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Published in:
SAE Technical Papers

2007

JSAE 20077035
SAE 2007-01-1855

Operation strategy of a Dual Fuel HCCI Engine with VGT

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ABSTRACT

HCCI combustion is well known and much results regarding its special properties have been published. Publications comparing the performance of different HCCI engines and comparing HCCI engines to conventional engines have indicated special features of HCCI engines regarding, among other things, emissions, efficiency and special feedback-control requirements. This paper attempts to contribute to the common knowledge of HCCI engines by describing an operational strategy suitable for a dual-fuel port-injected Heavy Duty HCCI engine equipped with a variable geometry turbo charger. Due to the special properties of HCCI combustion a specific operational strategy has to be adopted for the engine operation parameters (in this case combustion phasing and boost pressure). The low exhaust temperature of HCCI engines limits the benefits of turbo charging and causes pumping losses which means that “the more the merrier” principle does not apply to intake pressure for HCCI engines. It is desirable not to use more boost pressure than necessary to avoid excessively rapid combustion and/or emissions of NO_x. It is also desirable to select a correct combustion phasing which, like the boost pressure, has a large influence on engine efficiency. The optimization problem that emerges between the need for boost pressure to avoid noise and emissions and, at the same time, avoiding an extensive decrease of efficiency because of pumping losses is the topic of this paper. The experiments were carried out on a 12 liter Heavy Duty Diesel engine converted to pure HCCI operation. Individually injected natural gas and n-Heptane with a nominal injection ratio of 85% natural gas and the rest n-Heptane (based on heating value) was used as fuel. The engine was under feedback combustion control during the experiments.

INTRODUCTION

Despite intense research efforts in recent years the HCCI engine is still less known than its relatives the Otto and the Diesel engine. Nevertheless it seems to be one of the most promising engine concepts of the future, combining high efficiency with ultra low emissions of nitrogen oxides. The first report of the concept of HCCI combustion was presented by Onishi et al. [1] and was primarily an attempt to reduce emissions of unburned hydrocarbons (HC) and improve part load efficiency for two stroke engines. After Onishi et al. published their results it took a while before further HCCI publications emerged. When they did the publication rate increased. Important early publications are for example [2]-[4].

The working principle of the HCCI engine is best understood as a hybrid between the Otto and Diesel engines. The HCCI engine operates with a premixed charge as the Otto engine, and it operates unthrottled with compression ignition, controlling the load with the global air/fuel equivalence ratio (λ) like the diesel engine. This operation principle makes it possible to increase the efficiency compared to the Otto engine due to the avoidance of throttling losses. At the same time the high soot and nitrogen oxide emissions of the diesel engine are avoided. Soot emission is avoided because of the homogeneity of the mixture and the absence of locally rich combustion zones. Nitrogen oxide emission is avoided because of the decreased peak in-cylinder temperature due to the diluted oper-

ation of the engine and the absence of stoichiometric undiluted zones.

HCCI combustion can be achieved in numerous ways, both in two and four stroke engines. High compression ratio can be applied [5], the inlet air can be pre heated [6]. HCCI combustion can be induced by unconventional valve strategies that retain hot residuals [7] and the octane number of the fuel can be altered [8] to modulate the ignition temperature. Since the HCCI combustion process in many operating points is unstable, feedback combustion control is needed to operate an HCCI engine in parts of its operating range. Such combustion control can be performed in numerous ways using different actuators and sensors.

Even though it has many good features, the HCCI engine also has some limitations besides the, just described, need for feedback combustion control. The operational principle unfortunately suffers from very high combustion rates, causing noise as well as wear of engine hardware. Another issue with the HCCI principle is low combustion efficiency at low load. This causes high emissions of unburned hydrocarbons and carbon monoxide.

Besides the development of the actual HCCI principle much work has been carried out addressing the great task of designing a feedback combustion control system for the HCCI engine. The highly non-linear nature of HCCI combustion imposes a great challenge to researchers in the field and many interesting results have been published [8]-[12]. Many of these publications share the property of model assisted controllers, i.e. a model of the combustion process is maintained in the control system and it is continuously updated to reflect the state the engine. Such models are used by the controllers to predict adequate control outputs.

This work addresses the control issue from a strategy perspective rather than from a single control variable or model perspective. A setup (described in [8]) was already available to the authors and slight modifications to this setup were carried out. The main fuel was altered to Natural Gas (NG). N-Heptane was added in small quantities as an ignition improver. Besides the change of fuels the turbo charger was altered, instead of the turbocharger used by Olsson a Variable Geometry Turbocharger was used. The change of turbo is to adopt the setup for NG operation. NG, from an HCCI perspective, is a poorly suited fuel! As previously well known, NG has a very high octane rating (the exact octane number depends on the origin of the gas) typically close to 120 which represents both RON and MON for methane (NG contains mainly methane). The high octane rating of NG makes it very difficult to compression ignite. When combustion is initiated on the other hand the rate of heat release during the combustion event is explosive. The reason for the high rate of heat release is left for the chemistry and the kinetics people to explain, but empirically this is a sound fact. For successful operation it is absolutely essential to operate an NG-fuelled HCCI engine leaner than if the same engine were operated with e.g. ethanol. Successful, in this case, means free of NO_x emissions and with a low enough pressure derivative to avoid noise and engine damage. At the same time the exhaust gas temperature of an HCCI engine is very low, which means that it is difficult to obtain high boost pressure from the turbo without adding pumping losses. An operating strategy that provides enough boost pressure to avoid noise and emissions, and at the same time avoids excessive efficiency penalty due to pumping losses is hence desirable. One such strategy will be developed in this paper.

EXPERIMENTAL APPARATUS

EXPERIMENTAL ENGINE

The engine system that was the basis for the investigation is presented in this section. It should be pointed out that numerous results have been presented from the same setup before, by for example Olsson et al. [8]. Since the last publication by Olsson et al. the laboratory setup has however been somewhat modified, for example the turbo charger has been altered and the fuel selection has been changed. The fuel combination has been changed from n-heptane/ethanol to n-heptane/NG. This affects the performance of the engine, particularly the engine out NO_x emissions and maximum achievable engine load are affected by the altered fuel properties.

The engine system used in these experiments (see Figure 1) is based on a $12\ l$ turbo charged Scania “Euro III” Heavy Duty Diesel engine originally intended for truck applications. The geometrical properties are accounted for through Table 1. The engine was converted to a port fuel injected HCCI engine through the removal of the direct injectors and modifications of the intake in order to mount sequential port fuel injectors. The engine is run on two different fuels, n-Heptane and Danish NG. Two independent fuel systems were needed in order to supply the engine with the two fuels and combustion phasing was controlled through the injection ratio of the different fuels. Each fuel was, for practical reasons, introduced in a separate port. This does, however, not have any major significance to the combustion according to previous results by Olsson et al. [8].

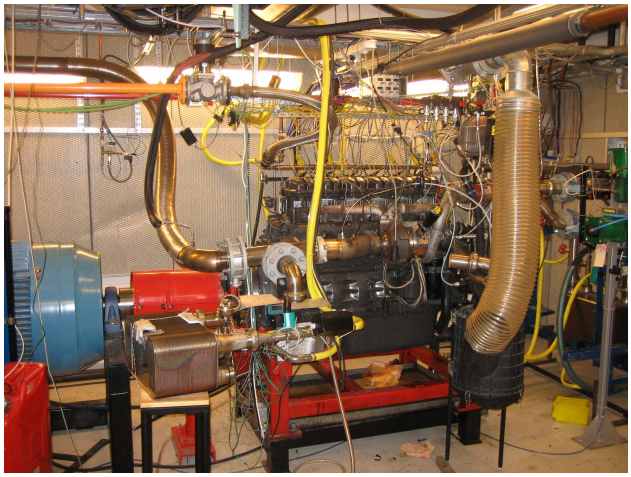

Figure 1: The test engine setup used in the experiments.

Furthermore the standard turbo was changed from the standard “diesel sized” turbo to a Variable Geometry Turbine (VGT) turbo charger There is a need for a VGT turbo in order to supply adequate boost pressure regardless of engine load without inferring extensive pumping losses at the high load operation points. Upstream the turbo the charge air is either cooled or heated depending on the current need for intake air condition.

The engine was, of course, “fully instrumented” with cylinder individual cylinder pressure sensors (CPS), thermocouples and pressure sensors in exhaust ports and at various locations in the intake system. The engine out emissions of O_2, CO, CO_2, NO_x and NO were all measured with relevant instruments.

The system used for control of the engine was a PC based system accounted for and developed by Olsson et al. [8]. The software was written “in-house” in Pascal/Delphi and implements an operator graphical interface which allows the user to monitor the engine operation as well as command fuel ratio, total fuel amount and intake heating/cooling. The control variables can be commanded open-loop by the operator, or by feedback controllers (gain scheduled PID controllers) controlling the combustion phasing by changing the cylinder individual ratios of n-Heptane to NG. Other variables that can be feedback controlled are intake temperature (by changing the power to the electric heaters) and $IMEP_n$ (through the total fuel amount).

RESULTS AND DISCUSSION

The first part of the results deals with engine operation with (almost) no boost pressure applied, we call this Minimum Intake Pressure (MIP) operation. During MIP operation the VGT was, of course, set for minimum boost pressure which also results in minimum exhaust back pressure. The reason for performing MIP tests was to provide reference data to compare the boosted operation with. MIP operation data were obtained prior to developing a boost strategy and conclusions drawn from the MIP data set were kept in mind when developing the boost strategy.

Table 1: Engine geometric properties.

Number of cylinders	6
Displaced Volume	$11705\ cm^3$
Bore	$127\ mm$
Stroke	$154\ mm$
Connecting Rod Length	$255\ mm$
Number of Valves	4
Compression Ratio	$18:1$
Exhaust valve open	$82\,^\circ$ BBDC
Exhaust valve close	$38\,^\circ$ ATDC
Inlet valve open	$39\,^\circ$ BTDC
Inlet valve close	$63\,^\circ$ ABDC
Fuel Supply	dual PFI

The engine setup used in this study has a large number of operating parameters and it is hence essential to apply a strategy for selecting these parameters in order to operate the engine in a good way and reduce the number of number of experiments that have to be performed. It is important to remember that the main objective with the VGT is to extend the operating range of the engine and the boosting strategy serves as a tool for selecting boost pressure to maximize efficiency and minimize emissions of NO_x throughout the extended operating range.

In order to limit the number of pages in this paper each explanation or statement can not be motivated by a diagram.

LIMITATIONS

In order to define the operational area of the engine limits on certain operational parameters were selected. An NO_x emission level of $0.2\ g/kWh$ was regarded as the maximum tolerated, based on the US2007 emission regulations. Peak Cylinder Pressure (PCP) was limited to $180\ bar$, “*ad-hoc*”,. This is a design parameter set by the engine manufacturer and $180\ bar$ is a reasonable PCP in HD applications. Max pressure derivative was limited to $15\ bar/CAD$, again “*ad-hoc*”, based on the emission of noise of the actual engine used. Fuel composition was limited as well. N-Heptane is more expensive than NG and the target consumption of n-heptane thus has to be limited. Since the experiments were run in a multi cylinder engine the fuel ratio condition had to be within a range, due to cylinder to cylinder variations and variation

Table 2: limitations of engine operation.

$CA50_{max} = arg\ max\ \eta_{nte}(CA50)$
$PCP \leq 180\ bar$
$dP/dCAD\ (PCDP) \leq 15\ bar/CAD$
$NO_x \leq 0.2\ g/kWh$
$80/20 \leq Q(NG)/Q(n-Heptane) \leq 85/15$
$T_{in} \leq 180C^{\circ}$

over time of the combustion. The fuel ratio was hence conditioned to be within 80% - 85% NG based on the heating value (the rest is n-Heptane). N-Heptane is not a commercially viable fuel. The fuel limitation is however still valid based on the assumption that you have one main fuel which is difficult to compression ignite (NG, ethanol, methanol or gasoline) by which most of the energy is supposed to be delivered. To the main fuel a fuel which is easy to compression ignite (DME, Diesel or n-Heptane) is added. The task of the secondary fuel is hence not to supply a major part of the energy, only to promote ignition. In a case where the secondary fuel is prized same as the primary fuel user convenience could be one reason to limit the usage of the secondary fuel. A situation where the secondary fuel is supplied at service only would e.g. be ideal. The intake temperature should, due to limitations of the hardware, not exceed $180\ ^{\circ}C$. Finally a limitation of the combustion phasing, CA50, was employed. In this work the crank angle degree where 50 % of the heat release has occurred (Crank Angle 50, CA50) is used as a measure of combustion phasing. CA50 should be set for Maximum net Indicated Efficiency (MIE). The reason for not choosing the standard way of determining best CA50 from the torque (Maximum Brake Toque, MBT) is that the torque sensor did not accurately enough resolve the difference in torque close to MIE. The set of conditions (rules of engine operation) are summarized in Table 2.

OPERATION WITH MINIMUM BOOST

Figure 2 shows net indicated efficiency as a function of CA50 for all speeds and loads during MIP operation. From the figure it is clear that there, for each load but the lowest one, exists a certain CA50 that yields maximum indicated efficiency, i.e. the MIE point. MIE occurs later and later in the cycle as the engine load increases. There are two phenomena that, when combined, result in this behavior. From an ideal-cycle point of view it would be beneficial to have the combustion occurring as close to Top Dead Center (TDC) as possible. This represents the best trade-off of minimized compression work without sacrificing expansion work. However as combustion moves closer to TDC the convective heat losses to the cylinder walls increase. The reason for increased heat loss is twofold. Both the maximum in-cylinder temperature and the maximum pressure derivative increase when the combustion moves closer to TDC. An increased pressure derivative causes pressure ringing in the cylinder, breaking down the thermal boundary layer close to the cylinder walls, which in turn increases the convective heat loss even further. Those two phenomena are responsible for the distinct maximum efficiency at a certain CA50 for all operation points but the very lowest load points. The efficiency at low load actually increases monotonically as the combustion is phased earlier. The reason for this behavior is that combustion efficiency is very sensitive to combustion phasing at low load. The increase of maximum temperature with earlier combustion phasing causes a drastic increase of the combustion efficiency. Since the low loads normally have very poor combustion efficiency, the effect of increased combustion efficiency is the most powerful phenomenon and hence we never find the local maxima for the lowest load points.

It is well known that combustion phasing is a very important parameter for the formation of NO_x (through peak temperature). Figure 3 shows NO_x emissions for the same operating points presented in Figure 2. It is easily noticed that the two major variables that determine the NO_x emissions are engine load and CA50. NO_x emissions increase with decreasing CA50 and increasing load. From Figure 4, showing NO_x emissions for the full operating range, it is obvious that the operating range with MIP operation is very limited. Maximum achievable engine load is just above $4\ bar$ net Indicated Mean Effective Pressure ($IMEP_{net}$). This is anything but impressive, but with the selected operating strategy this is the maximum engine load. Interesting to note is also that it is the limitation of engine out NO_x that sets the maximum load limit, not any other operational parameter. The emission of noise (pressure derivative) is for example far below its limit, it is never more than half the limit value within the engine operating range. The same thing can be said about the other operating parameters.

A strategy for retarding combustion phasing to limit NO_x emissions could be applied but it comes to a large cost in net indicated efficiency (3-4%). Since combustion has to be phased very late to decrease NO_x below the 0.2g/kWh limit, (CA50 between 12 and 15 CAD after TDC) the cycle to cycle variations of engine load and combustion phasing will hence be significantly larger compared to if CA50 is set for MIE. In some cases it might not even be practically feasible to run the engine with such late combustion phasing. A better way seems to be to apply boost pressure. Even though some boost was present even with the

Table 3: Color and sign encoding belonging to NO_x and indicated efficiency figures of the unboosted engine experiments (Figure 2 and Figure 3).

Load [bar]						
6						□
5	*	*	*	*	*	*
4	+	+	+	+	+	+
3	×	×	×	×	×	×
2	○	○	○	○	○	
Speed [rpm]	800	1000	1200	1400	1600	1800

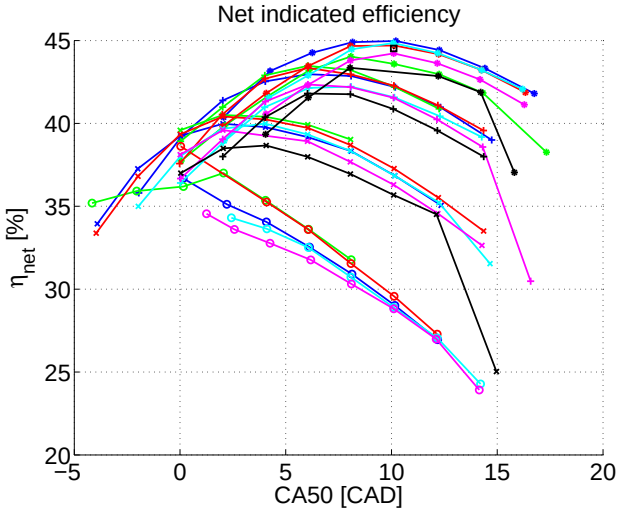

Figure 2: Net indicated efficiency as a function of combustion phasing (MIP operation). Legend can be found in Table 3.

VGT set for MIP (intake pressure and gas exchange efficiency are shown in Figure 5 and Figure 6), this level of boost pressure is negligible and does not infer any net indicated efficiency decrease due to pumping losses in any of the operating points. It does, however, contribute slightly to the increased maximum $IMEP$ at $1800\ rpm$. The option of further increasing the intake pressure to a greater extent using the VGT is dealt with in the following section.

BOOSTED ENGINE OPERATION

For each combination of load, speed and combustion phasing the minimum boost pressure that resulted in acceptable NO_x emission was applied. In practice this meant that the boost pressure was gradually increased, maintaining constant combustion phasing, until the NO_x criterion was met. The combustion phasing controllers adjusted the fuel composition automatically to maintain the combustion phasing at the setpoint. Finally intake temperature, being a slow actuator, was used to "bias" the fuel composition decided by the combustion phasing controllers. The combustion phasing controllers have, as stated before, the capability to affect CA50% by altering the fuel composition on a cycle to cycle basis. However

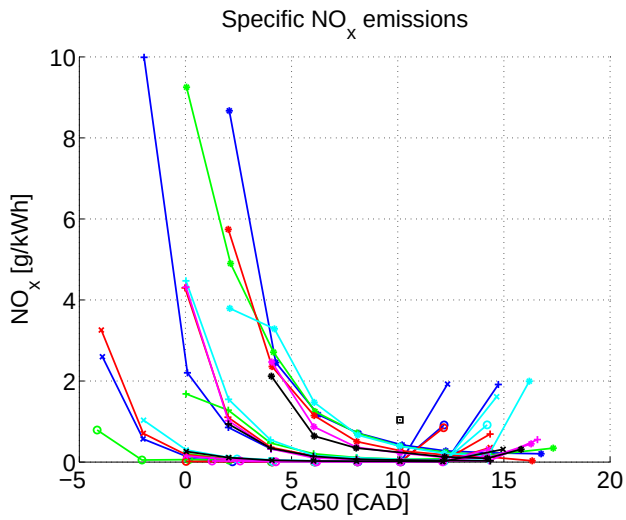

Figure 3: NO_x emissions as a function of combustion phasing (MIP operation). Legend can be found in Table 3.

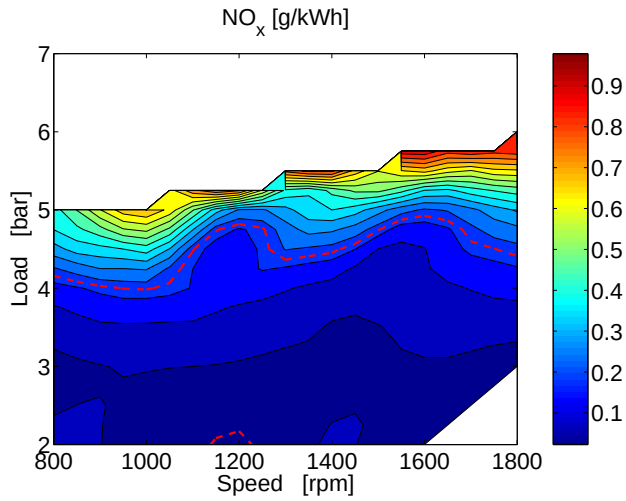

Figure 4: NO_x emissions (MIP operation). Red dashed line indicate limit of $0.2\ g/kWh$.

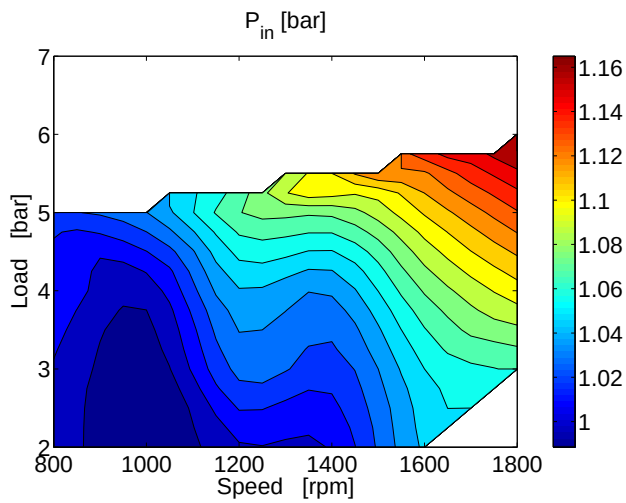

Figure 5: Boost pressure during operation with minimum boost pressure.

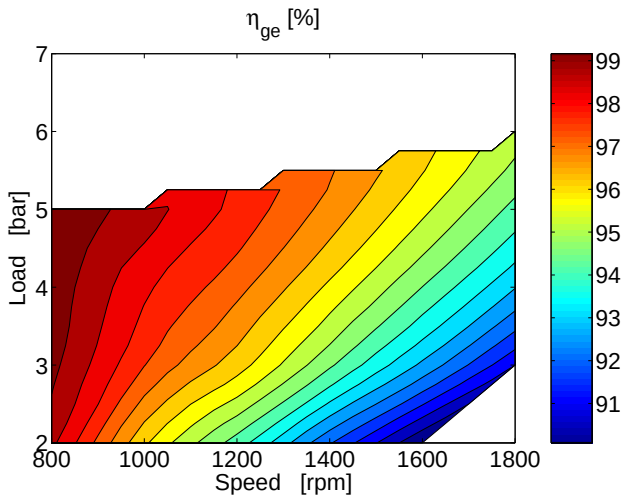

Figure 6: Gas exchange efficiency during operation with minimum boost pressure.

to maintain the fuel composition criterion in all of the six cylinders simultaneously intake temperature was used to "force" each of the phasing controllers to use a fuel composition within the limits of the fuel criterion. Since the controllers adjust the fuel ratios to the cylinders to maintain combustion phasing at the setpoint, a change of intake temperature will result in changed fuel ratios. In reality the intake temperature was increased until all of the cylinders fulfilled the fuel composition criterion. In this way the minimum intake temperature needed was used (meaning maximum thermal efficiency). The process was iterated until suitable conditions were obtained (meaning that the limitations from Table 2 are fulfilled). All of the measurements were of course carried out when the engine had reached steady state conditions. In this way the lowest possible boost pressure was used in order to suppress the NO_x formation. This strategy ensures the minimum efficiency penalty, in the form of pumping losses, for controlling the NO_x emissions. Combustion phasing was varied from the earliest possible to the latest possible in the same manner as during MIP conditions. For each load and speed the best efficiency point (MIE) was again used for putting together performance maps. MIE combustion phasing are not exactly same during boosted and MIP conditions. This is due to the fact that MIE now also depends on the gas exchange efficiency and boost pressure. The suggested boost strategy represents a constructive way of utilizing the extra degree of freedom that the VGT offers.

The results from this operating principle are shown in Figure 7. The principal behavior is similar to MIP operation with efficiency decreasing for early and late combustion phasing except at very low loads. A third effect of CA50 on the efficiency is added using the turbo to gain boost. With early combustion,

Table 4: Color and sign encoding belonging to NO_x and indicated efficiency figures of the boosted engine experiments (Figure 7 and Figure 8).

Load [bar]						
7	◇	◇	◇	◇	◇	
6	□	□	□	□	□	□
5	*	*	*	*	*	*
4	+	+	+	+	+	+
3	×	×	×	×	×	×
2	○	○	○	○	○	
Speed [rpm]	800	1000	1200	1400	1600	1800

the exhaust temperature is reduced which increases the back pressure requirement for maintained boost. Thus early combustion phasing means higher pumping losses. The pumping losses with early combustion are further increased by the fact that early combustion generates higher maximum temperature and thus NO_x which further increases the boost requirement. Because of this it is necessary to apply more boost pressure in order to lower the maximum temperature, hence net indicated efficiency is decreased slightly for early combustion phasings by the increased pumping losses. As a result MIE is phased later with boost than with MIP operation.

The reason for having the turbo charger is, as mentioned previously, to extend the operating range of the engine with the limitations from Table 2 in mind. As visible from the $0.2\ g/kWh\ NO_x$ line in Figure 9 the operating range was expanded by almost $2\ bar\ IMEP_{net}$ with the use of the turbo charger. The limiting factor of engine load is as with MIP operation the engine out emissions of NO_x. A comparison of Figure 3 and Figure 8 reveals that the VGT turbo makes it possible to "move" the NO_x formation curves significantly to the "left". This means that, for all the loads that were possible to operate without boost, NO_x formation requires much earlier combustion phasing with boost than without. It is hence possible to maintain MIE combustion phasing at higher loads without violating the NO_x limit. Almost all of the indicated operating points in Figure 8 have been run with a significant amount of boost to reduce NO_x emissions below the limit. The boost pressure used and the corresponding gas exchange efficiency are shown in Figure 10 and Figure 11. In the event that it was not possible to reduce NO_x below the limit the turbo was set for maximum boost pressure (as is the case with some high load and low speed curves).

NET INDICATED EFFICIENCY

The most important result of this investigation is the comparison of the overall η_{net} shown in Figure 12 and Figure 13 for MIP and boosted operation respectively. It is found that η_{net} for the two cases are equal up to

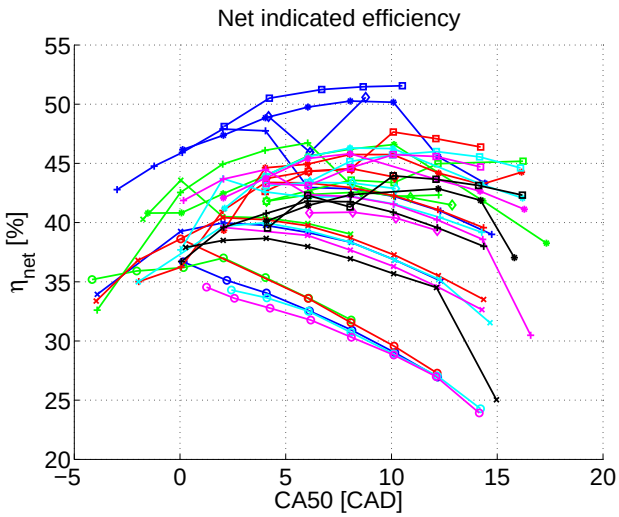

Figure 7: Net indicated efficiency as a function of combustion phasing. Intake pressure according to boost strategy. Legends can be found in Table 4.

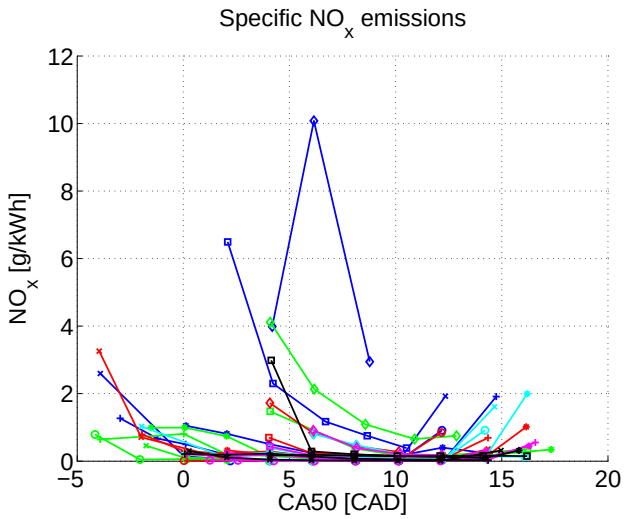

Figure 8: NO_x emissions as a function of combustion phasing. Intake pressure according to boost strategy. Legends can be found in Table 4.

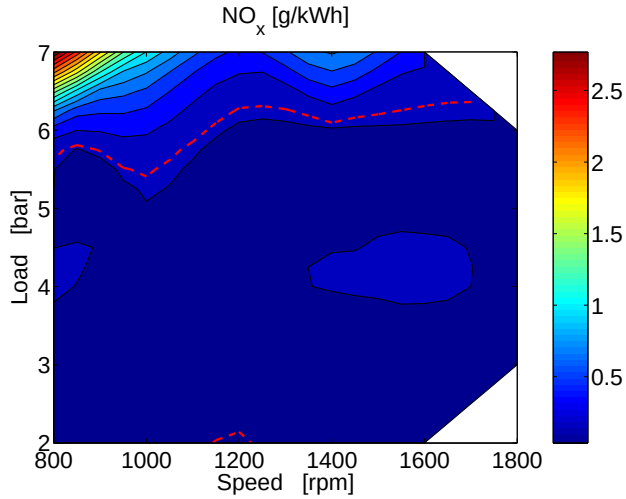

Figure 9: NO_x emissions. Engine run according to operational strategy. Red dashed line indicates limit of $0.2\ g/kWh$.

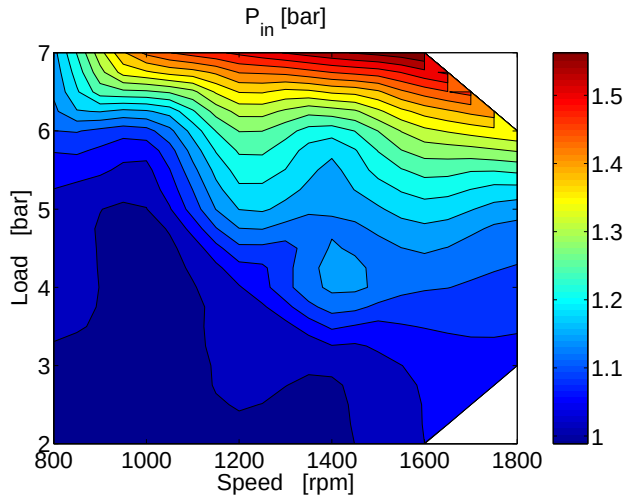

Figure 10: Boost pressure with boosted operation according to strategy.

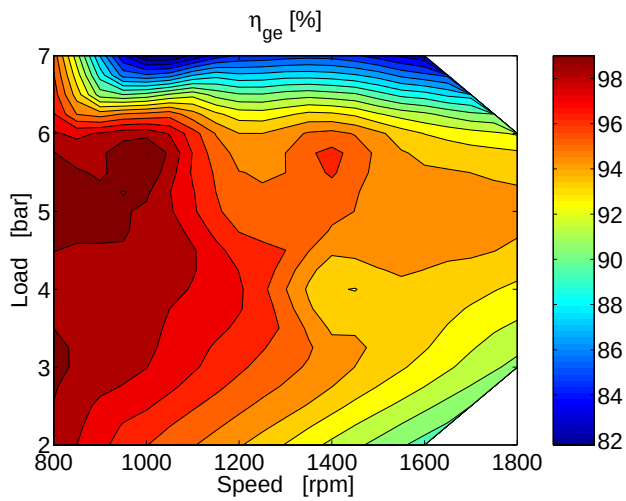

Figure 11: Gas exchange efficiency with boosted operation according to strategy.

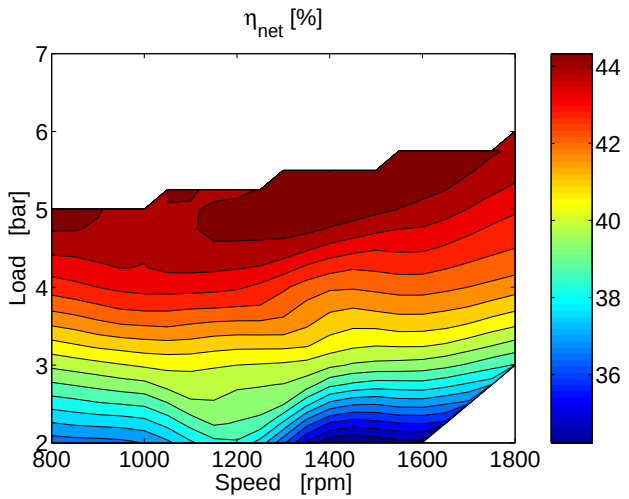

Figure 12: Net indicated efficiency (MIP operation).

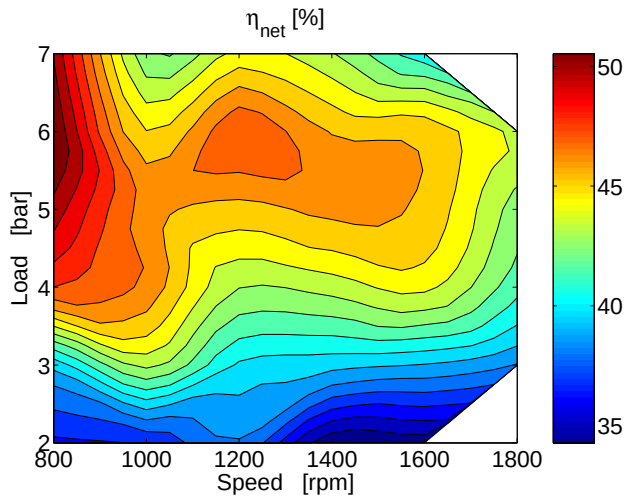

Figure 13: Net indicated efficiency. Engine run according to boost strategy.

roughly $4\ bar\ IMEP_{net}$. This is to be expected from the reasoning before since no boost is required to fulfill the operational criterion below this load level. In Figure 4 we find that $4\ bar$ is the maximum achievable load in the MIP condition. If it is desirable to run loads above $4\ bar\ IMEP_{net}$ an elevated boost pressure is needed in order to suppress the formation of NO_x. This agrees well with what we find in Figure 13. When the high load region of Figure 13 is studied we find that η_{net} at first increases with load due to increased combustion efficiency. At about $6\ bar$ the maximum efficiency is obtained. This also happens to be the high-load limit due to NO_x formation as can be seen in Figure 9. Above the $6\ bar$ load limit η_{net} drops rapidly. The efficiency drop is attributed to extensive pumping losses and heat losses due to breakdown of the thermal boundary layer caused by pressure oscillation. We have reached the limit of what is possible to achieve with the turbo charger. In the region between 4 and $6\ bar$ where MIE is not possible to achieve with MIP operation the efficiency is improved with boosted operation. This is expected since boosted operation allows us to operate at MIE phasing. Finally it should be pointed out that the very high η_{net} indicated in Figure 13 at $800\ rpm\ 6\ bar\ IMEP$ is regarded as an outlier, it was and should be overlooked.

CONCLUSIONS

One possibility to increase the operating range of a natural gas fuelled HCCI engine is to apply some kind of supercharging. Supercharging does, however, come at a cost. In this case, using a VGT turbo, the cost is increased pumping losses. For the operating points where combustion phasing is NO_x limited without turbo charging the efficiency can actually be increased with turbo charging however. It is hence essential to elevate intake pressure with care. If a variable turbocharger is present in the system some operating strategy is necessary.

One such operating strategy, based on constraints on control variables as well as cylinder pressure and emissions, was developed and analysed. The strategy in terms of turbo charging is to apply as little boost as possible while still meeting the NO_x requirements. The developed strategy enables structured engine operation maximizing the net indicated efficiency within the set constrains. Maximum achievable load was increased from $4bar$ to $6bar$ of $IMEP_{net}$ by the usage of the variable turbocharger and the applied operating strategy. Over all net indicated efficiency at high load between 45 and 50% was obtained with turbo charged operation. The effect of turbo charging on maximum efficiency combustion phasing was also discussed, illustrated and included in the operational strategy, both from an NO_x emissions perspective and from an overall efficiency perspective.

Natural gas seems to be an ill-suited fuel for pure HCCI operation due to its high ignition temperature and rapid combustion. The high-load limit of the operating range was due to excessive NO_x emissions and one reason is the small temperature span available with natural gas between auto ignition and NO_x formation.

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APPENDIX

DEFINITION OF CA50%

The definition of CA50% (the crank angle where 50% of the total heat release is obtained) is evident from figure 14. CA50% is best understood as the instance where half of the total heat which will be released during the cycle are released. CA50% were in this work as well as in much other research, used as an measurement on where during the engine cycle combustion actually occur.

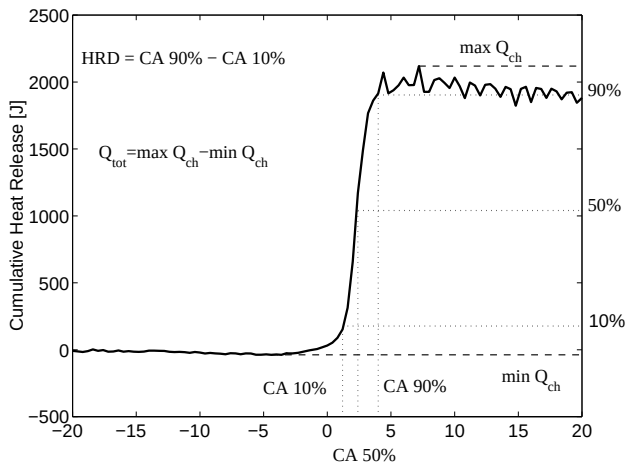

Figure 14: A typical accumulated heat release curve (adopted from [13]). The definition of CA50% is indicated in the figure, the unit of the horizontal axis are Crank Angle Degrees (CAD).