A Conceptual Dynamic Model for Electronic Braking System

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ABSTRACT

Advanced Driver Assistance Systems (ADAS) has been one of the fastest-growing sectors in the automotive industry. They are particularly designed to increase car safety and also driving comfort and experience. ADAS technologies embrace many functions like Cruise Control (CC), Adaptive Cruise Control (ACC), Electronic Stability Control (ESC), Assistant Parking System (APS), Lane Keeping Assistant (LKA), Anti-lock Brake System (ABS), among others. Distributed under several networked control modules, those functions acquire sensor data continuously to predict undesired situations and acting on vehicle subsystems on avoiding unsafe and uncomfortable driving conditions. One of the most promising ADAS to prevent or mitigate the severity of a crash is the Electronic Braking System (EBS). An EBS acts to optimize drive and braking operation adjusting braking force and torque for each respective braking situation. It receives sensor data (i.e., radar and/or camera) to detect front objects, braking distance, and speed acting once unsafe conditions are detected. Taking into account the importance of EBS function, this paper aims to design a parameterized dynamic model (PDM) of the automobile braking systems with ABS/ESC functionalities. In particular, this paper highlights the development of modelling equations for each braking system components (from brake pedal up to caliper assembly). The proposed model will be designed and implemented using Matlab/Simulink integrated environment. Simulation and test bench outcomes highlight successful results to represent the Fiat Strada brake systems with ESC support.

Keywords: Eletronic Braking, Dynamic Model, Simulation, ABS/ESC.

RESUMO

O Advanced Driver Assistance Systems (ADAS) tem sido um dos setores de crescimento mais rápido na indústria automotiva. Eles são especialmente promovidos para aumentar a segurança do carro e também o conforto e a experiência de direção. As tecnologias ADAS abrangem muitas funções como Cruise Control (CC), Adaptive Cruise Control (ACC), Electronic Stability Control (ESC), Assistant Parking System (APS), Lanekeeping Assistant (LKA), Antibloqueio Brake System (ABS), entre outras. Distribuídas em vários módulos de controle em rede, essas funções adquirem dados do sensor continuamente para prever indesejadas e atuando nos subsistemas evitando condições de direção inseguras e desconfortáveis. Um dos ADAS mais promissores para prevenir ou mitigar a gravidade de uma colisão é o Sistema de Frenagem Eletrônico (EBS). Um EBS atua para otimizar uma operação de tração e frenagem, ajustando a força e o torque de frenagem para cada situação de frenagem. Ele recebe dados do sensor (ou seja, radar e / ou câmera) para detectar objetos frontais, distância de frenagem e velocidade agindo assim que condições inseguras são detectadas. Levando em consideração a importância da função EBS, este trabalho tem como objetivo o projeto de modelo dinâmico parametrizado (PDM) dos sistemas de frenagem de automóveis com características ABS / ESC. Em particular, este destaca o desenvolvimento das equações de modelagem para cada componente do sistema de frenagem (desde o pedal do freio até a montagem da pinça). O modelo proposto escolhido e implementado em ambiente integrado Matlab / Simulink. Os resultados da simulação e da bancada de teste destacam os resultados bem-sucedidos para representar os sistemas de freio Fiat Strada com suporte ESC.

Palavras Chave: Frenagem Eletrônica, Modelo Dinâmico, Simulação, ABS/ESC.

INTRODUCTION

The braking system is a crucial vehicle safety system to stop the car according to the driver's intention without losing control of the vehicle. Functions such as ACC and ESC have been increasingly common with the responsibility of keeping the vehicle at a safe distance from the vehicle in front ensuring stability, avoiding loss of control, among other functions.

This paper introduces a conceptual non-linear Dynamic Model, in order to improve the understanding of the braking system as a whole so that it is possible to perform an electronic braking through an ABS/ESC module, motivated by its use in vehicles equipped with the ADAS system. and in autonomous cars.

LITERATURE REVIEW

The ADAS concept has been discussed for more than 40 years, in the 1980s and 1990s the Platoon concept was presented, which is the concept of a platoon of vehicles guided by a leading vehicle, where the leading car gradually increased its speed and the platoon of vehicles followed this increase in speed [1], another proposal was for a platoon of vehicles that suggested the possibility of an automated highway and the Platoon would be guided through that highway, with the possibility of changing lanes for the entire platoon [2].

Currently, concepts such as autonomous vehicles and ACC are already well consolidated in the automotive environment, with several sensors and cameras around the car so that a good reading of the environment in which it is located is made so that a good strategy for driving the vehicle can be drawn up [3] acting on the accelerator, engine torque, brake, and other parameters.

The automotive braking system consists of mechanical, hydraulic and electronic components, where ABS is the electronic component that has access to the vehicle's four-wheel valves, which makes it the main element for carrying out a control strategy and maintaining the system original car braking [1] [2] [3] [4].

SYSTEM MODELING

In light vehicles, an electronic braking system is often made up by the following components (see Figure 1):

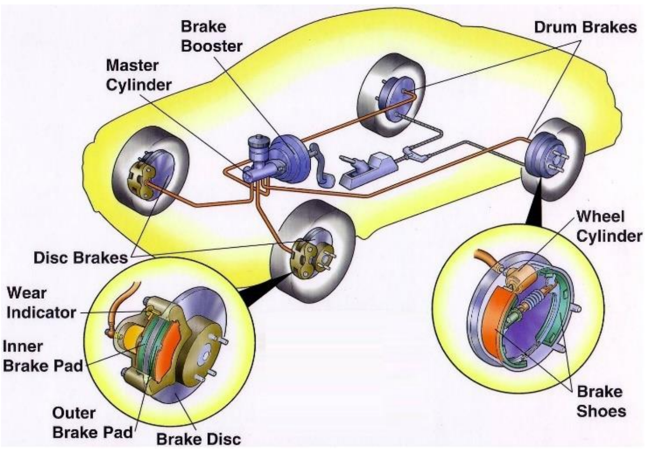


Figure 1 - Automotive braking system [5]

Taking into account those components, we can represent them as a physical model of the system can be seen in figure 2, adapted from [2]:

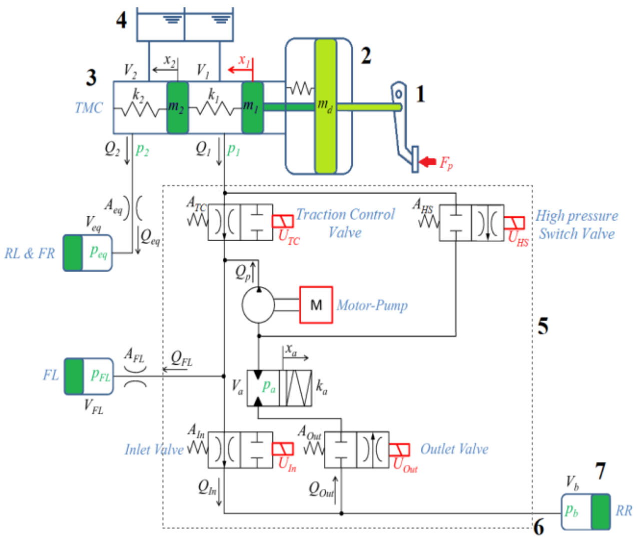


Figure 2 - Physical model of the braking system

Components usually found on braking system are: (1) brake pedal; (2) brake booster; (3) master cylinder; (4) oil tank; (5) ABS / ESC module; (6) throttle, which can be rigid and flexible; (7) wheel brake, which can be brake caliper or drum brake. Following this physical representation, a conceptual non-linear mathematical model was designed to improve the understanding of the braking system.

BRAKE PEDAL - The brake pedal is the interface between the driver and the braking system. It uses the principle of the lever as the first multiplier of the force applied by the driver on the pedal. We represent the force of the driver by applying a signal with a constant steady-state gain (K) and a time constant τ_p. Regardless of mechanical delays that may occur and the driver's reaction delay, we can define brake pedal force using the following equation [1]:

$$F_p = \frac{K}{1 + \tau_p s} \qquad (1)$$

The representation of the pedal is seen in Figure 3:

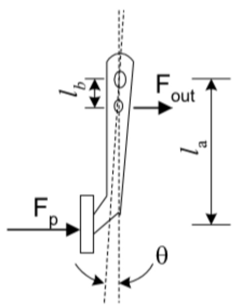


Figure 3 - Brake pedal free body diagram [1]

Described by the following equation [1]:

$$\frac{F_{out}}{F_p} = \frac{l_a}{l_b} = g_p \quad (2)$$

Where: F_p is the brake pedal force, $F_{\_out}$ is the output force multiplied by the gain g_p.

BRAKE BOOSTER - The brake booster is the second force multiplier of the braking system, it is a component divided into two chambers by a diaphragm in which the first chamber has atmospheric pressure and in the second chamber it has a vacuum from the engine intake manifold, and it is precisely this pressure difference between the chambers that is responsible for the multiplication of the input force, which can be described by the following equation [1]:

$$F_m = \frac{K_a}{1+\tau_a s} F_{out} \quad (3)$$

Where: K_a is the stationary gain, τ_a is the time constant and F_m is the force transferred by the brake booster to the master cylinder.

MASTER CYLINDER - The function of the master cylinder is to convert the brake force applied by the driver to the brake pedal and convert it into proportional hydraulic pressure [9]. The hydraulic fluid ducts are divided into two circuits, and the master cylinder, in turn, has two pressure chambers, each responsible for feeding a hydraulic fluid circuit. The physical representation of the master cylinder can be seen in Figure 4 (adapted from [2]):

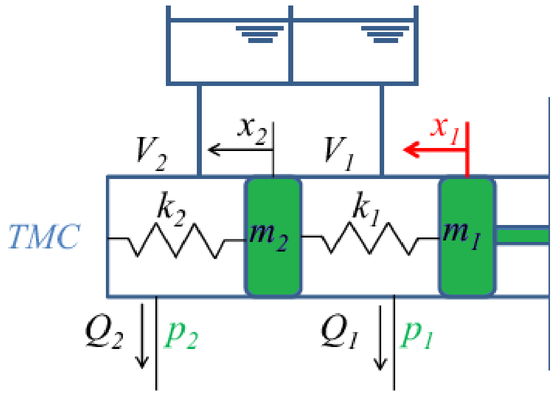


Figure 4 - Representation of the physical model of the master cylinder (adapted from [2])

The master cylinder has axial dynamics described as a system with two degrees of freedom [5], but since x_1 is the displacement of the primary piston, which is directly connected to the pressure rod of the brake booster, represent an input, so the system becomes only a degree of freedom. With a single equation of states described as [2]:

$$\begin{aligned} S(P_1 - P_2) - F_{f2} sng\dot{x}_2 + b_1(\dot{x}_1 - \dot{x}_2) \\ + k_1(x_1 - x_2) - b_2\dot{x}_2 - k_2x_2 \\ = m_2\ddot{x}_2 \end{aligned} \quad (4)$$

Where: S is the hydraulic cylinder surface, k_i is the stiffness of the i_{th} chamber spring of the cylinder, b_i is the oil viscous damping coefficient of the i_{th} chamber, m_i is the mass of the cylinder of the i_{th} chamber, P_i is the pressure inside the i_{th} chamber and F_{f2} is Coulomb's friction coefficient. The volume of the two chambers of the master cylinder varies according to the performance of the brake pedal, and can be described using the following equations [2]:

$$V_1 = V_{10} + S(x_2 - x_1) \quad (5)$$

$$V_2 = V_{20} - Sx_2 \quad (6)$$

Where: V_{10} and V_{20} are the initial volumes of the first and second chamber respectively.

ABS MODULE - The ABS module avoids the locking of the car wheels using a magnetic wheel sensor, a hydraulic pressure modulator with a pair of valves for each wheel, and an electronic control unit. So far, many vehicles now have electronic stability control. In these cases, the hydraulic modulator has four more valves to perform the stability control, and for the strategy to be effective, it is necessary to add other sensors and actuators, such as steering angle sensor, yaw sensor, pressure sensor, among others. The ABS / ESC module is responsible for controlling the valve set of the hydraulic pressure modulator. It can be seen in its physical representation in Figure 5:

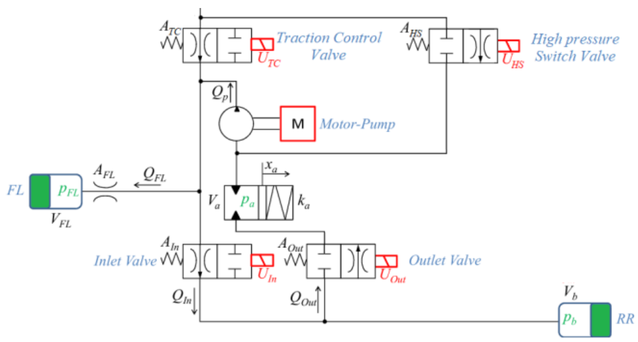


Figure 5 - Representation of the physical model of the ABS / ESC hydraulic module

The hydraulic fluid transfers the pressure from the master cylinder to the brake caliper, and this pressure is transferred through the flow rate Q_i, which is the volumetric capacity of fluid that passes through a pipe in a unit of time considered and is described like [1] [2] [3] [4]:

$$Q_{in} = C_{qin} A_{in} \sqrt{\frac{2|p_1 - p_b|}{\rho}} sgn(p_1 - p_b) \quad (7)$$

$$Q_{out} = C_{qout} A_{out} \sqrt{\frac{2|p_b - p_a|}{\rho}} sgn(p_b - p_a) \quad (8)$$

$$Q_{FL} = C_{qFL} A_{FL} \sqrt{\frac{2|p_1 - p_{FL}|}{\rho}} sgn(p_1 - p_{FL}) \quad (9)$$

$$Q_{eq} = Q_2 = C_{qeq} A_{eq} \sqrt{\frac{2|p_2 - p_{eq}|}{\rho}} sgn(p_2 - p_{eq}) \quad (10)$$

$$Q_1 = Q_{in} + Q_{FL} - Q_p \quad (11)$$

Where: Q_i is the flow rate of the i_{th} element, A_i is the area of the i_{th} element, C_{qi} is the flow coefficient and depends on the pressure drop Δp by the hydraulic resistance, and can be calculated by [2]:

$$C_{qi} = C_{qmax} \tanh\left(2 \frac{\lambda}{\lambda_{cr}}\right) \quad (12)$$

$$\lambda = \frac{h_d}{v} \sqrt{\frac{2|\Delta p|}{\rho}} \quad (13)$$

Where: h_d is the hydraulic diameter, v is the kinetic viscosity, ρ is the oil density, C_{qmax} is the maximum value of the flow coefficient, λ_{cr} is the critical flow number at which the laminar transition occurs turbulent flows. The flow rate generated by the electric pump (Q_p) which is influenced by the accumulator pressure (p_a). Pump flow is described by the following equation [2]:

$$Q_p = Q_{ss}[1 - e^{-3p_a/p_{th}}] \quad (14)$$

Where: Q_{ss} is the steady-state value of the flow rate provided by the electric pump and p_{th} is the pressure threshold at which the flow rate drops to zero. The hydraulic accumulator also has an axial dynamics, which can be written using Newton's second law [1] [2]:

$$S_a P_a - F_{fa} sng \dot{x}_a - b_a \dot{x}_a - k_a x_a = m_a \ddot{x}_a \quad (15)$$

Where: S_a is the accumulator cylinder surface, P_a is the pressure in the accumulator, F_{fa} is the Coulomb coefficient, b_a is the viscous damping coefficient, k_a is the spring stiffness and m_a is the mass of the cylinder. The hydraulic pressure gradient in the braking system is described by [1] [2]:

$$\frac{dp_i}{dt} = \frac{\beta_i}{V_i}(\pm Q_i - \dot{V}) \quad (16)$$

Where: β is the bulk modulus of hydraulic brake fluid that can be described as following [2]:

$$\beta_i = \beta_n \frac{1 + \alpha\left(\frac{p_{atm}}{p_{atm} + p_i}\right)^{\frac{1}{n}}}{1 + \alpha \frac{(p_{atm})^{\frac{1}{n}}}{n(p_{atm} + p_i)^{\frac{n+1}{n}}} \beta_n} \quad (17)$$

Therefore, the hydraulic pressure in the accumulator is [1] [2]:

$$\frac{dp_a}{dt} = \frac{\beta_a}{V_a}(Q_{out} - Q_p - \dot{V}_a) \quad (18)$$

The same equation can be rewritten in a simplified form such as [1]:

$$\frac{dp_a}{dt} = \frac{\beta_a \dot{V}_a}{V_a} \quad (19)$$

Some geometric characteristics such as volume and area are defined by [2]:

$$V_{FL} = q_{FL} V_b \quad (20)$$

$$V_{eq} = V_{FL} + V_b \quad (21)$$

$$A_{FL} = A_{in} \quad (22)$$

$$A_{eq} = 2A_{in} \quad (23)$$

Where: q_{FL} is the coefficient that represents the proportional relationship between the volume of the front and rear hydraulic cylinders.

BRAKE CALIPER - The brake caliper is the component used in the disc brake, with an application of radial force on the brake disc connected to the vehicle wheel, the representation of the brake caliper can be seen in Figure 6:

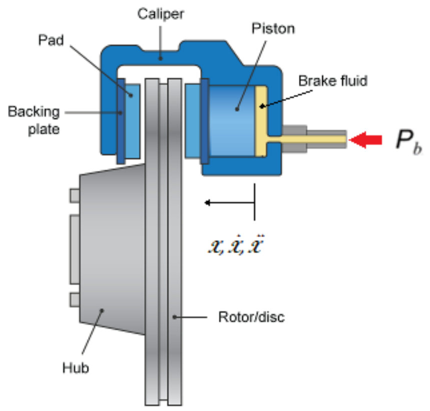


Figure 6 - Diagram of the brake caliper [6]

Resulting in an equation [1]:

$$S_b P_b - b_b \dot{x}_b - k_b x_b = m_b \ddot{x}_b \tag{24}$$

Where: S_b is the brake caliper piston surface, P_b is the pressure in the brake caliper, b_b is the damping coefficient and k_b is the effective spring constant of the brake caliper. And to get the pressure on the brake caliper that is our focus, we have the pressure gradient equation on the brake caliper, which can be described by [1] [2]:

$$\frac{dp_b}{dt} = \frac{\beta_b}{V_b}\left(Q_{in} - Q_{out} - \dot{V}_b\right) \tag{25}$$

Which can also be described in a simplified way [1]:

$$\frac{dp_b}{dt} = \frac{\beta_b \dot{V}_b}{V_b} \tag{26}$$

And obtaining the pressure on the brake caliper, we then find the brake torque on the brake caliper, described by [1]:

$$\frac{T_b(s)}{P_b(s)} = \frac{G_b}{1 + (2\zeta/\omega_n)s + (1/\omega_n{}^2)s^2} \tag{27}$$

Where: T_b is the Laplace transform of the braking torque, P_b is the Laplace transform of the pressure in the brake caliper, G_b is the gain of the braking torque, ζ is the damping ratio and ω_n is the natural frequency.

DRUM BRAKE - The drum brake is commonly used in Brazilian vehicles on the rear wheels, and is a system in which the brake shoe generates a radial force in the drum connected to the vehicle wheel. Figure 7 shows the forces acting in a drum brake:

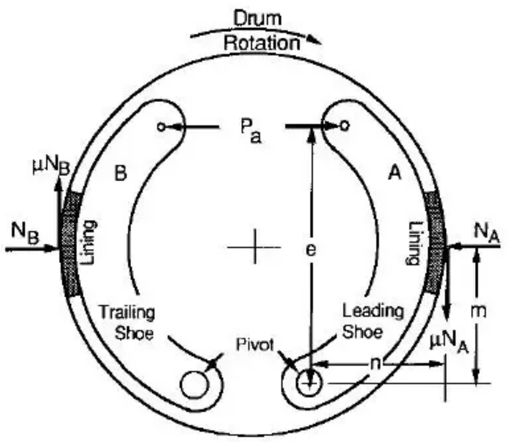


Figure 7 - Forces acting on the drum brake [7]

Where: e is the perpendicular distance between the application force (P_a) to the pivot; m is the perpendicular distance between the normal force (N_a or N_b) to the pivot; n is the perpendicular distance between the frictional force (μN_a or μN_b) to the pivot; N_a is the normal force exerted by the brake drum on the brake liner A; N_b is the normal force exerted by the brake drum on the brake liner B; P_a is the actuation force exerted by the wheel cylinder on the brake shoe (pressure P_b multiplied by the piston area of the cylinder); μ is the friction coefficient between the brake liner and the brake. Defining the frictional forces and the sum of the moments in the pivot for brake shoes A and B it is possible to arrive at the following equations [7]:

$$\frac{F_a}{P_a} = \frac{\mu e}{m + \mu n} \tag{28}$$

$$\frac{F_b}{P_b} = \frac{\mu e}{m + \mu n} \tag{29}$$

This relationship between application force and braking force is what is called the brake factor (C*), resulting in the sum of the components of the two brake shoes [7]:

$$C^* = \frac{F_a}{P_a} + \frac{F_b}{P_b} = \frac{\mu e}{m + \mu n} + \frac{\mu e}{m + \mu n} \tag{30}$$

And how we can perceive the braking torque of the disc brake depends on geometric factors.

EXPERIMENTS AND RESULTS

The model described in the previous section [1] [2] was implemented in a Matlab/Simulink environment, in which an input step from equation (1) was applied, for an analysis of how the system would behave. The strategy used by the vehicle braking system to activate the ABS strategy depends on the reading value of the wheel speed sensor. In the proposed conceptual dynamic, there is no connection between wheel speeds and the aforementioned dynamic model far.

This interface is made by the electronic control unit, therefore, for the switching of the valves, a state machine activates several cases implemented in Simulink with pre-defined time intervals. Each stage that is activated by the state machine represents a phase of the ABS strategy, which opens and closes the inlet and outlet valves of the hydraulic pressure controller to check the pressure profile on the brake caliper.

Table 1 highlights the input data used for experiments. The Conceptual Dynamic Model for Electronic Braking System is shown in Figures 8 and 9. Figure 8 shows a top view from the proposed model whilst Figure 9 shows a bottom view including (1) brake actuation pedal, (2) brake booster, (3) master cylinder, (4) hydraulic unit and (5) brake caliper.

Par	Value	Par	Value
K	$2N$	$C_{q,max}$	0,7
τ_p	$0,5s$	v	$10e^{-6}kg/(ms)$
g_p	4	λ_{cr}	100
K_a	$8N$	Q_{ss}	$0,26l/min$
τ_a	$1,5s$	p_{th}	$0,6bar$
S	$5,07cm^2$	S_a	$2,54cm^2$
$b_1 = b_2$	$100Nms/rad$	b_a	$85Nms/rad$
k_1	$2222,2N/m$	k_a	$35N/m$
k_2	$4000N/m$	m_a	$10g$
F_{f2}	$14N$	β_n	$27000bar$
m_2	$40g$	α	0,02
V_{10}	$18,8cm^3$	n	1,4
V_{20}	$12,5cm^3$	S_b	$45cm^2$
ρ	$1070kg/m^3$	b_b	$100Nms/rad$
A_{in}	$0,29mm^2$	k_b	$2000N/m$
A_{out}	$0,59mm^2$	m_b	$100g$

Table 1: values of parameters used in the experiment.

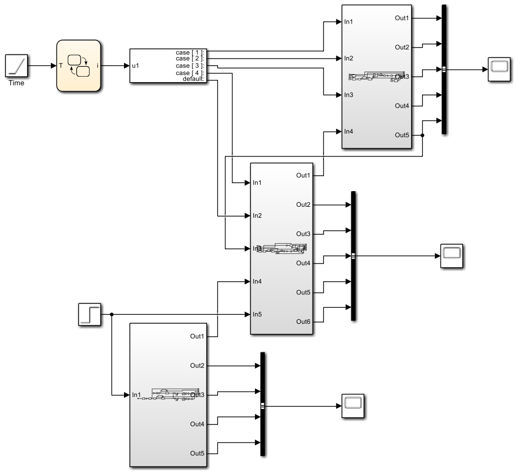
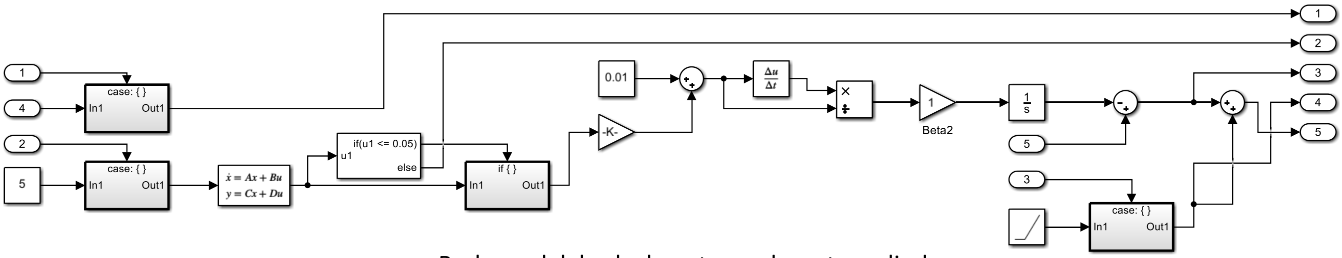
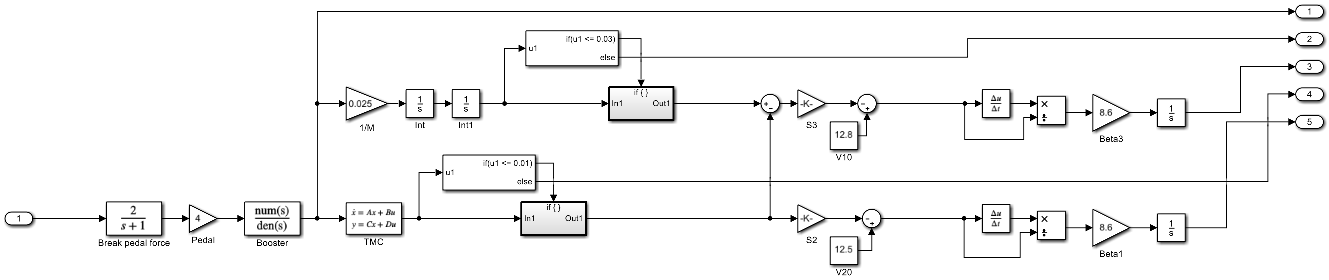


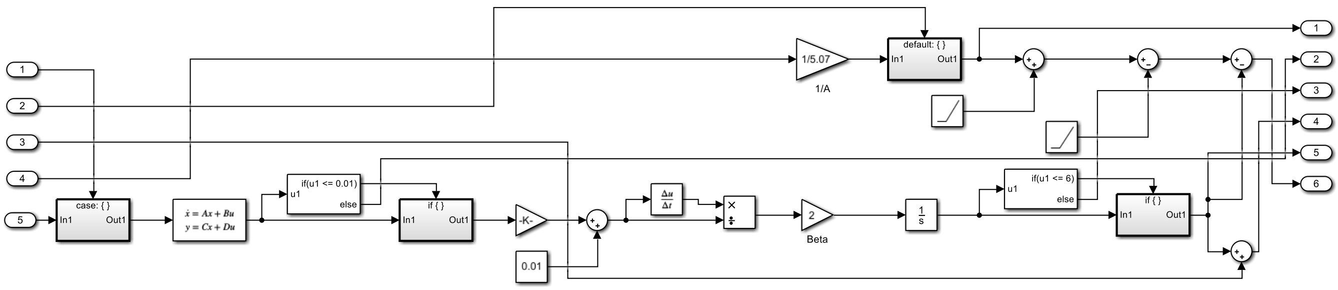
Figure 8 – Dynamic Model for Electronic Braking System



Brake pedal, brake booster and master cylinder



Hydraulic unit



Brake capiler

Figure 9 – Dynamic Model for Electronic Braking System (bottom view)

It is possible to verify a gain of 32.5 times the application of force in the brake pedal, by the first force multipliers, pedal and brake booster in Figure 10:

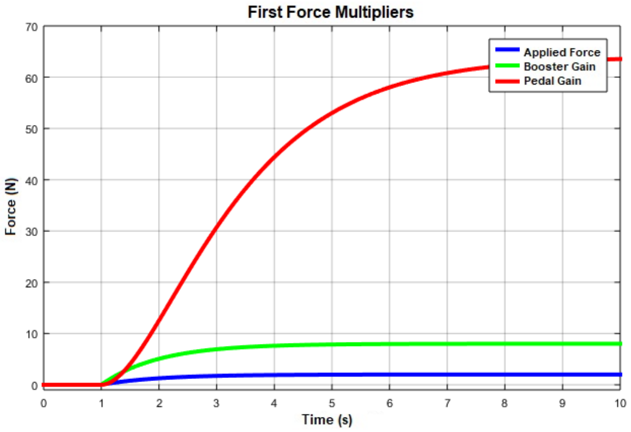


Figure 10 - Input force and mechanical gains

The strength gain of the booster is the force value that reaches the master cylinder and then starts to generate the braking pressure. With that, the four ABS stages were simulated as previously described, and the result obtained can be seen in Figure 11:

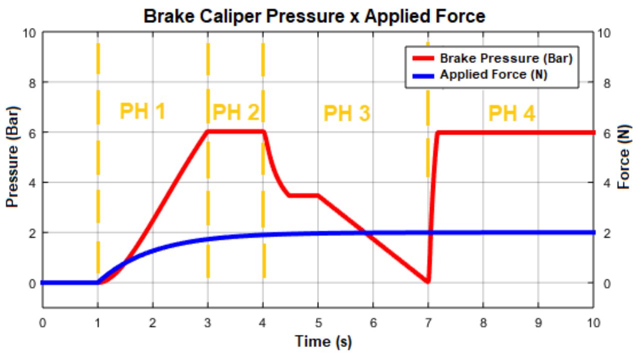


Figure 11 - Pressure profile on the brake caliper after ABS steps x Applied Force

PHASE 1 (PH 1) - In this stage, we have the nominal braking, following the Pascal principle for pressure in a hydrostatic system. Here the valves are in their initial positions allowing the fluid to flow freely over the ducts. Up to the 3 seconds (see Figure 12), it is simulating the pressure on the brake caliper according to the driver's performance.

PHASE 2 (PH 2) - It is the pressure retention stage, in which the ABS detects the imminence of a wheel lock and closes the inlet valve referring to that wheel, keeping the pressure constant. This step occurs due to the speed of the wheel, but as in the physical model of the brake we have no way to detect this phenomenon, all steps were done through predetermined time intervals. Then we have the brake caliper pressure at this stage:

Between the 3 and 4 seconds of Figure 11, the pressure in the brake caliper was kept constant, simulating the pressure locking step on the wheel.

PHASE 3 (PH 3) - This is the phase of emptying the brake caliper to relieve brake pressure, causing the wheel to turn again. The system opens the outlet valve and starts the electric motor to start the pump so that the oil is thrown back into the circuit. In the simulation seen in Figure 11, this stage was divided into 2 parts, with an interval of 4 to 5 seconds, only the outlet valve was opened, so that the accumulator absorbs the pressure of the hydraulic fluid that was in the brake caliper, and between moments 5 and 7 seconds, the electric motor was started so that the hydraulic pump removes the brake fluid that was in the brake caliper.

PHASE 4 (PH 4) - Finally, this is the phase of filling the brake caliper again, in which the valves return to their initial state and the brake caliper is filled with hydraulic fluid again as shown in Figure 11. As we can see, in this stage the brake caliper filling is much more abrupt than at the beginning of the simulation, because in that instant of 7 seconds when the inlet valve is opened again, the system is already at its maximum pressure, whereas in the instant initial simulation the pressure was gradual.

To verify the efficiency of this braking system, a longitudinal model built by [8] will be used, in which the vehicle's acceleration and gear values are chosen, and the model returns the vehicle's speed. The pressure profile shown in this article was included in the test at key points, and the result compared to validate the conceptual braking model presented in this article. As we can see in figure 12.

The first graph shows the vehicle speed in the simulation with the braking system acting (red) and the vehicle speed without the braking system acting (blue).

While the second graph shows three braking situations. In which the first situation occurs for a time of 30 seconds with the total braking capacity used, the second situation occurs for a time of 20 seconds with 50% of the braking capacity, and in the third situation we have a performance of 20 seconds with total braking capacity.

It is possible to verify that in the three situations the brake system is able to reduce the vehicle speed to zero, in the first case it reduces from 140km/h to 0km/h and in the second and third it reduces it around 40km/h until it stops completely vehicle.

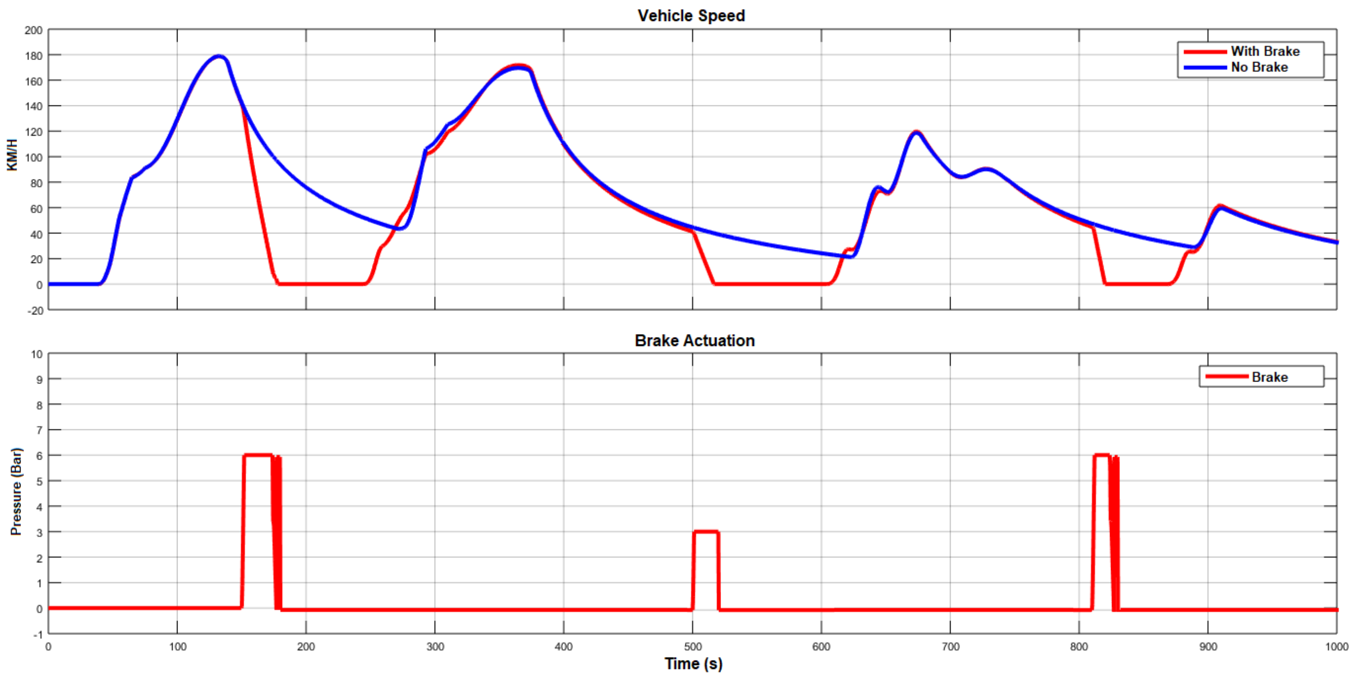


Figure 12 - Comparative test with and without braking action.

Then, we analyze only the first braking situation and the comparison of the speed curves. Being able to identify in the braking graph the four ABS steps explained previously along with the vehicle's speed drop:

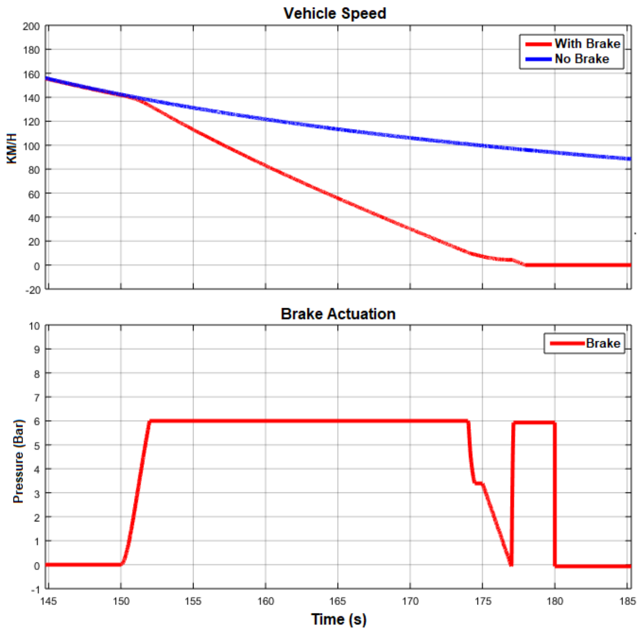


Figure 13 - Speed comparison during braking.

Analyzing the result obtained in figure 13, it is possible to verify a great similarity with what happens with the pressure on the brake caliper in a real situation of vehicle braking, as seen in the Figure 14.

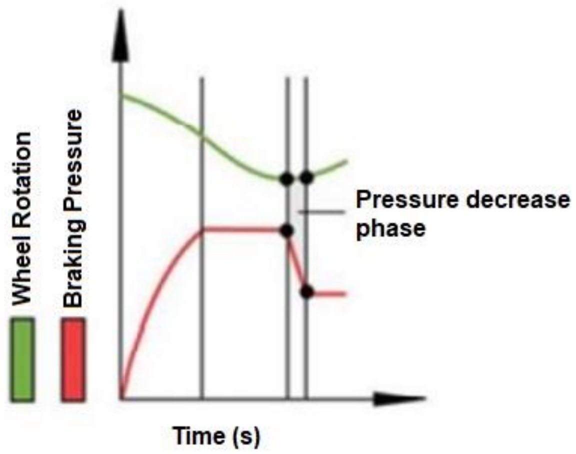


Figure 14 - ABS performance in pressure reduction [9]

In this test, the rotation of the wheel was monitored, so that the ABS can enter into action, showing the pressure in the brake caliper in phases 1, 2, and 3 of ABS operation, which has the same characteristic profile of the experiment carried out in this work.

CONCLUSIONS

The objective of this work was to present a conceptual nonlinear Dynamic Model of an automotive brake system and to simulate it in a computational environment. In this it was possible to verify the model of the brake system in computational experiments acting on a simplified longitudinal dynamic model of a vehicle, where it was possible to verify the speed drop when the system was activated.

The results achieved will be important for the design and assembly of an electronic braking system that will be applied to a platform vehicle for the development of ADAS functions, which will be implemented from a Fiat Strada model vehicle. Initially, a platform will be built consisting of the original braking system and an ABS / ESC module in which a "black box" model of the system will be obtained. This will allow, together with vehicle dynamics software, to develop braking control at the simulation level. Subsequently, the brake caliper pressure will be obtained from the physical brake platform and transported to a LABCAR platform where the same vehicle dynamics software will be installed, responding in real time, thus constituting Hardware in the Loop (HiL). This HiL will allow a first calibration and validation of the control unit. Finally, the system will be transported to the vehicle, for a final calibration and validation phase.

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