



Improving off-road vehicle handling using an active anti-roll bar

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Abstract

To design a vehicle's suspension system for a specific, well defined road type or manoeuvre is not a challenge any more. As the application profile of the vehicle becomes wider, it becomes more difficult to find spring and damper characteristics to achieve an acceptable compromise between ride comfort and handling. For vehicles that require both good on- and off-road capabilities, suspension design poses a significant challenge. Vehicles with good off-road capabilities usually suffer from poor on-road handling. These vehicles are designed with a high centre of gravity due to the increased ground clearance, soft suspension systems and large wheel travel to increase ride comfort and ensure traction on all the wheels. All of these characteristics contribute to bad handling and increased rollover propensity even on good level roads. It is expected from these vehicles to have the same handling characteristics as a normal on-road vehicle. This paper analyses the use of an active anti-roll bar as a means of improving the handling of an off-road vehicle without sacrificing ride comfort. The proposed solution is simulated, designed, manufactured, implemented and tested to quantify the effect of the active anti-roll bar on both the handling and ride comfort of an off-road vehicle.

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1. Introduction

The purpose of a vehicle's suspension system is to isolate the vehicle from the uncomfortable vibrations transmitted from the road through the tyres and to transmit the control forces back to the tyres so that the driver can keep the vehicle under control.

The popularity of sport utility vehicles (SUVs) has grown significantly over the past few years. Although mainly used on-road, SUVs are often designed for off-road conditions. This creates great concern over the handling and rollover of SUVs and opens an area for research.

According to Dukkipati et al. [9], SUVs contributed on average 32% to the number of rollover fatalities per million registered vehicles per year in the United States for the period 1985–1990. SUVs also contributed 10.7% to the total

number of fatal passenger vehicle crashes in the United States during 1988. This statistic increased to 25.6% for 2002. This is a dramatic increase in fatal crashes of SUVs and can be ascribed to the increase in popularity of SUVs over the years.

According to these statistics, it is concluded that the rollover of SUVs causes a safety concern that urgently needs to be addressed.

“Handling” is a widely used term. It can be expressed as the response of the vehicle to an input given by the driver through the steering wheel. This is a closed-loop control system. The driver determines the desired path of the vehicle and gives the input to the vehicle through the steering wheel. The vehicle responds to the input from the driver, road and surroundings and feedback is given to the driver who notes the response of the vehicle and corrects the input to achieve the desired path.

During high speed cornering, the tyres generate lateral forces that act on the vehicle. According to Newton's second law, this imposes a lateral acceleration on the centre of gravity (CG) point. Due to the fact that the height of CG (h_{CG}) is generally more than the height of the roll centre on a standard vehicle, the lateral force imposes a

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moment around the roll centre. This moment results in the body leaning to the outside during turning. This decreases the vertical loads on the inside wheels and increases the vertical loads on the outside wheels by the same amount. Due to the fact that the relationship between the lateral force and the vertical load on the tyre is a non-linear relationship, the change in vertical load between the inside and outside wheels of the vehicle results in a decrease in the lateral force that can be generated. Increasing the lateral load transfer (e.g. by using stiff springs or an anti-roll bar) decreases the lateral force that can be generated by the tyres [14].

To determine if a vehicle with rigid suspension will slide before it will roll on a flat road, the Static Stability Factor (SSF) is investigated. During rollover the rigid vehicle rotates around the contact point of the outside wheel. According to Forkenbrock et al. [12], the SSF around this point during cornering, if steady state conditions and rigid suspension is assumed, is given by:

$$SSF = \frac{t}{2h_{CG}}$$

where t = track width (m) and h_{CG} = height of CG point from ground (m).

It can be shown that if the SSF is larger than the friction coefficient between the tyres and the road, the vehicle will slide before it will roll and *vice versa*.

From this theoretical analysis of the lateral dynamics of a simplified linear vehicle model, it can be concluded that for improving the handling capabilities of a vehicle the lateral acceleration that can be generated by the vehicle must be increased. But, if the lateral acceleration is taken as the optimizing variable, the lateral force between the tyres and the road will increase, which will result in the vehicle rolling before sliding. This situation is undesirable, because it compromises the safety of the vehicle. This is the case with most vehicles with high CG's such as SUVs.

Uys et al. [25] conducted a study on what parameter(s) should be used to quantify and optimize the handling of a vehicle. The study strongly suggested that roll angle is a suitable variable to quantify handling. It is also suitable for the optimization of suspension settings given a prescribed road and manoeuvre.

The aim of improving the “handling” of a SUV, as treated in this study, is to reduce the roll over propensity of the vehicle or to design the suspension system so that the vehicle will rather slide than roll over.

Based on the recommendation by Uys et al. [25], the roll angle will be used to quantify and optimize the handling of the vehicle used in the current study. Reducing the roll angle during a handling manoeuvre requires an increase in the roll stiffness of the vehicle, which increases the vertical load transfer on the tyres. This will result in a lower lateral force that can be generated by the tyres, thereby reducing the risk of rollover. In other words, optimizing the roll angle of the vehicle improves the safety and reduces the rollover tendency of the vehicle.

Optimizing the roll angle during a handling manoeuvre can be achieved by one of the following three methods:

- (1) Lowering the CG height of the vehicle will increase the SSF, increasing the tendency of the vehicle to slide before it will roll. A lower CG point can be obtained by semi-active, slow-active or active suspension systems with ride height control.
- (2) Increasing the suspension stiffness and/or damping will reduce the body roll of the vehicle, thereby increasing vertical load transfer on the tyres and decreasing the lateral force between the tyres and the road. This can be achieved with passive, semi-active or active suspension systems.
- (3) An additional system can be added to increase the roll stiffness of the suspension. This will increase the vertical load transfer on the tyres and decrease the lateral force between the tyres and the road. This can be achieved with semi-active and active anti-roll bars, semi-active suspension and active suspension.

2. Literature study

From the literature various solutions to the ride comfort vs. handling compromise were obtained. These solutions include passive suspension systems, semi-active suspension systems, active anti-roll bars and active suspension systems.

The Land Rover Defender 110 test vehicle used in the current study is fitted with the four State Semi-active Suspension System, or 4S₄, developed at the University of Pretoria by Els [10]. This is a hydro-pneumatic suspension system designed to switch between two discrete spring characteristics, and two discrete damper characteristics. The switching is done by channelling hydraulic fluid by means of solenoid valves. The system switches between the soft setting (soft spring and low damping) for ride comfort and the stiff setting (stiff spring and high damping) for handling.

This system has been thoroughly tested and the stiff setting reduces the maximum body roll angle by 78% over the factory vehicle during a DLC (double-lane-change) manoeuvre at 70 km/h and by 90% over the soft setting. The soft setting performed similarly to the factory vehicle in terms of ride comfort on the Belgian paving and showed a 50–80% improvement in ride comfort over the stiff setting.

The control system automatically switches between “ride comfort” and “handling” modes by evaluating several vehicle parameters. Ride height control is also built into the system to level the vehicle when the suspension struts are loaded with gas [10].

Wilde et al. [27] proposed a passive interconnected suspension system. It consists of four hydraulic cylinders, two accumulators and connection pipes. This system counters body roll during handling manoeuvres and vehicle articula-

tion. It was tested on a Honda CRV by means of the National Highway Traffic Safety Association fishhook manoeuvre. The maximum speed the vehicle can manage without lifting two wheels two inches from the ground or a rim touching the ground was measured and used to determine the performance of the system. This system improved the vehicle's speed in the fishhook manoeuvre from 43 mph to more than 60 mph.

Kim et al. [18] describes Mando's continuously semi-active suspension system (SDC-20 model). This system controls the damping of the suspension between two settings by means of variable hydraulic dampers. The variable damping is used to control the response of the vehicle body during handling manoeuvres, severe road irregularities, nose dive, squat, yaw, pitch and heave. Different settings are also available such as auto, sport and comfort to change the ride experience. The aim of the design was to: (1) optimize the design of the control valve and to keep it compact and light with fast response and (2) to enhance the control performance, functionality, handling and safety aspects in co-operation with ESP (electronic stability program). Only simulation results of this system are presented. A handling simulation was done where a single cycle sinusoidal input with amplitude of 90° is given to the steering wheel. The roll velocity of the vehicle is taken as the measuring variable. The SDC-20 system showed up to a 50% improvement in peak roll velocity during the described test. Similar solutions are given by Ahmadian and Simon [1] and Yoon et al. [28].

Darling and Hickson [6] designed an active anti-roll bar system, which consisted of two anti-roll bars actuated by means of two hydraulic actuators. Simulations were done to predict the response of the system. Finally the system was fitted to a Ford Fiesta Mark II to verify the simulations. Two tests were done. The first test was a steady-state handling test, where the vehicle speed was increased while a constant steering angle was maintained. During this test the lateral acceleration on the vehicle ranged from 0.5 to 8 m/s^2 . The second test was a dynamic handling test. During this test the vehicle was driven at constant speed and step inputs, of magnitudes 90° , 180° , 270° and 360° , were applied to the steering wheel. The handling performance of the vehicle was rated by measuring the body roll angle during the test manoeuvres. The results showed an improvement in body roll during the steady state test of over 80% and an improvement in the peak body roll angle during the dynamic handling test of approximately 80%. Similar solutions are given by Everett et al. [11], Darling et al. [8], Cimba et al. [4], Danesin et al. [5], Sorniotti [21] and Kim and Park [17].

Shuttlewood et al. [20] proposed a non-linear model of a hydro-pneumatic suspension system. The suspension struts are actuated by moving oil between the accumulators and the struts by means of a hydraulic pump. The control system uses the lateral acceleration to determine the desired response of the system. Only simulations were done to acquire results of the proposed system. Simulation results

were not verified by actual tests. A J-turn manoeuvre is simulated up to a lateral acceleration of 0.8 g. The body roll angle was measured to quantify the handling of the vehicle. One of the problems experienced is that the roll movement of the body shows low damping and oscillates when the vehicle enters the turn. This could be reduced by moving the accelerometer forward a distance of 0.6 m in front of the CG. The simulations showed that the system improves the body roll angle by up to 80% during the J-turn manoeuvre. Similar solutions are given by Packer [19], Tillback and Brodd [23], Darling and Rosam [7] and Hubert and Kumar [15].

As SUVs are designed for on-road as well as off-road conditions, an improvement of handling on these vehicles should not be done at the cost of ride comfort.

The purpose of the present study is to implement a solution to improve the handling of an SUV without compromising the ride comfort over rough terrain. This is done by investigating the handling and the ride comfort of the test vehicle during a handling manoeuvre on a flat road as well as a handling manoeuvre on a rough road. No current literature could be found focusing on this topic.

Due to the fact that the test vehicle used for the current study is already fitted with the 4S₄, the two logical concepts worth additional investigation are:

- (i) By replacing the valves on the 4S₄ suspension with proportional valves and using an active control system, this system can be changed to an active height control system. The advantages of this system are that it can easily be adapted to not only do roll control, but also some form of slow-active or band-limited active suspension. High cost, complexity of the system and high pressure and flow requirements are drawbacks.
- (ii) Add an active anti-roll bar. The advantages of this system are low cost and low pressure and flow required, but this system is not very adaptable to improve other characteristics.

The rest of this study focuses on adding an active anti-roll bar to the existing 4S₄ suspension system to improve the handling of the vehicle without sacrificing ride comfort on rough roads. The following solution was generated: use the soft suspension of the 4S₄ (soft springs and soft dampers) to absorb the irregularities in the road and control body roll with an active anti-roll bar (AARB). Due to space constraints, the AARB will only be implemented on the rear of the vehicle.

3. Simulations

Simulations were used to test the feasibility of the active anti-roll bar, predict the results of the system and obtain some key design parameters.

An existing ADAMS/View 2005 model of the Land Rover 110 test vehicle, as developed by Thoresson [22]

and Uys [24], was used in this study. This is a non-linear full vehicle model and has 15 unconstrained degrees of freedom. The ADAMS/Controls interface was used to link ADAMS/View with Matlab and Simulink. Vehicle dynamics were solved in ADAMS while complex calculations for suspension characteristics and AARB control were done in Matlab and Simulink. The roll, pitch and yaw moments of inertia as well as the CG point of the vehicle was determined by experimental measurements as described by Uys et al. [26]. The test vehicle and simulation model use the 4S₄. The front suspension is modelled by means of a rigid axle which is fixed longitudinally by two leading arms connected to the body with rubber bushes. The stiffness of these bushes was measured and included in the ADAMS model. The rigid axle is fixed laterally with a Panhard rod. The rear suspension is modelled with a rigid axle with two trailing arms. An A-arm connected with a revolute joint to the body and a spherical joint to the axle fixes the axle in the lateral direction. The stiffness of the trailing arm rubber bushes is also included in the simulation model.

The simulation model was validated for ride comfort and handling by Thoreson [22] and for rollover by Uys [24]. Some of the characteristics of the initial simulation model were fine tuned for the present study to give a better representation of the test vehicle. These changes include: spring and damper characteristics, tyre characteristics, CG height and chassis torsional stiffness.

Simulations are performed by using the steering angle, vehicle speed and actuator displacement (only when the

ARB is passive and active), measured during vehicle tests, as inputs to the model. The measured data as well as the simulated results for a double-lane-change (DLC) manoeuvre at 70 km/h are compared in Fig. 1 (4S₄ on ride comfort setting) and Fig. 2 (4S₄ on handling setting). The four suspension displacements, lateral acceleration and average body roll angle were used to verify the model. The average body roll angle was defined as the average of the front and the rear body roll angle for each time step as calculated from the suspension displacement.

Good correlation is found, but it is noticed that the body of the simulated vehicle moves more easily than the body of the test vehicle. This is due to friction at the joints, connection points and seals that are not modelled in the simulation model, because the characteristics of these joints and connections points are unknown.

The validated simulation model is now used to evaluate the proposed active anti-roll bar solution. The ARB was modelled by two L shaped arms connected by a torsion spring representing the torsional stiffness of the ARB. A model of the ARB is shown in Fig. 3 and a schematic of how the ARB was modelled is shown in Fig. 4. The two L shaped arms were connected to the rear body with two revolute joints to allow rotation only in the Y-direction. At the right hand side of the ARB two dummy bodies with zero mass were inserted to be able to model the correct joint configuration. The end of the right L arm was connected to the bottom dummy body with a spherical joint. The two dummy bodies were then connected with a trans-

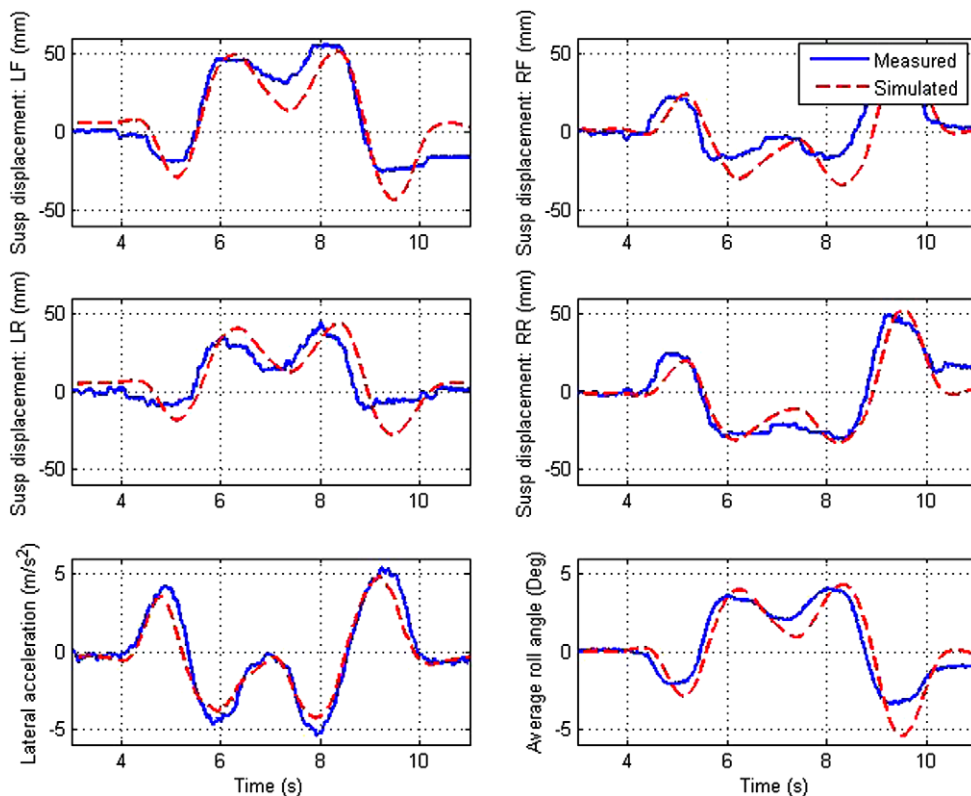


Fig. 1. Results of baseline vehicle and simulation during DLC manoeuvre with soft suspension (4S₄ on ride comfort setting).

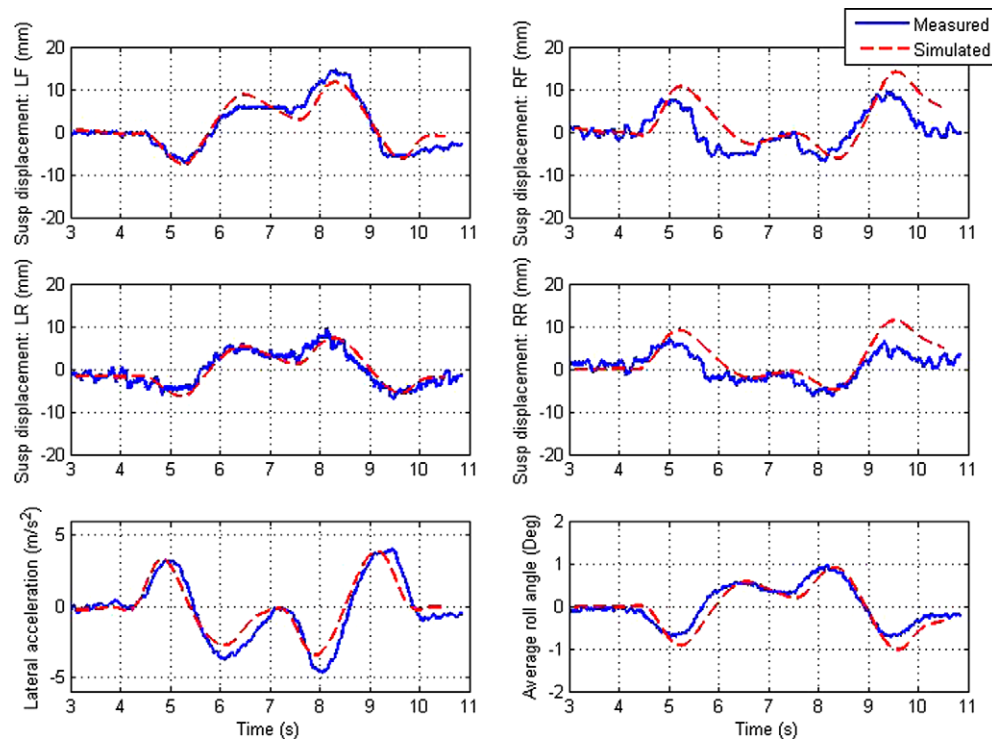


Fig. 2. Results of baseline vehicle and simulation during DLC manoeuvre with stiff suspension (4S₄ on handling setting).

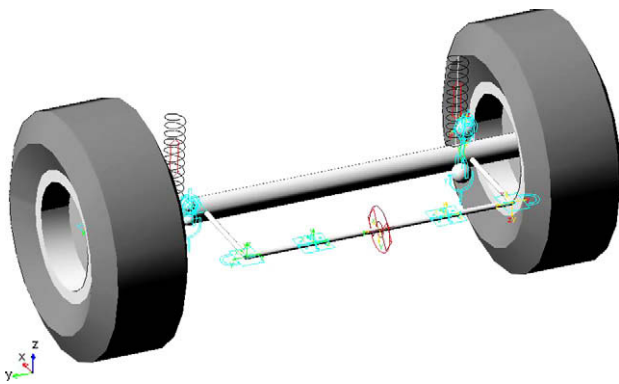


Fig. 3. Anti-roll bar modelled in ADAMS.

lational joint and a motion was imposed on this joint. This represented the actuator. The top dummy body was connected with a Hooks joint to the rear axle.

The left hand side of the ARB was connected in the same way as the right hand side with the only difference that the two dummy bodies were locked to each other. In other words, the two dummy bodies formed a linkage.

In order to simulate the proposed solution, a function was written in Matlab which calculates the desired displacement of the actuator. This was initially done by filtering the simulated lateral acceleration with a Bessel low pass filter (8th order, 6 Hz cut off frequency). After some tests, this filter was removed from the simulations as well as the test vehicle because the effect of the phase lag of the filter on the response of the system was too large. After filtering the lateral acceleration the signal was then limited to an

absolute maximum of 0.4 g. The limiter was implemented so that the AARB (Active anti-roll bar) system reduces the body roll angle to zero up to a lateral acceleration of 0.4 g and from there the body roll per lateral acceleration increases at the same rate as with the passive ARB. This was done to start warning the driver that the linearity limit of the vehicle is being approached. A value of 0.4 g was chosen because it is about half of the maximum lateral acceleration the vehicle can generate. It is therefore a safe limit and all the related solutions found in the literature were limited between 0.35 and 0.5 g lateral acceleration.

The filtered lateral acceleration was multiplied by a gain (26 mm/m/s²) to convert it to actuator displacement. The gain was obtained through trial and error with the simulations. Finally a velocity limiter was built in with a value of 0.3 m/s, to be able to limit the maximum velocity of the actuator. This was done to determine the maximum speed needed for the system to operate properly. The desired actuator displacement was then sent back to ADAMS to be used in the simulation.

It should be noted that both the 0.4 g lateral acceleration limit setting, as well as the gain (degree to which roll is cancelled out) are software settings and can therefore be changed easily based on the specific requirement, as long as the hardware capabilities are not exceeded.

In order to evaluate the proposed AARB solution, two road profiles were used, a flat concrete road and the Belgian paving as found at the Gerotek Test Facilities [13]. The profile of the Belgian paving as used in the simulations was measured by Becker [2].

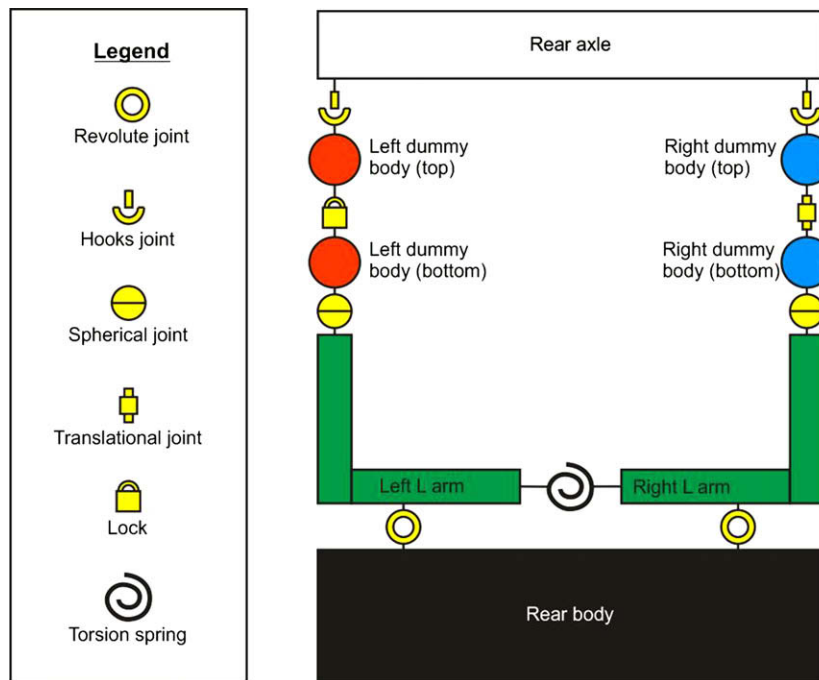


Fig. 4. ARB modelling schematic.

The simulations were done and the system was optimized until the best results were obtained. The results for the test vehicle with and without the AARB system on the soft suspension setting driving over the flat concrete road are shown in Fig. 5. The same simulations driving over the Belgian paving are shown in Fig. 6.

Fig. 5 shows an 80% improvement in body roll angle on a smooth road during a double-lane-change manoeuvre at 80 km/h. Fig. 6 indicates a 41% improvement on the

Belgian paving during a double-lane-change manoeuvre at 50 km/h.

The design specifications for the active anti-roll bar, shown in Table 1, were obtained from the initial simulations. These values were optimized with the capability of the hardware in mind, so that it did not result in unrealistic values being chosen.

Simulations were also done where a DLC manoeuvre was done on the Belgian paving to evaluate the effect of

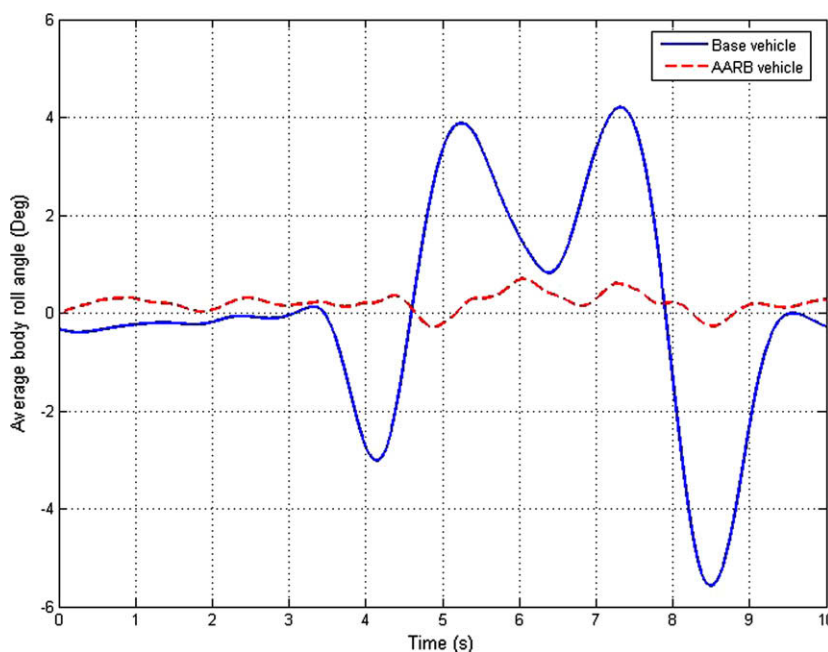


Fig. 5. Average body roll angle of the test vehicle during a double-lane-change manoeuvre at 80 km/h on a smooth road.

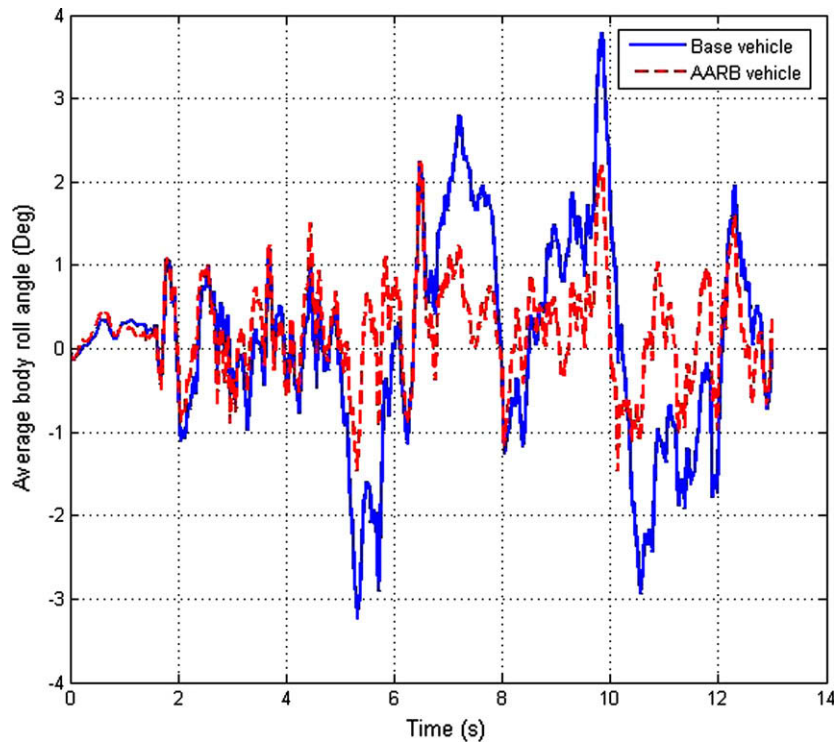


Fig. 6. Average body roll angle of the test vehicle during a double-lane-change manoeuvre at 50 km/h on the Belgian paving.

Table 1

Design parameters obtained from simulations.

Design parameter	Design value
Maximum torque in the ARB	1000 Nm
ARB torsional stiffness	85 Nm/°
ARB arm length	0.45 m
Actuator maximum speed	0.3 m/s
Actuator displacement	200 mm
Controller filters	Bessel low pass filter (8th order, 6 Hz cut off frequency)
Controller gains	26 mm/m s ⁻² (actuator displacement per lateral acceleration)
Controller dependant sensors	Lateral acceleration

the proposed system on the ride comfort of the vehicle during a handling manoeuvre. These simulations were done at a speed of 50 km/h, which is slow enough to represent a scenario of driving on a rough road and fast enough so that the handling manoeuvre has an effect on the dynamics of the vehicle. The road used for these simulations is the measured profile of the Belgian paving which is stretched by a factor of 2 over the width to obtain a 6 m × 100 m road. Unfortunately there is not a wide enough rough test track available to perform a correlation test.

It must also be noted that Becker [2] showed that the Pacejka '89 tyre model, as used in the current simulations, do not transmit frequencies higher than 8 Hz. Thus to calculate the ride comfort, the vertical acceleration is filtered by a 8 Hz low pass filter, then it is weighted according to the BS 6841:1987 standard [3] and finally the RMS is calculated to represent the ride comfort.

Initially simulations were done to ensure that the ride comfort over the stretched road correlated with the measured Belgian paving. These simulations showed a 0.3% difference in ride comfort, which is acceptable. Performing a DLC manoeuvre on the Belgian paving presented results as given in Table 2.

From Table 2 it can be seen that the differences between the ride comfort during the different simulations are negligible and that the AARB system does not have detrimental effect on the ride comfort of the vehicle even during a handling manoeuvre on a rough road.

4. Vehicle implementation

An active anti-roll bar, meeting the specifications set out in Table 1, was designed, manufactured and tested. A test setup was built in the laboratory to measure the torsional stiffness of the ARB and to calibrate the strain gauge which measures the torque in the ARB.

The measured stiffness of the ARB is 81.5 Nm/°. The difference between the measured and the calculated stiffness (89.9Nm/°) is due to the deflection of the arms and joints.

Table 2

Results of weighted RMS on vertical acceleration for Belgian paving test runs.

Simulation no.	Suspension setting	ARB setting	Weighted RMS
1	Soft	Disconnected	1.11 m/s ²
2	Soft	Connected	1.10 m/s ²
3	Soft	Active	1.13 m/s ²

This also shows that the deflection of the arms and joints contributes 9.3% to torsional stiffness of the whole ARB.

The ARB was fitted to the test vehicle (Fig. 7). The hydraulic cylinder was inserted so that it can move freely under all circumstances. The vehicle was fitted with a PC 104 form factor data acquisition and control computer. This computer was used to record all the measured data, switch the hydraulic pump and the hydraulic relief valve and compute the desired actuator displacement.

5. Testing

All the tests were done at Gerotek Test Facilities [13]. Outriggers were fitted to the test vehicle to prevent rollover and ensure safety.

5.1. Constant radius test

The constant radius test with a radius of 40 m was performed on a concrete surface. All the tests were done by slowly accelerating the vehicle from standstill. Vehicle speed was gradually increased until the outrigger hit the ground or it became impossible to maintain the constant radius. Six runs were done as shown in Table 3. The disconnected ARB means the ARB is bolted loose from the axle, the passive ARB means the hydraulic cylinder is kept at a fixed position during the tests and the active ARB means the system is fully active. The soft suspension setting means soft spring and low damping while the stiff suspension setting means stiff spring and high damping.



Fig. 7. The manufactured ARB.

Table 3
Constant radius test runs done.

Test no.	ARB setting	4S ₄ suspension setting
1	Disconnected	Soft
2	Disconnected	Stiff
3	Passive	Soft
4	Passive	Stiff
5	Active	Soft
6	Active	Stiff

The body roll angle vs. lateral acceleration for the three settings with the soft suspension are shown in Fig. 8 and with the stiff suspension in Fig. 9 (note different Y-axis scaling). The case with the disconnected ARB shows the most body roll. The case with the passive ARB shows significantly less body roll and the case with the AARB show zero body roll up to about 0.4 g. Above 0.4 g, body roll increases at the same rate as the connected ARB.

5.2. Double-lane-change-test

The DLC test was done with guidance from International Organization for Standardization [16]. A speed of 70 km/h was chosen for these tests, because it is about the limit where this test can safely be performed with all the suspension settings. The speed of the vehicle was kept as constant as possible by the driver and the same driver was used for all tests. The same configurations were used as for the constant radius test (Table 3).

The body roll angle vs. time of the three runs for the soft suspension (Fig. 10) and the stiff suspension (Fig. 11) were investigated (note different Y-axis scaling). For the soft suspension there was an improvement in maximum body roll angle between the disconnected ARB and the AARB of 74% and for the stiff suspension an improvement of 45%. This is a substantial improvement and very close to the 80% improvement predicted by the simulations. For the soft suspension setting the AARB shows a 55% improvement in maximum body roll over the passive ARB and an improvement of 40% with the stiff suspension setting.

5.3. Belgian paving test

Three runs were done on the Belgian paving as shown in Table 4. No tests were done on the stiff suspension setting, because the ride is highly uncomfortable. The measured vertical acceleration was weighted according to the BS 6841 standard [3] for vertical whole-body mechanical vibration on a seated person. The RMS of the weighted vertical acceleration was calculated to provide a value proportional to the ride comfort of the test. The results of these tests are given in Table 5.

From Table 5 it can be concluded that the AARB system has no detrimental effect on the ride comfort of the vehicle. This means that the AARB system can significantly improve vehicle handling, both on smooth and rough roads, without decreasing the ride comfort of the vehicle.

6. Conclusion

The present study successfully implemented an active anti-roll bar to improve the handling of an off-road vehicle without a detrimental effect on ride comfort.

An existing ADAMS model of the test vehicle (Land Rover Defender 110) was validated against double-lane-change-tests performed with the baseline vehicle. The ADAMS model was modified until good correlation was

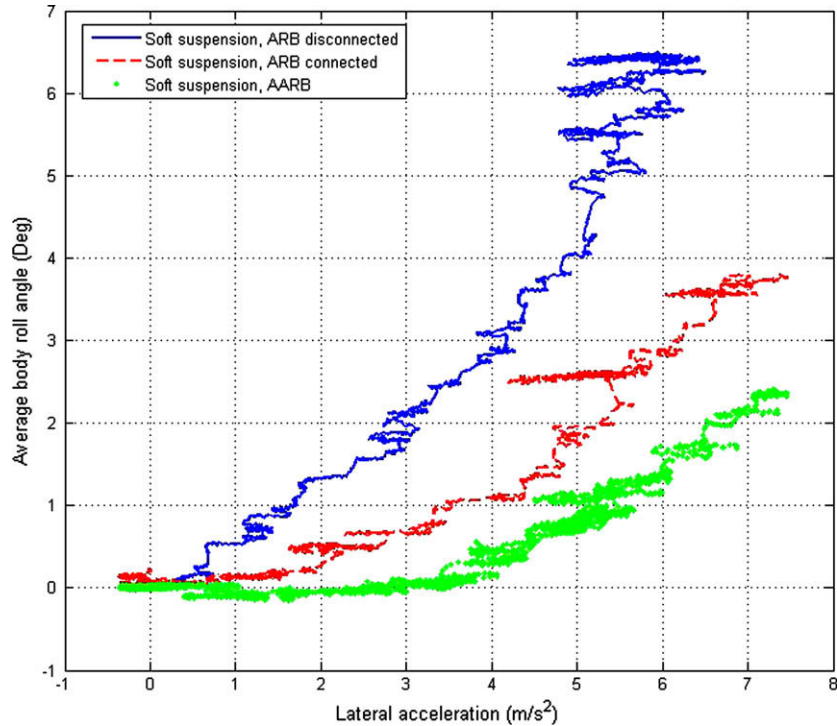


Fig. 8. Constant radius test: average body roll angle vs. lateral acceleration with soft suspension.

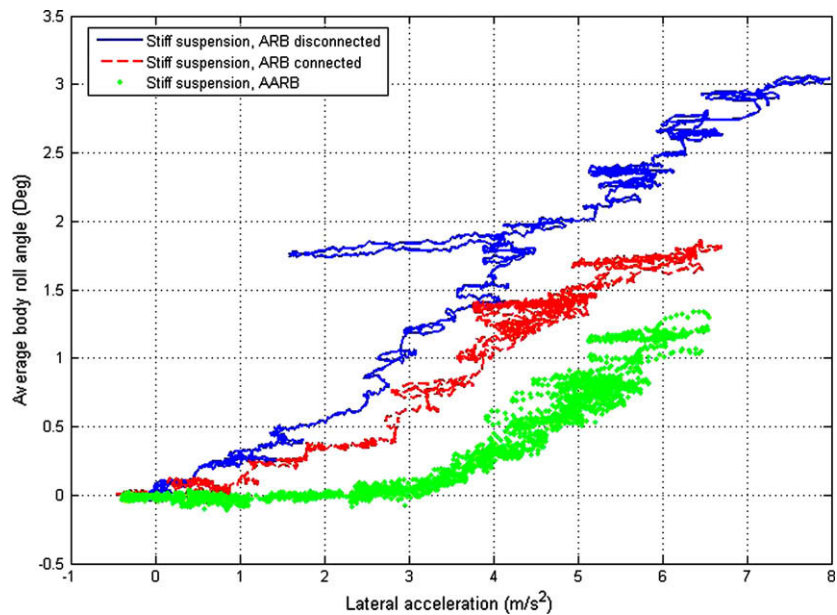


Fig. 9. Constant radius test: average body roll angle vs. lateral acceleration with stiff suspension.

achieved. The active anti-roll bar system was modelled in ADAMS to determine values for key design variables, designed, manufactured and tested.

Three types of tests were done to quantify any improvements in vehicle handling and ride comfort. Steady-state handling (constant radius) test results indicate that the AARB successfully eliminates body roll up to the pre-programmed 0.4 g lateral acceleration limit. Above 0.4 g lateral acceleration, body roll angle is still significantly

lower than that of the baseline vehicle, but increases at the same rate as for the baseline vehicle. This is done intentionally to give the driver advanced warning that the linearity limits of handling are being approached.

Dynamic handling (double-lane-change) test results, at a vehicle speed of 70 km/h, indicate improvements of between 40% and 74% in body roll angle depending on the settings of the test vehicles semi-active suspension system.

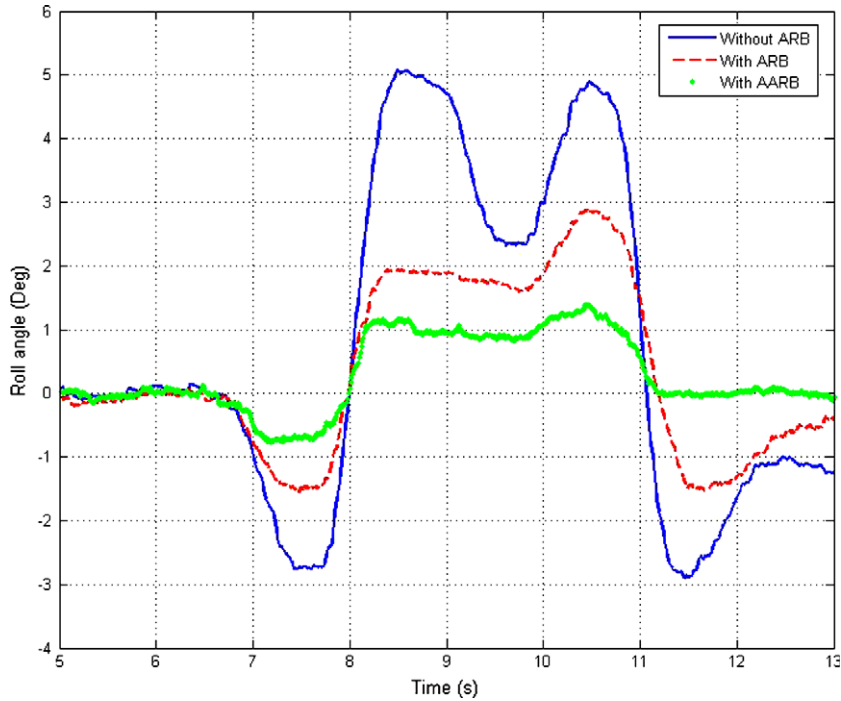


Fig. 10. DLC test: average body roll angle vs. time with soft suspension.

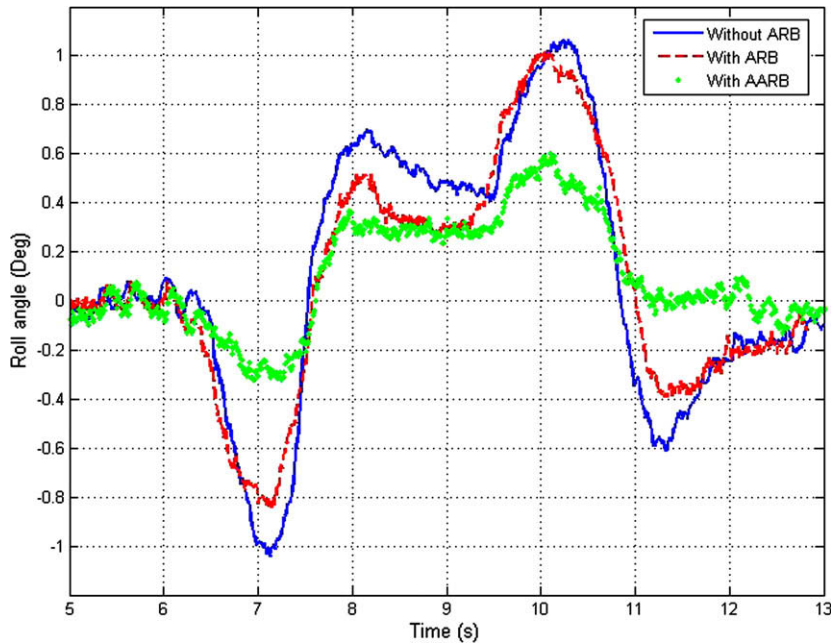


Fig. 11. DLC test: average body roll angle vs. time with stiff suspension.

Table 4
Belgian paving test runs done.

Test no.	ARB setting	Suspension setting
1	Disconnected	Soft
2	Passive	Soft
3	Active	Soft

Table 5
Results of weighted RMS on vertical acceleration for Belgian paving test runs.

Test run no.	Suspension setting	ARB setting	Weighted RMS
1	Soft	Disconnected	1.43 m/s ²
2	Soft	Connected	1.41 m/s ²
3	Soft	Active	1.44 m/s ²

Tests over Belgian paving indicates that ride comfort, based on BS6841 weighted RMS vertical acceleration, is not influenced by the AARB.

It is concluded that the active anti-roll bar system can dramatically improve the handling of an off-road vehicle without sacrificing ride comfort.

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