A SELF-TUNING ABS CONTROL FOR ELECTROMECHANICAL BRAKING SYSTEMS

Riccardo Morselli* Roberto Zanasi*

* DII, University of Modena and Reggio Emilia
Via Vignolese 905, 41100 Modena, Italy
Phone: +39 59 2056161, Fax +39 59 2056126
e-mail: morselli.riccardo@unimore.it

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. Copyright © 2006 IFAC

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 $\lambda_{opt}$ 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 Zavieh, 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 $\lambda$, the slip angle $\alpha$, the camber angle $\gamma$ and the vertical load $N_z$. The static maps are obtained by interpolating experimental data. The longitudinal slip rate $\dot{\lambda}$ during braking is defined as:

$$\dot{\lambda} = \frac{\omega R_e - v_x}{v_x}$$  \hspace{1cm} (1)

Fig. 1. Basic tire behaviour.

where $\omega$ 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:

$$\mu(\lambda) = D \sin(C\tan(B\lambda - E(B\lambda - \tan(B\lambda))))$$  \hspace{1cm} (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 $t=t$ the curve $\overline{F}_x(\lambda) = F_x(\lambda,\overline{t})$ has a unique minimum.

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

$$J_w \ddot{\omega} = -K_{brk}(t) F - R_e F_x(\lambda,t)$$  \hspace{1cm} (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:

$$M \ddot{v}_x = F_x(\lambda,t) - F_a$$  \hspace{1cm} (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 $\lambda$ 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 force-slip curve has always a minimum, it is respect to this minimum. Then the brake actuator can be seen as a minimum-seek algorithm. Since slip value can be easily found to be _v_ vehicle. Note that when _v_ is constant and _v_ 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 _ω_ ≥ 0 then _λ_ > 0. Proof: the sign of the slip derivative _λ_ is the sign of the term _ω_ _v_ _ω_ _v_ _ω_ during braking _v_ _ω_ ≤ 0 and _ω_ is limited by the vehicle speed: _v_ _ω_ ≥ _R_ _ω_ ≥ 0. If _ω_ ≥ 0 then _λ_ > 0 and the slip _λ_ increases.

**Property 2:** it exists a limited angular acceleration value _ω_ _N_ such that if _ω_ ≤ _ω_ _N_ then _λ_ < 0. Proof: since the minimum longitudinal acceleration _v_ _ω_ (maximum braking at the best conditions) is limited _A_ _ω_ _min_ ≤ _v_ _ω_ ≤ 0 and during braking _v_ _ω_ ≥ _R_ _ω_ ≥ 0, it exists a limited angular acceleration value such that _λ_ < 0 is ensured. This acceleration value can be easily found to be _ω_ _N_ = _A_ _ω_ _min_ / _R_ _e_ indeed:

\[
_ω_ < \frac{A_\omega^{\text{min}}}{R_e} = \frac{A_\omega^{\text{min}}R_\omega}{R_e} \leq \frac{A_\omega^{\text{min}}R_\omega}{v_x} \leq \frac{\dot{v}_xR_\omega}{v_x} \Rightarrow \dot{\lambda} < 0
\]

Let _ω_ _p_ ≥ 0 be a design parameter. If _ω_ ≥ _ω_ _p_ then, thanks to Property 1, _λ_ > 0. Let _ω_ _n_ ≤ _ω_ _N_ < 0 be another design parameter, thanks to Property 2, if _ω_ ≤ _ω_ _n_ then _λ_ < 0. Both _ω_ _p_ and _ω_ _n_ are “free” parameters that can be tuned to achieve the best possible braking performance.

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

\[
J_w\ddot{\omega} = -K_{brk} \dot{F} - \dot{K}_{brk} F - R_e \frac{\partial F_x}{\partial \lambda} \dot{\lambda} - R_e \frac{\partial F_x}{\partial t} (6)
\]

The operating point is in the stable region if _∂F_x/∂λ_ > 0 (see Fig. 1). Assuming _λ_ ≠ 0, _∂F_x/∂λ_ is obtained from (6):

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

The unknown terms in equation (7) are _K_{brk}(t)_ , _\dot{K}_{brk}(t)_ and _∂F_x/∂t_ , however if _F_ is constant the term _K_{brk} F_ is zero.

**Property 3:** if the force _F_ is constant and _ω_ ≥ _ω_ _p_ then:

\[
\frac{\partial F_x}{\partial \lambda} > 0 \Leftrightarrow \left[ J_w \ddot{\omega} + K_{brk} F + R_e \frac{\partial F_x}{\partial t} \right] < 0 (8)
\]

Proof: since _ω_ ≥ _ω_ _p_ from Property 1 follows _λ_ > 0. The above relation can now be derived from equation (7) whit _F_ = 0. \footnote{Property 3}

**Property 4:** if the force _F_ is constant and _ω_ ≤ _ω_ _n_ then:

\[
\frac{\partial F_x}{\partial \lambda} < 0 \Leftrightarrow \left[ J_w \ddot{\omega} + K_{brk} F + R_e \frac{\partial F_x}{\partial t} \right] < 0 (9)
\]

Proof: since _ω_ ≤ _ω_ _n_ from Property 2 follows _λ_ < 0. The above relation can now be derived from equation (7) whit _F_ = 0. \footnote{Property 4}

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 _ω_ is measured or estimated.

A.3) The tire characteristic _F_ _x_ (λ, t) has a unique minimum for _−1_ ≤ _λ_ < 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 _Δω(k)_ is measured instead of the second derivative _ω_. A method to get a reliable measure, is to compute _Δω(k)_ by linearly interpolating the acceleration values _ω(i)_ for _i = k - n_h, ..., k_ where _n_h_ ∈ _N_ is a design parameter that denotes the number of sampling periods that are needed to get a reliable measure of _Δω(k)_.

The key idea of the control strategy is to use properties 3 and 4 (with _Δω(k)_ instead of _ω_ ) 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 _Δω(k)_ ≤ 0 is used to approximately detect the instant when _λ_ = _λ_ _opt_ (the two inequalities involving the square brackets have the
same form). By this way, the two unknown terms in (8) and (9) and the error between $\dot{\omega}(k)$ and $\dot{\omega}$ 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 $\Delta \dot{\omega}(k) \leq 0$ is essentially exact.

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

**DECREASE:** $F_r(k+1) = F_r(k) + \Delta F_{\text{dec}} T$

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:** $F_r(k+1) = F_r(k) + \Delta F_{\text{inc}} T$

increase the force $F$ on the brake pads.

The parameter $\Delta F_{\text{dec}} < 0$ gives the decreasing rate, $\Delta F_{\text{inc}} > 0$ 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 $F$-$\lambda$ plane, see Fig. 3. This simplified representation is obtained computing the force $F$ form equation (3) when $F_r(\lambda, t) = F_0(\lambda)$ and $K_{\text{brk}} = 0$. For a constant value of $\dot{\omega} = a$ the curve $F(\lambda, a)$ has the same shape of the curve $F_0(\lambda)$, 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 $\lambda = \lambda_{\text{opt}}$.

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 $\dot{\omega} \geq \dot{\omega}_p$ 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 $R_c F_r(\lambda, t)$ 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 $\dot{\omega} \leq \dot{\omega}_n$ then next state = (6).
- if $(k - k_0) \geq n_h$ and $\Delta \dot{\omega}(k) \leq 0$ 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 $(k - k_0) \geq n_h$ the measure of $\Delta \dot{\omega}(k)$ can be considered as reliable.

Property 3 is used to approximately detect when $\lambda > \lambda_{\text{opt}}$. The two cases (d) and (e) of
Fig 3 are possible with respect to the sign of \( \Delta \omega(k) \). The second operation may be helpful in case of abrupt changes of adhesion.

(4) Control command: INCREASE

Events:
- if \( \omega \leq \omega_n \) 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 \( \omega > \omega_n \) then next state = (3).
- if \( (k - k_0) \geq n_h \) and \( \Delta \omega(k) \leq 0 \) then next state = (1). Case (a) of Fig 3.
- if \( \omega > \omega_n \) 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 \( \lambda < 0 \). 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 \( \lambda \) closer to \( \lambda_{opt} \). 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 \( \lambda_{opt} \) and the optimal wheel speed \( \omega_{opt} \) 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 \( T = 2\text{ms}, n_h = 2 \), road condition dry-wet-dry with abrupt adhesion variation. Simulation 2 has a greater sampling time \( T = 5\text{ms} \), \( n_h = 2 \), road condition dry-wet-dry with gradual variations. The simulation results are shown from Fig. 4 to Fig. 10.

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
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.

REFERENCES


