

Selection of the Optimal Gearbox Layout for an Electric Vehicle

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ABSTRACT

The paper describes the advantages due to the adoption of multi-speed transmission systems within fully electric vehicles. In particular, the article compares a conventional single-speed transmission layout, a 2-speed layout based on a novel gearbox architecture capable of seamless gearshifts, and a Continuously Variable Transmission layout. The selection of the optimal gear ratios for the 2-speed system has been based on an optimization procedure, taking into account the efficiency characteristics of the components of the whole vehicle powertrain. The control system for the Continuously Variable Transmission system has been designed with the aim of maximizing the efficiency of the operating points of the electric motor. The results show that there is a significant advantage in adopting a 2-speed transmission system over the single-speed layout, and, despite a reduction in motor losses, the typically lower efficiency characteristics of Continuously Variable Transmissions do not result in lower energy consumption for the four case study vehicles.

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 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 conventional multi-speed, helical gear transmission system is generally lower than the efficiency of a similar single-speed transmission, however this variation is usually marginal in comparison with the potential benefit from the viewpoint of the overall powertrain efficiency. In theory, therefore, the benefit in terms of the electric motor/inverter operating point distribution can be maximized through the adoption of a Continuously Variable Transmission (CVT) system. However, due to their mechanical layout and the need to provide the high pressure hydraulic fluid for the actuation system, CVTs transmission efficiency [6, 7, 8] is typically lower and they can suffer from

poor speed of response, particularly at launch [9], for example.

Some studies have proposed or experimentally demonstrated alternative approaches to optimizing electric vehicle motor operational efficiency. For example, [10] suggests the adoption of two driven axles in order to optimize the efficiency of the front and rear motoring units so that the overall powertrain efficiency remains high for the different vehicle operating conditions. A more advanced implementation of the same idea is presented in [11], where a method for optimizing the torque applied by each motor of a dual drive system is discussed and demonstrated. The two motor units are located on the same axle and are coupled to the differential through a novel transmission system. Although these solutions have their merits, their drawbacks include the inability to torque vector despite employing two motors in the case of [11], significantly greater volume and component count, control complexity, and therefore cost. [12] analyzes the effect of the DC bus voltage on the efficiency of the motoring unit, with the findings based on a case study of a PWM driven brushless DC electric machine. The outcome is that a higher bus voltage improves powertrain efficiency in some, but not all, operating conditions.

In conclusion, the literature review shows that it is desirable to be able to control the operating condition of the motoring unit for an assigned driver demand - which is a power demand - and that, at the current stage of technology development, the most cost effective solution for achieving this target is based on mechanical multi-speed gearboxes. The adoption of multi-speed transmission systems could enhance the competitiveness of on-board motor solutions compared to combined in-wheel motor and single-speed transmission solutions that are currently under development by several manufacturers [13]. In-wheel solutions still represent a challenge from the viewpoint of the package and the additional contribution to the vehicle unsprung mass, although the latter has been evaluated as marginal in [14] and [15]. In any case, it is currently not practicable to equip vehicles with in-wheel motor-based solutions that employ multiple-speed transmission systems for, predominantly, reasons of package. As a consequence, the adoption of multiple-speed electric axles, with either a single central motor or twin-motor layout [16] for torque vectoring, can be competitive against the in-wheel layouts, from the viewpoint of their energy efficiency.

Ren et al. [17] have so far carried out the only simulation-based analysis of alternative transmission layouts for fully electric vehicles. However, the vehicle and powertrain parameters (in particular, the efficiency maps) adopted for the research were not related to any particular real world application, hence, due to the lack of appropriate data, can not give an insight into the overall system sensitivity to the variation of transmission parameters. [18] and [19] are the

first quantitative publications - based on the data of a real vehicle - of optimizing the gear ratios for a fully electric vehicle. However, they are limited to a single case study vehicle, and they do not discuss the merits of a CVT, although [18] does introduce the possibility of motor downsizing - in terms of its peak torque - due to the adoption of a multiple-speed gearbox.

This paper provides the comparison, in terms of energy consumption and performance, between vehicles equipped with either a single-speed gearbox, or an innovative seamless 2-speed gearbox (described in detail in the paper), or a CVT system, for electric axle layouts with a single central motoring unit. The analysis is carried out by adopting vehicle models including specific procedures for the optimization of vehicle performance and energy efficiency, through the variation of gearbox parameters (ratios and gearshift maps). The comparison between the three alternative transmission systems is carried out for four different case study vehicles, three of them characterized by a front-wheel-drive layout, and one of them characterized by a rear-wheel-drive layout. Three of the four vehicles are equipped with the same electric motor/inverter unit. The paper describes the details of the procedures, the adopted cost functions, discusses the results, and justifies the selection of the optimal transmission layout.

THE SINGLE AND 2-SPEED TRANSMISSION ARCHITECTURES

A 2-speed transmission, which employs a dual-stage helical-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 (Figure 1) that has been previously installed in a light commercial electric vehicle demonstrator. In order to change gear, a seamless shift system that is both simple and compact, and which is able to achieve excellent shift quality without the need for a complex control system, has been developed (Figure 2). This, therefore, overcomes one of the major drawbacks of existing multiple-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 3; the input torque being transmitted by the sprag clutch whilst in first gear, and by the friction clutch whilst in second gear.

The clutch utilizes sintered metal friction material and it is electro-hydraulically actuated using a remote brushless motor driven actuator. 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



Figure 1. Single-speed transmission cutaway



Figure 2. 2-speed transmission cutaway

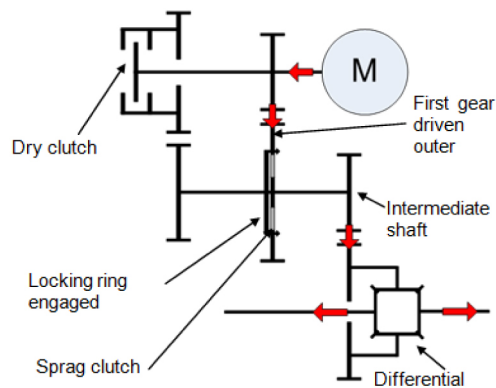


Figure 3. 2-speed transmission stick diagram

lock mechanism and actuator. The seamless gearshift process is illustrated in [Figure 4](#), where the vehicle starts in first gear with the sprag clutch engaged and an upshift and a downshift are implemented under constant electric motor torque.

- In order to shift to second gear, the locking ring is disengaged followed by progressive engagement of the

friction clutch - phase (1) in [Figure 4](#) - to match the engine torque. This provokes a reduction of the torque transmitted through the sprag clutch;

- The motor torque is then reduced in order to provoke the required speed reduction - phase (2) in [Figure 4](#) - therefore causing the sprag clutch to overrun and finally the friction clutch is fully engaged;

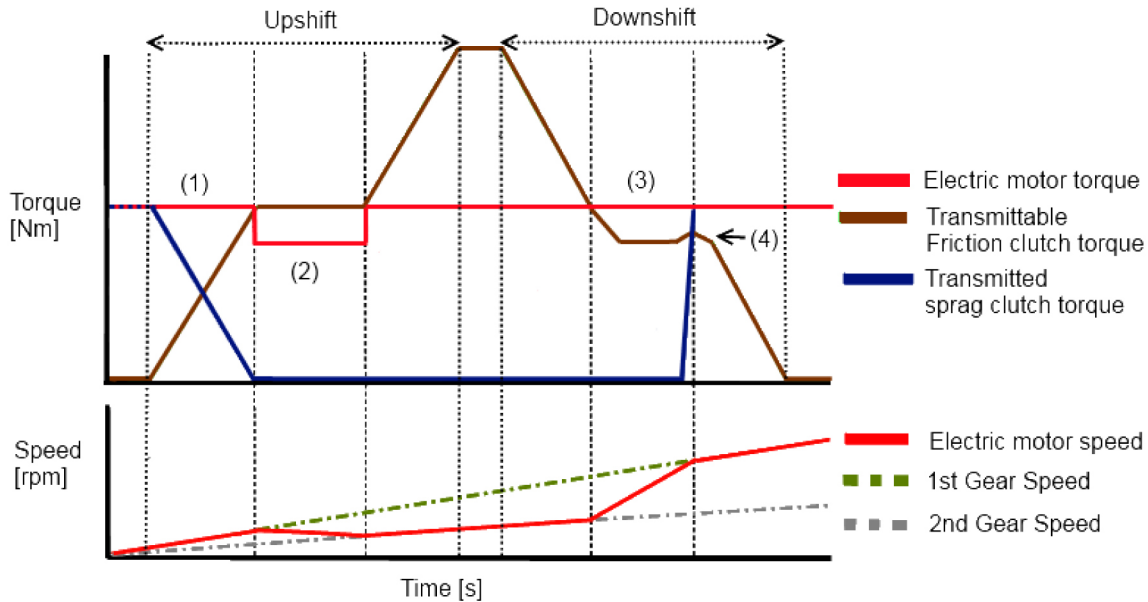


Figure 4. Illustration of a sequence of seamless upshift and downshift, in conditions of constant torque demand

- Downshifts are accomplished by opening the friction clutch which, being at a constant motor torque, increases the motor speed - phase (3) in [Figure 4](#);
- 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 4](#).

THE CASE STUDY VEHICLES

The activity has considered the conversion of 4 internal combustion engine powered vehicles into full electric vehicles. These vehicles are manufactured by 2 different OEMs and include:

- A front-wheel-drive electric minibus, which will be referred to as case study 1, to be equipped with a 70 kW (peak), 300 Nm (peak) electric motor unit, and is characterized by being able to operate in constant power above its base speed. The minimum required top speed for this vehicle is 120 km/h. A single-speed prototype of this vehicle is currently under test, whilst a 2-speed prototype - equipped with the optimized 2-speed transmission system - has been assessed during this project;
- A rear-wheel-drive urban delivery van, which will be referred to as case study 2, to be equipped with a 60 kW (peak), 260 Nm (peak) electric motor unit. Above base speed the motor unit is characterized by a constant power region (up to 5000 rpm), and a linearly decreasing power region up to 30 kW peak power at 9000 rpm. The electric motor and the transmission are to be longitudinally mounted and coupled to the differential through a bevel gear. The vehicle must be capable of a minimum operational top speed of 70 km/h. Once again, a single-speed prototype vehicle is already under test;

- A front-wheel-drive urban delivery van, characterized by the same performance requirements and electric motor unit as case study 2. This vehicle has been considered in two chassis variants, named respectively as case study 3 and case study 4.

THE SIMULATION MODELS

The activity is based on the adoption of the following three mathematical models:

- A basic model for the estimation and the optimization of the energy efficiency and, therefore, energy consumption of the system over standard and experimental driving schedules;
- A model for the estimation and the optimization of vehicle performance, considering its longitudinal acceleration, maximum speed, and the maximum road grade that the vehicle can ascend at a given velocity;
- An advanced model for the estimation of energy consumption, which is based on the model for the estimation of vehicle performance, but with the addition of an integrated feedback/feedforward driver model.

ENERGY EFFICIENCY MODEL

The model adopted for the estimation of the energy efficiency along driving schedules is, for reasons of computational efficiency, a backward-facing model, calculating the required electric motor torque, starting from the velocity profile of the assigned driving schedule. It predicts the power dissipation within the battery/supercapacitor/DC-DC converter, the electric motor and inverter, the gearbox (separated into layshaft and differential losses), the tires, the brakes, and that due to the aerodynamic losses. This model, in the same way as the most sophisticated models presented in the next

paragraphs, considers the individual component efficiencies as functions of temperature, transmitted torque and velocity, as well as the individual inertias of the electric motor, the gearbox components, and the half-shafts. The generation of the efficiency map of the transmission has been carried out by an experimentally validated efficiency model developed by Oerlikon Graziano. The efficiency maps of the electric motor and their temperature dynamics estimation are based on the experimental data obtained by the motor manufacturers on a test rig.

The energy efficiency model runs fast simulations and carries out a full optimization procedure on a standard Personal Computer (PC) in a few hours (typically 10 hours for each gearshift map). For example, an NEDC (duration 1160 sec) can be simulated in 90 sec on a PC equipped with a 2.3 GHz Dual Core Processor and 4 Gb RAM.

VEHICLE PERFORMANCE MODEL

This model is adopted for simulating longitudinal acceleration tests up to the maximum speed of the vehicle, at different road grades, within the performance optimization procedure. Due to the limited time duration of these tests, the model can be characterized by additional complexity in comparison with the energy efficiency model. The vehicle performance model is a forward-facing model. In addition to the features already present in the energy efficiency model, it includes:

- Tire longitudinal dynamics, through Pacejka's Magic Formula and a tire relaxation length model [20];
- The first order transmission dynamics, due to the simulation of the torsional vibrations of the half-shafts [21].

The combination of these characteristics permits to simulate, during a longitudinal acceleration test, the correct longitudinal acceleration/jerk dynamics of the system. For the specific vehicle application, the implementation of a Traction Control system [22] has been necessary in order to prevent wheel spinning in conditions of maximum electric motor torque and low payload.

Figures 5 and 6 are examples of the experimental validation of the vehicle dynamic performance model, which has been carried out on the single-speed vehicle of case study 2. Two test drivers have carried out several acceleration tests from standstill conditions. The results obtained in the different experimental tests have been averaged. Figures 5 and 6 plot the comparison between the experimental and simulated values of velocity, for a given distance covered by the vehicle from the starting line. The error is below the variance induced by the experimental procedure carried out by the test drivers. Potentially, this model also simulates low frequency drivability and can be used for the analysis of vehicle jerk dynamics.

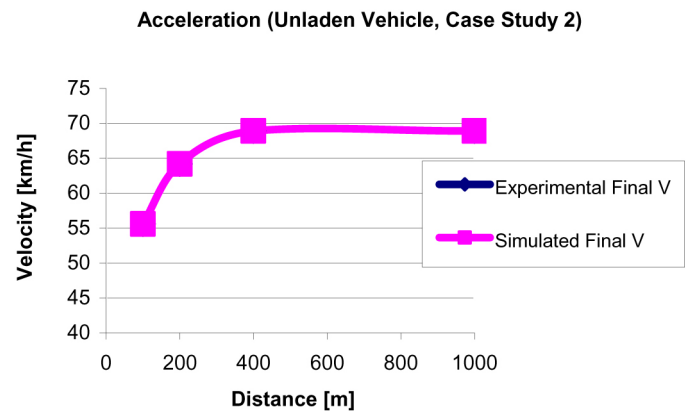


Figure 5. Experimental validation of the vehicle performance model

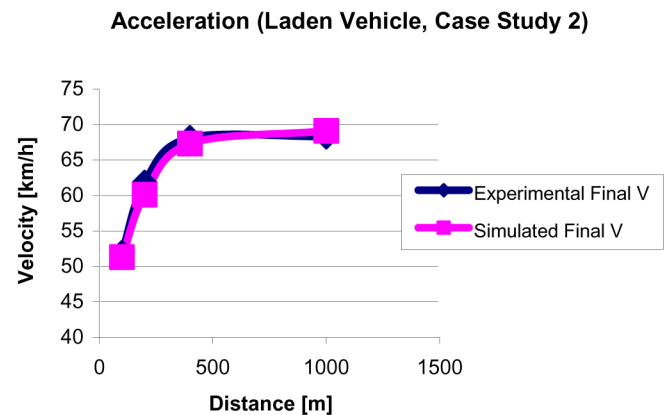


Figure 6. Experimental validation of the vehicle performance model

ENERGY EFFICIENCY MODEL INCLUDING DRIVER

In order to carry a more comprehensive analysis of the results and check their robustness, the driving schedules analyzed within the energy consumption optimization procedure have been re-simulated with a selection of gear ratio values close to the optimal ones, by using the forward-facing vehicle model, including a driver model.

The driver model has been implemented through a combination of a feedforward controller, generating a driver torque demand following the torque profile required to achieve the reference time history of vehicle speed, and of a feedback controller (Proportional, Integrative and Derivative, PID) comparing the actual speed and the reference speed. The controller parameters have been tuned in order to follow (where possible for the vehicle) the velocity profile within the tolerance limits specified by the regulations of the driving cycles, without significant and unrealistic oscillations of the torque demand. An automated warning is generated by the