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HCCI Operating Range in a Turbo-charged Multi Cylinder Engine with VVT and Spray-Guided DI

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ABSTRACT

Homogenous charge compression ignition (HCCI) has been identified as a promising way to increase the efficiency of the spark-ignited engine, while maintaining low emissions. The challenge with HCCI combustion is excessive pressure rise rate, quantified here with Ringing Intensity. Turbocharging enables increased dilution of the charge and thus a reduction of the Ringing Intensity. The engine used is an SI four cylinder base with 2.2*L* displacement and is equipped with a turbocharger. Combustion phasing control is achieved with individual intake/ exhaust cam phasing. Fuel injection with spray guided design is used. Cycle resolved combustion state is monitored and used for controlling the engine either in closed or open loop where balancing of cylinder to cylinder variations has to be done to run the engine at high HCCI load. When load is increased the NOx levels rise, the engine is then run in stoichiometric HCCI mode to be able to use a simple three-way catalyst. The fuel used is 95 RON pump gasoline and injection strategies are evaluated in order to maintain low soot levels and high efficiency. Limitations and benefits on operating range are examined between 1000 and 3000 rpm. This paper investigates how to extend the HCCI range and how to reduce the high pressure rise rate with: increased boost from turbocharging, external EGR and different injection strategies. A higher boost pressure was found to extend the load range. It is shown that the limitation from high RI, NOx or soot is not the same in all engine speed and load points. By turbocharging the engine in HCCI mode there is greater flexibility to increase the range of practical operating points.

INTRODUCTION

The HCCI mode offer advantages in high thermal efficiency and low emissions but so far has the possible load range been limited compared to conventional spark ignition (SI) engines. HCCI can be a complement to the throttled SI engine during part load to increase engine efficiency. By boosting the engine in HCCI mode the possible load range can be extended [1, 2, 3, 4] and total efficiency can be improved. The need for mode switches between HCCI and SI is reduced which simplifies the control task. The control of combustion timing in internal combustion engines has always been important to maximize efficiency and avoid damages. The SI engine uses spark to ignite the air/fuel mixture and therefore it relies on a suitable fuel type with high octane number to resist autoignition or knock. The drawback is that in order to use spark ignition the engine must be operated near stoichiometric conditions [5] with a three-way catalyst (TWC). Thus the SI engine has to be throttled at lower loads leading to efficiency losses, alternatively the SI engine can have a stratified charge to reduce the throttling losses but this demands a more complicated exhaust after-treatment if operated lean.

The high temperature in the flame front in the SI engine raises the nitrogen-oxide (NOx) emissions. The NOx emissions can be reduced in a normal TWC if the engine is run stoichiometric. In the HCCI engine the temperature of the fuel/air mixture is raised during the compression stroke until it is autoignited around top dead center (TDC). The combustion temperature is lower than in an SI engine and this reduces NOx emission formation.

Since the heat release rate with autoignition is much higher than from a normal SI flame front the time window is very short for successful timing and all the right conditions have to be set during the intake stroke. The autoignition can be made with lean air/fuel mixtures and therefore throttling losses are reduced.

The implementation of HCCI on a SI engine is usually done with minor modification to the SI engine structure but this might need to be changed to maximize the potential of HCCI. The scope of this paper is to investigate the upper load range limit for a turbocharged HCCI engine by imposing some operating limits. The boost pressure versus back pressure is considered for best efficiency and balanced against the combustion noise reduction the increased boost has.

The HCCI combustion has been researched on the last decades now and there is published a lot of results but the operating limitations is often just a few. This paper shows which limitations one can expect the HCCI has to fulfill and how it influences the operating range.

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 is direct injected (DI) with a slightly canted, centrally placed injector and is of the Spray-Guided type. The eight-hole solenoid fuel injector has a cone angle of 60^o. The canted spark plug is located close to the fuel injector and has an extended tip. To achieve HCCI combustion the engine is run with negative valve overlap (NVO), with short lift and short duration camshafts designed for a naturally aspirated (NA) HCCI engine. In Figure 1 the minimum and maximum valve timings is plotted. The variable valve timing (VVT) is controlled by hydraulic actuators at the camshafts, meaning there is a separate 50 crank angle degree (CAD) adjustment on both intake and exhaust valve timings. The engine is turbocharged by a variable geometry turbine (VGT), the VGT position is controlled by an electric actuator. 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 charge intercooler. The aluminum intake manifold has short intake runners and a small volume, it is mounted tight against the cylinder head 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.

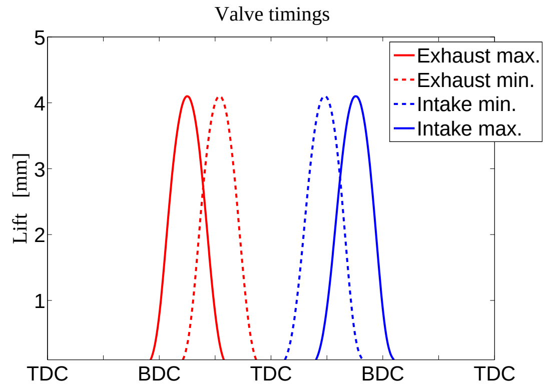

Figure 1: Valve timing range and setup

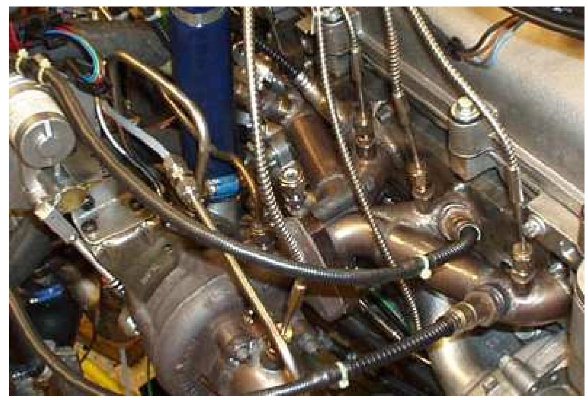

Figure 2: Exhaust manifold

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 timing for 50% heat released (CA50), peak cylinder pressure (PCP) and peak pressure derivative (dP/CAD) etc. are used for cylinder individual control in closed or open loop.

Emission analysis and soot measurement is collected on an external PC and the data is sent to the control unit. 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
Camshaft duration	125 CAD
Camshaft lift	4.1 mm
NVO range	44–144 CAD
Turbocharger	B&W BV35
Fuel Supply	DI, up to 20 MPa
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

RESULTS AND DISCUSSION

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 for example turbocharging the audible noise is suppressed and the HCCI load range can be extended. By increasing possible load range in HCCI the total efficiency can be increased and there will be less need for HCCI-SI mode switching that imposes in itself some challenging control tasks. The mode switching can for example involve mechanical switching between SI and HCCI camshaft profiles. In fairness the boosted mode switches HCCI-SI adds more complexity since the intake manifold pressure is different in HCCI and SI mode.

Normally the pressure rise rate is measured in bar/CAD or MPa/ms [6] 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 [7] on the other hand compares the pressure rise rate to peak cylinder pressure and engine speed. In this study RI has been used as a criterion to limit the pressure rise rate. In Figure 4 there is a comparison between these different pressure rise rate expressions when the load is constant and boost pressure is varied.

To be able to increase the load range we want as high intake pressure as possible to keep combustion noise at a low level, meaning we want a low RI number, preferably below 6 MW/m^2, a limit Eng[7] found that Lund used in earlier tests. It is common to operate the NVO HCCI with almost symmetrical valve timings, meaning that the intake valve opening (IVO) follows the exhaust valve closing (EVC). As the load gets higher less EVC is needed to keep the combustion timing and with these short duration camshafts the intake valve closing (IVC) will then be before bottom dead center (BDC). As seen in Figure 5 the boost pressure is increased by delaying IVC with asymmetrical valve timing to increase cylinder filling.

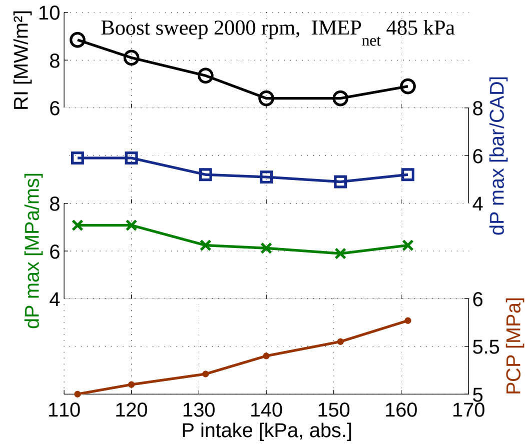

Figure 4: Boost sweep with different pressure rise rate expressions

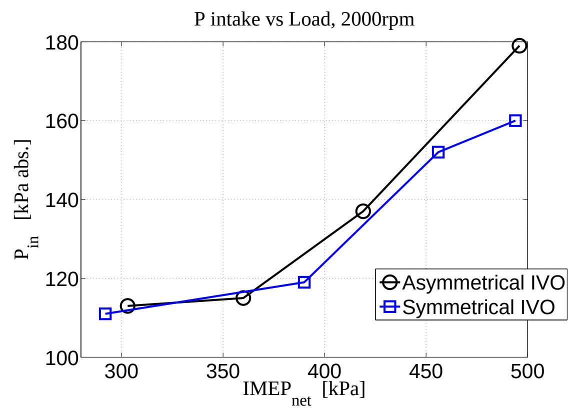

Figure 5: Load sweep with symmetrical valve timing and IVO locked at 72^oaTDC

The achievable boost pressure has to be weighted against pumping losses due to turbocharger inefficiency and throttling losses over the valves, to keep engine efficiency as high as possible– see Figure 6. Is there a downfall with high boost pressure beside efficiency concerns?

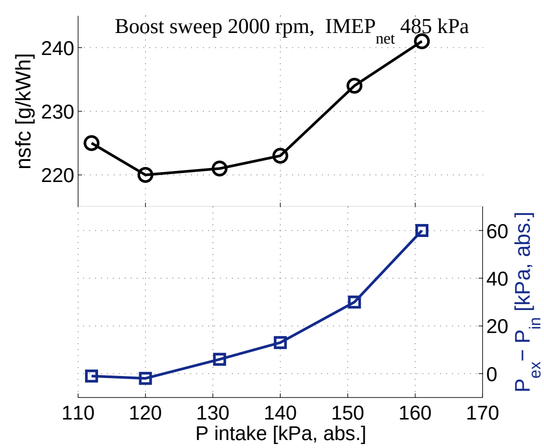

Figure 6: Boost sweep versus nsfc and back pressure

Since PCP scales with the intake pressure– see Figure 4, it can become a limiting factor. There is risk for structural damage of the engine if the design limit is exceeded. An example of consequences of high PCP can be seen in

Figure 7. The engine was operated at 2000 rpm and 600 kPa indicated mean effective pressure (IMEP$_{net}$), intake pressure at 220 kPa (abs.) and PCP at 10.5 MPa, RI at 8, when a piston failure occurred. Finite element analysis

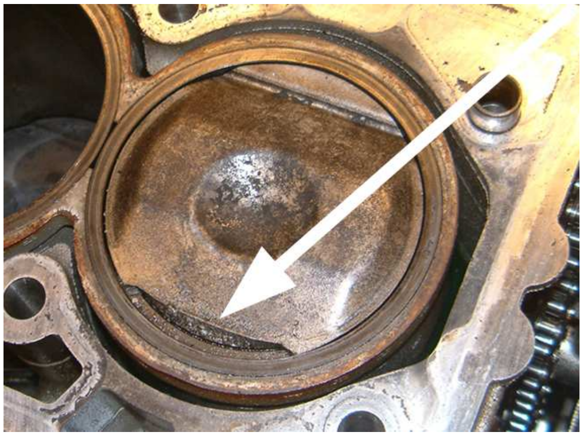

Figure 7: Damaged piston due to high PCP

(FEA) reveals that the piston stress level was above the fatigue limit as seen in Figure 8. A redesign of the piston was made with increased top land height and the PCP limit was reduced to 8.5 MPa.

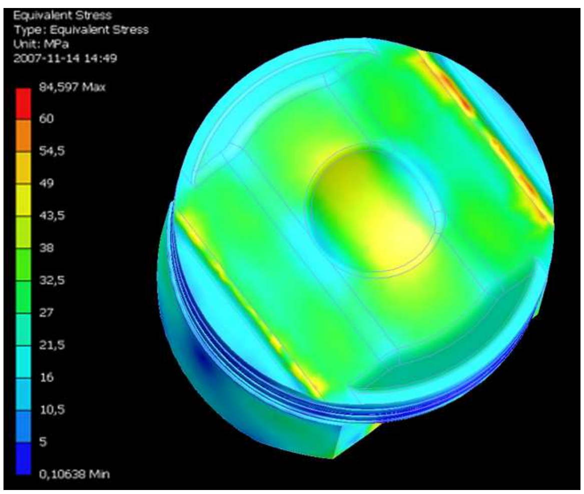

Figure 8: Piston failure FEA

The combustion phasing– CA50 is the main operating parameter when running in HCCI mode, a later CA50 can raise upper load limit by decreasing the RI and PCP as seen in Figure 9, 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, here the turbocharger sizing plays an important role. The boosted HCCI engine can on the other hand for a given RI use a better phased combustion and hence improve thermal efficiency compared to a naturally aspirated HCCI engine. The temperature and pressure decreases after TDC, and since the autoignition is dependent on both to start the chemical reactions, there is an increase in cycle to cycle variances (CoV) with late combustion phasing. Is there a risk with too late combustion timing?

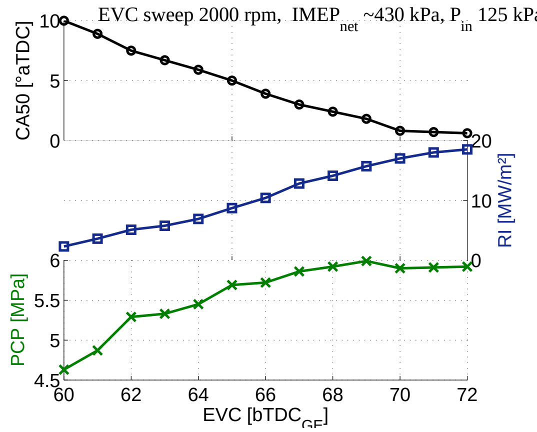

Figure 9: EVC sweep vs. CA50 and PCP

HCCI is known for its low CoV of IMEP$_{net}$ compared to SI engines but when load is maximized with a late CA50 the CoV of IMEP$_{net}$ increases. 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 possibility 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.

To increase the operating range it is vital that all the cylinders operate identically, meaning that they have the same CA50 position, IMEP$_{net}$ and RI. 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 therefore advances the combustion timing, it will be more advanced if the fuel is injected in the beginning of the NVO than in the end, as seen in Figure 10 where a start of injection (SOI) sweep is done with fixed valve timings. The excess air during NVO injection influences the fuel reformation and the effect of combustion phasing is more pronounced at lean mixtures [8].

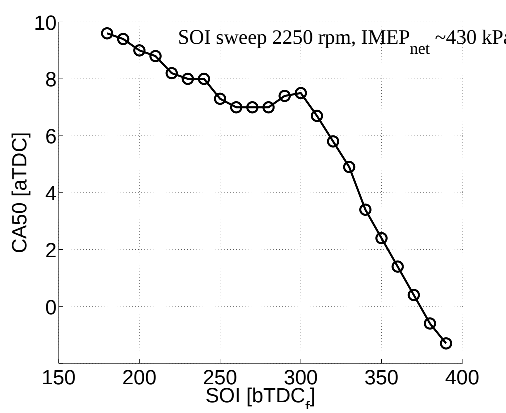

Figure 10: CA50 timing vs. SOI with single injection

The fuel injection timing has an influence on efficiency and soot. For example, injecting the fuel in the NVO sector will require a later EVC to keep the CA50 position correct. This means there will be less energy to the turbine and higher pumping losses from the exhaust blow-down with this type of short duration camshaft operating at high load. Injecting the fuel too late will also decrease efficiency as seen in Figure 11 due to insufficient time for vaporization. By splitting up the fuel injection and injecting a part of the

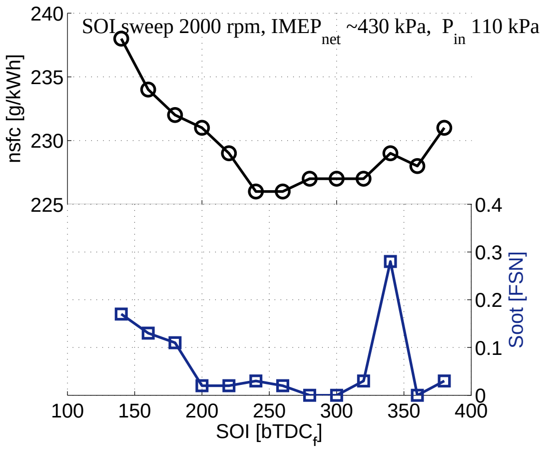

Figure 11: SOI sweep with single fuel injection

fuel in the NVO and the rest later, it is possible to have CB and injection timing with the highest fuel efficiency as seen in Figure 12, here 30% of the fuel is injected in the first injection.

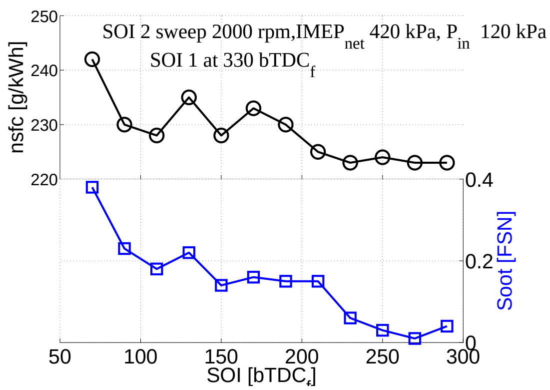

Figure 12: SOI sweep with double fuel injection timings

The control system sets the CA50 position and uses the cylinder with the most advanced CA50 position for EVC set point, the rest of the 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. As 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$_{net}$ between the cylinders. If we want 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, so the CB has to be constantly monitored and adjusted by the control system.

Injecting the fuel around TDC when the piston is close to the injector raises the risk for fuel impingement on the piston surface, leaving a liquid fuel film that can lead to an increase in soot levels. As can be seen in Figure 11 and Figure 12, a late injection timing will also increase the soot level and lower the efficiency. These tests were done with 12 MPa fuel pressure. The soot level can be reduced by increasing the fuel pressure to enhance fuel vaporization, the fuel injector design together with piston design [9] will also influence soot formation. To get a 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 $\lambda = 1.5$ the Smoke opacity arrives roughly at 0.25 %[10] which corresponds to a FSN value of 0.05 [11].

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$_{fuel}$. With higher intake pressure the dilution increases and the NOx level is reduced as seen in Figure 13.

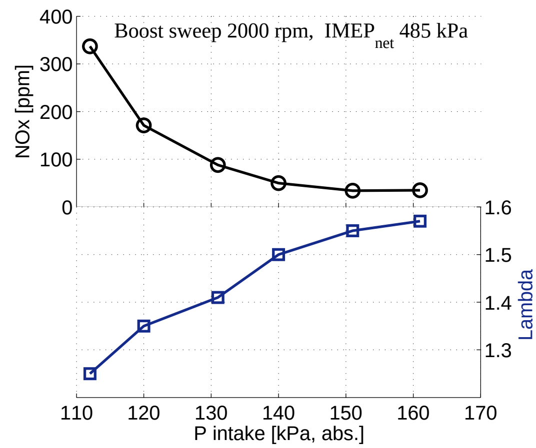

Figure 13: Boost sweep vs. NOx and Lambda

If the NOx level gets too high the engine has to be operated stoichiometric to meet the NOx legislation, there are some concerns about the low temperature in HCCI mode and operating with a TWC but in this engine the need for a TWC is at high load where the light off temperature of the TWC is reached. To run the engine stoichiometrically this engine has external exhaust gas recirculation (EGR). Two types of EGR routing are common; short and long, the short EGR bleeds off exhaust gas before the turbine inlet and route it to the compressor outlet on the intake side. This mean there is a loss of exhaust energy to the turbine– something that is already low in HCCI mode due to the low combustion temperature, leading to an unwanted decrease of intake boost. The long route EGR takes exhaust gas after the turbine and directs 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 can be beneficial due to the thermodynamic cooling effect [12]. Advanced EVC timing is required in order to maintain constant CA50. Pumping losses are thus reduced due to more suitable valve timing at high load. On the other hand the advanced EVC timing increases the heat losses during recompression. The result in the end will depend on speed and load. A drawback with EGR in HCCI mode is the relatively slow response in a system that is normally cycle to cycle controlled, especially during transients. Another drawback is the added weight and cost. Instead of using external EGR the boost pressure should be as high as possible to increase the dilution and suppress the NOx emissions but it has to be weighed against the pumping losses.

When operating the engine at a global $\lambda = 1$ in conjunction with above described cylinder balancing of IMEP$_{net}$ there will be local difference in cylinder to cylinder lambda that can lead to increased CoV, and even to misfire from the cylinder running richest. The lambda difference between all the cylinders when balancing with IMEP$_{net}$ or lambda in two tests is shown in Figure 14. So when operating near $\lambda = 1$ there should be lambda balancing [13] instead of IMEP$_{net}$ balancing for stable engine running.

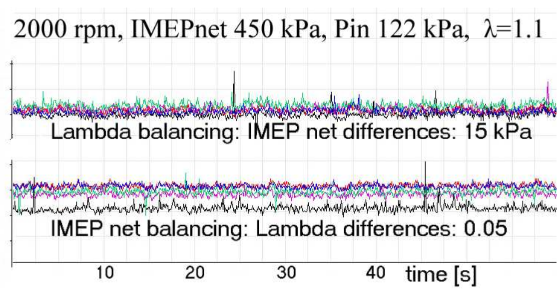

Figure 14: Cylinder balancing with IMEP$_{net}$ and Lambda

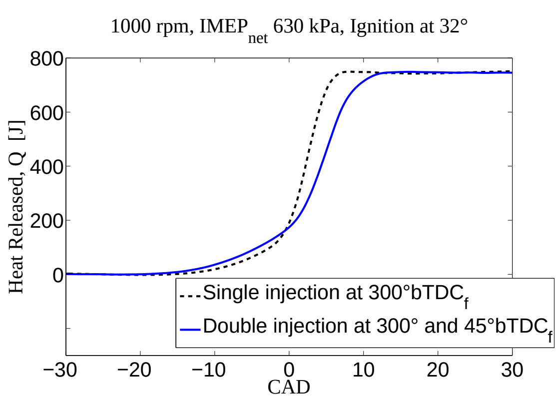

Figure 15: Heat release comparison

The HCCI operating is normally done without a spark to ignite the charge but we have seen that spark assist (SA) can give a lower CoV and thus enable a later combustion phasing. Especially at low engine speed and high load a later phasing is possible with low CoV with SA and thereby increasing possible load, here the NOx emissions is high and thus the engine is already operated stoichiometric. This means that the ignition timing is always adjusted depending on load and engine speed for best result but the influence from SA on the NOx emissions [14] has to be considered. The SA HCCI used in these tests has a normal steep HCCI heat release; in Figure 15 there is a heat release comparison between two injection strategies, the double injection with a small late second injection shows a slow initial heat release, that later becomes steeper. Even if RI is decreased here from 5.5 to 3.5 MW/m^2 with this double injection, the soot level is increased from 0.09 to unacceptable 1.39 FSN.

SPEED AND LOAD TESTING In order to make speed and load sweeps we set some operating limitations outlined previously that one can expect the HCCI engine has to fulfill. The operating limitations are set as following:

– RI below 6 MW/m^2
– CoV below 3.5%
– PCP below 8.5 MPa
– Soot average 0.05 FSN
– EI NOx below 1.1 g/kg$_{fuel}$ if lean, unlimited at stoichiometric

The turbocharged HCCI engine is tested from 1000 to 3000 rpm, from 300 kPa IMEP$_{net}$ and up to the point where operating limitations hinder further increase in possible load. The goal is to have low net specific fuel consumption (nsfc). We used cylinder balancing of combustion phasing and load with late combustion phasing without excessive fluctuations in IMEP$_{net}$. The split injection timings is adapted to every test point. Ignition timing is adjusted for low CoV or late CA50 as long there is no NOx penalty. IVO and EVC timing is most of the time asymmetrical set to increase cylinder filling with this turbo set-up. Fuel pressure during these tests are between 12–16 MPa. Desired fuel amount and VGT position are set manually.

The intake pressure as seen in Figure 16 scales with load and engine speed. Since the turbocharger is a bit large for this HCCI engine the highest intake pressure is around 180 kPa (abs.). The CA50 set-point has to be delayed from MBT timing to maximize exhaust energy and the VGT position is then adjusted for maximum boost. The short duration camshafts on this engine means that for higher loads, less EVC is needed and the exhaust valve opening (EVO) is then close to BDC where the available pressure to the turbine is small. This decreases possible boost pressure and increases pumping losses. When the boost pressure goes up the intake temperature also goes up, as seen in Figure 17. The intake temperature is measured inside the intake channel, 40 mm upstream the intake valve. This temperature is without any intercooler effect during these tests and scales to compressor efficiency plus the intake manifold that is integrated to the cylinder head in this engine set-up.

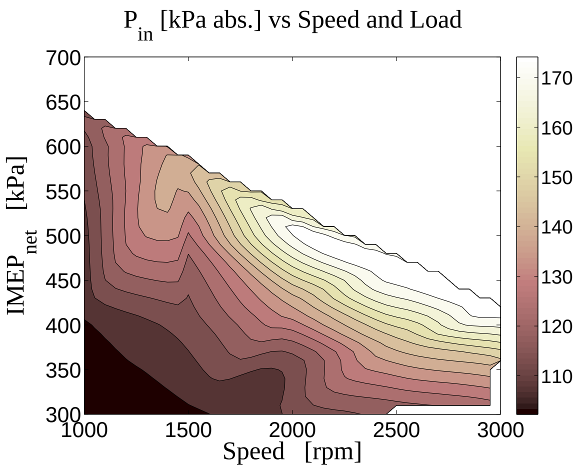

Figure 16: Speed and load testing– Intake pressure

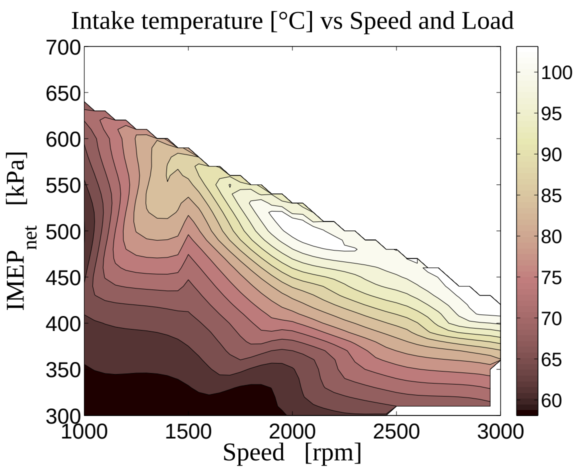

Figure 17: Speed and load testing– Intake temperature

When the emission index (EI) of NOx emission raises above the limitation at 1.1 g/kg$_{fuel}$ the engine is operated with cooled external EGR from the long route system. The EGR percentage is measured as the CO_2 ratio between the exhaust and the intake manifold. In Figure 18 the needed EGR is used to suppress NOx emissions, it is also used to decrease throttling losses over the valves at high engine speed by the increased EVC that is needed with EGR. At low engine speeds the exhaust flow to this turbine is insufficient to increase the boost pressure and therefore the air dilution is low, leading to high NOx levels and then the engine is operated stoichiometric.

The combustion phasing seen in Figure 19 is set for stable engine running without misfires. It is in most cases delayed from MBT to decrease peak pressure rise and to increase available exhaust energy to the turbine. The corresponding EVC position is plotted in Figure 20. When the EVC position is less than 55 CAD, EVO is after BDC.

The resulting nsfc is seen in Figure 21, the efficiency is best at 2000 rpm and up, at low engine speed the engine is run stoichiometric due to high NOx emissions. The increased in-cylinder heat capacity when operating stoichiometric leads to that the EVO phasing has to be advanced, further increasing the heat losses from the recompression. At this engine speed there is no gain in reducing

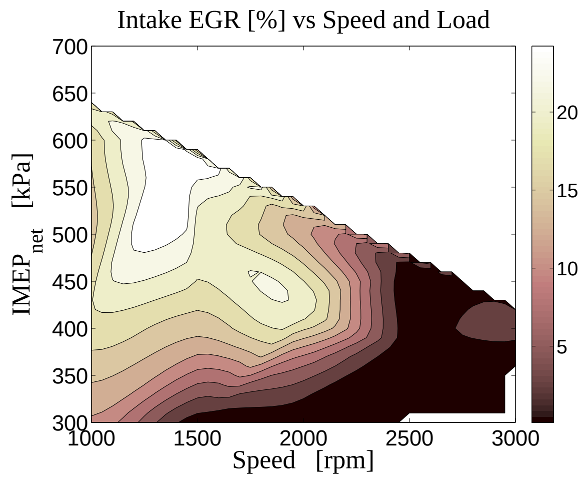

Figure 18: Speed and load testing– Intake EGR %

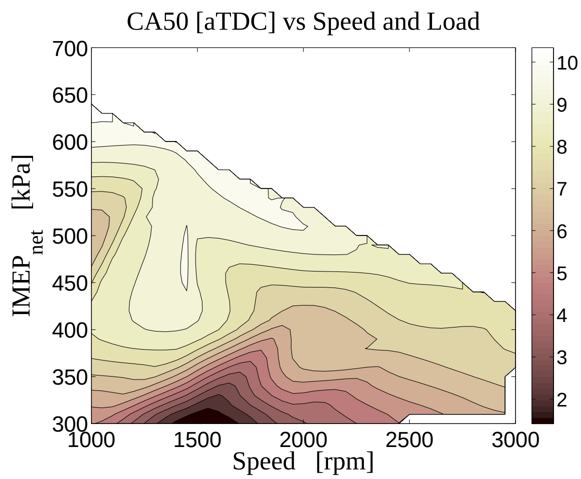

Figure 19: Speed and load testing– CA50 position

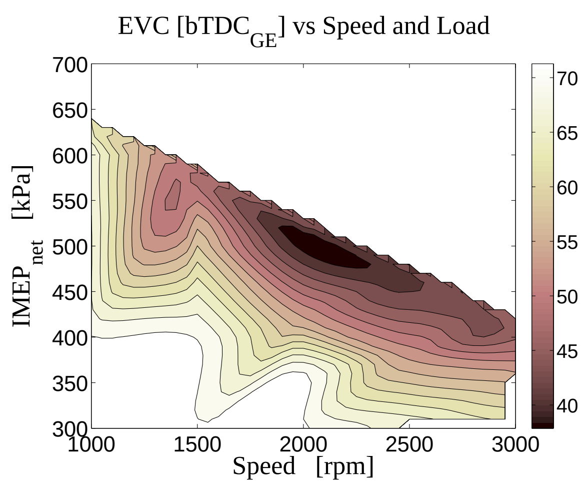

Figure 20: EVC position

pumping losses by advancing the EVC timing. At some low load points the EVC range is not enough to set the combustion phasing right and all the fuel is injected during the NVO. The testing is done so RI scales with load, meaning that there is room for improvement in nsfc at the cost of a higher RI.

The high pressure rise rate is the main limiting factor during these tests and is presented as RI in Figure 22. The

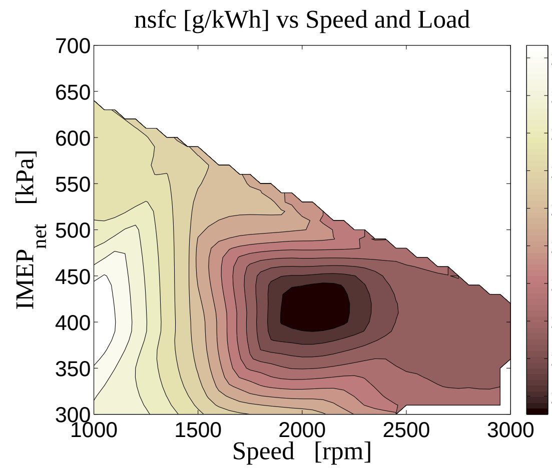

Figure 21: Speed and load testing– nsfc

maximum pressure rise in MPa/ms in Figure 23 has a similar shape as the RI. The corresponding maximum dP in bar/CAD is shown in Figure 24. The peak cylinder pressure is not a limiting factor in these tests, as seen in Figure 25.

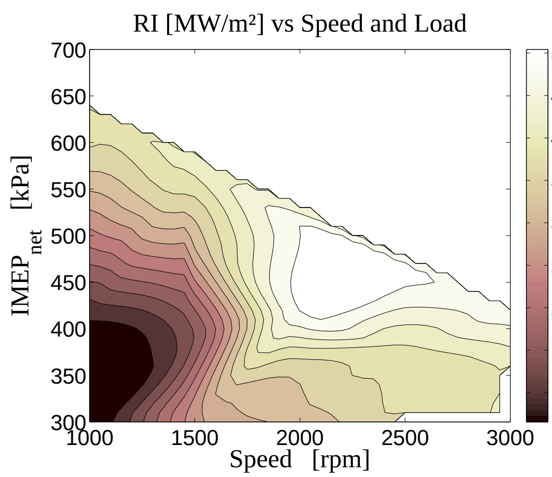

Figure 22: Speed and load testing– RI

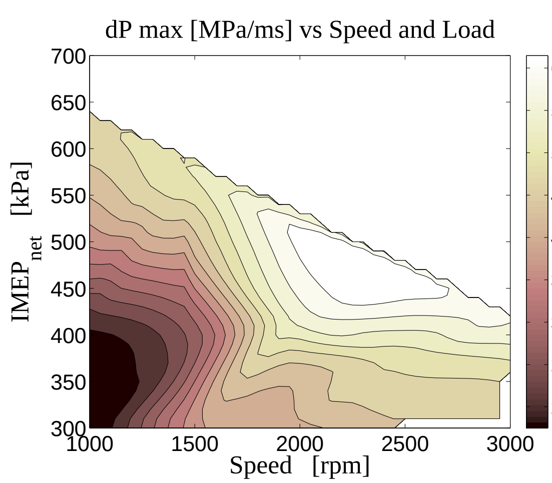

Figure 23: Speed and load testing– dP in Mpa/ms

In the highest speed range the CoV is a major factor against a late combustion timing even if is not obvious by

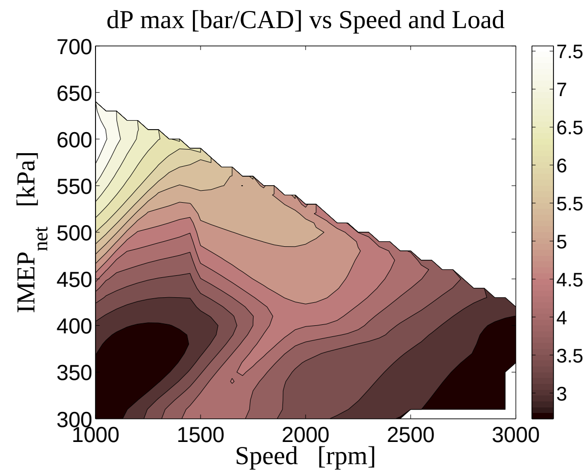

Figure 24: Speed and load testing– dP in bar/CAD

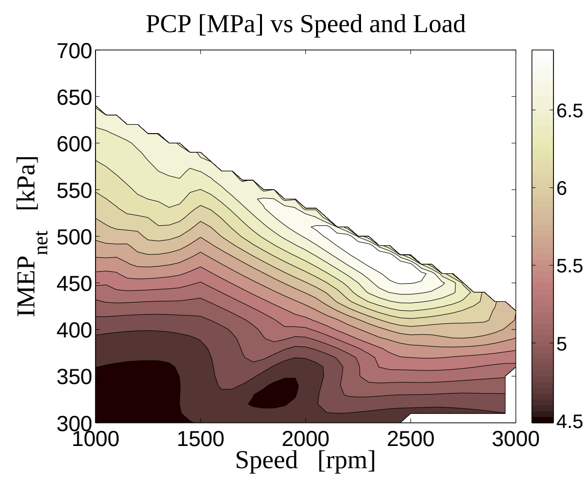

Figure 25: Speed and load testing– Peak cylinder pressure

looking at the Figure 26, here a tiny change of CA50 can drastically increase the CoV. The CoV is quite low except at 1500rpm/ 300 kPa IMEP$_{net}$, where both fuel injections had to be advanced into the NVO to get desired combustion phasing. At Figure 27 the CA10 to CA90 duration in that test point is short so here the cycle to cycle variations is high with all the fuel injected in the NVO. The area with longer CA10 to CA90 duration coincides also where the external EGR is used.

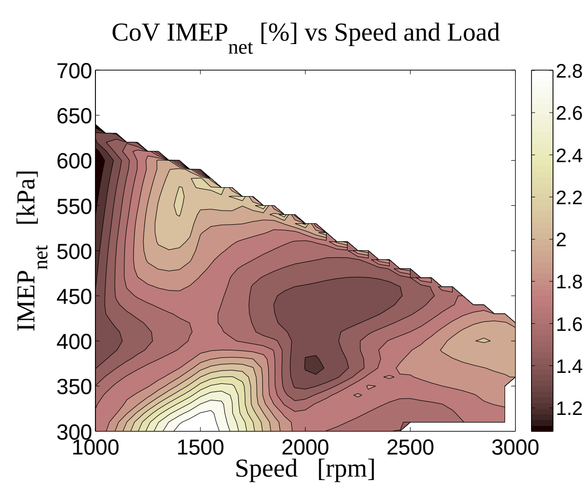

Figure 26: Speed and load testing– CoV of IMEP$_{net}$

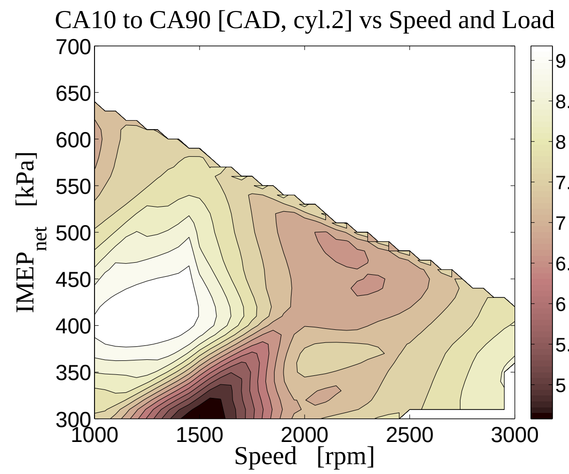

Figure 27: Speed and load testing– CA10 to CA90 duration

At high load and low engine speed the NOx emission as seen in Figure 28 increases and the engine is operated stoichiometric, see Figure 29, with cooled EGR from the long route system. The soot level in Figure 30 limited the possible load at 1000 rpm even if the fuel pressure here was increased from 12 MPa to 16 MPa.

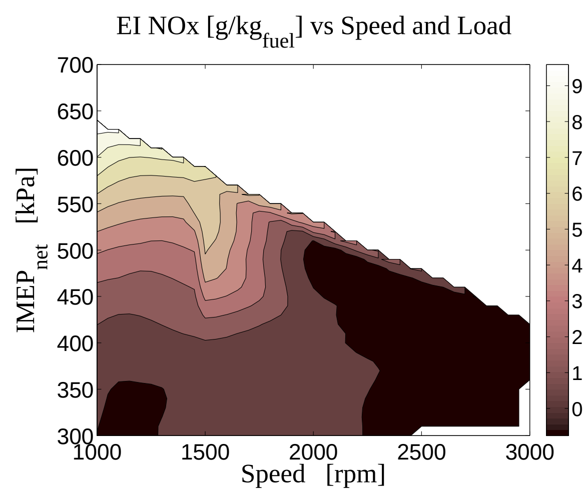

Figure 28: Speed and load testing– NOx emissions

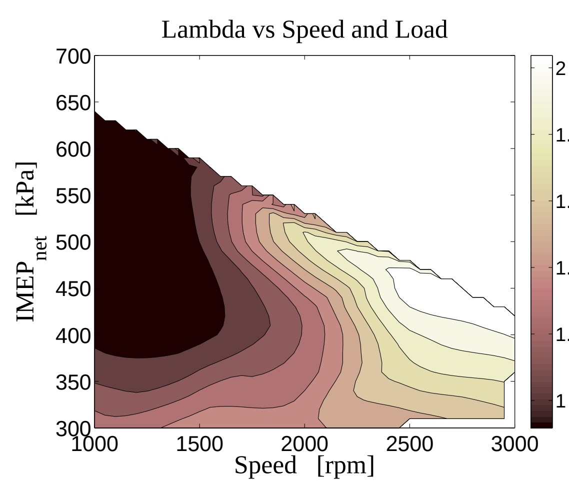

Figure 29: Speed and load testing– Lambda

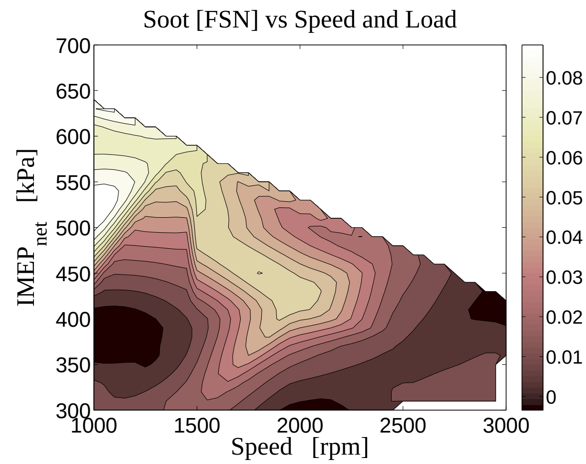

Figure 30: Speed and load testing– Soot

SPEED AND LOAD TESTING, EFFICIENCY BREAK-DOWN The engine efficiency can be divided into small fractions as seen in Figure 31. The combustion efficiency is defined as the relationship between heat released (Q_{hr}MEP) and normalized fuel energy/ cycle (FuelMEP). The thermodynamic efficiency is then the relationship between IMEP$_{gross}$ and Q_{hr}MEP. Gas exchange efficiency is defined as the relationship between IMEP$_{gross}$ and IMEP$_{net}$. Indicated efficiency is the relationship between IMEP$_{net}$ and FuelMEP.

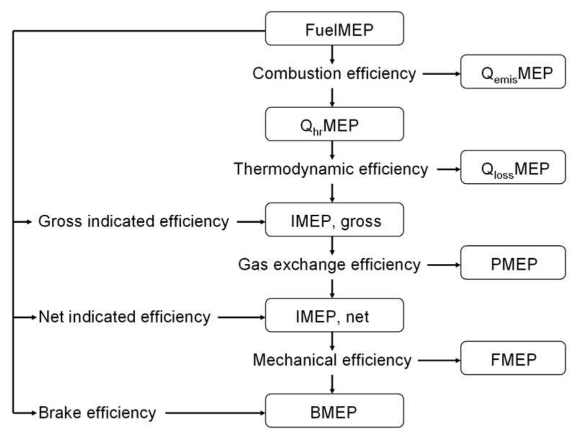

Figure 31: Engine efficiency break-down

The combustion efficiency in Figure 32 is stable throughout the speed and load test. The thermal efficiency in Figure 33 on the other hand show the increasing heat losses at low engine speed. Here the engine is operated stoichiometric which increases the recompression losses when the EVC timing has to be advanced due to the higher heat capacity from the increased external EGR.

Looking at the gas exchange efficiency in Figure 34, this turbocharged engine set-up can be improved on. By separating the pumping losses in two parts the reason to the low efficiency can be seen:
a. The pressure loss over turbine/compressor: $P_{ex} - P_{in}$ in Figure 35.
b. The pressure loss over inlet and exhaust valves: valve throttling in Figure 36.
With a turbocharger one usually encounters higher back

pressure than boost pressure, here the turbocharger is a bit large and therefore the VGT position has to be controlled for highest possible boost. The high throttling losses originates from the short duration exhaust camshaft, since late EVC is needed at high HCCI load the EVO can be close to BDC, leading to increased pumping work from the blow-down process. The resulting indicated efficiency is seen in Figure 37 for this turbocharged engine.

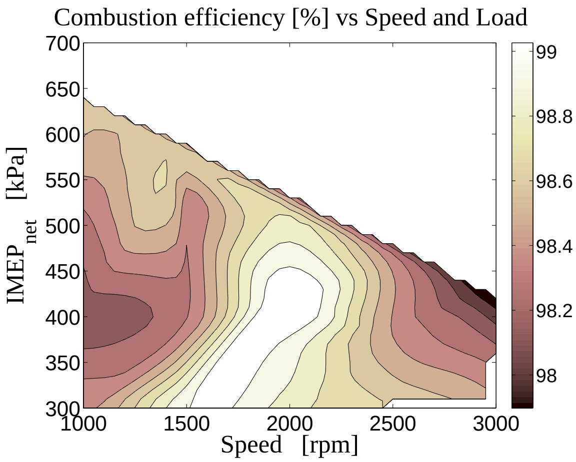

Figure 32: Combustion efficiency

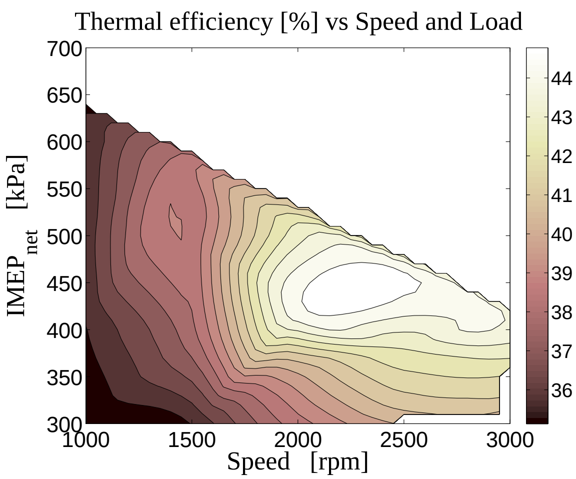

Figure 33: Thermal efficiency

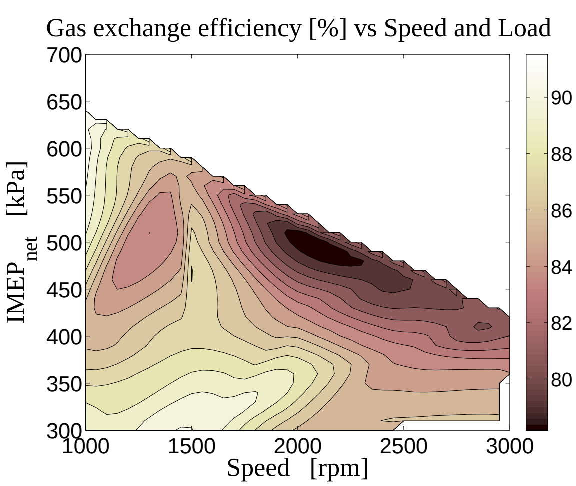

Figure 34: Gas exchange efficiency

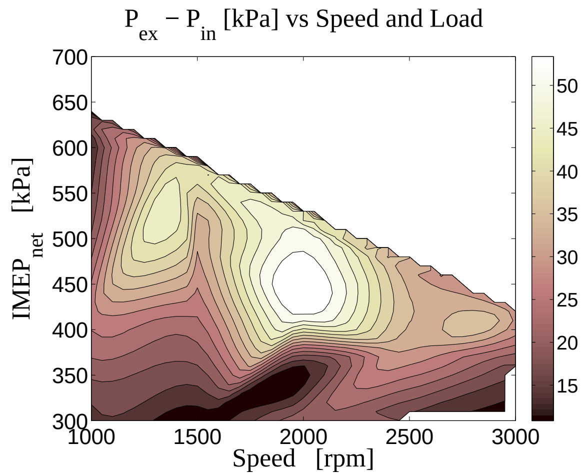

Figure 35: Pressure loss over turbine/compressor: $P_{ex} - P_{in}$

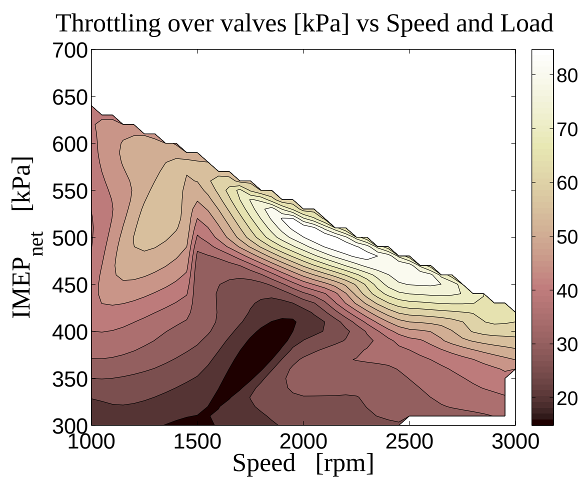

Figure 36: Pressure loss over inlet and exhaust valves: valve throttling

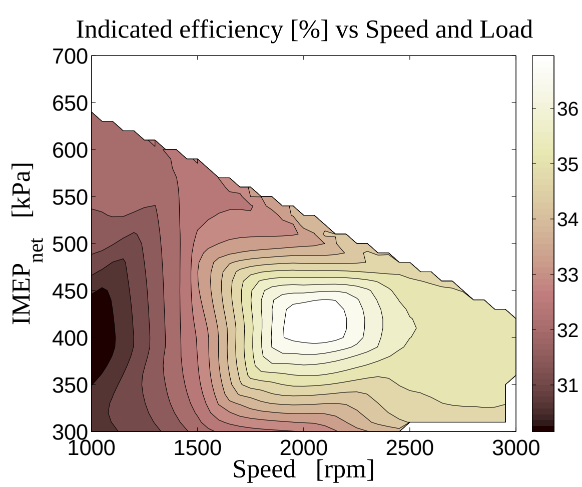

Figure 37: Indicated efficiency

DISCUSSION

When load is increased by this turbocharged HCCI engine the operating parameters all the time has to be balanced against each other. For example if we need to increase the load range where the pressure rise rate is too high the CA50 can be delayed if the CoV is stays low, the back pressure goes up from the delayed combustion timing and therefore the intake pressure and temperature is increased. The camshaft timing has to be changed to keep CA50 position meaning a later EVC is needed and therefore EVO comes later and there is risk for increasing throttling losses with these short duration camshafts. If the CA50 is delayed from MBT there will also be a loss in efficiency. On the other hand if we compare to a NA HCCI engine, we can operate on a combustion timing closer to MBT at high load with this boosted HCCI engine for the same RI number, leading to improved efficiency with right turbocharger sizing.

Finally in Figure 38 there is a comparison of possible load range between our turbocharged set-up and a naturally aspirated HCCI set-up; with an OEM style exhaust manifold. The limitations are the same and the control strategy and operating is the same, by turbocharging the possible load range is increased in the whole speed range.

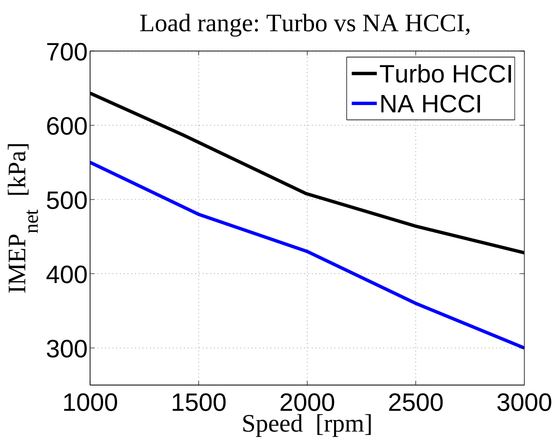

Figure 38: Maximum load: turbocharged vs. NA HCCI

CONCLUSIONS

- The pressure rise induced combustion noise is reduced with intake boost, leading to an increase of the capable HCCI operating range.
- The peak cylinder pressure can be a limiting factor when the boost pressure is high.
- Combustion phasing with cylinder balancing of CA50 with split injection together with cylinder balancing of IMEP$_{net}$ improve engine operating and maximize the load range.
- By delaying the combustion phasing to decrease the pressure rise rate there is risk of high CoV of IMEP$_{net}$ and the CoV has to be kept on a low level to suppress the possibility of misfire.
- Fuel pressure and injection timing control soot formation, and liquid film on the piston top should be avoided.
- When operating with a normal three-way catalyst the NOx emission at high load leads to that the engine must be operated stoichiometric with external EGR. Due to the slow response in the EGR system the air dilution by higher boost pressure should be preferred to suppress NOx formation.
- There is scope to increase the efficiency with better turbocharger sizing and valve timings that is better suited for this turbocharged HCCI engine.

ACKNOWLEDGMENTS

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DEFINITIONS AND ABBREVIATIONS

abs.:	Absolute
BDC:	Bottom Dead Center
BMEP:	Brake Mean Effevtive 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
EVO:	Exhaust Valve Open
FEA:	Finite Element Analysis
FMEP:	Friction Mean Effevtive Pressure
FSN:	Filter Smoke Number
FuelMEP:	Fuel Mean Effective Pressure
HCCI:	Homogeneous Charge Compression Ignition
IMEP$_{gross}$:	Indicated Mean Effective Pressure, gross
IMEP$_{net}$:	Indicated Mean Effective Pressure, net
IVC:	Intake Valve Closing
IVO:	Intake Valve Opening
NA:	Naturally Aspirated
NOx:	Nitrogen Oxide
nsfc:	Net Specific Fuel Consumption
NVO:	Negative Valve Overlap
MBT:	Maximum Brake Torque
OEM:	Original Equipment Manufacturer
P ex:	Pressure Exhaust
P in:	Pressure Intake
PCP:	Peak Cylinder Pressure
PMEP:	Pumping Mean Effevtive Pressure
Q$_{emis}$MEP:	Emission Mean Effevtive Pressure
Q$_{hr}$MEP:	Heat Release Mean Effevtive Pressure
Q$_{loss}$MEP:	Heat losses Mean Effevtive Pressure
RON:	Research Octane Number
RI:	Ringing Intensity
SA:	Spark Assist
SI:	Spark Ignition
SOI:	Start of Injection
TDC:	Top Dead Center
TDC$_f$:	Top Dead Center, firing
TDC$_{GE}$:	Top Dead Center, gas exchange
TWC:	Three-Way Catalyst
VGT:	Variable Geometry Turbine
VVT:	Variable Valve Timing