

High Pressure Gasoline Direct Injection in Spark Ignition Engines - Efficiency Optimization through Detailed Process Analyses

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ABSTRACT

At part load and wide open throttle operation with stratified charge and lean mixture conditions the Direct Injection Spark Ignition (DISI) engine offers similar efficiency levels compared to compression ignition engines

The present paper reports on results of recent studies on the impact of the in-cylinder processes in DISI engines e. g. the injection, the in-cylinder flow, the mixture preparation and the ignition on the combustion, the energy conversion and the exhaust emission behavior. The analyses of the spray behavior, of the in-cylinder flow during compression as well as of the flame propagation have been carried out applying advanced optical measurement techniques. The results enable a targeted optimization of the combustion process with respect to engine efficiency and exhaust emissions. The benefits of an increase in fuel injection pressures up to 100 MPa are discussed. The investigations have been mainly performed in a single cylinder research engine featuring a spray-guided injection process with central injector location. The considered operating range includes part load conditions with both stratified and homogeneous mixture preparation as well as full load operation with homogeneous mixture preparation. The results obtained from the research work are very promising and hence a 4-cylinder production engine is modified to operate with a suitable injection system with rail pressures in the range of 50 MPa to 60 MPa.

Additionally, irregular combustion events in the full load regime such as knocking combustion and low speed pre-ignition have been investigated in a 4-cylinder DISI production engine. The results support the development of mitigation strategies for both engine knock and low speed pre-ignition (LSPI).

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INTRODUCTION

Today, most Direct Injection Spark Ignition (DISI) engines are developed as downsizing concepts featuring homogeneous mixtures and boosting (turbocharging). These engines exhibit increased power output, improved torque behavior as well as lower fuel consumption in typical test cycles compared to conventional gasoline engines with Multi-Point Port Fuel Injection (MPFI).

Future emission legislations will focus on real driving conditions that generally result in increased fuel consumption and exhaust emissions compared to the present testing conditions. Thus, a considerable optimization of the combustion process has to be achieved in order to meet future emission standards under real driving conditions and/or in the World Harmonized Test Procedure (WLTP).

A considerable improvement in engine efficiency and emissions can be achieved implementing a mixture stratification strategy in the part load regime in combination with boosting. However, in high to full

load operation a stoichiometric homogeneous mixture is still required. Moreover mixture enrichment at full load as a means to reduce the thermal loads in critical parts has to be avoided.

In-cylinder mixture preparation enables a reduction of in-cylinder temperatures due to the vaporization of the fuel which yields to higher knock resistance in comparison with conventional Spark Ignition (SI) engines operating with intake manifold injection (MPFI). As a consequence the compression ratio of these engines can be increased.

The current status of DISI engine technology has been achieved through advances in high-pressure fuel injection systems like common-rail-injection, powerful electronic control systems, the development of new measurement techniques and improved numerical simulation providing more accurate analyses and a deeper understanding of the combustion process. This is intensified development activities during the last 20 years, up to now a gasoline

direct injection concept, that allows the combination of mixture stratification and homogeneous mixture preparation in the entire engine operation area in combination with high boosting has not been introduced yet. To meet the future requirements concerning fuel economy and exhaust emissions a control in-of the in-cylinder gas motion, an optimization of the fuel injection system as well as a stable combustion process are required. Additionally, catalytic converters and particulate filters must be developed to enable an easy exhaust gas after-treatment over the entire engine map.

CONCEPTS OF DISI ENGINES

Two different injection strategies are established in DISI production engines which are mainly characterized by the distance between the locations of the injector and the spark plug, [Figure 1](#).

The DISI concept with a lateral injector is the predominant in today's production engines [1, 2, 3, 4]. It is characterized by a wide spacing of the locations of the injector and the spark plug. The injected fuel is guided along a combustion chamber wall, usually along a shaped piston crown. The impingement of the fuel on the piston effects the atomization of the droplets and accelerates evaporation due to the piston surface temperature. The mixture transport to the spark plug, driven both by the combustion chamber geometry and the charge motion, requires an extended time period for mixture preparation and reduces the level of stratification in premixed regimes. In addition, a controlled gas motion i.e. a swirl, a tumble or a reverse tumble is required. Due to the impingement of fuel, a wall film of liquid fuel is developed on the piston crown. Particularly during cold-start and warm-up operation the wall film will not evaporate entirely and lead to non-premixed and incomplete combustion promoting an increase in emission of unburned hydrocarbons (HC) and particulates. These engines feature homogeneous mixture with stoichiometric air-fuel ratio under part load and enriched mixture conditions for example at higher and full loads, as well as during cold-start and engine warm-up phases. To compensate efficiency penalties of SI engines compared to compression ignition engines a combination of gasoline direct injection and boosting have been introduced into the market recently.

The DISI engine concept featuring an injector located in the center and referred to as "spray-guided direct injection", has been introduced recently in production engines. Both mixture formation strategies i.e. mixture stratification [5, 6] at part load and homogeneous mixture preparation at higher loads including full load or homogeneous mixture preparation over the entire engine map [7] can be realized. This strategy provides a flammable mixture in the vicinity of the spark plug in a very short time interval enabling high levels of stratification with ideal ignition phasing. Under optimal conditions, any contact between liquid fuel and combustion chamber walls is avoided. At full load operation the fuel injector position in the center of the cylinder head provides excellent homogeneous fuel distribution. In contrast, at part load a very lean average mixture can be realized.

[Figure 2](#) shows the different operating modes of spray-guided direct injection. Gasoline injection during the intake stroke results in a homogeneous mixture distribution at ignition phasing exhibiting an approximately constant air-fuel-ratio throughout the entire

combustion chamber. In the stratified mode during part load the fuel is injected in the compression stroke immediately before ignition (left). Ideally, an ignitable gaseous mixture cloud develops in the center of the combustion chamber around the spark plug electrodes. Moreover, the ignitable mixture cloud should be embedded in pure air and/or a mixture of air and residual gas that is not required for the combustion. In addition, widely opened throttle operation is required.



Air-guided



Spray-guided

Figure 1. Predominant direct injection concepts.

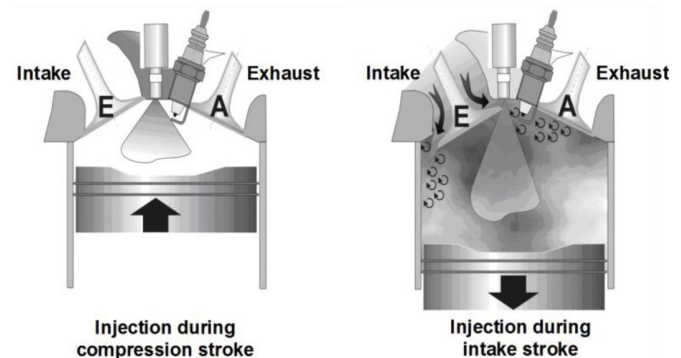


Figure 2. Mixture formation modes

Spray-Guided Direct Injection (Stratified Charge)

The spray-guided direct injection and charge stratification at part load was first introduced in the market by Daimler in 2006 in a 3.5 l V6 engine with a piezo injector placed at the center in a four-valve cylinder roof [5]. The piezo injector featured an outwardly opening nozzle, known as A-nozzle. The fuel has been injected into the combustion chamber with a pressure of 20 MPa. The potential of fuel economy of this engine compared to engines with port fuel injection and stoichiometric mixture is shown in [Figure 3](#).

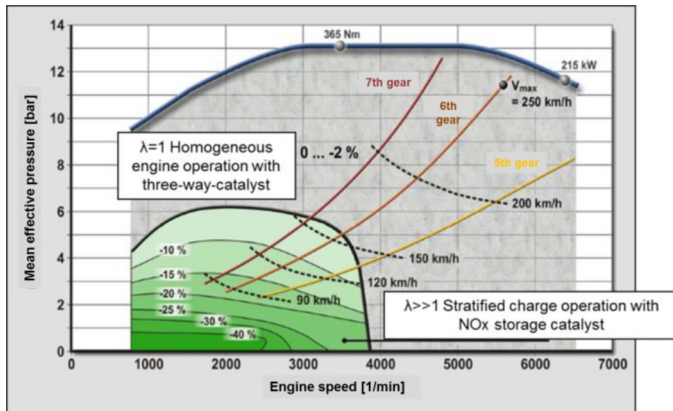


Figure 3. Operating range with stratification and fuel economy compared to conventional MPI engines.

Significant fuel economy can be obtained throughout the entire engine map. In test cycle, fuel consumption was approx. 10% lower than the fuel consumption of comparable engines with port fuel injection. A specific fuel consumption of 290 g/kWh at the operating point of BMEP = 2 bar at an engine speed of 2000 rpm still represents a benchmark for gasoline engines.

The level of stratification is generally limited by two aspects: Excessive particulate emissions as a result of the rich fuel-air mixture at center and increased nitrogen oxide emissions produced in the outskirts of the mixture cloud featuring lean combustion. To extend the area of mixture stratification in the engine map, the mixture preparation and the inflammation of the fuel-air mixture have to be optimized as displayed in Figure 4. Detailed investigations of the spray behavior, different injection strategies, and the interaction of the flow field and the spray have been performed. The investigations were supported by measurements of the flow field inside the combustion chamber, carried out using high-speed particle image velocity (HS-PIV), as described in [7].

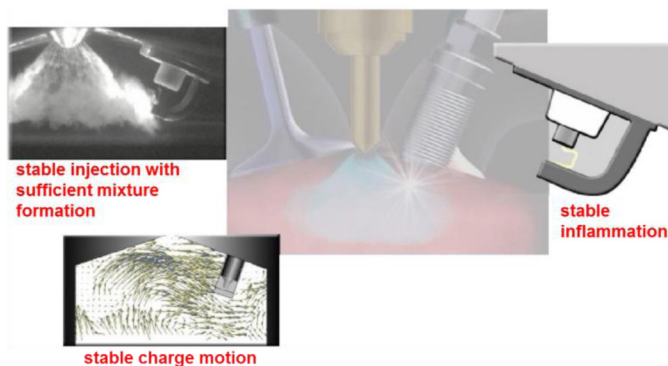


Figure 4. Detailed process analyses.

Figure 5 shows an example of results obtained by a combination of fuel spray visualization and HS-PIV of the flow fields inside the combustion chamber.

Independent of cycle-to-cycle variations in individual combustion cycles, the averaged frames computed out of 100 individual combustion cycles show a distinctively uniform flow field. The first image at 46 °CA BTDC displays the development of the tumble

motion immediately before the first injection. At that crank angle the tumble vortex center is located underneath the spark plug. The spray propagation during the first injection is depicted in the second picture representing the flow field at 43 °CA BTDC. The instantaneous flow field between the first and the second injection 35 °CA BTDC is illustrated in the third picture. Three distinguishable vortices form due to the transfer of momentum from the spray on the flow (see red crosses).

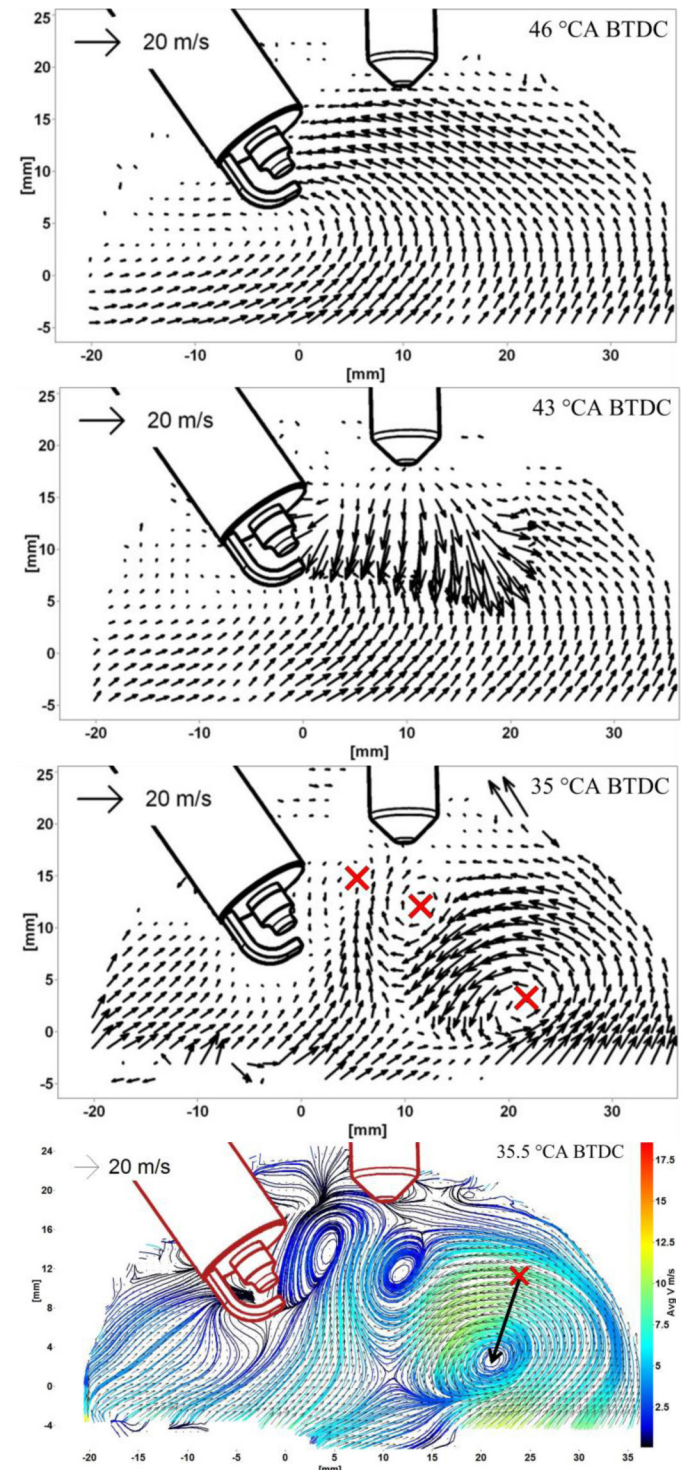


Figure 5. Flow field characteristics (averaged over 100 cycles).
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