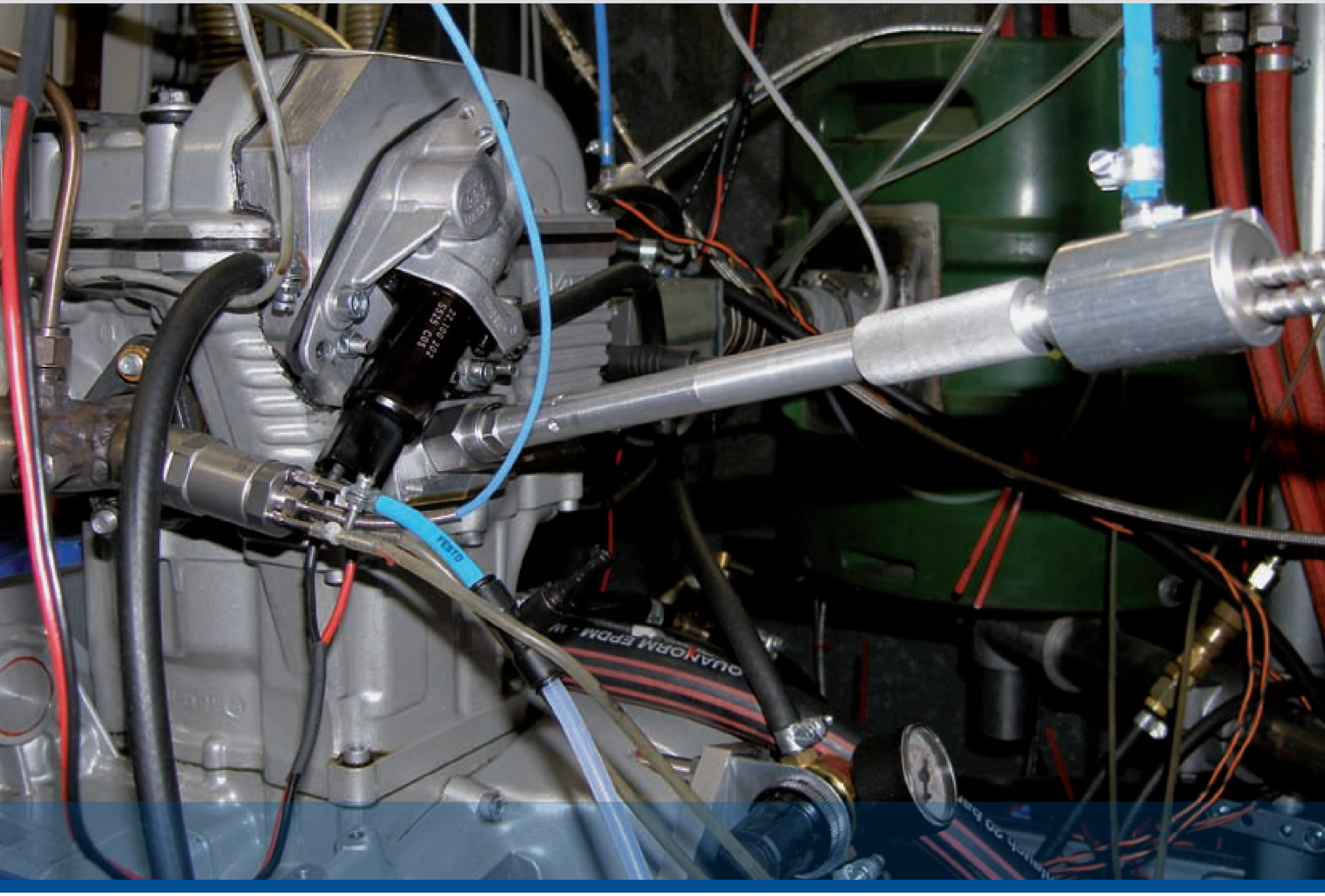

Controlled Auto-Ignition – Reduction of Engine Emissions by Controlled Auto-Ignition in Gasoline Engines

The controlled auto-ignition in gasoline engines is a promising concept to reduce both emissions and fuel consumption simultaneously. Within the FVV-Vorhaben „Benzinselbstzündung“ experimental and numerical investigations have been carried out at the Institut für Kolbenmaschinen (IFKM) and at the Institut für Technische Thermodynamik (ITT) to estimate the realisation of this combustion process in passenger car engines.

1 Introduction

Auto-ignition of gasoline is realised by compressing a homogeneous fuel air mixture until auto-ignition occurs. The required temperatures are obtained by trapping hot residual gas in the combustion chamber. In contrast, combustion in conventional gasoline engines is induced by spark ignition which leads to a spherical flame front propagation and a sequential combustion of the homogeneous fuel air mixture. Since conventional combustion of gasoline air mixtures with low cyclic fluctuations leads to high temperatures in the flame front, high NO_x-emissions cannot be avoided. In conventional diesel engines the diffusion combustion leads to high soot and NO_x-emissions. Due to the HCCI combustion a fuel conversion without flame front propagation or diffusion combustion is feasible, therefore a reduction of the peak temperatures in the combustion chamber results and NO_x formation at high temperatures can be avoided. With HCCI short combustion durations and high efficiencies are accomplished [1], [3].

Thermodynamic and numeric analyses of the engine operation in HCCI mode have been carried out, based on experimental investigations using a single cylinder engine with optical access. The adaption of a three-dimensional visualisation system enabled an insight on the real process inside the engine. An efficient model for treating the chemical reactions was developed and coupled with a commercial CFD-Code, to describe the spatial progress of the reactions.

2 Test Engine and special experimental Measurement Techniques

The single-cylinder engine used for this study, **Table 1**, is a modified BMW/Rotax F650 four-stroke motorcycle engine. A vane type camphaser and direct injection have been adapted to the engine. Auto-ignition is realised by symmetric negative valve overlap [1]. The investigations have been carried out with a synthetic fuel consisting of different components so that it matches the characteristics of 4-star petrol.

The three-dimensional visualisation system [2] consists of three fibreoptical endoscopes. In **Figure 1** the endoscopes adapted to the test engine and the images of the combustion chamber obtained by the endoscopes are shown. The used camera system is characterised by a high sensitivity in UV-wavelength. The high temporal resolution of the measurement system allows the recording of ten pictures each crank angle degree at a speed of 2000 rpm [5].

Table 1: Technical specifications of the test engine

Basic engine	BMW / Rotax F650
Engine Type	Single-cylinder, four-valve pentroof
Compression ratio	11.5
Displacement	652 cm^3
Bore x Stroke	100 mm x 84 mm
Engine Speed	2000 rpm
Mixture formation	Swirl injector (sideways located) Multi-hole-injector (centrally located)
Injection pressure	125 bar
Maximum lift, intake valves	2 mm
Maximum lift, exhaust valves	1.5 mm
Opening duration, intake valves	120 CAD
Opening duration, exhaust valves	110 CAD
Intake valve closing	182 CAD BTDC – 140 CAD BTDC
Exhaust valve closing	272 CAD ATDC – 324 CAD ATDC

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3 Chemical Kinetic Model to describe the Auto-Ignition

The time of auto-ignition depends neither only on the average temperature nor only on the global mixture composition. It depends on the local thermodynamic conditions. Hence zero- or one-dimensional tools to describe the process and the chemical reactions inside the combustion chamber especially in a HCCI engine are insufficient.

Different formulations exist to incorporate the auto-ignition and combustion in three-dimensional simulations realistically. Simple models are computationally inexpensive, the accuracy is however quite low. Using detailed reaction mechanisms the chemical source terms can be computed accurately depending on the species concentration, pressure and temperature. However this method is computationally prohibitive. In this work an efficient reduced model for treating the chemical reactions in a CFD-Code was developed. The model is based on a detailed reaction mechanism [6].

Prior to calculating the temporal rate of the progress variable, the development of a detailed chemical mechanism, for the investigated fuel, is required. This was accomplished by a automatic mechanism generator [4].

The link between the CFD-Code and the chemical kinetic model is realised by a tabulation method. A progress variable for the chemical progress is defined and the rate of this variable is calculated and tabulated depending on the physical and chemical conditions. The values of the progress variable are between 0 (fresh mixture of fuel and air before start on any reactions) and 1 (chemical equilibrium after combustion). The arc length of the reaction trajectory in the state space was selected as adequate progress variable χ based on detailed kinetics. **Figure 2** shows an example for a reaction trajectory. The development of the ignition and the combustion are shown for a CO_2, CO and CH_2O mass fraction subspace. At the start of the reaction the mass fraction is zero for all three species. Shortly before the beginning of the ignition the CH_2O mass fraction has a maximum. CO is another intermediate species, which is partially transformed to CO_2 while the process continues. The mass fraction of CO_2 increases during the combustion process. The colour of the reaction trajectory represents the temperature during the combustion process.

For the implementation of the chemical model in the CFD-Code StarCD additional governing equations for the progress variable and for the mass fraction of different species (CO_2, CO, CH_2O, fuel) have to be solved. For given values of the pressure, air-fuel ratio, enthalpy of the gas mixture and chemical progress variable the rate of the chemical progress and the chemical mixture are looked up in the tabulation of the detailed chemistry and handed over to the CFD-Code. A detailed description of the combustion model can be found in [6].

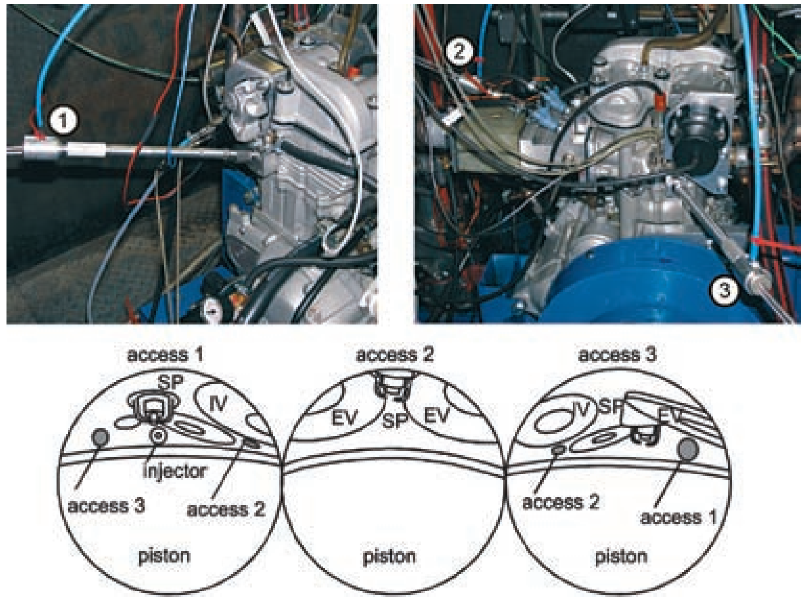

Figure 1: Adaption of the three fibreoptical endoscopes at the test engine and the images of the combustion chamber captured by the endoscopes 1 - 3

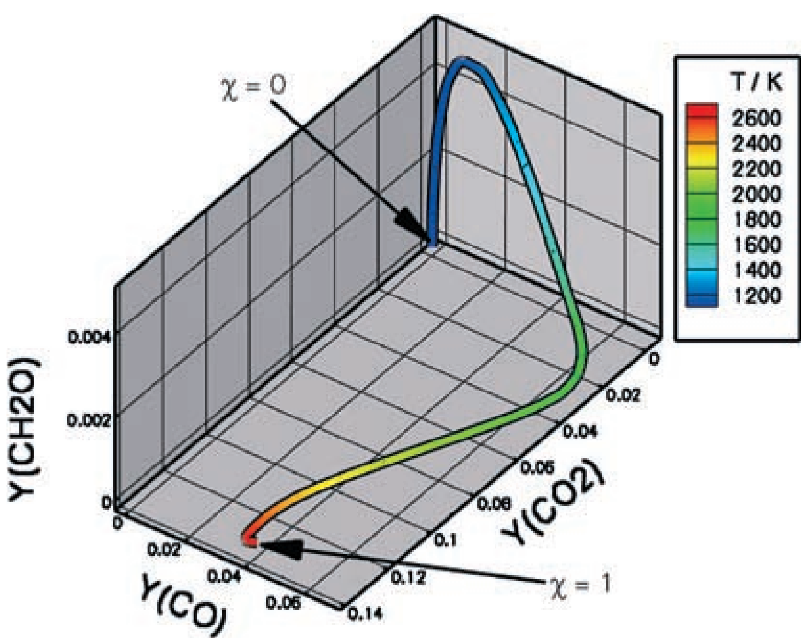

Figure 2: Reaction trajectory in the CO_2, CO and CH_2O mass fraction subspace

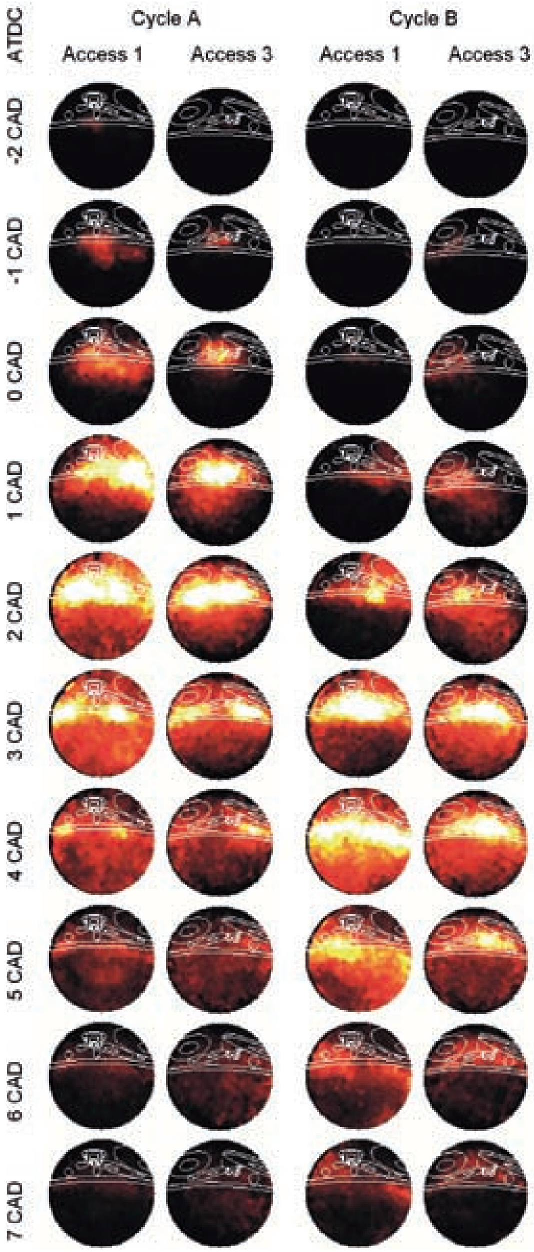

Figure 3: Visualisation of the combustion process of two selected cycles

4 Phenomenology of the Controlled Auto-Ignition

In fundamental investigations phenomenological processes in the HCCI engine were analysed.

4.1 Preferred Ignition Spots and Sequential Auto-Ignition

In **Figure 3** the ignition of two selected cycles is shown. For both cycles an ignition spot in the centre of the combustion chamber can be identified. It can be seen clearly that the ignition of the fuel air mixture does not occur in the whole combustion chamber at the same time. Due to the local mixture and temperature distribution, auto-ignition occurs in some regions early and in other regions with a temporal delay. The so called preferred ignition spots are not located close to the combustion chamber wall, but rather in the centre of the combustion chamber. Starting with these ignition spots, a sequential auto-ignition process can be seen, which ceases in the outer regions of the combustion chamber beneath the valves. From cycle to cycle high fluctuations of shape and location of the ignition spots were determined.

4.2 Effect of Spark Ignition

Experimental investigations illustrated that spark assistance has no effect on the combustion in operating points with high EGR rates. While it is possible to extend the operating range using a spark assistance in operating points with low EGR rates [6]. To analyse the influence of the spark assistance on the combustion, two operating points have been compared: operating point 1 with high amount of residual gas trapped in the combustion chamber, operating point 2 with a low amount of residual gas, **Table 2**. Pictures of the combustion of selected cycles of the operating points 1 and 2 are shown in **Figure 4**. On the pictures taken by endoscope 1, the ignition spark is visible in both operating points. Whilst the pictures captured by endoscope 3 the ignition spark is covered by the spark plug electrode on.

In operating point 1, the combustion starts suddenly at 4 CAD BTDC, followed by a fast combustion which is characteristic for HCCI mode. The energy source of the spark plug has no influence on the combustion process.

In operating point 2, by phasing the spark timing earlier, the combustion timing is shifted earlier as well [6]. The effects of the spark ignition on the combustion timing are illustrated by the pictures of the combustion, Figure 4. The

Table 2: Operating points to investigate the influence of spark assistance

Operating point	1	2
Speed	2000 rpm	2000 rpm
Indicated mean effective pressure	2 bar	3 bar
Exhaust gas recirculation	70 %	55 %
Air-fuel ratio	1.0	1.0
Spark timing	42 CAD BTDC	42 CAD BTDC

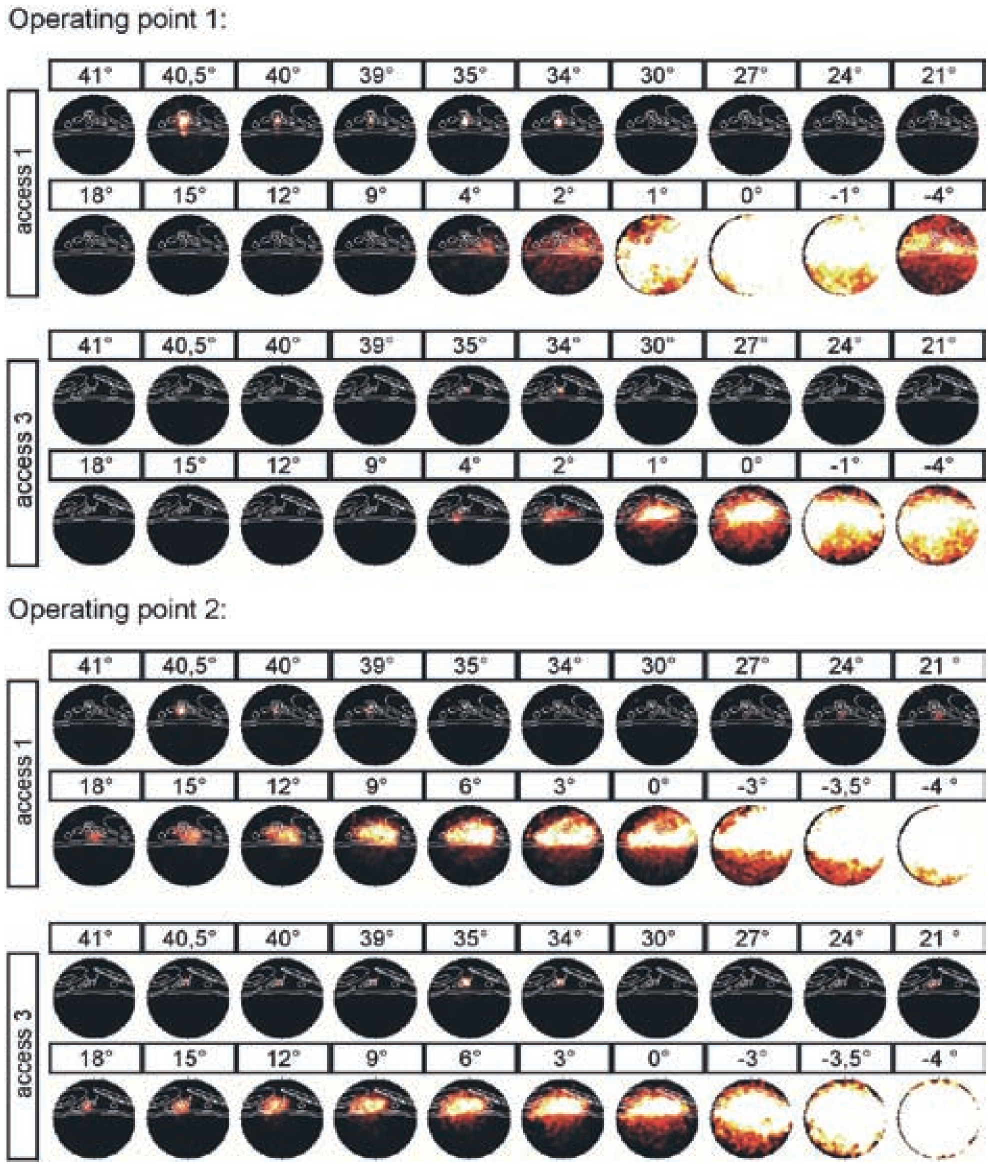

Figure 4: Visualisation of the combustion in operating points 1 and 2

ignition at the spark plug induces a flame front which propagates slowly at the beginning. Afterwards the flame propagation is accelerated and the remaining unburned fuel air mixture reacts suddenly. The high rate of reaction leads to the conclusion that sequential auto-ignition occurs in this period of the combustion.

5 Effect of Injection Timing

It was illustrated from the experimental investigations that the point in which the start of injection occurred, had a major effect on the operating behaviour in HCCI mode, **Figure 5**. By varying the start of injection, the 50 % mass fraction burnt point can be adjusted and a minimisation of the maximum pressure rise (dp/dα) is feasible.

As a result of this investigation, it was worked out that the shifting of the 50 % mass fraction burnt point is mainly related to the temperatures in the

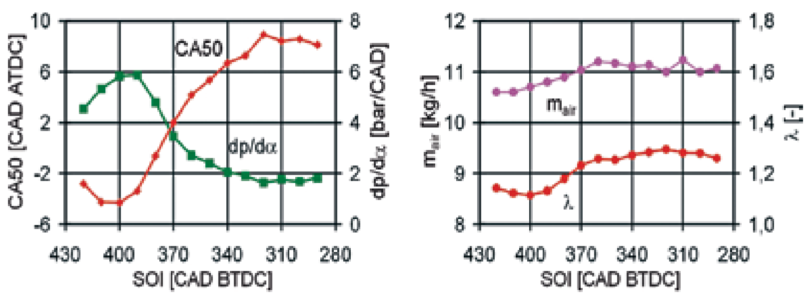

Figure 5: 50% heat released (CA50), maximum pressure rise (dp/dα), mass flow rate of fresh air (mair) and air fuel ratio (λ) as a function of start of injection (SOI), IMEP=2 bar

combustion chamber at the end of compression, which are effected by the start of injection. Retarding the start of injection, leads to an increased amount of fresh air in the cylinder at intake valve closing, Figure 5. Hence the mixture has lower temperatures which leads to later auto-ignition.

The required amount of hot residual gas is trapped in the cylinder by an early exhaust valve closing and a late intake valve opening. This leads to a negative valve overlap and a compression of the residual gas with maximum temperatures of about T=1400 K and maximum pressures of about p=15 bar. If an early start of injection is chosen, the fuel is injected into the hot residual gas which may lead to pre reactions. By detecting the radiation emitted during the negative valve overlap, it was shown that in operating points with lean air-fuel ratios and early fuel injection, a small amount of fuel reacts during the compression after the closing of the exhaust valve, **Figure 6**. In this operating point the injection was divided into a pilot injection at 450 CAD BTDC and a late main injection.

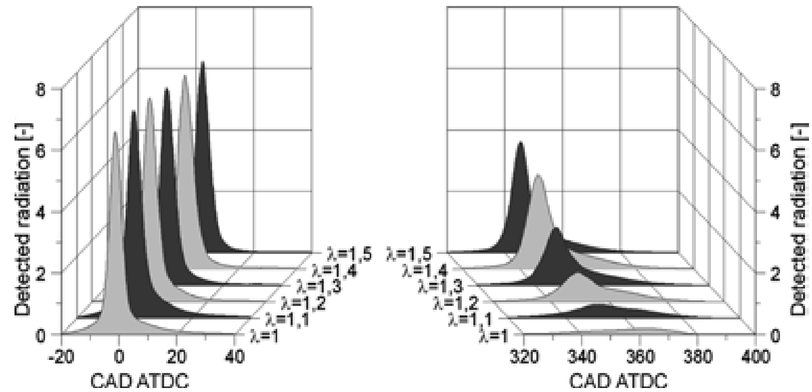

Figure 6: Detected radiation emitted at TDC and emitted during the compression after exhaust valve closing for a variation of the air-fuel ratio (λ)

Although the retarding of the start of injection leads to a worse mixture preparation, at very late start of injection it was observed that the temperatures calculated for the end of compression do not correlate with the 50 % mass fraction burnt point anymore.

To determine the influence of the homogeneity of the mixture at auto-ignition timing on the progress of the self ignition, the local and temporal progress of the auto-ignition was analysed for three different starts of injection (SOI=380 CAD, 320 CAD and 260 CAD BTDC), by three-dimensional visualisation technique and three-dimensional simulation. In all cases the same air fuel ratio (λ=1.15) was adjusted. With early start of injection at 380 CAD BTDC the fuel was injected during the compression of the hot residual gas after the exhaust valve closing. The middle injection was set to 320 CAD BTDC before the intake valve opening and the late injection was set to 280 CAD BTDC during the intake stroke.

Figure 7 shows a comparison of the measured and simulated pressure curves. A good comparison is achieved by using the chemical kinetic model developed in this work.

Figure 8 shows the residual gas, the fuel, the temperature distribution and the local air-fuel ratio before auto-ignition in all three cases. The difference regarding the residual gas distribution are negligible. Regarding the fuel distribution the early and middle start of injection lead to a lot more uniform fuel distribution than the late start of injection. A late start of injection leads to a highly inhomogeneous local air fuel ratio, with very rich and very lean regions. The temperature distribution in the combustion chamber shows, that in regions with low mass fractions of fuel and high mass fractions of residual gas the highest temperatures occur. Late start of injection leads to the most inhomogeneous temperature distribution, while the regions with a high amount of fuel have the lowest temperatures. At early start of injection the regions with high amounts of fuel and high mass fractions of residual gas overlap partially or are located side by side. Thus a homogeneous temperature distribution is achieved.

In **Figure 9** the experimental determined locations of the auto-ignition spots are shown. A variation of the start of injection leads to a dislocation. With an early start of injection the ignition spots are located on the opposite side of the injector. Retarding the start of injection shifts the preferred ignition spots to the centre of the combustion chamber. The results of the three-dimensional simulation confirm this trend, **Figure 10**. In case of an early start of injection the ignition spot is located near the squish region, with middle start of injection two ignition spots are centrally located in the combustion chamber and in case of late start of injection the ignition spot is located beneath the intake valve near the injector.

The comparison of the reaction progress in Figure 10 for early, middle and late start of injection shows that the homogeneous temperature and fuel distribution in case of the early in-

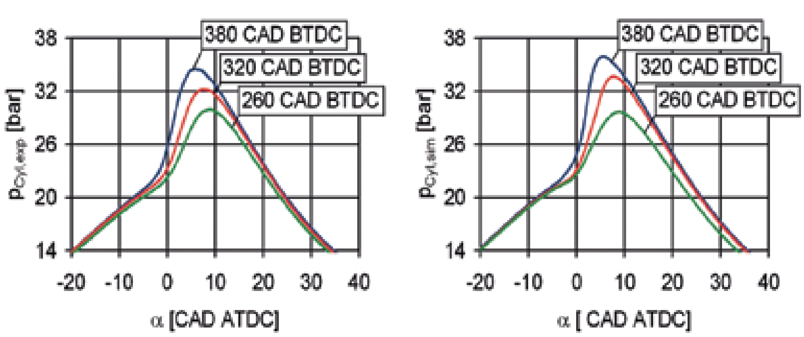

Figure 7: Comparison of measured ($p_{Cyl,exp}$) and simulated ($p_{Cyl,sim}$) pressure curves.

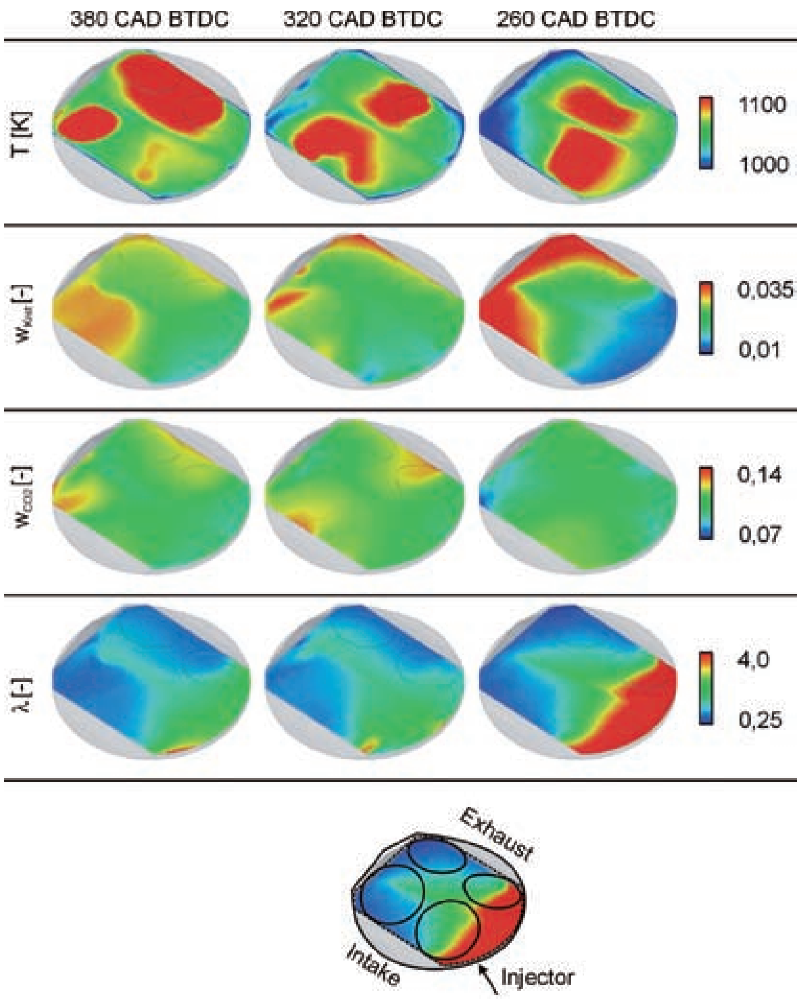

Figure 8: Residual gas (w_{CO2}), fuel (w_{fuel}), temperature distribution (T) and local air fuel ratio (λ) before auto-ignition

jection timing leads to a very fast fuel conversion, whereas the inhomogeneous mixture in case of late start of injection leads to a slow reaction progress. In all cases the ignition spots are located in the regions of highest temperature. With late injection timing the combustion is retarded so drastically that in lean regions a complete fuel conversion is not achieved.

By simulating the variation of the injection timing it was shown that the progress of the combustion can be influenced by a specific variation of the fuel distribution. By stratifying the fuel an increased stability of ignition, a shift of the ignition timing and a control of the combustion duration can be achieved.

6 Summary and Outlook

Within this work the process of auto-ignition was investigated in a combined project of engine experiments and simulation. The development of a chemical kinetic model enabled the analysis of the engine behaviour by three-dimensional simulations. For the first time the preferred ignition spots have been located three-dimensionally by the application of an endoscopic fibreoptic visualisation system. In addition to extensive investigations of the effect of the valve timing and start of injection on the combustion the influence of the spark assistance on the combustion has been analysed.

As well as the injection timing the spark assistance is a feasible control parameter for the adjustment of the 50 % mass fraction burnt point. The influence of the ignition spark on the combustion timing and on the possible operating map increases with decreased residual gas rates. With increasing load, a superposition of flame front propagation at the spark plug followed by sequential auto-ignition was determined. The direct injection of gasoline is an important control parameter for the optimisation of operating points, since the combustion timing is influenced by the start of injection.

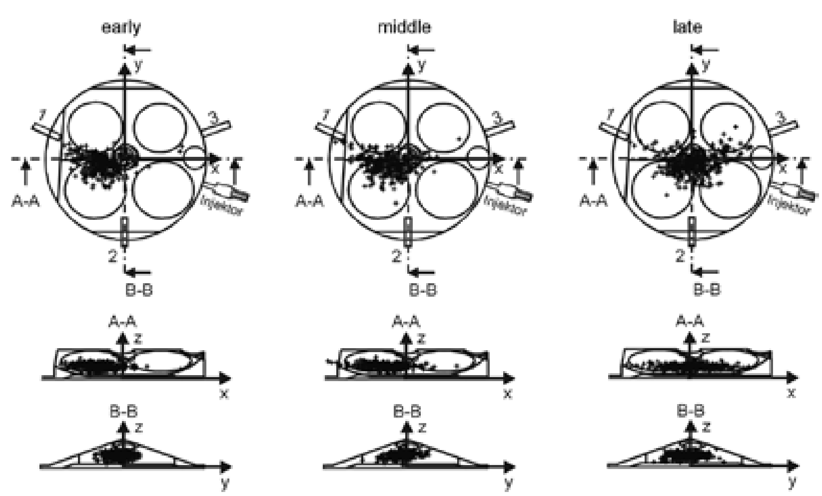

Figure 9: Experimental determined locations of the auto-ignition spots with early, middle and late start of injection

7 Acknowledgement

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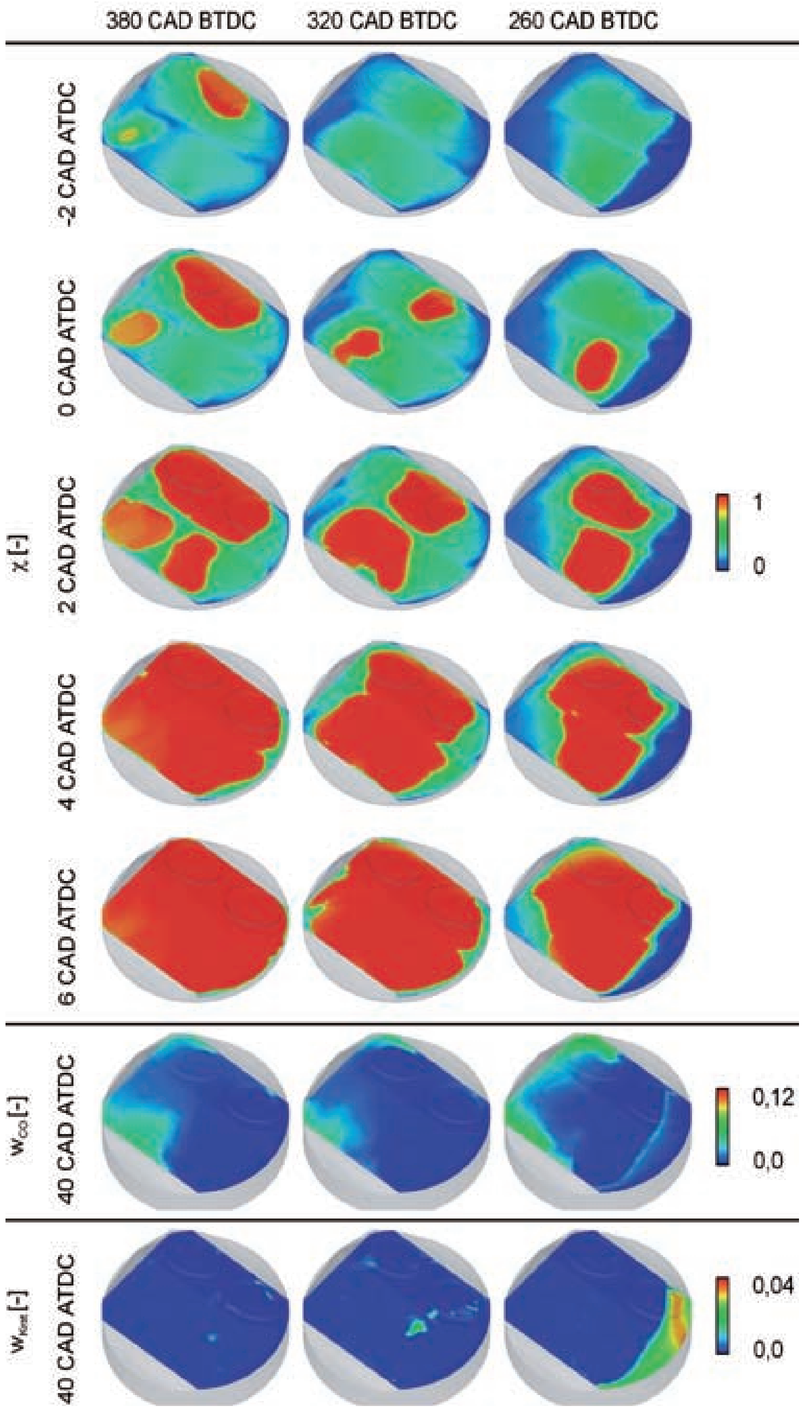

Figure 10: Progress of the reactions in case of early, middle and late starts of injection; CO mass fraction (wCO) and fuel concentration (wfuel) after combustion