

On the Nature of Particulate Emissions from DISI Engines at Cold-Fast-Idle

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ABSTRACT

Particulate emissions from a production gasoline direct injection spark ignition engine were studied under a typical cold-fast-idle condition (1200 rpm, 2 bar NIMEP). The particle number (PN) density in the 22 to 365 nm range was measured as a function of the injection timing with single pulse injection and with split injection. Very low PN emissions were observed when injection took place in the mid intake stroke because of the fast fuel evaporation and mixing processes which were facilitated by the high turbulent kinetic energy created by the intake charge motion. Under these conditions, substantial liquid fuel film formation on the combustion chamber surfaces was avoided. PN emissions increased when injection took place in the compression stroke, and increased substantially when the fuel spray hit the piston. A conceptual model was established for the particulate matter (PM) formation process in which PM is formed by pyrolysis after the normal premixed flame passage in fuel rich plumes originating from liquid films on the cylinder walls. The pyrolysis process is supported by heat conducted from the hot burned gases outside the plume and by the energy released by the pyrolysis reactions. Thus, the "pool fire" often observed is not a diffusion flame since the small amount of residual oxygen in the burned gases cannot support such a flame. The luminosity is radiation from the hot soot particles which are not oxidized after being formed in the pyrolysis reactions. This model was supported by the PN data obtained from sweeping the charge equivalence ratio from lean to rich.

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INTRODUCTION

Direct injection (DI) is an attractive option for spark ignition (SI) engines. The in-flight fuel evaporation cools the charge, thereby improving the volumetric efficiency and extending the knock margin. These attributes have made the technology an enabler for highly boosted, downsized engines which offer substantial gains in fuel economy [1, 2, 3, 4].

An important issue for DISI engines is the particulate matter emissions [5, 6, 7, 8]. Particulate number (PN) emissions for small particles are especially of concern, as stringent limits have been established [9]. PN emissions during cold start in DISI engines are substantially higher than those in port-fuel-injection (PFI) engines [5]. For the latter, when the intake valve opens at part-load, the back flow from the cylinder to the intake manifold blows the liquid fuel in the port back into the intake runner, thereby suppressing liquid fuel film formation in the combustion chamber [10]. In DISI engines, however, the in-cylinder fuel jet directly deposits liquid fuel on the cylinder walls. The fuel vapor that emerges from the fuel films produces

fuel rich regions which are responsible for PM formation. Visible soot particle radiation, often referred to as "pool fires", has been observed [11, 12, 13].

The goals of this paper are:

1. To connect the PN emissions from a production DISI engine to the geometry of the injection process; and
2. To establish a conceptual model describing the process through which particulates form from the vapor emerging from wall fuel films.

EXPERIMENTAL SET UP

Engine

The experiments were carried out in a production DISI engine (the GM LNF engine). The engine specifications are detailed in [Table 1](#). The engine has cam phasers for both the intake and exhaust valves; the valve timings were fixed at the values given in [Table 1](#). There was essentially no valve-overlap. The spark and injection timings will be specified in the descriptions of the different experiments that were performed.

Table 1. Engine specifications for the GM 2L LNF DISI engine.

Engine type	Inline 4 cylinder
Displacement [cc]	1998
Bore [mm]	86
Stroke [mm]	86
Compression ratio	9.2:1
Fuel system	Side-mounted direct injection
Valve configuration	16 valve DOHC,
Intake valve open/close (0.25 mm lift)	1° / 231° (atdc-intake)
Exhaust valve /open close (0.25 mm lift)	-222 / 0° (atdc-intake)
Max lift (both intake and exhaust)	10.3 mm

The injection geometry is shown in Fig. 1. There is a cup in the piston to allow wall-guided injection. The fuel injector is located between the two intake valves. The valves do not materially interfere with the spray since there is no substantial change in the engine behavior (in terms of PN and HC emissions) when the start-of-injection (SOI) timing is varied over the lift profile.



Figure 1. Piston and injector arrangement ©GM Company.

The fuel used in these experiments was Haltermann HF0 437 calibration gasoline with RON = 96.6 and RVP = 0.61 bar (9 psi). The fuel was supplied by a pressurized cylinder at 5 MPa. The fuel temperature was regulated by a single pass counterflow shell-in-tube heat exchanger and was maintained at a temperature of 20°C.

The engine coolant temperature (ECT) was regulated by a chiller/ heater system. The nominal operating temperature was 20°C. Experiments at ECT = 80°C were also conducted to assess the sensitivity of the results to ECT.

Data were obtained at 1200 rpm and a net indicated mean effective pressure (NIMEP) of 2 bar. This condition is typical of cold-fast-idle in the startup process leading to catalyst light off. Because the operation is at steady state, repeatability and data quality are much better than data obtained during the cranking and speed flare processes. The liner temperature is close to the ECT. The piston temperature is estimated to be 120°C for ECT at 20°C using the simulation results from reference [14]. It should, however be noted that while the piston temperature has reached equilibrium of about 120°C in these experiments, in actual cold start, it takes hundreds of seconds for the piston

temperature to get to steady state. So the piston temperature in the fast idle period in an actual cold start would be significantly lower.

The air equivalence ratio (λ) is set at stoichiometric for cylinder #4, which is the cylinder from which all measurements were made. This λ value was obtained by carbon balance from the CO₂/CO measurements in the exhaust runner of cylinder #4. The engine was operated with the same injection pulse width to all four cylinders. There was a substantial cylinder-to-cylinder λ variation (up to 10%) because of differences in volumetric efficiency such that when λ for cylinder #4 was stoichiometric, the overall λ , as measured by the exhaust UEGO sensor was lean.

Measurements

In-cylinder pressure was measured by a Kistler 6125 pressure transducer with a flame arrestor. Exhaust CO₂/CO was measured in the exhaust runner of cylinder #4 by an NDIR detector (Horiba MEXA 584L).

PN Measurements

Particle size distributions were measured by a scanning mobility particle sizer (TSI 3934 SMPS). The flow into this analyzer first went through an impactor to remove particles larger than 365 nm in diameter. A radioactive source was then used to impart a known electrical charge on the aerosol sample. The particles were then sorted by a mobility classifier and counted by a condensation particle counter. The measurement range was from 9.3 to 365 nm. In the results to be presented, the total particle number (PN) refers to the integral of the particle size distribution from 22 to 365 nm.

To prevent particle agglomeration in the sample and instrument saturation, the sample was diluted by a stream of nitrogen. The dilution ratio was measured by the CO₂ concentration measurements at the exhaust port sampling point by a Horiba MEXA 584L CO₂ meter, and at the SMPS inlet by a more sensitive Licor LI-820 CO₂ analyzer. The typical dilution ratio was approximately 100:1. All of the values reported in this paper have been corrected for dilution.

The exhaust sampling point was at 10 cm from the exhaust valve of cylinder #4. There are significant pressure pulsations at this location. To maintain a steady sampling flow to the dilution system, a heated "T" design similar to that used in the fast-response HC analyzer [15] was used (see Fig. 2). The exhaust gas flow rate is limited by orifice tube 1. The flow through this orifice tube pulsates corresponding to the exhaust pressure. At the inlet of orifice tube 2, however, the pressure is essentially constant at atmospheric; thus, with the dilution tunnel under a slight vacuum, a constant sample flow rate into the dilution system could be maintained. Both orifice tubes have an ID of 0.5 mm and length of 5 cm.

To minimize condensation and particle loss through thermophoresis, the entire system was heated to 200°C.

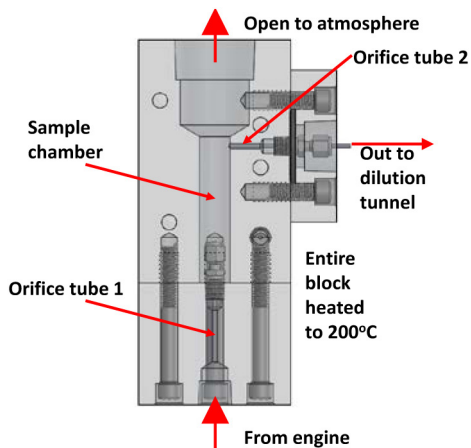


Figure 2. Arrangement to isolate exhaust pressure pulsations in order to achieve a constant sample flow to the sample dilution system.

RESULTS

Single Injection

The PN emissions (integration of the size spectrum from 22-365 nm) were measured at the exhaust runner of cylinder #4 as a function of start of injection (SOI) timing at a fast idle condition (1200 rpm, 2 bar NIMEP) with ECT = 20°C. The injection duration varied from 0.76 to 1.05 ms (5.5 to 7.6° crank angle); the fuel mass varied from 9.2 to 12.3 mg per cylinder per cycle, with the latter value being injected late (330° atdc-intake) in the compression stroke. The exhaust λ from a carbon balance was stoichiometric. Since NIMEP and exhaust λ were kept constant, the change in the amount of fuel supplied reflects the amount of fuel not oxidized and, to a lesser extent, the change in thermal efficiency due to changes in combustion phasing. The unconsumed fuel could go into the lubrication system or exit with the exhaust gas as hydrocarbons.

The PN emissions map as a function of SOI and spark timing is shown in Fig. 3. For injections early in the intake stroke, emissions were high; they intensified with spark retard. Emissions decreased with advanced SOI and attained a very low level when the SOI was in the mid compression stroke. The data at a spark timing of -25° atdc-compression are representative of the general trends and are used for further discussion. (Note that normal cold idle runs at much more retarded spark timing (10 to 20° atdc) to obtain a hot exhaust which facilitates catalyst warm up. However, with retarded spark, the engine becomes unstable at later SOI. The early spark timing has been chosen so that a larger SOI sweep data set could be obtained.)

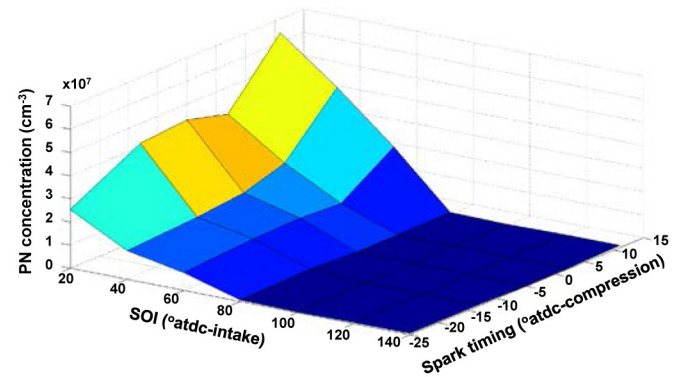


Figure 3. Cold-fast-idle PN emissions as a function of SOI and spark timing; ECT = 20°C.

The PN emissions at -25° atdc-compression spark as a function of SOI are shown in Fig. 4 together with the injection geometry at selected points. The range of SOI timings has been expanded to 330° atdc-intake. Further SOI retard was not attempted because the spark timing is at 335° atdc-intake.

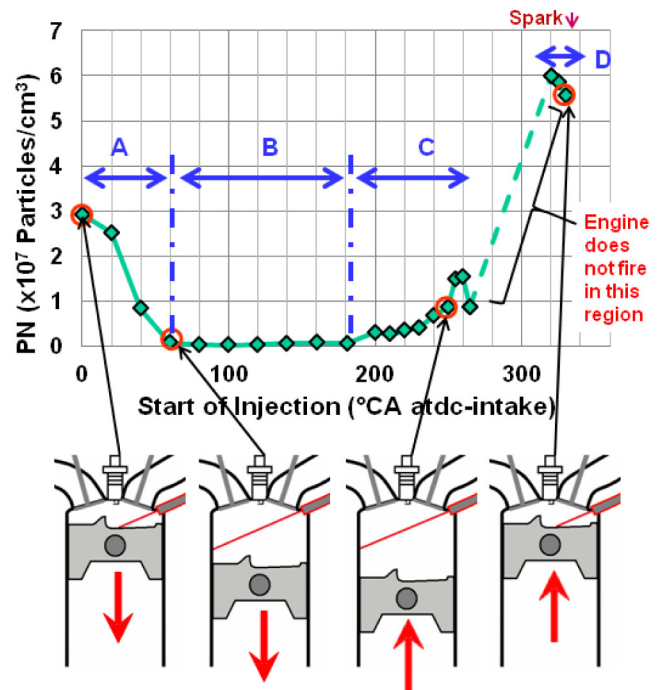


Figure 4. PN as a function of SOI and injection geometry at selected points. Cold-fast-idle at ECT=20°C.

The piston positions in Fig. 4 are drawn at the end of injection. The arrival of the spray at the walls (piston and liner) depends on the spray penetration speed and piston positions at the various SOI timings. CFD results (that finite amount of fuel has landed on the walls) indicate that the spray has reached the walls at the end of injection.

The corresponding COV of NIMEP values are shown in Fig. 5. The values were low for SOI in the intake stroke, but increased drastically at SOI timings later than 230° atdc-intake. The