XLVI CILAMCE
VITÓRIA - ES - BRAZIL - 2025
XLVI IBERO-LATIN AMERICAN CONGRESS ON COMPUTATIONAL METHODS IN ENGENEERING

Comparative analysis of passive and active anti-roll bar strategies for vehicle roll stability

Vitor Machado de Toledo[1], Rafael Rodrigues da Silva[1], Suzana Moreira Avila[1]

[1]*Faculdade do Gama, Universidade de Brasília*
St. Leste, Gama Leste Projeção A - Pte. Alta Norte (Gama), 72444-240, Brasília/DF, Brasil
vitormttoledo@gmail.com, rafael.rodrigues@unb.br, avilas@unb.br

Abstract. Vehicle roll stability is a critical aspect of automotive safety and handling, particularly under high lateral acceleration conditions that may lead to rollover. This study compares passive and active anti-roll bar (AARB) strategies in a virtual 14-degree-of-freedom vehicle model developed in VI-CarRealTime (VI-CRT). The passive anti-roll bar serves as a baseline, while two active configurations, one controlled by a PI controller and the other by a LQR controller, apply controlled torques to reduce body roll. The control logic was implemented via co-simulation with MATLAB/Simulink, interfacing with the full vehicle model. Controller performance was primarily evaluated using the Fishhook manoeuvre, as specified by the NHTSA dynamic rollover resistance test protocol, which imposes severe lateral loading. Performance metrics included peak and RMS roll angles and actuator torque demand. The findings reinforce the advantages of model-based state-feedback control for enhancing vehicle stability and reducing rollover risk in dynamic driving conditions.

Keywords: Active anti-roll bar, Vehicle dynamics, PI, LQR, Roll stability.

1 Introduction

Ensuring vehicle roll stability is fundamental for maintaining control and passenger safety during extreme steering events and high lateral acceleration scenarios. Excessive lateral load transfer under such conditions can lead to inner-wheel lift and, in severe cases, rollover, a phenomenon associated with a high fatality rate in road accidents [1].. Regulatory initiatives such as FMVSS 216 and Euro NCAP stability protocols have therefore emphasised the need for dynamic loading assessments to verify resistance to excessive roll and tyre unloading [U.S. Department of Transportation].

Passive anti-roll bars (PARBs) have traditionally been employed to couple suspension arms torsionally across an axle, effectively reducing both transient and steady-state roll. However, their fixed stiffness imposes an inherent compromise between handling stability and ride comfort [3]. Additionally, passive systems can increase understeer and reduce transient responsiveness due to lateral load transfer during cornering.

The need for adaptability has driven the development of active anti-roll bar (AARB) technologies, which employ electromechanical or hydraulic actuators to apply controlled torques, dynamically adjusting roll stiffness in real time. These systems have shown the potential to mitigate roll angle, optimise load distribution, and enhance vehicle response during rapid steering manoeuvres such as lane changes and slalom driving [4, 5]. The overall effectiveness, however, depends strongly on the chosen control strategy.

Proportional–Integral (PI) controllers remain prevalent due to their simplicity and ease of integration into automotive ECUs [6]. However, the coupled yaw–roll dynamics of modern vehicles suggest that full-state feedback techniques, such as Linear Quadratic Regulator (LQR), may provide superior roll suppression and energy efficiency [7, 8]. While both approaches have seen application in suspension and stability control, comparative studies using consistent modelling and standardised manoeuvres are limited.

This study addresses this gap by evaluating and comparing PI and LQR AARB strategies using a virtual 14-degree-of-freedom vehicle model in VI-CarRealTime (VI-CRT). Controllers were integrated via co-simulation with MATLAB/Simulink for real-time torque application. Analysis focuses on the Fishhook manoeuvre, defined by the NHTSA dynamic rollover resistance protocol, and assesses peak and RMS roll angles, roll rate, and control torque, thereby clarifying the trade-offs between control simplicity and optimality in next-generation chassis systems.

2 Vehicle modelling

The virtual vehicle model was developed using VI-CarRealTime (VI-CRT), a high-fidelity simulation platform widely adopted for real-time vehicle dynamics analysis. The model features 14 degrees of freedom (DOF), encompassing five rigid bodies: the sprung mass (chassis) and four unsprung masses (wheels). The chassis provides six DOF for translational and rotational motion, while each wheel has two DOF for suspension travel and rotation. Powertrain, braking, and steering subsystems are modelled through algebraic and differential equations, as documented in VI-CarRealTime [9].

Default suspension definitions were retained to focus the analysis on the influence of AARB control on the lateral and roll dynamics of a mid-size sedan. The main parameters used in the simulations, sourced from the default mid-size sedan template in VI-CarRealTime, are summarised in Table 1.

Table 1. Vehicle and suspension parameters

Parameter	Value	Description
m_s	1760 kg	Sprung mass (vehicle chassis)
m_{uf}, m_{ur}	61 kg, 53 kg	Front and rear unsprung masses
I_{xx}	527.93 kg·m^2	Sprung mass roll inertia
I_{zz}	2943.61 kg·m^2	Yaw inertia
I_{xz}	0.059 kg·m^2	Yaw–roll inertia product
h_s	0.626 m	Sprung mass CG height
h_u	0.300 m	Unsprung mass CG height
w	2.874 m	Wheelbase
l_f, l_r	1.332 m, 1.542 m	CG distances to front and rear axles
$c_{\phi f}$, $c_{\phi r}$	6.121 N·m·s/rad, 3.162 N·m·s/rad	Roll damping (suspension) front and rear
$k_{AO,f}$, $k_{AO,r}$	2.500 Nm/rad, 1.500 Nm/rad	Target roll stiffness of AARB (front, rear)

The AARB system was integrated through a co-simulation framework with MATLAB/Simulink. While VI-CRT resolves the complete 14-degree-of-freedom vehicle dynamics, including vertical load transfer and body roll states, Simulink implements the control logic for the AARB, either via a Proportional–Integral (PI) controller or a Linear Quadratic Regulator (LQR). The resulting control torques were applied in real time to the front and rear anti-roll bars as external moments. This configuration allows direct evaluation of each control strategy's effect on roll suppression and lateral stability.

The design of the LQR controller was based on the yaw–roll reduced-order formulation presented by Vu et al. [4]. The PI controller, in contrast, acts directly on the roll angle signal within VI-CRT without relying on a reduced-order model.

3 Control design

The active anti-roll bar (AARB) control logic was implemented in MATLAB/Simulink and integrated with the high-fidelity vehicle model in VI-CarRealTime. This co-simulation framework, depicted in Figure 1, allows for the real-time application of control torques to the vehicle's front and rear axles based on its dynamic states. Two distinct control strategies were developed and compared: a model-based Linear Quadratic Regulator (LQR) and an optimised Proportional-Integral (PI) controller.

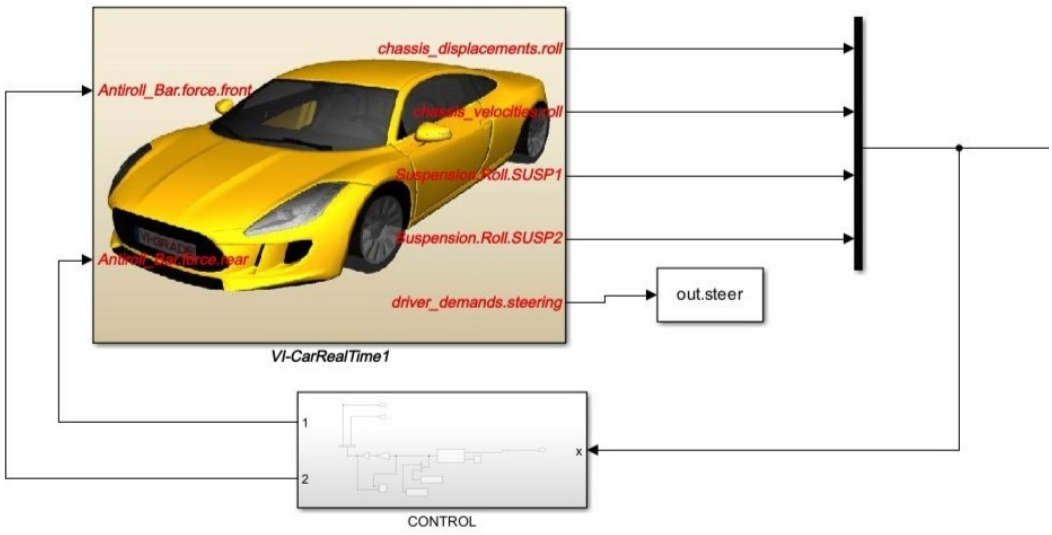


Figure 1. Block diagram implemented in MATLAB/Simulink.

First, the Linear Quadratic Regulator (LQR) was designed as a modern, state-feedback optimal control approach. The controller's objective was to minimise a quadratic cost function, J, which penalises both state deviations and control effort. The weighting matrices, Q (for states) and R (for inputs), were systematically determined using Bryson's Rule. This method scales the weights based on the maximum acceptable values for the states and control inputs. The maximum state values (x_{max}) and actuator torques (u_{max}) were defined as:

$$x_{max} = [0.087, 0.5, 0.052, 0.052]^\top, \quad u_{max} = [2000, 2000]^\top.$$

The final matrices were then tuned to heavily prioritise roll angle suppression, resulting in the following formulations:

$$Q = \text{diag}(50, 1, 1, 1) \cdot \text{diag}(1./x_{max}^2),$$
$$R = 0.34 \cdot \text{diag}(1./u_{max}^2).$$

By solving the continuous-time Algebraic Riccati Equation (ARE), the optimal state-feedback gain matrix was computed from these weights [7].

In contrast, the second strategy employed a more classical Proportional-Integral (PI) controller, with the derivative term set to zero ($D = 0$). Rather than relying on manual tuning, the PI gains were determined through a systematic optimisation process. A randomised parameter sweep was conducted over a predefined grid of possible values for the proportional (K_p) and integral (K_i) gains. The optimisation objective was to find the gain combination that minimised the Root Mean Square (RMS) value of the sprung mass roll angle. The optimal gains yielding the best performance were found to be:

- Front Axle: $K_p = 105,000$, $K_i = 1500$.
- Rear Axle: $K_p = 110,000$, $K_i = 1500$.

4 Simulation methodology

All simulations used a co-simulation architecture combining MATLAB/Simulink and VI-CarRealTime (VI-CRT). VI-CRT resolved the full 14-DOF vehicle dynamics; Simulink provided the active anti-roll bar (AARB) control logic. A fixed-step solver (10 ms) ensured stability and accuracy during transients.

The Fishhook manoeuvre, as per the NHTSA protocol, was adopted for its capacity to induce severe lateral load and roll. The steering profile (Figure 2) represents a double-swerve.

Two active configurations were evaluated: (a) PI-AARB and (b) LQR-AARB, under identical conditions. The passive ARB served as the baseline for all percentage-based comparisons.

Performance was assessed via peak and RMS sprung mass roll angle and peak actuator torque. This approach ensures all differences reflect the control strategy alone.

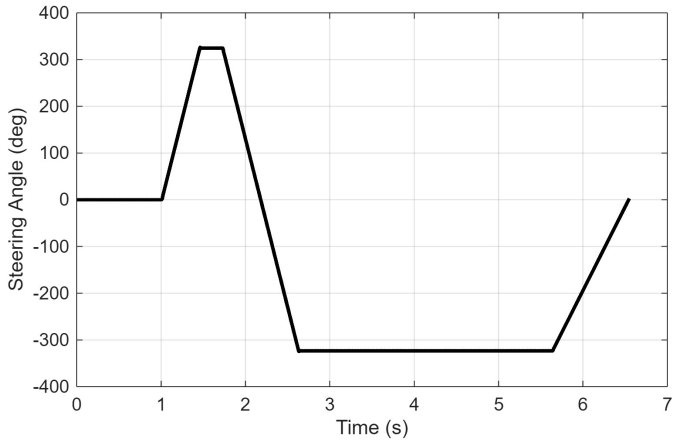


Figure 2. Steering for Fishhook manoeuvre

5 Results

This section presents the performance of the PI and LQR active controllers compared to the passive baseline during the demanding Fishhook manoeuvre. Key metrics, including roll angle and actuator torque, are analysed to highlight the effectiveness of each strategy.

First, the LQR-AARB system demonstrated a highly effective response. As shown by the sprung mass roll angle in Figure 3, the controller achieved a smooth transient response with minimal overshoot, effectively suppressing body roll.

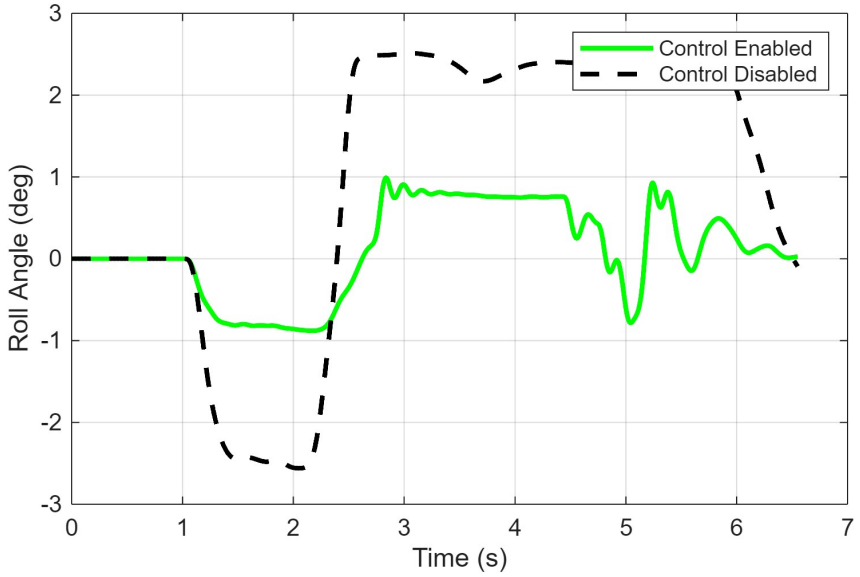


Figure 3. Sprung mass roll angle for LQR-AARB

Figure 4 shows the actuator torques for LQR-AARB, peaking at 3040 Nm with balanced front and rear axle loads.

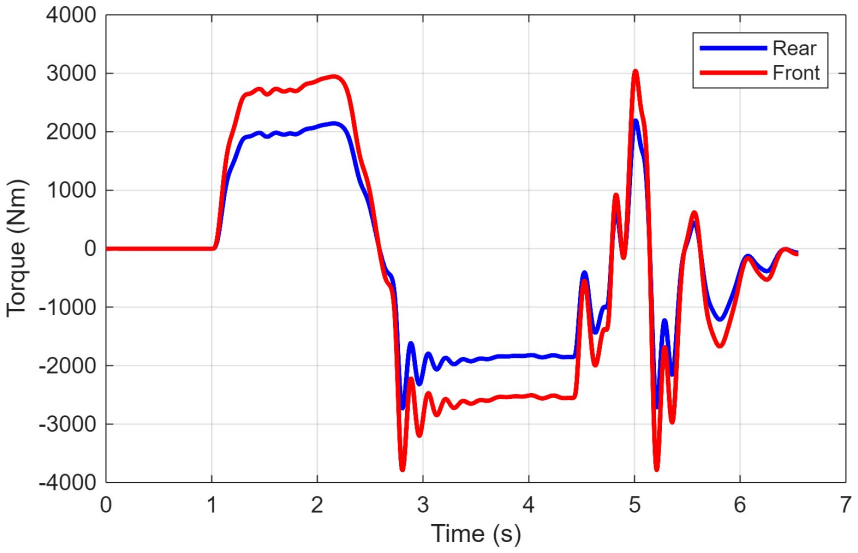


Figure 4. Actuator torques for LLQR-AARB

Figure 5 illustrates the suspended mass roll angle for PI-AARB. Although the roll was reduced, oscillations were more pronounced than with LQR-AARB.

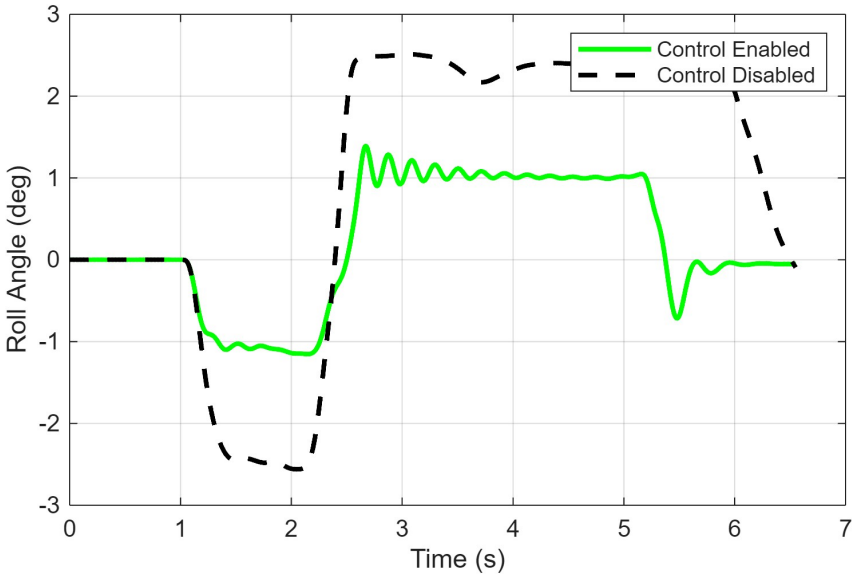


Figure 5. Sprung mass roll angle for PI-AARB

Figure 6 shows actuator torques for PI-AARB, reaching a peak of 2237 Nm and showing less balanced axle distribution than LQR-AARB.

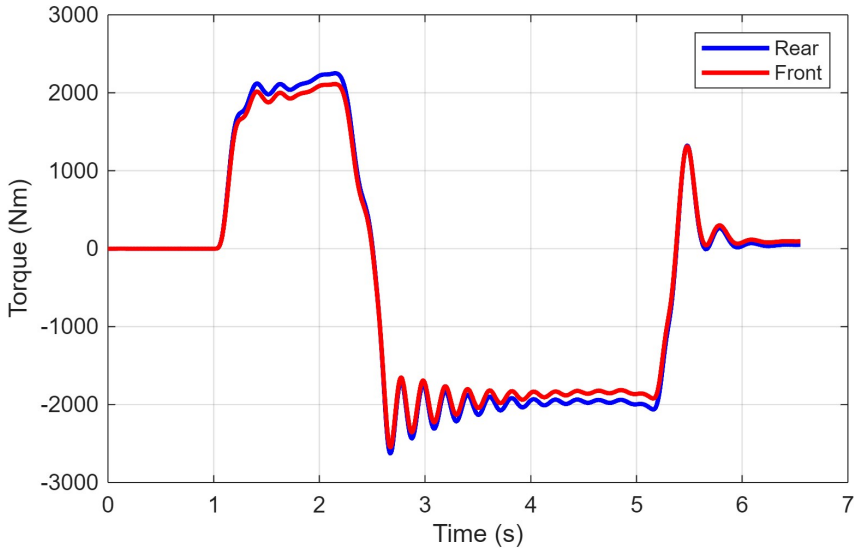


Figure 6. Actuator torques for PI-AARB

Table 2 summarises RMS and peak roll angles and peak actuator torques, with percentage reductions relative to PARB.

Table 2. Roll performance metrics during Fishhook manoeuvre.

Configuration	**RMS Roll (deg)**	**Reduction (%)**	**Peak Roll (deg)**	**Peak Torque (Nm)**
PARB	2.02	–	2.509	–
PI-AARB	0.81	60	1.391	2237
LQR-AARB	0.58	71	0.990	3040

Both were compared to the passive anti-roll bar (PARB), which served as the baseline. LQR-AARB achieved the lowest roll angles and smoother dynamics, while PI-AARB required lower peak torque but had more oscillations.

6 Conclusions

This study evaluated two active anti-roll bar (AARB) control strategies—PI and LQR—using a co-simulation approach with VI-CarRealTime and MATLAB/Simulink for a mid-size sedan. Performance was assessed during the Fishhook manoeuvre, with a passive anti-roll bar as the reference for comparisons.

Both PI and LQR controllers effectively enhanced roll stability. The PI-AARB reduced the RMS roll angle by 60%, from 2.02 deg to 0.81 deg, while the LQR-AARB achieved a 71% reduction, to 0.58 deg. Peak roll angles were similarly reduced, with PI achieving 1.391 deg (a 45% reduction) and LQR achieving 0.990 deg (a 61% reduction).

The LQR controller required a higher maximum actuator torque (3040 Nm) than the PI controller (2237 Nm). Nevertheless, the torque distribution of LQR was smoother and more balanced between the vehicle axles, resulting in a more consistent and stable vehicle response.

These findings indicate that state-feedback control methods such as LQR provide substantial advantages in roll suppression and vehicle stability, despite higher peak actuator demands. Future studies will incorporate actuator dynamics, evaluate different loading conditions, and validate the controllers through hardware-in-the-loop (HIL) testing for enhanced applicability in real-world automotive applications.

Acknowledgements. The authors gratefully acknowledge financial support provided by Fundação de Apoio à Pesquisa do Distrito Federal (FAPDF).

Authorship statement. The authors hereby confirm that they are the sole liable persons responsible for the authorship of this work, and that all material that has been herein included as part of the present paper is either the

property (and authorship) of the authors, or has the permission of the owners to be included here.

References

[1] P. Gaspar, Z. Szabó, and J. Bokor. Model-based analysis of vehicle dynamics and roll stability. In *Proceedings of the 6th Mini Conference on Vehicle System Dynamics, Identification and Anomalies*, pp. 85–92, Budapest, 2004.
[U.S. Department of Transportation] U.S. Department of Transportation. Federal motor vehicle safety standard no. 216: Roof crush resistance. Technical report, U.S. Department of Transportation / Euro NCAP. Includes Euro NCAP Test Protocols, —.
[3] M. J. L. Boada, B. L. Boada, and V. G. Díaz. Improved vehicle stability and comfort by active suspension system based on fuzzy logic. *Mechatronics*, vol. 19, n. 3, pp. 513–523, 2009.
[4] T. Vu, M. Tanelli, and S. Savaresi. Active roll control for heavy–duty vehicles: Performance evaluation and controller comparison. In *IFAC Symposium on System Structure and Control (SSSC)*, pp. 255–260, 2016.
[5] T. Vu, M. Tanelli, and S. Savaresi. Rollover prediction and prevention for heavy trucks via active suspensions. *Vehicle System Dynamics*, vol. 55, n. 7, pp. 1039–1060, 2017.
[6] K. Ogata. *Modern Control Engineering*. Prentice Hall, Upper Saddle River, NJ, 5 edition, 2010.
[7] B. D. O. Anderson and J. B. Moore. *Optimal Control: Linear Quadratic Methods*. Prentice Hall, Englewood Cliffs, NJ, 1990.
[8] Q. D. Nguyen, J. Kim, and H. S. Kim. Design of active stabiliser bar control using linear parameter varying approach for vehicle rollover prevention. *IEEE Transactions on Vehicular Technology*, vol. 71, n. 1, pp. 123–135, 2022.
[9] VI-grade GmbH. *VI-CarRealTime 2025.1 Documentation*. VI-grade GmbH, Darmstadt, Germany. Available from VI-grade official software documentation, 2025.