Application of an Anti-roll Bar System for Enhanced Vehicle Handling

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***Abstract*—An anti-roll bar (ARB) is a component of the car's suspension system designed to reduce the body roll angle ϕ of the vehicle. When the vehicle turns, roll acceleration is initiated towards the outside of the turn. This results in normal load transfer which leads to reduction in lateral acceleration of the vehicle. To improve performance, an ARB is introduced. The ARB, an U-shaped rod, provides a moment in the direction opposite to that of the body roll by deflection of the rod. The torsional stiffness of the ARB therefore defines its ability to counteract roll motion. This results in improved comfort during driving, less body roll and better traction performance while taking a turn. In this paper, an 8 DOF vehicle model is proposed which accounts for the planar motion along with the roll dynamics of the car. The effectiveness of ARB in reducing body roll is investigated by performing a NHTSA Fishhook manoeuvre. Simulations done on MATLAB/Simulink are presented which highlight the characteristics of body roll. Finally, several conclusions are made about the applicability of the ARB and future improvements.**

I. Introduction

According to a research from the year of 2011 by the *National Highway Traffic Safety Administration* (NHTSA), 46.4 per cent of the total traffic accidents within the United States that resulted in fatalities, was due to rollover of the vehicle [1]. This type of accident is more likely to result in fatalities than any other type of crash. According to this same research, SUV-type vehicles most often get involved in these type of accidents. The study also declares that these numbers are increasing and that in most cases this inauspicious event occurs as a result of a crash-avoidance manoeuvre where an abrupt steering input is applied. In virtue of this alarming scenario the need for improvement of vehicle dynamics has lead to the development of a solution, that has been mainly defined as *Anti-Roll Bar (ARB)*.

An ARB is a mechanical component that has been designed in order to cope with influence of roll dynamics on vehicle handling. The Canadian inventor Stephen Coleman obtained the first patent for an ARB [2]. Most simple ARBs applied to vehicles nowadays, still use the model defined by Coleman. The ARB is usually constructed out of a cylindrical U-shaped steel bar that connects two points; the left- and right-hand sides of the suspension. A visual example is given in figure 1, where the component under discussion has been highlighted in red.

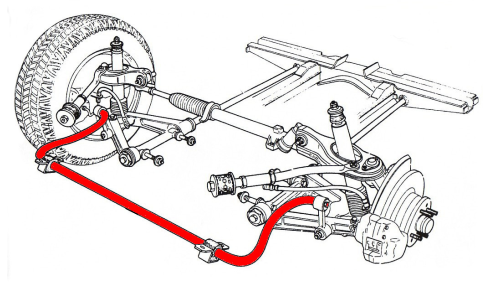

Fig. 1. Suspension system of the Alfa Romeo Alfetta (1972 - 1984) highlighting the applied passive ARB in red. (adapted from [3])

Through this connection, the element is capable of stabilizing the vehicle with respect to roll movements, either static or dynamic, creating counteracting moments due to it's own torsional stiffness. Through the years, the ARB has been an effective solution to the roll control problem for various reasons; the cost-efficiency of the solution with respect to active suspensions, the ease with which its stiffness can be designed, being inversely proportional to the length of the arms, and eventually the possibility, through its implementation, to keep spring stiffness in the suspensions relatively low, reducing consequently the vibration transmission in the middle-high frequency range.

The current automotive market also provides an alternative solution for the roll behaviour control of a vehicle, represented by the so called *Active Suspension*. This is a type of automotive suspension that controls the vertical movement of the wheels relative to the chassis, in opposition to the passive counterpart, where the movement is being determined entirely by the road surface. Active suspensions are generally divided into two classes, purely active and adaptive/semi-active suspensions, the latter by means of a variable shock absorber adapt to changing road or dynamic conditions, while the former use some type of actuator to raise and lower the chassis independently at each wheel.

On one hand this solution is able of virtually eliminating body roll and pitch variation, in many driving situations. Including cornering, accelerating, and braking. On the other hand this solution is extremely expensive, since implementation of such an architecture requires a high level of redundancy, both at sensor and actuator level, in addition to the expensive implementation of the control logic.

This drawback essentially restraints the application of these suspensions to the niche of the luxury automotive production; not being at the current state of the art a cost-efficient solution, the Anti-Roll Bar is generally preferred as solution to the body roll problem.

The influence and effectiveness of the ARB is measured on the basis of the three basic key performance indicators (KPIs): comfort, road holding and ride handling.

A. Comfort

Ride comfort is assessed through vibration analysis. Compensation of body roll reduces accelerations in roll angle $\frac{d^2}{dt^2}\phi$, reducing problems related to motion sickness of the passengers [4].

B. Road holding

The introduction of an ARB allows, as previously stated, to modulate the roll stiffness without modifying spring stiffness in suspension. This detail is of crucial importance since spring stiffness affects vertical dynamics, hence solving the roll problem with stiffer springs will lead to lower damping of road vibrations. Proper implementation of an ARB allows to achieve an acceptable level of tire contact with the road, ensuring sufficient contact patch area, therefore traction. During the design process, it is possible to make use of a smaller camber angle in the suspension setting, due to the fact that the body will suffer less from lateral load transfer and hence less camber compensation will be required during cornering. This characteristic consequently improves longitudinal dynamics, because of a larger contact patch, and will also result in longer life of the tire because of uniform wear.

C. Ride handling

The presence of an ARB allows the driver to receive proper feedback from the vehicle during dynamic excitation, allowing a better performance even during challenging manoeuvres, keeping at the same time an appropriate level of stability of the chassis. For feedback purposes, total compensation of the body roll is undesired. This can be ensured by properly designing the stiffness of the ARB achieving a reasonable trade-off. Body roll induces lateral load transfer, decreasing the acceleration a_y of the vehicle. To compensate for this and get better traction performance, an ARB is introduced.

In the majority of passenger cars the ARB is mounted on the front to increase the relative rolling stiffness of the axle and to reduce the lateral load transfer on it. The main purpose is the achievement of a higher level of overall lateral force F_y. Because of non-linearity of the tire dynamics, the maximum level of lateral force, and consequently the maximum level of lateral acceleration a_y, are decreased even with the slightest variation of the normal load on the tires. The relation between the non-linear tire characteristic and loss of lateral force are shown in figure 2. A reduction in the average value of lateral force F_y is directly proportional to the magnitude of the load transfer ΔF_z.

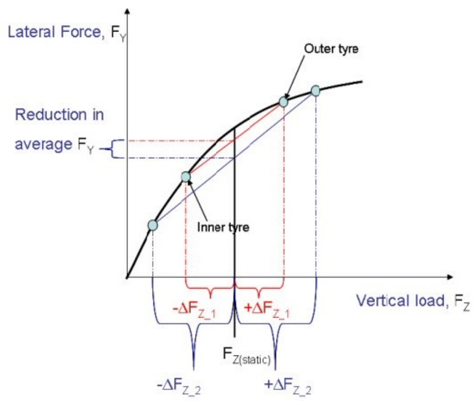

Fig. 2. Non-linear tire characteristic showing the proportional relation between changing normal loads ΔF_z and the change in lateral force ΔF_y of the tire, which results from linear interpolation between the two vertical loads (adapted from [5]).

In virtue of this reasoning, vehicle performances can be further improved by assembling a second ARB on the rear, which will also be assessed in this paper.

However, the way in which the passive ARB (P-ARB) reduces the roll of the vehicle has a side effect. If one of the wheels would hit an obstacle then, oscillations induced on the suspension in this wheel will induce torsion of the ARB and hence both wheels will be affected. This is often referred to as the *copying effect*. The copying effect has negative impact on ride comfort relative to the situation where both wheels (of front or rear axle) could move independently from each other. To cope with this problem, a decoupling of the axle is performed through the implementation of an *Active Anti-Roll Bar* (A-ARB).

The A-ARB solution consists of decoupling the two wheels by means of separation of the two arms of the bar, in the middle of which an electric motor is mounted. The latter provides, when insisted by the ECU, a torque $M_{ARB,act}$ proportional to the roll disturbance. A visual example of such a system is shown in figure 3.

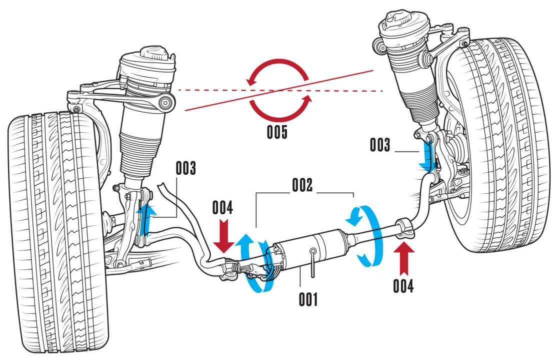

Fig. 3. A-ARB layout of the Bentley Bentayga (model 2016) also showing visual representation of dynamic effects [6]. The electric motor with planetary gearset (1) applies a torque to the ARB (2) which is transferred to the dampers (3). This creating a reaction force (4) and so decreasing body roll (5).

II. Vehicle Model

To assess body roll behaviour of the vehicle, implementation of a sufficiently complex model, capable of expressing the lateral load transfer induced by roll and lateral accelerations, is required. For the proposed purpose an 8-DOFs model is chosen where, in addition to the usual motions in xy-plane, wheel- and roll dynamics are also taken into account.

A schematic representation of the vehicle model is provided in figure 4.

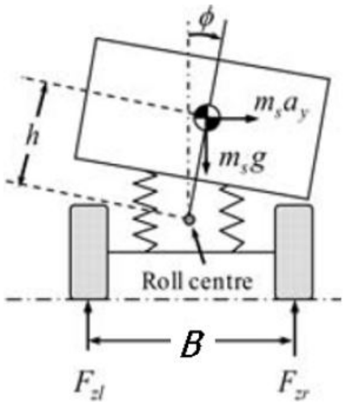

Fig. 4. Schematic representation of the 8-DOFs vehicle model, which forms the basis for the simulation on roll dynamics executed in this article.

In this paper, it is assumed that

1) The total mass of the vehicle is approximated only with the sprung mass, i.e. $m = m_s$,
2) The vehicle does not perform vertical motion (heave),
3) Rolling resistance and drag forces are negligible,
4) The vehicle does not perform pitch motion,
5) The vehicle is symmetric about the xz-plane,
6) Since a Dugoff tire model is used, the self-aligning moment M_{zwi} has been neglected. .

Equations related to the model can be derived from the *Euler equation*. Building of the model is started from the definition of the normal load per tire, which is affected by the introduction of the roll dynamics. The normal load is expressed through the following relations

$$\begin{aligned} F_{zfl} &= mg\frac{l_r}{2L} - F_z^{long} - F_{z,f}^{lat} \\ F_{zfr} &= mg\frac{l_r}{2L} - F_z^{long} + F_{z,f}^{lat} \\ F_{zrl} &= mg\frac{l_f}{2L} + F_z^{long} - F_{z,r}^{lat} \\ F_{zrr} &= mg\frac{l_f}{2L} + F_z^{long} + F_{z,r}^{lat} \end{aligned} \tag{1}$$

where the load transfer terms are defined as

$$\begin{aligned} F_z^{long} &= m\frac{a_x}{2L}h_{cg} \\ F_z^{lat} &= \frac{mgh\sin(\phi) + ma_y h\cos(\phi)}{B} \end{aligned}$$

and h is the actual distance of the center of gravity (CoG) to the roll center, i.e. the term that effectively characterizes front and rear moment arm, defined as

$$h = \frac{h_{cg} - h_{rc}}{\cos(\phi)}$$

It has to be noted that, upon substitution of h, the amount of lateral load transfer F_z^{lat} will be proportional to the roll angle ϕ, a trend that will be proven in simulations and corrected with the implementation and tuning of the (A-)ARB.

These relations can be used to calculate resulting normal forces. Using basic kinematic equations it is possible to write the *Newton Euler Equations* for the vehicle dynamics, for which, as has previously been pointed out, pitch and heave behaviour are assumed to be negligible and the vehicle is assumed to be symmetric with respect to the xz-plane. The resulting relations are the following

$$\begin{aligned} m(\dot{u} - vr) - m_s \dot{r}\dot{\phi}h\cos(\phi) &= \Sigma F_{x_1}, \\ F_{x,fl} + F_{x,fr} + F_{x,rl} + F_{x,rr} &= \Sigma F_{x_2}, \\ \Sigma F_{x_1} + \Sigma F_{x_2} &= \Sigma F_x, \\ m(\dot{v} - ur) - m_s \ddot{\phi}h\cos(\phi) &= \Sigma F_{y_1}, \\ F_{y,fl} + F_{y,fr} + F_{y,rl} + F_{y,rr} &= \Sigma F_{y_2}, \\ \Sigma F_{y_1} + \Sigma F_{y_2} &= \Sigma F_y, \\ I_{zz}\dot{r} - I_{xz}\ddot{\phi} + m_s a_x h_s \sin(\phi) &= \Sigma M_{z_1}, \\ l_f\left(F_{y,fl} + F_{y,fr}\right) - l_r\left(F_{y,rl} + F_{y,rr}\right) &+ \\ -0.5B\left(F_{x,fl} - F_{x,fr} + F_{x,rl} - F_{x,rr}\right) + \cancel{\sum_{i=1}^{4} M_{zwi}} &= \Sigma M_{z_2}, \\ \Sigma M_{z_1} + \Sigma M_{z_2} &= \Sigma M_z, \\ I_{xx}\ddot{\phi} - I_{xz}\dot{r} + K_{\phi_{eq}}\sin(\phi) &+ \\ +\left(C_{\phi f} + C_{\phi r}\right)\frac{B^2}{2}\dot{\phi}\cos(\phi) + M_{ARB,act} &= \Sigma M_x \end{aligned} \tag{2}$$

where for calculating the total moment ΣM_z, assumption 6 is applied. It has to be pointed out that the influence of the ARB in this model has been introduced both explicitly, through an eventual torque coming from an A-ARB ($M_{ARB,act}$), as well as implicitly, in the equivalent passive rolling stiffness term $K_{\phi_{eq}}$, defined as

$$K_{\phi_{eq}} = \tfrac{K_{\phi f} + K_{\phi r}}{2}B^2 + K_{ARB_f} + K_{ARB_r} \tag{3}$$

which is clearly affected by the assembling of the bar. Moreover, no combination of different types of ARBs have been performed and/or analyzed; in this way is possible to show and distinguish better the characteristic features of the passive and active versions.

III. Assessment Method, Simulation Results and Comparison

Using given vehicle model from section II, the setup is ready for simulation. This paper will focus on the assessment of dynamic roll stability, hence no further investigation on the *static stability factor* (SSF) will be done. The dynamic assessment is performed through the *NHTSA Fishhook Test*.

The fishhook test is a comprehensive experiment that investigates the ability of performing an emergency manoeuvre to avoid an obstacle, defining in this way the roll stability limit of the vehicle.

A. Test Method

The standard is described by the NHTSA using the following test procedure [7]

1) The test-vehicle is driven in a straight line
2) A first steering input is given and hold until the roll rate of the body is less or equal than $1.5deg/sec$
3) A counter-steer action is held for 3 seconds
4) The steering wheel is returned to it's natural position ($\delta_{SW} = 0deg$)

The magnitude of the steering actions described in steps 2 and 3 of the test procedure are symmetric, equal to 6.5 times the δ_{SW} that generates a steady state lateral acceleration of $0.3g$ at $50km/h$ on pavement. The steering rate $\dot{\delta}_{SW}$ is set to $720deg/s$.

The steering angle versus time is shown in figure 5 and the resulting manoeuvre in the xy-plane is shown in figure 6.

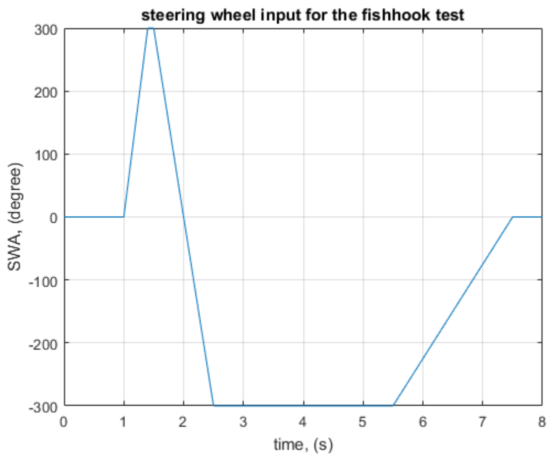

Fig. 5. Steering wheel input δ_{SW} versus time for the fishhook manoeuvre as described in the NHTSA-standard [7].

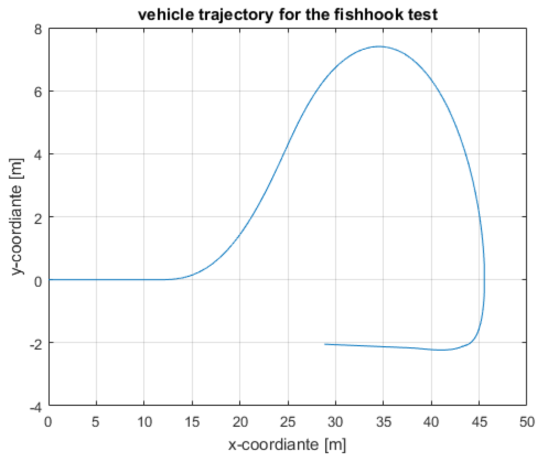

Fig. 6. Top-down view (projection on the xy-plane) of the vehicle's trajectory when performing the fishhook manoeuvre.

B. Simulation Data

The test vehicle considered for the simulations is a passenger car, whose relevant data is shown in table I.

TABLE I
Vehicle parameters used for simulating vehicle response.

Symbol	Description	Value	Unit
I_{xx}	Inertia along x-axis	580	$[kgm^2]$
I_{zz}	Inertia along z-axis	3,240	$[kgm^2]$
I_{xz}	Coupling Inertia	50	$[kgm^2]$
r_w	Effective Wheel Radius	0.24	$[m]$
l_f	Front Wheelbase	1.253	$[m]$
l_r	Rear Wheelbase	1.508	$[m]$
B	Track	1.5	$[m]$
h_{cg}	Height of CoG	0.75	$[m]$
h_r	Height of Rear Roll Center	0.4	$[m]$
h_f	Height of Front Roll Center	0.4	$[m]$
C_{df}	Front Damping Coefficient	2,000	$[kg/s]$
C_{dr}	Rear Damping Coefficient	2,000	$[kg/s]$
K_f	Front Spring Stiffness	20,000	$[kg/s^2]$
K_r	Rear Spring Stiffness	20,000	$[kg/s^2]$
C_f	Front Cornering Stiffness	57,000	$[kg \cdot m/s^2]$
C_r	Rear Cornering Stiffness	47,000	$[kg \cdot m/s^2]$
C_κ	Front/Rear Long. Slip Coeff.	200,000	$[-]$
i_s	Steering Ratio	17	$[-]$

The P-ARB practically affects the vehicle dynamics by means of a constant enhanced roll stiffness, whereas the A-ARB adds a variable opposite torsional torque. The latter has been modelled using a PDD-type controller, in particular as a *Double-Lead Compensator* of the form

$$G(s) = K_p \frac{(s+a)(s+b)}{(s+c)(s+d)}. \tag{4}$$

The controller generates the correcting torque $M_{ARB,act}$ using the roll offset $\Delta\phi$ as an input. The choice of the type of controller is mainly justified by the need of a high level of responsiveness on the input command (being the manoeuvre very fast), a trend ensured by a high phase value in the Bode plot of the transfer function. All relevant data regarding the ARBs is shown in table II on the following page.

TABLE II
ANTI-ROLL BAR PARAMETERS: PASSIVE AND ACTIVE CASE

Symbol	Description	Value	Unit
K_{ϕ_f}	ARB_p Front Roll stiffness	10,000	$[kg \cdot m^2/s^2 \cdot rad]$
K_{ϕ_r}	ARB_p Rear Roll stiffness	7,500	$[kg \cdot m^2/s^2 \cdot rad]$
K_p	Proportional Gain	$1.7 \cdot 10^6$	$[-]$
a	Zero of G(s)	3	$[1/s]$
b	Zero of G(s)	1	$[1/s]$
c	Pole of G(s)	9	$[1/s]$
d	Pole of G(s)	7	$[1/s]$

C. Simulations

Efficiency of the ARB can be measured through the roll angle ϕ. As stated previously, decision upon the number and position of implemented ARBs is up to the car designer. However, it is useful to assess to what changes each design choice could lead. Figure 7 shows the effect of the ARB on the roll feedback (all simulations have been performed at an initial speed of 40km/h).

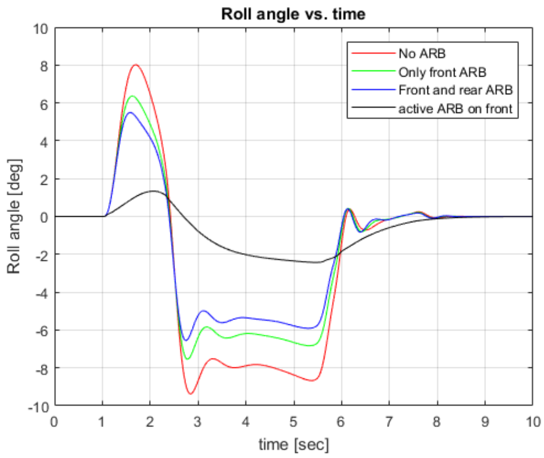

Fig. 7. Simulation results on the influence of both the P-ARB and A-ARB on the roll angle ϕ of the vehicle. As can be seen, implementation of an ARB reduces the body roll and jerkiness in body roll of the vehicle, improving the ride comfort. Four design choices are considered: (1) no use of ARB, (2) only P-ARB on the front of the vehicle, (3) P-ARB on both the front and the rear of the vehicle and (4) A-ARB on the front of the vehicle.

From figure 7, it can be seen that vehicle performance increases when one or two ARBs are applied. Most improvement in performance is obtained from application of an A-ARB, which at the same time decreases the copying effect. In addition, the A-ARB succeeds in dampening roll oscillations and therefore roll accelerations, improving the driving comfort of the vehicle. It has to be noted that in order to preserve an understeer tendency of the vehicle, it is highly recommended to preserve a certain ratio in roll stiffness of the front axle to the roll stiffness of the rear one.

Apart from the the direct effect that the implementation of the ARB has on the roll dynamics of the vehicle, it is very interesting to analyze how this reduction influences vehicle dynamics on a more general level. Looking at figure 4, it can be seen that roll over an angle $\phi \neq 0$ induces a misalignment of the CoG with respect to the mid-plane xz of the vehicle. Rotation of the body is performed about the (virtual) roll-axle of the vehicle, of which the position is mainly set through suspension settings. This misalignment of the CoG induces an uneven distribution of the normal loads on the tires, in favour of the outer ones that, coupled with tire characteristics of the form shown in figure 2, will induce an decrease of lateral force (i.e. loss of directional control). As a matter of fact a correct setting of the ARB can significantly improve this situation, reducing the overall load transfer of the vehicle. The effect of such implementations are shown in figure 8, making use of the concept of *Lateral Load Transfer Ratio* (LTR), expressed as

$$LTR = \frac{F_{zr} - F_{zl}}{F_{zr} + F_{zl}}$$

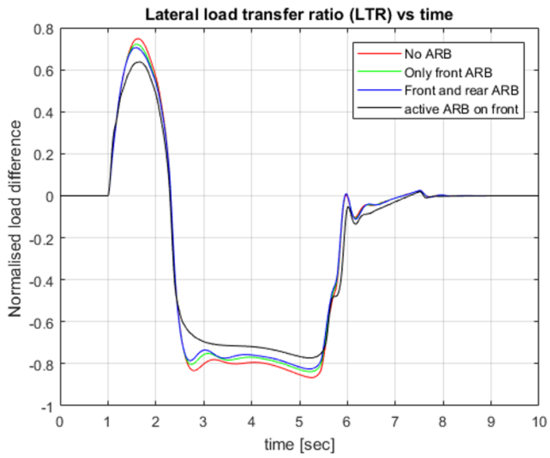

Fig. 8. Simulation results on the influence of both the P-ARB and A-ARB on the load transfer ratio of the vehicle. As can be seen, implementation of an ARB reduces the magnitude of the load transfer, improving vehicle handling due to larger possible lateral forces, but also dampens oscillations in the load transfer, thereby allowing a more constant contact patch with the road, thus increasing the road holding capacities of the tires. Four design choices are considered, see figure 7.

The improvement in lateral load transfer ΔF_y of the vehicle is again proportional to the number of ARBs assembled. However, improvements by using the A-ARB are not as high as in the case of roll angle investigation. The main contribution of the active bar is not only related to the mere reduction of the load transfer, but also to the related wheel attitude control. In virtue of the previous statement, the main advantage of application of an A-ARB is represented by the reduction of the normal load variations, a crucial property that affects the road holding performance of the vehicle. With reduced roll accelerations the tires are able to better follow the ground surface, maintaining a higher level of traction and lateral control.

Improved persistence of road contact translates into better continuity of the force transmission. This last detail is of crucial importance especially in challenging cornering manoeuvres, as the crash avoiding ones simulated in the fishhook manoeuvre. Thanks to the implementation of an ARB it is possible to achieve higher values of maximum lateral acceleration a_y in this manoeuvre. Relevant simulation results are shown in figure 9.

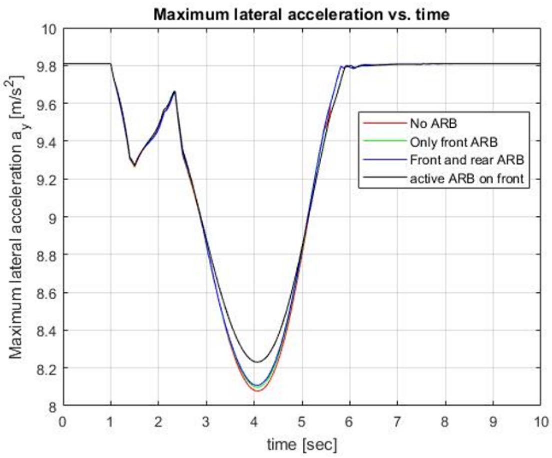

Fig. 9. Simulation results on the influence of both the P-ARB and A-ARB on the loss of maximum achievable lateral acceleration a_y of the vehicle (with respect to $1g$). As can be seen, implementation an A-ARB improves the maximum achievable lateral acceleration of the vehicle and therefore directional control, improving the ride handling. Four design choises are considered, see figure 7.

Once again the best performance is achieved with an A-ARB, thanks to which the natural loss of acceleration due to roll is significantly reduced.

IV. Conclusions

In this paper, starting from problem statement regarding road accidents caused by vehicle rollover, the need for both an effective, as well as cost-efficient, solution has been highlighted. Among all the alternatives proposed by the market the ARB has revealed to be the most cost-effective one.

On the basis of several assessment criteria (road holding, ride comfort and ride handling), the requisites for an effective improvement have been defined. Subsequently an 8 DOF vehicle model has been implemented to better catch the relevant dynamic parameters for the performance assessment, like body roll angle ϕ and normal load distribution. On the basis of these parameters, the simulation analysis of an emergency manoeuvre, the Fishhook test, has brought the conclusion that the overall dynamic performance of the vehicle improves on implementation of the ARB. In particular it has been observed that the improvement is directly proportional to the number of assembled ARBs. Furthermore, the problems intrinsically related to the purely mechanical nature of the P-ARB, like copying effect and oscillatory response of the vehicle, can be solved and further improved by means of an A-ARB. The A-ARB is capable of providing the right amount of correcting action when needed, in the form of compensating moment. This smart implementation results in a less stringent trade-off in between ride comfort and road holding, since a higher level of lateral acceleration is achievable, while at the same time keeping the overall jerk limited.

V. Recommendations

Many test procedures for testing vehicle performance are available. The test used in this report, the Fishhook test from the NHTSA, is mostly focused on evaluating light vehicle dynamic rollover propensity [7]. The test gives good insights in the maximum roll angle that the test-vehicle will achieve in different vehicle manoeuvres. Examples of other available test procedures are: the Moose or double lane change test, described in the ISO standard [8], which focus on handling performance in a more real-life situation in which an obstacle has to be avoided in a safe manner, and the J-turn, described in the ISO standard [9], which focuses on investigating lateral transient characteristics and the rough roads. Each has it's own (dis-)advantages. Combining different test procedures leads to better insights in vehicle performance in multiple driving situations.

Regarding the A-ARB, there are many types of control schemes available, including LQR, PID, PDD and many others. Each having multiple parameters to tune and different performance in different test scenarios. Other controllers also have to be implemented to be able to better capture the A-ARB effects on vehicle performances. Therefore, conclusions in this report are limited to qualitative obtained results instead.

Nowadays, ARBs are often implemented in vehicles with the purpose of changing under-/oversteer and aerodynamic characteristics. Whereas the report mainly focused on investigation on possible improvements in the road holding, ride handling and ride comfort of the vehicle. Further research is able to give more complete insights into the possible gains regarding these topics.

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Appendix
Nomenclature

a_y	Lateral acceleration	m/s^2
B	Track	m
C_{df}	Front Damping Coefficient	kg/s
C_{dr}	Rear Damping Coefficient	kg/s
C_f	Front Cornering Stiffness	$kg \cdot m/s^2$
C_r	Rear Cornering Stiffness	$kg \cdot m/s^2$
C_κ	Front/Rear Long. Slip Coeff.	$-$
F_x	Longitudinal force	N
F_y	Lateral force	N
F_z	Normal force	N
h	Distance CoG to roll center	m
h_{cg}	Height of CoG	m
h_f	Height of Front Roll Center	m
h_r	Height of Rear Roll Center	m
i_s	Steering Ratio	$-$
I_{xx}	Inertia along x-axis	kgm^2
I_{xz}	Coupling Inertia	kgm^2
I_{zz}	Inertia along z-axis	kgm^2
K_f	Front Spring Stiffness	kg/s^2
K_r	Rear Spring Stiffness	kg/s^2
$K_{\phi f}$	ARB_p front roll stiffness	$kg \cdot m^2/s^2 \cdot rad$
$K_{\phi r}$	ARB_p rear roll stiffness	$kg \cdot m^2/s^2 \cdot rad$
l_f	Front Wheelbase	m
l_r	Rear Wheelbase	m
m_s	Sprung vehicle mass	kg
r	Yaw	rad
r_w	Effective Wheel Radius	m
u		m/s
v		m/s
δ_{SW}	Steering wheel angle	deg
ϕ	Roll angle	deg