

Enhancing Light Load HCCI Combustion in a Direct Injection Gasoline Engine by Fuel Reforming During Recompression

Nicole Wermuth, Hanho Yun and Paul Najt
General Motors Research

Copyright © 2009 SAE International

ABSTRACT

Homogeneous charge compression ignition (HCCI) engines have the potential for high fuel efficiency and low NO_x emissions. The major disadvantage of HCCI remains the narrow operating range. One way to extend the operating range of HCCI combustion to lower load is to inject part of the total fuel mass into the hot gas during recompression. With even lower engine load, part of the fuel can also be injected late in the main compression and ignited by a spark. The propagating flame further compresses the remaining fuel-air mixture until auto-ignition occurs (spark-assisted HCCI).

In this study we investigated the effect of fuel reforming and spark assist in a gasoline engine with direct fuel injection and negative valve overlap. We performed experiments with different injection quantities and varying injection timings during recompression. Besides understanding the effect of reforming on engine performance, combustion stability and NO_x emissions we focused on methods that can be used to help monitor and control the amount of fuel reforming in a vehicle. It was found that fuel reforming reduces NO_x emissions but increases cyclic variability. Therefore the amount of fuel reforming should be closely monitored to achieve the desired trade-off between stability and emissions. The study shows that combustion phasing does not show a good correlation and should therefore not be used to monitor the amount fuel reforming. It was found, however, that PMEP shows a strong linear correlation with the fuel mass that is burned during recompression. Finally it is shown that the amount of fuel reforming can be controlled by injection timing and split ratio.

INTRODUCTION

Due to the increasing cost of gasoline and the ongoing discussion to limit CO_2 emissions fuel consumption reduction is the main focus of gasoline engine development. A variety of technologies have the potential to reduce fuel consumption significantly, e.g. non-throttled load control, down-sizing or lean stratified combustion. Homogeneous Charge Compression Ignition (HCCI) concepts offer the benefit of reduced fuel consumption while emitting only low levels of nitric oxide, therefore avoiding expensive exhaust after-treatment which is necessary for most lean combustion engines. The main advantages of HCCI combustion over traditional stoichiometric SI combustion depend on the chosen operating strategy and can include the un-throttling of the engine, the combustion of a lean air fuel mixture, a fast combustion that approaches the ideal constant volume cycle and reduced heat losses due to low in-cylinder temperatures [1].

Since the first study of a gasoline fueled four-stroke HCCI engine published in 1983 by Najt and Foster [2], HCCI technology has been receiving ever-increasing attention. The challenge for an HCCI combustion concept is to control the combustion phasing across a wide range of operating conditions. Auto-ignition is mainly governed by chemical kinetics which is influenced by the mixture composition, and the pressure-temperature history of the charge. Methods to control HCCI combustion phasing include the variation of the intake air temperature [3-5], the amount of exhaust gas retained [6, 7], and the amount of temperature and/or species stratification [8-10].

The useful HCCI operating range is limited by excessive rates of pressure rise at high load and by misfires at low

The Engineering Meetings Board has approved this paper for publication. It has successfully completed SAE's peer review process under the supervision of the session organizer. This process requires a minimum of three (3) reviews by industry experts.

All rights reserved. No part of this publication may be reproduced, stored in a retrieval system, or transmitted, in any form or by any means, electronic, mechanical, photocopying, recording, or otherwise, without the prior written permission of SAE.

ISSN 0148-7191

Positions and opinions advanced in this paper are those of the author(s) and not necessarily those of SAE. The author is solely responsible for the content of the paper.

SAE Customer Service: Tel: 877-606-7323 (inside USA and Canada)
Tel: 724-776-4970 (outside USA)
Fax: 724-776-0790
Email: CustomerService@sae.org
<http://www.sae.org>

SAE Web Address:

<http://www.sae.org>

Printed in USA

Full document details: SAE 2009-01-0923



load. Current research focuses on operating range extension and the transition between HCCI and SI operation [11, 12]. Extension of the operating range to higher loads can be achieved with turbo-charging or supercharging [13 - 15].

The extension of the low load limit for an engine using the recompression strategy is the focus of this study. This strategy employs negative valve overlap (NVO) to retain a high fraction of residual combustion products from the previous cycle to provide enough thermal energy for auto-ignition of a highly diluted mixture. With decreasing load a limit is reached when the retained exhaust gas gets too cold to initiate auto-ignition. Recent studies have shown that the use of direct fuel injection during the negative valve overlap period can be used for promoting and controlling auto-ignition in the main compression [16, 17]. With a split injection strategy where only a part of the total required fuel is injected in the hot gas during recompression and goes through partial oxidation (reforming) the operating limit can be extended to lower loads. The fuel reforming during recompression is a combination of chemical (fragmentation of the parent fuel) and thermal effects (charge cooling due to fuel evaporation and charge heating due to heat release from chemical reactions).

With even lower engine load (and therefore lower temperature in the cylinder) reforming of a portion of the fuel during recompression may not be enough to trigger auto-ignition. In this operating range (near idle operation) part of the total fuel mass can be injected late in the main compression rather than during intake stroke. The stratified part of the fuel can be ignited by a spark and the propagating flame compresses the remaining fuel-air mixture further to reach auto-ignition (spark-assisted HCCI). This study investigates fuel reforming during recompression in conjunction with stratified combustion during the main compression for an operating condition at 1000 RPM engine speed and approximately 130 kPa net mean effective pressure. This operating condition was chosen to represent a low speed vehicle cruise and as an important step towards HCCI idle operation. The amount of heat release during recompression is quantified for various injection patterns and the influence on the main heat release is studied. Furthermore a simple method that estimates the degree of fuel reforming is presented.

EXPERIMENTAL SETUP

The engine used in this experiment is a single cylinder four-stroke engine. The specifications of the engine are given in Table 1.

The prototype cylinder head is equipped with four valves and direct fuel injection. The compression ratio is 12:1. The engine is operated with fixed overhead cams and 60 crank angle degree authority hydraulic cam phasers. The

intake and exhaust cam shafts for this experiment were designed for negative valve overlap operation. The negative valve overlap (NVO) is defined as the crank angle duration between the closing of the exhaust valves and the opening of the intake valves. The peak valve lift is 4.1 mm and the opening duration 120 crank angle degrees. The configuration of piston, spark plug position and injector position was designed to enable spray-guided stratified charge combustion. The injector used in this experiment is a solenoid 8-hole injector with 60 degree spray angle. The fuel injection pressure is 11 MPa. The fuel used is a fully blended gasoline with octane number (RON+MON)/2 of 87. The engine coolant and engine oil are set to 90 deg C and the intake air is heated to 90 deg C.

Table 1: Engine specifications

Engine Type	4 valve DOHC gasoline
Bore	86.0 mm
Stroke	94.6 mm
Displacement	550 cm ³
Compression Ratio	12.0 :1
Intake / Exhaust Valve Lift	4.1 mm
Valve Opening Duration	120 CAD

The cylinder pressure is measured with a Kistler 6125A pressure transducer. Exhaust emissions are sampled from a mixing plenum in the exhaust system and analyzed by a Horiba Mexa 7000 exhaust gas analyzer. The air fuel ratio in the exhaust is measured with a wide-range air-fuel sensor (in addition to the calculation from the exhaust emissions).

The operating conditions for the present experiments are listed in Table 2. For all experiments shown in this paper the negative valve overlap duration is 196 degree crank angle. The intake manifold pressure is kept at 98 kPa. The exhaust pressure is held at 101.5 kPa. The experiments are performed with constant engine speed of 1000 RPM. No external exhaust gas recirculation is used in this set of experiments.

Table 2: Engine operating conditions

Engine Speed	1000 RPM
Engine Load	~130 kPa NMEP
Intake Temperature	90 °C
Coolant temperature	90 °C
Oil Temperature	90 °C
Intake manifold pressure	98 kPa
NVO	196 CAD

The experiments utilize a combination of fuel reforming during recompression and stratified combustion. The injection and ignition strategy for this spark-assisted HCCI operation is schematically shown in Figure 1.

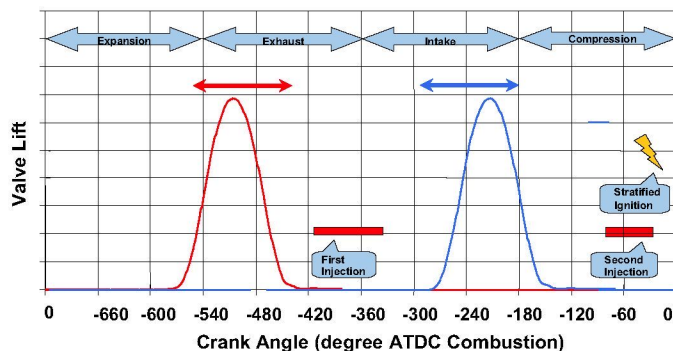


Figure 1: Injection and ignition strategy for spark-assisted HCCI operation

The experiments are performed with a variable first injection into the cylinder during the recompression period and a fixed second injection close to main compression TDC which is followed by a spark discharge. The bars in Figure 1 indicate the range where the injections typically occur. The total amount of fuel injected is 6 mg in all cases. In the first set of experiments the fuel is split evenly between the first and the second injection (injection duration 0.73 ms). In the second set of experiments the injection during the negative valve overlap is reduced to 1 mg (0.6 ms injection duration) and the late injection is increased to 5 mg (0.84 ms duration) of fuel.

In both sets of experiments the timing of the first injection is varied over a range of approximately 40 CAD. The latest injection timing around or after TDC recompression serves as reference to quantify how much heat release occurs due to left over fuel and oxygen from the previous engine cycle. It will be shown later that for very late injection the cooling due to fuel evaporation dominates and slows down or even stops the reactions of left over species. Changing amounts of heat release during the recompression result in changing in-cylinder conditions at IVO and subsequently in different fresh air masses that are inducted for the succeeding main compression and therefore in a varying air-fuel ratio. The residual gas fraction for these experiments ranges from 65 – 75 % of the total in-cylinder mass.

DATA ANALYSIS

The heat release calculation during the recompression and the main compression is performed according to the first law of thermodynamics. The heat transfer is accounted for whereas blow-by is assumed to be negligible and not taken into account for the heat release calculation. The heat transfer calculation uses the heat transfer coefficient correlation proposed by Woschni [18] with a slight modification of the velocity term. The residual gas fraction is estimated using the correlation published by Yun and Mirsky [19].

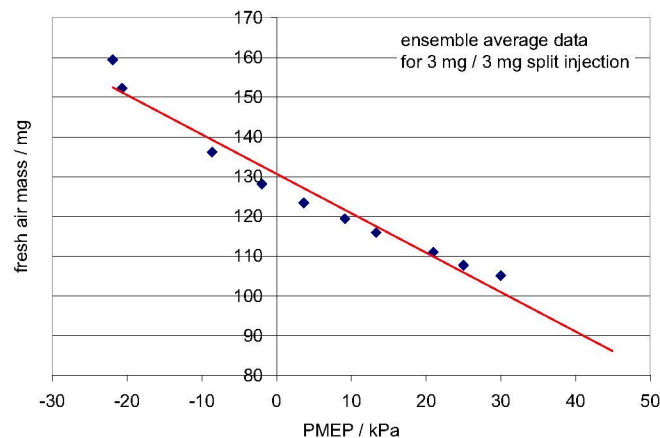


Figure 2: Fresh air mass as a function of ensemble average PMEP for 3 mg / 3mg split injection with varying first injection timing

The ensemble average data show that the inducted air mass and the air-fuel ratio vary significantly with the amount of heat release during the recompression stroke. Due to the high cyclic variability of the reforming process it is necessary to also estimate the inducted air mass for each individual cycle, rather than assuming that the air mass stays constant from cycle to cycle. A correlation is established between the inducted air mass and the pumping work on an ensemble average basis (Figure 2). This correlation is used to estimate the inducted air mass in each intake stroke based on the PMEP of the preceding gas exchange/recompression stroke.

To estimate the time of auto-ignition during the main compression the auto-ignition integral from Livengood / Wu [20] is used. The ignition delay time is calculated using the correlation from He et al [21] for iso-octane.

Due to the fact that the auto-ignition will more likely (depending on mixture composition, temperature and pressure history) occur in the hottest gas in the cylinder, the mass average temperature is not adequate for the calculation of the ignition delay time. The adiabatic core temperature and a pseudo-boundary layer temperature are therefore calculated [22] to estimate when auto-ignition first occurs in the cylinder and to check if the boundary layer also reached auto-ignition conditions. In a real engine the temperature distribution is always stratified due to wall heat losses and random processes in addition to any temperature stratification that might be left over from the previous reforming event. Lacking any quantitative data about the temperature distribution a homogenous temperature distribution at IVC is assumed.

RESULTS

OVERALL ENGINE PERFORMANCE - The influence of first injection timing (EOI1) and fuel split ratio on overall engine performance is shown in Figure 3.

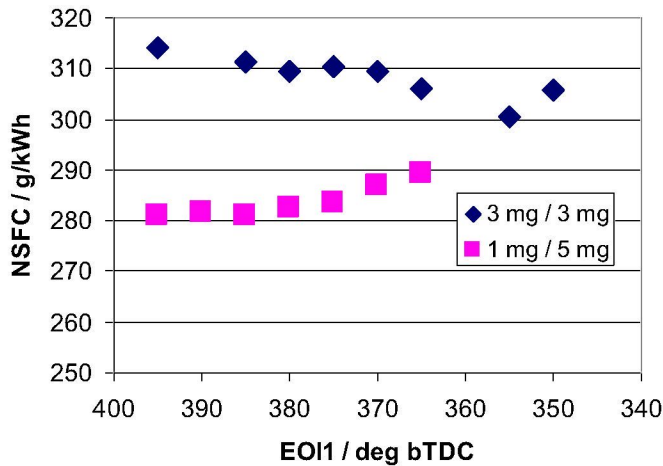


Figure 3: Ensemble average net specific fuel consumption NSFC as a function of first injection timing for 3 mg / 3mg and 1 mg / 5 mg split injection

The net specific fuel consumption NSFC¹ changes by about 3 % with the injection timing of the first injection (Figure 3). Overall the case with a 1 mg / 5 mg split of the fuel injection shows slightly lower fuel consumption which can be attributed to the combined effects of the higher air/fuel ratio (Figure 4), the lower heat loss (Figure 8), different combustion phasing (Figure 9), higher combustion efficiency (Figure 10) and the lower cyclic variability of the recompression heat release as well as the main combustion (Figure 5).

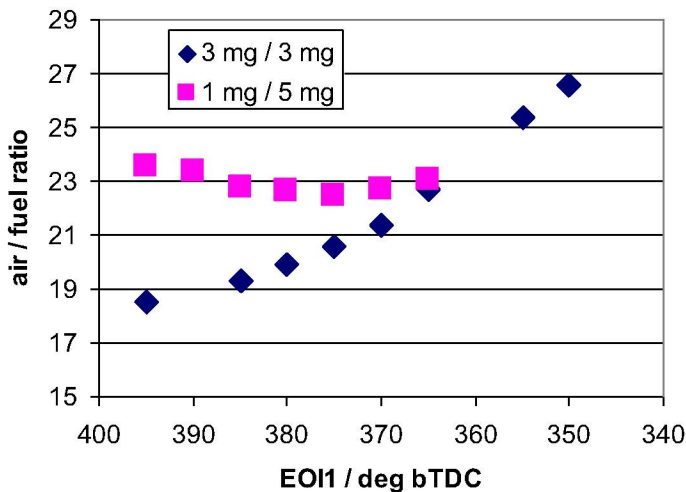


Figure 4: Average air fuel ratio as a function of first injection timing for 3 mg / 3mg and 1 mg / 5 mg split injection

¹ Net specific fuel consumption NSFC is defined as the fuel flow rate per net indicated power output. Net mean effective pressure NMEP is defined as the work delivered to the piston over the entire four-stroke cycle.

The air fuel ratio in this set of experiments is determined by the manifold air pressure, the valve timing and the amount of heat release during the recompression period. The more heat is released during the recompression period the higher is the temperature and pressure when the intake valve opens, which reduces the amount of fresh air inducted into the cylinder. Figure 4 shows that the air/fuel ratio for the 1 mg / 5 mg hardly changes at all, whereas the air/fuel ratio for the 3 mg / 3 mg case falls significantly with earlier injection timing. This indicates that the earlier injection of 3 mg of fuel results in more heat release during the recompression.

The cyclic variability of IMEP does not change with injection timing for the 1 mg / 5 mg case but it changes dramatically for the 3 mg / 3 mg case, increasing from 2 to 3 % with late injection timing to around 10 % for the earliest injection timing (Figure 5). This shows how strongly the recompression heat release influences the main combustion.

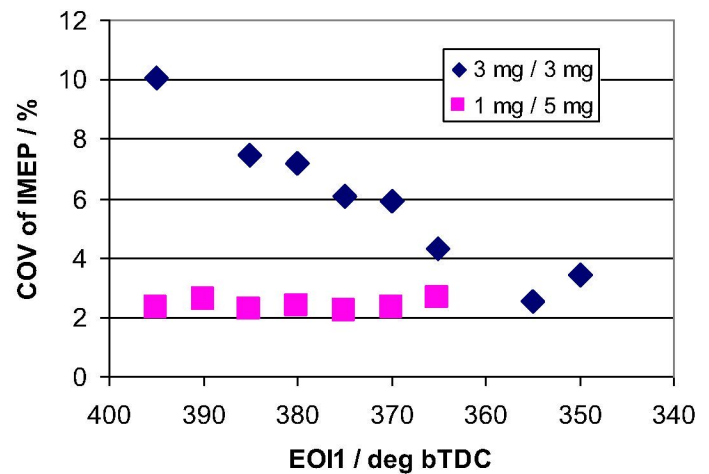


Figure 5: COV of IMEP as a function of first injection timing for 3 mg / 3mg and 1 mg / 5 mg split injection

The benefit of the early injection timing and the high amount of heat release during the recompression phase can be seen in the NO_x emission (Figure 6 shows the emission index NO_x in g NO_x / kg fuel). For the 1 mg / 5 mg case as well as for the 3 mg / 3 mg case the NO_x emission are reduced with earlier injection. For the 1 mg / 5 mg case however the emission index never drops below 1 g NO_x / 1 kg fuel and with very early injection the NO_x emission even increases again. With 3 mg fuel injection during the negative valve overlap period and early injection it is possible to reach levels well below 1 g NO_x / 1 kg fuel. As will be seen later this behavior is due to the amount of fuel that burns in a propagating flame before auto-ignition starts.

Another indicator that the amount of heat released during the recompression changes with injection timing is the significant influence of injection timing on the pumping work P_{MEP} (Figure 7).

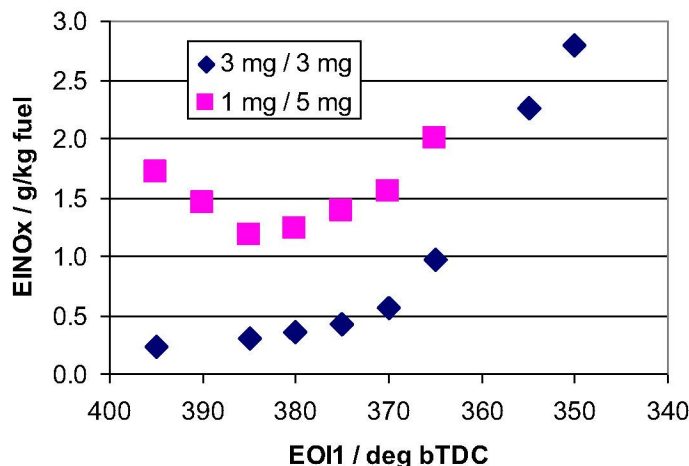


Figure 6: Average NO_x emission index (EINO_x in g NO_x / kg fuel) as a function of first injection timing for 3 mg / 3mg and 1 mg / 5 mg split injection

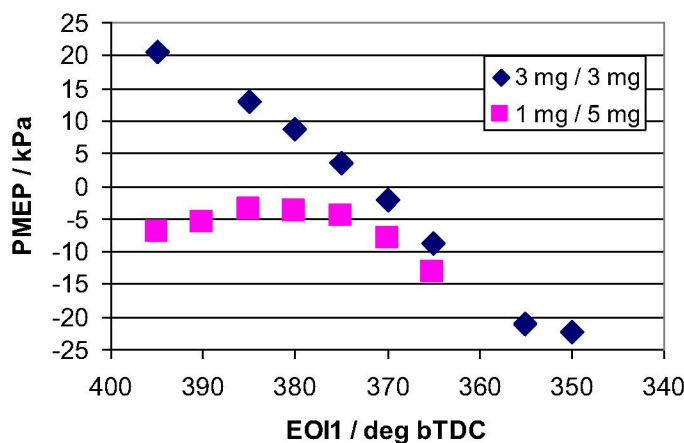


Figure 7: Ensemble average pumping work PMEP as a function of first injection timing for 3 mg / 3mg and 1 mg / 5 mg split injection

With late injection which does not lead to a significant heat release during the NVO period the pumping work is around -20 kPa. For the 1 mg / 5 mg case the pumping work increases with earlier injection timing but never reaches positive values. At very early injection the PMEP falls again, indicating lower levels of heat release during the recompression (the effect of which is also seen in the rising NO_x emissions with very early injection). This behavior is correlated to the main combustion event. The main combustion phasing (Figure 9) advances with the most advanced injection timing, resulting in lower exhaust gas temperature and reduced heat release during the recompression. For the 3 mg / 3 mg case the pumping work for early injection is positive, indicating high levels of heat release during the recompression. Note that the NSFC (and therefore the NMEP) hardly changes with injection timing because the IMEP decreases in cases where a lot fuel is consumed in the recompression period.

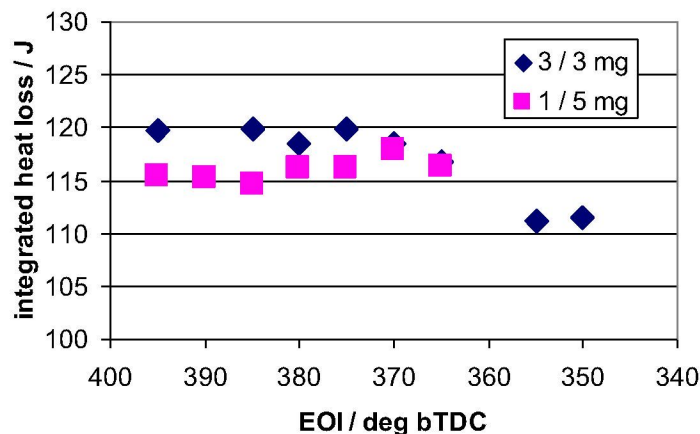


Figure 8: Ensemble average integrated heat loss as a function of first injection timing for 3 mg / 3mg and 1 mg / 5 mg split injection

The integrated heat loss (Figure 8) changes only for the 3 mg / 3 mg split ratio and shows higher values for earlier injection timing. This increase is the result of higher gas temperatures during the recompression period due to the increased amount of heat release with earlier injection timing.

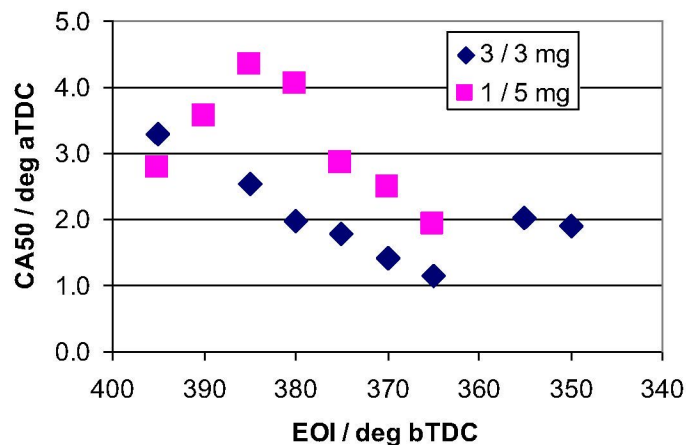


Figure 9: Ensemble average main combustion phasing (given as the crank angle when 50% of the fuel mass is burned, CA50) as a function of first injection timing for 3 mg / 3mg and 1 mg / 5 mg split injection

The combustion phasing of the main combustion event (shown in Figure 9 as the crank angle when 50 % of the fuel mass is burned, CA50) changes by approximately 2 crank angle degrees for each split ratio case and shows overall slightly earlier combustion phasing for the 3 mg / 3 mg split. These changes alone cannot explain the significant change of the NO_x emissions (Figure 6). It is therefore necessary to further investigate how the heat release progresses over time for the different injection strategies.