

Active Roll and Stability Control

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ABSTRACT

Computer Simulation was extensively utilized in the design and development of the Active Roll Control (ARC) system on LandRover 4X4 vehicle. An ADAMS model was developed integrating the electronic controller, hydraulic activation and vehicle model into one system of various degrees of complexity. Simulation results not only correlated well with vehicle test results, but also provided invaluable design guidelines crucial for solving key stability issues and successful product launch.

INTRODUCTION

The ARC system has been considered to offer relatively high benefit / cost ratio among different forms of active vehicle suspension systems. The ARC system reduces vehicle body roll at steady state cornering hence improves ride comfort level. For vehicles with passive anti-roll bar, its roll angle increases with lateral acceleration, resulting in excessive roll angle and reduced drivability at high lateral acceleration. In the ARC system, actuators can be inserted in various part of the passive bar, providing active anti roll forces and keeping the roll angle lower than that with a passive anti-roll bar (Fig 1).

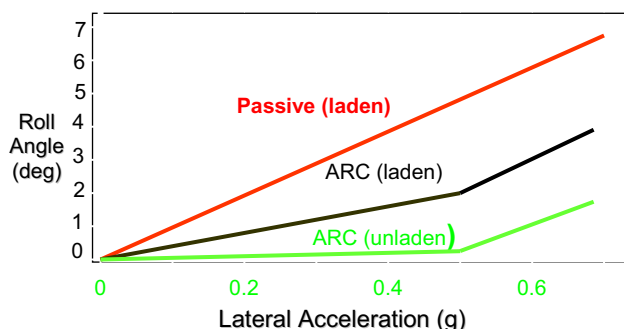


Figure 1 Passive and active roll

One type of ARC as shown in Figure 2 (a) replaces the vertical linkage by an actuator which changes its length at cornering. The second type shown in (b) inserts a rotational actuator at the center of the lateral bar which twists through certain angle at cornering¹. In the third type shown in (c), one longitudinal arm is pivoted on a

vertical lever which is the rigid extension of the lateral bar; the top end of the lever and the far end of the arm are connected by an actuator. Active roll control may also be achieved by so called "active suspension" adjusting the actuator length at the four corners.

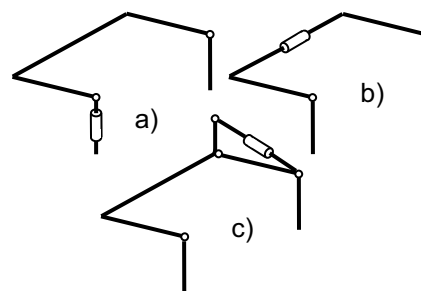


Fig. 2 c) Different type of ARC systems

The ARC system on the LandRover vehicle shown in Fig. 3 is of type (c). When activated, the extension or contraction of the actuator will push down or pull up the longitudinal pivoting arm, providing anti roll torque in either directions. The ARC is fitted on both front and rear axles which are of different width; their ratio is designed to provide optimal roll stiffness at nominal loading condition.

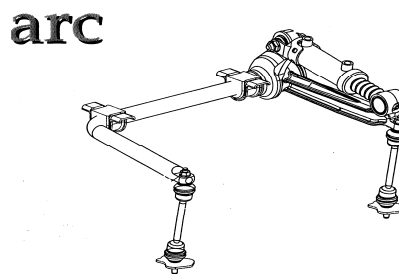


Fig. 3 ARC system on LandRover

Referring to Fig 4, the active roll stiffness K_r can be derived from the torsion bar stiffness K_t as $K_r = (L/l)^2 \times K_t$, where L is bar width and l is the distance between axle force and the bar.

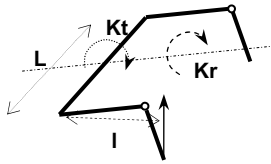


Fig. 4 ARC anti roll stiffness distribution on LandRover

The hydraulic valve block of ARC is shown in Figure 5, driving both the front and rear actuators. It consists of two directional valves and one pressure control valve, controlled by signals from an electronic controller. The four ports of the directional control valve are connected to the pump (P), tank (T), actuator top (A) and bottom (B) chambers respectively. The top chambers of the two actuators are permanently connected through the valve block, so are the two bottom chambers. In a straight driving condition, all three valves are fully open. Hydraulic fluid from the pump flow straight back to the tank through pressure control valve, resulting in a very low system pressure. The fully opened directional valves allow free actuator displacement to accommodate vehicle dynamic movement and road unevenness. In a cornering situation, one of the directional valves will be closed, depending on the turning direction. The pressure control valve is partially or completely closed, depending on the severity of the turn. This will force part or all of the pump fluid to go into relevant chambers, causing hydraulic pressure to rise to a desired level.

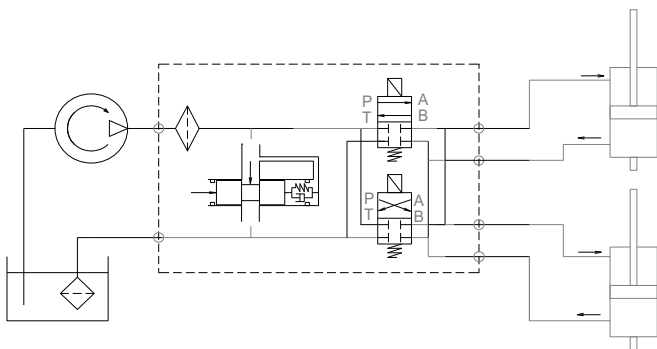


Fig. 5 Diagram of hydraulic circuit

During the development of ARC, test results of the prototype vehicle demonstrated good quasi-static performance at low frequency cornering. However for fast lane change, the anti-roll action provided by the ARC system seemed to lag behind, resulting in a fair amount of out-of-phase body roll. It was unclear whether this was due to insufficient pump/valve capacity, too much pipeline damping, compliance in the body/bushes, or deficient control logic. The performance of the ARC system under roll-rock road input and single bump input at turning was also unknown. In the latter case, the actuator may displace at high speed hence the lifecycle of the hydraulic seal is of concern if vacuum is created due to insufficient flow capacity. In the late development stage, the ARC vehicle showed some lift off oversteer tendency, threatening the successful launch of the product in time. During the same time period, there was

a growing concern in the general public, vehicle manufacturers and government regulators over vehicle rollover stability following high profile publications of hundreds of fatalities involving SUV rollover accidents in the US. Vehicle manufacturers were under pressure to understand the rollover mechanism and improve rollover resistance. The LandRover ARC project team needed to ensure that ARC system could improve rollover stability and prevent un-tripped rollover in forward and reverse driving conditions. All these required the development of a comprehensive computer simulation model to assist the ARC project work.

COMPUTER MODEL

The Design Analysis Department was requested to provide technical support in developing an integrated ADAMS model of LandRover vehicle fitted with the ARC system, optimizing the control algorithm and predicting performance in all relevant maneuvers. The integrated ADAMS model is a combination of detailed vehicle model, electronic controller and hydraulic activation system.

Vehicle Model

The LandRover vehicle model is represented in ADAMS by standard components such as mass, damper, spring, force, tires etc. A generic, detailed and parameterized vehicle model was created over years of development work. Various components can be inserted, replaced, swapped or modified easily. Detailed information of the vehicle model can be found in ADAMS user manuals. For modeling the active anti-roll force, a single directional force is applied between the two connection points on the lever and longitudinal rotating arm (Fig. 2c), with amplitude proportional to the actuator pressure.

As an excellent tool for detailed vehicle modeling and dynamic analysis, ADAMS did not have standard components and blocks for hydraulic systems and electronic controllers at the time of this study. The Electro-hydraulic ARC system then had to be described by differential equations, transfer functions, user subroutines, etc. in the integrated model.

Electronic Controller Model

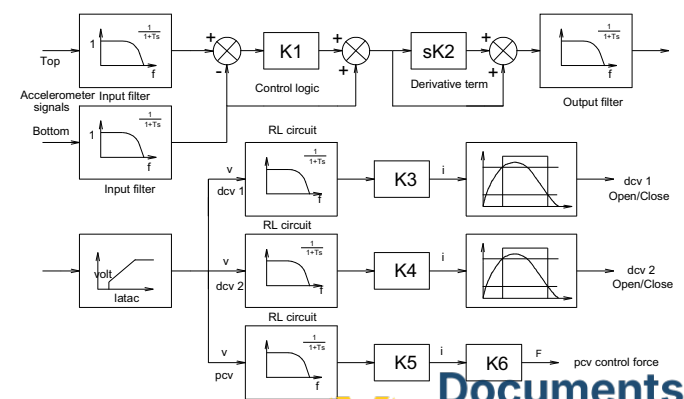


Fig. 6 Diagram of the ARC controller

The model of the electronic control unit for the hydraulic system shown in Fig. 6 consists of filters, a differentiator, a lookup table, dead band, and hysteresis etc. The controller manipulates the input acceleration signals and provides output control forces to activate the hydraulic valve block. The lateral acceleration signals come from the top and bottom accelerometers located on the roof and chassis respectively. The signals are filtered by first order low-pass filters at appropriate cutoff frequencies to eliminate noise. The bottom accelerator provides the basic control signal, which is then modified by a fraction of the difference between the top and bottom lateral accelerations, in order to reduce body roll in transient situation. The combined signal is then modified by a derivative term to provide phase advance. A second first order low-pass filter is applied to eliminate parasitic noise generated during differentiation.

The voltage signals are derived from the filtered lateral acceleration based on a lookup table. At very low lateral acceleration, the output voltage is zeroed to eliminate noise and false activation during straight driving condition. At acceleration levels higher than the dead band, the voltage signal increases linearly with acceleration, and is top limited to a saturation level. The voltage signal is then converted to current through an R-L circuit, described by the following differential equation:

$$L_{in} \frac{di}{dt} + R_e i = v$$

where L_{in} is inductance and R_e the resistance of the electric circuit. The transfer function between current and voltage is then:

$$\frac{i}{v} = \frac{1/R_e}{sL_{in}/R_e + 1}$$

A control force signal proportional to the currents is applied to the valve block. Two solenoid forces turn on/off the directional valves with hysteresis. If the force is below the lower threshold, the direction valve will be closed. If the force is above the higher limit, the valve will be open. If the force is in between, the valve will remain unchanged. The third solenoid force opens or closes the pressure control valve, changing its orifice size which in turn controls the flow rate and the hydraulic pressure.

Hydraulic System Model

To model the hydraulic system, the complete circuit is divided into eight regions or controlled volumes, as shown in different colors in Fig. 7. The pressure in each region is assumed uniform, described by a single differential equation. All regions are connected by either orifices flows or pipeline flows with arrows indicating the positive flow direction. Laboratory tests were also carried out on the valve block to extract empirical relationships

between pressures, flow rates and valve opening characteristics used in the model. The ARC model is fully parameterized, capable of quick parametric study.

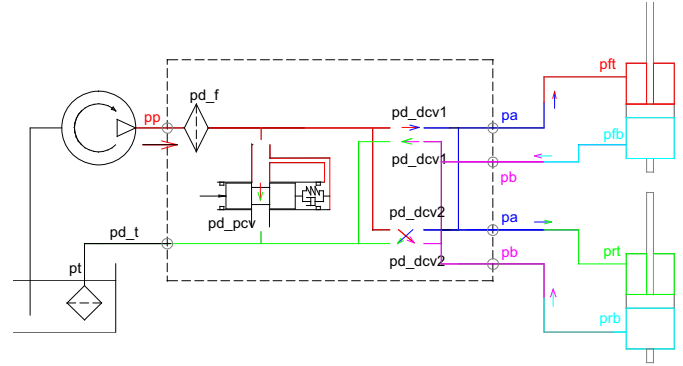


Fig. 7 Control volumes and flows in ARC hydraulic circuit

For each controlled hydraulic volume, the total input and output flow rates contribute to changes in both pressure and volume, assuming a constant temperature. This relationship can be described by the general continuity equation²:

$$\sum Q_{in} - \sum Q_{out} - \sum Q_{leak} = \frac{V\dot{P}}{\beta} + \dot{V}$$

where V and $\dot{V}$ are controlled volume and rate, P and $\dot{P}$ are pressure and its rate, β is the effective system bulk modulus (including entrapped air and mechanical compliance of chambers). The leakage term is ignored as test results have shown an insignificant effect on ARC system performance.

The flow rate through the orifice of pressure control valve in Fig. 8 is calculated by:

$$Q_o = C_d w x \sqrt{\frac{2(P_1 - P_2)}{\rho}}$$

where C_d is the discharge coefficient based on the orifice shape, x is valve displacement, w is area width, P_1 and P_2 are pressures at up and down streams, ρ is mass density.

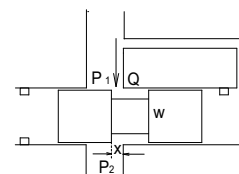


Fig. 8 Orifice flow rate

The flow rate Q through a pipeline as shown in Fig. 9 is calculated by:

$$Q = \frac{\pi D^4}{128 \mu L} (P_1 - P_2)$$

where μ is the friction coefficient, D and L are pipe ID and length, P_1 and P_2 are pressures at up and down streams.

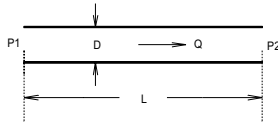


Fig. 9 Pipeline flow rate

Due to lack of detailed information, the pressure control valve is represented in the model by an idealized mass-spring-damper system shown in Fig. 10.

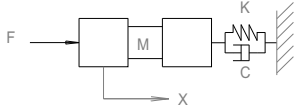


Fig. 10 Pressure control valve model

Assuming an equivalent spool valve mass M, the values of equivalent stiffness K and damping C were estimated based on optimal system response. The system can be described by the following 2nd order differential equation:

$$M\ddot{X} + C\dot{X} + KX = F$$

The transfer function between valve displacement X and driving force F is then written as:

$$\frac{X}{F} = \frac{1}{Ms^2 + Cs + K} = \frac{1/K}{\frac{M}{K}s^2 + \frac{C}{K}s + 1} = \frac{1/K}{\frac{s^2}{\omega^2} + \frac{2\xi s}{\omega} + 1}$$

where $\xi = 0.7$ is the critical damping, $\omega = 2\pi f$, f is the resonant frequency sufficiently high to meet the requirement for response speed. K can then be calculated by $M\omega^2$, and C by $2\xi K / \omega$. Force F is the summary of the electronic control force, the pressure feedback force, bump stop forces etc. The pressure feedback force is calculated by $P_{pp} \times A_v$ where P_{pp} is pump circuit pressure, A_v is the end area of spool valve. Bump stops are added in the model at both ends of the spool valve to prevent the spool from over-shooting.

Simplified ARC Model

In computer simulations, a physical system can often be described by either simplified or detailed mathematical models, depending on the objective of the study. From the experience of the author, a simple model should always be the first choice unless the answer may only be revealed by more complex models. Even if a detailed model is necessary, the simple model can still be useful particularly at the beginning of the work to provide guidelines for detailed model, to obtain quick answers at high level, or to correlate results of different models.

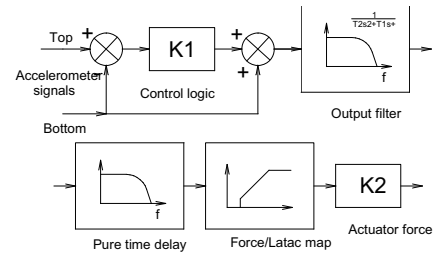


Fig. 11 diagram of the simplified ARC model

One version of the simplified ARC model shown in Fig. 11 was integrated into the vehicle model to provide quick analysis of the effect of controller time delay on ARC performance. The model does not include the hydraulic circuit but only an actuator force proportional to the control signal and applied at vehicle points with certain time delay. The controller was simplified as a 2nd order low pass filter followed a pure time delay representing the dynamics and delay in the whole electro-hydraulic system. The pure time delay is represented by a fourth order Pade approximation of the following form:

$$e^{-Ds} \doteq \frac{a_0 - a_1s + a_2s^2 - a_3s^3 + a_4s^4}{a_0 + a_1s + a_2s^2 + a_3s^3 + a_4s^4}$$

where D_i is the pure time delay in second, and the constants are defined as:

$$a_0 = 1072 / D_i^4, \quad a_1 = 536 / D_i^3, \quad a_2 = 120 / D_i^2, \\ a_3 = 13.35 / D_i, \quad a_4 = 1.$$

RESULTS CORRELATION AND ANALYSIS

Simulation was carried out in time domain to predict ARC system performance in the following maneuvers:

- steady state cornering with single wheel input
- sine and swept sine steer
- roll-rock surface
- tilting table
- "KU" swerve
- reverse steer
- lift off (turn in) over-steer

Steady state cornering with single wheel input

In this maneuver, the vehicle travels straight ahead at constant speed for one second and then turns right with a step steering arm input of 2.5 degree in 0.1 second. After holding this position for certain time, the outside (left) wheel hits a trapezium-shaped bump of 20 mm high, 1m long at top and 2m at the bottom. When the wheel hit the single bump, its vertical movement is transferred to a significant ARC actuator displacement, resulting in the compression of one chamber and expansion of the other with consequent pressure