HCCI Heat Release Data for Combustion Simulation, based on Results from a Turbocharged Multi Cylinder Engine

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ABSTRACT

When simulating homogenous charge compression ignition or HCCI using one-dimensional models it is important to have the right combustion parameters. When operating in HCCI the heat release parameters will have a high influence on the simulation result due to the rapid combustion rate, especially if the engine is turbocharged. In this paper an extensive testing data base is used for showing the combustion data from a turbocharged engine operating in HCCI mode. The experimental data cover a wide range, which span from 1000 rpm to 3000 rpm and engine loads between 100 kPa up to over 600 kPa indicated mean effective pressure in this engine speed range. The combustion data presented are: used combustion timing, combustion duration and heat release rate. The combustion timing follows the load and a trend line is presented that is used for engine simulation. The combustion duration in time is fairly constant at different load and engine speeds for the chosen combustion timings here. The heat release rate is fitted to a Wiebe function where the heat release parameter *m* is found. It is shown that this parameter *m* scale to the load and the presented trend line is used for simulating the heat release. When the engine is operated with negative valve overlap the mass flow is reduced through the engine. In an engine simulation the valve timings has to be estimated for different intake temperatures and boost pressure levels. By using the intake temperature at intake valve closing as a prediction tool for the temperature at top dead center, the exhaust valve closing timing can be estimated and will then follow the real test results closely as shown in a GT-Power simulation. The turbocharged test engine is an in-line four cylinder gasoline engine with a total displacement of 2.2 *l*. The engine is direct injected of spray-guided type. To achieve HCCI combustion the engine is operated with low lift and short duration valve timings where the variable negative valve overlap is used for combustion control.

INTRODUCTION

The homogenous charge compression ignition (HCCI) combustion can offer high thermodynamic efficiency, low throttling losses and low emissions. The limited operating range in HCCI mode makes it an alternative to the spark ignited (SI) engine at low load to improve the efficiency of the SI engine. In HCCI mode the air-fuel charge is auto ignited by controlling the temperature and pressure history inside the engine cylinder to initiate and time the combustion phasing at the right moment.

To reach auto ignition in a four-stroke engine the in-cylinder temperature is normally increased by exhaust gas recirculation (EGR), air heating or both as outlined in the eighties by Thring [1]. The compression ratio can also be used to control the HCCI combustion timing [2,3] The concept with negative valve overlap (NVO) uses early exhaust valve closing to trap a certain amount of hot residuals is widely accepted to control combustion timing. Here the variable valve timing (VVT) is accurately controlled either mechanically or electrically.

The concept of re-breathing [4] might be more mechanically complicated but has an advantage in reduced heat losses since it can be operated without recompression. The combustion timing in HCCI can also be adjusted with the intake temperature by a thermal management system [5,6], but this requires a heat recovery system.

By boosting the engine in HCCI mode the possible load range can be extended [7,8,9,10,11]. The boost pressure can be controlled directly by the intake valve timing [11] which gives large freedom to operate the engine with optimum efficiency, emissions, combustion noise, cyclic variations and control strategies. With NVO the intake temperature directly affects how much internal EGR is needed for a given combustion timing and therefore the in-cylinder relative burn gas fraction. By increasing the intake temperature, less internal EGR is needed which lead to increased mass flow that can lead to higher boost pressure [12]. In addition to the internal EGR, external cooled EGR can be used and it was found that it could improve combustion efficiency while there seem to be low influence on the combustion duration [13] in HCCI mode.

The main operating parameter in HCCI mode is the combustion timing here represented by the crank angle of 50 % heat released (CA50). The CA50 timing will affect the engine performance substantially and need to be controlled exactly to run the HCCI engine successfully. If engine speed goes up or if load is increased the CA50 window for stable engine operating will be narrowed [12], making the engine control more challenging.

When simulating HCCI it is vital to have the right combustion parameters, especially if the engine is turbocharged. The combustion timing and heat release rate has to be predicted together appropriate valve timings. If the engine is operated with NVO and short duration valve timings there can be high throttling losses over the valves [12] and this need to be predicted when simulating the engine performance. Since the boost pressure is used to decrease the combustion noise it is vital that the turbocharger performance is predicted accurately. The scope of this paper is to present data from extensive engine testing and show how it can be used in an engine simulating program.

EXPERIMENTAL SET-UP

ENGINE SYSTEM

The test engine is an in-line four cylinder gasoline engine with a total displacement of 2.2 *l*. The cylinder head is a 4-valve design with a pent-roof combustion chamber. There are some small squish areas on the intake and exhaust sides. The piston has a raised piston dome with a small bowl in the center. The intake channel is side-drafted with a low tumble design. The engine has direct injection (DI) of a Spray-Guided type slightly with a centrally placed injector. The eight-hole solenoid fuel injector has a cone angle of 60°. The spark plug is located close to the fuel injector and has an extended tip. The spark is always operated as a safety measurement against misfires and it can also improve the combustion stability at late combustion timings.

To achieve HCCI combustion the engine is operated with NVO with low lift and short duration camshafts as seen in Figure 1 where the minimum and maximum valve timings are plotted. The variable valve timing (VVT) is controlled by hydraulic actuators at the camshafts giving a separate 50 crank angle degree (CAD) adjustment on both intake and exhaust valve timings. The engine is turbocharged by a fixed geometry turbine. The exhaust manifold is of pulse type with short individual runners straight to the turbine inlet as seen in Figure 2. On the intake side there is a water cooled intercooler. The heated aluminum intake manifold has short intake runners and a small volume as seen in Figure 3. The cooling system has an electric driven water pump and the coolant temperature is adjusted by the control system. Engine specifications are listed in Table 1.

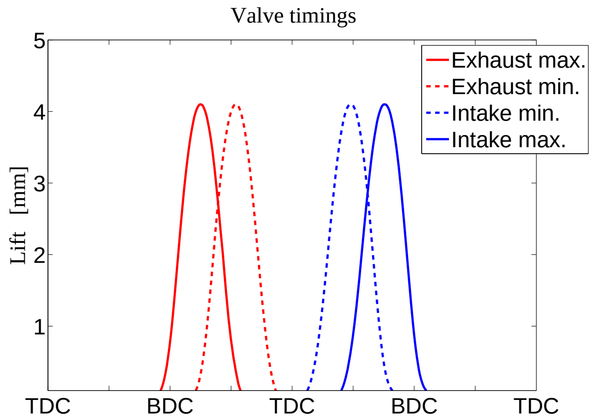

Figure 1: Valve timing range

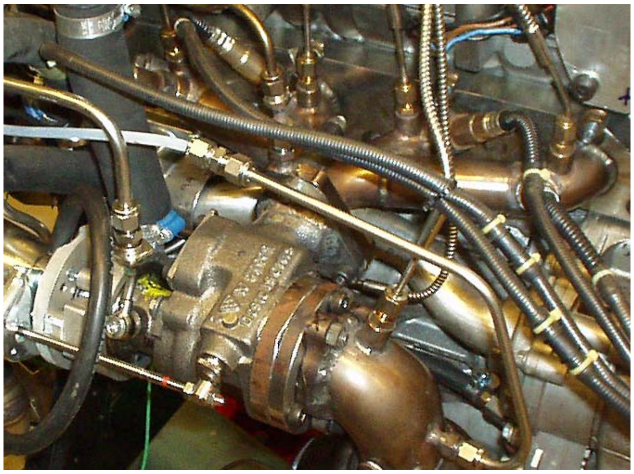

Figure 2: Exhaust manifold

Figure 3: Intake manifold

MEASUREMENT AND CONTROL SYSTEM

The control system is a combined data acquisition and engine control unit from dSPACE. The signals from a fully instrumented engine set-up are collected and analyzed. For combustion feedback there are individual cylinder pressure sensors. The control system has an in-cycle resolved heat release calculation where main parameters like CA50, peak cylinder pressure (PCP) and peak pressure derivative (dP/CAD) etc. are used for cylinder individual control in closed loop. The crank angle decoder has a resolution of 0.2 CAD. Emission analysis and soot measurement are collected on an external PC and the data is sent to the control unit.

To increase the operating range it is important that all the cylinders operate identically, meaning that they have the same CA50 position, indicated mean effective pressure (IMEP) and peak pressure rise rate. This means that the control system has to perform cylinder balancing. Injection of the fuel in NVO can give a reformation of the fuel, resulting in an increase of charge temperature and a reduction in effective octane number leading to more advanced combustion timing. The combustion timing will be more advanced if the fuel is injected in the beginning of the NVO than in the end. The control system sets the CA50 position and uses the cylinder with the most advanced CA50 position for the exhaust valve closing (EVC) set point. The other cylinders are then advanced by adjusting the injection timing and/or fuel amount in the first injection, until all cylinders have the same CA50 position. Since there will be some differences in injection timing when using CA50 balancing there will also be some differences in efficiency between the cylinders. This results in different load, IMEP in the cylinders. To maximize the load range there has to be cylinder balancing of IMEP as well. This is done at the main injection event. Obviously this affect the CA50 balancing since increasing the fuel amount will raise the combustion heat, leading to earlier CA50 for that cylinder. Therefore the cylinder balancing has to be constantly monitored and adjusted by the actuator set points. Table 2 shows the measurement and control equipment.

Table 1: Engine specifications

Number of cylinders	4
Displacement	2198 cm^3
Bore x Stroke	86 mm x 94.6 mm
Compression Ratio	11.75:1
Valve duration IN	125 CAD
Valve duration EX	155 CAD
NVO range	90 -190 CAD
Turbocharger	B&W KP31
Fuel Supply	DI, up to 20 MPa pressure
Fuel Type	Gasoline, 95 RON

Table 2: Measurement and control system

ECU and data logging	dSPACE Rapid Pro
Cylinder pressure sensor	Kistler 6043Asp
Charge amplifier	Kistler 5011B
Emission analyzer	Horiba 9100 MEXA
Soot analyzer	AVL 415 S

TEMPERATURE AND EGR CONTROL SYSTEM

The intake temperature is controlled by an electrical throttle in the bypass routing to the water-cooled intercooler circuit, see Figure 4. When the throttle is fully open the main air flow goes through the bypass routing since it has less flow resistance than the intercooler circuit. By closing the throttle the airflow is forced through the intercooler circuit.

The long route EGR system takes exhaust gas after the turbine and route it to the compressor inlet. To get a positive displacement of exhaust gas there is a throttle in the exhaust system to increase back pressure. The compressor inlet can be used as an ejector to enhance the EGR flow, so the needed back pressure is quite small. The external EGR routing is cooled by the engine cooling circuit and is controlled by an EGR valve near the turbocharger, see Figure 4. The external EGR is used when the engine is operated stoichiometric or when the pumping losses need to be reduced. The pumping losses can be reduced with these types of short duration valve timings by adding cooled EGR to get more beneficial valve timings. If the turbocharger builds more back pressure than boost pressure due to a choked turbine, addition of external EGR will reduce the mass flow. The reason to this is that more internal EGR is needed for a given combustion timing when the heat capacity of the intake charge is increased.

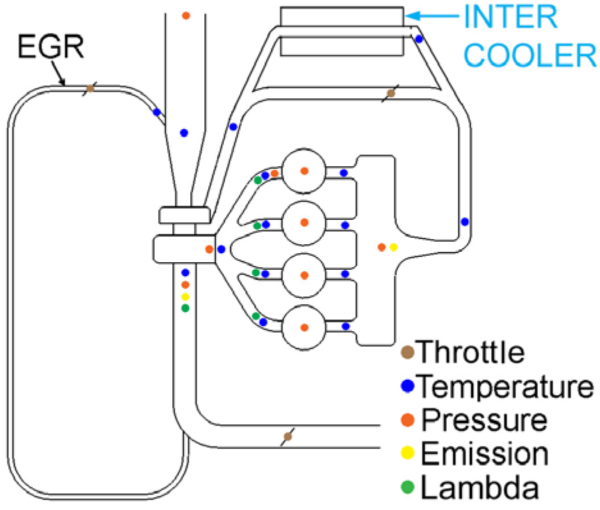

Figure 4: Engine layout

RESULTS AND DISCUSSION

OPERATING LIMITATIONS

When operating in HCCI mode one has to consider operating limitations due to combustion noise, combustion stability, soot level, emission level and peak cylinder pressure. These are outlined in author's previous paper [11] but here is a short background.

Combustion Noise: The maximum load in HCCI is often limited by the fast combustion rate where the combustion induced noise is a major factor. By increasing the air dilution in the cylinder using for example a turbocharger, the audible noise is suppressed and the HCCI load range can be extended. Normally the pressure rise rate is measured in bar/CAD or MPa/ms [14] and is used to impose some operating limits but neither of these catch the real combustion noise when the engine is supercharged. The Ringing Intensity (RI)- MW/m^2 [15] on the other hand compares the pressure rise rate to peak cylinder pressure and engine speed and therefore gives an good indication of the combustion noise. The RI level should be less than 6 MW/m^2.

Coefficient of variation (CoV): HCCI is known for its low CoV of IMEP compared to SI engines but with a late combustion timing the CoV of IMEP increases rapid in HCCI. At higher engine speeds where the time window for proper combustion phasing is reduced, it is more difficult to retard the combustion. We have found that there is less risk for misfire when the CoV is kept under 3.5 %. Misfire at high HCCI load results in violent combustion the following cycles and must be avoided at all times. At low load ie IMEP around 100 kPa the standard deviation (STD) of IMEP is used as limitation instead of CoV and the limit is set to 15 kPa.

Soot level: To get a filter smoke number (FSN) limit on soot we look at the Euro 5 and 6 legislation which limits the soot level to 0.005 g/km, if we assume a vehicle with a fuel consumption of 0.06 l/km or 44 g/km and operating at λ=1.5 the Smoke opacity arrives roughly at 0.25 [16] which corresponds to a FSN value of 0.05 [17].

Nitrogen oxides (NOx) level: HCCI is known for the low NOx emission levels but as the load increases, NOx emissions increase due to higher temperature and relative less dilution, meaning there will be a NOx limit when operating lean HCCI (and using a normal TWC). The Euro 5 and 6 limits the NOx emissions at 0.06 g/km, if we again assume a fuel consumption of 44 g/km, the corresponding NOx level will be 1.36 g/kg_{fuel}. To be on the safe side our NOx limit is set 20 % below this value at 1.1 g/kg_{fuel}. If this limit is reached the HCCI engine is operated stoichiometric with external EGR.

Peak cylinder pressure (PCP): Since PCP scales with the intake pressure it can become a limiting factor when the engine is turbocharged. There is risk for structural damage of the engine if the design limit is exceeded and for this engine the maximum limit is set to 9.5 MPa.

The results from the above described limitations can be found in the Appendix where also the intake pressure and peak pressure rise is included, since the RI is a function of these.

RESULTS FROM SPEED AND LOAD TEST

The engine is tested from lowest to highest possible load between 1000 to 3000 rpm. The load is represented here with $IMEP_{gross}$ since this is the input for the engine simulations. The test is done in load steps of 50 kPa and engine speed steps of 500 rpm. It is done with the limitations as seen in Table 3. The goal is to have highest possible efficiency in the whole operating range. When the gas exchange efficiency is included for this engine the maximum $IMEP_{net}$ stays above 600 kPa and the resulting BMEP is above 500 kPa between 1000 to 3000 rpm.

Table 3: Engine operating limitations

Combustion noise	RI	6 MW/m^2
Combustion stability	CoV of IMEP or STD of IMEP	3.5% or 15 kPa
Soot emission	FSN	0.05 average
NOx emission	EI NOx	1.1 g/kg_{fuel}
Peak cylinder pressure	PCP	9.5 Mpa

COMBUSTION TIMING

The combustion timing- CA50 is the main operating parameter when running in HCCI mode, a later CA50 can raise the upper load limit by decreasing the RI and PCP. This has to be weighted against an efficiency loss when moving away from the timing for maximum brake torque (MBT). A later CA50 will increase the energy to the turbine with higher boost as a result. This is the result from the increased mass flow since less EVC is needed for a later CA50 timing, therfore the turbocharger sizing plays an important role. The boosted HCCI engine can on the other hand for a given RI at high load use a better phased combustion and hence improve thermal efficiency compared to a naturally aspirated (NA) HCCI engine. In Figure 10 the used CA50 timing can be seen. At low load the CA50 timing is advanced before TDC to stabilize the combustion and as load gets higher the CA50 timing is retarded. The ignition timing is adjusted when it has some influence on combustion stability, both at low and high load as seen in Figure 6. The combustion timing is chosen for maximum efficiency and turbocharger performance. In Figure 11 the CA50 is plotted against the load and a trend line is fitted. The trend line equation for CA50 can then be used as a result dependent function in the simulation program. At high load and engine speed the CA50 timing could not be retarded more due choking of the rather small turbine in this set-up. A later CA50 would mean that the EVC timing has to be retarded, this will increase the mass flow which here leads to an increase in pumping losses (backpressure goes up with no gain in boost pressure).

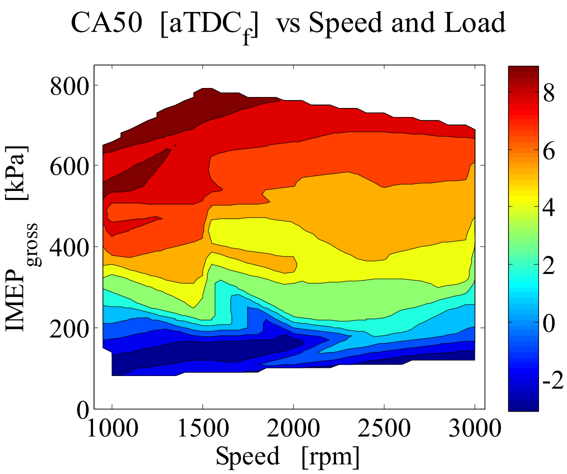

Figure 5: Combustion timing, CA50

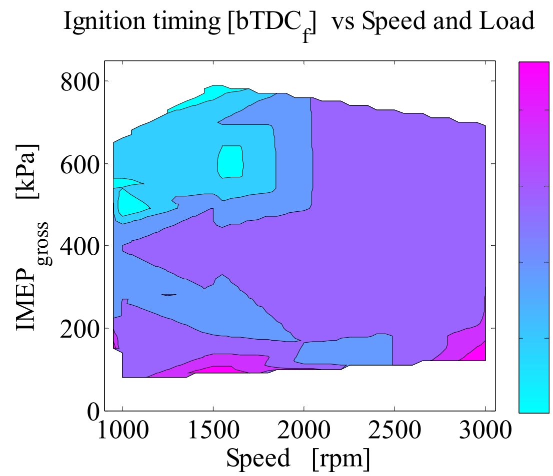

Figure 6: Ignition timing

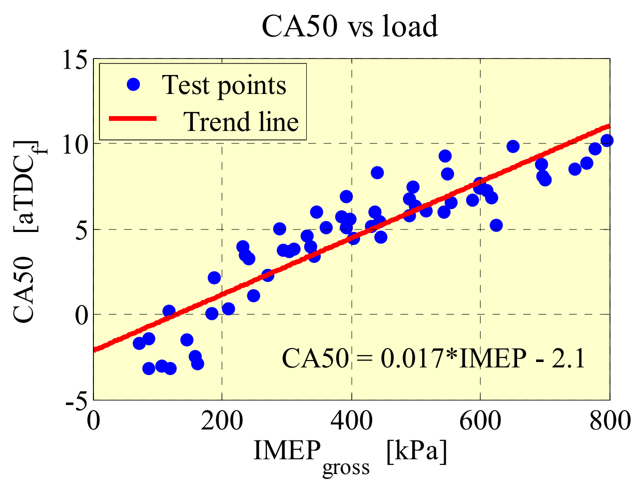

Figure 7: Combustion timing vs. load

COMBUSTION DURATION

The combustion duration between CA10 to CA90 is seen in Figure 8 and it scales to the engine speed. As engine speed is increased the combustion duration is increased in crank angle degrees but in time it stays fairly constant as seen in Figure 9. At low engine speed the combustion time is increased which reflects the stronger influence from spark assisting that is used here. The average time for the combustion duration (CA10 to CA90) in most of the operating range is around 0.7 ms and this can then be used in the simulation program.

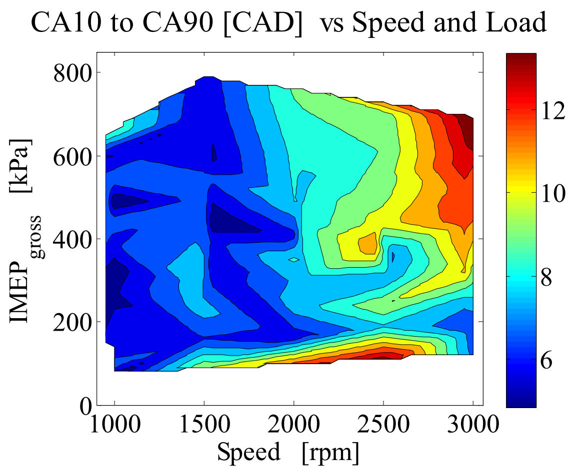

Figure 8: Combustion duration in crank angle degrees

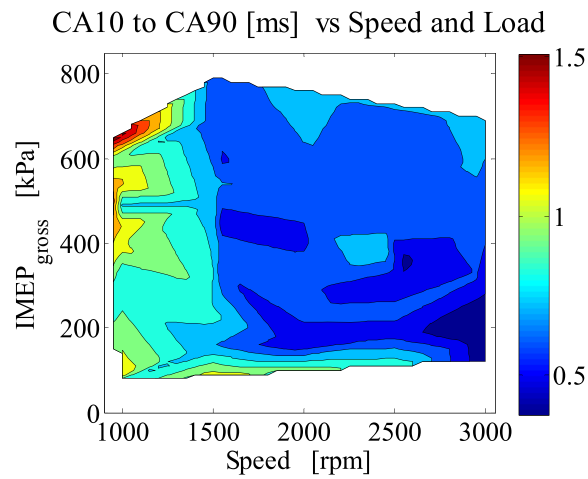

Figure 9: Combustion duration in time

THE WIEBE FUNCTION, FOR HEAT RELEASE SIMULATION

When simulating with single zone combustion models the heat release rate is fitted over the combustion duration and can give a good approximating of the heat release rate. The most common function is the Wiebe function that can be used for the heat release rate in HCCI mode. As seen in Equation 1 there is two unknown parameters, a and m.

$$x_b = 1 - \exp\left[-a\left(\frac{\theta-\theta_{SOC}}{\Delta\theta}\right)^{m+1}\right] \qquad (1)$$

The parameter a estimates the completeness of the combustion process. Typically a is set to 5 CAD [18] which corresponds to 99.3 % of burned mass in the function. The parameter m depends on the heat release rate, meaning that a small value of m will have a high initial heat release rate. A common value of m is 2 [18] which give an S-shaped heat release. From this engine testing results, the real heat release is fitted with a Wiebe function where the a parameter is set to 5 and the m parameter is found by a non linear least square function between 0 and 10 for best fit between the heat release and the Wiebe function. This is done to the heat release from 0.5 % to 99.3 % burned. An example of a real heat release and the corresponding m constant in the Wiebe function can be seen in Figure 10. This optimized single Wiebe function does follow the real heat release closely. The result of the m parameter for the whole operating range can be seen in Figure 11. The m constant is scaling to the load and in Figure 12 the trend line for m is plotted against the load. The trend line equation for m can then be used as a result dependent function in the simulation program.

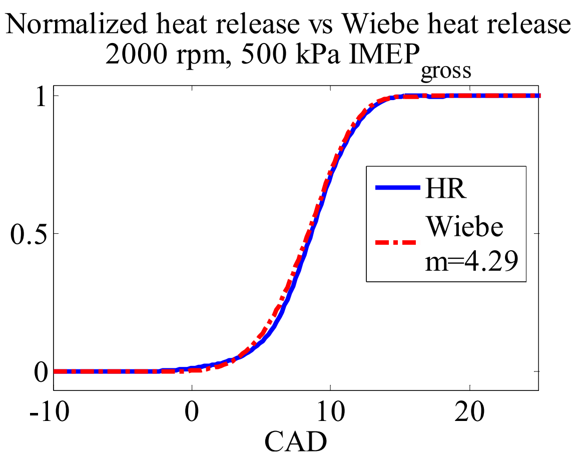

Figure 10: Heat release comparison

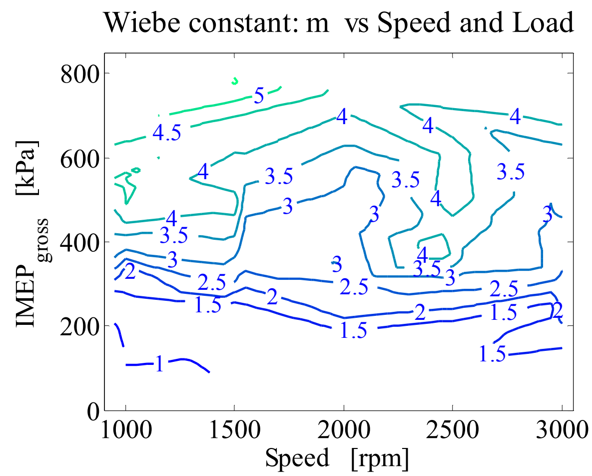

Figure 11: Wiebe constant *m* vs speed and load

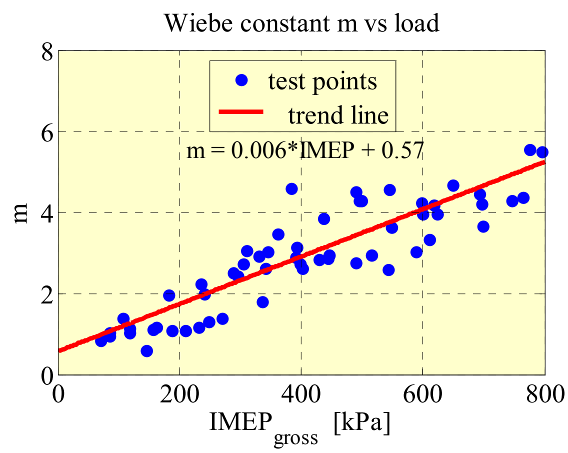

Figure 12: Wiebe constant *m* vs load

ESTIMATING THE EVC POSITION

To set the right condition for auto ignition with NVO some exhaust gases are kept in the cylinder by appropriate exhaust valve timing. The charge is then heated during compression until combustion around TDC. The EVC position controls the combustion timing but at the same time it also influences the mass flow in the engine. To be able to simulate a turbocharged HCCI engine it is important to set the EVC position correct as seen in Figure 13. Here an EVC sweep is done in GT-Power and a change of EVC position of 3 CAD makes the boost pressure differ 12 kPa at this load and speed (2250 rpm and $IMEP_{net}$ 400 kPa). Since the combustions noise relates to the RI level it is vital to predict the boost pressure to choose a suitable sized turbocharger.

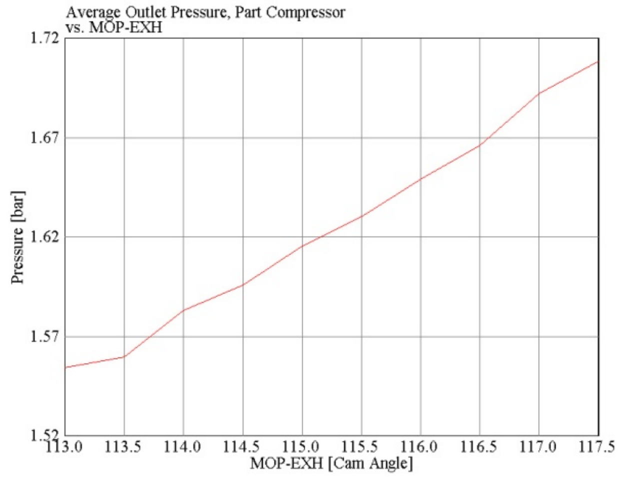

Figure 13: Intake pressure vs. EVC (MOP-EXH) in an engine simulation program

For simulating it is advisable to have the temperature close to TDC as a boundary condition for controlling the EVC position. From the engine testing the calculated temperature at intake valve closing (T_{IVC}) is seen in Figure 14. To estimate this temperature- T_{IVC}, the EVC position sets the trapped volume of internal EGR. The exhaust density is calculated from the lambda level that gives the chemical composition, these species is then adjusted by the exhaust temperature and pressure to calculate the internal EGR mass. The inducted mass is estimated using the intake temperature and pressure together with the lambda level, the measured fuel flow and the external EGR ratio. The mass fractions of the internal EGR and the inducted mass with their respective temperatures are then summarized. By adding the fuel vaporization cooling effect and Woschni heat transfer model the T_{IVC} can be estimated by an ideal cycle calculation. As can be seen the T_{IVC} variation is almost 100 degrees.

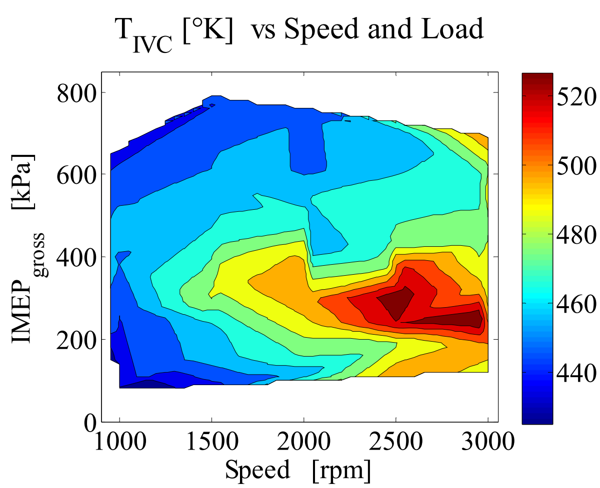

Figure 14: Temperature at intake valve closing, T_{IVC}

The same calculation gives the burned gas fraction (x_b) since the T_{IVC} calculation gives the respective mass ratio. This ratio can be seen in Figure 15. At low load it might look like there is no oxygen left to burn with a burned gas fraction around 90 % but the engine is operated lean here and the internal oxygen level is about 8 %. By setting the specific heat ratio as a function of the burned gas fraction as seen in Figure 16, the specific heat ratio varies between 1.30 to 1.36. By using the law of ideal gases, the predicted temperature at TDC (T_{TDC}) is calculated without regard to combustion as seen in Figure 17. This simplified method for estimating the temperature at TDC is normally an option for controlling the engine and it indicates that a temperature of ~ 1000 K is reached at TDC in most of the test area. This is also in line with earlier findings. At low load the engine has more part of the fuel injected in NVO period for fuel reformation due to limitation in EVC range in this high load set-up. This temperature increase is not reflected in the estimated T_{TDC}. At low engine speed there is also influence from the spark assisting, which is not reflected in the T_{TDC}. In Figure 18 the used EVC timing from this engine testing is seen.

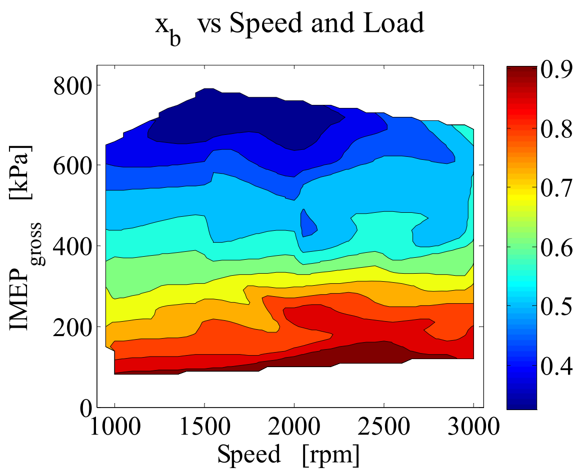

Figure 15: Burned gas fraction, x_b

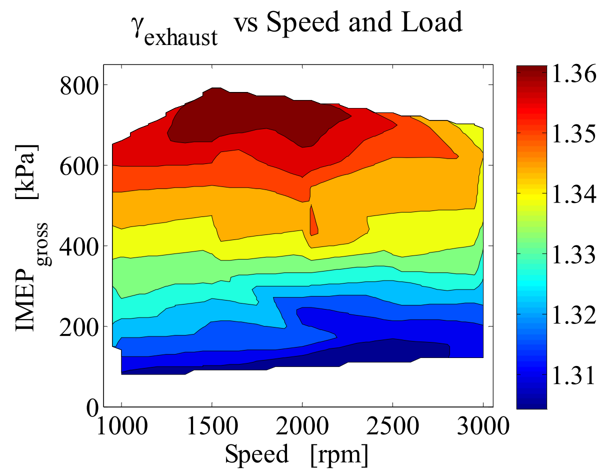

Figure 16: Specific heat ratio, γ

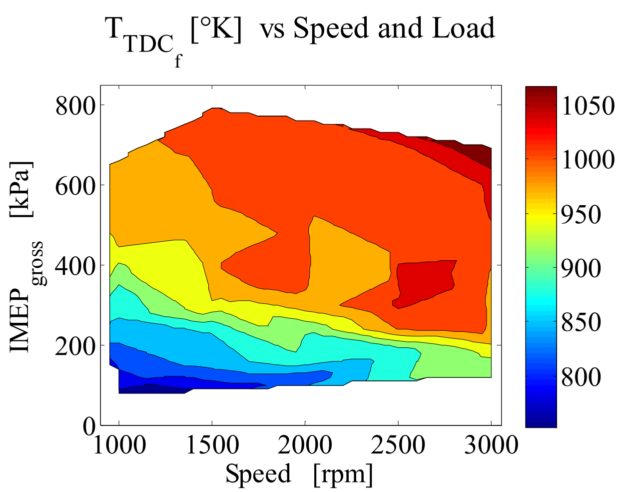

Figure 17: Temperature at TDC, T_{TDC}

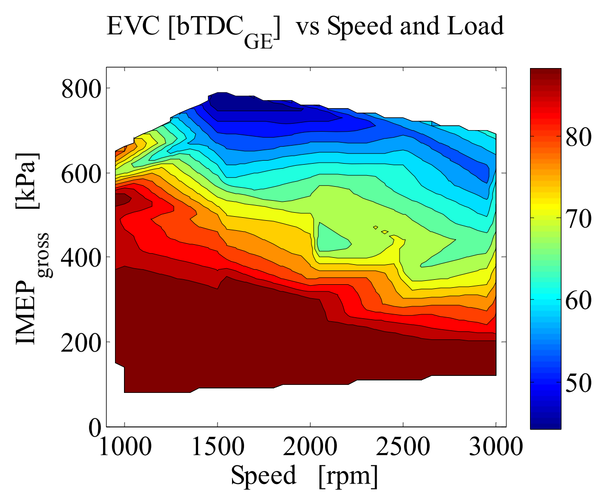

Figure 18: Exhaust valve closing timing, EVC

When simulating this turbocharged HCCI engine at normal and high load, the EVC position is set by the T_{TDC} by simple control blocks in the GT-Power simulating program as seen in Figure 19. This has shown that the predicted EVC position and therefore the boost pressure follow the real engine tests closely.

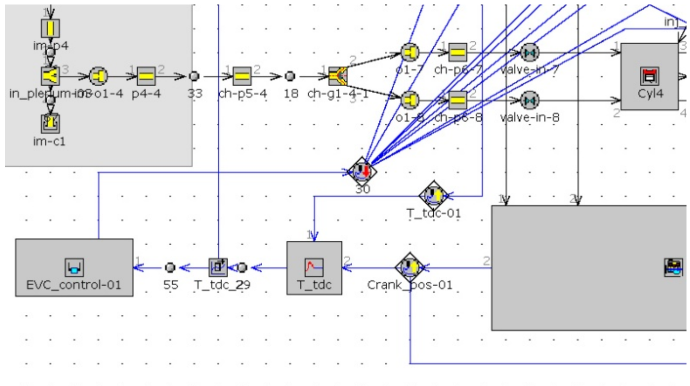

Figure 19: Temperature control in the engine simulating program

By using the combustion parameters shown above from this extensive operating range, it is possible to accurately predict this turbocharged HCCI engine using an engine simulating program as seen in Figure 20 and Figure 21. In this GT-Power simulation, the CA50 timing, the CA10 to CA90 duration and Wiebe constant follows the result dependent functions described above. The EVC position is controlled to have 1000 K at 5 CAD before TDC_f. The simulation is a full factorial [19] run where the engine speed and fuel amount are varied. The only variable that was map based from real engine testing was an imposed intake temperature. The difference at highest intake pressures (compared to Figure A-7) is due to simulation outside the turbocharger map range. These combustion functions have been used to evaluate different camshaft profiles and turbochargers etc. before any changes have been made to the test engine. Even if the presented data is unique for this engine set-up it can be used as a starting point for simulation of other HCCI concepts until enough data is collected to calibrate a suitable model. In this paper it is described how it can be done and it might then serve as a guideline for other HCCI simulations.

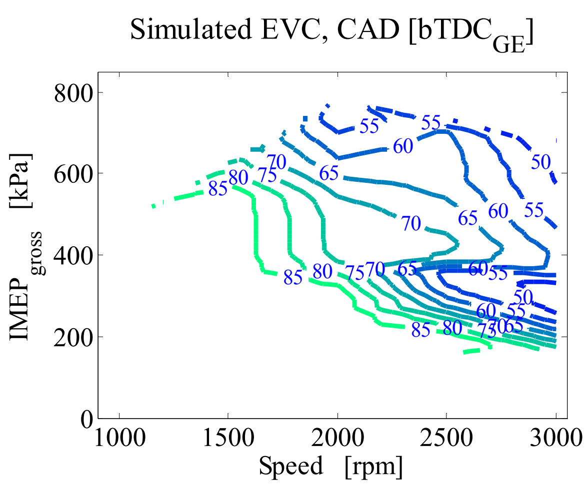

Figure 20: Simulated exhaust valve closing angle, EVC

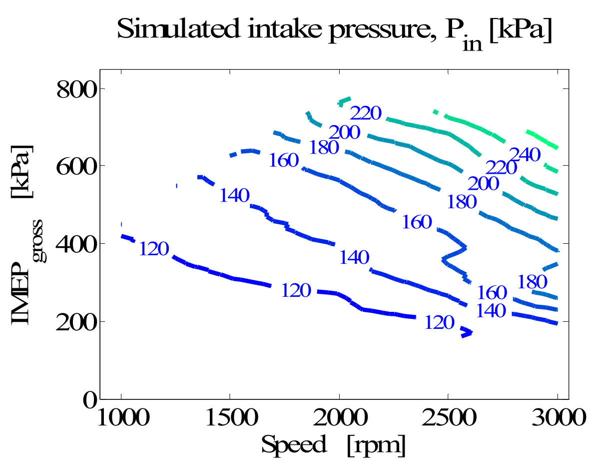

Figure 21: Simulated intake pressure, Pin

SUMMARY/CONCLUSIONS

In this paper it has been shown which limitations one can expect a HCCI engine will operate under. The results showed that it was the high soot level that limited the high load range at low engine speed. At medium speed it was limited by the combustion noise and peak cylinder pressure, at high engine speed the load range was limited by the turbocharger performance.

The combustion timing is an important parameter when operating in HCCI mode and it is even more important in a turbocharged engine since it also controls the mass flow in the engine for a given load. If the combustion timing is advanced for a given load more internal EGR is needed by an advanced exhaust valve closing timing and therefore the mass flow is reduced. It has been shown that the used CA50 timing scales to load and a trend line is presented that is used for simulation. Since the load and burned gas fraction are directly connected here, the CA50 could also been made to scale to the burned gas fraction but in a simulations it is usually easier to predict what kind of load that is going to be simulated.

The combustion duration is shown to increase with engine speed when looking at crank angle degrees but in the time domain it stays fairly constant for the used combustion timings in this test. The heat release from this operating range is used to find the Wiebe function constant; *m,* that can be used to describe the heat release rate for this turbocharged HCCI engine. The *m* constant is found by a non linear least square function which gives a Wiebe heat release that closely follows the real heat release. It is shown that the *m* constant a scale to load and a trend line is presented that is used for one-dimensional heat release simulations.

The exhaust valve closing timing needs to be closely predicted since it sets the combustion timing and also influences the mass flow in the engine by setting how much internal EGR is used. The calculated temperature at intake valve closing is used together with the burned gas fraction to predict in a simple way the temperature at TDC by using the ideal gas law. This calculation shows a temperature at ~1000 K at TDC. This temperature is used as a boundary condition for controlling the exhaust valve timing in the engine simulation and it will then predict the exhaust valve closing timing and therefore the intake pressure accurately.

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DEFINITIONS/ABBREVIATIONS

BMEP	Brake Mean Effective Pressure
CA10	Crank angle 10 % burned
CA50	Crank angle 50 % burned
CA90	Crank angle 90 % burned
CB	Cylinder Balancing
CAD	Crank Angle Degrees
CoV	Coefficient of Variation
DI	Direct Injection
dP	Pressure derivate
EI	Emission Index
EGR	Exhaust Gas Recirculation
EVC	Exhaust Valve Closing
FSN	Filter Smoke Number
HCCI	Homogeneous Charge Compression Ignition
$IMEP_{gross}$	Indicated Mean Effective Pressure, gross
$IMEP_{net}$	Indicated Mean Effective Pressure, net
IVC	Intake Valve Closing
NA	Naturally Aspirated
NOx	Nitrogen Oxide
NVO	Negative Valve Overlap
MBT	Maximum Brake Torque
PCP	Peak Cylinder Pressure
RON	Research Octane Number
RI	Ringing Intensity
SI	Spark Ignition
STD	Standard Deviation
TDC	Top Dead Center
TDC_f	Top Dead Center, firing
TDC_{GE}	Top dead center, gas exchange
T_{IVC}	Temperature at IVC
T_{TDC}	Temperature at TDC
TWC	Three Way Catalyst
VVT	Variable Valve Timing
x_b	Burned gas fraction

APPENDIX

RESULTS FROM SPEED AND LOAD TESTING

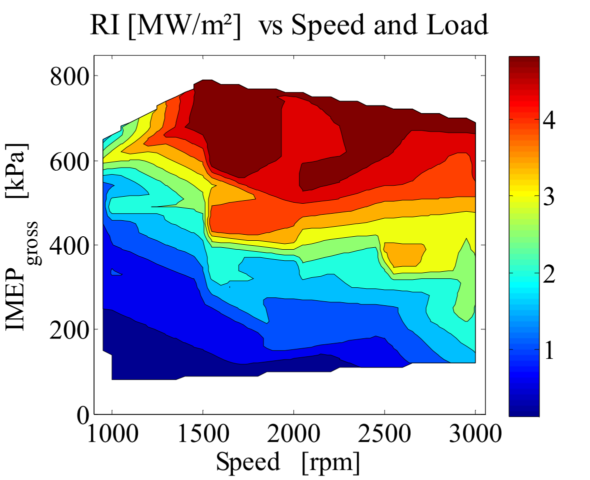

Figure A-1: Combustion noise, RI

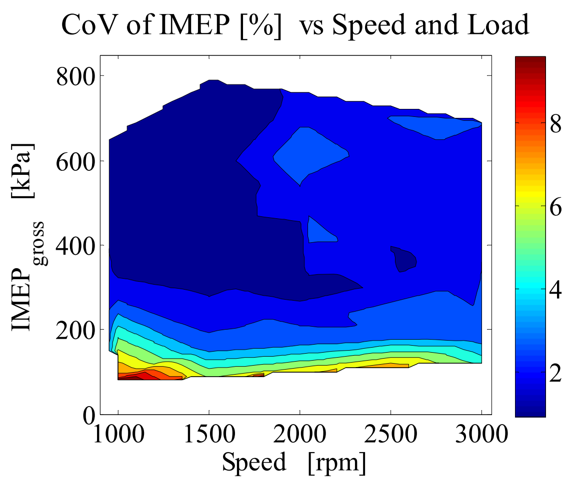

Figure A-2: Combustion stability, CoV of IMEP

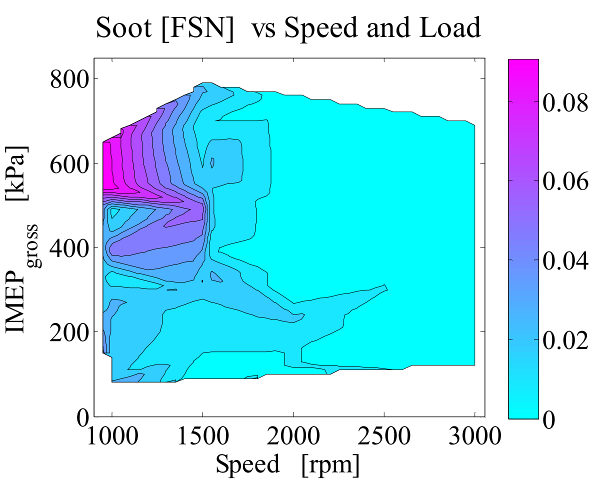

Figure A-3: Soot level, FSN

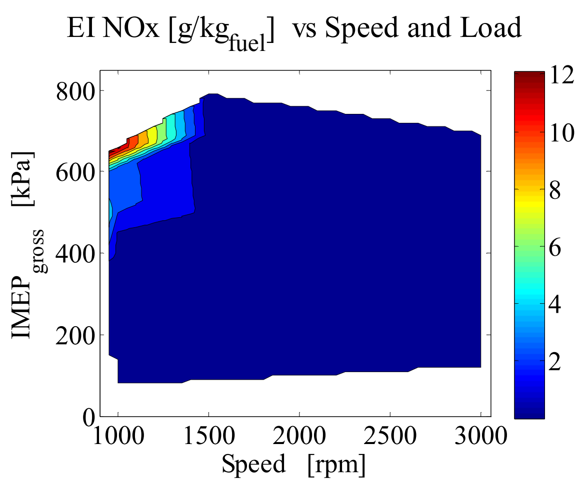

Figure A-4: Nitrogen oxides emission, EI NOx

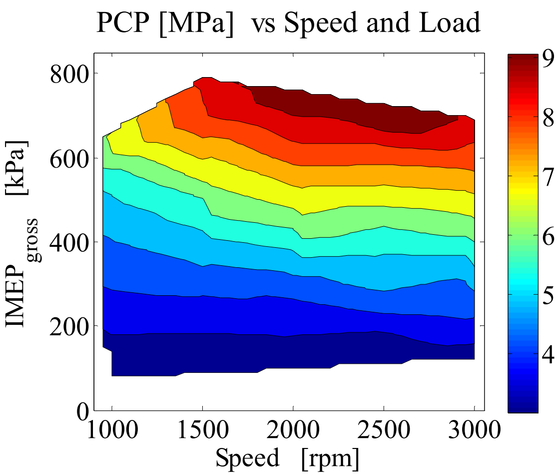

Figure A-5: Peak cylinder pressure, PCP

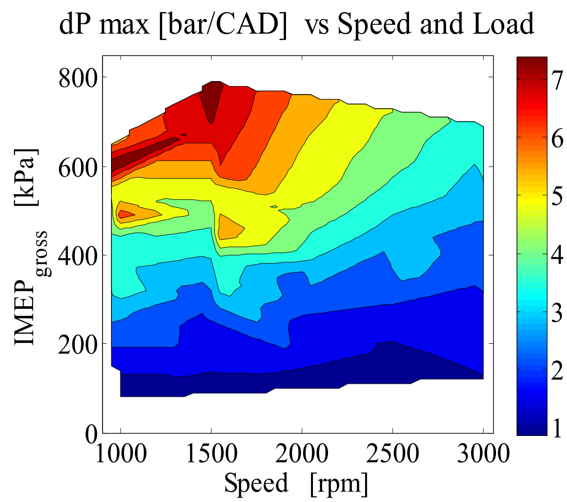

Figure A-6: Maximum pressure rise rate

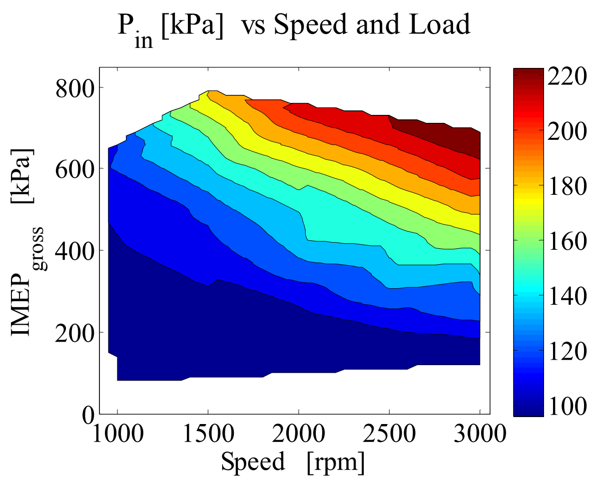

Figure A-7: Intake pressure, P_{in}