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Dynamic Modeling and Steering Performance Analysis of Active Front Steering System

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Abstract

The kinematic models of planetary gear set and steering gear are established, based on the analysis of the transmission mechanism of angle superposition with Active Front Steering system (AFS). A controller of variable steering ratio for Active Front Steering system is designed, and virtual road tests are made in CarMaker driver-vehicle-road simulation environment. The results of simulation tests validate the controller performance and the advantage of variable steering ratio function, also show that the driving comfort is improved at low speed especially, due to the Active Front Steering system alters the steering ratio according to the driving situation.

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Keyword: Active Front Steering system; steering performance; dynamic model; variable steering gear ratio

1. Introduction

The steering system acts a significant role of making car convenient to handle and enhance the vehicle stability. In the past one hundred years, the development of steering system has experienced many stages, and the Steer-by-Wire system (SBW) is the newest technology of steering system for passenger cars. But the Steer-by-Wire system has not yet accepted by public consumers and permitted by state regulations, in consideration of the reliability and safety of the system. Active Front Steering (AFS) is a newly technology for passenger cars developed by BMW, that implements an electronically controlled superposition of an angle to the hand steering wheel angle that is prescribed by the driver. However, the

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permanent mechanical connection between steering wheel and road wheels remains [1,2]. AFS could adjust the vehicle performance by means of intervening the road wheel angle in condition of the driver have top priority, which avoid the people's concerns about the Steer-by-Wire system.

A lot of studies about the active steering system are carried out all over the world. But most of the studies focus on the stability of vehicle, which apply AFS, as in [3-10]. As a basic function of active steering system, the driver will experience the variable steering gear ratio function at first, and perceive the improvement of steering portability. AFS enables continuous and situation-dependent variation of the steering ratio according to the vehicle's motion state, therefore AFS improve the maneuverability of the vehicle at low speed and the stability at high speed. The performances of stability improvement with active steering system depend upon the quality of variation of the steering ratio to a certain extent. Thence, it is significant to investigate the variable steering gear ratio function. So we established kinematic model of planetary gear set for active steering system and steering system model with angle input, moreover, some typical tests are accomplished in CarMaker.

2. Model of active steering system

Compared with traditional mechanical steering system, active steering system is comprised of a double planetary gear and an electric actuator motor additionally, besides the primary mechanical steering system. Because all the links from the steering wheel to road wheel are mechanical, it is indubitable that AFS is reliable and safe. AFS could ensure that the vehicle is under driver's control all the time, and make driver have a clear road feel.

The planetary gear of AFS have two degrees of freedom (DOF), the output of the planetary gear connects with the steering gear's pinion, and one input connects with the steering wheel, and the other input connects with an electric actuator motor, as shown in Fig.1.

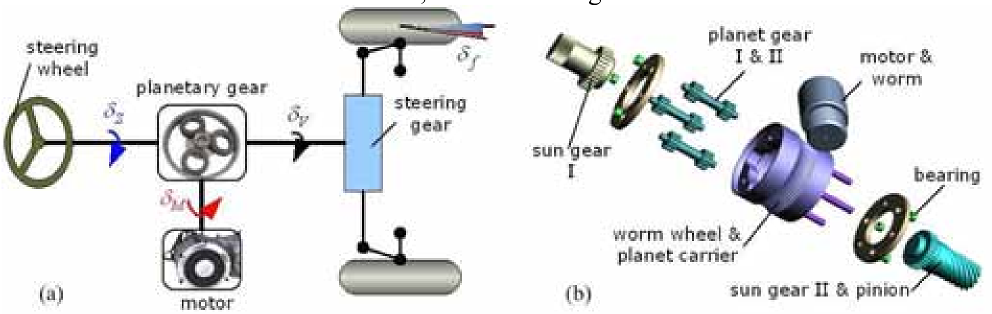

Fig. 1. (a) Schematic view of AFS system; (b) 3D-model of planetary gear set and electric motor

The steering gear's pinion angle δ_V is dependent up on the steering wheel angle δ_S and motor angle δ_M, and there is a nonlinear relationship between the average front road wheel steering angle δ_f and the pinion angle δ_V . Hence, the front wheel angle δ_f which relies on the pinion angle δ_V would be changed with variation of the motor angle δ_M when given a certain hand steering wheel angle δ_S, such is the function of variable steering gear ratio [11]. The above relationships are showed as follow:

$$\delta_V(t) = i_V \cdot \delta_S(t) + i_S \cdot \delta_M(t) \tag{1}$$

$$\delta_f(t) = F(\delta_V(t)) \tag{2}$$

$$\delta_f(t) = 1 / i \cdot \delta_S(t) \tag{3}$$

Where δ_V denotes the steering gear's pinion angle, δ_S the hand steering wheel angle, δ_M the motor angle, δ_f the average front road wheel angle, i_S the factor of conversion from hand steering wheel angle to the steering gear's pinion angle, i_M the factor of conversion from motor angle to the steering gear's pinion angle, $F(\bullet)$ denotes the nonlinear relationship between the average front road wheel steering angle δ_f and the pinion angle δ_V, i is defined as the steer ratio of active steering system, $i = \delta_S / \delta_f$.

2.1. Model of planetary gear

Active front steering system is primarily comprised of a rack-and-pinion steering system, a double planetary gear, an electric actuator motor, a worm gear, an electromagnetic locking unit in case of a safety relevant malfunction, and an electronic control unit as brain. The hand steering wheel connects with the sun gear I, the sun gear I meshes with the planetary gears I, at the same time, the motor rotation is transmitted to the planet carrier by the worm gear, then the planet carrier angle couples with the planetary gears I result in the rotation of the planetary gears II, and planetary gears II meshes with the sun gear II, the sun gear II connect with steering gear's pinion solidly, finally the motion transfers to front road wheels through steering gear and the linkage between steering gear and road wheel. Each planet gear I is jointed to planet gear II solidly, so each planet gear I and planet gear II rotate synchronously, the worm is connected with motor rotor, worm wheel and the planet carrier is integrated into one part, as shown in Fig.2.

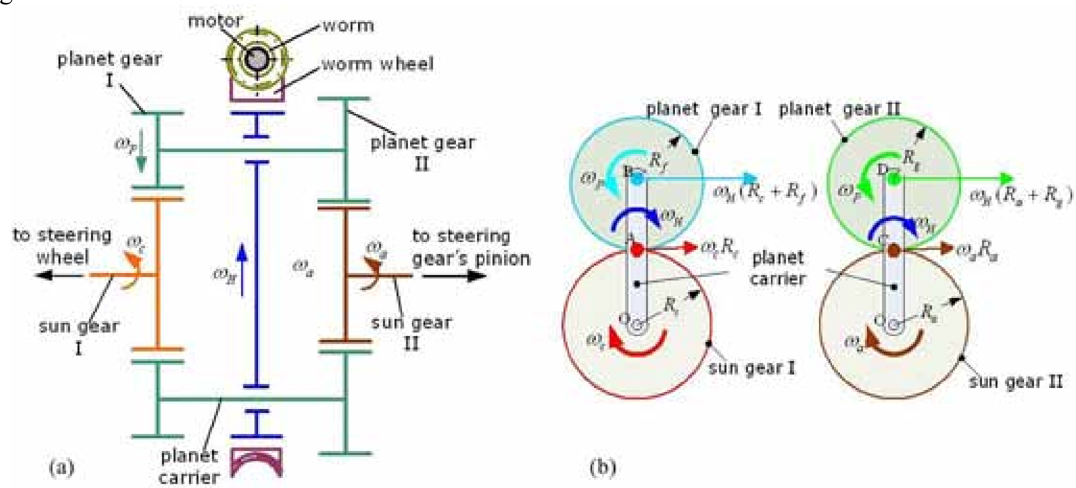

Fig. 2. (a) 3D-model of planetary gear set and electric motor; (b) Structure of planetary gear set.

The worm will be locked if the system is shut down and in case of a safety relevant malfunction. In this case the driver is able to further steer with a constant steering ratio like a traditional steering system. Therefore, the safety of system is certified.

Fig.2(b) shows the mechanisms of planetary gear, the core of AFS. The angular velocity of sun gear I, sun gear II, and planet carrier are denoted by ω_c, ω_a, ω_H respectively. The angular rotation velocity of planet gear is denoted by ω_p.

We assume that the angular velocity of sun gear I equal to the hand steering wheel, the angular velocity of planet carrier equal to motor gained by worm gear, and the sun gear II equal to the steering gear's pinion. The elastic deformation of all the steering linkages and the fluctuation of angular velocity caused by non-constant velocity universal joints are not considered in the model. The above relationship is described as that:

$$\begin{aligned}\dot{\delta}_S &= \omega_c \\ \dot{\delta}_M / i_W &= \omega_H \\ \dot{\delta}_V &= \omega_a\end{aligned} \tag{4}$$

Where i_w denote the ratio of worm gear.

Based on analyzing transmission of sun gear I to planet gear I, and planet carrier to sun gear II, we derived the relationship among the steering gear's pinion angle δ_V, the hand steering wheel angle δ_S, and the electrical motor angle δ_M.

We infer from motion of planet gear I and II two equations:

$$\omega_c R_c = \omega_H (R_c + R_f) + \omega_P R_f \tag{5}$$

$$\omega_a R_a = \omega_H (R_a + R_g) + \omega_P R_g \tag{6}$$

Where ω_c denote the angular velocity of sun gear I, ω_a the sun gear II angular velocity, ω_H the angular velocity of planet carrier, ω_p the rotational angular velocity of planet gear, R_c the sun gear I pitch radius, R_f the planet gear I pitch radius, R_a the sun gear II pitch radius, R_g the planet gear II pitch radius.

Equation (7) is derived from (5) and (6).

$$\omega_a \cdot (R_a / R_g) = \omega_c \cdot (R_c / R_f) + \omega_H \cdot (R_a / R_g - R_c / R_f) \tag{7}$$

Equation (8) is derived from (4) and (7).

$$\dot{\delta}_V = (R_c R_g / R_a R_f) \cdot \dot{\delta}_S + (1 - R_c R_g / R_a R_f) / i_W \cdot \dot{\delta}_M \tag{8}$$

The initial state is zero:

$$\dot{\delta}_V(0) = \dot{\delta}_S(0) = \dot{\delta}_M(0) = 0 \tag{9}$$

It can be inferred from (8) and (9) that:

$$\delta_V = (R_c R_g / R_a R_f) \cdot \delta_S + (1 - R_c R_g / R_a R_f) / i_W \cdot \delta_M \tag{10}$$

3. Simulation and result analysis

In the paper, we have proposed a simple steering ratio dependent vehicle speed to verify the function of variable steering ratio, as shown in Fig. 3. Steering ratio is divided into three parts, steering ratio is set to minimum i_{min} at low speed, steering ratio is set to maximum i_{max} at high speed, steering ratio increase linearly with the speed at normal speed. The steering ratio i is defined like that:

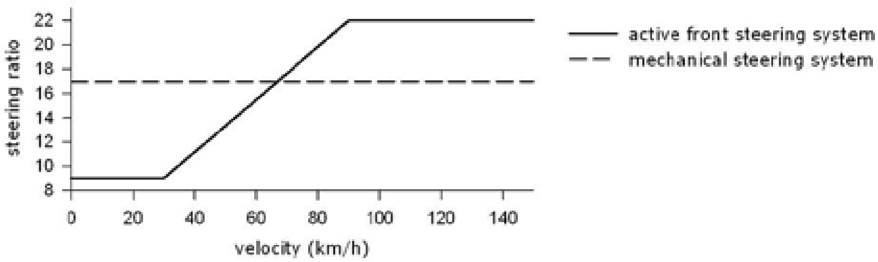

Fig.3 Velocity dependent steering gear ratio

$$i = \begin{cases} i_{\min} & u \le u_1 \\ (i_{\max} - i_{\min}) / (u_2 - u_1) \cdot u & u_1 \le u \le u_2 \\ i_{\max} & u \ge u_2 \end{cases} \tag{11}$$

Based on the above analysis, the model is established in Simulink and a PID controller is designed for variable steering ratio function. Simulations are based on the IPG's CarMaker, CarMaker include vehicle model, road model and driver model, all the parts build up a virtual simulation and test environment. In the paper, vehicle model adopted IPG's CarMaker original vehicle model except steering system.

3.1. Static steering test

The performance of static steering mostly represents the convenience of vehicle steering. So the static steering test is a basic test for steering system performance. The steering wheel angle input is two cycles sine of 0.2Hz, 270 degree, as shown in Fig. 4(a). Fig. 4(b) illustrates the result of static steering simulation. The result shows that front road wheel angle of vehicle with AFS is larger than without, that is to say, AFS turn more directly when static and at low speed. Active front steering system makes the maximum steering wheel angle of the test vehicle decrease from 1.3 to 0.75 turns.

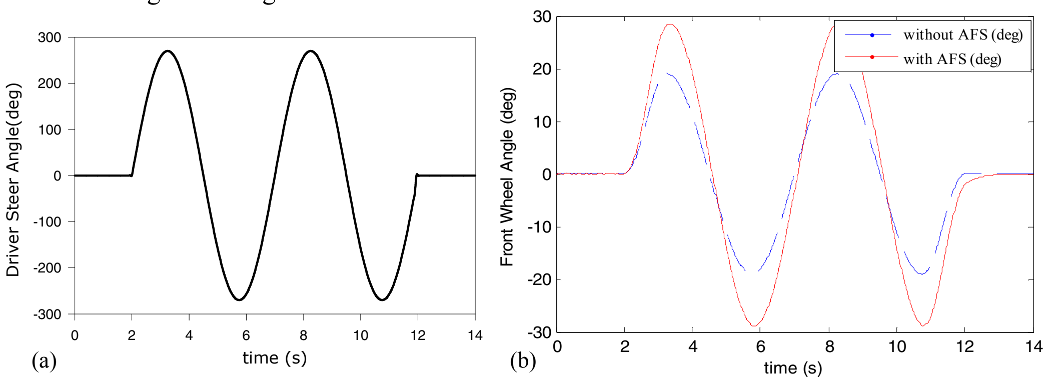

Fig.4 (a) Time history of steering wheel angle; (b) Front wheel angle with AFS vs without AFS

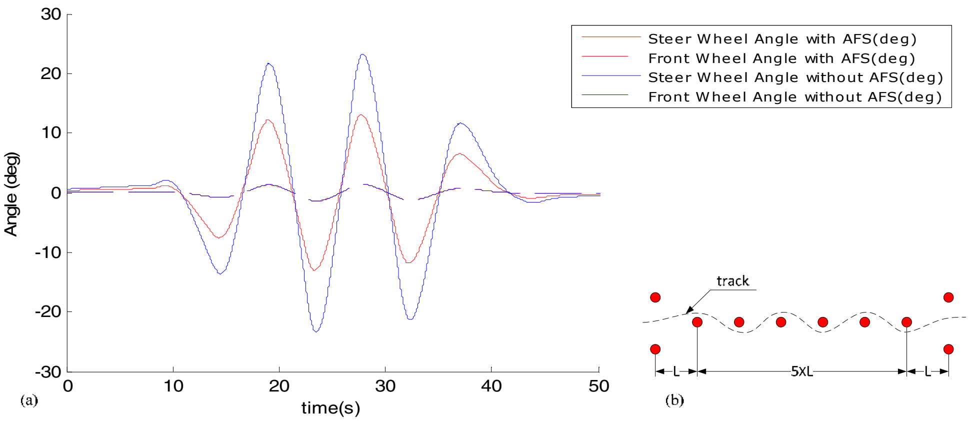

Fig. 5. (a) Steering wheel angle and front wheel angle of slalom test; (b) Pylon Markers of slalom test.

3.2. Slalom test

The slalom test is used for examining the vehicle turning performance at low speed. The pattern of the test road is shown in Fig. 5(b). Pylons space L=36m, vehicle speed is 30 km/h.

Fig. 5(a) shows the simulation result: In case of slalom test, whether the vehicle assembles active steering system or not, front wheel angle is almost the same, but steering wheel angle of vehicle with AFS is even smaller than without. In this case, vehicles with AFS can cut off the steering wheel angle about 40% at 0-30km/h. So the vehicle with AFS let driver easier to handle particularly on winding path.

4. Conclusion

By analyzing the structure of BMW's Active Front Steering System, the active front steering gear model is completed, which includes the planetary gear model and steering gear model. Taking account of the vehicle steering sensitivity and maneuverability, a simple linear steering ratio is proposed. And a PID controller for variable steering ratio function is designed in Simulink. The simulation tests are carried out in CarMaker to verify the function of variable steering ratio. The results confirm that AFS can improve the vehicle maneuverability especially at low speed.

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