Fuel carbon pathway in the first cranking cycle of a gasoline direct injection engine

The MIT Faculty has made this article openly available. Please share how this access benefits you. Your story matters.
Fuel Carbon Pathway in the First Cranking Cycle of a GDI Engine

J.F. Rodríguez*, W.K. Cheng

Sloan Automotive Laboratory, Massachusetts Institute of Technology, Cambridge, MA 02139, USA

*Corresponding author: jfrb@mit.edu; 77 Massachusetts Avenue, MIT Rm 41-205, Cambridge, MA 02139, USA

Abstract. The fuel carbon pathway for the cold-start first cranking cycle in a gasoline-direct-injection (GDI) engine is characterized quantitatively. The engine is fired for a single cycle in one cylinder at a specified cranking speed and at a coolant temperature of 20°C. The fuel carbon is accounted for from measurements of the exhaust carbon (CO2, CO, and HC). The remaining carbon is assumed to go into the oil and crank case. The parameters studied are the amount of injected fuel, the injection timing, the intake pressure, the injection pressure and the cranking speed. Substantial fuel enrichment is needed to produce stable combustion in the first cycle, with significant residual fuel that goes into preparing the mixture of the second cycle and into the oil and crank case. The first cycle HC emissions as a fraction of the fuel are not sensitive to the fuel enrichment, the manifold absolute pressure, and the injection pressure.

Keywords: Cranking, cold start, GDI, hydrocarbon emissions, fuel accounting

1. Introduction

Lowering fuel consumption and CO2 emissions have become the leading agenda for engine development around the world. This eminence is a result of the stringent fuel economy standards aiming for CO2 emissions to around 100gCO2/km (normalized to NEDC) by 2025 [1]. Turbocharged gasoline-direct-injection (GDI) is a promising technology for spark ignition (SI) engines towards this goal. Compared to the naturally aspirating engines of the same performance, the turbo-charged GDI engines have a CO2 reduction potential of up to 27% according to EPA estimates [2]. The market penetration of GDI engines in the USA is up to 30% of the gasoline engine sales in 2012 [3]; a prognosticated market share of up to 97% by 2025 has been reported [2].

The GDI technology has both advantages and drawbacks. Benefits include better knock resistance through charge cooling of the fuel spray, higher volumetric efficiency via cooling the intake air, and potential of pumping loss reduction via stratified lean part load operation. Directly injecting fuel into the combustion chamber, however, results in significant emission challenges for unburned hydrocarbons and particulates because of the substantial presence of liquid films on the combustion chamber walls. The challenges are particularly severe during the cold-start and warm-up phases of the engine operation. Because of the inactivity of the catalyst during a significant part of the cold-start, up to 80% of the allowed HC emissions in the FTP stem from the first minute of operation [4]. The contribution from the cranking process is particularly important because cranking has two distinctive characteristics which are more severe than the rest of the certification cycle: coldest cylinder wall temperatures and lowest engine speed.

Direct injection onto the cold cylinder walls results in formation of fuel films with low evaporation. As a result, the amount of fuel that needs to be injected for the combustion events is significantly higher than what is required to prepare a stoichiometric charge with full evaporation. The large residual fuel that escapes the main combustion event becomes a source of HC emissions. Furthermore, the cold walls reduce post flame oxidation of the unburned HC.

The low engine cranking speed results in reduced charge motion, which in turn, causes poor air-fuel mixing. To avoid flame extinction due to lean pockets in the non-uniform mixture, it is necessary to even further increase the amount of injected fuel, with the associated negative impact on the HC emissions.

We have embarked on a systematic study of the emissions in the cold cranking process of a GDI engine. This paper reports on the fuel carbon pathway and HC emissions in the first cycle of the cranking process. (The associated particulate emission is negligibly small in terms of the fuel carbon mass. The subject is addressed in a separate paper [5].) A carbon accounting analysis is used to deconstruct the amounts of fuel participating in combustion, being exhausted as HC emissions, staying in the combustion chamber for the second combustion event, and being absorbed in the oil or lost through blow-by.
2. Experimental Methodology

2.1 Engine and Engine Control

The engine used for this study is a GM Ecotec LNF engine; specification is given in Table 1. The engine features a centrally mounted spark plug and 4 valves per cylinder with variable valve timing for both intake and exhaust valves. The injection system consists of 4 side-mounted multi-hole injectors from Bosch driven by driver from Siemens.

The fuel, intake air and coolant temperature are fully conditioned. For the fueling system, a hydro-pneumatic accumulator pressurized by nitrogen was used, allowing pressures ranging from 30 to 110 bar to be set independent of the engine operation.

The engine control is achieved by an in-house developed controller, allowing a full customization of the engine parameters such as injection and spark timing, injection duration, split injection ratio and intake/exhaust cam phasing.

<table>
<thead>
<tr>
<th>Table 1. Engine geometry and features</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Basic geometry</strong></td>
</tr>
<tr>
<td>Engine type</td>
</tr>
<tr>
<td>Displacement [cc]</td>
</tr>
<tr>
<td>Bore [mm]</td>
</tr>
<tr>
<td>Stroke [mm]</td>
</tr>
<tr>
<td>Connecting rod [mm]</td>
</tr>
<tr>
<td>Compression ratio</td>
</tr>
<tr>
<td><strong>Valve timing during cranking</strong></td>
</tr>
<tr>
<td>IVO</td>
</tr>
<tr>
<td>IVC</td>
</tr>
<tr>
<td>Max. intake valve lift</td>
</tr>
<tr>
<td>EVO</td>
</tr>
<tr>
<td>EVC</td>
</tr>
<tr>
<td>Max. exhaust valve lift</td>
</tr>
<tr>
<td><strong>Fuel</strong></td>
</tr>
<tr>
<td>Tier II EEE certification fuel</td>
</tr>
<tr>
<td>carbon mass fraction 86.5%</td>
</tr>
</tbody>
</table>

2.2 Experiment Description

The engine crank-start is a highly transient process, with engine speed variations of up to 200 rpm within a combustion cycle. The sequence of events depends heavily on the first combustion event. Therefore, as a first step to fully understand the cold cranking emissions, the first firing cycle is the subject of this study.

To recreate the conditions during cold cranking, the engine coolant, engine oil, fuel and intake air temperatures were kept at 20° C. The results of a 1-D simulation performed with a commercially available software show a cylinder liner temperature of 25°C, a piston crown temperature of 29°C and a piston skirt temperature of 26°C under the steady-state motoring conditions at 280 rpm. Even though the engine is equipped with variable cam timing, both camshafts were kept at their parked positions (Table 1) since the low oil pressure at cranking speed does not allow the operation of the system. The parked position of the camshafts corresponds to a negative valve overlap of 20° CA.

The experiment started with the engine being motored at the desired cranking speed (280 rpm except for the engine speed sweep experiments) until the temperature and exhaust HC concentration reached steady state to ensure the purging of residual hydrocarbons stored in the engine and lubricant, and to measure the background HC concentration. The throttle plate position was kept at a fixed position. Except for the MAP sweep experiments, the position corresponded to that at fast idle (2 bar NIMEP at 1200 rpm). The resulting MAP at cranking was at 0.9 bar, which was slightly lower than the typical value (1 bar) for the actual first cranking cycle because the engine had been motored for some time. (It will be shown in the MAP sweep experiments that this difference has no material impact on the results.) After the purge, a single injection and ignition event took place in cylinder #4: a metered amount of fuel was injected at 50 bar followed by combustion. In real applications the fuel is pressurized by a positive displacement pump, and therefore the 1 st fuel injection event is heavily dependent on engine speed and on the pressure fluctuations caused in the fuel rail. In the experimental setup used in this study, the high pressure hydro-pneumatic accumulator ensured a constant fuel pressure and minimized the pressure fluctuations during injection.

After the single combustion event the engine continued to be motored at cranking speed and the exhaust stream measured, until the steady state was achieved again (Figs. 1a and 1b). Throughout
the complete experiment, the exhaust composition was measured and recorded during this time interval with 1° crank angle resolution. This procedure was repeated 5 times for each experimental condition and the results reported are the averaged values.

The exhaust composition was measured with fast-response analyzers, sampling directly from the exhaust runner of the fired cylinder (cylinder #4). The hydrocarbon concentration was measured using a fast FID unit (Cambustion model HFR400), with a response time $t_{10:90}$ of 10ms, a $< \pm 1\%$ linearity over the full scale (50000 ppmC1), and a sampling position 6cm from the exhaust valve. The CO and CO$_2$ concentration was measured with a fast NDIR unit (Cambustion model NDIR500), with a response time $t_{10:90}$ of 20ms, a $< \pm 2\%$ linearity over the full scale (15%CO$_2$ & 10%CO respectively) and a sampling position 8cm from the exhaust valve. The $\lambda$ value was not directly measured, since the response time of conventional $\lambda$ sensors is too slow (~150 ms). The $\lambda$ values reported were calculated from carbon balance using the CO and CO$_2$ measurements [6].

![Engine speed (a), cylinder pressure (b) and normalized cumulative emissions (c) as a function of the cycle number for a representative single fire experiment. A flow diagram of the engine setup is shown in (d).](image)

### 2.3 Fuel Accounting

A fuel carbon accounting analysis is performed to quantify the fuel carbon participating in combustion, the amount being exhausted as unburned hydrocarbons, and the amount that cannot be accounted for; the latter represents the fuel that goes into engine oil dilution and blow-by losses. The analysis is done by translating the fuel and exhaust mass flows to equivalent carbon mass flows, and then performing a control volume analysis around the cylinder (Eq. 1). (See Appendix 1 for notation.)

$$m_{C,\text{in}} = m_{C,\text{out}} + m_{C,\text{Eng}}$$  \hspace{1cm} (1)

The “in” flow corresponds exclusively to the fuel carbon injected, while the “out” flow encompasses the carbon mass flow due to the CO$_2$, CO and HC content in the exhaust over the entire experiment. The majority of the CO$_2$ and CO outflow occurs during the 1$^{\text{st}}$ cycle exhaust stroke. On the other hand, most of the HC outflow happens after the 1$^{\text{st}}$ cycle due to the evaporation of fuel films and the desorption of HC from the oil layer (see Fig. 1c for details). The “engine” component of the carbon accounting represents the carbon mass that cannot be accounted for solely by the intake and exhaust flows; this amount corresponds to the fuel that goes into oil dilution and blow-by losses.

To relate the concentration measurements to mass flows, the exhaust mass flow has to be correctly synchronized with the fast analyzer signals. The exhaust mass flow was calculated using the cylinder pressure and the piston position. The model assumes the gas is ideal, and that the charge within the cylinder expands isentropically to expel the exhausted gas. Then the exhaust gas flow rate is obtained from continuity.
\[ \dot{m}_{\text{exh}} = -\left( \frac{1}{\gamma \cdot p_{\text{cyl}}} \cdot \frac{dP_{\text{cyl}}}{dt} + \frac{1}{V_{\text{cyl}}} \cdot \frac{dV_{\text{cyl}}}{dt} \right) \cdot m_{\text{cyl}} \] (2)

To align the exhaust flow with the fast analyzer measurements, the transit time of the exhaust gas from the exhaust valve to the sampling point and the internal delay of the instrument need to be considered. The transit time is determined from the cumulative exhaust gas volume between the sampling point and the exhaust valve. The carbon mass flow due to each constituent \( y \) can then be calculated by multiplying the measured wet molar concentration of the component by the exhaust molar flow using \( MW_{\text{exh}} = 28.9 \text{ g/mol} \) and the molecular weight of carbon \( (MW_C) \):

\[ m_{c,y} = \dot{x}_y \cdot \frac{\dot{m}_{\text{exh}}}{MW_{\text{exh}}} \cdot MW_C \] (3)

For the motored cycles after the single cycle firing, the cylinder pressure is lower than the exhaust pressure when the exhaust valve opens. Thus the exhaust flow is initially backwards until the pressure equilibrates and the displacement flow commences (Fig. 2). By measuring the HC mole fraction during the reverse flow, it is possible to account for the HC content flowing from the exhaust manifold into the cylinder during this period, thus avoiding to double-count the emission for the reverse flow in the motored cycles.

![Fig. 2. Exhaust flow of the fired cycle and a motored cycle](image)

2.4 Fuel Enrichment Factor

Since not all of the injected fuel goes into the charge mixture, a large amount of fuel has to be injected to achieve stable combustion in the first cycle. To quantify the amount of additional fuel required, a fuel enrichment factor (FEF) based on a speed-density calibration is defined:

\[ FEF = \frac{m_{f,\text{cyl}}}{V_{\text{cyl}} \cdot \eta_{\text{vol}} \cdot \rho_{\text{int}} \cdot (F/A)_{\text{stoich}}} \] (4)

Thus the fuel mass injected \( (m_{f,\text{cyl}}) \) is equal to FEF times the amount required to prepare a stoichiometric mixture with the inducted air. The volumetric efficiency is referenced to the intake manifold conditions and takes the value at cranking speed. Under motoring conditions, the value was measured to be 80%, which corresponded to a fuel mass of 29.4mg for a FEF=1 at an intake temperature and pressure of 293K and 900mbar respectively.

2.5 Experimental Matrix

The scope of this paper focuses on six different engine parameters using a one-variable-at-a-time approach. The variables studied are: First cycle fuel enrichment factor (FEF, as defined in section 2.4), injection timing (single injection was used), spark timing, first cycle intake manifold pressure, fuel pressure, and engine cranking speed (Table 2). For the sweep of each parameter, the remaining parameters were held fixed at the nominal values.
Table 2. Experimental scope. For each sweep, the remaining parameters were kept at the nominal values

<table>
<thead>
<tr>
<th>Variable</th>
<th>Sweep range</th>
<th>Nominal value</th>
</tr>
</thead>
<tbody>
<tr>
<td>FEF [-]</td>
<td>1.7 to 3.5 with 0.2 increments</td>
<td>2.5</td>
</tr>
<tr>
<td>Injection timing [°CA ATDC]</td>
<td>30 to 315 with 15 °CA increments</td>
<td>90</td>
</tr>
<tr>
<td>Spark timing [°CA ATDC]</td>
<td>-45 to 20 with 5 °CA increments</td>
<td>-10</td>
</tr>
<tr>
<td>MAP [mbar]</td>
<td>650 to 1000 with 50 mbar increments</td>
<td>900</td>
</tr>
<tr>
<td>Fuel pressure [mbar]</td>
<td>30 to 110 with 20 bar increments</td>
<td>50</td>
</tr>
<tr>
<td>Cranking speed [RPM]</td>
<td>280, 700, 1200</td>
<td>280</td>
</tr>
</tbody>
</table>

3. Results and Discussion

The goal of the experiment is to deconstruct the pathways for the single cycle injected fuel and to examine how the engine parameters affect the pathways. As such, the individual carbon exhaust mass flows due to each component (CO₂, CO, and HC) are integrated for the firing and the subsequent motoring cycles, and the cycle-resolved results are summarized in three sets of numbers:

- The first set is the HC emissions respectively of the first, second and sum of the third-and-beyond cycles. The 1st (firing) cycle HC emissions give information of the emission performance; the 2nd (motoring) cycle HC emissions indicate the amount of the in-cylinder retained fuel from the first cycle that would contribute to the combustible mixture of the second cycle. The HC emissions from the 3rd cycle on, are caused by the evaporation of fuel films and the desorption of HC from the oil layer. The accounting of this HC content is necessary to estimate the amount of fuel going into the engine oil and crankcase through a total carbon balance.
- The second set is the integrated CO2 and CO over the complete set of recorded cycles (approximately 50). These carbons represent the burned fuel, and are used to compute the overall combustible mixture λ value.
- The third set is the difference between the fuel carbon from the injected fuel and the cumulative carbon from the CO2, CO and HC measurements over the complete set of recorded cycles. This difference, labelled as the unaccounted for fuel carbons, represents the fuel that goes into the engine oil and crankcase.

3.1 FEF sweep

The outputs of the single-cycle-fired engine as a function of the FEF are shown in Fig. 3. The NIMEP increases with FEF (Fig. 3a), since more fuel goes into the charge mixture. This observation is confirmed by the decrease in λ values with FEF (Fig. 3b). The progression, however, is not linear – the increment of the fuel going into the combustible charge is large at low FEF, but the increment diminishes with increase in FEF.

Fig. 3. Outputs of the single-cycle-fired engine as function of FEF as follows: a) 1st cycle NIMEP. b) 1st cycle lambda. c) 1st cycle CO emissions. d) 1st cycle relative HC emissions. e) 2nd cycle relative HC emissions. f) 2nd cycle HC relative emissions as a percentage of the fuel mass for lambda=1. Dashed lines correspond to a one standard deviation envelope.
Since the $\lambda$ value is calculated from carbon balance using the exhaust carbon values [6], it may be interpreted as the overall $\lambda$ value of the burned mixture. For FEF increasing from 1.7 to 3.5, $\lambda$ decreases from 2.8 to 1. Thus for the whole range of FEF, the burned mixture is overall lean. However, there is substantial CO emission (Fig. 3c), which is especially high (8 mg which corresponds to $\sim$4% wet CO) at overall $\lambda = 1$. The high value suggests that the mixture was significantly inhomogeneous and rich burning pockets were present during combustion.

For HC emissions, Fig. 3d shows the relative (in terms of fuel carbon) HC emissions with respect to the fuel injected. With the exception of the point at FEF=1.7 which resulted from a partial burn, the fraction of the fuel injected coming out of the engine as unburned HC remains roughly constant with increasing FEF at values between 3-4%, with a slight increase for higher FEF. This first cycle HC emissions will contribute directly to the tailpipe emissions, since the catalyst has not reached light-off temperature.

Fig. 3e shows the relative HC emissions for the second cycle, which is a motored cycle. This value represents the fuel fraction that would be available for second cycle combustion, and thus, it can be used to correct the second injection event, to avoid over enrichment. The relative HC emissions for the second cycle stay roughly constant at around 8% for FEF>1.9. To understand the impact that the residual HC could have on the second combustion event, the HC emissions are also plotted relative to the fuel necessary (based on a speed-density calibration) to achieve stoichiometric combustion. Fig. 3f shows that more than 20% of the fuel necessary to achieve stoichiometric combustion for the second combustion event is already in the cylinder in the form of residual HC for FEF>2.3.

The fuel carbon pathway as a function of FEF is shown in Fig. 4. The combined CO$_2$ and CO fraction, which represents the fraction of fuel carbon burned, is approximately 32%. Thus for the range of FEF values tested for the first cranking cycle, roughly 1/3 of the fuel participates in combustion. The HC emissions coming out of the engine on and after the third cycle, account to approximately 30%. Approximately 25% of the fuel cannot be accounted for. This fuel ends up in the oil or crankcase.

3.2 Injection timing sweep

Injection timing has a strong influence on mixture formation. Under warm operation conditions, early injection timings result in more homogeneous mixtures, while late injection timings result in higher heterogeneity and charge turbulence prior to ignition [7]. At cold engine temperatures, the injection timing also determines the amount of liquid and location of the fuel impinging on the walls of the combustion chamber, resulting in liquid fuel films.
The first zone, characterized by high CO emissions (Fig. 5c). The first zone, characterized by interaction between the injection spray and the intake valve. The maximum HC emissions (4.5% of injected fuel) occur between a SOI of 120°CA and 135°CA corresponding with the maximum lift of the intake valve. To the left and right of that maximum, HC emissions follow the intake valve lift, achieving a minimum of 1.9% of injected fuel when the intake valve lift is lower than 4mm. The third zone corresponds to SOI=180 to 240°CA, that is in the initial part of the compression stroke. In this zone there is not only a better utilization of the fuel, but also a flat region at low values of HC emissions. Finally, the fourth zone corresponds to late injection timings into the compression stroke, where the interaction between piston and injection spray gains importance, resulting in high HC emissions. Contrary to the first zone, the interaction between the fuel spray and the upwardly-moving piston occurs just before the start of combustion, resulting in a rich burning mixture around the spark plug. From the high CO emissions observed in this SOI region, the higher HC emissions are inferred to be the result of incomplete combustion.

The fuel available for the second cycle combustion varies significantly with the SOI of the first cycle (Fig. 5d), achieving a minimum value in the same range as does the first cycle HC emissions, that is, SOI during the early compression stroke.

Figure 6 shows the individual contributions of CO₂, CO and HC to the fuel carbon accounting. The combined total CO₂ and CO emissions show a monotonic increase with late injection timings. Although the combustion efficiency decreases with later injection, more fuel is taking part in combustion, suggesting that a lower FEF could be used to achieve an ignitable fuel-air mixture. The lower HC emissions after the third cycle, and the lower fraction of the unaccounted fuel for late injection, suggest that the diffusion of fuel in the oil is reduced, since the injection spray interacts less with the cylinder liner.
3.3 Ignition timing sweep

The use of retarded spark timing has been established as a common practice in the industry to reduce both catalyst light-off time and engine-out HC emissions [4,9]. In contrast, the influence of ignition timing on the first combustion cycle characteristics has only been examined in a few studies, for a reduced set on spark timings before TDC [10, 11].

The first cycle NIMEP as a function of spark timing is shown in Fig. 7a. For both extremes of the spark timing spectrum there is a reduction in work output, with a higher impact on the side of late ignition. The maximum NIMEP is achieved in a plateau region from ignition =−25 to −5˚CA ATDC. Due to the low, but rapidly changing, engine speed, the usual notion of proper combustion phasing for MBT (CA50=7˚CA ATDC and Pmax=15˚CA ATDC) does not apply [12]. The NIMEP plateau region is a result of the competing effects between heat transfer to the cold cylinder walls and the location of the maximum cylinder pressure.

As shown in Fig. 7b, the CO production of the first combustion cycle has a high sensitivity to spark timing. Given that the amount of fuel, SOI, fuel pressure and engine speed were kept constant, mixture formation was similar. The production of CO is a result of locally rich mixture being burned. Retarding the spark has two effects. First, there is more time for mixing so there is less fuel rich pocket for CO production. Second, the charge temperature in the expansion stroke is higher so that post-flame oxidation of CO is promoted. Both effects reduce the CO emission.

The first cycle HC emissions are also favored by later ignition timing, with a lower sensitivity compared to the CO production (Fig. 7c). The higher burned gas temperatures resulting from spark retard, favor the post-flame oxidation of unburned HC in the bulk gas [13]. The net effect is a decrease of HC as a fraction of the fuel from ~3.7 to 2.8%. The fuel available for the second cycle combustion...
(Fig. 7d) increases slightly with retarded spark timing. The lower HC emissions after the third cycle and the lower fraction of the unaccounted fuel for advanced spark timings (Fig. 8) are a direct consequence from the increased fraction of fuel taking part in combustion during the first cycle.

![Fuel pathway as a function of spark timing](image)

**Fig. 8** Fuel pathway as a function of spark timing

### 3.4 Manifold Absolute Pressure sweep

Lower MAP is beneficial to mixture preparation since less vaporized fuel is needed to form a combustible mixture [14]. During cranking, MAP is a function of engine speed and throttle position. In the experiment, the engine speed was kept constant and the throttle position was varied, achieving a minimum MAP value of 650mbar at a cranking speed of 280rpm for a fully closed throttle.

Since the engine is calibrated using a speed-density approach, at the same FEF, the injected amount of fuel, and the cycle work output, are functions of MAP (Figs. 9a and 9c). To assess the fuel evaporation behavior at different MAP values, the gross indicated fuel conversion efficiency $\eta_{i,g}$ may be used. In the MAP sweep, the combustion phasing, and hence the gross thermal efficiency of the burned fuel, are approximately the same, hence $\eta_{i,g}$ becomes the ratio of the mass of fuel burned to the injected fuel – a measure of the fuel utilization efficiency. For increasing MAP values from 650 to 1000mbar, $\eta_{i,g}$ decreases slightly from 9.5% to 8.8%. Thus the fuel utilization decreases with more injected fuel.

![Outputs of the single-cycle-fired engine as function of MAP](image)

**Fig. 9.** Outputs of the single-cycle-fired engine as function of MAP as follows: a) 1st cycle NIMEP. b) 1st cycle gross indicated efficiency. c) 1st cycle injected fuel mass. d) 1st cycle HC relative emissions. e) 1st cycle HC absolute emissions. Dashed lines correspond to a one standard deviation envelope.

The relative first cycle engine-out HC emissions only increases modestly with MAP (Fig. 9d). In absolute terms, however, lower MAP allows a reduction in the injected fuel amount without over-leaning of the mixture, and resulting in lower absolute HC emissions for the first cycle (Fig. 9e).

The distribution of the fuel carbon as a function of MAP is shown in Fig. 10. As MAP increases, while the relative HC emissions stay relatively constant for first and second cycle, the amount for third
cycle on increases. This observation may be explained by the increase of the oil layer temperature because more fuel is burned with the increased MAP so that there is more desorption in the subsequent motoring cycle. The corresponding decrease of the oil dilution and blow-by amount lends support to this explanation.

![Graph showing fuel unaccounted for and CO2, HC after 3rd cycle, HC 2nd cycle, 1st cycle, CO as a function of MAP.](image)

**Fig. 10.** Fuel pathway as a function of MAP

### 3.5 Fuel pressure sweep

Injection pressure influences the mixture formation of GDI engines through different mechanisms. First, a higher fuel pressure renders smaller fuel droplet diameter. The smaller droplets have better evaporation. They also have a lower inertia and exchange momentum readily with the charge; thus spray penetration and wall impingement are reduced. Second, higher fuel pressures result in higher nozzle velocities, and therefore higher momentum and penetration of the injection spray. The balance between these conflicting effects has been studied in the fuel pressure range from 5 to 40 bar [15, 16, 17]. In this range, droplet size decreases sharply with increasing injection pressure from 100 µm at 5 bar to roughly 30 µm at 40 bar and the advantages in reduction of droplet size are more significant than the disadvantages of increased spray penetration, resulting in better mixture formation and lower emissions. In this study, the fuel pressure range of 30 to 110 bar is examined. The lower value is what modern fuel pump could supply for the first injection cycle [15]; however, only modest reductions in fuel droplet size are achievable in this pressure range (e.g. from ~35 to 18 µm, as reported by Landenfeld et al. [15]). The fuel pulse width was adjusted to account for the different injection pressure.

![Graph showing NIMEP, CO mass emissions, and HC out, 1st/Fuel in as a function of fuel pressure.](image)

**Fig. 11.** Outputs of the single-cycle-fired engine as function of fuel injection pressure as follows: a) 1st cycle NIMEP. b) 1st cycle CO emissions. c) 1st cycle HC relative emissions. d) 2nd cycle HC relative emissions. Dashed lines correspond to a one standard deviation envelope.

The NIMEP increases with injection pressure (Fig. 11a). This observation suggests that mixture formation is favored by higher injection pressures. The CO emission also increases (Fig. 11b), indicating
that there may be more fuel rich regions. The HC emission as a fraction of the fuel, however, is not sensitive to the fuel pressure (Fig. 11c).

The second cycle HC emissions (Fig. 11d), as well as the HC emissions after the third cycle (Fig. 12), indicate that despite the higher fuel pressure and the corresponding potential increase in spray penetration, the fuel remaining in the combustion chamber after the first combustion event decreases with higher fuel pressures. The overall effect is that for injection pressure increasing from 30 to 110 bar, the fraction of fuel burned (as indicated by the fuel carbon used) only increases modestly, from 33 to 37%; the first cycle HC emissions as fraction of fuel remains approximately constant.

![Fig. 11. Fuel pathway as a function of fuel pressure](image)

**Fig. 12.** Fuel pathway as a function of fuel pressure

### 3.6 Engine speed sweep

With start-stop systems and hybrid powertrains, the engine cranking speed is no longer limited by the low speed of conventional starters. Hence a further dimension for engine start optimization could be explored. Three different engine speeds were assessed. These speeds correspond to conventional cranking speed (280rpm), idle (700rpm), and fast idle (1200rpm). The results are shown in Figures 13 and 14.

Since the mass burn per crank angle changes with engine speed due to the effect of crank angle speed and turbulence [18], combustion phasing is a function of engine speed. To decouple the effect of engine speed from that of combustion phasing on mixture preparation, the spark timing was modified for each engine speed to achieve a comparable CA50 (Fig. 13b).

![Fig. 13. Outputs of the single-cycle-fired engine as function of cranking speed timing as follows: a) 1st cycle NIMEP. b) 1st cycle CA50. c) 1st cycle HC relative emissions. d) 2nd cycle HC relative emissions. Dashed lines correspond to a one standard deviation envelope](image)

**Fig. 13.** Outputs of the single-cycle-fired engine as function of cranking speed timing as follows: a) 1st cycle NIMEP. b) 1st cycle CA50. c) 1st cycle HC relative emissions. d) 2nd cycle HC relative emissions. Dashed lines correspond to a one standard deviation envelope.

With the cranking speed increase from 280 to 1200 rpm, the NIMEP increases by 49% (from 5.5 to 8.2 bar); see Fig. 13a. The relative HC emissions, however, only increases modestly, from 3.2 to
3.8%; see Fig. 13c. The HC retained for the 2\textsuperscript{nd} cycle remains approximately the same. However, it should be noted that if a certain value of NIMEP is targeted for the first cycle, the amount of fuel can be reduced for higher engine speeds, resulting in lower absolute HC emissions.

The fuel utilization, as measured by the fraction of fuel carbon conversion to CO and CO\textsubscript{2}, improves by 20\% (from 0.35 to 0.42); see Fig. 14. This increase is a result of the improved mixture formation from the higher turbulence, which more than compensates for the shorter preparation time at the higher speed. This improvement partially explains the amount of NIMEP increase. The remaining improvement is due to the reduction in heat loss with increase of engine speed.

![Fuel pathway as a function of cranking speed](image-url)

**Fig. 14.** Fuel pathway as a function of cranking speed

4. Conclusions

The first combustion cycle of a gasoline direct injection engine at cranking speed is characterized under cold start condition (air and coolant temperatures both at 20° C). Using a fuel carbon accounting analysis, this study quantifies the amount of fuel participating in combustion, the amount of HC emissions from the first combustion cycle, the residual fuel that goes into the mixture of the second combustion cycle, and the residual fuel that goes into the lubrication oil and crankcase.

A one-parameter-at-a-time approach is used to isolate the effects of the different parameters on the first cycle combustion and emissions behaviors. The parameters being studied are: fuel enrichment, injection timing, ignition timing, manifold absolute pressure (MAP), fuel injection pressure, and engine speed. The key findings are:

- When the injected fuel is increased from 1.7 to 3.5 times the amount required for forming a stoichiometric mixture with the ingested air, the overall \( \lambda \) of the burned mixture decreases from 2.8 to 1. Although the \( \lambda \) value is lean, there is significant CO emission, indicating that the charge is non-uniform with fuel rich regions. Approximately 1/3 of the fuel participates in combustion. The first cycle relative HC emissions (to the injected fuel), however, are not sensitive to the enrichment if there is no partial burn or misfire. The residual fuel contributes to 10 to 30\% of the amount required to constitute a stoichiometric mixture for the second cycle.

- The injection timing determines the type and degree of interaction between the injection spray and the elements of the combustion chamber. For this particular engine configuration, injection during the early compression stroke results in lower relative HC emissions, due to reduced interaction with the intake valve and piston crown.

- Retarded spark timing provides a hotter charge in the expansion stroke and benefits post-flame oxidation. The results are reduced CO and HC emissions at the cost of lower work output.

- At lower MAP, less air is ingested, thus less fuel is needed to prepare a mixture of a given stoichiometric ratio. The fuel utilization (ratio of the burned fuel carbon to the injected fuel carbon) improves with reduced fuel. The relative HC emissions are not sensitive to MAP. Because of the reduced fuel required at the lower MAP, the absolute HC emissions are reduced.
• Fuel utilization increases with injection pressure. The relative HC emissions, however, are not sensitive to injection pressure.

• Higher engine speed promotes turbulence. The improvement in turbulent mass transfer more than compensates for the reduction in mixture preparation time; hence fuel utilization improves with engine speed. The relative HC emissions increase modestly with higher engine speeds. At the same combustion phasing, NIMEP increases with engine speed because of the better fuel utilization and of the heat loss reduction. For a targeted NIMEP value, therefore, less fuel is needed at the higher engine speed, resulting in reduction of the absolute HC emissions.

**Acknowledgements**

The authors would like to acknowledge the support for this research by Borg-Warner, Fiat Chrysler Automobiles, Ford Motor Company, and General Motors Company through a Consortium on Engine and Fuels Research.

**References**


Appendix 1

Notation

$a_{TDC}^{compression}$ After top dead center of the compression stroke

$a_{TDC}^{intake}$ After top dead center of the intake stroke

EPA Environmental Protection Agency

$\eta_{vol}$ Volumetric efficiency

$(F/A)_{stoich}$ Stoichiometric fuel to air mass ratio

FEF Fuel enrichment factor

FTP Federal test procedure

$\gamma$ Heat capacity ratio

GDI Gasoline direct injection

HC Hydrocarbon

$\lambda$ Air equivalence ratio

MAP Manifold absolute pressure

$\dot{m}_{c,in}$ Inflow of carbon in the form of fuel

$\dot{m}_{c,Eng}$ Unaccounted carbon

$\dot{m}_{c,out}$ Outflow of carbon in the form of emissions

$m_{cyl}$ Mass of carbon

$\dot{m}_{exh}$ Exhaust mass flow

$m_{f,cyl}$ Fuel mass injected

$MW_{exh}$ Molecular weight exhaust

NDIR Non-dispersive infrared

NEDC New European driving cycle

NIMEP Net indicated mean effective pressure

$p_{cyl}$ Cylinder pressure

$\rho_{int}$ Intake density

SI Spark Ignition

SOI Start of injection

$t$ Time

$t_{10-90}$ Time between 10% and 90% of the full response

$V_{cyl}$ Cylinder volume

$\tilde{x}_y$ Mole fraction of species $y$