

Study of the Effects upon Vehicle Stability Exerted by Tie Rod End Clearance Under Slalom Maneuver

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ABSTRACT

Tie rod end clearance is an important parameter influencing automobile stability under slalom maneuver. In this paper the steering mechanism is simplified into a plane linkage mechanism and an analysis of the effects on vehicle stability exerted by kinematic pair clearance under slalom maneuver is also presented. A 4DOF mathematical model of vehicle maneuvering system is thus being built. On the basis of this model, we adopt the numerical analysis method to conduct a simulated analysis about the stability of prototype vehicle side slip angle as the clearance parameter changes. According to the results, vehicle slalom dynamics behaviors manifest itself in shifting from single cycle to chaos directly. With the increase in clearance, nearly no change is displayed in the upper critical frequency of vehicle slalom instability. However, an increasing rise is shown in the lower critical frequency. The instability frequency bandwidth, accordingly, bears an increase as well and a marked difference manifests itself with regard to the characteristics of window dynamics in chaotic areas.

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INTRODUCTION

The stability of vehicle steering is one of the most important performances in the study of steering dynamics. Since the 1930s people began to pay attention to the study of this aspect. At the moment, the study on linear system has come to nearly maturity while that on non-linear system remains somewhat backward, within which the study of vehicle steady state steering is far more than that of unsteady state [1, 2, 3, 4]. For example, slalom maneuver is a common working condition in the process of vehicle moving and accidents take place from time to time during road transport [5, 6, 7]. Thus extensive and in-depth researches about the slalom maneuver have been carried out by scholars throughout the world. A prototype vehicle model [8] is made by application of VEDYNA and slalom maneuver simulation is conducted via hardware-in-the-loop experiment to give an analysis of the effects on the stability of slalom maneuver exerted by the change of vehicle mass center. The conclusion is reached that an overly high mass center can result in a lower stability in vehicle moving and a likely turnover under slalom steering condition. A multi-DOF system model of the electric vehicle with independent-motorized-wheel-drive is being built. And an analysis is made about the effects the moving velocity has on the steering stability under slalom maneuver [9]. A two-degree-of-freedom vehicle system model is made to carry out a

simulation experiment of sinusoidal steering wheel input with different amplitudes. The conclusion is that an excessively high amplitude of sinusoidal steering will lower the driving stability when the vehicle is under slalom maneuver [10]. A two-degree-of-freedom vehicle system model is also made to conduct a simulation experiment of the steering wheel sinusoidal input under different cycles and the conclusion is drawn that the frequency of steering wheel sinusoidal input has a relatively significant influence on the driving stability under slalom maneuver [11].

A vehicle-pavement system model of multi-DOF is built to conduct simulation experiments on system models of two, three and four degrees of freedom respectively under slalom maneuver. Through the comparison among experiment results of prototype vehicles, the conclusion comes to that vehicle road surface system model of both three and four degrees of freedom can better reflect vehicle slalom maneuver [12]. A four-degree-of-freedom vehicle maneuvering system model with regard to the four-wheel alignment parameter is established and the effects on the vehicle stability under slalom maneuver exerted by four-wheel alignment parameter are therefore acquired by means of numerical simulation analysis [13]. Many strategies aim to control handling stability are put forward in line with models of different vehicle systems. And

the simulation experiment under slalom maneuver has proved that these control strategies aiming at wheel slip ratio, side slip angle and yaw rate can effectively increase vehicle driving stability under slalom maneuver [14, 15, 16].

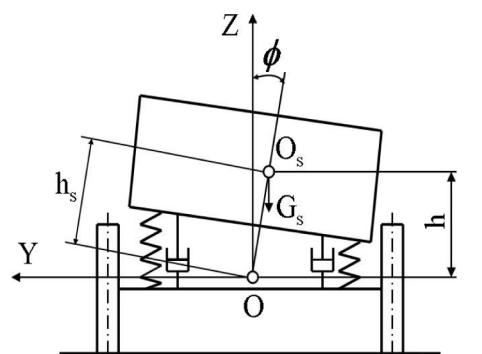
The literature review illustrates that most scholars' researches above concerns mainly about the impacts which working condition parameters (such as driving velocity V and steering wheel angle δ), vehicle mass center and control strategies have on slalom stability, while there is little attention paid to the influences on the frequency characteristics under slalom maneuver exerted by tie rod end clearance. Just as Guo writes, "it is so far still difficult to find certain literatures analyzing the influences on vehicle maneuvering of tie rod end clearance" [1]. And it is shown by our preliminary studies that the effects Clearance have on maneuvering movement cannot be neglected [13, 17].

As a result, in this paper, nonlinear factors of the tie rod end clearance are taken into consideration on the basis of researches above and our previous studies to build a 4DOF vehicle model characterized by steering nonlinear dynamics, including the roll motion of the vehicle body. A numerical analysis plus calculation is made based on this model so as to understand what effects the clearance has on the frequency feature of vehicle under slalom maneuver and to find out the law of clearance value's influences on the slalom stability, hoping to further enrich nonlinear dynamics theory of vehicle steering condition.

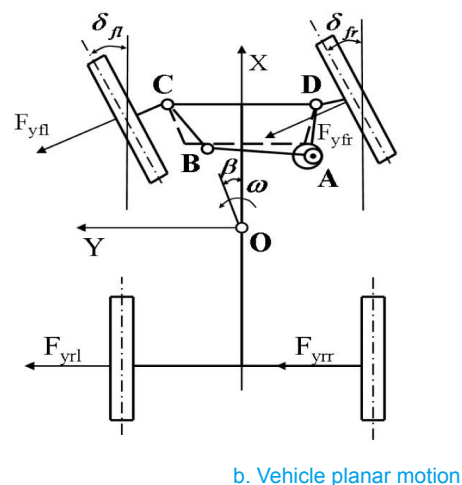
SYSTEM DYNAMICS MODEL

System Dynamics Model Taking Clearance into Consideration

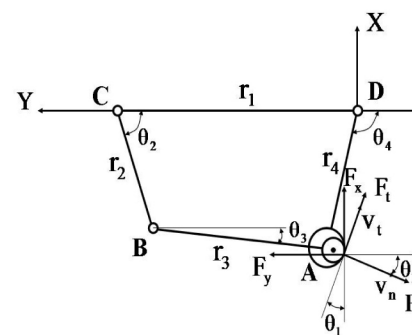
On the basis of a dynamics model of vehicle steering system built by previous work, this paper, taking into account the effects the tie rod end clearance has on the steering stability, establishes a dynamics model of prototype vehicle steering system and a coordinate system like what Figures 1(a), (b), (c) has demonstrated.



a. Vehicle body rolling motion



b. Vehicle planar motion



c. Steering trapezium considering clearance

Figure1. (cont.) Dynamics model of vehicle steering

Because the tie rod end clearance is considered and the influences of the front wheel alignment parameters are ignored, the steering trapezium of the prototype vehicle will be simplified into a crank-connecting rod mechanism like Figure 1(c) shows [18,19]. In order to simplify the analysis, we only consider that some Clearance exist in the joint between the trapezoid arm (right) and the tie rod and that all bars will be regarded as rigid bodies. The vehicle is under slalom maneuver at a constant pace V with OXYZ as a coordinate system fixed in roll center. XOY parallels the road surface while the X-axis points to where the vehicle is driving to and the Z-axis indicates upwards perpendicularly. Lastly, the Y-axis points to the left side according to the right-hand rule.

This system dynamics model makes assumptions as following:

- 1) exclude air resistance;
- 2) ignore the influences of front wheels' alignment parameters and make the steering trapezium parallel to XOY coordinate planes;
- 3) the front has the same suspension roll center as that of the rear.

Four DOF are used to present the vehicle steering system: yaw rate ω , side slip angle β , vehicle roll angle ϕ , left front wheel steer angle δ_l .

Figure1.

The Differential Equations of Vehicle Steering Taking Tie Rod End Clearance into Consideration

According to the dynamics model of vehicle steering in the above Figure 1(a), (b), (c), using D'Alembert theorem to establish the differential equation of vehicle steering motion.

The Equations of Vehicle Steering Motion

The moment equilibrium equation of the vehicle around the Z-axis is

$$I_z \dot{\omega} + I_{xz} \ddot{\phi} = F_{yfl} \cdot l_f \cdot \cos \delta_l + F_{yfr} \cdot l_f \cdot \cos \delta_r - (F_{yrl} + F_{yrr}) \cdot l_r \quad (1)$$

The equilibrium equation of the vehicle along the Y-axis is

$$mv(\dot{\beta} + \omega) \cos \beta - m_s h_s \ddot{\phi} = F_{yfl} \cdot \cos \delta_l + F_{yfr} \cdot \cos \delta_r + F_{yrl} + F_{yrr} \quad (2)$$

The moment equilibrium equation of the vehicle body rolling around the X-axis is

$$I_x \ddot{\phi} + c_\phi \dot{\phi} + k_\phi \phi + I_{xz} \omega = m_s h_s v(\dot{\beta} + \omega) \cos \beta + m_s h_s g \phi \quad (3)$$

The moment equilibrium equation of the right front wheel around the kingpin is

$$M_p - J_c \alpha_4 - F_{flr} \cdot D_w = 0 \quad (4)$$

where F_{yfl} is the cornering force of the left front wheel, F_{yfr} is the cornering force of the right front wheel, F_{yrl} is the cornering force of the left rear wheel, F_{yrr} is the cornering force of the right rear wheel, δ_r is the right front wheel steer angle, l_f is from front axle to CG, l_r is from CG to rear axle, m is the mass of the complete vehicle, m_s is the sprung mass, h_s is the distance from the sprung mass center to the roll axis, I_x is the moment of inertia of the sprung mass around the X-axis, I_{xz} is the product of inertia of the sprung mass around the X- and Z-axes, I_z is the moment of inertia of the vehicle around Z-axis, C_ϕ is the equivalent rolling angular damping of the body, k_ϕ is the equivalent rolling angular stiffness of the body, J_c is the moment of inertia of the right trapezoid arm going around the right kingpin, D_w is the pneumatic trail, M_p is the impact moment of clearance to the kingpin, and α_4 is the angular acceleration of the rotation of the right trapezoid arm.

The Expression of Tire Cornering Force

The cornering force can be expressed as a simplified magic formula [20] as

$$F_y = D \sin(C \cdot \arctan(B\alpha - E(B\alpha - \arctan(B\alpha)))) \quad (5)$$

$$\text{with } \begin{cases} D = a_1 \cdot F_Z^2 + a_2 \cdot F_Z \\ B = \frac{a_3 \cdot \sin(a_4 \cdot (\arctan(a_5 \cdot F_Z)))}{C \cdot D} \\ E = a_6 \cdot F_Z^2 + a_7 \cdot F_Z + a_8 \end{cases}$$

where B, C, D, E represent stiffness, shape, peak and curvature factor in the magic formula respectively, $a_1, a_2, a_3, a_4, a_5, a_6, a_7, a_8$ are the parameters obtained by experimental fitting, and F_Z is the load on the tires, α is the tire slip angle.

Table1. Magic formula parameters

Symbol	a_1	a_2	a_3	a_4
Value	-22.1	1011	1078	1.82
Symbol	a_5	a_6	a_7	a_8
Value	0.208	0	-0.354	0.707

The relation schema of the prototype vehicle's cornering force F_y and the slip angle can be worked out from the expression of tire cornering force and table 1, as is shown in Figure 2.

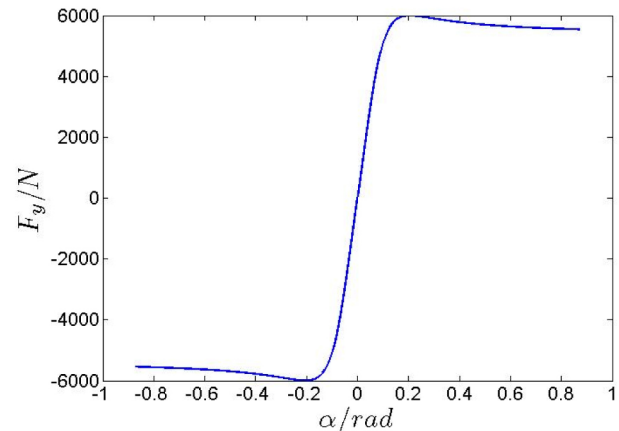


Figure 2. Cornering force as a function of the tire slip angle

The expression of front and rear tires' slip angles are

$$\left. \begin{aligned} \alpha_{fl} &= \delta_{fl} - a \tan\left(\frac{l_f \cdot \omega}{v \cdot \cos \beta} + \tan \beta\right) \\ \alpha_{fr} &= \delta_{fr} - a \tan\left(\frac{l_f \cdot \omega}{v \cdot \cos \beta} + \tan \beta\right) \\ \alpha_r &= a \tan\left(\frac{l_f \cdot \omega}{v \cdot \cos \beta} - \tan \beta\right) \end{aligned} \right\}$$

(6)