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Torque Vectoring for Electric Vehicles with Individually Controlled Motors: State-of-the-Art and Future Developments

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Abstract

This paper deals with the description of current and future vehicle technology related to yaw moment control, anti-lock braking and traction control through the employment of effective torque vectoring strategies for electric vehicles. In particular, the adoption of individually controlled electric powertrains with the aim of tuning the vehicle dynamic characteristics in steady-state and transient conditions is discussed. This subject is currently investigated within the European Union (EU) funded Seventh Framework Programme (FP7) consortium E-VECTOORC, focused on the development and experimental testing of novel control strategies. Through a comprehensive literature review, the article outlines the state-of-the-art of torque vectoring control for fully electric vehicles and presents the philosophy and the potential impact of the E-VECTOORC control structure from the viewpoint of torque vectoring for vehicle dynamics enhancement.

Keywords: Electric Vehicle, Vehicle Performance, Braking, Traction Control, European Union

1 Introduction

Over the last decades, the environmental problems related to greenhouse and polluting gases emissions have stimulated the research of alternative energy sources for automotive vehicle propulsion [1, 2]. In recent years, the focus of attention has moved into the development of fully electric vehicles (FEVs), which promise to provide a personal mobility solution with zero emissions. Moreover, owing to significant advancements in energy storage units and electric motors in terms of power density, this promise of modern FEVs may become a viable option for the mass market.

With these prospects, novel concepts of electric vehicle layouts are gaining more and more importance. The first generation of fully electric vehicles was based on the conversion of internal combustion engine driven vehicles into electric vehicles, by replacing the drivetrains, while keeping the same driveline structure; that is, one electric motor drive, which is located centrally between the driven wheels, and a single-speed mechanical transmission including a differential. Such a design solution is going to be gradually substituted by a novel vehicle architecture, based on the adoption of individually controlled electric powertrains, with the unique possibility to improve the vehicle dynamics control because of their intrinsic high and independent controllability. The active control of electric powertrains allows the regulation of the distribution of the driving torques in order to achieve desired steady-state and transient vehicle dynamics characteristics. At the same time, if implemented through in-wheel motors, these architectural solutions allow an improvement of the overall vehicle packaging as less space is required by the powertrain.

Current electric vehicle research is investigating different powertrain configurations, constituted by one, two, three or four electric motors with different performance in terms of vehicle dynamic behaviour and energy saving targets [3, 4].

This paper presents an extensive review of torque vectoring and torque modulation techniques for the improvement of the dynamic performance of fully electric vehicles. Also, these techniques are subject of the research work carried out within the European Union funded Seventh Framework Programme (FP7) E-VECTOORC (Electric-Vehicle Control of Individual Wheel Torque for On- and Off-Road Conditions) project.

2 The project E-VECTOORC

The potential advantage of individual motor control for vehicle propulsion to enhance safety, comfort and fun-to-drive in both on- and off-road driving conditions is investigated by the three-year long E-VECTOORC project that started on 1st September 2011. The E-VECTOORC project brings together eleven complementary partners from industrial and research backgrounds to address the following key objectives:

- Development and demonstration of yaw rate and sideslip angle control algorithms based on the combination of front-to-rear and left-to-right torque vectoring to improve overall vehicle dynamic performance.
- Development and demonstration of novel strategies for the modulation of the torque output of the individual electric motors to enhance brake energy recuperation, anti-lock brake (ABS) and traction control (TC) functions. The benefits of these strategies include reductions in: i) vehicle energy consumption, ii) stopping distance, and iii) acceleration times.

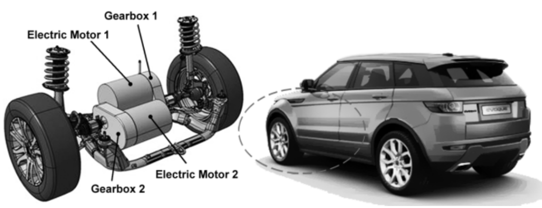

Figure 1: Front electric axle architecture of the Land Rover Evoque vehicle demonstrator

To achieve these targets, advanced torque vectoring control strategies for vehicle layouts characterised by one (in case of adoption of a torque vectoring differential) to four individually controlled electric motors are being developed for an optimal distribution (with respect to vehicle dynamics and energy efficiency targets) of the required driving torque between the two vehicle axles and within the individual axles.

The activity is carried out using vehicle dynamics simulations and Hardware-In-the-Loop (HIL) testing of vehicle components and subsystems. At full vehicle scale, experimental testing of the entire system will be performed using a highly versatile vehicle demonstrator (see Fig. 1) that can represent drivetrain architectures with one, two, three or four electric motors. The demonstrator vehicle will provide comprehensive information for quantifying the benefits of the proposed control system in both on-road and off-road driving conditions.

3 Torque vectoring control in steady-state conditions

3.1 The variation of the understeer characteristic

An extensive body of scientific literature presents and thoroughly discusses theoretical and experimental investigations on the cornering characteristics of automotive vehicles in steady-state conditions [5-10]. An overview of the important findings and insights is provided here.

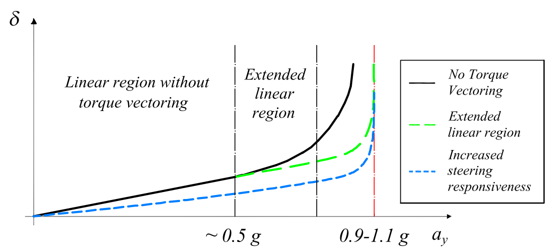

Figure 2: Potential modifications of the vehicle understeer characteristic achievable through torque vectoring with individually controlled powertrains

The evaluation of the vehicle cornering performance is carried out through the analysis of the trend of the steering-wheel angle, δ, as a function of the vehicle lateral acceleration, a_y (see Fig. 2). In particular, the vehicle response to a steering input is linear within a certain lateral acceleration threshold, which is usually about 0.5 g at constant vehicle velocity. Beyond this threshold value, the response becomes and remains non-linear until the maximum lateral acceleration of the vehicle, i.e. its steady-state cornering limit, is reached (see the black solid curve in Fig. 2). The two dashed curves in Fig. 2 represent possible targets that can be achieved through the implementation of individual electric motor control. For instance, the linear region can be extended above the lateral acceleration limit of 0.5 g (see the green dashed curve in Fig. 2). Also, the understeer gradient can be reduced in order to enhance vehicle responsiveness (see the blue dashed curve in Fig. 2). In addition, the maximum level of lateral acceleration can be increased as is shown for both the controlled vehicles of Fig. 2.

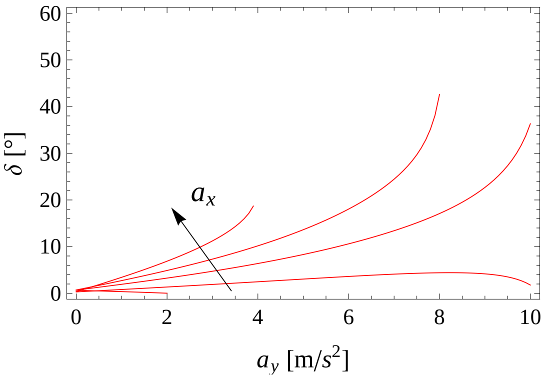

Figure 3: The steering-wheel angle δ[°] as a function of the lateral acceleration a_y [m/s^2], considering a constant torque distribution for different values of the longitudinal acceleration a_x [m/s^2], from a_x = -5 m/s^2 to a_x = 5 m/s^2 in steps of 2.5 m/s^2

A possible further implication of such individual motor control is that the variation of the cornering behaviour while accelerating or braking can be reduced. In doing so, robustness of vehicle response against vehicle longitudinal dynamics can be achieved.

The variation of the vehicle understeer characteristic as a function of longitudinal acceleration is highlighted in Fig. 3 by showing the understeer characteristics for a four-wheel-drive (4WD) vehicle with a constant traction force distribution (50% front/total, 50% left/front, 50% left/rear in traction, 75% front/total, 50% left/front, 50% left/rear in braking) at five different values of longitudinal acceleration. These simulations show that, for the specified vehicle parameters, positive longitudinal acceleration reduces the linear vehicle response region, and increases vehicle understeer. During braking, the linear response region is reduced as well, but the vehicle behaviour changes to oversteer in the non-linear region.

3.2 The E-VECTOORC approach

The authors of this paper have developed an *ad hoc* 4WD vehicle model simulator employing a quasi-static approach [5] and non-linear tyre characteristics. Three different torque vectoring strategies which summarise the strategies explained in [5] and [11] have been implemented: i) constant torque distribution (referred to as baseline vehicle); ii) torque proportional to the wheel vertical load; iii) torque distribution which allows achieving the same longitudinal slip ratio on each wheel.

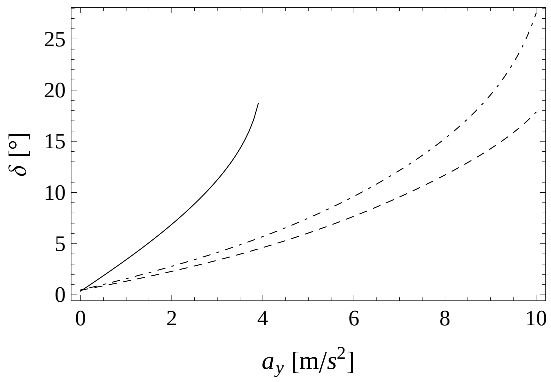

Figure 4: The steering-wheel angle δ[°] as a function of the lateral acceleration a_y [m/s^2] for the three torque distribution strategies, evaluated at a value of the longitudinal acceleration a_x = 5 m/s^2. The solid curve refers to strategy i), the dashed curve refers to strategy ii), and the dot-dashed curve refers to strategy iii).

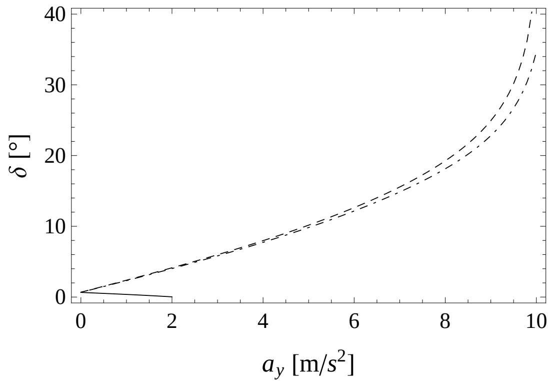

Figure 5: The steering-wheel angle δ[°] as a function of the lateral acceleration a_y [m/s^2] for the three torque distribution strategies, evaluated at a value of the longitudinal acceleration a_x = -5 m/s^2. The solid curve refers to strategy i), the dashed curve refers to strategy ii), and the dot-dashed curve refers to strategy iii).

The results show that strategies ii) and iii) effectively reduce the variation of the understeer gradient with the longitudinal acceleration and increase the linear region of the characteristics with respect to the baseline vehicle. However, vehicle understeer is increased in braking conditions and reduced in acceleration in comparison with strategy i) with the parameters of Fig. 3 (see Figs. 4 and 5). Therefore, in traction conditions, the vehicle dynamic behaviour achieved through strategies ii) and iii) could yield significant oscillations during transients, which are not acceptable for a normal driver. As a remedy for these oscillations, a feedforward controller in the frequency domain, together with feedback control, is necessary.

Recently, the authors of this article have developed a novel algorithm for the automated design of the torque vectoring strategy in steady-state conditions, which is based on an optimisation technique. This approach consists of the definition of a target understeer characteristic, which can be usually achieved with an infinite set of alternative wheel torque distributions in case of vehicle architectures with multiple electric motor drives. The selection of the most suitable wheel torque distribution for achieving the desired understeer characteristic can be carried out by solving an optimisation problem, by calculating the set of torque vectoring factors that minimises a defined objective function. In particular, the numerical procedure requires the following steps:

1 choice of the desired understeer characteristic parameters (e.g., understeer gradient in the linear region, extension of the linear region and maximum lateral acceleration);
2 definition of the objective function: for the purpose of energy efficiency, the authors have chosen to minimise the overall input motor power, which is computed by the simulation model considering the efficiency and inertial characteristics of the drivetrain components. Tyre slip losses are included in the calculation;
3 start of the optimisation routine by means of an algorithm based on the interior-reflective Newton method [12];
4 the outputs of the numerical procedure are the torque distribution factors which satisfy the assigned constraints and minimise the objective function.

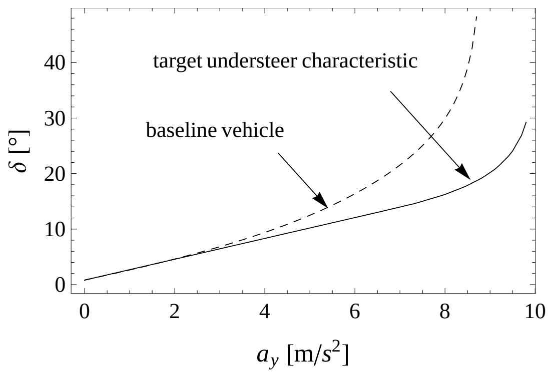

Figure 6: The understeer characteristic of the baseline vehicle (dashed line) and the desired understeer characteristic (solid line) evaluated at V=90 km/h and a_x=2 m/s^2

As an example of the developed optimisation methodology, we have considered a case study 4WD vehicle, equipped with four individually controlled electric motors, which travels at a velocity $V = 90$ km/h, and accelerates in the longitudinal direction at a constant value of $a_x = 2$ m/s^2. The understeer characteristic of the baseline vehicle in these conditions is shown with dashed line in Fig. 6: the understeer gradient in the linear part K_g is equal to K_g = 18 deg/g and the linear part of the characteristic ends at a value of lateral acceleration of about a^*_y = 0.2 g. Then the trend of the characteristic deviates from the linear behaviour up to the asymptotic maximum lateral acceleration achievable, which is a_{ymax} = 0.87 g. Thus the authors have defined a target understeer characteristic (solid line in Fig. 6) with the same value of the understeer gradient as the baseline vehicle, but with an increased linear part (a^*_y = 0.6 g) and a higher maximum lateral acceleration (a_{ymax} = 1 g).

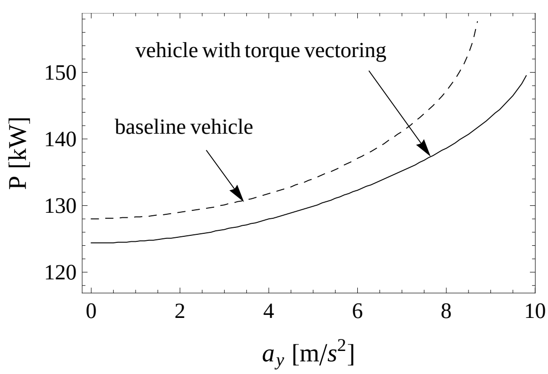

Figure 7: The overall motor input power P [kW] as a function of lateral acceleration a_y [m/s^2], evaluated for the baseline vehicle (dashed line) and for the vehicle with torque vectoring (solid line) at V=90 km/h and a_x=2 m/s^2

The results of the numerical iterations are shown in Fig. 7: the dashed line represents the overall input motor power evaluated for the baseline vehicle, whereas the solid line represents the overall motor input power of the vehicle provided with the torque vectoring distribution that allows achieving the desired cornering behaviour. The vehicle with torque vectoring requires less power than the baseline vehicle. This result is remarkable as the outlined torque vectoring strategy not only allows achieving the desired vehicle dynamic behaviour, but also allows optimal use of the battery energy for vehicle propulsion.

4 Torque vectoring control in transient conditions

4.1 Torque vectoring principles

The fundamental physical principles of effective torque vectoring systems are outlined in [5, 6], where the so-called β-method is explained in detail. This method is based on the analysis of the variation of the available vehicle yaw moment as a function of vehicle sideslip angle β. The authors of [5, 6] have focused their analysis on the compensation of vehicle dynamic response variation induced by longitudinal acceleration and braking. For the condition of zero yaw moment (i.e., $M_z = 0$), the gradient $dM_z/d\beta$ represents the static margin of the vehicle. It follows that the vehicle tends to understeer if $dM_z/d\beta > 0$, and tends to oversteer if $dM_z/d\beta < 0$.

Fig. 8 shows the trend of the stabilizing yaw moment M_z as a function of the sideslip angle β at zero steering-wheel angle and with constant vehicle velocity (green dashed line), for the conditions of longitudinal acceleration (black dashed line) and deceleration (red solid line) for a baseline vehicle. The controllability limits in the direction of understeer increase are represented by the red dot-dashed line and the blue dot-dashed line in case of acceleration and deceleration respectively. The target of the torque vectoring control is to reduce the offset between the curves of the yaw moment at different longitudinal acceleration values (taking into account the controllability limits), in order to reduce the variation of the vehicle dynamic response induced by the longitudinal dynamics. In deceleration conditions, the effect of the yaw moment variation cannot be fully compensated because the steady-state curve intersects the controllability limit during braking (see the blue dot-dashed line in Fig. 8).

According to [5], such a compensation can be achieved by means of three different actuations: i) a differentiation of the wheel torques within the rear axle (left-to-right torque vectoring technique); ii) an active roll control system capable of varying the lateral load transfer distribution between the two axles; and iii) a four-wheel-steering (4WS) system. The conclusions of the analysis are that the in-axle torque vectoring methodology (for the specific case study vehicle) is able to fully compensate the load transfer and the tyre longitudinal/lateral interaction effects due to vehicle acceleration/deceleration (a range of +/-2 m/s^2 is considered in the reference). Also, this method proves to be much more effective in the compensation than the Active Roll Control system and the 4WS system described in [5]. In particular, for the case study presented in [5], Active Roll Control is effective only for sideslip angle values of more than 5° in deceleration and 3° in acceleration. Below this threshold, the system is unable to compensate the effect of vehicle acceleration/deceleration. In contrast, the 4WS system is capable of generating the required compensation effect only for low values of β.

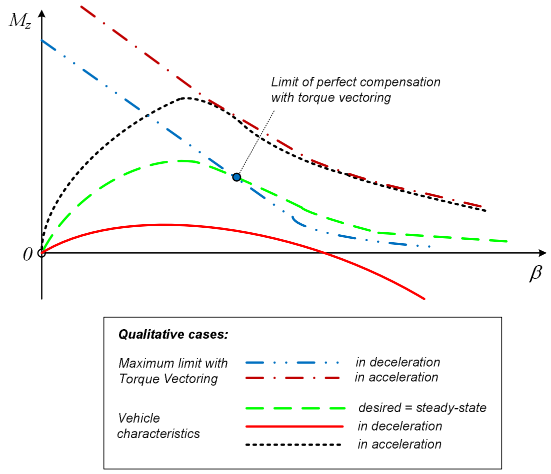

Figure 8: Stabilizing yaw moment as a function of vehicle sideslip angle in conditions of constant velocity and vehicle acceleration / deceleration (torque vectoring on the rear axle)

In [11] the authors describe the principles of the Mitsubishi Super-All-Wheel-Control, which is a direct yaw moment control (DYC) strategy obtained through the distribution of longitudinal forces and lateral forces among the four tyres. This torque-vectoring strategy is implemented through the employment of torque-vectoring differentials, comprising planetary gears and two clutches or brakes, in order to transfer torque from the left wheel to the right wheel and vice-versa, independently from the location of the faster wheel (within limits relating to the differential layout). According to the Mitsubishi algorithm, depending on the variation of the traction coefficient, a more balanced distribution of longitudinal and lateral forces between the left and right wheels can be achieved during cornering.

4.2 Torque vectoring control during emergency manoeuvres

According to the ISO and SAE regulations, vehicle dynamic performance can be evaluated through dynamic tests such as step steer or double step steer manoeuvres.

Figs. 9 and 10 show simulation results obtained by the authors for the dynamic response (in terms of sideslip angle) of a 4WD vehicle to a step input of the steering-wheel angle (0°-100°). The 4WD is simulated with two different torque distribution strategies: a constant torque distribution, as explained in Section 3.1 (indicated by the dashed line in Figs. 9 and 10 and referred to as the baseline vehicle) and a torque vectoring strategy, where the wheel torque is proportional to the wheel vertical load (denoted by the solid line in Figs. 9 and 10 and referred to as the torque vectoring vehicle).

During acceleration (see Fig. 9), large sideslip angles are experienced by the torque vectoring vehicle. Clearly, the dynamic response of the torque vectoring vehicle is not acceptable for a passenger car.

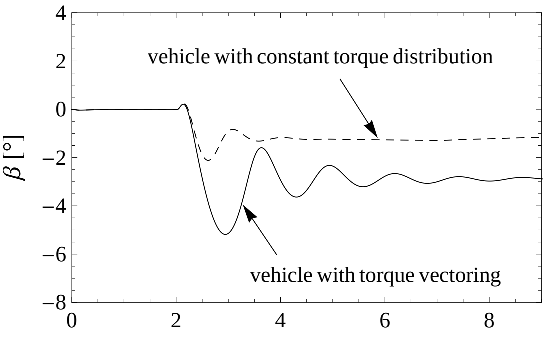

Figure 9: Step steer response (0°-100°) evaluated at a_x=3 m/s^2 and V=90 km/h. The dashed curve refers to a vehicle with a constant torque distribution whereas the black curve refers to a torque vectoring strategy that consists in the torque distribution proportional to the vertical load acting on the wheel

In braking conditions (see Fig. 10), the torque-vectoring strategy improves the dynamic behaviour of the vehicle, since the overshoot of the sideslip angle is strongly reduced with respect to the baseline vehicle. The important conclusion that can be drawn from our simulations is that the distribution of the wheel torques proportionally to the wheel vertical load is effective in braking conditions. However, in traction conditions, an advanced feedforward controller in the frequency domain is required (in addition to a commonly used feedback controller) in order to generate the desired yaw moment dynamics during the manoeuvre.

The subject of feedback yaw moment control has been addressed previously, e.g., in [13, 14]. In particular in [13], the authors have shown that, by employing a sliding-mode control approach to a single-track vehicle model and defining a sliding surface, direct yaw moment control combined with active wheel steering can maximise the stability limit for quick lane changes.

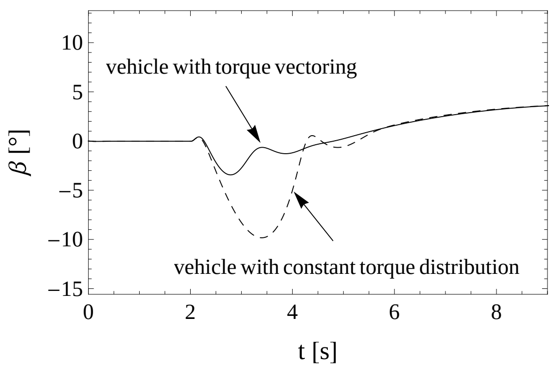

Figure 10: Step steer response (0°-100°) evaluated at a_x=-3 m/s^2 and V=90 km/h. The dashed curve refers to a vehicle with a constant torque distribution whereas the black curve refers to a torque vectoring strategy that consists in the torque distribution proportional to the vertical load acting on the wheel

Sliding-mode control is a useful control design technique to deal with non-linearities and uncertainties of the plant model. Therefore, it has been largely adopted in vehicle dynamics control. A good comparison of the performance obtained with the different types of sliding surfaces can be found in [15]. In [16] feedback yaw moment control is designed using a differential braking strategy for vehicle stability control. The controller has been developed using a three degree-of-freedom non-linear vehicle model. The performance of the sliding mode controller has been compared with that of a direct yaw moment controller (DYC), where vehicle motion is regulated by a yaw moment generated by the distribution of the tyre longitudinal forces [14]. Simulation results have shown that the proposed controller can provide a vehicle with superior performance with respect to brake actuation and system smoothness, and can minimise the acceleration and jerk without compromising stability at high speed and large steering angle inputs. Also, improved robustness to road conditions was reported [16].

4.3 Torque vectoring control in off-road conditions

The E-VECTOORC project addresses not only the on-road but also the off-road mobility of electric vehicles. In this context, an advanced torque vectoring strategy can significantly improve power efficiency and cross-country performance of FEVs while driving on various deformable surfaces, such as dry and wet ground or snow.

Off-road conditions impose several issues to be taken into account during the development of the torque vectoring controller. A first factor is the tyre rolling resistance coefficient C_r. For instance, average values of C_r are of about 0.01 on a conventional highway surface, 0.015 on snow, 0.02 on gravel road, 0.08 on wet earth road, and up to 0.3 on sand [17]. As a result, the rolling resistance losses can influence the power flows between the driveline and the wheels and must be taken into account for the development of the torque vectoring control strategy. A second source of tyre power loss is constituted by the slip ratio, which can reach particularly high values in off-road conditions, in comparison with on-road conditions. In off-road, values of slip ratio up to 0.4-0.5 are quite common [18]. Hence, the off-road torque vectoring control should achieve a trade-off between traction capability and the power losses due to wheel slip and rolling resistance.

Several preliminary investigations [19, 20] point out an essential positive effect of torque vectoring on the off-road performance of front-wheel-drive electric vehicles: the vehicle with torque vectoring control achieves a reduction of tyre friction power dissipation in conditions of rough terrain.

5 Torque modulation in ABS and TC

A further innovative feature of the E-VECTOORC project is the enhancement of the performance of ABS and TC systems. This enhancement is achieved through the adoption of wheel slip ratio control carried out by the electric motor drives and based on estimated friction conditions, rather than through the modulation of the hydraulic brake pressures and friction brake torques as implemented in conventional solutions currently found on FEVs.

The target for the development of these two systems is to increase the frequency range and precision of torque modulation, which is achievable through the use of electric drives. This high frequency would permit a reduction of the amplitude of slip ratio oscillations during ABS and TC interventions, which are detrimental to system performance (in terms of stopping distance and vehicle acceleration times).

The interaction between regenerative braking and friction brakes in ABS systems for fully electric vehicles is linked to the brake torque modulation rate, which, for typical commercial ABS systems, ranges between 3 and 7 Hz. Recently, an ABS modulation strategy based on the fluctuation of the electric motor torque generated by in-wheel motors has been presented [21]. This system achieves a frequency of ABS torque fluctuations of at least 10 Hz and a delay in the actuation of the desired torque of only a few milliseconds. The main benefit of the increase in torque actuation response is the reduction of the stopping distance of the vehicle of approximately 7%.

Common strategies for actuating ABS in electric vehicles include: i) a reduction of the regenerative braking torque as a function of the coefficient of friction of the surface on which the vehicle is travelling, or ii) regulation of regenerative braking in relation to the rate at which wheel slip is changing. Also, if it is sensed that the wheel is on a low-friction surface, regenerative braking is commonly removed as soon as ABS is activated [22].

Actuation of the ABS function through the electric motors requires development of the control algorithm, which is traditionally based on strict on/off rules. A significant body of literature deals with optimal ABS control algorithms based on the combination of wheel acceleration control and slip ratio control. For example, [23] describes a Linear Quadratic Regulator (LQR) applied to ABS control with vehicle velocity used as the look-up variable for gain scheduling within the controller. A strong set of experimental results for different friction conditions is presented based on an internal combustion engine driven test vehicle equipped with electro-mechanical brake callipers, which have an actuation bandwidth of 72 rad/s. The article does not provide details on two of the major issues in ABS control systems: i) the slip ratio setpoint, and ii) the estimation of vehicle velocity. These two characteristics can potentially be determined by keeping the on/off ABS cycling algorithm for the rear axle active, which, however, may have a small negative impact on the overall stopping distance.

The authors of [24] present a composite ABS control based on a simplified model, similar to the one in [23]. Here, the vehicle speed is assumed to be an input to the controller, which is supplied independently from the ABS control system. The composite controller consists of a robust controller that governs wheel dynamics when the slip ratio is between specified thresholds, and a rule-based controller (similar to the controllers implemented in the commercial systems) that is active when the values of the slip ratios are outside of the specified boundaries. In [25], the authors have addressed the problem of slip estimation and the setup of the optimum slip ratio in an integrated slip control structure based on the actuation of a conventional hydraulic ABS unit, without an a-priori knowledge of tyre characteristics. However, the paper does not provide detailed experimental results of the implementation of the designed control system.

Particularly interesting is [26], which presents a pragmatic approach that is being followed for the development of the electric motor based ABS control of E-VECTOORC. Indeed, this article describes the use of standard observers adopted within commercial ABS control units to subject the rear wheels to a sequence of pressure increase/decrease/maintenance phases (as in conventional ABS systems) in order to correctly estimate the vehicle velocity. Front wheel speeds and slip ratios are continuously monitored through a Proportional Derivative (PD) controller. This simple control structure has undergone extensive vehicle testing, and a comparison with conventional ABS algorithms is presented in terms of the average vehicle deceleration and the brake fluid flow rates through the ABS control unit (these two quantities are an objective index for measuring the quality of tyre slip control).

6 Actuation problems

In conventional road cars, vehicle dynamics control and ABS/TC are actuated through an electro-hydraulic brake unit, which generates friction brake torques independently from the driver input. This method introduces actuation delays, which can deteriorate the system performance. In particular, Figs. 11 and 12 demonstrate that for the case of extreme step steer manoeuvres the effect of the experimentally measured (for a vehicle with a conventional brake system with vacuum booster) actuation delays is very relevant to the overall vehicle dynamics [27, 28]. These delays are related to several factors, including the limited volume displacement of the piston pump, the pressure drops in the valves and the compliances in the hydraulic system. Better actuation performance should be provided by the modern electro-hydraulic brake system units (EHB), which substitute the brake booster with a pump and a high pressure accumulator [29].

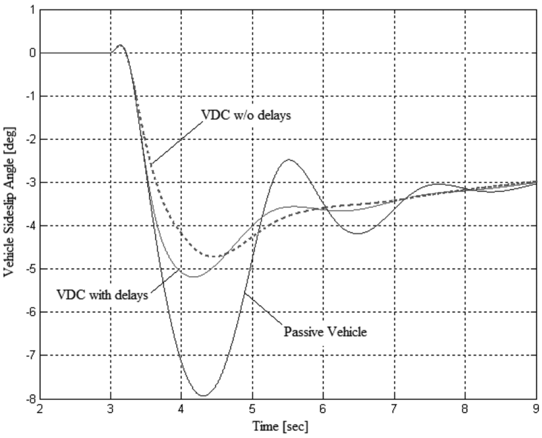

Figure 11: Comparison of vehicle response (vehicle sideslip angle) during step steer for a passive vehicle, a vehicle including an ideal VDC without delays in the actuation system, and a vehicle actuated by a commercial VDC unit (HIL test)

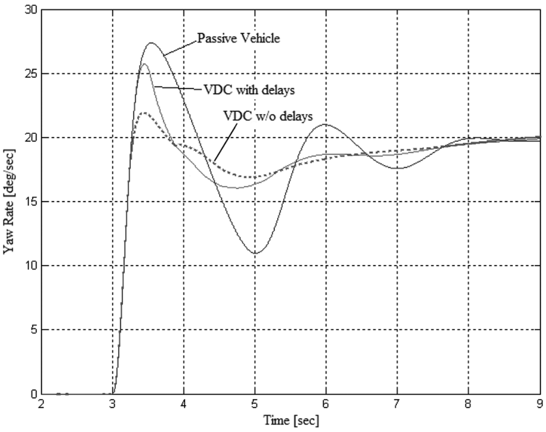

Figure 12: Comparison of vehicle response (yaw rate) during step steer for a passive vehicle, a vehicle including an ideal VDC without delays in the actuation system, and a vehicle actuated by a commercial VDC unit (HIL test)

The actuation solution that is being developed in the E-VECTOORC project relies on the individual control of the electric motor drive units to generate the target yaw moment, employing highly responsive (and characterised by low moment of inertia) switched reluctance electric motors.
The benefits of the high frequency range of torque modulation achievable with electric drive units may be compromised by the adopted drivetrain layout. In general, the design of FEVs is evolving towards the adoption of individually controlled electric motors, which can be configured as (i) in-wheel motors or (ii) in-board motors. In-wheel motors are particularly interesting for exploring new concepts for the layout of the passenger compartment [30]. However, they present immediate technical limitations because of problems related to packaging and increased unsprung mass, which restrict their potential vehicle dynamics capability, due to the increased vertical force oscillations that would occur on an uneven road profile, affecting the tyre-road contact. Moreover, current motor technology is limited in terms of power and torque density, which makes in-wheel powertrains, with their motor drive unit incorporated into the vehicle unsprung mass, a viable solution only for small and medium size cars, with relatively low performance [2, 30]. Therefore, a possible solution for the implementation of individual electric powertrains, without being subjected to the limitations of the in-wheel layout, is the adoption of in-board electric powertrains. Because of the physical offset of the motors from the wheels, half-shafts have to be employed to transmit torque to the wheels. The main disadvantage of this kind of choice arises from the torsional dynamics of the half-shaft and the subsequent first torsional mode of the drivetrain, together with the dynamics of the electric powertrain mounting system. Moreover, the torsional dynamics of the system is significantly affected by the slip ratio dynamics of the tyres, due to the combination of tyre steady-state non-linear characteristics and relaxation length. All these phenomena could interfere with vehicle drivability by affecting jerk dynamics (for internal combustion engines the first natural frequency of the powertrain is between 4 and 7 Hz) and could also reduce the effectiveness of ABS and TC actuation.
The E-VECTOORC consortium is well aware of this issue and a detailed analysis of the design specifications for the half-shafts, the powertrain mounting system and the drivetrain of FEVs with in-board motors (in order to achieve a dynamic response target compatible with effective ABS/TC actuation) is currently being carried out [31].
In [31] a very interesting dynamic analysis of in-board electric powertrains in both the time and frequency domains is presented. A feedback control system, incorporating state estimation through an extended Kalman filter has been implemented in order to compensate the effect of half-shaft dynamics and generate a smooth half-shaft torque profile. The effectiveness of the new controller is demonstrated through the analysis of the performance improvement of a traction control system based on direct slip control. The comparison of the performance of the passive vehicle, the vehicle equipped with the TC and the vehicle equipped with the TC and the half-shaft torque control system is shown in Fig. 13 for a tip-in test from an initial speed of 50 km/h. The wheel torque level during the first part of the tip-in manoeuvre is beyond the friction limit of the tyres, which implies significant wheel spinning for both the passive vehicle and the vehicle with TC only. The TC without half-shaft torque control achieves a more irregular slip control dynamics in comparison with the system including the half-shaft torque control loop. In contrast, the vehicle equipped with the novel controller and a basic proportional TC is characterised by the absence of any significant wheel spinning (maximum slip ratio of about 16%) and a higher velocity profile than the other three vehicles.

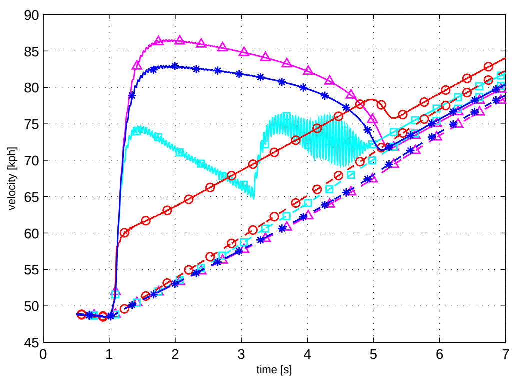

Figure13: Wheel (continuous line) and vehicle velocities (dashed line) during a tip-in manoeuvre. Vehicle without control system (△); vehicle with traction control based on a proportional gain (*); vehicle with traction control based on a PID (Proportional-Integral-Derivative) controller (□); vehicle with traction control based on a proportional gain and the novel half-shaft torque control (o)

Thanks to the enhancement of the performance of the traction control system, this methodology is being implemented into the E-VECTOORC vehicle demonstrator.

7 Conclusions

Fully electric vehicles allow an implementation of sophisticated torque vectoring strategies.

The subject of individual motor control is currently investigated by the EU-funded FP7 project E-VECTOORC. This project is aimed at the development and experimental testing of novel control algorithms. The first results of the E-VECTOORC research activity have shown that the steady-state and transient dynamic characteristics of a fully electric vehicle can be 'designed' through the active control of the electric powertrains, rather than indirectly tuned via the common chassis parameters such as mass distribution and suspension elasto-kinematics.

Furthermore, the low response time and high controllability of electric motor drives can bring significant benefits for the feedback control of vehicle yaw rate and sideslip angle in emergency conditions. However, the advantages of the high frequency range of torque modulation achievable with electric drive units may be compromised by the adoption of in-board motors, since the influence of the torsional dynamics of the powertrain and its mounting system should be taken into account for the implementation of TC/ABS systems.

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