Ignition- and combustion concepts for lean operated passenger car natural gas engines

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Abstract: Currently, passenger car CNG engines are based on boosted petrol engines. Such engines have typically restrictions, e.g. combustion peak pressures, which prevent from exploiting the potential of methane based fuels. Additionally, the use of cost-efficient three-way-catalysis limits the engine operation to $\lambda=1$. Here, we present the efficiency potential and the raw emission characteristics for passenger car CNG engines without sticking to combustion peak pressure and λ limitations. Lean combustion reduces the knocking tendency but, because of the higher pressure levels, increases the ignition energy demand. Therefore, different ignition systems (spark plug, prechamber, diesel pilot) have been used and compared.

1 Introduction

Natural gas is of increasing interest in the mobility sector as this low-carbon-fuel offers distinct CO_2 advantages [1]. Additionally, renewable methane can be produced and stored in cost-effective ways which gives biogenic and synthetic natural gas an ecologic and economic long-term perspective. Natural gas is also an attractive fuel for the automotive industry [2] which is faced with increasingly stricter CO_2 and emission regulations worldwide.

Today's commercially available natural gas engines for passenger cars are based on petrol engines, ideally with some adaptations (e.g. increased compression ratio, increased boost pressure, adapted valve seats, high-temperature turbines). Those adaptations do not fully take the advantageous properties of natural gas into account as for example the peak combustion pressure limitation of typically around 100 bar remains from the basic petrol engine. For pollutant emission reasons, passenger car natural gas engines are nowadays operated stoichiometrically which leads in combination with three-way-catalysis to very low emissions, also in real-world operation [3], and natural gas has the potential for practically zero emissions [4]. Stoichiometric operation, however, leads to reduced efficiencies compared to lean operation.

In the project described here, the above-mentioned limitations (combustion peak pressure, stoichiometric operation) were omitted to find the potentials and limitations for natural gas combustion for engines of passenger car size. To do so, a diesel engine was used as an experimental basis as modern diesel engines can cope with peak pressures in the magnitude of 200 bar. High combustion pressures involve also high ignition energies and a special focus was therefore put on the ignition systems. Three very different ignition systems were used: an inductive ignition system using a well-insulated spark plug (engine 1), an inductive ignition system in a prechamber which could be used with or without prechamber gas injection (engine 2), and a diesel pilot injection system (engine 3). The main goal was to come as close as possible to Diesel engine efficiency levels by combining lean premixed combustion and Diesel-like compression ratios.

2 Engines and Experimental Setup

The main characteristics of the three engines discussed here are listed in Table 1. The engines for the spark ignited versions had gone through extensive modifications (inserts for spark plug or prechamber instead of the diesel injection system, modified valve seats, modified swirl level, modified pistons, different turbocharger, etc.). The diesel pilot engine was only slightly modified (implementation of a PFI CNG supply system), all other details were identical from the serial production diesel engine. All engines were operated with rapid prototyping ECUs and in all engines, closed-loop centre of combustion (COC) control was implemented.

Table 1: Main characteristics of the used engines.

Parameter	Engine 1 Spark Plug Engine	Engine 2 Prechamber Engine	Engine 3 Diesel Pilot Engine
# of cylinders / valves per cylinder	4 / 4	4 / 4	4 / 4
Displacement [cm^3]	1968	1968	1968
Bore/stroke [mm]	81 / 95.5	81 / 95.5	81 / 95.5
Compression ratio	14.5	14.5	16.5
Ignition system	Inductive	Inductive	-
Spark plugs	NGK M12 in open chamber	NGK M10 in prechamber	-
Diesel injection system	-	-	Common Rail with Piezo Injectors
Gas port fuel injectors	Bosch NGI2 (via mixer)	Bosch NGI2 (via mixer)	Bosch NGI2 (MPI)
Prechamber injectors	-	Special design	-
EGR	Not installed	Not installed	Not used

2.1 Spark Plug Engine

Figure 1 shows the intake manifold with the upstream throttle and gas mixer. The gas mixer was chosen to enable a perfect mixing of methane and air. Transient behaviour was not an issue since this work concentrates on steady-state operation only. Equal lambda across all cylinders could be achieved (each cylinder was equipped with an own lambda sensor to monitor lambda differences).

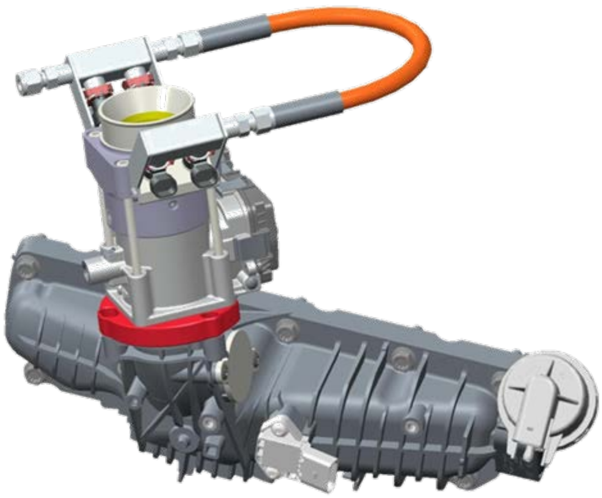

Figure 1: CAD visualisation of gas mixer, throttle and intake manifold.

Figure 2 shows a cross-cut through the combustion chamber. It shows the hemispherical piston bowl, the M12 spark plug insert and the flush-mounted cylinder pressure sensor.

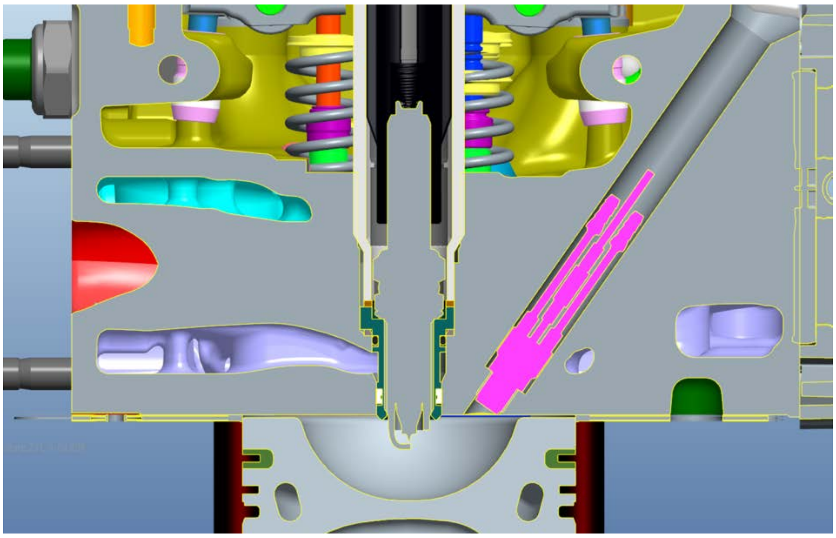

Figure 2: CAD visualisation of the combustion chamber of the spark plug engine.

2.2 Prechamber Engine

The Prechamber engine was built on the same basis as the spark plug engine with the difference, that the cylinder head was equipped with a specifically designed prechamber[1], see Figure 3. The prechamber can be operated passively (i.e. without gas injection to the prechamber) or scavenged (i.e. with gas supply to the prechamber). For space reasons, the prechamber was equipped with a M10 spark plug. A check valve at the prechamber entrance enabled the use of a recessed dosing valve. The rest of the engine (intake manifold, gas mixer turbocharger) was identical to the spark plug version.

[1] The prechamber was designed by Volkswagen together with the project partner Ricardo Software.

A model was created and implemented in the ECU which estimated the λ in the prechamber for the scavenged operation mode. This is not trivial as during compression, a mixture with a certain λ is pushed back into the prechamber and gas is injected into the prechamber. So, the resulting λ at spark timing depends on the λ in the main chamber, of the amount of gas injected to the prechamber and on the spark timing. λ control was implemented to control the prechamber at λ=1 at spark timing, based on this estimated prechamber λ. In passive prechamber operation (without gas injection to the prechamber), λ in the prechamber was the same as λ in the main chamber. In scavenged prechamber operation, the injection timing was optimized for best efficiency and lowest THC emissions which turned out to be the case for early injection (start of prechamber injection around 300 °CA before TDC).

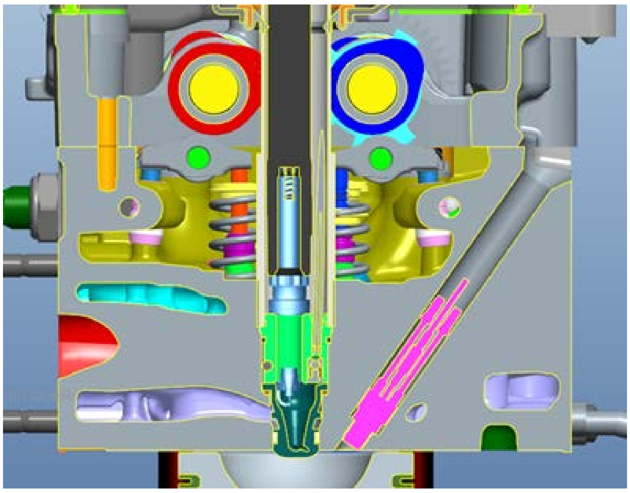

Figure 3: CAD visualisation of the combustion chamber of the prechamber engine (left), picture of combustion-chamber side of the prechamber (right).

2.3 Diesel Pilot Engine

The basic diesel engine was only very slightly modified for diesel pilot operation: Gas injectors were added to the swirl flap adapter just before the engine's intake channels (Figure 4). Diesel was directly injected into the cylinders using the standard Diesel injection system. Once compression ignited, the Diesel provided ignition centres for the premixed natural gas. The diesel injection parameters, i.e. start and duration of injection, were chosen such that the desired combustion phasing was achieved using the least amount of Diesel possible [5]. At high loads, the mechanical limitation on the maximum cylinder pressure prohibited the air-to-fuel ratios that exceed $\lambda = 1.43$.

Figure 4: Four PFI gas injectors mounted on the swirl flap adapter of the original diesel engine

3 Results

Albeit the engines were operated in wide ranges of speed and load, we will concentrate here on two operation points:

- One at low load: (1400 rpm, 50 Nm brake torque = 3.2 bar bmep).
- One at higher load: (2000 rpm, 220 Nm brake torque = 14.0 bar bmep).

Those two operating points cover all the dominant effects seen across the engine map. It has to be noted that the turbochargers are not able to cover all possible operating conditions from λ=1 until the lean limits at all engine speed/torque combinations. Especially at very lean, high load operation and low engine speed, boost pressure limitations occurred which led to a power loss. Such operating conditions are marked in the following Figures.

3.1 Brake Engine Efficiencies

Figure 5 and Figure 6 show the brake engine efficiencies against λ. Best efficiencies showed to be in the λ range of 1.4 … 1.7. The spark plug engine showed efficiencies very similar to the prechamber engine in active and passive operation, but at slightly lower λ. The missing efficiency advantage of the prechamber concept is attributed to increased wall heat losses and a next engine with reduced wall heat losses will be designed based on these results. It has to be noted that at the high load point the comparison between the different combustion concepts at λ values towards 1 is more difficult because different center of combustion settings were needed for knock prevention. However, the general trends are also found here. Overall, all combustion concepts showed similar peak efficiencies at high and low load points except for the diesel pilot engine. The throttled operation with high diesel quantities needed in low load operation led to lower efficiencies than for all the spark ignited concepts. However, at high load the diesel pilot engine showed the highest efficiency levels which is mainly attributed to its higher compression ratio.

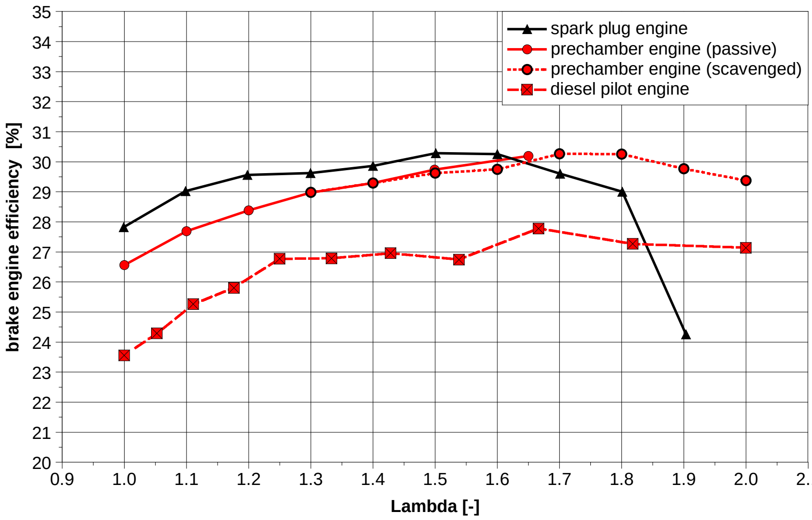

Figure 5: Brake Engine Efficiencies for the operating point 1400 rpm / 50 Nm

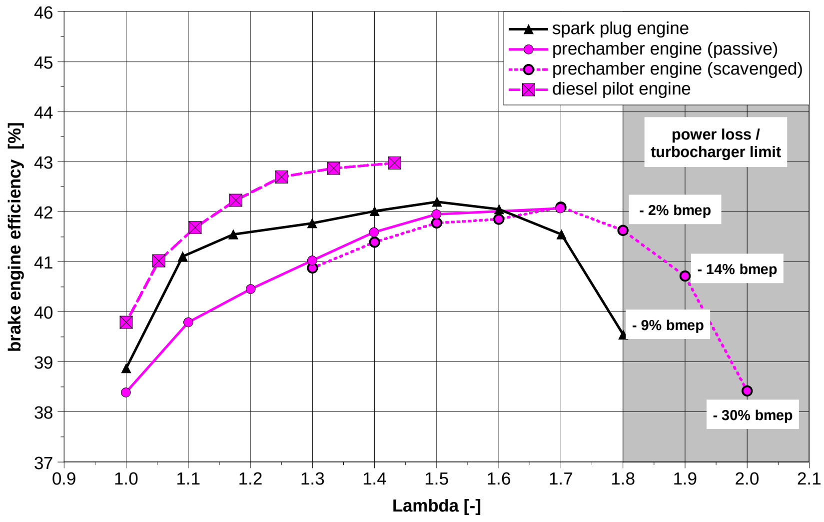

Figure 6: Brake Engine Efficiencies for the operating point 2000 rpm / 220 Nm

3.2 Raw NO_x Emissions

The results for raw NO_x emissions shown in Figure 7 and Figure 8 indicate clear benefits for the passive/active prechamber engine concepts in the range of maximum efficiency between lambda 1.4 and 1.7. For the higher load operating point, however, part of the nearly 50% advantage for the passive/active prechamber results from the retarded COC to prevent knock in the prechamber. Very lean operation revealed, as one would expect, the lowest NO_x emissions. Combustion concepts which enable very lean natural gas combustion are favourable in respect to raw NO_x; they showed raw NO_x levels in the magnitude of 1 g/kWh.

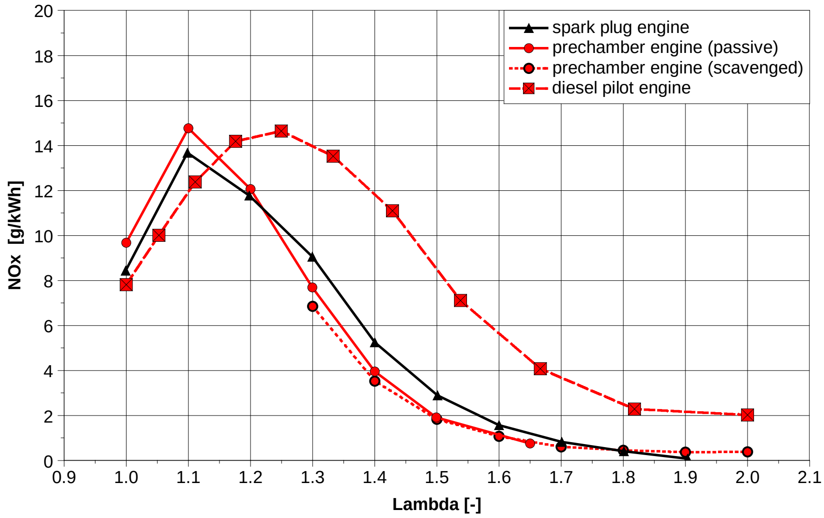

Figure 7: Raw NOx emissions for operating point 1400 rpm / 50 Nm

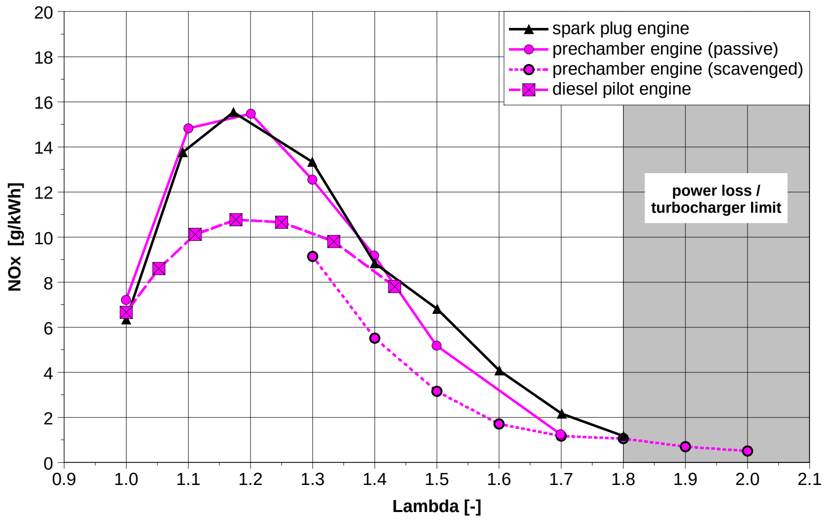

Figure 8: Raw NOx emissions for operating point 2000 rpm / 220 Nm

3.3 Raw THC Emissions

The results of the raw THC emission are shown in Figure 9 and Figure 10 and indicate noticeable benefits for the passive/active prechamber engine concepts in the range of maximum efficiency between lambda 1.4 and 1.7. The Diesel pilot engine did not show any andvantages regarding THC emissions versus the spark ignited concepts especially at lower load points.

Despite the reduction of THC emissions with the active/passive prechamber operation the absolute levels of unburnt hydrocarbons remain the unsolved challenge for exhaust aftertreatment at lean burn operation with levels ranging from 10…30g/kWh in the region with highest efficiency (lambda = 1.5-1.7).

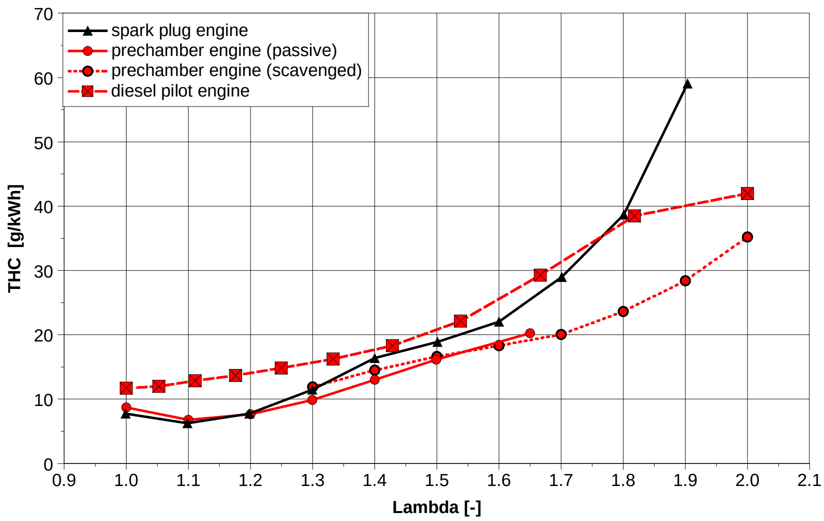

Figure 9: Raw THC emissions for operating point 1400 rpm / 50 Nm

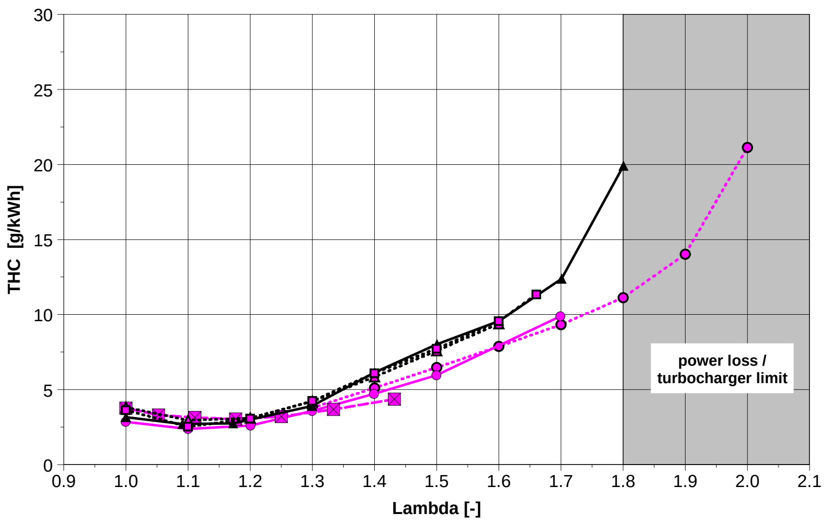

Figure 10: Raw THC emissions for operating point 2000 rpm / 220 Nm

3.4 Raw CO Emissions

The emissions results in Figure 11 and Figure 12 for CO were similar for all spark plug based combustion concepts. Only at lower load points there are clearly higher emissions from the Diesel pilot engine, especially pronounced at λ=1. However, CO emissions are generally not an issue if an oxidation (or three-way) catalyst is used.

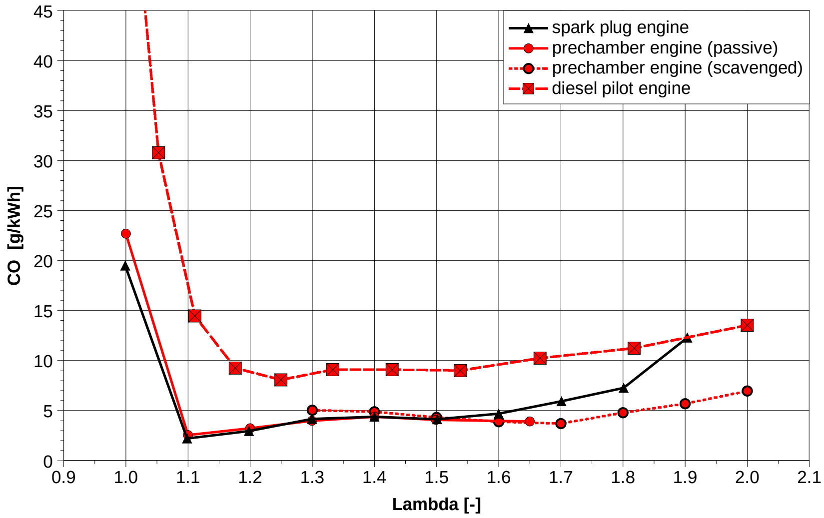

Figure 11: Raw CO emissions for operating point 1400 rpm / 50 Nm

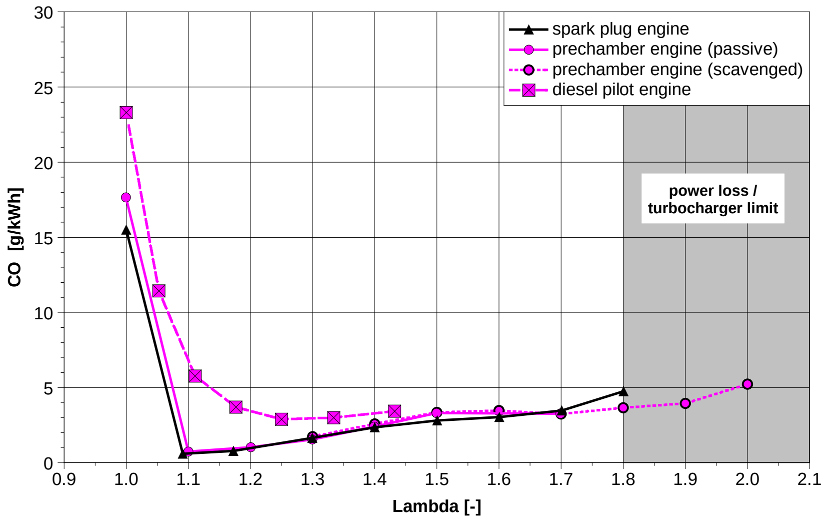

Figure 12: Raw CO emissions for operating point 2000 rpm / 220 Nm

3.5 Exhaust Temperature Levels

Figure 13 and Figure 14 show the measured temperature levels of the exhaust gases at turbine exit. This represents a location upstream a potential exhaust aftertreatment and indicates the thermal range of operation which could be expected for such a device. Due to the increasing air excess at lean burn operation, temperatures drop significantly and the temperature levels can become challenging for catalytic conversion, especially for methane.

At lower load points the lowest exhaust gas temperature can be observed for the prechamber version. This is because the prechamber operated engine showed higher wall heat losses (as described earlier). Significantly higher exhaust gas temperature levels for low load operation can be seen for the diesel pilot engine. This is in line with the drawback in brake efficiency levels for such low load operation, as discussed in section 3.1.

The lower temperature level for lean combustion leads to challenges for exhaust gas aftertreatment but it reduces the thermal requirements for the turbocharger so that for example affordable variable turbine geometries can be used.

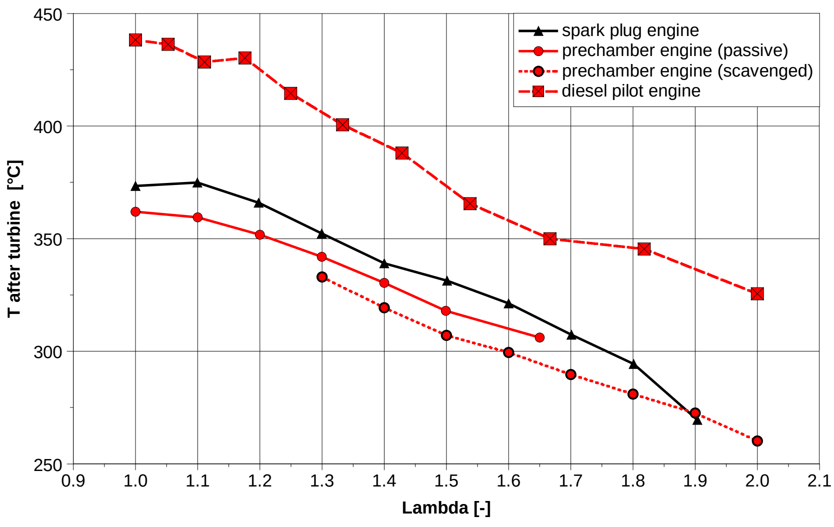

Figure 13: Temperature after turbine for operating point 1400 rpm / 50 Nm

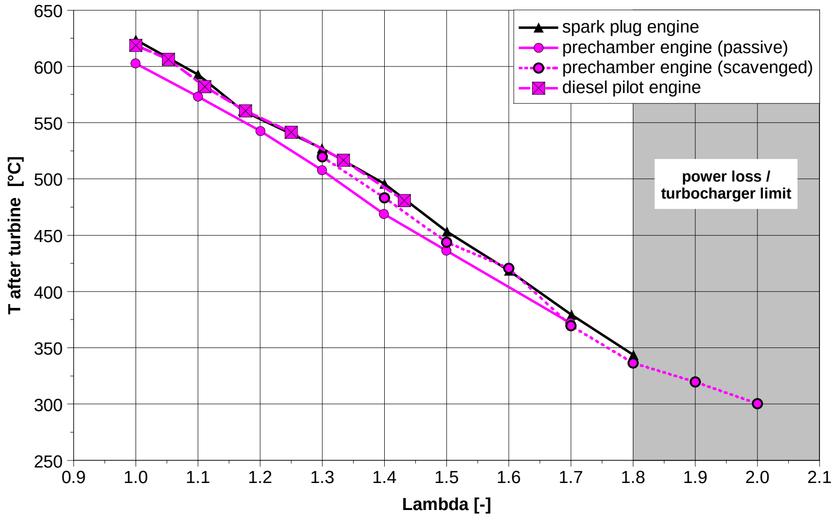

Figure 14: Temperature after turbine for operating point 2000 rpm / 220 Nm

3.6 Net Heat Release Examples

Figure 15 shows the comparison of the net heat release rates for λ=1.5 operation. Another operating point (1500 rpm / 150 Nm) is chosen here because all concepts could be operated here at good settings (no knock limitations and therefore centre of combustion at 8 °CA, minimal diesel amount for the diesel pilot concept).

The largely different net heat release rate of the spark plug engine versus all the other concepts is eye-catching: The spark plug engine has a long ignition delay and a comparably slow combustion. All other concepts show much faster combustion. For scavenged prechamber operation, the heat released from the prechamber can clearly be seen in the overall net heat release rate. Scavenged prechamber operation leads also to an extremely short ignition delay of only 1-2 °CA. The reason for this is that the ignition conditions in the prechamber are very good which enables an almost initial build-up of the flame kernel in the prechamber. However, the diesel pilot engine showed the fastest late combustion phase.

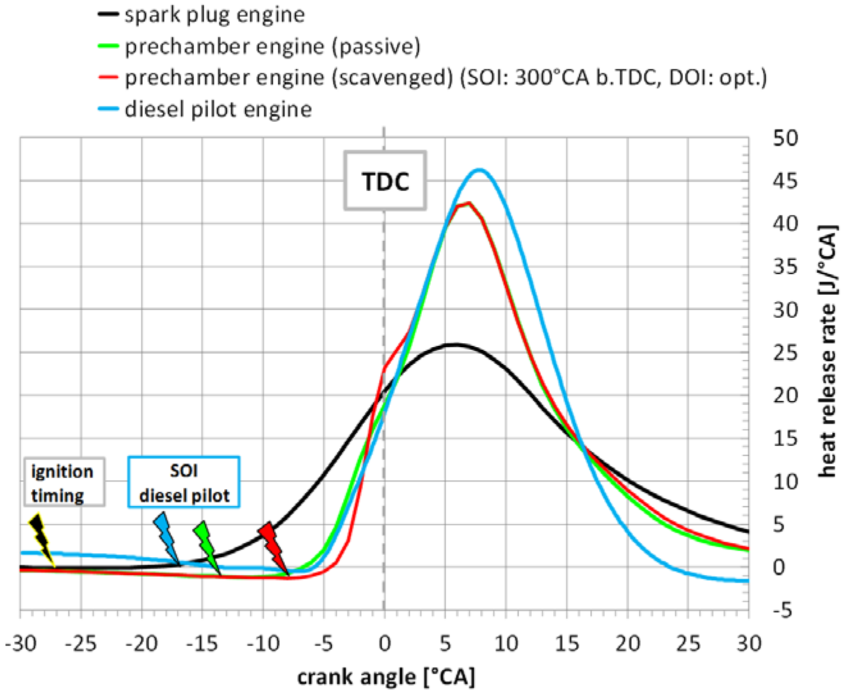

Figure 15: Net heat release rates for operating point 1500 rpm / 100 Nm and λ=1.5

4 Conclusions

Among the engines considered here, the Diesel pilot engine showed nominally the highest peak brake efficiency of around 43% at λ around 1.4 at the high load operation. There, stable combustion could be reached with only small Diesel pilot energies of about 1%. With decreasing load, especially at throttled operation, the Diesel pilot quantity had to be increased (e.g. to approx. 70% at loads around 2 bar bmep) to enable stable ignition and combustion. Extremely low loads made diesel pilot operation impossible anymore and the engine had there to be operated in pure diesel mode. Therefore, lean Diesel pilot combustion proved to be a fuel-efficient concept for mainly high load operation.

The spark ignited concepts showed efficiencies very close to the diesel pilot engine. These very similar peak efficiencies of all concepts are encouraging since the spark ignited engines had a considerably lower compression ratio (about 14.5 instead of 16.5). In addition, low load operation of both spark ignited concepts was possible without any problems and at good efficiency levels.

NO_x and THC emissions are the main concern for such efficient lean combustion concepts. NO_x and THC emissions show opposing trends: Whereas NO_x decreases with increasing air excess, THC increases. Comparably low NO_x levels can be achieved at lean operation which could be reduced using an SCR system or a NO_x storage catalyst. However, lean THC (methane) reduction at comparably low temperature levels is yet the missing aftertreatment component. So, the scientific community is encouraged to perform R&D on lean methane oxidation catalysis.

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