Optimization of a Multi-Speed Electric Axle as a Function of the Electric Motor Properties

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***Abstract*-This paper deals with the advantages of a multi-speed transmission system for an electric axle, in comparison with a single-speed layout. The selection of the gear ratios for the multi-speed application is described in detail for a case study vehicle, for two different electric motor units.**

I. INTRODUCTION

Due to the nature of the torque/speed characteristic of typical electric vehicle traction motors, which 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]-[3]. However, despite the wide operational speed range of such traction motors (e.g. from 0 to 8,000-12,000 rpm), a 2-speed gearbox 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 whilst also facilitating a reasonable top speed [4]-[5]. Furthermore, as in some cases the efficiency of the electric motor and inverter may vary significantly (even more than in an internal combustion engine) as a function of the operating torque and speed, the adoption of a 2-speed gearbox can also provide 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. Thus the selection of the gear ratio is as important as that of a conventional vehicle driven by an internal combustion engine. The 2-speed gearbox adopted within the electric axle presented in this article is characterized by an innovative mechanical layout capable of seamless gearshifts which do not interrupt the transmission of the motor torque to the road wheels.

A clear procedure for the selection of the optimal gear ratios for an electric vehicle application is still absent in the existing literature. In addition, an analysis of the influence of the torque and efficiency characteristics of the motor on the choice of the gear ratios is missing. Hence, this paper presents an optimization procedure for the selection of the gear ratios for an electric axle application, equipped with two alternative real-world electric motors (called motor A and motor B) for automotive traction, which have significantly different torque characteristics (maximum torque and base speed), and levels of regenerative torque, equal to the maximum motoring torque in one case, and approximately to one sixth of the maximum motoring torque in the second case, despite similar power values (70 kW and 75 kW).

The cost functions implemented within the optimization procedure have been defined in order to improve the dynamic performance whilst also maximizing the system operating efficiency. They are based on the cumulative analysis of the results of several maneuvers (e.g. longitudinal acceleration tests at different payloads/road gradients and standard driving schedules).

The results for the two motors will be compared to each other. In particular, the correlation between the optimal gear ratios and the motor torque characteristic will be analyzed. The effectiveness of different levels of regenerative motor torques will also be discussed during typical driving schedules.

II. THE MODELS

The activity is based on 3 models:

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

A. 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, 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

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efficiency map of the transmission has been carried out by an experimentally validated efficiency model developed by Oerlikon Graziano – Automotive. The efficiency map of the electric motor and its 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.

B. 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 reduced time duration of these maneuvers, the model for simulating them can be more complex than the energy efficiency model, and can include the 1st order dynamics of the system.

The vehicle performance model is a forward-facing model. In addition to the features already present in the energy efficiency model, it includes:

1. Tire longitudinal dynamics, through Pacejka's Magic Formula and a tire relaxation length model [6];
2. The first order transmission dynamics, due to the simulation of the torsional vibrations of the half-shafts [7].

The combination of these characteristics allows the simulation of the longitudinal acceleration/jerk dynamics of the system during a longitudinal acceleration test. For the specific vehicle application, the implementation of a Traction Control system [8] has been necessary in order to prevent wheel spinning in conditions of maximum electric motor torque and low payload.

C. Energy Efficiency Model Including Driver

In order to carry out a more comprehensive analysis of the results, the driving schedules analyzed within the optimization procedure in *A.* have been re-simulated with a selection of gear ratio values close to the optimal ones by using the forward-facing vehicle model, but with the addition of a driver model. This has been implemented through the 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, and without significant and unrealistic oscillations of the torque demand. An automated warning is generated by the model in the areas where the vehicle cannot manage to follow the velocity profile of the assigned driving schedule.

III. The Optimization Procedures

This paragraph describes the optimization procedures carried out during this research project, based on the minimization of two cost functions, a Dynamic Performance Cost Function (*DPCF*), and an Energy Consumption Cost Function (*ECCF*).

A. Dynamic Performance Cost Function (DPCF)

In this section the 2-speed drivetrain vehicle, with either motor A or B, is compared with the same electric vehicle equipped with motor A and the single-speed transmission system. Motor A is the electric motor installed in a single-speed vehicle prototype currently under testing, the results are then used as a reference for the 2-speed optimization procedure detailed in this paper. The different masses of the electric powertrain components for the single-speed and the 2-speed vehicles are taken into account in the simulation data (e.g. the 2-speed layout implies a mass increase of 19 kg), along with the different efficiency maps of the two electric motors (which will be discussed in a later section) and gearboxes. The average efficiency of the first gear of the 2-speed system is 1.9% lower than the average efficiency of the single-speed gearbox, whilst the average efficiency of the second gear of the 2-speed system is only marginally lower (0.1%) than the average efficiency of the single-speed gearbox.

Hence, each combination of gear ratios for the 2-speed vehicle is evaluated (against the reference single-speed layout with motor A) with respect to the following parameters:

1. Maximum ascendable road grade at 0, 20 and 40 kph;
2. Vehicle acceleration times for 0-50 kph, 0-70 kph, 0-100 kph, 10-50 kph, 20-70 kph, 20-100 kph, 40-100 kph, on horizontal road and on a 10% road grade;
3. Maximum vehicle speed on horizontal road and on a 10% road grade.

The tests described in 1)-3) were conducted for both the unladen and the fully laden vehicle cases, and the following cost function was adopted for the evaluation of vehicle performance for a given road grade (expressed in *x*%):

$$CF_{x\%,perf.} = \sum_{j=1}^{7} w_j \frac{t_{j,x\%,2\text{-}speed}}{t_{j,x\%,single\text{-}speed,mot.A}} + w_{Vmax} \frac{V_{x\%,single\text{-}speed,mot.A}}{V_{x\%,2\text{-}speed}} \qquad (1)$$

where w_j and w_{Vmax} are the relative weighting factors adopted for each test within the procedure (their sum being equal to 1), $t_{j,x\%,2\text{-}speed}$ being the acceleration time for the 2-speed system during an arbitrary maneuver *j*, $t_{j,x\%,single\text{-}speed,mot.A}$ being the acceleration of the single-speed system with motor A in the same arbitrary maneuver *j*, and $V_{x\%,single\text{-}speed,mot.A}$ and $V_{x\%,2\text{-}speed}$ being the maximum speeds achievable for the single (with motor A) and 2-speed systems, respectively. In particular, within the analysis of the case study vehicle, $j=1$ relates to a 0-50 kph acceleration test, $j=2$ relates to a 0-70 kph test, $j=3$ relates to a 0-100 kph test, $j=4$ relates to a 10-50 kph test, $j=5$ relates to a 20-70 kph test, $j=6$ relates to a 20-100 kph test and $j=7$ relates to a 40-100 kph test. All the weighting factors adopted in this paragraph and the next have to be discussed and selected together with the vehicle manufacturer as functions of the specific typical life cycle.

The cost function $CF_{RG,i}$ due to the maximum road grade achievable at a velocity of i kph is calculated by:

$$CF_{RG,i} = \frac{\alpha_{i,single\text{-}speed,mot.A}}{\alpha_{i,2\text{-}speed}} \tag{2}$$

where $\alpha_{i,single\text{-}speed,mot.A}$ and $\alpha_{i,2\text{-}speed}$ are the maximum road grades the single-speed (with motor A) and the 2-speed vehicles can ascend, respectively. Hence, the cost function for either the unladen or the fully laden vehicle is given by:

$$\begin{aligned} DPCF &= w_{0\%} \cdot CF_{0\%,perf.} + w_{10\%} \cdot CF_{10\%,perf.} + \\ &+ w_{RG,0} \cdot CF_{RG,0} + w_{RG,20} \cdot CF_{RG,20} + \\ &+ w_{RG,40} \cdot CF_{RG,40} \end{aligned} \tag{3}$$

where $w_{x\%}$ and $w_{RG,i}$ are the relative weighting factors of the cost function. Finally, the overall cost function related to the dynamic performance of the vehicle is the arithmetic average of the cost functions for both the unladen and fully laden conditions.

B. Energy Consumption Cost Function (ECCF)

A similar procedure has been implemented for the computation of the cost function for the evaluation of the energy consumption of the system. The NEDC, FTP, SC03 and UDDS driving schedules have been adopted for the analysis of the system performance in terms of energy efficiency. The overall cost function for each cycle is:

$$CF_{cycle} = \frac{Cons_{cycle,2\text{-}speed} + \Delta Cons_{cycle,2\text{-}speed}}{Cons_{cycle,single\text{-}speed,mot.A} + \Delta Cons_{cycle,single\text{-}speed,mot.A}} \tag{4}$$

$Cons_{cycle,single\text{-}speed,mot.A/2\text{-}speed}$ being the predicted battery energy consumed during the driving schedule for either the single (with motor A) or 2-speed drivetrains. When the vehicle is incapable of following the velocity profile specified by the driving schedule due to, for example, the mass of the payload, the penalty terms $\Delta Cons_{cycle,single\text{-}speed,mot.A/2\text{-}speed}$, which represent an estimation of the additional energy that would be required in order to follow the velocity profile of the driving schedule, are added to $Cons_{cycle,single\text{-}speed,mot.A/2\text{-}speed}$. These are calculated using:

$$\begin{aligned} \Delta Cons_{cycle,single-speed,mot.A/2-speed} &= \int \Big[\big(T_{theoretical,traction}(t) + \\ &- T_{actual,traction}(t) \big) \cdot \omega_{motor}(t) \cdot \frac{1}{\eta_{battery}(t) \cdot \eta_{motor}(t)} \Big] dt \end{aligned} \tag{5}$$

where $T_{theoretical,traction}(t)$ is the electric motor torque required in order to follow the reference velocity profile, $T_{actual,traction}(t)$ is the actual maximum electric motor torque that is available, $\eta_{battery}(t)$ and $\eta_{motor}(t)$ are the estimated efficiencies of the battery and the electric motor/inverter (as the system is working beyond its peak torque), and $\omega_{motor}(t)$ is the electric motor angular velocity.

The overall cost function $ECCF$ for the fully laden or unladen vehicle is given by:

$$\begin{aligned} ECCF &= w_{FTP} \cdot CF_{FTP} + w_{UDDS} \cdot CF_{UDDS} + \\ &+ w_{SC03} \cdot CF_{SC03} + w_{NEDC} \cdot CF_{NEDC} \end{aligned} \tag{6}$$

where w_{FTP}, w_{UDDS}, w_{SC03} and w_{NEDC} are the weighting factors for the selected driving cycles (their sum being equal to 1). Similarly to what was explained for the $DPCF$, the overall cost function $ECCF$ for the vehicle is the arithmetic average of the cost functions for the unladen and the fully laden configurations.

C. Basic Criteria for the Selection of the Gear Ratios and the Gearshift Map

The optimization procedure computes the optimal gear ratios and gearshift map; however, the understanding of the sensitivity of the system to the variation of the main parameters is necessary for interpreting the results and reducing the optimization time by constraining the range of the parameters.

The generation of the cost function values is based on a brute force algorithm, according to the constraints and the discretization steps specified by the user. The resultant cost function values for various combinations of gear ratios are represented in the form of 3-D surfaces, each one for an assigned gearshift map. The optimization problem consists of finding the absolute minima for the cost functions.

The gearshift map defines the values of vehicle velocity at which the upshifts and the downshifts are carried out, as functions of the driver torque demand. The downshift is carried out at a lower vehicle velocity than the upshift for an assigned level of driver demand. Fig. 1 shows two examples of gearshift maps ('Map STD' and 'Map 1') adopted during the optimization procedure.

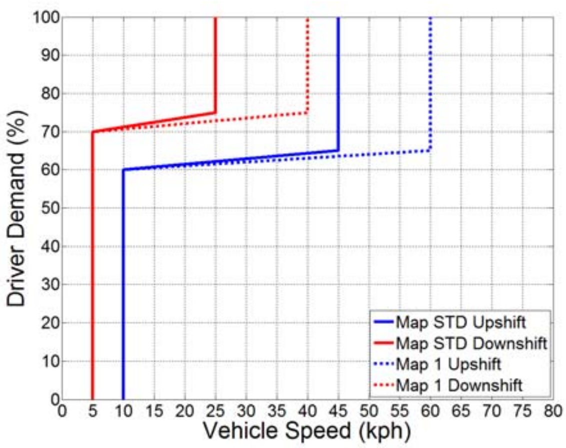

Fig. 1. Overlap of two gearshift maps adopted during the optimization procedure.

Generally speaking, a high value of τ_1 (first gear ratio) increases the value of the achievable road gradient, unless this is not limited by the available tire traction. A low value of τ_2 (second gear ratio) reduces the energy consumption, without influencing straightline performance. As a consequence, and as a general rule, a high step ratio τ_1/τ_2 represents the best solution for achieving high performance and low energy consumption; however the maximum value of the step ratio is often constrained by the mechanical layout of the gearbox.

Within electric vehicles, the selection of the velocity (Fig. 1) for the upshift in conditions of high torque demand is not as relevant as for internal combustion engine driven vehicles due to the significant extension of the constant power region. When considering the case of an electric vehicle during an acceleration test, as a first approximation rule it is better to perform the upshift at an electric motor speed higher than $n_{base}\frac{\tau_1}{\tau_2}$, so that after the upshift the motor is still working in the constant power region (above the base speed n_{base}). Apart from this, the effect the other parameters have on the optimal upshift point at high torque demands is marginal, but not negligible. In particular, within a 2-speed gearbox, the efficiency for τ_2 is higher than for τ_1, due to the lower gear ratio. Secondly, the equivalent vehicle mass increase $\Delta m_{eq,motor}$ induced by the electric motor inertia is equal to $\frac{J_{motor}\tau_{1/2}^2\tau_{diff}^2}{R_W^2}$, where J_{motor} is the moment of inertia of the electric motor unit, τ_{diff} is the differential ratio and R_W is the wheel radius. For typical data of the specific vehicle application, $\Delta m_{eq,motor}$ can be over 200 kg in τ_1 and below 60 kg in τ_2, with a sensible effect on vehicle performance and inertial load.

In conclusion, if the gearshift for high values of driver demand is performed so that the electric motor unit works in the constant power region before and after the gearshift, it is convenient to anticipate it as much as possible, in order to achieve the maximum benefit due to the higher efficiency and the lower inertia associated with τ_2.

Some electric motors for automotive traction are characterized by a third area for high speed values, in addition to the constant torque and constant power areas. This area is characterized by decreasing values of electric motor power, starting from the speed value $n_{decreasing\_power}$. For these units, it is possible to work in the constant power region before and after the gearshift only if $n_{base}\frac{\tau_1}{\tau_2} \le n_{decreasing\_power}$.

IV. RESULTS

The optimization procedure has been run for the case study of a light commercial vehicle, characterized by a mass of about 2.1 tons and a maximum payload of 850 kg. The 2-speed gearbox has been optimized for two different electric motor/inverter units for automotive traction (developed by different manufacturers), called A and B in this paper, having respectively 70 kW and 75 kW of peak power.

Fig. 2 and Fig. 3 contain the efficiency maps and the operating points along an NEDC driving cycle (laden conditions). The points indicated as 'UPSHIFT' in both figures are positioned at the electric motor speed where the upshift is performed in case of 100% torque demand ('map STD' of Fig. 1).

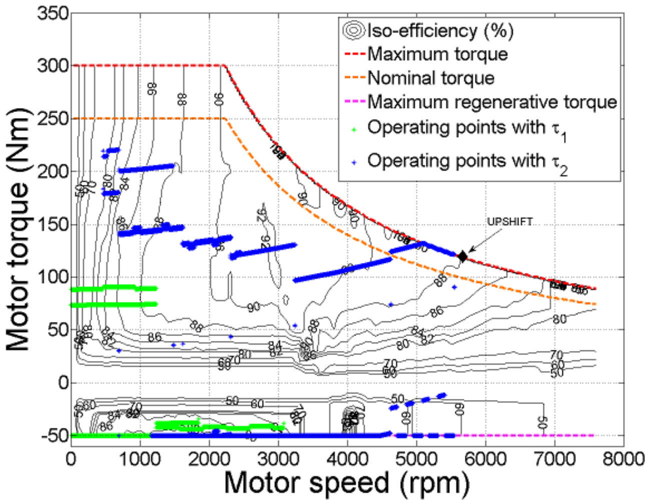

Fig. 2. Motor A (70 kW) efficiency map, with maximum, nominal and regenerative torques. Operating points for an NEDC driving cycle in conditions of fully laden vehicle ('map STD').

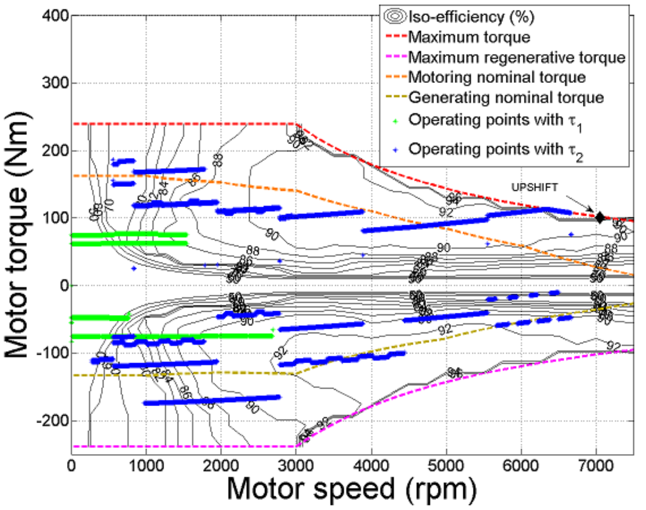

Fig. 3. Motor B (75 kW) efficiency map, with maximum, nominal and regenerative torques. Operating points for an NEDC driving cycle in conditions of fully laden vehicle ('map STD').

Motor B, despite the fact it generates a higher peak power, is characterized by a lower peak torque (240 Nm against 300 Nm) in traction, which results in a much higher base speed (approximately 3000 rpm against 2200 rpm). Due to its low torque and similar maximum speed (in comparison with motor A), motor B can be adopted for this case study vehicle, only if coupled with a 2-speed gearbox, otherwise vehicle performance would be affected either for the road grade or the maximum achievable speed.

As motor inertia is related to the peak torque rather than the peak power, motor B has lower inertia than motor A.

In addition, the iso-efficiency plots of the 2 units are different, as the maximum efficiency level is reached by motor B along the entire constant power region, whereas for motor A the efficiency at high torque demand tends to drop above 4300 rpm. Motor A can achieve a maximum level of regenerative braking torque of -50 Nm, against the symmetrical peak torque characteristic of motor B. For both units, under n_{base} the efficiency is mainly dependent on electric motor speed. In fact, the iso-efficiency curves are quite vertical, especially for motor A, in particular, η_{motor} is an increasing function of motor speed.

Above n_{base} the efficiency is mainly an increasing function of the torque level (the iso-efficiency curves are quite horizontal). During a standard driving cycles with an unladen vehicle, the electric motor units work at so low torques during traction, where the iso-efficiency curves are so close to each other, that it is quite difficult to predict the best gearshift option without the adoption of the simulator.

Motor A is characterized by a constant ratio (equal to 1.2) between the peak torque and the nominal torque, whereas motor B is characterized by a variable ratio (close to 1.5 for low electric motor speeds), decreasing as a function of electric motor speed. The model automatically checks that the unit does not work at a higher torque level than the nominal one for a continuous amount of time exceeding the manufacturer specification. However, experimental tests on actual motor units have shown that the concept of nominal torque can be quite relaxed for automotive traction motors, due to the additional cooling induced by vehicle motion, so the peak torque is more relevant than the nominal torque for this application.

A couple of examples of optimization surfaces for *ECCF* (for an assigned gearshift map, 'map STD' in Fig. 1) are presented in Fig. 4 as functions of the gear ratios $\tau_{1/2}$; with $\tau_{1/2}$ being normalized to $\tau_{1STD/2STD}$, the values of the gear ratios designed for the 2-speed gearbox coupled with motor A, according to the standard procedure adopted by Oerlikon Graziano – Automotive. For $\tau_{1/2}$=$\tau_{1STD/2STD}$, *ECCF* for motor A is between 0.96 and 0.97.

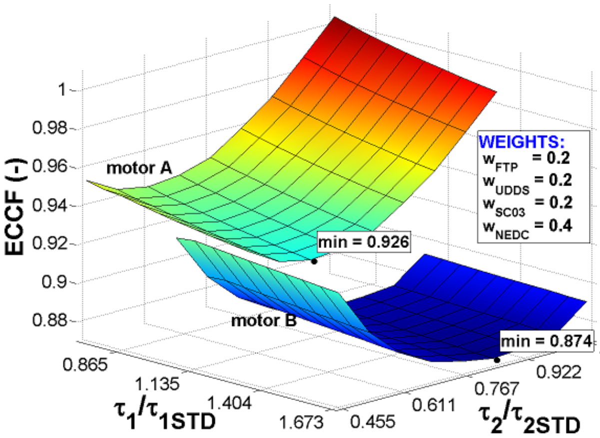

Fig. 4. Energy consumption cost function for motors A and B, for normalized values of the first (τ_1) and second (τ_2) gear ratio ('map STD').

The benefits due to the adoption of the optimized 2-speed transmission system instead of the conventional single-speed gearbox with motor A are about 10% with motor A (7.5% by limiting the analysis to only the 'map STD', Fig. 4) and 12.5% with motor B. This benefit represents the weighted average energy consumption reduction between the 4 driving cycles selected for the analysis. The advantage of motor B is due to its higher regenerative capability, and not due to the better efficiency for positive torque values.

Fig. 5 is an example of the distribution of the electric motor operating points during an NEDC schedule simulated using model *C.*, and can be used as a post-hoc validation of the optimization based on the model without driver dynamics. The simulation results of the two models (in II *A.* and II *C.*) are very similar, and the optimizations are independent of the parameterization of the feedback driver model, for the driving cycles where the velocity profile can be followed by the vehicle within the specified tolerances. The models including driver dynamics usually give origin to lower energy consumption values, due to the smoothening of the driver input in terms of the torque demand profile (as is depicted in Fig. 2 and Fig. 5).

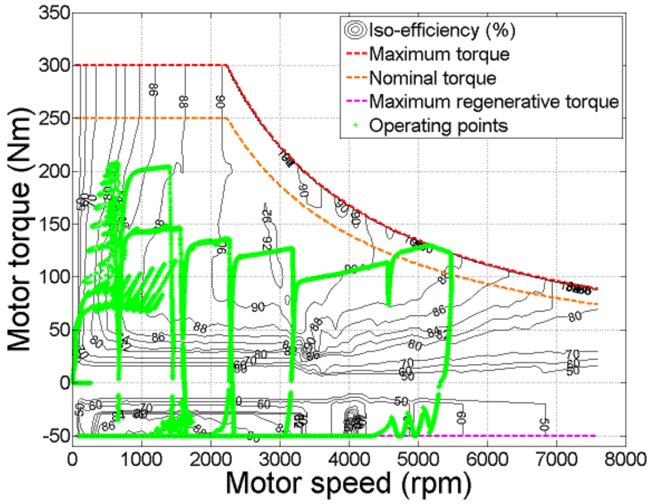

Fig. 5. Operating points within the electric motor efficiency map for an NEDC schedule simulated by the model including driver feedback dynamics (motor A, fully laden vehicle, 'map STD').

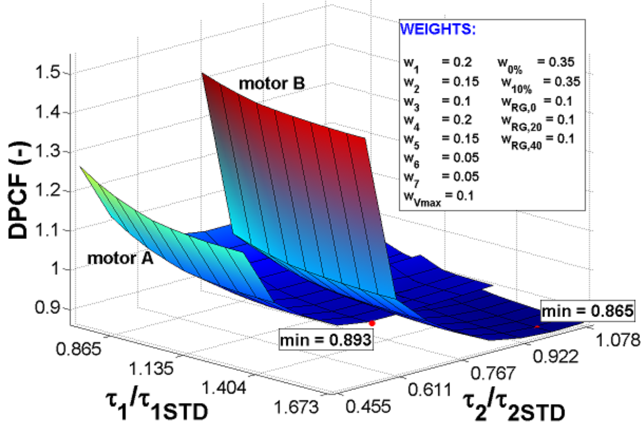

Fig. 6. Dynamic performance cost function for motors A and B, for normalized values of the first (τ_1) and second (τ_2) gear ratio ('map STD').

Fig. 6 is an example of an optimization surface for the performance of both motoring units ('map STD'), showing an overall improvement in vehicle performance of greater than 10% for both the motors, in comparison with the single-speed vehicle with motor A. In particular, the minimum of the cost function for motor B is lower than for motor A. This is consistent with the peak power values of the two units.

However, motor B, because of its lower peak torque, is more affected by the variation of the gear ratios, which results in a steeper gradient of *DPCF*. From Fig. 4 and Fig. 6 (similar trends can be derived for any gearshift map), the optimal gear ratios as functions of performance and energy consumption are not at variance, despite the fact that *DPCF* specifies higher $\tau_{1/2}$ than *ECCF*. The final gear ratio for the vehicle and the weighting between the two cost functions is chosen in collaboration with the industrial partners and the automotive manufacturer.

As an example, Table I summarizes the optimal ratios according to the consumption and performance procedures, and the final selected values, for an assigned gearshift map ('map STD'). In the case of motor A with the 2-speed transmission, the maximum speed for the laden vehicle is limited by the available motor torque, whereas for the other two cases it is limited by the maximum motor speed. As expected, the ratio between the optimal ratios of motor A and motor B is close to the ratio between the peak torques of the two units. The optimal step ratios τ_1/τ_2 are higher than those suggested by the standard design procedures of Oerlikon Graziano – Automotive.

Table II shows a sample of the energy consumption and dynamic performance results (for the laden vehicle), comparing the single-speed vehicle with the 2-speed vehicle (with both motor A and B), adopting the gear ratios indicated in Table I ('map STD').

TABLE I
OPTIMAL AND SELECTED GEAR RATIOS ('MAP STD')

		τ_1/τ_{1STD}	τ_2/τ_{2STD}
Motor A	Consumption	1.337	0.611
	Performance	1.337	0.767
	Selected layout	1.337	0.767
Motor B	Consumption	1.673	0.844
	Performance	1.673	1.000
	Selected layout	1.673	0.922

TABLE II
RESULTS ('MAP STD', LADEN VEHICLE)

	Motor A single-speed	Motor A 2-speed	Motor B 2-speed
NEDC consumption (kWh)	3.710	3.638	3.566
FTP consumption (kWh)	6.258	5.977	5.319
Road angle 0kph (°)	13.7	25.6	25.5
Road angle 20kph (°)	13.5	22.9	24.8
0-50 kph (s)	7.11	6.20	6.27
0-100 kph (s)	27.53	26.42	24.73
Vmax (kph)	117.4	135.3	133.6

Fig. 7 plots the energy consumption reduction (in comparison with the 2-speed vehicle equipped with electric motor B, and without any regenerative capability) vs. the maximum regenerative torque of the motoring unit. This trend, common to all the main driving schedules, shows that beyond 100-150 Nm (for the vehicle with motor B) of maximum regenerative torque, the benefits of regeneration are not so evident, given the speed profile of standard driving cycles.

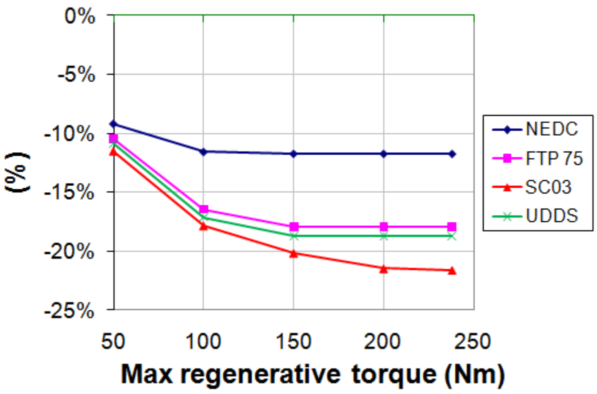

Fig. 7. Energy consumption reduction (in comparison with the 2-speed vehicle equipped with electric motor B, without any regenerative capability) during several driving cycles (NEDC, FTP, SC03 and UDDS), for different values of the maximum regenerative torque (motor B).

V. CONCLUSION

This activity has resulted in the following conclusions:

1. An optimization procedure has been implemented in order to automatically select the values of the gear ratios and the optimal gearshift map, for an assigned vehicle application;
2. For the case study vehicle equipped with motor A, the energy consumption is improved by 3% through the adoption of the 2-speed gearbox with standard gear ratios and further improved by 4-7% (depending on the gearshift map) by means of the optimization of the gear ratios;
3. The optimization procedure results in a significant improvement of the performance, in particular vehicle top speed and maximum ascendable road gradient;
4. The 2-speed gearbox permits the adoption of a low torque, low inertia, high regeneration electric motor unit having a comparable power level with the unit initially considered, with a further energy saving;
5. During standard driving cycles, the energy saving achievable through regenerative braking is effective only up to a threshold which is significantly lower than the maximum electric motor torque in traction.

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