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Fleet-Level Modeling of Real World Factors Influencing Greenhouse Gas Emission Simulation in ALPHA

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ABSTRACT

The 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 internal energy flows in the model.

In preparation for the midterm evaluation (MTE) of the 2017-2025 light-duty GHG emissions rule, ALPHA has been updated utilizing newly acquired data from model year 2013-2016 engines and vehicles. Simulations conducted with ALPHA provide data on the effectiveness of various GHG reduction technologies, and reveal synergies that exist between technologies. The ALPHA model has been validated against a variety of vehicles with different powertrain configurations and GHG reduction technologies.

This paper will present an overview of the laboratory benchmarking that was done to support validation of the ALPHA model. The paper discusses a variety of real world factors that influence the simulation of fuel economy and GHG emissions that are often overlooked. Updates have been made to the ALPHA model to reflect additional losses such as tire slip and more detailed representations of the electrical system and accessory loads. The characterization of a core set of future technologies is examined, focusing on developing generic calibrations for driver behavior, transmission gear selection and torque converter lockup that are representative across a wide range of vehicles and transmissions. Finally, the paper illustrates how a set of core future technologies can be used to model GHG emissions from future vehicle fleets.

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INTRODUCTION

During the development of the light-duty (LD) greenhouse gas (GHG) standards for the model years (MY) 2017-2025, Environmental Protection Agency (EPA) utilized a 2011 LD vehicle simulation study [1] from the global engineering consulting firm, Ricardo, Inc. This study provided a round of vehicle simulations to predict the effectiveness of future advanced technologies. Use of data from this study for the final rule is documented in the August 2012 EPA and NHTSA Joint Technical Support Document [3].

The 2017-2025 LD GHG rule required that a comprehensive advanced technology review, known as the midterm evaluation (MTE), be performed to assess any potential changes to the cost and the effectiveness of advanced technologies available to manufacturers during MY2022-2025. EPA developed the Advanced Light-Duty Powertrain and Hybrid Analysis (ALPHA) model [2] to enable the simulation of current and future vehicles, and as a tool for understanding vehicle behavior, greenhouse gas emissions and the effectiveness of various powertrain technologies. For GHG emissions, ALPHA calculates a vehicle's grams of CO_2 per mile based on test fuel properties and vehicle fuel consumption. No other emissions are calculated at the present time, but future work on other emissions is not precluded.

The work behind this paper was performed by EPA's National Center for Advanced Technology (NCAT), located at its National Vehicle and Fuel Emissions Laboratory in Ann Arbor, Michigan. NCAT engineers have further developed ALPHA as an in-house research tool to explore in detail current and future advanced vehicle technologies. ALPHA is being updated to more accurately model light-duty vehicle behavior. To validate the performance of ALPHA, EPA has performed in-depth vehicle benchmarking involving conventional and hybrid vehicles.

Focus

This paper reviews EPA's overall program to benchmark and validate ALPHA using more than 25 recent model year vehicles, engines and transmissions. It also focuses in detail on the approach of using ALPHA with a core set of current and future technologies to model CO_2 emissions from future fleets. After closely matching the behavior of specific validation vehicles, observed losses are converted into a simpler, more generic form to establish more general patterns of vehicle operation and fuel use. These general patterns are applied in ALPHA to allow for a clearer comparison of component effectiveness across various technology packages and vehicle classes. The calibration of the model essentially blends the unique control strategies and behavior of the individual vehicles used to validate the model (without overlooking fuel required to meet vehicle and powertrain performance requirements observed during vehicle benchmarking) and provides a more appropriate calibration for fleet level GHG modeling.

While ALPHA has been used to confirm and update, where necessary, efficiency data from the 2011 LD vehicle study, its primary use as an in-house research tool for the MTE is to study in detail the operation of current model year vehicles and to model advanced vehicle technology application in future vehicle fleets. Using this approach allows EPA engineers to configure characterizations of future technologies, while following manufacturers' operating and integration rules discovered through laboratory testing.

It should be noted, while the estimates of CO_2 reduction potential presented in this paper show a promising potential for vehicles to achieve the light-duty GHG standards for model years 2022 to 2025, the data and analyses in this paper were not used to make any conclusions or decisions regarding the MTE for the light-duty GHG rule. While this paper does include data similar to that used in the MTE to provide context, the purpose of the paper is to share information about EPA's approach to modeling and the progress made towards predicting CO_2 emissions from various technologies for future fleets, not to provide final results associated with the MTE.

The overarching goal of the paper is to explain ALPHA's approach to modeling to predict GHG emissions from future vehicles discussing the fundamental four following steps:

Step 1. **Benchmarking recent model year vehicles, engines, and transmissions:** This paper begins with a brief review of the specific vehicles, engines and transmissions that were benchmarked in EPA's laboratory in support of this work.

Step 2. **Validating ALPHA using data from specific vehicles, engines and transmissions**: To validate the performance of ALPHA, EPA completed many individual vehicle validations using in-depth newly acquired vehicle, engine, and transmission benchmarking data from 25 conventional and hybrid vehicles from MY2013-2016. The paper discusses a variety of factors that influence the simulated fuel economy and GHG emissions. ALPHA has been refined and updated to more accurately model light-duty vehicle behavior, to include new technologies and accurately simulate their losses.

Step 3. **Characterizing a core set of future engine and transmission technology**: The paper describes the core set of engines and transmissions that were used to make predictions of CO_2 emissions from future vehicles.

Step 4. **Using ALPHA to model GHG emissions from future fleets equipped with core future technologies:** This paper illustrates how ALPHA was configured to use the set of core future technologies to predictively model future vehicle fleets. It focuses on the generic calibrations for driver behavior, engine operation, transmission gear selection and torque converter lockup that are representative across a wide range of vehicles, engines and transmissions.

BENCHMARKING RECENT MY VEHICLES, ENGINES, AND TRANSMISSIONS (STEP 1)

In preparation for the MTE, EPA refined and revalidated ALPHA using newly acquired data from MY2013-2016 engines and vehicles. During 2014-2016, EPA has been involved with the testing of over 25 different types of conventional and hybrid vehicles/engines across a wide range of powertrains and segments. An overview of the types of vehicles, engines and transmissions in EPA's testing program was provided in an earlier SAE paper describing powertrain technology package modeling [4], and is summarized below.

- 8 vehicles benchmarked with naturally aspirated (NA) gasoline direct injection (GDI) engines (4 w/Atkinson)
- 7 vehicles benchmarked with gasoline turbocharged engines
- 3 vehicles benchmarked with diesel turbocharged engines
- 4 vehicles benchmarked with port fuel injection (PFI) engines
- 4 engine dyno benchmarks with NA engines
- 5 engine dyno benchmarks with turbocharged engines
- 5 transmission benchmarks with six-speed automatic (6AT) and eight-speed automatic (8AT) transmissions

The vehicles/engines were chosen based on EPA's need to evaluate key technologies in cars, SUVs and pickup trucks. Each vehicle was extensively instrumented to better characterize powertrain inefficiencies, and ultimately yield better predictions of technology effectiveness. To explain its vehicle benchmarking process EPA authored a paper describing the test process and benchmarking data of a 2013 GM Malibu [4]. Since this paper was published, EPA has continued benchmarking more vehicles and refining its test methods.

In addition to vehicle level benchmarking EPA also conducted component-level benchmarking of several engines in EPA engine dynamometer test cells. This testing made it possible to generate full fuel consumption maps for use in ALPHA. Several papers have been authored, describing test process and data from several engines [6, 7, 8].

As part of its transmission benchmarking program, EPA has tested five transmissions to specifically obtain torque loss data for use in ALPHA. EPA has authored several papers describing its transmission benchmarking and subsequent analysis [9, 10, 11].

VALIDATING ALPHA & COMPONENT INPUTS (STEP 2)

To validate the ALPHA model a subset of the benchmarked vehicles from step 1 were selected based on the advanced technologies employed and the completeness and robustness of test data. The vehicles are presented in Table 1 along with references regarding their benchmarking and/or validation.

Table 1. Vehicles and data sources for ALPHA validation activities

		Conventional Vehicle	Engine	Trans	Ref
Turbocharged Engines	Car	2015 Volvo S60 T5	2.0*l* I4 Turbo	8AT	
		2013 Mercedes E350	3.0*l* V6 ETEC Diesel	7AT	[12]
	Truck SUV	2013 Ford Escape	1.6*l* I4 EcoBoost	6AT	
		2015 Ford F-150	2.7*l* V6 EcoBoost	6AT	[14]
Naturally Aspirated Engines	Car	2013 Chevrolet Malibu	2.5*l* I4 Ecotec GDI	6AT GM 6T40	[4]
		2013 Chevrolet Malibu Eco	2.4*l* I4 Ecotec GDI	6AT GM 6T40	
		2013 Altima SV	2.5*l* I4	Jatco CVT8	[13]
		2014 Mazda 3 (US)	2.0*l* I4 SKYACTIV	6AT	[7]
		2014 Dodge Charger 5-spd	3.6*l* V6 Pentastar	5AT w5A580	[9]
		2014 Dodge Charger 8-spd	3.6*l* V6 Pentastar	8AT 845RE	[9]
	Truck SUV	2014 RAM 1500 HFE	3.6*l* V6 Pentastar	8AT 845RE	
		2014 Chevy Silverado 1500 2WD	4.3*l* Ecotec V6	6AT 6L80	[8,11]

The goal when validating individual vehicles within ALPHA is to verify that the model accurately represents the test data at both the vehicle and component levels. To this end, the validation compares not just overall UDDS and HWFET fuel consumption, but time series data for fuel, speed, and torque measurements throughout the powertrain.

The initial step in ALPHA validation involves integrating the available benchmark data for the appropriate parameters. Two types of inputs are necessary for ALPHA simulation. The first input data type consists of direct measurements of physical quantities. Examples of physical quantities include the engine's steady-state fuel map, transmission gear ratios, and transmission gear loss information. For most of these quantities a component-level benchmarking is preferred to gather the most detailed and highest quality data.

The second type of input needed for the validation is vehicle-level data that defines vehicle behavior. This input data is generally related to how the manufacturer chose to calibrate the electronic controls within the vehicle. An example of this type of input would be the transmission gear selection strategy. These inputs generally require calibration to create similar behavior. The specific calibration uses test data metrics to characterize the similarity of the behavior along with sufficient test data to cover the range of operating conditions that might be simulated.

All vehicle behavioral inputs, including transmission gear & torque converter lockup strategies, are used to ensure the simulation matches test data as closely as possible. Beginning at the wheels and working up to the engine, the available benchmarking test data is compared with speed and torque values from the simulation. Discrepancies are identified, examined further, and corrected. After the speeds, torques and fuel rates are corroborated, the behavioral inputs can be replaced with appropriate algorithms, such as ALPHAshift for gear selection, and calibrated to match the observed behavior.

ALPHA Model Subsystem Descriptions

Using the validation results, a number of the subsystem models within ALPHA were refined to better emulate the physics and more easily tune parameters within the model with available benchmarking data. Many ALPHA inputs play a role in determining powertrain efficiency and fuel consumption. The following sections provide a description of the various component models, looking at what data are used as inputs to ALPHA, and which of the ALPHA inputs play a significant role in determining fuel consumption. An overall breakdown of energy flows in the model, showing where the various losses are, can be found at the end of this section.

Dynamic Lookup Tables

One unique feature of ALPHA is the use of dynamic lookup tables. These special lookup tables provide interpolation similar to a normal Simulink 1D or 2D lookup, but allow the dimensionality and signals used for lookup to be determined at run time. This allows for component loss data within the model to be parameterized in a manner that corresponds directly to the available test data. For example, a detailed transmission map may have had its losses characterized by gear number, input speed, input torque, hydraulic line pressure and temperature using a five dimensional lookup, while other testing might have yielded a much simpler two dimensional map utilizing only input torque and speed. ALPHA can accept either map without physically altering the Simulink structure. Dynamic lookup tables are a powerful tool for improving model fidelity when highly detailed data is available, but also allow the model to run with coarse or simplified data when needed.

Comparison of Simulation with Torque Measurements

It is important to note that the torque signals within the model cannot be directly compared with test data from a torque meter. The method ALPHA uses to compute angular acceleration involves passing the torque and inertia measurements down to the integrator for that rotational body. To compare with measured test data, the torque signal must be compensated with the acceleration and inertia of the upstream components.

Engine Subsystem

The engine model is based around a steady-state fuel map covering all engine speed and load conditions. Curves representing maximum and minimum torque versus engine speed constrain the use of the map to points that lie within the engine's operating envelope. To determine the location on the fuel map used at each time step of the simulation, first the engine speed is calculated from the physics of the downstream speeds. Next, the torque demand is calculated from the driver model accelerator demand, an idle speed controller, and requests relating to torque management during transmission shifts.

A simple engine torque response model is implemented to emulate the air-path dynamics and turbocharger behavior. The model consists of two first order transfer functions, the time constants of which were calculated from transient testing on an engine dynamometer with a variety of engines. The first corresponds to boosted operation and limits increases in engine torque production with a time constant of around 0.7 seconds. The second transfer function represents the air path dynamics of throttle and intake manifold with a time constant of approximately 0.2 seconds. The in-cylinder combustion processes are not modeled.

The operating speed and torque are used to interpolate a steady state fuel map. A deeper explanation of the engine model and how some of the fueling adjustments are derived is available in [14]. The following description summarizes the methodologies described in that paper.

One fuel consumption adjustment included in the ALPHA engine subsystem is a power-rate based transient adjustment. Use of a steady-state fuel map for simulation has been shown to underestimate fuel consumption. Actuator dynamics and maintaining stable operation lead to this reduced efficiency. Within ALPHA this adjustment is characterized by the rate of change in engine power as a multiplier on the base steady-state fuel map. During more aggressive driving cycles such as the US06 this adjustment increases fuel consumption 2.0% to 2.5%, while for the less aggressive cycles such as the FTP or HWFET it only adds 0.6% to 1.1%.

A second fuel consumption adjustment is included for transmission upshifts. During shift events, modern vehicles modify engine torque output in coordination with the transmission to provide a smooth shift, but at the cost of increased fuel consumption. During a transmission upshift, the decelerating inertia of the engine via the transmission clutches would yield a brief surge in torque at the wheels accelerating the vehicle. To smooth out this behavior the engine torque is reduced by a similar amount. On a spark ignition (SI) engine this is usually accomplished via retarding spark timing, as it allows the engine to return to normal operation quickly. This change in spark timing causes the engine to operate with lower efficiency. In ALPHA this behavior is simulated by reducing torque to smooth out the gear shift, but using the unadjusted torque for interpolating the fuel map. This results in overall cycle fuel consumption increases of 0.2% to 1.0%, depending upon the vehicle and test cycle.

A third fueling adjustment within the engine model is associated with deceleration fuel cut-off (DFCO). Following longer periods of DFCO additional fuel is required to maintain proper catalyst operation. This additional fueling is characterized from time series chassis test data as a multiplier on the base fuel that decays over a specified time period, usually in the range of a few seconds. Depending upon the vehicle and drive cycle this adjustment adds 0.2% to 0.5% to the cycle fuel consumption.

For engines that utilize cylinder deactivation (CDA) to reduce fuel consumption, separate fuel maps were created with and without cylinder deactivation. Multiple options are available to determine when transitions between the two maps occur. Logic can be constructed to switch based on appropriate speed and load, with limitations for vehicle conditions like transmission gear. Alternatively, recorded test data can be used to command the transitions to match what was observed during chassis dynamometer testing. Another simple option is to estimate the percentage of time CDA was active when operating in an area it is available, then interpolate the two maps. All three methods were developed for ALPHA and yield similar results, but for simplicity, the two-map interpolation method was chosen for EPA fleet modeling.

It should also be noted that the engine subsystem includes an adjustment for CDA transitions. Spark timing is retarded to smooth out the transition while cylinders are deactivated or brought back online which leads to momentarily reduced efficiency and additional fuel consumption. Based upon EPA's testing of the 2014 Chevrolet Silverado about 0.4% additional fuel was consumed as result of these transitions.

Transmission Subsystem

The ALPHA transmission subsystem features different variants representing the major types of transmissions (automatic transmission [AT], manual transmission [MT], continuously variable transmission [CVT], and dual clutch transmission [DCT]) that are currently in use in LD vehicles. The different transmission models are built from similar components, but each features a unique control algorithm to emulate behaviors observed during vehicle benchmarking.

ALPHA features multiple speed integrators, located at each of the points in the driveline where rotational inertias may become decoupled. The torque and inertia from each component pass downstream from the engine through the transmission to the wheels. Whenever a disconnection point, such as the transmission gearbox, becomes decoupled the integrator at that location activates, computing the speed for the upstream components. If a coupling is locked up, the torques and inertias continue to be passed down to the next disconnection point or the vehicle speed integrator. This allows the physics of the system to be accurately simulated, losses associated with clutch slip to be computed, and the energy audit to be properly accounted.

Clutch & Dual Clutch

The ALPHA clutch model can be modulated during launch and provides an appropriate delay during engagement. Torque is conserved across the clutch during engagement, with the speed differential between input and output representing energy loss. The clutch modulation during launch necessitates a control algorithm to manage clutch slip and is calibrated based on observed data. Two clutches are bundled together to create the dual clutch module for the dual clutch transmission.

Gearbox

The gearbox model for ALPHA has been developed with the goal of simulating realistic operation during shifts for all types of transmissions. The gearbox contains gear ratios which properly scale both torque and rotational inertia through the ratio change. Power loss within the gearbox can be applied via dynamic lookup tables for either torque loss or efficiency. These loss tables are generally parameterized via commanded gear, input speed, input torque, and/or line pressure. Data used to fill the loss tables typically come from component benchmarking, examples of which can be found in [9, 11].

The gearbox rotational inertias are split between a common input inertia, common output inertia and a gear specific inertia. The common inertias represent rotational inertia always coupled to the input or output shafts. The gear specific inertias, used for planetary automatic transmissions, are added or removed as gears are engaged or disengaged. There is an additional load placed on the powertrain associated with spinning up each gear specific inertia, and when each gear is disengaged the kinetic energy contained within the gear specific inertia is discarded and treated as a loss.

Hydrodynamic Torque Converter

The torque converter model in ALPHA simulates a lockup-type torque converter. The torque multiplication and resulting engine load are calculated via torque ratio and K-factor curves that vary as a function of speed ratio across the torque converter. The lockup behavior of the torque converter is accomplished by integrating a clutch model similar to the one discussed earlier in this section. While controlled torque converter clutch slip can be simulated, it is simpler to model this behavior as lockup with slightly reduced efficiency.

Proper torque converter selection plays a role in determining fuel economy as it determines the engine speed and load during launch. When available for validation activities, component benchmarking data of torque converter torque ratio and K-factor curves versus slip ratio are used. Otherwise measurements at stall are used to scale generic curves. It has been observed that torque converters are more similar than different and minor differences in K-factor curves have little effect on fuel consumption as long as the stall K-factor is reasonable for a given application.

The torque converter model also contains a pump loss torque that is implemented via a dynamic lookup table to simulate the power required to operate the pump on an AT or CVT. When possible, this loss is measured separately during the component benchmarking process, and is generally represented as a function of torque converter input speed and transmission line pressure.

Transmission Gear Selection & Torque Converter Lockup

Transmission gear selection and torque converter lockup strategies represent inputs that must be calibrated to match observed data. ALPHA includes a variety of forms in which the calibrations can be represented. Observed data can be applied directly, shifting when it was observed in the test vehicle. Classic table based shift algorithms are available as well as options like the EPA developed ALPHAshift algorithm.

The ALPHAshift algorithm employs a rule based approach utilizing the engine torque curve and fuel map to select the minimum fuel consumption gear, but still provide torque and power reserves as a traditional transmission calibration would. The algorithm also allows downshifts due to high driver demand. A detailed description of the basic shifting strategy can be seen in [15] although development has continued since publication. ALPHAshift calibration parameters can be quickly tuned to create a shifting strategy that attempts to optimize efficiency and emulate benchmark data for a particular engine and transmission combination. This calibration can then be applied to simulations of different drive cycles or altered vehicle characteristics

The CVT transmission model uses a similar ALPHAshift-CVT algorithm for determining gear ratio selection. The algorithm identifies the operational points that minimize fuel consumed for any requested power, and then attempts to maintain operation on those points. This method also has constraints for minimum engine speed and the rate at which the gear ratio can be changed.

Driveline Subsystem

The driveline subsystem encompasses all the components that connect the transmission to apply force at the wheels. ALPHA has the capability to simulate multiple axle configurations, consisting of both driven and passive axles. The simulation of multiple axles does not add fidelity to the results as concepts like weight transfer are not simulated. As a result, most simulations are conducted with a mathematically equivalent single axle to improve simulation efficiency.

Axle Losses

The axle model within ALPHA has loss maps that can be configured as component efficiency or as a direct torque loss via dynamic lookup tables. For front wheel drive vehicles, these axle/final drive losses are generally contained within the transmission, as the transaxle is tested as a single unit with the transmission itself. Rear wheel drive vehicles are simulated with separate axle losses, often a fixed mechanical efficiency.

The axle loss maps within ALPHA could be utilized to simulate additional spin and churning losses within the axle. However, for light-duty vehicles ALPHA usually simulates roadload forces via the coastdown target ABCs. The axle spin losses are present within the ABCs as they represent one of the forces decelerating the vehicle during the coastdown test. Simulating them within the axle model would double count the losses. The ALPHA model does offer the ability to simulate roadload via aerodynamic drag and tire rolling resistance instead of ABCs. In this context including measured axle spin losses would be prudent.

Brakes

The brake model applies a torque directly proportional to the brake pedal position signal from the driver model. The maximum braking torque available is scaled with vehicle mass and tire radius to yield an equivalent maximum deceleration rate. The model also includes logic to coordinate braking when used in a hybrid to maximize the energy recovered during regenerative braking events.

Tires

The tire model handles the transfer of torque from the axles to force at the interface of the tire and driving surface. Two losses are present within the tire model. A rolling resistance force applied using the tire's rolling resistance coefficient and a portion of the vehicle mass carried by the axle. The rolling resistance is omitted when simulating roadload via coastdown ABC coefficients.

The ALPHA tire model also accounts for losses associated with slip that occur between the tire and the surface on which it is traveling. The relationship between tractive force and tire slip was derived from literature [16] and has been show to correlate well with test data. An example of this relationship is plotted in Figure 1. For fuel economy simulation within ALPHA the wheel forces and tire slip are relatively low, allowing the tire slip to be computed from wheel torque by interpolating the red portion of the curve, which corresponds wheel slip values of -14% to 14%. Force at the wheel is normalized by the vehicle mass such that the maximum tractive force corresponds to a 1 g acceleration. The normalization factor is tunable if the simulated tire slip differs from observations during testing.

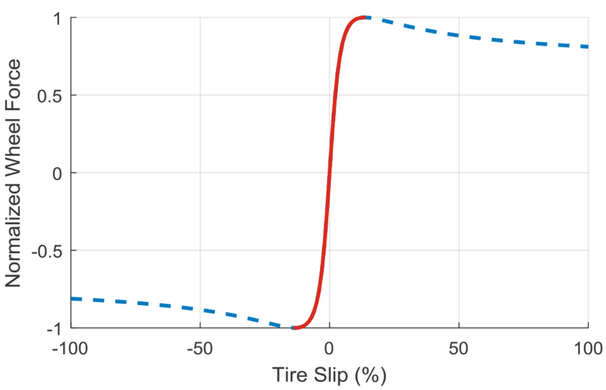

Figure 1. Sample relationship of wheel force versus tire slip, ALPHA uses the solid red portion of the line

Vehicle Subsystem

The vehicle subsystem represents the chassis of the vehicle being simulated. Roadload losses may be represented either by aerodynamic drag coefficient, frontal area and rolling resistance in the tire model, or as it is commonly generalized for light duty vehicles via coastdown target ABCs. For emission testing, the roadload force for a vehicle is expressed by a formula being equal to $A + Bv + Cv^2$, where v is the vehicle speed and the A, B and C coefficients are derived from an actual vehicle coastdown process and are used for setting up a chassis dynamometer to emulate the vehicle's on-road operation. ALPHA includes the ability to simulate road grade as well, however this feature is not used for this study. After applying these forces, the net force at the wheels can be used to compute vehicle speed and distance traveled.

When using transmission component testing loss data within ALPHA, an important consideration is that a portion of the transmission spin losses measured in component testing also contribute to the vehicle coastdown target ABC coefficients. If raw transmission test data and the target ABC coefficients are both used in vehicle simulation, this portion of the drivetrain losses will be double-counted.

The losses in question can be characterized on the test stand by measuring the spin loss at the transmission output while the transmission is in neutral, as it would be during the coastdown. Using these torque measurements, a corresponding force at the tires versus vehicle speed can be obtained. This relationship can then be used to adjust to original coastdown ABC coefficients for use within ALPHA simulations.

Table 2 shows examples of the original ABC coefficients for a select few vehicles that have been simulated within ALPHA, and their ABC values after being adjusted for use in ALPHA simulations based on transmission spin loss testing. Correcting the ABC coefficients for use in ALPHA is preferred to altering the transmission losses as it enables actual transmission output shaft torque measurements to be compared with the simulation data.

Table 2. Examples of original and neutral-drag adjusted ABC roadload coefficients

Vehicle Transmission		A lbs	B lbs/mph	C lbs/mph^2
2012 Chevrolet Malibu 6T40	Orig	28.62	0.1872	0.01828
	Adj	28.058	0.11183	0.01828
2014 Chevrolet Silverado 6L80	Orig	31.030	0.6234	0.03087
	Adj	30.288	0.26161	0.03397
2014 Dodge Charger 845RE	Orig	41.87	0.0217	0.01993
	Adj	33.527	0.1308	0.0190

The shift in the A and B coefficients for the 845RE transmission appears to be quite large. This change appears to be related to different neutral transmission clutch arrangements observed as vehicle speed changed during the coastdown.

Electrical & Accessories Subsystems

To properly account for losses associated with electrical accessory loads and the results of technologies like start-stop, ALPHA includes the ability to simulate the complete electrical system including battery, starter motor, alternator and accessory electrical loads.

Battery Model

The battery model for ALPHA was created after extensive literature review of battery models, particularly for hybrid vehicle applications. The same battery model structure is used for both conventional and hybrid vehicles, with different calibrations used to simulate different battery chemistries. Calibrations were generated from published literature or benchmark testing for both open circuit voltage and transient behavior. The simulated battery also features a thermal model, with the output current limited at extremes in temperature or state of charge. [17, 18, 19, 20]

Starter

The engine starter is a simplified electric motor with fixed efficiency and is commanded via a Boolean activation signal. The operation of the starter is characterized by a desired cranking speed and a maximum available torque, which is scaled to match the engine specifications. Cranking speed is maintained by a proportional integral controller with the output limited by the torque capacity. The mechanical power required and efficiency then determine the electrical current consumed. During starting, the transmission is placed in neutral to prevent excessive load on the electrical system as would occur in an AT if the transmission were in gear and the torque converter were stalled while trying to crank the engine. The starter is disabled once the engine's idle speed controller has started fueling the engine after a predetermined crank time.

Alternator

The engine alternator utilizes a fixed efficiency. The electrical output current is determined by a charging controller. The efficiency and electrical power output can then be used to compute the mechanical load applied to the engine. The charging controller can operate in two different modes. In a basic mode it charges the battery to a fixed voltage target. In alternator regen mode, alternator output varies the target voltage and thus the load applied to match driving conditions. Lower electrical output is provided during cruising, enough to maintain a minimal state of charge. During decelerations electrical output and thus mechanical load are increased to capture energy that would otherwise be dissipated via the brakes. ALPHA's default calibration provides a response similar to what has been observed on vehicles from a variety of manufacturers. Further calibration of the alternator regen strategy to match observed alternator output currents is possible, but requires use of more detailed accessory loss models.

Accessory Loads

ALPHA has placeholders for 4 different accessory load types: engine cooling fans, air-conditioning, power steering and a generic loss to cover additional loads. Each accessory load can apply mechanical loads directly to the engine and/or to the electrical system. The dynamic lookup blocks within the model allow each load to be characterized to relevant vehicle parameters such as engine or vehicle speed. Additionally, time series test data can be fed directly into tables.

Model Energy Auditing

One of the quality control components within the ALPHA model is an auditing report of all the energy flows. This auditing enables verification that the physics represented in the model is done correctly, generally resulting in a simulation energy error less than a few hundredths of a percent. An example of an ALPHA energy audit report for a current production sedan is shown in the Figure 2 below. This type of report is available from ALPHA when needed as a quality check for individual simulation runs.

The audit reports can be compared between simulations to verify that individual component losses are reasonable when compared to baseline packages or products that may feature similar technologies. As a quality check, this report is helpful at identifying and highlighting any erroneous input data. The percentage numbers represent the approximate amount of engine output energy consumed by that particular line item as a percentage of the energy provided by the engine.

Examination of this sample report illustrates the lack of final drive losses is attributable to the vehicle having front wheel drive where the final drive losses are included within the transmission gearbox (for rear wheel drive [RWD] vehicles the final drive losses are separately associated with the rear differential). The report also shows that a single generic electrical load was used during this specific validation versus the alternative of characterizing each of the accessory loads on the vehicle (fan, air conditioning, and power steering). The simulation error line in the report is calculated by subtracting each of the losses the model from the total input energy. Simulation errors are generally the result of discontinuities in the calculation of the component angular velocities that occur when elements of the powertrain are disconnected and recoupled such as transmission shifting.

```
         ---- Energy Audit Report ----
Total Energy Consumed              = 150146.30 kJ
    Fuel Energy                    = 150146.30 kJ
    Stored Energy                  =      0.00 kJ
         Battery Internal Losses   =    308.29 kJ        0.21%
    Kinetic Energy                 =      0.00 kJ
    Potential Energy               =      0.00 kJ

Usable System Energy Provided      = 47750.53 kJ
    Engine Energy                  = 47750.53 kJ
         Engine Efficiency         =    31.80 %
    Stored Energy                  =     0.00 kJ
    Kinetic Energy                 =     0.00 kJ
    Potential Energy               =     0.00 kJ

Energy Consumed by ABC roadload    = 31858.90 kJ      66.71%
Energy Consumed by gradient        =    -0.00 kJ      -0.00%
Energy Consumed by brakes          =  9806.78 kJ      20.53%
Energy Consumed by Accessories     =  2013.42 kJ       4.22%
    Starter                        =     0.26 kJ       0.00%
    Alternator                     =   604.10 kJ       1.26%
    Battery Stored Charge          =   313.57 kJ       0.66%
    Generic Loss                   =  1095.49 kJ       2.29%
         Electrical                =  1095.49 kJ       2.29%
         Mechanical                =    -0.00 kJ      -0.00%
    Total Electrical Accessories   =  1095.49 kJ       2.29%
    Total Mechanical Accessories   =    -0.00 kJ      -0.00%
Energy Consumed by Driveline       =  4077.57 kJ       8.54%
     Launch Device                 =   140.35 kJ       0.29%
     Gearbox                       =  3256.98 kJ       6.82%
         Pump Loss                 =   630.10 kJ       1.32%
         Spin Loss                 =   795.23 kJ       1.67%
         Gear/Inertia Loss         =  1831.65 kJ       3.84%
     Final Drive                   =     0.00 kJ       0.00%
     Tire Slip                     =   680.24 kJ       1.42%
Net System Kinetic Energy Change   =     0.53 kJ       0.00%
                                     ------------
Total Loss Energy                  = 47757.20 kJ
Simulation Error                   =    -6.67 kJ
Energy Conservation                =  100.014 %
```

Figure 2. A sample of the ALPHA energy audit report from a validation analysis to illustrate the report format.

CHARACTERIZE A CORE SET OF TECHNOLOGIES (STEP 3)

To perform a prediction of GHG emissions from future fleet of vehicles, ALPHA needs a defined core set of technologies to use in the matrix modeling. For this study four major aspects are considered, engines, transmissions, electrical systems and accessory loads, and improvements in vehicle roadload. The first three require sample data representing the technologies.

Engines

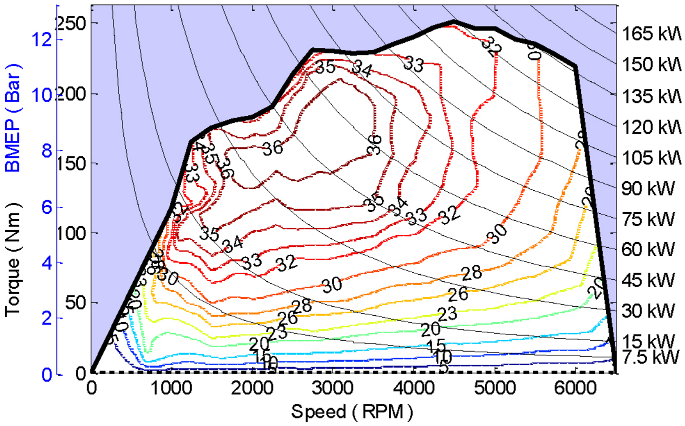

Figure 3. Brake thermal efficiency map for the 2013 GM Ecotec 2.5*l* engine from the Chevy Malibu (AKI 93 fuel)

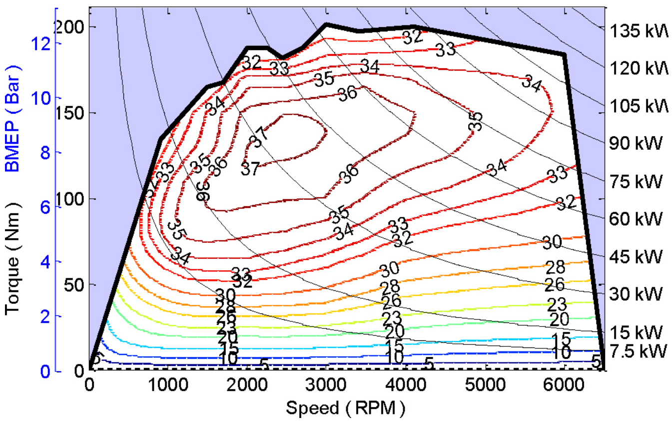

Figure 4. Brake thermal efficiency map for a 2014 Mazda 2.0*l* SKYACTIV 13:1 compression ratio engine (AKI 93 fuel)

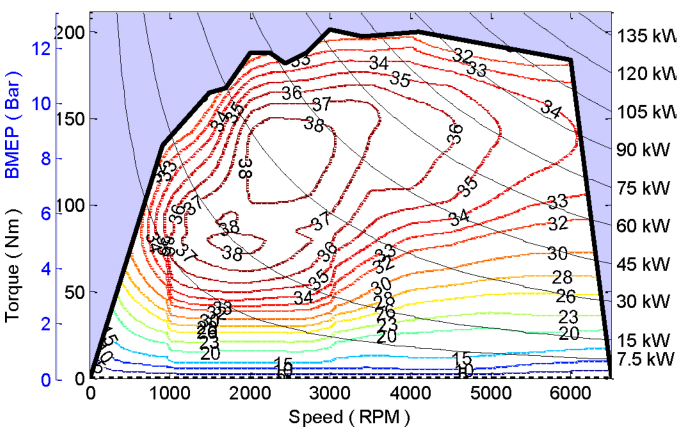

Figure 5. Brake thermal efficiency map for the future Atkinson naturally aspirated engine (AKI 93 fuel)

The four engines that were utilized during the prediction described in this paper are listed below with their associated thermal efficiency maps. The first two are production engines that were benchmarked in EPA's engine program. The final two represent possible future engines that EPA believes may be of the type and efficiency used in vehicles by the 2025 timeframe. These engine maps were generated assuming that the fuel utilized Tier 2 certification fuel (anti-knock index [AKI] 93).

- **2013 GM Ecotec 2.5*l*** naturally aspirated I4 GDI engine from the Chevrolet Malibu. [4]
- **2014 Mazda SKYACTIV 2.0*l*** naturally aspirated I4 engine with 13:1 compression ratio from the Mazda 3. [7]
- **Future Atkinson** naturally aspirated I4 engine with 14:1 compression ratio, cooled EGR and cylinder deactivation (CDA) developed via GT-POWER simulation. [17] The range of operating speeds and loads where CDA is employed, along with the frequency with which CDA was activated, was derived from benchmarking data on the GM Ecotec 4.3l V6 engine in the Chevrolet Silverado.
- **Future 24-bar turbo-downsized (TDS)** I4 engine with cooled EGR, derived from the 2010 Ricardo analysis for LD GHG Federal Rulemaking [3]. The engine map in Figure 6 was created by cropping its peak torque line to 24-bar maximum brake mean effective pressure (BMEP) versus the original Ricardo 27-bar.

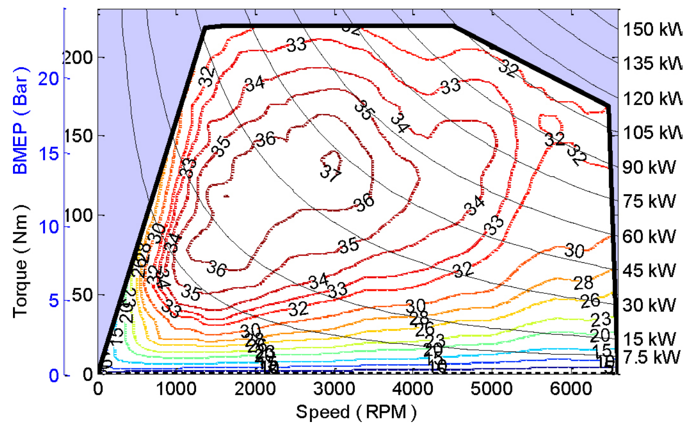

Figure 6. The thermal efficiency map for a future advanced 24-bar turbo-downsized engine (AKI 93 fuel)

Transmissions & Final Drive

While ALPHA is capable of simulating a wide variety of transmission types (AT, MT, CVT, and DCT) the examples in this paper will focus on the traditional torque converter automatic transmission. The automatic with torque converter lockup remains the most popular configuration in the light duty fleet. The three transmissions that were utilized during the fleet-level modeling described in this paper are listed below. The first two transmissions are real technologies that were benchmarked in EPA's transmission program. The final transmission is derived using supplier information to represent a future transmission that EPA believes may be used in vehicles by the 2025 timeframe.

- **2013 6AT** - a set of 6-speed transmissions (front wheel drive [FWD] and rear wheel drive [RWD]) based on data from benchmarking the GM6T40 front wheel drive in a 2013 Chevrolet Malibu [4]
- **2014 8AT** - a set of 8-speed transmissions (FWD and RWD) based on data from benchmarking the FCA 845RE rear wheel drive 8AT in the 2015 Dodge Charger [9]
- **Future 8AT**- a set of 8-speed transmissions (FWD and RWD) based on an update of the 2014 AT8 transmission data per information from ZF's SAE paper about futuring AT8 transmissions [9, 22].

For its fleet-level modeling analysis feature, ALPHA uses both the FWD and RWD versions of each of these three types of transmissions. It is important to note that ALPHA accounts for the torque loss associated with the vehicle's differential slightly differently with FWD and RWD transmissions. The differential gear mesh efficiency, which is an integral part of a FWD transmission, is assumed to be 98 percent. Consequently, when the RWD based transmission, which has a separate differential, is applied to a FWD vehicle an additional 2 percent torque loss must be applied to the transmission. Since the differential losses are included as part of the transmission for FWD vehicles no additional axles losses are simulated. For RWD vehicles, the differential is treated as a separate component, with a gear mesh efficiency of 96.2 percent. Therefore, when applying a FWD based transmission to a RWD vehicle the assumed 2 percent (FWD) differential loss is removed.

It should be noted that additional spin losses exist within the axle beyond the gear mesh efficiency for both FWD and RWD vehicles related to items such as bearing drag and churning of the lubricant. These losses are naturally included in the roadload coastdown measurements and thus do not need to be simulated separately.

Electrical System & Accessory Loading

To simplify simulation for this study, the accessory load is represented by a constant power electrical load. Two sets of inputs are considered, representing present and future scenarios.

For present vintage electrical systems, the load is 390W, representing the measurements captured in vehicle benchmarking shown in Table 3. This scenario also uses an alternator efficiency of 65%.

Simulations of future vehicles use 290W of electrical load with an alternator efficiency of 75%. Future powertrains include an implementation of alternator regen where the alternator output voltage is boosted during braking events as well as upshift and torque converter lockup events.

Table 3. Representative alternator loads for recently benchmarked vehicles

Vehicle	Average Alternator Output (W)		
	UDDS	HWFET	US06
2014 Chevrolet Silverado	538	441	442
2015 Ford F150	338	288	313
2014 Dodge Ram 1500 HFE	439	403	441
2016 Honda Civic	224	260	263
2013 Chevrolet Malibu	343	248	266
2014 Mazda 3	220	205	234

DEVELOPMENT OF THE FLEET-LEVEL MODEL (STEP 4)

Conversion of the validated vehicle and component data into estimates of GHG emissions for future vehicles consists of first establishing the platforms to which the technology will be applied. Next, rules for converting examples of particular technologies are developed and applied. Finally, composite vehicles can be constructed and simulated.

Definition of Exemplar Vehicles

The determination of the most appropriate values for technology effectiveness depends on the characteristics of the particular vehicle to which the technologies are applied. Variations in engine-power-to-vehicle-weight-ratio and vehicle-roadload-power provide a useful way to group vehicles when assessing technology effectiveness.

To determine the characteristics of each vehicle grouping, vehicles in the 2015 U.S. fleet were sorted into six categories based on their engine-power-to-vehicle-weight-ratio and vehicle-roadload-power characteristics. Within ALPHA these category groups are known as the vehicle classes. For each vehicle class the sales weighted average of the vehicles in that class becomes the weight of the "average" vehicle for that class. This average vehicle is known the exemplar vehicle. Table 4 contains the definition of each vehicle class and also shows the key characteristics of each exemplar vehicle for each class (engine power, equivalent test weight [ETW] and roadload coefficients). Figure 7 shows the distribution of engine-power-to-vehicle-weight-ratios" for each vehicle class based on the vehicles in the 2015 U.S. fleet.

Table 4. Vehicle classes and exemplar vehicle specifications

Vehicle Class	Vehicles within Vehicle Class	Specifications of Specific **Exemplar Vehicles**				
		Engine Power (hp)	Vehicle ETW (lbs)	Roadload Coefficients		
				A (lbf)	B (lbf/mph)	C (lbf/mph²)
LPW_LRL Low Power/Weight & Low Roadload	Fiesta Focus Yaris	137	3257	26.56	0.0630	0.01879
MPW_LRL Medium Power / Weight & Low Roadload	Fusion Taurus Camry	190	3626	32.27	0.0754	0.01993
HPW High Power / Weight	Chrysler 300 Mustang	314	4401	35.76	0.3414	0.02086
LPW_HRL Low Power / Weight &High Roadload	Escape Rav4 Tacoma	172	3855	34.95	0.0875	0.02526
MPW_HRL Medium Power / Weight &High Roadload	Explorer 4Runner Dodge Caravan	275	4849	39.30	0.3348	0.02721
TRUCK	F-150 Tundra	324	5303	39.62	0.4641	0.03222

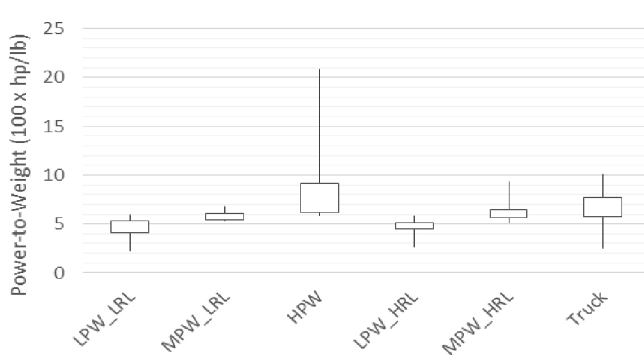

Figure 7. Distribution of engine-power-to-vehicle-weight-ratios for each of the six vehicle classes in MY2015, box area represents +/- 1 standard deviation from mean, stems indicate range of values

When calculating vehicle effectiveness for a fleet-level analysis simulation, ALPHA uses the definitions of the appropriate exemplar vehicle as the specifications for the baseline vehicle in the comparison.

Adaptation & Generalization of Test Data

In order to simulate the various technologies within the different vehicle groups, standard methods of converting the various test data needed to be developed. Engines and transmissions must be scaled to match vehicle requirements, a process that is not generally linear.

Engine Scaling

The engines in the exemplar vehicles of different classes have different peak powers. In addition, as powertrain technology packages or vehicle roadloads are changed, the corresponding peak engine power required for similar performance also changes. To reflect the range of engine powers required in different modelling scenarios, ALPHA needs the capability to produce fuel consumption maps corresponding to a range of engine displacements. However, simply multiplying the torque and fuel map parameters of the engine by a scale factor lacks robustness, and overlooks details of and constraints on combustion.

The overall engine scaling process consists of a progression that proportionally resizes the engine and then makes three separate adjustments to bring the BSFC values in all areas of the map more in line with actual engines; a heat transfer adjustment, a friction adjustment and a knock sensitivity adjustment.

Proportional Engine Resizing

There are two types of complementary steps to proportionally resizing an engine's fuel consumption map to obtain the required engine power: 1) change the engine architecture, if necessary, and 2) resize the cylinder displacement. Changing the engine architecture involves adding or removing cylinders to match the performance requirements on a coarse level. Altering the engine's architecture yields changes to the friction of the engine and resulting fuel consumption which will be discussed further later in this section. After the engine architecture is selected the engine's cylinder displacement may need to be resized. During this step the cylinder displacement is proportionally adjusted while maintaining the bore to stroke ratio. The engine scaling process is now ready to proceed with the first of the three BSFC adjustments.

Heat Transfer Adjustment

Increases in cylinder volume decrease the surface area to volume ratio (S/V) and as a result less combustion energy is dissipated into the engine head and block resulting in higher efficiency. Novak & Blumberg [23], as quoted by Heywood [24] examined this phenomenon in computer simulation showing a 13% reduction in S/V yielded a 3.4% reduction in BSFC. Using the limited data presented by Novak & Blumberg, along with the knowledge that the adjustment should be reversible, EPA constructed a base modifier curve, which agreed very well with the adjustment used in the 2011 LD vehicle simulation study [1].

Two issues were noted with the Novak & Blumberg study. One was that only a single operating point was examined (1250 RPM & 3.7 Bar IMEP). Variation in the heat transfer at different speed and load points were not examined. On the basis that heat transfer is not an instantaneous process, and a longer engine cycle duration would allow more heat transfer to occur, a factor scaling the adjustment with cycle period was added to the Novak & Blumberg data, resulting in a fuel map modifier of:

$$fadj_{HL} = 1 + \left(\left(\frac{V_{new}}{V_{orig}} \right)^{-1/11.5} - 1 \right) \cdot \frac{1250}{\omega} \tag{1}$$

Where,

V_{new} = desired displacement per cylinder

rV_{orig} = original displacement per cylinder

ω = engine speed in RPM

A second criticism of the Novak & Blumberg study is that it is a rather old computer simulation, and while the physics have not changed, the precision of the simulation may benefit from newer, more detailed software now available. As part of its ongoing work, EPA plans to address both of these concerns by conducting GT-POWER simulations to further examine scaling effects on engine efficiency at a variety of speed and load points.

Friction Adjustment

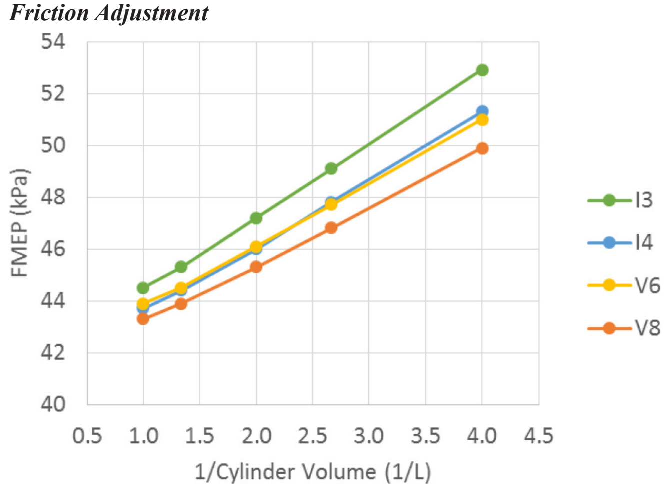

Figure 8. Estimates of FMEP for various engine architectures and displacements

Changes in engine architecture lead to changes in engine friction as a result of changes to the number and size of bearings, or additional cam shafts when moving from an inline to a V engine configuration. The scaling of cylinder volume also alters friction as piston ring contact area is a major source of engine friction. Two useful studies [25, 26] examine the many sources of friction within current engines and provide a template for estimating friction based on various design parameters. Using the methodology presented in [26] and inputs representative of current production engines, the friction estimates were computed for various engine displacements and architectures, the results of which are shown in Figure 8.

For each engine architecture the change in FMEP is quite linear with the reciprocal of cylinder volume. This yields the following engine friction estimates in kPa as a function of individual cylinder volume:

$$FMEP = \begin{cases} 41.60 + {}^{2.82}/_{V_{cyl}}, & I3 \\ 41.02 + {}^{2.55}/_{V_{cyl}}, & I4 \\ 41.38 + {}^{2.34}/_{V_{cyl}}, & V6 \\ 40.97 + {}^{2.21}/_{V_{cyl}}, & V8 \end{cases} \tag{2}$$

Where,

V_{cyl} = cylinder displacement in liters

Using equation 2, the estimates of FMEP for the original engine and the FMEP after scaling can be computed. The difference between these two values is then converted to a torque for application to the following parameters:

- Maximum (Wide Open Throttle) Torque Curve
- Minimum (Closed Throttle) Torque Curve
- Naturally Aspirated (Fast Torque Response) Torque Curve
- Fuel Map Torque Axis Breakpoints

Knock Sensitivity Adjustment

The previous adjustment factors are bi-directional, in that they are applied as engine cylinder size is either increased or decreased. However, when cylinder size is increased, there is also an increased tendency for knock to occur in the region of high load and low speed as highlighted in Figure 9.

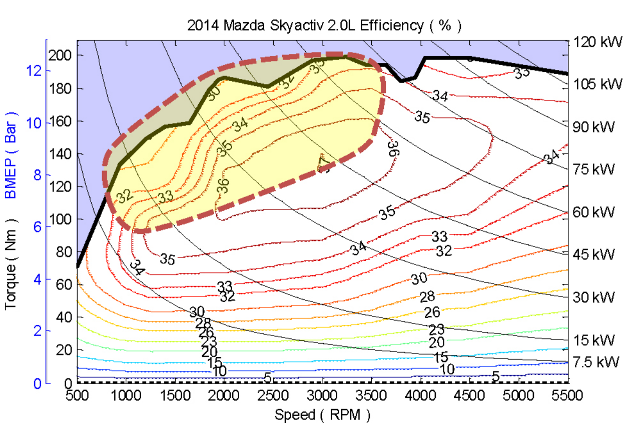

Figure 9. The region of high load and low speed operation that is sensitive to knock when engine size is increased.

The exact conditions leading to the onset of knock is the subject of much study, and is well beyond the scope of this paper. Although the region of engine operation affected by potential knock tends to be somewhat above the area of operation over the Federal Test Procedure (FTP) and highway fuel economy test (HWFET) cycles, there is some operation within this region. Ignoring the potential effect of knock when upscaling cylinder size may underestimate fuel consumption to some extent, thus the inclusion of an adjustment which accounts for fuel consumed to mitigate knock is preferable.

This section of the paper describes the current approach to account for the increase in fuel use associated with greater knock-tendency as engine size increases, and is not an attempt to build a knock model or otherwise simulate the physical reactions that occur. Douaud and Eyzat characterized the knock autoignition delay using the following equation [27]:

$$\tau = A\, p^{-n} e^{B/T} \tag{3}$$

Where,

p = cylinder pressure

T = cylinder temperature

A, B, n = calibration constants

Note that τ is not directly influenced by the cylinder dimensions. Rather, the tendency of the engine to knock is a result of an increase in cylinder temperatures and pressures by altering S/V and heat transfer during compression. By scaling up an engine, less heat is dissipated during compression resulting in higher temperature and pressure and earlier autoignition leading to increased knock tendency.

The ALPHA engine model does not simulate in-cylinder combustion, so the autoignition delay cannot be directly applied. The core of the engine model is a steady-state fuel map. Adjustments for knock sensitivity must be related to the increased fuel consumption to counteract knock via the retarding of spark timing. To quantify the additional fueling, the relationship between cylinder intake mass and surface area that influence the pressure and temperature in equation 3 must be examined. For a given scale factor, each point in the fuel map after scaling can be connected to a point in the original map with a similar mass to surface area ratio.

$$\frac{m_i \cdot x}{SA \cdot x^{2/3}} = \frac{m_i \cdot x^{1/3}}{SA} \tag{4}$$

Where,

m_i = intake mass of air and fuel

SA = surface area

x = engine cylinder volume scale factor

The intake air and fuel mass is roughly proportional to indicated torque under stoichiometric operation and MBT timing. Thus, the knock sensitivity of a load $\tau' = \tau \cdot x$ on the scaled map is similar to the knock sensitivity at $\tau \cdot x^{1/3}$ in the original fuel map. The knock adjustment for a point is the fueling at the point of similar knock sensitivity relative to the fueling had the fuel scaled linearly with torque as seen in equation 5.

$$fratio_{KNK} = \frac{f(\omega, \tau \cdot x^{1/3})}{f(\omega, \tau) \cdot x^{1/3}} \tag{5}$$

Where,

$f(\ldots)$ = engine fuel map

ω = engine speed

τ = engine indicated torque

x = engine cylinder volume scale factor

The engine fuel maps within ALPHA are represented using engine speed and brake torque, not indicated torque as in the above analysis. To approximate the difference between these torque measurements, the closed throttle torque is used to apply the appropriate shift at each speed and compute a knock modifier for each point in the fuel map.

Specific points where $fratio_{KNK}$ is greater than one, are considered to be in knock limited operation. Using only these data points, a regression equation is fitted to the data using equation 6. This step is included to provide a smooth adjustment, whereas the variation in the comparison of two points in the original fuel map could result in undesired variability.

$$\widehat{fratio}_{KNK} = b_1 + b_2 \cdot \tau + b_3 \cdot \omega + b_4 \cdot \tau \cdot \omega \tag{6}$$

Where,

ω = engine speed

τ = engine indicated torque

b_i = regression coefficients

The final knock sensitivity fuel adjustment is computed using the regression coefficients with the output limited to values greater than one as in equation 7. This ensures the adjustment only increases fuel consumption. To be conservative, the knock sensitivity adjustment is only included when engine cylinder volume is increased, decreasing engine efficiency for low speed, high load operation.

$$fadj_{KNK} = \max(1, b_1 + b_2 \cdot \tau + b_3 \cdot \omega + b_4 \cdot \tau \cdot \omega) \tag{7}$$

Where,

ω = engine speed

τ = engine indicated torque

b_i = regression coefficients

The fairly simple function represented by Equation 7 is unlikely to completely capture the complex causes and effects of knock. However, it does result in a reasonable modifier for the area of the engine maps affected by knock and closely approximates the on-cycle operating regime of the engine.

Sample Engine Scaling

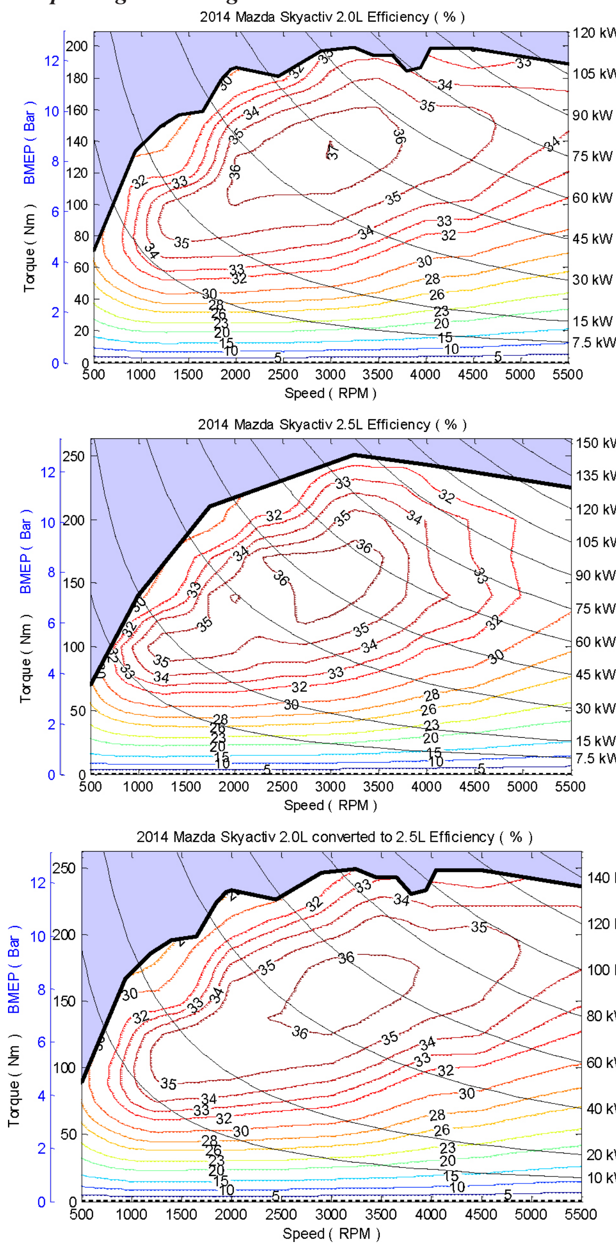

Figure 10. Engine efficiency for the actual 2.0*l* (top) and the 2.5*l* (middle) Mazda SKYACTIV engines, compared with sample scaling of the actual 2.0*l* (top) engine up to a 2.5*l* engine (bottom)

To check the robustness of the overall engine scaling process, a sample of this scaling process is presented in Figure 10. For this example of engine scaling, all of the fuel consumption maps shown are based on using Tier 3 certification fuel. The two Mazda SKYACTIV engines shown in the example are available in 4 cylinder configurations at 2.0*l* and 2.5*l* displacements. The engines feature very similar technology packages. The top two efficiency maps shown in Figure 10 are for the actual 2.0*l* and 2.5*l* engines, respectively. The bottom map shows the result of scaling the actual 2.0*l* engine map up to a 2.5*l* engine map. To arrive at the scaled engine map, the base 2.0*l* engine map was resized and the adjusted for heat transfer, friction and knock sensitivity. When comparing the scaled 2.5*l* engine map to the actual 2.5*l* engine map, the peak efficiency agrees well in both magnitude, and location. At lower loads the scaled map shows efficiency improvements relative to the actual 2.0*l* engine, and similar to the actual 2.5*l* engine map.

Combined Effect of the BSFC Adjustments

Figure 11 below illustrates the combined impact of the three BSFC adjustments (heat transfer, friction and knock sensitivity) applied when scaling the Mazda SKYACTIV 2.0*l* engine map (top map in Figure 11) to a scaled 2.5*l* engine (bottom map in Figure 11). The chart displays the percentage difference in BSFC resulting from the various adjustments for this particular example. Positive values indicate the adjustments result in higher fuel consumption.

After applying the three adjustments to the proportionally resized engine's fuel consumption map, the light load fuel consumption is reduced by 1-2%, while fuel consumption in the knock constrained region increases by as much as 5%. Note that the adjustment for the knock constrained region is only applied when upscaling engines.

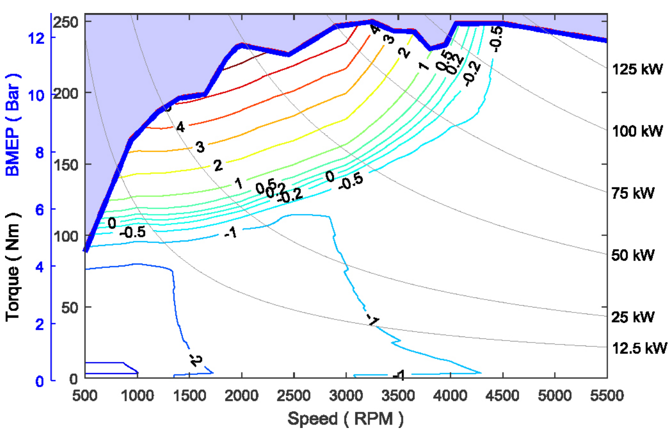

Figure 11. Cumulative effect of the three BSFC adjustments made to a scaled SKYACTIV 2.0l engine upsized to 2.5l

Sample Simulations Using the Actual and Scaled Maps

Table 5. Comparison of simulation of 2.5l Mazda SKYACTIV engine versus 2.0l Mazda SKYACTIV engine scaled up to 2.5l in a midsize sedan

Cycle	Mazda SKYACTIV 2.5lg CO2 / mi	Mazda SKYACTIV 2.0l scaled to 2.5lg CO2 / mi	Difference
UDDS Bag 1	239.4	240.9	0.67%
UDDS Bag 2	277.9	281.8	1.40%
HWFET	177.3	176.4	-0.52%
US06 Bag 1	425.2	429.8	1.09%
US06 Bag 2	246.5	247.4	0.35%

The comparison of two simulations using data from the actual Mazda SKYACTIV 2.5*l* engine map and the scaled 2.5*l* engine map in a generic midsize sedan is presented in Table 5 shows good agreement as well.

Transmission Scaling

The primary consideration in scaling transmission losses is based on the transmission's rated torque, on the observation that transmission efficiency curves plotted as a function of normalized load (fraction of rated torque) are comparable between different sized transmissions.

For purposes of this study, the transmission's rated torque is set to 115% of engine maximum torque. The transmission scale factor is defined as the ratio of the engine-based rated torque to the source transmission's rated torque. The transmission input torque loss, pump loss and line pressure torque index (for an AT) are multiplied by the scale factor. For ATs, the torque converter inertia is also multiplied by the scale factor. Gear-specific inertias are multiplied by the scale factor to the 1.5^{th} power, based on the assumption that the inertias will scale at a somewhat higher than linear rate based on torque rating.

Torque Converter K Factor

For trucks, the K-factor is set based on a stall speed of 3075 RPM at transmission rated torque. For other vehicles, the K-factor is based on a stall speed of 3250 RPM at transmission rated torque.

For trucks, the stall torque ratio is 2.3:1 and for all other vehicles the stall torque ratio is 2.2:1.

Generalized Behavioral Calibration

In addition to scaling measured test data, a common method for calibration of vehicle operation is necessary to avoid calibration differences tainting the calculation of technology effectiveness. This section will examine shifting, torque converter lockup and final drive ratio.

ALPHAshift

The following parameters are adjusted for each simulation case based on the engine's torque curve and the nominal vintage of the powertrain being modeled:

- Minimum (engine and/or transmission input depending on lockup state) speed in gear for past and present powertrains varies linearly by increasing gear number from 800 RPM to 1200 RPM.
- Minimum speed in gear for future powertrains varies linearly by increasing gear number from 800 RPM to 1050 RPM and represents a slight downspeeding trend.
- Minimum speed in gear is not less than engine idle speed.
- Minimum speed after upshift is at least 10 radians/sec above the minimum speed in gear and not less than 1200 RPM for current and past powertrains, and 1050 RPM for future powertrains.

- Consumption-based downshifts are enabled for future powertrains and disabled for past and present powertrains.
- The minimum required benefit for all powertrain vintages to make a consumption-based shift is a 3% decrease in consumption.
- For all vintages, engine redline speed is defined as the lesser of the engine's maximum speed minus 650 RPM and the highest speed at which the engine makes 98% of rated power. The maximum speed after downshift is limited to 80% of redline speed.

Torque Converter Lockup

Torque converter lockup strategy can have a noticeable impact on fuel consumption and therefore must be treated in a systematic manner in order to provide consistent results. For this paper, the torque converter lockup strategy is simplified to on/off behavior based purely on transmission gear and changes with powertrain vintage as noted below. The lockup gear range is defined by an always-locked and always-unlocked gear. The converter clutch will always be locked at or above the always locked gear and will always be unlocked at or below the always unlocked gear.

- For present vintage powertrains, the always-locked gear is 3rd and the always-unlocked gear is 2nd.
- For future vintage powertrains, the always-locked gear is 2nd and the always-unlocked gear is 1st.

To compensate for the losses omitted by not simulating controlled torque converter clutch slip each transmission has an efficiency associated with operation while the torque converter clutch is locked. This factor progresses with vintage ranging from 98 to 100 percent. Future work is planned to more accurately simulate this behavior and develop a generalized control strategy suitable for fleet level simulation.

Final Drive Ratio

When constructing a series of technology packages which substitute various engines and transmissions it is important to assure that there is reasonable gradeability and drivability across simulations. This helps ensure that technology effectiveness is neither over nor under estimated at the expense of gradeability or drivability. The simplest way to maintain this is to adjust the final drive ratio for the simulation of each technology package. The alternative of keeping a fixed final drive ratio for all exemplar vehicle packages would result in varying vehicle performance and less reliable technology effectiveness metrics.

To accomplish this, the final drive ratio is selected to provide a predetermined engine speed in top gear at 60 MPH as a function of engine displacement, which is based on EPA test car N/V data (ratio of engine RPM in top gear to vehicle speed in MPH) for the 2016 fleet shown in Figure 12. The red curve in Figure 12 represents the target engine speed at 60 MPH used by ALPHA for past and present vintage powertrains based on the 2016 test car data. The green curve is 250 RPM lower than the red curve and represents ALPHA's projected target engine speed at 60 MPH for future powertrains. The green curve highlights an expected slight downspeeding trend for future vehicles, recognizing that some vehicles in the 2016 fleet with high gear number transmissions are already at or below this expected trend.

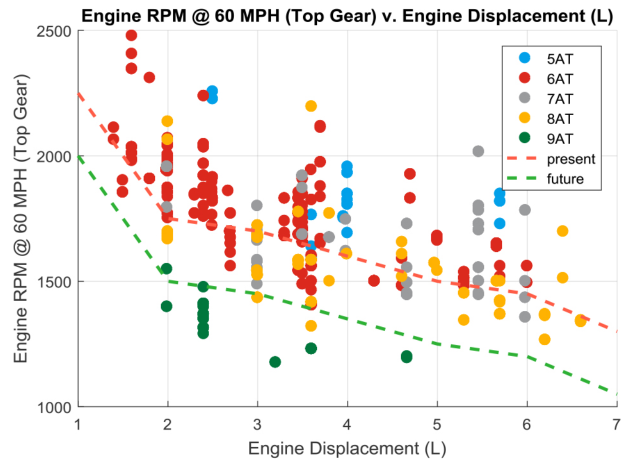

Figure 12. The data points are the engine RPM at 60 MPH versus displacement (L) based on data from the EPA 2016 test car, the red line represents the target engine speed at 60 MPH used by ALPHA for present vintage powertrains, the green line is 250 RPM lower than the red curve and represents ALPHA's projected target engine speed at 60 MPH for future powertrains.

Roadloads for Future Vehicles

One of the changes to be examined for future vehicles are improvements as result of reducing vehicle mass, tire rolling resistance and/or aerodynamic drag. Simulating with the coastdown target ABCs presents a complication as these factors are all lumped together. Vehicle mass reductions are done on a percentage basis, allowing the chassis static mass, ETW, and simulated dynamic mass which includes pertinent inertias to be scaled together.

Since the aerodynamic drag force increases with the square of vehicle speed it is assumed that the C term represents all the aerodynamic losses. Percentage improvements in aerodynamic drag are directly applied to yield a new C term as in equation 8.

$$C_{new} = C_{orig} * (1 - imp_{aero}) \tag{8}$$

Where,

C_{new} = future roadload C term

C_{orig} = exemplar roadload C term

imp_{aero} = aerodynamic drag percentage improvement

Improvements in rolling resistance are somewhat complicated by the nature of coastdown testing. While classical physics would imply that rolling resistance should be contained in the A term there is a large amount of variation in calculated target A values. This is in part attributable to coastdown procedures, which end at 15 km/hr, thus the A term is the extrapolated y-intercept. Another complicating factor at low speed can be variations in transmission drag as the speed across the clutches approaches zero. To generate consistent changes in roadload rather than proportionately scaling the A term a baseline tire coefficient of rolling resistance (Crr) of 0.01 is assumed. Along with changes in vehicle mass a rolling resistance force before and after can be calculated yielding the adjustment to the A term shown in equation 9.

$$A_{new} = A_{orig} - 0.01 * (m_{orig} - (1 - imp_{Crr}) * m_{new}) \quad (9)$$

Where,

A_{new} = future roadload A term

A_{orig} = exemplar vehicle roadload A term

m_{orig} = exemplar vehicle static mass

m_{new} = future vehicle mass after applying mass reduction

imp_{Crr} = percentage improvement in C_{rr}

Cold-Start Adjustment for FTP Simulation

ALPHA incorporates a cold-start adjustment when simulating the FTP, representing the additional fuel consumed by cold-start conditions compared to the as-simulated warm component losses. This correction is intended to estimate additional fueling associated with catalyst warmup, as well as additional powertrain losses associated with higher fluid viscosities. This adjustment is applied post-simulation, and it increases estimated fuel consumption and GHG emissions.

Vehicle Warm-Up Profile

To evaluate the warm-up of the vehicle, literature offers a "typical" warm-up profile of engine and transmission fluids during the FTP cycle [28]. This referenced paper presents the warm-up profile for engine oil, engine coolant, and transmission fluid. Engine oil and coolant are generally at operating temperature by the end of bag 1. However, the automatic transmission fluid (ATF) is not up to temperature until roughly the end of bag 2. This suggests that the adjustment for current vehicles should apply to both bag 1 and bag 2.

Cold-Start Adjustment Factors for Past & Present Vehicles

To determine the appropriate value of the cold-start adjustment, EPA test car data [29] from a range of model years was analyzed. Test car data from 2010 onwards is presented bag by bag, so fuel consumed during bag 1 (cold) can be compared to fuel consumed during bag 3 (warm). Figure 13 compares additional fuel consumption during bag 1 for all non-hybrid gasoline vehicles from the 2016 test car list.

The orange lines on the graph show the percentage difference in fuel consumption of bag 1 versus bag 3, corresponding to the median, highest 10% and lowest 10% of the test data. For 2016, the median cold-start adjustment percentage was 15.25%, while the highest and lowest 10% were at 20.25% and 11.07% respectively. For 2010, the median cold-start adjustment percentage was 16.74%, while the highest and lowest 10% were at 21.58% and 11.95% respectively. This demonstrates a noticeable, but small, decrease in the additional bag 1 fuel over time.

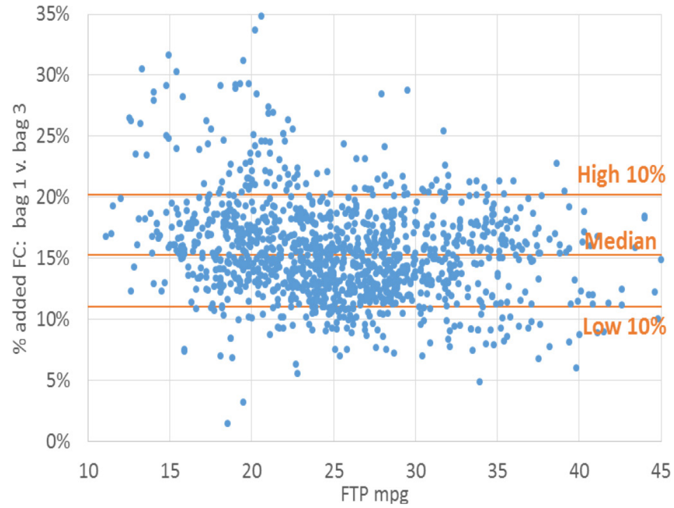

Figure 13. Ratio of FTP-bag-1 to FTP-bag-3 fuel consumption as a function of FTP mpg for the MY2016 EPA Test Car data.

For bag 2, much less data is available to draw from. However, test data for both bag 2 and a comparable warm bag 4 were available for 14 vehicles. These vehicles, from MY2012-2014, were tested at EPA [30] and Argonne National Laboratory [31]. The additional fuel used during bag 2 compared to bag 4 ranged from 0.7% to 4.4%, with an average of 2.6%.

Cold-Start Adjustment Factors for Future Vehicles

Literature also provides estimates of the improvements possible with technologies that speed up engine and transmission warm up. [28] Incorporating an early warm-up strategy into future engine and transmission packages can be expected to reduce the bag 1 adjustment by roughly 5% and reduce the bag 2 adjustment by roughly 2.5%. Compared to present values, this would show potential future adjustments could be set at roughly 10% to 12% for bag 1 and roughly 0% for bag 2.

Comparison of Cold-Start Adjustment Factors

ALPHA's default fleet-averaged FTP bag1 and bag2 cold-start adjustment factors for the vintage of each vehicle package are constructed from the data above are presented in Table 6.

Other models may apply a cold-start adjustment factor only to bag 1 results, so for as a reference and comparison purposes, the equivalent weighted bag1-only-adjustment factor is also shown in the table. The bag1-only-adjustment factor takes into consideration that fuel consumption from bag 2 is weighted more heavily in the final FTP calculation. Thus, when constructing a cold-start adjustment applied only to bag 1, additional fuel consumption during bag 2 would need to be weighted by a factor of approximately 3 (i.e., 2.6% additional fuel consumption during bag 2 is equivalent to 7.8% additional fuel consumption during bag 1).

Table 6. Fleet-averaged cold-start adjustments according to a vehicle's vintage

Package Vintage	FTP Cold-Start Adjustment		
	Bag1 Adj.	Bag2 Adj.	Equivalent Bag1-Only-Adj.
Past (c. 2008)	17%	2.5%	24.5%
Present (c. 2014)	15%	2.5%	22.5%
Future. (2022+)	11%	0.0%	11%

Performance Neutrality

Objective comparisons of the effectiveness of different technology packages can only be made when overall vehicle performance is held as constant as possible. To hold performance constant when comparing packages, ALPHA selects the engine size that produces the lowest CO2 while maintaining a performance metric equal to or better than the baseline exemplar vehicle. For this analysis, the performance metric was defined as the sum of four acceleration times (0-60 time, ¼ mile time, 30-50 passing time, and 50-70 passing time).

To determine the "performance sum" for each baseline exemplar vehicle, ALPHA simulates a "performance cycle" from which the four acceleration times are extracted and summed.

For any alternate vehicle technology package, ALPHA constructs a series of packages with a bracketing suite of engine sizes using the engine fuel consumption maps which have been scaled according to the engine scaling procedures described earlier in this paper. ALPHA then iterates to select the "right sized" engine (with its associated transmission and final drive ratio) that gives the lowest CO2 while maintaining a performance sum equal to or better than the baseline exemplar vehicle. Following this process ensures that the engine size with the lowest CO2 emissions is chosen, even if it is not the smallest engine.

EXAMPLE OF MODELING FOR FUTURE FLEETS

ALPHA is just one of the modeling tools that EPA uses to predict GHG emissions from the MY2022-2025 vehicle fleet. In addition to ALPHA, two other modeling tools are used in the MTE process: the Lumped Parameter Model (LPM) and the Optimization Model for reducing Emissions of Greenhouse gases from Automobiles (OMEGA) model [32, 33].

The LPM is calibrated to closely reflect the outputs of the ALPHA full vehicle simulation model, with the added flexibility to allow appropriate adjustments to technology effectiveness based on additional confidential or publically available information. The LPM also has the advantage of being easier to configure and run to generate technology effectiveness inputs to OMEGA, which subsequently analyzes around 100,000 vehicle configurations along with technology cost data to predict potential cost-effective technology pathways to achieve the MY2025 GHG standards for approximately 1900 vehicle models in the U.S. fleet.

For the example process in this paper, ALPHA was used to predict CO_2 emissions (in g/mi) from only the most promising of the possible future vehicle technology packages that are expected to be present in the MY2025 vehicle fleet. For each of the six vehicle classes modeled, a matrix of about 20 different vehicle technology packages was run through ALPHA. Each of the six matrices was configured to represent the typical configuration of the average vehicle for its power-weight/roadload group. These distinct vehicle classes allow EPA to uniquely model the different power and performance requirements of different groups of vehicles such as small cars, large cars, SUVs and pickup trucks. The six vehicle classes are identified as:

Non-Pickup Trucks

- LPW_LRL - Low power-weight ratio & low roadload
- MPW_LRL - Medium power-weight ratio & low roadload
- LPW_HRL - Low power-weight ratio & high roadload
- MPW_HRL - Medium power-weight ratio & high roadload
- HPW - High power-weight ratio

Pickup Trucks

- Truck

The ALPHA matrix simulation for each vehicle class contains a sweep of vehicle technology packages that successively add technology to a configuration of an exemplar vehicle representing a 2015 sales weighted average vehicle. In total for this exercise, ALPHA simulates about 120 different vehicle technology packages which are then used to calibrate the LPM.

Table 7 shows the CO_2 emission results of a sample of 36 of these 120 vehicle packages, six for each vehicle power-weight group. The column headings for Table 6 are listed below:

Vehicle Class	Exemplar vehicle's power-weight/roadload category (LPW_LRL, MPW_LRL, HPW, LPW_HRL, MPW_HRL, and Truck)
Tech Package	Unique numeric identifier for use in Figure 14 and Figure 15
ENGINE	Specific engine utilized in a technology package
TRANS	Transmission utilized in a specific technology package
ELECTRIC	Alternator loading vintage that is utilized in a specific technology package
MR	Percentage mass reduction utilized in a specific technology package
AR	Percentage aerodynamic-drag reduction utilized in a specific technology package
RR	Percentage tire rolling-resistance reduction utilized in a specific technology package
Combined FE	Fuel economy results (mpg) from the specific technology package simulated over the combined FTP and HwFET cycles

Combined CO_2	Emission results (CO_2 grams per mile) from the specific technology package simulated over the combined FTP and HwFET cycles
Effectiveness	Net percentage effectiveness from the specific technology package simulated over the combined FTP and HwFET cycles

It is important to note that the data provided in the example in Table 7 are for illustrative purposes only. They serve as an example of results from ALPHA's fleet modeling capability for this paper, and do not represent final technology configurations or the final numbers used for EPA's MTE process. Final numbers for the MTE will be published separately as part of the MTE process. In addition, the result values are displayed with two decimal digits to help analysts explore slight variations between vehicle packages, not to imply accuracy to the 100th of a gram. For the purpose of this example, the fuel was assumed to be Tier 2 certification fuel.

Table 7. Example ALPHA results from fleet-level modeling to predict CO2 emissions from future vehicle packages.

Vehicle Class	Tech Packag	ENGINE	TRANS	ELCTRIC	MR	AR	RR	Combined FE (CFR MPG)	Combined CO2 (gCO2/mi)	Effectiveness (percent)
LPW_LRL	1	**2013 Malibu 2.5L**	**2013 6AT**	**Present**	**0**	**0**	**0**	**38.16**	**232.86**	**0.00**
	2	2013 SKYACTIV 2.0 13:1	2013 6AT	Present	0	0	0	40.17	221.21	5.00
	3	2013 SKYACTIV 2.0 13:1	2014 8AT	Present	0	0	0	43.39	204.81	12.05
	4	Future ATK 14:1 cEGR CDA	Future 8AT	Future	0	0	0	51.41	172.86	25.77
	5	Future ATK 14:1 cEGR CDA	Future 8AT	Future	10	20	20	60.20	147.62	36.61
	6	Future TDS 24-bar	Future 8AT	Future	10	20	20	59.97	148.18	36.37
MPW_LRL	1	**2013 Malibu 2.5L**	**2013 6AT**	**Present**	**0**	**0**	**0**	**33.84**	**262.64**	**0.00**
	2	2013 SKYACTIV 2.0 13:1	2013 6AT	Present	0	0	0	34.96	254.20	3.21
	3	2013 SKYACTIV 2.0 13:1	2014 8AT	Present	0	0	0	38.19	232.71	11.40
	4	Future ATK 14:1 cEGR CDA	Future 8AT	Future	0	0	0	46.29	191.98	26.90
	5	Future ATK 14:1 cEGR CDA	Future 8AT	Future	10	20	20	54.30	163.67	37.68
	6	Future TDS 24-bar	Future 8AT	Future	10	20	20	53.26	166.86	36.47
LPW_HRL	1	**2013 Malibu 2.5L**	**2013 6AT**	**Present**	**0**	**0**	**0**	**31.63**	**281.00**	**0.00**
	2	2013 SKYACTIV 2.0 13:1	2013 6AT	Present	0	0	0	33.03	269.07	4.25
	3	2013 SKYACTIV 2.0 13:1	2014 8AT	Present	0	0	0	35.77	248.45	11.58
	4	Future ATK 14:1 cEGR CDA	Future 8AT	Future	0	0	0	42.09	211.12	24.87
	5	Future ATK 14:1 cEGR CDA	Future 8AT	Future	10	20	20	49.68	178.89	36.34
	6	Future TDS 24-bar	Future 8AT	Future	10	20	20	48.94	181.58	35.38
HPW	1	**2013 Malibu 2.5L**	**2013 6AT**	**Present**	**0**	**0**	**0**	**25.22**	**352.34**	**0.00**
	2	2013 SKYACTIV 2.0 13:1	2013 6AT	Present	0	0	0	26.04	341.26	3.14
	3	2013 SKYACTIV 2.0 13:1	2014 8AT	Present	0	0	0	29.39	302.43	14.17
	4	Future ATK 14:1 cEGR CDA	Future 8AT	Future	0	0	0	36.28	244.95	30.48
	5	Future ATK 14:1 cEGR CDA	Future 8AT	Future	10	20	20	42.36	209.81	40.45
	6	Future TDS 24-bar	Future 8AT	Future	10	20	20	43.32	205.15	41.78
MPW_HRL	1	**2013 Malibu 2.5L**	**2013 6AT**	**Present**	**0**	**0**	**0**	**24.54**	**362.14**	**0.00**
	2	2013 SKYACTIV 2.0 13:1	2013 6AT	Present	0	0	0	25.53	348.09	3.88
	3	2013 SKYACTIV 2.0 13:1	2014 8AT	Present	0	0	0	28.00	317.43	12.35
	4	Future ATK 14:1 cEGR CDA	Future 8AT	Future	0	0	0	33.71	263.60	27.21
	5	Future ATK 14:1 cEGR CDA	Future 8AT	Future	10	20	20	39.36	225.77	37.66
	6	Future TDS 24-bar	Future 8AT	Future	10	20	20	39.66	224.08	38.12
Truck	1	**2013 Malibu 2.5L**	**2013 6AT**	**Present**	**0**	**0**	**0**	**21.43**	**414.66**	**0.00**
	2	2013 SKYACTIV 2.0 13:1	2013 6AT	Present	0	0	0	22.17	400.94	3.31
	3	2013 SKYACTIV 2.0 13:1	2014 8AT	Present	0	0	0	24.94	356.33	14.07
	4	Future ATK 14:1 cEGR CDA	Future 8AT	Future	0	0	0	29.37	302.61	27.02
	5	Future ATK 14:1 cEGR CDA	Future 8AT	Future	10	20	20	34.35	258.74	37.60
	6	Future TDS 24-bar	Future 8AT	Future	10	20	20	35.24	252.21	39.18

The first vehicle package of each vehicle class in the Table 7 is highlighted in orange and represents the class' baseline exemplar vehicle. The table contains the vehicle package's predicted fuel economy in mpg and its GHG emissions in grams of CO_2 per mile, as well as an estimate of the GHG effectiveness in percent (with respect to the exemplar vehicle).

Figure 14 contains a summary graph of the combined EPA city and highway CO_2 emissions for each of the 36 vehicles grouped by its power-weight/roadload.

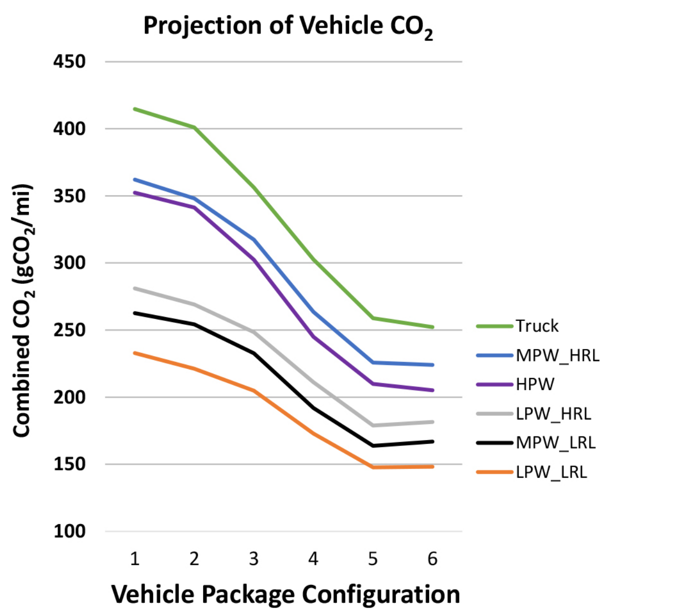

Figure 14. Combined EPA city and highway CO_2 grams per mile emissions for each of the vehicle power-weight/roadload groups.

Figure 15 graphs the percent CO_2 reduction effectiveness of each technology package with respect to its baseline exemplar vehicle.

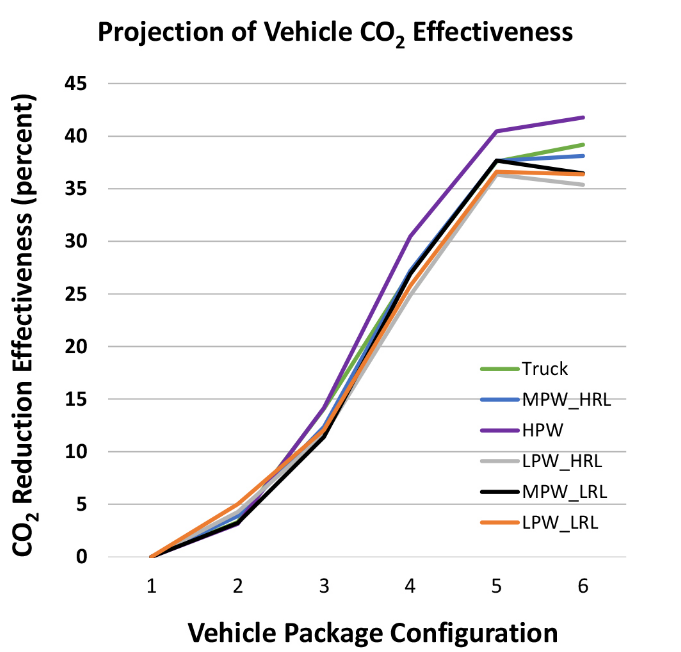

Figure 15. CO_2 reduction effectiveness for each technology package

SUMMARY

In preparation for the midterm evaluation (MTE) of the GHG standards for 2022-2025 MYs, ALPHA has been updated utilizing newly acquired data from MY2013-2016 engines and vehicles. EPA developed an approach to use its ALPHA model with a core set of current and future technologies to model CO_2 emissions from future fleet vehicles. Simulations conducted with ALPHA provide data on the effectiveness of various GHG reduction technologies, and reveal synergies that exist between technologies.

Through careful benchmarking and validations, EPA closely studied the behavior of many specific vehicles, engines and transmissions, so that observed losses could be converted into a simpler, more generic forms to establish more general patterns of vehicle operation and fuel use. These general patterns were applied in ALPHA to allow for a clear comparison of component effectiveness across various technology packages and vehicle groups.

A fleet-level matrix model capability was created to blend the unique control strategies and behavior of the individual vehicles used to validate the model to provide appropriate fleet-level GHG modeling. ALPHA was also configured to consider all the fuel required to meet vehicle and powertrain performance observed during benchmarking.

ALPHA has become EPA's primary in-house research tool to assess technology effectiveness for the MTE. It is used to study in detail the operation of current model year vehicles and to model advanced vehicle technology application in future fleet vehicles. Using this approach allows EPA engineers to follow manufacturers' operating and integration rules discovered through laboratory testing when predicting GHG emissions from future advanced technology vehicles.

With the model enhancements described in this paper, ALPHA can now be used to appropriately and accurately predict performance-neutral vehicle technology package effectiveness for MY2025 vehicles to inform calibration of EPA's Lumped Parameter Model (LPM). The outputs of the LPM can then be used in EPA's OMEGA model to determine possible compliance pathways for future fleets.

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DEFINITIONS/ABBREVIATIONS

6AT - six-speed automatic transmission

8AT - eight-speed automatic transmission

ALPHA - Advanced Light-Duty Powertrain and Hybrid Analysis tool

AKI - anti-knock index

AT - automatic transmission

ATF - automatic transmission fluid

BMEP - brake mean effective pressure

CDA - cylinder de-activation

Crr - coefficient of rolling resistance

CVT - continuously variable transmission

DCT - dual clutch transmission

EGR - exhaust gas recirculation

FMEP - friction mean effective pressure

FTP - US EPA Federal Test Procedure

FWD - front wheel drive

GDI - Gasoline direct injection

HWFET - US EPA Highway Fuel Economy Test

K-factor - capacity factor *K* of a transmission torque converter (equals the input speed divided by the square root of the input torque)

NA - naturally aspirated

MAP - manifold air pressure

MBT - Maximum Brake Torque

MT - manual transmission

MTE - midterm evaluation

MY - model year

OBD - On-Board Diagnostics

PFI - port fuel injected

RWD - for rear wheel drive

SI - spark injected

TC - turbocharger or turbocharger-boosted

UDDS - US EPA Urban Dynamometer Driving Schedule

US06 - US EPA US06 Supplemental Federal Test Procedure