The GCI Combustion Process from Volkswagen

The first road traffic driveable vehicle with gasoline engine HCCI combustion (GCI) was recently presented to the public by Volkswagen AG. The GCI combustion process is activated in a sub area of the stoichiometric operation map area of the engine. Perceptible advantages in fuel consumption are noticeable. Low engine-out emissions allow the use of a standard 3-way catalytic converter system. This contribution discusses the measured engine raw emissions from the GCI combustion process.

1 Introduction

The spark-ignition engine operated with a homogeneous stoichiometric mixture is and remains worldwide the dominating drive system for motorized passenger transportation. Its advantages lie in the outstanding smooth running properties, the high specific displacement performance, the good compatibility with different qualities of fuel combined with the opportunity of complying with the strictest of exhaust emissions thresholds as well with trusted 3-way catalytic converter technology. In the part load range, quantity regulation has certain disadvantages in fuel consumption, however, due to throttle losses.

With the so-called GCI combustion process, which is also generally known as HCCI combustion process (Homogeneous Charge Compression Ignition), it has been possible to develop a new part load combustion process for the spark-ignition engine. The GCI combustion process uses homogeneous compression ignition, in order that fuel consumption and also the engine-out emissions can be reduced at the same time compared with conventional throttled operation [1-4].

During work on the GCI project, comprehensive 1-D and 3-D calculations were carried out to enhance understanding of the complex correlations of the new combustion process. Relevant consumption and engine-out emissions potentials from the combustion process were proven on the engine test bench [5, 6]. As a prototype, a complete engine with series production similar technologies was specifically used from the outset. In this way, the balanced GCI combustion process could be transferred directly from the test bench to a vehicle. In conjunction with the newly developed control algorithms, command over the GCI combustion process in real vehicular operation was proven.

2 Prototype Design

As a prototype, the 1.6 l 85 kW FSI engine of Volkswagen AG was chosen, which is powered as standard by homogeneous stoichiometric air-fuel mixture, **Table 1**.

Moderate modifications had to be carried out on the standard engine chosen, in order to make the air-fuel mixture auto ignite during the compression stroke. The valve drive of the 4-cylinder prototype was enhanced by the Group's in-house Audi valvelift system AVS [7]. The AVS technology allows a freely selectable, operating point dependent shift between two different cam contours on the inlet and on the exhaust side. In this way, the cam contours designed to maintain standard FSI operation can also be configured in the same way as the special cam contours for GCI operation.

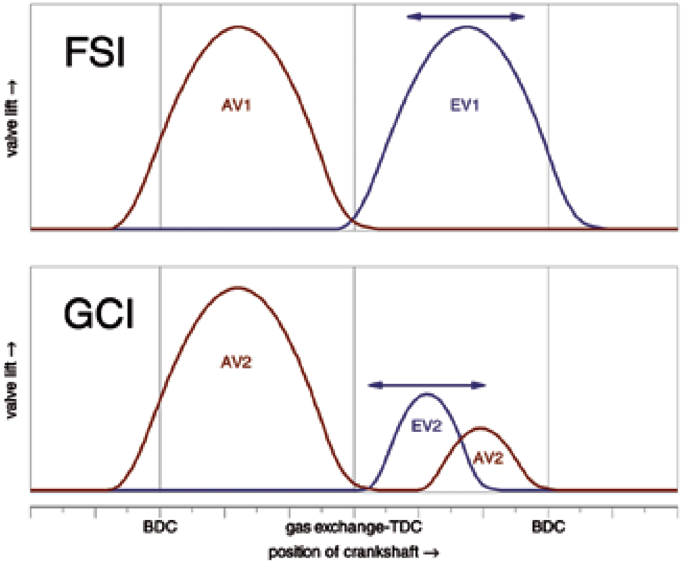

Figure 1: Valve lift curves in operating modes FSI and GCI

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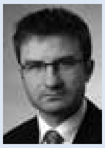

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Table 1: Technical data of 1.6 l FSI engine

number of cylinders	4
number of valves per cylinder	4
displacement	1598 cm^3
bore	76.5 mm
stroke	86.9 mm
compression ratio	12 : 1
injection	direct injection
fuel	Super (RON 95)
maximum power output	85 kW at 6000 rpm
maximum torque	155 Nm at 4000 rpm

In GCI mode, the exhaust valve – unlike for FSI operation (single cam contour AV1) – is opened and closed by a double cam contour (AV2) after opening and closing in the exhaust stroke a second time during the subsequent inlet stroke, **Figure 1**. Using this strategy of gas exchange, a large amount of hot exhaust gases are drawn back into the combustion chamber in GCI mode. As a consequence, the temperature level of the cylinder charge is raised so that controlled auto-ignition of the mixture occurs at the desired point during the compression stroke. The GCI inlet valve stroke (EV2) is reduced due to a lower air requirement in GCI mode (only part load, no maximum load possible) compared with the FSI inlet valve stroke (EV1).

Beyond the camshaft adaptations described, no further changes were instigated on the prototype. Special attention is drawn to the fact that no additional sensors were attached to the engine. The control algorithms newly developed for the engine management system do not rely on the signal from a cylinder pressure sensor. The compression ratio remains unchanged at 12:1. Normal gasoline fuel with an octane rating of RON 95 is used.

Table 2: Comparison of emission values and fuel consumption at operating point examined (speed 2500 rpm, brake mean effective pressure 2.0 bar)

	FSI in series configuration	GCI
brake specific NO_x emissions	6 g/kWh	0.1 g/kWh
brake specific CO emissions	24 g/kWh	5 g/kWh
brake specific HC emissions	7 g/kWh	7 g/kWh
FSN (soot)	0	0
brake specific fuel consumption	371 g/kWh	315 g/kWh

Without any restrictions in the performance compared with the unaltered engine, the spark-ignited FSI mode in the GCI prototype shown is combined with the part load range of the compression ignited GCI mode.

3 Results

Seen in ideal terms, combustion with an HCCI process starts at the same time in the complete combustion chamber (spacial ignition) and differs fundamentally in this way from conventional spark-ignited combustion. With spark-ignited combustion, the front of the flame passes through the combustion chamber spherically from the point of ignition. In real HCCI engine mode, there is a mixture with predominant features of spacial ignition and low percentages of flame spread. The reaction process is essentially dependent on the local thermal and chemical conditions and not heat transportation. In this way there are differences in the burning process and the formation of pollutants between HCCI and spark-ignited combustion.

The following focuses on the influences of different parameters on the exhaust gas components nitrogen oxides (NO_x), carbon monoxides (CO), hydrocarbons (HC) and soot in the GCI operating mode. The observation is made based on the example of the operating point: speed 2500 rpm, brake mean effective pressure 2.0 bar. Firstly, a direct comparison is presented of the emission values and the fuel consumption measured in the test between operating modes FSI (series configuration) and GCI for the nominated operating point, **Table 2**.

3.1 Nitrogen Oxide Emissions

With the aid of gasoline HCCI combustion, the nitrogen oxide engine-out emissions can be significantly reduced over spark-ignited gasoline engine combustion [8]. The very low nitrogen oxide engine-out emissions are due to the extremely high dilution of the charge from exhaust gas reintroduced into the combustion chamber and also, in the event of relevant lean combustion, by the excess air. The combustion temperature is considerably reduced by major dilution of the charge.

As energy is converted in the combustion chamber, approximately 90% to 95% of the nitrogen oxides are formed via the thermal NO mechanism [9]. The reaction-

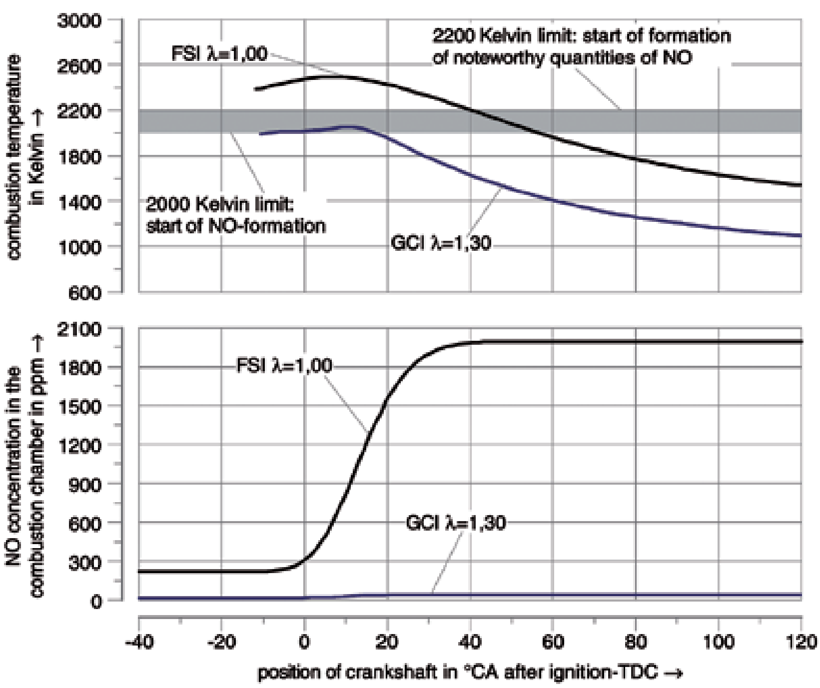

Figure 2: Combustion temperature and NO concentration in the combustion chamber, calculated from measured pressure curves, operating point: speed 2500 rpm, brake mean effective pressure 2.0 bar

kinetic controlled NO formation after the thermal NO mechanism, also referred to as extended Zeldovich mechanism, rises overproportionately as the temperature rises. Noteworthy quantities of NO are not formed until a temperature of more than approximately 2,200 K [10] is reached. At temperatures below approximately 2,000 K, on the other hand, practically no more thermal nitrogen monoxide is formed [11]. In addition to the level of combustion temperature, the temperature profile period is also decisive for the NO formation.

The following engine-out nitrogen emissions of the GCI process are explained by way of the selected operating point example (speed 2500 rpm, brake mean effective pressure 2.0 bar).

As **Figure 2** shows, the combustion temperature in GCI operating mode is considerably lower than in FSI mode and reaches the 2,000 K limit just for a short period. With spark-ignited FSI combustion on the other hand, the 2,200 K limit is clearly surpassed for a longer period. The NO formation in the combustion chamber, also shown in Figure 2, therefore spreads in the GCI operating mode much slower and only lasts about half as long as that in FSI mode, which leads overall to significantly lower NO_x engine-out emissions.

Within the GCI operating mode, the same mechanisms apply for influence of the NO formation as in FSI mode. All measures that cause the combustion temperature to fall also lead to a lower NO formation and thereby to reduction in the NO_x engine-out emissions. Thus, both a retarded shift in the point of 50% mass fraction burned and a dilution of the charge through exhaust gas or from a relevant high quantity of air leads to a reduction in the combustion temperature.

In GCI mode, the charge mass in the combustion chamber is about twice as much compared with spark-ignited operation with the same injected mass of fuel, **Figure 3**. As the results of a 1-D calculation show, the significantly higher quantity of charge mass falls upon the increased residual quantity of gas in GCI mode.

In contrast to the spark-ignited FSI mode, the relative air-fuel ratio and the point of 50% mass fraction burned in GCI mode cannot be varied independently of each other. The point of 50% mass fraction burned depends directly on the set relative air-fuel ratio. As a rule, the point of 50% mass fraction burned has a more retarded setting depending on how lean the mixture is, Figure 4. An influence of the point of 50% mass fraction burned by the point of ignition – as on the classic gasoline engine – or from start of injection – as on the classic diesel engine – is only possible to a certain degree. **Figure 4** shows further that it is possible to reduce the NO_x emissions by an increase in the relative air-fuel ratio. As familiar from the spark-ignited gasoline engine, the pronounced maximum of NO_x emissions in the area between relative air-fuel ratio figures of 1.0 and 1.1 should not be observed.

3.2 Carbon Monoxide Emissions

Carbon monoxide forms as the result of incomplete oxidation due to oxygen deficiency. Even with a global stoichiometric relative air-fuel ratio in the combustion chamber, insufficient mixture preparation can give rise to local understoichiometric zones in the combustion chamber. As the relative air-fuel ratio rises, the occurrence of zones with air deficiency decreases and thereby also the CO emissions become less. Part of the carbon monoxide produced is oxidized to CO_2 in the working stroke at temperatures above 1,700 K.

The inlet ports of the engine in question normally generate a tumble effect. In GCI mode, the small inlet valve strokes combined with the short opening period do not allow formation of a pronounced tumble flow, however. The affected CFD calculations then show during the GCI intake stroke highly turbulent air flow conditions in the combustion chamber. The temperature of the cylinder charge is raised considerably in this instance by reintroduction of the hot exhaust gas.

Due to the high turbulence and high gas temperatures, there are good conditions for an outstanding mixture preparation and thereby also for a high level of homogeneity in the cylinder charge. This is confirmed by CFD calculations and optical examinations on the engine. In this way, start of injection and injection pressure can be varied without significant effects on the quality of mixture in a decision way.

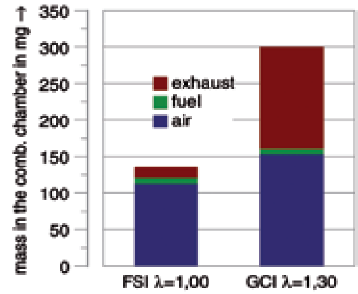

Figure 3: Comparison of charge composition in GCI and in FSI mode calculated from measured pressure curves, operating point: speed 2500 rpm, brake mean effective pressure 2.0 bar

The CO emissions are principally lower in overstoichiometric GCI mode than in stoichiometric FSI mode. Furthermore, the selected strategy of gas exchange guarantees a very good mixture preparation and thereby once again a notable reduction in CO emissions. **Figure 5** shows the measured CO emissions depending on the relative air-fuel ratio for the selected operating point (speed 2500 rpm, brake mean effective pressure 2.0 bar). With an advanced position of the point of 50% mass fraction burned (5 to 10°CA after ignition TDC), the dependence familiar from spark-ignited mode can be observed: starting from the stoichiometric mixture, the CO emissions fall as the relative air-fuel ratio rises. With an increase in the retarded position of combustion, however, there is again a rise in emissions. This is attributable to the falling combustion and combustion chamber temperatures with continually retarded position of combustion. In the working stroke, the prevailing temperatures in the combustion chamber then drop more and more below the temperatures, necessary for the CO oxidization reactions, of at least 1700 K.

3.3 Hydrocarbon Emissions

Hydrocarbons in the engine exhaust gas can always be attributed to zones that are not covered, or only in part, by combustion. Various mechanisms are known to be the causes of this on spark-ignited gasoline engines.

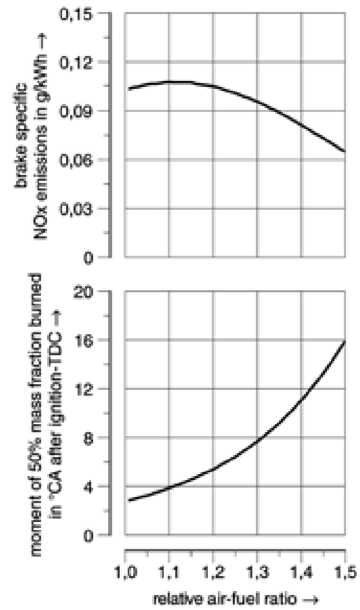

Figure 4: Point of 50% mass fraction burned and NO_x engine-out emissions in dependence on the relative air-fuel ratio in GCI mode, calculated from DOE data, operating point: speed 2500 rpm, brake mean effective pressure 2.0 bar

In proximity of the wall of a combustion chamber, the gas temperature falls considerably compared with the centre of the combustion chamber. In the gasoline engine with spark-ignition, this means a reduction in burning speed in the zones close to the wall, the flame extinguishes before it reaches the wall. This so-called wall quenching can also be found in a similar way to HCCI combustion because there are slow reaction zones in the vicinity of the combustion chamber walls due to the lower temperature level. These zones ultimately do not participate in combustion with the HCCI process either.

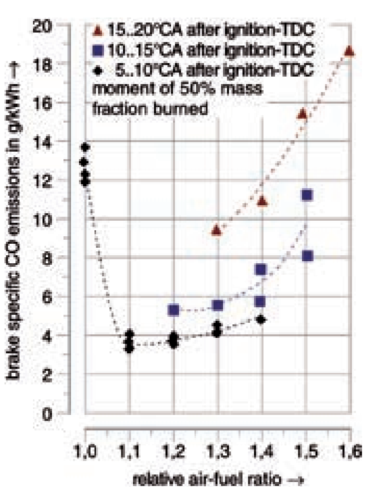

Figure 5: CO engine-out emissions in GCI mode depending on relative air-fuel ratio and point of 50% mass fraction burned, operating point: speed 2500 rpm, brake mean effective pressure 2.0 bar, phase angle of intake camshaft varies

With flame quenching, the flame extinguishes in too rich or too lean mixture areas. The flame quenching should only play a subordinate role in HCCI combustion as the flame paths are extremely short due to the diverse ignition locations.

On the other hand, the more or less large intermediately stored fuel components released after combustion from the gap areas, from the piston crown and the combustion chamber roof are deemed to be possible HC sources. Even fuel components released through desorption from the lubrication film contribute towards HC emissions. These causes for HC emissions are also principally evident with the HCCI combustion process.

The measured emissions of hydrocarbons are shown in **Figure 6** for GCI mode in the familiar operating point (speed 2500 rpm, brake mean effective pressure 2.0 bar) over the relative air-fuel ratio. The classic dependence of HC emissions, familiar from spark-ignited combustion, on the relative air-fuel ratio with minimum at lightly overstoichiometric mixture is also evident in GCI combustion. In addition, as already seen in observation of the CO emissions (Figure 5), the influence of the point of 50% mass fraction burned can be seen. With increasingly retarded combustion, the HC emissions rise dramatically.

This makes it possible in GCI mode to influence the HC and CO emissions independently of the adjustment parameters by the point of 50% mass fraction burned. Compared with spark-ignited mode, lower or higher HC and CO emissions can be “set”. This explains the often contradictory information in publications regarding the extent of HC and CO emissions in HCCI mode. If the point of 50% mass fraction burned is not taken into account during evaluation of the trial results, both lower and also much higher emissions can be attained in comparison with spark-ignited mode.

Further examinations carried out underpin the theory that the position of energy turnover with GCI combustion represents the dominant influencing factor for the observed HC engine-out emissions. Variations were carried out in the engine setting parameters, such as ignition angle, start of injection, injection pressure, relative air-fuel ratio, coolant temperature, amount of external EGR, intake air pressure and temperature, phase angle of the intake camshaft, engine speed and load. The results of the examinations make it evident that with constant injection quantity, the emission of hydrocarbons also remains practically constant as long as the position of energy conversion can remain constant – and that independent of the selected setting parameters, **Figure 7**. This association is also applicable for various engine speeds during the examinations carried out from 1500 to 3750 rpm.

As the injection quantity increases, the HC emissions level falls and dependence on the position of the point of 50% mass fraction burned becomes less. With small injection quantities, the projection of HC emissions in Figure 7 is as expected greater due to the heavily fluctuating power output from the variations. From

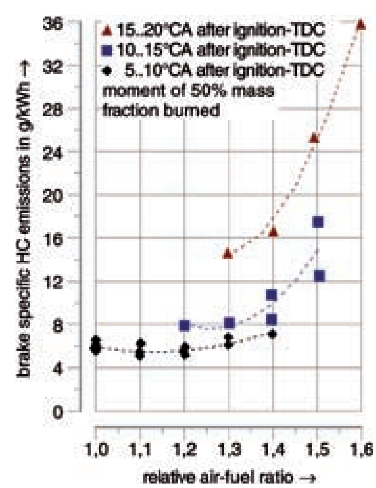

Figure 6: HC engine out emissions in GCI mode depending on relative air-fuel ratio and point of 50% mass fraction burned, operating point: speed 2500 rpm, brake mean effective pressure 2.0 bar, phase angle of intake camshaft varies

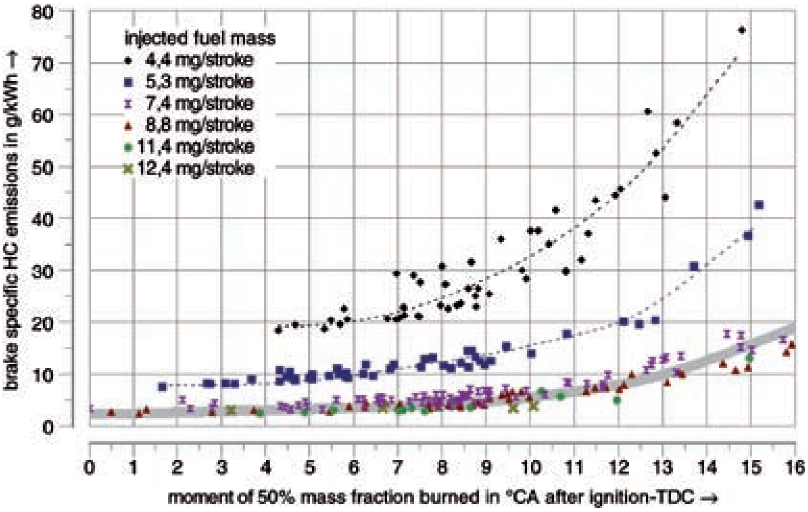

Figure 7: HC engine-out emissions in GCI mode depending on point of 50% mass fraction burned and injection quantity with variation of different engine setting parameters

an injection quantity of 7.4 mg/stroke, however, almost all results are on one line on the prototype selected. The influence of the point of 50% mass fraction burned on HC emissions is so dominant that, for example, neither mixture formation nor the engine speed or load can influence the HC emissions as long as the point of 50% mass fraction burned can be kept constant.

A complete analysis of the causes for hydrocarbon emissions for the GCI combustion process can be found in a dissertation written within the scope of the GCI project [12].

3.4 Soot Emissions

Soot particulates are caused during combustion as a result of extreme air deficiency, caused either by a fundamentally low relative air-fuel ratio or as with CO in global stoichiometric but (local) inhomogeneous mixture by local rich zones. As already described under section 3.2, the engine works in GCI mode with excess air and also with a predominantly homogeneous mixture. As a result, the soot emissions are at the detection limit in all operating conditions.

4 Summary and Outlook

The newly developed GCI combustion process allows, in conjunction with the selected strategy of gas exchange in a part load area of the operation map, low fuel consumption combined with low exhaust emissions. The emissions behaviour of combustion with homogeneous compression ignition differs in this instance, in part quite considerably, from that of spark-ignited combustion.

Extremely low NO_x emissions are achieved in GCI mode due to the principle. The emission of carbon monoxide can be reduced clearly due to an overstoichiometric relative air-fuel ratio and a good mixture preparation in comparision to the spark-ignited mode. The HC emissions can be maintained in GCI mode and in spark-ignited mode at a low level. The dependence of CO and HC emissions on the point of 50% mass fraction burned could be worked out by the examinations. The more retarded combustion is, the higher the emissions are. The emission of soot particulates is at the detection limit.

In addition to the advantages gained, the GCI combustion process is also influenced by new challenges. As an example, the combustion process reacts unusually sensitively even to very small changes in the setting parameters. Control of the dynamic mode is thereby dissimilarly harder to master than with conventional combustion processes.

Enhancement of the previously shown GCI operating range is desirable. With the use of a fully variable valve drive system, variable compression ratio, boosting and, not least in conjunction with externally cooled exhaust gas recirculation, the part load range can be extended moderately even today. The results of the previous examinations show, however, that the best cost-benefit efficiency is not achieved through massive use of costly technologies but much more by the application of proven and practical solutions.

The Group Research of Volkswagen AG transferred the GCI combustion process into the drive train of a series produced vehicle and evaluated the potential of tomorrow‘s technology today in real driving operation.

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