

Investigations on the Effects of the Ignition Spark with Controlled Autoignition (CAI)

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ABSTRACT

Controlled Autoignition (CAI) is a very promising technology for simultaneous reduction of fuel consumption and engine-out emissions [3, 4, 9, 16]. But the operating range of this combustion mode is limited on the one hand by high pressure gradients with the subsequent occurrence of knocking, increasing NO_x -emissions and cyclic variations, and on the other hand by limited operating stability due to low mixture temperatures. At higher loads the required amount of internal EGR decrease to reach self-ignition conditions decrease and hence the influence of the ignition spark gain. The timing of the ignition spark highly influence the combustion process at higher loads. With the ignition spark, pre-reactions are initialized with a defined heat release. Thus the location of inflammation and flame propagation can be strongly influenced and cyclic variations at higher loads can be reduced.

INTRODUCTION

In light of the ongoing discussions of CO_2 -emissions, future gasoline engine designs require improved fuel consumption. Furthermore, they have to meet increasingly stringent emission standards. Research engines running in CAI mode were operated with effective efficiencies higher 40 % [13]. The fuel consumption can be reduced for example in the NEDC of about 13 % without the need for a deNOX catalyst [10] because the NO_x -emissions can be reduced over 90 % at lower loads [11].

The operating range of this combustion mode is limited to higher loads by high pressure gradients with subsequent noise emissions and component stress and to lower loads by limited operating stability due to low in-cylinder temperatures. In order to expand the operating regime, current research is being performed on various concepts, such as forced induction, external EGR, cooling system modifications, fuel stratification, and variable compression [1, 2, 17]. Most of these investigations show that charge inhomogeneities reduce the pressure gradients at high loads. With higher loads, the required mass of hot residuals to reach self-ignition conditions decrease due to higher exhaust gas temperatures and therefore higher exhaust gas enthalpy. This leads to a decreasing dilution of the in-cylinder charge, whereby the influence of a ignition spark becomes more important. With the ignition timing, another parameter is given to take influence on the location of the 50 % MFB. Homogeneous external EGR positively affects the pressure gradients but negatively affects the operating stability [9]. In order to reduce the pressure gradients at high loads, stratified external EGR can be used to create an inhomogeneous temperature distribution at the start of combustion. With lower external EGR concentration at the spark plug region, the importance of the ignition spark can be retained.

In this paper, experimental and numerical investigations on the effect of the ignition timing were performed at different loads on a single-cylinder engine. The engine is equipped with a mechanical VVT system on both, the intake and exhaust side. In addition to thermodynamic

analyses, 3-D CFD simulations and optical investigations were performed. The calculations include the exhaust gas and fuel-mixture distributions and the combustion. The 3-D simulation of the combustion process was performed with an efficient chemical model, based on a detailed reaction mechanism. This model was coupled with the CFD-Code.

The optical investigations were performed with two measurement techniques. The first one was a high-speed camera coupled with an image intensifier and the second one was an assembly of two high-speed photomultiplier-cameras which are connected and synchronized in order to detect the flame radiation during the inflammation from different perspectives via optical fiber endoscopes. Using a newly developed software tool, a three-dimensional reconstruction of the flame propagation and the mapping of the ignition spots was possible for the different loads.

The results show, that the timing of the ignition spark highly influence the combustion process at higher loads. With the ignition spark, pre-reactions are initialized with a defined heat release. Thus the location of inflammation and flame propagation can be strongly influenced and cyclic variations at higher loads can be reduced. The amount of heat release caused by the spark ignition highly depends on the air/fuel ratio. To analyze the role of the spark ignition at high external EGR-rates, a stratification of the external EGR was realized with impulse charge valves and a modified intake manifold. Therewith a temporally and spatially separated intake of fresh air and exhaust gas was realized. The external EGR was added to the inter-nal EGR, which was retained during NVO. With lower external EGR-rates at the spark plug region, the timing of self-ignition at the outer regions with high external EGR-rates can be determined with the amount of the heat re-lease rate of the pre-reactions caused by the ignition spark.

EXPERIMENTAL SETUP

A typical in-cylinder pressure curve and the corresponding valve lifts are shown in figure 1.

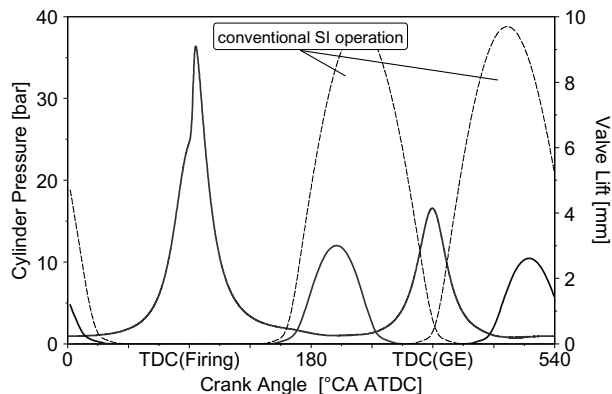


Figure 1: Cylinder pressure curve and valve lifts (CAI)

To reach self ignition combustion conditions a negative valve overlap with early EVC was realized. The cylinder pressure curve for this strategy shows the typical characteristic compression at TDC(GE). The trapped residual exhaust gas is compressed until TDC(GE) during negative valve overlap and so self ignition conditions are obtained near TDC(Firing).

The valve lifts of a conventional SI operation are also shown in this figure to demonstrate the needed modifications for CAI. To realize engine operation with a good efficiency the valve times need to be adjust due to the reduced maximum valve lifts, for the outlet side as well as for the inlet side. These investigations were performed in advance for all operating points shown in this paper. The required valve train flexibility was reached with a fully variable mechanical valve train adapted at the inlet and the outlet side. The test engine was a four stroke, pent-roof, single cylinder engine with two inlet and two outlet valves. The main engine specifications are listed in table 1.

Displacement [cm ³]	498
Number of valves	4
Bore [mm]	84
Stroke [mm]	90
Compression ratio	12:1
Fuel	Super Plus gasoline 98.4 RON
Injection pressure	120 bar
Injection System	Central mounted DI

Table 1: Single cylinder engine specifications

The test bench is fully equipped with measurement devices for all basic operation parameters. The air and fuel mass flow were measured with a Bronckhorst Hot-film air mass flow sensor and a Pierburg PLU for the fuel measurement. The pressures and temperatures were measured at all relevant positions in the intake and outlet ports. The exhaust gas concentration measurements for HC, CO, NO_x, CO₂ and O₂ were done with a Pierburg AMA 2000 and the exhaust A/F ratio with a wide range lambda sensor from ETAS. In addition, the engine is equipped with low pressure transducers (Kistler 4045 and 4075) mounted in the intake and the exhaust manifold close to the cylinder. The cycle-resolved pressure profiles of the intake and exhaust manifolds were also used for initialization of the 3-D CFD calculations. The in-cylinder pressure is recorded using a water-cooled pressure transducer (Kistler 6041), which permits a detailed analysis of the gas exchange and the high-pressure cycle using a high resolution incremental shaft encoder (3600 pulses/ revolution) and a Smetec Combi-Pro indication system. Oil pressure and coolant circulation are ensured by external pumps; both oil and water temperature can be adjusted by a conditioning system. The oil temperature was conditioned to 90 °C and the coolant temperature to 95 °C. The fuel used in the experiments has an octane number of 98.4 RON and 87.7

MON and was injected into the combustion chamber with a multi-hole injector placed in central position with an injection pressure of 12 MPa driven by an external fuel pump. The same fuel was used during all shown investigations.

STRATIFIED EXTERNAL EGR - During this investigations, the external EGR was added to the internal EGR, which is retained in the combustion chamber during the NVO period in the TDC of the gas exchange. Different possibilities of external EGR stratification are shown in figure 2.

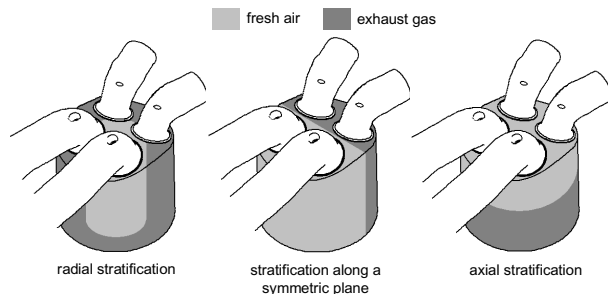


Figure 2: EGR-Stratification Modes

A strict separation of air and exhaust gas (ideal exhaust gas stratification) seems to be difficult in case of an intensive in-cylinder flow. A lower external EGR-concentration in regions of ignition spots and in the spark plug region has to be realized for high combustion stability. Therefore a high exhaust gas concentration gradient until the end of the compression stroke has to be obtained with lower EGR-concentration in the middle of the combustion chamber. Through an external port, the external EGR was supplied into the right inlet port, which was vertically separated by an air baffle. This air baffle in the inlet port, which reaches nearly up to the inlet valve, prevents an early mixing of air and exhaust gas. To reach the same flow structure of the fresh air through both inlet ports, another air baffle was adapted into the left inlet port. With this layout, a radial stratification can be achieved in the combustion chamber. The intake of exhaust gas and fresh air is temporally separated by Impulse Charge Valves (ICV's) developed by the company MAHLE International GmbH. These ICV's feature a high flexibility in adjusting the valve timing, short switching times and a low leakage ($V_L = 0.24 \text{ m}^3/\text{h}$ at $\Delta p = 60 \text{ kPa}$). With this basic principle, various intake strategies of both fluids can be realized, which can be read in [9]. With CAI first the external EGR flows in and afterwards fresh air. Otherwise a backflow of residual gas into the intake manifold occurs due to the overpressure in TDC(GE). Hence the air mass to reach higher loads can't be delivered. The basic principle of external EGR stratification is shown in figure 3. In these plots the lift-profiles of the ICV's are also shown. The analysis with homogeneous external EGR were realized with a standard EGR-valve mounted near to the inlet manifold. The orifices of the EGR-valve are distributed over the circumference of the inlet manifold, so a nearly ideal mixing of recirculated exhaust gas and fresh air is reached. The

homogeneous external EGR rate was measured with a CO_2 -measurement of the mixture in the inlet manifold and a CO_2 -measurement of the exhaust gas. This kind of determining the external EGR-rate was not possible during measurements with stratified external EGR due to the spatial separation of exhaust gas and fresh air. In this case the mass flow rate of the external EGR was measured with a bellows-type gas flow meter to determine the external EGR-rate.

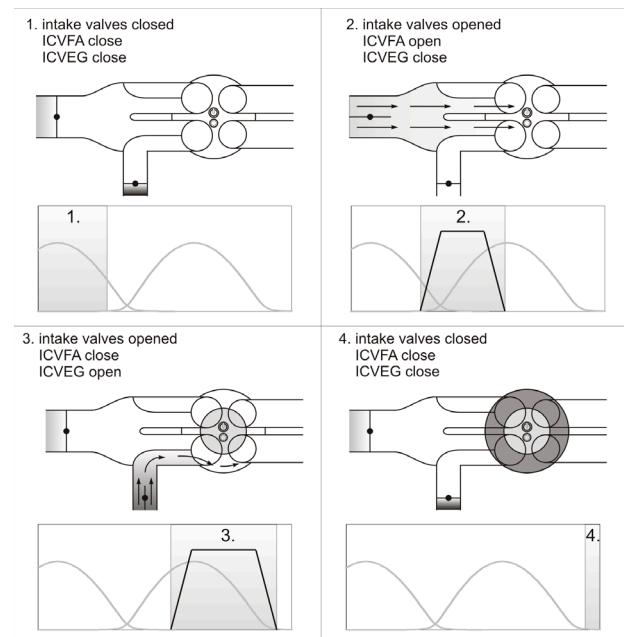


Figure 3: Principle of external EGR stratification

ANALYSIS TECHNIQUES

3-D CFD CALCULATION - A modular system of the calculation mesh was build for the 3D-CFD-calculations. Different intake meshes can be coupled to the existing cylinder mesh. The cylinder mesh with the standard intake manifold for the investigations with homogeneous EGR is shown in figure 4.

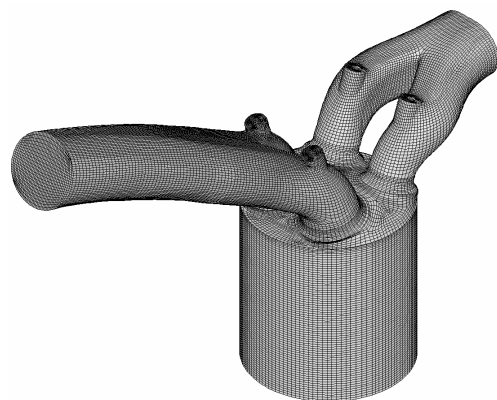


Figure 4: Calculation mesh for 3-D CFD calculations

The basic cylinder mesh without inlet and outlet cells includes approx. 210.000 cells in the BDC. For the calculations, the software Star-CD from Computational Dynamics was used. The gas exchange calculations with fuel injection were done from 20 °CA BTDC (charge cycle) until 0 °CA BTDC (firing). At the starting point, a latent gas mass was assumed for the whole calculation area. The combustion chamber and the exhaust ports were initialized with exhaust gas. Depending on the case of calculation, the inlet ports were initialized only with air, or with air and a corresponding level of exhaust gas. The calculations until combustion were performed with a time step of 0.1 °CA. The temporal discretization was implemented fully implicit. MARS algorithm was used for the spatial discretization. The standard k-ε-model was used for the turbulence modeling. The solution algorithm was the PISO-Algorithm.

Chemical kinetic model - Calculating the combustion in 3D-CFD simulation is a trade-off between calculation time and accuracy. Calculating detailed reaction mechanism delivers best results but is quite time-consuming. With simple models the calculating time can be reduced, but the accuracy is quite poor. An efficient method for modeling the chemical reactions in a CFD code was developed [7, 8] based on detailed reaction mechanism, which was generated for the used fuel by an automatic mechanism generator [12]. The mechanism meets the characteristics of a synthetic fuel consisting of the three main components iso-dodecane, cyclohexane and iso-octane. The mechanism consists of 1027 species and 5681 reactions [8]. The link between the CFD code and the chemical kinetics model is realized by defining a chemical progress variable whose source term is tabulated as a function of local conditions. This chemical progress variable is defined and the rate of this variable is pre-calculated for a homogenous reactor and tabulated, depending on the physical and chemical conditions, using detailed chemistry. The values of the progress variable range from 0 (fresh mixture of fuel and air before start of any reactions) and 1 (chemical equilibrium after combustion). The arc length of the reaction trajectory based on detailed kinetics in the state space was selected as progress variable χ . Figure 5 shows an example of such a reaction trajectory.

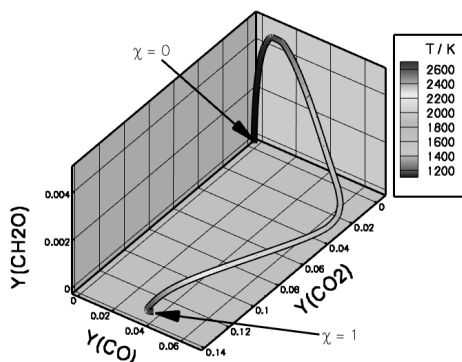


Figure 5: Reaction trajectory in the CO₂, CO and CH₂O mass fraction subspace

The implementation of the chemistry model in the CFD code Star-CD requires the solution of additional governing equations for the progress variable and for the mass fractions of various species (CO₂, CO, CH₂O, fuel). Given values of the pressure, air-fuel ratio, enthalpy of the gas mixture, the chemical progress variable, the chemical rate of the progress variable and the chemical mixture are looked up in pre-calculated tables of the detailed chemistry and passed to the CFD code. A more detailed description of the combustion model can be found in [7, 8].

OPTICAL ANALYSES - Two methods were used for combustion visualization. The first one of them was a high speed endoscopic imaging via an image intensifier. The used camera was a HighSpeedStar 6 from and the image intensifier an IRO-HS both from LaVision. To detect the main combustion, the OH-radiation was visualized using an UV capable endoscope and a bandpass filter with a range of 290-320 nm. Figure 6 shows the experimental setup for this technique.

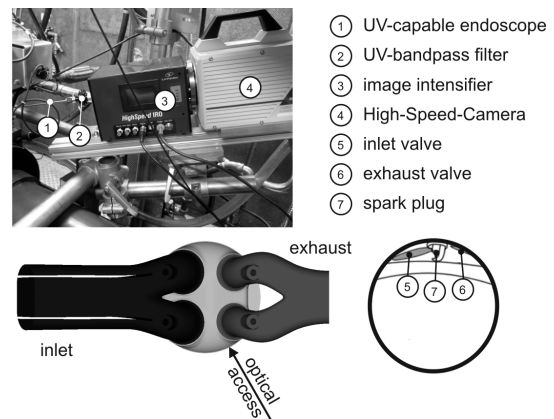


Figure 6: Experimental setup for high-speed visualization

The second method was an assembly of two high-speed photomultiplier-cameras and corresponding UV-capable optical fibre endoscopes. The highest sensitivity of the photomultiplier-camera is in the UV-range between 300 and 350 nm. But also radiation in the visible range is transmitted. These two devices were connected to a multi-camera system. To realize a simultaneous detection of the combustion process, these two systems were synchronized by a common indication system. Two computers, operating in a master-slave mode, were set up for data transfer and processing of control commands [5]. In Figure 7 the field of view for each endoscope and the location of both endoscopes are shown.

For the three-dimensional allocation of the ignition and flame propagation the voxel-based backprojection approach using discrete volume elements or voxels was used for both reconstruction and result output [6]. A CAD-based model is initialized with a mesh consisting of volume elements called voxels. Every single voxel inside the virtual combustion chamber corresponds to specific geometrical coordinates. These absolute coordinates are transformed into each camera coordinate system by