suggest that relationship is not linear at spark timings close to MBT. The smaller bore engine also appears to have a higher spark timing coefficient of 1.4 degrees per octane number. Similar tests conducted at Ford (Russ, Ref.16) on a single cylinder PFI engine with a 4 valve head, bore size of 90.2mm and a CR of 10 showed that there was a 5 to 6 degree difference in knock limited spark advance between two fuels with 97 and 91 RON (MON not given), showing the same coefficient of 1 degree advance per 1 octane number increase found by Kalghatgi in Ref. 4, but Russ (Ref. 16) found this difference to be near constant from 1000 to 3000 RPM. He also cites other research confirming this 1 degree per 1 octane number coefficient.

The analysis by Russ on the relationship of octane number to other variables in Ref. 16 examined the effect of CR on spark advance with a 91 RON fuel over the 1000 to 3000 RPM speed range. He found that the 9.1CR engine allowed about 5 to 6 degrees spark advance at KLSA over the 10CR engine, with the plots essentially mirroring the plot of spark advance with RPM for the 97 and 91 RON gasoline. Russ concludes that a unit increase in CR changed the engine ONR by about 5 octane numbers and also cited work much older work by Caris and Nelson (Ref.17) who found that increasing the CR from 9 to 11 resulted in approximately a 9 octane number increase in the engine OR. Russ also examined the effect of enrichment on engine ONR. He concluded that the ONR is highest at λ= 0.95 and declines by 3 octane numbers for every 0.1λ increase or decrease from the 0.95 value.

A similar study by Haghgooie (Ref. 18) at Ford on a single cylinder 4-valve engine with 94mm bore and a CR of 8.7 reported that the KLSA difference between a 91 and 97 RON fuel was around 8 degrees at 1500 rpm and λ of 1, which is higher than the 1 degree per octane point reported in other studies, suggesting that the CR may affect the relationship as well. His analysis also found that increasing inlet air temperature by 60°C resulted in KLSA being retarded by about 8 degrees. Mittal and Heywood (2008,Ref.6) at MIT examined the influence of octane on a single cylinder 4 valve engine of 83mm bore at three different CR values of 9.8, 11.8 and 13.4 (the engine being similar to the one used in Ref. 3). The Mittal and Heywood study (Ref. 6) focused on the issue of octane index by testing fuels with several different MON values but nearly identical RON values. Testing conducted at 1500 RPM and 1 bar intake air pressure (essentially at WOT) with a λ of 1 showed that a 4 increase in fuel octane index resulted in KLSA being advanced by a little less than 6 degrees (CR not provided), which is more consistent with the result from Haghgooie. The relationship of KLSA to fuel OI was found to be linear. In a study by Millo, et al., (Ref.19), using a production Fiat engine with a bore size of 86.4mm, a CR of 9.2, tests were conducted at a wide range of RPM at WOT and a λ of 0.9. At 2000 RPM, the KLSA difference between a 94 and 98 octane number PRF was about 7°, and was nearly constant to 4000 RPM where the difference was about 8°. Hence, the KLSA to ON relationship appears to be higher on average than 1 degree per ON suggested with the other studies reporting a sensitivity of 1.5 to 2 degrees spark advance per ON.

It is not clear if some of these differences in the octane number to spark advance relationship are due to differences in measurement methods, since the specific knock intensity level which is considered the "knock limit" is not precisely defined. In addition, the MBT spark advance is difficult to determine precisely since the torque peak is broad and timing changes of plus or minus 2 degrees may be observed in measurement of MBT spark advance as seen in Figure 2-3. These same concerns would apply to the measurement of ONR for a particular engine.

3.3 ENGINE ONR

Due to the many factors affecting the ONR for a particular engine, the data available do not have the uniformity in test conditions and engine conditions to derive a robust relationship between engine CR and bore size to octane number requirement. Data from Kalghatgi (Ref.4) offers the most consistent series of measurements on the ONR, and the data are shown in the table below for tests conducted at WOT with a λ of 1 on port fuel injected engines.

Engine	**Bore Size**	**CR**	**RPM**	**KLSA(MBT)**	**ONR**
A	86	10.5	1200	9°	105.1
A	86	10.5	2000	12°	103.3
A	86	10.5	3000	12.5°	101.0
A	86	8	1200	15°	88.1
B	75	10.5	1200	14°	102
B	75	10.5	2000	15°	96.4

The data for Engine A indicates that an increase of CR from 8 to 10.5 increases the engine ONR by 17 or about 6.8 octane numbers per CR, while a reduction in bore size by 11mm at the same CR reduces the ONR by 3.1 octane numbers at 1200 RPM but nearly 7 at 2000 RPM. As noted in Section 3.2, the study by Russ(Ref. 16) showed a KLSA sensitivity of about 5° to 6° for a 0.9 increase in CR, which translates into a 4 to 5 octane number increase in his analysis. In addition, the ONR declines by about 2 octane numbers per 1000 RPM increase for Engine A, but by about 6 octane numbers per 1000 RPM for Engine B. The paper by Millo et al., (Ref. 19) shows a change in ONR of about 4 per 1000 RPM at constant spark advance but it is not clear if these relationships are linear near MBT timing.

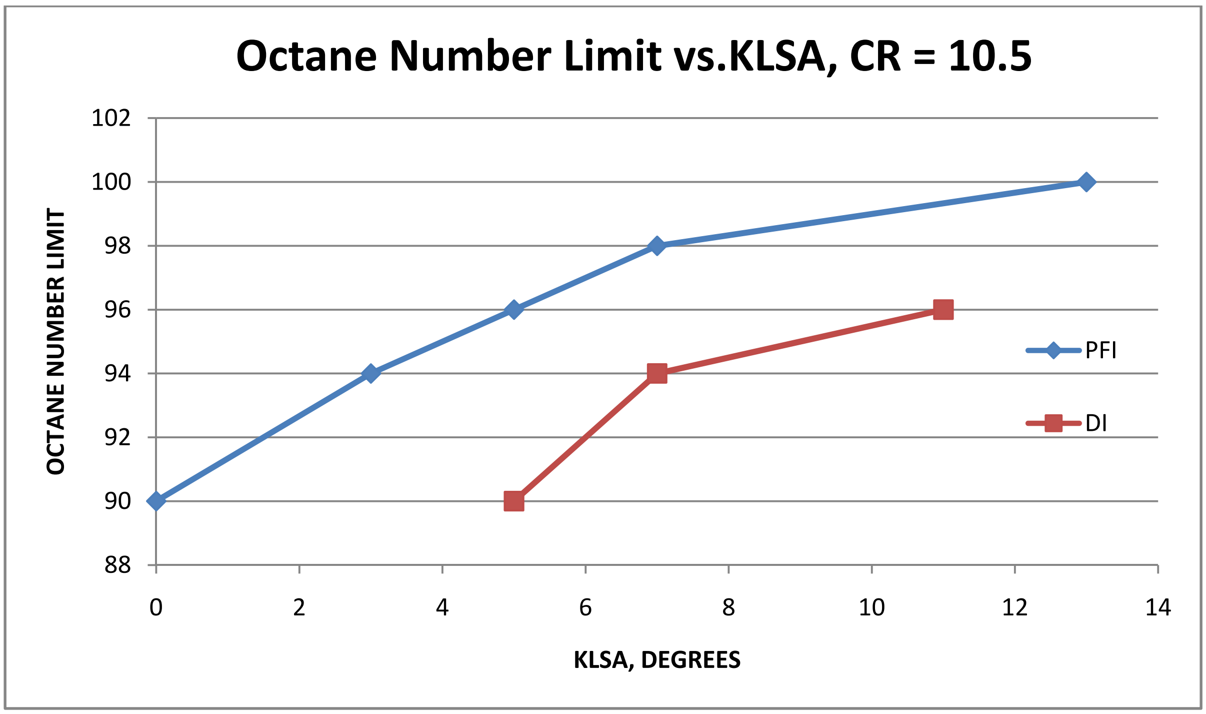

Figure 3-6

A detailed study conducted by Nippon Oil and Nissan (Okamoto, et al., Ref. 20) contrasted the performance and ONR of the same engine over a range of CR values using DI and PFI and provides the most extensive data on a controlled comparison between the octane number requirements of DI and PFI engines. The experimental engine had a 4-valve pent -roof combustion chamber and 86mm bore diameter, and tests were conducted at CR values of 10.5, 12, 13.5, and 15. Tests conditions were 1200 RPM at WOT and $\lambda$was set at 0.8. Figure 3-6 shows that at the same level of spark advance, the ONL of the DI engine is 4 to 5 octane numbers lower than the ONL of the PFI engine. Conversely, at the same fuel octane number, the DI engine had a KLSA that was 5 to 6 degrees more advanced than the KLSA with PFI. The effect of this larger spark advance on torque is non-linear (as shown in Figure 2-3) so that the increase in torque at low octane numbers with DI is quite large; at 90 RON, the DI system provides about 15% increase in peak torque, but at 96 RON, the increase is about 9%.

The ONR for PFI and DI engines is shown in Figure 3-7 and except for one data point for the DI engine at a CR of 12, the other data points indicate that DI has a 4 octane number reduction in ONR at 10.5CR,and the advantage increases with increasing CR. No fuel with an ON over 114 was used so that the ONR for the PFI engine could not be determined but appears to be over 120 from linear extrapolation of the trend line (which may not be accurate if the trend is non-linear). The 4 octane number reduction in ONR was confirmed by auto-manufacturers relative to current CR values for PFI engines which are in the 10 to 10.5 range.

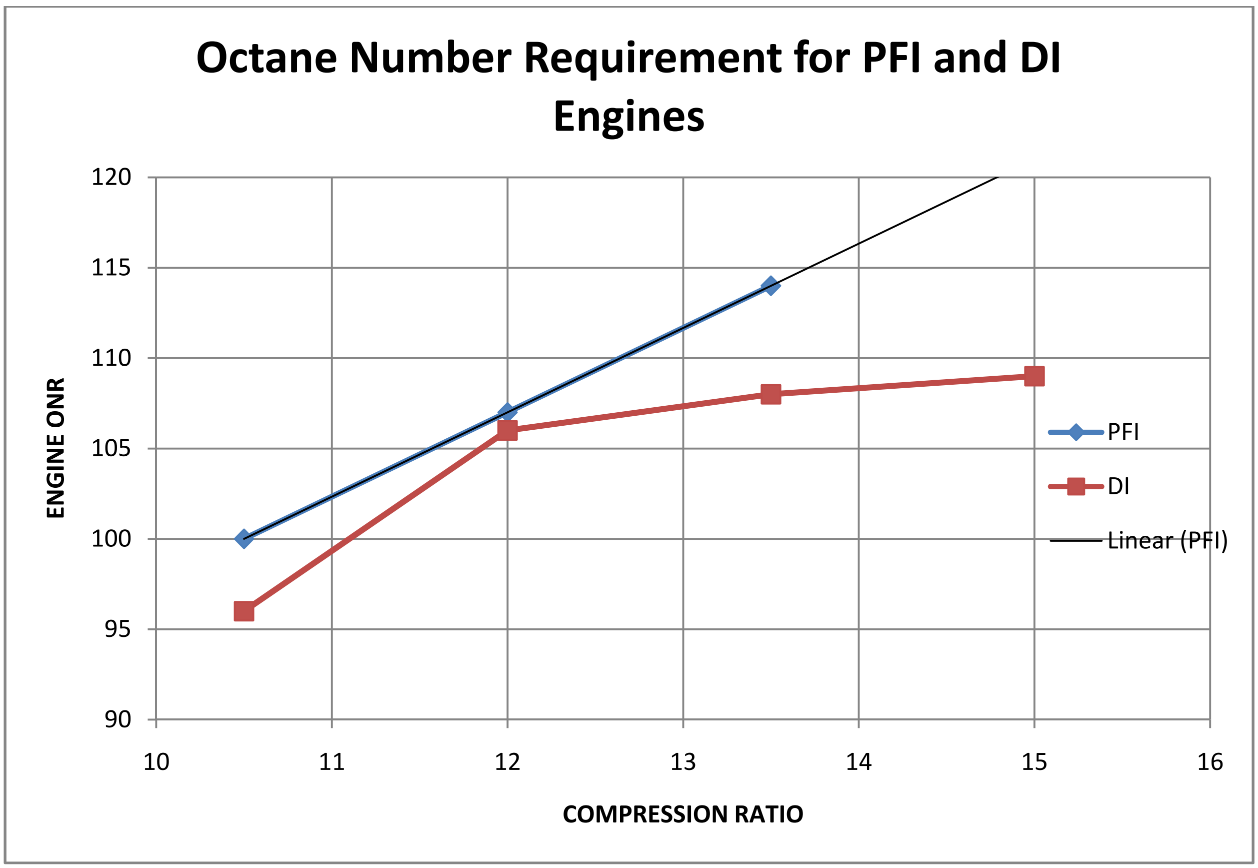

Figure 3-7

The Nippon Oil/Nissan paper (Ref. 20) also confirmed that brake specific fuel consumption of PFI and DI engines were near identical at the same CR, providing that both engines operated at MBT (i.e., spark was not retarded due to knock). The data were generated at 40 Nm, 80 Nm and 120 Nm torque values (which we estimate corresponds to about 25%, 50% and 75% of maximum torque) and at 1200 and 2400 RPM. Hence, the thermal efficiency benefits of increasing CR described in Section 3.1 of this report should apply equally to both PFI and DI engines.

A few other reports have provided ONR values for DI engines although many of the papers from the late 1990sto early 2000s have also used prototype engines or early production DI engines. In a joint paper by Shell and Toyota (Kalghatgi, Nakata and Mogi, Ref. 5), the ONR of a prototype DI engine (unspecified bore size) was determined at different CR values of 11.0 and 12.5 over a range of engine speeds from 1200 RPM to 6000 RPM. At 1200 RPM, the ONR at 11CR was found to be 106.8 and ata CR of 12.5, the ONR was found to be 112.8. The values are

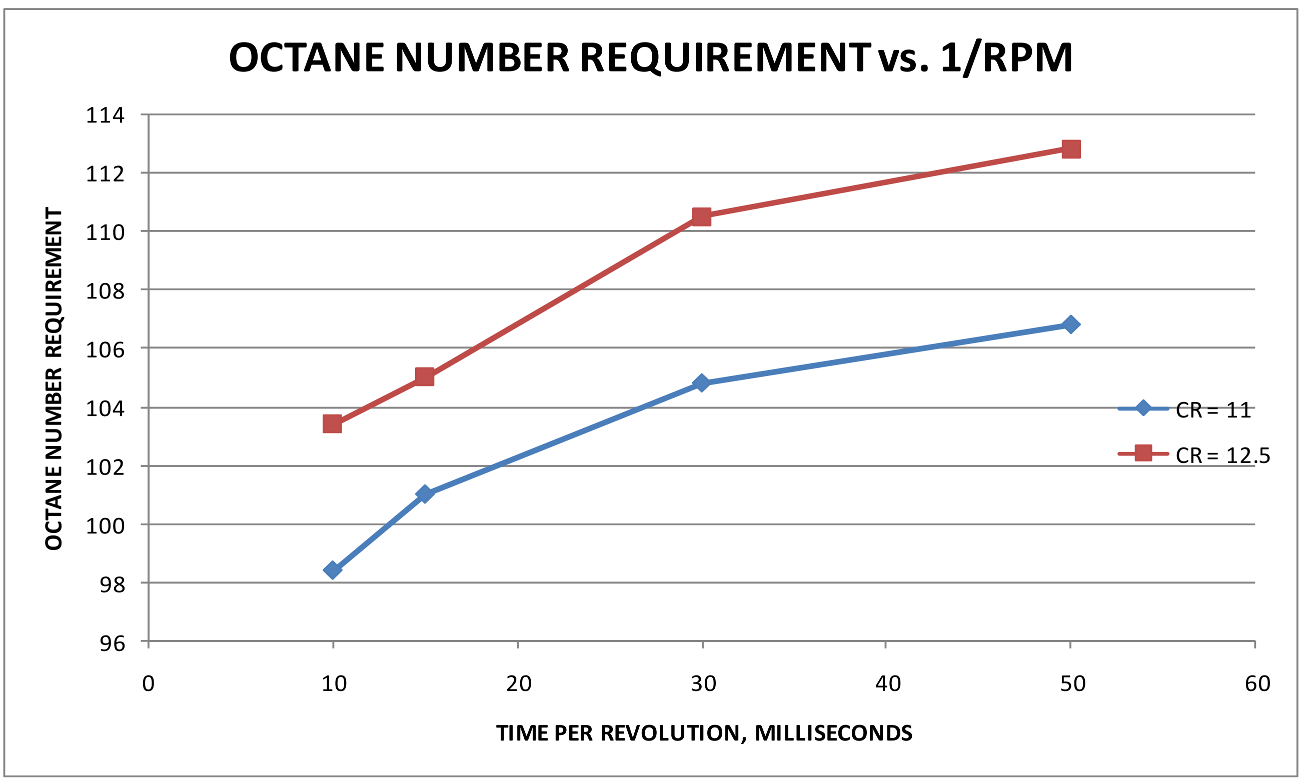

Figure 3-8

significantly higher than those reported by Nissan, especially considering that the ONR values are at a λof 0.85 which should reduce the requirement by 4 to 5 octane numbers relative to the

ONR at stoichiometric air-fuel ratio. The relationship of ONR with the time per revolution (inverse of RPM) appears to flatten out at low RPM, as shown in Figure 3-8.

In a study of fuel octane number effects on a DI engine by Toyota and ExxonMobil(Akihama, et al., Ref. 21) an engine with a CR of 13 and 86mm bore size was tested at 2000 and 4000 RPM, WOT and a λ of 0.87 with a range of fuels. Although the ONR of the engine was not directly determined, the KLSA at 100 octane number (PRF fuel) was retarded by about 11 degrees from MBT at 2000 RPM and by 6 degrees at 4000 RPM, suggesting an ONR of about 106 to 107 at 2000 RPM and 103 to 104 at 4000 RPM. This is about 2 to 3 octane numbers lower than the ONR of the 12.5CR engine shown in the figure above, consistent with the expectation of a 4 to 5 number ONR increase per CR. In another study by the Cosmo Research Institute (Fukui, et al., Ref. 22) a Mitsubishi DI engine with an 81mm bore and a CR of 12 was tested with several fuels. Although the engine ONR was not explicitly determined, the data show that the ONR at 1200 RPM is about 100 as isooctane allowed operation with MBT spark timing at borderline knock conditions. This ONR level is lower than those reported by the two papers cited above.

A study by BP and Toyota (Williams, et al., Ref. 28) reported on the efficiency as a function of fuel octane number for a PFI Atkinson Cycle engine with a geometric CR of 13. According to Toyota, the actual CR is about 9.5 at 1500 RPM and 10.6 at 2800 RPM. Tests at WOT and λ=1 show the engine to have maximum thermal efficiency at a fuel RON of 98 at 1300 RPM and about 100 RON at 2800 RPM

The following ranges of ONR as a function of CR at low RPM (<1500) for a bore size of about 85mm and a λ of 0.9 to 1, are approximate averages of the papers reviewed and manufacturer inputs received:

CR	ONR for PFI	ONR for DI
10	100 – 102	94 – 96
11	105 – 107	99 -101
12	109 – 111	103 -105
13	113 – 115	107 – 109
14	117 – 119	110 – 112

3.4 EFFECT OF LATENT HEAT OF VAPORIZATION OF FUEL

As noted in Section 2.5, the latent heat of evaporation results in a lower air-fuel mixture temperature for a DI engine which improves the volumetric efficiency. In addition, the cooler mixture allows a higher CR with the same fuel octane number. The reduction in ONR by 4 due to DI was discussed in Section 3.3. The Nippon Oil/ Nissan paper (Akihama, et al., Ref. 21) showed that the increased volumetric efficiency results in a 5% to 6% improvement in torque with additional improvement from the CR increase which depends on the fuel octane number and spark timing. Figure 3-9 shows the net torque improvement at KLSA comparing the same engine with DI and PFI at 4 different CR values. A net improvement of torque of about 7% if observed with DI over a range of CR values. With the 114 RON fuel, the DI engine was not knock limited except at 15 CR, while the PFI engine was knock limited at CR values of 13.5 and higher. In the case of the 90 RON fuel, both the DI and PFI engines were knock limited at all CR values.

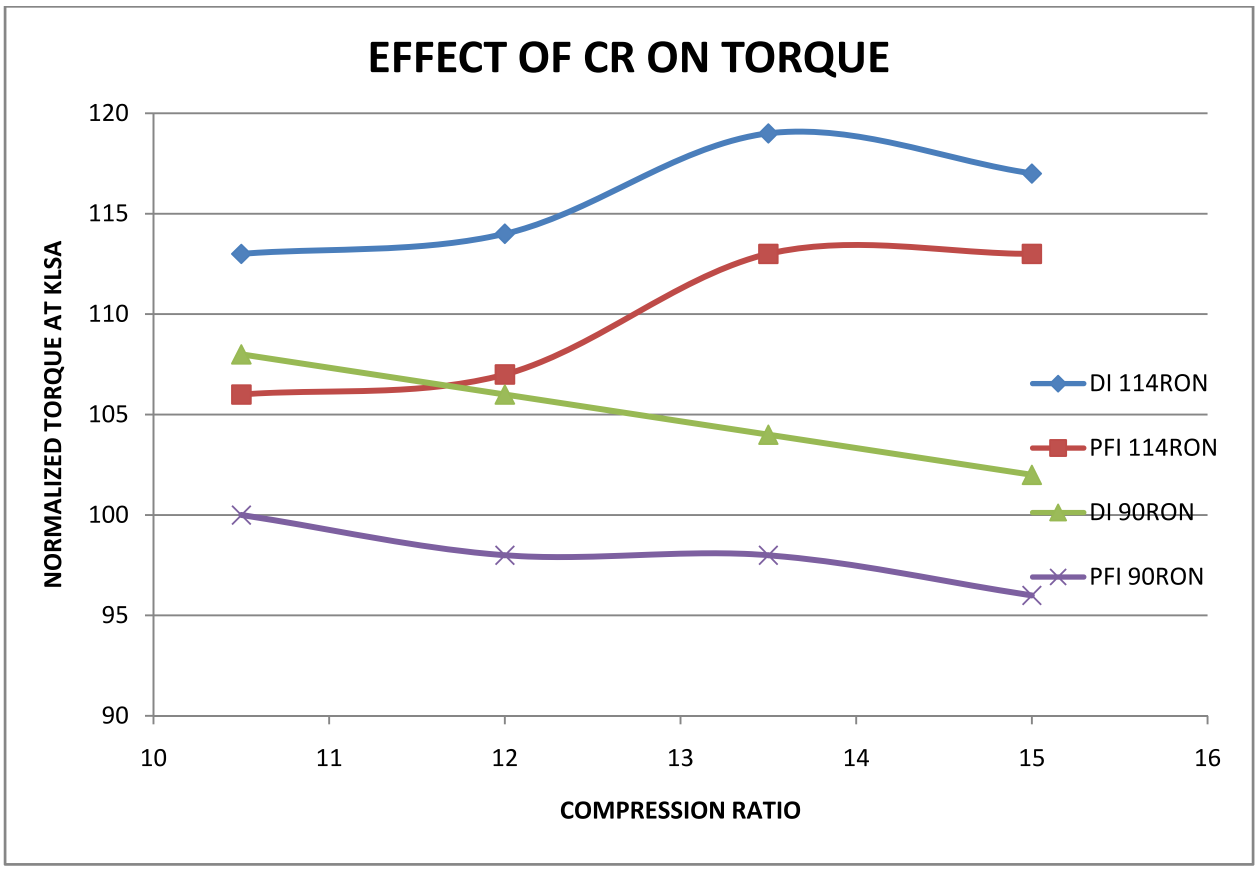

Figure 3-9

In an analysis by Toyota (Nakata, et al., 2006, Ref. 23), the 1.5L Atkinson Cycle PFI engine used in the Prius with a geometric CR of 13 and a bore of 75mm was tested with different levels of ethanol blends with a base regular gasoline of 91.5 RON and with a Japanese premium gasoline with a RON of 99.6. According to data provided by Toyota, the actual CR of the engine at 2800 RPM is 10.6. The paper reported that the octane number difference between premium and regular gasoline resulted in only a small improvement in thermal efficiency because the Atkinson Cycle limits the volumetric efficiency. With ethanol blends, the volumetric efficiency can be improved due to charge cooling even with a PFI system if the ethanol is inducted into the cylinder mostly as a liquid, and the engine can run at MBT timing with E50 and higher blends. With neat ethanol (E100), the paper reported that absolute efficiency approached 39.9% which was 5.6% relative improvement over the base (E0) efficiency level. The data are shown in Figure 3-10 below for tests conducted at 2800 RPM and $\lambda = 1$.

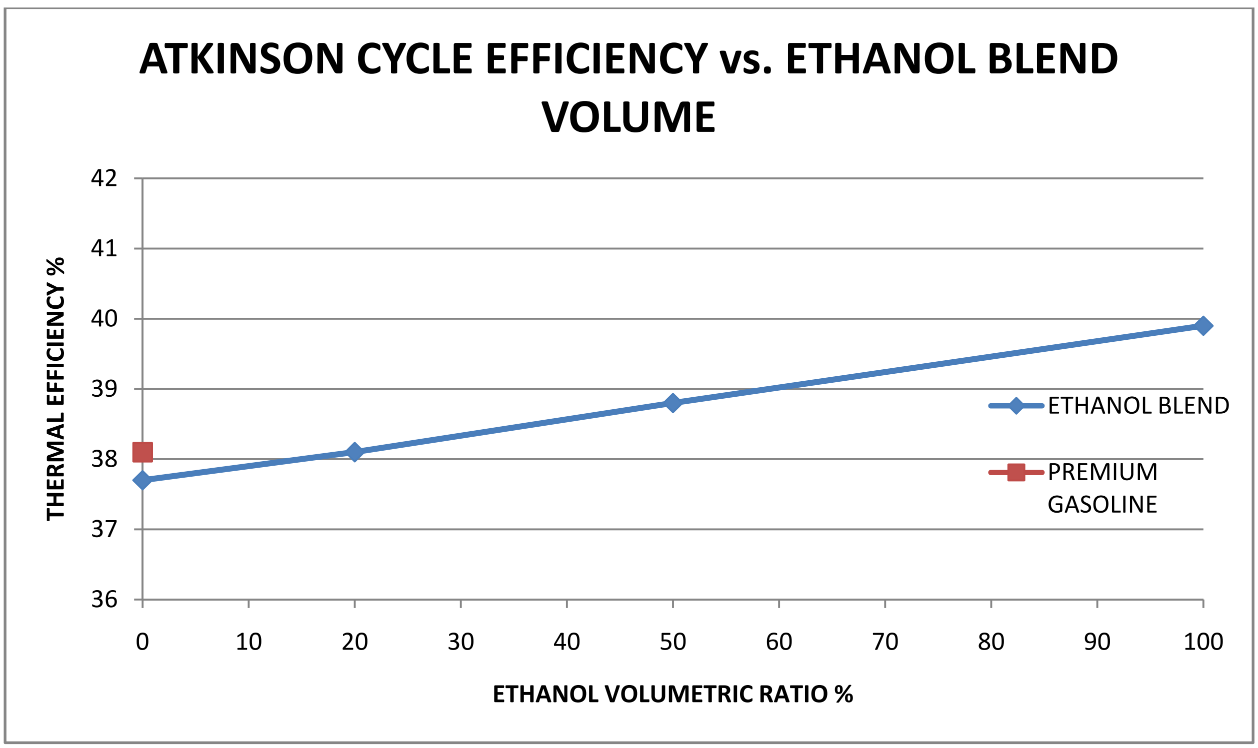

Figure 3-10

In an earlier analysis by Toyota (Nakata, etal., 2006, Ref. 12) similar tests were conducted on the same engine with a CR of 13 without the valve timing set up for Atkinson Cycle operation. In this case, volumetric efficiency was affected by increased ethanol concentration, decreasing by 2% for E50 and higher concentration blends. However, thermal efficiency for E0 (i.e., the base 91 RON gasoline) was reported as 31.7% absolute; thermal efficiency for E100 was reported as 39.6% absolute, nearly the same as that of the Atkinson Cycle engine operated on E100, and also higher than the 37.9% reported on 100 RON gasoline. These efficiency values show that use of E100 resulted in 24.9% relative efficiency gain, and the gain with E50 appeared to be on the order of 20% relative, significantly higher than with the Atkinson Cycle engine.

In another paper by Toyota (Taniguchi, et al., Ref.24), the use of E100 was examined in a DI engine with a high CR of 13 and a bore size of 87.5mm. At WOT, torque increased by 7.6% relative to the same DI engine with a CR of 11.5 running on gasoline with a RON of 96.4. The volumetric efficiency <u>improved</u> by up to 3.8% in contrast to the PFI case. The increase in torque was ascribed to both the increase in volumetric efficiency and the increase in CR made possible by the higher RON of ethanol and the lower mixture temperature. Surprisingly, the thermal efficiency with E100 was found to be only 0.5% (relative) better than with E0 which Toyota traced to oil dilution by ethanol. During the test, blow-by gases were vented and the lost ethanol was not recovered; whereas, the blow-by gas would be re-used in the normal situation due to positive crankcase ventilation. The higher volume of E100 required per injection created 3 to 4% higher fuel losses from oil dilution on the cylinder walls relative to gasoline, especially at high RPM. In addition, the lower flame speeds with ethanol contributed to a 0.7% decline in relative efficiency, which contributed to the small net effect observed.

A study by Caton, Hamilton and Cowart (Ref. 26) on a CFR engine equipped with PFI used regular gasoline, E10, and E85 to examine the effects of ethanol at a constant speed of 900 RPM. At a CR of 9 and a $\lambda$of 0.9, E10 allowed a 0.3 bar increase in IMEP (about 3%) by allowing about 5 degrees spark advance over regular gasoline at KLSA, while E85 was not knock limited and provided a very small IMEP increase over E10. In a sweep of different CR values, the authors found the E10 blend allowed a 0.5 CR increase over regular gasoline, and knock control required spark retard of about 5 degrees per unit CR increase. E85 was knock limited at CR ratios over 12 but required only 2 degrees of spark retard per unit CR increase. Torque with E85 increased by 2% per unit CR increase, but torque was flat to negative with CR increases using regular gasoline or E10 due to the need for spark retard to avoid knock.

It should be noted that the higher torque with ethanol and ethanol blends can influence vehicle fuel efficiency over and above changes in engine efficiency by allowing some engine size reduction without loss of power.

3.6 LEAN OPERATING CONDITIONS

As the relative air-fuel ratios are increased beyond stoichiometric, engine efficiency depends strongly on combustion chamber design, turbulence in the chamber and compression ratio (Heywood, Ref. 1). The flame speeds under lean conditions are slower as shown in Figure 2-8 but this is compensated for by other factors. Ayala, Gerty and Heywood (Ref. 3) investigated the factors that affect indicated efficiency as a function of air-fuel ratio, and reported on four factors that shape the relationship:

- The value of γ increases towards that of air as the mixture gets leaner.
- Pumping losses are reduced at part throttle conditions due to a higher airflow requirement to produce the same torque.
- Heat transfer losses are reduced due to lower combustion temperature.
- The burn duration increases with increasing air-fuel ratio.

The first three factors contribute to positive increases in efficiency while the fourth is a negative factor. The effect of increasing γ is relatively flat at air-fuel ratios over 1.4 so that as burn duration increases, the loss in efficiency offsets the gain from reduced pumping and heat transfer loss. Based on the data shown in Ref. 3, Figure 3-11 shows the efficiency effects of the different components at a light load point of NIMEP at 3.5 bar (which we estimate would typically correspond to a BMEP of about 2.3 bar). In this case, net indicated efficiency peaks at an air-fuel ratio of about 1.5. The authors note that the effect of burn duration becomes particularly pronounced when the duration exceeds 30 crank angle degrees, and is in good agreement with the other literature reviewed here.

The influence of the mass burning period on IMEP and indicated efficiency has been confirmed in other papers. Research at the JOMO Technical Center (Hashimoto, Inaba, and Akasaka, Ref. 25) where a stratified charge DI engine with a CR of 10 and a bore size of 86mm was tested at WOT at a λ of 1, 2 and 3 with a wide variety of fuels found a linear relationship between the 90% mass burn period and IMEP at a λ of 1. They also found that at lean conditions, the IMEP was much less sensitive to spark timing due to the long duration of combustion. Olefins had shorter mass burning periods than paraffin and aromatic compounds at lean conditions, resulting in higher IMEP.

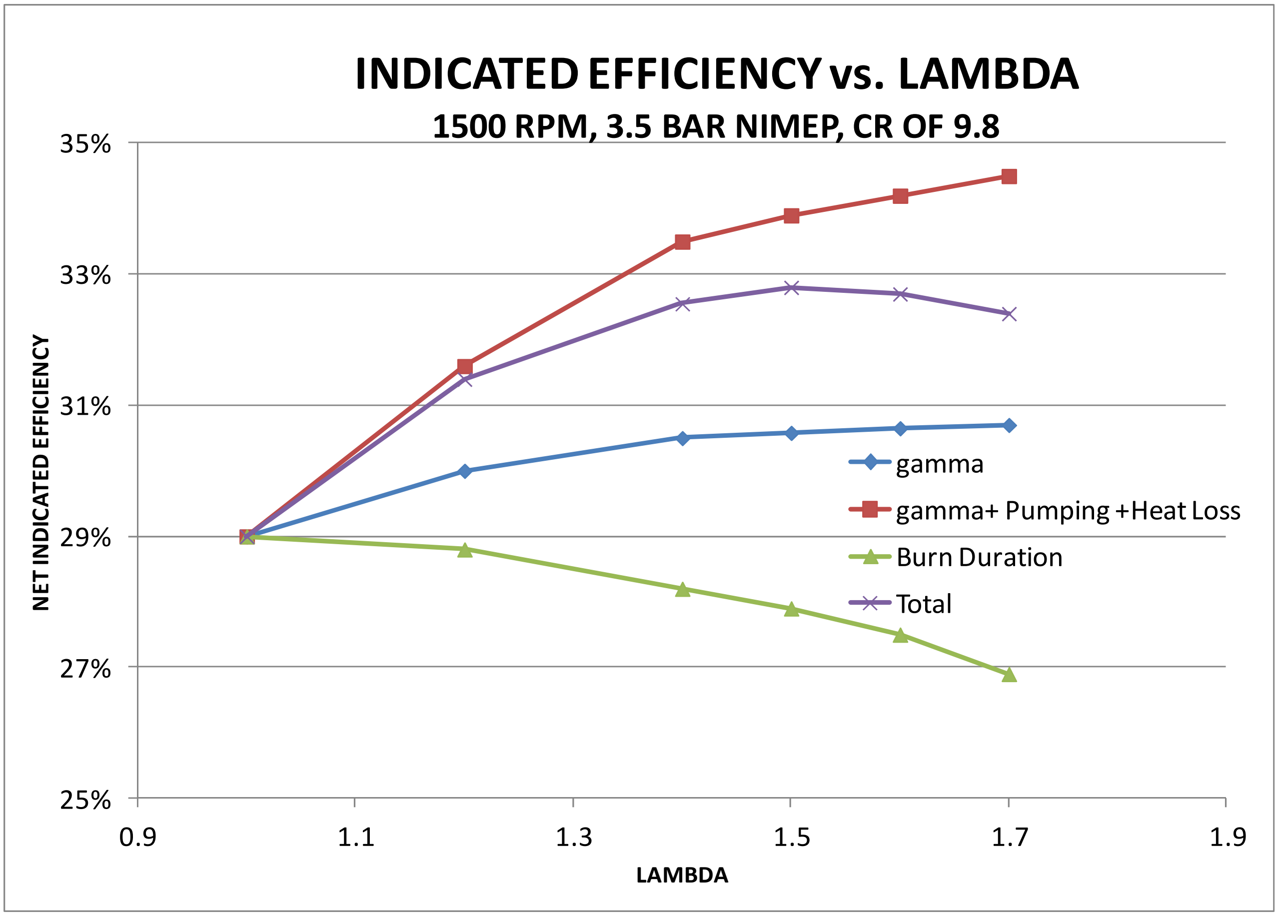

Figure 3-11

The effect of octane number at lean combustion conditions was investigated by the Cosmo Research Institute (Fukui, Komoriya, and Shimizu, Ref. 22) on a single cylinder version of a Mitsubishi DI engine with a CR of 12 and a bore size of 81mm. A wide variety of fuels with RON ranging from 67.5 to 102.2 were tested on this engine at 1200 RPM, 67 and 87 kPa manifold pressure and a λ of about 1.8 or 2. At these conditions, the indicated efficiency was highest at a low RON around 70 as shown in Figure 3-12. At these low RON values, the refinery feed-stocks had high auto-ignition intensity but single component fuels had relatively low intensity. The authors speculated that the low octane number fuels had better efficiency as the fuel mixture near the walls auto-ignited with low octane number fuels, providing more complete combustion. The increase in HC emissions with higher octane number fuels confirmed this hypothesis and the authors also observed a correlation between auto-ignition and thermal efficiency.

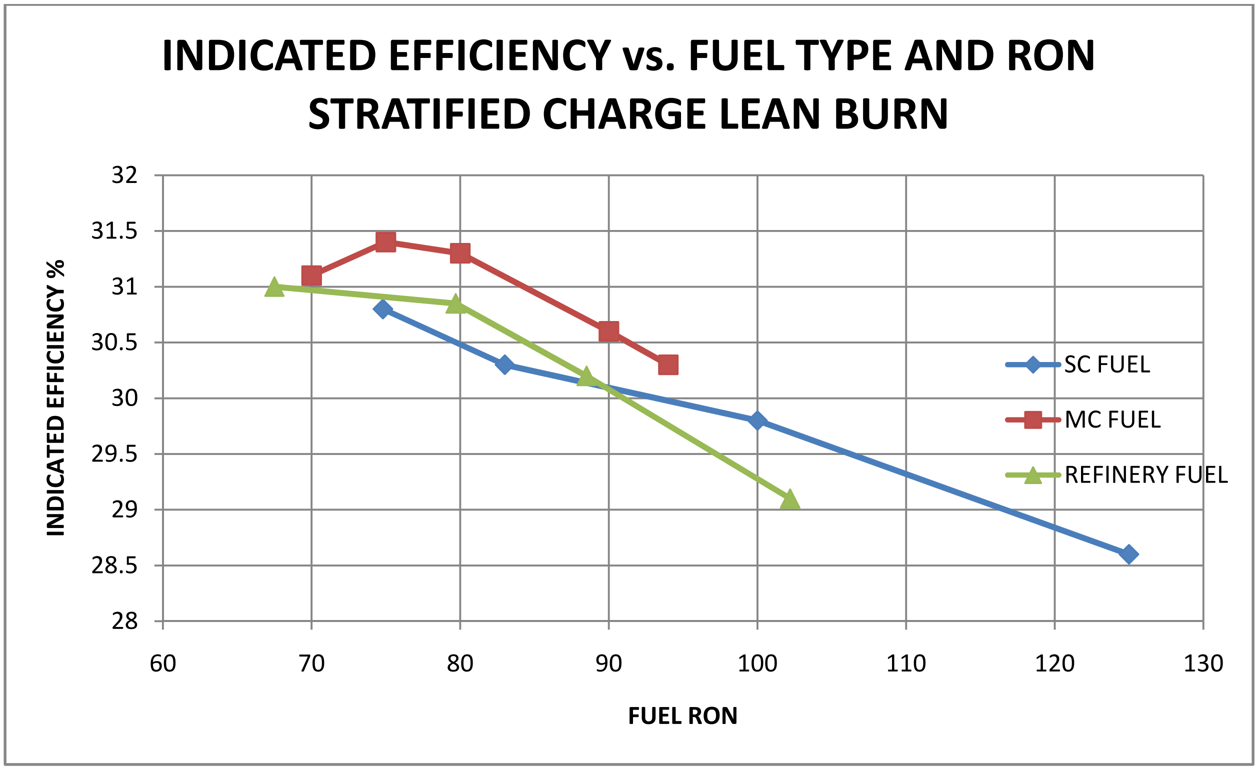

Figure 3-12

A similar result was obtained by a Toyota-Exxon-Mobil study (Akihama, et al., Ref. 21) with DI engine having a CR of 13 and a bore size of 86mm. Tests were conducted at a λ of about 1.9 and 1200 RPM engine speed and a “slightly throttled” intake. Two fuels were used, a toluene standardization fuel (TSF) with a RON of 83.8 and Japanese regular gasoline with a RON of 91.7. The data presented in the paper show that the lower octane number TSF provided about 5% greater torque and had about 5% greater brake efficiency than the Japanese regular fuel. The indicator diagrams showed that there was a two stage heat release on the TSF, with one corresponding to the flame propagation from the spark plug and the second corresponding to spark induced compression ignition. The authors stated that this auto-ignition differed from knock in that the pressure rise was not excessively rapid. The authors surmised that if CR is increased further, this type of spark induced compression ignition could occur at RON values similar to regular gasoline (i.e., at 91 RON).

During the interviews, manufacturers confirmed that low octane number fuels were better suited to very lean combustion that approached Homogeneous Charge Compression Ignition (HCCI) combustion. On the other hand, they also stated that at high loads, such lean air-fuel ratios could not be used without very large power loss so that this phenomenon was not directly relevant to production engines, as low octane number fuels could not be used across the operating range.

4. TURBOCHARGED PORT INJECTION AND DIRECT INJECTION ENGINES

4.1 OVERVIEW

Turbocharged engines are treated separately since the use of turbocharging generally requires the reduction of CR for a given level of fuel octane number, which hurts engine thermal efficiency. However, the use of turbocharging allows the use of a significantly smaller engine to replace a naturally aspirated engine, and the mechanism that improves vehicle fuel economy is the reduced pumping and cooling loss of a smaller engine operating at higher load to provide the same output to the vehicle.

Turbocharged engines also operate against different constraints. At low RPM (<1200), the turbocharger does not provide significant boost, and maximum boost is usually not available until 1800 to 2000 RPM or higher. Hence the WOT condition at low RPM may not be the point where the engine ONR is the highest. In addition, boost may be limited at high RPM due to maximum cylinder pressure constraints, and spark timing and air-fuel ratio can be constrained by turbine inlet temperature constraints. Hence, the impact of fuel octane number may be dissimilar between naturally aspirated and turbocharged engines. Turbocharged engines are knock limited over more of their operating range than naturally aspirated engines, (see Figure 4-1 below), and octane number benefits in terms of vehicle efficiency may be larger.

The efficiency of the turbocharged engine and its relationship to CR and spark advance are also not represented by the equations shown in Section 3, since the reciprocating engine and turbo unit act as a system. If the engine is not boost limited, the optimal combustion phasing for best torque is delayed relative to that for a naturally aspirated engine since there is a compromise between enthalpy flow to the turbine and the best torque achieved by the engine. At high RPM, the boost, air-fuel ratio, and spark advance at WOT have to be limited not only to avoid knock but also to limit turbine inlet temperature, so that fuel octane number effects are very non-linear.

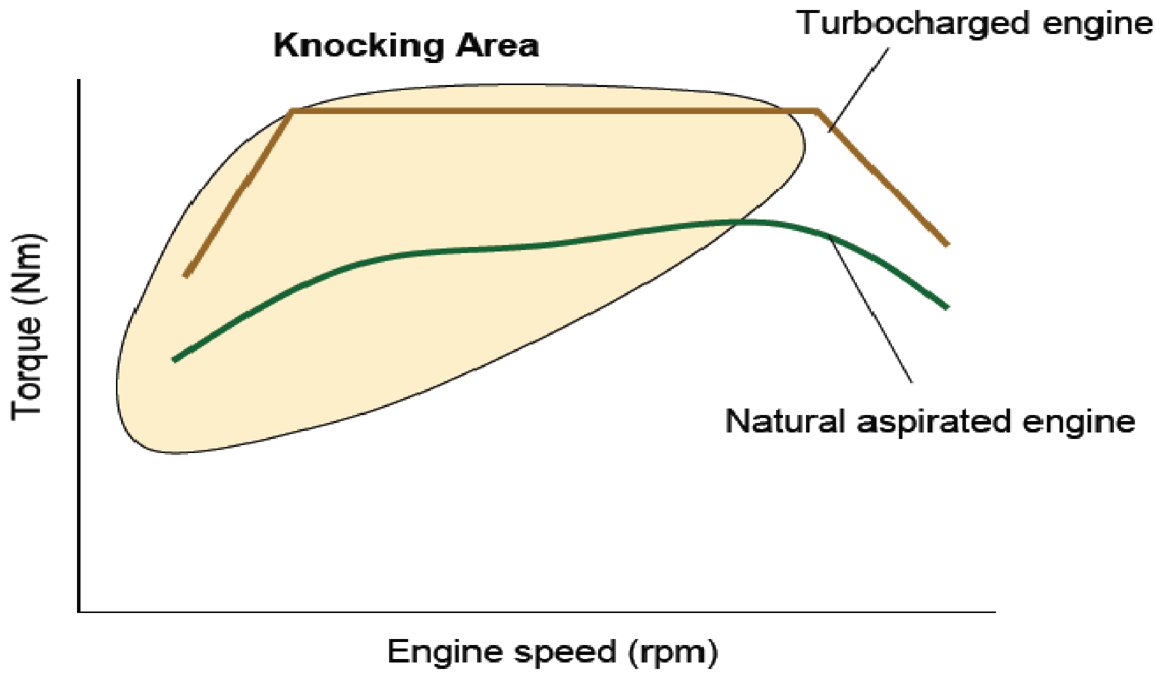

Figure 4-1

4.2 PORT FUEL INJECTED ENGINES

Turbocharged PFI engines have had modest penetration in the U.S., largely because of weak low RPM torque. An analysis of a 2L PFI engine with turbocharging using fuels of different octane numbers was reported in a paper by Renault and TOTAL (Duchaussoy, Barbier, and Schmelzle, Ref. 27). The engine had a 9.5 CR and a bore size of 82.7mm and was tested with hydrocarbon fuels ranging in RON from 92.5 to 106.5. Sensitivity for all of the fuels was about 10. The engine was tested at WOT and the air-fuel ratio was held to stoichiometric across the entire operating range of RPM from 1200 to 5000. The results are shown in Figure 4-2, and the reduced low RPM torque is obvious. Due to lack of boost, the different octane number fuels have almost no effect at RPM below 1500 RPM but by 2000 RPM, the 106.5 RON fuel shows a 11.5% torque advantage over the 92.5 RON fuel. However, the effects are quite non-linear with RON as the 98.3 RON fuel shows only a 3.9% torque advantage. At 3000 RPM, the torque differences between the various fuels are quite large, but decline at higher RPM as turbine inlet temperature limits control available timing retard and boost. At 4000 RPM, the 106.5 RON fuel offers a 43% torque advantage over the 92.5 RON fuel.

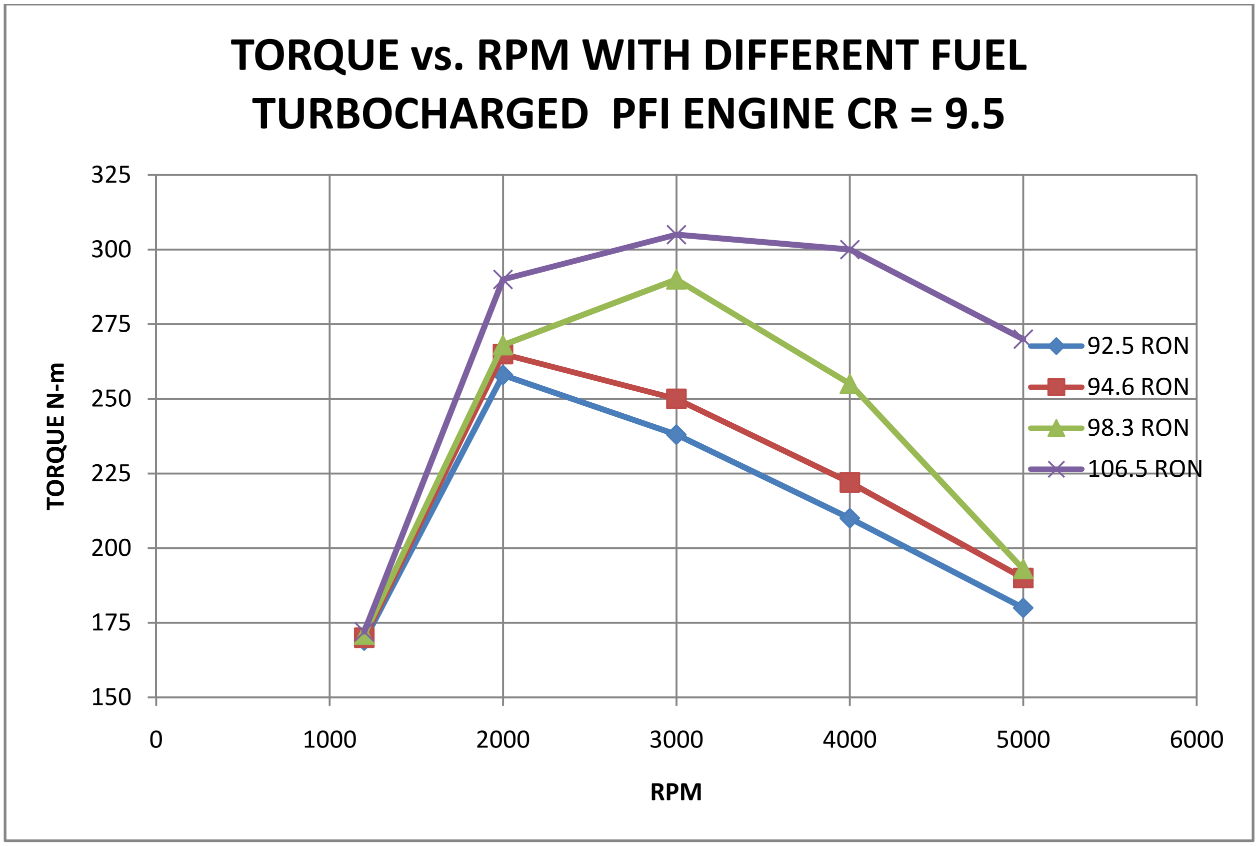

Figure 4-2

Reference 27 also examined the effect on increasing MON by decreasing the Sensitivity of the fuels at constant RON from 10 to 7 (i.e., a 3 increase in MON). The authors found that the 3 MON increase resulted in about 3% increase in torque at 1500 RPM, but the differences were quite small over 3000 RPM. Hence, MON increases were found to be helpful at low RPM, which is exactly opposite from the result for naturally aspirated engines.

By plotting the results shown in Figure 4-2 against RON with RPM as a parameter, the complex response of turbocharged PFI engines to fuel octane number is illustrated. Figure 4-3 shows the torque as a function of the RON at three different RPM levels, and the sensitivity to RON changes as different constraints come into play – boost constraints at low RPM, knock constraints at mid-RPM and turbine inlet temperature constraints at high RPM. Also, the efficiency does not increase with torque as boost varies across the fuels but the ranking of fuels for 1/BSFC (i.e., a measure of efficiency) was identical to the torque based ranking. BSFC varied by about 15% across fuels. Each octane number increase in fuel RON provides a 2% increase in peak torque at 3000 RPM, and a 3% increase at 4000 RPM.

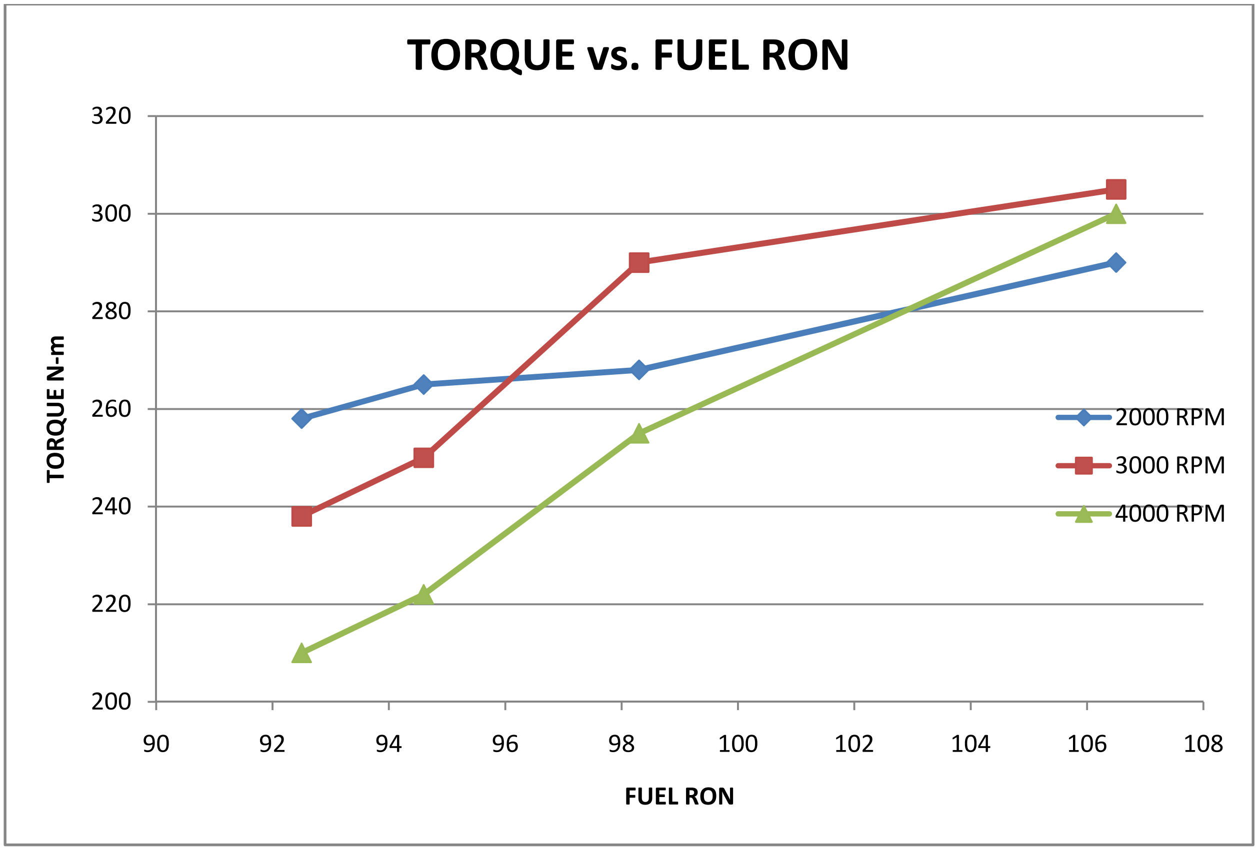

Figure 4-3

BP and Toyota (Williams, et al., Ref. 28) also investigated a 1.8L turbocharged PFI engine with a bore size of 80.5mm, but the engine had a very high CR of 13 for a turbocharged version. The paper provided details on engine efficiency at WOT and 2800 RPM with a λ of 1 and 1.6, but no other details such as the boost level and torque output are provided. Two reference fuels with 100 and 102 RON were tested with a wide variety of additives (such as diethyl ether and nitro-methane) toimprove the burn rate. The fuel matrix also included E85 and a high aromatic content 108 RON fuel. The efficiency improvements measured at the two air-fuel ratios are shown in Figure 4-4 below. The authors attributed part of the data scatter to the large confidence bands generated by the test method used to measure the calorific value of fuels.

Operation at the highest RON value of 108 required reduction of boost to meet cylinder pressure limits which reduced efficiency. However, Figure 4-4 shows that relative efficiency increases about 0.69% per octane point at λ= 1 and by 0.85% per octane point at λ = 1.6. It should be noted that at 2800 RPM, there are likely no constraints on available boost or turbine inlet temperature for these operating conditions.

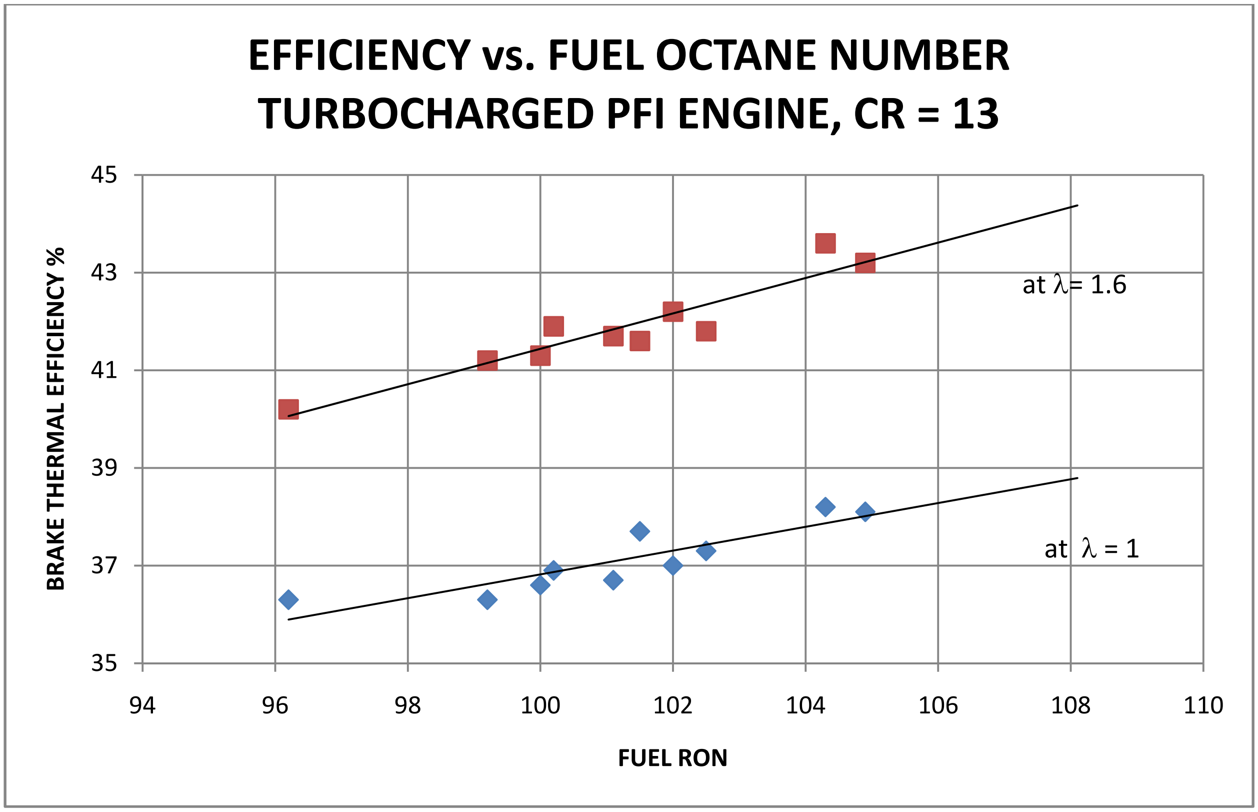

Figure 4-4

In an earlier paper by BP and Toyota (Williams, et al., 2009, Ref. 29), the same engine was tested with butanol-gasoline and ethanol-gasoline blends, as well as neat ethanol and butanol. The fuels spanned a range of RON numbers from 97.6 to 111(for neat ethanol). The efficiencies increased linearly with RON much like Figure 4-4. Indicated sensitivity of thermal efficiency for both the λ equal to 1 and 1.6 cases was in the order 0.9% to 1% per octane number, a little higher than for the case with additive improved RON found in Reference 28.

A more recent paper from Toyota (Nakata, et al., 2011, Ref. 30) provided additional details on the engine operating parameters with increased RON fuels. The paper showed that intake manifold pressure at the operating condition of 2800 RPM, WOT, and λ of 1.6 increased from about 30kPa with a 92 RON fuel to about 80kPa with a 105 RON fuel and BMEP increasing from 0.75MPa to about 1.4MPa. The increases in both intake manifold pressure and BMEP appeared to be almost linear with increasing octane number. Hence, a 15 point increase in RON provides a near doubling of torque and boost at the lean operating condition, and the authors stated that a diesel like 44% brake thermal efficiency was possible with a lean boosted engine operating on neat ethanol. Some of this benefit was potentially due to the cooling effect of ethanol which would permit more boost.

4.3 DIRECT INJECTION ENGINES

As with naturally aspirated engines, turbocharged DI engines are influenced by the reduction in mixture temperature due to the latent heat of evaporation of the fuel. A French consortium study (Milpied, et al., Ref. 31) examined the effects of octane number and latent heat of vaporization on an experimental single cylinder 4-valve engine with a 69mm bore and a CR of 9.5. Six hydrocarbon fuels with RON values ranging from 91.6 to 100.4 and sensitivity from 6.4 to 12.7 were tested to study the effects of RON and MON. Three of these six fuels were blended with ethanol and ETBE (ethyl tertiary butyl ether) to obtain a RON of about 100 but to have a different heat of vaporization. Tests were conducted at 1000, 2000, 3500, and 5500 RPM at WOT, with the spark timing set to KLSA (and most likely, at a λ of 1). Test reproducibility was given as 0.6 bar of IMEP. However, no information on the boost level, engine BSFC, or efficiency is provided with only the IMEP documented at the different fuels tested.

Figure 4-5 shows the results of the tests, and the authors (Ref. 31) found a strong effect of RON on the IMEP. However, no clear impact of MON was isolated, and the impacts were both positive and negative, but relatively small. As with the turbocharged PFI engines, the greatest impact was found at the 3500 RPM level. The trend lines plotted by us in Figure 4-5 show that IMEP increases by 2 bar for every 3 point increase in octane number at 2000 and 3500 RPM, higher than the 2 bar BMEP increase for every 4 octane numbers registered by the Toyota lean boosted PFI engine described in Section 4.2.

Data on the effect of the cooling power shows that the IMEP increase associated with an E30 blend relative to an ETBE blend with the same RON value of 100.5 was about 2.5 bar. The difference in cooling power (defined as latent heat/stoichiometric air-fuel ratio) was about 17 kJ/kg of air, indicating that a cooling power of about 6.85 kJ/kg allows a 1 bar increase in IMEP. This level of increase is equivalent to the level from a 1.5 increase in RON suggesting that a 4.5 kJ/kg cooling power increase is equivalent to a unit increase in RON. Reference 31 provides a wide range of 2 to 8 kJ/kg cooling power as being equivalent to one octane number increase. Since the difference in cooling power between E100 and gasoline is about 77 kJ/kg, this suggests that the cooling power of E100 is equivalent to an additional 17 $\pm$ 7 octane numbers in RON in a turbocharged DI engine.

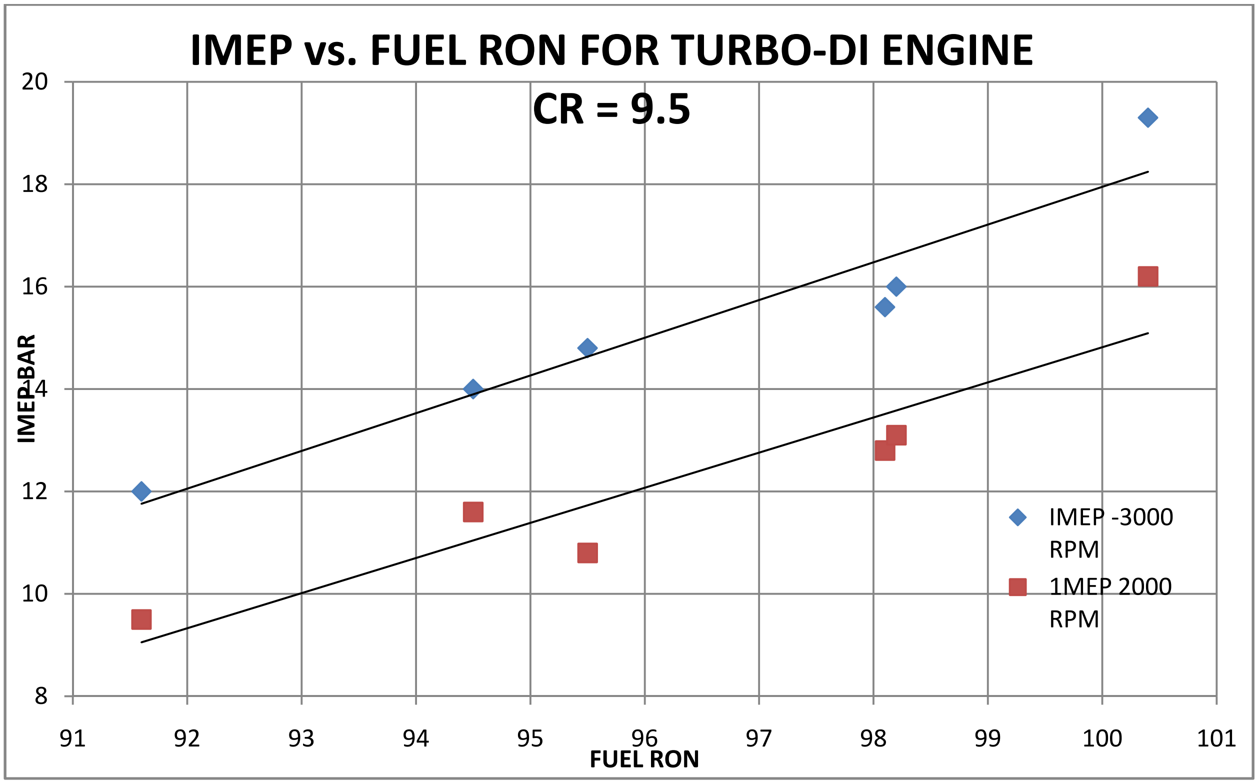

Figure 4-5

A more detailed analysis of the individual effects of octane number and cooling power was conducted by a consortium the included Ford, AVL, Deere, and BP (Stein, et al., Ref. 32). The tests were conducted on a single cylinder engine with the CR varied between 10, 12, and 14. A wide variety of ethanol blend concentrations with different base gasoline RON levels were tested at 1500 and 2500 RPM with some additional RPM tests at a CR of 12. The intake system was set up to allow a pre-heated intake to maintain constant intake mixture temperature with all fuels, as well as PFI and DI systems. Turbocharging was simulated by an intake pressure booster and an exhaust orifice to simulate turbine back pressure, and exhaust temperature limits were observed to simulate turbine inlet limits. A peak pressure limit of 160 bar was also used to simulate production engine constraints.

Measurements were carried out with an 88 RON base gasoline blend stock with E10, E20, E30, E50 and E75 blends. Spark retard was used to allow more boost and increase the maximum IMEP. Figure 4-6 shows the maximum IMEP with the base gasoline at 88 RON (E0) and the E50 blend with a 105 RON rating introduced with upstream fuel injection (UFI)before the mixture pre-heater, with the mixture at the controlled temperature of 52°C, and also the E50 blend directly injected into the cylinder. The difference in the performance of the engine in the first two cases is due to the octane number difference between the fuels, while the performance difference between UFI and DI cases for E50 is due to the cooling effect.

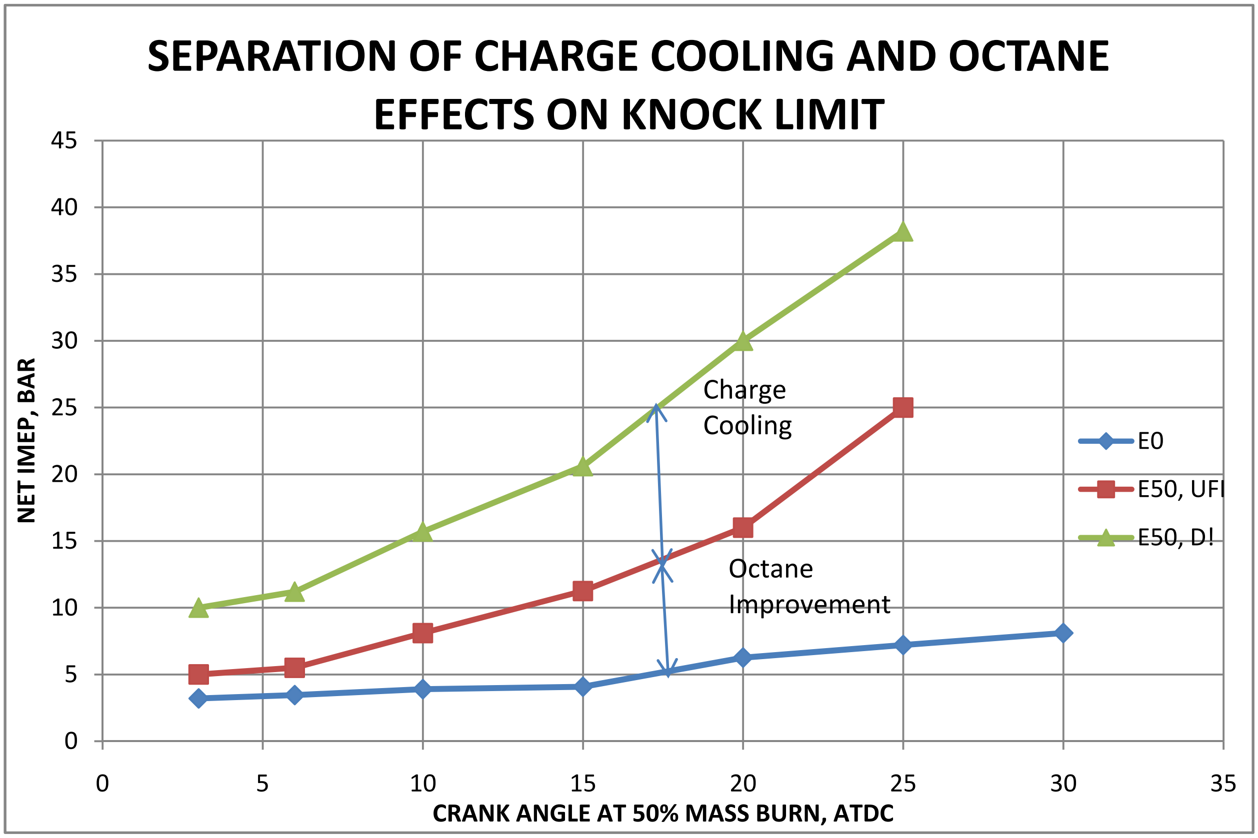

Figure 4-6

Reference 32 finds that for a given gasoline blend stock, increasing ethanol content significantly increases NIMEP as well as thermal efficiency especially at retarded combustion phasing.The magnitude of the improvement of NIMEP with E50 is striking as shown in Figure 4-6. E0 has a maximum IMEP of about 8 bar, while for E50 in a DI engine it is 38 bar. The authors of Ref. 32 attribute the improved performance at retarded timing to the high sensitivity of auto-ignition kinetics to temperature for ethanol which has a Sensitivity of 18 octane numbers. The authors also find that both the octane number effect and charge cooling effect of high ethanol concentration blends are very significant for DI engines and are of about equal magnitude. In the figure above, the difference in RON between the base gasoline and E50 blend is 17 octane numbers. If the cooling effect is of the same size or about 17 numbers for E50, the cooling effect is almost twice as large as estimated in Ref. 31.

Another set of tests with a turbocharged DI engine were conducted by Orbital (Brewster, Ref. 33) using a unique air assisted injector. Tests were conducted on a 2L 4 cylinder engine with a bore size of 86mm and a CR of 10.4. Test conditions were at 2000 RPM WOT with engine airflow controlled by the turbocharger waste-gate. Boost pressures were relatively low at 36 to 40kPa. The study found that E100 enabled 5 to 10 degrees more spark advance and provided about 12 to 13 more N-m of torque relative to a 98 RON gasoline which is about a 5% torque increase. These numbers are very modest compared to the results in Ref. 32, and it is not clear why timing retard and increased boost caused a net reduction in torque with E100 in this study. It may be that the turbocharger was not fully effective in using the enthalpy of the exhaust gas to increase boost at an engine RPM of 2000, and this may be a real limitation that is not accounted for in the previous study. At other RPM points, the study used the performance with the base gasoline fuel as a benchmark and optimized spark and boost calibration with E100 to maximize energy economy at the same torque. The study found that brake thermal efficiency is improved between 7% and 13%, increasing with RPM. With E100, the same torque level is achieved with lower boost and with lower exhaust temperatures. The author also pointed to some combustion instability issues at high boost conditions related to the slower burn rate of ethanol.

In an analysis of ethanol hot spot pre-ignition, Hamilton, et al., (Ref.38) examined the tendency of E85 and E100 fuels to pre-ignite using a CFR engine with a diesel glow plug to provide a temperature controlled hot spot. Tests were conducted at 900 RPM using CR values of 12. 13.5 and 15. The authors concluded that pre-ignition occurs at 620° to 680°C at lambda values of 0.85 to 1.05, and higher CR leads to lower pre-ignition temperatures. They found that E100 pre-ignites at 10° to 20° C lower glow plug temperatures than E85, and very high knock intensity occurs when per-ignition starts at 14 to 8 CAD before TDC. The range of temperatures for pre-ignition are much lower for E100 and E85 than for hydrocarbon gasolines reported in Ref.37 (where the range was 800° to 900° C).

MIT's Laboratory for Energy and the Environment has been developing the concept of direct injection of ethanol coupled with port injection of gasoline to tailor the net octane number benefit to engine load and speed. The simulation and experimental work have been reported in references 34, 35, and 13 (Bromberg, Cohn and Heywood, 2006; Bromberg and Cohn, 2007; Bromberg and Cohn, 2008). The modeling in Ref. 35 suggested that direct injection of E100 would be effective in preventing knock. For a turbocharged DI engine with a CR of 10, no ethanol was required until the intake manifold pressure rose beyond 0.8bar, but about 18% of the energy from E100 was required at 1 bar and 80% at 3 bar. At a CR of 12, the ethanol fraction as a function of manifold pressure was parallel to the curve at 10CR but shifted 0.2 bar lower intake manifold pressure. Bromberg, et al., (Ref. 13) conclude that this strategy of injecting ethanol directly as needed results in a very high "effective" octane rating. When 10% of the energy is provided by ethanol, (i.e., 14% by volume) the effective octane number of the E100 was found to be 175.Since normal driving is mostly at light load, high CR engines can be used with gasoline to minimize consumption of ethanol. The authors modeled vehicular use of ethanol on the U.S. EPA Federal Test Procedure (FTP) and Highway driving cycles, and found that less than 2% of fuel volume was ethanol on these cycles using a 10 CR PFI+DI engine operating with a maximum boost of 2.5 bar.

In a subsequent paper (Stein, House, and Leone, Ref. 36), Ford reported on an engine developed along the lines modeled by Bromberg and Cohn. This engine was a prototype 3.5L "Ecoboost" V-6 with a 9.8 CR; itwas a turbocharged engine where the gasoline was introduced with a PFI system and the ethanol through a DI system. This particular engine was constrained to a peak pressure of 100 bar. A BMEP sweep showed that 98 RON gasoline was knock limited above 8 bar BMEP and spark timing had to be retarded by 15° at the maximum BMEP of about 18 bar, when the turbine inlet temperature limit of 950°C was reached. With E85, no timing retard was required to about 18 bar BMEP and a peak BMEP of 21 bar was realized, being limited by engine peak pressure limitations. When employing a gasoline + ethanol fueling strategy, it was found that to avoid knock at MBT spark timing, the mass fraction of E85 at 10 bar BMEP was 40% and rose to 50% at 15 bar after which spark timing had to be retarded to meet engine peak pressure limits. The research also found that spark retard by about 5° was optimal to reduce the combined BSFC of gasoline + ethanol to the minimum. Using this fueling strategy and measured values of ethanol consumption, Ford confirmed that E85 consumption for a Ford F-150 truck would be less about 1% of total fuel mass on the U.S. EPA FTP and Highway test cycles at the truck's test weight but would increase to 19% and 30%, respectively, at the truck's maximum weight with full payload.

5. FINDINGS AND CONCLUSIONS

5.1 MAJOR FINDINGS

The effect of increasing fuel octane number on SI engines and vehicle energy efficiency cannot be quantified in a simple way but requires a complex, nuanced answer. The following findings from the literature review and analysis are the basis for the conclusions in Section 5.3 of this report.

A. Relationship between Engine Efficiency and Fuel Octane Number

1) The thermal efficiency of naturally aspirated engines primarily depends on the engine's CR, but is also adversely affected by increasing surface-to-volume ratio of the combustion chamber which depends directly on the CR and inversely on the cylinder displacement (and typically, the bore size). As a result, maximum indicated thermal efficiency is almost constant beyond CR values of 14 for small bore engines (<70mm) and almost constant beyond a CR of 16 for bore size over 85mm. The marginal thermal efficiency benefit of increasing CR declines continuously with increasing CR.
2) The engine octane number requirement (ONR) is the lowest fuel octane number that results in borderline knock at wide open throttle when the spark timing is set to MBT. The engine octane number requirement is primarily a function of the CR and to a slightly lesser extent, operating conditions of speed and air-fuel ratio. It is also a function of engine design variables such as the shape of the combustion chamber, efficacy of cooling, and bore diameter[7]. The highest octane number requirement occurs at low RPM (<1500) and wide open throttle (WOT) for naturally aspirated engines.

B. Naturally Aspirated Engines

1) The typical ONR for a port fuel injected naturally aspirated engine with a CR of 10 and a bore diameter of about 85mmoperating atλ= 1, 1500 RPM is 100 to 102 RON at WOT. Smaller bore engines have a lower octane requirement while larger bore engines have a higher octane requirement. The relationship between ONR and bore diameter is

[7] The variables listed are illustrative but this is not a comprehensive list.

not well quantified in the literature but may be around 2 to 3 octane numbers per 10mm. based on the data collected for this study. Hence, a small bore engine (70mm) at the same conditions can have an ONR of 96 to 99 while a large bore engine (100mm) can have an ONR of 103 to 106.

2) The ONR increases with engine CR by about 4 to 5 octane numbers per unit CR increase. (It is not clear if this relationship holds at higher CR values over 13.) The ONR decreases with RPM and the ONR dependence appears to be approximately linear with time per revolution (the inverse of RPM) to about 40 ms (1500 RPM), but not all studies confirm this. The ONR declines by approximately 2 octane numbers per 10ms decline in the time per revolution. Hence, the ONR declines by about 4 octane numbers between 1500 RPM and 3000 RPM. (40ms to 20ms per revolution).
3) The engine actual octane number limit (ONL) is sensitive to spark timing advance and air-fuel ratio and these are the two primary control levers used to prevent knock in production naturally aspirated engines. The ONL is reduced with spark retard from MBT values and this relationship is approximately linear, with the ONL being reduced by 1 octane point for every 1.5 $\pm$ 0.5 degrees spark retard. The relationship becomes more non-linear at retarded timing very close to MBT and at timing advance beyond MBT. The ONL at WOT is highest at λ of 0.95 to 1 and every 0.1λ richer or leaner than this level leads to about a 3 point reduction in ONL.
4) The influence of spark timing on torque and engine efficiency is very non linear with the first 5 CAD retard from MBT reducing torque and relative efficiency by only about 1.3%, while a 10 CAD retard causes a 5.1% reduction. Hence, spark retard by less than 10° from MBT can reduce the engine ONL significantly while having modest effects on available torque and thermal efficiency.
5) With direct injection of the fuel into the cylinder, the latent heat of evaporation of the fuel cools the air-fuel mixture. The cooling power of gasoline for a stoichiometric mixture of air and fuel is about 24 kJ/kg air. The reduced temperature reduces the octane number requirement by one octane point for about 4.5 to 6 kJ/kg of cooling power. Hence, direct injection of gasoline reduces the ONR by about 4 to 5 octane numbers.

C. Turbocharged Engines

1) The relationship between engine CR, spark timing and engine ONL is more complex in a turbocharged engine since the reciprocating engine and turbocharger act as system. If the engine is not boost limited, the optimal combustion phasing for best torque is delayed relative to that for a naturally aspirated engine since there is a compromise between enthalpy flow to the turbine and the best torque achieved by the engine. At high RPM, the boost, air-fuel ratio, and spark timing at WOT have to be limited not only to avoid knock but also to limit turbine inlet temperature(which limits the employment of spark retard) and engine peak pressure, so that fuel octane number effects are very varied across different operating conditions.
2) The available data show that increasing fuel RON increases mid-range torque (~ 3000 to 4000 RPM) by 2% to 3% per octane number increase in RON for a turbocharged PFI engine, while increasing engine thermal efficiency by about 0.7% per octane point increase. Low RPM and high RPM torque are usually limited by other factors as well, so that the effect of octane number increase at these RPM levels may be engine design specific.
3) Increasing octane number can have a very significant effect on mid-range torque for turbocharged DI engines that are not boost limited. Limited data on DI engines shows that an 8 increase in fuel RON can increase 3000 RPM torque by about 50%, or about 5% per octane number. Torque benefits at high and low RPM may be much smaller as they are subject to the same constraints as for PFI engines.
4) The use of high ethanol concentration blends (E30 and higher)may be particularly beneficial to turbocharged DI engines for two reasons. Research shows that with high volume ethanol blends, the combination of high boost and retarded timing allows knock-free operation at very high BMEP levels. Recent experimental work has shown the capability of an E50 blend to allow engine operation at 38 bar to 40 bar IMEP, which is about twice the IMEP level of current gasoline turbocharged DI engines. These high IMEP values will not likely be reproduced in production engines, as ethanol has other limitations such as reduced hot spot pre-ignition temperature relative to hydrocarbon gasoline.

D. Special conditions

1) The use of lean burn offers a substantial fuel economy opportunity especially at part load conditions, but is not used in the U.S. due to difficulties in meeting emission standards. Under lean conditions with a homogeneous mixture, lower RON fuels have been found to give better engine efficiency as the lower octane number leads to a combustion mode described as spark initiated compression ignition of end gases that are not fully combusted with high octane number fuels. If lean combustion becomes viable, lower RON fuels may be more desirable. However, if the limiting condition for engine ONR is at full load where stoichiometric or rich operation is desirable for maximum power, the fuel octane number will have to be tailored to this condition in actual practice.

E. Fuel Issues

On the fuels side, the review and analysis provide the following conclusions:

1) The existing test procedure to determine fuel octane number (both Research and Motor) uses test conditions outside the typical operating range of modern engines.
2) At low RPM conditions (<1500 RPM) and WOT, the effective octane number of a retail gasoline (whose MON is 8 to 10 numbers lower than the RON) is actually higher than RON. The fuel Octane Index, which is a weighted sum of RON and MON that represents the performance of the fuel in the engine, uses a negative weight for MON so that as MON decreases, the fuel has a higher Octane Index. (See conclusion 5 below.) The MON weighting factor, K, is estimated to be about -0.2 to -0.4 for modern engines at low RPM.
3) Available data suggest that the greater the ONR of the engine, the more negative the K value becomes. Hence, high CR turbocharged engines will have a more negative K value on average than a moderate CR naturally aspirated engine. This implies that fuels with high RON and low MON are well suited to high ONR engines, but the data show a lot of scatter in the K values for individual engines.
4) With increasing RPM, K appears to increase linearly with time per revolution (or 1/RPM). The increase implies that MON becomes more important at higher RPM. Higher

MON is also desirable for suppressing other unusual combustion modes. In particular, hot spot pre-ignition temperature is better correlated to MON than to RON.

5) Olefin and aromatic compounds have higher octane Sensitivity than paraffin compounds because their combustion kinetics are slower at lower end gas temperatures (similar to the end gas temperatures in modern engines) and faster at high temperatures (similar to the end gas temperature on the MON test) relative to paraffin compounds. Olefin compounds have the highest burning velocity, and high olefin content fuels can increase engine relative efficiency by 1% to 2%.
6) The latent heat of vaporization is an important factor that improves the knock resistance of a fuel in a DI engine. The cooling power, as measured by the latent heat of vaporization divided by the stoichiometric air-fuel ratio, is an indicator of the benefit. In this context, the cooling power of ethanol is about 4 times as high as gasoline.
7) Ethanol also offers high RON (111) and high Sensitivity (17) so that its use in turbocharged DI engines (where K is most negative) allows very large increases in engine output and efficiency. Experimental results suggest that the benefit in knock resistance from the evaporative cooling effect is as large as the benefit from the higher RON for high concentration ethanol blends (E30 and higher). However, ethanol also has much lower hot spot pre-ignition temperature, consistent with its lower MON rating, which will limit the benefit of ethanol use in production engines.

5.2 STATUS OF THE CURRENT U.S. NEW VEHICLE FLEET

The benefits of increasing octane number are very dependent on the starting point of CR, efficiency, and octane number. In the current US market, manufacturers designate vehicle gasoline requirements as “regular,” “premium recommended,” and “premium required,” but there are no regulatory requirements that must be met to have a premium fuel designations. Regular gasoline has a RON of about 90 to 91 and Sensitivity (RON – MON) of about 8 to 10. Premium gasoline has a RON of 95 to 96 and a similar Sensitivity. All of the vehicles where premium gasoline is recommended or required are high performance sports cars and luxury vehicles, with the exception of some mass market DI vehicles designated for premium gasoline by VW. The 2009 vehicle data was analyzed to derive the CR for PFI, DI, and Turbocharged engines by fuel octane number (regular or premium) as H-D Systems had access to a comprehensive database on the technical specifications of all vehicles sold in that model year.

A regression analysis was conducted with CR as the dependent variable and bore size, turbocharging, and the use of DI as independent variables. Unfortunately, there were only 2 DI engines with regular gasoline designation and one was turbocharged, so that no statistically significant coefficient was obtained for the regression of data on cars designated for regular gasoline. In the case of cars designated for premium gasoline (either recommended or required), the spread of bore sizes was quite small and no significant coefficient for bore size was obtained. The resulting regressions (t-statistic in parentheses) are shown below:

CR	=	10.88	– 0.01(Bore, mm)	-0.72 (Turbo) for regular gasoline
		(42.4)	(-3.36)	(-10.8)
CR	=	10.96	- 1.89 (Turbo)	+ 0.99 (DI) for premium gasoline
		(20.0)	(22.5)	(11.2)

It should be noted that most of the bore sizes are concentrated in the 85 to 100mm range in the U.S. so that the regression is not capturing the effect of small bore sizes in the 65 to 75 mm range. An evaluation of the CR levels at a constant bore size of 86 mm shows the following levels of CR by engine type and fuel requirement:

GRADE	NA –PFI	NA-DI	TURBO-PFI	TURBO- DI
REGULAR	10.1	-	9.35	-
PREMIUM	11.0	12.0	9.2	9.0

The actual data in the table are in excellent agreement with the findings. The 4 to 5 RON advantage of premium fuel over regular gasoline results in average CR increasing by 0.9 in production vehicles that require premium, relative to the CR for production vehicles that allow the use of regular. The “cooling effect” advantage of DI results in exactly 1 unit increase in CR. The reason that the CR for turbocharged engines for regular and premium gasoline are nearly identical is due to high performance engines which use higher levels of boost being certified for premium.

The regression shows that 10.1 CR engines are operating on 91 RON gasoline, which may appear to be inconsistent with the finding that the ONR of engines at that CR was found to be 100 to 102 at low RPM. Manufacturers acknowledge that current production engines use a combination of fuel enrichment and spark retard to avoid knock at full load. Our estimate is that the 9 to 11 octane number gap between engine ONR and gasoline RON is managed as follows:

Improvement in fuel octane index due to negative K:	2 to 3
Spark retard by 6° to 8° from MBT:	3 to 4
Fuel enrichment to λ of 0.8 to 0.85:	3 to 4

At higher RPM, the improvement from the K factor goes to zero or is negative, but this is offset by the decline in engine OR. For example, the data shows that increasing the RPM from 1500 to 3000 reduces engine OR by 4 octane numbers, but the K factor increases from -0.25 to +0.1 so that the OI of the fuel decreases by 3 numbers largely offsetting the decrease in OR.

The 2012 baseline data for CR is not expected to be very different from 2009, although there are many more DI models, both naturally aspirated and turbocharged, available for use with regular grade gasoline. In addition, there have been some market introductions of small displacement engines (<1.5L) that may allow better determination of the effect of bore size on CR. In Japan, where small displacement engine are more common, small bore 4 cylinder engines (~70mm bore) with displacement of 1L to 1.3L have a CR of 11.5, while 2L engines (~85mm bore size) have a CR of 10.6 for engines intended for use with regular gasoline (also 91 RON in Japan) according to data provided by Toyota to this project. The 0.9 CR increase should increase the OR by about 4 octane numbers at constant bore size, suggesting that the estimate of 2 to 3 octane point reduction per 10 mm bore size decrease is in the correct range as the 15mm bore size decrease offsets the effect of a 0.9 CR increase.

CR also has increased over time even as the RON and MON of regular gasoline has stayed constant. In the model year 1980 time frame, CR was about 8.5 and in 2010 it was about 10 suggesting an increase of 0.5 CR per decade. Mittal and Heywood (Ref. 6) have documented the reasons behind this increase and improved cooling, elimination of intake air preheat, replacement of carburetors with fuel injection, etc., have contributed. It is not clear if this trend will continue but we anticipate that the average will continue to improve at 0.5 CR per decade, and may improve even more quickly if the use of DI is widespread by 2020. It is also worth noting that Mazda has recently introduced a small bore DI engine with a CR of 14 in Japan that uses regular gasoline with a minimum RON of 89 (no MON limit is specified). Other Mazda engines sold in Europe and the U.S. have CR values of 12 to 13 on regular gasoline. In the US, Mazda offers a 2L engine with a 13 CR for use with regular gasoline. No detailed data on the engine are available yet but Mazda brochures suggest very good exhaust scavenging to reduce residual gas and hence, mixture temperature as well as some additional spark retard has been used to permit the high CR. The 2L engine is rated at 154 HP while the previous Mazda 2L engine with a10 CR had a rating of 148 HP. Hence Mazda has managed to increase CR by 3 with no sacrifice in the power rating. The potential for new technology to increase CR with no additional fuel octane number requirement is demonstrated by this technology. Another example of technology that can reduce the ONR is the use of cooled EGR at full load for turbocharged engines. Public statements by several auto-manufacturers suggest that cooled EGR at full load will be used in production engines in the near future.

5.3 CONCLUSIONS

One of the objectives of the project was to assess the impact of increasing fuel octane number on engine and vehicle efficiency. The conclusions for a **4 to 5 RON** increase are as follows:

1) In port fuel injected, naturally aspirated engines with mid-size (~85mm) and larger bore sizes (that account for most of the U.S. market today) and a CR of 10, this octane number increase will facilitate a 1 point increase in CR, which can provide a 2% relative improvement in engine efficiency. For small bore sizes (~70mm), the increase in CR will be similar but the improvement in engine efficiency will be smaller at 1.3% to 1.4%.
2) For midsize and large bore naturally aspirated DI engines that have a CR of 11, this octane number increase will facilitate a 1 point increase in CR and allow engine efficiency to increase by 1.5% to 1.6%. Small bore engines will have an improvement of 0.8% to 0.85%.
3) Vehicle efficiency gains for vehicles with naturally aspirated engines can be somewhat larger than engine efficiency gains due to increased engine torque when CR is increased, permitting engine down-speeding. This can provide a 1% to 2% fuel economy gain over and above engine efficiency gains.

4) In turbocharged engines, the effect of this improved octane number could be through the increase in engine peak torque if the engine is not boost limited. In PFI engines, the 4 to 5 RON increase can provide 10% to 15%more torque. In a DI engine, this increase in octane number can translate into a 20% to 25% increase in torque. This torque increase is for operation at 2000 to 4000 RPM; at engine speeds below 1500 RPM and above 4000 RPM, other factors control available torque increases so that no general conclusion is possible for all speeds.
5) The torque increases with turbocharged engines could allow significant downsizing and down-speeding of the engine but the extent of downsizing and down-speeding is dependent on engine low RPM performance. As an example of the size of the potential benefit, a 5% to 7% fuel economy improvement is possible if the engine can be downsized by 20% to 25% to keep absolute mid-range torque constant.
6) If the octane number increase were to be derived by addition of ethanol to gasoline, larger torque increases may be possible in turbocharged DI engines due to the high latent heat of vaporization of ethanol. An E30 blend with 91 RON gasoline blend-stock could potentially improve torque by 50% but this level of increase is not proven in any vehicle application, and could be limited by ethanol's lower resistance to hot spot pre-ignition.
7) Developments in engine technology will allow CR to increase in the future with no change to fuel RON and MON. The increased use of DI technology will allow a CR increase of 1 unit. Engine downsizing, with resulting smaller bore sizes, as well as the use of cooled EGR and improved exhaust scavenging, may allow continuing increases in CR over the next decade with unchanged fuel ON.

6. DATA GAPS AND RESEARCH RECOMMENDATIONS

The detailed literature review reveals a limited selection of studies in the public domain on the relationship between fuel octane number and engine/vehicle efficiency, as only 45 papers relevant to this topic were found after a reasonably exhaustive search. Of these 45, many were only tangentially related to the subject of fuel octane number and engine efficiency, but helped fill some gaps in the complex connections between the two variables. In addition, many of the directly relevant papers were from a small number of organizations that reported on different fuels and test modes for the same engine. For example, there are nine papers from Toyota, and the papers have used the same set of research engines for their studies.

Two types of data gaps have been identified. The first type is where the data exists but is not public. For example, the data set is on actual engine efficiency and octane number requirement is very limited across engine sizes and across the RPM range. Most papers have used engines in the 85 $\pm$ 5 mm bore size range, and most of the data are for engine speeds of 1200 to 2000 RPM. Other key variables with limited data include the effect of spark timing as well as the effect of air-fuel ratio on fuel octane number requirements across a range of engine bore sizes and RPM. These data clearly exist for a wide variety of production engines at each auto-manufacturer, and creating a database of these variables is not difficult if manufacturers participate in sharing the data (the origin of the data can be obscured to maintain confidentiality). Such a database could pave the way for developing more robust relationships between efficiency and fuel octane number for different bore sizes and operating conditions.

The second type of gap is where the data available are too limited to provide specific conclusions. The data on turbocharged DI engines (especially those with small displacement), and their response to fuel octane number and heat of vaporization in terms of efficiency and torque improvement across a range of engine speeds (especially at low RPM) are very thin and usually derived from experimental engines, where the interaction of the reciprocating unit with the turbocharger is not measured but simulated. Octane number improvements and improvements from ethanol blends appear to hold significant promise, but the available data are inadequate to estimate the benefits in a real vehicle application.

Newer technologies are also not represented in the database. The latest DI engines in Europe have sequential turbocharging with 2 stage cooling; experiments have been conducted with cooled EGR at all loads and speeds, but little information exists on the interaction with fuel octane number.

The testing of turbocharged DI engines of different sizes, and those equipped with newer technology such as sequential turbocharging, as well cooled EGR, can permit a look forward into the future interaction of fuel octane number and engine and vehicle efficiency. CRC sponsored testing research should focus on these newest technologies so that findings remain useful for the future.

LIST OF ACRONYMS

BP	- British Petroleum
BMEP	- Brake Mean Effective Pressure
BSFC	- Brake Specific Fuel Consumption
CAD	- Crank Angle Degrees
CFR	- Co-operative Fuels Research
CR	- Compression Ratio
CRC	- Coordinating Research Council
DI	- Direct Injection
Dyno	- Dynamometer
EGR	- Exhaust Gas Recirculation
ETBE	- Ethyl tertiary butyl ether
E*xx*	- Ethanol gasoline blend with the volumetric content of ethanol as a percent, *xx*
FC	- Fuel Consumption
HCCI	- Homogeneous Charge Compression Ignition
IMEP	- Indicated Mean Effective Pressure
ISFC	- Indicated Specific Fuel Consumption
K	- Weighting factor of MON in the Octane Index
KLSA	- Knock Limited Spark Advance
kPa	- kilo Pascal
LHV	- Latent Heat of Vaporization
MAP	- Manifold Air Pressure
MBT	- Maximum Brake Torque
MIT	- Massachusetts Institute of Technology
MON	- Motor Octane Number
MPa	- Mega Pascal
NA	- Naturally Aspirated
OI	- Octane Index of fuel = (1- K)*RON + K*MON
ONL	-Octane Number Limit

ONR	- Octane Number Requirement of engine
PFI	- Port Fuel Injection
PRF	- Primary Reference fuel
RON	- Research Octane number
RPM	- Revolutions Per Minute
S	- Sensitivity of the fuel defined as RON - MON
SAE	- Society of Automotive Engineers
SI	- Spark ignition
SIDI	- Spark Ignition Direct Injection
TDC	- Top Dead Center (piston position)

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