

Investigation on the Effect of Very High Fuel Injection Pressure on Soot-NO_x Emissions at High Load in a Passenger Car Diesel Engine

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ABSTRACT

Previous research has shown that elevating fuel injection pressure results in better air-fuel mixture formation, allowing for a further increase in maximum exhaust gas recirculation (EGR) rate while consequently reducing NO_x emissions. The aim of this paper is to find out whether there is an optimum injection pressure for lowest soot-NO_x emissions at a given boost pressure in high-speed diesel engines.

Experiments are carried out on a single-cylinder research engine with a prototype common-rail system, capable of more than 200 MPa injection pressure. The effect of injection pressure on soot-NO_x formation is investigated for a variety of boost conditions, representing the conditions of single to multi-stage turbocharger systems. Analysis of the data is performed at the application relevant soot to NO_x ratio of approximately 1:10.

It is observed that above a critical injection pressure, soot-NO_x emissions are not reduced any further. The value of this critical injection pressure increases with load or boost pressure. When increasing the injection pressure above the critical value at a low boost pressure, soot-NO_x emissions may rise. The relative air-fuel ratio and EGR rate at the soot to NO_x ratio of 1:10 also show effects of saturation when increasing the fuel injection pressure beyond the critical point.

INTRODUCTION

In various publications, fuel injection pressure is supposed to be a superior enabler to improve the diesel combustion process. Pauer [1] stated that in terms of air-

fuel mixture formation there is no limitation to the potential of injection pressure. Optical investigations revealed the increase in both penetration length of the gaseous phase of the fuel jet and spray tip velocity with increasing injection pressure. Especially the spray tip velocity has a crucial impact on air-entrainment which disperses the diesel spray. The air-entrainment is controlling the fuel vaporization and thus the diesel combustion after the premixed burn period, stated Siebers [2]. Rajalingam and Farrell [3] observed a conical control surface around diesel sprays and reported a rising air mass flux across this area with increased injection pressures. Likewise, Bruneaux [4] and Shundoh et al. [5] demonstrated that increasing fuel injection pressure leads to enhanced air-entrainment and higher air-fuel mixing velocities. A local change in the mixing rate with increased injection pressure is not detected. The benefit is rather based on an advanced global air-entrainment shortening the time span of mixing formation for a constant fuel mass.

The lift-off length, the distance between nozzle tip and stabilized near-nozzle flame front, is believed to play an important role in the formation of soot emissions. Siebers and Higgins [6] reported that increasing the injection pressure elongates the lift-off length. Because the liquid length of the fuel spray is not affected by the injection pressure, there is probably a lower interaction between liquid phase and the volume captured by the flame. Furthermore, higher injection pressures increase the percentage of regions with stoichiometric air-fuel ratio in a cross-sectional area of the fuel jet. This corresponds to Laguitton et al. [7], illustrating that the ignition takes place in the downstream vapor zone with higher injection pressures compared to the periphery of the liquid spray core when applying only low injection pressures. The relevance of the lift-off length is pointed

out by Siebers and Pickett [8], saying that the amount of air and fuel premixed upstream of the near-nozzle flame front is a prerequisite for low soot concentrations in the fuel jet. By increasing the cross-sectional average relative air-fuel ratio to more than 0.5 upstream of the flame front, soot concentration is negligible. Peak soot concentration scales inversely proportional with injection velocity, i.e. square root of the pressure drop across the nozzle hole.

Evaporation of the fuel spray is affected by the in-cylinder temperature as well, [7]. The temperature can be increased by mixing the fresh air charge with EGR. According to Nikolic and Iida [9], EGR also leads to reduced average flame temperatures which decreases NO_x emissions. Exhaust gas recirculation has an influence on soot formation by means of increasing the lift-off length. Siebers et al. [10] reported an inversely proportional dependency of the lift-off length on the in-cylinder oxygen concentration. Yet another influencing parameter on combustion and soot-NO_x emissions is the boost pressure. As Aoyagi et al. [11] stated, elevated boost pressure improves combustion and thermal efficiency and reduces soot emissions.

A combination of high injection and boost pressures and elevated EGR rates was investigated by Wakisaka et al. [12]. A possible simultaneous reduction in soot and NO_x emissions is reported. Flames are less luminous, due to lower flame temperatures and an increased soot oxidation. The bandwidth of the flame temperatures is smaller under aforementioned boundary conditions, probably thanks to a more homogeneous spray and combustion respectively. In a relative air-fuel ratio-temperature-map, the mixture tends to higher relative air-fuel ratios and lower temperatures (away from the core regions of soot and NO_x formation) with an increase in boost and injection pressure and increased EGR rates.

The vast number of investigations concerning fuel injection pressure and soot-NO_x emissions has a common tenor attesting injection pressure a virtually unlimited potential to reduce emissions (especially soot). Only Wakisaka et al. [12] clearly postulated that a further simultaneous reduction in soot and NO_x emissions can only be obtained by increasing injection pressure, boost pressure and EGR rate at the same time. The object of this work is to identify possible limits of the potential of fuel injection pressure on soot-NO_x emissions reduction. Particular attention is paid to the relationship between injection pressure and boost conditions.

EXPERIMENTAL SETUP

Experiments were conducted using a passenger car series production cylinder head mounted on a research single-cylinder engine block. Temperature of engine

coolant and oil were conditioned externally to 80 °C. Combustion air was supplied from a boost conditioning system. It was dry and its temperature upstream of the EGR feed was controlled to 50 °C. Table 1 shows the research engine specifications.

Table 1: Research engine specifications.

Cylinder head	4 cylinder
Valves per cylinder	2 inlet, 2 exhaust
Displacement per cylinder	499 cm ³
Bore and stroke	84 x 90 mm
Connecting rod length	136 mm
Compression ratio	16 : 1
Swirl at max. valve lift	2.1

The piston used for these investigations was characterized by a modern ω -bowl design, Figure 1. The ratio of the bowls maximum diameter to bore was 0.63. The ratio between height and maximum diameter was 0.29.

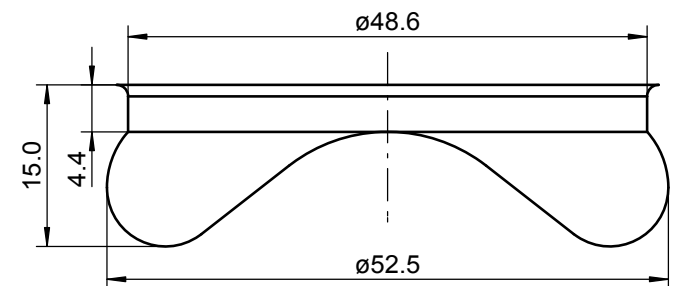


Figure 1: Piston bowl specifications.

The fuel injection equipment (FIE) was a common-rail prototype setup, capable of generating injection pressure levels considerably over 200 MPa. Table 2 lists information about the injector nozzle specifications. The engine was supplied with diesel fuel type EN590. The diesel fuel properties are given in Table 3.

Table 2: Nozzle specifications.

Nozzle type	Mini sac
Hydraulic flow rate	310 cm ³ / 30 s/ 10 MPa
Number of holes	6
Spray angle	160 °
Spray hole diameter	0.127 mm
Spray hole length	0.95 mm
k-Factor	1.5

Table 3: Diesel fuel properties.

Density (at 285 K)		822.1 kg/m³	
Viscosity (at 313 K)		2.03 mm²/s	
Cetane number		55.2	
Water mass fraction		79 * 10 ⁻⁶ kg/kg	
Distillation curve			
10 %	50 %	90 %	95 %
466 K	511 K	587 K	604 K

Emissions were measured and averaged over a 60 s time period using a Horiba MEXA 7100D exhaust gas analyzer. Smoke was detected via an AVL Smokemeter 415S. In this paper, all emission data are shown in gram per kilogram fuel. Hereby, soot emissions were calculated on basis of Bosch Filter Smoke Number (FSN) values as follows:

$$\text{soot}[\text{kg} / \text{m}^3] = \frac{5.32 \cdot \text{FSN} \cdot e^{0.31 \cdot \text{FSN}}}{0.405} \cdot 10^{-6} \cdot \frac{p_{\text{norm}}}{T_{\text{norm}}} \cdot \frac{T_{\text{meas}}}{p_{\text{meas}}}, \quad (1)$$

$$\text{soot}[\text{g} / \text{kg fuel}] = 1000 \cdot \frac{\text{soot}[\text{kg} / \text{m}^3] \cdot \dot{m}_{\text{exh}}[\text{m}^3 / \text{s}] \cdot t[\text{s}]}{m_{\text{fuel}}[\text{kg}]}, \quad (2)$$

where T_{norm} , p_{norm} are standard temperature and pressure (273.15 K, 0.101325 MPa) and T_{meas} , p_{meas} are measurement reference temperature and pressure (298.15 K, 0.1 MPa). The exhaust mass flow is $\dot{m}_{\text{exh}}$ and m_{fuel} is the injected fuel mass. Cylinder pressure was measured with a water cooled Kistler 6043A60 pressure transducer and the signal was recorded using AVL Indicom, also responsible for calculating the heat release rate and the associated mass fraction burned (MFB_x%) values and crank angles of mass fraction burned (alpha_MFB_x%). Each measurement was averaged over 25 cycles.

The upcoming trend of engine downsizing results in a load shift towards higher loads of the engine operating points in emissions certification driving cycles. To address this future demand of diesel engine development, this paper focused on high loads above 1.2 MPa indicated mean effective pressure (IMEP). Typical boost conditions from multi cylinder engines with different turbo-charger systems were used to define intake boost and exhaust back pressure, Table 4. The boost condition BC1 corresponded to a standard 1-stage turbo-charger system. BC2 represented an optimized 1-stage system and BC3 (BC4 at 1.6 MPa IMEP) was derived from a 2-stage system. The boost conditions at 1.2 and 1.6 MPa IMEP marked with an asterisk (BC4*, BC3*) were the same as BC2 at 2.0 MPa IMEP. They extended the boost pressure range at the lowest load by a virtual multi-stage turbo-charger boost condition and realized the opportunity to compare different loads at

constant boost pressure. Engine speed was 2000 rpm throughout the tests.

Table 4: Engine loads and boost conditions; BC4* (1.2 MPa IMEP) and BC3* (1.6 MPa IMEP) comply with BC2 (2.0 MPa IMEP).

Load (IMEP) [MPa]	Boost condition	Boost pressure [kPa]	Back pressure [kPa]
1.2	BC1	153	169
	BC2	174	192
	BC3	210	274
	BC4*	233	255
1.6	BC1	195	214
	BC2	201	211
	BC3*	233	255
	BC4	260	366
2.0	BC1	212	228
	BC2	233	255
	BC3	270	370

To derive comprehensive knowledge from the engine tests, the variable parameters (e.g. number of injections and separation) had to be constrained as much as possible. Thus, only single main injections were carried out eliminating the influence of variations in pilot injection combustion. Correspondingly, noise emissions could not be regarded. Mean injection pressure ($p_{\text{Inj_mean}}$) shown in the following diagrams was calculated by averaging the nozzle sac hole pressure over injection duration. This pressure trace was generated by means of 1-D hydraulic simulations of the entire injector using LMS AMESim and a Bosch internal nozzle model.

BACKGROUND AND TEST PROCEDURE

When applying EGR, the exhaust gases, fed back into the intake air system, replace a fraction of the fresh air charge. Due to the lower oxygen concentration in exhaust gases, the oxygen partial pressure in the cylinder charge is decreased. To maintain a constant IMEP, the same mass of fuel and oxygen has to be burned with a comparable heat release rate. Hence, a greater cylinder charge mass has to be transferred through the flame front in order to deliver equal oxygen mass such as in the case without EGR. This leads to an increased mass behind the flame front heated up by the combustion resulting in lower average in-cylinder gas temperatures. The higher specific heat capacity of exhaust gases compared to air also contributes to the reduction of the average in-cylinder gas temperature, but only to a considerably lower amount, [13]. By decreasing the gas and consequently the local peak temperature level, the formation of thermal NO_x emissions is significantly reduced. On the other hand, the decreased oxygen concentration leads to higher soot emissions due to further rich zones (soot formation) and a retarded

combustion (lower gas temperature, i.e. decreased soot oxidation), [14].

Figure 2 shows the effect of EGR on soot and NOx emissions by means of EGR sweeps of an injection pressure variation (refer to Figure 13 in the appendix for cylinder pressures and heat release rates). Each sweep starts with a measurement point without exhaust gas recirculation. The EGR rate is then gradually increased. The significant decrease in NOx with increasing EGR can be observed. By raising the injection pressure, the initial NOx emissions are increased, but due to higher possible EGR rates without increased soot level (EGR tolerance) the increase is overcompensated in comparison to lower injection pressure sweeps. At a constant relative air-fuel ratio value, a slight drawback in NOx can be identified for higher injection pressures. Concerning soot emissions, Figure 2 demonstrates the rise with increasing EGR. An elevated injection pressure shifts this slope towards lower relative air-fuel ratios resulting in decreased soot emissions at a constant air-fuel ratio. Hence, a higher injection pressure implies an improved EGR tolerance. The rise at the end of an EGR sweep becomes steeper with increasing injection pressure.

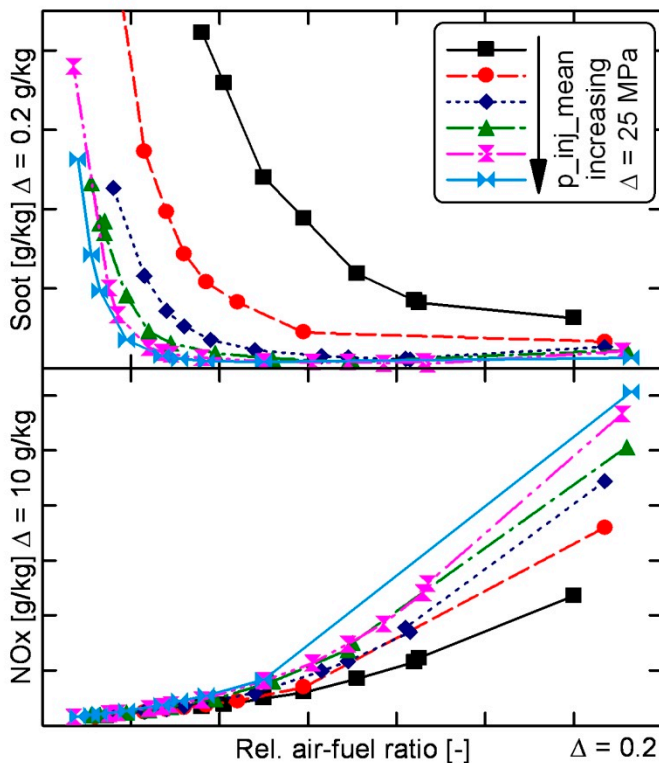


Figure 2: Soot and NOx emissions over relative air-fuel ratio, EGR sweeps of injection pressure variation; $n = 2000$ rpm, IMEP = 1.2 MPa, BC4*, SOE = 1 °CA aTDC.

Combining soot and NOx emissions in one diagram, the soot-NOx trade-off is obtained. It illustrates the conflict between reduced NOx and increased soot emissions by increasing the EGR rate. The injection pressure variation from Figure 2 is shown as soot-NOx trade-offs in Figure

3. With increasing injection pressure, the trade-offs move towards smaller soot and NOx values. However, the grade of improvement of each injection pressure step decreases with higher pressure levels. All trade-offs cross the “1:10” line indicating the constant ratio of soot to NOx emissions values of 1:10. Until the actual European emissions legislation Euro 4 [15], the emission limits of soot and NOx have always been defined accordingly to this ratio. Even with more challenging future emission limits requiring a particulate filter, soot-NOx ratio of 1:10 should not be exceeded significantly to avoid fuel consumption penalties due to increased regeneration intervals. Therefore, engine results at a soot-NOx ratio of 1:10 were examined throughout in this investigation.

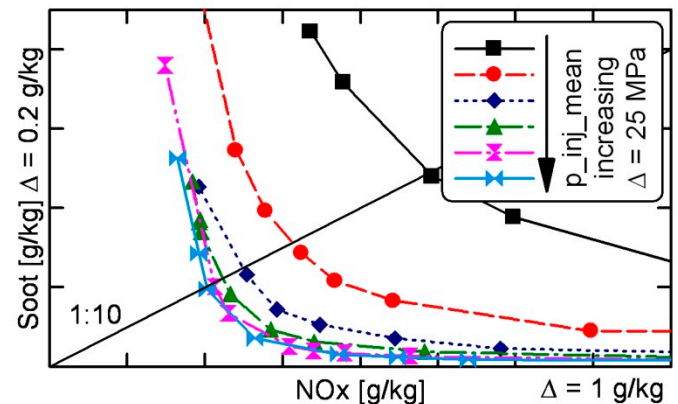


Figure 3: Soot over NOx emissions, EGR sweeps of injection pressure variation; $n = 2000$ rpm, IMEP = 1.2 MPa, BC4*, SOE = 1 °CA aTDC.

RESULTS AND DISCUSSION

The data shown in this section are based on EGR sweeps with different injection pressures under varied boost conditions. For each injection pressure variation the achieved intersections of the sweeps with the 1:10 line were determined and combined to a curve dependent on load and boost pressure. These curves therefore show the effect of mean injection pressure increase on soot and NOx emissions at the ratio of 1:10. Note, that the EGR sweeps were not approximated by a curve before determining the 1:10 values using a linear interpolation. Thus, depending on the distance between the two measurement points before and after the 1:10 line, these values could slightly differ from the real curve values. Once the intersections with the 1:10 line were determined, other parameters such as EGR rate, $\alpha_{MFB_x\%}$ and relative air-fuel ratio could be calculated for these specific NOx and respective soot values. Start of energizing (SOE) was kept constant throughout an injection pressure variation realizing a maximum deviation of the start of injection (SOI) of ± 0.15 °CA.