Effects of camber angle control of front suspension on vehicle dynamic behaviors [†]

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Abstract

This paper proposes a new electronic camber suspension mechanism in which the suspension geometry rather than brake and driving torques is controlled to improve the cornering performance. The new camber angle mechanism is made available in a double wishbone type suspension. The bicycle model is employed to establish the control strategy for the camber angle mechanism. The referenced yaw rate is derived to make the control input. To carry out the control simulation of the full car, ADAMS/Car and MATLAB/Simulink are used. The result of fishhook simulation indicates that the proposed mechanism reduces the camber angle generation and decreases the camber thrust. This makes the vehicle move straight forward easily. The single lane change simulation indicates that the change of camber angle is small and it contributes to the short cornering and fast straight forward ability.

Keywords: Vehicle dynamics; Electronic active suspension; Camber angle; Handling performance

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1. Introduction

The engine, tire, and suspension are the important components that determine the vehicle performance. In recent years, the information about the vehicle running state is gathered by the electronic control unit (ECU) which enables the vehicle to overcome dangerous conditions by the proper control logic. Bong et al. [1] have studied the intelligent torque distribution control for the 4 wheel driving car. Song et al. [2] carried out the performance evaluation of electronic stability program (ESP) systems for enhancing the lateral stability during cornering. Zhange et al. [3] used the quantitative feedback theory to develop the active front steering system (AFS). The electrical stability control (ESC) gives the vehicle lateral stability in severe conditions through the independent control of each brake. ESC gives good performance in severe conditions and low friction road conditions, but the velocity decreases and the comfort is reduced during operation. The continuous damping control (CDC) uses the semi-damping system to improve the ride ability but it has little effects on yaw rate control on the low road condition [4]. The requirement on vehicle behavior relating to tire performance has become complicated. Tire makers should develop exclusive tire for each new vehicle because of the suspensions characteristics such as camber angle and toe angle. These suspension characteristics have

constraints on the tire alignments which lead to the tire's loss of potential. While a vehicle is turning, the lateral forces are used. A lateral acceleration increases along with a roll angle, the camber angle leans toward an external direction, which decreases lateral forces and eventually limits tire performance as well. If the camber angle is directly controlled, the vehicle can make the most of the tires potential. This is the basic concept of the variable camber suspension which maximizes vehicle handling performance. If the active camber suspension mechanism is used, the handling performance can be enhanced without the loss of velocity unlike ESC.

Isao et al. [5] proposed a variable camber suspension model to improve the handling performance of vehicle. They suggested the double knuckle type suspension that had no control algorithm for the variable camber suspension.

In this study, we developed a new active camber suspension mechanism called "bicycle model" which controls the camber angle of the front suspension. To verify the active camber suspension mechanism, the fishhook and single lane change simulations are carried out. ADAMS/Car program is employed to perform the full vehicle simulation. The bicycle model is used to make the control algorithm, and ADAMS/ control simulation is carried out under the MATLAB/Simulink environment.

This paper is organized as follows. Section 2 proposes and describes a new electronic camber suspension mechanism. The control algorithm for camber geometry is shown in section 3. The full car simulation is explained in section 4. Finally, conclusions are presented in section 5.
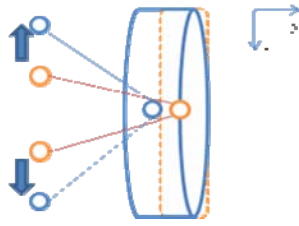

Fig. 1. Concept of generating camber angle.
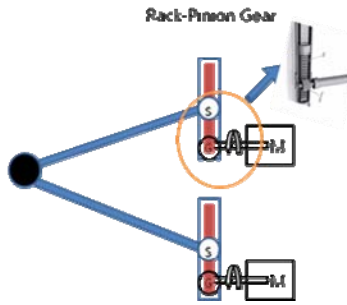
Fig. 2. Active camber mechanism of rack-pinion type.
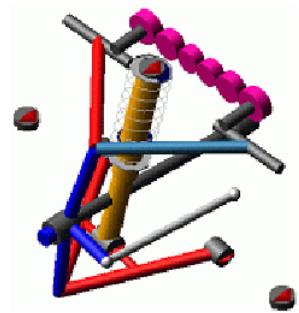
Fig. 3. Active camber suspension with gear sets.

Table 1. Calculation of degrees of freedom.

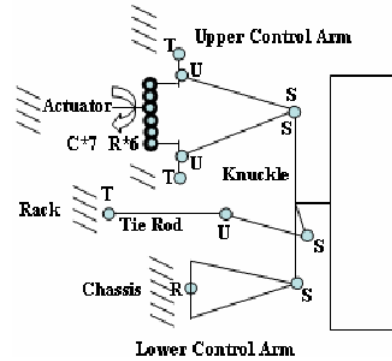| coordinates	15 bodies x 6 coords./body = 90 coords.
constraints	4 spherical x 3 const./joint = -12 coords.
	3 translationalx5 const./joint = 15 coords.
	7 revolute x 5 const./joint = -35 coords.
	3 universal x 4 const./joint = -12 coords.
	7 coupler x 1 const./joint = -7 coords.
	1 ground x -6 const./joint = -6 coords.
D.O.F	3 D.O.F

Fig. 4. Kinematic of diagram.

Fig. 5. Concept of camber angle control.

2. Active camber suspension

When the points connecting the chassis and upper control arm move along with the longitudinal direction which is the vehicle moving direction, the camber angle is generated as shown in Fig. 1. In the proposed mechanism, many gears are used to make the translational displacement along with longitudinal direction. As seen in the Fig. 3, when the rotary actuator is rotated, two links at upper arm move in longitudinal direction. Then the upper arm inclines to the chassis and the camber angle is generated. The suspension type is a double wishbone type, and the upper control arm is modified to make the active comber angle by a rotary motor. The kinematic diagram of front suspension with variable camber suspension is illustrated in Fig. 4. The degrees of freedom of front suspension are listed in Table 1. The proposed electronic camber suspension mechanism has 3 D.O.F. such as bump/rebound, steering angle, and camber angle. The concept of variable camber angle control is illustrated in Fig. 5. The steering wheel angle and vehicle velocity are transferred to the ECU to calculate the proper camber angle. Then, the actuator is operated to make the variable camber angle.
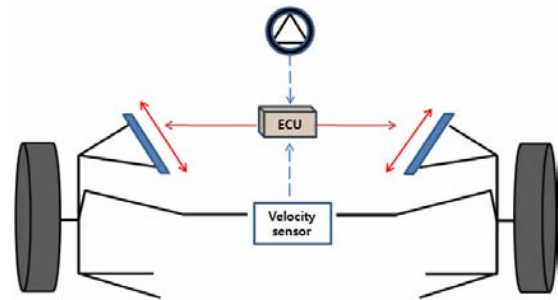
3. Control algorithm for active camber angle

3.1 Bicycle model

The 2 degree freedom linear model represents the lateral motion of the vehicle simply and accurately. Fig. 6 shows the 2 degree freedom linear bicycle model.

If the longitudinal tire force is ignored and the longitudinal velocity is constant, the system equations are as follows [6]:

$$mv_x(\dot{\beta}+\gamma) = 2F_{yf} + 2F_{yr} \tag{1}$$

$$I_z\dot{\gamma} = 2l_fF_{yf} + 2l_rF_{yr} . \tag{2}$$

To linearize the vehicle system, if the slip angle is small, the lateral tire forces can be represented as Eq. (3).

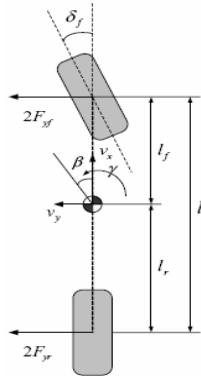$$F_{yf} = C_f\alpha_f, \qquad F_{yr} = C_r\alpha_r \tag{3}$$

Fig. 6. Bicycle model.

The slip angle of front and rear suspension can be expressed as follows:

$$\alpha_f = \delta_f - \left(\beta + \frac{l_f \cdot \gamma}{v_x} \right) \tag{4}$$

$$\alpha_r = -\beta + \frac{l_r \cdot \gamma}{v_x}. \tag{5}$$

These equations can be finally represented as follows:

$$\dot{X} = A \cdot X + B \cdot \delta_f \tag{6}$$

$$A = \begin{bmatrix} -\dfrac{2(C_f + C_r)}{m \cdot v_x} & -1 - \dfrac{2(l_f \cdot C_f - l_r \cdot C_r)}{m \cdot v_x^{\,2}} \\[2ex] -\dfrac{2(l_f \cdot C_f - l_r \cdot C_r)}{I_z} & -\dfrac{2(l_f^{\,2} \cdot C_f - l_r^{\,2} \cdot C_r)}{I_z \cdot v_x} \end{bmatrix} \tag{7}$$

$$B = \begin{bmatrix} \dfrac{2 \cdot C_f}{m \cdot v_x} \\[2ex] \dfrac{2 \cdot l_f \cdot C_f}{I_z} \end{bmatrix}, \quad X = \begin{bmatrix} \beta & \gamma \end{bmatrix}^T. \tag{8}$$

3.2 Estimation of referenced yaw rate and control strategy

In the bicycle model, when the vehicle runs at the steady state cornering, the referenced yaw rate can be calculated as Eq. (10) by applying the steady state condition as in Eq. (9) to the state equation.

$$\dot{\beta} = 0, \qquad \dot{\gamma} = 0 \tag{9}$$

$$\gamma_{ref} = \frac{1}{\left(1 - \dfrac{m}{2l^2} \dfrac{l_f \cdot C_f - l_r C_r}{C_f \cdot C_r} v_x^{\,2} \right)} \frac{v_x}{l} \delta_f \tag{10}$$

The current velocity and steering angle of the referenced model are used to calculate the referenced yaw rate [7]. Since the estimation equation of referenced yaw rate is calculated by ignoring the road condition, the referenced yaw rate is calcu-
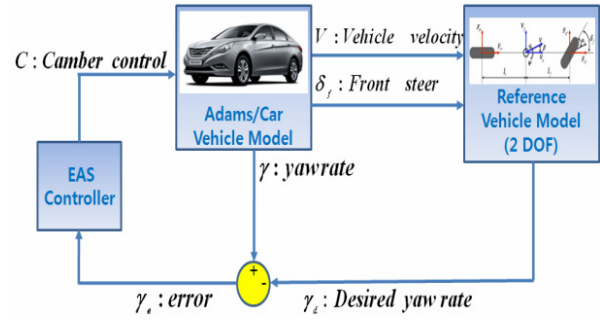

Fig. 7. Camber angle control strategy.

Fig. 8. Full car model.

lated at the road of high friction state. However, in the low friction state, when the referenced yaw rate exceeds the limited lateral acceleration, the yaw rate in the maximum lateral acceleration is set to the maximum referenced yaw rate.

Fig. 7 shows the camber control strategy. The referenced yaw rate from the signal of yaw rate and steer angle sensors are calculated. The referenced yaw rate is compared with the vehicle yaw rate. The control input proportional to errors is applied to the active camber mechanism.

4. Full car simulation

4.1 Fishhook simulation

Fig. 8 shows the full car model. The front suspension is the double wishbone type with the proposed active camber mechanism and the rear suspension is the double wishbone type. ADAMS/User template is employed to make variable camber suspension in the ADAMS/Car model. The total degrees of freedom is 25.

In this study, the fishhook and single lane change test are carried out. The fishhook test is used to evaluate the rollover stability. The vehicle starts at entrance speed between 35 ~ 50 mph and is given by rapid steering input with -270° then is done by rapid steering input with -540° in the reverse direction as shown in Fig. 9. Fig. 10 shows the steer input angle used in this study. The vehicle velocity is 70 km/h and steer angular velocities are 600 deg/s and 750 deg/s, respectively.

Fig. 11 presents the trajectory and the upper part shows the trajectory of uncontrolled car. When the vehicle does not have controlled camber angle, the vehicle is drifted because it loses its stability. The controlled car has camber angles smaller than non-controlled car model as shown in Figs. 13 and 14.
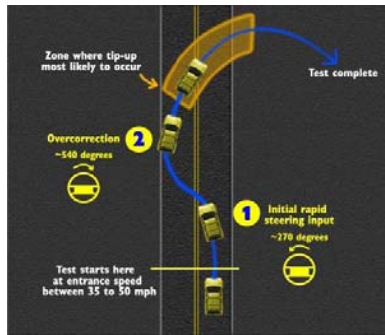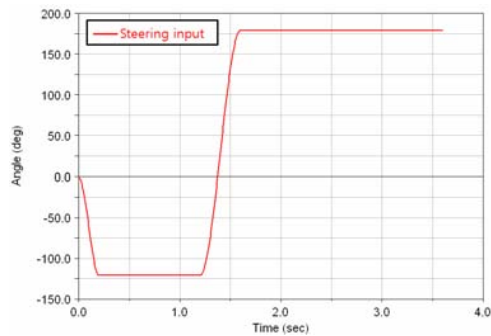
Fig. 9. Fishhook test regulation (NHTSA).

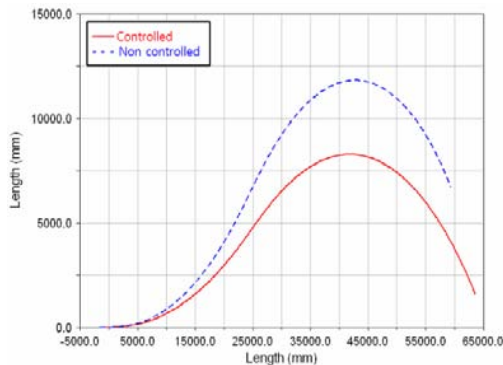Fig. 10. Steering input of fishhook.

Fig. 11. Trajectory of car.

Fig. 12. Path of car.

In this study, the initial front camber angle is -0.5 deg. When the steering input is applied at the front wheel, the load transfer occurs, and the cornering outer wheel has negative camber and the inner wheel has positive camber angle. Then, the cornering force increases due to camber thrust. In the case of a controlled car, the camber angle generation is reduced and the camber thrust decreases. This makes the vehicle move straight forward easily.
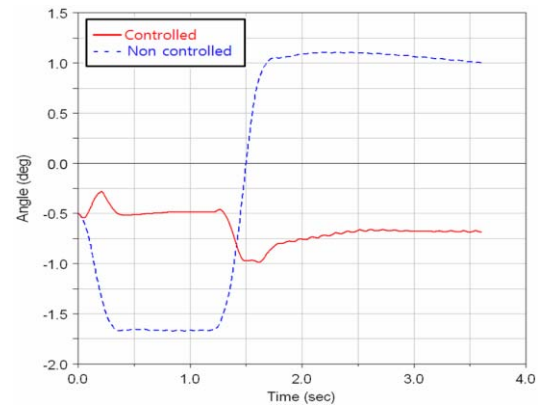

Fig. 13. Camber angle of left wheel at fishhook.
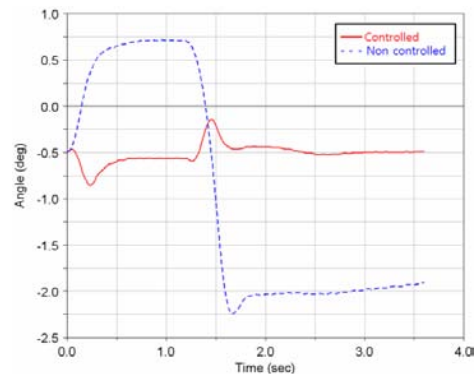

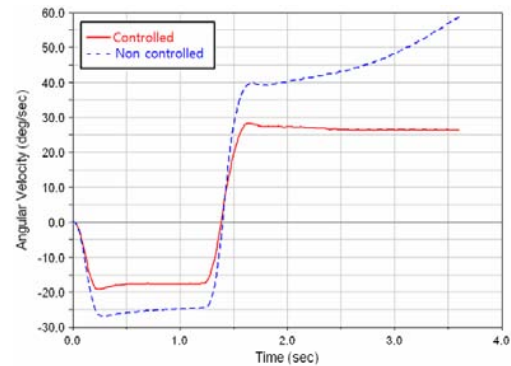Fig. 14. Camber angle of right wheel at fishhook.

Fig. 15. Yaw rate of car at fishhook.

The yaw rate of controlled car shows stable tendency but non-controlled car shows the drift out motion as shown in Fig. 15. Therefore, side slip angle of non-controlled car rapidly increases as shown in Fig. 16, The roll angle and the lateral acceleration of controlled car is smaller than non-controlled car. Table 1 lists the maximum and minimum value of yaw rate, roll angle and lateral acceleration of controlled and non-controlled car. Note that the difference of yaw rate between controlled and non-controlled is the biggest.

According to above results, in case of non controlled car,

Table 2. Maximum and minimum of yaw rate, roll angle, lateral acceleration at fishhook simulation.

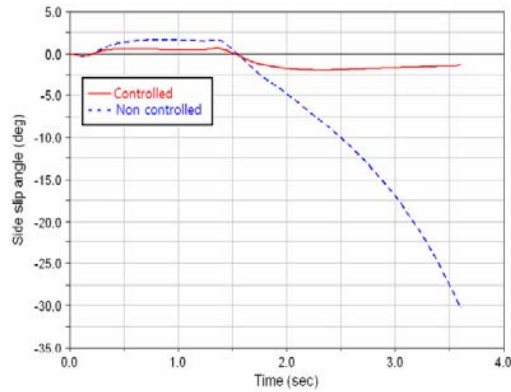| Conponents	Non controlled		Controlled	
	Max.	Min.	Max.	Min.
Yaw (deg/s)	58.72	-26.92	28.22	-19.26
Roll (deg)	3.76	-4.56	2.60	-3.91
Lat. Acc. (g)	0.85	-1.00	0.61	-0.89

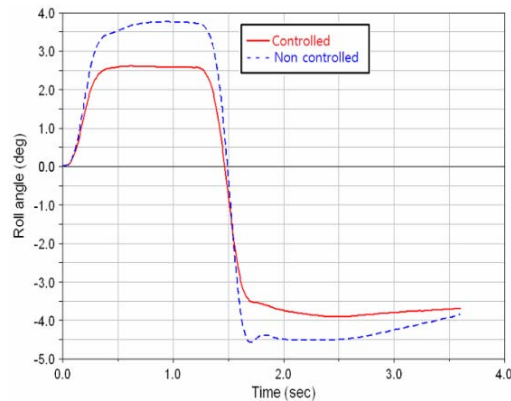Fig. 16. Side slip angle of car at fishhook.

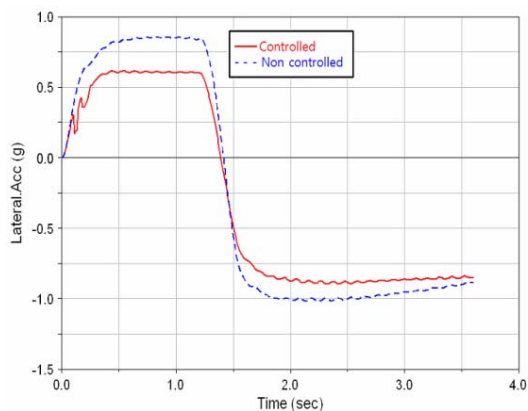Fig. 17. Roll angle of car at fishhook.

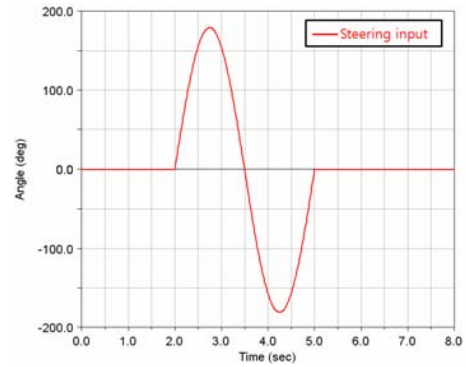Fig. 18. Lateral acceleration of car at fishhook.

Fig. 19. Steering input at single lane change.
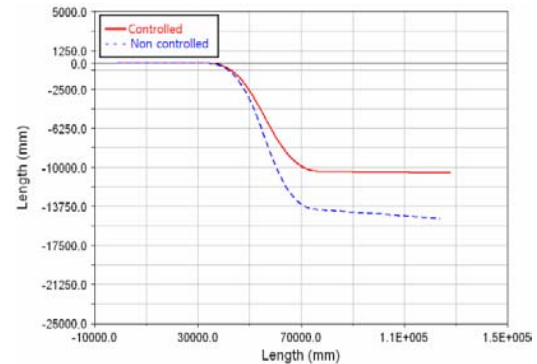

Fig. 20. Trajectory of single lane change.

Fig. 21. Path of car at single lane change simulation.

when the fishhook input is applied to steering system, the yaw rate and side slip angle increase and the instability increase.

4.2 Single lane change simulation

The single lane change test is used to evaluate the performance of obstacle avoiding. In this study, the velocity is 60 km/h and 180 degree of steer angle is applied at 2 sec. Fig. 19 represents the steer angle input. Figs. 20 and 21 show the trajectory and path of the car, respectively. Figs. 22 and 23 shows the comparison of camber angle between the controlled and non controlled car of left wheel and right wheel at the single lane change simulation, respectively.

According to above results, in the case of the controlled car, the change of camber angle is small and it contributes to the short cornering and fast straight forward ability. Figs. 24-27 show the yaw rate, side slip angler, roll angle and lateral acceleration, respectively. Table 3 lists the maximum and minimum for yaw rate, roll angle, and lateral acceleration.

The yaw rate of controlled car shows smaller amplitude as shown in Fig. 24. Therefore, side slip angle of controlled car shows small value as shown in Fig. 25, Note that the roll angle

Table 3. Maximum and minimum of yaw rate, roll angle, lateral acceleration at single lane change simulation.

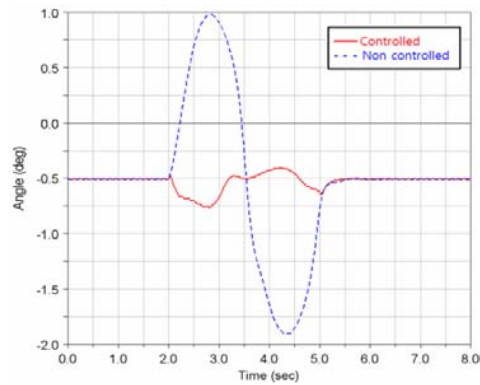| Conponents	Non controlled		Controlled	
	Max.	Min.	Max.	Min.
Yaw (deg/s)	35.68	-35.19	26.24	-26.11
Roll (deg)	4.02	-4.08	3.13	-3.12
Lat. acc. (g)	0.90	-0.92	0.72	-0.73

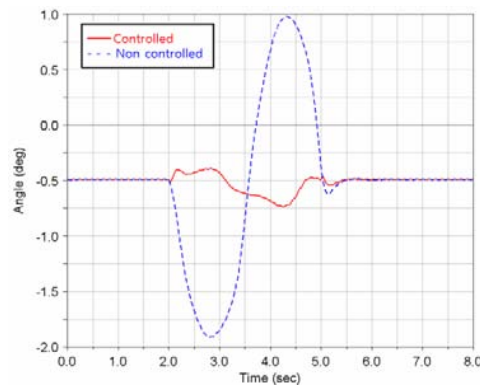Fig. 22. Camber angle of left wheel at single lane change.

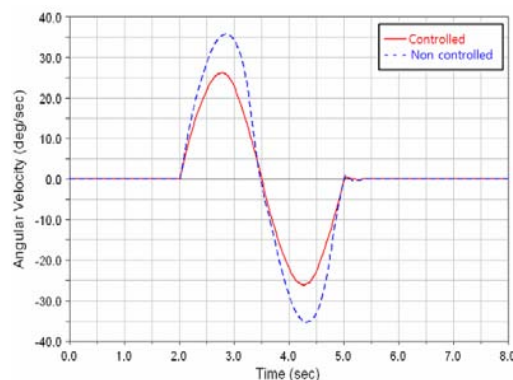Fig. 23. Camber angle of right wheel at single lane change.

Fig. 24. Yaw rate of car at single lane change.

and the lateral acceleration of controlled car has smaller than non-controlled car. Table 2 lists the maximum and minimum value of yaw rate, roll angle and lateral acceleration of con-
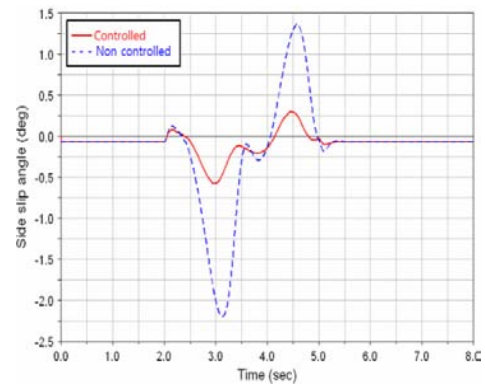

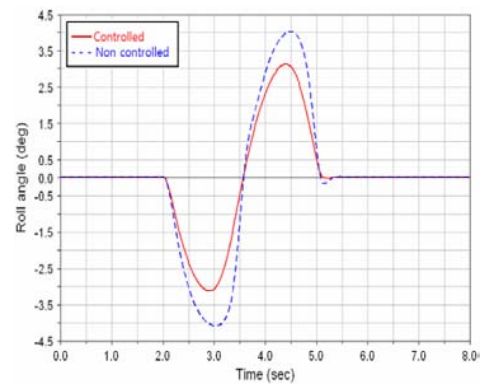Fig. 25. Side slip angle of car at single lane change.

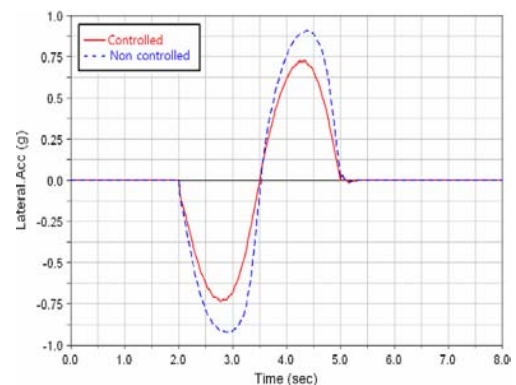Fig. 26. Roll angle of car at single lane change.

Fig. 27. Lateral acceleration of car at single lane change.

trolled and non-controlled car. The peak values of controlled car are small and its differences are about 20%.

5. Conclusion

This study developed a new mechanism called bicycle model that controls the camber angle of the front suspension. The suspension geometry rather than brake or driving torques is controlled to improve the cornering performance. The results of fishhook simulation and single lane change simulation verified that when a car model that controls the camber angle is used, the camber angle generation is reduced and the cam-

ber thrust decreases. In addition, the small change in the camber angle contributes to the short cornering and fast straight forward ability. This makes the vehicle move straight forward easily. The proposed bicycle model enabled the controlled car to show better results than passive car by improving the yaw rate, lateral acceleration and dynamic behavior. After several additional tests, it may be possible to use the proposed mechanism in the real vehicle.

Acknowledgement

Nomenclature——————————————————

β : Vehicle body slip agle (rad)

γ : Yaw rate (rad/s)

F_y : Lateral forces of wheels (N)

F_x : Longitudinal forces of wheels (N)

δ_f : Steering angle at front wheel (rad)

l_f : Distance between C.O.G and front axle (m)

l_r : Distance between C.O.G and rear axle (m)

l : Wheel base ($l = l_f + l_r$)(m)

C_f : Cornering stiffness of front wheel (N/m)

C_r : Cornering stiffness of rear wheel (N/m)

v_x : Longitudinal velocity (m/s)

v_y : Lateral velocity (m/s)

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