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Towards a generic control-oriented model for HEV predictive energy management

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Abstract: The increased interest in model-based control techniques for the energy management of hybrid electric vehicles has led to a focus on generic modeling in order to allow an effective integration in model-based design procedures. An architecture-free model ensures modularity with a view to control design. Under mild assumptions, a generic model for the torque and the rotational speed of a hybrid electric vehicle is proposed in this article. In addition, the mathematical formulation of the fuel consumption minimization and the intrinsic problem of torque split are addressed. In order to illustrate its ability to handle the constraints, a model predictive control formulation is presented and applied to a dual-clutch transmission hybrid architecture.

Keywords: hybrid electric vehicle, generic model, energy management, MPC

1. INTRODUCTION

In recent years, stringent CO₂ emission regulations, as well as the depletion of fossil fuel resources have triggered a determined orientation toward alternative transportation technologies, such as hybrid electric vehicles (HEV).

For HEV, one of the most challenging problems concerning fuel consumption optimization is computation of the torque split between the engine and the motor. In the literature several techniques have been proposed: dynamic programming, which provides the global optimum, under the assumption of perfect information (Sundstrom et al., 2008), rule-based (Goerke et al., 2015), ECMS - Equivalent Consumption Minimization Strategy (Sciaretta et al., 2004) and its variations: A-ECMS (Musardo et al., 2005), T-ECMS (Sciarratta et al., 2004); fuzzy, genetic algorithms, game theory, deterministic Model Predictive Control (MPC) (Cairano et al., 2011), (Borhan et al., 2012), (Lu et al., 2013) and stochastic MPC (Ripaccioli et al., 2010), (Josevski and Abel, 2014). The ability to cope with constraints and also the possibility nowadays to collect information about the future speed and road profile have made the latter a very attractive method.

Model-based methods have drawn attention due to the inherent robustness of the solution, but they depend on the powertrain architecture, i.e. electric machine position and transmission type. In Rizzoni et al. (1999) a generalized model for HEV components and energy flow was proposed and in Chen et al. (2009) a global model for different vehicle types (conventional, hybrid, electric) was described, but the functional modes for each configuration were not detailed. The torque values at different levels (wheel, gearbox, crankshaft) and rotational speed of components are mandatory for a supervisory control and in the absence of a comprehensive parameterization, the control would have to be adapted for each particular configuration. In this paper, a generic control-oriented model is proposed, that will complete the HEV architecture description and will also make a contribution from a control point of view.

The improvements in fuel consumption by the use of telemetry data have been the object of several publications, such as (Kim et al., 2008) or (Manzie et al., 2012). In the former, a maximum headway of 400 m was tested, whereas in the latter the prediction was specified on a time headway, with optimal values found to be on the order of 30 s. However, within the use of the MPC method, smaller prediction horizon values are used, mostly below 20 s.

In this paper, the velocity profile and the gear engaged are assumed known in advance for a given time horizon and therefore, only the torque split problem will be addressed.

For a charge sustaining operation, a state-of-charge (SOC) balance condition is usually imposed; appropriate penalty factors can solve this problem, but they depend on the driving cycle characteristics. MPC final state constraints can lead to conservative results for short prediction horizons, as SOC varies in a small range. In this respect, the present paper proposes a new approach, that uses distance-varying limits for SOC, enforcing the convergence to the initial value, as the vehicle reaches its destination.

The article is organized as follows: first, a generic model with the associated assumptions, equations and configurations will be presented and validated on a case-study architecture. In the second part, the MPC - based energy
The proposed model aims to cover a large class of HEV architectures under the following assumptions:

(i) potential EM connected to the crankshaft, primary shaft, secondary shaft and to the wheel, as in Fig. 1
(ii) one battery
(iii) one gear-shaft EM (an EM to the either odd or even shaft, but not both - for a dual-clutch transmission). The notation conventions assign the shaft index 2 to the EM connected to the primary. If the EM is connected to the odd shaft, the indexes will be switched.
(iv) gear and clutch dynamic not considered
(v) gear efficiency dependent only on rotational speed

2. GENERIC HEV CONTROL-ORIENTED MODEL

Table 1 gives a description of these configurations, with EM connected to $\text{pos} \subseteq \{\text{crk}, \text{prim}, \text{sec}, \text{w}\}$. This implies that for $\text{pos} \notin \text{pos}, \text{rat}_{\text{pos}} = 0$.

<table>
<thead>
<tr>
<th>Configuration</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>$C_1 = 0, C_2 \in {0,1}$</td>
</tr>
<tr>
<td>2</td>
<td>$C_1, C_2 \in {0,1}, \text{prim} \in \text{pos}$</td>
</tr>
<tr>
<td>3</td>
<td>$C_1 = 0, C_2 = 1$</td>
</tr>
</tbody>
</table>

2.2 Torque and rotational speed expressions

From an energy management point of view, the torque expression at the wheel level includes the complete information about a HEV architecture (EM position and gearbox). For a detailed characterization, the equations below describe the relationship between the components torque and the torque delivered at the 4 shafts (crk, prim, sec, w) and also the rotational speeds, as functions of the vehicle speed ($v$) and $\omega_{\text{ICE}}$. 

$$
\begin{bmatrix}
T_f \\
T_g
\end{bmatrix} =
M_T
\begin{bmatrix}
T_{\text{ICE}} \\
T_{\text{mot}}
\end{bmatrix},
\begin{bmatrix}
\omega_{\text{ICE}} \\
\omega_{\text{mot}}
\end{bmatrix} =
M
\begin{bmatrix}
\omega_f \\
\omega_g
\end{bmatrix},
$$

(1)

where $f, g, \text{mot}$ stand for final driveline, generator and motor, respectively. In what follows, it will be shown that our proposed model integrates and extends (1).

2.1 Model configuration

As depicted in Fig. 1, the EM position determines at least 4 possible configurations ($\text{crk}, \text{prim}, \text{sec}, \text{w}$, but also combinations: $\text{crk}$ and $\text{w}$, $\text{crk}$ and $\text{prim/sec}$), whereas the clutch determines 3 possible configurations:

(1) one-clutch (EM everywhere), as in Fig. 3, top
(2) dual-clutch (particular case for EM connected to the primary shaft, as in Fig. 2)
(3) no clutch (EM everywhere), as in Fig. 3, bottom

2. GENERIC HEV CONTROL-ORIENTED MODEL

The proposed model aims to cover a large class of HEV architectures under the following assumptions:

(i) potential EM connected to the crankshaft, primary shaft, secondary shaft and to the wheel, as in Fig. 1
(ii) one battery
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(iv) gear and clutch dynamic not considered
(v) gear efficiency dependent only on rotational speed

NOTATIONS

- ICE - internal combustion engine
- EM - electric machine
- DCT - dual-clutch transmission
- $R_i$ - gear ratio engaged on $i$th shaft (includes neutral definition), $i \in \{1 : \text{odd}, 2 : \text{even}\}$
- $C_i$ - clutch status (0 - open, 1 - closed)
- $N_i = \min(R_i, 1)$ - used to define the case where one of the shafts is decoupled
- $FD_{R_i}$ - axle ratio corresponding to $i$th shaft
- $\text{crk, prim, sec, w}$ - crankshaft, primary, secondary and wheel, respectively
- $\text{rat}_{\text{pos}}_{\text{EM}}$ - ratio between the EM and the corresponding shaft where it is connected (pos - position)
- $\omega_{\text{ctrl}}_{\text{ICE}}$ - engine speed for a series architecture; for the others: idle speed or 0 rpm, in case of engine stop
- $R_w$ - wheel radius

First, a parallel architecture will be considered and subsequently it will be shown that the model can be employed for other architectures. Recall also that in Chapter 4 of Guzzella and Sciarretta (2007) the power-split case was treated and a very compact model was proposed in the form:

$$
\begin{bmatrix}
T_f \\
T_g
\end{bmatrix} =
M_T
\begin{bmatrix}
T_{\text{ICE}} \\
T_{\text{mot}}
\end{bmatrix},
\begin{bmatrix}
\omega_{\text{ICE}} \\
\omega_{\text{mot}}
\end{bmatrix} =
M
\begin{bmatrix}
\omega_f \\
\omega_g
\end{bmatrix},
$$

(1)

Fig. 1. Generic representation of a hybrid electric powertrain

Fig. 2. Configuration (2)

management problem will be formulated; the simulation results obtained and the conclusions complete the paper.

NOTATIONS

- ICE - internal combustion engine
- EM - electric machine
- DCT - dual-clutch transmission
- $R_i$ - gear ratio engaged on $i$th shaft (includes neutral definition), $i \in \{1 : \text{odd}, 2 : \text{even}\}$
- $C_i$ - clutch status (0 - open, 1 - closed)
- $N_i = \min(R_i, 1)$ - used to define the case where one of the shafts is decoupled
- $FD_{R_i}$ - axle ratio corresponding to $i$th shaft
- $\text{crk, prim, sec, w}$ - crankshaft, primary, secondary and wheel, respectively
- $\text{rat}_{\text{pos}}_{\text{EM}}$ - ratio between the EM and the corresponding shaft where it is connected (pos - position)
- $\omega_{\text{ctrl}}_{\text{ICE}}$ - engine speed for a series architecture; for the others: idle speed or 0 rpm, in case of engine stop
- $R_w$ - wheel radius

Fig. 3. Configuration (1) and (3): schematic representation
Note that the proposed model can also describe the series architecture. In this case, the engine is decoupled from the drivetrain and there are two EM: one for traction, and another that acts as a generator, to convert the engine mechanical output into electricity. From Fig. 1 this implies: 
C₁ = 0, C₂ = 0, the traction motor is EM² prim and the generator- EM² crk. The engine and generator speed will be exclusively defined by ω² ICE (r² ICE = 0).

The terms in matrices A₁ and A₂ will produce products between the 2 clutches in order to appropriately integrate special use-cases of a DCT, as detailed in Table 3. For instance, (6) defines the ratio between the torque at primary and at crankshaft and it is the arithmetical expression for the logical OR operation. Equation (10) is detailed in Table 2, which is well defined for rad² EM ≠ 0. For a DCT, if both clutches are closed, the EM will run at the same speed as the ICE, the operating point being defined by the vehicle speed and the gear engaged on the odd shaft (R₁).

The condition that ensures the engine is disconnected from the drive (its speed being therefore defined by ω² ICE) is C₁ + N₂C₂ = 0. This covers the cases where both clutches are open and also the charge at standstill mode (C₁ = 0, C₂ = 1, N₂ = 0). The latter is defined for an architecture with one EM to the primary shaft and involves decoupling the engine from the wheel (N₁ = 0) and closing the clutch between the engine and the EM² prim (C₂ = 1).

In the case of a DCT, C₁ will be set to zero, according to assumption (iii) from section 2. Thus, the EM rotational speed will be equal to ω² ctrl and this is defined in the ω² EM expression by the term C₂(1 - C₁)(1 - N₂).

### 2.3 Case-study: hybrid architecture with a DCT

In this section, a DCT hybrid architecture is considered, with one EM connected to the primary shaft. This configuration allows both clutches to be simultaneously closed in some particular cases: take-off and charge during driving, in parallel mode (the ICE and EM run at the same speed). In this case, the even shaft must be decoupled from the wheel (R₂ = 0). The Table 3 summarizes the use-cases with respect to clutch states and even shaft (coupled/d decoupled).

### Table 3. Hybrid DCT functional modes

<table>
<thead>
<tr>
<th>C₁</th>
<th>C₂</th>
<th>r² EM</th>
<th>r² prim</th>
<th>Configuration</th>
</tr>
</thead>
<tbody>
<tr>
<td>0</td>
<td>0</td>
<td>R₂</td>
<td>1, 2</td>
<td></td>
</tr>
<tr>
<td>0</td>
<td>1</td>
<td>R₂</td>
<td>1, 2, 3</td>
<td></td>
</tr>
<tr>
<td>1</td>
<td>0</td>
<td>R₂</td>
<td>2</td>
<td></td>
</tr>
<tr>
<td>1</td>
<td>1</td>
<td>R₁</td>
<td>2</td>
<td></td>
</tr>
</tbody>
</table>

The charge during driving case can appear under 2 forms: torque split and parallel mode, as presented in Fig. 4. In

### Table 2. EM to primary shaft, (10) description

<table>
<thead>
<tr>
<th>C₁</th>
<th>C₂</th>
<th>r² EM</th>
<th>r² prim</th>
<th>Configuration</th>
</tr>
</thead>
<tbody>
<tr>
<td>0</td>
<td>0</td>
<td>R₂</td>
<td>1</td>
<td></td>
</tr>
<tr>
<td>0</td>
<td>1</td>
<td>R₂</td>
<td>1, 2</td>
<td></td>
</tr>
<tr>
<td>1</td>
<td>0</td>
<td>R₂</td>
<td>2</td>
<td></td>
</tr>
<tr>
<td>1</td>
<td>1</td>
<td>R₁</td>
<td>2</td>
<td></td>
</tr>
</tbody>
</table>

The charge during driving case can appear under 2 forms: torque split and parallel mode, as presented in Fig. 4.
3. ENERGY MANAGEMENT - PROBLEM FORMULATION AND RESULTS

In this section, the energy management problem will be addressed and it will be shown how the model introduced in the previous section can be exploited for its formulation. The main objective for the design point of view is the fuel consumption minimization which requires a torque distribution between the ICE and the EM. Fuel consumption is given by a nonlinear static map, as a function of engine torque and speed, but for control design, an analytical approximation is employed. In de Jager et al. (2013) a convex model is proposed for the consumption of an SI engine:

\[ P_f (\omega, P_p) = \gamma_{p,0} + \gamma_{p,1} P_p + \gamma_{2,2} P_p^2, \gamma_{2,2} \geq 0. \]  

(15)

where \( P_p \) is the engine power. In the present paper, 3 alternative polynomial formulations (16a)-(16c) are introduced, which are compared to an approximation directly resulting from (15). The coefficients \( p_i, i = 0 \ldots 2 \) are interpolated values and \( \tilde{m}_f \) is the instantaneous fuel consumption:

\[ \tilde{m}_f = p_1 (\omega ICE) T_{ICE} + p_0 (\omega ICE) \]  

(16a)

\[ \tilde{m}_f = p_2 (\omega ICE) T_{ICE} + p_1 (\omega ICE) T_{ICE} + p_0 (\omega ICE) \]  

(16b)

\[ \tilde{m}_f = p_2 (T_{ICE}) \omega_{ICE}^2 + p_1 (T_{ICE}) \omega_{ICE} + p_0 (T_{ICE}) \]  

(16c)

\[ \tilde{m}_f = \gamma_2 P_p^2 + \gamma_1 P_p + \gamma_0 \omega_{ICE}^2 + \gamma_0,2 \omega_{ICE} + \gamma_0,0 \]  

(16d)

Fuel consumption minimization favors an explicit relation in engine torque, as in (16a), (16b) or (16d). However, these approximations are less accurate than relation (16c), as can be seen in Table 4 where the RMSE (Root Mean Square Error) associated to validation data are summarized for a SI and CI engine, respectively, and also in Table 5, where the RMSE was calculated for different driving cycles (for the CI engine, only data for NEDC are available). The polynomial (16c) was therefore used, as it was validated with the best results for all the cases, with the remark that for the CI engine the polynomials in torque are also accurate.

As can be seen in Fig. 5, the torque dependence of the polynomial parameters in (16c) is highly nonlinear, but a piecewise linear approximation is possible (the curves in red) whose accuracy is quantified in the last column of Table 4. This approximation leads to a simplified version of (16c):

\[ \tilde{m}_f = \left( a_1^{(i)} \omega_{ICE}^2 + a_0^{(i)} \right) T_{ICE} \]  

(17)

where \( p_j = a_0^{(i)} T_{ICE} + b_0^{(i)}, j = 0 \ldots 2 \) and considering the expression for the fuel consumption variation is given by (13). Another quadratic criterion is currently under study, where the electrical consumption is expressed by the difference between the electrochemical power and its minimal value, defined by constraints. This new criterion takes into account cases where \( P_r \) is negative.

For a charge-sustaining operation mode, a particular attention must be paid to the SOC balance. This could be ensured by choosing appropriate values for the penalty factor \( \lambda \), but it is cycle-dependent (which is in turn characterized by speed profile and total duration). In this paper, distance-varying min-max limits for SOC were used:

\[ SOC_{min}(k) = SOC_0 - (SOC_0 - SOC_{min}) e^{-\frac{k}{T_{dist}}} \]  

(19)

\[ SOC_{max}(k) = SOC_0 + (SOC_{max} - SOC_0) e^{-\frac{k}{T_{dist}}} \]  

(20)
The problem is reformulated as a quadratic programming, the trajectory (SOC allowing it a wide range of variation in the first part of

The model complexity is given by the time-variance of

A Linear Time Varying (LTV) system is obtained:

\[ x_{k+1} = A_k x_k + B_k u_k + D_k \]  

where \( x = \text{SOC}, A_k = 1, u_k = T_{ICE}(k) \) and \( D_k \) is a residual term due to linearization. It can be noticed that \( A_k \) is constant, which reflects the integral behavior of SOC. The model complexity is given by the time-variance of \( B_k \) and \( D_k \), which depend on EM rotational speed and torque demand. After linear algebra manipulations, we have:

\[ X_k = \Phi_k x_k + \Psi_k U_k + V_k \]  

\[ \Phi_k(i) = \text{ones}(N,1), \Psi_k(i,j) = \begin{cases} B_{k+j-1} & j \leq i \\ 0 & \text{else} \end{cases} \]  

\[ V_k^T = \begin{bmatrix} D_k & D_k + D_{k+1} & \ldots & \sum_{i=1}^{N} D_{k+i-1} \end{bmatrix} \]  

where \( x_k \) is the SOC value at instant \( k \) and \( X_k, U_k \) the vectors of future states and commands, respectively.

As in (19), \( \Delta \text{SOC} \) has to appear in an explicit form:

\[ \Delta X_k = D_\Delta (\Phi_k x_k + \Psi_k U_k + V_k) + X_{k0} \]  

\[ X_{k0} = \begin{bmatrix} x_k \\ 0 \end{bmatrix}, \quad D_\Delta (i,j) = \begin{cases} -1 & i = j \\ 1 & i = j + 1 \\ 0 & \text{else} \end{cases} \]  

From (17), we have:

\[ \bar{m}_f = \alpha(\omega_{ICE}) T_{ICE} + \beta(\omega_{ICE}) \]  

Therefore, the optimization criterion (18) is reduced to a quadratic formulation as in (22), where the positivity requirement is always verified:

\[ H = \Psi_k^T D_\Delta^T Q D_\Delta \Psi_k + \bar{\alpha}^2 \]  

\[ F = \Psi_k^T D_\Delta^T Q \left( D_\Delta (\Phi_k x_k + V_k) + X_{k0} \right) + \bar{\alpha} \bar{\beta} \]  

with \( \bar{\alpha} = \text{diag}(\alpha(\omega_{ICE,k+1-1})), \beta(i) = \beta(\omega_{ICE,k+i-1}) \) and \( Q = \text{diag} \left( \lambda \left( \frac{1}{\eta_{ICE} \text{SOC}} \right)^2 \right) \). Future vehicle reference speed and gear engaged are considered known for the next \( N \) s and therefore, from (3) and (5) we can determine the rotational speeds for ICE and EM:

\[ \omega_{ICE} = \frac{r_{ICE}^w}{R_w} \frac{v}{R_w} + (1 - C_1 - N_2 C_2) \omega_{ICE}^{\text{prim}} \]  

\[ \omega_{EM} = \frac{r_{EM}}{R_w} \frac{v}{R_w} + r_{at} \omega_{EM}^{\text{prim}} C_2 (1 - C_1) (1 - N_2) \omega_{ICE}^{\text{prim}} \]  

The rotational speeds establish the torque and power constraints for each prediction step that can be easily
expressed from (2), whereas SOC constraints (20) are imposed from (24a):

\[
T_{\text{ICE}} = \frac{T_w - r_{EM}^w T_{EM}}{r_{\text{ICE}}} \tag{29a}
\]

\[
T_{\text{ICE}}^{\text{min}}(\omega_{\text{ICE}}) \leq T_{\text{ICE}} \leq T_{\text{ICE}}^{\text{max}}(\omega_{\text{ICE}}) \tag{29b}
\]

\[
T_{\text{EM}}^{\text{min}}(\omega_{\text{EM}}) \leq T_{\text{EM}} \leq T_{\text{EM}}^{\text{max}}(\omega_{\text{EM}}) \tag{29c}
\]

\[
P_{\text{EM}}^{\text{min}}(\omega_{\text{EM}}) \leq \frac{\pi}{30} \omega_{\text{EM}} T_{\text{EM}} \leq P_{\text{EM}}^{\text{max}}(\omega_{\text{EM}}) \tag{29d}
\]

\[
SOC_{\text{min}}^{(k)} - \epsilon_k \leq X_k \leq SOC_{\text{max}}^{(k)} + \epsilon_k \tag{29e}
\]

where the superscript \( \text{prim} \) was neglected, due to the presence of a single EM and \( \epsilon_k \) are slack variables. All the studied driving cycles have power requirements achievable with the conventional mode (ICE only), so torque and power limitations can be imposed as hard constraints, without affecting the recursive feasibility of the receding horizon optimization. On the contrary, in the case of distance-varying SOC limitations, infeasibility might occur, especially toward the reset distance, where there is a narrow range of variation. These limitations are therefore included as soft constraints.

### 3.2 Simulation and results

The control law was tested under Matlab/Simulink and the behavior of the vehicle was simulated by a high-fidelity model, designed in AMEsim. The total distance was considered known \textit{a priori} and a prediction horizon \( N = 5s \) was used. The results are summarized in Table 6, where the comparison is made with the consumption in conventional mode; in the case of a final SOC value different from the initial one, a consumption correction was performed. For all the scenarios the control parameters are the same. It can be seen that the hybrid vehicle with a MPC strategy offers a consumption gain up to 26%.

<table>
<thead>
<tr>
<th>Cycle</th>
<th>Conventional</th>
<th>Hybrid</th>
<th>Gain</th>
</tr>
</thead>
<tbody>
<tr>
<td>ARTEMIS road</td>
<td>5.8</td>
<td>5.32</td>
<td>8.2%</td>
</tr>
<tr>
<td>ARTEMIS urban</td>
<td>9.38</td>
<td>6.9</td>
<td>26.4%</td>
</tr>
<tr>
<td>ARTEMIS highway</td>
<td>8.2</td>
<td>8.01</td>
<td>2.3%</td>
</tr>
<tr>
<td>FTP-75</td>
<td>6.22</td>
<td>5.28</td>
<td>15.1%</td>
</tr>
<tr>
<td>NEDC</td>
<td>5.76</td>
<td>5.01</td>
<td>13%</td>
</tr>
</tbody>
</table>

### 4. CONCLUSIONS AND OUTLOOK

A generic parameterization under certain assumptions has been proposed for a hybrid powertrain, with detailed expressions for series and parallel architectures. For the energy management part, an MPC - based strategy has been presented and validated in simulation (Matlab and AMEsim) on a hybrid DCT case study. Current simulations show that encouraging results can be obtained with a MPC strategy. The quadratic formulation is still under investigation. In addition, a sensitivity analysis with respect to driver behavior and to tuning parameters (prediction horizon, penalty factor, reset distance), as well as a stability analysis will be the object of a future study.

### REFERENCES


