Evaluation Method for Steer Assist Feeling around Steering Center and Control Design on EPS∗

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Abstract
In this research, a method of designing an EPS control system that achieved a good steer feeling without using an actual vehicle was examined. Firstly, evaluation functions to evaluate steer feeling theoretically were set. And it was described that steer feelings such as “inertial feeling” and “viscous feeling” could be expressed by the evaluation functions. Secondly, a target characteristic that made the steer feeling good was determined on the evaluation function, and control methods to theoretically achieve this target characteristic were described. Finally, the designed control system was installed in an EPS vehicle. As a result, it was confirmed that the steer feeling in an actual vehicle was fine with no “inertial and viscous feeling”.

Key words: EPS (Electric Power Steering), Steer Feeling, Steer Feeling Evaluation, EPS Control, Motion Control, Automobile, Maneuverability

1. Introduction

As shown in Fig.1, EPS is a control system that applies motor power directly to the steering so as to reduce steering torque(1),(2). The motor is driven only when needed, so power loss is able to be reduced compared to conventional hydraulic systems that are constantly driven by the engine. As a result, EPS contributes positively to the environment by enhancing fuel economy, reducing CO₂ emissions and eliminating waste oil. EPS also contributes to the creation of new added value as an applied technology for practical uses such as systems to enhance preventive safety and comfort(3),(4). For these reasons, the use of EPS is increasing worldwide.
On the other hand, EPS has the inertia moment of the motor in the steering shaft. This means that the inertia moment of the motor can have a noticeable effect and degrade the steer feeling when steering operation takes place in the manual steering region around the steering wheel center, for example, during high speed driving. Therefore, various methods were used in the past to limit the influence of the inertia moment on steer feeling by correcting the motor assist signal, such as by applying correction to the motor assist signal using the steering rotation signal (1) or the differentiated steering torque signal (5). The steer feeling was then matched to the vehicle characteristics by adjusting these correction amounts in combination with the assist amount settings.

However, when correcting the assist signal, for example when designing control using control filters, no standard criteria was available for determining the achievement of good steer feeling free from the influence of the inertia moment. Therefore, the control designers had to set correction amounts based on their own experience and steer feeling checks made using actual vehicles in a process of trial and error. As a result, the development of EPS with good steer feeling required an extremely long development process time.

The above-mentioned spread of EPS use called for more efficient development, and methods that enable theoretical design of control systems that achieve a good steer feeling are desired.

From this perspective, authors had proposed a method for theoretically evaluating the influence of the inertia moment of the motor exerted on steer feeling (8), to enable development of a control system that enhances steer feeling without trial and error using actual vehicles. This paper proposes a control design method that uses above evaluation method to obtain a good steer feeling. In addition, an EPS vehicle was manufactured to confirm the validity of these methods, and the results are reported herein.

2. Basic EPS Control

As shown in Fig.1, EPS works to reduce the steering torque $T_s$ by having the assist torque $T_a$, which originates from the motor power, bear part of the load torque $T_w$.

Manual steering operation transmits the steering wheel input angle $\theta_s$ and steering torque $T_s$ directly to the pinion shaft of the rack and pinion gear mechanism via the universal joint. These inputs are converted by the rack and pinion gear into thrust force $F_T$ and displacement in the direction of the rack shaft, and then further converted by the tie rod and knuckle into the pivot angle displacement $\delta_T$ of the tire.
In addition to the manual steering operations noted above, the detected steering torque $T_{\text{det}}$ on the pinion shaft that is detected by the steering torque sensor, and the vehicle speed signal $V_S$ that is detected by the vehicle speed sensor are used by the ECU to determine the assist torque target value that will achieve the optimum steering characteristics. The ECU then performs motor drive control so that the generated torque matches this target value. The torque $T_M$ generated by the motor is boosted by the worm gear and the worm wheel gear, and is applied to the pinion shaft. The assist torque $T_A$ that is obtained in this way bears part of the load torque $T_W$, thus reducing steering torque $T_S$.

When the EPS model shown in Fig.2 is considered in a static manner that ignores the inertia moment, viscosity, the control filter $H(S)$, and other dynamic factors, this model yields the relationships shown in Eqs.(1) to (3) below.

$$T_S = K_T S(\theta_S - \theta_P) \quad (1)$$

$$K_T S(\theta_S - \theta_P) = T_W - T_A \quad (2)$$

$$T_A = n_M T_M$$

$$= n_M G_A T_{\text{det}} \quad (3)$$

These indicate that setting the assist gain $G_A$ to an appropriate value reduces the steering torque $T_S$ to $1/(1 + n_M G_A)$ as shown in Eq.(4).

$$T_S = \frac{T_W}{1 + n_M G_A} \quad (4)$$

### 3. Identifying the Issues

When considered statically without taking into account the inertia moment, viscosity and other dynamic factors, EPS assist operation is as shown in Eq.(4). However, actual EPS systems are influenced by the inertia moment of the motor and by viscosity occurring around the rotating shaft of the motor, so the extent of these influences was examined using the model in Fig.2.

The assist torque $T_A^*$ (ignoring the influence of friction) that can be obtained from the motor is influenced by the inertia moment $J_M$ and viscosity $C_M$ of the motor. Therefore, its relationship to the motor torque $T_M$ is expressed as follows.

$$T_A^* = T_M - \xi, \quad [\xi \equiv J_M \dot{\theta}_M + C_M \dot{\theta}_M] \quad (5)$$

From Eq.(5), the actual assist torque $T_A^*$ is obtained as follows.

$$T_A^* = n_M T_M$$

$$= n_M (T_M - \xi) \quad (6)$$

From this, Eq.(4) can be described as Eq.(7) below.

$$T_S = T_W - T_A^* = T_W - (n_M G_A T_S - n_M \xi)$$

$$\therefore T_S = \frac{T_W + n_M \xi}{1 + n_M G_A} \quad (7)$$

Equation(7) indicates that in EPS, the influence of $n_M \xi$ decreases as $G_A$ increases. When $G_A = 0$ as shown in Fig.3, that is to say when the driver operates the steering wheel around the steering wheel center during high speed driving, $n_M \xi$ manifests itself directly as the “delayed
feeling”, “viscous feeling” and “inertial feeling” described hereafter, and causes the steer feeling to deteriorate. However, no method that allows the degree of influence that \( nM \xi \) has on steer feeling to be theoretically evaluated currently exists, so steer feeling must be confirmed using an actual vehicle. In addition, when designing control systems that correct the influence of \( nM \xi \), the control system must be mounted in an actual vehicle to confirm the effects of the control on steer feeling, which requires an extremely long development process.

Therefore, this study recognized the above matters as the issues, and examined both a method for theoretically evaluating steer feeling, and one for designing a control system that uses this evaluation method to provide good steer feeling. However, the vehicle steer feeling is determined by the relationship between \( T_S \) and \( T_W \) in Eq.(7), so this paper labels the influence of \( nM \xi \) as “a steer assist feeling evaluation item” to distinguish the steer feeling due to the influence of \( nM \xi \) from the vehicle steer feeling.

![Fig. 3 Power assist map](image)

![Fig. 4 Block diagram of EPS system](image)

### 4. Modeling of EPS System and Equations of Motion

Substituting \( \theta_M = nM\theta_P \) and \( \delta_T = \theta_P/nG \) in Fig.2 gives the following equations of motion.

\[
T_S = J_S \ddot{\theta}_S + C_S \dot{\theta}_S + K_{TS} (\theta_S - \theta_P)
\]

\[
n_MT_M = \left( K_{TS} + \frac{K_T}{nG^2} \right) \theta_P + \left( nM^2 C_M + \frac{C_W}{nG^2} \right) \ddot{\theta}_P
+ \left( nM^2 J_M + \frac{J_W}{nG^2} \right) \dot{\theta}_P - K_{TS} \theta_S
\]

\[
T_{det} = K_{TS} (\theta_S - \theta_P)
\]

Given these equations of motion, the block diagram of the EPS model can be described as the EPS Plant within the dotted lines in Fig.4. The transfer functions \( G_S(S) \) and \( G_G(S) \) in Fig.4 can be expressed by Eqs.(11) and (12), respectively.

\[
G_S(S) = \frac{1}{J_S S^2 + C_S S}
\]

\[
G_G(S) = \frac{1}{J_G S^2 + C_G S + K_T/nG^2}
\]
5. Examination of Method for Evaluating Steer Assist Feeling

5.1. Setting the Steer Assist Feeling Evaluation Functions

When evaluating the EPS steer assist feeling (delayed feeling, viscous feeling, inertial feeling) using an actual vehicle, steering torque $T_S$ is input to the steering wheel at a relatively high frequency of approximately 2 Hz, and the steer assist feeling is evaluated from the relationship between the steering angle $\theta_S$ and the steering reaction torque which is equal to the detected torque $T_{det}$ that appears at that time. Therefore, relational Eq. (15) between the steering torque $T_S$ and the detected torque $T_{det}$ and relational Eq. (16) between the steering torque $T_S$ and the steering angle $\theta_S$ were set as evaluation functions for expressing the steer assist feeling. And, the steer assist feeling would be evaluated from frequency responses of these relational expressions and relationship of each other.

$$E_{V1} = \frac{T_{det}}{T_S}$$  \hspace{1cm} (15)

$$E_{V2} = \frac{\theta_S}{T_S}$$  \hspace{1cm} (16)

5.2. Steer Assist Feeling Evaluation Items in the Evaluation Functions

Figure 6 compares the frequency characteristics of evaluation functions $E_{V1}$ and $E_{V2}$ when the assist gain $G_A$, control gain $G_H$ and damping gain $G_D$ in the model in Fig. 2 are all set to zero (hereafter referred to as EPS*; without EPS control), and the frequency characteristics of the evaluation functions of manual steering which removed the motor further from EPS*. The results show a clear difference in the frequency characteristics of $E_{V1}$ and $E_{V2}$ over the range from 1 to 100 Hz.

The control filter $H(S)$ configuration was varied in an actual vehicle equipped with EPS, and the steer assist feeling evaluation items and features of the frequency characteristics of
evaluation functions $E_{V1}$ and $E_{V2}$ at that time were examined. The results showed that the “delayed feeling”, “viscous feeling” and “inertial feeling” in an actual vehicle could be expressed using the following features within the frequency characteristics of evaluation functions $E_{V1}$ and $E_{V2}$.

i. Delayed feeling: The feeling that the steering angle phase is delayed relative to the steering reaction torque is called the delayed feeling, and appears in the phase characteristics of evaluation functions $E_{V1}$ and $E_{V2}$ as the phase delay of $E_{V2}$ relative to $E_{V1}$.

ii. Viscous feeling: The feeling that the steering reaction torque increases with the steering angular velocity due to the influence of viscosity is called the viscous feeling, and appears in the gain characteristics of evaluation function $E_{V2}$ as a region with a $-20$ dB/decade slope (first-order lag).

iii. Inertial feeling: The feeling that the steering reaction torque increases at a particular frequency due to resonance in the spring-inertia moment system is called the inertial feeling, and appears as a resonance peak in the gain characteristics of evaluation function $E_{V1}$.

### 6. EPS Control Design

The EPS steer assist feeling is influenced most by the inertia moment and viscosity of the motor when the assist gain $G_A = 0$, so EPS control must enhance the steer assist feeling particularly in the region where $G_A = 0$ (power-assist dead zone). For this reason, EPS control is designed to obtain a good steer assist feeling without inertial and viscous feelings when $G_A = 0$.

#### 6.1. Setting Control Targets

As mentioned in the previous section, the steer assist feeling evaluation items can be expressed by the frequency characteristics of evaluation functions $E_{V1}$ and $E_{V2}$. The comparison of EPS* and manual steering in Fig.6 shows that in the $E_{V1}$ and $E_{V2}$ phase characteristics from 1 to 10 Hz, EPS* has a greater $E_{V2}$ phase delay relative to $E_{V1}$, and this $E_{V2}$ delay produces a delayed feeling. In addition, in the $E_{V2}$ gain characteristics from 1 to 10 Hz, EPS* has a slope of $-20$ dB/decade, indicating that viscosity produces a viscous feeling. Furthermore, the $E_{V1}$ gain characteristics of EPS* have a peak around 24 Hz, indicating that the influence of the inertia moment produces an inertial feeling.

A delay feeling, viscous feeling and inertial feeling are caused by the influence of $n_M\xi$ (inertia moment and viscosity of the motor). Therefore, it was thought that a good steer assist feeling that does not result in a delayed feeling, viscous feeling or inertial feeling, can be...
obtained by setting characteristics that ensure little difference between $E_{V1}$ and $E_{V2}$ and do not have a first-order lag region or a gain peak within the frequency range up to the $E_{V1}$ and $E_{V2}$ cutoff frequency (approximately 30 Hz), like manual steering which has no $n_M\xi$.

The first-order lag and resonance-induced gain peak that appear in the frequency characteristics are determined by the pole-zero plots of the transfer functions. When the pole-zero plots have real poles, first-order lag characteristics appear. When the pole-zero plots have complex conjugate poles and the ratio between the real part and the imaginary part (the damping ratio) is small, a resonance peak appears. In addition, Fig.4 shows that $E_{V1}$ and $E_{V2}$ have identical open loops, so they have identical poles and only the zero points differ. Therefore, the $E_{V1}$ and $E_{V2}$ frequency characteristics are shaped using the pole-zero plots that is straightforward to lay down a design principle because control targets can be set as regions.

Examination of the $E_{V1}$ and $E_{V2}$ frequency characteristics of manual steering (Fig.6) and the pole-zero plots in the frequency range up to 30 Hz shown in Fig.7 reveals that the shape of the frequency characteristics is largely determined by the poles at $-30\pm27i$. In addition, these poles have a large damping ratio of 0.74, so the pole-zero plot shows that it does not have a peak.

Therefore, the following three items were set as control design targets to be achieved within the $E_{V1}$ and $E_{V2}$ pole-zero plots.

1) Reduction of delayed feeling:
Making the $E_{V1}$ and $E_{V2}$ pole-zero plots roughly the same up to the nearest pole from the origin that is not canceled by zero points to reduce the difference between the $E_{V1}$ and $E_{V2}$ characteristics.

2) Reduction of viscous feeling:
Making the above-mentioned nearest pole from the origin multiple pole or complex conjugate poles so that $E_{V2}$ does not have first-order lag.

3) Reduction of inertial feeling:
When the pole-zero plot has complex conjugate poles, making the size of the imaginary part 1.73 times or less than that of the real part (damping ratio $>0.5$; region within the dotted lines in Figs.7 to 9) so that the peak gain is sufficiently small (1.2 dB or less).

6.2. Control Design Method
In the pole-zero plots of evaluation functions $E_{V1}$ and $E_{V2}$ for EPS* (Fig.8), the $E_{V1}$ pole at $-11$ is canceled by a zero point, but the $E_{V2}$ pole at $-11$ is not canceled and remains, so first-order lag characteristics appear only for $E_{V2}$. In addition, the poles at $-20\pm146i$ are located outside the dotted line region, and cause resonance.

Fig. 7 Pole-zero plots of $E_{V1}$ and $E_{V2}$ of manual steering
When changing these pole-zero plots, according to the block diagram shown in Fig.4 ($G_H = 1, G_A = G_D = 0$), $E_{V1}$ and $E_{V2}$ are expressed by Eqs.(17) and (18) below, and the $E_{V1}$ zero points cannot be moved.

\[
E_{V1} = \frac{G_S(S)K_{TS}}{1 + K_{TS}G_G(S)[nM_G(S)H(S) + 1] + G_S(S)K_{TS}} \quad (17)
\]

\[
E_{V2} = \frac{G_S(S)(1 + K_{TS}G_G(S)[nM_G(S)H(S) + 1])}{1 + K_{TS}G_G(S)[nM_G(S)H(S) + 1] + G_S(S)K_{TS}} \quad (18)
\]

Therefore, to eliminate the viscous feeling and inertial feeling of EPS*, control must be performed to change the pole-zero plots so that $E_{V2}$ has a zero point at $-1$ and to move the poles at $-20 \pm 146i$ to the inside of the dotted lines in Fig.8.

![Fig. 8 Pole-zero plots of $E_{V1}$ and $E_{V2}$ of EPS*](image1)

![Fig. 9 Pole-zero plots of $E_{V1}$ and $E_{V2}$ with control](image2)

To satisfy these demands, a control filter $H(S)$ that shapes the $E_{V1}$ and $E_{V2}$ transfer functions as shown in Eqs.(19) and (20) was appropriately derived and applied to EPS*.

\[
E_{V1} = \frac{(S + 11)\cdots}{(S + 11)(S + 20 + 146i)(S + 20 - 146i)\cdots} \Rightarrow \frac{(S + 11)\cdots}{(S + 11)(S + 29 + 34i)(S + 29 - 34i)\cdots} \quad (19)
\]

\[
E_{V2} = \frac{(S + 11)(S + 20 + 146i)(S + 20 - 146i)\cdots}{(S + 11)(S + 29 + 34i)(S + 29 - 34i)\cdots} \Rightarrow \frac{(S + 11)\cdots}{(S + 11)(S + 29 + 34i)(S + 29 - 34i)\cdots} \quad (20)
\]
As a result, the pole-zero plots of $E_{V1}$ and $E_{V2}$ with control changed as shown in Fig.9. The pole that remained at $-11$ only for $E_{V2}$ was canceled by zero points for both $E_{V1}$ and $E_{V2}$, and the $E_{V1}$ and $E_{V2}$ pole-zero plots up to the nearest complex conjugate poles ($-29 \pm 34i$) from the origin that were not canceled by zero points were confirmed to be approximately the same. In addition, the poles at $-20 \pm 146i$ were moved to $-29 \pm 34i$, and the size of the imaginary part became 1.73 times or less than that of the real part (damping ratio $> 0.5$). This achieved all three items set as control targets.

![Fig. 10 Frequency characteristics of $E_{V1}$ and $E_{V2}$ with control](image)

6.3. Verification of Control Effects

Figure 10 shows the frequency characteristics of $E_{V1}$ and $E_{V2}$ when control that achieved the pole-zero plot control targets was applied. The results show that good $E_{V1}$ and $E_{V2}$ characteristics without an $E_{V2}$ phase delay relative to $E_{V1}$, an $E_{V2}$ first-order lag region, or a resonance peak in the $E_{V1}$ gain characteristics were obtained in the range up to several tens of Hz that influences the steer assist feeling. These frequency characteristics are similar to the frequency characteristics of the manual steering shown in Fig.6. So, it was confirmed that the control filter $H(S)$ suppressed the influence of $n_M\xi$ (the inertia moment and viscosity of the motor).

![Fig. 11 Gain characteristics of $E_{V1}$ and $E_{V2}$ in an actual vehicle](image)

Therefore, this control was applied to an actual EPS vehicle, and the $E_{V1}$ and $E_{V2}$ gain characteristics at when impulse shaped steering torque was inputed to the steering wheel were measured by FFT analysis. Figure 11 shows the results. This confirmed that even in an actual vehicle, $E_{V1}$ and $E_{V2}$ characteristics that roughly match the calculated values (Fig.10) are obtained. In addition, the results of evaluating the steer assist feeling in an actual vehicle confirmed that an extremely good steer assist feeling equivalent to that of manual steering was obtained, and that there was no delayed feeling, viscous feeling or inertial feeling whatsoever, which had not been possible to date with EPS control.
7. Conclusion

A study was made of a method for theoretically designing an EPS control system that obtains a good steer assist feeling, and the following conclusions were reached.

1) The steer assist feeling can be evaluated using two transfer functions: the transfer function \( EV_1: T_{det}/T_S \) from the steering input torque to the steering reaction torque, and the transfer function \( EV_2: \theta_S/T_S \) from the steering input torque to the steering angle.

2) The control design targets within the \( EV_1 \) and \( EV_2 \) pole-zero plots for obtaining good steer assist feeling are: “Making the \( EV_1 \) and \( EV_2 \) pole-zero plots roughly the same up to the nearest pole from the origin that is not canceled by zero points”, “Making the above-mentioned nearest pole from the origin multiple pole or complex conjugate poles”, and “When the pole-zero plot has complex conjugate poles, making the size of the imaginary part 1.73 times or less than that of the real part (damping ratio > 0.5)”.

3) The effects of control that achieved the pole-zero plot targets were confirmed in an actual vehicle, and the results showed that the system obtained good steer assist feeling without a delayed feeling, viscous feeling or inertial feeling.

The above confirmed the effectiveness of the proposed steer assist feeling evaluation method using \( EV_1 \) and \( EV_2 \), and the control method that set pole-zero plot targets. This makes it possible to theoretically design control to obtain good steer assist feeling.

References


Nomenclature

\( T_S \): Steering torque [Nm]
\( T_{det} \): Detected torque [Nm]
\( T_W \): Load torque [Nm]
\( T_A \): Assist torque [Nm]
\( T_A^* \): Effective assist torque [Nm]
\( T_M \): Motor torque [Nm]
\( T_M^* \): Effective motor torque [Nm]
\( \theta_S \): Steering angle [rad]
\( \theta_P \): Pinion shaft angle [rad]
\( \theta_M \): Motor angle [rad]
\( \delta_T \): Tire pivot angle [rad]
\( V_S \): Vehicle speed [km/h]
\( E_{V1} \): Evaluation function 1
\( E_{V2} \): Evaluation function 2
\( K_{TS} \): Torsion-bar stiffness
  \( 3.2 \times 10^2 \) [Nm/rad]
\( K_T \): Tire rubber stiffness
  \( 1.25 \times 10^4 \) [Nm/rad]
\( J_S \): Steering wheel inertia
  \( 5.56 \times 10^{-2} \) [kgm\(^2\)]
\( J_W \): Tire inertia
  \( 0.600 \) [kgm\(^2\)]
\( J_M \): Motor inertia
  \( 1.29 \times 10^{-4} \) [kgm\(^2\)]
\( C_S \): Steering shaft viscosity
  \( 0.294 \) [Nms/rad]
\( C_M \): Motor viscosity
  \( 9.81 \times 10^{-3} \) [Nms/rad]
\( C_W \): G/Box & suspension viscosity
  \( 5.58 \times 10^{2} \) [Nms/rad]
\( n_M \): Reduction gear ratio
  \( 27 \)
\( n_G \): Steering ratio
  \( 15.6 \)
\( G_A \): Assist gain
\( G_H \): Control gain
\( G_D \): Damping gain [Nms/rad]
\( T_{MT} \): Target torque [Nm]
\( H(S) \): Transfer function
  (EPS compensator)
\( G_M(S) \): Transfer function
  (Motor-torque control)