

Compression Ratio and Derived Cetane Number Effects on Gasoline Compression Ignition Engine Running with Naphtha Fuels

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ABSTRACT

In the context of stringent future emission standards as well as the need to reduce emissions of CO₂ on a global scale, the cost of manufacturing engines is increasing. Naphtha has been shown to have beneficial properties for its use as a fuel in the transportation sector. Well to tank CO₂ emissions from the production of Naphtha are lower than any other fuel produced in the refinery due to its lower processing requisites. Moreover, under current technology trends the demand for diesel is expected to increase leading to a possible surplus of light fuels in the future. Recent research has demonstrated that significant fuel consumption reduction is possible based on a direct injection gasoline engine system, when a low quality gasoline stream such as Naphtha is used in compression ignition mode. With this fuel, the engine will be at least as efficient and clean as current diesel engines but will be more cost effective (lower injection pressure, HC/CO after-treatment rather than NOx).

In a previous publication, it was demonstrated that a 0.5liter single-cylinder DISI engine of a compression ratio (CR) of 12 could be run in CI mode using a heavy cut of Naphtha with a derived cetane number (DCN) of 41. Over a range of operating points covering the FTP cycle, average fuel consumption was improved by 19% compared to running the base engine in SI mode on 91 RON gasoline. In this study, both light and heavy Naphtha from one of Saudi Aramco's refineries were successfully run in GCI mode with regular valve events and intake charge boosting at six engine running conditions representative of a typical urban driving cycle including idle. The presented work highlights the potential for improvement and the limitations of such a configuration.

In this paper, the effect of increasing the CR on this type of operation is investigated. Three different pistons were designed to increase CR from 12 to 13 and 14 respectively with a diesel type bowl feature. Light Naphtha and Heavy Naphtha cuts with a DCN of 31 and 41 were run over the same operating points covering the FTP cycle with low soot and NOx with each piston. Fuel economy improvements over SI operation vary from 15% with CR12 to 26% with CR14 with light naphtha. Heavy Naphtha did not benefit as much from the increased compression ratio and the greatest fuel economy improvement in this case was 22% at CR13. Hence, the efficiency of a DISI engine could be significantly increased by fueling it with Naphtha with partially premixed combustion.

The load range of these engine/fuel configurations was assessed across engine speeds from 1000RPM to 3000RPM. The maximum load achieved was 10bar NMEP with light Naphtha at CR12, however it did not provide good combustion stability at loads below 6bar NMEP. At all three CRs, Heavy Naphtha allows operating with good combustion stability on the load range below 2bar NMEP but fails to reach loads above 7bar NMEP on the high end due to high pressure rise rates.

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INTRODUCTION

Global transport energy demand is expected to increase, primarily in non-OECD countries, by around 40% by 2040 compared to today [1,2,3,4]. Around 95% of this energy is currently supplied by petroleum based liquid fuels [2]. The demand for transport fuels is very large and in spite of the growth in alternatives, even by 2040, around 90% of transport

energy is expected to come from petroleum. This growth will not be constrained by the supply of oil [3,4] - oil reserves have been increasing at an unprecedented rate in recent years. The growth in demand is expected to be very heavily skewed towards commercial transport which uses heavier fuels like diesel and jet fuel compared to gasoline [2,3,4,5]. For instance, the ratio of (diesel+jet) demand to gasoline demand

is expected to move to between 2.4 and 3.8 compared to the current value of around 1.5 [5]. For future fuel demand and supply to be in balance, investment will be required in fuels manufacture. At the same time, future compression ignition engines will need to substantially use fuels other than current diesel fuels, e.g., "poor quality" gasoline such as naphtha, CNG (compressed natural gas) and GTL (gas to liquid) fuels.

A lot of recent research, e.g., [3,4,6, 7, 8, 9, 10, 11, 12, 13] has demonstrated that using low octane gasoline in diesel engines has great potential to achieve very high efficiency and reduce the cost of diesel engines (lower injection pressures and after-treatment focus on HC and CO oxidation rather than NOx and soot control). This approach, using fuels such as naphtha or straight run gasoline from the initial distillation of crude, which are less processed compared to current diesel or gasoline, can be termed Gasoline Compression Ignition (GCI). GCI technology has the following advantages:

- Opens up a path to mitigate expected demand imbalance between heavy and light fuels.
- Offers diesel-like efficiency at lower cost.
- Uses fuels that are "less processed" compared to current gasoline and diesel and uses light components that might be in surplus otherwise

Initially, GCI technology has to be developed using existing market fuels. In general, practical gasoline fuels have RON > 90 (cetane number, CN < ~19) while diesel fuels have CN > 40 (RON < ~40) in most markets. Sellnau and co-workers, e.g., [13] have been developing a GCI system to run on U.S. regular gasoline - (RON+MON)/2 of 87. They have changed the base SI engine by increasing the compression ratio to 15, designing a new injection system and injectors and redesigning the combustion chamber. As a result, they have demonstrated significant improvement in fuel efficiency compared to the base SI engine. Reitz and co-workers have been using a dual-fuel approach, which they term Reactivity Controlled CI or RCCI [e.g., 14, 15]. In this concept, the injection strategy is changed by using port-fuel injection of a market gasoline, and direct injection of a market diesel and indicated efficiencies of greater than 50% are achieved, while around 90% of the fuel used is gasoline rather than diesel. Another possible approach is to use a mixture of gasoline and diesel, e.g., 10% diesel fuel + 90% gasoline and run an engine in CI mode [e.g., 16]. All these concepts will allow "diesel" engines to mostly use gasoline rather than diesel fuel and thus open a path to mitigate the expected demand imbalance between gasoline and diesel. Other research suggests that for the GCI concept, the optimum fuel is a much less processed fuel - e.g., a "low quality" gasoline, with RON between 70 [11] and 85 [8,9], which does not need to have a final boiling point that is as low as today's gasolines.

Naphtha is the generic term used to describe fuel streams produced from different processes in the refinery in the gasoline boiling range. Naphtha needs to be processed further, at the cost of more energy and CO₂, to increase its octane number to convert it into useful gasoline components. Hence it is important to develop GCI systems, which use less processed fuels such as naphtha as this will also help save energy in fuels manufacture. A small step in this direction was described in [17] where it was demonstrated that, with some redesign (higher compression ratio) and a sophisticated control system, a modern diesel engine can be run on narrow-cut naphtha with a cetane number of 40 (rather than a European diesel fuel of cetane number > 52) and meet Euro 6 emissions standards and other requirements such as acceptable noise levels.

Almost all the work in GCI has started with diesel engines except perhaps the approach by Sellnau and co-workers [13]. In our previous work [18,19], we have started with a single-cylinder direct injection SI engine with a DISI injection system with a maximum injection pressure of 15 MPa - around 10 times lower than in a typical diesel injection system. Initially, with a compression ratio (CR) of 12, we could demonstrate a fuel efficiency benefit of 19% over an FTP cycle by running the engine in CI mode on heavy naphtha of 38 DCN (derived cetane number) when compared to running the engine on SI mode on 91 RON gasoline [18]. We could then increase the efficiency benefit to 26% by increasing the CR to 14 using a redesigned piston and running the engine on GCI mode on a light naphtha with a DCN of 31. In this work we present a more systematic study on the effect of CR (12,13,14) and the DCN of naphtha on GCI operation in this single-cylinder engine. The speed range covered is between 1000 rpm and 3000 rpm and the load range is up to a maximum of 10 bar (1 MPa). As expected, the heavy naphtha, with higher cetane, makes low load operation even at the lower compression ratio, easier but the light naphtha enables the engine to be run at a higher compression ratio at higher loads and gives higher efficiency.

EXPERIMENTAL SETUP

Data Acquisition System

Pressure is measured using Kistler pressure sensors and charge amplifiers along with a Leine and Linde 500 series crank angle encoder. The sampling resolution was adjusted to 0.3 Crank Angle Degree (CAD). Indicated mean effective pressure (IMEP) net mean effective pressure (NMEP), pumping mean effective pressure (PMEP), heat release rate and mass fraction burned are calculated with CAS using the commercial formulae.

Table 1. Crank angle resolved sensors description

Measurement	Location	Type
Intake Pressure	200 mm back of valve	4045A (piezo resistive)
Cylinder Pressure	Flush mounted Yellow spot in Figure 1	6061B (quartz water cooled)
Exhaust pressure	160 mm back of valve	4045A (piezo resistive water cooled)

Emission and EGR data are obtained through the Horiba MEXA-7500D-EGR. Exhaust gases are sampled with heated lines at 190°C. NOx and HC are measured in wet condition. Air fuel ratio as well as EGR rate is calculated by the Horiba emission bench. Lambda is calculated using the Brettschneider/Spindt formula and the EGR calculation assumes an ambient air containing 400 ppm of CO₂. The EGR rate refers to the volume fraction of exhaust gases found at the intake as a percentage of the total inlet charge. Smoke emissions are obtained by means of an AVL415S smoke meter and probe inserted into the exhaust gas stream. The exhaust temperature is measured via a K type thermocouple whereas all others temperatures are measured with PT100 thermocouples. Engine coolant and lubricant temperature is maintained at 90°C. Air temperature in the engine intake manifold is adjusted to 30°C without EGR activation and from that point increases gradually as a function of the EGR rate. A Siemens DI1.5 flow meter relying on the Coriolis principle measures the fuel mass flow supplied to the engine. Airflow measurement is covered by the ABB SensyFlow hotwire mass flow meter.

Base Engine Configuration

A single cylinder 4 valve engine with a 12:1 geometric compression ratio is used in this investigation. Table 2 shows details the engine specifications. The combustion chamber is originally designed to accommodate stratified charge spark ignition combustion. An outwardly opening piezo-electric production gasoline direct injector is centrally mounted adjacent to the spark plug. In Gasoline Compression Ignition (GCI) combustion tests, the spark is disabled. As can be seen in Figure 1, the injector is located between two intake valves and slightly skewed from the vertical direction. Fuel is introduced by an outwardly opening piezo-electric injector with hollow cone spray. Spray is more widely distributed with less penetration compared to spray-jet style multi-hole type gasoline direct injector. The operating fuel injection pressure range is 50-150 bars. Because fuel injection pressure is about 10 times lower than conventional Diesel injector, spray atomization and penetration are expected to be quite different. Although the injector and piston design might not be optimum for GCI combustion, 19% fuel consumption reduction had been achieved via this piston configuration with heavy naphtha fuel [15].

Table 2. Single Cylinder Engine Specifications

Cylinder	1
Number of Valves	4
Displacement (cm ³)	499.0
Bore (mm)	84
Stroke (mm)	90
Compression Ratio	12
Fuel Injector	Outwardly Opening Piezo-Electric
Fuel Pump / Range	External Motor Driven / 40-150bar
Intake System	Conditioned Air with External Boost
External EGR Line	Uncooled High Pressure Line
Exhaust System	No Catalyst (Back Pressure Control Valve)

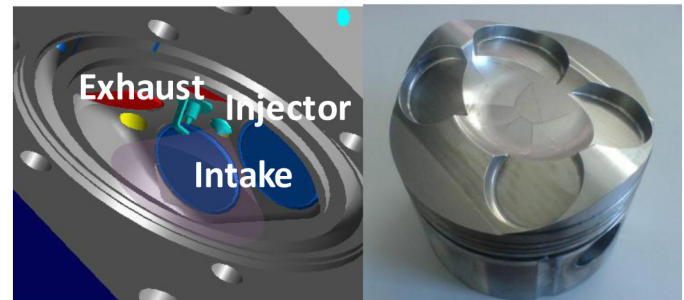


Figure 1. Base engine Combustion Chamber Shape: Pent-Roof Style 4 Valve Head (left), Piston with Valve-Cut and Shallow Oval Bowl Feature (right)

Compression Ratio Variation

For the purpose of this study a new set of pistons was designed and manufactured. To improve fuel containment, a diesel bowl-like feature was added. It also has a reentry feature to ensure that spray interaction is happening inside the bowl. In Table 3 the piston dimensions highlight how the compression ratio varies by adjusting the bowl design, essentially by reducing bowl and throat diameter. Bowl depth is little changed as the CR increases from 12 to 14. Squish height is 1.5mm (yellow). Since the combustion chamber is originally made for a gasoline engine (pent-roof style), the piston height is quite tall to match the CR and compensate the bowl volume. It also results in a smaller clearance height than a conventional gasoline piston. As can be seen in Figure 2 (side view), the spark plug is still installed even though it was not utilized in this study. The crevice in the spark plug can be considered as a major source of HC emissions if fuel spray is steered near the spark plug.

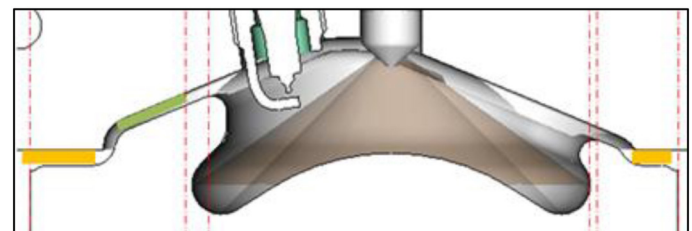


Figure 2. Side view picture of compression ratio 12.5:1 incorporating diesel bowl reentry feature



Figure 3. Top and full views of the three different GCI pistons. From top to bottom CR12, 13 and 14

Table 3. PPCI Pistons dimensions as a function of the compression ratio

Bowl Dimensions (mm)	CR12	CR13	CR14
Throat Diameter	49.2	47	43
Bowl Diameter	55.4	53	47.8
Bowl Depth	18.5	17.4	17.2

GCI testing was done with fixed intake and exhaust valve events and symmetric lift profiles (8 mm peak lift and 207 CAD duration), see Figure 4. This is considered to be similar to conventional production Diesel valve lift profiles. An external EGR line is routed from the exhaust to the intake plenum. An exhaust back pressure control valve is used to create the pressure differential to drive the EGR.

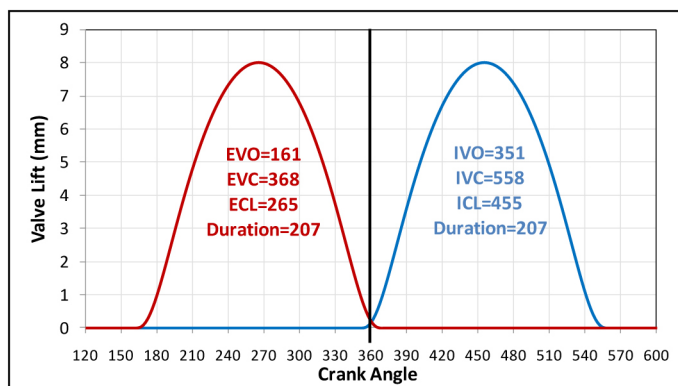


Figure 4. GCI Valve Lift Profile: 8mm Peak Lift, no Variable Valve Timing with Minimum Valve Overlap

Naphtha Derived Cetane Number Variation

Naphtha with Derived Cetane number (DCN) of 31 and 41 were used in this study. Both naphthas are obtained from a Saudi Aramco refinery in Saudi Arabia. Figure 5 shows ASTM D86 distillation curves for the fuels used. Even though they have the same initial boiling point, Light Naphtha's final boiling point is lower than that of a typical gasoline. Heavy Naphtha is the opposite with a boiling range slightly higher than that of gasoline and finishes at around the initial boiling point of diesel fuel. In Table 4 the differences between both naphthas can be highlighted. First of all the cetane number difference between both fuels is 10, which will lead to significant measurable changes in ignition delay and following burn location in an engine. The energy content is also slightly different with 9.5 MJ/kg more energy in the case of the heavy naphtha.

Table 4. Fuel Analysis Result: All Fuels Supplied from Saudi Aramco Refinery in Saudi Arabia

Fuel Name	Saudi Aramco Light Naphtha	Saudi Aramco Heavy Naphtha	Saudi Aramco Gasoline 91RON
DCN from IQT	31	41	16
RON	68	<40	93
MON	63	<40	84
Specific Gravity	0.66	0.76	0.74
Gross HV, MJ/kg	41.8	45	42.3
LHV or Net HV, MJ/kg	41.4	44.6	41.9
Normal paraffins	55	33	21
Iso-paraffins	38	30	30
Aromatics	1	27	31
Naphthenes	6	7	5
Olefins	0	2	1
Oxygenates	0	0	12

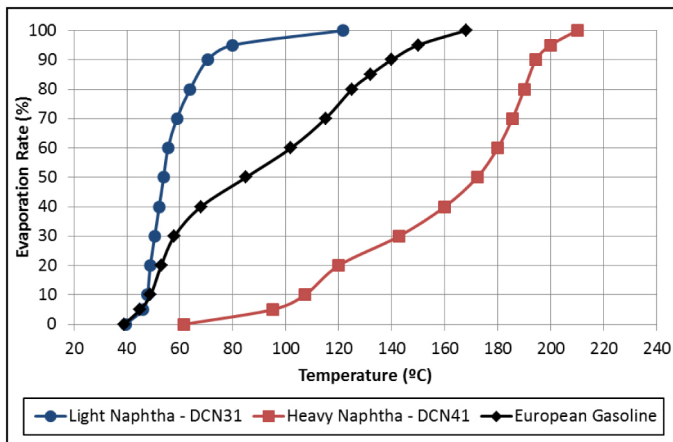


Figure 5. Evaporation Curves using ASTM D86 Method. Light Naphtha, Heavy Naphtha and Gasoline

Combustion Control Strategies

Two distinct control strategies were applied throughout the test with all hardware configurations. One was multiple injection to cover both light and high load operation, and another was to vary intake valve closing timing to change the effective compression ratio. Both were explained in detail in our previous publication [19]. For each one of the configurations tested, the control of the combustion was achieved by optimizing the number of injections, their timing and the amount of EGR introduced in the cylinder. In the operating range covered in this study, the lower loads relied on single and double injection in a region close to TDC between 45 to 10 CAD bTDC. The double injection at low load addressed mainly combustion stability issues and allowed keeping the fuel within a smaller space promoting combustion reactions that would otherwise extinguish themselves by excess of air. Each injection was separated by 15 to 20 CAD with approximately 30% of the quantity injected in the first injection while the remainder was injected in the second injection. As the load

increased the stability issues were moderated by a richer mixture and the double injection progressively stopped yielding benefits. Toward the highest loads achieved, the in cylinder pressure rise rate became a limitation.

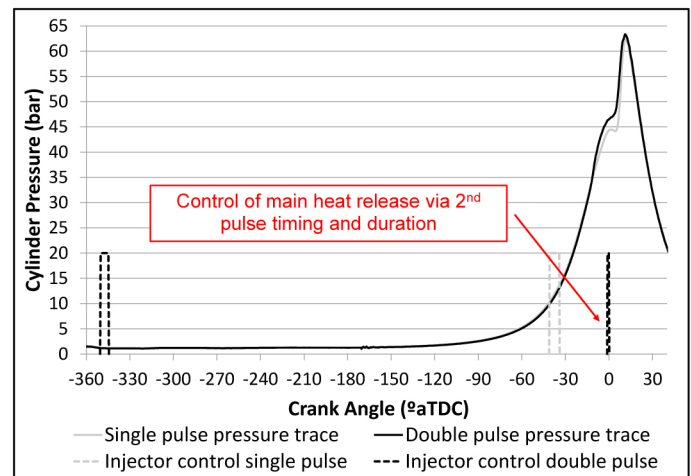


Figure 6. Difference of the combustion pressure profile between single and double pulse injection strategies

Figure 6 stresses another aspect of enabling high load operation, which was the utilization of a double pulse injection strategy. An early injection promoted homogeneous mixing while a second injection introduced some stratification. The combination of both injections lead to a burn rate that can be controlled within the test constraints. Approximately 80% of the fuel was injected early on during the intake stroke creating an homogeneous mixture, and then the remainder was injected close to TDC, to create some level of stratification in the combustion chamber and slow down the burning rate. In an opposite manner, the second injection was also used as a combustion enabler by adjusting its timing in a window from 40 to 20 CAD. before TDC. This strategy was also necessary to reduce smoke emissions at high loads for which the fuel mixing was not adequate with this injection system, with the injection pressure limited to 150 bar. The quantity in the second pulse was determined by a compromise between the pressure rise rate control and the increase in smoke emissions. The second injection close to TDC was suffering from mixing issues because of both the reduced pressure difference between the injector nozzle and the combustion chamber and because of the nature of the GDI injector spray (hollow cone). Another limiting factor at high load is the pressure rise rate. Reducing the effective compression ratio was achieved via closing one intake valve later, i.e., splaying two intake valves. It helped reducing in cylinder temperature and therefore increases the ignition delay. This allowed delaying the combustion and reducing the maximum pressure rise rate, and thus enabling further increase in load. This strategy was utilized only at the high load point when reducing effective compression ratio was necessary.