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Characterizing Factors Influencing SI Engine Transient Fuel Consumption for Vehicle Simulation in ALPHA

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ABSTRACT

The U.S. Environmental Protection Agency's (EPA's) Advanced Light-Duty Powertrain and Hybrid Analysis (ALPHA) tool was created to estimate greenhouse gas (GHG) emissions from light-duty vehicles. ALPHA is a physics-based, forward-looking, full vehicle computer simulation capable of analyzing various vehicle types with different powertrain technologies, showing realistic vehicle behavior, and auditing of all energy flows in the model. In preparation for the midterm evaluation (MTE) of the 2017-2025 light-duty GHG emissions rule, ALPHA has been refined and revalidated using newly acquired data from model year 2013-2016 engines and vehicles.

The robustness of EPA's vehicle and engine testing for the MTE coupled with further validation of the ALPHA model has highlighted some areas where additional data can be used to add fidelity to the engine model within ALPHA. A simple model based only on a steady-state fuel map will yield fuel consumption and GHG emissions lower than what is measured during a chassis dynamometer test due to a variety of factors present during transient operation.

This paper examines a) typical transient engine operation encountered over the EPA city and highway drive cycles, b) EPA's vehicle and engine testing to characterize that transient fuel usage, and c) changes made to ALPHA to better model transient engine operation. Topics examined in this paper include spark retardation for powertrain torque management, an engine power rate based fuel adjustment, additional fueling associated with deceleration fuel cut-off, and cylinder deactivation management.

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INTRODUCTION

Vehicle simulation is an established and effective method to predict a vehicle's fuel economy (FE) and greenhouse gas (GHG) emissions from a specific set of vehicle technologies. Accurate testing and analysis of fuel consumption from currently available vehicle technologies is key to validating the simulation model. Once validated the model can then be used to estimate the potential of various combinations of vehicle technologies to meet future GHG standards [1].

EPA has developed the Advanced Light-duty Powertrain and Hybrid Analysis (ALPHA) tool to inform upcoming GHG standards [2]. ALPHA is a physics-based, forward-looking, full vehicle computer simulation capable of (1) analyzing various vehicle types combined with different powertrain technologies and (2) evaluating the synergistic effects of the powertrain technologies to FE and GHG. The software tool is a MATLAB/Simulink based application. Recently, ALPHA has been enhanced to capture more realistic vehicle behaviors, including advanced gear selection and torque converter strategies, driver behavior, tire slip, and other dynamic effects in addition to the engine improvements discussed in this paper [3].

One of the key features in the development approach for ALPHA is the identification and implementation of adjustments for various transient behaviors seen in the powertrain. To properly identify technology effectiveness and synergy, model validation simulations attempt to match not just the overall test fuel economies, but a variety of time series data as well. Observations recorded across the many engine and vehicle test programs in support of the MTE has provided valuable data enabling more robust simulation of engine behavior and rules which incorporate various adjustment factors that can influence fuel economy. One motivation for the implementation of these features and adjustment factors in ALPHA is in response to the observation that vehicle simulation models relying solely on static input maps might tend to under predict fuel consumption and GHG emissions [4, 5, 6, 7, 8]. In reality, engine operation is too quick for a quasi-steady-state fuel consumption map to be fully representative of the engine's entire fuel consumption. Research indicates a significant difference can occur between vehicle simulations using a steady-state fueling map and experiments over various driving cycles [4]. As might be expected, EPA's experience is that the shortfall in fuel consumption is highly drive cycle dependent, with simulations of the more transient US06 cycle having a greater fuel consumption shortfall than simulations on EPA city/highway cycles. This paper will discuss some of the technical operating conditions that must be considered, and their individual effects on the determination of vehicle fuel economy.

One factor excluded from this study is the simulation of the additional fuel consumption that occurs during engine cold start conditions. Since both engine and transmission efficiency are reduced during cold start, thermal management strategies are employed on the combined powertrain system. Thus it is more appropriate to examine them in that context rather than in an engine centric study. Further information on one of the methods available in ALPHA to adjust simulation results for cold start operation required when simulating certification testing using FTP drive cycles is provided for reference [3].

The approach taken when tuning ALPHA parameters and comparing the results against chassis test data is to match the instantaneous data taken throughout the test with comparable simulation parameters, rather than just matching the cumulative overall test results. For example, ALPHA validations typically compare time dependent, engine torque, speed, and fuel flow along with other powertrain quantities rather than only total cycle fuel consumption. As a result, new technologies, whose effects may be dynamic and sometimes intermittent, can be more appropriately simulated within ALPHA. This provides additional confidence when various technologies are combined to simulate possible future technology packages, to determine the technologies' GHG effectiveness when used on different platforms or in synergy with other technologies.

BENCHMARKING VEHICLES AND ENGINES

To characterize engine transient fueling behavior a variety of vehicles and engines have been studied. This paper will focus on two vehicles; a Chevrolet Silverado with the 4.3*l* Ecotec engine, and a Ford F-150 with the 2.7*l* EcoBoost engine. Additional details of their specifications are listed in Table 1.

Table 1. Test Vehicle Specifications

Vehicle	2014 Chevrolet Silverado	2015 Ford F-150
Engine	GM Ecotec LV3	Ford EcoBoost
Displacement	4.3 liter	2.7 liter
Induction	Naturally aspirated	Turbocharger boosted
Rated Power	213 kW @ 5300 RPM	242 kW @ 5750 RPM
Rated Torque	413 Nm @ 3900 RPM	508 Nm @ 3000 RPM
Engine Technologies	VVT	Dual VVT
	Cylinder Deactivation	
Transmission	6L80 Automatic	6R80 Automatic
Other Technologies		Start-Stop

The vehicle chassis dynamometer and engine dynamometer benchmark testing for these two vehicles was performed at the National Vehicle Fuels and Emissions Laboratory (NVFEL) in Ann Arbor, Michigan. In separate tests the engines were instrumented, and many powertrain signals were recorded to capture each engine's detailed operating conditions. This data was used to construct steady-state fuel consumption maps for each engine which were used to validate ALPHA. More details about test cell equipment, instrumentation and test methods are available in [9, 10, 11].

Vehicle Testing on Chassis Dynamometer

Chassis dynamometer testing was conducted on both vehicles over the UDDS, HFET and US06 drive cycles. Triplicate tests over each drive cycle were conducted and only warm tests were considered when correlating test results to ALPHA. Inefficiencies (and additional fuel usage) related to cold start operation are influenced by both the engine and transmission. To simplify analysis, they were excluded from this study. Information on how ALPHA implements an adjustment to estimate fuel consumption during cold start conditions is available [3]. The average test results for each cycle are shown in Table 2.

Table 2. Chevrolet Silverado and Ford F-150 Fuel Economy Test Results

Vehicle	Test Cycle	Phase 1 MPG	Phase 2 MPG
Chevrolet Silverado	UDDS Average	22.30	19.66
	HWFET Average	30.47	-
	US06 Average	13.24	21.83
Ford F-150	UDDS Average	25.92	24.10
	HWFET Average	35.77	-
	US06 Average	13.54	24.35

Engine Mapping Using an Engine Dynamometer

Two testing processes were used to map each engine while it was on an engine dynamometer and tethered to its respective vehicle, intending to cover both steady-state and transient operation. During the first engine mapping process, each engine was operated at "steady-state" conditions over a range of speed and load points; more description of this process can be found in [9].

The steady-state operation was designed to gather data used to develop a nonlinear quasi-steady fueling map. The resulting BSFC maps of the GM Ecotec 4.3*l* and Ford EcoBoost 2.7*l* engines can be seen in Figures 1 and 2, respectively. Testing of the GM Ecotec 4.3*l* engine was conducted with the cylinder deactivation (CDA) technology both enabled and disabled. The testing also characterized the range of engine speed and torque where operation of cylinder deactivation was possible. Maps were developed for this engine in both modes but only the disabled map is shown below. Additional maps can be found in [9].

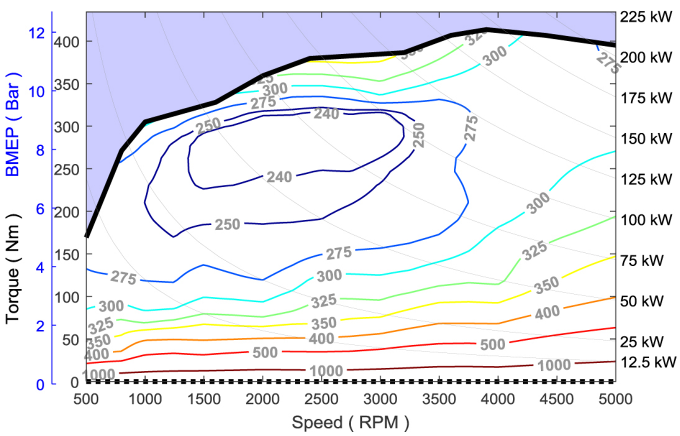


Figure 1. BSFC (g/kW·hr) map of the Silverado 4.3*l* Ecotec V6 with cylinder deactivation disabled.

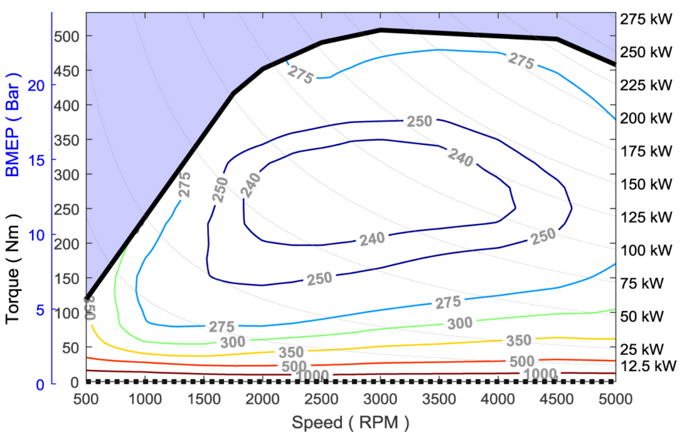


Figure 2. BSFC (g/kW·hr) map of the F-150 2.7*l* EcoBoost V6

The resulting steady state maps conducted on these and other engines, along with documentation covering the conversion of the data into a format suitable for simulation within ALPHA are available on the EPA website at: https://www.epa.gov/regulations-emissions-vehicles-and-engines/advanced-light-duty-powertrain-and-hybrid-analysis-alpha

Steady-state mapping also provided an opportunity to gather data to calibrate each engine's fuel injection quantity versus injection duration and fuel rail pressure. This calibration could then be used to examine fueling during the transient engine tests or chassis dynamometer drive cycles. A description of this calibration is included in the next section of this paper.

During the second engine mapping process, data from each engine was taken while operating during "ramp-up" conditions. The engine "ramp-up" tests were designed to emulate changes in engine torque seen during transient operation and to capture the inefficiency associated with the transient operation relative to the steady state map.

During the "ramp-up" tests, accelerator pedal position was increased from various starting speed and load points at different rates. The ramp-up test durations ranged from a quasi-steady-state 30 seconds to a near step input of 0.1 seconds. Dyno speed was held constant on most tests to mitigate the effect of the engine's rotational inertia on the torque measurement. After characterizing the rotational inertia of the engine and associated test apparatus, additional testing was conducted that varied speed as well. A more detailed examination of these data and their application to the engine model is discussed in the transient adjustment section of this paper.

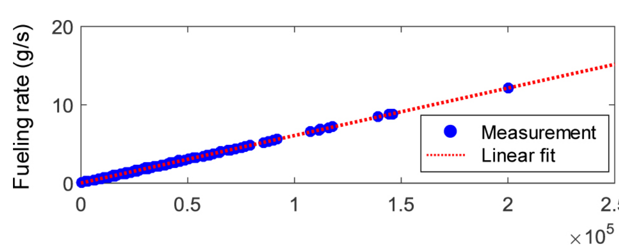
QUANTIFYING FUEL FLOW USING FUEL INJECTION

During steady-state mapping, an engine's fueling rate is usually measured with an external fuel flow meter. This approach works well for totalizing the amount of fuel consumed over steady-state or long-term operation such as that observed during drive cycle operation. However, because of the dynamics of an engine's fuel rail system, the instantaneous amount of fuel measured using the external fuel flow meter may not be the same as the actual amount of fuel injected to cylinders during a specific short time period.

Therefore, another technique was needed to determine the amount of fuel consumed by the engine during a short duration transient event. Using measurements of fuel rail pressure and commanded fuel injection duration from the steady state mapping, the relationship can be accurately characterized for analyzing short duration transient events. In general, the flow through a fuel injector can be modeled as flow through an orifice, so the fuel amount passing through the injector is proportional to the square root of the rail pressure, P_{rail} (as in the classic orifice equation) and the injection duration, Δt_{inj}. The total measured fueling rate for the engine, $\dot{m}_f$, is then proportional to the flow through each injector and the engine speed ω_{eng}. The general proportionality function is shown in equation 1.

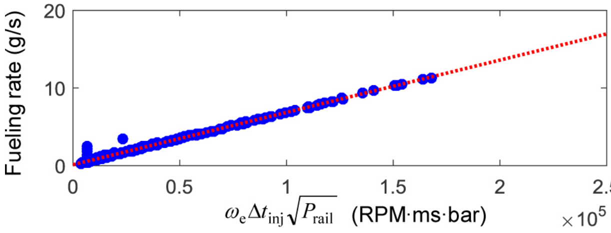
$$\dot{m}_f \propto \omega_{eng} \Delta t_{inj} \sqrt{P_{rail}} \tag{1}$$

To determine each engine's proportionality constant, data from the steady-state engine tests for both engines were obtained, and the measured fuel rate from the external flow meter was plotted against $\omega_{eng} \Delta t_{inj} \sqrt{P_{rail}}$. Figure 3 shows the linear fits that were used to determine the proportionality constants for the two engines which were used to compute the estimated *actual* amount of injected fuel during the duration of specific transient events.



a. GM Ecotec

Figure 3. Linear relation between fueling rate and a nonlinear function of engine speed, injection duration and rail pressure: (a) GM Ecotec engine and (b) Ford EcoBoost engine. Coefficients of the linear functions are identified by using a least square regression.



b. Ford Ecoboost

Figure 3. (cont.) Linear relation between fueling rate and a nonlinear function of engine speed, injection duration and rail pressure: (a) GM Ecotec engine and (b) Ford EcoBoost engine. Coefficients of the linear functions are identified by using a least square regression.

DETERMINE FUEL CONSUMPTION ADJUSTMENT FACTORS FOR TYPICAL TRANSIENT EVENTS

An engine operating in either regulatory drive cycle tests or real world driving situations is constrained by many factors such as meeting a driver's demand, satisfying tail-pipe emissions limits, mitigating NVH concerns, and maintaining combustion stability. As stated earlier a quasi-steady-state fuel map is not fully representative of actual engine fuel consumption and becomes increasingly less representative with increasingly aggressive driving. This section describes the implementation of four specific examples of transient fuel consumption adjustments that are built into ALPHA to deal with the individual effects on vehicle fuel consumption for:

- Powertrain torque management
- Changes in engine power (tip-in demand)
- Deceleration fuel cutoff
- Cylinder deactivation transitions

Fuel Adjustments for Powertrain Torque Management

Modern powertrains often involve multiple electronic devices that coordinate activities to provide the driver with a safe and comfortable driving experience. The primary interaction visible over regulatory drive cycles is coordination between the engine and transmission to maintain smooth operation during transmission upshifts. During an upshift, the rotational inertia of the engine, torque converter and other components connected to the transmission input shaft must be decelerated to match the angular velocity of the new gear ratio. This transfers angular momentum through the powertrain providing a momentary acceleration, which can lead to rough shift feel if not mitigated by powertrain torque management.

The engines and transmission can coordinate torque output during shifting to smooth out the bump in torque through changes in spark timing. Figure 4 contains data showing this shift smoothing behavior, as recorded from the Chevrolet Silverado. Fuel flow was calculated using the captured engine torque and speed with the steady-state map ($\dot{m}_{f,\,ss}$) and using the measured injector timing and calibration ($\dot{m}_{f,\,cal}$). The two calculations align close enough to provide an indication that the steady-state map in general provides an accurate reflection of fuel flow rate. During transmission upshifts, however, there is a significant deviation between the two quantities. The measured fuel flow is reasonably consistent through the gear shift while the steady-state map output shows a significant reduction attributable to the reduction in engine torque production. The lower chart in Figure 4 shows the source of the reduction in torque, a retarded spark timing during the shift events, a transient phenomenon that is not reflected in the steady-state engine maps.

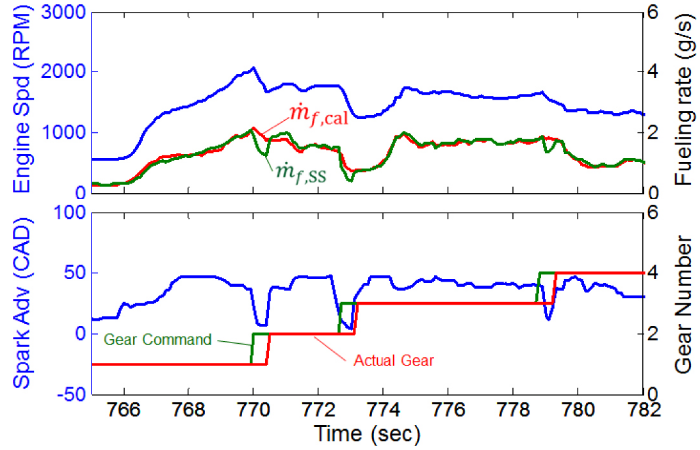


Figure 4. Fueling rates during a series of shift events from the Chevrolet Silverado. The fuel rate $\dot{m}_{f,\,cal}$ is calculated from measured injector duration using the injection calibration data, and $\dot{m}_{f,\,ss}$ is calculated using measured engine speed and load on the quasi-steady engine map.

The ALPHA engine model does not simulate engine operation at the spark timing level, but these deviations in timing do have an effect on fuel consumption. To account for the observed behavior during upshifts, a separate torque path was constructed to reduce output torque during these events. For transmission upshifts, the quantity of the torque reduction was calculated from the engine rotational inertia and rate of deceleration. This torque adjustment is applied to the engine output, but the unadjusted torque is used when interpolating the steady-state fuel map. This application is synonymous with the fast path torque used in engine controls, reducing torque by shifting spark timing away from maximum brake torque (MBT).

Engine torque reductions for transmission coordination in ALPHA are activated during transmission upshifts or torque converter lockup. Torque reductions can be disabled or reduced during situations of high driver demand such as a zero to sixty acceleration test when the additional shift harshness may be acceptable.

Currently ALPHA only utilizes this fast path torque for transmission upshifting and torque converter lockup. Other events where short duration rapid torque reduction events may be encountered, such as stability control, could be included as well but are not necessary for most fuel economy simulations.

Fuel Adjustments for Changes in Engine Power

Complete modeling of all interactions and dynamics associated with engine combustion and torque generation would be complex, time and resource intensive, and not practical for EPA's GHG simulation needs. Consequently, EPA examined prior literature for methods to accurately predict fuel consumption impacts of transient changes in engine load.

a. One option presented by Li, et al. in [5], a simple adjustment factor, which is proportional to a square of engine torque rate, was added to the base fueling rate calculated with a quasi-steady map. This adjustment factor was estimated based on correlating total fuel consumption and engine torque rate, however, it was not validated through experiments.
b. In another option presented by Chiara, et al. in [6], the authors considered air-path dynamics of an engine and characterized adjustment factors to estimate actual air mass flow rate. Although the accuracy was not quantitatively reported, better agreement could be seen when comparing modeled fuel consumption to measured time-series data. However, the model was developed for engine start-up operations only, and hence it has limitations in application over a wider range of engine operation.
c. In earlier EPA work, Newman, et al. [11] developed a transient fueling adjustment within ALPHA based on engine acceleration and showed that the difference in fuel consumption between measurement and simulation could be significantly reduced. However, that particular fuel adjustment development was based upon measurements using an external fuel flow meter. Data available later in this paper will highlight a limitation of using a fuel flow to characterize these short transient effects.
d. Lindgren in [7] and Mizushima, et al. in [8] discuss an adjustment factors was developed as a multiplier to a quasi-steady fueling rate. This adjustment factor was experimentally determined by analyzing engine data during transients. In [7], the author designed an experimental setup to account for various engine transients whereas authors in [8] utilized engine data collected over a World-Wide Harmonized Transient Cycle. The former approach is very simple and found to be accurate and hence it is adopted in our study. However, the previous research was conducted with off-road diesel equipment. Because of differences between gasoline and diesel engines and between off-road and on-road vehicles, the adjustment factors provided in [7] would likely not be representative of engines in the light duty fleet.

Careful study of these options lead EPA to conduct the "ramp-up" experiments described previously in the "Engine Mapping Using an Engine Dynamometer" section of this paper. The testing program was designed to quantify the transient fuel consumed during ramp-up operation, and examine differences in air-path and fuel-to-torque dynamics for naturally-aspirated and turbocharger-boosted induction systems.

The ramp-up engine dynamometer test provides a glimpse of how engine operation is altered during rapid transients. Figure 5 shows two separate transient tests covering the same load and speed range, but at two different transition periods of 30 seconds and 0.1 second ramp-ups. The slower transition shows throttle, spark, intake and exhaust cam actuations that change smoothly and predictably as might be found during steady-state testing. The faster transition shows more aggressive actuator movement as the ECU attempts to deliver the requested torque while maintaining stable operation and controlling emissions. This deviation relative to the steady-state actuator calibration leads to less efficient engine operation, and thus additional fuel consumption, which must be considered in vehicle simulation for accurate fuel economy evaluation.

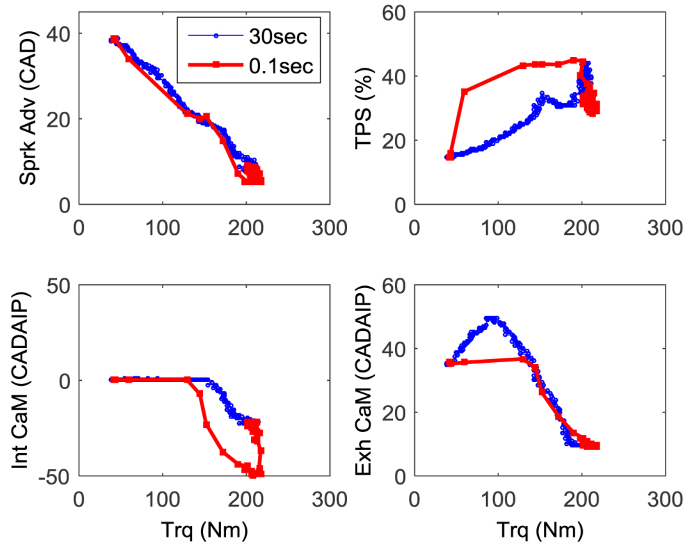


Figure 5. Measured actuator behaviors on the 2.7*l* EcoBoost engine during transition from 40 Nm to 210 Nm at two different pedal transition rates. While the actuator start and end points match, the fast transition features a more aggressive throttle opening and larger variation in intake cam phasing. These deviations represent attempts to meet the requested torque, but are probably less efficient than the slow (quasi steady state) transient.

Determination of Power Rate Adjustment Factor

To account for the lower engine efficiency during transient operation associated with engine ramp-up operation, the transient fueling rate $\dot{m}_{\mathrm{f,tr}}$ was modeled as a simple function given by:

$$\dot{m}_{f,\mathrm{tr}} = \mathrm{AF} \cdot \dot{m}_{f,\mathrm{ss}} \tag{2}$$

Where,

AF = Adjustment Factor (to be determined)

$\dot{m}_{f,\mathrm{SS}}$ = fueling rate from steady-state map

The adjustment factor (AF) for a given transient test is computed by comparing the actual fuel consumption as measured using the injector data with the predicted steady-state fuel consumption during a ramp-up operation as follows:

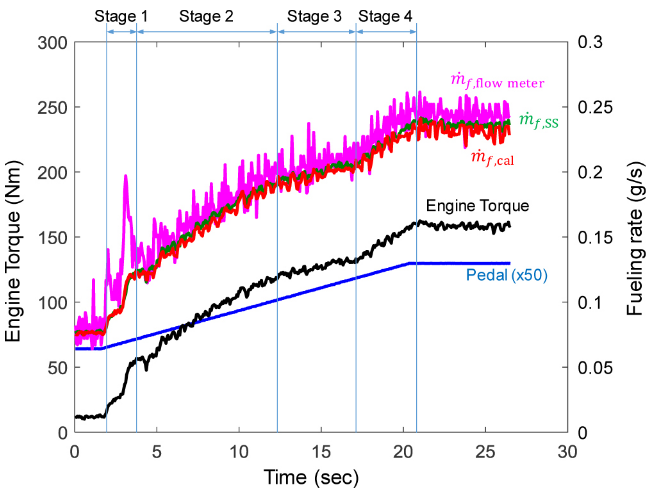
$$\mathrm{AF} = \frac{\int \dot{m}_{f,\mathrm{cal}} dt}{\int \dot{m}_{f,\mathrm{SS}} dt} \tag{3}$$

Where,

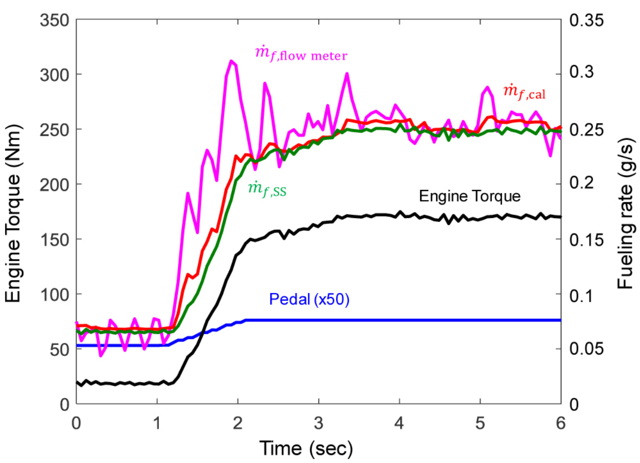
$\dot{m}_{f,\mathrm{cal}}$= fuel rate by injection calibration

$\dot{m}_{f,\mathrm{SS}}$= fuel rate from steady-state map

As mentioned in the experimental set-up section, the driver's acceleration pedal position was used as a control input. Fuel injection duration was measured for each cylinder and engine cycle using the relationship discussed above. For the Silverado Ecotec engine, a nonlinear relation between the pedal position and engine torque was observed and hence multiple power rates were observed during a linear ramp-up in accelerator pedal as shown in Figure 6(a). Both examples in Figure 6 show the significant deviation between fuel consumption measured using the injector calibration and fuel flow meter. The additional fuel consumption observed by the flow meter is related to the pressure increase of the fuel rail.



a.



b.

Figure 6. Engine response during ramp-up operation: (a) 20 second ramp-up test on the GM Ecotec and (b) 1 second ramp-up test on the Ford EcoBoost; For the purpose of signal comparison, Acc. Pedal position and fueling rates (fuel flow meter, steady-state fuel map and fuel injector calibration) are scaled.

The data for each ramp up test conducted on both vehicles was divided into segments of similar rate of change in measured engine torque. The adjustment factor was computed for each segment using equation 3. The data showed correlation with the rate of change in engine power as seen in Figure 7. However, the significant amount of scatter in the data, especially at the shorter ramp durations, which feature higher rates of change, do highlight the need for further testing.

Characterizing the transitions by power rate involves speed in the calculation. Tests run with the engine dynamometer at constant speed mitigate the influence of the engine rotational inertia on measured torque yielding a good dataset. After characterizing the engine inertia, additional tests were run conducting similar pedal transitions while engine speed was ramped up or down. These tests showed good agreement with, but greater scatter than the constant speed data.

In order to compare this adjustment with respect to various engine displacements the rate of change in engine power was normalized by the maximum engine power yielding normalized engine power rate:

$$\dot{P}_n = \frac{\dot{P}}{P_{max}} \tag{4}$$

Where,

$\dot{P}$ = rate of change in engine power

P_{max} = maximum engine power

Once additional test data is available and the power rate based adjustment factors can be computed for additional engines, an improved normalization method may yield an improved method for comparing efficiency degradation resulting from this type of transient operation.

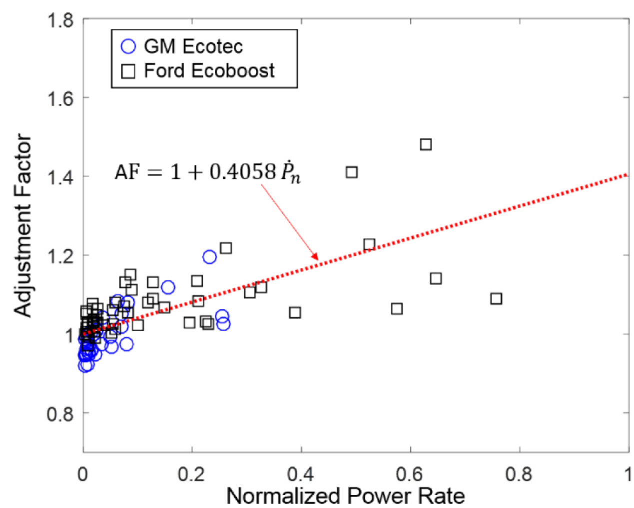


Figure 7. Computed adjustment factors for GM Ecotec (black circles) and Ford EcoBoost (blue squares) and a linear curve-fit.

Power Rates Observed over Regulatory Drive Cycles

To estimate the effect of incorporating a power rate based transient fuel adjustment factor on fuel consumption during vehicle simulations, it is important to determine the "range" of engine power rates observed during actual vehicle testing over the regulatory drive cycles. To this end, a statistical analysis of engine transients measured during vehicle testing on a chassis dynamometer was conducted. During the steady state mapping performed on the engine dynamometer, the engine's OBD torque data was found to corroborate well with the engine dynamometer's measurements. Consequently, we were comfortable using engine's estimated power rates encountered during the chassis testing (derived using the measured engine speed and the engine torque signal available from the ECU via OBD). These estimated rates were adequate for determining the overall distribution of engine power rates encountered for each of the three test cycles: the UDDS, HWFET, and US06.

Due to the presence of high frequency noise in the measured torque signal, the torque signal was filtered with a low-pass filter having a cut-off frequency of 1 Hz. Measurements taken during gear shift events were excluded from the computation to eliminate torque fluctuations due to powertrain management that could distort the results. Finally, a cumulative probability distribution of power rate over three driving cycles on the Chevrolet Silverado was computed as shown in Figure 8. The values of power rate covering 95%, 98% and 99% of engine operations for each drive cycle are also shown in the figure. The data show that nearly all engine transient power rates, even during the more aggressive US06 cycles, are below 100 kW/s for the Chevrolet Silverado. This corresponds to a normalized power rate of approximately 0.4 when comparing to the data in Figure 7.

Based on this statistical analysis, only data taken at power rates under 100 kW/s were used to develop the transient fuel adjustment model. Using these data, the following function was obtained by a least square regression shown in Equation 5:

$$AF = \begin{cases} 1 + 0.4058\,\dot{P}_{\mathrm{eng},n} & \dot{P}_{\mathrm{eng},n} \geq 0 \\ 1 & \text{o.w.} \end{cases} \tag{5}$$

Where,

$\dot{P}_{\mathrm{eng,n}}$ = normalized power rate

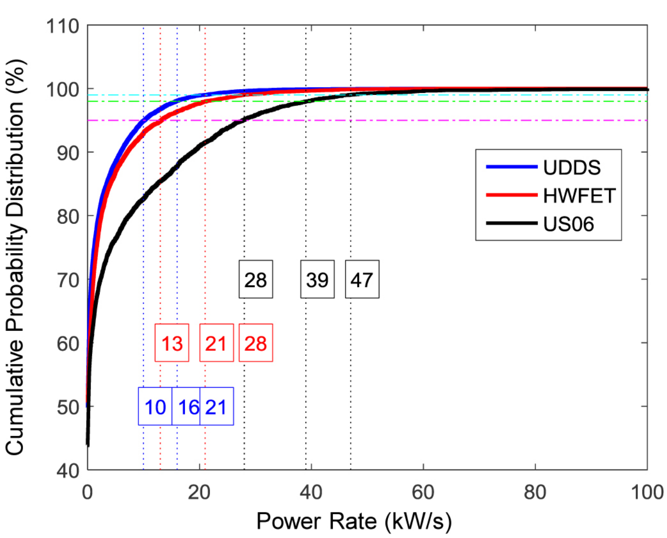


Figure 8. Cumulative probability distribution of engine power rate during three federal drive cycles on the Chevrolet Silverado: UDDS, HWFET, and US06. Three values represent power rates covering 95, 98 and 99% of engine operations, respectively.

Using Power Rate Adjustment Appropriately in ALPHA

The transient power rate adjustment factor calculation discussed above can be easily implemented in ALPHA. However, to utilize a power rate in vehicle simulation, it is important to ensure the power rates calculated during simulation are similar to those in actual vehicle operation. Calculation of power rate involves taking a first order derivative, and can be much more sensitive to small modeling changes.

In older versions of ALPHA, engine output torque, which would also be used for interpolation of the steady-state map, was determined directly from the simulated driver accelerator pedal position. Thus, the only constraint limiting engine torque production was the calibration of the driver model. While the simulated driver calibration may still contribute to the scale of additional fueling for transient conditions, constraining the engine's transient response seeks to reduce this effect.

To ensure the robustness of the power rate based transient fuel adjustment factor, and to more appropriately handle the torque response of turbocharged engines, the engine's "ramp up" test data was used to characterize the air-path dynamics within the engine. The actual torque generation process is highly influenced by nonlinear dynamics of air-mass flow, Engine Control Unit (ECU) commands and actual responses of the various actuators. Since it is difficult to model those behaviors individually, the approach presented in [6] was adopted in this work; that is, the torque generation process is modeled as a first-order system given by:

$$\dot{P}_{\mathrm{tr}} = \gamma P_{\mathrm{tr}} + P_{\mathrm{com}} \tag{6}$$

Where,

γ = time constant of torque generation process

P_{com} = commanded engine power

P_{tr} = transient output power

The time constant γ is identified from the torque response when a step change in pedal signal is applied. The time constant is defined as the time the system output reaches 63.2% of its final value when a step input is applied to the system.

Figure 9 shows the identified time constant for each of the torque response ramp tests using the 0.1 second pedal transition. The naturally aspirated 4.3*l* Ecotec shows a consistent response time constant around 0.2 seconds. The 2.7*l* EcoBoost shows a similar response time constant at light load, but considerably slower response when operating under boosted conditions.

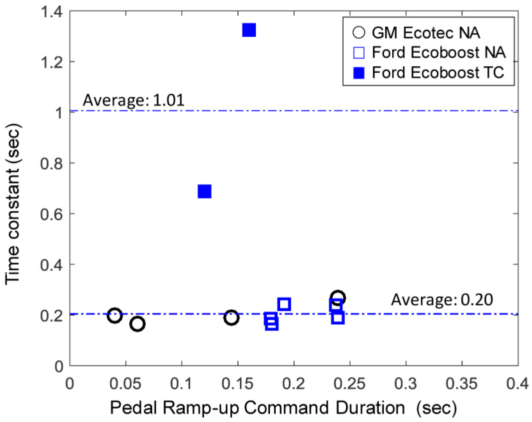


Figure 9. Identified time constants of torque generation process for naturally aspirated and turbocharger-boosted operation.

Adding the appropriate delays to the engine response within ALPHA was accomplished by implementing two separate transfer functions; one associated with the engine's air-path dynamics, and one for its boosted response. The delay associated with primary air-path dynamics are represented by a first order response on engine power with a 0.20 second time constant matching the test data. The delay associated with boosted response is implemented as a moving upper limit on the available engine torque. A lookup table versus engine speed provides a minimum torque that is always available, corresponding to the naturally aspirated operating zone. Figure 10 shows the measured manifold pressure (MAP) during steady-state mapping highlighting the regions where the faster naturally aspirated response would be expected.

The simulation control diagram in Figure 11 shows the relationship of the torque generation dynamics and adjustment factor to predicting transient fueling rates. Figure 12 shows a comparison of the simulated first order response to a step input with measured data from the Silverado Ecotec engine.

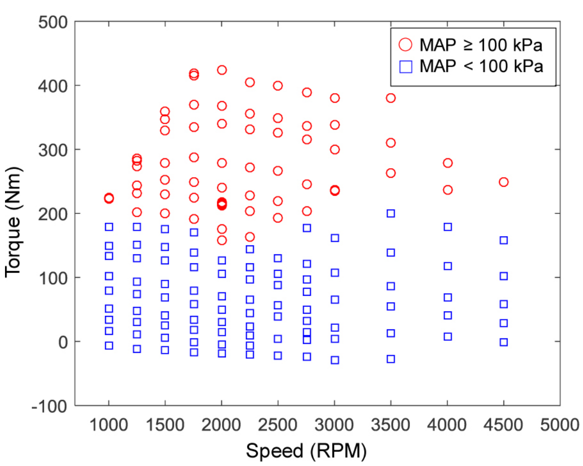


Figure 10. Measured manifold air pressure data from 2.7*l* EcoBoost engine during steady-state operation

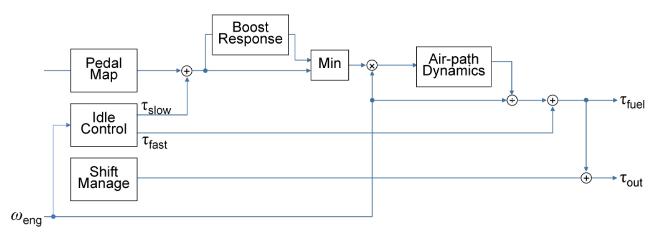


Figure 11. Implementation of the torque generation dynamics and adjustment factor in predicting transient fueling rates.

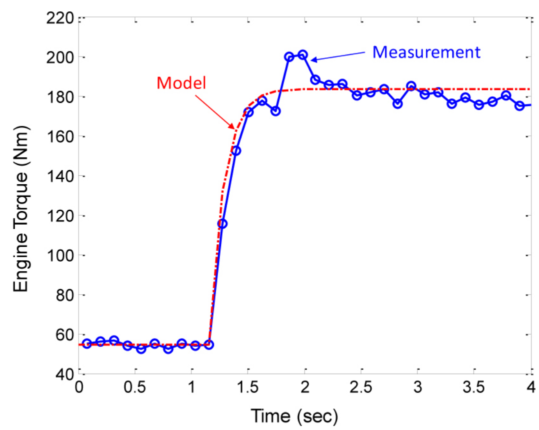


Figure 12. Comparison of simulated and measured torque responses due to a step input on Ecotec 4.3*l* engine at 2000 RPM

Torque requests above this threshold will cause the available torque to increase with a time constant of 0.77 seconds up to the maximum torque observed during the steady-state mapping. The cascading of these two time constants yields a response similar to the observed 1.01 second. Figure 13 show a comparison of this response during a 0-60 acceleration simulation for both turbocharged and naturally aspirated engines. The available torque increases quickly, then at time 20 the turbocharged example is limited by its available boost. For both induction systems, the output torque response is fast, but the delay and smoothing are noticeable.

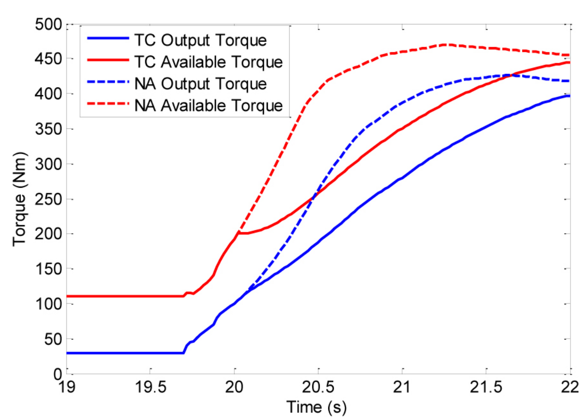


Figure 13. Comparing response rate of NA and TC during 0 - 60 simulation

Adding delays to the engine torque production creates an additional complication regarding idle speed control. Without considering air path dynamics, the torque requested by the idle speed was maintained by a simple Proportional-Integral-Derivative (PID) control. Since the engine torque production was instantaneous, idle speed could be precisely controlled. However, slowing the torque response caused the model to exhibit unsatisfactory idle speed control. The idle speed control was revised to provide separate fast path and slow path torque targets. This mirrors the idle speed control utilized in modern engine controls. Additional slow path (throttle controlled) torque was added to provide the fast path (spark timing controlled) torque the appropriate control authority. Unlike the fast path torque utilized to reduce torque during shifting, the idle speed control fast path torque was included when determining the torque for lookup in the steady-state fuel map as it would have been active when the steady-state map was collected near idle conditions.

The power rate based transient fuel adjustment seeks to estimate differences between actual engine operation and its representation via a steady-state map due many factors. Relative to the other adjustments discussed in this paper it is more difficult to directly observe in the chassis test time series data. The ramp-up tests and resulting adjustment factor improve the correlation between ALPHA and test data. Further study of additional engines may help lead to a more detailed understanding of this specific adjustment.

Fuel Adjustments for Deceleration Fuel Cutoff

An area where engine fueling during chassis testing deviated from simulation using only the steady-state fuel map was during the time after deceleration fuel cut-off (DFCO) events. Extended operation in DFCO causes catalyst cooling and excess levels of oxygen that will inhibit the functionality of the catalyst to decompose NOx when fueling resumes. To maintain proper emission control after a DFCO event, additional fueling may be required to quickly restore optimal catalyst operation.

Time series data from a variety of cars and trucks were compared against the simulation results using the steady-state map and transient adjustment factor. Deviations between the measured fuel consumption and results of the steady-state fuel map were compared. Figure 14 shows an example of the deviation. The fuel flow estimate from the steady-state map agrees well prior to fuel cutoff, however on resumption of fueling there is a substantial discrepancy. The bottom chart shows the ratio of observed and steady-state map flow rates, with observed fueling exceeding the steady-state value by a factor of 3 then tapering down over approximately two seconds. It should also be noted that depending on the catalyst and engine calibration, additional fueling may not be required for every DFCO event.

DFCO within ALPHA is triggered when the torque requested from the driver and idle speed control are both zero. Additional constraints can be included as well, such as minimum transmission gear or vehicle speed. A timer is used to measure the duration of the DFCO event. Upon exiting DFCO additional fueling is only triggered if the event was longer than the calibrated minimum duration. The adjustment itself is implemented as a time based multiplier on the base fuel map. The scale of the multiplier and the rate at which it ramps out is calibrated to match test data. An upper limit on the additional fueling is included as another configurable parameter. Table 3 shows the ALPHA DFCO adjustment calibration for a few different vehicles.

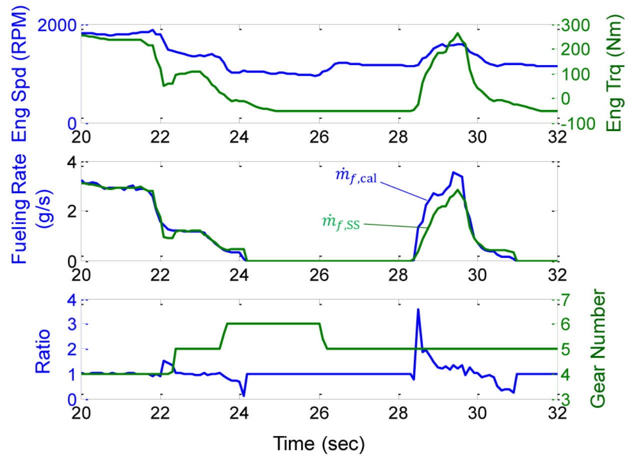


Figure 14. Measured fueling before and after DFCO event showing disagreement with steady state map.

Table 3. Alpha DFCO calibration for various vehicles/engines

Vehicle Engine	Maximum Multiplier	Ramp Out Duration (seconds)	DFCO Minimum Duration (seconds)
2014 Chevy Silverado Ecotec 4.3*l* V6	2.0	2.0	2.0
2015 Ford F-150 EcoBoost 2.7*l* V6	1.3	4.0	2.0
2013 Ford Escape EcoBoost 1.6*l* I4	1.4	4.0	0.5

Fuel Adjustments for Cylinder Deactivation Transition

Cylinder deactivation (CDA) is a technology enabling the GM 4.3*l* Ecotec and other engines to meet current and future GHG standards. It was observed in chassis dynamometer testing that when operating within the speed and load range where CDA is available, it was active only 60% of the time [9]. Thus, running a simulation using a steady-state engine map collected with cylinder deactivation active throughout its available range would dramatically overstate its effectiveness in GHG reduction.

To better simulate CDA, separate fuel consumption maps with and without CDA active were added to the ALPHA engine model. A variety of methods for transitioning between the two maps are available in ALPHA. A simplified strategy utilizing a constant interpolation of the two maps represents a generic solution useful for examining the technology effectiveness on different vehicle platforms or mated with a different powertrain where the specific constraints of CDA are unknown. A configurable cylinder deactivation control logic is also under development, but test data from additional vehicles is necessary to develop a robust strategy applicable across multiple vehicle platforms.

Test data were also analyzed to characterize any fuel consumption adjustment associated with transitions into and out of cylinder deactivation. Figure 15 shows data from a test conducted on the Silverado at steady speed and load. Utilizing an OBD scan tool, the CDA was activated and deactivated multiple times. The steady-state fueling rates are accurately predicted for each mode. However, the transitions consume additional fuel. When CDA is activated it takes approximately one second for the fueling rate to match levels from the steady state map. This duration is consistent with the transition in cam phasing. Upon exiting CDA, substantially more fuel is consumed than indicated by the steady state map, consistent with the observed retarding of spark timing to reduce torque as the two cylinders are brought back into operation.

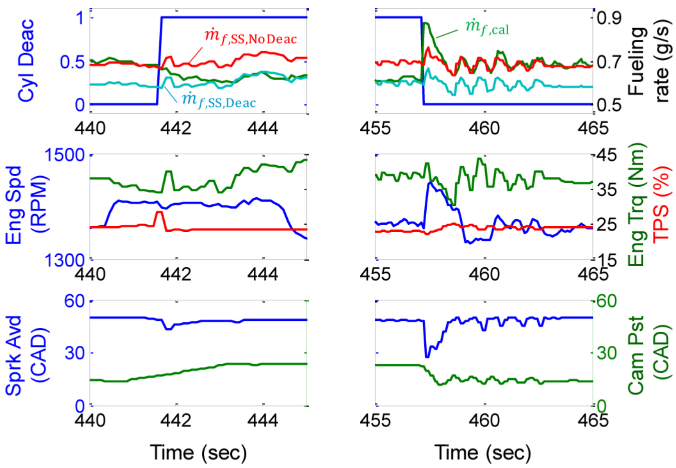


Figure 15. Measured actuator behaviors during transition from full cylinder to CDA mode (left) and CDA to full cylinder mode (right).

Implementation of the adjustments for transitions into and out of CDA within ALPHA consist of two separate components. The transition into CDA is not simulated as an instantaneous step change, but rather as a blended transition. The outputs of the two steady state fuel maps are blended over a specified duration. The time value used for the Ecotec 4.3*l* is 1.0 seconds. Transitions out of CDA consume additional fuel, so its adjustment uses a multiplier similar to that used for DFCO. The calibration for the Ecotec 4.3*l* is a maximum multiplier of 1.8 ramped out over 2 seconds.

FUEL ECONOMY EVALUATION AND ANALYSIS

To verify the result of including the effects of the fuel consumption adjustments addressed above, ALPHA fuel economy simulations were performed over various drive cycles such as UDDS, HWFET and US06. In ALPHA, engine torque is determined from the accelerator pedal linearly mapping to engine power. An additional torque request is provided by an idle speed control. Engine speed is determined from the physics as dictated by the downstream components. Transmission, accessory and road load losses were mapped based on measurement data. Separate simulations were run for each chassis dynamometer test using the vehicle speed, transmission shift points and torque converter lockup schedule that was observed during benchmarking of each vehicle over the same cycles.

The comparison of the average fuel economy results from warm cycle simulations and warm chassis dynamometer tests can be found in Table 4. Over the warm UDDS and HWFET cycles (which are core elements of the cycles used for the light-duty vehicle GHG standards), the maximum difference was 1.45%. Therefore, it is reasonable to conclude that fuel economy results from ALPHA simulation with inclusion of transient engine behaviors are sufficiently accurate for intended purposes.

Table 4. Comparison of fuel economy results obtained from warm chassis dynamometer tests versus ALPHA simulations

Vehicle	Drive Cycle	Chassis Test (avg mpg)	ALPHA Simulation (avg mpg)	FE Diff (%)
Chevrolet Silverado	UDDS Phase 1	22.30	21.95	-1.5
	UDDS Phase 2	19.66	19.47	-1.0
	UDDS Total	20.85	20.59	-1.2
	HWFET	30.47	30.48	+0.1
	US06 Phase 1	13.24	13.05	-1.4
	US06 Phase 2	21.83	21.39	-2.0
	US06 Total	19.08	18.73	-1.9
Ford F-150	UDDS Phase 1	25.92	25.95	+0.1
	UDDS Phase 2	24.10	23.84	-1.1
	UDDS Total	24.95	24.82	-0.5
	HWFET	35.77	35.24	-1.5
	US06 Phase 1	13.54	13.64	+0.8
	US06 Phase 2	24.35	24.01	-1.4
	US06 Total	20.72	20.65	-0.3

As a sensitivity study to see how well the transient fuel consumption adjustments worked with more aggressive drive cycles, EPA also ran simulations using the US06 cycle, a more aggressive test cycle with high fuel consumption per mile which is not used in the simulation of GHG emissions for the light-duty GHG standards. The average difference between ALPHA simulations and average test data using the US06 cycle was 0.8% to 2.0%, and the maximum difference was only 1.4% to 2.0% for Phase 1 of the US06 cycle. Since phase 1 of the US06 cycle is a very short cycle with high transient rates, this level of agreement is remarkable. EPA also believes that with some additional effort the transient fuel adjustment algorithms could be further improved for the more aggressive cycles like the US06.

Furthermore, fuel consumption breakdown over the three drive cycles is provided in Table 5. Due to rounding, some of the percentages may not add up to exactly 100%.On average, the transient fuel adjustment contributions of power rate changes, transmission shifting, CDA transitions and DFCO are 1.4%, 0.4%, 0.3%, and 0.3%, respectively, and transient engine behaviors affect total fuel consumption by 1.3 to 3.5%.

The transmission gear shifting transient fuel adjustment is most noticeable on the UDDS. This should be expected, as this cycle has the largest total number of gear shifts (for example, 103, 17, and 70 counts were observed in UDDS, HWFET, and US06, respectively, in chassis dynamometer tests for Chevrolet Silverado). The power-rate adjustment is most noticeable on the US06, due to the aggressive transients in US06 compared to the other drive cycles.

The CDA transient fuel adjustment is around 0.34% for all three driving cycles. There are a greater number of CDA transition events in the UDDS and HWFET cycles than observed in the US06 cycle (UDDS: 143, HWFET: 104, US06: 60). As can be seen in Figure 16, the CDA transitions during the US06 occur at points of higher engine operating power. Since the CDA transition fuel adjustment factor is a multiplier on the base steady-state fuel map, the adjustments on a per transition basis will be larger for the US06 cycle.

Lastly as shown in Table 5, the DFCO transient fuel adjustment is also most noticeable on the US06, as DFCO tends to occur at higher vehicle speed and/or in higher transmission gears. This is furthered by more aggressive accelerations following the DFCO events, which increase the effect of the adjustment.

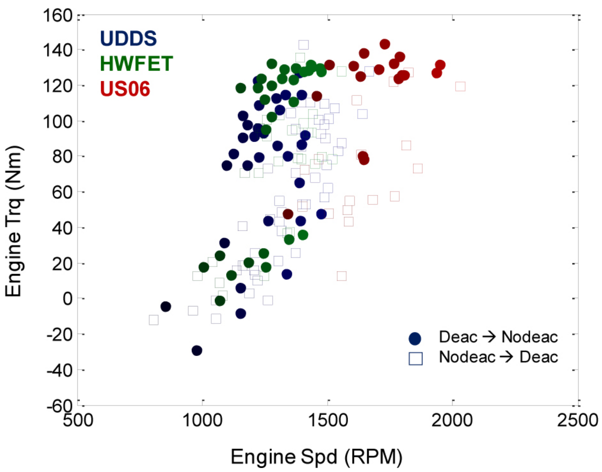


Figure 16. Engine speed and torque during CDA transitions

CONCLUSIONS

This paper presents an approach to determine rules that account for dynamic fuel consumption during engine transients, and implement the appropriate adjustments into the ALPHA model. The engines of interest were GM's 4.3*l* Ecotec and Ford's 2.7*l* EcoBoost which are installed in a 2014 Silverado and a 2015 F-150, respectively. EPA's test procedures for these engines provided data to:

1. Calibrate the engine's fuel injectors to enable the use of acquired injector data to calculate a more precise instantaneous measurement of fuel consumption during the engine's transient operations,
2. Determine the time constant for the engine's air-path and fuel-to-torque dynamics, and
3. Ascertain the transient fuel adjustment factors, multipliers to the steady-state fuel consumption data, to account for extra fueling associated with engine ramp-ups when increasing engine torque, transmission shifting, cylinder deactivation transitions, and fuel shutoff during decelerations (DFCO). Each of these fuel adjustment factors can be characterized by the time series test data and help improve the resulting correlation between simulation and chassis testing results.

Two different air induction systems were investigated: naturally-aspirated vs. turbocharger-boosted engines. The engine test programs provided the data necessary to develop the adjustment factors and improve correlation between ALPHA and test data. ALPHA simulation results show that the fuel economy prediction could be significantly improved by including modules to emulate engine dynamics and to predict transient fuel consumption. Procedures have been added to EPA's engine benchmarking procedures to gather further data so ALPHA can determine the appropriate level of transient fuel consumption adjustments as result of new technology.

Overall, the scale of the transient fuel adjustment factors range between 1.2% and 3.3% of total fuel consumed. Surprisingly, they were lower than the results from other literature that predicted differences ranging from 2% up to 8% [4, 5]. Some of this difference may be attributable to the specific engines examined, their respective technologies such as direct injection and the level of development for their electronic controls managing transient operation. Our technique of calculating fuel flow from an injector calibration rather than a fuel flow meter is likely to play a role as well. The ramp-up testing showed significantly more consumption when measured via the fuel flow meter than via the injectors attributable to changes in fuel rail pressure. It is also important to note that the factors discussed in this study represent only a portion of the total vehicle losses that are affected during more aggressive transient operation. For example, duty cycles like those found in the US06 drive schedule affect gear selection thus altering speed and load operating points, yielding further increases in fuel consumption.

Table 5. Fuel consumption breakdown over drive cycles from ALPHA simulation

Vehicle	**Drive Cycle**	**Steady-State Map** Fuel Mass (g) / %	**Power Rate Transient Adj.** Fuel Mass (g) / %	**Transmission Shifting Adj.** Fuel Mass (g) / %	**CDA Transition Adjustment** Fuel Mass (g) / %	**Post DFCO Adjustment** Fuel Mass (g) / %	**Total** Fuel Mass (g) / %
Chevrolet Silverado	UDDS	995.03 / 97.68	11.87 / 1.17	7.12 / 0.70	2.90 / 0.28	1.75 / 0.17	1018.67 / 100
	HWFET	932.99 / 98.19	9.00 / 0.95	2.48 / 0.26	3.67 / 0.39	2.03 / 0.21	950.17 / 100
	US06	1167.95 / 96.48	29.02 / 2.40	4.92 / 0.41	4.06 / 0.34	4.65 / 0.38	1210.60 / 100
Ford F-150	UDDS	830.11 / 98.04	8.07 / 0.95	5.98 / 0.71	N/A	2.53 / 0.30	846.71 / 100
	HWFET	807.57 / 98.73	5.08 / 0.62	1.48 / 0.19	N/A	3.79 / 0.46	817.92 / 100
	US06	1052.23 / 96.61	28.46 / 2.61	3.02 / 0.28	N/A	5.46 / 0.50	1089.17 / 100

EPA's analysis of the vehicle chassis test data has also highlighted that the predominant factor in determining engine power, and thus power rate of change, is the driver technique. While the overall engine power rates within ALPHA are similar to those seen in the vehicles it has been observed that the model's driver behaviors are somewhat different than real driver behaviors. Future work may include quantifying these differences and possibly developing a revised driver model.

Future work may also focus on better characterizing the transient power rate based adjustment factor. The two engines characterized in this study are deployed in similar vehicles, full size trucks. Testing on additional engines will help to better characterize and quantify the fueling adjustment required for transient operation of other engines, and may eventually highlight trends with regard to engine displacement, vehicle segment or differences attributable to individual technologies. Both of the primary engines for this study featured direct injection. A similar study of a port injected engine may highlight an additional benefit of GDI. Additionally, the influence of decoupling changes in engine speed and torque transients on fueling rate require additional investigation. As additional engines are tested the characterization of additional fueling adjustments may identify revisions to the generic assumed values for each adjustment within ALPHA.

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DEFINITIONS/ABBREVIATIONS

ALPHA - Advanced Light-Duty Powertrain and Hybrid Analysis tool

CDA - Cylinder Deactivation

DFCO - Deceleration Fuel Cut Off

HWFET - US EPA Highway Fuel Economy Test

NA - Naturally aspirated

MAP - Manifold Air Pressure

MBT - Maximum Brake Torque

MTE - Midterm Evaluation of the 2017-2025 Light-Duty GHG Emissions Rule

OBD - On Board Diagnostics

TC - Turbocharger or Turbocharger-boosted

UDDS - US EPA Urban Dynamometer Driving Schedule

US06 - US EPA US06 Supplemental Federal Test Procedure