Homogeneous Charge Compression Ignition – the future of IC engines?

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ABSTRACT
The Homogeneous Charge Compression Ignition Engine, HCCI, has the potential to combine the best of the Spark Ignition and Compression Ignition Engines. With high octane number fuel the engine operates with high compression ratio and lean mixtures giving CI engine equivalent fuel consumption or better. Due to premixed charge without rich or stoichiometric zones, the production of soot and NOx can be avoided. This paper presents some results from advanced laser diagnostics showing the fundamental behaviour of the process from a close to homogeneous combustion onset towards a very stratified process at around 20-50% heat released. The need for active combustion control is shown and possible means of control are discussed. Results with multi-cylinder engines using negative valve overlap, variable compression ratio, fast inlet temperature control as well as dual fuel are given.

INTRODUCTION
The internal combustion engine is the key to the modern society. Without the transportation performed by the millions of vehicles on road and at sea we would not have reached the living standard of today. We have two types of internal combustion engines, the spark ignition, SI, and the compression ignition, CI. Both have their merits. The SI engine is a rather simple product and hence has a lower first cost. This engine type can also be made very clean as the three-way catalyst, TWC, is effective for exhaust aftertreatment. The problem with the SI engine is the poor part load efficiency due to large losses during gas exchange and low combustion and thermodynamical efficiency. The CI engine is much more fuel efficient and hence the natural choice in applications where fuel cost is more important than first cost. The problem with the CI engine is the emissions of nitrogen oxides, NOx, and particulates, PM. Aftertreatment to reduce NOx and particulates is expensive and still not generally available on the market. The obvious ideal combination would be to find an engine type with the high efficiency of the CI engine and the very low emissions of the SI engine with TWC. One such candidate is named Homogeneous Charge Compression Ignition, HCCI.

The fuel efficiency of HCCI has been compared to that of normal SI operation by Stockinger et al. [1]. Figure 1 shows that they noted an improvement of fuel efficiency from 15% to 30% at 1.5 bar BMEP. This is an improvement of 100% equivalent to a reduction of fuel consumption with 50%. More recently Yang et al. presented a comparison between HCCI, denoted OKP, and normal SI and direct injected SI concepts, DISI. He found a much higher fuel consumption benefit for HCCI than for DISI concepts.

The major benefit of HCCI compared to CI is the low emissions of NOx and PM. The CI engine normally has a trade-off between particulates and NOx. If the engine operates at conditions with higher in-cylinder peak temperature, the oxidization of soot will be good but the thermal production of NO will increase. If on the other hand the engine is operated with lower temperature NO can be suppressed but PM will be high due to bad oxidation. Figure 3 shows this trade-off and also the allowed emissions in EU and US today and in the near future. Clearly the CI engine must use exhaust aftertreatment of NOx and/or PM.

In the CI engine, NO is formed in the very hot zones with close to stoichiometric conditions and the soot is formed in the fuel rich spray core. The in-cylinder average air/fuel ratio is always lean but the combustion process is not. This means that we have a large potential to reduce emissions of NOx and PM by simply mixing fuel and air before combustion. In Figure 3 the normal emission level from an HCCI engine is also displayed. The NOx is normally less than 1/500 of the CI level and no PM is generated by combustion.

HCCI FUNDAMENTALS

THE HCCI PRINCIPLE – HCCI means that the fuel and air should be mixed before combustion starts and that the mixture is autoignited due to the increase in temperature from the compression stroke. Thus HCCI is similar to SI in the sense that both engines use a premixed charge and HCCI is similar to CI as both rely on autoignition for combustion initiation. However, the combustion process is totally different for the three types.

Figure 4 shows the difference between (a) SI combustion and (b) HCCI. In the SI engine we have three zones, a burnt zone, a unburned zone and between them a thin reaction zone where the chemistry takes place. This reaction zone propagates through the combustion chamber and thus we have a flame propagation. Even though the reactions are fast in the reaction zone, the combustion process will take some time as the zone must propagate from spark plug (zero mass) to the far liner wall (mass w_i). With the HCCI
process the entire mass in the cylinder will react at once. The right part of Figure 4 shows HCCI, or as Onishi called it Active Thermo-Atmosphere Combustion, ATAC. We see that the entire mass is active but the reaction rate is low both locally and globally. This means that the combustion process will take some time even if all the charge is active. The total amount of heat released, \( Q \), will be the same for both processes. It could be noted that the combustion process can have the same duration even though HCCI normally has a faster burn rate. Initial tests in Lund on a two stroke engine revealed the fundamental difference between these two types of engines. Figure 5 shows normal flame propagation from two spark plugs at the rated speed of 9000 rpm. We see two well defined flames and a sharp border between burned and unburned zones. Figure 6 shows the same engine when HCCI combustion was triggered by using regular gasoline (RON 95) instead of iso-octane. The engine speed was increased up to 17000 rpm and a more distributed chemiluminescence image resulted.

REQUIREMENTS FOR HCCI – The HCCI combustion process puts two major requirements on the conditions in the cylinder:

(a) The temperature after compression stroke should equal the autoignition temperature of the fuel/air mixture.

(b) The mixture should be diluted enough to give reasonable burn rate.

Figure 7 shows the autoignition temperature for a few fuel as a function of \( \lambda \). The autoignition temperature has some correlation with the fuels’ resistance of knock in SI engines and thus the octane number. For iso-octane, the autoignition temperature is roughly 1000K. This means that the temperature in the cylinder should be 1000 K at the end of the compression stroke where the reactions should start. This temperature can be reached in two ways, either the temperature in the cylinder at the start of compression is controlled or the increase in temperature due to compression i.e. compression ratio is controlled. It could be interesting to note that the autoignition temperature is a very weak function of air/fuel ratio. The change in autoignition temperature for iso-octane is only 50K with a factor 2 change in \( \lambda \). Figure 7 also shows the normal rich and lean limits found with HCCI. With a too rich mixture the reactivity of the charge is too high. This means that the burn rate becomes extremely high with richer mixtures. If an HCCI engine is run too rich the entire charge can be consumed within a fraction of a crank angle. This gives rise to extreme pressure rise rates and hence mechanical stress and noise. With a high autoignition temperature like that of natural gas, it is also possible that formation of NOx can be the load limiting factor. Figure 8 shows the NO formation as a function of maximum temperature. Very low emission levels are measured with ethanol. If the combustion starts at a higher temperature like with natural gas, the temperature after combustion will also be higher for a given amount of heat released.

On the lean side, the temperature increase from the combustion is too low to have complete combustion. Partial oxidation of fuel to CO can occur at extremely lean mixtures; \( \lambda \) above 14 has been tested. However, the oxidation of CO to CO2 requires a temperature of 1400-1500 K. As a summary, HCCI is governed by three temperatures. We need to reach the autoignition temperature to get things started; the combustion should then increase the temperature to at least 1400 K to have good combustion efficiency but it should not be increased to more that 1800 K to prevent NO formation.

HCCI COMBUSTION PROCESS IN DETAIL

The above description of HCCI gives just a rough idea about the requirements and conditions of the combustion process. It is also of greatest interest to acquire detailed knowledge of the process. In order to get such information, laser based diagnostics is of crucial importance. Some of the activities in this field from Lund University will thus be presented.

INHOMOGENEOUS COMBUSTION – The first experiments with laser based diagnostics were performed to analyze the difference in combustion between a perfectly homogeneous fuel/air mixture and one with small gradients. Laser induced fluorescence of fuel tracer or OH was used to mark the combustion process. Figure 9 shows the system setup with a laser generating a vertical laser sheet. Figure 10 shows the fuel distribution for the two cases with an inhomogeneity of approximately 5% in the case of port fuel injection and homogeneity within the detection limit for the case with a mixing tank and fuel injection far upstream. Figure 11 and Figure 12 shows the fuel concentration with half the heat released. We can from these images conclude that the combustion is far from homogeneous. There are islands with much fuel remaining and close to them regions with very little fuel left. Figure 13 shows the same behavior for the concentration of OH. Zones with much OH are close to zones with no OH and the gradients are steep. Each individual cycle was also found to be unique. The four cycles displayed are randomly picked samples. No preferred type of structure could be detected.

SINGLE CYCLE INFORMATION – A major limitation with the information from Figure 11 to Figure 13 is that only one image can be captured from each cycle. Due to the very large cycle to cycle variation in the process, it is impossible to extract information on possible expansion of zones with intense reactions i.e. flame propagation. To overcome this problem a unique laser system was used. Four individual lasers which can generate eight laser pulses were combined with a framing camera using eight
individual CCD chips. This system was used in an optical Scania engine with transparent liner and a window in the extended piston. The setup can be seen in Figure 14. The measured area was 95x55 mm thus enabling distinction between local and global effects. Figure 15 shows a sequence of fuel LIF images captured with 0.5 CAD time separations at 1200 rpm. The images are from 20% to 50% heat release. From these and numerous similar mini-movies it was possible to conclude that the combustion changed behavior during the process. In the initial phase a slow but stable decrease in the fuel LIF signal was detected. This was interpreted as a slow and rather homogeneous start of the process. At around 20-30% heat released the fuel LIF image changed. Then even the smallest structures found before were amplified to give an image with more intense gradients. The gradients were found to be amplified even more as the process evolved and at approx. 50% heat release the structures found earlier during the single shot experiments were clear. From 50% heat release and onwards the structures were stable and the fuel signal disappeared not long after that. This single cycle observation of the process leads to a phenomenological description of the HCCI combustion process.

THE PHENOMENOLOGICAL MODEL OF HCCI COMBUSTION – The HCCI combustion process is assumed to start with a gradual decomposition of the fuel with well distributed reactions. The reactions will become significantly exothermic when a critical temperature is approached. At this critical condition the reaction rate will be very sensitive to the temperature of the charge. Even the smallest variations in temperature will thus influence the reaction rates. As we will have random variation in temperature in the cylinder, some locations will have more favorable conditions. In those locations, sometimes denoted “hot spots” the reactions thus will start a bit earlier. As the exothermic reactions start the temperature is increased and thus reactions become even faster. We thus have a local positive feedback in temperature. Figure 16 shows an attempt to illustrate this. As the local positive feedback is fast, there will not be sufficient time to distribute all the heat to the surrounding cold bulk. Thus we have a gradual amplification of small inhomogeneities generating the very large structures seen in the experiments. The size of the hot spots was found to be of the same order as the integral length scale of turbulence in the cylinder. In the Scania engine, this was 4-6 mm.

Flame propagation? - It could be argued that the “hot-spots” grow as a function of time and this growth rate could be translated to a reaction zone propagation or in other words flame front. However, after studying numerous individual cycles it was concluded that the concept of flame propagation in HCCI could not be supported. There will be a time lag between combustion starting point at different zones but new “hot-spots” show up randomly and the structures seen in the images are rather fixed i.e. do not move from image to image. If we would use the term flame speed for a case where two hot spots show up at exactly the same time we would also have a problem as the flame speed then would be infinity.

THE NEED FOR CONTROL

For better understanding of the combustion process, laser diagnostics is needed and this knowledge can be used to optimize the system. However, the HCCI process is very sensitive to disturbances. It can be sufficient to change the inlet temperature 2°C to move from a very good operating point to a total misfire. This sensitivity makes the HCCI engine require closed loop combustion control, CLCC. Closed loop control requires as always a sensor, control algorithm and control means.

The main parameter to control for HCCI is the combustion timing i.e. when in the cycle combustion takes place. Figure 17 shows the rate of heat release for a range of timings. With early phasing the rate of heat release is higher and as it is phased later the burn rate goes down. With combustion before top dead center, TDC, the temperature will be increased both by the chemical reactions and the compression due to piston motion. Thus for a given autoignition temperature, combustion onset before TDC will result in faster reactions. With the conditions changed to give combustion onset close to TDC, the temperature will not be increased by piston motion, the only temperature driver would be the chemical reactions. This gives a more sensitive system and the later the combustion phasing the more sensitive the system is. This is the underlying problem with HCCI combustion control. We want a late combustion phasing to reduce burn rate and hence pressure rise rate and peak pressure but on the other hand we can not accept too much variations in the combustion process. How late we can go depends on the quality of the control system. With a fast and accurate control system we can go later and hence reduce the noise and mechanical loads of the engine.

COMBUSTION SENSOR – The most accurate and reliable signal for combustion is the in-cylinder pressure. With the standard heat release equation it is very easy to extract the combustion onset etc. The most usable parameter for combustion phasing is the crank angle of 50% of the heat released. Figure 18 shows the procedure to extract this 50% heat released point denoted CA50.

The cylinder pressure is a very stable and robust signal but the cost of such sensors is still too high for production engines. One alternative could be an ion current measurement system. The ion current can be measured by applying a voltage on the electrodes of a normal spark plug. The technique has been used by SAAB Automobile in production since 1993 for the detection of knock and misfire in SI engines, but the application on HCCI is not straightforward. The signal
intensity is very sensitive to the temperature in the cylinder and thus lean burn HCCI give low signal. Figure 19 shows a typical ion current measurement system and Figure 20 shows the typical signal obtained in HCCI mode. The best representation of combustion phasing was found by extracting the crank angle at which 50% of the maximum amplitude was detected. This gave good correlation to the crank angle of 50% heat released, CA50 as shown in Figure 21. Two individual operating points are shown, one with a relatively early timing and hence less cycle to cycle variations and one with late timing. For both cases, a small phase difference was detected between the crank angle at 50% of maximum ion signal and CA50 but this can easily be compensated by the controller.

CONTROL MEANS - The HCCI combustion control can be considered as a balance in temperature. With low temperature at TDC the combustion will be late and with high temperature at TDC the combustion will start early. To control temperature, three major parameters can be used. Inlet temperature and compression ratio will directly change the TDC temperature. The third parameter is the amount of residual gas retained in the cylinder from the previous cycle. A fourth possible way of controlling the process is to change the required autoignition temperature by adjusting the fuel quality. Figure 22 shows possible combinations of inlet temperature, compression ratio and fuel octane number for combustion onset at TDC for a 1.6 liter single cylinder Volvo Truck engine. The figure shows that a higher octane fuel needs higher inlet temperature or higher compression ratio to reach autoignition at TDC. Figure 23 shows similar combinations but here the two fuels are regular gasoline and diesel oil instead of the primary reference fuels n-heptane and iso-octane.

A very popular concept for achieving HCCI in SI engines at part load is the use of negative valve overlap. With this concept the exhaust valves close early and thus hot burnt gas is trapped in the cylinder. After a short compression and expansion the inlet valve is opened late. This type of process often denoted Controlled Autoignition, CAI, gives good performance but in a limited operating range. Figure 24 shows the operating range of a 6-cylinder 3 litre Volvo Cars engine. A better way of controlling the process is by applying variable compression ratio or fast inlet air temperature control. With this concept it is possible to run at idle at all engine speeds between 600 and 3000 rpm. Maximum load is the same as for CAI but it can be maintained also for higher engine speeds. Figure 26 shows the operating range for a SAAB 1.6 liter 5-cylinder variable compression engine using fast thermal management as shown in Figure 25. It should be noted that the BMEP is presented in contrast to the IMEP for CAI in Figure 24.

A possible way of HCCI combustion control can also be the use of dual fuels. Using two fuel tanks could cause some problems with customer acceptance but it is possible to generate two fuels from one using a reformer. Experiments with dual fuel in Lund have shown that it is a very powerful control means. Figure 27 shows the operating range possible with a Scania 12-liter 6-cylinder truck engine running on a mixture of ethanol and n-heptane.

CONTROLLER - In order to achieve the high loads reported for the SAAB and Scania multi cylinder engines, it is absolutely necessary to use closed loop control with a well tuned controller. To make the controller usable over the entire speed and load range, the gain of the controller must be changed in accordance with the change of gain of the process. Figure 28 shows the combustion phasing, CA50, as a function of octane number for the Scania dual fuel engine at different operating conditions. With early combustion timing and conditions requiring low octane number, the slope of the curves are low. This means that a large change of octane number is needed to change the combustion timing one crank angle. Thus we should have a large gain of the controller in these operating conditions. If we then look at conditions with high octane number and late combustion phasing, the required change in octane number to change phasing a crank angle is much less. With this higher gain of the process we must reduce the gain of the controller; otherwise the system will become unstable. Tuning the gain of the controller to compensate for changes in the process can be done by using gain scheduling. With this it is possible achieve close to optimal performance for all operating conditions. In fact it is even possible to operate an HCCI engine at unstable operating points with the closed loop combustion control active. Figure 29 shows one such case.

TABLES AND FIGURES

![Figure 1: The fuel efficiency of HCCI and SI engine configurations. Open diamond =SI at $\lambda=1$, $R_c=18.7$, Open triangle = HCCI lean burn, Filled diamond= HCCI with EGR and Filled circle= SI at $\lambda=1$ and $R_c=9.5:1$ [1]](image_url)
Figure 2: Net indicated specific fuel consumption of four different combustion types [2]

Figure 3: The NOx-PM trade-off for a standard diesel engine, the future emission regulations and the emissions of HCCI (Green)

Figure 4: The difference between SI and HCCI combustion process. $Q =$ total amount of heat, $q =$ heat per mass unit, $w =$ mass [3]

Figure 5: SI flame propagation in 2-Stroke engine at 9000 rpm [4].

Figure 6: HCCI combustion in 2-Stroke engine at 17000 rpm.

Figure 7: Ignition temperature for a few fuels as a function of dilution ($\lambda$).
Figure 8: NOx as a function maximum temperature evaluated from the pressure-trace [5].

Figure 9: First laser system [6].

Figure 10: Fuel distribution with port fuel injection (left) and far upstream (right) just before combustion starts. Four individual cycles are shown. [6]

Figure 11: Fuel distribution at approx. 50% heat released with port fuel injection [6].

Figure 12: Fuel distribution at approx. 50% heat released with mixing tank [6].

Figure 13: OH signal at approximately 50% heat released. To the left with port fuel injection and to the right with mixing tank [6].
Figure 14: Optical system for high speed fuel LIF [7].

Figure 15: Fuel concentration from 2 CAD ATDC with 0.5 CAD step [7].

Figure 16: Temperature at three instants of time [8].

Figure 17: Rate of heat release with a change in inlet temperature and thus combustion phasing [9].

Figure 18: Cylinder pressure trace and corresponding heat release [10].

Figure 19: Ion current measurement system [11].
Figure 20: Ion current signal with a change in combustion timing. The average of 300 cycles is shown [11].

Figure 21: Crank angle for 50% of maximum ion signal vs. crank angle at 50% heat released for two individual operating points [11].

Figure 22: Combinations of fuel octane number, compression ratio and inlet temperature to give combustion onset at TDC [12].

Figure 23: Combinations of percentage gasoline, compression ratio and inlet temperature to give combustion onset at TDC [12].

Figure 24: Operating range of Controlled Autoignition type of HCCI in a 6-cylinder Volvo Car engine [13].

Figure 25: Fast Thermal Management, FTM [14].
CONCLUSION

The Homogeneous Charge Compression Ignition, HCCI, combustion process is an interesting alternative to the conventional Spark Ignition and Compression Ignition processes. The potential benefit of HCCI is high with simultaneous ultra low emissions of NOx and PM and low fuel consumption. Thus it can combine the best features of the SI (with TWC) and CI engines. To better understand the process, laser based techniques must be used. Such measurements in Lund have revealed that the combustion process is rather homogeneous in the initial stage but it gradually transfers into a highly inhomogeneous process with steep gradients between reacting and non-reacting zones.

The HCCI engine requires active control of the combustion process. Such closed loop combustion control has been demonstrated in a number of multi-cylinder HCCI engines in Lund. Use of negative overlap is possible but often generates a limited operating range. The use of variable compression ratio is a very powerful control means but can have some problems to reach production for cost reasons. Fast Thermal Management can perhaps be the key technology to be used for HCCI combustion control.

The maximum engine speed for HCCI in Lund is 17000 rpm and the maximum load is 20.4 bar IMEP/16 bar BMEP. This indicates that most interesting speeds and loads can be reached with HCCI.

ACKNOWLEDGMENTS

The results presented in this paper are a summary of results of the HCCI activities in Lund. I thank all fellow researchers, Ph.D. students and technicians for generating the results. I would also like to thank our sponsors: The Swedish Energy Administration, The Swedish Gas Centre, Volvo Cars, Volvo Trucks, Volvo...
Penta, Scania CV, Saab Automobile, Fiat-GM Powertrain, Caterpillar, Cummins, Toyota, Nissan and Hino.

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