

Forward-Looking Simulation of the GM Front-Wheel Drive Two-Mode Power-Split HEV Using a Dynamic Programming-Informed Equivalent Cost Minimization Strategy

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ABSTRACT

This paper presents a forward-looking simulation (FLS) approach for the front wheel drive (FWD) General Motors Allison Hybrid System II (GM AHS-II). The supervisory control approach is based on a dynamic programming-informed Equivalent Cost Minimization Strategy (ECMS). The controller development uses backward-looking simulations (BLS), which execute quickly by neglecting component transients while assuming exact adherence to a specified drive cycle. Since ECMS sometimes prescribes control strategies with rapid component transients, its efficacy remains unknown until these transients are modeled. This is addressed by porting the ECMS controller to a forward-looking simulation where component transients are modeled in high fidelity. Techniques of implementing the ECMS controller and commanding the various power plants in the GM AHS-II for FLS are discussed. It is shown that FLS-derived component states agree well with states commanded using the BLS-derived robust control strategy, with any difference being accounted for by transient effects. Fuel economy results from FLS decrease, as to be expected, by approximately 3-7% from that of BLS due to the increase in propulsion energy required by component transients. Overall, these two points of good agreement demonstrate the viability of the DP-informed ECMS as an online-implementable supervisory control strategy.

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INTRODUCTION

A hybrid electric vehicle (HEV) uses an internal combustion (IC) engine and one or more electric machines (EM) as power plants. IC engines have a narrow range of efficient operating conditions, typically under full throttle and mid-RPM operation. Inclusion of an EM in the powertrain avoids inefficient low power engine operations. Additionally, an EM supplements the IC engine in terms of its torque characteristics since an EM's full torque is available from near zero speed; this ultimately leads to sizing of smaller and more efficient engines. Lastly, the EM can recapture some of the vehicle's braking energy that would be lost otherwise. Efficient cooperation of the two types of power plants must be ensured to optimize the overall fuel usage of the powertrain.

There are three major types of HEV architectures: series, parallel and power-split [1,2]. A series HEV employs a full size engine to drive an electric generator which in turn

powers a traction motor. The engine-generator combination on the series architecture can always operate at its maximum efficiency point independent of wheel speed or torque requirement. Since only the traction motor drives the wheels, it must be full-sized. Additionally, the engine and the generator must be full-sized as well, unless the powertrain includes a large battery, to guarantee range and drivability of the series HEV. The aforementioned factors lead to higher cost of the powertrain and lower flexibility in EM sizing. Parallel HEVs use a full-sized or reduced-sized engine, augmented with an electric machine. Fuel economy (FE) improvement is achieved through downsizing of the engine and through regenerative braking. However, the engine is not decoupled from the wheels, therefore inefficient engine operations can still occur. The Power-split HEVs combine the features of the series and parallel architectures by using two EMs with a continuously variable transmission (CVT). Typically, the CVT is based on planetary gear (PG) sets with the gear ratio varied by one EM, while the other EM provides

extra traction torque. This type of transmissions is often termed an Electrically Variable Transmission (EVT) [3,4]. The most prominent example of this architecture is the Toyota Hybrid System II (THS-II) used in the Toyota Prius, and it uses one planetary gear set to realize the EVT function. While the THS-II powertrain is optimized for city driving, its highway performance suffers due to excessive power circulation loops at high speed driving [4]. On the other hand, General Motors developed a 2-mode EVT, the Allison Hybrid System II (AHS-II). The front-wheel drive (FWD) version of the 2-mode EVT utilizes two planetary gear sets and four clutches, yielding two power-split modes as well as four fixed gear modes [5,6]. The incorporation of two EVT modes allows optimization for both low speed and high speed driving, as well as reducing power requirement the electric machines [7]. However, Arata et. al. found that, with similar power plant parameters, the THS-II and AHS-II powertrains perform similarly in terms of FE [3].

The FE performance of an HEV depends heavily on the control strategy. Rule-based controllers produce good results and are easy to implement, but require extensive tuning of many parameters and rely on heuristics and engineering intuition [8,9,10]. Optimization based controllers provide a systematic approach to the HEV control problem, resulting in global optimal FE as in the case of dynamic programming (DP) [12,13], or near global optimal as in the case of the Equivalent Cost Minimization Strategy (ECMS) [3,11]. However, these controllers are not always online implementable due to the strategies being either non-causal (e.g., DP) or drive cycle specific (e.g., ECMS). Previous work addresses these problems by developing a robust ECMS controller with excellent cross drive cycle robustness against drive cycle and component specification variance [14]. Its FE performance is near the globally optimal solution. The efficacy of the controller has been verified with backward-looking simulations (BLS) on parallel and THS-II power-split architectures. Backward-looking simulations assume perfect cycle tracking and compute the steady-state energy or torque balance to arrive at component operation states. It is widely used in the literature for efficient control strategy verification and architecture comparison [3,11,12]. However, backward-looking simulations often neglect fast component transients and do not accurately represent the causality relationship in an actual vehicle, therefore they do not provide enough insight into the control strategy's viability for online implementation.

This work addresses these inadequacies by performing forward-looking simulations (FLS) on the GM AHS-II power-split architecture using the robust ECMS controller developed in previous work [14]. This will further verify the cross-platform applicability of the robust ECMS controller on a complex powertrain. Forward-looking simulations closely mimic the command structure of an actual vehicle in which a driver command ultimately results in a torque output at the wheels. Component dynamics are simulated in FLS, and the effects of component transients on FE will be analyzed. The

FLS controller uses component operation lookup maps generated by the backward-looking model. The effectiveness of this approach will also be discussed. Finally, this paper will give insight into the online implementation of an optimization-based controller on the GM AHS-II architecture, and demonstrate the efficacy of the robust ECMS controller.

The organization of the paper is as follows: first an overview of the GM FWD 2-mode architecture will be presented. The backward-looking simulation approach and application of the backward-looking derived control strategy in forward-looking simulation will then be described. Finally, simulation results from both the BLS and the FLS models will be compared.

GM AHS-II ARCHITECTURE OVERVIEW

In this section, an overview of the GM AHS-II architecture will be given. Operation and kinematics of the two EVT modes will be discussed. Bounds on both establish steady-state constraints needed in the backward-looking simulations.

Operation of the Planetary Gear Sets

The heart of the GM AHS-II architecture is two planetary gear sets. Unlike in the THS-II architecture where only one simple planetary is used, the GM AHS-II architecture uses one simple and one compound PG sets. Figure 1 depicts the simple planetary as used in the GM AHS-II. It is comprised of an outer ring gear and an inner sun gear and three pinion gears situated in the space between the ring and the sun gear. The three pinion gears are connected by a carrier arm (not shown in Fig. 1), which rotates about the same axis as the ring and sun gears.

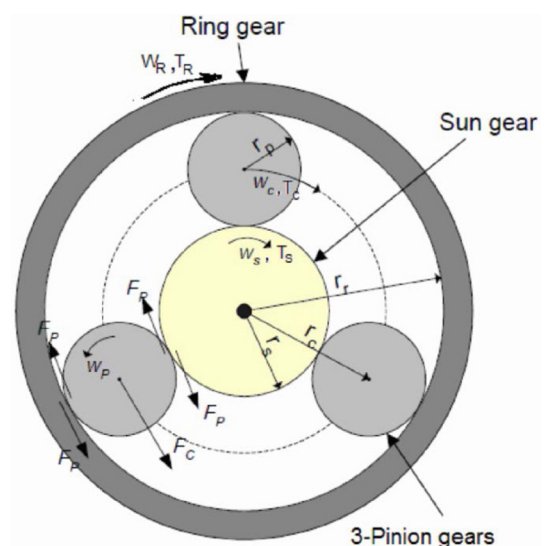


Figure 1. Simple planetary gear set, reproduced from [5]

Figure 2 depicts the compound planetary gear set used in the GM AHS-II. Three sets of two coupled pinion gears sit between the sun and ring gears. A carrier arm also joins the three sets of pinion gears and rotates about the same axis as the ring and the sun gears. The compound planetary and the simple planetary have similar, but different steady-state kinematic and torque relationships.

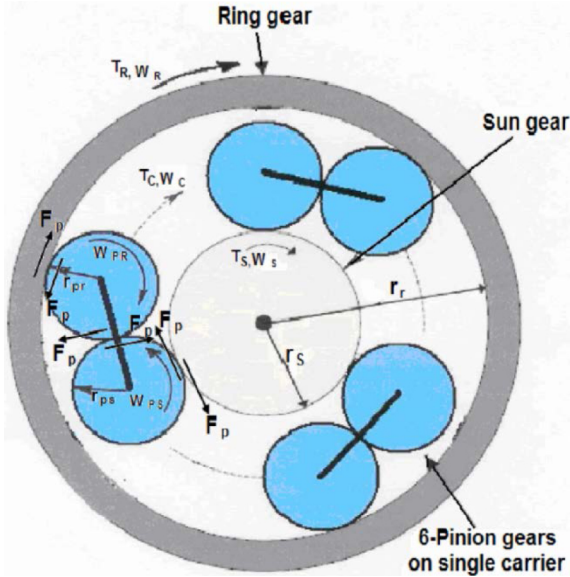


Figure 2. Compound planetary gear set, reproduced from [5]

Figure 3 depicts a schematic of the FWD GM AHS-II architecture and its arrangement of the planetary gear sets. The planetary PG1 is a compound planetary while PG2 is a simple planetary. The letters R, C, and S denote the ring, carrier and sun connection ports on a planetary gear set, while a succeeding number represents the respective planetary gear set. The two planetary gear sets and four clutches (CL) enable different modes of operation. The powertrain incorporates an IC engine and two electric machines (denoted as EMs) as power plants.

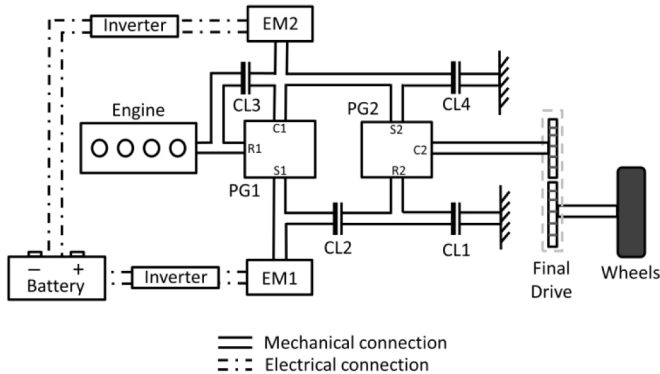


Figure 3. Schematic of the GM AHS-II architecture

Each planetary gear set has a characteristic ratio ρ defined as the ratio of the tooth-number of the sun gear to the tooth-

number of the ring gear. The characteristic ratios defined for the two planetary gear sets have the following relationship,

$$\rho_1 = \frac{N_{S1}}{N_{R1}} = \frac{r_{S1}}{r_{R1}} \quad (1)$$

$$\rho_2 = \frac{N_{S2}}{N_{R2}} = \frac{r_{S2}}{r_{R2}} \quad (2)$$

where N denotes the tooth-number and r the radius. Reference [5] provides the steady-state angular velocity and torque relationships in each of the planetary gear sets. The results will be summarized here. In summary, by assuming equal velocities at the contact points, the steady-state angular velocity relationships of the planetary gear sets become,

$$(1 - \rho_1)\omega_{C1} + \rho_1\omega_{S1} - \omega_{R1} = 0 \quad (3)$$

$$(1 + \rho_2)\omega_{C2} - \rho_2\omega_{S2} - \omega_{R2} = 0 \quad (4)$$

where ω denotes angular velocity. Note the differences in signs of the two equations. The steady state torque relationships arise from neglecting gear acceleration and performing an energy balance on a gear set, resulting in,

$$T_{R1} = -\frac{1}{\rho_1}T_{S1} = -\frac{1}{1-\rho_1}T_{C1} \quad (5)$$

$$T_{R2} = \frac{1}{\rho_2}T_{S2} = -\frac{1}{1+\rho_2}T_{C2} \quad (6)$$

In the AHS-II architecture, connection ports C1 and S2 on the two planetary gear sets are always rigidly connected. Port C2 is always connected to the final drive and its torque and angular velocity are typically the outputs of the EVT. Different modes of operations are enabled by engaging different combinations of the powertrain clutches. EVT-1 mode is activated by engaging only CL1, which grounds the R2 port to the frame of the vehicle. EVT-2 mode is activated by engaging only CL2, which connects port S1 and port R2.

Table 1. Clutch engagements for different operation modes

	CL1	CL2	CL3	CL4
EVT-1	X			
EVT-2		X		
FG-1	X		X	
FG-2	X	X		
FG-3		X	X	
FG-4		X		X

In addition to the EVT modes, there are four fixed gear (FG) modes, made possible by engaging two clutches at the same time. The fixed gear modes enable power transmission

characteristics similar to that of a conventional discrete gear automatic transmission. Furthermore, FG-2 mode uses both CL1 and CL2, and serves as the transition between the two EVT modes. This guarantees the transition between the two EVT modes occurs only when the angular velocity of the EM1 is zero. Table 1 shows the modes of operation and the required clutch engagements. Notice each fixed gear mode uses at least one of the clutches already utilized by either EVT-1 or EVT-2 mode. Therefore, the FG modes can be treated as subsets of the EVT modes with additional kinematic constraints. For this reason, the fuel economy simulations for AHS-II are performed using only EVT modes in this paper.

EVT-1 Kinematics

In EVT-1 mode, only CL1 engages and the powertrain is in its input split mode. Engine power splits at PG1, while PG2 acts as a simple gear reduction due to the grounding of its ring gear. Figure 4 depicts the GM AHS-II architecture in EVT-1 mode and its mechanical and electrical connections. In this mode, EM1 acts primarily as a generator while EM2 acts primarily as a motor delivering power to PG1 [3]. The constraints between the two planetary gear sets are $\omega_{C1} = \omega_{S2}$, $T_{C1} = T_{S2}$, and $\omega_{R2} = 0$. The resultant steady-state torque and angular velocity relationships in the powertrain are,

$$\omega_{EM1} = \frac{1}{\rho_1} \omega_e - \frac{(1-\rho_1)(1+\rho_2)}{\rho_1 \rho_2} \omega_{C2} \quad (7)$$

$$\omega_{EM2} = \frac{1+\rho_2}{\rho_2} \omega_{C2} \quad (8)$$

$$T_{EM1} = -\rho_1 T_e \quad (9)$$

$$T_{EM2} = -(1-\rho_1)T_e + \frac{\rho_2}{1+\rho_2} T_{C2} \quad (10)$$

where subscript e denotes the engine.

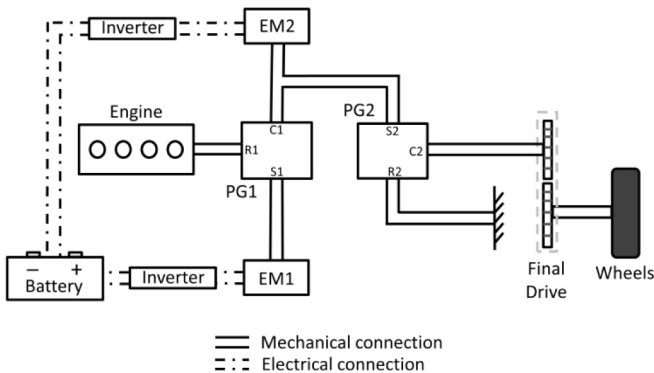


Figure 4. Schematic of the GM AHS-II architecture in EVT-1 mode

EVT-2 Kinematics

In EVT-2 mode, only CL2 engages and the powertrain is in its compound split mode. Engine power splits at PG1 and recombines with EM1 and EM2 power at PG2. Figure 5 depicts the GM AHS-II architecture in EVT-2 mode and its mechanical and electrical connections. The roles of the electric machines are reversed: EM-1 acts primarily as a motor while EM-2 acts primarily as a generator [3]. The constraints between the two planetary gear sets are $\omega_{C1} = \omega_{S2}$, $T_{C1} = T_{S2}$, $\omega_{S1} = \omega_{R2}$, and $T_{S1} = T_{R2}$. The resultant steady-state torque and angular velocity relationships in the powertrain are,

$$\omega_{EM1} = -\frac{\rho_2}{1-\rho_1-\rho_1\rho_2} \omega_e + \frac{(1-\rho_1)(1+\rho_2)}{1-\rho_1-\rho_1\rho_2} \omega_{C2} \quad (11)$$

$$\omega_{EM2} = -\frac{\rho_2}{1-\rho_1-\rho_1\rho_2} \omega_e + \frac{(1-\rho_1)(1+\rho_2)}{1-\rho_1-\rho_1\rho_2} \omega_{C2} \quad (12)$$

$$T_{EM1} = -\rho_1 T_e + \frac{\rho_2}{1+\rho_2} T_{C2} \quad (13)$$

$$T_{EM2} = -(1-\rho_1)T_e + \frac{\rho_2}{1+\rho_2} T_{C2} \quad (14)$$

Compared to EVT-1, EVT-2 mode allows lower engine speeds and avoids power circulation loops in the electrical path at high vehicle speeds. This helps maintain engine operation near its most efficient point, and maintains overall efficient power transmission through the EVT.

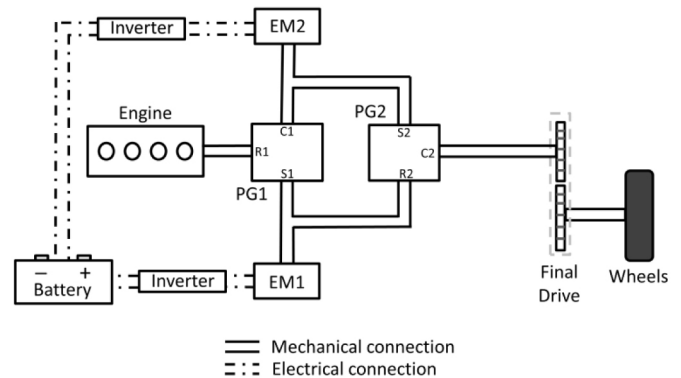


Figure 5. Schematic of the GM AHS-II architecture in EVT-2 mode

BACKWARD-LOOKING SIMULATIONS

The backward-looking simulations are conducted in MATLAB. This section will discuss the methodology of the backward-looking simulations before then developing the robust ECMS controller [14] for the GM AHS-II architecture.