

A Novel Seamless 2-Speed Transmission System for Electric Vehicles: Principles and Simulation Results

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ABSTRACT

This article deals with a novel 2-speed transmission system specifically designed for electric axle applications. The design of this transmission permits seamless gearshifts and is characterized by a simple mechanical layout. The equations governing the overall system dynamics are presented in the paper. The principles of the control system for the seamless gearshifts achievable by the novel transmission prototype - currently under experimental testing at the University of Surrey and on a prototype vehicle - are analytically demonstrated and detailed through advanced simulation tools. The simulation results and sensitivity analyses for the main parameters affecting the overall system dynamics are presented and discussed.

INTRODUCTION

Due to the nature of the torque/speed characteristic of typical electric vehicle traction motors, which usually exhibit a high constant torque from zero to base speed then entering into a constant power region for higher angular velocities, electric vehicles are usually equipped with only a single-speed gearbox in order to minimize the drivetrain mass, volume, losses and cost [1, 2, 3].

Despite the wide operational speed range of such traction motors (e.g. from 0 to 8,000-12,000 rpm), a 2-speed gearbox [4-5] may be employed in order to increase the wheel torque at low vehicle velocity and, therefore, increase the maximum

road gradient that the vehicle can ascend when transporting heavy payloads, for example, whilst also facilitating a reasonable top speed. Moreover, as in some cases the efficiency of the electric motor and inverter may vary significantly as a function of the operating torque and speed, the adoption of a 2-speed gearbox can also facilitate significant benefits in terms of the energy consumption by optimizing the distribution of the operating points of the electric motor/inverter over a given driving schedule; the selection of the gear ratio being 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; however this variation, for conventional transmission systems based on helical gears, is usually marginal in comparison with the potential benefit from the viewpoint of the overall powertrain efficiency. The benefit in terms of distribution of the operating points of the electric motor/inverter can be maximized through the adoption of a Continuously Variable Transmission (CVT) system, which, however, presents issues from the viewpoint of speed of response at launch [6], and transmission efficiency [7, 8, 9], due to the mechanical layout of CVTs and their constantly running electrically-driven hydraulic pump to provide the high pressure hydraulic fluid for the actuation system. 2-speed transmission systems appear to be the most suitable solution for efficient electric axles [10]. [11] and [12] summarize the optimization procedures implemented by the authors in order to achieve the best compromise between dynamic performance and energy

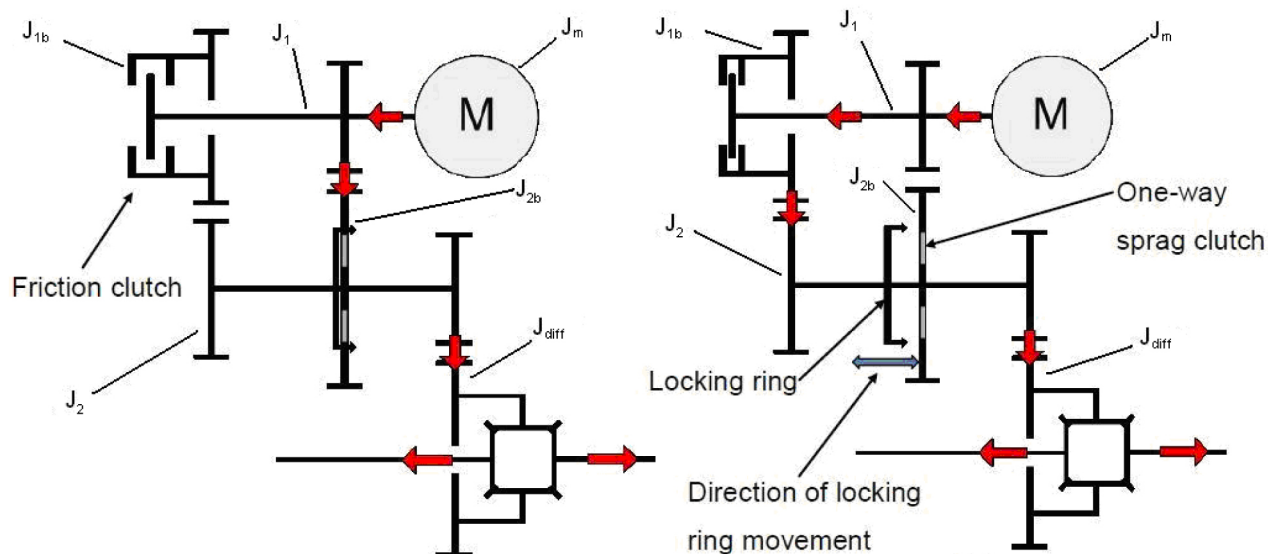


Figure 1. Schematic of the gearbox operation in 1st and 2nd gear

consumption in fully electric vehicles, through the design of the gear ratios and the gearshift map for 2-speed transmission systems for electric axles.

An important characteristic for multi-speed transmission systems to be competitive in fully electric vehicles, or in electric axles for parallel-through-the-road hybrid electric vehicles, is to permit seamless gearshifts, especially because a significant part of the gearshifts are going to be carried out in the constant power region of the motor characteristic, where a torque gap during an upshift could provoke a performance decay in comparison with a single-speed transmission. Seamless gearshifts can be achieved by using a cascade of planetary gear sets, as presented in [13], adopting a hydraulic actuation system for the control of the two friction clutches required for the second gear, the reverse gear and regeneration. The transaxle transmission in [13] is characterized by a first gear ratio of 15.52 and a second gear ratio of 10.15. A one-way clutch is also adopted to provide the torque reaction in first gear during traction. The motor rotor shaft is hollow allowing one axle shaft to pass through its center from the differential bevel gear assembly. An alternative seamless multi-speed gearbox layout for electric vehicle applications can be based on the adoption of a Dual Clutch system on a layshaft type gearbox, as described in [14]. This solution is becoming common for internal combustion engine driven vehicles, but is characterized by a significant mechanical complexity and requires a careful control of the actuation of the on-going and off-going friction clutches [15]. As a consequence, this layout is suitable for transmission systems characterized by at least 3 gear ratios, but specific more cost-effective solutions can be evaluated for seamless 2-speed transmissions for electric vehicles. In particular, [16] stresses the fact that 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. Alternative design solutions for the implementation of seamless gearshifts based on a novel design of the synchronizing rings and their actuation have not been subjected to large-scale testing and adoption yet [17]. The novel 2-speed transmission system of this paper combines the mechanical simplicity and low cost of a layshaft type gearbox, with the high quality of a clutch-to-clutch gearshift accomplished through a one-way clutch (sprag clutch) and a properly developed control system. The next paragraphs describe the mechanical layout of this novel transmission system, its governing dynamic equations, and analyze in detail the principles of the gearshift actuation control system, through a set of dynamic simulations.

THE NOVEL 2-SPEED TRANSMISSION

A 2-speed transmission, which employs a dual-stage spur-gear reduction to achieve the necessary gear ratios within a compact package envelope, has been developed from a VOCIS/Oerlikon Graziano-designed single-speed unit that has been previously installed in a light commercial electric vehicle demonstrator. In order to change gear, a patent-pending seamless shift system [18] that is both simple and compact, and which is able to achieve excellent shift quality, has been developed (Figure 1). This, therefore, overcomes one of the major drawbacks of existing multi-speed electric vehicle transmissions. The primary components of the shift system are a one-way sprag clutch and a friction clutch as displayed in Figure 1; the input torque being transmitted by the sprag clutch whilst in first gear, and by the friction clutch whilst in second gear.

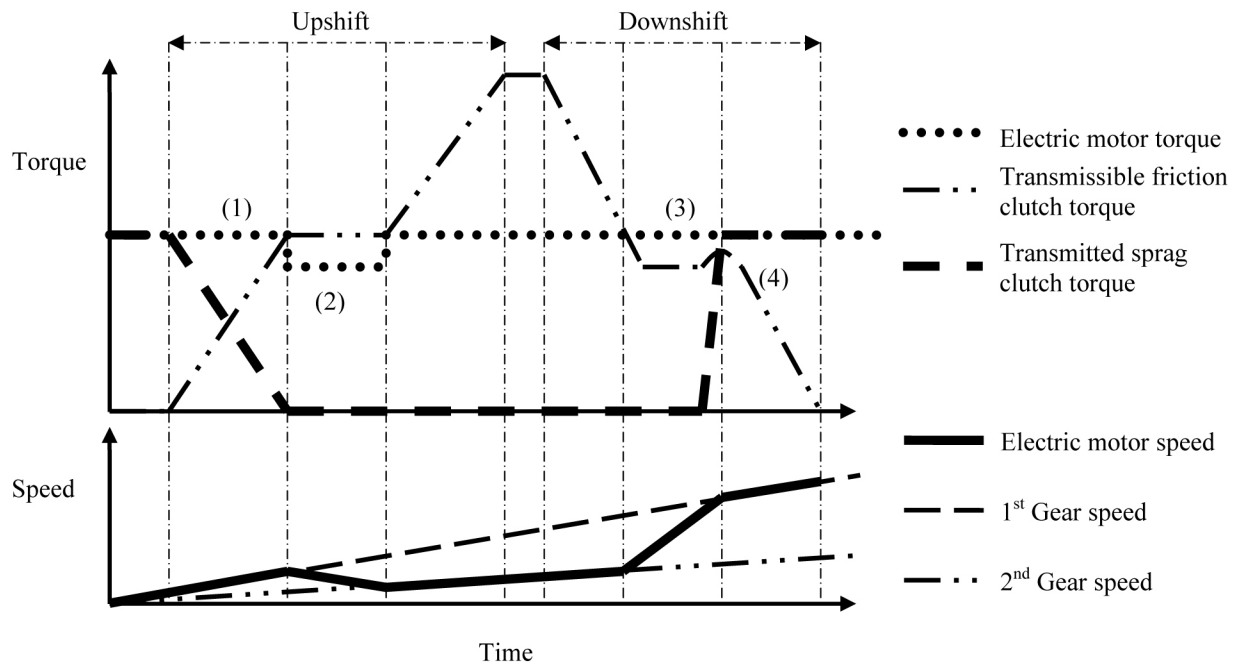


Figure 2. Illustration of a simplified sequence of seamless upshift and downshift, in conditions of constant torque demand

The clutch utilizes sintered metal friction material and is electro-hydraulically controlled using a remote brushless motor driven actuator, pressurizing a master cylinder mechanically connected to the Belleville spring of the friction clutch. In order to allow regenerative energy recovery whilst decelerating in first gear, the engagement of a locking ring 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 seamless gearshift process is illustrated in [Figure 2](#), where the vehicle starts in first gear with the sprag clutch engaged and an upshift and a downshift are implemented under constant driver torque demand.

- In order to shift to second gear, the locking ring is disengaged followed by progressive engagement of the friction clutch - phase (1) in [Figure 2](#) - to match the engine torque. This provokes a reduction of the torque transmitted through the sprag clutch (torque phase of the upshift).
- The motor torque is then reduced in order to provoke the required speed reduction - phase (2) in [Figure 2](#) - therefore causing the sprag clutch to overrun (inertia phase of the upshift) and finally the friction clutch is fully engaged.
- Downshifts in power-on are accomplished by opening the friction clutch which, being at a constant motor torque, increases the motor speed - phase (3) in [Figure 2](#) (inertia phase of the downshift).
- The friction clutch is momentarily re-engaged when the sprag clutch is about to connect, in order to dampen the re-

engagement of the sprag clutch and reduce any jerk - phase (4) in [Figure 2](#).

THE MODEL

A specific simulation model has been implemented in order to verify the dynamic performance of the transmission and design the control system for the novel gearbox layout. The implemented model describes the whole powertrain, including inverter, electric motor, the whole system thermal dynamics (particularly relevant from the viewpoint of the efficiency computation), and the vehicle longitudinal dynamics. The level of detail of the simulator is sufficient to enable the evaluation of the overall vehicle drivability, influenced by the torsion dynamics of the half-shafts and tire slip dynamics, modeled through Pacejka Magic Formula and a relaxation length model [19]. In this article, only the transmission model is detailed, as it constitutes the focus of the research.

The transmission can work in the following states:

- First gear, with locking ring engaged or disengaged, and friction clutch at zero torque or transmitting torque in condition of slip;
- Gearshift;
- Second gear.

The next paragraphs detail the equations describing the transmission system dynamics in the 3 different states.

FIRST GEAR

From the viewpoint of the modeling equations (but not of the transmission control unit), the system is in first gear when the ratio between the speed of the output and the input shaft of the transmission is equal to the first gear ratio i_1 . The primary shaft of the transmission, having a moment of inertia J_1 , which rotates together with the motor shaft (with inertia J_m

and an acceleration level $\ddot{\theta}_m$), is subjected to the motor torque T_m and the friction clutch torque T_{fc} , according to the following torque balance equation:

$$T_m - T_{fc} - T_{g1} = (J_m + J_1)\ddot{\theta}_m \quad (1)$$

where T_{g1} is the torque through the first gear. In first gear, the driven inertia J_{2b} and the secondary shaft inertia J_2 rotate together. As a consequence, the moment balance equation for the secondary shaft (experiencing an acceleration $\ddot{\theta}_2$) of the transmission is:

$$T_{g1}i_1\eta_1 + T_{g2}i_2\eta_2 - T_{output_shaft} = (J_2 + J_{2b})\ddot{\theta}_2 \quad (2)$$

where η_1 is the first gear efficiency and η_2 is the second gear (having a ratio i_2) efficiency. T_{output_shaft} is the output torque of the secondary (output) shaft of the gearbox. The torque balance equation for the inertia J_{1b} is:

$$T_{fc} - T_{g2} = J_{1b}\ddot{\theta}_{1b} \quad (3)$$

The moment balance equation for the differential (including half of the moment of inertia of the half-shafts $0.5 J_{LHS} + 0.5 J_{RHS}$, whilst the other half is incorporated in the wheel moment balance equation, not detailed in this paper) is:

$$T_{output_shaft}i_{diff}\eta_{diff} - T_h = (J_{diff} + 0.5J_{LHS} + 0.5J_{RHS})\ddot{\theta}_{diff} \quad (4)$$

where T_h is the half-shaft torque (sum of the torques of the left and right half-shafts, having moments of inertia respectively equal to J_{LHS} and J_{RHS}), i_{diff} is the final reduction ratio and η_{diff} is its efficiency. By combining Equations (1), (2), (3), (4) and using the definition of the gear ratios and the related kinematical relationships ($\ddot{\theta}_m = \ddot{\theta}_2i_1 = \ddot{\theta}_{diff}i_{diff}i_1$, $\ddot{\theta}_{1b} = \ddot{\theta}_2i_2$), the overall equation for the differential acceleration can be obtained:

$$\ddot{\theta}_{diff} = \frac{T_m i_1 \eta_1 i_{diff} \eta_{diff} + T_{fc} i_2 \eta_2 i_{diff} \eta_{diff} - T_{fc} i_1 \eta_1 i_{diff} \eta_{diff} - T_h}{(J_{diff} + 0.5J_{LHS} + 0.5J_{RHS}) + (J_m + J_1)i_1^2 \eta_1 i_{diff}^2 \eta_{diff} + J_{1b}i_2^2 \eta_2 i_{diff}^2 \eta_{diff} + (J_2 + J_{2b})i_{diff}^2 \eta_{diff}} \quad (5)$$

During normal operation of the first gear, $T_{fc} = 0$. In order to activate an upshift to the second gear, T_{fc} is ramped up through the electro-hydraulic actuation system, whilst the locking ring is disengaged. The system still operates in first gear until the sprag clutch torque T_{sc} is positive. During the ramp up phase (torque phase) of the friction clutch torque, the system experiences a torque variation at the differential due to the term $T_{fc}i_2\eta_2i_{diff}\eta_{diff} - T_{fc}i_1\eta_1i_{diff}\eta_{diff}$ of Equation (5), as part of the torque is transmitted to the output shaft of the transmission through gear 1 (torque contribution T_{g1}) and partially through gear 2 (torque contribution T_{g2}). This provokes (in case of a constant motor torque) a progressive torque decrease on the output shaft as $i_1 > i_2$, so that, when T_{sc} is null, the system transmits the entire torque through the second gear.

In first gear, the efficiencies η_1 and η_2 are mapped as functions of the torques T_{g1} and T_{g2} transmitted by the corresponding gears, the primary shaft speed and the estimated transmission temperature. In particular, T_{g1} can be computed from Equation (1), whilst T_{g2} from Equation (3). The model reverses the value of the efficiency as a function of the direction of the mechanical power flux. A similar procedure is adopted for η_{diff} . The efficiency values adopted for the simulation study presented in this paper have been derived from a detailed efficiency model developed by Oerlikon Graziano [20] and validated through experimental tests carried out on other transmission systems. This efficiency model considers the efficiency of the actual gearing, the losses due to the transmission bearings and the losses due to oil churning and windage. This permits to calculate an overall efficiency that is then reassigned, within the dynamic simulator, to a single gear coupling. This implies an approximation which will be overcome by the next generation of transmission models, which are going to include the computation of the single loss factor in the dynamic equations of the system.

GEARSHIFT (INERTIA PHASE)

From the viewpoint of the modelling equations, the gearshift phase is coincident with the inertia phase of the overall gearshift actuation process, when the ratio between motor speed and output shaft speed is intermediate between i_2 and i_1 . The features of this novel transmission permit to eliminate

the torque gap during upshift and downshift in power-on. An upshift in power-on starts when the friction clutch torque leads to the absence of any torque transmitted by the sprag clutch:

$$T_{sc} = [T_m - T_{fc} - (J_m + J_1)\ddot{\theta}_m]i_1\eta_1 - J_{2b}\ddot{\theta}_{diff}i_{diff} = 0 \quad (6)$$

As a consequence, the angular speed of the motor starts being different from the value it should have in the first gear ratio and the motor and differential dynamics differ from each other as well. The upshift is completed, from the viewpoint of the model, when the speed of the electric motor, properly controlled by the transmission control unit, reaches the value for the second gear and the friction coefficient of the clutch experiences the transition from its dynamic value μ_{dyn} to its static value μ_{static} , which is significantly higher than μ_{dyn} .

A downshift in power-on starts when the friction clutch torque is reduced until $T_{fc} - T_m$ is negative, so that the motor accelerates up to the level of the first gear and the sprag clutch starts transmitting torque, as per Figure 2. During the gearshift, the transmission system can be described as a two degree of freedom system (in the other two states, the transmission has only one degree of freedom). In this condition, the electric motor and primary shaft equation

$$T_{g1} = \frac{J_{2b}\ddot{\theta}_m}{i_1^2\eta_1}$$

remains the same as per Equation (1) whilst a consequence, the equation describing motor dynamics is:

$$\ddot{\theta}_m = \frac{T_m - T_{fc}}{\frac{J_{2b}}{i_1^2\eta_1} + J_m + J_1} \quad (7)$$

From Equation (7), the electric motor dynamics during the gearshift can be controlled by the difference between T_m and T_{fc} . The moment balance equation on the secondary shaft of the transmission is:

$$T_{g2}i_2\eta_2 - T_{output_shaft} = J_2\ddot{\theta}_2 \quad (8)$$

By combining Equations (8), (3) and (4), the dynamic equation of the differential during the gearshift is obtained:

$$\ddot{\theta}_{diff} = \frac{T_{fc}i_2\eta_2i_{diff}\eta_{diff} - T_h}{(J_{diff} + 0.5J_{LHS} + 0.5J_{RHS}) + J_{1b}i_2^2\eta_2i_{diff}^2\eta_{diff} + J_2i_{diff}^2\eta_{diff}} \quad (9)$$

The differential receives the friction clutch torque through the second gear. The friction clutch torque is the parameter which needs to be controlled during the gearshift in order to achieve the desired vehicle acceleration profile, without any torque gap (or marginal torque gap in actual conditions). The moment of inertia of the system in this condition is much lower than in Equation (5), mainly because Equation (9) does not include motor inertia. In particular, for the specific case study vehicle dealt with in this article, the equivalent moment of inertia of the transmission at the differential during the gearshift is about 1/30 of the moment of inertia in first gear and about 1/10 of the moment of inertia in second gear. This makes the natural frequency of the transmission during the gearshift much higher than when the system works either in first or second gear. In conclusions, the fundamental parameters to be controlled during upshifts and downshifts are the electric motor torque T_m and the friction clutch torque T_{fc} . The model considers the electric dynamics of the motor through a first order transfer function (time constant equal to 0.04 s for all the simulation results presented in this paper), whilst the dynamic properties of the friction clutch actuation system are considered through the experimental reaction time (8 ms for all the simulation results of this paper) and the rise time of the actuator. The model includes the details of the clutch actuation system, controlling the position and permitting a precise modulation of the friction clutch torque through the Belleville spring. The model permits to analyze the variation of the actuation system motion as a function of the wear condition of the disks of the friction clutch.

SECOND GEAR

In second gear, the overall transmission is characterized by one degree of freedom, as in first gear. The moment balance equation on the primary shaft is:

$$T_m - T_{g2} - T_{g1} = (J_{1b} + J_m + J_1)\ddot{\theta}_m \quad (10)$$

The dynamics of the outer part of the sprag clutch is described by:

$$T_{g1}i_1\eta_1 = J_{2b}\ddot{\theta}_{2b} \quad (11)$$

The secondary shaft dynamics is modelled by:

$$T_{g2}i_2\eta_{g2} - T_{output_shaft} = J_2\ddot{\theta}_2 \quad (12)$$