

Charge Cooling Effects on Knock Limits in SI DI Engines Using Gasoline/Ethanol Blends: Part 2-Effective Octane Numbers

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ABSTRACT

Spark Ignited Direct Injection (SI DI) of fuel extends engine knock limits compared to Port Fuel Injection (PFI) by utilizing the large in-cylinder charge cooling effect due to fuel evaporation. The use of gasoline/ethanol blends in direct injection (DI) is therefore especially advantageous due to the high heat of vaporization of ethanol. In addition to the thermal benefit due to charge cooling, ethanol blends also display superior chemical resistance to autoignition, therefore allowing the further extension of knock limits. Unlike the charge cooling benefit which is realized mostly in SI DI engines, the chemical benefit of ethanol blends exists in Port Fuel Injected (PFI) engines as well. The aim of this study is to separate and quantify the effect of fuel chemistry and charge cooling on knock.

Using a turbocharged SI engine with both PFI and DI, knock limits were measured for both injection types and five gasoline-ethanol blends. A methodology was developed and used to calculate unburned mixture temperatures including charge cooling for each fuel.

The experimental results were used in conjunction with a knock limit predictive model. The Livengood-Wu autoignition integral approach was employed in the model along with the Douad-Eyzat correlation for autoignition time. The octane number (ON) in the pre-exponential term of the correlation was adjusted for each ethanol blend to fit the experimental results. An "Effective Octane Number" (ON_{EFF}) is thus obtained for each blend. The increase in ON_{EFF} is a measure of the chemical benefit from increasing ethanol content. The ON_{EFF} results show diminishing returns in ethanol chemical benefit above about E40 (40% ethanol by volume).

A methodology is also presented to quantify the thermal benefit from charge cooling in SI DI engines as an "Evaporative Octane Number". The autoignition integral is used to predict what octane rating would be needed to prevent knock if there were no charge cooling. The results show that the antiknock thermal benefit increases monotonically with ethanol content from 5 octane numbers for DI gasoline to 18 octane numbers for E85.

This paper is the second part of a two paper series looking at the chemical and charge cooling effects of gasoline ethanol blends on engine knock.

CITATION: Kasseris, E. and Heywood, J., "Charge Cooling Effects on Knock Limits in SI DI Engines Using Gasoline/Ethanol Blends: Part 2-Effective Octane Numbers," *SAE Int. J. Fuels Lubr.* 5(2):2012, doi:10.4271/2012-01-1284.

INTRODUCTION

The automotive industry has been widely adopting the use of direct fuel injection (DI) in production engines over the last few years. DI has a significant advantage over conventional port fuel injection (PFI) in that much more of the heat of vaporization of the fuel can be extracted from the charge, thus cooling it. This substantial charge cooling effect can be used to increase engine efficiency by improving knock limits. Improved knock limits enable higher compression ratios, boost pressures and more optimized spark.

Ethanol-gasoline blends have been advocated in the U.S. as an alternative fuel [1]. Due to the high heat of vaporization

of ethanol, the use of ethanol-gasoline blends is especially effective in preventing knock when used in direct injection engines that can utilize the higher charge cooling effect [2, 3]. This is clear in [Table 1](#), where properties of gasoline (E0) and ethanol (E100) are presented. Ethanol has more than four times the heat of vaporization of gasoline per unit of stoichiometric mixture. As seen in a companion paper [4], the maximum theoretical charge cooling possible based on mixing thermodynamics is about 19° K for gasoline and about 81° K for pure ethanol. Additionally, as seen in [Table 1](#), the purely chemical resistance to knock as indicated by the Research Octane Number (RON) is significantly higher for ethanol compared to gasoline.

Since ethanol is a limited resource, there is an important need to evaluate how the anti-knock resistance benefit due to chemistry and charge cooling varies with ethanol content. It is also important to develop a knock limit predictive model that covers the full range of ethanol blends. In order to achieve these goals, the chemical and charge cooling anti-knock effects need to be separated. The charge cooling effect is much greater in DI engines whereas the chemistry effect is present in both PFI and DI engines. Separating and quantifying the two knock suppressing effects of ethanol is necessary to guide policy making on how to best use ethanol as a fuel. Additionally, a knock limit predictive model can be used to optimize engine concepts that use gasoline/ethanol blends.

Table 1. Ethanol and Gasoline Properties-The notation E'X' symbolizes a gasoline/ethanol blend that is 'X' % ethanol by volume.

	Gasoline (E0)	Ethanol (E100)	Notes
$F/(F+A)_{\text{stoich}}$	0.064	0.1	Stoichiometric Fuel to fuel and air ratio
h_{iv} (kJ/kg fuel)	305	885	Heat of Vaporization, Source:[5]
h_{iv}^* $F/(F+A)_{\text{stoich}}$ (kJ/kg mixture)	19.5	88.5	Vap.Heat/ kg mixture
Max ΔT charge cooling (K)	19	81	Instantaneous Evaporation during Intake. Source: [4]
Research ON	97 (premium)	109	Source: [6]

In terms of quantifying the chemical antiknock benefit of ethanol blending, the simplest approach is to look at the RON. The RON vs. ethanol volumetric content for a gasoline-ethanol blend is presented in Figure 1 for four types of base gasolines with different starting RON [7]. It is clear from Figure 1, that the RON doesn't increase linearly with volumetric ethanol content but rather reaches an asymptote. The addition of ethanol shows diminishing returns in terms of its chemical benefit, especially for premium gasoline. Reference [6] suggested that the RON vs. ethanol curve becomes linear when using molar ethanol concentrations instead of volumetric concentrations.

References [3] and [8] attempted to quantify the effect of charge cooling on knock. The full thermodynamic charge cooling potential was assumed and a chemical kinetics code was used to quantify the benefit of charge cooling on knock. Reference [9] looked at the experimental knock limited load of fuels that have the same RON but different heat of vaporization.

In terms of knock limit predictive models for engines that cover ethanol blends, reference [10] should be noted. A knock limit integral approach was used with the same autoignition time correlation for both gasoline and gasoline-ethanol blends.

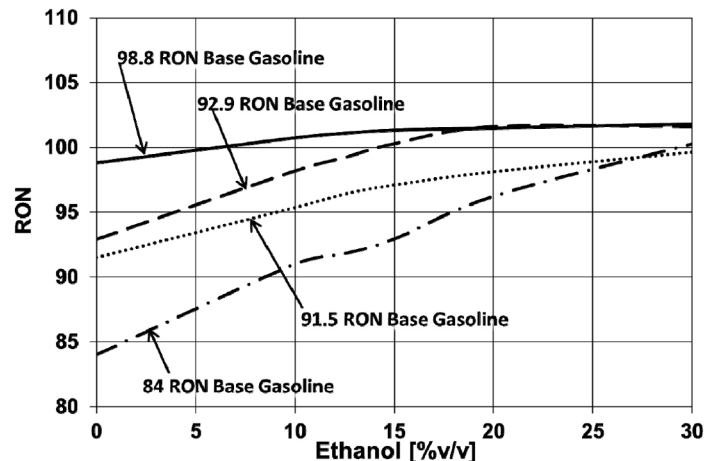


Figure 1. Research Octane Number vs. Volumetric Ethanol Content for Four Types of Base Gasoline- Source: [7]

The methodology used in this paper is an integrated approach: Unlike what has been tried in the literature so far, the amount of charge cooling that takes place is quantified experimentally instead of assuming it is equal to the maximum thermodynamic potential. A novel methodology was developed for that purpose. The methodology uses knock as a diagnostic of charge cooling and is described in the companion paper [4]. This methodology is important because due to heat and mass transfer limitations on fuel evaporation, there is always some fuel that impinges on the cylinder walls or the piston and evaporates extracting heat from the metal walls instead of the charge. The experimental results are subsequently fed to a one dimensional gas dynamics and thermodynamics engine simulation to calculate the evolution of end-gas temperature over time. Using the end-gas pressure and temperature, an autoignition time correlation is adjusted to match the experimental results by means of an "Effective Octane Number"- (ON_{EFF}). The amount of adjustment needed in the ON_{EFF} is a quantitative measure of the fuel's purely chemical resistance to knock and, as will be seen, is a property of the fuel alone and not of the injection type or the amount of charge cooling realized. Finally the hypothetical question is posed: How much would the ON_{EFF} need to be raised if there were no charge cooling to get the same

antiknock performance? The result is essentially a measure of charge cooling expressed as an increase in octane number which we will call the “Evaporative Octane Number”- ON_{EVAP} . The charge cooling antiknock effect can as a result be directly compared to the chemical antiknock effect.

EXPERIMENTAL RESULTS

Using a boosted engine equipped with both direct and port fuel injection systems, measurements were obtained at borderline knock for several gasoline-ethanol blends at different intake air temperatures. All the measurements were obtained at:

- Constant engine speed, 2000 rpm
- Constant stoichiometry, $\lambda=1$
- Constant engine geometry (Wide Open Throttle, with the wastegate bypassing as much flow as possible): This leads to almost constant MAP (Manifold Intake Pressure).
- Constant valve and fuel injection timing.

The maximum pressure at knock onset is plotted vs. intake air temperature for every fuel and injection type in Figure 2. The average of only the knocking cycles is used. Because all data was obtained at knock onset, each line represents borderline knock for that fuel and injection type. As intake air temperature increases, the end-gas is more likely to autoignite. As a result, the maximum pressure at knock drops. In PFI, where there is little charge cooling due to fuel evaporation, the knock borderline moves to higher pressures and temperatures with increasing ethanol content. This is due to the chemistry effect of ethanol in suppressing knock. Additionally, for every fuel, the knock boundary in DI is higher than that for the same fuel in PFI. This is clearly due to the effect of charge cooling. This diagram can therefore be used to quantify how much more charge cooling took place in DI compared to PFI.

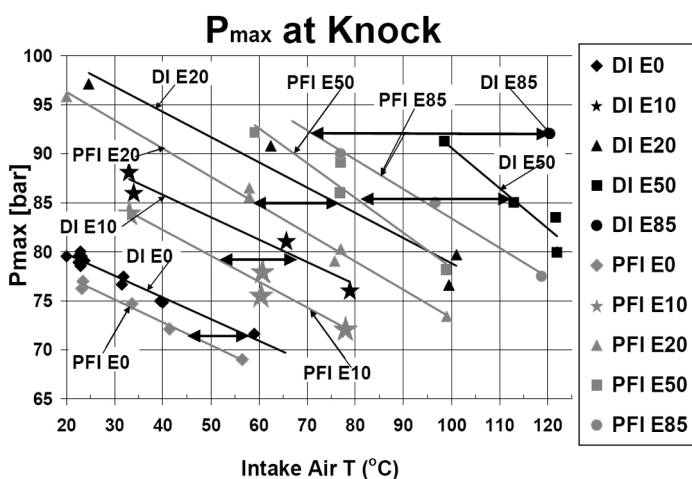


Figure 2. Maximum Pressure at Borderline Knock vs. Intake Air Temperature for Gasoline-Ethanol Blends in DI and PFI- The Average of Only the Knocking Cycles is Used.

To quantify how much charge cooling took place, knock is used as a diagnostic. The question asked is: “How much does the intake air need to be heated in DI to cancel the additional charge cooling out and make DI knock at the same conditions as PFI? As can be seen in Figure 2, for the same fuel, the intake air needs to be heated in DI to be at borderline knock at the same maximum pressure as PFI. This difference in intake air temperature at the same maximum pressure at borderline knock measures the effective difference in charge cooling between DI and PFI and is indicated by arrows in Figure 2. The resulting effective charge cooling numbers are presented in Figure 3 vs. ethanol content. These results and methodology have already been extensively presented in a companion paper [4]. It is also seen in [4], that when the intake air is heated to make the maximum pressure in DI the same as in PFI for the same fuel, the maximum unburned mixture temperature is also essentially the same. Furthermore, the maximum pressure and unburned mixture temperature occur at about the same crank position in DI and PFI at knock onset, so the time available for end-gas autoignition is the same. Essentially, after charge cooling has been cancelled out, conditions at knock are equivalent in DI and PFI.

It is important at this point to emphasize that the amount of additional charge cooling in DI measured using this methodology is an *effective difference in charge*. The additional charge cooling in a DI engine affects in-cylinder processes in multiple ways. The amount of cooling due to fuel evaporation that takes place during intake affects the air flow rate and trapped air mass. The amount of charge cooling during compression affects its polytropic exponent. By using knock as a diagnostic and heating the intake air in DI until the maximum pressures of the knocking cycles at knock onset are the same as PFI, the in-cylinder processes do become more similar. So do air flow rates and trapped air masses as explained in [4] and [11]. The DI and PFI in-cylinder processes have not however necessarily become perfectly matched one to one. It was seen nevertheless that this methodology does result into almost the same end-gas conditions (pressure and temperature) for DI vs. PFI. And in terms of determining knock onset, this is the metric that matters. In a sense, therefore, the charge cooling numbers obtained using this methodology are “effective” charge cooling numbers that collapse all the complexity of the effect of additional charge cooling in DI into a single temperature difference number that matters for knock.

It should also be noted that for the experimental conditions used and described in detail in [4], enough time was available for fuel evaporation and mixing, that the fuel-air mixture before spark was homogeneous without significant stratification present. This was supported by the low CO emission levels and very stable engine operation documented in [11]. Finally, it should be mentioned that even if some stratification is present, it doesn't affect the validity of the methodology which as explained produces “effective”

charge cooling numbers including all the effects that make DI different than PFI.

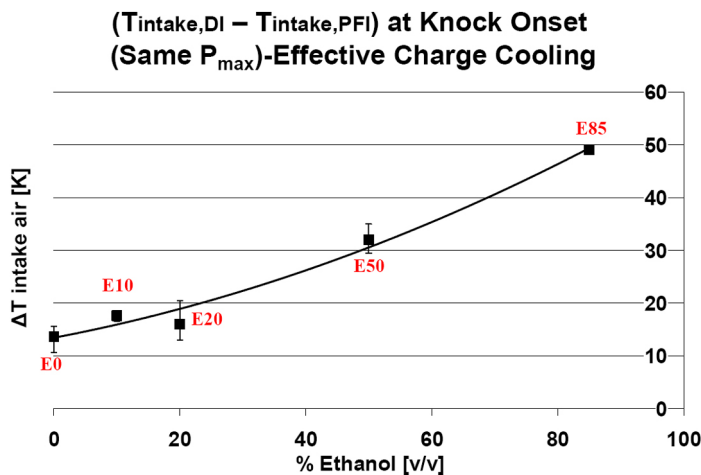


Figure 3. $T_{\text{Intake Air (DI-PFI)}$ at Same Maximum Pressure at Knock Onset-“Effective Charge Cooling” vs. Ethanol Content.

The experimental measurements presented in [4] and Figure 2 were obtained at a constant engine speed of 2000 rpm and base boost engine geometry. Base boost engine geometry means that the engine throttle was wide open and the turbocharger wastegate is also wide open, bypassing as much flow from the turbine as possible. In addition to the experimental data in Figure 2, borderline knock measurements were obtained at different speeds and load/engine geometry using the experimental setup described in [4]. For higher loads the wastegate was partly closed; for lower loads, the throttle was partly closed. The intake air temperature for these tests was kept constant at 30° C for gasoline (E0), 100° C for E50 and 120° C for E85. The higher air temperatures were chosen for E50 and E85 because as explained in [4], it is very difficult to get higher ethanol blends to knock with lower intake air temperatures in an engine designed with a compression ratio for gasoline. The maximum pressure at borderline knock for the three fuel blends in DI and PFI is presented as a function of Net Indicative Mean Effective Pressure (NIMEP) in Figure 4 and as a function of engine speed in Figure 5.

The maximum pressure at borderline knock rises as MAP and NIMEP increase. Air flow rate increases and more mixture mass is compressed. As a result, end-gas pressures and temperatures increase with higher load. Consequently, the knock limited spark advance becomes more retarded at higher load. Because spark timing is more retarded at higher loads, pressure (and temperature) need to be higher for the end-gas to autoignite as there is less time available before the flame consumes it. This explains why the maximum pressure rises with higher load.

The maximum pressure at knock onset is presented in Figure 5 vs. engine speed. It can be seen that the minimum required maximum pressure to get the engine to knock is at

the lowest speed tested-1750 rpm. Lowering rpm has two effects on knock: First, time available for the end-gas to autoignite before it is consumed by the flame increases. This increases the chances of knock occurrence. Second, heat transfer per cycle increases at lower rpm, cooling the end-gas and decreasing the chances of knock [12]. The worst rotating speed in terms of likelihood of knock occurrence is usually between 1500 rpm and 2500 rpm in modern engines. For the speeds tested, the worst was 1750 rpm.

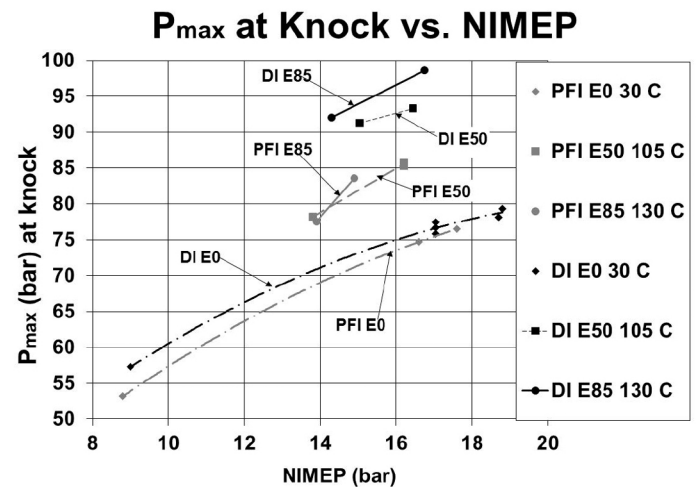


Figure 4. Maximum Pressure at Knock Onset vs. NIMEP

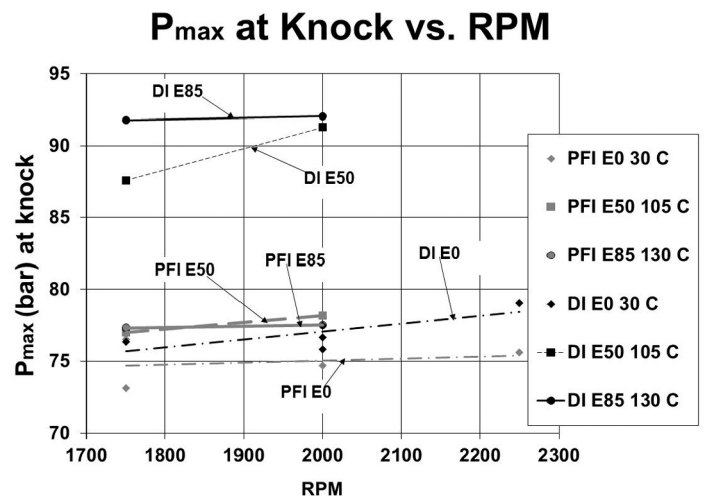


Figure 5. Maximum Pressure at Knock Onset vs. RPM

KNOCK MODELING METHODOLOGY

The methodology for predicting knock consists of two steps:

- Using a 1-D Gas Dynamics and thermodynamics engine simulation code to predict the evolution of end-gas temperature over time.



Documents

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