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ESTIMATION OF VEHICLE STATE AND ROAD BANK ANGLE FOR DRIVER ASSISTANCE SYSTEMS

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ABSTRACT–This paper presents vehicle state and road bank angle estimators for applications to driver assistance systems. Combined road bank angle estimator and vehicle lateral velocity is used to obtain good estimate of the vehicle side slip angle. The estimator is evaluated in test tracks using a test vehicle. For an application of driver assistance systems which consist of adaptive cruise controller and vehicle stability controller, driving simulator based evaluation is conducted under an obstacle-avoidance driving situations on a bank road. With bank angle estimator, the vehicle stability control logic and its desired values are modified to compensate the bank angle effects. The simulation result shows that the combined vehicle state and bank angle estimators ensure the performance of proposed controller on severe driving condition.

KEY WORDS : State estimator, Bank angle, Adaptive cruise control, Vehicle stability control

1. INTRODUCTION

Driver assistant systems are widely commercialized by many motor companies. A Vehicle lateral velocity, and corresponding vehicle body slip angle and tire slip angles, are important indicators of vehicle handling operating region and road condition, especially during limit handling (Van Zanten, 2002). For the VSC (vehicle stability control) system, accurate vehicle side slip angle detection is one of key issues to confirm the system performance (Nishio and Tozu, 2001; Farrelly and Wellstead, 1996; Kaminaga and Naito, 1998; Lee, 2004).

Road bank angle have a direct influence on vehicle dynamics and lateral acceleration measurement. When the vehicle runs on a banked road, lateral acceleration sensor detects a component of the gravity due to the road bank angle. Therefore, the lateral acceleration offset caused by bank angle have to be compensated to ensure a robustness of lateral velocity estimator and vehicle stability controller. Fukuda presented the importance of road bank angle to estimate the side slip angle in his work (Fukuda, 1998). Hahn *et al.* (2002) proposed bank angle estimator using disturbance observer, but it needs an accurate estimate of lateral velocity which is also affected by road bank (Hahn *et al.*, 2002). Cho presented bank angle detection algorithm to hold an unnecessary control intervention (Cho and Kwak, 2003). However, vehicle stability controller should be needed in critical driving situation such as lane change maneuver at high speed on the bank road. The dynamic bank angle estimator to decouple the lateral dynamics and road disturbances using transfer function approach was proposed by Tseng, and is applied in this research (Tseng, 2002). The bank angle estimator is evaluated via vehicle test under various driving conditions on a banked test road. In this study, the vehicle planar motion model based sliding controller is modified with bank angle estimate (Yi and Chung, 2003). The desired vehicle states are obtained by steady state values of 2-D linear vehicle model with disturbance induced by bank angle.

The control threshold of the VSC is determined by using phase plane analysis (Inagaki *et al.*, 1994). VSC systems connect the human driver, controller and vehicle behavior. Since VSC always works with the driver, the overall vehicle performance will depend on not how well the VSC works, but rather its interaction with the human driver. It is hard to predict driver input with a mathematical driver model. It is also difficult to test an actual vehicle due to time and costs. Therefore, we applied a driving simulator to evaluate the system. If the driving simulator can simulate the actual driving situations well, the human-in-the-loop evaluation makes sense. The driving simulator consists of a three dimensional vehicle dynamic model, driver interface and a visual display. It

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was validated by the driver's input data of actual vehicle test data (Chung, 2004). Critical driving situations, such as misjudgment of cornering curvature and emergency lane change, are happened while cornering on a bank road with high speed, therefore the performance of both lateral velocity estimation and the control logic are important to eliminate the bank angle disturbance. Since the most driver assistant systems are interacting vehicle-driver-controller, a combination of driver assistant systems is hard to evaluate. In this study, human-in-the-loop simulation using driving simulator is performed to validate the performance of proposed controller.

2. EXPERIMENTAL VALIDATIONS OF COMBINED BANK ANGLE AND LATERAL VELOCITY ESTIMATOR

Under an assumption of a 2-D linear vehicle model, a vehicle lateral motion with bank angle effect can be simplified as follows.

$$\dot{v}=a_y-u\gamma+g\sin\varphi \tag{1}$$

where, v is lateral vehicle velocity, u is longitudinal vehicle speed, g is yaw rate, a_y is lateral acceleration, and g is gravitational acceleration. In steady state cornering, vehicle lateral velocity can be assumed as constant ($\dot{v}=0$), and road bank angle is estimated as follows.

$$\varphi_{ss}=\sin^{-1}\left(\frac{u\gamma-a_y}{g}\right) \tag{2}$$

where, φ_{ss} represents steady state estimate of bank angle. It is reasonable for holding controller when the steady state bank angle is detected in high speed. However, the estimate could contain a bias in critical maneuvers since the lateral acceleration is neglected.

2-D linear vehicle model with gravitational force is presented as following equation.

$$x=[v\ \gamma]^T$$

$$\dot{x}=Ax+B\delta+E\sin\varphi \tag{3}$$

The state variables vehilcle lateral velocity, v, and yaw rate, γ. The bank angle, φ is working as a disturbance, and steering angle, δ, as an input of the system.

$$A=\begin{bmatrix}-\dfrac{C_f+C_r}{mu} & -u-\dfrac{aC_f-bC_r}{mu}\\ -\dfrac{aC_f-bC_r}{I_z u} & -\dfrac{a^2C_f+b^2C_r}{I_z u}\end{bmatrix} \tag{4}$$

$$B=\begin{bmatrix}\dfrac{C_f}{m} & \dfrac{aC_f}{I_z}\end{bmatrix}^T \qquad E=\begin{bmatrix}g & 0\end{bmatrix}^T$$

where m is vehicle mass, and I_z is moment of inertia of z-axis. a and b are distance from center of gravity to front and real axle shafts. C_f and C_r are front and rear cornering stiffness of the tire.

The dynamic bank angle estimator which is proposed by Tseng is shown as follows [9]

$$\varphi_{dyn}=\left(\sin^{-1}\sin\varphi_{ss}\cdot\max\left[0,1-|DFC|-\left|\frac{d\sin\varphi_{ss}}{dt}\right|\right]\right) \tag{5}$$

where DFC denotes a dynamic factor coefficient related to the transient lateral dynamics. Transfer function study between inputs, steering wheel angle and bank angle, and measurements, yaw rate and lateral acceleration, is applied to obtain dynamic factor coefficient information which is used in the final estimation as a multiplicative factor to reduce the bias in steady state bank estimate. As the changes in lateral dynamics grow more excessive, the dynamic estimate of bank angle becomes conservative. In this paper, φ_{dyn} is used for the lateral velocity estimator and the VSC logic improvement. The vehicle lateral velocity is estimated with state observer as follows

$$\dot{\hat{x}}=A\hat{x}+B\delta+K_{obs}(y-C\hat{x}-d\delta) \tag{6}$$

where,

$$y=[a_{y,m}\ \gamma_m]^T$$

$$C=\begin{bmatrix}-\dfrac{C_f+C_r}{mu} & -\dfrac{a\cdot C_f-b\cdot C_r}{mu}\\ 0 & 1\end{bmatrix} \qquad D=\begin{bmatrix}\dfrac{C_f}{m}\\ 0\end{bmatrix}$$

The observer gain, K_{obs}, is by choosing eigenvalues which determine observer characteristics. $a_{y,m}$ is modified measurement of lateral acceleration.

$$a_{y,m}=a_{y,mo}-g\ \mathrm{ain}\,\hat{\varphi} \tag{7}$$

where, $a_{y,mo}$ is measured lateral acceleration which includes bank angle effect. Conventionally, state observer in equation 6 without bank angle estimate shows bias when the vehicle on the bank road. Both bank angle and lateral velocity estimators are tested on a proving ground shown in the Figure 1. It consists of 4 lanes with 300 m radius tracks. The lane width is 3.8 meters, and the bank angle of lane 4 is about 35 degrees.

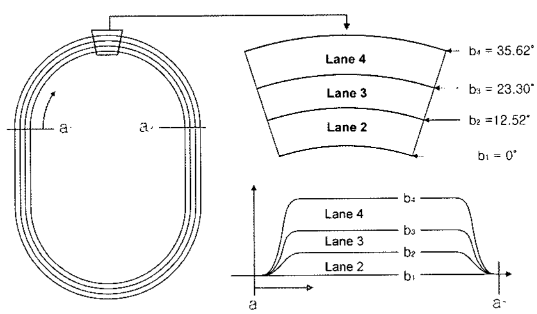

Figure 1. Proving ground configuration.

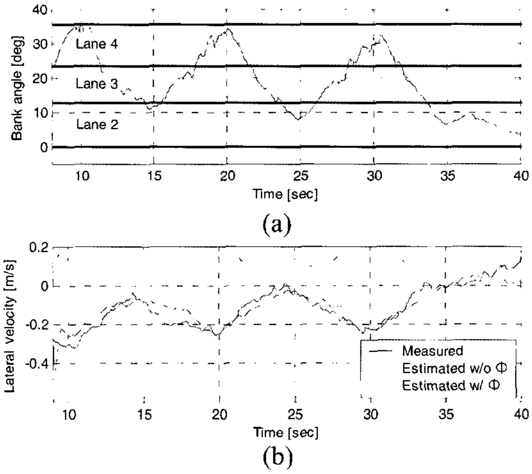

Figure 2. Estimation results (slalom 1).

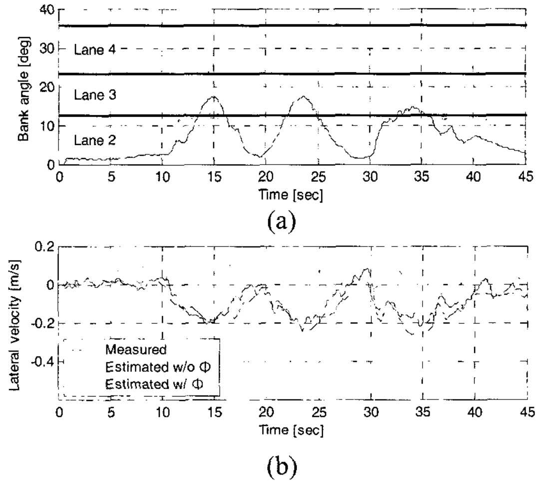

Figure 3. Estimation results (slalom 2).

Test vehicle is 2,500 cc of engine displacement FF passenger vehicle with automatic transmission and ABS/TCS. A PC based data acquisition is used for low-cost vehicle test condition. To acquire sensor signals within 5 ms for vehicle dynamics application, CAN analyzer and C167 controller are used. By using PC, the experiment consumes relatively low cost. Throttle position sensor, wheel speed sensors, engine RPM, and reaction carrier RPM is measured by existing sensors in the ABS/TCS installed vehicle. Yaw rate and lateral and longitudinal acceleration are measured by combinations sensors, and an optical absolute velocity measurement device is installed on rear bumper of the vehicle and its measurements are assumed to be the actual velocities.

Figure 2 shows slalom test on a bank road by varying vehicle speed. After approaching the lane 4 at high speed, the driver repeats the maneuver reducing vehicle speed to change lane toward lower lanes, and increasing vehicle speed toward lane 4, the highest lane. The steering wheel angle during the maneuver is less than 5 degrees. The lateral velocity varies relatively smaller on the bank road than on the flat road under lane changing situations. As shown in the Figure 2(b), vehicle lateral velocity estimation results with bank angle estimate (Figure 2(a)) shows good tracking performance to the measured signal. Without considerations of bank angle, the lateral velocity estimate has bias resulting from gravitational forces induced by bank angle.

The slalom test on a bank road by sinusoidal steering input is shown in the Figure 3. After approaching the bank road region, the driver steers toward left and right without changing vehicle speeds. The vehicle speeds during the maneuver is around 90 to 95 kph. Since dynamic bank angle estimator confirms reduce the bias in steady state bank estimate on transient lateral dynamics, the dynamic estimate is said to be representing actual bank angle rather than the steady state estimate which is affected by unknown dynamics. Like previous case, the lateral velocity estimation result with bank angle information is more accurate than the result without bank angle. If the VSC logic needs not only yaw rate but lateral velocity information, bank angle estimator possibly increases the robustness of the lateral velocity estimator.

3. DIFFERENTIAL BRAKING STRATEGY USING BANK ANGLE INFORMATION

If an excessive deviation between actual and driver intended lateral response exists or is developing, the VSC intervenes a driver's operation through actuators to keep the vehicle stable. The main concept of the study is that a driver steers similarly while in critical driving situation whether the bank road or flat road. To compensate the bank angle effect, desired value and control law of VSC are modified with bank angle estimate. The following Figure 4 shows the schematic of proposed control system.

Although the driver may not recognize an actual bank angle and its effect, the driver's intention has to be modified as follows. The steady state values of differential equation 3 are calculated as follows

$$x_{ss}=[\beta_{ss}\ \ \gamma_{ss}]^{T}=-A^{-1}(B\delta+E\hat{\varphi})$$

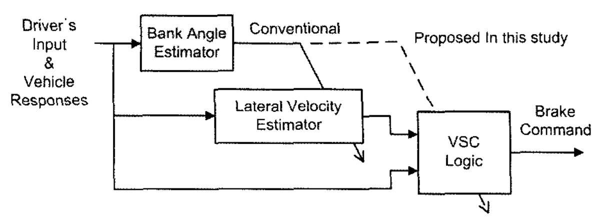

Figure 4. the VSC proposed in this study.

$$\beta_{ss}=\frac{\{2b(a+b)C_fC_r-mu^2aC_f\}\delta+(a^2C_f+b^2C_r)mg\hat{\varphi}}{2C_fC_r(a+b)^2-mu^2(aC_f-bC_r)}$$
$$\gamma_{ss}=\frac{2(a+b)C_fC_ru\delta+(aC_f+bC_r)mug\hat{\varphi}}{2C_fC_r(a+b)^2-mu^2(aC_f-bC_r)} \tag{8}$$

where $\hat{\varphi}$ is bank angle estimate. Desired yaw rate have to be limited under tire-road friction condition [1]. With bank angle estimate, the maximum value of the desired yaw rate is modified by using bank angle information as follows

$$\gamma_{des}=\begin{cases} \gamma_{ss} & if\ |\gamma_{ss}|<\dfrac{\mu(g+\sin\hat{\varphi})}{u} \\ \dfrac{\mu(g+\sin\hat{\varphi})}{u}sgn(\gamma_{ss}) & if\ |\gamma_{ss}|\geq\dfrac{\mu(g+\sin\hat{\varphi})}{u} \end{cases} \tag{9}$$

where μ is tire road friction coefficient. The vehicle yaw plane model is shown in the following equations. The state variables are longitudinal velocity, u, lateral velocity, v, and yaw rate, γ. It is assumed that the bank angle affects lateral direction only for simplicity.

$$m\dot{u}=F_{xr}+F_{xf}\cos\delta-F_{yf}\sin\delta+mrv$$
$$m\dot{v}=F_{yr}+F_{xf}\sin\delta+F_{yf}\cos\delta-mrv+mg\sin\varphi$$
$$I_z\dot{r}=aF_{xf}\sin\delta+aF_{yf}\cos\delta-bF_{yr}$$
$$+\frac{d}{2}(F_{xfr}-F_{xfl})\cos\delta+\frac{d}{2}(F_{xrr}-F_{xrl}) \tag{10}$$

where F_{xf} and F_{xr} are front and rear longitudinal tire forces. F_{yf} and F_{yr} are front and rear lateral tire forces. d is track width. By assuming small wheel slip angles, the lateral tire forces are given as follows:

$$F_{yf}=C_f\alpha_f=C_f\left(\delta-\frac{v+\gamma a}{u}\right)$$
$$F_{yr}=C_r\alpha_r=C_f\left(\frac{\gamma b-v}{u}\right) \tag{11}$$

where α_f and α_r are front and rear tire slip angle. Substituting the expressions into equation (10) and using small angle approximations, the vehicle model is represented as

$$m\dot{u}=F_{xr}+F_{xf}-C_f\left(\delta-\frac{v+\gamma a}{u}\right)\delta+m\gamma v$$
$$m\dot{v}=C_r\frac{\gamma b-v}{u}-C_f\frac{v+\gamma a}{u}+(C_f+F_{xf})\delta-m\gamma u+mg\sin\varphi$$
$$I_z\dot{r}=aF_{xf}\delta+a\cdot C_f\left(\delta-\frac{v+\gamma a}{u}\right)-b\cdot C_r\frac{\gamma b-v}{u}$$
$$+\frac{d}{2}(F_{xfr}-F_{xfl})+\frac{d}{2}(F_{xrr}-F_{xrl}) \tag{12}$$

The tire longitudinal forces can be controlled by the wheel braking control. Since the wheel dynamics are much faster than the vehicle body dynamics, under front wheel driving, the tire longitudinal forces can be approximated as

$$F_{xf}=F_{xfl}+F_{xfr}$$
$$\cong\left(\frac{1}{2}\frac{T_s}{r_{wf}}-\frac{K_{Bf}}{r_{wf}}P_{Bfl}\right)+\left(\frac{1}{2}\frac{T_s}{r_{wf}}-\frac{K_{Bf}}{r_{wf}}P_{Bfr}\right) \tag{13}$$
$$=\frac{T_s}{r_{wf}}-\frac{K_{Bf}}{r_{wf}}(P_{Bfl}+P_{Bfr})$$
$$F_{xr}=F_{xrl}+F_{xrr}\cong-\frac{K_{Br}}{r_{wr}}P_{Brl}--\frac{K_{Br}}{r_{wr}}P_{Brr}$$

where F_{xfl} is longtitudinal tire force of front left wheel, F_{xfr} is front right, F_{xrl} is rear left, and F_{xrr} is rear right. T_s is shaft torque, K_{Bf} and K_{Br} are front and rear brake gains, and P_{Bfl}, P_{Bfr}, P_{Brl}, P_{Brr} are brake pressures of 4 wheels. r_{wf} and r_{wr} are front and rear wheel effective radii. For brake proportioning, the brake pressure of the rear wheels needs to be controlled as follows:

$$P_{Br}=\frac{K_{Bf}ag+a_xh}{K_{Br}bg-a_xh}P_{Bf}=f_B(a_x)P_{Bf} \tag{14}$$

where a_x is longitudinal acceleration, and h is height of center of gravity. Above equation is derived from free body diagram of longitudinal rigid vehicle. The external forces applied to the model are lateral, longitudinal tire forces, and a disturbance induced by the bank angle. Previous studies have indicated that both slip angle and yaw rate should be combined in the control objective for stability control. With desired yaw rate from equation 8, the sliding surface is defined as weighted combinations of yaw rate error and side slip angle.

$$s=\frac{1}{2}(\gamma_{des}-\gamma)^2+\frac{1}{2}\rho\cdot\beta^2 \tag{15}$$

where, ρ is positive constant. The control objective is to keep the scalar s at zero. This can be achieved by choosing the control law such that

$$\frac{1}{2}\frac{d}{dt}s^2=s\dot{s}\leq-\eta\cdot s^2 \tag{16}$$

where η is a positive constant (Slotine and Li, 1991). This equation states that the squared distance to the surface, as represented by s^2, decreases along all system state trajectories. Thus, it constrains trajectories to point toward the surface s=0, and once the state is on the surface, it remains on the surface. The derivative of the scalar variable s along the trajectories is given by

$$s=(\gamma_{des}-\gamma)(\dot{\gamma}_{des}-\dot{\gamma})+\rho\beta\dot{\beta}$$
$$=\Gamma(\delta,u,v,\gamma)+e_\gamma\left(\frac{A^*}{m}-\frac{a}{I_z}\right)\frac{\delta T_s}{r_{wf}}$$
$$+P_{Bfl}e_\gamma\left(\left(-\frac{A^*\delta}{m}-\frac{d}{2I_Z}\right)\frac{K_{Bf}}{r_{wf}}+\frac{a\delta K_{Bf}}{I_zr_{wf}}-\frac{\rho\delta K_{Bf}}{mur_{wf}}\right)$$

$$+P_{Brl}e_{\gamma}\left(-\frac{A^{*}\delta}{m}-\frac{d}{2I_Z}\right)\frac{K_{Bf}}{r_{wf}}$$

$$+P_{Bfl}e_{\gamma}\left(\left(-\frac{A^{*}\delta}{m}+\frac{d}{2I_Z}\right)\frac{K_{Bf}}{r_{wf}}+\frac{a\delta K_{Bf}}{I_z r_{wf}}-\frac{\rho\delta K_{Bf}}{mur_{wf}}\right)$$

$$+P_{Brl}e_{\gamma}\left(-\frac{A^{*}\delta}{m}+\frac{d}{2I_Z}\right)\frac{K_{Bf}}{r_{wf}}$$

$$+\rho\beta\left[\frac{1}{mu}\left\{C_r\alpha_r+C_f\alpha_f+\frac{T_s}{r_{wf}}\delta\right\}-\gamma+\frac{g\sin\varphi}{u}\right] \tag{17}$$

where,

$$\Gamma(\delta,u,v,r)=A^{*}rv\delta-A^{*}\delta^{2}\frac{1}{m}C_f\alpha_f-\frac{a}{I_z}C_f\alpha_f+\frac{b}{I_z}C_r\alpha_r$$

$$A^{*}=\frac{(1-u^2/u_c^2)}{(a+b)(1+u^2/u_c^2)^2}$$

u_c is characteristic speed of a vehicle.

Yaw moment for stability control can be produced using front and rear wheel differential braking control. There exists a sliding controller gain K such that

$$\dot{s}=-K\cdot s\leq-\eta\cdot s \tag{18}$$

By substituting equation (15) and (17) into equation (18), front left and right brake inputs are derived with simplified tire model. The brake inputs obtained by the equation (16) with bank angle estimate are calculated in different ways from the logic without bank angle estimate (Yi and Chung, 2003). For an example of right turning, the desired value or limited value of yaw rate a negative (clockwise) bank angle is lessened with bank angle estimate. If a driver steers inside of the clockwise curved road and VSC is needed to prevent understeer, the control inputs obtained by the proposed logic are less than that of conventional logic which neglects bank angle effects. On the contrary, if a driver steers outside, the proposed control inputs are more than the conventional one. The modified sliding controller is expected for eliminating bank angle effects by using bank angle estimator. The control threshold of VSC is determined by using $\beta-\dot{\beta}$ phase plane with three steps to prevent unnecessary control intervention induced by the bank angle. First, a sensitivity gap of $\dot{\beta}$ is considered to prevent measure noise and bank angle effect. Second, the algorithm checks if the trajectory in the phase plane deviates from designed threshold surface which is smaller than the physical stability limit. Finally, the VSC is activated when the direction of the trajectory is outward from the threshold surface. By using bank angle estimate, the controller is not activated while steady cornering on a bank road. In critical driving situation on a bank road, excessive yaw rate error and side slip angle exist or are developing, then the VSC activated to reduce those variables of the sliding controller.

4. DRIVING SIMULATION WITH DRIVER ASSISTANCE SYSTEMS

The driver assistance systems considered in this study are adaptive cruise control system to hold vehicle speed automatically and vehicle stability control system to maintain vehicle lateral stability in critical driving situation. Figure 5 shows the schematic diagram of the simulation study. A driver steers the vehicle by his or her own decision which is induced by driving situation and environment. Since throttle and brake inputs are generated by driver assistance system, a driver input or reaction is only a steering wheel angle.

An unknown disturbance, which is not perceivable information for a normal average driver, is treated by driver assistance system. The bank angle is considered as unknown disturbance, and the proposed vehicle stability control system compensate the effects by using bank angle estimator. Simulation study is conducted using a vehicle simulation package validated using vehicle test data (Chung *et al.*, 2004). To evaluate stability controller and driver reaction at critical driving conditions, a driving simulator based evaluation is conducted with severe driving scenario. Figure 6 shows the schematics of driving scenario in this study. The driver sets vehicle speed at 80kph while on 150m-radius cornering (lane 2), and faces a cubic-shaped obstacle. To avoid the obstacle, the driver steers a lane change maneuver toward inside (lane 1) or outside lane (lane 2). The bank angle of each lanes is 9 degrees.

The Figure 7 shows the steering inputs and vehicle

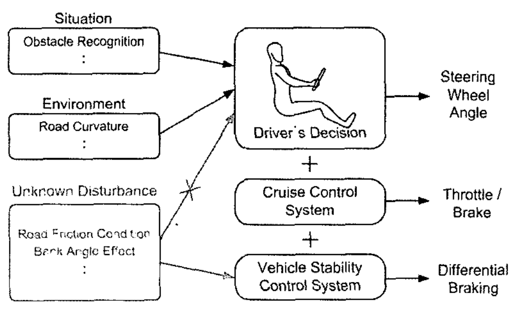

Figure 5. Driver perception and reaction.

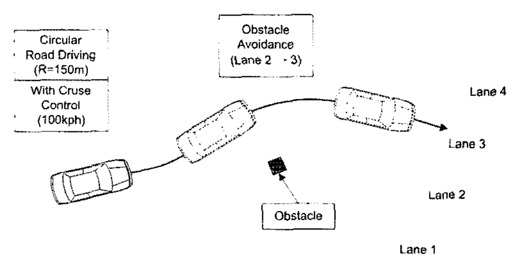

Figure 6. Driving scenario.

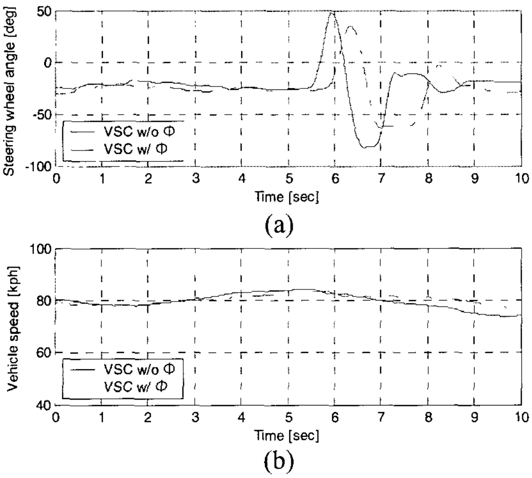

Figure 7. Steering wheel angle and vehicle speed.

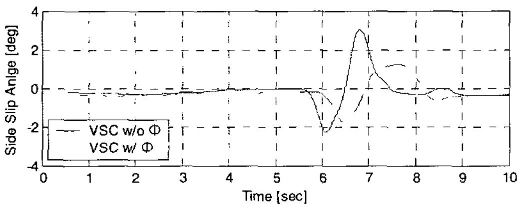

Figure 8. Side slip angle.

speed of an obstacle avoidance maneuver to the outside lane. As shown in the Figure 7, the VSC logic with bank angle estimator makes driver steer less than the conventional one. The steering input of VSC without bank angle shows excessive during first avoidance and objective lane keeping maneuver. The side slip angle of the case with bank angle is caused less than that of case without bank angle as shown in the Figure 8. Figure 9 shows that the desired and actual yaw rate. Without bank angle estimator, more yaw rate is induced than the case with bank angle (Figure 9(a) and 9(b)).

Figure 10 presents brake inputs of each case. As shown in the Figure 9(b), the case with bank angle estimate shows less yaw rate error than the case without bank angle. However, the magnitude of control inputs of both cases is similar around 900 Nm (Figure 10(a) and 10(b)). It means that the proposed controller, the case with bank angle information, magnifies the brake input to make vehicle follow the desired dynamics which include the bank angle effect.

As shown in the Figure 11, the trajectory of the proposed VSC is more reliable to avoid an obstacle than that of the conventional case.

5. CONCLUSION

While in the critical driving situation, the driver-VSC-vehicle interacts each other. A driver's misjudgment, mainly of steering input, may bring severe accident in the situation. Since a normal and average driver makes decision in quite short period, a VSC have to negotiate the situation without loosing consistency of the driver input. The bank angle of the curved road may cause inconsistency of the driver's reaction, and work as a disturbance of the driver-VSC-vehicle system. In the proposed VSC logic, bank angle estimator is applied not only to estimate

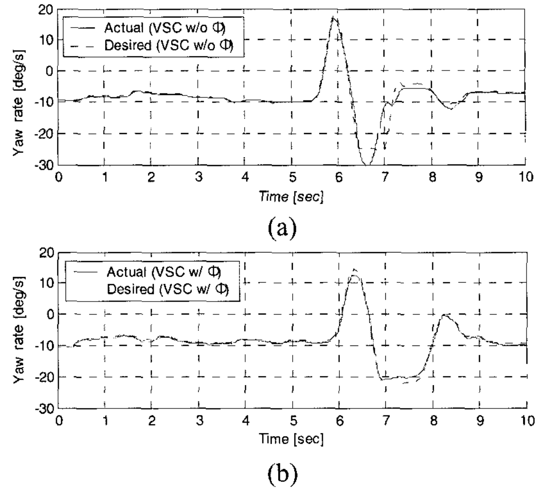

Figure 9. Yaw rate response (VSC with $\hat{\varphi}$).

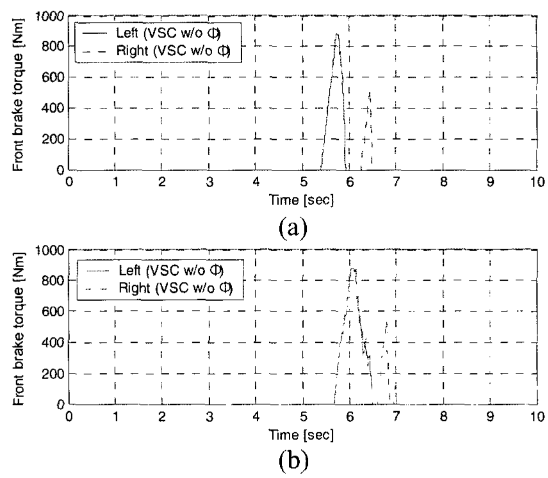

Figure 10. Brake input (VSC with $\hat{\varphi}$).

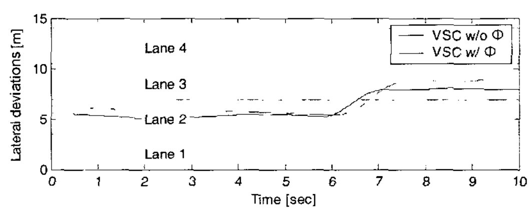

Figure 11. Vehicle trajectories.

the vehicle side slip angle but also the VSC logic itself. Thus, the dynamic bank angle estimator is needed to have robustness, and is evaluated by the vehicle tests with various driving situation. To eliminate the bank angle effects on driver-VSC-vehicle system, the VSC logic and its desired values are modified by using the bank angle estimate. The proposed controller is validated by using driving simulator, which enables more practical assessment of the VSC than an open-loop simulation. The driving simulation results of the critical driving situations, inside and outside collision avoidance while cornering, show that the proposed controller makes driver interact with the situation whether flat or bank road. The proposed controller is expected to reduce a possibility of a driver's misjudgment induced by the bank disturbance. In this point of view, the proposed controller is said to be better than the conventional VSC.

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