Optimization and Simulation Analysis of Loader Driving Stability System based on Differential Connection

ABSTRACT: Taking wheel loader as the research object, for the vibration problem under the driving condition of wheel loader, study an optimized differential connection loader driving stability system, model the whole vibration system of wheel loader with three degrees of freedom dynamics, and establish the mathematical model by using Lagrange's equation, combine with the stochastic engineering road excitation model under E-class road surface, and carry out the simulation analysis of the vibration system by MATLAB/Simulink module. Simulation analysis of the vibration system is carried out, and the results show that the body droop acceleration, body angular acceleration and work device angular acceleration under the driving condition of the wheel loader are significantly reduced after adding the driving stability system, and the three kinds of acceleration are further reduced after optimizing to the differential connection driving stability system, which indicates that the differential connection loader driving stability system further improves the driving smoothness of the loader.
Key words: Wheel Loader; driving stability system; Smoothness of driving; Three-degree-of-freedom vibration systems

1. INTRODUCTION

Wheel loader is a very widely used construction machinery, often in a variety of sites driving operations, in the loader work process in order to improve operational efficiency, often need the loader according to operational requirements in a full or no load state with a high driving speed to and from the operating site. Most of the mechanical structures of wheel loaders use rigid or semi-rigid suspensions, and due to the usually complex environment of the operating site, the loader is affected by the unevenness of the road surface and its own operating load, there will be a large pitch vibration during the driving process of the loader, which has a negative impact on the physical and mental health of the operator and the efficiency of engineering operations[1].

To address this problem, the early domestic mainly started from the body structure of the loader and optimized in the frame and wheels of the loader to achieve the effect of vibration damping. Currently, the group of Zhao et al[2] has studied a mechanical elastic wheel, which has a large lateral deflection stiffness and can enhance the handling stability of the vehicle under complex road surfaces. As early as in the late 1960s, Professor D.C. Karnopp from the Department of Mechanical Engineering of the University of California, Davis, invented an oleo-air damper and proposed the concept of oleo-air suspension[3]. After improvement and development at home and abroad, the oleo-air suspension system for engineering vehicles still has problems such as not applicable to multiple working conditions and unsatisfactory damping effect, and has not been widely used.

Domestic technicians design a kind of wheel loader driving stability system, and they add a kind of driving stability system to the original hydraulic system of the wheel loader based on the oil-pneumatic suspension technology, which is based on the theory of power absorption and uses the anti-resonance characteristics of the multi-degree-of-freedom system to transfer the vibration energy of the system to the additional elastic element, so as to achieve the control of the vibration of the main system[4][5]. After adding this driving stability system in the wheel loader, when the stiffness and damping of the system are selected reasonably, the violent vibration caused by road excitation when the whole vehicle is fully loaded and in transit operation can be effectively attenuated, and at the same time, the reliability and comfort of the loader are improved, but the structure of this device is relatively simple, the stiffness within the system varies greatly during operation, and the vibration damping effect is limited.

In this paper, on the basis of the above loader driving stability system, a loader driving stability system based on differential connection is studied. The system includes a skin-type accumulator (nitrogen gas as gas spring medium), electromagnetic reversing valve, oil damper, circuit controlled by switch, multi-way valve movable arm controller and some corresponding oil auxiliary parts, the opening and closing of the system can be controlled by the driver, different from the original loader driving stability system is that the differential connection driving stability system when the loader is displaced by road excitation, the piston rod of the movable arm lifting oil cylinder do reciprocating movement, moving arm lifting cylinder compression stroke, part of the oil in the rodless cavity flows into the rod cavity, another part flows into the accumulator, when stretching stroke, the oil flows back to the rodless cavity from the accumulator and the rod cavity, the effective cross-sectional area is the cross-sectional area of the piston rod, so on and so forth, because the accumulator can absorb the impact energy, the damping effect of the oil damper can gradually convert the kinetic energy of the oil flow into heat energy to dissipate, thus achieve the purpose of vibration damping.

2. DYNAMICS MODELING OF THREE-DEGREE-OF-FREEDOM VIBRATION SYSTEMS

2.1 Physical Modeling

To physically model the whole machine vibration system of wheel loader, first of all, the model should be reasonably simplified, the driving stability system acts as the stiffness and damping element in the whole machine vibration system, the total weight of the working device is concentrated in the position of the center of mass of the working device, the body as a whole is regarded as a rigid beam, the mass is concentrated in the position of the center of mass of the body, the whole vehicle structure is symmetrical about the neutral surface, the four tires of the whole vehicle are equivalent to two tires, which located on the neutral surface of the whole vehicle, each tire is the same, with the same stiffness and damping, and always in good contact with the ground, the ground excitation acts on the contact part between the ground and the center axis of the tire. Taking the full-load working condition as an example, the physical model of the whole loader vibration system can be established, as shown in Fig. 1.

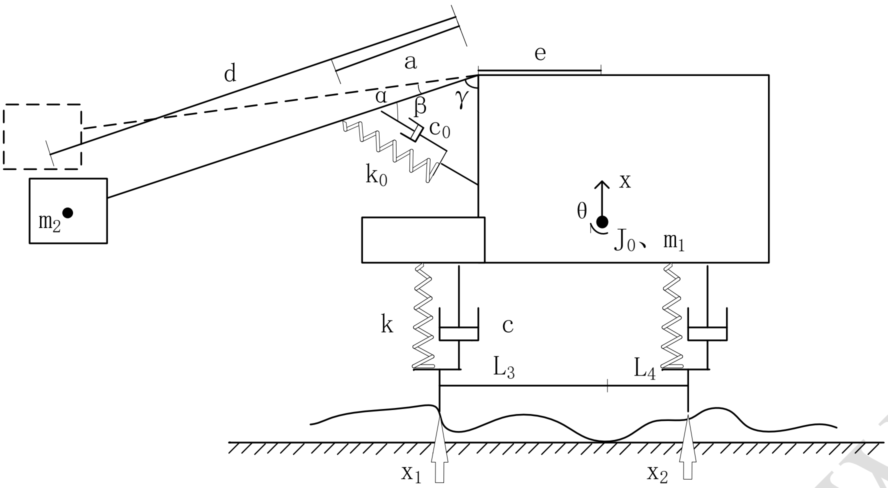


Fig. 1. Physical model of the whole machine vibration of wheel loader with the addition of driving stability system

In the figure: m_2 is the mass of the loader working device, m_1 is the mass of the loader body, J_0 is the rotational inertia of the loader body around the transverse axis of the center of mass, θ is the angular displacement of the loader body around the transverse axis of the center of mass, x is the vertical excitation displacement of the loader machine when it is excited to vibrate from the equilibrium position, α is the angular displacement of the center of mass of the loader working device from the equilibrium position, k is the stiffness of the loader tires, c is the damping of the loader tire, k_0 is the stiffness of the loader driving stability system, c_0 is the damping of the driving stability system of the loader, L_3 is the horizontal distance from the front axle of the loader to the center of mass of the body, L_4 is the horizontal distance from the rear axle of the loader to the center of mass of the body, e is the horizontal distance between the loader movable arm and the hinge point on the front frame to the center of mass of the frame, d is the distance from the hinge point on the loader movable arm and the front frame to the center of mass of the loader working device, a is the distance from the hinge point on the loader movable arm and the front frame to the hinge point between the loader movable arm lift cylinder and the boom, β is the angle between the loader movable arm lifting cylinder and the movable arm, γ is the angle between the vertical direction and the loader movable arm, x_1 is the road excitation on the front wheel of the loader, x_2 is the road excitation to the rear wheels of the loader.

2.2 Mathematical Modeling

After the wheel loader joins the driving stability system, the frame and the working device become elastically connected, and the working device can make angular displacement around the movable arm and the hinge point on the front frame under the action of road excitation. In this paper, the Lagrange equation is used to describe the motion of the loader vibration system. The Lagrange equation uses the relationship between the system energy and the generalized force to model the dynamics of the vibration system mathematically, and the Lagrange equation of the n-degree of freedom vibration system can be expressed as[6][7][8]:

$$\frac{d}{dt}\left(\frac{\partial T}{\partial \dot{q}_i}\right)-\frac{\partial T}{\partial q_i}+\frac{\partial U}{\partial q_i}+\frac{\partial D}{\partial \dot{q}_i}=Q_i \qquad (i=1,2,\sim,n) \tag{1}$$

In the formula: T is the kinetic energy of the system, U is the potential energy of the system, D is the dissipative energy of the system, q_i is the respective generalized coordinate, $\dot{q}_i$ is the respective generalized velocity, and Q_i is the generalized force corresponding to q_i.

The angular displacement θ around the transverse axis of the center of mass, the vertical excitation displacement x, and the angular displacement α of the working device can be taken as generalized coordinates, with the positive direction of x in the vertical upward direction and the positive directions of θ and α in the clockwise direction, and the differential equations of vibration of the three-degree-of-freedom system can be listed according to Lagrange's equation:

The kinetic energy T of the system is:

$$\mathrm{T} = \frac{1}{2}m_1\dot{x}^2 + \frac{1}{2}m_2\left(\dot{x} - e\dot{\theta} + d\dot{\alpha}\sin\gamma\right)^2 + \frac{1}{2}J_0\dot{\theta}^2 + \frac{1}{2}m_2(d\dot{\alpha}\cos\gamma)^2 \tag{2}$$

The potential energy U of the system is:

$$\mathrm{U} = \frac{1}{2}k(x + L_3\theta - x_1)^2 + \frac{1}{2}k(x - L_4\theta - x_2)^2 + \frac{1}{2}k_0[a(\theta + \alpha)\sin\beta]^2 \tag{3}$$

The dissipated energy D of the system is:

$$\mathrm{D} = \frac{1}{2}c\left(\dot{x} + L_3\dot{\theta} - \dot{x}_1\right)^2 + \frac{1}{2}c\left(\dot{x} - L_4\dot{\theta} - \dot{x}_2\right)^2 + \frac{1}{2}c_0\left[a\left(\dot{\theta} + \dot{\alpha}\right)\sin\beta\right]^2 \tag{4}$$

Bring equations (2), (3) and (4) into the Lagrange equation and express the system of vibration differential equations in matrix form:

$$\mathrm{M}\ddot{X} + C\dot{X} + KX = F \tag{5}$$

The mass matrix M, the damping matrix C and the stiffness matrix K can be obtained by collating:

$$M = \begin{bmatrix} m_1 + m_2 & -em_2 & dm_2\sin\gamma \\ -em_2 & J_0 + e^2m_2 & -edm_2\sin\gamma \\ dm_2\sin\gamma & -edm_2\sin\gamma & d^2m_2 \end{bmatrix} \tag{6}$$

$$C = \begin{bmatrix} 2c & (L_3 - L_4)c & 0 \\ (L_3 - L_4)c & (L_3^2 + L_4^2)c + a^2(\sin\beta)^2c_0 & a^2(\sin\beta)^2c_0 \\ 0 & a^2(\sin\beta)^2c_0 & a^2(\sin\beta)^2c_0 \end{bmatrix} \tag{7}$$

$$K = \begin{bmatrix} 2k & (L_3 - L_4)k & 0 \\ (L_3 - L_4)k & (L_3^2 + L_4^2)k + a^2(\sin\beta)^2k_0 & a^2(\sin\beta)^2k_0 \\ 0 & a^2(\sin\beta)^2k_0 & a^2(\sin\beta)^2k_0 \end{bmatrix} \tag{8}$$

where the vectors X and F can be expressed as:

$$\mathrm{X} = [\mathrm{x} \quad \theta \quad \alpha]^T \tag{9}$$

$$\mathrm{F} = [f_1 \quad f_2 \quad 0]^T \tag{10}$$

$$f_1 = k(x_1 + x_2) + c(\dot{x}_1 + \dot{x}_2) \tag{11}$$

$$f_2 = k(L_3x_1 - L_4x_2) + c(L_3\dot{x}_1 - L_4\dot{x}_2) \tag{12}$$

3. PAVEMENT EXCITATION MODEL

The working environment of wheel loaders is complex and the operating road conditions are variable, so the pavement model built in this section is the random engineering pavement model of wheel loaders in regular operation. The IOS/TC08/SC2N67 document "Draft Pavement Unevenness Representation Method" developed by the International Organization for Standardization in 1984 and the domestic GB/T7031-1986 "Vehicle Vibration Input - Pavement Unevenness Representation" standard drafted by Changchun Automobile Research Institute both suggest that the pavement power spectral density $G_q(n)$ is expressed by the fitted expression[9][10]， as follows:

$$G_q(n) = G_q(n_0)\left(\frac{n}{n_0}\right)^{-\omega} \tag{13}$$

Where: n is the spatial frequency, n_0 is the reference spatial frequency (taken as n_0=0.1m^{-1}), $G_q(n_0)$ is the pavement power spectral density at spatial frequency n_0, ω is the frequency index (taken as ω = 2).

The spatial frequency spectral density $G_q(n)$ is transformed into the temporal frequency spectral density $G_q(f)$ as follows:

$$f = nv \tag{14}$$

Where: v is the vehicle speed, f is the time frequency, unit Hz.

The relationship between the spatial spectrum and the temporal spectrum is as follows:

$$G_q(f) = \frac{1}{v} G_q(n) \tag{15}$$

The equation for calculating the time-frequency spectral density can be obtained:

$$G_q(f) = G_q(n_0) n_0^2 \frac{v}{f^2} \tag{16}$$

The first order differentiation of the road excitation is the velocity power spectral density, as follows:

$$\dot{G_q}(f) = (2\pi f)^2 G_q(f) = 4\pi^2 G_q(n_0) n_0^2 v \tag{17}$$

The white noise is input as the pavement velocity into the above equation as the excitation model for the pavement as follows:

$$\dot{x}(t) = -2\pi f_0 x(t) + 2\pi n_0 w(t) \sqrt{G_q(n_0) v} \tag{18}$$

Where: x(t) is the pavement excitation input, w(t) is the pavement white noise, and f_0 is the lower cutoff frequency of the filter ($f_0 \approx$ 0.01Hz).

The working environment of wheel loader is poor, and according to the national standard document, the pavement grade can be roughly divided into 8 grades A~H according to the power spectrum density[11]. In this paper, Class E pavement is selected as the working pavement with the maximum vehicle speed v=14 km/h. The pavement excitation model is built in MATLAB/Simulink, and the pavement excitation displacement curve is shown in Fig. 2.

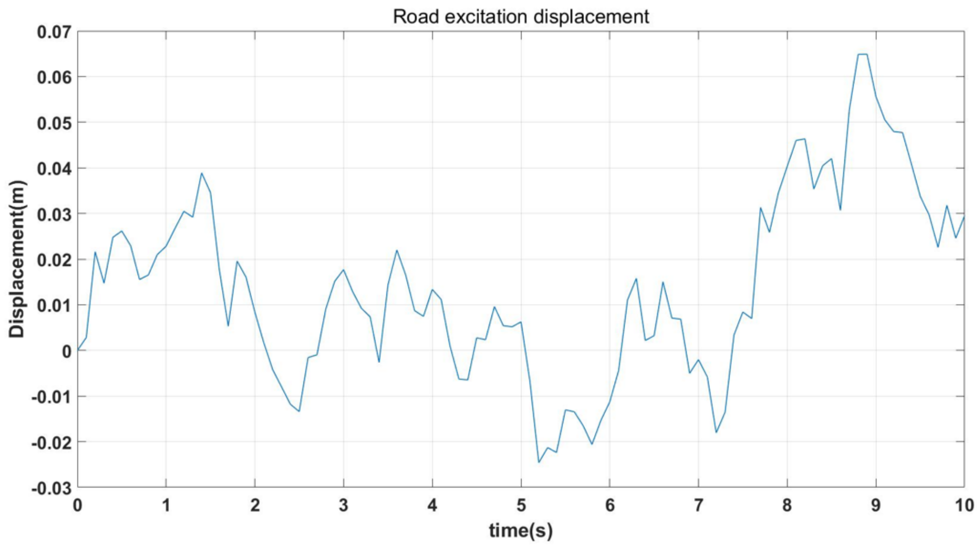


Fig. 2. Excitation displacement of road surface

4. SIMULATION ANALYSIS

4.1 Simulation Modeling

In order to facilitate the establishment of the simulation model, the matrix form of the mathematical model of the whole loader vibration system established above should first be processed, which can be obtained from equation (5):

$$\ddot{X} = M^{-1}\left(F - C\dot{X} - KX\right) \tag{19}$$

From this, a simulation model of the whole loader vibration system can be established, as shown in Fig. 3.

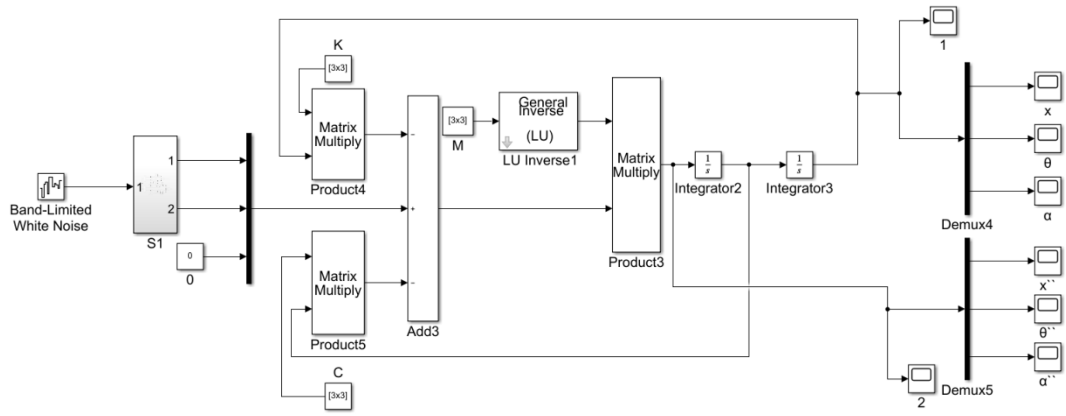


Fig. 3. Simulink simulation block diagram of the whole loader vibration system with the addition of driving stability system

The "S1" module is the subsystem module of vector F. The subsystem takes white noise as input and two elements f_1 and f_2 of vector F as output, and the block diagram of the subsystem is shown in Fig. 4. The delay module is added to simulate the excitation displacement of the front and rear wheels.

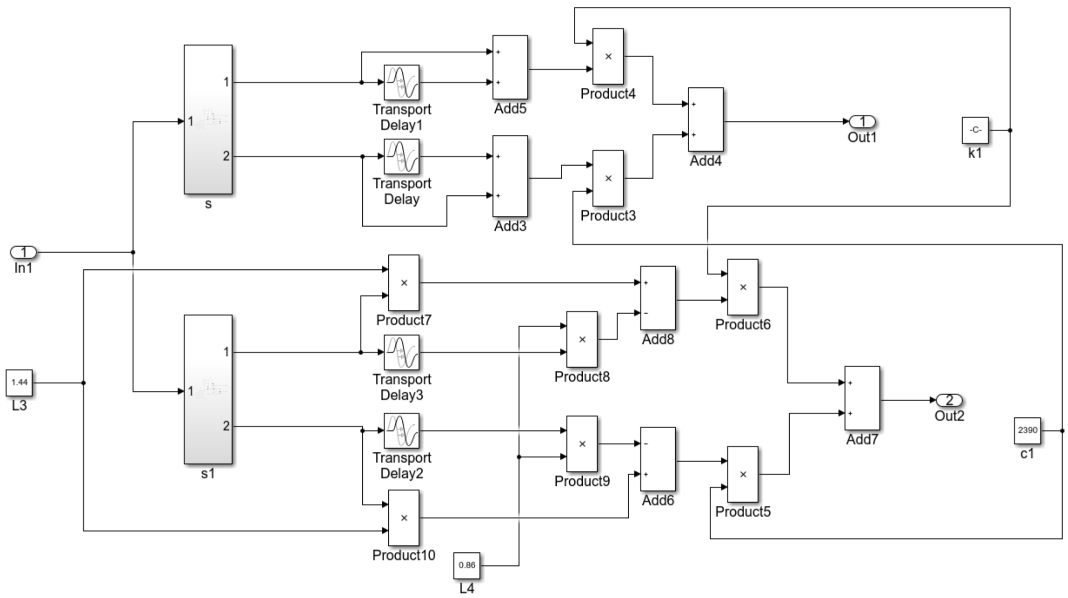
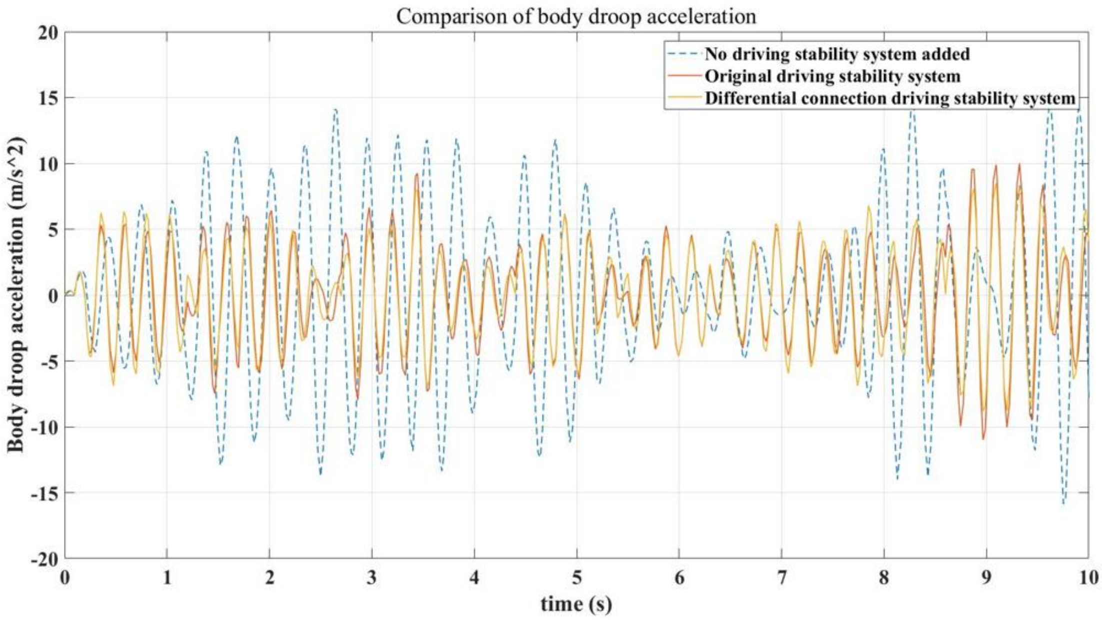


Fig. 4. Subsystem diagram of vector F

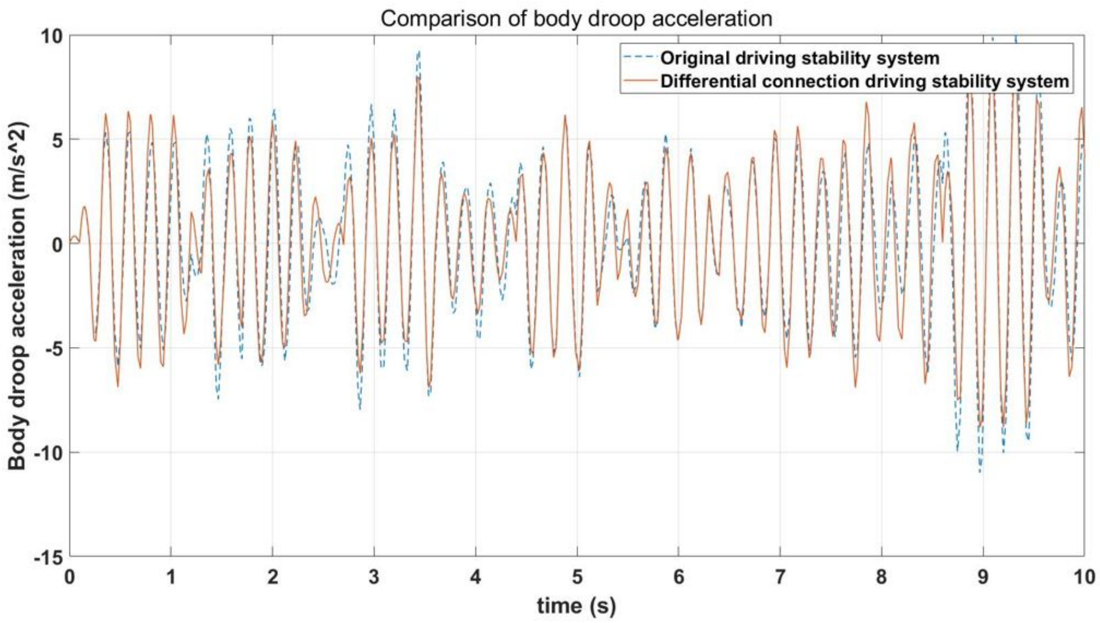
4.2 Simulation Results

In this paper, we use the 3D modeling software Solidworks to establish a properly simplified model, including the working device, front frame, rear frame, cab, tires and other structures of the wheel loader, and for the convenience of simulation analysis, the structures of the wheel loader that have no obvious influence on the simulation results are represented by the equivalent mass, and the mass, center of gravity position, rotational inertia and the size of each part of the model can be obtained by using some functions of the software. For the stiffness and damping of the driving stability system, they can be obtained within a suitable range by pre-setting, and other parameters can be estimated by consulting the sample data of the loader manufacturer, combining with relevant literature and theoretical calculations.

The three-degree-of-freedom vibration physical model is established, and the body vertical acceleration, body angular acceleration and work device angular acceleration are used as the indexes to evaluate the smoothness of the loader driving, and the simulation model takes the three kinds of acceleration as the output, the simulation step is 0.01, the simulation time is 10s, and the simulation results under the E-level random road excitation are shown below.

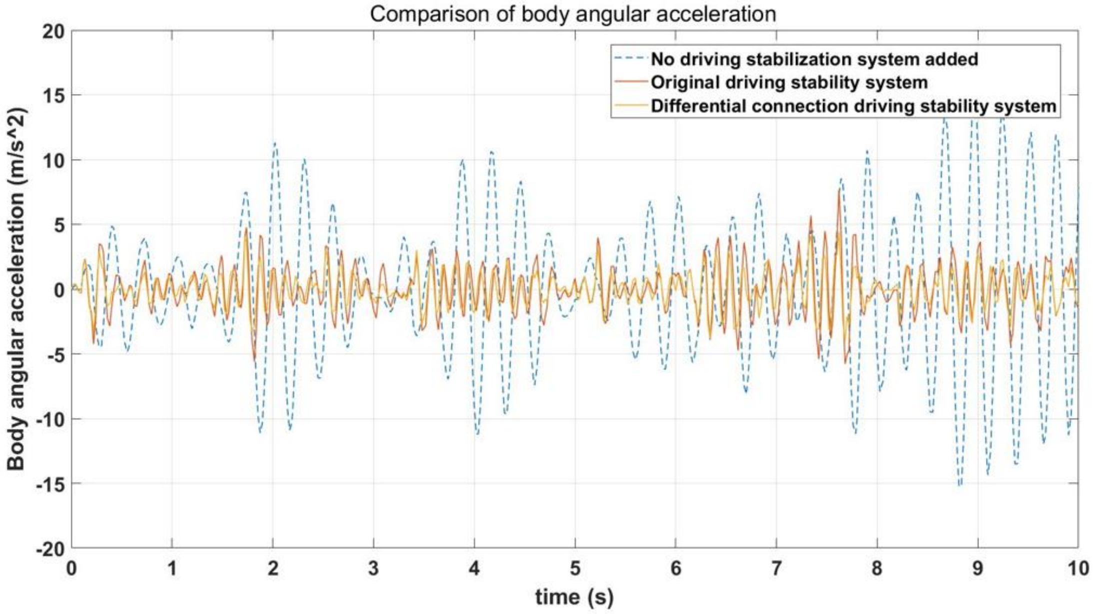


(1)

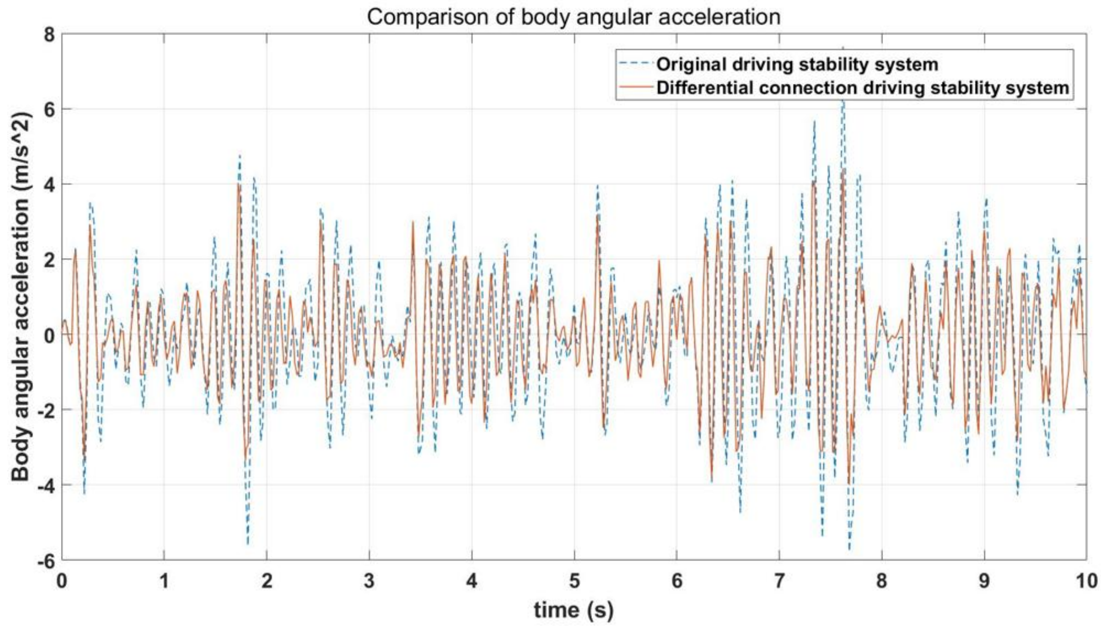


(2)

Fig. 5. Comparison of simulation results of body vertical acceleration



(1)



(2)

Fig. 6. Comparison of body angular acceleration simulation results

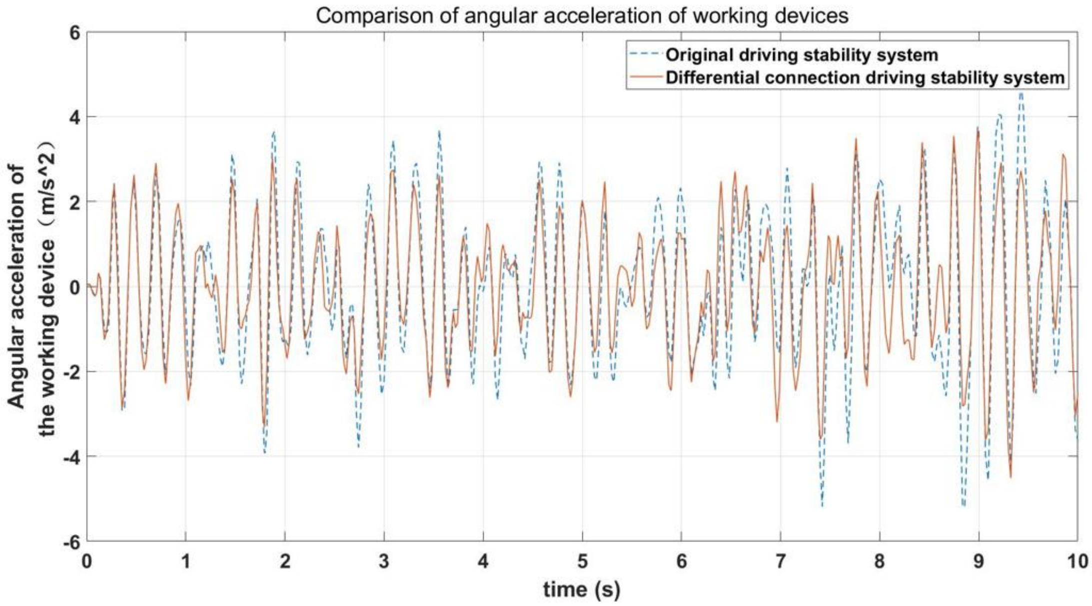


Fig. 7. Comparison of simulation results of angular acceleration of working devices

The simulation results can be concluded that under the effect of road excitation, the peak of body droop acceleration decreases by 30.71% and 44.39% respectively after adding the original driving stability system and adding the differential connection driving stability system; the peak of body angular acceleration decreases by 49.84% and 71% respectively. After optimizing the original driving stability system into differential connection driving stability system, the peak of body droop acceleration further decreased by 19.75%; the peak of body angular acceleration further decreased by 42.18%; the peak of working device angular acceleration decreased by 13.1%.

After adding the driving stability system, the root mean square value of body droop acceleration was reduced from 7.983 to 4.438, a reduction of 44.41%; the root mean square value of body angular acceleration was reduced from 7.858 to 1.571, a reduction of 80%, and the root mean square values of body droop acceleration and body angular acceleration were further reduced by 7.82% and 25.02%, respectively, after optimizing to the differential connection driving stability system. The root mean square value of angular acceleration of the working device was reduced by 30.73%.

5. CONCLUSION

Under the action of random vibration excitation displacement of E grade road surface in the driving condition of the loader after adding the driving stability system, the body vertical acceleration, body angular acceleration and work device angular acceleration are all significantly reduced, which indicates that the driving stability system of the loader can help improve the smoothness of the loader driving. After adding the original driving stability system and differential connection driving stability system respectively, the three kinds of acceleration of the loader are further reduced, and the root mean square of acceleration is also significantly reduced, among which the most significant is the body angle acceleration, which shows that the driving stability system can effectively control the pitch vibration of the body under the driving condition of the loader. Differential connection driving stability system compared with the original driving stability system work when the pressure in the system is greater, can improve the initial filling pressure of accumulator, further control vibration, more conducive to the smoothness of wheel loader driving.

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