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REAL WORLD CONTROLS CALIBRATION, TUNING, AND DATA CORRELATION FOR HYBRID THROUGH THE ROAD VEHICLE

Jeffery B Holtz
AVL Powertrain
Engineering Inc,
Plymouth, MI

Faisal J Uppal, PhD
AVL Powertrain
Engineering Inc,
Plymouth, MI

Andrew Cook
AVL Powertrain
Engineering Inc,
Plymouth, MI

ABSTRACT

A novel energy management strategy has been developed by AVL Powertrain Engineering, Inc. (PEI) that includes a unique power split and optimization approach and is used successfully for the FED BRAVO program. In this program, AVL is responsible for developing and delivering the full hybrid propulsion system integrated into the Fuel Efficient Demonstrator (FED) Bravo vehicle, designed by PRIMUS. This paper presents control system development and tuning using both simulations and vehicle testing carried out at multiple proving grounds. It summarizes important lessons learnt, in particular balancing fuel economy and drivability. It presents correlation results of AVL CRUISE simulations with data from vehicle testing at Chelsea Proving Grounds (CPG) and Aberdeen Proving Grounds (APG).

INTRODUCTION

This paper is a follow-up of [1, 2 & 3]. AVL has developed a unique energy management scheme for Parallel-Through The Road (TTR) Hybrid vehicles. This scheme includes a road load based power split and real-time optimization methodology. This methodology is applied to the PRIMUS FED BRAVO vehicle. This represents part of a continual effort by AVL using AVL's state of the art simulation and control tools such as AVL CRUISE, AVL BOOST, AVL DRIVE and AVL HYBRID DESIGN toolkit to design energy efficient vehicles. The main objective of these tools is to help designers strike the right balance between fuel economy, performance, emissions and drive quality.

The main goal of the algorithm development for FED Bravo is to improve fuel economy by optimizing the overall hybrid system efficiency while maintaining vehicle drivability and performance. The AVL-developed energy management algorithm calculates component energy availability and driver demanded torque, and manages the distribution of power between propulsion components. This includes a real-time road load calculated power split between the three propulsion sources, namely the Internal Combustion Engine (ICE), Integrated Starter Generator (ISG) and Front Motor (FMOT). Additionally, unique challenges of power split arose between the different propulsion sources due to the particular powertrain architecture selected for this vehicle, i.e. a combined through the road and parallel hybrid structure. For the optimization task, an objective function is constructed that reflects the overall potential power losses from the main powertrain components.

Content of this paper includes vehicle powertrain architecture, introduction to the road load based efficient power split strategy, control strategy updates, potential future improvements and a comparison of simulation and vehicle test results.

FED BRAVO POWERTRAIN AND CONTROLS ARCHITECTURE

An overview of the FED Bravo hybrid system layout and the main powertrain components from a systems viewpoint is illustrated in Figure 1. As mentioned before, there are three propulsion sources in this vehicle: ICE, ISG and FMOT. At the rear axle, the engine is coupled to the ISG via an electronically controlled clutch (engine disconnect clutch). The ISG is coupled to the rear differential via a six speed fully automatic torque converter-based transmission. A differential connects the transmission output shaft to the final drives. At the front, there is an electric motor directly coupled to the front differential through a two speed manual gearbox with pneumatic shift actuator. The differential connects to the final drives at the front axle. There are wheel end reduction units (WERU) at front and also at rear connecting to the wheels. The wheels enable the transfer of energy between the two axles together through the road.

A high power high capacity Li-Ion battery is used to supply power to the electric motors and other HV components, and to store the recuperated energy from the electric motors while regenerative braking or simulated engine braking. The AVL Hybrid Control Unit (HCU) coordinates and controls all system components as laid out in Figure 1, and the HCU is responsible for the power split and energy management functions.

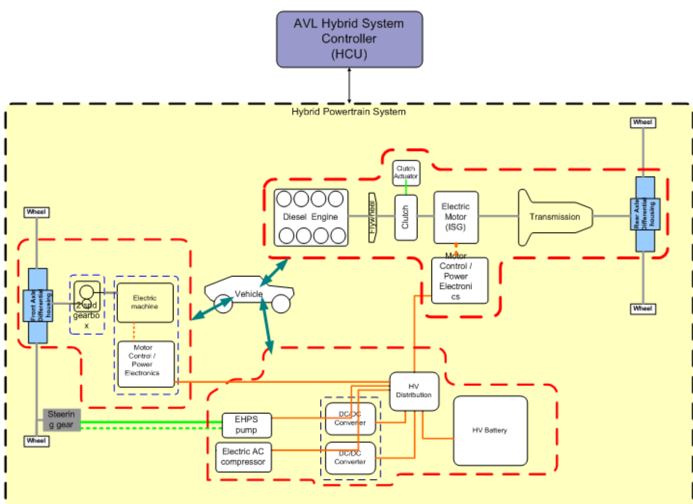

Figure 1: FED BRAVO Hybrid System Layout

The ICE and ISG constitute a parallel hybrid system, whereas the inclusion of the FMOT adds Through The Road (TTR) hybrid functionality. The main task of the energy management and control design is to utilize all three propulsion sources in the most fuel efficient manner while ensuring a minimal impact on performance characteristics.

A high level overview of main control tasks in the vehicle is outlined in Figure 2. These tasks consist of signal conditioning and powertrain management functions including driver demand calculation, torque management, safety limit monitoring and fault tolerance, component/local and system/global efficiency calculations, power split based on energy management, and real-time optimization.

The three main user selectable modes of powertrain operation are 'Engine Only', 'EV Only', and 'Hybrid'. There is a great emphasis of smooth transitions between these different modes under varying driving conditions.

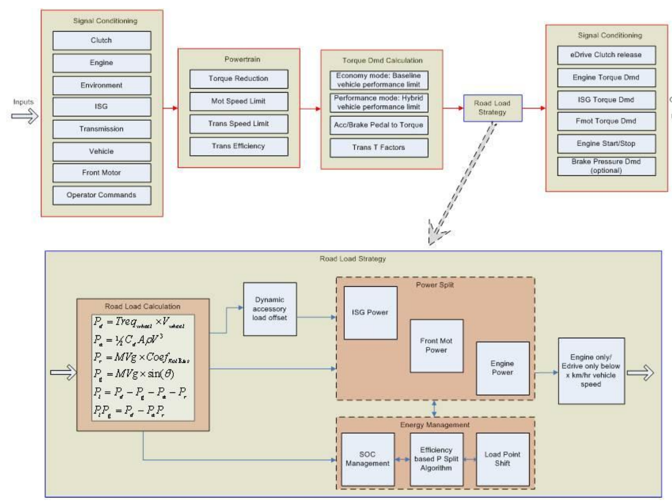

Figure 2: Control Strategy Overview

The following sections describe the three main components of the control strategy as shown in Figure 3. First there is a summary of the Torque Demand Calculation, second the Road Load based Power Split Strategy, and third the Powertrain/Energy Management Functions.

Torque Demand calculation

Depending on whether the user selects 'performance' or 'economy' mode, two separate methods are used for the torque demand calculation in forward transmission gears. The first is Full load based torque demand ($T_{dem\_FullLoad}$) and the second one is Road load based torque demand ($T_{dem\_RoadLoad}$). While 'performance' mode allows the vehicle to achieve the maximum power the physical hardware components are capable of providing, 'economy' mode

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limits the power at the wheels to the plausible power output of the existing HMMWV given inputs based on vehicle speed, acceleration pedal position, and brake pedal position. From this driver requested power, the power needed by the hybrid power pack is translated from the wheels through the driveline components, while considering the instantaneous efficiency of each driveline component.

$T_{dem\_FullLoad}$ is based on the full performance capability of the vehicle, whereas $T_{Dem\_RoadLoad}$ is essentially based on the max torque of the reference vehicle (a HMMWV in this case), taking into account a) Torque characteristics of the base 6.5L HMMWV engine, b) base HMMWV gear ratios of the 4 speed transmission, c) acceleration and brake pedal pressed for the hybrid vehicle, and d) Torque converter's pump and turbine characteristics for both locked/unlocked modes for the base vehicle. This is summarized in Eq. 1.

$$T_{dem\_FullLoad} = \begin{cases} \left(T_{\underset{eng}{\min}} + T_{\underset{isg}{\min}} + T_{\underset{fmot}{\min}}\right) Acc_{\underset{pos}{ped}}, & Br_{\underset{pos}{ped}} < \mu_b \\ \left(T_{\underset{eng}{\max}} + T_{\underset{isg}{\max}} + T_{\underset{fmot}{\max}}\right) Br_{pedpos} - T_{\underset{eng}{\min}}, & Br_{\underset{pos}{ped}} \geq \mu_b \end{cases} \quad (1)$$

INTRODUCTION TO ROAD LOAD BASED POWER SPLIT STRATEGY

Developed by AVL North America, AVL's Road Load Based Power Split Strategy for HEVs is a novel approach based on vehicle road loads that can be used to split the power demand between propulsion sources in a hybrid electric vehicle.

The main idea behind this approach is that slow varying or dynamically stable loads are supported by the engine which generally exhibits higher efficiency at these stable load conditions, whereas rapidly varying or transient loads are supported by the electric motors which are capable of higher efficiencies at these varying loads.

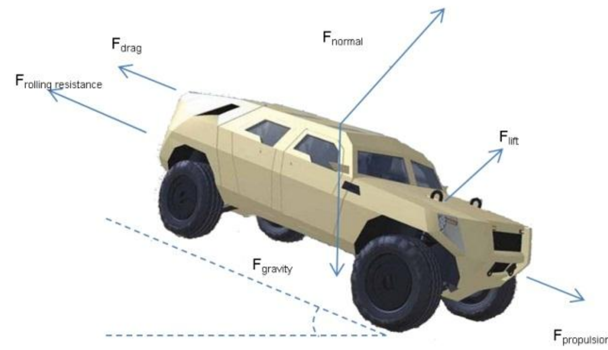

Figure 3: Forces acting on vehicle

To provide some background, the forces acting on a vehicle will be examined. As shown in Figure 3, six of these main forces are [4, 5]:

1) Rolling Resistance: F_r is a resistance force due to tire deformation in contact with the road surface:

$$F_r = C_{r1}v + C_{r2}F_N \quad (2)$$

C_{r1} and C_{r2} can be experimentally estimated and are usually provided by the tire manufacture.

2) Aerodynamic Drag: This is caused by the loss of momentum of air particles as air flows over the vehicle, and depends on the vehicle frontal area, shape, vehicle speed and air density.

$$F_a = \frac{1}{2}v^2 A_f C_d \rho \quad (3)$$

3) Aerodynamic Lift: Similar to an aircraft, aerodynamic lift is a force caused by different pressures between the top and bottom of the vehicle due to different air paths.

$$F_l = \frac{1}{2}v^2 A_l C_l \rho \quad (4)$$

where A_l is the area affecting the lift.

4) Gravitational force: The gravitational force can be decomposed into two components. The first is normal to the road surface, and the second is in the dimension of vehicle travel. The component of gravitational force in the dimension of vehicle movement is calculated from trigonometric relations as:

$$F_{g1} = \sin\{\alpha\} mg \quad \text{and} \quad \alpha = \tan^{-1}(G) = \tan^{-1}\left(\frac{c}{d}\right) \quad (5)$$

$$F_{g1} = \sin\left\{\tan^{-1}\left(\frac{c}{d}\right)\right\} mg$$

5) Normal force: The normal force is the force exerted by the road on the tires, the magnitude of which is equal to that of the gravitational force normal to the road.

$$F_N = F_{g2} - F_l = \cos\left\{\tan^{-1}\left(\frac{c}{d}\right)\right\} mg - \frac{1}{2} v^2 A_l C_l \rho \quad (6)$$

6) Propulsion force:

$$F_p = F_{isg\_w} + F_{fmot\_w} + F_{eng\_w} \quad (7)$$

where F_{isg_w}, F_{fmot_w} and F_{eng_w} are ISG, FMOT and ENG equivalent propulsion forces at the wheels.

Power Split Strategy

Considering the total power demand, for the vehicle to achieve a target speed v_t and ignoring the aerodynamic lift

$$P_d = P_i + P_g + P_a + P_r \quad (8)$$

$$P_i = Fvl_d = mavl_d = mvl_d \frac{dv}{dt} \quad (9)$$

$$P_g = \sin(\alpha)\, mgvl_d \quad (10)$$

$$P_a = \frac{1}{2} v^3 A_f C_d \rho l_d \quad (11)$$

$$P_r = mgvC_r l_d \quad (12)$$

$$P_d = mavl_d + \sin(\alpha)\, mgvl_d + \frac{1}{2} v^3 A_f C_d \rho l_d + mgvC_r l_d$$

$$= vl_d \left(m\frac{dv}{dt} + \sin\{\alpha\} mg + \frac{1}{2} v^2 A_f C_d \rho + mgC_r \right) \quad (13)$$

$$P_d = T_{Dem} \omega_{wheels} \quad (14)$$

where

P_d: Total computed power demand for the vehicle to overcome resistive forces and in order to maintain or achieve the target speed v_t

P_i: Power required to overcome inertia in order to achieve target speed (W)

P_g: Power required to overcome gravitational forces due to grade changes (W)

P_a: Power required to overcome air drag force (W)

P_r: Power required to overcome rolling resistance (W)

T_{Dem}: Calculated torque demand (Nm)

ω_{wheels}: Wheel speed (Rad/sec)

m: Equivalent mass of the vehicle

g: Standard gravity, g = 9.81m/s^2

F: Force required by the vehicle mass m, for acceleration a in order to achieve target speed (N)

v: Current vehicle velocity (m/s)

l_d: Vehicle driveline loss factor

G: Road grade, $G=c/d$, c: vertical distance and d: horizontal distance, α is the angle related to G

A_f: Vehicle equivalent frontal area (m) for drag computation

C_d: Vehicle drag coefficient

C_r: Vehicle rolling resistance coefficient or vehicle road coefficient

μ_b: Threshold for detecting brake pedal pressed

$$\alpha = \tan^{-1}(G) = \tan^{-1}\left(\frac{c}{d}\right)$$

ρ: Air mass density, $\rho = m_a/V_a$ where m_a is mass of air within test volume V_a. At 20°C and at 101kPa, the density of air is approximately 1.2041 kg/m^3.

It is interesting to note that the grade sensor is not needed for the engine and motor power split, as P_i+P_g can be computed as:

$$P_i + P_g = P_d - (P_a + P_r) \quad (15)$$

As mentioned earlier, in an attempt to minimize engine load variations or load transients, rapidly varying loads such as P_i and P_g are demanded from the electric motors whereas relatively slow varying loads such as P_a and P_r and demanded from the engine:

$$P_{mot} = \left| \left| P_i + P_g \right|_{L_{m-}}^{L_{m+}} \right|_{L_{b-}}^{L_{b+}} \quad (16)$$

$$P_{isg} = \left| P_{mot} - P_{accload} \right|_{L_{isg-}}^{L_{isg+}} \quad (17)$$

$$P_{fmot} = \left| P_{mot} - P_{isg} + \left| P_{accload} \right|_{L_{isg-}}^{L_{isg+}} \right|_{L_{fmot-}}^{L_{fmot+}} \quad (18)$$

$$P_{eng} = P_a + P_r + \left\{ \left\{ P_i + P_g \right\} - \left\{ P_{isg} + P_{fmot} \right\} \right\} \quad (19)$$

where $\{P_i + P_g\} - P_{mot}$ is the remaining part of $P_i + P_g$ (P_{mot} overflow) that the motors could not support because of either motor torque/current/temperature limits or battery current/temperature/SOC limits.

L_{m+}, L_{m-} are combined motors' positive and negative limits, peak or continuous.

L_{b+}, L_{b-} are battery positive and negative limits peak or continuous.

Objective Function for Optimization for an efficiency based motor power split

For the optimization task, an objective function is formed that reflects the overall potential power losses from the main powertrain components. This constitutes a minimization problem that requires evaluation over several iterations.

An objective function Ψ_t may be constructed which represents all main power losses in the powertrain system from minimization point of view.

$\Psi = f(\chi)$, $\chi = [0, 1/\eta, 2/\eta, \ldots \eta/\eta]$, η= number of equally spaced points that define the power split between ISG and FMOT. χ =0 corresponds to all of the motor power demand P_{mot} assigned to *ISG* and χ =1 corresponds to all motor power demand assigned to *FMOT.*

$$P_{isg} = \{P_{mot} \times [0, 1/\eta, 2/\eta, \ldots \eta/\eta]\} - P_{accload} \quad (20)$$

$$P_{fmot} = \{P_{mot} \times [\eta/\eta, \ldots, 2/\eta, 1/\eta, 0]\} \quad (21)$$

$$P_{eng} = P_d - [P_{isg} + P_{fmot}] \quad (22)$$

$$\Psi_t = |\Psi_{fmot}| + |\Psi_{isg}| + |\Psi_{eng}| + |\Psi_{fgbox}| + |\Psi_{rgbox}| \quad (23)$$

where $P_{accload}$ represents total high voltage load at the HV battery including export power, DCDC, HV hydraulic pump, HV air conditioning unit etc. Ψ_{fmot}, Ψ_{isg}, Ψ_{eng}, Ψ_{fgbox} and Ψ_{rgbox} are losses associated with the front motor, ISG, Engine, front gearbox and rear gearbox respectively and are calculated based on pre-determined efficiency maps stored in controller memory.

Minimizing Ψ_t wrt χ yields:

$$\chi_{opt} = \underset{\chi \epsilon [0,1/\eta,2/\eta,\ldots 1],\eta \epsilon \mathbb{I}}{\arg\min} \{\Psi_t(\chi)\}$$

$$= \underset{\chi \epsilon [0,1/\eta,2/\eta,\ldots 1],\eta \epsilon \mathbb{I}}{\arg\min} \{|\Psi_{fmot}| + |\Psi_{isg}| + |\Psi_{eng}| + |\Psi_{fgbox}| + |\Psi_{rgbox}|\} \quad (24)$$

such that

$$\|\chi - \chi_{opt}\| \leq \delta \text{ for } \delta > 0$$
$$\Psi_t(\chi_{opt}) \leq \Psi_t(\chi) \quad (25)$$

holds true; i.e. on some region around χ_{opt} all of the function values Ψ_t are greater than or equal to the value at that point (from standard form of an optimization problem) [7].

Note that χ_{opt} may be a local or global minimum depending on the objective function surface and the optimization algorithm used.

There are a large number of algorithms available for solving non-convex problems with some methods that are more complex but are better at finding global minimum than getting stuck at a local one. There are derivative based or search based methods.

In this application, for computational convenience and for dealing with possible discontinuity, the Nelder-Mead Simplex Method (Walsh, 1975) was used. The algorithm was initialized with a grid of uniformly spaced values. Nelder-Mead Simplex is an unconstrained non-linear optimization method. It is a non-gradient based direct search method which is generally less efficient for problems of higher orders but is more robust for problems which are highly discontinuous. It can be used to solve non-differentiable problems. However, this method may only give local solutions so it is important to start with a good initial estimation or a fine grid.

In order to avoid unnecessary switching between Front Motor and ISG or to avoid rapid changes or oscillations on the torque demands, a hysteresis loop is formed around power split changes. Calibrated threshold values are used for a minimum objective function change for the power shift to take place either from front to rear or from rear to front.

For implementing such an optimization in real time, it is important to stay well within the limits of processing requirements for the hardware target, while maintaining acceptable algorithm accuracy. That essentially means that in order to minimize the computational effort, a careful compromise is required between the number of iterations and the minimization goal.

Powertain Management/ Energy Optimization Management Scheme

Important functions of powertrain management/energy management optimization include SOC management, power split method selection, HV Accessory load offset, powertrain component safety limit management, powertrain mode management that decides when to allow any user demanded powertrain mode, manage idle speed regulation, compute torque and power factors, store and maintain component characteristics including efficiency tables and full load curves. Some of these functions are described in more detail subsequently.

SOC Management

One of the main functions of the energy management component is controlling the high voltage battery state-of-charge (SOC). SOC management aims to maximize vehicle fuel economy while maintaining SOC within safe and acceptable limits. Also, in order to maximize battery life and usable capacity for propulsion and regeneration, it is generally desirable to operate within tightly controlled bounds around the mid-range. However, in order to obtain a long EV range, a high initial SOC is required. In determining the compromise between these two objectives, a number of drive cycles were selected specifically for this vehicle's desired application and used during the simulations for determining this tradeoff.

To deal with this, SOC management upper and lower variable bounds are defined, within which the battery SOC is maintained. While always allowing maximum possible regeneration, the e-motor propulsion power limit is varied relative to the maximum and minimum allowable SOC bounds.

Two SOC States are defined and use different limits for the motor maximum power factor:

1. Charge Depletion State. In this state, higher power is allowed for propulsion, and as a result more battery power is utilized.

2. Charge Acquisition State. In this state less motor propulsion power is allowed so that battery can acquire and store charge from regeneration.

HV Accessory Load Offset

Accessory load offset is the amount of power demand that is offset from the ISG to the engine in order to take into account all the accessory-related electrical loads of the HV battery. This is computed taking into account a history of both high voltage load and load offset projected motor and battery efficiencies, battery capacity and motor limits, and a vehicle velocity dependent scaling factor.

Hybrid Controller Powertrain Manager (HCPM)

For reasons of safety, performance, and energy availability, the vehicle level powertrain management determines whether or not to a) allow 'Engine Only', 'EV Only' or 'Hybrid' powertrain modes, b) allow 4x4 propulsion mode and c) allow 'performance' mode.

Safety limit management includes maximum, minimum, and continuous motor, engine, transmission, and front gearbox torque and speed limits at current operating conditions, vehicle speed based limits, special limits in case of component warnings and component heat ups, and limits for special maneuvers like step climb and 60% grade.

HCPM also manages system idle speed regulation in different powertrain propulsion modes. The transmission requires a minimum input shaft speed in order to generate pressure for its operation. Idle speed is regulated by ISG, Engine, or both depending on the propulsion mode. In 'EV Only' mode it is regulated by the ISG as the engine disconnect clutch is disengaged. In 'Engine Only' mode the engine is used to regulate idle speed, and in 'Hybrid' mode, both engine and ISG are responsible for regulating idle speed. Special care has to be taken for ISG regeneration torque control near idle speed in order to avoid dipping below idle speed and stalling the motor. Smooth and controlled reduction in ISG regeneration torque close to idle speed is required otherwise torque oscillations or instability can occur.

HCPM computes Torque factors and Power factors based on transmission, differential, and gear reduction ratios and efficiencies.

HCPM stores and can also maintain efficiency tables for Transmission, Front Motor, ISG and Engine. It can compare the current component efficiency to that of pre-determined maps and report any inconsistencies. This information can be very helpful in diagnosing faults and issues in these components.

CONTROL STRATEGY UPDATES

Hybrid control strategy and engine management system were continuously updated for drivability, fuel economy, thermal management and engine/vehicle operation improvements. These updates were based on real world driving tests carried out on city style roads, country style roads, and moderate off road/ soft soil situations in both forward and reverse, moving up and down on grades up to +/-32%. These tests were carried out at Chelsea Proving Grounds (CPG, Chelsea MI) and Aberdeen Proving Grounds (APG, Aberdeen MD). The updates include both Hybrid Control Unit (HCU) and Engine Control Unit (ECU) input/output interface and control algorithm/strategy sections as well as calibration parameters and limits.

In order to minimize the development time and effort, simulation and vehicle tests on proving grounds were carried out in parallel. The same controls model was used in both the vehicle control unit in real-time and off-line simulations. Feedback from proving grounds, in tandem with simulation runs, made controls improvement much quicker, as most of the issues identified in the field could be replicated in simulation.

Some of the key updates/modifications based on track testing are listed below.

- HV Battery Control Updates
  - Ground Fault detection (GFD). Detection thresholds and timer constants.
  - HV filter boxes modified to reduce the overall Y-Cap values in the HV system.

- 4 x 4 Control
  - Power split ratios between front and rear axle and also between ISG and Engine updated.

- Cooling/Thermal Control
  - Cooling Strategy development
  - Fan control
  - Pump control
  - Battery Chiller/3-way valve Control
  - AC
  - Radiator Fan control

- DCDC12 Control
  - LV Battery charging

- HV Battery Stationary Charge
  - This is a new feature developed to maintain HV battery SOC while the vehicle is stationary. It includes engine start stop, HV battery charging, clutch slip detection, interface for engine speed control, ISG/Engine torque control, and SOC and stationary charge conditions monitoring.

- EHPS/Brake pump control
  - FED Bravo initially utilized a single HV electro-hydraulic pump for maintaining pressure for the correct operation of power steering and hydraulic brakes. This introduced excessive noise/ constant power drain and added complexity when considering vehicle startup and brake safety.
  - The hydraulic circuit was modified and a secondary LV pump was added only for the power steering.
  - Accumulators were used in the brake hydraulic circuit. The HV pump was only assigned to maintain pressure in these accumulators through intermittent operation. This was achieved through accumulator pressure feedback control.

- Clutch slip detection
  - Continuous clutch slip detection was implemented in all different powertrain modes.

- Engine Control
  - For improving engine cold start characteristics, glow plugs were added.
  - This needed new interfaces and updates to the engine start-up strategy.
  - Automatic selection of engine start mode was implemented based on HV battery SOC conditions. For low SOC of the HV battery, starter motor was used whereas for normal SOC ISG is used.

- Front Motor Control
  - Active motor speed protection development; critical when front two speed gearbox is in low gear.
  - Overall vehicle speed protection; critical when vehicle is travelling downhill.
  - Front motor regenerative braking tuning
  - Torque control improvements especially when the front GB is in low gear. This included rate limiting torque demand and further smoothing out startup torque in 4x4 mode.

- Powertrain Manager Updates
  - Engine, ISG and Front Motor torque and power limits, maximum and minimum, calculation updated
  - Powertrain Mode (‘Hybrid’/‘EV Only’/‘Engine Only’) selection
  - Powertrain enable conditions for safety that includes clutch slip detection
  - Traction mode control
  - Hybrid operation state machine updates
  - Over-speed/torque limit protection for Front Motor, ISG, Engine, Transmission Front gearbox). Imposing vehicle speed limits.
  - Powertrain stall protection at idle speed
  - HV Battery SOC management improvements

- HV Battery SOC based component disabling/imposing power/torque limits. For example AC selection by the user is disabled below 30% SOC. Also, In EV mode no propulsion torque is allowed below 20% SOC. This is to manage battery SOC and to prevent it from falling below critical levels.
- Electric launch assist development

- ABS/Slip Prevention Control
- This included a wheel slip monitor and reduction of applied torque in response to a slip/stability event.

- Drivability
  - Trade-off between launch torque and performance on grades

- HMI Control
  - Component power indicators
  - Powertrain mode selection
  - Traction mode selection
  - Diff-lock control
  - Ride height control
  - Energy flow indicators
  - Fuel economy/Miles to empty indicators

- HVJB control
  - HV bus discharge

- ISG Control
  - Interface with stationary charging strategy
  - Control of regenerative braking when ISG is approaching idle speed

- Torque Demand Calculation
  - Parameter updates e.g. ISG starter Torque
  - Interface with engine start mode
  - Interface with SOC management
  - Torque demand in forward gears (updates)
  - Torque demand in reverse gear (new feature)

- Power/Torque Split
  - Road load calculation and power split parameter updated
  - Accessory load offset control improved
  - Full load torque calculation update includes front motor limits and is based on powertrain modes.

- Transmission Control
  - Calibration/parameter updates

- Vehicle Control
  - Front gearbox gear switch enable for safety.
  - Rear-diff switch enable

- Input Interface
  - Analog input calibration
  - Two new analog channels added for monitoring pressure at two accumulators in the hydraulic brake circuit.
  - CAN updates
  - Digital in modifications for ride height, AC request by the user, front gearbox and rear-diff instrument panel interface

- Output Interface
  - Transmission of simulation parameters via CAN for control monitoring and power calculations
  - HMI Energy Flow calculations
  - HMI MPG and fuel rate monitors

- Input Signal Conditioning
  - Electrical load observer: Different methods for load estimation investigated. Parameters tuned to get smoother and reliable load value necessary for correct HV Battery SOC management.
  - Engine observer: Power/Torque limits, Engine 'Wait To Start Lamp' addition
  - Export power observer
  - Front Moor/ISG observer: Power/Torque limits calculation updated. Current motor efficiency calculation updated.
  - Vehicle observer: Engine glow plug control interface/ timers, Crank signal control updates

- Miscellaneous updates
  - Control/torque split improvements on rough terrain
  - 4x4 mode updates
  - Startup/shutdown sequences updates
  - Fuel rail pressure regulation
  - Engine turbo (VGT) tuning

FURTHER IMPROVEMENTS

Open source transmission calibration can further improve the fuel economy, drivability and performance for this vehicle. Examples include calibration of shift schedule and torque converter clutch engagement. Transmission engagement improvement may also be possible by reducing torque converter pressure at idle speed in order to eliminate rough engagement of the transmission from park.

Any hybrid prototype vehicle development involves a great deal of electrical signal probing for the purpose of testing and debugging. Future improvements could include further debugging considerations for the electrical wiring and connectors. Possible improvements could include use of electrical break out boxes for hybrid and engine controllers.

Future powertrain design and integration effort can be enhanced by an increased testing/validation of HV components and their interaction in the dyno or test lab prior to the vehicle integration.

Better functional enhancement of the engine separation clutch would allow better operation and blending of hybrid and EV modes.

If, for a future hybrid vehicle, the fuel economy is the main concern and a large EV range is not required, the HV battery system could be selected having a higher power density but smaller capacity. For maximum fuel economy and battery life, the HV battery SOC is currently maintained within a 10-20% of SOC band. Selecting a higher peak power battery could result in more braking energy being regenerated while minimizing battery weight and cost. There is also a possibility to improve power characteristics by using super capacitors if added complexity can be tolerated.

For functional improvement, an external Ground Fault Detection (GFD) module could be employed in addition to the GFD performed in the Battery Management System (BMS).

In order to minimize development effort and down time due to component failures, sufficient spare parts should be procured. Parallel development of two prototype vehicles instead of one can greatly improve the development process. On this vehicle there was some downtime due to mechanical component issues, air suspension system and HV/LV battery issues.

A COMPARISON OF SIMULATION AND VEHICLE TEST RESULTS

In order to ensure confidence in the predictive capability of the AVL CRUISE FED Bravo vehicle model, a correlation effort was necessary. By correlating the model, it can be definitively shown that operating the simulated vehicle over any combination of terrain and driving profiles will yield an overall fuel economy similar to that of the actual vehicle. This correlation provides predictive capability whereby the vehicle can be rated over a certain driving cycle in a virtual environment, and not require a driver, vehicle, or any of the associated overhead of real-world testing.

Previous simulation results, as reported in [3], show an overall 41- 92 % improvement over the baseline vehicle simulation that includes a 7 - 28% improvement due to hybridization.

For the correlation effort, the vehicle was instrumented to collect instantaneous fuel consumption data, velocity, and the operating conditions of every major component which contributed to fuel consumption and vehicle performance. With the vehicle fully instrumented, it was operated over several known drive cycles, traversing a known terrain profile for each, and continuously logging data. The resulting velocity/time/distance/terrain data was then imported into an AVL CRUISE model with the expectation that the simulated fuel consumption would align with what was seen in the real vehicle driving in the real world.

A visual aid for this process is illustrated in Figure 4. By controlling the tests and insuring that all vehicle and environmental conditions are met by the software, any differences between the simulated and observed fuel economy can be attributed to either noise, data collection error, or the misrepresentation of components and controls in the CRUISE space.

Figure 4: The vehicle model correlation process

To prove correlation, fuel consumption over the whole drive cycle must be accounted for and associated with each of the energy consumers. There is an infinite number of vehicle configurations and drive cycles which could all yield the same fuel economy as an end result, so in order to prove actual correlation, items like the battery SOC, electric motor torques and engine load should all line up for both the vehicle and simulation with respect to time, terrain, and distance traveled.

The vehicle and model were both run over a series of three drive cycles, each consisting entirely of either primary roads, secondary roads, or trails – henceforth referred to as the Primary, Secondary and Trails cycles – and for the purpose of proving this correlation methodology, the Secondary drive cycle has been taken as a detailed example. Figures 5a and 5b show the ISG torque/speed curve along with the efficiency contours and operating point clusters for both the vehicle test and model execution over the Secondary cycle. For starters it is clear that except for a few outliers which can be attributed to noise and the slightly different pedal responses of the vehicle driver and simulated driver, the operating points over the cycle are all clustered in about the same area for both cases. This indicates that the machine model spends about the same amount of time at the same operating points as the actual machine. The same can be said for the Engine as illustrated in Figures 6a and 6b.

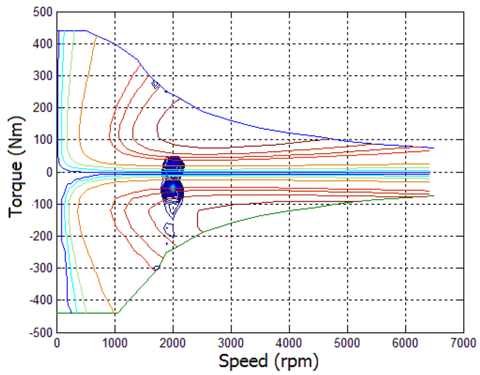

Figure 5a: ISG operating points (CRUISE)

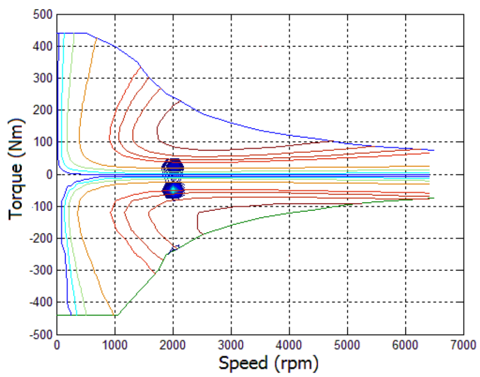

Figure 5b: ISG operating points (Vehicle)

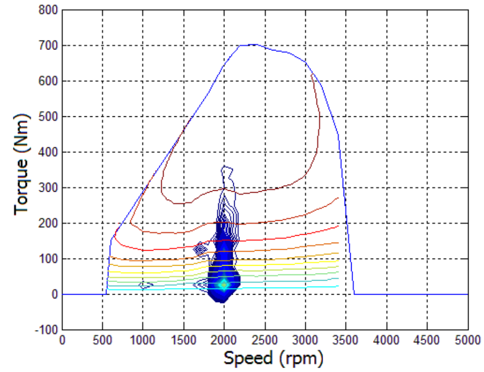

Figure 6a: Engine operating points (CRUISE)

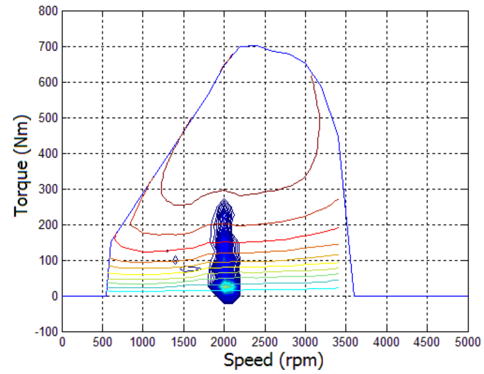

Figure 6b: Engine operating points (Vehicle)

The next logical step to the correlation effort is to observe the instantaneous torque for the important consumers over the whole cycle. For electric machines and diesel engines, torque is largely proportional to current draw and instantaneous fuel consumption with respect to shaft speed, and can therefore be used as a check on vehicle-level energy consumption over the cycle. Figures 7a and 7b illustrate the instantaneous torque over the Secondary cycle for both the ISG and diesel engine. Similar figures for the front-mounted electric motor, battery pack, and several other components were generated and used in the model validation process, but these have been omitted for the sake of brevity.

From the instantaneous ISG torque shown in Figure 7a it should be clear that, throughout the entire cycle, the model torque demand is nearly identical to that of the actual vehicle ISG, and the same can be said for the Engine torque in Figure 7b. Both vehicle and model are therefore shown to require the same energy to move the vehicle at a certain speed over a certain terrain, and it is apparent that this torque split is governed by similar control functions in both cases.

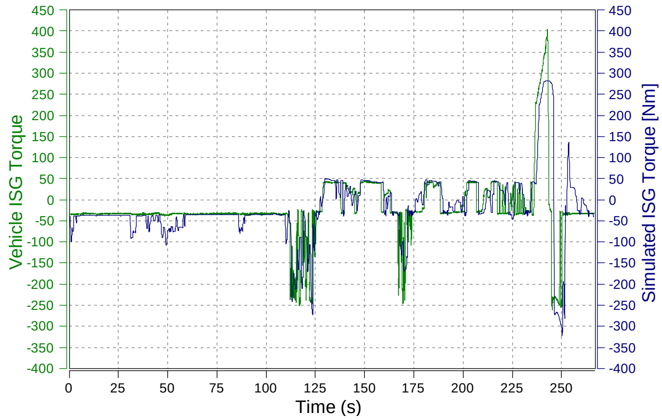

Figure 7a: Secondary Cycle ISG Torque/Speed

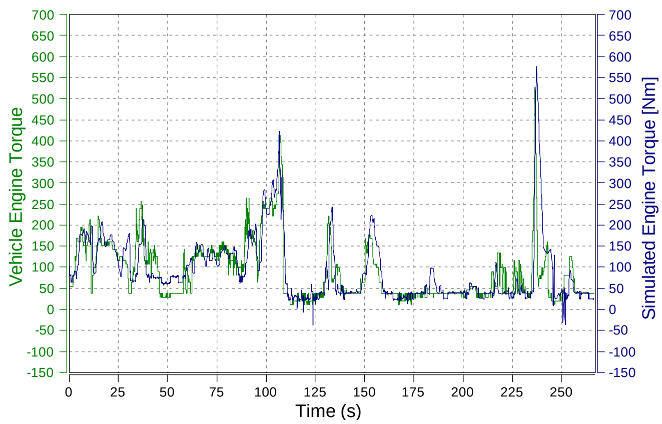

Figure 7b: Secondary Cycle Engine Torque/Speed

Finally, the integration of fuel consumption over time for the drive cycle is represented in Figure 8. Not only is the final value for total fuel consumed within 1.3% for both the vehicle and model, but identical paths were taken over the cycle. This final step of vehicle correlation ensures that the controls and component performance and efficiencies were modeled as accurately as necessary to have a representative vehicle model, and that this model can be operated over nearly any driving cycle to provide similar results to the real-world vehicle.

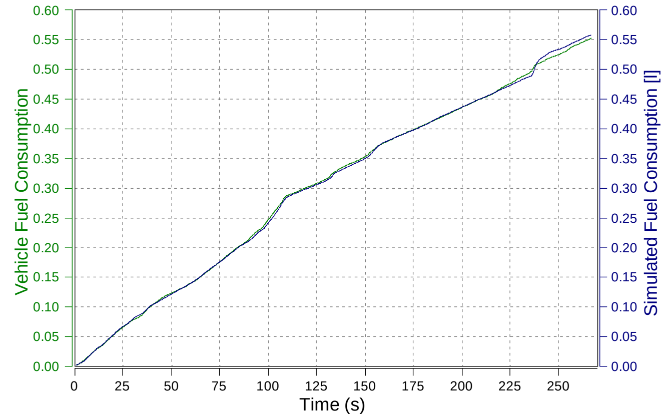

Figure 8: Fuel consumption correlation

The three cycles taken into consideration for this correlation were each distinct in their grade and terrain profiles, average and maximum speeds, and total cycle length. The intention of this variety was to add an extra level of confidence in the versatility and reliability of the model. The results of the model runs as compared to actual vehicle performance can be found in Table 1 below.

Drive Cycle	Model	Vehicle	Delta	Delta
Primary	12.78 mpg	13.09 mpg	0.31 mpg	2.4 %
Secondary	10.07 mpg	10.20 mpg	0.13 mpg	1.3 %

Table 1: Vehicle/Model FE Correlation Results

The CRUISE model was easily able to meet the ‘Primary’ and ‘Secondary’ drive cycles, following the same terrain and speed profile as the instrumented vehicle, and providing accelerator and brake pedal actuation similar to the actual human driver. Data collection for the ‘Trails’ cycle however, proved to be inadequate for use in the model validation process. The terrain was so rough and variable, and the inclination profile so aggressive, that it was not collected at a high enough fidelity to be useful as a CRUISE model input. Since a substantial portion of the vehicle drive energy is lost to crawling over the rough terrain and travelling up and down various inclinations, not having a good handle on this data made the correlation impossible. A future step of this effort will be to repeat the execution of this cycle with a greater attention paid to collecting the environmental data.

In all, the vehicle model has been shown to represent the actual vehicle very closely. Most correlation efforts are considered successful if the model reaches within 5-10% accuracy of actual vehicle performance, but since both the vehicle and vehicle model for this project were built in tandem, from scratch, the model has been found to be very highly representative.

CONCLUSIONS

This paper presents an efficient energy management strategy that includes a unique power split & energy distribution.

It presents the control strategy updates and further improvement potential based on the track testing performed at Chelsea and Aberdeen proving grounds.

A comparison of vehicle simulation and proving grounds test results is also shown. The fuel economy results from testing at APG correlate well with the simulation results and show that the modeling accuracy is within 3%.

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