

Effect of Operation Strategy on First Cycle CO, HC, and PM/PN Emissions in a GDI Engine

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ABSTRACT

The impact of the operating strategy on emissions from the first combustion cycle during cranking was studied quantitatively in a production gasoline direct injection engine. A single injection early in the compression cycle after IVC gives the best tradeoff between HC, particulate mass (PM) and number (PN) emissions and net indicated effective pressure (NIMEP). Retarding the spark timing, it does not materially affect the HC emissions, but lowers the PM/PN emissions substantially. Increasing the injection pressure (at constant fuel mass) increases the NIMEP but also the PM/PN emissions.

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INTRODUCTION

As a result of more stringent fuel economy standards aiming for CO₂ emissions of around 100gCO₂/km by 2025 [1], GDI engines have gained market penetration in the USA, with a 30% share of the gasoline engine sales in 2012 [2], and a prognosticated market share of up to 97% by 2025 [3]. GDI engines have better knock resistance through charge cooling; allow for aggressive scavenging to improve the low-end torque of turbo-charged engines and offer capability to increase the lean operation limit. On the other hand, liquid injection into the combustion chamber increases the emissions of unburned hydrocarbons (HC) and particulates (PM/PN), especially during cold-start. Figure 1 shows the normalized engine-out and tailpipe HC emissions during the FTP-75 for a gasoline engine. Due to the inactivity of the catalyst, more than 90% of the HC emissions occur during the first 2 minutes of operation.

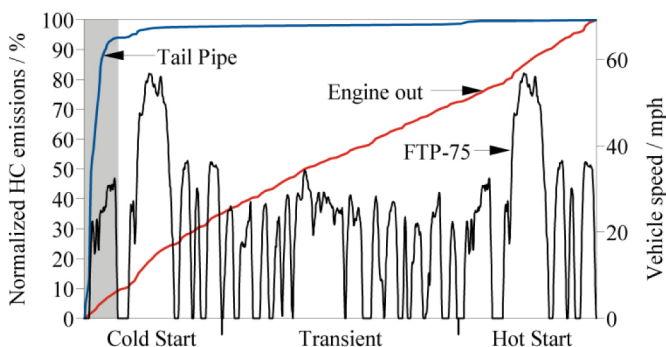


Figure 1. Cumulative engine-out and tailpipe HC emissions over the FTP-75 cycle for a gasoline engine.

In an effort to understand better the cold-start emissions, it is necessary to quantify the contribution of the cold crank-start process. Figure 2 shows the cumulative HC and PM emissions during the crank-start of a

production GDI engine as a percentage of the maximum allowable emission limits for the U.S. standard T3B50 / ULEV50, assuming the same NMOG/NO_x ratio as the T2B5 standard.

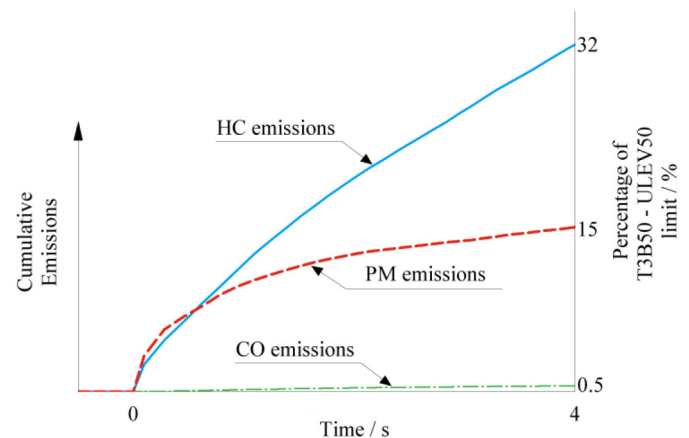


Figure 2. Cumulative emissions as percentage of the T3B50/ULEV50 limit

During a typical crank-start, the emissions account to 110 mg of unburned HC corresponding to 32% of the T3B50 limit and 5 mg of particulates corresponding to 15% of the T3B50. Additionally, the number of particles emitted are 2.7×10^{12} corresponding to 40% of the Euro6 limit (U.S. regulations do not include a limit for the number of particles). Given the significant impact of the cold cranking process in the overall emissions, it is necessary to gain a better understanding of the formation mechanism of the pollutants during engine crank-start and their dependence to the injection strategy during cranking.

The cold wall temperature and low engine speed during the crank-start process pose significant constraints on the mixture formation process. The cold cylinder walls result in the formation of fuel films which, in turn, are partly responsible for the high levels of HC and PM emissions during cold cranking.

PM/PN emissions. Additionally, the cold temperature promotes the absorption of fuel into the oil layer prior to combustion. After combustion, the hot burned gas with low HC concentration favors desorption of the fuel back into the bulk gases, providing an additional source for HC emissions. The low wall temperature also results in increased heat transfer away from the flame front during combustion causing a larger flame quenching distance, increasing the amount of gaseous fuel escaping combustion. Finally, the post-flame oxidation rate of the HC that managed to escape combustion and of the particles formed during combustion is reduced due to increased heat transfer to the cold cylinder walls.

The low engine speed impacts the degree of charge motion during cranking. This results in poor mixture formation, in the form of reduced evaporation of the fuel and reduced homogeneity of the mixture. Additionally, the post-flame oxidation is affected by the mixing process of the unburned HC layer in the vicinity of the walls with the hot bulk gases. This mixing process is a function of engine speed and it is negatively impacted during crank-start.

The cold cranking process for GDI engines has been studied experimentally and numerically in the past. Fan et al. studied experimentally the effect of split injection on the combustion characteristics of the first cycle [4] and on the HC emissions behavior of the complete cranking process [5]. Whitaker et al. [6] studied in an optical engine the effect of fuel pressure on spray penetration and droplet size at cranking speed as well its effect of PM emissions. Geringer et al. [7] studied the startability at extreme cold temperatures using E85, and the effect of throttling on the evaporation of the fuel and the run-up time. Wiemer et al. [8] measured the effect of fuel quantity, injection timing and spark timing on the run-up speed trace and HC emissions.

Using CFD analysis Kim et al [9] studied the AFR distribution during cranking at subzero conditions for different injection timings. Similarly, Malaguti et al. performed numerical analysis for the first [10] and second [11] combustion cycles, focusing on spray-wall interaction and its effect in fuel evaporation and liquid fuel film formation. Also, Xu et al. [12] modeled the engine cranking and run-up for split injection strategies and their impact on the AFR around the spark plug.

This paper also builds upon the methodology developed at MIT for studying the first combustion cycle and the cranking process in PFI-SI engines [13, 14, 15, 16, 17, 18].

Of all the cycles during the engine crank-start, the first combustion event is the most problematic since it experiences the lowest temperature and engine speed. Additionally, the absence of hot residual gases reduces the evaporation of the fuel spray compared to the next combustion cycles during crank-start. The goal of this study is to quantify the individual contributions of a broad set of different variables affecting the mixture formation process, and their impact on the combustion behavior and on the emissions of CO, HC and PM for that first combustion event during cranking.

EXPERIMENTAL METHODOLOGY

Engine Set Up

The experiments were carried out in a GM LNF engine; with the specifications given in Table 1. The engine features a centrally mounted spark plug and 4 valves per cylinder. The valve timing corresponds to the parked position of the camshafts, resulting in a negative valve overlap of 20°CA. This valve timing reduces the volumetric efficiency and effective compression ratio and increases the residual gas fraction during the cranking process. The injection system (Fig. 3) consists of side-mounted, 6-hole electromagnetic injectors with a horizontal inclination of 47° and a cone angle of 52°.

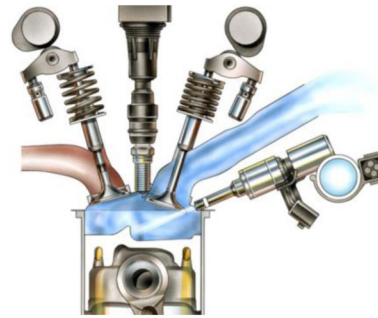


Figure 3. Combustion system of GM's LNF engine ©GM

Table 1. Engine specifications

Displacement	1998 cc
Bore / Stroke	86 / 86 mm
Connecting Rod	145.5 mm
Compression ratio	9.2:1
Inlet Valve Open / Close	11°aTDC / 61°aBDC @ 0.2 mm lift
Max. intake valve lift	10.3 mm @ 126°aTDC
Exhaust Valve Open / Close	52°bBDC / 10°bTDC @ 0.2 mm lift
Max. exhaust valve lift	10.3 mm @ 125°aTDC

The fuel, intake air and coolant temperature are kept at 20°C. The fuel is pressurized by a hydro-pneumatic accumulator using high pressure nitrogen, allowing the fuel pressure to be set independently from engine operation. The fuel used is a Tier II EEE certification gasoline with a carbon mass fraction of 86.5%.

The exhaust composition is measured using fast response analyzers from Cambustion. The HC concentration is measured using an FFID unit (HFR400), with a response time t_{10-90} of 1ms, and a sampling position 6cm from the exhaust valve. The CO and CO₂ concentrations are measured with a fast NDIR unit (NDIR500), with a response time t_{10-90} of 8ms, and a sampling position 8cm from the exhaust valve. The PM and PN are measured using a fast particle spectrometer (DMS500) with a t_{10-90} of 300ms a sampling point 15cm from the exhaust valve.

The DMS500 classifies particles in a discrete manner based on their electrical mobility, i.e. their drag to charge ratio. A typical particle concentration vs. size spectrum is shown in Figure 4. The discrete measurement is fitted by two log normal distributions, corresponding to the nucleation mode and to the accumulation mode. The log normal

distribution provides the best empirical fit in aerosol statistics. The distribution is described by the geometric standard deviation and the geometric mean of the diameters, which due to the symmetry of the distribution also corresponds to the median. In aerosol characterization, that median receives the name of count median diameter (CMD, see Fig. 4).

Since the European Particle Measurement Program (PMP) requires the removal of the volatile fraction, and given the low contribution of volatile particles to PM, it was chosen to consider the accumulation mode exclusively. In addition the sampling temperature was kept at 150°C to prevent volatiles condensation on the accumulation mode.

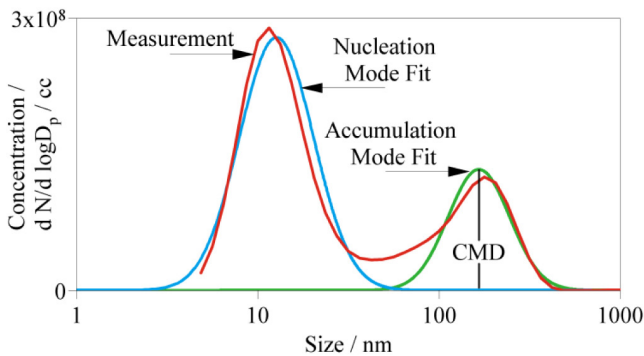


Figure 4. Particle distribution measurement for a representative first cycle

Experiment Description

The experiment carried out to study the first cycle emissions during the crank-start process is presented in Figure 5. In order to recreate the conditions at cold cranking, the engine coolant, fuel and intake air temperatures are kept at 20°C. The experiment starts with the engine being motored at a cranking speed of 280 rpm for several cycles in order to evacuate the residual HC stored in the combustion chamber and to measure the HC background concentration. Then a metered amount of Tier II EEE certification gasoline is injected followed by a combustion event. The engine continues to be motored for 50 additional engine cycles at 280 rpm, while the HC, CO, CO₂ and PN concentrations are measured and recorded throughout the process.

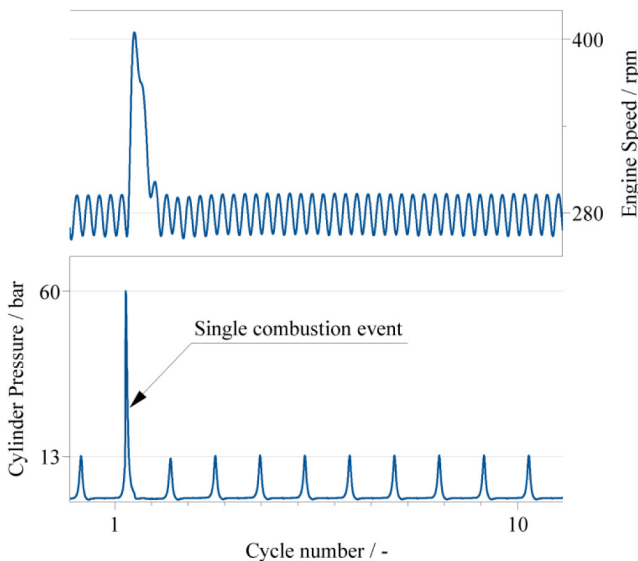


Figure 5. Engine speed and cylinder pressure during the single fire experiment

To assess the impact of the first cycle emissions it is necessary to translate the concentration measurements to exhausted mass of pollutants. To achieve this, further information about the exhaust mass flow rate is required. This information can be extracted from the cylinder pressure trace and the piston position by idealizing the exhaust process as an isentropic expansion of an ideal gas as shown in Eq. 1.

$$\dot{m}_{exh} = - \left(\frac{1}{\gamma} \frac{dP_{cyl}}{P_{cyl} dt} + \frac{1}{V_{cyl}} \frac{dV_{cyl}}{dt} \right) m_{cyl} \quad (1)$$

Figure 6 shows two typical mass flow rate traces for a fired and a motored exhaust stroke. Both traces have opposite behaviors between EVO and BDC. The fired cycle displays a blowdown process as the exhaust valve opens, while a reverse flow occurs for the motored cycle. In order to account for this difference, the phasing of concentration signals is corrected considering the velocity of the exhaust gas inside the runner. Then, the corrected signal is integrated over the exhaust event as shown in Eq. 2.

$$m_{k,cycle} = \int_{t_{EVO}}^{t_{EVC}} \hat{x}_k(t + \tau(t)) \frac{M_k}{M_{exh}} \frac{dm_{exh}}{dt} dt \quad (2)$$

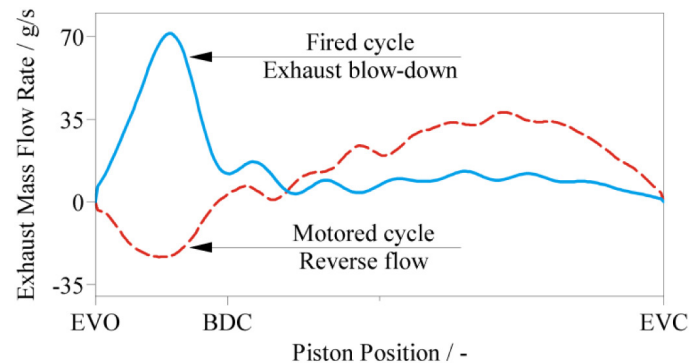


Figure 6. Mass flow rates for fired and motored exhaust events

The concentration of number of particles per standard cubic centimeter (#/scc) can be found by integrating the lognormal distribution for the accumulation mode (see Fig. 4 and Eq. 3). The total number of particles can be calculated by integrating the product of particle concentration with the standard volumetric exhaust flow (Eq. 3).

$$\hat{N} \left[\frac{\#}{scc} \right] = \int_{1 \text{ nm}}^{1000 \text{ nm}} \frac{dN}{d \log D_p} d \log D_p$$

$$PN[\#] = \int_{t_{EVO}}^{t_{EVC}} \hat{N} \frac{1}{\rho_{std}} \frac{dm_{exh}}{dt} dt \quad (3)$$

The calculation of the PM from the PN and size spectrum measurement requires additional consideration of the morphology of the soot particles and of their specific gravity. For GDI engines the recommended fractal dimension $D_f = 3$ (so that volume is proportional to d^{D_f}), i.e. spherical particles, and the recommended specific gravity