Estimation of Vehicle Sideslip Angle for Four–Wheel Steering Passenger Cars

Hwan–Seong Kim and Sam–Sang You

Abstract: This paper deals with an estimation method for sideslip angle by using an unknown disturbance observation technique in 4WS passenger car systems. Firstly, a 4WS vehicle model with 3DOF is derived under the constant velocity and same tyre's properties. The vehicle dynamics is transformed into the linear state space model with considering the external disturbances. Secondly, an unknown disturbance observer is introduced and its property which estimating the states of system without any disturbance information is shown. Lastly, the estimated sideslip angle of the 4WS vehicle system is verified through numerical simulation.

Keywords: sideslip angle, 4WS, steering angle, unknown disturbance, observer

I. Introduction

The research and development of vehicle control systems for highway automation have been focused over recent decades [1]-[12]. Among these control fields, the vehicle steering control is most basic control for improving ride comfort and manoeuvrability [2]-[5].

In vehicle steering control fields, the simplest model, which leads to a fundamental understanding of vehicle handling, has two degrees of freedom(2DOF): the lateral and yaw velocities as state variables [6]-[10].

The vehicle systems use the vehicle body sideslip angle as the state variable for control purposes. It is possible to measure the sideslip angle directly by using dedicated measuring devices capable of performing optical measurements. Sometimes it is impractical to use this hardware in control system for resasons of cost, accuracy, or avilability. Thus, it is necessary to estimate the sideslip angle on the basis of information obtained with sensor for detecting such parameters as yaw rate or lateral acceleration.

With 2 DOF vehicle model a few sideslip estimation methods are proposed, and the estimated value is applied to improve the direct yaw moment control. In 1997, an estimation method for lateral vehicle dynamics using non-linear observer was presented [6]. A combined method which is used by model observer and direct integration is proposed where two kind of values of the side force of the wheels are assumed [7]. An estimation method for vehicle position is developed by using the Kalman filter and disturbance observer [8]. An adaptive observer for estimating the vehicle body slip angle is proposed [9]. Also the sliding mode control theory applies to improving the direct yaw moment control, where the sideslip angle, and the road fraction are estimated by using conventional observer [10].

However, it is difficult to express complex vehicle motions accurately with a 2DOF model. It is also well known that one way to enhance the steering performance is a four-wheel steering(4WS) system [11][12]. The 4WS systems for automobiles are being widely studied as a means of improving stability and handling characteristics and have been the subject of various investigations. For estimating the sideslip angle in 4WS vehicle systems, an adaptive yaw–rate feedback control system which involves a tyre and road fraction estimator is presented [11]. A robust active rear steering system is presented, where the side slip angle is estimated using conventional observer with frequency filter [12].

In this paper, we propose an estimation method for sideslip angle by using an unknown disturbance observation technique in 4WS passenger cars. Firstly, a 4WS vehicle model with 3DOF is derived under the constant velocity and same tyre's properties. The vehicle dynamics is transformed into the linear state space model with external disturbances. Secondly, an unknown disturbance observer is introduced and its properties are shown. As one of the properties, the states of system can be estimated without any disturbance information. Lastly, the estimated sideslip angle of 4WS systems is verified through numerical simulations by using proposed estimation method.

II. 4WS Vehicle model

Generally, the 4WS vehicle model includes steering and roll motion, tyre dynamics, and drive train. With those dynamics and motions, a 4WS system with 3DOF is modeled by planar model, and the simplified roll motion is given in Fig. 1 and 2, respectively [4].

To obtain the vehicle's dynamic equations, we assume the following properties:

A1) The vehicle is symmetrical about the $x - z$ plane.
A2) Total vehicle mass m_v is lumped.
A3) The roll axis is fixed with $m_{uf}d_f \approx m_{ur}d_r$
A4) The road surface conditions are the same.
A5) The small angle approximations of the vehicle motion apply.

The dynamics of 4WS vehicle system are divided into three parts, *i.e.*, tyre side forces, yaw moments and roll moments, and those equations are described by Eqs. (1)-(3)

$$Y_r = \Sigma \text{ tyre side forces}$$

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Hwan–Seong Kim: Dept. of Logistics Eng., Korea Maritime University, KOREA (kimhs@hanara.kmaritime.ac.kr)

Sam–Sang You : School of Mechanical & Information Eng., Korea Maritime University, KOREA(ssyou@hanara.kmaritime.ac.kr)

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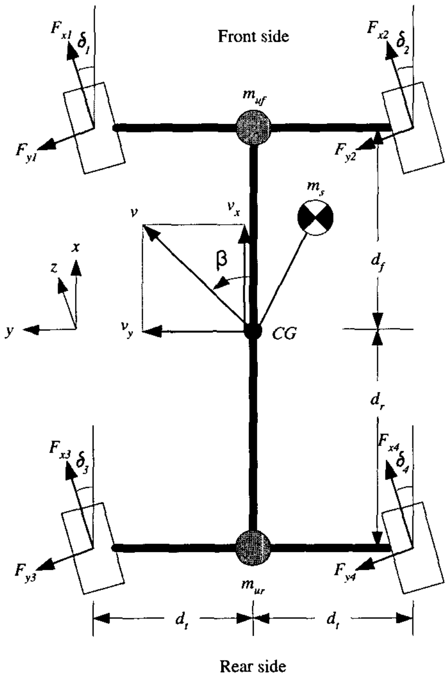

Fig. 1. Planar model of 4WS vehicle system.

$$
\begin{aligned}
&= m_v(\dot{v}_y + v_x r) + m_s h_s \dot{p} \\
&= Y_{Fi} + w_y \qquad (1) \\
M_z &= \Sigma \text{ yaw moments} \\
&= J_{zz}\dot{r} - J_{xz}\dot{p} \\
&= d_r(Y_{F3} + Y_{F4}) - d_f(Y_{F1} + Y_{F2}) + w_z \\
&\quad + d_t(X_{F2} + X_{F4}) - d_t(X_{F1} + X_{F3}) \qquad (2) \\
L_x &= \Sigma \text{ sprung mass roll moments} \\
&= J_{xx}\dot{p} - J_{xz}\dot{r} \\
&= -C_\phi p - (\kappa_\phi - m_s g h_s)\phi \\
&\quad + m_s h_s v(\dot{\beta} + r)\cos\phi + w_x \qquad (3)
\end{aligned}
$$

where,

$$
\begin{bmatrix} X_{Fi} \\ Y_{Fi} \end{bmatrix} = \begin{bmatrix} \cos\delta_i & -\sin\delta_i \\ \sin\delta_i & \cos\delta_i \end{bmatrix} \begin{bmatrix} F_{xi} \\ F_{yi} \end{bmatrix}, i = 1, \cdots, 4
$$

$C_\phi = C_{\phi f} + C_{\phi r}$, $\kappa_\phi = \kappa_{\phi f} + \kappa_{\phi r}$.

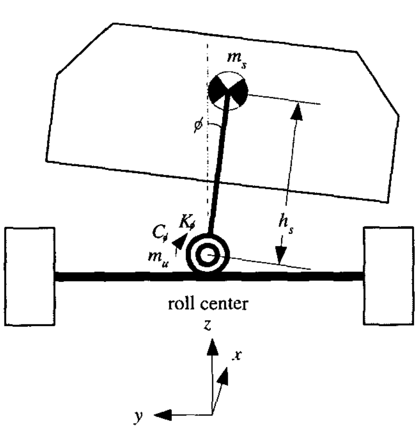

Fig. 2. Schematic diagram of the simplified roll model.

The variables(w_y, w_z, w_x) are the disturbance force and torques in the lateral, yaw and roll directions which result, for example, from the drag effect, loads, side wind gusts, a flat tyre, braking on ice, an uneven road, modelling uncertainties, etc., which are considered as an external disturbances. Under certain circumstances, these external disturbances may exert a significant influence on lateral/directional and roll handling.

The longitudinal forces, F_{xi}, are related with wheel rotational model. These forces applied to the four wheels from the engine and/or braking are given as

$$J_t \dot{\omega}_{ti} = \pm \tau_{bi} \mp R_t F_{xi}, \; i = 1, \cdots, 4, \qquad (4)$$

where J_t is the effective rotational inertia which includes all drivetrain effects.

And the lateral forces, F_{yi}, are nonlinear functions given by "magic formula"[3] as

$$
\begin{aligned}
F_{yi} &= a_1 \sin(a_2 \arctan\{a_3(1 - a_4)(\alpha + a_5) \\
&\quad + a_4 \arctan[a_3(\alpha + a_5)]\}) + a_6, \qquad (5)
\end{aligned}
$$

where $a_i (i = 1, \cdots, 6)$ are six coefficients which depend on the vehicle load F_z and on camber angle γ, and α.

To study the dynamic behavior of vehicle under the assumption of small sideslip angle in normal driving condition, Eq.(5) can be linearized as follows:

$$F_{yi} = -C_f \alpha_f, \; i = 1, 2 \qquad (6)$$

$$F_{yi} = -C_r \alpha_r, \; i = 3, 4. \qquad (7)$$

The tire slip angles can be written as

$$
\begin{bmatrix} \alpha_f \\ \alpha_r \end{bmatrix} = v \begin{bmatrix} 1 + \dfrac{d_f}{v} \\ 1 - \dfrac{d_r}{v} \end{bmatrix} \begin{bmatrix} \beta \\ r \end{bmatrix} - \begin{bmatrix} \delta_f \\ \delta_r \end{bmatrix} - \begin{bmatrix} R_f \\ R_r \end{bmatrix} \phi \qquad (8)
$$

With a small roll angle or $\sin\phi \approx \phi$ and $\cos\phi = 1$, the above equations can be linearized as follows:

$$
\begin{aligned}
Y_r &= m_v(\dot{v}_y + v_x r) + m_s h_s \dot{p} \\
&= m_v v(\dot{\beta} + r) + m_s h_s \dot{p} \\
&= -2C_f(\beta + \frac{d_f}{v} r - \delta_f - R_f \phi) \\
&\quad -2C_r(\beta - \frac{d_r}{v} r - \delta_r - R_r \phi) + w_y \qquad (9) \\
M_z &= J_{zz}\dot{r} - J_{xz}\dot{p} \\
&= -2d_f C_f(\beta + \frac{d_f}{v} r - \delta_f - R_f \phi) \\
&\quad +2d_r C_r(\beta - \frac{d_r}{v} r - \delta_r - R_r \phi) + w_z \qquad (10) \\
L_x &= J_{xx}\dot{p} - J_{xz}\dot{r} \\
&= -C_\phi p - (\kappa_\phi - m_s g h_s)\phi + m_s h_s v(\dot{\beta} + r) + w_x . (11)
\end{aligned}
$$

At constant velocity and steady-state condition with $\dot{\omega}_{ti} = 0$, the longitudal force F_{xi} for all wheel break in Eq.(4) is approximated as

$$F_{xi} = \frac{\tau_{bi}}{R_t}, \; i = 1, \cdots, 4. \qquad (12)$$

Also, the actuator dynamics are represented as linear first-order lag systems:

$$b_f \dot{\delta}_f = \delta_{sf} - \delta_f \qquad (13)$$

$$b_f \dot{\delta}_r = \delta_{sr} - \delta_r \qquad (14)$$

where $b_f = \frac{C_{sf}}{\kappa_{sf}}$ and $b_r = \frac{C_{sr}}{\kappa_{sr}}$.

By using the above dynamics equations (6) – (14), a descriptor system form is obtained as

$$E\dot{x} = \tilde{A}x + \tilde{B}u + \tilde{D}w \quad (15)$$

$$y = Cx \quad (16)$$

where,

$$E = \begin{bmatrix} m_v v & 0 & m_s h_s & 0 & 0 & 0 \\ 0 & J_{zz} & -J_{xz} & 0 & 0 & 0 \\ -m_s h_s v & -J_{xz} & J_{xx} & 0 & 0 & 0 \\ 0 & 0 & 0 & 1 & 0 & 0 \\ 0 & 0 & 0 & 0 & C_{sf} & 0 \\ 0 & 0 & 0 & 0 & 0 & C_{sr} \end{bmatrix},$$

$$\tilde{A} = \begin{bmatrix} a_{11} & a_{12} & 0 & a_{14} & 2C_f & 2C_r \\ a_{21} & a_{22} & 0 & a_{24} & 2C_f d_f & -2C_r d_r \\ 0 & m_s h_s v & -C_\phi & a_{34} & 0 & 0 \\ 0 & 0 & 1 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & -\kappa_{sf} & 0 \\ 0 & 0 & 0 & 0 & 0 & -\kappa_{sr} \end{bmatrix},$$

$$\tilde{B} = \begin{bmatrix} 0 & 0 \\ 0 & 0 \\ 0 & 0 \\ 0 & 0 \\ \kappa_{sf} & 0 \\ 0 & \kappa_{sr} \end{bmatrix}, \tilde{D} = \begin{bmatrix} 1 & 0 & 0 \\ 0 & 1 & 0 \\ 0 & 0 & 1 \\ 0 & 0 & 0 \\ 0 & 0 & 0 \\ 0 & 0 & 0 \end{bmatrix}, x = \begin{bmatrix} \beta \\ r \\ p \\ \phi \\ \delta_f \\ \delta_r \end{bmatrix},$$

$$u = \begin{bmatrix} \delta_{sf} \\ \delta_{sr} \end{bmatrix}, w = \begin{bmatrix} w_y \\ w_z \\ w_x \end{bmatrix}, a_{11} = -2(C_f + C_r),$$

$$a_{12} = \left(2\frac{C_r d_r}{v} - 2\frac{C_f d_f}{v} - m_v v\right),$$

$$a_{14} = 2(C_f R_f + C_r R_r), a_{21} = 2(C_r d_r - C_f d_f),$$

$$a_{22} = -2\left(\frac{C_f d_f^2}{v} + \frac{C_r d_r^2}{v}\right), a_{34} = (m_s g h_s - \kappa_\phi),$$

$$a_{24} = 2(C_f d_f R_f - C_r d_r R_r).$$

Using the matrix inverse operation, we obtain a linear time–invariant system described by

$$\Sigma : \begin{cases} \dot{x} = Ax + Bu + Dw \\ y = Cx \end{cases} \quad (17)$$

where $x \in R^n$, $u \in R^q$, $w \in R^m$ and $y \in R^p$ are the state vector, the known input vector, the unknown disturbance vector, and the output vector of system, respectively.

III. Unknown disturbance observer

For constructing the unknown disturbance observer, we assume that $p \geq m$ and without loss of generality, rank $D = m$ and rank $C = p$.

Consider a full-order observer described by

$$\Sigma_{UD} : \begin{cases} \dot{z} = \hat{A}z + \hat{B}y + \hat{J}u \\ \hat{x} = \hat{C}z + \hat{D}y \end{cases} \quad (18)$$

where $z \in R^n$ and $\hat{x} \in R^n$ denote a transformed estimated vector and an estimated state vector, respectively. $\hat{A}$, $\hat{B}$, $\hat{C}$, $\hat{D}$, and $\hat{J}$ are unknown matrices of appropriate dimensions.

Definition 1: The system Σ_{UD} is said to be an unknown disturbance full order observer for the linear systems Σ if and only if

$$\lim_{t \to \infty} e(t) = 0, \ \forall x(0_-), z(0_-), u(\cdot)$$

where $e(t) = \hat{x}(t) - x(t)$ represents the observer error.

Define an estimation error as

$$\xi = z - Ux. \quad (19)$$

Then, the dynamics of observation error is given by

$$\begin{aligned} \dot{\xi} &= \dot{z} + U\dot{x} \\ &= \hat{A}\xi + (\hat{A}U + \hat{B}C - UA)x \\ &\quad + (\hat{J} - UB)u - UDv \quad (20) \\ \hat{x} &= \hat{C}\xi + (\hat{C}U - \hat{D}C)x. \quad (21) \end{aligned}$$

If there exists a matrix U which satisfies the following conditions:

$$\hat{A}U + \hat{D}C = UA$$
$$\hat{J} = UB$$
$$UD = 0$$
$$\hat{C}U - \hat{D}C = I_n,$$

then, (20) and (21) are rewritten as

$$\dot{\xi} = \hat{A}\xi \quad (22)$$

$$\hat{x} = \hat{C}\xi + x. \quad (23)$$

From the above equations, if $\hat{A}$ is stable, then $\xi \to 0(\mathrm{t} \to \infty)$ and $x - \hat{x} = 0$.

Thus, the system Σ_{UD} is an unknown disturbance full order observer for the system Σ with unknown disturbance vector. From the above statements, we obtain the following theorem.

Theorem 1: [13]: The system Σ_{UD} is an unknown disturbance observer for the system Σ with unknown disturbance vector, if $\hat{A}$ is stable and there exists a matrix $U \in R^n$ which satisfies the following conditions:

$$\hat{A}U + \hat{D}C = UA \quad (24)$$

$$\hat{J} = UB \quad (25)$$

$$UD = 0 \quad (26)$$

$$\hat{C}U + \hat{D}C = I_n \quad (27)$$

■

Here let $\hat{C} = I_n$ for simplicity. Then, from (27), we obtain

$$U = I_n - \hat{D}C \quad (28)$$

By substitution of (28) into (24), we have

$$\hat{A} = UA - KC \quad (29)$$

$$\hat{B} = \hat{A}\hat{D} + K \quad (30)$$

where $K = \hat{B} - \hat{A}\hat{D}$.

Furthermore substituting (28) into (26), we have

$$\hat{D}CD = D. \quad (31)$$

In order to guarantee the matrix $\hat{D}$ satisfying (31), the following condition should be hold,

$$\text{rank}\, CD = \text{rank}\, D = m. \tag{32}$$

The condition (32) requires that $p \geq m$, *i.e.*, the number of measured output must be greater than or equal to that of the external disturbance input.

The general solution of (31) can be obtained as

$$\hat{D} = D(CD)^{+} + G(I_p - CD(CD)^{+}) \tag{33}$$

where the superscript + indicates the generalized inversion and G is an arbitrary matrix.

By substituting (33) into (28), we can get

$$U = (I_n - GC)\left\{I_n - D(CD)^{+}C\right\}. \tag{34}$$

From the above equation, there exists a matrix G, which makes $(I_n - GC)$ nonsingular, and then the rank $U = n - m$.

Since rank $D = m$, there exists the left-inverse of matrix D, *i.e.*, $D^{+}D = I_m$.

Under the condition of rank $U = n - m$, we have Ker $U \cap$ Ker $D^{+} = 0$, *i.e.*,

$$\text{rank}\begin{bmatrix} U \\ D^{+} \end{bmatrix} = n. \tag{35}$$

Then, we have the following relation

$$\text{rank}\begin{bmatrix} sI_n - A & D \\ C & 0 \end{bmatrix} = \text{rank}\begin{bmatrix} sI_n - UA \\ C \end{bmatrix} + m. \tag{36}$$

Consequently, for $\forall s \in \mathbf{C}$, where $\mathbf{C}$ denotes the complex space,

$$\text{rank}\begin{bmatrix} sI_n - A & D \\ C & 0 \end{bmatrix} = n + m, \forall s \in \mathbf{C}, \tag{37}$$

which means that the invariant zeros of the system $(A, D, C, 0)$ must be stable. From the above statements, we summarize the following theorem.

Theorem 2: The unknown disturbance full order observer Σ_{UD} for the system Σ can be realized if

$$\text{i) rank}\, CD = \text{rank}\, D = m \tag{38}$$

$$\text{ii) rank}\begin{bmatrix} sI_n - A & D \\ C & 0 \end{bmatrix} = n + m, \forall s \in \mathbf{C}. \tag{39}$$

For constructing an unknown disturbance observer, based on Theorem 2, the 4WS vehicle system requires that the number of sensor output should be great or equal to that of disturbance input and the invariant zeros of system shoud be stable.

IV. Simulation results

To show the effectiveness of the proposed estimation method, we consider a model parameters for a typical passenger vehicle in Table 1.

When considering the two sensors, or roll and yaw–rate sensors, Eq.(38) is not guaranteed in this system. Because the number of sensor is less than the number of disturbance input.

Table 1. Parameters of typical passenger vehicle.

$m_v = 1067$ [kg]	$C_f = 55000$ [N/rad]
$m_s = 900$ [kg]	$C_r = 45000$ [N/rad]
$m_{uf} = 100$ [kg]	$C_{\phi f} = 1100$ [Nms/rad]
$m_{ur} = 67$ [kg]	$C_{\phi r} = 1000$ [Nms/rad]
$g = 9.81$ [m/s^2]	$C_\phi = 2100$ [Nms/rad]
$v = 60$ [km/h]	$\kappa_{\phi f} = 15450$ [Nms/rad]
$h_s = 0.55$ [m]	$\kappa_{\phi r} = 15450$ [Nms/rad]
$d_f = 1.0$ [m]	$\kappa_\phi = 65690$ [Nms/rad]
$d_r = 1.5$ [m]	$J_{zz} = 2130$ [kgm^2]
$d_t = 0.65$ [m]	$J_{xx} = 500$ [kgm^2]
$b_f = 0.122$ [s]	$J_{xz} = 4750$ [kgm^2]
$C_{sf} = 10$ [Nms/rad]	$R_f = -0.17$
$C_{sr} = 13$ [Nms/rad]	$R_r = 0.15$
$\kappa_{sf} = 82$ [Nm/rad]	$R_t = 0.33$ [m]
$\kappa_{sr} = 80$ [Nm/rad]	$b_r = 0.1625$ [s]
$J_t = 0.07$ [kg m^2]	$\gamma = 5$ [deg]
$a_1 = 2321.7$	$a2 = 1.6929$
$a_3 = 0.2292$	$a4 = 0.225$
$a_5 = 0.0911$	$a6 = 3317$

Let us assume the disturbance input $w_y = 0$, then the disturbance matrix $\tilde{D}$ can be rewritten as

$$\tilde{D} = \begin{bmatrix} 0 & 0 & 1 & 0 & 0 & 0 \\ 0 & 1 & 0 & 0 & 0 & 0 \end{bmatrix}^T.$$

Thus, the conditions of theorem 2 are all satisfied, and the states of vehicle system can be estimated by using unknown disturbance observer.

From Theorem 1, the unknown disturbance observer is designed, and the eigenvalues of the $\hat{A}$ are given by using LQG method as

$$\lambda_i = \{-11.376 \quad -5.827 \pm 7.311i \quad -7.390 \quad -2.268 \quad -3.046 \pm 5.277i \quad -1.2825\}.$$

For tire dynamic simulation, we obtain the lateral forces, F_{yi}, by using "magic formula" [3], and it is given by Fig. 3.

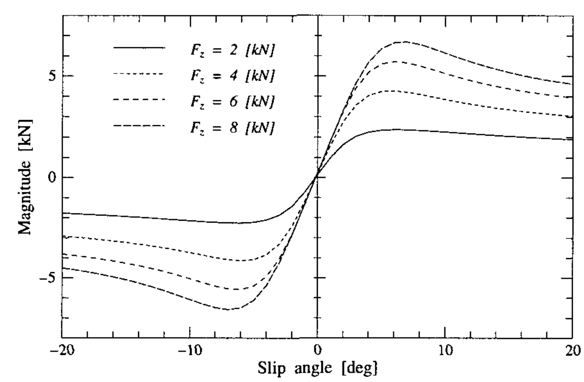

Fig. 3. Nonlinear lateral force curves $F_y(\alpha)$ obtained by using "magic formular".

In this simulation, the sampling time is 0.001[sec] and the initial state values are assumed zero. Also, we assume that the steering angle is changed into 0.2, -0.2 and 0 radians at 0, 3, and

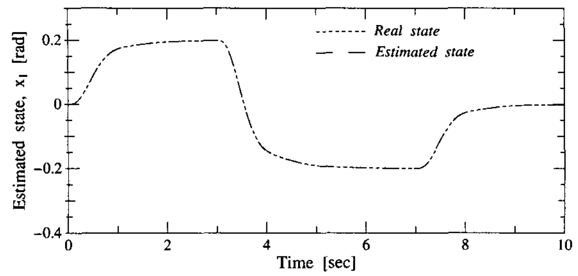

Fig. 4. Estimated sideslip angle, x_1 in case of non–linear lateral forces.

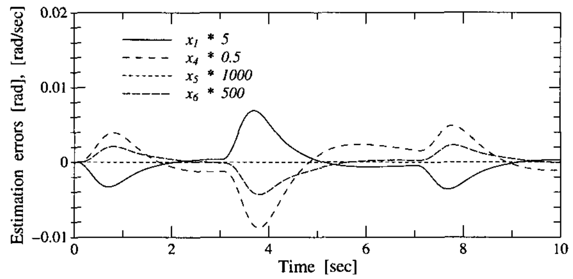

Fig. 5. Estimated states errors of x_1, x_4, x_5 and x_6 in case of non–linear lateral forces.

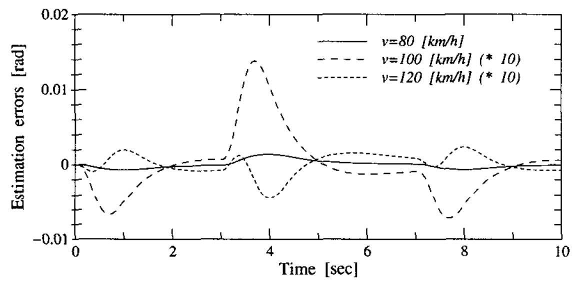

Fig. 6. Estimated errors of sideslip angle on vehicle velocity change in case of non–linear lateral forces.

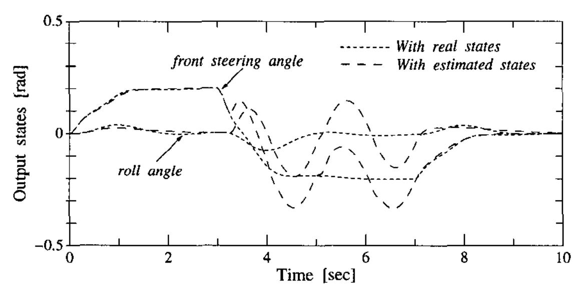

Fig. 7. Controlled output variables.

6 seconds with unknown input disturbance, respectively, as

$$w = \begin{bmatrix} w_z \\ w_x \end{bmatrix} = \begin{bmatrix} 0.1 \\ 0.1 \end{bmatrix}.$$

The estimated sideslip angle and each estimated state errors are shown in Fig. 4 and 5, respectively. From those results, we verified that the estimated error of sideslip angle is within 5% of real state value. The estimation errors of sideslip angle for varying the vehicle velocities on 80, 100, and 120 [km/h] are shown in Fig. 6

To verify the applicability of the estimated state vector for vehicle steering control, a robust servo controller is now designed. The input of servo controller is given as

$$\begin{aligned} u &= K_1 x + K_2 \int_0^t (ref_y - y)dt \\ &\approx K_1 \hat{x} + \int_0^t (ref_y - y)dt \end{aligned}$$

where ref_y denotes the reference of output, and K_1 and K_2 represent the state feedback gain and integral gains, respectively.

$$K_1 = \begin{bmatrix} -0.755 & 0.191 & -0.496 & -3.257 \\ -0.463 & -0.235 & 0.743 & 3.665 \end{bmatrix.}$$

$$\begin{matrix} 1.146 & -0.927 \\ -0.695 & 1.739 \end{matrix} \Bigg] ,$$

$$K_2 = \begin{bmatrix} -3.669 & 5.810 \\ 2.159 & 4.030 \end{bmatrix}.$$

Fig. 7 shows that the performances of robust servo controller with real state vector and estimated state vector are almost same under small sideslip angle. However, the performance is generally deteriorated at large sideslip angle. We need other robust control approaches, such as H_∞ theory etc..

V. Conclusions

In this paper, we have proposed an estimation method of vehicle sideslip angle by using unknown disturbance observation technique for four wheel steering passenger cars. With lateral, yaw and roll motions, a 3 DOF model for 4WS vehicle systems has been presented, and using the model an estimation method of sideslip angle is proposed. Also, the estimated sideslip angle is verified by comparing the real state value, and a robust servo controller can be constructed by the estimated states.

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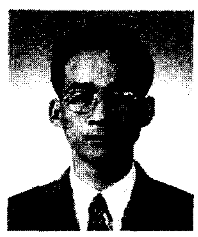
Nomenclature

b_f, b_r	time constant of (front, rear) steering actuator; $b_f = C_{sf}/\kappa_{sf}$, $b_r = C_{sr}/\kappa_{sr}$
C_f, C_r	(front, rear) tyre cornering stiffness
C_{sf}, C_{sr}.	steering stiffness of (front, rear) actuator
$C_{\phi f}, C_{\phi r}$	(front, rear) roll damping
d_f, d_r	distance from (front, rear) axis to vehicle center of gravity (CG)
F_{xi}, F_{yi}	longitudinal and lateral forces, $i \in \{1,4\}$ $(F_{xi} \approx \tau_{bi}/R_t)$
F_z	load force
g	gravitational acceleration
h_s	height of sprung mass CG on roll axis
J_{xx}	roll moment of inertia of the sprung mass above roll axis
J_{xz}	product of inertia of the sprung mass about the roll and yaw axes
J_{zz}	principal yaw moment of inertia
m_s	sprung mass
m_{uf}, m_{ur}	(front, rear) unsprung mass
m_v	total lumped mass = $m_s + m_{uf} + m_{ur}$
p	angular velocity of the roll
r	angular velocity of the yaw
R_t	wheel rolling radius
v	velocity of vehicle = $\sqrt{v_x^2 + v_y^2}$
v_x, v_y	velocity of (x, y) axes
α_f, α_r	slip angle of the (front, rear) tyre
β	sideslip angle
δ_f, δ_r	steering angle of the (front, rear) tyre
δ_{sf}, δ_{sr}	commend steering angle of the (front, rear) tyre
κ_{sf}, κ_{sr}	rotary compliances of the (front, rear) steering actuator
$\kappa_{\phi f}, \kappa_{\phi r}$	roll stiffness of the (front, rear)
ϕ	roll angle
τ_{bi}	traction torque at the ith wheel

Hwan–Seong Kim
He was born in Koera on June 13, 1965. He received the BS and MS degrees from National Fisheries University of Pusan, Korea, in 1991 and 1993, respectively. He received the Ph.D. degree from Kumamoto University, Kumamoto, Japan, in 1996. He is now an assistant professor with the department of logistics engineering, Korea Maritime University, Korea. His main research interests are in the areas of observer, fault detection and isolation, and industrial application.



Sam–Sang You
He was born in Korea on June 16, 1958. He received the BS from National Fisheries University of Pusan, Korea, in 1983. He received the MS degree from University of Wisconsin-Madison, USA, in 1990. He received the Ph.D. degree from Iowa State University, USA, in 1994. He is now an assistant professor with the department of mechanical and information engineering, Korea Maritime University, Korea. His main research interests are in the areas of robotics, vehicle dynamics , and robust control.