Proceedings of the 4th WSEAS International Conference on Fluid Mechanics and Aerodynamics, Elounda, Greece, August 21-23, 2006 (pp275-281)

A Methodology to Investigate the Dynamic Characteristics of ESP Hydraulic Units - Part II: Hardware-In-the-Loop Tests

Aldo Sorniotti
Politecnico di Torino, Department of Mechanics
Corso Duca degli Abruzzi 24
10129 Torino
ITALY

Prof. Nikos E. Mastorakis
WSEAS, Agiou Ioannou Theologou 17-23,
15773, Zografou, Athens,
GREECE
mastor@wseas.org
http://www.wseas.org/mastorakis

Abstract - The paper deals with the Hardware-In-the-Loop based methodology which was adopted to evaluate the dynamic characteristics of Electronic Stability Program (ESP) and Electro-Hydraulic Brake (EHB) system components. Firstly, it permits the identification of the time delays due to the hardware of the actuation system. Secondly, the link between the hardware of the hydraulic unit and a vehicle model running in real time permits the objective evaluation of the performance induced by the single components of different hydraulic units in terms of vehicle dynamics. The second part of this paper suggests the Hardware-In-the-Loop (HIL) tests which can be adopted to evaluate the influence exerted by the hydraulic hardware of the brake system on vehicle dynamics and handling.

Keywords – Vehicle model, Electronic Stability Program, Delays

1. The Connection between the Hardware and the Vehicle Model

This paper describes the procedure for the evaluation of the performance of ESP hydraulic units and their effect on vehicle dynamics. The instrument is Politecnico di Torino HIL braking systems test bench [1], as stated in the first part. It is characterized by the hardware of a whole brake system. The bench hydraulic unit permits the actuation of booster input rod, which can be controlled both in force and displacement. Pressure sensors are located in correspondence of the main components of the brake system. Pressure sensors at the wheels calipers send their signals to the vehicle model [2] which runs in real time on a dSpace® card. On the basis of pressure sensors, brake torques at the wheels are computed and given as an input to the vehicle model. The vehicle model is properly implemented for this HIL application. For example, it considers in detail the equivalent inertia of the engine computed at the wheels, to simulate the correct wheel dynamics during emergency brake maneuvers carried out at different gear ratios. Tire relaxation length variation as a function of slips and vertical load is taken into account. Interaction between lateral and longitudinal forces between tires and ground is considered, through the adoption of Pacejka Magic Formula. During braking with Anti-lock Brake System (ABS), temporary locking phenomena of the wheels can happen. To obtain realistic results, it is necessary to consider the transitions from kinetic to static friction and vice versa between brake pads and discs. When wheels are not locked, brake torque T_{BRAKE} for each wheel is computed as:

$$T_{BRAKE} = (p_l - p_0) \cdot A_{WC} \cdot \eta_C \cdot BF \cdot r \quad (1)$$

where p_l is line pressure (measured at the calipers), p_0 is pushout pressure, A_{WC} is the equivalent area of the wheel cylinder, η_c is the efficiency of the wheel cylinder, r is the equivalent radius of the disc or drum, BF is the brake factor (the constant factor between brake actuation force and brake drag force). BF corresponds to two times the friction coefficient between pads and disc in the case of a disc brake, whereas it depends both on friction and the geometry for a drum brake. When wheels are not locked, BF can be considered either a constant or a function of pressure, sliding speed and temperature, according to the target of the test. When wheels are locked, it is necessary to compute BF so that the wheel remains locked without turning in the opposite direction (in comparison with the direction of the motion before wheel locking). This task can be achieved by computing the brake factor during static friction with the following formula:

$$BF_{static} = \frac{T_m - F_x \cdot R_l}{(p_l - p_0) \cdot A_{WC} \cdot \eta_C \cdot r} \quad (2)$$

where T_m is the torque from the differential, R_l is the loaded radius of the wheel, F_x is the longitudinal force between the tire and the ground. During static friction, the following condition has to be satisfied:

$$BF_{static} \le BF_{static,\max} \quad (3)$$

where $BF_{static,\max}$ is the maximum value that the brake factor can assume in static conditions of friction between the pads and the disc. If (3) is not satisfied, the brake factor is equal to the value corresponding to kinetic friction. The computed value of the brake factor for static and kinetic friction is inserted in the equation of the rotational equilibrium of the wheel:

$$T_m - T_{BRAKE} - F_x R_l - T_{res} = \dot{\omega} I \quad (4)$$

where T_{res} is tires drag torque, $\dot{\omega}$ is wheel rotational acceleration, I is the inertial momentum of the wheel. The vehicle model behaves as a consequence of the experimental pressures measured at the calipers. This activity is devoted to the evaluation of the main parameters of the hardware of the hydraulic unit which can have an influence on ESP performance from the point of view of vehicle dynamics. As a consequence, the work here presented consists in activating the motor pump and the electro-valves of different hydraulic units, by-passing their control algorithms. The adopted electronic hardware for this target is described in [1]. Together with the vehicle model, devoted control algorithms (developed by the author) run in real time and are linked to the tested ESP hydraulic unit. The typical time histories for testing the components of the ESP are automatically implemented by these control algorithms. The behavior of each component of ESP hydraulics can be objectively characterized, first of all independently of the vehicle model, as shown in the first paper about this activity. Then more sophisticated control algorithms are adopted to simulate the behavior of commercial ESP software. In this configuration, the hardware of the brake system is linked to the vehicle model and can be used for the HIL evaluation of the influence of ESP hydraulics on vehicle dynamics. The results which will be presented in the following pages about EHB applied to ABS (Anti-lock Brake System), Traction Control and body yaw rate control are simulation results on the basis of EHB experimental actuation delays measured on an EHB bench [3]. The results about ESP hydraulic units are experimental results of Politecnico di Torino braking systems HIL test bench.

2. The Influence of the Hydraulic Unit Performance on Vehicle Dynamics

In this paragraph an ESP control algorithm is implemented on the brake system HIL test bench. The vehicle model is linked to the hardware of the brake system, as described in the first paragraph. The performance variation due to the adoption of different hydraulic units will be considered. In particular, the performance improvement connected with the adoption of an EHB system over an ESP will be described. The implemented ESP algorithm is based on feedback yaw rate control [1].

THE EFFECT OF ESP HYDRAULIC UNIT ON ABS PERFORMANCE

The actuation algorithm is based on the succession of pressure increase, maintenance and decay phases for ABS, characterized by a 4-channel control algorithm developed by the Vehicle Dynamics Research Team of Politecnico di Torino. Figures 1 and 2 are related to the implementation of the ABS algorithm by adopting an old generation ABS hydraulic unit. Figure 1 plots the time history of Left Rear (LR) and Right Front (RF) caliper pressures during an emergency brake. The two calipers belong to the same hydraulic circuit (the system has a 'X' configuration). Figure 2 plots the signals for valves '1' and '2' (refer to Figure 1 of the first part) during the same maneuver. When the control algorithm requires a contemporary pressure reduction phase for both the calipers, RF caliper pressure decreases slowly whereas LR caliper pressure increases. It is clearly due to a not sufficient flow rate guaranteed by the motor pump unit. This limit of the component can be discovered only through HIL simulation of vehicle dynamics linked to the hydraulics of the brake system. The HIL test bench permitted to carry out some experimental tests by adopting the same hydraulic unit, but using the original control algorithm developed by the supplier of the ABS.

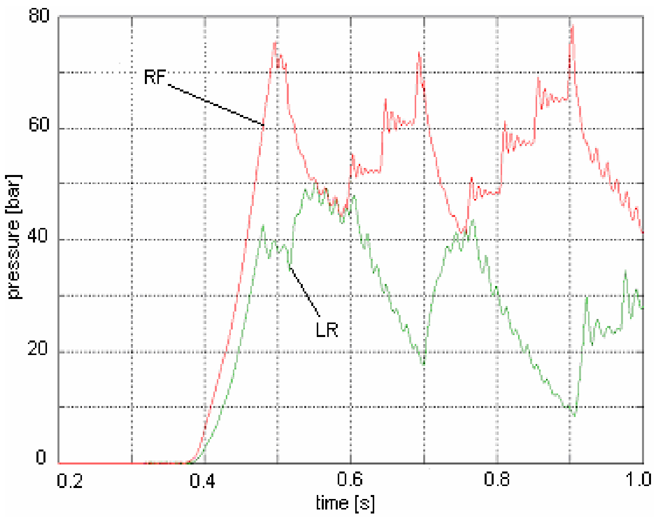

Figure 1 – Example of the problems related to a not sufficient pump volume displacement during ABS modulation

It was experimentally verified that the original control algorithm did not give origin to a contemporary pressure reduction phase for more than one caliper for each hydraulic circuit. A similar form of limitation was imposed also on the control algorithm conceived by Politecnico di Torino and pressure modulation became correct. In any case, the limitation to the contemporary pressure reduction of the two calipers of the same hydraulic circuit is a consistent inconvenience for the efficiency of the ABS system, which has to be taken in account by the car manufacturer and can be easily verified through HIL tests, which at the moment are not a standard between car manufacturers, at the level of the hydraulic components.

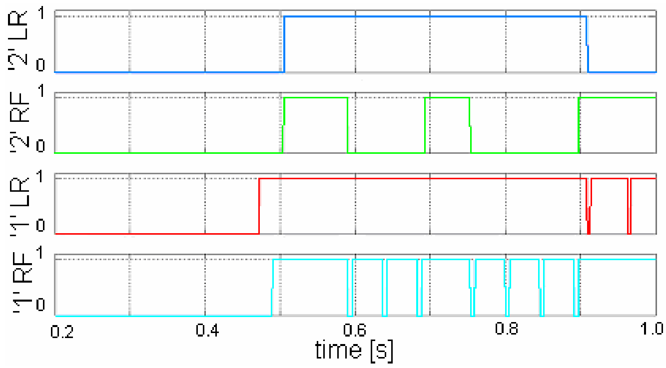

Figure 2 – Modulation of valves '1' and '2' (Figure 1) for Right Front (RF) and Left Rear (LR) wheels during the same brake maneuver of Figure 1

Figure 3 shows the results related to the conceived ABS algorithm, the same of Figure 1, implemented on the hardware of a new generation ESP hydraulic unit. Pressure modulation gives origin to the desired sequence of phases, independently from the number of calipers for which a contemporary pressure reduction phase is requested. The fundamental importance of the properties of the motor pumps of conventional ESP hydraulic units to have a good ABS control is demonstrated.

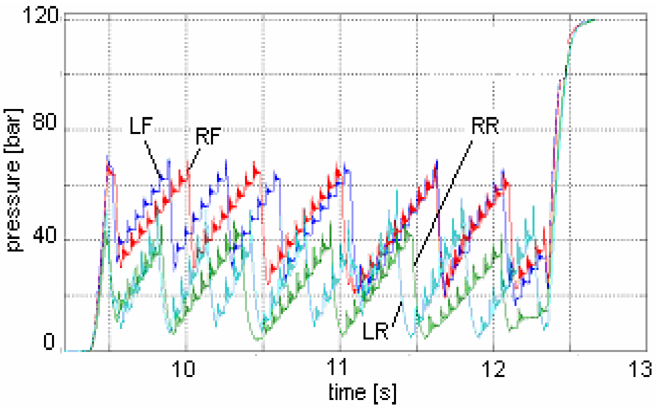

Figure 3 – Example of the implementation of the conceived ABS algorithm on a new generation commercial ESP unit (its pump has a larger flow rate)

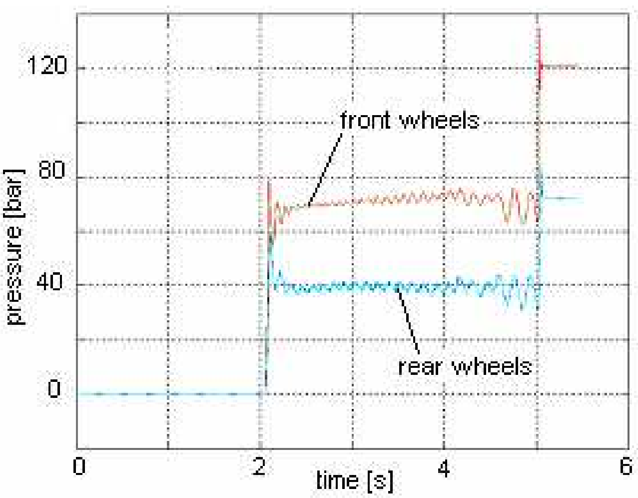

Figure 4 – Example of improvement of the ABS performance related to the adoption of an EHB hydraulic unit [3]

Figure 4 shows the benefits related to the adoption of an EHB hydraulic unit [3]. EHB hydraulic units can be useful to improve ABS performance not only from the hydraulic point of view, but from the point of view of the control algorithm. In fact, pressure sensors (necessary for pressure modulation) at the output ports of EHB hydraulic units can be used for a better estimation of friction coefficient between tires and ground. A locking tendency at a low pressure corresponds to low friction, the opposite for a locking tendency at a high pressure level. Secondly, ABS reference pressure level can be imposed as a continuously varying function of wheels peripheral acceleration and estimated slips, and not only as a sequence of discrete states of pressure reduction, maintenance and increase. Figure 4 was obtained with the same basic ABS control algorithm of Figures 1 and 3, with the mentioned improvements due to EHB implementation. Pressure oscillations entity during the maneuver is consistently reduced, from an average level of more than 20 bar for conventional ESP units, to a maximum level of 10 bar for EHB.

THE EFFECT OF ESP HYDRAULIC UNIT ON THE PERFORMANCE OF TRACTION CONTROL AND BODY YAW RATE CONTROL

This paragraph deals with the effect of the performance of ESP hydraulics on Traction Control (TC) and body yaw rate control.

The implemented actuation algorithm

The case which is focused here is that one of ESP interventions when the driver is not pushing the brake pedal. This case implies the same kind of actuation both for body yaw rate and traction control. In literature, several solutions for ESP actuation are presented, for example based on a feedback control of tires longitudinal slips [4] to generate the desired yaw torque. In any case, an estimation of the forces between the tires and the ground is performed by the ESP control algorithm, on the basis of the estimated pressure generated at the calipers by the hydraulic unit. The actuation algorithm implemented during this activity is capable of generating the desired pressure at the caliper according to an open-loop control algorithm, without using caliper pressure signals. Caliper pressures are not measured by conventional ESP hydraulic units for reasons of cost. This actuation algorithm was adopted for the comparison between the performance of different commercial hydraulic units. A simplified version of this control algorithm was presented in [1]. During the pressure increase phase, a continuous estimation of the actual pressure level p is performed, on the basis of a table, having as inputs two variables, $p_{auxiliary}$ and $t_{activation}$.

$$p = f(p_{auxiliary}, t_{activation}) \tag{5}$$

$t_{activation}$ is computed by a counter which starts at the instant in which the motor pump is activated and stops when the pump is switched off. During pressure maintenance and pressure increase phases, $p_{auxiliary}$ is equal to the value of the estimated pressure p at the end of the last activation of the motor pump:

$$p_{auxiliary} = p_{end\_activation} \tag{6}$$

During pressure reduction phases, it is:

$$p_{auxiliary} = p_{reference} \tag{7}$$

In such a way, a first approximation estimation of caliper pressure during ESP actuation is performed. ESP intervenes on the two calipers of the same side for yaw rate control and on one or two calipers of the same axle for TC. The considered vehicle is equipped with a ‘X’ configuration of the brake system. As a consequence, a contemporary actuation of more than one caliper of the same hydraulic circuit cannot happen. The minimum duration of motor pump intervention is imposed on the basis of a table (reported in Figure 5) defined according to experimental tests like those summarized in the first paper about this activity.

$$t_{\min} = f_1(p_{auxiliary}) \tag{8}$$

During pressure decay, ‘2’ valves (look at Figure 1 of the first part) are subjected to PWM modulation, as described in [1], on the basis of actual pressure level and the desired pressure gradient.

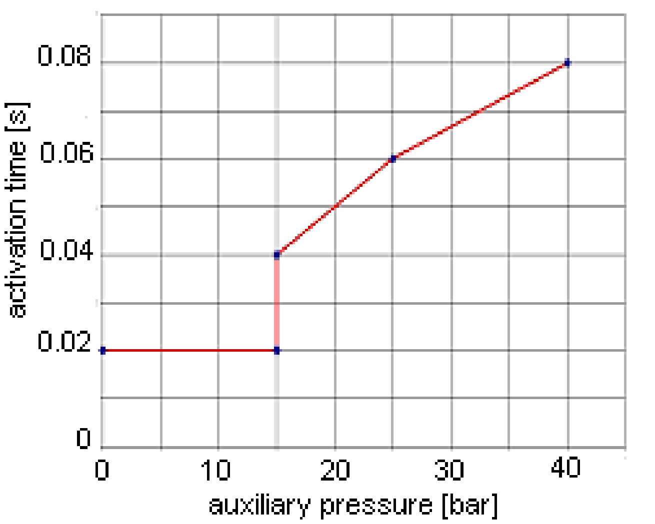

Figure 5 – Minimum duration of pump intervention as a function of $p_{auxiliary}$

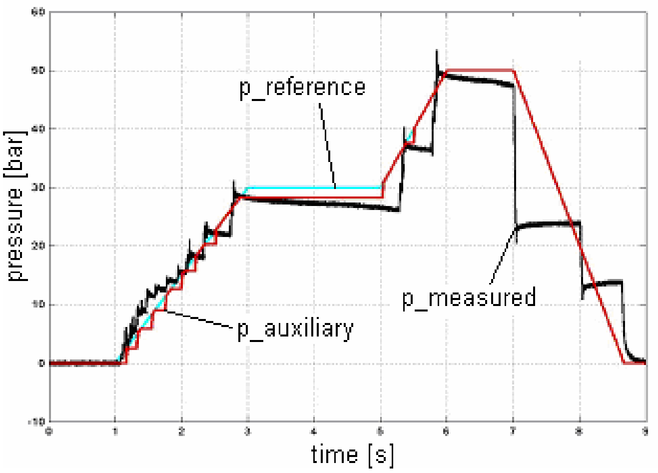

Figure 6 – Example of test to verify the performance of the actuation algorithm

Figures 6 and 7 show comparisons between the desired and the obtained pressure level. It is evident the consistent approximation in pressure modulation, especially during the pressure reduction phase, due to the slow dynamics of the '2' valves of the considered hydraulic unit. For an EHB unit the reference caliper pressures of Figures 6 and 7 would correspond to the equivalent of a base brake maneuver (decided by the driver) and would be performed with a nearly null offset between reference and measured pressures (thanks to the efficiency of EHB valves and to the pressure sensors adopted inside EHB hydraulics to measure pressure levels at the output ports of the hydraulic unit).

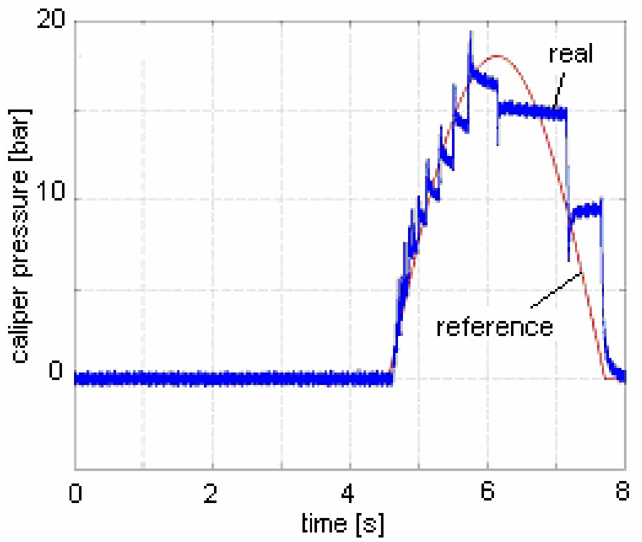

Figure 7 – Example of test to verify the performance of the actuation algorithm: comparison between reference and measured pressures

The effect of ESP hydraulics for TC performance

Figures 8 and 9 are about a start-up maneuver in split-μ conditions. They compare the experimental behavior (measured at the test bench) of a vehicle equipped with a commercial ESP unit, actuated according to the algorithm described in the former paragraph, and a vehicle equipped with a simulated EHB hydraulic unit. At about 8 s, at the end of the brakes intervention, the vehicle with the commercial ESP has obtained only the 70% of the useful effect in terms of longitudinal speed, in comparison to an EHB unit.

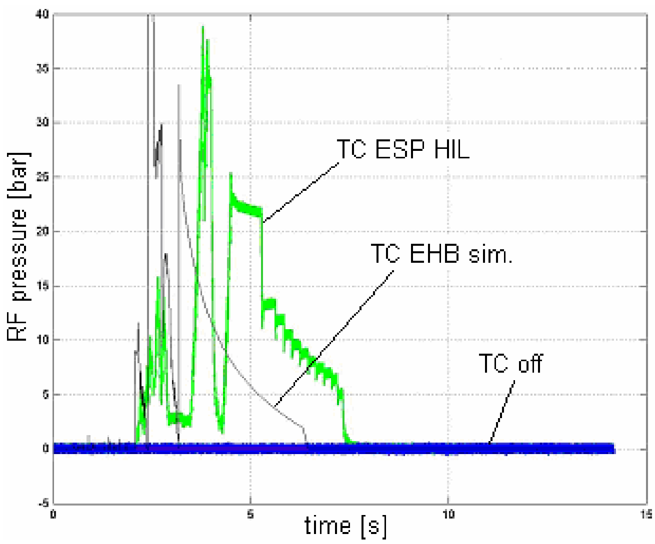

Figure 8 – Time history of low adherence caliper pressure during a start-up maneuver in split-μ conditions

The efficiency of the hydraulic unit with the control algorithm can be computed by the following index:

$$I_1 = \frac{V_{active,end\_brake} - V_{passive,end\_brake}}{V_{passive,end\_brake}} \qquad (9)$$

where V_{active,end_brake} is the longitudinal speed of the vehicle with TC at the end of the brake intervention carried out by TC and $V_{passive,end\_brake}$ is the longitudinal speed of the passive vehicle in correspondence of the end of brakes intervention for the active vehicle (this kind of comparison between a passive and an active vehicle is possible only thanks to HIL simulation, it would be too difficult in the form of road tests). The higher is the value of the index, the more efficient is the evaluated TC system.
Conventional ESP units appear critical during the slow pressure decrease and pressure modulation phases typical of traction control.

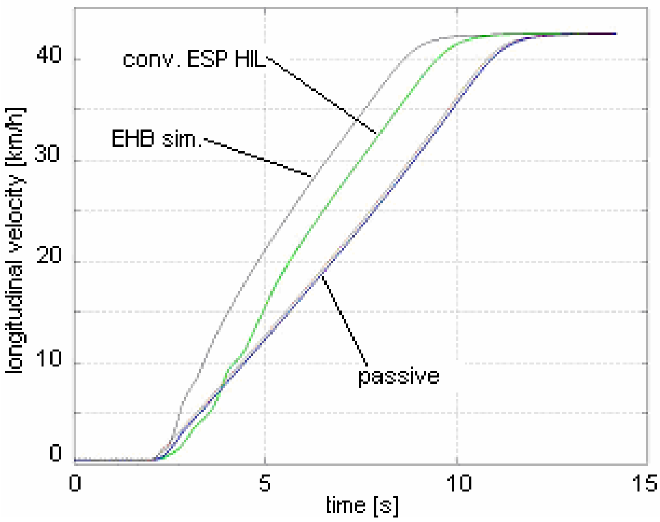

Figure 9 – The effect of the performance of the ESP hydraulic unit from the point of view of vehicle longitudinal speed

The effect of ESP hydraulics for yaw rate control performance

The same procedure was carried out also for body yaw rate control. Figure 10 compares reference and actual pressures measured at Politecnico di Torino brake systems HIL test bench during a double step steer maneuver, in the condition of high friction between tires and ground. The maneuver was performed by a conventional ESP hydraulic unit. Pressure increase phases appear to be critical due to the very high requested dynamics, whereas pressure decrease phases can be followed quite well by the ESP unit due to the quite consistent pressure gradients. Figures 11 and 12 show the HIL results related to a step steer in the condition of a low friction coefficient between tires and ground. Figures 13 and 14 are the comparison of the behavior, in terms of body yaw rate and body sideslip angle, of a vehicle equipped with a commercial ESP hydraulic unit and a vehicle equipped with an EHB hydraulic unit, governed by the same yaw rate feedback control. The maneuver is an extreme step steer. It is possible to define the efficiency of the body yaw rate control by adopting the following index:

$$I_2 = \frac{\dot{\psi}_{max} - \dot{\psi}_{min}}{\dot{\psi}_{end\_of\_maneuver}} \qquad (10)$$

where $\dot{\psi}_{max}$ and $\dot{\psi}_{min}$ are the maximum and the minimum values of body yaw rate, $\dot{\psi}_{end\_of\_maneuver}$ is the value of body yaw rate after vehicle stabilization. A small value of I_2 corresponds to a good vehicle behavior.

	Passive	ESP	EHB
I_2	0.75	0.5	0.25

Chart 1 – Values of I_2 for the passive vehicle, the vehicle with conventional ESP hydraulics and the vehicle with EHB during the extreme step steer maneuver of Figures 13 and 14

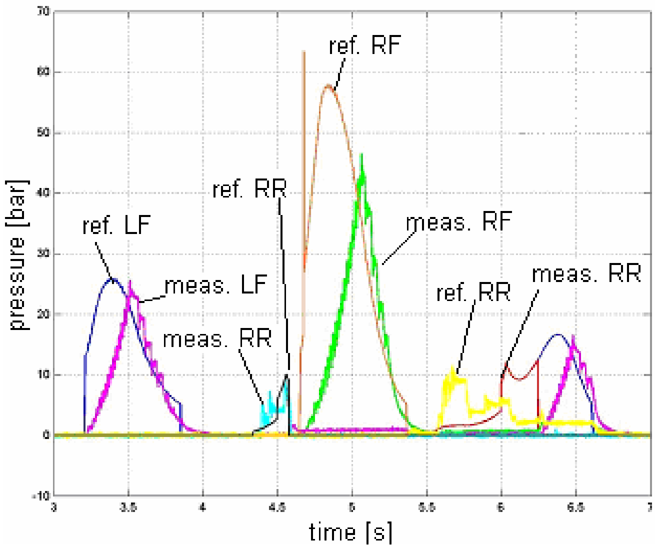

Figure 10 – Time history of reference ('ref.') and measured ('meas.') calipers pressures during an extreme double step steer maneuver

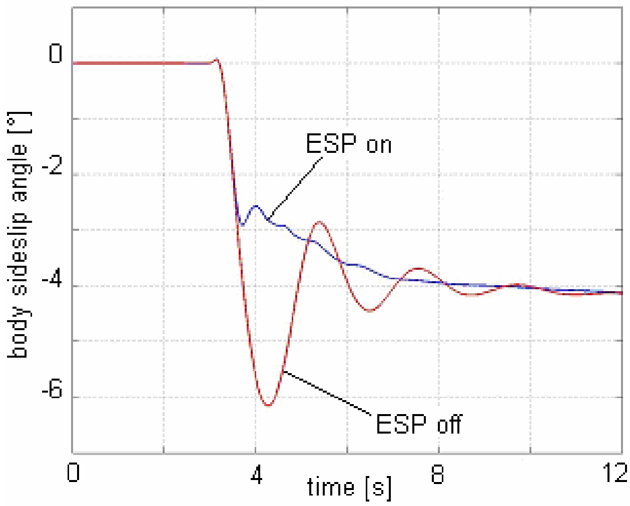

Figure 11 – Time history of body sideslip angle with and without ESP for the same vehicle; step steer in the condition of a low friction coefficient between tires and ground

In terms of I_2 (chart 1), the conventional ESP unit can use only the 50% of the possible improvement of vehicle dynamics during the step steer maneuver. The biggest efficiency of EHB is due to the largest obtainable pressure gradients during the pressure increase phases, whereas no substantial difference can be observed during the pressure decrease phase, due to the high values of the requested pressure gradients. Large pressure gradients during pressure decrease can be followed quite well also by conventional ESP units. Conventional ESP hydraulic units appear to have consistent chances of improvement, much more than those related to the control algorithm, which, in this kind of maneuver, already decides a consistent intervention of the brakes which could not be further anticipated by the software. The most significant improvements could be related to an increased flow rate of the pump to improve the response during the transient and to more precise electro-valves during modulation with low pressure gradients.

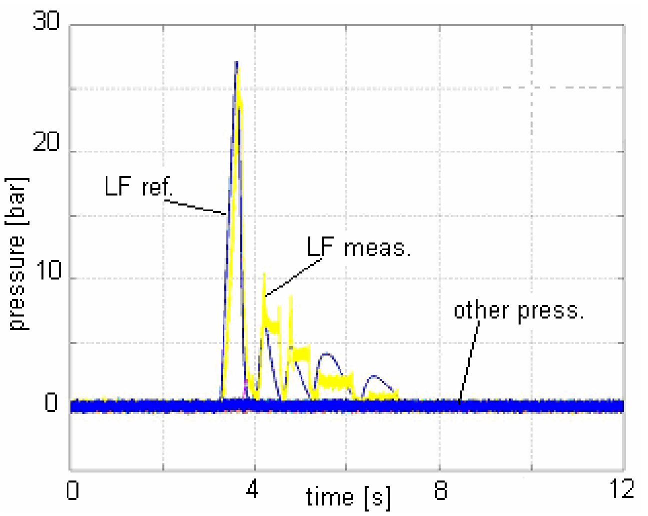

Figure 12 – Time history of reference (‘ref.’) and measured (‘meas.’) Left Front (‘LF’) and other (‘other press.’) calipers pressures during the same maneuver of Figure 11

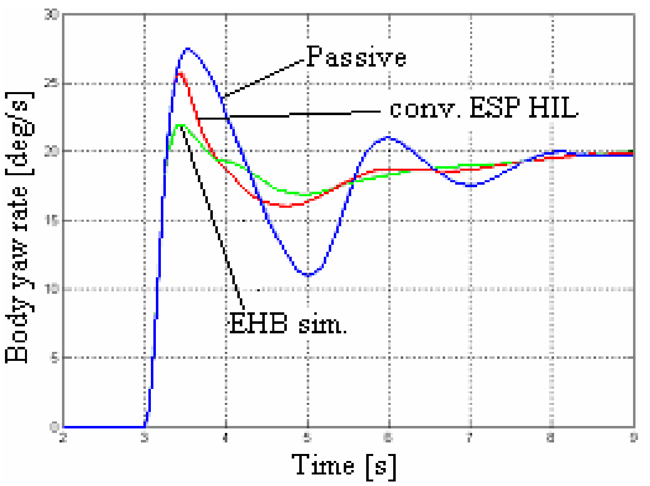

Figure 13 – Time history of body yaw rate during an extreme step steer maneuver (high friction coefficient between tires and ground); comparison between a passive vehicle (‘Passive’), a vehicle with a conventional ESP hydraulic unit (‘conv. ESP HIL’) and a vehicle with an EHB (‘EHB sim.’) unit

3. Conclusions

The maximum possible frequencies of brake actuation are not much higher in comparison to those of vehicle body. The hardware of the brake system has a fundamental weight in determining the performance of an ESP in terms of vehicle dynamics. HIL simulation can provide an objective evaluation of the performance of ESP hydraulic units, especially from the point of view of the influence of hydraulic parameters on vehicle dynamics. The effect of the main hydraulic components on vehicle dynamics and handling is explained. Motor pump displacement is fundamental for body yaw rate control performance whereas precision in valves modulation is particularly important for Traction Control. The possible margins of improvement which characterize commercial ESP units are demonstrated. EHB leads to consistent advantages over conventional ESP, from the point of view of ABS function, Traction Control function and body yaw rate control.

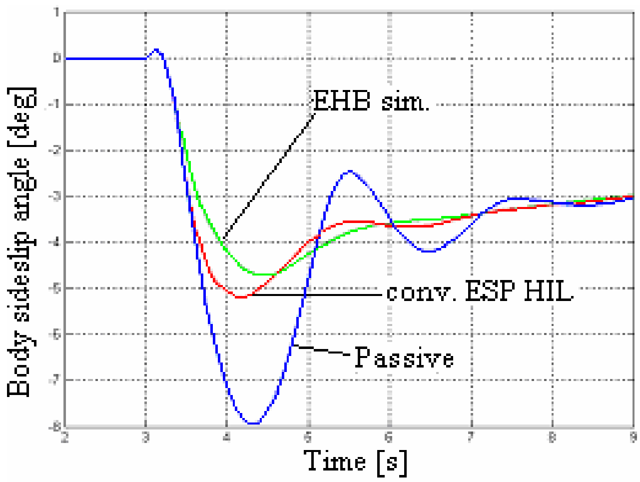

Figure 14 – Time history of body sideslip angle during an extreme step steer maneuver (the same maneuver of Figure 13); comparison between a passive vehicle (‘Passive’), a vehicle with a conventional ESP hydraulic unit (‘conv. ESP HIL’) and a vehicle with an EHB (‘EHB sim.’) unit

References

1. M. Velardocchia, A. Sorniotti, ‘Hardware-In-the-Loop (HIL) Testing of ESP (Electronic Stability Program) Commercial Hydraulic Units and Implementation of New Control Strategies’, ‘SAE 2004 Transactions – Journal of Passenger Cars – Mechanical Systems’, Vol. 113, Section 6, Ed. SAE International, Warrendale, ISBN: 0-7680-1555-3, pp. 1177-1185.
2. N. D’Alfio, A. Morgando, A. Sorniotti, M. Velardocchia, ‘Base Model Simulator (BMS) - A Vehicle Dynamics Model to Evaluate Chassis Control Systems Performance’, SAE Paper 2005-01-0401, Vehicle Dynamics and Simulation 2005, ISBN 0-7680-1561-8.
3. N. D’Alfio, A. Morgando, A. Sorniotti, ‘Electro-Hydraulic Brake System: Design and Test through Hardware-In-the-Loop Simulation’, 19th IAVSD Symposium, Dynamics of Vehicles on Road and Tracks, Milan, Italy.
4. A. T. van Zanten, ‘Bosch ESP Systems: 5 Years of Experience’, SAE Paper 2000-01-1633.

Contact
Aldo Sorniotti, Politecnico di Torino, Department of Mechanics, Corso Duca degli Abruzzi 24, 10129 Torino, ITALY, email: aldo.sorniotti@polito.it