Analysis on the Stability and Anti-Interference Performance of EHPS System in Pure Electric Bus

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Abstract

This paper studies the stability and anti-interference performance of the EHPS system of the pure electric bus, analyzes the influence of the disturbance of road excitation and the sensor's noise measurement on system performance, combines the traditional PD control algorithm and the H∞ control theory and applies it to the research of steering control system and design of the controller. The computer simulation results suggest that EHPS system of pure electric bus designed boasts good stability, and can effectively inhibit the influence of the disturbance of high-frequency road excitation and the measurement noise of the torque sensor on the system.

Key words: STEERING SYSTEM, STABILITY, ANTI-INTERFERENCE, H∞ CONTROL THEORY

1. Introduction

With the rapid development of electronic and computer techniques, electromechanical integration technique has been applied more and more to autos[1]. The application of electromechanical integration technique has greatly improved the quality of auto products. Auto steering technique has undergone the development process from mechanical steering to power steering[2]. Currently, the hydraulic power steering technique has been widely used[3]. To further improve the performance of the dynamic steering system, EHPS (Electro Hydraulic Power Steering) system of the pure electric bus featuring mechatronic characteristics has become the current research focus[4].China is a late-starter researching into the EHPS system of the pure electric bus, still being in the exploration period of hardware development and control methods[5]. Outside China, foreign scholars mainly adopted PID (proportion, integration and differentiation) control algorithm to study the electric power steering control system in the past[6]. The electric power steering system developed can achieve basic assistance for steering, but the assistance torque is small and the assistance scope is narrow, which is confined to the steering assistance for vehicles driving in low speed[7], and its stability and anti-interference performance is poor. In recent years, many foreign scholars have been exploring about how to apply advanced control theories to the research into the EHPS system of the pure electric bus, such as applying H∞ control theories, fuzzy control theories, fuzzy PID control theories and neural network control theories to the steering control system, and about how to obtain different steering road senses, inhibit measurement noise of the sensor, improve the robustness and tracking performance of the system, and achieve aligning control and so on.

Stability and anti-interference are important indexes to evaluate the performance of the EHPS system of the pure electric bus, which have a close bearing on autos' driveability. An instable steering system with poor anti-interference performance is vulnerable to the influence of the external disturbance, causing vibration of the steering wheel and reducing the comfortability and safety of driving. Influenced by the disturbance of high-frequency road excitation and the measurement noise of the sensor during steering, the EHPS system of the pure electric bus is interfered in terms of its output, causing a greatly influence on the steering operation. Thus, to improve the stability of the steering system and inhibit the disturbance of high-frequency road excitation and sensor's measurement noises are the research focus of the EHPS system of the pure electric bus[8].

This paper studies the stability and anti-interference of the EHPS system of the pure electric bus, analy-zes the causes of the system instability, the influence of the disturbance of high-frequency road excitation and sensor's measurement noise on the steering system[9]. In order to effectively inhibit such influence, this paper combines the PD control and the H∞ control theories to design the control strategies for the electric power steering system.

2. Description of the EHPS system of the pure electric bus

2.1. Description of the system's structural model

The structural model of the EHPS system of the pure electric bus is shown in Fig. 1. The steering system belongs to the pinion-and-track EHPS system assisted by the steering shaft of the pure electric bus. In Fig. 1, T_s stands for the steering moment of the steering wheel; T_m is the assisting torque of the electric machine; T_e stands for the steering drag torque imposed on the steering gear; k_c stands for the stiffness of the torsion bar; b_c stands for the damping efficient of the steering shaft. After the assisting torque of the power is augmented by the reducer, it is imposed on the steering shaft. The value of assisting torque is related not only to the steering moment of the steering wheel and the driving speed of autos. At different speeds, the EHPS system of the pure electric bus should work according to different assisting force ratios so as to achieve a favorable driveability. Since the steering moment of the steering wheel cannot be directly measured, the torque sensor installed on the steering shaft can be employed to approximately calculate the steering moment of the steering wheel through the measurement of the torsion bar and the torsion angle. The control goal of the EHPS system of the pure electric bus is to reduce the steering moment of the steering wheel, and to improve the stability and anti-interference of the steering system.

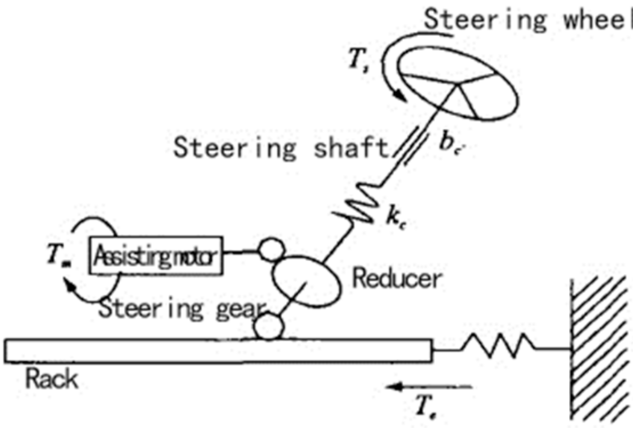

Figure 1. Structural diagram of the EHPS system of the pure electric bus

2.2. Assisting characteristics of the EHPS system of the pure electric bus

In order to obtain good driveability, the value of the steering torque of the steering wheel should be different at different speeds. In terms of steering at a low speed or in the original place, the steering moment of the steering wheel should be as small as possible so as to achieve convenient steering. In terms of steering at a high speed, the steering torque of the steering wheel should be relatively large so as to prevent the steering waving. Fig. 2 is the MAP chart showing the assistance characteristics of the EHPS system of the pure electric bus. From Fig. 2, it can be seen that, with the decrease of the vehicle speed, the ratio (assisting ratio) of the assisting torque to the steering moment of the steering wheel gradually increases so as to achieve convenient steering at a low vehicle speed. Since the steering drag torque decreases along with the increase of the vehicle speed, in order to obtain a relatively stable road sense of steering, the assisting ratio should gradually decrease along with the decrease of the vehicle speed so as to ensure a relatively large steering torque at a high vehicle speed. The assisting characteristics MAP chart is not exclusive. Through the change of the assisting ration at different vehicle speeds, different steering road senses can be achieved.

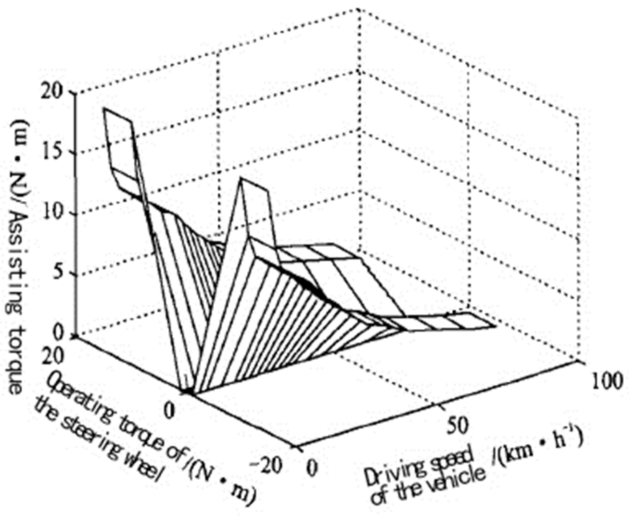

Figure 2. Assisting characteristics MAP chart

3. Analysis of the system's stability

During steering, the steering drag torque imposed on the steering pinion is imposed on the steering wheel through the steering shaft, leading to the generation of the reverse torque of the steering wheel. The reverse torque changes of the steering wheel directly reflect the road sense of steering. If the EHPS system of the pure electric bus is an instable system, the reverse torque will undergo huge vibration during steering to cause vibration of the steering wheel. To study the changes of the reverse torque of the steering wheel under the influence of the steering drag torque can contribute to the analysis of the stability of the EHPS system of the pure electric bus. In order to study the stability of the pure electric bus, the steering wheel is fixed. At the moment, the stability of the system can reflect the stability of the system during steering. Based on Fig. 1, the differential equation of the dynamic characteristics of the steering shaft when the steering wheel is fixed:

$$J_c\theta_c = T_e - T_m - b_c\theta_c - k_c\theta_c \quad (1)$$

$$T_s = -T_r = -k_c\theta_c \quad (2)$$

Where, J_c stands for equivalent rotational inertia of the steering shaft; θ_c stands for the angle of the torsion bar; T_r stands for the reverse torque of the steering wheel. The relationship between the steering assisting torque, T_m, and the armature current of the assisting motor is shown below:

$$T_m = ikl \quad (3)$$

Where, *i* and *k* stands for the transmission ratio of the reducer and the moment coefficient of the motor.

Based on the assisting characteristic MAP chart, it can be learned that, in terms of steering at given vehicle speed, the assisting torque and the steering moment of the steering, and the assisting torque and the steering torque of the steering wheel are both positively related to each other. The proportion control algorithm is adopted. The proportion coefficient is decided according to the assisting ratio. The relationship between the current of the assisting motor and the angle of the torsion bar is as below:

$$I = k_c\theta_c \quad (4)$$

Where, k_p stands for the proportion coefficient.

Based on Eq. (1) and Eq. (4), the transfer function between T_e and the reverse torque of the steering wheel, T_r, is:

$$G(S) = \frac{K_c}{J_cS^2 + b_cS + (K_c + ikk_p)} \quad (5)$$

From Eq. (5), it can be seen that the system is a second-order one. When the assisting ratio at a medium speed assisting ratio in Fig. 3 is "1," the amplitude-frequency characteristic of the transfer function, G(jk), shows that the transmission gain at 61.2 rad/s reaches the maximum; the system's bandwidth is about 95 rad /s. The system's bandwidth is relatively wide, which allows the high-frequency components of the input torque on the pinion to be conveyed to the steering wheel. The high-frequency components are in essence the disturbance of the high-frequency road excitation. During steering, the disturbance of high-frequency road excitation will destroy the system's stability, causing the decrease of driving comfortability and the vibration of the steering wheel. Therefore, to confirm the current of the assisting motor just based on the ratio of the motor current and the angle of the torsion bar cannot meet the practical steering demands. The control algorithm must be improved.

Since the EHPS system of the pure electric bus is a second-order one, the increase of the system's damping is an important approach to improve the system's stability. Differentiation control can increase the system's damping and improve the system's stability, so differentiation control can be increased to improve the control algorithm under the prerequisite of maintaining the proportional gain unchanged. In other words, PD control algorithm is employed to decide the current of the assisting motor. After the increase of the differentiation control, the relationship between the current of the assisting motor and the angle of the torsion bar is shown below:

$$I = k_p\theta_c + k_d\theta_c \quad (6)$$

Where, that is the differential coefficient.

The relative stability of the system can be improved through the choice of a proper differential coefficient. Fig. 3 is the system's amplitude-frequency characteristic chart after the increase of the PD control. From Fig. 3, it can be seen that, the gain at 61. 2rad/s is weakened; the system's bandwidth is obviously reduced; the transmission of the high-frequency components of the input torque of the steering pinion is inhibited, thus reducing the influence of the disturbance of high-frequency road excitation on the EHPS system of the pure electric bus. Thus, the stability of the EHPS system is significantly improved after the adoption of the PD control.

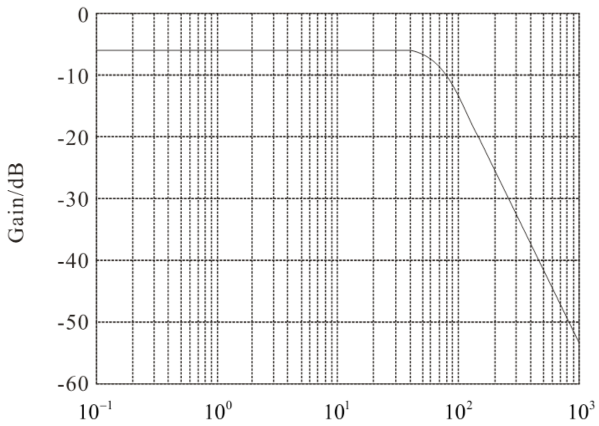

Figure 3. Amplitude-frequency characteristic chart (under the differentiation control)

Fig. 4 shows the time domain response of the angle of the torsion bar to the unit step input of the measurement noise of the torque sensor. From Fig. 4, it can be seen that the unit step input causes the 0.5rad steady-state output of the angle of the torsion bar. The steady-state output is relatively huge, suggesting that PD control cannot effectively inhibit the influence of the measurement noise of the torque sensor on the system.

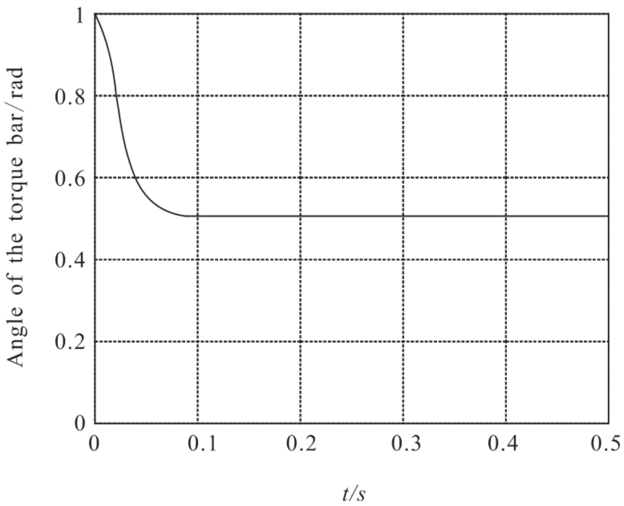

Figure 4. Time domain response of the angle of the torsion bar to the unit step input of the measurement noise of the torque sensor

4. H∞ controller design based on PD control

H∞ optimal control theory is an effective approach to improve the system's anti-inference performance, which adopts the H∞ norm of the transmission function between certain signals within the control system as the optimization performance index. In order to inhibit the influence of the measurement noise of the torque sensor and the disturbance of the road excitation, H∞ optimal control theory is applied to studying the anti-inference performance of the EHPS system of the pure electric bus, and designing H∞ controller based on the PD control. Fig. 6 is the system chart of the electric steering control system based on the PD control algorithm.

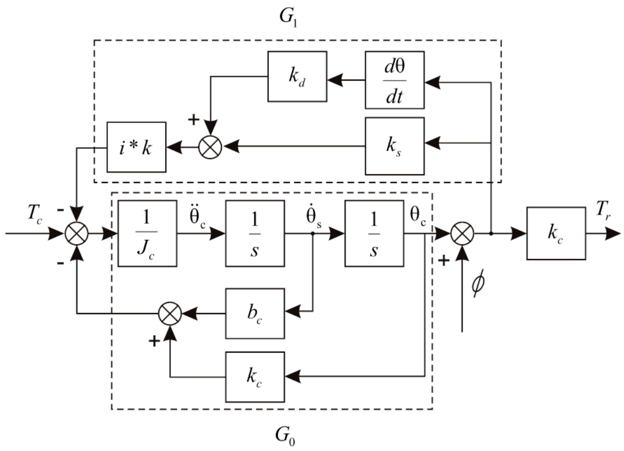

Figure 5. Control system chart

In Fig. 5, Φ stands for the measurement noise of the torque sensor. The following relationship can be found from Fig. 5:

$$G_0 = \frac{1}{J_c S^2 + b_c S + k_c} \tag{7}$$

$$G_1 = \mathrm{ik}\left(k_p + k_d s\right) + \tag{8}$$

Assuming that the measurement noise of the torque sensor refers to the system disturbance input formed by a disturbance signal, g, through the weight function, W (s), namely O=W(s)g. The frequency characteristic of the weight function W (s) reflects the frequency characteristic of the measurement noise of the sensor. It is defined that y is the measurement output of the angle of the torsion bar. "$y=\theta_c+Q_z$" stands for output quantity to be invoked, and z=y; K(s) is the controller; u stands for the control input, and u=K(s) y; the external input signal is w= $[g\ T_e]^T$. The structural chart of the H∞ optimal control is shown in Fig. 6. According to Fig. 6, the following relationship expression can be obtained.

$$y = \begin{bmatrix} \frac{W}{1+G_1G_0} & \frac{G_0}{1+G_1G_0} \end{bmatrix} \begin{bmatrix} \varsigma \\ T_e \end{bmatrix} - \frac{W}{1+G_1G_0} u \tag{9}$$

$$z=\begin{bmatrix}\dfrac{W}{1+G_1G_0} & \dfrac{G_0}{1+G_1G_0}\end{bmatrix}\begin{bmatrix}\varsigma \\ T_e\end{bmatrix}-\dfrac{G_0}{1+G_1G_0}u \tag{10}$$

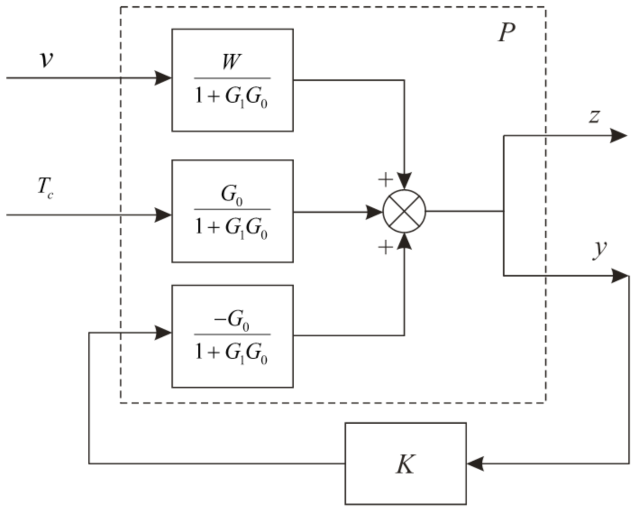

Figure 6. Structural diagram of H∞ control

The transfer function matrix of the controlled object in broad sense can be obtained based on Eq. (9) and Eq. (10):

$$P=\begin{bmatrix}\begin{bmatrix}\dfrac{W}{1+G_1G_0} & \dfrac{G_0}{1+G_1G_0}\end{bmatrix} & \dfrac{-G_0}{1+G_1G_0} \\ \begin{bmatrix}\dfrac{W}{1+G_1G_0} & \dfrac{G_0}{1+G_1G_0}\end{bmatrix} & \dfrac{-G_0}{1+G_1G_0}\end{bmatrix} \tag{11}$$

Adopt the H∞ norm of the transfer function from the measurement noise of the torque sensor and the steering drag torque to the angle of the torsion bar is optimization performance index of the optimal H∞ design; employs comprehensive tools of the H∞ controller based on the linear matrix in equivalent expression handling method provided by MATLAB and LMI to design the optimal H∞ controller, K (s).

5. Simulation of H∞ control

The assisting ratio at a given vehicle speed is decided by the assisting characteristic MAP chart; the proportion coefficient and the differentiation coefficient of the PD control is decided by the assisting ratio. After the confirmation of the proportion coefficient and the differentiation coefficient, the corresponding H∞ controller, K (s), of the vehicle speed can be worked out. Therefore, different vehicle speeds are corresponding to different proportion coefficients, differentiation coefficients and H∞ controllers. The major parameters of the system are shown in Table 1. The controller is designed with the assisting ratio as "1," and the designed controller is:

$$\mathrm{K}(s)=\frac{1.5ls+41.7}{s^2+12ls+14260} \tag{12}$$

Based on the PD control algorithm, the optimal H∞ controller, K (s), is added, and the computer simulation is conducted of the control system. Fig. 7 stands for the time domain step response of the angle of the torsion bar to the step input of the measurement noise of the torque sensor. From Fig. 7, it can be seen that compared with the simple PD control algorithm, the unit step torque of the measurement noise of the torque sensor leads to the steady-state output of the angle of the torsion bar from 0.5rad to zero, which effectively inhibits the influence of the measurement noise of the torque sensor on the reverse torque of the steering wheel. Fig. 8 shows the time domain response of the angle of the torsion bar to the input torque of the step pulse input of the steering pinion. From Fig. 8, it can be seen that the vibration changes of the angle of the torsion bar caused by the unit pulse input of the steering gear are effectively inhibited, which will not change the vibration of the reverse torque of the steering wheel. This suggests that, after the increase of the H∞ optimal controller, K (s), the electric power steering control system still has a favorable stability and resistance against the disturbance of the road extraction.

Table 1. Electric power steering system parameters

Parameters	Symbols	Numerical value
Equivalent rotational inertia of the steering shaft /(kg°m²)	J_c	0 06
Damping coefficient of the steering shaft /(N°m°(rad°s⁻¹)⁻¹)	b_c	0 36
Rigidity of the torsion bar /(N°m°rad⁻¹)	k_c	123

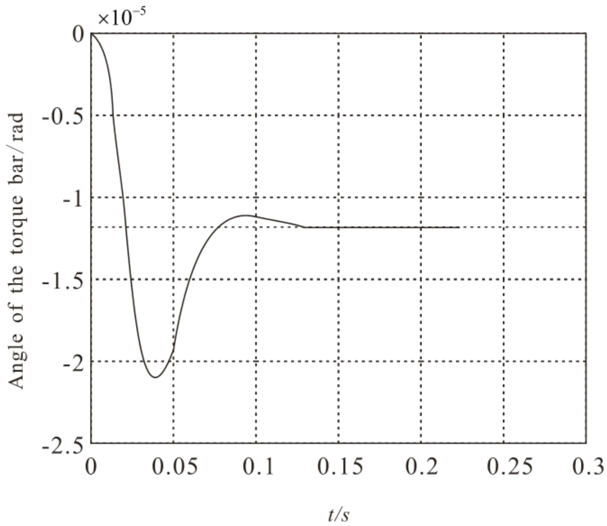

Figure 7. Time domain response of the angle of the torsion bar to the unit step input of the measurement noise of the torque sensor (under the H∞ control)

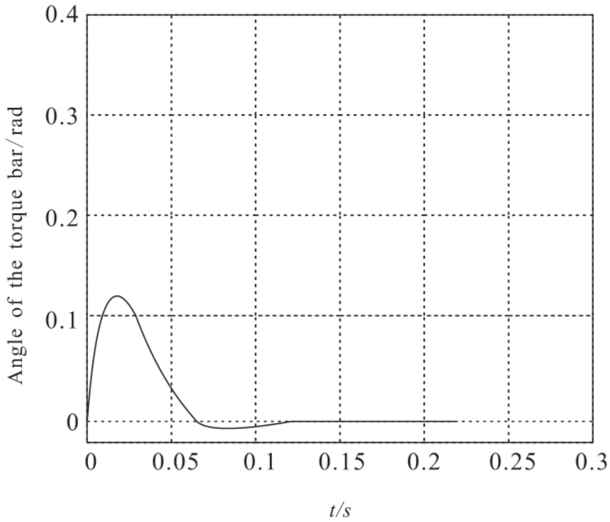

Figure 8. Time domain response of the angle of the torsion bar to the unit step pulse input torque

6. Conclusions

Stability and anti-inference are important indexes to evaluate the performance of the EHPS system of the pure electric bus. The instability of the rotational system can cause the vibration of the steering wheel to directly influence the driveability of autos. This paper adopts the fixed steering wheel to study the stability and the anti-inference of the EHPS system of the pure electric bus. PD control algorithm can significantly improve the relative stability of the system and inhibit the disturbance of the high-frequency road excitation, but its inhibiting effect of the measurement noise of the torque sensor is not obvious. To solve the problem, H∞ optimization control theory is applied to studying the anti-inference of the EHPS system of the pure electric bus. Based on the PD control, H∞ optimization controller is designed. The simulation results suggest that the control combining PD control and H∞ optimization control can effectively improve the stability of the EHPS system of the pure electric bus, and inhibit the influence of the disturbance of the high-frequency road excitation and the measurement noise of the torque sensor on the steering system.

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