

Assessment of the Influence of GDI Injection System Parameters on Soot Emission and Combustion Stability through a Numerical and Experimental Approach

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ABSTRACT

The next steps of the current European and US legislation, EURO 6c and LEV III, and the incoming new test cycles will impose more severe restrictions on pollutant emissions for Gasoline Direct Injection (GDI) engines. In particular, soot emission limits will represent a challenge for the development of this kind of engine concept, if injection and after-treatment systems costs are to be minimized at the same time. The paper illustrates the results obtained by means of a numerical and experimental approach, in terms of soot emissions and combustion stability assessment and control, especially during catalyst-heating conditions, where the main soot quantity in the test cycle is produced. A number of injector configurations has been designed by means of a CAD geometrical analysis, considering the main effects of the spray target on wall impingement. The numerical CFD simulation has helped the definition of the injection system and of its control settings for a given operating condition, in terms of start of injection, injection pressure and number of pulses per stroke. Engine test bench experiments have finally been used to validate the numerical results, and to further optimize the injection system calibration in order to minimize soot emissions, while respecting combustion stability constraints. The main results are presented in the paper.

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INTRODUCTION

The emission legislation is constantly changing in all the regions of the world [1]: the global development of CO₂ fleet targets for new passenger cars and light trucks is one of the main drivers for the downsizing and downspeeding of internal combustion engines. For Spark Ignition (SI) engines, the reduction of engine displacement (downsizing) and of engine speed (downspeeding) leads to a significant decrease of fuel consumption, while gasoline direct injection, along with turbocharging, is commonly used to maintain or even increase engine performance. Considering the future emission legislation, the most challenging limit for GDI engines is the particulate matter emission. In addition to the conventional toxic emission species like CO, NO_x and HCs, and the particle mass (PM), Euro 6 emission standards introduce a limit on the particulate number (PN) emissions for GDI engine-powered vehicles starting from September 2014 (Euro 6b), and a much more stringent limitation (Euro 6c) will follow in September 2017 (see [Table 1](#)).

Table 1. European and US legislation, particle emissions

		GDI PC-PDV PARTICULATE MATTER REGULATION TREND									
		2013	2014	2015	2016	2017	2018	2019	2020	2021...	... 2025
Europe	Level	Euro 5	Euro 6b			Euro6c			Euro 7 ?		
	PM	5 mg/km	4.5 mg/km								
	PN	-	6 x 10 ¹² #/km				6 x 10 ¹¹ #/km				
		CO2 140 g/km				CO2 120 g/km			CO2 95		
	Cycle	NEDC TEST CYCLE					WLTC + RDE TEST CYCLE				
USA	Level	LEV 2	LEV 3			LEV 3			LEV 3	LEV 3	
	PM	10 mg/mile	6 mg/mile			3 mg/mile			1 mg/mile		
	Cycle	FTP									

Many benchmarks of Euro 4 and Euro 5 GDI engine-powered vehicles in the New European Driving Cycle (NEDC) demonstrate that today's particulate mass emissions are already below the future limit. Considering the particles number legislation, the limit is much more challenging. To fulfil the emission legislation, a further optimization of the combustion system of GDI engines is needed, in particular focusing on injection system capabilities and calibration. The introduction of new driving cycles is under discussion: Worldwide Harmonized Light-Duty Vehicles Test Procedure (WLTP) and Real Driving Emission (RDE) test procedures are in fact being

considered as possible new standards. Worldwide Harmonized Light-Duty Test Cycle (WLTC) will be characterized by higher engine loads and transient conditions, so that meeting the particulate number limit could become more challenging. Besides, Real Driving Emission procedure, to evaluate the engine emissions in a wider engine operating range and outside homologation cycle conditions, is under definition. USA LEV III legislation is also focused on particulate emission. LEV III and Tier 3 regulations reduce the PM mass standard to 3 mg/mi, to phase-in over the 2017-2021 model years, with the LEV III level dropping to 1 mg/mi in 2025. These emission levels have to be met over 150 kmiles of vehicle operation. Considering the emission cycle, WLTC and FTP75 are quite comparable in terms of acceleration profiles, while the maximum speed is lower for the US cycle. In the United States, adopting PN standards is currently under debate, while Chinese emission legislation is moving towards very stringent limits: in particular, Beijing VI is oriented towards severe limitations as the US ones, with the strict limitation of the European legislation on particle number.

As it is well known, the cold start phase is the most critical for emissions production. Actually, the formation mechanisms of the various emission species are very different, and the role of the after-treatment system has to be carefully considered. CO, HC and NO_x emissions dramatically decrease after the catalyst lights off, so that specific strategies for fast catalyst light-off are applied immediately after engine start. In order to reduce light-off time, charge stratification available in GDI engines is a very important feature: the ignitability of the mixture near the spark plug allows using delayed spark timing while maintaining a stable combustion, so that the exhaust enthalpy flow can be significantly increased [2, 3, 4, 5]. On the other hand, particle emissions are strongly increased under cold conditions due to wall film formation and mixture inhomogeneity. After-treatment devices like Gasoline Particulate Filter (GPF) to reduce particle emission are available, and they do not need a light-off time to be activated; anyways, the adoption of the GPF leads to additional costs and performance issues that have to be avoided if possible.

Considering the injection system, many layouts are available for GDI engines [6], with different behaviors in terms of stratification capability and soot emissions production [7]. Side-mount multi-hole systems as the one under analysis are widely used in GDI engines: they guarantee a great flexibility in terms of holes number and direction, and they can be optimized focusing on different objectives [5, 8, 9, 10, 11]. Usually, the main aim of a spray targeting activity is the avoidance of wall-wetting, and a good mixing with air: besides, a good behavior in terms of mixture stratification is required for catalyst heating strategies. The characteristics of the injector in terms of spray atomization and hydraulics and its control are very important factors for improving engine performance and fulfilling emissions legislation.

In this work, an evaluation of the potential improvement attainable by modifying the injection system and its control laws has been performed, focusing on particle emissions and combustion stability. A GDI engine has been tested in its Euro 4 baseline configuration, after having modified injector characteristics, maximum system fuel pressure and injection control strategies. The experimental tests

conducted during this project have been designed to demonstrate the correlation between modified injector characteristics and corresponding reduction of smoke emissions. Smoke emissions have been qualitatively assessed with one of the most robust and widely adopted opacity indexes (FSN, Filter Smoke Number [12]), to validate a fast and efficient methodology for improved injectors design, based on geometric CAD analysis, CFD simulations, and fast and low-cost experimental tests to be conducted directly on the engine test bench. The work presented in this paper represents therefore the first part of a longer and more complex activity. Once the proposed approach has confirmed the robustness of the CAD/CFD-based analysis, a further (and finer) design phase will follow, and the complete system calibration will then take place using much more accurate and complex emissions measurement devices (particle counter, soot concentration sensors, ...), on the test bench and finally on the roll dynamometer.

EXPERIMENTAL SETUP

The engine used for this study is a 4 cylinder 1.4L turbocharged EURO 4 direct injection engine, for passenger car applications, commercially available in 2007. Its technical data are summarized in [Table 2](#).

Table 2. Technical data of the engine used during experimental tests

Displacement	1390 cm ³
Stroke	75.6 mm
Bore	76.5 mm
Compression ratio	10:1
Number of Valves	4 per cylinder
Max Torque	220 Nm @ 1500-4000 rpm
Max Power	103 kW @ 6000 rpm
Injection System	Direct Injection (up to 200 bar in the Magneti Marelli variant)

The series production engine is provided with a Magneti Marelli side-mounted multi-hole injector and it has been equipped for this work with a complete Magneti Marelli GDI system (pump, injectors and ECU). The new Magneti Marelli GDI injection system has been designed to operate to up to 200 bar of rail pressure, and the control functions implemented in the new ECU allow much greater injection flexibility, especially in terms of increased number of injections per cycle and of their accurate phasing. This system has been used to test different Magneti Marelli injectors and to verify their performance with respect to CAD/CFD-based analysis, in particular in terms of combustion stability and smoke emissions.

During the experiments, carried out on the test bench under steady state conditions, in-cylinder pressure has been measured for each cylinder by means of Kistler 6052A piezoelectric sensors, at high frequency (200 kHz), whereas the most important ECU variables have been recorded at low frequency (50 Hz). Smoke emissions have been measured by means of AVL 415s Smoke Meter, and reported in terms of Filter Smoke Number (FSN). The engine operating points that have been taken into consideration during the tests are summarized in the following [Table 3](#).

Table 3. Engine operating points analyzed during the tests

Engine Speed	BMEP
1200 rpm	idle – catalyst heating
1500 rpm	9 bar

Particular focus has been placed on catalyst heating conditions, due to the challenging optimization in terms of particulate emissions, fuel consumption and exhaust gas enthalpy. The operating condition 1500×9 (1500 rpm and 9 bar of BMEP) is on the other hand particularly relevant since it represents the maximum torque levels achieved during NEDC, at relatively low speed. It is therefore extremely interesting since it allows isolating the effects of modified geometrical targets due to relatively low in-cylinder turbulence levels and large injected mass.

SPRAY TARGET ANALYSIS

The spray target analysis has been carried out considering consolidated technologies (e.g. multi-hole injectors with equal hole diameters, EDM technique) with the aim of identifying performance limits of well-known (and thus low-cost) technologies. For this study, the Magneti Marelli multi-hole injector of the commercially available Euro4 engine has been considered as a baseline for the spray targeting process. The first step of the spray targeting activity consists in a geometrical analysis of the fitting between the combustion chamber and the baseline side-mounted injector, neglecting air motion, in-cylinder pressure and temperature conditions. Since the study is focused on the effects produced on smoke emissions by different spray patterns, piston and intake valves impingement needs to be accurately controlled, according to Magneti Marelli experience. Consequently, the modification process of the baseline spray pattern followed the guidelines of strongly limiting impingement on intake valves, increasing the jet-to-piston available distance and, in general, better employing the available volume inside the combustion chamber through a wider spray and a reduced jet-to-jet interaction. Nevertheless, it must be pointed out that, for GDI spray, a trade-off solution needs to be chosen, as the reduction of wall impingement on each wall is not achievable in most cases. Moreover, small engine bore, intake valves maximum lift value and intake valves-injector tip relative positioning are key aspects in determining such compromise solution.

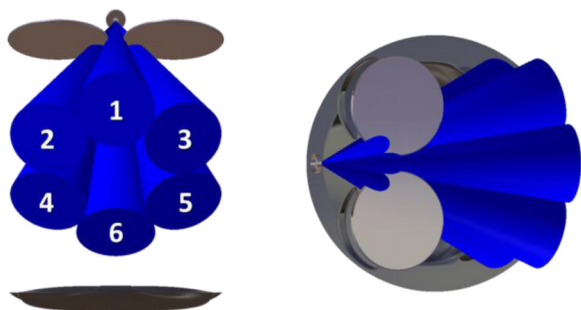


Figure 1. Baseline spray pattern front and top views

According to the above-described guidelines, the Magneti Marelli geometrical spray targeting model pointed out some possible issues of the baseline configuration, in terms of wall wetting. In particular,

under low speed and low/medium load conditions, considerable intake valves and piston impingement are expected. This aspect becomes clearly visible when evaluating CAD spray images (Figure 1): jets 2 and 3 show a considerable interference with intake valves at maximum lift and in proximity to such condition, while the inclination of jets 4, 5 and 6 tends to direct them towards the piston. In order to accurately assess the risk of piston and liner impingement, but still ignoring, at this stage, air pressure, temperature and motion conditions, the jet-to-wall available distances have been analyzed as a function of crank angle for each jet, neglecting the possibility of intake valves impingement or, in other words, under the hypothesis of closed intake valves. Such assumption is needed in order to focus the analysis on spray behavior in terms of piston and liner wetting, whereas the information on intake valves wetting risk is already determined through the above-mentioned observation of the CAD spray images. The result of this evaluation is the graph shown in Figure 2, which displays the normalized distance that each jet can travel before hitting the piston (supposed to be flat-head) or the liner, as a function of crank angle (or piston) position. As it can be seen in the graph, short jet-to-piston distances are available for jets 4, 5 and 6 up to about 100 CAdeg before combustion TDC (or, equally, from 100 CAdeg after overlap TDC). Maximum distance values are instead mostly limited by liner impingement, which is typically associated to oil dilution problems (not analyzed in this study) rather than smoke emissions.

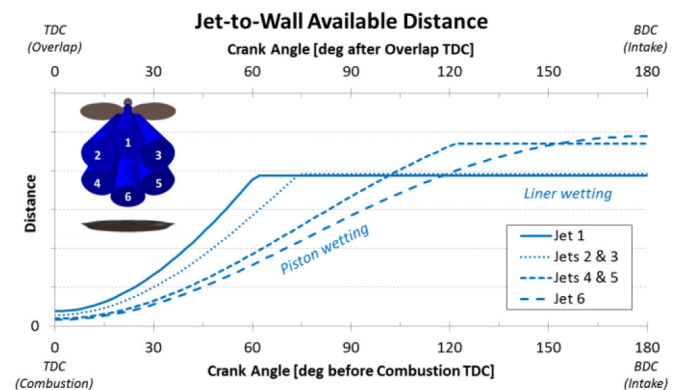


Figure 2. Available jet distances for baseline spray pattern

With the aim of improving the spray behavior in terms of wall wetting, the geometrical spray targeting model was employed to rapidly evaluate a number of different configurations through a process of variation of spray pattern geometrical parameters and analysis of model response. Such geometrical analysis showed a 5-hole pattern as the most promising one for the engine under study (Figure 3).

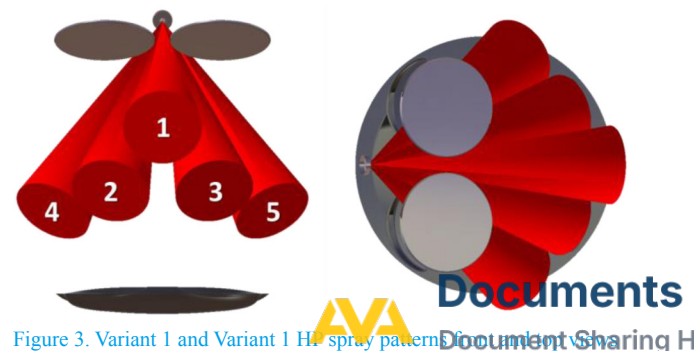


Figure 3. Variant 1 and Variant 1 H spray pattern front and top views

As shown in Figure 3, in injector Variant 1 jets have been diverted from intake valves, strongly limiting their impingement, which also depends on actual jet cone angle and thus on air and injection conditions. Moreover, the close-to-piston jets characterizing the baseline injector are not visible anymore, as a consequence of the removal of baseline jet 6 and the opening of baseline jets 4 and 5. These modifications have produced an improvement in terms of jet-to-piston available distance, reducing the low-distance angular range to about 70° before combustion TDC (or from 70° after overlap TDC), as shown in Figure 4.

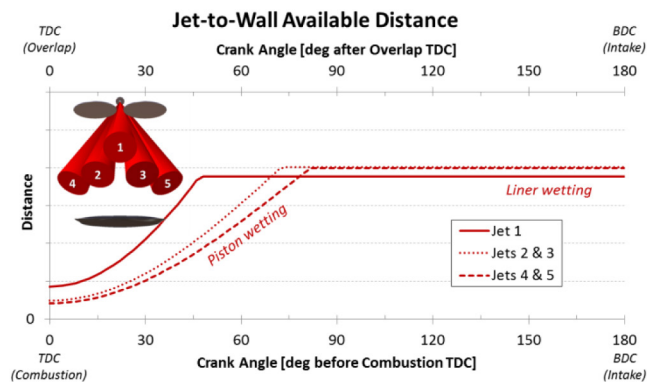


Figure 4. Available jet distances for Variant 1 and Variant 1 HP spray patterns

Further, the reduction/avoidance of intake valve interference is achieved lowering baseline jets 2 and 3 towards the piston, which is expected to slightly increase the piston wetting risk when SOI occurs in the low distance-to-piston angular range, e.g. under high load conditions and when a compression injection is performed. Nevertheless, the limitation/avoidance of intake valves wetting is expected to produce, under such conditions, a more significant benefit with respect to the penalties introduced by the slight increase on piston wetting. Finally, as a result of jets orientation modifications, variant 1 is expected to produce higher liner wetting, which is directly related to oil dilution, as mentioned before. However, lower priority has been assigned to this aspect in the project, by simply considering an upper oil dilution limit not to be exceeded.

The choice of a 5-hole pattern determines an increase of holes diameter if static flow rate is to be maintained at the baseline injector level, which may worsen spray atomization. An overall improvement can be reached through the use of a lower static flow rate injector, combined with a higher injection pressure level (Variant 1 HP, see Table 4). The reduction of the static flow rate, made possible by maximum injection pressure increase from 150 to 200 bar, allowed the reduction of Variant 1 HP holes diameter to the same value of the Baseline injector. The effects of the higher injection pressure on spray penetration were controlled through the hole length-to-hole diameter ratio.

Beyond the effects in terms of wall wetting, the use of a wider 5 jet spray pattern is also expected to provide a better charge homogenization. With this regard, Variant 1 jets orientation have been chosen taking also into account jet-to-jet interaction, which is likely to increase switching from a 6-hole to a 5-hole pattern, due to the increase in cone angles emerging from the larger hole diameters (needed to maintain the static flow rate at the baseline level), and to

the lower hole length-to-diameter ratio (required to control the higher jet penetration caused by increased per-hole static flow rate). The reduction of the static flow rate for Variant 1 HP should provide a further improvement also in terms of jet-to-jet interaction, thanks to smaller hole diameters and thus reduced cone angles.

Table 4. Injector characteristics

Injector	Position	Number of Holes	Maximum Injection Pressure	Static Flow Rate
Baseline	Side	6	150 bar	Reference value
Variant 1	Side	5	150 bar	=
Variant 1 HP	Side	5	200 bar	-20%

Table 5 summarizes the qualitative behavior expected for each injector at the geometrical analysis stage. It must be pointed out that the geometrical spray targeting model response becomes closer to real spray behavior as the considered engine operating condition is characterized by lower air velocity. However, even at this stage, some preliminary estimations can be made on spray behavior under higher air velocity conditions, simply considering that the main effect on spray direction should be provided by tumble motion, which is expected to bend downwards the spray jets with respect to their original directions. For some spray jets, this may lead to an alteration from liner to piston wetting, with respect to the behavior forecasted by the geometrical spray targeting model.

Table 5. Expected qualitative injectors behavior

		Baseline	Variant 1	Variant 1 HP
Liner wetting	Low speed	Medium	↑↑	↑
	Med./high speed	Low-medium	≈	↓
Intake Valves wetting	Low speed	High	↓↓	↓↓
	Med./high speed	Low	↓	↓
Piston wetting*	Low speed	Medium-high	↓	↓
	Med./high speed	Medium-high	↑	≈
Jet-to-jet interaction		Low	≈	↓

The effectiveness of the geometrical spray targeting methodology has been then evaluated via 3D CFD simulation in a virtual engine model through the estimation of wall impingement, and via experimental engine tests by means of FSN measurements and comparisons.