

Fuel Economy Potential of Variable Compression Ratio for Light Duty Vehicles

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ABSTRACT

Increasing compression ratio (CR) is one of the most fundamental ways to improve engine efficiency, but the CR of practical spark ignition engines is limited by knock and spark retard at high loads. A variable CR mechanism could improve efficiency by using higher CR at low loads, and lower CR (with less spark retard) at high loads.

This paper quantifies the potential efficiency benefits of applying variable CR to a modern downsized, boosted gasoline engine. Load sweeps were conducted experimentally on a multi-cylinder gasoline turbocharged direct injection (GTDI) engine at several CRs.

Experimental results were compared to efficiency versus CR correlations from the literature and were used to estimate the fuel economy benefits of 2-step and continuously variable CR concepts on several engine/vehicle combinations, for various drive cycles.

For vehicles with automatic transmissions, the fuel economy benefit of 2-step variable CR was estimated to be 2.5-3.1% on the EPA Metro-Highway cycle, and 0.8-1.2% on the US06 cycle relative to a fixed 10:1 CR baseline. The benefit of continuously variable CR for the same vehicles was estimated to be 2.7-3.3% EPA Metro-Highway, and 1.7-2.1% US06. The effects of CR on exhaust gas temperature, peak cylinder pressure, burn rate, low-speed torque, and high-speed power were also examined.

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INTRODUCTION

Many mechanisms have been proposed to change the CR of an operating engine. A review of various concepts was published by Asthana et. al. [1]. Some proposed mechanisms allow the CR to be varied in a continuous fashion between the maximum and minimum ratio. Other systems are limited to two fixed CRs with the ability to switch as the operating conditions change.

Automotive implementation of variable CR is challenging due to mechanism durability, cost, size, weight, NVH, friction, and other practical issues. However, Nissan has announced plans to implement variable CR on an upcoming production engine [2].

This study evaluated the fuel economy potential of 2-step and continuously variable CR concepts relative to a conventional fixed CR GTDI engine. The results presented here were obtained using published correlations for efficiency versus CR and efficiency versus crank angle of 50% mass fraction burned (CA50), along with experimentally determined values for knock-limited CA50 as a function of speed, load and CR. By combining the experimentally derived CA50 values with the two efficiency correlations, the effect

of variable CR was estimated for a wide range of operating conditions. For a 2-step variable CR system, an iterative process was used to select the two CRs that provided the best EPA Metro-Highway (M/H) drive cycle fuel economy improvement. This study did not evaluate possible secondary effects of a variable CR mechanism such as friction, parasitic losses, crevice losses, and compromised combustion chamber shape (heat transfer losses, burn rate, air-fuel mixing, etc.)

TESTING METHODOLOGY

Experimental results were obtained by conducting load sweeps on a conventional fixed CR 60° V6 GTDI engine (see Table 1 for specifications). The baseline CR was 10:1. Experiments were conducted at 8:1 and 9:1 by means of replacing the connecting rods with shorter rods and using the baseline pistons. Higher CRs, 11.9:1 and 13:1, were achieved by using the production connecting rods with prototype pistons (Figure 1). During each CR change, the engine was disassembled enough to remove and replace the connecting rods or pistons, keeping the head and block assembly intact. This methodology preserved the baseline mechanical friction as much as possible.

Table 1. Engine information.

Displaced volume	3496 cc
Stroke	86.7 mm
Bore	92.5 mm
Connecting rod length	149.9 mm: 8:1 CR 151.5 mm: 9:1 CR 152.7 mm: 10:1, 11.9:1, 13:1
Compression ratios	8:1, 9:1, 10:1, 11.9:1, 13:1
Valves per cylinder	4
Cam durations	Int: 248°, Exh 263° (at top of constant velocity ramp)
Fuel system	Side direct injection
Fuel	91 RON 10%v ethanol (for cycle fuel economy results)

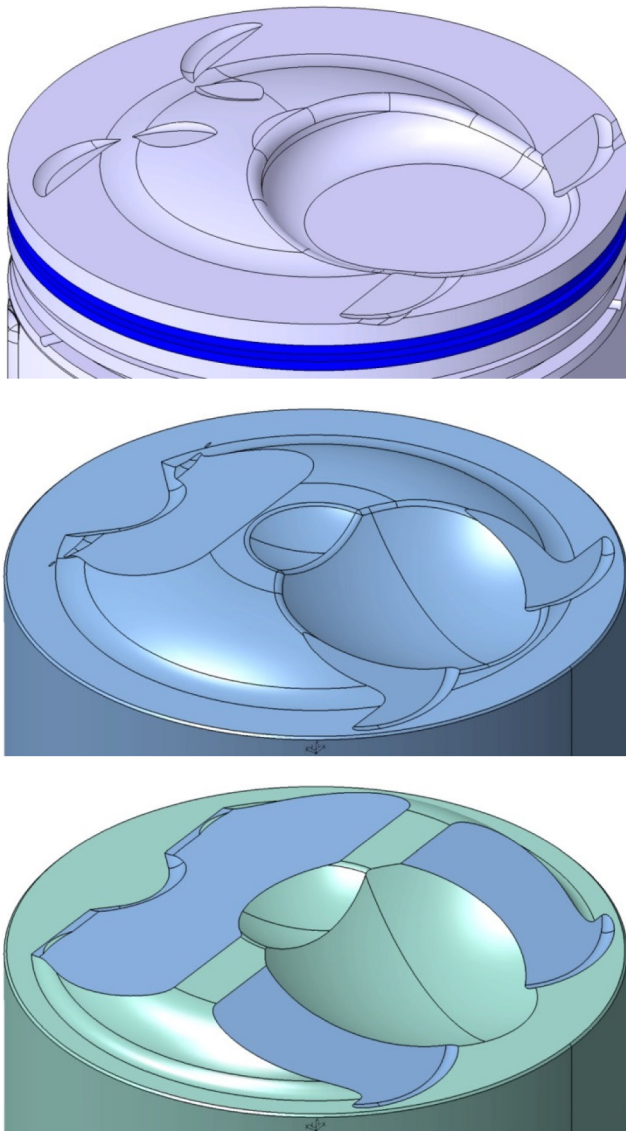


Figure 1. Pistons for 8:1, 9:1, 10:1 (top), 11.9:1 (middle), and 13:1 CR (bottom).

Load sweeps were conducted at 1000, 1500, 2000, and 2500 rpm in order to evaluate the impact of variable CR over the range of engine operation relevant to the fuel economy drive cycles. Load sweeps

were also conducted at 5000 rpm to evaluate the change in performance near peak power. Data points were collected at a given engine speed with gradually increasing brake mean effective pressure (BMEP) as shown in Figure 2 until the BMEP could no longer be increased due to one or more of the constraints listed in Table 2. For the lower engine speeds, the maximum BMEP was also constrained by the available boost pressure supplied by the turbocharger system.

The baseline engine has twin independent variable cam timing. In order to simplify the test plan, a cam timing schedule was selected at 10:1 CR for each RPM – BMEP set point, and that same schedule was applied to all other CRs. Further cam timing optimization may slightly improve the results at the non-baseline CRs.

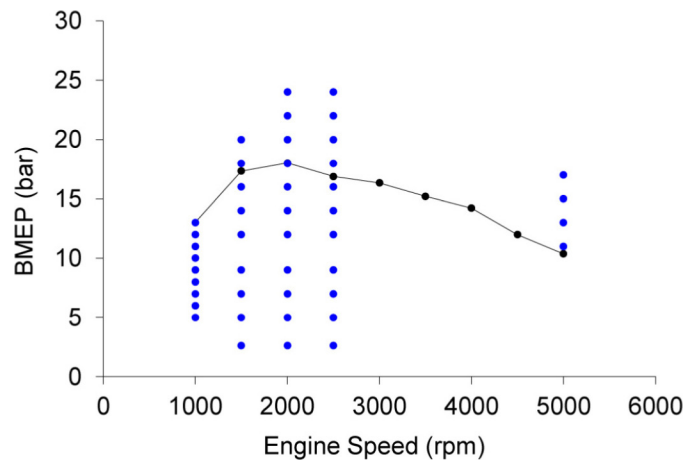


Figure 2. Test points (symbols) and maximum load at stoichiometry (line) for 91 RON E10 at 10:1 CR.

Because significant time and effort was required to change the CR, it was not possible to get back-to-back measurements. In order to maintain a consistent determination of borderline (knock-limited) spark timing throughout the test plan, one experienced technician conducted all of the testing. A combination of cylinder pressure based knock detection and audible knock was used to determine the borderline spark advance.

Table 2. Engine operating conditions and constraints.

Maximum turbine inlet temperature	950 °C
Peak cylinder pressure (mean + 3σ)	<120 bar
Lambda	> 0.75
Crank angle of 50% mass fraction burned	<30° ATC
Maximum turbocharger shaft speed	205 krpm
Pre-ignition	Sweep halted if pre-ignition encountered
Engine coolant outlet temperature	90°C
Target charge air cooler outlet temperature	38°C

Some pre-ignition events were encountered while testing at high load with the low CR configurations. This prevented fully exploring the additional BMEP potential of lower CR engines.

EXPERIMENTAL RESULTS

The experimental portion of this work was conducted in 2012, 2013, and 2015. The same physical engine was used for all the testing, but the long period of time between tests necessitated multiple batches of fuel. Variations in heating value of the fuel were taken into account.

Engine Efficiency (Unconstrained by Knock)

The effect of CR on indicated and brake thermal efficiency (ITE and BTE) was evaluated and compared to previously published correlations. For the thermal efficiency calculations, data collected as part of previous studies with high octane fuel [3,4] was included to increase the BMEP range of data points at MBT and to provide more data to mitigate measurement uncertainty. The following steps were completed as part of the data analysis:

1. ITE and BTE were calculated at each RPM and BMEP set point for each fuel and CR.
2. The efficiency values were adjusted to stoichiometric lambda, nominal BMEP, and MBT combustion phasing.
3. The adjusted values were then regressed at each BMEP and engine speed as a function of CR. A second order regression was used at most speed load points. There was insufficient curvature at 2000 rpm 16 bar BMEP to fit a 2nd order equation, so a linear regression was used.
4. The regressions were then used to compute the percent change in BTE and ITE as a function of CR relative to 10:1.

The adjustments in step two were used to remove small variations in the BMEP and lambda settings present in the experimental data. The MBT phasing adjustment allows the comparisons to extend to BMEP levels that are knock limited at the higher CRs. The MBT phasing adjustment was done using a correlation published by Ayala, Gerty and Heywood [6].

ITE Results

Indicated thermal efficiency is defined as:

$$\text{ITE} = \frac{\text{Indicated work per cycle}}{m_{\text{fuel}} Q_{\text{LHV}}}$$

The ITE is influenced by heat transfer, properties of the working fluid, CR, combustion efficiency, combustion duration, combustion phasing, and to some extent, the timing of exhaust valve opening and intake valve closing. The measured ITE varied significantly as a function of speed and load, but there was considerably less variation when plotted as ITE change versus CR (Figure 3). The data showed less benefit for high CR at 2.62 and 5 bar BMEP (2.62 bar data was not available at 1000 rpm). The data collected at 14 bar BMEP, 1000 rpm was obtained with blow-through and is not included in the ITE plot. 16 bar BMEP was not achievable at 1000 rpm with the experimental hardware used.

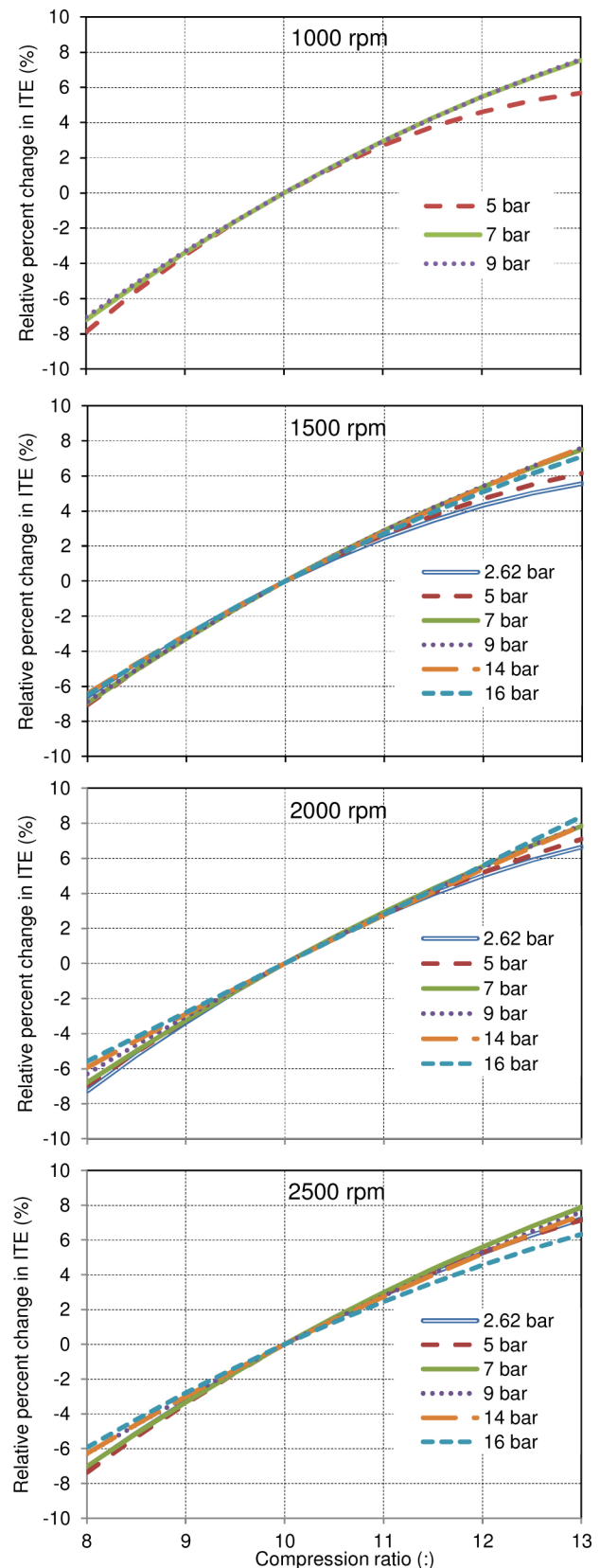


Figure 3. Percent change in ITE relative to 10:1 CR at various engine speeds (adjusted to MBT combustion phasing where knock-limited).

There was almost no variation in relative percent change in ITE versus CR between engine speeds at the same BMEP at 7 bar and above, but there was some variation at 5 and 2.62 bar BMEP. The data for 5 bar BMEP is shown in Figure 4. In general, light load operation at low engine speeds benefited slightly less from increased CR, likely due to heat transfer being a larger fraction of the heat release at light load. The 5th curve shown in Figure 4 is the fuel conversion efficiency of an ideal gas cycle with $\gamma = 1.3$. When expressed as percent change relative to 10:1 CR, the experimental data is very similar to the ideal curve. This indicates that the combustion chamber shape at high CR is not yet having a significant adverse effect on the flame propagation.

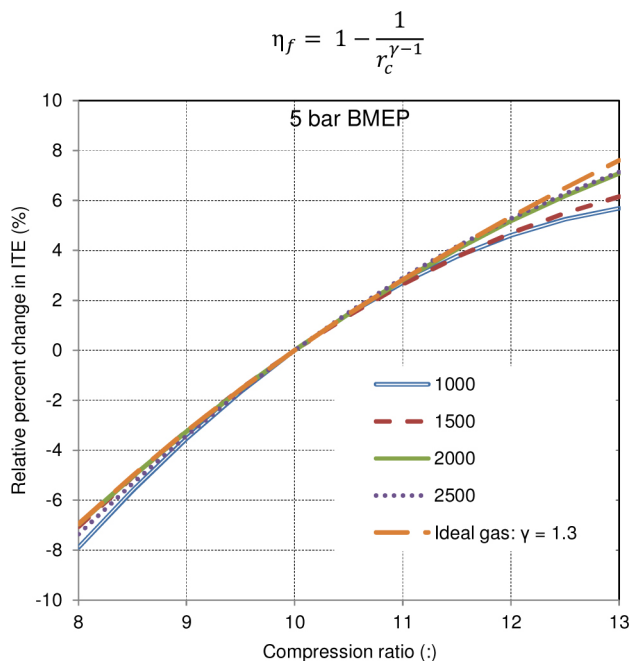


Figure 4. Percent change in ITE relative to 10:1 CR at 5 bar BMEP.

BTE Results

Similar regressions were also done for brake thermal efficiency (BTE) versus CR (Figure 5). Mechanical friction tends to increase as CR is increased due to higher cylinder pressures resulting in higher piston ring loads, higher piston side loads, and higher bearing loads. To evaluate the potential of variable CR for fuel economy improvement, BTE regressions were used in the later analysis.

The measured change in fuel consumption as a function of CR agreed well with published results from Caris and Nelson [5] and from Smith, Heywood, and Cheng [7]. Note: data from this work was one of the data sources used by Smith et al. To avoid reliance on a single data set, the Smith et al. BTE results (an average of seven data sources) were used for fuel economy analysis in this paper.

The “Smith et al.” line in Figure 5 was derived by fitting a 2nd order regression through the “Relative Brake Eff. Change” graph in figure 6 of the reference. The “Caris and Nelson” curves in Figure 5 were generated by digitizing figure 10 of the reference and fitting a 3rd order

polynomial through the points. The regressions were used to calculate the relative percent change in BTE over a range of CRs and the results are presented as percent change in BTE relative to 10:1 CR.

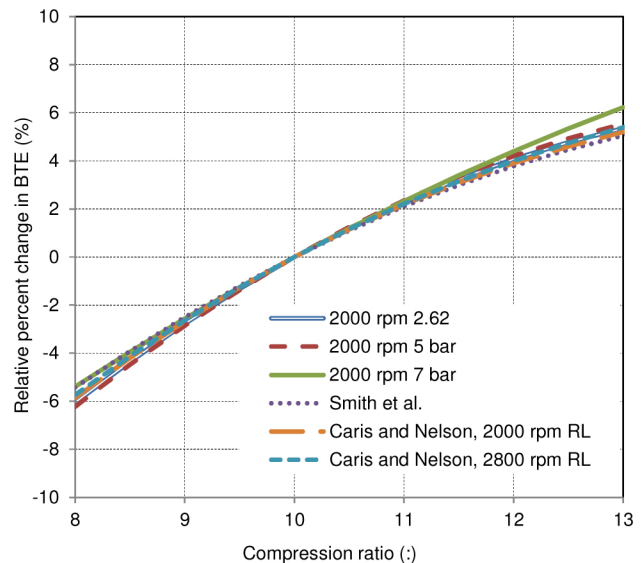


Figure 5. Percent change in BTE relative to 10:1 CR. Compared to figure 6 of Smith et al. [7] and figure 10 of Caris and Nelson [5].

Borderline CA50 as a Function of CR

The primary constraint for high CR spark ignition engines is knock at high loads. To avoid noise (and possible engine damage) from knock, spark timing is retarded to avoid knock at high loads. This retard from MBT results in combustion phasing which is later than optimum for engine efficiency. Combustion phasing is quantified as CA50, the crank angle at which 50% of the fuel mass has burned.

In order to quantify the fuel economy potential of variable CR, the impact of CR on knock-limited CA50 was mapped as a function of engine speed, BMEP, and CR (using 91 RON E10 fuel). Load sweeps were conducted with each CR and the spark timing was adjusted at each load to MBT or to the onset of knock. In an effort to avoid low speed pre-ignition, the amount of spark retard was constrained to CA50 < 30° ATC. Once knock-limited CA50 reached 30° ATC at stoichiometric A/F, enrichment was used to mitigate knock up to the enrichment limit of 0.75 lambda. Sample data at 2000 rpm are shown in Figure 6 (additional speeds are in the appendix).

Figure 7 shows linear regressions of the knock-limited CA50 values as a function of BMEP. The data are the same as Figure 6, except that the points with enrichment, blow through, and MBT combustion phasing have been eliminated. Regressions of knock-limited CA50 versus BMEP were generated for each CR at each engine speed, to be used later in the fuel economy analysis.

As expected, higher CR makes the engine more knock-limited at high loads (Figure 7). This results in later combustion phasing (higher CA50) and reduced efficiency at high loads, which will be quantified in the next section. Note also that the slope of CA50 vs BMEP is steeper at high CR. Changing CR does not result in a simple offset in knock limited CA50.

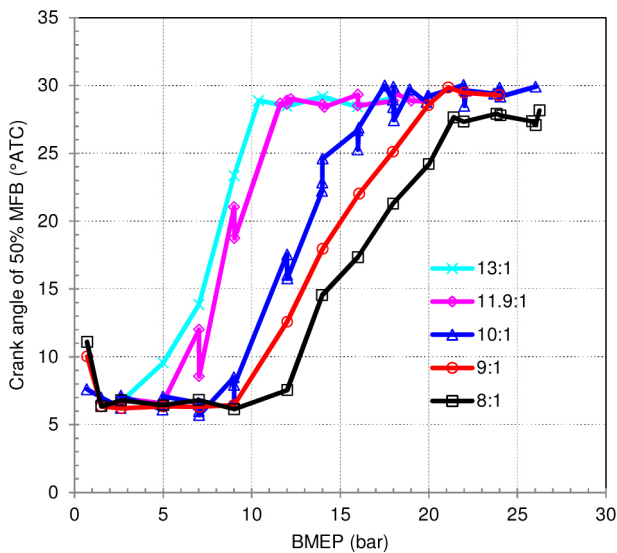


Figure 6. Measured knock-limited CA50 versus BMEP including all points, 2000 rpm, 91 RON fuel.

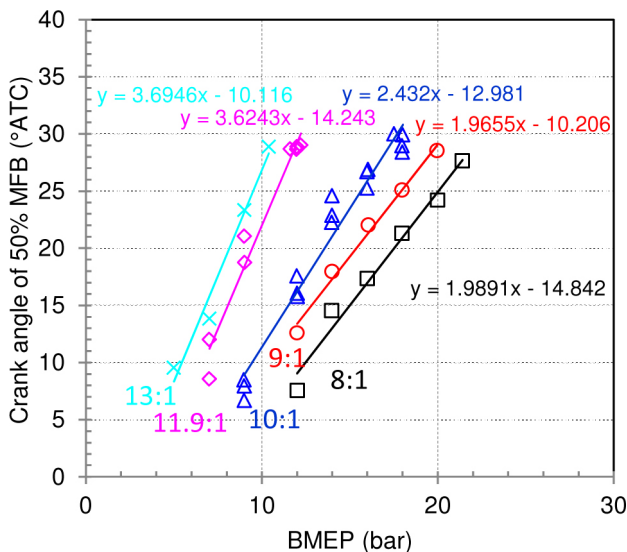


Figure 7. Knock-limited CA50 versus BMEP including only points at stoichiometry without blow through, 2000 rpm, 91 RON fuel.

VEHICLE FUEL ECONOMY ANALYSIS

Method

As described above, CR affects fuel consumption differently at different loads. At low to mid loads, where the engine is not knock-limited, higher CR improves fuel consumption. As the engine becomes knock limited at higher loads, the optimum CR is a balance between the efficiency penalty of retarded combustion phasing and the efficiency benefit of higher CR. At very high loads the optimum CR is lower than the baseline engine CR. At very light loads the charge is fully expanded at the baseline CR resulting in less fuel economy improvement via increased CR. The vehicle fuel economy analysis evaluates these effects separately to identify the optimum CR for each operating condition, and to quantify the net effect on fuel consumption at that operating condition.

The opposing effects on fuel consumption are quantified using the Smith et al. regression for BTE changes versus CR (including the friction effects for a conventional crank-slider mechanism), combined with an analysis of the fuel consumption impact of knock limited combustion phasing at high loads.

The analysis of knock limits at high loads starts with knock-limited CA50 regressions (example in Figure 7 for 2000rpm). Optimum CA50 (i.e. MBT combustion phasing) is assumed to be 7°ATC. The values shown in Figure 8 represent the amount of CA50 retard which is required to avoid knock. Each line represents a different CR. Data was collected at 8,9,10,11.9, and 13:1 and then linearly regressed to reduce scatter (points with blow through or significant enrichment were excluded from the regression). Linear interpolation was used to estimate knock-limited CA50 for other CRs (e.g. 8.5, 9.5, 10.5:1 shown with dotted lines).

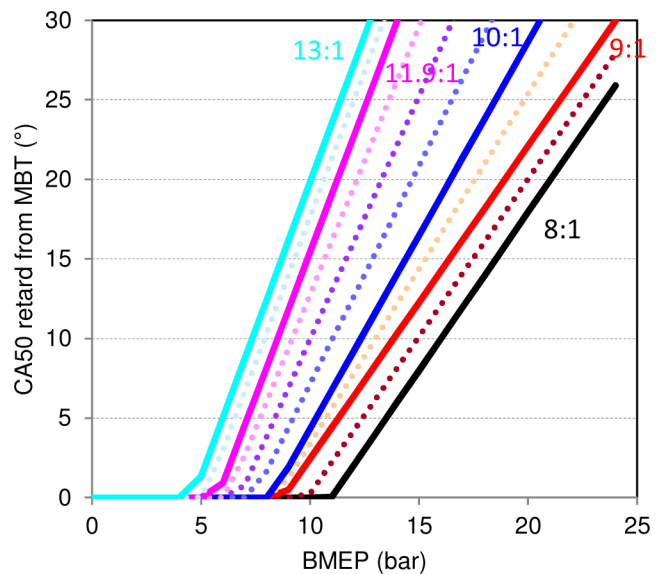


Figure 8. CA50 retard relative to MBT required to avoid knock at 2000 rpm, 91 RON fuel. Solid lines represent regressions of the data. Dotted lines are linear interpolations between CRs.

The fuel consumption impact of combustion phasing (i.e. spark retard due to knock) was quantified using a regression of net specific fuel consumption (NSFC) versus combustion phasing published by Ayala et al. [6]. As shown in the reference, this regression fits experimental data for a wide range of operating conditions. It has also been extensively validated by Ford researchers on several engines. A sample data set at 8:1 CR was acquired confirming agreement on the engine used for this study. The regression from [6] was expressed in terms of relative fuel consumption and has been plotted in Figure 9.

The increase in fuel consumption resulting from operating retarded from MBT was applied to the CA50 results shown in Figure 8 to create Figure 10, which shows fuel consumption relative to 10:1 CR considering only the variation in CA50 due to knock limits at higher loads (not including the efficiency benefit of higher CR).