Active Roll Control (ARC): System Design and Hardware-In-the-Loop Test Bench

A. SORNIOTTI, A. MORGANDO and M. VELARDOCCHIA*

Politecnico di Torino, Department of Mechanics

Correspondence

*Corresponding author. Email: mauro.velardocchia@polito.it

Abstract. The first part of the paper describes the targets related to the design of an Active Roll Control (ARC) system, based on the hydraulic actuation of the anti-roll bars of an automobile. Then the basic static and dynamic design principles of the system are commented in detail. The second part of the paper presents the Hardware-In-the-Loop (HIL) test bench implemented to evaluate the designed system. In the end, the main experimental results are summarized and discussed, also from the point of view of the integration of ARC with Electronic Stability Program (ESP).

Keywords: Active Roll Control; Hardware-In-the-Loop; Handling; Comfort

1 Targets and Fundamentals

The first target of the Active Roll Control (ARC) system is the reduction of the roll angle for small values of vehicle body lateral acceleration during semi-stationary manoeuvres. This target improves the comfort feeling transmitted by the vehicle to the passengers. In addition, it reduces the variation of the characteristic angles, especially camber angle, between the tires and the road plane during vehicle turning. It can provoke, according to the characteristics of the suspensions, a substantial improvement of vehicle dynamics in semi-stationary manoeuvres. The second target of the ARC system is the reduction of body sideslip angle and body yaw rate oscillations during dynamic manoeuvres, like step steer or double lane change. This target can be reached through a dynamic variation of the roll stiffness distribution between the two axles of the car. This property was described in several papers ([1--2]) and is based on the non-linearity of tires characteristics. In particular it is theoretically founded on the behaviour of tires lateral stiffness as a function of vertical load ([3]). An increased stiffness of an anti-roll bar provokes a decay of the total lateral stiffness of the axle. Especially for high values of lateral acceleration, the decrease in lateral stiffness of the tire internal to the bend is not fully compensated by a corresponding increase in lateral stiffness of the tire external to the bend. The result is less understeer if the rear bar is pre-charged by the ARC system and more understeer if the front bar is pre-charged by the ARC. ARC has to cancel the effect of the anti-roll bar during the straight ahead travel of the automobile, to reduce the dynamic forces on vehicle body induced by road bumps. ARC actuation can be performed in several ways, for example by the introduction of a rotational actuator in the middle of the bar, or substituting a linear actuator for one of the rods connecting the bar with the suspension system. The actuation can be either hydraulic or electro-mechanical. This paper is focused on the development of a hydraulic ARC system based on a linear actuator substituting one of the two rods connecting the bar to the suspension.

2 Active Roll Control (ARC) Design

2.1 Basic Calculations

The first step in the design process of an ARC system consists in deciding the number of active bars, their geometry and the number of channels (in the case of two active bars) of the hydraulic

system. A single channel system is a layout which guarantees a constant ratio between front and rear anti-roll torques. Double channel ARC can give origin to a variable anti-roll torques ratio between front and rear axles. Two channels and single channel solutions were compared. Firstly, it is necessary to define the basic dimensions of the bar, in terms of diameter, length and geometry of the lever arms. In order to have a dimension of the actuator which is coherent with its mounting on a real car, it is necessary to design active bars stiffer than the corresponding passive bars. In fact, a stiffer bar permits a larger variation of the anti-roll torque for the same displacement of the actuator. Figure 1 plots, as an example, roll characteristics (roll angle as a function of lateral acceleration) which can be obtained by adopting different dimensions for the rear bars. The bar indicated as '2' is characterized by a larger diameter than the bar '1'.

Figure 1 – Roll angle as a function of lateral acceleration for different values of anti-roll bars stiffness

Figure 2 – Pressure difference in the chambers of the ARC actuator as a function of vehicle lateral acceleration

$$\vartheta_{T,MAX} = \frac{M_{T,MAX}}{K_T} \quad (1)$$

$$\alpha = \frac{\pi}{2} - \frac{1}{2}(\pi - \theta_T) \quad (2)$$

$$\beta = \tan^{-1} \left(\frac{AD(OA, \vartheta_T)}{CD(OA, \vartheta_T)} \cdot \tan(\alpha) \right) \quad (3)$$

$$CB = \frac{CD}{\cos(\beta)} \quad (4)$$

$$\Delta l_{MAX} = CB_{MAX} - AC = f(\vartheta_{T,MAX}) \quad (5)$$

$$\frac{l}{2} \cdot \sin(\vartheta_{roll}) = \Delta l_1 \quad (6)$$

$$\Delta l_{TOT} = \Delta l_2 + \Delta l_1 \approx 2 \cdot \Delta l_1 \quad (7)$$

Figure 3 – A scheme of the anti-roll bar equipped with the hydraulic actuator together with the basic kinematical formulae

The second fundamental parameter which has to be considered at the beginning of the design is the maximum desired pressure inside the hydraulic circuit. For middle size cars, it is usually possible to limit the maximum pressure level at less than 100 bars if the actuation regards a rear bar only. If a front bar is actuated, pressure levels can reach 200 bars. Another fundamental point is the typical asymmetry of the linear hydraulic actuator, which is characterized by different active areas for the motion in the two directions. This characteristic is underlined by Figure 2, which plots the difference in pressure between the two chambers of an ARC hydraulic actuator as a function of lateral acceleration, for semi-stationary steering manoeuvres on the right and on the left. Equations (1)-(5) in Figure 3 contain the summary of geometrical calculations for a first approximation prediction of the maximum actuator length variation Δl_{MAX} , during the intervention of the hydraulic

system for a null value of body roll angle. It is the most critical condition from the point of view of the stroke of the actuator since the twist angle of the bar is entirely provoked by the active system and not by the roll motion of the vehicle. $M_{T,MAX}$ is the maximum twist torque desired for the system (at a null roll angle), K_T is the torsion stiffness of the bar and $\mathcal{G}_T$ is the bar twist angle induced by the hydraulic actuator. Usually, during extreme dynamic manoeuvres, it is necessary that the system guarantees a free motion of the actuator without a torsion of the bar; for example, in the case of a rear bar, it prevents oversteer. If the chosen bar is sufficiently stiff, the length of the hydraulic actuator is determined by the maximum value of body roll angle, in conditions of free motion of the actuator, without the contact of the piston of the actuator with the endstop. Equations (6) and (7) permit the first approximation computation of the actuator displacement variation induced by roll motion $\mathcal{G}_{roll}$ with a free-moving actuator, having both the chambers connected to the tank. It considers two contributions: the first one, Δl_1 , is due to the displacement of the extremity of the anti-roll bar (length l , Figure 4) induced by the roll motion of the vehicle, the second one, Δl_2 , considers the rotation of the torsion bar around its own axis induced by the passive rod opposite to the hydraulic actuator. It can be demonstrated that the two contributions are in first approximation equal. Through these stationary calculations, it is possible to define the main geometric parameters for the rod and the diameter of the chambers of the actuator. It has to include integrated spherical joints at its ends for the connection with the bar and the suspensions.

Figure 4 – A scheme of the displacement during the free motion of the anti-roll bar

Figure 5 – The variation of the normal force between tires and ground due to the effect of the ARC system considering different anti-roll bars

Figure 6 – An experimental time history of lateral acceleration for a vehicle (courtesy of Fiat Auto)

Figure 7 – A time history for the active bar, ARC control logic dependent, in terms of anti-roll generated torque

To perform a first approximation design of the actuation for an ARC system, it is necessary to evaluate the useful effect of the active system in comparison to a passive bar. The evaluation can be managed, for example, considering the maximum variation of vertical force between the tires and the ground permitted by the active system in comparison with the passive bars. Figure 5 compares the useful effect in term of vertical force increase due to two different ARC configurations. This

calculation, fundamental to consider the effect of the system on vehicle dynamics, has to take in account the geometry of the suspension system. It can be evaluated through multi-body softwares or, like in this design, on the basis of the experimentally measured elasto-kinematic characteristics of the suspensions of the car. The following task consists in performing the basic calculations to verify the structural behaviour of the bar. Stationary calculations have to consider the torsion effort inside the bar and the flexibility of the lever arms, which can reduce of about the 10% the theoretical roll stiffness of the anti-roll bar. The fatigue calculations can consider an experimental life cycle of an automobile (Figure 6), in terms of a time history of lateral acceleration. Fatigue calculations have to take in account the effect of ARC control algorithm on the bar, because different tunings of the ARC control algorithm give origin to different time histories of the anti-roll torques generated by the hardware of the same active bar. A possible time history, considered in this work, is shown in Figure 7. As a consequence, the designer has to know from the first steps of the design process which will be the fundamentals of the adopted control algorithm. The reduction of roll angles in semi-stationary manoeuvres can imply consistent strains in the active bar also for low values of lateral acceleration, in not extreme drive conditions.

2.2 Simulations

Simulations were adopted to foresee the hydraulic performance of the ARC system in dynamic conditions. For the first prototype, conventional hydraulic components were evaluated. It was chosen a low volume displacement pumping unit for the system, joined with a hydraulic accumulator capable of guaranteeing a sufficient volume of pressurized fluid for extreme manoeuvres. A devoted mono-dimensional software was adopted to model the hydraulic system. Valve body displacement dynamics is considered through a second order transfer function; the transition from laminar to turbulent motion of the oil through the valve is taken in account. Fluid compressibility due to gas bubbles inside the fluid is simulated, together with fluid inertial effects. Figure 8 shows a model of a two channels ARC system. Typical step, ramp and sweep tests were performed through these kind of models. The following step in the simulation process consists in linking the hydraulic circuit to a vehicle model and to the control algorithm of the ARC system, in a co-simulation process. In this way, a first prediction of the effect of the ARC system on vehicle dynamics can be obtained, even if the mono-dimensionality of the models of the bar can give origin to different displacements of the actuators, if compared to those of the real system. On a real car, there is the motion both of the attachments of the bar to the vehicle body and the effect of the rods connecting the bar to the suspensions. In a mono-dimensional model, the bar is modelled through a torsion spring with an equivalent rotation of its ends, which has to summarize all the effects typical of actual bars. Figure 9 presents a qualitative evaluation of the possible increase in handling and comfort performance related to different layouts of ARC. Two channels systems can contemporarily have a consistent effect both on roll dynamics and vehicle handling, single channel

Figure 8 – An example of model of a two channels ARC actuation system, used also for co-simulation with a whole vehicle model

ARC CONFIGURATION	HANDLING	COMFORT
Front	0	++
Rear	++	+
Front/rear single channel	0	+++
Front/rear double channel	+++	+++

Figure 9 – A chart summarizing the typical qualitative advantages (+) guaranteed by the different layouts of ARC systems

and two bars systems have a consistent effect on roll dynamics, rear single bar systems have an effect both on roll dynamics and handling, single front bar systems can have an influence mainly on roll dynamics.

3 ARC Hardware-In-the-Loop Test Bench

On the basis of vehicle dynamics simulations, it was implemented a rear bar based Active Roll Control system. The diameter of the bar is 25 mm and its measured roll stiffness is equal to 41000 Nm/rad. It was tested through a Hardware-In-the-Loop test bench. The bench was conceived to reproduce the roll dynamics of the vehicle. Figure 10 is a sketch of the mechanical structure of the bench. It consists of a system capable of rotating around a pivot, which corresponds to the roll centre of the suspension of the vehicle. The bar is fixed to the rotating part of the bench.

Figure 10 – The basic structure of the test bench

Figure 11 – Schematic of the Hardware-In-the-Loop concept applied to the ARC system

Figure 12 – The ARC HIL test bench

Figure 13 – The pressure accumulator for the ARC system

Figure 11 shows the logical connections between the elements of the bench. A vehicle mathematical model simulates vehicle dynamics. It computes a reference roll angle for the bench; it derives from the forces measured by devoted sensors located between the bar attachments and the bench. The reference roll angle is compared with the effective roll angle measured by a potentiometer located on the bench. A control system based on hydraulic components gives origin to the desired roll angle value on the bench. In the meantime, the vehicle model sends the input signals, like steering wheel angle, lateral acceleration, body yaw rate, etc..., to the ARC control algorithm, which sets a reference anti-roll torque and gives the proper input to the valves. It is the same loop used for the co-simulation process. Figures 12-14 show some of the components of the bench. The ARC hydraulic actuator was equipped with both displacement and pressure sensors, since two kinds of actuation algorithms for ARC were conceived during the activity, alternatively based on pressure

control and displacement control. The ARC hydraulic circuit implemented on the bench consists of a pump, a hydraulic accumulator, a proportional valve to control the actuator and a by-pass valve (to connect the chambers of the actuator with the tank) to give origin to very low force level during the straight ahead travel of the vehicle. Figures 15 and 16 are examples of the tests which were performed to validate the performance of the test bench before mounting the ARC system. Figure 15 is a frequency response test; the bench follows the typical frequencies induced by a driver for the roll motion of the vehicle, corresponding to a maximum of about 3 Hz. Figure 16 shows a comparison between the reference roll angle computed by the vehicle model and the real roll angle generated by the bench during an extreme step steer manoeuvre by adopting a passive anti-roll bar. The bench was adopted to test a specifically developed ARC system but could be used, without substantial modifications, to evaluate commercial ARC systems, also in the double channel configuration.

Figure 14 – The force sensor between the bar attachment and the bench

Figure 15 – A test to evaluate the performance of the bench in reproducing the desired roll angle

Figure 16 – A comparison between the reference and the actual values of body roll angle during a step steer manoeuvre

Figure 17 – Anti-roll torque as a function of roll angle during the opening of the by-pass valve

4 Experimental Tests

This paragraph contains some of the experimental results which can be obtained through the ARC test bench.

4.1 Basic Tests

Firstly, the typical tests include the characterization of the hydraulic behaviour of the ARC system, independently on the control algorithm based on vehicle dynamics and roll motion. Figure 17 is about a test in which both the chambers of the actuator are connected with the tank and a roll angle is generated by the bench, both towards left and right: the hysteretic behaviour in the curve of Figure 17 corresponds to the friction forces inside the ARC actuator. Figures 18 and 19 are about a test performed through a force control of the linear hydraulic actuator; a PID controller gives origin to the desired anti-roll torque for a fixed value of the roll angle at the bench. Stick-slip friction

phenomena inside the actuator are evident and can provoke an unpleasant feeling on the passengers of the vehicle. The curve plotting the anti-roll torque as a function of the actuator displacement appears to be much more regular in the case of a displacement control of the hydraulic actuator of the ARC system (Figures 20 and 21). Several tests were performed to evaluate the dynamics of the ARC system; response times were always under 0.1 s.

Figure 18 – ARC system characterization: a comparison between the reference force for the actuator and the estimated force on the basis of the pressure levels in the chambers of the actuator

Figure 20 – ARC system characterization: a comparison between the reference and the measured displacement of the actuator

Figure 19 – Measured actuator force as a function of displacement

Figure 21 – Measured anti-roll torque (directly proportional to actuator force) as a function of actuator displacement

4.2 High Level Control Algorithm

This paragraph describes the basic principle of the adopted control algorithms before presenting the results obtained with the full HIL system. The ARC high level control algorithm, on the basis of vehicle dynamics, gives origin to a reference value of the anti-roll torque. It consists of two contributions. The first one, useful in semi-stationary manoeuvres, computes the reference anti-roll torque through a table as a function of vehicle lateral acceleration. As a consequence, $M_{roll} = f(a_y)$.

M_{roll} is the contribution of anti-roll torque necessary to reduce the roll angle especially for middle-low values of body lateral acceleration. The second contribution gives origin to $M_{dynamic}$, as a function of the measured and a reference body yaw rate, computed by the control algorithm in the same way adopted by commercial Electronic Stability Programs (ESP). The anti-roll reference torque for the ARC system is given by:

$$M_{reference, high} = G_1 \cdot M_{dynamic} + (1 - G_1) \cdot M_{roll} \quad (8)$$

where G_1 takes in account the actual condition of the vehicle. If there is a large difference between the real and the reference body yaw rate, it can be supposed that the vehicle is in dynamic conditions and priority is given to the body yaw rate based control algorithm. Other additional conditions can be added to improve the performance of the system.

$$G_1 = k_1 \cdot \Delta \psi \quad (9)$$

$M_{reference,high}$ is directly proportional to the force requested at the level of the hydraulic actuator. The ARC actuation controller, based on force or position, produces the desired anti-roll torque through the actuation of the electro-valves of the system.

4.3 ARC Force Control

In the case of the force control of the ARC system, the torque is produced through PID controllers having as input the reference and the estimated anti-roll torques generated by the ARC actuator. The load cells of the bench can be used only for the HIL process, to make run the vehicle model; as a consequence, the anti-roll torque due to ARC is estimated on the basis of the pressures p_1 and p_2 measured in the chambers of the hydraulic actuator (or in the exit ports of the ARC hydraulic unit towards the chambers of the actuator).

$$M_{estimated} = k \cdot (p_1 A_1 - p_2 A_2) \quad (10)$$

The second input to the PID controllers corresponds to the desired anti-roll torque; it is determined by the control algorithm based on roll control and vehicle dynamics. The second term in the equation (11) corresponds to a compensation of friction phenomena (Figure 19) inside the actuator.

$$M_{reference} = M_{reference,high} + M_{friction} \cdot \text{sign} \left(\frac{dM_{reference}}{dt} - \frac{dM_{passive}}{dt} \right) \quad (11)$$

This term is added in the case $\left| \frac{dM_{reference}}{dt} - \frac{dM_{passive}}{dt} \right| > threshold$. This expression can be used to identify

the condition of motion of the system; it must be active during the motion of the actuator, to have a smooth correspondence between the reference and the measured torque. $M_{passive}$ is the torque which would be generated by a passive bar, having the same stiffness Γ_{bar} of that used for the ARC actuation, in the real condition of roll angle of the active vehicle.

$$M_{passive} = \Gamma_{bar} \cdot \mathcal{G}_{roll,estimated} \quad (12)$$

The roll angle of the active vehicle can be expressed as the sum of two terms, the first one, $\mathcal{G}_{roll,1}$, corresponding to the passive car without ARC and the second one, $\mathcal{G}_{roll,2}$, corresponding to the variation of roll angle due to the contribution of the active bar.

$$\mathcal{G}_{roll,estimated} = \mathcal{G}_{roll,1} + \mathcal{G}_{roll,2} \quad (13)$$

Transfer functions can be adopted for the estimation of $\mathcal{G}_{roll,1}$ and $\mathcal{G}_{roll,2}$ on the basis of the measured lateral acceleration a_y and the reference anti-roll torque. In the following equations m is vehicle mass, H is the lever arm of inertial force in roll motion, J_x , c and Γ are the inertial, damping and stiffness parameters of the passive vehicle without the rear bar.

$$\frac{\mathcal{G}_{roll,1}}{a_y} = \frac{mH}{J_x s^2 + cs + \Gamma}, \quad \frac{\mathcal{G}_{roll,2}}{M_{reference}} = \frac{1}{J_x s^2 + cs + \Gamma} \quad (14), (15)$$

4.4 ARC Displacement Control

The second chance for ARC actuation is based on displacement control. In this case, the pressure sensors at the ARC actuator are not used any more. A devoted displacement sensor permits to determine the effective position of the actuator, which is compared, by PID controllers, with the reference displacement. ARC reference displacement is proportional to the reference anti-roll torque only for a fixed value of body roll angle, otherwise the reference displacement of the actuator has to take in account the anti-roll torque due to the torsion of the active bar for a null displacement of the actuator. In formulae:

$$x_{reference} = k_2 (M_{reference} - M_{passive}) \quad (16)$$

where k_2 is the constant coefficient to pass from the anti-roll torque to the actuator displacement for a fixed value of vehicle body roll angle. It can be computed by using the theoretical formulae on the basis of the geometry of the anti-roll bar or through the data of experimental tests, like that one of Figures 20 and 21.

4.5 Experimental Results with ARC tested on HIL test bench

The following tests were performed by using the full HIL bench, connecting the hardware of the bench with the whole vehicle dynamics model and the ARC control algorithm. Figure 22 compares the reference and the measured anti-roll torques during a ramp steer manoeuvre. The system controlled in force is characterized by an irregular behaviour in the first part of the manoeuvre, due to the stick-slip phenomena inside the actuator, which can only partially be compensated through the algorithm corresponding to (11). Displacement control gives origin to an automated compensation of friction forces inside the actuator, at the condition that roll angle is correctly estimated. In any case, also considering the performance of commercial systems evaluated through road tests (courtesy of Fiat Auto), the irregularity of the motion should not be perceived by the passengers of the vehicle also in the case of the force control. In dynamic manoeuvres on a flat road, stick slip phenomena disappear and no substantial difference can be perceived between force and displacement control (Figure 26). On the other hand, force control permits an automated compensation of the anti-roll torque disturbances related to road irregularities, which provoke an additional torsion of the bar, perceived only through the pressure sensors used by the force control. Figure 23 shows the variation (due to ARC) of the understeer characteristic of the vehicle during a ramp steer manoeuvre in high adherence conditions. Figure 24 shows actuator displacement during a ramp steer manoeuvre; the curve is completely different in the shape from that of Figure 22, due to the torsion which roll angle induces on the active bar. Figure 25 compares the roll characteristics of the passive and the active vehicles during a ramp steer manoeuvre in low adherence conditions; the effects due to ARC adoption are evident. Figures 22, 24 and 25 are about an extremely not linear kind of ARC actuation in semi-stationary conditions, with a nearly null value of body roll angle for low lateral accelerations and a decrease of roll stiffness for medium-high lateral accelerations. It should make the driver perceive the proximity of the adherence limits between the tires and the road. In any case, the same control algorithm can give origin to a large variety of roll characteristics, according to the specific application. Figures 26 and 27 show an example of integration between ARC and a pre-existing ESP control algorithm experimented at the HIL test bench during a double step steer manoeuvre. The vehicle with the integration of ESP and ARC is characterized by reduced oscillations in terms of body yaw rate in comparison to the vehicle with the only ESP. In addition, also roll angle values are less consistent. Of course, the results are greatly variable according to the specifications followed in the integration process; it is possible to give the priority to roll angle reduction or yaw rate and sideslip angle oscillations control, according to the tuning of the control algorithm. The integration adopted in the tests of Figures 26 and 27 was based

Figure 22 – Ramp steer manoeuvre: comparison between the displacement and the force control, high adherence

Figure 23 – The effect of ARC during a ramp steer manoeuvre, high adherence

Figure 24 – ARC actuator displacement during a ramp steer manoeuvre, high adherence

Figure 25 – The effect of ARC during a ramp steer manoeuvre in low adherence conditions

Figure 26 – Double step steer: comparison between ESP, ARC and ESP integrated with ARC (1: force control, 2: displacement control)

Figure 27 – Double step steer: comparison between ESP, ARC and ESP integrated with ARC

on low values of k_l (Equation 9) and gave origin only to a small reduction of ARC anti-roll torque in correspondence with the interventions of ESP to reduce yaw rate. Different parameters could lead to an improvement in yaw rate oscillations with a drawback from the point of view of roll angle and maximum ARC actuator displacement during the manoeuvre.

5 Conclusions

The paper describes the design procedure used to conceive an ARC together with the Hardware-In-the-Loop test bench to verify the performance of the system. A comparison between a force control and a position control of the hydraulic actuator of the ARC system is presented. The possible integration of ARC with ESP is shown in an example of extreme dynamic manoeuvre. Future work will regard the implementation of a double-channel system, the dimensional optimisation of the components of the presented system, in particular the actuator, and the final verification of the system performance through road tests.

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