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ABSTRACT

Homogeneous Charge Compression Ignition (HCCI) holds great promises for good fuel economy and low emissions of NO_X and soot. The concept of HCCI is premixed combustion of a highly diluted mixture. The dilution limits the combustion temperature and thus prevents extensive NO_X production. Load is controlled by altering the quality of the charge, rather than the quantity. No throttling together with a high compression ratio to facilitate auto ignition and lean mixtures results in good brake thermal efficiency. However, HCCI also presents challenges like how to control the combustion and how to achieve an acceptable load range. This work is focused on solutions to the latter problem.

The high dilution required to avoid NO_X production limits the mass of fuel relative to the mass of air or EGR. For a given size of the engine the only way to recover the loss of power due to dilution is to force more mass through the engine. This paper shows that this can be done by the use of turbo charging or a mechanically driven compressor. The cost of forcing more air through the engine and the higher peak pressure requirements are discussed and quantified by simple engine modeling supported by experimental data.

INTRODUCTION

Homogeneous Charge Compression Ignition (HCCI) is a hybrid between the well-known Spark Ignition (SI) and Compression Ignition (CI) engine concepts. As in an SI engine a, more or less, homogeneous charge of fuel and air is created in the inlet system. During the compression stroke the temperature of the mixture increases and reaches the point of auto ignition; i.e. the mixture burns without the help of any ignition system, just like in a CI engine. The differences compared to the conventional concepts are the lack of direct means to control the onset of combustion and the simultaneous combustion in the whole cylinder.

The first studies of this phenomenon in engines were performed on 2-stroke engines [1-6]. The primary purpose of using HCCI combustion in 2-stroke engines is to reduce the Hydro Carbon (HC) emissions at part load operation. Later studies on 4-stroke engines have shown that it is possible to achieve high efficiency and low NO_X emissions by using a high compression ratio and lean mixtures [7-30]. In the 4-stroke case, a number of experiments have been performed where the HCCI combustion in itself is studied. This has mostly been done with single cylinder engines, which normally do not provide brake values. However, Stockinger et al. [23]. demonstrated brake efficiency of 35% on a 4-cylinder 1.6 liter engine at 5 bar Brake Mean Effective Pressure (BMEP). Later studies have shown brake thermal efficiencies above 40% at 6 bar BMEP [25].

Since the homogeneous mixture auto ignites, combustion starts more or less simultaneously in the entire cylinder. To limit the rate of combustion under these conditions, the mixture must be highly diluted. In this study a highly diluted mixture is achieved by the use of excess air. Controlling the amount of residual gas is another effective way to produce a diluted mixture. Without sufficient mixture dilution, problems associated with extremely rapid combustion and knocking-like phenomena will occur, as well as excessive NO_X production. On the other hand, an overly lean mixture will result in incomplete combustion or even misfire.

It is commonly accepted that the onset of combustion is controlled by chemical kinetics [18, 24 30, 31]: As the mixture in the cylinder is compressed, the temperature and pressure increase. Temperature and pressure histories, oxygen concentration, fuel quantity and composition as well as combustion products govern how combustion is initiated. As a consequence, combustion timing will be strongly influenced by air-fuel ratio, inlet temperature, compression ratio and residual gas fraction.

One of the main motivations for HCCI is the low engine out NO_X emissions. This is achieved by the high dilution of the mixture, preventing high gas temperatures during combustion. However, the high dilution also limits the amount of fuel that can be added for a given mass of charge, i.e. engine power is low relative to the mass flow through the engine.

The obvious way to win back the load is to boost the engine. The authors have previously shown how this can be done by turbo charging the engine [28], but as shown in that study and also pointed out by Hiltner et al. [32], turbo charging is associated with costs. This study focuses on the energy flow for different boost creation strategies. Most of the discussion is based on a simple cycle simulation but some experimental examples of turbo charged HCCI are included as well.

DEFINITIONS

The most important parameter for HCCI operation is perhaps the relative air-fuel ratio, λ.

$$\lambda \equiv \frac{\left(\frac{\dot{m}_{AIR}}{\dot{m}_{FUEL}}\right)_{ACTUAL}}{\left(\frac{\dot{m}_{AIR}}{\dot{m}_{FUEL}}\right)_{STOICH}} \qquad \text{(Eq. 1)}$$

In this study the gas exchange is very important and sometimes it makes sense to split the pumping work between intake and exhaust. Normalized pumping work, PMEP, is defined as positive when work is lost. The following definitions suggest how normalized pumping work can be divided between intake and exhaust:

$$PMEP_{INTAKE} \equiv \frac{1}{V_{DISP}} \int_{GETDC}^{GEBDC} (p_{ATM} - p) \cdot dV \qquad \text{(Eq. 2)}$$

$$PMEP_{EXHAUST} \equiv \frac{1}{V_{DISP}} \int_{EXPBDC}^{GETDC} (p - p_{ATM}) \cdot dV \qquad \text{(Eq. 3)}$$

$$PMEP \equiv PMEP_{INTAKE} + PMEP_{EXHAUST} \qquad \text{(Eq. 4)}$$

Here V_{DISP} is the total displacement volume of the engine.

When analyzing turbo charging another very useful variable is the in-cylinder available exhaust energy:

$$AvailExhMEP \equiv \frac{1}{V_{DISP}} \int_{EVO}^{EVC} \left(h(cyl) - h_{S,CYL}(p_{ATM})\right) dm_{EV} \qquad \text{(Eq. 5)}$$

h(cyl) is the enthalpy of the in-cylinder gas. $h_{S,CYL}(p_{ATM})$ is the enthalpy of the in-cylinder gas if it is isentropically expanded to atmospheric pressure. Some of this available energy is lost over the exhaust valve and in the exhaust plenum. The energy available to the turbine is thus:

$$AvailTurbMEP \equiv \frac{120}{n \cdot V_{DISP}} \int^{Cycle} \left(h(turb) - h_{S,PORT}(p_{ATM})\right) dm_{TURB}$$
$$\approx \frac{120}{n \cdot V_{DISP}} \cdot \left(h(T_{TURB}, p_{TURB}) - h_{S,PORT}(p_{ATM})\right) \dot{m}_{EXH} \qquad \text{(Eq. 6)}$$

The first equation requires the instant state of the exhaust gas in the turbine port to be determined. This is almost impossible to measure, and thus an assumption of a constant state in the exhaust plenum is used for the simplified equation. T_{TURB} and p_{TURB} are the temperature and pressure measured in the turbine ports. $h_{S,PORT}(p_{ATM})$ is the enthalpy of the gas after an isentropic expansion from the state in the turbine port to atmospheric pressure.

Also when analyzing mechanically driven compressors it makes sense to normalize the work required to drive them:

$$CompMEP \equiv \frac{P_{MECH} \cdot 120}{n \cdot V_{DISP}} \qquad \text{(Eq. 7)}$$

Friction Mean Effective Pressure (FMEP) quantifies the mechanical losses and is, in this study, defined as the difference between IMEPn and BMEP minus CompMEP, i.e. not including pumping losses or the work to drive the compressor:

$$FMEP \equiv IMEPn - BMEP - CompMEP \qquad \text{(Eq. 8)}$$

The MEP variables are all directly comparable, for a more complete review of the different MEP:s, see [36].

Brake thermal efficiency is the overall conversion ratio from fuel heat to mechanical power:

$$\eta_{BRAKE} = \frac{P}{\dot{m}_f \cdot Q_{LHV}} \qquad \text{(Eq. 9)}$$

Gas exchange efficiency, equation 10, is the ratio between the indicated work during the complete cycle, net IMEP, and the closed part of the cycle, gross IMEP. For successful turbo charging, where energy in the exhaust is recovered outside of the cylinder to be used for gas exchange, the gas exchange efficiency may be above 100%.

$$\eta_{GE} \equiv \frac{IMEPn}{IMEPg} = \frac{IMEPg - PMEP}{IMEPg} \qquad \text{(Eq. 10)}$$

MODELS AND SIMPLIFICATIONS

Since it is not very easy or practical to change the boost generating system, modeling and simulations are the main tools for this study. This section describes the assumptions and simplifications used for the simulations.

The first assumption concerns the relationship between intake pressure and $PMEP_{INTAKE}$ and exhaust back pressure and $PMEP_{EXHAUST}$:

$$PMEP_{INTAKE} \approx p_{ATM} - p_{INTAKE} \qquad \text{(Eq. 11)}$$

$$PMEP_{EXHAUST} \approx p_{EXHAUST} - p_{ATM} \qquad \text{(Eq. 12)}$$

Further it is assumed that the pressures in both exhaust and intake plenums are constant and that the exhaust gas is well mixed before it reaches the turbine, i.e. the temperature in the turbine inlet is assumed to be constant.

To estimate variables from the closed part of the cycle, e.g. peak cylinder temperature and peak cylinder pressure as well as quantitative information about the exhaust stroke, a very simple simulation tool is used. The gas is modeled as one gas with a temperature dependent γ. Combustion is modeled as added heat according to a Vibe function fitted to a measured high load HCCI cycle. Heat losses are model by the Woschni correlation. The cycle is closed from BDC to BDC and followed by blow down and exhaust stroke with heat losses modeled by a constant heat transfer coefficient. It is all very simple, but the results are realistic and predict the correct trends.

Since the paper discusses high load HCCI, all simulations are performed at the NO_X controlled load limit, i.e. λ is adjusted to limit cycle temperature to 1800 K.

Table 1 Ignition temperature for three different fuels [35].

Fuel	***Ignition Temperature***
Ethanol	950 K
Isooctane	1000 K
Natural Gas	1150 K

In most feasible HCCI implementations charge temperature has to be moderated in some way, either by heating or trapping of residuals. It is therefore realistic to perform simulations with a constant compression temperature close to the ignition temperature of the fuel. In a detailed study of the auto ignition process there is no such thing as the ignition temperature of the fuel. However, in practice it turns out that for a given fuel the exothermic reactions start when the in-cylinder temperature reaches a certain threshold. This temperature threshold varies only slightly with speed and load, but is highly dependent on the fuel. Table 1 shows the temperature thresholds, measured in an engine, for some fuels.

A low ignition temperature is preferable to allow for much heat release without reaching the NO_X critical temperature. At the same time auto ignition at desirable timing should not require unrealistically low inlet temperature or too low compression ratio. Ethanol combines low ignition temperature with high octane rating, due to the strong cooling effect from vaporization. This is why ethanol is such a good HCCI fuel.

In the simulations inlet temperature is adjusted so that the temperature at TDC would be 1000 K for a motored cycle.

EXPERIMENTAL APPARATUS

The test engine is a modified six-cylinder Scania D12 turbo charged diesel engine. The engine, in its original configuration, is mainly used in truck applications. The original diesel injection system is removed and replaced by a low-pressure sequential port fuel injection system for two separate fuels. The engine has four-valve cylinder heads and also two inlet ports per cylinder. The injection system can thereby supply two fuels to each cylinder, one in each port. This way, the amount of each fuel can be individually adjusted for each cylinder from a controlling computer. The inlet manifold is extended to supply space for the injectors.

Figure 1 shows a schematic of the engine system. Essentially the same engine setup has been used in previous studies by the authors [25, 26, 28, 36].

Because of the high dilution of the charge required for HCCI operation, exhaust gas temperature is often several hundred K lower than for a comparable DICI engine. Therefore, the diesel-engine turbo charger has to be replaced in order to obtain boost pressure. The new compressor has essentially the same characteristics as the one used in the truck application, whereas the turbine is different. The low temperature of the exhaust gas results in high density and thus low volume flow. In order to extract a certain amount of work to compress the inlet air from this low volume flow, the turbine has to be small and work with a higher pressure-ratio than would have been necessary in a standard CI engine. A Holset ZA13JC8 turbine is selected. As a comparison this turbine has an inlet area of 13 cm^2, while the turbine used in the truck application has an inlet area of 25 cm^2.

Apart from these changes the engine is in its original configuration, with pistons and cylinder heads unchanged. The properties of the engine are summarized in Table 2.

In earlier studies with this engine, [25, 26, 28], combinations of n-heptane and gasoline or isooctane, have been used. In this study, the combination of n-heptane and ethanol is used because of the high octane rating of ethanol.

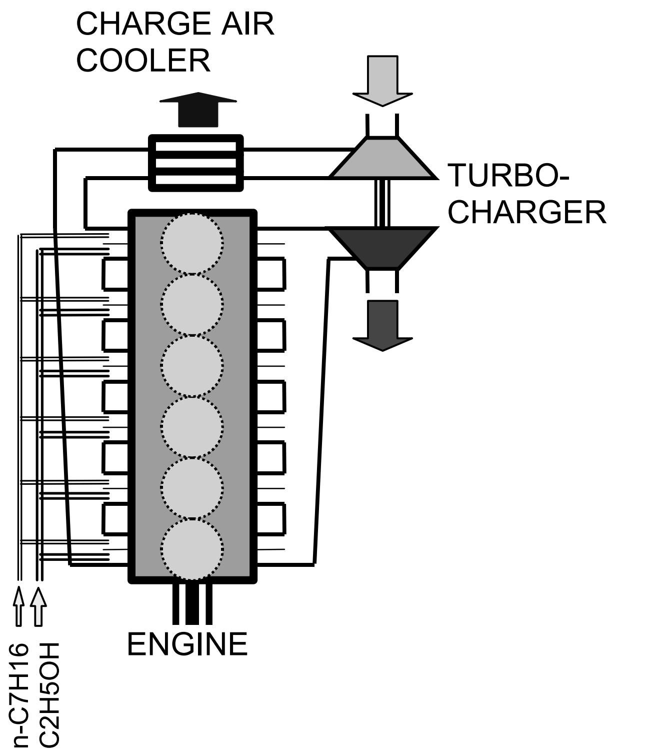

Figure 1 The engine system consisting of an engine with a double fuel injection system, turbo charger and charge-air cooler.

Table 2. Geometric specifications of the engine. Valve timings refer to 0.15 mm lift plus lash.

Displacement volume	11 705 cm^3
Compression Ratio	18:1
Bore	127 mm
Stroke	154 mm
Connecting Rod	255 mm
Exhaust Valve Open	82° BBDC
Exhaust Valve Close	38° ATDC
Intake Valve Open	39° BTDC
Intake Valve Close	63° ABDC

THE HCCI LOAD POTENTIAL

To avoid extensive NO_X production, in-cylinder temperature has to be limited to somewhere between 1800 and 1850 K, depending on engine speed [34]. Another limiting factor is the maximum allowed Peak Cylinder Pressure (PCP).

Figure 2 (simulation) shows the maximum load for HCCI compared to a diesel-like engine as a function of PCP for different compression ratios. It also shows how much boost is needed to hit the peak load for the given PCP limit. The diesel-like engine is simulated using the same rate of heat release and λ = 1.2. The difference in IMEP between compression ratios is smaller for HCCI than the λ=1.2 engine. This is due to the fact that a lower compression ratio requires higher intake temperature and thus the charge density is lower for the lower compression ratio.

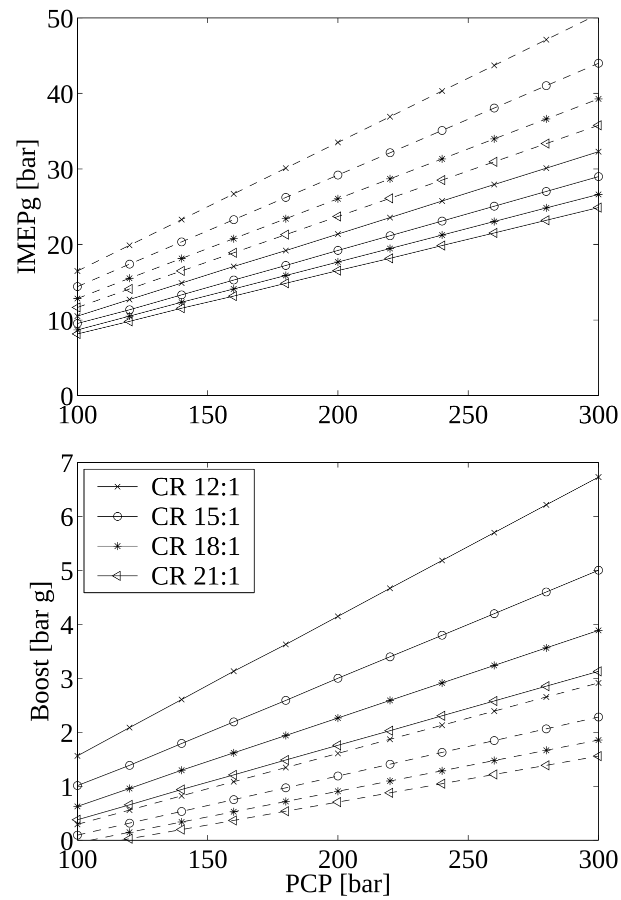

Figure 2 Maximum IMEP gross and the boost required for HCCI with compression temperature of 1000K. Dashed lines illustrates λ=1.2 and inlet temperature 25°C.

Figure 3 is a zoom-in of Figure 2 complemented with experimental results. The simulations are all performed at 1500 rpm, but to vary boost at peak load with the present turbo charger, speed had to be changed in the experiments. The model overestimates the boost that can be tolerated for a given PCP. One reason for this is the pressure oscillations that occur at high load, these are not modeled and therefore do not contribute to higher cylinder pressure in simulations. The loads correspond well, which indicates a higher λ in the experimental case. The temperature limit chosen for simulations, 1800 K, is probably somewhat conservative, but maybe somewhat too high NO_X emissions are tolerated for the experiments. Brake specific NO_X varies between 0.03 and 0.06 g/kWh. This is low, but maybe not low enough for an engine of the future without after treatment.

Another effect that occurs in real life but is not modeled is the increased wall temperature. At high load, pressure oscillations result in high heat losses. The wall temperature then increases and to avoid too early ignition intake temperature is lowered and thus the charge density is increased.

Obviously boost can be used to recover some of the load that is lost by the high dilution. However, to achieve loads around 25 bar BMEP would require an engine that could tolerate very high cylinder pressure and the engine would need extensive boost. It will be shown below that it is not realistic to boost the engine to the maximum load of a present-day Diesel engine.

THE COST OF BOOST

There are several possibilities for generating boost. The most common method is turbo charging, other possibilities are mechanically driven compressors or pressure wave chargers.

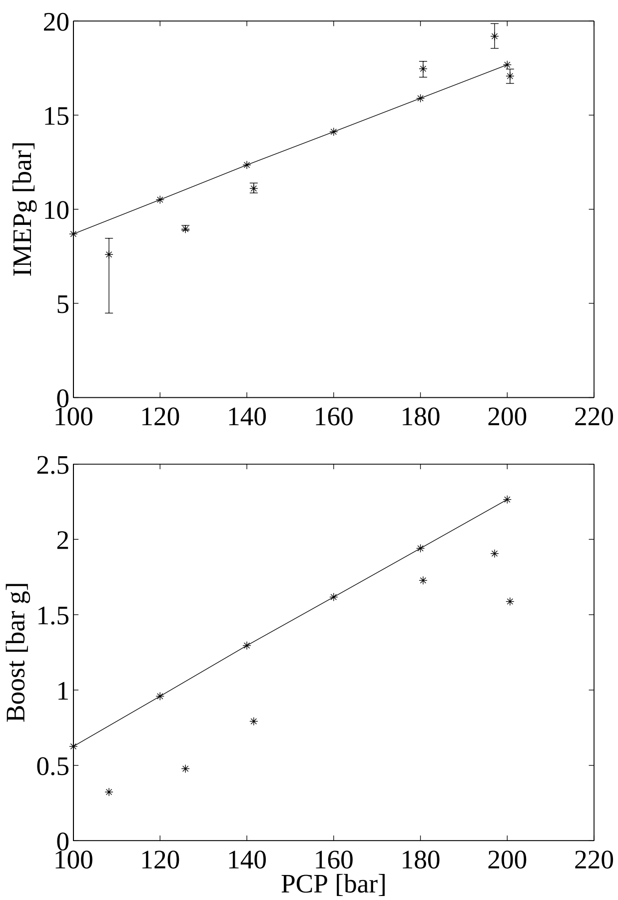

Figure 3 Simulated maximum IMEP gross, lines, compared to measured values, single stars. The bars indicates the cylinder to cylinder variation in the experiment.

MECHANICALLY DRIVEN COMPRESSORS

These are mainly used when brake thermal efficiency is not important, or when immediate response and low speed torque is important. Some passenger car manufacturers, e.g. Mercedes, use mechanically driven compressors.

The high boost required by HCCI makes the roots blower type compressors unsuitable. These do not feature internal compression and have very poor efficiency for high pressure ratios. Centrifugal compressors and screw compressors reach efficiencies around or above 70%. Rewriting Eq. 7 to use compressor efficiency and intake pressure gives:

$$CompMEP = \frac{\gamma}{\gamma - 1} \cdot \frac{\eta_{VOL} \cdot T_{AMB}}{\eta_{COMP} \cdot T_{INLET}} \cdot \left(\left(\frac{p_{INLET}}{p_{ATM}} \right)^{\frac{\gamma-1}{\gamma}} - 1 \right) \cdot p_{INLET}$$

Figure 4 shows the penalty for using a mechanically driven compressor to generate boost. On the y-axis $PMEP_{INTAKE}$ (a negative number for a boosted engine) is added to CompMEP. The reason for this is that some of the work for driving the compressor is recovered during the intake stroke. The figure also illustrates the possible gain by using two-stage compression with inter-stage cooling. This is especially important for high pressure ratios. At 3 bar boost about 1 bar of BMEP can be saved by applying inter-stage cooling.

TURBO CHARGING

Classically turbo charging is known to increase the efficiency of the engine, by utilizing exhaust energy that otherwise would be lost. In many applications this is more or less true, but there is always a price to pay for the increased intake pressure – higher exhaust backpressure.

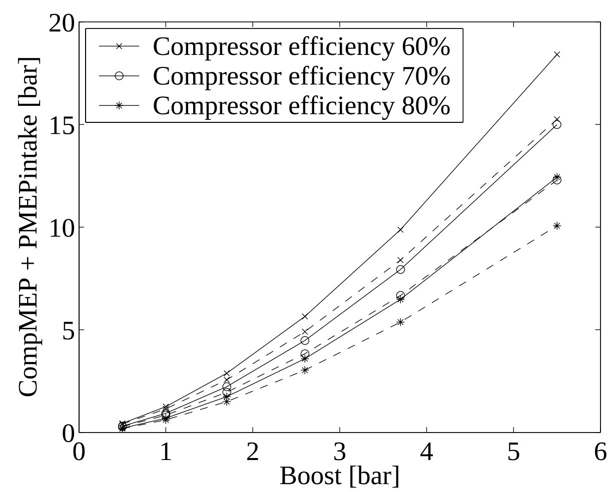

Figure 4 The work required to drive the compressor, compensated for the gain in pumping work. One stage compression solid lines and two stage compression dashed lines.

As long as the increase in intake pressure is higher than the increase in backpressure, turbo charging is beneficial for fuel economy. However, for engines operating with very diluted mixtures, like the HCCI engine, exhaust temperature is low and high backpressure is needed to generate the boost. At the same time more boost is needed for a given load. Figure 5 shows how exhaust backpressure varies with intake pressure for some different exhaust temperatures. For HCCI, temperatures between 250°C and 450°C are typical. The dashed lines show two-stage compression with inter-stage cooling. However more complex, the extra effort pays off well for high boost pressures.

Exhaust backpressure and intake pressure, for a varied load with Overall Turbo Efficiency (OTE) of 55% is shown in Figure 6. A prerequisite for this graph is that the boost can be perfectly controlled and OTE maintained as load is varied. This is very hard, maybe impossible, to achieve in practice, especially at different speeds. The important thing to read out of Figure 6 is that the more boost is needed, the bigger the difference between intake pressure and exhaust backpressure.

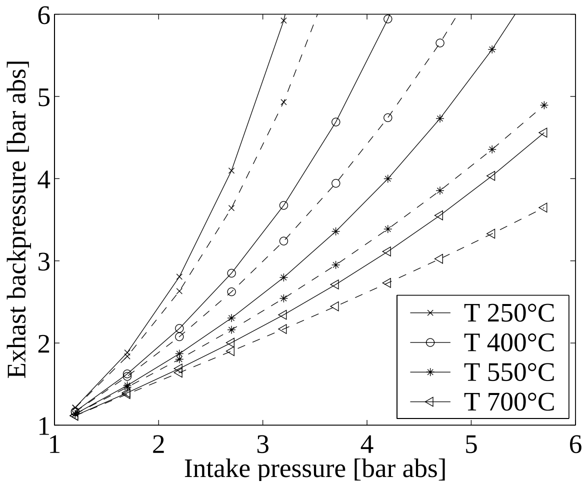

Figure 5 Exhaust backpressure versus intake pressure for different exhaust temperatures, assuming overall turbo efficiency of 55%. One stage compressor solid lines and two stage compressor dashed lines.

Figure 7 is zoomed in to a more realistic range of inlet pressures and the simulations are compared to experimental readings. In the experiments the OTE was typically around 50% or below, and thus the backpressure “lifts off” the 1:1 line earlier and more markedly. This illustrates the importance of high OTE. The higher the turbo charger efficiency, the higher boost can be achieved before pumping losses get severe.

In some of the figures above is mentioned two-stage compression with inter-stage cooling. This is very good for reducing the work required to drive the compressor at high pressure-ratios. It is a strategy very suitable for improving the gas exchange of the HCCI engine at high load.

Figure 8 illustrates two effects of applying two-stage turbo charging. The first thing is that lines for different compression ratios basically collapse to one line for loads up to about 17 bar BMEP. This is because the penalty for the lower compression ratio, in terms of higher boost, is lowered and thus fully compensated by the higher exhaust temperature.

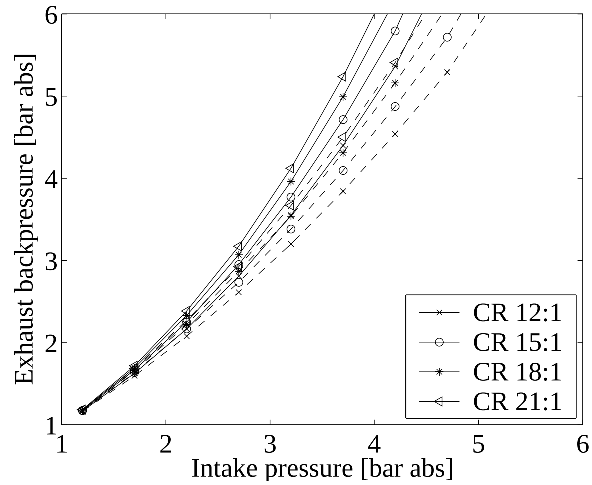

Figure 6 Exhaust backpressure versus intake pressure if load is increased to fully utilize the boost. One-stage compression solid lines and two-stage compression dashed lines.

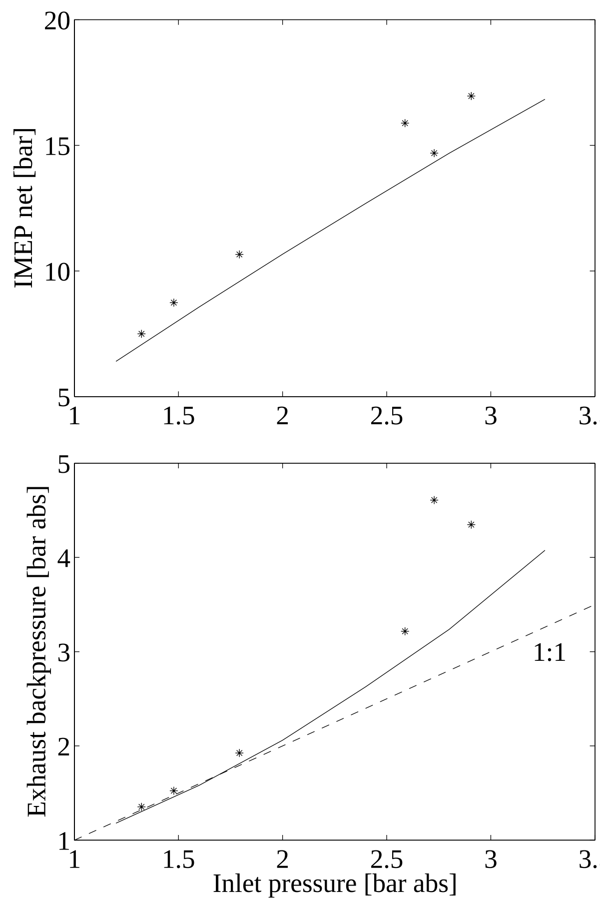

Figure 7 Simulated, line, and measured, stars, load and exhaust backpressure for various inlet pressures.

The other thing to note is the significantly lower pumping work at highly boosted conditions. Even for moderate loads, a small gain is possible. Again, OTE of 55% is used and the results are dependent of this. If one bar of pumping losses can be accepted, then 22-23 bar of IMEP net can be reached with one of the higher compression ratios. The simulation is performed for PCP up to 300 bar. From PCP 200 bar a marker is used for every 20 bar of PCP.

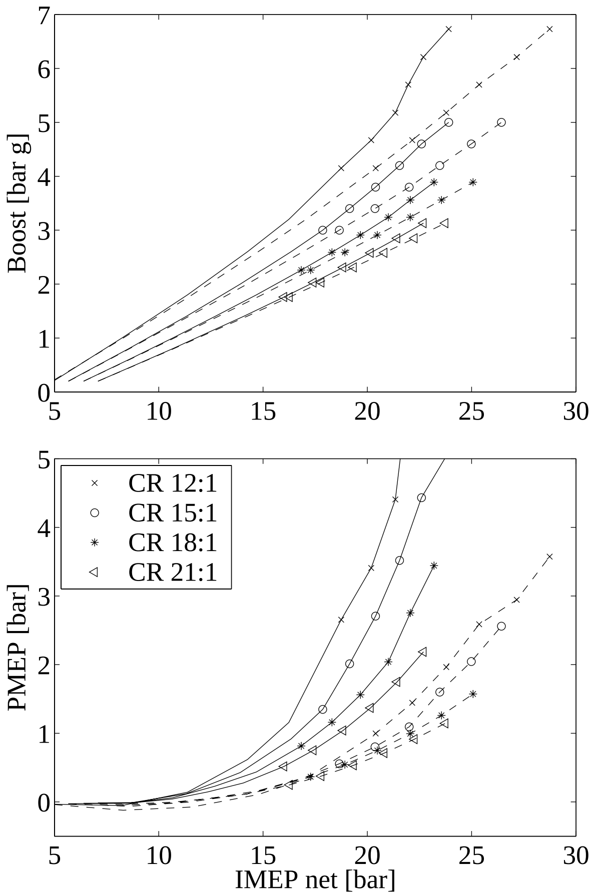

Figure 8 Boost and pumping work as load is increased towards the PCP limit. One-stage compressor solid lines, two-stage compressor dashed lines.

Relating pumping losses to load, η_{GE}, is a very relevant measure of the cost of boost. Figure 9 illustrates how η_{GE} drops off for high loads, i.e. high boost levels. Compared to Figure 8 the effect does not look as drastic, but the essence is the same – for acceptable losses at very high load, two-stage turbo charging is necessary.

HIGH LOAD BOOST STRATEGY

For really high load HCCI, even the two-stage inter-cooled turbocharger solution causes high pumping losses. This is because the blow-down energy, that is available for free, is such a small part of the energy needed to drive the turbine. The piston then has to drive the turbine during the exhaust stroke and due to the far from 100% efficiency of the turbine, this is an expensive way to produce power for the compressor.

However far fetched, for an automobile application where low speed torque and high peak load are important and high load fuel economy is not, one could imagine a system with both compressor and turbo charger. Even if this is not a very probable real-life solution, it is still interesting from an academic point of view to study how the engine could benefit from such a solution.

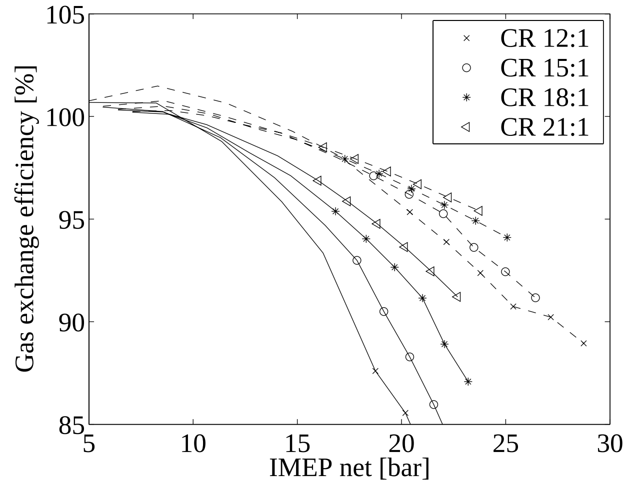

Figure 9 Gas exchange efficiency as load is increased. Solid lines for one-stage turbo charging and dashed lines for two-stage turbo charging.

One way to reduce, but far from eliminate, the pumping losses at really high load, is to use a mechanically driven compressor for one of the two stages. This is illustrated in Figure 10. Turbine driven only compressors are compared to a strategy of one turbine driven and one mechanically driven compressor. When both compressors are turbine driven the intermediate pressure is selected to minimize total compressor power only, i.e. the two pressure ratios are selected to be equal. Dashed lines show the penalty when one of the stages is mechanically driven. Here the intermediate pressure has to be selected through optimization. The higher the boost pressure, the larger the part of the work that should be done by the mechanical compressor.

A probably more realistic approach would be to use electrically assisted turbo charging. However, simulations showed that this is beneficial only for extremely high boost. The simulations used a mechanical-to-electricity-to-mechanical efficiency of 70% and the strategy was beneficial in terms of efficiency only for 5 bar boost or more.

PRACTICAL CONSIDERATIONS

COMBUSTION AND BACK PRESSURE

Turbo charging for high boost pressure typically results in high backpressure. High backpressure prevents an effective emptying of the cylinder and also raises the temperature of the residual gases. This is not very good for the control of combustion phasing and engine design.

To optimize efficiency and minimize NO_X emissions compression ratio should be selected as high as possible, so that at peak load the control variables almost saturate [29]. If intake temperature is used for control, the lowest intake temperature that can always be guaranteed must be low enough to prevent too early combustion at full load.High backpressure, and thereby high fraction of hot residual gases, at full load will significantly increase the charge temperature and thereby limit the compression ratio. Also a lower compression ratio results in higher residual fraction.

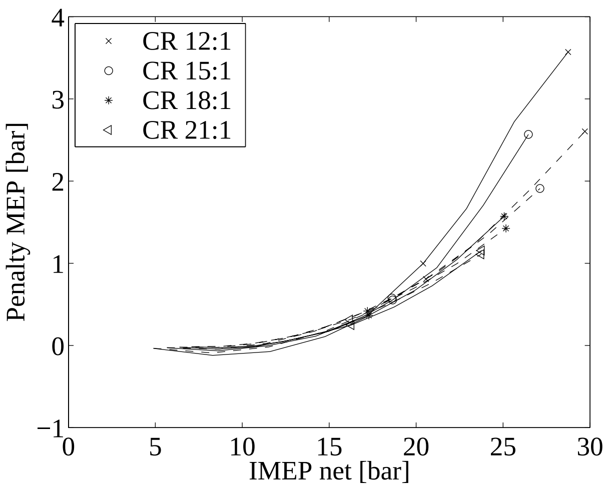

Figure 10 Two-stage compression. Both compressors turbine driven, solid lines and one stage mechanically drive, dashed lines.

BOOST SOLUTION

So far in the paper it has been assumed that a turbo charger or compressor is able to generate whatever boost is required at pretty good efficiency. In reality a positive displacement compressor will have an internal compression ratio and thus an optimal pressure ratio. Any deviations from that lower its efficiency. Also a practical mechanical drive of a compressor probably has a fixed gear ratio, limiting the ability to adjust boost. Turbo chargers tend to generate low boost at low speed and high boost at high speed.

All of these problems are apparent also for conventional engines, but for boost pressures below 2 bar they are not as prominent. Furthermore with higher exhaust temperatures turbo charger over-capacity is not as expensive.

The best boosting solution for an HCCI engine very much depends on the application. For single speed applications a turbo charger can be very well tuned. In such a case two-stage compression could be used and the gas exchange efficiency would probably not be load limiting. In stead PCP would be the design parameter.

For an automobile application high part-load efficiency and good transient performance is required. Full torque should be available from low to high speed. Here a two-stage turbo-charging concept is probably not easily designed. A VGT turbo charger could be a good choice if able to accommodate boost requirements in the full speed range. A mechanical compressor would provide boost at any speed, but at high cost. Many of the car manufacturers are looking into the possibilities of mode change to SI at high load – a way of avoiding the boost problem.

The truck application probably provides the biggest challenge for HCCI. Here full load is required over the entire speed range and high load efficiency is very important. A charging system for an HCCI truck engine would have to cope with the problem of both avoiding over boost at low load and high speed be able to supply full boost at low speed.

CONCLUSION

The potential of high load HCCI through boost is evaluated and discussed, through simulations and in some cases supported by experiments. It is shown that the dilution required to limit NO_X emissions from the engine, forces much more boost to be used compared to conventional engines for the same load. The high boost also generates higher pressure during the cycle.

Turbo charger efficiency is shown to be very important for HCCI to reduce pumping losses, especially when high boost is required. Two-stage turbo charging with inter-stage cooling is found to be very attractive for high load HCCI applications.

It is certainly possible to achieve high load HCCI by applying boost. If brake thermal efficiency is not very important at peak load, PCP will probably be the load-limiting factor. However, in most cases gas exchange efficiency will be load limiting. In these cases nothing will be gained by extending the PCP limit to the exotic levels above 250 bar mentioned in the paper.

All the work, experiments and simulations, is done at, or close to, the NO_X limit of cycle temperature. It is very important for the HCCI engine to stay at this line when boost is required, i.e. not to “over boost” and waste energy on pumping work that is not needed. This requires good control of the turbo charger, e.g. by the use of VGT.

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DEFINITIONS, ACRONYMS, ABBREVIATIONS

BDC	Bottom Dead Center
BMEP	Brake Mean Effective Pressure
c_p	Specific heat at constant pressure
c_v	Specific heat at constant volume
CI	Compression Ignition
DICI	Direct Injection Compression Ignition
EGR	Exhaust Gas Recirculation
FuelMEP	Fuel Mean Effective Pressure
FMEP	Friction Mean Effective Pressure
HC	Hydro Carbons (unburned)
HCCI	Homogeneous Charge Compression Ignition
IMEP	Indicated Mean Effective Pressure
IMEPg	Gross Indicated Mean Effective Pressure
IMEPn	Net Indicated Mean Effective Pressure
NO_X	The sum of nitrogen oxides
OTE	Overall Turbo Efficiency
PCP	Peak Cylinder Pressure
PMEP	Pumping Mean Effective Pressure
QMEP	Heat Release Mean Effective Pressure
SI	Spark Ignition
TDC	Top Dead Center
γ	Ratio of specific heats (=c_p/c_v)
η_{BRAKE}	Brake thermal efficiency
η_{COMB}	Combustion efficiency
η_{MECH}	Mechanical efficiency
η_{GE}	Gas exchange efficiency
η_{THERM}	Thermodynamic efficiency
λ	Relative air-fuel ratio