Proceedings of the 4th WSEAS International Conference on Fluid Mechanics and Aerodynamics, Elounda, Greece, August 21-23, 2006 (pp266-274)

A Methodology to Investigate the Dynamic Characteristics of ESP Hydraulic Units - Part I: Hydraulic Tests

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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) 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 first part of this paper suggests the main parameters and tests which can help the car manufacturer in evaluating ESP hydraulic units, without expensive road tests.

Keywords – Electronic Stability Program, Electro-valves, Pump, Delays

1. Introduction

ESP is getting to be the fundamental system for the improvement of vehicle dynamics. One of the main targets of the vehicle designer consists in choosing the best actuation unit for vehicle dynamics control. In fact, from experimental tests it is evident that the quality of the performance of an ESP unit is consistently determined by its hardware and not only by its control algorithm. Car manufacturers usually evaluate the performance of the passive elements of the brake system, like booster, tandem master cylinder and calipers, often without considering at all the efficiency of the ESP hydraulic unit. Most of the sources in literature describe ESP control algorithms without considering the limitations connected to the hardware of the system. Hardware-In-the-Loop (HIL) testing can be a useful solution for this problem. This paper describes the procedure for the evaluation of the performance of ESP hydraulic units and their effects on vehicle dynamics. The instrument is Politecnico di Torino HIL braking systems test bench [1]. 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 on the main components of the brake system. Pressure sensors at the wheels calipers send their signals to a 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 input to the vehicle model. The vehicle model is properly implemented for this HIL application. A detailed description of the connection between the vehicle model and the hardware of the braking system will be presented in the second paper about this research. The first part of this activity is devoted to the evaluation of the main parameters of the hardware of the hydraulic unit which can have an influence on the ESP performance from the point of view of vehicle dynamics.

2. The performance of the components of conventional ESP units

The first tests consist in activating the single valves of ESP commercial hydraulic units, to detect the dynamic performance of each component, in terms of precision, delay and inertia. At this step, the interface between the vehicle model and the braking system components is not yet adopted. Figure 1 shows the typical scheme of an ESP hydraulic unit, characterized by 12 electro-valves and a motor pump. More details about its internal work are included in [1]. The most important valves to be characterized inside the ESP units are those indicated as '1' and '2' in Figure 1. There is a couple of these valves for each wheel. They are used for pressure modulation inside wheel calipers. There is one valve '3' and one valve '4' for each of the two hydraulic circuits of the ESP unit.

VALVE '1'

Figure 2 shows an example of valve connecting tandem master cylinder with the brake calipers ('1'). Figure 3 is a typical sketch of the internal components of the valve. The specific one represented in the figure is characterized by a uni-directional spherical valve to provoke a quicker pressure release at the brake caliper at the end of each brake maneuver. Valve '1' is fundamental for pressure modulation during ABS interventions. It gets open during pressure increase phases. It has to be characterized by a fast response when getting fully open, to give origin to a quick pressure increase when requested by the control algorithm. At the same time, it has to be characterized by a consistent precision for a smooth pressure modulation, for example during the interventions of ESP when the driver is not pushing the brake pedal. In this case, high pressure oscillations induced by motion of the valve plunger can lead to an uncomfortable feeling of the whole vehicle. Pressure oscillations during the modulation can be considered unperceivable by the passengers when they are minor than 0.3-0.5 bar. Common production '1' valves for ESP provoke minimum pressure oscillations of about 1-1.5 bar.

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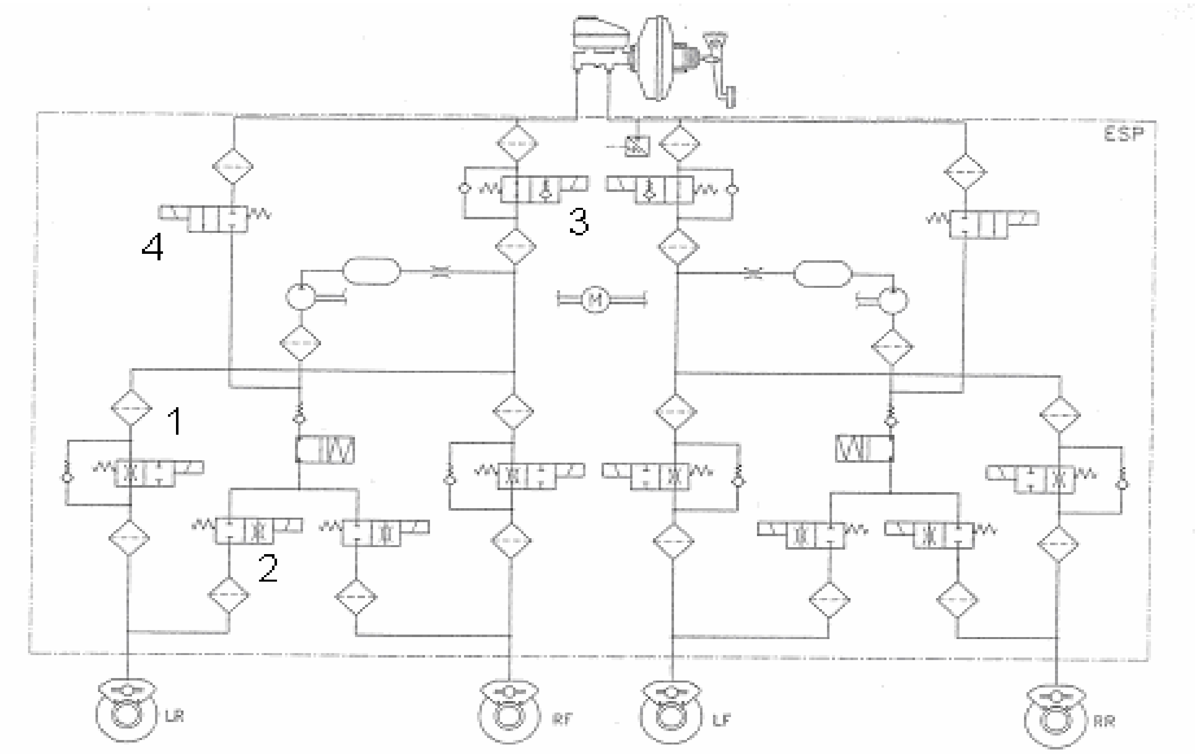

Figure 1 – Scheme of an ESP hydraulic unit

Typical tests consist in imposing a desired displacement of booster input rod on the HIL test bench and in generating a step signal for the valve '1' of Figure 1 when tandem master cylinder pressure has reached a pre-defined level. These tests can be summarized in charts like that one of Figure 4, which plots the pressure gradient obtainable at front and rear calipers as a function of master cylinder pressure. These charts permit an objective prediction of the efficiency of the ABS system to recover wheel slip during pressure increase phases.

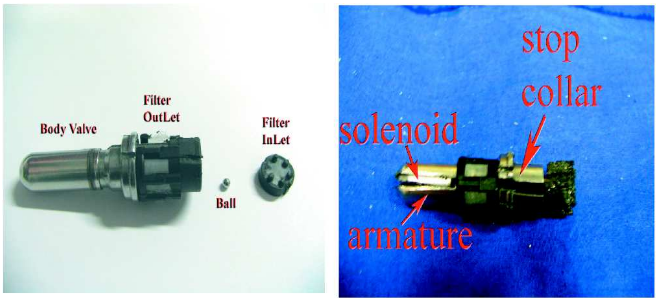

Figure 2 – Elements of one of the valves indicated as '1' in Figure 1

Figure 5 shows examples of modulation of the valve '1', typical condition of body yaw rate or traction control. Tandem master cylinder is maintained at a pre-defined pressure level, whereas the valve is subjected to a fixed number of electric pulses (each of which having a duration of 0.001 s) every 0.01 s. The obtained characteristics show the possible precision during ESP modulation. In the specific case, pressure oscillations are within 1.5 bar. Another important characteristic is the minimum excitation time t_{min} necessary to observe an effect in terms of calipers pressure variation. Usual values for t_{min} are about 0.001 s.

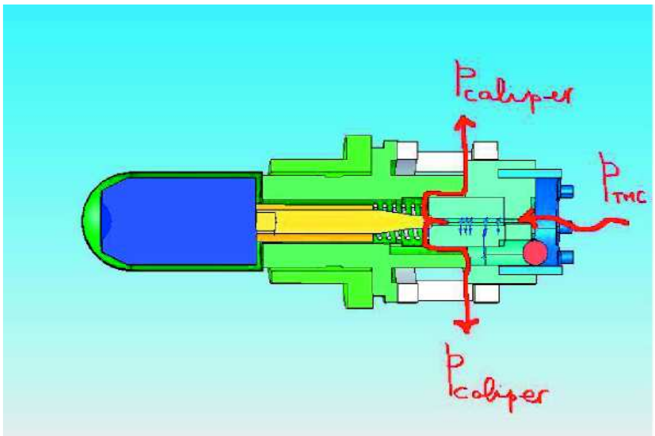

Figure 3 – Sketch of a commercial '1' valve

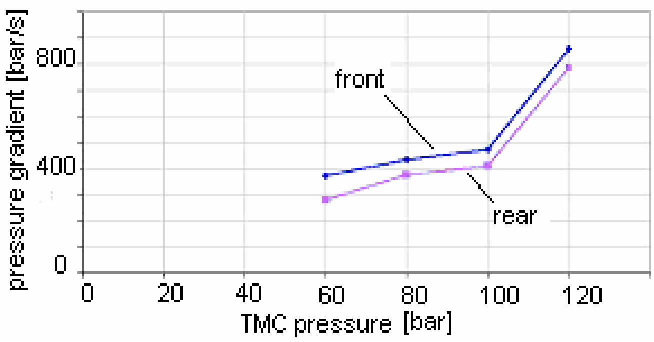

Figure 4 – Example of results of the tests with the full opening of the valve '1', for a front and a rear caliper

A small time corresponds to a valve with fast dynamics. Valve '1' can be characterized also from the point of view of its unidirectional valve visible in Figure 3.

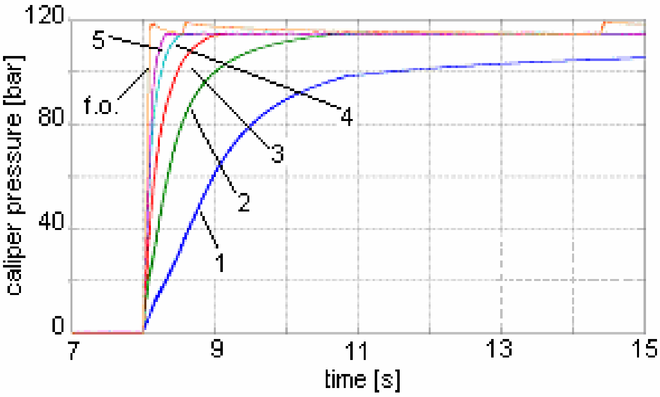

Figure 5 – Examples of modulation carried out with a valve ‘1’ in connection with a front caliper; 1: 1 pulse each 0.01 s, 2: 2 pulses each 0.01 s, 3: 3 pulses each 0.01 s, 4: 4 pulses each 0.01 s, 5: 5 pulses each 0.01 s, f.o.: fully open

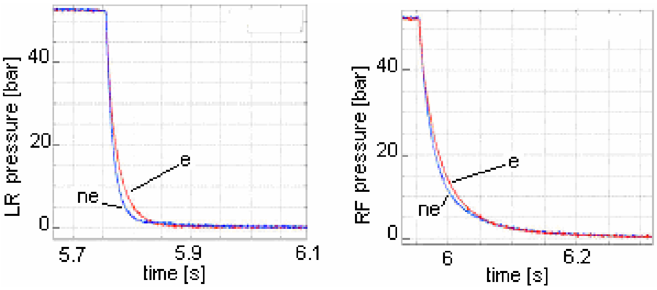

Figure 6 – Wheel caliper pressure as a function of time; characterization of the uni-directional valve inside the ‘1’ valve, e: valve ‘1’ excited, ne: valve ‘1’ not excited

Figure 6 compares pressure drops at Left Rear (LR) and Right Front (RF) calipers in two tests. They consist in pushing booster input rod until a pre-defined pressure level is reached at the caliper. Firstly valve ‘3’ of Figure 1 is closed to maintain caliper pressure and then it is opened after booster input rod is retracted to the initial position. In the first case, valve ‘1’ is maintained closed (it is excited) during the whole discharge process, in the second case it is maintained open. By observing the difference between the pressure drop with and without valve ‘1’ activation, it is possible to obtain the characteristics of both the uni-directional valve and the plunger valve, in terms of their equivalent area and hydraulic diameter. Pressure drop is smaller when valve ‘1’ is excited, since the whole flow rate passes through the spherical unidirectional valve.

VALVE ‘2’

Similar tests are carried out also for the valve ‘2’ (Figure 1), which connects the wheel caliper with the low pressure accumulator and the pump intake port. Figure 7 corresponds to tests devoted to identify the pressure gradients obtainable through the activation of the valve, without switching on the motor pump. Figure 7 is obtained through the activation of the valve ‘2’ of the Left Rear (LR) caliper, Figure 8 is obtained through the activation of the valves ‘2’ of both the calipers of the same hydraulic circuit. In this case, pressure does not reach a null value at the end of the experiment, due to the limited volume of the low pressure accumulator. During the experimental characterization of commercial ESP hydraulic units, valves ‘2’ usually demonstrate to be less precise in modulation in comparison to valves ‘1’. Valves ‘2’ of commercial ESP hydraulic units have to be excited for a consistent time to get open. In this case, they provoke pressure drops which have an amount of several bars.

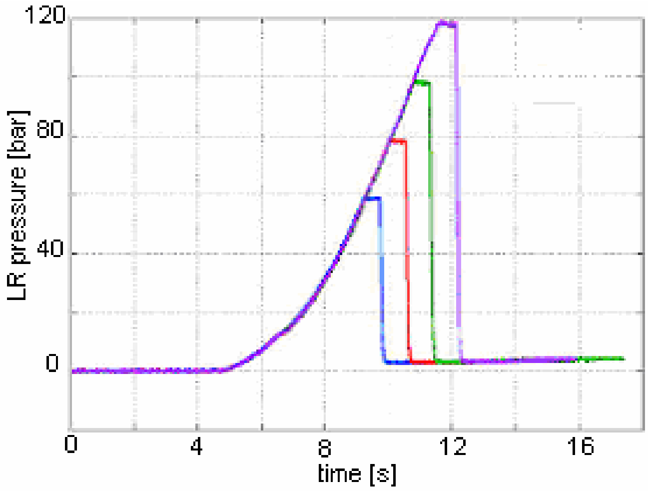

Figure 7 – Tests devoted to measuring the pressure gradients guaranteed by the valve ‘2’ from different initial pressure levels on the Left Rear (LR) caliper

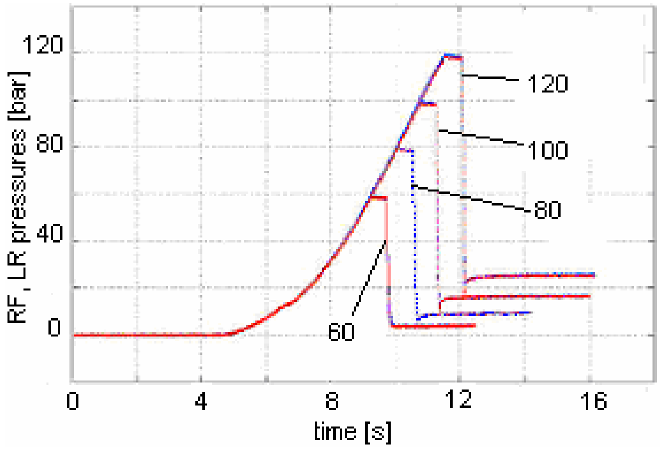

Figure 8 - Tests devoted to measuring the pressure gradients guaranteed by the valves ‘2’ of Right Front (RF) and Left Rear (LR) calipers, from different initial pressures (120, 100, 80, 60 bars)

To obtain a generic desired time history of pressure reduction (Figure 9a) through valves ‘2’, it is necessary a variation of the PWM (Pulse Width Modulation) modulation frequency, as shown in Figure 9b. During pressure modulation tests, it is fundamental to observe the variation of the results as a function of the duration of the former activation history of the solenoids.

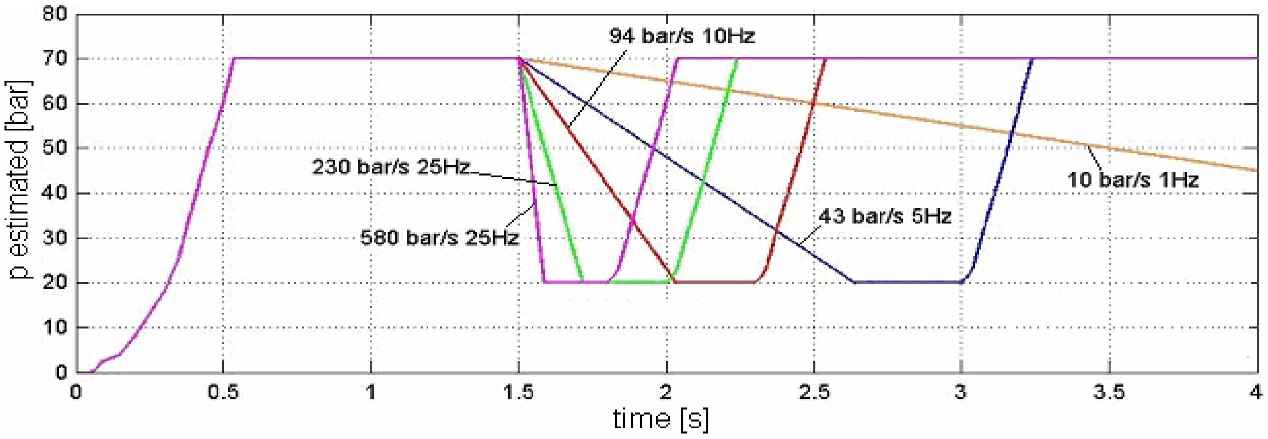

Figure 9a – Example of modulation tests of the '2' valve: time history of reference pressure ('p estimated')

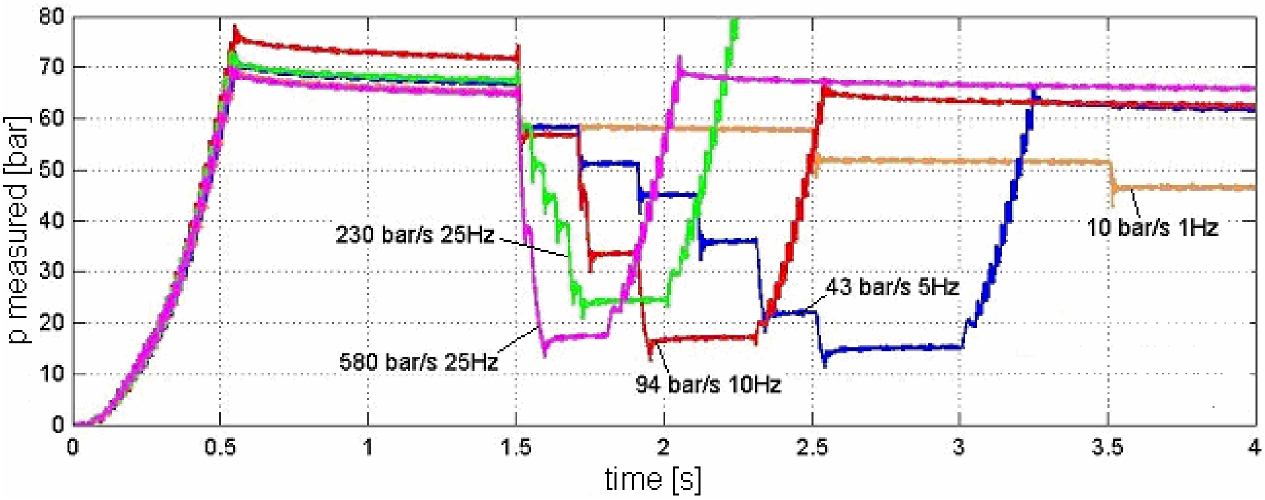

Figure 9b – Examples of modulation tests of the '2' valve: time history of wheel caliper pressure; PWM frequency has to be varied to obtain the desired pressure gradient

Temperature variation can provoke a consistent variability of the performance of the valve, which can only partially be compensated by the ESP control algorithm. ESP interventions should not be very frequent and their duration should not be very long. As a consequence, thermal phenomena inside the hydraulic unit should not represent a fundamental problem. Tests similar to those described for '1' and '2' valves were performed also for '3' and '4' valves. These valves should not be subjected to an intensive modulation like '1' and '2' valves.

MOTOR PUMP

The same procedure was adopted for the motor pump. Its performance is particularly important, from the point of view of vehicle dynamics, in the case that ESP control algorithm decides a pressure increase for some calipers when the driver is not pressing the brake pedal. Consistent delays between the activation of the motor pump and the effective pressure increase could make the ESP intervention unsafe in recovering excessive under/over-steer. Figure 10 shows a typical experimental test of the motor pump. Valves '3' and '4' are activated to generate a pressure increase at the wheel caliper. The minimum time necessary to have a pressure level at the calipers of about 30-35 bar (the typical values for an intervention to contrast over-steer) cannot be neglected in comparison to the period of the typical yaw rate free oscillations of vehicle body after a step steer. The motor pump has to provoke a fast pressure increase during each of the increase phases of body yaw rate, to limit the trend towards over-steer. This pressure increase has to regard the wheels located on the outer side of the bend. The motor pump also gives origin to wheels pressures on the internal wheels in correspondence of the lower peaks of body yaw rate. By linearization of vehicle behavior, it is possible to consider the following formula [3] for the estimation of the typical natural undamped frequencies of body yaw rate motion (which correspond, in first approximation, to the activation frequencies of the motor pump during step steer or double step steer maneuvers):

$$\omega_n^2 = \frac{C_F C_R l^2}{m^2 k^2} \frac{1 + K{V_{vehicle}}^2}{{V_{vehicle}}^2} \qquad (1)$$

where ω_n is the natural pulsation of yaw rate motion, C_F and C_R are front and rear side stiffness (they include tires and suspensions behavior), m is vehicle mass, k is vehicle body inertial radius, K is vehicle understeer coefficient and $V_{vehicle}$ is vehicle speed. The typical frequencies which can be obtained through (1) are usually between 1 and 3 Hz and vary as a function of vehicle speed. Larger vehicle speeds involve lower frequencies for the yaw motion. As a consequence, for an ESP hydraulic unit a step steer with a larger steering wheel angle at a lower speed could be more critical than a step steer with a limited steering wheel angle amplitude at a very high speed. This statement is valid from the point of view of the involved frequencies, without considering the dependency of yaw damping on vehicle velocity. Also the maximum frequencies of excitation of vehicle yaw motion through the steering wheel cannot exceed 3-4 Hz, even by professional test drivers. Drivers tend to turn the steering wheel in a disordered way when perceiving a sudden obstacle. The ESP must be capable of controlling this kind of events through sufficiently rapid pressure increase phases generated by the motor pump on the wheels of the two sides of the car. An important parameter which can have an influence over vehicle dynamics is the inertia of the motor pump. Figure 10 shows that pressure increase starts with a consistent delay in comparison with motor pump activation and that pressure increase ceases more than 0.1 s after the end of motor pump activation. In the test, caliper valve '1' is not closed at the end of the motor pump activation to identify inertial phenomena, which provoke a further pressure increase at the caliper. HIL tests make see that it can be convenient, for a reduced response time, to put in rotation the motor pump before exciting valves '3' and '4', in order to compensate the initial inertial phenomena. The activation of the motor pump could be performed when the difference between the desired yaw rate and the real one is consistent but below the activation threshold of yaw rate control. When activation threshold is reached, also '3' and '4' valves can be closed. Figure 11 is an example of the possible performance improvement related to this kind of control algorithm. Figure 12 plots the time history of pressure oscillations at the calipers due to the intervention of the motor pump. By observing the frequency of these oscillations, it is possible an estimation of the rotational speed of the pump unit. The flow rate can be estimated by multiplying the calculated rotational speed by the volume displacement of the pump, which can be deduced by the geometrical characteristics of the piston and the cam of the pump. ESP pumps are radial pistons pumps. They are characterized by two pistons, one for each hydraulic circuit. Their volume (for a single circuit of the brake system) displacement V corresponds to the product between the area of the piston A_{piston} and the piston displacement c_{piston}. Equation (2) permits the estimation of pump flow rate.

$$Q = \omega \cdot V = \omega \cdot A_{piston} \cdot c_{piston} \tag{2}$$

Through the described experimental tests, it is possible to deduce pump flow rate also on the basis of the volume displacement diagram of the calipers of the brake system. It is shown in Figure 13. It plots caliper volume displacement as a function of tandem master cylinder pressure (which is equal to caliper pressure during this test). This curve can be obtained by disconnecting, one after the other, all the components of a brake system and by repeating a slow brake application after each disconnection [4]. The result is constituted by several curves plotting booster input rod or tandem master cylinder primary piston displacement $c_{rod}(p)$ as a function of pressure p. In first approximation, the displacement corresponds to the volume absorption by the components of the brake system:

$$V_{displacement}(p) = A_{TMC} \cdot c_{rod}(p) \tag{3}$$

where A_{TMC} is tandem master cylinder area. The volume displacement for each component can be computed by subtracting the volume displacement of the brake system without the considered component to the volume displacement with the considered component. In formulae:

$$V_{displacement,component}(p) = V_{displacement,with\_component}(p) - V_{displacement,without\_component}(p) \tag{4}$$

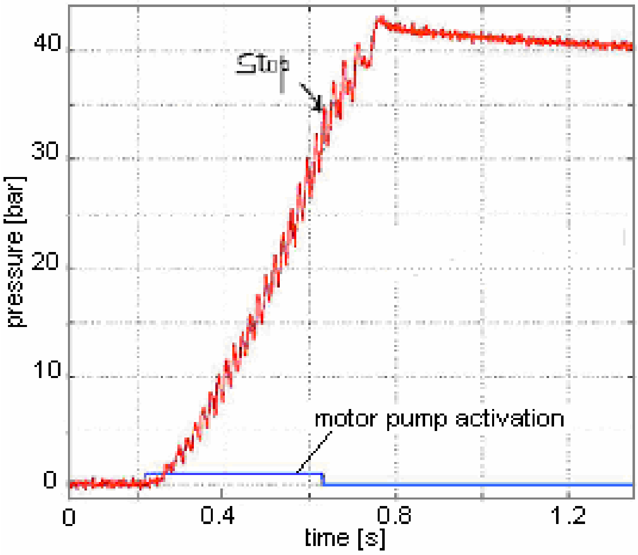

Figure 10 – A typical experimental test of the motor pump: evaluation of the inertial effects

By the tests on the motor pump, it is possible to find out the time $t_{1\to2}$ necessary to pass from a pressure level p_1 to a pressure level p_2 at the wheel caliper. From the volume displacement characteristic of Figure 13 (it belongs to a front caliper), it is possible to compute the volume $V_{1\to2}$ absorbed between the two pressure levels. The average pump flow rate between p_1 and p_2 can be computed as:

$$Q = \frac{V_{1\to2}}{t_{1\to2}} \tag{5}$$

By comparing the results from (5), (3) and (2), the effective flow rate guaranteed by the pump mounted on the brake system can be found out.

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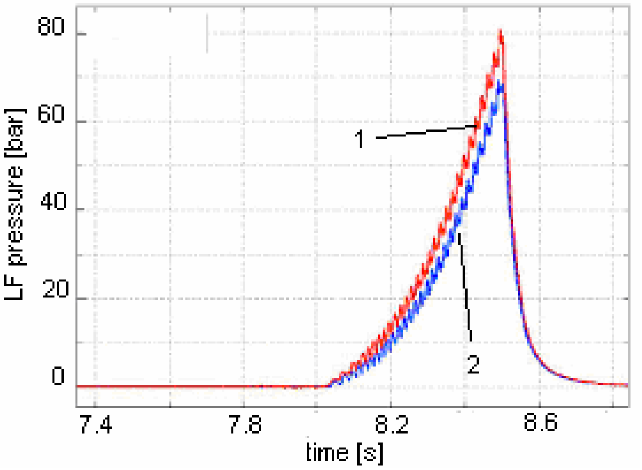

Figure 11 – Left Front (LF) caliper pressures as a function of time: effect of the anticipated activation of the motor pump (line 1) in comparison to the contemporary activation of the valves and the motor pump (line 2) during a pressure increase phase

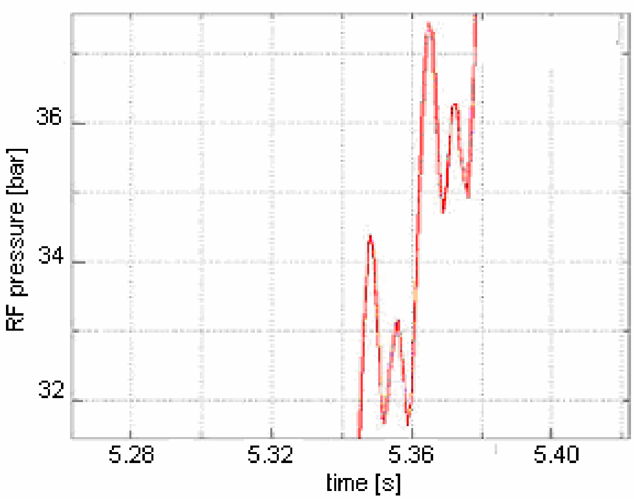

Figure 12 – Time history of Right Front (RF) wheel caliper pressure during motor pump activation

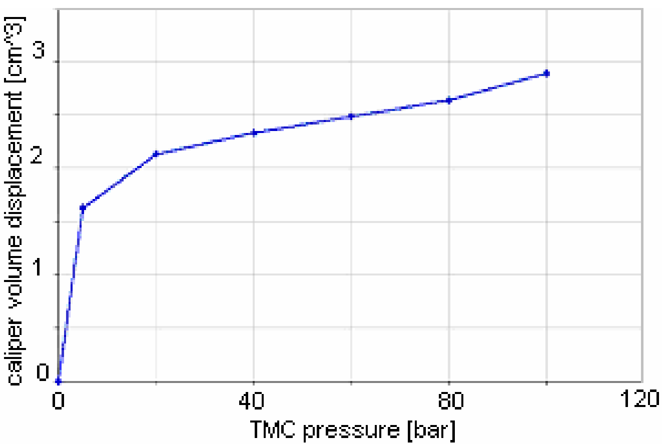

Figure 13 – Volume displacement as a function of pressure for a front caliper of a middle-size sedan

Figure 14 is about a typical characterization of the motor pump, to investigate the chance for a PWM modulation. This technique was presented in literature [5] and should give origin to:

- a smoother pressure modulation at the wheel caliper during ESP activation when the driver is not pushing the brake pedal
- a reduced level of pedal vibration during ABS activation.

The tests of Figure 14 consider a constant time interval of 0.2 s between two activations of the motor pump and a variable duration of the activation of the motor pump. It is possible to see that the minimum activation time to have a pressure increase at the caliper must be prolonged with pressure, otherwise the effect of the motor pump cannot be perceived in terms of pressure increase.

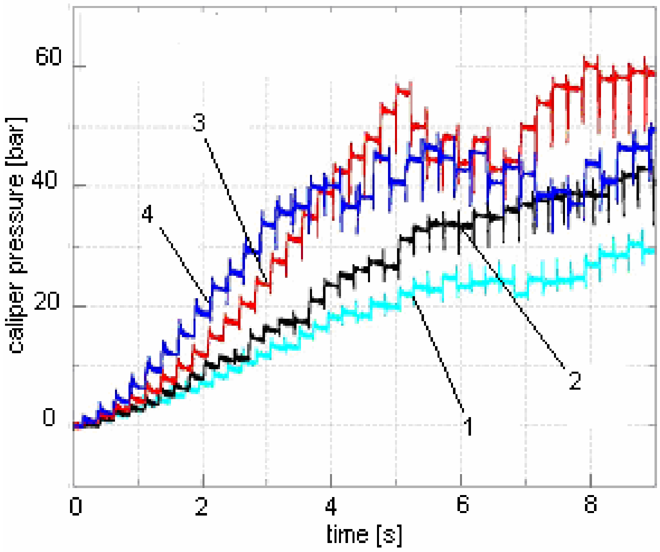

Figure 14 – Tests of modulated activation of the motor pump; 1: activation of 0.02 s every 0.2 s, 2: activation of 0.03 s every 0.2 s, 3: activation of 0.04 s every 0.2 s, 4: activation of 0.05 s every 0.2 s

LOW PRESSURE ACCUMULATOR

The same procedure was applied for the characterization of the low pressure accumulator (Figure 15). It is typically a spring-charged piston sliding inside a cylinder in the carcass of the ESP hydraulic unit. The low pressure accumulator is used to have an immediate pressure reduction at the wheel caliper after the activation of the ‘2’ valve, in spite of the inertia of the motor pump. In addition to this, low pressure accumulator is adopted for pressure reduction phases during Electronic Brake Distribution interventions, to prevent consistent brake pedal oscillations provoked by the motor pump activation. As a consequence, low pressure accumulator must have an internal volume corresponding to the full discharge of the rear calipers during a brake maneuver. The volume displacement of the low pressure accumulator (as a function of pressure) can be measured through tests at the HIL bench.

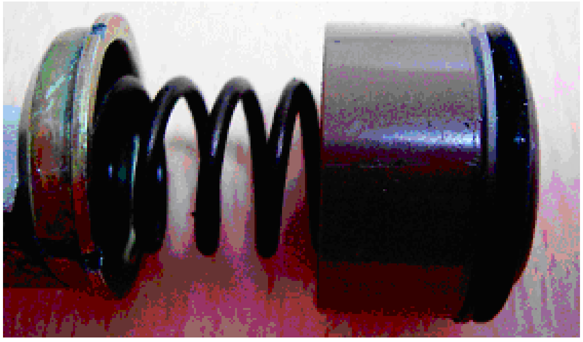

Figure 15 – An example of low pressure accumulator of a commercial ESP hydraulic unit

Figure 16 is an example of experimental result. The motor pump unit is activated for a pre-defined time interval, to be sure that the low pressure accumulator is empty. Then a semi-stationary brake maneuver (booster input rod speed of 3 mm/s) is simulated by the HIL bench, with the contemporary activation of the valves '2' (Figure 1) connecting the calipers to one of the low pressure accumulators. By comparing booster input rod displacement obtained in this test with the input rod displacement during the same test without valves activation, it is possible, as stated by equations (3) and (4), to obtain the volume displacement of the low pressure accumulator as a function of pressure.

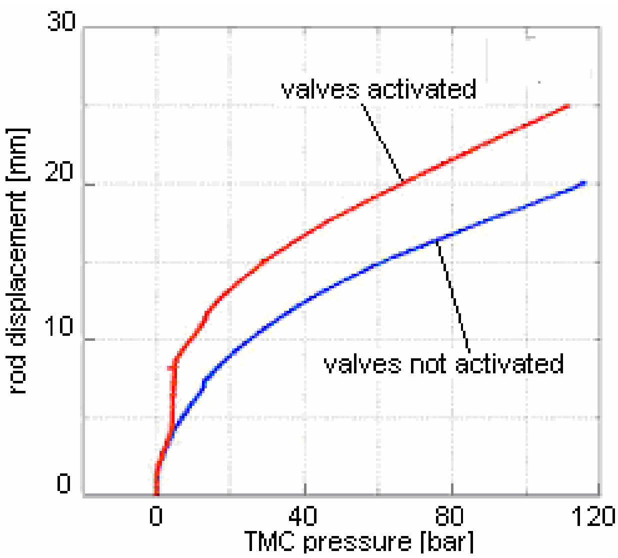

Figure 16 – Example of experimental characterization of the low pressure accumulator of a commercial ESP unit at Politecnico di Torino HIL test bench: the difference between the input rod displacements corresponds to the volume absorbed by the low pressure accumulator

THE EFFECT OF ESP ORIFICES ON VEHICLE DYNAMICS AND PEDAL FEELING

ESP orifices provoke consistent drops [6] in pressure between tandem master cylinder and the brake caliper during panic brake maneuvers, even if ESP valves are not activated. Figure 17 plots the estimated flow rate (on the basis of booster input rod displacement) through the ESP hydraulic unit as a function of the pressure drop between tandem master cylinder and the brake caliper, during an emergency brake maneuver simulated on the HIL test bench. After the initial transient with the deformation of the internal elements of the booster (from point O to point A), the pressure drop *Δp* vs. flow rate *Q* characteristic follows the typical law of turbulent motion (independently of booster vacuum level – the test was repeated for 0 and –0.8 bar of vacuum level):

$$Q = C_q \cdot A_{equivalent} \cdot \sqrt{\frac{2 \cdot \Delta p}{\rho}} \qquad (6)$$

where C_q is flow coefficient, $A_{equivalent}$ is the equivalent orifice area of the ESP unit, ρ is fluid density. In particular, this test permits to carry out an estimation of $A_{equivalent}$ and compare it with the theoretical value from the geometrical characteristics of the valve '1' (Figure 3). Valve '1' is the main responsible for this pressure drop. The HIL test bench permits the objective and repeatable evaluation of the vehicle stopping distance as a function of the different configurations of the brake system (the connection between the vehicle model and the hardware of the brake system will be explained in detail in the second paper about this work). For example, Figure 18, obtained through the HIL bench, shows that ESP hydraulic unit provokes an increase of stopping distance, due to the slower caliper pressure increase at the beginning of the brake maneuver, in comparison to the same brake system without ESP unit. It can have an amount of over 60 cm during a brake maneuver from an initial vehicle speed of 100 km/h in high adherence conditions. During the tests of Figures 17 and 18, ESP valves were not activated.

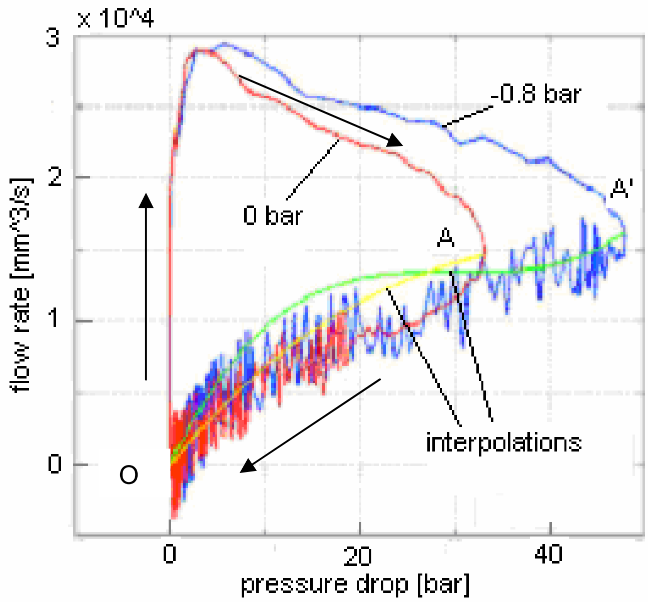

Figure 17 – Test result to evaluate the effect of ESP hydraulic unit on brake system performance

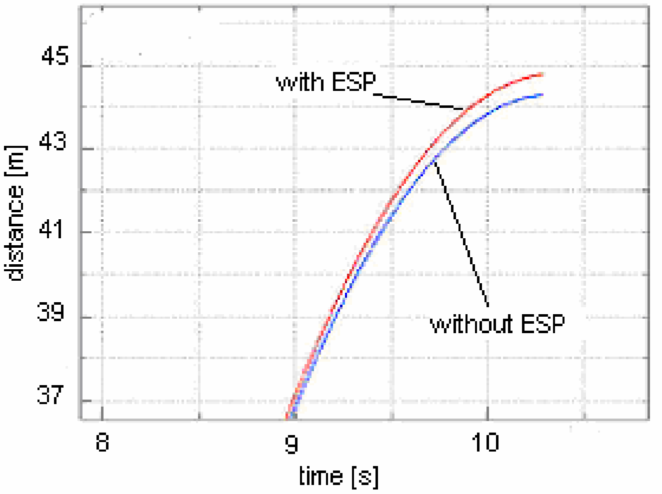

Figure 18 – Variation of the stopping distance induced by the ESP hydraulic unit

3. Conclusions

On the basis of the presented tests and HIL simulations, it can be concluded that:

1. ESP electro-valves should have orifices as large as possible, not to provoke a decay of pedal feeling and vehicle deceleration during the first instants of an emergency brake maneuver;
2. ESP electro-valves (especially those indicated as ‘1’ and ‘2’) should guarantee a smooth modulation of wheels pressures and should be characterized by very high dynamics;
3. ESP motor pump should guarantee a consistent flow rate to generate a quick pressure increase at the calipers when the driver is not pushing the brake pedal. In addition, during ABS interventions, it could happen a contemporary request of pressure decrease at different wheels calipers. In these conditions, it is necessary a consistent flow rate to guarantee efficacious pressure reductions for all the calipers;
4. ESP motor pump should have a reduced inertia not to provoke delays and disturbances during pressure modulation especially when ESP is active and the driver is not pushing the brake pedal.

Targets 1-2 and 3-4 can be in contrast during the design of the ESP unit. They have to be considered also by the designer of ESP control algorithm. Through this process of experimental tests, it is possible to find out the first approximation parameters for evaluating the hydraulic components of the ESP hydraulic unit. The development of the control algorithm, which is often performed by the car manufacturer, has to take in account the delays and the approximations due to the actuation system. They are fundamental to determine the performance of the ESP unit, which will be demonstrated in the next paper about this activity. The same procedure of characterization of the hydraulic unit was carried out also for a prototype of Electro-Hydraulic Brake unit. The main results obtained are described in [7] and [8]. The base brake function for an EHB system was implemented on a HIL test bench. For an EHB system, the main hydraulic components having an influence on vehicle dynamics are the valves connecting the brake calipers to the pressure accumulator and the brake fluid reservoir. In any case, specifications 1-4 are much better satisfied than for conventional ESP units, due to the optimized valves for the exploitation of the base brake function, which requests a very precise modulation without pressure oscillations, and due to the presence of the high pressure accumulator, which guarantees very high dynamics (1000 bar/s) during the pressure increase phases, even when the driver is not pressing the brake pedal. At the moment, EHB can be considered the optimal solution for ESP actuation.

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