

A Thermodynamic Study on Boosted HCCI: Motivation, Analysis and Potential

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ABSTRACT

Due to the increasingly stricter emission legislation and growing demands for lower fuel consumption, there have been significant efforts to improve combustion efficiency while satisfying the emission requirements. Homogeneous Charge Compression Ignition (HCCI) combined with turbo/supercharging on gasoline engines provides a particularly promising and, at the same time, a challenging approach.

Naturally aspirated (n.a.) HCCI has already shown a considerable potential of about 14% in the New European Driving Cycle (NEDC) compared with a conventional 4-cylinder 2.0 liter gasoline Port Fuel Injection (PFI) engine without any advanced valve-train technology. The HCCI n.a. operation range is air breathing limited due to the hot residuals required for the self-ignition and to slow down reaction kinetics, and therefore is limited to a part-load operation area.

Considering the future gasoline engine market with growing potentials identified on downsized gasoline engines, it becomes necessary to thoroughly investigate the synergies and challenges of boosted HCCI. Additional HCCI potentials can be realized through the additional breathing, resulting in extension of the operation map and further optimization of fuel efficiency. This provides additional potentials regarding test cycle efficiency. These potentials will be addressed in this work by thermodynamic analysis, drive-cycle simulations and split of losses. These analysis and simulations are based on single-cylinder and 4-cylinder engine data. As an outlook the new challenges associated with the boosted HCCI will also be discussed.

INTRODUCTION

A thermodynamic analysis and benchmark of various gasoline naturally aspirated combustion concepts at those part-load conditions that are frequently visited during the standard drive-cycles was performed in [1], in which the benefits and challenges of each combustion concept are benchmarked based on the analysis on thermodynamic split of losses [2]. Conventional stoichiometric spark ignition (SI) combustion enables higher fuel efficiency for example through variable valve actuation (SI-VVA). The HCCI combustion concept, also called controlled auto ignition (CAI), enables lean combustion that achieves high efficiency with very low NO_x emissions. This can be attributed to the dethrottling of the thermodynamic process and, at the same time, the high mixture dilution which leads to homogeneous combustion at lower temperatures. This combustion concept allows lean combustion without a deNO_x catalyst or other means of NO_x after-treatment. Using spray-guided stratified combustion (SGDI), one can achieve best fuel efficiency out of the SI engine due to the full thermodynamic dethrottling. The thermodynamic dethrottling is a consequence of fresh charge entering the process at atmospheric pressure. With HCCI, due to the high amount of residuals, the thermodynamic properties of the charge are not as positive for the process as inducting only fresh air with SGDI. HCCI combustion approach combines high thermal efficiency with low NO_x emissions. Furthermore, this concept offers the chance to meet future emission standards regarding CO and HC emissions in lean mode. Use of a standard three-way catalyst is sufficient since NO_x emission is very low.

In contrast to conventional gasoline engines, the start of the ignition during HCCI operations cannot be induced by the ignition timing. To trigger gasoline HCCI combustion compression of an air and fuel mixture has to achieve a specific temperature and pressure that will drive reaction

kinetics of the mixture into self ignition and combustion. The necessary thermodynamic mixture properties can be achieved by residual trapping or reinduction [3] or even air preheating [4]. During compression and after achieving self-ignition temperature, the mixture will start to ignite at various places simultaneously purely driven by reaction kinetics which are dependent on the pressure, the spacial temperature distribution, the mixture quality field and the residuals field. The heat release takes place without high local temperatures, which drastically reduces the formation of thermal nitrogen oxide. To reduce the high heat release rate from HCCI combustion, a high level of dilution is absolutely necessary to avoid high noise generation and mechanical stress. This, however, also indicates that gasoline HCCI is not applicable for full load. Due to the high dilution level used for gasoline HCCI, the dethrottling effect actually results from the reduction of the pumping losses.

Therefore, in this paper, the conditions for self ignition of the mixture were ensured by adjusting the necessary amount of fresh air, trapping hot residual gases and adapting fuel injection strategy and timing. Variable valve actuation in combination with direct gasoline injection is the best knob for HCCI combustion control [5, 6]. For transient application, cycle-to-cycle closed-loop control will be required to guarantee robust operation. In this case, the best way to control the right amount of hot residuals on a cycle-to-cycle basis is to use the fully variable valve actuation. It is worth noting that by using different breathing strategies, it is possible to get different mixture enthalpies and homogenization levels [7].

Gasoline Direct Injection plays a major role in gasoline HCCI combustion. It allows the use of various injection strategies which have direct influence on temperature, i.e. evaporation enthalpy, and also the temperature profile by varying injection timing during the residual trapping or compression stroke. This is achieved by varying the mixture heat capacity and therefore the specific heat ratio (polytropic coefficient) of the process changes. Additionally, direct fuel injection also has a strong effect on the reformation of the fuel when it is injected during the residuals trapping phase [7]. At a higher compression ratio and temperature, partial oxidation of the fuel could significantly change the temperature profile. Thus, the gasoline direct injection strategies offer multiple possibilities for direct control of ignition and combustion phasing. Additionally, multiple injections combined with the ignition of a small stratified-charge injection can be used to achieve a highly stable idling behavior [8].

In the presented work, the main strategy employed is to use a single injection event and residuals trapping via negative valve overlap (recompression). The timing for EVC, that determines the amount of trapped residuals, is relevant for load and combustion phasing together with injection timing. This is because of the direct influence on mixture quality

which, along with temperature profile, determines combustion phasing. In addition, IVC timing is optimized for maximum effective compression ratio.

The thermodynamic analysis comparing the HCCI combustion concept to other spark ignition combustion concepts used on gasoline engines underscores the advantages of HCCI [1]. Dethrottling of the combustion process (i.e., a large fraction of the mass participating in the thermodynamic process and the drastic reduction of the gas-exchange losses), the thermodynamic properties advantages resulting from the low-temperature heat release, and the fast burn rate allow for an approximation of the ideal constant-volume cycle lead to an increased thermal efficiency. As the auto-ignition process and the subsequent combustion event start simultaneously at multiple locations within the combustion chamber not being restricted to a single local constraint, different from that occurring for the flame propagation process during spark ignition. As a result, the cycle-to-cycle variations are less pronounced during HCCI operations [5].

As HCCI is applicable only to a limited region of the operation map, two combustion concepts must be combined within an engine. For that purpose, compromises have to be made such as the compression ratio. Although numerous parameters can be adjusted, HCCI cannot cover the entire engine map. As a result, the engine has to have the capability of switching between the HCCI and the conventional spark-ignition modes during real driving operations. Due to the extremely short combustion time and the early start of combustion, either a lean mixture or a high EGR rate is required in order to prevent high mechanical stress. Hence the load range of HCCI is restricted to the charging capability that can be achieved with a given residual-gas mass or a given air/fuel ratio. The lower load limit is determined by the ignition delay time beyond which the auto-ignition process cannot be initiated. As mentioned above, the lower load limit can be extended towards idling by a multiple-injection strategy combined with igniting a small stratified-charge to trigger HCCI [8].

MOTIVATION

For a vehicle of the upper compact class, the characteristic region of the urban part of the new European driving cycle can be covered by HCCI. Compared to a gasoline engine with port fuel injection without the advanced valve-train technologies, HCCI combustion combined with Direct Injection (DI) operation can achieve a fuel savings of up to approx. 14% [5] (Fig. 1). Specific HCCI operating points even achieve fuel savings up to 30% as brake specific fuel consumption plots in Fig. 2 (lower plot) show. The data shown there are based on engine measurements. A residuals recompression strategy using a two-step valve-lift system combined with direct injection was used for HCCI.

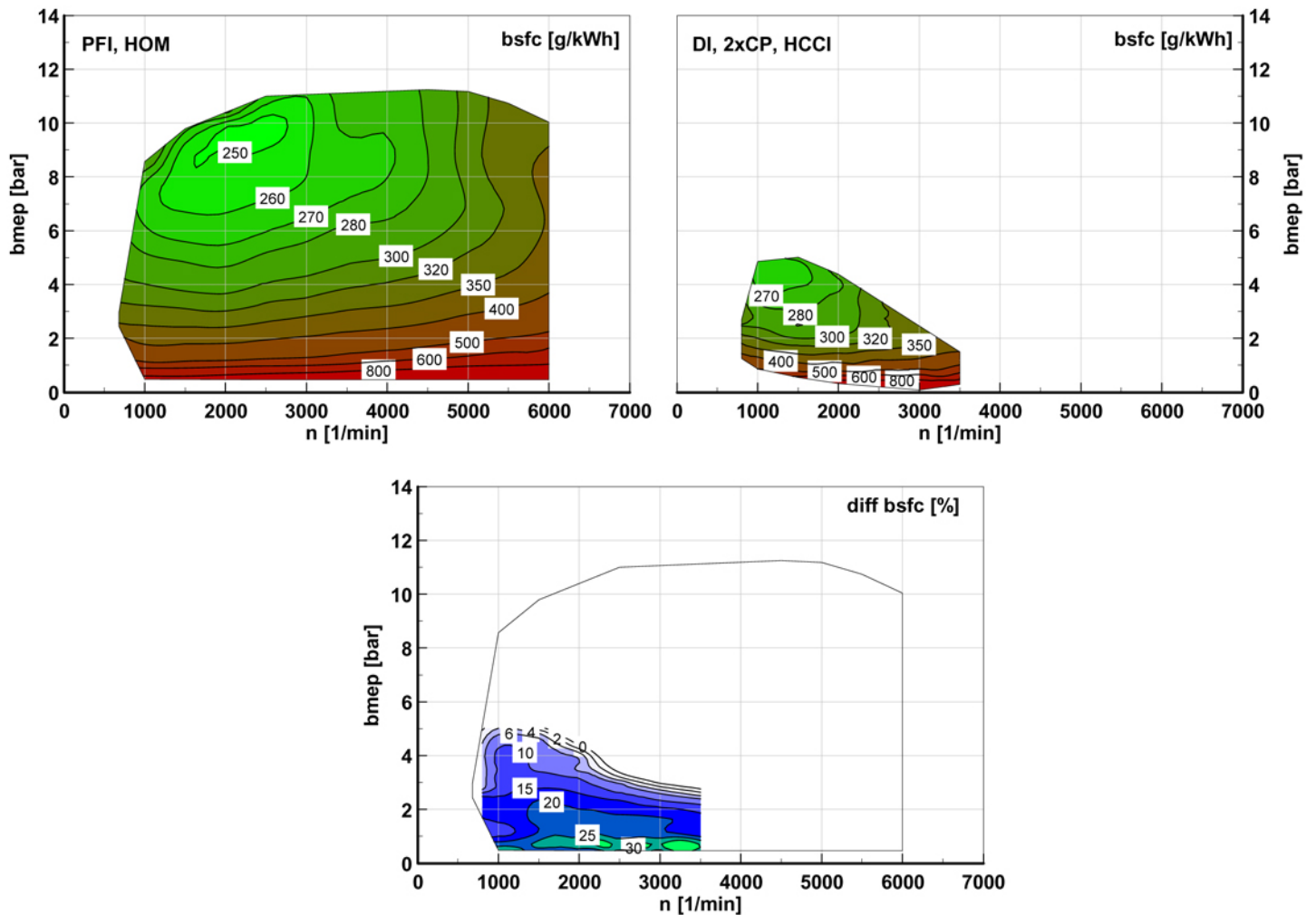


Figure 2. Brake specific fuel consumption map, PFI stoichiometric (upper left graph), DI HCCI using recompression strategy (upper right graph) and the potential achieved with HCCI (lower graph)

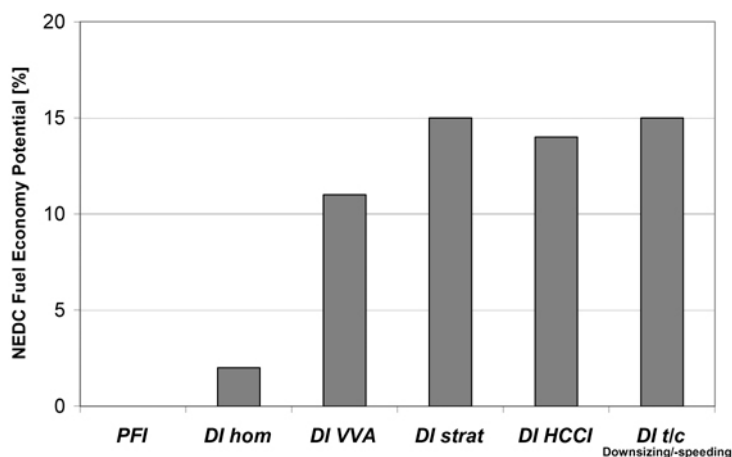


Figure 1. Fuel economy potential of competing combustion concepts for a middle-class vehicle with a 2.0l engine (NEDC, incl. cold start)

Even using a state-of-the-art DI engine with two cam-phasers (4-cylinder engine, data as in Table 1) fuel efficiency potentials up to 25% are still possible, but the efficiency potential map area shrinks (Fig. 3).

In Fig. 1 HCCI is compared with the basis PFI engine, along with stoichiometric direct injection (DI), variable valve actuation (DI VVA), lean stratified charge (DI strat) and direct injection with stoichiometric combustion combined with turbo-charging and downsizing/-speeding (DI t/c). From this result, an interesting question arises: what happens when combining HCCI with turbo/super-charging? Certainly many of the thermodynamic advantages, like dethrottling, are already addressed by both combustion concepts and cannot be added on top. Additionally, downsizing demands for higher loads and, adding to that, lower compression ratios for knocking suppression are needed due to the higher pressure and temperatures in the combustion chamber at boosted load points. These issues lead to reduced fuel efficiency on the HCCI side. Nevertheless, benefits will result from such

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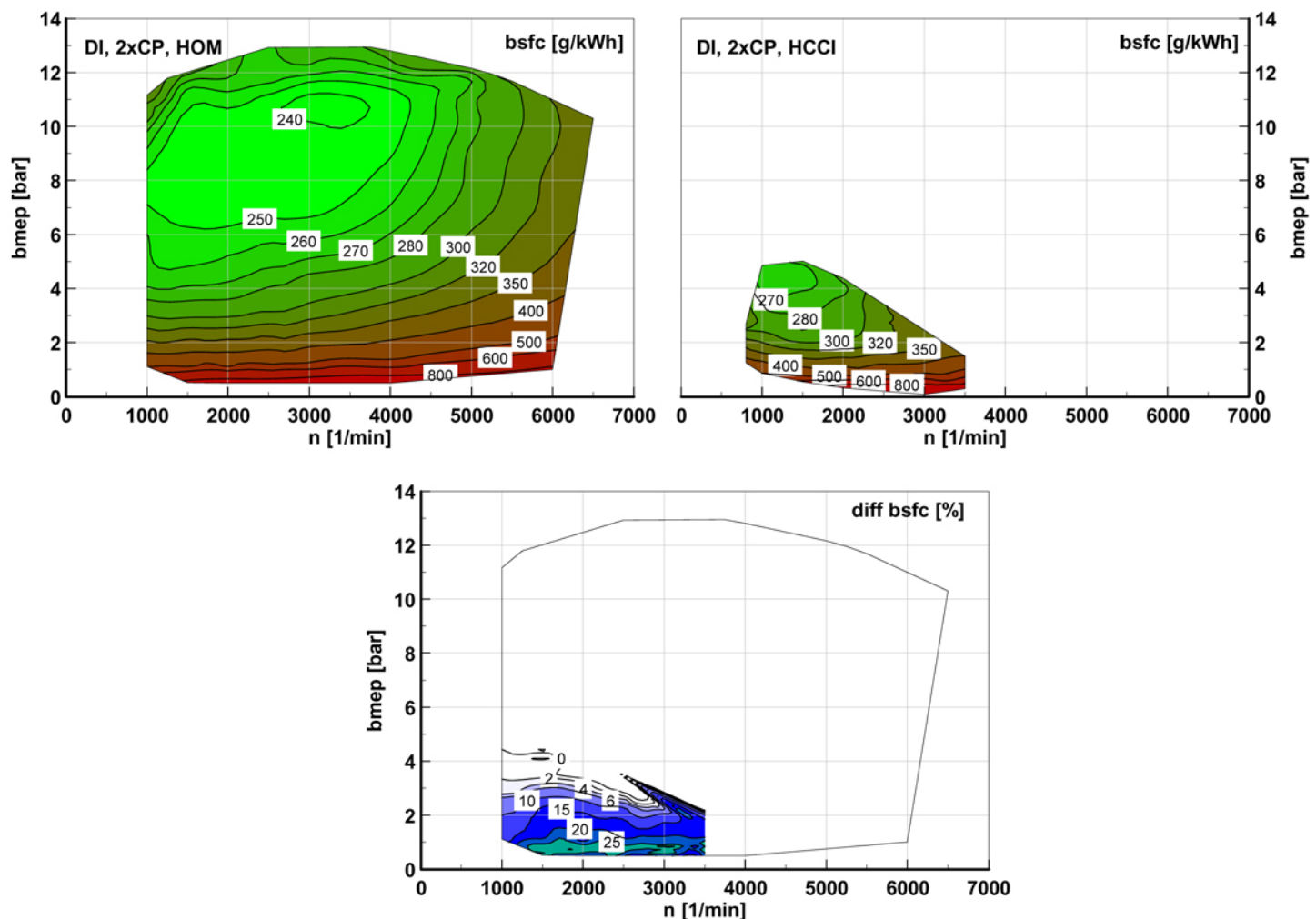


Figure 3. Brake specific fuel consumption map, DI w/ 2 cam-phasers stoichiometric (upper left graph), DI HCCI using recompression strategy (upper right graph) and the potential achieved with HCCI (lower graph)

combination, e.g., the ability of driving higher dilution rates (higher air/fuel-ratio) allows for an additional measure against NO_x at higher HCCI loads as a result of the lower combustion temperature. Higher dilution can also result in an additional fuel efficiency benefit. These advantages allow for higher HCCI loads, which could make HCCI even more interesting in a drive cycle, as the HCCI operation region could be extended and less combustion mode switches would be needed.

<figure 3 here>

All these and other effects will be addressed using analysis based on a single-cylinder research engine experimental set-up and a near-series 4 cylinder engine. Additionally, simulations will be used to extrapolate the possible potential outcome from boosted HCCI.

ENGINE SET-UP

A single-cylinder research engine was used in this work. An electro-hydraulic valve-train system, EHVS [9], was used which allows for variable lift and timings. This is the best way to control the right amount of residuals on a cycle-to-cycle basis. Additionally, Gasoline Direct Injection is employed which is highly flexible for cycle-to-cycle and cylinder-individual control [10]. It also offers indirect control on HCCI ignition and combustion phasing. The experimental data of the single cylinder engine used for the analysis in this article are presented in Table 1 and Figure 4. Also a 4-cylinder research engine was used to provide more realistic understanding of the combustion concept in a near series engine. Engine data is also presented in Table 1. For both engines full pressure indication was applied. For the in-cylinder pressure measurements, a conditioned (cooled) piezo-electrical pressure sensor was utilized. Additionally, manifold pressure indication was employed. In this case, conditioned piezo-resistive pressure sensors were used.

Table 1. Engine Data

	Single-Cylinder Research Engine	4-Cylinder Research Engine
Stroke [mm]	85.0	94.6
Bore [mm]	82.0	86.0
Compression Ratio [-]	11.5	12.1
Displacement [cm ³]	449.0	549.5
Connecting Rod [mm]	145.0	145.5
Fuel	EuroSuper 95 RON	EuroSuper 95 RON
Injection System	centrally mounted direct injection (DI)	centrally mounted direct injection (DI)
Injector	Bosch HDEV MHI 1.2 high pressure multi-hole	Bosch HDEV MHI 5.1, high pressure multi-hole
Valve-Train	Bosch EHVS 0.7 electro-hydraulic valve-train fully variable lift and phasing	cam phaser in/out 2-step valve lift in/out 100 °ca cam authority 3.5 mm lift cam for HCCI HCCI run with NVO strat.

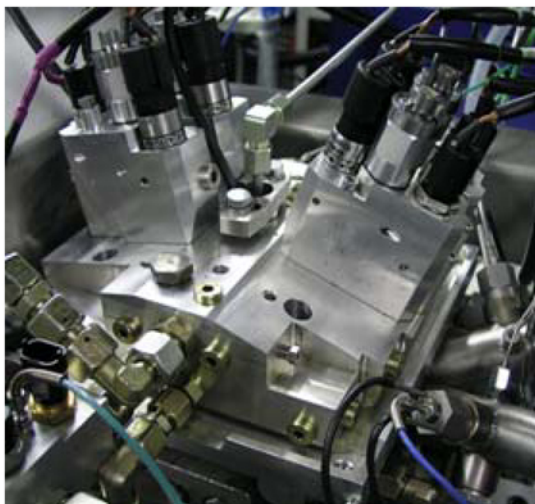


Figure 4. Single cylinder GDI research engine with centrally-mounted injector and full variable valve-train

ANALYSIS: THREE WAYS OF RUNNING HCCI - LEAN, STOICHIOMETRIC, BOOSTED LEAN

From former analysis using the single-cylinder engine [11], three possible strategies for running HCCI, such as 1) lean; 2) stoichiometric using external EGR and 3) lean with supercharging or turbo charging, can be further analyzed. In addition to the fact that the main knobs for HCCI control are SOI and EVC, EGR and supercharging or turbo charging can further extend the NO_x free load range. An operating point at 2000 rpm and a load of about 4.6 bar NMEP in HCCI mode was used to analyze the effects of external EGR on the controlled auto-ignition. [Figure 5a](#) shows the operation settings used for the comparison of lean, stoichiometric and boosted lean HCCI operation. At a constant fuel injection mass of 12.1 mg, various levels of external EGR was undertaken. Start of injection timing (SOI) was mainly used to adjust the combustion phasing fairly constant at about 9 °ca after TC. Additionally, when SOI stopped having any effect on combustion phasing, exhaust valve closing time (EVC) had to be adapted for higher internal EGR rates.

The trend of having to compensate the increase of external EGR with earlier SOI and earlier EVC can be explained through chemical kinetics. Adding external EGR reduces the air mass, i.e., the inert gas in the combustion chamber increases and the oxygen decreases, resulting in a lower specific heat ratio of the charge for the compression phase. The higher inert gas concentration, lower amount of oxygen and the lower specific heat ratio lead to slower reaction kinetics in the combustion chamber despite of slightly higher mixture temperature at IVC due to exchange of air with external EGR. It is observed that the introduction of external EGR during HCCI operations can result in about 6% of fuel consumption reduction in terms of ISFC. NMEP shows a similar decrease ([Figure 5b](#)) due to the constant injected fuel mass. HC-emissions stays nearly constant during the external EGR variation. However, there is a significant increase in CO-emissions when running $\lambda = 1$ due to the slight inhomogeneities in mixture quality. Probably due to the lack of excess oxygen for NO_x-formation, the NO_x-emissions show a reduction at $\lambda = 1$. Although no significant changes can be found in the maximum pressure gradient, the combustion stability, i.e., the standard deviation of NMEP (sigma NMEP), shows a significant improvement, from 0.10 to 0.06 bar, when adding external EGR, driving at $\lambda = 1$.

To analyze the supercharging effect on HCCI, an operating point at 2000 rpm and a load of about 5.1 bar NMEP in HCCI mode without external EGR was selected. At a constant fuel injection mass of 13.4 mg, the supercharged air was gradually added. EVC was changed to keep enough internal EGR for the auto-ignition in the combustion chamber. Additionally, SOI was corrected to keep constant combustion phasing.

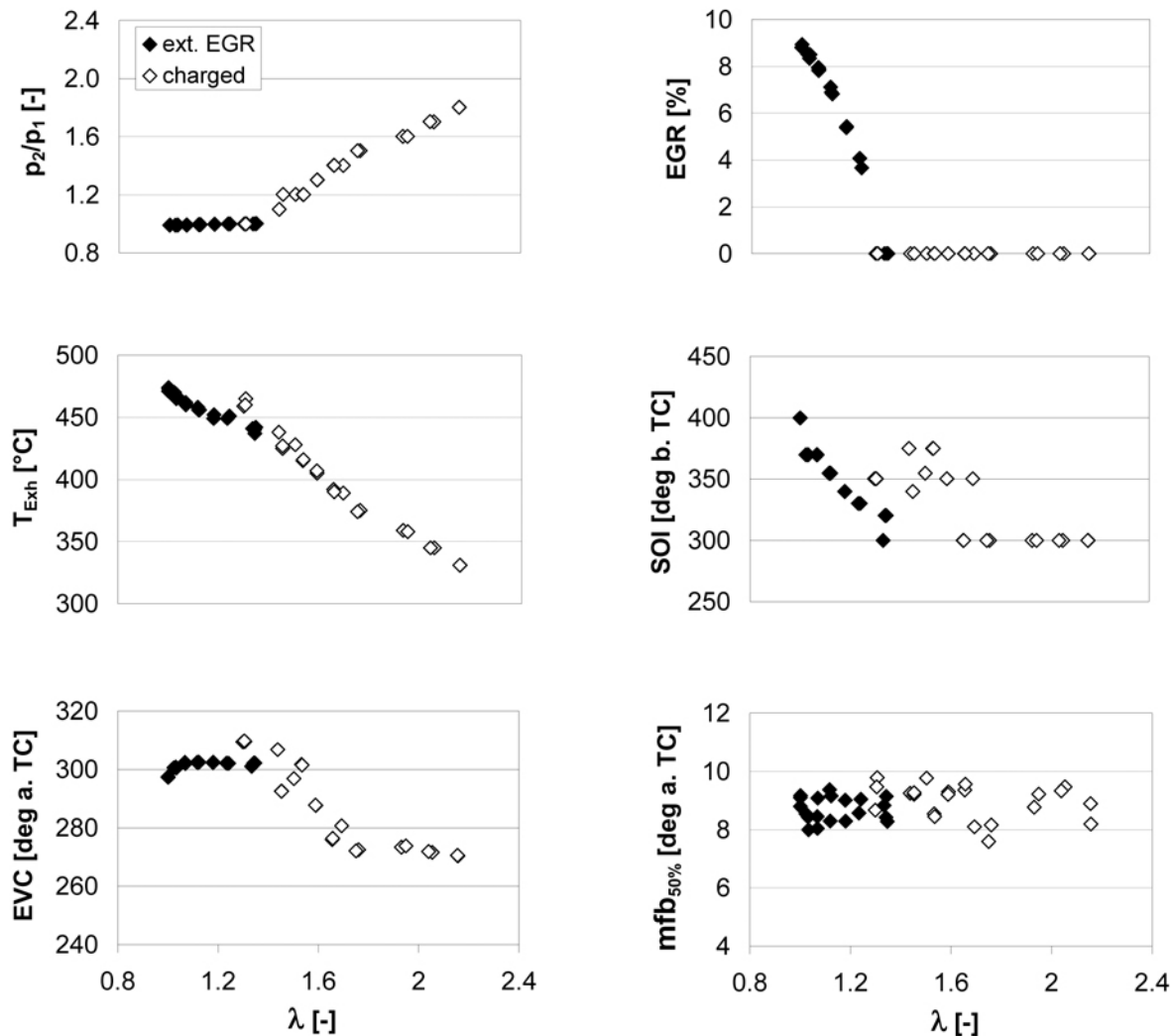


Figure 5a. Operation settings using EGR- or Supercharging-Strategy for HCCI at 2000 rpm in the load range of 4.3 to 5.6 bar NMEP over λ

Driving HCCI with boosting causes an increase of fuel efficiency of about 9% as shown in Figure 5b, where ISFC represents the indicated specific fuel consumption. Each point in the plot represents the measurements, averaged over 200 cycles, collected at a different operation setting. Note that in this measurement campaign, the energy needed for charging is not taken into account. The higher charging leads to a remarkable increase of CO-emissions as a result of the lower combustion temperatures. The effects of lower combustion temperatures can also be observed in the NOx-emissions, which are reduced to a near zero level, even at the high HCCI load operation. Up to a certain charging level, the increasing CO-emissions limit the fuel efficiency improvement. The maximum pressure gradient also shows significant reduction at a higher charging level. By combining supercharging with the HCCI combustion concept, the pressure gradient decreases from 4.5 bar/°ca to 2 bar/°ca. The combustion stability shows a trend towards slightly deteriorated values in σ_{imep} , the standard deviation of imep, from 0.09 to 0.17 bar

at an extremely supercharged level. This correlates with the CO-emissions and lower combustion temperatures.

<figure 5a here>

<figure 5b here>

Despite the non-negligible change at $\lambda = 1.3$ due to the load jump and the associated change in injected fuel amount, the effects of external EGR and supercharging against λ are compared in Figure 5a and 5b. As already mentioned, the increase in external EGR reduces efficiency by means of reduction in λ . On the other hand increasing air charge increases λ and efficiency. In both cases, the efficiency increase correlates with λ within a certain range. The advantages of supercharging in comparison to the external EGR are clear: higher λ and therefore higher efficiency; higher air charge in the process, reducing further the combustion temperature, which can be interpreted from the