

Valve Profile Adaptation, Stratification, Boosting and 2-Stroke Strategies for Raising Loads of Gasoline HCCI Engines

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ABSTRACT

The development of high efficiency powertrains is a key objective for car manufacturers. One approach for improving the efficiency of gasoline engines is based on homogeneous charge compression ignition, HCCI, which provides higher efficiency than conventional strategies. However, HCCI is only currently viable at relatively low loads, primarily because at high loads it involves rapid combustion that generates pressure oscillations in the cylinder (ringing), and partly because it gives rise to relatively high NO_x emissions.

This paper describes studies aimed at increasing the viability of HCCI combustion at higher loads by using fully flexible valve trains, direct injection with charge stratification (SCCI), and intake air boosting. These approaches were complemented by using EGR to control NO_x emissions by stoichiometric operation, which enables the use of a three-way catalyst. Experiments were carried out using a single-cylinder engine of passenger car size running on gasoline and controlled with negative valve overlap. By adapting the valve profiles (lift, duration and phasing) for high loads, a fuel saving of 3% at constant load or a load increase of 6% could be achieved for lean HCCI compared to those obtained using camshafts that were not adapted for high load operation. Further, using intake pressures up to 180 kPa provided almost linear increases in load for lean HCCI, stoichiometric HCCI and stoichiometric SCCI. However, lean SCCI did not profit from boosting because the charge became too lean and stratification lost its effect as a ringing inhibitor. At intake pressures exceeding 140 kPa, stoichiometric HCCI operation becomes redundant since NO_x ceases to be a limiting factor.

Additionally, promising results were obtained in initial tests of two-stroke operation, which yielded higher maximum loads, lower fuel consumption and lower NO_x emissions than the other strategies.

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INTRODUCTION

There is great interest in reducing the CO₂ emissions and fuel consumption of road vehicles. For example, the European Union (EU) has stipulated that by 2020 the fleet average CO₂ emissions for all cars sold by a given manufacturer should be no more than 95 g CO₂/km. In 2009, the typical fleet average was 147.5 g/km, so substantial increases in the efficiency of cars' powertrains are urgently needed [1]. However, legislation is not the only factor driving the desire to develop cars with reduced fuel consumption: consumers are increasingly concerned with their environmental impact and the rising price of fuel.

Most modern gasoline engines use homogeneous charge spark-ignited (SI) combustion. This approach offers poor

efficiency under partial load because of intake air throttling and high thermal losses, but is popular because of its ease of control and favorable emissions profile when used in conjunction with a three-way catalyst. Homogeneous charge compression ignition (HCCI) is an alternative combustion strategy that offers superior engine efficiency. Systematic investigations of this combustion strategy using two-stroke engines were first reported in 1979, by Onishi et al. [2] and Noguchi et al. [3], and its application in four-stroke engines was initially described by Najt and Foster [4] in 1983. The absence of flame front propagation in HCCI combustion means that it can be operated with air dilution and residual gases, reducing both thermal and pumping losses. The main barrier currently preventing the commercial uptake of HCCI relates to combustion control: while the timing of combustion



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events in spark-ignited combustion is determined by the spark timing, there is no direct way of controlling the combustion timing in HCCI. HCCI engines thus require indirect control strategies, which in turn require some degree of combustion feedback. For this purpose, the authors have focused on a method known as negative valve overlap (NVO), in which the exhaust valve closes before top dead center (TDC) in the gas exchange phase. This retains hot residuals in the combustion chamber, which influence the combustion timing [5, 6, 7].

At low loads, HCCI combustion proceeds at low temperatures and thus has the advantage of generating very small levels of NO_x emissions. This could make the use of a lean NO_x aftertreatment system redundant, since a standard three-way catalyst (TWC) alone might be sufficient for oxidizing carbon monoxide (CO) and unburned hydrocarbon (UHC) emissions during lean operation. However, this might require improvement of the TWC's low temperature conversion efficiency since the temperature can be too low for full oxidation, especially of UHCs, when using standard catalysts designed for SI operation [8]. The small NO_x emission levels are the strongest incentive for implementing the more complex HCCI strategy rather than lean SI combustion (homogeneous or stratified), which requires extra lean NO_x aftertreatment [9].

However, for naturally aspirated HCCI controlled with NVO the combustion becomes too rapid at high loads, causing pressure oscillations, and too hot, generating high NO_x emission levels. The pressure oscillations observed during HCCI combustion are referred to as "ringing" [10], and are similar to knock in SI engines. Detailed examination of ringing using high speed video [11] has shown that the phenomenon is due to the creation of localized zones of excessively rapid combustion, giving rise to acoustic modes in the cylinder that vary according to the location of the zones.

One potential solution for the problem of ringing in HCCI combustion involves fuel stratification. Since this combustion strategy eliminates the homogeneity of the charge (one defining characteristic of HCCI), it is referred to as stratified charge compression ignition (SCCI). In SCCI, most of the fuel is injected at the end of the NVO period as in HCCI, with a smaller portion being injected near TDC firing, normally between 30 and 50 crank angle degrees (CAD) bTDC. This strategy was studied in [12], using a multi-cylinder engine to assess the potential for using SCCI to increase the upper load limit of HCCI-type combustion. The conclusion was that by using SCCI the upper load limit can be increased substantially, by up to 1 bar BMEP (brake mean effective pressure). However, this improvement came at the expense of increased NO_x emissions. That problem was eliminated by using exhaust gas recirculation (EGR) to maintain stoichiometric conditions, allowing use of a three-way catalyst to control emissions. While this strategy did not afford better fuel consumption than SI combustion, it could

potentially be used to delay the switch from HCCI-type combustion to SI combustion. Other researchers have also demonstrated that using unmixed charges reduces the rate of heat release (ROHR) [13, 14, 15, 16, 17, 18, 19]. In tests using a primary reference fuel that exhibits two-stage ignition, Sjöberg and Dec [17] showed that fuel stratification is an effective strategy for smoothing the heat release profile and reducing the rate of pressure increase. Conversely, Yang et al. [18] have argued that charge stratification with single-stage ignition fuels increases, rather than decreases, rates of pressure increase. However, they subsequently extended these studies, using boosting, and concluded that at high boost pressures (above 160 kPa abs.) charge stratification can also effectively reduce pressure-rise rates for operation with conventional gasoline [19]. This is related to a finding by Dec and Yang [20], that gasoline has intermediate temperature heat release (ITHR) at high boost pressures and thus may become sensitive to fuel concentration. The overall conclusion from the results presented by Dec and co-workers [17, 18, 19, 20] is that a fuel's auto ignition chemistry must be sensitive to fuel concentration in order for fuel stratification to be effective, but the cited studies were performed using a diesel engine with high compression ratio, and thus high cylinder pressures which negate any requirement for using NVO. In contrast, in the study presented in [12] and here, an engine with a low compression ratio and low cylinder pressures with NVO for combustion control and moderately air-diluted charges were used.

Fuel stratification can only increase an engine's upper load limit up to a certain point; ultimately, the maximum load is restricted by the amount of air available in the cylinder. The air flow can be increased by boosting the intake charge. This has been done in combination with NVO using either a compressor [21] or turbochargers [22, 23]. In turbocharger systems, the compressor is driven by the turbine, which in turn takes energy from the exhaust gases. Mechanically-driven compressors are an alternative to turbocharger systems; compressors driven by the crankshaft have traditionally been used to increase maximum torque, at the expense of reduced engine efficiency. In addition, electrically-driven compressors can be used for short durations when additional air is required. In HCCI, they could provide a useful way of avoiding mode shifts during accelerations, provided the accelerations are not too rapid.

The objective of the three-part study reported here was to increase the upper load limit of a gasoline HCCI engine controlled primarily by NVO. The first part of the study focused on adapting valve profiles for high-load HCCI operation using a fully flexible valve system. In the second part, the maximum loads for HCCI and SCCI, both lean and stoichiometric, were investigated at different intake air pressures. In the third part, the potential for operating the engine in two-stroke HCCI mode was briefly investigated since the flexibility of the valve system allowed for some preliminary tests.

EXPERIMENTAL SETUP & PROCEDURE

ENGINE

A single-cylinder AVL research engine was used for the experiments, with: a Volvo Car Corporation single-cylinder head, combustion chamber with pentroof-type geometry, a centrally placed sparkplug and solenoid-driven multi-hole injector that generated a horse-shoe shaped spray pattern. The piston used was designed for spray-guided stratified charge combustion and thus had a small bowl. The fuel pressure was set to 15 MPa in all experiments and a compression ratio of 11.4 was used. The bore and stroke were 83.8 mm and 90 mm respectively, giving a displaced volume of 496 cm³. A Roots blower-type compressor driven by an electric motor was used to supply boost pressure. Boosting the intake air raised the temperature of the charge, but the compressor was located several meters upstream of the engine, hence it was never more than 15°C warmer at the intake. Unless otherwise stated, the consequences of this effect were mitigated by maintaining the temperature at the intake port at 50°C using an electrical heater in the experiments involving boosting. A long-route EGR system was used, and the EGR level was defined as the measured concentration of CO₂ in the intake divided by the measured CO₂ concentration in the exhaust. A gasoline certification fuel (of well controlled specifications) with a research octane number of 101 was used in all experiments unless otherwise stated. The engine was run at a speed of 2000 rpm in all experiments except for some of the two-stroke trials, which included tests at lower speeds. A hydraulically actuated valve system manufactured by Lotus engineering was used, which provides complete flexibility when designing valve profiles and can switch between different profiles in real time. A schematic diagram of the setup (incorporating a photograph of the engine with the active valve train, AVT, system mounted) is shown in [Figure 1](#).

For comparative purposes, the engine was also operated using normal stoichiometric direct injected spark ignition. In these cases the spark timing was set for maximal efficiency and standard valve timings were used. The valve timings were: EVO=160 CAD aTDC, EVC=365 CAD aTDC, IVO=360 CAD bTDC and IVC=167 CAD bTDC with a lift of 10 mm. These valve timings provided moderate overlap and the engine's fuel consumption could perhaps have been improved by a few percent by tuning them to optimize the amounts of residuals. The valve timings reported in this paper are defined as points corresponding to 0.5 mm valve lift (during opening and closing) and the reference timing is at TDC firing (0 CAD) hereafter.

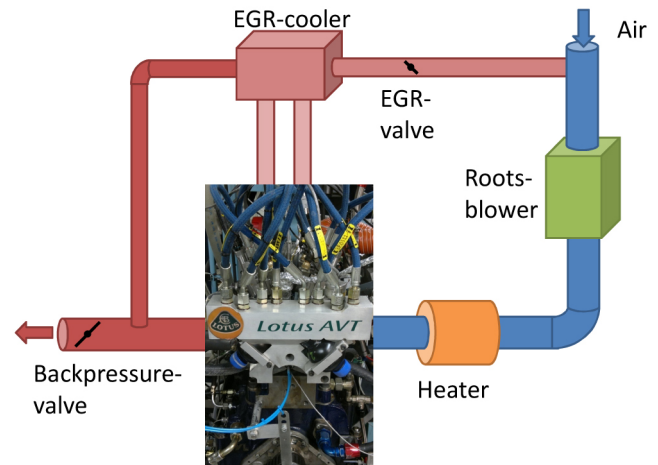


Figure 1. Schematic diagram of the setup incorporating a photograph of the engine with the active valve train (AVT) system mounted.

MEASURING EQUIPMENT

The emission measuring instruments consisted of a flame ionization detector (calibrated using propane) for UHC (J.U.M. Engineering VE7), non-dispersive infrared radiation detectors for CO (Maihak UNOR 611) and CO₂ (Maihak UNOR 6N), a chemiluminescence analyzer for NO_x (Eco Physics CLD 700 EL ht), a paramagnetic susceptibility analyzer for O₂ (Maihak UNOR 611), and an AVL Micro Soot sensor for particulate matter. The fuel flow was measured using a Micro Motion meter, which obtains estimates of the fuel flow by measuring the Coriolis force. Equivalence ratios were measured using a lambda meter from Horiba and verified by calculations using measured concentrations of exhaust gases. In-cylinder pressures were measured over 300 cycles at a resolution of 0.1 CAD around TDC and 1 CAD otherwise, using a cooled Kistler 6061b pressure sensor connected to a Kistler 5011 charge amplifier. An AVL Indimaster 670 system connected to a PC running the AVL IndiCom® 1.1 software package was used for data acquisition and real-time combustion analysis. A MatLab® code developed in-house was used for post-processing combustion analysis.

LOAD CONSTRAINTS

The constraints used to define maximum load, normally achieved by using the latest possible combustion timing, which is limited by combustion stability, are shown in [Table 1](#). Early combustion is limited by the maximum allowed ringing intensity (RI), here defined as the maximum amplitude of the high-pass filtered pressure trace for each cycle, averaged over 300 cycles. A cut-off frequency of 4000 Hz was found to be appropriate for this engine, since it captured all the fluctuations while keeping all other combustion information intact. The limit for RI was set to 1 bar, at which a clearly audible “knocking” sound appears.

The high load limit was thus set by the point at which the region between the ringing (early combustion) and unstable (late combustion) thresholds became too narrow for viable operation. The third load constraint, which was most important in the charge stratification experiments, related to the quantity of soot emitted. The Euro VI standards for passenger cars stipulate that soot emissions must not exceed 0.005 g/km, which translates to approximately 35 mg/kWh in the NEDC (New European Drive Cycle) for a medium-sized passenger car. In previous studies [12], a load constraint for soot emissions of 0.1 FSN (Filter Smoke Number) was employed; in simplified terms, this is equivalent to 17 mg/kWh. However, for the study reported here, an upper limit of 20 mg/kWh in net indicated specific soot (ISSoot) was used.

Table 1. Load constraints.

Parameter	Limit
RI [bar]	1
$\sigma(\text{IMEP}_n)$ [bar] for $\text{IMEP}_n < 3.3$ bar	0.1
CV(IMEP _n) [%] for $\text{IMEP}_n > 3.3$ bar	3
ISSoot [mg/kWh]	20

RESULTS

VALVE TIMING ADAPTATION

The first experiments examined the effects of varying valve timing parameters, such as valve opening/closing and valve lift, to see whether it would be possible to increase the maximum load by adapting the valve profiles. The initial valve settings were taken from a previous study conducted by our group in which a multi-cylinder engine was used [12]. Valve profiles matching those used for the multi-cylinder engine were created and phased in the same manner. A maximum load of 3.98 bar IMEP_n was reached in the single-cylinder engine using those profiles, while the corresponding load was 3.88 bar IMEP_n in the multi-cylinder engine. By phasing the cam profiles to optimal settings for the engine used here, a maximum load of 4.27 bar IMEP_n was achieved. This difference between the engines is probably due to differences in the geometries of their combustion chambers, and associated effects on ringing and the engines' gas exchange capabilities. In addition, it would be more difficult to achieve maximum load in every cylinder of the multi-cylinder engine than in the one cylinder of the single-cylinder engine. Finally, there may also have been differences in the responses and calibration of the pressure transducers used in this and the previous study.

A number of valve sweeps (in randomized order, most in duplicate) were performed at the outset of the study. Figure 2

shows data obtained from three different IVC timing sweeps; two with 3 mm lift and one with 5 mm lift, all three with fixed exhaust cam profiles. In experiments using a constant amount of injected fuel, the airflow and load were maximal at IVC=160–170 CAD bTDC. The range of viable IVC values is limited at the early end by excessive ringing (due to inadequate air supply, causing high residual concentrations and thus a hotter charge) and at the late end by combustion instability (due to low effective compression ratios delaying combustion). Since the amount of fuel injected was fixed, and the load was maximal within this IVC range, it was also optimal in terms of fuel economy. When the amount of fuel injected was allowed to vary to maximize the load, the airflow peaked at an IVC of around 170 CAD bTDC. However, with a 3 mm lift and variable amount of injected fuel, the maximum load occurred at a much later IVC, but resulted in increased fuel consumption.

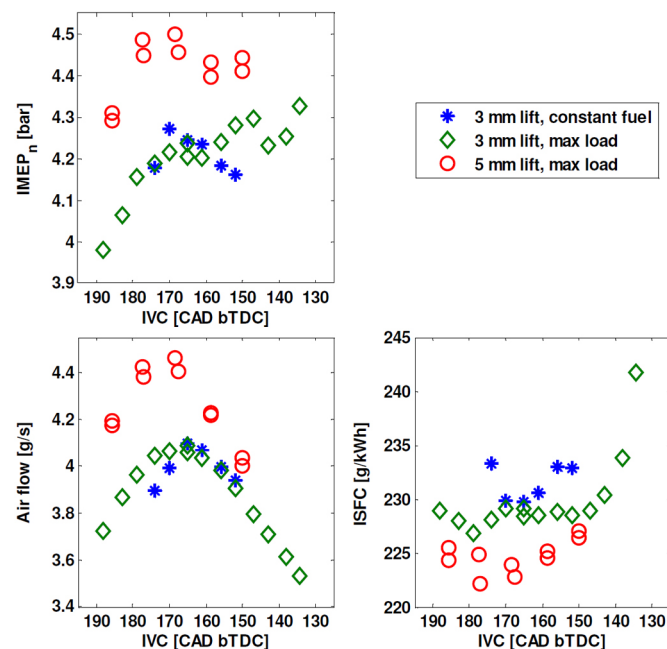


Figure 2. Effects of varying the IVC timing on IMEP_n (upper plot), air flow (lower left plot), and ISFC (lower right plot) at 3 mm lift with a fixed amount of injected fuel, at 3 mm lift under maximum load, and at 5 mm lift under maximum load.

An intake valve lift sweep was also carried out to see if higher loads could be achieved by increasing the lift from the 3 mm used in our previous studies. The full results of these experiments are not shown, but those displayed in Figure 2 clearly demonstrate that increasing the lift to 5 mm resulted in increased airflow and load. However, increasing the lift beyond 5 mm did not result in further improvements.

The standard NVO strategy is to position the opening of the intake valve and closing of the exhaust valve symmetrically around TDC. This is normally done to minimize the pumping losses associated with too early or too