

Cycle-by-Cycle Analysis of Cold Crank-Start in a GDI Engine

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ABSTRACT

The first 3 cycles in the cold crank-start process at 20°C are studied in a GDI engine. The focus is on the dependence of the HC and PM/PN emissions of each cycle on the injection strategy and combustion phasing of the current and previous cycles. The PM/PN emissions per cycle decrease by more than an order of magnitude as the crank-start progresses from the 1st to the 3rd cycle, while the HC emissions stay relatively constant. The wall heat transfer, as controlled by the combustion phasing, during the previous cycles has a more significant influence on the mixture formation process for the current cycle than the amount of residual fuel. The results show that the rise in HC emissions caused by the injection spray interacting with the intake valves and piston crown is reduced as the cranking process progresses. Combustion phasing retard significantly reduces the PM emission. The HC emissions, however, are relatively not sensitive to combustion phasing in the range of interest.

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INTRODUCTION

In response to growing concerns over global warming and climate change, the CO₂ emissions legislations around the world have become increasingly stringent. Figure 1 shows the evolution of the CO₂ legislation for light duty vehicles (LDV) in different parts of the world as they strive towards a goal of approximately 100 grams of CO₂ per kilometer during the next decade.

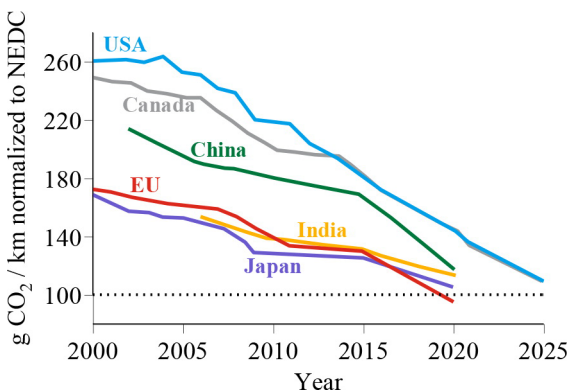


Figure 1. Light duty vehicles CO₂ emissions regulations around the world. Data normalized to NEDC. Adapted from [1]

The tightening of the fuel consumption standards has been a driver for the adoption of new technologies during the past decade; among them gasoline direct injection (GDI) engines. According to EPA and NHTSA estimates [2], downsized turbocharged GDI engines with cooled EGR have a potential for CO₂ emissions reduction between 16% and 25% compared to the 2008 gasoline engine (PFI, NA) used as a baseline. As this potential has been tapped during the past decade the market penetration of GDI engines has grown at an accelerated

pace (see Fig. 2). Despite the advantages in fuel consumption of GDI engines, the liquid fuel injection into the combustion chamber poses emissions challenges, particularly during the cold-start phase where over 95% of the HC [5] and over 75% of the PM/PN [6] tailpipe emissions take place.

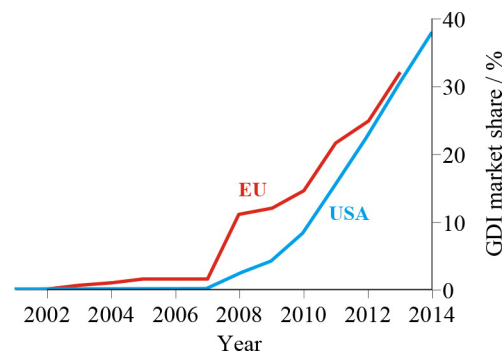


Figure 2. Market penetration of GDI engines in the US and the EU for the past decade. Data source: US [3]; EU [4]

Figure 3 shows the engine behavior during a typical cold crank-start of a GDI engine. Prior to the first combustion cycle the engine is driven by the starter motor at approximately 280 rpm. After the 1st combustion event the engine speed increases rapidly and reaches its maximum speed (a.k.a. speed flare) within 1 second. After the speed flare, the engine speed decreases to the targeted cold fast-idle speed. As can be seen from the cumulative engine out emissions in Fig. 4, the engine cold crank-start is responsible for a disproportionate amount of the cold-start phase emissions. During the initial 4 seconds (0.2% of the total FTP-75 duration) 110 mg of HC and 5 mg of PM were emitted, corresponding to 32% and 19% of the T3B50 limit respectively.

In comparison to all the other operation points during the certification driving cycle, the cold crank-start is subject to the lowest temperature and lowest engine speed. As a result, the fuel evaporation and mixture formation process are compromised and significant over-fueling is necessary to produce an ignitable fuel-air mixture. Because of the large amount of fuel injected, a significant portion of the injected fuel lands on the cold combustion chamber surfaces, resulting in fuel films that fail to evaporate before combustion. Optical investigations conducted by Costanzo et al. [7] provide evidence of the significant role that these fuel films play on HC emissions and PM formation. Additionally, the low temperature impacts negatively the classical HC emissions mechanisms identified by Cheng et al. [8]; low temperature results in larger crevice volumes, increases the solubility of hydrocarbons into the lubricant oil layer, and intensifies the heat transfer rates from the hot gases into the cylinder walls. The latter increases the flame quenching distance and reduces the post-flame oxidation rate of the resulting pollutants after combustion.

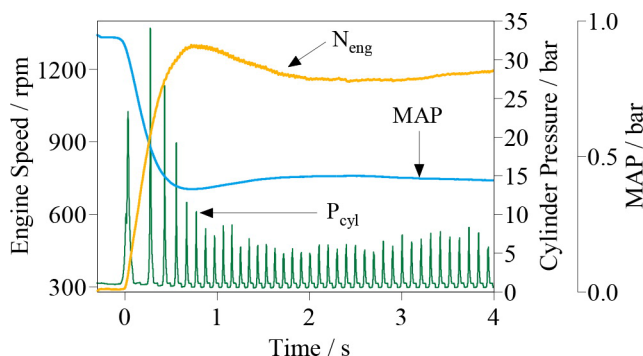


Figure 3. Cylinder pressure, engine speed and intake manifold pressure traces during cold crank-start.

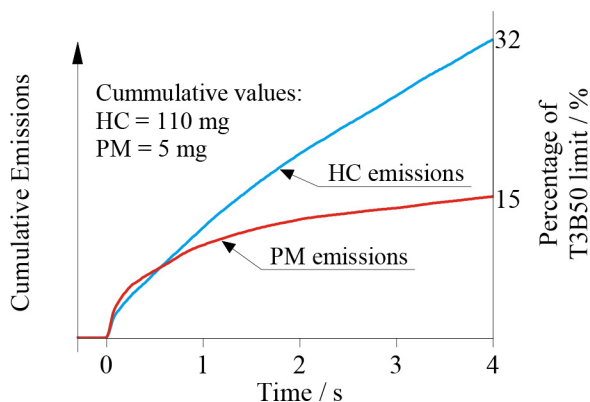


Figure 4. Cold crank-start cumulative emissions as percentage of the T3B50/ULEV50 limit [9]. The HC limit of T3B50 assumes the same HC/NOx ratio as the T2B5 standard

This study seeks to expand the understanding of the HC and PM/PN emissions behavior during the cold crank-start process of a GDI engine. For that purpose the initial 3 engine cycles are analyzed individually under a set of parameters that include the mass of injected fuel, the start of injection (SOI), and the ignition timing. Additionally, the interactions between cycles are also considered by studying the impact that the heat transfer history and the residual fuel from previous injection events have on the mixture formation and fuel enrichment requirement of the current cycle.

The research presented on this paper is a natural extension of previous studies by the authors focusing on 1st cycle emissions and fuel pathway during cold crank-start [9, 10]. This study also builds upon the methodology developed at the Sloan Automotive Laboratory at MIT for experimentally studying the first combustion cycle and the cranking process in PFI engines [11, 12, 13, 14, 15, 16, 17].

EXPERIMENTAL METHODOLOGY

Engine Set Up

The experiments were carried out using a commercial 4-cylinder, 2 liter GDI engine featuring side-mounted electromagnetic injectors, with a 52° cone angle, a 25° inclination from the horizontal and 6 holes. The variable valve timing (VVT) system was deactivated and a fixed valve timing was used for all of the experiments (see Table 1).

Table 1. Specifications of the GM - LNF engine

Displacement	1998 cc
Bore / Stroke	86 / 86 mm
Connecting Rod	145.5 mm
Compression ratio	9.2:1
Fuel pressure	50 bar
Intake 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

In real GDI applications the typical fuel pressure for the initial injection events during cranking ranges between 30 and 70 bar [18, 19] and is heavily dependent on engine speed. In the experimental setup used in this study, the fuel pressure was kept independent from engine operation and was maintained at a constant value of 50 bar by a hydro-pneumatic accumulator (see Fig. 6). A Tier II EEE certification gasoline was used with a carbon mass fraction of 86.5%, 29% aromatics content and a Reid vapor pressure of 62.7 kPa. The distillation curve can be found in Fig. 5. Three independent chillers for the fuel, intake air, engine oil and coolant were used to keep the cold crank-start conditions, with all temperatures at 20°C.

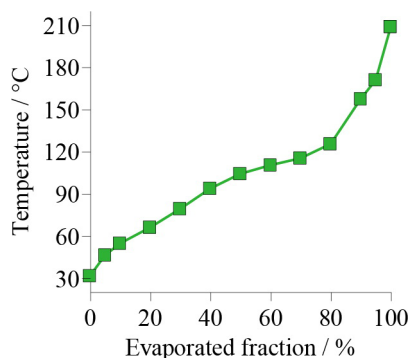


Figure 5. Distillation curve of the used Tier II EEE certification gasoline

The pollutant emissions were monitored at the exhaust runner of cylinder 4 using fast response analyzers. The water content in the exhaust was measured by a fast FID (Combustion HER400) with a 10 to 90%
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time response of 1 ms. The CO and CO₂ wet mole fractions were measured by a fast NDIR (Cambustion NDIR500) with a 10 to 90% time response of 8 ms. Lastly, the particulate spectrum was measured using a differential mobility spectrometer (Cambustion DMS500) with a size range of 5-1000 nm and a 10 to 90% response of 300 ms.

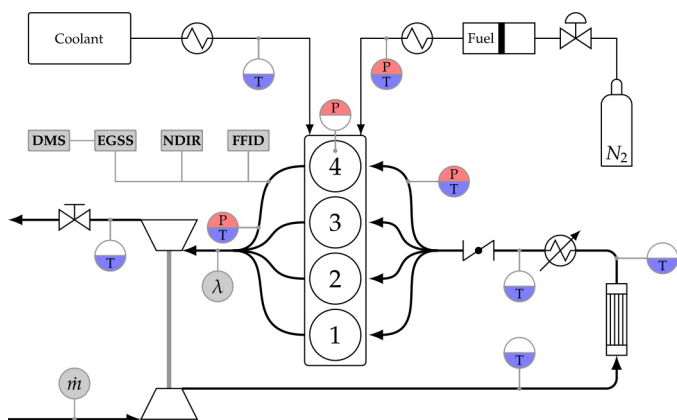


Figure 6. Diagram of the experimental setup and the sensor locations

In comparison to the fast FID and fast NDIR analyzers, the time response of the DMS is two orders of magnitude slower, precluding crank-angle resolved measurements of the particle spectrum. Still, at $t_{10-90\%} = 300$ ms, the instrument's response time is sufficient to allow cycle-resolved measurement. However, as the engine speed increases during crank-start, the time interval between two consecutive exhaust events might not be long enough to allow the DMS to completely differentiate between the emissions of two adjacent engine cycles. To overcome this obstacle for performing a cycle-resolved analysis of the initial 3 cycles, an exhaust gas sampling system (EGSS) is used.

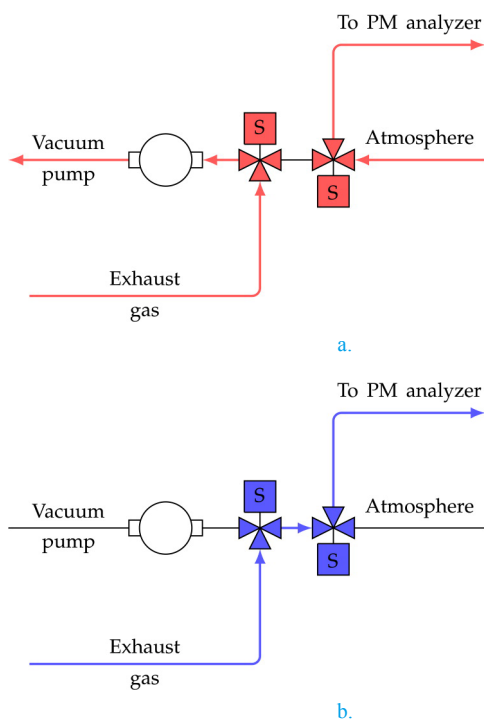


Figure 7. Exhaust gas sampling system (EGSS) for cycle-resolved particle spectrum measurement. a) Closed position feeds ambient air to the analyzer and circulates the exhaust gases to minimize the dead volume. b) Open position feeds exhaust sample to the analyzer

The objective of the EGSS is to supply to the analyzer only the exhaust gases from the desired cycle, while blocking the exhaust gases from previous and following cycles. The EGSS is located upstream of the DMS (Fig. 6) and consists of two three-way solenoid valves connected in series. In the “closed” position (Fig. 7-a), the exhaust gas is circulated around the system by a rotary vane vacuum pump in order to minimize dead volumes, while the DMS is allowed to sample ambient air in order to maintain its internal pressure within range. In the “open” position (Fig. 7-b) the exhaust and sample flows are connected, and the DMS is able to perform the particle spectrum measurement for the desired engine cycle. The solenoid valves have a 3/32” orifice and an opening time of 10 ms, which is fast enough to capture the exhaust flow period of 30 - 100 ms. The EGSS is controlled by a programmable microcontroller that uses the crankshaft encoder signal as an input for the timely triggering of the solenoid valves.

The emission values reported in this paper are presented in terms of mass per cylinder per cycle. Due to the transient nature of the experiments, it is difficult to measure the time-resolved exhaust flow rate using an air-flow meter. Instead, the in-cylinder pressure during the exhaust stroke was used to calculate the exhaust mass flow rate. The latter was then integrated with the concentration measurements over the exhaust stroke. Further details on the methodology used to convert the mole fraction and concentration values to mass emissions per cycle can be found in Ref. [9].

Experiment Description

As pointed out in the experimental layout description (see Fig. 6), the study was done via cylinder #4 with the engine operating in single-cylinder mode. A direct consequence of this approach is the inability to achieve a speed transient that is representative of a 4-cylinder engine during crank-start. To circumvent this, the engine was torque-assisted by a 10hp electric motor during the simulated crank-start experiments. The motor control and speed-ramp were adjusted to achieve similar speed traces between multi-cylinder and single-cylinder crank-start (Fig. 8).

In all cases, an experimental run started with the engine motoring at cranking speed while the temperatures and the exhaust HC background concentration were monitored until steady state condition was achieved. After the steady-state motoring, and depending on the engine cycle of interest, one, two or three combustion events took place in cylinder 4 (see Fig. 8 a-c). Thereafter, engine was motored at a fixed speed for 50 additional engine cycles while the exhaust flow composition and temperatures were recorded. The engine was brought again to cranking-speed motoring until the purging of residual HC was completed and the steady-state condition was attained. Five experiments were performed for each experimental condition; the average values are reported.

Throughout all of the experiments the throttle opening was fixed at the fast-idle position (2 bar NIMEP, 1200 rpm). The engine operated in the open-loop mode with the fuel amount being a function of the demanded enrichment, the engine speed and the intake manifold pressure (speed-density calibration). Table 2 shows a summary of the typical values of the relevant parameters for the initial 3 engine cycles