

A Parallel Hybrid Electric Drivetrain Layout with Torque-Fill Capability

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ABSTRACT

This paper discusses the torque-fill capability of a novel hybrid electric drivetrain for a high-performance passenger car, originally equipped with a dual-clutch transmission system, driven by an internal combustion engine. The paper presents the simulation models of the two drivetrains, including examples of experimental validation during upshifts. An important functionality of the electric motor drive within the novel drivetrain is to provide torque-fill during gearshifts when the vehicle is engine-driven. A gearshift performance indicator is introduced in the paper, and the two drivetrain layouts are assessed in terms of gearshift quality performance for a range of maneuvers.

CITATION: Mehdizadeh Gavgani, A., Bingham, T., Sorniotti, A., Doherty, J. et al., "A Parallel Hybrid Electric Drivetrain Layout with Torque-Fill Capability," *SAE Int. J. Passeng. Cars - Mech. Syst.* 8(2):2015, doi:10.4271/2015-01-9108.

INTRODUCTION

Dual-clutch transmission systems have encountered significant success in recent years, especially in the European market, as an effective solution, combining high mechanical efficiency, smooth gearshift actuation, and reasonable mechanical complexity [1, 2, 3, 4, 5, 6, 7, 8, 9].

[1, 10] present a comparison, in terms of energy efficiency, between manual transmissions (MTs), automated manual transmissions (AMTs), automatic transmissions (ATs), continuously variable transmissions (CVTs) and dual-clutch transmissions (DCTs). Despite AMTs being characterized by significant efficiency and mechanical simplicity, they do not fulfill the requirement of torque gap reduction during gearshifts [1, 4]. Nowadays this aspect is of uttermost importance for meeting customers' expectations in advanced automotive markets, especially for premium vehicles.

Several technical solutions have been proposed, in order to improve the gearshift performance of AMTs. For example, [11, 12, 13] suggest the introduction of an epicyclic gearset between the engine shaft and the secondary shaft of the transmission. According to this layout, either the ring or sun gear of the epicyclic gearset is connected to a friction brake. The control of the friction brake torque allows a torque transfer to the secondary shaft, thus reducing the torque gap during the process. The main issue of this solution is the controllability of the friction brake torque, together with the associated thermal aspects, i.e., the variation of the friction coefficient as a function of the

operating condition of the friction material. [14, 15] suggest the implementation of a wet clutch in place of the fifth gear synchronizer, in order to transmit the torque to the wheels during gearshifts. [16] illustrates a torque-fill layout based on a flywheel (adopted mainly for energy efficiency reasons) and a planetary gearset. The flywheel is connected to the secondary shaft of the gearbox through the epicyclic gearset and is energized by the internal combustion engine. Throughout the shift the flywheel (provided that it is characterized by an adequate energy level) can transfer its kinetic energy to the secondary shaft. The downsides of this method relate to: i) the increased transmission inertia and hence reduced vehicle acceleration in condition of constant gear ratio; and ii) the safety issues implied by the implementation of a flywheel on the vehicle. DCTs overcome these problems at the cost of increased mechanical complexity with respect to AMTs, which is even more significant if drivetrain hybridization is considered.

Nowadays hybridization is an important requirement for achieving the energy efficiency and emission improvements required by future regulations [17]. This aspect is significant even in high-performance vehicle applications, located at the top-end of the automotive market. In this paper a novel drivetrain layout is proposed, designed around a hybridized AMT-based drivetrain for high-performance passenger cars. The resulting drivetrain is simpler than the corresponding hybridized DCT. Moreover, the control of the electric motor (EM) torque, which is characterized by faster dynamics than the internal combustion engine (ICE) torque [18, 19], allows the reduction

of the torque gap during gearshifts in most operating conditions. A qualitative summary of the possibilities of torque-fill within hybrid electric drivetrains is provided in [20, 21].

The paper is organized as follows:

- The layout of the novel drivetrain is described;
- The models of the baseline vehicle with the DCT (coupled with the ICE only) and the hybrid electric vehicle with the novel drivetrain are presented, including their experimental validation based on test vehicle results;
- The performance of the novel and DCT drivetrains is compared based on an extensive set of simulation results. A gearshift performance indicator is introduced for the assessment of the DCT and the novel transmission system.

THE NOVEL HYBRID ELECTRIC DRIVETRAIN

The hybrid electric drivetrain consists of six gear ratios associated with the internal combustion engine ('ICE gears' in Figure 1) and two gear ratios associated with the electric motor drive ('EM gears'). Hence it will be referred to as '6+2' in the remainder of the paper. Figure 1 displays the schematic of the layout. The engine transmits the torque through a dry clutch, a primary shaft and a secondary shaft, according to a conventional AMT configuration. The electric motor drive can be linked to the secondary shaft of the transmission through the following route:

- A planetary gearset, a sequence of two gear drops ('drop 1' and 'drop 2' in Figure 1) and a hybrid coupling (indicated in Figure 1), implemented through a dog clutch, in the case of gear 1. A synchronizer is not required on the electric motor side of the transmission, as the synchronization process can be electrically controlled through the accurate modulation of the motor torque and closed-loop control of motor speed;
- A dog clutch (connecting the electric motor drive directly to 'Drop 1'), the sequence of two gear drops and the hybrid coupling, in the case of gear 2.

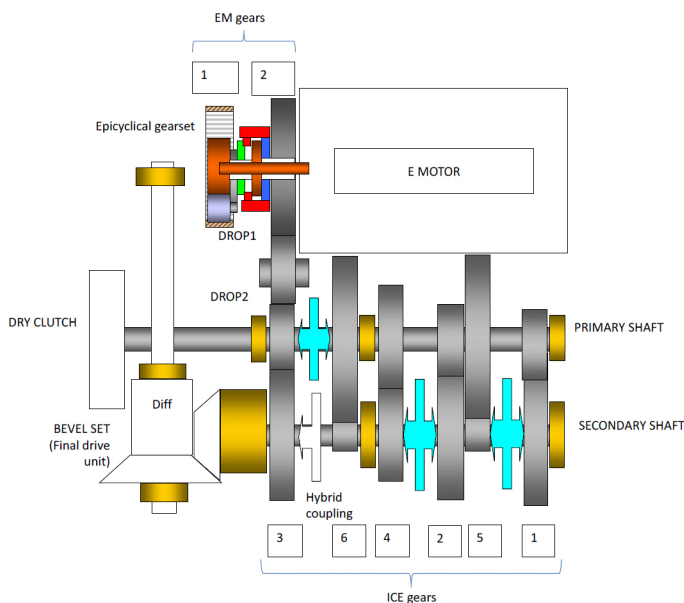


Figure 1. Schematic of the '6+2' hybrid electric drivetrain

The system is actuated hydraulically. Figure 2 is the three-dimensional representation of the mechanical components of the assembly and the overall packaging.



Figure 2. Three-dimensional view of the '6+2' hybrid electric drivetrain

The drivetrain can operate in:

- ICE mode, with the electric motor drive used only for torque-filling during gearshifts;
- Electric mode, with the electric motor drive providing the whole drivetrain torque. This mode is suitable for urban driving and low-speed vehicle operation (i.e., initial launch at low torque demands);
- Hybrid mode, based on the cooperation of ICE and EM, for improved energy efficiency.

During gearshifts in a conventional AMT the dry clutch is disengaged and therefore the engine is not transmitting any torque to the wheels. This produces a torque gap, which can be partially or totally compensated (depending on the operating condition of the vehicle) by an increase of the electric motor drive torque in the '6+2' layout. This feature gives origin to the torque-fill capability of the '6+2' drivetrain.

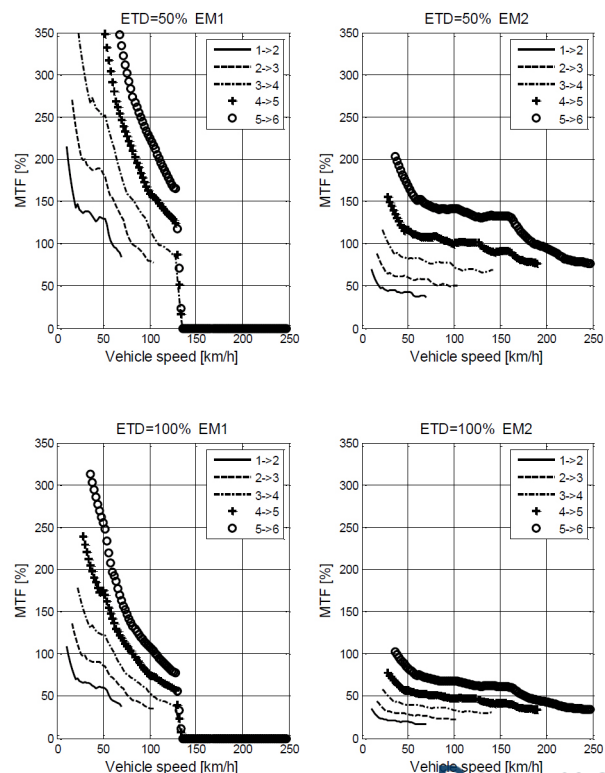


Figure 3. MTF as a function of vehicle speed, selected gear and engine torque demand (ETD)

A crude estimation of the maximum torque-fill capability is required for initial analysis and design. To this purpose, the maximum torque-fill parameter (*MTF*) is defined, as the ratio between: i) the maximum deliverable torque to the wheels by the electric motor drive during the ICE gearshift; and ii) the sum of the engine and motor torques referred to the wheels, according to the expected operating conditions at the completion of the ICE-related gearshift. These final conditions are expressed in terms of final torque demands for the ICE and the EM, ETD_{final} and $EMTD_{final}$ respectively, and final speed values, $\dot{\vartheta}_{E,final}$ and $\dot{\vartheta}_{EM,final}$, where:

$$ETD_{final} = 100 T_{E,final} / T_{E,MAX}(\dot{\vartheta}_{E,final}) \quad (1)$$

$$EMTD_{final} = 100 T_{EM,final} / T_{EM,MAX}(\dot{\vartheta}_{EM,final}) \quad (2)$$

Hence the equation for *MTF* is:

$$MTF = \frac{T_{EM}(100\%, \dot{\vartheta}_{EM}) \tau_{EM}}{T_E(ETD_{final}, \dot{\vartheta}_{E,final}) \tau_{E,final} + T_{EM}(EMTD_{final}, \dot{\vartheta}_{EM,final}) \tau_{EM}} 100 \quad (3)$$

Figure 3 plots the values of *MTF* derived from Equation (3), based on the assumption that the drivetrain is operating in ICE mode, and therefore $T_{EM}(EMTD_{final}, \dot{\vartheta}_{EM,final}) = 0$ in the denominator of Equation (3), for *ETD* equal to 50% and 100%. 'EM1' and 'EM2' in Figure 3 respectively refer to the first and second gear ratio of the EM side of the drivetrain. As expected, the graphs show better torque-fill capability for low engine torque demands, low speed and high gear conditions on the ICE side. The shape of the graphs depends on the torque characteristics of the engine and the electric motor drive. For the case 'EM1', which is the most effective gear for torque-fill, the variation of the *MTF* profile is evident when the base speed (corresponding to about 5700 rpm in the specific application) of the electric motor drive is reached. Given the maximum values of ICE and EM power for the specific case study rear-wheel-drive vehicle (the main parameters are provided in the appendix), respectively equal to 422 kW and 120 kW, it is normal that in conditions of significant torque demand and low engine gear, the values of *MTF* are well below 100%. However the torque-fill effect is significant in most of typical real-world driving conditions. The torque-fill effect disappears at vehicle speeds corresponding to the maximum allowed speed of the electric motor. This is the cause of the step reduction of *MTF* for 'EM1' between 100 km/h and 150 km/h.

Figure 4 plots the maximum deliverable torque to the wheels as a function of vehicle speed, for all the states of the powertrain, purely ICE-driven ('ICE1', ..., 'ICE6' in Figure 4) and purely EM-driven ('EM1' and 'EM2'), with the overall envelope profile ('Hybrid' in Figure 4) corresponding to the hybrid mode. This preliminary static analysis of the '6+2' torque-fill performance will be complemented by the dynamic analysis presented in the following sections.

For the specific high-performance vehicle application, the DCT must adopt two wet clutches, because of the power dissipation in the clutches during dual-clutch gearshifts, requiring significant cooling capability of the clutch assembly. On the contrary, the '6+2'

drivetrain can adopt a dry clutch, because of the single clutch gearshifts, which are less demanding in terms of energy dissipation. Moreover, apart from the clutch assembly, the overall layout of the '6+2' is intrinsically simpler, lighter and more efficient than that of the equivalent DCT. For example, Figure 5 provides an experimental comparison of the time histories of fuel consumption along a New European Driving Cycle (NEDC) for a vehicle with the DCT (production vehicle configuration with ICE only) and the same vehicle with the '6+2' (prototype version), where the '6+2' is used in purely ICE mode for a fair comparison of the results. These measurements were carried out on the rolling road facility at MIRA proving ground (United Kingdom). The actual benefit of the enhanced transmission efficiency of the '6+2' is even more significant than it appears from this experiment, as the mass of the two vehicles under testing was different (higher for the '6+2' vehicle prototype, because of the EM drive and the battery pack).

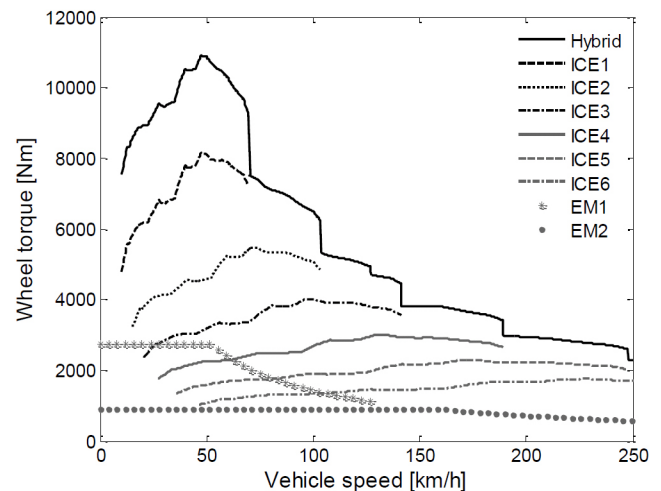


Figure 4. Maximum deliverable wheel torque for all possible states of the powertrain

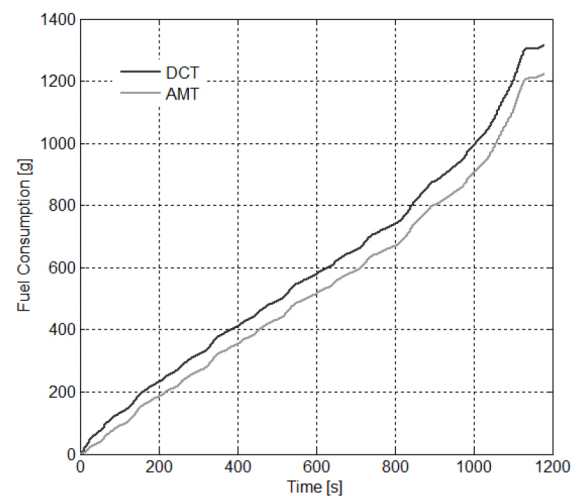


Figure 5. Comparison of the experimentally measured fuel consumption along the NEDC driving cycle for the DCT and AMT versions of the same vehicle, driven in ICE mode

DCT MODELING, CONTROL AND EXPERIMENTAL VALIDATION

Vehicle and Drivetrain Model with DCT

The objective of the model is the reliable simulation (in Matlab Simulink) of the gearshift performance in terms of resulting vehicle acceleration profile, with a limited number of degrees of freedom and model parameters. Figure 6 is the simplified schematic of the model of the DCT drivetrain originally installed on the ICE-driven vehicle. A transfer function and a pure time delay are used for modeling the dynamic performance of the ICE, e.g., the effect of the air intake system. The number of the mechanical degrees of freedom of the implemented simulation model is variable between 4 and 5, depending on the status of the clutches, which can be either engaged or disengaged. As the gear selection process is carried out in advance of the gearshift (pre-selection of the gears), its dynamics are neglected in this section.

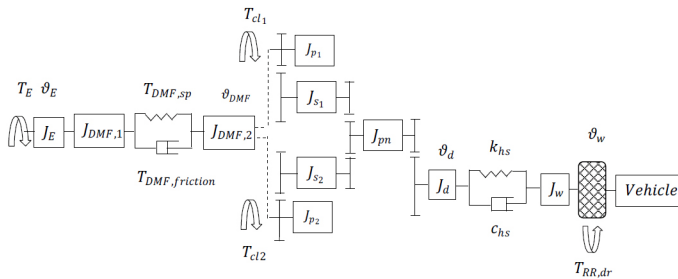


Figure 6. Simplified schematic of the DCT drivetrain originally installed on the baseline vehicle

The engine shaft dynamics are governed by the following equation:

$$T_E - T_{DMF} = (J_E + J_{DMF,1})\ddot{\theta}_E \quad (4)$$

where T_{DMF} is the torque of the torsionally compliant components (i.e., what is commonly defined as the clutch damper) of the dualmass flywheel. T_{DMF} is given by the sum of the contributions caused by the internal springs, ($T_{DMF,sp}$, non-linear function of the torsion angle of the dual-mass flywheel) and the internal friction torque of the flywheel ($T_{DMF,friction}$, function of the torsion angle and torsion rate of the component):

$$T_{DMF} = T_{DMF,sp}(\theta_E - \theta_{DMF}) + T_{DMF,friction}(\theta_E - \theta_{DMF})\text{sign}(\dot{\theta}_E - \dot{\theta}_{DMF}) \quad (5)$$

The mass moment of inertia of the dual-mass flywheel is subject to T_{DMF} and the clutch torques, T_{cl1} and T_{cl2} :

$$T_{DMF} - T_{cl1} - T_{cl2} = J_{DMF,2}\ddot{\theta}_{DMF} \quad (6)$$

In case of dry clutches, clutch torque is controlled only during gearshifts, as a zero-slip condition is passively maintained through an appropriate axial pre-load when a gear is engaged. In the case of wet clutches, continuous clutch slip control can be implemented, so that a 'micro-slip' is constantly present on the clutch associated with the engaged gear ratio, in order to improve the driveline dynamic response (i.e., vehicle drivability) in conditions of a fixed gear ratio.

As this is a model of vehicle longitudinal dynamics, the dynamics of the internal parts of the differential can be neglected and a single equivalent half-shaft can be considered within the model formulation, with equivalent stiffness and damping properties corresponding to those of the actual half-shafts installed on the vehicle. The drivetrain dynamics are expressed by:

$$T_{cl1}\tau_{E1}\tau_{pn}\tau_d + T_{cl2}\tau_{E2}\tau_{pn}\tau_d - T_{hs} = J_{eq,DCT}\ddot{\theta}_d \quad (7)$$

$J_{eq,DCT}$ is the equivalent mass moment of the rotating parts of the DCT, referred to the wheels:

$$J_{eq,DCT} = J_d + J_{pn}\tau_d^2 + (J_{s1} + J_{s2})\tau_{pn}^2\tau_d^2 + J_{p1}\tau_{E1}^2\tau_{pn}^2\tau_d^2 + J_{p2}\tau_{E2}^2\tau_{pn}^2\tau_d^2 \quad (8)$$

The efficiency of the gears is neglected in the equations of this simplified model formulation, which is focused instead on the analysis of the system dynamics. The transmission system manufacturer has developed a detailed model of the transmission power loss contributions for the different states, including the effect of the gears, bearings, synchronizers, churning and windage, as functions of the operating conditions of the system. In the simulation analyses (not reported here) focused on the assessment of the energy efficiency aspect, the efficiency was included as a lumped contribution into the simulation model. As during gearshifts sign variations of the drivetrain torque are possible during/after the clutch disengagement phase (if not properly actuated, since normally the DCT should reduce/eliminate the wheel torque gap), the model includes a lumped-parameter formulation of the mechanical plays of the drivetrain. The plays are considered at the interface between the transmission output and the equivalent half-shaft. As a consequence, the half-shaft torque is:

$$T_{hs} = k_{hs}\Delta\theta_{hs} + c_{hs}\Delta\dot{\theta}_{hs} \quad (9)$$

where the actual torsion angle of the half-shaft, $\Delta\theta_{hs}$, corrected by the amount of equivalent drivetrain backlash, $\vartheta_{backlash}$, is expressed by:

$$\Delta\theta_{hs} = \begin{cases} 0, & \text{if } |\vartheta_d - \vartheta_w| < \vartheta_{backlash} \\ (|\vartheta_d - \vartheta_w| - \vartheta_{backlash})\text{sign}(\vartheta_d - \vartheta_w), & \text{if } |\vartheta_d - \vartheta_w| \geq \vartheta_{backlash} \end{cases} \quad (10)$$