

# Pressure Following Strategy for Conventional Braking Control Applied to a HIL Test Bench

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## ABSTRACT

Brake systems represent important components for passenger cars since they are strictly related to vehicle safety: Anti-lock Braking Systems (ABS) and Electronic Stability Control (ESC) are the most well-known examples. The paper is focused on the characterization of the braking hydraulic plant and on the design of a pressure following control strategy. This strategy is aimed at pursuing performances and/or comfort objectives beyond the typical safety task. The low-level logic (focus of the paper) consists of a Feedforward and Proportional Integral controller. A Hardware In the Loop (HIL) braking test bench is adopted for pressure controller validation by providing some realistic reference pressure histories evaluated by a high-level controller. Results prove that innovative control strategies can be applied to conventional braking systems for achieving targets not limited to braking issues, i.e., comfort or NVH tasks.

**CITATION:** Galvagno, E., Tota, A., Vigliani, A., and Velardocchia, M., "Pressure Following Strategy for Conventional Braking Control Applied to a HIL Test Bench," *SAE Int. J. Passeng. Cars - Mech. Syst.* 10(3):2017, doi:10.4271/2017-01-2496.

## INTRODUCTION

In the 1950s J.E. Francis [1] introduced Anti-lock Braking Systems (ABS) hydraulic components in the automotive world with the aim of "Slip Prevention Regulator for Rail Vehicles". ABS control logic intends to control wheel slip ratio in order to reduce the stopping distance and providing lateral stability and vehicle directional control by preventing wheels locking effect. Several control strategies have been designed and proposed by different authors in order to take into account non-linear unknown parameters [2]-[3] or to exploit new information coming from innovative sensors [4]-[5].

Recently, the advent of electric vehicles and autonomous driving renew the way of conceiving braking systems, well beyond the safety purposes achieved with ABS and Electronic Stability Control (ESC) logic. Obstacle Collision Avoidance systems (OCA), i.e. for autonomous driving applications, requires the intervention of braking system for enhancing the overall vehicle safety. In these specific scenarios, the braking system is required to provide a desired brake torque for avoiding the obstacle collision thus moving the target from slip ratio control in ABS/ESC to brake pressure regulation. Authors in [6]-[8] use on-off braking logic meanwhile in [9] analogic solenoid-valves replace conventional digital ABS/ESC valves. Braking pressure control is also adopted in electro-hydraulic braking (EHB) applications for regenerative braking system in hybrid and electric vehicles: [10] proposes the use of linear solenoid valves that are the best hardware solution for brake pressure control but not economically advantageous for conventional passenger vehicles. Furthermore, authors in [11] show an innovative exploitation of braking system where a pressure following control is designed for reducing noises and vibrations in vehicle transmissions.

All previous cases represent only few examples of exploring new applications for vehicles braking system. Another important focus is how to exploit the conventional hydraulic hardware available on passenger vehicles for implementing a pressure following controller. Several paper have presented different models for simulating the hydraulic dynamics of a conventional braking system composed by digital On-Off valves commonly used for ABS/ESC control strategies. A detailed electro-mechanical model of on/off valves used to regulate brake pressure is presented in [12] and [13] and pulse with modulation (PWM) signals are usually selected to drive them [14]-[15]. Furthermore, due to the highly non-linear dynamics of hydraulic circuit, an experimental characterization of ABS/ESC digital valves is carried out by [16] through a Hardware-In-the-Loop (HIL) test rig and same authors proves in [17] that a PWM signal with constant frequency and variable Duty Cycle represent a feasible solution for a continuous brake pressure tracking controller.

The present paper proposes a methodology to experimentally design a pressure following controller by considering a conventional ABS/ESC electro-hydraulic hardware through PWM signals. The controller is composed by a FeedForward (FF) contribution and by a Proportional Integral (PI) term. The FF contribution is evaluated by inverting experimental maps that are characteristic of brake pressure dynamics meanwhile the PI gains are designed for compensating FF uncertainties and sensors noises. A HIL test bench is described and it is used for pressure dynamics characterization and for validating the PI+FF controller in tracking different reference pressure trends.

The structure of the paper is divided into seven sections including the present introduction: the “Hydraulic circuit analysis” section describes a typical automotive hydraulic braking system also providing the most important equations about braking pressure dynamics. In the third section, the HIL is introduced and an experimental characterization of brake pressure dynamics is presented. The PI+FF controller is designed and implemented in the fourth section. Some simulation results of pressure following controller are shown in fifth section meanwhile the controller experimental validation by using the HIL is analyzed in the sixth section. Finally, some conclusions are drawn in the last section.

## HYDRAULIC CIRCUIT ANALYSIS

Conventional braking systems consists of hydraulic and electro-mechanical components. Figure 1 shows a schematic view of the hydraulic circuit between the Tandem Master Cylinder (TMC) and brake calipers. TMC input rod is pushed by the driver brake pedal through a brake booster (not represented in figure) thus increasing pressures  $p_1$  and  $p_2$  inside its two chambers. ABS electro valves are introduced to control brake calipers pressure during emergency braking maneuvers. Inlet valves (normally open) let oil flow to increase pressure in the calipers, while Outlet valves (normally close) are activated together with a motor-pump (MP) for reducing pressure. The presence of a spring accumulator (SA) is necessary for providing a reference pressure value to the MP suction port.

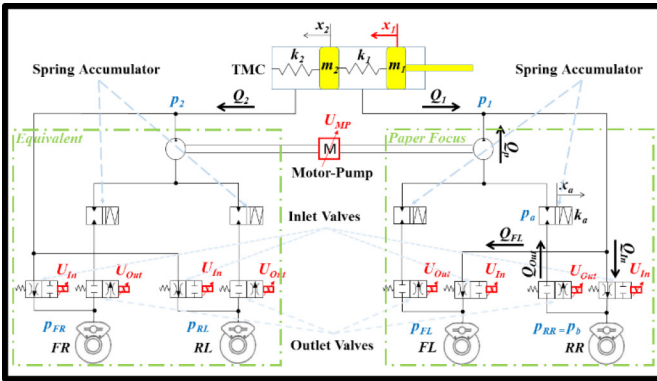


Figure 1. General scheme for the hydraulic circuit between TMC rod and brake caliper chambers.

The paper focus is the experimental characterization of the pressure dynamics inside a brake caliper when a PWM digital signal is applied to Inlet or Outlet valves; assumptions are that the TMC rod is pushed until  $p_1$  and  $p_2$  reach a desired pressure level and that the Inlet and Outlet valves are alternatively commanded for respectively increasing or decreasing braking pressure. MP is activating only when Outlet valves receive a command.

In the following description, only one hydraulic branch is considered without any loss of generality since the same procedure can be replicated for other calipers. By considering the Rear Right (RR) caliper, two flow rates may affect the pressure dynamics  $p_b$ :

- $Q_{In}$  : Ingoing flow rate regulated by Inlet valve regulation

- $Q_{Out}$  : Outgoing flow rate regulated by Outlet valve and MP intervention

Inlet and Outlet valves represent hydraulic resistances and their corresponding flowrates are expressed by Equations (1) and (2):

$$Q_{In} = c_q A_{In} \sqrt{\frac{2|p_1 - p_b|}{\rho}} \text{sign}(p_1 - p_b) \quad (1)$$

$$Q_{Out} = c_q A_{Out} \sqrt{\frac{2|p_b - p_a|}{\rho}} \text{sign}(p_b - p_a) \quad (2)$$

where  $c_q$  is the flow coefficient,  $A_{In}$  and  $A_{Out}$  are the corresponding Inlet and Outlet valves flow area,  $p_b$  the RR braking pressure,  $p_1$  the TMC pressure,  $p_a$  the spring accumulator pressure and  $\rho$  the oil density Effective flow areas  $A_{In}$  and  $A_{Out}$  are controlled by applying a digital command  $U_{In}$  or  $U_{Out}$  to each valve.

Each brake caliper behaves as a hydraulic chamber, so that  $p_b$  dynamics is described by:

$$\dot{p}_b = \frac{\beta}{V} (Q_{In} - Q_{Out}) \quad (3)$$

where  $\beta$  is the oil bulk modulus and  $V$  the brake cylinder volume.

In the present activity, a PWM signal with constant frequency and variable Duty Cycle (DC) is selected to command the valves: this solution allows to have a continuous control of flow areas and so a direct regulation of ingoing and outgoing flowrates:

$$Q_{In} = f_{In}(U_{In}, \Delta_{In}) \quad (4)$$

$$Q_{Out} = f_{Out}(U_{Out}, \Delta_{Out}) \quad (5)$$

where  $\Delta_{In} = p_1 - p_b$  is the pressure drop across the Inlet valve and  $\Delta_{Out} = p_b - p_a$  is that across the Outlet valve which is approximated to  $\Delta_{Out} \approx p_b$ , as mentioned by [16]. Since  $f_{In}$  and  $f_{Out}$  are non-linear functions and some parameters (i.e.  $c_q$ ,  $\rho$ ,  $\beta$ ) are not constant properties and are not easy to evaluate, the effect of PWM Duty Cycle on  $p_b$  is experimentally characterized as explained in next section.

## EXPERIMENTAL CHARACTERIZATION OF BRAKE PRESSURE DYNAMICS

In order to evaluate the non-linear functions  $f_{In}$  and  $f_{Out}$ , experimental tests are carried out on the HIL braking test bench shown in Figure 2.

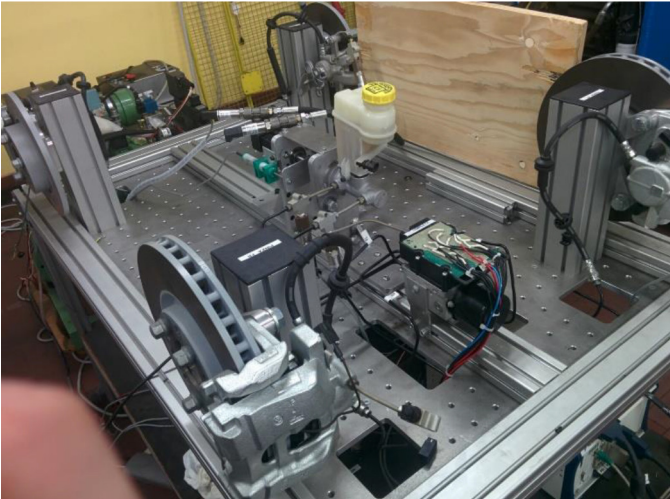


Figure 2. HIL Test Bench where are visible the four brake calipers, the TMC and the customized ABS/ESC unit.

The test bench is composed of:

- a conventional braking hydraulic circuit from the TMC rod to the brake calipers
- a Hydraulic Power Unit (HPU) mounted for providing the desired pushing force on the TMC rod through the brake pedal cylinder
- a Customized Electronic Board (CEB) for providing the desired command to ABS/ESC electro valves and MP
- a Solid-state Relay Box (SRB) for converting the command signal (0-5 V) into a power signal (0-12 V) for each electro valves and MP
- eight pressure sensors: two for brake pedal cylinder chambers, two for TMC chambers ( $p_1$ ,  $p_2$ ) and four for brake calipers pressure ( $p_{FR}$ ,  $p_{FL}$ ,  $p_{RR}$ ,  $p_{RL}$ )
- one potentiometer for TMC rod position  $x_1$
- a National Instruments (NI) PXI for real-time PWM signals generation and sensors data acquisition.

The NI Labview® software is used for experimental characterization process meanwhile NI Veristand® is selected for control purposes.

The controller strategy design requires a constant frequency PWM signal with a variable DC calculated for tracking a reference braking pressure. Specific tests are executed on the HIL test bench to obtain a complete description of pressure dynamics: the test consists in activating the brake pedal cylinder until a desired pressure level is reached in TMC chambers and inside brake calipers (Inlet and Outlet valves are kept in their normal configurations). When the steady-state condition is achieved, the Inlet valve of RR caliper is closed and a PWM signal (with a specific DC) is applied to its Outlet valve together with the activation of MP for emptying the caliper chamber. When braking pressure  $p_b$  falls below a low threshold, the Outlet valve is closed, the MP is switched off and a PWM signal (with a specific DC) is applied to the Inlet valve thus reverting to the initial steady-state condition.

An example of braking pressure response is shown in Figure 3 when a PWM signal of 50 Hz is applied to Inlet valve (DC of 55%) and to Outlet valve (DC of 35%).

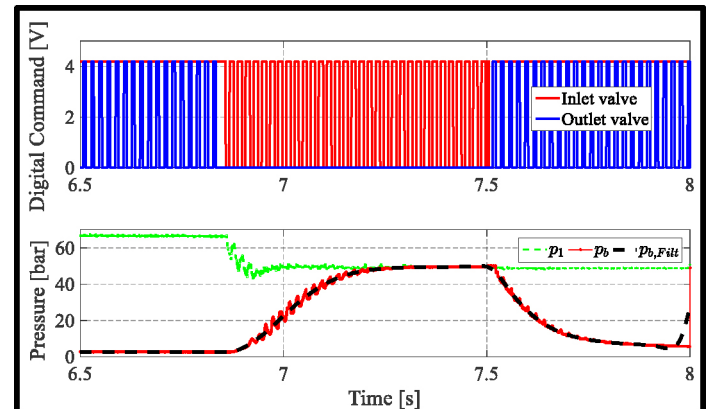


Figure 3. Example of braking response when a PWM signal of 50 Hz is applied to Inlet valve (DC of 55%) and to Outlet valve (DC of 35 %).

The frequency of the PWM signal influences the brake pressure oscillations and a methodology for its selection is described in [17]. A PWM frequency of 900 Hz and 50 Hz are respectively chosen for Inlet and Outlet valves: a different frequency selection is justified by the presence of a SA and a MP in the hydraulic branch where Outlet valve is placed.

Once the PWM frequency is fixed, the same test is repeated for different values of DC for both Inlet and Outlet valves and results are shown in Figure 4: DC directly affects the derivative of brake pressure  $\dot{p}_b$  as stated by Equations (3), (4), (5).

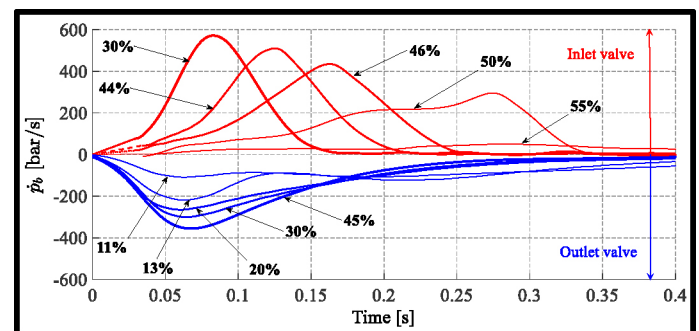


Figure 4. Brake pressure gradient for different values of DC for both Inlet and Outlet valves.

For a better characterization of the brake pressure dynamics, its derivative  $\dot{p}_b$  must be correlated to the pressure drop across each valve thus obtaining the experimental representation of states space shown in the map of Figure 5 where  $\Delta_p = \Delta_{In}$  for Inlet valve and  $\Delta_p = \Delta_{Out}$  for Outlet valve.