Body Slip Angle Estimation and Control for Electric Vehicle with In-Wheel Motors

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***Abstract* – In this paper, by making use of the important merit that electric vehicle's motor torque can be measured and controlled accurately, a body slip angle (β angle) observer based on nonlinear tire model and a novel control method of body slip angle are introduced. Experimental results by UOT March II are shown to verify the effectiveness of the proposed observer. Simulations and tests are conducted to test the control method. The torque difference can be generated directly with in-wheel motors to control body slip angle at its desired value to keep vehicle's stability.**

I. INTRODUCTION

For the solution of air pollution problems from automobiles, low emission vehicles such as electric vehicles or hybrid electric vehicles are intensely researched on. This paper focuses on designing of an electric vehicle to enhance active safety in critical driving situations.

Electric Vehicles (EVs) are inherently more suitable for active safety over Internal Combustion engine Vehicles (ICVs) [1]. First, motor's torque generation is fast and accurate, in the both directions of driving and braking. Electric motor's torque response is only several milliseconds, which is 10-100 times as fast as combustion engine's. This advantage can enable us to realize high performance control of EVs with a much shorter control period than ICVs'. Second, motor torque can be known precisely. Therefore we can easily estimate driving and braking forces between tire and road surface in real-time. This advantage can be used to realize novel control based on precise vehicle states estimation. Third, in-wheel motors can be installed in EVs' each rear and front tires, which can control the traction and braking forces independently. Based on these structural merits, vehicle motion can be stabilized by additional yaw moment generated as a result of the difference in tire driving or braking forces between the right and left side of the vehicle, which is so called 'Direct Yaw-moment Control' (DYC).

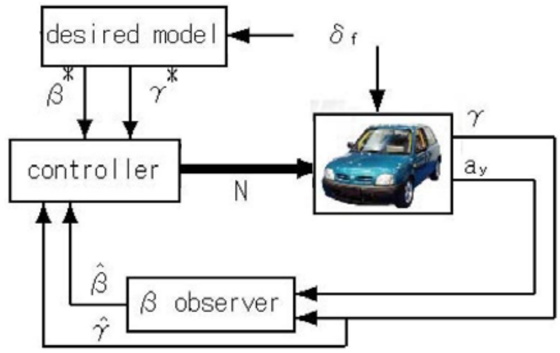


Fig.1 Vehicle stability control diagram

This paper proposes a chassis control system utilizing DYC by using the model matching control method and optimal control method. The main concept of this system is to maintain driver's handling ability at the physical limit of adhesion between the tires and the road by making the vehicle become as user friendly as it is well below that limit. The control diagram is shown as Fig.1. For the dynamics of two-freedom-linear vehicle model can describe driver's familiar characteristics under normal driving condition, vehicle motion state variables of body side slip angle β and yaw rate γ calculated from this two-freedom-linear model are taken as the desired behaviour of the vehicle [2]. With the application of model matching control technique, the control system consists of yaw moment (N) optimal decision based on state deviations feedback compensator of β and γ from their desired values.

For the state deviations feedback, the real states variable values of β and γ are needed. As sensors to measure β angle directly are very expensive, it needs to estimate β angle from only variables measurable. The most difficult for β angle estimation is that the non-linear characteristics of vehicle and uncertainty of vehicle model parameters. Especially, non-linear force characteristic of tire makes vehicle states change greatly as vehicle cornering in nonlinear area compared with that in linear area. To solve the above problems, this paper proposes a nonlinear observer for β angle estimation in which a nonlinear tire model is adopted for the observer model design. This nonlinear tire model has higher accuracy over linear tire model to describe the tire characteristics as tire slip angle becomes large in vehicle's nonlinear cornering conditions. In this paper, the observer model is also described in the form of an equivalent linear two freedom model by adopting the value of extended tire cornering power so that the linear observer design method is applied.

Simulations and field tests analysis demonstrate the effectiveness of β observer and controller.

II. BODY SLIP ANGLE OBSERVER BASED ON NONLINEAR TIRE MODEL

A. β Observer Structure and Description

The nonlinear observer structure is shown as fig.2. The estimate of β angle is computed as predicted value from states equation corrected by output feedback.

The observer's states equation is:

$$\dot{\hat{x}} = f(u, \hat{x}) - K \cdot (\hat{y} - y) \quad (1)$$

In the observer, $f(u, \hat{x})$ describes the states equation with nonlinear tire model. The observer's states variables, input variables and output variables are:

$$x=\begin{bmatrix}\beta\\ \gamma\end{bmatrix},\quad u=\begin{bmatrix}\delta_f\\ N\end{bmatrix},\quad y=\begin{bmatrix}\gamma\\ a_y\end{bmatrix}$$

where δ_f denotes steering angle of front wheel, a_y denotes vehicle lateral acceleration, N is direct yaw moment caused by differential longitudinal forces among tires, K feedback matrix of observer.

The observer's output equation is:

$$\begin{cases}\hat{\gamma}=\hat{\gamma}\\ \hat{a}_y=v(\dot{\hat{\beta}}+\hat{\gamma})\end{cases}\qquad(2)$$

where v denotes the velocity of vehicle.

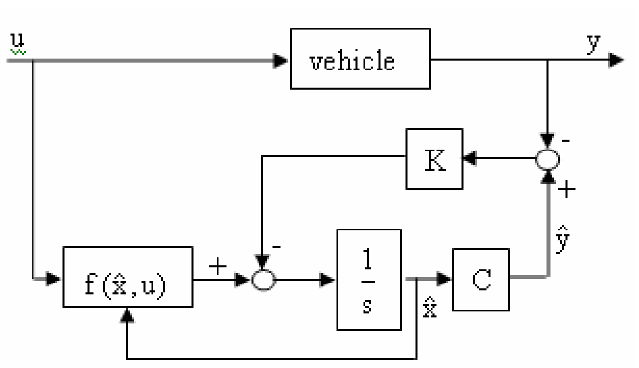


Fig.2 β observer with nonlinear tire model

The following tire model is adopted in the observer to describe the nonlinear characteristics of tire lateral force [3]:

$$F_{yi}=k_{xi}\frac{2}{\pi}\mu F_{zi}\tan^{-1}\left(\frac{\pi}{2\mu F_{zi}}C_i\alpha_i\right)\qquad(3)$$

Where F_{yi} denotes tire lateral force, α_i side slip angle of tires, C_i tire cornering stiffness, F_{zi} wheel vertical load, μ road friction coefficient, k_{xi} influence coefficient of tire longitudinal force, i index of tires.

Compared with linear tire model, this nonlinear model can describe the saturation characteristics of tire lateral force as tire slip angle gets large, and also it can reflect the influence of tire vertical load and road friction to the tire lateral force.

For simplification, a two-freedom vehicle model is adopted as observer' vehicle model (fig.3).

The dynamics of vehicle is described as:

$$\begin{cases}ma_y=F_{xf}\sin\delta_f+F_{yf}\cos\delta_f+F_{yr}\\ I_z\dot{\gamma}=l_fF_{xf}\sin\delta_f+l_fF_{yf}\cos\delta_f-l_rF_{yr}+N\end{cases}\qquad(4)$$

Where m is mass of vehicle, Iz yaw inertia moment, lf distance between mass center and front axle, lr distance between mass center and rear axle, F_{Xf} longitudinal forces of front tires, F_{Yf} and F_{Yr} lateral forces of front and rear tires which can be calculated according to above nonlinear tire model.

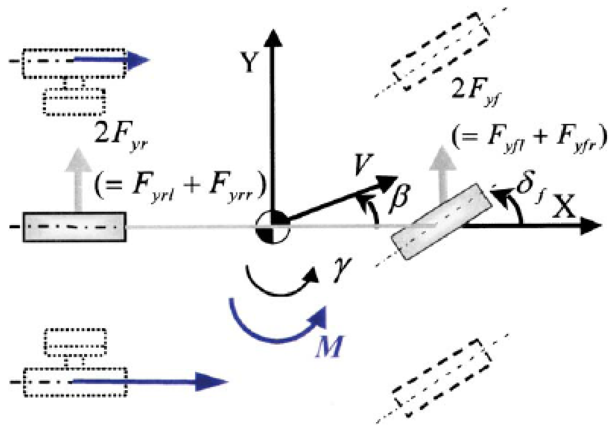


Fig.3 two-freedom vehicle model for observer design

The dynamics of vehicle is described as:

$$\begin{cases}ma_y=F_{Xf}\sin\delta_f+F_{Yf}\cos\delta_f+F_{Yf}\\ I_z\dot{\gamma}=-l_fF_{Xf}\sin\delta_fl_f+l_fF_{Yf}\cos\delta_f-l_rF_{Yr}+N\end{cases}\qquad(4)$$

Where m is mass of vehicle, Iz yaw inertia moment, lf distance between mass center and front axle, lr distance between mass center and rear axle, F_{Xf} longitudinal forces of front tires, F_{Yf} and F_{Yr} lateral forces of front and rear tires which can be calculated according to above nonlinear tire model.

Considering the kinematics relationship as equation 2 and that δ_f value is relatively small in the vehicle's high speed situations, the observer's nonlinear states equations are derived as:

$$\begin{cases}\dot{\hat{\beta}}=\dfrac{1}{mv}(F_{Yf}+F_{Yr})-\hat{\gamma}\\ \dot{\hat{\gamma}}=\dfrac{1}{I_z}(l_fF_{Yf}-l_rF_{Yr}+N)\end{cases}\qquad(5)$$

The nonlinear observer's design and application are much difficult. So this paper tries applying linear observer design method to solve the nonlinear problem. To do this, the observer model is changed into the form of an equivalent linear two freedom model by adopting the value of extended tire cornering power c'_p, which is defined as:

$$c'_p=\frac{F_y}{\alpha}\qquad(6)$$

Where F_y is the tire lateral force and slip angle α is the tire slip angle at its operating point.

When the observer running, c'_p is updated according to the calculated α value according to vehicle kinematics varibles and F_y value from tire lateral force model (3).

The design method of linear β observer refers to paper [4]. By adopting tire cornering power c'_p, the nonlinear observer model (5) can be described as an equivalent linear model (7) at each operating point:

$$\dot{x} = Ax + Bu \qquad (7)$$

In which,

$$A = \begin{bmatrix} \dfrac{-(C'_{fl} + C'_{fr} + C'_{rr} + C'_{rl})}{mv} & \dfrac{-l_f(C'_{fl} + C'_{fr}) + l_r(C'_{rl} + C'_{rr})}{mv^2} - 1 \\ \dfrac{-l_f(C'_{fl} + C'_{fr}) + l_r(C'_{rl} + C'_{rr})}{I_z} & \dfrac{-l_f^2(C'_{fl} + C'_{fr}) - l_r^2(C'_{rl} + C'_{rr})}{I_z v} \end{bmatrix}$$

$$B = \begin{bmatrix} \dfrac{C'_{fl} + C'_{fr}}{mv} & 0 \\ \dfrac{l_f(C'_{fl} + C'_{fr})}{I_z} & \dfrac{1}{I_z} \end{bmatrix}, \quad x = \begin{bmatrix} \beta \\ \gamma \end{bmatrix}, \quad u = \begin{bmatrix} \delta_f \\ N \end{bmatrix}$$

Where, $C'_{fl} \sim C'_{rr}$ are the extended cornering power values of tires.

The above equations have the same structures as the linear observer of reference paper [4]. So the same design method of gain matrix K can also be adopted.

According to [4], K is selected as following, for high response and robustness purposes.

$$K = \begin{bmatrix} \dfrac{[l_f(C'_{fr} + C'_{fl}) - l_r(C'_{rr} + C'_{rl})]\lambda_1\lambda_2 I_z}{(C'_{fr} + C'_{fl})(C'_{rr} + C'_{rl})(l_f + l_r)^2} - 1 & \dfrac{1}{v} \\ -\lambda_1 - \lambda_2 & \dfrac{m((C'_{fr} + C'_{fl})l_f^2 + (C'_{rr} + C'_{rl})l_r^2)}{[(C'_{fr} + C'_{fl})l_f - (C'_{rr} + C'_{rl})l_r]I_z} \end{bmatrix}$$

In which, λ_1 and λ_2 are the assigned pole values of the observer.

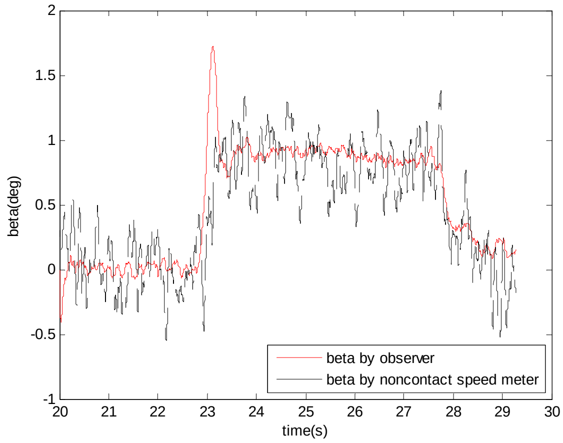
B. Experiments of β Observer

To test the proposed β observer, field tests are conducted in our experimental EV, UOT March II. UOT March II is equipped with acceleration sensor, gyro sensor and noncontact speed meter which enable us to measure real ay, γ and β values. Table.1 explains the sensors specifications.

TABLE I. Equipments for β observer field tests

Acceleration sensor	ANALOG DEVICES ADXL202
Yaw rate sensor	HITACHI OPTICAL FIBER GYROSCOPE HOFG-CLI(A)
Slip angle senser	Noncontact Optical sensor CORREVIT S-400

Results of field test shown in fig. 4 and fig.5 demonstrate the effectiveness of the nonlinear observer in both the linear and nonlinear cornering situations of the experimental vehicle.



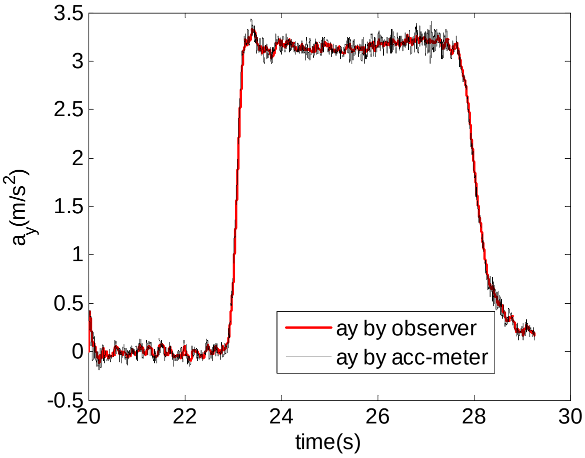
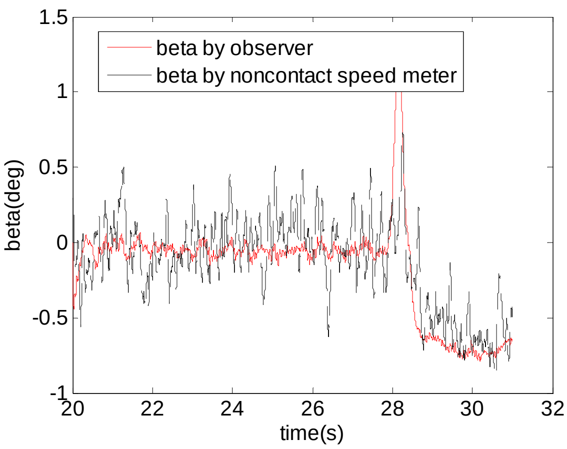


Fig.4 Field test results of slip angle observer

Linear area (driver steering angle=90° , v=40Km/h)



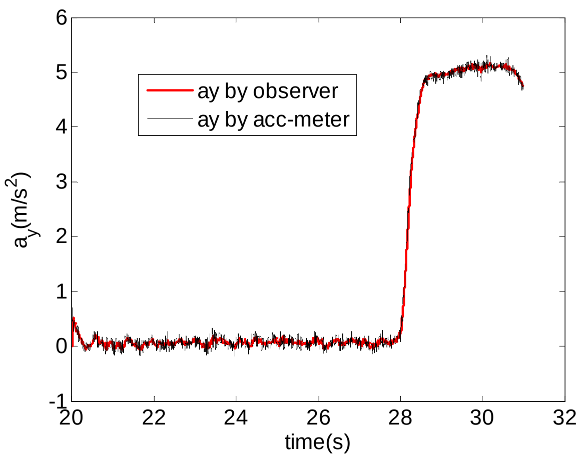


Fig.5 Field test results of slip angle observer
Nonlinear area (driver steering angle=90°, v=60Km/h)

III. BODY SLIP ANGLE CONTROL BASED ON OPTIMAL YAW MOMENT DECISION

A. Desired Model and State Deviations Equation

The desired state variables of β and γ are determined by two-freedom-linear model according to (7) as follows:

$$\begin{bmatrix} \dot{\beta}_d \\ \dot{\gamma}_d \end{bmatrix} = A\begin{bmatrix} \beta_d \\ \gamma_d \end{bmatrix} + \begin{bmatrix} b_{11} \\ b_{21} \end{bmatrix}\delta_f \qquad (8)$$

The state deviations variable between the desired value X_d and actual value X is assumed to be as follows:

$$E = X - X_d = \begin{bmatrix} \Delta\beta \\ \Delta\gamma \end{bmatrix} = \begin{bmatrix} \beta - \beta_d \\ \gamma - \gamma_d \end{bmatrix}$$

According to (7) and (8), the differentiated value of the above equation can be obtained as follows:

$$\dot{E} = \dot{X} - \dot{X}_d = A \cdot E + \begin{bmatrix} b_{12} \\ b_{22} \end{bmatrix} \cdot N \quad (9)$$

Equation (9) describes the dynamic relationship between the direct yaw moment and vehicle motion state deviations. It shows that when vehicle motion deviations appear, exerting direct yaw moment can reduce them to make vehicle restore stable.

B. Optimal Yaw Moment Decision with Changing Weight of β deviation

Based on LQR method, the optimal control input of N can be calculated by state deviations feedback as follows:

$$N^* = -k \cdot E(t) = -k_1(\beta - \beta_d) - k_2(\gamma - \gamma_d) \quad (10)$$

Where the feedback gain k_1 and k_2 are determined to minimize the following performance index:

$$J = \frac{1}{2}\int_0^\infty [q_1\Delta\beta^2(t) + q_2\Delta\gamma^2(t) + N(t)]dt \quad (11)$$

Where q_1 and q_2 are the weighting coefficients of state deviations. For the choice of q_1 and q_2, the coefficient ω_β ($0 \le \omega_\beta \le 1$) is introduced to describe the weighting of performance index on β deviation. Define $q_1 = q^2\omega_\beta$ and $q_2 = q^2(1-\omega_\beta)$, (11) can be rewritten as:

$$J = \frac{q^2}{2}\int_0^\infty [\omega_\beta\Delta\beta^2(t) + (1-\omega_\beta)\Delta\gamma^2(t) + N(t)]dt \quad (12)$$

To get high vehicle stability, when β is small, γ matching control is more important. Otherwise, when β gets larger, β control becomes more important. In addition, vehicle stability is more sensitive to β deviation under low adhesion road conditions than that under high adhesion road conditions. Therefore, ω_β should be change with β value and road friction coefficient μ which is chosen as follows:

$$\omega_\beta = \begin{cases} \dfrac{|\beta|}{\mu \cdot \beta_0} & \text{if } |\beta| < \mu \cdot \beta_0 \\ 1 & \text{else} \end{cases} \quad (13)$$

Where β_0 is a threshold value with value of 10° by experience.

The changing of ω_β according to β is shown as fig. 6.

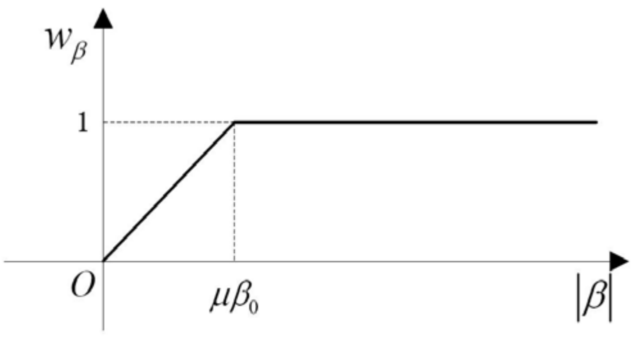


Fig.6 weight of body slip angle deviation for optimal yaw moment decision

B. Simulation Results of Body Slip angle controlling

Fig.7 is are the simulation results with sinusoidal front steering angle input when the road friction coefficient is 0.2 and vehicle velocity is 30m/s. This can represent the critical driving situation of continuous lane change maneuver on snow-covered road. With β control, yaw rate γ can match the desired value well and the body slip angle β is limited. In the case without β control, β gets so large that the vehicle loses stability and the driver can not accomplish the lane change as normal (fig. 8).

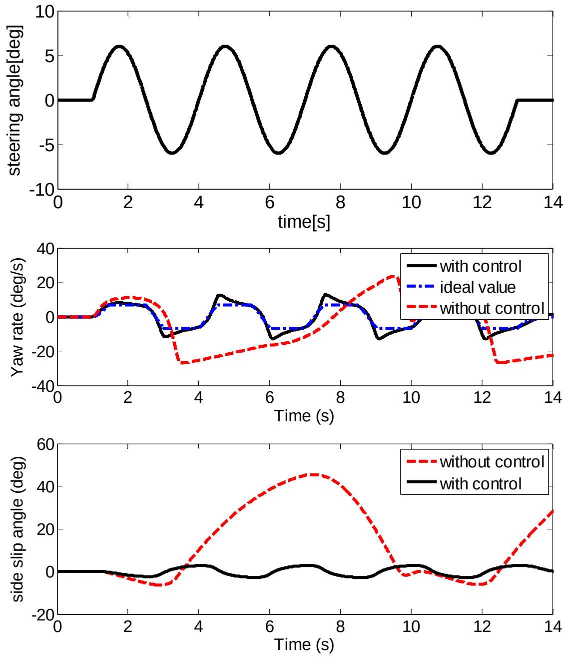


Fig.7 simulation results of β control

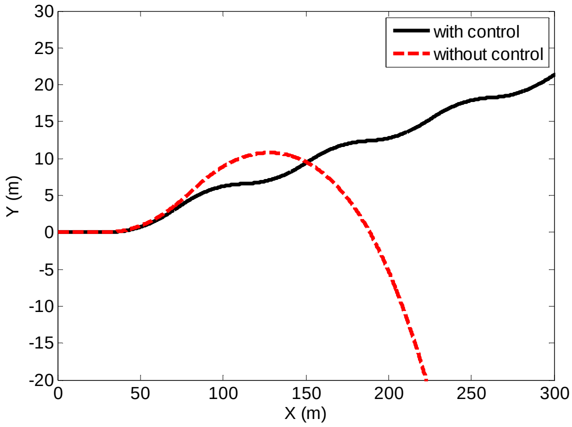


Fig.8 simulation results of vehicle trajectory

Fig.9 is the β-γ phase plane trajectory of the simulation results. With β control, a limit loop comes into being within the stable area. Without β control, the trajectory of β-γ phase plane is much larger to be out of the stable area.

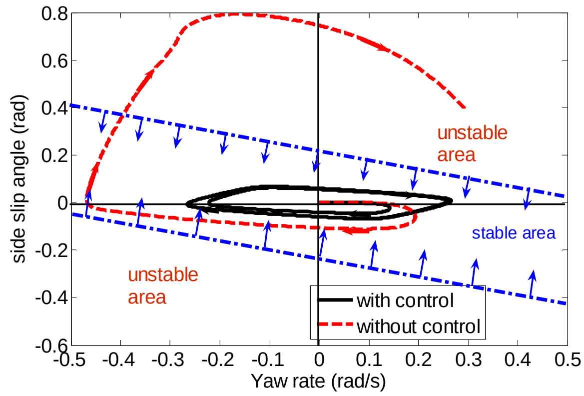


Fig.9 β-γ phase plane trajectory of the simulation results

IV. CONCLUSION

In this paper, we proposed novel methods to estimate β and to control β for EVs, which utilizes the direct yaw moment to maintain vehicle stability by making the full inherent advantage of motor-driven vehicle. The improved observer is based on nonlinear tire model and robust feedback gain design. The results of experiments by using UOT March II demonstrated that the proposed observer could estimate β exactly and robustly. The DYC is utilized by using the model matching control method and optimal control method. Vehicle motion state variables of body side slip angle β and yaw rate γ calculated from this two-freedom-linear model are taken as the desired behaviour of the vehicle. With the application of model matching control technique, the yaw moment (N) is optimally decided based on state deviations feedback compensator of β and γ from their desired values. Simulation studies clarify the control method can enhance the stability during critical driving situations.

V. REFERENCES

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