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ROLLOVER INDEX-BASED ROLLOVER MITIGATION CONTROL SYSTEM

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ABSTRACT−This paper describes a rollover index (RI)-based rollover mitigation control (RMC) system. A rollover index which indicates an impending rollover has been developed by a roll dynamics phase plane analysis. The rollover index is calculated using the roll angle, the roll rate, the lateral acceleration and time to wheel lift (TTWL). A differential braking control law based on a 2-D bicycle model has been designed using the direct yaw control (DYC) method. An RMC threshold has been determined from the rollover index. The performance of the RMC scheme and the effectiveness of the proposed rollover index are illustrated using a vehicle simulator.

KEY WORDS : Differential braking, Direct yaw moment control (DYC), Phase plane analysis, Rollover index (RI), Rollover mitigation control (RMC)

1. INTRODUCTION

Rollover accidents are significant safety problems for all classes of light vehicles such as light trucks, vans and SUVs. Even though rollovers constitute a small percentage of accidents, they have a disproportionately large contribution to severe and fatal injuries (Chen and Peng, 2001; Jang and Marimythu, 2006). For this reason, rollover mitigation control (RMC) systems that prevent rollover in vehicles have been studied by many researchers. Rollover prevention systems can be classified into two stages: detection of the possibility of rollover, and development of a mitigation control algorithm. In the early studies on detection of vehicle rollover, the concept of a static rollover threshold was used, but this is only useful at steady state. Chen and Peng proposed Time-To-Rollover (TTR) to estimate the time until rollover occurs and performed direct yaw moment control using differential braking (Chen and Peng, 1999;Ungoren and Peng, 2001; Chen and Peng, 2001). Hac and Martens described a rollover index using a model-based roll estimator (Hac *et al.*, 2004; Hac, 2002). Yang and Liu also presented a rollover index which is a combination of respective rollover indices from influential factors such as the position of vehicle's center of gravity (CG), the energy of rollover, and vertical tire forces (Yang and Liu, 2003).

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Kim and Oh (2006) proposed two main rollover criteria, Rotational Kinetic Energy (IKE), based on simple physical model.

This paper describes a rollover index (RI)-based vehicle rollover mitigation control scheme. The roll angle, roll rate and lateral acceleration are used to determine the wheel lift threshold and RI. That is, a RI which indicates rollover danger has been developed using roll angle and roll rate ($\phi-\dot{\phi}$) phase plane analysis and wheel lift threshold. The differential braking control law has been applied to the RMC and its threshold has been designed based on the RI. According to an increase in rollover danger, the RI increases to near unity and mitigation control input is applied to prevent rollover. A differential braking control law has been designed using a 2-D vehicle model. The brake control inputs have been directly derived from the sliding control law based on a 2-D vehicle model with differential braking. If a differential braking control input is applied to the vehicle, it generates the moment to mitigate the roll motion of the vehicle by the direct yaw control (DYC) method. The performance of the RMC and the effectiveness of the proposed rollover index are investigated through computer simulations.

2. ROLLOVER INDEX

In this study, we introduce the rollover index, a dimensionless number, to determine the danger of rollover. To

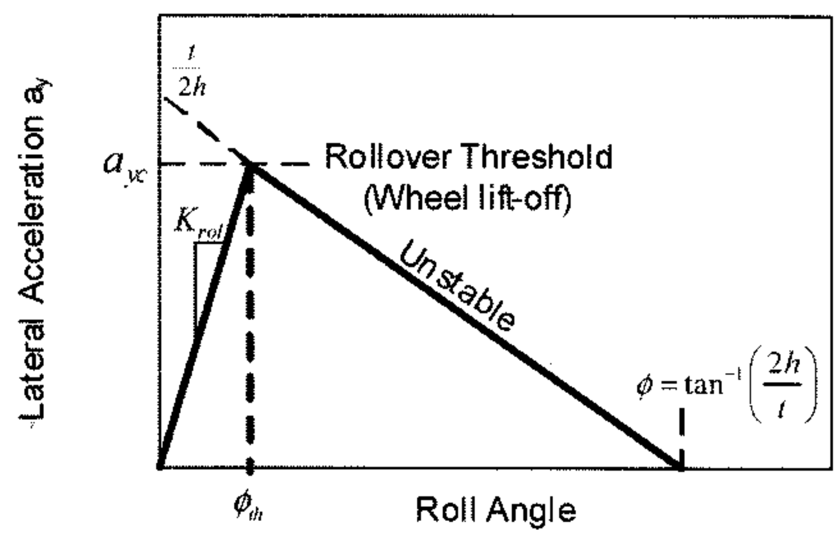


Figure 1. Static rollover characteristics of a suspended vehicle.

determine the RI, a rollover threshold should be defined and significant factors which affect rollover should be determined. Once wheel lift is occurring, rollover mitigation control to reduce the roll motion of the vehicle is very difficult, so a wheel lift threshold is applied for detection of rollover.

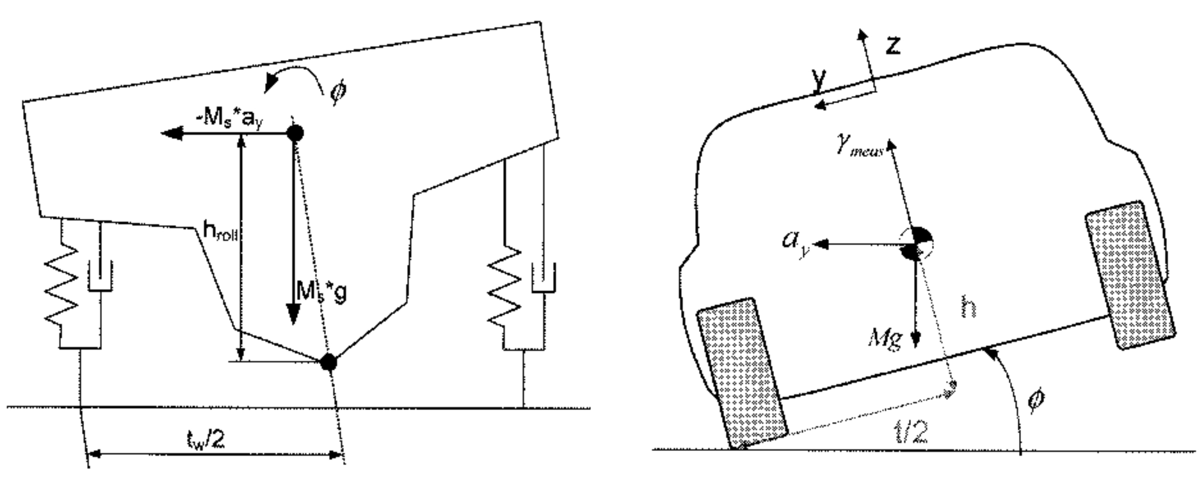
Figure 1 shows static rollover analysis of a suspended vehicle (Gillespie, 1992; Schubert *et al.*, 2004), that is the relation between vehicle parameters such as track width, t, height of CG, h, and roll stiffness, K_{roll}, and vehicle states such as roll angle and lateral acceleration. Through the above analysis of static rollover, we can estimate the roll angle threshold, ϕ_{th}, and the critical lateral acceleration, a_{yc}, that may induce rollover.

Figure 2 shows the vehicle model for two conditions: before and after wheel-lift-off. The roll dynamics of the vehicle can be represented by two different equations of motion depending on tire contact with ground, i.e., before and after wheel-lift-off. The equations of motion for the vehicle model are expressed as follows:

$$\ddot{\phi} = -\frac{C_{roll}}{I_{x1}}\dot{\phi} - \frac{K_{roll}}{I_{x1}}\phi + \frac{M_s(a_y + g\sin\phi)\cdot h_{roll}}{I_{x1}} \quad (\phi < \phi_{th}) \tag{1}$$

$$\ddot{\phi} = \frac{M(a_y + g\sin\phi)\cdot h}{I_{x2}} - \frac{Mg(t/2)\cos\phi}{I_{x2}} \quad (\phi \geq \phi_{th}) \tag{2}$$

where, I_{x1} and I_{x2} are the moment of inertia before and after wheel lift-off respectively. M is the total mass and M_s is the sprung mass of the vehicle. h_{roll} is the height of the roll center.



(a) Before wheel lift off (b) After wheel lift off

Figure 2. Vehicle model.

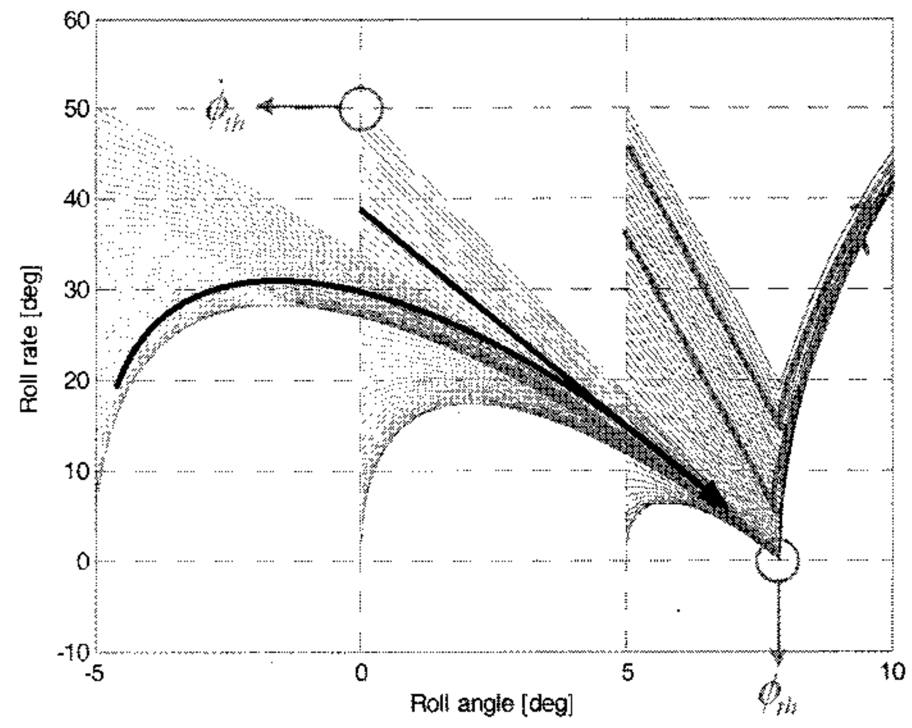


Figure 3. Phase plane analysis for different initial conditions.

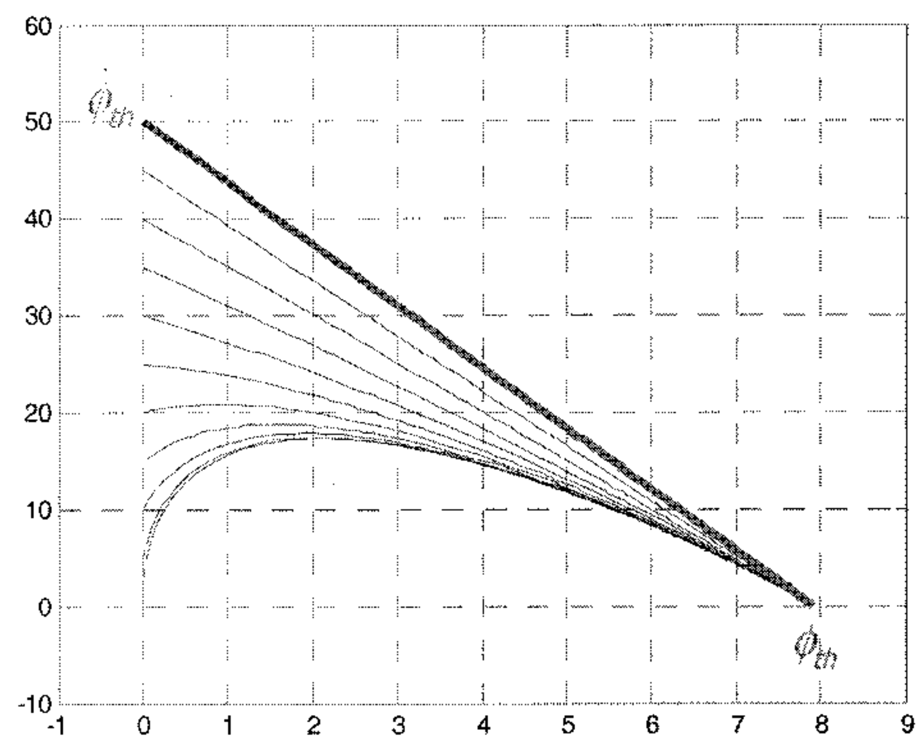


Figure 4. Wheel lift threshold in phase plane.

Using the critical lateral acceleration and the equation (1) and (2), we have performed ($\phi - \dot{\phi}$) phase plane analysis as illustrated in Figure 3 to investigate the relation between roll angle and roll rate in the phase plane with various initial conditions. The default SUV in the CARSIM was used as a subject vehicle. The lateral acceleration is assumed to be the critical acceleration. The roll angle and roll rate ($\phi - \dot{\phi}$) phase plane analysis can offer useful information about the roll state at the critical lateral acceleration.

From the phase plane analysis, wheel-lift-threshold can be obtained as illustrated in Figure 4. If the roll state of the vehicle is below the wheel lift threshold, the vehicle is stable, but when it is near or exceeds the wheel lift threshold, the danger of rollover increases.

Let us introduce a RI that can indicate the danger of rollover using the wheel lift threshold. Significant factors which can determine the RI from the present state of the vehicle are classified into three categories: (1) present states of roll angle and roll rate of the vehicle; (2) present lateral acceleration of the vehicle; and 3) time to wheel lift.

Factor 1) Present states of roll angle and roll rate of the vehicle: when the roll state of the vehicle is near the

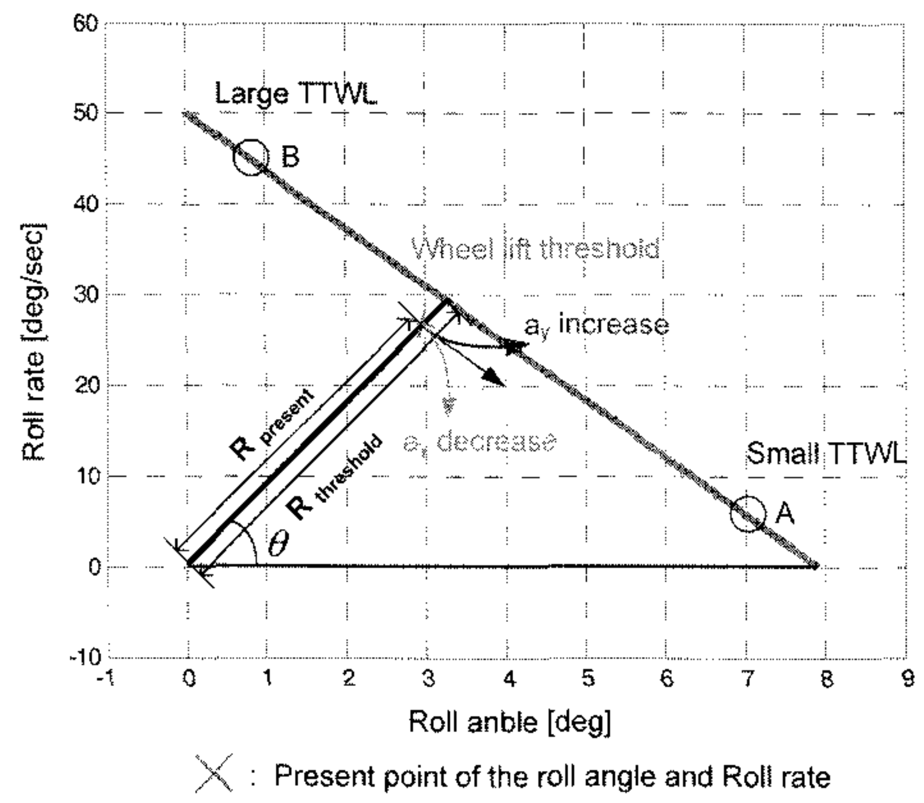


Figure 5. Significant factors in rollover index.

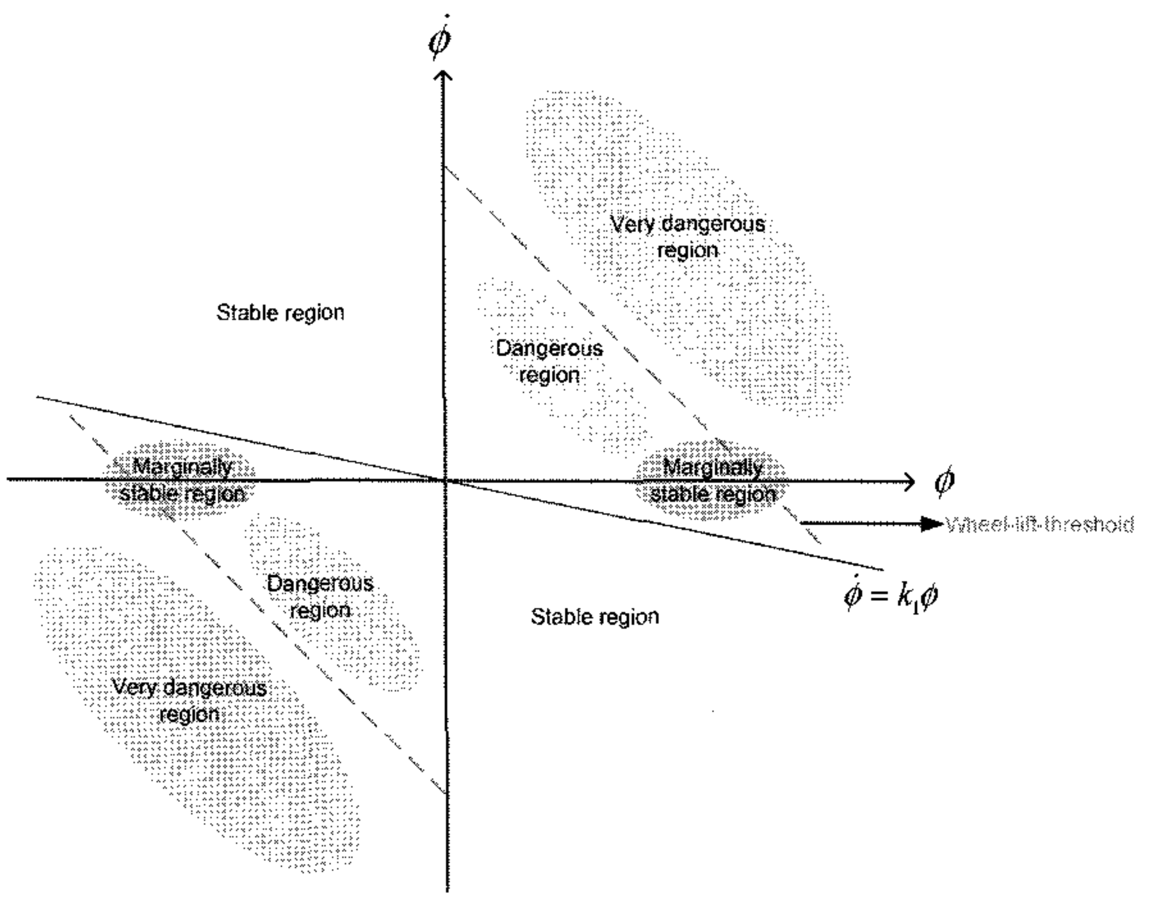


Figure 6. Stable/Dangerous regions on the phase plane domain.

wheel lift threshold, the danger of rollover increases as illustrated in Figure 5.

Factor 2) Present lateral acceleration of the vehicle: The trajectory of the roll state depends on the lateral acceleration. Although the roll state of the vehicle is small, large lateral acceleration may lead the trajectory of the vehicle toward the wheel-lift-threshold as illustrated in Figure 5.

Factor 3) Time to wheel lift (TTWL): We have introduced the TTWL to improve the accuracy of the RI. Although the roll states of the point A and B in Figure 5 are on the wheel-lift-threshold, the TTWL are different from each other. Compared to the point B, the wheel lift is impending at the point A, i.e., the time to wheel lift at the point A is smaller than the point B.

Figure 6 shows the stable and dangerous region on the phase plane. In the quadrant 1 and 3, the magnitude of the roll angle increases. In the quadrant 2 and 4, because the signs of the roll angle and roll rate are opposite, the magnitude of the roll angle decreases. In the region of large roll angle, since the wheel-lift-off can happen depending on the lateral acceleration, the vehicle is marginally stable.

The RI is defined as follows:

$$
\begin{aligned}
RI &= f(a_y, \phi, \dot{\phi}, a_{yc}, \phi_{th}, \dot{\phi}_{th}) \\
&= C_1\left(\frac{|\phi(t)|\dot{\phi}_{th} + |\dot{\phi}(t)|\phi_{th}}{\phi_{th}\dot{\phi}_{th}}\right) + C_2\left(\frac{|a_y|}{a_{yc}}\right) \\
&\quad + (1 - C_1 - C_2)\left(\frac{|\phi(t)|}{\sqrt{(\phi(t))^2 + (\dot{\phi}(t))^2}}\right), \quad \phi(\dot{\phi} - k_1\phi) > 0 \\
RI &= 0 \qquad\qquad\qquad\qquad\qquad\qquad\qquad\quad , \phi(\dot{\phi} - k_1\phi) \le 0
\end{aligned}
\tag{3}
$$

where, C_1, C_2 and k_1 are positive constants.

$(0 < C_1 < 1,\ 0 < C_2 < 1)$

The proposed RI indicates the danger of rollover. The RI of 1 indicates the wheel-lift-off.

3. ROLLOVER MITIGATION CONTROL

In this study, a rollover mitigation control scheme has been designed based on the RI as shown in Figure 6.

The purpose of the RMC is insuring vehicle stability during roll motion using a differential braking law. If the RI increases to a predefined RI threshold, a RMC input is applied to mitigate the roll motion of the vehicle. The control target is defined by the desired RI. The RMC scheme has five procedures: 1) define the RI_{des}, desired rollover index; 2) determine the $a_{y,des}$, desired lateral acceleration, which can reduce the RI to the RI_{des}; 3) determine the γ_{des}, desired yaw rate, which can produce the desired acceleration; 4) determine the required moment using a sliding controller and desired yaw rate; and 5) determine the brake pressure which can produce the required moment.

First of all, the desired RI, RI_{des}, was defined. To reduce the present RI to the desired RI, the lateral acceleration

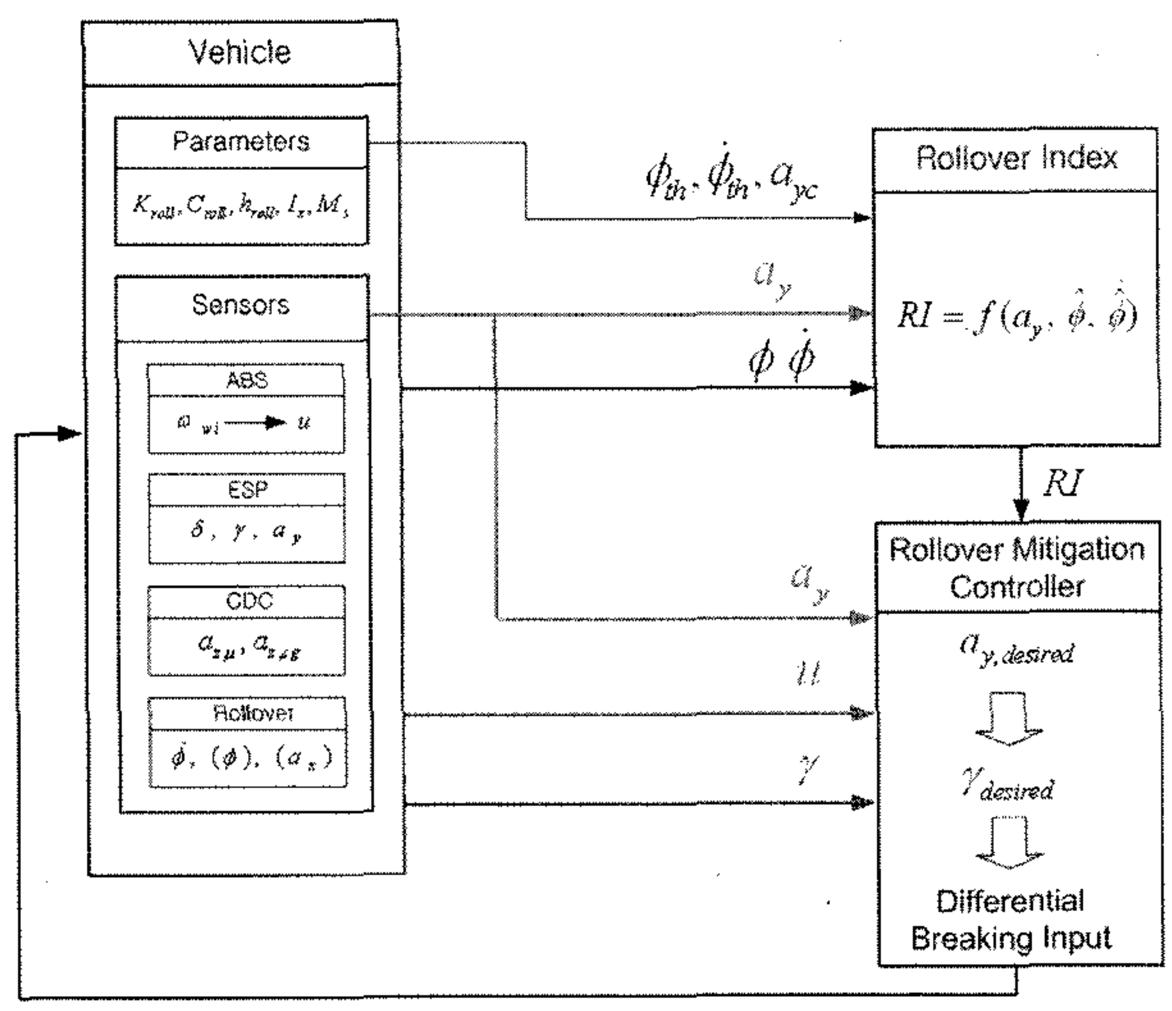


Figure 7. Schematic diagram of a RI-based RMC scheme.

must be reduced to the desired value. The desired lateral acceleration can be determined using the equation (3). If the RI and a_y susbstitute for the RI_{des} and $a_{y,des}$ in the equation (3), the desired lateral acceleration which can reduce the present RI to the RI_{des} is calculated from equation (3) using a predefined RI_{des} as follows:

$$a_{y,des} = \frac{1}{C_2}\left(\begin{array}{l} RI_{des} - C_1\left(\frac{|\phi(t)|\dot{\phi}_{th} + |\dot{\phi}(t)|\phi_{th}}{\phi_{th}\dot{\phi}_{th}}\right) \\ -(1 - C_1 - C_2)\left(\frac{|\phi(t)|}{\sqrt{(\phi(t))^2 + (\dot{\phi}(t))^2}}\right) \end{array}\right) \times a_{y,c} \tag{4}$$

where, $a_{y,des}$ is the desired lateral acceleration and $a_{y,c}$ is the predefined critical lateral acceleration.

When only the plane motion of the vehicle is considered, the derivative of the lateral velocity with the vehicle body fixed coordinate is defined as:

$$\dot{v} = a_{y,m} - u\gamma \tag{5}$$

where, $a_{y,m}$ is the sensor measurement of the lateral acceleration at the CG. u is the longitudinal velocity, and γ is the yaw rate.

If the desired yaw rate is exist, the desired lateral accele-ration of the vehicle can be expressed kinematically as follows:

$$a_{y,des} = \dot{v} + u\gamma_{des} \tag{6}$$

Because the $\dot{v}$ is the lateral acceleration with the vehicle body fixed coordinate in the equations (5) and (6), using equations (5) and (6), the desired yaw rate which can produce the desired lateral acceleration is calculated as follows:

$$\gamma_{des} = \frac{1}{u}\{a_{y,des} - (a_{y,m} - u\gamma)\} \tag{7}$$

where $a_{y,m}$: measurement a_y by a sensor.

Figure 8 shows the 2-D vehicle model including direct yaw moment (M_z) and the dynamic equations can be expressed as follows:

$$\begin{aligned} mu(\beta + \gamma) &= F_{yf} + F_{yr} \\ I_z\dot{\gamma} &= aF_{yf} - bF_{yr} + M_z \end{aligned} \tag{8}$$

The sliding controllers have been designed based on a two degrees-of-freedom vehicle dynamic model, and the

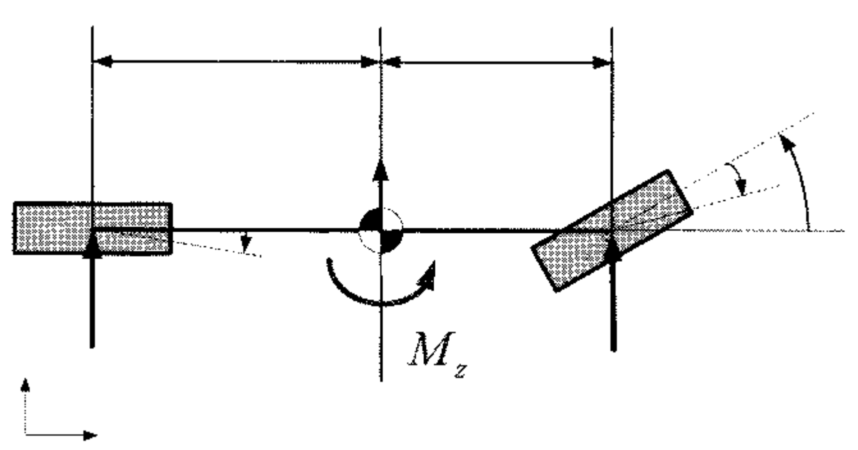


Figure 8. 2-D vehicle model.

performance characteristics of several sliding surfaces for stability control have been investigated using planar vehicle dynamics and a nonlinear tire model (Gillespie, 1992; Schubert *et al.*, 2004). Previous studies have indicated that both slip angle and yaw rate should be combined in the control objective for stability control. The scalar s is defined as a combination of the sideslip angle β and the difference between the desired target yaw rate and actual yaw rate as follows:

$$s = \frac{1}{2}(\gamma_{des} - \gamma)^2 + \frac{1}{2}\rho \cdot \beta^2 \tag{9}$$

where ρ is a positive constant.

The control objective is to keep the scalar s at zero. This can be achieved by choosing the control law such that:

$$\frac{1}{2}\frac{d}{dt}s^2 = s\dot{s} \leq -\eta \cdot s^2 \tag{10}$$

where η is a positive constant.

There exists a sliding controller gain K such that:

$$\dot{s} = -K \cdot s \leq -\eta \cdot s \tag{11}$$

Therefore, the required moment is calculated as follows:

$$\begin{aligned} M_z = {} & I_z\dot{\gamma}_{des} - aF_{yf} + bF_{yr} \\ & + \frac{\rho I_z\beta}{e_\gamma}\left(\frac{C_{\alpha f}\alpha_f + C_{\alpha r}\alpha_r}{mu} - \gamma\right) + \frac{I_zK}{2}\left(e_\gamma + \frac{\rho\beta^2}{e_\gamma}\right) \end{aligned} \tag{12}$$

and the brake pressure to produce the direct yaw moment can be determined as follows:

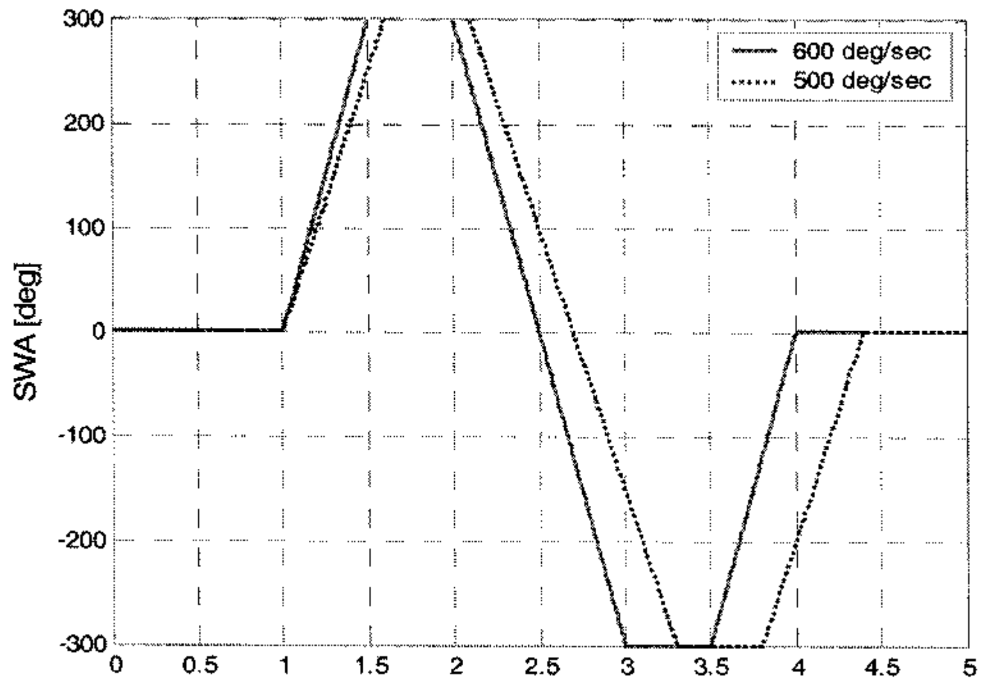
$$\begin{aligned} P_{B,left,com} &= \frac{M_z}{K_B}\frac{r_{wf}}{L/2} + \frac{T_s}{K_B} \qquad (M_z \geq 0) \\ P_{B,right,com} &= -\frac{M_z}{K_B}\frac{r_{wf}}{L/2} + \frac{T_s}{K_B} \qquad (M_z < 0) \end{aligned} \tag{13}$$

4. SIMULATION STUDY

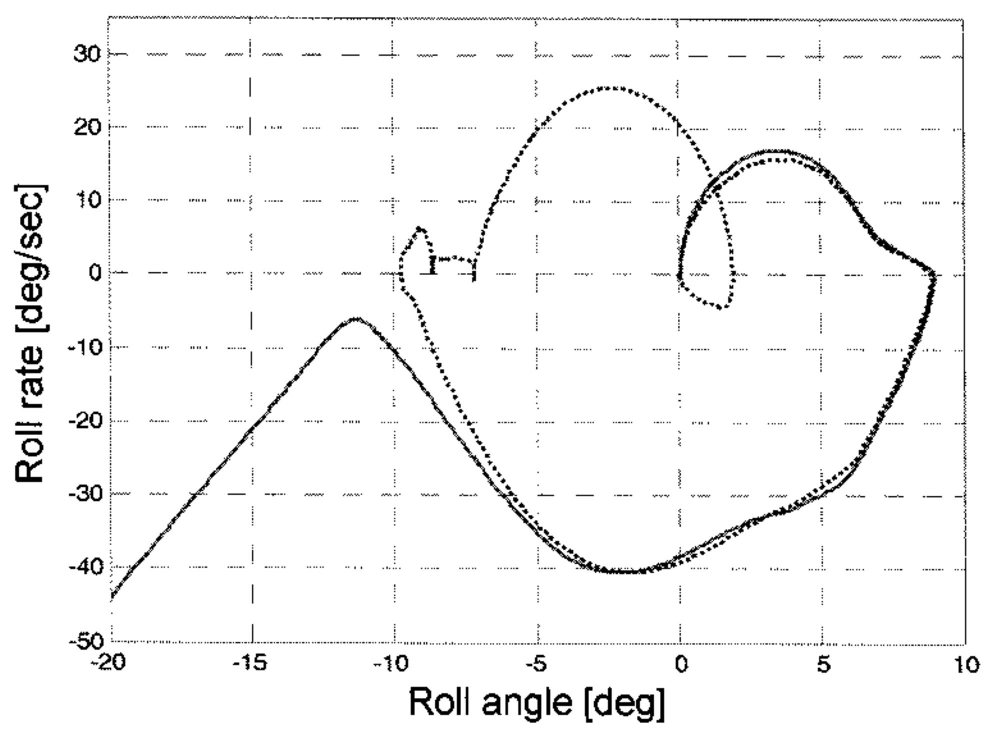
In this study, a simulation is conducted using vehicle simulation software, CARSIM, and MATLAB/Simulink. The estimator and RMC scheme are implemented with MATLAB/Simulink.

4.1. Rollover Index

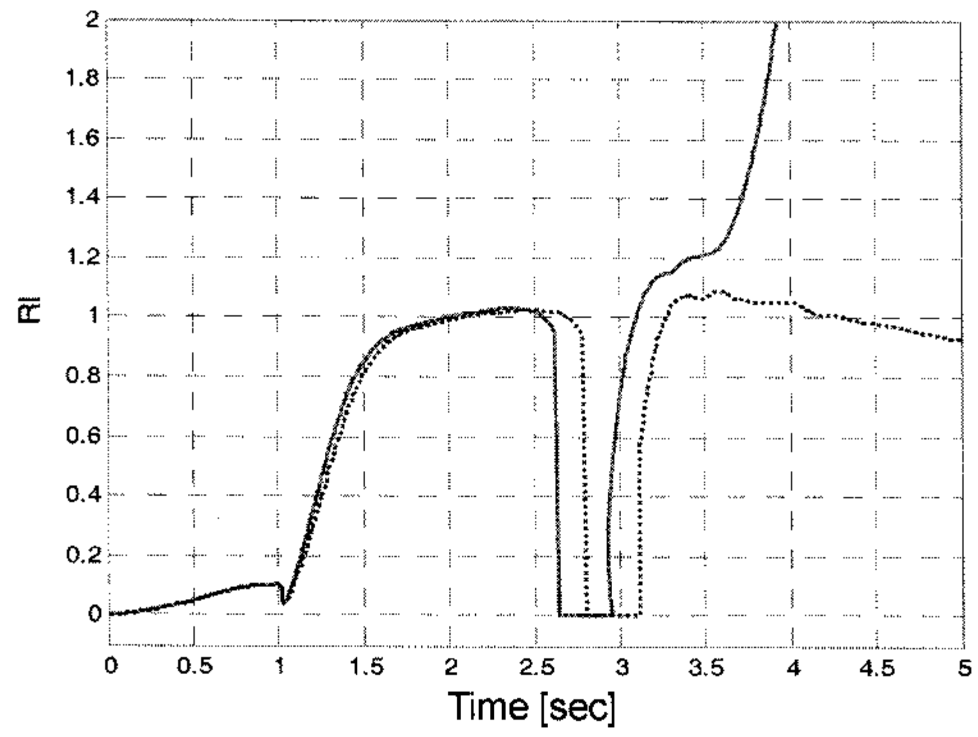
Figure 9 shows the simulation results of the fishhook test for two different steering wheel angular rates: 1) 600 deg/sec and 2) 500 deg/sec. These two simulation cases are very dangerous situations that may cause the rollover. In case of the 600 deg/sec, rollover is occurred. But in case of the 500 deg/sec, rollover is not occurred. In the rollover case, when the counter steering is applied, the roll angle and roll rate increase, and the RI increases over unity suddenly. But in the stable case, the RI is maintained



(a) SWA



(b) Phase plane plot



(c) Rollover index

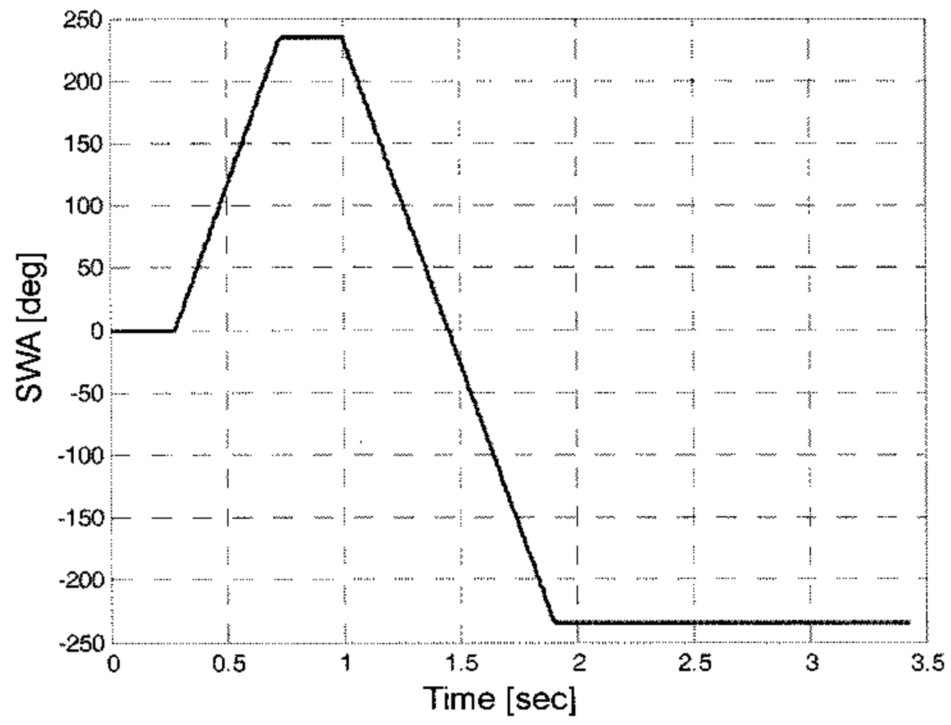
Figure 9. Phase plane plot and rollover index (Fishhook @80 kph).

around the unity. From the simulation results, it has been illustrated that the RI can be a good measure indicating the danger of rollover.

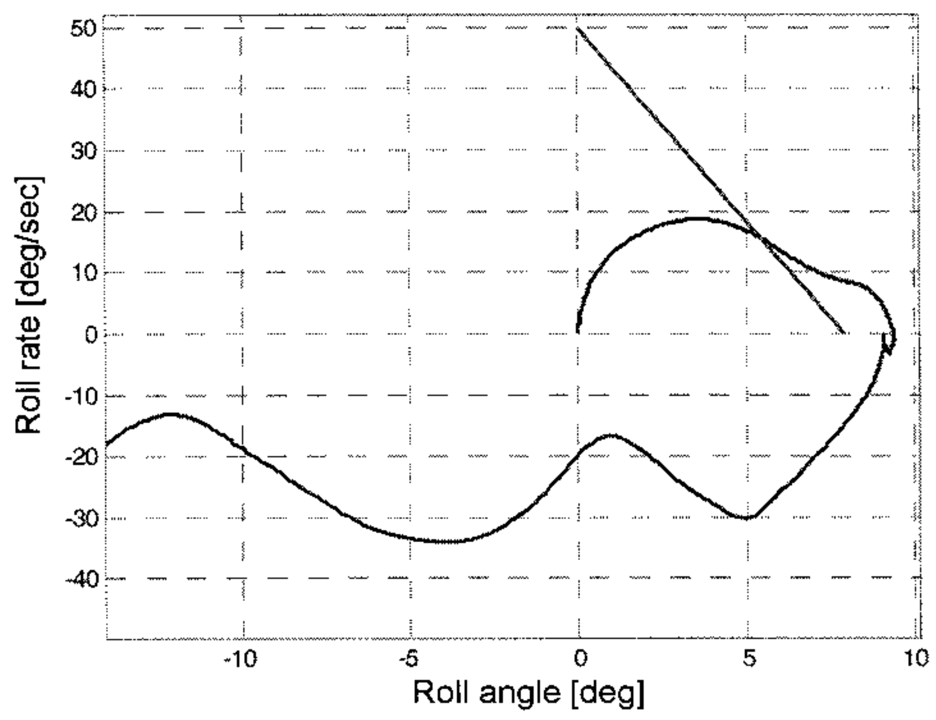
4.2. Rollover Mitigation Control

To verify the performance of the proposed RMC scheme, fishhook test is performed. The test maneuver conforms to NHTSA standards.

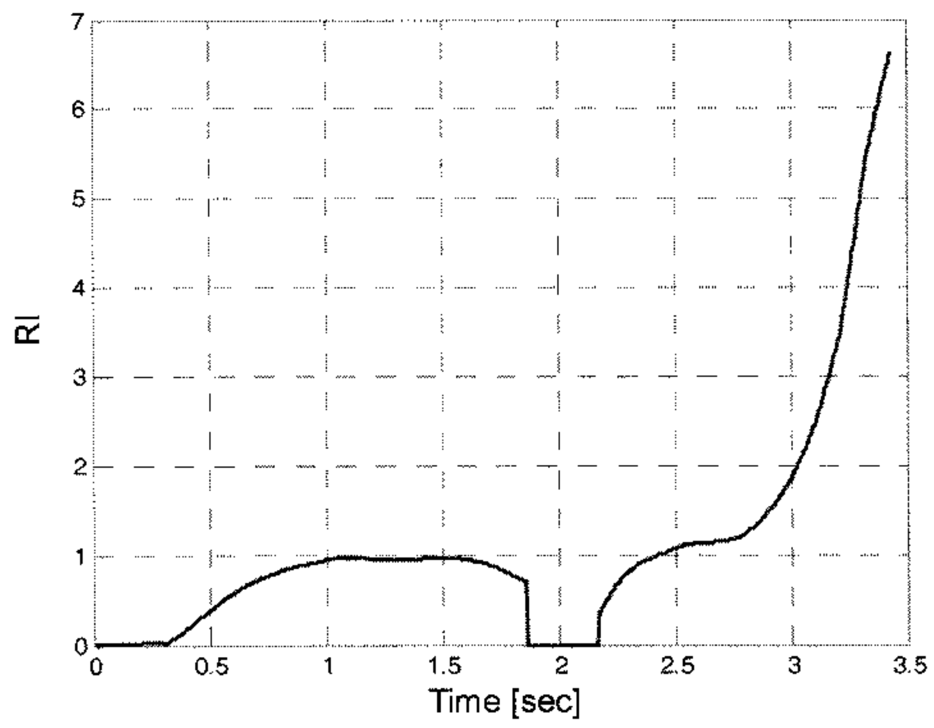
Figure 10 shows NHTSA fishhook test simulation results without RMC. When the steering is applied, although the RI increases to the unity and two wheels of the vehicle are lifted at 1.1 seconds. When the counter



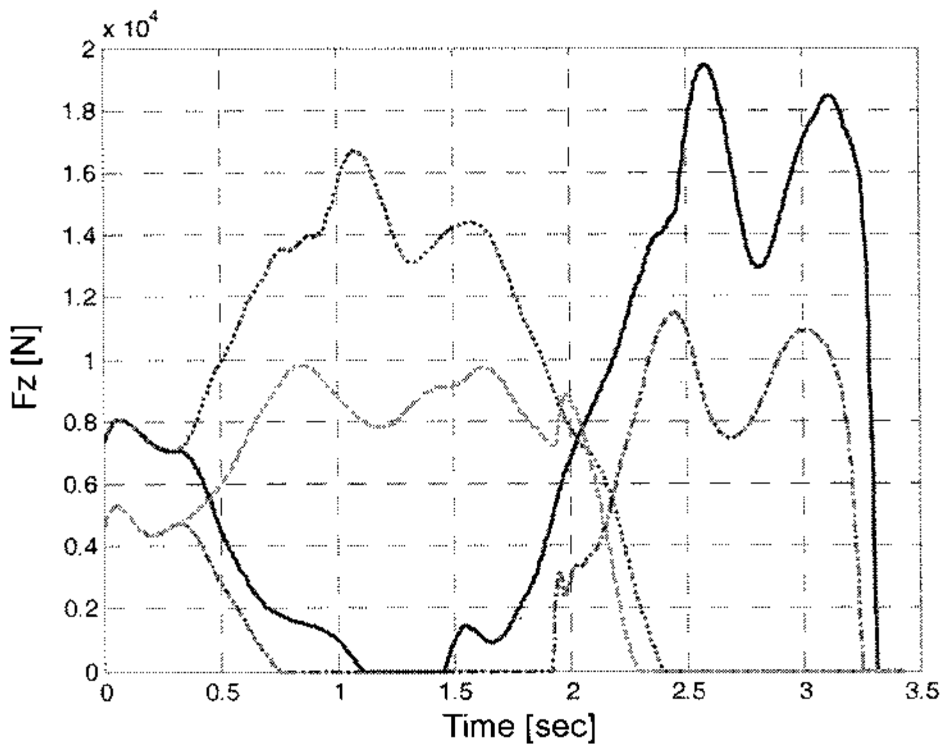
(a) SWA



(b) Phase plane plot

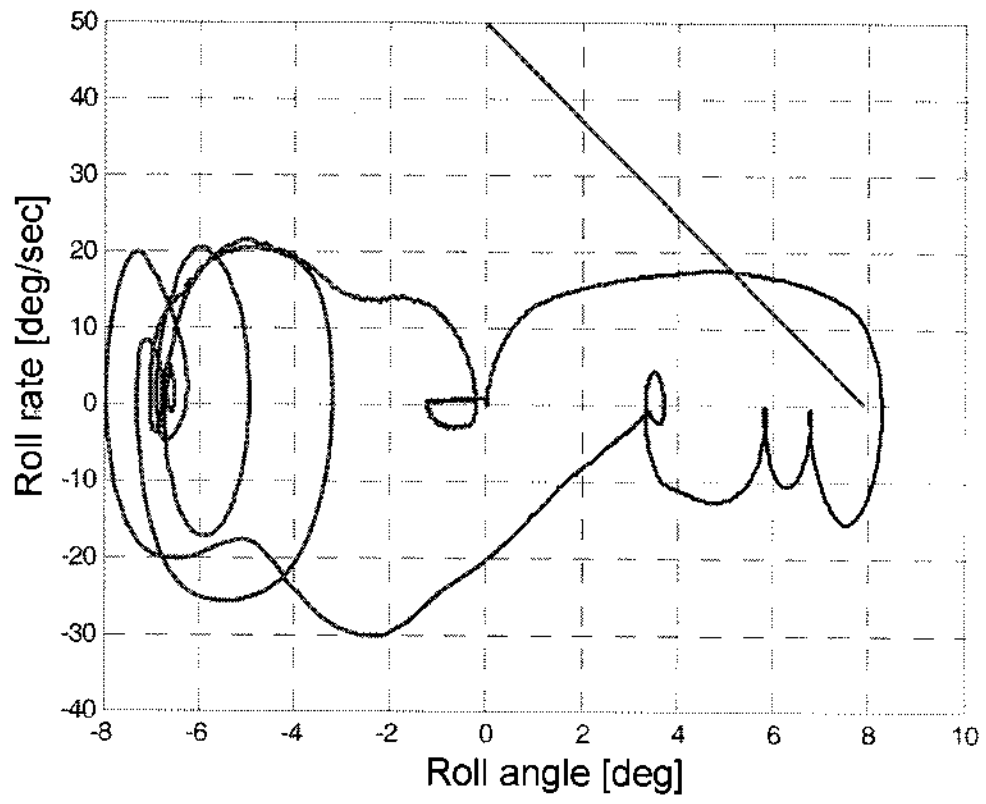


(c) Rollover index

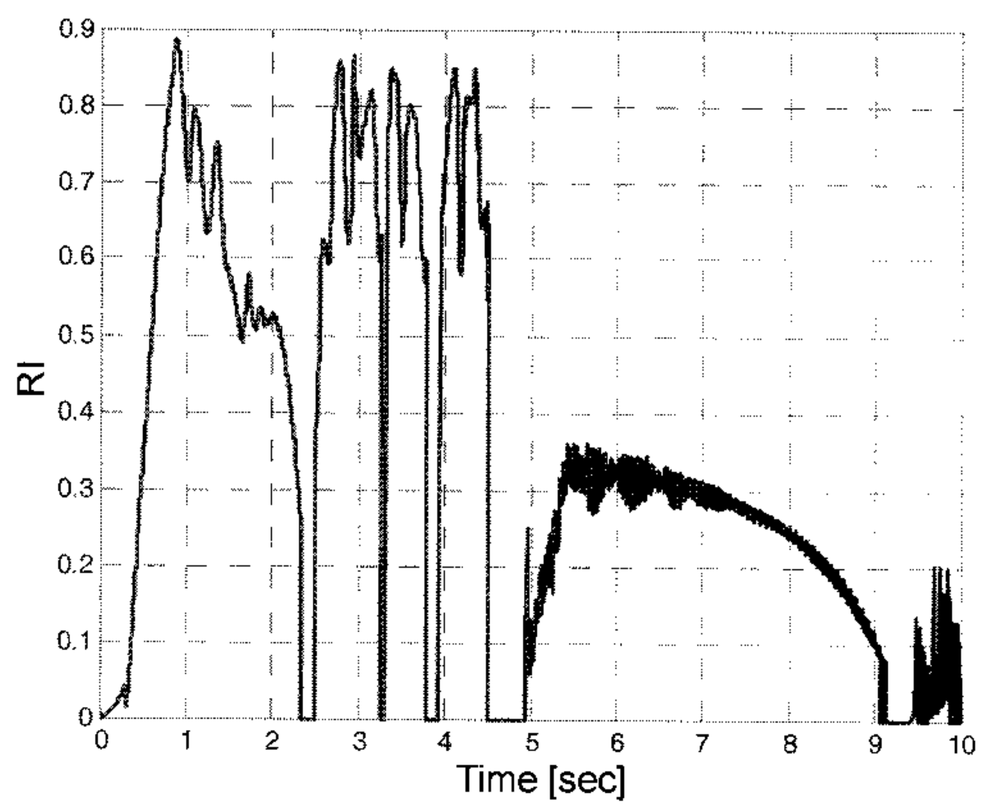


(d) Vertical tire forces

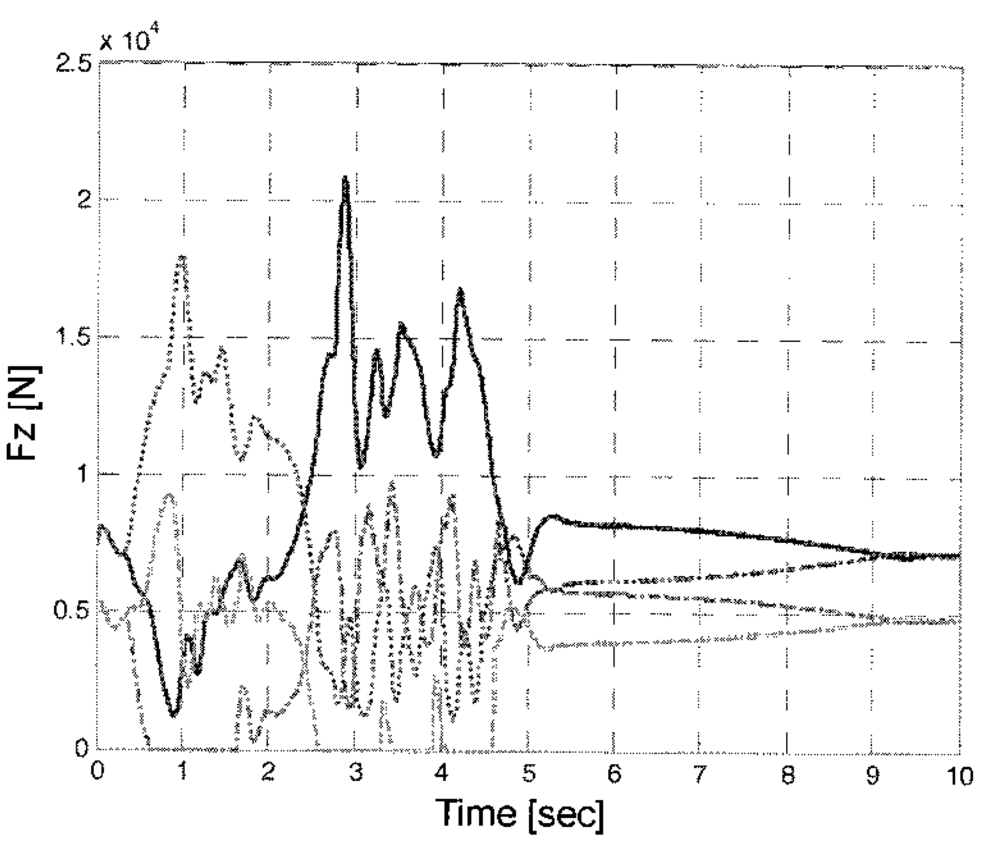
Figure 10. Simulation results of NHTSA fishhook test@ 80 kph (without RMC).



(a) Phase plane plot



(b) Rollover index



(c) Vertical tire forces

Figure 11. Simulation results of NHTSA fishhook test @80 kph (with RMC).

steering is applied, the RI increases over the unity and two wheels are lifted, then rollover is occurred at 2.4 seconds.

Figure 11 shows NHTSA fishhook test results with the RMC. Although one wheel is lifted, rollover is not occurred. The RI is maintained below the unity. Through the simulation results, it has been illustrated that the RMC can mitigate the roll motion of the vehicle and can prevent rollover accidents.

5. CONCLUSIONS

This paper has presented a rollover index-based rollover mitigation control scheme. A rollover index which indicates impending rollover has been proposed. A control threshold has been designed using the RI. The performance of the RI and the RMC are investigated by computer simulations. The simulation results illustrate that the proposed RI can be a good measure of the danger of rollover. The RI-based RMC scheme presented in this paper can efficiently reduce the risk of a rollover.

ACKNOWLEDGEMENT–This work has been supported by Mando Corp.

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