

High Cetane Fuel Combustion Performance in a Conventional Military Diesel Engine

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Jim Cowart, Matthew Carr, Pat Caton, Lars Stoulig, Dianne Luning-Prak, Andrew Moore and Leonard Hamilton
US Naval Academy

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ABSTRACT

Synthetic diesel fuels from Fischer-Tropsch or hydrotreating processes have high cetane numbers with respect to conventional diesel fuel. This study investigates diesel combustion characteristics with these high cetane fuels. A military jet fuel (JP-5 specification), a Fischer-Tropsch (FT) synthetic diesel, and normal hexadecane (C16), a pure component fuel with defined cetane number of 100, are compared with operation of conventional military diesel fuel (F-76 specification). The fuels are tested in a AM General GEP HMMWV engine, an indirect-injection, largely mechanically-controlled diesel engine. Hundreds of thousands of these are in current use and are projected to be in service for many years to come. Experimental testing showed that satisfactory operation could be achieved across the speed-load operating map even for the highest cetane fuel (normal hexadecane). The JP-5, FT, and C16 fuels all showed later injection timing. Despite having a significantly higher cetane number, the FT fuel showed a longer ignition delay, probably due to the lower density of this synthetic fuel, which leads to slower penetration into the chamber. Thus, ignition delay was not found to correlate directly with cetane number, and fuel density effects were able to counteract the much higher cetane number of the FT fuel. Peak pressure was lower with JP-5, FT, and C16, relative to diesel, due to both differences in ignition delay and combustion duration. BMEP with the JP-5, FT, and C16 changed little relative to diesel, but was reduced with FT operation (10-20%) in the high-speed, low-load region of the operating map. BSFC was improved for JP-5 and C16 (approximately 5%), but slightly worse for FT fuel (approximately 5%) particularly in the high-speed, low-load region.

BACKGROUND AND INTRODUCTION

Fischer-Tropsch (FT) synthetic fuels have been tested extensively as a replacement for conventional diesel fuel, especially over the past two decades. FT fuel processing details as well as fuel feedstock vary with GTL (Gas To Liquid) formulations often using natural gas as a feedstock, and CTL (Coal to Liquid) formulations using abundant world-wide coal reserves to produce these liquid FT fuels. While FT fuels are an established alternative diesel engine fuel they have not reached widespread implementation due to cost challenges as compared to conventional diesel.

FT fuels are not all created equal, with differences in fuel formation resulting from processing differences. FT fuels in general are comprised principally of normal and branched alkanes. FT fuels do not have significant sulfur, aromatics, or cyclo-alkanes, which are characteristic of conventional diesel [1]. FT fuels also have different fractions of branched and normal alkanes. One study compared five commercially available FT fuels from 3 different manufactures. The resulting branched alkane composition from these five different FT fuels varied from twenty-five to nearly ninety percent with the balance being normal alkanes. The distribution of the branched alkanes from C8 to C18 also varied significantly with C10 to C14 being the approximate mean of these fuels tested [2]. Thus every FT fuel is unique in its chemical composition.

Hydrotreated Vegetable Oils (HVOs) are a newer variant of alternative fuel. HVOs use vegetable based oils that are then hydrotreated to produce a fuel that looks very similar to FT fuels [3]. The US Navy is looking into hydrotreated algae oil as a diesel fuel replacement as well as hydrotreated camelina (mustard seed) oil as a jet fuel replacement. These HVOs are

not the ester based ‘bio-diesel’ of recent years, but rather pure hydro-carbon fuels without oxygen in the fuel. One recent study of HVOs by Aatola et al. [3] shows that these fuels have very similar properties to FTs, including high cetane values.

FT fuels characteristically have a very high cetane number. One study looked at 6 GTL FTs with cetane numbers in the 70's to 80's [4]. Thus, the absence of aromatics and cyclo-alkanes clearly aids in raising the cetane number of FT fuels [5]. The question exists as to how a conventional engine will perform with very high cetane fuels characteristic of FT and HVO based alternative diesel fuels as ignition might occur too early?

Few prior studies have looked at diesel engine operation with a wide range of cetane fuels. Olree and Lenane [6] looked at combustion characteristics with diesel fuel cetane ranging from 35-55. Hardenberg and Hase [7] studied a broader cetane range up to 80, with these fuels showing early combustion.

The authors have looked at numerous FT fuels with their associated ignition delays in a single cylinder research engine. Figure 1 shows results from a prior paper in which 5 FT fuels (‘F’) are compared to diesel fuel (‘D’) as well as jet fuel (JP5 = ‘J’) [8]. The military is somewhat unique in frequently using jet fuel in its diesel engines in order to simplify fuel logistics. Diesel fuel cannot be used in jets due to its high temperature distillation range, so jet fuel is the chosen single tactical fuel. The diesel fuel ignition delay is approximately 10 crank angle degrees (at 900 rpm) which is longer than 4 of the 5 FT fuels tested. It can also be seen that jet fuel is slightly quicker to ignite as compared to diesel fuel. Also shown in Figure 1 are results from pure component fuels testing. The n-alkane fuels (displayed as open triangles in the figure) have very short ignition delays, followed by n-alkenes, and the cycloalkanes and aromatic molecules have much longer ignition delays due to a more stable molecular structure.

Viewing these data from the perspective of carbon chain length is shown in Figure 2. This figure shows that longer straight chain alkanes have the shortest ignition delay with the shortest being C16 (normal hexadecane), which is by definition a 100 cetane reference fuel.

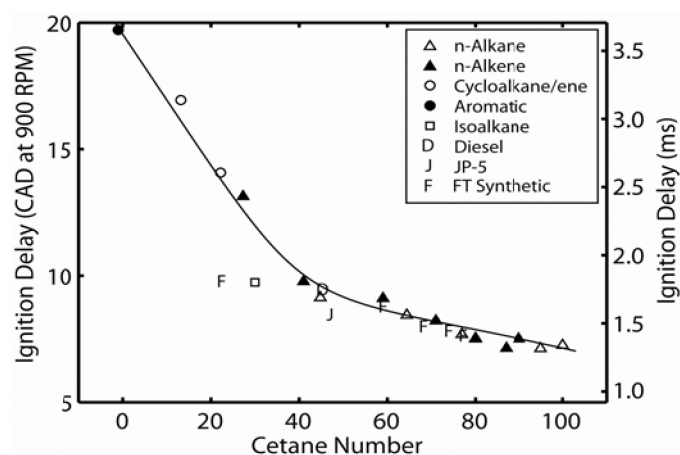


Figure 1. Ignition delay in crank angle degrees (CAD) and ms of various pure component and conventional and synthetic fuel mixtures with respect to cetane number.

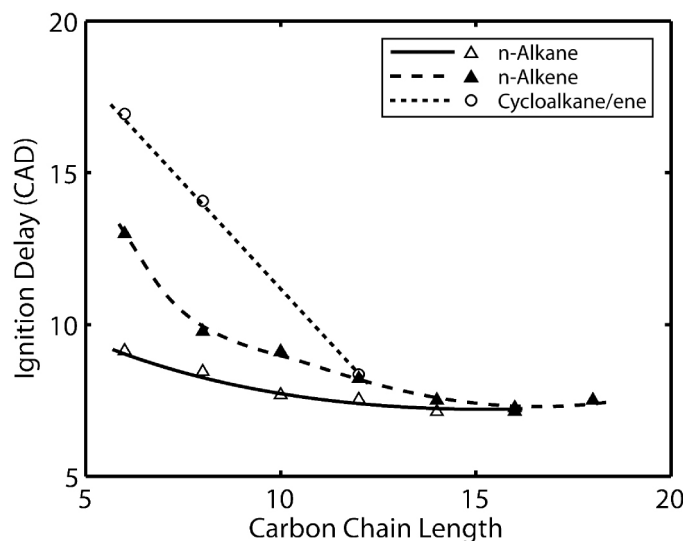


Figure 2. Ignition delay (CAD and ms) of normal alkanes, normal alkenes, and cycloalkanes/enes with respect to the chain length.

The objectives of this work are to evaluate the performance of very high cetane fuel in a current military diesel engine. Since military engines have long service lives of many decades, new fuels must be compatible with legacy engines. This study seeks to quantify the changes in combustion characteristics of very high cetane fuels, including the highest C16. In this paper, the performance characteristics of a conventional FT with relatively high cetane is compared to operation with conventional petroleum-based diesel fuel (military specification F-76), operation with jet fuel, and with high cetane (100) C16.

EXPERIMENTAL SETUP

The engine used in this study is an AM General Engine Products (GEP) 6.5 L V-8 HMMWV “Humvee” turbocharged diesel engine, manufactured in 2007. The engine is an indirect injection diesel with no electronic control in order to maximize reliability and durability in the field. It was equipped with 150-bar single hole injectors. Table 1 lists the basic specifications of this engine and the operating conditions used in this study. A Kistler™ 6054 pressure transducer was mounted in the engine head and used in conjunction with a Kistler™ 5010B charge amplifier. A Kistler™ 4046 fuel line pressure transducer was utilized to determine start of injection. Airflow was measured with a Meriam laminar flow element. A line diagram of the engine configuration is shown in Figure 3. Data were collected with a National Instruments™ Lab View data acquisition system sampling at a rate of 50 kHz.

A conventional heat release analysis modeled after MIT's Single Zone approach [5, 9, 10] was used to further analyze the engine cylinder pressure data. This single zone model uses the first law of thermodynamics with combined unburned and burned single zone average properties to determine the rate of the fuels' energy release. Time domain sampling was converted to crank angle degrees and engine wall heat transfer was accomplished with the conventional instantaneous spatially averaged Woschni coefficient. It was of interest to see if significant variations in burn rate were observed with the different fuels.

Table 1. Experimental engine parameters.

Type	Indirect-injection diesel
Bore	103 mm
Stroke	97 mm
Maximum Power	142 kW at 3600 RPM
Maximum Torque	522 Nm at 1800 RPM
Compression Ratio	21
Coolant	water cooled
Intake Air	turbocharged, 0 – 7 psig boost
Fuels	normal hexadecane (C ₁₆ H ₃₄) U.S. Navy conventional jet fuel (JP-5) U.S. Navy/Marine Corps. conventional diesel fuel (F-76) synthetic diesel (FT)

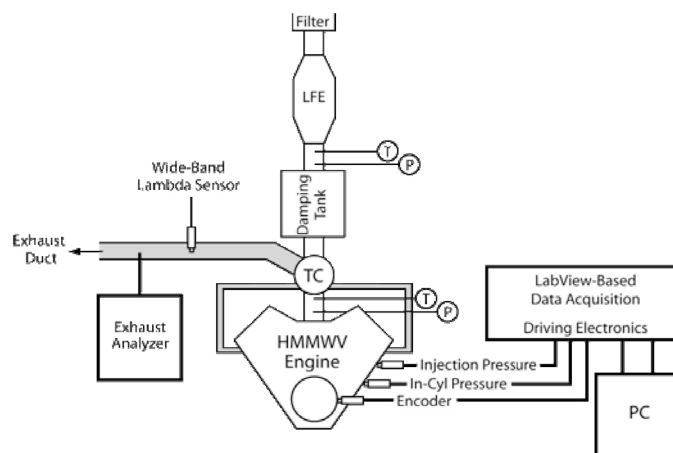


Figure 3. HMMWV engine experimental setup.

Exhaust gas was analyzed using a steady-state analyzer system and sampled from the exhaust flow using a heated filter approximately 10 cm downstream of the exhaust port. Hydrocarbons (HC) were measured on a wet basis using a CAI™ model 600 heated flame ionization detector (HFID). Oxygen (O₂), carbon dioxide (CO₂), and carbon monoxide (CO) were measured with a CAI™ model 602P combined non-dispersive infrared absorption (NDIR) and paramagnetic sensor. Nitric oxide (NO) emissions were measured using a CAI™ model 600 heated chemiluminescent analyzer on a dry basis. An ETAS LA4.2 wide-band oxygen sensor was used to measure the air-fuel ratio at all operating conditions; combined with the air-flow measurement, this measurement allowed calculation of the fuel flow rate.

Table 2 lists the fuels used in this study with some selected properties. The liquid fuel density and kinematic viscosity of all fuels in this study were measured in-house using a SVM 3000 Stabinger™ Viscometer (Anton Paar) at 20.00°C. For each fuel, at least two samples were analyzed. The accuracy of the measurement system was tested with a certified viscosity reference standard (Standard 83, Cannon Instrument Company). The percent difference between the measured and actual values at 20.00°C was less than 1.4% for viscosity and less than 1% for density. The precision of the instrument was determined from duplicate samples, and the percent deviation from the mean was found to be less than 1%. A Kruss™ DS100 drop shape analyzer was used to measure the liquid-air surface tension of the fuels. In this analyzer, a droplet of the fuel is formed in the air with a micro syringe. The system then takes an image of the droplet, enlarges it, and analyzes the droplet shape by fitting it with the Young-LaPlace equation.

Table 2. Fuels utilized and selected properties at 20°C.

Class and Formula		surface tension (mN/m)	kinematic viscosity (mm ² /s)	density (kg/m ³)	cetane	LHV (MJ/kg)	Notes
Pure Component Fuels							
n-hexadecane	(C ₁₆ H ₃₄)	26.3	4.47 ± 0.003	774	100	43.9	1,2
Conventional Fuels							
JP-5	C _{12.2}	24.9	1.85 ± 0.001	800	46.2	43.2	1,2,3
F-76	C _{13.5}	26.9	3.94 ± 0.005	856	44	42.6	1,2,3
Synthetic Fuels							
FT #2266	C _{11.6}	N/A	4.503	753	75	43.9	1,2,3

1 Accuracy specifications for surface tension are ±0.1 mN/m for all measurements.

2 Accuracy specifications for density are ±0.1 kg/m³ for all measurements.

3 Estimation of carbon number based on measured hydrogen content and estimation of molecular weight based on interpolation between similar, known blends or from correlations of boiling point data. Procedure described in Ref [8] and has been experimentally verified for all fuels tested. Cetane number measurements using ASTM D976 [11] specification from Fuels Division of Naval Air Systems Command, Patuxent River, MD.

EXPERIMENTAL METHOD

Data were collected at approximately two dozen speed-load points. Throughout the paper, these data are typically reported as contour lines on a plane of fuel-air equivalence ratio (Φ) plotted on the horizontal axis with the engine speed on the vertical axis. Note that the equivalence ratio is proportional to the engine's load, so approximately, the x-axis represents engine load. The dependent variable of interest will be shown in contour form.

To conduct the tests, the fuel must be switched in a relatively large engine. After testing with an initial fuel, the fuel system was drained, refilled with the next test fuel, and operated until fuel was pushed completely through the engine. This “flushing fuel” was discarded and then testing continued by refilling the engine fuel tank with 5 gallons of new test fuel. Data were collected once the engine was allowed to warm up to 170°F ± 2°F.

RESULTS

START OF INJECTION (SOI)

The actual SOI as measured by the rapid rise in fuel line pressure (followed by injector opening) is shown for the base diesel (F76) in Figure 4. SOI occurs from 9 to 12 degrees BTC across the baseline map with slightly later (retarded) injection times occurring at higher engine speeds and increased equivalence ratios. This slightly later actual SOI is a characteristic of the compressibility of the fuel taking a short, but significant real amount of time to reach injector opening pressures, in conjunction with modest mechanical timing changes affected by engine speed. The mechanical start of injection timing also has modest throttle based

adjustment. The stock GEP pump settings were fixed for all of the testing.

The measured SOI timings for JP-5, FT and C16 are show in Figure 5, Figure 6 and Figure 7, respectively. In these figures, the SOI is shown as relative to conventional diesel operation. Negative numbers indicate later, or retarded, SOI events. For all three alternative fuels, SOI occurs slightly later (retarded) by 0.5 to 1.5 degrees. This effect is believed due to the lower densities of each of these three alternative fuels, with specific gravities of 0.80, 0.75 and 0.77, for JP5, FT and C16. The base diesel fuel has a specific gravity of 0.86.

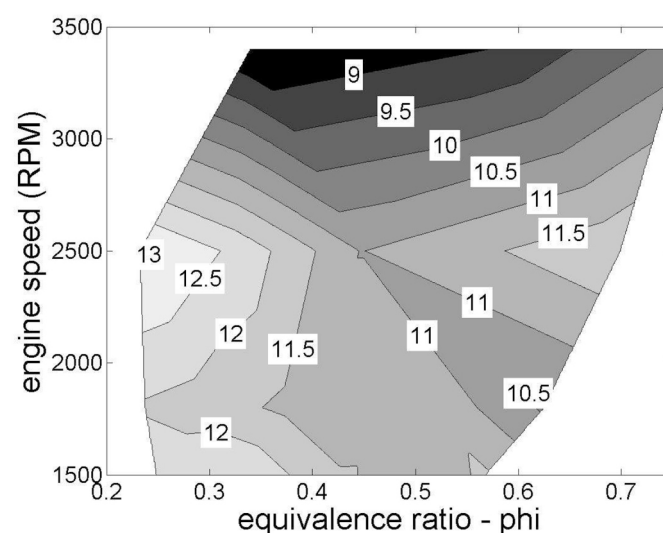


Figure 4. Start of injection (°BTC) for operation with conventional diesel (F-76).

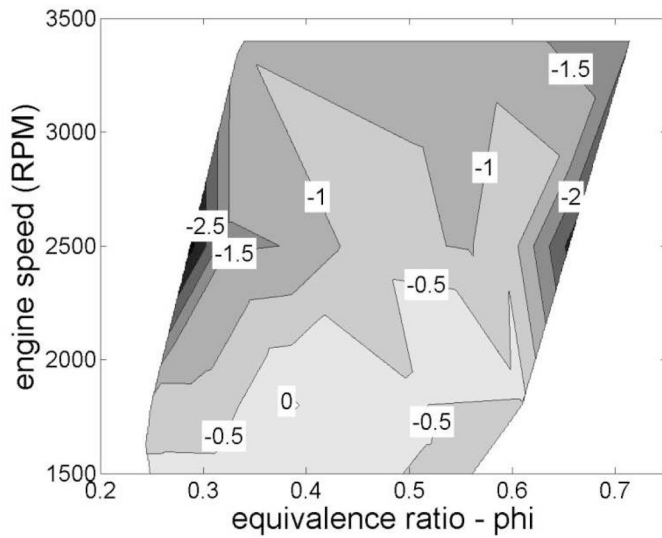


Figure 5. Start of injection for operation with JP-5, relative to conventional diesel (CAD).

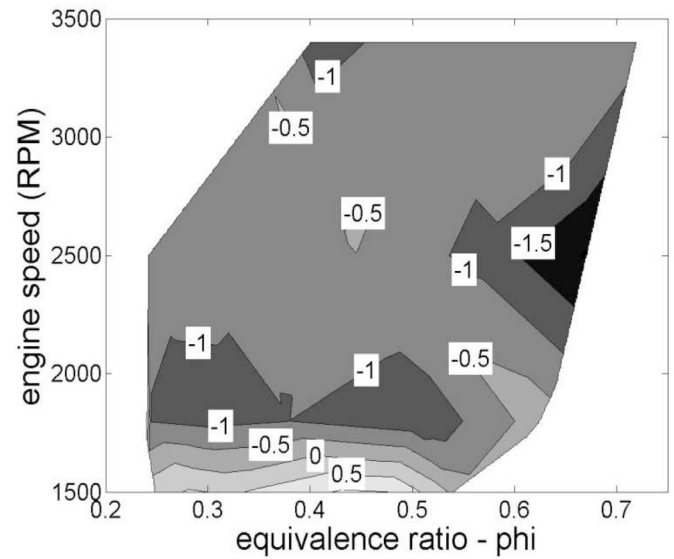


Figure 7. Start of injection for operation with C16, relative to conventional diesel (CAD).

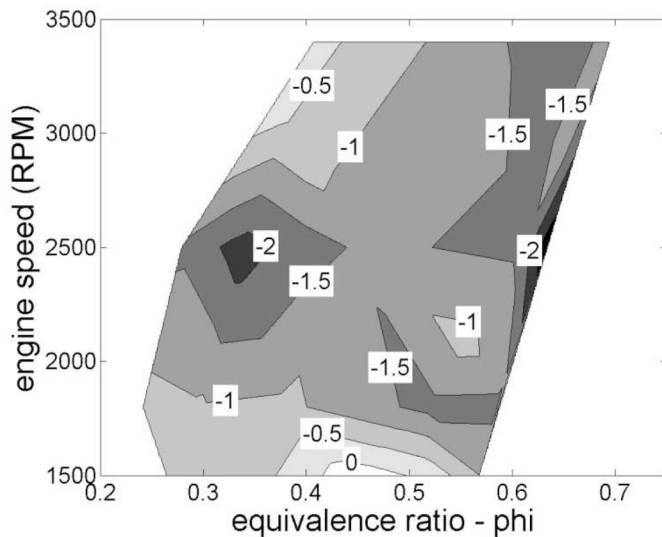


Figure 6. Start of injection for operation with FT, relative to conventional diesel (CAD).

IGNITION DELAY (IGD)

The IGD map for the HMMWV engine operating on diesel fuel is shown in [Figure 8](#). Ignition delay was calculated in this study as the difference in time from SOI to the 10% mass fraction burned point (CAD10). [Figure 8](#) shows the IGD data in units of time (msec) while [Figure 9](#) shows the same data in terms of crank angle degrees. For this engine, IGD is from 0.7 to 1.3 msec in duration. The ignition delay decreases by approximately 50% as the engine speed is doubled. This effect is due principally to increased turbulence in the pre-chamber of this engine as well as less charge thermal losses per cycle at higher engine speeds. In terms of crank angle degrees, the IGD does not change much as a function of engine speed, and only modestly increases with increased ϕ or load. This load effect could be due to increased charge cooling with increased fuel injection amount.