Steering Assistance System for Driver Characteristics using Gain Scheduling Control

Yukihiro Fujiwara† and Shuichi Adachi††

†Honda R&D Co., Ltd. Tochigi Center / Utsunomiya University
4630 Shimotakanezawa, Haga-machi, Haga-gun, Tochigi, 321-3393, Japan
Fax: +81-28-677-6730
Phone: +81-28-677-7259
e-mail: yukihiro_fujiwara@n.t.rd.honda.co.jp

††Department of Electrical and Electronic Engineering, Utsunomiya University
7-1-2 Yoto, Utsunomiya, Tochigi, 321-8585, Japan
Fax/Phone: +81-28-689-6125
e-mail: adachis@cc.utsunomiya-u.ac.jp

Keywords:Driver support system, Vehicle control, $\mathcal{H}_\infty$ control, Gain-scheduling control, LMI.

Abstract

This paper proposes a new driver support system which uses both the steering wheel angle and the steering torque of the vehicle so as to take into account the characteristics of the driver's operation. The proposed method is based on a gain scheduling control to accomplish the cooperation between the driver and the support system. By numerical experiments which simulate obstacle avoidance, it is shown that the proposed method has a good tracking capability and cooperates well with driver's operation.

1 Introduction

The prevention of traffic accidents is one of the most important social requests. The accident prevention systems have been developed by a variety of schemes, for example, by adding some intelligence to the vehicles[1]. Among the research on the intelligent vehicles, an automated steering system has been studied[2]~[4]. However, it has many problems such as recognition of driving situation and stabilization of vehicle dynamics.

As a preliminary stage to the automated steering system, driver support systems have been investigated. In order to realize the driver support systems, it is necessary to take into account the characteristics of the driver's operation. In the conventional studies on the support systems[5, 6], the steering torque is managed so that the support system cooperates with the driver's operation. However, the system has a drawback that the road tracking performance was deteriorated.

In order to improve the road tracking performance, this paper proposes a new control method of the support system which uses both the steering wheel angle and the steering torque. The proposed method is based on a gain scheduling control so as to accomplish the cooperation between the driver and the support system. By numerical experiments which simulate an obstacle avoidance, it is shown that the proposed method has a good tracking capability and cooperates well with driver's operation.

2 Problem formulation

In this paper, a system configuration[7] shown in Fig.1 is considered. The notation in Fig.1 is briefly explained.

As a steering actuator which drives the front wheels, an Electric Power Steering (EPS)[8] which consists of a DC-motor and a ball screw mechanism is employed. A camera is installed in a vehicle to recognize the road shape. From the camera image, a feature value, X_{sum} is calculated.

The vehicle model which is a 1-input and 2-outputs model describes the input-output relationship from the actual wheel angle, θ_f, to side slip angle, β, and the yaw-rate, γ. β and γ can be considered as the state variables of the vehicle.

The control system contains two controllers. One is a guidance controller, K, to determine a reference wheel angle, A_{cmd}, using the feature value, X_{sum}, relating to lane shape. The other is an angle controller, $\boldsymbol{R}$,

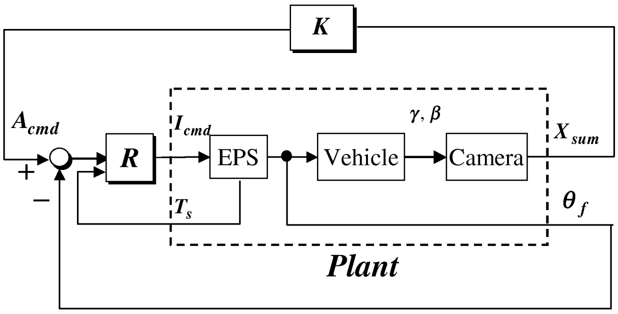

Figure 1: System configuration

to determine the reference current, I_{cmd}, of the motor which is built in the EPS based on the error between the reference angle and the actual wheel angle. The angle controller is redesigned as a steering controller, which uses both the steering wheel angle and the steering torque.

The performance of the system is examined by numerical experiments which simulates a double-lane change maneuver that recovers straight-line travel after obstacle avoidance. Usually, the maneuver results from the driver's operation.

3 System modeling

In this section, two models for controller design are built. One is an integrated model for the guidance controller design, which consists of the vehicle dynamics and the camera characteristics. The other is a model of the EPS for the steering controller, which includes the steering torque characteristics.

3.1 Integrated model

The integrated model for designing the guidance controller is described by

$$\begin{array}{rcl} \dot{\boldsymbol{x}}_{\boldsymbol{p}} &=& \boldsymbol{A_p x_p} + \boldsymbol{B_p}\theta_f + \boldsymbol{D_p}, \\ y &=& \boldsymbol{C_p x_p}, \end{array} \tag{1}$$

where

$$\boldsymbol{x_p} = \left[\begin{array}{ccc} \gamma & \beta & X_{sum} \end{array}\right]^T,$$

$$\boldsymbol{A_p} = \left[\begin{array}{ccc} \frac{-2(K_f l_f^2 + K_r l_r^2)}{I_z V} & \frac{-2(K_f l_f - K_r l_r)}{I_z} & 0 \\ \frac{-2(K_f l_f - K_r l_r)}{mV^2} - 1 & \frac{-2(K_f + K_r)}{mV} & 0 \\ \frac{f}{\lambda}\Sigma_{i=1}^n a_i & -\frac{V}{H}\Sigma_{i=1}^n a_i y_{pi} & 0 \end{array}\right],$$

$$\boldsymbol{B_p} = \left[\begin{array}{c} \frac{2K_f l_f}{I_z} \\ \frac{2K_f}{mV} \\ 0 \end{array}\right], \quad \boldsymbol{C_p} = \left[\begin{array}{ccc} 0 & 0 & 1 \end{array}\right],$$

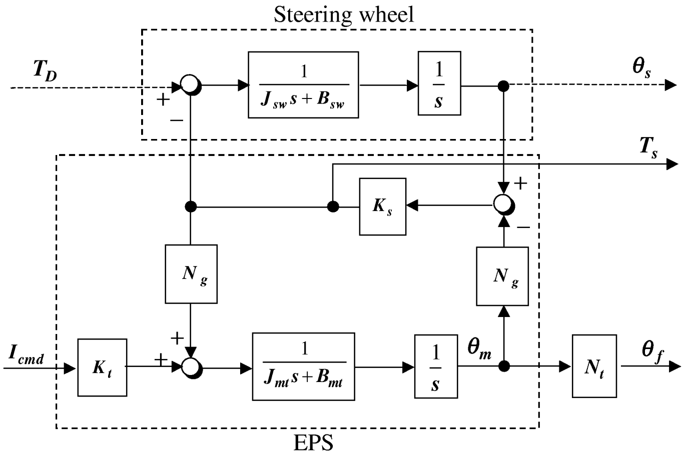

Figure 2: Model for steering controller

$$\boldsymbol{D_p} = \left[\begin{array}{c} 0 \\ 0 \\ \frac{1}{H}\Sigma_{i=1}^n a_i y_{pi} T_{xi} \end{array}\right].$$

l_f is distance between the center of gravity and the front axle, l_r is distance between the center of gravity and the rear axle, K_f is the cornering power of a front tire, K_r is the cornering power of a rear tire, I_z is the yaw inertia moment, m is the vehicle weight, V is the vehicle speed, and θ_f is the front wheel angle.

3.2 Steering model

In order to take the driver's operation applied to the vehicle into account, dynamics of the steering wheel is incorporated into the EPS model, as shown in Fig.2.

Then,

$$\frac{d}{dt}\left[\begin{array}{c} \boldsymbol{x_m} \\ \boldsymbol{x_s} \end{array}\right] = \left[\begin{array}{cc} A_m - N_g^2 K_s B_m C_m & N_g K_s B_m C_s \\ K_s N_g B_s C_m & A_s - K_s B_s C_s \end{array}\right]\left[\begin{array}{c} x_m \\ x_s \end{array}\right] + \left[\begin{array}{cc} 0 & B_m K_t \\ B_s & 0 \end{array}\right]\left[\begin{array}{c} T_D \\ I_{cmd} \end{array}\right] \tag{2}$$

$$\left[\begin{array}{c} \theta_s \\ T_s \\ \theta_f \end{array}\right] = \left[\begin{array}{cc} 0 & C_s \\ -K_s N_g C_m & K_s C_s \\ N_t C_m & 0 \end{array}\right]\left[\begin{array}{c} x_m \\ x_s \end{array}\right] \tag{3}$$

where

$$\boldsymbol{A_m} = \left[\begin{array}{cc} 0 & 1 \\ 0 & -\frac{B_{mt}}{J_{mt}} \end{array}\right], \quad \boldsymbol{B_m} = \left[\begin{array}{c} 0 \\ \frac{1}{J_{mt}} \end{array}\right],$$

$$\boldsymbol{C_m} = \left[\begin{array}{cc} 1 & 0 \end{array}\right], \quad \boldsymbol{x_m} = \left[\begin{array}{cc} \theta_m & \dot{\theta}_m \end{array}\right]^T$$

$$\boldsymbol{A_s} = \left[\begin{array}{cc} 0 & 1 \\ 0 & -\frac{B_{sw}}{J_{sw}} \end{array}\right], \quad \boldsymbol{B_s} = \left[\begin{array}{c} 0 \\ \frac{1}{J_{sw}} \end{array}\right],$$

$$\boldsymbol{C_s} = \left[\begin{array}{cc} 1 & 0 \end{array}\right], \quad \boldsymbol{x_s} = \left[\begin{array}{cc} \theta_s & \dot{\theta}_s \end{array}\right]^T.$$

J_{mt} is the inertia moment of the motor, B_{mt} is the damping constant of the motor, J_{sw} is the inertia moment of the steering wheel, B_{sw} is the damping constant of the steering wheel, θ_m is the motor rotational angle, $\dot{\theta}_m$ is the motor rotational rate, θ_s is the steering wheel rotational angle, $\dot{\theta}_s$ is the steering wheel rotational rate, N_g is the gear ratio between the motor and the steering wheel, N_t is the gear ratio between the rotational angle of the motor and of the front wheel, K_t is the torque constant of the motor, K_s is the spring constant of torque sensor, T_D is the steering torque applied by the driver operation.

We suppose that the driver characteristics is time varying parameter described by

$$T_D = \delta(t) q \theta_s, \quad (4)$$
$$|\delta(t)| < 1. \quad (5)$$

where, $\delta(t)$ is time varying parameter and q is constant value. From eq.(2) $\sim$ (5), we obtain

$$G_a \begin{cases} \frac{d}{dt}\boldsymbol{x_a} = \boldsymbol{A_a x_a} + \boldsymbol{B_a} I_{cmd} \\ \begin{bmatrix} T_s \\ \theta_f \end{bmatrix} = \boldsymbol{C_a x_a} \end{cases} \quad (6)$$

where,

$$\boldsymbol{A_a} = \begin{bmatrix} A_m - N_g^2 K_s B_m C_m & N_g K_s B_m C_s \\ K_s N_g B_s C_m & A_s - K_s B_s C_s - \delta(t) B_s q C_s \end{bmatrix},$$
$$\boldsymbol{B_a} = \begin{bmatrix} B_m K_t \\ 0 \end{bmatrix}, \quad \boldsymbol{C_a} = \begin{bmatrix} -K_s N_g C_m & K_s C_s \\ N_t & 0 \end{bmatrix},$$
$$\boldsymbol{x_a} = \begin{bmatrix} x_m \\ x_s \end{bmatrix}.$$

3.3 Steering model with driver characteristics

In order to predict the conflict between the driver operation and the system's assistance operation, we defined the following parameter.

$$p(t) = p^{max}\left(1 - \frac{|T_s|}{T_{max}}\right) + p^{min}\frac{|T_s|}{T_{max}} \quad (7)$$

where T_{max} is maximum value of steering torque T_s. Please note that its value is small if the steering torque is big and that its value is big if the steering torque is small. The following frequency weight is introduced in order to limit the control input to the low frequency.

$$W_g(T_s) \begin{cases} \dot{x}_e = a_e x_e + b_e v \\ I_{cmd} = p(t) c_e x_e \end{cases} \quad (8)$$

where, x_e and v_e denote the filter state and the imaginary input, respectively, and a_e, b_e, c_e denote the frequency weighting constants. Augmenting eq.(6) by using the frequency weight, we obtain

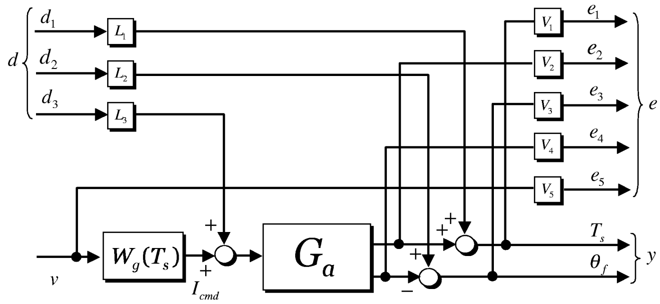

Figure 3: Block structure of generalized plant

$$\frac{d}{dt}\begin{bmatrix} \boldsymbol{x_a} \\ x_e \end{bmatrix} = \begin{bmatrix} \boldsymbol{A_a} & p(t)\boldsymbol{B_a} c_e \\ 0 & a_e \end{bmatrix}\begin{bmatrix} \boldsymbol{x_a} \\ x_e \end{bmatrix} + \begin{bmatrix} 0 \\ b_e \end{bmatrix} v. \quad (9)$$

4 Controller design

4.1 Guidance controller design

The guidance controller, $\boldsymbol{K}$, is designed based on the integrated model eq.(1). Due to the space limitation, the detail is omitted, see [9].

4.2 Steering controller design

The angle controller, $\boldsymbol{R}$, shown in Fig.1 is redesigned as the steering controller based on the model depicted in Fig.2.

The control specifications are

S1: To reduce the influence of the road reaction torque d_1 on the steering torque motor θ_m, which corresponds to the disturbance rejection performance.

S2: To reduce the influence of reference angle d_2 on motor rotational angle e_3, which corresponds to the reference tracking performance.

Then, we obtain the generalized plant shown in Filg.3, where d_3 are disturbance signals,L_1, L_2, L_3 are constant values for the disturbance signals, e_1, e_2, e_3, e_4, e_5 are evaluated signals, and V_1, V_2, V_3, V_4, V_5 are constant values for the evaluated signals. The generalized plant is described by the following equation.

$$\begin{bmatrix} \boldsymbol{x_{aa}} \\ \boldsymbol{e} \\ \boldsymbol{y} \end{bmatrix} = \left[\begin{array}{c|cc} \boldsymbol{A}(p(t), \delta(t)) & \boldsymbol{B_1} & \boldsymbol{B_2} \\ \hline \boldsymbol{C_1} & \boldsymbol{D_{11}} & \boldsymbol{D12} \\ \boldsymbol{C_2} & \boldsymbol{D_{21}} & D_{22} \end{array}\right]\begin{bmatrix} \boldsymbol{d} \\ I_{cmd} \end{bmatrix} \quad (10)$$

where

$$
\boldsymbol{A}(p(t),\delta(t)) = \boldsymbol{A_0} + p\boldsymbol{A_1} + \delta\boldsymbol{A_2}
$$

$$
\boldsymbol{A_0} = \begin{bmatrix} A_m - N_g^2 K_s B_m C_m & N_g K_s B_m C_s & 0 \\ N_g K_s B_s C_m & A_s - K_s B_s C_s & 0 \\ 0 & 0 & A_e \end{bmatrix},
$$

$$
\boldsymbol{A_1} = \begin{bmatrix} 0 & 0 & B_m K_t C_e \\ 0 & 0 & 0 \\ 0 & 0 & 0 \end{bmatrix}, \quad \boldsymbol{A_2} = \begin{bmatrix} 0 & 0 & 0 \\ 0 & qC_s & 0 \\ 0 & 0 & 0 \end{bmatrix},
$$

$$
\boldsymbol{B_1} = \begin{bmatrix} 0 & 0 & B_m K_t L_3 \\ 0 & 0 & 0 \\ 0 & 0 & 0 \\ 0 & 0 & 0 \end{bmatrix}, \quad \boldsymbol{B_2} = \begin{bmatrix} 0 \\ 0 \\ 0 \\ B_e \end{bmatrix},
$$

$$
\boldsymbol{C_1} = \begin{bmatrix} V_1 K_s N_g C_m & V_1 K_s C_s & 0 \\ -V_2 K_s N_g C_m & V_2 K_s C_s & 0 \\ -V_3 N_t C_m & 0 & 0 \\ V_4 N_t C_m & 0 & 0 \end{bmatrix},
$$

$$
\boldsymbol{C_2} = \begin{bmatrix} -K_s N_g C_m & K_s C_s & 0 \\ -N_t C_m & 0 & 0 \end{bmatrix},
$$

$$
\boldsymbol{D_{11}} = \begin{bmatrix} V_1 L_1 & 0 & 0 \\ 0 & 0 & 0 \\ 0 & V_3 L_2 & 0 \\ 0 & 0 & 0 \\ 0 & 0 & 0 \end{bmatrix}, \quad \boldsymbol{D_{12}} = \begin{bmatrix} 0 \\ 0 \\ 0 \\ 0 \\ V_5 \end{bmatrix},
$$

$$
\boldsymbol{D_{21}} = \begin{bmatrix} L_1 & 0 & 0 \\ 0 & L_2 & 0 \end{bmatrix}, \quad \boldsymbol{D_{22}} = \begin{bmatrix} 0 \\ 0 \end{bmatrix},
$$

$$
\boldsymbol{x_{aa}} = \begin{bmatrix} \boldsymbol{x_a} & x_e \end{bmatrix}^T .
$$

The steering controller is obtained by applying gain scheduling H_∞ control method to the generalized plant eq.(10). Gain scheduling output feedback controllers satisfying

- Internal stability for closed loop system
- $\|\boldsymbol{T_{ed}}\|_{l2} < \gamma$

for all the $p(t), \dot{p}(t)$ in the generalized plant can be obtained if there exist the symmetric matrices $\boldsymbol{X}(p)$, $\boldsymbol{Y}(p)$ and the four matrices $\boldsymbol{A_k}(p)$,$\boldsymbol{B_k}(p)$,$\boldsymbol{C_k}(p)$, $\boldsymbol{D_k}(p)$ satisfying the following two LMIs

$$
\begin{bmatrix} X & I \\ I & Y \end{bmatrix} > 0 \tag{11}
$$

$$
\begin{bmatrix} \dot{X} + XA + B_k C_2 + (XA + B_k C_2)^T & * & * & * \\ A_k^T + A + B_2 D_k C_2 & -\dot{Y} + AY + B_2 C_k + (AY + B_2 C_k)^T & * & * \\ (XB_1 + B_k D_{21})^T & (B_1 + B_2 D_k D_{21})^T & -\gamma I & * \\ C_1 + D_{12} D_k C_2 & C_1 Y + D_{12} C_k & D_{11} + D_{12} D_k D_{21} & -\gamma I \end{bmatrix} < 0 \tag{12}
$$

$$
p(t) \in [p^{max} \quad p^{min}] \quad for^{\forall} t \in [0 \quad \infty), \quad \dot{p}(t) \in [pd^{max} \quad pd^{min}] \quad for^{\forall} t \in [0 \quad \infty)
$$

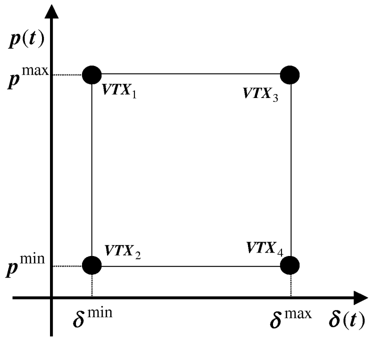

Figure 4: Vertices in paramer space

where '*' denotes

$$
\begin{bmatrix} Q & * \\ M & P \end{bmatrix} = \begin{bmatrix} Q & M^T \\ M & P \end{bmatrix}, \tag{13}
$$

$\boldsymbol{T_{ed}}$ is the closed loop transfer function from disturbance signal, $\boldsymbol{d}$, to controlled signal, $\boldsymbol{e}$. Notation of t and p is omitted.

Then, we obtain a gain scheduling H_∞ output feedback controller [10] described by

$$
\begin{aligned}
\boldsymbol{A_c}(p,\dot{p}) &= \{I - X(p)Y(p)\}^{-1}[X(p)\dot{Y}(p)\} \\
&+ A_k(p) - X(p)\{A(p) - B_2(p)D_k(p)C_2(p)\}Y(p) \\
&- B_k(p)C_2(p)Y(p) - X(p)B_2(p)C_k(p)] \qquad (14) \\
\boldsymbol{B_c}(p) &= \{I - X(p)Y(p)\}^{-1} \\
&\quad \{B_k(p) - X(p)B_2(p)D_k(p)\} \qquad (15) \\
\boldsymbol{C_c}(p) &= C_k(p) - D_k(p)C_2(p)Y(p) \qquad (16) \\
\boldsymbol{D_c}(p) &= D_k(p) \qquad (17)
\end{aligned}
$$

In order to solve the LMIs, we suppose the controller is described by

$$
\begin{aligned}
\boldsymbol{X}(p) &= \boldsymbol{X_0} + p\boldsymbol{X_1}, \quad \boldsymbol{Y} = \boldsymbol{Y_0} \\
\boldsymbol{A_k}(p) &= \boldsymbol{A_{k0}} + p\boldsymbol{A_{k1}}, \\
\boldsymbol{B_k}(p) &= \boldsymbol{B_{k0}} + p\boldsymbol{B_{k1}}, \\
\boldsymbol{C_k}(p) &= \boldsymbol{C_{k0}} + p\boldsymbol{C_{k1}}, \\
\boldsymbol{D_k}(p) &= \boldsymbol{D_{k0}} + p\boldsymbol{D_{k1}}.
\end{aligned}
$$

and we design a controller of the structure by solving the LMIs corresponding to the parameters p and δ at the vertices in Fig.4.

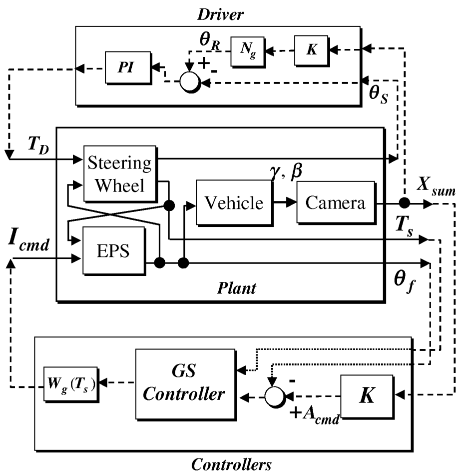

Figure 5: System structure for numerical experiments

5 Numerical Experiments

Numerical experiments were carried out in order to evaluate the performance of the proposed controllers. The parameter values for a compact-class vehicle were used in the experiments. The driver characteristics is modeled as proportional and integral elements whose input is error between the desired wheel angle, θ_R, and the actual one. The desired wheel angle is calculated by controller $\boldsymbol{K}$ based on the feature value, shown in Fig.5. In the numerical experiments, a maneuver which recovers straight-line travel after obstacle avoidance, which is called a double-lane change maneuver shown in Fig.6, was used.

The results of the numerical experiments are shown in Figs, 7 and 8. In order to compare with performance of gain scheduling controller, H_∞ constant gain controller designed at the situation in vertices of VTX_1 and VTX_2 is prepared. It is clear from the dashed line in Fig.7, the vehicle dynamics is unstable when the H_∞ constant gain controller assists for driver's operation. On the other hand, from solid lines in Fig.7, the vehicle dynamics become stable when the proposed driver's assist methods are applied. Especially, it is clear from the solid line in Fig.7, the scheduling steering controller makes the trajectory of the vehicle smooth.

Fig.8 shows the steering wheel torque signals for each controllers. It is clear that the scheduling steering controller generates a more smooth steering wheel signal.

Fig.9 shows the open loop transfer functions in each vertices ($VTX_1 \sim VTX_4$). In a case which employ the gain scheduling controller, In low frequency range less than 1 Hz, the gain scheduling controller has the open loop characteristics in VTX_2. Therefore, the road tracking performance is carried out by high gain characteristic of the loop transfer function. In middle frequency range from 1 Hz to 20 Hz, the scheduling controller has the characteristic in VTX_4. The robustness for disturbance applied the steering torque signal is performed by low gain characteristic of transfer function in VTX_4.

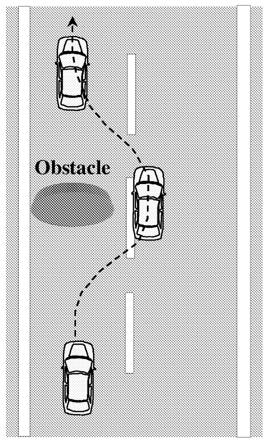

Figure 6: Manuver for numerical experiments

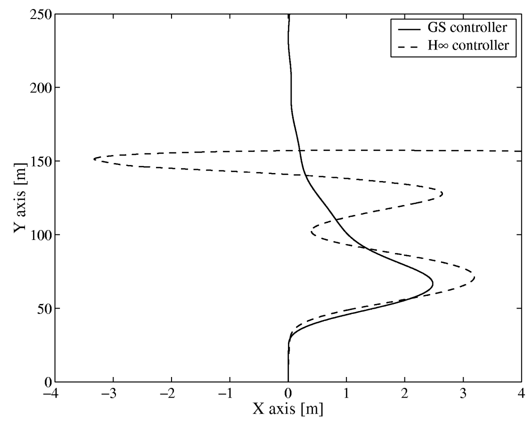

Figure 7: Trajectory of center of gravity on vehicle

From these results and discussions, it is shown that the scheduling steering controller is effective not only from the vehicle stability point of view but also the cooperation with the drive's operation point of view.

6 Conclusions

The new driver support system which uses both the steering wheel angle and the steering torque of the vehicle has been proposed. The steering controller was designed by considering the driver's characteristics as uncertainty and was extended to the gain scheduling controller where the scheduling parameter was the steering torque. The effectiveness of the proposed controller was shown by the numerical experiments. The examination of the proposed method by an actual vehicle remains to be done.

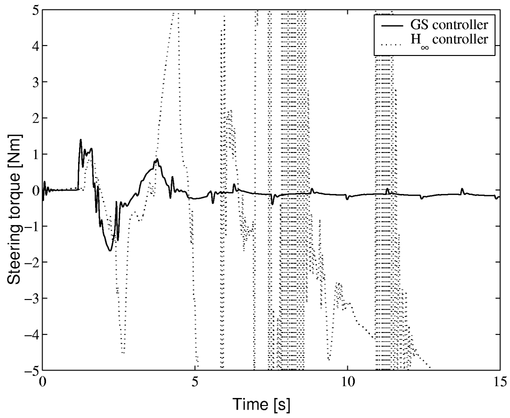

Figure 8: Steering torque signals of steering controller

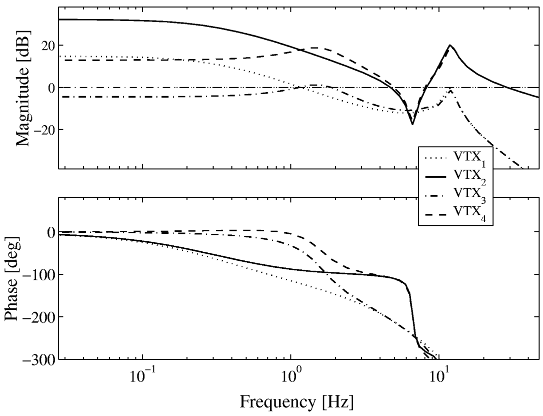

Figure 9: Open loop transfer functions of steering control system

References

[1] T. Miyazaki : Plan for promotion of the development of advanced safety vehicle (ASV) (in Japanese), *The Journal of SICE*, Vol.36, No.3, pp.165–167, 1997.

[2] H. Mouri and H. Furusho : Investigation of automatic path tracking —Comparison the performance of LQ control with that of PD control— (in Japanese), *Proc. of JSAE*, Vol.30, No.1, pp.121–126, 1999.

[3] S. Nohtomi and S. Horiuchi : A path tracking control system using generalized predictive control theory (in Japanese), *Proc. of JSAE*, No.62–98, pp.9–12, 1998.

[4] H. Kumamoto, I. Sasamoto, K. Tenmoku and H. Shimoura : Vehicles steering control by reduced-Dimension Sliding mode theory (in Japanese), *Trans. of the SICE*, Vol.34, No.5, pp.393–399, 1998.

[5] H. Mouri and M. Nagai : Study on automatic path tracking with steering torque input —1st report, comparison between control method with steering angle input and with steering torque input (in Japanese), *Trans. of the JSME*, Vol.67-C, No.664, pp.3836–3843, 2001.

[6] M. Nagai, K. Shitamitsu, H. Yosida and H. Mouri : Over-ride characteristics of lane-keeping control system using steering torque input —Driving simulator study on lateral wind response— (in Japanese), *Proc. of JSAE*, No.82-01, pp.9–12, 2001.

[7] T. Komori, Y. Fujiwara, M. Fujita and K. Uchida : Automated driving by visual servoing (in Japanese), *Trans. IEE of Japan*, Vol.120-C, No.4, pp.501–506, 2000.

[8] N. Sugitani, Y. Fujiwara, K. Uchida and M. Fujita : Electric power steering with H-infinity control designed to obtain road information, *Proc. of the ACC*, 1997.

[9] Y. Fujiwara, M. Yoshii and S. Adachi : Automated steering control system design for passenger vehicle in consideration of steering actuator dynamics, *Proc. of the ACC*, pp.857–862, 2002.

[10] P.Apkarian : On the discretization of LMI-synthesized linear parameter-varying controllers, *Automatica*, Vol.33, No.4, pp.655–661, 1997.