HCCI combustion control using advanced gasoline direct injection techniques

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Abstract. Homogeneous Charge Compression Ignition (HCCI) is a promising low temperature combustion technology for reciprocating engines that offers high fuel efficiency and extremely low exhaust emissions. However, combustion control should be improved and operating range should be widened for the technology to achieve production level. In this study an overview of different direct gasoline injection control approaches, applied to improve stability at low engine loads and to reduce pressure rise rates at high load regime, is presented. The tests are performed on a single-cylinder research engine operated in a negative valve overlap (NVO) mode for residual gasses trapping. The investigated direct injection schemes included: (i) fuel injection during the NVO period to improve mixture reactivity and take an advantage of exhaust-fuel reactions thermal effects, (ii) fuel injection during intake stroke to create homogeneous charge and (iii) late fuel injection during compression stroke to create stratified charge. The results showed that application of early NVO injection enables active control of combustion timing at nearly idle conditions. The late fuel injection, during the compression stroke, enabled mitigation of excessive pressure rise rates at high engine load regime.

1 Introduction

The recent progress in engine development towards full system flexibility, and achievements in model based engine control, greatly relaxed the constrains on system complexity. With fully flexible valve trains and variable compression ratio systems becoming standard features for next generation automotive engines [1, 2], the feasibility for advanced low temperature combustion systems like homogeneous charge compression ignition (HCCI) is being re-considered.

In HCCI, the combustion process is kinetically controlled, resulting in very fast heat release rates (HRR). This allows for optimizing the engine efficiency close to the theoretical limits of the ideal Otto cycle [3]. The premixed mixture, with uniform temperature distribution across the cylinder, supports achieving ultra-low nitrogen oxides (NO_X) and particulate matters emissions [4, 5]. On the other hand, assuring low-load combustion stability and limiting excessive pressure rise rates (PRRs) at high-loads are main challenges for the HCCI concept [6, 7].

Exhaust gas trapping via negative valve overlap (NVO) enables variety of control mechanisms that can be used to tackle the above challenges of HCCI [8, 9]. This technique introduces internal exhaust gas recreation (EGR) enabling sufficiently high intake valve closing (IVC) temperatures for gasoline auto-ignition at compression ratios low enough to avoid knock. Additionally, the EGR slows down the reaction rates and increases fuel dilution, thus contributing to further NO_X emissions reduction [10].

Note that NVO operation differs from the typical 4-stroke engine cycle (with positive valve overlap), by introducing exhaust gas recompression between exhaust and intake processes. An example of a full in-cylinder pressure trace recorded for an HCCI engine with NVO mode is shown in Fig. 1

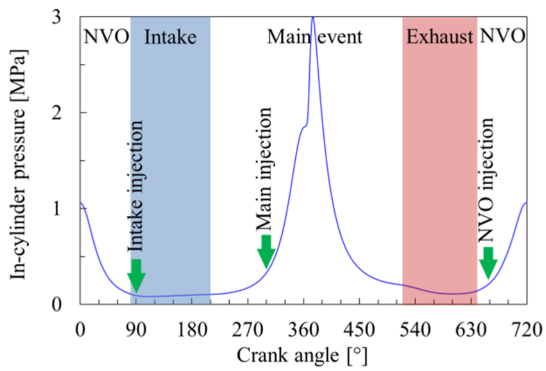

Fig. 1. Example of in-cylinder pressure and injection timings.

Further extending of the HCCI engine operating range towards low loads can be attained with direct injection of a certain portion of fuel early during the NVO period, as shown in Fig. 1. This injection strategy results in the so-called fuel reforming – a chemical process which produces highly reactive species like: ethane, ethylene, formaldehyde, methanol and acetylene [11, 12]. The mix of highly reactive reformed gasses with the air and the fuel injected during the induction process provides proper mixture ignitability in the main compression event.

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Additionally, combustion stability is improved and THC emission reduces. Furthermore, lean operation limit can be extended, positively influencing low-load engine performance and NO_X emissions [13].

Aside of the above chemical effects of fuel reforming, thermal effects of NVO injection can be used for combustion phasing control across all the engine operating conditions [14, 15]. Note however that the effects highly depend on NVO injection timing and mixture strength.

From the perspective of the expansion of engine operation area towards higher loads extensive PRRs can be avoided by introducing mixture stratification via late fuel injection during the main event. The mechanism relays on distributing the ignition of in-cylinder mixture in space and time thus elongating the combustion and making it smoother [15-17].

Bearing in mind the recent trends in engine development and the resulting, increased feasibility for advanced low temperature combustion concepts, the present work aims to provide a concise demonstration of the effects of different direct injection strategies on residually affected HCCI combustion characteristics. Namely, following techniques are showcased: (i) fuel injection during the NVO period to improve mixture reactivity and take an advantage of exhaust-fuel reactions thermal effects, (ii) fuel injection during intake stroke to create homogeneous charge and (iii) late fuel injection during compression stroke to create stratified charge. The engine experiments, supporting this demonstration, cover the whole range of engine loads attainable in the HCCI mode.

2 Experimental approach

2.1 Research engine and test stand

The discussed results are produced on a single-cylinder research engine installed on a test bed with a direct current dynamometer. The engine is equipped with a fully variable valvetrain, which enables regulation of valves lifts and timings. A detailed description of the engine and the valvetrain is presented in refs. [14, 15]. At this point only a concise decryption of the functionalities relevant to the present tests are provided.

The fully variable valvetrain allows to obtain internal EGR with the use of the NVO technique. Fuel is introduced into the cylinder with the use of a side-mounted, single-stream, swirl-type injector. An electric motor driven vane compressor and air temperature management system are installed in the intake runner to provide required boost levels. The main engine geometric parameters, along with the utilized valvetrain settings are specified in Table 1. To simplify the imaging of crank angle based figures, all timings are given in terms of degrees after top dead centre (TDC) during NVO in accordance with Fig. 1.

The engine test bench is equipped with a standard, in-cylinder pressure based combustion analysis system, by AVL. The acquisition for the in-cylinder pressure and the injector excitation current is triggered by an optical crankshaft encoder, with an angular resolution of 0,1° of crank angle. In order to provide proper input for combustion analysis, the intake mass flow rates, temperature and pressures are recorded along with the fuel consumption (measured gravimetrically). Additionally, to estimate the overall main event excess ratio (λ), a wide band lambda probe, installed in the exhaust runner, is utilized.

Table 1. Test engine data.

Parameter	Value
Swept volume, cm^3	498,5
Bore, mm	84
Stroke, mm	90
Compression ratio, -	11,7
Fuel	Gasoline 95 RON
Fuel injector	Direct swirl type
Fuel pressure, MPa	10
Boost	Vane compressor
Number of valves	2
Intake valve opening (IVO), °	80
Intake valve closing (IVC), °	210
Intake valve lift, mm	3,6
Exhaust valve opening (EVO), °	520
Exhaust valve closing (EVC), °	640
Exhaust valve lift, mm	2,9

2.2 Experimental conditions and procedure

The experiments are conducted at the engine speed of 1500 rev/min with fully open throttle. The intake air temperature is set to 40 ± 1 °C. The temperature of cooling liquid at the engine outlet is maintained constant at 90 ± 1 °C. The fuel is introduced into the cylinder at strategically selected start of injection (SOI) timings, which functions were verified in our previous works [14-16]. To enable fuel reforming, the NVO injection is applied at SOI = 660°. Intake injection, applied at 90° aims at creating a homogeneous mixture without any thermal nor chemical effects which are associated to NVO injections. The role of the main injection, applied at 300°, is to create a stratified mixture in order to slow-down the combustion rates at elevated loads.

The experimental matrix includes three operating points covering the whole attainable HCCI load range. The operating parameters, along with the applied injection strategies are shown in Table 2. The set points of λ and boost pressure are adjusted to optimal conditions for a given load case. Those settings were determined in previous research endeavours as the best trade-offs between efficiency and NOx emissions [14, 15]. Note that the valve settings used to achieve NVO in the present study are not per-case optimized. This is maintained the same in all operating points, in order to provide a common benchmark for comparing the effects of NVO injection.

The internal EGR ratio definition used in the present work (Table 2) relays on compering the calculated mass of trapped residuals at exhaust valve closing (EVC) with the same quantity at the IVC event. The mass indexes are computed using the AVL Boost software, basing on the measured in-cylinder and manifold conditions. The same software is further used for the HRR analysis. The net

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HRR traces presented in the results section are calculated according to the first law of thermodynamics, with consideration of the effects of temperature and composition on a the instantaneous specific heats ratio.

The Equivalence ratio (Φ) is calculated using the following formula:

$$\Phi = \frac{m_F(1+L_t)}{m_{Air}+m_{Exh}}, \tag{1}$$

where m_F , m_{Air} and m_{Exh} are masses of fuel, fresh air and residuals respectively, and L_t is the theoretical air-fuel ratio for gasoline. One should note that the above definition represents the total fuel dilution by air and exhaust gasses. The values of Φ, presented in Table 2 show how extensive is the dilution in each operating point despite nearly stoichiometric excess air. It should be further noted , that some values shown in Table 2 are averaged for different injection strategies for the sake of clarity of the presentation. In fact, these values were changing in some extent due to effects of fuel injection on mass of aspirated air, temperature, etc.

Table 2. Detailed engine settings for analysed cases.

Parameter		Case 1	Case 2	Case 3
IMEP, MPa		0,19	0,31	0,61
λ, -		1,25	1,0	1,07
EGR, %		56	47	29
Φ [-]		0,354	0,507	0,693
p_{int}, kPa		101	101	152
SOI, °	A	660	660	660
	B	660; 90	660; 90	660; 90; 300
	C	90	90	660; 300
m_F, mg	A	7,79	11,05	20,65
	B	2,6; 5,03	2,6; 8,06	2,6; 14,9; 2,6
	C	7,95	9,86	2,6; 19,59
η_{therm}	A	0,28	0,32	0,35
	B	0,31	0,33	0,37
	C	0,29	0,33	0,34

3 Results and discussion

At the lowest investigated engine load (case 1 in Table 2) the fuel is injected in a single dose early during NVO (injection strategy A), in a single dose during the intake stroke (strategy C) and with split fuel injection (strategy B). At the latter case, only a small portion of the fuel (so-called pilot fuel), is injected during the NVO phase. In order to better illustrate the effects of variable injection strategies on in-cylinder processes, the pressure-volume diagrams in logarithmic scale are shown in Fig. 2. Careful analysis of the p-V diagram, for the injection strategy A, shows that the system performs positive work around NVO TDC. This is the result of partial fuel oxidation during NVO. Since the mixture is lean at this operating point, the trapped exhaust gases contain some amounts of oxygen, which allows for exothermic reactions to occur. While the in-cylinder volume increases, the work becomes negative, with values comparable to those obtained for the single intake injection strategy. For the split injection case, a positive loop near TDC can observed, similarly to strategy A. Note however, that the negative work at higher cylinder volumes is much smaller than in the remaining cases. This observation allows to state that the reduction of the NVO injected fuel mitigates the re-compression loop losses in comparison to strategy A. It should be also noted from Fig. 2 that the intake injection strategy results in much higher expansion pressure. It is an effect of improved cylinder aspiration and the resulting higher energy introduced with the fuel at constant λ.

Furthermore, the main compression-expansion loops in Fig. 2 reveal the effect of injection strategies on combustion timing. To provide more insight into combustion evolution, Fig. 3 shows HRR curves for the analysed case 1. It should be noted that NVO fuel injection results in very early combustion. Nearly all fuel is oxidized before TDC, which increases compression work and deteriorates overall cycle efficiency. Shifting the injection from NVO period to the intake stroke enables delaying the start of combustion by approximately 10 °CA. Additionally, the reaction rates are substantially reduced, which is manifested by lower peak HRR value and longer combustion duration. Note that the split fuel injection (strategy B), provides moderate peak HRR values and optimal combustion timing. The properly aligned combustion timing and discussed above reduced NVO losses provide improvement of overall thermal efficiency of the cycle by 10%, comparing to all fuel injection during NVO (see Table 2).

At mid-load (case 2 in Table 2) the engine was operated at stoichiometric mixture. At given valve timings, this was the maximum attainable load under naturally aspirated conditions. The applied injection strategies were the same as in the previously discussed case 1.

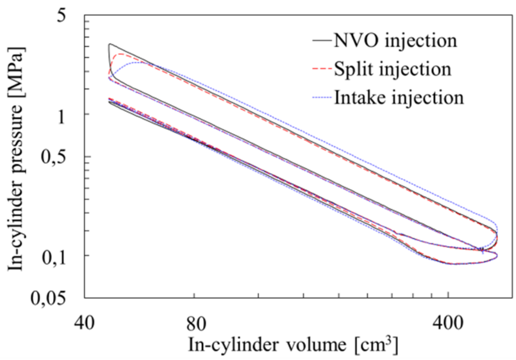

Fig. 2. Pressure-volume diagrams for case 1.

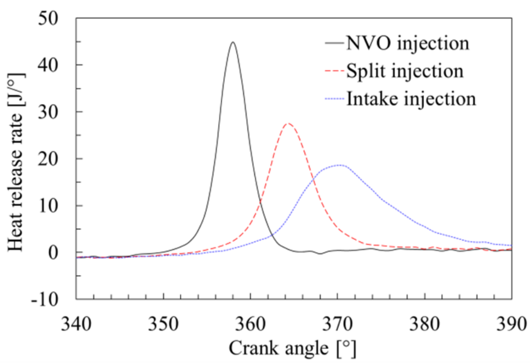

Fig. 3. Heat release rates for case 1.

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Pressure-volume diagrams in Fig. 4 show that the effects of fuel injection on the performed work are substantially different than for lean mixture realized in case 1. The strategy A, realizing a single NVO injection, results in high recompression losses. It is evident from Fig. 4, that the pressure rise during the NVO compression is diminished by fuel vaporisation. It is not compensated by partial fuel oxidation due to the lack of oxygen in the exhaust, while operating close to the stoichiometric conditions. Without additional oxidation the evaporative cooling deteriorates cycle efficiency, as shown in Table 2. Note however, that at the same time, the strategy reduces temperature and pressure at IVO which improves fresh charge aspiration. This effect in some extent compensates for efficiency drop, as engine load (amount of fuel injected) itself increases.

Following the same reasoning, with all the fuel injected during the intake stroke (strategy C), the recompression loop is much thinner, while, adding a small pilot injection (strategy B) increases the pressure delta between NVO compression and expansion.

Fig. 5 shows peculiar behaviour it terms of the auto-ignition timing. Despite in Fig. 4, the thermal effect associated to different fuel injection strategies are clearly visible, the ignition timing is nearly the same no matter the SOI. It is plausible that the NVO fuel reforming produces some auto-ignition promoting species, however, changes in mixture reactivity are balanced-out by the thermal effects of fuel evaporation. Nevertheless, the combustion timing for the discussed mid-load, stoichiometric case is perfectly aligned from the cycle efficiency point of view. The reaction rates are substantially affected by the NVO injection, similarly to the lean mixture operation in case 1. This finally suggests that the mixture reactivity improves as a result of injection strategy A, even at stoichiometric conditions. For more details on quantification of reactive species production as a result of NVO fuel injection the reader is referred to the following works [11, 18].

In order to further increase the engine load, precluded in case 2 by insufficient fresh air aspiration, it was necessary to elevate the intake pressure beyond the ambient level. For the case 3, this is set to the absolute value of 152 kPa. As it was earlier demonstrated in [15], HCCI combustion under loads excessing 0,5 MPa in terms of IMEP produces unacceptable PRRs. Thus, in this case investigated injection strategies are altered in comparison to the lower load cases, where exceeding the PRR limit does not pose an issue. At high load, late injection at SOI = 300° is incorporated to take the advantage of introducing in-cylinder mixture stratification during the main event. Two stratification patterns were realized via different amounts of fuel injected during the compression stroke. The injection strategies for case 3 and fuel division between injections are provided in Table 2. To avoid excessive emissions of CO and unburned hydrocarbons from locally fuel-rich zones which might arise as a result of stratification, the engine is operated under overall slightly leaner mixture compared to case 2. It should be also noted that during boosted operation, the engine was unable to realize autonomous HCCI operation without the support of the NVO pilot injection. The reason for this behaviour is the substantial reduction of IVC temperatures, caused by trapped residuals being replaced by the colder intake air. To compensate for that drop in temperature, the mixture reactivity has to be improved by incorporating NVO fuel reforming effects.

Fig. 6 shows how impactful are the effects of applied fuel injection schemes at high load. Similarly to the medium load case, strategy A (NVO injection) produces high recompression losses. These losses can be minimized by reduction of the NVO fuel amount, as evident when analysing the Fig. 6 for strategies B and C (split injection with different levels of stratification). However, for two cases with the same pilot fuel injection recompression loops are different. It is a result of different amounts of trapped residuals and different exhaust temperatures. Namely, the share of the thermal losses for large stratification was higher, which deteriorated the cycle efficiency, as shown in Table 2.

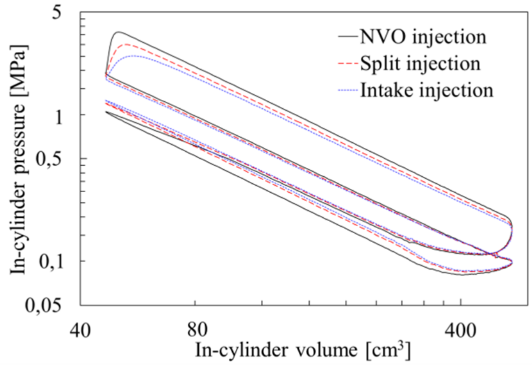

Fig. 4. Pressure-volume diagrams for case 2.

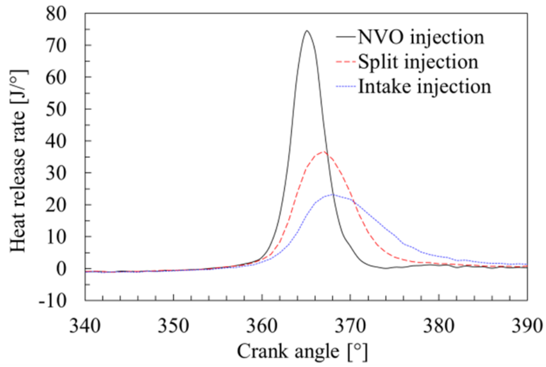

Fig. 5. Heat release rates for case 2.

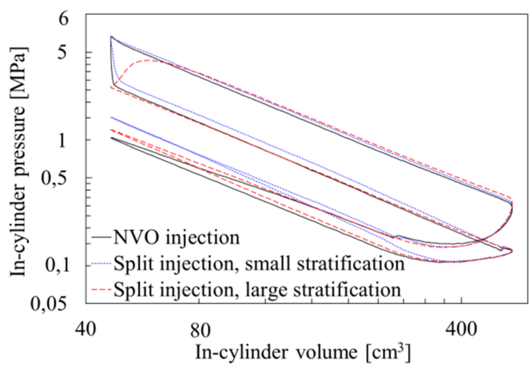

Fig. 6. Pressure-volume diagrams for case 3.

Fig. 7 shows that for the primary, NVO injection strategy A the maximum HRR exceeds 200 J/°. Additionally, this point appears at TDC, which causes very high peak pressure and excessive PRR at a level of 13 GPa/s, where commonly acceptable limit is approximately 5 GPa/s [19]. Interestingly, the small stratification achieved with strategy B indeed reduces combustion rates, but advances auto-ignition. In contrast, large stratification reduces the combustion rates and delays the auto-ignition at the same time. This peculiar effect can be explained as follows. With relatively short injection during the compression phase, the fuel has sufficient time to evaporate and largely premix before the auto-ignition starts. Still, the evaporating fuel creates thermal stratification with locally hot zones acting as ignition sources. Nevertheless, the following combustion process is slower because cold zones ignite later. This thesis was proven on the basis of modelling and experimental studies [20-22]. At extensively stratified mixture (strategy C) the fuel does not premix well before the auto-ignition. Thus, the mixture is stratified not only thermally, but also compositionally. As a result, in the fuel-rich zones, that are more prone to auto-ignition temperature is heavily impacted by the fuel vaporization. In contrast in the regions of high temperature, the mixture is too lean to initiate the combustion. This introduces ignition delay time required for the mixture to reach optimal auto-ignition properties (so-called physical ignition delay related to mixture preparation). The combustion rates are reduced on the basis of the same mechanisms as for mild stratification. In this case, however, the effect is reinforced by the combustion being retarded.

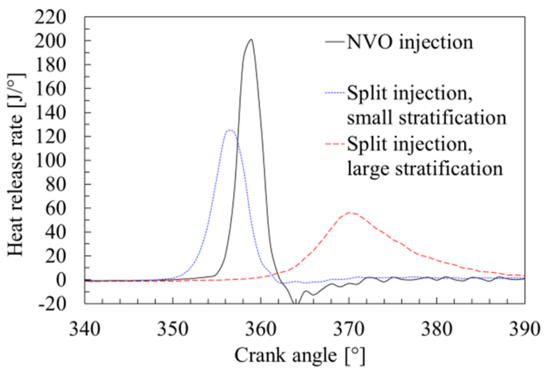

Fig. 7. Heat release rates for case 3.

Summary

The present work provides a concise summary of the authors experiences with residually affected HCCI concept development related to combustion controllability and load-range extension. The results show that application of early NVO injection enables active control of combustion timing at nearly idle conditions. The late fuel injection during the compression stroke enables mitigation of excessive pressure rise rates at high engine load regime. Furthermore, applying injection split between NVO and intake phases can be an effective control measure to attain optimal combustion phasing with fast response times not attainable with air-path related measures.

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