

Effect of Ethanol on Part Load Thermal Efficiency and CO₂ Emissions of SI Engines

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ABSTRACT

This paper presents engine dynamometer testing and modeling analysis of ethanol compared to gasoline at part load conditions where the engine was not knock-limited with either fuel. The purpose of this work was to confirm the efficiency improvement for ethanol reported in published papers, and to quantify the components of the improvement.

Testing comparing E85 to E0 gasoline was conducted in an alternating back-to-back manner with multiple data points for each fuel to establish high confidence in the measured results. Approximately 4% relative improvement in brake thermal efficiency (BTE) was measured at three speed-load points.

Effects on BTE due to pumping work and emissions were quantified based on the measured engine data, and accounted for only a small portion of the difference. Approximately half of the improvement was accounted for by the fact that the heat of vaporization (HoV) of the fuel detracts from the heat release measured in the combustion bomb used in the determination of heating value, but does not detract from the heat released during combustion in the engine. Engine modeling indicated that the remaining difference in BTE is due to lower burned gas temperatures and consequently lower heat transfer losses. The lower temperatures are due to greater charge cooling and to lower adiabatic flame temperature.

CO₂ emissions at part load are reduced about 7% for ethanol compared to gasoline. Approximately 4% CO₂ benefit is due to improved thermal efficiency, and about 3% is due to the increased hydrogen-to-carbon ratio (H/C) of ethanol.

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INTRODUCTION

CO₂ emissions from automotive engines are a significant contributor to greenhouse gases. It is anticipated that renewable fuels will play an important role in reducing these emissions. Renewable fuels can have an impact on CO₂ emissions through "field-to-tank" effects and through "tank-to-wheels" effects. This paper addresses one aspect of the tank-to-wheels CO₂ emissions of ethanol.

It is well known that ethanol has very high effective octane and knock resistance, which provides opportunity to improve engine thermal efficiency and reduce CO₂ emissions. This can be achieved through downsizing and downspeeding enabled by increased full load performance, and through increased compression ratio [1, 2, 3, 4, 5, 6].

Additionally, several studies have reported improved thermal efficiency with increased ethanol content at part load conditions [6, 7, 8], where combustion phasing can be maintained at the thermodynamic optimum timing with the baseline gasoline fuel, and therefore the increased knock resistance of ethanol does not play a role. A paper by GM reported thermal efficiency improvements of 3% to 6% with E85 compared to an optimized direct injected (DI) gasoline baseline, at certain brake mean effective pressure (BMEP) and engine speed operating conditions [7]. These gains were attributed to a reduction in heat transfer losses and to increased dilution tolerance, which was available with variable valve timing. Similarly, a paper by Toyota documented efficiency improvements of 3% by increasing ethanol content from zero to 100 percent at equal combustion

phasing with a port fuel injected (PFI) engine, and attributed this improvement to reduced heat transfer losses [8]. A study by Oak Ridge National Laboratory (ORNL) and Delphi also documented efficiency improvements for increasing ethanol content, which was attributed to an increased number of moles in the combustion products for ethanol, consequently resulting in lower burned gas temperature and heat transfer losses [6].

The purpose of the work documented in this paper was to confirm the reported efficiency improvement at part load conditions for ethanol compared to gasoline, and to identify and quantify the contributing components of the improvement. The approach taken was to acquire careful engine dynamometer measurements at part load conditions and to exercise Ford's general engine simulation (GESIM) model at the same conditions to provide a detailed understanding of heat transfer and thermodynamic effects.

ENGINE AND MODEL DESCRIPTION

Engine Hardware

A 5.0L V8 gasoline turbocharged direct injection (GTDI) engine was used for this testing (Table 1). The engine was equipped with twin mono-scroll turbochargers with log style exhaust manifolds and twin independent variable cam timing (TiVCT). The engine combustion system is described in [1], and detailed engine specifications are provided in [3].

Table 1. Engine description.

Displaced volume	4951 cc
Bore	88.5 mm
Stroke	100.6 mm
Intake cam duration	243 °CA
Exhaust cam duration	253 °CA
Compression ratio	9.5:1

This engine was used for a dual fuel (E85 and gasoline) investigation that was partially funded by the Department of Energy [1, 2, 3]. The data presented in this paper was acquired during a side study to investigate BTE values for E85 which were higher than those of gasoline at equivalent engine operating conditions. The engine dynamometer test results are at specific part load engine speed and BMEP conditions where the engine was not knock-limited. Thus, the conclusions derived from analysis of the data are expected to be applicable to automotive engines in general, with only minor differences in the efficiency improvement of ethanol depending on the details of the engine design and the type of fuel system, as discussed in a later section of this paper.

Engine Model

GESIM is a lumped, single-cylinder, quasi-dimensional spark ignition engine model whose capabilities are described in detail in [9, 10, 11, 12]. Only the features which are crucial to an accurate prediction of the in-cylinder heat transfer losses are noted here:

- GESIM incorporates a turbulent-entrainment, eddy-combustion flame propagation model based on the measured temperature and pressure dependence of the laminar flame speed for the particular fuel supplied. Once calibrated for one operating condition and fuel, the model can accurately predict the combustion rate at other operating conditions for a variety of fuels.

- GESIM includes a multi-zone model of the in-cylinder gases. From exhaust valve opening (EVO) to ignition the cylinder contents are treated as a single unburned zone. From ignition to EVO, an adiabatic core zone and a boundary-layer zone adjacent to the chamber walls are added to represent the burned gases behind the flame front. Each zone is characterized by its own temperature, turbulence intensity, and composition of fuel, air, and combustion products.

- In contrast to strictly zero-dimensional models, the quasi-dimensional GESIM can account for the effects of combustion chamber geometry on flame development and heat transfer. The surface areas of the head, piston, and bore wall in contact with the unburned and burned gases as well as the flame area at any time are obtained from tables generated from a solid model of the combustion chamber of the actual engine under the assumption that the burned gases behind the flame form a truncated sphere centered at the spark plug.

- The unburned and burned gases have distinct models for their heat transfer with the chamber walls. Both depend on the instantaneous turbulence within their respective zones. For the unburned zone, the heat transfer is purely convective, but for the burned gases, the heat path from the adiabatic core to the walls goes through the boundary layer whose overall thermal resistance (due to both molecular and convective effects) is calculated using an integration method developed by Borgnakke et. al. [13]. This approach has been found to capture observed trends in thermal efficiency with engine speed and load quite well.

- The path of the rejected heat through the block, head, and piston to the coolant and oil is modeled as a thermal resistance network which is solved every cycle for the temperatures at nodes representing the gas-side surfaces of the chamber, thus guaranteeing consistency between those temperatures and the heat transferred as operating conditions are changed.

Fuel Properties

For this study, pure ethanol was used in the model as a substitute for E85. Fuel properties for the gasoline and E85 used in the engine tests and for pure ethanol are shown in Table 2 [4,14]. The H/C and oxygen-to-carbon ratio (O/C) were determined using ASTM procedures D5291 and D5622, respectively. Net heating value (NHV) was measured using ASTM procedure D240. The heat of vaporization (HoV) for E85 was estimated from those of gasoline and pure ethanol using the weight fractions of gasoline and ethanol in the fuel.

Table 2. Fuel properties.

Property	Gasoline	E85	Ethanol
Ethanol (%v/v)	0	82.7	100
H/C (mole/mole)	1.83	2.72	3.0
O/C (mole/mole)	0	0.387	0.5
RON	90.7	109	109
MON	83.4	90	90
Sensitivity	7.3	19	19
NHV (MJ/kg fuel)	43.4	29.2	26.9
(MJ/L fuel) ^a	32.5	22.9	21.4
(MJ/kg air at stoich)	2.98	2.98	3.00
(MJ/kg stoich mixture)	2.79	2.70	2.69
HoV ^b (kJ/kg fuel)	350	850	920
(kJ/MJ heating value)	8.1	29.1	34.2
Stoichiometric A/F	14.54	9.81	9.00
Density ^a (kg/L)	0.748	0.785	0.796

^a at 15.6 °C (60 °F)^b at 25 °C

TEST DATA AND MODEL RESULTS

Test Data

Alternating back-to-back testing with E85 and E0 gasoline was conducted using DI at three part load points: 1500 rpm / 2.62 bar BMEP, 1500 rpm / 5 bar BMEP, and 2000 rpm / 5 bar BMEP. These points were chosen so that E0 91 RON gasoline fuel could be run at the minimum spark advance for best torque (MBT) without knock. Spark advance was adjusted as needed to keep the combustion phasing at MBT, with a crank angle of 50% mass fraction burned (CA50) between 7 and 8°CA aTDC. Air-fuel ratio (A/F) was adjusted to maintain stoichiometry. The engine coolant temperature was kept constant at 90°C and the charge air cooler (intercooler) outlet temperature was kept constant at 38°C. TiVCT settings were adjusted for minimum brake specific fuel consumption (BSFC) with gasoline at each speed/load point and then maintained constant for the alternating back-to-back testing.

As the fuel was changed from E85 to gasoline and back to E85 during this alternating back-to-back testing, consistent change in BTE was observed, as shown in [Figure 1](#). (In [Figure 1](#), [Figure 4,5,6](#), and [Figure 14](#), the lines connect data points for the same speed-load point. The data points alternate between E85 and gasoline.)

The average difference in BTE is about 1.2% absolute (3.8% to 4.6% relative difference), with higher BTE for E85 than for E0 gasoline. The traditional BSFC metric is based on the mass flow rate of fuel, so it is affected by differences in the energy density of the fuel, as well as differences in BTE. E85 has 32.7% lower NHV than gasoline, but the average difference in BSFC is only 30.0%, which reflects the efficiency benefit of E85. In some references, brake specific energy consumption (BSEC), the rate of fuel energy input divided by brake power output, is used as an alternate

measure of engine efficiency instead of BTE. BSEC is the inverse of BTE multiplied by a constant, and therefore conveys the same information.

Although each fuel line had its own fuel flow measurement meter, the fuel meters were interchanged to verify the results at the 1500 rpm / 5 bar BMEP point with excellent repeatability (within 0.1% absolute difference in average BTE), ruling out fuel flow measurement as a possible cause of the measured difference in BTE.

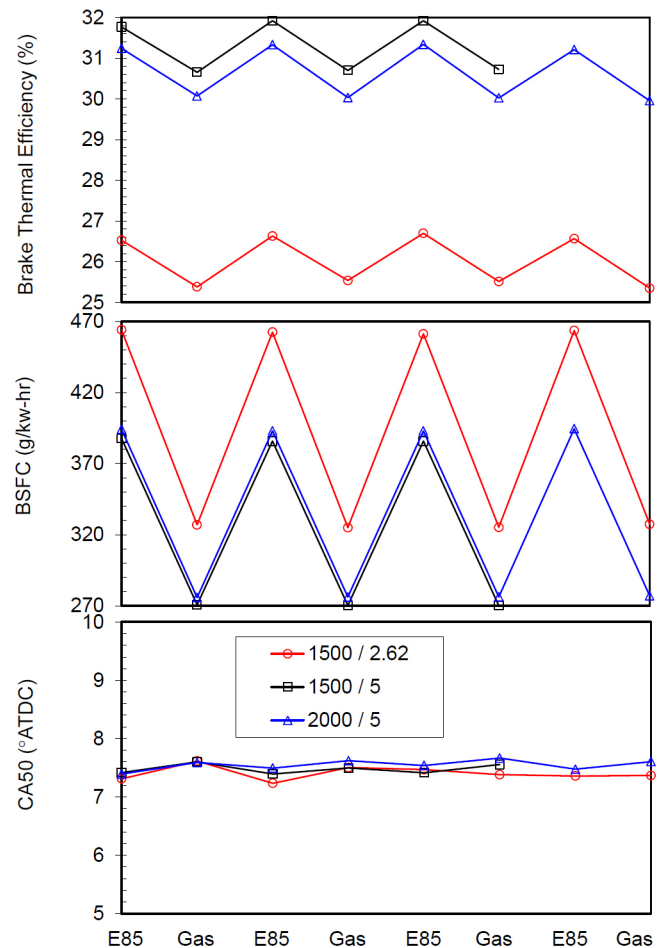


Figure 1. Brake thermal efficiency, BSFC and CA50 of E85 and gasoline at various speed/loads.

Model Results

Model calibration was performed at the 1500 rpm / 2.62 bar BMEP operating condition for gasoline. GESIM's flame propagation and heat transfer submodels were adjusted so that the calculated MBT spark advance and thermal efficiency nominally matched the measured values. Without further adjustment, GESIM was then applied to the other two operating conditions for gasoline and all three operating conditions with pure ethanol (E100) as a surrogate for E85. Comparison of model calculations to the engine data for MBT spark advance and BTE are shown in [Figure 2](#) and [Figure 3](#), respectively.

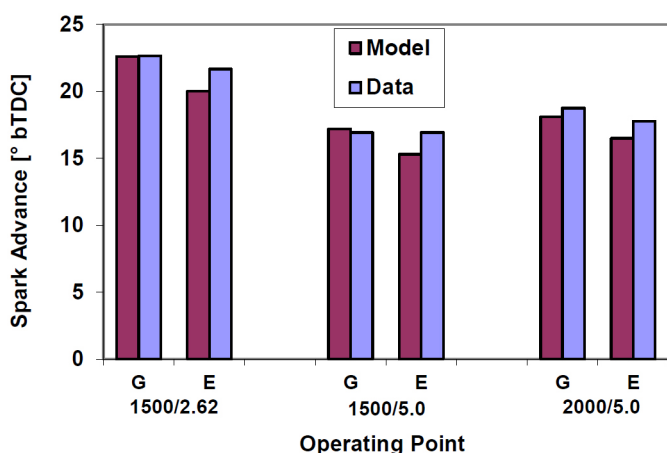


Figure 2. Comparison of MBT from model and data for gasoline (G) and ethanol (E).

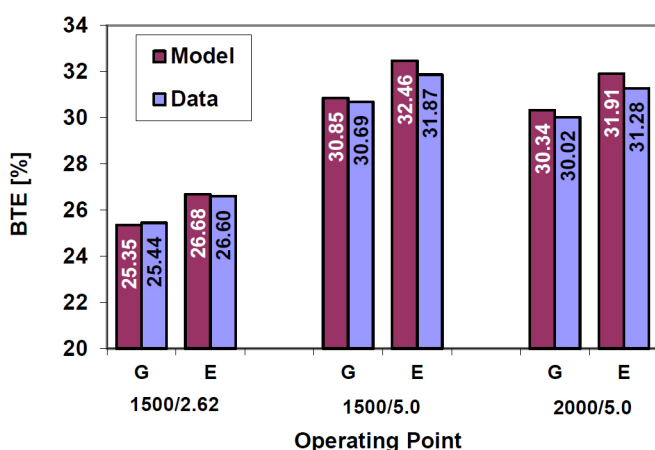


Figure 3. Comparison of BTE from model and data for gasoline (G) and ethanol (E).

In both cases there is excellent agreement for all conditions with gasoline as fuel. GESIM predictions for E100 also correspond quite well with the data for E85, but, as might be expected, any changes due to changing fuel from gasoline to E85 are slightly exaggerated by switching to E100 instead. A corresponding comparison of the relative percentage difference in BTE between ethanol and gasoline is shown in Table 3. The E85 used in the engine tests was 82.7 % ethanol by volume, which corresponds to about 92% ethanol by molar fraction [4]. Hence the values calculated by GESIM are multiplied by 0.92 for comparison to the engine data.

The quality of this agreement between model calculations and measured quantities serves as an indicator that GESIM is properly simulating the engine at the selected operating conditions and that the calculations of unmeasured quantities such as heat loss are reliable as well.

Table 3. Relative percent differences in BTE between ethanol and gasoline from model and engine data.

	Engine Speed/BMEP		
% BTE (relative)	1500/2.62	1500/5	2000/5
Model with ethanol	5.0	5.0	4.9
Model x 0.92	4.6	4.6	4.5
Engine data with E85	4.6	3.8	4.2

ANALYSIS OF EFFECTS ON BTE

Sum of Measured Components

As a first step in investigating the improvement in BTE with ethanol compared to gasoline, the effects of differences in BSCO, BSHC, and pumping work on BTE were quantified based on the measured engine dynamometer data. BSCO and BSHC represent lost fuel energy, and pumping work is a loss which also detracts from BTE. Plots of BSCO, BSHC, and pumping mean effective pressure (PMEP) are shown in Figure 4. No clear trend in BSCO is evident at the 1500/2.62 bar BMEP condition. At the 5 bar BMEP conditions, BSCO is lower for E85 than for gasoline. At stoichiometry CO is formed in rich regions due to imperfect fuel-air mixing. CO is also a result of chemical equilibrium at high temperatures, and lower combustion temperatures result in more complete combustion of CO to CO₂ [14,15]. Thus, lower BSCO for E85 indicates that the air-fuel homogeneity is improved with E85 compared to gasoline at these conditions and/or that lower combustion temperature with E85 results in reduced equilibrium levels.

It should be noted that CO formation is highly sensitive to lambda (the ratio of the actual A/F to the stoichiometric A/F). Using the H/C and O/C of each fuel, the stoichiometric A/F and measured A/F based on SPINDT analysis [16] of the exhaust constituents were calculated. As shown in Figure 5, differences in CO between E85 and gasoline are reflected as differences in O₂, where higher levels of CO result in higher levels of O₂ in the exhaust gas. Also as shown, lambda was held constant within very tight limits for these data. Therefore the differences in CO are not due to differences in lambda and can be attributed to the aforementioned air-fuel homogeneity and chemical equilibrium effects.

As mentioned previously, BSHC and BSCO represent lost fuel energy. The effects of BSHC and BSCO on BTE can be estimated by expressing the mass flow of HC and CO as an emissions index (mass flow rate of the emission divided by the mass flow rate of the fuel). For HC, this implicitly makes the simplifying assumption that the heating value of unburned HC is the same as that of the fuel. For gasoline fuel, the CO emissions index is multiplied by 0.3 to account for the heating value of CO and the H₂ associated with CO relative to that of gasoline [17]. For ethanol fuel, the CO emissions

index is multiplied by 0.4 instead of 0.3 because the NHV is lower for ethanol than for gasoline.

Components of the fuel efficiency difference calculated in this manner are shown in Table 4. There is about a 0.6% to 0.7% BTE improvement with E85 due to lower BSHC. Note that the BSHC measurements were adjusted using the FID response correction equation of [18]. Also as shown, lower BSCO for E85 corresponds to 0.2% and 0.5% improvement in BTE at 1500 rpm / 5 bar BMEP and 2000 rpm / 5 bar BMEP, respectively.

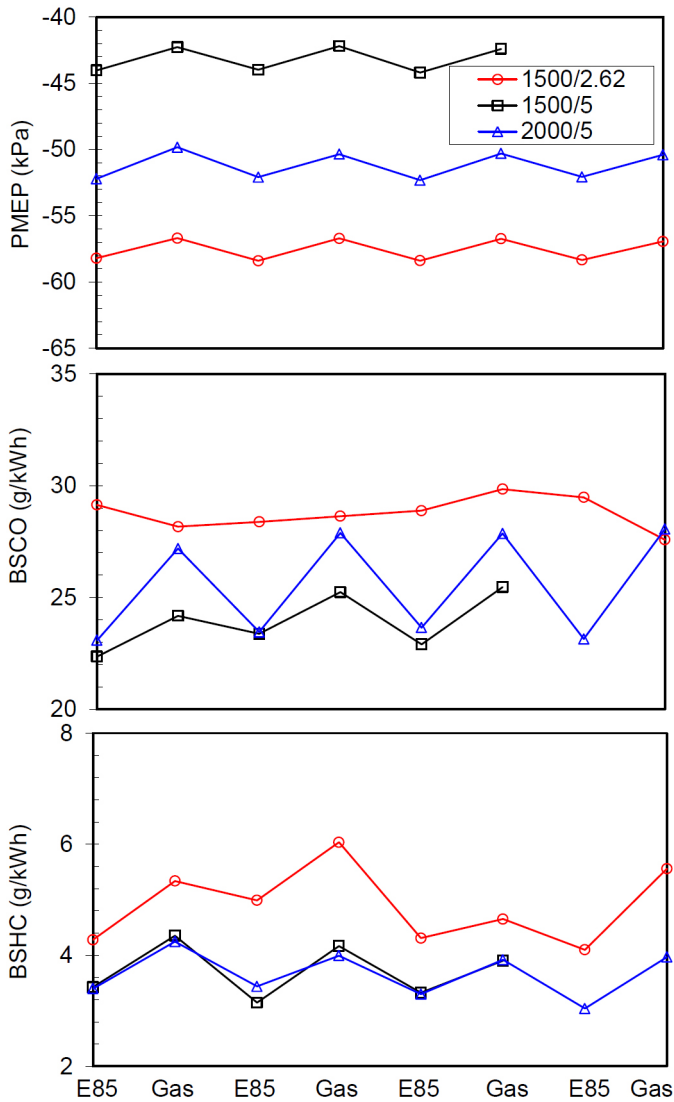


Figure 4. PMEP, BSCO and BSHC of E85 and gasoline at various speed/loads.

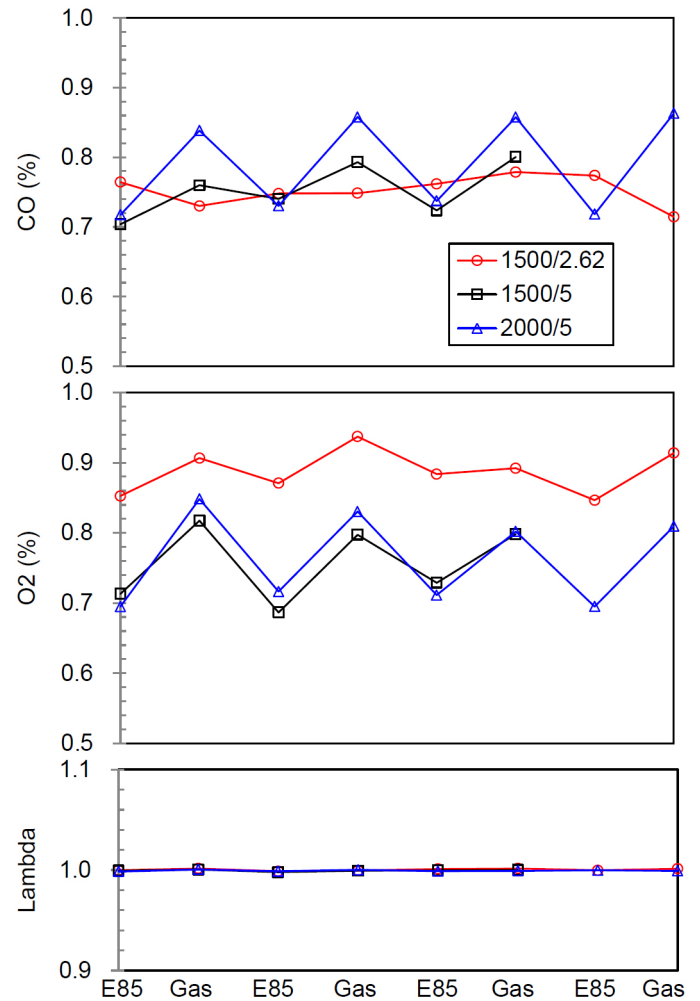


Figure 5. Engine out CO, O2 and lambda of E85 and gasoline at various speed/loads.

Table 4. Measured components of BTE differences between E85 and gasoline expressed as relative percent.

% BTE (relative)	Engine Speed/BMEP		
	1500/2.62	1500/5	2000/5
BSCO	- 0.1	0.3	0.6
BSHC	0.7	0.7	0.6
PMEP	- 0.4	- 0.3	- 0.3
Sum of Components	0.2	0.7	0.9