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ABSTRACT

Homogeneous Charge Compression Ignition, HCCI, has the attractive feature of low particulate and low NOx emission combined with high efficiency. The principle is a combination of an Otto and a Diesel engine in that a premixed charge is ignited by the compression heat.

One of the main challenges with the HCCI combustion system is to control the combustion timing/phasing for varying load and external conditions. A method to achieve this on a cycle-by-cycle basis is to vary the valve timing based on a feedback signal from the combustion timing of previous cycles.

A combined engine and control simulation is performed. The simulations are accomplished with a commercial cycle simulation code linked with a commercial control simulation code. The simulations are iteratively verified against engine test data. Engine tests are conducted on a single cylinder engine equipped with at hydraulic valve system that allows a high degree of freedom in choosing the valve timings. A novel very fast hybrid hardware-software system to evaluate combustion timing is described and implemented on the test bed. Combustion timing estimated with this new system has been used as feedback to control the valve timings for the oncoming cycle.

The results show that combustion phasing successfully can be controlled by aid of variable valve timing during varying external conditions. Engine tests describing the control performance achieved in initial experiments are reported. Net Indicated Mean Effective Pressure, net IMEP, of up to 5.6 bar has been tested. The work described forms the basis for further studies of HCCI control.

INTRODUCTION

HCCI COMBUSTION

HCCI has the attractive feature of low particulate and low NOx emission combined with high efficiency. The principle is a combination of the Otto and Diesel engine. As in an Otto engine, the combustion charge is premixed. The mixing can either be done, as for example here, by injectors in the intake ports or by early direct injection as for example in [1]. The mixture is during the compression event exposed to high pressure and high temperature causing it to reach the point of ignition. Hereby the charge is ignited by the compression heat as in a Diesel engine.

The following combustion of the charge is neither Otto nor Diesel like. If the charge is homogeneous enough optical analyses in [2] show that the charge burns from a number of points in the combustion chamber. The combustion starts where heterogeneities in temperature and air/fuel ratio have created favorable conditions. During the combustion event new kernels are formed at places were the global pressure rise has lead to ignitable conditions.

The speed of the combustion is governed by chemical kinetics and hereby strongly influenced by concentration of the reactive species. As a result, only lean mixtures are possible to burn with respect to limitations in pressure rise and pressure levels. The chemical kinetics also causes too lean mixtures to misfire.

The fact that the combustion takes place in the whole combustion chamber at lean conditions results in moderate maximum temperature and thus results in very low NOx emissions. Charge that has reached down in narrow crevices will not burn and be one of the main contributors to the engine out HC emissions. The result is that the combustion system shows a, by comparison with a Diesel combustion system, relatively high engine out HC emissions and low combustion efficiency.

The reactions that lead to the actual combustion are governed by temperature, pressure, concentrations of the participating species and time along the compression event. Different kinds of behavior can be seen during this stage. For certain fuels, mixture ranges, pressure ranges, and temperature ranges there might be a pre-reaction behavior called cool flame occurring as a step toward the actual combustion.

The above dependencies results in a rather narrow operating window that makes timing of the combustion event, over a wide range of inlet conditions, one of the main obstacles to deal with when running a HCCI-engine. This is especially emphasized during transient conditions.

CONTROL OF HCCI COMBUSTION

To be able to run a HCCI engine during transient conditions timing control of the combustion is the basic requirement. Different approaches have been proposed.

In [3] a method called thermal management was analyzed through simulation. The idea was to use exhaust energy to heat the intake charge.

In [4] the combustion were timed by aid of dual-fuels with radically different octane numbers. The results showed that reasonably fast transient behavior could be reached. In [5] a thermal efficiency of 41.2 % and a load of 16 bar Brake Mean Effective Pressure, BMEP, were achieved with the same control structure.

In [6], [7] and [8] it has been shown that the combustion can be controlled by capturing hot residual gases by aid of modified or flexible valve trains over a wide range of operating conditions.

This far only [4] and [5] have shown real transient capabilities. In [4] and [5] the control structure was a gain scheduled PID-controller. The controlled variable was the fifty percent burnt angle, CA50.

The task to correctly time the combustion requires feedback from the combustion chamber in some way or other. The most direct way of achieving this is by in-cylinder pressure measurements. This method is today expensive and not possible to realize in series production. A drive to make the whole engine control system cylinder pressure based [9], [10] and [11] makes small/cheap in-cylinder pressure transducers foreseeable in the future. The analysis of the cylinder pressure can be done by direct heat release analysis, as in [4], or less computer expensive based on the Rassweiler and Withrow relationship, described in [12], as in [10] and [11].

SCOPE OF WORK

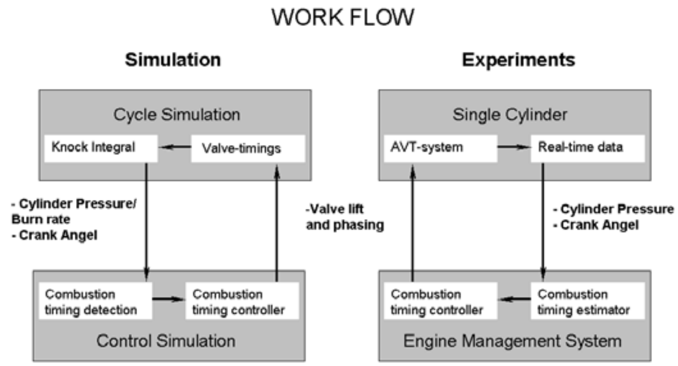

Figure 1. Work flow.

In order to be able to control the HCCI combustion during transient operation a feedback controller is required.

In this early stage it is necessary to investigate the basic engine behavior on the test bed as a basis for control design.

The task to construct a combustion timing controller is greatly assisted by simulations. Simulations can hereby be used to provide ideal signals, basic control parameter tuning and contribute to increased dynamic understanding thereby providing a base for future model based control algorithms.

The scope of the work described in this paper is shown in figure 1 and can be summarized as:

- Initial engine tests to demonstrate the control possibilities and the principal behavior.
- Investigation of a method to perform transient HCCI cycle simulations.
- Closing of the control loop in engine test as well as in simulation.

EQUIPMENT

TEST BED

The test bed that has been used is located at the division of Internal Combustion Engines, department of Machine Design at the Royal Institute of Technology (KTH). It comprises a single cylinder engine equipped with a Lotus Active Valve Train System, AVT-System. The engine, shown in figure 2, is based on a SCANIA D12 cylinder with the following data.

Table 1. Engine data.

Swept volume	1.95 dm^3
Compression ratio	18.0
Bore	127 mm
Stroke	154 mm
Connecting rod length	255 mm
Number of valves	4

The test bed is equipped with an external compressor that can provide up to 6 bar of boost pressure. On the exhaust side makes an exhaust pressure governor it possible to reach about the same level of exhaust backpressure.

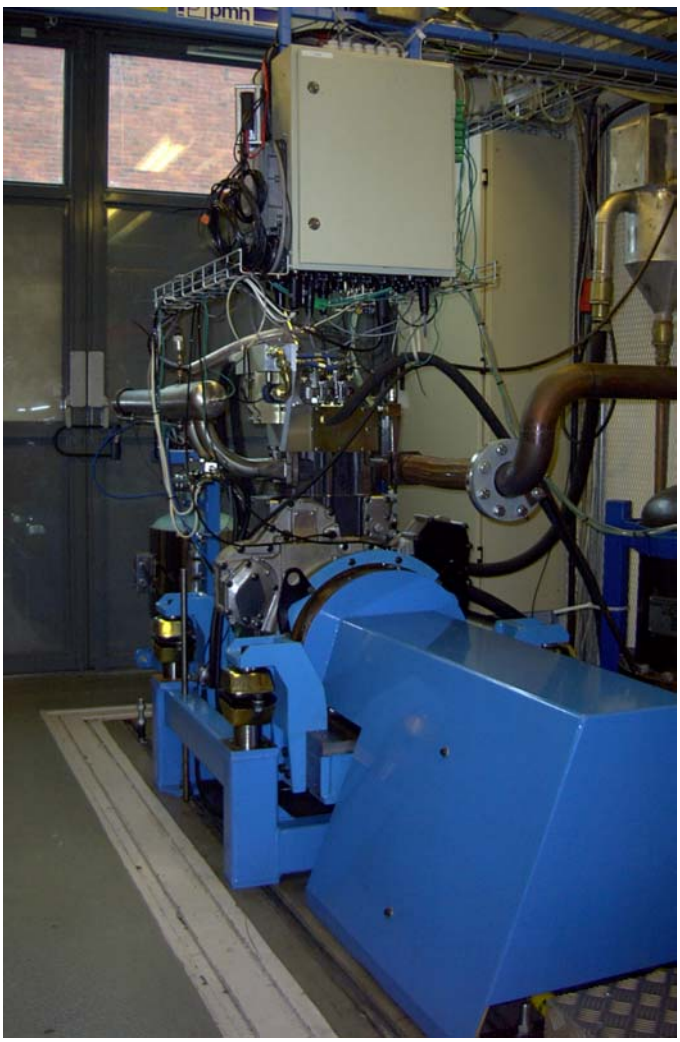

Figure 2. Test bed picture.

For the engine tests reported in this paper the fuel system is port injection and the fuel used is commercial gasoline 98 RON. The combustion chamber geometry is a flat cylinder head combined with a flat piston crown. The cylinder pressure has been sampled with an increment of 0.1 Crank Angle Degree, CAD, using an AVL 12Qp505clk piezo electrical transducer. The inlet pressure has ranged from 1.0 bar to 1.6 bar absolute while inlet temperature has been in the range of 70 °C to 95 °C.

The test bed is managed through an in house developed test bed management system. The management system comprises of five PIC micro controllers and one PC. The user interacts with the system through the PC, which communicates with the micro controllers. The micro controllers in turn handle the various “engine close” tasks for example fuelling, inlet pressure control, speed control etc. This system has a combustion timing estimator and a combustion timing controller. The combustion timing controller controls the combustion phasing by actuating the valve timings.

<u>AVT-System</u>

The AVT-System is a hydraulic valve system supplied by an external subcontractor and is in principal a stand-alone system. The AVT hardware is briefly described in [7]. The system allows a high degree of freedom in choosing the valve timings.

In the AVT-system there is space for 255 predefined valve lift profiles. The system allows switching between these profiles and phasing of them “on the run” individually for the four valves. A CAN link connects the test bed management system and the AVT-system allowing the test bed management system to choose and phase valve profiles on a cycle-by-cycle basis.

SIMULATION

The simulation is performed with a commercial cycle simulation software, GT-Power, linked with a commercial control simulation software, MATLAB®/Simulink. The two codes are linked through the supplied interface. The principle is that the cycle simulation and the control simulation run independently, managed from the control simulation, and exchange information at a fixed time interval. From the cycle simulation, cylinder pressure/burn rate and crank angle are passed to the control simulation. Based on these quantities the control simulation calculates required valve-timing which is passed back to the cycle simulation.

BASIS FOR ENGINE TEST AND SIMULATION

VALVE TIMINGS

Two different valve-timing strategies have been used to control the combustion phasing.

The first strategy used is varying the negative valve overlap, the overlap-method. Overlap is defined as the crank angle difference according to equation 1.

$$\textbf{Overlap} = \textbf{EVC} - \textbf{IVO} \quad [1]$$

EVC - Exhaust Valve Closure.

IVO - Inlet Valve Opening.

The strategy has been reported by other authors in [6], [7] and [8]. As shown in figure 3 and 4 an increased negative overlap phases the combustion earlier. The cause is that the residual gases, i.e. internal EGR, that are trapped from the previous cycle are hot and thereby promoting reactions [8].

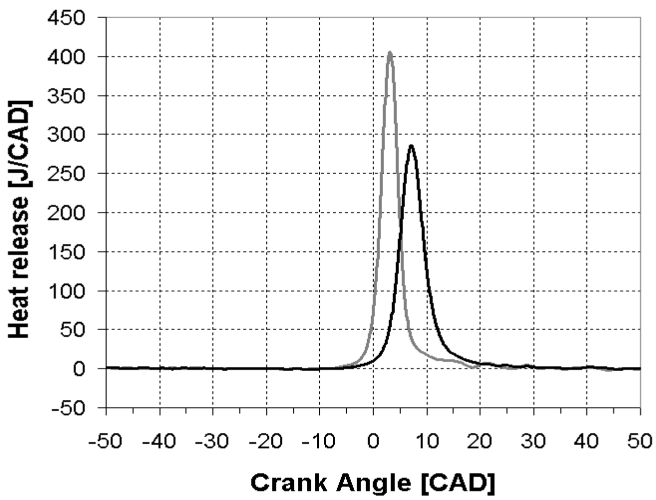

Figure 3. Heat releases, from engine tests, for the two different negative overlap that are shown in figure 4.

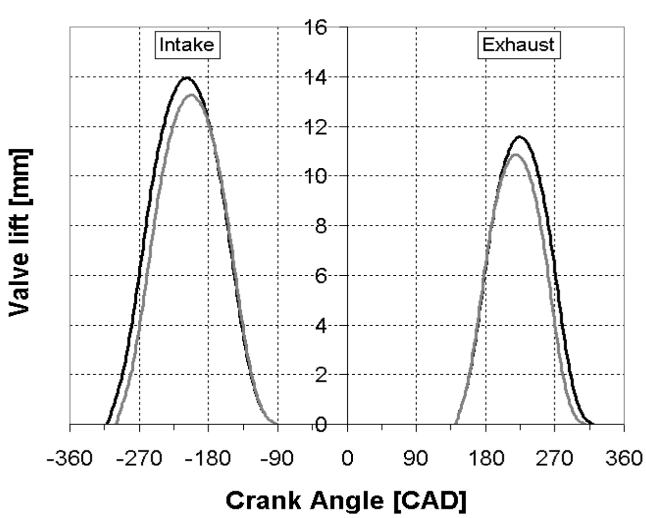

Figure 4. Valve timings to achieve the heat releases in figure 3.

The second strategy used is varying the Inlet Valve Closure, IVC. The strategy is referred to as the IVC-method and is described in figure 5 and 6. The strategy works by affecting the effective compression ratio.

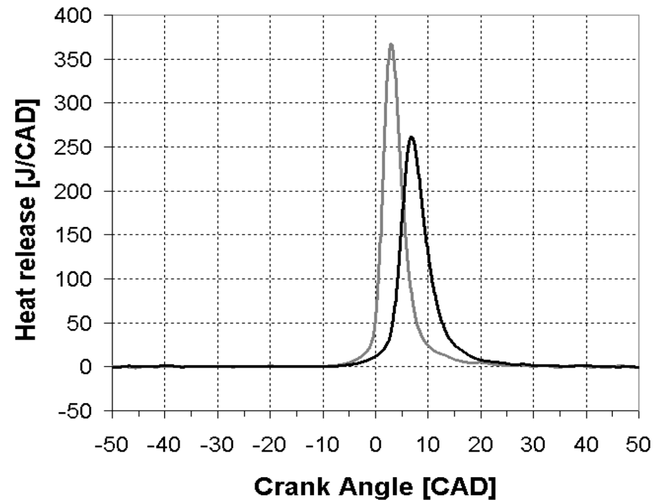

Figure 5. Heat releases, from engine tests, for the two different inlet valve closures that are shown in figure 6.

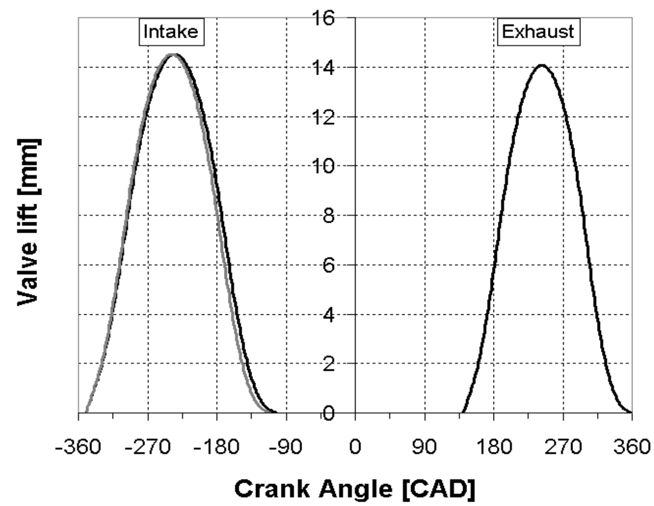

Figure 6. Valve timings to achieve the heat releases in figure 5. Observe that the exhaust lift is unaltered for the two cases.

By examining figures 3 to 6 one can be lead to believe that the overlap-method is a weak control strategy, requiring a large amount overlap change to impose a certain combustion timing change, and that the IVC-method is a strong control strategy, requiring a small amount IVC-change to impose a certain combustion timing change. That conclusion is correct for these particular valve timings but should not be generalized. The strong behavior of the IVC-method is in this case a consequence of the fact the IVC-area is placed in the crank angel region where the piston velocity peaks. The result is that an IVC change results in a relatively large change in the effective compression ratio. The late IVC area is in turn a result of the decision to run a rather high compression ratio and a will to test the two strategies in the same valve timing region, while being able to conduct various inlet condition disturbances. The table below explains the valve timing used for the different control methods.

In table 2 the following unexplained abbreviation is used:

EVO - Exhaust Valve Opening.

Table 2. Valve timing ranges for the two different control methods.

Control method	**Overlap**	**IVC**
IVO [CAD]	-350 to –266	-350
IVC [CAD]	-90	-140 to –56
EVO [CAD]	141	141
EVC [CAD]	276 to 360	360

The ability to store 255 different valve profiles in the AVT-system and the valve timing ranges in table 2 results in that the overlap can be chosen in steps of 2 CAD between –10 CAD and –178 CAD. The IVC can from the same set of valve profiles be chosen in steps of 1 CAD between –140 CAD and –56 CAD.

TEST BED

Combustion timing estimator

In the KTH test bed a single chip computer, micro controller, has been programmed in a way that essentially uses the Rassweiler/Withrow [12] observation that mass fraction burnt is approximately equal to the percent pressure rise due to combustion. In contrast to Rassweiler/Withrow is the volume change during combustion not correctly accounted for. This approximation is although small for the fast HCCI combustion that burns in a narrow crank angle region with close to constant volume. The method does not account for the combustion efficiency.

The I/O interface in the micro controller measures crank angle, inlet pressure and cylinder pressure. Based on this the micro controller continuously calculates the compression pressure from a starting point before TDC according to the relationship below.

$$P = P_0 \left(\frac{V_0}{V} \right)^n \qquad [2]$$

P - Compression pressure

P_0 - Cylinder pressure at the starting point

V_0 - Cylinder volume at the starting point

V - Cylinder volume

n - Polytropic exponent

V is a function of the crank angle. To speed up the calculation $(V_0/V)^n$ has been stored as a table in the single chip computers ROM-memory. P_0 is read from the pressure sensor at V_o. Due to the fact that the cylinder pressure is measured with a piezo electrical transducer it must be adjusted in level. This is done once per cycle by leveling it with the inlet pressure at an angle close to bottom dead center.

The calculated cylinder pressure is from –30 CAD in hardware subtracted from the measured cylinder pressure. This is accomplished by that a voltage proportional to the calculated pressure are generated and electronically subtracted from the voltage that represents the measured cylinder pressure. The voltage is generated through a D/A converter. When combustion occurs there will be a sharp rise from the zero level of the subtracted voltage. Through comparison in a comparator with a reference voltage, is the combustion event detected as a toggle in a digital level. This toggle occurs at the instant when the combustion is detected and is thereby a very fast way to indicate combustion timing. The result is available to a control algorithm at this instant, thereby giving maximum time to control calculations. The toggle level can be set constant or as a percentage of the previous cycles maximum voltage rise and thereby closely resemble CA50 calculated by a direct heat release analysis. The principle is shown in figure 7.

A problem in the above mentioned compression pressure calculations is that the polytropic exponent n in equation 2 varies due to load, speed, engine temperature, wear state and so on. This might lead to deviations from the ideally flat line corresponding to the voltage difference during the compression. The deviations can reach values that require an algorithm for exponent adjustment in order to get the system to work. A implemented solution is to tabulate $(V_0/V)^n$ as two tables. One corresponding to a high n and one corresponding to a low n. The compression pressure is then calculated as the sum of the compression pressure calculated with both exponents weighted with a factor. The deviation from the straight line in the previous cycle is hereby used to determine the weight factor for the oncoming cycle. This is done by forcing the ideally straight line to a determined voltage reduction as shown in figure 7. Essentially a to high n is deliberately used.

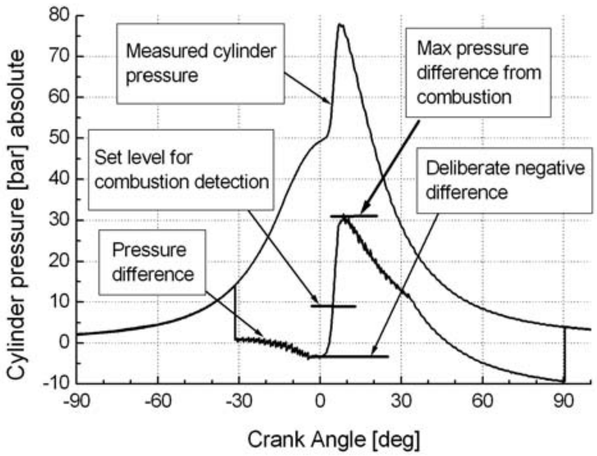

Figure 7. Explanation of the relationships in the combustion timing estimator.

Combustion timing controller

The combustion-timing controller is of PID-structure with the control law according to equation 3.

$$\mathbf{u} = \mathbf{K_P} * \mathbf{e}(\mathbf{t}) + \mathbf{K_I} * \int_0^t \mathbf{e}(\mathbf{s})\mathbf{ds} + \mathbf{K_D}\frac{\mathbf{d}}{\mathbf{dt}}\mathbf{e}(\mathbf{t}) \quad [3]$$

u - Actuated quantity

e - Error, difference between the desired value and the actual value, feedback value, of the controlled parameter

K_P - Proportional gain

K_I - Integral gain

K_D - Derivative gain

The gains determine the behavior of the controller with respect to speed, stability and accuracy. Generally speaking higher gains result in a fast controller and small gains result in a slow controller. A fast controller is although more likely to show an unstable behavior which thereby limits the possible gains. In the combustion timing controller only the P and I part have been used, K_D=0.

The feedback signal is the estimated combustion timing signal averaged over the last five cycles. The reason for averaging is that the combustion has cycle-to-cycle variations. These variations have to be damped to make the controller stable with reasonable performance. The averaging is obviously not a very good way to filter the combustion timing signal but was the fastest way to get the system in operation. The desired combustion phasing is given as input and the controller actuates the valve timing to achieve the desired combustion timing over the CAN link. Due to the fact that the valve timing has a discrete resolution as discussed in the "Valve timings" section, which the controller output has not, the controller output is rounded to the nearest integer, or the nearest even integer in the overlap case, before being passed to the AVT-system.

SIMULATION

Cycle-simulation

The cycle simulations are performed with a commercial cycle simulation program. Hereby not only the gas dynamics are solved but also the structural thermal dynamics.

The Start Of Combustion, SOC, for the HCCI combustion is calculated according to the knock-integral-method described in [13]. After ignition is detected the following heat release is implemented as a Vibe correlation. The reason for choosing this method is the following:

- No access to HCCI-combustion code for gasoline.
- Easy to use with few tuning parameters.
- The SOC calculation maintains some physicality outside of explored regions compared to the alternative of tabulating the heat release.

The knock-integral-method has primarily been used to predict knock in SI-engines and ignition delay in diesel engines [14]. The use for HCCI-combustion has been proposed in [15] and [16]. The basics in the method is that the ignition delay, τ, can be estimated through an Arrhenius correlation.

$$\tau = \mathbf{A} * \mathbf{p}^{-\mathbf{n}} * \mathbf{e}^{\frac{\mathbf{B}}{\mathbf{T}}} \quad [4]$$

A, n and B - coefficients.

p - pressure

T - temperature

If pressure and temperature varies during the ignition delay the time to ignition can be estimated in accordance with Livengood and Wu as

$$\int_{t_{IVC}}^{t_{IVC}+t_{SOC}} \frac{1}{\tau(s)} ds = 1 \qquad [5]$$

t_{IVC}- time at inlet valve closure

$t_{IVC}+ t_{soc}$- corresponds to the time when enough radicals are present to ignite the charge

Suggestions of values for the coefficients in the Arrhenius correlation can be found in [14] and [15]. For the simulations in this paper the values suggested in [15] has been used. The values used are reprinted below.

A=2.67E-5

n=1.35

B=8330

These values require that the pressure is stated in atmospheres and that the temperature is stated in Kelvin. The necessary adjustments for the two different valve timing strategies and the lowest inlet pressures have been conducted simply by tuning of the A coefficient which has been scaled with factors in the 1.0 to 1.25 range.

The following heat release is implemented as a Vibe correlation as suggested in [16].

$$x_b = 1 - e^{-a\left(\frac{\theta-\theta_0}{\Delta\theta}\right)^{m+1}} \qquad [6]$$

x_b - Heat release fraction

a - Vibe parameter

θ - Crank Angle

θ_0 - Crank Angle at combustion start

$\Delta\theta$ - Combustion duration

m - Shape parameter

Assumption of complete combustion when x_b=0.999 results in a=6.908.

The duration and shape parameter is tuned against experimental data. For the transient simulations the Vibe duration is mapped against SOC, calculated from the knock integral, and trapped fuel mass or engine speed. This is done to resemble the behavior that earlier combustion burns faster than later combustion, that a richer charge burns faster than a lean charge and that the combustion is extended in crank angle at higher engine speed.

Naturally the method comes with some drawbacks:

- The knock integral coefficients are not valid in a wide region. To resemble the engine tests different coefficients have to be used for the two different valve-timing strategies and the lowest inlet pressures.
- The Vibe function does not resemble the HCCI-combustion very closely.
- No physical coupling in duration and shape behavior. Simple expressions for duration and shape are today not present. This leads to that "non" physical mapping has to be used for these parameters.

In the experiments that form the basis for this paper no cool flame behavior has been detected. If a cool flame had been evident it would have been treated with an extra set of knock-integral and Vibe correlation.

<u>Control simulations</u>

The control structure is the same as in the engine tests.

The control simulation in general consists of a trigger block that captures the CA50 from the cycle simulation. The CA50 is in turn fed as the current value for the combustion timing to the combustion controller. In order to account for the averaging of the combustion timing signal in the test bed a delay is used on the feedback signal. The set point for the combustion timing is given as input for the desired operating point. The controller actuates the desired valve timing to achieve the set point in combustion timing through a valve timing logics block. As in the test bed the output from the controller is rounded to the nearest integer, or the nearest even integer in the overlap case, to achieve the discrete valve timing resolution.

<u>Combined simulation</u>

The combined simulation is performed as two independent simulations that exchange data at a defined interval. The interval is chosen to correspond to the integration step of the cycle simulation of 0.2 CAD. The combined simulation is managed from the control simulation.

Data passed from the cycle-simulation to the control simulation:

- Cylinder pressure/ Percent burnt.
- Crank Angle.

Data passed from the control simulation to the cycle-simulation:

- Valve profile number corresponding to the valve profile number in the AVT-system.
- Phasing of the profile.

RESULTS

TRANSIENT ENGINE TEST

In order not to show a lot of diagrams twice only the results from the overlap controller are depicted in the following section. For transient behavior of the IVC controller see “Comparison of cycle simulation and engine test in transient”-section.

In the following the behavior of the engine controller will be shown during a number of common engine transients. The notation “CA50_Engine_Test” is used for the averaged estimated combustion-timing signal.

A basic engine control requirement is to choose combustion timing and to move between different combustion timings in a fast and controlled manner. Figure 8 shows the response to a set point step in CA50 from 3 CAD to 7 CAD at 1000 rpm for the engine. The transition takes about 1 second that is corresponding to 8 cycles.

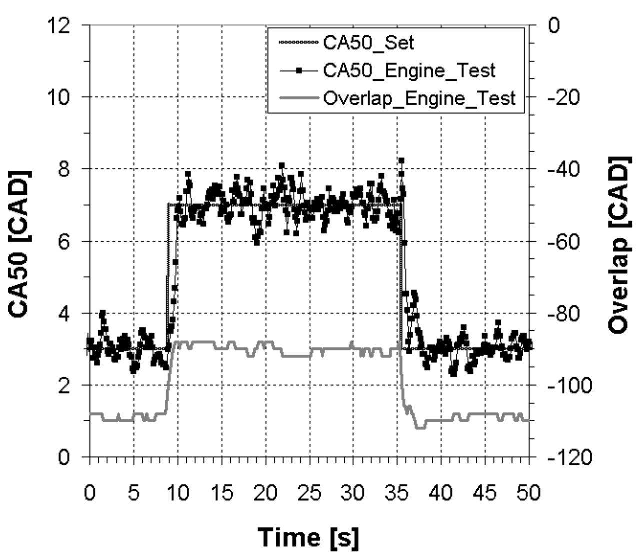

Figure 8. Combustion control overlap. Positive and negative step in demanded CA50 angle and the resulting actual CA50 angle along with the actuated overlap. The engine is here operated at 1000 rpm, the net IMEP is 4.2 bar, the inlet pressure is 1.6 bar absolute and the inlet temperature is about 85 ^{O}C.

Another basic engine control requirement is to be able to change load in a controlled manner. Figure 9 shows an example of a change in injected fuel. As can be seen the response in net IMEP is not step like indicating that a wall film phenomenon is present in the inlet manifold. The load step is from 4.2 bar to 5.6 bar net IMEP. One notices a dip in the combustion angle at the positive load step and a peek in the combustion angle at the negative load step. The dips are caused by fuel concentration variation that results in faster and slower burns. The actuated overlap is also shown.

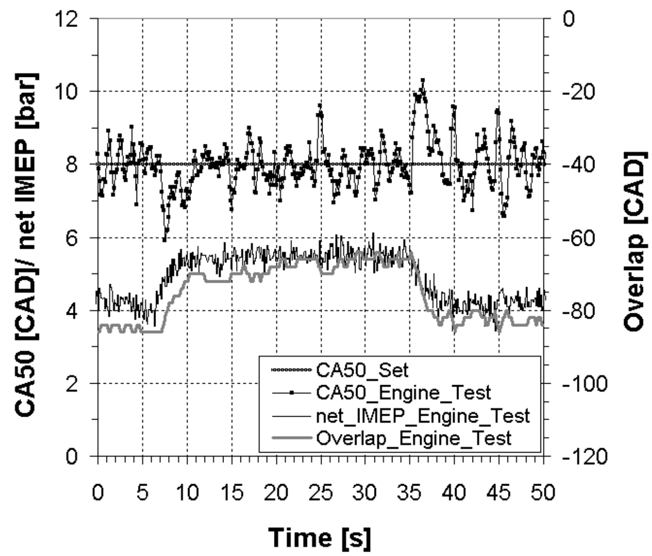

Figure 9. Combustion control overlap. Positive and a negative step in injected fuel. The engine is operated at 1000 rpm with an inlet pressure of 1.6 bar absolute and about 90 ^{O}C in inlet temperature. The fuel step is recognized through the net IMEP rise and fall. Also shown is the actuated overlap.

In figure 10 the behavior without a combustion controller during a similar load change is shown. The combustion is here advanced about 2 CAD at the positive step and then continues to advance during the high fuel time probably due to a cylinder wall temperature rise. When the fuel amount is lowered to the initial amount the combustion retards but not immediately to the original value.

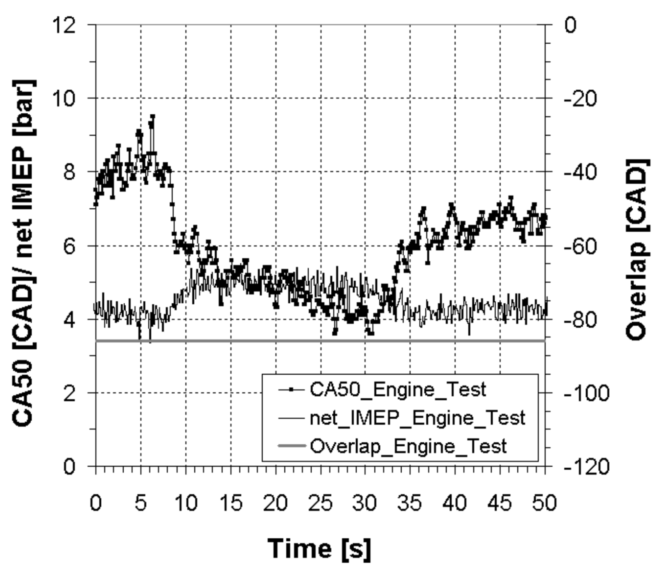

Figure 10. Response to a fuel step without combustion control.

In figure 11 the behavior during an inlet pressure change is depicted. The pressure response is slow due to the fact that the inlet comprises of large piping combined with the fact that the massflow is low because of the negative overlap. The behavior with the IVC controller during a similar inlet pressure reduction is not this good. The cause is that the pressure drop is steeper in the IVC case, which the controller is to slow to cope with. The reason for the steeper pressure drop is that the massflow is higher in the IVC case compared to the overlap case.

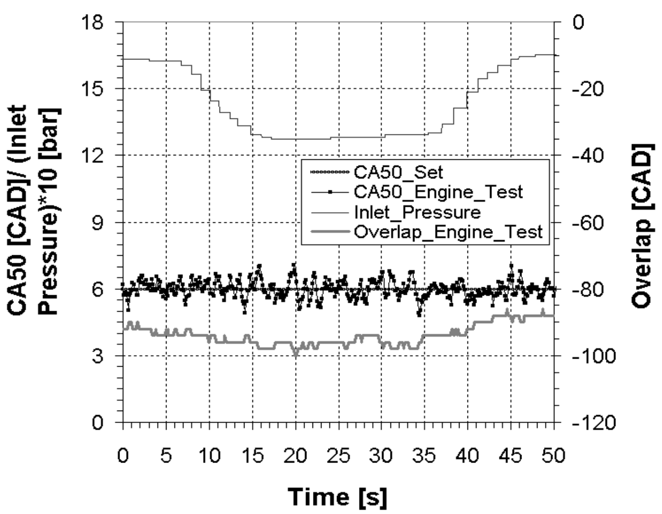

Figure 11. Combustion control overlap. Behavior when the inlet pressure is lowered from 1.6 bar absolute to 1.3 bar absolute. The engine is operated at 1000 rpm at an net IMEP of about 4.4 bar.

Figure 12 describes the behavior when the inlet pressure is lowered without a combustion controller. The combustion is here retarded about 2.5 CAD during the low-pressure time.

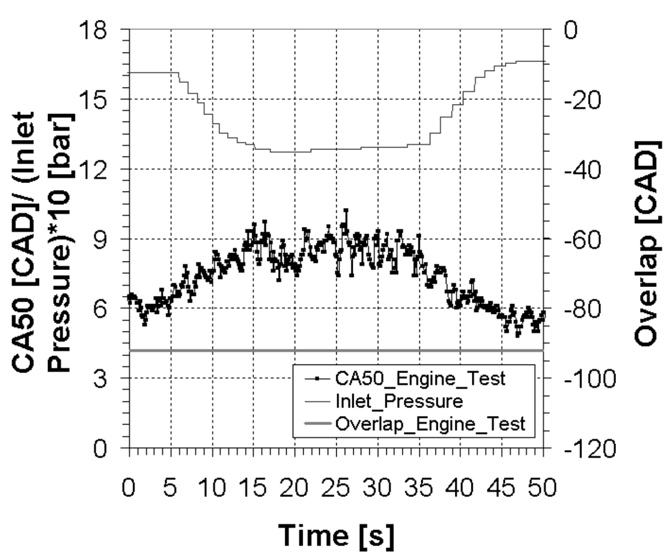

Figure 12. Behavior during an inlet pressure reduction without combustion controller.

The last common engine transient depicted here is an engine speed change. Figure 13 shows the behavior. The engine speed rises from 1000 rpm to 1500 rpm during approximately 8 seconds and falls after roughly 35 seconds from 1500 rpm back to 1000 rpm on 4 seconds. The reason for the different speed ramp lengths is that the engine does not manage to accelerate faster during the positive ramp with the present fuelling. During the negative ramp the dynamometer forces the engine to follow the prescribed ramp. As seen, two quite large drops are present in the combustion angle during the accelerating speed ramp. The behavior is even more evident at the faster negative ramp. A probable cause to the behavior is presented in the "Simulated and tested transients in comparison" section. As can be seen the overlap is continuously increasing during the high-speed portion.

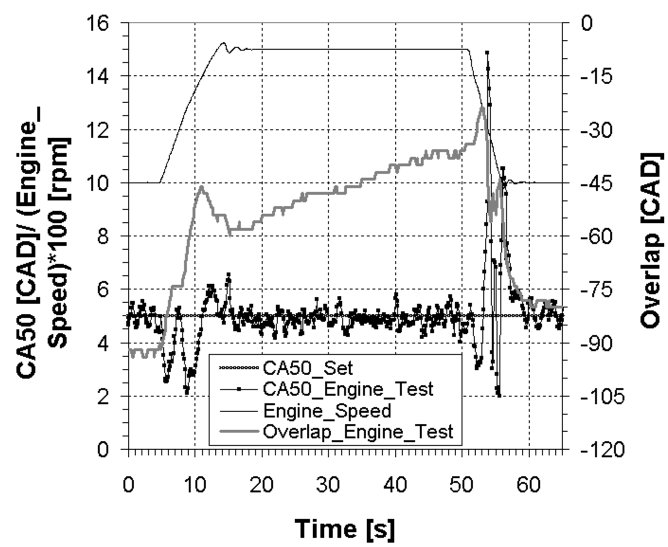

Figure 13. Combustion control overlap. Control behavior during a speed ramp. The positive ramp is from 1000 rpm to 1500 rpm during 8 seconds while the negative speed ramp are from 1500 to 1000 rpm on 4 seconds. The inlet pressure set point is maintained at 1.6 bar absolute and the inlet temperature is about 75 ^{O}C. The net IMEP is about 4.5 bar.

COMPARISON OF CYCLE SIMULATION AND ENGINE TEST AT STEADY STATE POINTS

In order to validate the cycle simulation experimental data from steady state operation, with fixed valve timings, were compared to simulated data from the same conditions. The points of operation used are chosen to represent start or end state of a transient.

As can be seen in the pressure traces, figure 14 and 15, the maximum pressures are overestimated in simulation. The main cause is that the Vibe heat release does not resemble the HCCI combustion closely. The behavior also affects the load prediction, as shown in figure 16.

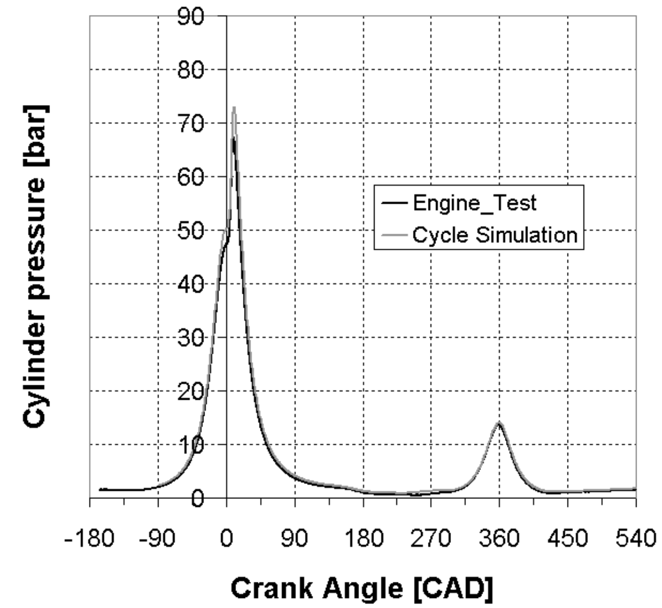

Figure 14. The figure shows the measured and calculated cylinder pressure for a load point at 1000 rpm with 1.6 bar of absolute inlet pressure, -112 deg of overlap, CA50 of 3.5 CAD and an net IMEP of 4.2 bar.

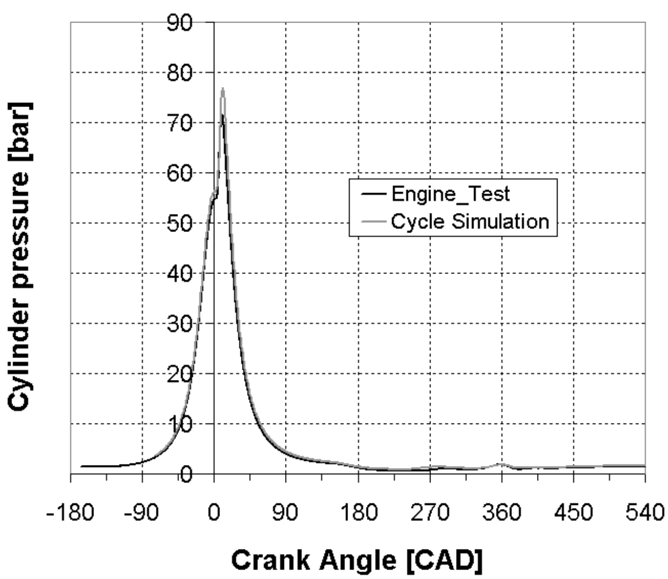

Figure 15. The figure shows the measured and calculated cylinder pressure for a load point at 1000 rpm with 1.6 bar of absolute inlet pressure, inlet valve closure at 618 CAD, CA50 of 7.6 CAD and an net IMEP of 4.6 bar.

The CA50 angle is quite accurately predicted. The results in figure 17 are achieved using different knock-integral parameters, different A in equation 4, for the overlap and IVC cases and for the lowest inlet pressure points.

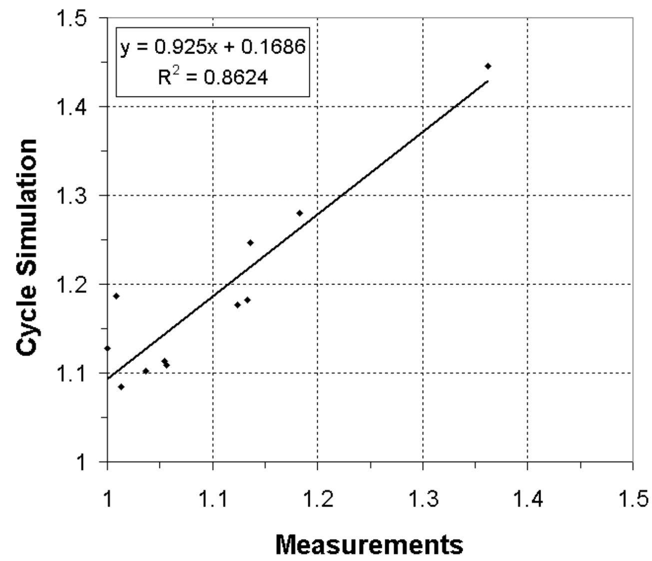

Figure 16. Normalized engine load for all calculated points, for both overlap and IVC control, and the corresponding engine test result.

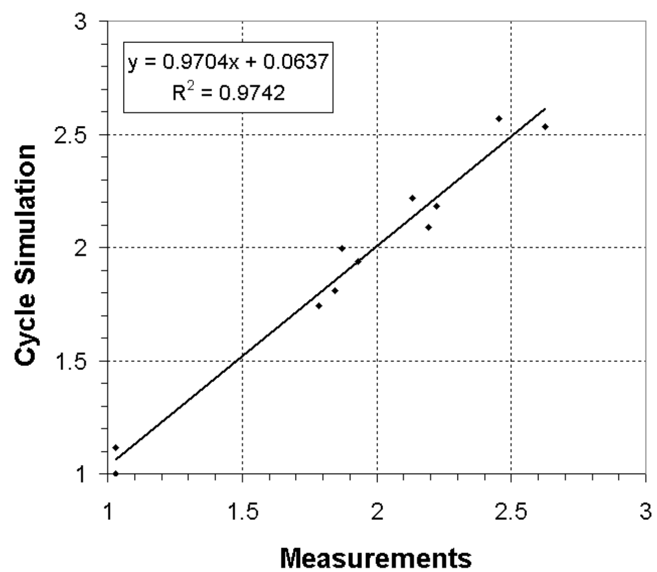

Figure 17. Normalized combustion timing for all calculated points, for both overlap and IVC control, and the corresponding engine test result.

COMPARISON OF CYCLE SIMULATION AND ENGINE TEST FOR TRANSIENTS

Controller behavior

Due to the lack of cycle-to-cycle variations in the simulation it is from stability point of view possible to use a faster controller in the simulations compared to the engine tests. As can be seen in table 3 it is not possible to use the simulation for precise controller tuning. If the simulation is supposed to be used for controller tuning cycle-to-cycle characteristics has to be included. The following maximal control parameters have been found through empirical tests.

Table 3. Control parameter overview.

Controller	Overlap	IVC
Simulation K_P **[CAD_Valve-timing/CAD_CA50]**	4	1
Test Bed K_P **[CAD_Valve-timing/CAD_CA50]**	2	1
Simulation K_I **[CAD_Valve-timing/ s*CAD_CA50]**	40	10
Test Bed K_I **[CAD_Valve-timing/ s*CAD_CA50]**	10	5

Simulated and tested transients in comparison

The following is obtained if the test bed parameters in table 3 are used in engine tests as well as in simulations.

In this section only the IVC controller is exemplified. The notation "CA50_Engine_Test" is used for the averaged estimated combustion-timing signal.

Figure 18 shows the response to a set point step in CA50 from 3 CAD to 7 CAD at 1000 rpm for the engine and the cycle simulation. In figure 19 the CA50 set point and the actuated IVC in engine test and cycle simulation are shown. As seen the simulation is quite closely reflecting the engine test in CA50 as well as in actuated IVC.

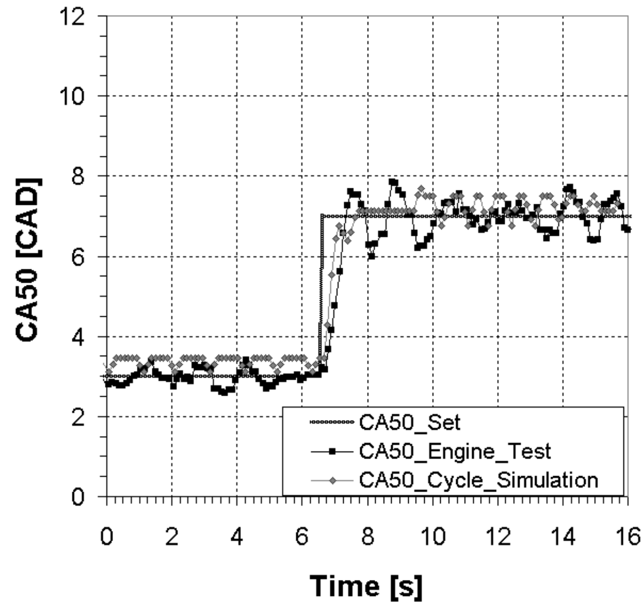

Figure 18. Combustion control IVC. Positive step in demanded combustion angle and the resulting actual combustion angle along with the actuated IVC. The engine is here operated at 1000 rpm, the net IMEP is 4.6 bar, the inlet pressure is 1.6 bar absolute and the inlet temperature is about 85 °C.

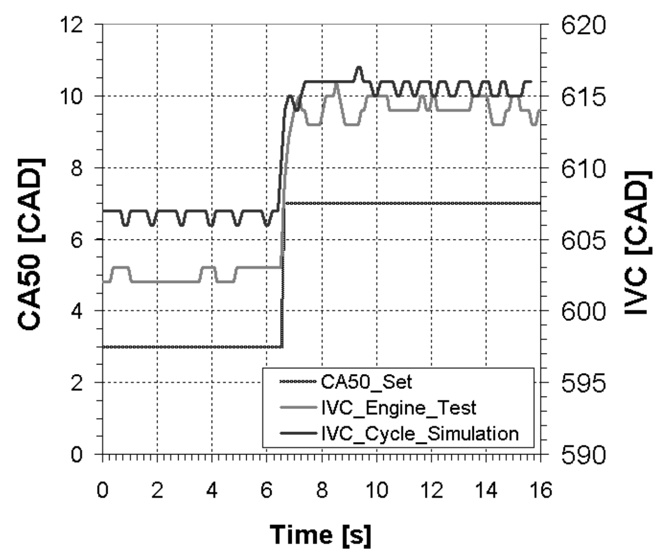

Figure 19. Actuated IVC for engine test and cycle simulation during the step in CA50 set value depicted in figure 18. Also shown is the CA50 set value.

Figures 20 and 21 describe the behavior during a load change. As seen there is a lag between demanded fuel increase and actual net IMEP rise in the IVC case. The lag is also present with the overlap controller as seen in figure 9. The cause is most probably a wall film phenomenon. Also shown in figure 20 is the predicted combustion angle and in figure 21 the actuated IVC in the cycle simulation along with the simulated net IMEP rise. The simulation suffers from the fact that it does not consider the wall film.

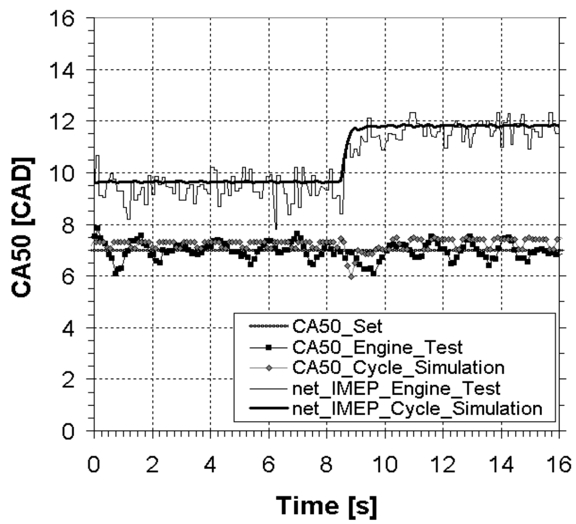

Figure 20. Combustion control IVC. Positive step in injected fuel, seen through an net IMEP rise, and the resulting actual combustion angle along with the actuated IVC. The engine is here operating at 1000 rpm, with an inlet pressure of 1.6 bar absolute and about 85 °C in inlet temperature.

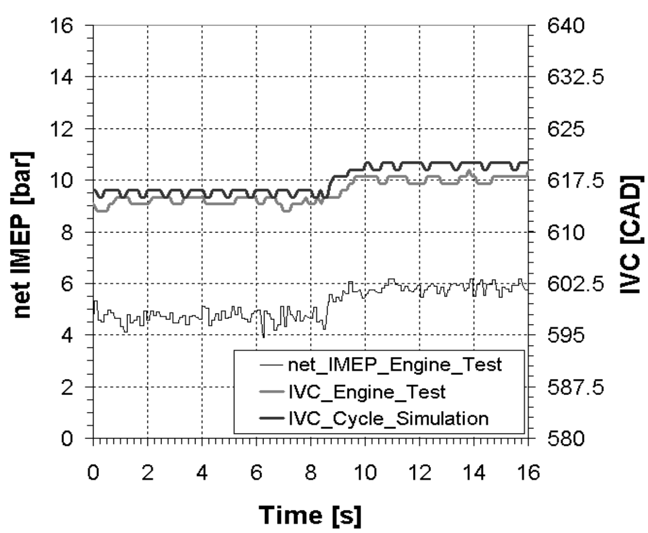

Figure 21. Actuated IVC for engine test and cycle simulation during the load step depicted in figure 20. Also shown is the measured net IMEP.

Figure 22 and figure 23 depicts the behavior during a speed ramp. As seen in figure 22 two quite large drops are present in the combustion angle. The drops are, as can be seen, also present in the cycle simulation. The simulation although does not predict the behavior between the drops correctly. The similar behavior was also shown for the overlap case in figure 13. The cause of the two drops is that the speed rise causes a rise in volumetric efficiency that in turn forces the combustion earlier. The behavior is in the simulation to two-thirds depending on the SOC prediction and to one-third dependant on the mapping of the Vibe combustion duration against predicted SOC. To get good prediction of the later drop one has to consider that the inlet pressure drops about 0.2 bar during the ramp and that the speed ramp is not absolutely straight. Figure 23 shows the actuated IVC for engine test and simulation.

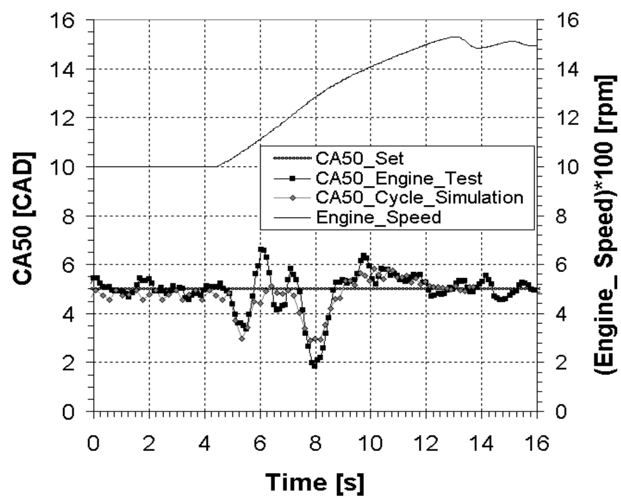

Figure 22. Behavior of the IVC controller during a speed ramp. The ramp is from 1000 rpm to 1500 rpm during 8 seconds. The inlet pressure set point is maintained at 1.6 bar absolute and the inlet temperature is about 90 °C. Also shown is the predicted combustion angel. The net IMEP is about 4.5 bar.

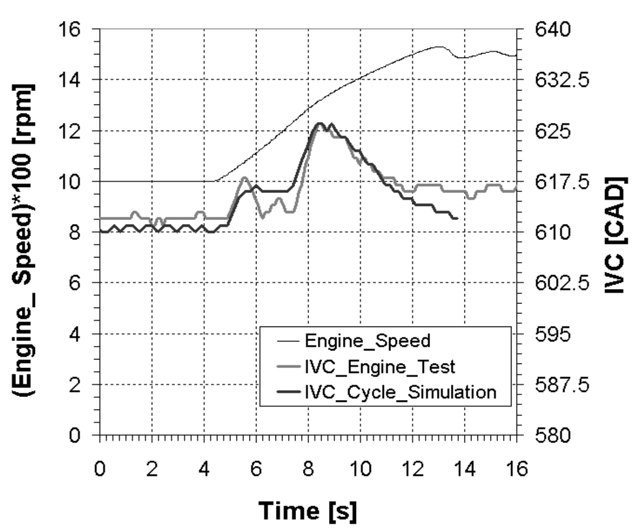

Figure 23. Actuated IVC for engine test and cycle simulation during the speed ramp depicted in figure 22. Also shown is the engine speed.

FUTURE WORK

Future work can be divided in a couple of branches.

One branch is the engine branch. Today the area of operation is hampered by the fact that the two valve strategies cannot be used in one controller. Therefore a first step is to combine the two strategies.

Another branch is the controller branch. The controller performance is today too slow to be of any real transient use. The first question that needs to be addressed is how to handle the combustion timing signal in a more refined way. Hereby the goal is to minimize the cycle-to-cycle variations in the signal while being able to predict the true combustion timing without time delay. A continuation of this branch is to change or enlarge the control structure.

The third branch is the simulation branch. A requirement is to increase the ability to tune the control parameters through simulation. In order to do this cycle-to-cycle characteristic has to be included in the simulation.

CONCLUSION

The conclusions from the presented work can be summarized as follows.

The two valve strategies, varying negative overlap and varying IVC, works to control the combustion timing.

A faster controller is required in order to be able to provide rapid transient engine performance.

The method to simulate transient HCCI behavior by aid of the knock-integral and mapped Vibe heat release coefficients works with high accuracy in narrow regions.

Combined cycle simulations and control simulations have been conducted during a number of common engine transients. The outcome indicates that in order to tune control parameters through simulation cycle-to-cycle characteristics have to be included in the simulation.

ACKNOWLEDGMENTS

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REFERENCES

1. M. Sjöberg, L.O. Edling, T. Eliassen, L. Magnusson, H.E. Ångström. ”GDI HCCI: Effects of Injection Timing and Air Swirl on Fuel Stratification, Combustion and Emission Formation”. SAE 2002-01-0106.
2. A. Hultqvist, “Characterization of the Homogeneous Charge Compression Ignition Process”. Doctoral Thesis, Lund Institute of Technology.
3. J. Martinez-Frias, S.M. Aceves, D. Flowers, J.R. Smith, R. Dibble, “HCCI Engine Control by Thermal Management”, SAE 2000-01-2869.
4. J.O. Olsson, P. Tunestål, B. Johansson. ”Closed-Loop Control of an HCCI Engine”. SAE 2001-01-1031.
5. J.O. Olsson, P. Tunestål, G. Haraldsson, B. Johansson. ”A Turbo Charged Dual Fuel HCCI Engine”. SAE 2001-01-1896.
6. N.B. Kaahaaina, A.J. Simon, P.A. Caton, C.F. Edwards. “Use of Dynamic Valving to Achieve Residual-Affected Combustion”. SAE 2001-01-0549.
7. D. Law, D. Kemp, J. Allen, G. Kirkpatrick, T. Copland. “Controlled Combustion in an IC-Engine with a fully Variable Valve Train”. SAE 2000-01-0251.
8. L. Koopmans, I. Denbratt. ”A Four Stroke Camless Engine, Operated in Homogeneous Charge Compression Ignition Mode with Commercial Gasoline”. SAE 2001-01-3610.
9. R. Muller, M. Hart, G. Krötz, M. Eickhoff, A. Truscott, A. Noble, C. Cavalloni, M. Gnielka, “Combustion Pressure Based Engine Management System”. SAE 2000-01-0928.
10. F. A. Matekunas, P. A. Battitston, C.F. Chang, D.R. Lancaster. ”Cylinder-Pressure-Based Engine Control Using Pressure-Ratio-Management and Low-Cost Non-Intrusive Cylinder Pressure Sensors”. SAE 2000-01-0932
11. M. Mladek, C.H. Onder, “A Model for the Estimation of Inducted Air Mass and the Residual Gas Fraction using Cylinder Pressure Measurements”. SAE 2000-01-0958.
12. G.M. Rassweiler, L. Withrow, “Motion Pictures of Engine Flames Correlated with Pressure Cards”. SAE Journal (Trans), Vol. 42, pg. 185-204, May 1938. (Landmark Reprint paper, SAE 800131).
13. J.C. Livengood, P.C. Wu. “Correlation of Autoignition Phenomenon in Internal Combustion Engines and Rapid Compression Machines”. Fifth Symposium (International) on Combustion, pp. 347-356, 1955.
14. J.B. Heywood. ”Internal Combustion Engine Fundamentals”, McGraw-Hill, Inc., ISBN 0-07-100499-8
15. Y. Ohyama. “Engine Control Using Combustion Model”. SAE 2000-01-0198.
16. M. Larsson “Homogeneous Charge Compression Ignition Conditions for Diesel Fuel”. Master Thesis, Machine Design KTH Internal Combustion Engines.

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DEFINITIONS, ACRONYMS, ABBREVIATIONS

BMEP: Break Mean Effective Pressure

CAD: Crank Angle Degree

CA50: Crank Angle Degree for 50 percent burnt

EVC: Exhaust Valve Closure

EVO: Exhaust Valve Opening

HCCI: Homogeneous Charge Compression Ignition

IVC: Inlet Valve Closure

IVO: Inlet Valve Opening

net IMEP: Net Indicated Mean Effective Pressure

SOC: Start Of Combustion