Improvement of Vehicle Directional Stability in Cornering Based on Yaw Moment Control

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In this research any abnormal motion of a vehicle is detected by utilizing the difference between the reference and actual yaw velocities as sell as the information on vehicle slip angle and slip angular velocity. This information is then used as a criterion for execution of the yaw moment control. A yaw moment control algorithm based on the brake control is proposed for improving the directional stability of the vehicle. The controller executes brake controls to provide each wheel with adequate brake pressures, which generate the needed yaw moment. It is shown that the proposed yaw moment control logic can provide excellent cornering capabilities even on low friction roads. This active control scheme can prevent a vehicle from behaving abnormally, and can assist normal drivers in coping with dangerous situations as well as experienced drivers.

Key Words : Yaw Moment Control, Vehicle Dynamics Control, Yaw Velocity, Vehicle Slip Angle, Brake Control

1. Introduction

Vehicle safety systems have been rapidly developing thanks to an increased awareness of the drivers' need for stability enhancement as well as advances in electronics and control technologies. Active safety systems such as ABS (anti-lock braking system), which improves stability of a vehicle by avoiding lock-up of the wheels during braking, and TCS (traction control system), which enhances acceleration performance and directional stability by preventing excessive spinning of the wheels on slippery roads, are currently commercially available. Recently more advanced and complex active safety systems that combine the functions of ABS and TCS, called VDC (vehicle dynamics control), have been developed. The VDC keeps a vehicle from behaving abnormally against the driver's intention and guarantees directional stability by controlling the dynamic characteristics of the vehicle. It was first introduced by Bosch (G. Heeb ? et al., 1988), but many automotive companies have now developed similar systems, for instance, VSC (vehicle stability control) by Toyota, AYC (active yaw control) and ASC (active stability control) by Mitubishi, DYC (direct yaw control) and VSA (vehicle stability assist) by Honda, and so on. However, no information on control logic has been released to date.

Van Zanten et al. (1998) determined the vehicle motion desired by the driver based on the steering angle, the accelerator pedal position, and the brake pedal position, and then conducted engine control and brake pressure modulation so that the actual vehicle motion follows the desired motion. Koibuchi et al. (1996) and Yasui et al. (1996) analyzed the vehicle stability based on the slip angle and slip angular velocity and obtained the relationship between the brake force and/or the

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slip ratio of each wheel and the change in the yaw moment. Then, they suggested the control algorithm that aims to improve the cornering stability through brake control. Song and Cha (1999) suggested the algorithm in which the spin condition is determined by using the difference between the target and actual yaw velocities and the resulting difference is reduced by conducting engine control and brake pressure modulation based on a control map. Ikusima and Sawase (1995) presented a torque distribution system aimed at improving the vehicles' cornering properties by yaw moment control based on the torque difference between the right and left wheels. Furukawa and Masato (1997) observed the effectiveness and limits of yaw moment control in 4-wheel steering vehicles, and pointed out that yaw moment control was more effective in the vehicle with larger slip angles and/or higher lateral accelerations.

In this research the abnormal motion of a vehicle is detected by utilizing the difference between the reference and actual yaw velocities and the information on vehicle slip angle and slip angular velocity. This information is then used as a criterion for execution of yaw moment control. The controller executes brake control to provide each wheel with adequate brake pressures, which generate the needed yaw moment computed through the PD control based on the errors in yaw velocity and vehicle slip angle. This active control scheme prevents a vehicle from behaving abnormally, and assists novice drivers in coping with dangerous situations as skillfully as experienced drivers.

Section 2 deals with a vehicle dynamics model, a powertrain model, and a brake model and introduces derivation of the target yaw velocity of a vehicle from the driver's inputs. Section 3 is concerned with the control decision part for detecting the abnormal motion of a vehicle and the yaw moment control part for generating the needed yaw moment by applying suitable brake control. Section 4 presents some simulation results, and evaluates the effectiveness of the proposed algorithm.

2. System Modeling

It is of great importance to develop mathematical models for the system as well as the components, since this research is mainly based on simulations of the vehicle behavior.

2.1 Vehicle dynamics model

The vehicle dynamics model consists of 7-DOF's; 3-DOF's for longitudinal, lateral, and yawing motions, and 4-DOF's for rotation of each wheel as shown Fig. 1. The equations of a 7-DOF vehicle model derived from the vehicle dynamics are as follows (Gillespie, 1992):

Longitudinal motion:

$$m_v(\dot{v}_x - v_y\Omega_z) = \sum F_x \tag{1}$$

Lateral motion:

$$m_v(\dot{v}_y + v_x\Omega_z) = \sum F_y \tag{2}$$

Yawing motion:

$$I_z\dot{\Omega}_z = \sum M_z \tag{3}$$

Wheel dynamics:

$$I_{wi}\dot{\omega}_i = T_{wi} - F_{xi}r_w - T_{bi} (i=1, 2, 3, 4) \tag{4}$$

where m_v represents the vehicle mass, v_x, v_y, Ω_z the longitudinal, lateral and yaw velocity, I_z the mass moment of inertia of the vehicle, and $\sum F_x$, $\sum F_y$, $\sum M_z$ the sum of the longitudinal, lateral force and yaw moment at the mass center,

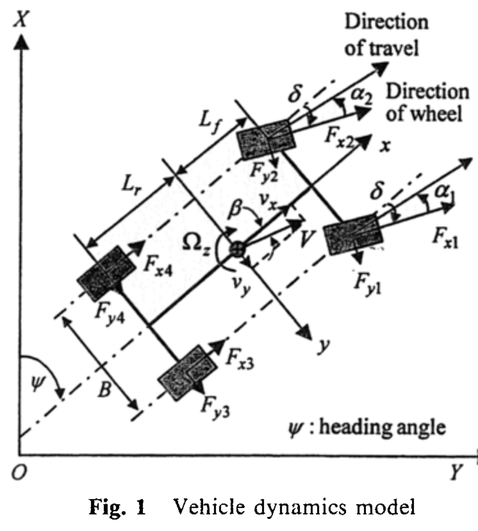

Fig. 1 Vehicle dynamics model

respectively. And I_w, T_w, ω_w, r_w, T_b represent the mass moment of inertia of the wheel, the driving torque applied to the driving axis from an engine, the angular velocity of the wheel, the wheel radius, and the braking torque, respectively, and subscript i denotes each wheel shown in Fig. 1.

The wheel slip angle α_i represents the angle between the direction of the wheel heading and wheel travel and is defined as

$$\alpha_i=\begin{cases}\tan^{-1}(v_{wyi}/v_{wxi})-\delta & \text{for } i=1,2 \\ \tan^{-1}(v_{wyi}/v_{wxi}) & \text{for } i=3,4\end{cases} \tag{5}$$

where v_{wxi} and v_{wyi} represent the longitudinal and lateral velocity of each wheel and δ the steering angle, respectively. The vehicle (or body) slip angle β denotes the angle between the longitudinal direction of a vehicle and the actual direction of travel and is defined as

$$\beta=v_y/v_x \tag{6}$$

Tire models are of great importance because the forces between the tire and the road play an important role in the vehicle behavior including turning, acceleration and braking operations. In this research the Dugoff tire model is employed, with the nonlinear tire behavior as described in Dugoff et al. (1970).

2.2 Powertrain and brake models

A gasoline engine can be modeled as a first-order system with transport delay, where the throttle angle and the engine torque are chosen as input and output, respectively (Song et al., 1999). The overall system model for the engine/automatic transmission in which the throttle angle θ and driving torque T_w are the input and output, respectively, is given by

$$\frac{T_w(s)}{\theta(s)}=\rho\cdot\frac{T_e(s)}{\theta(s)}=\rho\cdot\frac{K_e}{1+\tau s}\cdot e^{-\tau_d s} \tag{7}$$

where T_e denotes the engine torque, ρ the overall speed ratio taking into account the torque converter, transmission and final drive, and K_e, τ, τ_d represents the steady-state gain, time constant and transport delay, respectively. For convenience, the effect of a torque converter is taken into account by the speed ratio of a pump to a turbine.

The brake model used in this research was established from the experimental study. The frequency response tests on a modulator and a brake line were conducted using an actual brake system, and the brake model was obtained by observing the characteristics of brake pressure generation for various duty cycles at the selected operating frequencies of a solenoid valve (Kim and Chang, 1996). Consequently, the brake system model used in simulations had the mode and the increase/decrease time of brake pressure as inputs and the brake pressure as an output.

2.3 Derivation of target yaw velocity and vehicle slip angle

In order to make the vehicle follow the trajectory set by the driver's inputs, the reference trajectory reflecting the intention of the driver should be determined first. Based on this reference, any abnormal motion of the vehicle is detected, and the vehicle is made to follow the reference trajectory by an adequate control algorithm. In this chapter this control reference is derived from the driver's inputs and the dynamic equations of the vehicle in turning.

Consider the behavior of a turning vehicle. The relationship between the steering angle δ, the turning radius R, the lateral acceleration a_y, and the vehicle forward speed v_x is given by Gillespie (1992) and Wong (1993)

$$\delta=\frac{L}{R}+K_{us}\frac{a_y}{g}=\frac{L}{R}+K_{us}\frac{v_x^2}{gR} \tag{8}$$

where L, K_{us}, and g represent the wheelbase, the understeer coefficient, and the gravitational acceleration, respectively. In the steady-state cornering of a vehicle, the relation $v_x=R\Omega_z$ holds. The yaw velocity is, therefore, expressed by

$$\Omega_z=\frac{v_x\delta}{L+K_{us}(v_x^2/g)} \tag{9}$$

The yaw velocity is computed using the steering angle and longitudinal velocity, which are the inputs reflecting the driver's intentions. The yaw velocity computed in Eq. (9) is used as the reference yaw velocity for control decision in the next chapter.

From the relation $a_y=v_x^2/R$ and Eq. (8), the lateral acceleration is expressed by

$$a_y = \frac{v_x^2 \delta}{L + K_{us}(v_x^2/g)} \tag{10}$$

Then, using the lateral velocity v_y obtained by integrating the lateral acceleration a_y, the reference vehicle slip angle β_r can be obtained by

$$\beta_r = v_y / v_x, \quad \text{where } v_y = \int a_y dt \tag{11}$$

This vehicle slip angle is used as a reference for yaw moment control.

3. Yaw Moment Control Algorithm

The driver having first-hand experiences with many emergency situations can cope with abnormal behavior of a vehicle, but normal drivers cannot manage these kinds of situations properly. More detailed explanation of these situations can be found in the research of Shibahata et al. (1993). Furthermore, they may operate the vehicle in such a way that the situation is aggravated by excessive steering or braking. For this reason, the yaw moment control system should assist normal drivers in coping with dangerous situations. Two aspects of directional stability, a yaw velocity error and a vehicle slip angle, are examined, and a yaw moment control algorithm based on the brake control is proposed in this chapter.

3.1 Decision of directional stability

In order to execute the yaw moment control proposed in this research, a decision of whether or not such control is necessary should be made by evaluating the directional stability of the vehicle. Two conditions are considered for this purpose: one is associated with the yaw velocity and the other the vehicle slip angle.

Let us consider the yaw velocity related condition first. The yaw velocity was selected as a parameter describing the vehicle behavior for deciding whether or not the vehicle is faithfully following the course set by the driver's inputs such as the steering, acceleration and braking operations. An analytical yaw velocity can be computed from the vehicle conditions as in Eq. (9), and an actual yaw velocity can be measured using a yaw rate sensor. The deviation from the designated course is detected by both the error between the reference yaw velocity Ω_{zr} and the actual one Ω_z, and its derivative. As a result, a yaw velocity related criterion for control decision is defined as

$$| C_1 \cdot e_\Omega + C_2 \cdot \dot{e}_\Omega | < \mathrm{SF1}, \text{ where } e_\Omega = \Omega_{zr} - \Omega_z \tag{12}$$

where C_1 and C_2 denote the weights for the yaw velocity error and its derivative, and SF1 is the factor for control decision. It is noted that use of the error derivative makes it possible to predict a tendency of a vehicle to deviate from the reference yaw velocity. A high level of SF1 allows the vehicle to get off course without the control action. Thus, the greater the value of SF1, the larger the amount of brake force required to correct the error; the vehicle may lose directional stability due to excessive brake force, especially on slippery roads. Hence the value of SF1 should be adjusted by taking into account the road conditions (e. g., friction coefficient), the vehicle velocity, and so on.

Now, let's consider the condition associated with the vehicle slip angle. Various simulations on different operating conditions were carried out to investigate loss of directional stability, and some results will be shown here. Suppose that the vehicle at the speed of 100km/hr is running on wet roads with the friction coefficient of 0.6 and is subject to sinusoidal steering input with the frequency of 0.5Hz and the steering angles of 4, 6, 8° as shown in Fig. 2. Figure 3 illustrates the simulation results of the vehicle slip angle and vehicle slip angular velocity.

It is observed from Fig. 3 that the vehicle slip angle grows with an increase of the steering angle for the identical vehicle speeds. It is also shown that the vehicle slip angle with the steering angle of 8° gradually increases, thus going into the unstable area in the end. This is because the vehicle turning on low friction roads tends to lose its stability due to excessive vehicle slip angle resulting from excessive slip at the wheels. Therefore, the loss of directional stability can be predicted by observing the vehicle slip angle β and the vehicle slip angular velocity $\dot{\beta}$, and should be prevented through adequate control. A

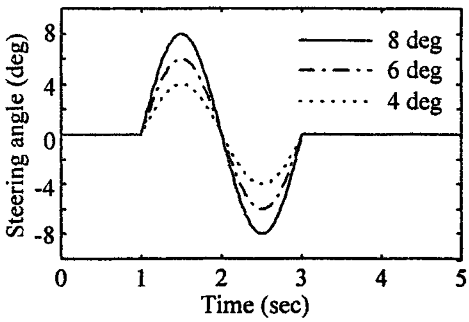

Fig. 2 Different steering angle inputs

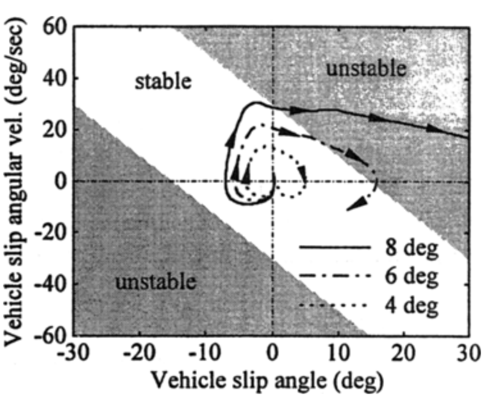

Fig. 3 Vehicle slip angular velocity vs vehicle slip angle

criterion for directional stability can be given in terms of the vehicle slip angle and the slip angular velocity of the vehicle with the corresponding weights, D_1 and D_2, which is similar to condition (12), as follows:

$$| D_1 \cdot \beta + D_2 \cdot \dot{\beta} | < \mu \cdot \mathrm{SF2} \qquad (13)$$

The value of can be determined by analyzing the vehicle stability through a series of simulations with different vehicle parameters as shown in Fig. 3. For example, from the various simulation results conducted for the normal roads (such as dry asphalt roads), the boundary between the stable and unstable regions can be determined as in Fig. 3. Then, SF2 can be computed from the boundary values. SF2 is then multiplied by the estimated friction coefficient μ to provide for a flexible control and to prevent the loss of vehicle stability.

In general, maneuverability is lost at the vehicle slip angle of approximately 10° on dry asphalt and of 4° on packed snow. Therefore, the value of SF2 is determined first for the normal roads, and then the friction coefficient μ of the road is multiplied to this SF2. That is, inclusion of μ enables a smaller level to be assigned for low friction roads to prevent the loss of vehicle stability. Condition (13), together with condition (12), is used to determine when the yaw moment control is executed.

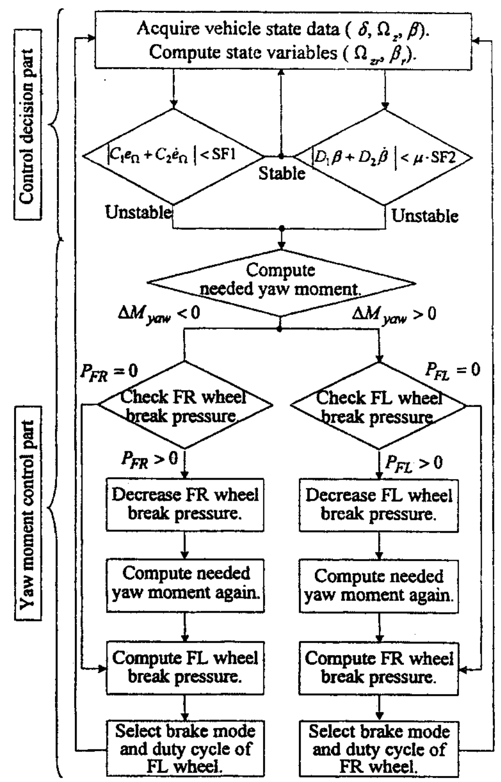

Fig. 4 Flowchart for yaw moment control scheme

3.2 Yaw moment control algorithm based on brake control

Figure 4 represents the flowchart for the yaw moment control through a brake control proposed in this research. If the conditions (12) and (13) are not satisfied, the yaw moment control is activated. First, the yaw moment needed for stabilizing the vehicle is computed in the PD control scheme as follows:

$$\Delta M_{yaw} = K_{P1} e_\Omega + K_{D1} \dot{e}_\Omega, \text{ where}$$
$$e_\Omega = \Omega_{zr} - \Omega_z \qquad (14)$$

$$\Delta M_{yaw}=K_{P2}e_\beta+K_{D2}\dot{e}_\beta,\ \text{where}$$
$$e_\beta=\beta_r-\beta \tag{15}$$

where K_{P1}, K_{D1}, K_{P2}, and K_{D2} are the proportional and derivative gains of the yaw velocity error and vehicle slip angle error, respectively. When the yaw moment needed is positive (which means a clockwise direction in Fig. 1), the brake pressure of the front right wheel is forced to increase. An increment of braking force at each wheel for generating the needed yaw moment is given by

$$\Delta F_{b1}=\Delta M_{yaw}\Big/\left(\frac{B}{2}\cos\delta-L_f\sin\delta\right)$$
$$\text{for } \Delta M_{yaw}>0 \tag{16}$$
$$\Delta F_{b2}=\Delta M_{yaw}\Big/\left(-\frac{B}{2}\cos\delta-L_f\sin\delta\right)$$
$$\text{for} \Delta M_{yaw}<0 \tag{17}$$

where ΔF_{b1} and ΔF_{b2} represent changes in braking forces at the front right and front left wheels, respectively. Note that the front-wheel driving is assumed throughout this paper. The computed increment in the braking force is converted to a change in the brake pressure by

$$\Delta P_{bi}=\Delta F_{bi}\cdot\frac{r_w}{A_b r_b}(i=1,\ 2) \tag{18}$$

where A_b and r_b denote the effective area and radius of a brake pad, respectively. If the brake control is executed based only on Eqs. (16) and (17), the brake pressure can only increase to create the required yaw moment, thus leading to a possible saturation of the brake pressure. As far as the yaw moment is concerned, however, an increase in the brake pressure of the front right wheel is equivalent to the same amount of decrease in the brake pressure of the front left wheel. Before increasing the brake pressure of one wheel, therefore, that of the opposite wheel is checked. If this wheel has a residual brake force, the compensating yaw moment calculated from Eqs. (16) or (17) is achieved by decreasing its brake pressure. After the residual brake pressure has been reduced to zero, for the control logic recalculates the needed the yaw moment except the already compensated amount of the yaw moment, and then increases the brake pressure of the original wheel accordingly. The yaw moment control part sends the braking mode and duty cycles to the brake system to generate adequate brake pressures computed above.

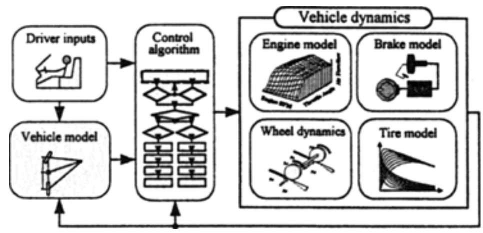

Fig. 5 Block diagram for simulations

4. Simulations and Results

Figure 5 shows the block diagram for the simulations performed in the present work. The vehicle model calculates the reference yaw velocity associated with the vehicle behavior to set the driver's inputs. A 2 DOF bicycle model is used. The reference yaw velocity is sent to the control algorithm part, which determines the directional stability of the vehicle and outputs the braking modes and duty cycles to the vehicle dynamics model. Simulations are carried out by the MATLAB package.

4.1 Lane change simulation

The vehicle and road conditions for a simulation shown in Fig. 6 are as follows. Suppose that a vehicle at the speed of 100km/hr changes a lane on wet roads to avoid collision with an obstacle. It is also assumed that the driver inputs the sinusoidal steering with the frequency of 0.5Hz and the magnitude of 7°. Figure 6 represents the results of the lane change simulation with and without the yaw moment control proposed here.

Without the proposed control, the yaw velocity error increases greatly from about 1.8sec. on. Even after around 2.5sec, the error is still not appreciably reduced and remains excessive for a long time. Consequently, the vehicle loses its directional stability and the vehicle slip angle reaches up to 20°. The trajectory of the vehicle for the lane change shown in Fig. 7 indicates loss of directional stability.

With the proposed yaw moment control activat-

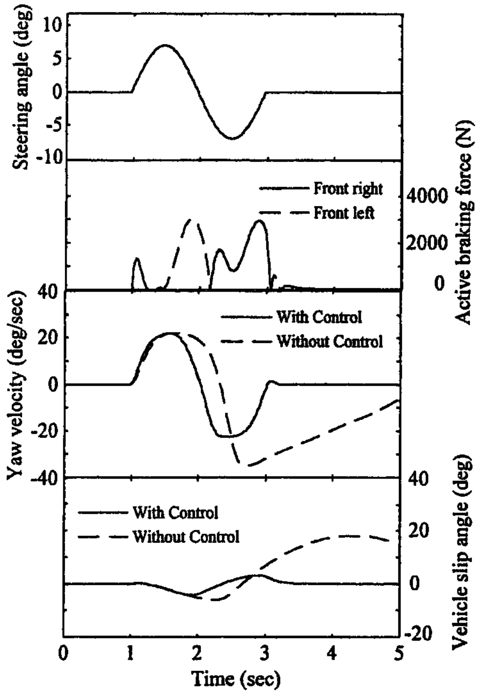

Fig. 6 Vehicle responses for lane change maneuver

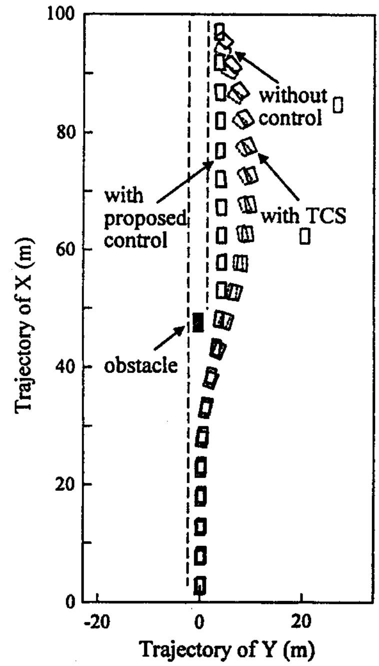

Fig. 7 Vehicle trajectory for lane change maneuver

ed, on the other hand, a brake force is generated to produce the needed yaw moment calculated by the error between the reference and actual yaw velocities. As a result, the vehicle can trace the desired course satisfactorily by following the reference yaw velocity. The vehicle slip angle also stays in the range of 4° to 5°, and the directional stability of the vehicle is maintained.

Figure 7 illustrates the vehicle trajectories for the simulation shown in Fig. 6. Without the control the vehicle does not follow the desired trajectory starting from approximately 50m on in the X direction. Finally, it loses the directional stability and slips laterally. With the TCS system in action, the vehicle cornering performance is not improved effectively because the TCS system controls the slip ratio alone without taking the yaw velocity and slip angle into account. With the TCS control in action, however, the trajectory error is reduced and the vehicle maintains the directional stability. Applying the proposed control, the vehicle effectively avoids an obstacle and changes the lane smoothly as illustrated in the figure.

4.2 J-turn simulation

In the simulation shown in Fig. 8, snowy roads with low friction coefficient are assumed. Suppose that the vehicle at the initial speed of 40km/hr is accelerated and then turns with the steering angle of 5° after 2sec. Figure 8 shows the result of this simulation. Without the control, the yaw velocity increases in accordavce with a change in the steering angle, but decreases after 2.5sec even though the steering angle does not change. Most passenger vehicles have understeer characteristics. For an understeer vehicle, when it is accelerated with the steering wheel fixed, the turning radius increases and the vehicle tends to get off course.

On the other hand, the vehicle with the proposed yaw moment control generates a braking force at the front right wheel to provide the vehicle with an appropriate yaw moment and follows the desired vehicle motion as shown in Fig. 8. Consequently, it maintains the yaw velocity to a higher level than the uncontrolled vehicle. As a result, it can overcome the understeer characteristics, and can faithfully trace the trajectory desired by the driver. Figure 9 shows that the cornering radius of the controlled vehicle in a J

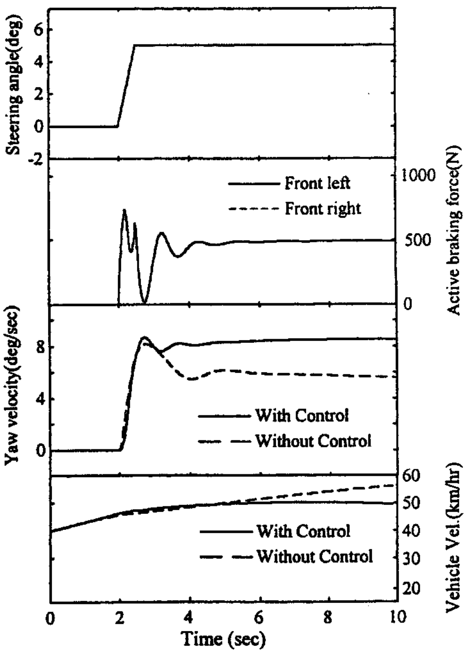

Fig. 8 Vehicle responses to J-turn maneuver

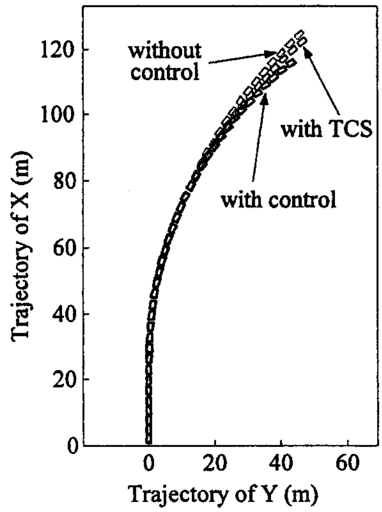

Fig. 9 Vehicle trajectory for J-turn maneuver

-turn maneuver is smaller than that of the vehicle with the TCS system activated as well as that of the uncontrolled vehicle.

5. Conclusions

In the present work the vehicle directional stability is determined bassed on the difference between the reference and actual yaw velocities as well as the vehicle slip angle and slip angular velocity. This information is then used as a criterion for execution of a yaw moment control. The yaw moment control algorithm based on the brake control was proposed for improving directional stability of the vehicle.

It was shown from various simulation studies that the proposed yaw moment control system prevents the vehicle from becoming unstable and provided excellent cornering capabilities even on low friction roads. When an understeer vehicle is accelerated while turning on slippery roads, the controlled brake force prevents the yaw velocity from decreasing and keeps the vehicle on the course set by the driver's inputs.

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