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Unified Chassis Control for Vehicle Rollover Prevention

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Abstract: This paper describes an unified chassis control (UCC) strategy to prevent vehicle rollover and improve maneuverability. In order to detect a danger of rollover, rollover index (RI) which indicates an impending rollover is determined. The rollover index is calculated using estimated roll angle, roll rate and measured lateral acceleration. Lateral and vertical model-based roll state estimators are designed and combined to obtain the vehicle roll state induced by maneuvering and road disturbances. The vehicle mass is adapted to improve the robustness of the roll state estimator. The RI-based rollover mitigation controller (RMC) is designed by integrating the electronic stability control (ESC), active front steering (AFS) and continuous damping control (CDC). The RI/lateral stability-based RMC is also designed to ensure maneuverability. Computer simulation is conducted to evaluate the proposed UCC scheme by using validated vehicle simulation software. From the simulation results, it is shown that the proposed UCC can prevent vehicle rollover and load to improvements in vehicle stability.

1. INTRODUCTION

Vehicle rollover is a serious problem in the transportation community. Even though rollovers constitute a small percentage of all accidents, they have a disproportionately large contribution to severe and fatal injuries (Jang *et al*., 2006). Of the nearly 11 million passenger car, SUV, pickup and van crashes in 2002, only 3% involved a rollover. However, rollovers accounted for nearly 33% of all deaths from passenger vehicle crashes (NHTSA, 2003). In order to help consumers understand a vehicle's likelihood to rollover, a Rollover Resistance Rating program was proposed by NHTSA. This program uses the Static Stability Factor (SSF), which is the ratio of one half the track width to the center of gravity (CG) height, to determine the rating. However, from automotive industries' comments, SSF was too simple because it did not consider the effects of suspension deflection, tire traction, and vehicle dynamics control. Therefore, NHTSA intends to publish another notice to present a tentative dynamic rollover test procedure in 2002 (NHTSA, 2001). 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 *et al*., 2001). Hac and Martens described a rollover index using a model-based roll estimator (Hac *et al*., 2004). Yang and Liu also presented a rollover index which is a combination of rollover indices from influential factors such as the position of vehicle's CG height, the energy of rollover and vertical tire forces (Yang *et al*., 2003). Kim and Oh proposed two main rollover criteria, Rotational Kinetic Energy (RKE) and Initial Kinetic Energy (IKE), based on simple physical model (Kim *et al*., 2006).

In this paper, UCC system is proposed to prevent vehicle rollover and improve maneuverability. A rollover index (RI) is introduced for detecting rollover. The RI provides an assessment of impending rollover danger, and thus could be the basis for rollover prevention. Because the roll angle and roll rate are required to calculate the RI, a model-based roll state estimator is designed. The proposed roll state estimator can obtain good estimates in situations in which both maneuvering and road disturbances affect the vehicle roll motions. The RI-based rollover mitigation controller (RMC) is designed by integrating the electronic stability control (ESC), active front steering (AFS) and continuous damping control (CDC). The RI/lateral stability-based RMC is also designed to ensure maneuverability. Fig. 1 shows a schematic diagram of UCC system for rollover prevention.

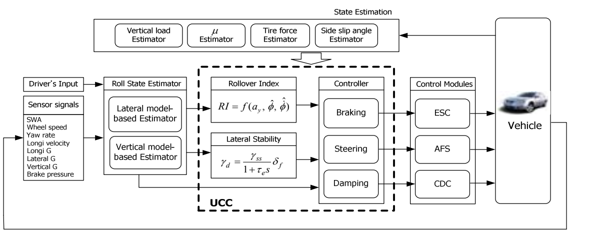


Fig. 1. Schematic diagram of UCC system for rollover prevention

The proposed UCC system is evaluated via computer simulations conducted using the vehicle dynamic software. Computer simulations of a closed-loop driver-vehicle-controller system subjected to circular turning are conducted to verify the performance of the proposed UCC system over individual chassis control systems.

2. Rollover Index (RI)

In this study, The RI is designed to detect the danger of rollover. It is a dimensionless number which indicate an impending rollover. Since the roll angle and roll rate is required to calculate the RI, a model-based roll state estimator is designed.

2.1 Model-based Roll State Estimator

Vehicle roll motions are generally induced by driver's maneuvering and road disturbances. In this study, a model-based roll state estimator is designed based on the lateral dynamics model and vertical dynamics model to estimate the roll angle and roll rate. An estimator is designed using sensor measurements such as steering angle, lateral acceleration, yaw rate and vertical accelerations of sprung/unsprung mass. These measurements are available on a vehicle equipped with ESC and CDC systems. Combining the lateral model estimator and the vertical model estimator, a road-disturbance decoupled roll state estimator is designed to obtain good estimates of roll angle and roll rate in driving situations in which roll motions are jointly induced by maneuvering and road disturbances (Yoon *et al*., 2007a). Vehicle parameters are very important factors to estimate the roll states. However, since vehicle parameter such as vehicle mass is variable caused by passengers and payload, the performance of the roll state estimator cannot be guarantee. For this reason, parameter adaptation algorithm is applied by using simple adaptive control law (Park *et al*., 2007). Fig. 2 shows schematic diagram of model-based roll state estimator.

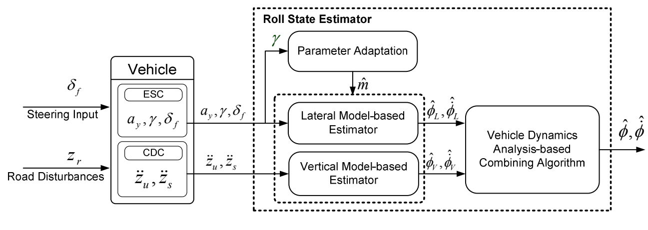


Fig. 2. Model-based roll state estimator

The lateral dynamics model-based roll state estimator is designed to estimate the maneuvering-induced roll states of vehicle. 2-D bicycle model and 2-D simple roll model are used to design the estimator. By using sensor signals such as lateral acceleration and yaw rate obtained by ESC are used to measurements. The estimator is designed as follows:

$$\begin{aligned} \dot{\hat{x}} &= A\hat{x} + Bu + K(y - \hat{y}) \\ \hat{y} &= C\hat{x} + Du \\ y &= \begin{bmatrix} a_y & \gamma \end{bmatrix}^T + \begin{bmatrix} n_{a_y} & n_y \end{bmatrix}^T \end{aligned} \tag{1}$$

When the only maneuvering input is applied, the lateral model-based roll state estimator works very well. However, when the road disturbances exist, it shows poor performance. As a result, it is not sufficient that the roll angle and roll rate can be estimated by using only the lateral dynamics model-based roll state estimator.

In order to compensate the lateral model-based estimator, the vertical model-based estimator is designed to estimate the roll motion induced by road disturbances. A four-degree-of freedom half-car suspension model is used to design the estimator. Measurement signals for the vertical model-based estimator are the accelerations of sprung and unsprung mass of the front-left and right side respectively. The accelerations of the unsprung mass can be measured easily from the conventional vehicle equipped with a CDC module. By using the vertical model-based roll state estimator, the roll angle and roll rate can be obtained as follows:

$$\begin{aligned} \hat{\dot{\phi}}(t) &= \frac{x_2 - x_4}{t} = \frac{(z_2 + z_5) - (z_4 + z_6)}{t} \\ \hat{\phi}(t) &= \frac{s}{s + w_c}\frac{1}{s}\hat{\dot{\phi}}(t) \end{aligned} \tag{2}$$

Where, $\hat{x}_2$ and $\hat{x}_4$ are velocities of sprung mass of left and right side respectively.

Since the proposed two estimators work only well in driving situations in which maneuvering and road disturbances affect the vehicle roll motions respectively, it is necessary to design the combined estimator to improve the performance. But it should not combine the proposed two estimators simply because they are coupled. The maneuvering is a dominant input of the lateral dynamics model-based estimator and the road disturbances are dominant inputs of the vertical dynamics model-based estimator. But when the maneuvering and road disturbances inputs exist simultaneously, each estimator is affected by them as shown in Fig. 3. In this study, the combining gains are selected adequately through the steady-state tuning and frequency analysis of the road disturbances.

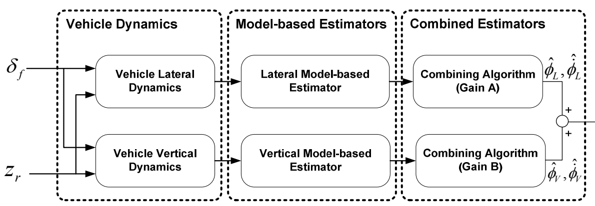


Fig. 3. Combining the model-based estimators

2.2 Rollover Index

The RI is calculated as a function of the measured lateral acceleration (a_y), estimated roll angle ($\hat{\phi}$), roll rate ($\hat{\dot{\phi}}$), and their critical and threshold values depend on vehicle parameters as follows:

$$RI = C_1\left(\frac{|a_y|}{a_{yc}}\right) + C_2\left(\frac{|\phi(t)|\dot{\phi}_{th} + |\dot{\phi}(t)|\phi_{th}}{\phi_{th}\dot{\phi}_{th}}\right) + (1 - C_1 - C_2)\left(\frac{|\phi(t)|}{\sqrt{(\phi(t))^2 + (\dot{\phi}(t))^2}}\right) \quad (3)$$

The critical lateral acceleration is defined as the maximum lateral acceleration achievable on a dry surface in a steady-state turn when one wheel is lift-off. The critical roll angle and roll rate are defined as the maximum roll angle and roll rate in a steady-state turn when one wheel is lift-off. The critical values are obtained through phase plane analysis using vehicle parameters. The function is tuned such that the RI of 1 indicates the wheel-lift-off (Yoon et al., 2007b).

3. Controller Design

Rollover mitigation controller is designed by integrating the modular chassis controller such as ESC, AFS and CDC. In this study, two of control techniques are investigated. The one is a RI-based RMC strategy, and the other is a RI/Lateral stability-based RMC strategy. The RI-based RMC operates to reduce the current RI to desired RI (RI_{des}). The RI/Lateral stability-based RMC is designed to satisfy that the RI is reduced to desired RI and vehicle lateral stability which can reduce the yaw rate error is guaranteed.

3.1 RI-based Rollover Mitigation Controller

The purpose of the RI-based RMC uses the RI as a control threshold and target values. When the RI exceeds a threshold value, RMC is activated to reduce the RI to RI_{des}. Fig. 4 shows schematic diagram of RI-based RMC strategy. An upper level controller calculates a desired yaw moment and roll moment from the RI and vehicle states. By using calculated desired yaw moment, a lower level controller determines the detailed control inputs such as differential braking pressures, additional front steering angle and current to apply to ESC, AFS and CDC respectively. Roll stability controller using CDC module always operate separately to minimize the vehicle roll motions. More details about the roll stability control can refer to reference (Yoon et al., 2007c). In this section, moment distribution to generate the desired yaw moment is only describes.

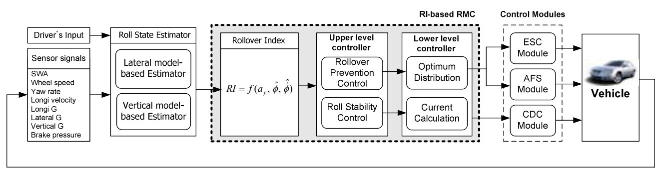


Fig. 4. RI-based RMC strategy

If the RI exceeds a predefined RI threshold (RI_{th}), the upper level controller is activated and calculates the desired yaw moment to reduce the RI to RI_{des}. From (3), a desired lateral acceleration can be calculated to reduce the present RI to RI_{des}. And then, a desired yaw rate is calculated through the vehicle lateral dynamics. Finally, a desired yaw moment for rollover prevention (M_z) is obtained to generate the desired yaw rate as follows (Yoon et al., 2007a):

$$M_{z,RMC} = I_z\,\dot{\gamma}_{des} - aC_f\alpha_f + bC_r\alpha_r + \frac{\rho I_z \beta}{\gamma_{des} - \gamma}\left(\frac{C_f\alpha_f + C_r\alpha_r}{mv_x} - \gamma\right) + \frac{I_z K_1}{2}\left((\gamma_{des} - \gamma) + \frac{\rho\beta^2}{(\gamma_{des} - \gamma)}\right) \quad (4)$$

The lower level controller deals with an optimum problem for moment distribution. In a conventional ESC, the desired yaw moment is generated by differential braking. The differential braking leads to significant longitudinal decelerations and pitching motions of the vehicle body. These could be sensed by the driver and thus lead to a degradation of ride comfort. In addition, braking control inputs could lead to wear of tires and brakes. Yaw moment generation by AFS can be a solution to these problems. In order to minimize the usage of the braking, an optimized coordination of the AFS and ESC has been proposed in this study. An optimal coordination of the active lateral and longitudinal tire forces ($\Delta F_x, \Delta F_y$) for the desired yaw moment are determined by Karush-Kuhn-Tucker (KKT) conditions. Fig. 5 shows coordinate system corresponding to resultant force. The longitudinal and lateral forces are computed depending on the sign of the desired yaw moment. If the desired yaw moment is positive, four variables ($\Delta F_{x1}, \Delta F_{y1}, \Delta F_{y2}, \Delta F_{x3}$) can be used to generate the yaw moment. Since the same active steering angle is used for both of the front tires, the active lateral force for the tire 2 can be represented as

$$\Delta F_{y2} = \frac{F_{z2}}{F_{z1}} \cdot F_{y1} \quad (5)$$

Using the following braking force distribution strategy, rear tire force can be represented as

$$\Delta F_{x3} = \frac{F_{z3}}{F_{z1}} \cdot \Delta F_{x1} \quad (6)$$

Two ($\Delta F_{y2}, \Delta F_{x3}$) of the four variables can be eliminated in the optimization problem. The cost function of the proposed optimization process is the magnitude of the additional longitudinal tire force by braking as follows:

$$L(\Delta F_x) = \Delta F_{x1}{}^2 \quad (7)$$

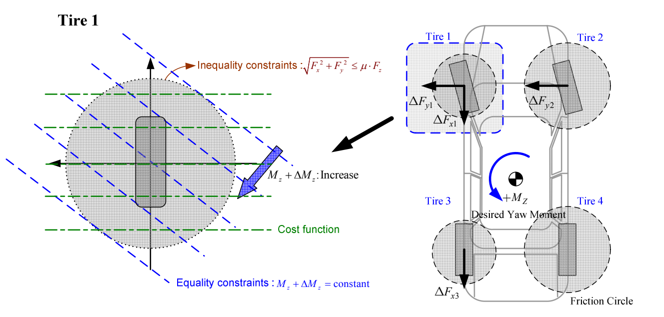


Fig. 5. Coordinate system corresponding to resultant force

This optimization problem has two variables ($\Delta F_{x1}, \Delta F_{y1}$), one equality constraint and one inequality constraint. Two of constraints are as follows:

$$f(x) = -\frac{t}{2} \cdot D_1 \cdot \Delta F_{x1} + l_f \cdot D_2 \cdot \Delta F_{y1} - \Delta M_Z = 0 \tag{8}$$

$$g(x) = \left(\Delta F_{x1} + F_{x1}\right)^2 + \left(\Delta F_{y1} + F_{y1}\right)^2 - \mu^2 \cdot F_{z1}{}^2 \le 0 \tag{9}$$

where, $D_1 = 1 + \frac{F_{z3}}{F_{z1}}$, $D_2 = 1 + \frac{F_{z2}}{F_{z1}}$

Using from (7) to (9), Hamiltonian is defined as follows:

$$\begin{aligned} &H(\Delta F_{x1}, \Delta F_{y1}, \lambda, \rho, c) \\ &= \Delta F_{x1}{}^2 + \lambda \cdot \left(-\frac{t}{2} \cdot D_1 \cdot \Delta F_{x1} + l_f \cdot D_2 \cdot \Delta F_{y1} - \Delta M_Z \right) \\ &\quad + \rho \left(\left(\Delta F_{x1} + F_{x1}\right)^2 + \left(\Delta F_{y1} + F_{y1}\right)^2 - \mu^2 \cdot F_{z1}{}^2 + c^2 \right) \end{aligned} \tag{10}$$

Where, λ is Lagrange multiplier, c is slack variable, and ρ is positive value.

First order necessary conditions about Hamiltonian are determined by Karush-Kuhn-Tucker (KKT) condition theory as follow:

$$\begin{aligned} &\frac{\partial H}{\partial \Delta F_{x1}} = 2 \cdot \Delta F_{x1} - \frac{t}{2} \cdot D_1 \cdot \lambda + 2\rho\left(\Delta F_{x1} + F_{x1}\right) = 0 \\ &\frac{\partial H}{\partial \Delta F_{y1}} = l_f \cdot D_2 \cdot \lambda + 2\rho\left(\Delta F_{y1} + F_{y1}\right) = 0 \\ &\frac{\partial H}{\partial \lambda} = -\frac{t}{2} \cdot D_1 \cdot \Delta F_{x1} + l_f \cdot D_2 \cdot \Delta F_{y1} - \Delta M_Z = 0 \\ &\rho \cdot g(x) = \rho \cdot \left(\left(\Delta F_{x1} + F_{x1}\right)^2 + \left(\Delta F_{y1} + F_{y1}\right)^2 - \mu^2 \cdot F_{z1}{}^2 \right) = 0 \end{aligned} \tag{11}$$

From (11), longitudinal and lateral tire forces for ESC and AFS are obtained. When the desired yaw moment is positive and ρ is zero, tire forces are calculated as

$$\Delta F_{x1} = 0, \quad \Delta F_{y1} = \frac{\Delta M_Z}{l_f \cdot D_2} \tag{12}$$

If ρ is positive, tire forces are calculated as follows:

$$\begin{cases} \Delta F_{x1} = \dfrac{-(F_{x1} + \kappa \cdot \zeta) + \sqrt{(1+\kappa^2) \cdot \mu^2 \cdot F_{z1}{}^2 - (\kappa \cdot F_{x1} - \zeta)^2}}{(1+\kappa^2)} \\ \Delta F_{y1} = \dfrac{t \cdot D_1}{2 \cdot l_f \cdot D_2} \Delta F_{x1} + \dfrac{1}{l_f \cdot D_2} \Delta M_Z \end{cases} \tag{13}$$

Where, $\kappa = \frac{t \cdot D_1}{2 \cdot l_f \cdot D_2}$, $\zeta = \frac{1}{l_f \cdot D_2} \Delta M_Z + F_{y1}$

When the desired yaw moment is negative, the tire forces can be obtained similar with (12) and (13).

3.2 RI/Lateral Stability-based Rollover Mitigation Controller

Although the RMC strategy considered only RI shows good performance for rollover prevention. However, it tends to control the vehicle opposite direction to driver's intention. This may cause that vehicle is departed from the road. For this reason, another approach is investigated in this section, that is, RI/Lateral stability-based RMC strategy. RI/Lateral stability-based RMC is designed to satisfy that the vehicle can follow an intended path of driver while reducing rollover danger. Fig. 6 shows a schematic diagram of the RI/Lateral stability-based RMC system. The desired braking force which should be subjected to vehicle is obtained from the RI, while at the same time calculating the desired yaw moment for lateral stability. By using the desired braking force and the desired yaw moment, braking forces of four wheels are calculated respectively.

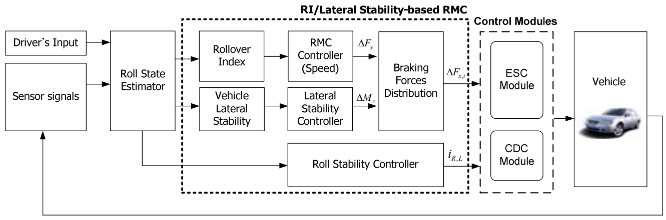


Fig. 6. RI/Lateral stability-based RMC strategy

By using the desired lateral acceleration calculated in previous section, a desired vehicle speed ($v_{x,des}$) can be obtained through the vehicle dynamics. The desired braking force to yield the desired vehicle speed is calculated using a planar model as shown in Fig. 7 and sliding mode control law.

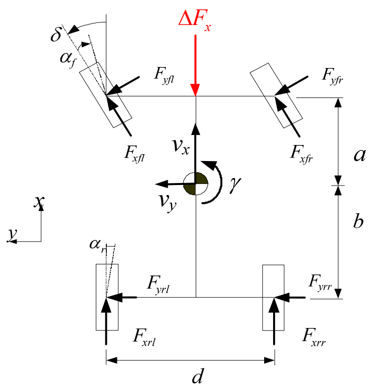


Fig. 7. Planar model including desired braking force

Dynamic equation of motion about longitudinal axis is represented as

$$m\dot{v}_x = F_{xr} + F_{xf}\cos\delta - F_{yf}\sin\delta + mv_y\gamma - \Delta F_x \tag{14}$$

In order to obtain the desired braking force, sliding surface and sliding condition are defined as follows:

$$s = v_x - v_{x,des}\ , \quad \frac{1}{2}\frac{d}{dt}s^2 = s\dot{s} \le -\eta s^2 \tag{15}$$

The desired braking force is obtained as follows:

$$\Delta F_x = F_{xf} + F_{xr} - F_{yf}\delta + m\left[v_y\gamma - \dot{v}_{x,des} + K\left(v_x - v_{x,des}\right)\right] \tag{16}$$

The desired yaw moment for lateral stability can be obtained from the ESC control algorithm which is used commonly as follows (Cho *et al.*, 2007):

$$M_{z,ESC} = -I_z \left(\frac{2\left(-l_f \hat{C}_f + l_r \hat{C}_r\right)}{I_z} \beta - \frac{2\left(l_f^2 \hat{C}_f + l_r^2 \hat{C}_r\right)}{I_z v_x} \gamma + \frac{2 l_f \hat{C}_f}{I_z} \delta_f \right) - K_2 sat\left(\frac{\gamma - \gamma_d}{\Phi} \right) \quad (17)$$

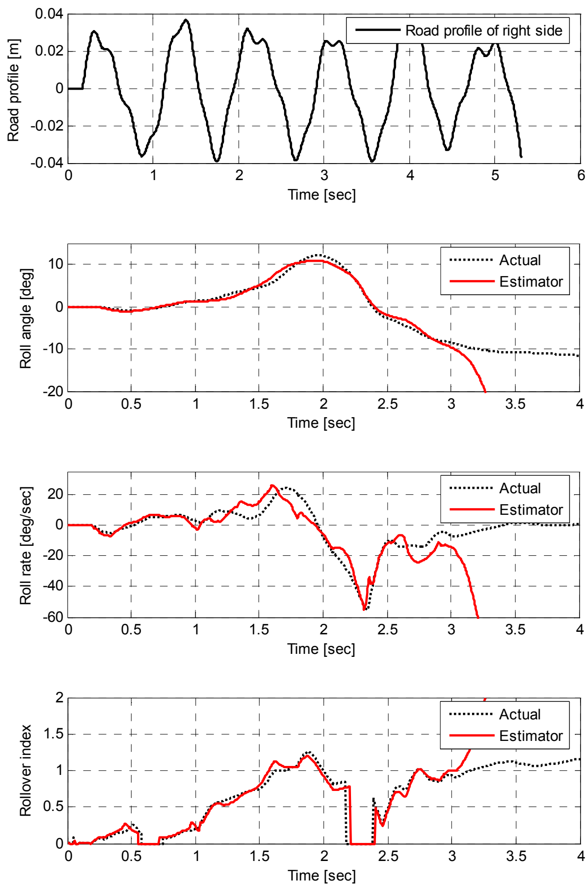
Using (16) and (17), braking forces of left and right side is obtained by vehicle dynamics as follows:

$$\Delta F_{x,l} = \frac{1}{2} \Delta F_x + \frac{M_z}{t}, \quad \Delta F_{x,r} = \frac{1}{2} \Delta F_x + \frac{M_z}{t} \quad (18)$$

4. Simulation Results

4.1 Roll State Estimator

In order to validate the performance of the proposed roll state estimator, NHTSA fishhook simulation is conducted with road disturbances. The road disturbance is shown in first plot of Fig. 8. In this simulation, rollover is occurred at about 3 seconds with two wheels-lift-off. As shown in fourth plot of Fig. 8, the RI increases over the unity at about 3 seconds. From the simulation results, it is shown that the proposed roll state estimator shows good performance before vehicle rollover although one wheel or two wheels are lifted-off as shown in fifth plot of Fig. 8.



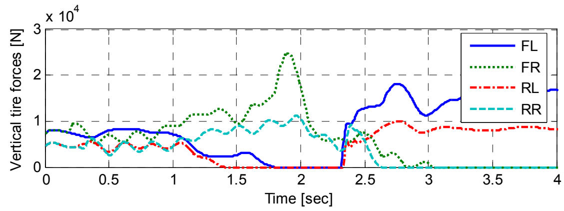
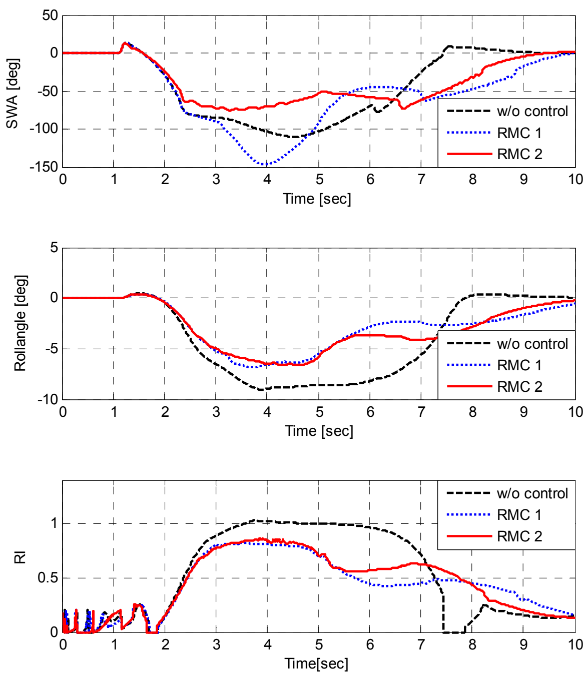


Fig. 8. Simulation results of the NHTSA fishhook @80kph

4.2 Rollover Mitigation Control

Closed-loop driver-vehicle-controller system simulation is conducted to investigate and compare the performance of the proposed RMC strategies. In this simulation, wheel steering angle is determined by a driver steering model developed to present human drivers in lane following situations (Kang *et al.*, 2006). A circular turning maneuver is simulated and the radius of curvature is 60 m. Rollover is occurred without RMC in this situation. However, when the RMC is activated, all of the RMC systems show good performance against rollover as shown in second and third plots of Fig. 9. In the Fig. 9, 'RMC 1' indicates the RI-based RMC and 'RMC 2' indicates the RI/Lateral stability-based RMC. Because the RI-based RMC system intends to control the vehicle opposite direction to driver's intention, wheel steering angle is larger than the RI/Lateral stability-based RMC system to follow the desired trajectory as shown in first plot of Fig. 9. Of course, the yaw rate error of the RI/Lateral stability-based RMC is maintained smaller than the RI-based RMC system as shown in last plot of Fig. 9.



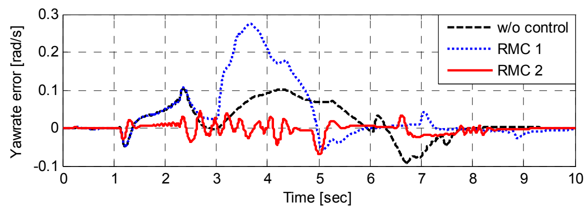


Fig. 9. Simulation results of circular turning @80kph

Fig. 10 shows tracking errors and Fig. 11 shows vehicle trajectories. The RI/Lateral stability-based RMC shows the best tracking performance. In the case of the RI-base RMC, tracking performance is worse rather than the no-control case.

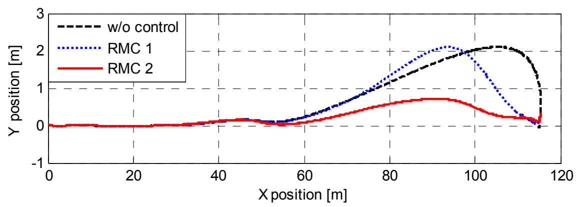


Fig. 10. Tracking errors

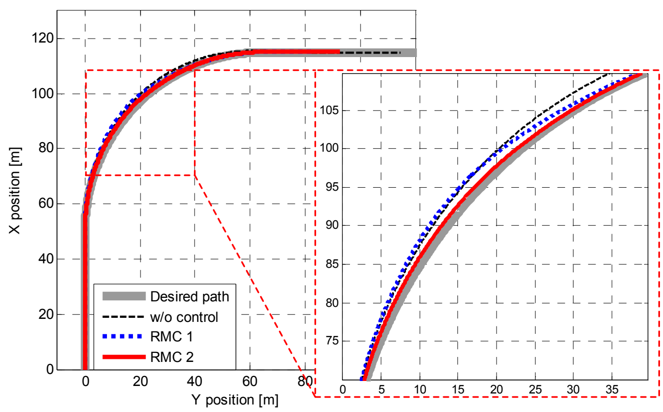


Fig. 11. Trajectories of vehicle

5. Conclusion

An UCC for vehicle rollover prevention is proposed in this paper. The RI is introduced to detect an impending rollover, and a model-based roll state estimator is also introduced. Two of the RMC systems are proposed. The one is the RI-based RMC and another is the RI/Lateral stability-based RMC. The individual chassis control modules such as ESC, AFS and CDC are integrated using an optimal method. An optimal distribution of longitudinal and lateral tire forces is achieved to minimize the deceleration. From the simulation results, it is verified that the proposed RMC systems show good performance against the rollover. However, From a view point of the lateral stability such as yaw rate error and tracking error, RI/Lateral stability-based RMC is better than the RI-based RMC. It implies that the maneuverability can be improved by the RI/Lateral stability-based RMC system.

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