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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 examed 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 (NOx) 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 2.2*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 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^o. 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.

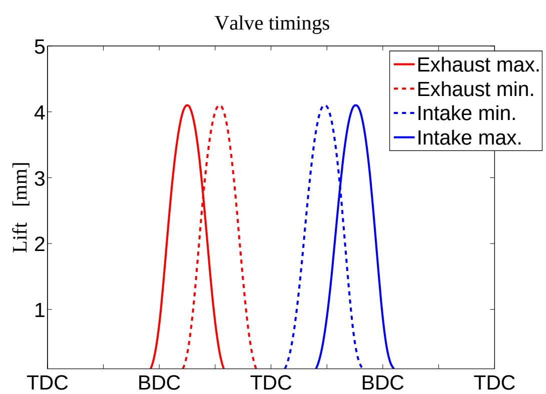

Figure 1: Valve timing range and setup

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 balancing 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.

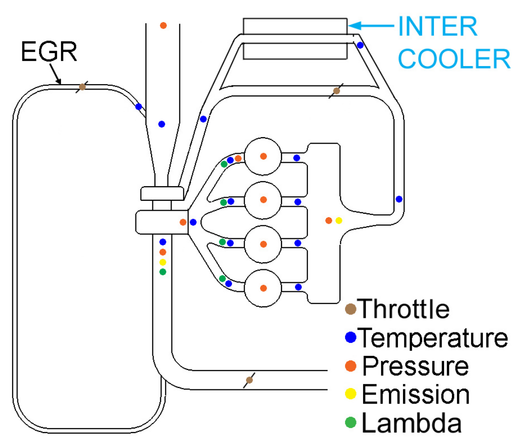

Figure 4: Engine layout

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 $\pm$ 3 kPa and in engine speed $\pm$ 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/m^2 [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 $\sim 7^o$ 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.

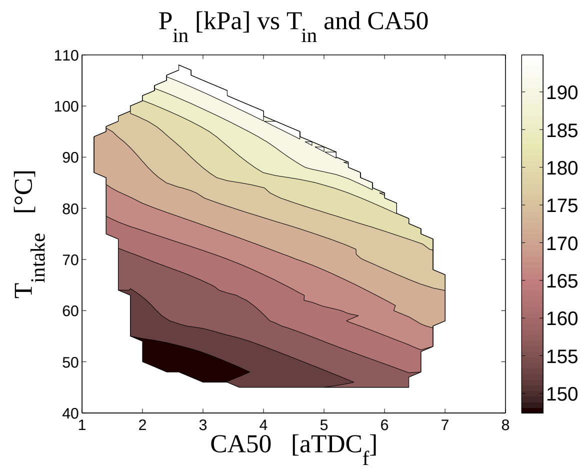

Figure 5: Intake pressure, absolute

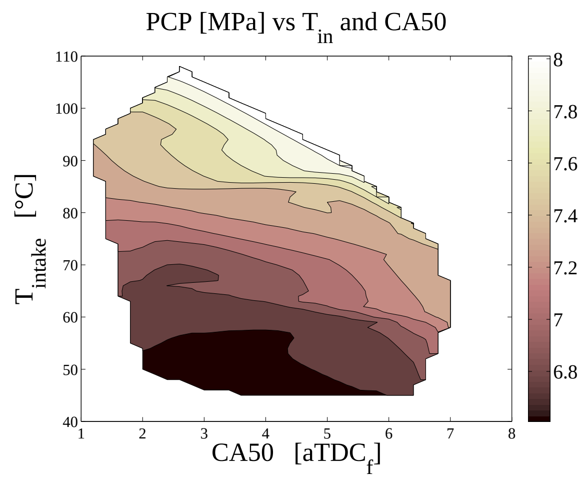

Figure 6: Peak cylinder pressure

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^o, 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.

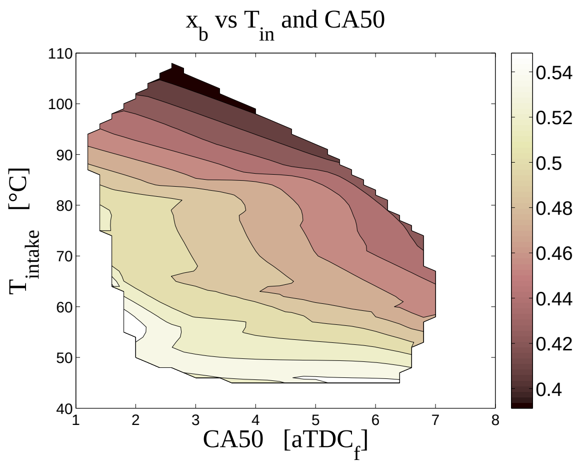

Figure 7: Burned gas fraction, x_b

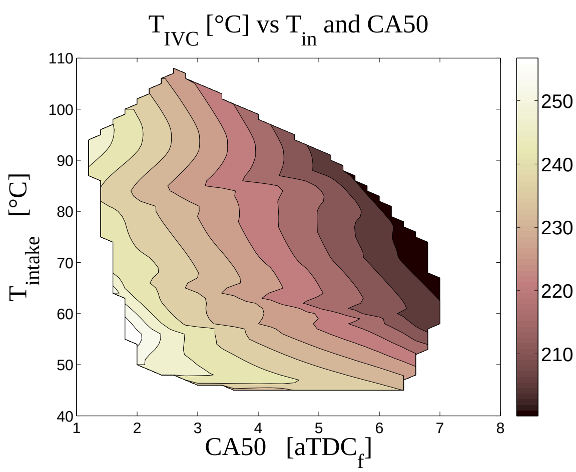

Figure 8: Temperature at IVC

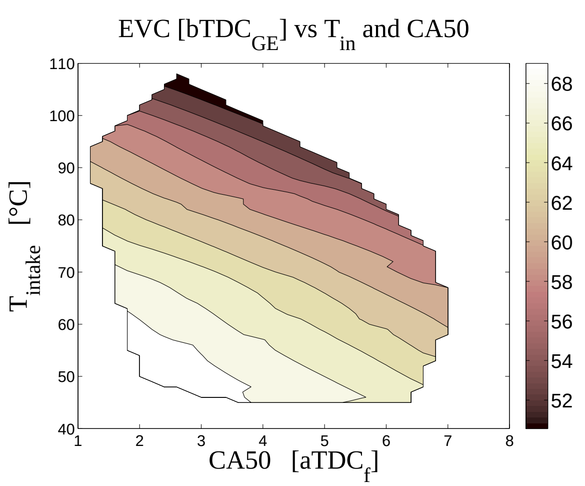

Figure 9: EVC position

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= Throttling$_{valve}$ + $(P_{ex} - P_{in})$

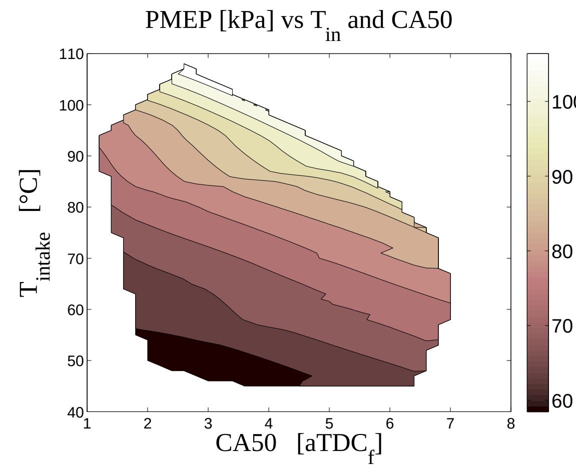

Figure 10: Pumping losses

$P_{ex} - P_{in}$: 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^o (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.

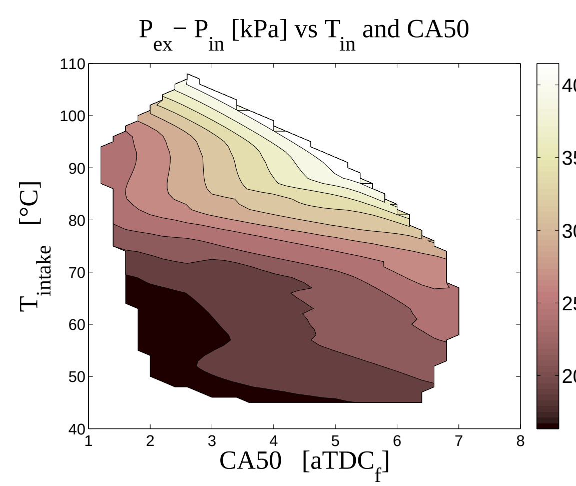

Figure 11: Pressure loss over turbine/ compressor

To decrease the engine noise the combustion timing can be delayed but this can increases 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

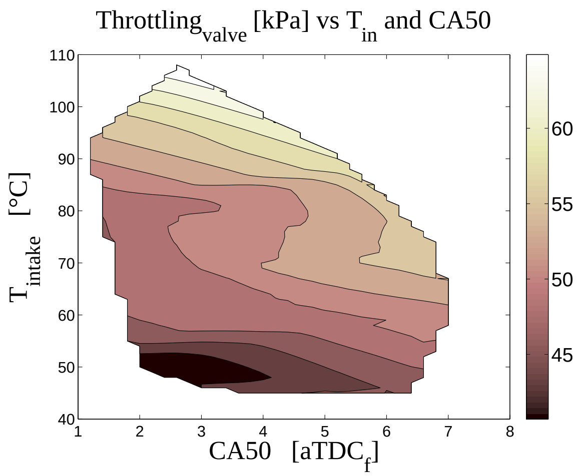

Figure 12: Valve throttling losses

combustion stability. But the STD of CA50 reveals that there is an issue when the intake temperature is low as well. This is also noticeable when listening to the engine running. It has a more ruff engine sound at low intake temperatures. At the lowest temperatures the burned gas fraction, as seen in Figure 7, is highest and the dilution from boost pressure is lowest, making the influence from small variations in T_{IVC} more pronounced, see Figure 8.

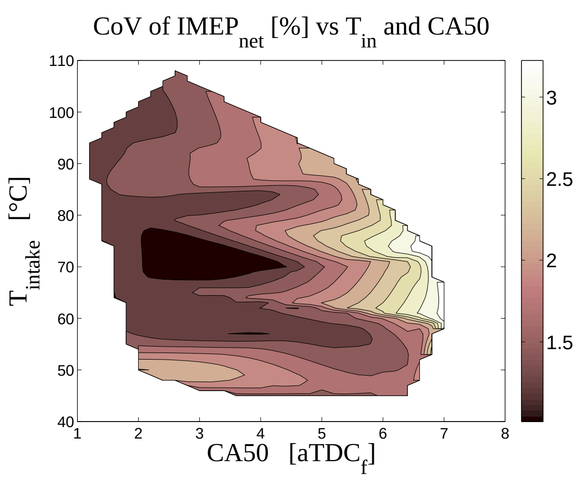

Figure 13: CoV of $IMEP_{net}$

The high pressure rise rate is the main limiting factor when the load is increased. In Figure 15 it can be seen that the CA50 position is the major control variable for the pressure rise rate. Recall that the intake pressure (Figure 5) increases with later CA50 position and high intake temperature. The pressure rise rate scale to the amount of dilution but the effect of this is masked to some extent because there is a need to burn more fuel at constant load when the pumping losses increases.

In Figure 16 the combustion duration for CA10 to CA90 is shown. The CA50 timing is the most important factor but there is some influence from the increased dilution with higher boost pressures as well. The residual gas (Figure 7) level has no direct influence on the combustion duration

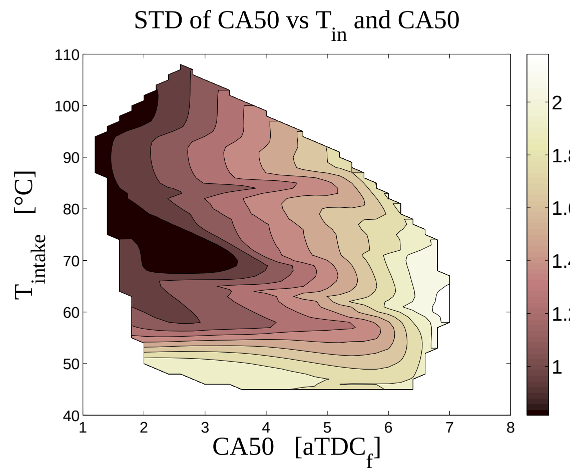

Figure 14: Standard deviation of CA50

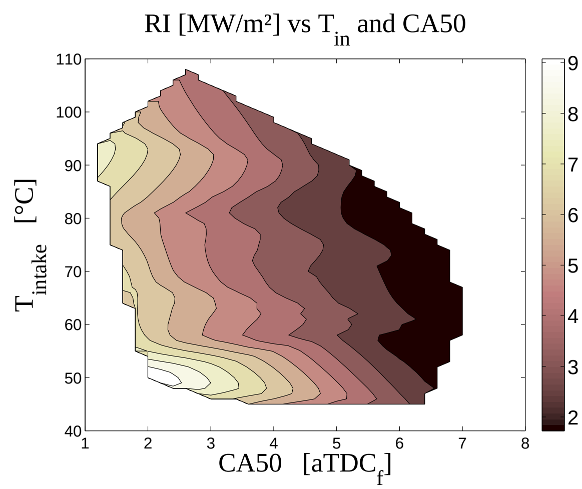

Figure 15: Ringing Intensity

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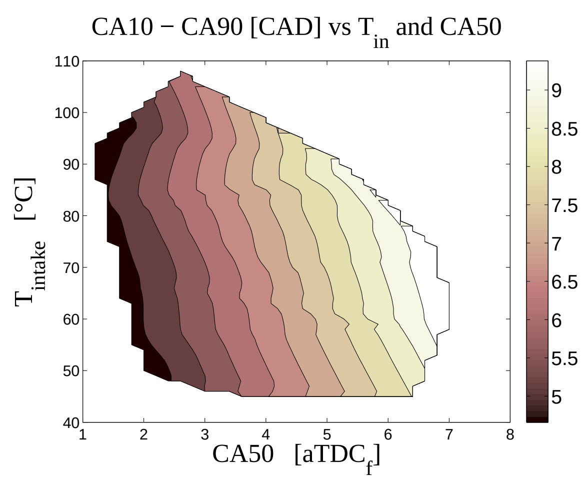

Figure 16: Combustion duration, CA10 to CA90

The NOx emissions can be an issue when the load is increased in HCCI mode but the NOx level at this engine speed and load is very low as seen in Figure 17. A later combustion timing decreases the combustion temperature with further reduction in the NOx level. The increased dilution from increased intake pressure is also a reason for the low NOx levels. In Figure 18 the corresponding lambda is plotted. Normally when the NOx level is increased above a set limit [16] the lambda level is used to operate the engine stoichiometric on cylinder to cylinder basis.

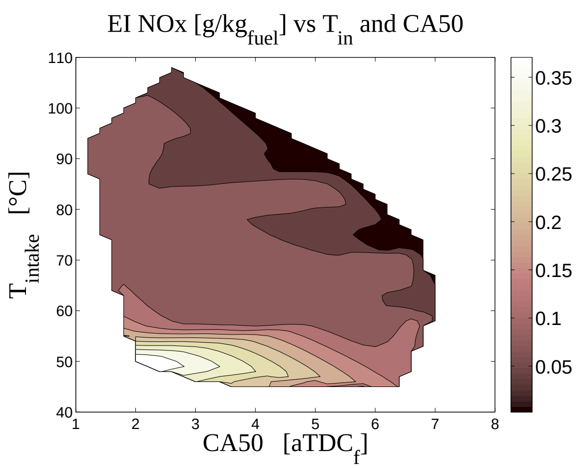

Figure 17: NOx emissions

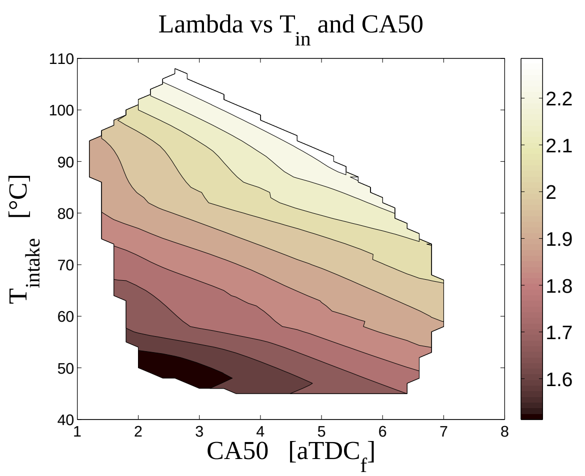

Figure 18: Lambda

The engine total efficiency can be divided into fractions [21] as seen in Figure 19. 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.

By using suitable injection timings the combustion efficiency in Figure 20 is high at this test point. The main deviation is from the combustion timing. The thermal efficiency in Figure 21 on the other hand show its highest efficiency at low intake temperatures where the mass flow is low. At this low intake temperature there is a need for advanced exhaust valve timing. The heat losses during the recompression phase can be an issue but this is more

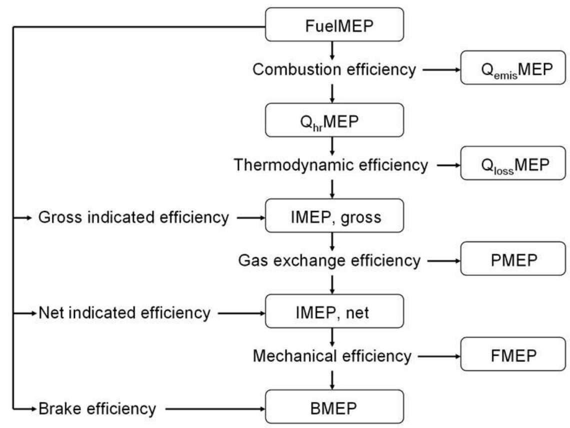

Figure 19: Engine efficency break-down

pronounced at low engine speeds. There is also an increase in thermal efficiency around $80^\circ\ C$. The gas exchange efficiency in Figure 22 scales to the intake pressure and originates from the turbocharger performance (Figure 11) and throttling losses (Figure 12) associated with these type of short duration valve lifts at increased load. The indicated efficiency Figure 23 is then the sum of the combustion, thermal and gas exchange efficiency. It is apparent that the intake temperature has strong influence on total efficiency where the operating temperature has to be balanced against combustion noise and stability.

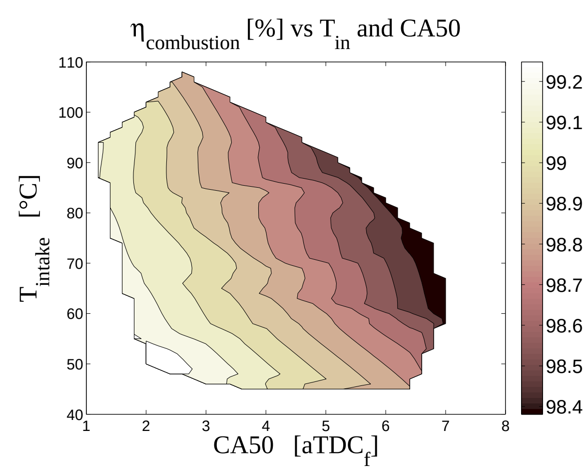

Figure 20: Combustion efficiency

The next test is on how external EGR influences engine operation. The EGR fraction is measured as the CO_2 ratio between the exhaust and the intake manifold.

EGR AND INTAKE TEMPERATURE SWEEP AT 2250 RPM AND 400 KPA IMEP$_{NET}$ The combustion timing during these tests has a CA50 position at 4^o (aTDC$_f$) except at the highest temperatures where no more than 3^o (aTDC$_f$) could be used due to increasing back pressure as described earlier. From this test there are some additional graphs that can be found in the APPENDIX.

In Figure 24 the available boost pressure decreases with

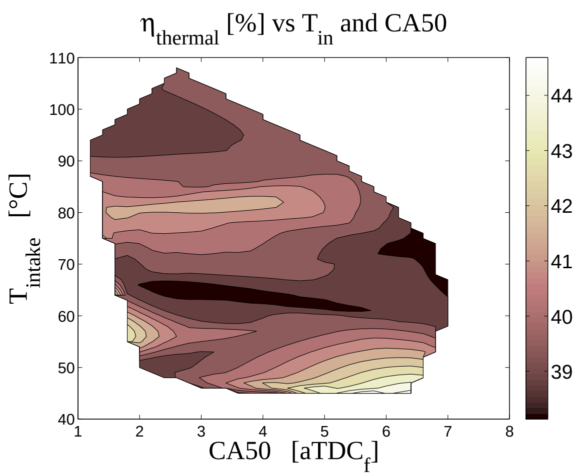

Figure 21: Thermal efficiency

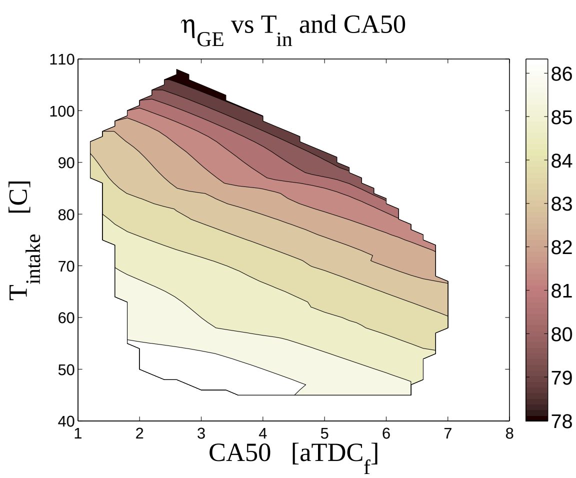

Figure 22: Gas exchange efficiency

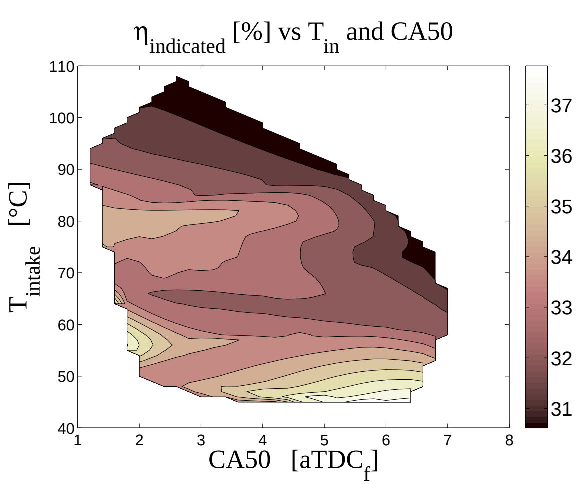

Figure 23: Indicated efficiency

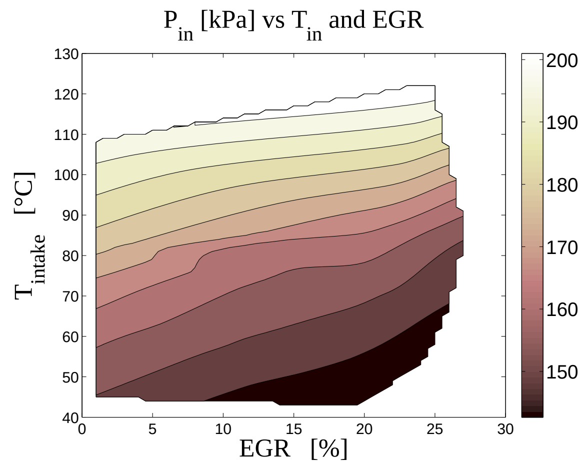

Figure 24: EGR sweep, Intake pressure

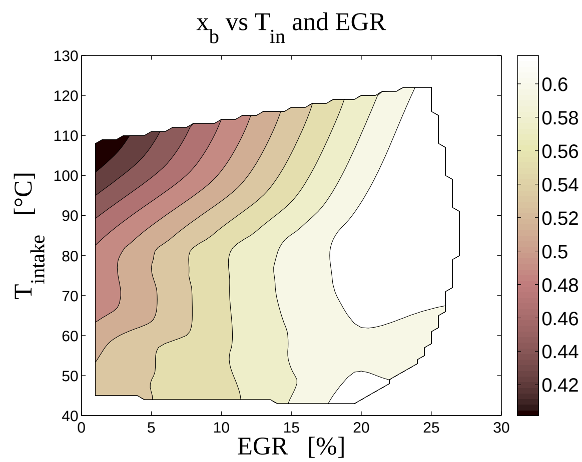

Figure 25: EGR sweep, Burned gas fraction

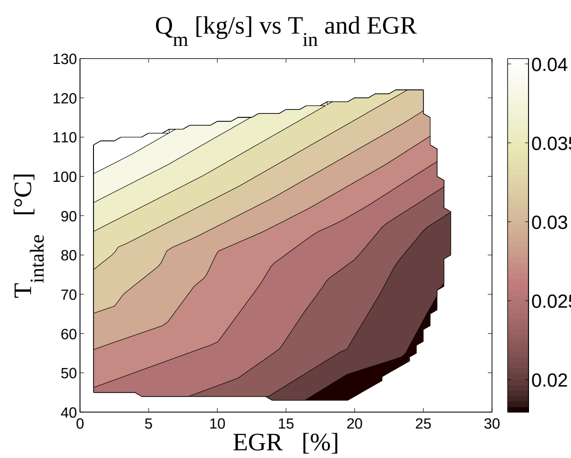

Figure 26: EGR sweep, Mass flow

increasing EGR levels. This is a function of the increased burned gas fraction, see Figure 25, where the charge will have an increased heat capacity. This leads to a decreased mass flow rate for a given combustion timing as seen in Figure 26.

The pumping losses are divided in two parts as described earlier. In Figure 27 the influence of increased EGR show some variation in turbocharger performance at lower intake temperatures. The largest decrease in throttling losses occur at low intake temperatures and high EGR levels, see Figure 28. Here the mass flow is reduced which suites current valve timings.

The combustion stability seems to have an optimum around an intake temperature of $60^\circ\ C$ when looking at the CoV of IMEP$_{net}$ in Figure 29. On the other hand, if we look at the standard deviation of CA50 in Figure 30 the picture broadens. Here an increase of EGR leads to less combustion fluctuations except at low intake temper-

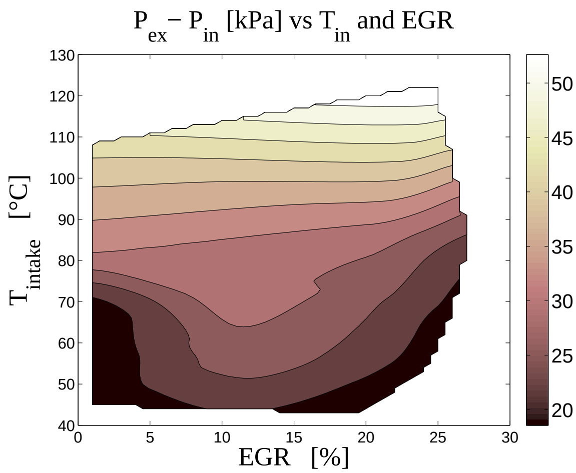

Figure 27: EGR sweep, Pressure losses over turbine–compressor

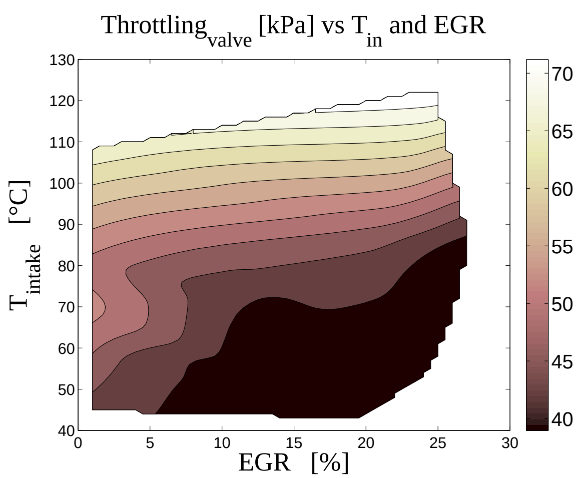

Figure 28: EGR sweep, Throttling losses

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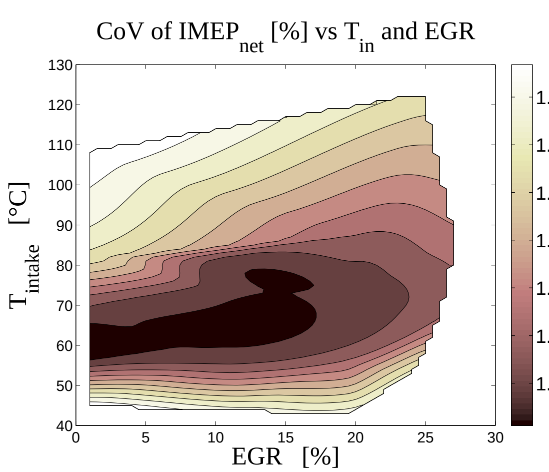

Figure 29: EGR sweep, CoV of IMEP$_{net}$

Since the combustion noise, quantified here with RI, is of major interest during turbocharged HCCI operation, it is seen that increased external EGR has different effects on RI depending on intake temperature, see Figure 31. At higher intake temperatures the RI level is stable even if the intake pressure is decreasing at the same time, see Fig-

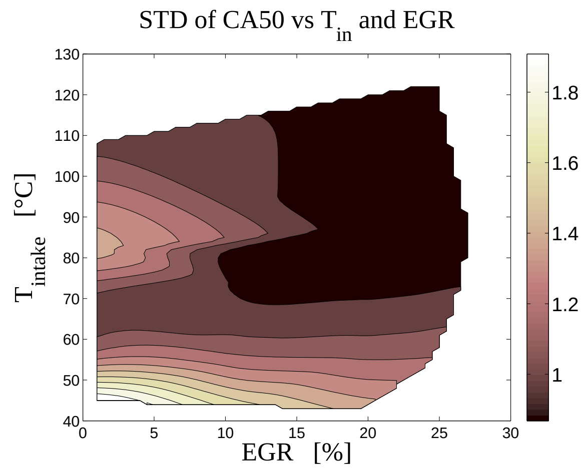

Figure 30: EGR sweep, Standard deviation of CA50

ure 24. At lower temperatures RI is increasing with higher EGR levels. The corresponding combustion duration in Figure 32 reflects this behavior. There is no clear connection between the combustion duration and the burned gas fraction shown in Figure 25. In normal engine operating an increase of EGR level will delay the combustion timing due to increased in-cylinder heat capacity, but here the combustion timing is kept constant. In Figure 16 it could be seen that the combustion timing is the main parameter for the combustion duration.

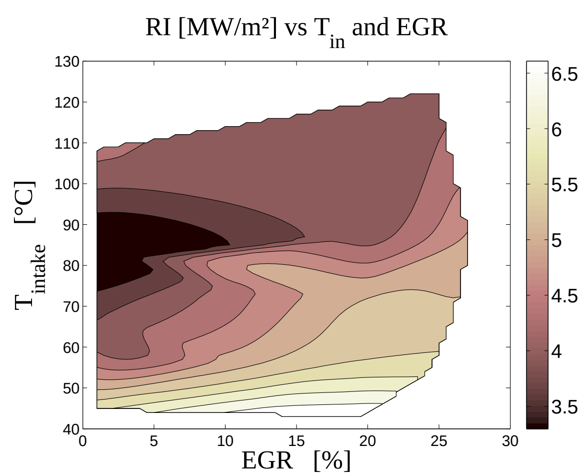

Figure 31: EGR sweep, Ringing Intensity

Finally the indicated efficiency in Figure 33 shows no real gain to operate with external EGR at this load and speed point (and these CA50 timings) except at the point with low intake temperature and high EGR level. But recall that the RI level and the cycle to cycle variations are high here.

Since the combustion timing is fixed during these tests the question rises of what happens if the combustion timing is swept with and without external EGR and the intake temperature is kept constant.

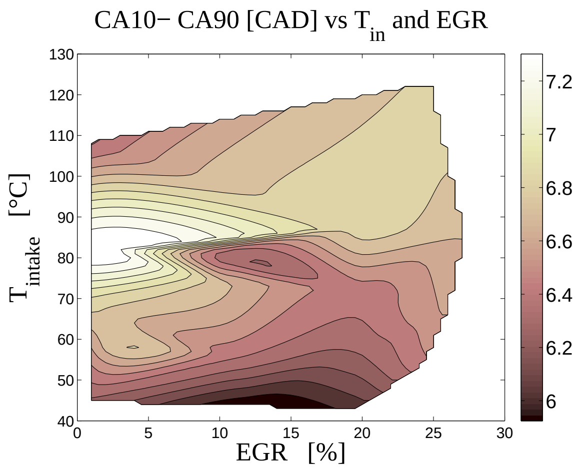

Figure 32: EGR sweep, Combustion duration, CA10 to CA90

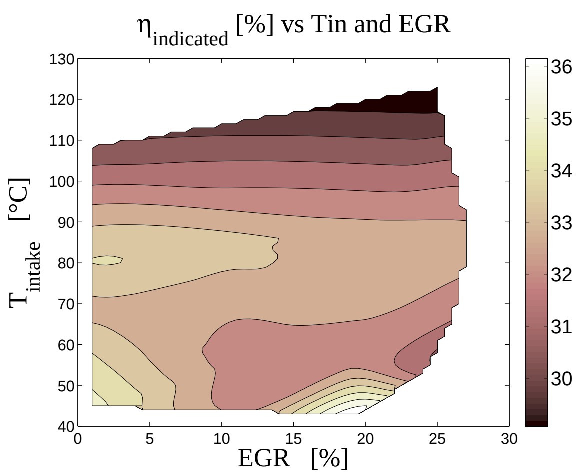

Figure 33: EGR sweep, Indicated efficiency

CA50 SWEEP WITH AND WITHOUT EGR AT 2250 RPM AND 400 KPA IMEP$_{NET}$ To further expand this load and speed point, a CA50 sweep is done with and without EGR at an intake temperature of $60°\ C$. This intake temperature was chosen since it has acceptable RI levels, low cycle to cycle variations, a broad combustion timing range and rather high total efficiency. The EGR level is set around 19%, this range of positive displacement could be reached without any increase in backpressure. In Figure 34 the boost pressure is decreasing with EGR and so is also the throttling losses. The pressure differential between the intake and exhaust ($P_{ex} - P_{in}$) is not really affected by the increased EGR here.

In Figure 35 the CoV of IMEP is reduced with EGR at late CA50 timings, this trend is even more pronounced when looking at the STD of CA50. Notable is also that the operating range is wider with EGR. The combustion duration, CA10 to CA90, shows no real influence of EGR.

The RI is higher at early CA50 positions when using EGR which reflects the relative lower dilution level, see Figure 36. The other pressure rise expressions, dP/CAD and dP/ms, also reflect this at a smaller scale.

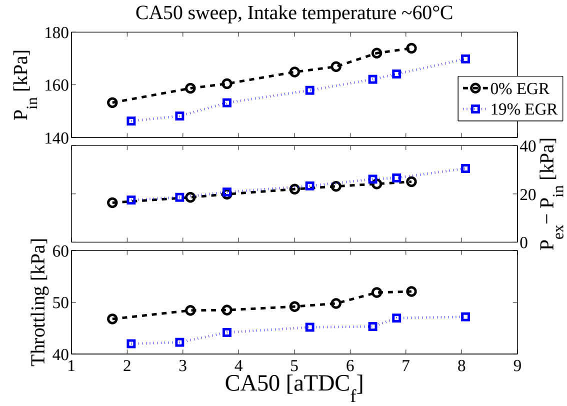

Figure 34: CA50 sweep, with and without EGR

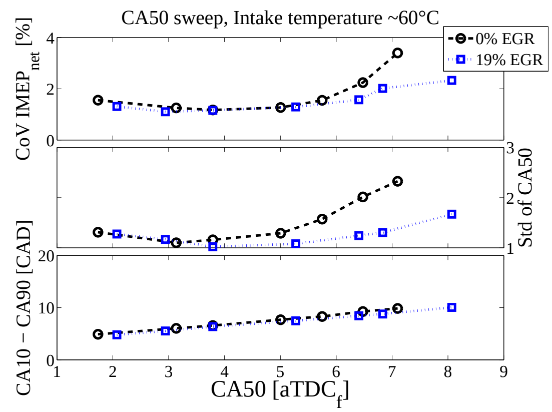

Figure 35: CA50 sweep, with and without EGR

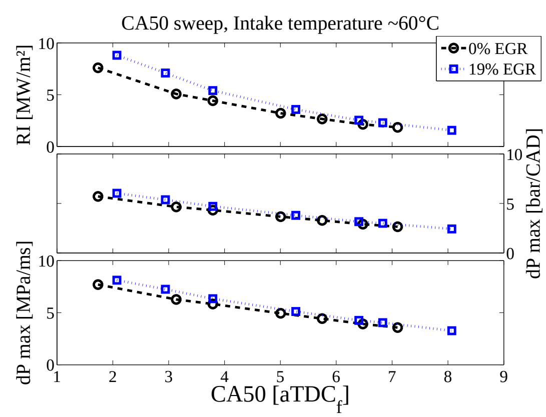

Figure 36: CA50 sweep, with and without EGR

In Figure 37 the combustion efficiency is higher with EGR but this effect is counteracted at early CA50 timings where the thermal efficiency is lower. The rise in thermal efficiencies at late CA50 timings reflects the more stable combustion position with EGR which has less cyclic fluctuations than without EGR. The gas exchange efficiency is improved slightly with EGR due to reduced throttling losses as shown in Figure 34. Finally the indicated efficiency shows that it can be beneficial to operate with EGR due to better combustion stability at late combustion timings.

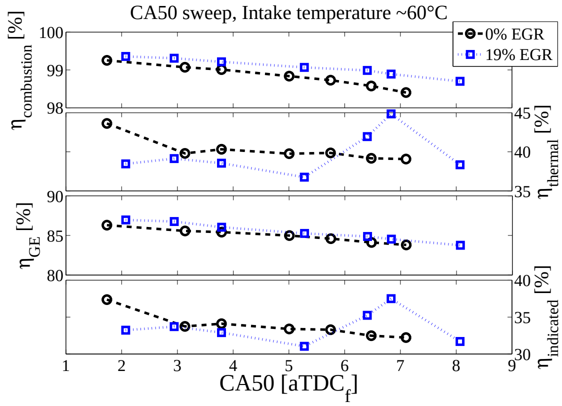

Figure 37: CA50 sweep, with and without EGR

DISCUSSION

When operating a turbocharged HCCI engine with NVO, the extra freedom to adjust the intake temperature is attractive from a combustion perspective where one can operate the engine at the most beneficial temperature. For example if the pressure rise rate is high we want a high intake pressure. Choosing a higher intake temperature leads to higher intake pressure and the combustion noise can be reduced. However, this has to be weighted against the increased pumping losses with reduction in efficiency. Too high intake pressure can result in loss of turbocharger performance and this was the case at the highest intake temperatures where the increasing backpressure limited the available combustion timing range. This can be circumvented by appropriate intake valve timing which is constrained in these tests. A shrinking combustion timing range puts even higher demands on the control system, especially during transients where a broad combustion timing range is wanted.

At this speed and load point, 2250 rpm and IMEP$_{net}$ 400 kPa, the best efficiency was reached with low intake temperatures but at the same time there is a loss of combustion stability. The loss of combustion stability is not really obvious when only using the CoV of IMEP$_{net}$ as a measurement. The standard deviation of CA50 visualizes the combustion stability more clearly. By using external EGR the combustion range and stability can be increased. EGR show no big influence on combustion duration and efficiency. The usage of an external EGR systems added weight, cost and control complexity, has to be considered.

The combustion timing is the major variable during these tests since it directly control the dilution level and the intake temperature level that can be reached. This gives the combustion timing a huge impact on test results in HCCI mode. This further emphasizes the importance of robust combustion timing control in turbocharged HCCI mode. The prediction of the in-cylinder charge temperature and properties at IVC can be used by the control system. Combined with a DI fuel system there is a possibility to influence the combustion timing at a late stage.

To find the optimum operating intake temperature and combustion timing one has to balance factors like pressure rise rate, efficiency, combustion stability/ range and emissions level against each other. This study is only performed at one engine speed and load point so the question is how the results are at other speed/ load points, how changes in turbocharger, valve timings etc. influences the result. This need to be investigated in future work.

CONCLUSIONS

From this test point at 2250 rpm and 400 kPa IMEP$_{net}$

- Available intake pressure scales directly with intake temperature and combustion timing, it can therefore be modulated with charge cooling and valve timings.
- An increase of intake pressure decreases peak pressure rise rate but has to be weighed against increasing pumping losses.
- The highest indicated efficiency at this study was found with the lowest intake temperature, but in practice, the high pressure rise rate and cyclic variations makes it an unsuitable working area.
- Standard deviation of the combustion phasing is a good measurement of combustion stability that can complement CoV of IMEP$_{net}$ for the cycle to cycle behavior.
- The usage of external EGR is shown to improve combustion stability and extend usable combustion timing range.
- Turbocharger performance and appropriate valve timings are vital for improving efficiency and operating range in a NVO HCCI engine.
- Test results are heavily influenced by the combustion timing.

ACKNOWLEDGMENTS

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DEFINITIONS AND ABBREVIATIONS

aTDC$_f$:	After Top Dead Center, firing
BDC:	Bottom Dead Center
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
$\eta_{combustion}$:	Efficiency, combustion
η_{GE}:	Efficiency, gas exchange
$\eta_{indicated}$:	Efficiency, indicated
$\eta_{thermal}$:	Efficiency, thermal
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
NVO:	Negative Valve Overlap
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
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, intake valve closing
VVT:	Variable Valve Timing
x$_b$:	Burned gas fraction

APPENDIX

Complementary graphs from CA50 and intake temperature sweep at 2250 rpm and 400 kPa IMEP$_{net}$

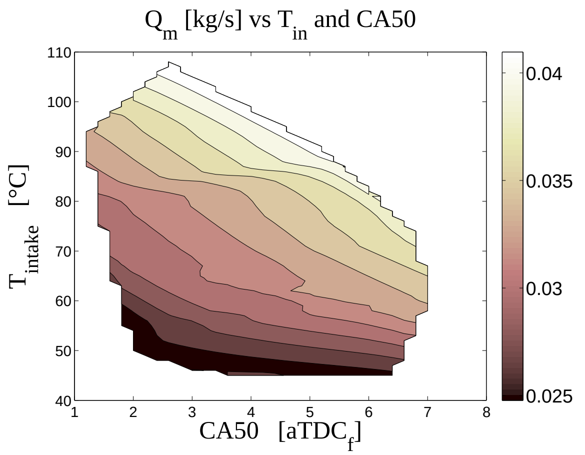

Figure 38: Mass flow

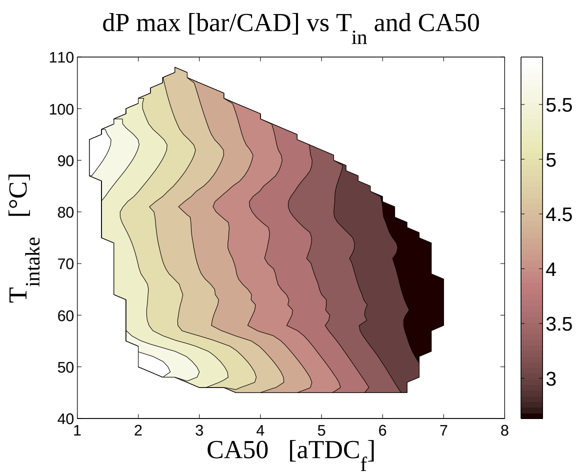

Figure 39: Pressure rise, bar/ CAD

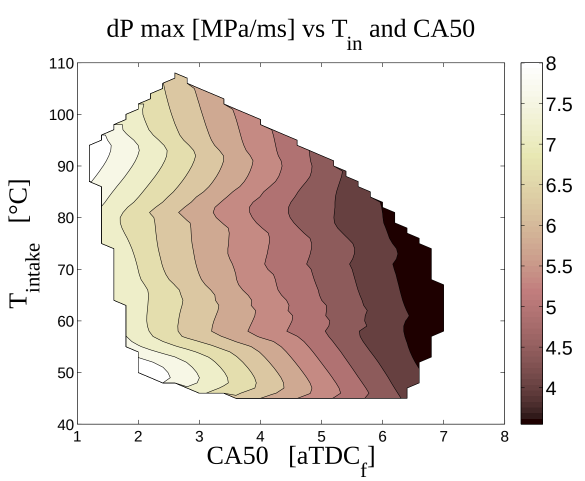

Figure 40: Pressure rise, MPa/ ms

Complementary graphs from EGR sweep at 2250 rpm and 400 kPa IMEP$_{net}$

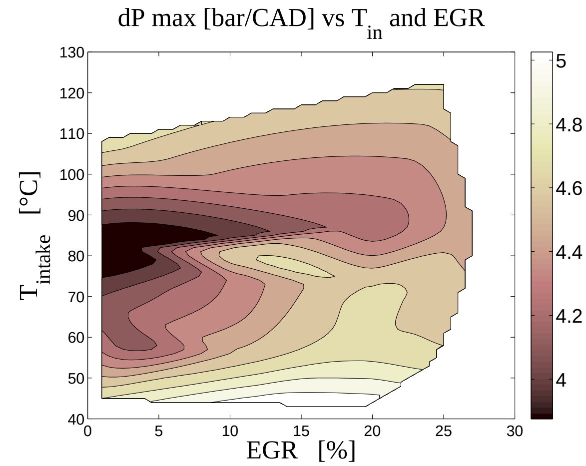

Figure 41: EGR sweep, Pressure rise dP/ CAD

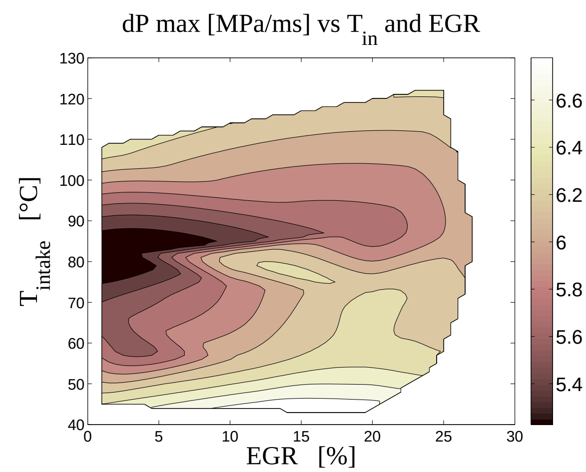

Figure 42: EGR sweep, Pressure rise dP/ ms

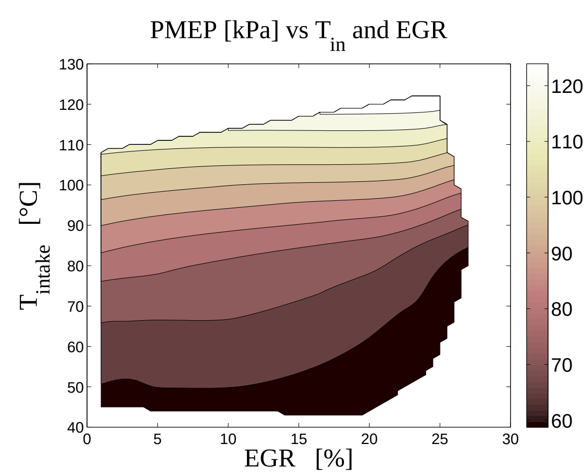

Figure 43: EGR sweep, Pumping losses

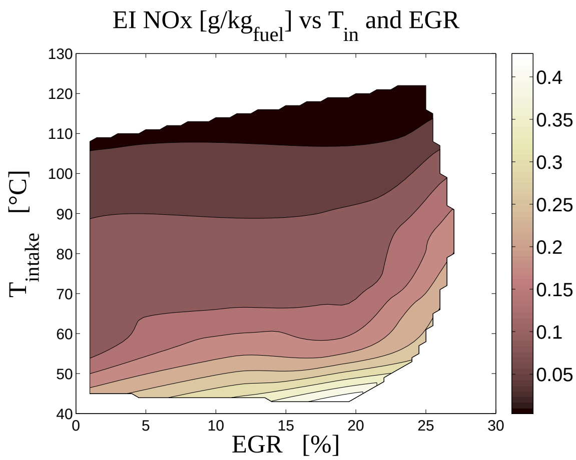

Figure 44: EGR sweep, NOx emissions

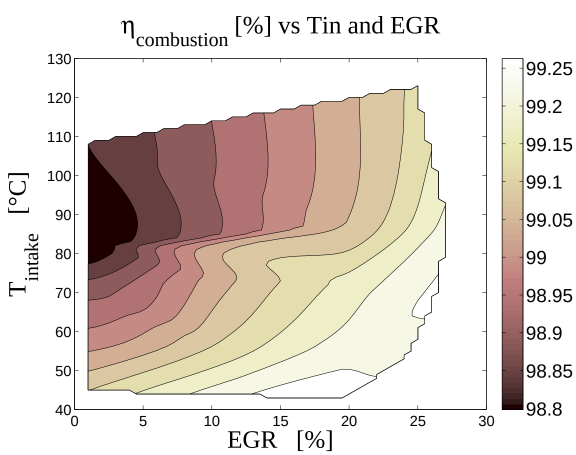

Figure 45: EGR sweep, Combustion efficiency

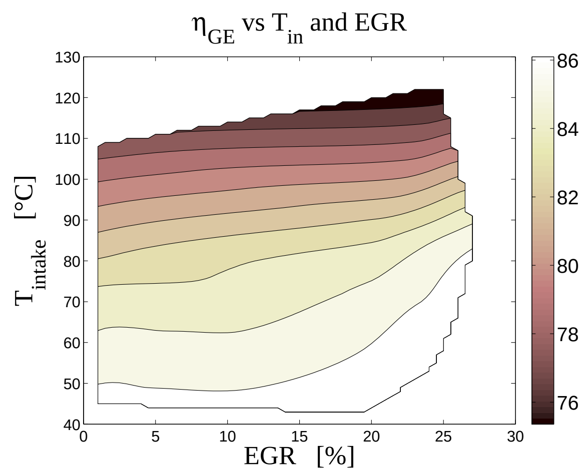

Figure 47: EGR sweep, Gas exchange efficiency

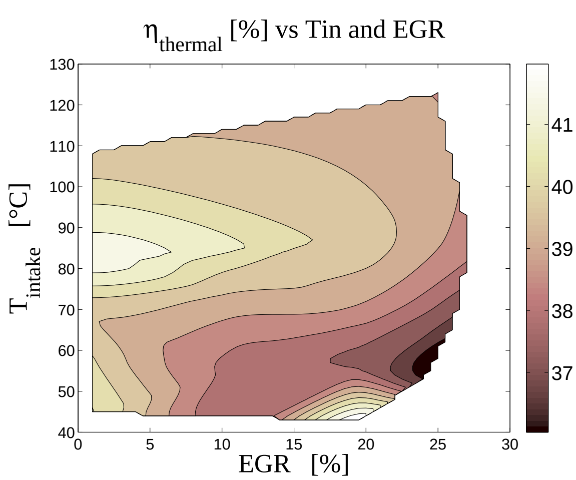

Figure 46: EGR sweep, Thermal efficiency