RCCI in Heavy Duty Engines

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Abstract. The simultaneous reduction of fuel consumption and pollutant emissions, namely NO_x and soot, is the predominant goal in modern engine development. In this context, low temperature combustion (LTC) concepts are believed to be the most promising approaches to resolve the above mentioned conflict of goals. Dual fuel, Natural Gas (NG)-diesel in Reactivity Controlled Compression Ignition (RCCI) is a promising high efficient combustion concept that combines EURO-VI engine-out NO_x and PM emissions with 25% CO_2 reduction compared to mono-fuel diesel and natural gas reference.

Main challenges are:

• CH_4 emission reduction: to meet EURO-VI tailpipe target of 0.5 g/kWh;
• Increasing load range: up to 23 bar BMEP;
• Combustion control to enable transient and robust operation under real-world conditions

Different natural gas-fuelled concepts exist. Stoichiometric natural gas SI combustion is widely accepted because of its maturity and simple aftertreatment. Driven by fuel costs and CO_2 emission targets, also conventional dual fuel concepts are introduced, in which natural gas is blended in standard diesel combustion. Depending on the injection timing of the diesel pilot injection, the process is either called liquid spark or RCCI. Conventional dual fuel concepts produce significant NO_x emissions as outcome of the diesel combustion. In RCCI the well homogenized mixture of the diesel pilot and the natural gas result in much cooler room ignition, controlled by injection timing and diesel injection amount.

So far, Single Point Injection and Port Fuel Injection of natural gas are applied in Natural Gas-Diesel RCCI. In this paper, the novel Direct-Injection (DI) Natural Gas technology is studied to reduce in-cylinder CH_4 emission and maximize thermal efficiency. On a heavy duty single cylinder engine the NO_x and soot engine out emissions below the EURO-VI targets have been undergone with ease. Efficiency in all investigated loadpoints is on a high level, even the challenging lowload area. Yet the CH_4-Emissions are about 50% over the target. This "CH_4-Slip" called CH_4-Emissions, result mostly from unburnt fuel. It was expected to result from the quenching area due to stopped flame propagation. A new RCCI-Pistonshape is introduced with raised compression as no knock occurres at all. This shape establishes no quenching and raised in cylinder temperatures to improve CH_4 combustion. The result from four most relevant variables are discussed in this paper. All these measurements are compared at the A50 loadpoint, which was predefined as 6300J/cycle injected fuel energy.

Notation

ATDC	*After top dead center*
BMEP	*Break mean effective pressure*
CA	*Crankshaft angle*
CFD	*Computational fluid dynamics*
CHI (χ)	*Energetic ratio*
CNG	*Compressed natural gas*
DI	*Direct injection*
EGR	*Exhaust gas recirculation*
EPS (ε)	*Compression ratio*
ETA (η)	*Thermal efficiency*
FID	*Flame ionization detector*
FSN	*Filter Smoke Number*
HRF	*High reactivity fuel*
IMEP	*Indicated mean effective pressure*
LAMBDA (λ)	*Air–fuel equivalence ratio*
LRF	*Low reactivity fuel*
LTC	*Low temperature combustion*

BMEP	*Mean effective pressure*
RCCI	*Reactivity Controlled Compression Ignition*
SOI	*Start of Injection*
TDC	*Top dead center*
W_CNG	*Start of CNG injection*
W_Diesel	*Start of Diesel injection*
X50	*center of mass conversion*
mf_CH_4	*Methane slip / Effective methane emission*
mf_NOx_g_e	*Effective NO_x emission*

1. Introduction

Intensified environmental requirements necessitate further improvements of combustion processes in heavy duty engines. NO_x and soot formation in particular are typically associated with conventional diesel engines. While further improvement of combustion efficiency in DI-Diesel engines leads to higher combustion temperatures and therefore an increase in NO_x formation, research is beginning to focus on new strategies for low temperature combustion to achieve further improvements in thermal efficiency and simultaneously minimize the formation of pollutants.

A promising approach is the implementation of a homogeneous compression charge ignition which makes it possible to achieve diesel engine-like thermal efficiency with gasoline engine-like emissions [1]. In HCCI the combustion process is initiated by an increase in chemical radicals during the compression process. Combustion starts simultaneously in multiple areas of the combustion chamber and proceeds without flame propagation, avoiding local temperature peaks that lead to thermal NO_x formation. Furthermore, the heat release rate is not limited by the speed of flame propagation which makes it possible to approach idealized Otto cycle. Figure 1 shows the model concept of ignition by exothermic centers. These exothermic centers, which can be described as elements of stoichiometric fuel air mixtures, continuously exchange heat with their environment. Keeping some distance between these centers can be beneficial for the prevention of local temperature peaks and therefore thermal NO_x-formation. On the other hand, the temperature of the surrounding gas must be high enough to initiate the decay of fuel molecules. A common way to achieve these temperatures is through the use of hot recirculated exhaust gas.

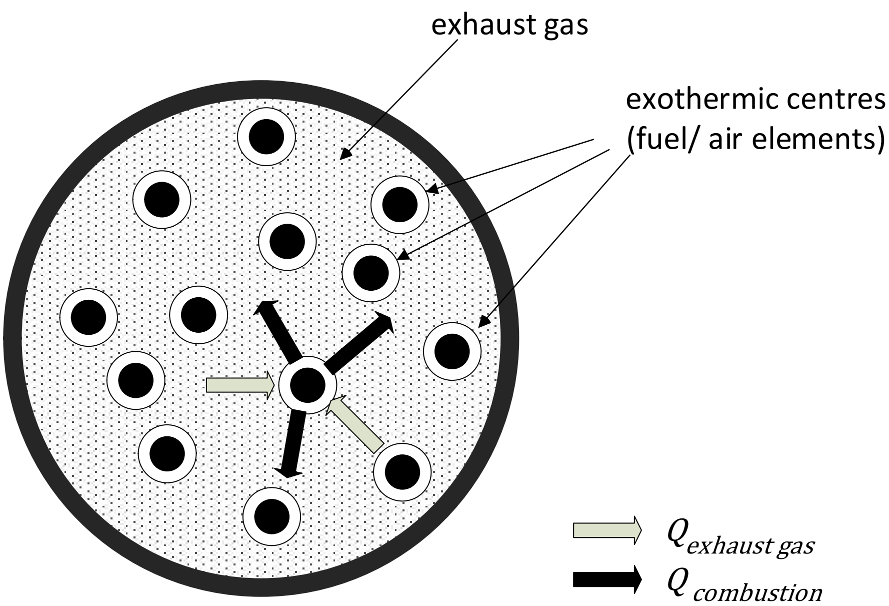

Fig. 1. Combustion control by auto ignition in exothermic centers, redrawn from [2]

To control the auto-ignition process, the rate of chemical decay can be influenced by increasing or lowering the intake charge temperature or varying the engines compression ratio. In comparison to conventional gasoline or diesel engines in which the ignition timing follows a trigger event such as spark timing or the beginning of diesel fuel injection, the control of HCCI is more difficult. Especially in cold start situations or at low loads, the required temperature for auto ignition cannot be achieved. At high loads on the other hand, the operation range is limited by the engine's resistance to high in-cylinder pressure. In general, the main Challenges of HCCI operation are the inability to reach high loads and the complex control of combustion timing [3].

A possible method to achieve the benefits of homogeneous auto-ignition over a wider operating range is the use of reactivity-controlled compression ignition (RCCI). A more efficient control over combustion processes has the potential to lower both fuel consumption and pollutant formation compared to HCCI [4].

In RCCI, Auto-ignition is controlled by varying the energetic ratio χ (CHI) of two different fuels which makes it possible to effectively control combustion phasing without negative implications on NO_x emissions [5].

$$\chi = \frac{H_{u,LRF}}{H_{u,LRF} + H_{u,HRF}}$$

A chemically homogeneous base mixture of a low reactivity fuel (LRF) is mixed with a small amount of high reactivity fuel (HRF), injected during compression stroke. By increasing the amount of high reactivity fuel, combustion can be initiated even at low loads, while lowering the reactivity increases knock-resistance at high loads. While the LRF delivers the majority of chemical energy, the injection strategy of HRF has a huge impact on ignition-timing and heat release behavior. Controlling injected mass and injection timing of HRF provides two additional degrees of freedom. Early Injection-timing increases the degree of homogenization while a late injection strategy causes a stratification of HRF. Eichmeier distinguishes between stratified diesel combustion, partly stratified diesel combustion and premixed diesel combustion [6]. An overview of the emission and combustion characteristics of these different operation strategies can be seen in Figure 2.

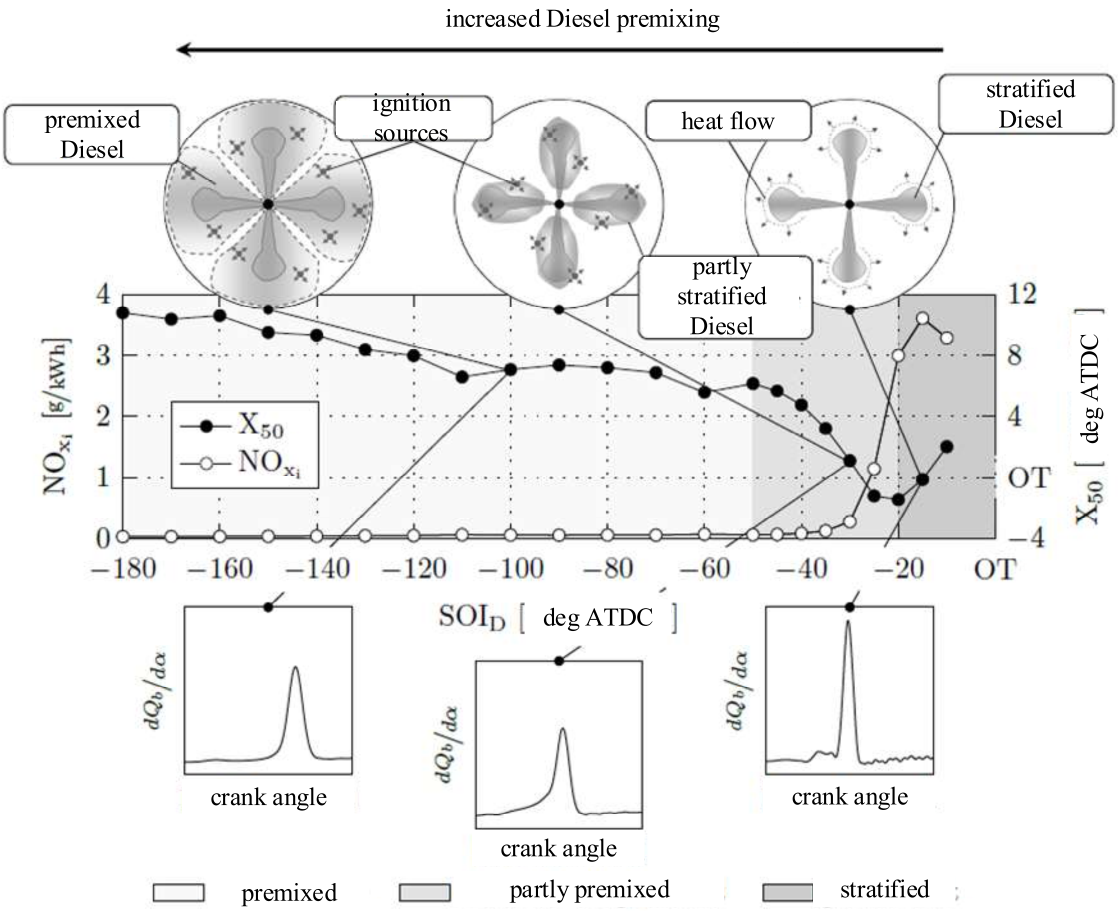

Fig. 2. Influence of HRF injection timing (SOI) on emissions and combustion behavior, PMI=10bar [6]

Stratified diesel combustion is characterized by high soot and NO_x formation due to the combustion of rich mixture parts and local temperature peaks. Heat release typically occurs in a pronounced two stage combustion. In the first stage, areas of rich diesel mixture oxidize under high soot formation. The resulting heat release leads to abrupt combustion of lean mixture after a short delay. Because of strong pressure oscillations combined with adverse emission characteristics, operating points with stratified diesel combustion should be avoided.

Shifting Diesel SOI earlier, the start and center of combustion move in the same direction until earlier start of injection does not lead to an earlier combustion. At this point, lower temperature and pressure levels cause an increase of ignition delay time. Areas containing HRF are larger and diesel concentrations in these areas are decreasing. These operating points of partly stratified diesel are characterized by an inverse behavior of SOI and start of combustion. Earlier injection causes an increase in ignition delay and therefore a later center of combustion. Homogenization increases while combustion temperatures decline. NO_x and soot formation are sinking until earlier injection causes a rise of unburned hydrocarbon concentration in exhaust gas [7].

In research carried out at the University of Wisconsin, RCCI has shown significant reduction in NO_x and soot emission in combination with an improvement in indicated efficiency. Simulations by Kokjohn et al. showed an efficiency improvement due to reduction of heat transfer and improved combustion control. [8]

2. Test procedure and equipment

2.1 Testbench

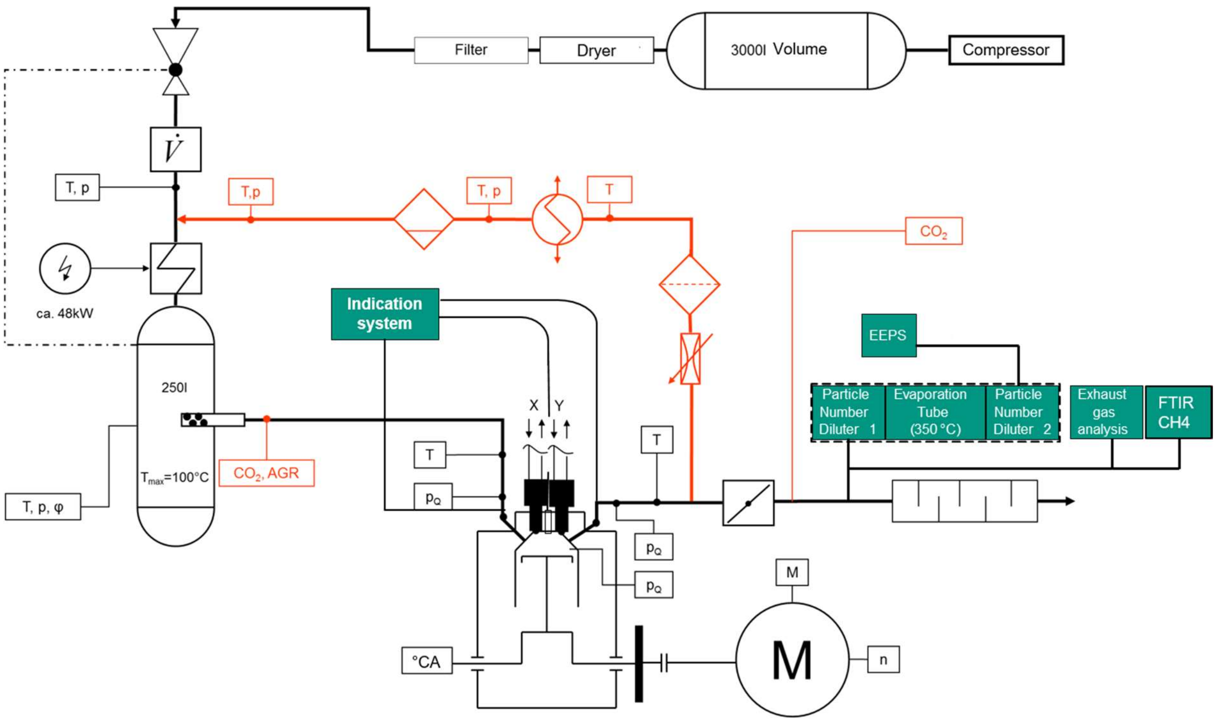

Fig. 3. Single Cylinder Testbench Scheme at KIT

The testbench of a single cylinder heavy duty engine used in this study is shown in Figure 3. The 4-quadrant electrical dynamometer starts and controls the engine. The base engine is built on the Daimler OM472 with adapted cylinder head to introduce a Dual-Fuel injection system. CNG and a liquid fuel (Diesel) are direct injected into the chamber by a second-generation needle in needle Westport HPDI² injector, the successor of the first generation HPDI injector. Full engine data is shown in Table 1. Cooling and lubrication are supplied externally and controlled to a maximum temperature deviation of 2K. Dry charge air supply is provided by the testlab and preheated to a desired engine intake mixture temperature with a deviation of 0.5K. Cooled high pressure EGR is introduced by using a backpressure flap in the exhaust and a standard EGR-Valve. EGR and charge air are mixed in dampening devices to ensure a homogenous mixture and to reduce the pressure fluctuations at the volume flow rate metering. The

EGR rate is calculated by ABB URAS, balancing the CO_2 amount in the inlet and exhaust. CNG-Rail pressure is adapted to match Diesel-Rail pressure by an external compressor station while constantly measuring gas quality. Intake, exhaust and incylinder pressure traces are recorded in the loop by Kistler Kibox indication system at a resolution of 0.1°CA.

Emissions are sampled with different 45° cut midstream probes. Soot is measured with AVL-Smokemeter as FSN. Particle Number concentration is counted in a Condensation Particle Counter AVL APC. The Particle Size Distribution is classified with an TSI EEPS. The gaseous emissions are analyzed in the AVL AMA 4000 with a 2 channel FID to separate the CH_4-Slip from the rest of the remaining hydrocarbons.

Table 1. Testbench data

ELIN Dyno	330kW, 4-Quadrant
Base engine	Daimler OM472
Bore x stroke	139 x 163mm
Injection System	Westport HPDI²
Railpressure	300bar CNG / Diesel
Charge air pressure	Up to 10bar
Intake temperature	20°C - 100°C
External cooled EGR	0% - 50%

2.2 Methodology

The measurement data from the automation System FEV Morphee 3 is averaged over a 60s period. Before sampling a stabilizing criteria and gas transportation time have to be complied. On trigger 100 consecutive cycles are taken out of the Indication systems ring buffer. Those two datasets are combined in Matlab and postprocessed. The indicated data is averaged and based on the fast heat release model, the necessary characteristics are calculated. The net indicated pressure IMEPn from this calculation is used in this study as the base engine power output. The break effective power can be misleading since friction losses, charge air and other media supply are externally on the single cylinder. Still other external devices like the backpressure flap influence the IMEPn too. In the shown sweeps the backpressure (P3) is taken from a turbocharger map and set to 150mbar. To improve comparability of the combined dataset, specific values are calculated and attached to it. One way is to use the EG88 directive for heavy duty gasoline engines. This directive is chosen since the main energy source in this study is CNG and the ignition process is controlled with the second fuel or often called pilot. In the means of, the main energy source is ignited by a different source. Secondary this directive supplies the method to calculate results comparable to EURO-VI legislation limits. For detailed combustion analysis a GT-Power single cylinder model is used. Woschni-Huber is used as the wall heat transfer model and external parameters are automatically setup for the cases in the Matlab routine. The setup then sets the boundary conditions for a Three-Pressure-Analysis (TPA) in GT-Power using the above mentioned three pressure traces. Single cycle events are parsed in Kibox Cockpit.

The sweeps presented in this study are calibrated on the energy amount injected per cycle. On the reference engine, load points were selected and the injected energy per cycle was calculated. This energy amount was always kept same for the comparing load points. As a result, an increased efficiency leads to a raised IMEPn. For the investigation of global effects, the center of mass conversion (X50) or the phasing was kept constant. Most of the diagrams are presented with the X50 at 8°ATDC. When necessary single parameter sweeps with X50 variations complement those sets.

As shown in the introduction and as Nieman et al [5] discussed, combustion losses on lean operated CNG engines result mainly from bulk losses in the squish region, crevice losses and blow-through losses. Since the base engine is derived from the standard diesel OM472 the shape of the combustion chamber is not optimized for Dual-Fuel mode. The base engine application for CNG ignited by a spark plug uses CNG port injection. CNG is distributed equally over the whole combustion chamber and a homogeneous mixture is achieved. While using the base camshaft configuration this results in some points in blow-through losses. This issue is directly solved by introducing CNG-DI and injection timings after the exhaust valve is closed. Still the injection in the overlap is part of the testing to show the constraints.

The combustion chamber shape is optimized in a three-step approach. Start is a port fuel and sparkplug adapted omega shape bowl with the compression ratio ε=13.1. The squish height here is 1mm. Following the previous work of Eichmeier et al the bean shaped RCCI bowl is introduced with the compression ratio ε=13.5. The squish region is significantly reduced with the new shape. Especially the squish height of now 7mm is expected to reduce bulk losses. The third step is lowering the distance between cylinder head and piston by 1,7mm. This raises ε to 15.4. On every shape and compression ratio the same sweeps are, if possible, performed to compare results. Both bowl shapes are displayed in Figure 4.

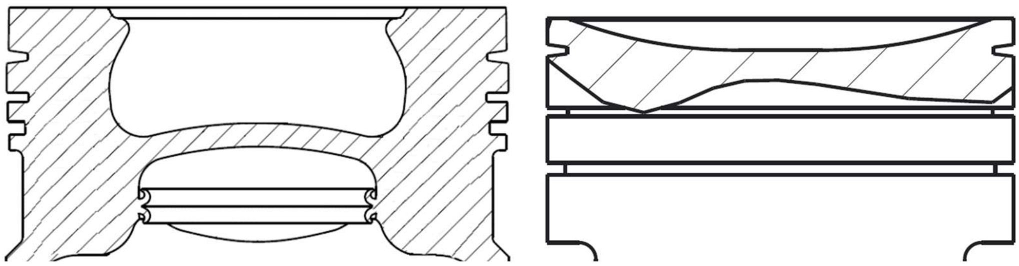

Fig. 4. Left stock port fuel piston ε=13.1, right RCCI piston ε=13.5 / 15.4

3. Results

Starting conservative on a low reactive mixture RCCI with CNG-DI is achieved. To avoid engine damage a switch from liquid spark to RCCI is performed at the 7,5bar BMEP loadpoint. By forwarding Pilot SOI the combustion phasing is forwarded close to an X50 of 0°ATDC. Going further from this point delays the phasing again and partial to full RCCI is reached. Most significant sign is the drop in NO_x emissions since combustion temperature decreases. Then the load is set to 10bar BMEP which equals 6300J/Cycle injected. All the shown sweeps are on this 6300J/cycle mode and referred to as A50. Filter Smoke Number (FSN) is measured but due to the homogenous mixture and the high CNG share it is always below FSN=0.04. After first optimizations it is below FSN=0.02 and won't be shown in the diagrams. Instead particle number measurements are introduced and shown if necessary. For every specific emission there is two main trends that influence the results aside from the raw emission in ppm. Reducing exhaust mass flow, reduces the specific value. While reducing thermal efficiency increases the specific emissions since the power output is reduced. The railpressure for Diesel and CNG is set to 300bar.

3.1 Fuel to air Ratio λ

One of the first project sweeps in RCCI mode is shown in figure 5 on the left side. At CHI=0,91 it is possible to stabilize the combustion. These λ variations are performed by reducing the charge air pressure (P2) and holding the fuel Energy at 6300J/cycle and the EGR-Rate at 30%. At P2=2500mbar the mixture is diluted to λ=2. By reducing P2 the dilution decreases and λ=1.1 is reached at P2=1450mbar. NO_x Emissions for lean combustion are at almost 0g/kWh equal to 2ppm and stay below 0.5g/kWh or 60ppm till λ=1.4. For rich mixture the NO_x threshold of 0.5g/kWh is exceeded. More striking is the CH_4-Slip (mf_CH_4). Operating at λ=2 results in 15% unburnt CNG emitted. Even at λ=1.1 emitting 5g/kWh means 2% of the CNG injected is not ignited. λ≤1 is not feasible due to an increasing number of cycles not igniting at all. In succession emitting the complete cylinder load into the exhaust and endangering the testbench. The efficiency already over 40% is promising, especially considering the amount of unburnt fuel. Figure 5 on the right shows the same sweeps but now optimized with CHI=0,83, T2=60°C and EGR=35%. Most significant change is the reduced CH_4-Slip (mf_CH_4) of 0.8g/kWh and the Pilot SOI (W_Diesel) that is forwarded to -70°ATDC. Even for λ=2 the CH_4-Slip with 2.5g/kWh is way below the start and the efficiency remains around 44%.

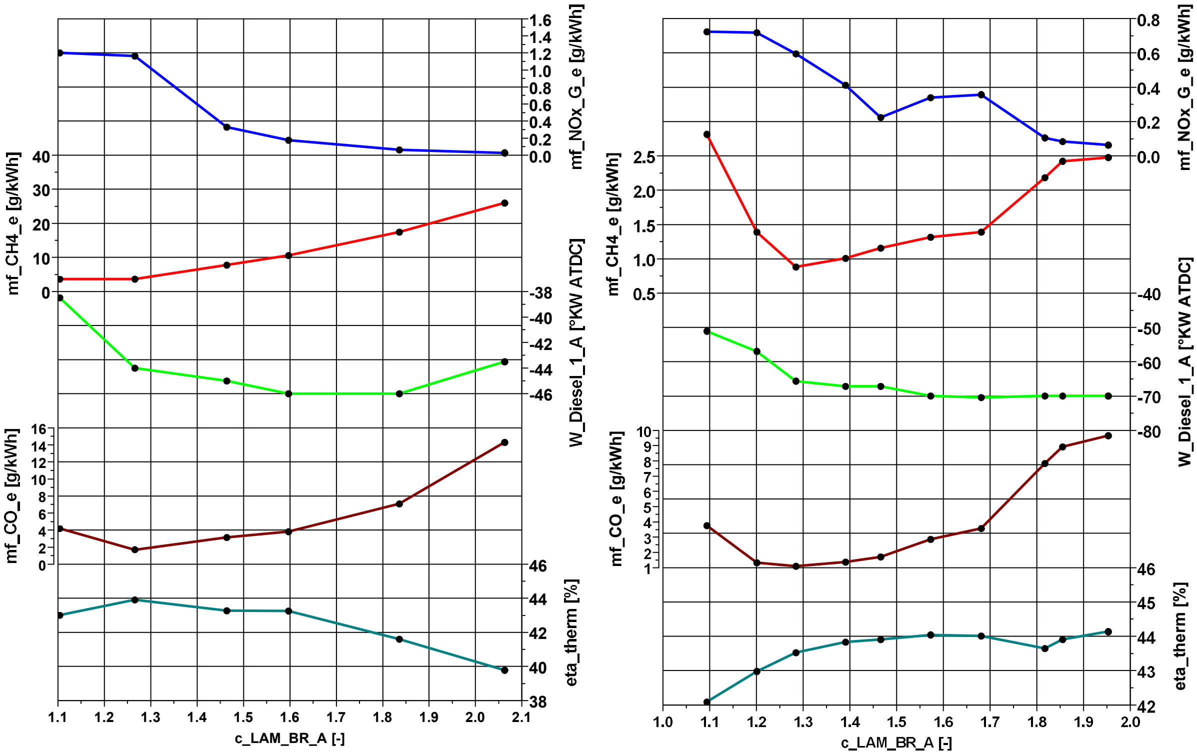

Fig. 5. λ-variation left at CHI=0.91, right optimized at CHI=0.83; P2=1450-2500mbar, 6300J/Cycle, ε=13.1

3.2 CNG Start of Injection

The major advantage of CNG-DI instead of PI is expected to be the degree of freedom for the injection timing and the in-cylinder mixture formation. A variation of the CNG start of injection (W_CNG) at ε=13.1

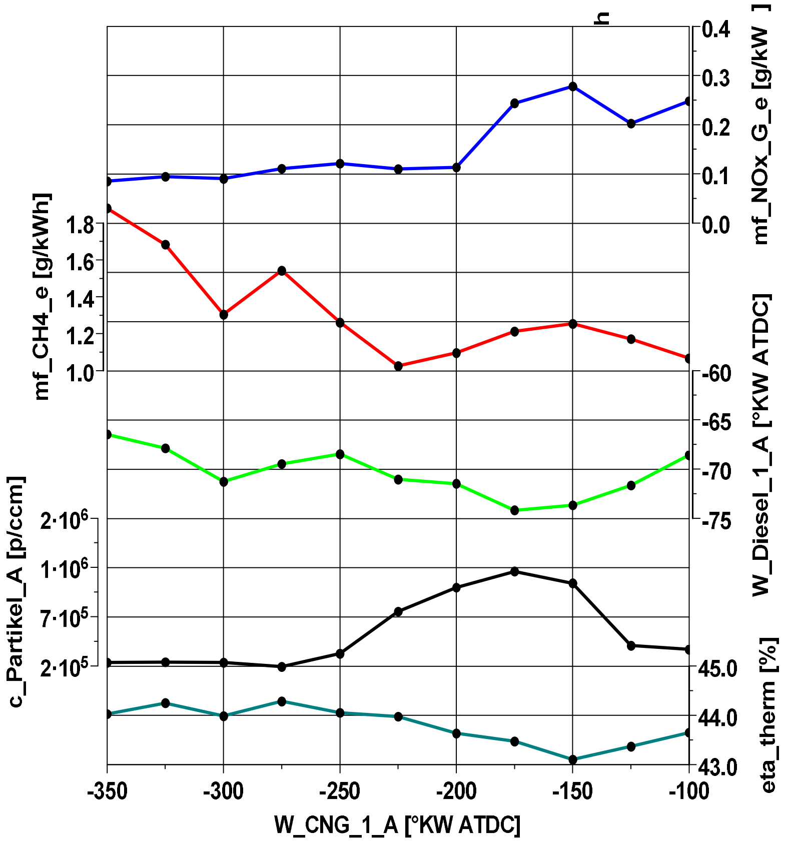

Fig. 6. CNG SOI sweep; λ=1.5, P2=2100mbar, EGR=35%, 6300J/Cycle, ε=13.1 stock bowl

and stock bowl is shown in figure 6. As seen before the region around λ=1.5 is a good compromise and is chosen for the shown figure. First to notice is the NO_x maximum (mf_NO_x) 40% below the EURO-VI-Target. At CNG-SOI -200°CA ATDC and earlier NO_x emissions stay at 0.1g/kWh. With Pilot SOI earlier then -65°CA ATDC the diesel spray homogenizes and the combustion temperature is low. This is achieved by reducing CHI to 0,83. The trends for the Diesel share and the Pilot SOI are discussed later in Figure 7. CH_4-Slip for early CNG-SOI is a result of the combination of the injector setup and valve timing. The CNG-jets are injected at an angle of 20° close to the flat cylinder head at a pressure of 300bar. Jets hitting the not fully closed exhaust valve can lead to emitted fresh load. Injecting later reduces the slip until a plateau starting -225°CA is reached. Retarding the CNG injection further stabilizes the slip between 1-1.2g/kWh or 400 to 500ppm. At late injections from -100°CA and further the process becomes instable. The injected diesel amount is fluctuating cycle to cycle and the phasing control is lost. This leads to high pressure rise rates at early X50 and misfire on the other side. A concentrated mixture only in the bowl is not possible to achieve on this setup. The efficiency only decreases for the earliest Pilot SOI that are necessary at the late CNG angles. Forwarding Pilot SOI was crucial here because the mixture became more reactive. For the further data shown -250°CA CNG SOI is used in the next sweeps.

3.3 Energetic CNG to Diesel share CHI

As there are unexplained issues with the CNG-SOI it is believed, that for certain points CNG is trapped in the squish and crevice region. Secondary it is possible that the low reactive CNG is separated from the pilot Diesel at late Pilot SOI. Only the CNG inside the bowl would be able to react and the flame would not ignite the volume in the squish region. Following this approach the diesel share is increased from 6% to 22%. The improved reactivity leads to a faster heat release and forwards the combustion phasing. As the phasing will be kept constant the Pilot SOI is forwarded. The Diesel stratification is reduced and the two fuels mix better. As long as stratification decreases along with the increasing Diesel share, NO_x Emissions stay at the ultra low level as expected for RCCI. Figure 7 a variation over CHI, the contrary of the Diesel share illustrates this.

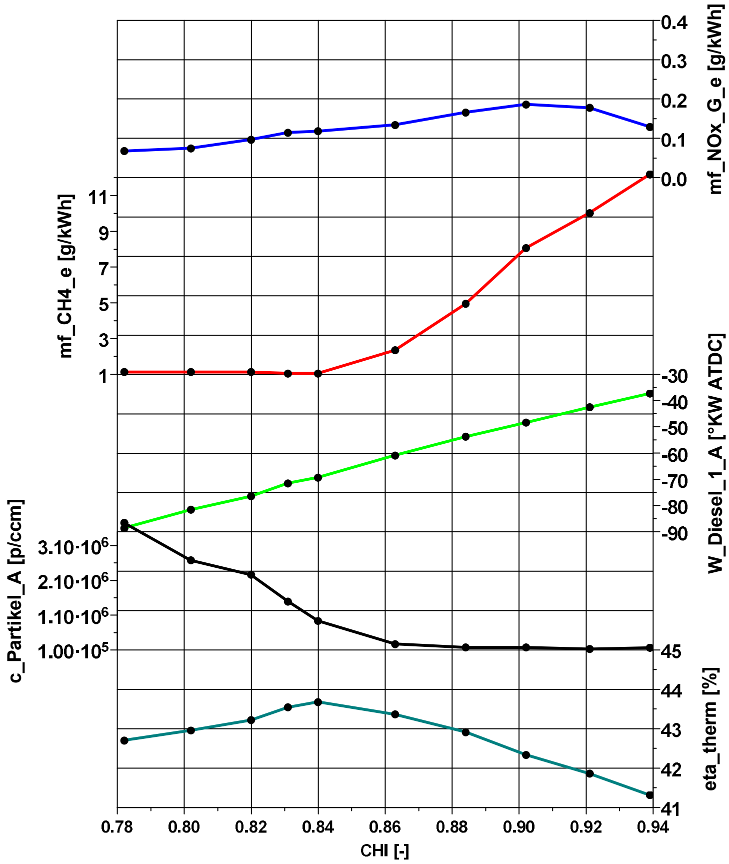

Fig. 7. CHI variation; λ=1.5, P2=2100mbar, EGR=35, 6300J/Cycle, ε=13.1 stock bowl

The Methane emissions decrease linear to reach the minimum of 0.9g/kWh at CHI 0.84. After this point more diesel only reduces efficiency. The pressure analysis shows a loss in combustion efficiency about 1% from CHI=0.84 to 0.78. The fact that the CH_4-Slip can't be reduced further shows that bulk losses are at their possible minimum with this setup. Especially the homogenised Diesel which is also in the squish region at the early SOIs helps to burn the trapped CNG in the later burn phase. The additional Diesel close to the walls not being burned completely at CHI<0.84 is the reason for efficiency loss, since CH_4-Slip is not increasing again. As Eichmeier et al discribed the CO emissions are also increasing. A big influence of CHI on the crevice losses is not expected. The new piston bowl with reduced squish heigt and fireland seem more promising. The burn duration is reduced from more than 40°CA to 20°CA and the pressure rise rates are still below the engine limits. Heat release traces are analysed in the next chapter. In figure 6 and 7 the rising particle number (c_Partikel) for Pilot SOIs earlier as -70°CA is recognisable. Analysis with the particle spectrometer EEPS show a growing peak for particles smaller than 30nm. In combination with CFD Simulations at VKA Aachen the assumption of Diesel hitting the cylinder walls and piston are underlined. To try to avoid this effect testing was done with separated pilot injections too. The result is an instable combustion. The energising time for the small Diesel amounts are too short and the needle stays in the ballistic region on this special injector. Subsequent optical investigations are supposed to improve the understanding of the particle trend better.

All of the seen phenomena lead to the introduction of the RCCI piston with a bigger squish height, lower fireland and a bean shape bowl.

3.4 Stock bowl compared to RCCI bowl

Figure 8 compares the above discussed results of ε=13.1 stock bowl in dashed lines and the ε=13.5 RCCI bowl. Fully matching ε was not possible due to constrution constrains but the difference is still in an accetable range. Expectedly on both sweeps the NO_x emission follows the same trend and stays on the same level. Only in the CNG-SOI sweep on the right at -100°CA the value is twice as high. The explanation can be found in the Pilot SOI. Injecting CNG at -100°CA and Diesel at -80°CA is at the edge of the injector control on this setup. In both diagrams the Pilot SOI follows the same trend and is forwarded between 5 to 10°CA. Reduced heat losses at the improved surface to volume ratio and the raised compression ratio make this necessary to keep the phasing constant at 8°CA ATDC. The new shape diminishes the influence of the CNG SOI on the Methane slip. The best point at -300°CA is only 0.2g/kWh or 100ppm below the worst point at -150°CA. Before the difference was around 1g/kWh or 400ppm. While being on the same level the best point on the new piston now is shifted to mixing with the entering charge air through the open inlet valves and fully homogenising. This behavior is reproducable over all the testing done with the new shape.

In Figure 8 on the left the most important change is the reduced Methane slip even for the high CHI region. The curve is flattened and mf_CH_4=1g/kWh is reached at CHI=0,86 instead of 0,84. The minimum of 0,9g/kWh can't be improved in this sweep. After this point introducing more diesel again only reduces combustion efficiency. On top more diesel reduces the sensitivity to Pilot SOI changes due to the early injections. Changing CHI is then the best solution to influence the X50 phasing.

The improved ignition of the CNG load is visible in the thermal efficiency which is always above 44%. From the point on where most of the CNG is combusting at CHI 0,9 and lower it is even over 45%. To make it comparable the efficiency was improved for 2 percentage points or 6% in total. This is directly achieved by changing the combustion chamber shape.

As seen in figure 9 the heat release for the RCCI bowl compared to the stock bowl is of the same shape. The low temperature reaction at -23°CA is at the same position only the combustion of the main load differs. The heterogenous spots ignite around the same angle (-12°CA) as on the stock bowl. Even though the incylinder pressure is around 1bar higher in that point. Yet the ignition delay on the more homogeneous rest mixture is longer and the main load then burns with a high heat release rate in a shorter duration. Thus, resulting in the same phasing. As a consequence, the peak pressure rises by 12bar to 130bar. The pressure rise rates are still within the engine limits.

In detail it can be seen that on the stock bowl the heat release flattens at 20°CA and cylinder load is still converting. On very high CHI this phase is prolonged and overlapped by a propagating flame up to 60°CA. This point is assumed to result from the squish region which will be verified in CFD and optical investigations. On the RCCI bowl with the increased squish height this phenomenon is barely visible.

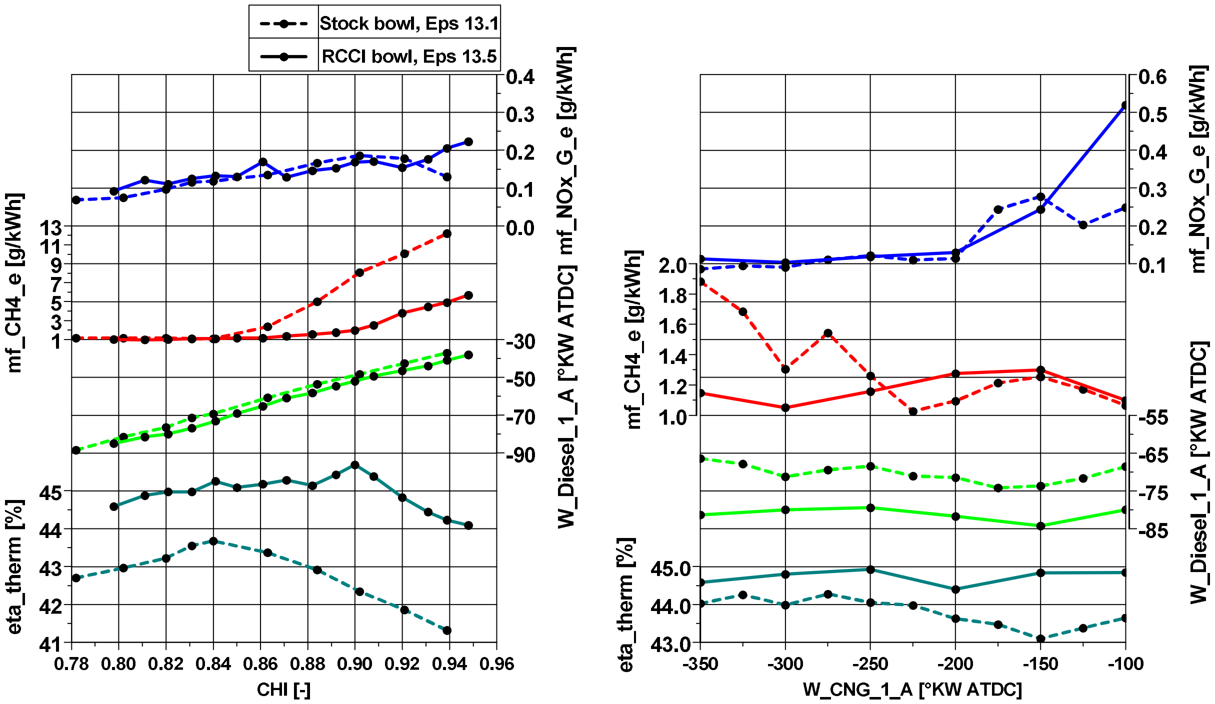

Fig. 8. CHI variation left, CNG SOI sweep right; Comparison of stock and RCCI bowl; λ=1.5, P2=2100mbar, EGR=35%, 6300J/Cycle, ε=13.1 / 13.5

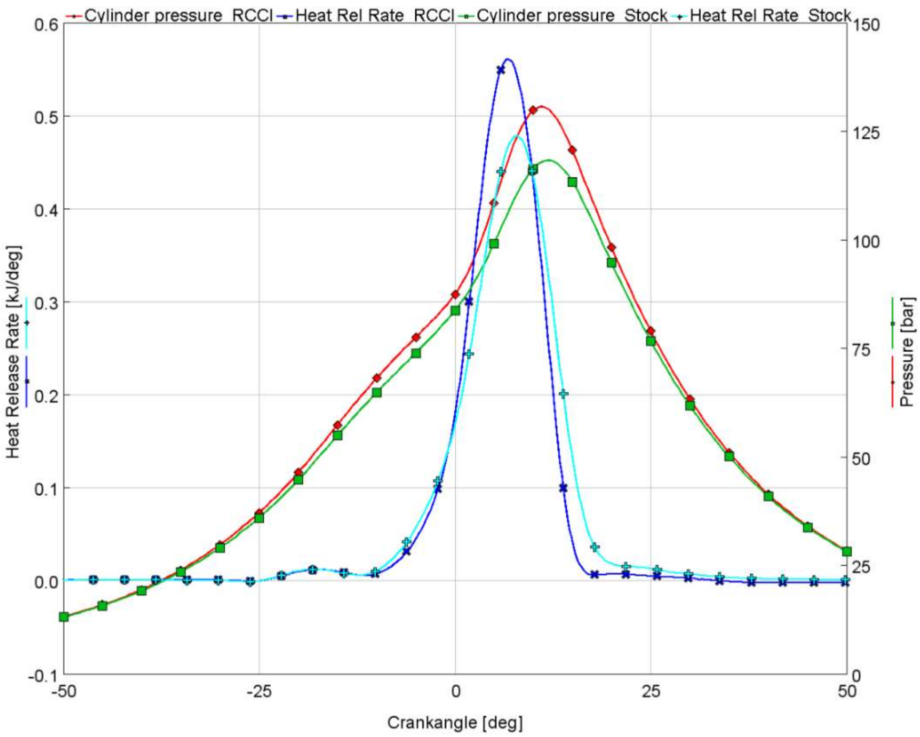

Fig. 9. Pressure and heat release trace over crankangle; λ=1.5, P2=2100mbar, EGR=35, CHI=0.83 6300J/Cylce, ε=13.1 stock and RCCI bowl with ε=13.5

3.5 Compression Ratio 15.5 and RCCI piston

In Figure 10 the CHI sweeps from the last two figure are displayed in comparison with the last setup. The cylinder is lowered by 1.7mm with the same RCCI piston. The resulting compression ratio is 15.5. This third setup is displayed dashed and with triangle markers. Again NO_x emissions are below 50% of the EURO-VI threshold and decrease with the increasing homogeneous Diesel distribution. Especially since the Pilot SOI has to be even earlier than -100°CA to control the combustion phasing. In detail it can be seen that the 15.5 setup has the lowest specific NO_x Emissions.
From the start at CHI=0.94 CH_4 emissions (mf_CH_4) are below the level achieved with the stock bowl at CHI=0.86. After CHI =0,88 there is no further improvement with the high compression. Main reason is the high reactivity leading to overstepping the engine limits for pressure rise rate. Also, beyond this point phasing control is limited. At the Pilot SOI of -100 to -120°CA the influence of changes to the phasing from the SOI is reduced dramatically. At ε=13.5 and a reactive mixture, changing Pilot SOI by 3°CA results in at least 1°CA phasing change. In this case changing the Pilot SOI by 20°CA results in almost 0 change on the phasing. Other effects like fluctuating EGR-Rates or a change in the load temperature dictate the combustion. As a consequence, the cycle to cycle deviation can suddenly increase in a way of not igniting cycles and then forwarding the phasing to 5°CA ATDC. In that moment engine limits are overstepped by 50% and the only useful and fast influence at hand is to increase CHI.

With an efficiency of more than 46% in a partial load point even high CHI seems sufficient. This high CHI region with CH_4-emissions under 2g/kWh and the yet highest measured efficiency is the direct consequence of the raised compression ratio. Regarding only the efficiency, CHI 0.93 has a good controllability, is enough homogenized but still stratified and needs less than 50% Diesel fuel compared to ε=13. Below CHI 0,9 the advantage is weakened. The CH_4-Slip stays on the same level but the combustion efficiency is reduced. Especially the here not shown CO emissions rise from this point on. This CO trend is visible for both compression ratios and it starts with the Pilot SOI of 75°CA and earlier. Possible flame / spray - wall interactions have to be examined closer on the optical setup.

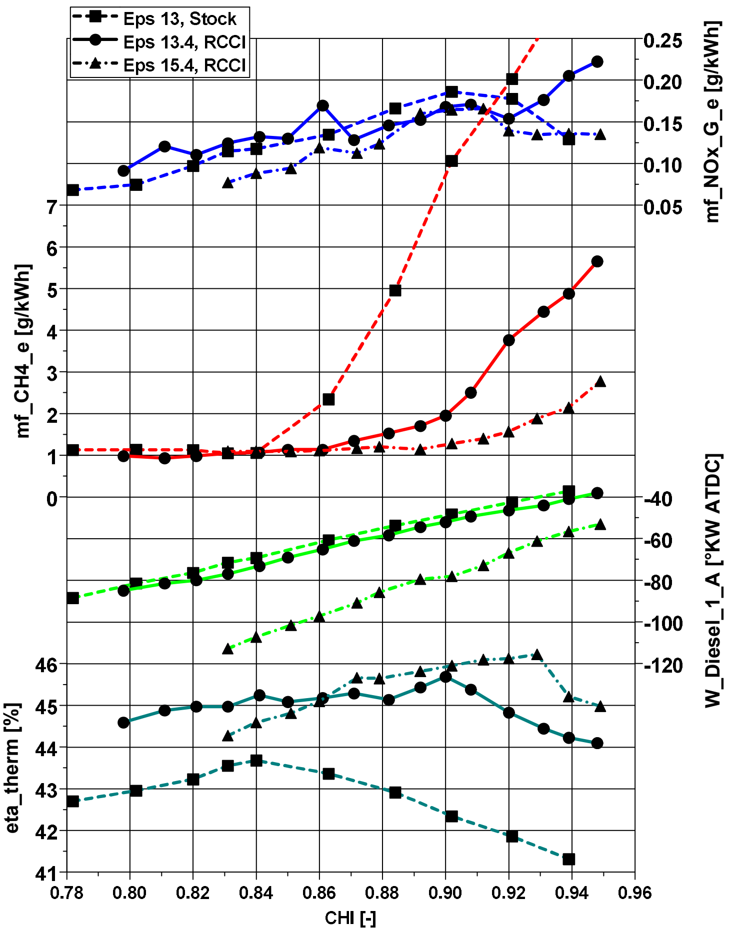

Fig. 10. CHI sweep; λ=1.5, P2=2100mbar, EGR=35, 6300J/Cycle, ε=13.1 stock and RCCI bowl ε=13.5/15.5

4. Conclusions

The data from the FVV Project “RCCI in Heavy Duty Engines” has shown the potential of RCCI with CNG / Diesel DI². Selected sweeps of the main parameters are displayed in this study. A cooled EGR-Rate of 35% is used which suppresses in combination with the early Diesel injections the NO_x formation. Less EGR is possible but then the Diesel SOI has to be retarded which leads to a more stratified Diesel inside the engine. Those rich regions ignite hotter forming more NO_x. At the correct EGR-Rate the inversed energetic Diesel share CHI does not influence the NO_x formation anymore. Increasing the Diesel share even reduces the NO_x emissions since the Pilot SOI has to be forwarded to keep the combustion phasing constant between 7 and 8°CA ATDC. NO_x emissions stay below the EURO-VI target for almost all tests using this strategy.

Lambda in the area of 1.3 to 1.5 shows the best results especially concerning the CH_4-Slip. The sweeps shown are measured at λ=1.5. Lower lambda has shown potential due to the reduced exhaust massflow but for high epsilon it was not possible to control the reactivity. For the comparability only λ=1.5 is displayed. By introducing sufficient Diesel the CH_4 emissions are minimized and more Diesel only reduces the efficiency. For higher compression ratios this Diesel amount is reduced, the trends stay the same and the overall Methane emissions are reduced. This is also the result of the improved combustion chamber layout and the increasing reactivity. As seen in figure 9 the heat release is higher and faster for the RCCI bowl

The used CNG has a Methane share of about 92% and has shown in this highly diluted process a demand of CHI in the region of 0,84 to be combusted by the homogenized but rich enough Diesel spots in the chamber. The EURO-VI target for methane emissions is not fully reached yet but with 0.75g/kWh in sight. The homogenization is enabled by Pilot SOIs between -80 and -60°CA. Those wide spread rich reaction spots ignite simultaneous all over the cylinder and in combination with the high methane number no knock occurred over all the testing. The only issue for the hardware can be the high pressure rise rates. This is mostly the case at the ε=15.5 sweeps which also need Pilot SOI earlier than -100°CA. With the lost phasing control this high compression ratio is excluded from further testing.

In figure 11 an overview of the best points of the sweeps is shown to close this paper. The most interesting figures are pointed out, namely the efficiency and the Methane emissions. In every point NO_x and soot emissions are below the targets. On the fullload points the limiting factor is the pressure rise rate. CHI=0,94 is the lowest possible. At 1000rpm and CHI=0.98 even 25bar BMEP are possible and realized but this testing is still ongoing.

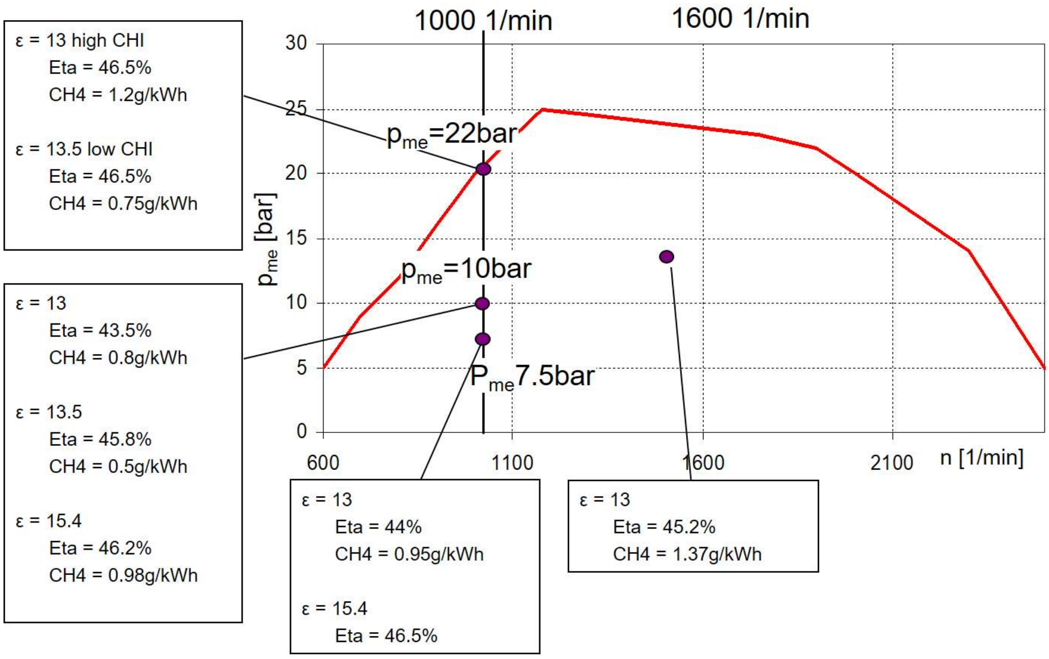

Fig. 11. Engine map with achieved best points in the project

Acknowledgement

This report is the scientific result of a research project undertaken by the FVV (The Research Association for Combustion Engines eV) and performed by Dipl.-Ing. Frederik Eise at Institute of Internal Combustion Engines at KIT under the direction of Prof. Dr. sc. techn. Thomas Koch.

The FVV would like to thank the professor Koch for the implementation of the project. Special thanks are due to the AiF (German Federation of Industrial Research Associations eV) for funding the project within the framework of the collective research networking (CORNET) programme. The project was conducted by an expert group led by Dr. Ingo Mikulic (Shell). We gratefully acknowledge the support received from the chairman/chairwoman and from all members of the project user committee.

In addition, we would like to thank the Daimler AG and Westport Inc for the provision of the single cylinder and the Injection system.

The research project was carried out in the framework of the industrial collective research programme (IGF/CORNET no. 123 EN). It was supported by the Federal Ministry for Economic Affairs and Energy (BMWi) through the AiF (German Federation of Industrial Research Associations eV) based on a decision taken by the German Bundestag.

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