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Modelling of a Versatile Vehicle Braking System with a Fuzzy PID Torque Controller

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Abstract— The paper concentrates on intelligent control of electric vehicles operated in different braking regimes, from gradual downhill motion to intensive antilock braking. A model of a versatile fuzzy PID torque controller is designed and assessed in MATLAB/Simulink™. The system successfully follows the driver's demands in such changing environmental obstacles as tire-road friction, road inclination, wind velocity, etc. The best energy recovery is ensured in all braking modes. Simulation results, compared to experimental curves obtained from the hardware-in-the-loop test rig, demonstrate consistently high braking quality and potentiality of the proposed technique.

Keywords— electric vehicles, torque control, brakes, fuzzy logic, PID controller, energy recovery.

I. INTRODUCTION

A degree of the competence in the control of braking processes largely affects energy saving of road electric vehicles (EV) [1], [2]. Despite braking management has been intensively studied in the last decade and already includes several books and research papers abound [3], [4], little attention is paid to versatile systems that might unite both the intensive and the gradual (normal) braking in a common module. Since the gradual braking system and the intensive antilock braking system (ABS) have different targets, they are usually separated. The goal of the gradual braking is to follow the driver's pedal request as close as possible. Therefore, several studies were addressed to the kinetic energy recovery during smooth braking and downhill driving [5], [6]. On the contrary, the primary target of the ABS is to reduce the braking distance and time. There can be found rather few solutions, in particular, [7] and [8], that are so versatile that equally succeed in both braking scenarios.

The core of a versatile system is a controller capable of managing both the gradual and the urgent braking modes along with maximal energy recovery.

The main contender to the gradual single-input single-output (SISO) controller is a proportional-integral-differential (PID) device. The linear PID controllers are easy to implement, and sufficient tuning rules are available for taking a compromise in approaching rapid response and small overshoot [9] based on the famous Ziegler-Nichols and the Refined Ziegler-Nichols formulas.

Since the setting of the PID parameters must take into account as minimum three gains simultaneously along with their proper relationship, these controllers cannot yield a fine performance in highly order, nonlinear, multi-input single-output (MISO) or multi-input multi-output (MIMO) environment [10]. As well, they are not effective and often completely fail in tuning the processes with relatively long

dead time. Regrettably, the vehicle braking systems belong to this category. For instance, the dynamic model of EV created in [11] with the help of the PID controller, determines energy consumption and the effect of parameters on the car performance, but it cannot optimize energy recycling because of its orientation on the fixed driving environment. The same could be said about solutions offered in [12] – [15].

Other control instrument is a fuzzy logic controller (FLC). This intelligent tool governs successfully responses to actions taken in the environment and ensures improved management in comparison with conventional PID controllers by successful processing noisy data and uncertainties. Fuzzy control does not require a knowledge of the equations governing the system behaviour, but just needs a linguistic description of how the inputs and outputs depends on each other. Some antilock braking system (ABS) applications demonstrate enhanced performance of high order linear, non-linear, and dead timing systems upon the fuzzy control. Despite lacking a design theory, different offline techniques have been developed by now for deciding the nonlinear transfer elements of the urgently braking cars with the help of FLC [16], [17].

However, there are disadvantages in this technique when dealing with continuous processes, such as the ones considered in gradual braking where a set of variables have to be addressed in order to generate control actions. Usually, the FLCs are designed case-by-case, instead of systematic construction, using the trial-and-error method guided by designers, experiences on fuzzy logic. This makes many automation tools infeasible for implementation in FLCs. Due to the complexity of the nonlinear input-output surface of conventional FLCs, identifying a large number of tuning parameters, derivation of scaling factors and linguistic rules, and shaping the fuzzy sets remains a difficult challenge [18] – [20]. Since the tuning of FLC require high skill specialists, the fuzzy logic approach does not play the leading role in the management of industrial processes [21]. Moreover, the general FLC has a steady-state static error and fails in dynamics, while the PID controller can eliminate static errors along with its simplicity, easy to use, and nice dynamic robustness.

A natural step forward along the line is to consider a fuzzy-PID (FPID) control, where two groups of parameters have to be derived, namely, the scaling factors, and the fuzzy rule base [22], [23]. Several SISO, MISO, and MIMO configurations of FPID controllers, capable to improve the dynamic and static quality of control were manufactured including single-input, dual-input, and three-input ones with one, two, and three outputs, [19], [24].

In [25], a change-in-error is used as the only input of fuzzy inference, and three PID gains are the outputs. Consequently, three fuzzy rule tables are prepared and MATLAB/Simulink™ simulation is done. In [26], the FLC gains are obtained by solving a nonlinear optimization problem. Two architectures are developed in this research: one where the constrained optimization problem is solved offline and another devoted to updating the gains in real time. In [27], an error and its change come as the dual-input FLC and the three PID gains serve as outputs. In [28], the same inputs and outputs are used in FLC, but the only error as a PID input is applied for the motor speed adjustment. In [29], the EV was co-simulated in MATLAB/Simulink™ and AMESim™ using a dual-input, three-output FPID controller. The fuzzy variables were all defined in seven levels, from negative big to positive big. In [9], an FPID control scheme for ABS is developed, where the proportional, integral, and derivate gains can be self-tuned online under the EV speed control. Three control improvements were approached in this design, namely, the reduced stopping time, the limited slip ratio, and the enhanced vehicle performance. Considering error and error derivation as inputs, this system has 27 free parameters that are actually the coefficients of the inputs linear combination to compose the rules. As well, there are 12 parameters for each membership function input, where each one is used for three modules. Therefore, the total number of free parameters is 93.

At the same time, due to the analytic complexity of MIMO FPID, its construction makes the design rather complicated in terms of rule base structure and implementation of inferencing [30]. For this reason, the intelligent ABSs are mainly governed by the single-input and dual-input fuzzy-PI and fuzzy-PD type controllers proposed by Mamdani [31], where the P mode provides the required tracking accuracy, the PD mode is used to speed up the response, whereas the PI mode is applied to eliminate the steady state offset [32]. In [33], an error of the slip ratio and its change are the FLC inputs, and only the error enters the PID controller, which output represents an increment of the hydraulic pressure. When the slip error is bigger than 0.06, the fuzzy control is closed, the integral constant is zeroed, and the proportional gain obtains a biggest value to shorten the response time.

The most painful issue of the intelligent braking control concerns the methods targeted to identification of the road surface under the tire. In most of the above references, this problem is omitted, and irregularity of the road surface is ignored. For example, in [15] the detailed ABS description in the MATLAB/Simulink™ environment is presented. Nevertheless, this solution, built on the fixed theoretical tire-road model with the ideal friction factor ratio of 0.2 to 1, cannot maximise energy recovery. Since the controller does not consider changeable road conditions, it must decrease the actuating torque when driving off-road. The same constrain reduces the quality of the designs proposed in [34], [9], and [35].

On the plus side, a novel ABS control approach is developed in [36] that couples the vehicle dynamics, wheel dynamics, and suspension dynamics in a cascaded structure considering the road slope and the effective tire radius as an estimated state. However, even though the scheme demonstrates good performance for critical braking cases, experimental validation was not produced by the authors and

suspension dynamic influences were not analysed there. In [34], the tire-road friction is derived relying on the velocity sensor signals and vehicle geometry. In [37], a perturbed sliding mode observer is introduced. Certain models of the friction-slip relations may be found in literature, such as Pacejka's "Magic Formula" [38], Burckhardt model, Rill model, and others. A method designed in [39], presumes adding a small oscillation to the assigned torque with following measuring the gain and the phase shift of the wheel speed oscillation. These values correlate to the transfer function of the vehicle model around an operating point, thus obtaining the slip ratio and the road characteristics. Regrettably, only simulations were used there, without experimental confirmation.

The objective of this study is to offer the model that ensures the best energy recovery in both the gradual (normal) and the intensive (ABS) EV braking modes at volatile road conditions. Its main contribution is in utilization of all attributes of the FPID controllers to provide the self-tuning vehicle control in such changing driving obstacles as the driver's demands, tire-road friction instability, road inclination, wind velocity, etc.

In Section II, the focus is on the general model of the EV braking process. More specifically the FPID design, which is particular to this research, is dealt with in Section III. Simulations of the system performance are postponed until Section IV, where they are compared to experiments in order to show the potentiality of the proposed technique and to validate the method. Finally, the results obtained are concluded.

II. MODEL OF THE BRAKING VEHICLE

Drawing on [1] and [3], the following mathematical model of the nonlinear dynamic behaviour of the braking single-wheel quarter-vehicle is considered in this study:

$$F_B^* = F_{air} + F_g + F_x, \quad (1)$$

$$F_{air} = 0.5\rho C_{air} Q (v_v + v_{wind})^2 \operatorname{sgn}(v_v + v_{wind}), \quad (2)$$

$$F_g = mg \sin(\beta), \quad (3)$$

$$F_x = \mu F_z, \quad (4)$$

$$F_z = mg \cos(\beta), \quad (5)$$

$$F = F_B^* + m \frac{dv_v}{dt} = F_B^* + ma, \quad (6)$$

$$T = Fr, \quad T_B^* = F_B^* r, \quad (7)$$

$$\lambda = \frac{v_v - v_w}{v_v} = \frac{v_v - \omega_w r}{v_v}, \quad (8)$$

where

- F_B^* – required braking force;
- F_{air} – air resistance (aerodynamic drag);
- F_g – climbing force;
- F_x – longitudinal force;
- ρ – air density;
- C_{air} – aerodynamic drag coefficient;
- Q – vehicle front area;
- v_v – vehicle longitudinal velocity;
- v_{wind} – wind velocity;
- m – quarter-vehicle mass;
- g – acceleration due to gravity;

β – climbing slope (road incline);
 μ – dimensionless friction factor;
 F_z – transversal force;
 F – application force developed by the drives;
 a – longitudinal deceleration of the vehicle;
 T – application torque developed by the drives;
 r – effective radius of the wheel;
 λ – wheel slip;
 v_w – wheel velocity;
 ω_w – angular speed of the wheel.

To slow down the vehicle moving at a certain initial velocity v_0 , the required braking force (1) must be developed. In order to find this force, the parameters of the equations (2) – (8) have to be identified. Most of them are available from the vehicle passport characteristics (m , Q , r) or can be acquired with on-board sensors (a , v_v , v_{wind} , ω_w). The driver's demand T_B^* takes into account the aerodynamic drag (2) and the road incline (3).

However, estimating the tire-road friction μ in (4) is a complex challenge because this parameter varies with such factors as velocity, load, torque, road surface roughness, tire radius, inflation, wear, etc., and these variations are very difficult to detect. Since the friction factor cannot be sensed, one or the other computational method is applied usually for its estimation.

The friction factor μ is strictly connected with the wheel slip (8). The maximal friction $\mu_{\max}$ takes place at the zero friction-slip gradient, which has a specific level for every road surface. Since $\mu_{\max}$ neither depends on the aerodynamic drag (2) nor on the road incline (3), equation (4) can be converted as

$$a = \mu g. \quad (9)$$

Using (9), replace (6) and (7) with

$$T = T_B^* + mra = T_B^* + mrg\mu. \quad (10)$$

Given that the application torque is easily measured with torque encoders and accurately adjusted in electric braking (EB) systems [40], the close loop torque control can be introduced aiming to develop an application torque T which meets the driver's demand T_B^* without violation any of the following restrictions:

$$\mu < \mu_{\max}, \quad a < a_{\max}, \quad \frac{d\mu}{d\lambda} \geq 0, \quad \frac{dT}{d\lambda} \leq 0 \quad (11)$$

In compliance with (11), two solutions of this task can be offered. In both cases, the controller has to produce an actuating braking torque T^* which, on the one hand, should follow the driver's demand T_B^* , but on the other hand, is to be restricted by an appropriate control parameter. In the first case, the control parameter is a deceleration. This is suitable for the pre-training of the FLC in order to find maximal possible deceleration values for simulation. Data collection can be done through the model running at different road surfaces in an intensive braking mode (imitation of ABS) [17]. In the second case, the control parameter is the torque gradient, which does not depend on the driver's demand. Therefore, this solution fully meets the versatile braking systems.

In Fig. 1, the proposed structure of the MATLAB/Simulink™ model of the versatile braking system is shown. The model is made up of three groups of blocks:

- *Vehicle Friction* group including *Tire-Road*, *Slip*, *Vehicle* and *Gradient* blocks;
- *Drives* group, including *Electric Drive*, *Friction Drive*, *Wheel*, and *Energy* blocks;
- *Control* group, including *FPID* and torque allocation (*TA*) blocks.

Fig. 1. MATLAB/Simulink™ model with FPID torque control.

The *Vehicle Friction* group implements the model (1) – (8). To imitate the on-board velocity sensors, a programmable road estimator is presented in the *Tire-Road* block, which contains a preliminary stored set of the friction-slip lookup tables $\mu(\lambda, v_v)$ acquired from the friction-slip characteristics. The input of this block is associated with the *Slip* block that receives the vehicle and wheel velocity signals v_v , v_w . In the *Vehicle* block, the deceleration rate a is estimated with (9) and the vehicle velocity v_v is counted by integrating the deceleration signal. In a real car, the *Tire-Road* and *Vehicle* blocks are omitted. An additional *Gradient* block aims to derive the torque gradient, which is further directed to the *Control* block.

In the *Drives* group, the *Friction Drive* block, integrated with friction brake (FB), and the adjustable electric drive block with the battery-regenerative EB produce separately their portions of the actuating braking torque T^* generated by the *Control* group. The *Friction Drive* is modelled as a first-order system with dead time. In turn, the electric drive is responsible for the direct torque control, electrical power supply, and energy regeneration. The AC6 Interior *PM Synchronous Motor Drive* block from the Specialized Power Systems MATLAB/Simulink™ library is composed of four main parts. They are: the electrical motor of 288 VDC, 100 kW, the three-phase voltage source inverter fed by a constant voltage from the modified active rectifier, the space vector modulated controller equipped with a three-phase current regulator, and the speed controller operated in the torque regulation mode. Motor torque, speed, power, current, and voltage signals are available at the output of the block.

The system has been discretised with a 20 μs time step. Space vector modulation uses that same sample time to simulate a microcontroller control device. Gear and vehicle inertia are represented by the *Wheel* block. The *Energy* block counts both the electrical fraction of braking power, and the braking energy regenerated back to the supply grid with efficiency η in a given time interval dt :

$$P_E = T_E \omega_w \quad (12)$$

$$W_E = \eta \int P_E dt \quad (13)$$

Two combo-boxes accompany the model, namely, T_B^* to choose the driver's request (*small, average, intensive (ABS)*) and *Road* to assign the current road surface (*dry, wet, ice*).

III. CONTROL SYSTEM DESIGN AND OPERATION

The *Control* group generates actuating braking torque T^* dependently of the driver's request T_B^* and application torque derivative with respect to slip λ , i.e. the gradient $dT/d\lambda$.

Given the complexity and nonlinearity of the system, several FPID master-slave MIMO topologies were tested. In the typical topology, the dual-input master FLC adjusts P, I, and D gains of the slave PID controller at once. In the simplified variant, the same FLC finds an optimal P gain, while I and D gains are adjusted separately. In the predictive controller, the FLC decreases the driver's demand signal any time when the torque gradient drops below the predefined threshold, whereas the PID-controller processes this corrected input.

In the current research, the choice fell on the typical topology of a master FLC and slave PD torque controller. Two input numerical variables (crisps) of the FLC are the torque gradient $dT/d\lambda$ and vehicle velocity v_v . Six fuzzy variable are labelled as Z (zero), VS (very small), S (small), M (mean), B (big), and VB (very big). All membership functions have a triangle shape. The velocity and gradient inputs are greatly nonlinear functions (Fig. 2).

Fig. 2. The *Gradient* membership function of the designed FLC.

Fig. 3. One of control surfaces of the designed FLC.

The Mamdani-style inference engine with “If-Then” modus ponens converts two fuzzy input sets to two fuzzy output sets, P and D, using 36 rules in the rule base. One of two nonlinear control surfaces of the FLC is shown in Fig. 3. An estimated linguistic P and D gain signals are turned back to the real-world output crisp with the help of the weighted average defuzzification method. The slave PD controller converts the torque error input signal into the actuating torque T^* output.

The torque allocation block (*TA*) algorithmically distributes T^* from the FPID output between the front and rear wheels in a fixed ratio [41] and allocates it between FB (T_F^*) and EB (T_E^*) taking into account the real-time state of charge, voltage, and permissible EB current I_E [17]. The T_F^* signal adjusts the FB pressure, while the T_E^* signal manages regeneration. The EB is activated once the control system recognises the actuating torque request T^* , whereas the FB torque does not appear until the electric motor power saturates. As soon as the solo EB becomes insufficient, the system runs FB and EB together. Only in the case when a state-of-charge level exceeds its boundaries, the solo FB remains due to the inability to regenerate. Therefore, the general feature of this strategy lies in including regeneration into all braking scenarios, including the heavy one with ABS.

IV. COMPARING SIMULATIONS TO EXPERIMENTAL RESULTS

In Fig. 4, the ABS simulation diagrams obtained from the designed model at the volatile driving conditions are shown. The performance variables evaluated are the regenerative energy (kJ), the vehicle and wheel velocities (vv, vw), the tire longitudinal slip (L%), and electrical and friction components of the application torque (TE, TF) experienced by the EV.

It was assumed that the EV is moving in a straight-line manoeuvre at 100 km/h fed by the motor with maximal permissible torque of 200 Nm, speed of 157 rad/s, and 2.1 kgm² inertia connected to the wheel through the gear of 10.5 ratio. Due to this transmission, the peak torque on the wheel at heavy braking approached 2000 Nm and wheel angular speed reached 15 rad/s. Mass of the studied EV is 2117 kg and wheel radius 0.2 m. In order to decelerate the EV from the initial velocity as fast as possible, the driver's demand $T_B^* = 550$ Nm was applied to the FPID, which generated an appropriate actuating braking torque T^* , further allocated between the electric T_E^* and friction T_F^* torque references.

At the beginning, the vehicle deceleration was around 30 m/s² on the dry surface. The wheel velocity v_w (light blue) followed the vehicle longitudinal velocity v_v (blue) with some slip (magenta). An electric torque T_E (red) has a limitation of 200 Nm, and the remaining part of the application torque is provided by the friction unit T_F (green).

After the vehicle velocity has been reduced to 50 km/h, the road surface changed suddenly from dry to icy. The system successfully recognises a change in road conditions based on the analysis of the torque-slip gradient. Since a new gradient was detected, the total application torque needed to ensure an intensive stop dropped to 50 Nm. From this instant, the friction torque T_F was no longer requested because the electric torque T_E is sufficient to decelerate the vehicle within the optimal wheel slip area. Therefore, only electric braking was produced further. However, once the

velocity has dropped below 10 km/h, the EB turned off, regeneration resumed, and the FB was launched alone. Using the recovered energy curve (kJ , black) and assuming 50% recuperative efficiency in (13), it turns out that nearly 22 kJ of energy was recovered during braking.

Fig. 4. Simulation diagrams.

To verify the offered methodology, simulation results were compared to the experimental curves.

A hardware-in-the-loop (HIL) electro-hydraulic test rig from ZF TRW Automotive (Koblenz, Germany) was granted for experimentation by TU Ilmenau (Germany). Its vehicle-oriented software IPG CarMaker® (Karlsruhe, Germany) ran the ABS imitator of an electric sport utility vehicle. Unlike the designed quarter-vehicle model described by (1) – (11), a full-vehicle model was used in the HIL test rig. Aerodynamic and climbing factors were neglected in that study.

Fig. 5. Experimental diagrams.

During experimentation, a gradient method and FPID controller were not still developed and a FLC was used instead of the FPID. To determine the road surface without gradient, an original method was proposed in [17]. Every second, starting from ABS activation, the FLC evaluated the maximal deceleration of the vehicle and compared it with the deceleration peaks preliminary calculated using (8). Such momentary friction reset did not affect driving comfort, as it was very rapid, but it indicated whether the road surface changes or remains the same as before. Since the optimal wheel slip looks alike for both the front and rear left and right wheels, the experimental braking diagrams in Fig. 5 introduce only one wheel velocity v_w of the front left wheel that follows the vehicle longitudinal velocity v_v , and appropriate EB and FB wheel torque curves.

As follows from Figs. 4 and 5, both the simulation and the experimentation demonstrate rather similar braking behaviour above 30 km/h. However, an evident chattering phenomenon at low velocity is seen in the experimental diagram. In fact, two reasons may explain its appearance. First, increase of friction due to its static component when the vehicle moves slowly and one of the wheels tends to slip. Second, increased ripples of the power converter responsible for electric drive speed adjustment. Such oscillations, reported also by other researchers, was considered and mitigated in the current study.

V. CONCLUSION

The versatile braking model reveal that by using the presented FPID controller, the control objective is met effectively with good performance specifications. Connection of a master FLC level with a slave PID controller effectively adjusts and stabilises the requested braking torque without overshoot and short settling time. Allocation between the electric and friction brakes allows integrating the advantages of both braking tools. Obtained simulation diagrams largely coincide with the experimental curves. They demonstrate consistently high braking quality regardless of the road surface and slope uncertainty, vehicle initial velocity, and air resistance.

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