

Injector Fouling and Its Impact on Engine Emissions and Spray Characteristics in Gasoline Direct Injection Engines

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ABSTRACT

In Gasoline Direct Injection engines, direct exposure of the injector to the flame can cause combustion products to accumulate on the nozzle, which can result in increased particulate emissions. This research observes the impact of injector fouling on particulate emissions and the associated injector spray pattern and shows how both can be reversed by utilising fuel detergency. For this purpose multi-hole injectors were deliberately fouled in a four-cylinder test engine with two different base fuels. During a four hour injector fouling cycle particulate numbers (PN) increased by up to two orders of magnitude. The drift could be reversed by switching to a fuel blend that contained a detergent additive. In addition, it was possible to completely avoid any PN increase, when the detergent containing fuel was used from the beginning of the test. Microscopy showed that increased injector fouling coincided with increased particulate emissions. Based on these results a selection of the injectors was installed in a laboratory injection chamber and the spray patterns were investigated with a high speed camera. Injectors corresponding to the largest PN drift produced the thinnest spray jets with the deepest penetration. These factors amplify the risk of wall wetting and provide an explanation for the increase of PN. The positive effect of the detergent was also reflected in the spray pattern analysis, which illustrates the potential benefits of such fuel additives.

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INTRODUCTION

Unlike in port fuel injection (PFI) systems, gasoline direct injection (GDI) engines deliver fuel directly into the combustion chamber. This opens up the possibility of new injection strategies and can improve efficiency and fuel consumption, especially under partial load [1, 2, 3]. An additional advantage is the so-called stratified charge operation, which allows adjustment of the injection and ignition timings of the engine, to further optimise engine operation [4]. In addition, the charge cooling effect observed during fuel injection allows for the use of higher compression ratios, with reduced risk of knocking. These benefits are gaining more attention with the continuing trend of downsizing and super- or turbocharging of engines, due to increasingly strict emissions regulations [5, 6, 7]. Additional benefits of GDI are the option to completely shut-off fuel supply while coasting [8] and improved cold start behaviour compared to PFI engines [9,10].

One major challenge of the GDI concept, however, is the potential for increased formation of particulate emissions. Recent emissions legislations such as EURO 6 in Europe and LEV III (Low Emissions

Vehicle Program) in the US pose very strict limitations on particle number and mass (PM) concentrations [11,12]. Methods to control particulate emissions from GDI engines are therefore important.

Introduction of an injector into the combustion chamber of a GDI engine results in direct exposure of the injector tip to the harsh conditions occurring during combustion. Because the flame directly interacts with the nozzle, combustion products can be deposited on the surface and inside cavities. This is usually referred to as injector fouling or coking and can have an adverse effect on fuel consumption, engine performance and both hydrocarbons and particulate emissions. Formation of deposits on the injector tip can result in alteration of spray pattern leading to increased spray impingement and therefore increased particulate emissions [13, 14, 15, 16].

The phenomenon of injector fouling and how it is influenced by fuel and engine parameters was the focus of a number of studies. Aradi *et al.* [17] investigated different fuel blends under several conditions in a four cylinder engine. They used rich fuel-to-air ratios in order to promote fouling. By measuring the temperature of the injector tip with an internal sensor, they saw that with increasing temperature

inside the engine the level of fouling also increased and the injector performance, measured by the flow loss, decreased. By changing the composition of the fuel blend and increasing its overall volatility, fouling was also promoted, as was increasing the olefin content. In later work of Aradi *et al.* [18,19] they showed that the formation of injector deposits can be controlled by the use of fuel additives. Depending on the detergent type, they were able to reverse or avoid any injector flow loss caused by fouling. Analogous effects of fuel additives were also reported by other groups [13,14,20,21]. These publications all show a decrease of injector performance caused by fouling and illustrate potential benefits of fuel additives. The mechanism behind the build-up and removal of deposits with detergents is complex and not in the scope of this investigation. Detailed information on this topic can however be found in the literature. [18,19,22, 23, 24]

One pressing issue associated with injector fouling is the increase of engine out emissions, which is a concern for direct injection systems. Arters *et al.* [23] performed vehicle and laboratory tests. During their experiments they saw a clear correlation between the level of injector fouling and the increase of particulate emissions. In the worst case particle mass was three-fold compared to the clean system. They also reported a negative effect on hydrocarbons, carbon monoxide and NO_x . By switching the engines to a detergent containing fuel they were able to reverse these changes. Research performed by Carlislei *et al.* [25] further demonstrated the link between injector fouling and increase in engine-out emissions. Both hydrocarbon and soot levels increased with the build-up of deposits. Under homogeneous operating conditions, increased T90 of the fuel and increased engine load promoted the fouling process under their conditions. When switching to stratified injection, a levelling out of the engine emissions was observed, most likely due to stabilisation of the fouling. Overall, the use of detergents was beneficial in most cases.

A possible reason for the increase in particulate numbers (PN) due to injector fouling, aside from the change in injector performance and fuel impingement, was given by Berndorfer *et al.* [26]. Their tests in a single-cylinder engine, equipped with an endoscope, showed the formation of a rich diffusion flame stabilizing on the injector tip during the combustion phase. The residue build-up on the surface acted like a sponge absorbing fuel during injection, which upon combustion resulted in a rich flame. The integral intensity of the flame was used as a measure of its extent and was directly correlated to the particulate numbers measured at the exhaust. Once they removed the deposits from the surface mechanically, they no longer observed these issues. Additionally, by increasing the injection pressure they were able to reduce the size of the flame and eventually completely avoid it. With this research, Berndorfer *et al.* provide additional explanation for the increase of particulate emissions, aside from the increased probability of wall wetting and pool fires. This phenomenon was also studied by Wang *et al.* [27].

The research presented in this paper is a collaborative effort. In the first phase of the study, engine tests were run with and without detergent additives to observe the effect on injector tip deposit

formation and particulate emissions. Different fuel blends (see Table 1) were used in an engine dirty-up cycle to deliberately foul injectors. Under engine operation some of these injectors were subsequently exposed to a fuel blend with a detergent additive. For all cases engine performance and emissions were monitored. In the second phase of the experiment both fouled and cleaned-up injectors were taken from the engine tests and analysed visually with a microscope before conducting spray pattern analysis. The intention was to correlate potential changes in the spray pattern to the extent of the fouling and emissions drift.

EXPERIMENTAL SETUP AND PROCEDURES

Engine Setup, Fuel Blends and Test Cycles

All engine tests were performed in a four cylinder VW EA211 GDI engine. The injectors (VW part number 04E 906 036 E) have a multi-hole design with five nozzles located around the centre of the tip. A photograph of one unused injector can be seen in Figure 1, with a close-up view of the injector tip and the numbering of each nozzle. Each of the holes has a different orientation relative to the injector normal, with an axis symmetric design between hole 1, 2 and 4, 5. These pairs create sprays that are pointed outwards at an angle. Hole 3 is orientated almost parallel to the injector normal. The injector is mounted inside the engine with hole #3 pointed towards the spark plug at the top and the 4 other holes targeted downwards. All nozzles have a diameter of about 200 μm and a static flow rate of 6.44 mg/ms at a rail pressure of 100 bar, 7.98 mg/ms at 150 bar and 9.23 mg/ms at 200 bar, based on the measurements of one injector.

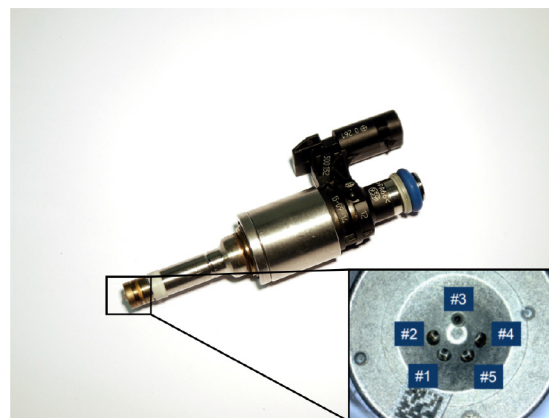


Figure 1. Multi-hole injector used in the EA211 engines with a close-up view of the injector tip and numbering of each nozzle.

Table 1. Overview of the fuel blends used during the engine cycles.

Fuel Blend	Detergent Content	Aromatic Content, vol%	T90, °C
A	0	40	134.3
B	Low	40	134.3
C	High	40	134.3
D	0	20	173.9

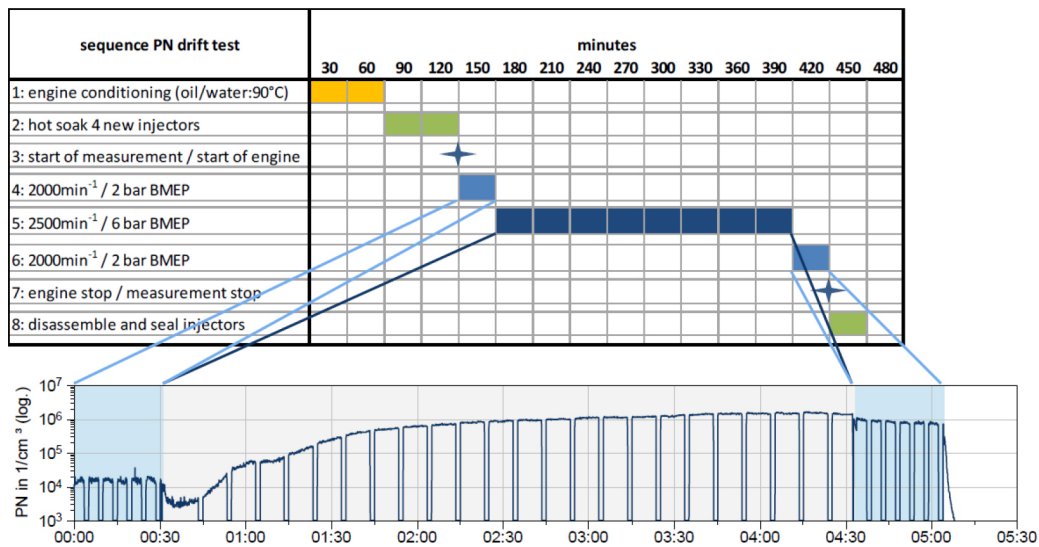


Figure 2. Engine parameters used during the injector fouling phase.

In total 32 injectors, in sets of four, were used in the engines with four different fuel blends. All injectors were selected at random from the manufacturing line but originated from same batch. An overview of the different blends is shown in Table 1. Two base fuels (Fuel A and Fuel D) were used with high and low aromatic content as well as different distillation behaviour (T90). By adding a detergent to Fuel A at a low and high concentration Fuel B and Fuel C were created, respectively.

The engine experiment was divided into two different stages, an injector fouling phase and an injector clean-up phase. During the former injectors were exposed to Fuels A, B, C and D. Each run produced a set of four injectors for each fuel blend. This yielded in a total number of 16 fouled injectors. The engine parameters and an overview of the injector fouling phase are shown in Figure 2.

After conditioning the engine and hot soaking of the injectors for 60 minutes in total, the fouling cycle and measurements were started. During the first 30 minutes the engine was operated at 2000 rpm at 2 bar BMEP and a rail pressure of 100 bar in order to create a reference point for engine performance and emission levels. After this, the engine speed was increased to 2500 rpm at 6 bar BMEP and 200 bar rail pressure. This phase lasted four hours. Finally, the operating conditions were changed back to 2000 rpm, 2 bar BMEP and 100 bar for 30 minutes. The start of ignition (SOI) was set to 260 °CA for the lower and 285 °CA for the higher operating point. After finalising the engine experiments the respective injectors were removed from the engines and sealed for later analysis.

The injector fouling cycle was repeated for a second run with sets of fresh injectors following the same procedure. Upon completion, however, the injectors remained in the engine and were used in the injector clean-up cycle. This was done by switching to Fuel C, which contained the detergent at high concentration. The operating procedure of the clean-up cycle followed that of the fouling cycle, with the exception that the experiment was run for eight hours instead of four. During both cycles, engine emissions in the form of particle number and particle mass were analysed with an AVL 489 particulate counter. The dilution ratios were adjusted to accommodate for an

expected increase of 2000-3500 in particle numbers during the fouling stage. Additionally, hydrocarbon, CO and NO_x emissions were recorded. The fuel trim was also monitored.

Upon completion of the engine experiments all injectors were sealed and the extent of their deposits were analysed by microscopy and compared to the emissions drift. After this, all injectors were installed in the laboratory injection chamber to record the spray patterns.

Injection Chamber and Recording Setup

The 32 injectors, 16 from the injector fouling cycle and 16 from the clean-up cycle were installed in a laboratory injection chamber in order to record and correlate the spray pattern changes to the fouling level and emissions drift observed during the engine experiments. An overview of the setup is shown in Figure 3.

The setup comprised of a cube-like injection chamber with a length of 300 mm, sealed by Perspex windows on the sides. The frame was mounted on a steel base plate with ports connected to the laboratory extraction system in order to remove fumes created by evaporating fuel. The rear of the chamber was closed with an aluminium plate, which formed the support for the transparent impingement plate, with its overall dimensions of 150x170 mm² and ports to connect a heat exchanger (Julabo ME-12, Julabo UK LTD), which was used to temperature control the surface. Top and bottom of the frame were sealed with quartz glass windows (UQG Optics). The thickness of the entire assembly was 20 mm. The liquid used in the heat exchangers was silicone oil (Rhodorsil 47V100, Bentley Advanced Materials).

The top of the injection chamber was closed with another aluminium panel, which formed the structural support of the injector mount. The orientation of the injector in this assembly was perpendicular to the impingement plate at a distance of 45 mm. The fuel delivery was achieved with a pneumatically actuated pump (MSF22, Maximator GmbH). In order to minimise pressure fluctuations caused by the injector needle, a 155 ml pressure cylinder (SvageLok Ltd.) was connected to the pressure side of the pump. During measurements, the rail pressure was set to 160 bar and a 1.25 ms injection width was