Active Systems Design: Hardware-In-the-Loop Simulation

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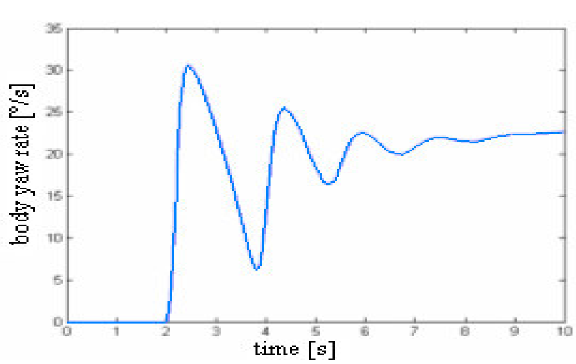
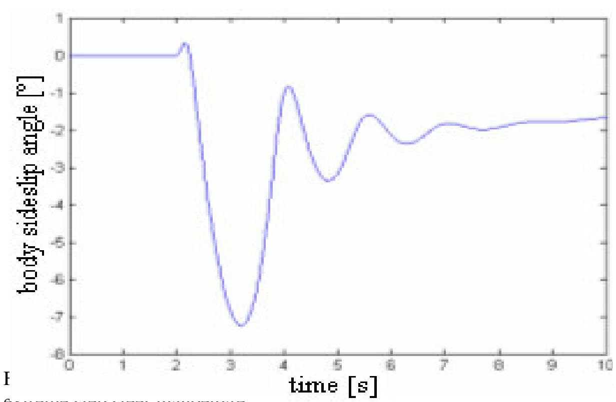
Abstract: - Improving safety and comfort of aeronautical and mechanical products is one of the most important goals of our research activity, with a particular attention on reliability problems. The paper presents the method our research group followed to design active systems improving vehicle dynamics, safety and comfort. It is based on several levels of simulation, depending on the kind of analysis requested by the design process. The highest level consists in Hardware-In-the-Loop (HIL) simulation, with the experimental tests concerning the hardware of the studied system, controlled by the devoted software. The procedure, applicable to both aeronautical and mechanical systems, is described in the paper relating to an automotive "case study" and in particular for the specific application of an Active Roll Control system for a rear torsion bar, actuated by a hydraulic or an electro-mechanical system.

Key-Words: - Optimization, simulation, safety, reliability, comfort, modeling, experimental validation.

1 Introduction: the model

The vehicle model adopted in the activity is quite simple from the point of view of the number of degrees of freedom: it has to be coherent with the dynamic of interest and as simple as possible, not to request an excessive computational power. In the application described in the paper we used aeronautical and mechanical know-how to achieve best results as possible in terms of safety, comfort and reliability of the final product. The vehicle model we used is characterized by 8 degrees of freedom: four for the car body (longitudinal and lateral translations, roll and yaw) and four (rotational) for the wheels. Suspensions were modelled by considering non linearities in roll stiffness and damping coefficient; compliances as a function of lateral forces were taken in account. Tires were modelled through Pacejka magic formula, considering also the variation of relaxation length as a function of tire vertical load and sideslip angle/longitudinal slip. The software model ran at a typical fixed step of integration of 0.001 sec.; it was implemented in Simulink® and experimentally validated through road tests. This vehicle model was used to perform all the typical manoeuvres to characterize handling performance, like ramp steer and step steer (Figures 1 and 2).





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extreme step steer manoeuvre

2 First applications to design the control algorithm of an ARC system

Fixed the above mentioned results as starting points, we began to design the control algorithm we needed. First approximation simulations for the design of an Active Roll Control (ARC) algorithm were performed ([1], [2]); ARC consists in the actuation of suspensions torsion springs, through electro-mechanical or hydraulic devices. The first target consists in modifying the semi-stationary roll characteristic of the vehicle, giving origin to reduced levels of body roll angle in correspondence of middle-low values of lateral acceleration (Figure 3). The second target consists in reducing body yaw rate and sideslip angle oscillations during extreme dynamic manoeuvres, thanks to a closed loop control algorithm based on yaw velocity (Figures 4 and 5). In this phase a system characterized by the only rear active bar was studied. An increased stiffness of the rear bar gives origin to a less understeering behaviour; the opposite happens as a consequence of a rear bar reduced stiffness. This activity is based on the actuation of the rear torsion bar. The available signals for the control algorithm are all those typical of Electronic Stability Program, which is a standard equipment of most European vehicles ([3]). In the first step of simulation activity, the delay due to the actuation system is not considered, and the output of the control algorithm is represented directly by the bar equivalent stiffness (Figure 6). In the following step of this basic level, it is possible to take in account the delays in the actuation process thanks to proper transfer functions.

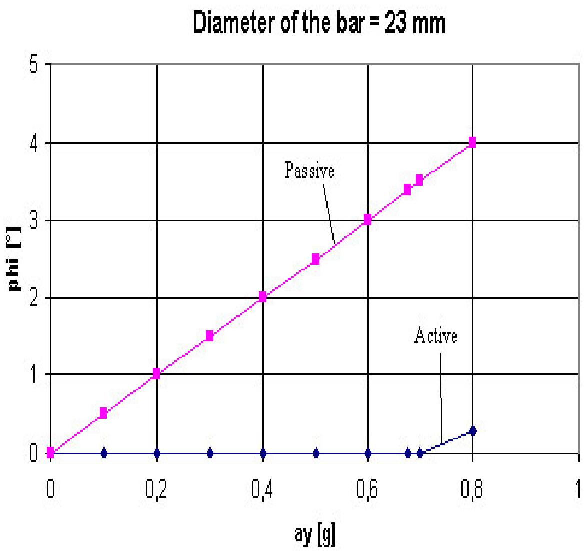
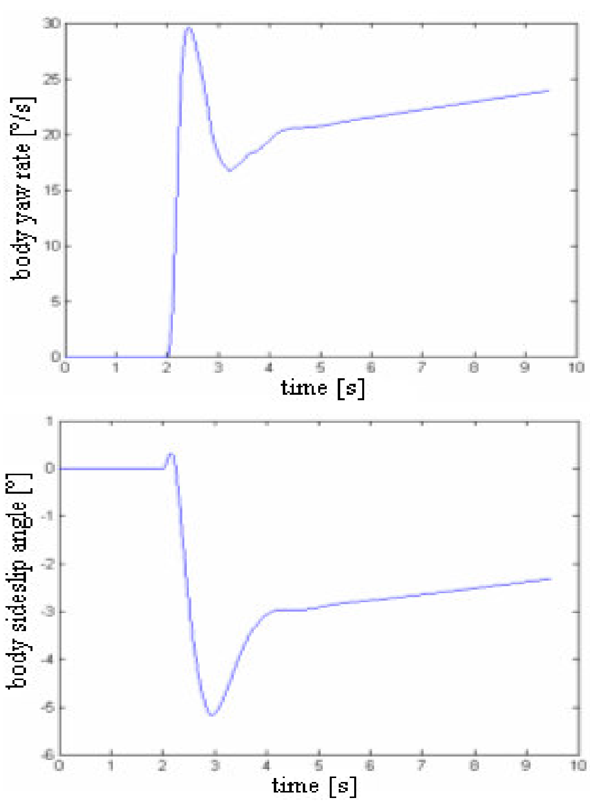


Figure 3 – Body roll angle versus lateral acceleration for a passive and an active vehicle



Figures 4, 5 – First approximation results obtained in simulation by using ARC in the same manoeuvre of Figures 1 and 2

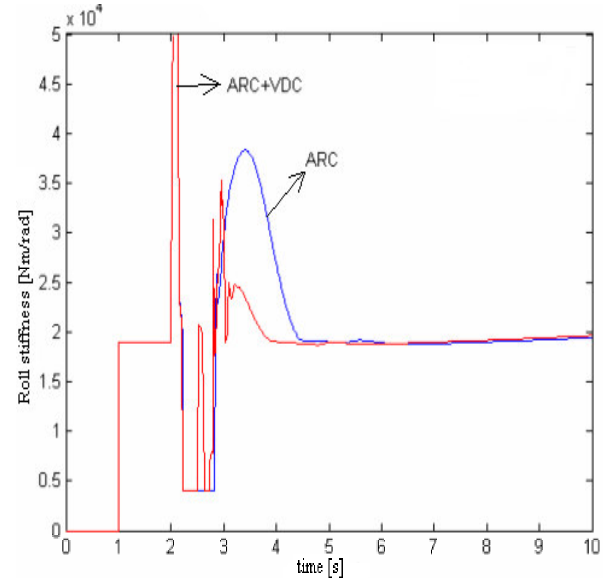


Figure 6 – Example of ARC intervention in a stand alone application and together with Vehicle Dynamics Control (VDC): rear bar equivalent stiffness versus time

Second approximation simulations for the design of an ARC system were performed by implementing a physical model of the actuation system, like that of Figure 7. This first example of actuation system here presented consists of a linear hydraulic actuator instead of the rod connecting the torsion spring to the suspension strut. It was conceived by using the software AMESim®, devoted to hydraulic systems simulation. It permits to take into account valve dynamics, the transition from laminar to turbulent motion of the fluid, friction phenomena inside the actuators, etc…

The hydraulic actuation system consists of a motor pump group (for example it could be the same of the power steering system), a hydraulic accumulator and a system of proportional electro-valves to control the actuator. Anti-roll bars are modelled as torsion springs. The model of the hydraulic circuit as described above is connected to a Simulink® vehicle model; it is a form of co-simulation, in which each software uses his own solver. The vehicle model runs with a fixed step size of 0.001 sec., whereas the hydraulic circuit uses a variable step algorithm, optimised for stiff systems. A proper communication interval was defined for the data exchange process between the two softwares. The vehicle model calculates the body roll angle and imposes it on the torsion spring model implemented in AMESim®, which gives origin to the reaction transmitted to the car body model. The control algorithm for the ARC system is implemented in Simulink®. It receives all the typical signals corresponding to the sensors of a real car: steering wheel angle, body yaw rate, lateral acceleration, longitudinal speed (from the Vehicle Dynamics Control) and the main values from the engine control unit. The so called ‘high level control algorithm’ determines the desired anti-roll torque from the bar, whereas the ‘low level control algorithm’ decides the actuation for the electro-valves and the motor pump to give origin to the desired effect. The actuation strategy is usually based on force control, since the hydraulic actuator has to generate a reaction proportional to the desired anti-roll moment.

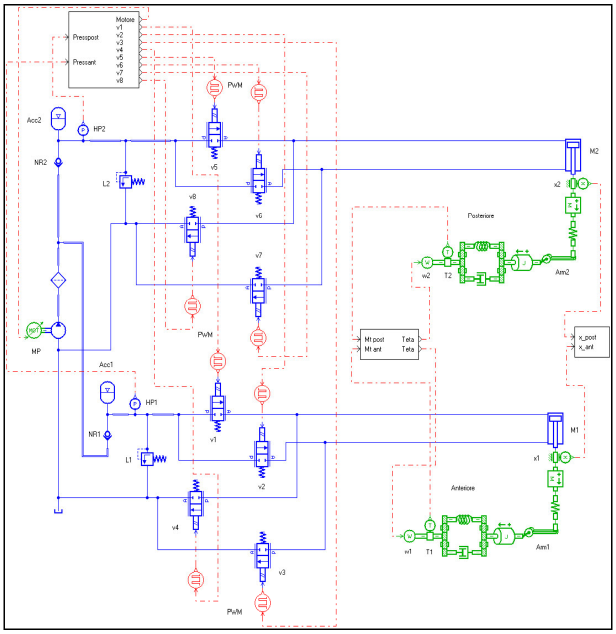


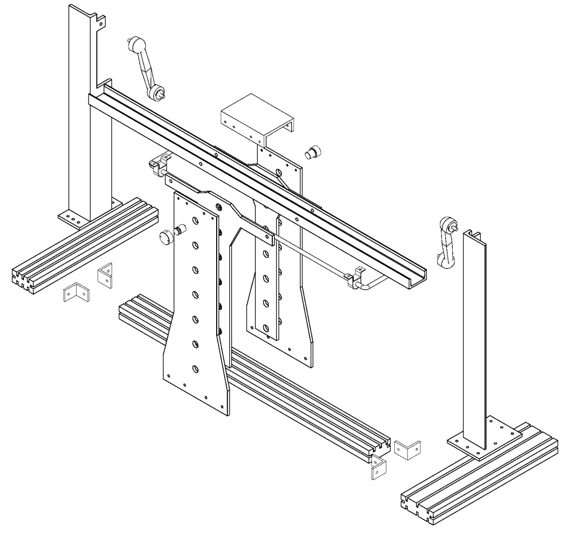
Figure 7 – First approximation model of an ARC actuation system for both the bars of the vehicle

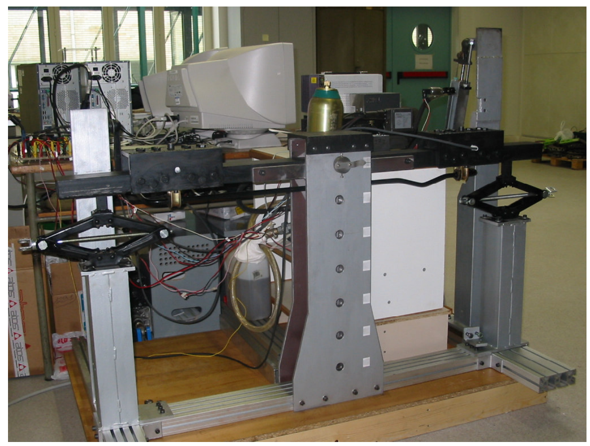
In the next paragraph we are going to analyse how it is possible to design the hydraulic ARC system using the algorithm we just mentioned in a Hardware-In-the-Loop simulation.

3 Hardware In the Loop simulation for the design of the hydraulic ARC system

This fundamental phase consisted in implementing the simulated system on a Hardware-In-the-Loop (HIL) test bench, capable of generating a dynamic roll angle on the physical bar (Figures 8 and 9) according to the output of a real time vehicle model. The bar was equipped with the designed ARC system. Devoted load cells (Figure 10), insensible to radial loads, were interposed between the bushings of the torsion bar and the bench, to measure the effect of the bar. Their signals were used by the real time vehicle model to compute the actual value of body roll angle (Figure 12). The procedure to generate the interface between the vehicle model and bench is the same which was followed for the connection between the vehicle model and the AMESim® model of the hydraulic circuit of the bench. The model of Figure 7 can be considered a virtual test bench for the ARC actuation system. The experimental activity consisted of several steps:

- Measurement of the roll stiffness of the passive bar, for fixed values of body roll angle. It was useful to evaluate bending effects due to the flexibility of the lever arms of the torsion bar;





Figures 8, 9 – Politecnico di Torino ARC test bench: a schematic and a photograph

- Measurement of the effect due to the linear hydraulic actuator of the ARC system in terms of torques between the bar and the vehicle body;

- Hardware In the Loop simulation of the roll behaviour of the car according to the vehicle model output, to verify the real work of the ARC system.

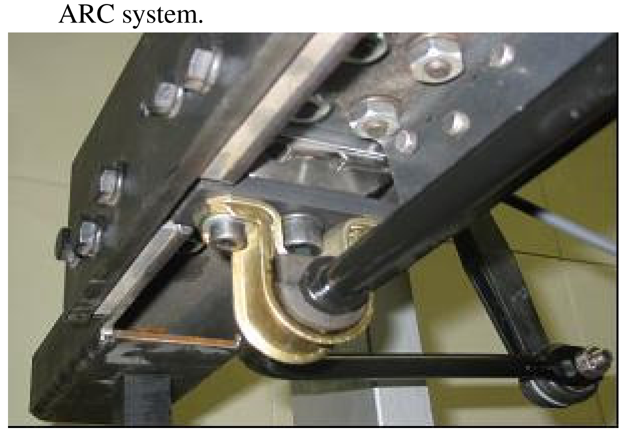


Figure 10 – ARC test bench: the mounting of the bar

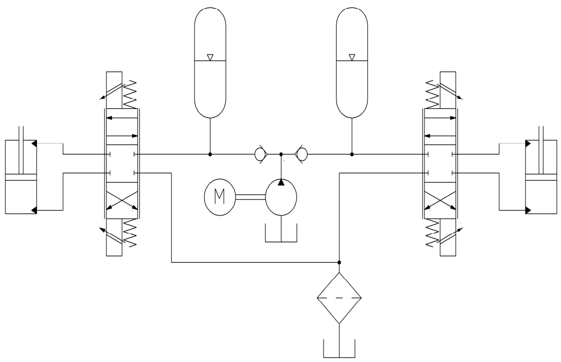


Figure 11 – ARC test bench: the hydraulic circuit

Figure 11 shows the hydraulic circuit implemented on the bench; it consists of two hydraulic actuators, the first used to generate the dynamic roll angle whereas the second corresponds to the ARC system.

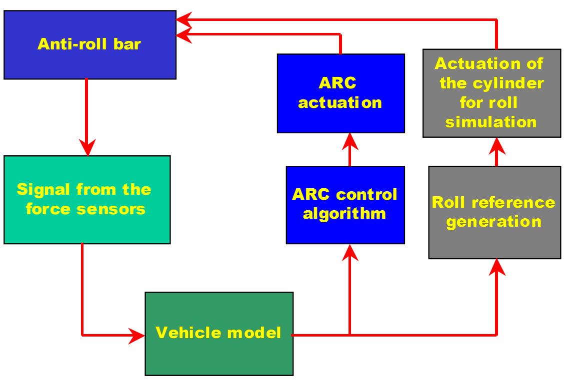
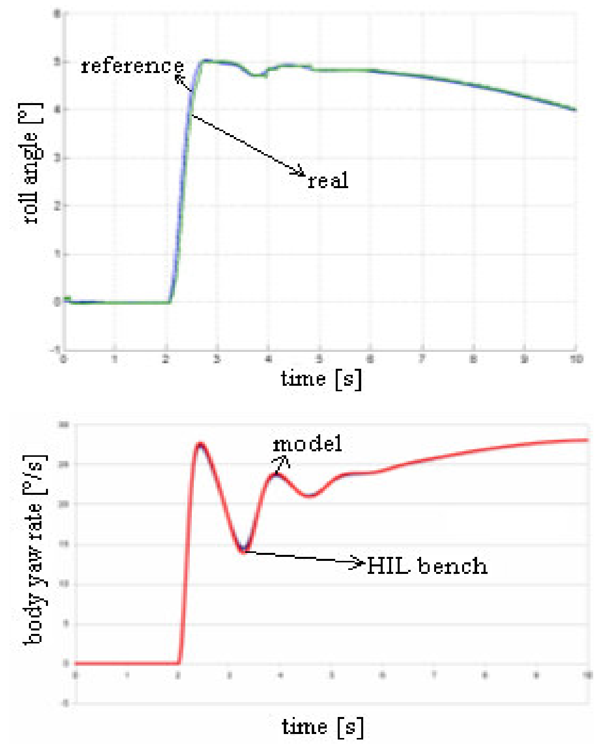


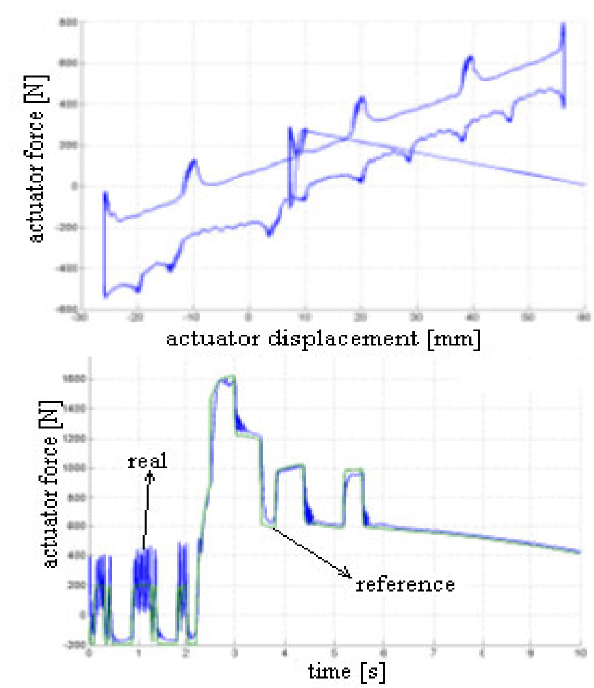
Figure 12 – ARC test bench: the scheme of the Hardware-In-the-Loop system

Figure 13 shows the results related to the generation of the dynamic roll angle on the test bench (whereas Figure 14 shows the effects in terms of body yaw rate), during a step steer manoeuvre without an ARC activation.
Figure 13 compares the reference body roll angle (imposed by the vehicle model) with the actual body roll angle generated by the bench; the dynamics of the experimental system is very good, since it seems to follow the virtual reference also during the low amplitude oscillations, immediately after the steering wheel rotation. The small difference between the desired and the effective roll angle does not provoke any perceivable difference from the point of view of body yaw rate during the same manoeuvre (Figure 14).
Figure 15 shows an example of characterization of the ARC actuator, in terms of force as a function of displacement. It is evident a hysteresis of about 200-300 N, due to the friction forces inside the actuator.
Figure 16 plots a comparison between the reference (imposed by the control algorithm) and the effective forces (measured on the physical bench) exchanged between the car body and the bushings of the ARC



Figures 13, 14 – Roll behaviour emulation performed by the ARC test bench; body yaw rate performance according to the stand-alone vehicle model and the vehicle model connected with the HIL test bench

system during a step steer manoeuvre (steering wheel rotation is performed at 2 sec). In the first part of the manoeuvre, the effect of the bar is reduced, to eliminate oversteer; on the opposite, in the second part of the manoeuvre, anti-roll bar equivalent stiffness is increased to reduce body roll angle.



Figures 15, 16 – Characterization of the ARC linear hydraulic actuator; comparison of the desired and emulated force of the ARC actuator

Figure 17 – Politecnico di Torino electro-mechanical ARC system

A devoted algorithm was conceived to reduce actuation errors connected with friction forces, by adding (to the reference force) a contribution, indicated with $F_{friction}$, according the estimated direction of motion of the actuator. A couple of devoted transfer functions (the first one estimates roll dynamics as a function of lateral acceleration, the other one calculates roll dynamics as a function of ARC actuation) is capable of estimating body roll angle also in dynamics conditions. The difference ΔM between the reference anti-roll moment for the bar and the estimated anti-roll moment of the passive rear bar is computed by the control algorithm.

$$\Delta M = M_{reference} - M_{estimated}$$

where $M_{estimated} = \Gamma_{rearbar} \cdot \varphi_{estimated}$.

$\Gamma_{rearbar}$ is the roll stiffness of the rear bar, considering a constant displacement of the linear actuator. A proper force $F_{friction}$, capable of compensating friction phenomena, is added to the reference force for the actuator, according to the estimated versus of motion for the actuator:

$$F_{reference,ARC\_actutuator} = F_{reference} + sign\left(\frac{d(\Delta M)}{dt}\right) F_{friction}$$

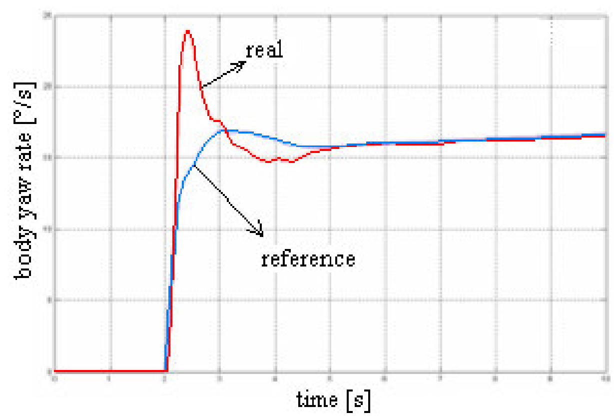
Of course, this control algorithm has to add $F_{friction}$ only if the time derivative of ΔM is over a defined threshold, not to have an unsteady behaviour of the linear actuator. This trick is fundamental to guarantee the reliability and the safety of the carried out system as specified in the introduction.

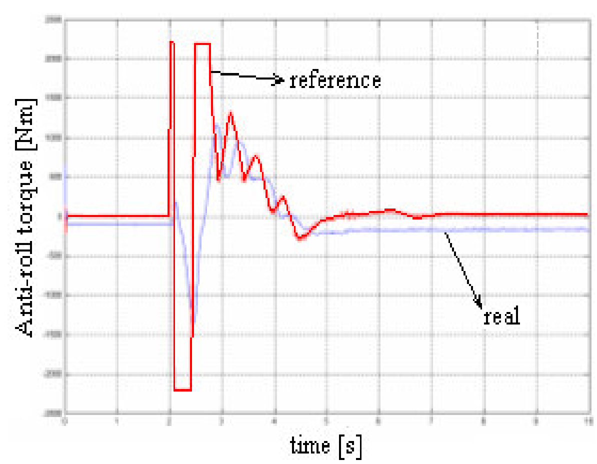
Further development of the system will consist in adding a bypass valve to connect both the chambers of the hydraulic actuator to the tank during the straight ahead travel of the vehicle. This valve will permit a comfort improvement, since it will eliminate the forces transmitted to the vehicle body due to asymmetric bumps.

Both the last shrewdness allow us to optimise the whole system and to get the desired goals in terms of safety, comfort and reliability. The illustrated kind of procedure can be obviously applied to every electromechanical device, both in aeronautical and in mechanical field ([4]) .

4 HIL applied to ARC

A second application of the same methodology and analysis procedure ([4]), linked with the "case study" we talked about, concerns the design of an electro-mechanical ARC actuator, for the same rear torsion bar. It is characterized by the separation of the rear torsion bar in two halves, having a vertical lever arm on which the linear actuator can exert the proper forces. The linear actuator is formed by a brushless motor controlled in torque, a gearbox and a recirculating balls screw. An experimental study on a test bench, similar to that described for the hydraulic system, had to be performed also for this solution. Figure 18 permits to appreciate the improvement of the active vehicle (with the electro-mechanical ARC only) in terms of body yaw rate during a step steer manoeuvre, the same of Figures 1 and 2. Figures 19 permits to see the delays in the actuation, typical of the electro-mechanical system, compensated by an efficient control algorithm. Devoted experimental tests were necessary to implement a compensation of the inertial and stick-slip phenomena of the actuator.





Figures 18, 19 – Step steer manoeuvre: reference and actual body yaw rate versus time; reference and actual anti-roll moments

5 Conclusions

The paper summarizes the simulation based procedures used to implement algorithms and design actuators for active chassis control systems.

It is basic to underline the importance of the methodology we used to approach and solve the problem. Knowledge comes from different fields (aeronautical and mechanical) and allowed us to get best results reducing testing time and saving money, achieving in the meantime the reliability and safety target. This procedure of models validation and testing through the feedback loop control can be widely applied to every complex electro mechanical system with the due modifications.

Obviously, as we showed about the analysed "case study", experimental test benches remain necessary to refine the design, especially of the control strategy, in particular from the point of view of the compensation of friction and stick-slip phenomena. Future publications will deal in detail with the physical layout of the components and the control algorithms of each of the ARC systems here presented. The same method was adopted by our research team also for other active systems, like Electronic Stability Program (ESP).

References:

[1] M. Saeger, A. Gartner, '*Simulation Environment for the Investigation of Active Roll Control in Combination with a Vehicle Dynamics Controller*' paper F2004F330, FISITA World Congress 2004, Barcellona;

[2] A. Sorniotti, M. Velardocchia, P. Krieff, D. Danesin, '*Active Roll Control to Increase Handling and Comfort*', SAE paper 2003-01-1062;

[3] A. Hac, '*Influence of Active Chassis Systems on Vehicle Propensity to Maneuver-Induced Rollovers*', SAE paper 2002-01-0967

[4] G. Repici, A. Sorniotti, '*A new approach to failsafe projects in mechanical field: experimental results*', WSEAS Conference 2003, New York

[5] Society of Automotive Engineers: *ARP-4761: Aerospace recommended practice: guidelines and methods for conducting safety assessment process on civil airborne systems and equipment*. 12th edition, SAE, USA,1996