

Electronic Power Steering Compensating Control for Influence of Vehicle Dynamics on Steering Torque

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ABSTRACT

The frequency response of steering effort torque changes according to the influence of vehicle dynamics. To help enhance feel, a new electronic power steering (EPS) control has been constructed. As the control can be achieved by the addition of a viscosity control with a filter to existing EPS systems, its structure is simple and easily installed. Actual vehicle tests have been performed to verify that the control is capable of enhancing convergence without adversely affecting response during steering.

INTRODUCTION

It is well known that the dynamic characteristics of vehicle behavior and the steering system have a mutual relationship and help contribute to vehicle control and steering convergence. In particular, during highspeed driving, vehicle dynamics may reduce the damping characteristics of the steering system as well as the damping characteristics of vehicle yaw rate response. Moreover, there has been a recent rise in vehicles equipped with electronic power steering (EPS) to improve fuel efficiency. The electrical motors of EPS systems apply a large inertia moment on the steering system, which has a reducing effect on damping characteristics. In other words, the dynamic characteristics of vehicle behavior and the steering system combine, and as an overall system, cause further deterioration to damping.⁽¹⁾⁽²⁾ For this reason, measures to ensure that the EPS has enough damping control, such as the application of friction to the steering

system have been adopted. However, excessive friction can be felt as a lack of smoothness by the driver and excessive damping can be transmitted as a feeling of stickiness. These characteristics can occur when torque is applied to the steering and are different when a certain steering wheel angle is maintained. Focusing on this point, several case studies have been reported of EPS control for improving the damping characteristics of the steering and vehicle systems with respect to steering torque.⁽³⁾⁽⁴⁾ There is also a published case study of EPS control that focuses on the dynamic characteristics of the steering system to help enhance damping characteristics while compensating for the influence of inertia moment.⁽⁵⁾

As it can be seen, it is important to consider the dynamic characteristics of the steering and vehicle systems with respect to steering torque. Vehicle response due to steering torque input from a steering robot has already been analyzed to identify the influence of vehicle dynamics on the steering system and vice versa.⁽⁶⁾⁽⁷⁾ This paper details the construction of an EPS control that compensates for the influence of vehicle dynamics on the steering system based on a novel approach. It also describes verification of the effectiveness of the control by simulations and actual vehicle tests using steering torque input, as well as by subjective evaluations from drivers of vehicles equipped with the control.

STEERING SYSTEM MODEL AND EQUATIONS OF MOTION

To observe the influence of vehicle dynamics on the steering system, the extremely simple steering and vehicle model shown in Fig. 1 was used.

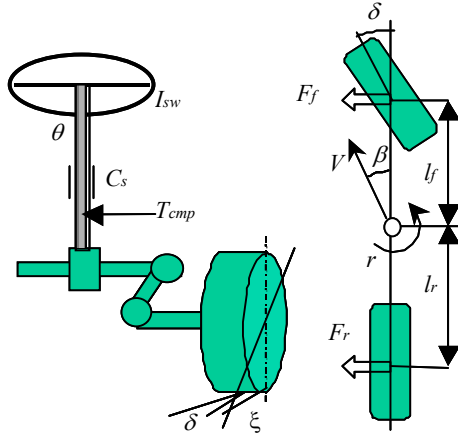


Fig. 1 Steering and Vehicle Model

The equations of motion are shown in equations 1 to 7 below after Laplace transformation.

$$mV(\beta \cdot s + r) = F_f + F_r \quad (1)$$

$$I_z r \cdot s^2 = F_f l_f - F_r l_r \quad (2)$$

$$F_f = K_f \left(\delta - \beta - \frac{l_f r}{V} \right) \quad (3)$$

$$F_r = K_r \left(-\beta + \frac{l_r r}{V} \right) \quad (4)$$

$$T_{sat} = \xi F_f \frac{1}{N_s} \quad (5)$$

$$\delta = \frac{\theta}{N_s} \quad (6)$$

$$I_{sw} \theta \cdot s^2 + C_s \theta \cdot s + T_{cmp} + T_{sat} = T_{sw} \quad (7)$$

Here, equations 8 to 12 were adopted to make the above equations easier to visualize. In addition vehicle yaw moment and cornering power were normalized, and I_{zN} was set to 1 for simplicity.

In the equations below, T_{cmp} is the torque that compensates for the influence of vehicle dynamics.

$$I_z = I_{zN} m l_f l_r \quad (8)$$

$$l_f = l(1 - D_{wf}) \quad (9)$$

$$l_r = l D_{wf} \quad (10)$$

$$K_f = C_f m D_{wf} g \quad (11)$$

$$K_r = C_r m(1 - D_{wf}) g \quad (12)$$

INFLUENCE OF VEHICLE DYNAMICS ON STEERING SYSTEM AND COMPENSATION

To study what influence vehicle dynamics exert on the steering system, the first step was to consider just the vehicle model only, supplemented by equation 6, which calculates the steer angle applied to the steering system. Next, equations 1 to 6 were solved to determine the self-aligning torque (SAT) from the vehicle with respect to the steering wheel angle θ . The SAT applied to the steering system, T_{sat} , forms a quadratic second order transfer function as shown in equation 13. Thus, by substituting in the specifications shown in Table 1 and calculating the frequency characteristics for SAT T_{sat} at 100 km/h, the SAT gain and SAT phase as shown in Fig. 2 can be obtained.

$$T_{sat} = \frac{\xi m D_{wf} g \cdot C_f (s^2 + B_1 \cdot s + B_0)}{s^2 + 2\zeta \omega_n \cdot s + \omega_n^2} \frac{\theta}{N_s^2} \quad (13)$$

$$2\zeta \omega_n = \frac{(C_f + C_r)g}{V}, \quad \omega_n^2 = \frac{C_f C_r g^2}{V^2} + \frac{(C_r - C_f)g}{l}$$

$$B_1 = \frac{C_r g}{V}, \quad B_0 = \frac{C_r g}{l}$$

Table 1

$m = 1600 \text{ kg}, \quad I_{zN} = 1, \quad D_{wf} = 0.55$
$l = 2.7, \quad C_f = 10, \quad C_r = 20, \quad N_s = 15$
$I_{sw} = 0.15 \text{ kgm}^2, \quad \xi = 0.05 \text{ m},$
$V = 100 \text{ km/h}, \quad g = 9.8 \text{ m/s}^2$

The SAT gain forms a trough at approximately 1.4 Hz, while phase lag and phase lead increase at low and high frequencies, respectively. These results were obtained at 100 km/h, but in general terms, the gain trough, and phase lag at low frequencies both increase in accordance with speed. These high-speed characteristics are experienced as fluctuations in control and convergence of the whole system including the steering, making it necessary to provide compensation. In contrast, the gain trough disappears below a certain speed, and the phase lag changes to phase lead. Thus, compensation for the fluctuations in the control and convergence of the whole system is not required at low speeds.

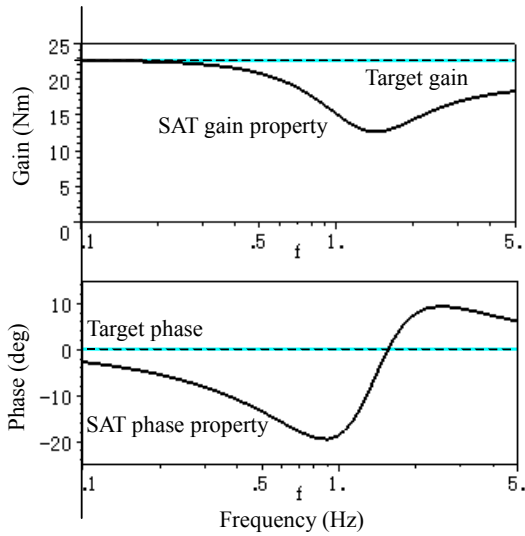


Fig. 2 Frequency Response of Torque

Steering characteristics are influenced by vehicle dynamics as described above, and change in accordance with vehicle speed and steering angle velocity. In response, a compensation control was considered in which target SAT characteristics applied to the steering system maintain a constant gain with no phase lag or phase lead regardless of frequency. The considered target torque was a steady-state solution for SAT obtained by equation 15 with s in equation 13 set to 0. The target gain and target phase shown in Fig. 2 has these characteristics. In other words, the goal is to calculate the compensation torque T_{cmp} that satisfies equation 14. This can be achieved by calculating equations 14 and 15.

$$T_{sat} + T_{cmp} = T_{sat0} \quad (14)$$

$$T_{sat0} = \frac{\xi m D_{wf} g \cdot C_f B_0}{\omega_n^2} \frac{\theta}{N_s^2} \quad (15)$$

$$\begin{aligned} T_{cmp} &= \frac{\omega_n^2 (1 + \tau_1 s)}{(s^2 + 2\zeta \omega_n \cdot s + \omega_n^2)} G \cdot s \cdot \theta \quad (16) \\ &= H(s) s \cdot \theta \\ \tau_1 &= \frac{V(V^2 - C_r g l)}{C_r g (2V^2 - C_r g l)} \\ G &= \frac{C_f^2 C_r m D_{wf} g l V (2V^2 - C_r g l)}{[C_f C_r g l + (C_r - C_f) V^2]^2} \frac{\xi}{N_s^2} \end{aligned}$$

The compensation torque can be obtained from equation 16, and the compensation term is a simple value obtained by multiplying the quadratic first order transfer function to the steering angle velocity.

Thus, it is possible to perform compensation control using a structure that applies a filter to damping control. To simplify the explanation, the compensation control is

calculated with $I_{ZN}=1$, but the calculation can be performed in the same way when I_{ZN} is not 1. However, as this produces complex mathematical expressions, this has been omitted.

Figure 3 shows the changes of G , and $T_1 G$ in accordance with vehicle speed. G is the constant gain from steering angle velocity to the compensation torque and $T_1 G$ is the constant gain from steering angle acceleration.

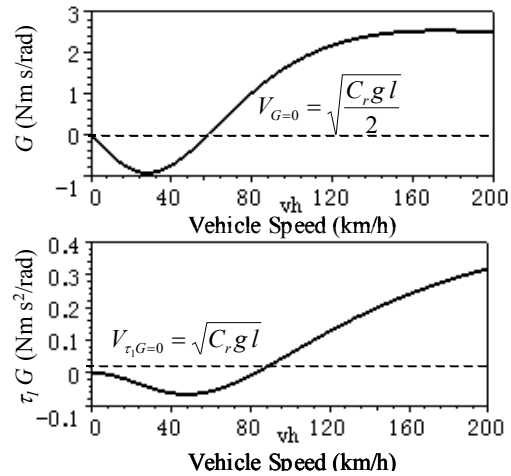


Fig. 3 G and $T_1 G$ Properties with Respect to Vehicle Speed

STRUCTURE OF CONTROL AND VERIFICATION OF EFFECTIVENESS

The description above focused on SAT under steering wheel angle input using only a vehicle model, but it is also necessary to include the steering system. When the dynamic characteristics of the steering system are included, steering torque input is required rather than steering wheel angle input. Therefore, equation 7 is solved as an equation of motion that considers torque input T_{sw} with compensation torque T_{cmp} added to steering system SAT T_{sat} . When the compensation torque T_{cmp} described in the previous section, i.e., equation 16, is substituted into equation 7, and equations 1 to 7 are solved using input torque T_{sw} , the characteristics equations for yaw angle velocity and the transfer characteristics of vehicle body slip angle are obtained as the 4th order of s . However, since the quadratics of the denominators are offset by the quadratics of the numerators, the characteristics equations for steering wheel angle and steer angle can be organized as quadratic expressions.

In this event, the transfer function for the steering wheel angle θ with respect to torque input is obtained as