Application of Regenerative Braking on Electric Vehicles

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***Abstract*—Regenerative braking influences several aspects of design and performances of electric vehicles. Improved performances of modern electric drives can be exploited to optimize vehicle efficiency, stability and environmental impact. Braking plant is devoted not only to stop the vehicle but also to the actuate many on board safety related systems. As a consequence of the application of regenerative braking, the system is redundant and over-actuated, so the application of electrical and mechanical efforts has to be carefully optimized. In this study a general approach is proposed and discussed, in terms of used engineering tools and obtained results.**

Keywords—Braking, Brake Blending, Torque Vectoring, Electric Vehicles EV, Energy Efficiency.

I. Introduction

The greatest opportunities for innovation in the automotive sector offered to designer concern the possibility of using electric traction system. Electric motors can be both speed or torque controlled quite precisely in wider operational ranges, respect to conventional Internal Combustion Engines (ICEs) [1]: modern electric drives allow a precise torque control even in near to standstill conditions [2] and enable, also, a simple four quadrant control, which offers the possibility of performing regenerative braking, allowing to recover a significant part of vehicle kinetic energy during the braking phase [3].

Instead, wear and heating of brake pads is due to dissipated energy [4].

Therefore, the application of regenerative braking is very important, since can reduce overall energy consumption (improving autonomy and efficiency), maintenance costs and environmental impact related to the worn brake debris, credited as first source of pollution not related to combustion [5], with not negligible consequences in terms of environmental impact, which is still difficult to be completely quantified. Also, the good bandwidth response of electric drives should be exploited by on board mechatronic systems, in order to maintain vehicle stability and safety, not only during braking and traction phase [6], [7], but also through cornering maneuvers [8]–[10].

High power density rate of electric units allows the usage of multiple traction motors to regulate torque efforts among wheels, in order to perform torque vectoring. On this last topic there is wide literature, which is mostly referred to vehicles with four in-wheel motors [11], [12].

For these reasons the system has to be designed in order to optimize the synergy between electric and friction braking [13], [14], not only in terms of vehicle longitudinal dynamics but also for lateral stability issues.

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Conventional friction brakes are constrained to work as passive components able only to dissipate vehicle kinetic energy [15]. On the other hand, the performances of the electric systems are constrained by thermal, current and power limitations of motors, drives and connected storage systems, as explained in the author's previously work [16].

The term *Brake Blending* (BB) is used to describe the way in which the action of both conventional and regenerative braking is applied. The action of the BB regulation has to be "*transparent*" for the user: the system has to compensate different performances and availability levels of electric and conventional actuators, maintaining a stable vehicle behavior.

Aim of this study is to optimize the blending strategy and torque allocation algorithm of electric and conventional friction brake efforts on vehicle's wheels, whit the objective to completely exploit the regenerative brake, while ensuring the provision of a minimum level of braking performance in every operational condition, according to specific driving safety specifications.

A general flexible modelling methodology is developed in order to be easily adapted to different vehicle powertrain and brake plant layouts. Proposed models are designed to be modular, in order to be reassembled and customized for different applications. Finally, implementation is optimized for fixed step integration and easier Real Time Implementation (RTI). These are fundamental features in order to perform Hardware/Software in the Loop (HIL/SIL) testing procedure.

The paper is organized as follow: section II relay on the adopted modelling procedure for the vehicle and related subsystem, section III focus on the vehicle benchmark description, section IV is dedicated to the simulation results and finally section V presents some final consideration regards conclusion and future work development.

II. Model Description

In Fig. 1 it's introduced a simplified scheme of the proposed approach: the brake plant is supposed to be controlled by a "*brake demand*", an abstraction of a signal, representing a braking torque reference desired by a human or an autonomous driver. This brake reference should be further modified by on-board subsystems, such as ABS or ESP, devoted to improving vehicle stability and safety.

As a consequence, brake demand is a vector whose scalar components are T_{ref_i}, each one representing the requested torque in Nm of the i-th vehicle wheel.

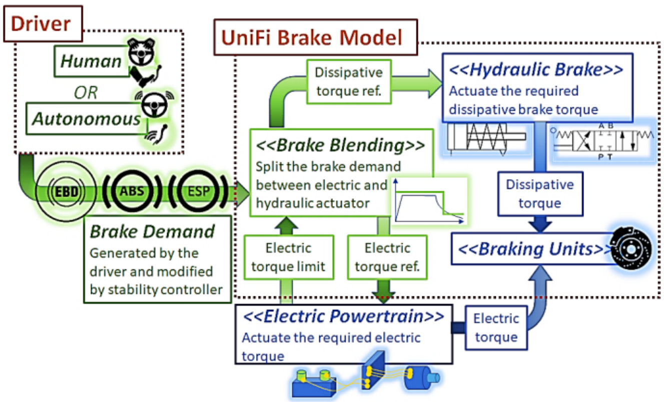

Fig. 1 Unifi Brake Model: layout and main subsystem.

According the scheme of Fig. 1, **UniFi Brake Model (UBM)** is composed by three sub-modules:

- *Brake Blending Controller (BBC)*: an algorithm devoted to executing the brake blending strategy.
- *Hydraulic Brake Plant*: it simulates the hydraulic actuation of the dissipative braking torque.
- *Braking Units*: which reproduce the application of braking torques on vehicle's wheels, including the wear and thermal calculations of disc and pad.

A. Brake Blending Controller

Brake Blending Controller (BBC) have to decide how to split the torque demand between the conventional brake ($T_{ref\_br\_i}$) and the regenerative one ($T_{ref\_reg\_i}$), according to (1).

$$T_{ref\_i} = T_{ref\_br\_i} + T_{ref\_reg\_i} \tag{1}$$

Adopted BB logic is well described by the Fig. 2. This strategy is also known in literatures as hybrid brake blending algorithm.

Main component of brake blending controller logic, for a single motorized wheel, is represented in Fig. 3, whose principal features is:

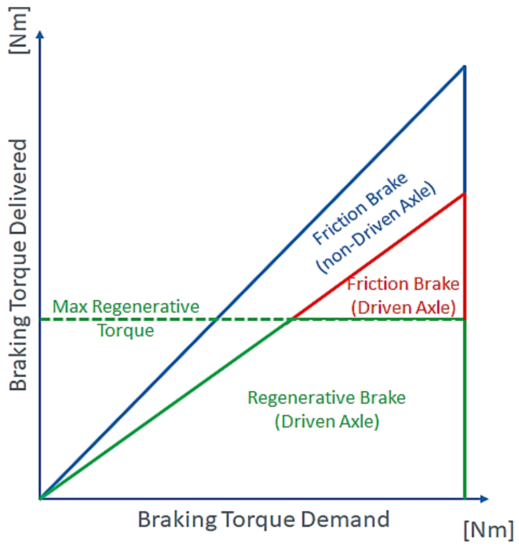

Fig. 2 Hybrid brake blending algorithm.

a) *Power and Current Limits*: according the state of traction and energy storage systems, the BBC evaluates the limitations in terms of maximum power and current, selecting the most cautious/restrictive condition.

b) *Torque Demand Creation*: a reference torque demand T_{ref_i} is evaluated according brake and traction commands of vehicle driver.

c) *Electrical Torque Saturation*: torque reference is supposed to be totally exerted by electric motors and saturated respect to power limitations previously calculated by a). As result, a regenerative torque reference $T_{ref\_reg\_i}$ is calculated. The aim is to maximize regenerative braking respect to the conventional one.

d) *Mechanical/Dissipative Braking Torque*: in order to satisfy the torque demand, the difference between desired braking torque T_{ref_i} and the one available for regenerative braking $T_{ref\_reg\_i}$ is used to calculate the desired torque $T_{ref\_br\_i}$ exerted by the conventional brake, as described in (1). In this way limited performances or availability of regenerative braking are compensated by the conventional plant.

e) Dynamic Compensation: electric and conventional brake plants should have a quite different dynamical behaviour. Electric braking torque reference signal is filtered in order to match the slower response of the conventional brake plant.

Brake Blending and Vehicle Stability Controllers

Stability controllers, like ESP™, perform vehicle torque vectoring by modulating braking torques applied on left and right wheels: in this way it's possible to apply an equivalent yaw torque M_{yaw} to the carbody, in order to correct its trajectory.

For this reason, brake blending has to be integrated in a more general optimal torque allocation policy of controllers devoted to keep the lateral/directional stability of the vehicle performing torque vectoring.

Assuming that applied longitudinal efforts can be modulated separately on each wheel, corresponding efforts should be calculated as the ratio between torques T_{ref_i} and wheel rolling radius r_w. Also, transversal distances y_i between tire contact patches and vehicle symmetry plane are supposed to be constant and known.

According over-cited simplifications, the correction torque M_{yaw} produced by the in-wheel motors is described by the following equation (2):

$$M_{yaw} = \sum \tag{2}$$

To find a near to optimal solution some constraints have to be respected.

First, total braking or traction demand should not be affected by the action of the stability controller: the exerted torques T_{ref_i}* has to be maintained as unaltered as possible respect to T_{ref_i}, the torques that should be applied to wheels without any intervention of the stability controller (3).

$$T_{ref\_i} \cong T_{ref\_i}{}^{*} \quad \forall i = 1,2,3,4 \tag{3}$$

Values of applied T_{ref_i} have to be constrained in order to respect known limitations of braking and traction units (4).

$$T_{\min\_i} \le T_{ref\_i} \le T_{\max\_i} \tag{4}$$

Finally "norm 2" of the applied correction has to be minimized (5) .

$$\left\| T_{ref\_i} - T_{ref\_i}{}^{*} \right\|_2 = \left(\sum_{i=1}^{n=4} \left(T_{ref\_i} - T_{ref\_i}{}^{*} \right)^2 \right)^{\frac{1}{2}} \tag{5}$$

Minimization of (5) contributes to find a solution that respect constraints and limitations described by (4). Also, a smoother dynamical behavior of applied actuations is expected.

Proposed implementation is described by following steps: first conditions, corresponding to relations (2) and (3), are implemented obtaining the linear system (6).

$$\overbrace{\begin{bmatrix} -\frac{y_1}{r_w} & \frac{y_2}{r_w} & -\frac{y_3}{r_w} & \frac{y_4}{r_w} \\ 1 & 1 & 1 & 1 \end{bmatrix}}\begin{bmatrix} T_{ref\_1} - T_{ref\_1*} \\ T_{ref\_2} - T_{ref\_2*} \\ T_{ref\_3} - T_{ref\_3*} \\ T_{ref\_4} - T_{ref\_3*} \end{bmatrix} = \begin{bmatrix} M_{yaw} \\ 0 \end{bmatrix} \tag{6}$$

By solving (6), using the Moore-Penrose Pseudo-Inverse matrix of A, it's possible to calculate the desired correction torque applied on every wheel. The use of Pseudo-Inverse assures the minimization of the norm 2 of the solution as stated by (5). Then it is possible to impose to each torque profile T_{ref_i} the saturation constrains (4), according (7):

$$\begin{cases} if\, T_{ref\_i} > T_{\max\_i} \Rightarrow T_{ref\_i} = T_{\max\_i} \\ if\, T_{ref\_i} < T_{\min\_i} \Rightarrow T_{ref\_i} = T_{\min\_i} \end{cases} \tag{7}$$

Calculation of (6) is repeated until a valid solution is found or alternatively when every torque profile T_{ref_i} is saturated. The resulting calculation is quite efficient, since in the worst case four iterations are needed (one per wheel).

B. Hydraulic Brake Plant

The Hydraulic Brake Plant model have to reproduce the behavior of the dissipative brake system, which converts the brake demand in real clamping forces of caliper pads to the discs, in order to produce the application of desired brake torques to wheels.

Brake unit adopted here is a hydraulic servo-amplification and actuation systems. The latter is also analyzed in terms of the functions that are performed by its different subsystems and then translated into an equivalent functional model, visible in Fig. 4. Adopted model maintains only some limited physical features of the simulated plant, i.e.:

- *Brake Demand Generation*: it's simulated as a converted and servo-amplified command signal which represents a clamping force reference and, consequently, a torque one.
- *Plant Configuration*: driver brake demand and system configuration are affected by mechatronics subsystems. According the current plant state, applied commands are modified to reproduce the response of corresponding fluid components.
- *Brake Modulation*: clamping pressure applied to brakes is typically regulated by electro-hydraulic valves that are able to connect the actuator with a pressure source or to discharge it, according to the adopted stability policies.
- *Brake Inexhaustibility*: safety of brake plant involves the availability of supply pressure in every working condition.

Sketches and Equations of Brake Plant

Forces applied to calipers are proportional to internal pressure of the actuator, whose dynamics is described by (8), where m_{act} is the mass of the piston, c_{act} the viscous coefficient, k_{act} the elastic coefficient, P the fluid pression, *preload* the initial spring force, S the cylinder frontal area and y the piston stroke:

(8)

Derivatives set to 0 when actuator hit endrun ($y \ge y_{\max}$ or $y \le y_{\min}$)

$F_{clamping}$ is calculated when interference with pad is verified ($y \ge y_{\max}$))

Equation (8) is used to model the motion of the piston caliper: once the pad reach the surface of the brake disc ($y \ge y_{max}$), derivatives of position y are set to zero and maximum run is saturated to y_{max}.

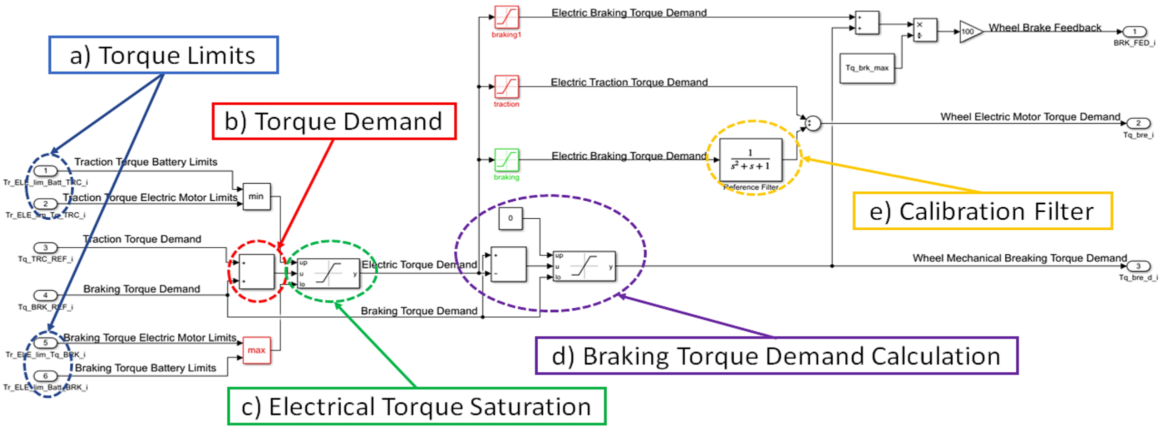

Fig. 3 Equivalent implementation in Matlab Simulink™ (release 2017a and 2018b) of a toy model of the BBC.

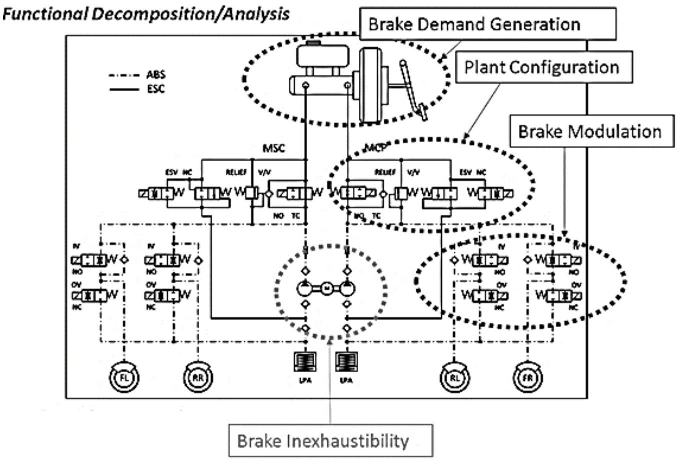

Fig. 4 Functional decomposition of the hydraulic braking plant.

Clamping force is then calculated solving a static problem: main advantage of this approach is to avoid the application of contact stiffness/damping terms, which often introduce high frequency poles that are difficult to be handled by fixed step solvers running at low frequencies (no more than 1 kHz) [17].

Neglecting thermal terms, pressure derivative is mostly a function of specific volume of the fluid. Chamber volume V and its derivatives (9) are known from calculated kinematics (8):

$$V = V_0 + Area\ y \Rightarrow \dot{V} \tag{9}$$

Fluid mass m inside the actuator volume V can be determinated by integrating the balance equation (10), where: mass flow sources due to valves (Q_{valves}) are calculated according (11), h_x is a corrective coefficient and x is the valve state (the dimensionless piston position between 0 and y_{max}).

$$\dot{m} \Rightarrow m(t) = \int_0^t Q_{valves}\, dt \tag{10}$$

$$- \leq \leq \quad \Delta = \quad \Delta \tag{11}$$

P_{max}, the maximum fluid pressure, is proportional to driver brake command: when the brake is activated by ESP or another safety related system during a traction/coasting maneuver, P_{max} is a fixed value decided by control logic state. P_{atm} represent the ambient pressure.

Real brake modulation valves have a finite response bandwidth, which is reproduced by inserting a second-order filter between input of the valve i_{valve} and the valve state x (12):

$$x / i_{valve} = \omega_n^2 / \left(s^2 + 2\xi\omega_n s + \omega_n^2\right) \tag{12}$$

C. Braking Unit

Braking Unit sub-model is able to calculate the power flows and corresponding energy integrals due to the application of braking efforts on wheels: knowing the amount of dissipated energy on each wheel, the model calculates corresponding thermal and wear behavior of brake friction components.

"Braking model" performs the following sub-functionalities that are described in the scheme of Fig. 5:

- Thermal behaviour of components: which calculate the temperature of the dissipative components.
- Wear of components: volume of pollutant debris produced in the braking phase is evaluated.
- Stability of friction/braking performances: torques applied to wheels are corrected taking count of the thermal behaviour of friction components.

Thermal Model

$T\_Br\_d\_i$ and w_w_i are respectively the dissipative torque applied on the i-th wheel and the relative rotational speed. Dissipated power on brake-components $W_{br\_d\_i}$ is calculated according (13).

$$W_{br\_d\_i} = T\_Br\_d\_i \cdot w\_w\_i \tag{13}$$

Energy is dissipated in the contact interface between pads and discs, so generated heat is transferred to both ones, being Q_{pad_i} and Q_{disc_i} respectively the heat flows transferred to pads and disc of the i-th wheel.

It's possible to define a heat flux distribution coefficient γ (14) in order to evaluate how transferred heat flow is divided between pads and discs. By adopting the coefficient γ, a decoupling of the thermal systems (pads and the discs) is introduced [18], [19].

$$\gamma = Q_{disc\_i} / Q_{pad\_i} \Rightarrow \begin{cases} Q_{disc\_i} = \gamma / (\gamma + 1) W_{br\_d\_i} \\ Q_{pad\_i} = 1 / (\gamma + 1) W_{br\_d\_i} \end{cases} \tag{14}$$

Once inlet heat flows for each brake component are calculated, it's possible to evaluate the mean temperatures T_{disc_i} (disc) and T_{pad_i} (pads), solving the lumped systems described by equations (15) and (16).

$$Q_{disc\_i} = C_{disc\_i} \dot{T}_{disc\_i} + \left(\overbrace{h_{conv\_d\_i}}^{convection} + \overbrace{h_{cond\_d\_i}}^{conduction} + \overbrace{h_{rad\_d\_i}}^{radiation} \right) \left(T_{disc\_i} - T_{amb} \right) \tag{15}$$

$$Q_{pad\_i} = C_{pad\_i} \dot{T}_{pad\_i} + \left(\overbrace{h_{conv\_p\_i}}^{convection} + \overbrace{h_{cond\_p\_i}}^{conduction} + \overbrace{h_{rad\_p\_i}}^{radiation} \right) \left(T_{pad\_i} - T_{amb} \right) \tag{16}$$

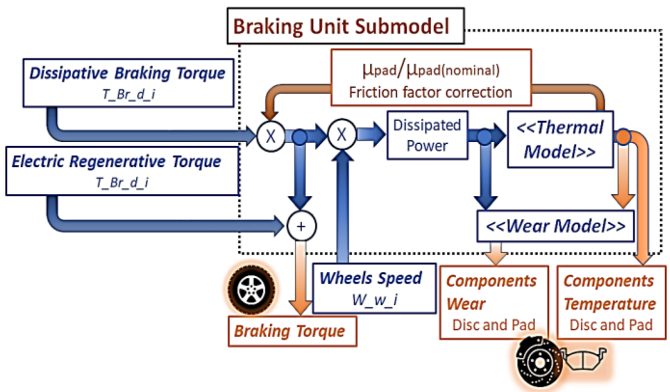

Fig. 5 Braking unit model and corresponding sub-models.

Wear Model

Wear of friction components it's calculated according an Archard approach corresponding to (17): it's supposed a proportionality of worn volumes of pads and discs (V_{pad}, V_{disc}) respect to dissipated energy (E_d).

$$\left.\begin{aligned} V_{pad} &= k_{pad} E_d \\ V_{disc} &= k_{disc} E_d \end{aligned}\right\} \Rightarrow V_{debris} = \left(k_{disc} + k_{pad}\right) E_d \tag{17}$$

Wear coefficients k_{pad} and k_{disc} are supposed to be known. However the main objective of the (17) is to evaluate the volume of pollutant debris (V_{debris}) produced by conventional braking. In this way it's possible to investigate how electric regeneration can produce a reduction of harmful micro-particles.

III. Proposed Benchmark Vehicle Model

Proposed approach was tested on a virtual model of a benchmark vehicle whose main parameters have been inspired by a known existing one. This data are freely available on line [20]. However, the latter were not exhaustive, so the model was completed with parameters derived from reasonable heuristic consideration.

For the prescribed benchmark authors considered two different powertrain configurations:

- *Single traction Motor*: a conventional powertrain layout in which single electrical motor is used to distribute power to frontal wheels through a differential distributor.
- *Four In-wheel Motors*: the same power of the previous case is divided between four identical motors, each one directly connected to a wheel. This second powertrain configuration is not related to any existing application and it's introduced only to comparatively evaluate possible advantages arising from different powertrain configurations.

Over described models have been developed in Matlab-Simulink™, where each subsystem is implemented as an independent model instance, allowing a separate execution of threads with different solvers and sampling frequencies. This feature allows an efficient execution and the investigation of computational issues due to the coupling of discrete and continuous subsystem, that also affect the real system.

For the vehicle body is adopted a planar 3 D.O.F. model (longitudinal and lateral motion with yaw rotation), also rotation of each wheel is considered. For modelling the tires a Pacejka [21] approach is adopted.

Also, high level control sub-systems are introduced, mutating existing simplified sub-models that are used by other simulation tools, such as Amesim™, and inspired by [22]:

- EBD: optimal distribution of braking forces between front and rear wheels, according to longitudinal load transfer.
- ABS/ASR: during the braking (ABS) and traction (ASR) phases, application of longitudinal forces respect to available tire-road adhesion is optimized.
- ESP: corrects longitudinal force applied to vehicle wheels in order to assure directional stability of the vehicle.
- Human Driver: simulates the behaviour of a pilot attempting to control the vehicle in order to perform a known mission profile.

The interchangeability of adopted Simulink™ sub-models respect to corresponding Siemens-Amesim™ ones was deliberately chosen to make easier integration and co-simulation between different simulation instruments. This seamless integration between different simulation instruments is a part of the objective of the OBELICS Project [23], which have financed this activity.

IV. Results

Some preliminary simulations have been performed. Aim of performed tests is not to produce validated results, but to demonstrate that the proposed approach is able to evaluate some fundamental features of electric vehicles.

In this sense proposed brake blending strategy proved to be flexible, since it was possible to use the same model for both the powertrain configurations of the benchmark vehicle model. In Fig. 6 a result of the applied BB strategies is shown: driver demands a constant braking torque of -700 Nm at the motorized wheels and the algorithm allocates the maximum available braking torque on the electric motor. Only the remaining one is applied through conventional brake system.

In Fig. 7 some results concerning control of brake caliper are exposed. The test consists in the application of a full braking demand followed by a modulation pattern, due to ABS intervention, with a duty cycle of 50%. Proposed model is clearly able to reproduce some typical features of the plant. It's interesting to notice how limited bandwidth of brake systems assures a relatively smooth behavior of applied clamping and braking forces.

Another interesting feature is the possibility of reproducing different mission profiles, in order to verify how proposed regeneration strategies, applied to different powertrains, should affect vehicle performances in terms of saved energy and wear of brake pads. Simulations have been repeated for different test cycles (NEDC, WLTP, FTP-75) in order to verify the robustness of obtained results.

Some results are visible in TABLE I: as expected, a four in-wheel powertrain is absolutely desirable in terms of regenerated energy. Another interesting result is represented by the evaluation of brake pad wear which is far lower for the four in-wheel powertrain.

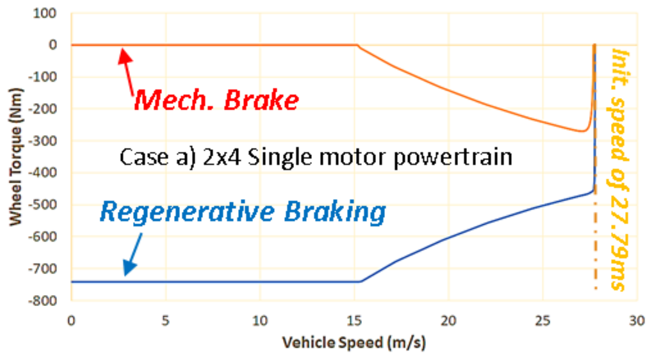

Fig. 6 Brake blending with a constant brake demand of 700 Nm.

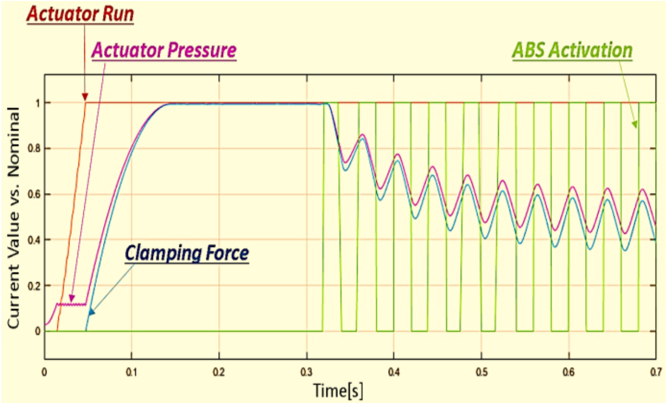

Fig. 7 Example of simulated hydraulic response of a brake calliper.

These results can be easily explained considering the limited decelerations associated to simulated test cycles: for the four in-wheel powertrain desired decelerations are almost completely assured by regenerative braking.

V. CONCLUSIONS AND FUTURE DEVELOPMENTS

In this work, some preliminary results concerning the development of modular brake models have been presented. Proposed models offer interesting features for preliminary sizing and optimization of brake blending policies for electric vehicles. The tool is also designed and optimized for real time implementation and hardware in the loop testing. Proposed models should be further calibrated and validated when some experimental data will be provided by others industrial OBELICS project's partners.

TABLE I. CONSUMED VS RECOVERED ENERGY AND PAD WEAR

Drive Cycle Simulation Test			
Drive Cycle	***Traction Layout***	***Recovered vs. Consumed Energy***	***Pad Wear Reduction***
NEDC	2x4 (A)	0,258	About 60%
	4x4 (B)	0,427	About 99%
WLTP	2x4 (A)	0,287	About 68%
	4x4 (B)	0,454	About 99%
FTP-75	2x4 (A)	0,393	About 66%
	4x4 (B)	0,601	About 99%

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