An ABS control logic based on wheel force measurement

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Abstract *The paper presents an ABS control logic based on the measurement of the longitudinal forces at the hub bearings. The availability of force information allows to design a logic that does not rely on the estimation of the tyre-road friction coefficient, since it continuously tries to exploit the maximum longitudinal tyre force.*

The logic is designed by means of computer simulation and then tested on a specific hardware in the loop (HIL) test bench: the experimental results confirm that measured wheel force can lead to a significant improvement of the ABS performances in terms of stopping distance also in presence of road with variable friction coefficient.

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1 Introduction

Anti Lock Braking systems are a highly effective means of preventing skidding related accidents (1) by taking maximum advantage of the frictional coefficient between road and tyre.

In order to understand how an ABS works, it is necessary to refer to the interaction between tyre and road. As a wheel rotates, there are always small slippages in the part of the tyre contacting the ground: a 100% slip ratio occurs when the wheel is locked, while the slip annihilates when the wheel rotates ideally on the ground. It is worth underlining that the maximum adhesion between road and tyre does not occur at a fixed value of the slip rate, but it lies in the range $5 \div 30\%$ ((2)), depending mainly on friction coefficient and vertical load.

Therefore ABS logic attempts to maximise adhesion between tyres and road by trying to keep the slippage in a limited range around the maximum value (3). One of the problems is the estimation of the friction coefficient, since not only the longitudinal force maximum depends on it, but also the slip value correspondent to the maximum is linked to this parameter (Fig. 1, (2)).

To control the system, traditional ABS logic (3) uses the wheel speed sensors to compute a number of control variables:

- wheel acceleration
- reference vehicle speed
- slip threshold

There is a significant amount of research in tyre-road friction, mostly focused on the estimation of the tyre-road friction coefficient in order to develop ABS control logics. Yi *et al.* (4) propose an observer-based control logic, using a LuGre friction model to estimate the tyre-road friction, while Solimon *et al.* (5) present another observed-based control logic that requires an electro-mechanic actuator to continuously adjust the braking force in order to maintain the slip at an arbitrary set point. An optimised braking system (A-ABS) able to recognise different type of road surfaces is described by Rattasiri *et al.* (6): the algorithm is based on discriminative hierarchical evolutionary fuzzy system (D-HEFS).

The estimations of tyre-road friction are often based on Kalman filter. Ray (7) proposes a Kalman-Bucy filtering (EKBF) to directly estimate the friction coefficient μ and to determine the tyre-road forces and slip σ which are then used to compute μ by means of an analytical tyre model. This approach requires only sensors usually available on normal production vehicles. The same algorithm can be used also to detect μ-split conditions. A Kalman filter is also used by Gustafsson (8), but the system proposed works only for two-wheel-driven (2WD) vehicles during normal cruise, not during braking phase: the filter selects the friction coefficient from a set of two or three values. The estimation is performed before braking, but the algorithm is equipped with a change detector in order to react in case a friction change should occur.

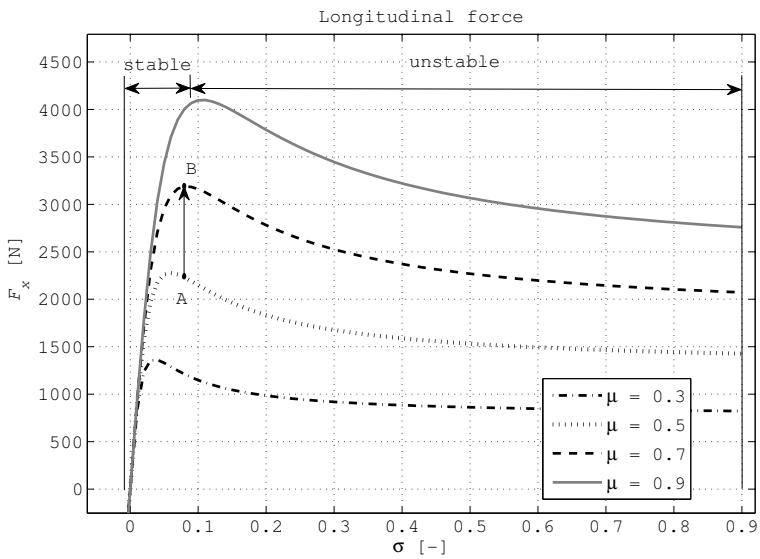

Figure 1: Longitudinal forces vs. slip for different tyre-road friction coefficients

Müller *et al.* (9) base the estimation on the hypothesis that the initial slope of the μ-slip curve is related to the maximum available friction coefficient; the algorithm presented is able to classify the road surface as dry or slippery.

To overcome the estimation of tyre-road coefficient, many algorithms are based on slip control; the disadvantage of these systems is that slip cannot be computed correctly, since the vehicle speed is just estimated and not measured. A solution is proposed by Choi (10) controlling the rear wheels to make them follow a cycle around the peak friction slip point and thus defining the optimal slip; the front wheels are also continuously controlled to track the reference velocity maintaining the front calliper pressure almost constant.

Nouillant *et al.* (11) compute the reference slip using feedforward and feedback controlled servo-valves, whereas Harifi *et al.* (12) present a sliding mode slip control provided with an integral switching surface to reduce chattering effect. An alternative solution to this problem is given by Wu and Shih (13), who use a pulse width modulation (PWM) of the calliper pressure.

Other ABS systems use the wheels decelerations as controlled parameters. Obviously the velocity variations depend on the behaviour of the wheels and are not directly linked to the friction coefficient; however they represent a widely used solution because they can be easily computed from the wheel speed sensors. Pasillas-Lépine (14) and Ait-Hammouda and Pasillas-Lépine (15) suggest two control logics: the first one works only on constant tyre-road friction surfaces, while the second can deal also with friction changes. Ait-Hammouda and Pasillas-Lépine (15) propose an algorithm able to overtake a drawback due to the independent control of the wheels: in fact, if the torques applied on the front

wheels are not equal, also the longitudinal forces are different, thus generating a moment that propagates along the steering column. The algorithm is able to synchronise the front wheels, while the rear wheels are independently controlled, thus allowing a more reliable estimation of the vehicle speed.

From the analysis of the literature it appears that one of the most challenging problems is to estimate tyre-road friction so that the brake forces can be fully exploited at their maximum. A typical situation in which traditional logics response is not satisfactory is represented by the passage from low to high adherence conditions: the ECU cannot identify with sufficient promptness the change of friction and consequently the wheels do not lock but the vehicle is under braked (6; 10; 11).

The availability of measured wheel forces could provide an effective method to solve the problem, without need of directly estimating friction; on the basis of innovative Load Sensing Hub Bearing Units (LS-HBU) developed by SKF and described in (16; 17), this paper presents a control algorithm which allows enhanced ABS performances in all braking conditions, with significant improvements in case of surfaces with variable friction (μ jump).

It is of interest observing that other authors (e.g., Botero *et al.* (18)) present an ABS logic based on sensed forces and moments, using a sliding-mode controller; their work aims at the online reconstruction of the actual force-slip curve, so that they track the optimal slip.

Moreover, Deur *et al.* (19) developed a strategy non-reliant on wheel slip information aiming at enhancing the traction control performances: their approach is based on the generation of a sawtooth excitation in order to adapt the slip for maximum tyre traction performance.

The present research is focused in exploiting the maximum longitudinal forces during braking, using wheel deceleration and slip to prevent wheel locking conditions.

Finally, some experimental results from a HIL test bench running both new control strategy and normal production logic are presented; from the comparison it appears that the proposed algorithm shows better performances in most case of straight braking on roads with different friction coefficients.

2 New ABS control logic

The idea at the basis of the control logic lies in a procedure to evaluate the maximum of longitudinal forces F_x without needing to estimate the value of the longitudinal slip σ at which the maximum $F_{\max}$ is located. If this condition can be identified, then it is possible to develop an algorithm to control the pressure in order to give the desired value of the wheel force, without the need of identifying the actual value of the tyre-road friction or of the correspondent longitudinal slip. *Moreover this procedure is independent of load transfer, since it aims only at estimating the maximum value of the longitudinal force, regardless of vertical load and tyre-road friction value* It is worth noting that the maximum of the force is relative to force-slip curve $F_x(\sigma)$; anyway, since during the initial phase

of braking the modulus of slip $\sigma(t)$ grows with time t (see e.g. Fig. 12) due to the increase of the TMC pressure and the consequent growth of the braking torque, also the longitudinal force $F_x(t)$ is a growing function of time, up to the point it reaches a maximum.

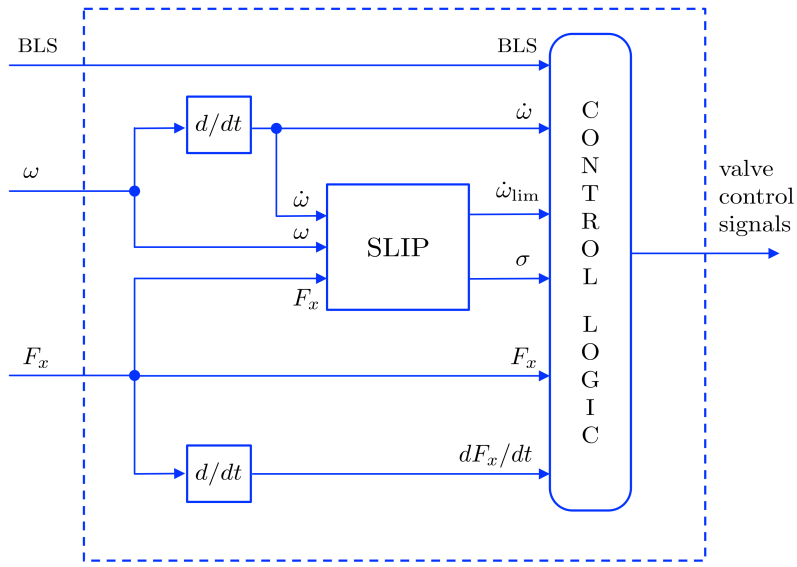

Figure 2: Schematic diagram of the ABS block

The proposed algorithm, described in Fig. 2-5, is as follows:

1. During the initial phase of the braking, after the brake light signal (BLS) is activated, the estimate of the maximum of the tyre longitudinal force is based on a well known mathematical procedure: the maximum of a function corresponds to the point where its first derivative is null. In fact, when the gradient is equal to zero the function is in a maximum or a minimum point. It is also well known that in case of a wheel undergoing braking, in the initial phase longitudinal forces increase with pressure, up to the point where the derivative of the longitudinal force reaches zero. Hence, to anticipate the identification of this maximum, it is possible to analyse the derivative of the longitudinal forces, which progressively decreases, vanishes and then becomes negative. The solution lies in setting a positive threshold $(dF_x/dt)^*$ for the derivative of the longitudinal force F_x provided by the LS-HBU, and to consider that condition as the start signal to activate the pressure hold phase:

$$\frac{dF_x}{dt} < \left(\frac{dF_x}{dt}\right)^\star \quad \text{and} \quad \frac{dF_x}{dt} > 0 \quad \text{at} \quad t = t^\star. \tag{1}$$

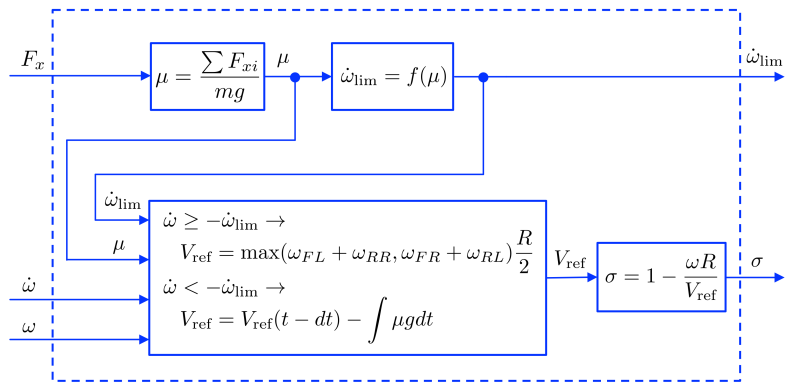

Figure 3: Slip evaluation block diagram

The value of the longitudinal force read at the time $t^\star$ when the previous condition is satisfied, is considered the maximum available force $F_{\max}$:

$$F_{\max} \simeq F_x\left(t^\star\right); \tag{2}$$

this value is stored until the next maximum recognition (Fig. 5, left diagram).

Since it is not possible to directly use the value of the longitudinal force provided by the sensors, it is necessary to perform some filtering: in the present work it is accomplished with a moving average technique, considering the last eight values of force F_x; the sampling frequency is the same as that of the model, i.e. 2 ms. The numerical derivatives are computed with suitable filtering in order to reduce oscillations.

2. Aiming at the best exploitation of the tyre characteristics, after the identification of the maximum of the longitudinal force as described in the first step, the pressure is kept constant (Fig. 4) until one of the following conditions is reached:

$$\dot{\omega} < \dot{\omega}_{\text{lim}} \quad \text{or} \quad \sigma > \sigma_{\text{lim}} \tag{3}$$

where $\sigma = 1 - \omega R/V_{\text{ref}}$ is the tyre longitudinal slip, R is wheel rolling radius (constant) and V_{ref} is the vehicle speed as estimated by the ABS control logic (Fig. 3). For the sake of simplicity, both $\dot{\omega}$ and $\dot{\omega}_{\text{lim}}$ represent the modulus of the acceleration (positive values).

Thresholds $\dot{\omega}_{\text{lim}}$ and σ_{lim} represent the maximum acceptable values of wheel deceleration and slip. Regarding the first one, it has been observed that its optimal value, i.e. the value granting the shortest stop distance, increases with friction coefficient (Fig. 6). In practice, the optimal value

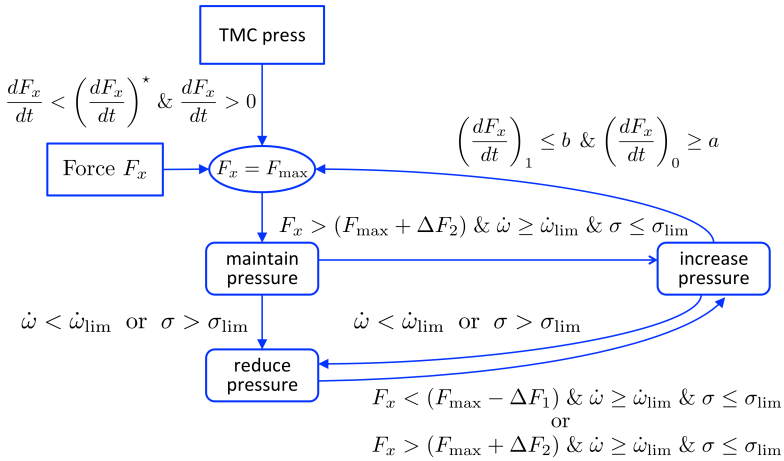

Figure 4: Flow diagram of the ABS control logic

of $\dot{\omega}_{\lim}$ correspondent to different values of road-tyre friction coefficient μ is registered through suitable numerical tests; then the curve visible in Fig. 6 is obtained interpolating the points through the following fourth order polynomial:

$$\dot{\omega}_{\lim} = k\left(c_1\mu^4 + c_2\mu^3 + c_3\mu^2 + c_4\mu + c_5\right), \tag{4}$$

where c_i are the interpolating coefficients and k is a scale parameter, used to adapt $\dot{\omega}_{\lim}$ to different vehicles without changing the function shape, which it is related to the tyre-road behaviour and almost independent of the car. Threshold $\dot{\omega}_{\lim}$ can be inserted into the control logic in the form of a discrete look-up table or can be evaluated by means of the interpolating function (4).

Threshold $\sigma_{\lim}$ is simply a fixed value.

The algorithm allowing to estimate the slip for each wheel (Fig. 3) is based on the definition of slip. The vehicle reference speed V_{ref} is chosen between the maximum of the faster car diagonal and the velocity computed in function of its preceding value: the choice is taken on the basis of the wheels deceleration (see, e.g., (3)). Moreover, it is worth observing that the friction coefficient, used to compute both vehicle speed V_{ref} and limit threshold $\dot{\omega}_{\lim}$ in equation (4), is estimated through the longitudinal forces read by the LS-HBU, as visible in the block diagram of Fig. 3.

In case of a friction coefficient variation after the maximum recognition, thus leading to a longitudinal force reduction, condition (3) causes the logic to jump immediately to the following point, without keeping the pressure constant.

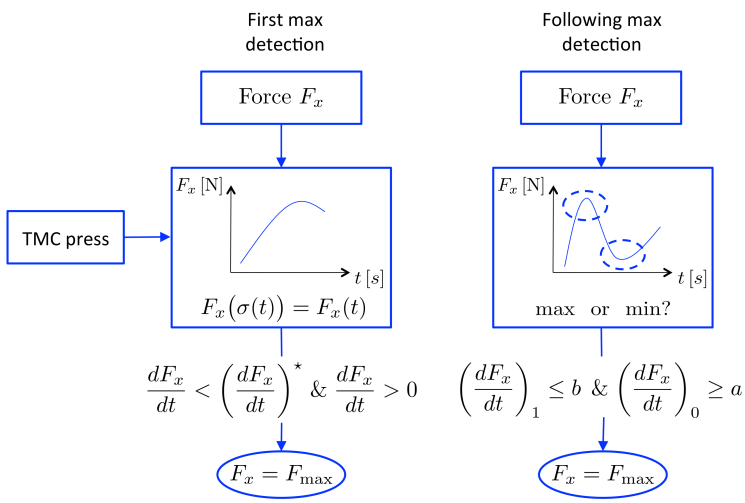

Figure 5: Flow diagram: detail of the identification of the force maxima

3. When one of the previous conditions (3) is verified, the wheel tends to lock; hence it is necessary to reduce the pressure at the calliper. This phase can be accomplished through a step or a continuous pressure reduction by suitable control of ABS valves (Fig. 4).

   It is of interest noting that when a pressure reduction is required at the front wheels, in order to prevent the braking plant from clogging up, if also the rear wheels are in a reduction phase, then they are switched to a pressure hold status.

4. The pressure reduction phase ends when one of the following conditions is satisfied:

$$F_x < (F_{\max} - \Delta F_1) \quad \& \quad \dot{\omega} \geq \dot{\omega}_{\lim} \quad \& \quad \sigma \leq \sigma_{\lim} \tag{5}$$

$$F_x > (F_{\max} + \Delta F_2) \quad \& \quad \dot{\omega} \geq \dot{\omega}_{\lim} \quad \& \quad \sigma \leq \sigma_{\lim} \tag{6}$$

   where ΔF_1 and ΔF_2 are two tuneable thresholds, as explained in the next paragraph (Fig. 4). The first condition is used to increase pressure in order to obtain a force as close as possible to the maximum, avoiding an excessive pressure drop. In fact, after some pressure reduction steps, if wheel deceleration and slip are both into acceptable limits, it is advisable to increase the pressure, since the possibility for the wheel to lock again is negligible. Condition (6) occurs when the available friction increases: the tyre longitudinal force grows almost immediately as a consequence of the friction rise, before the variations due to load longitudinal transfer take place. In fact, under constant normal load, a friction coefficient variation is the only parameter that can justify a sudden increase of the longitudinal force.

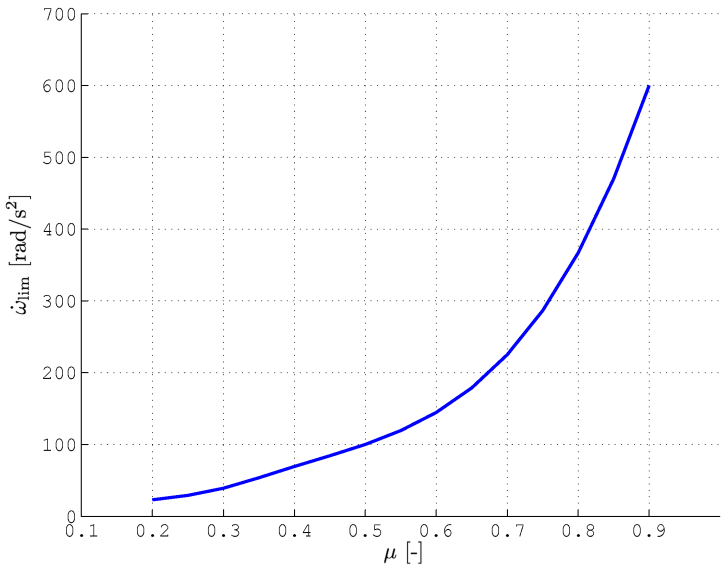

Figure 6: Interpolating function for threshold $\dot{\omega}_{\text{lim}}$

A suitable value for threshold ΔF_2 can be provided analysing the tyre force-slip curve. Figure 1 shows different curves for various friction coefficient with the same normal load and ΔF_2 is the difference between the maxima: when a sudden increase of friction coefficient takes place, the operating point shifts suddenly, e.g., from A to B. An appropriate value can be chosen comparing curves with a friction coefficient difference equal to 0.1. The pressure is then increased by steps in order to avoid a sudden wheel lock. The choice of ΔF_1 has no physical basis, but its value is tuned performing numerical tests. The idea behind it is that after the pressure reduction due to one of conditions (3), in absence of friction coefficient reduction, it is advisable to increase the pressure again as soon as possible, aiming at reducing the braking distance. It was observed that if only conditions $\sigma \leq \sigma_{\text{lim}}$ and $\dot{\omega} \geq \dot{\omega}_{\text{lim}}$ were used to increase pressure again, a large number of pressure and force oscillations would occur, probably because the activation signals are too fast with respect to the wheel-road dynamics. Hence, it proves advisable to let the longitudinal forces drop of an amount ΔF_1 before the next valve intervention, thus granting a consistent margin before the maximum force is reached again.

5. During the pressure increase phase, a maximum for the tyre longitudinal force will be reached again; its value can be identified through the following procedure (Fig. 5, right diagram). First define two different values of the force derivatives dF_x/dt, e.g. a and b, with $a > b$; obviously, in proximity of a minimum b comes before a, while close to a maximum a anticipates b

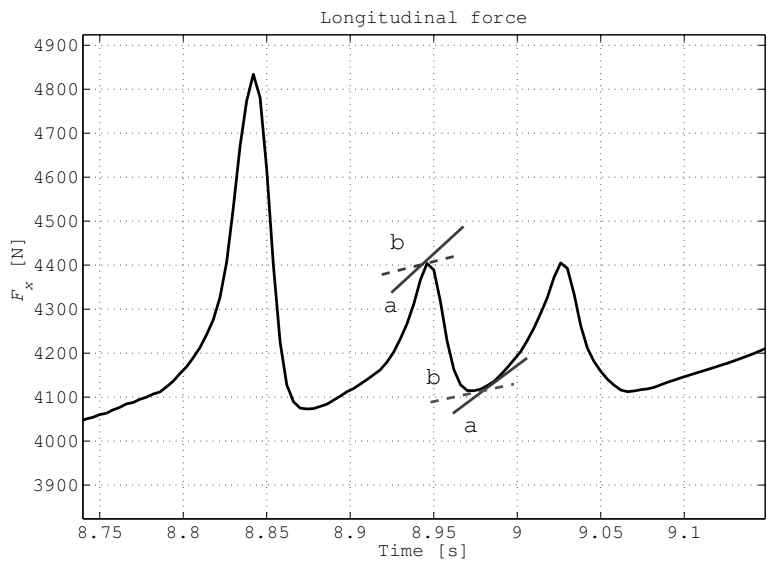

Figure 7: Longitudinal force vs. time: technique to identify local maxima and minima

(Fig. 7). Hence, comparing the time when the computed derivative of the force reaches values a and b, it is possible to distinguish local minima and maxima. The condition for a local maximum is

$$\left(\frac{dF_x}{dt}\right)_1 \leq b \quad \text{and} \quad \left(\frac{dF_x}{dt}\right)_0 \geq a \tag{7}$$

where $(dF_x/dt)_1 = dF_x/dt$ at the actual time $t = t_1$ and $(dF_x/dt)_0 = dF_x/dt$ at previous time $t_0 = t_1 - dt$. Obviously, the derivatives a and b are two tuneable parameters used for the identification of the maxima of the F_x curve. Afterwards the cycle is repeated from step 2.

6. In order to enhance the performance of the logic in terms of its ability to quickly respond to possible changes of tyre-road friction, the two conditions already described in points 2 and 4 give satisfactory results. In particular condition (6) allows the direct passage from point 2 to point 4, i.e., from pressure hold to pressure increase phase, in case a friction increase is detected (Fig. 4). On the contrary, if one of the (3) is satisfied then, during phase 4, the logic can jump directly to phase 3, i.e. to pressure reduction, even if no maximum value of the force has been identified; this condition proves useful in order to prevent the wheel from locking when a sudden reduction of the friction coefficient takes place.

3 ABS HIL experimental bench

3.1 Vehicle model

The proposed ABS logic is tested on a vehicle model having 14 degrees of freedom: six for the car body and two for each wheel (rotation and vertical travel); suspensions kinematics and compliances (K&C) and dynamic behaviour are fully described. The hydraulic and electric characteristics of the ECU, as well as the booster and tandem master cylinder dynamic behaviour, are determined through experiments on the test bench described in the following section, thus allowing to have an accurate model of the braking system. The experimental results of the tests carried on ECU valves, motor pump unit and booster allow to describe the dynamic behaviour of the entire braking system with suitable transfer functions. Moreover also the characteristics of LS-HBU sensors (e.g. 200 Hz and accuracy about 5% of full scale as reported in (17)) are considered in order to completely model the hardware dynamics.

The model is implemented in Simulink® with a block oriented approach and is validated for a european middle sedan, through comparison with on-road tests. The code for the ABS control logic (Fig. 4) is written with Stateflow®.

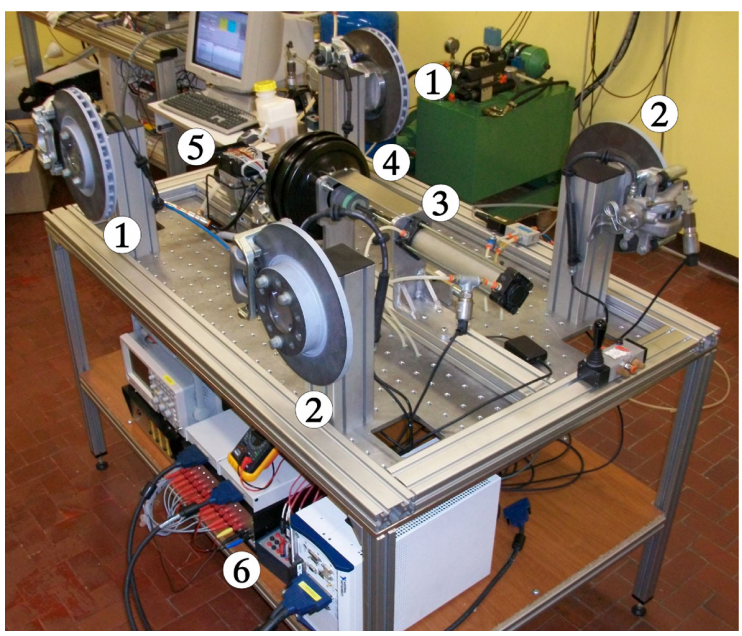

Figure 8: Brake system test bench: front (1) and rear (2) disks, pneumatic actuator (3), booster and master cylinder (4), ABS hydraulic control unit (5) and PXI with acquisition boards and power unit (6)

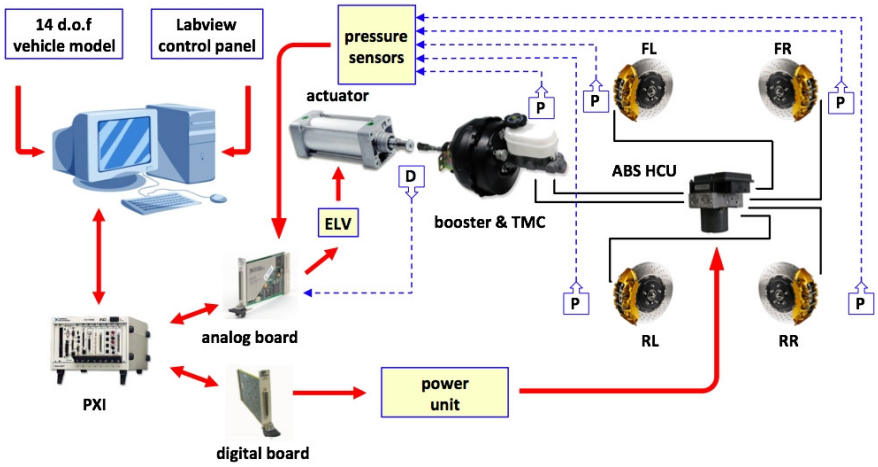

Figure 9: Layout of the experimental test bench, showing connections between brake plant and vehicle model (D: displacement and speed transducer, P: pressure transducers)

3.2 Test bench

Test experimentation is based on the use of a hardware in the loop (HIL) test bench (Fig. 8 and 9). The test rig is designed to mount the original components of the brake system of the vehicle used to validate the mathematical model. It consists of the entire brake system, comprehensive of vacuum booster, tandem master cylinder (TMC), electronic stability control unit, all rigid and flexible pipes mounted on the real car and four wheel discs with brake callipers; the discs are fixed to the bench. A controlled actuator pushes the booster input rod, allowing simulating both semi-stationary and panic brake manoeuvres. A displacement sensor is mounted on the actuator to measure the feedback of the PID controller, which can work as displacement, speed or pressure controller. Vacuum levels are measured and maintained constant inside the booster. Dedicated sensors measure the pressure at wheel brake cylinders and at TMC. A modified version of vehicle model described above runs in real time on a dedicated platform, equipped with data acquisition and signal generation boards (Fig. 9) to allow data and signal dialog between the software vehicle model and the real brake components. In particular, during the tests on the bench, the Simulink® model sends signals to drive the actuator acting on the booster rod and to activate the ECU valves according to the ABS control logic; moreover it receives the measured values of TMC and calliper pressures, which are used to estimate the braking force at the wheels.

The original ECU and power electronics can be bypassed in order to test alternative control logics. The solenoids devoted to the actuation of the 12 electro-valves lay under the circuit of the ECU; the circuit is removed and substituted with a plate containing only the welding spots to the solenoid pins. The

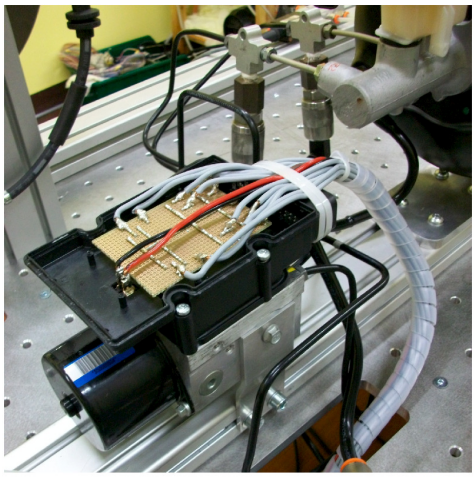

Figure 10: Open ESC unit with wirings to directly control the electro-valves and the motor pump

plate in Fig. 10 shows the connections with the electro-valves and a couple of cables devoted to transmit the power signal to the motor pump.

Since the real time platform signals need to be turned into power signals, a dedicated box, equipped with high performance solid-state relays, substitutes the motor and valves relays.

The bench allows to test the modified control logic based on wheel forces: during the experiments, the ABS inputs are the simulated values of the wheel speeds and the longitudinal values of the forces exchanged at the wheel hubs. The outputs of the logic are the signal commands to activate or de-activate the motor pump and the electro-valves of the hydraulic circuit. Obviously it is possible to record a number of additional signals that are not available on commercial vehicles, e.g. the pressure at the four callipers and the force on the brake pedal, thus allowing to directly monitor the system states.

The bench also permits to run test using the ABS logic currently available on cars, also known as normal production (NP) logic, aiming at evaluating the improvements of new control strategies. In order to test NP ABS it is necessary to supply the ECU with the reconstructed values of the wheel speeds, light brake signal (LBS) and other information usually available on the vehicle CAN that are requested by the control unit to work properly, without fault messages preventing NP logic to be actuated.

4 Results

To test the performances of the new ABS logic many tests were run, both in case of standard or panic braking, for different values of the friction coefficient

between tyres and ground.

Particular attention has been devoted to verify the performances during μ jump experiments, i.e. when a sudden transition high-low friction (or vice-versa) takes place, since these conditions are particularly critical for driving safety.

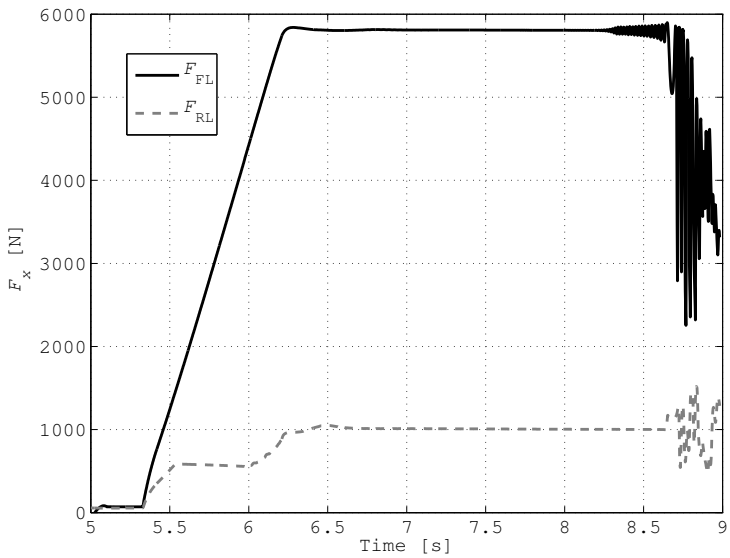

Figure 11: Longitudinal forces at front and rear left wheels in high adherence ($\mu = 0.9$)

4.1 High friction

In case of high adherence ($\mu = 0.9$), the new ABS control logic is able to detect the presence of a maximum of the longitudinal force F_x (see Fig. 11, showing the left side tyre forces) and to regulate pressure in order to hold this value until $t = 8.5$ s, when some wheel oscillations occur and then a pressure reduction is needed. The following oscillations are due to the friction model that is used to model the pad-disc contact: it is a stiff system, due to the nonlinear behaviour of the friction coefficient close to null relative speed. The oscillations are particularly evident in case of high friction, due to the high values of the forces exchanged between tyre and road. Obviously, the force exchanged by the rear wheel is lower than the front one, both for the load transfer and the EBD intervention; it is of interest noting that the EBD logic developed at the Politecnico di Torino (PdT) is different from the NP logic.

Figure 12 plots the values of the slip of the left wheels for the same test of Fig. 11: it can be observed that both wheels present a limited value of the slip (less than 10%) up to final part of the braking, when the ABS control logic is de-activated because the estimated vehicle velocity has fallen below the lower

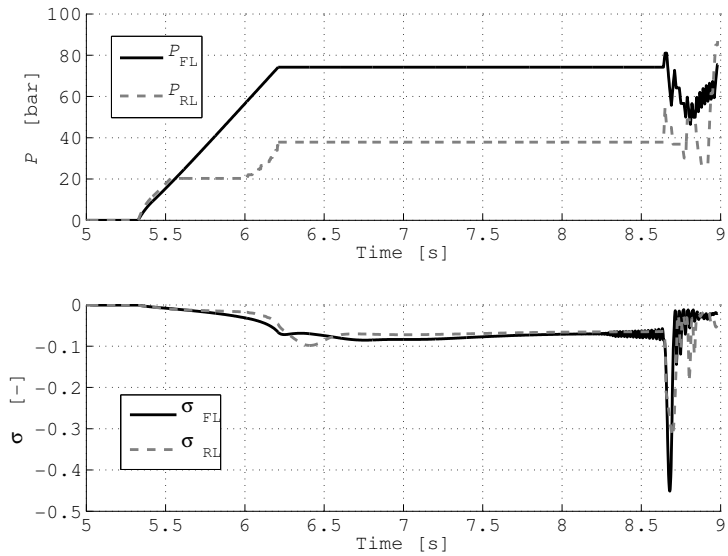

Figure 12: Pressure and slip at front and rear left wheels in high adherence ($\mu = 0.9$)

speed threshold. The limited slip value also proves that the force-based control logic is effective in keeping the tyre in the stable region of the force-slip curve; moreover it grants that the slip estimation performed by the control logic gives reliable results, since the wheels do not approach the locking condition.

Finally, also the effects of disturbances on the force signals have been investigated, by introducing noise of amplitude from 1% to 10% of the sensors full scale (FS); during the simulations, the FS value has been set equal to 15 kN. The analysis of the results allows to state that the control logic proves satisfactory in presence of disturbances up to 5% FS (see, e.g., Fig. 13): under these conditions, the stop distance obtained with the PdT logic is still smaller than that achieved with NP logic. On the contrary, it fails to identify the longitudinal force maxima above 8% FS.

Comparing the results shown in Fig. 12 and 13, the plot in absence of noise appears smoother, thus indicating that the control logic succeeds in maintaining a stable working condition, requiring less regulations. The performance is slightly worse in presence of noise: anyway, though the brake force at the front wheels is lower, the rear wheels experience a greater brake force, thus partially compensating the front reduction.

We have only explored some limits of the logic, without investigating thoroughly the effects of noise, since this analysis is beyond the aim of this paper.

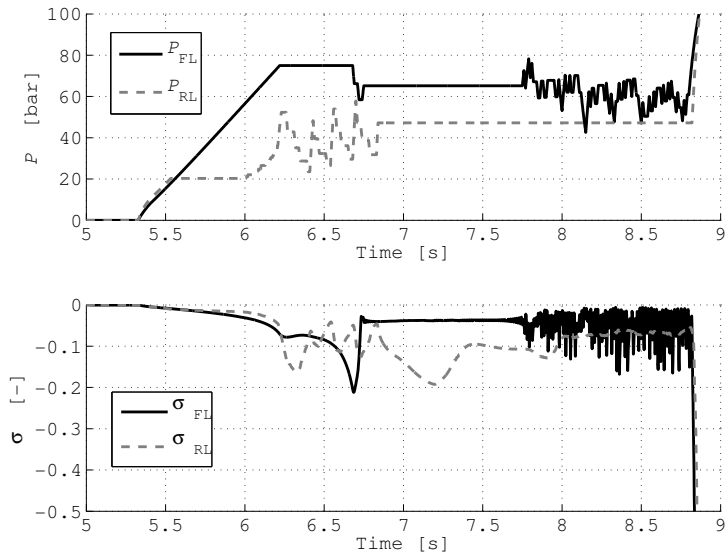

Figure 13: Pressure and slip at front and rear left wheels in high adherence ($\mu = 0.9$), in presence of disturbances on the force signals (5% full scale)

4.2 High-low friction jump

In case of sudden transition from high to low adherence, the control logic reacts very quickly, thus limiting the time in which the wheels remain in the unstable region of the force-slip characteristic.

The callipers pressure time history is visible in Fig. 14: at the beginning the friction coefficient between road and tyre is 0.9 so the pressure reaches the maximum value. At second 7 the friction drops to 0.3: this abrupt change causes the wheels to oscillate and then the system has to control this tyre instability (the tyre starts working on the unstable branch of the force-slip curve) by varying the pressure continuously. The pressures will then grow up to the TMC values at the end of the braking action (i.e., after 13 s), when the vehicle has come to a full stop. A similar situation takes place on the vehicle right side.

These results can be compared with those due to normal production logic, as reported in Fig. 15. The sudden friction transition causes an oscillatory behaviour of the pressures, but the new logic developped at the Politecnico di Torino (PdT) proves more effective because it allows to halt the vehicle in a shorter time (about 2.5 s in advance) and in a shorter distance (see Table 1).

4.3 Low-high friction jump

In most cases PdT logic provides better results in terms of stop distances; anyway, for the sake of completeness, we also present a case in which the new logic

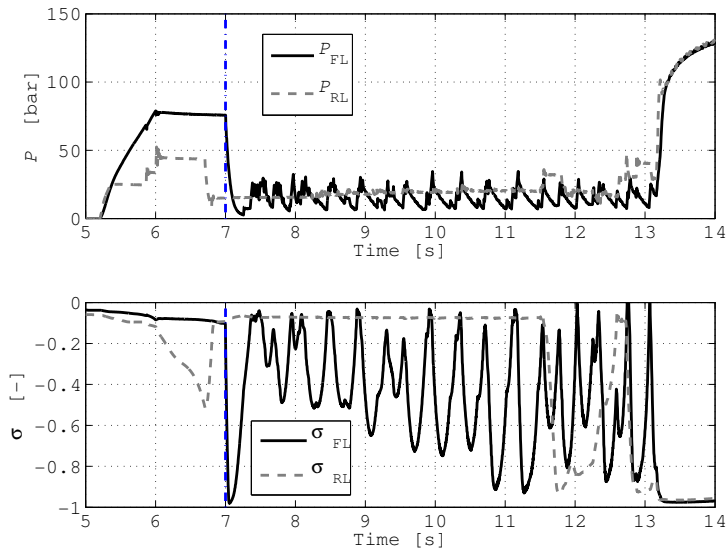

Figure 14: Pressure and slip at front and rear callipers in μ jump conditions (0.9÷0.3) - PdT logic

gives performance worser than the traditional logic, i.e. , when the friction coefficient between road and tyre changes from 0.3 to 0.9 at time $t = 7$ s (Fig. 16 and 17).

It is evident that the system recognises this variation and immediately increases the calliper pressure, thus allowing to exploit the maximum available friction. The front wheels pressure is then kept constant until, at time $t = 9.3$ s, the ABS has to intervene, with a pressure reduction followed by repeated intervention; on the contrary the rear wheels pressure is kept constant up to 10 s.

From the comparison of the results it is evident that the friction change is recognised more promptly by the PdT logic; nevertheless the global performance is better for the NP logic, probably due to the fact that it exploits more effectively the braking at the rear wheels, where the pressure starts to grow soon after 7 s.

A possible explanation of the fact that the rear wheels are under braked is that, in order to avoid oil plant saturation, the PdT logic does not allow the rear wheels to discharge pressure when front wheels are already cutting pressure. Consequently the logic is more conservative for the rear wheels, for which the activation parameters have been set to give a not "aggressive" behaviour. Also the NP logic suffers from a similar problem, but in a reduced way.

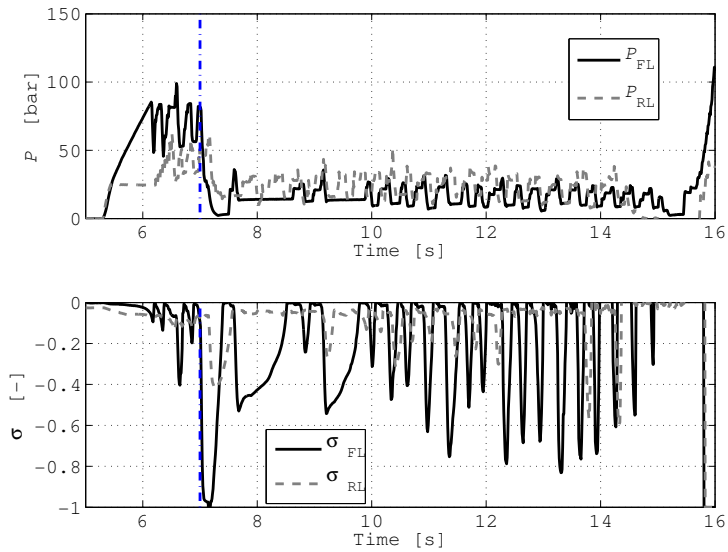

Figure 15: Pressure and slip at front and rear callipers in μ jump conditions (0.9÷0.3) - normal production logic

4.4 Double friction change

The control logic has also been tested in highly dynamic conditions, i.e., in presence of repeated variations of the road-tyre adherence. Fig. 18 shows the time history of the left wheel pressures when braking on a road with a low friction zone (about 20 m, correspondent to a time interval of 1 s, between 7 and 8 seconds).

It is possible to observe that the double friction change is immediately identified by the algorithm, which reacts regulating the pressure in all the callipers. Also in this case, the rear wheel pressure is kept almost constant after the second friction variation; a refinement of some parameters and thresholds could lead to exploit more successfully the rear brakes.

4.5 Comparison with normal production (NP) logic

To evaluate the performances of the proposed logic, the distance required for the vehicle to come to a complete halt has been chosen as the most significant parameter.

The tests were run according to the methodologies currently adopted in automotive world: with the vehicle travelling at 110 km/h, the driver applies full braking, and the stopping distance is measured starting from the instant in which the car crosses 100 km/h.

The results from tests for different values of tyre-road friction are reported in Tab. 1, showing improvements in most of the experiments.

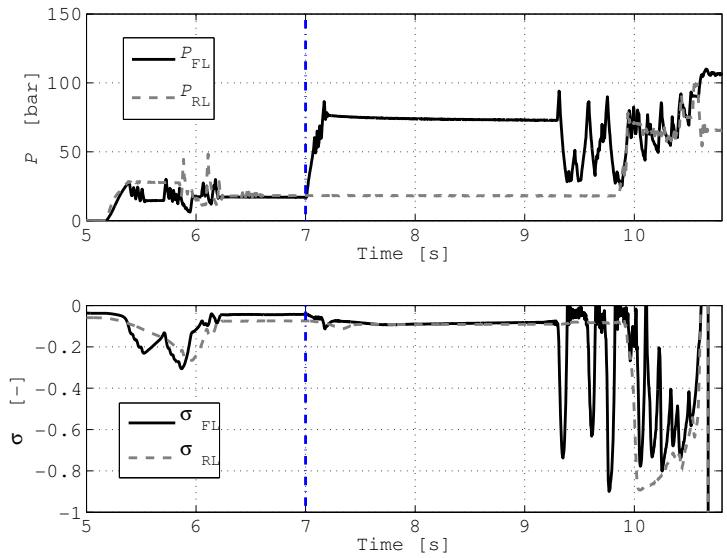

Figure 16: Pressure and slip at front and rear left callipers in μ jump conditions (0.3÷0.9) - PdT logic

5 Conclusions

Commercial ABS systems are based on the control of wheel deceleration or on tyre slip; unfortunately both the methods present drawbacks. In fact, deceleration is not related only to the tyre performance and hence it is not possible to have the tyre working at the top of longitudinal force vs. slip curve. Theoretically, the slip control logic could reach this goal; however the problem is that the slip is not measured but computed using the estimated longitudinal velocity of vehicle. Thus such estimate is affected by errors and so is the computed slip. Furthermore the optimal value of slip depends on tyre, vertical load and road, which are usually unknowns.

The proposed control logic will allow overcoming the aforementioned problems, and hence optimising the braking phase in any tyre or road condition. Moreover, thanks to the "dynamic" estimate of the maximum value of the longitudinal force, the algorithm can adapt to the effective tyre-road friction coefficient without needing to directly evaluate such coefficient. The effects of disturbances on measured force signals have been investigated, showing the robustness of the algorithm if the noise level does not exceed 5% of the sensors full scale.

Finally, in terms of stop distance, this adaptive algorithm is able to give better performances than those of the existing ABS units in most of the examined cases, on road with different adherence conditions, even in case of surfaces with sudden friction changes (e.g. transition from dry to icy asphalt).

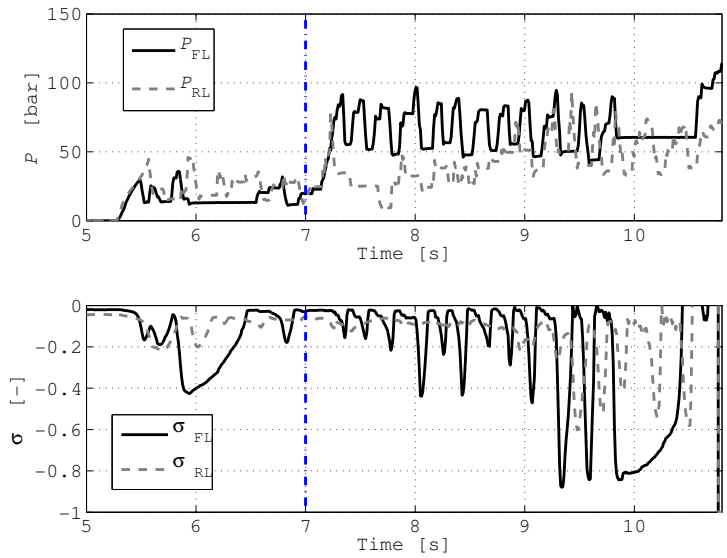

Figure 17: Pressure and slip at front and rear callipers in μ jump conditions (0.3÷0.9) - normal production logic

The additional conditions on tyre acceleration and longitudinal slip allow a greater robustness of the control logic, so that it is possible to quickly and safely identify a wheel lock pending situation. Consequently, performances are likely to be improved in all road conditions by tuning the logic parameters (σ_{lim}, ΔF_1, ...) aiming at obtaining an aggressive braking action but avoiding wheel lock conditions.

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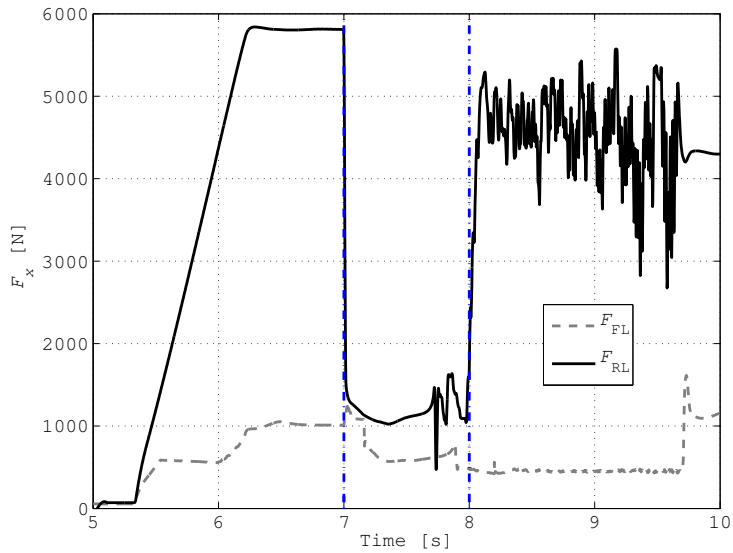

Figure 18: Longitudinal forces at front and rear left wheels in heavily dynamic environment: $\mu = 0.9$ up to $t = 7$ s, then $\mu = 0.3$ up to $t = 8$ s, finally $\mu = 0.9$ again, until the end of the test

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Table 1: Stopping distances with commercial normal production (NP) and force based (Politecnico di Torino - PdT) control logic

road friction μ [-]	NP [m]	PdT [m]	variation [-]
0,9	50,5	46,7	-7,5%
0,5	90,1	81,7	-9,3%
0,3	153,5	144,8	-5,7%
0,9 $\rightarrow$ 0,5	71,6	66,3	-7,4%
0,9 $\rightarrow$ 0,3	106,1	97,2	-8,4%
0,3 $\rightarrow$ 0,9	65,3	67	2,6%
0,5 $\rightarrow$ 0,9	64,7	59,2	-8,5%

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