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ANTI-LOCK BRAKING SYSTEM MODEL

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Abstract. *This paper presents the modeling of an anti-lock braking system (ABS). The conception process and construction of the model are described step by step. The ABS model was attached to a vehicle model to simulate possible braking conditions. The simulations were performed for low, medium and high adherences between the ground and the tire. The parameters used to verify the performance advantage obtained with ABS braking in relation to passive braking were the stopping distance, the wheel slip and the ABS index of performance (ABSIP). The simulation results using a standard anti-lock braking system (ABS) control logic allow us to conclude that the vehicle stopping distance with the anti-lock braking system is reduced when compared to the passive braking case (according to the system purpose) and that the wheel slipping is much closed to the desired wheel slip, confirming the appropriate working of the tested anti-lock braking system control logic.*

Keywords: *brake modeling, passive braking, ABS braking.*

1. Introduction

Nowadays anti-lock braking systems (ABS) integrate automotive braking systems of most of automotive vehicles being manufactured. Its objective is to increase the braking efficiency, improving the directional control and reducing the stopping distance of the vehicles in braking actions. Basically, as the wheel slip increases past a critical point where it is possible that lateral stability, and hence the ability to steer the vehicle could be lost, the controller releases the pressure of brakes. The basic idea of ABS is to attain the optimum vehicle deceleration rate without sacrificing the necessary stability and steer ability of the vehicle while maintaining the maximum safety to the driver and passengers.

In anti-lock braking systems, the main objective of the control algorithms is to limit the wheel braking torque, so, most of the ABS systems in the published literature utilize the wheel slip estimate to control the wheel cylinder pressure through a set of valves and a pump in order to regulate the wheel braking torque (Anwar and Ashrafi, 2003). One of the major problems in designing the controller for the vehicle anti-lock braking systems is finding the appropriate control algorithms to rejecting the parameter uncertainties such as friction coefficient, road elevation, wind gust, vehicle absolute speed, and so on (Kueon and Bedi, 1995). See Lennon and Passino (1999) for more information about automobile brake system automatic controllers.

When activated correctly, ABS can provide drivers with the ability to stop a vehicle in shorter distances and allow for more vehicle control under heavy braking than conventional brake systems. This is especially true under dry or wet conditions. However, it is believed that many drivers are either unaware of the correct method of activation or they revert back to the old method of pumping the brakes when they are faced with a hard braking situation (Mollenhauer et. Al., 1997).

The influence of several uncertain parameters in ABS control algorithms as mentioned above, the improper brake activation and the current technology available in brake systems development suggest a previous virtual modeling to evaluate and make the necessary adjustments of the ABS algorithms before its experimental test validation.

In this paper, it is presented the development of a dynamic model of an ABS system. This model was developed in such a way that it can be used to simulate many possible braking conditions allowing ABS control logics evaluating through mathematical simulation. We present simulation results using a standard control logic described in Semmler et al. (2004).

Automobile ABS conventional systems usually have 4 angular speed sensors, while the control logic adopted would need 10 sensors: 4 for wheels angular speed, 4 for calipers pressure measurement and 2 for longitudinal and transverse acceleration of the vehicle. The ABS modeling seeks to support a preview evaluation about the system performance before its experimental implementation in the test bench (see Figure 1). The next step is to provide a passenger car with the necessary equipment and sensors to perform the ABS braking tests.

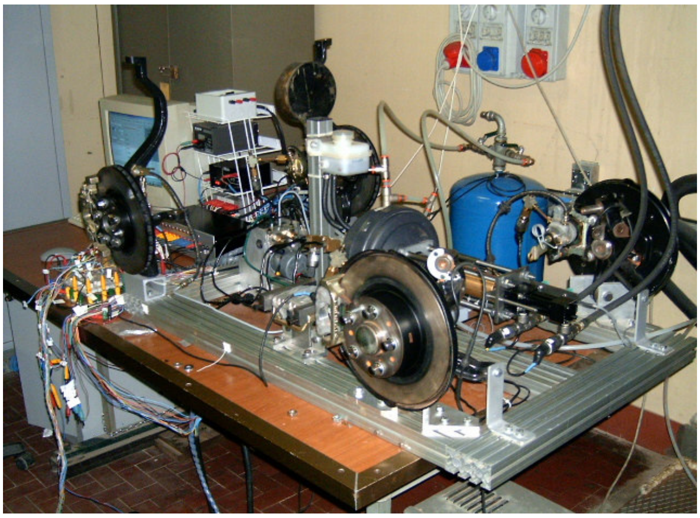

Figure 1 - Braking test bench, Politecnico di Torino (Turin, Italy)

This paper is based on the work of Infantini (2003) [6] in the brake systems development at Politecnico di Torino in partnership with FIAT Research Centre (CRF), Turin, Italy.

2. ABS Control Logic

To design a system to control the slipping of the wheels it is necessary to take into account the behavior of some system states, such as: torque supplied by the engine (M_a) and by the brakes (M_b), the longitudinal force (F_x) that acts on the wheel, and the polar moment of inertia (J). Eq. (1) presents the equilibrium equation for the torques on the wheel:

$$J\dot{w} = r_{dyn}F_x - eF_n - M_f + M_a - M_b \tag{1}$$

where: r_{dyn} is the rolling radius, w is the angular velocity of the wheel, F_n is the normal force on the wheel and M_f is the bearing friction torque.

Figure 2 presents a graphic interpretation of Eq. (1):

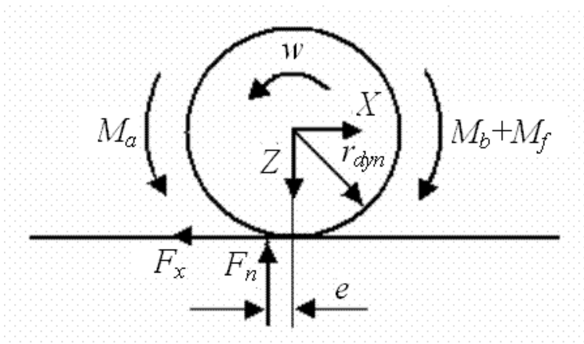

Figure 2. Forces and torques acting on the wheel

The term $eF_n + M_f$ can be substituted for:

$$M^* = eF_n + M_f \tag{2}$$

Then the Eq. (1) can be rewritten as:

$$J\dot{w} = r_{dyn}F_x - M^* + M_a - M_b \tag{3}$$

The applied torque ($M_{b(mc)}$) on the wheel is given by:

$$M_{b(mc)} = Ac^{*}.r_b p_b = k_b p_b \tag{4}$$

where: A is the effective piston area, c^* is the brake factor, r_b is the effective friction radius and p_b is the brake pressure. The wheel slip (S) is defined according to the two following situations: braking or acceleration of the wheel. In the braking case:

$$S = \frac{v - wr_{dyn}}{v} \tag{5}$$

According to the above presented equations, through the linearization of Eq. (3), neglecting M^*, is obtained the following control law that, as we will demonstrated by means of simulations, is able to keep the wheel closed to the desired slip (S_d):

$$M_{b(abs)} = M_a + r_{dyn}F_x - \frac{J\dot{v}}{r_{dyn}}(S-1) + \frac{Jv}{r_{dyn}}\xi \tag{6}$$

where the sub index abs refers to ABS system and ξ is defined by

$$\xi = k_p(S_d - S) + k_d(\dot{S}_d - \dot{S}) \tag{7}$$

and v is the wheel longitudinal velocity, k_p>0 is a proportional factor and k_d>0 is a derivative factor (Semmler et al., 2004).

3. ABS Control and Modeling

The ABS control logic was modeled in the software Matlab/Simulink® (CAVALLO; SETOLA; VASCA, 2002). It was attached to a vehicle model of an Alfa Romeo 166 (Cavallo et al, 2002).

The ABS control logic presented in the section 2 is obtained by the subsystem shows in Figure 3. This subsystem calculates $M_{b(abs)}$.

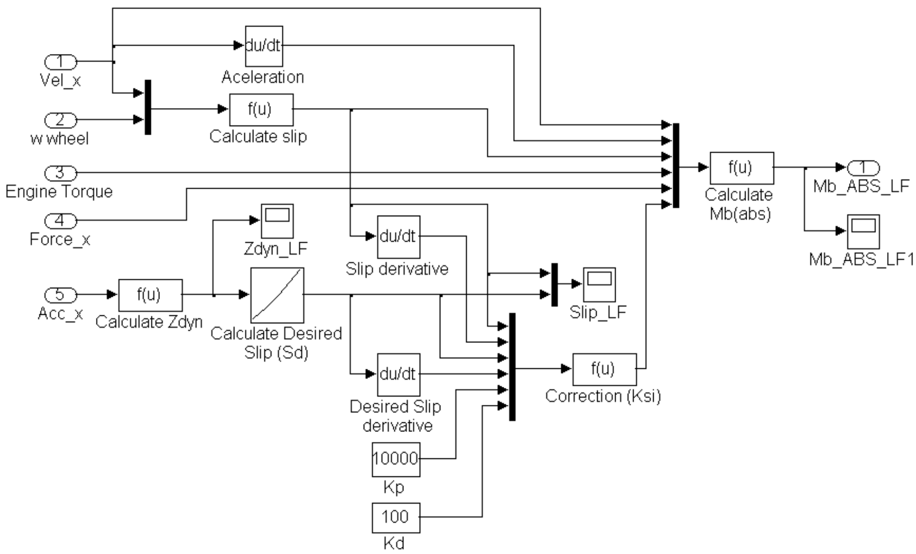

Figure 3 - ABS control subsystem (for $M_{b(abs)}$ computation)

The ABS control system compares the value of $M_{b(abs)}$ calculated by Eq. (6) with the value of $M_{b(mc)}$ calculated by Eq. (4) and, in view of the result, increase, reduce or maintain the pressure in the calipters of the wheels. The table below presents this control logic.

Table 1: ABS control logic table

Conditions	Signals sent to the solenoid valves	
	ISO	DUMP
If $M_{b(mc)} \leq M_{b(abs)}$ and $p_{ref} \geq 5bar \Rightarrow$ increase the pressure	0	0
If $M_{b(mc)} > M_{b(abs)}$ and $p_{ref} \geq 5bar \Rightarrow$ reduce the pressure	1	1
If $p_{ref} < 5bar \Rightarrow$ keep the pressure	1	0

The reference pressure (p_{ref}) is obtained by:

$$p_{ref} = \left| p_b - p_{abs} \right| \tag{8}$$

where p_{abs} can be calculated by Eq. (4) replacing $M_{b(mc)}$ by $M_{b(abs)}$.

3.1. Characterization of the desired wheel slip

The characterization of the desired wheel slip (S_{opt}) is very important in order to take advantage of the maximum adherence between ground and tires. According to Eq. (9)

$$\mu_x = \frac{F_x}{Z} \tag{9}$$

So, how bigger the longitudinal force that acts on the tire is, bigger is the adherence (observe that the vertical load (Z) is constant).

The longitudinal force can be calculated by the experimental equation of Pacejka as presented by Genta, 2000.

$$F_x = D\sin(Carctg(B(1-E)(S+S_h) + Earctg(B(S+S_h)))) + S_v \tag{10}$$

where:

$$B = \left(\frac{K}{\alpha + d} \right)^{\frac{1}{n}} \tag{11}$$

$$C = b_0 \tag{12}$$

$$D = \mu_p Z \tag{13}$$

$$E = b_6 Z^2 + b_7 Z + b_8 \tag{14}$$

$$S_h = b_9 Z + b_{10} \tag{15}$$

$$S_v = 0 \tag{16}$$

where the parameters b_i are functions of the tire, and K, d and n are functions of experimental curves.
Equation (10) is used to scan the maximum longitudinal forces, and consequently, the maximum adherence through the systematic variation of the vertical loads (Z) and wheel slip values.

So, according with this methodology, it was changed the wheel slip value from -1 to 1 to each peak adherence (μ_p) and each load as showed in Table 2 to obtain the maximum longitudinal force acting on the wheel and consequently, the optimum wheel slip.

Table 2: Optimum wheel slip

Z [KN]	μ_p	S	S_{opt}
1	0.1,0.2,0.3,...,0.9	-1 to 1	0.2
3	0.1,0.2,0.3,...,0.9	-1 to 1	0.24
5	0.1,0.2,0.3,...,0.9	-1 to 1	0.28
8	0.1,0.2,0.3,...,0.9	-1 to 1	0.36

The graphic below shows the results obtained for a load of 5kN. It is observed that the optimum wheel slip is the same for each load and different adherence conditions. This behavior is explained by the tire characteristics.

It is then possible to calculate the desired wheel slip as a function of the variation of the vertical load on the wheels during the braking situation. According to the forces diagram schematized in Figure 5 it is obtained the equations (19) and (22) for the equilibrium condition. It is observed that the variation of the load on the wheels is a function of the variation of the vehicle acceleration during the braking.

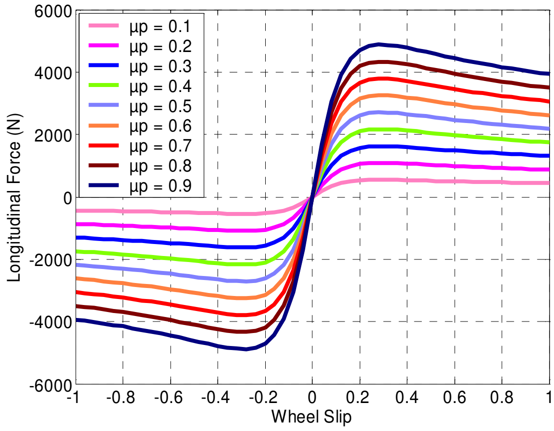

Figure 4 - Longitudinal Forces versus wheel slip for Z=5kN.

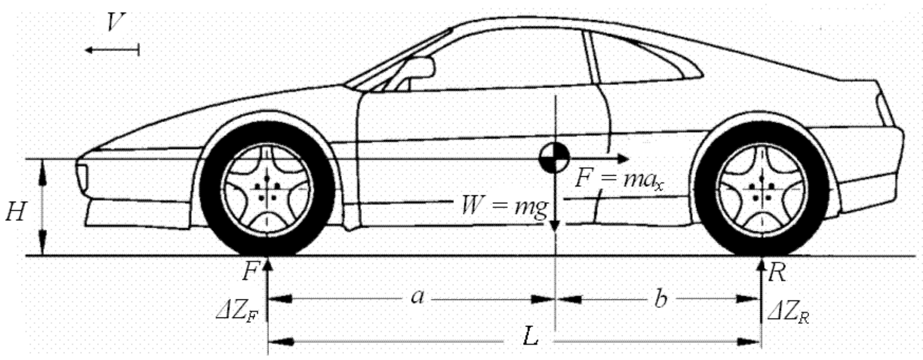

Figure 5 - Force diagram in braking condition

$$\sum M_R = 0 \tag{17}$$

$$mgb - ma_xH - \Delta Z_F L = 0 \tag{18}$$

For each front wheel:

$$\Delta Z_F = \frac{mgb - ma_xH}{2L} \tag{19}$$

$$\sum M_F = 0 \tag{20}$$

$$-mga_x - ma_xH + \Delta Z_R L = 0 \tag{21}$$

For each back wheel:

$$\Delta Z_R = \frac{mgb + ma_xH}{2L} \tag{22}$$

The desired wheel slip is necessary for the calculation of the correction factor (ξ) in Eq. (7). This equation controls the oscillation of the torque to be applied to the wheel by the ABS system according to its control logic. More details about the ABS control and the slip optimization can be found in Infantini (2003, 2004).

4. Results

Some simulations were performed to verify the operation of the ABS model and to evaluate the gain obtained in ABS braking respect to passive braking. The parameters taken into consideration to evaluate its performance were the wheel slip, the stopping distance and the ABS index of performance (*ABSIP*).

The *ABSIP* is defined by:

$$ABSIP = \frac{a_{abs}}{a_{ps}} \tag{23}$$

where a_{abs} and a_{ps} are the average acceleration in ABS braking and passive braking respectively. The *ABSIP* consists in a convenient way of comparing the passive braking with ABS braking. Therefore, it is not an absolute measure of braking performance. An *ABSIP*, such as 1.17, represents an improvement of 17% of ABS braking upon passive braking (Marshek and Cuderman II, 2002).

The tests were performed with a vehicle initial velocity (v_0) of 100 Km/h and three cases of adherence: snow (μ = 0.3), wet asphalt (μ = 0.6) and dry asphalt (μ = 0.9). The vehicle brakes 2 seconds after the simulation starting.

As it was expected, the wheels lock in passive braking (S = 1). It can be seen in Figure 6. In ABS braking condition it is verified that the wheel slip converge to the desired slip after 6s. The wheel slip achieves a peak after the brake pedal is activated and takes about 4s to reach the desired wheel slip (see Figure 7).

The stopping distance is reduced from 147.8 m to 122.4 m with the ABS action and there is an improvement of 22.8% in ABS braking over passive braking calculated through the *ABSIP* in snow slipping conditions.

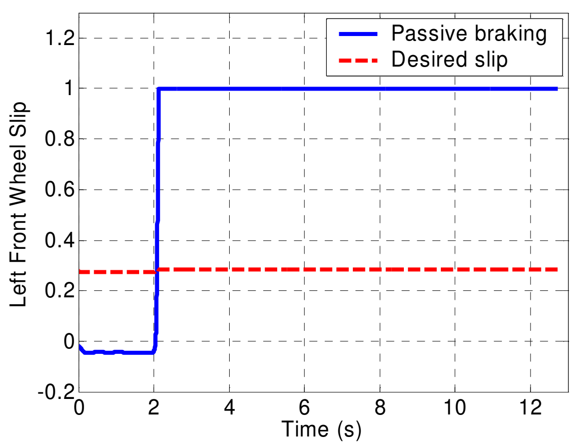

Figure 6 - Passive braking, snow.

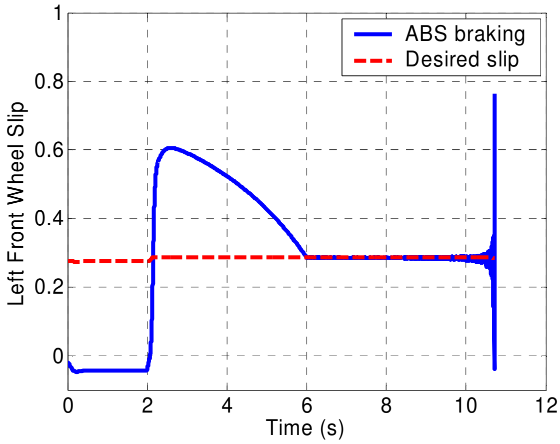

Figure 7 - ABS braking, snow.

Figure 8 shows the locking of the wheel (S = 1) in passive braking. Figure 9 shows a wheel slip peak less accentuated than in Figure 7. The wheel slip reaches the desired wheel slip quickly.

The stopping distance is reduced from 75.4m to 62.3m with the ABS action and there is an improvement of 22.9% in ABS braking over passive braking calculated through the *ABSIP* in wet asphalt.

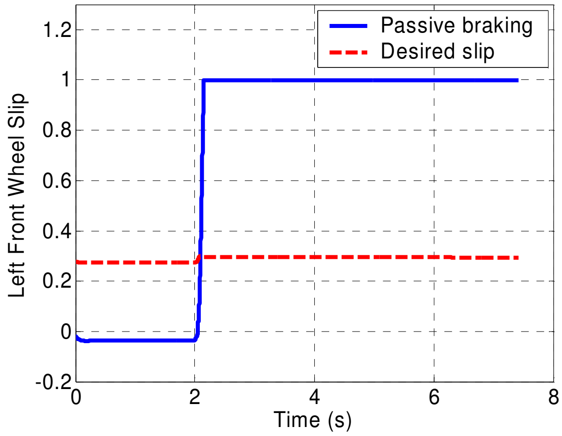

Figure 8 - Passive braking, wet asphalt.

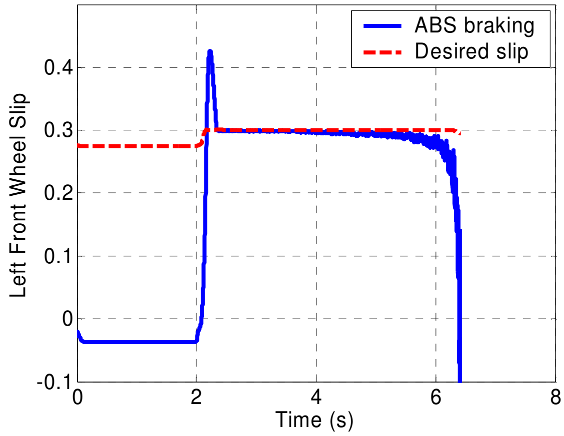

Figure 9 - ABS braking, wet asphalt.

Figure 10 shows the locking of the wheel (S = 1) in passive braking. Figure 11 shows a wheel slip peak less accentuated than in Figure 9 and Figure 7. The wheel slip reaches the desired slip quickly.

The stopping distance is reduced from 50.61m to 43.0m with the ABS action and there is an improvement of 19.9% in ABS braking over passive braking calculated through the *ABSIP* in dry asphalt.

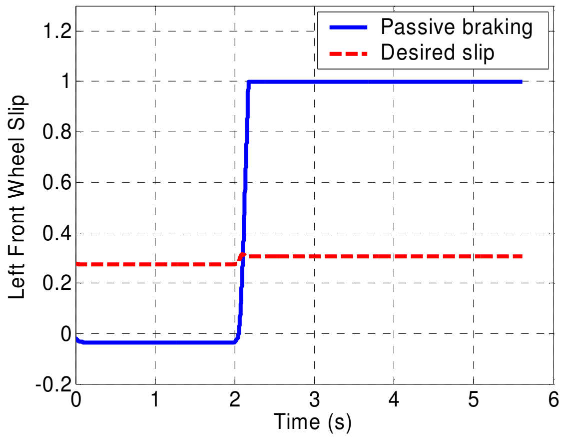

Figure 10 - Passive braking, dry asphalt.

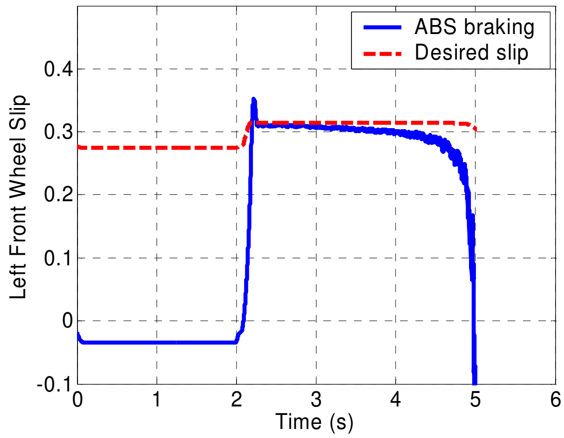

Figure 11 - ABS braking, dry asphalt.

5. Conclusion

The development of an anti-lock braking system (ABS) model was presented. The ABS model was attached to a vehicle model (Alfa Romeo 166 data) to simulate possible braking conditions.

The simulations results presented in this work are the left front wheel slip in three ground conditions: snow, wet and dry asphalt, for both, passive and ABS braking. The behavior of the wheel slip for the others wheels is quite similar to this one, and hence, it is not showed. These simulations compare the hard (full stroke) brake situation for passive and ABS cases. The parameters taken into account to evaluate its performance were the wheel slip, the stopping distance and the ABS index of performance (*ABSIP*).

As it was expected, for the passive case, there is wheel locking while in the ABS case, the slipping curves converge to the desired value, due to the feedback control logic action, reducing the braking distance and, therefore, improving the braking efficiency for all simulated cases.

Furthermore, the control logic studied avoids the wheels locking for every adherence conditions simulated, increasing the vehicle stability and safety. The ABS performance improves for lower adherence conditions as it can be verified by the higher values of *ABSIP* for the snow case when compared to the dry and wet asphalt cases.

Future works will attain simulations for others vehicle initial velocities, steering input and varying the way of brake pedal activating.. Furthermore, the ABS control logic will be applied in the experimental test bench showed in Figure 1 and its results compared with the simulation data.

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