ABSTRACT

Since the introduction of the EURO 5 emission legislation particulate matter emissions are no longer only a concern in the development of Diesel engine powertrains. In addition to particulate mass (PM) requirements the new European legislation will also foresee the implementation of a particulate number (PN) requirement for all spark ignition (SI) vehicles with the introduction of EURO 6. Measurements with state of the art gasoline engine powered vehicles show that conventional MPFI engines are already below the future proposed limits while gasoline engines with direct injection are above these limits and will require additional development efforts.

This paper discusses both fuel system component requirements as well as control strategies in support of reducing particulate emissions. On the component side, mixture formation in regard to evaporation rate and penetration is a key factor. On the control side, injection timing and injection splitting are important parameters, especially under cold catalyst heating conditions. Encouraging test results show that significant improvements in regard to particulate matter emissions can be made. For the particulate mass emission a value significantly lower than the proposed limit can be achieved, while the proposed particulate number limit is significantly more challenging. The demonstrated vehicle results detailed below show that the proposed EURO 6 targets can be met by gasoline engines with direct injection by careful further optimization of the involved hardware and calibration without adding an additional after-treatment system.

INTRODUCTION

Improving the overall efficiency of gasoline powered passenger cars is the main focus of today's engine combustion development work. A remarkable number of different gasoline engine concepts are already in production or under development depending on market specific requirements. No common gasoline engine technology and architecture is identified yet. Gasoline engines operated under stoichiometric conditions can fulfill all current global exhaust gas emission requirements due to the very effective exhaust after treatment systems based on a three-way catalyst (TWC). For lean stratified engines the NOx emissions will remain a challenge both from the point of view of conversion efficiency over lifetime and overall system cost. One of the major current gasoline engine trends is to downsize and boost in combination with direct gasoline injection. These measures lead to the usage of higher load points with improved combustion efficiency, while maintaining the drivability and NVH characteristics typical for gasoline engines.

The new emission standard EURO 5a introduced in September 2009 defines the particulate matter mass emission limits for Diesel and for the first time also for gasoline fueled engines with direct injection (GDi). The more stringent emission limits of EURO 5a compared to EURO 4 have led to the introduction of particulate filters in many Diesel applications while for gasoline engines a refined calibration is sufficient to be significantly below the limit. With the introduction of EURO 5b in September 2011, a limit for particulate number for Diesel powertrains will be introduced [1]. Although under discussion it is expected that the same limit will be introduced for gasoline powered vehicles with EURO 6 as shown in Table 1.
Looking at particulate mass and number emissions measured on production vehicles with EURO 4 calibration in Figure 1, it becomes obvious that the discussed particulate number targets will be a significant challenge for gasoline engines with direct injection - independently from homogenous or stratified operation modes - while the particulate mass limits can be easily achieved.

For port fueled injection engines (MPFI) the PM is already more than an order of magnitude lower, while the PN limit will require some refined calibration work to be sufficiently below the target value. Figure 1 also reflects a weak correlation between PM and PN for gasoline powered vehicles for very low emissions.

Looking to the time-resolved particulate emissions of a EURO 4 calibrated vehicle with gasoline direct injection during the European Driving Cycle (NEDC) in Figure 2, several conclusions can be derived. A major amount of particulate mass and number is emitted in the early phase when the engine is cold and the catalyst system is still not fully operational. All the transient load and speed steps cause a steep increase of particulate emissions.

Since particulate emissions were not a priority in the calibration process previous to EURO 5, it can be assumed that a significant reduction will be achieved by optimization of engine control parameters with respect to particulate mass and number.

The injection system as a key element in the combustion system is responsible for the delivery of the demanded fuel mass at the correct timing and with the appropriate quality of atomization. All three parameters interact with other engine

| Table 1. Past, current and proposed future European Emission Standards |
|---|---|---|---|---|---|---|---|---|---|---|
| | Gasoline | | | | | | | | | |
| | EURO4 | EURO 5a | EURO 5b | EURO 6 | EURO4 | EURO 5a | EURO 5b | EURO 6 | | |
| THC mg/km | 100 | 100 | 100 | 100 | - | - | - | - | | |
| NMHC mg/km | - | 68 | 68 | 68 | - | - | - | - | | |
| HC+NOx mg/km | - | - | - | - | 300 | 230 | 230 | 170 | | |
| NOx mg/km | 80 | 60 | 60 | 60 | 250 | 180 | 180 | 80 | | |
| CO mg/km | 1000 | 1000 | 1000 | 1000 | 500 | 500 | 500 | 500 | | |
| PM mg/km | - | 5* | 4.5* | 4.5* | 25 | 5 | 4.5 | 4.5 | | |
| PN# #/km | - | - | - | TBD** | - | - | 6.0E+11 | 6.0E+11 | Status: September 2010 |

*for GDi engines only
**for all gasoline engines

Figure 1. Vehicle emission proposal for EURO 6 (first step engineering target: 70%) - Status June 2010
parameters such as air delivery, charge motion, etc. and determine the mixture preparation, which ultimately determines the combustion process and the emission characteristics. It is generally anticipated that the shorter time for mixture preparation - evaporation and homogenization - and in-cylinder spray-wall interactions are responsible for the higher particulate emission of GDi engine powered vehicles. The present article gives an overview of the development work performed at Delphi Powertrain Systems with the focus of reducing particulate engine out emissions of GDi engines.

PARTICULATE MATTER FORMATION IN GASOLINE ENGINES

The exhaust gas of internal combustion engines is a complex mixture of gaseous compounds, liquid and solid species. All components of the exhaust gas that can be trapped by a filter are considered to be particulate matter. These particles consist of elemental carbon (EC) also known as soot, an organic fraction with high and low volatility, ash from lubricants and metallic particles from wear [2]. While the particulate formation process for Diesel combustion is well investigated and documented [2], only very limited published material is available dealing with particulate formation during the combustion process of spark ignited engines [3][4].

EC can be formed during the combustion process in areas of locally rich combustion conditions. In the absence of oxygen and at temperatures above 400°C the hydrocarbon (HC) compounds from fuel or lubrication oil are decomposed into radicals and smaller molecules (pyrolysis), in particular acetylene. Those first radicals start to create polycyclic molecules (PAH's), which are the precursors to soot particles. The combination of these pre-cursors with HC molecules and other species form small soot particles (inception). Further dehydrogenation and combination of the species lead to a fast growth to larger particles (surface growth). In the coagulation phase larger particles are formed by accumulation governed by physical processes. These larger particles form agglomerates of a non spherical shape [5]. This step is characterized by a reduction of the particle number. Later in the combustion process most of the generated soot is oxidized and so typically only a smaller amount of soot is entering the exhaust system [3]. The conditions in the combustion chamber will determine the degree of oxidation: high temperatures and sufficient oxidation promoters like oxygen or OH radicals are the main factors of influence.

Pre-mixed combustion is dominant in gasoline engines, nevertheless inhomogeneities in the mixture can lead to locally rich areas which cause the formation of particulate matter [6]. Especially with GDi engines rich regimes can be found due to insufficient evaporation and mixing with air forming EC and volatile organic fractions. Mixture
deficiencies are also found close to combustion chamber walls when fuel is deposited and excessive soot formation occurs due to so called pool fires. In addition particulate matter consisting of volatile organic fraction is generated due to incomplete oxidation processes as the flame extinguishes. In the vicinity of the combustion chamber walls the in-cylinder oxidation of the generated particulates is limited and combustion ends consequently with a high amount of engine out particulates. Especially at cold start and warm-up engine conditions this trend is amplified as the evaporation of the fuel wall film is reduced and the turbulent mixing with the air is limited due to low engine speeds.

Overall, gasoline engine exhaust particles consist of similar materials and have a comparable morphology to Diesel particles. However, the number size distribution shows for homogeneous gasoline engines compared to Diesel engines typically a higher number of particles at smaller sizes [6]. Contrary to “typical” Diesel particulate matter, the majority of the particulates in gasoline engines is not composed of elementary carbon, but consists of high and low volatile compounds which are highly dependent on the operating conditions [7]. In Figure 3 the composition of deposits for two different coking tests in a GDi engine are shown. This data confirms the strong dependence of the deposit composition on the engine operation conditions.

The success of the mixture formation process determines the final engine out particulate emissions. Therefore the optimization of the injection process and the subsequent mixing while minimizing wall films are global guidelines for low particulate emissions. With respect to fuel wall films, GDi engine architectures with a central mount injector configuration should therefore have an advantage over side mounted configurations where the piston wall is used to guide injected fuel towards the center of the combustion chamber specifically in the catalyst heating mode.

PARTICULATE EMISSION OF GASOLINE ENGINES

Reduced particulate matter emissions result from minimizing the particle formation itself, as well as from optimizing particle oxidation in the combustion chamber and in the exhaust system. Clearly the injection process greatly influences mixture formation through mixture homogeneity and the potential presence of non-vaporized droplets and wall films. Other factors however, do also contribute to particulate emissions [8]. Figure 4 outlines the most important high level functions in order to minimize the particulate matter tailpipe emissions of a GDi engine.

Generally high exhaust gas temperatures and the presence of residual oxygen are required for an efficient oxidation at the end of the combustion process both in the combustion chamber and also in the exhaust system. This implies that engine control parameters promoting those two key engine characteristics for low particulate emissions need to be fully utilized. At a given engine load and speed retarding the spark timing leads to a delayed combustion with an increased exhaust gas temperature thus lowering particulates. However retarded ignition leads to a fuel consumption penalty and is therefore only used during catalyst light off.

An engine configuration with a fast warm-up of the combustion chamber walls helps to minimize the stored liquid fuel in wall films on the piston surface and the cylinder liner and in deposit layers. Engine oil in the combustion chamber introduced either via the lubrication path at the cylinder walls or via the intake ports contributes to particle formation. The higher evaporation temperature of the oil

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**Figure 3. Coking Deposit Composition at different Operating Conditions of a GDi engine**

![Deposit Composition Graph]

- **Volatile Residue [wt%]**
- **Oxidized Hydrocarbons [wt%]**
- **Elemental Carbon (Soot) [wt%]**
- **Ash Residue [wt%]**
constituents results in incomplete vaporization and a less effective mixture with air.

The cold start and warm-up phase contributes strongly to the overall particulate matter emissions as the engine is not yet at operating temperature and therefore particles consisting of volatile residues cannot be effectively oxidized. In consequence, similar to conventional gaseous emissions, any measure to shorten the warm-up phase will help to reduce both the particulate mass and the particulate number [9].

In principle, particulate filters already in standard use on modern Diesel engine powered passenger cars can be applied to gasoline engines as well [10]. The typically smaller particles generated by gasoline engines require a finer filter characteristics (e.g. cell density, mean pore size and porosity) which consequently leads to a higher exhaust system backpressure with a negative impact on performance, fuel consumption and CO2 emissions. The addition of a particulate filter system would cause a significant increase of the overall after-treatment system complexity and cost.

PARTICULATE REDUCTION BY OPTIMIZED OPERATION

VALVE TRAIN
Mixture quality is critically influenced by the successful atomization and subsequent evaporation of fuel and an adequate mixing with air. In addition to the injection process the optimization of the air path is important in order to provide fresh air and adequate charge motion levels for mixing and flame front propagation. To understand the influence of cam timing on particulate number emissions a series of tests were performed on a single cylinder engine. During warm engine operating conditions particulate mass and particulate number emissions were at a very low level. Influences of either intake or exhaust cam timing variation were negligible. However, at cold operation (50°C coolant temperature) a more significant effect of cam timing could be observed. At low part load conditions (2000 rpm and 3 bar NMEP) an increase in hot internal EGR, established by trapping exhaust gas in the combustion chamber with early EVC, was identified to reduce particulate number emissions by almost 40%. Increasing hot internal EGR by means of recuperating exhaust gas from the outlet port with late EVC did also show a reduction of particulate number emissions, but to a lesser degree of 20%. Both effects diminish when going to higher load and speed conditions. A higher percentage of hot EGR in the lower load condition provides higher mixture temperatures with better vaporization and helps to reduce particulate number emissions under these cold conditions. Varying intake valve timing, with the exhaust cam in the mid position, showed significant decrease in particulate number emissions of up to 55% with late IVC. As can be seen in Figure 5 in this low load/speed conditions the variation of the valve timing had little effect on HC emissions in contrast to the strong influence on particulate matter emissions.

ENGINE TEMPERATURE
Single cylinder engine testing has been performed at various fixed engine oil and coolant temperatures and at different injection timings for single pulse operation. Figure 6 shows the particulate emissions at two steady state engine
temperatures (coolant and oil): 60°C and 90°C and at two injection timings: 300°CA and 260°CA for a low load point operation. The total number of particulates represents the sum of both accumulation and nucleation modes. The nucleation mode consists of nano-particles formed from volatile precursors while the accumulation mode consists mainly of larger carbonaceous agglomerates that have survived the combustion process [11]. An increased engine temperature strongly reduces the particulate emissions. The more pronounced impact on the nucleation mode particles can be explained by the strong effect of temperature on its volatile precursors.

The single cylinder engine tests showed that the optimum injection timing is independent of engine temperature and therefore particulate emissions are best optimized at cold temperature and during the warm-up phase with focus on the control of the injection system. Engine level warm-up optimization (for example split cooling or switchable water pump system) is a key factor to limiting the particulate emissions during this phase.
Fuel pressure has a strong influence on particulate emissions as shown in Figure 7. Lower particulate numbers are observed with increasing fuel pressure due to improved mixture preparation. This pressure effect is almost entirely seen in the decrease of the nucleation mode particulates while the accumulation mode particulates remain almost constant. It has also been observed that the measurement stability improves at higher fuel pressures.

In addition, the effect of the probe position in the exhaust line on the particulate measurement has been evaluated. Lower particulate numbers are observed when the probe is far away from the cylinder head. The oxidation and coagulation process continues within the exhaust line and leads to a further reduction of the PN emissions. This means that in addition to the measurement device itself the setup and methodology of the measurement system is critical in order to obtain consistent measurement results. Various measurement systems exist based on different technologies [12] and a standardization is currently underway [13], [14].

**INJECTION STRATEGY**

A test campaign to reduce the particulate emissions by optimization of the injection strategy has been performed at cold engine conditions simulating the warm-up phase. Figure 8 shows results of the optimized calibration for various injection strategies. A strong reduction of particulates was achieved while improving the combustion stability (represented as COV %).

Multipulse operation helps to improve mixture preparation and therefore to minimize the particulate formation within the combustion chamber. In addition to an appropriate homogenization in the intake and compression stroke, a short pulse delivering a very small quantity of fuel prior to the ignition is beneficial to stabilize the combustion while maintaining the particulate emission at an acceptable low level [15].

The quantity required for the late injection after TDC requires the control of small pulses in the ballistic operation mode of the injector. In order to reliably control the injector over life time it is required to extend the currently standard EMS control strategies.

In Figure 9 CFD results using the optimized injection strategy (timings and injected fuel mass detailed in Figure 8) are shown for a piston position of 25° crank angle aTDC. The overall slightly lean mixture was obtained by the early injection into the intake stroke, where the majority of the fuel was introduced. A small amount of fuel mass was injected during the compression stroke to establish stratification in the center of the combustion chamber. The injection of a very small quantity after TDC leads to a locally rich mixture
which develops asymmetrically towards the spark plug due to the in-cylinder charge motion. The turbulent kinetic energy shows that the last injected quantity does not have a strong impact on the turbulence energy level overall and around the spark plug position in particular. This leads to the conclusion that the primary stabilizing effect of the late injection is due to the creation of a locally slightly rich mixture.

PARTICULATE REDUCTION BY OPTIMIZED MIXTURE FORMATION

The results in the previous chapter show that in addition to the accurate control of small pulse fuel delivery, improved mixture preparation is a key parameter in order to minimize engine out particulate emissions. In this chapter the role of the injector in the mixture preparation is further investigated. As shown in Figure 1 current production vehicles using low pressure MPFI can achieve particulate numbers below the proposed EURO 6 targets. In a MPFI application the fuel is injected on the intake ports. Heat from the ports and intake valve facilitate fuel vaporization, and then the air fuel mixture is entrained with high velocity into the combustion chamber where a high degree of mixture homogeneity is achieved at the moment of the spark event. In a direct fuel injection application, it is more challenging to achieve a sufficiently homogeneous base mixture because of a more limited time for the injection and subsequent evaporation of fuel, the risk of wall wetting and a less efficient mixing of air and fuel compared to MPFI. Issues of wall wetting can be addressed by optimized spray targeting and by the use of multiple pulse injection strategies in order to limit spray width and penetration.
The overall mixture preparation is improved by an increased evaporation rate. For a given nozzle geometry faster evaporation is promoted by higher system pressure and an increased injection rate \[15\], but also the detail of the nozzle design can have a major influence \[17\]. In this study the focus is on the comparison of two basic nozzle types: multihole and outwardly opening valve.

The multihole nozzle achieves spray atomization by a number of individual spray holes (typically 5 to 7) \[18\]. In fully open valve conditions most of the pressure drop is across these flow holes of typical diameters in the 150-200 µm range. The overall mixture formation can be optimized using the targeting of the individual flow holes but also by the adaptation of flow parameters of the individual holes, i.e. length to diameter ratio and inflow conditions \[17\]. During the injector opening and closing phases the available differential pressure across the flow holes is strongly reduced due to the pressure loss in the valve to seat gap and the required change of flow direction into the individual flow holes. This effect can have an impact on spray atomization and results in the formation of larger droplets both at the spray front and tail. When shortening injection pulses an increasingly larger portion of the overall spray is affected by the opening and closing events.

The outwardly opening nozzle generates the spray in a conical slit which is directly fed by the high pressure fuel \[19\]. For a typical valve stroke of 40 µm and a 90° spray angle the resulting slit height is around 30 µm. During all injector operating conditions the complete pressure drop is across the conical nozzle without deviation of the main flow direction.

Figure 10 shows the Sauter mean diameter D32 and the Dv90 over system pressure comparing two outwardly opening nozzles (Multec 20 Homogeneous Injector labeled M20 HI and 400 bar capable High Performance Injector labeled M20 HPI) and a multihole nozzle. The outwardly opening injectors are based on the injector nozzle developed for the M20 base injector \[19\]. Results were obtained by a time-averaged laser diffraction measurement. As specified in SAE J2715 \[20\] the measurement volume is at 50 mm downstream of the nozzle exit and N-heptane fuel is used. Injection times were adjusted to generate 7.5 mg fuel delivery, which represents typical fuel consumption for idle operation in warm-up conditions. The impact of pressure on the generated droplet sizes is very pronounced below 200 bar. Tests using the piezo driven M20 HPI show a weakening impact of system pressure but without reaching asymptotic conditions at 400 bar. The smaller droplet size results of the Multec 20 HI compared to the Multec 20 HPI is mainly due to a reduced penetration of the Multec 20 HI spray and a resulting increased effect of the recirculation of small droplets in the measurement volume.

In Figure 11 the temporal evolution of the Dv90 droplet size distribution is compared between the multihole and the two outwardly opening injectors M20 HPI and HI. The time axis is normalized with respect to the arrival of the first droplets into the measurement volume. This compensates differences in opening delay and travel time of the initial spray front. The initial peak is generated as the progression of the spray front reaches the measurement volume. In the initial spray front a volume portion of 10% of the droplets is larger than ~24 µm for the 3 nozzles compared. The evolution in the tail of the spray shows a steady increase in Dv90 for the multihole nozzle while the outwardly opening valves stay flat. This effect can be explained by the arrival of larger droplets into the measurement volume due to the injector closing event. Due to the reduced exit velocity of the droplets in the spray tail the travel time into the measurement volume increases. Figure 11 also shows the link of the temporal evolution to the instantaneous cumulative droplet distribution at a late timing of 7 ms after the start of injection. The multihole results are shifted towards larger droplet sizes across the whole distribution compared to the two outwardly opening valves. From the graph with the cumulative droplet distribution at 7 ms Dv90, Dv50 and Dv10 can be directly obtained. The distribution above Dv90 shows that the largest drops for the multihole injector spray are up to 20 µm larger than those for the outwardly opening injector sprays.
In summary the laser diffraction measurement results show larger droplets in the spray tail for the multihole injector. Nevertheless this difference in size distribution is only one aspect in the explanation of the differences in evaporation rates. Not only the basic ability of a nozzle to atomize fuel, but also its ability to transport fuel into the combustion chamber and contribute to the formation of an overall homogenous mixture is required to optimize particulate number emissions. One aspect in this context is the overall size of the spray characterized by the penetration and width.

The spray penetration results in Figure 12 were obtained by shadowgraphy and show a similar initial spray front velocity up to about 30 mm penetration for all three injectors. After that point the outwardly opening Multec 20 HPI and HI develop substantially less penetration than the multihole.

Besides the width and penetration the multihole and the outwardly opening injector differ in the overall spray volume to spray surface ratio. Whether these considerations can help to quantify the differences in evaporation behavior between injector nozzles needs to be verified in a future investigation. Also, as a next step, the development of the spray close to the nozzle exit will be investigated with a special focus on the opening and closing events.
VEHICLE TEST DATA
The strategies, which had been explained in the previous chapters were adapted and verified at Delphi Powertrain on a prototype vehicle with a complete GDi Delphi EMS described in [21]. The internal development goal for this first step was to reach with adapted calibration strategies less than 70% of the expected EURO 6 particulate mass and number targets in homogenous operation, while the conventional gaseous emissions should be kept below the EURO 6 level. The selected vehicle is equipped with a 3.5 l V6 engine (bore ~93 mm) and a central injector location.

The main focus of the application work in the vehicle was during cold start and warm-up. A design of experiment approach was followed to optimize important parameters including fuel rail pressure, injection targeting, injection split ratio, A/F ratio, spark angle timings. After the warm-up phase the calibration focus was on the higher load acceleration phases. In the following the current development status is summarized which enables to meet both the particulate mass as well as the planned particulate number emission targets. An emissions trace of PM and PN over the NEDC cycle is shown in Figure 13, reflecting the performance resulting from the application work.

START
The engine start was performed with a 1-pulse strategy into the compression stroke (high pressure start). Previous tests had already shown the positive influence on stoichiometric or slightly lean A/F ratio of a stratified start where fuel is injected at a high fuel rail pressure. These results were used as a base for the particulate number investigation. The rail pressure for the start was finally set to 80 bar. Increasing the rail pressure during the start had no significant positive impact on particulate numbers but increased the start time. After the start the rail pressure was immediately brought to 200 bar and was kept there for the rest of the test. During the start the A/F ratio was kept close to stoichiometric.

WARM-UP PHASE
For the mixture preparation during the combustion chamber and converter light-off phase a 2 pulse strategy was applied with a pulse in the late intake phase and a pulse in the late compression phase. This initial approach is substantially simpler than the 5 pulse strategy found to be optimum in the single cylinder engine test campaign. After the converter light-off phase the split injection was kept up to 200 seconds.

For catalyst heating a spark retard strategy was applied down to 20 degree aTDC spark angle during the first idle phase. This spark retard was limited to a few seconds after the start in order to give the combustion chamber and exhaust system time to heat up and minimize the negative impact on fuel consumption. The retarded spark strategy was stopped at the end of the first idle, where the converter reached a temperature of 600°C. Normal spark has been applied during the first acceleration to create max torque at the lowest injected fuel mass and with reduced particulate number emissions.

The air excess ratio Lambda during the first idle after the start was kept slightly lean between 1.05 - 1.1 to avoid local rich zones in the combustion chamber. This Lambda was retained during the first ECE driving cycle. Afterwards stoichiometric operation was started.

REMAINING PART OF THE NEDC EMISSION CYCLE
After the first ECE driving cycle the coolant temperature was at around 60°C and the engine and the combustion chamber was sufficiently heated up to vaporize efficiently the fuel with one pulse only. Injection timing for this pulse was early in the intake to have the maximum time for mixture preparation. With this strategy particulate number emission was almost flat for the rest of the ECE and EUDC cycle.

To reduce particulate numbers during the transients (mainly the acceleration phases) it was necessary to implement smooth transitions from idle/low load to higher loads of all combustion related parameters like cam phasing angles, injection timing(s) and spark angles. In addition it was assured that by appropriate algorithms that the 3-way converter was purged from oxygen during the idle phases, so that no significant rich fuel bias was necessary during the acceleration phases to avoid NOx breakthrough. This rich bias would also increase the particulate number emission by local rich zones in the combustion chamber.

The internal development goal for participate number and particulate mass emissions in the vehicle was met. Further progress will be driven by the transfer of the optimized strategies developed on the single cylinder to the multi-cylinder engine. In particular, the strategy to inject a very small pulse shortly before the spark event will be implemented and tested on the multi cylinder engine in the next step. In order to achieve robust and reproducible operation of small pulses below 1-2 mg an enhanced cylinder-individual learning software is being developed.
SUMMARY AND OUTLOOK

The introduction of a legislation for particulate emission of gasoline powered vehicles adds a new challenge to the engine development and vehicle calibration. Similar to Diesel engines the formation of particulates is caused by an imperfect mixture formation inside the combustion chamber. Gasoline engines with direct injection have a significantly reduced time for mixture formation compared to conventional MPFI engines and therefore have more difficulties to meet the proposed future targets.

Very similar to the conventional gaseous emissions, the majority of the tailpipe emitted particulates are from the cold start and warm-up phase of the engine (Figure 13). Any measure to accelerate the warm-up of the engine and the three-way catalyst system contributes consequently to lower vehicle emissions. For low particulate matter emissions a detailed optimization of the entire air as well as the fuel path and the interaction between each other is required. The direct injection fuel system needs to: enable precise metering and timing of small fuel masses in order to provide a very effective atomization with reduced penetration, prevent rich regions inside the combustion chamber and avoid spray wall-interaction. Fast engine control architecture and sophisticated algorithms is the key to handle successfully the additionally required control tasks.

Gasoline engine exhaust gases can also be filtered effectively with the addition of a costly particulate filter system, but at the same time the increased exhaust backpressure will negatively impact the vehicle performance, fuel consumption and CO2 emissions. The demonstrated vehicle results show that the proposed EURO 6 targets can be met by gasoline engines with direct injection by further optimization of the relevant hardware and calibration without adding after-treatment systems. For the particulate mass emission a value significantly lower than proposed target can be achieved, while the proposed particulate number limit is more challenging. Although the particulate emissions of MPFI engines are lower today, the ongoing development activities combined with the inherent higher control flexibility of GDi engines indicate similar particulate emission potential.

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DEFINITIONS/ABBREVIATIONS

aTDC  
after TDC

CA  
Crank Angle

CFD  
Computational Fluid Dynamics

D32  
Sauter Mean Diameter (SMD)

Dv90  
droplet diameter (90% of total liquid volume is in droplets of smaller diameter)

EC  
Elemental Carbon (Soot)

GDi  
Gasoline Direct Injection

MPFI  
Multi Point Fuel Injection

TDC  
Top Dead Center

rpm  
revolutions per minute

PM  
Particulate Mass

PN  
Particulate Number