A SELF-TUNING ABS CONTROL FOR ELECTROMECHANICAL BRAKING SYSTEMS

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Abstract: One of the main issues of any control strategy for braking systems is to face the many uncertainties due to the strong spread of the system's parameters: road conditions, actuator dynamics, tire behaviour, etc. This paper proposes a self-tuning control for an electromechanical braking system. Electromechanical brake system is a promising replacement for hydraulic brakes in the automotive industry. The proposed control can be seen as a minimum seek algorithm in a highly uncertain situation. Only the measure of the wheel angular speed and a rough measure (or estimate) of the actuator force on the brake pads are required. The proposed control is tested by simulation studies.

Keywords: Active brake control, Vehicle dynamics, Safety, Tires, Actuators.

1. INTRODUCTION

Antilock braking systems (ABS) are now a commonly installed feature in road vehicles. They are designed to stop vehicles as safely and quickly as possible. Safety is achieved by maintaining the steering effectiveness and trying to reduce braking distances over the case where the brakes are controlled by the driver during a "panic stop", see (Robert Bosch GmbH, 2000).

The ABS control systems are based on the typical tire behaviour described in (Pacejka, 2002) and briefly shown in Fig. 1. As demonstrated in (Tsiotras and Canudas de Wit, 2000), optimal braking (in terms of minimum traveled distance) occurs when the longitudinal force F_x operates at its minimum value along the force-slip curve. The slip value corresponding to the minimum longitudinal force F_x depends also on the road conditions, vehicle speed, the normal force, the tire temperature, the steering angle, etc. The main issue of the ABS control strategies is to track the optimal slip value λ_{opt} corresponding to the minimum longitudinal force F_x using the smallest number of sensors, using the cheapest hardware and facing the uncertainties due to both the aging of components and the unknown working and environmental conditions.

Many strategies are based on the slip control, see (Lin and Hsu, 2003), (Tan and Tomizuka, 1990), (Armeni and Mosca, 2003), (Kazemi and Zaviyeh, 2001) and (Savaresi *et al.*, 2005). Theoretically, the method of slip control is the ideal method. However, two problems arise: the (unknown) optimal slip value must be identified and the vehicle speed must be measured or estimated in a low cost and reliable way. To overcome these problems, either pressure measurement have been proposed (Drakunow *et al.*, 1994) or the braking torque is supposed to be known (Lennon and Passino, 1999), (Chamaillard *et al.*, 1994), (Ünsal and Kachroo, 1999). These solutions lead to very good performances, but do not fit the cost requirements. Moreover, many papers give important theoretical results but do not deal with the dynamics of the actuators. Currently most commercial ABSs use a look-up tabular approach based on wheel acceleration thresholds, see (Robert Bosch GmbH, 2000), (Kiencke and Nielsen, 2000) and (Wellstead and Pettit, 1997). These tables are calibrated through iterative laboratory experiments and engineering field tests. Therefore, these systems are not adaptive and issues such as robustness are not addressed.

In recent years the automotive industry is prompting "by wire" mechanism to increase comfort, efficiency and safety. One of such system is the electromechanical brake system, a promising replacement for the hydraulic brakes. An example of electromechanical brake system can be found in (Krishnamurthy *et al.*, 2005).

The work proposed in this paper shows that for electromechanical brakes it is possible to track the optimal slip value by measuring only the wheel speed and knowing when the force on the brake pads is constant. The proposed control can be seen as a minimum seek algorithm in a highly uncertain situation: the slip is neither measured nor estimated, the tire longitudinal force as a function of the tire slip is unknown, time varying, speed varying and it is only supposed to have always a unique minimum; the actuator dynamics is taken into account however it is not *a priori* known by the controller; the brake fading effect is considered. These phenomena are not considered all together in the cited papers.

The paper is organized as follows: the dynamic models of a standard braking system are described in Sec. 2. Based on this model, the basic operating principle of the proposed control is explained in Sec. 3. The control strategy is then described in Sec. 4 and tested by simulations in Sec. 5. Finally some conclusions are drawn.

2. MODEL OF A BRAKING SYSTEM

The braking system consists of three interconnected subsystems: the tires, the vehicle and the electromechanical actuator.

One of the most widely used tire model is the Pacejka's "magic formula", see (Pacejka, 2002). This is a set of static maps which give the tire forces as a function of the longitudinal slip λ, the slip angle α, the camber angle γ and the vertical load $\mathbf{N}_z$. The static maps are obtained by interpolating experimental data. The longitudinal slip rate λ during braking is defined as:

$$\lambda = \frac{\omega\, R_e - v_x}{v_x} \tag{1}$$

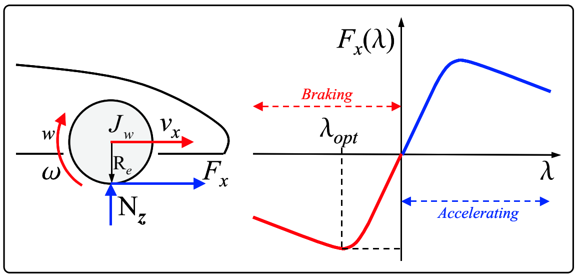

Fig. 1. Basic tire behaviour.

where ω denotes the wheel angular speed, R_e is the rolling radius and v_x is the longitudinal speed of the wheel center in forward direction, see Fig. 1. For a longitudinal braking with constant camber and slip angles, v_x is the vehicle speed and the longitudinal force F_x is $F_x(\lambda) = N_z\,\mu(\lambda)$ where the friction coefficient $\mu(\lambda)$ is given by:

$$\mu(\lambda) = D\sin(C\text{atan}(B\lambda - E(B\lambda - \text{atan}(B\lambda)))) \tag{2}$$

The constants B, C, D and E are chosen to match the experimental data. A qualitative example of the curve $F_x(\lambda)$ is shown in Fig. 1. The curve $F_x(\lambda)$ may be time-varying due to numerous factors, in the sequel of the paper we sum all these effects by considering the longitudinal force as a function of the slip and of the time: $F_x = F_x(\lambda, t)$. The assumption is that for any instant $t = \bar{t}$ the curve $\overline{F}_x(\lambda) = F_x(\lambda, \bar{t})$ has a unique minimum.

The dynamic behaviour of a wheel during braking is described by the differential equation:

$$J_w \dot{\omega} = -K_{brk}(t)\, F - R_e\, F_x(\lambda, t) \tag{3}$$

where F is the force on the brake pads, $K_{brk}(t)$ denotes the brake gain, $\tau_w = -K_{brk}(t)\, F$ is the braking torque. The gain K_{brk} is time-varying due to the brake fading effect: due to the heat generated while braking the brake disk temperature increases and, for common commercial brakes, the gain K_{brk} decreases.

In this work we consider a simplified model of a single wheel braking vehicle, the dynamics of this quarter vehicle model is described by:

$$M \dot{v}_x = F_x(\lambda, t) - F_a \tag{4}$$

where M is the mass of the quarter vehicle and F_a is the aerodynamic drag force. During braking both $F_x(\lambda, t)$ and λ are negative.

The control strategy (proposed in Section Sec. 4) is almost independent from the actuator structure, however to obtain reasonable simulation results, the actuator dynamics is the same as in (Krishnamurthy *et al.*, 2005).

3. BASIC PROPERTIES

Optimal braking occurs when the longitudinal force F_x operates at its minimum value along the force-slip curve. The proposed ABS control can be seen as a minimum-seek algorithm. Since the force-slip curve has always a minimum, it is first necessary to determine whether the operating point lies in the left or in the right region with respect to this minimum. Then the brake actuator is operated to switch from one region to the other. By this way, a "limit cycle" around the optimal slip value arises and it guarantees that the longitudinal force F_x varies around its minimum.

From (1) the time derivative of the slip λ is:

$$\dot{\lambda} = R_e \frac{\dot{\omega}\, v_x - \omega\, \dot{v}_x}{v_x^2} \tag{5}$$

where $\dot{v}_x$ is the longitudinal acceleration A_x of the vehicle. Note that when v_x is reaching zero, the slip λ can vary faster than at high longitudinal speeds. This explains why the worst performance of the ABS controllers happens usually at low speed.

Property 1: *if* $\dot{\omega} \geq 0$ *then* $\dot{\lambda} > 0$. Proof: the sign of the slip derivative $\dot{\lambda}$ is the sign of the term $\dot{\omega}\, v_x - \omega\, \dot{v}_x$. During braking $\dot{v}_x \leq 0$ and ω is limited by the vehicle speed: $v_x \geq R_e\omega \geq 0$. If $\dot{\omega} \geq 0$ then $\dot{\lambda} > 0$ and the slip λ increases. $\diamond$

Property 2: *it exists a limited angular acceleration value* $\dot{\omega}_N$ *such that if* $\dot{\omega} \leq \dot{\omega}_N$ *then* $\dot{\lambda} < 0$. Proof: since the minimum longitudinal acceleration $\dot{v}_x$ (maximum braking at the best conditions) is limited $A_x^{min} \leq \dot{v}_x \leq 0$ and during braking $v_x \geq R_e\omega \geq 0$, it exists a limited angular acceleration value such that $\dot{\lambda} < 0$ is ensured. This acceleration value can be easily found to be $\dot{\omega}_N = A_x^{min}/R_e$ indeed:

$$\dot{\omega} < \frac{A_x^{min}}{R_e} = \frac{A_x^{min}\omega}{R_e\omega} \leq \frac{A_x^{min}\omega}{v_x} \leq \frac{\dot{v}_x\omega}{v_x} \quad \Rightarrow \quad \dot{\lambda} < 0$$

$\diamond$

Let $\dot{\omega}_p \geq 0$ be a design parameter. If $\dot{\omega} \geq \dot{\omega}_p$ then, thanks to Property 1, $\dot{\lambda} > 0$. Let $\dot{\omega}_n \leq \dot{\omega}_N < 0$ be another design parameter, thanks to Property 2, if $\dot{\omega} \leq \dot{\omega}_n$ then $\dot{\lambda} < 0$. Both $\dot{\omega}_p$ and $\dot{\omega}_n$ are "free" parameters that can be tuned to achieve the best possible braking performance.

Let derive equation (3) with respect to the time:

$$J_w\ddot{\omega} = -K_{brk}\,\dot{F} - \dot{K}_{brk}\,F - R_e\,\frac{\partial F_x}{\partial \lambda}\,\dot{\lambda} - R_e\,\frac{\partial F_x}{\partial t} \tag{6}$$

The operating point is in the stable region if $\partial F_x/\partial\lambda > 0$ (see Fig. 1). Assuming $\dot{\lambda} \neq 0$, $\partial F_x/\partial\lambda$ is obtained from (6):

$$\frac{\partial F_x}{\partial \lambda} = -\frac{1}{R_e\,\dot{\lambda}}\left[J_w\ddot{\omega} + K_{brk}\,\dot{F} + \dot{K}_{brk}\,F + R_e\,\frac{\partial F_x}{\partial t}\right] \tag{7}$$

The unknown terms in equation (7) are $K_{brk}(t)$, $\dot{K}_{brk}(t)$ and $\partial F_x/\partial t$, however if F is constant the term $K_{brk}\,\dot{F}$ is zero.

Property 3: *if the force* F *is constant and* $\dot{\omega} \geq \dot{\omega}_p$ *then:*

$$\frac{\partial F_x}{\partial \lambda} > 0 \Leftrightarrow \left[J_w\ddot{\omega} + \dot{K}_{brk}\,F + R_e\,\frac{\partial F_x}{\partial t}\right] < 0 \tag{8}$$

Proof: since $\dot{\omega} \geq \dot{\omega}_p$ from Property 1 follows $\dot{\lambda} > 0$. The above relation can now be derived from equation (7) whit $\dot{F} = 0$. $\diamond$

Property 4: *if the force* F *is constant and* $\dot{\omega} \leq \dot{\omega}_n$ *then:*

$$\frac{\partial F_x}{\partial \lambda} < 0 \Leftrightarrow \left[J_w\ddot{\omega} + \dot{K}_{brk}\,F + R_e\,\frac{\partial F_x}{\partial t}\right] < 0 \tag{9}$$

Proof: since $\dot{\omega} \leq \dot{\omega}_n$ from Property 2 follows $\dot{\lambda} < 0$. The above relation can now be derived from equation (7) whit $\dot{F} = 0$. $\diamond$

4. SELF-TUNING CONTROL

The proposed control strategy is based on the four properties presented above and on the following assumptions or requirements:

A.1) The electromechanical actuator can hold a constant force F and it is possible to detect if the force F on the brake pads is constant.
A.2) The wheel angular speed ω is measured. The wheel angular acceleration $\dot{\omega}$ is measured or estimated.
A.3) The tire characteristic $F_x(\lambda, t)$ has a unique minimum for $-1 \leq \lambda < 0$ at any time.

Let k denote the current sampling instant and let T be the sampling period of the controller.

Properties 3) and 4) of the previous section requires the second derivative of the wheel speed. This is a problem in a real applications where only the wheel speed is measured. To overcome this problem, the acceleration variation $\Delta\dot{\omega}(k)$ is measured instead of the second derivative $\ddot{\omega}$. A method to get a reliable measure, is to compute $\Delta\dot{\omega}(k)$ by linearly interpolating the acceleration values $\dot{\omega}(i)$ for $i = k - n_h, ..., k$ where $n_h \in \mathbb{N}$ is a design parameter that denotes the number of sampling periods that are needed to get a reliable measure of $\Delta\dot{\omega}(k)$. With a small n_h, if the acceleration variation is small the measurement noise will affect the measure. With good low-noise sensors n_h can be small.

The key idea of the control strategy is to use properties 3 and 4 (with $\Delta\dot{\omega}(k)$ instead of $\ddot{\omega}$) to detect when the operating point is about passing from the stable to the unstable region (property 4) or vice versa (property 3) assuming a constant force F on the brake pads.

Due to the unknown terms in (8) and (9), the threshold $\Delta\dot{\omega}(k) \leq 0$ is used to approximately detect the instant when $\lambda = \lambda_{opt}$ (the two inequalities involving the square brackets have the

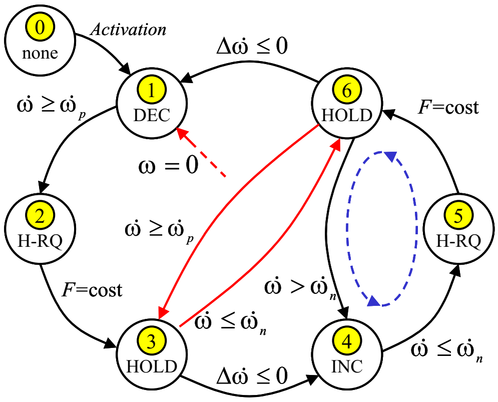

Fig. 2. State chart of the proposed control.

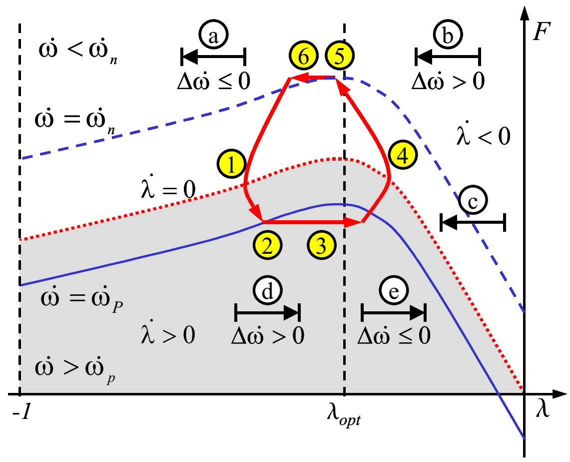

Fig. 3. Basic working cycle on the F-λ plane.

same form). By this way, the two unknown terms in (8) and (9) and the error between $\Delta\dot\omega(k)$ and $\ddot\omega$ have the effect to advance or to delay the corresponding control action. However if the unknown terms in (8) and (9) are sufficiently small, this delay/advance is not significative and does not affect the efficiency of the proposed control (see the simulation examples given next). If the two unknown terms are neglected (as often is in the literature) the switching of the control action when $\Delta\dot\omega(k) \le 0$ is essentially exact.

Let F_R denote the force reference for the electromechanical actuator. The three basic control action are the following:

DECREASE: $F_r(k+1) = F_r(k) + \Delta F_{dec} T$ decrease the force F on the brake pads;
HOLD: $F_R(k+1) = F_R(k)$ maintain a constant force F on the brake pads;
INCREASE: $F_r(k+1) = F_r(k) + \Delta F_{inc} T$ increase the force F on the brake pads.

The parameter $\Delta F_{dec} < 0$ gives the decreasing rate, $\Delta F_{inc} > 0$ gives the increasing rate, both has to take into account the actuator dynamics.

The proposed control is based on a 7 state algorithm. The state chart of the algorithm is shown in Fig. 2. The basic working cycle is given by the sequence of states (1)-(2)-(3)-(4)-(5)-(6)-(1). A schematic representation of the basic working cycle is represented on the F-λ plane, see Fig. 3 . This simplified representation is obtained computing the force F form equation (3) when $F_x(\lambda, t) = \overline{F}_x(\lambda)$ and $\dot K_{brk} = 0$. For a constant value of $\dot\omega = a$ the curve $F(\lambda, a)$ has the same shape of the curve $\overline{F}_x(\lambda)$, moreover if $a_2 > a_1$ then $F(\lambda, a_2) < F(\lambda, a_1)$ consequently any acceleration $\dot\omega$ defines an unique curve $F(\lambda, \dot\omega)$ that does not intersect any other $F(\lambda, \dot\omega)$ curve. For any $\dot\omega$ the peak of the curve $F(\lambda, \dot\omega)$ happens for $\lambda = \lambda_{opt}$.

The description of the 7 states is the following (the events within each state are checked following the given sequence; for some states, a simple initialization assignment is executed once when the algorithm enters the state):

(0) *Control action*: normal braking.
Operations:
if "emergency brake" then next state = (1).
Description: The ABS control is not active. If a "emergency brake" is detected the ABS control is activated. The activation mode does not affect the behaviour of the proposed control and it is out of the scope of the paper.

(1) *Control action*: DECREASE
Events:
- if $\dot\omega \ge \dot\omega_p$ then next state = (2).

Description: This control action is established as soon as the wheel operating point is supposed to be in the unstable region or when the wheel is locked. By decreasing the force on the brake pads, the term $R_e F_x(\lambda, t)$ will become dominant in equation (3) and the wheel acceleration will become positive.

(2) *Control action*: HOLD
Events:
- if $\omega = 0$ then next state = (1).
- if F is constant then next state = (3).

Description: The actuators are requested to hold a constant value for F. Due to the actuator dynamics, this state lasts until the force F is detected to be constant.

(3) *Control action*: HOLD
Initialization: $k_0 := k$
Events:
- if $\omega = 0$ then next state = (1).
- if $\dot\omega \le \dot\omega_n$ then next state = (6).
- if $(k - k_0) \ge n_h$ and $\Delta\dot\omega(k) \le 0$ then next state = (1). Case (e) of Fig 3.

Description: Actuators delay was compensated while in state (2) or (5), therefore the force F can be considered constant.
If $(k - k_0) \ge n_h$ the measure of $\Delta\dot\omega(k)$ can be considered as reliable.
Property 3 is used to approximately detect when $\lambda > \lambda_{opt}$. The two cases (d) and (e) of

Fig 3 are possible with respect to the sign of $\Delta\dot{\omega}(k)$. The second operation may be helpful in case of abrupt changes of adhesion.

(4) *Control command*: INCREASE
Events:
- if $\dot{\omega} \leq \dot{\omega}_N$ then next state = (5).

Description: The INCREASE control action is established as soon as the operating point is supposed to be in the stable region. By increasing the brake pressure, the term $K_{brk}\, F$ will become dominant in equation (3) and the wheel acceleration will become negative.

(5) *Control command*: HOLD
Initialization: $k_0 := k$
Events:
- if $\omega = 0$ then next state = (1).
- if F is constant then next state = (6).

Description: The same as state (2).

(6) *Control action*: HOLD
Initialization: $k_0 := k$
Events:
- if $\omega = 0$ then next state = (1).
- if $\dot{\omega} \geq \dot{\omega}_p$ then next state = (3).
- if $(k - k_0) \geq n_h$ and $\Delta\dot{\omega}(k) \leq 0$ then next state = (1). Case (a) of Fig 3.
- if $\dot{\omega} > \dot{\omega}_n$ then next state = (4). Case (c) of Fig 3.

Description: Analog to state (3). Property 4 is used to approximately detect when $\lambda < \lambda_{opt}$. The cases (a), (b) and (c) of Fig 3 are possible. Case (a) corresponds to property 4. In case (b) the HOLD control is kept since $\lambda > \lambda_{opt}$ and $\dot{\lambda} < 0$. Case (c) allows to re-establish an acceleration lower than $\dot{\omega}_n$. By this way a sub-cycle (6)-(4)-(5)-(6) can arise to make λ closer to λ_{opt}. The second condition may be helpful in case of abrupt changes of adhesion.

5. SIMULATION RESULTS

To obtain reasonable simulation results the actuator dynamics is the same as in (Krishnamurthy *et al.*, 2005). To verify the self-tuning properties, braking on varying road conditions have been considered. Fig. 7 shows the two force-slip curves that represent the tire behaviour in the two extreme conditions. The transition between the two conditions is obtained by modifying the coefficients of equation (2) along the traveled distance. Due to the speed and road condition variation, the optimal slip λ_{opt} and the optimal wheel speed ω_{opt} are not constant. The control is activated when the traveled distance is greater than 5m. The two simulations described next consider a fading effect that introduce a gain reduction of 1/3 along the traveled distance. Simulation 1 has the sampling time $T = 2$ms, $n_h = 2$, road condition dry-wet-dry with abrupt adhesion variation. Simulation 2 has a greater sampling time $T = 5$ms, $n_h = 2$, road condition dry-wet-dry with gradual variations. The simulation results are shown from Fig. 4 to Fig. 10.

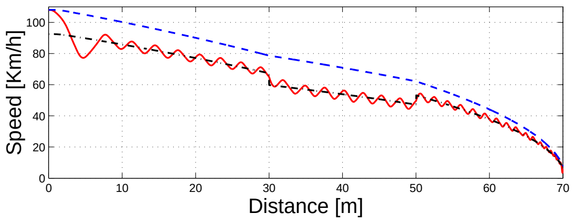

Fig. 4. Simulation 1: vehicle speed v_x (--), wheel peripheral speed $R_e\,\omega$ (–) and computed optimal wheel speed $R_e\,\omega_{opt}$ (–·).

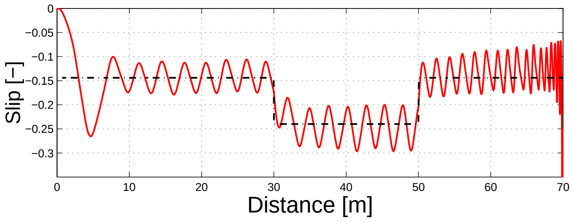

Fig. 5. Simulation 1: slip λ (–) and computed optimal slip λ_{opt} (–·).

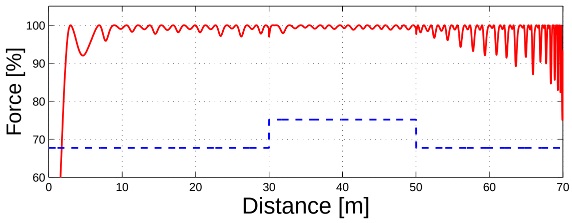

Fig. 6. Simulation 1: ratio between the longitudinal force F_x and the maximum achievable force F_{opt}(–). Comparison with the same ratio at wheel lock-up (--).

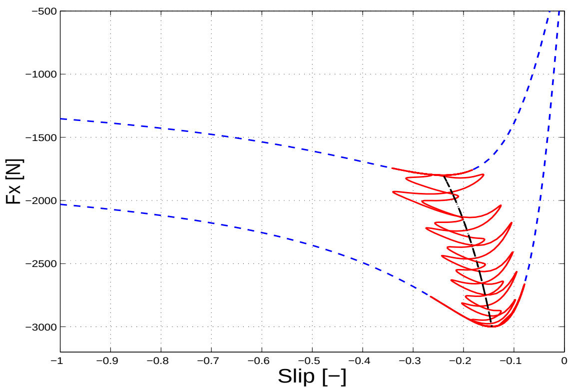

Fig. 7. Simulation 2: tire longitudinal force F_x(–), tire characteristics $F_{x1}(\lambda)$ and $F_{x2}(\lambda)$ (--), optimal operating points (–·).

Both simulations allows to verify that it is possible to track the optimal slip and the wheel speed values by using the proposed control strategy. The good tracking of the optimal slip ensures that the braking force is always around its maximum. The performances decay only at very low speed when the wheel locks-up, this is due to the rapid slip

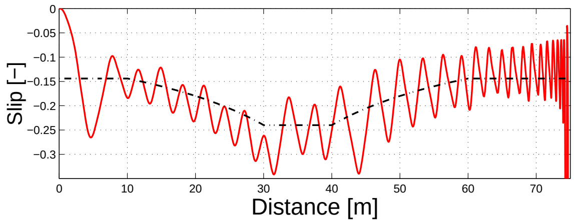

Fig. 8. Simulation 2: slip λ (–) and computed optimal slip λ_{opt} (−·).

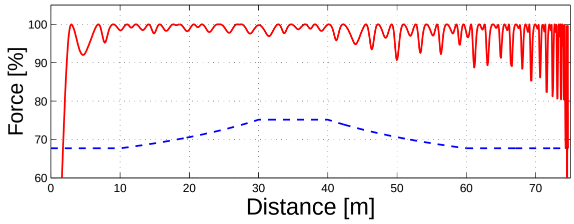

Fig. 9. Simulation 2: ratio between the longitudinal force F_x and the maximum achievable force F_{opt}(–). Comparison with the same ratio at wheel lock-up (- -).

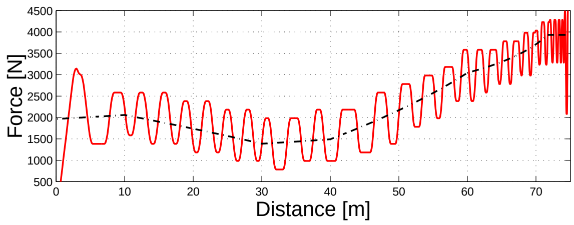

Fig. 10. Simulation 2: braking force F (–) and optimal braking force F_{opt} (−·)

variation at low speed due to equation (5). As well known, the controllers based on acceleration thresholds induce oscillations on the braking pressure (see Fig. 10), the average increase (especially in the last 15m) of the braking force is the consequence of the fading effect.

6. CONCLUSIONS

A self-tuning ABS control electromechanical braking systems has been proposed. The paper has shown that it is possible to track the optimal slip value by measuring only the wheel speed and estimating the wheel acceleration. As shown in the simulation studies, the proposed control strategy is almost hardware independent, robust with parameters variations and takes into account the dynamics of the actuators.

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