

Design Optimization of Motor/Generator Full-load Characteristics in Two-mode Hybrid Vehicles

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ABSTRACT

An important benefit in adopting multi-mode hybrid powertrains is the possibility of downsizing the electric motor/generator (M/G) units. This paper discusses design optimization of M/G full-load operation envelopes. A three-dimensional operation space analysis is introduced and the full-load speed-torque envelopes of M/G units appear as constraints in the powertrain operation space. The analysis shows how the change in the M/G characteristics affects the powertrain full-load performance and how the M/G units can be downsized by combining multiple modes. When the acceleration performance of a two-mode hybrid vehicle is estimated using this analysis method, the trade-off between enhancing performance and reducing M/G full-load capacities is formulated as an optimization problem. The results show significant improvement in dynamic performance at reduced M/G cost.

INTRODUCTION

Power-split hybrid electric vehicles (HEVs) have been the most successful configurations since their pioneering debut in the marketplace. However, relatively low highway fuel efficiency and high manufacturing costs are drawbacks for this configuration. In an effort to overcome these deficiencies, manufacturers have developed advanced versions of the power-split HEV, referred to as multi-mode hybrid systems.

The multi-mode hybrid systems combine more than one power-split operation mode. For example, a system can use both input-split and compound-split modes; it can have additional parallel hybrid operation options, engineered to improve acceleration and cruise

performance. These systems also aim at improving fuel efficiency and reducing manufacturing costs. According to the previous research on the merits of these systems, the improvements in dynamic performance and the effects of M/G downsizing look far more promising than the increase in fuel efficiency [1, 2]. In particular, the requirements for M/G units may be greatly decreased, resulting in lower weight and cost.

This paper investigates these potential benefits of a two-mode hybrid system by focusing on the effects of the M/G full-load characteristics on the vehicle acceleration performance. To this end, two optimization frameworks are developed for powertrain operation and for component characteristics design, which will later be integrated in one problem.

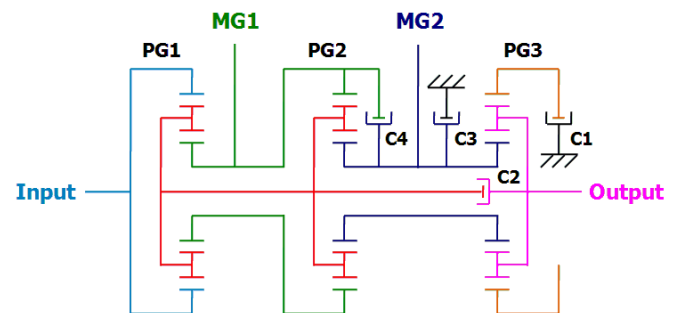


Fig. 1: Schematic of the two-mode hybrid powertrain

SYSTEM DESCRIPTION

A patented two-mode hybrid powertrain with fixed-gear options will be investigated in the analyses to follow [3]. Its schematic view is shown in Fig. 1. It uses three

planetary gear sets (PG1, PG2, and PG3) and four wet clutches (C1 through C4) to allow for six different operation modes: two electrically variable transmission (EVT) modes and four fixed-gear (FG) modes. Clutch operations and quasi-static equations of motion for each mode of the system have been detailed in the literature [1, 4, 5]. Here, a brief description of the system and generalized representations of the equations of motion are presented for facilitate the discussions in the following sections.

When only C1 is engaged, the transmission functions as an input-split device (EVT1), and PG3 acts merely as a single-ratio reduction gear. During this operation, C4 can be engaged to realize a parallel hybrid mechanism (FG1). As the vehicle velocity increases, the powertrain shifts modes from EVT1 to EVT2, which is a compound-split mode. FG2 is an inherent mode between the two EVT modes, and enables the synchronous shift. During this shift, C2 is engaged and C1 is released, minimizing a possible shock. In this mode, the ring gear of PG3 is free-wheeling, functioning neither as a power-split device nor as a single-ratio gear. The other two fixed-gear modes will not be discussed in this paper. (See references [1, 4, 5] for further details.)

Before presenting the generalized quasi-static equations of motion, the generalized representation of the power-split system is introduced [4, 6]. The lever diagram in Fig. 2 visualizes the relationships among elements by interpreting the rotational motions into linear ones. The representation can comprise all three power-split cases. The lever lengths, α and β , are the distances from the output to M/G units when the lever length of the input is normalized to 1.

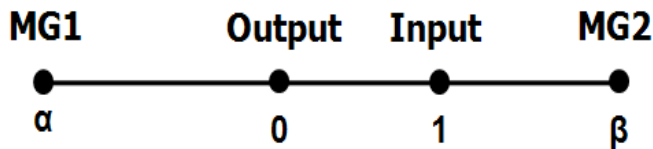


Fig. 2: Combined lever diagram

Using this representation, speeds and torques of M/G units are expressed with respect to those of the input and output in Table 1. In the case of EVT1, the output speed in the equations is not identical but proportional to the actual output speed by the reduction ratio of PG3. For FG modes, another set of generalized equations are given. There are four equations with respect to the eight component state variables in both cases. The last equation shared by EVT and FG modes is the generalized power conversion equation, which takes the inverse of the M/G conversion efficiency when the M/G works as a generator or the M/G power is negative. The generalized lever lengths for the two EVT modes, the reduction ratio for EVT1 mode, and the coefficients in the FG equations are given in Equations (1) to (6).

Table 1: Generalized quasi-static equations of motion

	<i>EVT</i>	<i>FG</i>
<i>Speed</i>	$\omega_1 = \alpha\omega_i + (1-\alpha)\omega_o$ $\omega_2 = \beta\omega_i + (1-\beta)\omega_o$	$\omega_i = k_i\omega_o$ $\omega_1 = k_1\omega_o$ $\omega_2 = k_2\omega_o$
<i>Torque</i>	$T_1 = \frac{1-\beta}{\beta-\alpha}T_i + \frac{\beta}{\beta-\alpha}T_o$ $T_2 = \frac{1-\alpha}{\alpha-\beta}T_i + \frac{\alpha}{\alpha-\beta}T_o$	$T_o = k_iT_i + k_1T_1 + k_2T_2$
<i>Power Conversion</i>	$P_b = \frac{\omega_1 T_1}{\eta_1(\omega_1, T_1)} + \frac{\omega_2 T_2}{\eta_2(\omega_2, T_2)}$	

$$\text{EVT1: } \alpha = \frac{1+r_2}{1-r_1r_2}, \quad \beta = 0, \quad \rho = \frac{1+r_3}{r_3} \quad (1)$$

$$\text{EVT2: } \alpha = -\frac{1}{r_1}, \quad \beta = \frac{1}{r_1r_2} \quad (2)$$

$$\text{FG1: } k_i = \frac{1+r_3}{r_3}, \quad k_1 = \frac{1+r_3}{r_3}, \quad k_2 = \frac{1+r_3}{r_3} \quad (3)$$

$$\text{FG2: } k_i = \frac{r_1r_2+r_3}{r_3}, \quad k_1 = \frac{r_3-r_2}{r_3}, \quad k_2 = \frac{1+r_3}{r_3} \quad (4)$$

$$\text{FG3: } k_i = 1, \quad k_1 = 1, \quad k_2 = 1 \quad (5)$$

$$\text{FG4: } k_i = 1-r_1r_2, \quad k_1 = 1+r_2, \quad k_2 = 0 \quad (6)$$

Here, P , T , and ω denote power (W), torque (Nm), and speed (rad/s) of each component, respectively, and η is the M/G conversion efficiency. α and β are the normalized locations of M/G units, and r_1 , r_2 , and r_3 are the planetary gear ratios of PG1, PG2, and PG3. ρ is the reduction ratio of PG3 in EVT1 mode, and k is the component speed ratio to the output in FG modes. 1, 2, i , o , and b are the subscripts for M/G1, M/G2, input, output, and battery, respectively.

POWERTRAIN FULL-LOAD OPERATION

The powertrain needs to be operated in such a way that the output shaft can transmit the maximum torque from the powertrain to the final drive. This can be regarded as an optimization problem, and is solved for the corresponding vehicle velocity. This optimization problem, however, is far more complicated than in conventional vehicles, where the engine exerts its wide-

open-throttle torque while the engine speed is determined by the vehicle velocity and transmission ratio. In two-mode hybrid powertrains, each mode has three degrees of freedom to determine the output torque, and each power component should be operated within its feasible operation domain or the full-load envelope. This can be formulated as either of two optimization problems depending on whether the operation is in EVT or FG mode. See Table 2.

The three independent operation variables are engine speed, engine torque, and M/G2 torque in the case of EVT modes, and engine, M/G1, and M/G2 torques for FG modes. The problem will be solved repeatedly, possibly several hundreds of times, along an output speed grid ranging from 0 to its maximum. The two cases will share the constraints on the component speeds, torques, and power, which will produce fairly different shapes in the respective three-dimensional design spaces.

Table 2: Optimization statements

	<i>EVT</i>	<i>FG</i>
Minimize	$\frac{1-\alpha}{\alpha}T_i + \frac{\beta-\alpha}{\alpha}T_2$	$-k_iT_i - k_1T_1 - k_2T_2$
With respect to	ω_i, T_i, T_2	T_i, T_1, T_2
Subject to	$\begin{aligned} \omega_{i,\min} &\leq \omega_i \leq \omega_{i,\max} \\ 0 &\leq T_i \leq T_{i,\max}(\omega_i) \\ \omega_{1,\min} &\leq \omega_1 \leq \omega_{1,\max} \\ T_{1,\min}(\omega_1) &\leq T_1 \leq T_{1,\max}(\omega_1) \\ \omega_{2,\min} &\leq \omega_2 \leq \omega_{2,\max} \\ T_{2,\min}(\omega_2) &\leq T_2 \leq T_{2,\max}(\omega_2) \\ P_{b,\min} &\leq P_b \leq P_{b,\max} \end{aligned}$	

DESIGN SPACE ANALYSIS

The optimization problems described in the previous section do not have as large a design space as other power management problems with tens or hundreds of variables. Furthermore, the problem size can be reduced further by applying dominance and monotonicity analysis in some of the cases. This kind of analysis of the design space will give us a better understanding of the relations and interactions among the components. Here, we analyze the three-dimensional design space of the operation in EVT2 mode, which is a compound-split mode and has the most complicated and nonlinear design space. The analysis in this section is presented with a given powertrain design, which is the result of the powertrain design optimization problem in the Results section. (See Table 4.)

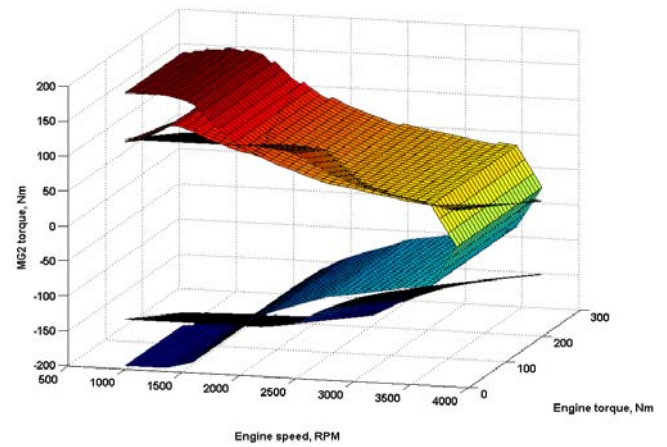


Fig. 3: Constraint surfaces in the design space

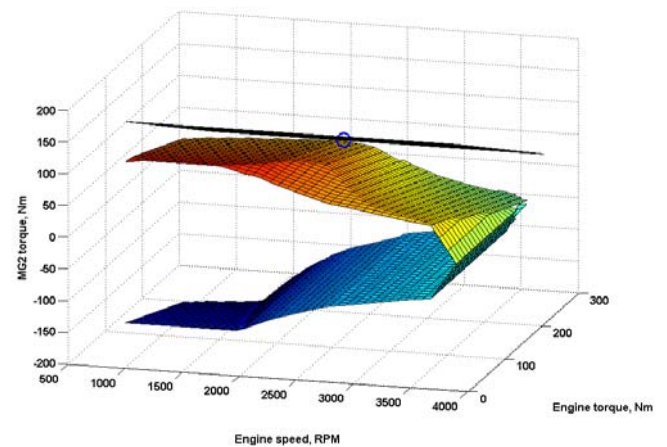


Fig. 4: Feasible domain and constant objective plane

Fig. 3 shows how some of the constraints appear in the design space. The plots are for a specific output speed (100 rad/s), and entirely different design space shapes can be observed for different output speeds. The upper three surfaces are the inequality constraints on M/G1 torque, M/G2 torque, and battery power, respectively. The lower two surfaces are derived from the other inequalities, but the constraint on minimum battery power does not appear because it is dominated by the others. The simple bound constraints would appear as vertical surfaces and are omitted for clarity.

The intersection set of all the constraints will form the feasible domain, which is shown in Fig. 4, together with a constant objective plane. As illustrated in the figure, the optimal solution will be found at the last point of contact between the plane and the feasible domain as the plane moves along the direction of minimization. This visualization shows that the active constraints are the M/G1 and engine torque upper limits, suggesting that increasing the components' torque capacities will improve the performance, at least for the corresponding output speed. Fig. 5 shows how the M/G1 upper torque limit constrains the maximum output torque as the vehicle accelerates.