

An Evaluation of Different Combustion Strategies for SI Engines in a Multi-Mode Combustion Engine

Daniel Dahl and Ingemar Denbratt
Chalmers University of Technology

Lucien Koopmans
Volvo Car Corporation

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ABSTRACT

Future pressures to reduce the fuel consumption of passenger cars may require the exploitation of alternative combustion strategies for gasoline engines to replace, or use in combination with the conventional stoichiometric spark ignition (SSI) strategy. Possible options include homogeneous lean charge spark ignition (HLCSI), stratified charge spark ignition (SCSI) and homogeneous charge compression ignition (HCCI), all of which are intended to reduce pumping and thermal losses.

In the work presented here four different combustion strategies were evaluated using the same engine: SSI, HLCSI, SCSI and HCCI. HLCSI was achieved by early injection and operating the engine lean, close to its stability limits. SCSI was achieved using the spray-guided technique with a centrally placed multi-hole injector and spark-plug. HCCI was achieved using a negative valve overlap to trap hot residuals and thus generate auto-ignition temperatures at the end of the compression stroke.

The experiments were performed using a 6 cylinder, 3.2 liter Volvo engine equipped with cam profile switching (CPS), variable cam timing (VCT) for both intake and exhaust valves, and a spray guided direct injection (SGDI) system. In conjunction with a fully programmable control unit these features allowed the engine to be run in all the tested modes without any hardware modifications.

Five operating points in the low load/speed zone of the engine map were optimized for fuel consumption using full factorial, 2-D and 3-D experimental designs (with centre points). The same operating points were used for each combustion strategy and their effects on exhaust emissions, combustion and fuel-consumption were evaluated.

INTRODUCTION

In 1998 the European Automobile Manufacturers Association, ACEA, signed an agreement to reduce the fleet average CO₂ emissions from new cars to 120 g/km by 2012 [1]. In 2005, the average value for new European cars was 160 g/km, meaning that a further 25% reduction was required [2]. There are numerous ways to reduce cars' CO₂ emissions, including reducing their mass, rolling resistance or aerodynamic drag coefficients, and improving the energy efficiency of components like power steering systems, air-conditioning systems, water pumps and generators. Unfortunately, however, cars are tending to become heavier rather than lighter, due partly to the extra weight of additional safety equipment and other accessories, and partly because cars are simply becoming larger due to customer preferences for increased comfort, roominess etc. This means that more effort has to be made to improve the efficiency of their propulsion systems, especially in the premium car segment where weight is of greatest concern. Legislation regarding nitrogen oxide (NO_x), unburned hydrocarbon (THC) and carbon monoxide (CO) emissions is also becoming increasingly stringent. For instance, in 2009, "Euro5" limits will start to apply in Europe, enforcing predicted emissions limits of 60, 100 and 1000 mg/km for NO_x, THC and CO, respectively (prognosis by Delphi [3]).

The first 4-stroke gasoline engine was run in 1876 [4] and today, over 130 years later, this type of engine is still the dominant prime mover for passenger cars. Furthermore, the stoichiometric spark ignition (SSI) combustion strategy used in the gasoline engine has not been seriously challenged for most of its history, due in large part to its relative simplicity. However, other more complex strategies have been developed recently that are competitive because of the low efficiency of the SSI strategy, especially in part load, which is mainly due to pumping losses caused by throttling of the intake air flow and the large heat losses from the high temperature combustion.

Furthermore, engine-out emissions of CO, THC and (especially) NO_x are relatively high when the SSI strategy is applied, although during the last 25 years these emissions have been successfully reduced using three-way catalysts, which will also probably be able to help meet anticipated future emission limits.

In the homogeneous lean charge spark ignition (HLCSI) strategy a leaner mixture is used, which reduces both pumping losses and the cycle temperature resulting in smaller exhaust and cooling losses. Since excess air is present during lean combustion THC and CO emissions are lower, unless the equivalence ratio (and thus the flame speed) is too low, in which case combustion becomes unstable and partial burns or complete misfires occur, thereby increasing THC emissions and fuel consumption. In contrast NO_x emissions peak when the engine is operated slightly lean since NO_x formation is dependant on the oxygen concentration and temperature. The equivalence ratio limit is dependent on various factors, e.g. residual gases, compression ratio and in-cylinder motion, and can vary somewhere between 0.8 to just under stoichiometric [4, 5].

In the stratified charge spark ignition (SCSI) strategy the fuel-air "cloud" is located solely around the spark-plug, surrounded by excess air with or without residual gases, at the time of ignition. No throttling at all is needed in this system, which reduces pumping losses to a minimal level. In addition, the excess air lowers the cycle temperature, thereby reducing thermal losses. SCSI, also called spray-guided gasoline direct injection (SGDI) is an improvement of the wall-guided GDI system developed during the 1990's, which is described in [6]. In the spray-guided system the time between the end of injection (EOI) and the spark is much shorter, leading to better stratification (thus improving THC emissions and fuel economy) than in the wall-guided system, in which fuel was injected earlier in the cycle and then guided by the piston up towards the spark-plug, leading to both over-mixing and piston wetting [7]. Because of the late EOI (when the piston is close to TDC) associated with the spray-guided GDI system, care must also be taken to avoid piston wetting when applying this strategy. The wall-guided GDI engine only showed a fuel consumption reduction of 5 to 12% compared to the SSI engine on the NEDC test cycle [8], while the BMW spray-guided GDI engine has shown a reduction of 20%, although other measures were also applied to reach these figures [9]. Compared to the other three combustion strategies the SCSI strategy is associated with greater risks of soot formation because the spark timing is limited to a small window near the end of injection, leaving little time for evaporation of fuel droplets and mixing [7], which may lead to diffusion combustion occurring in some parts of the combustion chamber.

While a spark is used to initiate combustion in the first three combustion strategies, auto-ignition is used in the homogeneous charge compression ignition (HCCI) strategy. A close to homogenous charge of air, fuel and (maybe) residuals from the previous cycle is compressed until the temperature is

sufficiently high for auto-ignition of the mixture to occur. One way of realizing this is by closing the exhaust valve early so that some of the hot residuals are retained in the cylinder, which helps to further increase the temperature in the compression stroke of the following cycle, as explained by Willand *et al.* [10] and Koopmans *et al.* [11]. By controlling the time when the exhaust valves closes, one also affects the temperature of the mixture and hence the time of ignition. A pilot injection in the gas exchange phase can also be used to increase the temperature of the mixture, and thus adjust the ignition timing, as discussed by Koopmans *et al.* [12]. As with SCSI, HCCI can run with wide open throttle (WOT), significantly reducing pumping losses. However, wall heat losses are higher in HCCI combustion than in SCSI combustion, due to the high mean temperature over the cycle [13]. Nevertheless, although the mean cycle temperature is high, the lack of flame propagation means that there is almost no temperature stratification, and thus NO_x emissions are very low in HCCI combustion [11, 12].

SCSI, HLCSI and HCCI are all globally lean strategies, and thus prone to a drawback associated with any lean-burn combustion strategy, i.e. that three-way catalysts do not work properly in operating conditions that are not close to stoichiometric because oxygen poisoning strongly inhibits removal of NO_x. Oxygen poisoning means that the surface of the catalyst becomes covered by oxygen, which must be removed by a reducing species before further absorption and dissociation of NO can occur [14]. A promising solution for this problem is to use a NO_x storage-reduction catalyst (NSR), also called lean NO_x-trap (LNT), to store or "trap" NO_x in the form of nitrate during lean burn operation and then release it during a short period of rich operation to react with HC, H₂ and CO [14,15]. The negative aspect of this strategy is that the mode of the engine has to be switched from time to time in order to reduce the nitrates. A second approach for reducing NO_x in a lean environment is selective catalytic reduction (SCR), in which reduction of NO_x competes with reduction of the relatively large amount of excess oxygen. HCs can be used as selective reductants, or ammonia, which has previously been used in power plants and heavy duty diesel engines, with the obvious drawback that a supply of ammonia must be maintained in the vehicle [14]. To avoid storage of the poisonous ammonia, urea can be used instead, which is then decomposed to ammonia in the system.

Another problem with lean burn strategies is that the exhaust temperature may be too low for efficient oxidation of THC and CO in the catalyst. Skoglundh *et al.* [16] discuss several methods for overcoming potential problems associated with low temperature exhaust gases, including variations in the composition and distribution of precious metals in catalysts, use of closed-coupled or heated catalysts, and trapping hydrocarbons.

It is very important to reduce engine-out emissions rather than merely emissions from the tail-pipe, since precious metals used in catalysts are expensive, and the size, cost and volume of the catalytic systems

needed are obviously correlated to the magnitude of required reductions in emissions.

Pressure traces for the four combustion strategies are shown in Figure 1. The high pressure shown for SCSi combustion is due to the WOT leaving more mass in the cylinder and the early combustion phasing, see Figure 2. HCCI combustion shows marked differences from the other strategies, with a rapid pressure rise around TDC caused by the relatively late start of combustion and very rapid heat release, see Figure 2. The pressure rise in the gas exchange phase caused by the negative valve overlap with early EVC and late IVO for HCCI combustion is also recognized in Figure 1.

Figure 3 shows in-cylinder pressure plotted against in-cylinder volume data obtained for each tested strategy. The areas enclosed under the lower parts of the plots correspond to the pumping losses of the engine. It can be seen that the area for HLCSi is somewhat smaller than the area for SSI combustion. This is due to the larger throttling in SSI combustion reducing the manifold pressure. The trace from SCSi combustion was taken when the engine was operated un-throttled, leading to almost zero pumping losses. Because of the compression in the gas exchange phase for HCCI combustion, the trace looks different from those obtained for the other strategies. In this particular case the pumping losses were positive, but in some cases when a large amount of pilot fuel is injected, the pumping losses may become negative, meaning that positive work is done on the piston in the gas exchange phase.

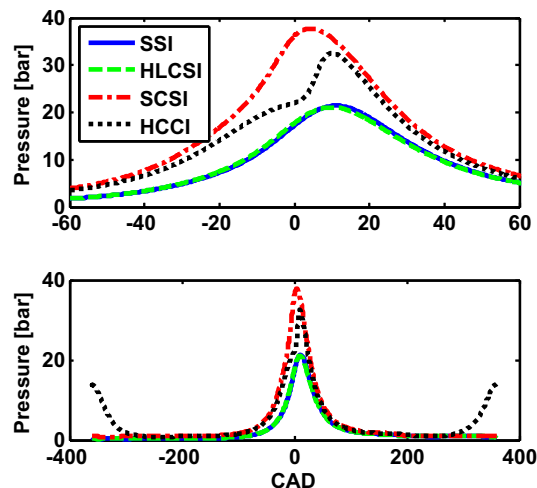


Figure 1 Average in-cylinder pressure traces obtained for the four combustion strategies at 2000 rpm, 2 bar BMEP from the six cylinders over 300 cycles. The upper panel of the figure shows the end of the compression stroke and start of the expansion stroke in detail, while the lower panel shows the whole engine cycle.

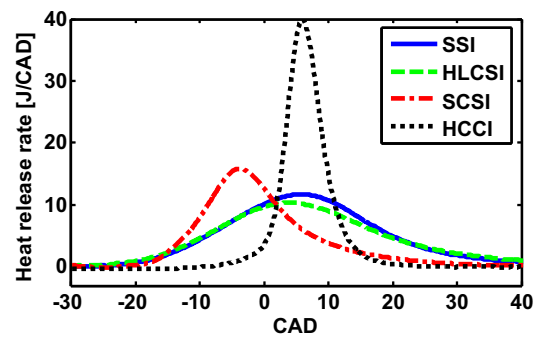


Figure 2 Average heat release rate obtained for the four combustion strategies from the six cylinders over 300 cycles at 2000 rpm, 2 bar BMEP.

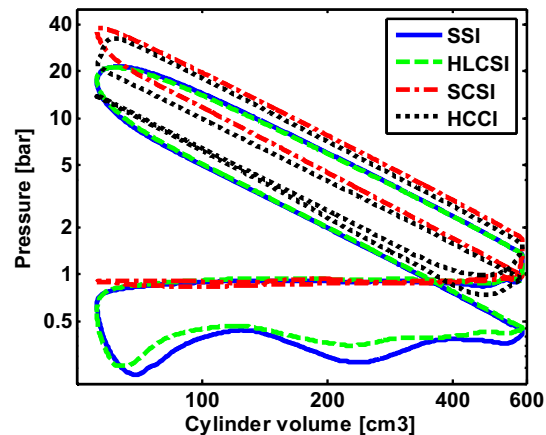


Figure 3 Average logarithmic in-cylinder pressure values obtained for each of the combustion strategies plotted versus logarithmic cylinder volume at 2000 rpm, 2 bar BMEP from the six cylinders over 300 cycles.

Previous comparisons of alternative combustion strategies for SI engines have been performed by [8], [13] & [17], but no experiments were performed by [8], experiments were performed on different engines (with a common base) by [17], and experiments were performed on a single cylinder research engine by [13]. In this work, all four of the combustion strategies were performed in the same engine capable of operating in all combustion modes with no hardware modifications. However, the engine was optimized for SSI and HCCI combustion with low in-cylinder gas motion and a flat piston, so the performance of the other strategies would probably have been better if engines optimized for them had been used. Unlike the previous investigations, this engine was also equipped with three-way catalysts to perform some analysis on conversion efficiencies of the emissions from the combustion strategies.

EXPERIMENTAL EQUIPMENT

ENGINE - The engine used for the experiments was a research 6 cylinder, 3.2 liter rebuilt Volvo engine equipped with cam profile switching (CPS) and variable cam timing (VCT) for both the intake and exhaust valves (Figure 4). The CPS system enabled two different cam profiles: one with low lift and one with high lift. The VCT system enabled the opening

and closing times of the valves to be varied 60 CAD on the intake side and 40 CAD on the exhaust side. The flexibility of the VCT/CPS system is illustrated in Figure 5.

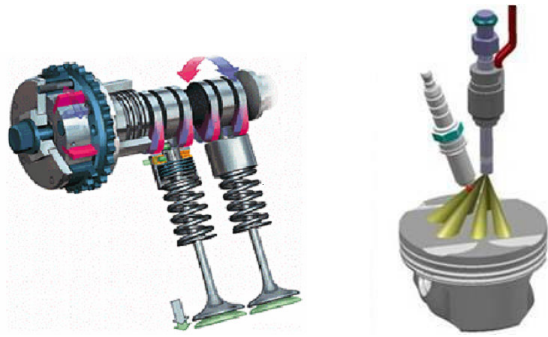


Figure 4 Illustration of the VCT/CPS system to the left and the combustion chamber to the right.

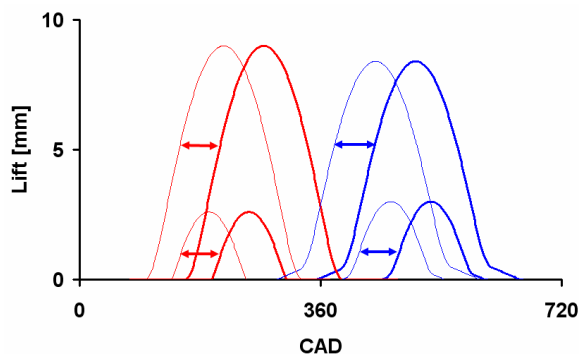


Figure 5 Possible valve profiles with CPS and VCT.

The engine had a fully programmable control unit controlled with a PC, allowing the spark timing, injection timing, injection pressure, injected fuel amounts, throttle and the VCT/CPS system to be controlled in real time and, hence, each of the combustion strategies (and shifts between them) to be controlled in real time without any hardware modifications. Engine parameters are listed in table 1.

The combustion chamber was not optimized for SCSi combustion since the engine had a flat piston, making it difficult to obtain a compact concentrated fuel/air mixture near the spark plug. The effects of this can be seen in the SCSi heat release rate in Figure 2, which very slowly decayed at the end of combustion, indicating that some of the fuel was over-mixed and hence burnt slowly. The injectors used were of multi-hole type and since the injectors were not guided and fixed in position when inserted into the cylinder head the spray-spark plug geometry could have slightly differed between the cylinders. Due to these two factors the engine is unlikely to have worked perfectly with a stratified charge. A schematic of the combustion chamber is illustrated in Figure 4.

The engine was equipped with two three-way production catalysts, designed to meet LEVII-ULEV emission legislations for SSI combustion, each treating emissions from three cylinders. A photograph of the Volvo 6 cylinder engine is displayed in Figure 6.

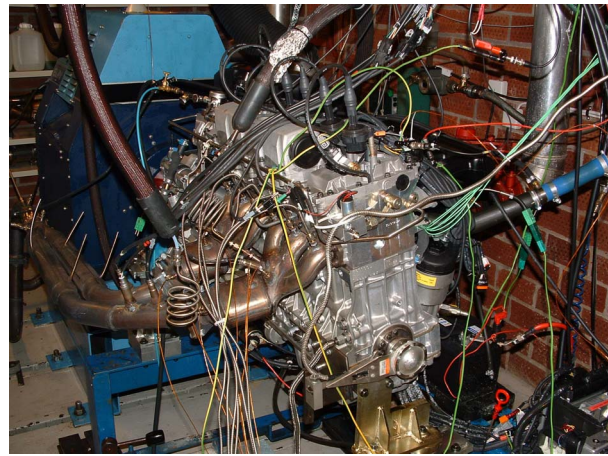


Figure 6 Photograph of the Volvo 6 cylinder engine

Table 1 Engine data.

Intake valve lift	3 and 8 mm
Exhaust valve lift	2 and 9 mm
Intake valve durations	130 & 230 CAD
Exhaust valve durations	110 & 230 CAD
Bore	84 mm
Stroke	96 mm
Compression ratio	11.5
Fuel (CEC legislative fuel)	RF-02-03, 99.5 RON
Inlet temperature	20-25 °C
Water / Oil temperature	90/97 °C

MEASURING EQUIPMENT - The emission measuring instruments consisted of a flame ionization detector for unburned THC, non-dispersive infrared radiation detectors for CO and CO₂, a chemiluminescence analyzer for NO_x, and a paramagnetic susceptibility analyzer for O₂. Soot was measured using an AVL 415 smoke meter. The gases for emission measurements were sampled both before and after the catalyst, except for smoke, which was only measured after it. The fuel flow was measured using both an AVL fuel balance, which measures changes in fuel weight over a certain period (in this case 100 seconds) and a Micro Motion meter, which obtains the fuel flow by measuring the Coriolis force. The Micro Motion gave more accurate, repeatable and real time fuel consumption values, so the fuel balance was only used as a backup system. Equivalence ratio values were measured using LA4 lambda meters from ETAS and a binary Lambda Sond™ was used for tuning towards stoichiometric mixtures. In-cylinder pressure was measured in each of the six cylinders at a resolution of 0.1 CAD, using Kistler 6053 pressure transducers connected to Kistler 5011 amplifiers. For data acquisition an AVL Indimaster 670 system was used which was connected to a PC loaded with IndiCom 1.5 software for real time combustion analysis. For the post-processing combustion analysis presented in this work an in-house MatLab code was used.