

Operation of a Gasoline Direct Injection Compression Ignition Engine on Naphtha and E10 Gasoline Fuels

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ABSTRACT

Gasoline Direct Injection Compression Ignition (GDCI) is a partially premixed low temperature combustion process that has demonstrated high fuel efficiency with full engine load range capabilities, while emitting very low levels of particulate matter (PM) and oxides of nitrogen (NO_x). In the current work, a comparison of engine combustion, performance, and emissions has been made among E10 gasoline and several full-boiling range naphtha fuels on a Gen 2 single-cylinder GDCI engine with compression ratio of 15:1. Initial results with naphtha demonstrated improved combustion and efficiency at low loads. With naphtha fuel, hydrocarbon and carbon monoxide emissions were generally reduced at low loads but tended to be higher at mid-loads despite the increased fuel reactivity. At higher loads, naphtha required less boost pressure compared to gasoline, however, up to 20% additional EGR was required to maintain combustion phasing. The higher reactivity of naphtha did cause some reduction of peak load for the operating strategy tested, when constrained to the targets used in this study. For all fuels, particle size distributions were bimodal and had similar numbers of particles larger than 40 nm, while there were some differences in numbers of smaller particles. Based on an efficiency loss analysis, the fuels exhibited generally similar performance over the load range.

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INTRODUCTION

Benefits of Naphtha Fuels

Naphtha, a term covering a range of light petroleum distillates, is a feedstock for market gasoline and other petrochemicals. In typical refineries today, straight-run naphtha is blended directly with upgraded (refined) naphtha to produce market gasoline [1]. Naphtha is significantly less processed than conventional gasoline [1]. The percentage of the fuel energy used from well to pump is reduced from 22% of the fuel energy of gasoline to 13% of fuel energy used of naphtha. Greenhouse gas emissions of the well to pump process is reduced, being 18 g/MJ fuel energy of gasoline and only 12 g/MJ fuel energy for naphtha [2]. Naphtha fuels have higher heating value and higher energy density than gasoline as well.

With China and India undergoing rapid economic development, the worldwide use of diesel fuel has increased steadily over recent years, while gasoline usage has been relatively stable [3]. This is leading to an increasing imbalanced use of gasoline and diesel fuels compared to the proportions they make up of a typical barrel of crude oil.

Therefore there is a great need for a high efficiency, low emissions engine which can operate on non-diesel portions of the crude oil distillation curve. Naphtha could be a key fuel for the future given its slightly lighter location in the crude oil distillation curve than diesel, higher energy density than gasoline, and lower refining costs and carbon dioxide (CO_2) emissions.

Naphtha Fuel Usage in Internal Combustion Engines

During the early days of internal combustion engine (ICE) development, several fuels were tested from different regions of the crude oil distillation curve [4]. In the late 1800s, a naphtha fuel was used in the "mineral spirits" engine, while heavier fuels were used in the "paraffin" engine (predecessor to the diesel engine). Today, a "diesel" region of the distillation curve with high cetane number was designated for its short ignition delay properties in compression ignition engines and a lighter "gasoline" portion of the crude oil distillation curve was established for its reduced auto-ignition (anti-knocking) properties and higher volatility in spark ignited engines. But wide-spread use of naphtha fuels in ICEs did not become a commercial option due to incompatibility with production spark-ignition and compression-ignition (diesel) engines. Recent

advancements in partially premixed compression ignition (PPCI) engine technology present a new opportunity to use naphtha for highly efficient, low CO₂ emissions compression ignition engines [5, 6, 7, 8, 9, 10, 11, 12, 13, 14, 15, 16]. Due to the lack of a naphtha fueling infrastructure, such PPCI engines may be required to initially operate on current market gasoline. This research explores the opportunities that a higher reactivity fuel may offer from both a wells-to-pump and pump-to-wheels energy and CO₂ emission perspective.

Compression Ignition of Gasoline-Like Fuels

Compression-ignition engines have inherent thermodynamic advantages over spark-ignition engines by being able to run lean, unthrottled, and at higher compression ratios. The main challenge with diesel compression-ignition engines has been higher emissions of particulate matter (PM) and oxides of nitrogen (NO_x) due to the nature of diesel diffusion flame combustion [17]. Homogeneous charge compression ignition (HCCI) is an engine combustion concept which leverages all of the thermodynamic advantages of compression-ignition engines, but with very low PM and NO_x emissions through a well-mixed fuel-air mixture [18, 19]. The challenges of HCCI are that the ignition and combustion phasing is predominantly controlled by chemical kinetics. This has made it difficult for HCCI engines to operate over a wide engine load range or through speed and load transients [20, 21].

With advances in gasoline direct injection (GDi) technology, some researchers began using direct injection of gasoline to obtain better control of the combustion phasing through fuel mixture and thermal stratification [22, 23, 24, 25, 26]. In this early work of direct-injection compression-ignition of gasoline, relatively low injection pressures limited the retard of the last injection timing before significantly increasing PM and NO_x emissions. With the use of much higher pressure common rail diesel injection hardware, it became possible to perform injections of gasoline at a wide variety of timings in the compression stroke for better control of mixture stratification and combustion phasing, while maintaining very low PM and NO_x emissions [27, 28, 29, 30, 31, 32, 33, 34, 35]. Continued advancements of GDi injection pressures and injector design have enabled injections during the compression stroke with well-controlled mixture stratification and simultaneously very low PM and NO_x emissions [36]. Gasoline compression ignition using a single-fuel and a single combustion mode was demonstrated over a wide range of engine speed and load [36, 37, 38, 39].

The objective of this work was to compare combustion characteristics, performance, and emissions of a Gen 2 Gasoline Direct Injection Compression Ignition (GDCI) engine running on baseline RON 91 E10 gasoline to operation with three different full-boiling range naphtha fuels of RON 80, RON 70, and RON 60. All fuels were tested, where possible, at eight speed-load engine operating conditions. System level parameters were modified for each fuel and test condition to meet combustion and emissions targets with the minimum fuel consumption and hydrocarbon (HC) emissions.

ENGINE DESCRIPTION

Experiments were performed on a single-cylinder Gen 2 GDCI engine at Delphi. The cylinder head, cam drive, valve train, and piston were identical to the multi-cylinder Gen 2 GDCI engine [40]. The cylinder head was mated to a Ricardo Hydra cylinder block, crankcase, and crankshaft. Only cylinder two of the head was operated. A photo of the engine installed in the test cell is shown in [Figure 1](#).

The GDCI cylinder head has four valves with double-overhead camshafts and central injection ([Figure 2](#) and [Figure 3](#)). The aluminum cylinder head is rated at 200 bar peak cylinder pressure. The injector is centrally-mounted without offset or inclination relative to the cylinder axis. The combustion chamber features a shallow pent roof with valves flush with the pent roof surface. The low pent roof angle was selected to achieve high compression ratio, while maintaining low chamber surface area and reasonable valve sizes. Intake and exhaust ports were developed for good flow characteristics but without the need for swirl, tumble, or squish. This was a significant benefit and produced good intake flow efficiency as a function of valve lift as measured on a steady flow bench. The engine features a Type II valve train with roller finger followers with low friction and compact packaging. No spark plug exists in the cylinder head but an automotive grade cylinder pressure sensor (CPS) was mounted nearly flush with the chamber surface under the intake valves. In addition to automotive grade cylinder pressure sensing, a Kistler cylinder pressure transducer was mounted vertically at the cylinder periphery. [Figure 4](#) shows the chamber layout. Engine specifications are listed in [Table 1](#).



Figure 1. Photo of the Gen 2 GDCI Single-Cylinder Engine Installed in the Delphi Test Cell.

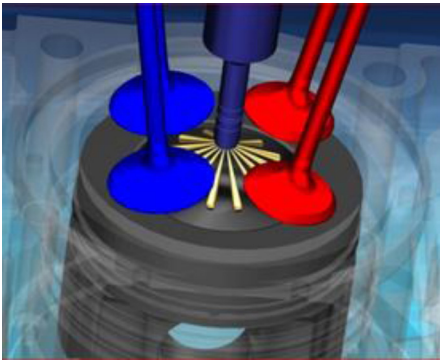


Figure 2. Packaging of Central Injector and Valves in Gen 2 GDCI Engine. Fuel Spray is not Representative as Shown.

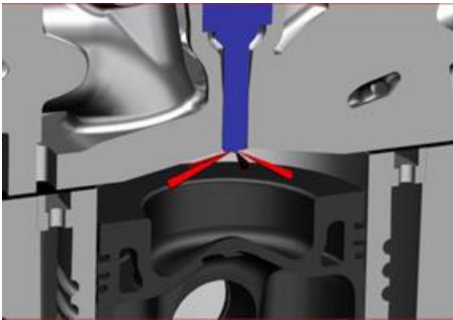


Figure 3. Cross-section of Gen 2 Cylinder Head Showing GDi Injector and Piston. Piston Bowl Design is not Representative as Shown.

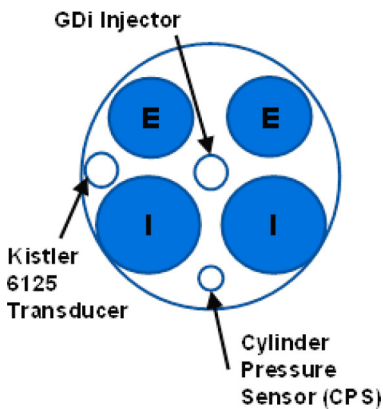


Figure 4. Layout of Combustion Chamber Showing Positions of GDi Injector, Cylinder Pressure Sensor, and Kistler Pressure Transducer.

Injection System

The GDCI injection system has evolved considerably. Simulation tools were used to develop injector spray characteristics that provide very fast vaporization and low spray penetration, while also providing the necessary high injection rate (flow rate). The multihole GDi injector features a symmetric spray pattern with the intent to minimize piston and liner wetting for all injections. The piston bowl shape was matched to injector spray characteristics and features an open bowl with low surface area for low heat losses.

The GDi fuel pump was mounted at the end of the cylinder head and was driven by the exhaust camshaft. The pump shaft has four lobes and controls rail pressure via modulation of the spill valve. The fuel rail has inlet and outlet orifices for dampening of pressure pulsations in the system. Injector drive waveforms were developed to support the multiple injection strategies used for GDCI. The injection system delivers accurate fuel quantities for multiple injections in the cylinder.

Table 1. Base Engine Specifications.

Engine Type	4-valve Compression Ignition
Chamber	Shallow 10-deg Pentroof
Displacement	450 cc per Cylinder
Bore x Stroke x Rod	82 × 85.2 x 144.7mm
Compression Ratio	≈15:1
Cyl. Pressure Rating	200 bar
GDi Injection	Multi-hole, Central Mount, 400 bar
Valve Train	DOHC Type II, Exhaust Rebreathing
Mixture Motion	Near Zero Swirl and Squish
Boost	External Compressor - Dry
EGR	High-Press-Loop w/ Cooler
Oil Jets	Two per Cylinder
Lubricant	Shell Rotella 15W-40

Valve Train System

Full-time GDCI combustion over the full operating map is achieved by using exhaust rebreathing (RB) at low loads and cooled EGR at medium to higher loads. Exhaust rebreathing is accomplished using the exhaust valve train system, which provides a secondary exhaust lift event during the intake stroke. It is an effective method to recuperate heat from hot exhaust gases in order to raise mixture temperatures. This heat promotes autoignition at low loads when boost pressure is zero or low. Rebreathing also reduces pumping work (intake and exhaust valves are open at the same time) and raises exhaust temperatures significantly [41]. In this way, a close-coupled catalyst will remain warm over a wide range of low loads without special maintenance heating [40].

The rebreath functionality is provided by a Type II, continuously-variable mechanical valve train system. The valve train has good packaging characteristics with low friction. An electric variable cam phaser (Delphi e-Phaser) is used to actuate the exhaust valve train and control the secondary valve lift with very fast response.

EXPERIMENTAL SETUP

A schematic diagram of the engine test setup is shown in Figure 5. Air flow was controlled and measured by a Flow Systems FC-500 system using a 7-nozzle sonic array. Dry air was supplied from building compressed air and heated by an electric heater. An EGR valve and cooler were located between intake and exhaust surge

tanks, which ensured that EGR was well mixed prior to entering the engine. This system enabled single-cylinder engine testing under simulated turbocharged operation.

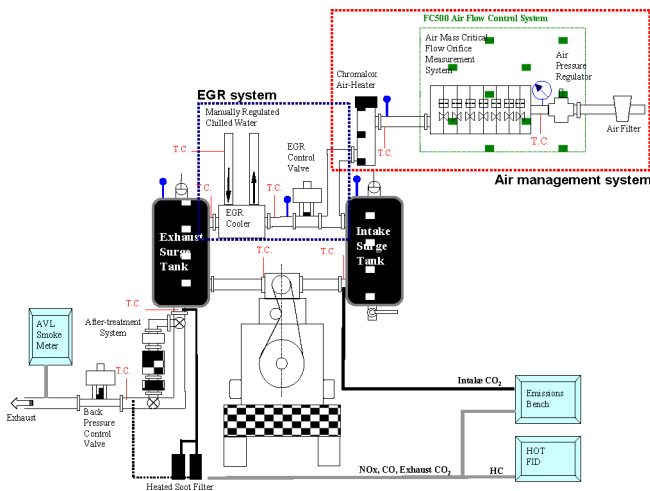


Figure 5. Schematic Diagram of Test Setup.

Fuel flow was measured with a high-precision Pierburg PLU103b fuel meter. The total error of this instrument as calibrated is 0.3 percent of reading down to 0.05 g/s fuel flow. Fuel flow data was corrected to that of an E00 gasoline equivalent based on the lower heating value of the test fuel. Indicated specific fuel consumption calculated using corrected fuel flow was reported as $ISFC_{REF}$.

All emissions were measured on a wet basis using heated sampling systems. Carbon species, nitrogen species, and some hydrocarbon species were measured with an AVL Sesam FTIR. Total hydrocarbon emissions were measured with a Horiba Fast FID.

Exhaust oxygen and intake CO_2 were measured with a Rosemount MLT-3 analyzer. Smoke was measured with an AVL 415S smoke meter. Particle size distributions were measured with a TSI 3090 Engine Exhaust Particle Size Spectrometer [42]. The sampling system for exhaust particles is shown in Figure 6. Exhaust particles were sampled from the exhaust runner using a heated sampling system. The sample probe was a “radial aperture” design with the intent to sample gas uniformly across the exhaust runner. Sample plumbing was operated at 191°C and could be purged periodically with high pressure nitrogen gas. The sample was diluted using a TSI 379030 Rotating Disc Thermodiluter, which included a volatile particle remover (VPR) to suppress measurement of volatile and semi-volatile particulate matter [43]. The VPR was operated at 300°C for the tests performed in this work. As seen in Figure 7, operation of the VPR at 350°C and 400°C only slightly altered the particle size distribution for a low load (higher HC emissions) operating condition.

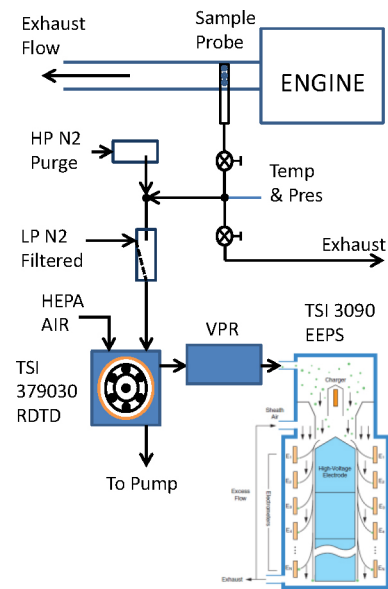


Figure 6. Schematic diagram of particle emissions sampling system.

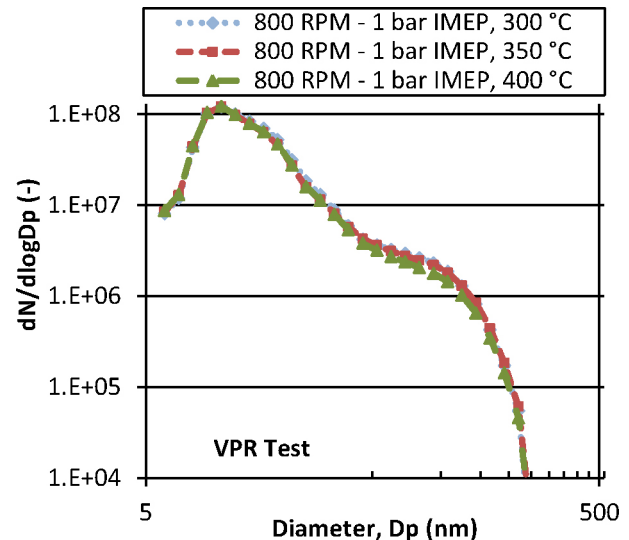


Figure 7. Effects of Changes to Volatile Particle Remover Temperatures from 300°C to 400°C.

Cylinder pressure measured with an in-cylinder, flush-mounted Kistler 6125CU20 pressure transducer was sampled with 0.5 crank angle degree (CAD) resolution using a Kistler 2613B crankshaft encoder. Indicated mean effective pressure (IMEP) was reported on a gross basis as IMEP_g.

A single-cylinder engine controller was developed with real-time heat release analysis capability. The controller is based on National Instruments hardware and Labview software, and was built by Drivven, Inc. [44,45]. Within the controller, a multiple-injection control utility was developed to determine and control the fuel quantity for each of the injection events. Fuel quantity was

determined for each injection using instantaneous rail pressure, cylinder pressure, and an embedded calibration map for the tested injector. Closed-loop PID control of total fuel delivery was used to maintain desired IMEP. Similarly, closed-loop PID control was used for intake and exhaust pressure, EGR flow, rail pressure, and intake air temperature.

Residual mass concentration in the cylinder was measured by a Delphi Residual Estimator Tool (RET) [46–47]. Integrated with GTPower [40], the RET provides accurate estimates of residuals based on measured cylinder pressure, airflow, and average temperature and pressure at the intake and exhaust ports. As part of the RET macro, simulations are conducted using GT-Power to perform an efficiency loss analysis.

All test data from all instruments used in these experiments was saved in binary files on the Driven engine controller. The data was post-processed using special macros to perform detailed combustion analysis, calculate residual mass, and determine efficiency losses, among other calculations. The crank-angle location of 50% fuel mass burned (CA50) represented combustion phasing, and the crank-angle duration from 10% to 75% fuel mass burned (CA10-75) represented combustion duration. In the efficiency loss analysis, the experimental indicated thermal efficiency (ITE_{EXP}) was compared with the simulated indicated thermal efficiency from the combustion macros (ITE_{RET}).

FUEL PROPERTIES

Naphtha is a liquid petroleum product that has a boiling point range from about 30°C to approximately 200°C. It consists of a complex mixture of hydrocarbon molecules generally having between 5 and 11 carbon atoms. Usually the research octane number (RON) of naphtha is around 70. Light naphtha is the fraction of naphtha with a boiling point between 30°C to 90°C and consists of molecules with five to six carbon atoms. Heavy naphtha has a boiling point between 90°C and 200°C, and consists of molecules with 7 to 11 carbon atoms. Light naphtha has higher octane number (lower cetane number) compared to heavy naphtha, but a range of octane numbers can be obtained by taking different cuts of distillates from different boiling point ranges.

Naphtha is anticipated to require less upgrading, offering potential CO₂ savings within the refining process, which means that the energy required to produce naphtha and its resulting CO₂ impact will be lower compared to conventional gasoline and diesel fuels. Moreover, lighter fractions of the crude oil, like naphtha, will be readily available and could be in surplus as the demand shifts toward heavier fuels used in commercial transport.

The fuels tested in this study included three naphtha fuels, with RON 60, RON 70, and RON 80 octane number ratings. A “RON 91” E10 gasoline, representative of U.S. grade “regular” market gasoline, was used as the baseline test fuel and has been used for GDCI research for several years [36,38,40,49,50,51]. The major fuel properties are listed in Table 2.

Table 2. Properties of Fuels Evaluated.

	Test Method	RON 60	RON 70	RON 80	RON 91 E10
Target Research Octane Number		60.0	70.0	80.0	N/A
Research Octane Number	ASTM D2699	58.8	71.4	80.3	92.6
Motor Octane Number	ASTM D2700	57.9	69.7	77.9	83.0
Derived Cetane Number	ASTM D6890	33.7	28.5	26.3	21.0
Specific Gravity at 15.56°C	ASTM D4052	0.7054	0.7456	0.7445	0.7199
Gross Heating Value [MJ/kg]	ASTM D240	46.574	46.917	46.762	45.075
Net Heating Value [MJ/kg]	ASTM D240	43.363	43.815	43.683	42.089
Carbon [wt%]	ASTM D5291	84.87	85.38	85.49	81.90
Hydrogen [wt%]	ASTM D5291	15.13	14.62	14.51	14.07
Oxygen [wt%]	ASTM D5622	0.00	0.00	0.00	4.03

The RON 60 fuel is a straight-run refinery naphtha stream, which is the most reactive fuel among those tested, showing the highest derived cetane number (DCN) and the lowest RON. Both RON 70 and RON 80 naphthas were formulated in order to match the target RON values. The test fuels were chosen and formulated in a way such that their autoignition qualities, in terms of RON, would vary widely by more than 30 units of RON, specifically from 58.8 to 92.6 RON. This difference in RON will lead to significant measurable change in ignition delay and combustion characteristics. The specific gravity of the RON 60 naphtha is slightly lower than that of RON 91 E10 gasoline, while the values of both RON 70 and RON 80 naphthas are higher because of an increased amount of heavier molecular weight components, mainly paraffinic and iso-paraffinic components, which were added to meet the design specifications. In addition, the energy content of the naphtha test fuels was somewhat higher than the baseline gasoline.

It was decided to use lubricity additives for the naphtha fuels in order to enable engine testing with either diesel or gasoline high pressure pumps. Based on the wear scar diameter testing results, a dosing concentration of 400 µg/kg was selected for the naphtha fuels in this study.

Figure 8 shows the distillation characteristics of the naphtha and baseline gasoline fuels tested. The dotted grey shaded area in the figure represents the distillation range of typical U.S. market gasolines [52]. The baseline RON 91 E10 gasoline used in this study was at the lighter end of the distillation range of typical U.S. market gasolines since this fuel was procured during the winter season. The RON 60 naphtha and RON 91 E10 gasoline had a similar initial boiling point, but the final boiling point of the RON 60 naphtha was lower than that of the RON 91 E10 gasoline. Both the RON 70 and RON 80 naphthas showed similar initial and final boiling points as the RON 91 E10 gasoline, but overall they showed less volatility throughout the distillation curve than the RON 91 E10 gasoline.

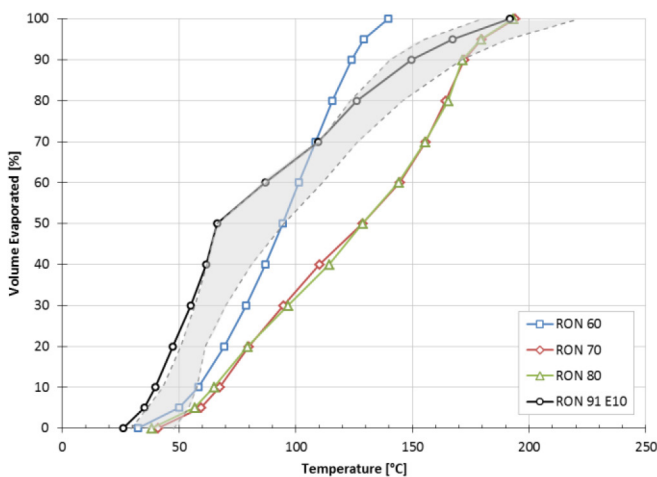


Figure 8. Distillation Characteristics of Fuels Tested.

Chemical compositions of the tested fuels were compared in Figure 9. As shown, the major hydrocarbon components of naphtha fuels are paraffins, isoparaffins, and naphthenes. Even though it was known that typical naphtha streams contain very low aromatic content, the tested naphtha fuels have relatively higher aromatic content. Both RON 70 and RON 80 naphthas showed comparable amounts of aromatic content to “E10 gasoline” in order to meet the desired RON targets.

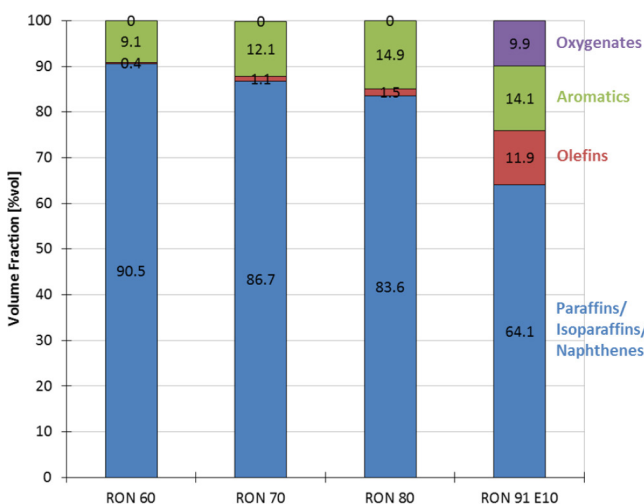


Figure 9. Chemical Compositions of Fuels Tested.

GDCI COMBUSTION BASICS

Timing of Last Injection Event

Since GDCI is not homogeneous charge compression ignition (HCCI), mixture stratification, through injection system parameters, is an important control of the combustion process. The last injection, which is normally in the compression stroke for GDCI operation, offers greater control over the combustion process and emissions than earlier injections which may take place during the intake stroke. To demonstrate the effects of the start of injection of the last injection timing (SOI_{last}) on the combustion and emissions, the results of a typical SOI_{last} sweep are shown in Figure 10 through Figure 12. The

engine was operated at 800 RPM - 2 bar IMEP using RON 70 naphtha fuel. At these low loads, a single injection strategy was used. So SOI_{last} was the only injection under these conditions. However the trends shown for SOI_{last} under these single injection conditions are similar to the trends observed with multiple injections. To maintain a constant 2 bar IMEP load, the fueling level was adjusted at each injection timing.

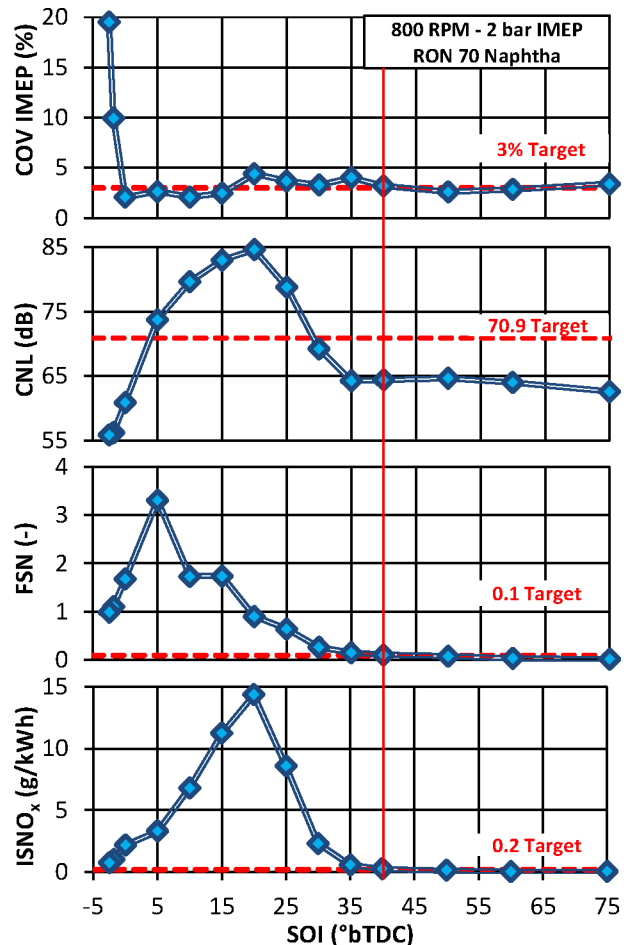


Figure 10. Effect of Last Injection Timing on Key Performance Targets at 800 RPM - 2 bar IMEP using Naphtha RON 70 fuel.

Good combustion stability, as indicated by 3% coefficient of variance (COV) of IMEP, was observed over nearly the entire SOI sweep, except for the latest injection timings. When SOI_{last} was delayed later than a critical timing, combustion noise level (CNL), filter smoke number (FSN), and indicated specific oxides of nitrogen ($ISNO_x$) all increased greatly. Increases in these performance parameters occurred when the ignition dwell, IDW (defined as the duration from end of injection until 10% fuel mass burned), was the shortest and the CA50 (crank angle of 50% fuel mass burned) was the most advanced. Shorter ignition dwell represents less mixing time for the air and fuel before ignition. Shorter mixing times lead to higher emissions of particulate matter (PM) and NO_x due to regions of higher local richness during the combustion process [53]. An early CA50 represents advanced combustion phasing, which promotes higher levels of combustion noise.

Because of the relatively high reactivity of the naphtha test fuel, it was possible to continue delaying injection timing as late as piston top dead center (TDC) while maintaining good combustion stability. Although PM, NO_x, and CNL were all significantly reduced as injection timing was delayed, the combustion became too unstable before PM and NO_x were reduced to their target values.

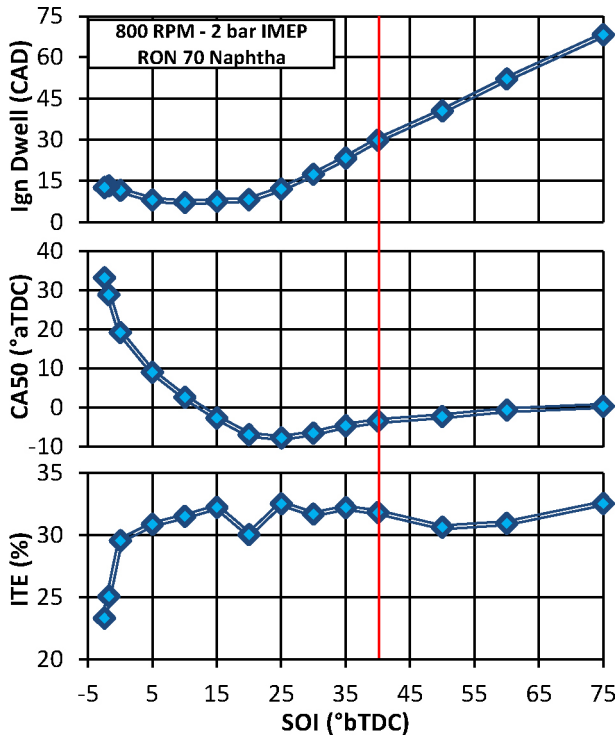


Figure 11. Effect of Last Injection Timing on Ignition Dwell, Combustion Phasing, and Indicated Thermal Efficiency at 800 RPM - 2 bar IMEP using Naphtha RON 70 fuel.

Although combustion efficiency decreased at the earliest injection timings, the engine's indicated thermal efficiency (ITE) remained quite similar out to the earliest injection timing tested. However, due to increased HC and carbon monoxide (CO) emissions, as shown in Figure 12, the optimum SOI was that where injection advance was minimized while still meeting all targets. Increasing the ignition dwell caused portions of the fuel to become overmixed. Regions in the combustion chamber which become overmixed, with local equivalence ratios much leaner than stoichiometry, will require longer time to oxidize due to slower chemical kinetics. When these overmixed lean regions do not have enough time to oxidize, they result in increased HC and CO emissions [54,38,39,]. For these engine operating conditions and fuel properties, the optimum SOI_{last} was approximately 40 degrees before TDC (°bTDC) and is designated by the vertical red line in each figure. These criteria were used to select the best points for all tests in the later fuel comparison tests.

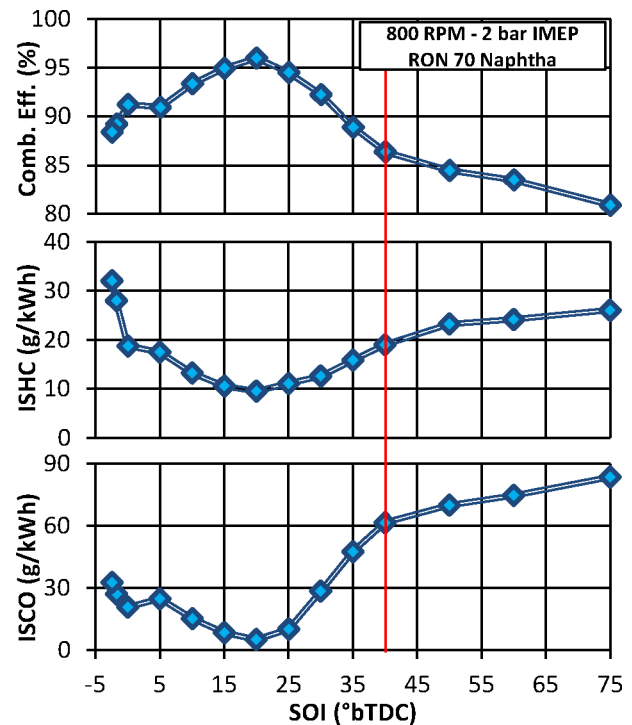


Figure 12. Effects of Injection Timing on ISHC, ISCO, and Combustion Conversion Efficiency at 800 RPM - 2 bar IMEP using Naphtha RON 70 fuel.

Injection Pressure

In addition to the timing of SOI_{last}, changes to the injection pressure can also affect the characteristics of the fuel-air mixture, combustion, and emissions. For the purpose of demonstrating these trends, an injection pressure sweep was conducted at 800 RPM - 3 bar IMEP with RON 80 naphtha using a single-injection strategy. At this relatively low load test condition, the injection pressure was varied from 100 to 250 bar. While the injection system was capable of higher pressures, it was limited to a practical level of 250 bar at this low speed and low load condition, which was sufficient to confirm its effects on the combustion and emissions.

In this experiment, injection timing advance was minimized at each injection pressure to achieve the lowest HC and CO emissions by minimizing IDW (as demonstrated by the SOI_{last} sweep in Figure 11 and Figure 12), while still meeting the PM, NO_x, and CNL targets. The main constraint factor at each injection pressure on minimizing the injection timing advance under these conditions was the 0.1 FSN limit.

In general, higher injection pressure increases air entrainment in the fuel spray and results in better mixing and a reduction of the regions of local richness in the combustion chamber. Where advancing the injection timing increases ignition dwell and mixing time, increasing injection pressure increases mixing quality. Figure 13 shows that as the injection pressure increased, the minimum injection timing