

Optimization of a Multiple-Speed Transmission for Downsizing the Motor of a Fully Electric Vehicle

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ABSTRACT

The research presented in this paper focuses on the effects of downsizing the electric motor drive of a fully electric vehicle through the adoption of a multiple-speed transmission system. The activity is based on the implementation of a simulation framework in Matlab / Simulink. The paper considers a rear wheel drive case study vehicle, with a baseline drivetrain configuration consisting of a single-speed transmission, which is compared with drivetrains adopting motors with identical peak power but higher base speeds and lower peak torques coupled with multiple-speed transmissions (double and three-speed), to analyze the benefits in terms of energy efficiency and performance. The gear ratios and gearshift maps for each multiple-speed case study are optimized through a procedure developed by the authors consisting of cost functions considering energy efficiency and performance evaluation. The cost functions are explained in the paper along with the models adopted for the research.

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1. INTRODUCTION

A typical Fully Electric Vehicle (FEV) traction motor is generally combined with a single-speed transmission due to the distinctive torque / speed characteristics of an electric motor, which consist of a high constant torque region from zero to base speed and a constant power region above base speed. However, the required torque range on a passenger car usually varies between 1,000 and 4,000 Nm at the axle, depending on the size and type of the vehicle, and consequently a relatively high torque electric motor has to be employed to provide the required torque on a single-speed electric axle, without affecting vehicle top speed. The peak motor torque is related to the motor volume as it is proportional to the effective area of the gap between the rotor and stator, specifically it varies quadratically with the diameter and linearly with the overall length of the active parts [1]. As a result, a motor that would provide the required power to move a vehicle for a single-speed transmission would be relatively voluminous and massive. In addition, for such a powertrain layout, the single gear ratio has to be accurately selected in order to provide the required operational torques and to achieve the optimal trade-off

between adequate acceleration, gradeability and maximum velocity.

This paper, therefore, analyzes if a downsized electric motor, characterized by the same peak power as the high torque single-speed unit but with a lower peak torque and higher base speed, can be adopted to improve a FEV's performance and energy consumption. The adoption of a downsized motor requires a multiple-speed transmission to be fitted, as the lower peak torque necessitates a higher gear ratio to provide adequate acceleration and achievable road grade but will subsequently lower the attainable top speed. As a result further gears are required to increase the vehicle top speed. Furthermore, due to the efficiency of the electric motor drive varying significantly as a function of the operating torque, speed, voltage and temperature, the adoption of a multiple-speed transmission can also facilitate significant benefits in terms of the energy consumption by optimizing (an example of an optimization procedure is described in [2]) the distribution of the operating points of the electric motor 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 [3], [1].

The research presented in this paper considers a case study vehicle, fitted with single, double and three-speed transmissions. Each vehicle drivetrain consists of a motor with identical peak power (160 kW) but with peak torques of 890 Nm, 450 Nm and 390 Nm, and base speeds of 1,700 rpm, 3,400 rpm and 3,900 rpm respectively. All the motor drives have a limited (due to the layout of their power electronics) regenerative capability of 50 Nm. Each of the multiple-speed transmission case study vehicles underwent a comprehensive gear ratio and gearshift map optimization procedure to maximize the dynamic performance and drivetrain operating efficiency. The performance is optimized through a cost function taking into account a wide range of maneuvers, whilst the energy efficiency cost function considers the energy consumption over a number of carefully selected driving cycles [4] and autonomy test results. The case study transmissions adopted for the research are based on prototypes under testing at Oerlikon Graziano and the University of Surrey, with the single and double-speed transmissions being installed on a hardware-in-the-loop test rig at the University of Surrey and on a test vehicle powered by a 70 kW motor [5].

2. THE MODELS

To analyze each case study electric powertrain and to optimize the gear ratios / gear shift maps for the multiple-speed transmissions, according to the cost functions explained in section 3, three simulation models were developed:

- A basic model for the estimation of the energy efficiency and therefore the energy consumption of the system over standard driving schedules and autonomy tests. This model was adopted for the energy consumption cost function.
- A model for the estimation of vehicle performance, considering the vehicles longitudinal acceleration and maximum speed, and the maximum road grade that the vehicle can ascend at a given velocity. This model was adopted for the dynamic performance cost function.
- An advanced model for the estimation of the energy consumption, which is based on the vehicle performance model but with the addition of an integrated feedback / feedforward driver model, developed and implemented for final validation of the results obtained through the basic energy efficiency model.

2.1. CONSUMPTION MODEL

The basic model adopted for the estimation of the energy efficiency along driving schedules is, for reasons of computational efficiency, a backward-facing model [6], calculating the required electric motor torque, starting from the velocity profile of the assigned driving schedule or autonomy test. It predicts the power dissipation within the battery, the electric motor and inverter, the gearbox (separated into layshaft and differential losses), the tires, the

brakes, and that due to the aerodynamic losses. The model 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 efficiency maps and temperature dynamics of the 450 Nm and 890 Nm electric motors are scaled from the experimental data obtained by the motor manufacturer on a test rig for the smaller motor unit. Each of the transmission efficiency maps were estimated from approximate values obtained through Oerlikon Graziano. The model accounts for the different masses of the transmissions and motors for each vehicle.

2.2. 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, using the driver torque demand as the model input. 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 [7].
- The first order transmission dynamics, due to the simulation of the torsion of the half-shafts [8].

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 [9] has been necessary in order to prevent wheel spinning especially in conditions of high electric motor torque and low payload.

3. OPTIMIZATION PROCEDURES

A simple procedure for the analysis of multiple-speed transmissions for fully electric vehicles was presented by Ren in 2009 [10], however this procedure did not optimize the gear ratios for a multiple-speed transmission. The gear ratio optimization was achieved in a paper authored by Sorniotti [11], which the work in this paper builds on. [10] and [11] discuss the benefits of multiple-speed transmission systems for an assigned electric motor drive. The optimization procedure presented here deals with a different aspect of the same problem, which is the possibility of significant energy consumption savings and performance benefits from the viewpoint of electric motor downsizing.

This section describes the optimization procedures carried out to maximize the vehicle performance for the case study vehicles fitted with multiple-speed transmissions through optimizing the gear ratios and gearshift maps based on the minimization of two cost functions; a Dynamic Performance



Cost Function (*DPCF*), and an Energy Consumption Cost Function (*ECCF*).

The generic optimization procedure adopted for each cost function is illustrated in [Figure 1](#). Initially, the maneuvers are defined for each cost function, along with any constraints such as a maximum top speed which must be reached or certain acceleration times to be achieved for the results to be considered. The system then computes the results for each maneuver for each gear shift map and selection of gear ratios. The same calculations are also carried out for an optimized single-speed transmission vehicle. The results of the multiple-speed vehicle are then compared with the results obtained for the single-speed vehicle through the generation of the cost functions, with the cost function result being 1 if the average results for the single and multiple-speed vehicles are identical. The lower the resulting cost function the larger the improvement over the single-speed. The vehicle parameters are the same for each of the case study vehicles, apart from marginal variations due to the different electric motor drives, and can be found in [Table A.1](#) of the [Appendix](#), whilst the gear ratio for the single-speed was selected from an initial design procedure.

The cost function generation procedure is characterized by a system wherein once the maneuvers / constraints have been selected and the simulations have been carried out, the weightings / constraints for the cost functions can be reselected and the cost functions recalculated without running the simulations again, reducing the computation time. A full explanation of the cost functions is presented in sections 3.1 and 3.2.

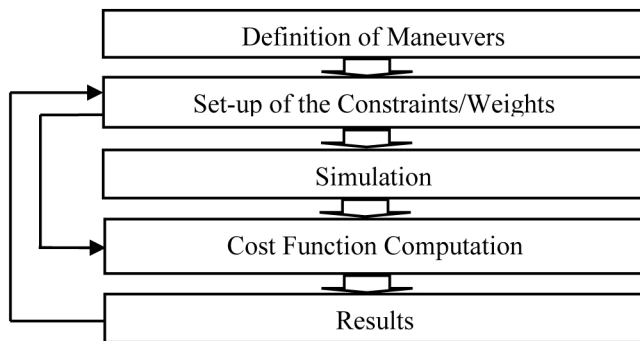


Figure 1. Framework for the derivation of the cost function values

3.1. DYNAMIC PERFORMANCE COST FUNCTION

The dynamic performance cost function (*DPCF*) is used to optimize the gear ratios and gear shift maps of the multiple-speed transmissions to maximize the vehicle performance over a number of set maneuvers. The maneuvers for the *DPCF* adopted in this research are as follows:

- Maximum ascendable road grade at 0, 20 and 40 kph.

- Vehicle acceleration times for a large number of maneuvers (as listed in [Table 1](#)) on horizontal road and on a 10% road grade.
- Maximum vehicle speed on horizontal road and on a 10% road grade.

The *DPCF* procedure optimizes the gear ratios for the double and three-speed transmission case study drivetrains. The double-speed vehicle requires the range between τ_1 (first gear ratio) and τ_2 (second gear ratio) to be set, whereas the three-speed vehicle requires τ_1 and τ_3 (third gear ratio) to be set. In the case of the three-speed transmission is τ_2 defined by the formula $(\tau_3/\tau_2)/(\tau_2/\tau_1)=X$, where the user has the option to fix X during the gear ratio optimization procedure or to optimize X within a range defined by the user. The optimization procedure will then test a range of gear ratio values around the initial values as set by the user.

The acceleration tests carried out for the *DPCF* are all accomplished at 100% of torque demand so only the upshift speed at 100% torque demand will be optimized through this procedure. In theory, the upshift is optimal if the electric motor remains in the constant power region above its base speed, n_{base} , after the shift. For example, for the double-speed transmission, the upshift must be performed at a minimum motor speed equal to $n_{base} \cdot (\tau_1/\tau_2)$, which in theory allows a seamless gearshift to take place, from the viewpoint of the vehicle acceleration levels before and after the upshift. The gear ratios and upshift points are constrained as the vehicle velocity, $v_{upshift}$, defining the upshift point needs to be at a motor speed lower than the maximum allowed $n_{max,motor}$, as defined by [Equation \(1\)](#).

$$\tau_{1max} = n_{max,motor} \cdot \frac{1}{v_{upshift}} \cdot \frac{R_w}{\tau_{diff}} \quad (1)$$

where R_w is the wheel radius and τ_{diff} is the differential ratio.

The following equation defines the cost function adopted for the evaluation of vehicle performance for a given road grade (expressed in x%):

$$CF_{x\%,perf.} = \sum_{j=1}^{17} w_j \frac{t_{j,x\%,multiple-speed}}{t_{j,x\%,single-speed}} + w_{Vmax} \frac{V_{x\%,single-speed}}{V_{x\%,multiple-speed}} \quad (2)$$

where w_j and w_{Vmax} are the relative weighting factors adopted for each test within the procedure (their sum being equal to 1), $t_{j,x\%,multiple-speed}$ being the acceleration time for the multiple-speed vehicle during an arbitrary maneuver j ($t_{j,x\%,multiple-speed}$).