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VEHICLE STABILITY CONTROL SYSTEM FOR ENHANCING STEERABILTY, LATERAL STABILITY, AND ROLL STABILITY

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ABSTRACT−The Vehicle stability control system is an active safety system designed to prevent accidents from occurring and to stabilize dynamic maneuvers of a vehicle by generating an artificial yaw moment using differential brakes. In this paper, in order to enhance vehicle steerability, lateral stability, and roll stability, each reference yaw rate is designed and combined into a target yaw rate depending on the driving situation. A yaw rate controller is designed to track the target yaw rate based on sliding mode control theory. To generate the total yaw moment required from the proposed yaw rate controller, each brake pressure is properly distributed with effective control wheel decision. Estimators are developed to identify the roll angle and body sideslip angle of a vehicle based on the simplified roll dynamics model and parameter adaptation approach. The performance of the proposed vehicle stability control system and estimation algorithms is verified with simulation results and experimental results.

KEY WORDS : Vehicle stability control system, Target yaw rate, Yaw rate controller, Brake pressure distribution, Roll angle estimator, Body sideslip angle estimator

1. INTRODUCTION

There is an ever increasing demand for active safety systems to prevent accidents from occurring through artificial intervention (You *et al*., 2006). This system extends beyond the passive safety notion of merely minimizing damage from accidents and it is increasingly becoming recognized in the market as a necessity. Therefore, diverse research on active safety systems for ground vehicles has been conducted in recent years. Although there have been other alternative candidates relating to actively securing vehicle stability, such as 4WS (Four Wheel Steer), AFS (Active Front Wheel Steer), rear wheel steering, and differential traction (Song *et al*., 2007), the recent mainstream active vehicle safety system is focused on braking intervention by differential braking. This appears to be mainly due to the hardware reliability and cost efficiency resulting from the existing technical outcome relating to ABS (Anti-lock Brake System) and TCS (Traction Control System), which hold wheel slip in the linear slip region during braking/acceleration.

Here, the vehicle is assumed to be equipped with the differential braking system, and, therefore, the vehicle stability control system proposed in this paper mainly focuses on controlling the vehicle by generating a yaw moment with differential braking intervention at each of the four wheels. This paper chooses the yaw rate to be a control variable because the body sideslip angle of a vehicle can be stabilized using yaw rate control with a proper reference yaw rate by changing the body sideslip angle dynamics into stable internal dynamics (You *et al*., 2006). Similarly, the risk of rollover can be mitigated by stabilizing the roll dynamics via yaw rate control. Consequently, the body sideslip angle reduction and the rollover prevention can be fulfilled, by yaw rate control, via the appropriate choice of reference yaw rate (Jo *et al*., 2006).

A yaw rate controller is designed to track the target yaw rate which is determined by combining each reference yaw rate. To generate the required yaw moment from the yaw rate controller, brake pressures at each of the four wheels are distributed based on effective control wheel decision.

In addition, one of the major obstacles to the implementation of the vehicle stability control system in a real-world application is the lack of information on the vehicle states, such as the body sideslip angle and the roll angle. The accuracy of the information greatly influences the performance and reliability of the vehicle stability control system. To address this problem, estimation algorithms to identify the roll angle and the body sideslip angle of a vehicle are developed.

The overall schematic diagram of the proposed vehicle stability control system is given in Figure 1.

The performance of the proposed vehicle stability control system and estimation algorithms is verified with simulations and experiments using a commercial SUV.

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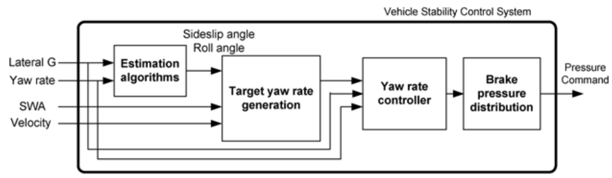

Figure 1. Schematic diagram of proposed system.

2. YAW RATE CONTROLLER DESIGN

If the steering input of a driver, in yaw rate dynamics, is assumed to be an uncontrollable element, then a yaw moment generated from differential braking, which is a single control input, cannot simultaneously control the two state variables, the body sideslip angle and yaw rate of the vehicle's lateral dynamics, in each trajectory for all values of time with an acceptable performance (You *et al*., 2006; Nagai *et al*., 2002). In order to enhance steerability, lateral stability, and roll stability, the body sideslip angle and roll angle as well as the yaw rate should be controlled by an artificial method. The body sideslip angle and roll angle of a vehicle can be stabilized via yaw rate control with proper choice of reference yaw rate by changing each dynamics into a stable one (You *et al*., 2006). In this respect, the target yaw rate is properly chosen from three different reference yaw rates and a yaw rate controller is designed to track the target yaw rate based on the sliding mode control theory.

2.1. Target Yaw Rate Design

Three different reference yaw rates are proposed in this section. By tracking each reference yaw rate, the steerability, lateral stability, and roll stability of a vehicle can be enhanced respectively.

2.1.1. Yaw rate tracking

The reference yaw rate to enhance the steerabilty of a vehicle is defined as the steady-state response of the bicycle model (Van Zanten *et al*., 1996). It is expressed as a function of a vehicle's longitudinal velocity and the driver's steering input as shown in (1):

$$r_{ref\_yaw}=\frac{1}{1-\dfrac{M(l_f C_f-l_r C_r)v_x^2}{2C_f C_r(l_f+l_r)^2}}\frac{v_x}{l_f+l_r}\delta \tag{1}$$

where r is yaw rate; M is vehicle mass; C_f, C_r, are front/rear cornering stiffnesses; l_f, l_r, are the distances between the vehicle C.G. and the front/rear axle; v_x is longitudinal velocity, and δ is steering wheel angle.

Moreover, in the case of driving on a banked road, the yaw rate due to the bank angle is compensated as shown in (3) to prevent the vehicle stability control system from malfunctioning:

$$r_{bank\_comp}=\frac{1}{1+\dfrac{2C_f C_r(l_f+l_r)^2}{Mv_x^2(l_r C_r+l_f C_f)}}\frac{g}{v_x}\sin\varphi \tag{2}$$

$$r_{ref\_yaw\_comp}=r_{ref\_yaw}+r_{bank\_comp} \tag{3}$$

where φ is the road's bank angle.

The compensated reference yaw rate is reasonably constrained by the physical limit of tire-road friction as shown in (4):

$$|Ma_y|=|Mv_x r|\le M\mu g,\ |r_{ref\_yaw\_comp}|\le\frac{\mu g}{v_x} \tag{4}$$

2.1.2. Body sideslip angle reduction

An excessive body sideslip angle of a vehicle makes the vehicle's yaw motion insensitive to the driver's steering input and threatens the lateral stability of a vehicle. Thus, the control strategy to limit the body sideslip angle is required in order to maintain lateral stability.

The reference yaw rate defined as (5), based on the bicycle model, changes the sideslip angle dynamics into stable dynamics as shown in (6), which implies that the body sideslip angle converges to zero asymptotically (Jo *et al*., 2006).

$$r_{ref\_lateral}=K_1\beta+\frac{F_{yr}+F_{yf}\cos\delta}{Mv_x}+\frac{g}{v_x}\sin\varphi \tag{5}$$

$$\dot{\beta}=-K_1\beta \tag{6}$$

The parameters are defined as follows: K_1 is a design parameter, and β is the body sideslip angle.

2.1.3. Rollover prevention

In the vehicle stability control system, it is required to consider roll stability as well as lateral stability. The reference yaw rate defined as (7) changes the roll dynamics into a stable one, as shown in (8), and, therefore, the risk of rollover can be mitigated with yaw rate control (Jo *et al*., 2006):

$$r_{ref\_roll}=\frac{I_{xx}}{M_s e_s v_x}\ddot{\phi}+\frac{g}{v_x}\varphi \tag{7}$$

$$\dot{\phi}=-\frac{K_\phi+M_s e_s g}{C_\phi}\phi \tag{8}$$

where ϕ is roll angle, I_{xx} is roll moment of inertia, M_s is sprung mass, and e_s is the distance between the roll axis and the center of gravity.

2.1.4. Target yaw rate design

Three different reference yaw rates are combined into a target yaw rate depending on the driving situation as shown in (9). Each weighting coefficient, σ_1, σ_2, and σ_3 is calculated based on lateral stability and roll stability control thresholds (Jo *et al*., 2006).

$$r_{target}=\sigma_1 r_{ref\_yaw\_comp}+\sigma_2 r_{ref\_lateral}+\sigma_3 r_{ref\_roll} \tag{9}$$

2.2. Yaw Rate Controller Design

In order to track the proposed target yaw rate, a yaw rate controller is designed based on sliding mode control theory using the bicycle model (10):

$$I_{zz}\dot{r}=l_f(F_{yfr}+F_{yfl})\cos\delta - l_r(F_{yrr}+F_{yrl})+M \tag{10}$$

$$s=r-r_{target} \tag{11}$$

$$\dot{s}=-Ks \tag{12}$$

where I_{zz} is the yaw moment of inertia and K is the control parameter.

The required yaw moment, a control input, is calculated through the designed yaw rate controller.

$$M=I_{zz}K(r_{target}-r)+I_{zz}\dot{r}_{target}+l_rF_{yr}-l_fF_{yf}\cos\delta \tag{13}$$

3. BRAKE PRESSURE DISTRIBUTION

The designed yaw rate controller calculates the required yaw moment to stabilize the dynamic maneuvers of a vehicle. In order to generate the required yaw moment, it needs to be decided how braking forces should be distributed to the individual wheels.

In this section, methods for determining the effective braking wheel and effective brake releasing wheel are established. Brake pressure distribution strategies during accelerating and braking are also presented.

3.1. Effective Control Wheel Decision

Longitudinal and lateral tire forces are coupled with each other such that lateral tire force decreases with the increase of longitudinal tire force. In this respect, it is noted that there exists one effective braking wheel, even between the same side wheels, that will yield the same directional yaw moment since braking pressure for generating the designed yaw moment affects the lateral tire force as well as the longitudinal tire force (Koibuchi *et al*., 1996).

Thereby, the effective braking wheel for the generation of a yaw moment by differential braking corresponding to respective cornering situations can be determined as shown in Figure 2.

So far, only the cases where drivers do not operate their brakes have been considered. However, most drivers employ the brake pedal in the critical driving situations; thus, the intervention of the yaw moment through releasing brake pressure should be considered separately from generating brake pressure. Braking situations created by the driver's intent involve the effective brake releasing wheel, according to the same principle as mentioned above.

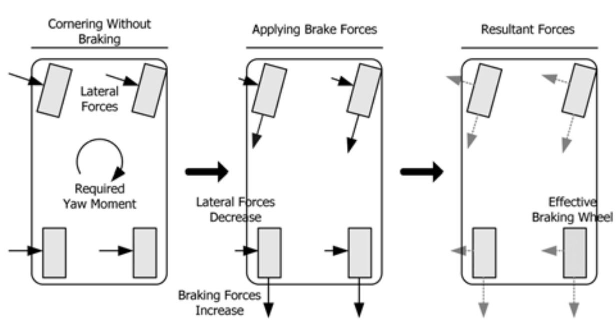

Figure 2. Effective braking wheel decision.

3.2. Brake Pressure Distribution

Longitudinal tire force that can be generated at the individual wheel is constrained with a physical limit. Therefore, while the driver is not applying brake input, there may be a case where the braking force at the effective braking wheel cannot fulfill the required yaw moment. In this case, the additional braking force on the same side wheel compensates for the shortage of the required yaw moment.

When the driver applies brake input, a yaw moment can be generated by the decreasing braking force at the effective brake releasing wheel as well as increasing the braking force at the effective braking wheel. Furthermore, in order to prevent the vehicle from decelerating more than the driver intends, the total braking forces should be constant. In case the braking forces at the effective braking wheel and effective brake releasing wheel cannot perform the required yaw moment, the additional braking forces at the remaining wheels compensate for the shortage of the required yaw moment.

4. ESTIMATION ALGORITHM DESIGN

One of the major obstacles to the implementation of the vehicle stability control system in a real-world application is the lack of information on the vehicle states and parameters. Therefore, estimation algorithms to identify the roll angle and body sideslip angle of a vehicle are essential for reliable operation of a vehicle stability control system, and they are developed herein.

4.1. Vehicle Roll Angle Estimation

As shown in (14), the actual measurements of lateral acceleration sensors are affected by the vehicle roll and road bank angle, as well as vehicle lateral acceleration (You *et al*., 2006).

$$a_{y,meas}=v_x\left(\frac{d\beta}{dt}+r\right)-g\sin(\varphi-\phi) \tag{14}$$

In this study, the roll angle is estimated by the following simplified roll dynamics model shown in Figure 3.

$$I_{xx}\ddot{\phi}+C_\phi\dot{\phi}+K_\phi\phi=M_se_s(v_xr+\dot{\beta})-M_se_sg\sin(\varphi-\phi)$$
$$=M_se_sa_{y,meas} \tag{15}$$

The estimated roll angle of a vehicle is determined as follows:

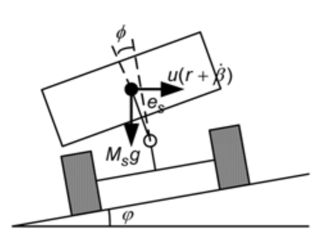

Figure 3. Simplified roll dynamics model.

$$\phi = \frac{M_s e_s}{I_{xx}s^2 + C_\phi s + K_\phi} a_{y,meas} \tag{16}$$

where s denotes the Laplace transform operator, C_ϕ is roll damping coefficient, and K_ϕ is roll stiffness.

4.2. Body Sideslip Angle Estimation

The front/rear lateral tire force acting on the vehicle body can be given as (17) and (18). By applying the linear lateral tire force model to the estimated lateral tire force, the following lateral tire force descriptions can be obtained.

$$F_{yf} = \frac{Ml_r a_y + I_{zz}\dot{r}}{l_f + l_r} = C_f \alpha_f = C_f\left(\delta - \beta - \frac{l_f}{v_x} r\right) \tag{17}$$

$$F_{yr} = \frac{Ml_f a_y - I_{zz}\dot{r}}{l_f + l_r} = C_r \alpha_r = C_r\left(-\beta + \frac{l_r}{v_x} r\right) \tag{18}$$

Then, the body sideslip angle can be algebraically determined from each front/rear lateral tire force equation, as shown in (19), if the correct cornering stiffness is known.

$$\beta_f = -\frac{F_{yf}}{C_f} + \delta - \frac{l_f}{v_x} r,\ \beta_r = -\frac{F_{yr}}{C_r} + \frac{l_r}{v_x} r \tag{19}$$

To identify the cornering stiffness, the following description can be manipulated by eliminating the yet-to-be-determined body sideslip angle in (20).

$$\frac{F_{yf}}{C_f} - \frac{F_{yr}}{C_r} = \delta - \frac{l_f + l_r}{v_x} r \tag{20}$$

Then, the unknown parameters can be linearly separated from the known regressor terms as follows.

$$z = \delta - \frac{l_f + l_r}{v_x} r,\ \Phi = [\phi_1\ \phi_2]^T = [F_{yf} - F_{yr}]^T$$

$$\Theta = [\theta_1\ \theta_2]^T = [C_f^{-1}\ C_r^{-1}]^T \tag{21}$$

To compensate for the effect of load transfer and traction/braking force, the nominal part should be separated from the perturbed part in the parametric model. Then a new parametric model can be defined as follows:

$$z = \frac{1}{s+\alpha}\left\{\delta - \frac{l_f + l_r}{v_x} r + F_{yr}\left(\frac{1}{C_r}\right)_n - F_{yf}\left(\frac{1}{C_f}\right)_n\right\} \tag{22}$$

$$\Theta = [\theta_1\ \theta_2]^T = \left[\Delta\left(\frac{1}{C_r}\right)\ \Delta\left(\frac{1}{C_f}\right)\right]^T \tag{23}$$

$$\Phi = [\phi_1\ \phi_2]^T = \frac{1}{s+\alpha}[F_{yf} - F_{yr}]^T \tag{24}$$

where α is a filter constant to be chosen appropriately. With the above parametrization, the following adaptive law can be obtained using the ordinary gradient algorithm (Ioannou and Sun, 1996):

$$\dot{\hat{\theta}}_1 = \gamma_1 \varepsilon \phi_1,\ \dot{\hat{\theta}}_2 = \gamma_2 \varepsilon \phi_2,\ \varepsilon = z - \hat{z} \tag{25}$$

where γ_1 and γ_2 are adaptive gains and ε is the output estimation error.

It can be noted that an understeering motion mostly results from lateral tire force saturation in the front wheels, which implies that the lateral slips of the front tires are in the nonlinear region. In contrast, an oversteering motion is mainly caused by lateral tire force saturation in the rear tires. In this respect, a suitable choice according to the specific cornering situation can exist. For example, it is desirable to use the rear tire model (19) in cases of understeering because the estimated sideslip angles from the lateral tire force equation at the front wheel are not valid due to the deviation from the linear friction region.

In cases where the tire sideslip angles of all wheels exceed the linear friction limit during vehicle cornering, the model-based estimation approach of the body sideslip angle no longer keeps its validity. Therefore, the estimated body sideslip angle via the integration of lateral acceleration sensor signal is properly combined with the above model-based estimated value.

5. SIMULATION RESULTS

Computer simulations are carried out to verify the effectiveness of the designed control system using CarSim software.

5.1. Slalom Simulation

A slalom maneuver is simulated with an initial vehicle speed of 100 km/h. In the simulation, the friction coefficient between the tires and the road is assumed to be 0.2.

The yaw rate and the body sideslip angle of the vehicle are shown in Figure 4 and Figure 5, respectively. The results show that the vehicle with the proposed control

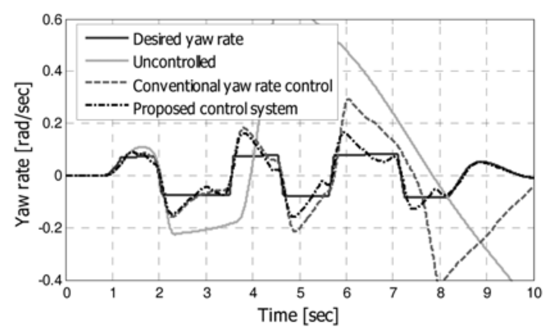

Figure 4. Yaw rate in slalom maneuver.

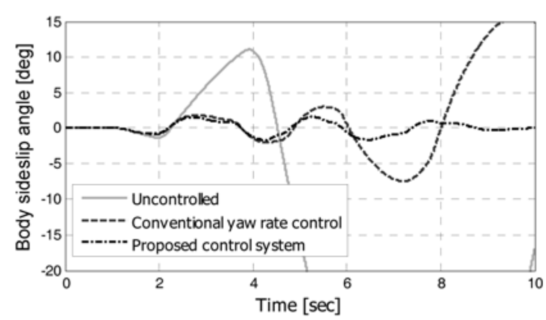

Figure 5. Body sideslip angle in slalom maneuver.

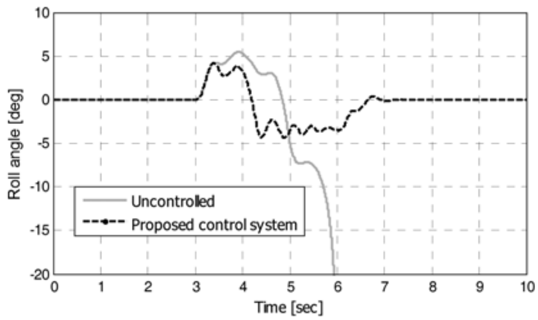

Figure 6. Roll angle in fishhook maneuver.

system exhibits a better performance than the uncontrolled vehicle and the vehicle with the conventional yaw rate control system.

5.2. Fishhook Simulation

A fishhook maneuver is simulated with an initial vehicle speed of 100 km/h. In the simulation, the friction coefficient between the tires and the road is assumed to be 1.1.

The roll angle of the vehicle is shown in Figure 6. The roll angle of the uncontrolled vehicle increases rapidly, and the vehicle finally rolls over. However, it can be seen that the roll angle of the vehicle with the proposed control system is reduced, thus successfully mitigating the risk of rollover.

6. EXPERIMENTAL RESULTS

This section investigates the performance of the proposed vehicle stability control system and the developed identification algorithms to estimate the body sideslip angle and roll angle via experimental results using a real vehicle. A commercial SUV equipped with a gyroscope, a steering wheel angle sensor, a lateral acceleration sensor, and a GPS-type lateral velocity and roll angle sensor was used for the experiments.

6.1. Roll Angle Estimation Results

The vehicle tests are conducted to evaluate the roll angle estimation algorithm. The experimental result shows that the roll angle is estimated satisfactorily.

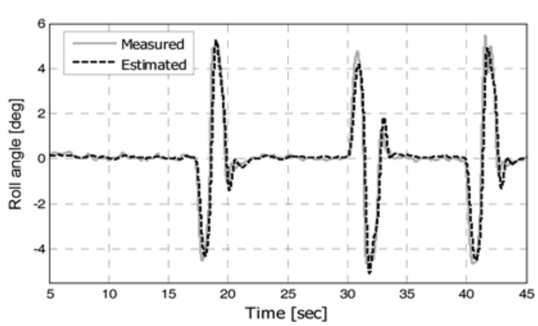

Figure 7. Roll angle estimation result.

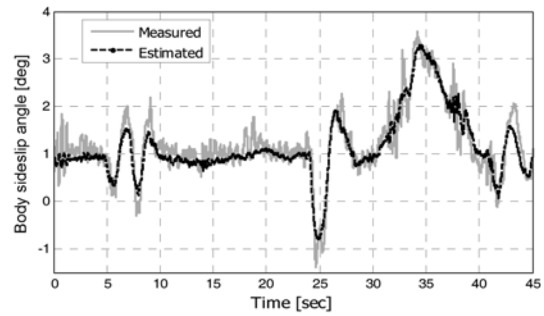

Figure 8. Body sideslip angle estimation result.

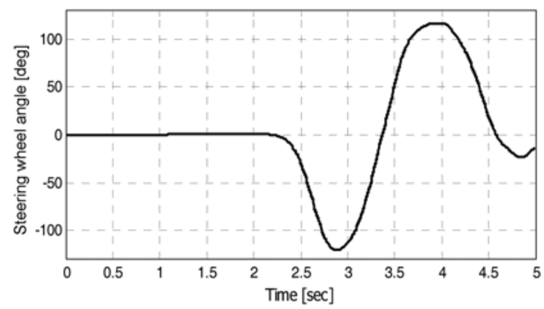

Figure 9. Steering wheel angle in single lane change.

6.2. Body Sideslip Angle Estimation Results

The experimental result is presented in Figure 8. It shows that the body sideslip angle is estimated well compared to the measured value.

6.3. Single Lane Change Results

Experiments with a real vehicle are carried out to verify the effectiveness of the proposed vehicle stability control system.

Figure 10 and Figure 11 show the experimental results of a single lane change maneuver on a dry asphalt road comparing the vehicle's behavior in the cases with and without the proposed vehicle stability control system. The vehicle was run at an initial speed of 90 km/h with the steering input as shown in Figure 9.

The vehicle without the control system became unstable as its body sideslip angle increased until it finally lost its steerability. The results of the vehicle with the vehicle

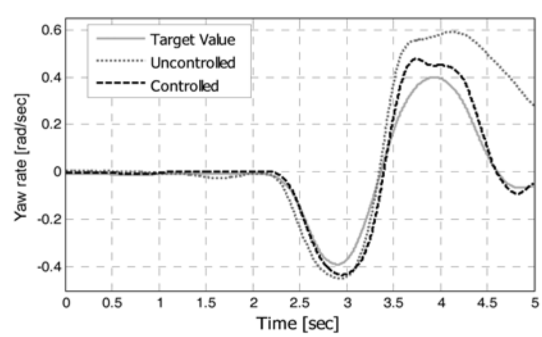

Figure 10. Yaw rate in single lane change.

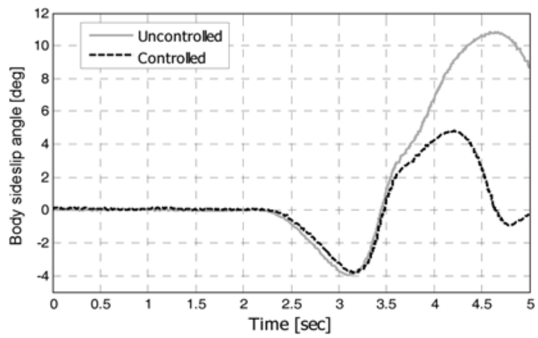

Figure 11. Body sideslip angle in single lane change.

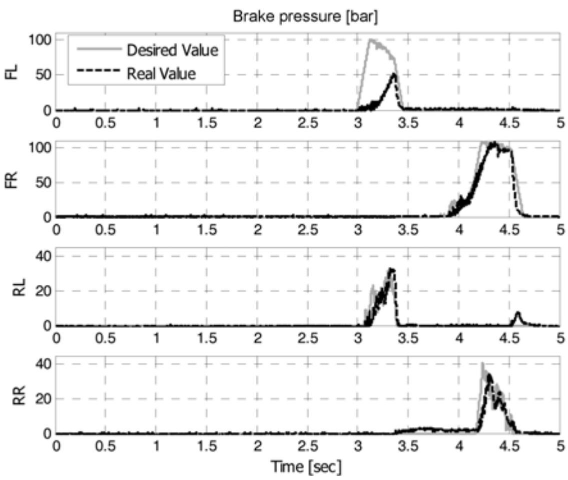

Figure 12. Brake pressure intervention.

stability control system show that the control system intervenes with active braking at the individual wheels, as shown in Figure 12, and that the vehicle tracks the target yaw rate satisfactorily while maintaining a small body sideslip angle.

7. CONCLUSION

In this paper, in order to enhance the steerability, lateral stability, and roll stability of a vehicle, a target yaw rate is properly chosen and a yaw rate controller is designed to track the designed target yaw rate. To generate the required yaw moment, brake pressures at four wheels are distributed based on effective control wheel decision. Estimation algorithms to identify the roll angle and the body sideslip angle of a vehicle are developed.

The satisfactory performance of the proposed vehicle stability control system and estimation algorithms is verified with simulation results using CarSim software and experimental results using a real vehicle, proving their practicability in real-world applications.

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