

Experimental Investigation of Natural Gas-Diesel Dual-Fuel RCCI in a Heavy-Duty Engine

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ABSTRACT

Studies have shown that premixed combustion concepts such as PCCI and RCCI can achieve high efficiencies while maintaining low NO_x and soot emissions. The RCCI (Reactivity Controlled Compression Ignition) concept use blending port-injected high-octane fuel with early direct injected high-cetane fuel to control auto-ignition. This paper describes studies on RCCI combustion using CNG and diesel as the high-octane and high-cetane fuels, respectively. The test was conducted on a heavy-duty single cylinder engine. The influence of injection timing and duration of the diesel injections was examined at 9 bar BMEP and 1200 rpm. In addition, experiments were conducted using two different compression ratios, (14 and 17) with different loads and engine speeds. Results show both low NO_x and almost zero soot emissions can be achieved but at the expense of increasing of unburned hydrocarbon emissions which could potentially be removed by catalytic after-treatment. CA50 generally occurred before TDC when using a compression ratio of 17. While the CA50 could be shifted to slightly after TDC by increasing the amount of EGR, this would lead to excessive HC emissions. A lower compression ratio of 14 was therefore used to retard the CA50 while maintaining acceptable UHC emissions.

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INTRODUCTION

Rising fuel costs, increasingly stringent emissions standards, and the need to reduce atmospheric greenhouse gas levels mean that it is necessary to increase engine efficiency and reduce exhaust emissions. One way of reducing CO_2 emissions would be to develop engines that burn "low-carbon" fuels such as CNG or bio-methane. CNG is a particularly attractive alternative to conventional fossil fuels because of the booming shale gas production in the US.

Many studies on Homogeneous Charge Compression Ignition (HCCI) and Premixed Charge Compression Ignition (PCCI) have shown that these combustion strategies can reduce soot and NO_x emissions when moderate or high levels of EGR are used [1,2,3,4]. Unfortunately, they also make it difficult to control combustion phasing and present challenges under high load conditions. These issues can potentially be avoided by using the Reactivity Controlled Compression Ignition (RCCI) combustion concept, which is also known as dual-fuel PCCI.

The RCCI combustion is controlled mainly by fuel's reactivity. This concept was first proposed by Inagaki [5] and elaborated by Reitz and coworkers [6-7]. It uses blending port-injected high-octane fuel with early direct injected high-cetane fuel in order to handle auto-ignition and tune the reactivity of the combustion mixture. The work of the Reitz group has largely focused on a split diesel injection strategy,

which involves an initial injection targeting the squish region of the combustion chamber followed by a second injection targeting the bowl region.

Both compression ignition and spark ignition natural gas fuelled engines for heavy duty applications have been studied. Compression ignition engines of this sort normally rely on conventional dual-fuel combustion with the combustion of the NG being initiated by the autoignition of the direct-injected diesel spray. To date, there have only been three published studies on NG/diesel RCCI combustion. The first was conducted by Reitz group [8] and focused on comparing single- and split-injection combustion schemes. The second was conducted by Doosje et al. [9], who investigated NG/Diesel RCCI combustion in a full-scale engine. The third was done by Reitz group again [10] and mainly made a comparison between NG/diesel and Gasoline/diesel RCCI combustion with single diesel injection strategy.

The goal of this study was to investigate an early split diesel injection strategy and the influence of compression ratio on the performance and emissions of natural gas/diesel RCCI combustion. In this paper, the effects of varying the first injection SOI timing and durations of the first and second diesel injection were investigated. The similar split diesel injection strategy was used in all of the experiments. The primarily focus will be on experimental rather than computational investigations for this paper: simulation-based studies will be presented in due course after the relevant models have been refined.

EXPERIMENTAL SETUP

All experiments were conducted using a single-cylinder, four-stroke, compression ignition research engine that was designed to mimic the characteristics of Volvo's heavy duty D12C engine. The engine was controlled using an AVL PUMA 5 test bed system; its most important specifications are listed in Table 1 and a schematic layout of the engine test cell is shown in Figure 1. The intake air goes through a screw compressor to establish the required boost pressure, and then passes through a dryer to remove water, after that an air conditioner to heat the air as required and finally a plenum to eliminate pressure oscillations. The recirculated exhaust gas is cooled by the engine's cooling water before entering the intake system. The temperature of the exhaust gas after cooling depends on the amount of EGR used and the engine load.

The exhaust concentrations of NO_x, CO, O₂, intake and exhaust CO₂ and UHC were measured using an AVL AMA i60 exhaust measurement system. The amount of EGR applied was calculated by measuring the intake and exhaust CO₂ concentrations. The detailed detector for each compound is shown in table 2. The exhaust sample first went through a preheated filter (190°C) to filter out particles, and then went through a heated sample line (190°C) to AVL AMA i60 FID and CLD detectors. The inlet temperature for FID and CLD were 190°C and 130°C, respectively. A cooldown sample from this heated sample line was fed to the rest of detectors in the AMA i60 analyzer see table 2. Soot emissions were measured using an AVL 415 smoke meter. It should be noted that this meter's accuracy is limited at low concentrations.

Table 1. Engine specifications

Engine type	Single-cylinder AVL 501
Bore [mm]	131
Stroke [mm]	150
Connecting Rod [mm]	260
Compression ratio	17:1 /14:1
Number of Valves	4
Exhaust Valve Open [*ATDC]	118.2
Exhaust Valve Close [*ATDC]	-343.4
Inlet Valve Open [*ATDC]	300.9
Inlet Valve Close [*ATDC]	-105.2

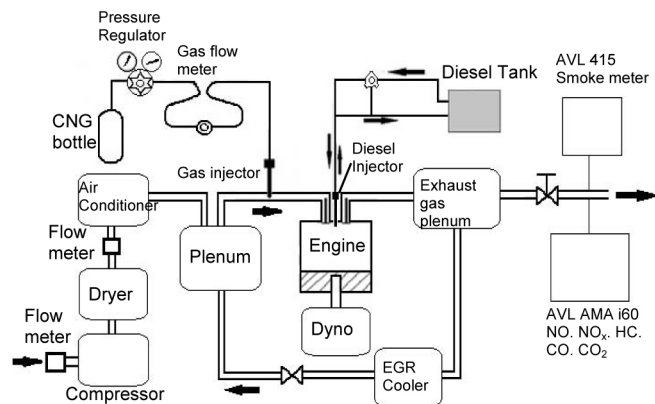


Figure 1. Layout of the engine test cell

Table 2. Exhaust gas measurement devices

NO _x	CLD i60 HHD SLQ
CO	IRD i60 CO H
CO ₂	IRD i60 CO ₂ H
O ₂	PMD i60 O ₂
HC	CUTTER FID i60 HHD
Soot	AVL 415 smoke meter

The low- and high-reactivity fuels employed in this work are CNG and diesel, respectively. The CNG composition is given in table 3. The diesel fuel selected for the test is Swedish environmental class 1 diesel (MK1). The key properties of both fuels are listed in Table 4, where MK1 denotes the MK1, 2010 specification (SS 155435) and CNG properties are calculated based on its composition. CNG density is obtained from ideal gas law; methane number is calculated based on equations in [11]; heating value can be calculated from its composition since each compound heating value is already known.

Table 3. Composition of CNG

Compound	Percent in CNG (v/v)
Methane	88.8%
Ethane	5.6%
Propane	2.4%
Isobutane	0.3%
Butane	0.5%
Pentanes	0.15%
Carbon dioxide	1.2%
Carbon monoxide	0.4%
Oxygen	0.15%
Nitrogen	0.5%

Table 4. Characteristics of the fuels

Fuel	NG	Diesel(MK1)
Viscosity [mm ² /s] (at 40°C)	-	1.4-4
Density [kg/m ³]	196.6 @ 20 MPa	800 - 830
Methane number [-]	79.88	-
Cetane number [-]	-	≥ 51
Lower heating value [MJ/kg]	46.68	43.1

To create a premixed NG/air mixture, a Volvo bus gas injector controlled by a Bosch control unit was mounted on the engine's intake manifold. The diesel fuel was delivered using a common rail type DI system that can deliver up to five injections per cycle. The specifications of the diesel injector are given in Table 5.

Table 5. Diesel injector specifications

Number of holes	5
Steady flow rate at 100 bar (L/min)	2
Hole diameter	0.256 mm
Injection pressure	100 MPa
Spray angle	145°

The cylinder pressure was recorded using a D2T Osiris data acquisition system, with a sampling interval of 0.1 CAD. The acquired cylinder pressure traces were averaged for 100 cycles.

smooth the pressure data. Heat losses were not considered in the rate of heat release calculations. Pressure traces related calculations were performed using the Osiris software.

METHODS

The diesel injection strategy used in this work is similar to the split injection strategy used by the Wisconsin group [6]. This approach makes it possible to separately manipulate the reactivity of the fuels in the squish and bowl regions of the combustion chamber. As shown in Figure 2, the first and second injections are referred to as pilot injection 1 (PI1) and pilot injection 2 (PI2), respectively. The SOI timing showed below are electrical signal. PI1 SOI timings include all the test cases in this paper; PI1 SOI timings for some cases were not presented.

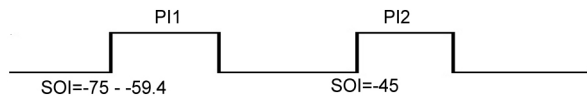


Figure 2. Diesel injection profile

Two different compression ratios were tested, 14 and 17. The initial study was to investigate the effects of varying the PI1 SOI timing and the amount of the fuel injected during PI1 and PI2 at 1200 rpm with a compression ratio (CR) of 17. Based on the results of these preliminary studies, further tests were conducted at two CRs and engine speeds of 1200, 1500 and 1800 rpm. Finally, the influence of load on the engine's performance and emissions was investigated at both CRs.

Given the high UHC and CO emissions observed in most of the experiments, the combustion efficiency was taken into account when calculating the engine's indicated thermal efficiency:

$$\eta_{thermal} = \frac{P_{indicated}}{\eta_{comb}(\dot{m}_D LHV_D + \dot{m}_{NG} LHV_{NG})} [\%]$$

$$\eta_{comb} = 100 \cdot \left(1 - \frac{\dot{m}_{CO} \cdot LHV_{CO} + \dot{m}_{HC} \cdot LHV_{HC}}{\dot{m}_D LHV_D + \dot{m}_{NG} LHV_{NG}} \right) [\%]$$

$$LHV_{HC} = \frac{\dot{m}_D LHV_D + \dot{m}_{NG} LHV_{NG}}{\dot{m}_D + \dot{m}_{NG}}$$

where LHV stands for the lower heating value, $\dot{m}$ stands for the mass flow rate, η_{comb} stands for the combustion efficiency, and the subscripts D, NG, CO, and HC refer to the diesel fuel, natural gas, carbon monoxide, and hydrocarbons, respectively.

The percentage of the total energy supplied by the NG is calculated as below:

$$EP = \frac{\dot{m}_{NG} \cdot LHV_{NG}}{\dot{m}_{NG} \cdot LHV_{NG} + \dot{m}_D \cdot LHV_D}$$

The effective expansion ratio (EER) is calculated as [12]:

$$EER = \frac{\int_{SOC}^{EOC} HHR(\theta) \cdot ER(\theta) \cdot d\theta}{\int_{SOC}^{EOC} HHR(\theta) \cdot d\theta}$$

where θ is the crank angle; $ER(\theta)$ is the instantaneous expansion ratio (the ratio of the volume at bottom dead center to the instantaneous volume); $HHR(\theta)$ is the instantaneous heat release rate, SOC is the start of combustion, and EOC is the end of combustion.

The fuel penetration is calculated based on the equation in [13].

$$L = \frac{b}{a} \cdot \sqrt{\frac{\rho_f}{\rho_a}} \cdot \frac{\sqrt{C_a}}{\tan\left(\frac{\theta}{2}\right)} \sqrt{\left(\frac{2}{B(T_a, P_a, T_f)} + 1\right)^2 - 1}$$

where

$$B = \frac{Z_a(T_s, P_a - P_s) \cdot P_s \cdot M_f}{Z_f(T_s, P_s) \cdot (P_a - P_s) \cdot M_a} = \frac{h_a(T_a, P_a) - h_a(T_s, P_a - P_s)}{h_f(T_s) - h_f(T_f, P_a)}$$

B equation is used to calculate T_s , then B value can be obtained by enthalpies at different temperature and pressure.

The terms a, b and C_a are constant with the value of 0.66, 0.41 and 0.9, respectively; ρ_f and ρ_a are diesel fuel and ambient gas density, respectively; θ is the spray spreading angle; d is the diesel injector nozzle diameter; Z_a and Z_f are the vaporized fuel and ambient gas compressibilities, respectively; P, T, M and h are pressure, temperature, molecular weight and enthalpy, respectively, and the subscripts a, s and f refer to ambient gas, saturated condition and diesel fuel.

Figure 3 shows a representative liquid length calculation. The blue line is the calculated liquid length. The red line is the distance from nozzle tip to cylinder wall, which can be calculated based on the cylinder bore and spray angle. Since split injection strategy was used for all test cases and also the injection durations were quite small, the fuel penetration equation can just provide a rough estimation. In this specific case, if injection SOI timing is before -36.5° ATDC, the spray has the risk to hit the wall; otherwise there will be no spray-wall effect. SOI timing at -36.5° ATDC is defined as critical SOI. As can be seen from Figure 2, there is a risk for wall wetting.

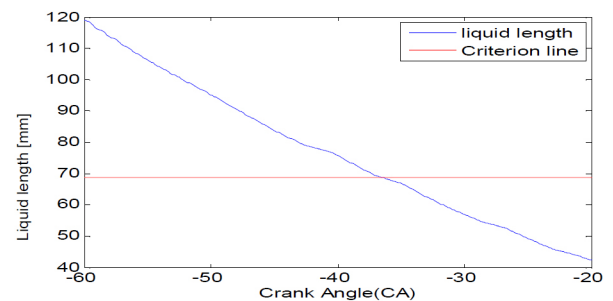


Figure 3. A representative liquid length calculation vs crank angle

The premixed percentage is calculated by comparing the heat release during the premixed combustion phase to that for the total combustion phase. Because the ascending flank of the rate of heat release curve for premixed combustion has a different gradient to the descending flank (and because the gradients of the ascending flank are not constant in some experiments), the ascending flank is modeled

using a polynomial fit in Matlab while the descending flank is described using a Gaussian profile [14]. The Gaussian profile is defined as:

$$G(x) = h \cdot e^{-\frac{(x-x_0)^2}{2\alpha^2}}$$

where x_0 is the central position of the peak, and h and α are the height and width of the Gaussian profile, respectively. Figure 4 shows a typical rate of heat release curve together with the corresponding polynomial & Gaussian fits.

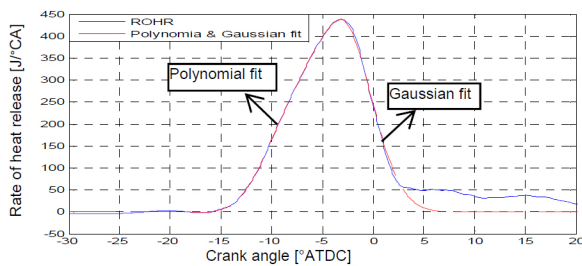


Figure 4. A typical rate of heat release curve with the corresponding polynomial & Gaussian fits

EXPERIMENTAL RESULTS

Impact of varying the PI1 SOI timing

The effects of varying the PI1 SOI timing were investigated at 9 bar BMEP and 1200 rpm, with a CR of 17. The details of the experimental conditions are provided in Table 6. The relative energy percentage of port-injected CNG and direct-injected diesel were held constant at 67% and 33%, respectively.

Table 6. Testing conditions for the PI1 SOI timing influence

BMEP (bar)	9			
Engine speed (rpm)	1200			
Intake pressure (bar)	1.8			
EGR (%)	42			
Compression ratio	17			
Diesel injection pressure (MPa)	100			
Indicated Diesel duration 1 (°CA)	2.8			
Indicated Diesel duration 2 (°CA)	3.6			
Intake charge temperature (°C)	41.5	41.3	40.7	41.6
EGR temperature (°C)	83.6	83.1	82.5	83.9
Indicated Diesel PI1 SOI (°ATDC)	-59.4	-62.3	-64.5	-66.6
Indicated Diesel PI2 SOI (°ATDC)	-45			

Figure 5 shows how the CA05, CA50, effective expansion ratio (EER), maximum pressure rising rate (PRR) and indicated thermal efficiency varies with the sweep of PI1 SOI timing. As the SOI was retarded, both the start of combustion (CA05) and CA50 advanced. These results resembled those reported by Leermakers [15] and probably occurred because of locally richer equivalence ratios and more reactive mixtures. The locally richer equivalence ratios at -59.4°ATDC could form for two reasons: the reduced time available for the mixing of the diesel spray with the surrounding air and CNG prior to autoignition and the reduced time available for mixing between the first and second injections because the timing of the second injection was not varied [16]. The maximum PRR increased

with the SOI sweep, which might reflect the formation of locally rich regions which result in fast combustion. Figure 6 shows the burning rate of SOI case at -59.4° ATDC was larger than that of other cases until -3°ATDC. Conversely, the indicated thermal efficiency declined as the SOI was retarded. Its decline on going from an SOI at -66.6°ATDC to -62.3°ATDC was mainly because the CA50 phasing became further advanced relative to TDC. However, the decline in thermal efficiency on going from -62.3°ATDC to -59.4°ATDC was due to the associated decline in the EER. The pure diesel case was also tested at the same BMEP, intake pressure and EGR mentioned before. The indicated thermal efficiency for pure diesel case was 46%, which is lower than all the testing cases in Figure 5.

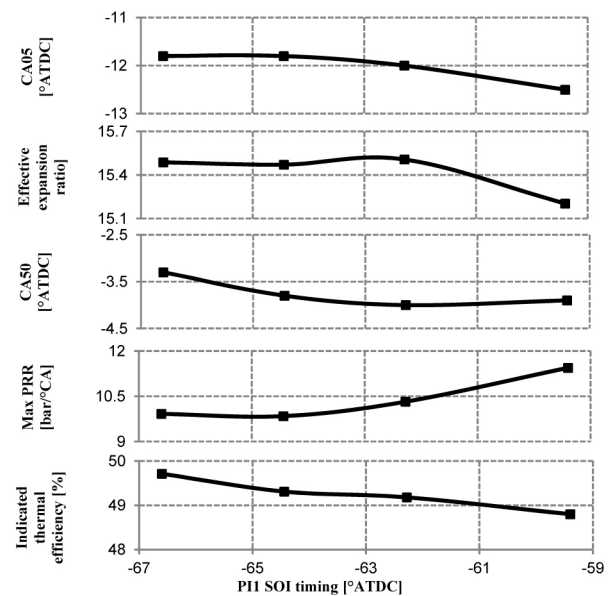


Figure 5. The influence of the PI1 SOI timing on performance

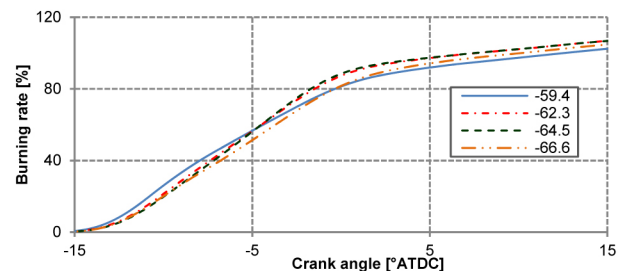


Figure 6. The influence of the PI1 SOI timing on the burning rate

The rate of heat release and pressure traces are shown in Figure 7a and b. Considering rate of heat release curves, all of them included both low temperature heat release (LTHR) and high temperature heat release (HTHR). LTHR shown in the right corner of Figure 7a indicated except SOI=-66.6° ATDC, the other three cases were roughly the same. HTHR curves indicated the combustion for all cases except SOI=-59.4° ATDC included two peaks: first one was premixed air, diesel fuel and CNG combustion; second peak was premixed natural gas combustion. Considering case SOI=-59.4° ATDC, it only got one peak which was due to the combustion of premixed air, diesel fuel and CNG and followed by a smaller proportion of pre-mixed CNG/air combustion than the other cases. All four cases in Figure 7a involved a late mixing control combustion