

Next Improvement Potentials for Heavy-Duty Diesel Engine - Tailor the Fuel Injection System to the Combustion Needs

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ABSTRACT

Future diesel engine legislation Tier 4 / Stage V and EU6d demand further improvements to reduce CO₂ while keeping the already low NO_x emissions levels. For US trucks a more strict limit of 0.2 g/bhp-hr NO_x emissions need to be achieved. In this trade-off, system costs and complexity of the after-treatment are defining the constraint in which the common rail fuel injection system layout has to be defined. The increase of rail pressure was in the past the major step to control the soot emissions in view of low engine-out NO_x emissions by applying massive EGR. With the on-going development of NO_x-aftertreatment by Selective Catalytic Reduction (SCR), conversion efficiencies of up to 97% allow to reduce the EGR usage and rail pressure usage. In that context, the steepness of injection rate, the nozzle flow rate and the injection pressure are remaining parameters to control the NO_x emissions. A shallow injection rate in combination with larger nozzle flow rates is beneficial to reduce the NO_x emissions thanks to a reduced premixing of fuel with air. To study this effect, the latest solenoid injector with improved magnetic actuation is utilized. The influence of the steepness of injection rate is studied on a 6 cylinder heavy-duty Diesel engine on 3 representative part- load points of the WLTC cycle. Depending on the engine-out NO_x emissions requirements, two scenarios are considered. In case of a strong NO_x aftertreatment, the shallow injection rate steepness is beneficial for the fuel consumption. In case of less NO_x aftertreatment, high EGR rates are required and soot emissions can be controlled through the steepness of injection rate.

CITATION: Weber, J., Herrmann, O., Puts, R., Kawamura, J. et al., "Next Improvement Potentials for Heavy-Duty Diesel Engine - Tailor the Fuel Injection System to the Combustion Needs," *SAE Int. J. Engines* 10(3):2017, doi:10.4271/2017-01-0705.

INTRODUCTION

In the past, the combustion process towards clean diesel engines was most likely driven by reducing the particulate emissions. The soot (PM)-NO_x trade-off is limited by the EGR usage and the available rail pressure as presented in Fig. 1.

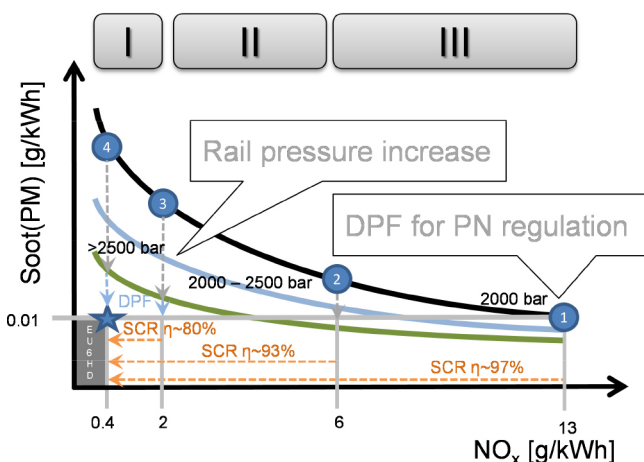


Fig. 1. Future Emissions Trade-off Scenario

The ultra-low NO_x levels < 2 g/kWh require the usage of massive EGR up to 50%. As soot emissions are increasing the high rail pressure up to 3,000 bar is required. Already such concept to countermeasure the soot emissions without compromising the fuel consumption was studied recently as presented in Fig. 2.

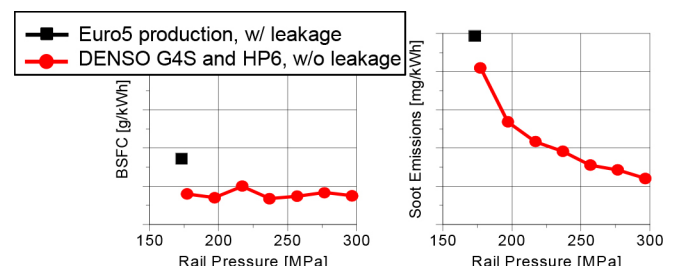


Fig. 2. Soot Reduction Potential with 300 MPa Rail Pressure [12]

In view of the highly transient cycles of WLTC and PEMS (real drive), raw engine NO_x reduction is even more challenging for off-road US EPA Tier 4 / EU Stage V and on-road EU 6d emissions levels. One possibility to countermeasure the NO_x engine out raw emissions is the NO_x aftertreatment by Selective Catalytic Reduction (SCR) which is now widely applied. In this case, the EGR system can

be simplified or even omitted to reduce the system costs. While operating such diesel engine without EGR or on very low rates, the oxygen content in the cylinder charge is sufficient to oxidize the soot emissions. Nevertheless for complying the Stage V/ EU6d emissions legislation, a Diesel Particulate Filter (DPF) is still required since the Particulate Number (PN) will be restricted. In such engine configuration, the demand for increasing the maximum rail pressure becomes less severe but requires larger nozzle flowrates which has a positive effect on the fuel consumption thanks to a shorter combustion duration. In Tab. 1 the different requirements resulting from different NO_x emissions scenarios are summarized.

Tab. 1. Requirements for the Injection System

Concept: NO _x emissions:	I < 2 g/kWh	II 2 – 6 g/kWh	III > 6 g/kWh
Target Rail Pressure	Up to 3000 bar	2500-2700 bar	<2500 bar
EGR	40-50%	20-30%	0-20%
Soot	DPF	DPF	DPF for PN (no regen.)
η_{SCR}	80%	95%	96-98%
Urea Consumption	3%	6%	8-15%
EGR	40-50% LP + HPEGR	20-30% HPEGR	0-20% Less/ w/o EGR
T/C	2 stage	1 stage	1 stage

From injection system viewpoint these various requirements need to be considered and therefore the injection system must be adapted to give the flexibility to these NO_x emissions scenarios within the CO₂ reduction constraints.

The objective of the current study is to investigate the injection rate strategy, the injection rate steepness and the influence of the nozzle hydraulic flowrate on the emissions performance and fuel consumption. In the following sections firstly possible improvements in injector technology are presented and confirmed by hydraulic measurements on a pump bench compared to G4S standard specification. Secondly the engine test results demonstrate the improvement of the beforehand mentioned objectives.

DIESEL FUEL INJECTOR TECHNOLOGY

The combustion process of the Diesel engine directly depends on the performance of the fuel injection system, particularly on the injector performance with regard to the steepness of the injection rate and minimum hydraulic interval, and the maximum rail pressure level. In the past the increase of rail pressure was the key to control the air entrainment and the premixed combustion portion. This premixing combustion increases the combustion noise but also reduces the soot emissions as the air is entrained into the fuel rich zones. The increase of injection rate steepness has a similar effect. By a fast needle opening, pressure losses by the needle seat throttle are reduced, especially in the case of Pilot injection events. The rail pressure is

utilized more efficient for the spray momentum and enhances the air entrainment of the spray. Similar to the rail pressure increase, this leads to the increase of a premixed combustion and reduces the soot emissions by entraining the air into the rich fuel areas either. Such fast injection opening and high rail pressure is required in engine concepts with high EGR usage to reduce the NO_x emissions as first objective but the combustion noise increases. To countermeasure the combustion noise, the premixed combustion therefore has to be limited even in the presence of high rail pressures. Such approach is realized by the flexibility of the injection system to increase the number of injection events and to achieve short hydraulic intervals. Already such concept were developed and tested by using Piezo servo valve technology [4,6] for passenger car applications and firstly studied by trial samples for solenoid injectors [7,8] for passenger car and heavy-duty diesel engine tests. Now this technology is furthermore improved through the on-going development of new solenoid actuators. The G4S injector functionality during start-of-injection and end-of-injection is displayed in Fig. 3.

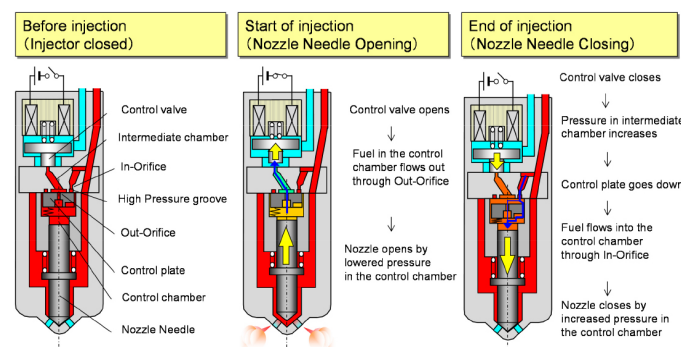


Fig. 3. Actuation Principle of the Solenoid Servo Valve of the G4S Injector

The needle controls the opening and the closing of the fuel passage to the nozzle hole. Its motion is controlled by the pressure force on top of the needle from the pressure in the control chamber and from the pressure force at the bottom of the needle from the pressure in the nozzle sac volume. By actuating the control valve, the pressure of the control chamber is released and the needle opens. At closing, the pressure chamber is refilled with fuel to build up the pressure. A small control plate separates the high and low pressure area and has the function of a three way valve. The switching is released through the movement of the control plate. At the start of the actuation, the control plate remains at its position. At the end of the actuation to fill-up the fluid in the pressure chamber quickly.

The pressure decrease is defined by the flow rate characteristics of the out-orifice which is part of the control plate. It is therefore the flow characteristics of the out-orifice to define the needle opening speed. To improve the needle opening speed, this flowrate characteristic of the out-orifice was increased while keeping the flowrate of the inlet orifice constantly.

In addition the intermediate chamber volume and sub-orifice diameter was increased to avoid downstream a throttling but causes an increase of the pressure forces on the control valve. As a consequence the

spring set force is increased against which the magnetic force acts during actuation. Therefore the magnetic force had to be increased by changing the solenoid material. As a result of these improvement steps the injection rate steepness could be increased while the closing speed controlled through the inlet orifice flowrate and motion of the three way valve was kept same.

By the increase of the out-orifice flowrate a shorter hydraulic interval could be achieved simultaneously. The standard solenoid injector for passenger car applications achieves in best case a hydraulic interval of 200 μ s. This interval is resulting from the movement of the control plate at the end of the actuation. Once the control valve closes the sub out-orifice, the control plate movement starts and the pressure in the intermediate chamber and control chamber is built up again by the fuel feed from the in-orifice throttle. Both chambers are pressure balanced by the out-orifice throttle. A small support spring pushes back the control plate to the original closed position when the pressure in the control chamber is restored. Now the injector can be actuated for the next injection event. As described the hydraulic interval mostly is defined by the time required for the lift of the control plate. This duration is shortened if the out-orifice flowrate is increased as the time to pressure balance the intermediate and the control chamber is shortened.

HYDRAULIC INJECTOR PERFORMANCE

The hydraulic performance was investigated on a pump bench. The supplier information and bench conditions are summarized in [Tab. 2](#).

Tab. 2. Pump bench for hydraulic performance assessment

Pump bench	Moehwald CA4000
Injection rate measurement	Injection Analyzer
Fuel: Test-oil	Shell Divinol CAF ISO 4133
Fuel temperature	40°C

[Fig. 4](#) shows the resulting performance of the injector equipped with different out orifices. As in the previous chapter discussed the steepness of injection rate was increased thanks to varying the out-orifice flowrate while keeping the same closing steepness of the injection rate.

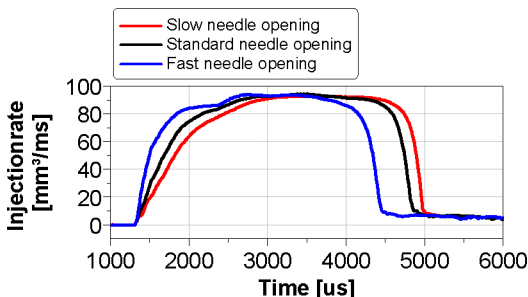


Fig. 4. Measured Injection Rate: HFR=1550 cc/min, Prail=120 MPa, Q=260 mm³/str

In [Fig. 5](#), the resulting hydraulic interval is presented. For analysing the minimum hydraulic interval, 3 consecutive pilot injections are applied followed by a main injection. Within the standard G4S specification, the min. hydraulic interval of 250 μ s is achieved. With increased out orifice diameters, a much shorter hydraulic interval of 100 μ s is feasible as shown on the pump bench. Such kind of short hydraulic intervals are in the passenger car diesel engine utilized to control the rate of heat release at part load conditions.

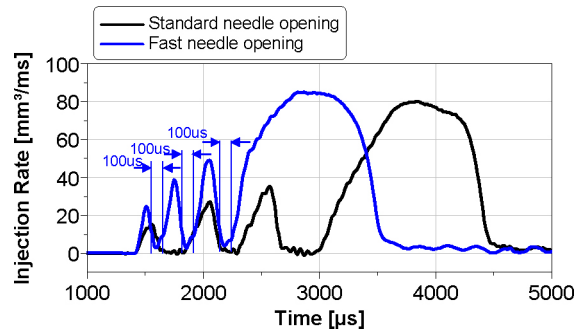


Fig. 5. Triple Pilot Multiple Injection Strategy, Prail = 120 MPa [10]

Overall the injector's hydraulic performance measured on the pump bench demonstrates that the 3-way-valve allows the adaption of the out-orifice flowrate independent of the in-orifice throttle by utilising a new solenoid actuator.

DIESEL ENGINE TESTING

The performance of the improved solenoid injector is evaluated on a heavy-duty EU6 Diesel engine. Details of the engine are given in [Tab. 3](#).

Tab. 3. Properties of the heavy-duty EU6 Diesel Engine

Displacement Volume	12L
No. of cylinders	6
Rated Power	353 kW
Max. Torque	2500 Nm
Peak Firing Pressure	190 bar
Compression Ratio	17
Turbo Charger	1 stage VNT
EGR	Cooled HP-EGR

The engine was equipped with the high-pressure fuel pump HP6 (2 cylinders, 3 lobes) with a max. rail pressure of 300 MPa. All cylinders are instrumented by pressure transducers (KISTLER 6053CCsp) to analyze the combustion cycle individually. For engine testing, five operating points of the world harmonized steady state test cycle (WHSC) were selected as shown in [Fig. 6](#). These points cover the area of low part-load, mid part-load and full-load engine operating conditions. NO_x emissions are controlled by utilizing the high-pressure exhaust gas recirculation (HP-EGR).