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자동 차선 유지 시스템의 전기식 파워 조향 시스템을 위한 슬라이딩 모드 제어기

Sliding Mode Control for an Electric Power Steering System in an Autonomous Lane Keeping System

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Abstract: In this paper, we develop a sliding mode control for steering wheel angle control based on torque overlay in order to resolve the problem of previous methods for Electric Power Steering (EPS) systems in the Lane Keeping System (LKS) of autonomous vehicles. For the controller design, we propose a 2nd order model of the electric power steering system in an autonomous LKS. The desired state model is designed to prevent a rapid change of the steering wheel angle. The sliding mode steering wheel angle controller is developed for the robustness of the disturbance. Since the proposed method is designed based on torque overlay, torque integration with basic functions of the EPS system for the steering wheel angle control is available for the driver's convenience. The performance of the proposed method was validated via experiments.

Keywords: electric power steering, torque overlay, sliding mode control

NOMENCLATURE

θ_h : Steering wheel angular position [rad]
θ_r : Reference steering wheel angular position [rad]
θ_p : Pinion angular position [rad]
θ_m : Motor angular position [rad]
x_r : Rack bar position [m]
i : Current input of the motor [A]
T : Input torque of EPS system ($T = K_t i$) [N·m]
τ_l : Load torque [N/m]
τ_p : Pinion torque [N/m]
K_t : Motor torque constant [N·m/A]
T_d : Driver's torque [N·m]
T_f : Friction torque [N·m]
T_r : Road reaction torque on the rack and pinion [N·m]
J_c : Steering column moment of inertia [Kg·m2]
B_c : Steering column viscous damping [N·m/(rad/s)]
K_c : Steering column stiffness [N·m/rad]
M_r : Mass of the rack [kg]
B_r : Viscous damping of the rack [N·m/(rad/s)]
R_p : Steering column pinion radius [m]
K_r : Tire spring rate [N/m]
J_m : Motor moment of inertia [kg·m2]
B_m : Motor shaft viscous damping [N·m/(rad/s)]
N : Motor gear ratio

I. INTRODUCTION

Autonomous vehicles are an important issue to enhance the safety and convenience of the driver in the automotive industry. Especially, longitudinal control and lateral control are main issues in the view point of vehicle motion control. The longitudinal control is studied for the vehicle following that desires to keep an appropriate headway between the leading vehicle and the controlled vehicle for collision avoidance [1,2]. The aim of the lateral control is to keep the vehicle between lanes [3,4]. Various lateral control methods have been studied for autonomous lane keeping system (LKS) [5-9]. In LKS, the reference steering wheel angle is derived by the lane-keeping control method. Then, the steering wheel angle is controlled by the power steering system.

Nowadays electric power steering (EPS) system is substituted for hydraulic power steering (HPS) system, since the EPS system is superior in several aspects including safety, cost, energy efficiency, environmental protection, and assembly compared with the traditional HPS system [3]. The schematic diagram of a column-mounted EPS system is depicted in Fig. 1. The EPS

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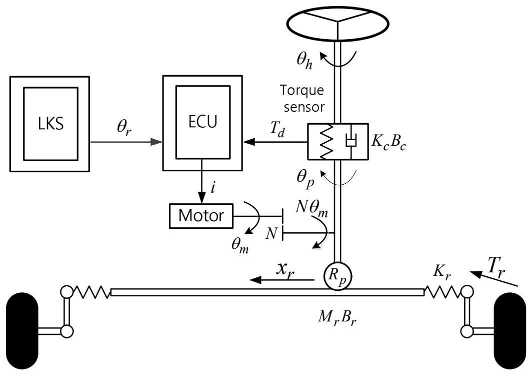


그림 1. 횡방향 제어를 위한 전기적 파워 조향 시스템의 구조.

Fig. 1. Structure of the electric power steering system for lateral control.

system consists of a steering wheel, an intermediate shaft, a motor, a torque sensor, a reduction gear, and a rack/pinion structure. When the driver manually handles the steering wheel, the main role of the EPS system is the torque control of the motor to generate the assistant torque. Various methods for the torque control in EPS have been studied [10-13].

In the LKS of autonomous vehicles, the main role of the EPS system is to make the steering wheel angle track the reference steering wheel angle derived by the lateral control method. The previous lateral control methods regarded the steering wheel angle as the system input [5-9]. These cases may use a DC motor instead of the power steering system or modify the EPS system for the steering wheel angle control. This angle overlay based approach has difficulties and limits since it does not allow the torque combination so that it is difficult to use all of the basic functions of the EPS system Furthermore, it is difficult for driver to smoothly take over the steering wheel control.

On the other hand, in the torque overlay based method, the EPS system can be used for the steering wheel angle control without any modification of the EPS so that a torque integration with basic functions of the EPS is available for the driver's convenience [14]. Thus, the driver can smoothly take over the steering wheel control from the LKS without uncomfortable feeling [15]. In the torque overlay approach, the steering wheel angle is controlled by the interaction between drivers and EPS system. The torque imposed by the driver is amplified by the basic function of the EPS system. For the LKS of the autonomous vehicle, the driver's torque could be affected as disturbance input to the steering wheel angle control and it results in the decrease of the control performance. The unsymmetrical hysteresis behavior occurred due to the structure and friction of the EPS system is also activated as the disturbance. Furthermore, the EPS model uncertainties make the control become more difficult. Thus the steering wheel angle control method should be designed based on torque overlay with the consideration of both the steering wheel angle tracking and the compensation of the model uncertainty and external disturbance (the steering torque imposed by the driver, the unsymmetrical hysteresis behavior, and the friction so on.)

In this paper, we develop a sliding mode control for steering wheel angle control based on torque overlay in order to resolve the problem of previous methods for EPS in the LKS of the autonomous vehicle. For the controller design, we propose 2^{nd} order model of the electric power steering system in the autonomous LKS. The desired state model is designed to prevent the rapid change of the steering wheel angle. The sliding mode steering wheel angle controller is developed for the robustness of the disturbance. Since the proposed method is designed based on torque overlay, a torque integration with basic functions of the EPS for the steering wheel angle control is available for the driver's convenience. The performance of the proposed method was validated via experiments.

II. MATHEMATICAL MODEL OF ELECTRIC POWER STEERING SYSTEM

By applying Newton's second law, the force and torque balance equations of the steering wheel, the motor, and the shaft are given by [12]

$$J_c\ddot{\theta}_h + B_c\dot{\theta}_h + K_c\left(\theta_h - \theta_p\right) = T_d \tag{1}$$

$$J_m\ddot{\theta}_m + B_m\dot{\theta}_m = K_t i - \tau_l \tag{2}$$

$$M_r\ddot{x}_r + B_r\dot{x}_r + K_r x_r = \frac{\tau_p}{R_p} - T_r \tag{3}$$

where $\theta_m = N\theta_p,\ x_r = R_p\theta_p,\ \tau_p = K_c\left(\theta_h - \theta_m / N\right) + N\tau_l.$ In the autonomous LKS, the motor of the EPS system directly control the angular position of the steering wheel for the reference angular position tracking without any intention of the driver. Furthermore, the torque bar is very stiff. Consequently, we can assume that $\theta_h \approx \theta_p$. Thus

$$\begin{aligned} &\theta_m = N\theta_p \approx N\theta_h \\ &x_r = R_p\theta_p \approx R_p\theta_h \\ &\tau_p = K_c\left(\theta_h - \frac{\theta_m}{N}\right) + N\tau_l \approx N\tau_l. \end{aligned} \tag{4}$$

From (1)-(4), we obtain

$$\ddot{\theta}_h = -\frac{K_r R_p^2}{J_{eq} N}\theta_h - \frac{B_{eq}}{J_{eq}}\dot{\theta}_h + \frac{K_t}{J_{eq}} i + \frac{1}{J_{eq}} d \tag{5}$$

where $B_{eq} = B_c + B_m N + \frac{B_r R_p^2}{N}\dot{\theta}_h$, $J_{eq} = J_c + J_m N + \frac{M_r R_p^2}{N}$, $d = T_d - \frac{R_p}{N}T_r$. With the definitions of the state and the input as

$$\begin{aligned} &x = \begin{bmatrix}\theta_h & \dot{\theta}_h\end{bmatrix}^T \\ &u = i. \end{aligned} \tag{6}$$

Equation (5) becomes

$$\dot{x} = \begin{bmatrix} 0 & 1 \\ -\frac{K_r R_p^2}{J_{eq} N} & -\frac{B_{eq}}{J_{eq}} \end{bmatrix} x + \begin{bmatrix} 0 \\ \frac{K_t}{J_{eq}} \end{bmatrix} u + \begin{bmatrix} 0 \\ \frac{1}{J_{eq}} \end{bmatrix} d. \tag{7}$$

III. CONTROLLER DESIGN

In the autonomous LKS, the reference steering wheel angle is generated for the lane keeping by the lateral control method [18]. If the lateral offset error at look down (or at look-ahead distance) is large, the steering wheel angle error becomes large. Thus for the fast convergence of the lateral offset error, the steering wheel angle rotates rapidly above the allowed rotation speed. In order to prevent this situation, we design the desired state x_d using the reference steering wheel angle θ_r as

$$\begin{bmatrix} \dot{x}_{1_d} \\ \dot{x}_{2_d} \end{bmatrix} = \underbrace{\begin{bmatrix} 0 & 1 \\ -k_1 & -k_2 \end{bmatrix}}_{A_d} \underbrace{\begin{bmatrix} x_{1_d} \\ x_{2_d} \end{bmatrix}}_{x_d} + \underbrace{\begin{bmatrix} 0 \\ k_r \end{bmatrix}}_{B_d} \theta_r$$
$$y_d = \underbrace{\begin{bmatrix} 1 & 0 \end{bmatrix}}_{C_d} \begin{bmatrix} x_{1_d} \\ x_{2_d} \end{bmatrix} \tag{8}$$

where k_1 and k_2 are chosen such that the matrix A_d is Hurwitz., and k_r is chosen such that $C_d(-A_d)^{-1}B_d = 1$. The transfer function from θ_r to y_d is

$$\frac{Y_d}{\Theta_r} = C_d\left(sI - A_d\right)^{-1} B_d. \tag{9}$$

Generally, the frequency of the steering wheel angle reference generated by LKS is below 0.05 Hz on the curved road of high way. Thus the parameters k_1 and k_2 are chosen such that the phage lag of (9) is very small below 0.05 Hz. If the steering wheel angle reference includes the high frequency component, the desired state model (8) makes the reference become smooth to prevent that the steering wheel angle rotates rapidly above the allowed rotation speed.

Equation (7) is rewritten as

$$\begin{aligned} \dot{x}_1 &= x_2 \\ \dot{x}_2 &= a_{21}x_1 + a_{22}x_2 + bu + d_1 \end{aligned} \tag{10}$$

where $a_{21} = -\frac{K_r R_p^2}{J_{eq} N}$, $a_{22} = -\frac{B_{eq}}{J_{eq}}$, $b = \frac{K_t}{J_{eq}}$, and $d_1 = \frac{1}{J_{eq}} d$.

For the reference tracking, the control input u is designed as

$$u = -\frac{a_{21}}{b}x_1 - \frac{a_{22}}{b}x_2 - \frac{k_1}{b}x_1 - \frac{k_2}{b}x_2 + \frac{1}{b}v + \frac{k_r}{b}\theta_r \tag{11}$$

where the sliding mode control law v will be designed later. Now we define the tracking error as

$$\begin{aligned} e_0 &= \int_0^t x_1 - x_{1_d} \, d\tau \\ e_1 &= x_1 - x_{1_d} \\ e_2 &= x_2 - x_{2_d}. \end{aligned} \tag{12}$$

Then the tracking error dynamics are

$$\begin{aligned} \dot{e}_0 &= e_1 \\ \dot{e}_1 &= e_2 \\ \dot{e}_2 &= -k_1 e_1 - k_2 e_2 + v + d. \end{aligned} \tag{13}$$

The sliding surface is designed as

$$s = k_{s0}e_0 + k_{s1}e_1 + e_2 \tag{14}$$

where k_{s0} and k_{s1} are constant. If $s = 0$, (14) becomes

$$e_2 = -k_{s0}e_0 - k_{s1}e_1 \tag{15}$$

From (13) and (15),

$$\begin{aligned} \dot{e}_0 &= e_1 \\ \dot{e}_1 &= e_2 \\ e_2 &= -k_{s0}e_0 - k_{s1}e_1. \end{aligned} \tag{16}$$

Thus (16) is simplified as

$$\begin{bmatrix} \dot{e}_0 \\ \dot{e}_1 \end{bmatrix} = \underbrace{\begin{bmatrix} 0 & 1 \\ -k_{s0} & -k_{s1} \end{bmatrix}}_{A_e} \begin{bmatrix} e_0 \\ e_1 \end{bmatrix}. \tag{17}$$

In (17), k_{s0} and k_{s1} should be chosen such that the matrix A_e is Hurwitz. The sliding mode control law v is

$$v = v_{eq} + v_s \tag{18}$$

where v_{eq} and v_s will be designed. We obtain the equivalent control v_{eq} to make $\dot{s}$ become zero as

$$\begin{aligned} \dot{s} &= k_{s0}\dot{e}_0 + k_{s1}\dot{e}_1 + \dot{e}_2 \\ &= k_{s0}e_1 + k_{s1}e_2 + \left(-k_1e_1 - k_2e_2 + v_{eq} + d\right) \\ &= \left(k_{s0} - k_1\right)e_1 + \left(k_{s1} - k_2\right)e_2 + \left(v_{eq} + d\right). \end{aligned} \tag{19}$$

From (19),

$$v_{eq} = -\left[\left(k_{s0} - k_1\right)e_1 + \left(k_{s1} - k_2\right)e_2\right]. \tag{20}$$

The disturbance d will be compensated for by v_s. The disturbance d is $d = T_d - \frac{R_p}{N}T_r$. Thus we can assume that $d_{\max}$ exists such that $d_{\max} \ge |d|$. In most actual systems, all state variables and external disturbances are physically bounded [16]. For compensation of d, v_s is designed as

$$v_s = -ks - \rho\,\mathrm{sgn}(s) \tag{21}$$

where $k > 0$ and $\rho \ge d_{\max}$. In order to prove the stability, we define the Lyapunov candidate function V [17] as

$$V = \frac{1}{2}s^2. \tag{22}$$

The derivative of V with respect to time is

$$\begin{aligned} \dot{V} &= s\dot{s} \\ &= s\left(k_{s0}\dot{e}_0 + k_{s1}\dot{e}_1 + \dot{e}_2\right) \\ &= s\left(-ks - \rho\,\mathrm{sgn}(s) + d\right) \\ &= -ks^2 - \rho s\,\mathrm{sgn}(s) + sd \\ &= -ks^2 - \rho|s| + sd < 0 \end{aligned} \tag{23}$$

Thus, s converges to zero. From (11), (20), and (21), the control input is

$$\begin{aligned} u = &-\frac{a_{21} + k_1}{b}x_1 - \frac{a_{22} + k_2}{b}x_2 + \frac{k_r}{b}\theta_r \\ &+ \frac{1}{b}\left[-[(k_{s0} - k_1)e_1 + (k_{s1} - k_2)e_2] - ks - \rho\,\mathrm{sgn}(s)\right] \end{aligned} \tag{24}$$

IV. EXPERIMENTAL RESULTS

Experiments were executed to evaluate the performances of the proposed method. The EPS hardware in the loop simulation (HILS) system is shown in Fig. 2. The EPS HILS system consisted of the EPS system, the spring system and the dSPACE. In this system, the mounted spring was used to emulate the self-alignment torque. DS1501 manufactured by dSPACE Inc. was used as an embedded real-time controller. The control sampling rate was 100 Hz. Since the numerical value of used EPS parameters is proprietary information, it is omitted.

The Bode plot of (9) is from Fig. 3, it is observed that the parameters k_1 and k_2 were chosen such that the phage lag of (9) is very small below 0.05 Hz.

Sinusoidal steering tests were performed to study the characteristics of EPS system. Fig. 4 shows the steering wheel angle response when a sinusoidal command torque, $T=3\sin(0.05\times 2\pi t)$ was injected artificially. In Fig. 4, the used input torque was amplified fifth times for comparison with the steering wheel angle. Due to the unsymmetrical structure and the friction, the steering wheel angle response was very poor and slow. For the detailed study, four sinusoidal command torques, $T=3\sin(0.5\times 2\pi t)$, $T=3\sin(0.1\times 2\pi t)$, $T=3\sin(0.2\times 2\pi t)$, and $T=3\sin(0.4\times 2\pi t)$ were injected artificially. The steering wheel angle responses are shown in Fig. 5. It was observed that the unsymmetrical hysteresis behaviors occurred.

Fig. 6 shows the steering wheel angle tracking performance of the proposed method without the driver's torque. It is observed that the desired state variable x_{1_d} tracked the reference x_r well. Although the disturbance exists as shown in Figs. 4 and 5, the steering wheel angle also tracked the desired state variable x_{1_d} well. The relatively large tracking errors near the zero velocity periods appeared due to the unsymmetrical hysteresis behaviors of EPS system. The high spring force in the experimental set up might be one of the main causes of the relatively large tracking errors near the zero velocity periods. To overcome the unsymmetrical hysteresis behaviors, the control input was also asymmetric as shown in Fig. 6(b). Since the driver's torque was not injected as the disturbance, the measured driver's torque was

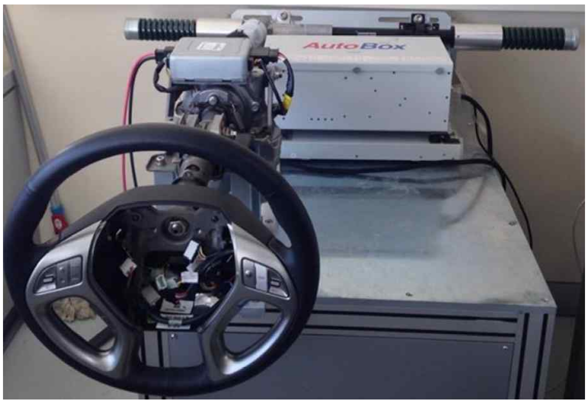


그림 2. 전기적 파워 조향 시스템의 모의 실험 장치.

Fig. 2. The EPS HILS system.

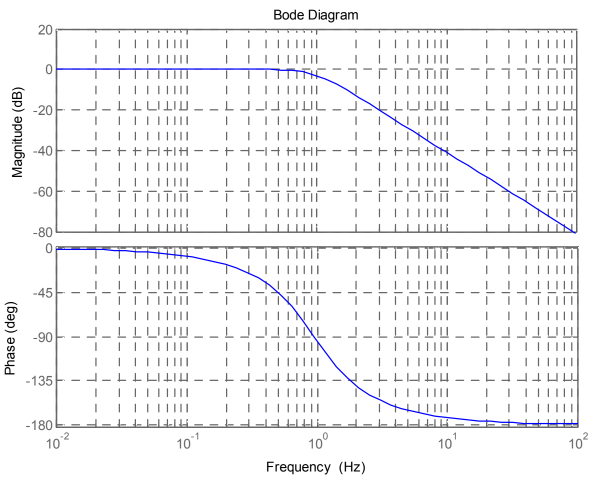


그림 3. (9)의 보드 선도.

Fig. 3. Bode plot of (9).

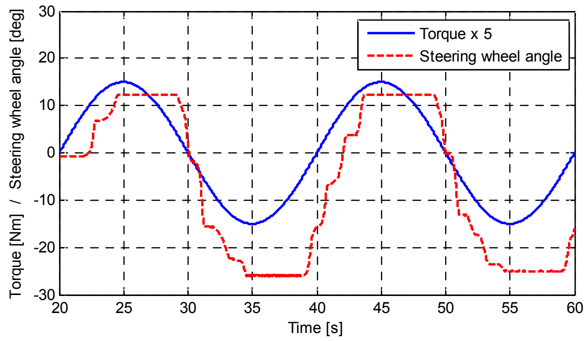


그림 4. $T=3\sin(0.05\times 2\pi t)$ 를 이용한 조향 테스트: 비교를 위하여 5배 증폭된 토크를 사용함.

Fig. 4. Sinusoidal steering test for $T=3\sin(0.05\times 2\pi t)$: For comparison, the used input torque was amplified five times.

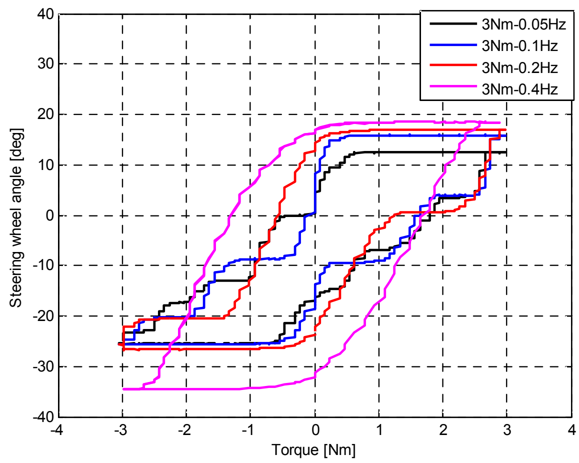
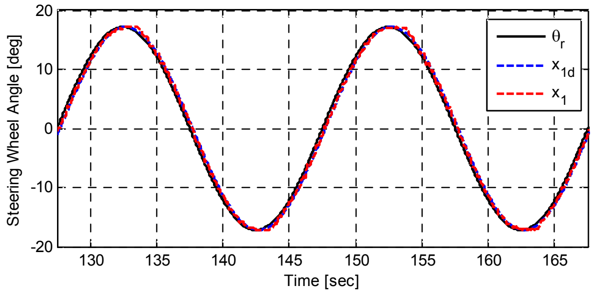
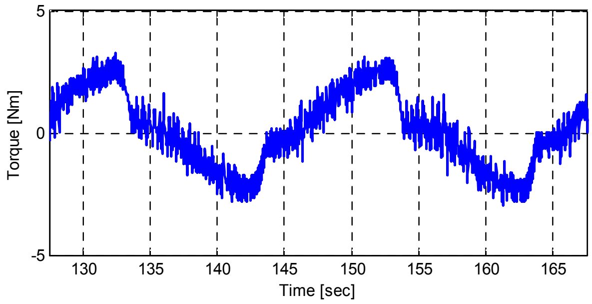


그림 5. $T=3\sin(0.05\times 2\pi t)$, $T=3\sin(0.1\times 2\pi t)$, $T=3\sin(0.2\times 2\pi t)$, 그리고 $T=3\sin(0.4\times 2\pi t)$ 을 이용한 조향 테스트.

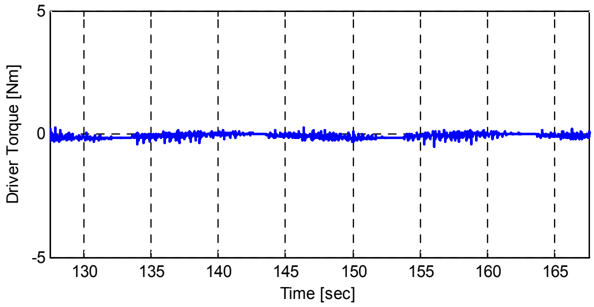
Fig. 5. Sinusoidal steering tests for $T=3\sin(0.05\times 2\pi t)$, $T=3\sin(0.1\times 2\pi t)$, $T=3\sin(0.2\times 2\pi t)$, and $T=3\sin(0.4\times 2\pi t)$.



(a) Steering wheel angle.



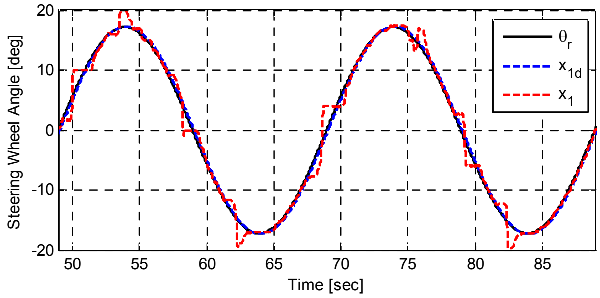
(b) Input.



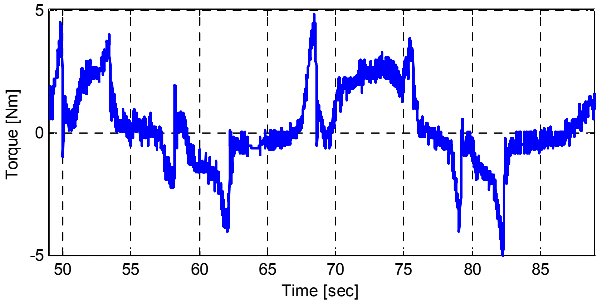
(c) Driver's torque.

그림 6. 운전자의 토크가 없는 경우의 제시된 방법의 조향각 추종 성능.

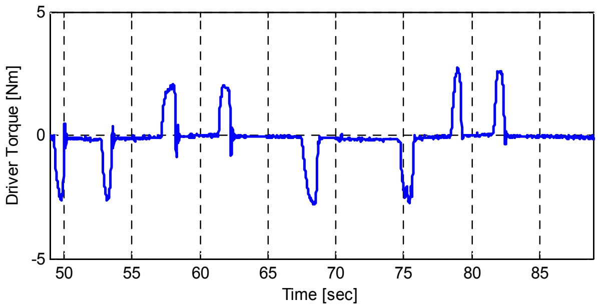
Fig. 6. Steering wheel angle tracking performance of the proposed method without the driver's torque.



(a) Steering wheel angle.



(b) Input.



(c) Driver's torque.

그림 7. 운전자의 토크가 있는 경우의 제시된 방법의 조향각 추종 성능.

Fig. 7. Steering wheel angle tracking performance of the proposed method with the driver's torque.

almost zero. Fig. 7 shows the steering wheel angle tracking performance of the proposed method with the driver's torque. In Fig. 7(c), when the driver tried to strongly hold the steering wheel, the measured driver's absolute torque went up to 3 Nm. To overcome driver's torque, the input torque also increased. Note that the shaft with torsion bar is nearly rigid in the EPS. Thus, the driver's torque to hold the steering wheel was activated as torque disturbance as well as angle disturbance in the torque overlay based steering wheel control Consequently, the steering wheel control cannot perfectly be free under the driver's torque although the driver's holding torque is compensated for in the torque overlay based steering wheel control. Thus the steering wheel tracking error was relatively larger, however, the performance was recovered after the driver released the steering wheel. It means that if the driver intentionally handles the steering wheel to avoid the emergent situation (i.e., the collision or the malfunction of LKS) then the driver can resist the steering wheel controlled by the EPS so that the driver can relatively smoothly easily take over the steering wheel control compared to the angle overlay based method.

V. CONCLUSION

In this paper, we developed the sliding mode control for steering wheel angle control based on torque overlay for the steering wheel angle tracking of EPS in the LKS of the autonomous vehicle. For the controller design, we proposed 2nd order model of the electric power steering system in the autonomous LKS. The desired state model was designed to prevent the rapid change of the steering wheel angle. The sliding mode steering wheel angle controller was developed for the robustness of the disturbance. Since the proposed method was designed based on torque overlay, a torque integration with basic functions of the EPS for the steering wheel angle control is available for the driver's convenience. The performance of the proposed method was validated via experiments. Although the disturbance exists, the proposed method made the steering wheel angle track the reference well.

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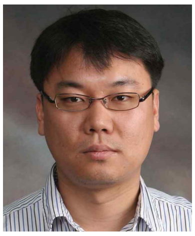

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