

Development of a High Turbulence, Low Particle Number, High Injection Pressure Gasoline Direct Injection Combustion System

Johann Peer and Fabian Backes
Technische Universitaet Muenchen

Henning Sauerland
Hitachi Europe GmbH

Martin Härtl and Georg Wachtmeister
Technische Universitaet Muenchen

ABSTRACT

In the present work the benefit of a 50 MPa gasoline direct injection system (GDI) in terms of particle number (PN) emissions as well as fuel consumption is shown on a 0.5 l single cylinder research engine in different engine operating conditions.

The investigations show a strong effect of injection timing on combustion duration. As fast combustion can be helpful to reduce fuel consumption, this effect should be investigated more in detail. Subsequent analysis with the method of particle image velocimetry (PIV) at the optical configuration of this engine and three dimensional (3D) computational fluid dynamics (CFD) calculations reveal the influence of injection timing on large scale charge motion (tumble) and the level of turbulent kinetic energy. Especially with delayed injection timing, high combustion velocities can be achieved.

At current series injection pressures, the particle number emissions increase at late injection timing. With up to 50 MPa it is possible to overcome this trade off. Higher injection pressures allow faster combustion by delaying injection timing without rising particle number emissions. Especially at high engine loads the enhanced combustion speed enables an extension of knock limit and the use of optimized spark timing. This results in higher thermal efficiency and reduced fuel consumption.

CITATION: Peer, J., Backes, F., Sauerland, H., Härtl, M. et al., "Development of a High Turbulence, Low Particle Number, High Injection Pressure Gasoline Direct Injection Combustion System," *SAE Int. J. Engines* 9(4):2016, doi:10.4271/2016-01-9046.

INTRODUCTION

Gasoline direct injection (GDI) was mainly introduced to reduce fuel consumption and to improve performance [1]. The main reason lies in the cooling effect of the directly injected fuel. This reduces the knock tendency which in turn increases power output and allows higher compression ratios. The trend towards Downsizing even pushed the introduction of this innovation as the direct injection concept is better suited for a combination with supercharging compared to port fuel injection. Especially when scavenging is applied, direct injection is advantageous, as it shows lower hydrocarbon emissions.

While showing a large potential in terms of reduced fuel consumption, particle number (PN) emissions are a challenge with GDI engines. As regulations on PN emissions get stricter [2] [3], the direct injection system demands for further improvement. On the one hand, the injection timing should be as early as possible for providing enough time for a homogeneous mixture formation. On the other hand, the injection timing needs to be optimized to avoid wetting of piston, valves or liner.

Another possibility to reduce PN emissions is to increase the injection pressure [4][5][6]. Higher injection pressure reduces the fuel droplet size promoting faster evaporation and thus reduces soot emissions. In today's series gasoline engines direct injection with fuel pressures of usually up to 25 MPa is applied [7]. A noteworthy reduction in fuel consumption by using injection pressure higher than 20 MPa could not yet be demonstrated [5].

Besides the positive effect of high pressure direct injection on emissions, in-cylinder phenomena get more complex with increasing fuel pressure. The interaction between the spray of a multi-hole injector and charge motion was investigated in [8]. This study shows a reduction of combustion duration with later start of injection (SOI). In this case, the tumble motion is increased by the momentum of the spray. If early injection timings are applied in order to increase the duration for mixture formation, the tumble ratio is reduced and thus the combustion duration gets longer. The limits for early and late injection timing are defined by increasing soot emissions caused by

wall wetting (early) and too short time for mixture preparation (late). This correlation between injection timing and PN emissions was also observed in [4].

In [9] the influence of injection timing on volumetric efficiency and on knocking at full-load torque of a direct injection spark ignition (DISI) engine was investigated using a swirl-type injector. As the knock tendency is lower for lower final charge temperature, a later start of injection is preferred. Whereas, for high volumetric efficiency end of injection and vaporization should be finished before the intake valves close. In this paper also split injection was investigated as a method to achieve high volumetric efficiency and low knock tendency.

In [10], split injection with one early injection during the intake stroke and a second injection at the end of the intake stroke was compared to single early injection at intake stroke regarding knock behaviour by means of high speed imaging and CFD simulation. These experiments were conducted at 10 MPa injection pressure. One of the main results was that the distribution of turbulent kinetic energy is different for the two injection strategies, also indicating a strong interaction between spray and charge motion.

The present work analyzes the flame propagation for different injection timings and injection pressures. Furthermore, the interaction between air motion and spray is investigated by means of particle image velocimetry (PIV) measurements and 3d CFD calculations. Finally, engine experiments at higher loads show the influence of injection on PN emissions and fuel consumption.

EXPERIMENTAL SETUP AND NUMERICAL METHODOLOGY

The experimental investigations were carried out using a single-cylinder engine [11] based on the Audi EA888 [12] [13]. Table 1 summarizes the basic specifications of the research engine, which is also used in an optical configuration by exchanging the cylinder head and piston.

Table 1. Specifications of the Research Engine

	Thermodynamic Engine	Optical Engine
Basic Engine Concept	Audi EA888 Gen2	
Bore	82.51 mm	
Stroke	86.6 mm	
Displacement	463 cm ³	
Geom. Compression Ratio	9.92	8.64
Injection System	Side mounted direct injection	
Valvetrain	Hitachi, full variable	
Exhaust Valves	Lift: 2.1...10.3 mm	
	Phase: 187...257°CA	
Intake Valves	Lift: 1.6...9.0 mm	
	Phase: 450...520°CA	
Max. Load (IMEP)	17 bar	5 bar
Fuel	Commercial Gasoline RON 95	

The research engine is equipped with a fully variable valve train for the inlet and exhaust valves, based on a Hitachi series system [14]. The valve lift can be continuously adjusted by an electro-mechanical Valve Event and Lift (VEL) system and the valve phasing is adjustable by a hydraulic Variable Timing Control (VTC) system.

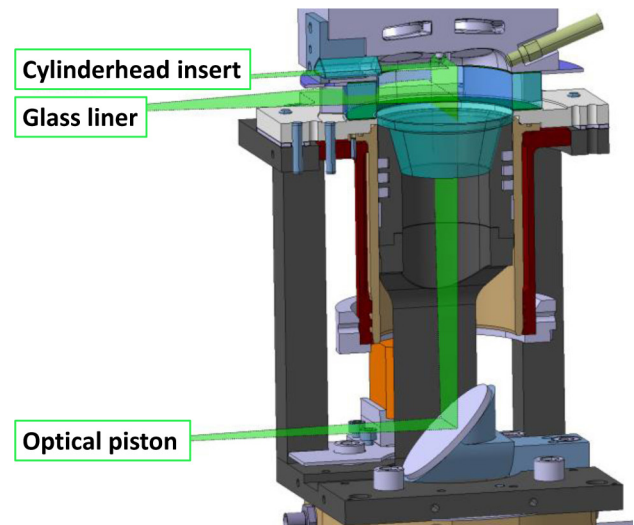


Figure 1. Optical configuration of the single cylinder research engine with three optical accesses

Figure 1 shows the accesses of the optical engine configuration. A glass insert allows to observe the roof-shaped combustion chamber. Additionally, in-cylinder phenomena can be viewed through a glass liner, 21.5 mm in height. Finally, by using an elongated piston with a glass insert it is possible to observe the combustion chamber through a mirror. In contrast to the metal piston, the optical piston shape is flat and does not feature valve recesses. Due to the differences in cylinder head and piston shape the compression ratio is slightly reduced in the optical configuration compared to the thermodynamic setup (cf. Table 1). However, the geometry of the inlet and outlet ducts remain the same for both configurations, hence, only minor differences in charge motion and combustion behaviour are expected.

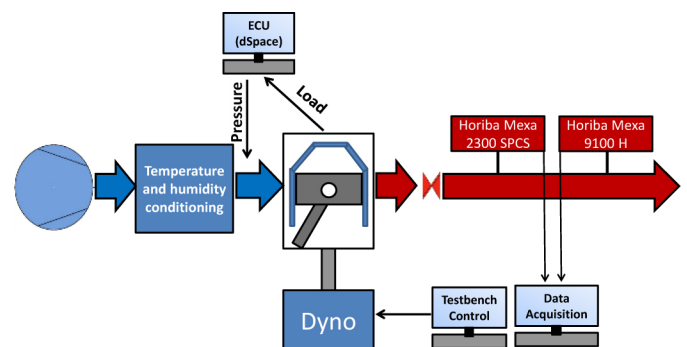


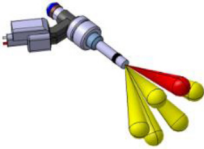
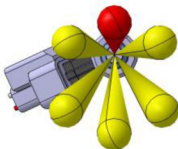
Figure 2. Test bench setup

Figure 2 depicts an overview of the test bench setup. The engine is controlled by a freely programmable ECU based on a dSPACE rapid prototyping system. A feedback controller for the engine load is used to adjust the intake pressure based on the cylinder pressure indication. The ECU communicates with the test bench control and data acquisition (National Instruments) via CAN. ECU data, temperatures,

pressures and mass flow rates are measured and recorded by the data acquisition. Based on the cylinder pressure indication system also rate of heat release and total heat release can be calculated. Conditioning of all media (fuel, coolant, oil, intake air, exhaust pressure) allows for a reproducible operation of the test bench. For analysis of the exhaust gas emissions and particle number emissions a Horiba Mexa 9100 H and a Mexa 2300 SPCS system is used.

In the following investigations an injector prototype by Hitachi is used, which is capable of up to 50 MPa injection pressure. The basic specifications of this side mounted injector are given in Table 2.

Table 2. Specification of the Hitachi injector

Injector type	Multi-hole	
Actuation	Solenoid	
Number of spray holes	6	
Max. Pressure [MPa]	50	
Stat. Flowrate at 10 MPa [cc/s] (n-Heptane)	10.1	

All investigations were performed in homogeneous stoichiometric combustion mode applying a single injection during the intake stroke. To determine the injection timings and all other engine settings for optical investigations, pretests at the thermodynamic engine were conducted. The influence of injection timing on combustion duration as investigated in [8] was also observed at the present research engine. For detailed investigations an early injection at 430°CA (290°CA before Top Dead Center firing (bTDCf)) and a late injection at 480°CA (240°CA bTDCf) were determined as representative points. SOI 430°CA is the earliest possible injection timing without piston wetting. The late injection starting at 480°CA shows the shortest combustion duration. Figure 3 shows these representative injection timings. The valve lifts and valve timings were kept constant for all measurements and are also shown in Figure 3.

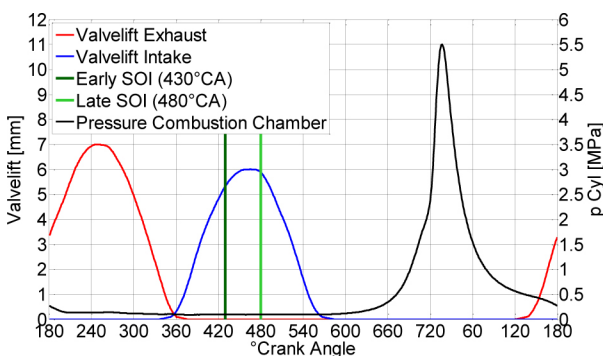


Figure 3. In-cylinder pressure curve, valvelift curves and injection timing of early (SOI430°CA) and late (SOI480°CA)

Setup for Optical Measurements

In this work, the optical engine configuration is used for flame luminosity and particle image velocimetry measurements. Details on both setups and the used equipment are explained in the following.

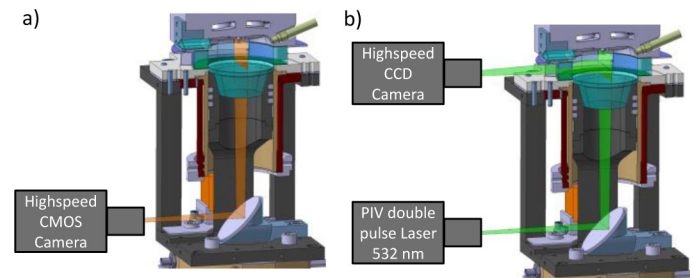


Figure 4. Setup of a) flame luminosity measurement and b) PIV at optical engine configuration

Flame Luminosity Measurements

In order to evaluate the flame propagation during combustion, the combustion chamber is observed with a high speed CMOS camera from the bottom of the piston via a mirror, cf. Figure 4 a). For each operation point 50 subsequent cycles are considered. Based on cycle-averaged data a threshold of intensity is set to separate the burning area from the unburned, with this procedure the flame area is determined. The high speed camera takes a picture each two degree crank angle, so the evaluation of flame area is done for every second degree crank angle. The field of view is limited by the minimum diameter of the glass inlay of the piston, that is, 53.2 mm compared to the bore diameter of 82.51 mm. Therefore, the area of the flame is expressed relative to the visible area in the following.

PIV Measurements

The flow field in the cylinder is measured by PIV. The main components of the measurement setup are a double pulsed laser (Litron Nano TI 200-15-PIV) and a CCD camera (pco.1600), cf. Figure 4, b). The laser beam passes a lens system to form a light sheet which is then guided into the combustion chamber by the piston mirror. For the present measurements the light sheet is adjusted to the cylinder midplane covering the intake and exhaust side of the combustion chamber. The intake air is seeded with oil (particle size nearly 1 nm) by a Laskin nozzle aerosol generator. In the chamber the light is scattered on the oil seeded intake air and finally captured by the camera through the glass liner. The latter restricts the field of view to an area of 51.5 x 21.5 mm. Furthermore, the glass liner is partly blocked by the piston from 53°CA bTDC on. The time separation between the two laser pulses is adjusted to the expected flow velocities, depending on the measured crank angle and engine speed, in the range of 7 to 20 μs. For each operating point 30 subsequent cycles are measured.

The evaluation of the PIV measurement data is performed using the commercial software Davis 7 by LaVision. The contrast of the background and the oil particles is enhanced by a sliding background subtraction (length 6 pixel). Thereby, the accuracy of the used cross correlation algorithm can be improved. The algorithm works iteratively from an interrogation window size of 128 x 128 pixel to a

size of 64 x 64 pixels with an overlap of 50 % each. Post-processing functions finally remove inappropriately calculated vectors by comparing each velocity vector with its surrounding vectors in terms of magnitude and orientation. To obtain the mean velocity field the 30 cycles of an operating point are averaged with Matlab. Additionally, the turbulent kinetic energy is calculated according to Reynolds decomposition from the difference of the velocity vectors of the single cycles and the mean velocity [15].

3D CFD Simulations

As mentioned before the PIV measurements allow to investigate only a limited field of view and limited time span before the piston blocks the glass. Furthermore, the measurements only show the in-plane velocities in the cylinder midplane. 3D CFD simulations are used in this work to complement the PIV measurements, thereby, allowing for detailed analysis in the complete combustion chamber and at all degree crank angles.

An in-house CFD code of Hitachi [16] is used for 3D in-cylinder simulations. The conservative equations of momentum, energy, and mass for a compressible fluid are discretized by means of a finite volume method on Cartesian meshes. A standard k- ϵ turbulence model is applied, that is, RANS (Reynolds-averaged Navier-Stokes) equations are solved and the discrete droplet method (DDM) is used to track the fuel spray behavior. The spray model parameters and boundary conditions are calibrated to spray measurements for the present study as shown in the [appendix](#).

COMBUSTION ANALYSIS

In the following, the influence of the start of injection on the combustion speed is evaluated with the help of flame luminance measurements and the cylinder pressure indication.

The operation condition was set to 3.3 bar indicated mean effective pressure (IMEP) and 2000 rpm as a representative operation point for low engine load. One representative SOI for 'early' and 'late' injection was chosen: 430°CA and 480°CA, respectively. In order to obtain comparable results the MFB50 was set to 8°CA aTDC for each case by adopting the spark timing.

[Figure 5](#) shows the cycle averaged flame luminance images for both SOI.

With the late SOI (480°CA) the flame propagation is faster than with the early SOI (430°CA). This becomes even more obvious in the quantitative evaluation of the flame propagation in [Figure 6](#).

Based on the already described evaluation of the flame area, the averaged radial flame velocity was estimated. Therefore a circle with the same area as the area defined as burning was assumed. The increase in radius of this circle between subsequent optical measurements defines the radial flame velocity. A significant increase in the flame velocity could be observed for the late SOI.

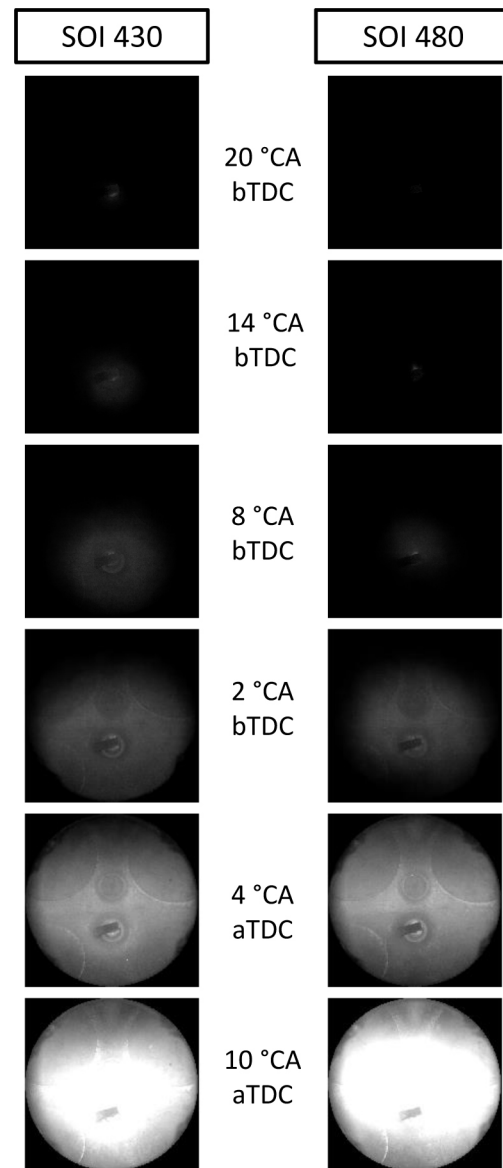


Figure 5. Flame luminosity measurements of SOI 430 and 480°CA at 2000 rpm, 3.3 bar IMEP and 50 MPa injection pressure

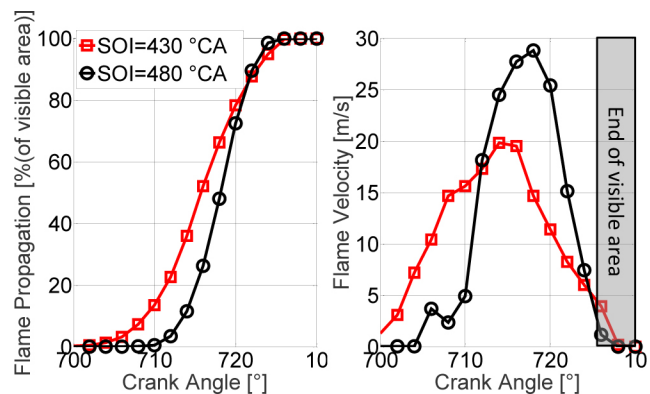


Figure 6. Evaluation of flame propagation and flame velocity at 3.3 bar IMEP, 2000 rpm for SOI 430°CA and 480°CA at 50 MPa injection pressure

[Figure 7](#) shows the combustion duration determined from the cylinder pressure indication which was tracked simultaneously during the optical measurements. In this case combustion duration means the