Mechanical Systems and Signal Processing 92 (2017) 146–155

Contents lists available at ScienceDirect

Mechanical Systems and Signal Processing

journal homepage: www.elsevier.com/locate/ymssp

Chaos vibration of pinion and rack steering trapezoidal mechanism containing two clearances

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ARTICLE INFO

Article history:
Received 11 January 2016
Received in revised form 15 January 2017
Accepted 18 January 2017
Available online 31 January 2017

Keywords:
Pinion and rack steering mechanism
Clearance
Chaos
Stability

ABSTRACT

The multi-clearances of breaking type steering trapezoidal mechanism joints influences vehicle steering stability. Hence, to ascertain the influence of clearance value on steering stability, this paper takes the steering mechanism of a certain vehicle type as a prototype that can be simplified into a planar six-bar linkage, then establishes the system dynamic differential equations after considering the two clearances of tie rods and the steering knuckle arms. The influence of the clearance parameters on the movement stability of the steering mechanism is studied using a numerical computation method. Results show that when the two clearances are equal, the planar movement of the tie rods changes from period-doubling to chaos as the clearances increase. When the two clearances are 0.25 mm and 1.5 mm respectively, the planar movements of the two side tie rods come into chaos, causing the steering stability to deteriorate. Moreover, with the increase of clearances, turning moment fluctuates more intensively and the peak value increases.

1. Introduction

The pinion and rack steering trapezoidal mechanism, which is a typical six-bar mechanism that contains five kinematic pair clearances, is widely used in cars nowadays. When steering, the contact-impact and separate process of pairs is similar to the impact process of a piece-wise linear system. Theoretically, a chaotic phenomenon may occur in the system, which will cause the motion stability of the steering mechanism to deteriorate. The steering stability is dependent on whether the design of the clearance parameter is reasonable. Therefore, research on the motion stability of vehicle steering mechanism with clearance can provide theoretical guidance for the optimal design and matching of steering mechanism and vehicle.

Many fruitful studies have been conducted on the kinematic and dynamic characteristics of planar linkage mechanism with clearances. Both motion accuracy and stability of mechanism have been improved through such research [1–5]. Seneviratne and Earles [6] studied system impact and collision vibration caused by the clearance in rod connection. By using the Poincare map and the initial value sensitivity of system, they successfully found the rod vibration caused by clearance. However, few articles focus on applying the findings to designing vehicle steering mechanism. Li [7] established a dynamic model of a truck's front wheel self-excited shimmy system with considering the integral steering trapezoidal clearance. He observed that a numerical increase in steering system clearance would lead to an increase in the front wheel shimmy ampli-

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http://dx.doi.org/10.1016/j.ymssp.2017.01.019
0888-3270/

tude. Gu [8] observed that front wheel shimmy could change from period-doubling bifurcation into chaos with the numerical increase of clearance. He established a model of a front wheel self-excited shimmy system that used a two-state clearance rotation pair model and considered the integral steering trapezoidal clearance. Lu et al. [9] analyzed the influences of system parameters on dynamics behavior of the vehicle shimmy system with consideration of clearance in the steering linkage mechanism and demonstrated that these parameters could make a coupled contribution to the dynamics response of the vehicle shimmy system. Sun [10], by establishing the front shimmy model with using a two-state clearance rotation pair model and considering the clearance in front wheel kingpin, also found that the front shimmy system would turn from aperiodic reciprocating vibration to periodic vibration with the numerical increase of clearance. Wang et al. [11] established a model of the car steering wheel shimmy system that used the discontinuous contact piece-wise nonlinear clearance model and considered the two clearances of a pinion and rack steering trapezoidal mechanism. Calculation results showed that the amplitude of the front wheel shimmy would increase significantly with the numerical increase of clearance.

Synthesizing the above studies, the effect of clearance on vehicle front wheel shimmy has been analyzed, but the mechanism used was the integral steering trapezoidal mechanism of dependent suspension [7–10]. Although Wang et al. [11] analyzed the dynamic characteristics of the independent suspension vehicle shimmy system with clearance, the clearance model was equivalent to a spring damper sliding pair model that could not reflect the characteristics of the ball joint rotation pair in steering mechanism. On the basis of our previous studies [12,13], a rotation pair model with two-state clearance that can better reflect the characteristics of the ball joint rotation pair in the steering mechanism is utilized in this paper. Additionally a dynamic model of pinion and rack steering mechanism containing two clearances is established. The numerical method is then applied to analyze the effects of clearance value on the dynamic characteristics of the steering mechanism, providing theoretical support for the design of pinion and rack steering mechanism.

2. Dynamic model of pinion and rack steering mechanism

2.1. Dynamic model of pinion and rack steering mechanism containing two clearances

According to the relative position of pinion and rack steering gear and steering linkage, front steering mechanism in an automobile has four layout forms: the steering gear can be located in the front or rear of the axle, while the steering linkage can either be in the front or rear. The prototype car is acquired from the cooperative automobile manufacturer—Zhejiang Geely Automobile Research Institute. The car uses the Macpherson front suspension, pinion and rack steering mechanism, steering gear located behind the front axle and steering linkage located in the rear. The force of the steering gear is inputted from the middle and outputted to both sides. In this paper, the steering system is simplified into a planar six linkage mechanism. The motion direction of the vehicle is defined as the x-axis, the center line of the front axle as the y-axis, and the middle of front axle as the origin point of coordinates, as shown in Fig. 1a.

As a part of the steering system, the pinion and rack steering mechanism is extracted in Fig. 1a. For a more convenient calculation, the steering mechanism is built as a dynamic model of six linkage mechanism. Force analysis is conducted at clearances B and E. The reason why we choose clearances B and E is that the motion pair at point B or E is where the left or right steering tie rod is directly connected to the left or right steering wheel. The motion pair at point B or E has a direct effect on the left or right steering wheel. Moreover, they are the last motion pairs in the transport chain of torque inputted by the driver. When conducting the force analysis, assume that the rack just moves along the x-axis. So the degrees of freedom of the system are as follows: θ_1-the angle between the left steering knuckle arm and the x-axis; θ_2-the angle between the left steering tie rod and the x-axis; x-the displacement of the rack; θ_3-the angle between the right steering tie rod and the x-axis; and θ_4-the angle between the right steering knuckle arm and the x-axis.

As shown in Fig. 1b, suppose that the clearances exist between the left steering tie rods 2 and the left steering knuckle arms 1 and between the right steering tie rod 4 and the right steering knuckle arms 5. The radius of the axle pins is R and the radius of the axle sleeves is R_0. Torques are applied on the rod 5 and the clearance distance of the motion pair is e. The two-state clearance model raised by Dubowsky [14] is adopted. Furthermore, assume that elastic and damping effects exist only on the surface of the axle sleeves. The stiffness coefficient, normal damping coefficient, tangential damping coefficient and friction coefficient are denoted as K, c, c_t and f respectively.

2.2. Differential equations of motion of the system

2.2.1. Driving torque

According to Fig. 1, differential equations of the motion of pinion and rack steering mechanism can be built. Assume that a driving torque, functionally expressed as (1), is applied on the right steering knuckle arm. The mathematical expressions of a, b are as presented in [15].

$$T = a - b \cdot \dot{\theta}_4 \tag{1}$$

$$\begin{cases} a = i\mu 9.8 n_c M_B/(n_C - n_B) \\ b = 60 i^2 9.8 M_B/[2\pi(n_C - n_B)] \end{cases} \tag{2}$$