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Published in:
SAE Special Publications

DOI:
10.4271/2003-01-0743

2003

2003-01-0743

The Effect of Cooled EGR on Emissions and Performance of a Turbocharged HCCI Engine

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ABSTRACT

This paper discusses the effects of cooled EGR on a turbo charged multi cylinder HCCI engine. A six cylinder, 12 liter, Scania D12 truck engine is modified for HCCI operation. It is fitted with port fuel injection of ethanol and n-heptane and cylinder pressure sensors for closed loop combustion control. The effects of EGR are studied in different operating regimes of the engine. During idle, low speed and no load, the focus is on the effects on combustion efficiency, emissions of unburned hydrocarbons and CO. At intermediate load, run without turbocharging to achieve a well defined experiment, combustion efficiency and emissions from incomplete combustion are still of interest. However the effect on NO_X and the thermodynamic effect on thermal efficiency, from a different gas composition, are studied as well. At high load and boost pressure the main focus is NO_X emissions and the ability to run high mean effective pressure without exceeding the physical constraints of the engine. In this case the effects of EGR on boost and combustion duration and phasing are of primary interest. It is shown that CO, HC and NO_X emissions in most cases all improve with EGR compared to lean burn. Combustion efficiency, which is computed based on exhaust gas analysis, increases with EGR due to lower emissions of CO and HC.

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 as in a CI engine. The differences 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 efficiencies 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 in combination with cooled EGR. Controlling the amount of internal 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 history, together with the concentration of O_2, different fuel contents and combustion products, govern how combustion is initiated. As a consequence, combustion timing will be strongly influenced by air-fuel ratio, inlet temperature, compression ratio, residual gases and EGR.

In the present study the effect of EGR is studied in a multi cylinder engine. The engine, originally a Scania

D12 truck-size diesel engine, is converted for HCCI operation. The changes to the engine include a double port fuel injection system for ethanol and n-heptane, a turbo charger with a smaller turbine to provide boost in spite of the low exhaust temperature of the HCCI process and cylinder pressure sensors for feedback control.

The EGR fraction is defined as the ratio between the mass flow of recirculated exhausts and the total mass flow entering the engine through the intake. EGR is calculated from the exhaust composition and the concentration of CO_2 in the intake system according to equation 1. [CO_2] is measured on dry gas, but the calculation compensates for the estimated fraction of water in the exhausts and the intake gases. However, if water is condensed and trapped somewhere in the system, this introduces an error in the calculated EGR fraction.

$$EGR = \frac{[CO_2]_{INTAKE}}{[CO_2]_{EXHAUST}} \qquad \text{(Eq. 1)}$$

Three different operating regimes are studied:

First, idle operation at 600 rpm. So far very little experimental data has been reported on HCCI running at idle. Modeling of the chemical kinetics of the combustion [30] has suggested that it is more or less impossible to run the highly diluted charges associated with idle, without producing excessive amounts of CO. Here this is experimentally investigated for different inlet temperatures of the charge and different EGR-ratios.

Second, intermediate load at naturally aspirated condition. These experiments are very similar to those previously reported by Christensen et al. [19]. In this case it is a different engine and the combustion timing is more accurately controlled and more flexible. In these experiments, the engine is run in naturally aspirated mode by using the standard turbo charger, designed for diesel operation. The reasons for running naturally aspirated are to be able to make a better comparison with the paper mentioned above and to make a more controlled experiment, without the possible effect of changing boost pressure. For these experiments the fundamental effects of EGR on efficiency, emissions and exhaust temperature are studied for a range of fuel injection rates.

Third high load at turbo charged condition is studied. At these conditions the main focus is on the ability to achieve high load within the physical limitations of the engine and without producing high levels of NO_X. An effect on exhaust temperature at these conditions will affect the ability to produce boost and thus engine performance.

EXPERIMENTAL APPARATUS

As mentioned above, 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.

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.

Figure 1 shows a schematic of the engine system. Basically the same engine setup has been used in previous papers by the authors [25, 26, 28]. The main difference is the EGR system and the arrangement of heaters and CAC.

The intake system is fitted with a city-water cooled Charge Air Cooler (CAC) and electrically powered heaters. A pair of butterfly valves controls the fraction of the air going through the heaters and the CAC respectively. At high intake temperature all the air is directed through the heaters and the heating power is controlled to achieve desired temperature. As the intake temperature set point is lowered, heating power will approach zero and when necessary, some of the air will be directed through the CAC in order to achieve an intake temperature lower than the temperature after the compressor. The water temperature is around 15°C and thus the temperature of the inlet air can be controlled in a range from 15°C up to about 200°C. The lowest achievable temperature will in practice be higher due to the limited size of the CAC and some leakage of air into the heater path. The highest temperature is limited in order to avoid damage to equipment fitted to the intake system. The intake temperature is measured at the entry of the intake manifold, downstream of the mixing of cooled and non-cooled charge of air and EGR but upstream of fuel injection.

Downstream from the turbine, a T-junction is fitted with a pair of butterfly valves controlling the amount of exhausts recirculated for EGR. This valve pair is manually controlled and not subject to any form of automatic control. Before the exhausts are mixed with fresh intake air, they are cooled in a heat exchanger. The cooling is necessary to avoid damage to the compressor due to excessive temperatures during and after compression. The cooling of the gas before compression also lowers the compression work needed to create boost, i.e. the pumping work is lowered.

However, if the EGR is cooled too much, water may condense as the EGR mixes with the cooler air. Condensed water would cause wear on the compressor and therefore the EGR temperature, after the cooler, is kept at 60°C.

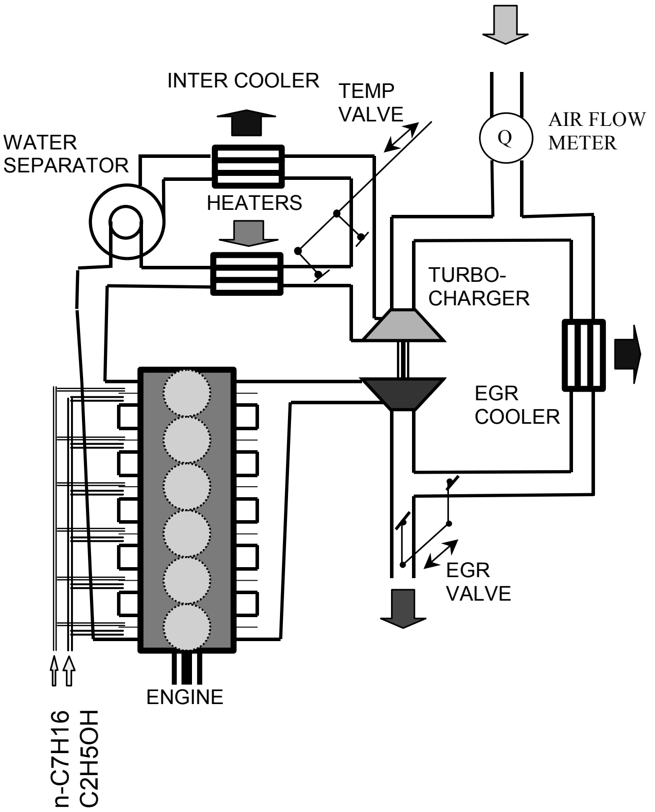

Figure 1. The engine system consisting of an engine with a double port fuel injection system, an EGR system and a system for intake temperature control.

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 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. 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 1.

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 system also measures the concentration of CO_2 in the intake to allow calculation of the EGR fraction. 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 [34]. 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.

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

Since the fuel is ethanol and HCCI can be expected to release other Oxygenated HydroCarbons (OHC) as well [30], the FID can be suspected to underestimate the concentration of organically bonded carbon in the exhausts. This is not compensated for.

When using EGR and charge-air cooling, some of the water is condensed, trapped and separated. This is not accounted for in the calculation of EGR fraction, resulting in an overestimation of EGR fraction at low intake temperatures. It can be expected that some of the HC and OHC, as well as NO_2, will condense and be separated as well. As a result combustion efficiency will be overestimated and NO_X emissions underestimated. In the present study the intake charge is cooled only for the high load turbo charged experiments, and thus the errors described only apply for this section.

Thermocouples, measuring the stagnation temperature, are installed in each exhaust port (one per cylinder), and in the two entrance ports of the turbine. When exhaust temperature is mentioned, it is the average temperature measured in the ports of the turbine, if nothing else is stated. This temperature is normally higher than the temperature measured in the ports, due to time averaging [33].

The fuel injection system allows the use of two different fuels. In earlier studies with the same engine, [25, 26, 28], combinations of n-heptane and gasoline or isooctane, have been used. In this study, combinations of n-heptane and ethanol are used because of the high octane rating of ethanol.

STRATEGY OF ENGINE OPERATION

To obtain a desired operating condition, in terms of speed and load, several options of intake temperature, combustion timing and EGR are available. To limit the number of test points required, a strategy for the intake temperature is applied.

Previous studies have indicated that combustion efficiency improves as inlet temperature is increased. On the other hand higher temperature at the start of the cycle results in higher NO_X emissions. When controlling the engine with two fuels of different ignition characteristics, higher intake temperature also reduces the necessary fraction of the "easy to ignite fuel". Obviously the inlet temperature should be selected as high as possible to achieve highest possible combustion efficiency, avoiding excessive NO_X production or loss of control authority.

The strategy used in this study is to adjust the intake temperature according to the cylinder using the lowest fraction of n-heptane, when the engine is running under closed loop control using fuel-mixing ratio. The temperature is adjusted to obtain 5% of the fuel heat in this cylinder from n-heptane. 5% is considered to provide the control authority needed both to advance and retard the combustion timing. If the fraction is lower, the inlet temperature is decreased, forcing the combustion controller to use more n-heptane to maintain combustion timing. If the fraction is higher, the temperature is increased. Inlet temperature is never allowed to exceed 200°C.

The electrical power required to heat the intake air is in most cases neither included in efficiency nor discussed. It is the opinion of the authors that this heat should be taken from the exhausts in a practical application. However, at idle, the intake temperature sometimes exceeds the exhaust temperature and this situation is discussed in more detail.

In most of the following experiments a combustion-timing sweep is performed without EGR, controlling the temperature as described above. The optimum combustion timing, i.e. the combustion timing giving the highest brake thermal efficiency, is then used for an EGR sweep. At the highest EGR fraction, timing is swept again. The temperature control results in a different inlet temperature for each operating point. EGR fraction is limited by the stoichiometric air/fuel ratio. Because of cylinder-to-cylinder variations, operation at stoichiometric conditions results in some cylinders running fuel rich, thus causing excessive emissions of HC and CO.

At high load and turbo charged condition, the engine operates close to the limit of acceptable NO_X emissions. Combustion stability and pressure rise rate are also close to their limits, preventing combustion timing from being significantly varied. The behavior of the engine is here studied in a sweep of intake temperature without EGR as well as with an EGR fraction of 45%.

EGR CHARACTERISTICS

In the field of HCCI, the main reason to use EGR has been to slow down the rate of combustion [19, 27, 30]. This could be used to achieve high load within the physical constraints of the engine. There are several reasons why EGR should slow down the rate of combustion:

- The recirculated gas contains three-atomic molecules, H_2O and CO_2, with lower ratio of specific heats, $\gamma = C_P/C_V$. As a result the temperature after compression is lower and the combustion is retarded. However, in this study the intention is to study the effect of EGR at a given (optimal) combustion timing. This can be obtained either by changing fuel composition or increasing intake temperature. In most cases presented in this paper, the intake temperature is increased for increased EGR ratio.

- The specific heats, c_p and c_v, of the recirculated gas are higher compared to air. Therefore the heat released by the combustion results in a lower increase in temperature. Since reactions are strongly temperature dependent this slows down the rate of combustion.

- The concentration of O_2, one of the reactants, is lowered and the concentrations of CO_2 and H_2O, the products, are increased. This slows down the reactions in the oxidizing direction and speeds up the reactions in the reverse direction.

- H_2O is a third body in a three body chain-terminating reaction ($H + O_2 + H_2O \rightarrow HO_2 + H_2O$) [30]

Considering these effects, suggesting a lower rate of heat release, EGR is obviously tempting to try at high load. But reasons to use EGR at intermediate and low loads exist:

Combustion efficiency, often below 95%, is an issue for HCCI engines regarding brake thermal efficiency. Previous studies have reported that EGR improves combustion efficiency, primarily by reducing the specific HC emissions [19, 27]. Considering the complete engine system, including the EGR loop, higher EGR fraction results in a lower mass flow through the engine system, thus at a constant concentration of an emission component in the exhausts, the specific emissions are reduced. To understand how the concentration is affected, the in cylinder phenomena have to be considered.

Consider an example of an engine operated either without EGR or with 50% EGR. Assume that the fuel in the cylinder is always 1g, and that 0.9g always burns and 0.1g leaves the cylinder unburned. Without EGR the fuel supplied through the fuel system is 1g and the combustion efficiency is 0.9g/1g = 90%. With 50% EGR, half of the unburned fuel is recirculated and only 0.95g is supplied from the fuel system. The combustion efficiency, in terms of burned fuel divided by added fuel, is then 0.9g/0.95g = 94.7%. This effect can be thought of as a second chance for the fuel to burn.

HC emissions from HCCI combustion at moderate and high load are shown to originate primarily from crevices [22]. The amount of HC leaving the cylinder is therefore, as a first order approximation, controlled by the amount of fuel trapped in the crevices at the time when bulk reactions freeze. The difference in the cylinder between EGR and no EGR is the oxygen pressure and the charge temperature. The oxygen pressure has a very small impact on the reaction kinetics as long as the mixture is lean. If the strategy for running the engine includes increased inlet temperature to maintain combustion timing when EGR is introduced, the difference in charge temperature is more or less negligible concerning the time for freezing reactions. Accordingly the concentration of HC in the exhaust is approximately the same with and without EGR. Note that this conclusion is valid only when inlet temperature is increased with increased EGR fraction.

The formation of CO is much more complex and the effect of EGR cannot easily be predicted. CO is believed to form close to the walls where the temperature is high enough for the oxidation of HC to start but the cooling from the wall prevents complete oxidation to CO_2 [31]. For this mechanism, wall temperature, mixing between boundary layer and bulk, and bulk temperature history all have a strong impact on CO emissions. For highly diluted mixtures bulk quenching of the reactions may occur, resulting in extremely high CO emissions [30]. Except maybe when bulk quenching occurs, recirculated CO should oxidize to CO_2, not contributing to increasing CO concentration. However, effects from EGR on temperature history, even if small, may have a significant impact on CO formation. Previous studies show that CO emissions sometimes increase and sometimes decrease when EGR is introduced [19]. Normally emissions of CO are low enough not to have a lot of impact on combustion efficiency, but as a regulated emission component in itself it is still important.

The rate of NO formation is normally controlled by the temperature history and the concentrations of N_2 and O_2. For lean mixtures the concentrations of N_2 and O_2 typically do not change by more than one order of magnitude. The temperature dependence though is exponential and the rate of NO formation changes with several orders of magnitude for a comparably small change in temperature. Since all operating points in this study are lean of stoichiometric, the effect of EGR on NO formation should primarily originate from its effect on temperature.

Apart from the potentially positive effects of slowing down the rate of combustion at high load and improving combustion efficiency, EGR also has one negative effect. As mentioned above EGR lowers γ and this has an impact on efficiency. For an ideal Otto cycle at compression ratio 18:1, the efficiency is lowered by approximately 10 percentage points when decreasing γ from 1.4 to 1.3.

DEFINITIONS

Since the test engine for the present study uses two different fuels, ethanol and n-heptane, with two very different heating values, presenting fuel on mass basis is not very relevant. A suitable measure of the fuel usage is the Fuel Mean Effective Pressure (FuelMEP), which is the normalized heat of fuel supplied from the fuel systems in each cycle, defined by equation 2.

$$FuelMEP \equiv \sum_{AllFuels} \frac{\dot{m}_{FUEL} \cdot Q_{LHV,FUEL}}{n/120 \cdot V_{DISP}} \qquad \text{(Eq. 2)}$$

For the present engine, a FuelMEP of 1 bar equals 195 J per cycle and cylinder. 195 J is equivalent to 7.7 mg of ethanol, 4.4 mg of n-heptane or any linear combination of these.

This definition means that FuelMEP is directly comparable to Indicated Mean Effective Pressure (IMEP), gross (IMEPg) or net (IMEPn), and BMEP. Here, net refers to full cycle and gross refers to compression/expansion only. QMEP, defined by equation 3, introduces a similar normalization of the total accumulated heat release per cycle per cylinder. Here Q_{HR} is the accumulated heat release and V_{DISP} is the displacement volume.

$$QMEP \equiv \frac{Q_{HR}}{V_{DISP}} \qquad \text{(Eq. 3)}$$

Friction Mean Effective Pressure (FMEP) is defined as the difference between net IMEP and BMEP, i.e. not including pumping losses:

$$FMEP \equiv IMEPn - BMEP \qquad \text{(Eq. 4)}$$

Since all relevant quantities are now defined on the MEP scale, they can be used to define the efficiencies in the discussion. Brake thermal efficiency is the overall conversion ratio from fuel heat to mechanical power:

$$\eta_{BRAKE} \equiv \frac{BMEP}{FuelMEP} = \frac{P}{\dot{m}_f Q_{HV}} \quad \text{(Eq. 5)}$$

Combustion efficiency is the ratio between accumulated heat release and the heat supplied by the fuel system and is defined by equation 6. However, in practice the total heat released in the cylinder is hard to measure accurately and combustion efficiency is calculated from the exhaust composition instead.

$$\eta_{COMB} \equiv \frac{QMEP}{FuelMEP} = \frac{H_R(T_A) - H_P(T_A)}{m_f Q_{HV}} \quad \text{(Eq. 6)}$$

The thermodynamic efficiency, equation 7, reflects how the released heat is converted to pressure-volume work on the piston for the closed part of the cycle. The thermodynamic efficiency will be affected by parameters such as the compression ratio, the shape of the heat release and heat losses.

$$\eta_{THERM} \equiv \frac{IMEPg}{QMEP} \quad \text{(Eq. 7)}$$

Gas exchange efficiency, equation 8, 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%.

$$\eta_{GE} \equiv \frac{IMEPn}{IMEPg} = \frac{IMEPg - PMEP}{IMEPg} \quad \text{(Eq. 8)}$$

The last step toward mechanical work out of the engine is to convert the indicated pressure work into mechanical work. The efficiency for this, the mechanical efficiency, is the ratio between mechanical work and total indicated work, equation 10.

$$\eta_{MECH} \equiv \frac{BMEP}{IMEPn} = \frac{IMEPn - FMEP}{IMEPn} \quad \text{(Eq. 9)}$$

The brake thermal efficiency can be calculated as the product of the other efficiencies according to equation 10.

$$\eta_{BRAKE} = \eta_{MECH} \cdot \eta_{GE} \cdot \eta_{THERM} \cdot \eta_{COMB} \quad \text{(Eq. 10)}$$

In equations 5 and 6 above, the efficiencies are also expressed in “conventional” terms. Nomenclature and expressions are from Heywood [33]. This is to show that the definitions do not deviate from the normal ones.

IDLE OPERATION

Even if no load is extracted from an idling engine, it is, in most applications, important for the engine to allow idle operation. In many test cycles for vehicles, idle is highly weighted.

One of the advantages of HCCI, compared to SI, is the lower pumping losses at low load. This advantage should be very pronounced at idle. However, simulations by J. Dec indicate vast emissions of CO, HC and OHC [30] for mixtures diluted to idle-like fuel concentrations. The reason is that the combustion will not provide enough energy, when distributed to the complete charge, to keep the temperature high enough long enough for complete oxidation of the fuel, i.e. bulk quenching will occur.

Dec suggests some possible solutions: First, extremely advanced timing results in higher peak temperatures and a longer time of high enough temperature for reactions to progress. The drawback is lower efficiency of the thermodynamic cycle when heat is released before Top Dead Center (TDC). Second, the use of a throttle to reduce the amount of air would enrich the mixture. This solution adds pumping losses, lowering the gas exchange efficiency. Third, the use of stratified charge could potentially provide a locally richer mixture and no fuel in crevices. This method has really no drawbacks except maybe for the practical problems of creating such a charge with the preferred concentration at different operating conditions.

In the present paper the objective is to study the effect of EGR. The engine is operated unthrottled, i.e. inlet pressure is atmospheric. The port fuel injection system does not provide the possibility to create a stratified charge. Instead, it is the method of advanced timing, combined with EGR that is deployed to control combustion efficiency.

In a previous study by the authors essentially the same engine was run at idle without pre heating of the intake air. Combustion timing was maintained by using high fractions of n-heptane [28]. At these conditions combustion efficiency deteriorated to below 65%. Figure 2 shows the combustion efficiency achieved in the present study with and without EGR and using pre heating of intake air. CA50 refers to the crank angle of 50% accumulated heat release.

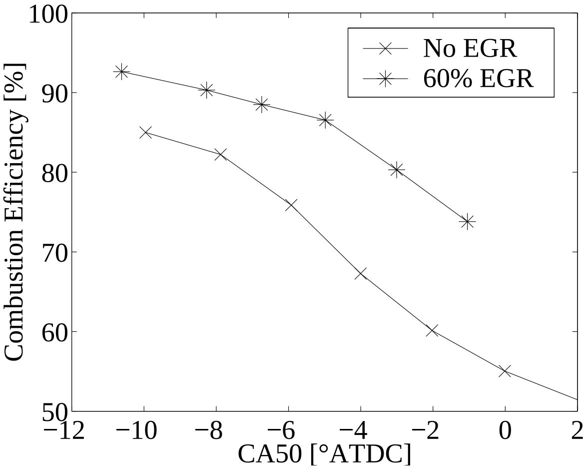

Figure 2. Combustion efficiency at idle with and without EGR.

Even when using extremely advanced combustion timing the combustion is far from complete. But by utilizing a high EGR fraction, combustion efficiency improves significantly. The reason for higher combustion efficiency is mainly the recirculation of unburned fuel. The strong influence of combustion timing is reflected in the FuelMEP needed to keep the engine running. Figure 3 shows FuelMEP with and without EGR.

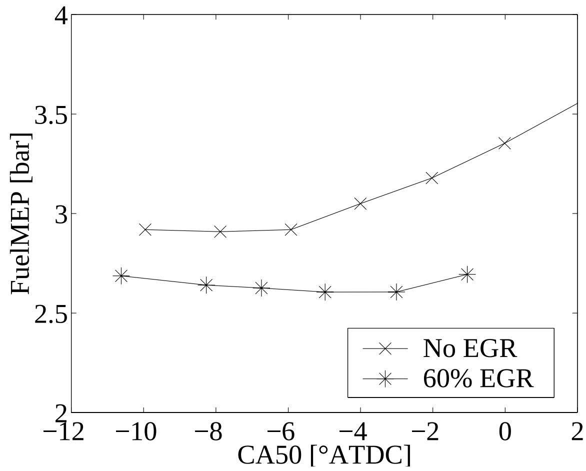

Figure 3. FuelMEP at idle with and without EGR.

The poor combustion efficiency at “late” combustion timing results in an optimum timing as early as 8.7° BTDC, when no EGR is used. The combustion timing giving the lowest fuel consumption is considered the optimum timing. When using EGR, the combustion efficiency trend is the same, although not quite as pronounced, resulting in a later timing as optimum. However, the engine might be run at very advanced timing anyway to improve emissions. Retarding the heat release towards TDC reduces heat losses and avoids compression of the heated gases. This is illustrated by the reduction of the released heat needed to turn the engine, Figure 4.

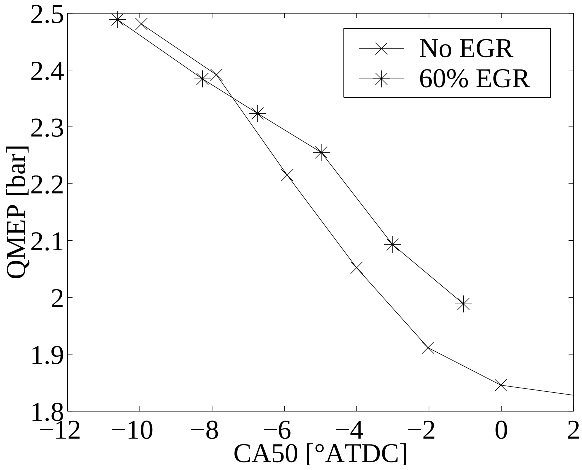

Figure 4. QMEP at idle with and without EGR.

QMEP grows strongly with advanced timing, to compensate for heat losses and lost effective compression ratio. At late timing, EGR requires higher QMEP due to the lower γ of the charge at this condition. At highly advanced timing this is compensated for by the lower heat losses due to the higher specific heat of the recirculated gases.

Using the strategy described above the intake air has to be heated to temperatures well above exhaust temperature for the more advanced combustion timings. Figure 5 and Figure 6 compare intake and exhaust temperatures without and with EGR respectively. Obvious from the graphs is also the design limit of 200°C.

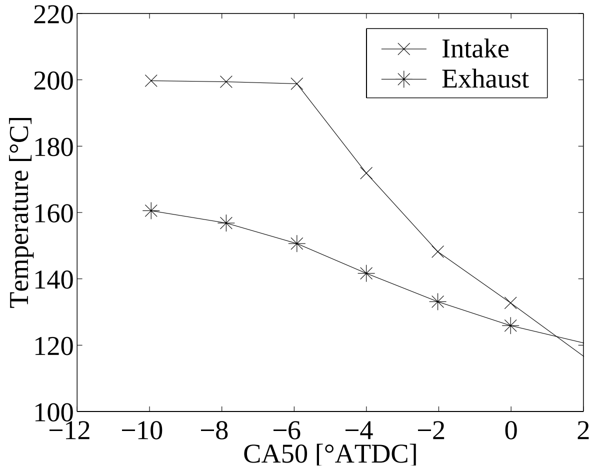

Figure 5. Intake and exhaust temperature at idle without EGR.

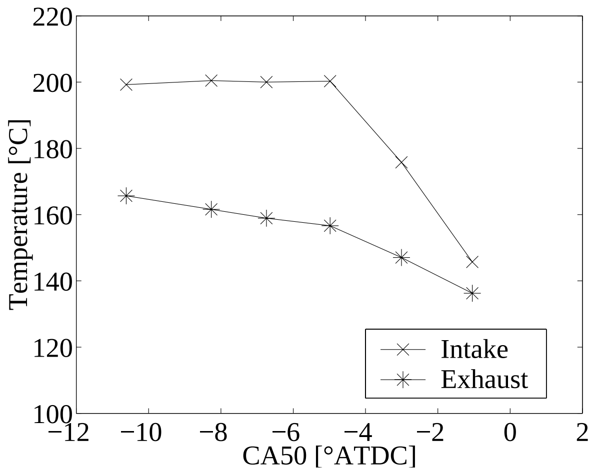

Figure 6. Intake and exhaust temperature at idle with 60% EGR.

The lower γ of EGR results in higher intake temperature requirement compared to no EGR. The difference is not huge since, at idle even 60% of EGR results in a very lean charge and the oxygen concentration in the intake is between 17 and 18%. The exhaust consists mainly of air.

In this study intake air has been heated according to the temperature strategy described above. This strategy results in very high intake temperature, above exhaust temperature. For a combustion engine this is a rare condition, but not very surprising. The released heat is very low, but with a very high intake temperature heat losses are still significant.

Heating the intake air to 200°C at 600 rpm and no significant compressor work requires about 10kW of electricity. Transformed to mean effective pressure it is 1.7 bar, i.e. on the order of 50% of the injected fuel. Of course some of the heat should be extracted from the exhausts, but it is still unlikely to operate the engine with an intake temperature exceeding the exhaust temperature in a practical application.

INTERMEDIATE LOAD

These experiments are very similar to the experiments performed by Christensen et al in 1998 [19]. However, in the present study, a different engine is used, the fuel is a combination of ethanol and n-heptane and combustion timing is controlled independently of EGR.

To make a better comparison with the old measurements, the standard turbo charger for diesel operation replaces the turbocharger described above. In this way the engine behaves like naturally aspirated at the selected speed of 1000 rpm. In [19] it was concluded that EGR makes it possible to retard the start of combustion and slow down the combustion. The emissions of HC improve, but emissions of CO are more dependent on preheating and load. The improvement of combustion efficiency results in higher thermal efficiency of the engine. These conclusions can be compared to the results in the present study.

Figure 7 shows the variation of lambda as EGR is increased and the total fuel heat, FuelMEP, is maintained. The two lines correspond to approximately $\lambda = 3.5$ and $\lambda = 2.5$ respectively at no EGR. The combustion timings for the two cases are CA50 = 0.9°ATDC for the lower fuel loading and CA50 = 7.2°ATDC for the higher fuel loading. The lower fuel loading results in a BMEP of around 2.3 bar and the higher fuel loading gives approximately 4.2 bar.

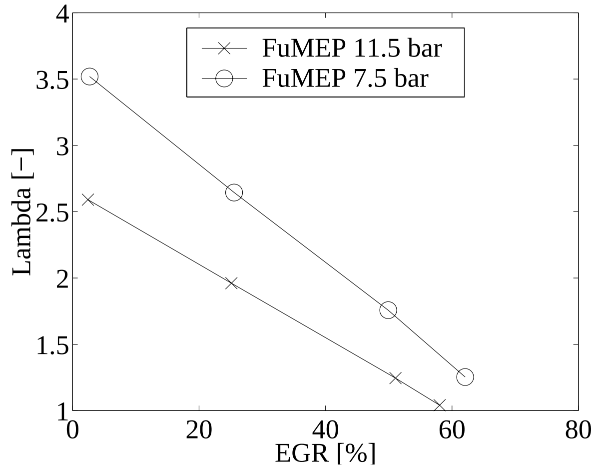

Figure 7. Relative air-fuel ratio as a function of EGR at two different fuel loadings.

The adopted temperature strategy results in intake temperatures as illustrated in Figure 8. Low fuel loading or high EGR fraction results in a higher intake temperature in order to maintain combustion timing with approximately the same fuel composition.

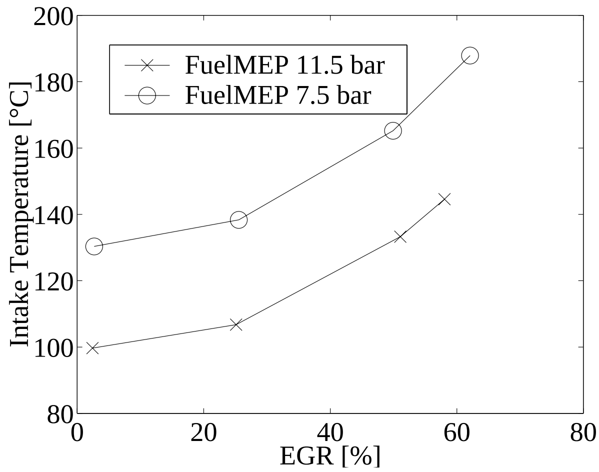

Figure 8 Applied intake temperature as a function of EGR fraction for two different fuel loadings.

In the present study combustion timing is maintained and the effect of EGR on combustion duration, crank angle between 10 and 90% burned, can be studied in Figure 9. The data suggests a longer duration for a higher EGR level, but the trend is not very strong. The effect of combustion timing can be studied together with the effect of EGR in Figure 10. Dashed lines refer to the highest EGR fraction tested for that fuel load. Heat release durations are very similar and it is even hard to distinguish the two lines. The effects of fuel loading and timing are much stronger.

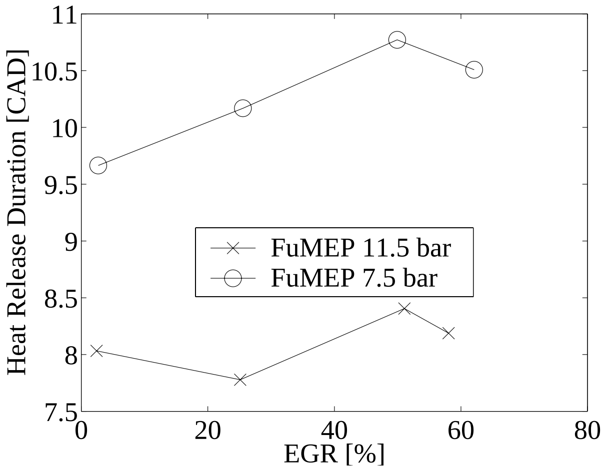

Figure 9. Heat release duration, expressed as the crank angle between 10 and 90% accumulated heat release, as function of EGR.

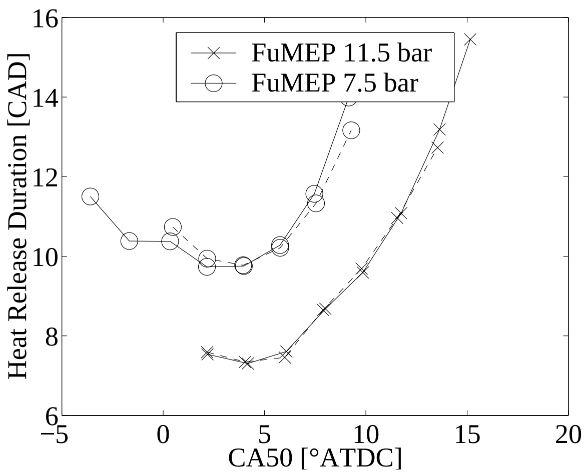

Figure 10. Heat release duration, expressed as crank angle between 10 and 90% accumulated heat release, as function of the 50% burned crank angle. Solid lines refer to no EGR and dashed lines refer to highest EGR fraction.

Recirculation of partially burned fuel potentially improves combustion efficiency. Figure 11 shows the relationship between emissions of HC and EGR ratio. Obviously the concentration of HC in the exhausts is almost constant as the flow decreases when more gas is recirculated. This corresponds well with the theory that the main source of HC in normal operation is crevices [22]. The concentration of fuel or partially burned products in the crevices will only increase marginally when EGR is applied.

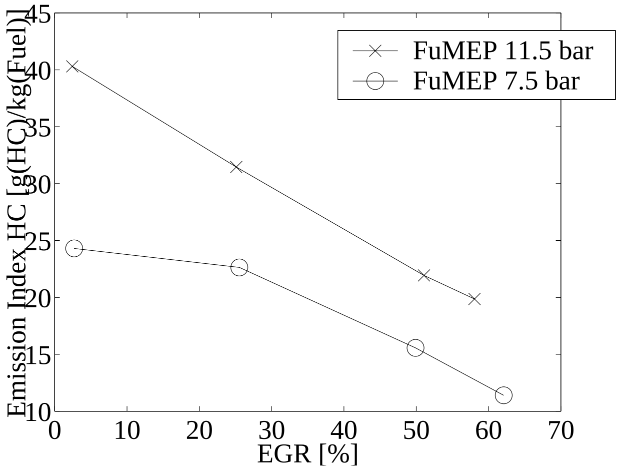

Figure 11. Emission index of HC as a function of EGR for two different fuel loadings.

The formation and oxidation of CO is known to be more complex. For CO to form it is required that oxidation of the fuel starts. For CO not to oxidize during combustion it is required that the reactions are quenched. This can happen either by cooling from the wall [31] or by bulk quenching during expansion [30]. This complex nature makes predictions hazardous and the results are very much dependent on exactly how the engine is operated. In this study, combustion timing is maintained and temperature is increased to keep a constant ratio of fuels for the cylinder using the lowest fraction of n-heptane. This is important to keep in mind when interpreting the results.

Figure 12 shows how emissions of CO change when EGR is introduced. Except for one data point the trend is very similar to HC. This suggests that, in this case, the recirculated CO is more or less completely oxidized during its “second combustion”. The diverting data point is probably due to cylinder-to-cylinder variation in λ. Overall the engine is very close to stoichiometric and one or two cylinders may be running rich, causing much higher levels of CO.

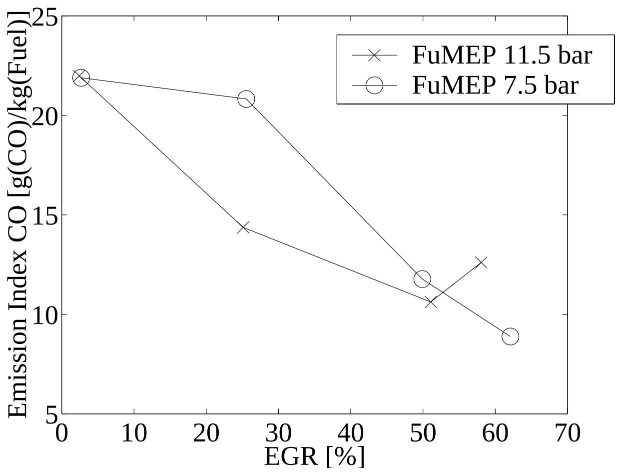

Figure 12. Emission index of CO as a function of EGR for two different fuel loadings.

Studying Figure 11 and Figure 12 it is obvious that combustion efficiency improves with EGR in this case. Figure 13 quantifies this improvement. For both fuel loadings the combustion efficiency improves by about two percentage points. In other words, the loss of fuel energy to the exhaust is reduced by 50%. The higher fuel loading has the lower combustion efficiency. This is because combustion timing is selected to give the highest brake thermal efficiency without EGR, which results in retarded timing for the higher fuel load to reduce heat losses.

Looking back at equation 10, gas exchange and mechanical efficiency should not change with EGR.

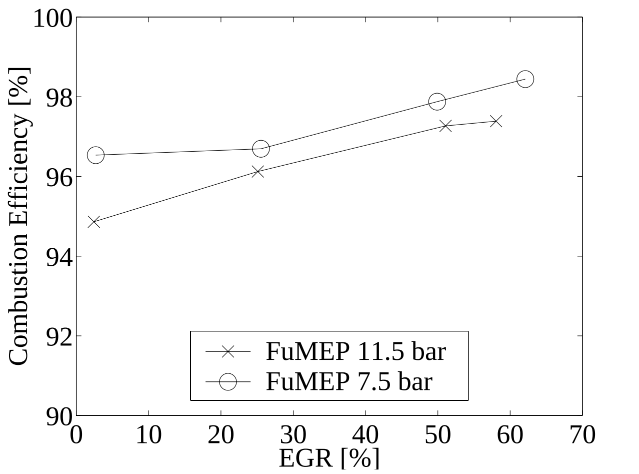

Figure 13. Combustion efficiency, based on exhaust analysis, as a function of EGR.

Since the effect on cylinder pressure is not huge, there is no reason to expect a change in the mechanical efficiency when EGR is introduced. Considering a naturally aspirated engine there should not be a significant effect on gas exchange efficiency either. Some additional pumping losses could be expected if large pressure drops are required to drive the recirculated flow. However this effect is negligible in the present study. PMEP varies less than 0.01 bar and with no obvious trend compared to the EGR fraction. According to equation 10 this leaves the overall effect on brake thermal efficiency to be governed by changes in combustion efficiency and thermodynamic efficiency.

Combustion efficiency is shown to improve, but left to discuss is the thermodynamic efficiency. The ratio between released heat and work performed by the in-cylinder gases. This efficiency depends on the γ-trace over the closed part of the cycle. The thermodynamic efficiency also depends on the change of the number of molecules in the charge during combustion as well as the phasing and duration of the combustion and the heat losses.

Since the combustion phasing, or timing, is maintained by closed loop control and it is shown above that the heat release duration is only marginally affected by EGR, these factors should not affect thermodynamic efficiency. This leaves heat losses and differences in γ as sources of changes in thermodynamic efficiency as the fraction of EGR is changed. Figure 14 shows how the change in γ, due to EGR, affects the thermodynamic efficiency. The graph originates from a simulation assuming perfect combustion according to a measured heat release profile, ideal gas data for air, ethanol and products and constant heat loss profile. The heat loss profile is calculated using the Woschni correlation [33] for the operating point without EGR. FuelMEP is 8.9 bar, corresponding to $\lambda = 3$ without EGR.

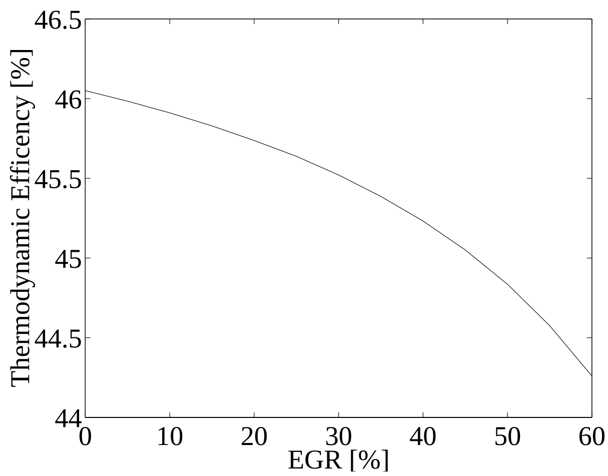

Figure 14. Simulated thermodynamic efficiency as a function of EGR, for a realistic cycle. Only γ dependence of EGR taken into account.

The change in γ, when moving from no EGR to 60% EGR, accounts for a decrease in the thermodynamic efficiency of just above 1.5 percentage points. Since the other efficiencies are close to unity, this translates to a decrease in brake thermal efficiency of the same magnitude.

In the simulation above the heat losses are constants, in order to isolate the effect of γ. Because of the higher specific heat of the EGR, a lower gas temperature and thus a lower heat loss could be expected for high EGR fractions. Unfortunately neither the experimental setup nor the analytical tools available provide accurate means of estimating the heat losses. Instead the thermodynamic efficiency is plotted in Figure 15.

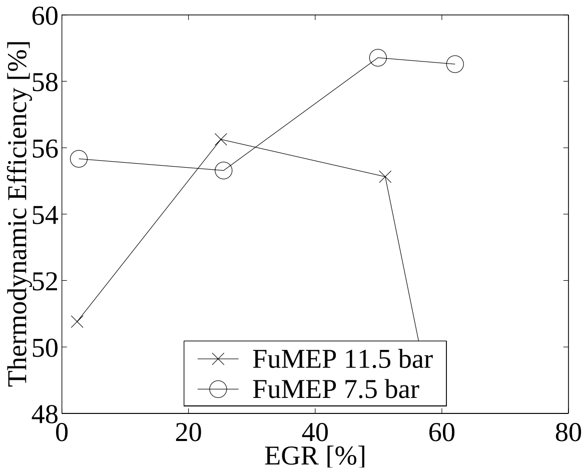

Figure 15. Thermodynamic efficiency as a function of EGR for two different fuel loadings.

In Figure 15 the thermodynamic efficiency is higher than in the simulation. In the experimental data QMEP is calculated from measured fuel flow and combustion efficiency calculated from exhaust composition. The deviation between the results could be due to overestimated heat loss in the simulation or an experimental error. However, as discussed above it is not very likely to underestimate the combustion efficiency.

For the experimental data on thermodynamic efficiency the trend versus EGR is not very clear. Apparently, the lower fuel loading gains efficiency with increasing EGR. This suggests that the positive effect of lower heat losses due to lower gas temperature dominates the negative effect of lower γ. For the higher fuel loading no such trend is established.

Figure 16 shows the overall effect of EGR on brake thermal efficiency.

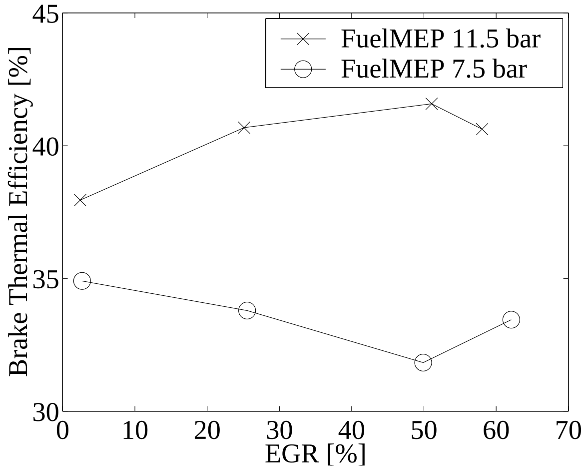

Figure 16. Brake thermal efficiency as a function of EGR fraction for two different fuel loadings.

So far the effect of EGR on efficiency has been discussed for constant combustion timing. This timing is selected to give the highest brake thermal efficiency for the operation without EGR. When studying the total effect of EGR on brake thermal efficiency, this approach would skew the results against EGR. Figure 17 shows brake thermal efficiency versus combustion timing with and without EGR for the two fuel loadings.

For the higher load, timing has to be retarded to achieve highest efficiency. This is probably caused by high heat losses for advanced timing without EGR. It appears that the effect of lower heat losses with EGR, due to the lower temperature, advances the optimum combustion timing. For this load EGR definitely improves brake thermal efficiency, the effects of higher combustion efficiency and lower heat losses is greater than the effect of lower γ.

For the lower fuel loading however, the effect is not as clear. At retarded combustion timing the difference in combustion efficiency, around 17 percentage points, dominates to the advantage of EGR. As timing is advanced the difference in combustion efficiency decreases and the thermodynamic efficiency with EGR drops, while the thermodynamic efficiency without EGR increases. As a total the EGR makes a small, if any, difference in brake thermal efficiency for the lower fuel loading.

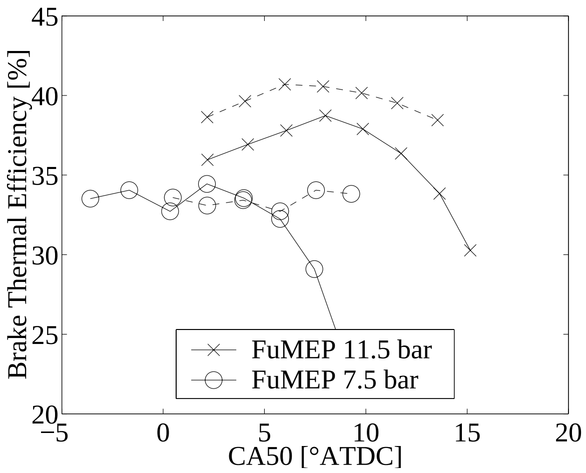

Figure 17. Brake thermal efficiency versus combustion timing for two different fuel loadings. Solid lines refer to no EGR and dashed lines to highest EGR fraction tested.

Figure 18 shows how NO_X emissions decrease with increasing EGR at the higher of the two fuel loadings discussed above. The most probable reason for this is the lower temperature after combustion, resulting from the higher specific heat of the recirculated gases. For the highest EGR ratio, the charge is close to stoichiometric and the lack of oxygen at the end of combustion may also reduce the rate of NO_X formation.

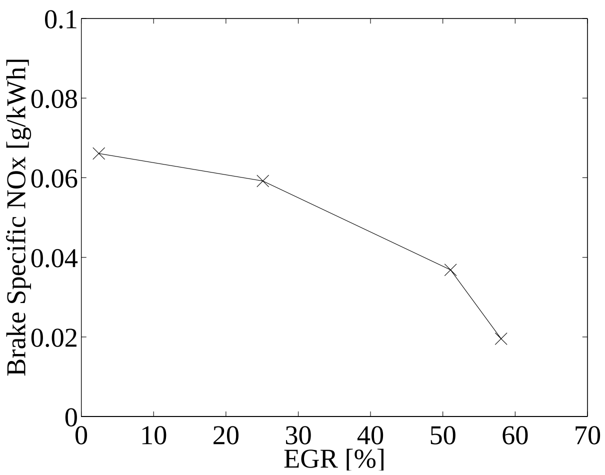

Figure 18. Brake specific NO_X emissions at FuelMEP 11.5 bar, as function of EGR fraction.

TURBO CHARGED OPERATION

For fundamental studies of some aspects of EGR, naturally aspirated condition is to prefer. However, for a heavy-duty engine, turbo charging is a tempting way to improve power density. It has previously been shown by the authors [28] that turbo charging is an effective means of increasing the achievable load, but at the cost of high pumping losses. In this section temperature sweeps without EGR and with 45% EGR at 1600 rpm, 11 bar BMEP and about 0.9 bar boost are examined.

Unfortunately it was not possible to keep a perfectly constant load. Figure 19 shows how BMEP, net IMEP, gross IMEP, QMEP and FuelMEP varied in the sweeps. The intention was to keep BMEP constant, but as is seen in the figure small variations occurred. This has to be kept in mind when analyzing the data.

Just as in the cases discussed above, recirculation of unburned products and the “second chance” for combustion improves combustion efficiency. Figure 20 shows the combustion efficiency at 1600 rpm and 11 bar BMEP as a function of intake temperature. EGR gives an improvement, but at this high load, combustion efficiency is well above 95% even without EGR. The improvement is on the order of one percentage point or a reduction by 30% of the unburned fuel.

Thermodynamic efficiency should not depend very much on moderate changes in intake temperature. The effect of QMEP, and its influence on heat losses, is stronger. Figure 21 shows the thermodynamic efficiency as a function of QMEP. The effect of QMEP is very strong and the difference between EGR and no EGR is small. Obviously the negative effect of lower γ for the EGR is compensated by the lower heat losses due to higher C_P. Higher QMEP leads to higher cycle temperatures and pressures. The violent combustion of HCCI in this end of the load range results in rapidly increasing heat losses with load.

Pumping losses are known to be severe for turbo charged HCCI, the reason being the low exhaust temperature [16]. Figure 22 shows PMEP for the two sweeps. The effect of EGR is not very pronounced. For the lowest intake temperature with EGR, pumping work is somewhat increased due to a negative mass flow balance over the turbo charger. The fraction of the intake air passing the CAC is high and much water is condensed and separated. As a result the compressor is exposed to a higher mass flow compared to the turbine.

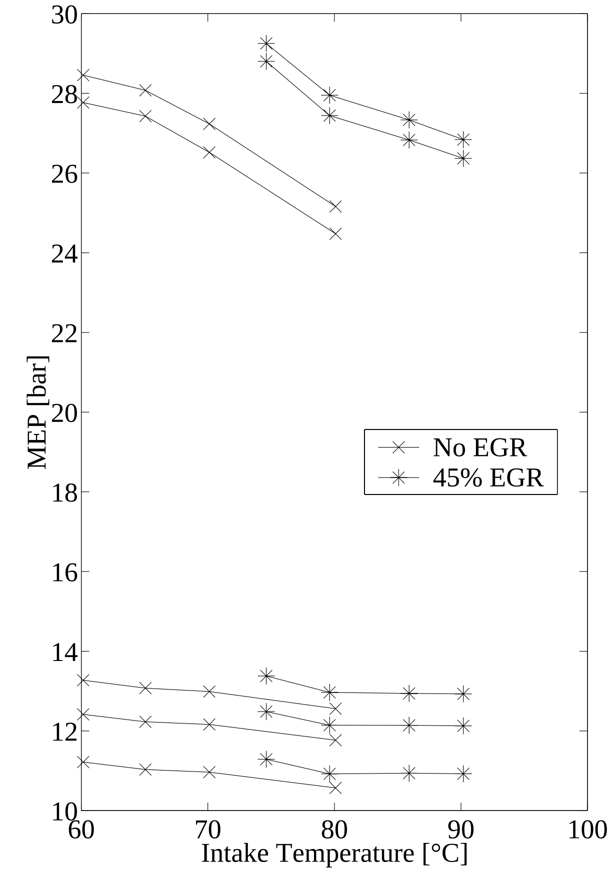

Figure 19 From top: FuelMEP, QMEP, gross IMEP, net IMEP and BMEP with and without EGR as a function of intake temperature.

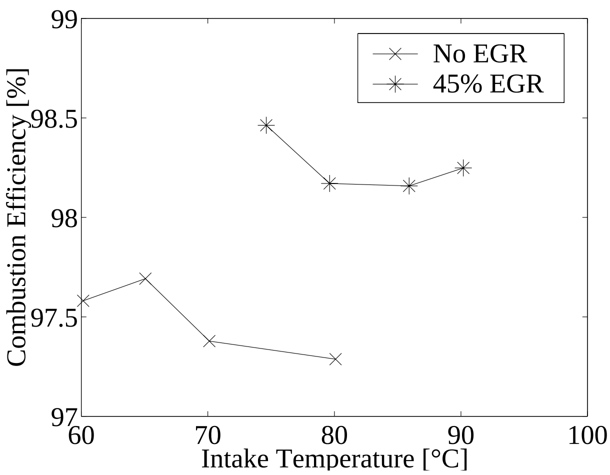

Figure 20. Combustion efficiency with and without EGR at turbo charged condition. 11 bar BMEP at 1600 rpm

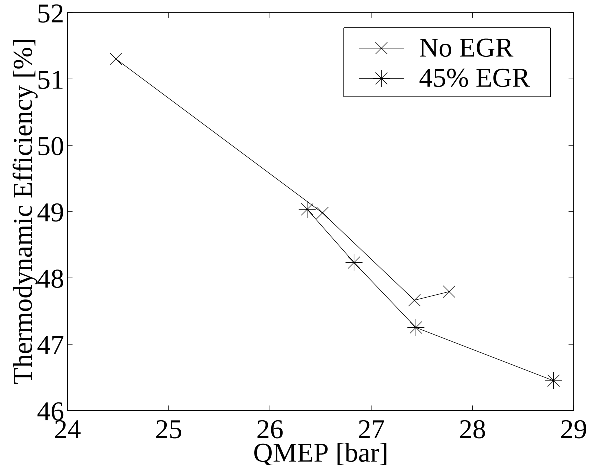

Figure 21. Thermodynamic efficiency with and without EGR at turbo charged condition. 11 bar BMEP at 1600 rpm

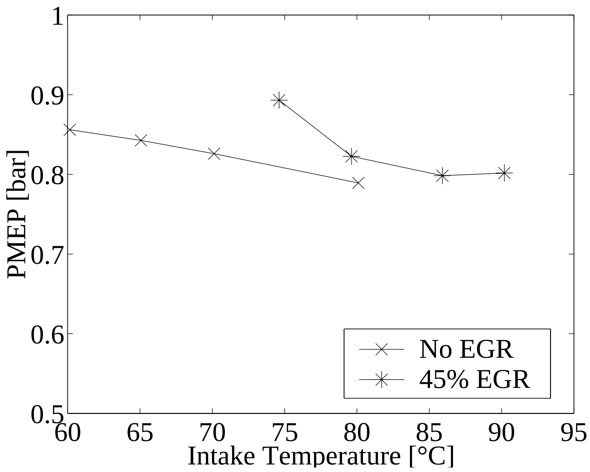

Figure 22. PMEP with and without EGR at turbo charged condition. 11 bar BMEP at 1600 rpm

The high PMEP originates from the low exhaust temperature of the HCCI process. Figure 23 shows that this temperature is on the order of 30°C higher with EGR. The reason for this is the lower temperature drop during expansion, due to lower γ of the closer to stoichiometric burned gases in the EGR case. Equation 11 shows the expression for the isentropic expansion work available to the turbine. Exhaust back pressure in this case is around 2.3 bar abs. Putting numbers in the equation gives that the 30°C hotter exhausts for EGR should result in a 0.15 bar lower back pressure. Unfortunately some of this gain is lost by the increased backpressure from the EGR system, Figure 24. The combined effects of changes in C_P and γ on equation 11 cancel out to within 0.5% for this operating condition.

$$\dot{W} = \dot{m} \cdot c_P \cdot T_{EXH} \cdot \left(1 - \left(\frac{p_{AMB}}{p_{EXH}} \right)^{\frac{\gamma-1}{\gamma}} \right) \qquad \text{(eq. 11)}$$

In the section “Experimental Apparatus” it is mentioned that some water is separated after the CAC. The amount is not measured, but can, in the cases when water is separated, be estimated by assuming saturated condition out of the water separator. Doing so for the operating points with EGR renders separated mass fractions of around 1%, except for at 75°C intake temperature, when the fraction is around 2%. This has a negative effect on gas exchange efficiency, but only accounts for about 0.05 bar of PMEP. The water separation also results in an over-estimation of the EGR fraction by 2 to 2.5 percentage points.

Increased back pressure and lost mass flow accounts for approximately 0.1 bar of the observed difference in PMEP. However, differences in operating condition for the exhaust turbine can easily change the efficiency of the turbine by several percentage points, thus perfect agreement cannot be expected.

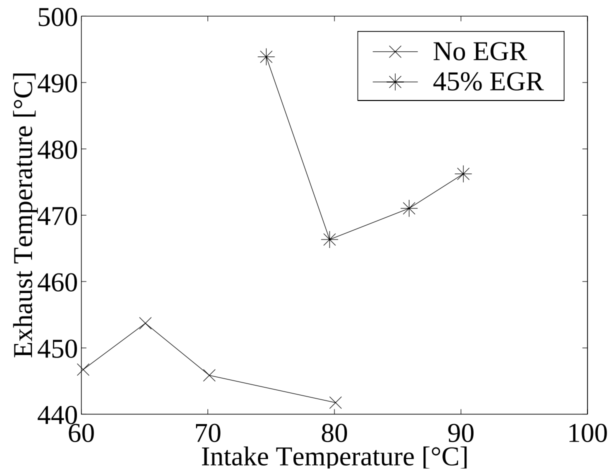

Figure 23. Exhaust temperature with and without EGR at turbocharged condition.

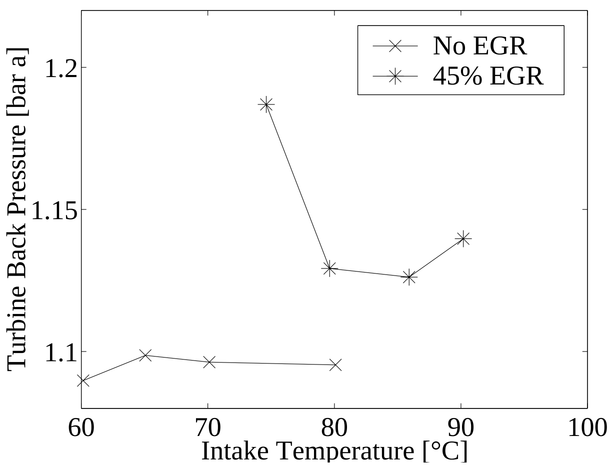

Figure 24. Turbine backpressure with and without EGR at turbo charged condition.

Figure 25 shows the gas exchange efficiency with and without EGR. The variation is very small, less than 1 percentage point. The gas exchange efficiency with EGR could be somewhat improved by redesign to reduce the pressure drop in the EGR system and eliminate the water separation. However, the possibilities for improvements are limited according to the discussion above.

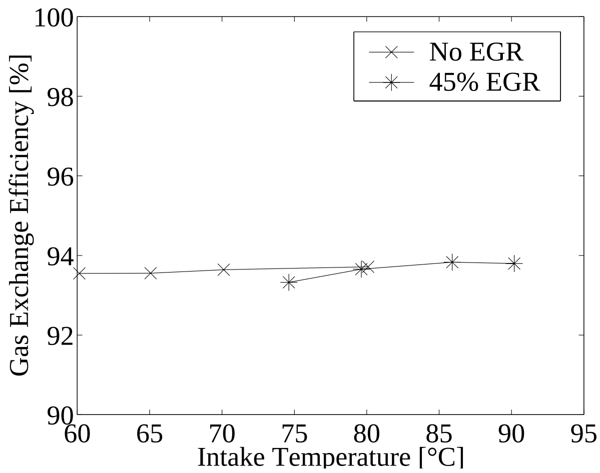

Figure 25. Gas exchange efficiency with and without EGR at turbo charged condition. 11 bar BMEP at 1600 rpm

At high load, NO_X emissions start to become significant. It is well known that the rate of NO_X formation is a very strong function of temperature and more or less proportional to the concentrations of N_2 and O_2. Figure 26 shows the concentration of NO_X and the result might look somewhat confusing at a first glance.

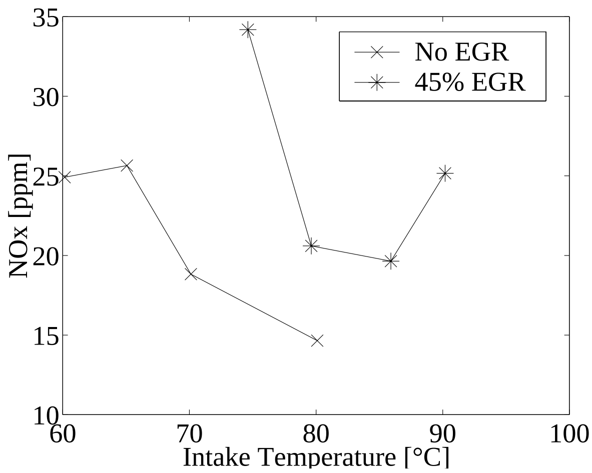

Figure 26. NO_X concentration in the exhausts with and without EGR at turbo charged conditions. 11 bar BMEP at 1600 rpm

Unlike emissions of HC, being results of incomplete combustion, NO_X is actually formed during the combustion. During the time when the temperature is high enough for significant NO_X formation rates, the NO_X concentration is much lower than equilibrium. Therefore, the presence of NO_X from recirculated gases does not in itself lower the NO_X formation rate. At constant NO_X

formation rate the NO_X concentration increases with increased recirculation rate.

The brake specific NO_X, saying more about the formation rate, is shown in Figure 27. Here the differences between EGR and no EGR are smaller, but, maybe surprisingly, not to the clear advantage of EGR. The main reason for the EGR not giving better results is the higher QMEP, Figure 19. Higher QMEP results in higher gas temperature, which counteracts the beneficial effect of higher C_P. The higher QMEP with EGR is mainly a result of the inability to maintain exactly the desired load. Actually Figure 27 shows that EGR allows a somewhat higher load with maintained NO_X emissions.

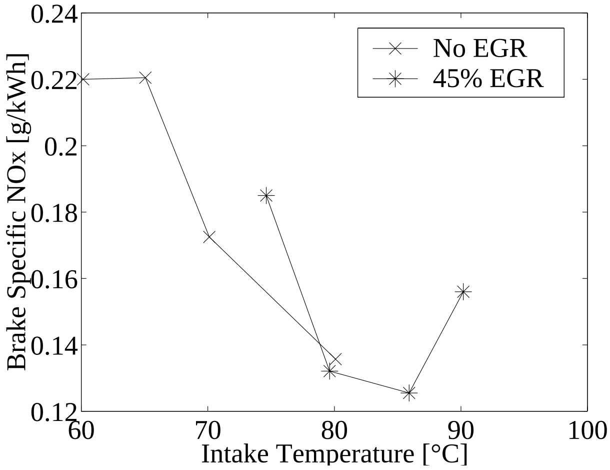

Figure 27. Brake specific NO_X production with and without EGR at turbocharged condition. 11 bar BMEP at 1600 rpm

Figure 28 also reveals why higher intake temperature gives lower NO_X emissions without EGR. The higher intake temperature also results in the higher intake pressure, thus higher lambda. In fact the NO_X results are almost the inverse of the intake pressure. Again the system design is responsible for the results. The water separator is mounted in series only with the CAC, resulting in a higher pressure drop in this circuit compared to the heater circuit. Lower intake temperature requires a higher flow through the CAC, thus increasing the pressure drop between the compressor and intake manifold.

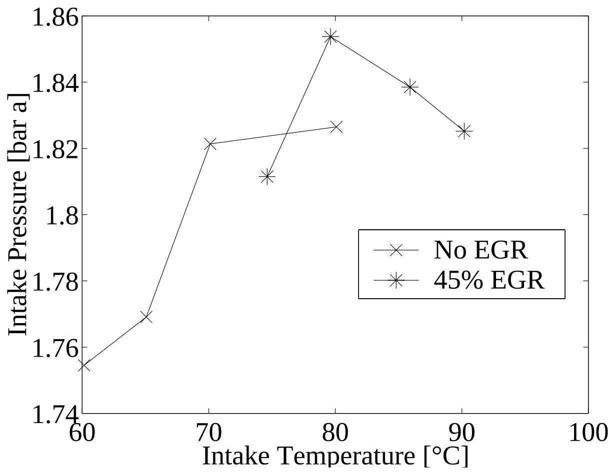

Figure 28. Intake pressure with and without EGR at turbocharged condition.

CONCLUSION

EGR is tested on a multi cylinder HCCI engine for a wide range of operating conditions. Some of the effects are clear and easy to understand, others are much dependent on the particular setup.

In all cases EGR improves combustion efficiency. This is most important at low load, where combustion in an HCCI engine often is far from complete. However, EGR requires higher intake temperature and if this cannot be arranged, results will not be as good.

The thermodynamic efficiency of the cycle is affected in two ways: The lower γ of the recirculated gas tends to lower the efficiency. The higher c_p results in lower temperature after combustion, decreasing heat losses. It is not clear, and probably depending on operating conditions, which of the effects is the strongest. In most cases studied in this paper the two effects esentially cancel out.

EGR was expected to slow down the combustion, and previous research has come to this conclusion [19, 30]. In this study the isolated effect of EGR on combustion duration is negligible and cannot be proven. This conclusion does however assume that combustion timing is kept constant as EGR is introduced.

In turbo charged mode the results are very much dependent on how the system is implemented. With the present setup the EGR system introduces a pressure drop and some mass flow is lost after compression. This counteracts the positive effect of higher exhaust temperature with EGR.

REFERENCES

1. S: Onishi, S. Hong Jo, K. Shoda, P Do Jo, S. Kato: ”Active Thermo-Atmosphere Combustion (ATAC) – A New Combustion Process for Internal Combustion Engines”, SAE790501
2. P.Duret, S.Venturi: ”Automotive Calibration of the IAPAC Fluid Dynamically Controlled Two-Stroke Combustion Process”, SAE960363
3. M. Noguchi, Y. Tanaka, T. Tanaka, Y. Takeuchi: ”A Study on Gasoline Engine Combustion by Observation of Intermediate Reactive Products during Combustion”, SAE790840
4. N. Iida: ”Combustion Analysis of Methanol-Fueled Active Thermo-Atmosphere Combustion (ATAC) Engine Using a Spectroscopic Observation” SAE940684
5. Y. Ishibashi, M. Asai: ”Improving the Exhaust Emissions of Two-Stroke Engines by Applying the Activated Radical Combustion”, SAE960742
6. R. Gentili, S. Frigo, L. Tognotti, P. Hapert, J. Lavy: ”Experimental study of ATAC (Active Thermo-Atmosphere Combustion) in a Two-Stroke Gasoline Engine”, SAE 970363
7. A. Hultqvist, M. Christensen, B. Johansson, A. Franke, M. Richter, M. Aldén: ”A Study of the Homogeneous Charge Compression Ignition Combustion Process by Chemiluminescence Imaging”, SAE1999-01-3680
8. P. Najt, D.E. Foster: ”Compression-Ignited Homogeneous Charge Combustion”, SAE830264
9. R.H. Thring: ”Homogeneous-Charge Compression-Ignition (HCCI) Engines”, SAE892068
10. T. Aoyama, Y. Hattori, J. Mizuta, Y. Sato: ”An Experimental Study on Premixed-Charge Compression Ignition Gasoline Engine”, SAE960081
11. T.W. Ryan, T.J. Callahan: ”Homogeneous Charge Compression Ignition of Diesel Fuel”, SAE961160
12. H. Suzuki, N. Koike, H. Ishii, M. Odaka: ”Exhaust Purification of Diesel Engines by Homogeneous Charge with Compression Ignition”, SAE970313, SAE970315
13. A. W. Gray III, T. W. Ryan III: ”Homogeneous Charge Compression Ignition (HCCI) of Diesel Fuel”, SAE971676
14. H. Suzuki, N. Koike, M. Odaka: ”Combustion Control Method of Homogeneous Charge Diesel Engines”, SAE980509
15. T. Seko, E. Kuroda: ”Methanol Lean Burn in an Auto-Ignition Engine”, SAE980531
16. A. Harada, N. Shimazaki, S. Sasaki, T. Miyamoto, H. Akagawa, K. Tsujimura: ”The Effects of Mixture Formation on Premixed Lean Diesel Combustion”, SAE980533
17. M. Christensen, P. Einewall, B. Johansson: ”Homogeneous Charge Compression Ignition (HCCI) Using Isooctane, Ethanol and Natural Gas – A Comparison to Spark Ignition Operation", SAE972874
18. M. Christensen, B. Johansson, P. Amnéus, F. Mauss: ”Supercharged Homogeneous Charge Compression Ignition”, SAE 980787
19. M. Christensen, B. Johansson: ”Influence of Mixture Quality on Homogeneous Charge Compression Ignition”, SAE982454
20. Christensen, M., Hultqvist, A., and Johansson, B., “Demonstrating the Multi Fuel Capability of a Homogeneous Charge Compression Ignition Engine with Variable Compression Ratio,” SAE 1999-01-3679
21. M. Christensen, B. Johansson: ”Homogeneous Charge Compression Ignition with Water Injection”, SAE1999-01-0182
22. M Christensen, A Hultqvist, B Johansson: “The Effect of Piston Topland Geometry on Emissions of Unburned Hydrocarbons From a Homogeneous Charge Compression Ignition (HCCI) Engine “, SAE 2001-01-1893
23. M. Stockinger, H. Schäpertöns, P. Kuhlmann, Versuche an einem gemischansaugenden mit Selbszündung, MTZ 53 (1992).
24. P Amneus, D Nilsson, M Christensen, B Johansson: “Homogeneous Charge Compression Ignition Engine: Experiments and Detailed Kinetic Calculations”, The Fourth Symposium on Diagnostics and Modeling of Combustion in Internal Combustion Engines, Comodia 98, pp. 567-572, 1998
25. J-O Olsson, O. Erlandsson, B. Johansson: “Experiments and Simulations of a Six-Cylinder Homogeneous Charge Compression Ignition (HCCI) Engine”, SAE2000-01-2867
26. J-O Olsson, P. Tunestål, B. Johansson, “Closed-Loop Control of an HCCI Engine”, SAE 2001-01-1031
27. M. Christensen, B. Johansson: ”Supercharged Homogeneous Charge Compression Ignition (HCCI) with Exhaust Gas Recirculation and Pilot Fuel”, SAE 2000-01-1835
28. J-O Olsson, P Tunestål, G Haraldsson, B Johansson: “A Turbo Charged Dual Fuel HCCI Engine”, SAE 2001-01-1896
29. J-O Olsson, P Tunestål, B Johansson, S Fiveland, R Agama, M Willi, D Assanis: “Compression Ratio Influence on Maximum Load of a Natural Gas Fueled HCCI Engine”, SAE 2002-01-0111
30. J Dec: “A Computational Study of the Effects of Low Fuel Loading and EGR on Heat Release Rates and Combustion Limits of HCCI Engines”, SAE 2002-01-1309
31. S Aceves, J Martinez-Frias, D Flowers, J Smith, R Dibble, J Wright, R Hessel: “A Decoupled Model of Detailed Fluid Mechanics Followed by Detailed Chemical Kinetics for Prediction of Iso-Octane HCCI Combustion”, SAE 2001-01-3612
32. J Hiltner, S Fiveland, R Agama, M Willi: “System Efficiency Issues for Natural Gas Fueled HCCI Engines in Heavy-Duty Stationary Applications”, SAE 2002-01-0417

33. John B. Heywood, Internal Combustion Engine Fundamentals ISBN 0-07-100499-8
34. Meß- & Analysentechnik GmbH, "Servicemenu Flame-Ionization-Detector Thermo-FID"

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DEFINITIONS, ACRONYMS, ABBREVIATIONS

BMEP	Brake Mean Effective Pressure
c_p	Specific heat at constant pressure
c_v	Specific heat at constant volume
CAC	Charge Air Cooler
CI	Compression Ignition
CO	Carbon monoxide
CO_2	Carbon dioxide
DICI	Direct Injection Compression Ignition
EGR	Exhaust Gas Recirculation
FID	Flame Ionization Detector
FuelMEP	Fuel Mean Effective Pressure
HC	HydroCarbons (unburned)
HCCI	Homogeneous Charge Compression Ignition
IMEP	Indicated Mean Effective Pressure
IMEPg	Gross Indicated Mean Effective Pressure
IMEPn	Net Indicated Mean Effective Pressure
N_2	Nitrogen
NO_X	The sum of nitrogen oxides
OHC	Oxygenated HydroCarbons
O_2	Oxygen
PMEP	Pumping Mean Effective Pressure
QMEP	Heat Release Mean Effective Pressure
SI	Spark Ignition
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