

Energy Analysis of Low-Load Low-Temperature Gasoline Combustion with Auxiliary-Fueled Negative Valve Overlap

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ABSTRACT

In-cylinder reforming of injected fuel during an auxiliary negative valve overlap (NVO) period can be used to optimize main-cycle auto-ignition phasing for low-load Low-Temperature Gasoline Combustion (LTGC), where highly dilute mixtures can lead to poor combustion stability. When mixed with fresh intake charge and fuel, these reformat streams can alter overall charge reactivity characteristics. The central issue remains large parasitic heat losses from the retention and compression of hot exhaust gases along with modest pumping losses that result from mixing hot NVO-period gases with the cooler intake charge. Accurate determination of total cycle energy utilization is complicated by the fact that NVO-period retained fuel energy is consumed during the subsequent main combustion period. For the present study, a full-cycle energy analysis was performed for a single-cylinder research engine undergoing LTGC with varying NVO auxiliary fueling rates and injection timing.

A custom alternate-fire sequence with 9 pre-conditioning cycles was used to generate a common exhaust temperature and composition boundary condition for a cycle-of-interest, with performance metrics recorded for each custom cycle. The NVO-period reformat stream and main-period exhaust stream of the cycles-of-interest were separately collected, with sample analysis by gas chromatography used to identify the retained and exhausted fuel energy in the respective periods. To facilitate gas sample analysis, experiments were performed using a 5-component gasoline surrogate (iso-octane, n-heptane, ethanol, 1-hexene, and toluene) that matched the molecular composition, 50% boiling point, and ignition characteristics of a research gasoline. The highest total cycle thermodynamic efficiencies occurred when auxiliary injection timings were early enough to allow sufficient residence time for slow reforming reactions to take place, but late enough to prevent significant fuel spray crevice quench. Increasing the fraction of total fuel energy injected into the NVO-period was also found to increase total cycle thermal efficiencies, in part due to a modest reduction in NVO-period heat loss from a combination of fuel-spray charge cooling and endothermic fuel decomposition by pyrolysis. The effect was most pronounced at the lowest loads where larger charge mass reformat fractions increased overall specific heat ratios and main-period combustion phasing advanced closer to top dead center. These effects improved both expansion efficiency and combustion stability.

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INTRODUCTION

Light-duty engines that utilize Low-Temperature Gasoline Combustion (LTGC)-where compression heating rather than a spark plug are used to initiate combustion-have long been explored as a way to improve cycle efficiencies with simultaneous reductions of engine-out nitrogen oxide and particulate matter emissions [1]. A major challenge has been to achieve stable engine combustion at low loads, where low in-cylinder temperatures and elevated dilution fractions can result in overly slow combustion with partial burns or even misfire [1]. The problem is exacerbated when high-octane fuels under consideration for future high-efficiency combustion strategies are used [2]. The Reactivity Controlled Compression Ignition (RCCI) strategy that uses highly reactive diesel pilot injections to initiate combustion can achieve robust phasing control across the load-speed

map [3], although new hardware requirements (e.g., extra injectors, tanks, pumps) add significant cost and complexity to the vehicle. For single-fuel LTGC, both spark-assist [4] and partial fuel stratification [5, 6] strategies show real promise to provide dynamic control authority at lower loads while maintaining good cycle emissions and efficiency characteristics. However, for the lowest loads with high un-throttled dilution fractions, ignition kernel quench or insufficient reactivity enhancement can limit the effectiveness of these strategies.

Modified valve timings used to achieve intake charge heating from exhaust re-breathing via positive valve overlap (PVO) [7] or retained exhaust enthalpy via negative valve overlap (NVO) [8, 9, 10] can also improve low-load combustion stability. For direct injected engines, further benefit of the NVO strategy is that the oxidation of an

auxiliary fuel injection during the recompression can be used to provide additional sensible enthalpy to the subsequent charge. Moreover, when NVO-periods with low excess oxygen (O_2) are used, a combination of kinetically limited fuel pyrolysis and partial oxidation has been observed to generate product streams laden with fuel fragment reformat [10, 12, 13]. Once mixed with fresh intake charge and fuel, these reformat streams have been observed to alter overall charge reactivity characteristics, particularly when significant concentrations of acetylene, allene, or acetaldehyde are present [11, 12]. Alger et al. [14] further demonstrated that reformat addition can increase cycle thermodynamic efficiency through increased charge specific heat ratios (γ). For LTGC with a fixed reformat fraction, combustion phasing has been found to advance as the main-period fueling decreases [11, 13]. Ekoto et al. [13] attributed the advance to a combination of increased bulk-gas temperature from higher charge γ and increased fuel reactivity from higher reformat fractions that offset slower kinetics of leaner mixtures.

The central issue with NVO operation is the large parasitic heat loss that results from the retention and compression of hot exhaust gases along with a pumping loss due to the mixing of hot NVO-period gases with the cooler intake charge. These adverse effects on total cycle efficiency can be compounded when auxiliary fuel is injected during the NVO-period due to poor fuel utilization from any resultant heat release. However, accurate determination of the fuel utilization is complicated by the fact that any retained fuel energy is consumed during the subsequent main combustion period. Ekoto et al. [10] quantified the fuel energy recovered as a function of auxiliary fuel injected and retained excess O_2 using dump-sample collection at the conclusion of the NVO-period with sample analysis performed by gas chromatography. From a complementary energy balance analysis, they found that the fuel energy recovered improved and the NVO-period heat losses decreased as the exhaust O_2 was lowered and auxiliary fueling rates were increased. Wolk et al. [11] performed similar measurements and analysis for a range of single-component fuels. While full-cycle energy analyses have been performed for conventional LTGC operation [15], no studies to date have quantified the sources of loss when NVO generated reformat addition is used to maintain stable operation at low engine loads.

In this work, the impact of in-cylinder generated NVO reformat on full-cycle engine performance and fuel energy utilization was investigated for the engine fueled by a gasoline surrogate. All experiments were performed in a single-cylinder research engine, with cycle performance metrics acquired through the use of a custom alternate-fire sequence that produced a common exhaust boundary condition for the NVO period of a cycle-of-interest. Note that as a consequence of the alternate-fire sequence, the cycles-of-interest do not reproduce steady operation since wall and exhaust temperatures generated from the previous cycle will not be representative. Custom NVO start of injection (SOI), NVO fuel quantity injected, and total cycle fueling rates were used for the cycle-of-interest. A 1-D engine model was developed to determine bulk-averaged cycle characteristics at each valve event for the alternate-fire sequence. To validate the engine model, exhaust dump samples were acquired and characterized via gas chromatography with the results compared to

model predictions of the unburned exhaust fuel energy. Additional dump samples were collected at the end of the NVO period to determine the composition and fuel energy content of the reformat. Finally, the validated engine model was used to evaluate the sources of heat transfer and work at each portion of the cycle, along with exhaust enthalpy losses and total cycle combustion efficiency.

EXPERIMENT SETUP

Sandia Single-Cylinder Research Engine

All experiments were conducted in a single-cylinder research engine with a flat-top piston and central direct injection from a pent-roof head. The Bosch, step-hole, valve covered orifice injector has 8 125 μm diameter nozzles positioned at a 60° cone angle. Since the engine is normally configured for optical measurements not needed for the current experiments, the quartz liner and optical piston viewing window were replaced by all-metal components. A consequence of the lower piston compression ring pack for the optical piston is that crevice volumes are roughly 4 times greater than a comparable production engine. A thermocouple embedded in the head was used to monitor wall surface temperatures. Custom low-lift intake and exhaust cams were used to generate a fixed 150 crank angle (CA) duration NVO recompression with a 5.5:1 recompression ratio centered at top dead center of the NVO period (TDC_{NVO}). The recompression led to roughly 46% of the exhaust gases to be retained into the next cycle. Relevant engine geometric and operating characteristics are summarized in Table 1, while a schematic of the engine setup is provided in Figure 1.

Table 1. Engine specifications and operating conditions.

Displaced volume [liter]	0.633
Bore/Stroke [mm]	92/95.25
Connecting Rod [mm]	166.7
Geometric Compression Ratio	11.3:1
NVO period Recompression Ratio	5.5:1
Intake Valve Open/Close [CA [†]]	-285 / -140
Exhaust Valve Open/Close [CA [†]]	+140 / +285
NVO Duration [CA [†]]	150
Valve Lift [mm]	3
Speed [rpm]	1200
Intake Pressure [kPa]	100
Intake Temperature [°C]	104
Fuel Pressure [bar]	100
Residual Gas Fraction [%]	~ 45 – 51
Exhaust Excess O_2 [%]	2.4
Exhaust Temperature [°C]	370
Total Inj. Fuel Energy (main+NVO) [J/cycle]	350 – 750
SOI [CA _{NVO} [†]]	-65, -50 -40, -25, -10
NVO Inj. Fuel Energy [J/cycle]	66, 133, 265
NVO Bulk-Averaged Equiv. Ratio	0.84, 1.7, 3.3
SOI _{MAIN} [CA [†]]	-270

[†]CA and CA_{NVO} respectively refer to main-period TDC and TDC_{NVO}.



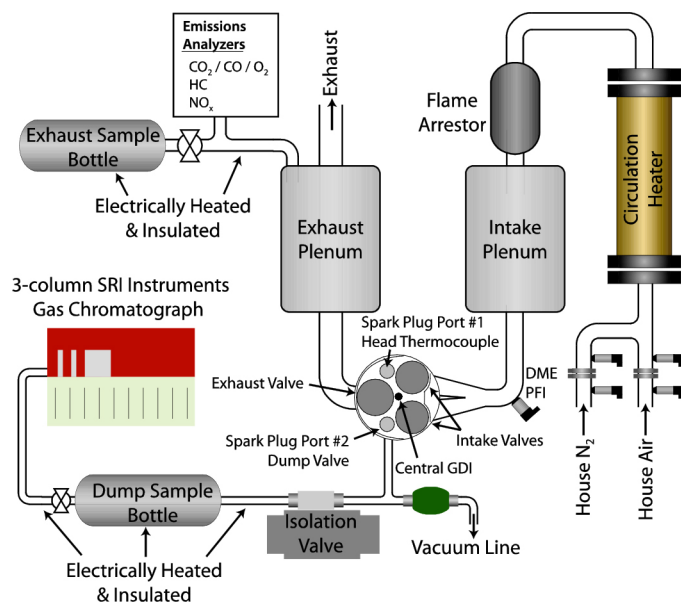


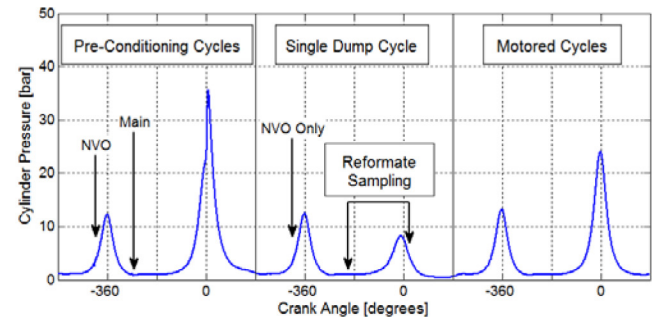
Figure 1. Schematic of the LTGC engine and dump sampling setup.

The operating conditions spanned low and intermediate engine loads where conventional LTGC combustion stability can be problematic. Cycle-of-interest indicated mean effective pressure (IMEP) varied between ~ 110 and 390 kPa, depending on the fueling rate. A motoring dynamometer was used to fix the engine speed at 1200 rpm, with the intake air pressure set to 100 kPa. Normal LTGC engines have compression ratios of around $14:1$, which result in greater TDC compression temperatures relative to the $11.3:1$ geometric compression ratio of the current engine. To more accurately replicate TDC temperatures of higher compression ratios, the intake charge was heated to 104°C . Exhaust O_2 concentration was adjusted to 2.4% through a combination of intake air dilution by nitrogen ($\sim 15.7\%$ intake O_2) and total cycle fueling so that partial oxidation and pyrolysis reactions were promoted during the NVO period for the current experiments. NVO-period SOI timings were spanned between -65 CA_{NVO} (relative to TDC_{NVO}), set just after exhaust valve closure (EVC) to generate the largest possible residence time, to -10 CA_{NVO} expected to result in significant piston wall wetting [8]. NVO injection quantities were also spanned, with NVO-period bulk-averaged equivalence ratios (ϕ) varied between 0.84 and 3.3 .

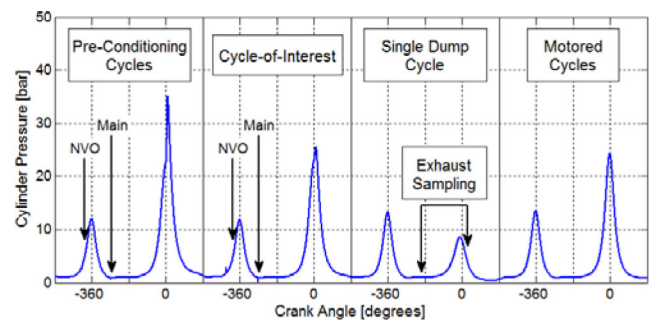
Dump-Sample Collection

Two distinct dump-sample collection strategies were employed, which are schematically illustrated in Figure 2: (a) Reformate sampling was used to characterize the constituents generated during the NVO period, while (b) Exhaust sampling was performed to accurately measure the exhausted fuel energy content for a particular cycle-of-interest. Because all dump samples were collected in bulk, they were sampled separately from the acquisition of engine performance metrics. Dump sampling made use of an in-cylinder dump-valve apparatus described previously [8, 10, 12, 16] that collected nearly all cylinder contents during the main-period compression stroke. The dump-sampling setup is also illustrated in

Figure 1. The dump-valve assembly mounted into an available 14 mm spark plug port, with collected sample directed through a manifold and into a vacuum evacuated, 1 -liter stainless-steel sample bottle. To minimize condensation of collected water or low-volatility hydrocarbons, all sampling system components were thermally insulated and electrically heated to 90°C .



a.



b.

Figure 2. Illustration of the (a) Reformate, and (b) and Exhaust dump-sample collection sequences.

For both dump-sample collection strategies, the engine was initially warmed-up by dimethyl ether port fuel injections (not shown in Figure 2) that readily auto-ignited regardless of the chamber wall temperatures. Once engine wall surfaces were sufficiently warm, the dimethyl ether injections were replaced by gasoline surrogate direct injections during both the NVO period and the intake stroke. Total fueling rates were adjusted such that a target 370°C exhaust temperature was achieved. The total cycle fueling rate was monitored using a Max Machinery model 213 volumetric flow meter, with the main/NVO split set from a predefined calibration map. Target total fueling from the specified main/NVO injection duration values was generally within 1% of the measured fueling rate. During post-processing, main and NVO injection fueling quantities were proportionally adjusted to match the measured fuel delivered. Once head surface temperatures reached a steady 127°C (fixed throughout for all experiments), dump sample collection was performed for a single cycle.

For the Reformate dump-sample collection sequence (Figure 2a), the single dump cycle performed after the last pre-conditioning cycle featured a custom NVO injection with main fueling cut during the

intake stroke and the dump-valve actuated just after intake valve opening (IVO). For the Exhaust dump-sample collection sequence (Figure 2b), an additional cycle-of-interest was added in between the common pre-conditioning cycle used in Figure 1a and the dump sample collection cycle. The cycle-of-interest featured an NVO-period injection that matched both the injection quantity and timing used for the corresponding Reformate dump-sample collection sequence. The main-period injection quantity was set such that the desired amount of total cycle fueling was achieved. Dump valve activation was performed during main-period compression of the following cycle; the main and NVO period fueling were cut during the dump-sample collection cycle. Both dump-sample collection sequences concluded with continuous motored cycles after the sampling event. The full dump-cycle sequence was repeated until the sample bottle pressure exceeded atmospheric (~7 dump-samples). Once the sample bottle was full, the engine was then stopped and the sample bottle isolated.

Gas Chromatograph Speciation

To characterize the dump samples, a 3-column SRI Instruments Multi-Gas Analyzer #3 gas chromatograph (GC) was used that could quantify 26 individual hydrocarbon constituents along with hydrogen (H_2), O_2 , carbon monoxide (CO), and carbon dioxide (CO_2). The GC contained 2 flame ionization detectors (FID). The first FID was fed by a fast eluting Restek MXT-1 60 meter capillary tube and was used to identify large hydrocarbons (C_3+). The second FID had a slower eluting 2 meter Restek Hayesep-D column that was used to measure smaller hydrocarbons (C_1 - C_2). The column also contained a methanizer used to measure CO and CO_2 . Finally, H_2 and O_2 were detected via a thermal conductivity detector (TCD) with sample supplied from a Restek Molecular Sieve-13x column connected in series to the second FID column. An estimate for the amount of water (H_2O) in the sample was obtained by performing a hydrogen atom balance, with all unaccounted fuel hydrogen attributed to H_2O . Nitrogen (N_2) was estimated from the molar balance of all remaining constituents.

Since acetylene (C_2H_2) and ethylene (C_2H_4) have similar column elution times, they could not be distinguished by FID#1. Similarly for FID#2, allene/propyne and p-m-xylene could not be distinguished. Instead the cumulative measurement for each set of species is used.

From a fuel energy basis, this assumption does not strongly influence the results.

Fuel Properties

The large number of constituents in real gasoline make gas sample analysis via the GC diagnostic intractable. To overcome this hurdle, a 5-component gasoline surrogate (54.9% iso-octane, 11.6% n-heptane, 9.9% ethanol, 4.7% 1-hexene, and 18.9% toluene by liquid volume) was used that matched the broad molecular composition of research-grade RD587 gasoline [16]. The surrogate and RD587 gasoline fuel properties are provided in Table 2; the results compare favorably aside from modest deviations in liquid density and volatility at the 10% (T10) and 90% (T90) boiling points.

Table 2. RD587 gasoline and surrogate fuel properties.

	RD587 Gasoline	RD587 Surrogate
Liquid Density [g/mL]	0.780	0.732
LHV [MJ/kg]	41.87	41.70
H/C ratio	1.97	2.02
Anti-Knock Index	88.4	89.3 [†]
T10 / T50 / T90 [°C]	57 / 98 / 156	91 / 96 / 103 [‡]

[†]Estimated using the approach by Mehl et al. [17]

[‡]Estimated using REFPROP [18]

Main-Period Engine Performance

The impact of the reformate stream on main-cycle combustion behavior was evaluated through the use of a 9-1 alternate-fire sequence with 1000 total cycles (i.e., 100 cycles-of-interest). A common EVC temperature and composition for the cycle-of-interest NVO period was generated by the 9 preceding “pre-conditioning” cycles that had the same main and NVO injection timings used for the dump-sample collection sequences. Note that injected fuel quantities are reported in terms of fuel energy content on a lower heating value (LHV) basis to facilitate comparisons of the parent fuel and reformate streams. Sweeps of NVO SOI (-65 - -10 CA_{NVO}) and NVO injection quantity (66 - 265 J) were performed, with total cycle fueling rates also swept (350 - 750 J) for each NVO condition. The baseline NVO condition has the common NVO fuel injection quantity (265 J) and SOI (-40 CA_{NVO}) for the respective sweeps.

ENERGY ANALYSIS MODEL

To obtain performance metrics for each cycle-of-interest, a 1-D engine model originally developed by Fitzgerald et al. [19] for steady operation was modified to account for the alternate-fire sequence. The original model accurately mapped the in-cylinder thermodynamic state through the use of the measured oxidizer stream and fuel mass flow rates, cylinder pressure, and the intake/exhaust pressure/temperature. The cycle charge composition was estimated using single-zone chemical kinetics simulations, with initial conditions adjusted until the simulation results matched the heat release and combustion efficiency measurements. Iterative models of the blow-down process were used to estimate the exhaust mass (m_{exh}). The residual gas fraction (RGF)-or the mass fraction of retained exhaust gas-was estimated from the ratio of the calculated masses during the closed cycle portion of the NVO and main periods. Because cyclic variations in oxidizer mass flow rate and exhaust temperature are not accounted for (these measurements were not cycle-resolved), the original model was ineffective for the alternate-fire sequence described here. Moreover, it was impractical to perform detailed kinetic modeling for each operating condition over the full 10 cycles in order to obtain an estimate for the composition.

Updated 1-D Engine Model

The purpose of the model is to estimate the state variables at each valve event based on the injection schedule and pressure record. Heat release characteristics are not determined by the model as these effects are explicitly accounted for from the input measurements.

measurement. To make the engine model suitable for the employed alternate-fire sequence, the RGF and the composition at each valve event are calculated through a series of atom-balance equations between the NVO and main portions of the cycle. The only species considered in the atom balance equations are O_2 , N_2 , H_2O , CO_2 , CO , and the parent fuel ($C_xH_yO_z$) since these species overwhelmingly governed the cycle charge mixture molecular weight and heat capacity characteristics. The RGF is defined here as:

$$RGF_i \equiv \frac{m_{NVO_i}}{m_{main_{i-1}}} \quad (1)$$

The variables m_{NVO} and m_{main} represent the charge mass for the closed NVO and main portions of the cycle respectively. The indices $i = 1, \dots, 9$ represent the pre-conditioning cycles within the alternate fire sequence, while $i = 10$ is the cycle-of-interest. Note that $i - 1$ refers to index 10 when $i = 1$, while $i + 1$ refers to 1 when $i = 10$. For each cycle, m_{main} is further defined as:

$$m_{main_i} \equiv m_{ind} + m_{fuel,main_i} + m_{NVO_i} \quad (2)$$

Here $m_{fuel,main}$ is the cycle specific amount of injected main-period fuel mass. For each cycle, the total inducted oxidizer mass (m_{ind}) measured from calibrated sonic flow nozzles is used in lieu of unavailable cycle resolved measurements of the intake charge mass. While small variations in the in-cylinder charge mass at IVO likely led to inversely proportional variations in the amount of intake mass inducted, these variations represented less than 1% of the total charge mass at intake valve closure (IVC), and thus are deemed to be negligible.

It is assumed that all O_2 for the globally-rich NVO periods was consumed. Hence, the NVO-period species balance equation is:

$$RGF_i(CO_2 + CO + H_2O + O_2 + N_2)_{main,out_{i-1}} + C_xH_yO_z_{NVO,in_i} \rightarrow (CO_2 + CO + H_2O + N_2 + C_xH_yO_z)_{NVO,out_i} \quad (3)$$

Note here that the subscripts *main* and *NVO* refer to the main or NVO portions of the cycle, while the subscripts *in* and *out* indicate whether the flow is into or out of the respective periods.

From Equation 3, four atom balance equations (not shown) are obtained for atomic carbon, hydrogen, oxygen, and nitrogen. An additional balance equation is obtained by assuming 90% of all excess fuel carbon out of the NVO period was converted to CO with the remainder left as pure fuel. Such an assumption produces charge mixture molecular weights and heat capacities at IVC that are consistent with previous dump sampling measurements [13]. Conversely, since the main period charge was globally lean, it is assumed that 95% of the charge mass carbon is converted into CO_2 , with the remainder converted to CO. Again, these assumptions are roughly consistent with previously reported combustion efficiency measurements from similar conditions [19]. The resultant main-period species balance equation is then:

$$(C_xH_yO_z + O_2 + N_2)_{main,in_i} + (CO_2 + CO + H_2O + N_2 + C_xH_yO_z)_{NVO,out_i} \rightarrow (CO_2 + CO + H_2O + O_2 + N_2)_{main,out_i} \quad (4)$$

From Equation 4 and the assumed fuel carbon conversion, 5 additional balance equations are obtained. Equations 1, 2, 3, 4 have 13 unknowns for each of the 10 cycles, with only 12 equations (Equations 1 and 2, along with the 10 balance equations). To overcome the fact that there was one more unknown than equation for each cycle, the RGF in each cycle was initially assumed to be 0.5. The rest of the cycle characteristics were calculated following the method described by Fitzgerald et al. [19]. The RGF for each cycle was recalculated based on the model output mass values for the NVO and main periods. The process was repeated until the RGF converged to a consistent value (2 iterations were usually sufficient).

Note that exhaust temperatures for the pre-conditioning cycles ($T_{exh,PC}$) were measured separately when the engine was continuously fired using pre-conditioning cycle injection commands. Cycle-of-interest exhaust temperatures ($T_{exh,COI}$) could not be directly measured due to the nature of the alternate-fire sequence, and were instead computed from the measured exhaust temperature (T_{exh}) as:

$$T_{exh,COI} = 10 \cdot (T_{exh} - 0.9 \cdot T_{exh,PC}) \quad (5)$$

As observed by Fitzgerald et al. [19], there was little sensitivity to the exhaust temperature on overall model outputs. A sensitivity study was also performed to analyze the influence of the assumed carbon conversion efficiencies at the end of the NVO and main portions of the cycle. The total amount of excess carbon converted to CO at the end of the NVO period was varied between 50% and 100%, while the percentage of main-period carbon converted to CO_2 was varied between 80% and 100%. In all instances, predicted RGF and state variables (e.g., heat capacity, temperature) at each valve event varied by less than 0.1% relative to model predictions using the assumed values of 90% and 95% respectively. Accordingly, the model likewise has very low sensitivity to these conversion efficiency values.

Wall Heat Transfer

Wall heat transfer (Q_{HT}) was estimated at each portion of the cycle using a generalized instantaneous bulk heat transfer expression:

$$Q_{HT} = Q_{HT,NVO} + Q_{HT,ind} + Q_{HT,main} + Q_{HT,exh} = \int hA(T - T_w)dt \quad (6)$$

For the heat transfer variables, the subscripts *NVO*, *ind*, *main*, and *exh* refer to the respective wall heat transfer contributions from the NVO, induction, main and exhaust portions of the cycle. The variable A is the instantaneous wall-surface area, T_w is the wall surface temperature measured from the embedded thermocouple, T is the instantaneous bulk-averaged charge temperature provided from the 1-D engine model, and h is the global heat transfer coefficient.



During the exhaust gas exchange period, values for h during both the expansion and recompression are calculated following the method described by Fitzgerald et al. [19]. A similar calculation is used for the intake gas exchange process, except that the 1-D model does not provide a bulk-averaged temperature during this period. Instead it assumes that the IVC temperature matches the bulk-gas temperature throughout the gas exchange process. This assumption is reasonable given that the value is near the mass averaged temperature of the retained NVO-period gases and the fresh intake charge.

For the closed portions of the cycle, the Woschni correlation is employed for h [20, 21], which is expressed as:

$$h \left(\frac{\text{watt}}{\text{m}^2 \cdot \text{K}} \right) = C_m B^a P^c T^b \left[C_1 \bar{S}_p + \frac{C_2}{6} (P - P_{\text{motor}}) \right]^c \quad (7)$$

Here B is the cylinder bore diameter, P is the measured cylinder pressure in units of bar, $\bar{S}_p$ is the mean piston speed, and P_{motor} is the motored pressure. Values for the coefficients in Equation 7 are provided in Table 3. Note that the main coefficient (C_m) and the constants a and b were adjusted such that a motored pressure trace best matched the results of a 1-D heat-transfer-corrected engine simulation cylinder-pressure; a and b deviate slightly from those reported by Chang et al. [21]. Moreover, the gas velocity coefficient (C_2) is constant due to the negligible flow swirl in the combustion chamber.

Table 3. Tabulated Woschni heat transfer correlation coefficients used in Equation 7.

Woschni Correlation Coefficient	Expansion	Compression
a	-0.25	-0.25
b	-0.71	-0.71
c	0.8	0.8
C_1	2.28	2.28
C_2	0	0.00324
C_m	18.5	18.5

It is impractical to generate a motored pressure trace in Equation 7 that matches the mixture composition throughout the cycle. Instead, P_{motor} is modeled as a polytropic compression using the measured pressure and cylinder volume (V) beginning at the start of the respective closed portion of the cycle:

$$P_{\text{motor,main}} = P_{\text{IVC}} \left(\frac{V_{\text{IVC}}}{V} \right)^\gamma; P_{\text{motor,NVO}} = P_{\text{EVC}} \left(\frac{V_{\text{EVC}}}{V} \right)^\gamma \quad (8)$$

The instantaneous value of γ is once again pulled from the 1-D engine model results using the estimated gas composition and temperature.

Cycle Work and Exhaust Losses

Total cycle work (W) is directly calculated for each portion of the cycle using the measured pressure. There is some positive work during the exhaust gas exchange that is offset by a slightly larger pumping loss during induction.

$$W = \int P dV = W_{\text{NVO}} + W_{\text{ind}} + W_{\text{main}} + W_{\text{exh}} \quad (9)$$

The cycle exhaust enthalpy (H_{exh}) is calculated from the difference in enthalpy between exhaust valve opening (EVO) and EVC of the following cycle. The enthalpy at each valve event is obtained from the 1-D engine model, with H_{exh} computed as the RGF corrected mixture enthalpy at EVO (H_{EVO}):

$$H_{\text{exh}_i} = H_{\text{EVO}_i} (1 - \text{RGF}_{i+1}) \quad (10)$$

Excess enthalpy retained into the subsequent cycle at EVC (H_{EVC}) is similarly given as:

$$H_{\text{EVC}_{i+1}} = H_{\text{EVO}_i} \text{RGF}_{i+1} \quad (11)$$

Since cycle resolved emissions measurements are not possible for the current engine, a direct measure of unburned fuel in the exhaust for the cycle-of-interest is similarly not possible due to the alternate fire sequence. It is possible, however, to compute the deficit in total energy after the cycle work, heat transfer losses, and exhaust energy losses are accounted for. For the entire cycle (starting at EVC) the balance of input and output energy is as follows:

$$U_{\text{fuel inj}_i} + H_{\text{EVC}_i} + U_{\text{fuel,EVC}_i} = H_{\text{EVO}_i} + W_i + Q_{\text{HT}_i} + U_{\text{fuel,EVO}_i} \quad (12)$$

System inputs on the left-hand side include the injected fuel energy ($U_{\text{fuel inj}}$) along with the EVC mixture enthalpy (H_{EVC}) and fuel energy retained from the previous cycle ($U_{\text{fuel,EVC}}$). System outputs on the right-hand side include the aforementioned H_{EVO} , W , and Q_{HT} terms along with any unburned fuel at EVO ($U_{\text{fuel,EVO}}$).

The amount of $U_{\text{fuel inj}}$ was specified during the experiment and the enthalpy at EVC and EVO is directly calculated from the 1-D engine model. The total cycle heat transfer and work terms are calculated from Equations 6, 7, 8, 9. To determine $U_{\text{fuel,EVC}}$, exhaust for the engine operated only on the pre-conditioning injection schedule was collected (see Figure 1), with the fuel energy content determined from GC measurements; GC measured values compared favorably to conventional emissions analyzer measurements. Finally, $U_{\text{fuel,EVO}}$ is calculated directly from Equation 12.

By expressing $U_{\text{fuel,EVO}}$ and H_{EVO} as the sum of the respective exhausted and retained components into the following cycle, Equation 12 can be expressed more intuitively as:

$$U_{\text{fuel inj}_i} + H_{\text{EVC}_i} + U_{\text{fuel,EVC}_i} = H_{\text{exh}_i} + H_{\text{EVC}_{i+1}} + W_i + Q_{\text{HT}_i} + U_{\text{fuel,exh}_i} + U_{\text{fuel,EVC}_{i+1}} \quad (13)$$

Here, the exhausted fuel energy ($U_{\text{fuel,exh}}$) and the fuel energy transferred to the subsequent cycle ($U_{\text{fuel,EVC}}$) can be related to $U_{\text{fuel,EVO}}$ through the RGF as: