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Research Article

Smooth Sliding Mode Control for Vehicle Rollover Prevention Using Active Antiroll Suspension

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The rollover accidents induced by severe maneuvers are very dangerous and mostly happen to vehicles with elevated center of gravity, such as heavy-duty trucks and pickup trucks. Unfortunately, it is hard for drivers of those vehicles to predict and prevent the trend of the maneuver-induced (untripped) rollover ahead of time. In this study, a lateral load transfer ratio which reflects the load distribution of left and right tires is used to indicate the rollover criticality. An antiroll controller is designed with smooth sliding mode control technique for vehicles, in which an active antiroll suspension is installed. A simplified second order roll dynamic model with additive sector bounded uncertainties is used for control design, followed by robust stability analysis. Combined with the vehicle dynamics simulation package TruckSim, MATLAB/Simulink is used for simulating experiment. The results show that the applied controller can improve the roll stability under some typical steering maneuvers, such as Fishhook and J-turn. This direct antiroll control method could be more effective for untripped rollover prevention when driver deceleration or steering is too late. It could also be extended to handle tripped rollovers.

1. Introduction

Vehicle rollover accidents typically contribute to fatal damage even though they account for a small percentage among all accidents. According to the statistics of U.S. National Highway Traffic Safety Administration (NHTSA), more than 280,000 rollover accidents and 10,000 injuries or deaths are reported annually. It has been marked as the second most dangerous accident following head-on collisions in the United States [1].

There are two fundamental problems in rollover prevention: to *detect/predict* rollover and to *control/prevent* it. Generally, most active control systems for vehicle roll stability in previous studies are based on the assumption of a good detection or prediction for vehicle rollovers. In other words, one has to accurately predict the impendence of rollover in time before sensible control actions. The Static Stability Factor (SSF) presented by the Society of Automotive Engineers (SAE) is a simple and practical method for vehicle roll stability analysis. This factor, which is the ratio between the half-track and height of the center of gravity (CG), indicates the maximum allowable lateral acceleration for vehicle cornering. It could be used to indicate the rollover propensity. However, this method has some drawbacks due to neglecting impacts of suspension and tire defections, as well as vehicle dynamic characteristics. Therefore, some dynamic rollover predicting methods are presented to remedy the limitation of the above static predicting method, such as Time-To-Rollover metric and Rollover Prevention Energy Reserve index.

In previous studies, some active antiroll control systems using braking or steering control are used to help in dealing with vehicle rollover threat and improving the vehicle stability and safety. Admittedly, those approaches are easy to implement since steering and brake are the most common

control actuators installed on vehicles. However, they do not control vehicle roll dynamics directly. Instead, the roll motion is controlled through coupling with lateral and longitudinal motions. Therefore, control bandwidth and performance of those approaches are limited. It may be effective for rollover control of passenger cars, but it is not effective for roll stabilization of high CG vehicles such as trucks fully loaded with straw/foam/clothing bales or other low density products.

Therefore, the main contributions of this study include the following: (1) an effective rollover index which reflects the load distribution of left and right tires is used to indicate the rollover criticality; (2) a simpler model-based control approach is used for stabilization of roll motion; (3) instead of using steering (lateral) and braking (longitudinal), direct antiroll control for high CG vehicles can be taken to improve control bandwidth and to reduce time delay; (4) the direct roll control method using the smooth sliding mode control technique with robust stability analysis is applied to a simplified roll dynamics model with uncertainties.

This paper is structured as follows: Section 2 discusses the critical threshold indicating the vehicle rollover impendence; in Section 3, a smooth sliding mode control design based on a roll dynamics model is presented; the performance of this control system (controlled vehicle) is analyzed in comparison with simulated experiments of the uncontrolled system (conventional vehicle) in Section 4; and this paper is concluded in Section 5.

2. Literature Review

This part presents a brief literature review on two core issues of rollover problems for vehicles with elevated CG, that is, rollover prediction and antiroll control.

Aiming at the problem of vehicle rollover prediction, Chen and Peng [2, 3] propose a Time-To-Rollover (TTR) metric. Antiroll control is activated to avoid the rollover if the TTR is less than a preset value. The TTR metric which helps to indicate the level of rollover threat will automatically "count down" toward rollover according to vehicle speed and steering patterns.

An algorithm that detects impending rollover evaluated the index including lateral acceleration, roll angle, and roll rate information which can be implemented to detect both maneuver-induced and road-induced rollovers [4]. One energy method using Rollover Prevention Energy Reserve (RPER) index is developed for vehicle rollover prevention [5]. The RPER index is the difference between the energy consumption of vehicle rollover and the rotational kinetic energy. Another energy method for vehicle rollover prevention is presented. The rollover index is defined from the vehicle lateral kinetic energy and a new concept of virtual gravity.

Trent and Greene [6] present a method for predicting rollover propensity by calculating tire deflections based on genetic algorithm. Tire deflections calculated through tire model are compared with their limited deflections estimated through genetic algorithm. If tire deflections exceed the threshold, rollover is assumed likely to happen.

Presently, most methods for the rollover prediction are based on the above static or dynamic roll stability analysis. Some of them are practically implemented on several types of chassis control systems for rollover prevention. At present, the practical vehicle control systems for preventing rollover accident are using active steering control and/or differential braking control. Odenthal et al. [7] implement steering and braking control to avoid vehicle rollover. Three feedback control loops including continuous operation steering control, emergency steering control, and emergency braking control are used. Moreover, a load transfer ratio (rollover coefficient) indicating rollover impendence is defined, which basically depends on the lateral acceleration and roll angle at CG. Schofield [1] presents a control strategy based on limitation of the roll angle following a yaw rate reference. A new computationally efficient control allocation strategy based on convex optimization is used to map the controller commands to individual braking forces with actuator constraints taken into account.

These active control systems using active steering control or differential braking control can help in dealing with vehicle rollover threat and improving the stability and safety of the vehicle. However, these systems do not control vehicle roll dynamics directly. In other words, they do not aim at vehicle roll stability primarily. Hence, the vehicle handling performance and riding quality brought about by active steering, braking, and suspension have to be compromised to some extent due to the system extension for roll stability. Therefore, few studies have considered the direct active roll control for improving vehicle roll stability and reducing rollover hazards through active roll or suspension systems.

Darling et al. [8] present a low cost active antiroll system with an "idealized" roll control mechanism aiming at controlling roll dynamics during several steering maneuvers and thus it improves vehicle ride comfort and safety. Shuttlewood et al. [9] evaluate the performance of a slow-active hydropneumatic suspension which is incorporated into a vehicle handling model for roll control during a cornering maneuver. It uses feedback measurements of the body motions for the roll control without compromising the ride performance. Lu and Hedrick [10] present a novel design of active suspension system using hydraulic actuator (a cylinder and a piston) for each suspension with a centralized control activated by a pump and servo valves. However, the dynamical behavior of the closed loop system is not yet analyzed in detail.

Yu et al. [11] developed an active suspension system based on the linear quadratic regulator (LQR) method. Yim et al. [12] designed an antirollover controller with both active suspension and ESP, and they also used linear quadratic method for the control design. Yang and Liu [13] presented an active suspension control system with the control technique of simple feedback control.

In this paper, a practical active suspension actuator (i.e., *active antiroll suspension*) is used in order to tackle the problems related to rollover prediction and antiroll control of vehicles with elevated CG. It can provide an emerging effective solution for reducing rollover accidents and improving vehicle handling performance. Specifically, this paper presents a rollover index and a sliding mode controller regulated this kind of suspension for rollover prevention.

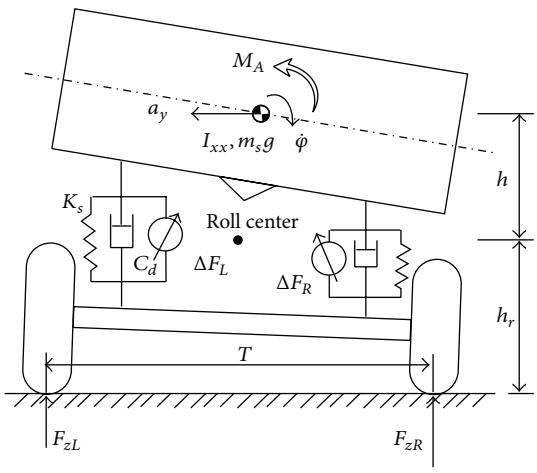


Figure 1: The induced roll dynamic model.

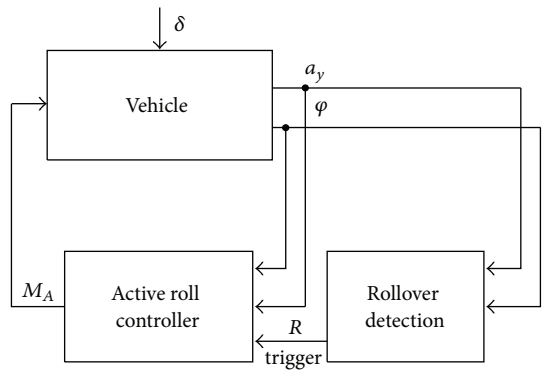


Figure 2: Sliding mode controller design for rollover prevention.

3. Methodology

For rollover prevention, a rollover index based on lateral load transfer ratio to monitor the rollover critical condition online is presented. When the rollover index exceeds a preset threshold value, the sliding mode controller will be triggered and generate appropriate antiroll moment which acted on the center of gravity. Therefore, this generated moment can drag the unstable vehicle to the equilibrium state.

3.1. Impending Rollover Prediction. The method using a load transfer ratio with measured vehicular lateral acceleration and roll rate can identify the situation of wheel contraction reduction/loss. This situation is often referred to as a "critical situation" which should be avoided for rollover prevention [7]. Suppose the vertical motion has been neglected; the sum of vertical forces is just the static vehicle weight. Hence, the lateral load transfer ratio (the so-called rollover index) can be calculated as (Figure 1)

$$R = \frac{F_{zR} - F_{zL}}{F_{zR} + F_{zL}}, \tag{1}$$

where the subscripts "L" and "R" represent the left and right tires. Moreover, tire vertical forces are denoted by F_{zL} and F_{zR} which can be obtained using Newton's Law in the vertical direction.

In (1), tire vertical forces can be calculated as follows:

$$\begin{aligned} F_{zR} + F_{zL} &= mg, \\ F_{zR} - F_{zL} &= \left(m_s gh \sin\varphi + m_s a_y \left(h\cos\varphi + h_r\right)\right) \cdot \frac{2}{T}. \end{aligned} \tag{2}$$

Combining (1) and (2), as well as assuming a small roll angle, the rollover index is approximated by the following equation (4):

$$R = \frac{2m_s}{mT}\left(\left(h_r + h\right)\frac{a_y}{g} + h\varphi\right). \tag{3}$$

During the straight driving, the lateral acceleration and roll angle are both zero, which leads to zero rollover index. In the case of a severe steering maneuver, the lateral acceleration and roll angle become large which may cause the absolute value of R to be saturated to one; that is, $|R| \leq 1$.

In the real world, the lateral acceleration (unit: 'g) is often much greater than the roll angle (unit: rad); that is, $a_y/g \gg \varphi$. Moreover, the sprung mass is often much greater than the unsprung mass for heavy-duty vehicles. That is to say, the sprung mass occupies the absolute proportion of the total mass; that is, $m_s \approx m$. So the calculation of the above rollover index can be transformed as follows:

$$R = \frac{2\left(h_r + h\right)a_y}{Tg}. \tag{4}$$

3.2. Control Design for Rollover Prevention. The objective of the roll control system is to generate a roll moment from active antiroll suspension for maximizing the roll stability of the vehicle. Consequently, the following vehicle roll dynamic model is obtained for control design:

$$\ddot{\varphi} = \frac{1}{I_{xx}}\left[-C_d\dot{\varphi} - \left(K_s - mgh\right)\varphi + ma_y h + M_A\right] + \Delta, \tag{5}$$

where M_A represents the control input and Δ represents the allowable uncertainties (such as road disturbances, nonlinearity, and unmodeled dynamics), which is sector bounded as follows:

$$\Delta = \Delta\left(\varphi, \dot{\varphi}\right) \leq \sigma_1 \left|\varphi\right| + \sigma_2 \left|\dot{\varphi}\right|, \tag{6}$$

where $\sigma_1 > 0$ and $\sigma_2 > 0$ are known positive real numbers.

Control design with the vehicle dynamics model will be used to control a loaded two-axle vehicle model in TruckSim for simulation. According to (5), the system input is roll moment from controlled suspension M_A and the control output is the roll angle. Thus, this active roll control system can be demonstrated in Figure 2.

The system input and output can be expressed, respectively, as follows:

$$\begin{aligned} u &= M_A, \\ y &= \varphi. \end{aligned} \tag{7}$$

The uncertainty in inequality (6) can be represented as

$$\Delta(y,\dot{y}) \le \sigma_1 |y| + \sigma_2 |\dot{y}|. \tag{8}$$

The main results regarding the stability are stated in the following.

Theorem 1. *For the vehicle dynamics in (5) with sector bounded uncertainty in inequality (6), if the controller and the corresponding gains are selected as*

$$\begin{aligned} u = &-((\lambda+\eta) I_{xx} - C_d)\dot{\varphi} - (\lambda\eta I_{xx} - (K_s - mgh))\varphi \\ &- ma_y h - I_{xx}(\varepsilon \cdot \operatorname{sgn}(\dot{\varphi}+\lambda\varphi) + \Delta) \\ &\eta > \sigma_2 \quad \varepsilon > (1+\sigma_1+\sigma_2\lambda) y_{\max}, \end{aligned} \tag{9}$$

the overall system is semiglobally asymptotically (exponentially) stable.

Proof. Define the sliding surface as

$$s = \dot{y} + \lambda y = \dot{\varphi} + \lambda\varphi, \tag{10}$$

where λ is a positive constant indicating the performance of system converging to the equilibrium point.

The derivative of sliding surface along the trajectory of the model in (5) is

$$\begin{aligned} \dot{s} &= \ddot{\varphi} + \lambda\dot{\varphi} \\ &= \frac{(ma_y h - ((K_s - mgh)\varphi + C_d\dot{\varphi}))}{I_{xx}} + \frac{u}{I_{xx}} + \lambda\dot{\varphi} \\ &\quad + \Delta. \end{aligned} \tag{11}$$

Choose the reaching condition which guarantees the asymptotic stability as follows:

$$\dot{s} = -\eta \cdot s - \varepsilon \cdot \operatorname{sgn}(s), \tag{12}$$

where the positive constants η and ε are the control gains to be determined, which represent the speed reaching the sliding surface, and sgn(·) is a sign function. Now the control input can be derived as

$$\begin{aligned} &\frac{(ma_y h - ((K_s - mgh)\varphi + C_d\dot{\varphi}))}{I_{xx}} + \frac{u}{I_{xx}} + \lambda\dot{\varphi} + \Delta \\ &\quad = -\eta(\dot{\varphi}+\lambda\varphi) - \varepsilon \cdot \operatorname{sgn}(\dot{\varphi}+\lambda\varphi), \end{aligned} \tag{13}$$

that is,

$$\begin{aligned} u = &-((\lambda+\eta) I_{xx} - C_d)\dot{\varphi} - (\lambda\eta I_{xx} - (K_s - mgh))\varphi \\ &- ma_y h - I_{xx}(\varepsilon \cdot \operatorname{sgn}(\dot{\varphi}+\lambda\varphi) + \Delta). \end{aligned} \tag{14}$$

With this control law, the original dynamical system, that is, (5) and (10), is equivalent to the following system in stability:

$$\begin{aligned} \dot{y} &= -\lambda y + s, \\ \dot{s} &= -\eta s - \varepsilon \cdot \operatorname{sgn}(s) + \Delta. \end{aligned} \tag{15}$$

Without loss of generality, it is assumed that the initial condition for $y(0)$ belongs to a compact domain: $y(0) \in Y \subset \Re$ (the real set). Then, there exists $y_{\max} > 0$ such that $y(0) \le y_{\max}$, for all $y \in Y$.

The following Lyapunov candidate function is selected:

$$V = \frac{1}{2}(y^2 + s^2). \tag{16}$$

The time derivative of this Lyapunov function along the state trajectory of (15) is as follows:

$$\begin{aligned} \dot{V} &= y \cdot \dot{y} + s \cdot \dot{s} \\ &= y \cdot (-\lambda y + s) + s \cdot (-\eta \cdot s - \varepsilon \cdot \operatorname{sgn}(s) + \Delta) \\ &= -\lambda \cdot y^2 + y \cdot s - \eta \cdot s^2 - \varepsilon \cdot |s| + \Delta \cdot s \\ &\le -\lambda \cdot y^2 + y \cdot s - \eta \cdot s^2 - \varepsilon \cdot |s| + s(\sigma_1 |y| + \sigma_2 |\dot{y}|) \\ &\le -\lambda \cdot y^2 + |y| \cdot |s| - \eta \cdot s^2 - \varepsilon \cdot |s| + \sigma_1 |y| |s| \\ &\quad + \sigma_2 (\lambda |y| + |s|) |s| \\ &\le -\lambda \cdot y^2 - (\eta - \sigma_2) \cdot s^2 \\ &\quad - (\varepsilon - (1+\sigma_1+\sigma_2\lambda) y_{\max}) |s|, \end{aligned} \tag{17}$$

which is strictly negative if

$$\begin{aligned} &\eta > \sigma_2, \\ &\varepsilon > (1+\sigma_1+\sigma_2\lambda) y_{\max}. \end{aligned} \tag{18}$$

We therefore have proved the above theorem regarding the robustness stability. □

From the sliding mode control theory, the sliding surface s defined by inequality (6) is attractive, leading s to converge towards 0 exponentially.

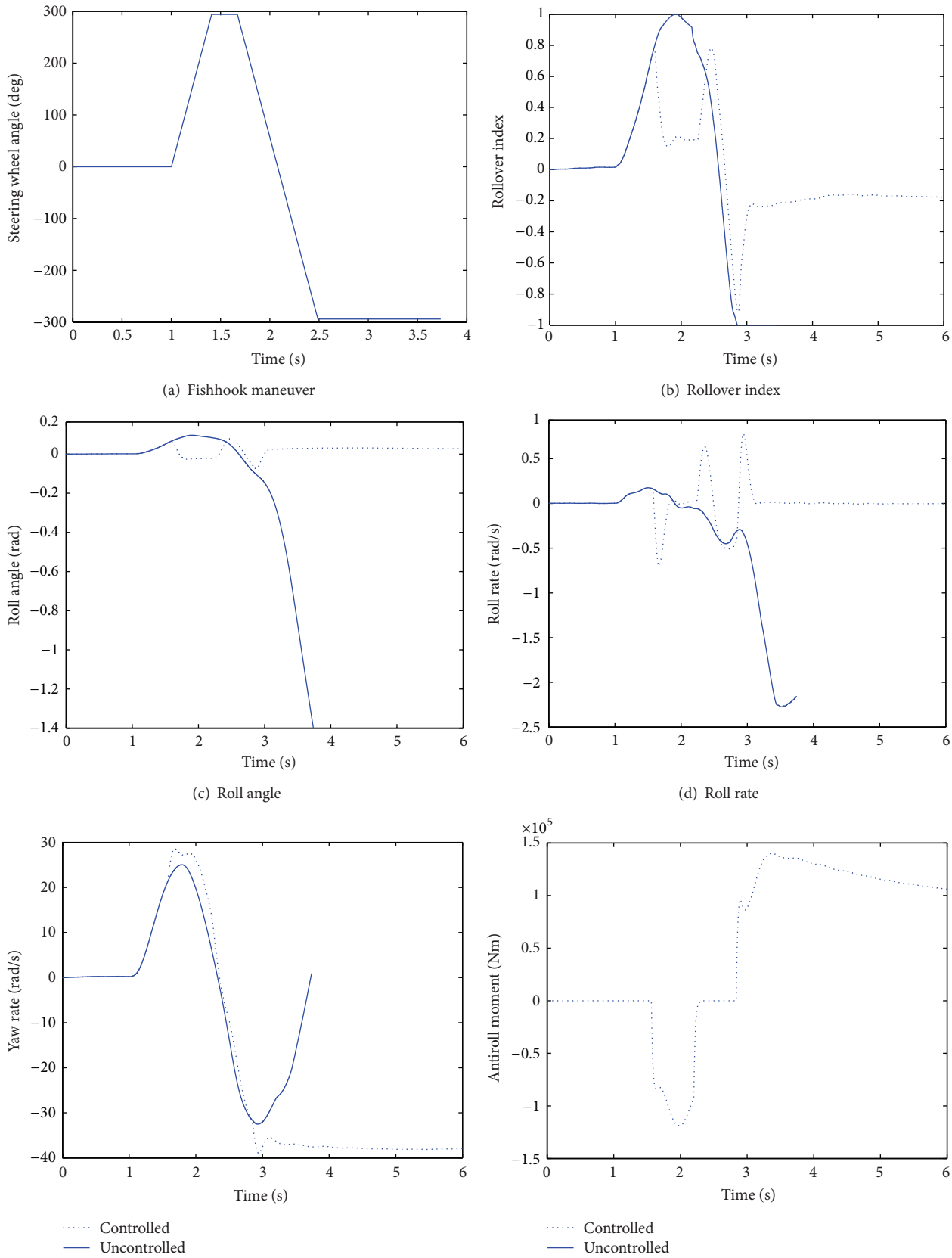
A saturation function is used to replace the sign function in (15). Consider

$$\operatorname{sat}\left(\frac{s}{b}\right) = \begin{cases} \dfrac{s}{b} & |s| \le b \\ \operatorname{sgn}\left(\dfrac{s}{b}\right) & |s| > b. \end{cases} \tag{19}$$

The verification of this control strategy will be presented in the following simulating experiment and analysis.

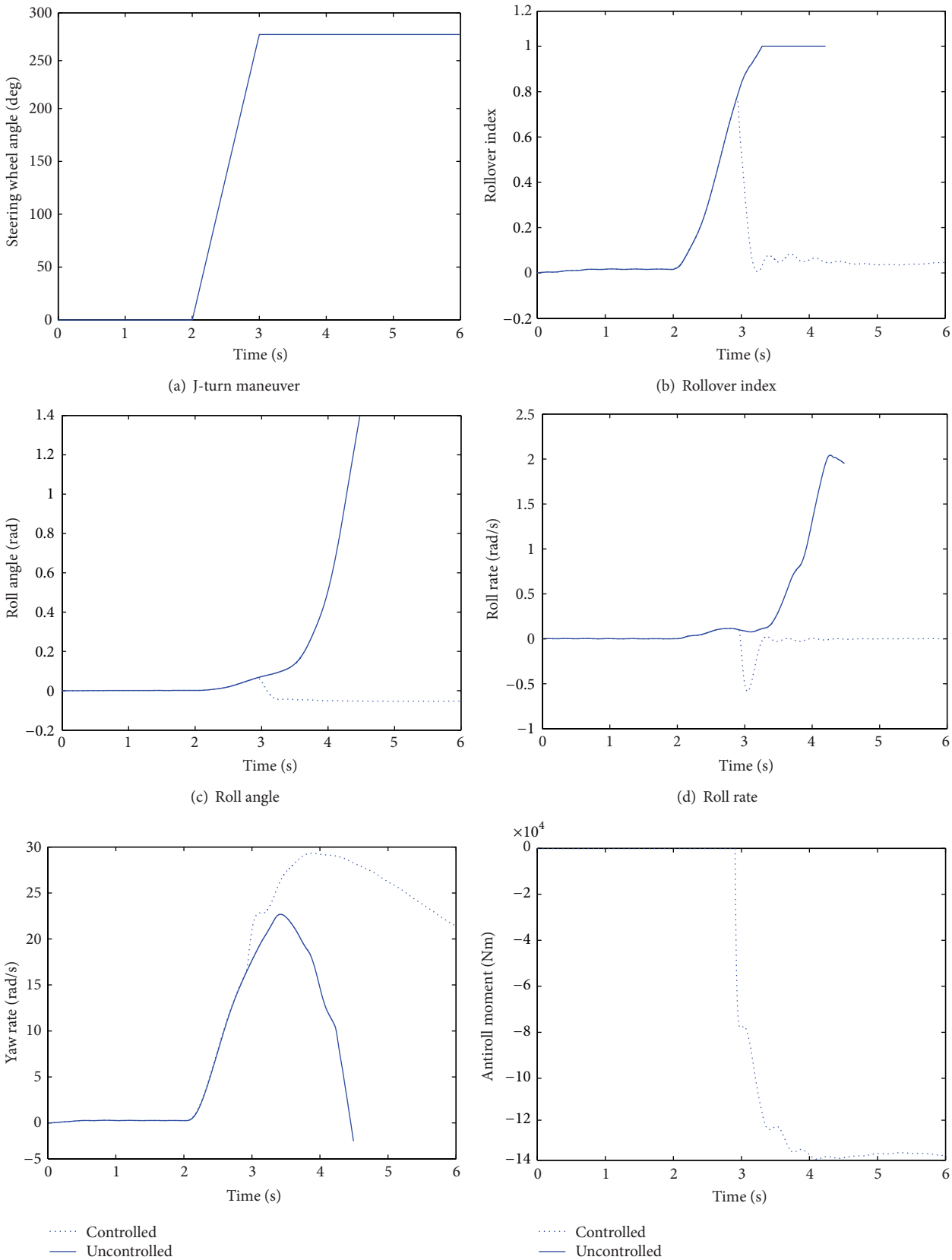
4. Experiment Results and Analysis

Several steering maneuvers have been developed by NHTSA to evaluate the roll stability of vehicles. Two typical maneuvers, that is, Fishhook in Figure 3(a) and J-turn in Figure 4(a), are used for verification of the sliding mode controller designed in this paper. It has been applied to the maneuvers to test the dynamic responses of uncontrolled and controlled systems at the speed of 80 km/h. The dynamic responses of the uncontrolled vehicle (*solid lines*) and the controlled vehicle (*dotted lines*) are shown in Figures 3(b)–3(f) and 4(b)–4(f) respectively. Specifically, Figures 3(b) and 4(b) show



(a) Fishhook maneuver (b) Rollover index (c) Roll angle (d) Roll rate (e) Yaw rate (f) Antiroll moment

FIGURE 3: Comparisons of the uncontrolled and the controlled vehicle dynamics responding to Fishhook experiment with sliding mode control parameters $\lambda = 16$, $\eta = 1.5$, $\varepsilon = 1.2$, $\sigma_1 = 0.1$, $\sigma_2 = 0.1$, and $y_{\max} = 0.2$ and sliding surface bound $b = 0.01$.



(a) J-turn maneuver (b) Rollover index (c) Roll angle (d) Roll rate (e) Yaw rate (f) Antiroll moment

FIGURE 4: Comparisons of the uncontrolled and the controlled vehicle dynamics responding to J-turn experiment with sliding mode control parameters $\lambda = 16$, $\eta = 1.5$, $\varepsilon = 1.2$, $\sigma_1 = 0.1$, $\sigma_2 = 0.1$, and $y_{\max} = 0.2$ and sliding surface bound $b = 0.01$.

the dynamic change of rollover index under steering maneuvers; Figures 3(c)-3(d) and 4(c)-4(d) show the dynamic responses of vehicle roll state; Figures 3(e) and 4(e) show the dynamic responses of vehicle yaw state; and Figures 3(f) and 4(f) show control outputs of the designed sliding mode controller. Moreover, in the simulating experiment, the threshold of the rollover index (R) has been set to 0.8. If there is no control performed when R has exceeded the threshold, the vehicle will be about to roll over.

Moreover, $y_{\max}$ is set to be 0.2 rad which is the limit of roll angle of the test vehicle model. For the simplification of experiments, σ_1 is equal to σ_2, and they are both set to be 0.1 mathematically. Therefore, inequality (18) is derived as follows:

$$\begin{aligned} &\eta > 0.1, \\ &\varepsilon > 0.22 + 0.02 \times \lambda. \end{aligned} \tag{20}$$

During Fishhook and J-turn experiments, if the above equation is satisfied, the designed sliding mode controller for the antirollover prevention system will be stable according to Lyapunov stability theory. However, according to the sliding mode control technique, there exists the chattering problem which is caused by frequently switching control along the sliding surface. The first method for smoothing out the chattering effect to some extent is using the saturation function shown in (19) instead of the sign function. This is the popularly used boundary layer sliding mode control, and the boundary parameter b in this case is set to 0.01 for narrowing the switching control boundary, which helps in reducing the chattering effect. The other one is tuning sliding mode parameters which are the positive constants λ, η, ε, and so forth.

Actually, these three constants determine the extent of chattering, and the appropriate combinations of these parameters are chosen by trial and error. Through repeated selection, it is found that parametric values around $\lambda = 16, \eta = 1.5$, and $\varepsilon = 1.2$ can reduce the chattering effect to an extent. When $\lambda < 8$ or $\lambda > 40$, the chattering will become severe. According to (18), $\eta > \sigma_1 = 0.1$, but an analogous chattering situation happens when $\eta > 5$. It also seems that ε does not exert too much influence on the chattering effect; but, according to (18), ε has to be larger than 0.54. Therefore, the appropriate chosen combinations of these parameters are shown in Figures 3 and 4.

For the Fishhook experiment shown in Figure 3, it can be observed that rollover index goes to −1 after 2.8 seconds, which indicates that tires on one side lift off the ground, and vehicle rollover happens in 3.6 seconds. Then, after a short time, the roll and yaw dynamics diverge rapidly and the program stops.

During the simulating experiment of the controlled vehicle, the rollover index has been dragged towards the set value. When the vehicle roll stability is about to be lost, the antiroll moment is applied. As shown in the dotted lines of Figures 3(c) and 3(e), the roll and yaw dynamics are maintained stable in equilibrium states.

Likewise, the result of J-turn experiment is shown in Figure 4. After 2 seconds when the severe steering maneuver is exerted, the magnitude of the rollover index goes up to 1 rapidly, which indicates that tires on one side lift off. Without the antiroll control, vehicle roll and yaw dynamic stabilities be lost within 4.4 seconds, and then the program stops. From the dotted lines of Figures 4(b)–4(e), antiroll moment in Figure 4(f) drags the rollover index to equilibrium state, and both roll and yaw dynamics are maintained stable as well.

5. Conclusions and Future Work

This paper presents a new method for rollover prevention, that is, the direct antiroll control. It is different from the methods with vehicle braking or steering which essentially controls the roll motion through coupling with lateral and longitudinal motions. Through direct roll motion control, it can potentially increase control bandwidth and reduce time delay so that it could handle untripped rollovers or even tripped rollovers for vehicles with elevated CG.

An antiroll sliding mode controller has been designed and verified through the joint simulation of TruckSim and MATLAB/Simulink. The underneath vehicle dynamics model in TruckSim is a full vehicle dynamics model. The simulation results are the control responses of the full truck model in TruckSim with respect to the feedback SMC derived from the simplified model. Vehicle robust roll stability is shown through simulations with Fishhook and J-turn steering maneuvers. For practical implementations, the lateral load transfer ratio which indicates the rollover impendence is used for control activation. The roll moment could be adjusted with the active antiroll suspension in real time. Compared with other indirect rollover prevention systems using braking or steering, there are some advantages for applying this direct roll control system:

(1) Control strategies become more and more complicated using differential braking or active steering to improve both the handling stability and the roll stability; besides, controlling roll motion through coupling with steering (lateral motion) and braking (longitudinal and later motions) is indirect, where control performance and bandwidth are limited; therefore, it is simpler and more effective to avoid the maneuver-induced rollover with direct roll control.

(2) Direct roll control strategy could potentially be used for tripped rollover prevention caused by road irregularities if tripped rollover could be accurately detected/predicted.

(3) Active antiroll suspension is a part of vehicle chassis system; therefore, the active roll control could also be extended to improve the riding comfort.

Practical implementations of this approach could improve both safety and riding comfort when control energy consumptions of the antiroll moment generation are limited within an allowable range, which will be the focus of the future work.

Nomenclature

h: Distance from CG to roll center, 1.26 m
h_r: Distance from roll center to the ground, 0.47 m
m: Vehicle total mass, 12562 kg
m_s: Vehicle sprung mass, 11257 kg
g: Acceleration due to gravity, 9.81 kg·m/s^2
I_{xx}: Moment of inertia about x-axis, 5114 kg·m^2
T: Track width, 2.03 m
C_d: Roll damping of passive suspension, 31000 N·m·s/rad
K_s: Roll stiffness of passive suspension, 347000 N·m/rad
R: Rollover index
a_y: Lateral acceleration
φ: Roll angle.

Conflict of Interests

The authors declare that there is no conflict of interests regarding the publication of this paper.

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