

SI Engine Control in the Cold-Fast-Idle Period for Low HC Emissions and Fast Catalyst Light Off

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ABSTRACT

The engine and its exhaust flow behaviors are investigated in a turbo-charged gasoline direct injection engine under simulated cold-fast-idle condition. The metrics of interest are the exhaust sensible and chemical enthalpy flows, and the exhaust temperature, all of which affect catalyst light off time. The exhaust sensible enthalpy flow is mainly a function of combustion phasing; the exhaust chemical enthalpy flow is mainly a function of equivalence ratio. High sensible and chemical enthalpy flow with acceptable engine stability could be obtained with retarded combustion and enrichment. When split injection is employed with one early and one later and smaller fuel pulse, combustion retards with early secondary injection in the compression stroke but advances with late secondary injection. Comparing gasoline to E85, the latter produces a lower exhaust temperature because of charge cooling effect and because of a faster combustion.

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INTRODUCTION

Downsized turbocharged direct-injection spark ignition (DISI) engines can achieve superior fuel economy with the same or better performance [1]. However, compared to similar naturally aspirated engines, there are significant cold-start emission issues. The focus of this paper is on hydrocarbon (HC) emissions.

SI engine emission control relies on three-way catalyst technology. The catalyst, however, is ineffective when cold [2]. For fast catalyst warm-up, conventional strategies such as using close-coupled catalysts and exhaust manifolds with coatings and/or low thermal inertia are less effective when applied to turbo DISI engines. The warm-up of the catalyst is hindered by the heat transfer to the turbocharger assembly which is typically located upstream of the catalyst. Furthermore, emission rates of unburned hydrocarbons during cold-start are higher because of the presence of substantial in-cylinder liquid fuel due to direct fuel injection. Hence, reducing the cumulative cold-start emissions of unburned hydrocarbons from a turbocharged DISI engine is more challenging than that in a naturally aspirated, port-fuel-injection (PFI) engine.

In this paper, the effects of fuel injection timing, split injection, ignition timing, and fuel ethanol content on the sensible and chemical enthalpy flow to the catalyst are characterized.

BACKGROUND

To improve cold-start performance, the metrics of interest are the exhaust sensible enthalpy flow, exhaust gas temperature, and exhaust chemical enthalpy flow. These metrics strongly influence catalyst warm-up time and exhaust emissions during warm-up.

In this research, combustion phasing is used to control exhaust sensible enthalpy flow and exhaust gas temperature; enrichment is used to stabilize combustion and to enhance chemical enthalpy flow for the potential benefit of secondary air injection; and split injection is used to locally enrich the in-cylinder mixture for combustion stability at retarded combustion phasing. Engine emissions with E85 fuel are also assessed.

Ignition Timing

Late ignition timing is a common strategy for fast catalyst warm-up. Retarded combustion reduces work transfer from cylinder gases to the piston resulting in higher exhaust temperature. For constant engine torque, the reduced fuel conversion efficiency associated with late combustion requires greater fuel and air flow rate. Thus late ignition leads to greater engine-out sensible enthalpy flow due to both higher temperatures and greater mass flow rates. Late combustion is

limited by increasing cycle-to-cycle variation. Very late combustion could lead to partial-burn or incomplete combustion, resulting in unacceptable level of HC emissions.

Mixture Enrichment

Rich mixtures are useful during cold start for two reasons. First, enrichment can improve combustion stability and reduce cycle-to-cycle combustion variations. Second, a strategy employing late ignition timing with rich mixtures and air injection to the exhaust manifold (i.e. secondary air injection) has been successful to simultaneously reduce feed-gas emissions and to accelerate catalyst warm-up [3]. This research does not experimentally investigate secondary air injection, but quantifies the chemical enthalpy flow rate which is proportional to the potential chemical heat release available when secondary air is used. Mixture enrichment has a potential drawback in terms of sensible enthalpy flow rate: rich mixtures lead to lower exhaust gas temperatures due to extra evaporative charge cooling and reduction in combustion efficiency.

Split Fuel Injection

Direct fuel injection affords the ability to have multiple injections per cycle. Split injection with an injection during the intake stroke, and a shorter injection late in the compression stroke can provide local enrichment near the spark plug to stabilize early combustion without global mixture enrichment. [4]. Split injection also diminishes fuel jet wall wetting, thereby reduces engine-out emissions. Adequate fuel pressure is required to provide sufficient atomization [1, 5]. There are concerns that late fuel injection could lead to excessive piston impingement and subsequent PM emissions due to the excessive liquid fuel presence on the combustion chamber surfaces.

Fuel Ethanol Content

Fuels containing significant ethanol content present an additional emissions challenge for turbocharged DISI engines. Ethanol has a substantially higher latent heat of vaporization than gasoline. The lower heating value of ethanol also requires a greater quantity of fuel to be injected for the same torque output. Both attributes would lead to significant charge cooling and hinders fuel vaporization, which, in turn, would worsen cold-start emissions.

EXPERIMENTAL SETUP

Engine experiments were carried out on a production GM 2.0L inline-4 16-valve turbocharged gasoline direct injection spark ignition engine (GM LNF series). The production engine as equipped with independent variable' cam phasing on the intake and exhaust camshafts. [Table 1](#), [2](#), [3](#) summarize the engine and valve specifications.

Table 1. Engine specifications

Engine type	Inline 4 cylinder
Displacement [cc]	1998
Bore [mm]	86
Stroke [mm]	86
Wrist pin offset [mm]	0.8
Connecting rod [mm]	145.5
Compression ratio	9.2:1
Fuel system	Side-mounted, gasoline direct injection
Valve configuration	16 valve DOHC, dual cam phaser, 35.1 mm intake valve diameter, 30.1 mm exhaust valve diameter

Table 2. Intake valve timing and lift information.

Intake Valve	Opens [ATDC _{gas ex}]	Max open [ATDC _{gas ex}]	Closes [ABDC _{compr}]
Base	+11°	+126°	+61°
Max advance	-39°	+76°	+11°
Lift [mm]	0.25mm	10.3mm	0.25mm

Table 3. Exhaust valve timing and lift information.

Exhaust Valve	Opens [BBDC _{exp}]	Max open [ATDC _{gas ex}]	Closes [ATDC _{gas ex}]
Base	+52°	-125°	-10°
Max retard	+2°	-75°	+40°
Lift [mm]	0.25mm	10.3mm	0.25mm

The fuel used is a Haltermann HF0437 Tier II EEE certification fuel with RON of 96.5 and MON of 88.6. For E85 experiments, HF0437 is splash-blended on-site with anhydrous 99.9% pure ethanol by volume.

Engine control was accomplished using a custom desktop PC control system. The hardware allowed real time control of fuel injector pulse width, ignition timing, intake camshaft timing, exhaust camshaft timing. Fuel supply pressure was held constant by a hydraulic accumulator driven by high pressure nitrogen gas at 5 MPa.

Exhaust gas temperatures are measured in the manifold approximately 3 cm downstream of the flange. The thermocouple is equipped with a custom-built aspirated radiation shield to reduce error due to radiation and exhaust mass flow rate pulses during the exhaust process.

Hydrocarbons are measured with a Cambustion HFR400 Fast Flame Ionization Detector (FFID). Measurements are conducted at two points. The first is exhaust manifold runner of cylinder 4 (nearest the flywheel) approximately 6 cm downstream of the flange. The second is downstream of the turbocharger outlet.

Carbon monoxide, and carbon dioxide are measured with a Horiba MEXA-554JU and a Horiba MEXA-584L. Samples are drawn through an ice-bath condenser and lab dessicant column from the exhaust manifold runner of cylinder 4. The dry measurements have been converted to wet values using the measured λ values.

Experimental Conditions

The experiments have been conducted at a speed/load operating point meant to represent the quasi-steady fast-idle period that occurs after cranking and speed flare during cold start. The engine operating conditions used for these experiments are based on values representing typical fast idle condition in practice, and are summarized in Table 4.

Table 4. Nominal operating condition for quasi-steady fast idle portion of cold start.

Engine parameter	Value	Units
Engine speed	1200	rpm
NIMEP	2.0	bar
Ignition	Varied	CAD ATDC
Fuel/air equivalence ratio (Φ)	1	-
External EGR	0	%
Coolant/oil temperature	20	°C
Fuel inj. timing	80	CAD ATDC _{gas ex}
Fuel inj. pressure	5.0	MPa
Intake air temperature	20	°C
Exhaust back pressure	5	kPa

Except the split injection experiments, experiments use single pulse fuel injection per cycle. All experiments used valve timing in the park position.

RESULTS

Exhaust Sensible Enthalpy Flow

When ignition timing is retarded from 7 CAD BTDC to 17 CAD ATDC, the 50% mass-fraction-burn (m.f.b.) crank angle (denoted by CA50) shifts from 36° to 84° ATDC. The exhaust gas temperature varies almost linearly from 525K to 750K. Figure 1 shows exhaust sensible enthalpy flow (referenced to 298K) increases from approximately 4.2 kW to 11.5kW over this range of combustion phasing which is characterized by the CA50 values. The variation is super-linear because NIMEP is held constant. Then more fuel and air are required as combustion is retarded because of the associated decrease in indicated fuel conversion efficiency.

Retarded combustion leads to higher sensible enthalpy flow, but also to larger cycle-to-cycle variability; see Fig. 2. At COV of GIMEP greater than approximately 10%, misfire was sometimes observed as short-lived lean spikes on the lambda sensor signal.

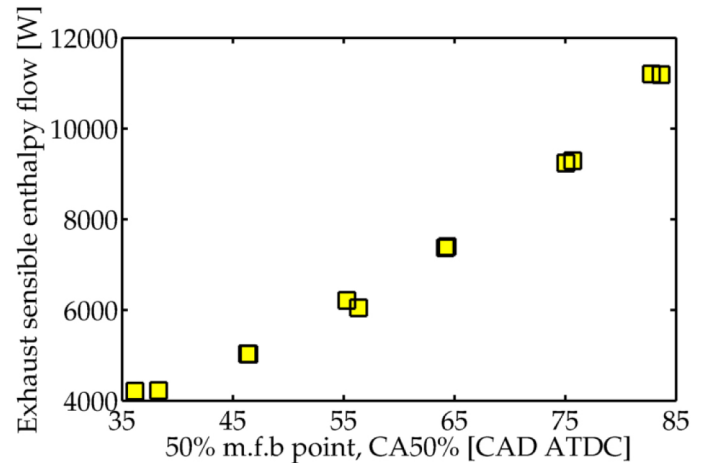


Figure 1. Exhaust thermal enthalpy flow increases from 4.2kW to 11.5kW while NIMEP and mixture stoichiometry are held constant as CA50 changes from 36° to 84° ATDC.

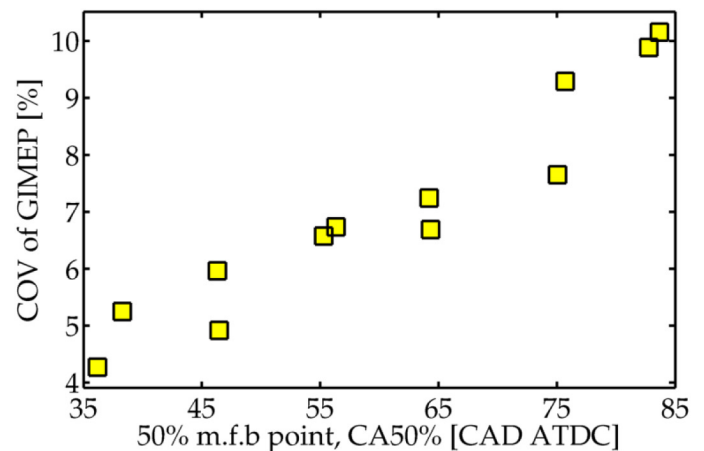


Figure 2. COV of GIMEP as a function of combustion phasing as indicated by the CA50 values.

For robust combustion at this cold-start fast-idle point, a maximum COV of GIMEP of 10% may be imposed. Then a maximum sensible enthalpy flow of 11.5 kW is readily achievable. This value is shown in the next section to be comparable to the chemical enthalpy flow achievable with mixture enrichment.

Exhaust Chemical Enthalpy Flow

The exhaust chemical enthalpy flow is the sum of the chemical enthalpy flow of the exhaust species of incomplete combustion. For each species, that value is the product of the individual mass flow rate and lower heating value. The components considered are CO, unburned hydrocarbons and hydrogen. The hydrogen concentration is not measured directly, but is estimated using an empirical relationship based on the water gas shift equation with an equilibrium constant of $K=3.5$.

The contour map of the exhaust chemical enthalpy flow is shown in Fig. 3. At constant CA50 (along a vertical line in the figure), the chemical enthalpy flow increases and is nearly linear with increasing enrichment. The values increase more rapidly