

Pre-Ignition Characteristics of Ethanol and E85 in a Spark Ignition Engine

L. J. Hamilton, M. G. Rostedt, P. A. Caton and J. S. Cowart

U.S. Naval Academy

ABSTRACT

Ethanol based fuels have seen increased use in recent years due to their renewable nature as well as increased governmental regulatory mandates. While offering performance advantages over gasoline, especially at high compression ratios, these fuels are more sensitive to pre-ignition (PI). Pre-ignition experiments using ethanol (E100) and E85 were performed in a CFR spark ignition engine using a diesel glow plug "hot spot" to induce PI. PI is found to occur over a specific air-fuel ratio range based on hot spot temperature. Additionally, increasing ethanol content or compression ratio (CR) decreases glow plug temperature thresholds for PI. A kinetics-based model was used to simulate pre-ignition of E100 and to elucidate sensitivities of pre-ignition to various operating parameters, including initial charge temperature, air dilution, and residual dilution. The model shows that the most violent cases of PI can be mitigated by switching to either lean or rich operation.

INTRODUCTION

Diminishing petroleum reserves and increasing gasoline prices have sparked interest in alternative fuels such as ethanol. According to the Biomass Research and Development Technical Advisory Committee established by the Biomass R&D Act of 2000 {P.L. 106-224}, "transportation fuels from biomass will increase significantly from 0.5 percent of U.S. transportation fuel consumption in 2001 to 4 percent of transportation fuel consumption in 2010, 10 percent in 2020, and 20 percent in 2030" [1]. Gasohol or E10, a blend of 10% ethanol and 90% gasoline, has been available for many years and can be run in unmodified automobile engines. Additionally, several "flex-fuel" vehicles have been produced that can run on blends of up to 85% ethanol-15% gasoline (E85).

In a previous study, the authors highlighted some significant performance benefits associated with ethanol-gasoline blends [2]. In this study, it was shown that E85 use resulted in significant thermal efficiency and torque

gains as its high octane permitted increases in compression ratio without knock. In general, both efficiency and torque increased about 2% per compression ratio (CR) unit.

There have been numerous studies that have focused on the performance characteristics of lower alcohol blends of ethanol with gasoline (less than E40) [3, 4, 5, 6]. The first two of these references showed that for a conventional (nominal for gasoline operation) fixed CR the brake power, torque, thermal efficiency, and indicated specific fuel consumption increase while stoichiometric air-fuel ratio decreases with increasing fractions of ethanol in gasoline. Reference 5 and 6 also included variable compression ratio experiments ranging from approximately 8 to 12 with increasing compression ratio providing best performance with increasing ethanol fraction.

While ethanol blends have significant performance advantages, they also have higher pre-ignition (PI) tendencies than gasoline. PI can be due to surface ignition that results in flame initiation and propagation before the spark [7]. Hot spots such as spark plugs, exhaust valves or carbon deposits are usually sources of PI [8, 9]. Because combustion is advanced before timing that gives maximum torque, high temperatures and pressures capable of damaging the engine can occur. Of all the surface ignition phenomena, PI is potentially the most damaging [9].

To quantify the PI tendency of a fuel, a 0-100 rating scale can be used based on the power required to cause PI using an in-cylinder heating coil. On this scale, 0 represents the PI character of cumene (which readily exhibits PI) and 100 represents iso-octane (resistant to PI) [10]. Using this scale, pure ethanol has a pre-ignition rating of -28 while gasoline has a rating of 70 [11].

Surprisingly, little research has been published on ethanol PI characteristics, probably due to the possibility of severe engine damage. Suga et al [8] investigated PI phenomena using a mix of 85% methanol and 15%

gasoline (M85) with a post-ignition spark interrupt technique. They found that M85 had a lower pre-ignition temperature and faster in-cylinder pressure rise than premium unleaded gasoline.

The primary goals of this study are twofold: (1) to determine the effect of various engine parameters on PI and (2) to quantify the PI characteristics of ethanol and E85 in a spark ignition engine.

EXPERIMENTAL SETUP

The engine used in this study is a 0.6L Cooperative Fuels Research (CFR) variable compression ratio (CR) gasoline engine. In this study, CR was varied from 12.0 to 15.0. This engine was originally designed for research method octane number (RON) testing and motoring method octane number (MON) testing. The CFR unit contains a motor-generator for maintaining the engine at a constant 900 rpm during steady state operation. Further information on the engine can be found in the Waukesha™ CFR manual [12].

The original CFR carburetor and ignition system were removed. A conventional automotive coil and a production Ford™ 4-hole fuel injector were installed. A new intake manifold was constructed using 30 mm diameter pipe containing a fuel injector port (directed at the intake valve), 3.7 liter transient damping tank and throttle (wide-open for all of the tests in this study). Figure 1 shows the experimental apparatus schematic.

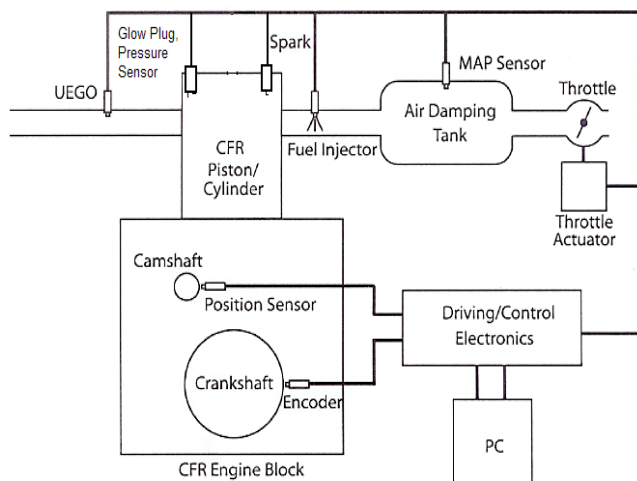


Figure 1: Experimental apparatus

The fuel injector and ignition coil were all controlled by a custom engine controller with associated high power switching electronics external to the PC. The PC was equipped with a Cyber Research™ CYDAS1600 A/D card that interfaces with C code run on the PC. A BEI™ shaft encoder and an optical cam encoder were used by

the PC for crank angle degree resolved engine position control. The engine was equipped with an automotive manifold absolute pressure (MAP) sensor on the intake damping tank as well as a universal exhaust gas oxygen (UEGO) sensor in the exhaust.

A General Motors Duramax™ diesel glow plug was used to create a local hot spot in the combustion chamber to cause PI. A custom instrumentation plug containing the glow plug and Kistler™ 6125B pressure transducer was inserted into “detonation pickup hole” in the CFR cylinder head. Figure 2, a graphic taken from the CFR manual, shows the location of this hole. Figure 3 shows a detailed drawing of the glow plug setup. The tip of the diesel glow plug emerged 2.5 mm into the combustion chamber. A thermocouple was used to measure glow plug shaft temperature. Due to geometric constraints, the thermocouple was placed 0.7 inches from the tip of the glow plug. While this does not provide actual tip temperature, companion experiments were completed to establish a suitable correlation as will be discussed later.

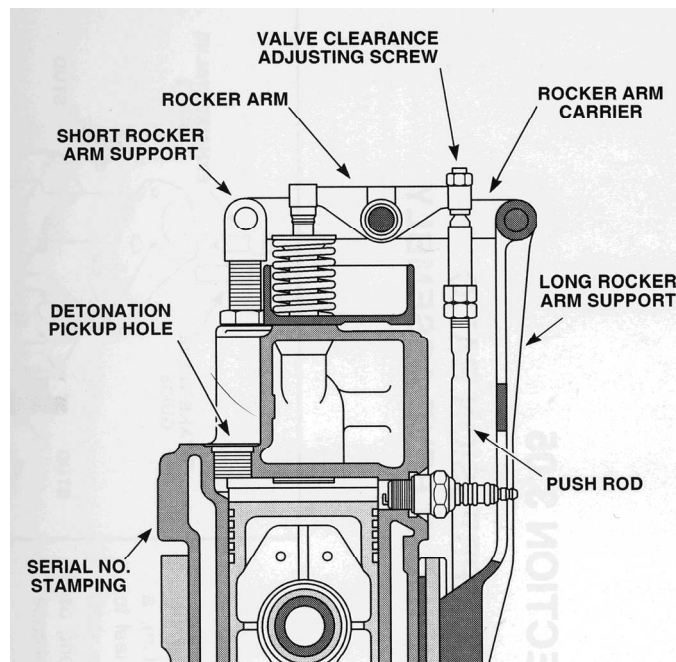


Figure 2: CFR engine detail. Pressure transducer and glow plug located in detonation pickup hole. Graphic taken from Ref [12].

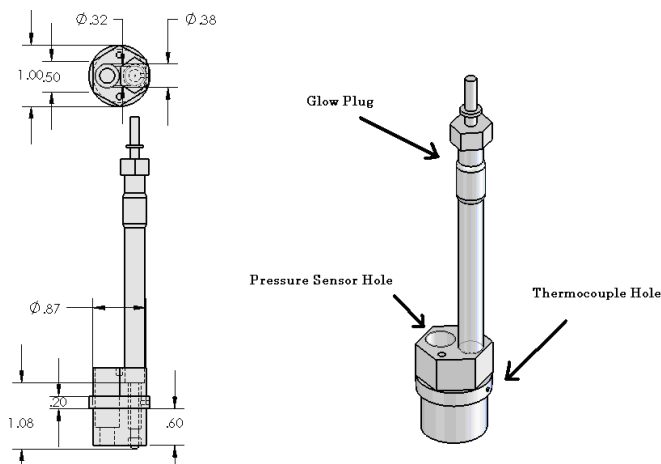


Figure 3: Glow plug drawing (dimensions in inches)

EXPERIMENTAL PROCEDURES

All tests were conducted at wide-open-throttle (WOT) with the CFR engine running at a 900 RPM. Both commercial E85 (locally tested at 84% ethanol content minimum) and pure ethanol were evaluated at CR = 12, 13.5 and 15. Spark timing was varied from 10° BTC to 5° ATC and the relative air-fuel ratio, ϕ , was swept from 0.6 to 1.2. For each experiment, the engine was stabilized prior to energizing the glow plug. Following glow plug warmup and subsequent audible detection of pre-ignition, fuel was cutoff to prevent engine damage. Data for approximately 25 cycles were collected for each test. In-cylinder pressure, glow plug temperature and UEGO were collected for a total of 90 test runs with 50 kHz data sampling.

For each test case, in-cylinder pressure and combustion phasing were used to determine the presence of pre-ignition. These data sets were evaluated using a MATLAB™ program developed by the authors to identify pre-ignition and quantify associated knock intensity. For the purposes of this study, a cycle was defined as pre-igniting if 10% of the total energy release occurred at or before 2 crank angle degrees after the normal spark event. Total energy release was calculated by summing energy lost to the cylinder walls, sensible heating of the in-cylinder charge, and expansion work performed. Energy loss to the walls was estimated using a heat transfer coefficient correlation developed by Woschni [13].

Figure 4 shows a representative pre-ignition cycle for E85 at CR = 12.0 with spark timing set at 0° BTC. In this case, 10% burn is achieved at 9° BTC. As confirmed by the pressure peak location, just prior to spark, this is a clear case of pre-ignition.

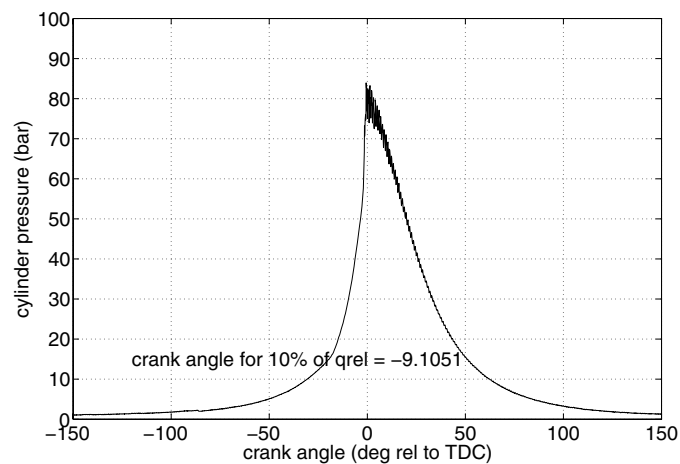


Figure 4: Pre-ignition example, E85, CR = 12, spark timing at 0° BTC

Peak to peak cylinder pressure fluctuations were used to characterize knock intensity. These fluctuations were obtained by conditioning the raw pressure data with a MATLAB™ band-pass filter. The filter used was a Chebyshev type 1 2nd order digital filter with 2 decibels peak-to-peak ripple allowed in the passband. Normalized passband frequency limits were 0.2 and 0.8 (times π rad/sample). Figure 5 shows a filtered data sample. In this example, the knock intensity of 4.68 bar refers to the maximum peak to peak pressure difference.

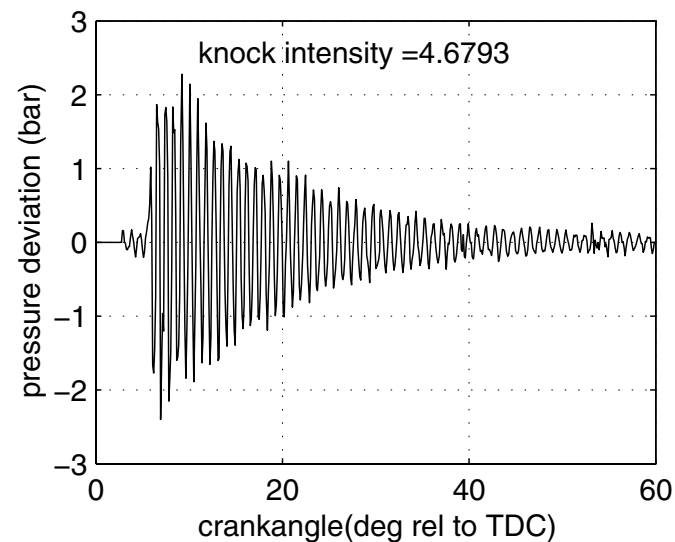


Figure 5: Definition of knock intensity by measuring maximum range of the bandpass filtered pressure.

Knock intensity (KI) provided a quantifiable indication for determining knock acceptability. KI greater than 2 bar was deemed unacceptable as it exceeded the “barely audible” threshold. KI greater than 10 bar was characterized as “very strong.”