

Effects of Ethanol Content on Gasohol PFI Engine Wide-Open-Throttle Operation

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ABSTRACT

The NO_x emission and knock characteristics of a PFI engine operating on ethanol/gasoline mixtures were assessed at 1500 and 2000 rpm with $\lambda = 1$ under Wide-Open-Throttle condition. There was no significant charge cooling due to fuel evaporation. The decrease in NO_x emission and exhaust temperature could be explained by the change in adiabatic flame temperature of the mixture. The fuel knock resistance improved significantly with the gasohol so that ignition could be timed at a value much closer or at MBT timing. Changing from 0% to 100% ethanol in the fuel, this combustion phasing improvement led to a 20% increase in NIMEP and 8 percentage points in fuel conversion efficiency at 1500 rpm. At 2000 rpm, where knocking was less severe, the improvement was about half (10% increase in NIMEP and 4 percentage points in fuel conversion efficiency). Because there was no significant change in the end gas temperature in these experiments, the gasohol knock resistance was attributed solely to the ignition chemistry of the ethanol.

INTRODUCTION

With the renewable fuel mandate required in US and the world, ethanol has been increasingly introduced as a supplement to the petroleum based gasoline (gasohol). Although most of the ethanol has been supplied as a low concentration blend to gasoline (E10), usages at high concentrations (E85 or E100) are also in practice in US and Brazil. It is therefore, of interest to assess the impact of gasohol on engine operation.

A significant attribute of ethanol is the high heat of vaporization compared to gasoline. Furthermore, compared to gasoline, significantly more ethanol is required for forming a stoichiometric mixture. Thus there could be substantial evaporative cooling of the fuel air mixture. The positive effects are the charge temperature is lower so that the knock threshold and the volumetric efficiencies could be improved. The chemistry of the ethanol also suppresses knock. The negative effect is a much more problematic cold start [1], which is not the topic of this investigation.

The cooling of the intake air very much depends on that the liquid fuel takes out the heat for evaporation from the air. This process is much more effective in direct injection engines [2], and the gasohol effects for such had been investigated [3]. As a matter of fact, there was a proposal to control engine knock by direct injection of ethanol on demand [4]. The effect of gasohol on Port-Fuel-Injection (PFI) engine, however, is less clear. In these engines, the majority of the injected fuel lands on the port wall and derived the heat for evaporation from the wall heat transfer. Thus there may not be a significant cooling of the intake air by the fuel. The case for PFI engines is important because the configuration still dominates over the US and world markets, and is expected to be so for the immediate future.

In this paper, the effects of gasohol on knock and NO_x emission for a PFI engine at Wide-Open-Throttle (WOT) operation were investigated. The WOT condition was chosen because knock and NO_x were most problematic at such. The objective was to quantify and understand

the relationship between the fuel ethanol content and these attributes.

BASIC CHARACTERISTICS OF ETHANOL/GASOLINE MIXTURES

The pertinent characteristics of gasoline and ethanol are shown in Table 1.

Table 1 Characteristics of gasoline and ethanol

	Gasoline	Ethanol
Representation	$(CH_{1.78})_n$	C_2H_5OH
Specific gravity	0.74	0.785
Stoichiometric A/F	14.6	9.00
Latent heat	305 kJ/kg	840 kJ/kg
Fuel LHV per mass	44 MJ/kg	26.9 MJ/kg
The following are for stoichiometric mixtures		
gaseous density (1bar, 298° K)	1.227 kg/m ³	1.214 kg/m ³
LHV per mass	2.82 MJ/kg	2.69 MJ/kg
LHV per vol.	34.61 MJ/m ³	32.66 MJ/m ³
Adiabatic flame temperature at 1 bar pressure (unburned mixture at 298° K)	2289° K	2234° K
Laminar flame speed (stoichiometric at 298° K, 1 bar)	0.33 m/s [5]	0.41 m/s [6]

Because ethanol is an oxygenate, the stoichiometric air/fuel ratio and the Lower Heating Value (LHV) are much lower than gasoline. Referring to fig. 1, as the fuel ethanol fraction increases, the heating value of the fuel decreases, while more fuel is needed to form a stoichiometric mixture (increase of stoichiometric fuel air ratio). The gaseous density of the mixture remains almost unchanged (see Table 1). The net effect is that the LHV per unit volume of the stoichiometric mixture only drops slightly (-6%). Thus if there is no evaporative cooling effect, the power density of the PFI engine would drop modestly.

Because of the high latent heat of vaporization, and that more fuel is required to form a stoichiometric mixture than gasoline, the evaporation process could substantially cool the charge. The upper limit of cooling is shown in Fig. 2, in which is shown the temperature drop in forming a stoichiometric mixture from gasohol with adiabatic evaporation. In practice, especially with PFI engine, this cooling may not be realized because much of the heat of vaporization is derived from wall heat transfer.

Because of the change of mixture composition with the fuel ethanol content, the thermodynamic properties of the fuel air mixture also changes. The impact on compression temperature, however, is small. This is illustrated in Fig. 3, in which the adiabatic compression temperatures as a function of the ethanol content are

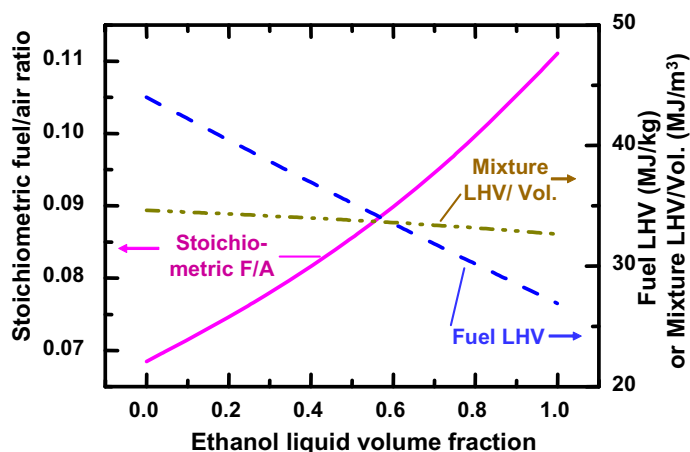


Fig.1 Stoichiometric fuel air ratio and Lower Heating Values as a function of the ethanol content in the gasohol.

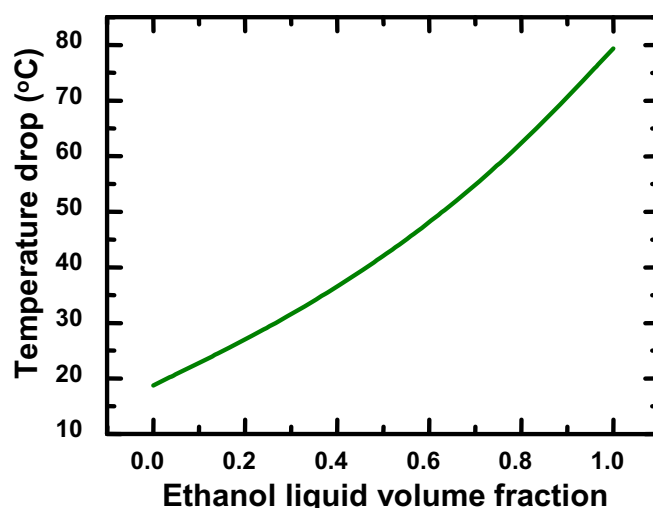


Fig. 2 Temperature drop in adiabatic evaporation of gasohol to form a stoichiometric mixture. Starting temperature was at 300° K.

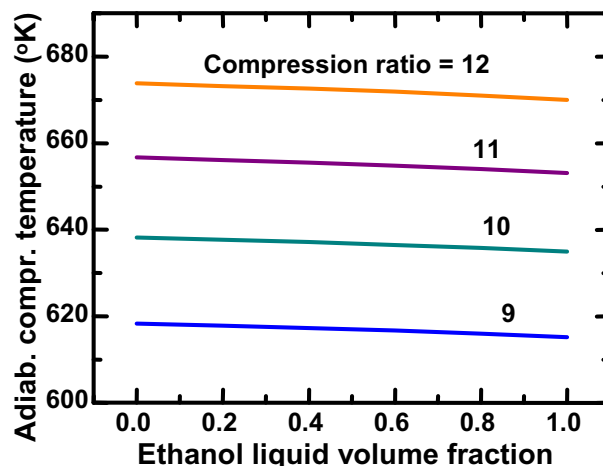


Fig. 3 Adiabatic compression temperature for gasohol/air stoichiometric mixtures at different compression ratios. Starting temperature of the gaseous mixture was 300° K.

plotted for different values of compression ratios. The compression temperature drop from E0 to E100 is about 3° K for compression ratios of 9 to 12.

The heating value per unit mass of the stoichiometric mixture for ethanol is 5% lower than that of gasoline. This is the major factor contributing to the lower flame temperature with gasohol fuel; see Fig. 4. Because of the sensitivity of NO_x formation to temperature, the 2.4% drop in flame temperature moving from gasoline to ethanol could substantially lower the NO_x emission.

The laminar flame speed of ethanol is 24% faster than that of gasoline (see Table 1). A faster combustion would increase the NO_x production. The impact on knock tendency is not clear: advancing combustion phasing would increase knocking, but faster burn out of the end gas would decrease knocking. For modern engine operating at moderate speed and high load, the combustion is usually limited by engine turbulence [5]. Whether the faster laminar flame speed affects the burn rate remains to be seen.

EXPERIMENTAL SET UP AND PROCEDURE

A 1.8L I-4 DOHC spark-ignition engine was modified for single cylinder operation on a motoring dynamometer. See Table 2 for the engine specifications. The engine featured a centrally located spark plug, a shallow bowl-in-piston, and it is representative of modern spark-ignition engines. The intake and exhaust systems were modified to separate the flow through one cylinder from the other three.

Table 2. Engine Specifications

Displacement	442.3cm ³
Compression Ratio	11.0
Bore	80mm
Stroke	88mm
Clearance Volume	208.1cm ³
Connecting Rod	140.5mm
Operating condition	WOT at $\lambda = 1$

A Kistler 6051A piezoelectric pressure transducer coupled to a Kistler 5010 charge amplifier was used for in-cylinder pressure measurements. In-cylinder pressure was pegged to the Manifold Absolute Pressure at BDC of the intake stroke.

All experiments were performed at WOT with $\lambda = 1$. The Horiba Mexa-720 Lambda-NO_x sensor was used for exhaust gas measurements. Engine coolant temperature was at 80° C.

The gasoline was Chevron Philips UTG91 (RON=91; MON=83; RVP=63.4 kPa). The ethanol was anhydrous at 99.5% purity level. The gasoline and ethanol were injected into the intake port with two separate injectors so that different amount of each fuel could be metered by the injection pulse width according to the pressure

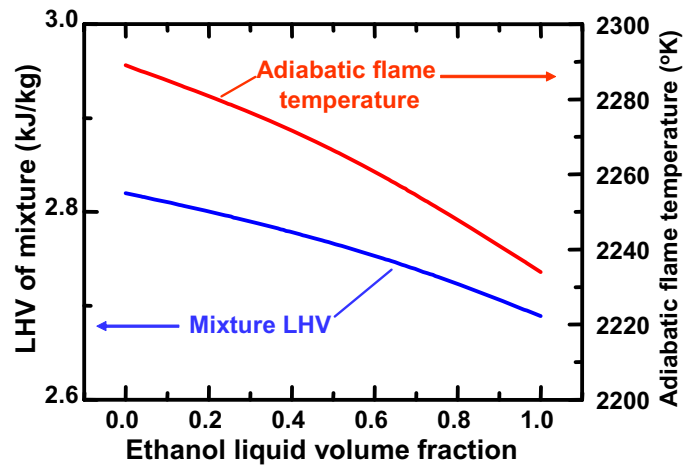


Fig. 4 LHV (per mass) of stoichiometric gasohol mixture and adiabatic flame temperature (at unburned mixture temperature of 298° K and pressure of 1 bar).

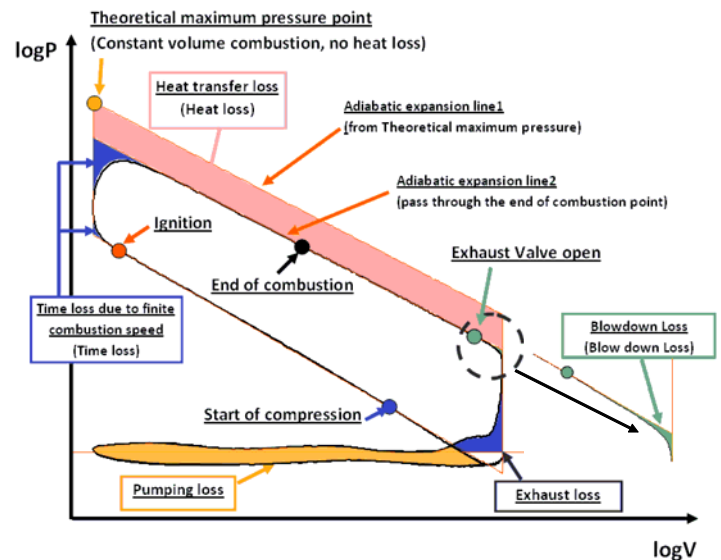


Fig. 5 Schematic of Log(P) versus Log(V) plot illustrating the various indicated loss terms of the engine cycle.

differential across the injector. The injectors were calibrated offline individually with the two fuels.

Heat release analysis was done using the Rassweiler and Withrow method [7]. The various losses were identified by comparing the measured cylinder pressure and the pressure of an idealized cycle in the log(p) versus log(v) plot as shown in Fig. 5. The indicated losses were computed by evaluating the $\int p dV$ terms corresponding to the different regions identified in the figure.

RESULTS AT FIXED SPARK TIMING

To assess the effect of ethanol content on the engine behavior, operation at fixed spark timing was studied first. At 1500 rpm, the spark timing was fixed at 10° BTDC.