

Impact of Cetane Number on Combustion of a Gasoline-Diesel Dual-Fuel Heavy-Duty Multi-Cylinder Engine

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ABSTRACT

Dual-fuel combustion using liquid fuels with differing reactivity has been shown to achieve low-temperature combustion with moderate peak pressure rise rates, low soot and NO_x emissions, and high indicated efficiency. Varying fractions of gasoline-type and diesel-type fuels enable operation across a range of low- and mid-load operating conditions. Expanding the operating range to cover the full operating range of a heavy-duty diesel engine, while maintaining the efficiency and emissions benefits, is a key objective. With dissimilar properties of the two utilized fuels lying at the heart of the dual-fuel concept, a tool for enabling this load range expansion is altering the properties of the two test fuels - this study focuses on altering the reactivity of the diesel fuel component.

Tests were conducted on a 13L six-cylinder heavy-duty diesel engine modified to run dual-fuel combustion with port gasoline injection to supplement the direct diesel injection. A single diesel injection strategy was used across a range of operating points. Diesel fuels with varying cetane number were tested, including a ULSD certification fuel; a low cetane, low aromatics, high volatility fuel from the Fuels for Advanced Combustion Engines matrix (FACE #1); and a high cetane Fischer-Tropsch diesel fuel.

Decreasing diesel fuel cetane number reduces overall mixture reactivity and extends the mixing time prior to onset of combustion, reducing soot emissions and shifting the overall operating window to a higher load range. Stable operation with high gasoline fractions across a wider load range is enabled by the high cetane fuel.

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INTRODUCTION

Strict new standards for CO₂ emissions and fuel consumption of heavy-duty engines and vehicles [1], coupled with the business interests of heavy-duty vehicle operators to reduce total fuel operating costs, drive an increased push for high-efficiency heavy-duty motive power. Stringent emissions standards also mandate low system-out (including engine and aftertreatment) levels of gaseous (including NO_x and NMHC) and particulate emissions [2]. Reflecting the need for higher efficiency heavy-duty engines, key objectives of the Department of Energy (DOE) SuperTruck program are the demonstration of a heavy-duty engine with 50% brake thermal efficiency and establishment of a pathway to 55% brake thermal efficiency [3]. Dual-Fuel combustion was identified as a possible enabling technology for high efficiency while also maintaining low engine-out emissions levels. Many previously developed low-temperature diesel combustion strategies produce low engine-out emissions but frequently at the penalty

of lower engine efficiency. However, increasingly effective aftertreatment systems for heavy-duty diesel engines have enabled higher overall engine efficiency [4] - combustion concepts designed to produce low engine-out emissions must therefore also deliver comparable efficiencies to remain competitive. Dual-Fuel combustion concepts were developed as a method of addressing the limitations of earlier low-temperature combustion strategies while providing high engine efficiency.

By introducing two fuels of differing reactivity into the combustion process separately, overall fuel reactivity levels can be adjusted with operating conditions. Further, separating the fuel delivery, port-injecting the low reactivity fuel and direct injecting the high reactivity fuel, has been shown to create in-cylinder reactivity stratification, slowing combustion and decreasing the pressure rise rate, a primary goal in the path for many HCCI type combustion strategies [5,6]. Dual-Fuel

strategies using gasoline and diesel have shown potential for high efficiency, both quantified by indicated thermal efficiency measurements on single-cylinder engines [5,6] and by brake thermal efficiency measurements on multi-cylinder engines [7,8]. By using a low compression ratio, it has been shown that a dual-fuel concept can cover the entire operating range of a heavy-duty engine [9]; efficiency with the low compression ratio, however, may not offer advantages over current single-fuel conventional diesel engines.

Different variations on the dual-fuel concept have been explored, with key differences being the delivery conditions of the diesel injection. The Reactivity Controlled Compression Ignition (RCCI) concept uses very early injection timing with relatively low injection pressure - start of combustion and subsequent combustion phasing is largely kinetically driven [5,6]. A similar concept, denoted as 'CM2' has also been reported in literature [8,10,11]. However, by using a more conventionally timed, high pressure, single diesel injection, greater control over the combustion phasing can be achieved and, depending on the thermal boundary conditions, potentially an expansion of the upper load limit of the operating range [7,8].

Impact of fuel properties on dual-fuel combustion has been explored primarily on the low-reactivity (gasoline-like) component. Using high-octane ethanol blends, with their lower reactivity relative to gasoline, enables higher efficiencies and expanded operating range [8,12,13]. Investigation of how the properties of the high reactivity (diesel-like) fuel component impact dual-fuel combustion modes has been limited to the exploration of using gasoline with a cetane improver for the high-reactivity component to implement a single fuel with additive strategy [14,15].

Building off prior work on a gasoline-diesel dual-fuel strategy employed on a multi-cylinder, heavy-duty diesel engine [8], this study explores the combustion and emissions impact of varying the cetane number of the diesel fuel. This work is undertaken with a specific focus of expanding the operating range and improving maximum efficiency.

EXPERIMENTAL DETAILS

Equipment

The test engine used in this study is a MY2010 Navistar MaxxForce 13 six-cylinder heavy-duty diesel engine. A schematic of the test engine configuration is shown in Figure 1, along with basic engine specifications in Table 1. The base engine features an air system configured for high EGR delivery, utilizing two-stage turbocharging (with interstage cooling) and a dual-pass EGR cooler with high and low temperature stages. The stock 2200 bar capable common-rail diesel fuel injection system is maintained.

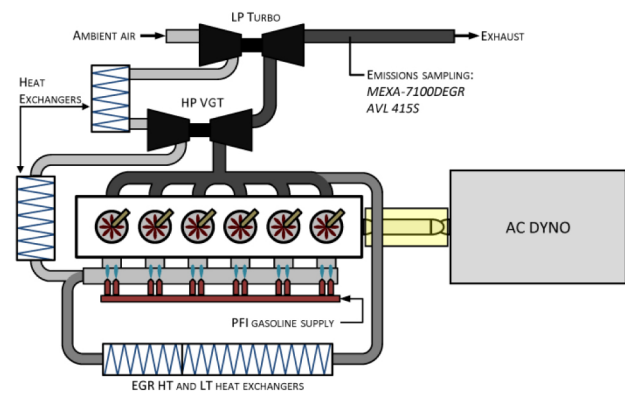


Figure 1. Schematic of test engine configuration.

Table 1. Engine Specifications.

Navistar 13L DF	
Cylinders [-]	6
Bore [mm]	126
Stroke [mm]	166
Displacement [L]	12.4
Compression Ratio [-]	14

Several alterations to the production engine were made to enable/enhance low-temperature dual-fuel combustion. Pistons with modified bowl geometry yield a reduced (14:1) compression ratio, reducing process temperatures and pressures, thereby suppressing tendency for early autoignition [11]. A rematched set of turbochargers, including a variable geometry (VGT) high-pressure turbocharger, was implemented to increase the EGR driving capacity. EGR is used to reduce combustion temperatures for lower NO_x emissions levels, to mitigate high peak pressure rise rates, and to increase the ignition delay after diesel injection. A modified intake manifold is used which incorporates two port fuel injection (PFI) gasoline injectors per cylinder. The gasoline fuel is injected at 6 bar above intake manifold pressure directed at the intake runner of each cylinder. Gasoline is supplied by an externally mounted, electrically-driven pump; energy consumed for gasoline fuel delivery is negligible and therefore not included in brake thermal efficiency calculations.

Stock EGR and interstage intake air cooling systems are maintained. A building process water system is used for cooling heat exchangers in the high and low temperature engine coolant loops, and to directly cool the high pressure charge air cooler. Cooling systems were tuned using Navistar internal metrics to mimic heat rejection available in a production vehicle. Intake and exhaust restrictions were likewise adjusted to simulate vehicle air filtration systems along with exhaust aftertreatment and muffler restrictions. Constant restrictions were used across all tested conditions.

Fuel flow, both gasoline and diesel, is measured with Coriolis-type flowmeters. Low speed measurements of engine operating parameters, including temperatures, pressures,

emissions, and fuel flow are logged at 10 Hz. Crank-angle resolved measurements of cylinder pressure are collected using Kistler 6125B transducers in all six cylinders at 0.1 crank angle degree resolution for 500 engine cycles. A 3 kHz low-pass filter is used to suppress significant cylinder pressure oscillation due to chamber resonance.

Start of injection timing, both as a reported value and used in computation of ignition delay, is determined from ECU command and not directly measured from injector current.

Gaseous emissions are measured using a standard 5-gas analyzer (Horiba MEXA-7100DEGR), which includes CO₂ measurements in the intake for EGR calculation. An AVL 415 Smoke Meter is used for soot emissions measurements.

Test Fuels

The dual-fuel operating strategy employed in this study uses two fuels concurrently: a low-reactivity fuel (gasoline) along with a high-reactivity fuel (diesel). The gasoline used in this study was EEE certification gasoline; relevant properties are listed in Table 2, along with properties of the other test fuels.

Table 2. Test Fuel Properties.

	Gasoline	ULSD	LTFT	FACE #1
Density [kg/L]	0.743	0.851	0.773	0.808
NetHV [MJ/kg]	42.5	42.8	43.7	42.8 ¹
RON [-]	97	-	-	-
MON [-]	89	-	-	-
CN [-]	-	44	>73	30
Aromatics [%]	32	28	<0.1	22
Olefins [%]	1	1	<0.1	4
Saturates [%]	68	71	>99.9	74
C:H Ratio	1.87	1.88	2.10	1.90
Sulfur [ppm]	28	10	1	2
T10 [°C]	50	214	213	156
T50 [°C]	104	259	271	190
T90 [°C]	158	317	347	269

¹FACE #1 net heating value from [16]

Three different diesel fuels were used, covering a range of autoignition characteristics, as highlighted by the spread in cetane numbers. The reference diesel was an ultra-low sulfur (ULSD) certification diesel fuel sourced from Haltermann with typical diesel fuel properties and a cetane number of 44. A low cetane diesel fuel was also tested, drawn from the Fuels for Advanced Combustion Engines (FACE) matrix of test fuels [16,17]. The FACE fuels cover high/low points on a matrix of cetane number, aromatic content, and volatility (represented by distillation temperature T90). FACE #1 is the low cetane number (30 CN), low-aromatics (20%), high-volatility (270°C T90) fuel. This fuel was selected because the low reactivity (low CN) should increase the ignition delay between diesel

injection and the start of combustion to potentially reduce soot formation, lower aromatic content is preferable for a reduction in polyaromatic soot precursor formation and decreased CO and unburned hydrocarbon levels, and the higher volatility enhances fuel vaporization and mixing [18, 19, 20, 21]. For the other extreme in cetane number, a high cetane number gas-to-liquid synthetic diesel fuel from Sasol was utilized. This low-temperature Fischer-Tropsch (LTFT) fuel has negligible aromatic content and a cetane number in excess of the ASTM test limits (>73) [22]. The high reactivity of the LTFT fuel should provide a robust ignition source while the negligible aromatic content may reduce soot formation.

Operating Conditions

Testing was conducted across a range of load conditions, detailed in Table 3, at a fixed speed of 1200 rpm. Important to note is that dual-fuel combustion within specified operating limits and combustion targets was not achieved at all load conditions for all test fuels - further discussion is highlighted in relevant results sections.

Table 3. Operating Conditions.

Speed [rpm]	BMEP [bar]	Engine Load [%]
1200	5.8	25
1200	8.2	35
1200	10.3	45
1200	11.6	50
1200	12.6	55

At each operating point, using each fuel combination, dual-fuel combustion was optimized for highest brake efficiency subject to a set of operating targets and limits. These targets are set to maintain reasonable bounds on the operation. Based on hardware limitations, cylinder pressure is limited to 220 bar while maximum rate of pressure rise (for each cylinder, measured on filtered cylinder pressure signal) is limited to less than 15 bar/degree. This limit is established as an outer boundary based on engine hardware durability - NVH concerns (combustion noise) is not the critical factor. NO_x emissions are limited to less than 0.27 g/kW-hr (0.20 g/hp-hr) based on EPA 2010 emissions standards for heavy-duty onroad engines. Soot emissions are targeted at less than 0.20 FSN, approximately corresponding to brake specific soot emissions of 0.01 g/kW-hr, the limit established in EPA 2010 emissions standards. This is a very restrictive target, one which may not be achievable at certain operating conditions. Reflecting this reality, and that some particulate aftertreatment may be required in a practical implementation, the soot target is relaxed at certain conditions. Both scenarios, with and without the 0.20 FSN soot target, are examined. Explicit limits are not placed on CO or HC emissions, though they are evaluated. Stable operation was maintained at each operating point, with COV of IMEP less than 5% for all tested conditions.

Due to the lengthy time required for complete fuel purging, the different diesel fuels were not tested within the same day. However, ambient air conditions including humidity were comparable between all tests.

Dual-Fuel Combustion Strategy

The dual-fuel combustion mode splits the fuel introduction between two fuels of differing reactivity: a low-reactivity gasoline fuel (introduced via port injection during the intake stroke) and a high-reactivity diesel fuel (introduced via direct injection into the cylinder during the compression stroke). Fifty to ninety percent of the delivered fuel mass is gasoline, with the exact split in fuel delivery dependent on load condition. A schematic of typical gasoline and diesel injection timings as related to a pressure trace is included in Figure 2.

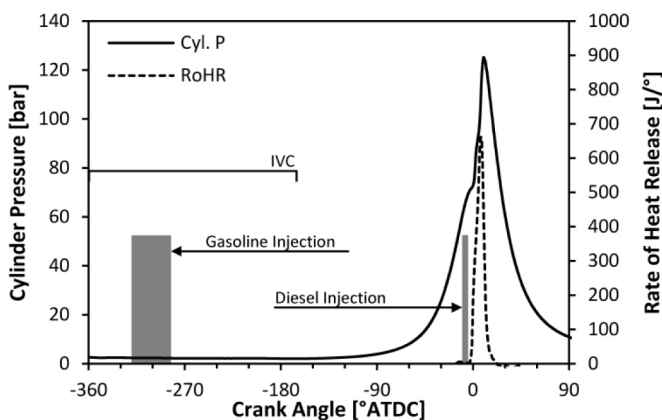


Figure 2. Representative dual-fuel combustion. 1200 rpm, 10.3 bar BMEP, gasoline with ULSD reference fuel.

The single gasoline port fuel injection occurs during the intake stroke, with a constant start of injection at 320°BTDC for all test cases. Gasoline injection duration covers the range of 3-6.6 ms (21-47°CA). Varying the start of the gasoline injection, across a range of 300-400°BTDC, did not measurably alter combustion or emissions behavior.

A single diesel injection strategy is used, with the injection phased close in proximity to top dead center (0-20°BTDC). This differs in strategy from notable previous works [5, 6] and is comparable to the “CM1” mode used in [8]. The timing of the diesel injection is used to control combustion phasing: combustion operates in a regime where diesel SOI and combustion phasing are closely coupled. Maintaining this direct relationship offers enhanced combustion control. Further, early phased diesel injections, whether single or multiple injections, are limited to lower load ranges than the single-injection strategy due to early autoignition. Reducing process temperatures may mitigate the early autoignition limitation but, as noted before, representative vehicle cooling levels were used in this study.

Cooled exhaust gas recirculation (EGR) was used for all tested operating conditions, its fraction increasing with load. Air dilute combustion strategies, though successfully employed in previous research at low-load conditions [5,12] were not considered here. EGR quantity varies across load conditions and fuels - the lowest possible EGR required to achieve the targeted NOx emissions and peak pressure rise rate (PPRR) levels was used to minimize efficiency losses from pumping work. Increasing EGR with increasing load is used to reduce charge reactivity, allowing greater time for diesel injection mixing and slowing combustion to limit peak pressure rise rate.

BASELINE OPERATING BEHAVIOR

Within this study, combustion at each load condition, and with each set of test fuels, is optimized to yield highest brake thermal efficiency. Accordingly, there is significant variation in combustion characteristics across the tested conditions. A base understanding of combustion behavior with a single fuel combination is necessary for complete comprehension of combustion changes resulting from fuel variations.

Combustion Characteristics

Varying the split in fuel fraction between diesel and gasoline fuel is a fundamental element of the dual-fuel strategy - the concept itself grew from the understanding of a varying ideal fuel reactivity dependent on operating load [5]. Within the present study, engine operation is optimized at each load condition for maximum efficiency subject to the operating limitations laid out prior. Accordingly, there is significant variation in engine control parameters, and therefore combustion behavior, across the range of tested conditions. Cylinder pressure and heat release characteristics for the number 1 (front-most, firing order one) cylinder of a baseline load sweep at 1200 rpm with certification gasoline and a ULSD certification diesel fuel are shown in Figure 3, with relevant operating parameters in Table 4. Traces are 500-cycle averages.

Immediately apparent in Figure 3 is the increase in peak compression pressure as load increases due to the increase in intake manifold pressure provided by the multi-stage turbocharger air system. Absolute intake manifold pressure rises from 1.6 bar to 2.2 bar as load covers the range shown. Combustion, and consequently combustion pressure rise and peak pressure location, retards as load increases, reflecting the required retard in diesel injection timing. This is driven both in response to the change in heat release characteristics, as discussed further below, and to maintain peak pressure rise rate within the specified operating boundary (<15 bar/degree).

Evolution of the heat release across the load range is also highlighted in Figure 3. At the lowest load, with the lowest gasoline fuel energy fraction (68%), initial heat release is