A Detailed Vehicle Simulation Process to Support CAFE and CO₂ Standards for the MY 2021–2026 Final Rule Analysis

Energy Systems Division

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March 31, 2020

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Acronyms and Abbreviations

AER	all-electric range
Argonne	Argonne National Laboratory
AMTL	Advanced Mobility Technology Laboratory
AU	automatic transmission
BEV	battery electric vehicle
BISG	belt-integrated starter generator
BMEP	brake mean effective pressure
BSFC	brake-specific fuel consumption
CAFE	corporate average fuel economy
CAFE model	CAFE Compliance and Effects Modeling System
CAN	controller area network bus
CD	charge-depleting
CISG	crankshaft-integrated starter generator
CS	charge-sustaining
CVT	continuously variable transmission
DCT	dual-clutch transmission
DEAC	cylinder deactivation
DI	direct injection
DM	manual transmission
DOE	U.S. Department of Energy
DOHC	dual overhead camshaft valvetrain
DOT	U.S. Department of Transportation
EGR	exhaust gas recirculation
EPA	U.S. Environmental Protection Agency
EPCA	Energy Policy and Conservation Act
EREV	extended range electric vehicle
EV	electric vehicle
FMEP	friction mean effective pressure
FRIA	Final Regulatory Impact Analysis
FRM	Final Rulemaking
FTP	Federal Test Procedure

GVW	gross vehicle weight
HEV	hybrid electric vehicle
HIL	hardware-in-the-loop
HWFET	Highway Fuel Economy Test
ICE	internal combustion engine
IMEP	indicated mean effective pressure
I/O	input(s)/output(s)
IVM	initial vehicle movement
LHV	lower heating value
MR	mass reduction
MY	model year
NEDC	New European Driving Cycle
NHTSA	National Highway Traffic Safety Administration
NPRM	Notice of Proposed Rulemaking
NVH	noise, vibration and harness
PEV	plugged-in electrified vehicle (PHEV and BEV)
PFI	port fuel injection
PHEV	plug-in hybrid electric vehicle
PI	proportional-integral
PMEP	pumping mean effective pressure
QQ	quantile-quantile
RANSAC	random sample consensus algorithm
RCP	rapid-control prototyping
SAE	Society of Automotive Engineers
SIL	software-in-the-loop
SOC	state of charge
SUV	sport utility vehicle
TAR	Technical Assessment Report
UDDS	Urban Dynamometer Driving Schedule

VPA	vehicle propulsion architecture
VVL	variable valve lift
VVT	variable valve timing
XML	extendible markup language

Units of Measure

Ah	ampere-hour(s)
h	hour(s)
kg	kilogram(s)
km	kilometer(s)
kW	kilowatt(s)
m ²	square meter(s)
mpg	mile(s) per gallon
mph	mile(s) per hour
rad	radian(s)
rpm	rotation(s) per minute
s, sec	second(s)
V	volt(s)
W	watt(s)
Wh	watt-hour(s)

1. Introduction

In 1975, Congress passed the Energy Policy and Conservation Act (EPCA), requiring standards for corporate average fuel economy (CAFE), and charging the U.S. Department of Transportation (DOT) with the establishment and enforcement of these standards. The Secretary of Transportation has delegated these responsibilities to the National Highway Traffic Safety Administration (NHTSA).

NHTSA has contracted the DOT Volpe National Transportation Systems Center (Volpe Center) to provide analytical support for NHTSA's regulatory and analytical activities related to fuel economy standards. Unlike long-standing safety and criteria pollutant emissions standards, fuel economy standards apply to manufacturers' overall fleets rather than to individual vehicle models. In developing the standards, NHTSA made use of the CAFE Compliance and Effects Modeling System (the "CAFE model"), which was developed by DOT's Volpe Center for the 2005–2007 CAFE rulemaking and has been continually updated since. The model is the primary tool used by the agency to evaluate potential CAFE stringency levels by applying technologies incrementally to each manufacturer's fleet until the requirements under consideration are met. The CAFE model relies on numerous technology-related and economic inputs such as market forecasts and technology cost and effectiveness estimates; these inputs are categorized by vehicle classification, technology synergies, phase-in rates, cost learning curve adjustments, and technology "decision trees." The Volpe Center assists NHTSA in the development of the engineering and economic inputs to the CAFE model by analyzing the application of potential technologies to the current automotive industry vehicle fleet to determine the feasibility of future CAFE standards, the associated costs, and the benefits of the standards.

Part of the CAFE model's function is to estimate CAFE improvements that a given manufacturer could achieve by applying additional technologies to specific vehicles in its product line. Because CAFE standards apply to the average fuel economy across manufacturers' entire fleets of new passenger cars and light trucks, the model, when simulating manufacturers' potential application of technology, considers the entire range of each manufacturer's product line. This typically involves accounting for more than 1,000 distinct vehicle models and variants, many more than can be practically examined using full vehicle simulation (or the other techniques mentioned above). Instead, the model uses estimates of the effectiveness of specific technologies for a representative vehicle in each vehicle class, and arranges technologies in decision trees defining logical progressions from lower to higher levels of cost, complexity, development requirements, and/or implementation challenges.

NHTSA has made use of full vehicle simulation results to update technology effectiveness estimates used by the CAFE model. In recent rulemakings, the decision trees have been significantly expanded for NHTSA to better track the incremental cost and effectiveness associated with each technology. A detailed description of the CAFE model can be found in NHTSA's Final Regulatory Impact Analysis (FRIA) supporting the 2012 rule establishing CAFE standards applicable beginning with model year (MY) 2017.

One significant update to the CAFE model since the 2012 rule is that the model now includes functionality to support the Environmental Protection Agency's carbon dioxide (CO₂) emissions standards rulemaking. This update and others are described in the NPRM and final rule for MYs 2021-2026 CAFE and CO₂ standards. All rulemaking documents, the model, source code, model documentation, and all model inputs and outputs are available at <http://www.nhtsa.gov/fuel-economy>.

All inputs to CAFE's decision tree model are related to the effectiveness (fuel consumption reduction) of each fuel-saving technology. Although vehicle testing could be used to estimate these factors, vehicle testing that spans many vehicle types and technology combinations could be prohibitively resource-intensive. Another alternative, either as a substitute for or as a complement to vehicle testing, is to make greater use of vehicle simulation. Full vehicle simulation tools use physics-based mathematical equations, engineering characteristics (e.g., engine maps, transmission shift points, hybrid vehicle control strategies), and explicit drive cycles to predict the effectiveness of individual fuel-saving technologies as well as their combinations.

Argonne National Laboratory, a U.S. DOE national laboratory, has developed a full vehicle simulation tool, Autonomie, which has become one of the industry's standard tools for analyzing vehicle performance, energy consumption, and technology effectiveness. Through an Inter Agency Agreement, the DOE Argonne Site Office and Argonne National Laboratory have been tasked with conducting full vehicle simulation to support NHTSA CAFE rulemaking.

This report describes the process that Argonne used to conduct full vehicle simulation for the MY 2021-2026 final CAFE and CO₂ standards rule.

2. Project Statement

The CAFE model currently relies on multiple decision trees to represent the component technology options identified below:

- Powertrain
- Engine
- Transmission
- Light-weighting
- Aerodynamics
- Rolling resistance

Figure 1 shows the variety of technology combinations adapted to represent current and potential future technologies that are simulated for the final rule.

Figure 1. Technologies Considered for the Argonne Autonomie Final Rulemaking (FRM) Analysis

3. Autonomie Simulation Process

From the draft Technical Assessment Report (TAR) through the Notice of Proposed Rulemaking (NPRM) analysis, NHTSA directed the Argonne simulation team to update inputs and processes based on an extensive review of comments, technical publications and meetings with stakeholders. In order to address the comments in the draft TAR, performance categories for the initial five vehicle classes simulated were added, with various performance times and estimated curb weights (and towing capacities for the pickup class) that would span a greater variety of vehicles on the road. The component weights were also updated using the A2Mac1 database (A2Mac1, 2018) across vehicle classes.

As mentioned, individual classes and performance categories have been simulated for every combination of vehicle, powertrain, and component technology. The combinations include the following:

- Five vehicle classes (compact, midsize, small SUV, midsize SUV, pickup)
- Two performance categories for each vehicle class: non-performance (base) and performance (premium)
- Twenty-two engine technologies
- Sixteen transmission technologies
- Seven light-weighting levels
- Five aerodynamic reduction levels
- Three rolling resistance reduction levels
- Two battery electric vehicle ranges

Along with the increased number of technology combinations, vehicle models were also updated in NPRM runs. Changes in the process also included technology inheritance (platform-sharing) of engines and transmissions.

The process developed includes the following steps:

1. Specify the vehicle technology specifications (VTS).
2. Select component assumptions.
3. Build the vehicle models.
4. Size the reference powertrains to meet the desired performance.
5. Build all the different vehicle combinations, including “inheritance.”
6. Simulated individual vehicles on the US standard driving cycles.
7. Perform QA/QC checks on the simulation results.
8. Create CAFE model database.

4. Autonomie Background

4.1. Overview

Autonomie is a Mathworks®-based software environment and framework for automotive control-system design, simulation, and analysis. The tool, sponsored by the DOE Vehicle Technologies Office (VTO), is designed for rapid and easy integration of models with varying levels of detail (low to high fidelity), abstraction (from subsystems to systems to entire architectures), and processes (e.g., calibration, validation). Developed by Argonne in collaboration with General Motors, Autonomie was designed to serve as a single tool that can be used to meet the requirements of automotive engineers throughout the development process—from modeling to control. Autonomie was built to:

- Estimate the energy, performance, and cost impact of advanced vehicle and powertrain technologies
- Support proper methods, from model-in-the-loop, software-in-the-loop (SIL), and hardware-in-the-loop (HIL) to rapid-control prototyping (RCP)
- Integrate math-based engineering activities through all stages of development — from feasibility studies to production release
- Promote re-use and exchange of models industry-wide through its modeling architecture and framework
- Support users' customization of the entire software package, including system architecture, processes, and post-processing
- Mix and match models with different levels of abstraction to facilitate execution efficiency with higher-fidelity models, for which analysis and high-detail understanding are critical
- Link with commercial off-the-shelf software applications, including GT-POWER, AMESim™, and CarSim®, for detailed, physically based models
- Protect proprietary models and processes

By building models automatically, Autonomie enables simulation of an unparalleled number of component technologies and powertrain configurations. Autonomie offers the following capabilities:

- Simulate subsystems, systems, or entire vehicles
- Predict and analyze energy efficiency and performance
- Perform analyses and tests for virtual calibration, verification, and validation of hardware models and algorithms
- Support system hardware and software requirements
- Link to optimization algorithms

- Supply libraries of models for propulsion architectures of conventional powertrains, as well as electric-drive vehicles

In this study, Autonomie is used to assess the energy consumption of advanced powertrain technologies across vehicle classes and configurations. Autonomie has been validated for all powertrain configurations on multiple validated vehicles currently in the market using dynamometer test data from Argonne’s Advanced Mobility Technology Laboratory (AMTL) formerly known as APRF (Argonne Research Powertrain Facility).¹

With hundreds of pre-defined powertrain configurations and controllers, Autonomie is a unique tool to analyze the impact of different technologies (e.g., powertrains, components, controls, etc.) across different vehicle classes.

Autonomie allows users to evaluate the impact of component sizing on energy consumption for different technologies, as well as to define the component requirements (e.g., power, energy) to maximize fuel displacement for a specific application. This is important for the current study because the use of validated plant models, vehicle controls, and complete vehicle models is critical to properly evaluating the benefit of any specific technology. The vehicle-level control algorithms (e.g., engine ON/OFF logic, component operating conditions algorithm) are crucial for properly quantifying technology impact, especially for electric drives. Argonne has been developing shifting algorithms for conventional vehicles based on different component characteristics (e.g., engine fuel flow rate, gear ratios), as well as developing vehicle-level controls for electrified powertrains, for more than 20 years.

The ability to simulate a large number of powertrain configurations, component technologies, and vehicle-level controls over numerous drive cycles has been used to support a large number of studies, focusing on energy efficiency, cost-benefit analysis, or greenhouse gases.²

¹ Autonomie Model Validation: <https://www.autonomie.net/currentprojects/ModelValidation.html>

² Lee, H., Lee, B., Kim, S., Kim, N., Rousseau, A., “Model-based fuel economy technology assessment,” SAE 2017-01-0532, SAE World Congress, Detroit, April 2017. R. Vijayagopal, N. Shidore, M. Reynolds, C. Folkerts, A. Rousseau, “Estimating the Fuel Displacement Potential of a Thermoelectric Generator in a Conventional Vehicle Using Simulation,” EVS27, Oct 2013, Barcelona. Jeong, J., Kim, N., Stutenberg, K., Rousseau, A., “Analysis and Model Validation of the Toyota Prius Prime,” SAE 2019-01-0369, SAE World Congress, Detroit, April 2019; Kim, N., Jeong, J., Rousseau, A. & Lohse-Busch, H. “Control Analysis and Thermal Model Development of PHEV,” SAE 2015-01-1157, SAE World Congress, Detroit, April 15; Kim, N., Rousseau, A. & Lohse-Busch, H. “Advanced Automatic Transmission Model Validation Using Dynamometer Test Data,” SAE 2014-01-1778, SAE World Congress, Detroit, Apr 14.; Lee, D., Rousseau, A. & Rask, E. “Development and Validation of the Ford Focus BEV Vehicle Model,” 2014-01-1809, SAE World Congress, Detroit, Apr 14; Kim, N., Kim, N., Rousseau, A., & Duoba, M. “Validating Volt PHEV Model with Dynamometer Test Data using Autonomie,” SAE 2013-01-1458, SAE World Congress, Detroit, Apr 13.; Kim, N., Rousseau, A., & Rask, E. “Autonomie Model Validation with Test Data for 2010 Toyota Prius,” SAE 2012-01-1040, SAE World Congress, Detroit, Apr 12; Karbowski, D., Rousseau, A., Pagerit, S., & Sharer, P. “Plug-in Vehicle Control Strategy - From Global Optimization to Real Time Application,” 22th International Electric Vehicle Symposium (EVS22), Yokohama, (October 2006)].

More than 250 companies and research entities, including major automotive companies, suppliers and research organizations, have licensed Autonomie to support advanced vehicle development programs.

4.2. Structure

Autonomie was designed for full plug-and-play support. Models in the standard format create building blocks, which are assembled at run time into a simulation model of a vehicle, system, or subsystem. All parts of the user interface are designed to be flexible to support architectures, systems, subsystems, and processes not yet envisioned. The software can be molded to individual uses, so it can grow as requirements increase and technical knowledge expands. This flexibility also allows for implementation of legacy models, including plant and controls.

Autonomie is based on standardized modeling architecture, on-demand model building, associated extendible markup language (XML) definition files, and user interfaces for managing models, including a file-versioning database, as seen in Figure 2.

Figure 2. Simulation Management Concepts

All systems in the vehicle architecture can be logically categorized as either a “containing system” or a “terminating system” (Figure 3). Containing systems consist of one or more subsystems, as well as optional files to define that system. They do not contain models; they only describe the structure of the interconnections among systems and subsystems. Terminating systems consist of a model that defines the behavior of the system and any files needed to provide inputs or calculate outputs. Terminating system models contain the equations that describe the mathematical functions of a system or subsystem.

Both types of systems are arranged in a hierarchical fashion to define the vehicle to be simulated. To avoid confusion, it is a best practice to mimic the structure of the hardware as much as possible. For example, low-level component controllers should be grouped with the components that they control, at different levels of the hierarchy (where applicable). Only systems that appear in the vehicle should be represented; in other words, there is no need for unused components or empty controllers. In addition to simplifying the architecture, this philosophy will allow for easy transfer of systems among users and will fully support HIL, SIL, and RCP.

Figure 3. Class Diagram of Container and Terminating Systems

At the top level is a vehicle system containing the following systems: environment, driver, vehicle propulsion controller for advanced powertrain vehicles, such as hybrid electric vehicles (HEVs) or plug-in hybrid electric vehicles (PHEVs), which require a vehicle level controller, and vehicle propulsion architecture (VPA) (Figure 4). The VPA system will contain the powertrain components that are required to simulate the vehicle, such as engine, battery, and wheels.

Figure 4. Top-Level Vehicle Layout

The model files created for the terminating systems are combined to allow simulation in Simulink®. One option is to create every possible combination of the systems and save each complete vehicle as a separate model file. Because of the staggering number of possible combinations, this option is not feasible. Combinations involve not only many different components, but also different levels of fidelity and model versions for each component. Changing the version of a single component model would result in a new version of the entire vehicle. This method is clearly storage-intensive and impractical.

A second option is to save every model in its own file and manage a library of the models. This would be an improvement over the first option; however, it still presents some difficulties. When users wish to create a new vehicle, they must select all the appropriate models from the library and connect them by hand into a vehicle context. Not only is this manual process time-consuming, but it introduces many opportunities for error. Consider an engine control unit model for auto code generation that can have more than 2,000 inputs and outputs (I/Os). Manually connecting all I/Os almost guarantees errors. It also requires some outside solution for model library management (e.g., searching, versioning, and ensuring compatibility).

Autonomie uses a patented approach that combines the second option with an automated building process, giving the user the flexibility of saving and versioning models independently without the potential pitfalls of manual connections. Users select the desired files in a user interface, and the

automatic building uses metadata associated with the models to create the correct connections, as shown in Figure 5.

Figure 5. Models Are Automatically Built

The ability to automatically build entire vehicle models is critical to the current study, as more than 1.5 million different vehicle models are considered.

4.3. Engine Model

Autonomie uses different engine models to support specific technologies (cylinder deactivation, turbocharging, etc.). For this study, five different engine models are used:

- Gasoline engine model
- Diesel engine model
- Gasoline turbocharged engine model
- Cylinder deactivation engine model
- Gasoline turbocharged with cylinder deactivation engine model

This section describes the baseline Autonomie engine model.

4.3.1. Model Description

The engine model simulates engine torque production, fuel consumption, etc. In this model, the fuel rate is expressed as a function of the engine's brake torque and speed.

The engine model is divided into multiple blocks: engine torque calculation, engine thermal calculation, and engine fuel rate calculation. Figure 6 shows the Autonomie spark-ignition engine model diagram, with these blocks outlined in red.

Figure 6. Autonomie Spark-Ignition Engine Model Diagram

Block A (Figure 7) calculates the engine torque by interpolating between the maximum and minimum torque curves, using the engine command from the controller.

Figure 7. Engine Model, Block A: Engine Torque Calculation

If $T_{cmd} > 0$ and $\omega_{eng} > 0$ or T_{cmd} equals 0 and $\omega_{eng} > 0$, then

$$(1) T_{out} = (1 - T_{cmd}) \cdot T_{CTT} + (T_{cmd} \cdot T_{WOT})$$

Otherwise, if T_{cmd} equals 0 and ω_{eng} equals 0, then

$$(1a) T_{out} = 0$$

$$(2) T_{cmd} = PWM_cmd \cdot Eng_ON \cdot (W_{eng} > W_{starting})$$

- T_{CTT} = minimum torque curve of the engine as a function of speed (closed-throttle torque curve).
- T_{WOT} = maximum torque curve of the engine as a function of speed (wide-open-throttle torque curve).
- PWM_cmd = engine command from controller, not modified by any other signals.
- $W_{starting}$ = engine starting speed threshold (if the engine speed is above this value, combustion is assumed to be stable, and the engine is considered to consuming fuel and capable of producing torque).

As this engine model is only for hot operation, the engine temperature, exhaust gas temperature, and warm-up coefficient are all set to constant values, as shown in Block B (Figure 8). The engine cold-start

The diagram illustrates the engine model's data flow. It starts with inputs **eng_spd** and **eng_trq** (red cylinders) passing through gain blocks into the **Autonomie Limited 2D Map** (containing a contour plot). The map outputs **Presid (J/kg)** to a multiplication block (**X**) along with **fuel rate** (red cylinder). The output of this block is summed (+) with another input from a multiplication block (which also takes **eng_trq** and a gain) to produce **heat rejection (W)**. This signal is then integrated by a **1/s** block to produce **eng_htrej_cum_simu**. Other inputs include **eng_temp_coeff_index** (1), **eng.plant.init.temp_operating** (5), and **eng.plant.init.ex.temp_operating** (2), which are used to calculate **eng_ex_temp_gas** and **eng_htrej_rate_simu**.

$$(3) \text{ Instantaneous_Heat_Rejection} = (\text{Fuel_rate} \cdot \text{Heat_rejected_per_unit_mass}) - (W_{eng} \cdot T_{eng})$$

- Fuel_rate = mass flow rate of the engine fuel.
- Heat_rejected_per_unit_mass = total energy released from the fuel by combustion per unit mass of fuel.

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Figure 9. Engine Model, Block C: Fuel Rate Model

Equation 6 interpolates the fuel rate when the engine is on and the torque delivered by the engine is below the torque boundary of the fuel rate map. Since no data are usually available in this region, the fuel rate is interpolated. When $T_{eng} = T_{min_map}$, the fuel rate at T_{min_map} is requested. When $T_{eng} = T_{CTT}$, the fuel rate is zero. The fuel rate is proportional to the engine torque fraction. Equation 7 ensures that no fuel is consumed when the engine is below its starting speed; the engine speed has to be increased up to starting speed by the starter motor before fuel can be injected. Note that the starting speed is lower than the engine idle speed.

If the engine is started and $T_{eng} > T_{min_map}$,

$$(5) \text{ Instantaneous_fuel_rate} = \text{function}(W_{eng}, T_{eng})$$

If the engine is started and $T_{eng} < T_{min_map}$,

$$(6) \text{ Instantaneous_fuel_rate} = \frac{\text{Fuel_rate} \cdot (T_{eng} - T_{CTT}) \cdot (T_{eng} - T_{min_map})}{T_{min_map} - T_{CTT}}$$

T_{min_map} = minimum torque index (minimum value in the vector eng.init.trq_fuel_hot_index).

This is the torque boundary for the fuel rate map. Below this value, unless data are available, the map must be interpolated.

Otherwise,

$$(7) \text{ Instantaneous_fuel_rate} = 0$$

Equation 8 calculates the total mass of fuel that went into the engine:

$$(8) \text{ Cumulative_fuel_rate} = \int \text{Instantaneous_fuel_rate} \cdot dt$$

4.3.2. Engine Technologies Evaluated

This section provides detail on the different engine technologies modeled in the gasoline and diesel engines.

4.3.2.1. Friction Reduction

Friction reduction has been shown to offer significant improvements in vehicle fuel consumption. Therefore, to evaluate the potential of friction reduction, engines can potentially be subjected to two levels of reduction in friction mean effective pressure (FMEP):

1. A reduction in FMEP by 0.1 bar across the entire engine speed range.
2. An extreme friction reduction (25% FMEP) across the entire speed range.

For the current study, only the first level of friction reduction has been considered. A predictive FMEP equation was calibrated from test data to allow for a smooth and systemic friction study, but it may under-predict FMEP at high loads with late combustion phasing.

4.3.2.2. Turbocharged Engines

In addition to the naturally aspirated engines, turbo engines were also mapped using GT-POWER, a commercially available engine simulation tool with detailed cylinder modeling and combustion analysis. With turbo engines, there is a “lag” in torque delivery because of the operation of the turbo charger. This lag impacts vehicle performance and vehicle shifting on aggressive cycles. Turbo lag has been modeled in Autonomie for turbo systems based on principles of a first order delay, where the turbo lag kicks in after the naturally aspirated torque limit of the turbo engines has been reached.

Figure 10 shows the brake mean effective pressure (BMEP) response of the turbo engine model to a step command.

Figure 10. BMEP Response of Turbo-Charged Engine to a Step Command

The turbo response varies with engine speed; i.e., at higher speeds, the turbo response is faster because of higher exhaust flow rates. It should be noted that the baseline engine maps for the naturally aspirated and the turbo engines were validated with test data. Maximum torque line on boosted engines is adjustable on the basis of boost pressure.

4.3.2.3. Cylinder Deactivation and Advanced Cylinder Deactivation

Autonomie also uses a specific engine model for cylinder deactivation, as this model uses an advanced fuel calculation subsystem with different maps. Due to noise, vibration and harness (NVH) considerations in production vehicles, cylinder deactivation operation is not performed during several vehicle operation modes, including vehicle warm-up, lower-gear operation, idle, and low engine speed. As a result, cylinder deactivation is disabled under the following conditions:

- If the engine idles or its speed is below 1000 RPM or above 3000 RPM
- If the vehicle is in the first or second gear
- If the engine load is above half the maximum BMEP of the engine (and a certain hysteresis is maintained to prevent constant activation and deactivation)

As noted above, cylinder deactivation is not typically performed during the vehicle warm-up phase, i.e., for a cold start. Since all the simulations considered in this study assume a “hot start,” where the engine coolant temperature is steady at around 95°C, the cold start condition was not considered for the simulations. In addition, changes in the transmission shifting calibration (like lugging speed limits) and additional torque converter slippage during cylinder deactivation, have also not been considered.

4.3.2.4. Engine Cylinder Deactivation Methodology

The cylinder deactivation state is implemented in a Stateflow® diagram as shown in Figure 11.

Figure 11. Stateflow Diagram for Cylinder Deactivation Regulator

In the engine plant model, the fuel rate maps for both cylinder deactivation and no cylinder deactivation are used by the Stateflow logic. Figure 12 shows the engine plant model with cylinder deactivation.

It shows how both the fuel maps (with and without cylinder deactivation) is used by the Stateflow logic in the engine plant model.

Figure 12. Engine Plant Model with Cylinder Deactivation

Figure 13 shows the Stateflow control that switches the fuel maps for the cylinder deactivation and no cylinder deactivation conditions on the basis of the cylinder deactivation signal.

Figure 13. Stateflow Diagram Showing the Capability for Switching Fuel Maps of Cylinder Deactivation

As shown in Figure 13, a state diagram is used to switch between the two different engine maps (state fuelling vs. fuelling with cylinder deactivation) based on whether or not the cylinder deactivation state is active (cylinder_deac_state).

4.3.2.5. Fuel Cutoff

Autonomie uses fuel cutoff methodology for the engine models considered in this study. This engine model has a specific torque calculation to calculate the torque loss when fuel is cut off during deceleration events. In general, engine models in Autonomie are of two types, throttled engines and unthrottled engines. As shown in Figure 14 and Figure 15, both types of models provide motoring torque when fuel is cut to the engine (e.g., when fuel is cut off during deceleration). With throttled engines, the motoring torque is a function of throttle position. Figure 14 shows the engine operating regions for throttled engines, and Figure 15 shows the engine operating regions for unthrottled engines.

Figure 14. Engine Operating Regions for Throttled Engines (WOT = Wide-Open Throttle)

Figure 15. Engine Operating Region for Unthrottled Engines

Engine fuel cutoff control algorithms used in the study have been developed on the basis of vehicle test data collected at the AMTL.³ The fuel cutoff controller is implemented for gasoline and diesel engines

³ 2017 Ford F150, 2016 Mazda CX9, 2014 Chevrolet Cruze Diesel, 2014 Mazda 3 iEloop, 2013 Dodge Ram 1500 HFE, 2013 Hyundai Sonata, 2013 Nissan Altima, 2013 Volkswagen Jetta TDI, 2012 Chrysler 300, 2012 Fiat 500 Sport, 2012 Ford F150 Ecoboost, 2012 Ford Focus, 2012 Ford Fusion V6, 2009 Volkswagen Jetta TDI.

<https://www.anl.gov/es/conventional-vehicle-testing>

through analysis, as shown in Figure 16. In Autonomie, engine control and plant blocks are organized for idle fuel rate and fuel-off conditions. Engine fuel is cut off under the following conditions:

- Vehicle is actively braking for a certain minimum time.
- Engine speed is above a minimum threshold (e.g., 1000 RPM).

Figure 16. Engine Fuel Cutoff Analysis Based on Test Data (Data Source: AMTL)

4.4. Transmission Model

The drivetrain is considered to be rigidly attached to the wheels. Since the wheel speed and acceleration are calculated in the wheel model and propagated backward throughout the rest of the drivetrain model, the gearbox unit is modeled as a sequence of mechanical torque gains. The torque and speed are multiplied and divided, respectively, by the current ratio for the selected gear. Furthermore, torque losses corresponding to the torque/speed operating point are subtracted from the torque input. Torque losses are defined on the basis of a three-dimensional efficiency lookup table (input shaft rotational speed, input shaft torque, and gear number). When a gear is selected, the input inertia is fed forward to the next component after being reflected to the output shaft using the square of the gear ratio. When the neutral gear is engaged, the input gearbox rotational speed is calculated on the basis of the input shaft inertia.

4.4.1. Automatic Transmission

For automatic transmissions, gear shifting occurs without having to pass through neutral and without a complete torque interruption at its output. The torque converter model is separate from the automatic gearbox model.

Figure 17 shows the I/O of the automatic-transmission model.

Figure 17. Autonomie Automatic Gearbox Model I/O

The top-level diagram of the automatic gearbox is composed of three main subsystems: speed calculation (Block A); torque calculation (Block B); and inertia calculation (Block C). Figure 18 shows the top-level diagram of the automatic gearbox model.

Figure 18. Top Level of the Automatic-Gearbox Model

Block A (Figure 19) calculates the gearbox shaft input speed based on the output speed, gear ratio, and whether the gearbox is in neutral. If the gearbox is in first gear, Equation 1 is used to calculate the input speed of the gearbox.

$$(1) W_{in} = K_{Ratio} \cdot W_{out}$$

Figure 19. Block A: Speed Calculation of the Automatic Transmission

In between gear choices during an automatic transmission shift, the output shaft speed lags the input shaft speed as one set of wet clutches opens and another closes, allowing for a smooth gear shift and no interruption in torque output. This speed characteristic of the automatic gearbox is modeled in Block A1 using a low-pass filter. However, this low-pass filter cannot be used during a shift from neutral into first gear. This shift occurs during a performance test for a limited subset of hybrid configurations that are in neutral when the vehicle is stopped instead of being in first gear, which is the state for a stopped conventional vehicle. The low-pass filter interferes with the performance test, slowing the vehicle down and leading to exaggerated 0-to-60 mph times. To address this issue, the filter is not used during performance to ensure that when the vehicle is shifting from neutral into first, the input speed is proportional to the output speed.

Block A1 (Figure 20) calculates the input speed when the gearbox is not in neutral. Any gear changes will result in the input speed lagging the output speed. Equation 2 implements the phase difference between input and output speeds.

Figure 20. Block A1: Speed Calculation When the Gearbox Is Not in Neutral

Equation 2 shows the calculation of the input speed when the gear number is higher than first.

$$(2) W_{in} = \frac{1}{1 + \tau S} \cdot (K_{Ratio} \cdot W_{out})$$

Block A2 (Figure 21) calculates the input speed when the gearbox is in neutral.

Figure 21. Block A2: Speed Calculation When the Gearbox Is in Neutral

When the gearbox is in neutral, the input shaft speed is calculated by using the first-order differential equation shown (Equation 3). If the input torque goes to 0, the shaft will continue to spin until the drag torque brings it to rest.

$$(3) W_{in} = \int \frac{T_{in} - T_{loss}}{J_{in}}$$

Block A2-1 (Figure 22) calculates the torque loss of the free-wheeling input shaft. The torque loss is proportional to the shaft speed, as illustrated by Equation 4.

$$(4) T_{loss} = K_{coeff} \cdot W_{in}$$

Figure 22. Block A2-1: When The Gearbox Is Not in Neutral, with Free-Wheeling Input Shaft Losses

Block B (Figure 23) calculates the gearbox output torque using the gearbox ratio and torque loss.

Figure 23. Block B: Output Torque Calculation

Equation 5 shows the output torque calculation based on input torque minus torque loss.

$$(5) \quad T_{out} = T_{in} - T_{loss}$$

Torque loss is a function of the gearbox input torque, speed, and ratio.

$$(6) \quad T_{loss} = f(T_{in}, W_{in}, K_{Ratio})$$

When the gearbox is in neutral, the torque output is forced to 0.

$$(6a) \quad T_{loss} = 0$$

Block C (Figure 24) gives the output inertia of the gearbox.

Figure 24. Block C: Output Inertia Calculation

The output inertia equals the upstream inertia reflected to the output shaft of the gearbox through the gear ratio plus the output shaft inertia of the gearbox.

$$(7) J_{out} = (J_{in} + J_{input_shaft_tx}) \cdot K_{ratio}^2 + J_{output_shaft_tx}$$

When the gearbox is in neutral, Equation 8 demonstrates that the output inertia of the gearbox is the inertia of the output shaft.

$$(8) J_{out} = J_{output_shaft_tx}$$

4.4.2. Manual Transmission

Figure 25 shows the main I/O of the manual transmission model.

Figure 25. Autonomie Manual Transmission Model I/O

The top-level diagram of the manual gearbox is composed of three main subsystems: speed calculation (Block A); torque calculation (Block B); and inertia calculation (Block C). Figure 26 shows the top-level diagram of the manual gearbox model.

Figure 26. Top Level of the Manual Gearbox Model

Block A (Figure 27) calculates the input shaft speed of the gearbox on the basis of the current output shaft speed, the current gear ratio, and whether the gearbox is in neutral.

Figure 27. Block A: Gearbox Input Shaft Speed Calculation

If the gearbox is in gear, the input shaft speed is proportional to the output shaft speed. If the gearbox is in neutral, the input shaft is free to spin. The spinning shaft produces drag losses that are proportional to its speed. Equation 1 shows the speed calculation when the gearbox is in gear.

$$(1) W_{in} = K_{Ratio} \cdot W_{out}$$

When the gearbox is in neutral, the input shaft speed is calculated by using the first-order differential equation shown below (Equation 2). If the input torque goes to 0, the shaft will continue to spin until the drag torque brings it to rest.

$$(2) \dot{W}_{in} = \int \frac{T_{in} - T_{loss}}{J_{in}}$$

Block A1 (Figure 28) shows the torque loss calculation when the gearbox is in neutral.

Figure 28. Block A1: Torque Loss Calculation When the Gearbox Is in Neutral

The torque losses are proportional to the input speed, as shown in Equation 3.

$$(3) T_{loss} = K_{coeff} \cdot W_{in}$$

Block B (Figure 29) calculates the torque output of the gearbox, accounting for the torque gain due to the ratio and the torque loss in the gearbox.

Figure 29. Block B: Speed Calculation When the Gearbox Is in Neutral

Equation 4 shows that the output torque before applying the ratio gain is the output torque minus the torque loss.

$$(4) T_{out} = T_{in} - T_{loss}$$

When the gearbox is in neutral, the torque output is forced to 0.

$$(4a) T_{out} = 0$$

The torque loss is a function of the gearbox input torque, speed, and ratio.

$$(5) T_{loss} = f(T_{in}, W_{in}, K_{ratio})$$

Block C (Figure 30) gives the output inertia of the gearbox.

Figure 30. Block C: Output Inertia Calculation

The output inertia equals the upstream inertia reflected to the output shaft of the gearbox through the gear ratio plus the output shaft inertia of the gearbox.

$$(6) J_{out} = (J_{in} + J_{input_shaft_tx}) \cdot K_{ratio}^2 + J_{output_shaft_tx}$$

When the gearbox is in neutral, Equation 7 demonstrates that the output inertia of the gearbox is the inertia of the output shaft.

$$(7) J_{out} = J_{output_shaft_tx}$$

4.4.3. Continuously Variable Transmission

4.4.3.1. Plant Model

The metal V-belt CVT model was developed by considering hydraulic and mechanical loss. The hydraulic loss constitutes the majority of the total loss at low vehicle speed, whereas the mechanical loss is the main source of inefficiency at high speed, as shown in Figure 31. The CVT model considers the oil pump efficiency as a function of the line pressure and input speed. The mechanical loss map, collected from experimental data, is also used as a function of the speed ratio, input torque, and vehicle speed.

Figure 31. Hydraulic Pump Loss and Mechanical Loss for Metal V-Belt Continuously Variable Transmission (Francis, 2006)

Generally, the operating conditions of the metal V-belt CVT system can be described by the following five parameters:

- Primary clamping force (F_p) or primary pressure (P_p)
- Secondary clamping force (F_s) or secondary pressure (P_s)
- Primary revolution speed (ω_p)
- Input torque (T_{IN})
- Pulley ratio (i)

On both the primary and the secondary pulleys, the belt is clamped by the forces produced by the hydraulic pressures in the cylinders. These two clamping forces, F_p and F_s , counteract each other. Therefore, when the pulley ratio is constant, there is a balance between F_p and F_s . A ratio change occurs when their balance is lost. In the next subsection, the relation between F_p and F_s in a state of balance is discussed. After that, di/dt is considered.

4.4.3.2. Variator Clamping Force Model

In the CVT system, the pulley ratio is controlled by the primary pulley. Therefore, we consider that the state of balance is produced by the primary clamping force F_p . The necessary primary clamping force for counterbalance is denoted by F_p^* , and the corresponding primary pressure is denoted by P_p^* .

The secondary actuator pressure P_s can be obtained for the given input torque T_{IN} and CVT ratio i from Fujii's formula:

$$(1) F_s = \frac{T_{IN}}{2\mu R_p} \cos \frac{\alpha}{2}$$

where μ is the friction coefficient between the belt and the pulley and R_p is the pitch radius of the primary pulley. In the CVT system design, F_s should be controlled to prevent slippage between the belt and the pulley.

The primary pressure P_p at steady state can be determined from the thrust ratio map. The thrust ratio shows the relationship between the steady-state primary force F_p and the secondary force F_s with respect to the speed ratio i and the torque ratio. The thrust ratio $F_p F_s$ is represented as

$$(2) \quad F_p F_s = \frac{F_p}{F_s} = f\left(i_{CVT}, \frac{T_{IN}}{T_{max}}\right)$$

For the given speed ratio i and the input torque T_{IN} , the secondary pressure P_s is calculated from Equation 1. The primary actuator force F_s is determined from the thrust ratio map for the given i and ratio $F_p F_s$. The primary pressure can be calculated from Equation 2. Figure 32 shows the $F_p F_s$ ratio map of the CVT.

Figure 32. $F_p F_s$ Ratio Map (Continuously Variable Transmission)

4.4.3.3. CVT Shift Dynamics Model

In order to obtain the dependency of the ratio changing speed di/dt on the five above-mentioned parameters, we consider the Ide's experiment formula (T.Ide, 1995), which is based on a test rig using only the belt and pulley set. The deviation of P_p from P_p^* is denoted by ΔP_p . Then, di/dt is expressed by the following equation:

$$(3) \quad \frac{di}{dt} = \beta(i) \cdot \omega_p \cdot (P_p - P_p^*)$$

The parameters of Ide's formula were determined from experiments reported in the literature. The linear relation is also obtained for all pulley ratios. The effect of P_p on di/dt is obtained from the five

parameters (P_p , P_s , ω_p , T_{IN} , i). This formula shows that di/dt is in a proportional relation with ΔP_p under loaded conditions as well, and the slope β hardly changes even if T_{IN} changes.

4.4.3.4. Controller

The CVT ratio control and clamping force control strategies, including the CVT shift dynamics, focus on the following:

- The demanded CVT ratio is determined from the engine best-efficiency line
- The secondary pressure is determined for the given input torque and CVT ratio
- The primary pressure needs to be controlled to meet the demanded CVT ratio

Figure 33 shows a block diagram of the model-based ratio control and plant.

Figure 33. Continuously Variable Transmission System Block Diagram

4.4.4. Dual Clutch Transmission

4.4.4.1. Plant Model

Dual clutch transmission (DCT) dynamic models include the clutch and gear-train, as shown in Figure 34.

Figure 34. Stick Diagram of a Dual-Clutch Transmission (Kim N. L.-B., 2014)

The following assumptions are made in the model development of the shifting process:

- All shafts within the powertrain are assumed to be rigid.
- The synchronization transition process of the synchronizer is ignored.
- The inertia of shaft1 is lumped with that of clutch1 (CL1), and the inertia of shaft2 is lumped with that of clutch2 (CL2).

4.4.4.2. System Operating Conditions

The gear pre-selection can be implemented by considering the DCT operating conditions, as shown in Table 1.

For example, if the first synchronizer is at the first gear position, and the third through fifth synchronizers are at the neutral position (as they must be), then the gear ratio between shaft1 and the output shaft is first gear. At the same time, the gear ratio between shaft2 and the output shaft can be selected in the same manner for the pre-selection mode. To achieve a desired input-output gear ratio, the corresponding synchronizer and clutch have to be applied.

Table 1. Dual-Clutch Transmission Operating Conditions

Dynamic Modeling

The schematic diagram in Figure 35 and the operating conditions in Table 1 were used to generate the equations for each mode, as described below.

Figure 35. Pre-Selection Mode Bond Graph

$$\omega_e = \omega_{s1} = N_{odd} \cdot \omega_{out}$$

$$\omega_{s2} = N_{even} \cdot \omega_{out}$$

$$T_{out} = N_{odd} \cdot \{T_e - (J_e + J_{c1} + J_{odd}) \cdot \dot{\omega}_e\} + N_{even} \cdot \{T_{c2} - T_{loss2} - (J_{c2} + J_{even}) \cdot \dot{\omega}_{s2}\} - J_{out} \cdot \dot{\omega}_{out}$$

J_e : lumped inertia of the engine, $\text{kg} \cdot \text{m}^2$
 J_c : inertia of the driving plates of the clutch, $\text{kg} \cdot \text{m}^2$
 J_{odd} : inertia of the odd geartrain, $\text{kg} \cdot \text{m}^2$
 J_{even} : inertia of the even geartrain, $\text{kg} \cdot \text{m}^2$
 J_{out} : inertia equivalent of the output shaft, $\text{kg} \cdot \text{m}^2$
 N_{odd} : gear ratio of the odd geartrain
 N_{even} : gear ratio of the even geartrain
 T_e : output torque of the engine, $\text{N} \cdot \text{m}$
 T_c : output torque of the clutch, $\text{N} \cdot \text{m}$

The synchronizer of shaft1 is at an odd gear position, and the synchronizer of shaft2 is at an even gear position for the pre-selection. CL1 or CL2 is applied to connect the input power path into shaft1 or

shaft2. Figure 35 shows the bond graph representation of the pre-selection mode when only CL1 is locked.

Figure 36 shows the details of the Autonomie gearbox plant model for the DCT.

Figure 36. Gearbox Plant Model for Dual-Clutch Transmission

The model and the governing equations for the clutch are described below:

If clutching,

$$T_c = T_{slip_max} \cdot Displacement(p_c(t)) \cdot sign(\omega_{in} - \omega_{out})$$

If declutching,

$$i) T_{slip_max} \cdot Displacement(p_c(t)) < T_{in}$$

$$T_c = T_{slip_max} \cdot Displacement(p_c(t)) \cdot sign(\omega_{in} - \omega_{out})$$

ii) otherwise

$$T_c = T_{in}$$

Figure 37. Model and Governing Equations for Clutch

Figure 38 shows the high-level DCT model in Autonomie.

Figure 38. Gearbox Plant Model for Dual-Clutch Transmission

Controller

The controller operating logic was defined based on the principle of DCT operation. The gearbox transient block coordinates all components during transient phases, as shown in Figure 39.

Figure 39. Dual-Clutch Transmission Controller

Gearbox Transient Controller

The component modes (i.e., engine mode, clutch mode, and electric motor mode) and signals of gear demand (odd/even) are generated when a gear shift is required or the engine turns on. The gear selection control (synchronizer control) algorithm is unique to DCT systems (Figure 40). Once the gearshift schedule algorithm generates a new desired gear command, the gear selection control algorithm controls the synchronizer actuators to pre-select the desired gear.

Example of gear shifting : 2nd → 3rd

- Check gear dmd if odd or even?
EV driving or HEV driving?
- Pre-selection : synchronizing 3rd in shaft1
- CL1 clutching / CL2 declutching
- CL1 locked / neutral of shaft2
- Finishing

Figure 40. State Diagram Representation of the Control Strategy for a Dual-Clutch Transmission

Clutch Controller

The clutch controls use time-based clutch position maps, as shown in Figure 41.

Figure 41. Clutch Controller for Dual-Clutch Transmission

4.4.5. Transmission Shifting

There are two parts to transmission shifting logic:

- Shift initializer
 - Shift initializer logic is implemented in the vehicle model. This is where it is decided when to change gears. In addition to the shift maps, there are different thresholds in place (i.e., gear upshift demand validation) to ensure proper drive quality. One of the key parameters of the shifting logic is the shift map. For a given transmission, the shifting logic is the same.

- Shift controller

The shift controller consists of the following:

- The shift map: There are two separate maps, one for upshift and one for downshift. The maps use vehicle speed and pedal position as inputs. Those shift maps are developed for each individual vehicle/powertrain/component combination using a generic shifting algorithm.
- The shifting algorithm: The shifting algorithm uses the component characteristics (e.g., engine BSFC, gear ratios, final drive ratio, wheel radius, etc.) to develop the shift maps. While energy consumption is very important, other drive quality metrics are also taken into account (e.g., avoid low/high vehicle speeds, high torque demands, etc.)

Transmission shifting has a significant impact on vehicle energy consumption and should be carefully designed to maximize the powertrain efficiency while maintaining acceptable drive quality. The logic used in the simulated conventional light-duty vehicle models relies on two components: (1) the shifting

controller, which provides the logic to select the appropriate gear during the simulation, and (2) the shifting initializer, the algorithm that defines the shifting maps (i.e., values of the parameters of the shifting controller) specific to a selected set of component assumptions. While the algorithm used to develop shifting maps is similar across all vehicles, the shifting maps are specific to each individual vehicle, as they are designed using component information such as gear and final drive ratios, engine brake specific fuel consumption (BSFC), etc.

4.4.5.1. Engine Lugging Limits

Engine lugging limit is a critical NVH parameter. The assumptions shown in Table 2 and Table 3 below describe the logic implemented in Autonomie to prevent lugging. The logic and values were developed based on AMTL vehicle test data analysis. Figure 42 shows the engine operating points for 2017 Ford F-150 during an urban dynamometer driving schedule (UDDS) cycle (Lohse-Busch, July 2018).

Figure 42. Engine Operating Points for 2017 Ford F150 (UDDS)

To ensure consistency with the different engine technologies using the vehicle test data, a distinction is made between the naturally aspirated and turbo-charged engines in terms of engine lugging limits across different geared transmissions.

Shift parameters are selected such that low-speed, high-torque operation is avoided. The selected shifting limits are based on test data observations relative to the number of gears available.

Table 2. Engine Lugging Limits for Different Transmissions (NA Engines)

	5-speed	6-speed	7-speed	8-speed	9-speed	10-speed
Lugging speed (rad/s)	140	130	120	110	110	110
Lugging speed (RPM)	1337	1241	1241	1050	1050	1050

Table 3. Engine Lugging Limits for Different Transmissions (Turbo Engines)

	5-speed	6-speed	7-speed	8-speed	9-speed	10-speed
Lugging speed (rad/s)	140	130	130	130	130	130
Lugging speed (RPM)	1337	1241	1241	1241	1241	1241

Figure 43 shows an example of how engine operating conditions are restricted to prevent lugging for 5-speed and 10-speed automatic transmissions on the UDDS cycle for naturally aspirated engine.

Figure 43. Example of Engine Operating Conditions for 5-Speed and 10-Speed Transmissions on UDDS Cycle for NA Engine

4.4.5.2. Shifting Initializer

The shifting controller uses shifting maps to compute the gear command. In the controller, the shift map is a two-dimensional (2D) lookup table indexed by vehicle speed and accelerator-pedal position. Defining such a map is equivalent to defining the “boundaries” of each gear area; those boundaries are the shifting speeds. Figure 44 illustrates that equivalence.

Figure 44. Upshifting Gear Map (Left); Upshifting Vehicle Speeds (Right)

For each shifting curve, there are two key points: the “economical” shifting speed (at very low pedal position) and the “performance” shifting speed (at high pedal position). The objective of the control engineer is to combine both goals of the shifting control to fulfill the driver’s expectations: minimization of energy consumption on the one hand and maximization of vehicle performance on the other.

The economical shifting speed for an upshift or a downshift is the speed at which the upshift/downshift occurs when the accelerator pedal is very lightly pressed: $V_{eco}^{k \rightarrow k+1}$ is the economical vehicle speed for upshifting from gear k to gear $k+1$, and $V_{eco}^{k+1 \rightarrow k}$ is the downshifting speed for this same set of gears. The vehicle speed shift points are computed from the engine shift points $\omega_{eco}^{k \rightarrow k+1}$ and $\omega_{eco}^{k+1 \rightarrow k}$. Figure 45 shows the engine speed shift points for an engine associated with a 5-speed transmission.

Figure 45. Example of Engine Speed Range in Economical Driving and Economical Shift

The initializing algorithm for the shifting controller computes the upshifting and downshifting speeds at zero pedal position based on the four “extreme” shift points: upshifting from lowest gear ($\omega_{eco}^{1 \rightarrow 2}$), upshifting into highest gear ($\omega_{eco}^{N-1 \rightarrow N}$), downshifting into lowest gear ($\omega_{eco}^{2 \rightarrow 1}$), and downshifting from highest gear ($\omega_{eco}^{N \rightarrow N-1}$). N is the number of gears. The speeds can be set by the user or left at their default values. Below is a description of their default values in Autonomie:

$$\omega_{eco}^{2 \rightarrow 1} = \omega_{idle} + \omega_{margin} [\omega_{idle}: \text{engine idle speed}; \omega_{margin}: \text{speed margin, } \approx 50\text{--}100 \text{ rpm}]$$

$$\omega_{eco}^{1 \rightarrow 2} = \omega_{idle} \frac{k_1}{k_2} (1 + \epsilon_{ud}) [k_1, k_2: \text{gear ratios for gears 1,2}; \epsilon_{ud}: \text{margin to avoid overlap, } \approx 0.05\text{--}0.1]$$

$$\omega_{eco}^{N-1 \rightarrow N}: \text{Engine speed at which best efficiency can be achieved}$$

$$\omega_{eco}^{N \rightarrow N-1} = \omega_{eco}^{N-1 \rightarrow N} - \omega_{\Delta} [\omega_{\Delta} \approx 1,000 \text{ rpm}]$$

Once those four speeds are computed, the remaining ones are computed by linear interpolation to allow consistent shifting patterns that are acceptable to the driver. For example, any upshifting speed is given by Equation 1:

$$(1) \omega_{eco}^{i \rightarrow i+1} = \frac{\omega_{eco}^{N-1 \rightarrow N} - \omega_{eco}^{1 \rightarrow 2}}{N-2} \cdot (i-1) + \omega_{eco}^{1 \rightarrow 2}, \quad 1 \leq i \leq N-1$$

In a shifting map, the vehicle upshifting speed from gear i to $i+1$ must be higher than the downshifting speed from gear $i+1$ to i . Otherwise, the downshifting speed will always request gear i while gear $i+1$ is engaged and vice-versa, resulting in oscillations between gears that would be unacceptable to the driver. For this study, the algorithm in the initialization file prevents that by making sure the following relation is true:

$$(2) \omega_{eco}^{i \rightarrow i+1} > \omega_{eco}^{i+1 \rightarrow i} \cdot \frac{k_1}{k_2} (1 + \epsilon_{ud}), \quad 1 \leq i \leq N-1$$

The values of the engine economical shifting speeds at lowest and highest gears are automatically defined on the basis of the engine and transmission characteristics.

Finally, the vehicle economical up- and downshifting speeds can be computed using the engine up- and downshifting speeds, the gear ratio, the final drive ratio and the wheel radius:

$$(3) V_{eco}^{i \rightarrow i+1} = \frac{\omega_{eco}^{i \rightarrow i+1}}{k_i k_{FD}} \cdot R_{wh}$$

where k_{FD} is the final drive ratio and R_{wh} is the wheel radius.

During performance, the gears are automatically selected to maximize the torque at the wheel. Figure 46 illustrates that gear selection, which consists of finding the point where the curve of engine peak torque (reported at the wheels) at gear k falls under the curve at gear $k+1$.

Figure 46. Maximum Engine Torque at Wheels and Performance Upshift Speeds

The performance downshifting speed is given by the performance upshifting speed and the difference between the economical shifting speeds:

$$\Delta V_{perf}^i = \alpha_{pf,ec} \cdot \Delta V_{eco}^i \Leftrightarrow V_{perf}^{i \rightarrow i+1} - V_{perf}^{i+1 \rightarrow i} = \alpha_{pf,ec} \cdot (V_{perf}^{i \rightarrow i+1} - V_{perf}^{i+1 \rightarrow i}) \quad (4)$$

The definition of the final shifting curves is critical to properly evaluating the benefits of technologies while maintaining acceptable performance. Figure 47 shows how a set of upshifting and downshifting curves for two adjacent gears is built, based on selected vehicle speeds and accelerator pedal positions. At low pedal positions (i.e., below a_{eco}^{up}), the upshifting speed is the economical upshifting speed. Similarly, below a_{eco}^{dn} , the downshifting speed is the economical downshifting speed. This approach ensures optimal engine operating conditions under gentle driving conditions. At high pedal positions (i.e., above a_{perf}), the shifting speed is the performance shifting speed, ensuring maximum torque at the wheels under aggressive driving conditions.

Figure 47. Design of Upshifting and Downshifting Speed Curves for Two Adjacent Gears

4.4.5.3. Traditional Gear Shifting Controller

The shifting controller determines the appropriate gear command at each simulation step using the shifting maps developed by the shifting initializer. A simplified schematic of the controller is shown in Figure 48. The letters and numbers in the discussion that follows correspond to those shown in circles in Figure 48.

Figure 48. Shifting Controller Schematic in Autonomie.

The controller is based on two main shifting maps — one for upshifting (a) or moving from a lower gear to a higher gear, and another one for downshifting (b) or moving from a higher gear to a lower gear — as well as a state-machine (c) that defines the status of the system (e.g., no shifting, upshifting). Each shifting map outputs a next-gear command $\gamma_{dn}(t)$ and $\gamma_{up}(t)$ based on the current accelerator pedal position $a(t)$ and vehicle speed $V(t)$. The state machine is composed of different states, of which only one is active at any time step; a change in state occurs whenever a transition condition from the active state becomes true (i.e., an upshift will occur only if a set of conditions is true). The state that is active most of the time is the hold-gear state (d), which makes sense because, most of the time, for drivability reasons, the vehicle should be in gear and not shifting. An upshift occurs when the upshifting gear $\gamma_{up}(t)$ is strictly higher than the current gear $\gamma(t)$ (1) (e.g., $\gamma_{up}(t) = 5$ and $\gamma(t) = 4$). For all vehicles, the shift does not necessarily happen instantly when the command to shift is given, depending on the current pedal position. In aggressive driving, i.e., at high accelerator-pedal positions (5), the shift happens as soon as the gear transition (1) becomes true, ensuring optimal performance. In contrast, in

“normal” driving, i.e., at low pedal positions (2), there is an intermediate state (e) that allows the shift only when the gear condition (1) is true for a minimum time τ . This constraint is imposed to avoid an excessive number of shifting events, which would lead to unacceptable drive quality and increased energy consumption. The upshifting itself is executed in state (f), in which the shift command $\gamma_{cmd}(t)$ is incremented (i.e., the next upper gear is selected); once the shifting is completed (6), the state machine comes back to the hold-gear state (d). Downshifting occurs in a similar way.

As an additional level of robustness in the Autonomie control algorithm, an upshift or downshift cannot occur if the resulting engine speed would be too low or too high, respectively. This approach ensures that the engine is not operated below idle or above its maximum rotational speed, as shown in Figure 49.

Figure 49. Shifting Calculations in Autonomie

4.4.5.4. Torque Control During Shifting Events for Automatic Transmission

Figure 50 shows the transmission clutch pressure, output torque, and engine speed curves during a change from first to second gear. The output torque experiences both a trough period (lower than the torque in the original gear) and a crest period (higher than the torque in the original gear). The trough period is called a torque hole, while the crest period is called a torque overshoot. The torque hole is defined by depth and width, where the depth is the difference between minimum torque and the torque in previous gear and the width is the half value of the maximum width of the torque hole.

Figure 50. Shift Process for Automatic Transmission

The bigger the torque hole, the larger the decrease of torque in torque phase, resulting in a more significant reduction in acceleration. Because the decrease in acceleration causes discomfort for both the driver and passengers, the torque hole should be as shallow and narrow as possible. Torque reduction behavior is a well-known phenomenon, observed during vehicle testing and referenced in several papers and presentations.

Autonomie integrates a low-level control algorithm that reproduces the torque hole phenomenon. Figure 51 illustrates, in detail, the behavior of the vehicle model for a short period of time (205 sec to 205.8 sec). The area highlighted by the oval outline indicates the torque hole during a shifting event.

Figure 51. Torque Hole in Autonomie During Shifting Event

4.4.5.5. Gear Skip-Shifting Strategy

For the final rule-making (FRM) runs, a gear skip-shifting strategy has been developed and implemented for 10-speed transmissions using the Ford F-150 10-speed benchmarking (Lohse-Busch, July 2018).

Table 4 below is presented in the report outlined for F-150 benchmarking:

Table 4. Detailed Gear-Shifting Events for Ford F-150 (Lohse-Busch, July 2018)

# of shifts	1-2	1-3	2-3	2-4	3-4	3-5	4-5	4-6	5-6	6-7	7-8	8-9	9-10	Total
UDDS		18			14	5	16	1	21	12	4	1	2	94
Highway		1			1		1		1	2	2	5	5	18
US06	1	6		1		7	1		9	5	7	5	11	53

From the table, a fairly consistent gear-skipping can be observed for the drive cycles considered in the study. A similar gear-skipping method was implemented for this analysis. Table 5 details the shifting events for the different cycles in simulation of Pickup-NonPerfo/Micro Hybrid/eng12/MR0/AERO0/ROLL0.

Table 5. Argonne Simulated Gear-Shifting Events for 10-Speed Transmission

Cycles	1-2	1-3	2-3	3-4	3-5	4-5	5-6	6-7	7-8	8-9	9-10	Total
UDDS		18			18		21	12	1	1	1	73
Highway		1			1		1	2	2	4	3	13
US06		7			8		8	7	5	6	4	46

4.4.6. Engine Start Control for Pre-Transmission HEV

The vehicle speed, engine speed, and electric motor torque for engine start-up are compared with the vehicle test results for t=21–24 sec in Figure 52. The single clutch (CPL2) is located inside the electric machine, between the engine and the electric machine. The clutch is engaged when starting the internal combustion engine (ICE).

Figure 52. Engine Start Transient Control Using Electric Machine

The electric machine torque is also controlled to quickly synchronize the clutch input and output speed. The controller reacts by setting a torque-increasing intervention that is added to the torque of the electric machine in the speed increase phase.

4.5. Torque Converter

A torque converter is a hydrodynamic fluid coupling used to transfer rotating power from a prime mover, such as an ICE, to a rotating driven load. It consists of an impeller (drive element), a turbine (driven component), and a stator (a set of stationary windings that generate a magnetic field), which assists the torque converter function. The torque converter is filled with oil and transmits the engine torque by means of the flowing force of the oil. The device compensates for speed differences between the engine and the other drivetrain components and is therefore ideally suited for the start-up function.

The torque converter is modeled as two separate rigid bodies when the coupling is unlocked and as one rigid body when the coupling is locked. The downstream portion of the torque converter unit is treated as being rigidly connected to the drivetrain. Therefore, there is only one degree of dynamic freedom, and the model has only one integrator. This integrator is reset when the coupling is locked, which corresponds to the loss of the degree of dynamic freedom. Figure 53 shows the efficiency of the torque converter used for the study.

The effective inertias are propagated downstream until the point where actual integration takes place. When the coupling is unlocked, the engine inertia is propagated up to the coupling input, where it is

used for calculating the rate of change of the input speed of the coupling. When the coupling is locked, the engine inertia is propagated all the way to the wheels.

Figure 53. Typical Torque Converter Efficiency

Figure 54 describes the conditions under which the torque converter will be locked. The same algorithm is used to represent current torque converter lockup logic as well as future aggressive lockup logic. The torque converter is used as a start-up device in first gear, with very low slip (torque ratio of 0.95) at higher speeds in first gear. Recent trends in torque-converter technology suggest operation in locked or controlled-slip mode in second and higher gears. In general, the torque converter is in controlled slip mode or mechanically locked, depending on vehicle speed and pedal position, for each gear apart from the first. To suggest advances in torque-converter technology, it was assumed that the torque converter would be in a mechanically locked state for the second and higher gears. This approach has been applied to all transmissions with six gears or more. The logic does not change between different automatic transmissions types (AU/AU+/AU++).

Figure 54. Torque Converter Lockup Control Algorithm

4.5.1. Plant Model

The torque converter is modeled as two separate rigid bodies when the coupling is unlocked and as one rigid body when the coupling is locked. The downstream portion of the torque converter unit is treated as being rigidly connected to the drivetrain. Therefore, there is only one degree of dynamic freedom, and the model has only one integrator. Figure 55 shows the main I/O of the torque converter model.

Figure 55. Autonomie Torque Converter Model I/O

As with the other components, the effective inertias are propagated downstream until the point where actual integration takes place. When the coupling is unlocked, the engine inertia is propagated up to the coupling input, where it is used to calculate the rate of change of the input speed of the coupling. When the coupling is locked, the engine inertia is propagated to the wheels.

The torque converter model is based on a lookup table, which determines the output torque depending on the lockup command. The upstream acceleration during slip and the downstream acceleration are taken into account in calculating the output speed.

4.5.2. Operational Modes

The different operational modes are described below.

- **Mode 1: Idle**
We enter this mode if $W_{in} < W_{eng_idle} + \text{Threshold}$. We quit the mode if $W_{in} > W_{eng_idle} + \text{Threshold}$ and T_{in} is increasing and positive.
- **Mode 2: Acceleration (transient calculation)**
We are in this mode if $T_{in} > 0$ and $\text{Speed_Ratio} < 0.1$ (meaning $W_{in} \gg W_{out}$).
- **Mode 3: Steady state**
We are in this mode if $[W_{in} > W_{eng_idle} + \text{Threshold} \text{ and } W_{in} \text{ close to } W_{out}] \text{ OR } \text{Speed Ratio} > 0.8 (W_{in} \text{ close to } W_{out}) \text{ OR } T_{in} < 0$.
- **Mode 4: Locked**
We enter this mode if the torque converter command is 1.

4.5.2.1. Steady-State Calculation

Figure 56 details the steady-state operational mode calculation of the torque converter model.

Figure 56. Steady-State Mode Calculation

$$(1) T_{impeller} = T_{in} - [K_{in} \cdot \omega_{impeller}^2 \cdot sign(\omega_{impeller})]$$

$$(2) T_{turbine} = T_{ratio} \cdot K_{in} \cdot \omega_{impeller}^2 \cdot sign(\omega_{impeller})$$

$$(3) K_{in} = \frac{1}{capacity_factor^2} = f(\omega_{ratio})$$

$$T_{ratio} = f(\omega_{ratio})$$

$$\omega_{impeller} = \int \frac{T_{impeller}}{J_{impeller}}$$

$$\omega_{ratio} = \frac{\omega_{turbine}}{\omega_{impeller}}$$

4.5.2.2. Transient Calculation

Figure 57 details the transient operational mode calculation of the torque converter model.

Figure 57. Transient Mode Calculation

$$(1) T_{turbine} = (T_{in} - T_{impeller}) \cdot T_{ratio}$$

$$(2) T_{impeller} = |T_{in}| \cdot \frac{\omega_{goal} - \omega_{impeller}}{gain}$$

$$(3) \omega_{goal} = capacity_factor \cdot T_{in}^{0.5}$$

$$(4) capacity_factor = \frac{\omega_{impeller}}{\sqrt{T_{impeller}}} = f(\omega_{ratio})$$

$$\omega_{impeller} = \int \frac{T_{impeller}}{J_{impeller}}$$

4.5.2.3. Idle Calculation

Figure 58 details the idle operational mode calculation of the torque converter model. Table 3 details the list of initialization parameters for the torque converter model in Autonomie.

Figure 58. Idle Mode Calculation

$$(1) T_{impeller} = T_{in} - T_{turbine}$$

$$(2) T_{turbine} = T_{in} \cdot \frac{\omega_{impeller} - \omega_{engine_idle}}{gain}$$

$$\omega_{impeller} = \int \frac{T_{impeller}}{J_{impeller}}$$

Table 6. Initialization Parameters

$$capacity_factor(k) = \frac{\omega_{impeller}}{\sqrt{T_{impeller}}} = f(\omega_{ratio})$$

Name	Unit	Description	Size
cpl.plant.init.trq_impeller.idx1_spd	rad/s	Impeller speed index used to index impeller torque map; used in calculating the "k" factor	1*n
cpl.plant.init.trq_impeller.map	Nm	Impeller torque map indexed by impeller speed index; used in calculating the "k" factor	1*n
cpl.plant.init.k_factor.idx1_spd_ratio	-	Speed ratio vector used to index the torque ratio map or k factor	1*n
cpl.plant.init.k_factor.map	(rad/s)/ sqrt(Nm)	Torque converter "k" value calculated from impeller speed index and impeller torque map used in the transient calculation block to calculate the desired impeller speed	1*n

4.6. Energy Storage Model

The energy storage system block models the battery pack as a charge reservoir and an equivalent circuit. The equivalent circuit accounts for the circuit parameters of the battery pack as if it were a perfect open-circuit voltage source in series with an internal resistance and two resistor-capacitor (RC) circuits that represent the polarization time constants. Figure 59 shows the main I/O of the energy storage model.

Figure 59. Autonomie Energy-Storage Model I/O

Autonomie includes several energy storage models for use in various high power and high energy applications:

- High-power application: One battery model is used as a charge reservoir. An equivalent circuit, whose parameters are a function of the remaining charge in the reservoir (also known as the state of charge [SOC]), accounts for circuit parameters of the battery pack as if it were a perfect open-circuit voltage source in series with an internal resistance.
- High-energy application: Another battery model in Autonomie is the one used for high-energy batteries. This model uses two time constants to represent the polarization behavior of the battery pack. This lumped parameter model can represent many different battery chemistries for internal resistances, capacitances, and open-circuit voltages, which are all maps based on SOC and, in some cases, temperature.

Another important aspect to consider for sizing is the pulse power limits of the battery pack. There are several different options to represent the maximum power of the battery. The main one represents maximum power as a function of SOC. Other models introduce a time constraint for the maximum power. These battery packs have different power limits for 10-second, 2-second, and continuous power. The model accounts for the duration of the pulse and limits power accordingly. This aspect is not necessarily a feature of the plant, but is handled by the low-level control and is dependent on the battery chemistry and plant's performance characteristics.

4.6.1. High-Power Battery Model (Used for Hybrid Electric Vehicles)

Figure 60 shows the top-level diagram of the high-power battery model.

Figure 60. High-Power Battery Model Diagram in Autonomie

The top-level diagram of the high-power model consists of three main blocks: the voltage calculation (Block A); the current calculation (Block B); and the SOC calculation (Block C).

Block A (Figure 61) calculates the output voltage by taking the open-circuit voltage, which depends on SOC, and subtracting three terms: the voltage drop due to the internal resistance of each cell, the voltage drop due to the polarization resistance of each cell, and the voltage drop due to an effective series capacitance, modeling the variation of output voltage with the time integral of the current.

Figure 61. Block A: High-Power Battery Model Output Voltage Calculation

The polarization current causes an additional voltage drop at the terminals of the battery. As current continues to flow in one direction through the cell, this voltage drop increases. As the ions migrate in a lithium-ion cell, it takes more work to keep the current flowing in the same direction. However, when the current reverses, the migrated ions facilitate current flow in the reverse direction.

$$(1) V_{out} = V_{oc} - OCV \times \int I_{load} \cdot dt - R_0 \times I_{load} - R_p \times I_p$$

where

I_{load} = current flowing into the load, that is, the input current from the voltage bus

OCV = series capacitance of the battery modeling the variation of output voltage with the time integral of the current

R_0 = series internal resistance of a cell

R_p = polar resistance of a cell

I_p = polar current in a cell

In Block B (Figure 62), the polarization current is calculated by solving differential Equation 2.

Figure 62. Block B: High-Power Battery Model Current Calculation

$$(2) \tau_p(SOC) \cdot \frac{dI_p}{dt} + I_p = I_{load}$$

In Block C (Figure 63), the SOC for the battery is calculated. If the SOC drops too low, the stop block automatically stops the simulation.

Figure 63. Block C: High-Power Energy Model SOC Calculation

$$(3) SOC = SOC_{init} + \Delta SOC$$

$$(4) \Delta SOC = - \int \frac{I_{in}}{Capacity_{max}}$$

where

SOC_{init} = initial SOC (initial value of the integrator)

I_{in} = input current into the battery from the bus

$Capacity_{max}$ = maximum charge capacity

The SOC is calculated by determining the variation of charge in the battery and dividing it by the maximum coulombs the battery can store. A value of 0 is the unattainable state of having no charging remaining in the battery, while a value of 1 is the unattainable state of having a perfectly charged battery. Attempting to reach either of these values in practice would damage the battery and result in a short life cycle.

In practice, different values of minimum and maximum SOC values are used depending on the battery chemistry and applications.

4.6.2. High Energy Battery Model (Used for PHEVs and Battery Electric Vehicles)

Unlike the high-power battery model, the high-energy battery model utilizes two polarization resistances. The top-level diagram of the high-energy battery model in Figure 64 consists of three main blocks: the voltage calculation (Block A), the current calculation (the two B blocks correspond to the two current polarizations), and the SOC calculation (Block C).

Figure 64. High-Energy Battery Model Diagram in Autonomie

4.7. Electric Machine Models

The electric machine transforms electrical power into mechanical power by creating a magnetic field that applies a force to current-carrying conductors. Electric machines that operate on this principle can be divided into two main categories: DC electric machines and AC electric machines. DC electric machines can be further divided into electric machines with and without brushes. AC electric machines can also be divided into two main categories: synchronous and asynchronous. The AC electric machine categories can be further subdivided based on the number of phases. Figure 65 shows the main I/O of the electric machine model in Autonomie.

Figure 65. Autonomie Electric-Machine Model I/O

All electric machines consist of two major parts: the stator, which is a set of stationary windings that generate a magnetic field that encompasses the second part, the rotor or armature, which is the rotating part.

Electric machine plant models in Autonomie use torque or power as the command and produce a torque output. The electric machine operating speed is determined by components connected to the electric machine. In a vehicle, the vehicle speed and gear ratios determine the electric machine operating speed. The lookup table used in an electric machine model estimates operational losses over the entire operating region.

Typically, an electric machine has a continuous operating region (a region under the continuous torque curve) and a transient region where the electric machine can operate for a short period of time. The peak torque capability of an electric machine is defined for a specific duration, e.g., 30 seconds. The maximum torque output gets de-rated to continuous torque levels when the electric machine's temperature increases. The electric machine model in Autonomie has this general logic built into it.

Figure 66 shows the general map based electric machine model used in Autonomie. It has three essential maps:

- Continuous torque as a function of speed
- Maximum torque as a function of speed
- A four-quadrant efficiency map as a function of speed and torque

A warm-up time constant is used to interpolate between the maximum and continuous torque curves of the electric machine. The maximum-torque curve and efficiency map do not depend on the electric machine input voltage. Except for the fuel-cell-only configuration, this is the default electric machine model used in all of configurations.

Figure 66. Autonomie Electric Machine Model in Simulink

Equation 1 calculates the electric machine current:

$$(1) I_{in} = \frac{P_{electrical}}{V_{in}}$$

Equation 2 computes the electric machine output torque by using the electric machine command and the maximum available torque of the electric machine at a given temperature and speed.

$$(2) T_{out} = T_{max} \cdot PWM_{cmd}$$

Figure 67 shows the diagram of block A for the max torque calculation of the electric machine model.

Figure 67. Block A: Electric Machine Model Max Torque Calculation

Equation 3 interpolates between the continuous-torque curve and the maximum-torque curve by using the heat index. If the electric machine is hot, then the continuous-torque curve is used (that is, the heat

index is 1). If the electric machine is at its operating temperature, then the maximum-torque curve is used.

$$(3) T_{max_{mechanicl}} = T_{cont} \cdot PWM_{heatindex} + T_{Peak} \cdot (1 - PWM_{heatindex})$$

Equations 4 through 8 show the dependence of the electric machine torque curves on electric machine speed.

$$(4) T_{cont} = T_{cont_{prop}} = function(W_{in})$$

$$(5) T_{Peak} = T_{peak_{prop}} = function(W_{in})$$

$$(6) T_{cont} = T_{cont_{regen}} = function(W_{in})$$

$$(7) T_{peak} = T_{peak_{regen}} = function(W_{in})$$

$$(8) T_{max_{electrical}} = function(W_{in}, P_{electrical})$$

Figure 68 shows the diagram of block A1 for the heat index calculation of the electric machine model.

Figure 68. Block A1: Heat Index

Equation 9 computes the heat index of the electric machine.

$$(9) PWM_{Heatindex} = -0.3 + \int \frac{0.3}{\tau} \cdot \left(\frac{T_{out}}{T_{cont}} - 1 \right) \cdot dt$$

4.8. Electrical Accessories Model

Electrical accessories, such as lamps, radiator fans, and wipers, obtain their energy from an electrical source. They represent dedicated auxiliary load systems.

The plant model of an electrical accessory loss uses a constant power draw. The model considers the electrical losses associated with the powertrain. The current losses are taken from the energy storage. While the accessory load varies under real-world driving conditions, it is fairly constant during standard driving cycles. Therefore, the approach of constant power draw is valid for the study. Figure 69 shows the main I/O of the electrical accessories model.

Figure 69. Autonomie Electrical Accessories Model I/O

Figure 70 shows the Simulink diagram for the electrical accessories model.

Figure 70. Top-Level Simulink Diagram of Electrical Accessories Model

Two equations govern the electrical-accessories plant model:

$$(1) I_{in} = I_{out} + \frac{P_{loss}}{V_{in}}$$

$$(2) V_{out} = V_{in}$$

The current drawn by the accessory load is the constant power loss constant divided by the bus voltage. As shown in Equation 1, the total current flowing in is the sum of the current sunk by the accessory load and the output current propagated back to this model.

Section 5.9 details the constant auxiliary loads used for this analysis.

4.9. Driver Model

The driver model uses a look-ahead controller to model the accelerator and brake pedals. The desired vehicle speed is compared with the current speed, and a controller is used to request more or less torque to the vehicle. Figure 71 shows the main I/O of the driver model.

Figure 71. Autonomie Driver Model

The look-ahead model estimates the additional torque needed to meet the speed trace in the next Δ seconds in order to provide a more realistic model of driver behavior. This approach avoids unrealistic high torque demands during acceleration. Figure 72 shows the top-level diagram of the driver model in Autonomie.

Figure 72. Top-Level Simulink Diagram of the Look-Ahead Driver Model

Figure 73 describes the equations involved in the look-ahead driver model.

$$T_{\Delta t}(t) = m \cdot \frac{r(t + \Delta t) - \omega(t)}{\Delta t} + m \cdot \dot{r}(t + \Delta t) + L(r(t + \Delta t))$$

ω : actual wheel speed
 r : reference wheel speed (trace)

Torque demand (points to $T_{\Delta t}(t)$)
 Estimation of additional torque needed to reach $r(t+\Delta t)$ (points to $m \cdot \frac{r(t + \Delta t) - \omega(t)}{\Delta t}$)
 Acceleration (points to $m \cdot \dot{r}(t + \Delta t)$)
 Vehicle losses (points to $L(r(t + \Delta t))$)
 Needed to meet trace @ $(t+\Delta t)$ (points to the entire equation)

Figure 73. Equations for the Look-Ahead Driver Model

4.10. Wheel Model

The wheel model accounts for the braking force at each wheel and the added inertia to the drivetrain of all the wheels.

Figure 74 shows the main I/O of the Autonomie wheel model.

Figure 74. Autonomie Wheel Model I/O

As shown in Figure 75, the wheel model can be divided into three subsystems: wheel angular speed calculation (Block A), output force calculation (Block B), and wheel mass calculation (Block C).

Figure 75. Autonomie Wheel Model Simulink Diagram

The wheel angular speed is calculated by dividing the linear speed of the vehicle by the radius of the wheel.

$$(1) W_{in} = \frac{V_{veh}}{K_{Radius}}$$

There is no slip calculation in the wheel model.

Block B (Figure 76) implements the wheel-force calculation.

Figure 76. Block B: Force Calculation

The wheel net torque is converted into a force by dividing by the wheel radius:

$$(2) F_{out} = \frac{T_{out}}{K_{Radius}}$$

The wheel net torque equals the torque in from the drive axle minus the brake torque minus the rolling resistance losses:

$$(3) T_{out} = T_{in} - T_{loss_{brakes}} - T_{loss_{tire}}$$

The maximum brake torque is calculated by multiplying the brake command from the controller by the maximum available brake torque. The maximum available brake torque is assumed to be a constant and is unchanged across different vehicle powertrains and classes.

$$(4) T_{loss_{brake}} = PWM_{brake} \cdot T_{max_{brake}}$$

The wheel losses are modeled by a fourth-degree polynomial that is a function of vehicle speed:

$$(5) T_{loss_{tire}} = \left[\cos(\alpha_{grade}) * (C_1 + C_2 * V_{veh} + C_3 * V_{veh}^2 + C_4 * V_{veh}^3) * M_{veh} * g * \eta_{weight_{traction}} * f\left(\frac{\omega}{\omega_{threshold}}\right) \right] * R_{wheel}$$

The powertrain rotating inertia is converted into an equivalent mass using the wheel radius, as seen in Figure 77.

Figure 77. Block C: Mass Calculation

The powertrain upstream inertia is added to the wheels inertia and then divided by the wheel radius squared to yield the equivalent powertrain mass.

$$(6) Mass_{inertia} = \frac{J_{in} + J_{wh}}{K_{Radius}^2}$$

4.11. Chassis Model

The chassis model calculates speed based on the fed-forward upstream inertia of the drivetrain, the chassis mass, and the drag and grade losses. Figure 78 shows the main chassis model I/O.

The equivalent chassis mass is the sum of the static chassis mass and the equivalent powertrain mass. The powertrain inertia is converted into an equivalent mass in the wheel model by dividing by the square of the wheel radius.

$$(4) \text{Mass}_{vehicle} = \text{Mass}_{staticframe} + \sum J$$

Block A (Figure 80) calculates the aerodynamic drag and grade loss.

Figure 80. Block A: Chassis Losses

The total chassis losses are the sum of the aerodynamic losses and the grade losses.

$$(5) F_{loss} = F_{aerodynamics} + F_{grade}$$

The grade loss is computed when the grade angle and the weight of the chassis are known. The grade angle is the inclination that the chassis makes with the horizontal (parallel to the ground).

$$(6) F_{grade} = \text{Mass}_{vehicle} \cdot g \cdot \sin(\text{grade})$$

The standard equation for aerodynamic drag is used. The sine function is used to ensure that the aerodynamic drag opposes chassis movement. The chassis should never move backwards in simulations, so this function is not necessarily needed.

$$(7) F_{aerodynamics} = -Kh \cdot \rho_{air} \cdot c_d \cdot f_a \cdot V_{veh}^2 \cdot \text{sign}(V_{veh})$$

Equation 8 shows the variation of air density with altitude, due to the compressibility of air. This equation is not currently implemented in the chassis model.

$$(8) \quad Kh = \frac{\rho}{2 \cdot g} = \begin{cases} 0.6 @ 200 \text{ meters} \\ 0.57 @ 500 \text{ meters} \\ 0.5 @ 1500 \text{ meters} \end{cases}$$

Equation 9 expresses air density as a function of air pressure and temperature. Like Equation 8, it is not currently implemented in the chassis model.

$$(9) \quad \rho_{air} = 1.225 \cdot \frac{Pr}{101.325} \cdot \frac{288.16}{273.16 + T_r}$$

To estimate the frontal area of a chassis from track width and chassis height, Equation 10 is used:

$$(10) f_a = 0.9 \cdot Width_{veh} \cdot Height_{veh}$$

4.12. Final Drive Model

Final drive gears are incorporated in vehicle driving axles and transaxles for the following reasons:

- To provide a right-angled drive from either the propeller shaft or the gearbox layshaft to the driven wheels.
- Alternatively, to provide a parallel drive from the gearbox layshaft to the driven wheels.
- To permit an additional and constant gear reduction in the transmission system.

The second requirement applies only in the case of passenger cars with transversely mounted engines (i.e., lying parallel to the axis of the driven front or rear wheels).

Two basic types of final drive reduction gearing have generally been used in vehicles with longitudinally mounted engines: bevel drive and worm drive. In the modern vehicle, the bevel drive has proven to be lighter, more efficient, less expensive, and equally quiet-running. For cars with transversely mounted engines, the final drive reduction gears are of the helical pinion type.

Figure 81 shows the main final drive model I/O.

Figure 81. Autonomie Final Drive Model I/O

Figure 82 shows the top-level Simulink diagram of the final drive model in Autonomie.

Figure 82. Simulink Diagram of the Final Drive Model in Autonomie

The top-level diagram of the final drive model is divided into three parts, highlighted in red above. Block A calculates the input and output speeds of the final drive, as shown in Figure 83. Block B calculates the output torque, as shown in Figure 84, and Block C calculates the inertia that is fed forward, as shown in Figure 85.

The input speed is the product of the final drive ratio and the output speed of the final drive. The output speed of the final drive is calculated downstream and fed back.

Figure 83. Block A: Final Drive Model Speed Calculation

$$(1) W_{in} = W_{out} \cdot K_{finaldrive}$$

where

W_{in} = final drive speed in

W_{out} = final drive speed out

K_{final_drive} = reduction gear ratio for the final drive

Block B (Figure 84) determines the torque output of the final drive by first subtracting the torque loss, which is a function of input torque and input speed, and then multiplying by the final drive ratio. The blending block ensures that at zero speed the torque loss is zero; otherwise, a negative torque at zero vehicle speed would lead the vehicle model to give the erroneous result of a negative vehicle acceleration and, consequently, a negative vehicle speed. Essentially, the vehicle model cannot discriminate between negative loss torque and negative propulsive torque.

Figure 84. Block B: Torque Calculation

$$(2) T_{out} = K_{finaldrive}(T_{in} - T_{loss})$$

$$(3) T_{loss} = function(W_{in}, T_{in})$$

where

T_{out} = final drive output torque speed

W_{in} = final drive input shaft speed

T_{in} = final drive input shaft torque

T_{loss} = final drive input torque loss

K_{final_drive} = reduction gear ratio of the final drive

Block C (Figure 85) and Equation 4 reflect the product of the drivetrain inertia upstream of the final drive using the final drive ratio into its equivalent inertia at the output of the final drive. This equivalent inertia is fed forward on a path to the vehicle model, where the lumped drivetrain inertia is used to calculate the vehicle speed. Drivetrain speeds are then fed back through the drivetrain model.

Figure 85. Block C: Inertia Calculation

$$(4) J_{out} = J_{in} \cdot K_{finaldrive}^2 + J_{FD}$$

where

J_{in} = final drive input inertia from upstream components

J_{out} = final drive output inertia fed forward to downstream components

J_{FD} = inertia of the final drive

K_{final_drive} = reduction gear ratio of the final drive

4.13. Vehicle-Level Control Algorithms

All the vehicle-level control algorithms used in the study have been developed from vehicle test data collected at AMTL. It is important to note that while the logic for the vehicle-level control algorithms was developed based on test data, only the logic has been used for the present study, since the calibration parameters have been adapted for each vehicle to ensure fuel consumption minimization with acceptable drive quality (i.e., acceptable number of engine on/off conditions).

4.13.1. Micro- and Mild HEV

The vehicle-level control strategies of the micro- and mild (i.e., BISG) HEVs are similar in many aspects because of the low peak power and energy available from the energy storage system.

For the micro-HEV case, the engine is turned off as soon as the vehicle is fully stopped and restarted as soon as the brake pedal is released. No regenerative braking is considered in this controller.

For the mild HEV cases, the engine is also turned off as soon as the vehicle is fully stopped. However, since some regenerative braking energy can be recovered, the vehicle is propelled by the electric machine during vehicle launch, allowing the engine to be restarted later. The electric machine also provides some limited assist during propelling to improve engine efficiency.

4.13.2. Power-Split HEV (Split HEV)

As shown in Figure 86, power-split hybrids combine the best aspects of both series and parallel hybrids to create an extremely efficient system. The most common configuration, called an input split, is composed of a power-split device (planetary gear transmission), two electric machines and an engine. Within this architecture, all these elements can operate separately. Indeed, the engine is not always ON, and the electricity from the generator may go directly to the wheels to help propel the vehicle or go through an inverter to be stored in the battery. The operational phases for an input split configuration are the following:

- During vehicle launch, when driving, or when the SOC of the battery is high enough, the ICE is not as efficient as electric drive, so the ICE is turned off and the electric machine alone propels the vehicle.
- During normal operation, the ICE output power is split, with part going to drive the vehicle and part used to generate electricity. The electricity goes either to the electric machine, which assists in propelling the vehicle, or to charge the energy storage system. The generator also acts as a starter for the engine.
- During full-throttle acceleration, the ICE and electric machine both power the vehicle, with the energy storage device (e.g., battery) providing extra energy.
- During deceleration or braking, the electric machine acts as a generator, transforming the kinetic energy of the wheels into electricity to charge the energy storage system.

The vehicle control algorithms were based on both the Toyota and Ford implementations.

Figure 86. Power-Split Hybrid Electric Vehicle

4.13.3. Single-Mode Power-Split PHEV (Split PHEV)

The vehicle-level control strategy of a single-mode power-split PHEV was based on the Toyota Prius Prime (Jeong, 2019). The control implemented can be divided into three areas: engine-on condition, battery SOC control, and engine operating condition. Each algorithm is described below.

4.13.3.1. *Engine-ON Condition*

The engine operation determines the mode, such as pure electric vehicle (PEV) mode or HEV mode. The engine is simply turned ON when the driver's power demand exceeds a predefined threshold. As shown in Figure 87, the engine is ON only when the battery SOC is under 17%. This means that only the electric energy is used when the battery SOC is over 17%, i.e., in charge-sustaining (CS) mode. Once the operating mode in terms of SOC is determined, the engine is turned on early if the driver's torque demand exceeds a predefined threshold, which means that the system is changed from PEV mode to HEV mode to meet the power demand.

Figure 87. Engine-ON Condition—2017 Prius Prime Example Based on 25 Test Cycles

4.13.3.2. SOC Control in CS Mode

The desired output power of the battery is closely related to the energy management strategy. When the vehicle is in HEV mode, the battery power is determined by the current SOC, as shown in Figure 88. The overall trend shows that the energy management strategy involves bringing the SOC back to a regular value close to 14%. When the battery SOC decreases below 13.5%, the battery is charged to 10 kW to sustain battery SOC. As battery SOC is increasing, the charging power is decreasing, and the battery is discharged when the battery SOC is over 14.5%. If the battery output power is determined, engine output power can be calculated.

Figure 88. SOC Regulation Algorithm—2017 Prius Prime Example Based on 25 Test Cycles

4.13.3.3. Engine Operation

The two previously described control concepts determine the power-split ratio. The concepts do not, however, generate the target speed or torque of the engine, because the power-split system could have infinite control targets that produce the same power. Therefore, an additional algorithm is needed to determine the engine torque operating points according to the engine speed, as shown in Figure 89. An engine operating line is defined on the basis of the best efficiency curve to select the optimum engine speed for a specific engine power demand.

Figure 89. Example of Engine Operating Target—2017 Prius Prime Example Based on 25 Test Cycles

In summary, the engine is turned ON on the basis of the power demand at the wheel along with the battery SOC. If the engine is turned on, the desired output power of the battery is determined on the basis of the current SOC, and the engine should provide appropriate power to drive the vehicle. The engine operating targets are determined by a predefined line, so the controller can produce required torque values for the motor and the generator on the basis of the engine speed and torque target.

4.13.4. Multi-Mode E-REV (EREV PHEV50)

The multi-mode EREV control is based on the Voltec powertrain from General Motors (GM Voltec powertrain, 2019).

The second-generation Voltec PHEV has one engine, two motor-generators, and one battery. The two electric machines are connected to a main transmission shaft using an individual planetary gear set, as shown in Figure 90. By activating the brake (BK) and clutches, the vehicle can be driven in various modes. Normally, electric machine 1 (MG1) drives the vehicle only by holding the BK. When the BK and one-way clutch (OWC) are locked, both electric machines can provide the maximum torque, putting the vehicle into “two-motor electric vehicle” (EV) mode. An additional planetary gear set is used for a compound power-split mode in extended-range operation. Depending on the clutch (CL) positions or the BK activation status, the input-split or the compound-split mode is determined. The input-split mode is activated by the BK by holding the ring gear of the second planetary gear set. The compound-split mode is activated by the CL when it connects the sun gear of the first planetary gear set to the ring gear of the second gear set.

Figure 90. Configuration of the Chevrolet Volt 2016 Powertrain System

Although a number of sophisticated control concepts were necessarily added to the supervisory control concepts, the main control flow of the vehicle, based on test data, can be summarized as shown in Figure 91. First, the engine ON/OFF control is determined by the battery SOC and the driver’s demand

power. During EV driving, the use of two electric machines allows for two EV driving modes to provide either maximum output torque or increased efficiency by torque distribution. If the engine is on after most of the battery energy has been depleted by EV driving, the operational state of the clutch or brakes is defined to select the extended-range mode. Energy management between the engine and the battery is controlled, depending on the powertrain operation mode. Once the operation mode is chosen, the battery power demand is determined by the proportional control power, which also determines the engine power demand by subtracting the battery power demand from the driver power demand. Then, each component operates according to an optimal target based on engine target and battery power demand. The entire powertrain model, including the vehicle-level controller, was implemented in Autonomie.

Figure 91. Summary of Control Analysis for the Second-Generation Voltec System

4.13.5. Parallel HEV (Par HEV)

In a parallel configuration, the vehicle can be directly propelled by either electrical or mechanical power. Direct connection between the power sources and the wheels leads to lower powertrain losses compared to the pure series configuration. However, since all of the components' speeds are linked to the vehicle's speed, the engine cannot consistently be operated close to its best efficiency curve. The pre-transmission configuration used in this study has an electric machine in between the engine and the transmission. The electric machine peak power usually ranges from 19 to 45 kW for light-duty applications, allowing the driver to propel the vehicle in electric-only mode as well as recover energy through regenerative braking. The pre-transmission parallel HEV configuration can take advantage of different gear ratios that allow the electric machine to operate at higher efficiency and provide high torque for a longer operating range. This configuration allows operation in electric mode during low and medium power demands in addition to the ICE ON/OFF operation. The main challenge for these configurations is being able to maintain a good drive quality because of the engine ON/OFF feature and the high components inertia during shifting events.

The vehicle-level control strategy for a parallel HEV is based on the 2013 Volkswagen Jetta HEV (Kim N. L.-B., 2014). The vehicle control behaviors are interpreted on the basis of the analyzed results in order to understand the overall control behaviors.

4.13.5.1. *Engine Operation*

The upper panel in Figure 92 shows the vehicle speed and wheel demand torque when the engine is turned on. The lower panel shows the operating area of pure electric driving in the same index.

Figure 92. Wheel Torque vs. Vehicle Speed of 2013 VW Jetta HEV

4.13.6. Fuel-Cell HEV

Unlike the other vehicle-level control algorithms previously discussed, the algorithm for fuel-cell HEVs is not derived from test data, owing to the lack of test vehicles. Instead, dynamic programming was used to define the optimum vehicle-level control algorithms for a fuel-cell vehicle. A rule-based control is then implemented to represent the rules issued from the dynamic programming. Overall, owing to the high efficiency of the fuel-cell system, energy storage only regains energy during deceleration and propels the vehicle under low-load operations; the fuel-cell system does not recharge the battery. Unlike in electric-drive powertrains with an engine, the battery does not smooth the transient demands. An example of fuel-cell hybrid operations is shown in Figure 93.

Figure 93. Component Operating Conditions of a Fuel-Cell Vehicle on the Urban European Drive Cycle Using Dynamic Programming

4.14. High-Level Overview of Autonomie Controllers

4.14.1. Supervisory Controller

In Autonomie, the supervisory controller manages the vehicle-level decision-making.

The main control decisions (i.e., gear shift demand, etc.) are sent to the vehicle propulsion architecture (VPA). The lower-level controls within the VPA execute the demands (e.g., different steps to perform a shifting event coordinated between engine, gearbox, etc.) The split of the control logic between the supervisory controller and component controllers depends on the vehicle configuration.

For conventional vehicles, all decisions are made at the component (including the driver) level.

For electrified vehicles, the supervisory controller manages most decisions, including engine ON/OFF strategy, engine demand, electric machine demand, and others, while the transients (for example, how

to shift a gear) are handled at the component level. Figure 94 shows the Autonomie supervisory controller.

Figure 94. Autonomie Supervisory Controller

The individual component controllers are part of the VPA.

Figure 95 shows the structure inside the VPA.

Figure 95. Autonomie Nested-Level Controllers

Each of the component models in Autonomie may contain several subsystems, consisting of either plant model only or plant and controller models. Figure 96 shows the component model layout in Autonomie.

Figure 96. Autonomie Component Model Layout

4.14.2. Component Controllers

A component controller can contain a single control model (ctrl), or a combination of two, three or four of the following sub-models:

- A demand model (dmd)
- A transient model (trs)
- A constraint model (cstr)
- A command model (cmd)

For example, the battery controller in Figure 97 consists of a single control model (ctrl).

Figure 97. Battery Controller Composed of a Single Control Model

In contrast, the engine controller in Figure 98 consists of four sub-models: dmd, trs, cstr and cmd.

Figure 98. Engine Controller Composed of Four Sub-Models

4.14.2.1. The Demand Block – dmd

The demand block in Figure 99 is the usual entry point of the low-level controllers. The request coming from the vehicle controller is translated into a request at the component level (for example, torque demand to pulse-width modulation [0–1]). The request then feeds into the transient block.

Figure 99. The Demand Block

4.14.2.2. The Transient Block – trs

The transient block in Figure 100 checks for conditions that may impact the request coming from prior blocks. For instance, if a shift is in progress, the engine torque may need to be reduced for the duration of the shift.

Figure 100. The Transient Block

4.14.2.3. The Constraint Block – *cstr*

The constraint block, shown in Figure 101, checks component limits at any given time. For example:

- The maximum engine torque available, given the current engine speed and engine condition
- The maximum battery charge and discharge current, given the current conditions (voltage, temperature, etc.)

The constraints are fed into the command block.

Figure 101. The Constraint Block

4.14.2.4. The Command Block – *cmd*

The command block, shown in Figure 102, receives inputs from the transient and constraint blocks and determines the final request that will be sent to the plant model.

Figure 102. The Command Block

4.15. Low-Level Controller Summary for Engine and Transmission Models

Autonomie allows for the simulation of hundreds of powertrains and technology combinations. That being said, numerous control functionalities are common across architectures. The following section describes the list of low-level controller option for both engine and transmission and how some are shared across multiple vehicle powertrains.

Figure 103 shows how the engine low-level controls are organized by functionality (transient, command, constraint, and demand) and re-used across transmissions and powertrains.

Figure 103. Low-Level Control for Engine

Figure 104 shows how the transmission low-level controls are organized by functionality (transient, demand, and command) and reused across transmission types and powertrains.

Figure 104. Low-Level Control for Transmission

Table 7 through Table 9 provide short descriptions of the component control blocks, their I/O, and lists of compatible plant models.

Table 7. Engine Control — Low-Level Demand Models

Model Name	Description	Powertrains Used	Compatible Plant Models	Inputs	Outputs
eng_ctrl_dmd	Translates the driver pedal position into a torque demand	Light-duty conventional powertrains with any transmission	eng_plant_hot_map_si_fuel_cut_off eng_plant_hot_map_si_thermal eng_plant_hot_map_ci eng_plant_hot_map_si_fuel_cut_off_wit h_cylinder_deac	drv_accel_dmd_simu gb_plant_ratio_simu chas_plant_lin_spd_out_simu eng_ctrl_cstr_trq_hot_max_simu eng_ctrl_cstr_trq_hot_min_simu eng_plant_spd_out_simu	eng_ctrl_trq_dmd_simu

Table 8. Engine Control — Low-Level Constraint Models

Model Name	Description	Powertrains Used	Compatible Plant Models	Inputs	Outputs
eng_ctrl_cstr_map_hot	Provides min and max torque capabilities based on engine speed	Used in all powertrain configurations	eng_plant_hot_map_si_fuel_cut_off eng_plant_hot_map_si_thermal eng_plant_hot_map_ci eng_plant_hot_map_si_fuel_cut_off_with_cylinder_deac eng_plant_hot_map_si eng_plant_hot_map eng_plant_hot_map_CS	eng_plant_spd_out_simu	eng_ctrl_cstr_trq_hot_max_simu eng_ctrl_cstr_trq_hot_min_simu

Table 9. Engine Control — Low-Level Transient Models

Model Name	Description	Powertrains Used	Compatible Plant Models	Inputs	Outputs
eng_ctrl_trs_au	Determines appropriate engine torque on basis of engine mode (idling, shift in progress, engine starting, etc.)	Conventional and hybrid powertrain with automatic transmission	eng_plant_hot_map_ci eng_plant_hot_map_si eng_plant_hot_map_CS	gb_ctrl_trs_eng_mode_simu eng_ctrl_dmd_trq_simu eng_ctrl_cmd_on_simu eng_plant_spd_out_simu accmech_plant_trq_simu cpl_plant_cmd_simu drv_accel_dmd_simu gb_ctrl_trs_previous_gear_simu gb_ctrl_dmd_gear_simu gb_ctrl_trs_sit_in_progress_simu	eng_ctrl_trs_trq_simu eng_ctrl_trs_on_simu

Model Name	Description	Powertrains Used	Compatible Plant Models	Inputs	Outputs
eng_ctrl_trs_au_with_str_fco	Determines appropriate engine torque on basis of engine mode (idling, shift in progress, engine starting, etc.)	Conventional powertrain with automatic transmission with fuel cut-off	eng_plant_hot_map_si_fuel_cut_off	gb_ctrl_trs_eng_mode_simu eng_ctrl_dmd_trq_simu eng_ctrl_cmd_on_simu eng_plant_spd_out_simu accmeh_plant_trq_simu cpl_plant_cmd_simu drv_accel_dmd_simu gb_plant_gear_simu gb_ctrl_dmd_gear_simu gb_ctrl_trs_sft_in_progress_simu drv_brk_dmd_simu gen_plant_trq_out_simu chas_plant_lin_spd_out_simu	eng_ctrl_trs_trq_simu eng_ctrl_trs_on_simu eng_ctrl_trs_fuel_cut_off_simu
eng_ctrl_trs_au_with_str_fco_with_vpc	Determines appropriate engine torque on the basis of engine mode (idling, shift in progress, engine starting, etc.)	Hybrid powertrain with automatic transmission with fuel cut-off	eng_plant_hot_map_si_fuel_cut_off	eng_mode vpc_eng_trq_dmd vpc_eng_on_dmd eng_spd accmech_trq cpl_position drv_acc_pedal previous_gear gb_gear_dmd shift_in_progress drv_brk_dmd	eng_ctrl_trs_trq_simu eng_ctrl_trs_on_simu eng_ctrl_trs_fuel_cut_off_simu
eng_ctrl_trs_au_with_str_fco_with_vpc_cyldeac	Determines appropriate engine torque on the basis of engine mode (idling, shift in progress, engine starting, etc.)	Hybrid powertrain with automatic transmission with fuel cut-off and cylinder deactivation	eng_plant_hot_map_si_fuel_cut_off_with_cylinder_deac	eng_mode vpc_eng_trq_dmd vpc_eng_on_dmd eng_spd accmech_trq cpl_position drv_acc_pedal previous_gear gb_gear_dmd shift_in_progress drv_brk_dmd	eng_ctrl_trs_trq_simu eng_ctrl_trs_on_simu eng_ctrl_trs_fuel_cut_off_simu eng_ctrl_trs_cylin_deac_state_simu

Model Name	Description	Powertrains Used	Compatible Plant Models	Inputs	Outputs
eng_ctrl_trs_au_with_str_fco_thermal	Determines appropriate engine torque on the basis of engine mode (idling, shift in progress, engine starting, etc.)	Conventional powertrain with automatic transmission and an engine with a temperature model	eng_plant_hot_map_si_thermal	gb_ctrl_trs_eng_mode_simu eng_ctrl_dmd_trq_simu eng_ctrl_cmd_on_simu eng_plant_spd_out_simu accmeh_plant_trq_simu cpl_plant_cmd_simu drv_accel_dmd_simu gb_plant_gear_simu gb_ctrl_dmd_gear_simu gb_ctrl_trs_sft_in_progress_simu drv_brk_dmd_simu gen_plant_trq_out_simu chas_plant_lin_spd_out_simu eng_plant_temp_simu	eng_ctrl_trs_trq_simu eng_ctrl_trs_on_simu eng_ctrl_trs_fuel_cut_off_simu
eng_ctrl_trs_fuel_cut_off_dm	Determines appropriate engine torque on the basis of engine mode (idling, shift in progress, engine starting, etc.)	Conventional powertrain with manual transmission with fuel cut-off	eng_plant_hot_map_si_fuel_cut_off	gb_ctrl_trs_eng_mode_simu eng_ctrl_dmd_trq_simu drv_key_on_dmd_simu eng_plant_spd_out_simu accmeh_plant_trq_simu gb_plant_gear_simu cpl_plant_cmd_simu gb_ctrl_dmd_gear_simu drv_accel_dmd_simu drv_brk_dmd_simu cpl_plant_spd_out_simu	eng_ctrl_trs_trq_simu eng_ctrl_trs_on_simu eng_ctrl_trs_fuel_cut_off_simu
eng_ctrl_trs_ct_with_str_fco	Determines appropriate engine torque on the basis of engine mode (idling, shift in progress, engine starting, etc.)	Conventional powertrain with a CVT and with fuel cut-off	eng_plant_hot_map_si_fuel_cut_off	gb_ctrl_trs_eng_mode_simu eng_ctrl_dmd_trq_simu eng_ctrl_cmd_on_simu eng_plant_spd_out_simu accmeh_plant_trq_simu cpl_plant_cmd_simu drv_accel_dmd_simu gb_ctrl_trs_cpl_mode_simu drv_brk_dmd gen_trq_out	eng_ctrl_trs_trq_simu eng_ctrl_trs_on_simu eng_ctrl_trs_fuel_cut_off_simu

Model Name	Description	Powertrains Used	Compatible Plant Models	Inputs	Outputs
				veh_speed	
eng_ctrl_trs_dct_with_str_fco	Determines appropriate engine torque on the basis of engine mode (idling, shift in progress, engine starting, etc.)	Conventional powertrain with a DCT and with fuel cut-off	eng_plant_hot_map_si_fuel_cut_off	gb_ctrl_trs_eng_mode_simu eng_ctrl_dmd_trq_simu eng_ctrl_cmd_on_simu eng_plant_spd_out_simu accmeh_plant_trq_simu cpl_plant_cmd1_simu cpl_plant_cmd2_simu drv_accel_dmd_simu gb_plant_spd_out_simu gb_ctrl_dmd_gear_simu gb_ctrl_trs_sft_in_progress_simu gen_plant_gear_simu drv_brk_dmd_simu	eng_ctrl_trs_trq_simu eng_ctrl_trs_on_simu eng_ctrl_trs_fuel_cut_off_simu
eng_ctrl_trs_split	Limits engine torque to the max engine capability	Hybrid configuration with a power-split powertrain, including 2 times 2-wheel drive, extended range, and multi mode vehicles	eng_plant_hot_map_si eng_plant_hot_map_ci eng_plant_hot_map_si_thermal	vpc_eng_trq_dmd_simu eng_ctrl_cstr_trq_hot_max_simu vpc_eng_on_dmd_simu	eng_ctrl_trs_trq_simu eng_ctrl_trs_on_simu
eng_ctrl_trs_amt	Determines appropriate engine torque on the basis of engine mode (idling, shift in progress, engine starting, etc.)	HEV and PHEV powertrain with an automated manual transmission	eng_plant_hot_map_si	gb_ctrl_trs_eng_mode_simu eng_ctrl_dmd_trq_simu eng_ctrl_cmd_on_simu eng_plant_spd_out_simu accmech_plant_trq_simu cpl_plant_cmd_simu drv_accel_dmd_simu gb_plant_spd_out_simu gb_ctrl_dmd_gear_simu gb_ctrl_trs_sft_in_progress_simu gb_plant_gear_simu	eng_ctrl_trs_trq_simu eng_ctrl_trs_on_simu

Model Name	Description	Powertrains Used	Compatible Plant Models	Inputs	Outputs
eng_ctrl_trs_ct	Determines appropriate engine torque on the basis of engine mode (idling, shift in progress, engine starting, etc.)	Parallel hybrid powertrain with CVT	eng_plant_hot_map_si	eng_ctrl_trs_eng_mode_simu eng_ctrl_dmd_trq_simu eng_ctrl_cmd_on_simu eng_plant_spd_out_simu accmech_plant_trq_simu cpl_plant_cmd_simu drv_accel_dmd_simu gb_ctrl_trs_cpl_mode_simu	eng_ctrl_trs_trq_simu eng_ctrl_trs_on_simu
eng_ctrl_trs_au_cylinder_deac_with_str_fco	Determines appropriate engine torque on the basis of engine mode (idling, shift in progress, engine starting, etc.)	Conventional powertrain with an automatic transmission, fuel cut-off, and cylinder deactivation	eng_plant_hot_map_si_fuel_cut_off_with_cylinder_deac	eng_mode vpc_eng_trq_dmd vpc_eng_on_dmd eng_spd accmech_trq cpl_position drv_acc_pedal previous_gear gb_gear_dmd shift_in_progress drv_brk_dmd gen_trq_out veh_speed	eng_ctrl_trs_trq_simu eng_ctrl_trs_on_simu eng_ctrl_trs_fuel_cut_off_simu eng_ctrl_trs_cylin_deac_state_simu

4.16. Autonomie Validation

Argonne has been validating vehicle models for almost 20 years, leveraging vehicle dynamometer test data from the AMTL. Test data have been collected at Argonne from more than 60 vehicles, ranging from model year 2000 to the present. A large number of signals were collected on each vehicle for specific control analysis. Signals include component efforts (torque, current, etc.) and flow (rotational speed, linear speed, etc.) as well as temperatures and direct fuel-flow measurement collected using sensors and high-speed controller area networks (CANs). These measurements were integrated and aligned into a single data acquisition system. Some additional parameters were estimated on the basis of measured data and other advanced vehicle technology (e.g., electric machine current as estimated from measured speed, torque and voltage). Each individual model was then independently validated. Vehicle system model validation was quantified using normalized cross correlation power (NCCP),⁴ over a large number of cycles.

As a result, a large number of Autonomie vehicle models have been validated within test-to-test repeatability for a wide range of technologies and powertrain configurations. The following section provides some validation examples, using AMTL data.

Below is a subset of references for vehicle and component model validations in Autonomie over the years:

- N. Kim, S. Choi, J. Jeong, R. Vijayagopal, K. Stutenberg and A. Rousseau, **“Vehicle Level Control Analysis for Voltec Powertrain.”** 30th International Electric Vehicle Symposium and Exhibition (EVS30), Stuttgart, Germany, October 9–11, 2017
- J. Jeong, W. Lee, N. Kim, K. Stutenberg, et al., **“Control Analysis and Model Validation for BMW i3 Range Extender.”** SAE Technical Paper 2017-01-1152, 2017, DOI:10.4271/2017-01-1152
- H. Son, N. Kim, S. Ko, A. Rousseau, and H. Kim, **“Development of Performance Simulator for a HEV with CVT and Validation with Dynamometer Test Data.”** 28th International Electric Vehicle Symposium (EVS28), Kintex, South Korea, 2015
- N. Kim, N. Kim, and A. Rousseau, **“Thermal Model Developments for Electrified Vehicles.”** 28th International Electric Vehicle Symposium (EVS28), Kintex, South Korea, 2015
- N. Kim, J. Jeong, A. Rousseau, and H. Lohse-Busch, **“Control Analysis and Thermal Model Development of PHEV.”** SAE 2015-01-1157, SAE World Congress, Detroit, April 15, 2015
- N. Kim, A. Rousseau, and H. Lohse-Busch, **“Advanced Automatic Transmission Model Validation Using Dynamometer Test Data.”** SAE 2014-01-1778, SAE World Congress, Detroit, April 2014

⁴ See SAE 2011-01-0881, “Test Correlation Framework for HEV System Model,” Ford Motor Company

- N. Kim, E. Rask and A. Rousseau, **“Control Analysis under Different Driving Conditions for Peugeot 3008 Hybrid 4.”** SAE 2014-01-1818, SAE World Congress, Detroit, April 2014
- D. Lee, A. Rousseau, E. Rask, **“Development and Validation of the Ford Focus BEV Vehicle Model.”** 2014-01-1809, SAE World Congress, Detroit, April 2014
- N. Kim, A. Rousseau, D. Lee, and H. Lohse-Busch, **“Thermal Model Development & Validation for the 2010 Toyota Prius.”** 2014-01-1784, SAE World Congress, Detroit, April 2014
- N. Kim, N. Kim, A. Rousseau, and M. Duoba, **“Validating Volt PHEV Model with Dynamometer Test Data using Autonomie.”** SAE 2013-01-1458, SAE World Congress, Detroit, April 2013
- N. Kim, A. Rousseau, and E. Rask, **“Autonomie Model Validation with Test Data for 2010 Toyota Prius.”** SAE 2012-01-1040, SAE World Congress, Detroit, April 2012
- N. Kim, R. Carlson, F. Jehlik, and A. Rousseau, **“Tahoe HEV Model Development in PSAT.”** SAE paper 2009-01-1307, SAE World Congress, Detroit, April 2009
- Q. Cao, S. Pagerit, R. Carlson, and A. Rousseau, **“PHEV Hymotion Prius model validation and control improvements.”** 23rd International Electric Vehicle Symposium (EVS23), Anaheim, CA, December 2007
- Rousseau, P. Sharer, S. Pagerit, and M. Duoba, **“Integrating Data, Performing Quality Assurance, and Validating the Vehicle Model for the 2004 Prius Using PSAT.”** SAE paper 2006-01-0667, SAE World Congress, Detroit, April 2006
- M. Pasquier, A. Rousseau, and M. Duoba, **“Validating Simulation Tools for Vehicle System Studies Using Advanced Control and Testing Procedures.”** 18th International Electric Vehicle Symposium (EVS18), Berlin, Germany, October 2001
- Rousseau, B. Deville, G. Zini, J. Kern, J. Anderson, and M. Duoba, **“Honda Insight Validation Using PSAT.”** 01–FTT49, Future Transportation Technology Conference, Costa-Mesa, August 2001
- Rousseau and M. Pasquier, **“Validation of a Hybrid Modeling Software (PSAT) Using Its Extension for Prototyping (PSAT-PRO).”** Global Powertrain Congress, Detroit, June 2001.

4.16.1. Conventional Vehicles

The main focus of conventional vehicles’ validation are the shifting algorithm, torque converter lockup and fuel cutoff. First, the simulated vehicle speed, engine speed, and engine torque are compared with test results. For example, Figure 105 shows the comparison for an automatic transmission on the UDDS cycle:

- Initial calibration (simulation 1)
- Calibrated algorithm using test data (simulation 2)

Figure 105. Simulation and Testing Results on UDDS Cycle (0–505 s) for 2013 Sonata Conventional 6ATX

In Figure 106, the gear numbers on the UDDS cycle are compared with the test results for both 6- and 8-speed transmissions. Both simulations are close in shifting performance to the test results. Figure 107 demonstrates the robust calibration of the shifting algorithm during the New European Driving cycle (NEDC).

Figure 106. Shifting Algorithm Validation for 2013 Sonata Conventional 6ATX (Left) and 2013 Chrysler 300 8ATX (Right) on the UDDS Cycle (0-505 s)

Figure 107. Shifting Algorithm Validation for 2012 Fusion Conv. 6ATX (left) and 2013 Chrysler 300 8ATX (Right) on the NEDC Cycle

The CVT model and shifting control strategy developed in Autonomie were also validated by comparing the simulation results with the experimental results from Argonne's AMTL. Figure 108 shows the validation results for the 2012 Honda Civic HEV system on the UDDS and Highway Fuel Economy (HWFET) cycles. The simulated vehicle speed, gear ratio, engine torque and battery SOC behave like the experimental results, demonstrating the validity of the simulation model and control strategy.

Figure 108. Comparison of Simulation and Test Data for 2012 Honda Civic CVT HEV on UDDS (Left) and HWFET (Right) Cycles

4.16.2. Power-Split HEV

Multiple versions of the power-split HEV have been tested and validated since early 2000. As with the other powertrains, we focus first and foremost on validating the component operating conditions throughout the driving cycles. For example, Figure 109 shows consistent engine ON/OFF, SOC control and component operating conditions (engine torque & speed) for the 2010 Toyota Prius on the UDDS cycle. The latest Toyota Prius HEV has been similarly validated.

Figure 109. Results of Simulation and Testing on UDDS Cycle for 2010 Toyota Prius HEV

4.16.3. Pre-Transmission HEV

The pre-transmission HEV control logic was validated using Argonne AMTL test data from the 2013 Jetta DCT hybrid. The simulation results for the vehicle speed, gear number, and battery SOC on the UDDS cycle, shown in Figure 110, showed good correlation with the test data.

Figure 110. Results of Simulation and Testing on UDDS cycle for 2013 Jetta DCT Hybrid

4.16.4. Range-Extender PHEV

The range-extender PHEV model was validated under different thermal conditions, using Argonne's AMTL test data from the second-generation Chevrolet Volt. The vehicle operating behavior, including vehicle speed, battery SOC, fuel consumption, and engine speed, torque, and temperature under ambient temperature were successfully compared with the testing results shown in Figure 111.

Figure 111. Results of Simulation and Testing on UDDS cycle for 2012 Chevrolet Volt PHEV

Additional configurations including start-stop, blended PHEV, and battery electric vehicles (BEVs) have also been validated.

5. Vehicle and Component Assumptions

Directed by NHTSA, based on the feedback received from reviewers on the midterm TAR, a number of steps were taken to improve the overall vehicle simulation process and consider additional vehicles to better replicate fleets in both existing and future markets.

The primary focus was to update the vehicle and component assumptions to represent the latest vehicles in the market. An extensive analysis was carried out on the existing vehicles in the market to evaluate the vehicle parameters and component weights across the considered vehicle classifications. Along with vehicle parameters (frontal area, drag coefficient, etc.), individual component weight assumptions (body, chassis, interior, etc.) have been analyzed. The vehicle attribute parameters have been updated using Argonne's internal vehicle technical database. Finally, the vehicle component weights have been updated using A2Mac1 (A2Mac1, 2018) teardown analysis of different representative vehicles.

This section elaborates on the method used to select each attribute. The attributes were defined across different vehicle classes and performance categories, along with different transmission types and number of gears. Table 10 details the different vehicle classifications along with the definition of the different performance categories.

Table 10. Vehicle Classification and Performance Categories

Vehicle Class	Performance Category	0-60 mph Time (s)
Compact	Base / Premium	10 / 8
Midsize	Base / Premium	9 / 6
Small SUV	Base / Premium	9 / 7
Midsize SUV	Base / Premium	10 / 7
Pickup	Base / Premium	7

Additional performance metrics have been developed:

- Gradeability: 6% grade at 65 mph
- Payload: 900 kg (Pickup Base/Premium only)
- Towing: 3000 kg (Pickup Base) and 4350 kg (Pickup Premium)

5.1. Vehicle-Level Attributes Selection

The assumptions for each vehicle class and performance category have been defined individually.

5.1.1. Drag Coefficient

The following section shows the distribution of the drag coefficient using the Argonne internal vehicle technology database. “ANL Value” represents the parameter value selected for the reference vehicle in Autonomie.

Figure 112 shows the distribution of drag coefficient values for the compact (base) vehicle class.

Figure 112. Distribution of Drag Coefficient Values of Compact (Base) Vehicle Class

Such analyses were performed across the different vehicle classes and performance categories and the reference values selected. Table 11 summarizes the values defined for the drag coefficient of the reference vehicles for each vehicle class and performance category.

Table 11. Drag Coefficient Ratio Summary Table

Vehicle Class	Performance Category	Reference Value
Compact	Base/Premium	0.31
Midsize	Base/Premium	0.30
Small SUV	Base/Premium	0.36
Midsize SUV	Base/Premium	0.38
Pickup	Base/Premium	0.42

5.1.2. Frontal Area

A similar detailed analysis was performed to determine the frontal area values for the reference vehicle across different vehicle classifications and performance categories. Table 12 summarizes values defined

for the frontal area of the reference vehicles for the different vehicle classes and performance categories.

Table 12. Frontal Area Summary Table

Vehicle Class	Performance Category	Reference Value (m²)
Compact	Base/Premium	2.3
Midsize	Base/Premium	2.35
Small SUV	Base/Premium	2.65
Midsize SUV	Base/Premium	2.85
Pickup	Base/Premium	3.25

5.1.3. Rolling Resistance

A constant value of 0.009 is used as the first coefficient term of the wheel rolling resistance for all classes and performance categories. The overall rolling resistance of the wheel is, however, speed-dependent, with a factor of 0.00012.

5.2. Vehicle Component Weight Selection

The vehicles in Autonomie are built from the ground up, using individual component weights. Powertrain-dependent component weights (engine, motor, fuel cell system, high-power/energy battery, etc.) are updated as a part of the sizing procedure. The following sections describe the process of selecting the non-powertrain weights, which are developed based on an analysis using the A2Mac1 database (A2Mac1, 2018).

5.2.1. Body Weight

Figure 113 shows the distribution of body weights for the compact (base) vehicle class.

Figure 113. Body Weight Selection for Compact (Base) Vehicles (A2Mac1, 2018)

Such analyses are performed across the different vehicle classes and performance categories and the reference values selected. Table 13 summarizes the values defined for the body weight of the reference vehicles for the different vehicle classes and performance categories.

Table 13. Body Weight Summary

Vehicle Class	Performance Category	Reference value (kg)
Compact	Base/Premium	525
Midsize	Base/Premium	650
Small SUV	Base/Premium	650
Midsize SUV	Base/Premium	650/750
Pickup	Base/Premium	650/800

5.2.2. Chassis Weight

Figure 114 shows the distribution of chassis weights for the compact (base) vehicle class.

Figure 114. Chassis Weight Selection for Compact (Base) Vehicles (A2Mac1, 2018)

Such analyses are performed across the different vehicle classes and performance categories and the reference values selected. Table 14 summarizes the values defined for the chassis weights of the reference vehicles for the different vehicle classes and performance categories.

Table 14. Chassis Weight Selection Summary

Vehicle Class	Performance Category	Reference Value (kg)
Compact	Base/Premium	160
Midsize	Base/Premium	200
Small SUV	Base /Premium	200
Midsize SUV	Base /Premium	200/225
Pickup	Base/Premium	300/350

5.2.3. Interior Weight

Figure 115 shows the distribution of interior weights for the compact (base) vehicle class.

Figure 115. Interior Weight Selection for Compact (Base) Vehicles (A2Mac1, 2018)

Such analyses are performed across the different vehicle classes and performance categories and the reference values selected. Table 15 summarizes the values defined for the interior weights of the reference vehicles for the different vehicle classes and performance categories.

Table 15. Interior Weight Selection Summary

Vehicle Class	Performance Category	Reference Value (kg)
Compact	Base/Premium	150/200
Midsize	Base/Premium	175/200
Small SUV	Base/Premium	180/220
Midsize SUV	Base/Premium	200/240
Pickup	Base/Premium	160/200

5.2.4. Safety System Weight

Figure 116 shows the distribution of safety-system weights for the compact (base) vehicle class.

Figure 116. Safety System Weight Selection for Compact (Base) Vehicle (A2Mac1, 2018)

Such analyses are performed across the different vehicle classes and performance categories and the reference values selected. Table 16 summarizes the values defined for the safety-system weights of the reference vehicles for the different vehicle classes and performance categories.

Table 16. Safety System Weight Selection Summary

Vehicle Class	Performance Category	Reference Value (kg)
Compact	Base/Premium	20/22
Midsize	Base/Premium	25/28
Small SUV	Base/Premium	25/28
Midsize SUV	Base/Premium	30
Pickup	Base/Premium	30

5.2.5. Thermal System Weight

Figure 117 shows the distribution of thermal-system weights for the compact (base) vehicle class.

Figure 117. Thermal System Weight Selection for Compact (Base) Vehicle (A2Mac1, 2018)

Such analyses are performed across the different vehicle classes and performance categories and the reference values selected. Table 17 summarizes the values defined for the thermal system weights of the reference vehicles for the different vehicle classes and performance categories.

Table 17. Thermal System Weight Selection Summary

Vehicle Class	Performance Category	Reference Value (kg)
Compact	Base/Premium	18/20
Midsize	Base/Premium	25/28
Small SUV	Base/Premium	25/28
Midsize SUV	Base/Premium	30
Pickup	Base/Premium	30

5.2.6. Brake Mechanism Weight

Figure 118 shows the distribution of brake mechanism weights for the compact (base) vehicle class.

Figure 118. Brake Mechanism Weight Selection for Compact (Base) Vehicle (A2Mac1, 2018)

Such analyses are performed across the different vehicle classes and performance categories and the reference values selected. Table 18 summarizes the values defined for the brake mechanism weights of the reference vehicles for the different vehicle classes and performance categories.

Table 18. Brake Mechanism Weight Selection Summary

Vehicle Class	Performance Category	Reference Value (kg)
Compact	Base/Premium	50/55
Midsize	Base/Premium	60/65
Small SUV	Base/Premium	60/75
Midsize SUV	Base/Premium	70/75
Pickup	Base/Premium	90/95

5.2.7. Steering System Weight

Figure 119 shows the distribution of steering system weights for the compact (base) vehicle class.

Figure 119. Steering System Weight Selection for Compact (Base) Vehicle (A2Mac1, 2018)

Such analyses are performed across the different vehicle classes and performance categories and the reference values selected. Table 19 summarizes the values defined for the steering system weights of the reference vehicles for the different vehicle classes and performance categories.

Table 19. Steering System Weight Selection Summary

Vehicle Class	Performance Category	Reference Value (kg)
Compact	Base/Premium	20/22
Midsize	Base/Premium	25/28
Small SUV	Base/Premium	25/28
Midsize SUV	Base/Premium	30
Pickup	Base/Premium	30

5.2.8. Electrical Accessories Weight

Figure 120 shows the distribution of electrical accessories weights for the compact (base) vehicle class.

Figure 120. Electrical Accessories Weight Selection for Compact (Base) Vehicle (A2Mac1, 2018)

Such analyses are performed across the different vehicle classes and performance categories and the reference values selected. Table 20 summarizes the values defined for the electrical accessories weights of the reference vehicles for the different vehicle classes and performance categories.

Table 20. Electrical Accessories Weight Selection Summary

Vehicle Class	Performance Category	Reference Value (kg)
Compact	Base/Premium	30/35
Midsize	Base/Premium	30/40
Small SUV	Base/Premium	30/40
Midsize SUV	Base/Premium	30/50
Pickup	Base/Premium	80/100

5.2.9. Engine Weight Determination

For the FRM runs, a detailed analysis of engine weight was done on A2mac1. Using the analysis, engines are divided into naturally aspirated engines and turbo-charged engines. Figure 121 shows the relationship between the engine weight and the engine peak power.

Figure 121. Engine Weight vs. Engine Peak Power (A2Mac1, 2018)

From the figure above, the following relationship is derived:

- Naturally aspirated engine:

$$\text{Weight} = -9.80403e-10 * \text{EnginePower}^2 + 0.000841923 * \text{EnginePower} + 32.7479$$

- Turbocharged engine:

$$\text{Weight} = -3.68948e-10 * \text{EnginePower}^2 + 0.00057385 * \text{EnginePower} + 49.8163$$

5.2.10. Electric Machines System Weight

For FRM runs, the electric machine weight computation was updated by analyzing the existing electric machine component weights and peak power output from A2mac1.

Figure 122. Electric Machine Weight vs. Peak Power Output (A2Mac1, 2018)

The linear equation derived from the plot above is used for the weight of electric machines. The equation is stated below:

$$Weight (kg) = 0.113639 \times Electric Motor Peak Power Output (kW) + 27.0705$$

5.3. Transmission Assumptions

5.3.1. Transmission Technology Definitions

Table 21 details the different transmission technologies used in the study.

Table 21. Transmission Technologies

Simulation Name	Transmission Type	Description/Source
5AU	5-speed automatic (base class)	1:1 ratio efficiency from 6AU (base) and use rule to generate the efficiency for other ratios
5AU	5-speed automatic (premium class)	1:1 ratio efficiency from 6AU (premium) and use rule to generate the efficiency for other ratios
6AU	6-speed automatic (base class)	Transmission used for low-torque engines. Source: U.S. EPA test data – GM 6T40 (U.S. EPA, 2016a)
6AU	6-speed automatic (premium class)	Transmission used for high-torque engines Source: (National Highway Traffic Safety Administration (NHTSA), 2016)

Simulation Name	Transmission Type	Description/Source
7AUp	7-speed automatic+	1:1 ratio efficiency from 8AU+ and use rule to generate the efficiency for other ratios
8AU	8-speed automatic	Source: U.S. EPA test data – Ram 845RE (U.S. EPA, 2016b)
8AUp	8-speed automatic+	845RE (8AU) with improved efficiency (NHTSA data)
8AUpp	8-speed automatic++	845RE (8AU) with improved efficiency (NHTSA data)
9AUp	9-speed automatic+	1:1 ratio efficiency from 8AU+ and use rule to generate the efficiency for other ratios
10AUp	10-speed automatic+	1:1 ratio efficiency from 8AU+ and use rule to generate the efficiency for other ratios
10AUpp	10-speed automatic++	1:1 ratio efficiency from 8AU++ and use rule to generate the efficiency for other ratios
5DM	5-speed manual	1:1 ratio efficiency from 6DCT and use rule to generate the efficiency for other ratios
6DM	6-speed manual	1:1 ratio efficiency from 6DCT and use rule to generate the efficiency for other ratios
7DM	7-speed manual (premium class)	1:1 ratio efficiency from 6DCT and use rule to generate the efficiency for other ratios
6DCT	6-speed DCT	Source: Argonne (Kim N. L.-B., 2014)
8DCT	8-speed DCT	1:1 ratio efficiency from 6DCT and use rule to generate the efficiency for other ratios
CVT	CVT	Source: Argonne (Hanho Son, 2015)
CVTp	CVT+	CVT with improved efficiency (NHTSA data)

Like engines, transmissions in the market always include multiple improvements from one generation to the next (such as increased gear number and efficiency). The objective of the transmission selection was to separate the benefits of increased gear number from those of improved efficiency. For example, 6AU to 8AU quantify the effectiveness of increased gear span and gear number, while 8AU to 8AU+ quantify the impact of efficiency. As a result, while the test data were used to model several transmissions, a rule was used to develop some transmission models to ensure appropriate effectiveness value.

5.3.1.1. Automatic transmission efficiency rule (USCAR, 2015)

In the equations below, τ is the normalized torque (torque/max rated input torque). In the specific data set that was used to generate these equations, the maximum torque was taken to be 450 Nm.

The maximum efficiency is given by:

$$(1) \eta = 100 - 1.385 \times \tau^{-1.0127}$$

The temperature dependence is considered as a function of torque for temperatures ranging from $T = 38^\circ\text{C}$ to $T = 93^\circ\text{C}$:

$$(2) \Delta\eta = 0.3612 \times \tau^{-0.9238}$$

The speed dependence is a function of input torque for speeds ranging from 500 rpm to 5000 rpm:

$$(3) \Delta\eta = 0.6394 \times \tau^{-1.3068}$$

The efficiency data is generated using the following steps:

Start with the “maximum efficiency curve,” which essentially represents the efficiency for direct drive (1:1 ratio) at 93°C .

The temperature offset is applied when calculating efficiency at 38°C .

The speed offset is applied.

The gear ratio other than the direct drive is scaled.

Figure 123 shows the plot of the efficiency for direct drive for the range of temperatures and speeds considered. For other gears, the results are scaled down by a factor ranging between 0.97 and 1.0.

Figure 123. Efficiency for Direct Drive, for the Range of Temperatures and Speeds Considered

5.3.1.2. Dual-clutch transmission efficiency rule

The efficiency of the DCT is broken down into a speed-dependent term (spin loss) and a load dependent term (gear train mechanical efficiency).

For the speed-dependent part, the turning torque (Nm) is given by the following equations through curve fit as a function of the overall gear ratio R:

At 93°C, 500 rpm

$$(1) \quad T = 4.89 \times \left(\frac{1}{R}\right)^2 + 0.135 \times \left(\frac{1}{R}\right) + 0.21$$

At 93°C, 5000 rpm

$$(2) \quad T = 23.5 \times \left(\frac{1}{R}\right)^2 + 1.4 \times \left(\frac{1}{R}\right) + 1.7$$

The turning torque is approximately linear between 500 rpm and 5000 rpm.

The gear mechanical efficiency is very high, and can be assumed to be in the range of 99% to 99.5% per gear mesh. The mesh efficiency is higher when the meshing gears are of similar size.

The efficiency data is generated by the following steps:

- The torque loss is subtracted from the input torque.
- The additional torque loss due to constant mechanical efficiency is calculated by multiplying the difference between the input torque and the torque loss by (1 - efficiency).

- The efficiency is calculated by taking the sum of the (spin) torque loss and the loss due to mechanical efficiency and dividing it by the input torque.

The data set is based on a dual-clutch transmission with a rated input torque of up to 250 Nm.

5.3.2. Transmission Attributes

5.3.2.1. Gear Span

Figure 124 shows the analysis done on the existing vehicle attributes to determine the gear span of a 6-speed automatic transmission.

Figure 124. Vehicle Attribute Analysis for Gear Span of 6AU (Source: Argonne Vehicle Database)

A similar analysis is conducted on other transmission types and numbers of gears considered for the study. Table 22 summarizes the gear span values implemented across the different transmission types and gear numbers.

While we recognize that some transmissions currently in the market have a higher gear span, the study focuses on selecting assumptions on the basis of the overall market. Separate studies will be conducted to quantify the impact of uncertainties related to variations of the assumptions.

5.3.2.2. Final Drive Ratio

Figure 125 shows the analysis done on the existing vehicle attributes to determine the final drive ratio of 6-speed automatic transmissions.

Figure 125. Vehicle Attribute Analysis for Final Drive Ratio of 6AU (Source: Argonne Vehicle Database)

A similar analysis was conducted on other transmission types and number of gears considered for the study. Table 22 summarizes the final drive ratio values implemented across the different transmission types and gear numbers.

5.3.2.3. Gear Ratio Selected for Each Transmission

Table 22 also summarizes the gear ratios selected for the different transmissions for the study.

Table 22. Gear Ratios, Gear Span and Final Drive Ratio Selected for Different Transmissions

Simulation Name	Gear										Gear Span	Final Drive
	1	2	3	4	5	6	7	8	9	10		
5AU	3.85	2.3262	1.5039	1.0403	0.77						5	3.31
5DM	3.85	2.2714	1.4339	0.9685	0.7						5.5	3.6
6AU	4.074	2.4867	1.6241	1.135	0.8487	0.679					6	3.65
6DM	4.074	2.4867	1.6241	1.135	0.8487	0.679					6	3.65
6DCT	4.074	2.4867	1.6241	1.135	0.8487	0.679					6	3.65
7AU+	4.78	3.10	1.98	1.37	1.00	0.87	0.78				6.16	3.13
7DM	4.298	2.624	1.7141	1.1981	0.8961	0.7171	0.614				7	3.5
8AU/+ / ++	4.284	2.6593	1.7763	1.2553	0.9546	0.7768	0.6763	0.63			6.8	3.6
8DCT	4.284	2.6593	1.7763	1.2553	0.9546	0.7768	0.6763	0.63			6.8	3.6
9AU+	4.69	2.902	1.9213	1.3611	1.0317	0.8368	0.7262	0.6743	0.67		7	3.3
10AU+	4.7	2.99	2.15	1.8	1.52	1.28	1	0.85	0.69	0.64	7.34	3.31
10AU++	5.25	3.27	2.19	1.6	1.3	1	0.78	0.65	0.58	0.52	10.10	3.55
CVT	Ratios from 0.529 to 3.172											4.44
CVTp	Ratios from 0.45 to 3.6											4.44
Planetary Gear	Sun = 30, Ring = 78											3.267
Voltec	Sun = 37, Ring = 83											3.02

5.3.3. Transmission Weights

Like the vehicle component weights, the weights for transmission system components were updated after the midterm TAR comments. The weights have been evaluated from the A2Mac1 database (A2Mac1, 2018) for different transmission types and numbers of gears for the different vehicle classes and performance categories.

5.3.3.1. Final Drive Weight

Figure 126 shows the distribution of final drive weights for 6-speed automatic transmissions for different vehicle classes.

Figure 126. Final Drive Weight Selection for 6AU (A2Mac1, 2018)

Such analyses were performed across different transmissions, numbers of gears, vehicle classes and performance categories and the values for the reference vehicles defined.

Table 23 (below) summarizes the values defined for the final drive weights of the reference vehicles with different transmissions across different numbers of gears, vehicle classes and performance categories.

5.3.3.2. Gearbox Weight

Gearbox weights have been re-evaluated from the A2Mac1 database (A2Mac1, 2018) and have been assigned across the different vehicle classifications and performance categories for the different transmission types and numbers of gears.

Figure 127 shows the distribution of gearbox weight for 6-speed automatic transmissions for the midsize vehicle class.

Figure 127. Gearbox Weight Selection for 6AU for Midsize Vehicles (A2Mac1, 2018)

Such analyses are performed across different transmissions, number of gears, vehicle classes and performance categories and the values for the reference vehicles defined. Table 24 summarizes the gearbox weights for different transmissions across different number of gears for the different vehicle classes and performance categories.

Table 23. Final Drive Weight Summary for All Transmission Types

Vehicle Class	Performance Category	Reference Value (kg)													
		5AU	6AU	7AU	8AU	9AU	10AU	5DM	6DM	7DM	6DCT	8DCT	CVT/+	Power-Split	EREV
Compact	Base/Premium	14	14	14	14	14	14	12	14	14	14	14	14	14	14
Midsize	Base/Premium	17	17	17	17	17	17	12	14	14	14	14	14	14	14
Small SUV	Base/Premium	20	22	22	22	22	22	24	24	24	24	24	24	24	24
Midsize SUV	Base/Premium	25	30	30	30	30	30	35	35	35	35	35	35	35	35
Pickup	Base/Premium	60	70	72	75	75	75	60	65	65	65	65	65	65	65

Table 24. Gearbox Weight Summary for All Transmission Types

Vehicle Class	Performance Category	Reference Value (kg)													
		5AU	6AU	7AU	8AU	9AU	10AU	5DM	6DM	7DM	6DCT	8DCT	CVT/+	Power-Split	EREV
Compact	Base/Premium	60	50	60	65	70	40	30	40	50	65	80	41	40	50
Midsize	Base/Premium	65	60	70	80	85	50	35	45	50	70	90	51	40	50
Small SUV	Base/Premium	70	65	72	80	90	55	45	50	50	75	90	56	50	60
Midsize SUV	Base/Premium	80	65	72	80	90	75	45	50	70	80	90	56	50	60
Pickup	Base/Premium	80	75	80	90	95	85	50	60	70	90	100	65	50	60

5.3.4. Transmission Performance Data

This section details the transmission losses for the different numbers of ratios for each of the transmission types considered in the study.

5.3.4.1. Automatic Transmission

Figure 128 shows the transmission efficiency maps for the different numbers of gears for automatic (AU/AU+/AU++) transmissions.

Figure 128. Automatic (AU/AU+/AU++) Transmission Efficiency Maps for Different Number of Gears

5.3.4.2. Manual Transmissions

Figure 129 shows the transmission efficiency maps for the different number of gears for manual (DM) transmissions.

Figure 129. Manual (DM) Transmission Efficiency Maps for Different Number of Gears

5.3.4.3. Dual-Clutch Transmission

Figure 130 shows the transmission efficiency maps for the different numbers of gears for dual clutch transmissions (DCTs).

Figure 130. DCT Efficiency Maps for Different Numbers of Gears

5.3.4.4. Continuously Variable Transmission

Continuously variable transmission (CVT) system loss consists of the hydraulic loss and the mechanical loss. The hydraulic loss comes from the pump loss to provide the CVT clamping force when the pump generates line pressure. The mechanical loss is caused by the torque loss from the slip between pulley and belt. The hydraulic loss has a dominant influence on the total CVT system efficiency when the vehicle is driving at low speed, while the mechanical loss is the main part of the CVT system loss at high speed. The CVT system model was developed by considering the hydraulic and mechanical losses, using experiment-based map data as mentioned earlier.

CVT Efficiency Maps

Figure 131 shows the oil pump efficiency map and the mechanical efficiency maps for different vehicle speeds for the CVT transmission type.

Figure 131. CVT Oil Pump and Mechanical Efficiency Maps

CVT+ Efficiency Maps

Figure 132 shows the oil pump efficiency map and the mechanical efficiency maps for different vehicle speeds for the CVT+ transmission type.

Figure 132. CVT+ Oil Pump and Mechanical Efficiency Maps

5.3.5. Gear-Shifting Maps

Examples of shifting maps for the midsize (base) category are presented below in Figure 133. Shifting maps have been developed to ensure minimum energy consumption across all transmissions while maintaining an acceptable drivability. While plant models with a higher degree of fidelity would be necessary to accurately model the impact of each technology on drivability, using such models was not appropriate for the current study. As a result, the work related to drive quality was focused on number of shifting events, time in between shifting events, engine time response, and engine torque reserve.

Figure 133Figure 133 shows the upshifting and downshifting maps for the different transmission types and numbers of gears for gasoline conventional powertrains in midsize (base) vehicles.

Figure 133. Maps of Transmission Upshifting (Solid Lines) and Downshifting (Dotted Lines)

5.4. Torque Converter Assumptions

5.4.1. Component Performance

Figure 134 shows the characteristic map for the torque converter data.

Figure 134. Autonomie Torque Converter Characteristic Map

The relationships of the different factors are as follows:

$$\text{Speed ratio} = \frac{\text{output speed}}{\text{input speed}}$$

$$\text{Torque ratio} = \frac{\text{output torque}}{\text{input torque}}$$

$$K \text{ factor} = \frac{\text{input speed}}{\text{input torque}}$$

5.4.2. Oil Pump Efficiency

The equation below calculates the pump torque loss map as a function of speed and torque for the torque converter.

$$(1) T_{\text{loss_map}_{i,j}} = (1 - \eta_{i,j}) T_{\text{loss_index}_i}$$

This torque loss represents an external pump driving the hydraulic fluid through the torque converter, not for the turbine.

$$(2) T_{in} = T_{eng} - T_{\text{loss_pump}}$$

Figure 135 shows the pump efficiency map for the torque converter model.

Figure 135. Pump Efficiency of the Torque Converter Model

5.5. Engine Technologies

5.5.1. Engine Technology Definitions

Table 25 details the list of different engine technologies studied, along with the associated simulation names and the reference peak power. The engine performance values were provided by IAV Automotive Engineering, Inc., except for the eng24 and eng25 values, which were provided by the U.S. EPA, and eng26 values, which were derived from Argonne AMTL test data.

Table 25. Engine Technologies with Reference Peak Power and Reference Displacement

Engine Simulation Name	Engine Technology	Engine Reference Peak Power (kW)	Engine Reference Displacement (L)	Engine Turbo Max Boost Level
eng01	DOHC VVT	108	2.0	
eng02	DOHC VVT+VVL	108	2.0	
eng03	DOHC VVT+VVL+GDI	113	2.0	
eng04	DOHC VVT+VVL+GDI+DEAC	113	2.0	
eng5b	SOHC VVT (level 1 reduced friction)	109	2.0	
eng6a	SOHC VVT+VVL (level 1 reduced friction)	109	2.0	
eng7a	SOHC VVT+VVL+GDI (level 1 reduced friction)	114	2.0	
eng8a	SOHC VVT+VVL+GDI+DEAC (level 1 reduced friction)	114	2.0	
eng12	DOHC Turbo 1.6l 18bar	132	1.6	1.5271
eng13	DOHC Turbo 1.2l 24bar	133	1.2	2.0409

Engine Simulation Name	Engine Technology	Engine Reference Peak Power (kW)	Engine Reference Displacement (L)	Engine Turbo Max Boost Level
eng14	DOHC Turbo 1.2l 24bar + cooled EGR	133	1.2	2.0409
eng17	Diesel	140	2.2	
eng18	DOHC VVT + SGDI	113	2.0	
eng19	DOHC VVT + DEAC	113	2.0	
eng20	DOHC VVT + VVL + DEAC	113	2.0	
eng21	DOHC VVT + SGDI + DEAC	113	2.0	
eng24	Current SkyActiv 2.0l 93AKI (U.S. EPA)	101	2.0	
eng25	Future SkyActiv 2.0l CEGR 93AKI+DEAC (U.S. EPA)	101	2.0	
eng26	Atkinson (Argonne) – PSHEV and PSPHEV only	73	1.8	
eng12Deac	DOHC Turbo 1.6l 18bar + DEAC	144	1.6	1.5271
eng22b	Atkinson 2.5L VVT PFI CR14	132	2.5	
eng23b	2.0 Miller VTG + VVT + VVL + DI + cEGR + CR12	139	2.0	
eng23c	Miller eCharger + VVT + DI + eEGR + CR12	139	2.0	
eng26a	VCR VVT + DI + Turbo + cEGR + CR9/12	180	2.0	2.0870

5.5.2. Engine Efficiency Maps

IAV Automotive Engineering modeled gasoline and diesel engine maps in GT-POWER and supplied those maps to Argonne for use in Autonomie.

IAV Automotive Engineering also provided wide-open-throttle engine performance values and brake-specific fuel consumption (BSFC) maps for future engine concepts. To properly quantify the benefits of individual technologies, each incremental technology was modeled using GT-POWER and validated with existing dynamometer measurements for several engines. GT-POWER is used to predict engine performance quantities such as power, torque, airflow, volumetric efficiency, fuel consumption, turbocharger performance, and matching and pumping losses. The models were trained over the entire engine operating range and have predictive combustion capability. This is essential, since the BSFC prediction needs to be accurate while the engine setup is subject to change.

Relevant engine geometries and parameters were measured and modeled with friction/flow losses, heat transfer, and other parameters and calibrated to match measurements. Displacement-normalized mechanical friction was modeled as a function of engine speed and specific load. A combustion model was trained to predict fuel heat release rates in response to physical effects such as cylinder geometries, pressure, temperature, turbulence, residual gas concentration and other parameters. A knock correlation based on in-cylinder conditions and fuel octane rating predicts whether knock will occur and

at what intensity. A combustion stability threshold prediction was trained using covariance of IMEP data and is used for understanding exhaust gas recirculation (EGR) tolerance, especially at low loads. Load controllers were developed for fuel/air path actuators and targeting controllers to drive optimal and knock-limited combustion phasing, just as in a physical engine. Careful modeling practice was used to provide confidence that calibrations will scale and predict reasonable/reliable results as parameters are changed throughout the various technology concept studies.

IAV Automotive Engineering provided 20 engine maps in total, for 15 naturally aspirated gasoline engines, four turbocharged gasoline engines, and one diesel engine. Two naturally aspirated gasoline engine maps (for eng24 and eng25) were provided by the U.S. Environmental Protection Agency (EPA). EPA benchmarked the Skyactiv 2.0L 13:1 compression ratio engine in an EPA engine test cell. Finally, one Atkinson engine map, generated using Argonne test data, was used for electrified vehicles with power-split architecture. Thus, the total number of engine maps used in the study is 19.

For all IAV Automotive Engineering engines, engine speed, BMEP, brake torque, fuel flow rate, PMEP, and FMEP data were provided in a standardized format to Argonne. These channels were provided from 1,000 RPM to the maximum engine speed and from 0 bar BMEP to full load to provide a full operation map. Fuel flow rates at zero output torque were provided separately from 650 RPM (defined idle) to 6000 RPM.

The following fuel specifications are used by IAV when modeling the engines in GT-POWER:

- Gasoline LHV = 43.1 MJ/kg
- $(R+M)/2 = 87$
- Ambient temperature, $T_{amb} = 25\text{ }^{\circ}\text{C}$
- Ambient pressure, $P_{amb} = 990\text{ mbar}$

Table 26 lists the detailed characteristics of fuel used by IAV Automotive Engineering when modeling the engines in GT-POWER.

Table 26. Characteristics of Fuel Used for Modeling Engines

Type Of Fluid	Composition	Molecular Formula			Density	Lower Heating Value
	Mass fraction	C	H	O	kg/m ³	MJ/kg
Hydrocarbon	0.903712493	8	14.851265	0	741.9	43.19
Ethanol	0.094801493	2	6	1	785	26.9
Water	0.001486014	0	2	1	1002.5	0

5.5.2.1. ENGINE (eng01) 2L_pfi_dohc_vvt_engine1_baseline

Engine 1 is a naturally aspirated PFI 2.0-L gasoline engine with variable valve timing engine technology (VVT) from a MY 2013 vehicle. A BSFC engine map was generated from dynamometer testing of the existing engine, which then served as the baseline BSFC map for all simulated naturally aspirated engines (Engines 1-8a, 18-21). The engine calibrations are fully optimized for best BSFC and maximum torque. Figure 136 shows the BSFC map for engine 1.

Figure 137. Engine 2 BSFC Map

5.5.2.3. ENGINE (eng03) 2L_di_engine3

For engine 3, the port fuel injection (PFI) from engine 2 was converted to direct injection (DI), and the compression ratio was raised from 10.2 to 11.0, with injection timing optimized. The benefit of this technology is that DI provides greater knock tolerance, allowing higher compression ratios and increased efficiency over the entire map. Figure 138 shows the BSFC map for engine 3.

Figure 138. Engine 3 BSFC Map

5.5.2.4. ENGINE (eng04) 2L_cylinder_deac_engine4

For engine 4, cylinder deactivation capabilities were added to engine 3. The engine fires only two cylinders at low loads and at speeds below 3000 RPM by deactivating valves on two cylinders. The additional benefit of this technology is that the effective load doubles on two cylinders, providing less pumping work and higher efficiency. Figure 139 shows the BSFC map for engine 4.

Figure 139. Engine 4 BSFC Map

5.5.2.5. ENGINE (eng5b) 2L_engine5b_SOHC_low_friction

For engine 5b, the valve timing was optimized for a fixed-overlap camshaft with a standard friction model from dual overhead camshaft (DOHC) concepts. Then the engine FMEP was reduced by 0.1 bar over the entire operation range to evaluate the friction benefit from a single overhead camshaft (SOHC). The benefit of this technology is reduced friction, which improves efficiency at all load points and also raises the full-load line. Figure 140 shows the BSFC map for engine 5b.

Figure 140. Engine 5b BSFC Map

5.5.2.6. ENGINE (eng6a) 2L_engine6a_VVL_low_friction

For engine 6a, the engine 2 FMEP was reduced by 0.1 bar over the entire operation range. The benefit of this change is similar to that for engine 5b, with improved efficiency at all load points and the full-load line raised compared with engine 2. Figure 141 shows the BSFC map for engine 6a.

Figure 141. Engine 6a BSFC Map

5.5.2.7. ENGINE (eng7a) 2L_engine7a_DI_low_friction

For engine 7a, the engine 3 FMEP was reduced by 0.1 bar over the entire operation range. As mentioned earlier, the reduced friction improves efficiency at all load points and raises the full-load line. Figure 142 shows the BSFC map for engine 7a.

Figure 142. Engine 7a BSFC Map

5.5.2.8. ENGINE (eng8a) 2L_engine8a_cylinder_deac_low_friction

For engine 8a, the engine 4 FMEP was reduced by 0.1 bar over the entire operation range. The reduced friction improves efficiency at all loads and raises the full-load line. Figure 143 shows the BSFC map for engine 8a.

Figure 143. Engine 8a BSFC Map

5.5.2.9. ENGINE (eng12) 1pt6L_engine12_turbo_DI_DOHC_VVT_VVL

Engine 12 is a 1.6-L, 4-cylinder, turbocharged gasoline engine with DI, DOHC, dual-cam variable valve timing (VVT), and intake variable valve lift (VVL). The calibrations for the engine are fully optimized for the best BSFC. Figure 144 shows the BSFC map for engine 12.

Figure 144. Engine 12 BSFC Map

5.5.2.10. ENGINE (eng13) 1pt2L_engine13_turbo_DI_DOHC_VVT_VVL

For engine 13, engine 12 was downsized to 1.2 L. The turbocharger maps were scaled to improve torque at low engine speeds. The downsizing allows for operation at a higher engine load point (increased efficiency) at a given vehicle torque demand. Figure 145 shows the BSFC map for engine 13.

Figure 145. Engine 13 BSFC Map

5.5.2.11. ENGINE (eng14) 1pt2L_engine14_turbo_external_cool_EGR

For engine 14, high-pressure cooled exhaust gas recirculation (EGR) was added to engine 13 (IAV Northville, 2014). The cooled EGR target set points were further optimized. There are multiple benefits from the application of cooled EGR:

- The cooled burned gas lowers in-cylinder temperatures, reducing knock tendency and thus improving combustion phasing.
- Reduced in-cylinder temperatures lead to reduced exhaust temperatures and therefore a reduced need for enrichment to protect exhaust components.

Figure 146 shows the BSFC map for engine 14.

Figure 146. Engine 14 BSFC Map

5.5.2.12. ENGINE (eng17) eng_plant_ci_2pt2L_IAV_Engine17

Engine 17 is a 2.2-L, 4-cylinder diesel engine. Figure 147 shows the BSFC map for engine 17.

Figure 147. Engine 17 BSFC map

5.5.2.13. ENGINE (eng18) 2L_di_dohc_vvt_engine18

Engine 18 is a 2.0-L, 4-cylinder, naturally aspirated gasoline engine with DOHC, dual VVT, and DI. The engine was developed from engine 1, with increased knock resistance and volumetric efficiency due to in-cylinder vaporization of the fuel. Also, open-valve injection and homogeneous operation were assumed (IAV Inc., 2016). Figure 148 shows the BSFC map for engine 18.

Figure 148. Engine 18 BSFC Map

5.5.2.14. ENGINE (eng19) 2L_cylinder_deac_engine19

Engine 19 is a 2.0-L, 4-cylinder, naturally aspirated PFI gasoline engine with DOHC, dual VVT, and cylinder deactivation (DEAC) capability. The engine was developed from engine 1, with the VVT timing map of active cylinders based on the cylinder indicated mean effective pressure (IMEP) of engine 1 (IAV Inc., 2016). Figure 149 shows the BSFC map for engine 19.

Figure 149. Engine 19 BSFC Map

5.5.2.15. ENGINE (eng20) 2L_cylinder_deac_engine20

Engine 20 is a 2.0-L, 4-cylinder, naturally aspirated PFI engine with DOHC, dual VVT, intake VVL, and DEAC abilities. The engine was developed from engine 2. The VVT maps and intake valve map lift of active cylinders are based on the cylinder IMEP of engine 2 (IAV Inc., 2016). Figure 150 shows the BSFC map for engine 20.

Figure 150. Engine 20 BSFC Map

5.5.2.16. ENGINE (eng21) 2L_cylinder_deac_engine21

Engine 21 is a 2.0-L, 4-cylinder, naturally aspirated engine with DOHC, dual-cam VVT, DI and DEAC abilities. The engine was developed from engine 18. The VVT timing map of active cylinders is based on the cylinder IMEP of engine 18 (IAV Inc., 2016). Figure 151 shows the BSFC map for engine 21.

Figure 151. Engine 21 BSFC Map

5.5.2.17. ENGINE (eng22b) 2L_engine22b_Atkinson_2.5L_VVT_CR14

Engine 22b is a 2.5-L, 4-cylinder, Atkinson naturally aspirated PFI engine with DOHC and dual cam VVT and a compression-ratio of 14. Figure 152 shows the BSFC map for engine 22b.

Figure 152. Engine 22b BSFC Map

5.5.2.18. ENGINE (eng23b) 2L_engine23b_VVT_VVL_DI_cEGR_CR12

Engine 23b is a 2.0-L, 4-cylinder, turbocharged DI engine with DOHC, dual cam VVT, intake VVL, and cooled EGR capabilities. The engine has a compression ratio of 12. Figure 153 shows the BSFC map for engine 23b.

Figure 153. Engine 23b BSFC Map

5.5.2.19. ENGINE (eng23c) 2L_engine23c_Miller_eCharger_VVT_DI_eEGR_CR12

Engine 23c is a 2.0-L, 4-cylinder, turbocharged DI engine with DOHC, dual cam VVT, and cooled EGR capabilities. The engine has a compression ratio of 12. The turbocharging technology has an eCharger to enhance e-boost. Figure 154 shows the engine BSFC map for engine 23c.

Figure 154. Engine 23c BSFC map

5.5.2.20. ENGINE (eng24) 2014_mazda_skyactiv_2p0L_USTier2

Engine 24 is a Mazda SKYACTIV® 2.0-L engine using Tier 2 fuel. The engine efficiency map has been provided by the U.S. EPA (Ellies, 2016). Figure 155 shows the BSFC map for engine 24.

Figure 155. Engine 24 BSFC Map

5.5.2.21. *ENGINE (eng25) future_mazda_skyactiv_2p0L_atkinson_cyl_deac*

Engine 25 is the future Atkinson engine with cooled EGR and DEAC. The data for the engine has been provided by the U.S. EPA (US Environmental Protection Agency, 2016). The engine is based on the Mazda SKYACTIV® 2.0-L engine. The EPA tested a GM 4.3-L EcoTec engine with cylinder deactivation to implement the DEAC abilities in the SKYACTIV® engine (Stuhldreher, 2016). Figure 156 shows the BSFC map for engine 25.

Figure 156. Engine 25 BSFC Map

5.5.2.22. ENGINE (eng26) Atkinson

Engine 26 is a 1.8-L, 4-cylinder 73kW Atkinson engine. The data for the engine comes from 2010 Toyota Prius AMTL test data. Figure 157 shows the BSFC map for engine 26.

Figure 157. Engine 26 BSFC Map

5.5.2.23. ENGINE (eng26a) 2L_engine26a_VCR_VVT_DI_Turbo_cEGR_CR9/12

Engine 26a is a 2.0-L, 4-cylinder, turbocharged DI engine with DOHC, dual cam VVT, and cooled EGR capabilities. The engine has a variable compression ratio of 9/12. Figure 158 shows the engine BSFC map for engine 26a. The BSFC map demonstrates the fuel map from merging the two separate maps of individual compression ratios.

Figure 158. Engine 26a BSFC Map

5.5.3. Incremental BSFC and Thermal Efficiency Difference of Engines

Figure 159 shows the incremental differences (%) in BSFC and thermal efficiency among the different engines.

5.5.4. Engine Efficiency Map Comparison With and Without DEAC Capability

Figure 160 shows the engine efficiency difference between engine 1 and engine 19 (with DEAC capability added to engine 1).

Figure 160. Engine Efficiency Difference Between Engine 19 and Engine 1

Figure 161 shows the engine efficiency difference between engine 2 and engine 20 (with DEAC capability added to engine 2).

Figure 161. Engine Efficiency Difference Between Engine 20 and Engine 2

Figure 162 shows the engine efficiency difference between engine 18 and engine 21 (with DEAC capability added to engine 18).

Figure 162. Engine Efficiency Difference Between Engine 21 and Engine 18

5.5.5. Idle Fuel Rate Update

For NPRM runs, Autonomie simulation extrapolated the engine map fuel rate data provided from IAV Automotive Engineering to determine the idle fuel flow rate based on Willans line method (Pachernegg, 1969). The Willans line extrapolation to a zero line of torque input provides a useful approximation of the idle fuel flow rate. For FRM runs, the values for the idle fuel flow rates have been updated to specific values provided by IAV. The difference in the idle fuel flow rates are summarized in Table 27.

Table 27. IAV Idle Fuel Rate for All Engines

Engine List	Comments	Fuel Flow (kg/hr) 650 – 700rpm Warm Idle	
		NPRM	FRM
Eng1	Gasoline, 2.0 l, 4 cyl, NA, PFI, DOHC, VVT	0.3394	0.364
Eng2	VVL added to Eng1	0.3403	0.355
Eng3	DI added to Eng2	0.3283	0.350
Eng4	Cylinder deactivation ability added to Eng3	0.270/0.3283	0.270/0.380
Eng5b	Eng5a with valvetrain friction reduction (small friction reduction)	0.3347	0.339
Eng6a	Eng2 with valvetrain friction reduction (small friction reduction)	0.3152	0.315
Eng7a	Eng3 with valvetrain friction reduction (small friction reduction)	0.3035	0.308
Eng8a	Eng4 with valvetrain friction reduction (small friction reduction)	0.3035	0.216
Eng12	Gasoline, 1.6 l, 4 cyl, turbocharged, DI, DOHC, VVT, VVL	0.2152	0.230
Eng12Deac	Cylinder deactivation ability added to Eng12	-	0.230
Eng13	Eng12 downsized to 1.2 l	0.1656	0.200
Eng14	Cooled external EGR added to Eng13	0.1656	0.200

Engine List	Comments	Fuel Flow (kg/hr)	
		650 – 700rpm Warm Idle	
Eng18	Gasoline, 2.0 l, 4 cyl, NA, DI, DOHC, VVT	0.3387	0.380
Eng19	Cylinder deactivation ability added to Eng1	0.270/0.3394	0.270/0.380
Eng20	Cylinder deactivation ability added to Eng2	0.270/0.3403	0.270/0.380
Eng21	Cylinder deactivation ability added to Eng18	0.270/0.3387	0.270/0.380
Eng22b	2.5liter Atkinson naturally aspirated	-	0.364
Eng23b	2.0liter VVL Miller + VTG + EGR	-	0.229
Eng23c	2.0liter VVT Miller echarge + EGR	-	0.229
Eng24	Current SkyActiv 2.0L 93AKI	0.3405	0.3405
Eng25	Future SkyActiv 2.0L CEGR 93AKI + DEAC	0.4075	0.4075
Eng26a	2.0liter variable compression ratio + EGR	-	0.229

5.6. Electric Machines

The electric machine in belt-integrated starter generator (BISG) hybrid vehicles captures regenerative braking energy and provides minimal assist to the engine during high-transient operating modes. Because the electric machine is linked to the engine through a belt, its power is usually limited. A value of 10 kW peak power was assigned to the BISG for this study.

The maps were developed assuming normal-temperature operating conditions. Electric machine inverter losses are included. The electric machine's power, like the engine's, is sized for the reference-sized powertrains. Table 28 details the electric machine efficiency map sources for the different powertrain configurations.

Table 28. Electric Machine Efficiency Map Sources for Different Powertrain Configurations

Powertrain Type	Source of Efficiency Map for Motor1 (Traction Motor) + Inverter	Source of Efficiency Map for Motor2 (Motor/Generator) + Inverter
Micro 12-V HEV, BISG	Camry EM1 data (Oak Ridge National Laboratory, 2008)	
Parallel HEV	Sonata HEV data (Oak Ridge National Laboratory, 2011)	
Split HEV and Blended PHEV	Camry EM1 data (Oak Ridge National Laboratory, 2008)	Camry EM2 data (Oak Ridge National Laboratory, 2008)
EREV PHEV	Camry EM1 data (Oak Ridge National Laboratory, 2008)	Sonata HEV data (Oak Ridge National Laboratory, 2011)
BEV	Chevrolet Bolt EM data (Faizul Momen, 2018)	

Powertrain Type	Source of Efficiency Map for Motor1 (Traction Motor) + Inverter	Source of Efficiency Map for Motor2 (Motor/Generator) + Inverter
Fuel-cell HEV	Nissan Leaf data (Oak Ridge National Laboratory, 2011)	

5.6.1. Electric Machine Efficiency Maps

Figure 163 shows the electric machine efficiency maps for different powertrains.

Figure 163. Electric Machines Efficiency Maps for Different Powertrains

5.6.2. Electric Machine Peak Efficiency scaling

For NPRM runs, the peak efficiency of electric machines was scaled according to DOE VTO targets for different electrified powertrains. For FRM runs, the peak efficiency of electric machines for different powertrains are scaled as shown in Table 29.

Table 29. Efficiency Scaling of Electric Machines

Vehicle Powertrain	Peak Efficiency Scaled (%)
Micro HEV / Mild Hybrid BISG / Split HEV / Par HEV	96
Split PHEV20 / EREV PHEV50 / Par PHEV20 / Par PHEV50	96
BEV & FCEV	98

A constant ratio was assumed between the continuous and peak torque curves, as follows:

2 for micro-HEV, BISG, and CISG

2 for motor 1 and 1.5 for the motor 2 of the power-split HEV, blended PHEV and Voltec PHEV

1 for BEVs and fuel-cell HEV

5.7. Fuel Cell System

The fuel cell system was modeled to represent hydrogen consumption as a function of produced power. The system's peak efficiency is 60%, including balance of plant, and represents normal-temperature operating conditions. The data set cannot be provided here because it is proprietary. The system's specific power is 650 W/kg.

The hydrogen storage technology selected is a high-pressure tank with a specific weight of 0.04 kg H₂/kg, sized to provide a 320-mile range on the combined cycle (UDDS + HWFET) on the basis of adjusted energy values.

Figure 164 shows the fuel cell efficiency curve with respect to fuel cell power.

Figure 164. Fuel Cell Efficiency vs. Power

5.8. Energy Storage System

Lithium-ion batteries are used for all different hybrid powertrains. Different useable SOC ranges have been selected based on the powertrain configuration:

10% SOC range for micro and mild HEVs

20% SOC range for full HEVs

70% SOC range for PHEVs

90% SOC range for BEVs

Vehicle test data have shown that for the U.S. standard drive cycles and test conditions considered, battery cooling does not draw a significant amount of energy for most of the vehicle powertrain architectures (Hanho Son, 2015). The exception is high-energy PHEVs (Namdoo Kim, 2017) and BEVs (Kim & Jeong, 2017), for which an additional constant power draw is used for battery cooling.

For FRM runs, the mass, capacities, and the voltages of the batteries across different vehicle classes and powertrains are computed using BatPaC model described in Section 5.9.

5.9. Battery Performance and Cost Model (BatPaC)

BatPaC, a lithium-ion battery performance and cost model for electric-drive vehicles, was developed by the Chemical Sciences and Engineering division at Argonne (CSE, ANL, 2017).

To accelerate the large-scale simulation process, a lookup table replaces the traditional BatPaC model initially developed in Microsoft Excel®. The lookup table is dependent on powertrain options. It consists of four dimensions in parallel — pack power, pack energy, number of cells, and number of cell modules. The cell chemistry and heat transfer fluid is fixed for each powertrain. Additional lookup tables for other parameters of the BatPaC model (e.g., positive thickness, negative thickness, battery open circuit voltage, and battery pack volume) are also generated as constraints.

5.9.1. BatPaC Process

The default BatPaC model spreadsheet comes with a set of default specifications comparing seven different batteries for HEVs, PHEVs and BEVs (Table 30).

Table 30. BatPaC Input Details

Key Inputs	Tab/Sheet	Row No.
Vehicle type (micro HEV, HEV-HP, PHEV, EV)	Battery Design	F59
Battery energy storage, kWh	Battery Design	202-204
Pulse power requirement, kW	Battery Design	62
Production volume, packs per year	Battery Design	76
Cathode/anode combination	Chem	E4

5.9.1.1. Cathode/Anode Combination Selection

When one of the seven electrode combinations in the Chem tab is studied, the selected electrode combination is copied from F4-L4 and pasted into E4. All the properties for this combination become the default values and will be used in the calculations. The default values used in the calculations associated with the selected electrode combination will be listed in Column E of the tab, and the user can override the default material property or price by entering a new value in Column N. Figure 165 shows the BatPaC model input for cathode/anode selection.

	A	B	C	D	E	F	G	H	I	J	K	L
1	Cell Chemistry											
2	Selected System						Default Values					
3												75%
4					NMC622-G	NCA-G	NMC622-G	NMC333-G	LFP-G	LMO-LTO	LMO-G	NMC/x%LMO-G

Figure 165. BatPaC Cathode Anode Selection

5.9.1.2. Vehicle Pack Specifications Selection

Vehicle and pack specifications are provided in the Battery Design tab shown in Figure 166.

	A	B	C	D	E	F	G	H	I	J	K	L
1	Program for Calculating Performance and Materials Requirements											
2	Li1.05(Ni0.6Mn0.2Co0.2)0.95O2-Graphite											
3						Battery 1	Battery 2	Battery 3	Battery 4	Battery 5	Battery 6	Battery 7
58	Battery Pack Parameters											
59	Vehicle type (microHEV, HEV-HP, PHEV, EV)					EV	EV	EV	EV	EV	EV	EV
60	Pack heat transfer fluid (EG-W, CA, CoolA)					EG-W	EG-W	EG-W	EG-W	EG-W	EG-W	EG-W
61	Designated duration of power pulse for application (10 or 2), s					10	10	10	10	10	10	10
62	Target battery pack power, kW					180	220	260	300	120	400	360
63	Approximate target power for 30-sec pulse (EV application), kW					172	210	249	287	115	383	344
64	Estimated pack power at target % OCV, kW					697	832	962	1086	329	1003	1633
65	Number of cells per module					12	12	12	12	14	12	18
66	Number of cells in parallel					1	1	1	1	1	1	2
67	Number of modules in row					5	5	5	5	5	5	4
68	Number of rows of modules per pack					4	4	4	4	2	4	6
69	Number of modules per battery pack					20	20	20	20	10	20	24
70	Number of modules in parallel					1	1	1	1	1	1	1
71	Cells per battery pack					240	240	240	240	140	240	432
72	Total cells per battery system					240	240	240	240	140	240	432
73	Voltage drop for bus bar for packs with one row of modules, V					0.03	0.03	0.03	0.03	0.03	0.03	0.03
74	Battery pack insulation thickness, mm					10	10	10	10	10	10	10
75	Battery jacket total thickness, mm					14	14	14	14	14	14	14
76	Number of battery systems manufactured per year					100,000	100,000	100,000	100,000	100,000	100,000	100,000

Figure 166. BatPaC Battery Design Tab

The cell and module configurations can be configured for different battery packs. These parameters affect the battery pack voltage, the cell dimensions and mass, the modules, and the pack.

Fast charging requirements could drive the battery cost by limiting the thickness of the electrodes. Figure 167 shows the BatPaC Fast Charging sheet. The additional cost of a fast charging constraint is displayed in Row 55 of the sheet. For FRM rulemaking runs, the fast charge constraint was disabled by setting cell E10 to 0.

1	Rapid Charging of Electric Vehicles (EVs)							
2								
3	Basic Pack Design to Achieve Discharge Requirements	Battery 1	Battery 2	Battery 3	Battery 4	Battery 5	Battery 6	Battery 7
10	Charging rate: 1C rate (0), desired charging time (1)	1						
24	Charging Time							
25	Maximum charger power, kW							
26	Charging time for 60% addition to SOC (15 to 75% SOC), minutes							
27	Desired charging time	10	10	10	10	10	10	10
33	Total time charging 80% SOC (15 to 95% SOC, 94% useable), min	15.6	15.6	15.6	15.6	15.6	15.6	15.6
37	Temperature Control							
38	Heat generated charging 60% SOC (15 to 75% SOC, 71% useable), kWh	1.531	1.909	2.299	2.702	0.644	2.373	3.565
39	Heat generated charging 80% SOC (15 to 95% SOC, 94% useable), kWh	1.850	2.306	2.778	3.264	0.778	2.867	4.308
43	Initial pack temperature, °C	15	15	15	15	15	15	15
44	Maximum allowed temperature at center of cells, °C	40	40	40	40	40	40	40
45	Estimated heat capacity of pack, J/g-°C	0.85	0.85	0.85	0.85	0.85	0.85	0.85
46	Mass of battery pack less 50% of jacket mass, kg	292.7	349.3	405.3	461.0	139.7	420.4	656.7
47	Battery temperature at end of charging 60% SOC with no cooling, °C	37.2	38.1	39.0	39.8	34.5	38.9	38.0
48	Battery temperature at end of charging 80% SOC with no cooling, °C	41.8	43.0	44.0	45.0	38.6	43.9	42.8
55	Extra cost of meeting fast charging requirements, \$	409	635	867	1,108	0	925	1,570

Figure 167. BatPaC Fast Charging Tab

BatPaC provides performance metric and cost parameters for the different battery packs designed, as summarized in Table 31.

Table 31. BatPaC Output Performance Metric

Performance Metrics	Cost Parameters
Cell, Module, and Pack Level	Cell, Module, and Pack
Dimensions	Cost of production
Mass	Cost breakdown
Volume	
Voltage	
Wh/kg, Wh/liter	
Material inventory	

5.9.2. BatPaC Examples from Existing Vehicles in the Market

To estimate battery performance For FRM runs, existing battery packs from vehicles in the market, ranging across different vehicle classes and powertrains, have been used to determine the assumptions set in BatPaC and the configuration of the battery packs. These configurations were studied using the default BatPac assumptions, except for battery pack configurations.

5.9.2.1. Full HEV

The battery cell teardown analysis reports for the following vehicles were used to evaluate the assumptions for full HEV:

- 2016 Toyota Prius
- 2013 Ford Fusion HEV
- 2014 Honda Civic HEV
- 2014 VW Jetta Hybrid

The battery pack parameters for these vehicles, determined using A2mac1 (A2Mac1, 2018) are shown in Table 32.

Table 32. Battery Pack Details for Full HEVs

Parameter	2016 Toyota Prius	2013 Ford Fusion HEV	2014 Honda Civic HEV	2014 VW Jetta Hybrid
Cells/module	28	38	10	15
Modules/pack	2	2	4	4
Total number of cells	56	76	40	60
Pack voltage (V)	204.4	280	144	220
Cathode type	NMC111	NMC	NMC	NMC 111
Cell capacity (Ah)	3.62	5	4.7	5
Cooling medium	Air cooled	Forced air	Forced air	Forced air
Pack mass (kg)	33.44	44.1	22	57

From BatPaC analysis, the results shown in Table 33 are deduced for these battery packs

Table 33. BatPaC Analysis of HEVs

Vehicle	Battery Pack Manuf. Cost (\$)	Battery Pack Mass (kg)	Battery Cell Capacity (Ah)	Battery Pack Nominal Voltage (V)
2016 Toyota Prius	1,190	16	6.4	205.6
2013 Ford Fusion HEV	1,457	21.9	5	279
2014 Honda Civic HEV	983	10.6	4.8	146.8
2014 VW Jetta Hybrid	1,177	13.44	5.1	220.3

5.9.2.2. PHEV20 AER

The battery cell teardown analysis reports from A2mac1 for the following vehicles were used to evaluate the assumptions used for plug-in hybrid 20 AERs:

- 2016 Prius Prime
- 2016 Mercedes-Benz GLE 550e PHEV
- 2016 BMW X5 xDrive40e PHEV

The results for these vehicles, determined using A2mac1 (A2Mac1, 2018), are shown in Table 34.

Table 34. Battery Pack Details for PHEV20 AERs

Parameter	2016 Toyota Prius Prime	2016 Mercedes-Benz GLE 550e PHEV	2016 BMW X5 xDrive40e PHEV
Cell/ module	-	20	16
Modules/pack	-	6	6
Total number of cells	95	120	96
Pack voltage (V)	351.5	380.4	355
Cathode type	NMC	LFP	NCA/NMC111
Cell capacity (Ah)	-	22.57	27.89
Cooling medium	Refrigerant	Liquid with EG	Refrigerant
Pack mass (kg)	120	133.675	115.51

From BatPaC analysis, the results shown in Table 35 are deduced for these battery packs.

Table 35. BatPaC Analysis of PHEV20 AER

Vehicle	Battery Pack Manuf. Cost (\$)	Battery Pack Mass (kg)	Battery Cell Capacity (Ah)	Battery Pack Nominal Voltage (V)
2016 BMW X5	2,848	71.7	25.6	360
2016 Mercedes-Benz GLE 550e PHEV	2,959	70.4	19.3	450

5.9.2.3. PHEV50 AER

The 2016 Chevrolet Volt battery cell teardown analysis from A2mac1 (A2Mac1, 2018) was used to evaluate the assumptions in BatPaC for PHEV50 AERs. The different battery pack parameters for the 2016 Chevrolet Volt are shown in Table 36.

Table 36. Battery Pack Details of PHEV50 AERs

Parameter	Value
Cell/ module	24 & 32
Modules/pack	7
Total number of cells	192
Cathode type	NMC523/LMO
Pack voltage (V)	351.4
Cell capacity (Ah)	25.27
Pack mass (kg)	215.5
Cooling medium	Liquid cooled

From BatPaC analysis, the results shown in Table 37 are deduced for the battery pack of the 2016 Chevrolet Volt.

Table 37: BatPaC Analysis of PHEV50 AER

Vehicle	Battery Pack Manuf. Cost (\$)	Battery Pack Mass (kg)	Battery Cell Capacity (Ah)	Battery Pack Nominal Voltage (V)
2016 Chevrolet Volt	4,335	167.5	25.6	360

5.9.2.4. BEV200 AER

The 2017 Chevrolet Bolt battery cell teardown analysis from A2mac1 (A2Mac1, 2018) was used to evaluate the assumptions in BatPaC for BEV200 AERs. The different battery pack parameters for the 2017 Chevrolet Bolt are shown in Table 38.

Table 38. BatPaC Analysis of BEV200 AERs

Parameter	Value
Total number of cells	288
Cathode type	NMC622
Pack voltage (V)	355
Cell capacity (Ah)	60
Pack mass (kg)	426.639
Cooling medium	Liquid cooled

From BatPaC analysis, the results shown in Table 39 are deduced.

Table 39. Battery Pack Details of BEV200 AERs

Vehicle	Battery Pack Manuf. Cost (\$)	Battery Pack Mass (kg)	Battery Cell Capacity (Ah)	Battery Pack Nom. Voltage (V)
2017 Chevrolet Bolt	8,583	342.5	55.9	360

5.9.2.5. BEV300 AER

The 2018 Hyundai Kona battery cell teardown analysis from A2mac1 (A2Mac1, 2018) was used to evaluate the assumptions in BatPaC for BEV300 AERs. From BatPaC analysis, the estimated results shown in Table 40 are deduced for the 2018 Hyundai Kona battery pack.

Table 40. BatPaC analysis of BEV300AERs

Vehicle	Battery Pack Manuf. Cost (\$)	Battery Pack Mass (kg)	Battery Cell Capacity (Ah)	Battery Pack Nom. Voltage (V)
2018 Hyundai Kona	9219	376.2	40.9	393.8

5.9.3. Use of BatPaC in Autonomie for FRM runs

For FRM runs, a significant change was made to the BatPaC inputs to Autonomie. Different sets of battery pack configurations were set for different vehicle classes and powertrains. These include the following:

- Number of cells per module
- Number of cells in parallel
- Number of modules in row
- Number of rows of modules per pack
- Number of modules per battery pack
- Number of modules in parallel

BatPaC provides Autonomie the battery pack mass and cost values. The generated lookup tables are used when sizing the batteries to get the accurate battery pack mass. BatPaC would further provide the cell capacity in ampere hours (Ah). Table 41 shows the different set of assumptions set in BatPaC for micro HEVs for five different vehicle classes.

Table 41. BatPaC Assumptions for Micro HEV

Vehicle Class	Compact – Midsize	Small SUV – Pickup
Cell chemistry	LFP-G	LFP-G
Cooling type	CoolA	CoolA
Numbers of cells per module	4	4
Number of cells in parallel	1	1
Number of modules in row	1	1
Number of rows of modules per pack	1	1
Number of modules per battery pack	1	1
Number of modules in parallel	1	1

Vehicle Class	Compact – Midsize	Small SUV – Pickup
Cells per battery pack	4	4
Maximum thickness limit, um	100	100
Number of battery produced per year per plant	100000	100000
Nominal pack voltage	13.1	13.1

Table 42 below provides the different set of assumptions set in BatPaC for mild hybrid BISG for five different vehicle classes.

Table 42. BatPaC Assumptions for Mild Hybrid BISG

Vehicle Class	Compact – Midsize	Small SUV – Pickup
Cell chemistry	LFP-G	LFP-G
Cooling type	CoolA	CoolA
Numbers of cells per module	14	14
Number of cells in parallel	1	1
Number of modules in row	1	1
Number of rows of modules per pack	1	1
Number of modules per battery pack	1	1
Number of modules in parallel	1	1
Cells per battery pack	14	14
Maximum thickness limit, um	100	100
Number of battery produced per year	100000	100000
Nominal pack voltage	45.9	45.9

Table 43 below provides the different set of assumptions set in BatPaC for full HEVs for five different vehicle classes.

Table 43. BatPaC Assumptions for Full HEVs

Vehicle Class	Compact – Midsize	Small SUV – Pickup
Cell chemistry	NMC622-G	NMC622-G
Cooling type	CoolA	CoolA
Numbers of cells per module	26	30
Number of cells in parallel	1	1
Number of modules in row	1	1
Number of rows of modules per pack	2	2
Number of modules per battery pack	2	2

Vehicle Class	Compact – Midsize	Small SUV – Pickup
Number of modules in parallel	1	1
Cells per battery pack	52	60
Maximum thickness limit, um	100	100
Plant efficiency (cell yield), %	85	85
Number of battery produced per year	25000	25000
Nominal pack voltage	195	225

Table 44 below provides the different set of assumptions set in BatPaC for power-split PHEV 20 AER/par PHEV 20 AER for five different vehicle classes.

Table 44. BatPaC Assumptions for PHEV20 AER

Vehicle Class	Compact – Midsize	Small SUV – Pickup
Cell chemistry	NMC622-G	NMC622-G
Cooling type	EG-W	EG-W
Numbers of cells per module	20	24
Number of cells in parallel	1	1
Number of modules in row	2	2
Number of rows of modules per pack	2	2
Number of modules per battery pack	4	4
Number of modules in parallel	1	1
Cells per battery pack	80	96
Maximum thickness limit, um	100	100
Plant efficiency (cell yield), %	85	85
Number of battery produced per year	25000	25000
Nominal pack voltage	300	360

Table 45 below provides the different set of assumptions set in BatPaC for PHEV 50 AERs for five different vehicle classes.

Table 45. BatPaC Assumptions for PHEV50 AER

Vehicle Class	Compact – Midsize	Small SUV – Pickup
Cell chemistry	NMC622-G	NMC622-G
Cooling type	EG-W	EG-W
Numbers of cells per module	24	25
Number of cells in parallel	2	2

Vehicle Class	Compact – Midsize	Small SUV – Pickup
Number of modules in row	2	2
Number of rows of modules per pack	4	4
Number of modules per battery pack	8	8
Number of modules in parallel	1	1
Cells per battery pack	192	200
Maximum thickness limit, um	100	100
Plant efficiency (cell yield), %	85	85
Number of battery produced per year	25000	25000
Nominal pack voltage	360	375

Table 46 below provides the different set of assumptions set in BatPaC for BEV 200 AERs for five different vehicle classes.

Table 46. BatPaC Assumptions for BEV200 AERs

Vehicle Class	Compact – Midsize	Small SUV – Pickup
Cell chemistry	NMC622-G	NMC622-G
Cooling type	EG-W	EG-W
Numbers of cells per module	16	19
Number of cells in parallel	4	4
Number of modules in row	5	5
Number of rows of modules per pack	4	4
Number of modules per battery pack	20	20
number of modules in parallel	1	1
Cells per battery pack	320	380
Maximum thickness limit, um	-	-
Plant efficiency (cell yield), %	85	85
Number of battery produced per year	25000	25000
Nominal pack voltage	300	356.3

Table 47 below provides the different set of assumptions set in BatPaC for BEV 300 AERs for five different vehicle classes.

Table 47. BatPaC Assumptions for BEV300 AERs

Vehicle Class	Compact – Midsize	Small SUV – Pickup
Cell chemistry	NMC622-G	NMC622-G
Cooling type	EG-W	EG-W
Numbers of cells per module	19	21
Number of cells in parallel	4	4
Number of modules in row	5	5
Number of rows of modules per pack	4	4
Number of modules per battery pack	20	20
number of modules in parallel	1	1
Cells per battery pack	380	420
Maximum thickness limit, um	-	-
Plant efficiency (cell yield), %	85	85
Number of battery produced per year	25000	25000
Nominal pack voltage	356.3	393.8

5.9.3.1. Setting Assumptions in BatPaC for FRM runs

This section details how the BatPaC files are set up for different powertrains.

Vehicle Type

In the Battery Design tab of the BatPaC file, the respective vehicle powertrain type (microHEV, HEV-HP, PHEV, EV) is copied and pasted into cells F59:L59.

	A	B	C	D	E	F	G	H	I	J	K	L
1	Program for Calculating Performance and Materials Requirements											
2	Li1.05(Ni0.6Mn0.2Co0.2)0.95O2-Graphite											
3						Battery 1	Battery 2	Battery 3	Battery 4	Battery 5	Battery 6	Battery 7
58	Battery Pack Parameters											
59	Vehicle type (microHEV, HEV-HP, PHEV, EV)					EV	EV	EV	EV	EV	EV	EV

Figure 168. Setting Vehicle Type

Cell Chemistry

In the Chem tab of the BatPaC file, the respective cell chemistries defined in the section above in the corresponding column are selected and copied to cell E4. For example, Figure 169 shows the cell chemistry selection for BEV200.

	A	B	C	D	E	F	G	H	I	J	K	L
1	Cell Chemistry											
2					Selected System	Default Values						
3												75%
4					NMC622-G	NCA-G	NMC622-G	NMC333-G	LFP-G	LMO-LTO	LMO-G	NMC/x%LMO-G
5	Positive Electrode											
6	Active material molecular weight				94.39	96.10	94.39	93.88	157.76	176.02	176.02	155.5
7	Active material capacity, mAh/g:				180.00	200	180	155	150	108	100	113.8
8	Weight %											
9	Active material				89	89	89	89	89	89	89	89
10	Carbon				6	6	6	6	6	6	6	6
11	Binder				5	5	5	5	5	5	5	5
12	Binder solvent				NMP	NMP	NMP	NMP	NMP	NMP	NMP	NMP
13	Void, Vol% %				32	32	32	32	50	32	32	32
14	Density, g/cm ³											
15	Active material				4.65	4.78	4.65	4.65	3.45	4.23	4.23	4.34
16	Carbon				1.825	1.825	1.825	1.825	1.825	1.825	1.825	1.825
17	Binder				1.77	1.77	1.77	1.77	1.77	1.77	1.77	1.77
18	Maximum thickness limit, µm				200							
19	Minimum thickness limit, µm				15							

Figure 169. Cell Chemistry for BEV200

Cooling Type

In the Battery Design tab of the BatPaC file, the cooling types corresponding to the different powertrains are placed in cells F60:L60.

	A	B	C	D	E	F	G	H	I	J	K	L
1	Program for Calculating Performance and Materials Requirements											
2	Li1.05(Ni0.6Mn0.2Co0.2)0.95O2-Graphite											
3						Battery 1	Battery 2	Battery 3	Battery 4	Battery 5	Battery 6	Battery 7
60	Pack heat transfer fluid (EG-W, CA, CoolA)					EG-W	EG-W	EG-W	EG-W	EG-W	EG-W	EG-W

Figure 170. Setting Cooling Type

Battery Design

In the Battery Design tab, the battery design information (number of cells per module, number of cells in parallel, number of modules in row, number of rows of modules per pack, number of modules per battery pack, number of modules in parallel, and cells per battery pack) is set in cells F65:L72.

	A	B	C	D	E	F	G	H	I	J	K	L
1	Program for Calculating Performance and Materials Requirements											
2	Li1.05(Ni0.6Mn0.2Co0.2)0.95O2-Graphite											
3						Battery 1	Battery 2	Battery 3	Battery 4	Battery 5	Battery 6	Battery 7
65	Number of cells per module					12	12	12	12	12	12	12
66	Number of cells in parallel					3	3	3	3	3	3	3
67	Number of modules in row					5	5	5	5	5	5	5
68	Number of rows of modules per pack					4	4	4	4	4	4	4
69	Number of modules per battery pack					20	20	20	20	20	20	20
70	Number of modules in parallel					1	1	1	1	1	1	1
71	Cells per battery pack					240	240	240	240	240	240	240
72	Total cells per battery system					240	240	240	240	240	240	240

Figure 171. Battery Design Information

Maximum Thickness Limit

Based on expert feedback from the developers of the BatPaC model, a maximum thickness limit is specified for some of the powertrain batteries. In the Chem tab of the BatPaC file, the maximum thickness value is placed in cell N18. This value was not overwritten for BEV200 and BEV300 AERs.

	A	B	C	D	E	F	G	H	I	J	K	L	M	N
1	Cell Chemistry													
2	Selected System							Default Values						
3												75%		Override
4					NMC622-G	NCA-G	NMC622-G	NMC333-G	LFP-G	LMO-LTO	LMO-G	NMC/x%LMO-G	Other	Values
18	Maximum thickness limit, μm				100									100

Figure 172. Setting Maximum Thickness

Plant Efficiency (%), Yield

In the Cost Input tab, the cell yield % value is placed in cell E4. A constant value of 85% is used for FRM analysis.

	A	B	C	D	E	F	G	H	I	J
1	Cost Input									
2										
3	Materials Yields During Processing				Cost inputs that vary with cell chemistry					
4	Cell yield, %				85	Cost inputs that are do not vary with chemistry				

Figure 173. Setting Plant Efficiency

Number of Battery Systems Manufactured per Year

In the Battery Design tab, the number of battery systems manufactured per year is placed in cell F76. A constant production volume of 25,000 battery systems is assumed across all the different battery types.

	A	B	C	D	E	F	G	H	I	J	K	L
1	Program for Calculating Performance and Materials Requirements											
2	Li1.05(Ni0.6Mn0.2Co0.2)0.95O2-Graphite											
3						Battery 1	Battery 2	Battery 3	Battery 4	Battery 5	Battery 6	Battery 7
76	Number of battery systems manufactured per year					25,000	25,000	25,000	25,000	25,000	25,000	25,000

Figure 174. Battery Systems Manufactured per Year

5.9.3.2. Inputs to BatPaC from Autonomie

The following inputs are send to BatPaC from Autonomie simulations:

Battery pack power (kW)

Total battery pack energy (kWh)

	A	B	C	D	E	F	G	H	I	J	K	L
1	Program for Calculating Performance and Materials Requirements											
2	Li1.05(Ni0.6Mn0.2Co0.2)0.95O2-Graphite											
3						Battery 1	Battery 2	Battery 3	Battery 4	Battery 5	Battery 6	Battery 7
62	Target battery pack power, kW					110	110	150	150	150	150	150

	A	B	C	D	E	F	G	H	I	J	K	L
1	Program for Calculating Performance and Materials Requirements											
2	Li1.05(Ni0.6Mn0.2Co0.2)0.95O2-Graphite											
3						Battery 1	Battery 2	Battery 3	Battery 4	Battery 5	Battery 6	Battery 7
196	Energy requirement of vehicle on UDDS cycle (default = 250), Wh/mile					250.0	250.0	250.0	250.0	250.0	250.0	250.0

Figure 175. Autonomie Inputs to BatPaC

5.9.3.3. Outputs from BatPaC into Autonomie

The following outputs are sent to Autonomie from BatPaC:

Estimated total battery cost (\$)

Total battery mass (kg)

Battery pack capacity (Ah)

Nominal battery system voltage (V)

	A	B	C	D	E	F	G	H	I	J	K	L
1	Program for Calculating Performance and Materials Requirements											
2	Li1.05(Ni0.6Mn0.2Co0.2)0.95O2-Graphite											
3						Battery 1	Battery 2	Battery 3	Battery 4	Battery 5	Battery 6	Battery 7
225			Estimated Total Battery Cost to OEM, US\$			8,427	8,427	9,711	9,711	9,711	9,711	9,711

	A	B	C	D	E	F	G	H	I	J	K	L
1	Program for Calculating Performance and Materials Requirements											
2	Li1.05(Ni0.6Mn0.2Co0.2)0.95O2-Graphite											
3						Battery 1	Battery 2	Battery 3	Battery 4	Battery 5	Battery 6	Battery 7
220			Mass, kg			346.0	346.0	419.7	419.7	419.7	419.7	419.7

	A	B	C	D	E	F	G	H	I	J	K	L
1	Program for Calculating Performance and Materials Requirements											
2	Li1.05(Ni0.6Mn0.2Co0.2)0.95O2-Graphite											
3						Battery 1	Battery 2	Battery 3	Battery 4	Battery 5	Battery 6	Battery 7
151	Calculated Battery Pack Parameters											
152	Battery pack capacity, Ah					211.3	211.3	268.6	268.6	268.6	268.6	268.6

	A	B	C	D	E	F	G	H	I	J	K	L
1	Program for Calculating Performance and Materials Requirements											
2	Li1.05(Ni0.6Mn0.2Co0.2)0.95O2-Graphite											
3						Battery 1	Battery 2	Battery 3	Battery 4	Battery 5	Battery 6	Battery 7
57	Nominal battery system voltage, V					300.0	300.0	300.0	300.0	300.0	300.0	300.0

Figure 176. BatPaC Outputs to Autonomie

5.9.4. BatPaC Lookup Tables

5.9.4.1. Micro HEV

For micro HEVs, the costs of additional pack level component (pack hardware, TMS, high voltage wiring, battery management unit) are subtracted. Table 48 below summarizes the final lookup tables generated for micro-HEVs for BatPaC cost and cell capacity.

Table 48. BatPaC Lookup Table for Micro HEVs

			Energy, kWh				
Micro HEV			0.6	0.8	1.0	1.2	1.4
BatPac Cost	Power, kW	0.5	\$ 211	\$ 255	\$ 283	\$ 320	\$ 354
		1.1	\$ 210	\$ 255	\$ 283	\$ 320	\$ 354
		1.5	\$ 210	\$ 255	\$ 283	\$ 320	\$ 354
		2.0	\$ 210	\$ 255	\$ 283	\$ 320	\$ 354
		2.5	\$ 210	\$ 255	\$ 283	\$ 320	\$ 354
		3.0	\$ 210	\$ 255	\$ 283	\$ 320	\$ 354
BatPac Cell Capacity (Ah)	Power, kW	0.5	46.1	64.5	76.9	92.3	107.7
		1.1	46.1	64.5	76.9	92.3	107.7
		1.5	46.1	64.5	76.9	92.3	107.7
		2.0	46.1	64.5	76.9	92.3	107.7
		2.5	46.1	64.5	76.9	92.3	107.7
		3.0	46.1	64.5	76.9	92.3	107.7

The battery mass of micro HEVs differs across classes and performance categories and is defined in the assumptions sheet, shown in Table 49.

Table 49. Mass Assumptions for Micro HEVs

Vehicle Class	Performance Category	Battery Mass (kg)	
		Gasoline	Diesel
Compact	Base	15	20
Compact	Premium	25	30
Midsize	Base	18	20
Midsize	Premium	28	30
Small SUV	Base	18	20
Small SUV	Premium	28	30
Midsize SUV	Base	20	25
Midsize SUV	Premium	30	35
Pickup	Base	20	20
Pickup	Premium	30	35

5.9.4.2. Mild Hybrid BISG

For mild hybrid BISGs, mass and costs of additional pack level components (pack hardware, TMS, high voltage wiring) are subtracted. Table 50 below summarizes the final lookup tables generated from BatPaC for mild hybrid BISG battery packs for cost, mass and cell capacity.

Table 50. BatPaC Lookup Table for Mild Hybrid BISGs

		Mild BISG	Energy, kWh					
			0.30	0.40	0.60	0.81	1.00	1.20
BatPac Cost	Power, kW	5.00	\$378	\$392	\$429	\$466	\$500	\$534
		6.00	\$383	\$397	\$429	\$466	\$500	\$534
		7.69	\$393	\$406	\$432	\$466	\$500	\$534
		8.00	\$394	\$407	\$433	\$466	\$500	\$534
		9.00	\$400	\$413	\$438	\$466	\$500	\$534
		10.00	\$406	\$418	\$443	\$469	\$500	\$534
BatPac Mass (kg)	Power, kW	5.00	4.9	5.5	7.1	8.7	10.2	11.8
		6.00	5.0	5.7	7.1	8.7	10.2	11.8
		7.69	5.3	5.9	7.2	8.7	10.2	11.8
		8.00	5.4	6.0	7.2	8.7	10.2	11.8
		9.00	5.5	6.1	7.4	8.7	10.2	11.8
		10.00	5.7	6.3	7.5	8.8	10.2	11.8
BatPac Cell Capacity (Ah)	Power, kW	5.00	6.56	8.83	13.15	17.67	21.93	26.32
		6.00	6.56	8.82	13.15	17.67	21.93	26.32
		7.69	6.55	8.81	13.15	17.67	21.93	26.32
		8.00	6.55	8.81	13.14	17.67	21.93	26.32
		9.00	6.55	8.80	13.13	17.67	21.93	26.32
		10.00	6.54	8.80	13.12	17.66	21.93	26.32

The mild hybrid BISGs battery packs considered in FRM rulemaking consist of 0.403 kWh energy with 7.69 kW power output. As a result, BatPaC provides a cost of \$406.00 with a pack mass value of 5.7 kg.

5.9.4.3. Full Hybrid HEV

Table 51 below summarizes the cost, mass and cell capacities generated for compact and midsize full hybrids from BatPaC using the assumptions specified in Table 51.

Table 51. BatPaC Lookup Table for Full Hybrids (Compact/Midsize)

	HEV		Energy, kWh													
			0.9	1.0	1.2	1.4	1.6	1.8	2.0	2.2	2.4	2.6	2.8	3.0		
BatPac Cost	Power, kW	10.0	\$1,295	\$1,314	\$1,349	\$1,384	\$1,418	\$1,451	\$1,484	\$1,516	\$1,548	\$1,579	\$1,610	\$1,641		
		20.0	\$1,311	\$1,322	\$1,354	\$1,388	\$1,422	\$1,455	\$1,488	\$1,520	\$1,552	\$1,583	\$1,614	\$1,645		
		30.0	\$1,356	\$1,365	\$1,384	\$1,405	\$1,427	\$1,459	\$1,492	\$1,524	\$1,556	\$1,587	\$1,618	\$1,649		
		40.0	\$1,416	\$1,418	\$1,429	\$1,446	\$1,465	\$1,485	\$1,506	\$1,529	\$1,560	\$1,591	\$1,622	\$1,653		
		60.0						\$1,552	\$1,558	\$1,570	\$1,585	\$1,602	\$1,620	\$1,639	\$1,658	\$1,678
		80.0									\$1,685	\$1,694	\$1,706	\$1,720	\$1,735	\$1,752
		100.0										\$1,811	\$1,817	\$1,826	\$1,838	
BatPac Mass (kg)	Power, kW	10.0	10.68	11.23	12.33	13.43	14.52	15.61	16.69	17.77	18.84	19.91	20.97	22.03		
		20.0	11.01	11.44	12.46	13.55	14.63	15.71	16.79	17.86	18.93	20.00	21.06	22.11		
		30.0	12.01	12.35	13.11	13.93	14.78	15.84	16.90	17.97	19.03	20.10	21.15	22.21		
		40.0	13.52	13.66	14.19	14.87	15.63	16.42	17.23	18.11	19.16	20.21	21.27	22.32		
		60.0						17.58	17.96	18.51	19.16	19.86	20.60	21.36	22.13	22.93
		80.0									21.78	22.23	22.78	23.41	24.09	24.79
		100.0										25.64	26.01	26.49	27.05	
BatPac Cell Capacity (Ah)	Power, kW	10.0	4.63	5.15	6.18	7.21	8.24	9.27	10.30	11.33	12.36	13.39	14.42	15.45		
		20.0	4.63	5.15	6.18	7.21	8.24	9.27	10.30	11.33	12.36	13.39	14.42	15.45		
		30.0	4.62	5.14	6.17	7.21	8.24	9.27	10.30	11.33	12.36	13.39	14.42	15.45		
		40.0	4.62	5.14	6.17	7.20	8.23	9.26	10.30	11.33	12.36	13.39	14.42	15.45		
		60.0						7.19	8.22	9.25	10.28	11.31	12.34	13.38	14.41	15.44
		80.0									10.27	11.30	12.33	13.36	14.39	15.43
		100.0										12.32	13.35	14.38	15.41	

Table 52 summarizes the cost, mass and cell capacities generated for full hybrid SUVs and pickups from BatPaC, using the assumptions specified.

Table 52. BatPaC Lookup Table for Full Hybrids (SUVs/Pickups)

	HEV		Energy, kWh											
			0.9	1.0	1.2	1.4	1.6	1.8	2.0	2.2	2.4	2.6	2.8	3.0
BatPac Cost	Power, kW	10.0	\$1,375	\$1,393	\$1,429	\$1,463	\$1,497	\$1,531	\$1,563	\$1,596	\$1,628	\$1,659	\$1,690	\$1,721
		20.0	\$1,390	\$1,401	\$1,433	\$1,467	\$1,501	\$1,534	\$1,567	\$1,599	\$1,631	\$1,663	\$1,694	\$1,724
		30.0	\$1,435	\$1,444	\$1,463	\$1,484	\$1,506	\$1,539	\$1,571	\$1,603	\$1,635	\$1,666	\$1,697	\$1,728
		40.0	\$1,495	\$1,497	\$1,508	\$1,525	\$1,543	\$1,563	\$1,584	\$1,607	\$1,639	\$1,668	\$1,699	\$1,730
		60.0				\$1,628	\$1,634	\$1,646	\$1,661	\$1,678	\$1,695	\$1,714	\$1,733	\$1,753
		80.0						\$1,757	\$1,760	\$1,768	\$1,780	\$1,794	\$1,810	\$1,826
		100.0									\$1,883	\$1,888	\$1,898	\$1,909
BatPac Mass (kg)	Power, kW	10.0	10.87	11.43	12.53	13.64	14.74	15.83	16.91	17.99	19.07	20.13	21.20	22.26
		20.0	11.18	11.62	12.64	13.74	14.83	15.91	17.00	18.07	19.15	20.21	21.27	22.34
		30.0	12.19	12.53	13.29	14.11	14.95	16.02	17.10	18.17	19.24	20.30	21.36	22.42
		40.0	13.71	13.85	14.37	15.05	15.80	16.59	17.40	18.28	19.34	20.40	21.46	22.51
		60.0				17.74	18.12	18.67	19.31	20.01	20.74	21.51	22.29	23.08
		80.0						21.64	21.91	22.35	22.91	23.53	24.20	24.90
		100.0									25.72	26.09	26.57	27.13
BatPac Cell Capacity (Ah)	Power, kW	10.0	4.02	4.46	5.36	6.25	7.14	8.03	8.93	9.82	10.71	11.61	12.50	13.39
		20.0	4.01	4.46	5.36	6.25	7.14	8.03	8.93	9.82	10.71	11.61	12.50	13.39
		30.0	4.01	4.45	5.35	6.24	7.14	8.03	8.93	9.82	10.71	11.61	12.50	13.39
		40.0	4.00	4.45	5.34	6.24	7.13	8.03	8.92	9.82	10.71	11.61	12.50	13.39
		60.0				6.23	7.12	8.02	8.91	9.80	10.70	11.59	12.49	13.39
		80.0						8.01	8.90	9.79	10.69	11.58	12.47	13.37
		100.0									10.68	11.57	12.47	13.36

5.9.4.4. PHEV 20 AER (Power-Split/Parallel)

Table 53 summarizes the cost, mass and cell capacities generated for compact and midsize PHEV20 AERs from BatPaC, using the assumptions specified.

Table 53. BatPaC Lookup Table for PHEV20 AERs (Compact/Midsize)

		PHEV20	Energy, kWh		
			5.0	10.0	20.0
BatPac Cost	Power, kW	30.0	\$2,760	\$3,434	\$4,588
		40.0	\$2,764	\$3,438	\$4,592
		60.0	\$2,770	\$3,444	\$4,598
		80.0	\$2,777	\$3,450	\$4,604
		100.0	\$2,786	\$3,456	\$4,610
		120.0	\$2,793	\$3,465	\$4,616
		140.0	\$2,866	\$3,472	\$4,625
		160.0	\$2,997	\$3,479	\$4,631
		200.0		\$3,524	\$4,647
		240.0		\$3,727	\$4,664
		280.0		\$3,960	\$4,684
BatPac Mass (kg)	Power, kW	30.0	45.7	72.6	121.6
		40.0	45.8	72.7	121.7
		60.0	46.0	72.9	121.9
		80.0	46.2	73.1	122.0
		100.0	46.5	73.3	122.2
		120.0	46.7	73.5	122.5
		140.0	48.7	73.8	122.7
		160.0	52.3	74.0	122.9
		200.0		75.3	123.4
		240.0		80.9	123.9
		280.0		87.5	124.5
BatPac Cell Capacity (Ah)	Power, kW	30.0	16.7	33.4	66.9
		40.0	16.7	33.4	66.9
		60.0	16.7	33.4	66.9
		80.0	16.7	33.4	66.9
		100.0	16.7	33.4	66.9
		120.0	16.7	33.4	66.9
		140.0	16.7	33.4	66.9
		160.0	16.7	33.4	66.9
		200.0		33.4	66.9
		240.0		33.4	66.9
		280.0		33.4	66.9

Table 54 summarizes the cost, mass and cell capacities generated for SUV and pickup PHEV20 AERs from BatPaC using the assumptions specified.

Table 54. BatPaC Lookup Table for PHEV20 AERs (SUVs/Pickup)

			Energy, kWh		
			5.0	10.0	20.0
BatPac Cost	Power, kW	30.0	\$2,910	\$3,586	\$4,742
		40.0	\$2,911	\$3,587	\$4,744
		60.0	\$2,917	\$3,593	\$4,749
		80.0	\$2,923	\$3,598	\$4,755
		100.0	\$2,931	\$3,604	\$4,760
		120.0	\$2,937	\$3,610	\$4,765
		140.0	\$3,002	\$3,618	\$4,771
		160.0	\$3,131	\$3,624	\$4,777
		200.0		\$3,652	\$4,791
		240.0		\$3,852	\$4,803
		280.0		\$4,077	\$4,819
BatPac Mass (kg)	Power, kW	30.0	47.2	74.4	123.8
		40.0	47.3	74.5	123.9
		60.0	47.4	74.6	124.0
		80.0	47.6	74.8	124.2
		100.0	47.8	75.0	124.4
		120.0	48.0	75.2	124.5
		140.0	49.8	75.4	124.7
		160.0	53.3	75.6	124.9
		200.0		76.5	125.3
		240.0		81.9	125.7
		280.0		88.2	126.2
BatPac Cell Capacity (Ah)	Power, kW	30.0	13.9	27.9	55.8
		40.0	13.9	27.9	55.8
		60.0	13.9	27.9	55.8
		80.0	13.9	27.9	55.8
		100.0	13.9	27.9	55.8
		120.0	13.9	27.9	55.8
		140.0	13.9	27.9	55.8
		160.0	13.9	27.9	55.8
		200.0		27.9	55.8
		240.0		27.8	55.8
		280.0		27.8	55.8

5.9.4.5. PHEV50 AER (Voltec EREV/Parallel)

Table 55 summarizes the cost, mass and cell capacities generated for compact and midsize PHEV50 AERs from BatPaC using the assumptions specified.

Table 55. BatPaC Lookup Table for PHEV50 AERs (Compact/Midsize)

			Energy, kWh					
PHEV50			10.0	20.0	30.0	40.0	50.0	60.0
BatPac Cost	Power, kW	60.0	\$ 4,344	\$ 5,525	\$ 6,557	\$ 7,528	\$ 8,458	\$ 9,393
		80.0	\$ 4,352	\$ 5,532	\$ 6,564	\$ 7,536	\$ 8,465	\$ 9,400
		100.0	\$ 4,360	\$ 5,540	\$ 6,572	\$ 7,544	\$ 8,473	\$ 9,408
		120.0	\$ 4,369	\$ 5,548	\$ 6,580	\$ 7,551	\$ 8,481	\$ 9,416
		140.0	\$ 4,377	\$ 5,557	\$ 6,588	\$ 7,559	\$ 8,489	\$ 9,423
		160.0	\$ 4,389	\$ 5,565	\$ 6,596	\$ 7,567	\$ 8,497	\$ 9,431
		200.0	\$ 4,409	\$ 5,582	\$ 6,613	\$ 7,584	\$ 8,513	\$ 9,447
		240.0	\$ 4,594	\$ 5,603	\$ 6,630	\$ 7,601	\$ 8,530	\$ 9,464
		280.0	\$ 4,818	\$ 5,623	\$ 6,651	\$ 7,618	\$ 8,547	\$ 9,481
BatPac Mass (kg)	Power, kW	60.0	83.49	136.23	181.83	226.25	269.46	311.93
		80.0	83.78	136.51	182.10	226.53	269.73	312.20
		100.0	84.08	136.79	182.38	226.80	270.00	312.47
		120.0	84.39	137.08	182.66	227.08	270.27	312.74
		140.0	84.73	137.38	182.95	227.36	270.56	313.02
		160.0	85.08	137.69	183.25	227.65	270.84	313.31
		200.0	85.87	138.34	183.86	228.25	271.43	313.89
		240.0	91.11	139.04	184.51	228.87	272.04	314.49
		280.0	97.74	139.81	185.20	229.52	272.67	315.11
BatPac Cell Capacity (Ah)	Power, kW	60.0	13.93	27.88	41.86	55.86	69.88	83.92
		80.0	13.93	27.88	41.86	55.86	69.88	83.92
		100.0	13.93	27.88	41.86	55.86	69.88	83.92
		120.0	13.93	27.88	41.86	55.86	69.88	83.92
		140.0	13.93	27.88	41.86	55.86	69.88	83.92
		160.0	13.93	27.88	41.86	55.86	69.88	83.92
		200.0	13.93	27.88	41.86	55.86	69.88	83.92
		240.0	13.92	27.88	41.86	55.86	69.88	83.92
		280.0	13.91	27.88	41.86	55.86	69.88	83.92

Table 56 summarizes the cost, mass and cell capacities generated for SUV and pickup PHEV50 AERs from BatPaC using the assumptions specified.

Table 56. BatPaC Lookup Table for PHEV50 AERs (SUVs/Pickups)

			Energy, kWh					
			10.0	20.0	30.0	40.0	50.0	60.0
BatPac Cost	Power, kW	60.0	\$4,399	\$5,581	\$6,613	\$7,585	\$8,515	\$9,451
		80.0	\$4,407	\$5,589	\$6,621	\$7,593	\$8,522	\$9,458
		100.0	\$4,415	\$5,596	\$6,628	\$7,600	\$8,530	\$9,465
		120.0	\$4,423	\$5,604	\$6,636	\$7,608	\$8,537	\$9,473
		140.0	\$4,432	\$5,612	\$6,644	\$7,615	\$8,545	\$9,480
		160.0	\$4,441	\$5,620	\$6,652	\$7,623	\$8,552	\$9,488
		200.0	\$4,463	\$5,637	\$6,668	\$7,639	\$8,568	\$9,501
		240.0	\$4,643	\$5,655	\$6,685	\$7,655	\$8,584	\$9,517
		280.0	\$4,867	\$5,676	\$6,702	\$7,672	\$8,601	\$9,534
BatPac Mass (kg)	Power, kW	60.0	84.13	137.05	182.61	227.09	270.33	312.83
		80.0	84.41	137.32	182.87	227.35	270.59	313.09
		100.0	84.70	137.59	183.14	227.61	270.85	313.35
		120.0	85.00	137.87	183.41	227.88	271.11	313.61
		140.0	85.32	138.16	183.69	228.15	271.38	313.88
		160.0	85.65	138.45	183.97	228.43	271.66	314.15
		200.0	86.41	139.07	184.56	229.00	272.22	314.71
		240.0	91.61	139.74	185.18	229.60	272.80	315.28
		280.0	98.21	140.48	185.84	230.23	273.41	315.88
BatPac Cell Capacity (Ah)	Power, kW	60.0	13.37	26.77	40.19	53.63	67.09	80.56
		80.0	13.37	26.77	40.19	53.63	67.09	80.56
		100.0	13.37	26.77	40.19	53.63	67.09	80.56
		120.0	13.37	26.77	40.19	53.63	67.08	80.56
		140.0	13.37	26.77	40.19	53.63	67.08	80.56
		160.0	13.37	26.77	40.19	53.63	67.08	80.56
		200.0	13.37	26.77	40.19	53.63	67.08	80.56
		240.0	13.36	26.77	40.19	53.63	67.08	80.56
		280.0	13.36	26.77	40.19	53.63	67.08	80.56

5.9.4.6. BEV200 AER

Table 57 summarizes the cost, mass and cell capacities generated for compact and midsize BEV200 AERs from BatPaC using the assumptions specified.

Table 57. BatPaC Lookup Table for BEV200 AERs (Compact/Midsize)

		BEV200	Energy, kWh			
			30.0	50.0	70.0	90.0
BatPac Cost	Power, kW	20.0	\$7,663	\$9,604	\$11,433	\$13,163
		40.0	\$7,680	\$9,621	\$11,450	\$13,180
		60.0	\$7,697	\$9,638	\$11,467	\$13,196
		80.0	\$7,714	\$9,655	\$11,484	\$13,213
		100.0	\$7,732	\$9,672	\$11,501	\$13,230
		120.0	\$7,749	\$9,690	\$11,518	\$13,247
		140.0	\$7,767	\$9,708	\$11,536	\$13,265
		160.0	\$7,786	\$9,725	\$11,553	\$13,282
		180.0	\$7,807	\$9,744	\$11,571	\$13,300
		200.0	\$7,826	\$9,764	\$11,591	\$13,320
		240.0	\$7,865	\$9,801	\$11,628	\$13,356
		280.0	\$7,908	\$9,842	\$11,665	\$13,393
		320.0	\$7,953	\$9,882	\$11,706	\$13,433
		400.0	\$8,052	\$9,969	\$11,787	\$13,512
BatPac Mass (kg)	Power, kW	20.0	207.85	297.53	383.40	468.34
		40.0	208.55	298.23	384.09	469.04
		60.0	209.27	298.93	384.80	469.74
		80.0	210.00	299.66	385.52	470.46
		100.0	210.74	300.39	386.25	471.18
		120.0	211.50	301.12	386.98	471.90
		140.0	212.27	301.87	387.72	472.64
		160.0	213.05	302.63	388.47	473.38
		180.0	213.85	303.41	389.22	474.13
		200.0	214.67	304.19	389.99	474.89
		240.0	216.36	305.78	391.55	476.43
		280.0	218.13	307.43	393.15	478.00
		320.0	220.00	309.14	394.80	479.61
		400.0	224.14	312.76	398.23	482.94
BatPac Cell Capacity (Ah)	Power, kW	20.0	25.10	41.89	58.71	75.57
		40.0	25.10	41.89	58.71	75.57
		60.0	25.10	41.89	58.71	75.57
		80.0	25.10	41.89	58.71	75.57
		100.0	25.10	41.89	58.71	75.56
		120.0	25.10	41.89	58.71	75.56
		140.0	25.10	41.89	58.71	75.56
		160.0	25.10	41.89	58.71	75.56
		180.0	25.10	41.89	58.71	75.56
		200.0	25.10	41.89	58.71	75.56
		240.0	25.10	41.89	58.71	75.56
		280.0	25.10	41.89	58.71	75.56
		320.0	25.10	41.89	58.71	75.56
		400.0	25.10	41.89	58.71	75.56

Table 58 summarizes the cost, mass and cell capacities generated for SUV and pickup BEV200 AERs from BatPaC using the assumptions specified.

Table 58. BatPaC Lookup Table for BEV200 AERs (SUVs/Pickup)

		BEV200	Energy, kWh				
			30.0	50.0	70.0	90.0	120.0
BatPac Cost	Power, kW	20.0	\$8,001	\$ 9,947	\$11,779	\$13,512	\$16,079
		40.0	\$8,015	\$ 9,962	\$11,794	\$13,526	\$16,093
		60.0	\$8,030	\$ 9,976	\$11,808	\$13,541	\$16,107
		80.0	\$8,045	\$ 9,991	\$11,823	\$13,555	\$16,122
		100.0	\$8,060	\$10,006	\$11,837	\$13,570	\$16,136
		120.0	\$8,075	\$10,021	\$11,852	\$13,585	\$16,151
		140.0	\$8,091	\$10,036	\$11,867	\$13,600	\$16,166
		160.0	\$8,107	\$10,051	\$11,883	\$13,615	\$16,181
		180.0	\$8,123	\$10,067	\$11,898	\$13,630	\$16,196
		200.0	\$8,139	\$10,083	\$11,913	\$13,645	\$16,211
		240.0	\$8,173	\$10,115	\$11,945	\$13,676	\$16,242
		280.0	\$8,210	\$10,148	\$11,977	\$13,708	\$16,273
		320.0	\$8,247	\$10,182	\$12,010	\$13,740	\$16,305
		400.0	\$8,329	\$10,255	\$12,078	\$13,807	\$16,370
BatPac Mass (kg)	Power, kW	20.0	215.23	305.78	392.25	476.97	601.47
		40.0	215.82	306.36	392.83	477.54	602.05
		60.0	216.42	306.96	393.42	478.13	602.64
		80.0	217.04	307.57	394.02	478.73	603.23
		100.0	217.66	308.18	394.63	479.33	603.84
		120.0	218.30	308.80	395.24	479.94	604.44
		140.0	218.95	309.43	395.87	480.56	605.06
		160.0	219.61	310.07	396.49	481.18	605.68
		180.0	220.28	310.72	397.13	481.81	606.30
		200.0	220.97	311.38	397.78	482.45	606.93
		240.0	222.39	312.72	399.09	483.74	608.20
		280.0	223.87	314.10	400.43	485.06	609.50
		320.0	225.44	315.54	401.81	486.41	610.83
		400.0	228.91	318.57	404.69	489.21	613.55
BatPac Cell Capacity (Ah)	Power, kW	20.0	21.14	35.28	49.44	63.63	84.95
		40.0	21.14	35.27	49.44	63.63	84.95
		60.0	21.14	35.27	49.44	63.63	84.95
		80.0	21.14	35.27	49.44	63.63	84.95
		100.0	21.14	35.27	49.44	63.63	84.95
		120.0	21.14	35.27	49.44	63.63	84.95
		140.0	21.14	35.27	49.44	63.63	84.95
		160.0	21.14	35.27	49.44	63.63	84.95
		180.0	21.14	35.27	49.44	63.63	84.95
		200.0	21.14	35.27	49.44	63.63	84.95
		240.0	21.14	35.27	49.44	63.63	84.95
		280.0	21.14	35.27	49.44	63.63	84.95
		320.0	21.14	35.27	49.44	63.63	84.95
		400.0	21.14	35.27	49.44	63.63	84.95

5.9.4.7. BEV300 AER

Table 59 summarizes the cost, mass and cell capacities generated for compact and midsize BEV300 AERs from BatPaC using the assumptions specified

Table 59. BatPaC Lookup Table for BEV300 AERs (Compact/Midsize)

			Energy, kWh				
			30.0	50.0	70.0	90.0	120.0
BatPac Cost	Power, kW	20.0	\$8,001	\$ 9,947	\$11,779	\$13,512	\$16,079
		40.0	\$8,015	\$ 9,962	\$11,794	\$13,526	\$16,093
		60.0	\$8,030	\$ 9,976	\$11,808	\$13,541	\$16,107
		80.0	\$8,045	\$ 9,991	\$11,823	\$13,555	\$16,122
		100.0	\$8,060	\$10,006	\$11,837	\$13,570	\$16,136
		120.0	\$8,075	\$10,021	\$11,852	\$13,585	\$16,151
		140.0	\$8,091	\$10,036	\$11,867	\$13,600	\$16,166
		160.0	\$8,107	\$10,051	\$11,883	\$13,615	\$16,181
		180.0	\$8,123	\$10,067	\$11,898	\$13,630	\$16,196
		200.0	\$8,139	\$10,083	\$11,913	\$13,645	\$16,211
		240.0	\$8,173	\$10,115	\$11,945	\$13,676	\$16,242
		280.0	\$8,210	\$10,148	\$11,977	\$13,708	\$16,273
		320.0	\$8,247	\$10,182	\$12,010	\$13,740	\$16,305
		400.0	\$8,329	\$10,255	\$12,078	\$13,807	\$16,370
BatPac Mass (kg)	Power, kW	20.0	215.23	305.78	392.25	476.97	601.47
		40.0	215.82	306.36	392.83	477.54	602.05
		60.0	216.42	306.96	393.42	478.13	602.64
		80.0	217.04	307.57	394.02	478.73	603.23
		100.0	217.66	308.18	394.63	479.33	603.84
		120.0	218.30	308.80	395.24	479.94	604.44
		140.0	218.95	309.43	395.87	480.56	605.06
		160.0	219.61	310.07	396.49	481.18	605.68
		180.0	220.28	310.72	397.13	481.81	606.30
		200.0	220.97	311.38	397.78	482.45	606.93
		240.0	222.39	312.72	399.09	483.74	608.20
		280.0	223.87	314.10	400.43	485.06	609.50
		320.0	225.44	315.54	401.81	486.41	610.83
		400.0	228.91	318.57	404.69	489.21	613.55
BatPac Cell Capacity (Ah)	Power, kW	20.0	21.14	35.28	49.44	63.63	84.95
		40.0	21.14	35.27	49.44	63.63	84.95
		60.0	21.14	35.27	49.44	63.63	84.95
		80.0	21.14	35.27	49.44	63.63	84.95
		100.0	21.14	35.27	49.44	63.63	84.95
		120.0	21.14	35.27	49.44	63.63	84.95
		140.0	21.14	35.27	49.44	63.63	84.95
		160.0	21.14	35.27	49.44	63.63	84.95
		180.0	21.14	35.27	49.44	63.63	84.95
		200.0	21.14	35.27	49.44	63.63	84.95
		240.0	21.14	35.27	49.44	63.63	84.95
		280.0	21.14	35.27	49.44	63.63	84.95
		320.0	21.14	35.27	49.44	63.63	84.95
		400.0	21.14	35.27	49.44	63.63	84.95

Table 60 summarizes the cost, mass and cell capacities generated for SUV and pickup BEV300 AERs from BatPaC using the assumptions specified.

Table 60. BatPaC Lookup Table for BEV300 AERs (SUVs/Pickups)

BEV300			Energy, kWh						
			30.0	50.0	70.0	90.0	120.0	140.0	160.0
BatPac Cost	Power, kW	20.0	\$8,220	\$10,170	\$12,004	\$13,740	\$16,252	\$17,927	\$19,530
		40.0	\$8,233	\$10,183	\$12,017	\$13,753	\$16,265	\$17,940	\$19,542
		60.0	\$8,247	\$10,197	\$12,031	\$13,766	\$16,278	\$17,953	\$19,556
		80.0	\$8,261	\$10,210	\$12,044	\$13,779	\$16,291	\$17,967	\$19,569
		100.0	\$8,274	\$10,224	\$12,058	\$13,793	\$16,305	\$17,980	\$19,582
		120.0	\$8,288	\$10,235	\$12,069	\$13,804	\$16,316	\$17,991	\$19,594
		140.0	\$8,301	\$10,249	\$12,083	\$13,818	\$16,330	\$18,005	\$19,607
		160.0	\$8,315	\$10,263	\$12,097	\$13,832	\$16,343	\$18,019	\$19,621
		180.0	\$8,330	\$10,278	\$12,111	\$13,846	\$16,357	\$18,032	\$19,634
		200.0	\$8,345	\$10,292	\$12,125	\$13,860	\$16,371	\$18,046	\$19,648
		240.0	\$8,376	\$10,322	\$12,154	\$13,888	\$16,399	\$18,074	\$19,676
		280.0	\$8,408	\$10,352	\$12,184	\$13,915	\$16,426	\$18,100	\$19,702
		320.0	\$8,441	\$10,383	\$12,211	\$13,945	\$16,455	\$18,129	\$19,731
		400.0	\$8,517	\$10,448	\$12,274	\$14,005	\$16,514	\$18,188	\$19,789
BatPac Mass (kg)	Power, kW	20.0	220.08	311.20	398.10	483.26	607.52	689.98	771.22
		40.0	220.61	311.73	398.62	483.77	608.05	690.49	771.73
		60.0	221.16	312.26	399.15	484.30	608.58	691.02	772.26
		80.0	221.71	312.81	399.69	484.84	609.12	691.56	772.79
		100.0	222.28	313.37	400.24	485.39	609.66	692.11	773.34
		120.0	222.85	313.93	400.80	485.94	610.21	692.65	773.88
		140.0	223.44	314.50	401.37	486.50	610.77	693.21	774.43
		160.0	224.04	315.08	401.93	487.06	611.33	693.76	774.99
		180.0	224.65	315.66	402.51	487.63	611.89	694.32	775.55
		200.0	225.27	316.26	403.09	488.21	612.46	694.89	776.11
		240.0	226.55	317.47	404.27	489.37	613.61	696.03	777.25
		280.0	227.89	318.72	405.49	490.57	614.78	697.20	778.41
		320.0	229.31	320.02	406.73	491.78	615.98	698.38	779.58
		400.0	232.43	322.75	409.33	494.31	618.44	700.82	782.00
BatPac Cell Capacity (Ah)	Power, kW	20.0	19.12	31.91	44.73	57.56	76.86	89.74	102.64
		40.0	19.12	31.91	44.73	57.56	76.85	89.74	102.64
		60.0	19.12	31.91	44.73	57.56	76.85	89.74	102.64
		80.0	19.12	31.91	44.73	57.56	76.85	89.74	102.64
		100.0	19.12	31.91	44.73	57.56	76.85	89.73	102.64
		120.0	19.12	31.91	44.73	57.56	76.85	89.73	102.64
		140.0	19.12	31.91	44.73	57.56	76.85	89.73	102.64
		160.0	19.12	31.91	44.73	57.56	76.85	89.73	102.64
		180.0	19.12	31.91	44.73	57.56	76.85	89.73	102.64
		200.0	19.12	31.91	44.73	57.56	76.85	89.73	102.64
		240.0	19.12	31.91	44.73	57.56	76.85	89.73	102.64
		280.0	19.12	31.91	44.73	57.56	76.85	89.73	102.63
		320.0	19.12	31.91	44.73	57.56	76.85	89.73	102.63
		400.0	19.12	31.91	44.73	57.56	76.85	89.73	102.63

5.10. Accessory Loads

Electrical and mechanical accessory base loads are assumed to be constant over the drive cycles, with values varying across powertrain types. Derived from data from the AMTL, this value is used to represent the average accessory load consumed during the standard urban FTP and EPA's HWFET drive-cycle dynamometer testing. For FRM rulemaking runs, NHTSA has directed Argonne to vary the base accessory loads for different vehicle classes and performance categories across the different vehicle powertrains. Table 61 details the base accessory load assumptions by vehicle class and performance category across the different vehicle powertrains.

Table 61. Base Accessory Load Assumptions

Vehicle Class	Performance Category	Vehicle Powertrain		
		Conventional	HEVs	PHEVs and BEVs
Compact	Base	250	275	375
Compact	Premium	300	375	475
Midsize	Base	250	275	375
Midsize	Premium	300	375	475
Small SUV	Base	300	325	425
Small SUV	Premium	300	375	475
Midsize SUV	Base	300	325	425
Midsize SUV	Premium	350	375	475
Pickup	Base	300	325	425
Pickup	Premium	300	375	475

5.11. Light-Weighting Technologies

For the NPRM analysis, DOT/NHTSA proposed that light-weighting be applied to the glider weight as defined by the body and chassis only. For final rulemaking, the agencies decided to use the definition of the glider weight defined by the following systems:

- Body
- Chassis
- Interior
- Safety system
- Brake mechanism
- Steering system
- Mechanical accessories

- Electrical accessories
- Wheels

Light-weighting is applied across all glider systems except for those related to safety.

The secondary effects of light-weighting (such as downsizing) are taken into account as part of the vehicle sizing algorithm. To emphasize technology platform sharing in the study, vehicles with lower levels of mass reduction (5% and 7.5%) inherit the sizing characteristics of the reference vehicle (0% mass reduction). Vehicles with higher levels of mass reduction are resized to meet the vehicle technical specifications. The percentages of mass reductions selected for the study are as follows:

- Light-weighting level 0 (MR0): 0% (reference vehicle is sized)
- Light-weighting level 1 (MR1): 5% (inherited from reference vehicle)
- Light-weighting level 2 (MR2): 7.5% (inherited from reference vehicle)
- Light-weighting level 3 (MR3): 10% (vehicle is sized)
- Light-weighting level 4 (MR4): 15% (vehicle is sized)
- Light-weighting level 5 (MR5): 20% (vehicle is sized)
- Light-weighting level 6 (MR6): 30% (vehicle is sized)

5.12. Aerodynamic Reduction Technologies

Along with different levels of light-weighting reductions, five levels of aerodynamic reduction have been applied. The percentages of aerodynamic reduction selected for the study are as follows:

- Aerodynamic reduction level 0 (AERO0): 0%
- Aerodynamic reduction level 1 (AERO1): 5%
- Aerodynamic reduction level 2 (AERO2): 10%
- Aerodynamic reduction level 3 (AERO3): 15%
- Aerodynamic reduction level 4 (AERO4): 20%

5.13. Rolling Resistance Reduction Technologies

For this study, three levels of rolling resistance reduction have been applied for each vehicle configuration. The percentages of rolling resistance reduction selected for the study are as follows:

- Rolling resistance-reduction level 0 (ROLL0): 0%
- Rolling resistance-reduction level 1 (ROLL1): 10%
- Rolling resistance-reduction level 2 (ROLL2): 20%

6. Test Procedure and Energy Consumption Calculations

All simulations were performed under hot conditions. Cold-start penalties were assessed after the simulations, based on test data collected at AMTL and EPA published values. A two-cycle test procedure, based on the Urban Dynamometer Driving Schedule (UDDS) and Highway Fuel Economy Driving Schedule (HWFET) drive cycles, was used.

6.1. Conventional Vehicles

The conventional vehicle test procedure follows the current EPA two-cycle test procedure (U.S. Environmental Protection Agency, 2018).

The urban cycle for a non-hybrid vehicle (Figure 177) has four parts:

1. Bag 1: cold start
2. Bag 2: stop and go
3. Engine OFF
4. Bag 3: hot start

Figure 177. The Urban Cycle for a Non-Hybrid Vehicle

The highway cycle for a non-hybrid vehicle has only one part, the HWFET (Figure 178).

Figure 178. The Highway Cycle for a Non-Hybrid Vehicle

6.2. Hybrid Electric Vehicles

The HEV procedure is similar to the conventional vehicle procedure except that the drive cycles are repeated until the initial and final battery SOC_s are within a tolerance of 0.5% (see Figure 179 and Figure 180). This is based on SAE J1711 procedure (Duoba, 2009). This procedure is applied for mild hybrid BISG, split HEV, par HEV and FC HEV vehicle powertrains.

Figure 179. The Urban Cycle for a Hybrid Vehicle

**Figure 180. The Highway Cycle for a Hybrid Vehicle
(Only The Results from the Second Cycle Were Used)**

6.2.1. Two-Cycle Procedure Calculations for Conventional and Hybrid Vehicles

6.2.1.1. Fuel Consumption

For the urban cycle, fuel consumption was computed using Equation 1:

$$(1) \text{ Fuel Consumption} = 0.43 \frac{V_{Fuel}^1 + V_{Fuel}^2}{Dist_1 + Dist_2} + 0.57 \frac{V_{Fuel}^3 + V_{Fuel}^Z}{Dist_3 + Dist_Z}$$

Where

V_{Fuel}^y = Volume of fuel from Bag y ,

$Dist_y$ = Distance driven by the vehicle for the Bag y part of the cycle, and

Z = Bag 2 for a non-hybrid vehicle and Bag 4 for a hybrid.

The same equation was used to compute the gas-equivalent fuel consumption as well as the SOC-adjusted fuel consumption by replacing V_{Fuel} with the corresponding physical quantity.

The highway cycle results were the same as for an urban cycle, except for the hybrid case, in which only the results from Bag 1 were used to compute the values:

$$(2) \text{ Fuel Consumption} = \frac{V_{Fuel}^2}{Dist}$$

Combined Fuel Consumption

The combined fuel consumption is a weighted value lying between the urban and highway cycles:

$$(3) FC^{combined} = 0.55 \times FC^{urban} + 0.45 \times FC^{highway}$$

6.2.2. Plug-in Hybrid Electric Vehicles

This section describes the methodology currently implemented in Autonomie for PHEVs. This procedure is used for split PHEV20, EREV PHEV50, par PHEV20 and par PHEV50 vehicle powertrains. The implementation is based on the J1711 procedure. The procedure is divided into several phases, as described below.

6.2.2.1. Charge Sustaining on the UDDS Cycle

1. Set battery SOC to charge sustaining (CS) value.
2. Run UDDS.
3. 10-minute soak with the key off.
4. Run UDDS.
5. Assume the cycle charge is balanced. Display warning if it does not meet 1%.

Weightings and Cold Factor Correction

The following equations show the cold compensation:

$$(4) M_{0-505}^* = \frac{M_{0-505}}{1-CF_{75F}}$$

Where

M_{0-505} = fuel mass consumed during the time window between 0 and 505 sec

CF_{75F} = cold-factor correction at 75°F

M_{0-505}^* = cold-corrected mass of fuel

$$(5) Vol_{0-505}^* = \frac{M_{0-505}^*}{\delta_{gasoline}}$$

Where

Vol_{0-505}^* = volume of fuel consumed during the time window between 0 and 505 sec, and

$\delta_{gasoline}$ = density of gasoline.

One can then calculate FC^{UDDS} , the fuel consumed on the UDDS cycle:

$$(6) FC^{UDDS} = 0.43 \times \left(\frac{Vol_{0-505}^* + Vol_{506-1372}}{D_{0-505} + D_{506-1372}} \right) + 0.57 \times \left(\frac{Vol_{1972-2477} + Vol_{2478-3340}}{D_{1972-2477} + D_{2478-3340}} \right)$$

6.2.2.2. Charge Sustaining on the HWFET Cycle

1. Set battery SOC to CS value.
2. Run HWFET.
3. Wait four seconds.
4. Run HWFET.
5. Assume the cycle is charge balanced.
6. Perform calculations on the second HWFET cycle.

$$(7) FC^{HWFET} = \frac{Vol_{765-1529}}{D_{765-1529}}$$

Where

$Vol_{765-1529}$ = volume of fuel consumed during the time window between 765 and 1,529 sec

$D_{765-1529}$ = distance traveled during the time window between 765 and 1,529 sec

FC^{HWFET} = highway fuel consumption

6.2.2.3. Charge Depleting on the UDDS and HWFET Cycles

1. The charge-depleting (CD) calculations are identical for the UDDS and HWFET cycles.
2. Set battery SOC to full charge test initial SOC.
3. Run UDDS (HWFET).
4. 10-minute soak with the key off (15-second pause with key ON).
5. Run UDDS (HWFET).
6. 10-minute soak with the key off (15-second pause with key ON).
7. Repeat until SOC reaches the CD/CS crossover point, and the last cycle is completed.
8. Round down the number of cycles unless the CD range is less than one cycle. In that case, round up the number of cycles. At least one CD cycle is required to run the analysis.

Cold Weighting Calculation

The user specifies the number of cycles over which to apply the cold correction factor:

$$(8) N_{cold} = \min(N_{cold}^{user}, N_{cd})$$

$$(9) N_{hot} = N_{cd} - N_{cold}$$

Where

N_{cold} = number of cold cycles

N_{hot} = number of hot cycles

N_{cold}^{user} = number of user-specified cold cycles

N_{cd} = total number of CD cycles

$$(10) M_{cd} = \left[\frac{\alpha_{cold} M_{cd-cold}^1}{1-CF_{75F}}, \dots, \frac{\alpha_{cold} M_{cd-cold}^{N_{cold}}}{1-CF_{75F}}, \alpha_{hot} M_{cd-hot}^1, \dots, \alpha_{hot} M_{cd-hot}^{N_{hot}} \right]^T$$

Where

$M_{cd-cold}^1$ = mass of fuel consumed during the first cold CD cycle

$M_{cd-cold}^{N_{cold}}$ = mass of fuel consumed during the last cold CD cycle

CF_{75F} = cold-start fuel economy penalty at 75°F

M_{cd-hot}^1 = mass of fuel consumed during the first hot CD cycle

$M_{cd-hot}^{N_{hot}}$ = mass of fuel consumed during the last hot CD cycle

α_{cold} = user-specified cold weighting factor (default value = 0.43)

α_{hot} = user-specified hot weighting factor (default value = 0.57)

M_{cd} = column vector of cold-corrected fuel mass

$$(11) Vol_{cd} = \frac{M_{cd}}{\delta_{gasoline}}$$

Where

Vol_{cd} = column vector of cold-corrected fuel volumes

Note that each element in the Vol_{cd} vector is divided by its respective distance:

$$(12) FC_{cd} = \frac{Vol_{cd}}{D_{udds}}$$

Where

FC_{cd} = column vector of cold-corrected fuel consumption

The net battery energy used was calculated for each cycle using the open-circuit voltage and the current:

$$(13) \text{ for } i = 1, \dots, N_{cd}; E_{cd}^i = \int_{(i-1)T_{udds}}^{(i)T_{udds}+t} V_{oc}(\tau) * I(\tau) d\tau$$

Where

E_{cd}^i = net battery energy used during the i^{th} CD cycle

T_{udds} = duration of the UDDS cycle + soak time or (HWFET + 15 sec)

i = index of the CD cycle

N_{cd} = total number of CD cycles

V_{oc} = open-circuit voltage as a function of time during the cycle

I = battery current as a function of time during the cycle

$$(14) E_{cd} = [E_{cd}^1, \dots, E_{cd}^{N_{cd}}]^T$$

Where

E_{cd} = column vector of net battery energy used on each cycle

Note that each element in the E_{cd} vector is divided by its respective distance:

$$(15) EC_{cd} = \frac{E_{cd}}{D_{udds} * \eta_{chg}^{ess} * \eta_{charger}}$$

Where

EC_{cd} = column vector of electrical-energy consumption in AC-Joules (wall outlet)

D_{udds} = distance traveled on a UDDS (or $HWFET - D_{HWFET}$) cycle

η_{chg}^{ess} = user-definable efficiency of the battery during charging (default value = 0.99)

$\eta_{charger}$ = user-definable efficiency of the charger (wall or in-vehicle) (default value = 0.88)

$$(16) \text{ for } i = 1, \dots, N_{cd}; \mu_i = \mu(i * D_{udds}^i) - \mu(i - 1) * D_{udds}^i$$

$$\mu_{cd} = [\mu_1, \dots, \mu_{N_{cd}}]$$

Where

μ_{cd} = row vector of utility factors

μ_1 = utility factor on the first CD cycle

μ_i = utility factor on the i^{th} CD cycle

$\mu_{N_{cd}}$ = utility factor on the last CD cycle

μ = fleet mileage fraction utility factor as a function of distance

$$(17) FC = \mu_{cd} FC_{cd} + (1 - \sum_i^{N_{cd}} \mu_i) FC_{cs}$$

Where

FC = fuel consumed on the city or highway portion of the PHEV procedure

$$(18) EC = \mu_{cd} EC_{cd}$$

Where

EC = electrical energy consumed during the city or highway portion of the PHEV procedure

6.3. Electric Vehicles (BEVs)

For BEVs, the SAE J1634 test procedure is followed (Light Duty Vehicle Performance and Economy Measure Committee, 2017). The procedure is to start the battery at full SOC and run until minimum SOC is reached. The electric consumption is then computed as:

$$(1) \text{ Electric Consumption} = \frac{\int V_{oc} * I_{ess}}{\eta_{ess} \eta_{charger}}$$

Where

η_{ess} = efficiency of the battery while charging
 $\eta_{charger}$ = average efficiency of the charger while charging

V_{oc} = open-circuit voltage as a function of time over the cycle

I_{ess} = current as a function of time over the cycle

6.3.1. Cold-Start Penalty

An analysis was conducted on the EPA database of MY 2016 light-duty vehicles (U.S. EPA, 2016c) for the different fuel economy values reported for the different bags. Following an average of the ratio of Bag 3 fuel economy over Bag 1 fuel economy, a 15% cold-start penalty was selected to be implemented. The cold-start penalty was applied for the fuel consumption during the FTP for conventional vehicles, HEVs, and PHEVs; 0% was applied for BEVs. Figure 181 shows the ratio of Bag 3 fuel economy to Bag 1 fuel economy. An average value of 15% was used as the cold-start penalty value for Bag 1. An additional penalty of 2.5% was applied to the Bag 2 fuel economy

Figure 181. Ratio of Bag 3 to Bag 1 Fuel Economy (MY 2016)

7. Vehicle Setup Process

Argonne's large-scale simulation process was developed to run a very large number of vehicles simulations quickly and effectively. It allows Argonne to quickly respond to DOT requests by simulating any technology combination in any vehicle class. The following subsections describe the different steps of the process.

7.1. Powertrain Template

Powertrain templates contain basic information such as name, class, and technology, as well as component information such as battery technology, engine technology, and transmission type. To automate the differentiated process of the different powertrain options, three different templates are defined: Conventional, Parallel Hybrids, and Other Hybrids (power-split HEV/PHEV, EREV, fuel cell, and BEV).

The template contains seven tabs: vehicle, parameter, control, sizing, run, translation, and assumptions. In each tab, columns outline vehicle configurations.

7.1.1. Vehicle Tab

The vehicle tab of the template defines the initialization files, the component models required for each vehicle, and the vehicle configuration selected. The initialization files selected will depend on the tree selection and the technological combination nominated for that vehicle. Figure 182 shows the vehicle tab of the Conventional template.

<div> <div>Clipboard</div> <div>Font</div> <div>Alignment</div> <div>Number</div> <div>General</div> <div>Conditional Formatting</div> <div>Format as Table</div> <div>Cell Styles</div> <div>Insert</div> <div>Delete</div> <div>Format</div> <div>AutoSum</div> <div>Fill</div> <div>Clear</div> <div>Sort & Filter</div> <div>Find & Select</div> </div>			
<div> <div>G177</div> <div> <div>✕</div> <div>✓</div> <div>fx</div> </div> <div>=C\$177</div> </div>			
	A	B	C
1			
2			
3		Run Number	Conventional
4		Template Vehicle (*a_vehicle)	conv_a_vehicle
5		AUTONOMIE Vehicle File (*a_vehicle)	conv_ClassName_EngName_TransName_Mrname_AEROname_RollName_a_vehicle
6	drv.ctrl	model	
7	drv.ctrl	init	
8	env	model	
9	env	init	
10	vpa	init	
11	vpa	preproc	
12	str.ctrl	model	
13	str.ctrl	init	
14	str.plant	model	
15	str.plant	init	
16	eng.ctrl	model	
17	eng.ctrl	init	
18	eng.ctrl	preproc	
19	eng.ctrl.dmd	model	
20	eng.ctrl.dmd	init	
21	eng.ctrl.dmd	preproc	
22	eng.ctrl.cstr	model	
23	eng.ctrl.cstr	init	
24	eng.ctrl.cstr	preproc	
25	eng.ctrl.trs	model	
26	eng.ctrl.trs	init	
27	eng.ctrl.trs	preproc	
28	eng.ctrl.cmd	model	
29	eng.ctrl.cmd	init	
30	eng.ctrl.cmd	preproc	
31	eng.plant	model	
32	eng.plant	init	EngFile
33	eng.plant	scale	
34	eng.plant	preproc	
35	exh.plant	model	
36	exh.plant	init	
37	exh.plant	preproc	
38	accmech.plant	model	
39	accmech.plant	init	
<div> <div>vehicle</div> <div>parameter</div> <div>control</div> <div>sizing</div> <div>run</div> <div>translation</div> <div>Assumptions</div> </div>			

Figure 182. Vehicle Tab of Conventional Template

7.1.2. Parameter Tab

The parameter tab of the template defines the values of the components specific to the vehicle designated (e.g., power, weight, performance constraints). Figure 183 shows the parameter tab of the Conventional template.

<div> <div> Cut Copy Paste Format Painter Clipboard </div> <div> Calibri 11 A⁺ A⁻ B I U </div> <div> Wrap Text Merge & Center </div> <div> General \$ % </div> <div> Conditional Formatting Table </div> <div> Normal 3 Normal 4 Good Neutral </div> </div>					
L19					
	A	B	C	D	E
7					
8			Conventional	Micro Hybrid	Mild Hybrid BISG
9					
10		Run Number	1	4	10
11					
12	Configuration				
13	Specific Comments (I.e. FC files, high case,...)		Mrname	Mrname	Mrname
	AUTONOMIE Vehicle File (*.a_vehicle)		conv_EngName_Trans Name_Mrname_AERO name_RollName.a_veh icle	micro_EngName_Tra nsName_Mrname_A EROName_RollName .a_vehicle	bisg_EngName_Trans Name_Mrname_AER OName_RollName.a vehicle
14	Sizing				
15					
16	AutoSizing		SizingType	Inherited	Inherited
17	PerfoCategory		PerfoCategory	PerfoCategory	PerfoCategory
18	Vehicle Class		ClassName	ClassName	ClassName
19	Vehicle Config		Conventional	Micro Hybrid	Mild Hybrid BISG
20	Engine Type		EngName	EngName	EngName
21	Mass Reduction Type		Mrname	Mrname	Mrname
22	Mass Reduction Percentage		MRValue	MRValue	MRValue
23	AERO Type		AEROName	AEROName	AEROName
24	AERO Reduction Percentage		AEROValue	AEROValue	AEROValue
25	Rolling Resistance Type		RollName	RollName	RollName
26	Rolling Resistance Reduction Percentage		ROLLValue	ROLLValue	ROLLValue
27	Constraints				
28	Acceleration				
29	From	mph	0	0	0
30	To	mph	60	60	60
31	In	sec	p_time	p_time	p_time
32	Shifting Time	sec	0.45	0.45	0.45
33	Constraints				
34	Passing Acceleration				
35	Passing From	mph	50	50	50
36	Passing To	mph	80	80	80
37	Passing In	sec	p_time	p_time	p_time
38	Perc of peak power required to meet VTS				
39	Engine or fuel cell sized to meet X%	%	70	70	70
40	Grade				
41	Minimum	%	6	6	6
42	At Speed	mph	65	65	65
43	Use Grade Mass equation		Yes	Yes	Yes
44	Maximum Payload	kg	payload_mass	payload_mass	payload_mass
45	Maximum Towing	kg	towing_mass	towing_mass	towing_mass
46	Max Vehicle Speed must be at least	mph	110	110	110
47	Range				
48	UDDS Fuel Range	miles	320	320	320
49	UDDS EV Range	miles			

Figure 183. Parameter Tab of Conventional Template

7.1.3. Control Tab

The control tab selects the appropriate controller for the designated vehicle. Figure 184 shows the control tab of the Hybrid template.

	A	B	G	I	L
1			Split SI	Split SI - 20mi AER	Erev SI - 50mi AER
2		Run Number	16	22	29
3	conso	vpc.prop		vpc.prop.init.ev_only	vpc.prop.init.ev_only
4	conso	vpc.prop		0	0
5	conso	vpc.prop		vpc.prop.init.stop_at_low_soc	vpc.prop.init.stop_at_low_soc
6	conso	vpc.prop		1	1
7	conso	vpc.prop		vpc.prop.init.whl_pwr_above_eng_turn_on_map_pwr6	vpc.prop.init.whl_pwr_above_eng_turn_on_map_pwr6
8	conso	vpc.prop		20000	20000
9	conso	vpc.prop		vpc.prop.init.whl_pwr_below_eng_turn_off_multiplier	vpc.prop.init.whl_pwr_below_eng_turn_off_multiplier
10	conso	vpc.prop		0.25	0.25
11	conso	vpc.prop		vpc.prop.init.whl_pwr_above_eng_turn_on_map_pwr7	vpc.prop.init.whl_pwr_above_eng_turn_on_map_pwr7
12	conso	vpc.prop		20000	20000
13	conso	gb.ctrl.trs			
14	conso	gb.ctrl.trs			
15	conso	gb.ctrl.dmd			
16	conso	gb.ctrl.dmd			
17	conso	gb.ctrl.dmd			
18	conso	gb.ctrl.dmd			
19	conso	gb.ctrl.dmd			
20	conso	gb.ctrl.dmd			
21	conso	gb.ctrl.dmd			
22	ev_only	vpc.prop		vpc.prop.init.ev_only	vpc.prop.init.ev_only
23	ev_only	vpc.prop		1	1
24	ev_only	vpc.prop		vpc.prop.init.stop_at_low_soc	vpc.prop.init.stop_at_low_soc
25	ev_only	vpc.prop		1	1
26	ev_only	vpc.prop			
27	ev_only	vpc.prop			
28	ev_only	vpc.prop			
29	ev_only	vpc.prop			
30	ev_only	gb.ctrl.trs			
31	ev_only	gb.ctrl.trs			
32	ev_only	gb.ctrl.trs			
33	ev_only	gb.ctrl.trs			
34	ev_only	gb.ctrl.trs			
35	ev_only	gb.ctrl.trs			
36	ev_only	gb.ctrl.trs			
37	ev_only	gb.ctrl.trs			
38	ev_only	gb.ctrl.dmd			
39	ev_only	gb.ctrl.dmd			
40	perfo	vpc.prop		vpc.prop.init.ev_only	vpc.prop.init.ev_only
41	perfo	vpc.prop		0	0
42	perfo	vpc.prop			
43	perfo	vpc.prop			
44	perfo	vpc.prop			
45	perfo	vpc.prop			
46	perfo	gb.ctrl.trs			
47	perfo	gb.ctrl.trs			
48	perfo	gb.ctrl.trs			
49	perfo	gb.ctrl.trs			
50	perfo	gb.ctrl.trs			
51	perfo	gb.ctrl.trs			
52	perfo	gb.ctrl.dmd			
53	perfo	gb.ctrl.dmd			
54	perfo	gb.ctrl.dmd			
55	perfo	gb.ctrl.dmd			

Figure 184. Control Tab of Hybrid Template

7.1.4. Sizing Tab

The sizing tab selects the appropriate sizing rule to size each component to match the required vehicle technical specifications. Figure 185 shows the sizing tab of the Hybrid template.

	A	B	G	J	L	M	N	O
1			Split SI	Split SI - 30mi AER	Erev SI - 50mi AER	FC HEV	BEV - 100	
2		Run Number	16	25	29	75	93	
3	AUTONOMIE Vehicle File (*.a_vehicle)							
4	Rule	rule_file	split_IVM_60mph_mot_70p_of_peak_eng_pEVXX_simu_test_70p_of_pef_PHEVXX_simu_70p_of_peifc_IVM_60mph_mot_70p_of_pesr_elec_simu_passing					
5								
6	Power							
7	Grade	grade_compo	eng	eng	eng	fc	mot_ess	
8	Acceleration	accl_compo	mot_eng	mot_eng	mot_eng	mot_fc	mot_ess	
9	Regen	regen_compo	mot_ess			mot_ess		
10	EV Only	ev_only_compo		mot_ess	mot_ess		mot_ess	
11	Energy							
12	Range	range_compo		ess	ess		ess	
13	Sizing Cycle for Regen / EV Only / Range							
14	Name	cycle_pwr_name	UDDS	UDDS	US06	US06	US06	
15	File	cycle_pwr_file	fuds1.mat	fuds1.mat	us06.mat	us06.mat	us06.mat	
16	Repear	cycle_pwr_number	1	1	1	1	1	
17	Scale	cycle_pwr_scale	1	1	1	1	1	
18	Grade	cycle_pwr_grade	0	0	0	0	0	
19	Sizing Cycle for EV Only Energy / Fuel							
20	Name	cycle_nrg_name	UDDS	UDDS	UDDS	UDDS	UDDS	
21	File	cycle_nrg_file	fuds1.mat	fuds1.mat	fuds1.mat	fuds1.mat	fuds1.mat	
22	Repear	cycle_nrg_number	1	1	1	1	1	
23	Scale	cycle_nrg_scale	1	1	1	1	1	
24	Grade	cycle_nrg_grade	0	0	0	0	0	
25	Control							
26	CD Range	ctrl_tuning		yes	yes			
27								

Figure 185. Sizing Tab of Hybrid Template

7.1.5. Run Tab

The run tab selects the drive cycle/procedure to be simulated for each powertrain options. Figure 186 shows the run tab of the Hybrid template.

J56									
	A	B	C	H	J	M	N	O	P
1		Run Number		Split SI	Split SI - 30mi AER	Erev SI - 50mi AER	FC HEV	BEV - 100	
2				15	22	29	75	99	
3	Cycle 1								
4	Cycle 1								
5	Name	cycle1	name	US06	US06	US06	US06	US06	
6	File	cycle1	file	us06.mat	us06.mat	us06.mat	us06.mat	us06.mat	
7	Repeat	cycle1	number	1	1	1	1	1	
8	Scale	cycle1	scale	1	1	1	1	1	
9	Grade	cycle1	grade	0	0	0	0	0	
10	Cycle 2								
11	Name	cycle2	name	Accel	Accel	Accel	Accel	Accel	
12	File	cycle2	file	accel.mat	accel.mat	accel.mat	accel.mat	accel.mat	
13	Repeat	cycle2	number	1	1	1	1	1	
14	Scale	cycle2	scale	1	1	1	1	1	
15	Grade	cycle2	grade	0	0	0	0	0	
16	Cycle 3								
17	Name	cycle3	name						
18	File	cycle3	file						
19	Repeat	cycle3	number						
20	Scale	cycle3	scale						
21	Grade	cycle3	grade						
22	Cycle 4								
23	Name	cycle4	name						
24	File	cycle4	file						
25	Repeat	cycle4	number						
26	Scale	cycle4	scale						
27	Grade	cycle4	grade						
28	Cycle 5								
29	Name	cycle5	name						
30	File	cycle5	file						
31	Repeat	cycle5	number						
32	Scale	cycle5	scale						
33	Grade	cycle5	grade						
34	Procedure								
35	Procedure Name	test_procedure	name	Unid Procedure (FTP+HWFET) with 1	PHEV Preliminary Procedures	PHEV Preliminary Procedures	Unid Procedure (FTP+HWFET) with 1	BEV Procedure	
36	Cold Start Penalty	test_procedure	cold_start_penalty_inhert	15	15	15	15	15	
37	Cold Start Penalty UDDS	test_procedure	cold_start_penalty_udds						
38	Cold Start Penalty Number of cold cycles	test_procedure	cold_start_penalty_nbr_cycle						
39	Charger efficiency	test_procedure	charger_eff		96	96		96	
40									
41									
42									
43									
44									

Figure 186. Run Tab of Hybrid Template

7.1.6. Translation Tab

The translation tab, shown in Figure 187, translates and transfers inputs into Autonomie to build the vehicle model.

	A	B	C	D	E	F	G	H	I
		location	component	variable type	variable name	location	component	variable type	variable name
2	Sizing								
3	AutoSizing	drivetrain	vpa	param_var	vpa.init.isSized	sizing	struct	isSized	
4	PerfoCategory	drivetrain	vpa	param_var	vpa.init.veh_PerfoCategory				
5	Vehicle Class	drivetrain	vpa	param_var	vpa.init.veh_class_name				
6	Vehicle Config	drivetrain	vpa	param_var	vpa.init.veh_config_type				
7	Engine Type	drivetrain	eng.plant	param_var	eng.plant.init.type				
8	Mass Reduction Type	drivetrain	vpa	param_var	vpa.init.veh_MR				
9	Mass Reduction Percentage	drivetrain	vpa	param_var	vpa.init.veh_MR_percentage_reduc				
10	AERO Type	drivetrain	vpa	param_var	vpa.init.veh_AERO				
11	AERO Reduction Percentage	drivetrain	vpa	param_var	vpa.init.veh_AERO_percentage_reduc				
12	Rolling Resistance Type	drivetrain	vpa	param_var	vpa.init.veh_ROLL				
13	Rolling Resistance Reduction Percentage	drivetrain	vpa	param_var	vpa.init.veh_ROLL_percentage_reduc				
14	Constraints								
15	Acceleration								
16	From	sizing	accel	from					
17	To	sizing	accel	to					
18	In	sizing	accel	time					
19	Shifting Time	sizing	accel	shift_time					
20	Constraints								
21	Passing Acceleration								
22	Passing From	sizing	passing	from					
23	Passing To	sizing	passing	to					
24	Passing In	sizing	passing	time					
25	Perf of peak power required to meet VTS								
26	Engine or fuel cell sized to meet X%	sizing	constraints	peak_pwr_perc					
27	Grade								
28	Minimum	sizing	grade	percent					
29	At Speed	sizing	grade	speed					
30	Use Grade Mass equation	sizing	grade	use_gwv_equation					
31	Maximum Payload	drivetrain	vpa	param_var	veh.plant.init.payload_mass	sizing	grade	max_payload	
32	Maximum Towing	drivetrain	vpa	param_var	veh.plant.init.towing_mass	sizing	grade	max_towing	
33	Max Vehicle Speed must be at least	sizing	speed	max					
34	Range								
35	UDDS Fuel Range	sizing	range	fuel					
36	UDDS EV Range	sizing	range	ev					
37	SOC								
38	Electrical Only								
39	EV SOC Init ESS #1	sizing	soc	ev_max					
40	EV SOC Final ESS #1	sizing	soc	ev_min					
41	Charge Depleting / Blending Mode								
42	CD SOC Init ESS #1	sizing	soc	cd_max					

Figure 187. Translation Tab of Template

7.2. Multi-Vehicle Template Expansion and Duplication

After the large-scale simulation process defines the list of all component and vehicle inputs, a multiplier code expands the reference/template file into as many as needed to define the vehicle's technological combinations based on the decision tree inputs. This step stores all of the template files in the respective folders for each vehicle class and performance category. Figure 188 shows the vehicle template files for the different vehicle classes and performance categories in their respective folders.

8. Vehicle Sizing Process

To ensure platform sharing among vehicles, a unique approach to sizing the vehicle powertrain components was taken. While the reference baseline vehicles are sized to meet specific sets of performance criteria, the rest of the vehicles inherit their properties from the baseline reference vehicle and may have variations in performance. This approach represents real-world platform sharing (i.e., OEMs have a limited number of engines that are used across classes). A vehicle with inherited properties is referred to as an inherited vehicle.

8.1. Inheritance

The process of inheritance retrieves the values of different vehicle parameters of interest (engine power, engine weights, etc.) from the respective baseline reference vehicle and updates the inherited vehicle with the inherited value.

8.1.1. Conventional Powertrain (Conventional/Micro-12V/Mild Hybrid BISG)

Figure 190 shows the conventional powertrain inheritance flowchart for the range of vehicles and the respective baseline vehicle from which they inherit.

Figure 190. Conventional Powertrain Inheritance Flowchart for eng01

For a given engine, all transmissions, AERO and ROLL combinations would inherit the engine power and mass from conventional/6AU/AERO0/ROLL0 for a given MR combination. The micro-hybrid and mild-hybrid powertrain inherits from conventional as well.

For each conventional inherited vehicle, the algorithm in Figure 191 is implemented.

Figure 191. Inheritance Algorithm for Conventional Vehicle

8.1.2. Hybrid Powertrains (Split HEV/Split PHEV/EREV PHEV/Fuel Cell HEV/BEV)

Figure 192 shows the hybrid powertrain inheritance flowchart for the range of vehicles studied and the respective baseline vehicle to inherit from for each of the respective hybrid powertrains.

Figure 192. Hybrid Powertrain Vehicle Inheritance Flowchart

8.1.2.1. Inheritance for Battery-Electric Vehicles

For each inherited BEV, the algorithm in Figure 193 is implemented.

Figure 193. Inheritance Algorithm for Battery-Electric Vehicles

8.1.2.2. Inheritance for Fuel-Cell HEVs

For each inherited fuel-cell HEV, the algorithm shown in Figure 194 is implemented.

Figure 194. Inheritance Algorithm for Fuel-Cell HEVs

8.1.2.3. Inheritance for EREVs

For each inherited EREV, the algorithm shown in Figure 195 is implemented.

Figure 195. Inheritance Algorithm for EREVs

8.1.2.4. Inheritance for Split PHEVs

For each inherited split PHEV, the algorithm in Figure 196 is implemented:

Figure 196. Inheritance Algorithm for Split PHEVs

8.1.2.5. Inheritance for Split HEVs

For each inherited split HEV, the algorithm shown in Figure 197 is implemented.

Figure 197. Inheritance Algorithm for Split HEVs

8.1.3. Parallel Hybrid Powertrains

8.1.3.1. Inheritance for Parallel HEVs

The flowchart in Figure 198 shows the range of inherited parallel hybrid vehicles and the reference baseline vehicles that the vehicles inherit from.

Figure 198. Parallel HEVs: Inheritance from Reference Baseline Vehicles

For each inherited parallel HEV, the algorithm in Figure 199 is implemented.

Figure 199. Inheritance Algorithm for Parallel HEVs

8.1.3.2. Inheritance for Parallel PHEVs

The flowchart in Figure 200 shows the range of inherited parallel hybrid vehicles and the reference baseline vehicles that the vehicles inherit from.

Figure 200. Parallel PHEVs: Inheritance from Reference Baseline Vehicles

For each inherited parallel HEV, the algorithm in Figure 201 is implemented.

Figure 201. Inheritance Algorithm for Parallel HEVs

8.2. Vehicle Technical Specifications

Each vehicle class and performance type has specific vehicle performance requirements. Table 62 shows the details of the different vehicle classifications and the corresponding performance times used to size the vehicles. The vehicles can exceed specific performance targets (e.g., 0-60 mph time) owing to other constraints (e.g., EV range and EV mode).

Table 62. Vehicle Class Performance Times

Vehicle Class	Performance Category	0-60 mph Time (s)	50-80 mph Time (s)
Compact	Base/Premium	10/8	10/8
Midsize	Base/Premium	9/6	9/6
Small SUV	Base/Premium	9/7	9/7

Vehicle Class	Performance Category	0-60 mph Time (s)	50-80 mph Time (s)
Midsize SUV	Base/Premium	10/7	10/7
Pickup	Base/Premium	7	7

Along with the initial vehicle movement time to 60 mph, all vehicles are sized to meet the following requirements at minimum:

- Maximum grade (gradeability) of 6% at 65 mph at gross vehicle weight (GVW).
- Maximum vehicle speed > 100 mph
- Payload: 900 kg (pickup base/premium only)
- Towing: 3000 kg (pickup base) and 4350 kg (pickup premium)

These requirements are a good representation of the current American automotive market and of American drivers' expectations. The relationship between curb weight and GVW for current technology-configuration-powertrain combinations is modeled from the existing vehicles in the market, and it forms the basis for estimating the GVWs of future vehicle scenarios. Figure 202 shows the GVWR vs. curb weight relationship.

Figure 202. Curb Weight vs. GVWR

Using the equation derived from the figure above, the relationship is determined as:

$$\text{GVWR} = 1.224 * (\text{curb weight}) + 279.59$$

8.3. Vehicle Powertrain Sizing Algorithms

Improperly sizing components will lead to differences in energy consumption and will influence the effectiveness results. Therefore, we have developed several automated sizing algorithms to provide a fair comparison between technologies. Algorithms have been defined depending on the powertrain (e.g., conventional, power-split, series, electric) and the application (e.g., HEV, PHEV).

All algorithms are based on the same concept: The vehicle is built from the bottom up, meaning each component assumption (e.g., specific power, efficiency) is taken into account to define the entire set of vehicle attributes (e.g., weight). This process is iterative, as the main component characteristics (e.g., maximum power, vehicle weight) are modified until all vehicle technical specifications are met. The transmission gear span or ratios are currently not modified to be optimized with specific engine technologies. On average, the algorithms take between five and 10 iterations to converge.

8.3.1. Conventional Vehicle Sizing Algorithm

A conventional vehicle is mainly defined by its internal combustion engine (ICE); its ability to follow a cycle or meet acceleration performance is directly linked to its power density. Therefore, the sizing algorithm focuses on calculating the mechanical power needed to meet the requirements. Figure 203 illustrates the steps in the sizing process. After a default vehicle is created, a simulation is performed to determine the engine peak power and vehicle weight.

First, the desired power is estimated to meet the gradeability and acceleration performance requirements, and engine power is updated with the maximum value.

Then, the sizing enters an acceleration loop to verify the performance (e.g., initial vehicle movement to 60 mph). The definition of initial vehicle movement (IVM) is that the vehicle must move 1 ft. (about 1/3 m) before the clock starts to record the performance time. This metric provides a more consistent result and removes phenomena that are difficult to model at initial acceleration — such as tire and clutch slip — from consideration.

Finally, the acceleration performance for passing (i.e., time to accelerate from 50 to 80 mph) is measured with the vehicle's updated parameters. At the end, the times to reach the target (0–60 mph and 50–80 mph) are compared with the simulated data, which is the main condition to exit the routine. Figure 203 shows the detailed steps of the conventional-vehicle sizing algorithm.

Figure 203. Conventional Powertrain Sizing Algorithm

8.3.2. Split HEV Sizing Algorithm

Figure 204 shows the detailed steps of the split HEV sizing algorithm.

Figure 204. Split Hybrid Electric Powertrain Sizing Algorithm

The main algorithm for split-HEV is as follows:

- The engine is sized to achieve at least 70% of the peak power required to meet VTS (acceleration performance or gradeability).
- The battery and electric machine (EM1) powers are sized for maximum regenerative braking.
- The vehicle weight is updated with respect to the engine peak power, electric machines (EM1,2) peak power and battery power.
- The electric machine (EM2) is sized as follows:
 - Start ICE at $V_{\max}$ (~57 mph for UDDS cycle). IC/E should be ON (i.e. EM2 peak power for engine start at top speed on UDDS cycle)
 - Control maximum power of engine at $V_{\text{spd}} = 0$ (i.e., EM2 peak power for engine control on performance)
 - Control ICE at maximum grade (i.e., EM2 continuous power for engine control on grade, engine power fraction going through electro-mechanical power path).

8.3.3. Parallel HEV Sizing Algorithm

Figure 205 shows the detailed steps of the parallel HEV (par HEV) sizing algorithm.

Figure 205. Parallel Hybrid Electric Powertrain Sizing Algorithm

The main algorithm for parallel HEV sizing is as follows:

- The engine is sized to achieve at least 70% of the peak power required to meet VTS (acceleration performance or gradeability).
- The battery and electric machine (EM1) powers are sized for maximum regenerative braking.
- The vehicle weight is updated with respect to the engine peak power, electric machine (EM1) peak power and battery power.

8.3.4. Parallel PHEV Sizing Algorithm

Figure 206 shows the detailed steps for the parallel PHEV (par PHEV20) sizing algorithm.

Figure 206. Parallel Plug-In Hybrid (Par PHEV20) Electric Vehicle Powertrain Sizing

Figure 207 shows the detailed steps for the parallel PHEV (par PHEV50) sizing algorithm.

Figure 207. Parallel Plug-In Hybrid (Par PHEV50) Electric Vehicle Powertrain Sizing

The main sizing algorithm for parallel PHEV sizing is as follows:

- The engine is sized to achieve at least 70% of the peak power required to meet VTS (acceleration performance or gradeability).
- The battery energy is sized to achieve the specified AER on the combined cycle (UDDS + HWFET), on the basis of adjusted energy values.
- The battery and electric machine (EM) powers are sized to follow the UDDS cycle in EV mode for Par PHEV20 and US06 cycle in EV mode for Par PHEV50 at low SOC (beginning of CS mode) or to meet the requirement of acceleration performance.
- The vehicle weight is updated with respect to the engine peak power, electric-machine (EM) peak power, and battery energy.

8.3.5. Split-PHEV Sizing Algorithm

Figure 208 shows the detailed steps of the split-PHEV (split PHEV20) sizing algorithm.

Figure 208. Split PHEV Sizing Algorithm

The main sizing algorithm for split-PHEV sizing is as follows:

- The engine is sized to achieve at least 70% of the peak power required to meet VTS (acceleration performance or gradeability).
- The battery energy is sized to achieve the specified AER on the combined cycle (UDDS + HWFET), on the basis of adjusted energy values.
- The battery and electric machine (EM1) powers are sized to follow the UDDS cycle in EV mode at low SOC (beginning of CS mode) or to meet the requirement of acceleration performance.
- The vehicle weight is updated with respect to the engine peak power, electric-machine (EM1,2) peak power, and battery energy.

- The electric machine (EM2) is sized as follows:
 - Start ICE at $V_{\max}$ (~57 mph for UDDS cycle). IC/E should be ON (i.e. EM2 peak power for engine start at top speed on UDDS cycle)
 - Control maximum power of engine at $V_{\text{spd}} = 0$ (i.e., EM2 peak power for engine control on performance)
 - Control ICE at maximum grade (i.e., EM2 continuous power for engine control on grade, engine power fraction going through electro-mechanical power path).

8.3.6. Voltec PHEV (Extended Range) Vehicle Sizing Algorithm

Figure 209 shows the detailed steps for the Voltec PHEV vehicle sizing algorithm. This algorithm is used for the EREV PHEV50 vehicle powertrain.

Figure 209. Voltec PHEV Vehicle Sizing Algorithm

The main sizing algorithm for Voltec extended-range PHEV sizing is as follows:

- The engine is sized to achieve at least 70% of the peak power required to meet VTS (acceleration performance or gradeability).
- The battery energy is sized to achieve the specified AER on the combined cycle (UDDS + HWFET), on the basis of adjusted energy values.

- The battery and electric machine (EM1) powers are sized to follow the US06 cycle in EV mode at low SOC (beginning of CS mode) or to meet the requirement of acceleration performance.
- The vehicle weight is updated with respect to the engine peak power, electric-machine (EM1,2) peak power, and battery energy.
- The electric machine (EM2) is sized as following:
 - Start ICE at $V_{\max}$ (~57 mph for UDDS cycle). ICE should be ON (i.e., EM2 peak power for engine start at top speed on UDDS cycle)
 - Control maximum power of engine at $V_{\text{spd}} = 0$ (i.e., EM2 peak power for engine control on performance)
 - Control ICE at max grade (i.e., EM2 continuous power for engine control on grade, engine power fraction going through electro-mechanical power path).

8.3.7. BEV Sizing Algorithm

Figure 210 shows the detailed steps of the BEV sizing algorithm. This algorithm is used for BEV200 and BEV300 vehicle powertrain.

Figure 210. BEV Sizing Algorithm

The main sizing algorithm for BEV sizing is as follows:

- Battery and electric machine (EM) powers are sized to be able to follow the US06 cycle at low SOC (beginning of CS mode) or to meet the requirement of acceleration performance.
- The battery energy is sized to achieve the specified AER on the combined cycle (UDDS + HWFET), on the basis of adjusted energy values.
- The vehicle weight is a function of the electric-machine (EM) peak power and battery energy

8.3.8. Fuel Cell Series HEV Sizing Algorithm

Figure 211 shows the detailed steps of the fuel cell HEV sizing algorithm:

Figure 211. Fuel Cell Series HEV Sizing Algorithm

The main sizing algorithm for fuel-cell HEV sizing is as follows:

- The fuel cell is sized to achieve at least 70% of the peak power required to meet the VTS (acceleration performance or gradeability).
- The hydrogen storage capacity is sized to achieve the specified AER on the combined cycle (UDDS + HWFET), on the basis of adjusted values.

- The vehicle weight is a function of the fuel-cell peak power, electric-machine peak power, and battery cell number.

8.3.9. Selection of Drive Cycles for Electric Machine Sizing for PHEVs

The selection of drive cycles for electric machine sizing for PHEVs is based on analysis of existing vehicles in the market. Figure 212 shows the relationship of electric machine sizes to battery energy for PHEV20s.

Figure 212. Electric Machine (kW) vs. Battery Energy (kWh) for PHEV20s

Figure 213 shows the relationship of electric machine sizes to battery energy for PHEV50s.

Figure 213. Electric Machine (kW) vs. Battery Energy (kWh) for PHEV50s

Figure 214 shows the relationship of electric machine sizes to battery energy for BEVs.

Figure 214. Electric Machine (kW) vs. Battery Energy (kWh) for BEVs

From the analyses, it can be seen that the low-range PHEVs usually have an electric machine power requirement of less than 100kW and the higher-range PHEVs have greater than 100kW (in-line with BEVs). The more aggressive power requirement could be established from following the US06 cycle in EV mode, compared to UDDS (which is applicable for low-range PHEVs).

8.4. Determining the Relationship Between Engine Displacement and Number of Cylinders

Figure 215 shows the relationship between the number of engine cylinders and engine displacement for existing vehicles in the U.S. market.

Figure 215. Relationship Between Engine Displacement and Number Of Engine Cylinders

Using this relationship, thresholds were created to define the number (and type) of engine cylinders for given engine displacements. The thresholds are shown in Table 63.

Table 63. Thresholds for Engine Displacement vs. Number of Engine Cylinders

Number and Configuration of Engine Cylinders	Engine Displacement (L)
4-cylinder, in-line (I4)	1.2, 1.4, 1.6, 1.8, 2.0, 2.2
6-cylinder, V6	2.5, 2.7, 2.9, 3.1, 3.3, 3.5, 3.7
8-cylinder, V8	4.0, 4.5, 5.0, 5.5, 6.0, 6.5, 7.0

The flowchart in Figure 216 shows the detailed method used to calculate the engine displacement and number of cylinders from the initial engine size derived from the engine map.

Figure 216. Method for Computing Engine Displacement and Number of Cylinders

Figure 217 shows the variation of engine displacement vs. number of cylinders from all simulation results across the different vehicle classes.

Figure 217. Engine Displacement vs. Number of Cylinders from Simulation Results

9. Distributed Computing Process

At this stage of the large-scale simulation process, all the vehicles are created and ready to be sized and simulated in Autonomie. Running 1.2 million vehicles requires more than 5 million simulations, from sizing algorithms — imposing recurrence and iteration/looping — to vehicle simulation on cycles and combined or PHEV procedures.

With the multitude of technology combinations to simulate and the objective of providing direct inputs from Autonomie to the CAFE model, the usual computing resources are no longer practical. Running all of the simulations on one computer would take several months or years to complete. However, thanks to advances in distributed computing, simulation time can be greatly reduced. Among the computing resources available at Argonne is a cluster of several thousand worker nodes dedicated to the Vehicle & Mobility System Group. A larger high-performance computing facility will be used in the future to further accelerate the simulations.

9.1. Setup

Autonomie is used as the simulation framework, synchronized by a cluster head node computer. The head computer extracts the data describing the different technology pathways from the vehicle files and distributes it, as diagrammed in Figure 218. An algorithm optimizes the distribution of jobs for vehicle simulations.

Figure 218. Distributed Computing Process

9.2. Distributed-Computing Flexibility

One of the biggest advantages of distributed computing is that it facilitates the quick reruns of simulations that occur many times during any study. Experience enabled Argonne to develop a new process: an ultimate large-scale simulation process that is functional, smooth, and flexible, with the ability to easily and quickly add and rerun as many vehicles and new technologies as needed. The generic process can automatically handle additional technologies without any code modification. As a result, the CAFE model's future technological needs can be easily and quickly integrated at any time, and new runs can be implemented to directly feed the model for CAFE rulemaking analyses.

10. Vehicle Simulation Process

Once the vehicles are sized or inherited to meet the desired vehicle technical specifications, they are simulated following the appropriate standard driving cycles and run procedures. It is important to properly store individual results as structured data because they will be reused to support database generation and ease of browsing. Figure 219 shows the folder organization for each individual simulation.

Figure 219. Vehicle Simulation Folder Organization

Each folder contains the results for one combination and characterizes one branch/path of the tree. Folders can contain up to five directories, depending on the vehicle technology and the type of run performed. Results are divided into directories representing the cycle or procedure simulated. For example, the combined procedure for conventional vehicles has two folders, containing the FTP and

HWFET runs, and the PHEV procedure has four folder, for the FTP and HWFET runs plus the CS and CD modes. The last directory is the sizing structure (performance test).

10.1. Data.mat File

Data.mat is the results file containing all of the vehicle parameters and the time-based signals. A sample of signals and parameters included in data.mat is shown in Figure 220.

Name	Value	Min	Max
drv	<1x1 struct>		
drv_accel_dmd_simu	<15291x1 double>	0	0.7650
drv_brk_dmd_simu	<15291x1 double>	-1	0
drv_ctrl_pwr_dmd_simu	<15291x1 double>	-3.443...	8.6684...
drv_key_on_dmd_simu	<15291x1 double>	1	1
drv_lin_spd_dmd_simu	<15291x1 double>	0	26.7777
drv_trq_dmd_simu	<15291x1 double>	-583.9...	1.5855...
eng	<1x1 struct>		
eng_ctrl_cmd_fuel_off_simu	<15291x1 double>	0	1
eng_ctrl_cmd_on_simu	<15291x1 double>	0	1
eng_ctrl_cmd_simu	<15291x1 double>	0	1
eng_ctrl_cstr_trq_hot_max_simu	<15291x1 double>	181.86...	231.95...
eng_ctrl_cstr_trq_hot_min_simu	<15291x1 double>	-37.43...	-13.56...
eng_ctrl_dmd_trq_simu	<15291x1 double>	-25.12...	221.81...
eng_ctrl_trs_fuel_cut_off_simu	<15291x1 double>	0	1
eng_ctrl_trs_on_simu	<15291x1 double>	0	1
eng_ctrl_trs_trq_simu	<15291x1 double>	-25.08...	221.77...
eng_plant_cmd_simu	<15291x1 double>	0	1
eng_plant_co_simu	<15291x1 double>	0	0
eng_plant_eff_total_simu	<15291x1 double>	0	2.4487
eng_plant_energy_in_total_simu	<15291x1 double>	0	5.4631...
eng_plant_energy_loss_total_simu	<15291x1 double>	-361.4...	4.0809...
eng_plant_energy_out_total_simu	<15291x1 double>	0	1.3872...
eng_plant_ex_equiv_simu	<15291x1 double>	0	1
eng_plant_ex_gas_flow_simu	<15291x1 double>	-4.629...	0.0962
eng_plant_ex_gas_temp_simu	<15291x1 double>	NaN	NaN
eng_plant_exterior_temp_simu	<15291x1 double>	90	90
eng_plant_fuel_cum_simu	<15291x1 double>	0	1.2746
eng_plant_fuel_rate_simu	<15291x1 double>	0	0.0039
eng_plant_hc_simu	<15291x1 double>	0	0
eng_plant_htrej_cum_simu	<15291x1 double>	0	0
eng_plant_htrej_rate_simu	<15291x1 double>	0	0
eng_plant_nox_simu	<15291x1 double>	0	0
eng_plant_o2_simu	<15291x1 double>	0	0

Figure 220. Autonomie Sample Result (data.mat)

10.2. XML Results File

As shown in Figure 221, simulation.a_result is an XML version of the results file that includes the main simulation inputs and outputs. This file is later used to generate the complete database.

```
1 <simulation DisplayName="conv_midsize_au_2wd_VOLPE" Description=" Generic midsize conventional with an automatic transmissi
2 <Version Author="amoawad" Date="03/27/2013 03:24:47.325 PM" />
3 <Property Name="Application" Value="Light Duty" />
4 <Software Name="Matlab" Version="R2010a (32-bit)" />
5 <system DisplayName="conv_midsize_au_2wd_VOLPE" name="veh" layout="vehicle.a_layout" layoutVersion="" configuration="veh.
2232 <sizing rule_file="sr_conv_accel_eng_pwr_passing" grade_compo="eng" path="\\vms-01\cluster\volpe_2013\users\amoawad\save
2314 <process DisplayName="US 2 Cycle with cost and GHG" Version="" name="us_2_cycle_with_cost_and_ghg" optional="false" FileN
2552 <setup decimation="10" gbl_stop_time="1529" />
2553 <results >
3434 <signals name="accelec_plant_curr_in_simu" />
3435 <signals name="accelec_plant_curr_out_simu" />
3436 <signals name="accelec_plant_pwr_simu" />
3437 <signals name="accelec_plant_volt_in_simu" />
3438 <signals name="accelec_plant_volt_out_simu" />
3439 <signals name="accmech_plant_pwr_simu" />
3440 <signals name="accmech_plant_spd_in_simu" />
3441 <signals name="accmech_plant_spd_out_simu" />
3442 <signals name="accmech_plant_trq_in_simu" />
3443 <signals name="accmech_plant_trq_out_simu" />
3444 <signals name="accmech_plant_trq_simu" />
3445 <signals name="chas_plant_distance_out_simu" />
3446 <signals name="chas_plant_force_grade_simu" />
3447 <signals name="chas_plant_force_in_simu" />
3448 <signals name="chas plant force loss simu" />
```

Figure 221. XML File of Autonomie Result (simulation.a_result)

11. Simulation Results Analysis and Detailed Validation Processes

The simulation results database analysis is performed using Tableau and MATLAB. This section highlights some examples of analyses performed on the simulation results. The objective is to automatically detect potential outliers in the simulations so as to perform more in-depth analysis on them.

11.1. Engine Operating Points Across Transmissions

Evaluating the engine density (i.e., operating points) across the different transmission types and numbers of gears would help us evaluate the impact of improved transmissions. The particular vehicle selected for this analysis is as follows:

- Vehicle class: Midsize
- Performance category: Non-performance
- Engine: Engine 01
- Mass reduction: MR Level 0 (MR0)
- Aerodynamic reduction: AERO Level 0 (AERO0)
- Rolling-resistance reduction: ROLL Level 0 (ROLL0)

As expected, the average engine speed decreases with increased gear number and gear span.

11.1.1. Manual Transmission

Figure 222 shows the engine operating points for 5- and 6-speed manual transmissions during UDDS and HWFET cycles.

Figure 222. Engine Operating Points for 5- and 6-Speed Manual Transmissions

11.1.2. Automatic Transmission

Figure 223 shows the engine operating points for automatic transmission types with different numbers of gears.

Figure 223. Engine Operating Points for Automatic Transmissions

11.1.3. Dual Clutch Transmissions

Figure 224 shows the engine operating points for 6- and 8-speed DCTs.

Figure 224. Engine Operating Points for 6- and 8-Speed DCT Transmissions

11.1.4. CVT transmission

Figure 225 shows the engine operating points for two CVT types.

Figure 225. Engine Operating Points for CVTs

11.2. Powertrain Efficiency Analysis

The evaluation of powertrain efficiency across different transmission types and numbers of gears as well individual powertrains would help us evaluate the impact of improved technologies.

In this section, the particular vehicle combinations selected for this analysis is as follows:

- Vehicle class: Midsize
- Vehicle powertrain: Conventional

- Performance category: Non-performance
- Aerodynamic reduction: AERO Level 0 (AERO0)
- Rolling-resistance reduction: ROLL Level 0 (ROLL0)

Figure 226 shows the powertrain efficiency values for both UDDS and HWFET cycles for the different engine types with automatic transmissions having different numbers of gears.

Figure 226. Powertrain Efficiency Values of Different Engine Types with Automatic Transmissions with Different Numbers of Gears

It can be seen that with increasing number of gears, the powertrain efficiency increases for both UDDS and HWFET. This increment is further emphasized with engines demonstrating higher efficiencies (for example, eng01 vs. eng26a).

Figure 227 shows the powertrain efficiency values for both UDDS and HWFET for different engine types with automatic+ (AU+) transmissions with different numbers of gears.

Figure 227. Powertrain Efficiency Values of Different Engine Types with AU+ Transmissions with Different Numbers of Gears

From the figure, it can be seen that going from 8-speed to 9-speed AU+ causes powertrain efficiency to decrease slightly for UDDS, due to increased shifting, as well as a drop in gearbox efficiency. However, for HWFET runs, powertrain efficiency picks up for increasing number of gears.

Figure 228 shows the powertrain efficiency values for different engine types with automatic++ (AU++) transmissions and different numbers of gears.

Figure 229 shows the powertrain efficiency values for the different engine types with DCTs and different numbers of gears.

Figure 229. Powertrain Efficiency Values of Different Engine Types with DCTs with Different Numbers of Gears

For DCT transmission, the powertrain efficiency increases with an increased number of gears for both UDDS and HWFET runs. Values also improve for engines demonstrating higher technology effectiveness (for example, eng01 vs. eng26a).

Figure 230 shows the powertrain efficiency values for different engine types with manual transmissions and different numbers of gears.

Figure 230. Powertrain Efficiency Values of Different Engine Types with Manual Transmissions with Different Numbers of Gears

It can be seen that with increasing number of gears, the powertrain efficiency increases for both UDDS and HWFET. This increment is further increased with engines demonstrating higher efficiencies (for example, eng01 vs. eng26a).

Figure 231 shows the powertrain efficiency values for different engine types with CVTs.

Figure 231. Powertrain Efficiency Values of Different Engine Types with CVT

For CVTs, the increment in powertrain efficiencies is increased with higher engine efficiencies (for example, eng01 vs. eng26a).

11.3. Total Number of Shifting Events with Different Transmission Types and Numbers of Gears

The total number of shifting events (upshift + downshift) determines the drive quality, so it is important to check the total number of shifting events across transmissions and numbers of gears against vehicle test data. Throughout the process, the following vehicle was used:

- Vehicle class: Midsize
- Performance category: Non-performance
- Powertrain type: Conventional
- Engine: Engine 01
- Aerodynamic reduction: AERO Level 0 (AERO0)
- Rolling-resistance reduction: ROLL Level 0 (ROLL0)

11.3.1. Automatic Transmission Types

Figure 232 shows the total number of shifting events for automatic (AU), automatic+ (AUp), and automatic++ (AUpp) transmissions with different numbers of gears during a UDDS cycle.

Figure 232. Total Number of Shifting Events for AU/AU+/AU+ Transmissions During UDDS Cycle

For UDDS runs, the total number of shifting events increases with increasing number of gears. However, for 10-speed transmissions, the total number of shifting events is similar to that of 8-speed transmission due to the implementation of gear-skipping methods.

Figure 233 shows the total number of shifting events for automatic (AU), automatic+ (AUp), and automatic++ (AUpp) transmissions with different numbers of gears during a US06 (a high acceleration aggressive driving schedule) cycle.

Figure 233. Total Number of Shifting Events for AU/AU+/AU++ Transmissions During US06 Cycle

For US06 runs, the total number of shifting events increases with increasing number of gears, with the exception of 10-speed transmissions due to the implementation of gear-skipping methods. The total number of shifting events for US06 runs is lower than for UDDS runs.

11.3.2. Manual Transmission Types

Figure 234 shows the total number of shifting events for manual (DM) transmissions with different numbers of gears during a UDDS cycle.

Figure 234. Total Number of Shifting Events for DM Transmissions During UDDS Cycle

For DM runs, the total number of shifting events increases with increasing number of gears. When compared to the automatic transmission type, the total number of shifting events for a 5-speed DM is higher due to the difference in gear span and final drive ratio. The total number of shifting events for a 6-speed DM is the same as for a 6-AU.

Figure 235 shows the total number of shifting events for manual (DM) transmissions with different numbers of gears during a US06 cycle.

Figure 235. Total Number of Shifting Events for DM Transmissions During US06 Cycle

As in UDDS runs, the total number of shifting events for DM transmissions during a US06 cycle run can be compared to the AU transmissions.

11.3.3. Dual Clutch Transmission Types

Figure 236 shows the total number of shifting events for DCTs with different numbers of gears during a UDDS cycle.

Figure 236. Total Number of Shifting Events for DCTs During UDDS Cycle

For UDDS runs, the total number of shifting events increases with an increasing number of gears for DCT transmissions, similar to what is observed for AU/DM transmission types.

Figure 237 shows the total number of shifting events for DCTs with different numbers of gears during a US06 cycle.

Figure 237. Total Number of Shifting Events for DCTs During US06 Cycle

11.4. Engine Power Inheritance Validation

As part of the vehicle inheritance validation, the engine power of the inherited vehicles was analyzed in comparison to baseline vehicles sized for the conventional powertrain.

Figure 238 shows the engine power of the conventional vehicles in the compact vehicle class (non-performance category) in response to the mass reduction step.

Figure 238. Engine Power vs. Mass Reduction Step (Compact, Non-Performance)

Figure 239 shows the engine power of the conventional vehicles in the compact vehicle class (performance category) in response to the mass reduction step.

Figure 239. Engine Power vs. Mass Reduction Step (Compact, Performance)

Figure 240 shows the engine power of the conventional vehicles in the midsize vehicle class (non-performance category) in response to the mass reduction step.

Figure 240. Engine Power vs. Mass Reduction Step (Midsize, Non-Performance)

Figure 241 shows the engine power of the conventional vehicles in the midsize vehicle class (performance category) in response to the mass reduction step.

Figure 241. Engine Power vs. Mass Reduction Step (Midsize, Performance)

Figure 242 shows the engine power of the conventional vehicles in the small SUV vehicle class (non-performance category) in response to the mass reduction step.

Figure 242. Engine Power vs. Mass Reduction Step (Small SUV, Non-Performance)

Figure 243 shows the engine power of conventional vehicles in the small SUV vehicle class (performance category) in response to the mass reduction step.

Figure 243. Engine Power vs. Mass Reduction Step (Small SUV, Performance)

Figure 244 shows the engine power of the conventional vehicles in the midsize SUV vehicle class (non-performance category) in response to the mass reduction step.

Figure 244. Engine Power vs. Mass Reduction Step (Midsize SUV, Non-Performance)

Figure 245 shows the engine power of the conventional vehicles in the midsize SUV vehicle class (performance category) in response to the mass reduction step.

Figure 245. Engine power vs. mass reduction step (midsize SUV, performance)

Figure 246 shows the engine power of the conventional vehicles in the pickup vehicle class (performance category) in response to the mass reduction step.

Figure 246. Engine Power vs. Mass Reduction Step (Pickup, Non-Performance)

Figure 247 shows the engine power of conventional vehicles in the pickup class (performance category) in response to the mass reduction step.

Figure 247. Engine Power vs. Mass Reduction Step (Pickup, Performance)

11.5. Vehicle Performance Analysis

As part of the performance neutrality analysis, vehicle performance (0–60 mph) vs. passing (50–80 mph) times were analyzed across the different vehicle classifications and performance categories.

11.5.1. Compact, Non-Performance

Figure 248 through Figure 256 show the vehicle performance vs. passing times of the conventional vehicles in the compact vehicle class (non-performance category). The maps of different engine types and numbers of gears across different transmission types are also highlighted.

Figure 248. Vehicle Performance (0-60 mph) vs. Passing (50-80 mph) Times (Compact, Non-Performance — Conventional)

Figure 249. Vehicle Performance (0-60 mph) vs. Passing (50-80 mph) Times (Compact, Non-Performance — Micro Hybrid)

Figure 250. Vehicle Performance (0-60 mph) vs. Passing (50-80 mph) Times (Compact, Non-Performance — Mild Hybrid)

Figure 251. Vehicle Performance (0-60 mph) vs. Passing (50-80 mph) Times (Compact, Non-Performance — Split HEV)

Figure 252. Vehicle Performance (0-60 mph) vs. Passing (50-80 mph) Times (Compact, Non-Performance — Par HEV)

Figure 253. Vehicle Performance (0-60 mph) vs. Passing (50-80 mph) Times (Compact, Non-Performance — Par PHEV20)

Figure 254. Vehicle Performance (0-60 mph) vs. Passing (50-80 mph) Times (Compact, Non-Performance — Par PHEV50)

Figure 256. Vehicle Performance (0-60 mph) vs. Passing (50-80 mph) Times (Compact, Non-Performance — EREV PHEV50)

11.5.2. Compact, Performance

Figure 257 shows the vehicle performance vs. passing times of conventional vehicles in the compact vehicle class (performance category). The maps of different engine types and numbers of gears across different transmission types are also highlighted.

Figure 257. Vehicle Performance (0-60 mph) vs. Passing (50-80 mph) Times (Compact, Performance)

11.5.3. Midsize, Non-Performance

Figure 258 shows the vehicle performance vs. passing times of conventional vehicles in the midsize vehicle class (non-performance category). The maps of different engine types and numbers of gears across different transmission types are also highlighted.

Figure 258. Vehicle Performance (0-60 mph) vs. Passing (50-80 mph) Times (Midsize, Non-Performance)

11.5.4. Midsize, Performance

Figure 259 shows the vehicle performance vs. passing times of conventional vehicles in the midsize vehicle class (performance category). The maps of different engine types and numbers of gears across different transmission types are also highlighted.

11.5.6. Small SUV, Performance

Figure 261 shows the vehicle performance vs. passing times of the conventional vehicles in the small SUV vehicle class (performance category). The maps of different engine types and numbers of gears across different transmission types are also highlighted.

Figure 261. Vehicle Performance (0-60 mph) vs. Passing (50-80 mph) Times (Small SUV, Performance)

11.5.7. Midsize SUV Non-Performance

Figure 262 shows the vehicle performance vs. passing times of conventional vehicles in the midsize SUV vehicle class (non-performance category). The maps of different engine types and numbers of gears across different transmission types are also highlighted.

Figure 262. Vehicle Performance (0-60 mph) vs. Passing (50-80 mph) Times (Midsize SUV, Non-Performance)

11.5.8. Midsize SUV, Performance

Figure 263 shows the vehicle performance vs. passing times of the conventional vehicles in the midsize SUV vehicle class (performance category). The maps of different engine types and numbers of gears across different transmission types are also highlighted.

Figure 263. Vehicle Performance (0-60 mph) vs. Passing (50-80 mph) Times (Midsize SUV, Performance)

Figure 264 shows the vehicle performance vs. passing times of conventional vehicles in the pickup vehicle class (non-performance category). The maps of different engine types and numbers of gears across different transmission types are also highlighted.

Figure 264. Vehicle Performance (0-60 mph) vs. Passing (50-80 mph) Times (Pickup, Non-Performance)

11.5.10. Pickup, Performance

Figure 265 shows the vehicle performance vs. passing times of conventional vehicles in the pickup vehicle class (performance category). The maps of different engine types and numbers of gears across different transmission types are also highlighted.

Figure 265. Vehicle Performance (0-60 mph) vs. Passing (50-80 mph) Times (Pickup, Performance)

11.6. Validation Against Vehicle Technology Database

The vehicle simulation results are further validated against existing vehicles in the market using an internal vehicle technology database from Argonne. The analysis is performed for fuel economy and acceleration times across the different vehicle classes and performance categories. The vehicle simulation results and the vehicles from the vehicle technology database represent all combinations of the conventional powertrain.

11.6.1. Fuel Economy, UDDS, Unadjusted (mpg)

Figure 266 shows the variation in fuel economy (unadjusted) on the UDDS cycle for the different vehicles in the market compared to vehicle simulations performed across the different vehicle classes and performance categories.

Fuel Economy, UDDS Unadj. (mpg) [Existing vehicles]

Fuel Economy, UDDS Unadj. (mpg) [Argonne simulation]

Figure 266. Fuel Economy (Unadj.) Compared on UDDS Cycle — Vehicles in the Market (Top Panel) and Vehicle Simulation Results (Bottom Panel)

11.6.2. Fuel Economy, HWFET, Unadjusted (mpg)

Figure 267 shows the variation in fuel economy (unadjusted) on the HWFET cycle for the different vehicles in the market compared to the vehicle simulations performed across the different vehicle classes and performance categories.

Fuel Economy, HWFET Unadj. (mpg) [Existing vehicles]

Fuel Economy, HWFET Unadj. (mpg) [Argonne simulation]

Figure 267. Fuel Economy (Unadj.) Compared on HWFET Cycle — Vehicles in the Market (Top Panel) and Vehicle Simulation Results (Bottom Panel)

11.6.3. Glider / Powertrain Weight vs. Vehicle Curb Weight Analysis

Figure 268 shows the relationship of glider weight to curb weight for compact vehicles in the market compared to the vehicles simulated as part of this analysis.

Figure 268. Glider Weight vs. Curb Weight Distribution (Compact Class)

Figure 269 shows the relationship between powertrain weight and curb weight for compact vehicles in the market compared with the vehicles simulated as part of this analysis.

Figure 269. Powertrain Weight vs. Curb Weight (Compact Class)

Figure 270 shows the relationship of percentage of powertrain weight to percentage of glider weight for compact vehicles in the market in comparison with the compact vehicles simulated as part of this analysis.

Figure 270. Percent Powertrain Weight vs. Percent Glider Weight (Compact Class)

Figure 271 shows the glider weight vs. curb weight distribution for midsize vehicles in the market compared with the simulated midsize vehicles used in this analysis.

Figure 271. Glider Weight vs. Curb Weight Distribution (Midsize Class)

Figure 272 shows powertrain weight vs. curb weight for midsize vehicles in the market compared with the midsize vehicles that are simulated as part of this analysis.

Figure 272. Powertrain Weight vs. Curb Weight (Midsize Class)

Figure 273 shows the relationship of percentage of powertrain weight to percentage of glider weight for midsize vehicles in the market compared with the compact vehicles simulated as part of this analysis.

Figure 273. Percent Powertrain Weight vs. Percent Glider Weight (Midsize Class)

Figure 274 shows the glider weight vs. curb weight distribution for midsize vehicles in the market compared with the simulated midsize vehicles used in this analysis

Figure 274. Glider Weight vs. Curb Weight Distribution (SUVs)

Figure 275 shows the distribution of powertrain weight vs. vehicle curb weight for different SUVs in the market compared to the simulated SUVs used in this analysis.

Figure 275. Powertrain Weight vs. Curb Weight (SUVs)

Figure 275 shows the relationship of percentage of powertrain weight to percentage of glider weight for different SUVs in the market compared with the simulated vehicles used in this analysis.

Figure 276. Percent Powertrain Weight vs. Percent Glider Weight (SUVs)

Figure 277 shows the glider weight vs. curb weight distribution for compact BEVs in the market compared with the simulated midsize vehicles used in this analysis.

Figure 277. Glider Weight vs. Curb Weight (Compact BEVs)

Figure 278 illustrates the powertrain weight vs. curb weight distribution for compact BEVs in the market compared to simulated vehicles used in this analysis.

Figure 278. Powertrain Weight vs. Curb Weight (Compact BEVs)

Figure 279 shows the percentage of powertrain weight vs. percentage of glider weight for compact BEVs in the market compared with simulated vehicles used in the analysis.

Figure 279. Percent Powertrain Weight vs. Percent Glider Weight for Compact BEVs

Figure 280 shows the glider weight vs. curb weight distribution for midsize BEVs in the market compared with the simulated midsize BEVs used in this analysis.

Figure 280. Glider Weight vs. Curb Weight (Midsize BEVs)

Figure 281 shows the powertrain weight vs. vehicle curb weight for midsize BEVs in the market compared with the simulated midsize BEVs used in this analysis.

Figure 281. Powertrain Weight vs. Curb weight (Midsize BEVs)

Figure 282 shows the distribution of percentage of powertrain weight vs. percentage of glider weight for midsize BEVs in the market compared with the simulated midsize BEVs used in this analysis.

Figure 282. Percent Powertrain Weight vs. Percent Glider Weight Distribution (Midsize BEVs)

Figure 283 shows the glider weight vs. curb weight distribution for SUV BEVs in the market compared with the simulated vehicles used in this analysis.

Figure 283. Glider Weight vs. Curb Weight Distribution (SUV BEVs)

Figure 284 shows the powertrain weight vs. vehicle curb weight for SUV BEVs in the market compared with the simulated SUV BEVs used in this analysis.

Figure 284. Powertrain Weight vs. Curb Weight (SUV BEVs)

Figure 285 shows the percentage of powertrain weight vs. percentage of glider weight for SUV BEVs in the market compared with simulated SUV BEVs used in this analysis.

Figure 285. Percent Powertrain Weight vs. Percent Glider Weight (SUV BEVs)

Figure 286 shows the 0-60 mph acceleration timespans across different powertrains of midsize vehicles in the market compared with the midsize vehicles simulated as part of this analysis.

Figure 286. Acceleration Timespans (0-60 mph) Comparison

Figure 287 shows the spread of acceleration times (0-60 mph), across different powertrains of midsize vehicles in the market and, in detail, the variability observed across different vehicle models with a given powertrain.

Figure 287. Acceleration Time Spread Across Powertrains in the Market

Figure 288 shows the adjusted electrical energy consumption vs. vehicle curb weight for existing BEVs in the market compared with BEV200 simulated vehicles used in this analysis.

Figure 288. Electrical Energy Consumption vs. Vehicle Curb Weight (BEV200)

Figure 289 shows the motor-specific power vs. vehicle acceleration time for existing BEVs in the market compared with simulated BEV200 vehicles used in this analysis.

Figure 289. Motor-Specific Power vs. Acceleration Time (BEV200)

Figure 290 shows the total battery energy vs. all-electric range for existing BEVs in the market compared with simulated BEV200 vehicles used in the analysis.

Figure 290. Battery Energy vs. Combined Range (BEV200)

Figure 291 shows the adjusted electrical energy consumption vs. vehicle curb weight for existing BEVs in the market compared with simulated BEV300 vehicles used in this analysis.

Figure 291. Electrical Energy Consumption vs. Vehicle Curb Weight (BEV300)

Figure 292 shows the total battery energy vs. all-electric range for existing BEVs in the market and compared with simulated BEV300 vehicles used in the analysis.

Figure 292. Battery Energy vs. Combined Range (BEV300)

Figure 293 shows the motor specific power vs. vehicle acceleration time for existing BEVs in the market compared with simulated BEV300 vehicles used in this analysis.

Figure 293. Motor-Specific Power vs. Acceleration Time (BEV300)

Figure 294 shows the distribution of total battery energy vs. vehicle curb weight for existing BEVs in the market compared with simulated BEV200 vehicles used in the analysis.

Figure 294. Battery Energy vs. Vehicle Curb Weight (BEV200)

Figure 295 shows the distribution of total battery energy vs. vehicle curb weight for existing BEVs in the market compared with simulated BEV300 vehicles used in this analysis.

Figure 295. Battery Energy vs. Vehicle Curb Weight (BEV300)

11.7. Machine Learning for Outlier Detection

To validate the simulation results using machine learning, a random sample consensus algorithm (RANSAC) method was used. The technique involves iteratively estimating the parameters and best model that fit sufficiently many points (supposed inliers). Outliers will not fit the true model within some error (maximum deviation) attributed to the effect of noise.

For example, Figure 296 shows that most simulation values fit the model within the given confidence interval bands.

Figure 296. Confidence Interval Bands of Simulation Results

Figure 297, however, shows a situation where some engineering sense is required to conclude that the point detected by the RANSAC method to be an outlier is actually not an outlier.

Figure 297. RANSAC Method Validation

The RANSAC-generated model is a reasonable approximation of the simulation results.

11.8. Statistical Methods

Most statistical tests rely on the assumption of normality, so it is crucial to first determine whether the generated data is normally distributed. For results validation, the quantile-quantile (QQ) plot is a simple way to graphically confirm whether the data come from a normal distribution. Figure 298 shows the QQ plot of the simulation results.

Figure 298. QQ Plot To Confirm the Normal Distribution of Results

12. Appendix: Summary of All Assumptions

[illegible]

Category	Run	Class Of Interest			
Class	1	Compact_NonPerfo			
Class	1	Compact_Perfo			
Class	1	Midsize_NonPerfo			
Class	1	Midsize_Perfo			
Class	1	Small_SUV_NonPerfo			
Class	1	Small_SUV_Perfo			
Class	1	Midsize_SUV_NonPerfo			
Class	1	Midsize_SUV_Perfo			
Class	1	Pickup_NonPerfo			
Class	1	Pickup_Perfo			
Category	Run	EngineList	EnginePower		
Engine	1	eng01	108002		
Engine	1	eng02	108002		
Engine	1	eng03	113064		
Engine	1	eng04	113064		
Engine	1	eng5b	114137		
Engine	1	eng6a	109001		
Engine	1	eng7a	114137		
Engine	1	eng8a	114137		
Engine	1	eng12	143998		
Engine	1	eng13	131999		
Engine	1	eng14	131999		
Engine	1	eng12Deac	143998		
Engine	1	eng17	140728		
Engine	1	eng18	113064		
Engine	1	eng19	108002		
Engine	1	eng20	108002		
Engine	1	eng21	113064		
Engine	1	eng22b	131774		
Engine	1	eng23b	139003		
Engine	1	eng23c	139003		
Engine	1	eng24	100531		
Engine	1	eng25	100531		
Engine	1	eng26a	180001		
Category	Run	TransmissionList	Transmission Init File NonPerfo	Transmission Init File Perfo	Earlylockup file
Transmission	1	5AU	gb_plant_5_au_base_volpe.m	gb_plant_5_au_premium_volpe.m	cpl_ctrl_au_earlyldcp_6spd_CAFE_init.m
Transmission	1	6AU	gb_plant_6_au_base_volpe.m	gb_plant_6_au_plus_volpe.m	cpl_ctrl_au_earlyldcp_6spd_plus_CAFE_init.m
Transmission	1	7AU	gb_plant_7_au_volpe.m	gb_plant_7_au_volpe.m	cpl_ctrl_au_earlyldcp_7spd_CAFE_init.m
Transmission	1	8AU	gb_plant_8_au_volpe.m	gb_plant_8_au_volpe.m	cpl_ctrl_au_earlyldcp_8spd_CAFE_init.m
Transmission / Pre-Transmission	1	8AU	gb_plant_8_au_plus_volpe.m	gb_plant_8_au_plus_volpe.m	cpl_ctrl_au_earlyldcp_8spd_plus_CAFE_init.m
Transmission	1	8AUpp	gb_plant_8_au_plusplus_volpe.m	gb_plant_8_au_plusplus_volpe.m	cpl_ctrl_au_earlyldcp_8spd_plus_CAFE_init.m
Transmission	1	9AU	gb_plant_9_au_volpe.m	gb_plant_9_au_volpe.m	cpl_ctrl_au_earlyldcp_9spd_CAFE_init.m
Transmission	1	10AU	gb_plant_10_au_plus_volpe.m	gb_plant_10_au_plus_volpe.m	cpl_ctrl_au_earlyldcp_10spd_plus_CAFE_init.m
Transmission	1	10AUpp	gb_plant_10_au_plusplus_volpe.m	gb_plant_10_au_plusplus_volpe.m	cpl_ctrl_au_earlyldcp_10spd_plus_CAFE_init.m
Transmission	1	6DCT	gb_plant_6_dct_volpe.m	gb_plant_6_dct_volpe.m	
Transmission	1	8DCT	gb_plant_8_dct_volpe.m	gb_plant_8_dct_volpe.m	
Transmission	1	5DM	gb_plant_5_dm_volpe.m	gb_plant_5_dm_volpe.m	
Transmission	1	6DM	gb_plant_6_dm_volpe.m	gb_plant_6_dm_volpe.m	
Transmission	1	7DM	gb_plant_7_dm_volpe.m	gb_plant_7_dm_volpe.m	
Transmission	1	CVT	gb_plant_ct_05_32_volpe.m	gb_plant_ct_05_32_volpe.m	
Transmission	1	CVTp	gb_plant_ct_045_36_plus_volpe.m	gb_plant_ct_045_36_plus_volpe.m	
Category	Run	MRList	Note		
MR	1	MRO	0		
MR	1	MR1	0.05		
MR	1	MR2	0.075		
MR	1	MR3	0.1		
MR	1	MR4	0.15		
MR	1	MR5	0.2		
MR	1	MR6	0.3		
Category	Run	AEROList	Aerodynamics		
AERO	1	AERO0	0		
AERO	1	AERO1	0.05		
AERO	1	AERO2	0.1		
AERO	1	AERO3	0.15		
AERO	1	AERO4	0.2		
Category	Run	ROLLList	RollingResistance		
ROLL	1	ROLL0	0		
ROLL	1	ROLL1	0.1		
ROLL	1	ROLL2	0.2		
Category	Run	Wheels	Ideal Wheel Radius		
Wheels	1	Compact_NonPerfo	0.31725		
Wheels	1	Compact_Perfo	0.31725		
Wheels	1	Midsize_NonPerfo	0.31725		
Wheels	1	Midsize_Perfo	0.31725		
Wheels	1	Small_SUV_NonPerfo	0.35925		
Wheels	1	Small_SUV_Perfo	0.35925		
Wheels	1	Midsize_SUV_NonPerfo	0.3677		
Wheels	1	Midsize_SUV_Perfo	0.3677		
Wheels	1	Pickup_NonPerfo	0.38165		
Wheels	1	Pickup_Perfo	0.38165		

		Units	Compact_base			Compact_premium			Midsize_base			Midsize_premium			Small_SUV_base			Small_SUV_premium			Midsize_SUV_base			Midsize_SUV_premium			Pickup_base			Pickup_premium		
			Conv-S	Conv-C	Hybrid	Conv-S	Conv-C	Hybrid	Conv-S	Conv-C	Hybrid	Conv-S	Conv-C	Hybrid	Conv-S	Conv-C	Hybrid	Conv-S	Conv-C	Hybrid	Conv-S	Conv-C	Hybrid	Conv-S	Conv-C	Hybrid	Conv-S	Conv-C	Hybrid	Conv-S	Conv-C	Hybrid
	Base Electrical Acc Load	W	250	250	275	300	300	375	250	250	275	300	300	375	300	300	325	300	300	375	300	300	325	350	350	375	300	300	325	300	300	375
	Additional Load for PHEVs & BEVs	W			100			100			100			100			100			100			100			100			100			100
	Cold Start Penalty	%	15	15	15	15	15	15	15	15	15	15	15	15	15	15	15	15	15	15	15	15	15	15	15	15	15	15	15	15	15	15
	Cold Start Penalty on Bag 2	%	2.5	2.5	2.5	2.5	2.5	2.5	2.5	2.5	2.5	2.5	2.5	2.5	2.5	2.5	2.5	2.5	2.5	2.5	2.5	2.5	2.5	2.5	2.5	2.5	2.5	2.5	2.5	2.5	2.5	2.5
	Charger Efficiency	%	96	96	96	96	96	96	96	96	96	96	96	96	96	96	96	96	96	96	96	96	96	96	96	96	96	96	96	96	96	96
	Atkinson Engine Peak Efficiency*	-			0.41			0.41			0.41			0.41			0.41			0.41			0.41			0.41			0.41			0.41
HEV High Power Applications	Energy Oversize	%	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
	Power Oversize	%	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
	SOC Max	%	70	70	70	70	70	70	70	70	70	70	70	70	70	70	70	70	70	70	70	70	70	70	70	70	70	70	70	70	70	70
	SOC Init	%	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60
	Specific Power @ 70% SOC**	W/kg																														
PHEV High Energy Applications	Energy Oversize	%	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
	Power Oversize	%	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
	SOC Max	%	95	95	95	95	95	95	95	95	95	95	95	95	95	95	95	95	95	95	95	95	95	95	95	95	95	95	95	95	95	95
	SOC Init	%	25	25	25	25	25	25	25	25	25	25	25	25	25	25	25	25	25	25	25	25	25	25	25	25	25	25	25	25	25	25
	Specific Power @ 70% SOC**	W/kg																														
AEV Applications	Energy Density **	Wh/kg																														
	SOC Max	%	95	95	95	95	95	95	95	95	95	95	95	95	95	95	95	95	95	95	95	95	95	95	95	95	95	95	95	95	95	95
	SOC Init	%	5	5	5	5	5	5	5	5	5	5	5	5	5	5	5	5	5	5	5	5	5	5	5	5	5	5	5	5	5	5
Electric Machine Applications	Energy Density **	Wh/kg																														
	Motor Specific Power****	W/kg																														
Fuel Cell System	Motor + Controller Peak Efficiency***	%	96	96	96	96	96	96	96	96	96	96	96	96	96	96	96	96	96	96	96	96	96	96	96	96	96	96	96	96	96	96
	Specific Power FC System	W/kg			650			650			650			650			650			650			650			650			650			650
	Peak Fuel Cell System Efficiency	%			60			60			60			60			60			60			60			60			60			60
	Fuel Mass HEV	kg			3			3			3			3			3			3			3			3			3			3
	Power Oversize	%			10			10			10			10			10			10			10			10			10			10
	Fuel Cell Tank Constant	kg			28			28			28			28			28			28			28			28			28			28
	Fuel Cell Tank Slope	-			21			21			21			21			21			21			21			21			21			21
	Percent H2 used in Tank	-			1			1			1			1			1			1			1			1			1			1
* Based on Prius 2017 Peak Efficiency																																
** Battery Energy/Power Density driven by BatPac																																
*** For FC HEV & BEVs, motor + controller efficiency scaled to 98%																																
**** For 12V Start-Stop, and BISG, Motor Specific Power is used at 562.5 W/kg. For other Hybrids, the equation used is Motor Weight (kg) = 0.113639 * (Electric Machine Peak Power [kW]) + 27.07																																

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