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Experimental Investigation and Analysis of Natural Gas RCCI on a Modified GDI Engine using NVO

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Abstract

Reactivity Controlled Compression Ignition with port-injected Natural Gas and direct-injected N-heptane is investigated on a modified production gasoline direct injection with symmetric Negative Valve Overlap (NVO). Two injection strategies are explored for the high-reactivity n-heptane direct injection. A single injection strategy is compared with a split injection strategy where half the high-reactivity fuel is injected during NVO. The heat release, combustion timing, and Burn Duration (BD) were calculated from the measured cylinder pressure trace and used to examine differences in the injection strategies. This work shows that the split injection strategy attains higher combustion and thermal efficiency, lower maximum pressure rise rate and a longer BD due to better in-cylinder charge homogeneity.

1 Introduction

Dual-fuel Reactivity Controlled Compression Ignition (RCCI) is an auto-ignition combustion strategy that utilizes varying the ratio between two fuels of different reactivity to control the combustion timing. RCCI has been known to yield longer combustion durations and lower maximum Pressure Rise Rates (PRR) when compared to Homogeneous Charge Compression Ignition (HCCI) and Premixed Charge Compression Ignition (PCCI) strategies [1]. This strategy provides a means for reducing engine out NOx and soot emissions. Typically the low-reactivity fuel is port-injected while the high-reactivity fuel is injected directly into the cylinder through the use of direct injectors. Injection of the high-reactivity fuel is used during the compression stroke as an ignition source giving a large amount of controlability over the main ignition timing [1]. However direct injections closer to TDC increase the NOx emission. Multiple direct injections can be used during the compression stroke to create a fuel reactivity stratification. This stratification causes lower PRR, higher gross indicated efficiencies and near-zero NOx production [2].

Negative Valve Overlap (NVO) can be utilized with HCCI/RCCI combustion to trap residual gasses in the cylinder, called internal exhaust gas recirculation, which raises the cylinder temperature at the Intake Valve Closing (IVC). This assists in achieving fuel auto-ignition with low compression ratio engines [3]. Using a direct liquid fuel injection during the NVO phase can cause a decrease in the temperature of the intake stroke due to the latent heat of evaporating the injected fuel. This decrease of temperature in the intake stroke increases the ignition delay and helps inhibit NOx production [4]. Early fuel injection during NVO recompression can experience endothermic fuel reforming while fuel injection after the TDC of recompression typically only absorb heat to evaporate the injected liquid [5].

In this work, RCCI with NVO using port injection Natural Gas (NG) and direct injection n-heptane is investigated on a production Gasoline Direct Injection (GDI) engine modified for this research. The use of fuels with larger differences in reactivity lead to higher thermal efficiencies due to the stratification [1]. Two direct injection strategies were considered for the n-heptane; first, a single injection that provides the ignition source; and second, an early injection at NVO and a second later injection during compression stroke that acts as the main ignition source.

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2 Experimental Setup

A 2.0L General Motors Ecotec II LNF GDI engine from a 2009 Chevrolet Cobalt, shown in Figure 1, is used in this work. Port fuel injectors were added to the engine to inject compressed natural gas as a high-octane low-reactivity fuel while the stock direct injectors were used to inject n-heptane, a low-octane high-reactivity fuel with a constant injection pressure of 100 bar. The pistons were replaced to raise the compression ratio of the engine to 11.1 from 9.2 of the stock engine. The engine's phasers were adjusted to utilize the Variable Valve Timing (VVT) technology of the engine to enable NVO. The engine's intercooler was replaced with an intake air heater to heat up the intake charge and facilitate auto-ignition. The LNF engine specifications can be found in Table 1.

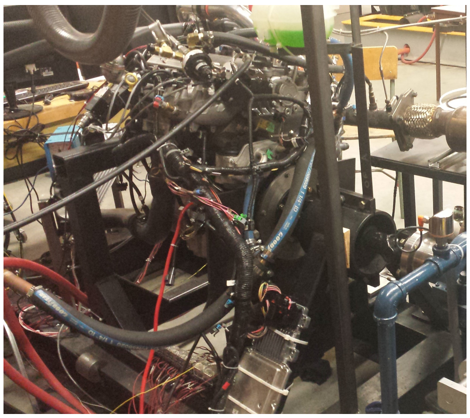

Fig. 1: The 2.0L General Motors Ecotec II LNF GDI Engine, modified with additional port injection, higher compression ratio and research engine control unit

Table 1: LNF engine specifications

Bore × Stroke	86 × 86 mm
Displacement	2.0 L
Modified Compression Ratio	11.1:1
Valves Per Cylinder	4
Max Valve Lift	10.3 mm
NVO Phasing Range (at 1 mm lift)	41.2 to 141.2 CAD

Table 2: RCCI engine operating conditions for single and split injection

	Single Injection	Split Injection
Negative Valve Overlap [CAD]	71.2 ± 2	71.2 ± 2
Engine Speed, n [RPM]	1200 ± 1	1200 ± 1
Intake Air Temperature, IAT [°C]	85.3	85.3
Engine Coolant Temperature, ECT [°C]	90	90
Air-Fuel Equivalence Ratio, λ	1.23	1.19
Injected Fuel Energy D-Inj1, IFE_{NH1} [kJ]	-	0.176
Injection Start Angle D-Inj1, θ_{NH1} [CAD bTDC]	-	360
Injected Fuel Energy D-Inj2, IFE_{NH2} [kJ]	0.377	0.163
Injection Start Angle D-Inj2, θ_{NH2} [CAD bTDC]	39	39
Injected Fuel Energy P-Inj1, IFE_{NG} [kJ]	0.314	0.314
Injection Start Angle P-Inj1, θ_{NG} [CAD bTDC]	310	310
Total Injected Fuel Energy, IFE_{tot} [kJ]	0.691	0.652

RCCI tests were only conducted on one test cylinder of the engine while the other three cylinders operate using port-injected natural gas with a spark. Cylinder pressure at 0.1 Crank Angle Degree (CAD) for 200 cycles was collected for each operating point. On the one test cylinder, a safety spark at 20 CAD after Top Dead Center (aTDC) was used for all RCCI tests to ensure that no misfires would occur. As there is not an individual exhaust port for the RCCI cylinder used for testing, emission data were not collected. The RCCI operating points were conducted at a constant engine speed of 1200 RPM with lean-burn operating conditions and 71.2 CAD of NVO. Since increasing the NVO duration delays the intake valve opening timing which increases the back pressure in the intake air manifold less air is aspirated. Higher NVO phasing is not possible with this experimental setup since intake air boosting via supercharging to the engine would be required at this speed to maintain acceptable air flow rates. The experiment operating conditions are shown in Table 2. The start of injection crank angle timings during the engine cycle are shown in Figure 2.

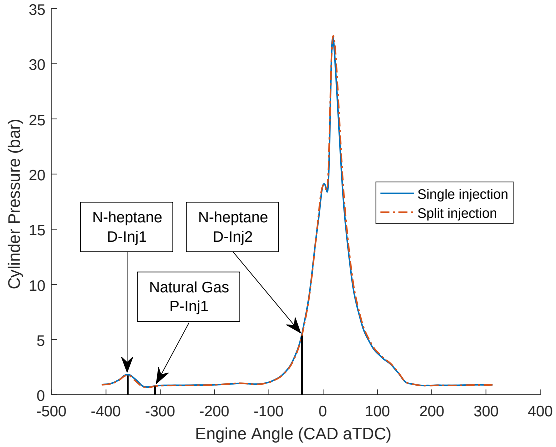

Fig. 2: Pressure trace comparison between single and split n-heptane injection strategies

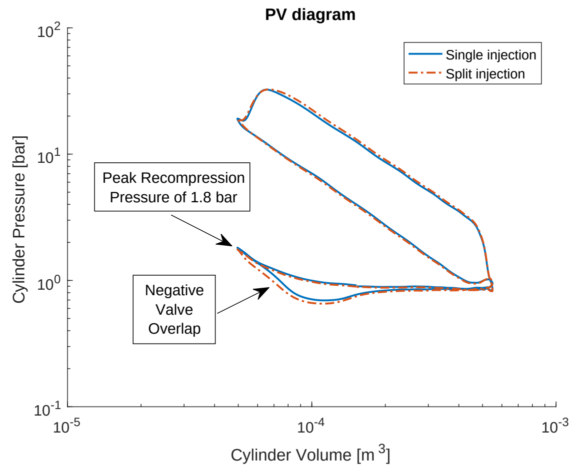

Fig. 3: PV diagram comparison between single and split n-heptane injection strategies

3 Engine Performance Analysis

The analysis is performed based on the measured in-cylinder pressure as follows. Indicated Mean Effective Pressure (IMEP) is calculated [3] as

$$IMEP = \frac{W_i}{V_d} \tag{1}$$

where P is the measured in-cylinder pressure, and V_d is the displacement volume. The indicated work, W_i is calculated [3] as

$$W_i = \int PdV \tag{2}$$

where V is the cylinder volume. The Heat Release Rate, HRR is calculated [3] as

$$HRR = \frac{dQ_{HR}}{d\theta} = \frac{1}{k-1}V\frac{dP}{d\theta} + \frac{k}{k-1}P\frac{dV}{d\theta} \tag{3}$$

where k, V, θ, and Q_{HR} are the specific heat ratio, in-cylinder volume, engine crank angle, and the cumulative heat release rate respectively. The thermal efficiency, η_{th} is calculated [3, 6] as

$$\eta_{th} = \frac{W_i}{IFE_{tot}} \tag{4}$$

where IFE_{tot} is the total Injected Fuel Energy calculated using the fuel mass injected per cycle times the Lower Heating Value (LHV) of the two fuel ($LHV_{NG} = 47.14$ MJ/kg and $LHV_{Nhept} = 44.57$ MJ/kg, where the subscripts *NG* and *Nhept* denote natural gas and n-heptane respectively.

4 Results and Discussion

Single injection and split injection strategies for direct injection n-heptane are detailed in this section. The pressure trace of both RCCI operating points are quite similar as shown in Figure 2. The pressure decrease during the NVO recompression, caused by the injected fuel evaporation, can be seen on the log-log PV diagram in Figure 3.

The performance results of the two injection strategies are listed in Table 3. The pressure drop of the pumping loop causes the pumping loss to increase by 23% due to the cooling of the trapped residual gases. However even with the 71.2 CAD of NVO, the pumping losses are negligible compared to the generated power since the power stroke is two orders of magnitude larger than the NVO losses. The cumulative heat release is 10% higher for the split injection case despite have 6% less injected fuel energy and is shown in Figure 4. Thus the split injection case has a higher combustion efficiency.

Table 3: Cylinder pressure and heat release comparison of single and split injection strategies

	Single Injection	Split Injection
Indicated Mean Effective Pressure, $IMEP$ [bar]	5.2 ± 0.2	5.6 ± 0.1
Pumping Loss [bar]	0.057 ± 0.015	0.07 ± 0.017
Max Pressure Rise Rate, PRR_{max} [bar/CAD]	2.6 ± 0.4	2.2 ± 0.3
Exhaust Gas Temperature, T_{exh} [°C]	310 ± 5	355 ± 5
Total Heat Released, Q [kJ]	0.583 ± 0.024	0.644 ± 0.012
Indicated Thermal Efficiency, η_{th}	37.8 ± 1.2	42.9 ± 0.4

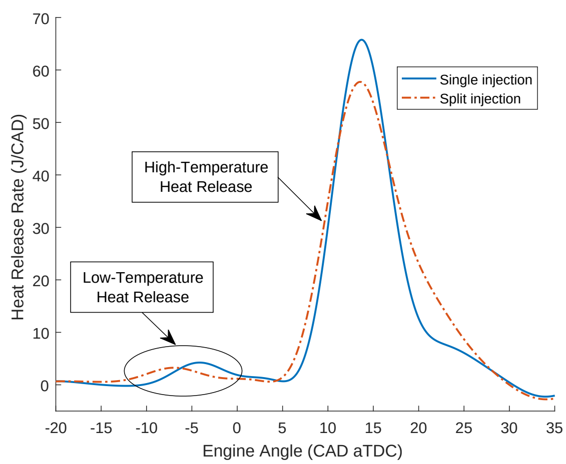

Fig. 4: Heat release rate comparison between single and split n-heptane injection strategies

The split injection operating point has an earlier low-temperature heat release and therefore shorter ignition delay, this could be attributed to possible fuel reforming since roughly half of the n-heptane was already mixed during NVO, thus advancing the start of combustion. The high-temperature heat release also begins earlier, does not reach as high as single injection high-temperature heat release, but declines more gradually. This corroborates with the measured exhaust gas temperature which is 45°C higher for the split injection case since more energy is released later in the cycle. It is important to note that the exhaust temperature is for all four cylinders in the exhaust manifold and not for the one individual test cylinder. The smoother lower maximum pressure rise rates, 2.2 bar/CAD of the split injection compared to 2.6 bar/CAD for the single injection case is desirable for reducing engine noise and wear and could allow for higher loads given a constraint on PRR [7].

From the HRR, the crank angle of fifty percent fuel mass fraction burned (CA50) and burn duration (BD) are calculated. Table 4 shows the average values of 200 cycles with one standard deviation while the individual values for 200 cycles are shown in Figure 5. The BD is calculated as the CAD from 10% to 90% combustion completion (CA10 to CA90). The burn duration for the split injection case is slightly longer for the same combustion timing and it is attributed to the multiple direct injections stratifying the charge in the cylinder.

In order for RCCI to be used in production engines, cyclic variability must be thoroughly understood and controlled [8]. The combustion parameters in Tables 3 and 4 indicate that the cyclic variability of the two operating points are similar.

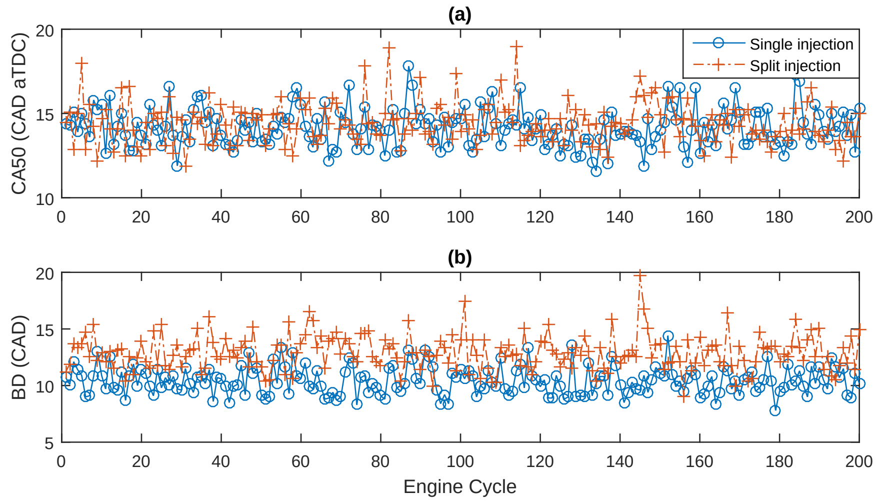

Fig. 5: Cycle by cycle comparison of the main ignition for single and split injections: (a) Combustion Timing (CA50), and (b) Burn Duration (BD)

Table 4: Heat release combustion parameters

Combustion Parameter [CAD aTDC]	Single Injection	Split Injection
Low-Temperature Heat Release, CA3	-3.1 ± 0.6	-6.1 ± 0.5
Combustion Timing, CA50	14.2 ± 1.2	14.4 ± 1.2
Burn Duration, CA10 to CA90	10.4 ± 1.2	12.9 ± 1.5

The heat release rate during NVO is shown and compared for both n-heptane injection strategies in Figure 6. The heat release during NVO differs after the fuel injection in split injection case compared to the single injection strategy where no fuel is injected during NVO. The dip in the HRR is due to the energy transferred of the injected n-heptane as it absorbs energy to vaporize the liquid fuel.

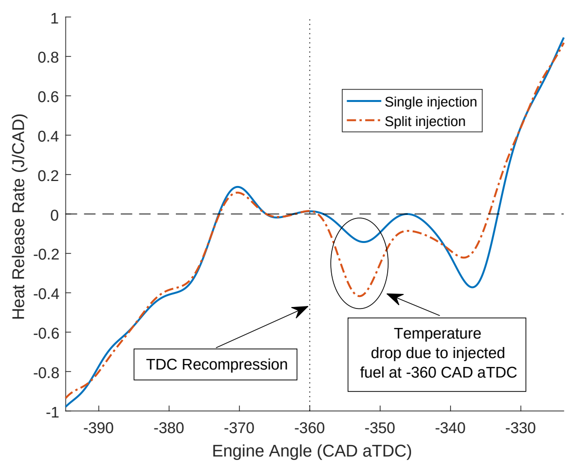

Fig. 6: HRR during NVO recompression

Based on the latent heat of evaporation of n-heptane, the energy required to vaporize the amount of fuel injected in NVO is 1.42J. The energy transferred after TDC of the NVO (from -360 to -330 CAD aTDC) was calculated to be -2.4J for the single injection case and -3.5J for the split injection case. Thus the difference between the energy transfer is 1.1J which is close to the calculated 1.42J. It can be concluded for split injection, a portion of the fuel does indeed vaporize which assists in fuel mixing causing a more homogeneous mixture. This should contribute to lower particulate matter formation, since localized rich regions in the main ignition is the main contributor to particulate formation for this type of engine combustion [9]. In addition, since the first split injection was not injected before TDC during the NVO compression, no fuel reforming is expected. This is confirmed by the fact that endothermic fuel reforming would take more energy than what is required to evaporate the fuel [5]. Higher NVO phasing than what was tested may be required to achieve the conditions needed for NVO fuel reforming.

5 Conclusions

Split direct injection is compared to the single direct injection strategy on an experimental RCCI engine fueled with natural gas and n-heptane. Symmetric NVO is used as the valve timing strategy and half of the n-heptane fuel is injected during NVO recompression for split direct injection strategy. The effects of direct injection strategies on RCCI combustion characteristics are detailed while all other engine parameters are kept constant. For the two direct injection strategies explored, the split injection has higher thermal and combustion efficiencies, lower pressure rise rates and longer burn duration. The n-heptane fuel injected during NVO evaporates; however, no endothermic fuel reforming occurs. Further studies regarding NVO fuel reforming effects on RCCI combustion are needed with longer NVO duration and with early NVO injection.

Acknowledgments

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