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DEVELOPMENT OF THE ECOCAR 3 PROPOSAL AND GUIDELINES FOR MODELING AND DESIGN IN YEAR ONE OF ECOCAR 3

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DEVELOPMENT OF THE ECOCAR 3 PROPOSAL AND GUIDELINES FOR MODELING AND DESIGN IN YEAR ONE OF ECOCAR 3

By

Tyler B. Daavettila

A REPORT

Submitted in partial fulfillment of the requirements for the degree of

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This report has been approved in partial fulfillment of the requirements for the Degree of MASTER OF SCIENCE in Mechanical Engineering.

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Preface

The modeling and design work discussed in this report was a collaborative effort between Amir Rezaei and I, Tyler Daavettila. Amir's work concentrated on the controls logic and the simulation system including creation of the Matlab/Simulink cosimulation files. The AMESim models of the physical vehicle system were created by Amir with discussion and suggestions from Tyler. Tyler's work focused on parameterizing and exercising the models and drawing conclusions from the results along with writing the majority of the report. Amir contributed to the writing contained in sections 4.1, 4.2, 4.3, 6.3, 7.1, 9.2.9, 9.2.10, and 9.2.11. Amir's contribution is greatly appreciated and his wealth of knowledge in modeling and controls has helped to make this report possible.

Mr. Bob Page contributed to the content and format of this report through the editing process. Mr. Page provided guidance and suggestions on the content of the report and provided editorial corrections and additions to the report. His assistance throughout the writing process made the quality of this final version possible.

Nomenclature

RFP	Request for Proposal
ICE	Internal Combustion Engine
BEV	Battery Electric Vehicle
HEV	Hybrid Electric Vehicle
EV	Electric Vehicle
CAFE	Corporate Average Fuel Economy
UDDS	Urban Dynamometer Driving Schedule
HWFET	Highway Fuel Economy Driving Schedule
US06	Supplemental Federal Test Procedure Driving Schedule
FE	Fuel Economy
EC	Energy Consumption
mpg	miles per gallon
GHG	Greenhouse Gas
WTW	Well-to-Wheels
HVAC	Heating, Ventilation, and Air Conditioning
CS	Charge Sustaining
CD	Charge Depleting
GVWR	Gross Vehicle Weight Rating
C_d	Drag Coefficient
A_f	Frontal Area
C_{rr}	Rolling Resistance Coefficient
M	Vehicle Mass
g	Acceleration Due to Gravity
α	Road Angle
F_{tr}	Tractive Force
M_i	Inertial Mass
a	Vehicle Acceleration
N_i	Gear Ratio
N_f	Final Drive Ratio

SI	Spark Ignition
CI	Compression Ignition
CO_2	Carbon Dioxide
EPA	Environmental Protection Agency
NAE	National Academy of Engineers
NREL	National Renewable Energy Laboratory
kW	Kilowatt
kph	Kilometers per Hour
mph	Miles per Hour
Nm	Newton-Meter
kg	Kilogram
W	Watt
Wh	Watt-hour
J	Joule
S-HEV	Series HEV
E-REV	Extended Range EV
P	Power
ε	Efficiency
SOC	State of Charge
E10	Gasoline with 10% Ethanol Content
E85	Gasoline with 85% Ethanol Content
B20	Biodiesel with 20% bio content

Abstract

This report summarizes the work done for the Vehicle Powertrain Modeling and Design Problem Proposal portion of the EcoCAR3 proposal as specified in the Request for Proposal from Argonne National Laboratory.

The results of the modeling exercises presented in the proposal showed that:

- An average conventional vehicle powered by a combustion engine could not meet the energy consumption target when the engine was sized to meet the acceleration target, due the relatively low thermal efficiency of the spark ignition engine.
- A battery electric vehicle could not meet the required range target of 320 km while keeping the vehicle weight below the gross vehicle weight rating of 2000 kg. This was due to the low energy density of the batteries which necessitated a large, and heavy, battery pack to provide enough energy to meet the range target.
- A series hybrid electric vehicle has the potential to meet the acceleration and energy consumption parameters when the components are optimally sized.
- A parallel hybrid electric vehicle has less energy conversion losses than a series hybrid electric vehicle which results in greater overall efficiency, lower energy consumption, and less emissions.

For EcoCAR3, Michigan Tech proposes to develop a plug-in parallel hybrid vehicle (PPHEV) powered by a small Diesel engine operating on B20 Bio-Diesel fuel. This architecture was chosen over other options due to its compact design, lower cost, and its ability to provide performance levels and energy efficiency that meet or exceed the design targets. While this powertrain configuration requires a more complex control system and strategy than others, the student engineering team at Michigan Tech has significant recent experience with this architecture and has confidence that it will perform well in the events planned for the EcoCAR3 competition.

1. **Introduction**

Diminishing fossil fuel reserves and increasing concern over greenhouse gas emissions drive the urgency for developing new vehicle propulsion technologies and more efficient vehicle powertrains. Federal agencies created ambitious new fuel economy regulations to push automotive companies to develop new technologies, reduce vehicle mass, and increase vehicle electrification[1]. Introducing new technologies and whole new architectures increases the number of options involved in design decisions and the complexity of vehicle designs. Evaluating each component combination with the old prototype and test method while still meeting time constraints is not possible. Engineers must use computer-aided engineering tools for system level vehicle modeling and simulation to evaluate the vast array of options and make well informed design decisions. Modeling and simulation can reduce development time and even cost while creating a more optimized design.

Models mathematically represent a physical system. A model's capability depends on how accurately the set of equations in a model match the physics governing the behavior of the system. An accurate model of a complex system demands a lot of development time, simulation time, and computing resources. Limited time and resources force engineers to determine the minimum level of model accuracy that will provide useful results. Vehicle system models used for architecture and power requirement evaluation require less accuracy than a model used for crankshaft design.

The design process begins with a detailed set of design targets. These targets provide an end goal and are used to evaluate each decision throughout the design process. The design targets provided in the RFP are shown in Table 1.1[2]. These targets are used in the modeling problems presented in this proposal.

Table 1.1: Vehicle Modeling and Design Targets

Performance/Utility Category	Vehicle Modeling Design Targets
Energy consumption	Better than 370 Wh/km combined city/highway (55% /45%, respectively)
GHG emissions (WTW combined city/highway)	Less than 120 g of CO_2 /km
Interior size/number of passengers	Minimum of four passengers
Luggage capacity	More than 230 L
Range	Greater than 320 km combined city and highway
Top Speed	Greater than 135 kph
Acceleration time of 0 to 97 kph (0 to 60 mph)	Less than 11 seconds
Highway gradeability (at gross vehicle weight rating [GVWR])	Greater than 3.5% grade at constant 97 kph for 20 minutes

1.1. Vehicle Types and Powertrain Architectures

A variety of powertrain architectures will be discussed in this proposal; therefore a brief summary is necessary to introduce each architecture. A conventional vehicle is propelled by an internal combustion engine (ICE) energized by liquid fuel. A battery electric vehicle (BEV) is propelled by an electric machine that draws energy from a battery. A hybrid vehicle uses two energy sources on board the vehicle. The most common hybrid vehicle is the hybrid electric vehicle (HEV) which typically utilizes liquid fuel and electric energy drawn from a battery. There are several different subtypes or architectures of HEVs, with the two most common being series and parallel. In a series HEV, the electric motor propels the vehicle while the ICE drives a generator to provide electricity. In a parallel HEV, the electric motor and ICE can simultaneously or independently propel the vehicle. Further definition of powertrain architecture and components is provided where necessary.

1.2. Energy Consumption

Fuel economy is a traditional US industry standard vehicle design target. It is important to understand the difference between *actual fuel economy*, *certified fuel economy*, and *reported fuel economy*. The *actual fuel economy* observed by a driver can vary dramatically since fuel economy is highly dependent on driving style and ambient conditions. *Certified fuel economy* and *reported fuel economy* are determined through rigorous vehicle testing under controlled conditions using standard driving cycles. The calculations for these values are defined by government standards. *Certified fuel economy* refers to corporate average fuel economy (CAFE) while *reported fuel economy* the number displayed on window stickers of new vehicles to inform consumers. The Urban Dynamometer Driving Schedule (UDDS), Highway Fuel Economy Driving Schedule (HWFET), and Supplemental Federal Test Procedure Driving Schedule (US06) are three drive cycles that are used to determine certified and reported fuel economy [21].

However, the EcoCAR3 design targets define energy consumption rather than fuel economy. Consumption is an inverse of economy, showing the energy consumed per distance rather than the distance per energy consumed. The values are determined similarly to the fuel economy numbers, but care must be taken in the calculation process.

The energy consumption values calculated and reported in this proposal are determined by the following calculation:

$$EC_{combined} = \frac{0.55}{EC_{city}} + \frac{0.45}{EC_{highway}} \tag{1}$$

Where EC_{city} and $EC_{highway}$ are the unadjusted energy consumption values for the UDDS and HWFET cycles, respectively [2]. The energy consumption in watt-hours per mile (Wh/mi) is related to fuel economy (mpg) by the following equation:

The energy consumption in watt-hours per mile (Wh/mi) is related to fuel economy (mpg) by the following equation:

$$\left(X\ \frac{mi}{gal}\right)^{-1} \cdot \rho \cdot LHV \cdot \frac{1000\ Wh}{3.6\ MJ} = Y\ \frac{Wh}{mi} \tag{2}$$

Where ρ and LHV are the density (kg/gallon) and lower heating value (MJ/kg) of the fuel, respectively. When performing this calculation for various fuels the correct density values should be substituted.

1.3. Emissions

Greenhouse gases (GHG) include carbon dioxide, methane, and nitrous oxide. These gases trap heat in the atmosphere and are the most scrutinized vehicle emissions gases when considering anthropogenic effects on climate [3]. The team quantitatively evaluated GHG emissions for each modeling task with NREL's GREET tool. Other vehicle emissions, such as particulate matter, are discussed qualitatively. The simulation tools used are capable of calculating emissions constituents over a wide range of engine operation if suitable data is available for that particular engine. It was determined to be beyond the scope of this modeling assignment, but will be part of the model that will be developed for EcoCAR3. Vehicle emissions are only one part of the total emissions that are accounted for by the well-to-wheels

(WTW) greenhouse gas (GHG) emissions calculation. The WTW method also takes into account the emissions that are generated from the process of extracting and refining the gasoline or the fuel that is used to generate electricity. The electricity generation emissions are only factored in when doing the analysis for an EV or HEV.

1.4. Consumer Expectations

While fuel economy is a primary design target, many consumers will not accept improved fuel economy at the expense of reduced performance or passenger comfort. Therefore, it is imperative to find ways to improve fuel economy while maintaining performance and comfort characteristics, such as cargo area and payload capacity, a spacious and quiet passenger compartment, power adjustable seats, entertainment/information center, and advanced HVAC. While these features represent a significant design challenge, such items are considered non-negotiable in order to maintain customer acceptance and, thus, sales.

1.5. Range

The total vehicle range is another key performance factor, since an acceptable operating range is necessary to alleviate consumer concerns regarding early production EV and HEV models. The maximum distance traveled with one tank of fuel or one full battery charge is the vehicle's range. Range becomes more complicated when analyzing hybrid vehicles since there are multiple energy sources, powertrain modes, and power flow paths. The range must be determined separately for each of these options. Hybrid vehicle operation is typically divided into charge sustaining (CS) mode and charge depleting (CD) mode. For an HEV in CD mode, the battery is the only energy source, so the vehicle is operating as an EV. In CS mode, the engine operates to maintain the battery charge at a constant level while the vehicle is driving. CD range is limited by the battery capacity while CS range is limited by the fuel tank capacity.

1.6. Acceleration

Acceleration is a primary performance factor used to evaluate vehicle drivability. Variations in measurement/calculation methods can lead to differences in reported acceleration time of up to a second or more. For the acceleration values reported in this proposal, the acceleration time is defined as the first instance of non-zero velocity to the time that the vehicle reaches 60 miles per hour (96.6 kph).

1.7. Top Speed

Vehicle top speed must meet the design target to ensure acceptable performance during highway driving. However, it was not used as a primary design factor during modeling. Instead, Michigan Tech's design process targeted energy consumption and acceleration, using top speed performance as a final check for consumer acceptability.

1.8. Gradeability

The transmission gear ratios and final drive ratio impact a vehicle's ability to negotiate a slope under a given set of conditions, such as velocity, cargo and passenger load, and road surface. It is a good measure of vehicle capability under sustained high load and is a key consumer expectation. Gradeability can be reported as a road angle or percent grade value. The grade value can be converted to road angle by:

$$\tan^{-1}\left(\frac{grade}{100}\right) = road\ angle \qquad (3)$$

where the grade is in percent and the road angle is in degrees. Gradeability performance was determined by simulating vehicle operation over successively steeper grades until the powertrain failed to move the vehicle at the target speed.

2. Model Development Methodology

2.1. Simulation Platform Selection

The selection of the modeling platform was governed by software capability, prior experience, and availability of technical resources. Since no single software package had the desired capabilities, a co-simulation environment was developed to combine the strengths of two different packages. LMS ImagineLabs's AMESim software was chosen for vehicle plant models. AMESim is widely used for modeling physical systems and it includes an extensive library of mechanical, electrical, thermal, and hydraulic components that can be assembled to form complex systems. Custom sub-models and components can be developed with the AMESet facility. The control strategy was implemented in Mathworks' MATLAB/Simulink, which is well suited for graphical block-based control logic development. Figure 2.1 shows the co-simulation system and the interactions between the systems.

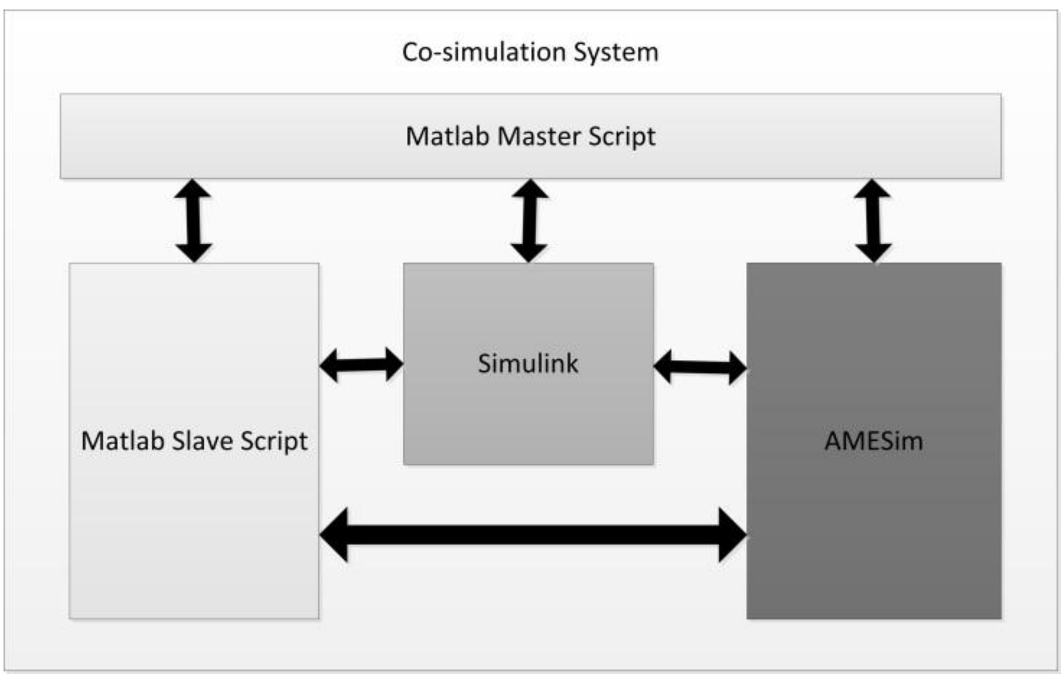


Figure 2.1: Co-simulation System

MATLAB scripts were developed to define drive cycles, environment and component parameters, and controller variables. A master script was written to call all of the necessary files, including the Simulink controller and AMESim vehicle model and execute the co-simulation for a single drive cycle or in batch runs with unique parameter sets for each drive cycle.

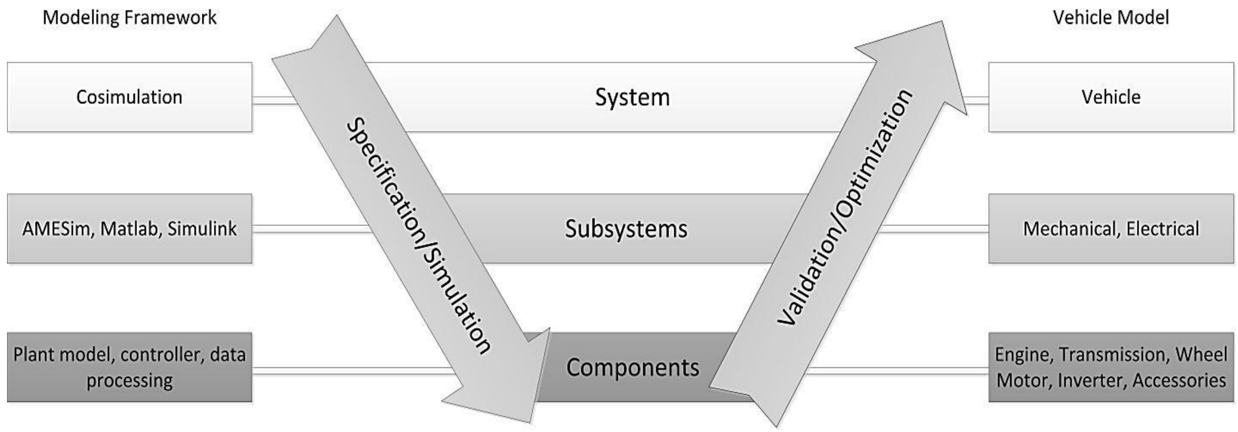


Figure 2.2. Model Development Process [3]

Vehicle modeling can be done with one of two different overall modeling methods: forward-looking or backward-looking. The best choice of method depends on the intended use and desired complexity of the model. Backward-looking modeling is the simpler method and starts with the result or action desired from the physical system and works back to determine what inputs are required to produce that result. Forward-looking modeling starts with providing inputs to the system and the result is dependent on how the model estimates the behavior of the system. Forward-looking modeling better represents how an actual vehicle system works: the driver and environment provide input and the vehicle system reacts to these inputs.

Backward-looking modeling is useful when determining powertrain requirements for following a certain velocity profile. Forward-looking modeling is useful when predicting maximum system response or reaction to a certain set of inputs, situations where the final action is unknown. Regardless of the method chosen , a knowledge of the overall vehicle parameters is necessary to determine the resistance force acting on the vehicle as it moves along the road. These vehicle parameters include vehicle body frontal area, drag coefficient, mass, tire type and road surface. The forces acting on the vehicle body are shown in Figure 3.1. When using the forward-looking method the tractive force is determined by the powertrain model response to the driver input. The acceleration is then calculated by solving the tractive force equation.

Both the backward and forward-looking methods were used to solve the modeling problems laid out in the RFP. The first problem requested the forces and power required to complete a drive cycle to be calculated. This problem was solved using the backward-looking method since the desired velocity is already known. The rest of the problems requested the energy needed to complete a given drive cycle when using a certain powertrain to provide the tractive force. The desired end result is not defined, rather, a set of inputs is defined and the model response must be determined. This type of problem is better suited for a forward-looking modeling approach, which is what was used to solve these problems.

While it would be possible to solve these mathematical models manually or with a generic solver, a specialized modeling software facilitates fast model development and simulation. Two modeling software were used for solving the problems in the RFP – Matlab/Simulink and LMS ImagineLab's AMESim.

2.2. Development Process

The models used in this proposal were developed using methodologies taught in the hybrid electric vehicle engineering curriculum at Michigan Tech. The development process follows a common V-diagram approach shown in Figure 2.2. The modeling framework and the individual vehicle models were developed with this process. The modeling framework development process began by specifying the co-simulation system requirements and then went on to determine the required interaction between the software packages (AMESim, MATLAB/Simulink). A basic vehicle model was created to use with the modeling framework development to provide a simple tool with minimal error sources. The modeling framework was iteratively developed following the V-diagram, with added complexity at each step, which was tested for complete functionality and validity of results. Using this process, complex tasks were developed, such as specialized drive cycles, separate code and function interactions, and data processing and calculations. The iterative process allowed streamlined troubleshooting to identify and eliminate sources of error. The vehicle plant model was developed in similar fashion, beginning with the overall vehicle specifications. The next step was to determine which components were required in the mechanical and electrical subsystems, followed by parameterizing each component. The number of components and the level of fidelity were increased through the iterative process. Each vehicle component was implemented through a pre-defined component block in AMESim that contained a verified framework of equations and calculations for modeling the behavior of the component. The component blocks were parameterized with data from industry sources, reports, and data sheets. The vehicle controller was the last step in the development process. Each discrete vehicle architecture required development of a unique vehicle model and controller. Refer to the Appendix for details on the MATLAB/Simulink/AMESim simulation process.

The modeling framework was structured to allow easy component changes and updates. A small library of component data was developed in MATLAB script files. By simply changing a number or text string, the master script will "upload" the corresponding data to the vehicle model. The post-processing code conveniently displays all parameters and calculations of interest. With this model framework, any AMESim or Simulink library component can be quickly implemented or a custom component can be built and integrated. With this process, Michigan Tech has developed a powerful and fully customizable test bed to evaluate different components and different vehicle architectures.

3. RFP Part 1 – Power and Energy Requirements at the Wheels

The first step in developing a powertrain design from a set of design targets is to characterize the approximate size, shape, and mass of the vehicle that the powertrain must propel. These parameters determine the resistive forces that act upon the vehicle as it moves down the road. The vehicle design engineer must then estimate the powertrain requirements to overcome these resistive forces, while concurrently meeting the performance targets. The size, shape, and mass are termed the vehicle glider characteristics and have been provided in the RFP as shown in Table 3.1. The size and approximate shape are accounted for in the drag coefficient and frontal area parameter.

Table 3.1: Vehicle Glider Characteristics

Vehicle equivalent test weight	1,500 kg
GVWR	2,000 kg
Road load coefficients for equivalent test weight	F_0 = 120 N F_1 = 1.46 N/(m/s) F_2 = 0.42 N/(m/s)^2
Drag × Frontal Area, C_dA_f	0.75 m^2
Coefficient of rolling resistance, C_{rr}	0.009

Powertrain force and torque requirements were determined with a force balance using the vehicle glider characteristics and performance targets. The vehicle free body diagram shown in Figure 3 shows the forces that act upon the vehicle. The resistance forces include the aerodynamic drag, rolling resistance and gravitational resistance. The force required to propel the vehicle at a given acceleration rate and velocity was determined by the standard road load equation. The road load equation calculates the required tractive force F_{Tr} from rolling resistance, aerodynamic drag, inertial force, and gravitational resistance as shown in equation 3.1.

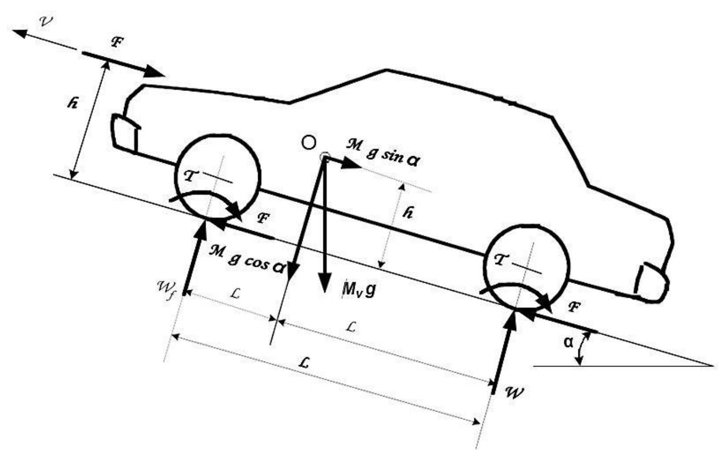


Figure 3.1: Free Body Diagram of Vehicle Forces [3]

The force required to propel the vehicle at a given acceleration rate and velocity can be determined by summing the forces shown in the free body diagram and breaking down each force to include the specific parameters that affect it. The resulting equation is called the road load equation and is a standard in vehicle simulation and design. The required tractive force, F_{tr}, is calculated from rolling resistance, aerodynamic drag, inertial force, and hill climbing resistance with the road load equation shown here:

$$F_{tr} = MgC_{rr}\cos(\alpha) + {}^{1}\!/_{2}\,\rho C_d A_f (v - v_w)^2 + F_{inertial} + Mgsin(\alpha) \qquad (4)[3]$$

where M is the mass, g is the acceleration due to gravity, C_{rr} is the coefficient of rolling resistance, ρ is the air density, C_d is the drag coefficient A_f is the vehicle frontal area, v is the vehicle velocity, v_w is the wind velocity, and α is the road angle [3]. The inertial force is defined as

$$F_{inertial} = M_i a \tag{5}[3]$$

where M_i is the inertial mass and a is the acceleration ($^{dv}/_{dt}$ from the drive cycle velocity). The inertial force includes the vehicle linear inertia as well as the rotational inertia of the wheels. The tractive force was calculated for each time interval over a given drive cycle. Since the velocity and tractive force are known, the required power can be determined. These calculations were carried out in MATLAB. Tractive force is directly related to powertrain torque as a function of the tire radius and gear ratios in the transmission and final drive.

Table 3.2: Results at the Wheels for Drive Cycles

Metric	UDDS	HwFET	US06
Positive propulsion energy required at the wheels (Wh/km)	121.93	114.81	193.17
Negative (braking) energy required at the wheels (Wh/km)	48.15	11.12	49.35
Net (road load) energy required at the wheels (Wh/km)	73.77	103.69	143.81
Average positive propulsion power at the wheels (kW)	3.83	8.9	14.91
Peak power output at the wheels (kW)	33.66	28.06	85.93
Peak tractive force at the wheels (kN)	2.41	2.35	5.91
Percent idle time (%)	17.74	0.65	6.67

Accounting for the mechanical efficiency and the gear ratios of the powertrain allows for the determination of the actual powertrain requirements, including the torque that the engine must generate at any given vehicle speed.

In addition to the standard drive cycles, there are also numerous special cases used to determine maximum powertrain requirements and capabilities. Two of the most common are the 0-60mph acceleration and the highway gradeability tests. The powertrain power power requirements from these tests are shown in Table 3.3.

Table 3.3 : Average Tractive Power Requirements

Metric	Result
Average power required to meet minimum acceleration time (kW)	54
Average power required to climb 3.5% grade at 60 mph at GVWR (kW)	32

The average acceleration required to reach 60 mph within 11 seconds was calculated with the kinematic equation:

$$V_f = V_i + at \tag{6}$$

Where V_f is the final velocity (m/s), V_i is the initial velocity (m/s), a is the acceleration (m/s^2), and t is the time (s). The same equation was used with the constant acceleration to determine the velocity at each time step. The road load power was calculated for each time step with the corresponding velocity and then averaged to obtain the result. To calculate the gradeability, the 3.5% grade was converted to a road angle value of 2.0 degrees using Eq. 3. A vehicle mass corresponding to GVWR was used to calculate the road load power to maintain 60 mph at a 2.0 degree incline.

4. **Vehicle Component Sizing**

"Amir Rezaei contributed to the writing of this section and all subsections within this section"

Initial component sizing was done with some estimation and calculations to determine the approximate specifications to satisfy the design targets.

4.1. Acceleration

The acceleration of the vehicle on a flat road can be calculated from Newton's second law and the road load equation, resulting in equation 6.

$$ma = F_{tr} - mgC_{rr} - \frac{1}{2}\rho C_D A_f v^2 - F_{inertial} \qquad (7)[3]$$

The inertial force was determined by accounting for the inertias of the wheel, transmission, engine and/or electric motor as well as the transmission and final drive gear ratios. The acceleration time was then calculated with equation 11.

To calculate $F_{inertial}$ the following approach has been used, refer to Figure 4.1 and equation 8.

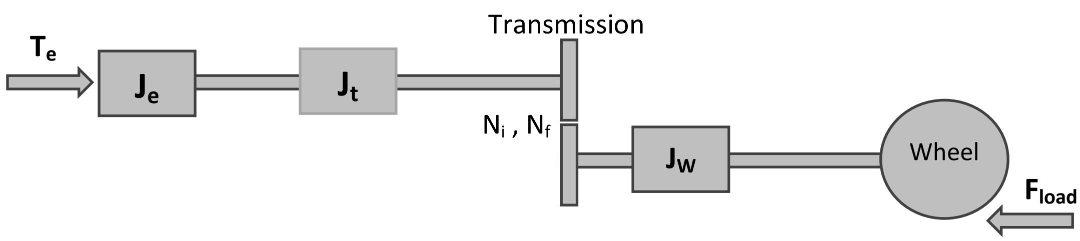


Figure 4.1: Powertrain and Drivetrain Inertia

$$T_t = T_e - J_e\alpha_e \ , \ T_d = \varepsilon_t N_i N_f (T_t - J_t\alpha_e) \ , \ T_w = T_d - J_w\alpha_w \ , \ \alpha_w = \frac{a}{r} = \frac{\alpha_e}{N_i N_f} \qquad (8)[3]$$

$$\Rightarrow \ F_{tractve} = \frac{T_e N_i N_f \varepsilon_t}{r} - \left((J_e + J_t)N^2 + J_w\right)\frac{a}{r^2}$$

Where ε_t is the total efficiency of the transmission, N_i is the i'th gear ratio, N_f is the final drive ratio, a is longitudinal acceleration, r is wheel radius and J_e ,J_t and J_w are electric motor, transmission and wheel rotational inertias respectively. And as a result the equation can be expressed as:

$$\left(m + \frac{(J_e + J_t){N_i}^2{N_f}^2 + J_w}{r^2}\right)a = m_e a = \frac{T_e N_i N_f \varepsilon_t}{r} - mgC_{rr} - \frac{1}{2}\rho C_D A_f v^2 \qquad (9)[3]$$

Using the above equation, the acceleration time can be obtained from:

$$t_a = \int_0^{V_{60}} \frac{m_e}{\frac{T_e N_i N_f \varepsilon_t}{r} - mgC_{rr} - \frac{1}{2}\rho C_D A_f v^2} dv \qquad (10)[3]$$

The typical torque profile of an electric motor operating as part of a powertrain in a vehicle is shown in the following figure:

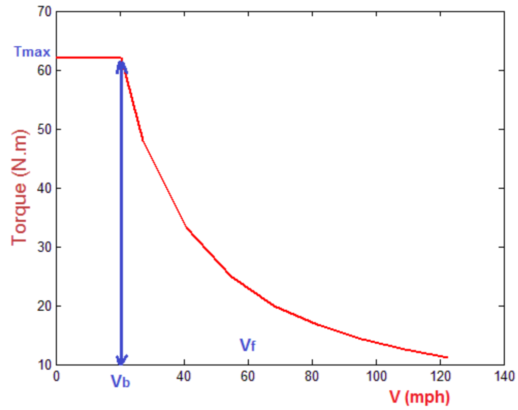


Figure 4.2: Typical Torque Profile of an Electric Motor (Motor Torque vs Vehicle Speed with single constant ratio)

Integrating the motor torque profile in a piecewise fashion, the above equation becomes:

$$t_a = \int_0^{V_b} \frac{m_e}{\frac{P_{e,max}\varepsilon_t}{v_b\, r} - mgC_{rr} - \frac{1}{2}\rho C_D A_f v^2} dv + \int_{V_b}^{V_f} \frac{m_e}{\frac{P_{e,max}\varepsilon_t}{v\, r} - mgC_{rr} - \frac{1}{2}\rho C_D A_f v^2} dv \qquad (11)[3]$$

By ignoring the road load, the acceleration time for a no load vehicle will be:

$$t_a = \frac{m_e}{2P_{e,max}\varepsilon_t}\left(v_f^2 + v_b^2\right) \quad \Rightarrow \quad P_{e,max} = \frac{m_e}{2\varepsilon_t t_a}\left(v_f^2 + v_b^2\right) \qquad (12)\ [3]$$

Now the average of road load power can be calculated from:

$$\bar{P}_{load} = \frac{1}{t_a}\int_0^{t_a} v.F_{load}dt = \frac{1}{t_a}\int_0^{t_a} mgC_{rr}v - \frac{1}{2}\rho C_D A_f v^3 dt \qquad (13)\ [3]$$

With a constant average power increment $\frac{mv^2}{2t} = \frac{mv_f^2}{2t_a}$ the solution of the above integral is:

$$\bar{P}_{load} = \frac{2}{3}mgC_{rr}v_f + \frac{1}{5}\rho C_D A_f v_f^3 \qquad (14)\ [3]$$

And finally an estimation of required power to achieve an acceleration time of t_a is:

$$P_{e,max} = \frac{m_e}{2\varepsilon_t t_a}\left(v_f^2 + v_b^2\right) + \frac{2}{3}mgC_{rr}v_f + \frac{1}{5}\rho C_D A_f v_f^3 \qquad (15)[3]$$

4.2. Top Speed

From equation 15 the maximum speed happens when:

$$\frac{T_e N_{i,min} N_f \varepsilon_t}{r} = mgC_{rr} + \frac{1}{2}\rho C_D A_f v_{max}^2 \quad \Rightarrow \qquad (16)[3]$$

$$P_{e,max} = \frac{1}{\varepsilon_t}\left(mgC_{rr}v_{max} + \frac{1}{2}\rho C_D A_f v_{max}^3\right)$$

If $P_{e,max}$ is very high, then maximum speed of the electric motor and transmission ratio restricts the maximum achievable speed:

$$v_{max} = \frac{r\ \omega_{max}}{N_{i,min}N_f} \qquad (17)[3]$$

Where r is wheel radius, ω_{max} is maximum electric motor speed in $\frac{rad}{s}$ and $N_{i,min}$ is the lowest transmission ratio.

4.3. Gradeability

For a vehicle with constant speed on a road with angle of θ, the equation (9) is:

$$\frac{T_e N_i N_f \varepsilon_t}{r} = mg\sin\theta + mgC_{rr}\cos\theta + \frac{1}{2}\rho C_D A_f v^2 \qquad (18)[3]$$

For small θ which is reasonable at the speed of $v_f = 60\ mph$:

$$\frac{T_e N_i N_f \varepsilon_t}{r} = \frac{\varepsilon_t P_{e,max}}{v_f} \quad \Rightarrow \quad \theta \approx \sin\theta = \frac{\frac{\varepsilon_t P_{e,max}}{v_f} - \frac{1}{2}\rho C_D A_f v_f^2}{mg} - C_{rr} \qquad (19)[3]$$

A high gradeability value is desirable for better performance and driveability.

5. RFP Part 2 – Conventional Vehicle Performance and Fuel Consumption

The conventional vehicle model includes a combustion engine coupled to a five speed manual transmission that provides power to the front axle of the vehicle. The powertrain configuration is shown in Figure 5.1. Standard engine maps and performance curves predict the engine performance and efficiency. A maximum torque versus speed map defines the maximum power of the engine, while a fuel consumption map indexed by speed and torque characterizes the efficiency of the engine. Engine friction is modeled by a friction mean effective pressure (FMEP) curve. The transmission model accounts for torque and speed changes according to the defined gear ratios and the efficiency of each gear. An inertia parameter in each model accounts for the rotational mass of the engine and transmission. The vehicle dynamics block models the vehicle body and chassis that interacts with the powertrain model. This block calculates the longitudinal dynamics according to the weight distribution between the front and rear axle, allowing calculation resistance forces, traction limits, vehicle velocity and acceleration. This vehicle system model includes all of the energy and tractive force sources and all of the primary resistive forces and energy sinks.

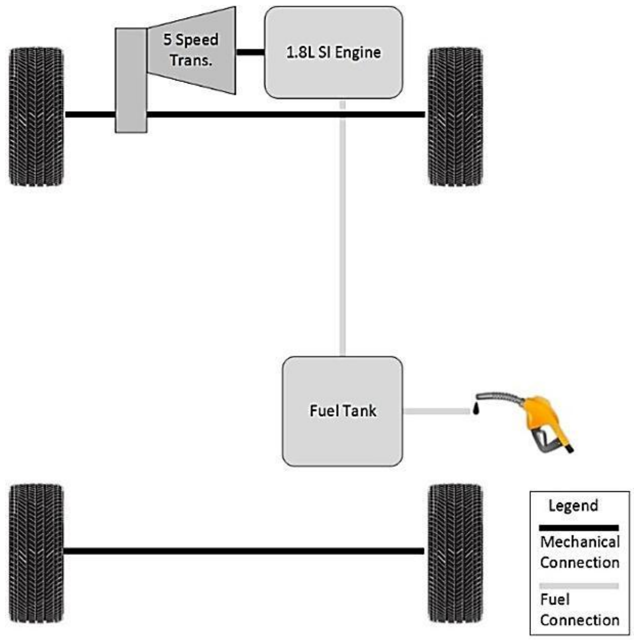


Figure 5.1: Conventional Vehicle Powertrain Configuration and Power Flow

5.1. Assumptions and Limitations

The vehicle system model was developed for quasi-static simulation only. It does not account for transient behavior in the engine, transmission, driveline, and chassis. When modeling, varying levels of fidelity are used to represent different phenomena – overall vehicle load and powertrain performance modeling requires only relatively low fidelity and accounts for average component behavior while ignoring much of the transient behavior that might actually occur in each component. Examples of component transient behavior include engine torque spikes, engine/motor/battery response times, shaft flex, and vehicle inertial dynamics during acceleration, braking, and turning. The effects of temperature in the engine and transmission are also ignored in the model. It is understood that cold temperatures will drive friction and other parasitic losses higher, requiring additional power to overcome the losses. Additionally, cold start correction factors in the engine controls would be necessary to handle emissions at start up, as well as drivability issues. Accounting for these issues and others is not necessary for long cycle energy consumption and acceleration estimation, but must be taken into account during actual vehicle development.

5.2. Baseline Engine Model

A 99 kW engine was selected which closely matches the baseline engine guidelines in the RFP. Typical transmission gearing for the 99 kW/1500 kg power-to-weight ratio was initially determined from that used in similar vehicles and then adjusted to meet the vehicle performance design targets. When adjusting the transmission gearing, the vehicle acceleration was tuned by changing the ratios of the low gears. Highway fuel economy performance was maintained by preserving the high gear values. The tank capacity was set at 45 liters (11.88 gallons), which is typical of a small- to mid-size sedan.

Table 5.1: Results and Powertrain Sizing

Test mass, kg	1500
Top speed, kph (mph)	181.5 (112.8)
Acceleration 0-60 mph, s	9.96
Highway gradeability at 60 mph at test mass, %	6
Powertrain configuration	Conventional SI Engine
Powertrain sizing:	
Engine peak power, kW	99
Transmission, gearing	3.45,1.92,1.28,0.88,0.67
Final Drive, ratio	5.13

Table 5.2: Drive Cycle Energy Consumption Results for Baseline Engine Powertrain

Test Mass (kg): 1,500 Engine Size (kW): 99	Unit	UDDS	HwFET	Combined	US06
Net tractive energy	Wh/km	127	113	121	187
Fuel energy	Wh/km	577	470	529	690
Battery energy	DC Wh/km	-	-	-	-
AC grid energy	AC Wh/km	-	-	-	-
GHG WTW	g CO_2 eq/km	180	147	165	216
Range	Km	700	860	764	586

The combined unadjusted energy consumption value for the baseline vehicle is displayed as fuel energy consumption in the combined category in Table 6 and has a value of 529.0 Wh/km which does not meet the target of 370 Wh/km (56.7 mpg). A 2010 Pontiac Vibe was chosen for a production vehicle comparison from the EPA test car list data which has vehicle type, rated horsepower, transmission type, number of gears, equivalent test weight, axle ratio, N/V ratio, and test fuel type specifications comparable to the baseline vehicle [4]. The HWFET fuel economy of the Vibe is 44.8 mpg which matches will with the 45 mpg HWFET economy of the baseline vehicle. The energy balance shown in Figure 5.2 displays the various losses in the powertrain and the balance of the sources and losses indicates that the model is accounting for all energy flowing through the system.

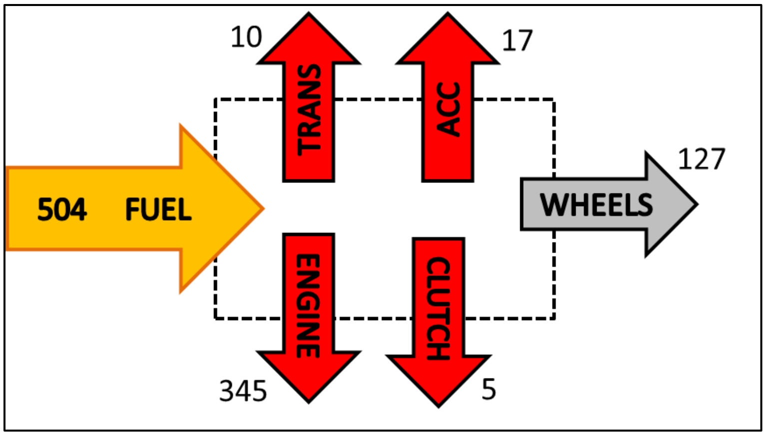


Figure 5.2: Baseline Vehicle Energy Balance (Wh/km)

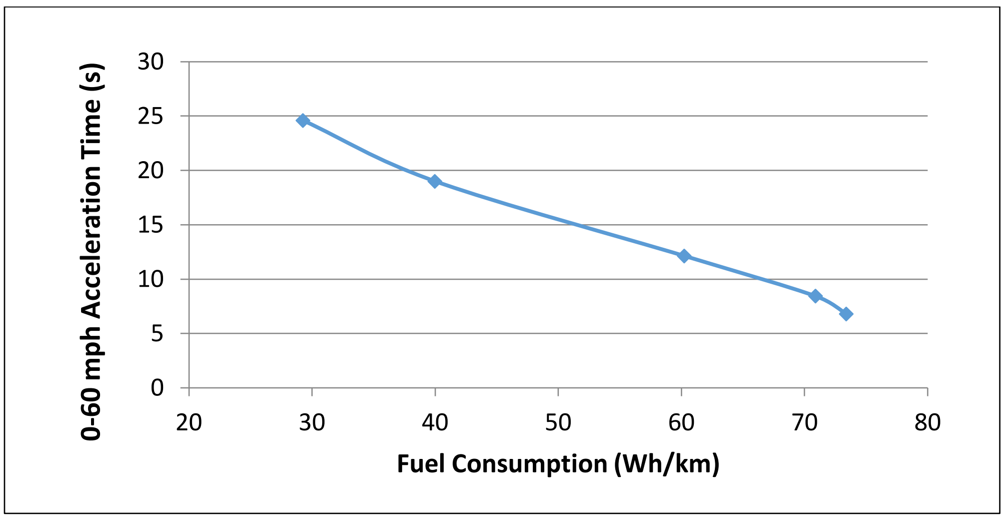


Figure 5.3: Acceleraton and Fuel Consumption Trends

As engine size and power are varied the fuel consumption and acceleration vary inversely from one another, with acceleration increasing as engine power increases. This trend is displayed in Figure 5.3 where engine power is increasing as you move down the y-axis and right on the x-axis. This demonstrates one of the design tradeoffs between power and fuel consumption. As engine power is increased the overall fuel consumption also increases.

5.3. Downsized Engine Model

Choosing a smaller engine, known as engine downsizing, is a common strategy used to improve fuel economy and is often accompanied by the addition of a turbocharger to maintain a power output similar to the larger predecessor. There are many reasons why this works, including lower engine mass, reduced friction, and reduced fuel flow requirements due to the smaller displacement. For this modeling exercise, the engine was downsized to the point where the vehicle was just able to meet the minimum acceleration target of 11 seconds. The engine was downsized by applying a scale factor to the engine torque data while applying an inverse scale factor to the brake specific fuel consumption (BSFC) data. If the engine mechanical design and compression ratio are held constant, then the BSFC will be similar across all engine sizes [5]. The downsized engine had a maximum power of 89.6 kW, with an approximate displacement of 1.63 L. The resulting acceleration time was 10.96 seconds, just under the 11 second limit.

Table 5.3: Drive Cycle Energy Consumption Results for Downsized Engine Powertrain

Test Mass (kg): 1,500 Engine Size (kW): 89.6	Unit	UDDS	HwFET	Combined	US06
Net tractive energy	Wh/km	126.9	113.3	120.8	186.5
Fuel energy	Wh/km	550.1	435.2	498.4	660.1
Battery energy	DC Wh/km	--	--	--	--
AC grid energy	AC Wh/km	--	--	--	--
GHG WTW	g CO_2 eq/km	174	140	182	208
Range	km	734	929	811	612

The combined unadjusted energy consumption value for the downsized engine vehicle is displayed as fuel energy consumption in the combined category in Table 5.3 and has a value of 498 Wh/km which is less than the baseline engine but still does not meet the target of 370 Wh/km.

5.4. Diesel Engine Model

Another option for reducing vehicle energy consumption is to replace the spark-ignition (SI) gasoline engine with a compression ignition (CI) Diesel engine. This replacement is not as straightforward as the downsized engine since CI engines are typically heavier than SI engines and require additional after-treatment devices to meet emissions regulations. However, CI Diesel engines offer an advantage over SI engines due to the greater energy density of Diesel fuel compared to gasoline and due to the lack of a throttle which eliminates some pumping losses at low loads. Diesel fuel properties also allow for greater compression ratios which give CI engines a higher thermal efficiency than SI engines (35-45% for CI compared to 20-30% for SI) [22]. For this exercise, the Diesel engine was scaled to adjust the size and power of the engine to match the acceleration time to the baseline engine acceleration time of 9.96 seconds.

Table 5.4: Results and Powertrain Sizing for Diesel Engine Powertrain

Test mass, kg	1500
Top speed, kph (mph)	159.3 (99.9)
Acceleration 0-60 mph, s	10.0
Highway gradeability at 60 mph at test mass, %	7
Powertrain configuration	Conventional SI Engine
Powertrain sizing:	
Engine peak power, kW	90
Transmission, gearing	3.45,1.92,1.28,0.88,0.67
Final drive ratio	5.13

The scaled engine achieved an acceleration of 10.0 seconds, 0.04 seconds off of the baseline.

Table 5.5: Drive Cycle Energy Consumption Results for Diesel Engine Powertrain

Test Mass (kg): 1,500 Engine Size (kW): 90	Unit	UDDS	HwFET	Combined	US06
Net tractive energy	Wh/km	126.8	113.4	120.7	187.8
Fuel energy	Wh/km	504	426	480	654
Battery energy	DC Wh/km	-	-	-	-
AC grid energy	AC Wh/km	-	-	-	-
GHG WTW	g CO_2 eq/km	138	116	127	178
Range	km	807	954	847	622

The combined unadjusted energy consumption value for the diesel engine vehicle is displayed as fuel energy consumption in the combined category in Table 5.5 and has a value of 551.2 Wh/km which is still higher than the target of 370 Wh/km (56.7 mpg). The fuel consumption and CO_2 emissions results for each engine option are compared in Table 5.6.

Table 5.6: Conventional Vehicle Engine Comparison

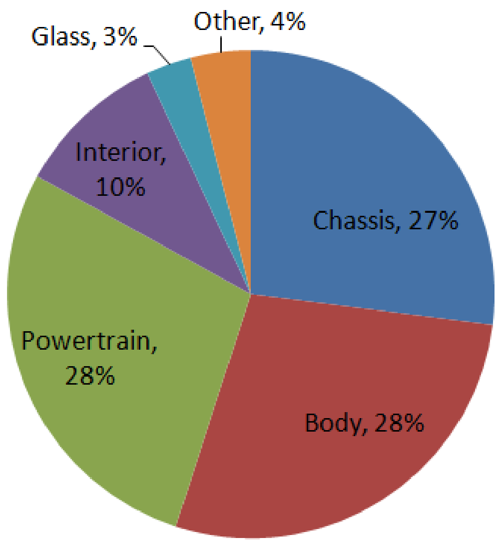
Test Mass (kg): 1,500 Engine Size (kW):	Unit	Baseline Gasoline	Downsized Gasoline	Diesel
Net tractive energy	Wh/km	120.9	120.8	120.7
Fuel energy	Wh/km	529.0	498.4	480
Battery energy	DC Wh/km	-	-	-
AC grid energy	AC Wh/km	-	-	-
GHG WTW	g CO_2 eq/km	165	182	127
Range	km	764	811	847

The Diesel engine powertrain delivered the lowest energy consumption and emissions results. This was attributed to the higher efficiency of the Diesel engine compared to the SI engine and the renewable energy content of the B20 Diesel fuel.

6. RFP Part 3 – Battery Electric Vehicle Performance and Energy Consumption

Powertrain electrification is one of the most commonly accepted methods for reducing vehicle energy consumption. Implementing full electrification such as with a battery electric vehicle (BEV) is actually simpler in terms of powertrain design and control than partial electrification as with a hybrid electric vehicle. The BEV powertrain consists of a simple gearbox and a motor powered by a high voltage battery and inverter. This modeling exercise explores BEV design parameters and their effect on performance, range, and energy consumption.

The BEV mass must be estimated before accurate drive cycle results can be obtained. This is an iterative process because the battery energy required to meet the range target depends on the vehicle mass, which is determined in part by the battery mass which is directly proportional to its energy capacity. The vehicle mass estimation starts by breaking the vehicle down into individual system and component masses. The weight breakdown of a conventional vehicle by system is shown in Figure 6.1 [12]:



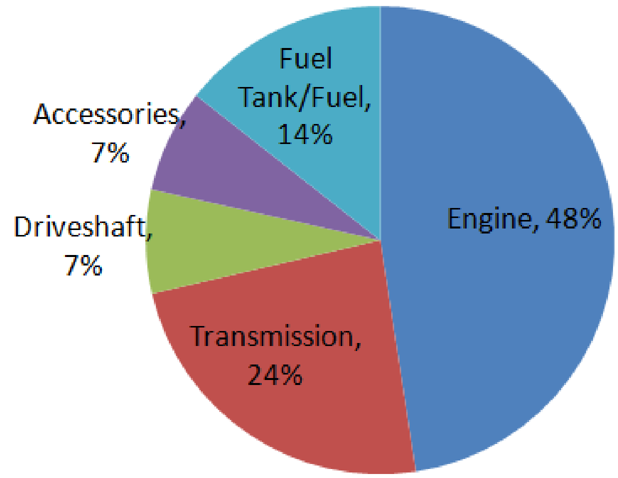


Figure 6.1: a.) Vehicle Mass Breakdown **b.) Powertrain Mass Breakdown**

A conventional vehicle test mass of 1500 kg includes 300 pounds (136.08 kg) added vehicle load according to the EPA standard [15], making the actual vehicle mass 1363.92 kg. The body of a BEV was found to be typically lighter than the conventional vehicle body by about 21% due to the higher aluminum content [12]. This results in a 75 kg reduction in overall mass. The powertrain mass will change according to the design and is further divided as shown in Figure 6.1(b).

Typical component weights were estimated by researching existing components. Table 6.1 shows the mass of each component factored into the total vehicle mass.

Table 6.1 : BEV Mass Estimation

Component	Mass (kg)	Notes
Chassis, interior, glass, and other components	600	Same as base vehicle
Conventional body	382	
EV body mass reduction	-60	Body redesigned for EV
Motor	80	
Transmission	23	
Inverter	20	
No intake and exhaust	-20	
Accessories	27	
Battery	100 Wh/kg	68 kWhr =680 kg
Driveshafts/Axles	27	
Passenger mass	80*4	

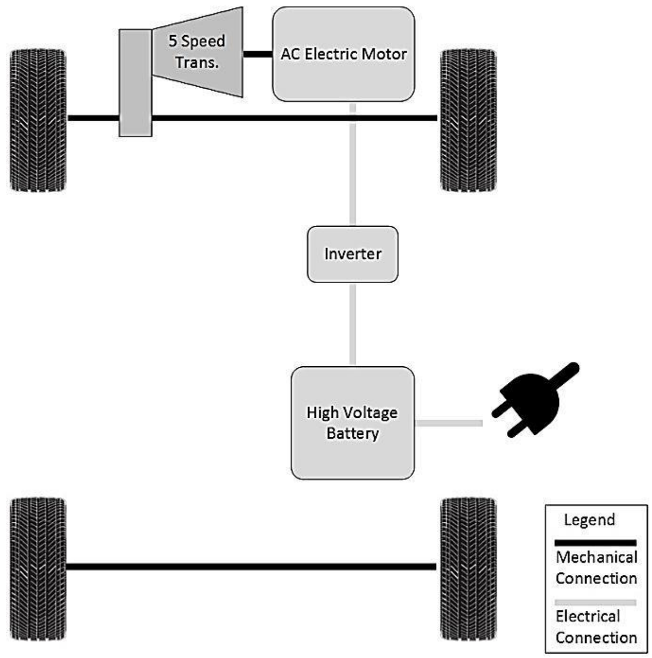


Figure 6.2: Battery Electric Vehicle Powertrain Diagram and Power Flow

The total vehicle mass is 1,727 kg unloaded and 2,047 kg with four passengers. Four passengers push the vehicle just over the 2000 kg GVWR design limit, even without cargo.

The BEV model includes an 85 kW electric motor coupled to a single speed manual transmission that provides power to the front axle of the vehicle, as shown in Figure 6.2. The model is designed for quasi-static simulation of the vehicle system and does not model transient behavior in the motor, transmission, driveline, or chassis. The motor performance and efficiency is predicted through the use of standard motor maps and performance curves. A maximum torque versus speed map defines the maximum power of the engine while an efficiency map indexed by speed and torque defines the power loss. The transmission model accounts for torque and speed changes according to the defined gear ratios and the efficiency of each gear. An inertia parameter in each model accounts for the rotational mass of the motor and transmission. The vehicle dynamics block models the vehicle body and chassis that interacts with the powertrain model. This block calculates the longitudinal dynamics according to the weight distribution between the front and rear axle and also calculates overall vehicle parameters such as the resistance forces, traction limits, vehicle velocity and acceleration. This vehicle system model includes all of the energy and tractive force sources and all of the primary resistive forces and energy sinks.

Table 6.2: Results and Powertrain Sizing for Electric Vehicle

Test mass, kg	1873
Top speed, kph (mph)	160
Acceleration 0-60 mph, s	9.2
Highway gradeability at 60 mph at test mass, %	7
Powertrain configuration	Battery electric
Powertrain sizing:	
Motor peak power, kW	85
Transmission, gearing	9
Battery energy capacity, kWh	68
Battery peak power, kW	95
Battery mass, kg	700

6.1. Assumptions and Limitations

The level of component detail in the EV model is similar to that in the conventional vehicle model. The effects of temperature in the battery, motor, and transmission are not included in the model and the performance is assumed constant across the range of operating temperatures. Taking temperature affects into account alters the performance of the battery and the electric motor more than the transmission. Low temperatures will increase battery internal resistance, reduce capacity and reduce motor efficiency. The torsional flex of the shafts, and other detailed material mechanics are not modeled. These are not necessary for long cycle energy consumption and acceleration estimation but for actual vehicle development these factors must be taken into account.

6.2. Modeling Results

Table 6.3: Modeling Results and Powertrain Sizing for Electric Vehicle

Test mass, kg	1737+136=1873
Top speed, kph (mph)	160
Acceleration 0-60 mph, s	9.2
Highway gradeability at 60 mph at test mass, %	7
Powertrain configuration	Battery electric
Powertrain sizing:	
Motor peak power, kW	85
Transmission, gearing	9
Battery energy capacity, kWh	68
Battery peak power, kW	95
Battery mass, kg	700

The electric motor in the BEV reduces energy consumption due to its higher operating efficiency and can offer improved acceleration performance with its greater low speed torque even though the peak power is lower than that of a conventional vehicle.

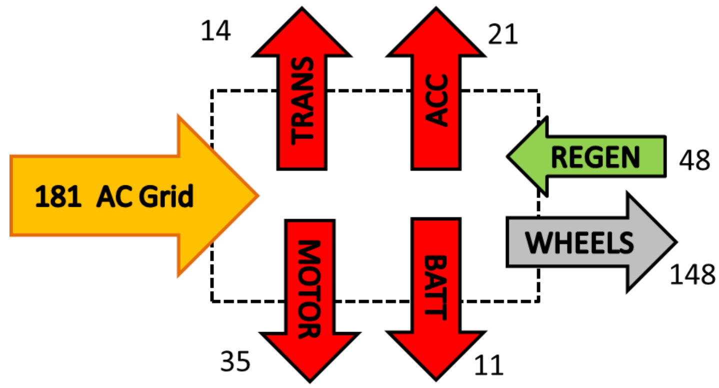


Figure 6.3: Electric Vehicle Energy Balance

6.3. Electric Vehicle Sizing

"Amir Rezaei contributed to the writing of this section and all subsections within this section"

6.3.1. Tractive Electric Motor Sizing

For passenger vehicles, the acceleration requirement dominates the sizing of the electric motor compared to the gradeability and top speed requirements. Assuming reasonable values for the inertia of rotational components, the value of effective vehicle mass during acceleration will be:

$$\begin{array}{ll} J_e = 0.054\ kg.m^2 & N_i N_f = 10 \\ J_t = 0.003\ kg.m^2 & r = 0.31\ m \\ J_w = 3.84\ kg.m^2 & m = 1873\ kg \end{array} \quad \Rightarrow \quad m_e = \mathrm{m} + \frac{(J_e + J_t){N_i}^2{N_f}^2 + J_w}{r^2} = 1972\ kg \qquad (14)$$

By considering an acceleration time of 10 seconds, the required tractive power $P_{e,max}$ for different values of electric motor base speed can be calculated from the equation 10. A motor power of 85 kW is shown to be sufficient. Since $P_{e,max}$, is known from equations 10 and 11, the values for top speed and gradeability can be checked to determine if they satisfy the minimum performance requirements. Since the tractive power satisfies the acceleration requirement, the gradeability and top speed requirements are also satisfied. An electric motor with a base speed of 2500 rpm was chosen since it has a reasonable balance between torque and power across its operating range.6.4 (A) represents calculated values for the tractive electric motor and expected vehicle performance. Since the tractive power has been calculated based on acceleration requirement, it can be seen that the gradeability and top speed requirements are also satisfied. An electric motor with a base speed of 2500 rpm was chosen since it has a reasonable balance between torque and power across its operating range, which is well matched with typical vehicle requirements. 6.4 (B) shows an estimation of the torque and power profile of the desired tractive electric motor.

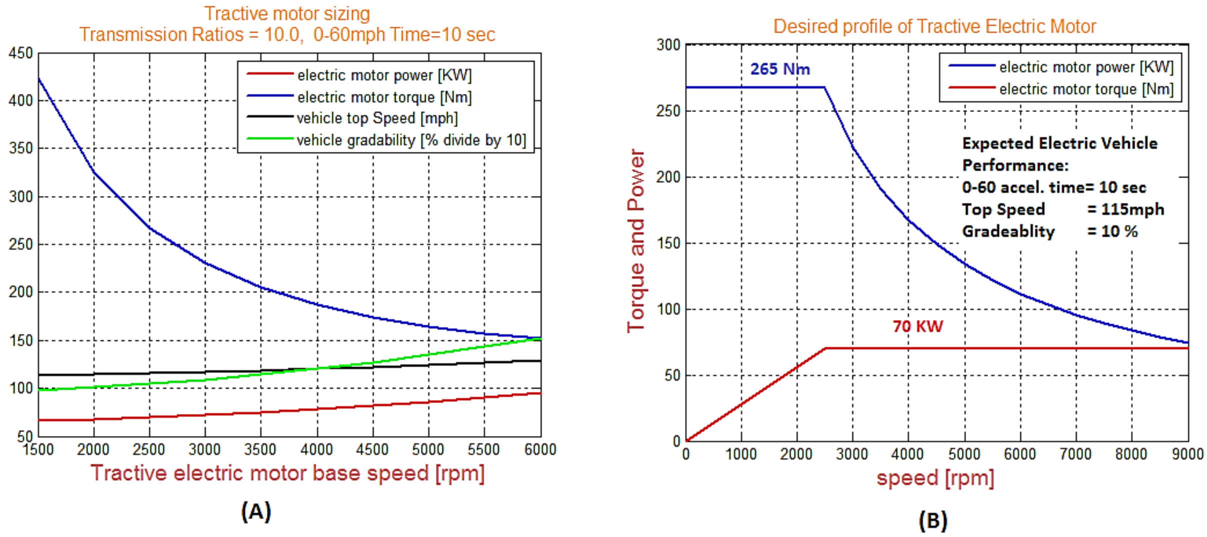


Figure 6.4: (A) Tractive motor power and torque vs. base speed and the electric vehicle performance (B) An estimation of the desired torque and power of the tractive electric motor

6.3.2. Transmission Sizing

From equations 7 and 8 it can be seen that a high gear ratio, increases effective mass during acceleration and consequently demands greater power. On the other hand, a low ratio requires an electric motor with very high torque. From previous part, a transmission ratio of 10 was found to be acceptable for a good acceleration with reasonable values of power and torque.

6.3.3. Battery Sizing

For the target electric vehicle, the battery should be able to provide the maximum power needed by the tractive electric motor. Assuming an average efficiency of $\varepsilon_{em} = 0.9$ for the electric motor, then the desired battery power is:

$$P_{battery,max} = \frac{P_{e,max}}{\varepsilon_{em}} = 77\ KW \tag{23}$$

In addition to the battery power requirement, the desired battery energy is also important to meet the range requirement of 320 km. The average energy at the wheels for was determined by:

$$E_{wheel,net} = \left(E_{wheel,propulsion} + \varepsilon_{regen} E_{wheel,brake}\right) = 97.1 \frac{Wh}{km} \tag{16}$$

Where $E_{wheel,propulsion}$ is combined positive propulsion energy at the wheels, $E_{wheel,brake}$ is combined negative energy at the wheels and ε_{regen}=0.7 is an estimation of the portion of negative energy that can be absorbed by regenerative braking. Then the total battery energy was calculated:

$$E_{bat,total} = \frac{320}{d}\frac{E_{motor}}{\varepsilon_{em}} = \frac{320}{d}\frac{E_{wheel,net}}{\varepsilon_t \varepsilon_{em}} \approx 50\ KWh \tag{17}$$

where d is the combined distance traveled for UDDS and HWFET drive cycles. It should be mentioned that in above calculations, the lower limit on the battery state of charge has been ignored.

However, since the EV was sized for the target acceleration and range, the battery energy should be evaluated again to make sure EV can also meet the target grade ability on highway for 20 minutes:

$$E_{bat} \geq \int_0^{20\times 60} \frac{P_{em}}{\varepsilon_{em}}\, dt = \int_0^{1200} \frac{v}{\varepsilon_{em}} \left(mg\sin\theta + \; mgC_{rr}\cos\theta + \frac{\rho C_D A_f v^2}{2} \right) dt \approx 12\ KWh \qquad (26)$$

That is less than the sized battery energy for EV and it can be concluded that the sized EV also will meet all target criteria.

6.3.4. Combined Energy Consumption

Assuming efficiency of $\varepsilon_{charger} = 0.8$ and $\varepsilon_{battery} = 0.8$ for the charger and battery respectively, the average expected AC grid energy on combined UDDS and HWFET drive cycles is expected to be:

$$E_{AC\ grid,combined} = \frac{E_{bat,total}}{320\ \varepsilon_{charger}\ \varepsilon_{battery}} = 145 \frac{Wh}{km} \qquad (27)$$

That satisfies the energy consumption requirement of the target vehicle.

Table 6.4: Drive Cycle Energy Consumption Results for Electric Vehicle

Test Mass (kg): 1,873 Motor Size (kW): 75	Unit	UDDS	HwFET	Combined	US06
Net tractive energy	Wh/km	148.1	124.4	136	238.6
Fuel energy	Wh/km	--	--	--	--
Battery energy	DC Wh/km	228.5	164.8	195	343.4
AC grid energy	AC Wh/km	180.7	154.7	167	309.6
GHG WTW	g CO_2 eq/km	142	124	133	247
Range	Km	296	412	345	197

The combined unadjusted energy consumption value for the electric vehicle is displayed as AC grid energy consumption in the combined category in Table 6.4 and has a value of 167 Wh/km which is significantly less than the target of 370 Wh/km. This is also significantly less than all of the conventional vehicle models. This is primarily due to the efficiency disparity between engine and electric motor, 35% peak compared to 95% peak. Battery electric vehicles offer low energy consumption and zero vehicle emissions but this is offset by the large, heavy battery back that is required to achieve sufficient range. The AC grid energy consumption is lower than the battery energy consumption due to regenerative braking that recharged the battery rather than AC energy input required.

7. RFP Part 4 – Series Hybrid Electric Vehicle Performance and Energy Consumption

Series HEVs (S-HEV) have been introduced as an innovative solution to address some of disadvantages of BEVs. While BEVs benefit from low energy consumption and zero vehicle emissions, they have not been widely accepted due to limited range, long charging time, and limited infrastructure for charging. Battery capacity can be increased to extend EV range, but this will lead to large increases in battery size and mass which may go beyond the acceptable limits.

Series HEVs have the potential to extend the range without increasing battery size and mass. This is accomplished with the addition of an engine and generator that can make use of energy dense liquid fuels to generate electricity to increase the range, hence the oft used moniker *extended range electric vehicle* or E-REV. The charging time is reduced by implementing a smaller battery since the battery does not need to provide all of the propulsion energy for the full driving cycle. The engine in E-REVs is mechanically separated from the power axle, so it is the electric motor that provides propulsive torque and largely determines the overall performance of the vehicle. The engine's separation from the axle can provide fuel consumption improvements in series HEVs compared to conventional vehicles because the engine can run at its most efficient operating points for the current load without any consideration of speed.

7.1. Sizing the Series Hybrid Electric Vehicle

"Amir Rezaei contributed to the writing of this section and all subsections within this section"

The vehicle's performance parameters are determined by the electric motor since it is the only tractive power source. However, improper sizing of the engine and generator can limit maximum motor performance and adversely affect the vehicle's overall performance. The same electric motor and transmission that was sized for the EV can also be used for series HEV to meet the target vehicle requirements.

Table 7.1 : Series HEV Mass Estimation

Component	Mass (kg)	Notes
Mass of chassis, interior, glass, and other components	600	Same as base vehicle
Conventional body	382	
HEV body mass reduction	-30	
Traction motor	80	
Transmission	25	
Power electronics	30	
Generator	60	
Engine	80	
Accessories	27	
Battery	80 Wh/kg	3.0 kWhr = 37.5 kg
Driveshafts/Axles	27	
Fuel and tank	25	

The total vehicle mass for the initial series model is 1344 kg, leading to a test mass of 1480 kg.

7.1.1. Generator Sizing

Highway driving situations have only a few braking events, so the potential for charging the battery through regenerative braking is much lower than in city driving situations. Therefore, the generator/engine system must be sized based on highway driving conditions in order to satisfy driver's demanded power on the highway even when battery is deeply discharged.

It can be concluded that a typical energy management controller for S-HEV must be designed in a way that in highway driving conditions the engine/generator system can provide enough power at high engine speed. The battery is used as a supplemental power source to help the engine/generator system for high acceleration and occasional top speed situations. Hence, from equation 10 the required generator power can be calculated:

$$P_{gen,highway} = \frac{1}{\varepsilon_t \varepsilon_{em}} \left(mgC_{rr}(0.8v_{max}) + \frac{1}{2}\rho C_D A_f (0.8v_{max})^3 \right) \approx 42\ KW \qquad (21)$$

In urban driving conditions, it is not possible to recover all of braking energy because when driver pushes the brake pedal deeply the mechanical friction braking system must become active. In addition, there are energy losses in delivering braking energy to the battery. Hence, there will be occasions that engine/generator system has to be turned on to propel the vehicle and simultaneously charge the battery. So the other criteria for generator power could be:

$$P_{gen,city} = \frac{1}{\varepsilon_t \varepsilon_{em}} \left(\bar{P}_{propel,city} + \left(1 - \varepsilon_{regen}\right)\bar{P}_{brake,city}\right) \approx 4\ KW \tag{22}$$

Where $\bar{P}_{propel,city}$ and $\bar{P}_{brake,city}$ are the average power required on the wheels during propelling and during braking respectively in UDDS drive cycle and $\left(1 - \varepsilon_{regen}\right)$ is an estimation of the portion of braking energy that cannot be recovered.

Now the generator power can be determined from the following statement:

$$P_{gen,max} = \max\left\{P_{gen,city}\,, P_{gen,highway}\right\} = 42\ KW \tag{23}$$

7.1.2. Engine Sizing

By sizing the generator maximum power, engine power can also be determined:

$$P_{engine,max} = \frac{1}{\varepsilon_{gen}}\, P_{gen,max} = 47\ KW \tag{24}$$

Where $\varepsilon_{gen} = 0.9$ is the average efficiency of the generator.

7.1.3. Battery Sizing

By knowing the maximum power of the tractive motor and maximum power that generator will provide, it is possible determine the lowest threshold of the battery power:

$$P_{battery,max} \geq \frac{P_{e,max}}{\varepsilon_{em}} - P_{gen,max} \qquad \Rightarrow P_{battery,max} \geq 35.7\ KW \tag{25}$$

$$\Rightarrow P_{battery,max} = 40\ KW$$

Sizing the battery energy for an HEV is highly dependent on the Energy Management Strategy (EMS) of controller. In HEVs, only a portion of the battery energy can be used because the battery efficiency is a function of State of Charge (SOC). Hence, for a SOC range of 0 to 1, the EMS tries to keep the battery in its most efficient range of SOC. In addition, maintaining battery life cycle is another reason for defining boundaries on SOC as numerous deep cycles will reduce battery life and efficiency.

From previous calculations it was assumed that only $\varepsilon_{regen} = 0.7$ of braking energy can be recovered in a city drive cycle. So one criterion for sizing the battery energy could be:

$$E_{wheel,net,UDDS} = \left(E_{wheel,propulsion} + \varepsilon_{regen} E_{wheel,brake}\right)_{UDDS} = 89\,\frac{Wh}{km} \tag{26}$$

$$E_{bat,total} = \frac{E_{wheel,net,UDDS}}{\varepsilon_t\ \varepsilon_{em}\ \Delta SOC}\, d \approx 4\ KWh \tag{27}$$

where $E_{wheel,net,UDDS}$ is the net energy required at the wheels to follow the UDDS drive cycle, $d\ is$ the displacement and ΔSOC=0.3 represents the allowed range of SOC in EMS. It should be emphasized again that the sizing of the battery is highly dependent on the EMS and the actual value of $E_{bat,total}$ should be selected after finalizing the EMS and running several simulations over different drive cycles.

7.2. Modeling Results

The sizing of components for the initial S-HEV was limited by the problem statement in the RFP, which stated that a 3.0 kWhr/50 kW battery should be used. The other components were sized according to the calculations shown above.

Table 7.2: Modeling Results and Powertrain Sizing for Initial S-HEV

Test mass, kg	1480
Top speed, kph (mph)	140
Acceleration 0-60 mph, s	27
Highway gradeability at 60 mph at test mass, %	5
Powertrain configuration	Series hybrid electric
Powertrain sizing:	
Engine peak power, kW	100
Generator peak power, kW	60
Motor peak power, kW	45
Transmission, gearing	9.0
Battery energy capacity, kWh	3.0
Battery peak power, kW	50
Battery mass, kg	37.5

Table 7.3: Drive Cycle Energy Consumption Results for Initial Series HEV

Test Mass (kg): 1,480 Engine Size (kW): 25	Unit	UDDS	HwFET	Combined	US06
Net tractive energy	Wh/km	120	116	118	183
Fuel energy	Wh/km	455	506	476	710
Battery energy	DC Wh/km	118	87	101	110
AC grid energy	AC Wh/km	-1.7	1.2	-.3	2.3
GHG WTW	g CO_2 eq/km	142	158	149	222
Range	Km	500	450	480	360

The combined unadjusted energy consumption value for the initial series hybrid electric vehicle is displayed as fuel energy consumption in the combined category in Table 7.3 and has a value of 476 Wh/km which is higher than the target of 370 Wh/km. This is due to the component size restriction which prevented any further reduction in fuel consumption.

The second part of the series exercise imposed no limits on component sizing and an optimal series HEV was designed with component parameters determined to meet each of the design targets. Battery capacity and power limited the traction motor sizing and the overall performance of the initial design.

Table 7.4: Modeling Results and Powertrain Sizing for Optimal S-HEV

Test mass, kg	1567.5
Top speed, kph (mph)	140
Acceleration 0-60 mph, s	9.6
Highway gradeability at 60 mph at test mass, %	5
Powertrain configuration	Opimized S-HEV
Powertrain sizing:	
Engine peak power, kW	25
Generator peak power, kW	50
Motor peak power, kW	75
Transmission, gearing	3.45, 0.67 FDR=5.13
Battery energy capacity, kWh	10
Battery peak power, kW	80
Battery mass, kg	125

Increased battery capacity and power and increased motor power differentiate this design from the initial.

Table 7.5: Drive Cycle Energy Consumption Results for Optimal Series HEV

Test Mass (kg): 1,567 Engine Size (kW): 25	Unit	UDDS	HwFET	Combined	US06
Net tractive energy	Wh/km	128	120	124	189
Fuel energy	Wh/km	294	293	294	397
Battery energy	DC Wh/km	149	90	367	156
AC grid energy	AC Wh/km	74	72	73	128
GHG WTW	g CO_2 eq/km	149	151	150	223
Range	Km	389	380	385	240

The combined unadjusted energy consumption value for the optimal series hybrid electric vehicle is sum of the fuel energy consumption and the AC grid energy in the combined category in Table 7.5 and has a value of 367 which is just meets the target of 370 Wh/km.

8. RFP Part 5 – Innovative Technologies to Reduce Energy Consumption

Considerable resources have been devoted to the development of optimized hybrid electric drive systems to provide motive power to the vehicle and to drive a myriad of accessories. Further improvements to the powertrain typically net only small gains in efficiency, usually at considerable cost. Instead of expending more time and effort for small returns in powertrain efficiency, it is time to pursue the reduction of the overall average load to reduce the energy consumption of the vehicle.

Michigan Tech proposes the following technology:

Electrified Cooling System with Intelligent Control

These potential efficiency improvements can be maximized while keeping the engine operating temperature at its optimal point by designing and controlling the cooling fan and coolant pump individually while optimizing the control strategy for overall system efficiency. Electrifying both the fan and the pump along with replacing the thermostat with an electronic control valve allows precise control of coolant flow and heat rejection while minimizing component power consumption. It has been demonstrated that electrified cooling systems in transit buses can improve fuel economy by 10.5% [9]. This efficiency gain was due to high efficiency electric components and improved cooling system control.

More precise temperature control allows engine operating temperatures to be increased with less risk of exceeding maximum temperature limits of fluid and components. Increased operating temperatures reduce the heat transfer from the combustion chamber which is one of the energy losses that reduce engine efficiency.

Some new engines trends are placing greater demands on the cooling system, particularly downsizing and turbocharging, increasing the need for cooling system improvements to squeeze the maximum fuel economy out of each vehicle. Engine downsizing increases engine loading which generates more heat. Turbochargers are being added to these smaller engines to maintain engine power and vehicle performance. Turbocharging engines generates more heat and places additional demand on cooling systems. In conventional cooling systems, the components need to be oversized compared to a non-turbo engine to handle these additional heat loads. These larger fans and water pumps are constantly turning leading to larger power losses. In an electrified cooling system, these components are sized to the maximum demand, but the ability to operate them at lower speeds during periods of low cooling demand significantly reduces power requirements. Also increasing the cooling system load of turbocharged engines is the charge air cooler which cools the air that comes from the turbocharger before it enters the engine. Along with the turbocharger, this further increases air density, allowing more fuel to be injected and increasing maximum potential power. Both the cooler itself and the additional power that it creates place increased demand on the cooling system. Creation of a separate electrified cooling loop for the charge air cooler would allow for more precise control over the charge air temperature, which would add another controllable parameter for use in overall engine optimization. Having control over the air flow through the turbocharger and the temperature of the air as it enters the engine allows for adjustment of the engine operating point and power output to maximize efficiency while matching the specific power demand.

9. RFP Part 6 – Proposed Powertrain Design to Meet EcoCAR 3 Design Targets

A careful review of the EcoCAR 3 requirements and guidelines provides critical direction and boundaries for the design process. While there are no specific powertrain design limitations, there are several points that provide boundaries for possible solutions. The first is the list of allowable fuels and energy carriers, which includes the gasoline-ethanol fuels E10 and E85, B20 biodiesel, and electricity. The second one is safety. Inherently hazardous technologies that do not have established safety practices should be avoided. And the third one is the physical design space constraint presented by the donor vehicle. Since the vehicle has not yet been specified, it must be possible to package the proposed powertrain design into the smallest vehicle platform – a compact sedan. Considering these three points, potential powertrain solutions become more evident. Through the lessons learned in the modeling exercises, it is clear that the EcoCAR3 targets cannot be met by a vehicle with conventional or downsized powertrain nor a BEV using available battery technology. Therefore, the solution is to use the available energy sources in combination in order to meet the design targets.

The team has focused on hybrid electric powertrains which combine the energy density of liquid fuel, necessary for long range capability, and the high efficiency of electricity needed for low overall energy consumption.

9.1. Potential Powertrain Designs to Meet EcoCAR 3 Targets

Numerous hybrid powertrain configuration have been developed, which can be categorized coarsely categorized as series, parallel, or series-parallel (i.e. power-split). The series and parallel configurations were explained in the introductory section. The series-parallel is a blending of the two other configurations to deliver capabilities of both. Figure 9.1 highlights the primary benefits and drawbacks of each configuration. Examining the configurations reveals that each has at least one significant advantage and drawback.

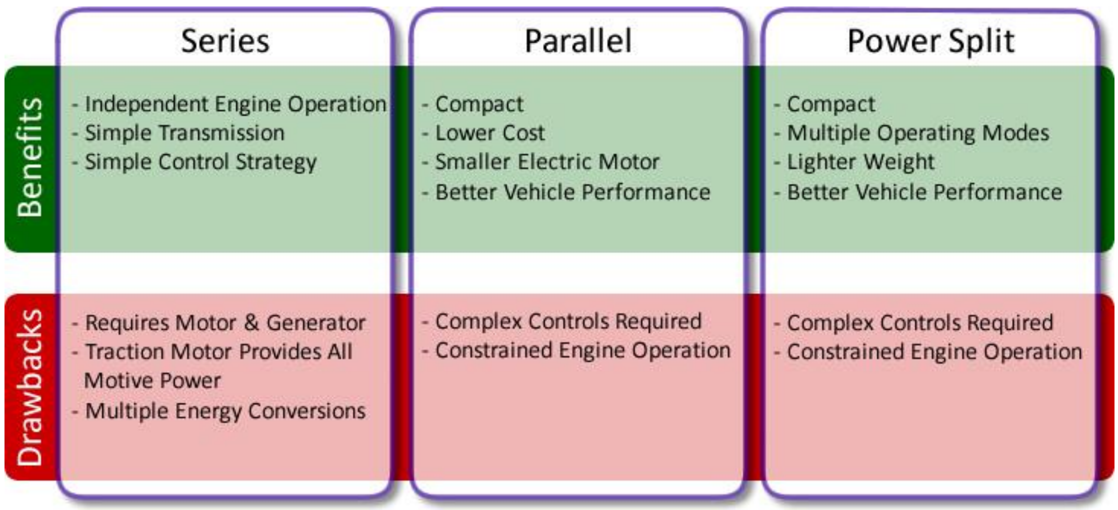


Figure 9.1: A Comparison of Major HEV Architectures

Due to the high component complexity, high controls complexity, and high cost, the power-split architecture was removed from consideration for the final proposed design. The two remaining architectures were modeled for further evaluation. An SI and CI engine option were considered for each architecture The CI series hybrid, CI parallel hybrid and E85 SI parallel hybrid were determined to have the lowest energy consumption. The modeling results are presented in the following tables.

9.2. Simulation Results for Powertrain Designs

The following tables present the simulation results for the three potential powertrain designs.

Table 9.1: Powertrain and Vehicle Characteristics for Diesel Series HEV

Test mass, kg	1680
Top speed, kph (mph)	150
Acceleration 0-60 mph, s	10.7
Highway gradeability at 60 mph at test mass, %	7
Powertrain configuration	Battery electric
Powertrain sizing:	
Engine peak power, kW	45
Generator peak power, kW	70
Motor peak power, kW	65
Transmission, gearing	9.0
Battery energy capacity, kWh	10
Battery peak power, kW	70
Battery mass, kg	125

Table 9.2: Drive Cycle Energy Consumption Results for Diesel Series HEV

Test Mass (kg): 1,680 Engine Size (kW): 99	Unit	UDDS	HwFET	Combined	US06
Net tractive energy	Wh/km	131	115	123	190
Fuel energy	Wh/km	345	440	382	720
Battery energy	DC Wh/km	130	53	79	210
AC grid energy	AC Wh/km	0.8	1.1	0.9	11
GHG WTW	g CO_2 eq/km	95	120	105	197
Range	km	424	333	383	203

The diesel series HEV showed improved energy consumption from the gasoline series HEV discussed in previous sections. However, the energy consumption still did not quite meet the target value.

Table 9.3: Parallel HEV Mass Estimation

Component	Mass (kg)	Notes
Mass of chassis, interior, glass, and other components	600	Unchanged from base vehicle
Body	355	
HEV body mass reduction	-75	
Motor	60	
Transmission	50	
Inverter	20	
Engine	70	This may be lighter
Accessories	27	
Battery	80 Wh/kg	20 kWhr = 250 kg
Driveshafts/Axles	27	
Passenger mass	80*4	

Table 9.4: Powertrain and Vehicle Characteristics for B20 Diesel Parallel HEV

Test mass, kg	1611
Top speed, kph (mph)	144
Acceleration 0-60 mph, s	9.77
Highway gradeability at 60 mph at test mass, %	6
Powertrain configuration	Parallel HEV
Powertrain sizing:	
Engine peak power, kW	30
Motor peak power, kW	40
Transmission, gearing	2.8,1.8,1.28,1.0,0.9
Battery energy capacity, kWh	20
Battery peak power, kW	50
Battery mass, kg	250

Table 9.5: Drive Cycle Energy Consumption Results for B20 Diesel Parallel HEV

Test Mass (kg): 1611 Engine Size (kW): 30	Unit	UDDS	HwFET	Combined	US06
Net tractive energy	Wh/km	127	115	121	178
Fuel energy	Wh/km	340	355	347	506
Battery energy	DC Wh/km	92	54	70	71
AC grid energy	AC Wh/km	0.4	-0.7	0	11
GHG WTW	g CO_2 eq/km	94	97	95	138
Range	km	470	450	460	315

Table 9.6: Powertrain and Vehicle Characteristics for E85 Parellel HEV

Test mass, kg	1590
Top speed, kph (mph)	128
Acceleration 0-60 mph, s	10.1
Highway gradeability at 60 mph at test mass, %	4
Powertrain configuration	Parallel HEV
Powertrain sizing:	
Engine peak power, kW	35
Motor peak power, kW	45
Transmission, gearing	3.15,1.95,1.28,1.0,0.87
Battery energy capacity, kWh	20
Battery peak power, kW	50
Battery mass, kg	250

Table 9.7: Drive Cycle Energy Consumption Results for E85 Parallel HEV

Test Mass (kg): 1,590 Engine Size (kW): 99	Unit	UDDS	HwFET	Combined	US06
Net tractive energy	Wh/km	126	114	120	173
Fuel energy	Wh/km	357	382	367	570
Battery energy	DC Wh/km	97	60	76	82
AC grid energy	AC Wh/km	0.2	0.5	0.3	13
GHG WTW	g CO_2 eq/km	78	86	81	148
Range	km	393	365	380	233

It was determined that the parallel hybrid configuration will best satisfy the design targets. This configuration showed the lowest energy consumption while still meeting all other design targets. The diesel parallel HEV also has a simpler, lighter, and less costly powertrain than the diesel series HEV and a more durable powertrain than the gasoline series HEV.

9.3. Proposed Design for EcoCAR3 Competition

The Michigan Tech Hybrid Electric Vehicle Enterprise team proposes to design and build a plug-in parallel hybrid electric vehicle (PP-HEV) that will meet or exceed each of the design targets for EcoCAR 3. The configuration is capable of efficient long range urban and highway driving and can provide high power acceleration and hill climbing while minimizing harmful emissions.

The following sections will discuss the detailed powertrain design, the evaluation and reasoning behind each component selection, the design tradeoffs, the control strategy, and the component sizing.

9.3.1. Powertrain Configuration

The proposed powertrain for EcoCAR 3 is a Plug-in Parallel Hybrid Electric Vehicle (PP-HEV). This can be achieved through several different configurations; however, this proposed design will use a “P2” configuration where the motor is located just before the transmission with a single through-shaft as shown in the Figure 9. The powertrain components are, in order, turbocharged diesel engine, dry clutch, permanent magnet alternating current electric motor, and automated manual transmission.

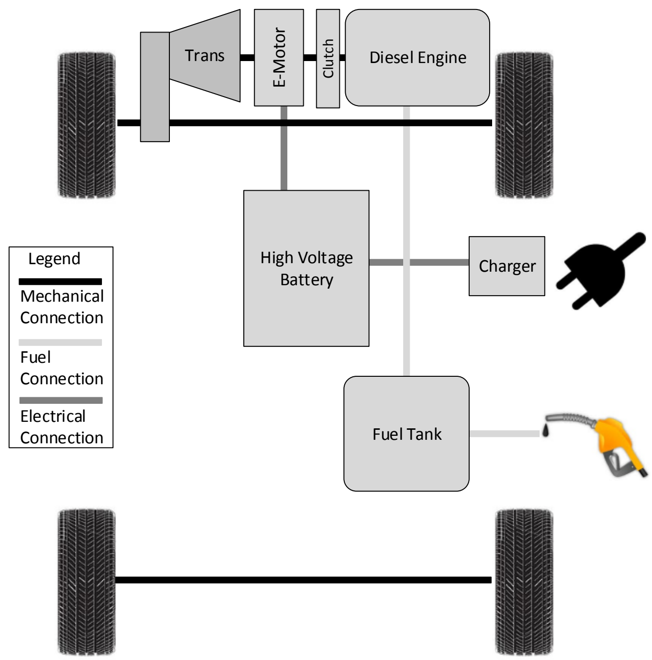


Figure 9.2: Plug-in Parallel Hybrid Electric Vehicle Configuration

The diesel engine is coupled to the electric motor through a clutch while the electric motor is directly coupled to the transmission. The electric motor has a through-shaft design that allows the engine to transmit torque to the wheels while the motor freewheels or applies a negative torque for opportunity charging of the battery when it is beneficial to increase engine load. Disengaging the clutch will allow for electric only operation without incurring parasitic losses from spinning the engine. The through-shaft motor and directly-coupled motor and transmission create a very compact powertrain package. The pre-transmission location of the motor allows for the motor speed and torque output to the wheels to be adjusted over a wide range if needed and allows the motor to operate in its most efficient range at all times. Even though the electric motor has a relatively high efficiency across its entire operating range, there is still a 20% variation, so the transmission allows for near maximum motor efficiency in all driving situations. In this location, the motor can be used for speed matching on the input side during shifting to reduce clutch and synchronizer wear and enable faster clutch engagement. Precise control of motor speed and torque eliminates the need for a clutch between the motor and transmission [10].

The plug-in feature of the vehicle allows charging of the battery with AC grid electric power. This allows the battery to be charged while the vehicle is parked and when coupled with the larger battery pack gives an extended electric only range for subsequent vehicle operation.

9.3.2. Cost Consideration

Estimating component cost, especially for advanced technology components, requires close communication with automotive suppliers. Lack of accurate cost estimation knowledge and sources led to the development of advanced technology evaluation reports, such as the 2009 report by the National Academy of Engineers [11]. While the cost estimations contained in this report are accurate according to the information available at the time, there are many other factors that can affect component costs, including materials and manufacturing methods. Once a price estimate and a potential energy consumption improvement have been determined then the technologies impact per retail cost dollar can be calculated and a consumer payback period can be determined. A simple ratio that can be used for technology comparisons is the cost per improvement, obtained by dividing the cost increase by the fuel consumption improvement. The technology with the lowest value for ratio is the most effective, when cost is the primary factor. To evaluate a technology's effectiveness for satisfying a given consumer payback period, the following inequalities can be used:

$$\frac{Tech.\ Cost}{(\Delta FC(\%))} \leq baseline\ FC \cdot \frac{miles\ driven}{year} \cdot \frac{cost}{gallon} \cdot length\ of\ ownership \tag{28}$$

For technology cost in USD, $\Delta FC(\%)$ is the change in fuel consumption as a percentage, the baseline fuel consumption in gallons/mile, cost per gallon in USD, and length of ownership in years. Recent data show that average length of new car ownership in the US is 5.95 years while the average miles driven per year is 13,476 miles [12]. Assuming an average fuel price of $4/gallon and a baseline fuel economy of 30 mpg yields a limiting right side value of $10,690 in equation 26. A technology that exceeds this value will not provide a return on investment within the vehicle ownership time. Ideally, the consumer payback period would be shorter than the ownership time so that the consumer will have a net savings in comparison to the baseline vehicle, providing motivation to buy the vehicle.

Evaluation of the technologies presented in the NAE report using their cost and fuel consumption estimates reveals that 22 of 38 falls below the limiting value of $10,690 calculated above. Of these 22, most of them only provide a consumption reduction of 2-3%, while only two provide a fuel consumption reduction of greater than five percent. The other technologies with the largest impact on consumption were also the most expensive to implement and did not satisfy the payback period. These technologies included the gasoline to diesel conversion, power-split hybrid, and the series hybrid. While this is a cause for concern, it should not be used as a primary decision factor for vehicle design decisions. A survey conducted by the University of California Transportation Center revealed that vehicle purchase decisions even for hybrid vehicles are largely driven by emotion and not by technical or financial evaluation [13]. A payback period or fuel savings cost was rarely calculated by consumers. The primary vehicle attribute that attracted consumers to fuel efficient vehicles was the fuel economy itself while the price difference or financial impact of this fuel economy was rarely considered. Also, while there are many variables in determining costs, a near certainty is that advanced technology costs will decrease over time as the technology matures and production volumes increase. This will improve the viability of these technologies in the future. For these reasons, more emphasis should be placed on fuel economy improvements rather than on the cost of the improvements. However cost must still be considered and vehicle purchase prices should be kept within reasonable limits. It is expected that these limits will be defined for the EcoCAR 3 competition.

9.3.3. Mass Consideration

Component mass is a critical parameter considered in the design process since it directly impacts acceleration and fuel consumption. Mass can also affect the ride comfort, handling, and braking of the vehicle. The primary contributors to mass in the final vehicle design are the vehicle chassis and body, battery pack, engine, transmission and electric motor. The vehicle chassis and body will not present opportunities for significant mass reduction since the team will be retrofitting an existing vehicle rather than designing a new vehicle. Some of the body components could potentially be replaced with lighter weight versions but, overall, the benefits would be minimal within the bounds of the competition rules. The battery pack, engine, transmission, and electric motor selections represent the biggest opportunity for careful consideration of weight.

9.3.4. Engine Selection

The engine selection process focused on small displacement turbocharged engines which provide packaging and fuel economy benefits while still providing acceptable performance. Turbo CI and Turbo SI engines both provide torque and power improvements but typically at the expense of fuel economy when running at high boost pressures. Two primary methods for decreasing engine fuel consumption and emissions are *downsizing* and *downspeeding* [16]. This can be done on both CI and SI engines. However, CI engines are a better candidate for these methodologies. High torque density allows downsized CI engines to provide sufficient performance and the greater efficiency and higher torque at low speeds allows for downspeeding. Downspeeding is achieved by increasing gear ratios or final drive ratio to bring engine speed down but maintain vehicle speed.

The fuel of choice is B20 biodiesel. Aside from the fact that B20 biodiesel was the only fuel choice compatible with the diesel engine chosen for the design, this fuel also has a few advantages over E85 and E10. B20 biodiesel is closer in energy content and fuel properties to standard No. 2 diesel than E85 or E10 is to gasoline. B20 contains 99% of the volumetric energy content of diesel while E85 and E10 contain only 73% and X% of the energy content of gasoline, respectively [17]. The energy content of E85 is 33% less than that of B20; however both fuels can have varying values due to the lack of close regulation on biofuel properties. B20 has shown improved lubricity with no negative impacts on engine performance or durability while E85 is corrosive to some materials found in the engine and fuel system. An NREL report that states no engine modifications are necessary for biodiesel blends up to B20 ensures that widely available diesel engines will meet the fuel requirements [18]. B20 provides reduced GHG emissions and reductions in all emissions except for NO_x, which remains approximately the same as the standard diesel emissions [19]. The oxygen content of biodiesel fuels allows for more complete combustion which is one of the factors for reduced emissions.

9.3.5. Battery and Electric Motor Selection

Electricity is the second energy carrier that will be used in the proposed EcoCAR 3 design. Electricity is an attractive choice due to the high conversion efficiencies for chemical to electrical and electrical to mechanical processes. Another advantage is the mass and volume of electric machines is less than that of combustion engines for equivalent power and torque ratings. The drawback of electricity is the currently available storage technologies are limited by size, weight, durability, and/or cost.

Current battery options for EVs and HEVs include nickel metal hydride (NiMH) and lithium ion (Li-ion) due to their superior power and energy density over other chemistries. A Li-ion battery was chosen since it has the highest power and energy density, while the cost is acceptable.

The battery and electric motor must work together with the help of an inverter. High performance motors typically have the option of purchasing paired inverter. This is advantageous as the inverter is tuned to work well with the motor and provide maximum performance and efficiency.

The electric motor selection process could include a vast array of available options, however, the necessity for high efficiency, high torque, efficient regeneration and precise control eliminate direct current (DC) motors and limits the choices to alternating current (AC) motors, specifically permanent magnet synchronous motors (typically called PMAC) and induction motors. Both motors types exhibit high efficiency, depending on the overall system. PMAC motors have two important advantages that were the deciding factors: they have higher torque density and are manufactured in greater quantities. PMAC motors are available in more power levels, package sizes, and related options. Size, weight, cost, and availability are all key factors for the proposed application which made a PMAC motor the definite choice.

9.3.6. Engine and Motor Pairing

The torque-speed curves of the engine and motor were matched during selection and sizing of these components. This is essential since they are constrained by a common shaft to run at the same speed. When the engine is running at city and highway speeds, it is desirable for the electric motor to provide high torque to assist acceleration for passing and other maneuvers. The motor will be used to provide instant throttle response in an acceleration event where the engine cannot respond quickly.

Pairing an electric motor and diesel engine allows vehicle emissions and fuel economy optimization. Hardware-in-the-loop tests have shown that a diesel parallel hybrid powertrain utilizing a control strategy with these two optimization targets is capable of reducing NOx emissions by up to 35% and fuel consumption by up to 15% when compared to a conventional diesel powertrain with the same engine[].

9.3.7. Transmission Selection

The current transmission options for transverse powertrains are automatic, manual, continuously variable, and dual clutch. All of these transmissions are capable of handling the motor torque and speed when properly sized. However, the manual transmission has the lowest cost and highest efficiency. The team plans to automate the manual transmission by designing and fabricating a custom electronic actuator shift system. This will provide the convenience of an automatic transmission with the high mechanical efficiency of a manual transmission. It will also allow gear selection to be handled by the vehicle control system to maximize the efficiency of the powertrain.

9.3.8. Component Integration in Compact Sedan

The packaging of powertrain components may present a greater challenge than the weight requirements. The most limiting constraint for the powertrain in a compact sedan is the overall length of the powertrain, as it is limited by the distance between the frame rails or any sheet metal contours that protrude into the engine bay. Packaging a hybrid powertrain will be challenging, but can be made possible with several other changes. The small diesel engine will be shorter than the standard I4 or V6 engine. Transmission and motor selection will focus on length to further reduce the overall length. Removing the FEAD from the engine will trim up to 100mm from the package requirements. Since the motor will be directly coupled to the transmission input shaft, more complete integration of the motor and transmission will receive serious consideration.

The battery location will be determined according to the geometry of the vehicle body and weight distribution. Safety will play a key role in determining battery location, as batteries must be kept away from crumple zones to avoid damage in a collision. The safest battery location is near the center of the vehicle since this area is least likely to be damaged in a collision. A central location also helps to preserve the front to rear weight distribution of the vehicle and maintain ride and handling characteristics and eliminate the need for new suspension components. It may be possible to split the battery pack into two or more units to allow for more packaging options. However, a split battery pack will have greater cooling system complexity and cabling/fusing requirements.

9.3.9. Simulation and Control Strategy

"Amir Rezaei contributed to the writing of this section"

A model has been developed in AMESim for the proposed powertrain. The EMS was designed in Simulink and by co-simulation capability of AMESim/Simulink, the whole system was simulated. Both plant model in AMESim and controller model in Simulink were designed with high flexibility for choosing any combination of several engines, electric motors, transmissions and etc.

The following figure represents designed EMS inputs/outputs. Several EMS have been developed for the proposed powertrain configuration: Rule-Based Controller (RBC), Instantaneous Optimal Controller (IOC) and Model Predictive Controller (MPC). However, at the moment of writing this proposal only RBC has been tested on the plant designed in AMESim. IOC and MPC have been tested on a simpler but similar plant designed in MATLAB.

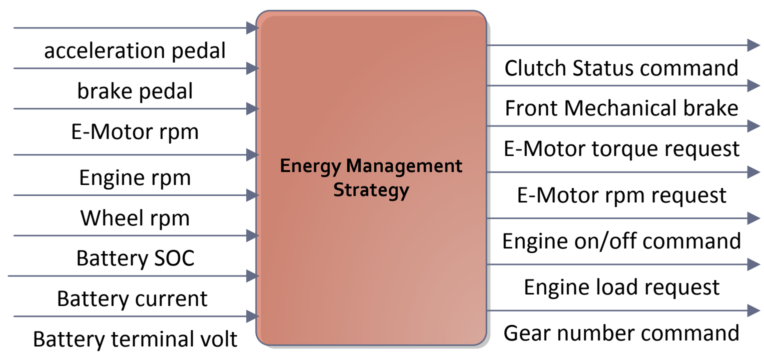


Figure 9.3: EMS Input/Output Designed in Simulink

All of designed EMSs, try to make optimal actions at each moment while not violating plant constraints as the following:

- Constraint on gear shifting frequency: 2 seconds interval
- Constraint on time duration of engine on or off state: 10 seconds
- Constraint on engine maximum/minimum rpm: manufacture's recommendation
- Constraint on e-motor maximum: manufacture's recommendation
- Constraint on time duration of e-motor peak power: manufacture's recommendation
- Constraint on battery pack maximum charge/discharge current: manufacture's recommendation
- Constraint on SOC boundaries in CS mode: based on battery most efficient region
- Constraint on engine maximum dynamic torque: based on engine torque profile
- Constraint on e-motor maximum dynamic torque: based on e-motor torque profile
- No regenerative braking when vehicle speed is less than 3 mph.

For the RBC, two modes have been considered: (A) economy mode in which EMS tries to minimize fuel consumption and (B) performance mode where EMS tries to maximize acceleration capabilities. In economy mode, RBC chooses the admissible gear (the gear that doesn't violate any of constraints) with the least fuel consumption and suggests that gear to EMS state machine. EMS waits until gear shifting interval is passed and then applies suggested gear number to the plant.

In CD mode, the engine becomes on only when e-motor cannot provide driver's demanded power but it is not allowed to charge the battery.

In CS mode, when gear shifting is done, the power split strategy is based on engine Optimum Operating Lines (OOLs) shown in the Figure 11 for the engine of Honda Civic IMA. On the engine BSFC map, OOLs are calculated offline for each speed based on the most efficient points at that speed. OOL1 contains the torque $T_{eng}$and fuel rate $\dot{m}_{fuel}$ of the most efficient engine operating points at each speed, OOL2 contains $\left(T_{eng}\ ,\ \dot{m}_{fuel}\ \right)$ pair of the second best operating points and so on for each progressive OOL. In hybrid mode, every 0.1 second the controller checks engine speed and then for that speed, interpolates the values of $\left(T_{eng}\ ,\ \dot{m}_{fuel}\ \right)$ from each OOL. Since the driver's demanded torque T_{total} on the wheels is known, the RBC can determine the required e-motor torque for each pair of$\left(T_{eng}\ ,\ \dot{m}_{fuel}\ \right)$:

$$T_{em} = T_{total} - T_{eng} \tag{36}$$

Then the RBC searches among $\left(T_{eng}\ ,\ \dot{m}_{fuel}\ , T_{em}\ \right)$ options in order to find the option that minimizes the following cost function:

$$\mathrm{C} = \ \mathrm{argmin}_{T_{eng}}\{\dot{m}_{fuel} + \mu P_{battery}\} \tag{37}$$

Where μ is a constant number and can be tuned for achieving the best performance. Then EMS sends selected T_{eng} and T_{em} as commands to the engine and e-motor, respectively. The algorithm that was just explained is a simplified but fast version of a well-known optimal controller for HEVs called Energy Consumption Minimization Strategy [18]

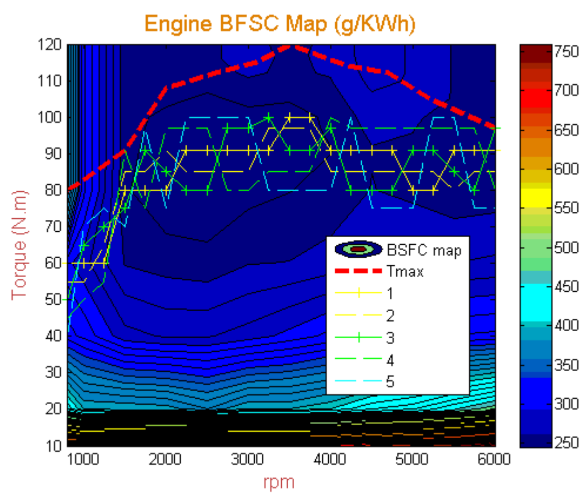


Figure 9.4: Optimal Operating Lines (OOLs) on BSFC map of an engine (Honda Civic)

It should be mentioned that while the above semi-optimal algorithm is the heart of the designed RBC, it will only be executed if the driver has chosen economic mode, engine is in ON state and EMS is in CS mode. There are some other situations and details that are not discussed for sake of proposal length.

As was mentioned IOC and MPC have not yet been tested on the plant model designed in AMESim. But a simple quasi static plant model has been designed in MATLAB that is based on the Honda Civic IMA [19]. The Honda Civic IMA is a mild HEV that has the same powertrain configuration as the proposed powertrain for EcoCar3. As a result it is reasonable to expect very similar improvements for the proposed powertrain after using IOC and MPC. The following figure represents the fuel economy improvement for the Honda powertrain by using IOC and MPC.

As can be seen in the following figure, EMS plays an important role for improving fuel consumption. In other words, EMS can considerably affect component sizing, component selection and final vehicle cost.

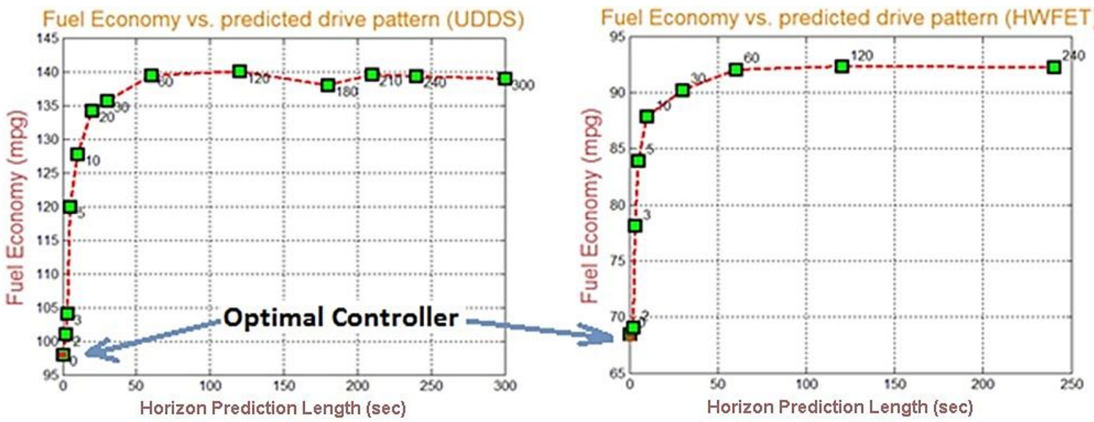


Figure 9.5: Effect of EMS on Fuel Consumption

9.3.10. Energy Consumption of the Proposed Powertrain

"Amir Rezaei contributed to the writing of this section"

The SAE recommends a balanced SOC in Charge Sustaining (CS) mode for accurate and repeatable determination of energy consumption in a typical drive cycle [SAE J1711 Standard]. However, in plug-in HEVs that begin the test with a fully charged battery, the EMS will continue to use electric energy until the battery charge is depleted to the minimum limit at which point it will transfer from CD to CS mode. In CD mode the engine may be turned on if the motor alone cannot satisfy driver's demand but unlike CS mode, fuel energy will not be used to charge the battery in CD mode.

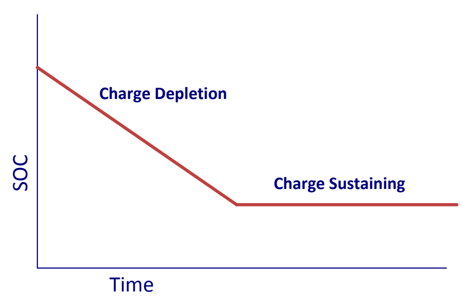


Figure 9.6: Battery SOC Trends in SD and SC Operation

Due to this complex combination of energy consumption it is difficult to determine the exact consumption and fuel economy for a standard cycle since the overall efficiency of the vehicle will vary as the battery SOC drops to a minimum and the vehicle enters CS mode. The SAE J17100 standard suggests using several drive cycles of one type in a row and then recording electric and fuel energy consumption for each cycle in 2-D plot as shown in the following figure. Now it is possible to estimate the balanced fuel consumption for that drive cycle.

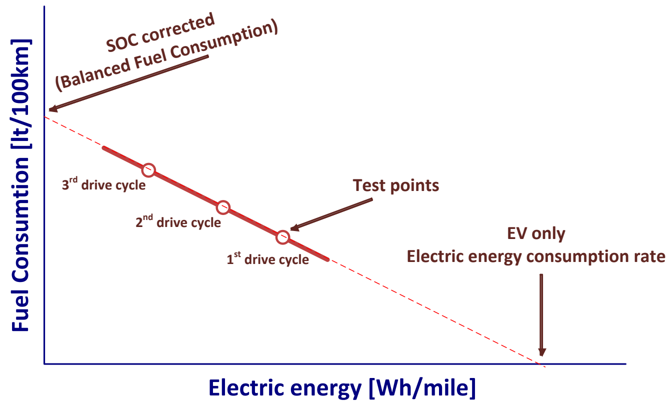


Figure 9.7: Fuel Consumption Estimation

This method is used for energy consumption determination for the PP-HEV powertrain.

9.3.11. Sizing the Proposed Powertrain

"Amir Rezaei contributed to the writing of this section"

The supervisory control strategy in a P-HEV plays an important role in maximum power requirement and energy consumption of each component. As a result, a precise sizing and component selection should be done after running extensive simulations under a variety of conditions. However, it is still possible to make an estimation of each component characteristic without factoring in each detail of the control strategy for a PP-HEV. The power distribution approach chosen is to allocate the engine for required cruising power and use the electric motor for providing or absorbing the dynamic power due to acceleration. Engine efficiency is much lower than electric motor and also the variation of engine efficiency map is higher than the electric motor. So while it may not be possible in practice, it is more beneficial to keep the engine working in a constant high efficient operating point. Hence, even without any prior knowledge about the final EMS, an effective approximation is to size the engine and electric motor based on average and dynamic power, respectively, on highway drive cycles.

9.3.11.1. Engine

"Amir Rezaei contributed to the writing of this section"

In a highway drive cycle, there are limited regeneration opportunities, so in the event that the battery SOC reaches its minimum value, the engine must be able to propel the vehicle at the desired speed. Since the target vehicle top speed must be greater than 85 mph the cruising speed $V_{hwy} = 75\ mph$ can be one criteria for determining engine power (assuming transmission and final drive efficiency $\varepsilon_t = 0.95$) :

$$P_{eng,highway} = \frac{q}{\varepsilon_t}\left(mgC_{rr}V_{hwy}. + \frac{1}{2}\rho C_D A_f {V_{hwy}}^3\right) \approx 30\ KW \tag{38}$$

The coefficient $q = 1.1$ is added in order to provide enough flexibility for EMS to have the option of charging battery while cruising. [20]

The other criterion for sizing engine power is having a balanced SOC in urban driving conditions. As was mentioned before it is not possible to recover all of potential and kinetic energy of the vehicle during a braking event because of safety and drivability issues [3]. So there will be occasions when the engine should be turned on to charge the battery and propel the vehicle simultaneously. So engine power should also be greater than:

$$P_{eng,city} \geq \frac{1}{\varepsilon_{em}\varepsilon_{inv}\varepsilon_{bat}} \left(\bar{P}_{propel,city} + (1-\varepsilon_{regen})\bar{P}_{brake,city}\right) \approx 6\ KW \qquad (39)$$

Where $\varepsilon_{regen} = 0.70$ is the percent of braking power that can be recovered, $\varepsilon_{em} = 0.9$ is the average efficiency of the motor/generator, $\varepsilon_{inv} = 0.85$ is the inverter average efficiency and ε_{bat} =0.75 represents the average charging efficiency of the battery. As can be seen the cruise power requirement is the major factor for indicating engine power:

$$P_{eng,max} = \max\{P_{eng,city}, P_{eng,highway}\} = 30\ KW \qquad (40)$$

9.3.11.2. Electric Motor

"Amir Rezaei contributed to the writing of this section"

As was mentioned before, the basic method for electric motor sizing is to satisfy the dynamic power requirements necessary for acceleration in different driving situations. The most important one is the 0-60 acceleration time design target. In the proposed configuration engine and motor speeds are the same but there is an absolute lower limit on engine speed that is the idle speed and there is a lower limit on engine speed below which it produces low torque at low efficiency. For these reasons the electric motor is used for electric only launch to provide the initial acceleration. When the vehicle approaches a specific speed, then engine can be connected to drive wheels through clutch to help the electric motor.

Since the torque/power profile of hybrid powertrain is complex, acquiring an analytical solution to the integral of equation X is difficult. A simplified approach for estimating the acceleration time is to assume a part of engine power will be used for overcoming the road load. As a result, by assuming only a single gear during acceleration from equation (), the resulting equation is: [3 Ehsani]:

$$P_{em,max} = \frac{m_e}{2\varepsilon_t t_a}\left(v_f^2 + v_b^2\right) - \frac{1}{t_a - t_i}\int_{t_i}^{t_a} \varepsilon_t P_{eng} - mgC_{rr}v - \frac{1}{2}\rho C_D A_f v^3 dt \qquad (41)$$

Where $P_{em,max}$ is an estimation of required power from electric motor, $m_e$is vehicle inertia mass from equation (source), t_a=9 is acceleration time, v_b is motor base speed, P_{eng} represents engine power profile and t_i represents the time in which engine can be connected to the wheels. In order to calculate the above equation, the following assumptions have been made:

- Electric motor base speed = 2000 rpm
- Engine minimum speed = 800 rpm
- Engine maximum speed = 6000 rpm
- Engine speed in which the power is maximum ω_{pmax} = 5200 rpm
- $P_{eng} \approx -\frac{P_{eng,max}}{\omega_{pmax}^2}\left(\frac{30N}{\pi}\frac{v}{r}\right)^2 + \frac{2\,P_{eng,max}}{\omega_{pmax}}\frac{30N}{\pi}\frac{v}{r}$ where $r = 0.31$ is wheel radius and N is gear ratio
- $v(t) = v_f sin\left(\frac{\pi}{2t_a}t\right) \quad , \quad 0 \leq t \leq t_a$

The following figure represents the electric motor power estimation to meet acceleration time of 9 seconds for different first gear ratios. As can be seen the gear ratio of 7 (both transmission and final drive) yields the least required power from the electric motor. So:

$$P_{em,max} = 45\ KW \qquad (42)$$

Again it should be emphasized that these estimations have to be confirmed by simulating the vehicle.

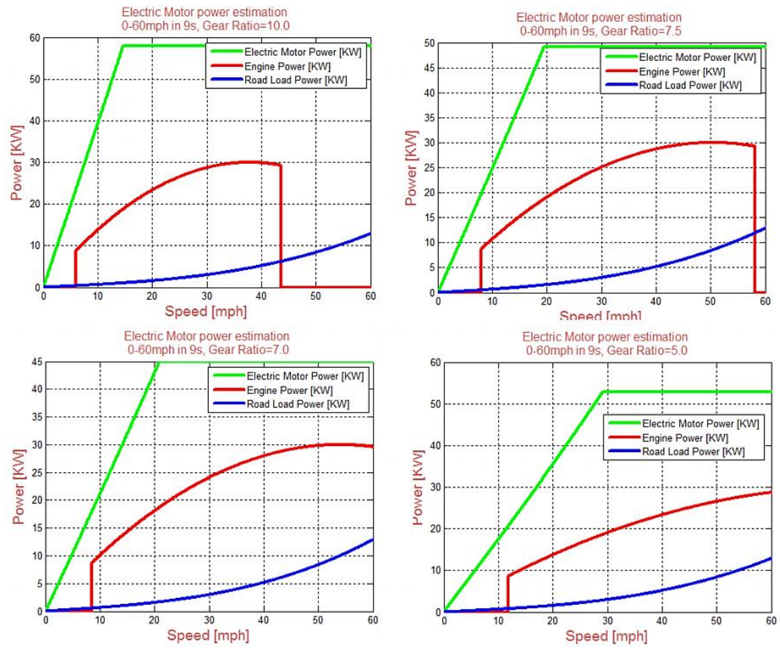


Figure 9.8: Electric motor power sizing for different first gear ratios

9.3.11.3. Transmission

"Amir Rezaei contributed to the writing of this section"

For the proposed configuration, since the engine is mechanically connected to the wheels a multi gear transmission is required for controlling engine speed. An estimation of the first gear ratio was calculated in the previous section. The last gear ratio can also be determined by matching the two upper limits of the speed: speed limit due to maximum power and speed limit due to maximum engine/motor rpm:

$$v_{max} = \frac{r\ \omega_{pmax}}{N_n} \quad , \qquad P_{max} = P_{em,max} + P_{eng,max} = \frac{1}{\varepsilon_t}\left(mgC_{rr}v_{max} + \frac{1}{2}\rho C_D A_f v_{max}^3\right) \ \Rightarrow$$

$$N_n \approx 0.4 \qquad \Rightarrow \quad v_{max} \approx 115\,mph \tag{43}$$

That meets the target top speed criterion of EcoCAR 3. The other gear ratios can be determined by simulating the vehicle.

9.3.11.4. Battery

"Amir Rezaei contributed to the writing of this section"

Like previous sections, the battery power can be determined by maximum electric motor power:

$$P_{battery,max} \geq \frac{P_{e,max}}{\varepsilon_{em}} \qquad \Rightarrow P_{battery,max} \approx 52KW \tag{44}$$

A battery power of 52 kW is sufficient to power the 45 kW electric motor with an efficiency of 87%.

10. **Summary and Conclusions**

The introduction of new technologies and wholly new architectures greatly increases the number of design options that must be handled during the development process. It is no longer feasible to use the traditional design-prototype-test process. Now, engineers must use computer-aided engineering tools for system level vehicle modeling and simulation to evaluate the vast array of options and to make well informed design decisions. Effectively used, modeling and simulation can significantly reduce development time and cost while leading to a more optimal design, especially when included as part of a comprehensive vehicle design process. The model development showcased in this proposal follows the proven V-diagram process structured to efficiently produce useful models and simulation results.

The models were developed for the purpose of complete vehicle system simulation and energy consumption estimation. The models utilize torque, fuel consumption, and efficiency maps to estimate the behavior of engines and electric motors. Battery behavior was estimated with voltage and internal resistance maps. The transmission model included gear ratios and efficiencies. Appropriate temperatures were selected for each component and temperature was assumed to be constant. This assumption leads the model to neglect cold start engine behavior, and cold motor and battery operation which have lower efficiency than steady state operating temperature. However, energy consumption for the UDDS and HWFET cycles does not include cold weather conditions and assumes the vehicle is already warmed up. More aggressive testing that is included in the determination of new vehicle sticker values does include these colder conditions and more aggressive testing but these tests were not required for energy consumption determination as defined in the RFP.

A quasi-static model that accounts for component force and power characteristics and efficiencies can be effectively used for powertrain system design. This has been validated by numerous researchers, showing the comparison of actual test data to modeled energy consumption was generally within 10%.

The modeling results show that an average conventional vehicle powered by a combustion engine cannot meet the energy consumption target when the engine is sized to meet the acceleration target. At this point, the engine's most efficient load range is greater than the average load during the urban and highway cycles. This prevents the engine from operating at maximum efficiency.

It was also determined that a battery electric vehicle could not meet the required range target of 320 km while keeping the vehicle weight below the GVWR. This is due to the low energy density of the batteries which leads to a large, heavy battery pack. The high electric motor efficiency enables the BEV to easily meet the energy consumption target despite the high vehicle mass, however.

A series hybrid vehicle is shown to have the potential to meet the acceleration and energy consumption parameters when the components are optimally sized. This is determined by balancing the engine and electric motor power characteristics to meet the steady power requirements and acceleration power requirements, respectively. A two speed transmission enables optimum motor performance while also allowing for the engine to accelerate the vehicle and cruise when low battery SOC prevents motor assist.

The initial modeling exercises led to the conclusion that a hybrid powertrain is required to meet all of the EcoCAR 3 design targets. The three major hybrid electric architectures, series, parallel, and power-split, were considered. The power-split architecture was removed from consideration since it has the greatest physical complexity, greatest controls complexity, and potentially highest cost. The series and parallel architectures were modeled for further evaluation, each with an SI and IC engine option. The results showed that each design could meet the targets.

The design proposed for EcoCAR 3 is a plug-in parallel hybrid vehicle (PPHEV). This design was chosen over the other designs due to its compact design and lower cost while still providing performance and economy that meets or exceeds the design targets. This design requires more complex controls but a thorough understanding of hybrid vehicle controls and powertrain behavior has enabled the development of an effective control strategy.

The modeling process discussed in this report along with the content and format of the report itself demonstrate key skills necessary for success in the EcoCAR 3 competition. Critical thinking and problem solving skills developed through a challenging engineering curriculum were used to select and execute an effective modelling and design process similar to what will be required in the initial stages of EcoCAR 3. These skills enable objective evaluation and informed decision making at each step in the development process, resulting in an accurate vehicle model and sophisticated but eloquent co-simulation system. The iterative, piece-wise fashion of the model development process ensures complete understanding of the model, its function, and its limitations.

The simulation system developed and demonstrated in this proposal provides the basis for a model based design process. This Simulink/AMESim co-simulation system serves as both a testing platform for various vehicle designs and as a model-in-the-loop development tool for an energy management controller. The controls logic developed in Simulink and utilized in the modeling exercises can be adapted to interact with hardware components and is then ready to download to a rapid prototyping controller.

The team at Michigan Tech has developed a depth of knowledge through an engineering curriculum including classes on vehicle component design, vehicle system design, vehicle and control system modeling, and vehicle testing, along with the standard mechanical engineering classes. Each of these vehicle classes has a strong focus on hybrid electric propulsion systems that has led to the development of a thorough understanding of the various architectures, control strategies, component options, and power flows. Combined with a strong multidisciplinary culture on the university campus, promoting cooperation and sharing, Michigan Tech is poised for success in the upcoming EcoCAR3 competition.

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