Analysis of assist characteristic of electric hydraulic power steering system

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Abstract

According to vehicle handling dynamics and multi-body dynamics, the dynamic steering torque geometric model considered the change of the front wheel ground was established. The vehicle mathematical model and the tire model for vehicle steering system were analyzed. Fuzzy PID control method was used to control influence assist steering of electric hydraulic power steering (EHPS). The results show that control method of EHPS achieves the adaptive change of assist steering. This method enhances the stability of driving vehicle and steering portability.

Keywords: geometric model, fuzzy PID, assist steering, EHPS

1 Introduction

Automobile steering performance directly affected vehicle handling stability. Alone with the rapid development of automobile electronic technology, automobile steering performance requirements were increasing, automotive steering systems from traditional mechanical steering, hydraulic power steering systems to electric control steering system [1-7].

Traditional hydraulic power steering system reduced the driver's labor intensity and improved steering sensitivity and driving safety. But there were also inadequate. Such as hydraulic power steering system was driven by the engine. It had been running even without steering the oil pump, which increased energy consumption. Traditional hydraulic power steering system required inconsistent with the actual driving. The ideal power steering should decrease with increasing vehicle speed to enable the driver to maintain a certain sense of the way in driving the process, to ensure vehicle handling and stability. In fact, when the vehicle was traveling, the greater assist power followed with the greater engine speed. Because the greater the actual flow power steering pump. When the pressure was greater than the pressure relief valve piping, some fluid back through the relief valve tank, not only caused the oil temperature rises, reducing the hydraulic power steering system energy efficiency, but also cause energy loss. Hydraulic steering system for conventional problems was resolved by electronic control unit controlling hydraulic system, which can maintain reliability of the original model of steering, while reducing energy consumption, the purpose of the precise position of the steering control. Electric hydraulic power steering system utilized a conventional hydraulic power steering system big power steering and reliable used in large buses.

Electric hydraulic power steering system provided the appropriate power according to vehicle speed and traffic information size [8]. For the inner front wheel steering caster and kingpin inclination affected, the tire with the front axle ground was changed. It resulted lateral force is applied aligning torque on the tire change. Hydraulic booster could not provide help according to the actual road conditions.

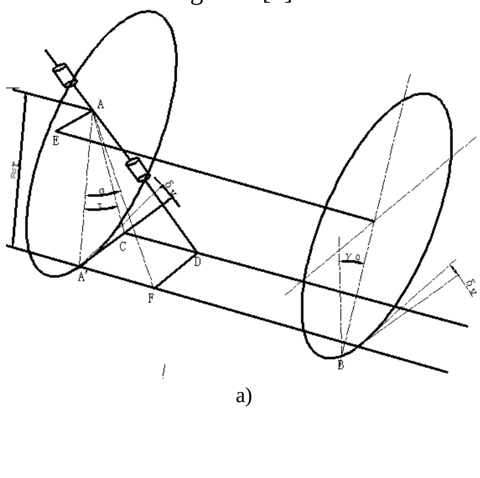
Aimed at the vehicle in return performance calculation did not consider changing tire ground, electric hydraulic pressure caused by providing power value calculation is not accurate, which influenced assist power control. This paper discussed the change power value for the tire ground when the electric hydraulic power steering.

2 Mathematical model of EHPS

Steering resistance torque were composed of aligning torque of the tire generated by the cornering force, the aligning torque generated by the gravity of the structure of the steering system, the internal friction torque of the steering system and the friction torque between tire and road surface. The acting force was not on ground point of the wheel because of the relationship of the kingpin caster angle and the kingpin front the ground. It generated the steering resistance was bigger.

2.1 GEOMETRIC RELATIONSHIP OF THE STEERING SYSTEM

The lateral force of the wheel forced on tires, torque was generated because of the distance between the lateral force and wheel axis. It was necessary to get the geometrical relationship. Geometry relationship of the front wheel and the kingpin was shown in Figure 1 [9].



a)

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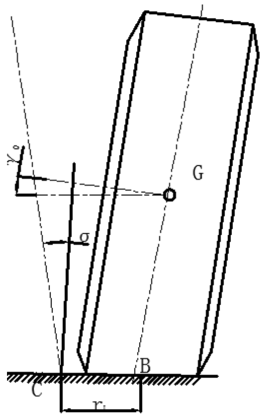


b)

FIGURE 1 Geometry relationship of thefront wheel and the kingpin

The kingpin inclination angle σ and caster angle τ were generated by the inclination and retroversion of the kingpin when the wheel was rotating the kingpin. The kingpin front the ground generated by the kingpin inclination angle improved the resistance torque acting on the wheel, and the torque increased with the distance of the kingpin front the ground. According to the geometric relationship, the kingpin front the ground was generated by the rear axle and kingpin caster angle, its formula was as follows:

$$n_K(\delta_V) = n_0 + r\tan\tau + r[\tan\sigma\sin\delta_V - (1-\cos\delta_V)\tan\tau]. \quad (1)$$

2.2 VEHICLE MODELS

The ground counterforce to the wheels forced by tire on the vehicle. There were wheel force in the forward direction for the tangential force F_{xV} and the vertical direction F_{xH}. There were the wheel force to the side force F_{yV} and F_{yH}. In this paper, steering radius r, vehicle quality m, front wheel steering angle δ_V were presumed. For the automobile steering, there was an angle β in the vehicle speed direction and longitudinal axis of vehicle and the vehicle yaw angle φ. The vehicle affected by lateral wind on the steering was not considered and vehicle dynamics model of centroid was thought on the road plane. The kinetic equation of vehicle was obtained.

The equation for the vehicle force balance in the longitudinal axis was as follows:

$$\begin{gathered} m\frac{v^2}{r}\sin\beta - m\dot{v}\cos\beta + F_{xH} - \\ F_{Lx} + F_{xV}\cos\delta_V - F_{yV}\sin\delta_V = 0. \end{gathered} \quad (2)$$

The equation for perpendicular to the vehicle longitudinal axis force was as follows:

$$\begin{gathered} m\frac{v^2}{r}\cos\beta + m\dot{v}\sin\beta - F_{yH} + F_{Lx} - \\ F_{xV}\sin\delta_V - F_{yV}\cos\delta_V = 0. \end{gathered} \quad (3)$$

The equation for around the centroid of the moment for vehicle was as follows:

$$J_z\ddot{\varphi} - (F_{yV}\cos\delta_V + F_{xV}c\sin_V)a + F_{yH}b = 0\ . \quad (4)$$

For the vehicle centroid was not on the road, which had a certain distance with road. Its height value was shown h. Torque around the longitudinal axis of the moment was as follows:

$$M = m\frac{v^2}{\rho}h = (F_{yH} + F_{yV})h\ . \quad (5)$$

The vertical load of wheel was changed along the longitudinal axis of the front wheel produced by winding, the outside wheel vertical load increased and the inner wheel vertical load reduced.

Aligning torque value for the gravity induced was decided by the positioning parameters of steering [10]. The calculation formula was as follows:

$$M_G = G(c + r\tan\varphi)\cos\varphi\sin\varphi\sin\delta\ . \quad (6)$$

2.3 TIRE MODELS

Tire pressure impacted grounding mark shape, pressure distribution of print, matrix stiffness and torsional stiffness, mechanical properties. So changed tire pressure impacted the tire mechanical properties and the tire aligning moment effect. Dugoff tire model was used to determine a single tire longitudinal force and lateral force, the mathematical model of the tire was as follows [11]:

$$\begin{cases} F_x = C_s sf(\lambda)/(1+s) \\ F_y = C_\alpha \tan\alpha f(\lambda)/(1+s) \end{cases}, \quad (7)$$

$$\lambda = \frac{\mu F_z(1+s)}{2\sqrt{(C_s s)^2 + (C_\alpha \tan\alpha)^2}}, \quad (8)$$

$$f(\lambda) = \begin{cases} (2-\lambda)\lambda, \lambda < 1 \\ 1, \lambda \geq 1 \end{cases}. \quad (9)$$

In the formula, C_s as the tire longitudinal stiffness; C_α as the cornering stiffness of tire, s as tire longitudinal slip rate, α as the tire side slip angle, λ as the tire model variables, μ as the adhesion coefficient.

3 Fuzzy PID controller design

3.1 PID CONTROLLER DESIGN

PID control method is a classic control algorithm, which had simple algorithm and high reliability. Especially when it applied to the invariant systems whose mathematical model could be established, the control effect was obvious.

The key point of PID controller was the research on the best parameter value. The ratio of the value of the parameter was selected to eliminate the system static error, which was acquired by trial and error that was based on experience and refer to the relevant steering characteristic parameters. The curve of it response was fast and overshoot was small could be obtained by adjusting the factor. Coefficient was adjusted continuously until the satisfactory control effect could be obtained.

3.2 FUZZY CONTROLLER DESIGN

PID control was difficult to controlling the dynamic characteristics and the static features at the same time and the poor anti-interference ability, fuzzy control could overcome these problems. Fuzzy control combined certain knowledge with the process state to decide the control behaviour. It adjusted PID parameters online according to the error signal and the error of differential of the system.

Fuzzy control system consists of fuzzy data, rule, fuzzy logic computer and defuzzifier. Fuzzy control system replaced the analog controller with fuzzy controller, which was different from the traditional closed-loop control system. Fuzzy control applied to control complex nonlinear system. Because it could describe the experience and knowledge of experts as rules, which can be expressed by language variables. Performance merits of the fuzzy control system depended on the structure of fuzzy controller, fuzzy rules, fuzzy reasoning algorithms and methods.

3.3 FUZZY PID CONTROL SYSTEM

In general fuzzy control system, the error and the error rate were often used as the language input variables. It acted like the PD controller. The system had the static error which was difficult to eliminate and at this time the advantages of PID control could be used to supplement. With the control strategy combined by the fuzzy controller and PID controller, steering torque difference was used as a comparison. When a value was greater than set threshold value, the fuzzy control system was used. When the error is less than a certain range, PID control was used to eliminate the system steady-state error [12, 13].

The steering force was used as an input variable by a two-parameter and converted into fuzzy variable. Fuzzy decisions were made by fuzzy control rules, which was based on the fuzzy information and then exact amount could be obtained after the anti-blur. It judged and controlled output hydraulic pressure current. Fuzzy PID control was to identify the fuzzy relationship among the PID three parameters, the error values and the error rate in order to meet the different requirements on control parameters of the control system. So the controlled object had a good dynamic and static performance. The structure of Fuzzy PID controller is shown in Figure 2.

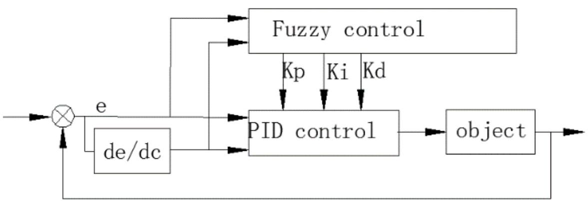


FIGURE 2 Structure of Fuzzy PID controller

Fuzzy control with double fuzzy input and single output selected steering torque difference rate of change of fuzzy quantitative values. Steering torque difference rate variable was described by state of three words. That was small, medium and large. Steering torque difference rate variable fuzzy quantitative values was described by state of three words. That was small, medium and large. The output variable for the steering hydraulic value was three words was also described small, medium and large. Gaussian membership function was used frequently in control engineering, which had has good sensitivity, stability and robustness. The fuzzy controller rules were:

When the steering torque deviation was bigger, the steering hydraulic value was taken large K_p to accelerate the response speed. At the same time, in order to avoid possible differential supersaturated and make the control action was beyond the scope at the beginning of the deviation of the instantaneous larger, the steering hydraulic value was taken a medium K_d. In order to prevent secondary element speed appear larger overshoot and produce integral saturation, the integral action was usually removed, $K_i = 0$.

When the steering torque deviation and deviation rate of change in a medium size K_p, this was in order to make the secondary element rotation speed had a smaller overshoot. K_i was moderate, K_d of the impact on the system response was bigger to ensure the system response speed.

4 Assisting curve design

The characteristic of a certain type of vehicle was considered. For the parameters of different types of vehicles were not identical. In different areas the driver of the same type vehicle preference for manipulation of steering wheel torque was not the same. As the change of speed and lateral acceleration, steering torque was different. The average expected by the driver under a certain speed and lateral acceleration expectations as a design of electric hydraulic booster reasonable curve of the important basis of the steering wheel torque.

Because the maximum steering moment occurred in the vehicle pivot steering vehicle, it must satisfy the steering portability requirements at the power steering. The static steering resisting moment for vehicle was as follows:

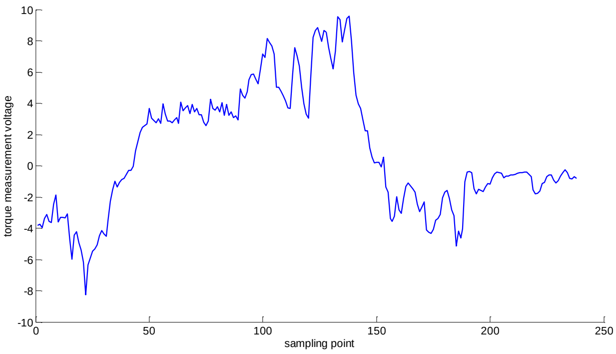
$$T = \frac{\mu}{3}\sqrt{\frac{G_f^{\ 3}}{P}}\,, \tag{10}$$

where G_f as Vertical load of front wheel, μ as tire and the ground friction, P as tire pressure.

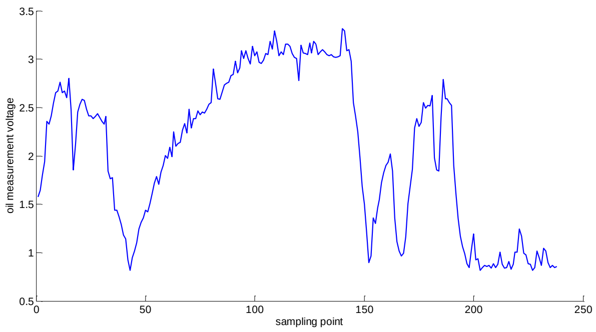
5 Analysis of experimental results

YS6120DG vehicle is a pure electric bus to the real vehicle test. The measurement range of oil pressure sensor used in the tests is 0~12MPa, which the output voltage is 0~5V. The steering wheel torque sensor measurement range is 0±50Nm, which the output voltage of ±12V. The speed signal measurement range is 0~300km/h, which output voltage is 0~5V.The pure electric bus steering experiments of observing changes of speed and torque and oil pressure, its results is showed in Figures 3-4.

Figure 3 and 4 show the experimental results show that when the vehicle turning in place, due to the resistance value, need power steering is increased, the hydraulic pressure is bigger than vehicle runs on the lemniscate road. And as you can see from Figure 3, when the vehicle turns in pivot steering, steering wheel torque with the steering angle increases.

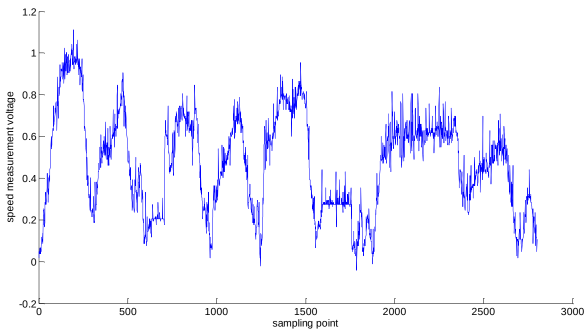


a) torque curve

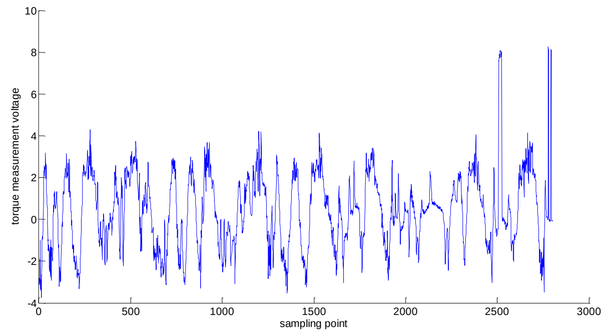


b) oil pressure curve

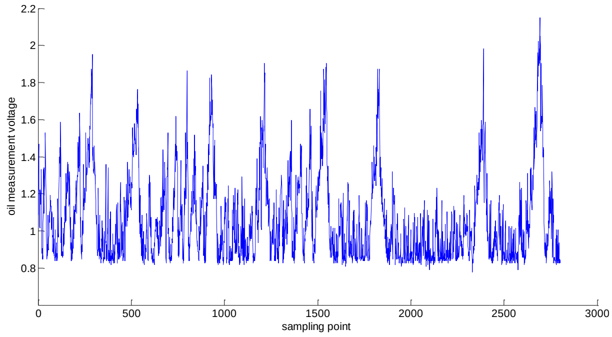
FIGURE 3 Vehicle standing angle change curve



a) speed change curve



b) torque curve



c) oil pressure curve

FIGURE 4 Rotation curve changes vehicle runs on the lemniscate road

6 Conclusions

According to analyze the geometric relationship, the kingpin front the ground was generated by the rear axle and kingpin caster angle influences the wheel turning torque. Through the theory the torque changes is analyzed by the principle of mechanics. It improves the steering portability and stability control.

Acknowledgments

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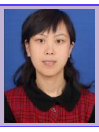	**Zhuan You, January 1977, Jiangsu, China** **Current position, grades**: vice professor at the Haian College of Information Technology. **University studies**: Master degree in vehicle engineering at Jiangsu university of Science and Technology, in 1995. **Scientific interests**: mechanical and electrical integration of vehicle.