**ANALYSIS AND SIMULATION OF THE GEARSHIFT METHODOLOGY FOR
A NOVEL TWO-SPEED TRANSMISSION SYSTEM FOR ELECTRIC
POWERTRAINS WITH CENTRAL MOTOR**

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Abstract

Electric vehicle powertrains traditionally consist of a central electric motor drive, a single-speed transmission and a differential. This electric powertrain layout, either for use in fully electric vehicles or through-the-road parallel hybrid electric vehicles, will be extensively adopted in the next few years, despite the ongoing research in electric vehicles with individually controlled motors. However, current research suggests that electric powertrains with a central electric motor drive can still be widely improved. For example, the installation of a seamless multiple-speed transmission instead of a single-speed can provoke an increase in vehicle performance, together with an enhancement of the overall efficiency of the electric powertrain. These novel transmission systems for electric powertrains require a specific design, in order to be efficient, compact, easy and robust to control, and cheap to manufacture. This article presents the mechanical layout and the control system of a novel two-speed transmission system designed by the authors, with particular focus on the achievement of optimal gearshift dynamics. The torque characteristics of typical electric motor drives require a different actuation of the seamless gearshifts, in comparison with the equivalent operation for a dual clutch transmission within an internal combustion engine driven powertrain.

Keywords

2-speed transmission; seamless gearshift; control; electric powertrain

1. The Benefits of Multiple-Speed Transmission Systems for Electric Powertrains

Due to the nature of the torque characteristics of typical electric vehicle traction motors, electric vehicles (or electric axles for through-the-road parallel hybrid electric vehicles) are generally equipped with a single-speed transmission. This is due to electric motors exhibiting a high constant torque from zero to base speed then entering into a constant power region for higher angular velocities. A further benefit of a central motor drive equipped with only a single-speed transmission is a reduction in the drivetrain mass, volume, losses and cost [1-3].

Despite the wide operational speed range of such traction motors (e.g. from 0 to 8 000-12 000 rpm), a multiple-speed transmission [4-6] may be employed to enhance the vehicle performance. In particular the wheel torque at low vehicle velocities is increased along with the maximum road gradient that the vehicle can ascend when transporting heavy payloads. In addition, the lower second gear ratio raises the vehicle top speed.

As an example, Figure 1 shows the wheel torque envelopes achievable through a single-speed transmission in traction and regeneration, and the same characteristics for each gear of a two-speed transmission system. The case study vehicle for Figures 1 and 2 is a prototype electric minibus wherein the electric motor drive is characterised by a limited regenerative capability due to the design of the power electronics. The two-speed transmission brings an increase of approximately 34% of the peak value of wheel torque over the single-speed axle, namely 800 Nm. Secondly, Figure 1 demonstrates the

benefit in vehicle top speed. At 0% of road gradient, the single-speed vehicle (in unladen conditions) is characterised by a top speed $V_{Max,SS,0\%}$ of 119 km/h, limited by the maximum speed of the electric motor (8 000 rpm for the case study motor). The top speed $V_{Max,TS,0\%}$ for the two-speed vehicle is 149 km/h, limited by the available motor power, but still showing a gain of 24% over the single-speed transmission vehicle.

Figure 1 – Potential performance benefit deriving from the adoption of a two-speed transmission system over a single-speed for a case study vehicle

Moreover, in most cases the overall efficiency of the electric motor drive and energy storage unit varies significantly as a function of the operating torque and speed. The adoption of a two-speed transmission can facilitate significant energy benefits through

the optimisation of the operating points of the electric powertrain over a given driving schedule. Consequently, the selection of the gear ratio is as important as in the case of a conventional vehicle driven by an internal combustion engine. The efficiency of a multiple-speed transmission system is generally lower than the efficiency of the single-speed transmission for the same vehicle at least due to the losses relating to the presence of an actuation system and increased mass. However this variation for conventional transmission systems based on a layshaft layout, is usually marginal in comparison with the potential increase in the overall powertrain efficiency. Evidence of this is comprehensively provided in [4] and [7]. As an example, Figure 2 presents the distribution of the operating points on the efficiency map of the electric motor drive during a NEDC (New European Driving Cycle). The results are compared for the same case study vehicle of Figure 1, equipped with a single-speed and a two-speed transmission. The two-speed transmission system predominantly operates in second gear at higher electric motor torques and lower speeds than the single-speed transmission, and consequently in a higher efficiency region of the motor drive. This conclusion is not only true for this singular case as the authors have derived diagrams and results similar to those in Figures 1 and 2 for very different vehicle typologies and electric powertrain characteristics [7].

Figure 2 – Potential energy efficiency benefit deriving from the adoption of a two-speed transmission system

The existing literature suggests that two-speed transmission systems are the most suitable solution for increasing the efficiency and performance of electric axles, [7-11], without adding significant complexity and mass. [9] and [10] summarise the optimisation procedures implemented for the choice of gear ratios and gearshift map for two-speed transmission systems for electric axles. In particular, along a set of typical driving schedules the energy consumption is reduced by approximately 10%, in comparison with the single-speed vehicle equipped with the same motor drive and energy storage unit. [11] adopts the same optimisation procedure of [9] and [10] in

order to demonstrate that a multiple-speed transmission system can permit a significant reduction of the electric motor drive peak torque, for the same value of its peak power. The downsizing (in peak torque) of the electric powertrain implies an additional benefit in terms of mass, energy consumption and manufacturing cost.

This article presents the description of a novel transmission system layout specifically designed for electric powertrains and the mathematical equations for modelling the overall system dynamics. In addition, the principles of the methodology developed for controlling the upshifts and downshifts with the new transmission are derived. The issues which need to be addressed when designing a seamless gearshift control for electric powertrains are discussed in great detail through a comprehensive set of simulation results.

2. The Novel Transmission System

A multiple-speed transmission system must be able to accomplish seamless gearshifts in order to be competitive against a single-speed transmission for electric axles. A seamless gearshift is particularly relevant as a large number of upshifts (in conditions of medium-high torque demands) are carried out in the constant power region of the electric motor which can result in a perceivable torque gap.

Seamless gearshifts can be achieved by using a cascade of planetary gear sets, as presented in [12], with a one-way clutch used to provide the torque reaction in first gear

during traction. A hydraulic actuation system is adopted for the control of the two friction clutches required for the second gear, the reverse gear and regeneration. An alternative seamless multiple-speed transmission layout for electric vehicle applications can be based on the adoption of a Dual Clutch system on a layshaft type transmission, as described in [13]. This solution is becoming common for internal combustion engine driven vehicles, but is characterised by a significant mechanical complexity and requires careful control of the actuation of the on-going and off-going friction clutches [14]. As a consequence, this layout is suitable for transmission systems characterised by at least three gear ratios, but more cost-effective solutions can be evaluated for seamless two-speed transmissions for electric vehicles. In particular, according to [15] and [16], the use of one-way clutches can improve shift quality to a great extent, as their ability to automatically disengage if the input torque becomes negative benefits the control of a clutch-to-clutch gearshift. An alternative design solution for the implementation of seamless gearshifts is based on a novel design of the synchronising rings and their actuation [17].

The novel two-speed transmission system of this paper combines the mechanical simplicity of a layshaft type transmission, with the high quality of a clutch-to-clutch gearshift. Its primary components are a one-way sprag clutch located on the secondary shaft and a friction clutch on the primary shaft along with an open differential, as displayed in Figure 3. The input torque is transmitted by the sprag clutch whilst in first

gear, and by the friction clutch whilst in second gear. The system can work either as a fully automated transmission, or as an automated manual transmission through a seamless shift system [18]. The friction clutch is applied to transfer torque from the sprag clutch during an upshift, and released to allow the sprag clutch to engage to accomplish a downshift. A full description of the gearshift methodology is presented in section 4.

Figure 3 – Schematic of the transmission operation in first gear (left) and second gear (right)

The multi-disc friction clutch utilises sintered metal friction material and is electro-hydraulically controlled using a remote brushless motor driven actuator, pressurising a master cylinder mechanically connected to the Belleville spring of the friction clutch. The friction clutch actuation system is controlled through a feedback loop based on actuator displacement. In order to allow regenerative energy recovery whilst

decelerating in first gear, the engagement of a locking ring, electro-mechanically actuated, prevents the one-way sprag clutch from overrunning when the direction of torque through the transmission is reversed. Once the vehicle has come to rest, the gearshift mechanism can also be used as a park lock by simultaneously engaging the locking ring and closing the friction clutch, thus eliminating the need for a separate park lock mechanism and actuator. The mechanical layout of the two-speed transmission results in a compact design, with the overall distance between the primary and differential shaft being about 200 mm for a premium passenger car. Furthermore, for this application the distance from the primary shaft to the secondary shaft is less than 110 mm and secondary shaft to the differential is approximately 125 mm. The transmission with this layout results in a mass of about 38 kg. The dimensions and weight are comparable with the single-speed unit (25 kg) from which this novel transmission was derived.

3. Transmission Dynamics: Governing Principles

The best methodology for developing the control system for the gearshift actuation is through the analysis of the equations governing the overall system dynamics [19]. Starting from the basic physical principles described in this section, the control system for the gearshift actuation will be described in detail in section 4.

The derivation of the equations governing the dynamics of the bespoke two-speed transmission system has been carried out in [20], therefore only the final dynamic equations are presented here. In the following description of the first order system dynamics the plays and internal torsion deformations of the transmission components, in particular of the engaged sprag clutch, are not considered as they are second approximation effects in relation to the gearshift control analysis which is the focus of the article. In fact, these characteristics do not affect the acceleration profile of the vehicle in conditions of constant gear or upshift / downshift. Experimental tests on the sprag clutch have revealed that its internal elastic deformation, proportional to the transmitted torque (with a maximum torsion angle of 2-3 deg), is negligible in comparison with the torsion dynamics of the half-shafts, from the viewpoint of the vehicle low frequency drivability.

The transmission system can work in three different states, each of which is characterised by different governing equations:

- *Engaged first gear.* In this condition the kinematical ratio between the electric motor shaft speed and the transmission output shaft speed is equal to the first gear ratio, and the friction clutch can be transmitting torque while slipping during a gearshift (torque phase of the gearshift).
- *Inertia phase of the gearshift.* In this condition both clutches are slipping, and the transmission system is characterised by two degrees of freedom, as the

electric motor dynamics are decoupled from the transmission output shaft dynamics.

- *Engaged second gear.* In this condition the friction clutch is engaged and no torque is transmitted through the sprag clutch, which is overrunning.

3.1 Engaged First Gear

Equation (3.1) governs the overall transmission dynamics in this condition, derived through each shaft moment balance equation:

$$\ddot{\theta}_{diff} = \frac{T_m \cdot i_1 \cdot \eta_1 \cdot i_{diff} \cdot \eta_{diff} - T_h + T_{fc} \cdot i_2 \cdot \eta_2 \cdot i_{diff} \cdot \eta_{diff} - T_{fc} \cdot i_1 \cdot \eta_1 \cdot i_{diff} \cdot \eta_{diff}}{J_{eq,trans,gear1}} \quad (3.1)$$

The efficiencies in the formulas (the η terms, having values between 0 and 1) can be reversed depending on the direction of the transmitted power through the coupling. With reference to the differential, the equivalent moment of inertia of the transmission in first gear is given by:

$$J_{eq,trans,gear1} = (J_{diff} + 0.5 \cdot J_{LHS} + 0.5 \cdot J_{RHS}) + (J_m + J_1) \cdot i_1^2 \cdot \eta_1 \cdot i_{diff}^2 \cdot \eta_{diff} + J_{1b} \cdot i_2^2 \cdot \eta_2 \cdot i_{diff}^2 \cdot \eta_{diff} + (J_2 + J_{2b}) \cdot i_{diff}^2 \cdot \eta_{diff} \quad (3.2)$$

During normal operation of the first gear, $T_{fc} = 0$, but throughout the torque phase of the gearshift, the friction clutch is transmitting increasing (in case of upshift) or decreasing (in case of downshift) torque T_{fc} while slipping. When the friction clutch torque is applied, the system experiences a torque variation at the differential due to the term

$T_{fc} \cdot i_{diff} \cdot \eta_{diff}$ ($i_2 \cdot \eta_2 \cdot i_1 \cdot \eta_1$) in the numerator of equation (3.1), as part of the transmission input torque is transmitted to the output shaft of the transmission through gear one (torque contribution $(T_m - T_{fc}) \cdot i_1 \cdot \eta_1 \cdot i_{diff} \cdot \eta_{diff}$), and part through gear two (torque contribution $T_{fc} \cdot i_2 \cdot \eta_2 \cdot i_{diff} \cdot \eta_{diff}$). If the locking ring is disengaged, the system still operates in first gear if the sprag clutch torque T_{sc} is positive (i.e. the system is in traction).

3.2 Inertia Phase of the Gearshift

During the inertia phase of the gearshift, the ratio between motor speed and output shaft speed is intermediate between i_2 and i_1 . In such a condition, the equation describing motor dynamics is:

$$\ddot{\theta}_m = \frac{T_m - T_{fc}}{\frac{J_{2b}}{i_1^2 \cdot \eta_1} + J_m + J_1} \quad (3.3)$$

whilst the equation for transmission dynamics is:

$$\ddot{\theta}_{diff} = \frac{T_{fc} \cdot i_2 \cdot \eta_2 \cdot i_{diff} \cdot \eta_{diff} - T_h}{J_{eq,trans,IP}} \quad (3.4)$$

where:

$$J_{eq,trans,IP} = (J_{diff} + 0.5J_{LHS} + 0.5J_{RHS}) + J_{1b} \cdot i_2^2 \cdot \eta_2 \cdot i_{diff}^2 \cdot \eta_{diff} + J_2 \cdot i_{diff}^2 \cdot \eta_{diff} \quad (3.5)$$

The equivalent moment of inertia of the transmission system in this condition is much lower than in equation (3.2), mainly because equation (3.4) does not include the electric motor inertia. This makes the first natural frequency of the transmission during the

inertia phase of the gearshift much higher than when the system works either in first or second gear, Figure 4. The high natural frequency can result in possible dynamic and NVH (Noise, Vibration and Harshness) problems in case of a non-optimal tuning of the friction clutch and its control.

3.3 Engaged Second Gear

In second gear, the overall transmission is characterised by one degree of freedom, as in first gear. The transmission system dynamics in second gear are governed by:

$$\ddot{\theta}_{diff} = \frac{T_m \cdot i_2 \cdot \eta_2 \cdot i_{diff} \cdot \eta_{diff} - T_h}{J_{eq,trans,gear2}} \quad (3.6)$$

Equation (3.7) is the expression of the equivalent moment of inertia of the transmission in second gear:

$$J_{eq,trans,gear2} = (J_{diff} + 0.5 \cdot J_{LHS} + 0.5 \cdot J_{RHS}) + \\ + (J_m + J_1 + J_{1b}) \cdot i_2^2 \cdot \eta_2 \cdot i_{diff}^2 \cdot \eta_{diff} + \frac{J_{2b} \cdot i_2^2 \cdot \eta_2 \cdot i_{diff}^2 \cdot \eta_{diff}}{i_1^2 \eta_1} + J_2 \cdot i_{diff}^2 \cdot \eta_{diff} \quad (3.7)$$

The equivalent moment of inertia described by equation (3.7) is lower than the one of equation (3.2), predominantly due to the fact that the moment of inertia of the electric motor drive is multiplied by i_2^2 instead of i_l^2 .

3.4 System Dynamics in the Frequency Domain

Figure 4 illustrates the overall system dynamics in the three states described in sections

3.1-3.3, considering the overall half-shaft torque as an output with the inputs being the reference electric motor drive torque when the gears are engaged, or the friction clutch torque during the inertia phase. These Bode diagrams have been obtained from a linearised model of the overall system, including the dynamics of the electric motor drive, friction clutch actuation system and tyre slip, the tyre relaxation length and half-shaft compliance.

Figure 4 – Frequency response of the system in first gear, second gear and during the inertia phase for system linearization at 10 m/s of vehicle speed

The variation of the first natural frequency of the driveline is evident from first to second gear due the lower value of the equivalent moment of inertia (at the wheels) of

the electric motor. The natural frequency is subjected to a further significant increase during the inertia phase, as the equivalent moment of inertia of the drivetrain is reduced when the electric motor is decoupled from the wheels. The amplitude of the frequency response is reduced between 1 and 25 Hz during the inertia phase, due to the dynamic characteristics (time constant) of the friction clutch actuator.

4. The Gearshift Control System

4.1 Upshift Control

The control of the upshifts for such a transmission system is similar to the methodology usually adopted for dual clutch transmission systems, with some differences to compensate for the characteristics of the electric powertrain. In this section, upshifts during power-on are analysed.

In order to shift to second gear, firstly the locking ring is disengaged. The torque phase is then initiated by the progressive engagement of the friction clutch at a rate limited by the dynamic properties (reaction time and rise time) of the electro-hydraulic actuation system. This provokes a reduction of the torque T_{sc} transmitted through the sprag clutch while the system is kinematically in first gear and consequently changes the transmission output torque from the first to the second gear value. The progressive transition reduces the drivetrain oscillations which can be observed in a conventional manual transmission system, where the disengagement of the clutch typically provokes

an interruption of the half-shaft torque and produces the consequent excitation of the torsional dynamics of the system. The basic version of the control system presented in this paper, called ‘Control 1’, does not manipulate the driver torque demand, which is directly sent to the motor drive, during the torque phase of the upshift.

At the end of the torque phase the sprag clutch torque T_{sc} becomes null for the friction clutch torque value identified by the following equation:

$$T_{fc,dis} = T_m - (J_m + J_1) \cdot \ddot{\theta}_m - \frac{J_{2b} \cdot \ddot{\theta}_{diff} \cdot i_{diff}}{i_1 \cdot \eta_1} \quad (4.1)$$

T_{fc} is estimated by the control system from the displacement of the friction clutch actuator. Equation (4.1) establishes the beginning of the inertia phase of the upshift, during which the electric motor speed has to be reduced from the value for the first gear ratio to the value for the second gear ratio, whilst keeping an adequate vehicle acceleration profile. The principles for the inertia phase control can be derived from equations (3.3) and (3.4). During the inertia phase, vehicle acceleration dynamics are controlled by the friction clutch torque (equation (3.4)), whilst the difference between electric motor torque and friction clutch torque affects the motor dynamics (equation (3.3)). As a consequence, the two degrees of freedom of the system can be independently controlled, provided that the electric motor drive is not working in conditions of saturation (on its peak torque characteristic, which represents the

constraint of the control system). This is the ‘golden rule’ for the control of the inertia phase of the gearshifts in such a system.

During the inertia phase of the upshift for ‘Control 1’, T_{fc} is ramped up according to an open-loop control system at the same rate as the torque phase to reduce any driveline oscillations during phase transitions. T_{fc} ramps to a reference level equal to the torque value the electric motor would produce for the actual condition of driver torque demand $DTD(t)$ (not manipulated by the controller) and electric motor speed. Additional terms compensate for the inertial torque of the main components of the system. $T_{fc,saturation,IP,US}$ is given by:

$$T_{fc,saturation,IP,US}(t) = T_m \left(DTD(t), \dot{\theta}_m(t) \right) - (J_m + J_1) \cdot \ddot{\theta}_m \quad (4.2)$$

In the meantime, electric motor dynamics are controlled by the combination of a feedforward and a feedback (Proportional Integral Derivative - PID) controller (Figure 5), based on a reference speed profile equal to:

$$\dot{\theta}_{m,ref} = \dot{\theta}_{diff} \cdot i_{diff} \cdot i_1 \cdot y(t_{IP}) \quad (4.3)$$

where the adimensional factor $y(t_{IP})$ is a normalisation parameter defining the reference speed profile, according to the qualitative shape in Figure 5. t_{IP} is the output of a counter which is activated by the transmission control unit at the beginning of the inertia phase. The initial value of y is 1, so that equation (4.3) provides an initial value of the reference motor speed equal to the actual speed of the unit at the beginning of the upshift. The

final value of y is equal to the step ratio i_2/i_1 , so that the final value of the reference motor speed is equal to the one required for the synchronization of the friction clutch.

Figure 5 – Block diagram of the feedforward / feedback control system of electric motor speed during the inertia phase of the upshift

The shape of $y(t_{IP})$ is designed so that the electric motor reference speed is at a maximum in the first part of the inertia phase, and reduces to zero (for the specific tuning shown in Figure 5) at the time $t_{IP,end}$ at the end of the inertia phase. The shape of the profile can be tuned to alter the duration of the inertia phase, depending on the vehicle parameters and required upshift performance, and the amount of perceived discontinuity at the engagement point of the friction clutch at $t_{IP,end}$. If the air-gap torque dynamics of the motor drive is modelled through a first order transfer function,

the open-loop transfer function of the feedback part of the controller of Figure 4 is given by:

$$\frac{\bar{\theta}_m(s)}{\bar{\theta}_{m,ref}(s) - \bar{\theta}_m(s)} = \left(K_P + K_D \cdot s + \frac{K_I}{s} \right) \cdot \frac{1}{J_m \cdot s \cdot (1 + s \cdot \tau_m)} \quad (4.4)$$

Hence the gains of the feedback part of the controller can be tuned according to the well known rules in terms of tracking capability (bandwidth) and phase margin [21].

Figure 6 – Bode diagram of the open-loop and closed-loop transfer functions for the feedback part of the electric motor control loop

Figure 6 shows an example of possible tuning of the PID control parameters and the relating open-loop (given by equation (4.4)) and closed-loop transfer functions. The transfer functions are affected by the time constant τ_m of the electric motor air-gap

torque dynamics (sometimes filtered for anti-jerk purposes). Consequently a sensitivity analysis has been added to Figure 6 to illustrate the effect of the motor time constant.

The parameters of the electric motor controller can vary depending on *DTD* and speed, in order to make the upshift quicker or more comfortable as a function of the specific driving situation. In any case, the motor controller has very low impact on the low frequency vehicle drivability during the upshift, as this is controlled through the friction clutch (equation (4.2)), consistently with the ‘golden rule’. Therefore, the performance of the system is very robust against the variation of the parameters of the feedback motor controller. Tests have been successfully carried out with only the feedforward system reducing the electric motor drive torque by a constant amount and so with no contribution from the PID, with little variation of the perceivable quality of the achieved results.

At the conclusion of the inertia phase, when the friction clutch engages, the friction clutch actuator is moved to its endstop where the transmissible clutch torque is at the nominal level, depending on the wear condition of the clutch disks.

Figure 7 – Possible conditions of an upshift manoeuvre on the electric motor drive torque characteristic, under the hypothesis of constant driver torque demand DTD during the manoeuvre

The upshift control system described until now (‘Control 1’) gives origin to a complete absence of torque gap during the upshift when the initial and the final operating points of the electric motor drive are in the constant torque region of the electric motor. For example, this condition is satisfied for the upshifts from point A to B or point C to D in Figure 7. For upshifts in the constant power region of the electric motor drive, such as those from E to F or G to H, the torque phase of the upshift, when operated as described for ‘Control 1’, involves a reduction of wheel torque. For the same conditions, the inertia phase implies a progressive increase of wheel torque, consistent with the reduction of electric motor speed and the subsequent increase of $T_{fc,saturation,IP,US}$

(equation (4.2)). These variations of wheel torque, especially the increase during the inertia phase, are progressive and provoke lower jerk (time derivative of vehicle acceleration) levels and significantly better acceleration profiles than those experienced in a conventional single-clutch transmission, which would provoke negative values of vehicle acceleration during the phase of the upshift characterised by the disengaged clutch. The control system defined as ‘Control 1’ is reliable, robust and relatively simple; therefore it is currently adopted on the first transmission prototypes mounted onto an electric minibus demonstrator and on the Hardware-In-the-Loop electric axle rig at the University of Surrey.

In order to significantly reduce the torque gap during the inertia phase of the upshift, it is possible to adopt the following control characteristic, here named ‘Control 2’, for the friction clutch saturation level:

$$T_{fc,saturation,IP,US,C2}(t) = T_m \left(DTD(t), \dot{\theta}_{diff}(t) \cdot i_{diff} \cdot i_2 \right) - (J_m + J_1) \cdot \ddot{\theta}_m \quad (4.5)$$

The control system provokes the friction clutch to transmit a torque level equal to the value the electric motor would generate if the system was already in second gear. The adoption of the friction clutch torque of equation (4.5) during the inertia phase of the upshift significantly improves vehicle acceleration, as it eliminates the partial wheel torque gap during the inertia phase of the upshift. However, it generates significant vehicle jerk in the transition between the torque phase and the inertia phase of the upshift. This is due to the fact that the torque phase of the upshift, when implemented

according to ‘Control 1’ and ‘Control 2’, intrinsically produces a reduction of the available wheel torque which is progressively recovered in case of ‘Control 1’ but quite abruptly recovered in case of ‘Control 2’. ‘Control 2’ could be adopted as a sport-oriented transmission control algorithm selectable by the driver.

In the case of an upshift in the constant power region of the electric motor, it is possible to compensate for the reduction of wheel torque, induced by the torque shift from the first to the second gear (torque phase of the upshift), by manipulating the electric motor torque demand. This variant of the control system is defined as ‘Control 3’ and is described by equation (4.6):

$$T_{m,ref,TP,US,C3}(t) = T_m(DTD, \dot{\theta}_m) + T_{fc,est} \cdot \left(1 - \frac{i_2}{i_1}\right) \quad (4.6)$$

With the implementation of such a control system during the entire torque phase, the disengagement of the sprag clutch would happen at a friction clutch torque level equal to:

$$T_{fc,dis} = \left[T_m(DTD, \dot{\theta}_m) - (J_m + J_1) \cdot \ddot{\theta}_m - \frac{J_{2b} \cdot \ddot{\theta}_{diff} \cdot i_{diff}}{i_1 \eta_1} \right] \cdot \frac{i_1}{i_2} \quad (4.7)$$

As a consequence, the final level of the friction clutch torque during the torque phase of the upshift could be higher than the desired friction clutch torque of equation (4.5) for the inertia phase of the upshift, which is the same for ‘Control 2’ and ‘Control 3’, in case of an upshift carried out in the constant torque region of the electric motor drive. In

order to prevent a significant negative jerk (due to the reduction of the friction clutch torque) at the transition between the torque phase and the inertia phase of the upshift, when $T_{fc,est} > T_{fc,saturation,IP,US,C2}$, $T_{m,ref,TP,US,C3}$ is switched back to the level imposed by the driver torque demand $DTD(t)$. During this transition careful tuning of the control parameters have to be carried out, paying particular attention to the dynamics of the friction clutch actuator and the electric motor. The system sensitivity to the clutch actuator dynamics has already been demonstrated in [20].

4.2 Downshift Control

As a first approximation, the downshift actuation sequence is a reverse of the upshift method, wherein the inertia phase precedes the torque phase. Downshifts in power-on are actuated following significant increases of driver torque demand DTD , in order to increase the amount of wheel torque. Due to the significantly lower frequency of downshifts in power-on in comparison with the upshifts (consequence of the usual gear selection algorithms), the control system adopted in the inertia phase is a simplified version of the one presented for the upshifts, even if potentially the control systems could be similar.

Downshifts in power-on are accomplished by initially opening the friction clutch at the rate allowed by the actuator dynamics. When the clutch transmissible torque is lower than the electric motor torque, the clutch starts slipping, giving origin to the inertia

phase. The motor torque is kept at the level requested by the user, whilst the friction clutch torque is controlled in order to produce the required acceleration level of the electric motor shaft. At the start of the inertia phase, the friction clutch actuator position should be carefully monitored due to the dynamic friction coefficient of the dry friction clutch being lower than the static friction coefficient. The manipulation of both motor torque and friction clutch torque (according to the ‘golden rule’ presented in section 4.1) during the inertia phase of the downshift in power-on would lead to the full controllability of both electric motor dynamics and vehicle acceleration, at the cost of a significantly increased complexity of the control system and the time required to tune its parameters for each vehicle application. The friction clutch can be momentarily re-engaged when the sprag clutch is about to connect (beginning of the torque phase of the downshift), in order to dampen the re-engagement of the sprag clutch and reduce any jerk.

5. Results

A non-linear vehicle simulation model has been implemented in order to evaluate the gearshift dynamics of the novel transmission system. The equations utilised to model the transmission dynamics have been presented in section 3.

The developed simulator is characterised by a detailed transmission efficiency model, based on the efficiency maps for the transmission and the final reduction ratio

(functions of torque, speed and operating temperature), created by the authors through specific models [22] and validated through experimental tests carried out on other transmission systems. These efficiency maps include the efficiency of the actual gear couplings, the losses due to the transmission bearings, the sprag clutch losses and the losses due to oil churning and windage. This permits to calculate an overall equivalent efficiency for the subsystem (transmission or final reduction ratio) that is then re-assigned, within the dynamic simulator, to the gear coupling of the first gear, second gear and final reduction ratio. Figures 8 and 9 plot the equivalent efficiency maps for the transmission (including the final reduction ratio) in first and second gear (characterised by the higher efficiency values).

Figure 8 – Efficiency map of the novel transmission system in first gear (including the efficiency relating to the final reduction ratio)

Figure 9 – Efficiency map of the novel transmission system in second gear (including the efficiency relating to the final reduction ratio)

The simulation model includes the estimation of the temperature dynamics of the electric powertrain components (air cooled transmission), and in particular the thermal dissipation within the friction clutch during the gearshifts, which are fed into the efficiency model. The model also considers the wear in the system and the play in the actuation of the clutch, evident in the delay between the motion of the pressure plate and the actual torque on the friction clutch. The transmission model includes the torsional dynamics of the half-shafts; the tyres are modelled using Pacejka's magic formula [23] with tyre relaxation length. The adopted vehicle dynamics simulator permits the detailed modelling of vehicle longitudinal dynamics and has already been experimentally validated by the authors on electric vehicle applications equipped with

single-speed transmission systems [7]. The main vehicle parameters adopted for the simulation results of Figures 10 to 15 are listed in Table A.1. The motor drive data was selected to have a high moment of inertia and slow reaction time to test the robustness of the proposed controllers in critical conditions.

5.1 Upshift

Figures 10 and 11 summarise the overall transmission system dynamics for an upshift at 80% of driver torque demand, carried out at a vehicle speed of 75 kph, in the constant power region of the electric motor drive and using control system ‘Control 3’.

Figure 10 – Electric motor drive dynamics and gear input torques during an upshift at 80% of driver torque demand

The phases of the upshift, namely ‘Upshift Request’ followed by the torque phase, inertia phase (defined by ‘Inertia Phase Start’ and ‘Inertia Phase End’) and the final motion of the actuator after the engagement of the second gear (defined by ‘Actuator Stop’), are evident in the graphs. The effect of the efficiencies and the moments of inertia of the components are also visible in the graphs, for example in the marginal difference between the electric motor torque and the input torques transmitted by gear one and two when a gear is engaged in Figure 10.

Figure 11 – Clutch dynamics during an upshift at 80% of driver torque demand

During the torque phase the electric motor torque demand is modified according to equation (4.6) and is increased in order to compensate for the reduction of vehicle acceleration induced by the torque transfer from first to second gear. Due to the sharp gradient in the reference electric motor speed at the start of the inertia phase, the feedforward and feedback controller shown in Figure 5 provokes a large decrease in electric motor torque (Figure 10). The friction clutch torque is ramped up to make the vehicle acceleration equal to the level of the vehicle acceleration in second gear, according to equation (4.5). In the second part of the inertia phase the rate of motor reference speed is reduced as shown in the graph of Figure 5, and consequently the

electric motor torque demand is increased. Moreover as the electric motor speed reduces, the maximum available torque increases due to the torque map of the electric motor (Figure 7). This justifies the difference between the first gear torque and the second gear torque in conditions of engaged gear in Figure 10, and represents the main peculiarity to be taken into account in the implementation of algorithms for gearshift control of electric powertrains.

The torque actually transmitted by the friction clutch and the maximum torque which can be potentially transmitted (transmissible torque) for an assigned clutch actuator displacement are the same when the clutch is slipping, as in Figure 11, whilst they differ when the clutch is engaged. This is evident after the inertia phase, when the transmissible torque of the friction clutch is increased due to the change between the dynamic and static friction coefficient of the clutch. Finally the actuator is moved to increase the friction clutch axial force and therefore the transmissible torque. Notably, Figure 11 shows that the friction clutch torque transmitted in second gear is lower than that transmitted by the sprag clutch in first gear due to the friction clutch being located on the primary shaft whilst the sprag clutch is located on the secondary shaft as illustrated in Figure 3.

In addition, Figure 11 illustrates that the pressure plate moves when the shift is initiated to take up any play between the clutch plates. After recovering the play, during the torque and inertia phases of the upshift there is an infinitesimal axial movement of the

pressure plate due to the high axial stiffness of the clutch plates, although the transmissible torque varies with the friction clutch actuator travel, which depends on the stiffness properties of the Belleville spring.

Figure 12 – Half-shaft torque dynamics during the same upshift of Figures 10 and 11 (constant power region of the electric motor), and during an upshift at 40% of driver torque demand and 30 kph (constant torque region of the electric motor)

Figure 12 plots the time history of half-shaft torques (sum of the torques of the left and right half-shafts) for the same manoeuvre as in Figures 10 and 11, and during an upshift at 40% of driver torque demand, carried out at 30 kph, in the constant torque region of

the electric motor drive. Both manoeuvres have been simulated by adopting 'Control 3'. The marginal torque gap induced by the torque phase of the upshift and recovered by the friction clutch control during the inertia phase is evident in the first manoeuvre. The second manoeuvre is characterised by the total absence of any torque gap. This characteristic is common to 'Control 1', 'Control 2' and 'Control 3', when the upshift is requested in the constant torque region of the electric motor unit. The equivalent moment of inertia of the transmission is very high in first gear and very low during the inertia phase and it is this change which provokes some marginal oscillations in the half-shaft torque.

Figure 13 compares the vehicle acceleration profiles achievable during two different upshifts at 40% (different manoeuvre from Figure 12) and 80% (identical manoeuvre to Figures 10 to 12) of driver torque demand, both in the constant power region of the electric motor drive. Table 3 provides an objective comparison of the three gearshift strategies during each manoeuvre.

Figure 13 – Upshifts in the constant power region of the electric motor drive, at 40% (different from the upshift in Figures 7-9) and 80% (the same as in Figures 7–9) of driver torque demand: vehicle acceleration profiles achievable with ‘Control 1’, ‘Control 2’ and ‘Control 3’

Torque Demand		40 – 100 km/h	70 – 100 km/h	Upshift time	Mean acceleration during upshift	Mean jerk during upshift
40%	Control 1	14.91 s	9.55 s	1.11 s	0.76 m/s ²	0.95 m/s ³
	Control 2	14.71 s	9.34 s	1.35 s	1.00 m/s ²	1.18 m/s ³
	Control 3	14.68 s	9.32 s	1.39 s	1.01 m/s ²	0.41 m/s ³
80%	Control 1	6.53 s	4.03 s	0.95 s	1.82 m/s ²	2.24 m/s ³
	Control 2	6.33 s	3.84 s	0.70 s	2.25 m/s ²	3.22 m/s ³
	Control 3	6.31 s	3.82 s	0.74 s	2.31 m/s ²	2.15 m/s ³

Table 1 – Comparison of ‘Control 1’, ‘Control 2’ and ‘Control 3’ during the same upshift manoeuvres of Figure 13 (in the constant power region)

‘Control 2’ shows most of the benefit from the viewpoint of the acceleration time (a gain of more than 0.2 s when compared to ‘Control 1’), however the mean of the absolute value of jerk (time derivative of vehicle acceleration) during the simulation of the upshift is higher than for ‘Control 1’, with an even more significant disadvantage in terms of peak value of jerk. ‘Control 3’ represents the best compromise between a high performance upshift and the requirement for the expected comfort level. At 80% of torque demand, the benefit of ‘Control 3’ is much more limited than at 40% of torque demand, due to the fact that the torque increase specified by ‘Control 3’ is saturated at the peak torque of the electric motor (at 100% of motor torque demand, ‘Control 3’ produces the same performance as ‘Control 2’).

5.2 Downshift

Figure 14 summarises the torque and speed dynamics for a power-on downshift (kick-down) during a tip-in test (a sudden driver torque demand request) from 25 kph where the final driver torque demand is 80%. The downshift takes place in the constant torque region of the electric motor, and so is performed to provoke an increase in available wheel torque. Figure 14 illustrates the oscillations (circled) in the speeds due to the effect of the tip-in manoeuvre on the torsional dynamics of the half-shafts. The transmitted second gear torque reduces at the start of the inertia phase due to the change in the friction coefficient from the static to the dynamic value. This is also evident in

Figure 15 which shows that the change is partially compensated by the motion of the friction clutch actuator. At the end of the inertia phase, when the motor speed is at the required level and the sprag clutch is engaged, the friction clutch torque is progressively reduced to zero during the torque phase.

Figure 14 – Speed and torque dynamics during a downshift in power-on for a tip in test at an initial speed of 25 kph and a final 80% driver torque demand

Figure 15 – Clutch dynamics during a downshift in power-on for a tip in test at an initial speed of 25 kph and a 80% driver torque demand

6. Conclusion

Multiple-speed transmissions for electric powertrains with a central electric motor give rise to significant performance and energy efficiency benefits in comparison with conventional single-speed transmissions. Specifically a two-speed transmission system represents the best compromise between the advantages of a multiple-speed transmission system and the simplicity of a compact and lightweight drivetrain. This article has presented a novel two-speed transmission system design, together with the equations governing its dynamics. Three alternative gearshift control systems have been outlined, with particular reference to the typical characteristics of electric powertrains,

which require novel control algorithms for a seamless management of the upshifts within the constant power region of the electric motor drive. A comprehensive set of simulation results has demonstrated the functionality of the implemented control system and mechanical hardware, currently under experimental testing.

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Appendix A – Main Vehicle Parameters

The simulation results presented in Figures 10-15 refer to a front-wheel-drive vehicle characterised by the following parameters.

	Unit	Value
m	[kg]	1785
<i>Mass Distribution</i>	[% _{front} /% _{rear}]	52/48
L	[m]	2.80
R_W	[m]	0.34
$T_{m,MAX}$	[Nm]	450
$n_{m,base}$	[rpm]	3400
i_1	[-]	3.12
i_2	[-]	1.74
i_{diff}	[-]	3.00
J_{mot}	[kgm ²]	0.18
τ_m	[ms]	120
τ_{FC}	[ms]	70
$k_{L,HS}$	[Nm/rad]	8500
$k_{R,HS}$	[Nm/rad]	11700

Table A.1 – List of the main vehicle and transmission parameters

Appendix B – List of Notations

DTD : driver torque demand (equivalent to the throttle command for a conventional internal combustion engine driven vehicle)

i_{diff} : final reduction ratio

i_1 : first gear ratio

i_2 : second gear ratio

n_{base} : electric motor base speed

J_{diff} : equivalent moment of inertia of the differential

$J_{eq,trans,gear1}$: equivalent moment of inertia (at the differential) of the transmission in first gear

$J_{eq,trans,gear2}$: equivalent moment of inertia (at the differential) of the transmission in second gear

$J_{eq,trans,IP}$: equivalent moment of inertia (at the differential) of the transmission during the inertia phase

J_{LHS} : moment of inertia of the left half-shaft

J_{mot} : moment of the electric motor

J_{RHS} : moment of inertia of the right half-shaft

J_I : moment of inertia of the input shaft of the transmission (primary shaft), as indicated in Figure 3

J_{Ib} : moment of inertia of the part of the friction clutch (located on the transmission input shaft) gearing with the secondary shaft of the transmission, as indicated in Figure 3

J_2 : moment of inertia of the output shaft of the gearbox (secondary shaft), as indicated in Figure 3

J_{2b} : moment of inertia of the part of the sprag clutch gearing with the primary shaft of the transmission, as indicated in Figure 3

$k_{L,HS}$: torsion stiffness of the left half-shaft

K_P, K_D, K_I : proportional, derivative and integral gains of the electric motor speed controller during the inertia phase of the upshift
 $k_{R,HS}$: torsion stiffness of the right half-shaft

L : vehicle wheelbase

m : vehicle mass

$n_{m,base}$: electric motor base speed

R_W : electric motor base speed

s : Laplace variable

t : time

T_{fc} : friction clutch torque

$T_{fc,dis}$: friction clutch torque value required for the disengagement of the sprag clutch during an upshift

$T_{fc,est}$: estimated friction clutch torque

$T_{fc,saturation,IP,US}$: saturation level of the reference friction clutch torque during the inertia phase of the upshift

$T_{fc,saturation,IP,US,C2}$: saturation level of the reference friction clutch torque during the inertia phase of the upshift, according to ‘Control 2’

T_h : half-shaft torque (sum of the left and right contributions)

t_{IP} : time from the inertia phase start

$t_{IP,end}$: expected duration of the inertia phase

T_m : electric motor torque

$T_{m,MAX}$: electric motor peak torque

$T_{m,ref}$: reference torque for the electric motor (Figure 4)

$T_{m,ref,TP,US,C3}$: electric motor reference torque during the torque phase of the upshift according to ‘Control 3’

T_{sc} : sprag clutch torque

$V_{Max,SS,0\%}$: maximum speed of the single-speed vehicle at 0% road grade

$V_{Max,TS,0\%}$: maximum speed of the two-speed vehicle at 0% road grade

y : adimensional parameter for the definition of the electric motor speed profile during upshift

η_1 : first gear efficiency

η_2 : second gear efficiency

η_{diff} : final reduction gear efficiency

$\dot{\theta}_{base}$: base speed of the electric motor drive (speed at the transition between the constant torque region and the constant power region)

θ_{diff} $\dot{\theta}_{diff}$ $\ddot{\theta}_{diff}$: differential angular position, velocity and acceleration

$\dot{\theta}_{m,ref}$: electric motor angular speed reference during the inertia phase of the upshift

$\dot{\theta}_m$ $\ddot{\theta}_m$: electric motor angular speed and acceleration

τ_{FC} : time constant of the friction clutch actuator

τ_m : time constant of the electric motor drive

In the frequency domain, the parameters have the same notation as in the time domain, with a horizontal line at the top.