

Effects of Secondary Air on the Exhaust Oxidation of Particulate Matters

Joseph Pritchard
GM Powertrain

Wai K. Cheng
MIT

ABSTRACT

The effects of secondary air on the exhaust oxidation of particulate matters (PM) have been assessed in a direct-injection-spark-ignition engine under fuel rich fast idle condition (1200 rpm; 2 bar NIMEP). Substantial oxidation of the unburned feed gas species (CO and HC) and significant reduction of both the particulate number (up to ~80%) and volume (up to ~90%) have been observed. The PM oxidation is attributed to the reactions between the PM and the radicals generated in the oxidation of the feed gas unburned species. This hypothesis is supported by the observation that the reduction in PM volume is proportional to the amount of heat release in the secondary oxidation.

CITATION: Pritchard, J. and Cheng, W., "Effects of Secondary Air on the Exhaust Oxidation of Particulate Matters," *SAE Int. J. Engines* 8(3):2015, doi:10.4271/2015-01-0886.

INTRODUCTION

Direct injection (DI) is an attractive option for spark ignition (SI) engines. The fuel charge cooling improves the volumetric efficiency and knock margin so that highly boosted, downsized engines which offer substantial gains in fuel economy could be devised [1,2,3,4].

An important issue for DISI engines is the particulate matter (PM) emissions [5,6,7,8,9]. Particulate number (PN) emissions for small particles are especially of concern, as stringent limits have been established [10]. PN emissions during cold start in DISI engines are substantially higher than those in port-fuel-injection (PFI) engines [5, 6] because the in-cylinder fuel jet directly deposits liquid fuel on the cold cylinder walls. The fuel vapor that emerges from the fuel films produces fuel rich regions which are responsible for PM formation [5].

An effective strategy for controlling the tailpipe emission is to introduce secondary air injection to the exhaust [11,12,13]. Then, by running the engine rich and by retarding the spark timing substantially [14, 15], the secondary air would react with the exhaust mixture, thereby oxidizing the unburned hydrocarbons and enhancing the enthalpy flow to the catalyst. Thus both the tailpipe-out emissions before the catalyst light-off and the catalyst light-off time are reduced.

The interaction between the secondary air and the engine-out particulate matter is the subject of this paper. The issues being addressed are:

1. Would the secondary air injection strategy help to reduce the PM emission?
2. If so, to what extent is the PM emission reduced.
3. What is the mechanism behind this reduction?

APPARATUS AND PROCEDURE

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.

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

Operating Condition

The engine operated at a condition typical of the cold fast idle period, at 1200 rpm and 2 bar net indicated mean effective pressure (NIMEP). The engine coolant temperature was regulated by a water chiller so that at the coolant inlet, the water temperature was at 20° C.

The fuel used in these experiments was Haltermann HF0437 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 counter flow shell-in-tube heat exchanger and was maintained at a temperature of 20°C.

The start of the direct fuel injection was at 80° aTDC-intake. This timing was chosen so that there was a significant amount of PM in the exhaust while the amount did not change rapidly with respect to the injection timing.

Secondary Air Injection

The secondary air injection (SAI) arrangement is shown in Fig. 1. The secondary air flow rate was controlled by the air injection pressure through critical orifices. An end-plugged 3.2 mm (1/8") OD tube with two 1.6 mm holes drilled cross-wise supplied the secondary air to each branch of the exhaust port. The air flow was perpendicular to the exhaust flow to promote mixing [16]. The air injection points were at 2 cm from the exhaust valve seat. Pictures of the injection arrangement are shown in Fig. 2.

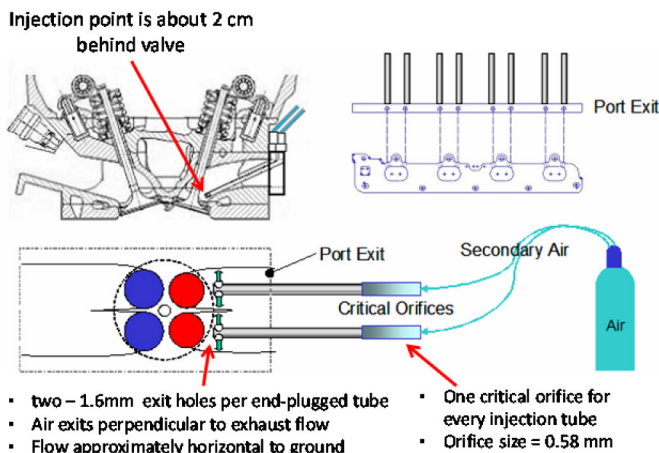


Figure 1. Secondary air injection arrangement

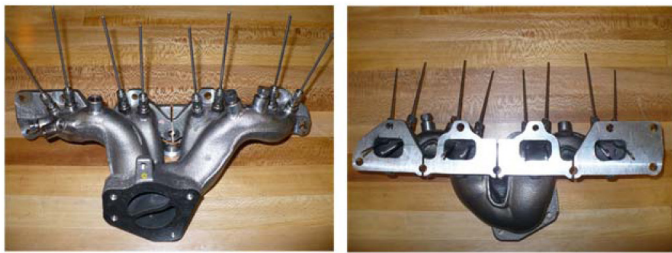


Figure 2. Secondary air injection hardware

Table 2. Representative secondary air flow value.

Representative running condition	
λ engine/ λ exhaust	0.8 / 1.0
Spark	15° atdc-compression
Speed/ NIMEP	1200 rpm / 2 bar
Representative secondary air flow	
Total (8 tubes)	198 SLPM; corresponds to 4.2 g/s
Per tube	24.8 SLPM; corresponds to 0.53 g/s
Velocity at exit hole	100 m/s
Volume occupied by injected air during EVC	0.03 L
Length of runner occupied by the above air	4 cm

The amount of secondary air was metered by the λ value measured at the exhaust (i.e. the λ value after the engine-out exhaust gas mixed with the secondary air). The representative secondary air flow is shown in Table 2. The air exit velocity was fast (~100 m/s) to enable fast mixing. Note that the secondary air injection was continuous, while the engine exhaust flow for each cylinder is intermittent. The

length of the exhaust runner occupied by the injected air during exhaust valve close was approximately 4 cm. This slug of air would mix with the burned gas of the next cycle and with that from the other cylinders. Therefore the actual mixing process is substantially non-uniform in nature.

Particle Measurements

The particle size spectrum was measured by a scanning mobility analyzer (TSI SMPS Model 3934). Each measurement took a scan time of 90 seconds with the engine running at steady state condition. The sampling location was at approximately 1 m downstream of the turbine to allow adequate time for the secondary air to mix and react with the exhaust stream. The catalyst has been taken out in these experiments. The sampling was done via a heated sample line and the sample was diluted by nitrogen. The dilution ratio was determined by measuring the CO₂ concentration at the sampling inlet with a Horiba MEXA 584L CO₂ meter, and at the SMPS inlet by a more sensitive Licor LI-820 CO₂ analyzer. The sampling system was heated to 200° C to prevent condensation. Because there were significant pressure pulsations at the sampling inlet, to maintain a steady sampling flow to the dilution system, a heated “T” design similar to that used in the fast-response HC analyzers [17] was used. The details have been described in Ref. [5].

Because of the presence of secondary air, the particulate matter concentrations have been corrected for both the dilution ratio and the temperature and pressure, and are reported as the undiluted values at the sampling inlet adjusted to the reference temperature of 300K and pressure of 1 bar.

Temperature Measurements

To determine the heat release from the secondary air reaction with the exhaust gas, the temperatures of the exhaust stream were measured at the locations shown in Fig. 3. The measurements were done with type K thermocouples inside a shielded aspiration probe to minimize radiation loss [18]. To estimate the heat loss in the different sections of the exhaust system, the wall temperatures were measured by an IR radiation thermometer.

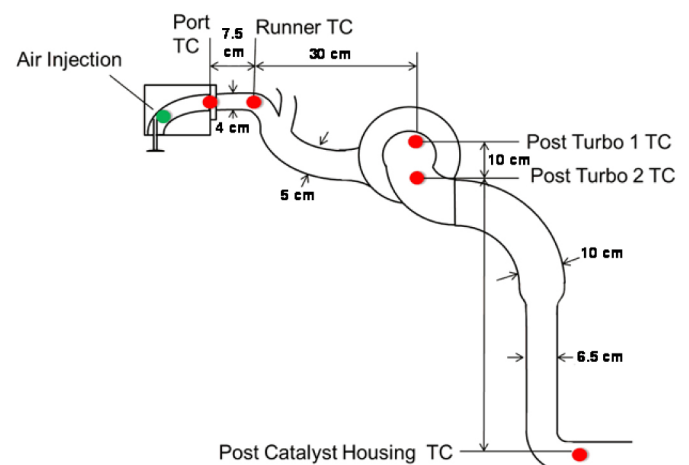


Figure 3. Locations of temperature measurements to assess the heat release from the reaction of secondary air and the engine exhaust gas

HEAT RELEASE FROM THE SECONDARY AIR AND EXHAUST GAS REACTION

The strategy was to run the engine rich (at engine $\lambda = 0.8$ and 0.9) to provide an engine-out feed-gas with significant chemical enthalpy, and then to oxidize this gas by the secondary air to enhance the sensible enthalpy flow. The in-cylinder air equivalence ratio will be referred to as the engine λ ; the air equivalence ratio of the mixture of feed gas and secondary air will be referred to as the exhaust λ . Thus the difference between the exhaust λ from the engine λ is a measure of the amount of secondary air injection.

The feed-gas chemical enthalpy was obtained by measuring the CO and HC concentrations at the entry of the exhaust runner of cylinder #4; the H₂ concentration was not measured but was calculated from the water gas shift equilibrium at 1740K. The values are shown in Table 3. The chemical enthalpy at $\lambda = 0.9$ was ~ 500 kJ/kg-exhaust gas, and at $\lambda = 0.8$ was ~ 1000 kJ/kg-exhaust gas. In comparison, the exhaust sensible enthalpy was approximately 1000 kJ/kg-exhaust gas.

Table 3. Engine out chemical enthalpy at 1200 rpm, 2 bar NIMEP. Exhaust sensible enthalpy ~ 1000 KJ/kg-exhaust gas.

Engine λ	Spark Timing [aTDC]	Feed-gas chemical enthalpy [kJ/kg exhaust]	Percent Enthalpy Contribution		
			CO	H ₂	HC
0.9	10	520	63%	16%	21%
0.9	15	510	64%	17%	19%
0.8	10	1070	65%	20%	15%
0.8	15	1050	66%	21%	14%

The CO and HC concentrations in a mixing tank approximately 1.5 m downstream from the turbine were measured. It was assumed that beyond this location the temperature was sufficiently low such that there was no further reaction between the secondary air and the exhaust. These values, referred to as the exhaust values, were corrected for the secondary air dilution, and therefore, were measures of the extent that the feed gas species were oxidized. These exhaust CO and HC mole fractions are shown in Figures 4 and 5. At exhaust λ equal to engine λ , there was no secondary air. Since the engine was operating rich, there was little oxygen in the exhaust stream; hence there was little oxidation of the CO and HC. As more secondary air was introduced (increase in exhaust λ), there was increasing oxidation. Beyond exhaust λ of 1.05, most of the CO and HC were oxidized.

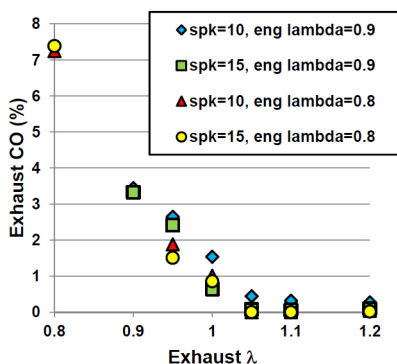


Figure 4. Exhaust CO mole fraction (corrected for secondary air dilution) as a function of the exhaust λ for various engine λ and spark timing. Engine operated at 1200 rpm and 2 bar NIMEP.

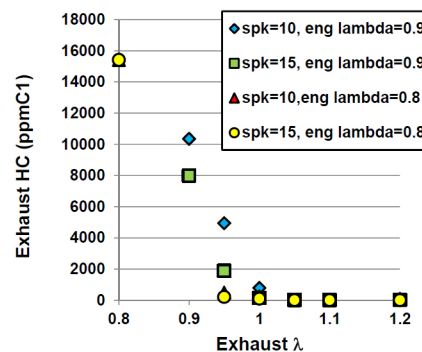


Figure 5. Exhaust HC mole fraction (corrected for secondary air dilution) as a function of the exhaust λ for various engine λ and spark timing. Engine operated at 1200 rpm and 2 bar NIMEP.

The temperature measurements at the locations shown in Fig. 3 are shown in Figures 6 and 7 for engine $\lambda = 0.9$ at spark timing of 10° and 15° aTDC. When no SAI was used (exhaust $\lambda =$ engine λ), the temperature decreases downstream due to heat loss. When SAI was applied, the decrease was less because of the heat release from the secondary reactions. The effect was more prominent at spark timing of 15° aTDC for which the exhaust temperature was higher so that the heat release reaction was faster.

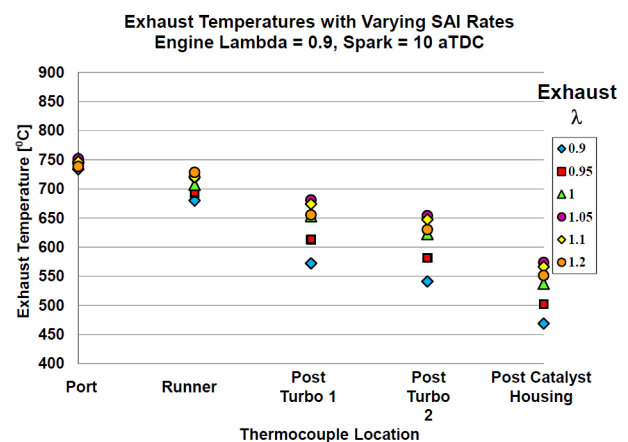


Figure 6. Exhaust gas temperatures at locations (shown in Fig. 3) for engine $\lambda = 0.9$ and varies exhaust λ values; spark timing at 10° aTDC.

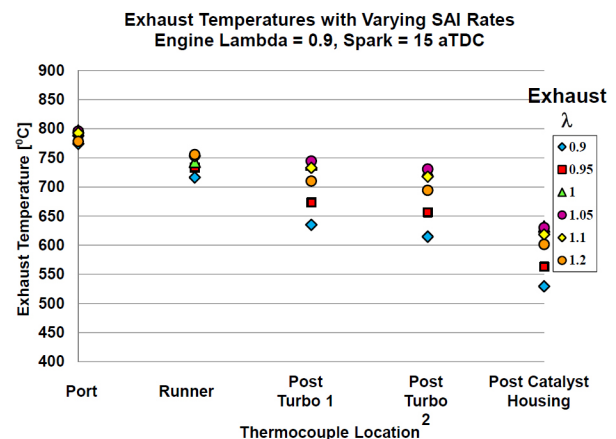


Figure 7. Exhaust gas temperatures at locations (shown in Fig. 3) for engine $\lambda = 0.9$ and varies exhaust λ values; spark timing at 15° aTDC.