

Research and Analysis on Vehicle Multi-body Dynamics Simulation

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Abstract: This article introduces the vehicle multi-body system dynamics and basis of combination, including multi-rigid body system and multi-flexible body system dynamics. By studying the two major problems, the article obtains the research methods: Lagrange equation method, Newton-Euler method and so on. The article first studies the theoretical foundation of multi-rigid body dynamics and multi-flexible body dynamics. And then it constructed the model of front suspension system and rear suspension system model of the vehicle. After that the article proposes the basic parameters of material properties, whole vehicle and suspension stiffness. Finally, it makes the simulation analysis on suspension system, stationary of manipulation and riding smoothness and braking performance which gets the relevant rules and conclusions of vehicle. The results have important guiding significance for practices. *Copyright © 2013 IFSA.*

Keywords: Vehicle suspension systems, Multi-body dynamics, Simulation analysis, Optimization.

1. Introduction

Multi-body system dynamics, which includes multi-rigid body system and multi-flexible body system dynamics, refers to the disciplines of studying the movement of multi-body system. It is developed on the basis of combining classical mechanics and computer, biomechanics, spacecraft control, robotics, vehicle design, mechanical dynamics and other disciplines [2].

In the field of engineering, research systems would face two major problems. One is related to the structural strength analysis of these complex systems [5]. The other is to solve the state's problems of such complex kinematic systems, dynamics and control. Such systems are characterized by the presence of a wide range of systematic components in relation to movement, the variety of connection with formation in topological constraints, complex control link, commonality is the system composed by

the relative motion of multiple objects, which are the reasons that it was called as many-body systems [3]. Multi-body dynamics research methods include: multi-rigid body systems and multi-flexible body systems. In addition, the research methods of multi-body are in common use Lagrange equation method, Newton-Euler method, Graph Theory method, Kane method, Variational method, Spinor method and so on.

The automotive vehicle dynamics studies the relationships between force and movements, which find the inner link and laws of vehicle performance. The development of vehicle dynamics cannot do without analytical mechanics, materials, mechanics, structural mechanics and other disciplines. In order to study vehicle dynamics, it is necessary to establish its dynamic model first (Physical and mathematical models) [4]. The research method of mathematical model plays an increasingly important part in the future of the vehicle dynamics. In a particular study,

the research object model, should be based on the complexity of the problems studied. But the complexity of the model will make a substantial increase in the number of differential equations describing the system motion, which would increase the difficulty of solving the system [1]. Therefore, multi-body theory and methods have become an urgent need when using the combination of classical dynamics of rigid, kinetic of analysis mechanics and computer.

Modeling, analysis and problem solving are always the keys of the research on vehicle dynamics in studying performances of the cars. Multi-body dynamics method is a high-efficient and high-precision analytical method. It is necessary to choose the most effective method according to the circumstances and the nature of the problem being studied. This is particularly important for more complex automotive systems. The steps of the application of the theory on multi-body system dynamics to solve practical engineering problems are as follows [7]: first, simplify the model of the actual vehicle multi-body system. Second, generate the equations of the multi-body system dynamics automatically. Third, accurately deal with the problems in the equations of the multi-body system dynamics.

Optimization design was the beginning of the suspension system in the field of automotive engineering [6]. Design of the automotive suspension systems has a great important influence on automobile ride comfort, handling stability as well as other auto parts. In recent years, with the expansion of the software features of the multi-body dynamics, and in the complex environment of vehicle multi-body system dynamics model, it is possible to consider more factors in the application of the vehicle multi-body system dynamics model, and then to analyze the optimization using a nonlinear model of vehicle. The vehicle active and semi-active suspension becomes a hot research area at home and abroad.

2. Modeling

2.1. Theoretical Basis

2.1.1. The Basic Theory of Multi-Rigid Body Dynamics

ADAMS software uses Cartesian coordinates and the parameters of Euler angles to describe the steric shape of the object and uses Jill Gear rigid integral to solve the problem of the sparse matrix. ADAMS Solver offers a variety of functional maturation solvers and the model can be kinematics, statics, and dynamics analysis. The kinetic equation selected in generalized coordinates is (1), Apply efficient sparse matrix methods to solve

$$q_1 = [x, y, z, \psi, \theta, \phi]^T, q = [q_1, q_2, q_3, \dots, q_n]^T \quad (1)$$

Using Lagrange multiplier method, it can create a system dynamics equation:

$$\frac{d}{dt} \left(\frac{\partial T}{\partial \dot{q}} \right)^T - \left(\frac{\partial T}{\partial q} \right)^T + f_q^T \rho + g^T \mu = Q \quad (2)$$

Completely constraint equation: $f(q, t) = 0$, non-complete constraint equation: $g(q, \dot{q}, t) = 0$. We can simplify the equation (2) as more general form:

$$\begin{cases} F(q, u, \dot{u}, \lambda, t) = 0 \\ G(u, q) = u - \dot{q} \\ \phi(q, t) = 0 \end{cases}$$

If considering the system kinematics, the study was zero degrees of freedom system position, velocity, acceleration and constraint forces, therefore, the system only needs to solve the constraint equation:

$$\phi(q, t) = 0 \quad (3)$$

Using Taylor series to estimate the system a time state value, it can get the equation:

$$y_{n+1} = y_n + \frac{\partial y_n}{\partial t} h + \frac{1}{2!} \frac{\partial^2 y_n}{\partial t^2} h^2 + \frac{1}{3!} \frac{\partial^3 y_n}{\partial t^3} h^3 + \dots,$$

Time step is $h = t_{n+1} - t$. Expanse the equation (3)

in $t = t_{n+1}$, it can obtain $\phi(q_{n+1}, t_{n+1}) = 0$. Then the derivation of the equations can be obtained $\left(\frac{\partial \phi}{\partial q} \right) \dot{q} = -\frac{\partial \phi}{\partial t}$,

$$\left(\frac{\partial \phi}{\partial q} \right) \ddot{q} = - \left\{ \frac{\partial^2 \phi}{\partial t^2} + \sum_{k=1}^n \sum_{i=1}^n \frac{\partial^2 \phi}{\partial q_k \partial q_i} \dot{q}_k \dot{q}_i + \frac{\partial}{\partial t} \left(\frac{\partial \phi}{\partial q} \right) \dot{q} + \frac{\partial}{\partial q} \left(\frac{\partial \phi}{\partial t} \right) \dot{q} \right\} \quad (4)$$

By using the Lagrange multipliers, it is possible to solve the equations:

$$\left(\frac{\partial \phi}{\partial q} \right)^T \lambda = \left\{ -\frac{d}{dt} \left(\frac{\partial \phi}{\partial \dot{q}} \right)^T + \left(\frac{\partial \phi}{\partial q} \right)^T + Q \right\} \quad (5)$$

If the system is static, set speed, acceleration is zero, then it can obtain the following equation of statics of system:

$$\begin{bmatrix} \frac{\partial F}{\partial q} & \left(\frac{\partial \phi}{\partial q} \right)^T \\ \frac{\partial \phi}{\partial q} & 0 \end{bmatrix} \begin{Bmatrix} \Delta q \\ \Delta \lambda \end{Bmatrix}_j = \begin{Bmatrix} -F \\ -\phi \end{Bmatrix}_j \quad (6)$$

Using ADAMS software to create multi-body dynamics model, kinetic equation is generally implicit nonlinear with differential-algebraic

equations mixed. By solving the equations, the dynamic model can be obtained boundary conditions of all the components (power, speed, acceleration). The methods of solving process and data flow can be seen in Fig. 1. ADAMS software can be used for computing time, reading the raw input data correctly and determining the degree of freedom of the whole system. If the system is not zero degree, the action of kinetic analysis or static analysis would be carried out. If the value of divergence, the calculation fails, and the software would check the model recalculated.

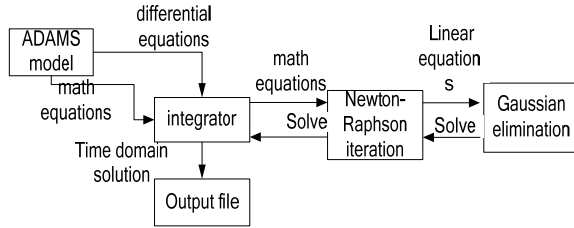


Fig. 1. Dynamics solving process and data flow.

2.1.2. The Basic Theory of Multi- Flexible Body Dynamics

In practical engineering, multi-body system composed in general by the rigid body of the interaction and flexible deformable body, each component in the system may have translational and rotational space. The theory of Flexible multi-body system dynamics is about the composition of systematic theoretical analysis of the rigid body and flexible body. The relevant flexible body theory and methods are as follows: flexible body discretion methods and modal integration method and a combination of many-body theory and finite element method. Discrete flexible body is consistent with the model of rigid nature.

The flexible body spatial vector position of the i node is determined as follows:

$$r_i = x + A(s_i + \Phi_i q), \quad (1)$$

where x is the spatial position vector of the local coordinate system, A is the origin direction cosine matrix of the local coordinate system, s_i is the spatial position vector of i node of the local coordinate system before deformation. Φ_i is the freedom of movement modal matrix sub-block of i node. q is the modal amplitude vector. The equation derivative to time, get the node speed [12]:

$$v_i = \dot{x} - A(\tilde{s}_i + \tilde{\Phi}_i q)w + A\Phi_i \dot{q} = [E - A(\tilde{s}_i + \tilde{\Phi}_i q)B + A\Phi_i] \dot{\xi} \quad (8)$$

At the same time, it can get the kinetic and potential energy equation:

$$T = \frac{1}{2} \sum_{i=1}^N m_i v_i^T v_i = \frac{1}{2} \dot{\xi}^T M(\xi) \dot{\xi} \quad (9)$$

$$V = \frac{1}{2} \dot{\xi}^T K(\xi) \dot{\xi}$$

Substituting into Lagrange equation, it can get ADAMS flexible body equations:

$$M \ddot{\xi} + M \dot{\xi} - \frac{1}{2} \left(\frac{\partial M}{\partial \xi} \dot{\xi} \right)^T \dot{\xi} + K \xi + f_g + D \dot{\xi} + \left(\frac{\partial \Phi}{\partial \xi} \right)^T = Q \quad (10)$$

The basic principle of modal integration method is flexible body as a collection of nodes of the finite element model, using the Lagrangian method to establish multi-body system dynamics. Then it can get the finite element equation:

$$[M]\{\ddot{x}\} + [C]\{\dot{x}\} + [k]\{x\} = \{f_a\} \quad (11)$$

$$\frac{\partial f}{\partial \dot{u}} \{\Delta \dot{u}\} + \frac{\partial f}{\partial \dot{q}} \{\Delta \dot{q}\} + \frac{\partial f}{\partial q} \{\Delta q\} = \{f_a\} \quad (12)$$

According to boundary conditions, the deformation, force and torque can be obtained.

2.2. The Modeling of Vehicle Multi-Body Dynamics

Vehicle is a complex multi-DOF [8] "quality-stiffness-damping" dynamics system. In the process of establishing vehicle dynamics simulation model, the main steps are: 1) move parts and constraints of the vehicle systems, obtain kinematic parameters of the model, establish kinematic model; 2) verify the DOF and correctness of the model; 3) Obtain kinetic parameters of the model; 4) The dynamic model validation, tuning, and simulation calculation; 5) Deal with the simulation results.

In the modeling process, the vehicle model has some reasonable simplifications, following the assumptions for the simulation model of dynamics analysis:

1) Vehicle systems, in addition to elastic components, rubber damping elements and stabilizer bar, the other parts considered to be rigid, its deformation are considered.

2) The structure of the car's front and rear suspension quality system are similar, which can imply the automobile suspension is symmetry in axis.

3) Roll bar is considered as flexible body or simplified into a rigid lever and torsion spring mechanism when performing dynamic analysis.

4) Cab, chassis and body interior accessories are considered as an object - sprung mass.

We can obtain front suspension system model and rear suspension system model after simplifying and abstraction, which can be seen in Fig. 2 and Fig. 3.

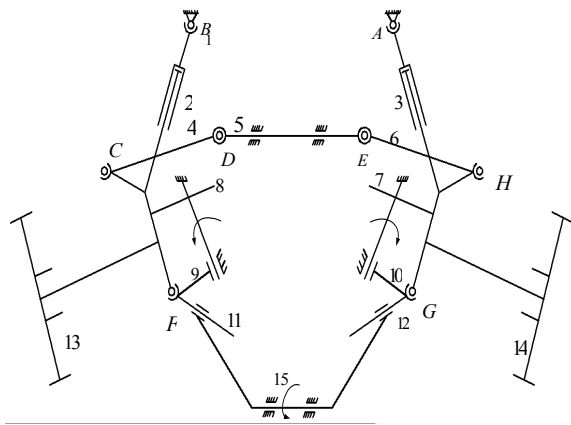


Fig. 2. Front suspension system model.

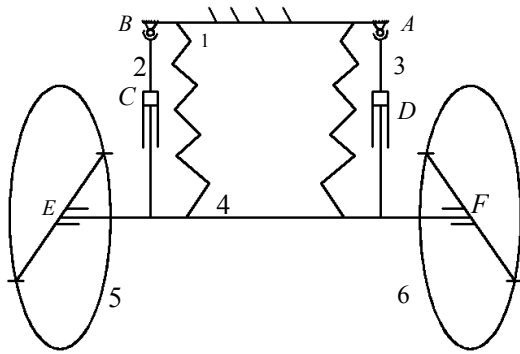


Fig. 3. Rear suspension system model.

For front suspension system, each gimbal hinge and hinge pillar gives four constraint equations, each ball joint gives three constraint equations and both of each pillar hinge and slip hinge gives five constraint

equations [9]. The front suspension system consists of constraint equations [11]: $m = 1 \times 6 + 6 \times 4 + 2 \times 4 + 6 \times 5 + 1 \times 5 + 6 \times 3 = 91$, system DOF: $DOF = 6n - m = 6 \times 16 - 91 = 5$. Respectively represent front left wheel axis, front right wheel axis, front left shock absorber, front right shock absorber, steering rack, left and right wheels.

For rear suspension system, there are six component members, which are two gimbal joints, two slip column hinges, two column hinges, so, the rear suspension system consists of constraint equations: $m = 1 \times 6 + 2 \times 4 + 2 \times 4 + 2 \times 5 = 32$, system DOF: $DOF = 6n - m = 6 \times 6 - 32 = 4$. Respectively represent axle moves up and down, swing, left and right wheel rotation.

2.3. The Determination of Basic Parameters

1) In order to get correct simulation, the article obtained the material properties of stabilizer bar of the vehicle after analyzing, which is shown in Table 1.

2) In order to improve the accuracy of simulation results, the article must identify the basic parameters of the simulation model of the car as accurately as possible. The main technical parameters of the whole vehicle are shown in Table 2.

3) Suspension stiffness and damping is mainly determined by the springs and shock absorbers. After analyzing, the article got the stiffness parameter values of vehicle suspension components, which can be seen in Table 3. Tire rolling resistance coefficient is 8.4578×10^{-3} .

Table 1. Material properties parameters.

Modulus of elasticity E	Density ρ	Poisson's ratio μ
2.1×10^{11} Pa	7.8×10^{-6} kg/mm ³	0.3

Table 2. The whole vehicle technical parameters.

Engine Model	Maximum engine power	Maximum torque	Engine displacement	Front axle load full load
SOFIM8140.27S	76 kW/3800 rpm	230 N.m/rpm	2.499 L	1590 kg
Rear axle load full load	Maximum speed	Load center of gravity height	Tire Size	Fully loaded vehicle total mass
2430 kg	120 km/h	1010 mm	6.5 R 16 C	4020 kg

Table 3. Suspension stiffness parameters.

Front suspension stiffness N.m/deg	Rear suspension spring stiffness N/mm		Radial stiffness of rubber bushing
	The main spring stiffness	Composite stiffness	
80.4	65	150	3500

3. The Simulation Analysis and Verification

3.1. Suspension System Simulation Analysis

1) Front suspension system simulation analysis.

In front suspension, as given around under arm corner displacement input function $\theta = 3/4 \sin(2\pi t)(\text{rad})$ [10], given the steering rack displacement input function $Z = 0(\text{mm})$. Kinematic analysis, simulation results are shown below. As can be seen from Fig. 4, arm angular displacement changes by sine law. In the direction of travel of the car, the front and rear wheel center displacement (X direction) is small, which will help improve the suspension performance at high speed of the car.

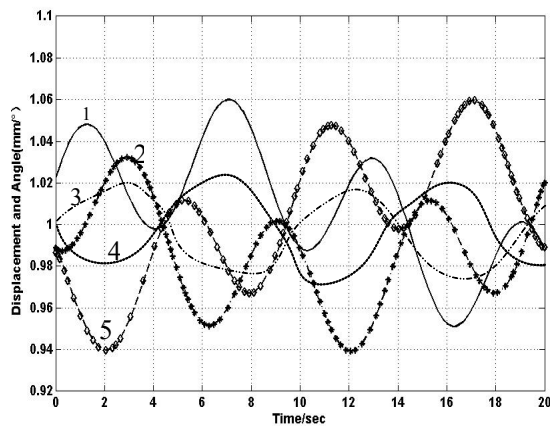


Fig. 4. Front suspension system simulation.

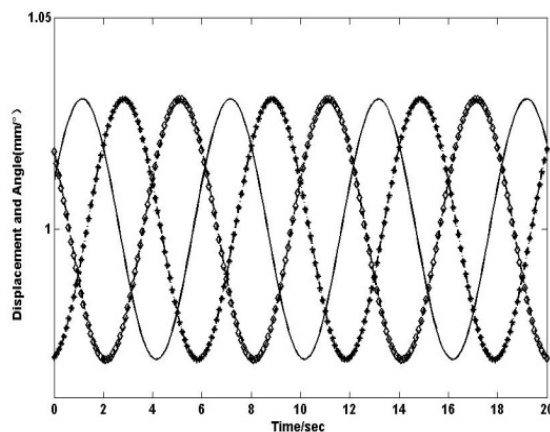


Fig. 5. Vertical stiffness run-out simulation.

Displacement of the wheel center in all directions changes with sine law, given the steering angle is zero, torsion stiffness of the spring itself is linear, however, due to the vertical stiffness of the front suspension is nonlinear. Vertical stiffness of the suspension of the wheel center simulation calculation results are shown in Fig. 5.

2) Rear suspension system simulation analysis.

In the rear suspension, given axle displacement input function $Y = 20 \sin(2\pi t) \text{mm}$. The system is zero degrees of freedom. Kinematic analysis, simulation results are shown in Fig. 6. From the simulation results it can see each displacement of the rear wheel center in sinusoidal rule. Vertical displacement movement remains substantially constant.

Body roll occurs in a small angle and left and right wheels are maintained vertical. A left and right rear wheel has linear relationship. The result can be seen in Fig. 7.

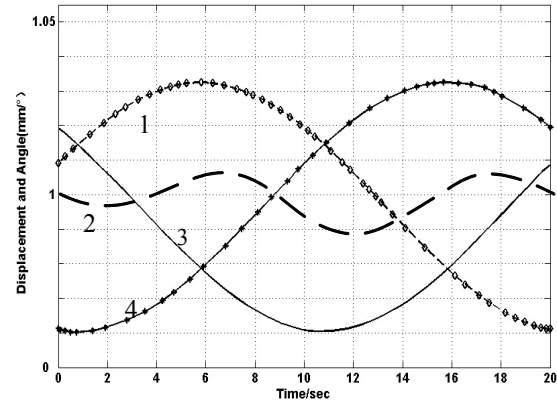


Fig. 6. Rear suspension system simulation.

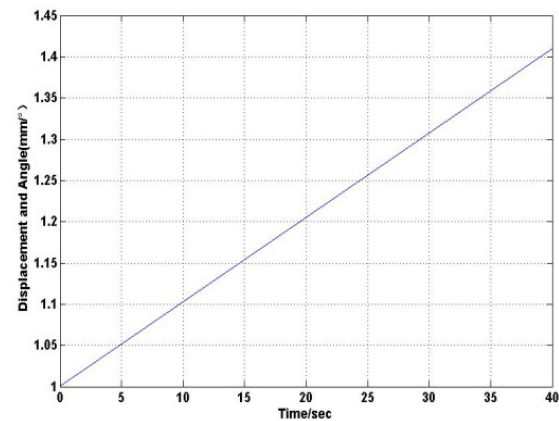


Fig. 7. Body roll linear relationship.

3.2. The Stationarity of Manipulation and Riding Smoothness Simulation Analysis and Verification

1) The stationarity of manipulation simulation analysis and verification.

Generally, only the appropriate vehicle under-steer has good handling and stability. Study on Stable Steering feature is an important aspect of handling and stability studies. By steady-state steering characteristics simulation, it can be seen that the car gradually stabilized shown in Fig. 8.

2) Riding smoothness of the simulation analysis and verification.

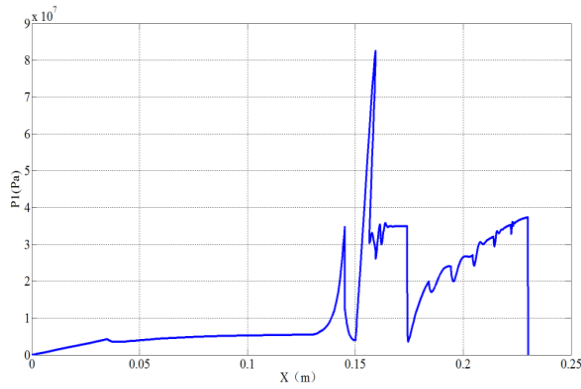


Fig. 8. Stationarity of manipulation simulation.

Sprung mass of the vehicle suspension system (body and frame) and the unsprung mass (axle and wheels) of vibration have a great impact on riding comfort car insurance. In order to verify the correctness of the model vehicle rigid evil together, the car riding simulation was made. Real vehicle test unload condition, according to specified method GB14783-94 test conduct, it can get the results (see Table 4).

From the Table 4, it can be seen that the calculation results are basically consistent with the experimental results. The natural frequency is 1.59 Hz, the system type is mainly vertical vibration of rear axle and the system type is mainly vertical vibration of front axle. Obtained simulation results as shown in Fig. 9, it shows that the cars gradually become smoother.

In order to further test the accuracy of vehicle dynamics simulation model of rigid-flexible coupling, the article compared the calculation results between vehicle smoothness simulation analysis and practical test results. The results can be seen in Table 5.

From the Table 5, it can be seen that the coupled dynamics model is reasonable. For the reasons why there is an error, on one hand, it is due to the

establishment of the simulation model in some components of the system which can be simplified; on the other hand, errors many be caused by numerical calculation and test measurements.

Table 4. Comparison of inherent characteristics and real test.

Program (unload)	Inherent characteristics	Real test data
Rear suspension sprung mass natural frequency (Hz)	2.15	2.33
Rear suspension damping Bibi	0.0770	0.0780
Rear axle (wheel) natural frequency (Hz)	13.96	13.10
Front suspension sprung mass natural frequency (Hz)	1.59	1.63
Front axle (wheel) natural frequency (Hz)	0.1423	0.1561
Front axle (wheel) natural frequency (Hz)	14.68	15.21

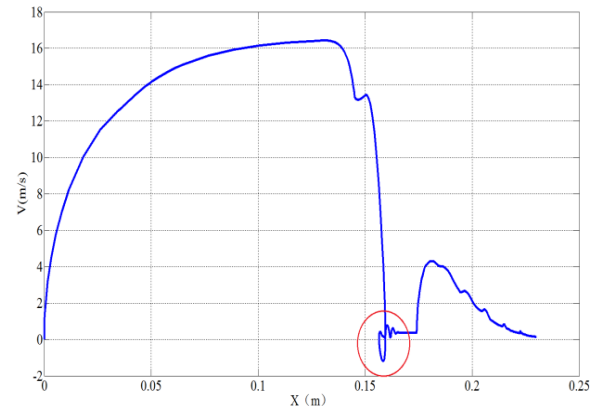


Fig. 9. Riding smoothness simulation.

The key evaluation indicators of car riding are the dynamic response results in the car body under random input. The result can be seen in Fig. 10.

Table 5. Comparison of simulation results and practical results.

Program(load)	Vibrates vertically momentum	Speed(Km/h)				
		40	50	60	70	80
Practical results		0.5248	0.7124	0.7786	0.9641	1.1234
Simulation results		0.5098	0.6916	0.7690	0.9413	1.0986

3.3. Braking Performance Simulation Analysis

3.3.1. Simulation Analysis of Linear Brake

Linear braking performance of the car is one of the main performances. There are three areas, which are braking efficiency, constant direction and braking performance. Function expression of control link is as follows:

$$\begin{cases} \phi = \phi_p e^{-0.00112Y^{-2226}} - Q \\ Q = \begin{cases} 0 & Y \leq 0.2 \\ 0.24(Y - 0.2) & Y > 0.2 \end{cases} \end{cases} \quad (12)$$

We can obtain the car speed and the angular response curve during braking after analysis, which is shown in Fig. 11.

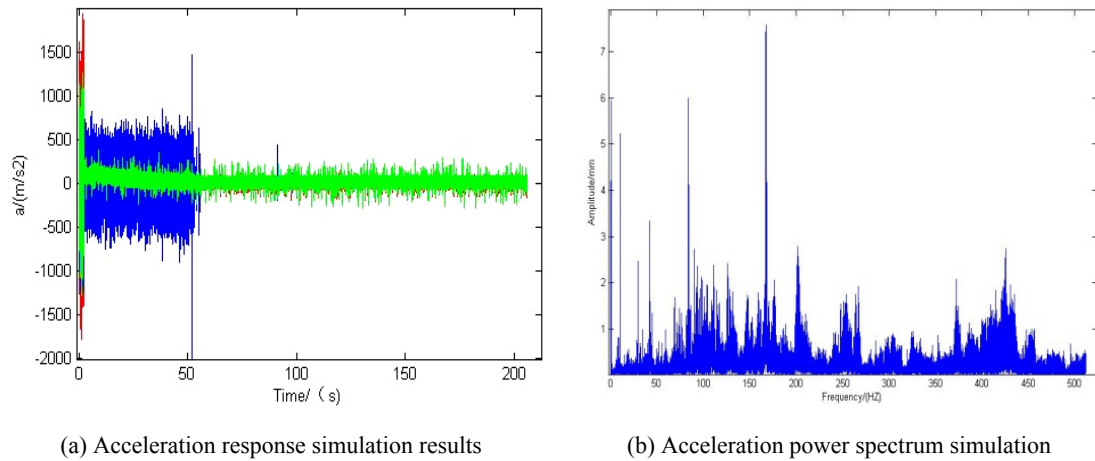


Fig. 10. Acceleration response simulation and power spectrum simulation.

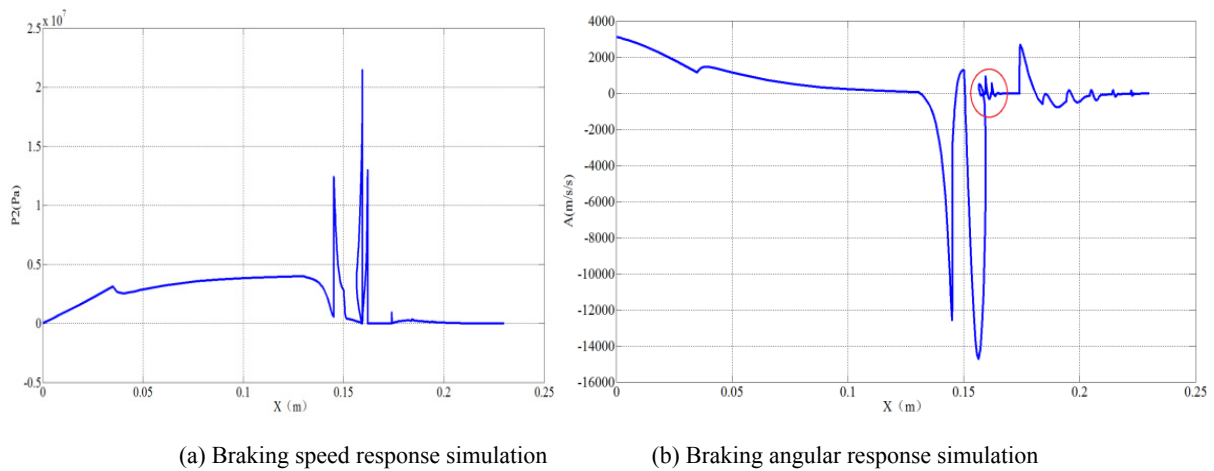


Fig. 11. Braking speed and angular response simulation.

3.3.2. Cornering Brake Simulation Analysis

The braking condition of car turning is often encountered in common driving life. In the simulation, the initial state of the car is the uniform linear motion. Giving the car a step input and steering wheel angle by adjusting the steering wheel angle and initial speed, it could make the car get into a

certain lateral acceleration steady of circular motion. Braking torque is applied to the brake. Road adhesion coefficient and suspension stiffness simulation results affect the car's braking stability (see Fig. 12).

The article also can get the simulation results of the body vertical acceleration response and body vertical acceleration power spectrum, which can be seen in Fig. 13.

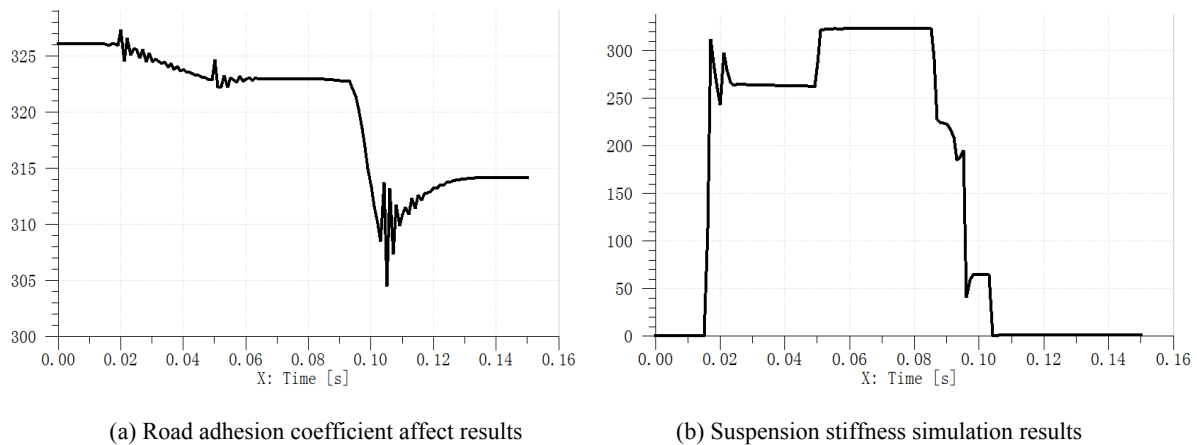
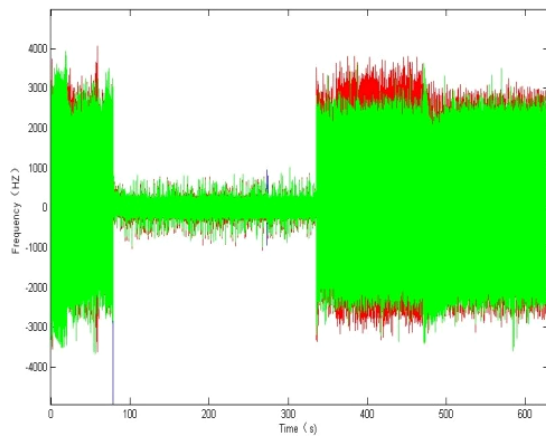
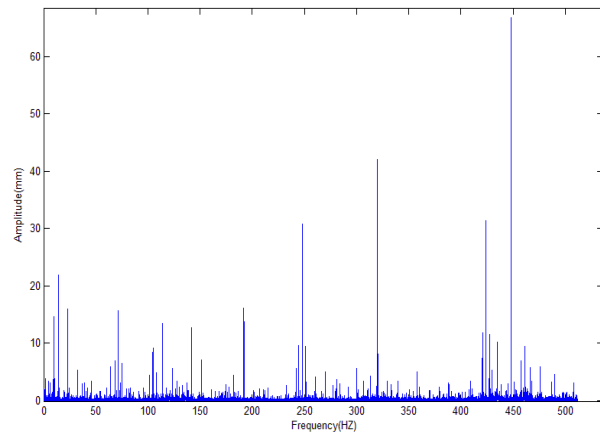


Fig. 12. Road adhesion coefficient and suspension stiffness affect results.



(a) Vertical acceleration response simulation



(b) Vertical acceleration power spectrum simulation

Fig. 13. Vertical acceleration response and power spectrum simulation.

4. Conclusion

In this article, it used the basic theory and method of multi-body dynamics to study the simulation of the performance vehicle dynamics. The article established a multi-body model with various parameters. The analysis result of the handling stability on kinematics simulation shows that riding comfort and braking performance can get a set of simulation results and influence of parameters, which can provide guidance for the car modification and design.

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