

The Effect of Intake Temperature in a Turbocharged Multi Cylinder Engine operating in HCCI mode

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ABSTRACT

The operating range in HCCI mode is limited by the excessive pressure rise rate and therefore high combustion induced noise. The HCCI range can be extended with turbocharging which enables increased dilution of the charge and thus a reduction of combustion noise. When the engine is turbocharged the intake charge will have a high temperature at increased boost pressure and can then be regulated in a cooling circuit. Limitations and benefits are examined at 2250 rpm and 400 kPa indicated mean effective pressure. It is shown that combustion stability, combustion noise and engine efficiency have to be balanced since they have optimums at different intake temperatures and combustion timings. The span for combustion timings with high combustion stability is narrower at some intake temperatures and the usage of external EGR can improve the combustion stability. It is found that the standard deviation of combustion timing is a useful tool for evaluating cycle to cycle variations. One of the benefits with HCCI is the low pumping losses, but when load and boost pressure is increased there is an increase in pumping losses when using negative valve overlap. The pumping losses can then be circumvented to some extent with a low intake temperature or EGR, leading to more beneficial valve timings at high load.

INTRODUCTION

The homogenous charge compression ignition (HCCI) combustion offers high thermal efficiency, low throttling losses and low emissions. The limited operating range in HCCI mode makes it to an alternative to the spark ignited (SI) engine at low load to improve the efficiency. On the other hand the diesel engine has shown some favorable properties when operated with fuels with long ignition delay like gasoline [1] leading to low soot and nitrogen oxides (NO_x) levels. Here the HCCI mode or a mix of it can be attractive to cover the whole operating range. 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.

When operating with a homogenous charge the right conditions for the combustion timing has to be set before intake valve closing (IVC). With the introduction of direct injection (DI) of the fuel there is an increased control space [2, 3] and even a possibility to change the operating conditions after IVC, so the preferred charge composition does not have to, or will be homogenous [4].

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 [5]. The compression ratio can also be used to control the HCCI combustion timing [6, 7]. The concept with negative valve overlap (NVO) with 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 [8] 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 [9, 10], but this need a heat recovery system.

The combination of intake heating and NVO [11] in a naturally aspirated (NA) engine showed that the intake temperature had a low impact on engine operating but increased temperature could be used to extend the low load limit. By boosting the engine in HCCI mode to extend the possible load range [12, 13, 14, 15, 16], the intake temperature will be elevated and can therefore be cooled down to a suitable level with an intercooler. The boost pressure can be controlled directly by the intake valve timing [16] 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 control how much internal EGR is needed for a given combustion timing and therefore the in-cylinder relative burn gas fraction. The chemical and thermodynamic properties of the EGR [17] will influence the combustion. In addition to the residual gas, 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 [18] 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 effect 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 running will be narrowed, making the engine control more challenging.

The scope of this paper is to investigate how combustion timing, intake temperature and EGR influence a turbocharged NVO HCCI engine performance. This is done by dividing the result in small fractions. By doing this the understanding on how to operate a turbocharged HCCI engine is expanded. Since the application of HCCI has to meet consumer demand, emission legislation and control capability the real operating range is always a balance between these. The result is also applicable to some extent for a NA HCCI engine.

EXPERIMENTAL SET-UP

ENGINE SYSTEM The test engine is an in-line four cylinder gasoline engine with a total displacement of 2.2l. 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 DI with a slightly canted, centrally placed injector of the Spray-Guided type. 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 designed for a NA HCCI engine. In Figure 1 the minimum and maximum valve timings are plotted.

The 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. On the intake side there is a water cooled intercooler. The heated aluminum intake manifold has short intake runners and a small volume. 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.

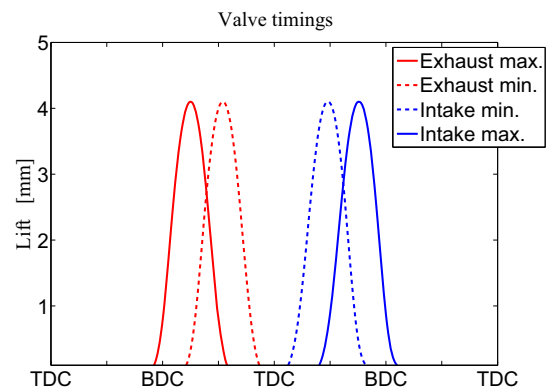


Figure 1: Valve timing range and setup

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	125 CAD
Valve lift	4.1 mm
NVO range	74–174 CAD
Turbocharger	B&W KP31
Fuel Supply	DI, up to 20 MPa
Fuel Type	Gasoline, 95 RON



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. 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 ($IMEP_{net}$) and peak pressure rise rate. This means that the control system has to perform cylinder balancing (CB). Injection of the fuel in NVO can give a reformation of the fuel with an increase in charge temperature and reduction in effective octane number leading to 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 bal-

ancing there will also be some differences in efficiency between the cylinders. This results in different load, $IMEP_{net}$ in the cylinders. To maximize the load range there has to be CB of $IMEP_{net}$ 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 CB has to be constantly monitored and adjusted by the control system. Table 2 shows the measurement and control equipment.

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 due to less flow discharge compared to 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.

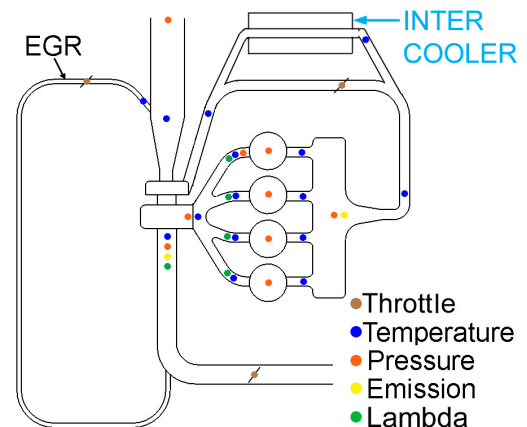


Figure 4: Engine layout

The presented results are based on average results from all four cylinders during 1000 cycles. The deviation in engine load between measured test points is ± 3 kPa and in engine speed ± 2 rpm. Engine coolant temperature is $\sim 80^\circ C$ out from the engine.

RESULTS AND DISCUSSION

The combustion phasing in HCCI mode is vital for stable engine running and efficiency. In the NVO concept the EVC timing is the major control variable for the combustion phasing since it controls the amount of trapped hot residual gas. The intake valve timing can also be used to control the combustion timing but the influence is weaker and not as predictive as the EVC timing which makes it less suitable for cycle to cycle combustion control. It is normal to operate with almost symmetrical valve timings on the intake and exhaust. With these short duration valve timings, a late intake valve closing will increase the available in-cylinder pressure when the engine is turbocharged. The combustion timing is even more important due to the direct influence it has on boost pressure and therefore the intake temperature. A later combustion timing will increase the available exhaust enthalpy for the exhaust turbine leading to higher boost pressure. This means that the intake temperature will also increase. There is then the freedom to adjust the intake charge temperature by the cooling circuit to obtain best overall performance depending on load and engine speed.

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 with turbocharging the audible noise is suppressed and therefore the operating range can be extended. The pressure rise rate can be measured in bar/CAD or MPa/ms [19] and is used to impose a operating limit. But neither of these catch the real combustion noise when the engine is supercharged. The Ringing Intensity (RI)– MW/m² [20], on the other hand compares the pressure rise rate to peak cylinder pressure and engine speed and is therefore used in these tests.

The engine tests in this study are performed at 2250 rpm and 400 kPa IMEP_{net}. This load and speed point was chosen due to the wide span of boost levels that can be reached here with alteration of the intake temperature. It also has different operating limitations due to combustion stability, high pressure rise rate and turbocharger performance. Even if it is possible to operate this turbocharged HCCI engine at a higher load at this engine speed, it is unsuitable for a wide CA50 sweep due to the high pressure rise rate at early combustion timings. In the first set of results the CA50 position and intake temperature are swept. Then there is an external EGR and intake temperature sweep at locked CA50 timing, and finally, there is a CA50 sweep with and without EGR at the same intake temperature. During these tests the intake valve timing is constrained to latest possible opening position to increase available in-cylinder pressure.

CA50 AND INTAKE TEMPERATURE SWEEP AT 2250 RPM AND 400 KPA IMEP_{NET} From this test there are some additional graphs that can be found in the APPENDIX. In Figure 5 the intake manifold pressure (P_{in}) is

shown at the possible operating range in this test point. The temperature can be altered from 110° C down to 45° C with the throttle controlled intercooler routing. The timing for CA50 can not be set to more than ~ 7° aTDC_f for stable engine operating at this test point. The importance for robust CA50 control is evident when operating this turbocharged HCCI engine since it is dependent on the boost pressure to decrease RI to manage high load. The resulting PCP reflects the wide intake pressure variations and span from 6.6 to 8.2 MPa, see Figure 6.

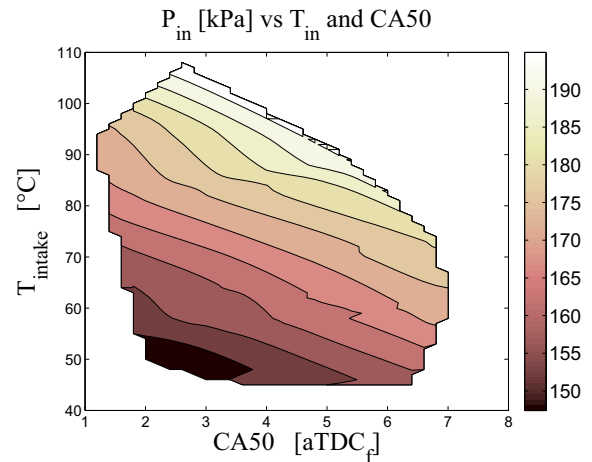


Figure 5: Intake pressure, absolute

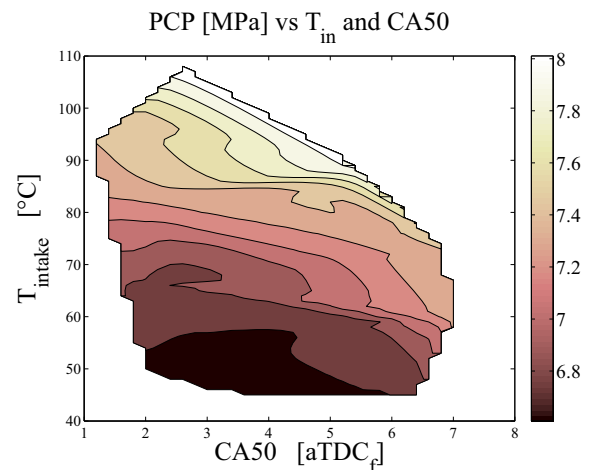


Figure 6: Peak cylinder pressure

The influence of the intake temperature is strong on achievable boost level. Since less burned gas fraction (x_b) is needed, see Figure 7, at higher intake temperatures for a given combustion phasing, the specific heat capacity in the charge is reduced and thus the mass flow rate is increased. This increased mass flow to the turbocharger increase the boost pressure.

The corresponding charge temperature at intake valve closing (T_{IVC}) is plotted in Figure 8. To shift the CA50 one degree, the T_{IVC} changes approximately 8°, except at low temperatures where the CA50 timing is very sensitive and changes rapidly to the T_{IVC} . The corresponding EVC position that is used to set the combustion phasing is seen in Figure 9.

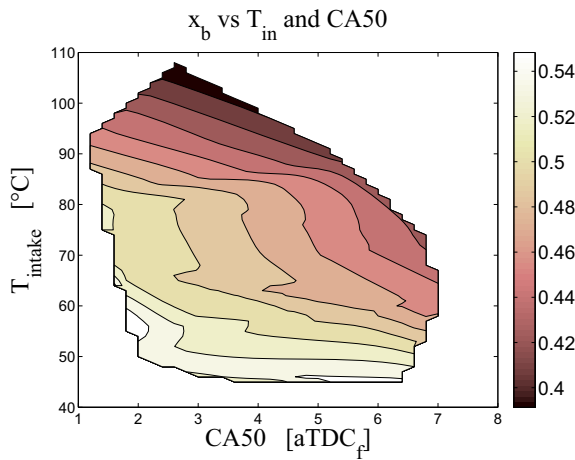


Figure 7: Burned gas fraction, x_b

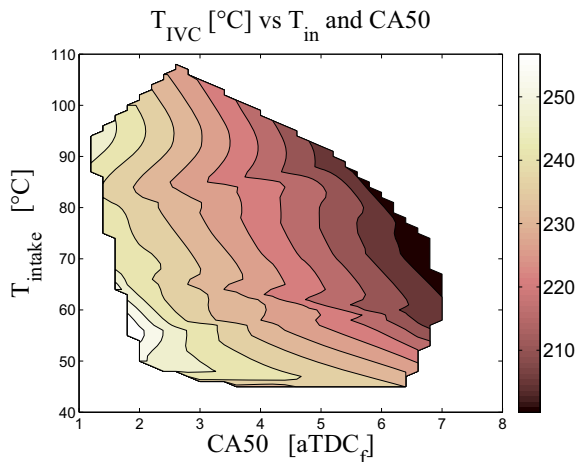


Figure 8: Temperature at IVC

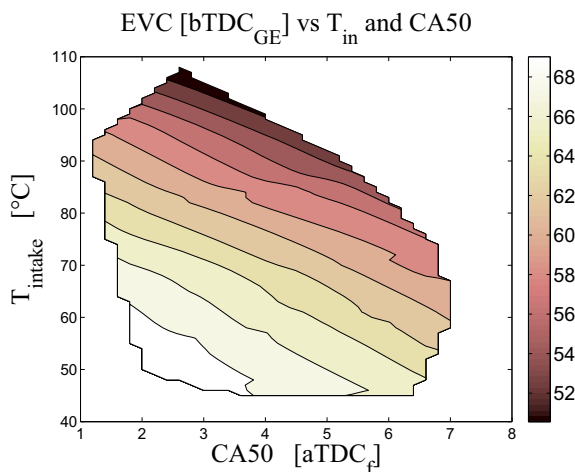


Figure 9: EVC position

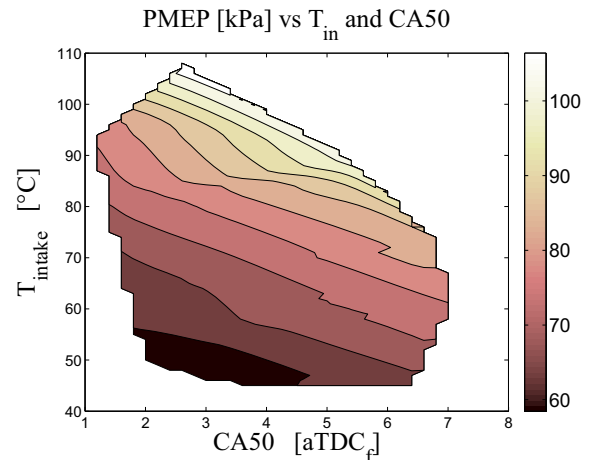


Figure 10: Pumping losses

$P_{ex} - P_{in}$: The pressure loss over turbine/compressor is shown in Figure 11.

Throttling_{valve}: The pressure loss over inlet and exhaust valves is shown in Figure 12.

With a turbocharger one usually encounters higher back pressure than boost pressure. Since the turbocharger is on the small side here, it was not possible to have a later CA50 timing than 3° (aTDC_f) the highest intake temperature. The reason to this is that the back pressure increased considerable which led to reduced possible load range. The high throttling losses originates from the short duration valve timings. For example, at high HCCI load a late EVC is needed which lead to that the exhaust valve opening (EVO) can be close to BDC. This will increase the pumping work from the blow-down process.

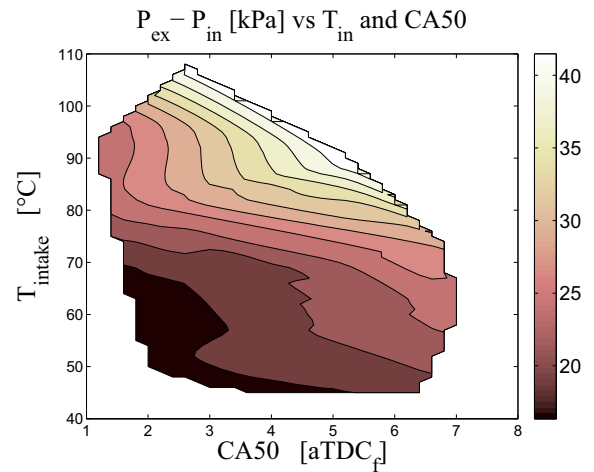


Figure 11: Pressure loss over turbine/ compressor

When the boost level is increased it often comes at a cost, in the form of increased pumping losses that is represented as pumping mean effective pressure (PMEP) in Figure 10. The achievable boost pressure has to be weighted against the pumping losses from the turbocharger inefficiency and throttling losses over the valves, to keep engine efficiency as high as possible.

By separating the pumping losses in two parts further information is available: $PMEP = \text{Throttling}_{\text{valve}} + (P_{ex} - P_{in})$

To decrease the engine noise the combustion timing can be delayed but this can increase the probability for misfire. The coefficient of variance (CoV) of $IMEP_{net}$ as seen in Figure 13 can be used as an indicator of combustion stability. Since the nature of HCCI combustion inhibits some cycle to cycle variations of combustion timing, the CoV of $IMEP_{net}$ need to be complemented by variations in combustion phasing as well. In Figure 14 the standard deviation (STD) of CA50 is plotted. Looking at the CoV graph, a later CA50 timing is the major influence of low