Compression Ratio and Intake Air Temperature Effect on the Fuel Flexibility of Compression Ignition Engine

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Abstract

The effect of compression ratio (CR) and intake air temperature on the combustion characteristics of fuels with different octane ratings were investigated on a single-cylinder heavy duty engine. The study focused on Primary Reference Fuels (PRFs) and commercial grade diesel with octane numbers ranging from 0 to 100. The engine was configured at a CR of 11.5:1, which is lower than typical heavy-duty compression ignition CI engines. This aims to compare the fuels' burning regime with recently reported measurements at CR17:1. Experiments were performed at different intake air temperatures of 20 to 80 °C and net indicated mean effective pressure (IMEP_{Net}) of 5 to 20 bar. The injection rates have been characterized to determine the hydraulic delay of the injector and thus define the actual ignition delay time. At low loads, diesel-like fuels were found to burn in partially premixed combustion (PPC) mode whereas high octane fuels did not ignite. At high loads, fuels combustion becomes diffusion driven regardless of their RON or MON values. The effect of intake air temperature on the combustion characteristics depended on the combination of the octane ratings and the engine load. At high loads, fuels with low octane numbers were insensitive to the change of the intake air temperature. The ignition delay time was short enough to maintain a diffusion driven combustion. At lower loads, it is more challenging to reach conditions where the combustion characteristics are invariant regardless of the fuel's RON and MON values (Fuel Flexible). At the low tested compression ratio of 11.5:1, the extent of fuel flexibility is limited to only high loads (IMEP_{Net} = 20 bar) whereas it is extended to intermediate loads (IMEP_{Net} = 10 and 15 bar) at CR17:1.

Introduction

Carbon dioxide emission has become the talking point in any sustainability discussion around energy and technology. The contribution of man-made CO₂ emissions to the increase of the average global temperature is becoming more apparent. The Intergovernmental Panel on Climate Change (IPCC) estimated in 2013 that if countries continue burning fossil fuels at the same rate, the Earth will likely warm up with 4.8 °C by the end

of the 21st century [1]. This alarming projection pushed 195 nations to agree at the 2015 United Nations Climate Change Conference held in Paris (COP 21) [2] to limit the increase in the average global temperature to 2 °C by the end of this century. Realizing the ambitious target set at the Paris agreement would require substantial effort in multiple platforms including the reduction of carbon intensive energy sources, enhancing carbon dioxide capture and utilization technologies, and improving the efficiency of current energy devices.

The International Energy Agency recently published a report [3] that recommended a pathway to meet various climate and air quality objectives through sustainable development. According to that report, one of the key enablers for sustainable development scenario is enhancing the capability of energy production devices in industry, transportation, and buildings to burn various fuels, which has been labelled as "Fuel Switching". With regards to the transport sector, fuel flexible internal combustion IC engines are attractive to solve many issues related to sustainability. First of all, it will pave the way for refineries to follow fuel production pathways with the lowest carbon intensity and cost without being restricted to adhere to exhaustive list of fuel standards. In addition, such flexibility offers protection for consumers from fluctuation in fuel prices where they can seek the cheapest fuel available at any given time. In the UK, the fluctuation of petrol and diesel prices between 2010-2019 was approximately 30% [4]. Driven by the aforementioned reasons, this paper is part of a project to design an engine configuration that is compatible to burn various types of fuels.

The capability of engines to adapt to fuels with different octane qualities has been studied by Christensen et al. [5] focusing in HCCI type combustion and by Manente et al. [6] who explored the extent of PPC operation. Christensen et al. [5] established a map that defines the required intake air temperature and compression ratio to reach optimum CA50, which depends on the fuel's octane number. According to Manente et al. [6], the engine can be considered flexible for fuels with RON < 70 where they do not encounter issues with auto-ignition. In addition, Paz et al. [7] examined the combustion characteristics

of diesel and gasoline in heavy-duty engine at high loads. It was concluded that the combustion of both fuels is dominated by diffusion. In a recent work by Xu and Wang [8], the Monte Carlo statistical method was used to study the combustion chemistry of multicomponent real fuels. The study concludes that the high temperature chemistry of fuels, with high number of components, are statistically identical.

Our research group recently explored the engine load ranges where the burning regime is invariant regardless of the fuel octane number [9]. Fuels with octane number values that range from 0 to 100 were tested in a heavy-duty engine operated at high compression ratio ($CR = 17:1$). At high loads ($IMEP_{Net} = 20$ bar), all fuels burn in typical CI mode and share similar characteristics such as ignition delay time, combustion duration, emissions, and indicated efficiency. Following our previous work, the extent of fuel flexibility is examined in this study for similar engine configuration but at a lower CR of 11.5:1. The study aims to explore the variation in the burning regimes of fuels with different octane qualities when the intake air temperature and compression ratio are varied. At the first part of this study, the combustion characteristics of fuels was explored as the net indicated mean effective pressure ($IMEP_{Net}$) is varied from 5 to 20 bar and the temperature of the air fed to engine is altered from 20 to 80 °C. In the second part, the experimental results collected in this study at 11.5:1 CR are compared to our previous work [9] ($CR = 17:1$) using various methods to match the conditions at the engine's top dead center (TDC), the start of injection (SOI), and the intake stroke of the cycle.

Experimental Setup

Single-cylinder Engine Test Bench

Experiments were performed on a modified heavy-duty Volvo D13C500 EUV 6-cylinder engine. Five of the six cylinders were deactivated by dismounting their intake and exhaust valves to mimic the operation of a single-cylinder engine. A schematic of the experimental setup is shown in Figure 1. Table 1 summarizes the specifications of the engine utilized in this study.

Figure 1. Schematic of the single-cylinder Volvo D13C500 EUV engine setup

Table 1. Volvo D13C500 EUV single-cylinder engine specifications

Engine Type	Single-cylinder Volvo D13F
Compression ratio	11.5:1
Bore	131 mm
Stroke	158 mm
Connecting Rod	255 mm
Displacement Volume	2.13 L
Valve Configuration	SOHC, 4 valves per cylinder
Exhaust Valve Open	39° bBDC @ 0.1 mm lift
Exhaust Valve Close	13° bTDC @ 0.1 mm lift
Inlet Valve Open	18° bTDC @ 0.1 mm lift
Inlet Valve Close	18° aBDC @ 0.1 mm lift
Fuel Injection System	Common Rail, Delphi F2E, $P_{max}=2700$ bar
Max Intake Boost Pressure (absolute)	3 bar
Max Intake Temperature	120 °C
Maximum Engine Speed	2400 rpm
Swirl Ratio	0
Coolant and Lubricant Temperature	80 °C

Air from external compressor was supplied to the engine with a capacity to raise the intake pressure up to 3 bar absolute. The flow was regulated through a pneumatic valve, coupled with flow meter to monitor air flow rate. The intake system contains air heater that has the capability to increase the supplied air temperature to a maximum of 120 °C. In order to simulate the operation of the standard turbocharger, a back-pressure valve was installed to pressurize the exhaust so that its pressure is higher than the intake. The engine setup is coupled with two port fuels injectors and exhaust gas recirculation (EGR) system that weren't utilized in this study. The pressure and temperature are monitored before and after the air heater, and at the intake and exhaust manifolds using appropriate sensors. The engine in-cylinder pressure was sampled at a resolution of 0.2 CAD

using Kistler 7061C M14 piezo-electric pressure transducer mounted on the cylinder head.

The exhaust concentrations of NO, NO₂, CO, CO₂, O₂, and THC and CH₄ were measured using HORIBA MEXA-1700DEGR Motor Exhaust Gas Analyzer. AVL micro soot sensor is connected to the exhaust line to measure soot concentration at a resolution of 0.01 µg/m³. The fuel injection system is adopted from Delphi F2E concept that has high pressure distributed pump common rail system [10]. The fuel was pressurized at the common rail using a pumping injector. The injector has seven holes, each with a nominal diameter of 225 µm, and has a static flow rate of 2.15 l/min at 100 bar. The fuel return is cooled down by external heat exchanger to avoid the evaporation of volatile fuels. At each experimental condition, data was acquired for 400 fired cycles and 100 motored cycles. Throughout the experiments, the coefficient of variance (COV) of the IMEP was maintained lower than 5%.

Injection Rate Measurement

According to previous studies [11-13], characterizing the rate of fuel injection can provide key information about the hydraulic delay of the injector, the mixing period, and the shape of the heat release. Therefore, an injection rate rig has been constructed on a modified heavy-duty Volvo D13C500 EUV single cylinder engine, similar to the aforementioned one. The engine test bench was originally adapted for optical studies by extending the piston and mounting a new cylinder with side windows. These parts were dismantled and replaced by the cylindrical injection rate rig. The rig is installed below the engine cylinder-head around the injector nozzle, as illustrated in the schematic in Figure 2.

Figure 2. Schematic of the injection rate rig

An AVL GU22C piezoelectric pressure sensor was mounted at the side of the cylindrical rig and oriented towards the spray direction of one of the injector holes. The distance between the pressure sensor and the injector nozzle is approximately 5 mm. The fuel was collected and used for the calibration of the measurement to find the fuel flow rate. The signal from the pressure sensor has been filtered, pegged, corrected from charge leak, and calibrated to obtain a quantitative injection rate.

The fuel container was kept at room temperature and no back-pressure was imposed while measuring the spray momentum. An open-end pipe was installed on the fuel collector to ensure an injection at a constant atmospheric pressure. The fuel line passes through the engine block, which has elevated temperature. Hence, the temperature of the rig increases with the fuel injection but not to great extent because of the low flow rate of the fuel and the short experimental time. Therefore, it is justified to claim ambient temperature and pressure conditions inside the rig throughout the injection rate measurements. Further details about the methodology to characterize the injector used herein can be found in Aljohani et al. [14].

Fuels Properties and Composition

Since the study is focused on the octane number and its effect on the combustion characteristics of a CI engine at various conditions, it is worthwhile to examine fuels that are considered the pillars of the octane number ratings. In the standard test methods for measuring the research octane number (RON) [15] and the motor octane number (MON) [16], the knocking tendency of fuels is compared that of Primary Reference Fuels (PRFs). PRFs are binary mixtures of n-heptane and isooctane where neat n-heptane represents the lower boundary of the octane number tests with high reactivity (RON = MON = 0) and isooctane characterizes the upper boundary with high resistivity to auto-ignition (RON = MON = 100). For example, a fuel with RON or MON of 80 has the same propensity to knock as PRF80 that is a mixture of 80% isooctane and 20% n-heptane by volume.

Most of the experiments performed herein were carried out using PRFs, namely PRF0 (n-heptane), PRF30, PRF70, and PRF100 (isooctane). Since the injection system is designed for diesel grade fuels, a small trace of lubricating additive (~500 ppm) was added to each PRF tested to compensate for the lubricity deficit. In addition, diesel fuel supplied by Coryton was tested in this study as a reference case. The key properties of this Diesel fuel are summarized in Table 2.

Table 2. Coryton Diesel properties and composition

Property	Value
Cetane Number	53.8
Net calorific value (MJ/kg)	43.21
H/C ratio	1.78
Aromatics (% Wt)	25.9
Fatty Acid Methyl Ester (% v/v)	<0.1

Part I: Effect of Engine Load and Intake Air Temperature

Test Procedure

In the first part of the study, the effect of engine load and intake air temperature was studied for PRF0, PRF30, PRF70, PRF100,

and diesel. This enables exploring the change of combustion characteristics of fuels with different octane ratings as the engine load and intake air temperature are varied. Table 3 summarizes the experimental conditions for this study. The examined engine loads were net indicated mean effective pressures (IMEP_{Net}) of 5, 10, 15, and 20 bar. For each of the aforementioned load points, the intake air temperatures of 20, 50, and 80 °C were examined giving a total of 12 cases for each fuel and hence 60 in total for the six fuels. The start of injection was adapted so that the CAD, where 50% of the heat is released (CA50), is fixed at 5 CAD aTDC. This value of CA50 is considered to be the optimum for achieving the highest indicated efficiency in CI mode according to Leermakers et al. [17].

Lambda was fixed at 3.5 for IMEP_{Net} of 5 and 10 bar but then had to be reduced to 2.1 for IMEP_{Net} = 15 bar and 1.8 for IMEP_{Net} = 20 bar due to the limitation of air supply. For the 20 °C intake air temperature case, the intake pressure was maintained at 1.5 bar for IMEP_{Net} = 5 bar, 2.5 bar for IMEP_{Net} = 10 and 15 bar, and 2.8 bar for IMEP_{Net} = 20 bar, leading to peak motored in-cylinder pressure of 42, 72, and 83 bar, respectively. The boost level was slightly adjusted for higher intake temperatures (50 and 80 °C) to maintain the same lambda. The fuel rail pressure was elevated depending on the boost level. The engine was operated at 1200 rpm. The back-pressure valve was adjusted so that the difference between the exhaust and intake pressures was ~0.2 bar. This pressure difference is intended to mimic the real pressure drop across a turbocharger.

Table 3. Experimental conditions of the engine load and intake air temperature analysis

IMEP _{Net} [bar]	5	10	15	20
T _{intake} [°C]	20, 50, and 80 (for each load)			
Fuels Tested	PRF0, PRF30, PRF70, PRF100, and Diesel			
P _{intake} [bar]	~1.5	~2.5	~2.5	~2.8
CA50 [aTDC]	5			
Engine Speed [rpm]	1200			
Rail Pressure [bar]	700	1200	1200	1400
Lambda	3.5	3.5	2.1	1.8
P _{exhaust} - P _{intake} [bar]	~ 0.2			

Injection Rate Measurements

The injection rate rig has been utilized to characterize the injector used for this experimental campaign. An extensive characterization of the injector was performed in Aljohani et al. [14] using diesel fuel. The sensitivity of the injection rate shape to different fuels has been examined by testing diesel and n-

heptane. For each load point described in Table 3, the injection rate was measured by inducing the same rail pressure and injection duration settings as the diesel and n-heptane experimental cases. We noticed that there is no pronounced alteration in the injection rate between diesel and n-heptane fuels. Therefore, diesel fuel was used to model the load sweep conditions of isooctane. Table A1 in the appendix highlights the rail pressure and injection duration for the loads examined in this study. Figure 3 illustrates the injection rates coupled with the injector signal for all the characterized cases. The results obtained herein were utilized to find the time lag between the energizing timing of the injector and the effective SOI, which is defined as the hydraulic delay of the injector. This parameter is useful to identify the actual CAD where fuel is injected, especially for the cases in which the injection has not been characterized (PRF30 and PRF70). Illustration of the method to calculate the hydraulic delay is shown in Figure A1. Figure 4 shows the hydraulic delay of the injector for all the characterized cases. The average of the hydraulic delay at each injection pressure leads to 3.3 CAD at P_{rail} = 700 bar, 3.0 CAD at P_{rail} = 1200 bar, and 2.8 CAD at P_{rail} = 1400 bar.

Figure 3. Injection rate measurements at various loads for n-heptane, isooctane, and diesel (* the fuel used for characterizing isooctane cases is diesel)

Figure 4. Hydraulic delay of the injector at various loads for n-heptane, isooctane, and diesel (* the fuel used for characterizing isooctane cases is diesel)

Results and Discussion

Rate of Heat Release (RoHR)

The rate of heat release (RoHR), the motored pressure, and the injector signal for the studied combinations of engine loads and intake air temperatures are shown in: Figures 5 to 7 for $IMEP_{Net} = 5$ bar; Figures 8 to 10 for $IMEP_{Net} = 10$ bar; Figures 11 to 13 for $IMEP_{Net} = 15$ bar; and Figures 14 to 16 for $IMEP_{Net} = 20$ bar. The figures illustrate for each engine load the slight increase in the maximum motored pressure when increasing the intake air temperature to maintain the same lambda. For the lowest load tested (5 bar $IMEP_{Net}$), most of the fuels burn in partially premixed combustion (PPC) mode, as the combustion starts fairly at the end of injection. The RoHR has the typical shape of a PPC mode with a high peak and not that much of a long tail representative of diffusion-controlled combustion. The temperature and pressure conditions were moderate so that neat isooctane did not burn even when T_{intake} was increased to 120° C. Although the SOI are very different, all the fuels share similar RoHR shapes and phasing at $T_{intake} = 20^\circ\text{C}$. Therefore, as long as the fuel is ignitable, all the fuels burn in PPC mode. PRF70 exhibits some low temperature reactivity at low temperatures indicating a negative temperature coefficient (NTC) behavior. The NTC regime is no longer observed at higher intake air temperatures. At $T_{intake} = 50^\circ\text{C}$, the RoHR has a more spread shape while the amplitude of the premixed spike increases with higher octane number fuels. Most of the fuels continue to burn in PPC mode. However, a diffusion tail starts to appear in n-heptane and diesel cases. An increase of T_{intake} to 80° C boosts the reactivity of most fuels to burn in typical CI mode. The only fuel that retains PPC burning characteristic is PRF70. The spread in the SOI to achieve optimum CA50 shrinks compared to the $T_{intake} = 20^\circ\text{C}$ case. It is worth noting that diesel has a higher reactivity than n-heptane.

Moving towards higher load point ($IMEP_{net} = 10$ bar), the conditions were adequate for isooctane combustion. There is a wide spread in the combustion characteristics that depends on the octane numbers of the fuels. Isooctane persists to burn in PPC mode for all the examined intake air temperatures. It has low temperature reactivity at $T_{intake} = 20^\circ\text{C}$ that disappears at elevated temperatures. Diesel, PRF0, and PRF30 burn as a typical CI mode while the peak of the premixed combustion decreases at elevated intake air temperatures. At $T_{intake} = 80^\circ\text{C}$, fuels that lie at the lower octane number boundary have a pronounced diffusion portion in the RoHR. PRF70 transforms from a combustion that is dominated by PPC to diffusion as the intake air temperature is increased. The variation in the SOI to reach the optimum CA50 is minimal for all the fuels except for isooctane at $T_{intake} = 20^\circ\text{C}$ where the SOI is advanced significantly.

The presence of PPC regime becomes limited as the load is increased to 15 bar (Figures 11 to 13). At $T_{intake} = 20^\circ\text{C}$, only PRF100 exhibits a pronounced premixed spike whereas the combustion of the other fuels is driven by diffusion. Similar trend is observed as the amplitude of the premixed spike is correlated to the fuel's octane rating. A diffusion tail at the RoHR starts to appear for PRF100 as the intake temperature is increased to 50° C. After that, the tail becomes more pronounced with the elevation of temperature to 80° C. The SOI maintains its value regardless of the fuel used.

At the highest load point tested in this study ($IMEP_{Net} = 20$ bar), all the fuels burn in typical CI mode without any sign of fuels burning in PPC mode. At $T_{intake} = 20^\circ\text{C}$, similar SOIs are used for all fuels to reach the optimum CA50. Nevertheless, the portion of diffusion to premixed combustion is quite different depending on the fuel octane number. The diffusion portion increases as the resistivity to auto-ignition reduces. The variation in diffusion to premixed combustion between fuels shrinks significantly as the intake air temperature increases. The fuels appear to share similar RoHR shapes and phasing at $T_{intake} = 80^\circ\text{C}$. However, the extent of fuel flexibility observed herein does not replicate a previously reported measurements at higher compression ratio (CR17:1) [9] where the RoHR of all the tested PRFs lied at the top of each other at $IMEP_{Net} = 20$ bar even with lower intake air temperature of 20° C.

Figure 5. Rate of heat release, motored pressure trace and injector signal of different fuels at $IMEP_{Net} = 5$ bar and $T_{intake} = 20^{\circ}C$

Figure 8. Rate of heat release, motored pressure trace and injector signal of different fuels at $IMEP_{Net} = 10$ bar and $T_{intake} = 20^{\circ}C$

Figure 6. Rate of heat release, motored pressure trace and injector signal of different fuels at $IMEP_{Net} = 5$ bar and $T_{intake} = 50^{\circ}C$

Figure 9. Rate of heat release, motored pressure trace and injector signal of different fuels at $IMEP_{Net} = 10$ bar and $T_{intake} = 50^{\circ}C$

Figure 7. Rate of heat release, motored pressure trace and injector signal of different fuels at $IMEP_{Net} = 5$ bar and $T_{intake} = 80^{\circ}C$

Figure 10. Rate of heat release, motored pressure trace and injector signal of different fuels at $IMEP_{Net} = 10$ bar and $T_{intake} = 80^{\circ}C$

Figure 11. Rate of heat release, motored pressure trace and injector signal of different fuels at $IMEP_{net} = 15$ bar and $T_{intake} = 20^{\circ}C$

Figure 14. Rate of heat release, motored pressure trace and injector signal of different fuels at $IMEP_{net} = 20$ bar and $T_{intake} = 20^{\circ}C$

Figure 12. Rate of heat release, motored pressure trace and injector signal of different fuels at $IMEP_{net} = 15$ bar and $T_{intake} = 50^{\circ}C$

Figure 15. Rate of heat release, motored pressure trace and injector signal of different fuels at $IMEP_{net} = 20$ bar and $T_{intake} = 50^{\circ}C$

Figure 13. Rate of heat release, motored pressure trace and injector signal of different fuels at $IMEP_{net} = 15$ bar and $T_{intake} = 80^{\circ}C$

Figure 16. Rate of heat release, motored pressure trace and injector signal of different fuels at $IMEP_{net} = 20$ bar and $T_{intake} = 80^{\circ}C$

Combustion Characteristics and Efficiency

Ignition delay time (IDT) is a commonly used indicator to evaluate the auto-ignition tendency of fuels. Another useful parameter that describes the level of fuel and air mixing would be the mixing period. In this study, signal-based IDT is defined as the CAD lag between the SOI and the initiation of the auto-ignition event identified as the lowest CAD with positive RoHR value. The hydraulic delay obtained from the injector characterization (Figure 4) is subtracted from the signal-based IDT to find the actual IDT. The reported IDT for the studied fuels herein are actual IDT. For PRF30 and PRF70, the hydraulic delay was estimated by averaging the delays from the injector characterization using n-heptane and diesel. The mixing period is defined as the difference in crank angle degrees between the end of injection and the start of heat release. Therefore, fuels burning with negative mixing period release heat while fuel is still being injected to the combustion chamber. It is worthwhile to note that the mixing period of PRF30 and PRF70 are not reported, as the characterization of the injector did not include these fuels. In addition, for the cases where fuels exhibit two stage heat release, the start of the main combustion event (second stage) is considered as the start of RoHR for both IDT and mixing period calculations. Figure A1 in the appendix exemplifies the method to calculate the actual IDT and the mixing period.

The IDT of the cases studied herein are illustrated in Figure 17. As expected, the charge reactivity elevates with the increase of either the engine load or the intake air temperature, as the IDT decreases. Fuels with higher octane numbers are more sensitive to the parameter variation performed in the current study. Diesel, PRF30, and PRF70, which exhibit a combustion dominated by diffusion, have a minimal change in IDT with intake temperature elevation. At the highest load tested, the maximum difference between the IDT of fuels is approximately 4 CAD at $T_{\text{intake}} = 20^\circ\text{C}$. The difference shrinks to 3 CAD when the intake temperature is increased to 80°C , making it challenging to reach fuel flexible conditions in terms of IDT at low CR engines.

Figure 17. Ignition delay time for the tested fuels at various loads and intake air temperatures

Figure 18 highlights the change in the mixing period (start of combustion – end of injection) as a function of load and intake air temperature. The mixing period linearly decrease for the reported fuels as a function of engine load. Similar but less pronounced decrease in the mixing period is observed when increasing the intake air temperature when compared to engine load elevation. Isooctane combustion at $T_{\text{intake}} = 20^\circ\text{C}$ and $\text{IMEP}_{\text{Net}} = 10$ bar is the only case where the mixing period has a positive value. This indicates that attaining pure PPC mode is challenging for loads beyond 10 bar and air temperatures beyond 20°C even with the low CR ratio examined in this study. The difference in the mixing period between the high octane isooctane and the low octane n-heptane and diesel shrinks at the highest loads tested.

Figure 18. Mixing period for the tested fuels at various loads and intake air temperatures

The gross indicated efficiency of the different fuels cases are compared in Figure 19 at different engine loads and intake air temperatures. There is no apparent correlation between the fuel's octane number and the indicated efficiency. The maximum difference in efficiency between the fuels is limited

to 4%. The increase in the heat transfer losses with intake air temperature elevation explains the minor decrease in efficiency. Most of the fuels follow the general efficiency trend in the load map. The efficiency increases as a function of load until a certain intermediate load point where the efficiency starts to drop due to heat transfer losses. All of the fuels follow the aforementioned trend other than PRF30 at IMEP_{Net} = 20 bar, in which it encounters an increase. Pure isooctane consistently appears at the highest boundary of efficiency, highlighting the correlation between the high levels of premixing with efficiency gain. Overall, the results do not highlight a significant change in the gross indicated efficiency when shifting fuels with different octane quality. Few differences in efficiency are observed, which can be attributed to the adaptation of SOI for optimum CA50.

Figure 19. Gross indicated efficiency at the effect of engine load and intake air temperature analysis

Emissions

The engine emissions associated with load and intake air temperature sweep cases are presented in Figures 20 to 23. Moving towards higher engine loads causes a steady increase in the NO_x emissions for all the fuels until IMEP_{Net} = 15 bar where the NO_x emissions start to plateau. The actual quantity of NO_x produced at load of 20 bar is actually higher than 15 bar but the scalability of increase is reduced by the normalization with the engine load as reported in specific emissions. It appears that engine loads of 15 and 20 bar share similar in-cylinder temperature conditions, which is the driver for thermal NO_x production. The increase of intake air temperature causes a moderate elevation in the NO_x created in the exhaust as expected. The fuels share similar NO_x emissions except for isooctane at T_{intake} = 20 °C and IMEP_{Net} = 10 and 15 bar. The combustion of isooctane at these conditions has a pronounced premixed spike (Figures 8 and 11), creating a sharper rise in RoHR and hence NO_x emissions rise.

The CO concentration rapidly drops for loads beyond 5 bar where the conditions stimulate CO to CO₂ conversion. This trend is observed with the heating of intake air. Fuels that are

dominated by PPC mode such as PRF70 and PRF100 at low loads create high amounts of CO. This is due to their short combustion duration that is insufficient to oxidize CO. The study reports small overall specific concentration of unburned hydrocarbons. This is justified as most of fuels burn in CI mode and so they have diesel-type combustion efficiency. As reported by the discussion of CO emissions, the unburned hydrocarbons increase for high octane fuels, which burn in PPC mode at low loads. This is attributed to the low temperature nature of PPC mode, which stands against reducing unburned hydrocarbons. The soot emissions of all PRFs follow each other where they encounter an increase as a function of load. There is no apparent correlation between the soot emissions of fuels with different octane ratings. The only outlier is the soot emissions of diesel which is higher than all the PRFs at all the temperature and load points. The aromatics content of diesel is the main precursor for polycyclic aromatic hydrocarbons (PAHs) which agglomerates to create soot particles. The high soot concentration observed at IMEP_{Net} = 5 bar can be attributed to the low injection pressure of the fuel (P_{rail} = 700 bar), creating poor fuel atomizing. Overall, all the emissions for the tested fuels appear to share similar concentrations at high loads even with the slight variation in their combustion regimes.

Figure 20. Specific NO_x emissions of PRFs and diesel at different engine loads and intake air temperatures

Part II: Effect of Compression Ratio

Test Procedure

Following our previous work [9] on studying the fuel flexibility of a CI engine at a CR of 17:1, we investigated the extent of that flexibility at a lower CR of 11.5:1. Since the in-cylinder pressure and temperature profiles cannot be matched for engines with different compression ratios, it is challenging to establish the same basis for comparing fuels. Therefore, we used different methodologies to compare the combustion characteristics of fuels at different compression ratios. These methodologies include matching the TDC, SOI, and intake conditions at CR11.5:1 as the ones analyzed in the CR17:1. Detailed description of these methodologies is presented below.

Low Engine Load ($IMEP_{Net} = 5$ bar)

The variation of the burning regimes of n-heptane (PRF0) and isooctane (PRF100) are investigated at CR11.5 and $IMEP_{Net} = 5$ bar. Results from previous work at CR17:1 [9] where the intake air pressure and temperature were respectively maintained at 1.5 bar and 20°C are used as reference cases. For the fuels tested, the SOI has been adapted so that the CA50 is maintained at 5 CAD aTDC. The first analysis proposed in this study using the lower CR aims to match the temperature and pressure conditions at TDC. This will be labeled in the coming sections as “TDC Match”. The average in-cylinder temperature is estimated using isentropic relations for an ideal gas considering variable gamma, which was obtained from NASA Polynomials of air. In order to match the TDC conditions at high CR, air must be fed at an intake pressure of 2.5 bar and intake temperature of 76°C for both n-heptane and isooctane.

The second tested methodology is to match the temperature condition when the fuel is injected, specifically at SOI. This analysis will be referred to as “SOI Match”. According to the temperature calculations, the elevation of intake temperature needed to match the SOI conditions of CR17:1 is 70°C. The intake pressure was maintained at 2.5 bar, which leads to the same TDC pressure as CR17:1 case. For the third analysis, the intake temperature and pressure are matched to the ones of CR17:1 and this is referred to as the “Intake Match”. Hence, the air was fed to the engine at 1.5 bar pressure and 20°C temperature. The motored in-cylinder pressure and temperature profiles of the tested conditions are shown in Figures 24 and 25.

Table 4 summarizes the experimental conditions of all the analysis approaches described above. In all cases, the same fuel quantity per cycle is injected. This leads to different lambda values since the air density varies for each case. The rail pressure was maintained for all the cases at 700 bar and the difference between the exhaust and intake pressures is maintained at approximately 0.2 bar. The engine speed is regulated at 1200 rpm for all cases.

Figure 21. Specific CO emissions of PRFs and diesel at different engine loads and intake air temperatures

Figure 22. Specific unburned hydrocarbon emissions of PRFs and diesel at different engine loads and intake air temperatures

Figure 23. Specific soot emissions of PRFs and diesel at different engine loads and intake air temperatures

Figure 24. Motored pressure trace of the tested conditions at the effect of compression ratio analysis for $IMEP_{Net} = 5$ bar (CR17:1 case obtained from AlRamadan et al.[9])

Figure 25. Motored temperature estimation of the tested conditions at the effect of compression ratio analysis for $IMEP_{Net} = 5$ bar (CR17:1 case obtained from AlRamadan et al. [9])

Table 4. Experimental conditions of the effect of compression ratio analysis at $IMEP_{Net} = 5$ bar

	CR17:1	CR11.5:1 TDC Match	CR11.5:1 SOI Match	CR11.5:1 Intake Match
T_{intake} [°C]	20	76	70	20
P_{intake} [bar]	1.5	2.5	2.5	1.5
Tested Fuels	PRF0 (n-heptane) and PRF100 (isooctane)			
P_{rail} [bar]	700			
Engine Speed	1200 rpm			
Lambda	3.5	5.6 PRF0 5 PRF100	5.7 PRF0 5 PRF100	3.5
$P_{exhaust} - P_{intake}$ [bar]	~ 0.2			

Intermediate Engine Loads ($IMEP_{Net} = 10$ and 15 bar)

Similar to the $IMEP_{Net} = 5$ bar, n-heptane and isooctane are compared at higher engine loads of 10 and 15 bar. At these loads, it is challenging to match the pressure conditions of previous study [9]. The current capacity of the engine air supply is limited to 3 bar absolute pressure. This leads at CR11.5:1 engine configuration to a peak motored pressure trace of 87 bar whereas the pressure peaks at 120 bar in the CR17:1 case for $IMEP_{Net}$ of 10 and 15 bar. Hence, the first analysis considered is to match SOI temperatures while fixing the intake pressure at its ceiling value of 3 bar. According to the temperature calculations obtained from isentropic compression, the intake air temperature needed for n-heptane is 72°C and for isooctane is 71°C at an engine load of 10 bar. At $IMEP_{Net} = 15$ bar, the intake temperature requirements reduce to 69°C for n-heptane and 67°C for isooctane. The second methodology considered to establish basis for comparison between data collected at different compression ratios is by matching the intake pressure and temperature conditions. For this analysis, air is fed to the engine at 2.5 bar pressure and 20°C temperature. The motored temperature and pressure profiles for the cases considered at engine loads of 10 and 15 bar for n-heptane are illustrated in Figures 26 and 27. Figures A2 and A3 in the appendix show the motored temperature and pressure traces of isooctane.

Tables 5 and 6 summarize the test conditions of the $IMEP_{Net} = 10$ and 15 bar cases. The lambda of CR17:1 and the CR11.5:1 intake match cases shares the same value whereas it increases in magnitude at the SOI match case due to the increase in air density. The rail pressure of all cases is kept constant at 1200

bar and the back-pressure valve is adjusted to maintain a positive pressure difference between the exhaust and intake valves. The engine was operated at 1200 rpm speed.

Figure 26. Motored pressure and temperature traces of the tested conditions at the effect of compression ratio analysis for n-heptane at $IMEP_{Net} = 10$ bar (CR17:1 case obtained from AlRamadan et al. [9])

Figure 27. Motored pressure and temperature traces of the tested conditions at the effect of compression ratio analysis for n-heptane at $IMEP_{Net} = 15$ bar (CR17:1 case obtained from AlRamadan et al. [9])

Table 5. Experimental conditions of the effect of compression ratio analysis at $IMEP_{Net} = 10$ bar

	CR17:1	CR11.5:1 SOI Match	CR11.5:1 Intake Match
Fuels Tested	PRF0 (n-heptane) and PRF100 (isooctane)		
T_{intake} [°C]	20	72 (PRF0) 71 (PRF100)	20
P_{intake} [bar]	2.5	3.0	2.5
P_{rail} [bar]	1200		
Engine Speed [rpm]	1200		
Lambda	3.5	3.6	3.5
$P_{exhaust} - P_{intake}$ [bar]	~ 0.2		

Table 6. Experimental conditions of the effect of compression ratio analysis at $IMEP_{Net} = 15$ bar

	CR17:1	CR11.5:1 SOI Match	CR11.5:1 Intake Match
Fuels Tested	PRF0 (n-heptane) and PRF100 (isooctane)		
T_{intake} [°C]	20	69 (PRF0) 67 (PRF100)	20
P_{intake} [bar]	2.5	3.0	2.5
P_{rail} [bar]	1200		
Engine Speed [rpm]	1200		
Lambda	2.1	2.3 (PRF0) 2.2 (PRF100)	2.1
$P_{exhaust} - P_{intake}$ [bar]	~ 0.2		

Results and Discussion

Rate of Heat Release (RoHR)

The RoHR at $IMEP_{Net} = 5$ bar for the CR analysis methodologies is plotted in Figure 28. N-heptane combustion at CR17:1 that is reported by AlRamadan et al. [9] burns in CI mode with extended diffusion portion observed on the RoHR. Conversely, for the compression ratio configuration studied herein (CR11.5:1), n-heptane burns in PPC mode when conditions in the intake are matched to the CR17:1 case. This confirms that matching the intake conditions is inadequate to reproduce the combustion characteristics of fuels at different compression ratios. However, when the TDC and SOI

conditions are matched, n-heptane shares similar burning characteristics as the CR17:1 case. The apparent variations are the retard in the combustion phasing and the increase in the amplitude of premixed spike observed for the CR11.5 TDC and SOI match cases. This is expected as the level of dilution increase an order of magnitude with TDC and SOI match cases, which reduces slightly the reactivity of the mixture (Lambda values at Table 4). The air fed to the engine at higher temperature when TDC conditions are matched compared to the SOI match case, which justifies the advanced combustion phasing of the TDC match analysis. All the examined cases with isooctane at $IMEP_{Net} = 5$ bar burns in PPC mode with the exception of the intake match case. When matching the pressure and temperature conditions at the intake of CR17:1 case, combustion is not initiated since isooctane has a large ignition delay time. The TDC and SOI match cases follow similar trends highlighted in the n-heptane case where the phasing is retarded compared to the CR17:1 case because of lambda increase. Overall, the combustion characteristics of both n-heptane and isooctane at CR17:1 are similar to the ones reported by either TDC or SOI match analyses.

Figure 28. RoHR for the different compression ratios tested at $IMEP_{Net} = 5$ bar

Figures 29 and 30 illustrate the effect of compression ratio change for n-heptane and isooctane at $IMEP_{Net}$ of 10 and 15 bar, respectively. For both load points, n-heptane burns with a pronounced diffusion portion observed on the RoHR. The initial premixed spike has the lowest amplitude at the CR17:1 case and $IMEP_{Net} = 10$ bar but then disappears at $IMEP_{Net} = 15$ bar. Matching the temperature conditions at the SOI leads to a comparable behavior to the CR17:1 case. In contrast, a greater deviation between CR17:1 and CR11.5 SOI match is observed for isooctane. At $IMEP_{Net} = 10$ bar, the temperature and pressure conditions were moderate so that isooctane burn in PPC mode at the SOI match analysis whereas it burns as conventional diesel combustion at CR17:1. This can be attributed to the fact that SOI analysis matches only the temperature conditions and under predicts the pressure settings because of the air compressor limitation of the setup. The deviation in the RoHR for isooctane between SOI match and

CR17:1 cases shrinks at $IMEP_{Net} = 15$ bar, where a diffusion tail starts to appear at the SOI match case. In summary, matching the temperature conditions at the SOI appears to be adequate for low octane fuels to burn in similar fashion at different compressions ratios. Higher deviations in the combustion characteristics at different compression ratios are reported for higher octane fuels that have longer ignition delay times.

Figure 29. RoHR for the different compression ratios tested at $IMEP_{Net} = 10$ bar

Figure 30. RoHR for the different compression ratios tested at $IMEP_{Net} = 15$ bar

Combustion Characteristics and Emissions

An overview of the change in IDT of n-heptane and isooctane when burned at different compression ratios (CR17:1 and CR11.5:1) is shown in Figure 31. It is apparent that matching the intake conditions of CR17:1 is not adequate to replicate the combustion characteristics at CR11.5. The deviations elevate with higher engine loads especially for isooctane where it transforms from CI mode at CR17:1 to PPC mode at CR11.5:1. Conversely, matching either the TDC or SOI conditions captures with good accuracy the IDT at CR17:1 for $IMEP_{Net} =$

5 bar. At higher engine loads, the gap between the motored pressure at fuel injection points between CR11.5 and CR17:1 at the SOI match analysis stands against reproducing the ignition delay time values.

Figure 31. Ignition delay time of the cases tested at the effect of compression ratio analysis

Traces of NO_x, CO, unburned hydrocarbons, and soot for the compression ratio analysis for IMEP_{Net} of 5, 10 and 15 bar are highlighted in Figures 32 to 34. The NO_x emissions at IMEP_{Net} = 5 bar share similar values for all the examined conditions. However, the concentration of NO_x elevates slightly for isooctane at higher loads when moving from CR17:1 case to SOI match case. This occurs because of the PPC nature of isooctane combustion at these loads points (Figure 29 and Figure 30) where the sharp premixed spike in the RoHR causes thermal NO_x production. As the combustion of n-heptane in the SOI and TDC match cases can be characterized as CI mode, low concentration of CO emissions for all load points are observed, which follows conventional diesel combustion. In contrast, the SOI and TDC match cases over-predicts the CO concentration for isooctane combustion compared to CR17:1 at IMEP_{Net} = 5 bar. The retarded combustion phasing that occurs with SOI and TDC analysis make isooctane burn in milder conditions that stand against CO oxidation. In fact, the same cause justifies the slight increase in the unburned hydrocarbon concentrations at CR11.5:1. The CO and unburned hydrocarbons concentration obtained from the SOI and CR17:1 analysis share similar values at elevated loads due the pronounced diffusion part of the RoHR. For all the loads, the soot concentrations are considered to be low except for n-heptane combustion at IMEP_{Net} = 5 bar and CR17:1. The potential cause of this behavior is the non-premixed burning of n-heptane and the poor atomizing of fuel at low rail pressures ($P_{rail} = 700$ bar). To summarize, matching either TDC or SOI pressure and temperature conditions to compare engine cases at different compression ratios appears to be adequate for comparable emissions results at high loads. The main deviation occurs at lower loads especially for high-octane fuels, which tends to shift from PPC mode at lower compression ratios to CI at higher compression ratios.

Figure 32. Specific emissions of the cases tested at the effect of compression ratio analysis at IMEP_{Net} = 5 bar

Figure 33. Specific emissions of the cases tested at the effect of compression ratio analysis at IMEP_{Net} = 10 bar

Figure 34. Specific emissions of the cases tested at the effect of compression ratio analysis at IMEP_{Net} = 15 bar

Conclusions

An experimental campaign was carried out on a single-cylinder heavy duty engine to study the effect of compression ratio and intake air temperature on altering the combustion characteristics of fuels with various octane ratings at different loads. The study initially investigates the sensitivity of fuels to the variation of intake air temperature at various engine loads. After that, it compares the extent of fuel flexibility measured herein at compression ratio of 11.5:1 to results reported in previous work [9] with engine set with 17:1 compression ratio. From all of these analyses, claiming absolute fuel flexibility for engines with low compression ratios is not possible as there is ignitability issues with the high octane fuels at lower loads and variation in the premixed/diffusion portion of the RoHR at higher loads that causes slight deviations in the emissions and indicated efficiency. Incorporating an ignition system to the low compression ratio engine is a major enabler to achieve fuel flexible operation where even the high octane fuels will burn at low loads. Additionally, the following conclusions has been drawn:

- As long as the fuel is ignitable, fuels tend to burn in PPC mode at low IMEP and intake air temperature with the CR11.5:1 engine configuration.
- It was previously shown that all fuels 0 to 100 octane number burned in similar manner at loads higher than 15 bar IMEP even with low inlet temperature. Reducing the compression ratio from 17:1 to 11.5:1 changed the trend dramatically. The level of fuel reactivity becomes important.
- There is a direct correlation between the portions of premixed to diffusion regions in the RoHR with the fuel's octane number at the examined compression ratio in this study (CR11.5:1).
- At CR11.5:1, most of the fuels favors PPC mode but starts to become driven by diffusion at IMEP = 20 bar.
- Although there was a slight variation in the combustion regimes of fuels at high loads with CR11.5:1 configuration, fuels with different octane ratings share similar emissions and indicated efficiencies.
- Matching the temperature conditions at the start of injection seems to be adequate for low octane fuels to burn in similar fashion even for engines with different compression ratios. However, fuels with higher resistance to auto-ignition experience higher deviations in the conditions because of their longer ignition delay times.

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PPC	Partially premixed combustion
PRF	Primary reference fuel
RoHR	Rate of heat release
RON	Research octane number
SOHC	Single overhead camshaft
SOI	Start of injection
TDC	Top dead center

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Definitions/Abbreviations

CAD	Crank angle degree
CA50	CAD at which 50% of the heat is released
EGR	Exhaust gas recirculation
HCCI	Homogenous charge compression ignition
IC	Internal combustion
IDT	Ignition delay time
IMEP_{Net}	Net indicated mean effective pressure
MON	Motor octane number

Appendix

Figure A1. Illustration to calculate hydraulic delay, ignition delay time, and mixing period from adopted from isooctane case at $T_{\text{intake}} = 20^{\circ}\text{C}$ and $\text{IMEP}_{\text{Net}} = 10 \text{ bar}$

Figure A2. Motored pressure and temperature traces of the tested conditions at the effect of compression ratio analysis for isooctane at $\text{IMEP}_{\text{Net}} = 10 \text{ bar}$ (CR17:1 case obtained from AlRamadan et al.[9])

Figure A3. Motored pressure and temperature traces of the tested conditions at the effect of compression ratio analysis for isooctane at $IMEP_{Net} = 15$ bar (CR17:1 case obtained from AlRamadan et al. [9])

Table A1. Rail pressure and injection duration settings for injector characterization at the examined load points

$IMEP_{Net}$ [bar]	Injection Settings	N- heptane	Isooctane*	Diesel
5	P_{rail} [bar]	700		
	Injection Duration [μs]	1085	1200	1085
10	P_{rail} [bar]	1200		
	Injection Duration [μs]	1350	1340	1235
15	P_{rail} [bar]	1200		
	Injection Duration [μs]	1980	2100	1980
20	P_{rail} [bar]	1400		
	Injection Duration [μs]	2430	2550	2430

* The fuel used for characterizing isooctane cases is diesel