

Investigations into the Effects of Thermal and Compositional Stratification on HCCI Combustion – Part II: Optical Engine Results

R. E. Herold, J. M. Krasselt, D. E. Foster, J. B. Ghandhi
Engine Research Center; University of Wisconsin – Madison

D. L. Reuss, P. M. Najt
General Motors Research

Copyright © 2009 GM Global Technology Operations, Inc. Published by SAE International with permission.

ABSTRACT

The effect that thermally and compositionally stratified flowfields have on the spatial progression of iso-octane-fueled homogeneous charge compression ignition (HCCI) combustion were directly observed using high-speed chemiluminescence imaging. The stratified in-cylinder conditions were produced by independently feeding the intake valves of a four-valve engine with thermally and compositionally different mixtures of air, vaporized fuel, and argon.

Results obtained under homogeneous conditions, acquired for comparison to stratified operation, showed a small natural progression of the combustion from the intake side to the exhaust side of the engine, a presumed result of natural thermal stratification created from heat transfer between the in-cylinder gases and the cylinder walls. Large differences in the spatial progression of the HCCI combustion were observed under stratified operating conditions. Qualitative observations of the manner in which the combustion proceeded indicated that ± 20 °C temperature stratification, $\pm 15\%$ fuel concentration stratification, and ± 5 air-fuel ratio stratification all similarly affected the combustion progression, with the combustion proceeding, in general, from high to low temperature, high to low fuel concentration, or high to low air-fuel ratio. Concurrent and opposite stratification of these parameters were often present and mitigated the effects that the stratified flowfields had on the manner in which the combustion progressed spatially.

The possibility of using the thermal and compositional stratification studied herein for HCCI combustion control was evaluated in a hypothetical recompression-type engine. The magnitude of stratification that could be maintained until combustion commenced was estimated to be small, and competing effects were often present, likely limiting the effects the stratification could have on the combustion.

INTRODUCTION

Mixture inhomogeneities play an important role on the homogeneous charge compression ignition (HCCI) combustion process. Because of the chemical-kinetic nature of the HCCI combustion, all of the fuel would ignite concurrently after compression if the in-cylinder charge were truly homogeneous in composition and temperature. Experiments, however, have shown that this does not occur. Sample images acquired by Richter *et al.* [1] using planar laser-induced fluorescence (PLIF) of the OH radical revealed a non-uniformly igniting charge in spite of efforts to make the most homogeneous fuel, temperature, and residual gas distribution possible for that engine configuration. Several other PLIF- and chemiluminescence-based experiments have shown similar results [2-9]. From these results, it appears that inhomogeneities dictate the spatial and temporal distribution of autoignition sites within an HCCI engine. As such, stratification – purposely introduced inhomogeneities – holds the potential to alter the autoignition characteristics and serve as a mechanism for controlling the start of combustion.

Natural inhomogeneities (primarily in temperature) have been suggested as the reason for the finite duration of HCCI combustion, as opposed to the nearly instantaneous combustion predicted by chemical kinetics [9]. Sjöberg *et al.* [10] found, using a multi-zone chemical kinetic model, that introducing a $\pm 5\%$ temperature stratification ($380\text{ K} \pm 10\text{ K}$, represented by 4 independent zones) at bottom dead center was sufficient to match the measured pressure rise rate from HCCI combustion. The kinetic model results revealed that the zone with highest initial temperature ignited first, compression heating the remaining zones which sequentially autoignited in order of decreasing initial zone temperature. By introducing a wider initial temperature distribution (*i.e.* larger amount of thermal stratification), a longer combustion duration resulted, suggesting the possibility of achieving high engine loads while still maintaining acceptable pressure-rise rates through the use of thermal stratification. Additionally, recent studies have found that modifying the mixture's fuel concentration, air-fuel ratio, and the ratio of specific heats strongly affect the HCCI combustion phasing [11-13]. These results introduce the possibility of using compositional stratification to control the rate of HCCI combustion, similar to the way that the thermal stratification resulted in longer combustion duration in Sjöberg's simulation results.

In the present work, thermal and compositional stratification were introduced in an engine by independently feeding the intake valves of a four-valve engine, and the effects that these stratifications had on HCCI combustion were analyzed using the combination of cylinder pressure, emissions, and high-speed chemiluminescence measurements. The cylinder pressure and emission data provided a macroscopic measure of how the stratified flowfield affected the HCCI combustion performance, and the chemiluminescence images provided a direct observation on how the stratification affected the spatial progression of the combustion. Using a detailed characterization of the mixing between the two intake streams [14] and the known intake conditions, the local in-cylinder conditions could be estimated and used to compare the amount of stratification in the engine flowfield to the effect it had on how the combustion proceeded spatially.

The experimentally obtained results were used to evaluate the potential of using introduced stratification to control HCCI combustion in a hypothetical, recompression-type engine. The engine was assumed to induct equal volumes of exhaust residual and fresh charge, and in-cylinder mixing characteristics were specified to be the same as determined for the real engine. The estimated range of mixture properties present at the time of combustion was compared to the experimental results to address the possibility of obtaining HCCI combustion control in a production-type engine.

EXPERIMENTAL APPARATUS

ENGINE

Experiments were performed in an optically accessible, Bowditch-type single cylinder research engine constructed with a drop-down liner assembly. A cross-sectional view of the engine is provided in Figure 1. The engine has a geometric compression ratio of 11:1, a bore of 86 mm, and a stroke of 94.6 mm, resulting in a displaced volume of 550 cm^3 . Optical access to the combustion chamber was achieved through a sapphire window mounted in the piston crown, and chemiluminescence was collected from the combustion chamber via a 45 degree imaging mirror mounted in the piston extension. The UV-grade fused silica ring window that normally replaced the top 20 mm of the cylinder liner was removed and a steel ring blank was installed in its place. Laser excitation was unnecessary for the present experiments, so the glass window was removed to prevent damage of the fragile engine component.

The cylinder head features a pent-roof geometry with two intake valves, two exhaust valves, and a Siamese-type intake port that had a mount for an intake port fuel injector. To allow each intake valve to be fed independently, the concave port fuel injector-mount cavity of the cylinder head was filled with a cast aluminum plug, and the septum that splits the intake port to feed each valve was extended to the port entrance by a thin aluminum plate. The aluminum plug was held in place by a screw that exited the intake port via the fuel injector entrance, and the gaps between the plug, extended septum, and intake port walls were filled with high temperature RTV. Additional details on hardware created to split the intake port can be found in reference [12].

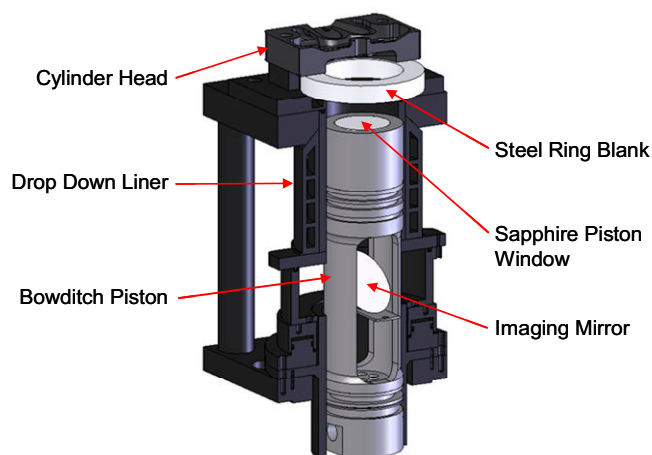


Figure 1. Optically accessible engine with drop-down liner assembly. Please note that color versions of all figures within this document can be found at <http://digital.library.wisc.edu/1793/31880>.

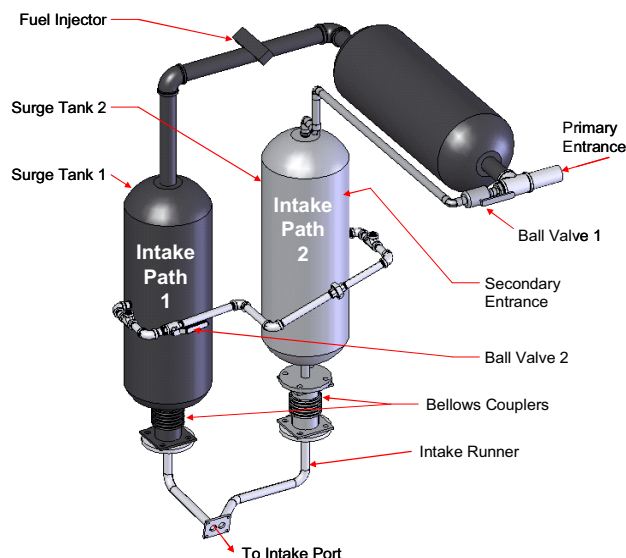


Figure 2. Intake system used to feed independent feed the optical engine's intake valves.

INTAKE CHARGE DELIVERY

The intake system for the optically accessible engine, shown in Figure 2, features two intake paths that allow each of the cylinder head's intake valves to be fed independently. Intake path 1, shown in dark gray, consists of two 37.8 L tanks connected with steel piping. An air-assisted fuel injector is permanently mounted in the piping between the two surge tanks. Intake path 2, shown in light gray, consists of a single 37.8 L tank and is connected to the primary entrance of the intake system with soldered copper tubing. The surge tanks of each intake path were connected to the intake runner with steel piping and flexible bellows couplers in order to isolate the intake system from the engine vibration. The intake runner was fabricated using stainless-steel tubing to connect the bellows couplers of each surge tank to one of the independent intake ports in the cylinder head.

The two ball valves in the intake system dictated the manner in which intake path 2 was supplied by intake gases. Opening ball valve 1 and closing ball valve 2 allowed intake path 2 to be fed with charge that entered the intake system at the primary entrance. Feeding intake path 2 in this manner prevented fuel from entering the engine through the intake valve fed by intake path 2. By closing ball valve 1 and opening ball valve 2, intake path 2 was fed with charge from intake path 1, allowing each intake valve to be fed with the same homogeneous mixture. Additionally, gas was fed directly into the surge tank 2 through the secondary entrance (a pipe fitting in the side of the tank), providing an additional method in which compositional stratification could be introduced to the engine.

To enhance the amount of stratification maintained throughout the compression process, top-directing flow baffles were mounted within each of the engine's intake ports. The top-directing baffles, shown schematically in Figure 3, were made out of 1.7 mm thick aluminum and

were machined to closely match the dimensions of the intake port, although no attempt was made to seal between the baffle and the intake port walls. A discussion of the improved bulk stratification that was achieved when using the top-directing baffles can be found in reference [14].

Air and argon were metered to the intake system using 1.17 mm and 1.18 mm calibrated critical flow orifices, respectively. The location into which the gases were delivered to the intake system varied depending on the compositional stratification that was introduced and will be discussed below. The fuel was introduced into the intake stream using a modified air-assisted fuel injector system mounted at the location shown in Figure 2. The air-assisted injector setup is directed such that the injection occurs in the direction opposite to the bulk intake flow, enhancing the mixing of the highly atomized spray provided by the air-assisted injection.

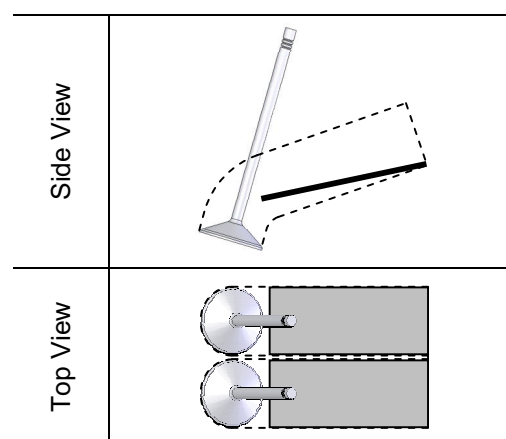


Figure 3. Top-directing flow baffles used to direct intake flows towards the top of the intake valves. The baffles were found to enhance the stratification maintained through compression [14].

INTAKE CHARGE HEATING

The intake charge was heated using six heating tapes wrapped on portions of the intake system. Two 63.5 mm wide, 3.0 m long (240 V, 2350 W) heating tapes were wrapped around the lower surge tanks of each intake path. The top heating tape on the lower surge tank of intake path 1 was extended upwards to wrap around the piping into which the fuel was injected to facilitate rapid vaporization of the injected liquid fuel. Additional heating was provided by 12.7 mm wide, 1.8 m long (240 V, 468 W) heating tapes wrapped around the tubes of the intake runner. An Omega CN616 six-zone temperature controller was used to control the six heater strips. Control temperatures for the heater strips around the surge tanks were measured using K-type thermocouples mounted within the surge tanks, whereas the control temperatures for the heater strips around the runner tubes were measured using K-type thermocouples mounted in the intake runner, just upstream of the cylinder head.

COMBUSTION DIAGNOSTIC COMPONENTS

CYLINDER PRESSURE & EMISSIONS

The cylinder pressure was measured with a Kistler 6125B piezoelectric pressure transducer coupled with a Kistler 5010B dual-mode charge amplifier. The pressure data were low-pass filtered at 8 kHz to remove high frequency noise prior to being recorded for 100 cycles by a 12-bit DSP 4012 data acquisition system. A BEI model H25 optical shaft encoder provided acquisition triggering at 0.5 crank angle increments. The measured cylinder pressure data were pegged at 190° bTDC to the intake surge tank pressure, which was measured with an Omega PX212-100AV pressure transducer.

A Horiba emissions analyzer was used to measure the steady-state concentration of five exhaust species: CO₂, CO, O₂, HC, and NO_x. The types of the individual analyzers are listed in Table 1. Each of the analyzers was calibrated using a span gas supply with known concentration of the desired exhaust species (see Table 1) and a Stec SGD-710C gas divider. Linear fits of the calibration data were used for all analyzers except the non-dispersive infrared type, which required third-degree polynomial fits.

Table 1. Types of the Horiba emissions analyzer for each emission species and the span range over which each analyzer was calibrated.

Emission	Analyzer Type	Span Range
CO ₂	Non-Dispersive Infrared	13%
CO	Non-Dispersive Infrared	0.997%
O ₂	Magnetopneumatic	15%
HC	Flame Ionization Detector	2550 ppm (C3H8)
NO _x	Chemiluminescence	98.2 ppm (NO)

CHEMILUMINESCENCE

Fifty chemiluminescence movies were acquired at each operating condition using the Phantom 7.1 high speed CMOS camera. The movies were acquired at a constant framing rate of 8000 frames per second, or approximately 1 frame per 0.45 crank angle degrees at the 600 rpm engine speed. The exposure duration of each frame of the movie was 120 μs, the maximum duration allowed for the specified framing rate. The camera acquisition was triggered by a Berkeley Nucleonics Model 565 function generator which, in turn, was triggered by the falling edge of a TTL pulse provided by the engine control unit (ECU). An f/1.8, f=50 mm standard glass-element camera lens was used to image the chemiluminescence, eliminating the possibility of capturing any ultraviolet wavelengths, including those emitted from the OH radical. The camera field of view allowed visualization of approximately 60 mm of the combustion chamber and is shown in Figure 4, with the intake valves, exhaust valves, and spark plug hole indicated for reference. This field of view provides visual access to ~50% of the clearance volume. A sample image sequence acquired at a homogeneous operating condition is provided in Figure 5.

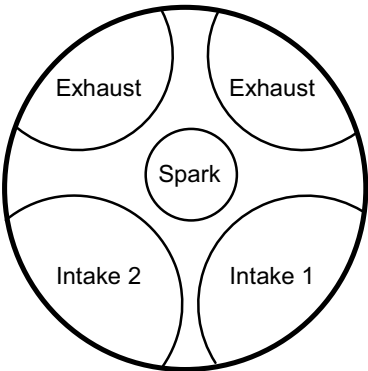


Figure 4. Field of view imaged when acquiring the chemiluminescence movies. The field of view is roughly centered within the combustion chamber and is ~60 mm in diameter.

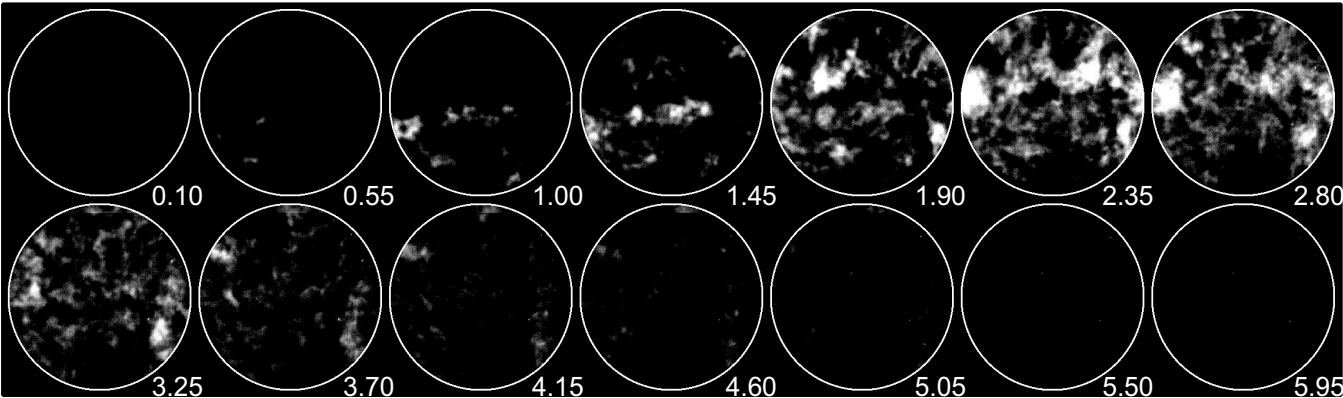


Figure 5. Sample image sequence of a chemiluminescence movie acquired under homogeneous operation. The number at the lower right of each frame indicates the crank angle at which the frame was acquired (aTDC).

The chemical species from which the chemiluminescence originated can be identified by their characteristic emissions spectra, although no effort was made to precisely identify the species in the present experiments. Several previous investigations have determined that species such as CH_2O^* , HCO^* , CH^* , C_2^* , CO_2^* , and OH^* are strong contributors to the chemiluminescence spectra [2,15-18]. Additionally, broadband emission from 350 nm to 550 nm has been observed and was attributed to the CO-O oxidation reaction [17,19,20]. The intensity of the chemiluminescence during HCCI combustion has been shown to strongly correlate with the rate of energy release, making the chemiluminescence signal an excellent, albeit spatially integrated, indicator of the location and intensity of HCCI combustion [2,6,7,9].

OPERATING CONDITIONS

BASELINE HCCI OPERATION

The combination of the engine's low (11:1) geometric compression ratio and the desired fuel's high octane number (iso-octane, octane number = 100) made achieving HCCI combustion in the optically accessible engine difficult. In previous metal engine experiments in our laboratory [21,22], excessively high intake charge temperatures were required to achieve acceptable HCCI combustion in an engine with the identical cylinder geometry. Furthermore, significant mass loss past the optical engine's non-metal rings lowered the peak cylinder pressure and exacerbated this issue. Since continuous operation of the optical engine under extremely high intake temperatures was not advisable, a strategy was required to lower the intake charge temperature to acceptable values, *i.e.* temperatures less than 150 °C.

As an alternative to increasing the engine's geometric compression ratio, which was difficult because of engine and the cylinder head hardware constraints, and using a lower octane number fuel, which was not advisable because of the possibility of fuel breakdown in the intake system [21], argon was used as a diluent. Because of its low number of energy storage modes, argon has a very low specific heat and a high ratio of specific heats, $\gamma \equiv c_p/c_v = 1.67$. The high γ value allowed the in-cylinder charge to compress to a higher pressure, and therefore higher temperature, than when excess air or exhaust gas residual (EGR) were used as a diluent. Because higher temperatures were achieved with the same engine geometry, HCCI combustion was achievable at acceptable intake charge temperatures. A comparison of motored engine operation when fed entirely with air and entirely with argon is shown in Figure 6. A ~500 kPa increase in peak motored cylinder pressure was achieved with argon dilution compared to air dilution at an atmospheric intake pressure.

The engine was operated at an engine speed of 600 rpm to sync with the 10 Hz repetition rate of a Nd:YAG laser

that was used in related experiments [23]. Natural aspiration was simulated by imposing a near-atmospheric intake pressure, and the physical connection between the intake surge tanks that fed each intake valve resulted in pressure continuity between the two intake paths. The engine was fueled with a mixture of 78% iso-octane, 20% 3-pentanone, and 2% triethylamine (TEA), which was selected based on its performance when used in a dual-tracer, PLIF-based temperature imaging strategy that was employed in subsequent experiments [23]. The mass flow rates that allowed stable HCCI combustion to be achieved at modest intake temperature, with acceptable pressure rise rates, and with sufficient combustion luminosity were 5 mg/cycle fuel, 110 mg/cycle air, and 590 mg/cycle argon. The intake charge temperature was varied to maintain a constant location of peak pressure (LPP) of ~5.3° aTDC.

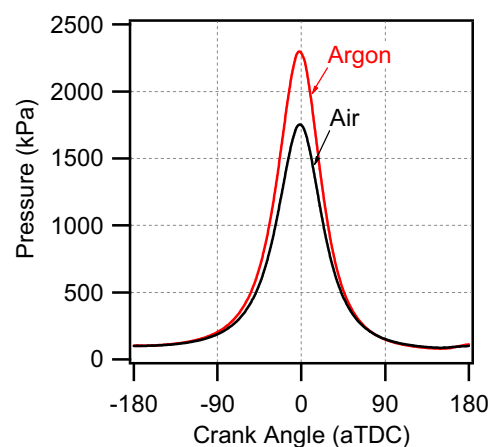


Figure 6. Measured cylinder pressure versus crank angle when feeding the engine with air or argon. Argon dilution results in a ~500 kPa increase in peak motored cylinder pressure than with air dilution because of its higher γ value ($\gamma_{\text{Ar}} = 1.67$; $\gamma_{\text{air}} = 1.4$).

STRATIFIED INLET CONDITIONS

The fuel, air, and argon mass flow rates were held constant when the engine was run with stratified inlet conditions; the locations of the intake system into which these fluids were introduced, however, changed and allowed for the compositional stratification. Additionally, the temperatures of the mixtures delivered to each intake valve were controlled independently via the heaters wrapped around the surge tanks and intake runner tubes, which allowed for thermal stratification to be introduced to the engine.

Details of the compositional and thermal stratifications considered are provided in Figure 7 and Table A in the appendix. Figure 7 provides schematics for the mass delivered through each surge tank per cycle for each of the stratified operating conditions, where the conditions denoted with a 'C' indicate a compositional stratification, and those denoted with a 'T' indicate a thermal stratification. Table A provides the

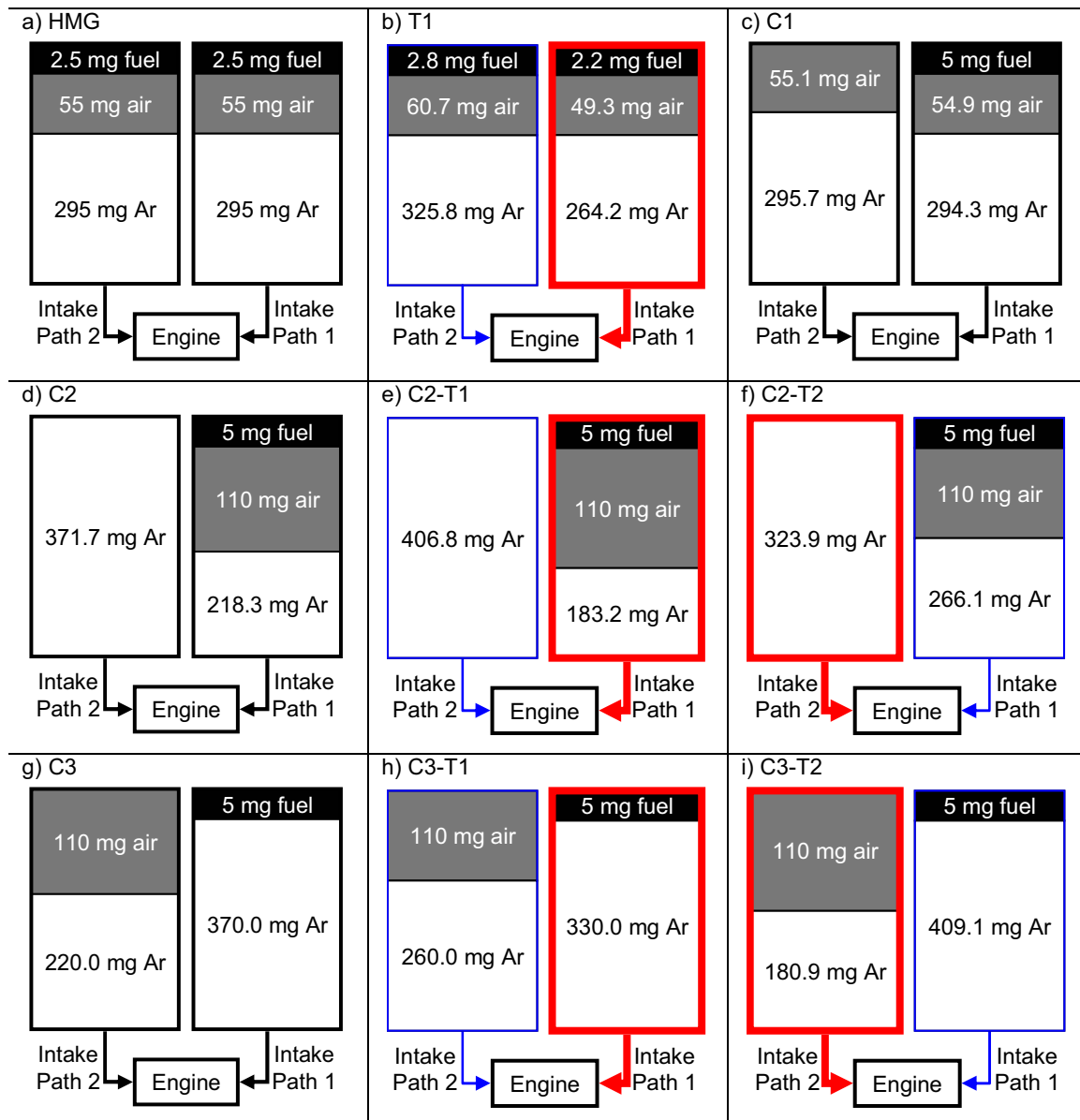


Figure 7. Schematics showing the mass flow rates per cycle delivered through each intake path for the stratified operating conditions. The bold line indicates the higher temperature path for the thermal stratification, and the narrow line indicates the lower temperature path.

ball valves located in the intake system (see Figure 2), the locations where the air and argon were introduced into the intake system, the intake charge temperatures entering each intake port, and the mixture properties in each surge tank for each of the stratified operating conditions. Note that the sizes of the rectangles in Figure 7 are not proportional to the gas volumes within each surge tank.

The mass distributions in Figure 7 and the mixture properties in Table A show the effect that the surge tank pressure balancing had on the stratified mixture properties. When comparing the homogeneous (HMG) and thermally stratified (T1) conditions, it can be seen that heating surge tank 1 to a higher temperature than surge tank 2 results in an uneven amount of fuel being delivered through each intake valve. This is a direct result of a density effect (higher temperatures result in lower densities); more mass is delivered to the engine through

intake valve 2. This effect is small and likely negligible considering the large mixing that occurs between the two intake streams during intake and compression, but it should be noted nonetheless.

LOCAL IN-CYLINDER CONDITIONS

The gradients in mixture properties produced for each of the stratified operating conditions described in Figure 7 and Table A were characterized using a single measured scalar distribution. This generic distribution (shown in Figure 8) was measured using laser-induced fluorescence (LIF) of a tracer molecule introduced through one intake valve only while operating with the same intake hardware, engine speed, and intake pressure as the present experiments [14,23]. The scalar profile was then converted to in-cylinder values of equivalent fuel mass, air-fuel ratio, ratio of specific heats, or estimated compression temperature.