Fuzzy Sliding Mode Control for Anti-lock Braking Systems

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Abstract—A combination of two control methods, sliding mode and fuzzy, for the wheel slip control in anti-lock braking system, is presented in this paper. The fuzzy block is used to determine the values of key parameters important for establishing switching function dynamics. It is demonstrated, via performed experiments, that proposed control algorithm gives good system performances.

I. INTRODUCTION

Nowadays vehicles cannot be imagined without anti-lock braking system (ABS), an electronic system, situated inside the vehicle control panel, which is activated when the possibility of blocking wheel is met. The main idea is that, during suddenly braking, the wheels do not block completely. On the other hand, if that happens, the vehicle control probably will be lost and it can skid in an unwanted direction. ABS provides that driver can normally operate with his vehicle, despite the fact that the brake pedal is pushed to the end. In that way, driver has control over the vehicle during suddenly braking and the stopping distance is also significantly reduced.

The coefficient which characterizes the adhesion between the wheels and the road surface is known as road adhesion coefficient μ. It is defined as the proportion between the friction force and the normal load of the vehicle. This coefficient is in nonlinear dependence on the wheel slip λ, defined as the relative speed difference between the wheel and vehicle. Most of the controllers are designed to regulate wheel slip on the pre-set level, in desired range, so that the road adhesion coefficient has its maximal value for that level of wheel slip.

The ABS dynamic has strong nonlinear nature, so a robust control method tends to be a logical choice. Sliding mode is of particular interest in nonlinear systems and its main goal is to force a system state to a certain prescribed manifold-sliding surface, determined by the so-called switching function. In many papers authors deal with ABS sliding mode control (SMC) problem. Traditional SMC enhanced by a grey system theory is proposed in [1]. The desired wheel slip is considered to be constant and the regulator problem is analyzed. The wheel slip control through the engine torque control is given in [2]. The moving sliding surface is used to ensure that the system state is always on sliding surface. Digital simulation results of ABS with conventional SMC, where hydraulic brake dynamics is included during the design, is shown in [3]. In [4], the SMC, allowing the maximum value of the wheel-road friction force during the braking phase, without a priori knowledge of optimal slip, is discussed. The adaptive SMC of vehicle traction is considered in [5].

There are also many papers describing applications of fuzzy control (FC) of ABS. Its main major advantage comes from applicability to the systems when the mathematical model is not exactly known, as in our case. One way to design a control algorithm is to use linearized models of ABS in the vicinity of operation points. Such approach in design of linear PI, gain-scheduling, and FC is thoroughly discussed in [6]. A self-tuning PID control scheme with an application to ABS via combinations of fuzzy and genetic algorithms, is developed in [7]. A Takagi-Sugeno fuzzy controller and an interpolative fuzzy controller for slip control in ABS are proposed in [8].

Over the last few years, there is a significant research effort in combining the two topologies [9] in a manner that serves to reduce the limitations of the sliding mode, while still maintaining the guarantees of global uniform stability and invariance to matched disturbances [10]. In this paper, we tend to improve control method presented in the paper [11], by using the combination of above mentioned control methods. The fuzzy block of the fuzzy-sliding mode control (FSMC) is used to determine key parameter values, used in the definition of SMC switching function. Its dynamics is defined by using the so-called reaching law method [12]. More precisely, we use the constant plus proportional variant of this control approach.

II. LABORATORY ABS EXPERIMENTAL SETUP

A. Description of ABS laboratory setup

The ABS experimental setup [13], shown in Fig. 1, consists of two wheels which are permanently in rolling contact. While the upper wheel is equipped with a tire, representing the vehicle wheel, the lower one, representing the relative road motion, has smooth surface which can be covered with some materials to simulate the surface of the road.

The encoders, mounted on the each wheel, measure the angles with accuracy of 2π/2048=0.175°. ABS experimental setup does not have sensors for direct measuring of wheel angular velocities, which are needed for determining of ABS mathematical model, so it is estimated by simplest Euler formula with the sample time period of 0.5ms. The upper wheel is equipped in the disk brake system connected via hydraulic coupling to the brake lever which by the tight side and tightening pulley is driven by the small dc motor. We would like to point out that the steel cord causes strong nonlinearity and limitation of control input signal on 50% of its maximum nominal value.
The lower wheel is coupled to the big flat dc motor whose task is to accelerate the wheel. During the braking phase its power supply is switched off. Both dc motors are controlled by pulse-width modulation (PWM) signals with frequency of 3.5 kHz. The system depicted in Fig. 1 is connected to PC via hardware interface corresponding to the control unit of traditional ABS.

B. ABS Mathematical model

Free-body diagram of ABS for quarter vehicle is given in Fig. 2 [13, 14]. The ABS model is quite simplified due to several assumptions: (i) we consider only longitudinal vehicle and angular wheel motions, while the lateral and vertical motions are neglected; (ii) rolling resistance force is ignored, as it is very small due to braking, and (iii) the interaction of wheels is also neglected. Despite this model simplification, it preserves fundamental characteristics of real system.

Let us introduce the following auxiliary variables:

\[ s = \text{sgn}(r_2 x_2 - r_1 x_1), \quad s_1 = \text{sgn}(x_1), \quad s_2 = \text{sgn}(x_2) \]  \hspace{1cm} (1)

where \( x_1 \) represents the angular velocity of the upper wheel, \( x_2 \) is the angular velocity of the lower wheel and \( r_1, r_2 \) represents the radius of the upper and lower wheel, respectively.

The equation of the upper wheel motion, based on graphical model from Fig. 2, can be obtained as:

\[ J_1 \ddot{x}_1 = F_x r_1 s \mu(\lambda) - d_1 x_1 - s_1 (M_{10} + M_1) \]  \hspace{1cm} (2)

by using the Newton’s second law, where: \( J_1 \) is the moment of inertia, \( d_1 \) is the viscous friction coefficient, \( M_{10} \) is the static friction of the upper wheel and \( M_1 \) represents the brake torque. According to (2), the friction force is assumed to be proportional to the normal pressing force \( F_n \), where \( \mu(\lambda) \) is the coefficient of proportion called the road adhesion coefficient.

In similar manner, the lower wheel motion can be described by:

\[ J_2 \ddot{x}_2 = -F_x r_2 s \mu(\lambda) - d_2 x_2 - s_2 M_{20} \]  \hspace{1cm} (3)

where \( J_2 \) is the moment of inertia, \( d_2 \) is the viscous friction coefficient and \( M_{20} \) is the static friction of the lower wheel. To derive the normal force \( F_n \), we write the sum of torques corresponding to the point A in Fig. 2 as:

\[ F_n L (\sin \varphi - s \mu(\lambda) \cos \varphi) = M_g + s_1 M_1 + s_1 M_{10} + d_1 x_1 \]  \hspace{1cm} (4)

yielding:

\[ F_n = \frac{M_g + s_1 M_1 + s_1 M_{10} + d_1 x_1}{L (\sin \varphi - s \mu(\lambda) \cos \varphi)} \]  \hspace{1cm} (5)

where \( M_g \) represents gravitational and shock absorber torques acting on the balance lever, \( L \) is the distance between the contact point of the wheels and the rotational axis of the balance and \( \varphi \) is the angle between the normal in the contact point and the line \( L \).

Replacing \( F_n \) from (5) in (2) and (3), the model becomes:

\[ J_1 \ddot{x}_1 = \frac{M_g + s_1 M_1 + s_1 M_{10} + d_1 x_1}{L (\sin \varphi - s \mu(\lambda) \cos \varphi)} - r_1 s \mu(\lambda) - s_1 M_{10} - d_1 x_1 \]  \hspace{1cm} (6)

\[ J_2 \ddot{x}_2 = \frac{M_g + s_1 M_1 + s_1 M_{10} + d_1 x_1}{L (\sin \varphi - s \mu(\lambda) \cos \varphi)} - r_2 s \mu(\lambda) - s_2 M_{20} \]  \hspace{1cm} (6)

While the wheel angular velocity would match the forward vehicle velocity in the normal operating conditions, in the course of the braking and the acceleration phases these velocities differs one from
other. Their difference is called a wheel slip \( \lambda \), and it is defined as:

\[
\lambda = \begin{cases} 
\frac{r_2 x_2 - r_1 x_1}{r_2 x_2}, & r_2 x_2 \geq r_1 x_1, \quad x_1 \geq 0, \quad x_2 \geq 0, \\
\frac{r_1 x_2 - r_2 x_1}{r_1 x_1}, & r_2 x_2 < r_1 x_1, \quad x_1 \geq 0, \quad x_2 \geq 0, \\
\frac{r_1 x_2 - r_2 x_1}{r_1 x_1}, & r_2 x_2 < r_1 x_1, \quad x_1 < 0, \quad x_2 < 0, \\
1, & x_1 < 0, \quad x_1 \geq 0, \\
1, & x_1 \geq 0, \quad x_2 < 0.
\end{cases}
\] (7)

for all model operating conditions. A zero wheel slip represents the equality of the wheel and the vehicle velocities, while the slip value equal to one tells us that the tire is not rotating and the wheels are skidding on the road surface, meaning that the vehicle is no more steerable.

The road adhesion coefficient \( \mu(\lambda) \) is a nonlinear function of wheel slip and other physical variables, and one of its models can be given by:

\[
\mu(\lambda) = \frac{w_A^p}{a + \lambda^p} + w_1 \lambda^q + w_2 \lambda^r + w_3 \lambda.
\] (8)

Equation (6) now can be rewritten as:

\[
\begin{aligned}
\dot{x}_1 &= S(\lambda, x_1, x_2)(c_{11} x_1 + c_{12}) + c_{13} x_1 + \\
&+ c_{14} + \left( c_{15} S(\lambda, x_1, x_2) + c_{16} \right) s_1 (x_1) M_1, \\
\dot{x}_2 &= S(\lambda, x_1, x_2)(c_{21} x_1 + c_{22}) + c_{23} x_1 + \\
&+ c_{24} + c_{25} S(\lambda, x_1, x_2) s_1 (x_1) M_1.
\end{aligned}
\] (9)

where

\[
\begin{aligned}
S(\lambda) &= \frac{\mu(\lambda)}{L (\sin \varphi - s \mu(\lambda) \cos \varphi)}, \\
c_{11} &= \frac{r_1 d_1}{J_1}, \\
c_{12} &= \frac{s_1 M_{10} + M_{10}}{J_1}, \\
c_{13} &= \frac{r_1 d_1}{J_1}, \\
c_{14} &= \frac{s_1 M_{10}}{J_1}, \\
c_{15} &= \frac{r_1}{J_1}, \\
c_{16} &= \frac{1}{J_1}, \\
c_{21} &= -\frac{r_2 d_1}{J_2}, \\
c_{22} &= -\frac{s_1 M_{10} + M_{10}}{J_1}, \\
c_{23} &= \frac{r_2 d_1}{J_2}, \\
c_{24} &= -\frac{s_1 M_{20}}{J_2}, \\
c_{25} &= -\frac{r_2}{J_2}.
\end{aligned}
\] (10)

The ABS braking process starts with acceleration phase in which the lower wheel is accelerated up to desired speed. After that, the braking phase begins. Now, the brakes are applied causing that the wheel speed decreases and the force acting on the wheel increases.

This, in other hand, causes the slippage between the tire and the road surface. In that case, the wheel speed will be lower than vehicle speed, i.e. \( r_2 x_2 \geq r_1 x_1 \) and \( x_1 > 0, \) \( x_2 > 0, \) so the wheel slip can be determined by:

\[
\lambda = \frac{r_2 x_2 - r_1 x_1}{r_2 x_2}.
\] (11)

In that case, since \( s = s_1 = s_2 = 1, \) (9) is rewritten as:

\[
\begin{aligned}
\dot{x}_1 &= S(\lambda)(c_{11} x_1 + c_{12}) + c_{13} x_1 + \\
&+ c_{14} + \left( c_{15} S(\lambda) + c_{16} \right) M_1, \\
\dot{x}_2 &= S(\lambda)(c_{21} x_1 + c_{22}) + c_{23} x_1 + \\
&+ c_{24} + c_{25} S(\lambda) M_1.
\end{aligned}
\] (12)

ABS controller is designed to maintain the wheel slip at a particular level, where the corresponding friction force (i.e. road adhesion coefficient) reaches its maximum value. That is why we need to determine quarter vehicle model with wheel slip as a controlled variable.

Differentiating (11) results in:

\[
\dot{\lambda} = -\frac{r_1}{r_2 x_2} \dot{x}_1 + \frac{r_2 x_1}{r_2 x_2} \dot{x}_2.
\] (13)

Putting (12) in (13) and taking into account that:

\[
\lambda = 1 - \frac{r_2 x_1}{r_2 x_2} \Rightarrow \dot{x}_1 = \frac{r_1}{r_2} (1 - \lambda) x_2,
\] (14)

finally yields:

\[
\dot{\lambda} = f(\lambda, x_2) + g(\lambda, x_2) M_1, \quad x_2 \neq 0,
\] (15)

where

\[
\begin{aligned}
f(\lambda, x_2) &= \frac{\left( S(\lambda) c_{11} + c_{13} \right) (1 - \lambda)}{r_1 x_2} + \\
&+ \left( c_{14} \right) \left( S(\lambda) c_{11} + c_{14} \right) M_1, \\
&+ \frac{1}{x_2} \left( S(\lambda) c_{21} \frac{r_2}{r_1} (1 - \lambda) + c_{23} \right) x_3 + \\
&+ S(\lambda) c_{22} + c_{24},
\end{aligned}
\] (16)

\[
\begin{aligned}
g(\lambda, x_2) &= \frac{r_1}{r_2 x_2} \left( c_{25} S(\lambda) + c_{16} - \frac{r_2}{r_1} c_{25} S(\lambda) (1 - \lambda) \right).
\end{aligned}
\] (17)

The equation (15) represents the final continuous-time ABS mathematical model. The parameters of used ABS mathematical model are presented in Table I, given below.
TABLE I
PARAMETERS OF ABS MODEL

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III. SLIDING MODE CONTROLLER DESIGN WITH ADDITIONAL FUZZY BLOCK

A. SMC based on Constant Plus Proportional Reaching Law

Since the system is of the first order, the switching function is selected as:

$$\sigma = \lambda - \lambda_s,$$  \hspace{1cm} (18)

where $\lambda_s$ is the constant reference wheel slip. The main control design objective is to find control providing

$$\sigma = 0$$   \hspace{1cm} and consequently     \hspace{1cm} $$\lambda = \lambda_s.$$ The switching function dynamics is defined via the constant plus proportional reaching law \[12\]:

$$\sigma = \left\{ \begin{array}{ll} \frac{g}{g_s} f + \frac{g}{g_s} \left( \frac{1 - g}{g_s} \right) \sigma - M_s \sigma^2 - M_s |\sigma| & \text{if } \sigma > 0, \\ \frac{g}{g_s} f + \frac{g}{g_s} \left( \frac{1 - g}{g_s} \right) \sigma - M_s \sigma^2 - M_s |\sigma| & \text{if } \sigma < 0, \\ \frac{g}{g_s} f + \frac{g}{g_s} \left( \frac{1 - g}{g_s} \right) \sigma - M_s \sigma^2 - M_s |\sigma| & \text{if } \sigma = 0, \end{array} \right.$$  \hspace{1cm} (19)

enabling the finite reaching time determined by:

$$t_r < \ln \left( \left( M_r |\sigma| + M_c \right) / |M_c| \right) / M_r.$$  \hspace{1cm} (20)

By substituting (19) in (15), the control torque becomes:

$$M_1 = -g(\lambda, x_2)^{-1} \left( f(\lambda, x_2) + M_c |\sigma| + M_r \text{sgn}(\sigma) \right).$$  \hspace{1cm} (21)

Since the nominal values of $f$ and $g$ are $f_s$ and $g_s$ respectively, the control torque implemented in real control is:

$$M_1 = -g_s(\lambda, x_2)^{-1} \left( f_s(\lambda, x_2) + M_c |\sigma| + M_r \text{sgn}(\sigma) \right).$$  \hspace{1cm} (22)

For the sake of simplicity, we denote $g(\lambda, x_2) = g_s(\lambda, x_2) = g_s, \quad f(\lambda, x_2) = f_s$ and $f_s(\lambda, x_2) = f_s$. We assume that     \hspace{1cm} $|f_s| < F, \quad |f - f_s| < \varepsilon_f$ and

$$\left| 1 - g/g_s \right| < \varepsilon_g < 1,$$ where $F, \varepsilon_f$ and $\varepsilon_g$ are the positive real constants, as well as $g/g_s > 0$.

Substituting (22) in (15) gives:

$$\sigma = f - f_s + f_s \left( 1 - g/g_s \right) - \frac{g}{g_s} M_s \sigma - \frac{g}{g_s} M_r \text{sgn}(\sigma).$$  \hspace{1cm} (23)

The existence condition of sliding mode:

$$\sigma = \left\{ \begin{array}{ll} g \left( f - f_s \right) \sigma + g \left( f_s - g \right) \sigma - M_s \sigma^2 - M_s |\sigma| & \text{if } \sigma > 0, \\ g \left( f - f_s \right) \sigma + g \left( f_s - g \right) \sigma - M_s \sigma^2 - M_s |\sigma| & \text{if } \sigma < 0, \\ g \left( f - f_s \right) \sigma + g \left( f_s - g \right) \sigma - M_s \sigma^2 - M_s |\sigma| & \text{if } \sigma = 0, \end{array} \right.$$  \hspace{1cm} (24)

is fulfilled, if $M_s$ is chosen in accordance with:

$$M_s > \frac{\varepsilon_f + F \varepsilon_g}{1 - \varepsilon_g},$$  \hspace{1cm} (25)

which is satisfied if

$$M_s > \frac{\varepsilon_f + F \varepsilon_g}{1 - \varepsilon_g}.$$  \hspace{1cm} (26)

Since

$$1 - \varepsilon_g < g/g_s < 1 + \varepsilon_g \Rightarrow 0 \Rightarrow \left| 1 + \varepsilon_g \right| g < g_s \Rightarrow 0.$$  \hspace{1cm} (27)

$M_1$ should be selected in such a manner to make the reaching phase as fast as possible.

B. Fuzzy control

In this paper, fuzzy blocks are used to evaluate the variable control gains $(M_{c}, M_{r})$ and the control signal depending on the distance of the state vector from the sliding surface and the velocity of approaching. The parameter values defining sliding mode controller with additional fuzzy block can be chosen in such a way to obtain the best system behavior with respect to specific criteria.

We observe only distance from sliding manifold, so the sign of the switching function is not important. Having this in mind, we define fuzzy rules based on absolute values of $\sigma$. In order to avoid the redundancy, both fuzzy blocks have only one input and one output. First one, used for determining $M_s$, as input uses the switching function $\sigma$, while second one uses the first
derivative of $\sigma$. The main idea for controller design is that a large control gain should be applied only when the state vector is far away from the sliding manifold, so that system moves towards the sliding manifold fast, but when the trajectory of the system is near the sliding manifold, the control signal should decrease smoothly. In such a way system becomes faster invariant to disturbances and chattering is suppressed.

Therefore, the complete rule base of the $M_{c}$-fuzzy block could take the following form:

\[
\begin{align*}
\text{if } & \sigma = Z \text{ then } M_{c} = Z, \\
\text{if } & \sigma = S \text{ then } M_{c} = S, \\
\text{if } & \sigma = B \text{ then } M_{c} = B.
\end{align*}
\]

The rule base for the $M_{p}$-fuzzy block is:

\[
\begin{align*}
\text{if } & \sigma = Z \text{ then } M_{p} = B, \\
\text{if } & \sigma = S \text{ then } M_{p} = S, \\
\text{if } & \sigma = B \text{ then } M_{p} = Z.
\end{align*}
\]

One choice for normalized fuzzy sets of the fuzzy variables $\sigma$, $\dot{\sigma}$, $M_{c}$, $M_{p} = \{Z, S, B\}$ is shown in Figs. 3 and 4.

The nonlinear mapping from the input to the output of the fuzzy block is done by the trial and error and experience basis. The membership functions are triangular with the following centers:

a) $M_{c}$-fuzzy block: $\{0, 0.05, 0.1\}$ for input $\sigma$ and $\{0, 5, 10\}$ for output $M_{c}$,

b) $M_{p}$-fuzzy block: $\{0, 50, 100\}$ for input $\sigma$ and $\{100, 125, 150\}$ for output $M_{p}$.

As defuzzification method, according to the control problem specificity, center of gravity (COG) method was chosen. The control law formed in such a way can be rewritten:

\[
M_{c} = -g_{c}(\lambda, x_{2})^{-1}\left(f_{c}(\lambda, x_{2}) + \frac{M_{p}}{M_{c}}\sigma + M_{c} \text{sgn}(\sigma) \right). \tag{30}
\]

IV. EXPERIMENTAL RESULTS

Practical verification of the proposed control algorithm is performed on the previously described experimental ABS setup (Fig. 1). As we mentioned earlier, this is the experimental environment in which the users verify their ABS control algorithms in real time using MATLAB and Simulink toolboxes. The results are given in Figs. 5 and 6. Each figure consists of three subplots representing the wheel and vehicle velocities, the wheel slip and the control brake torque, respectively. The reference wheel slip is constant $\lambda_{r} = 0.2$. Control signal is limited to the range 0.2-0.4 in order to avoid saturation. Sampling period is set to 5ms.

![Figure 5. ABS responses with the sliding mode control](image1)

![Figure 6. ABS responses with the fuzzy sliding mode control](image2)

First, the experimental results obtained by using sliding mode control (22) are given in Fig. 5. $M_{c}$ and $M_{p}$ are chosen to be 120 and 5, respectively, according to the derived condition (26) in Section III. As it can be seen, the sliding mode exists, but the chattering is not too large since the constant component of reaching law control is taken to be as small as possible. The proportional term, ensuring existence of sliding motion with faster reaching time, enables such selection of $M_{c}$.

The experimental results with applied fuzzy sliding control method (30) are given in Fig. 6. These results are slightly better in comparison to the previous. As we can see, the reference slip is better tracked thanks to the
variable control gains. Moreover, the chattering is reduced, because of the fact that sliding dynamics significantly slows with reaching the sliding manifold.

V. CONCLUSION

In this paper we proposed a new control method, based on the combination of fuzzy and sliding mode control, for wheel slip control in anti-lock braking system (ABS). The mathematical model of ABS in continuous-time domain is developed first, and, then, the proposed control algorithm is applied to ABS model. Finally, the verification of the control algorithm is performed comparing it with other sliding mode control algorithm. Experimental results confirmed good performances of the given control and it is shown that chattering is significantly reduced, and the tracking behavior is also improved.

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