

Development of Dual-Fuel Low Temperature Combustion Strategy in a Multi-Cylinder Heavy-Duty Compression Ignition Engine Using Conventional and Alternative Fuels

Yu Zhang, Ilya Sagalovich, and William De Ojeda
Navistar

Andrew Ickes and Thomas Wallner
Argonne National Laboratory

David D. Wickman
Wisconsin Engine Research Consultants

ABSTRACT

Low temperature combustion through in-cylinder blending of fuels with different reactivity offers the potential to improve engine efficiency while yielding low engine-out NO_x and soot emissions. A Navistar MaxxForce 13 heavy-duty compression ignition engine was modified to run with two separate fuel systems, aiming to utilize fuel reactivity to demonstrate a technical path towards high engine efficiency.

The dual-fuel engine has a geometric compression ratio of 14 and uses sequential, multi-port-injection of a low reactivity fuel in combination with in-cylinder direct injection of diesel. Through control of in-cylinder charge reactivity and reactivity stratification, the engine combustion process can be tailored towards high efficiency and low engine-out emissions.

Engine testing was conducted at 1200 rpm over a load sweep. In addition to conventional gasoline and diesel, a blend of ethanol in gasoline on a level of 85% by volume (E85) was also investigated to examine the impact of reduced charge reactivity and enhanced reactivity stratification on load extension and engine efficiency. At each test point, engine operation was optimized for best brake thermal efficiency (BTE) within the constraints of NO_x < 0.2 g/bhp-hr and pressure rise rate < 15 bar/deg, by varying diesel injection strategy, percent of fuel delivered through port fuel injection (PFI%), and air system operating conditions. Using gasoline and diesel, dual-fuel low temperature combustion (LTC) operation reached 11.6 bar BMEP with NO_x < 0.2 g/bhp-hr and a best brake thermal efficiency (BTE) of 43.6%. The use of E85 further extended LTC operation to 19 bar BMEP (21.7 bar gIMEP) with a best BTE of 45.1%. When using E85, throughout the test range, NO_x was below 0.2 g/bhp-hr and smoke was below 0.2 FSN.

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INTRODUCTION

The increasingly stringent regulations for vehicle exhaust emissions and the rising cost of petroleum-derived transportation fuels have motivated the development of high efficiency, clean internal combustion (IC) engines. With the intent to attain low engine-out emissions while achieving high fuel efficiency, various combustion strategies, such as

homogeneous charge compression ignition (HCCI) [1, 2, 3, 4] and partially premixed combustion (PPC) [5, 6, 7, 8], have been proposed and developed in recent years.

The essence of these combustion strategies is the enabling of low temperature, premixed combustion. Through modulation of the pressure and temperature of in-cylinder charge as well as fuel ignition propensity, simultaneous reduction in NO_x and soot emissions can be attained. One

major challenge for these combustion strategies is the excessively high pressure rise rate associated with premixed combustion. Therefore, these combustion strategies can be operated only within a limited load range. To address this issue, researchers have made efforts to enhance the stratification of in-cylinder temperature and fuel concentration. However, despite these efforts, controlling the pressure rise rate remains as the main challenge that prevents the practical application of these low temperature combustion strategies.

In the past few years, a reactivity-controlled compression ignition (RCCI) concept has been developed [9, 10, 11, 12, 13, 14, 15, 16, 17, 18, 19]. A unique feature of this concept is the introduction of fuel-reactivity stratification across the engine combustion chamber. This is typically achieved by combining in-cylinder direct injection of a high reactivity fuel with port-injection of a low reactivity fuel. As a result, combustion proceeds sequentially from the high reactivity regions to the low reactivity regions. Pressure rise rate is therefore effectively lowered. Moreover, by manipulating the ratio between the high reactivity fuel and the low reactivity fuel, RCCI offers more flexibility in tailoring the combustion process as compared to the other low temperature combustion strategies.

Splitter et al. [14] have recently demonstrated RCCI operation in a single-cylinder compression ignition engine that has a geometric compression ratio of 16 coupled with intake valve closing (IVC) modulation. Through in-cylinder blending of gasoline and diesel, RCCI operation achieved up to 14.5 bar gIMEP at an EGR rate of 57%. A peak gross indicated thermal efficiency of 55% was attained, while engine-out NO_x and soot emissions were below US EPA 2010 heavy-duty standards. Further load extension was enabled by using ethanol to substitute gasoline to lower charge reactivity and enhance reactivity stratification. With the use of E85 and diesel, RCCI operation was extended to 16.5 bar gIMEP. More recently, by reducing compression ratio to 11.7, Dempsey and Reitz [16] showed RCCI operation up to 23 bar gIMEP through combustion simulation using KIVACHEMKIN coupled to a multi-objective genetic algorithm. Engine thermal efficiency, however, was compromised due to the low compression ratio.

The current study is part of our on-going effort to develop a viable dual-fuel low temperature combustion strategy for heavy-duty applications. Prior to the experimental investigation, computational efforts helped identify a proper compression ratio, suitable fueling strategy, and recommended air system targets [20].

In this study, dual-fuel combustion strategy development was conducted using a multi-cylinder heavy-duty compression ignition engine with a geometric compression ratio of 14. In addition to the conventional gasoline and diesel, a blend of ethanol in gasoline on a level of 85% by volume (E85) was also studied as a gasoline replacement to examine the impact of reduced fuel reactivity and enhanced reactivity stratification on load extension and engine

performance. Engine operating parameters including fuel injection strategy, the ratio between low reactivity fuel and high reactivity fuel, and air system operating conditions were optimized for best engine brake thermal efficiency (BTE) while maintaining NO_x below 0.2g/bhp-hr and pressure rise rate below 15 bar/deg.

EXPERIMENTAL SETUP

Engine and Lab Instrumentation

The engine used in this study was based on 2010 Navistar MaxxForce 13. Several modifications were made to the base engine for dual-fuel operation: (1) engine geometric compression ratio was reduced from 17.2 to 14 to allow for better control over the temperature and pressure of in-cylinder charge. The drop in compression ratio was achieved by scaling up the baseline piston bowl; (2) engine intake manifold was modified to accommodate a sequential, multi-port-fuel injection (PFI) system with two PFI injectors for each engine cylinder; (3) a dedicated dual-fuel control system was developed to synchronize and control both port-fuel injection and in-cylinder direct injection; (4) the production 2-stage fixed geometry turbocharger system was replaced by a 2-stage variable geometry turbocharger (VGT) system with variable vanes incorporated on the high pressure turbine. Engine specifications are listed in Table 1.

Table 1. Engine specifications

Displacement Volume	12.4 L
Number of Cylinders	6
Stroke	166 mm
Bore	126 mm
Connecting Rod	251 mm
Compression Ratio	14
Diesel Fuel System	2200 bar common rail
Port Fuel Injection System	multi-port, manifold injection
Air System	2-stage turbocharger (variable vanes on the high pressure turbine) 2-stage high-pressure EGR cooling inter-stage cooler charge-air cooler

Cylinder pressure measurements were attained using Kistler 6125B pressure transducers in all six engine cylinder at a resolution of 0.1 CA°. Gaseous emissions, including CO, CO₂, NO_x, O₂, and unburned hydrocarbons, were measured using a Horiba MEXA 7100DEGR emissions bench. Intake CO₂ concentration was also measured for EGR calculation. Engine-out soot emissions were measured in terms of filter smoke number (FSN) through an AVL 415 Smoke Meter. Engine cooling system and air system restrictions were tuned

to be representative of the conditions encountered in production vehicles.

Fuels

Emissions certification ultra low sulfur diesel and Tier II EEE gasoline were used in the present study. The EEE gasoline has no ethanol content. E85 was prepared by splash blending between ethanol and EEE gasoline with an ethanol content level of ~85% by volume. Properties of the test fuels are detailed in Table 2.

Table 2. Properties of test fuels

	Diesel	Gasoline	E85
Density [kg/m ³]	0.85	0.74	0.78*
Lower Heating Value [MJ/kg]	42.8	42.5	29
Cetane Number [-]	44.2	-	-
RON [-]	-	97	104
MON [-]	-	89	-
H/C Ratio [-]	1.88	1.87	2.8
Sulfur Content [ppm]	10	28	4*

* calculated values

Test Conditions

In the present study, port fuel injection started during the intake stroke, after intake valve opening (IVO) and stopped before the piston reached bottom dead center (BDC). A pressure difference of ~7 bar across the PFI injectors was maintained. As detailed below, two different combustion modes were explored in terms of the strategy of diesel in-cylinder direct injection (DI).

- Combustion Mode 1 (CM1): diesel injection occurs near top dead center (TDC) and serves as an ignition source. Combustion is initiated through the combustion of diesel and entrained gasoline/E85. This approach offers robust control over start of combustion and provides good combustion stability.
- Combustion Mode 2 (CM2): diesel injection starts appreciably earlier than in CM1 during the compression stroke. Combustion process is driven by the reactivity and temperature distribution of in-cylinder charge. This approach essentially follows the concept of RCCI.

Dual-fuel operation was studied at 1200 rpm by sweeping the engine load from 5.5 bar BMEP to the highest possible load. The key criteria for dual-fuel operation were: (1) NO_x < 0.2 g/bhp-hr; (2) maximum pressure rise rate (MPRR) < 15 bar/deg. Engine-out soot level was targeted below 0.2 FSN. The target soot level was relaxed for gasoline-diesel CM1 operation to extend load limit. The key engine operating boundary conditions are provided in Table 3.

Table 3. Key operating boundary conditions

Engine Speed: 1200 rpm			
Fuels	Gasoline-Diesel		E85-Diesel
Combustion Mode	CM1	CM2	CM1
EGR Rate [%]	44...46	44...47	41...43
AFR	15...18	15...19	16...19
PFI% [%]	76...86	71...93	51...90
PFI Fuel Type	gasoline	gasoline	E85
# of Diesel Injections	1	1	1
Diesel SOI [°BTDC]	7...20	47...60	12...14
Diesel Injection Pressure [bar]	1500...2200	800	1900...2000
Highest Load Achieved [bar BMEP]	11.6	9.9	19
Best BTE [%]	43.6	42.1	45.1

RESULTS AND DISCUSSION

Gasoline-Diesel Combustion

Figure 1 shows the key air system conditions from both CM1 and CM2 operation as a function of load. The current air system reached the maximum capacity for EGR delivery at an EGR level of 46~47%. As engine load increased, dual-fuel operation migrated from fuel-lean to near stoichiometric.

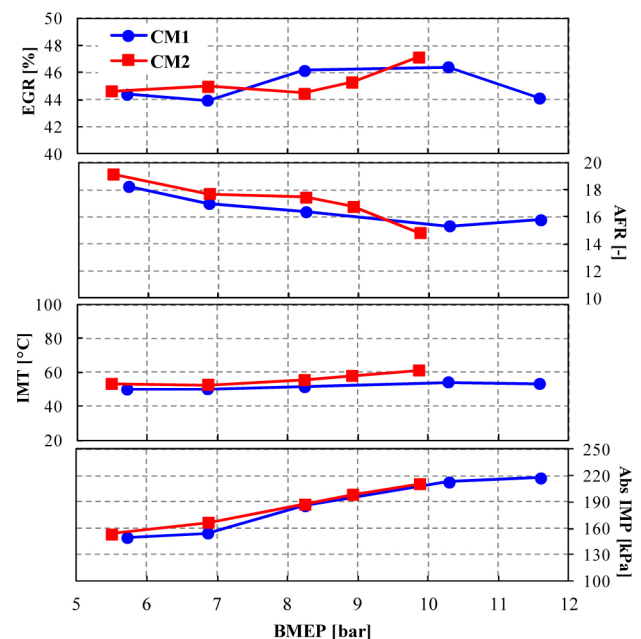


Figure 1. Air system boundary conditions during gasoline-diesel Operation

Fueling conditions during gasoline-diesel operation are detailed in Figure 2. As discussed previously, CM1 and CM2 differ primarily in diesel injection strategy. Through near-TDC diesel injection, CM1 utilizes diesel fuel as an ignition source. In contrast, with early diesel injection, CM2 relies on