

Benchmarking Hybrid Concepts: On-Line vs. Off-Line Fuel Economy Optimization for Different Hybrid Architectures

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ABSTRACT

The recent advance in the development of various hybrid vehicle technologies comes along with the need of establishing optimal energy management strategies, in order to minimize both fuel economy and pollutant emissions, while taking into account an increasing number of state and control variables, depending on the adopted hybrid architecture.

One of the objectives of this research was to establish benchmarking performance, in terms of fuel economy, for real time on-board management strategies, such as ECMS (Equivalent Consumption Minimization Strategy), whose structure has been implemented in a SIMULINK model for different hybrid vehicle concepts. The results obtained from these simulations have then been compared with those derived from a general purpose, off-line optimization technique, based on deterministic DP (Dynamic Programming), and capable of finding the numerical global optimum and of generating the optimal cycle-based control trajectory over a discretized multidimensional grid of the selected state and control variables. The paper investigates the structure of the DP problem and its interactions with the specific hybrid architecture, especially in terms of the most appropriate selection of state and control variables. The implications of the chosen modeling approach are also critically evaluated, searching for the best compromise between accurate simulation results and reliable comparisons between off-line and on-line optimization results.

One of the outcomes is that the system model should be designed in order to be compatible with efficient DP techniques implementation, with the objective of obtaining robust optimal control policies while achieving acceptable computational costs.

The concepts that have been analyzed in this work are the following two parallel hybrid architectures: HEV (Hybrid Electric Vehicle), normally applied in current hybrid vehicles production, and HSF-HV (High Speed Flywheel Hybrid Vehicle), an interesting and promising hybrid mechanical solution. An example of the influence of the selected gear has also been investigated by implementing a multi-dimensional DP optimization routine. In order to perform this analysis, a general purpose DP MATLAB function, including specifically designed algorithms to avoid numerical interpolation issues that typically occur in constrained problems, has been modified to run any SIMULINK-based engine-vehicle model.

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INTRODUCTION

The continuously increasing interest on fuel economy improvement, together with pollutant emissions reduction are nowadays the most important drivers of the automotive market, due to upcoming legislations. Hybrid vehicle

technologies certainly play a key role within this context, gathering remarkable benefits both in the fields of the internal combustion engine operation, due to downsizing and load point shifting, and vehicle kinetic energy recuperation due to regenerative braking.

Different hybrid powertrain solutions are currently studied, and in some cases already available on the market. They basically differ for the propulsion topology, classified as parallel or serial, for the secondary energy source nature, e.g. electrical energy in a battery, kinetic in a flywheel, oil or gas pressure in hydraulic or pneumatic accumulators, and for the hybrid propeller size, namely full or mild. The potential Fuel Economy (FE) performance of electric, kinetic, hydraulic and pneumatic hybrid concepts were compared with respect to a common baseline conventional vehicle in [1, 2], considering a parallel configuration. The purpose of that study was to evaluate the fuel reduction potential of different hybrid solutions, with respect to a conventional thermal engine vehicle, adopting a version of the ECMS algorithm (Equivalent Consumption Minimization Strategy), as on-line optimization supervisor for the Energy Management Strategy (EMS). Various versions of the ECMS algorithm can be found in literature, as reported in [3], mainly characterized by different criteria for the equivalence factor $s(t)$ calculation along the prescribed mission. Particularly notable results can be achieved by adopting procedures based on future driving conditions prediction, as demonstrated in [4], or simply on the signal feedback of secondary energy source state of charge (SOC), as reported in [1, [5] and [6].

From the latter considerations, it immediately arises the importance of determining the benchmark FE value that each hybrid concept can achieve, both to obtain an unbiased assessment of the investigated solutions, and to fairly evaluate the potential of the selected real-time EMS over different homologation cycles. For this purpose, it is necessary to employ a method capable of calculating the optimal control policies that an ideal supervisor Hybrid Control Unit (HCU) would apply. Given the nature of the analyzed control problem, characterized by a prescribed finite time horizon (i.e. the driving cycle) and by systems that can be modeled due to a limited number of state and control variables, the implementation of deterministic Dynamic Programming (DP, see also [7] and [8]) looks feasible, as explained in [9], and has been applied in several recent studies.

The simulation models that allowed the comparative study carried out in [1] with an ECMS strategy approach were built in IAV VeloDyn® environment, shown in Figure (1), a straightforward and general purpose modular longitudinal dynamic vehicle simulator, comprising the behavior of a driver. The outcomes of such simulation tools had already been validated experimentally on an engine test bench, through software-in-the-loop implementation of the hybrid propulsion system (e.g. Electric Motor/Generator EMG and battery). As explained in the following sections, the application of DP algorithms requires a simpler powertrain model to reduce the computational cost of the overall procedure. Moreover, since its calculation proceeds backwards in time, it must not rely on information about past events (i.e. no driver model). As a consequence, a quasi-static vehicle modeling technique, as described in [10], has been

adopted for the aim of this research. Such kind of model for the powertrain behavior is widely adopted for fuel consumption-based analysis, and is based on the assumption that the vehicle will exactly follow the prescribed speed profile. That permits a direct calculation of the car accelerations and, once the selected gear is provided, it allows a backward-facing reconstruction of the instantaneous mechanical power demanded to the (hybrid) propulsion system. Throughout this paper, all the presented results involving quasi-static simulation tools have to be regarded as validated with respect to the equivalent original straightforward VeloDyn® ones. The DP control strategies to optimally drive the EMS of such models were computed via a general purpose MATLAB® function, published by ETH Zurich and presented in [11], which implements a vectorized version of the basic DP algorithm to improve calculation speed. Although this solution substantially reduces the computational cost of the overall offline optimization, it also requires some caution when dealing with SIMULINK® blocks that don't accept multi-dimensional inputs, e.g. Lookup tables.

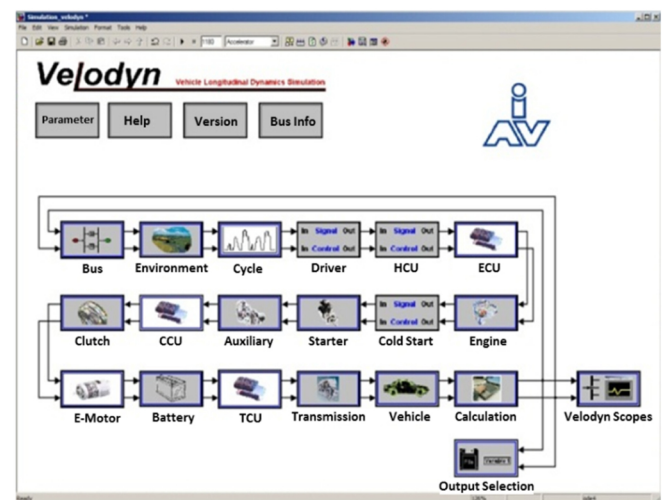


Figure 1. IAV VeloDyn® simulation environment.

Among the four previously mentioned hybrid vehicle concepts, only the two considered more appealing were selected and submitted to the comparison with the optimal solution from DP, that is the Hybrid Electric Vehicle (HEV) and the Mechanical Hybrid Vehicle, named HSF-HV (High Speed Flywheel Hybrid Vehicle). On one hand, the electric one is widely recognized as being the most technologically well-established, as its presence on the market from several years confirms. On the other hand the mechanical concept, whose main disadvantage is represented by the limited energy storage capacity, has other interesting features. As highlighted in the topological study carried out in [12], it can be considered a very competitive technology in terms of trade-off between increased FE and investment cost. Moreover, it is not affected by well-known problems concerning batteries for automotive use, that is ageing, life

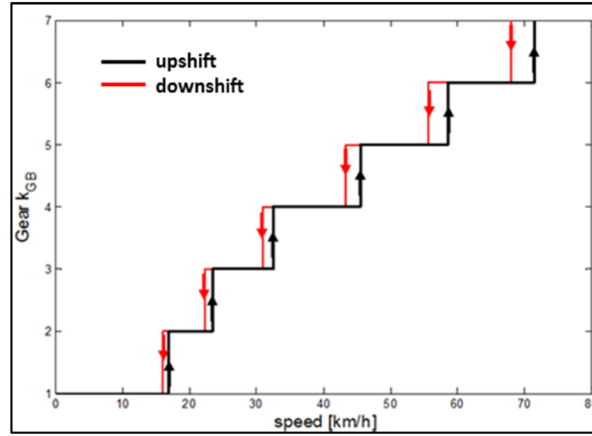


Figure 2. AMT Gear-shifting policy example.

length, safety and ecological issues related to their disposal. Further, the energetic roundtrip does not involve low energy conversion efficiencies, since the only present source is the mechanical one, in the form of kinetic energy. Finally, a benchmark for the innovative mechanical hybrid topology evaluated in [1] and first introduced in [13] is considered a very useful tool to assess its actual potential performances.

Interesting aspects regarding the DP-based modeling of the HSF-HV, which is a more complex system than the HEV from a control-oriented point of view, will be discussed throughout this paper.

HYBRID VEHICLES MODELING

This section provides a detailed description of the main equations governing the hybrid vehicle system modeling, both for the HEV and the HSF-HV concepts. As previously mentioned, the following modeling approach belongs to quasi-static (backward-facing) classification, so that it can be a useful tool to compare fuel consumption for the same system with different control strategies. However, it neglects basic features regarding vehicle longitudinal dynamics, shaft elasticity effects, damping phenomena and others. Those characteristics are not supposed to substantially affect the reliability of the provided performance indicators, regarding relative values of fuel economy, which are the core of this research. The fundamental vehicle specifications are reported in Table 5 in the Appendix section.

Common Path

The HEV and HSF-HV architectures can be considered equivalent to a certain extent, both being full parallel hybrids, and only distinguished by the nature of the secondary energy source, so that they include different components, i.e. the EMG and Li-Ion Battery for the HEV and HSF clutch, Continuously Variable Transmission (CVT), final gear and HSF, with vacuum chamber housing for the HSF-HV. At each time instant t , the vehicle speed $v(t)$ and acceleration $\dot{v}(t)$ are supposed to be exactly known, as well as the road

gradient α , while the transmission ratio τ_{gb} of the Automated Manual Transmission (AMT) is a consequence of a predetermined gear shift-pattern, where the gear k_{gb} is only dependent on the vehicle speed. An example of such shift policy is provided in Figure (2).

The requested traction effort at the wheel F_t can then be computed as the sum of inertial force F_i , aerodynamic drag resistance F_a , rolling resistance F_r and gravitational force F_g , so that:

$$F_t = F_i + F_a + F_r + F_g \quad (1)$$

where:

$$\begin{aligned} F_i &= m_v \cdot \dot{v}(t) \\ F_a &= \frac{1}{2} \rho_a A_f c_x v^2(t) \\ F_r &= \left(\sum_{n=0}^4 c_{r,n} v^n(t) \right) \cdot m_v g \cos(\alpha) \\ F_g &= m_v g \sin(\alpha) \end{aligned} \quad (2)$$

In the previous equations, the term m_v is the vehicle mass, ρ_a the air density, c_x the aerodynamic drag coefficient, $c_{r,n}$ the rolling friction parameters, g the gravitational acceleration and A_f the cross-sectional frontal area. The torque T_{WH} , as well as the wheel angular speed ω_{WH} and acceleration $\dot{\omega}_{WH}$ required at the wheel can then be calculated as follows:

$$\begin{aligned} \omega_{WH} &= \frac{v(t)}{r_{WH}} \\ \dot{\omega}_{WH} &= \frac{\dot{v}(t)}{r_{WH}} \\ T_{WH} &= \begin{cases} F_t \cdot r_{WH} + J_{WH} \cdot \dot{\omega}_{WH} & (v > 0) \\ 0 & (v = 0) \end{cases} \end{aligned} \quad (3)$$

where r_{WH} represents the effective rolling wheel radius, J_{WH} the wheels' inertia. The requested torque T_{Req} at the gearbox primary shaft can then be calculated, including the final drive

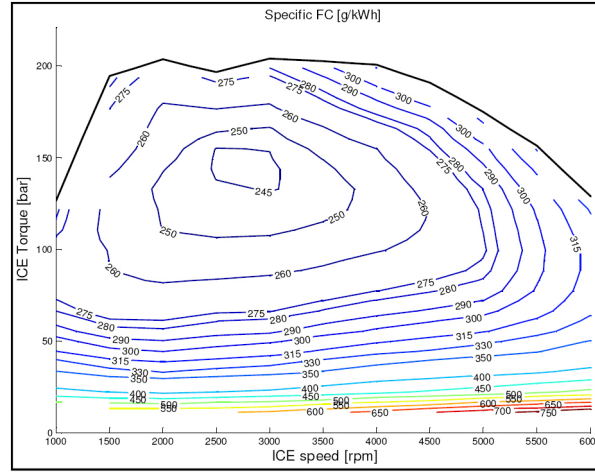


Figure 3. Map-based representation of the Internal Combustion Engine (ICE) efficiency.

ratio in the AMT transmission ratios τ_{gb} . The mechanical losses are computed due to an efficiency η_{gb} map as a function of the selected gear, taking into account the power flow direction.

$$\begin{aligned}\omega_{gb} &= \tau_{gb}(k_{gb}) \cdot \omega_{WH} \\ \dot{\omega}_{gb} &= \tau_{gb}(k_{gb}) \cdot \dot{\omega}_{WH} \\ T_{Req} &= \frac{T_{WH}}{\tau_{gb}(k_{gb}) \cdot \eta_{gb}^{sign(T_{WH})}}\end{aligned}\quad (4)$$

The requested torque is considered positive during traction operation and negative during braking. The gearbox primary shaft can be modeled as the node at which the torques coming from the hybrid propulsion system T_{Hyb} and from the ICE T_{ICE} converge, so that $T_{Req} = T_{Hyb} + T_{ICE}$ and the core control variable for the EMS, the split factor $u(t)$, is defined:

$$u(t) = \frac{T_{Hyb}}{T_{Req}} \quad (5)$$

The general notation T_{Hyb} will be replaced by the torque requested from the EMG T_{EMG} for the HEV and by T_{HSF} for the mechanical hybrid vehicle. The ICE clutch slipping dynamics is not modeled, as it is considered an on/off element, so that if ϕ_e is that state of the engine clutch, where 0 is open and 1 is engaged, the split factor u defines its state, along with the ICE speed ω_{ICE} as follows:

$$\begin{aligned}\phi_e &= \begin{cases} 1, & \text{if } u < 1 \\ 0, & \text{if } u = 1 \end{cases} \\ (\omega_{ICE}, \dot{\omega}_{ICE}) &= \begin{cases} (\omega_{gb}, \dot{\omega}_{gb}) & \text{if } u < 1 \\ (0, 0) & \text{if } u = 1 \end{cases} \\ \omega_{ICE} &\in [\omega_{ICE, idle}, \omega_{ICE, max}]\end{aligned}\quad (6)$$

The torque requested to the ICE derives from Equation 5 and can be written as $T_{ICE} = (1 - u) \cdot T_{Req}$ and the internal

requested torque accounting for the inertial term $T_{ICE,i} = T_{ICE} + J_{ICE}\dot{\omega}_{ICE}$, with $T_{ICE,i} \in [0, T_{ICE,max}]$. It follows the fuel mass flow rate $\dot{m}_f(t, u)$, calculated from a steady-state consumption map, neglecting engine cold start effects, as depicted in Figure (3):

$$\dot{m}_f(t, u) = \begin{cases} \dot{m}_f(T_{ICE,i}, \omega_{ICE}), & \text{if } u < 1 \\ 0, & \text{if } u = 1 \end{cases} \quad (7)$$

Hybrid Electric Vehicle (HEV) Model

Similarly to the ICE, the EMG is modeled via a quasi-stationary map, illustrated in Figure (5). EMGs and batteries data can be found in Table 7 in Appendix.

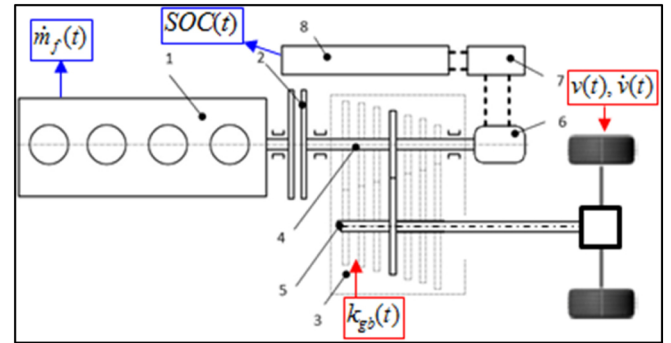


Figure 4. Schematic of HEV system: 1-ICE 2-ICE clutch 3-Automated Manual Transmission (AMT) 4-Primary AMT shaft 5-Secondary AMT shaft 6-Electric Motor-Generator (EMG) 7-Power electronics (PE and BMS) 8-Li-Ion battery a-Power flow to the differential-front axle

Once the T_{EMG} is increased by the inertial term related to the electric machine inertia J_{EMG} , the effective EMG torque

is $T_{EMG,i} = T_{EMG} + J_{EMG}\dot{\omega}_{gb}$, with $T_{EMG,i} \in [T_{EMG,min}(\omega_{gb}), T_{EMG,max}(\omega_{gb})]$.

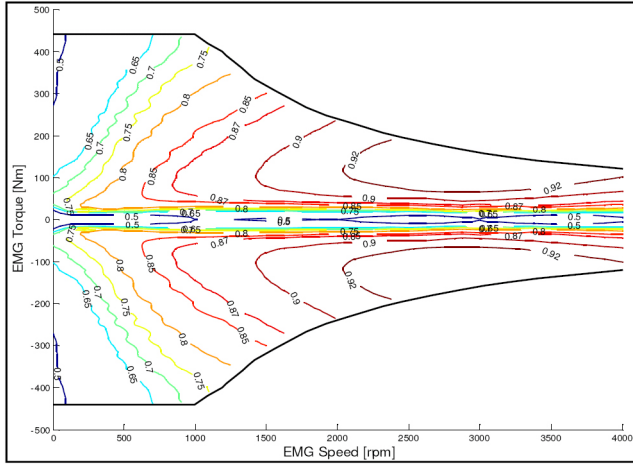


Figure 5. Map-based representation of the EMG efficiency and Torque range limitations (Full HEV, see Table 7).

The battery requested power can be calculated through the efficiency map as $P_{bat} = f(T_{EMG,i}, \omega_{gb})$. Figure (6) represents the resistive equivalent circuit schematization of the battery, characterized by an open-circuit voltage $V_{oc}(SOC)$ and variable internal resistance $R_i(SOC, sign(I_{bat}))$ where the battery current I_{bat} is considered to be positive when discharging. Thermal effects regarding the battery internal resistance have been neglected throughout the reported simulations.

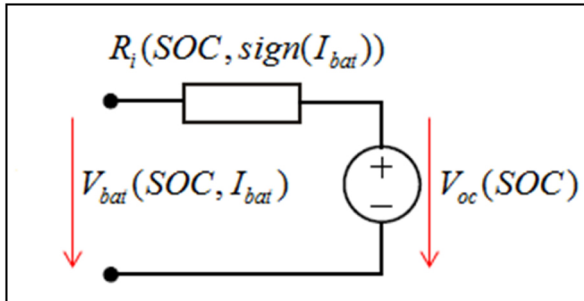


Figure 6. Equivalent circuit representation of Li-Ion battery.

The battery terminal voltage results:

$$V_{bat}(SOC, I_{bat}) = V_{oc}(SOC) - R_i(SOC, sign(I_{bat})) \cdot I_{bat} \quad (8)$$

Since the terminal battery power $P_{bat} = V_{bat}(\cdot) \cdot I_{bat}(\cdot)$, then by combining the latter expression with Equation 8, and conveniently solving with respect to the battery current in the quasi-static simulation model, it follows:

$$I_{bat}(\cdot) = \frac{V_{oc}(\cdot) - \sqrt{V_{oc}^2(\cdot) - 4R_i(\cdot) \cdot P_{bat}}}{2R_i(\cdot)} \quad (9)$$

Finally, if Q_0 is the battery nominal capacity, the state-of-charge derivative can be expressed by:

$$\frac{dSOC(t)}{dt} = -\frac{I_{bat}}{Q_0} \quad (10)$$

High Speed Flywheel-Hybrid Vehicle (HSF-HV) Model

As shown in Figure (7), the mechanical hybrid propulsion system is formed by a clutch, a CVT, a fixed gear and a HSF, installed in a vacuum carter (HSF data can be found in Table 6 in Appendix). The HSF (a small inertia flywheel that rotates at very high speed) is coupled to the driveline at the primary shaft of the AMT gearbox [1]. Unlike the engine-side clutch, neglecting the losses associated to the HSF clutch would lead to unfair comparisons between the different hybrid concepts. If $\eta_{c,H}$ is the HSF clutch efficiency, and taking into account the inertial term due to the clutch outer disk inertia $J_{c,Hout}$, the power demanded at the CVT primary shaft (element 7 in Figure (7)), may be expressed as:

$P_{c,H} = (P_{HSF} + J_{c,Hout}\omega_{gb}\dot{\omega}_{gb})/\eta_{c,H}^{sign(P_{HSF})}$. The clutch efficiency is defined as a function of the slip speed, and it depends on the power flow direction, so that:

$$\eta_{c,H} = \begin{cases} 1 - \frac{|\omega_{cvt,1} - \omega_{gb}|}{\omega_{cvt,1}}, & \text{if } (P_{HSF} > 0) \\ 1 - \frac{|\omega_{cvt,1} - \omega_{gb}|}{\omega_{gb}}, & \text{if } (P_{HSF} < 0) \end{cases} \quad (11)$$

Since the kinematic relationship between the primary and secondary CVT shafts leads to $\omega_{cvt,2}(t) = \omega_{cvt,1}(t) \cdot \tau_{cvt}(t)$, the following dynamic balance equations can be written:

$$\begin{aligned} \dot{\omega}_{cvt,1} \cdot (J_{c,Hin} + J_{cvt,1}) &= T_{cvt,1} - T_{c,H} \\ \dot{\omega}_{cvt,2} \cdot J_{cvt,2} &= T_{HSF,2} - T_{cvt,2} \\ \dot{\omega}_{cvt,2} &= \dot{\omega}_{cvt,1} \cdot \tau_{cvt} + \omega_{cvt,1} \cdot \dot{\tau}_{cvt} \\ T_{cvt,1} &= T_{cvt,2} \cdot \tau_{cvt} \cdot \eta_{cvt}^{sign(T_{HSF})} \end{aligned} \quad (12)$$

where $T_{cvt,1}$ and $T_{cvt,2}$ represent the torques that the CVT applies on the primary and secondary shafts respectively, $T_{HSF,2}$ the torque that the HSF applies on the CVT secondary shaft through the fixed gear. The inertial terms $J_{cvt,1}$, $J_{cvt,2}$, $J_{c,Hin}$ refer to the variable transmission shafts and the clutch inner disk respectively. For the sake of simplicity, the transmission efficiency η_{cvt} is computed as a function of the transmission ratio, although a more detailed description would include the effects of torque and speed, as reported in