# Robust Roll Stability Control of Narrow Tilting Vehicle Based on Disturbance Observer

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**ABSTRACT**: The automotive industry faces challenges like emission limits, traffic congestion, and limited parking demand for compact vehicles. However, the narrow design of these vehicles increases the risk of rollovers. This paper addresses rollover safety in Narrow Tilting Vehicles (NTVs), using the stability criterion of the Lateral Load Transfer Ratio (LTR). Detailed roll dynamics of an NTV with MacPherson strut suspension are presented. A cascade control structure with a robust position controller is proposed to enhance stability against disturbances. To validate the proposed tilting control strategy, a Hardware-in-the-loop (HIL) simulation was conducted, integrating a real-time vehicle simulator with CarSim to replicate roll dynamics under various driving conditions.

**KEY WORDS**: narrow tilting vehicle(NTV), lateral load transfer ratio(LTR), robust controller, cascade control, disturbance observer(DOB), Hardware-in-the-loop simulation(HILS)

## 1. INTRODUCTION

As global populations grow and economies rapidly develop, the demand for automobiles is rising, leading to transportation issues like parking shortages and air pollution. Compact vehicles have gained attention as a solution to these problems,[(1,2)] but their narrow design increases the risk of rollovers compared to regular cars. Various studies have explored methods to improve the stability of these vehicles, with many focusing on adding tilting mechanisms to enhance safety.[(3-5)] Such vehicles, known as Narrow Tilting Vehicles (NTVs), tilt to improve cornering stability, similar to motorcycles. However, tilt alone is insufficient for determining stability, requiring numerical models. The Lateral Load Transfer Ratio (LTR) is a key measure for assessing roll stability.[(6-8)]

To achieve effective roll control, it is crucial to analyze the vehicle's dynamic characteristics. Lateral and roll dynamics are derived from basic vehicle equations,[(9)] incorporating the effects of tilting actuators. Previous studies have explored NTVs with actuators that apply torque directly to the vehicle body.[(3,10)] This paper focuses on a tilting system integrated with a MacPherson strut suspension.

Understanding roll dynamics is essential for developing control strategies. While some studies focus solely on actuator torque,[(11-14)] others propose combined control of steering and tilting mechanisms.[(15-16)] This paper introduces a stability-enhancing method using a robust internal controller to address disturbances like crosswinds.

To verify the proposed control strategy and its real-world applicability, a Hardware-in-the-loop (HIL) simulation environment was implemented. HIL testing allows real-time interaction between a physical hardware system and a virtual driving environment. The HIL system consists of a simulator with an integrated tilting mechanism, actuators replicating roll moments, and a Veristand-based real-time processing unit linked to CarSim. This setup enables realistic replication of dynamic vehicle behavior while facilitating precise control analysis under varying driving conditions.

## 2. SYSTEM MODELING

### 2.1. Lateral dynamics for tilting vehicle

In conventional vehicles, the roll dynamics is basically based on the difference in load between the left and right suspension due to lateral force in cornering situations. In contrast, tilting vehicles can actively make the roll angle through actuators. Based on the lateral dynamics of a general vehicle and adding the effect of the tilting structure, the dynamics equation for the tilting vehicle can be derived.

Lateral dynamics of tilting vehicle, as depicted in Fig 1, can be expressed in terms of lateral forces acting on the ground, lateral acceleration, and the vehicle's roll angle. Equation of the dynamics for the lateral force $R_y$ and vertical force $R_z$ acting on the tire from the ground is derived as Equation (1) and (2).

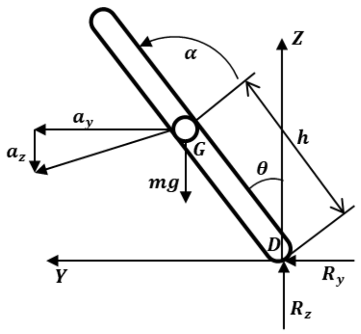


Fig. 1 Simplified vehicle roll dynamics model

$$R_y = m\ddot{y} + m\dot{\varphi}V + mh\ddot{\theta}cos\theta - mh\dot{\theta}^2 sin\theta \tag{1}$$

$$R_z = mg - mh\ddot{\theta}sin\theta - mh\dot{\theta}^2 cos\theta \tag{2}$$

When considering the rotational moment generated by the actuator at the center of gravity $G$ as $M_{tilt}$, the equation of the roll dynamics is derived as follows:

$$\begin{aligned} I_x\ddot{\theta} &= R_z hsin\theta - R_y hcos\theta + M_{tilt} \\ &= mghsin\theta - mh^2\ddot{\theta}sin^2\theta - mh^2\dot{\theta}^2 cos\theta sin\theta \\ &\quad - ma_y hcos\theta + M_{tilt} \end{aligned} \tag{3}$$

**2.2. Tilting mechanism dynamics**

The tilting mechanism proposed in this paper is mechanically connected to a MacPherson strut suspension via a rotating plate. Unlike traditional systems where the motor generates torque directly on the vehicle body, this design uses a more complex 2DOF structure, involving both the suspension and body components. In this system, torque is distributed to both the rotor and stator sides rather than being fixed to a single point. The motor's stator housing is attached to the vehicle's body, representing the sprung mass, while the rotor and shaft are connected to a plate mounted on the suspension's upper part. The layout of the tilting mechanism is illustrated in Fig 2.

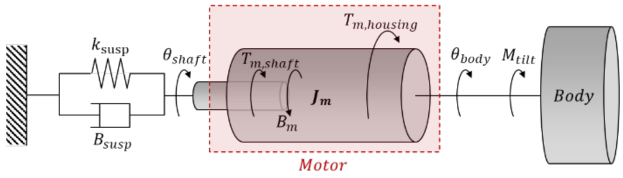


Fig. 2 Simplified dynamics model of tilting structure

Tilting structure can be defined by the equation of the angle of the motor shaft($\theta_{shaft}$) and the roll angle of the body($\theta_{body}$), where the difference between the two angles is $\theta_m$, defined as $\theta_m = \theta_{shaft} - \theta_{body}$. Based on this relationship, the dynamics of the tilting actuator can be derived as Equation (4).

$$\begin{aligned} J_m\ddot{\theta}_{shaft} &= -B_m\dot{\theta}_m - B_{susp}\dot{\theta}_{shaft} \\ &\quad -k_{susp}\theta_{shaft} + T_m + M_{tilt} \end{aligned} \tag{4}$$

**3. Controller Design**

The control block diagram is structured in a cascade structure for roll stability and can be represented as Figure 3. The inner loop controller is designed as a roll angle control loop to track the roll angle, while the outer loop is composed of a control loop generating the target roll angle based on the vehicle's state.

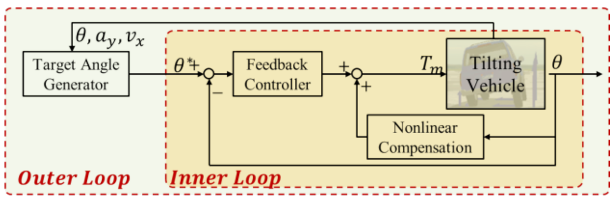


Fig. 3 Block diagram of the vehicle controller

**3.1. Roll angle controller**

The vehicle's roll angle controller is designed based on the dynamic model that links the roll angle to motor torque. The previously defined 2DOF relationship between the motor angle and roll angle is simplified into a 1DOF rotational model for easier control. The nominal model used in the control logic is kept as simple as possible, with an additional robust controller applied for stability. In this model, $J_n$ represents the nominal roll inertia of the entire vehicle, while $B_n$ accounts for nominal damping, such as friction from bushings and the reduction gear. The dynamic equation of the system is expressed as follows:

$$\frac{\theta(s)}{T(s)} = \frac{1}{J_n s^2 + B_n s} \tag{5}$$

The vehicle's roll angle controller is composed of three key components: Feedback, Feedforward, and a Disturbance Observer (DOB), all based on the nominal model. The complete block diagram of the controller is shown in Figure 4.

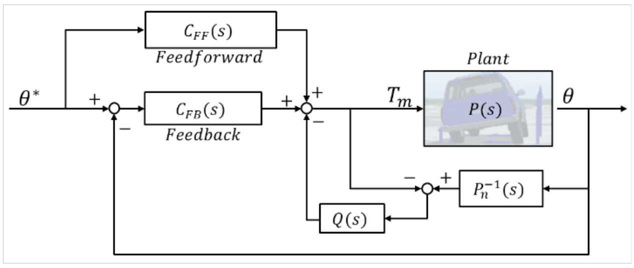


Fig. 4 Block diagram of the roll angle controller

The feedback controller is designed using pole-zero cancellation based on the nominal model. To enhance the response time of the position control, the feedforward controller is designed by inverting the nominal model and applying a second-order low-pass filter for stability. To account for model uncertainties and disturbances, a Disturbance Observer (DOB) is implemented. The DOB predicts the input to the nominal model and compares it with the actual input, identifying any differences as disturbances. These disturbances are passed through a second-order low-pass filter, represented as $Q(s)$, which allows the observation of disturbances within a specific frequency range while filtering out high frequency signals.

### 3.2. Roll stability controller

The target roll angle is calculated to converge lateral acceleration to zero based on this equation. For simplicity, the lateral acceleration caused by side slip $\ddot{y}$ and the angular acceleration $\ddot{\theta}$ are assumed to be zero in the equation. The vehicle's yaw rate $\gamma$ is calculated as $\gamma = v_x/R$, and the cornering radius $R$ can be expressed in relation to the steering angle $\delta_f$ and the wheelbase $L$. The formula for calculating the target roll angle is represented as a function of the vehicle's longitudinal velocity and steering angle, as shown in equation (6).

$$\bar{\theta}_d \approx \tan^{-1}\left(\frac{a_y}{g}\right) = \tan^{-1}\left(\frac{v_x^2}{gR}\right) \approx \frac{v_x^2 \delta_f}{gL} \tag{6}$$

Directly using the values generated by vehicle speed and steering can result in too rapid response in comparison to the steering angle, potentially causing discomfort for the driver. To control the response speed of the target roll angle, a steering response coefficient τ has been added to the controller. Additionally, to prevent the excessive roll angle, k has been included as a tuning parameter for the target roll angle. The final formula for generating the target roll angle including tuning factor is represented as shown in Equation (7).

$$\theta_d = \bar{\theta}_d H(s) = \frac{v_x^2 \delta_f}{gL}\left(\frac{k}{\tau s + 1}\right) \tag{7}$$

## 4. HARDWARE-IN-THE-LOOP SIMULATION

To validate the proposed tilting mechanism and control logic, Hardware-in-the-loop (HIL) simulation was implemented by the simulator that including the tilting mechanism. In order to make the roll moment acting on the vehicle while driving, an external actuator was integrated into the system. The HIL environment was developed using a Veristand-based platform, which was linked with CarSim to enable real-time interaction between the hardware and the simulation software.

Fig. 5 Hardware-in-the-loop environment

The fundamental vehicle inputs, such as speed and steering angle, were derived from the CarSim simulation, while the roll moment generated during driving was also obtained from CarSim and applied to the load actuator within the HIL system. The roll angle observed in the hardware was then transmitted back to the CarSim model to ensure that the simulated vehicle maintained the same roll motion as the hardware.

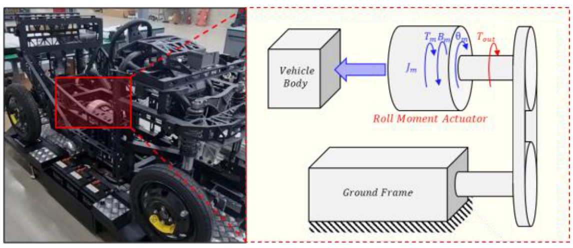


Fig. 6 Load actuator structure

To examine both control performance and driver experience, the HIL simulation was tested under two different modes: an autonomous driving mode, where the vehicle followed predefined motion and speed profiles to facilitate an objective performance assessment, and a manual driving mode, in which a human driver directly operated the simulator to evaluate the real-time response of the tilting control system. Through these two testing approaches, comprehensive validation was achieved by incorporating both quantitative performance metrics and qualitative driver feedback.

## 5. SIMULATION RESULT

The performance of the proposed tilting control was assessed through simulations conducted under two different test scenarios: a double lane change maneuver, which evaluates lateral stability during rapid lane switching, and a constant radius cornering test, which examines roll control performance in steady state turning conditions. Additionally, manual driving experiments were carried out on curved roads to assess real-time driver interaction with the tilting mechanism.

For a comparative evaluation, three different control strategies were implemented: a baseline case where the motor shaft angle was fixed at zero degrees, making the vehicle behave like a conventional non-tilting system; a second case in which the tilting control was activated without the disturbance observer (DOB); and a third case where the same tilting control was employed with the addition of a DOB to enhance robustness. To quantify the roll stability, the lateral acceleration and the lateral load transfer ratio (LTR) were analyzed, with the latter serving as an indicator of weight difference between the left and right wheels, where values closer to zero signify improved roll stability.

The results indicate that, in the absence of tilting control, both lateral acceleration and LTR increased, demonstrating lower stability compared to cases where the tilting mechanism was actively controlled. The application of the tilting control significantly reduced both values, confirming its effectiveness in enhancing roll stability. While the presence of the DOB did not result in noticeable improvements in stability, it effectively minimized control errors, thereby improving the accuracy of the tilting response. Given that the experiments were conducted under controlled simulation conditions without external disturbances, it is expected that the impact of the DOB will become more pronounced in real-world driving scenarios where external factors such as crosswinds or road irregularities introduce additional uncertainties.

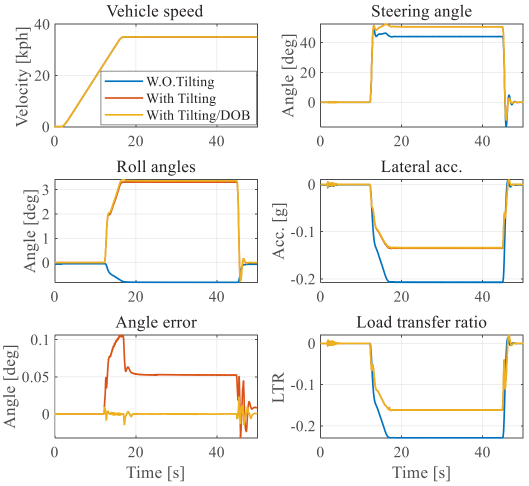


Fig. 7 The Simulation results of constant radius cornering

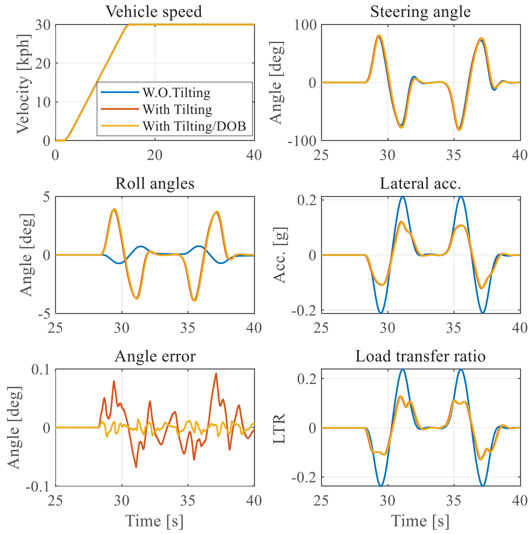


Fig. 8 The Simulation results of double lane change

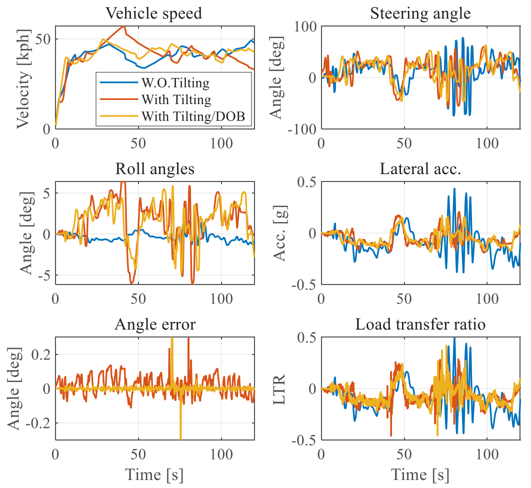


Fig. 9 The Simulation results of free driving condition

## 6. CONCLUSIONS

This paper presents a roll stability control strategy for a Narrow Tilting Vehicle (NTV) with a novel tilting structure, developed using a real vehicle model. The control system uses a cascade structure, with the inner loop ensuring robust angle control and the outer loop focusing on roll stability. By simplifying the roll angle controller, the model reduces instability and improves real-time performance. A Disturbance Observer (DOB), which compensates for external disturbances and model uncertainties, ensures stable control in dynamic driving conditions. Simulation results show that this approach effectively enhances vehicle stability and reduces rollover risk, offering a reliable solution for improving the safety of NTVs.

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