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A Turbo Charged Dual Fuel HCCI Engine

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ABSTRACT

A 6-cylinder truck engine is modified for turbo charged dual fuel Homogeneous Charge Compression Ignition (HCCI) engine operation. Two different fuels, ethanol and n-heptane, are used to control the ignition timing. The objective of this study is to demonstrate high load operation of a full size HCCI engine and to discuss some of the typical constraints associated with HCCI operation. This study proves the possibility to achieve high loads, up to 16 bar Brake Mean Effective Pressure (BMEP), and ultra low NO_x emissions, using turbo charging and dual fuel. Although the system shows great potential, it is obvious that the lack of inlet air pre heating is a drawback at low loads, where combustion efficiency suffers. At high loads, the low exhaust temperature provides little energy for turbo charging, thus causing pump losses higher than for a comparable diesel engine. Design of turbo charger therefore, is a key issue in order to achieve high loads in combination with high efficiency. In spite of these limitations, brake thermal efficiencies and power rating close to those of the original diesel engine are achieved with significant reduction in NO_x emissions. The maximum efficiency is 41.2%, which is slightly lower than for the original diesel engine.

INTRODUCTION

Homogeneous Charge Compression Ignition (HCCI) is a hybrid of the well-known Spark Ignition (SI) and Compression Ignition (CI) engine concepts. As in an SI engine, a homogeneous fuel-air mixture 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 as in a CI engine. 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 HC emissions at part load operation. Later studies on 4-stroke engines have shown that it is possible to achieve high efficiencies and low NO_x emissions by using a high compression ratio and lean mixtures [7-21]. 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. [21]. 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 [22].

Since the homogeneous mixture auto ignites, combustion starts more or less simultaneously in the whole 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. Without sufficient mixture dilution, problems associated with extremely rapid combustion and knocking-like phenomena will occur. On the other hand, an overly lean mixture will result in incomplete combustion or even misfire.

The air-fuel ratio affects the timing of the combustion onset, which is also strongly influenced by the inlet temperature, fuel properties and compression ratio. In this study the compression ratio is constant, the inlet temperature is uncontrolled but fairly constant, and fuel properties are varied to maintain correct ignition timing.

The objective of this study is to demonstrate the potential of high-load, turbo-charged, HCCI and to discuss how the characteristics of HCCI affect the design of auxiliary systems such as turbo charger, Charge Air Cooler (CAC) etc. The performance is mapped for the tested engine, and compared to the performance of the original diesel engine. Emissions characteristics under high load HCCI are also investigated.

EXPERIMENTAL APPARATUS

The engine used is a modified six-cylinder Scania DSC12 turbo charged diesel engine. The engine is mainly used for truck applications. The original diesel injection system is removed and replaced by a low-pressure sequential port fuel injection system. The engine has four-valve cylinder head and therefore two inlet ports per cylinder. The injection system can thereby supply two fuels to each cylinder, one in each port. In 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.

Since the two fuels are injected in different ports, stratification between the two fuels could be expected. Engine tests comparing operation with a mixture injected into one port to operation with the same mixture injected as separate fuels into two ports do not show any significant difference in engine behavior though.

The intake system is fitted with a city-water cooled CAC. The water temperature is around 15°C and the temperature of the inlet air ranges between 15 and 55°C depending on speed and load, due to the limited size of the CAC. At lower loads water circulation is turned off to allow higher inlet temperature and improved combustion efficiency.

In previous studies, [22, 25], inlet air pre heaters have been used, but because of leakage problems at high inlet pressure, the heaters are excluded from this study. Figure 1 shows the schematic sketch of the engine system.

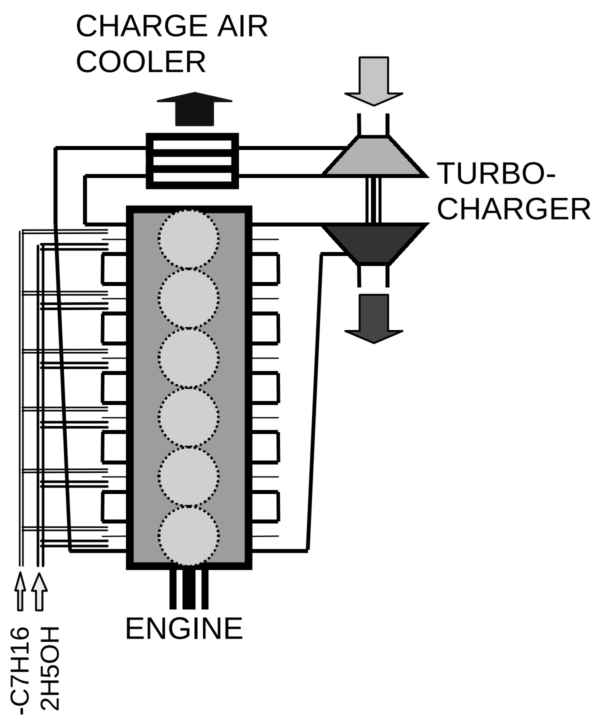

Figure 1 Engine system.

Each cylinder is equipped with a cylinder pressure sensor to allow monitoring of the combustion. The pressure data are also used online for combustion control and computation of indicated parameters.

Because of the low exhaust gas temperature, the diesel-engine turbo charger is replaced in order to generate 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 a low volume flow. 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 both pistons and cylinder heads unchanged. The properties of the engine are summarized in Table 1.

Table 1 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

An emission measurement system sampling exhaust after the turbine is used to measure the concentrations of O_2, CO, CO_2, HC, NO_x and NO. The Flame Ionization Detector (FID), measuring HC, is calibrated using CH_4. The response factor, compared to the reference of C_3H_8, of CH_4 is 1.06, according to the manufacturer [23]. Ethanol has a response factor of 0.84 and n-heptane 1.03. These factors have not been taken into account in the calculations below.

Thermocouples, measuring the stagnation temperature, are installed in each exhaust port (one per cylinder), providing readings of exhaust temperature for each cylinder individually.

The fuel injection system allows the use of two different fuels. In earlier studies with the same engine, [22] and [25], the combinations of n-heptane and gasoline, have been used. However, these combinations do not allow octane numbers high enough for the present study. With a compression ratio of 18:1 pure isooctane or gasoline will auto ignite too early at high load. In this study, combinations of n-heptane and ethanol are used because of the high octane rating of ethanol.

STRATEGY OF ENGINE OPERATION

Since no inlet air heating is available, the only way to control the inlet air temperature is by charge-air cooling using the CAC described above. No attempt is made to use this for active inlet temperature control. For very low loads, the cooling water flow is turned off and for other cases maximum cooling is applied.

In the absence of inlet air temperature control the proportion between the two fuels, n-heptane and ethanol, is the only means of controlling the combustion timing for a given load. This control is performed automatically by the closed loop control system used earlier by Olsson et al. in [25]. The control system uses cylinder pressure to calculate the angle of 50% heat release. The ratio between the two fuels is adjusted to move this angle to the set point. Adjusting the total amount of fuel performs load control.

This control system is optimized for the fuel combination of n-heptane and isooctane. Since it is not tuned for the fuel combination used here the dynamic performance of the control system is very poor. However, for static operation the performance is good enough and provides very stable combustion timing.

Normally the choice of combustion timing is a trade-off between thermodynamic use of heat, heat losses, and combustion efficiency in order to achieve maximum brake thermal efficiency. Assuming isentropic expansion, the heat release should appear around Top Dead Center (TDC) for maximum efficiency. Retarded combustion timing decreases combustion efficiency, but also decreases heat losses. Normally these factors result in an optimum angle for 50% heat release a few degrees after TDC. However, for high load situations, engine noise, rate of pressure increase and peak pressure approach their limits. To further increase the load, the combustion timing has to be retarded from the optimum in efficiency.

At high loads combustion timing is selected as early as possible subject to the limits on peak pressure and rate of pressure increase. At lower loads timing is selected at an earlier angle, but no effort is made to find the optimum value. It should be mentioned that timing of combustion affects efficiency less in an HCCI engine than in an SI or CI engine. This is due to the fast heat release of HCCI, which makes a change in combustion timing from optimum comparable to a small change in compression ratio for ideal constant-volume combustion.

The total amount of fuel per cylinder and cycle is set equal for all cylinders and is selected to achieve the desired load. In this way, the load control part of the closed loop control system is disabled.

The engine is started by having the dynamometer motor the engine at desired speed while gradually increasing the injection of n-heptane until combustion occurs. The total amount of fuel is then increased slowly, while the timing control loop changes the fuel proportions, until the desired load is reached.

RESULTS

PERFORMANCE

Turbo charging of the engine increases inlet pressure and allows higher loads to be reached. Figure 2 shows how boost pressure is developed as engine speed and load increase. A maximum of 16 bar BMEP is reached at 1800 rpm, which is the same as the maximum rated BMEP for the Scania diesel engine at 1900 rpm.

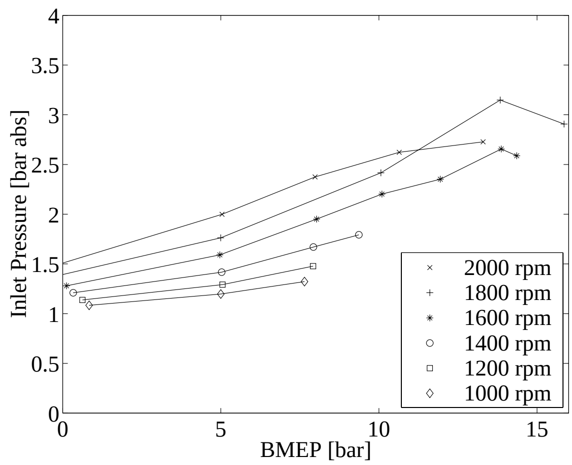

Figure 2 The effect of turbo charging is shown as an increased inlet pressure at high engine speeds and high loads.

Brake torque increases with speed and reaches its maximum at 1800 rpm. At this point maximum fuel capacity of the system was utilized, and therfore maximum brake torque decreased with further increase of engine speed. This limitation is due to the fuel injectors selected for the application. A different choice of injectors would allow for increased torque at higher speeds as well. At high speeds and loads, inlet temperature is higher, due to limitations of the CAC, and the point is reached where auto-ignition at high load, occurs without the use of n-heptane. In this situation load cannot be increased further, since this would cause premature ignition and engine damage.

Inlet temperature is an essential factor for HCCI operation, since auto ignition is very temperature dependent. Figure 4 shows how inlet temperature varies with engine speed and load. The temperature is a function of the characteristics of the CAC and the flow and temperature of the air leaving the turbo compressor.

At higher loads, the fuel mixture consists mainly of ethanol, and vaporization of the fuel will lower the charge temperature significantly. However, since the fuel is port injected, vaporization takes place both in the port and in the cylinder. The temperature decrease due to this effect is not measured.

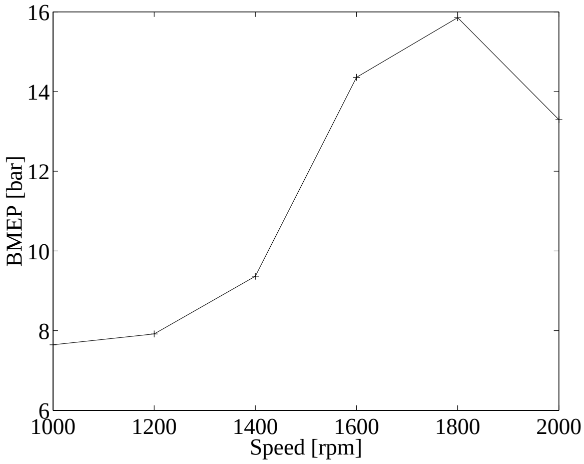

Figure 3 Maximum load versus engine speed.

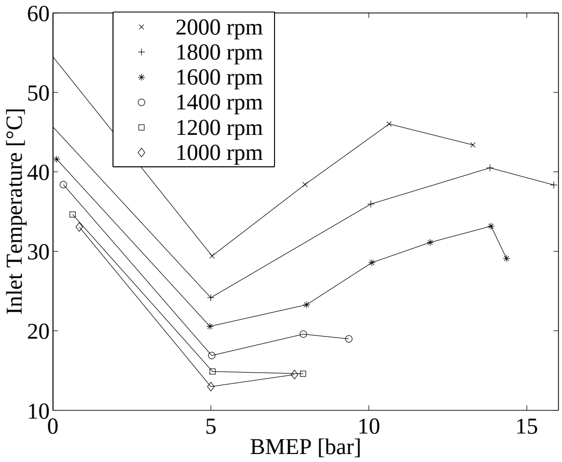

Figure 4 Inlet air temperature for different speeds and loads. At the lowest loads, cooling is turned off and the inlet temperature is the same temperature as after the turbo compressor.

The relative air-fuel ratio, lambda, shown in Figure 5, approaches 2 for the highest loads. This is rich compared to previous reported values for HCCI e.g. in [18, 20]. The limiting factors here are of course the maximum cylinder pressure, the maximum rate of pressure increase, the noise, and the NO_x emissions. All of these deteriorate as the mixture gets richer. However, if the combustion timing can be retarded, and still be stable, the limit can be extended.

For low loads, extremely lean mixtures are used, with lambda close to 7.5. These mixtures are possible to run, but combustion efficiency deteriorates. Because of the small turbine, significant boost is achieved, at 2000 rpm around 0.5 bar, even at zero torque. By use of a waste gate or Variable Turbine Geometry (VTG), this boost could be removed, thus making the mixture richer and improving combustion quality.

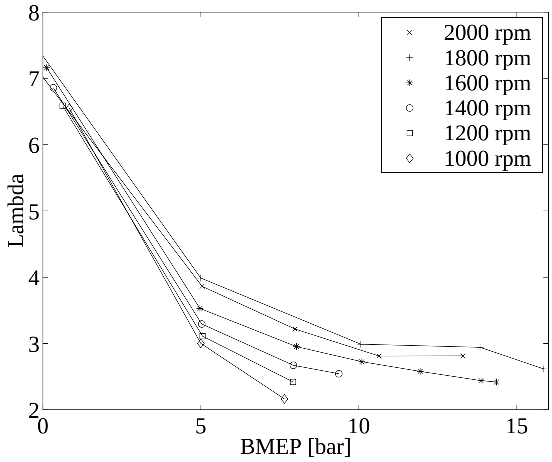

Figure 5 Relative air-fuel ratio, lambda, shown as a function of load and speed.

Brake thermal efficiency is shown in Figure 6. The efficiency peaks at 41.2% at 1000 RPM and 9 bar BMEP, and is comparable to that of CI engines and previously reported results for HCCI [22]. At high loads and speeds the efficiency ranges between 35 and 40%, which is somewhat lower than expected, but pump losses limit the efficiency. The same engine in a diesel truck application claims a maximum brake thermal efficiency of 45.3%.

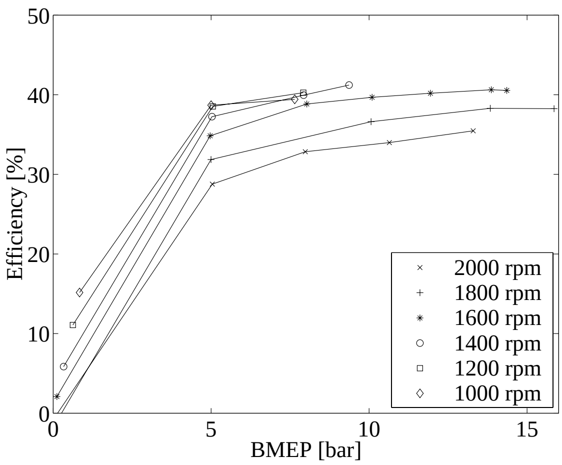

Figure 6 Brake thermal efficiency for different loads and speeds.

As mentioned above, an HCCI engine is limited in load by the rough running and knock-like behaviour at low lambda values. To obtain loads as high as 16 bar BMEP at an absolute inlet pressure of 3 bar and without the use of Exhaust Gas Recirculation (EGR), fairly high stresses on the engine have to be accepted.

Figure 7 shows a P-V diagram for a typical cycle at high load. Note that the graph is not an average of several cycles, but one cycle typical for this load and speed. Figure 8 shows the late timing of combustion, 50% of the heat release was set to 13° after TDC. This allows peak pressure to be kept around 200 bar. The late timing also explains the shape of the curve just after TDC, where the pressure actually decreases before it starts to increase due to combustion.

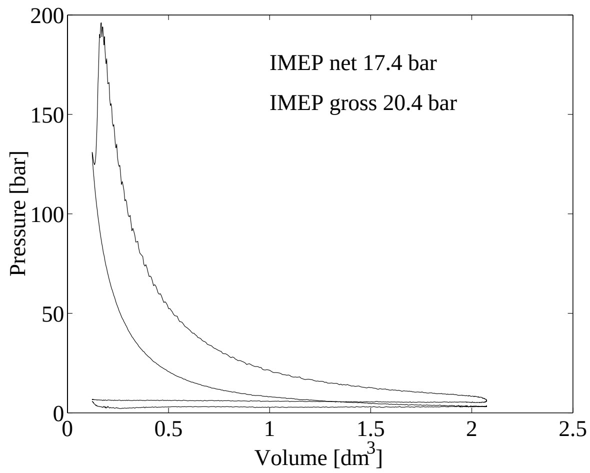

Figure 7 Pressure - Volume diagram for a typical cycle at high load and 1800 rpm.

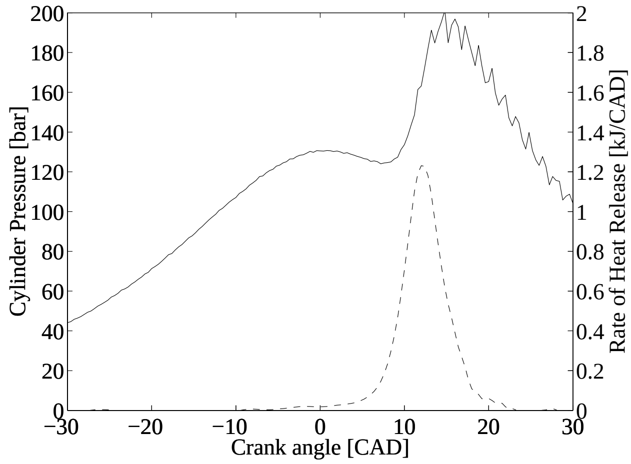

Figure 8 Cylinder pressure (solid line) and rate of heat release (dashed line) as functions of crank angle. Same cycle as Figure 7. The rate of heat release is filtered whereas the cylinder pressure is not.

EMISSIONS

This study investigates the dependence of pollutant levels on engine load and speed. Some of the apparent randomness in the pollutant levels is due to the fact that the exhaust composition depends on the combustion timing as well. This dependence was investigated by J-O Olsson et al. in [22]. In order to keep the presentation clear, this study is limited to the load-speed dependence of the pollutants.

The specific HC emissions are shown in Figure 9. The general trend is decreasing specific HC emissions with increasing load. Some anomalies are caused by the communication problems associated with the fuel injection controllers. This problem, which is discussed in [25], causes errors in the injected fuel mass and thus increased HC emissions and, in severe cases, even misfire, see Figure 10. The problem is more likely to occur for high loads, when injection durations are long, which reduces the time window in which communication can take place between computer and controllers.

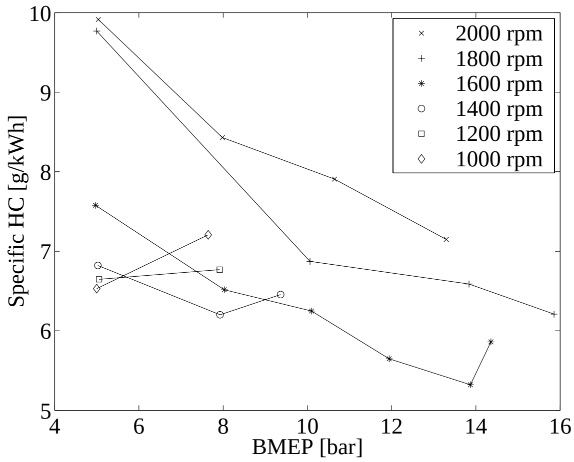

Figure 9 Specific HC emissions as a function of BMEP for different speeds.

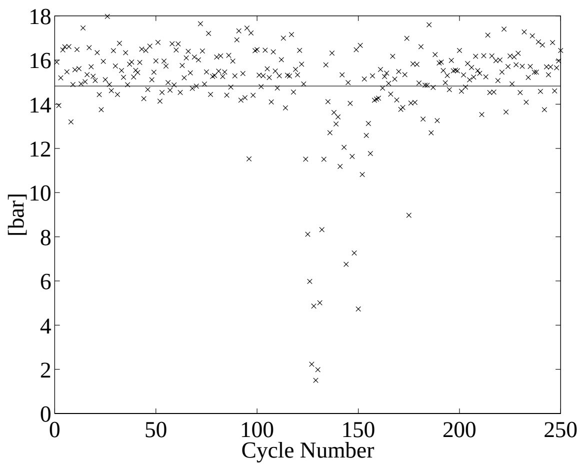

Figure 10 IMEP net versus cycle number. The variation in IMEP illustrates the communication problem associated with the injection controllers.

The specific CO emissions are shown in Figure 11. The same trend as for HC emissions is found but even more pronounced.

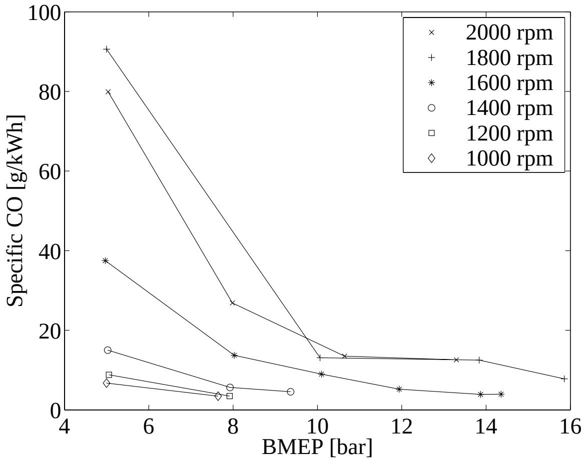

Figure 11 Specific CO emissions as a function of BMEP for different speeds.

Previous studies [22] with moderate engine load indicate a potential for extremely low NO_x emissions using the HCCI concept. This present study shows that this property of HCCI engines holds even for high values of BMEP. Figure 12 shows the dependence of specific NO_x on engine speed and load. A clear trend of increasing specific NO_x with engine load is noted, although the level of specific NO_x is extremely low even at 16 bar BMEP. The highest recorded NO_x concentration in this study is below 4 ppm.

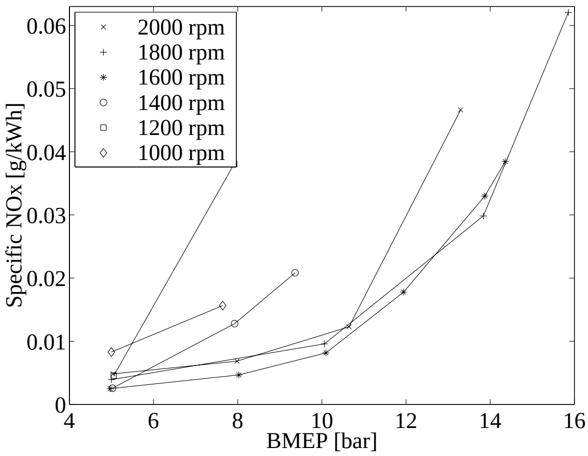

Figure 12 Specific NO_x emissions as a function of BMEP for different speeds.

DISCUSSION

GAS EXCHANGE

In these experiments efficiency suffers from high pump losses. Comparing inlet pressure to exhaust pressure, Figure 13, exposes the turbo charging as an important factor. In a diesel engine, turbo charging is known to be able to produce an inlet pressure higher than the exhaust back pressure. In the studied HCCI engine the opposite occurs.

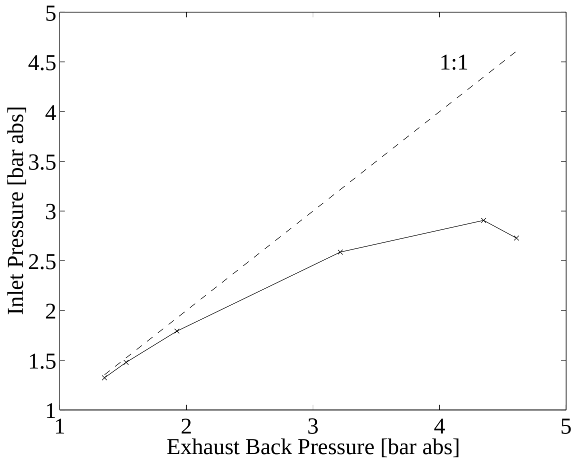

Figure 13 Inlet pressure versus exhaust back pressure at full load as speed increases from 1000 to 2000 rpm.

Exhaust back pressure higher than inlet pressure results in pumping work. In Figure 14 the Pump Mean Effective Pressure is compared to the BMEP. Comparing Figure 13 and Figure 14 shows that most of the pump work originates from the difference between inlet and exhaust pressure.

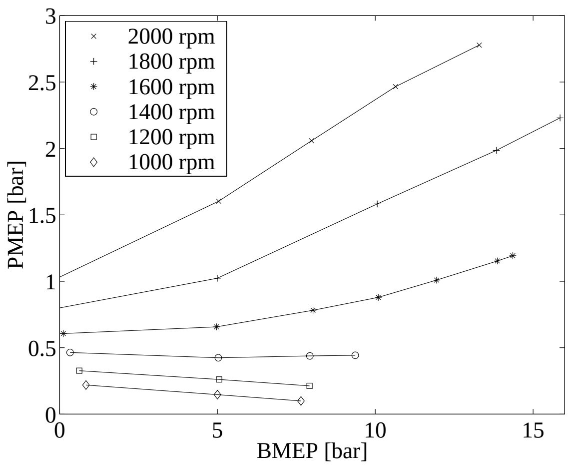

Figure 14 PMEP for different engine speeds and loads.

The pump losses result in a big difference between net and gross indicated thermal efficiency for the engine. Figure 15 shows the gross indicated efficiency and Figure 16 the net indicated efficiency. Comparing the two graphs gives an understanding for how the pump losses affect the thermal efficiency of the engine. The dip in gross indicated efficiency at 1000 RPM, 8 bar BMEP is a result of the injector communication problem causing misfire on some cylinder. This intermittent problem has a heavy impact on the indicated efficiencies, while hardly affecting the BMEP due to the low-pass character of the dynamometer. Another interesting characteristic is that the gross indicated efficiency is fairly independent of engine speed whereas the net indicated efficiency shows a strong dependence on engine speed. This is a result of the increase in exhaust back pressure with speed, caused by the small turbine.

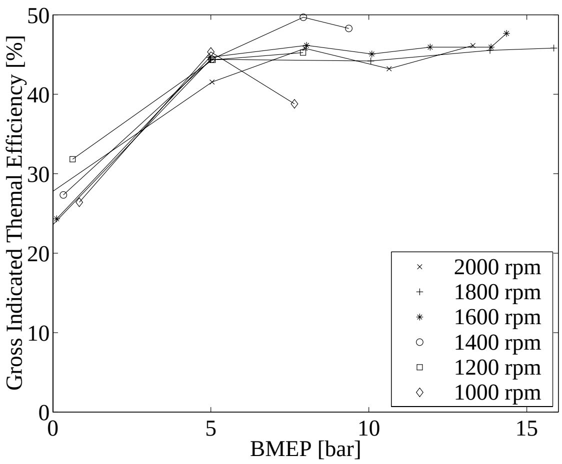

Figure 15 Gross indicated thermal efficiency for different engine speeds and loads.

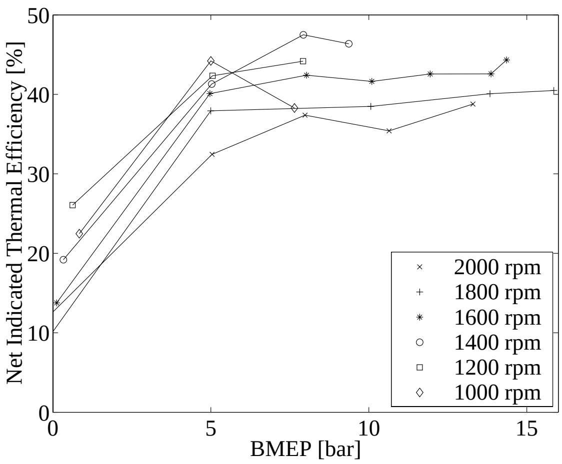

Figure 16 Net indicated thermal efficiency for different engine speeds and loads.

The big difference between inlet and exhaust pressure is mainly explained by the low exhaust temperature, Figure 17. The low temperature is due to the high dilution of the charge and the rapid combustion with very low heat release late in the expansion stroke, both characteristics of HCCI. To extract a certain amount of work, the pressure drop has to be higher for a cold gas than for a hot gas.

One way to understand this is to consider the expansion work:

$$\dot{W}_{Turbo} = \frac{\partial}{\partial \tau}\int_{p_{exh}}^{p_{amb}} p \cdot dV = \dot{V}_{exh} \cdot \frac{p_{exh}}{\gamma - 1} \cdot \left(p_{exh}^{\frac{\gamma-1}{\gamma}} - p_{amb}^{\frac{\gamma-1}{\gamma}} \right)$$

Clearly, for a given mass flow through the engine the volume flow will be lower if exhaust temperature is low. To compensate for this a higher pressure drop is needed to obtain the same work.

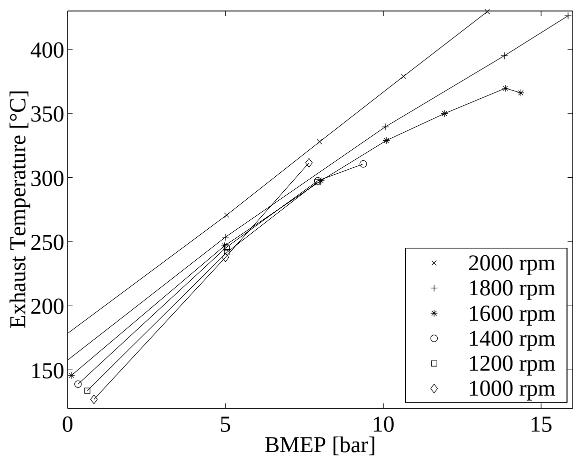

Figure 17 Exhaust temperature measured in the exhaust ports. The thermocouples used measures stagnation temperature.

Also the efficiency of the turbo charger is important for the pump losses. Figure 18 shows estimated total efficiency of the turbo charger. The definition of this efficiency is as follows:

$$\eta_{turbo} = \frac{\dot{m}_{air} \cdot \Delta h_{s,air}}{\dot{m}_{exh} \cdot \Delta h_{s,exh}}$$

where:

$\Delta h_{s,\,air}$ isentropic enthalpy increase of the inlet air from atmospheric conditions to the measured inlet pressure.

$\Delta h_{s,exh}$ isentropic enthalpy decrease from the measured exhaust pressure and temperature before turbine to atmospheric pressure.

$\dot{m}_{air}$ air mass flow through compressor

$\dot{m}_{exh}$ exhaust mass flow through turbine

Unfortunately, the turbo charger is not equipped with means for monitoring rotor speed. Thus, it is not possible to see where in the turbo maps the engine is operating. Only the overall performance of the turbo charger can be discussed.

The values found indicate surprisingly high efficiencies for the turbo charger. It should be noted though, that available exhaust enthalpy is based on exhaust temperature measured in the exhaust ports. Temperatures measured with thermocouples are lower than the mass-averaged temperature. Heywood [24] estimates the difference to about 100 K for SI engines. The difference should be somewhat smaller for an HCCI engine because the lower temperature during expansion results in lower pressure at exhaust valve opening and therefore a smaller mass fraction in the high-temperature blow-down gases. The result of this systematic error is an overestimation of turbo charger efficiency in the order of a few percentage points.

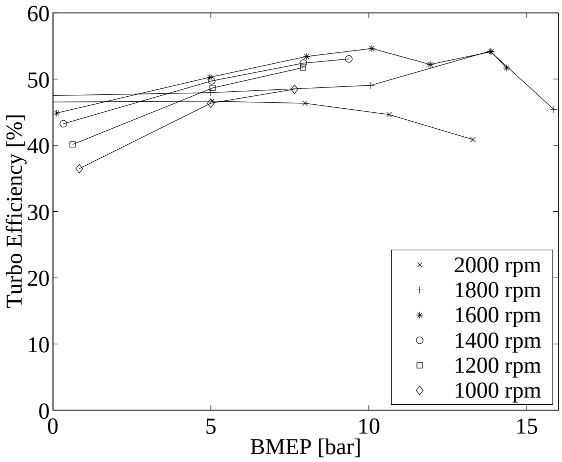

Figure 18 Total efficiency of the turbo charger.

The maximum efficiency of the turbine is 75% and the compressor peaks at 78%. This gives a maximum overall turbo efficiency of about 58%. Compared to the turbo efficiencies measured at 1600 rpm the conclusion must be that a better matched turbo charger probably could somewhat decrease pump losses, but not very much. Considering the low exhaust temperatures, turbo charging will most likely always give high pump losses at some operating points.

In a CI engine, turbo charging is normally a good thing even if the higher inlet pressure is not needed to produce the desired torque – pump losses are normally negative. In an HCCI engine though, where turbo charging results in high pump losses, it is important not to “over-boost” to keep efficiency high.

For an engine operating over a wide range of speed and load, e.g. in a vehicle application, pump losses from turbo charging will cause special constraints. The characteristics of a standard turbo charger, designed to give desired torque versus speed performance of the engine, will result in very high pump losses for all operating points where “over-boost” is produced. For this reason it might be of interest to consider VTG or electrically assisted turbo chargers in some applications.

Another angle to attack the problem is to maximize the power output for a given mass flow through the engine, i.e. to use less diluted charges. One solution, that has proven to be successful, is the use of EGR. Christensen et al. [19] showed that for a naturally aspirated engine the load could be increased by about 25% by the use of EGR. At the same time combustion efficiency improved and exhaust temperature increased. For the turbo-charged case Chrisensen and Johansson [26] showed that specific HC emissions also could be reduced by the use of EGR.

COMBUSTION EFFICIENCY

Due to leakage problems at high intake pressures, preheating of the inlet air is excluded from this study. Previous measurements, [22], indicate that CO emission levels drop significantly with increased inlet air temperature, which should be kept in mind when evaluating the results of this study.

The low inlet temperature can explain the high specific CO emissions for low loads Figure 11. Previous measurements with a setup corresponding to [22], where the inlet temperature is varied can be seen in Figure 19. These should be compared with the results from the present study, Figure 20. The specific HC emissions in Figure 9 shows the same trend as for CO but not quite so pronounced.

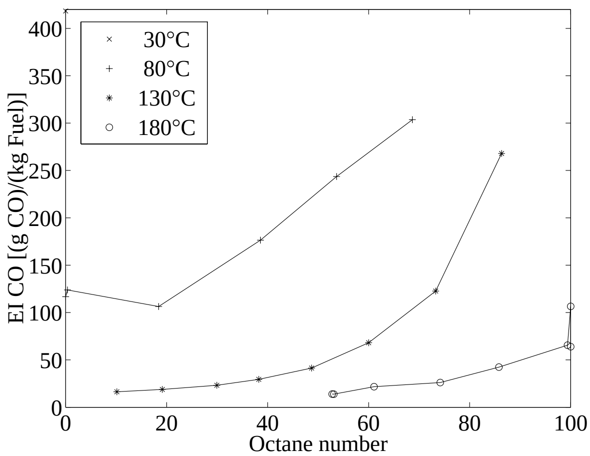

Figure 19 Emission index as a function of octane number for four different inlet temperatures. The load is held constant at 1.6 bar BMEP and timing is altered by changing octane number. Note the data point in the upper left corner.

The exhaust temperature in Figure 17 can be used to get a rough estimate of the combustion temperature, by assuming polytropic expansion from TDC to Exhaust Valve Opening (EVO).

$$\frac{T_{EVO}}{T_{TDC}} = \left(\frac{V_{EVO}}{V_{TDC}}\right)^{1-\kappa}$$

At low loads the exhaust temperature is approximately 150°C, which according to the discussion above corresponds to a cylinder gas temperature of approximately 250°C at EVO. Applying the polytropic equation yields a temperature at TDC of 1100°C. This temperature is extremely low, and explains the poor combustion efficiency at low loads. The polytropic exponent of 1.4 was used for this estimation.

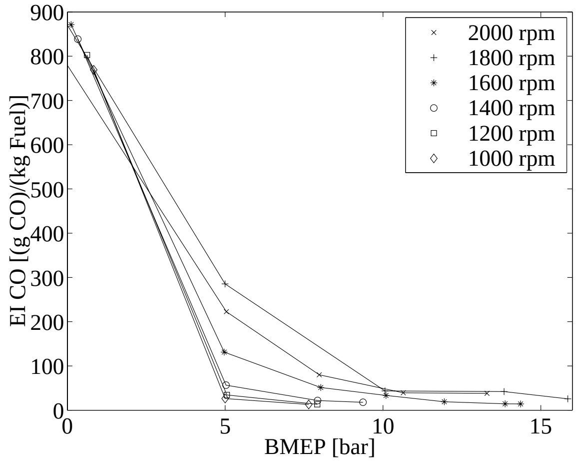

Figure 20 Emission Index for CO as a function of BMEP for different speeds.

It is clear from the figures that the inlet temperature has a very strong influence on CO emissions at low loads and to obtain reasonable levels of CO and HC, inlet air preheating should be used. An increase of inlet temperature means higher combustion temperature and lower lambda, which improves combustion efficiency. Also higher overall temperature is desirable when a catalyst is used for exhaust gas after-treatment.

Figure 21 shows that combustion efficiency improves rapidly with increasing load. For high loads efficiencies around 95% are reached.

HIGH LOAD EMISSIONS

The specific NO_x emissions are very low, even though fairly high loads are achieved. One reason for the low NO_x emissions is the fact that lambda always exceeds 2, even for the highest loads. With such a diluted and homogeneous mixture a larger mass of bulk gas takes up the heat from the combustion. This results in lower maximum combustion temperature, which results in the extremely low NO_x emissions shown in Figure 12. Another reason for low NO_x is the homogeneous charge in HCCI with no hot spots that increase NO_x.

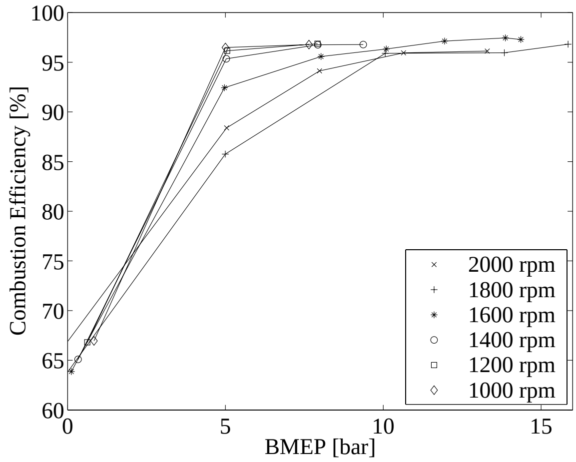

Figure 21 Combustion efficiency at different loads and speeds.

The timing at high loads in this test is retarded to limit the maximum cylinder pressure. This also lowers the combustion temperature and hence the NO_x emissions. But previous studies, [22], with optimized combustion timing also show very low NO_x emissions. This proves that the HCCI concept is very effective overall in producing low NO_x emissions. Figure 17 shows exhaust temperatures for a range of loads and engine speeds and they are low compared to an Otto engine where exhaust temperatures typically range from 400-600°C [24].

CONCLUSIONS

The study shows that using turbo charging on an HCCI engine is an effective way to reach high loads. A BMEP of 16 bar is reached, compared to 21 bar for the same engine, with diesel cycle operation in a truck application. Efficiency peaks at 41.2% compared to 45.3% in a truck application. The reduction is mainly due to pump work.

The ability of HCCI to provide high loads with very low NO_x emissions is proven. Concentrations never exceed 4 ppm and brake specific emissions of NO_x keep well below 0.1 g/kWh.

The low exhaust temperature, which is characteristic for HCCI, combined with turbo charging causes pump losses. To keep efficiency high, it is therefore important not to use higher boost than needed to reach desired load.

To keep CO emissions down at low loads, heating of inlet air is needed.

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DEFINITIONS, ACRONYMS, ABBREVIATIONS

ABDC	After Bottom Dead Center
ATDC	After Top Dead Center
BBDC	Before Bottom Dead Center
BTDC	Before Top Dead Center
BMEP	Brake Mean Effective Pressure
CAC	Charge Air Cooler
CI	Compression Ignition
EGR	Exhaust Gas Recirculation
EVO	Exhaust Valve Opening
FID	Flame Ionization Detector
Net IMEP	Net Indicated Mean Effective Pressure
SI	Spark Ignited
TDC	Top Dead Center
VTG	Variable Turbo Geometr