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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.

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In this paper, the effects of gasohol on knock and NOx emission for a PFI engine at Wide-Open-Throttle (WOT) operation were investigated. The WOT condition was chosen because knock and NOx 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
Theoretical relative air intake at $\lambda=1$	1	0.973
LHV per mass	2.82 MJ/kg	2.69 MJ/kg
LHV per vol.	3.461 MJ/m ³	3.266 MJ/m ³
Adiabatic flame temperature at 1 bar pressure (unburned mixture at 298° K)	2289° K	2234° K
Laminar flame speed	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

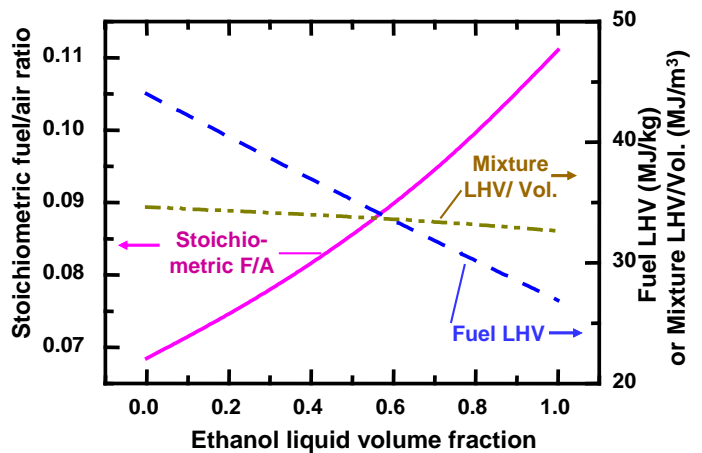


Fig.1 Stoichiometric fuel air ratio and Lower Heating Values as a function of the ethanol content in the gasohol.

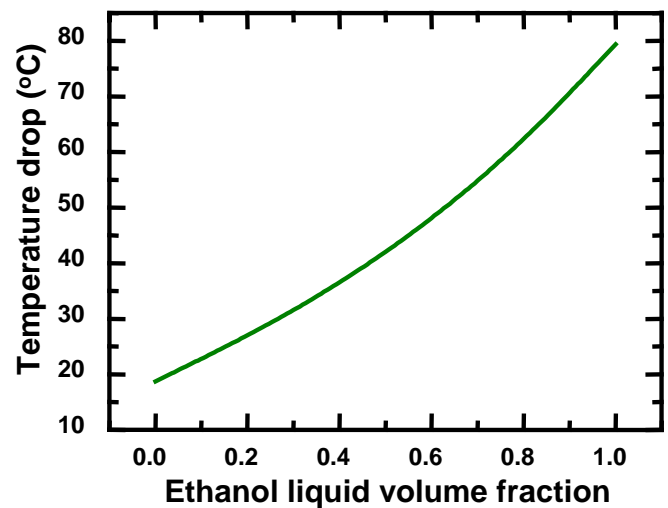


Fig. 2 Temperature drop in adiabatic evaporation of gasohol to form a stoichiometric mixture. Starting temperature was at 300° K.

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 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 NOx formation to temperature, the 2.4% drop in flame temperature moving from gasoline to ethanol could substantially lower the NOx emission.

The laminar flame speed of ethanol is 24% faster than that of gasoline (see Table 1). A faster combustion would increase the NOx 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 Length	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-NOx sensor was used for exhaust gas measurements. Engine coolant temperature was at 80° C.

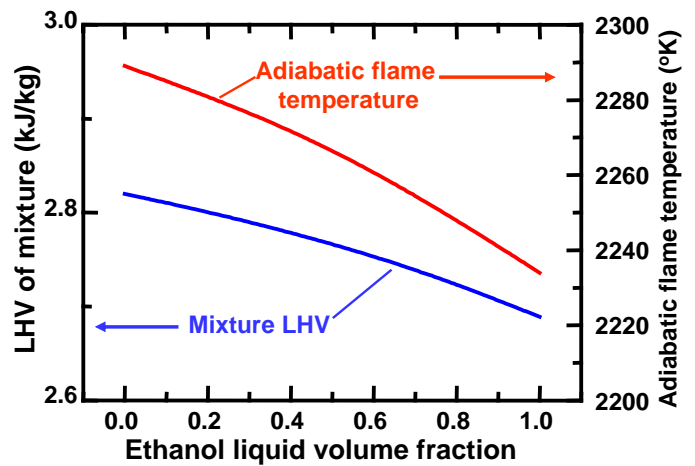


Fig. 4 LHV (per mass) of stoichiometric gasohol mixture and adiabatic flame temperature (at unburned mixture temperature of 298° K and pressure of 1 bar).

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

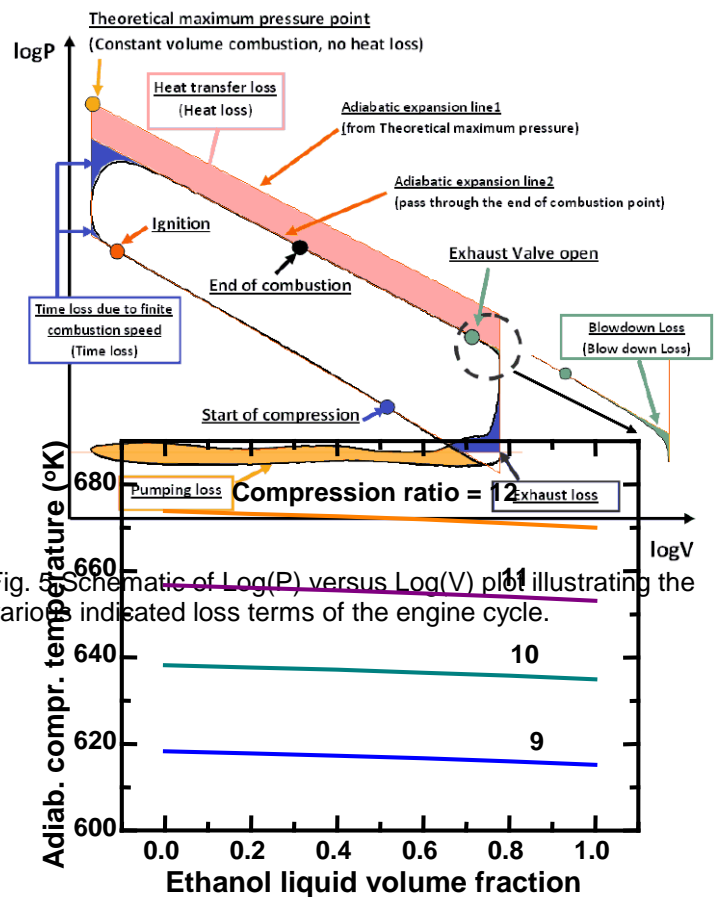


Fig. 5 Schematic of Log(P) versus Log(V) plot illustrating the various indicated loss terms of the engine cycle.

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

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 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 $\int pdV$ 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 chosen to be at -2° BTDC, a value sufficiently retard so that the engine would not knock for the range of gasohol fuels considered.

The fuel, air and energy (LHV of fuel mixture) input per cycle are shown in Fig. 6 for the engine operating at $\lambda = 1$ and WOT. As the ethanol content went up to 100%, the induced air mass was reduced by a small amount (2%). This observation is consistent with the fact that the theoretical displacement effect reduces the air flow by 2.7% (see Table 1). Thus within the margin of experimental error, there was no change in charge cooling due to the evaporation of the fuel. If the evaporation process were adiabatic, there would have been a 60° C change in mixture temperature (Fig. 2), which would correspond to approximately a 20% increase in air flow. Hence it may be concluded that for engines at WOT, the PFI fuel derived the majority of the heat of vaporization from the port walls and the effect of charge cooling was negligible.

The combustion durations are shown in Fig. 7. At ethanol liquid volume fraction higher than 30%, the combustion durations were independent of the ethanol content. Thus the presence of ethanol did not affect the combustion rate. The observation is consistent with the fact that combustion was limited by turbulence rather than laminar burning velocity. For volume fraction less than 30%, however, combustion was found to be slightly slower with more ethanol, in spite of the fact that ethanol has a faster laminar flame speed. The slow down was primarily due to a longer (by 2 to 3° crank angle) initial (0-10%) burn duration.

The NIMEP and net indicated fuel conversion efficiency are shown in Fig. 8. The drop in NIMEP and efficiency when the ethanol content was increased from 0 to 30% was due to the slower combustion. Beyond 30% ethanol

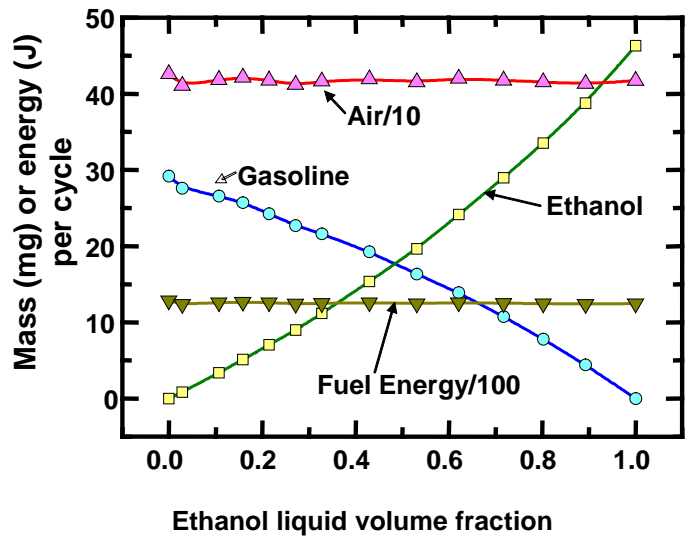


Fig. 6 Fuel, air and fuel energy supplied per cycle; 1500 rpm; WOT; $\lambda = 1$; spark at -2° BTDC.

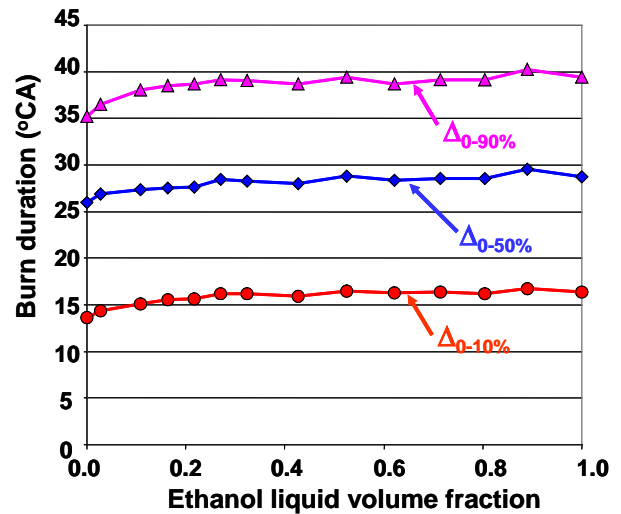


Fig. 7 Combustion durations for gasohol fuels; 1500 rpm; WOT; $\lambda = 1$; spark at -2° BTDC.

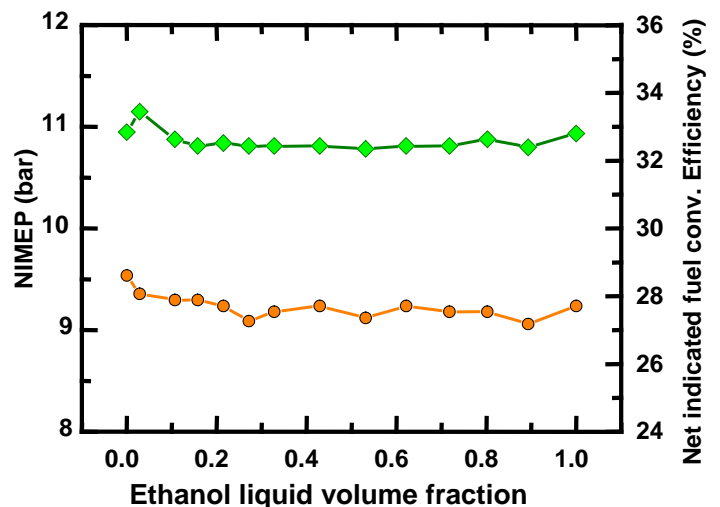


Fig. 8 NIMEP and net indicated fuel conversion efficiency for gasohol fuels; 1500 rpm; WOT; $\lambda = 1$; spark at -2° BTDC.

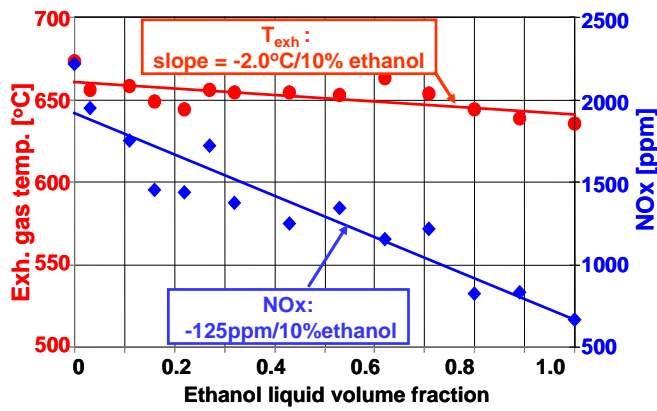


Fig. 9 Exhaust temperature and NOx emission for gasohol fuels; 1500 rpm; WOT; $\lambda = 1$; spark at -2° BTDC.

volume fraction, both values were independent of the ethanol content.

The exhaust temperature and NOx emission as a function of ethanol content are shown in Fig. 9. Both values decreased with increase of ethanol content. From E10 to E100, the exhaust temperature dropped from 660 to 640° C. The % temperature drop ($20/[660+273] = 2.1\%$) is consistent with the drop in adiabatic flame temperature ($[2289-2234]/2289 = 2.4\%$; see Table 1).

Because of the sensitivity of NOx formation to burned gas temperature, the % drop in NOx emission was much larger. The slope was -125 ppm NOx per 10% increase in liquid volume fraction of ethanol in the gasohol. This decrease is mainly due to the reduced temperature of the burned gas with the lower heating value per unit mass of stoichiometric mixture; see Fig. 4.

KNOCK BEHAVIOR

The spark timing at incipient audible knock and MBT timing for the gasohol fueled engine at 1500 rpm, WOT and $\lambda = 1$ are shown in Fig. 10. For gasoline, substantial ignition retard was required to avoid knocking. The amount of retard reduced as the ethanol content increased. At beyond 30% ethanol liquid volume fraction, the ignition timing at incipient knock was more advanced than the MBT timing and knock was no longer the limiting factor for ignition timing. Over the range of the gasohol fuels, the sensitivity of the incipient knock timing to composition was 5.6° CA advance per 10% increase of liquid volume of ethanol for E10-E30, and 1.2° CA per 10% increase for E30 to E100.

The case at 2000 rpm is shown in Fig. 11. The general behavior was similar to that at 1500 rpm. Here, because

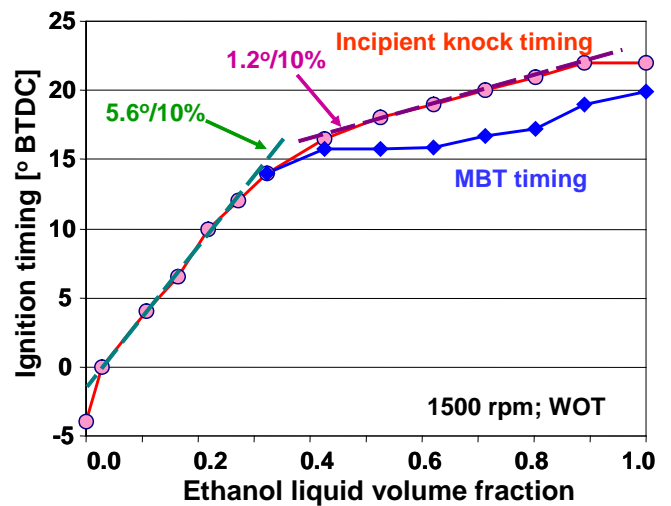


Fig. 10 Ignition timing at incipient knock and MBT timing for PFI gasohol engine at 1500 rpm, WOT, and $\lambda = 1$. MBT timing was not achievable with ethanol volume fraction below 30%.

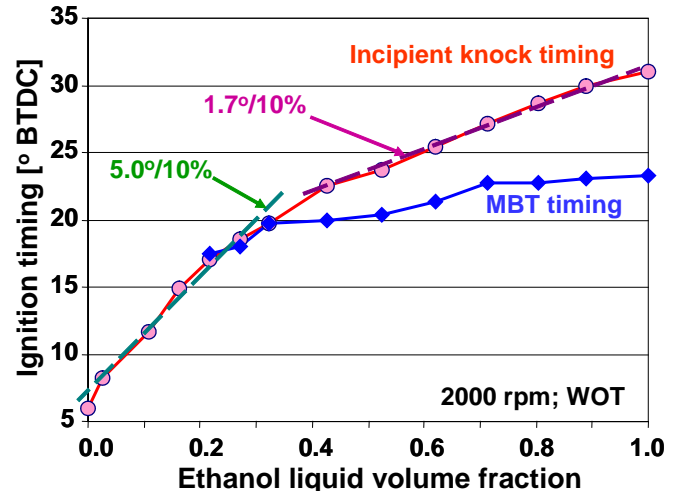


Fig. 11 Same plot as Fig. 10; except at 2000 rpm. MBT timing was not achievable with ethanol volume fraction below 20%.

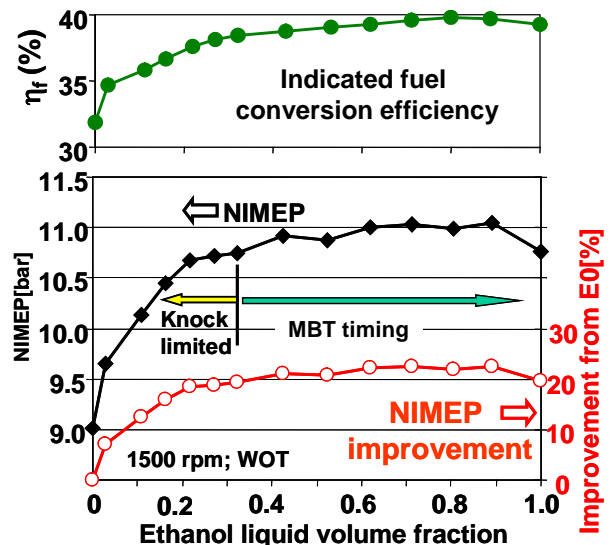


Fig. 12 Indicated fuel conversion efficiency and NIMEP at 1500 rpm; WOT; $\lambda = 1$.

of the less time available for end gas ignition, the incipient knock timing values were more advanced. The threshold of ethanol content below which MBT timing was not achievable was thus lower, at E20. There were also two ranges with different sensitivities of the incipient knock timing to ethanol content; the sensitivity values were also similar.

Because the gasohol fuel could be operated much closer to MBT (or at MBT, if knock was no longer the limiting factor for ignition timing), the torque output and fuel conversion efficiency could be improved substantially. This point is illustrated in Fig. 12. At 1500 rpm, going from E0 to E100, the NIMEP could be improved by 20% and the indicated fuel conversion efficiency (η_f) could be improved by 8 percentage points. Most of the improvement came from the more optimal timing (closer to MBT) when the operation was knock limited.

The last point may be illustrated by the combustion analysis described in Fig. 5. The results are shown in Fig. 13, in which the various losses associated with the data points in Figures 10 and 12 are tabulated. (The blowdown losses were too small to be legible in the plot.) As the ethanol content increased, the spark timing could be advanced closer to MBT. The time loss and exhaust loss (as defined in Fig. 5) reduced accordingly, although there was an increase in heat loss due to the earlier combustion phasing. The net effect was that the total loss was reduced by 8 percentage points (from 68 to 60%) and the fuel conversion efficiency improved correspondingly.

The indicated fuel conversion efficiency and NIMEP are shown in Fig. 14 for gasohol fuels at 2000 rpm; WOT; $\lambda = 1$. Because knocking was less severe at this higher speed, spark timing was closer to MBT with E0. Therefore, the improvement in η_f (by 4 percentage points) and NIMEP (by 10%) were only half of the values at 1500 rpm.

The distribution of the losses at 2000 rpm is shown in Fig. 15. Because ignition timings for the knock limited operation were closer to MBT timing, the time losses there were much smaller than the corresponding values at 1500 rpm.

Discussion: The gasohol fuels were found to be substantially more knock resistant than gasoline. When the fuel was changed from E0 to E100, the incipient knock ignition timing improved from -4° to 22° BTDC at 1500 rpm (WOT and $\lambda = 1$) and from 6° to 31° BTDC at 2000 rpm. Within the margin of experimental error, the charge was not cooled by the evaporation of the ethanol (from E10 to E100 the observed 1% decrease in air flow was accounted for by the 1% decrease in gaseous density of the mixture). Furthermore, the compression temperature difference due to differences in specific heat ratios is small ($<3^\circ$ K). Since the combustion phasing did not change appreciably, the sole explanation for the

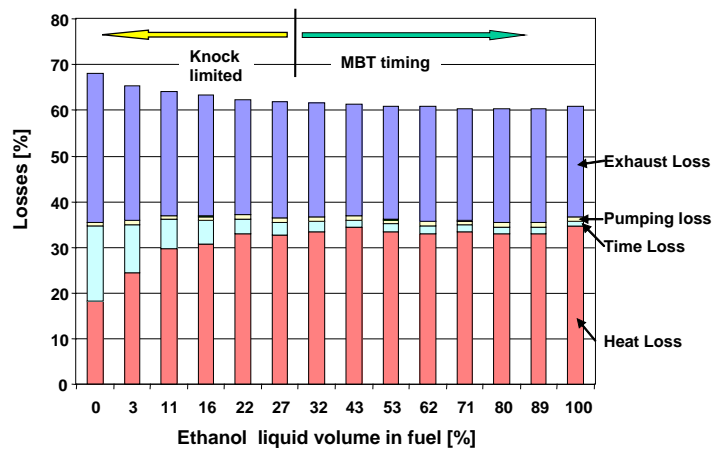


Fig. 13 Various losses (as defined in Fig. 5) for gasohol engine at 1500 rpm; WOT; $\lambda = 1$. Spark timing was optimized at either incipient knock or at MBT timing.

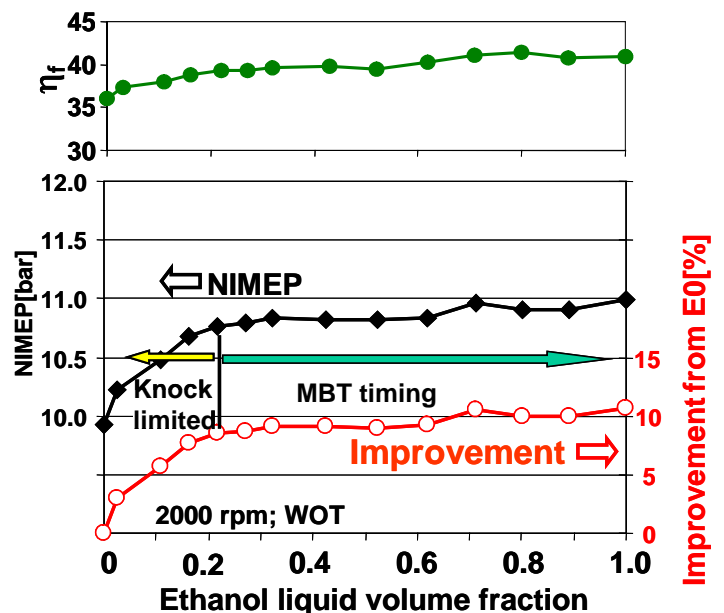


Fig. 14 Indicated fuel conversion efficiency and NIMEP at 2000 rpm; WOT; $\lambda = 1$.

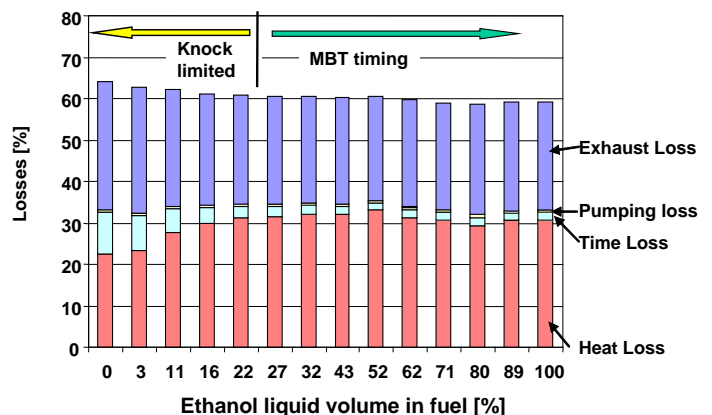


Fig. 15 Various losses (as defined in Fig. 5) for gasohol engine at 2000 rpm; WOT; $\lambda = 1$. Spark timing was optimized at either incipient knock or at MBT timing.

knock resistance of the gasohol fuel has to be in the ignition chemistry of ethanol.

CONCLUSION

The NO_x emission and knock characteristics of a PFI engine operating on ethanol/gasoline mixtures were assessed at 1500 and 2000 rpm with $\lambda = 1$, WOT.

1. With the ethanol volume fraction changing from 0% (E0) to 100% (E100), the air intake per cycle decreased by 2%, which could be accounted for by the change in air flow due to composition change of the stoichiometric mixture. It was thus concluded that there was no significant cooling of the intake charge by the ethanol evaporation process in a PFI engine.
2. At fixed ignition timing:
 - A change from E0 to E30 resulted in an increase of the 0-10% burn duration by 3° CA, while the 10-90% duration remained approximately the same. For ethanol content from E30 to E100, there was no change in the burn duration. Thus the overall impact of ethanol on burn rate was not substantial.
 - The NO_x emission decreased with a slope of -125 ppm per 10% increase in liquid volume fraction of ethanol in the fuel. The exhaust temperature also decreased, but with a more modest slope — at -2° C per 10% increase. These decreases could be explained by the lower adiabatic flame temperature of the mixture.
3. The presence of ethanol in the fuel significantly increases the fuel knock resistance. At 1500 rpm, the incipient knock timing could be advanced at 5.6° per 10% increase in ethanol fuel liquid volume content in the E0 to E30 range, and at 1.2° per 10% increase in the E30 to E100 range. At 2000 rpm, the respective value became 5° and 1.7° per 10 % increase.
4. The improvement of fuel knock resistance enabled operation much closer or at MBT ignition timing. Thus going from E0 to E100, the NIMEP could be improved by 20% and η_f by 8 percentage points (from 32 to 40). At 2000 rpm, because knocking was less severe and the operation with E0 was closer to MBT ignition, the improvements were half of that of at 1500 rpm: NIMEP improved by 10% and η_f by 4 percentage points.
5. Since the unburned charge temperature and combustion phasing did not appreciably change with the gasohol mixtures, it was concluded that the improvement in knock resistance was solely due to the ethanol chemistry.

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DEFINITIONS, ACRONYMS, ABBREVIATIONS

BTDC Before-Top-Dead-Center

E0, E10, ... Ethanol liquid volume fraction in gasohol;
E0 is 0% ethanol; E100 is 100% ethanol
LHV Lower Heating Value
MBT Maximum Brake Torque
NIMEP Net Indicated Mean Effective Pressure
PFI Port-Fuel-Injection

RVP Reid Vapor Pressure
WOT Wide-Open-Throttle operating condition
 λ Air-to-fuel equivalence ratio
 η_f Indicated net fuel conversion efficiency