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Mathematical Modeling Techniques and Development of a Blended Model for Hybrid Electric Vehicle Powertrain

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Abstract

The gradual decline trend of oil resources and increasing global warming around the world have created an urgent need to search for alternate options for crude oil. Electric Vehicles (EVs) can counter the need for crude oil but they have range anxiety. Hybrid Electric Vehicles (HEVs) have proved to be a viable option for ensuring improved fuel economy and reduced emissions. The performance of the vehicle, energy consumption, and emissions depend upon the selection of different vehicle topologies.

Before manufacturing an actual HEV prototype and testing the same in the laboratory, on test tracks, and the actual field, it is important to give an appropriate consideration towards the modeling of it in a simulation environment. There exist three main stages of computational modeling in the development activity of HEVs, viz., model in the loop (MiL), software in loop (SiL) and hardware in the loop (HiL). Development of a MiL can further be classified into three main modeling approaches, viz., kinematic modeling, quasi-static modeling, and dynamic modeling. The development of a virtual simulation model is a pre-requisite for the development of an efficient control strategy for HEVs, which ultimately leads to an optimized load-leveling amongst the power plant. This paper presents a brief review of the above-mentioned modeling approaches of HEVs. The research work describes a blend of forward and backward modeling approaches for a full parallel hybrid electric powertrain. Finally, the results of fuel consumption and energy management are discussed in detail.

Keywords: Hybrid Electric Vehicle; Modelling; Simulation

Introduction

The automobile industry has experienced innumerable advances since its conception over the last 100 years. However, ever-increasing greenhouse gas emissions and depleting oil resources demands for automobiles with alternate fuel and alternate power plants. Hybrid electric vehicles (HEVs) are emerging out as strong candidates in this tectonic shift in the automotive market. Optimizing powertrain component size for desired vehicle class and a usage profile is beneficial for reducing tailpipe emissions. Simulation is playing a key role in the development of new transportation systems. The development cycle of any product is comprising of 3 phases of simulations, viz., Model in Loop (MiL), Software in Loop (SIL), and Hardware in Loop (HiL) [1]. In the case of MiL simulations, a virtual model of the system is developed in the modeling software.

HEV modeling can be performed using 3 different techniques, viz., the kinematic or backward technique, the quasi-static or forward technique, and the dynamic technique. In the backward approach, the simulation is realized from the objective (driving cycle) to the cause (required energy of the power plant to propel the vehicle) [2]. In the forward approach, the simulation is realized from the cause (energy supplied by the power plant) to the effect (velocity of the vehicle).

During managing the modeling processes, it is important that the direct energy flow can have a forward as well as backward direction, corresponding with driving or braking the vehicle. Backward or 'effect-cause' simulation method is often used for energy consumption assessment in vehicles [3].

This paper explains about various modeling techniques used for HEV. The blending of the backward and the forward methods has been developed in this research work for modeling HEVs. This paper also explains the simulation results of a developed model by applying a simple control strategy.

Modeling techniques

In the early stages of research and development, before the prototype building stage, it is necessary to develop a mathematical model of the systems, engine, electric motors, converters, inverters, entire vehicles, etc. This stage endeavors a detailed study of the behavioral patterns of the systems and facilitates further optimization. One of the main objectives of building mathematical models of the complete vehicle in a simulation environment is to estimate fuel consumption and emissions while negotiating different driving cycles. Different modeling techniques of the vehicles are discussed in the next section.

The backward approach

The backward approach of modeling of the vehicle is also known as Kinematic modeling methodology. In this technique, the input variables are the speed of the vehicle and the grade angle of the road [4]. The principal assumption of this backward-facing model is the ability of the vehicle model to meet the demands of the drive cycle. Here, the engine speed is determined using simple kinematic relationships starting from the wheel revolution speed, back to the axle speed, back to the transmission output and input speeds, and finally to the speed of powertrain at clutch (Fig. 1). The tractive torque that should be provided to the wheels to propel the vehicle according to the chosen speed profile can be calculated from the vehicle longitudinal dynamics and propagated back to the ICE via drivetrain. Once engine speed and torques are estimated, instantaneous fuel consumption and emission rates are estimated and then are integrated over the driving cycle to obtain cumulative outcomes.

The backward approach assumes that the vehicle meets the target performance so that the vehicle speed is supposedly known a priori. The kinematic modeling technique ensures that the driving speed profile will be exactly followed. However, there exist no guarantees that the given vehicle will actually be able to meet the desired speed trace since the power request is directly computed from the speed and not checked against the actual powertrain capabilities. This approach also neglects the thermal modeling of powertrain components, viz., engine, motor, battery, etc. In the simulation, the backward approach includes a "fail-safe" feature which stops the simulation run if the required torque exceeds the maximum torque available (from the electric motor and engine). The simulation models built by these techniques rely on efficiency maps that were created based on torque and speed data and usually produced during steady-state real-world testing. This results in relatively simpler calculations than the other vehicle modeling techniques [1].

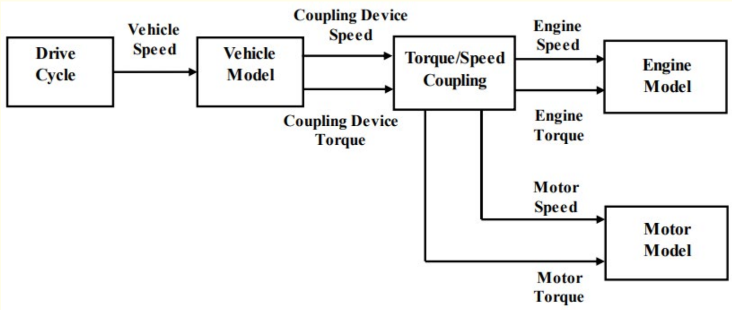

Figure 1: Backward modeling approach.

The forward approach

The forward approach of modeling of the vehicle is also known as a quasi-static modeling methodology. This technique of modeling of the vehicle makes use of a driver model, typically a PID, which compares the target vehicle speed (speed demand from driving cycle), with the actual speed of the vehicle and then generates a power demand, needed to follow the target vehicle speed profile.

Unlike backward-facing kinematic models, the speed trace is not imposed onto the vehicle model in forward-facing or quasi-static models. This augments and inevitably a small margin of error between the actual vehicle speed and the speed trace. Here the role of the driver model comes into the picture to minimize this margin of error (Fig. 2). This is analogous to the role of a real-world test driver carrying out an emissions test for vehicle type-approval at the chassis dynamometer in the laboratory.

Once the propulsion torque and speed of the engine have been determined, instantaneous fuel consumption can be estimated similarly as explained in the previous section [5]. The suitability and accuracy of the forward modeling approach depend very much on the nature of simulation studies to be conducted. The forward modeling approach provides reasonable accuracy when it is used for evaluation of the fuel consumption and NOx emissions [6]. With these techniques, the prediction of the soot emissions is difficult as it is dependent on the acceleration transients and phenomena like turbo lag.

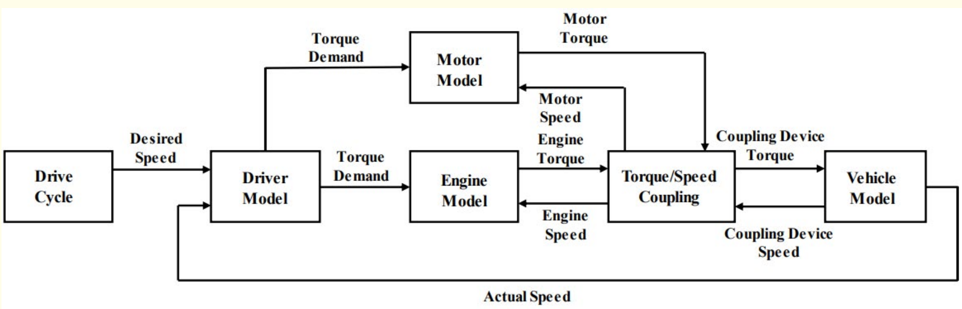

Figure 2: Forward modeling approach.

The dynamic approach

In the case of dynamic modeling of the vehicle, apart from the development of the longitudinal model of the vehicle, an in-detailed model of the power plants (engine / electric motor/fuel cells) is developed which is capable of capturing all the acceleration transients. The solution of the equations governing the conservation of mass, momentum, and energy of the flow for each element of the network can then obtained using a finite difference technique.

In this way, highly dynamic events, such as abrupt vehicle accelerations during tipin maneuvers can be properly and reliably simulated with reasonable accuracy [7].

Development of HEV model in simulation environment

In this research paper, a mathematical model of a full parallel hybrid electric vehicle is developed for the simulation by blending the backward and forward modeling approaches. It augments the advantages of both of the modeling methods. Forward modeling helps to trace the results whereas backward modeling reduces computation burden. This method makes use of the driver model to tune the PID controller whereas the drive cycle and vehicle model to compute the load requirement of the vehicle.

The driver model generates the acceleration and deceleration command and it is supplied to the supervisory controller which in turn operates both the powertrains, viz., engine, and electric motor by comparing actual and desired speeds. The generated torque is then passed forward through the powertrain to the wheel. The advantage of this modeling technique is that the speed profile trace can be done accurately and torque value at any point in the powertrain can be estimated rapidly.

The powertrain components include driver model and drive cycle data, supervisory control unit, vehicle dynamic load calculations, conventional diesel engine, electric motor, battery, transmission, torque coupling device, and tire.

The concept of torque coupling is used to couple the two powertrains [8]. The mathematical model of the HEV powertrain is shown below in Fig. 3.

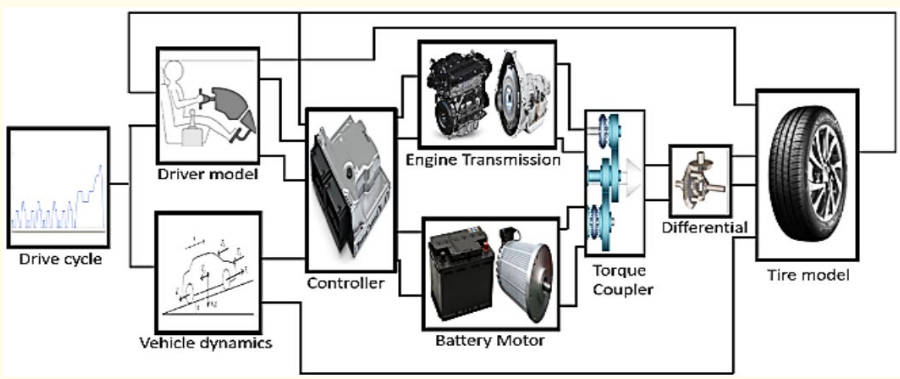

Figure 3: Parallel HEV powertrain model.

Driver model

The driver model exclusively consists of the PID controller (Fig. 4). It receives the reference speed profile as an input from the driving cycle or external device and the feedback of the actual speed profile from the wheels. PID controller then calculates the error between these two-speed profiles received and generates an acceleration and deceleration commands accordingly.

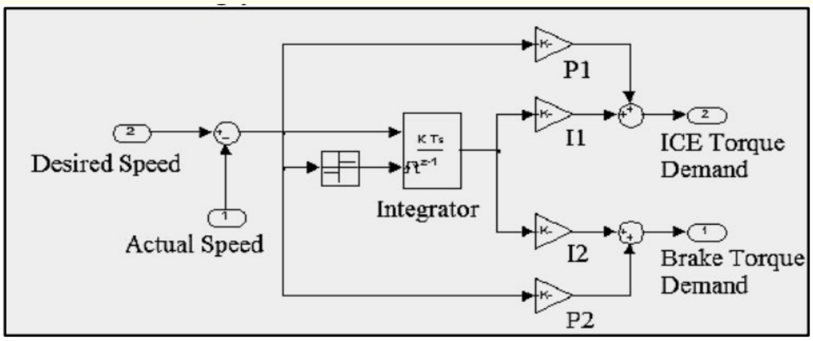

Figure 4: PID controller driver model.

Longitudinal vehicle dynamics

To develop an integrated model of the longitudinal dynamics for HEV, the detailed dynamics equations of the vehicle in motion are used. The state-space logic functions are developed to calculates vehicle rolling resistance, air drag resistance, gradient resistance, and acceleration resistance. These resistances provide the requirement of load torque which is to be fulfilled by powertrains (either by engine/electric motor alone or by the hybrid configuration of both the power sources; as per developed controller) to propel the vehicle.

The equilibrium equation of a force balance for vehicle longitudinal dynamics is represented as:

$$m\ddot{x} = \sum(F_{traction\ force} - F_{aero} - F_{rolling\ resistance} \pm F_{gradient}) \qquad (1)$$

Where $F_{traction\ force}$ is the longitudinal traction force at the wheels, F_{aero} is the equivalent longitudinal aerodynamic drag force, $F_{rolling\ resistance}$ is the force due to rolling resistance at the wheels, $F_{gradient}$ is the force due to gradient at the wheels, m is the equivalent mass of the vehicle, $\ddot{x}$ is the vehicle longitudinal acceleration.

Engine and transmission

The IC engine can be modeled 1D model or 3D detailed CFD model. However, 1D simulations are today expanding in the area of control system modeling towards Software in the Loop (SiL), which is nowadays becoming a popular activity in control system development before prototype hardware availability. In this research work, the 1D model of the engine is developed, coupled with the transmission. An alternate way of modeling the engine is by using maps, consisting of engine torque, speed characteristic curve along with BSFC maps (Fig. 5). The transmission consists of 1D lookup table presentation vehicle speed vs transmission gear ratios used.

The mass of fuel consumption is given as,

$$\dot{m}_f = f(T_{ICE}, N_{ICE}) \qquad (2)$$

Since the gear ratios are fixed and are determined by kinematic constraints, the power losses imply the reduction of the torque at the output shaft:

$$\begin{cases} \omega_{out} = \frac{\omega_{in}}{GR} \\ T_{out} = \eta GR T_{in} \end{cases} \qquad (3)$$

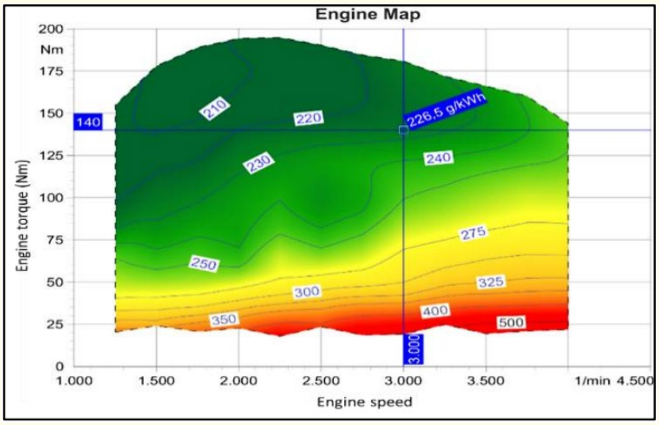

Figure 5: Engine Speed-Torque map.

Where ω is the revolution speed, T the torque, GR the transmission ratio, η the gear efficiency, and the subscripts in and out refer to the input and output shafts according to the power flow. The corresponding power loss is then calculated as [4]:

$$P_{loss} = \bar{\omega}_{in} T_{in}(1-\eta) \qquad (4)$$

Electric powertrain

The electric powertrain comprises of 3-phase induction motor model, a battery model, and an inverter model. The SoC of the battery is one of the constraints for the optimization of vehicle performance and fuel consumption.

$$P_{batt} = V_{oc} i_{batt} + R_{batt}(i_{batt})^2 \qquad (5)$$

$$SOC = SOC_0 - \int \frac{idt}{Q(i)} \qquad (6)$$

Where P_{batt} is battery power, V_{oc} is the open-circuit voltage at battery terminals, i_{batt} is a battery current, R_{batt} is resistance in the battery, SOC_0 is initial SOC of battery, and $Q(i)$ is battery capacity in Wh.

Torque coupling

Mechanical torque coupling is used to couple the torques of the engine and electric motor and provides a total torque to the driven wheels. But the speeds of the engine, motor, and vehicle are linked together with a fixed relationship and cannot be independently controlled because of the power conservation restraints.

Driver throttle command, actual vehicle speed, desired vehicle speed, or other longitudinal control criteria are used to determine the required total drive torque Td. This drive torque Td needs to be provided by a parallel combination of the engine and the motor through transmission is computed as:

$$T_d = \frac{1}{R}(T_{ICE} + T_{EM}) \qquad (7)$$

Where R is the transmission gear ratio, T_{EM} is the torque from the electric motor and T_{ICE} is the torque from the IC engine.

Final drive and wheel model

The final drive is modeled as a gear ratio multiplier, connected to the output of the torque coupling device. The wheel model contains a 3-DoF wheel model from MATLAB/Simulink library. This model takes input as speed and torque from the final drive. Wheel rotational speed is converted to vehicle linear speed considering the slip of the wheel.

$$\mu(\sigma) = D \sin(C \tan^{-1}(B\sigma - E(B\sigma - \tan^{-1}(B\sigma)))) \qquad (8)$$

The value of B, C, D, and E are constants and are dependent on different road types [9]. Computing the friction coefficient $\mu(\sigma)$ and the normal force Fz exerting on the wheel, the longitudinal traction force Fx [10] is then calculated as follows:

$$Fx = \mu(\sigma)Fz \qquad (9)$$

$$\text{Wheel slip, } \sigma = \frac{(r\omega - V)}{V} \qquad (10)$$

Supervisory Controller

There are different types of control strategies that need to be explicitly developed for the optimum usage of both the power plants. In this research work, to begin with, and as an initial iteration, PID based strategy is developed. Depending upon acceleration and deceleration(braking) commands from the driver model, power commands are generated for the respective power plants.

Results and discussions

The simulation was run for the Modified Indian drive cycle (MIDC) which represents vehicle speed profile with respect to time (Fig. 6). Appropriate step size is chosen with the appropriate solver to obtain the desired results. PID controllers are tuned accordingly to trace the desired and actual speed profiles.

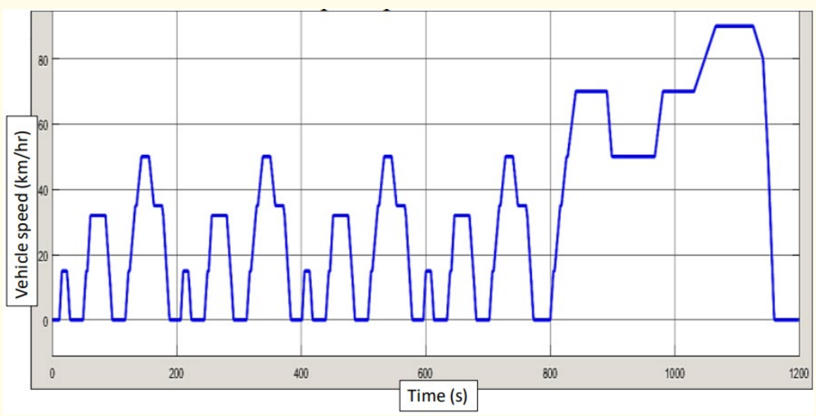

Figure 6: Modified Indian Drive Cycle (MIDC).

The requirement of load torque necessary to propel the vehicle is computed as shown in Fig. 7. This load torque requirement is satisfied by both the power plants by taking appropriate signals from the supervisory controller (Fig. 8).

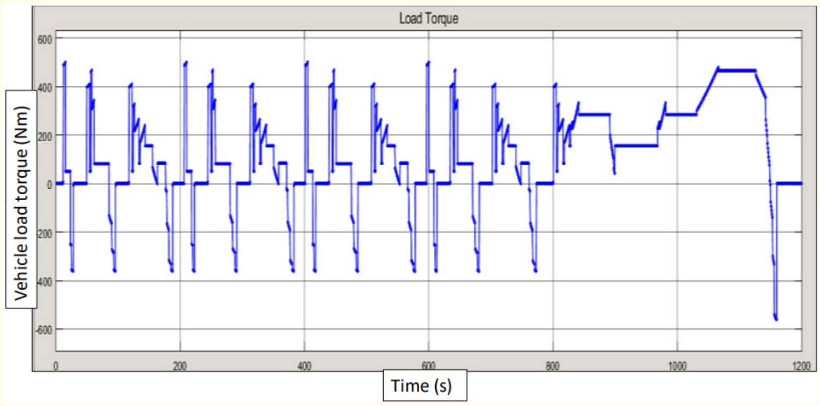

Figure 7: Load torque requirement.

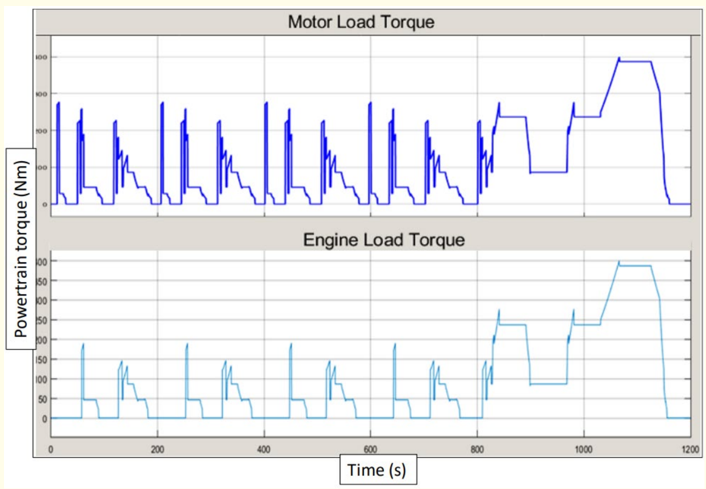

Figure 8: Variation of electric motor and IC engine torques with time.

Battery SoC while negotiating the driving cycle is represented in Fig. 9. The graph shows that regeneration taking place effectively by charging the battery whenever the vehicle is decelerated.

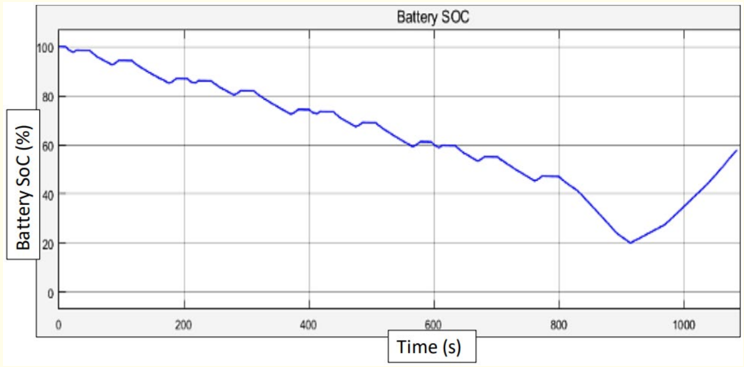

Figure 9: Variation of Battery SoC with time.

Conclusions

An efficient modeling and simulation technique facilitates fast, accurate, and cost-effective design and development of advanced power plants and vehicles. To endeavor the same, different modeling techniques can be followed, depending upon the simulation targets and accuracy requirements, ranging from the black box models to the highly dynamic, high fidelity models. Hence, it is very crucial to set the targets as clearly as possible for the simulation activity, which allows the selection of the most suitable modeling technique, in order to achieve the best compromise between computational requirements and the actual experimental results.

This investigation has given some insight into the possibility of using forward-backward facing models for the purpose of powertrain size optimization and getting an appropriate result from the same. Results have suggested that it is necessary to optimize the driver model by tuning the PID controller's inappropriate manners along with the powertrain components. The additional simulation time overhead imposed by forwardbackward facing models can also be reduced over time with ever-increasing computing power.

Therefore, the prospect of using forward-backward facing models for the purpose of powertrain component sizing is indeed achievable with acceptable simulation results.

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