

Research on Ride Comfort Model of Wheel Motor Driving Vehicle Based on Matlab/Simulink

Wenwen Xiao*, Feihu Xuan, Huanghuang Zhang

Shanghai University of Engineering Science, Shanghai, China

Email: *1663173083@qq.com

How to cite this paper: Xiao, W.W., Xuan, F.H. and Zhang, H.H. (2017) Research on Ride Comfort Model of Wheel Motor Driving Vehicle Based on Matlab/Simulink. *Engineering*, 9, 950-961.
<https://doi.org/10.4236/eng.2017.911057>

Received: September 11, 2017

Accepted: November 26, 2017

Published: November 29, 2017

Copyright © 2017 by authors and Scientific Research Publishing Inc. This work is licensed under the Creative Commons Attribution International License (CC BY 4.0).

<http://creativecommons.org/licenses/by/4.0/>



Open Access

Abstract

Simulink is a visual simulation tool in MATLAB; through Simulink software, to establish a model can reduce the amount of programming workload, and improve the efficiency of the establishment of automotive models. The ride comfort of the vehicle is a measure of the most basic indicators of a car performance. By establishing a ride comfort model in Matlab/Simulink, the wheel motor electric vehicle mainly affects the smoothness of the car mainly in the following aspects: pavement, tire, suspension, motor and so on. Through the establishment of the above model, we can effectively study the wheel motor drive electric vehicle ride comfort research.

Keywords

Ride Comfort, Simulation Model, Hub Motor

1. Introduction

With the development of society and scientific progress, people's requirements on the ride comfort (that is, the car ride comfort) are getting higher and higher; the car's ride comfort is mainly to keep the car in the process of vibration and shock environment. The impact on the comfort of the occupants is within a certain limit [1]. Due to the limited number of experimental equipment, a lot of experiments need to simulate the simulation to establish a virtual simulation environment, as well as virtual conditions. However, the most important point of simulation is the establishment of simulation model. The closer the simulation model is to the real conditions, the more conducive the experiment will be. In 2009, Zhang Lu of Wuhan University of Technology put forward different me-

thods to analyze and optimize the ride comfort of wheel motor, and improved the ride comfort of the vehicle [2], on the car ride up to the account from the car seat, seat, fully considering the factors affecting the ride comfort [3]. To control the quality of the motor is given a quantitative index of the wheel assembly of the suspension; the suspension structure selection and control strategy are given. To solve the problem of deterioration of reference of an electric vehicle ride comfort, from the choice of the motor, such as the application of lightweight materials of the motor, and the degree of integration of motor and wheel, we improved suspension to other direction, compared with the traditional automobile hub motor effectively saves a lot of energy to meet the needs of modern life [4] [5]. In recent years, an electric vehicle has a rapid development in foreign countries, there are some researches on several properties of electric wheel, the car ride for outstanding, whether domestic or foreign people on the vehicle comfort put forward higher requirements, but the wheel motor drive electric vehicle becoming a hot research direction.

2. The Establishment of Random Pavement

Now that the road surface roughness of the time domain model methods are harmonic superposition method, white noise method and other methods [6] [7] These methods have made some assumptions on the road, assuming that the road contour in line with the Gaussian distribution, due to white The noise method is realized by the differential equation, and the calculation is relatively small, using MATLAB/Simunlink to facilitate modeling. As shown in **Figure 1**.

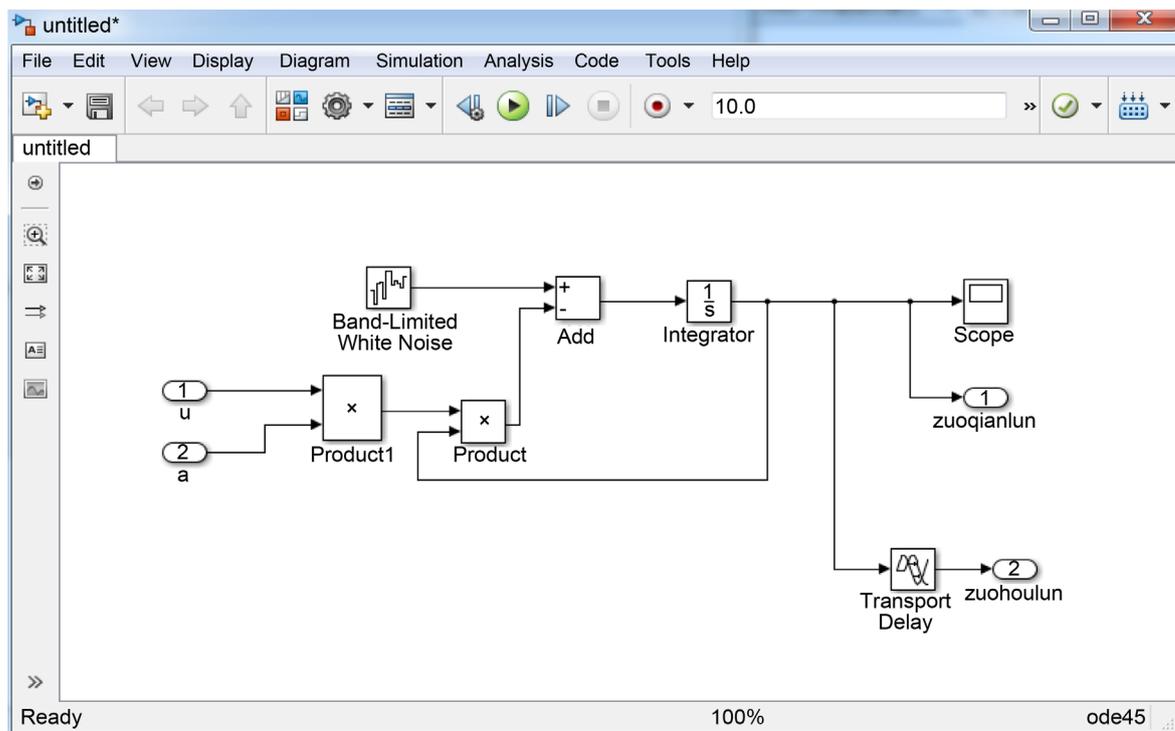


Figure 1. Pavement model connection diagram.

In order to reflect the authenticity of the road, a white noise model with cutoff frequency is adopted. The model can be used to reflect the actual characteristics of the road spectrum closer to the horizontal line at low frequencies.

The following formula can be used to analyze the pavement in time domain for different pavement grades and vehicle speeds.

$$\dot{x}_{of}(t) = -2\pi n_{oo} u x_{of}(t) + 2\pi n_o \sqrt{G(n_o)} u w(t) \tag{1}$$

This formula, $x_{of}(t)$ is a random displacement of the front wheel, unit m; n_{oo} is the cut-off space frequency, $n_{oo} = 0.011 \text{ m}^{-1}$. u is the speed of the vehicle, unit m/s; n_o is the reference space frequency, $n_o = 0.1 \text{ m}^{-1}$; $G(n_o)$ is the reference space frequency under the road power density, which is the road roughness coefficient, unit m^3 ; $w(t)$ is the Gaussian white noise with mean zero [8].

3. Reference Coordinate System Selection

Before the simulation model is built, the reference coordinate system should be developed so that the movement state of the vehicle can be better described. Commonly used reference coordinate system mainly has tire coordinate system, vehicle coordinate system and earth coordinate system. The vehicle coordinate system is a moving coordinate system, used to describe the movement of the car state, which is consolidated in the movement of the car [9].

Figure 2 shows the consolidation of the car in the oyez right-angled coordinate system is the vehicle coordinate system. When the vehicle is in a stationary state on the horizontal road, the x-axis is parallel to the ground, and the y-axis points to the left side of the driver. The z-axis passes through the center of the center, and the origin of the coordinate system. Can make it coincide with the center of mass [10].

Figure 3 for the tire coordinate system, perpendicular to the wheel rotation axis of the tire in the sub-plane known as the wheel plane, Z axis perpendicular to the ground plane, the provisions of the above point is positive. Y axis in the ground plane, the provisions of the direction of the direction of the wheel when

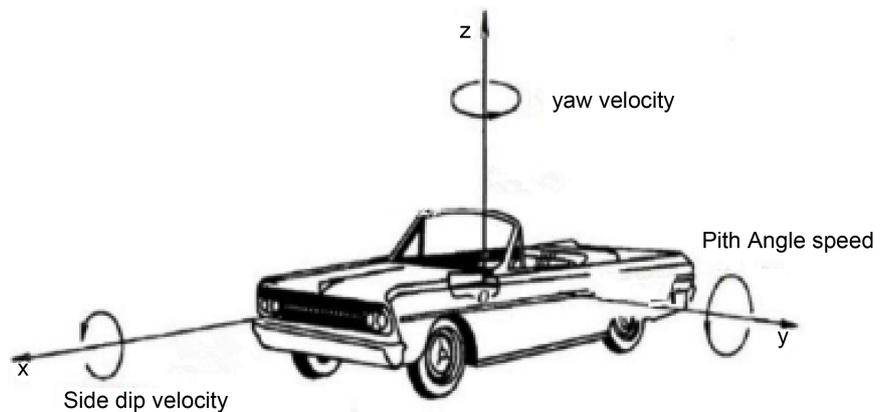


Figure 2. Vehicle coordinate system.

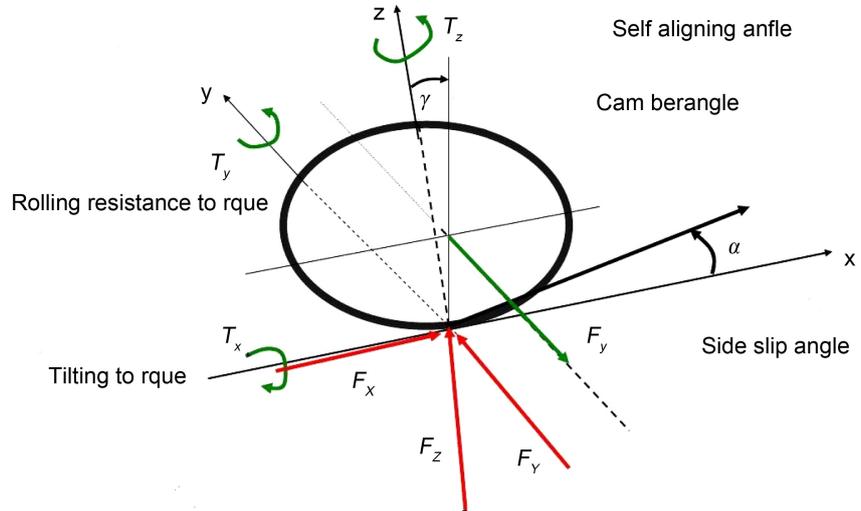


Figure 3. Tire coordinate system.

the direction of the left is positive, the wheel plane and the ground plane of the intersection of the X axis, the provisions of forward is [11]. The origin of the coordinate system is the intersection of the intersection of the wheel plane and the ground plane and the projection line of the wheel rotation axis on the ground plane.

The geodetic coordinate system is used to describe the absolute motion of the vehicle. Its origin is fixed in a certain coordinate point on the ground. The positive direction of the Z axis is the direction perpendicular to the ground, the positive direction of the X axis is the direction of the initial state when the vehicle is running. The positive direction of the Y axis is the left side of the vehicle's initial state direction.

4. Body Dynamics Model

According to the vehicle in **Figure 4** in the XY plane within the force analysis and coordinate system requirements, in the horizontal plane, the car's motion equation is:

$$M_i(\dot{u} + qw - rv) = M_i h'(\dot{q} + pr) + \sum_{i=1}^4 F_{xi} - M_i g \sin a_g - \frac{1}{2} C_D A \rho_a u^2 - F_f \quad (2)$$

$$M_i(\dot{v} + ru - pw) = -M_i h'(\dot{p} + qr) + \sum_{i=1}^4 F_{yi} \quad (3)$$

$$I_{zz} \dot{r} = (I_{xx} - I_{yy})pq + I_f(F_{y1} + F_{y3}) - I_r(F_{y2} + F_{y4}) + \frac{T}{2}(F_{x1} + F_{x2}) - \frac{T}{2}(F_{x3} + F_{x4}) \quad (4)$$

$$F_{xi} = F_{ti} \cos \delta_{Ti} - F_{si} \sin \delta_{Ti} \quad (5)$$

$$F_{yi} = F_{ti} \sin \delta_{Ti} + F_{si} \cos \delta_{Ti} \quad (6)$$

$$I_{yys} \dot{q} + (I_{xss} - I_{zss})pr = L_f \left(\frac{F_{y1}}{B_1} + \frac{F_{y3}}{B_3} \right) + L_r \left(\frac{F_{y2}}{B_2} + \frac{F_{y4}}{B_4} \right) + h \sum_{i=1}^4 F_{xi} \quad (7)$$

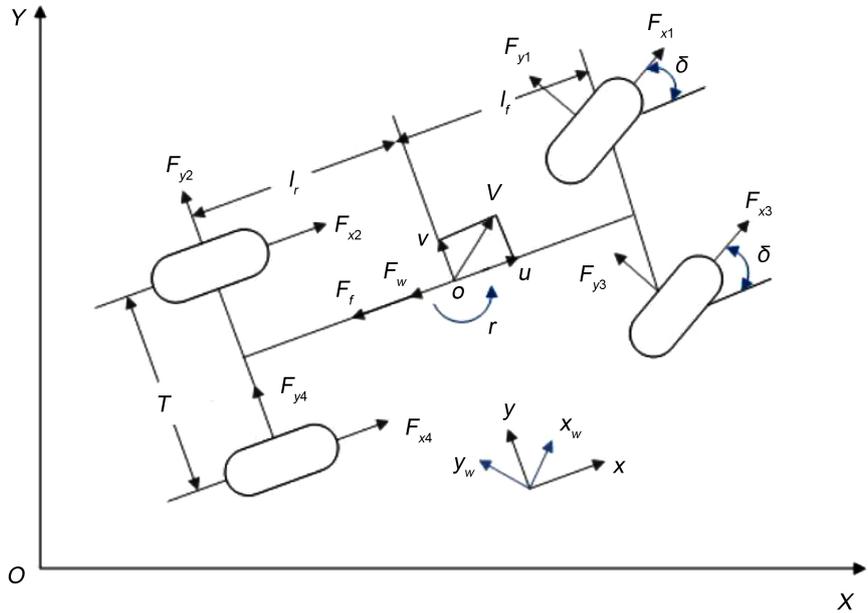


Figure 4. The force analysis of vehicle in the XY plane.

$$M_s (\dot{w} + pv - qu) = \sum_{i=1}^4 \frac{F_{y_i}}{B_i} \tag{8}$$

$$I_{xxs} \dot{p} + (I_{zss} - I_{yys}) qr = \sum_{i=1}^4 A_i F_{v_i} + M_s gh' \sin f - M_s h' (\dot{v} + ru - pw) \tag{9}$$

$$F_{v_1} = K_{s1} (Z_{u1} - Z_s + L_f \sin f + d_1 \sin f) + C_{s1} (w_{u1} - w + L_f q \cos f + d_1 p \cos \theta) \tag{10}$$

$$F_{v_2} = K_{s2} (Z_{u2} - Z_s - L_r \sin \theta + d_2 \sin f) + C_{s2} (w_{u2} - w + L_r q \cos f + d_2 p \cos \theta) \tag{11}$$

$$F_{v_3} = K_{s3} (Z_{u3} - Z_s - L_f \sin \theta + d_3 \sin f) + C_{s3} (w_{u3} - w + L_f q \cos \theta + d_3 p \cos \theta) \tag{12}$$

$$F_{v_4} = K_{s4} (Z_{u4} - Z_s - L_r \sin \theta + d_4 \sin f) + C_{s4} (w_{u4} - w - L_r q \cos \theta - d_4 p \cos \theta) \tag{13}$$

$$\text{Let } B_i = \frac{a_i + b_i}{b_i}, \quad A_i = \pm \left(d_i - (d_i - b_i) \frac{a_i}{a_i + b_i} \right) \tag{14}$$

This formula, i is the wheel number ($i=1,2,3,4$); M_s is the vehicle's sprung mass; M_i is the total mass of the car; δ_{Ti} is the wheel steering angle, rad; F_f is the tire rolling resistance; ρ_a is the air density, $\text{kg}\cdot\text{m}^3$; I_{xx} , I_{yy} , I_{zz} Respectively, for the vehicle around the x, y, z axis of inertia; g is the gravitational acceleration; w_{ui} is the vertical speed of the first wheel, $i=1,2,3,4$; B_i is the vehicle suspension related parameters; a_g is the front projection area of the car; a_g is the vertical velocity of the body; a_g is the slope of the road; F_{y_i} is the tire lateral force, $\text{kg}\cdot\text{m}^2$; I_{xxs} , I_{zss} , I_{yys} are the moment of inertia of the sprung mass around their respective axes of rotation, $\text{kg}\cdot\text{m}^2$; For the corresponding one of the wheel suspension geometry parameters; K_{si} The action of the first wheel on the suspension; K_{si} is the stiffness of each suspension; C_{si} is the damping of each suspension; C_D is the air resistance coefficient; θ is the pitch angle; φ is the roll angle; d_i Distance from the center of the sus-

pension to the center of mass of the sprung mass; z_{ui} The center of mass of the non-sprung mass; v is the lateral velocity of the body; z_s is the centroid height of the sprung mass; p is the roll angle; q is the pitch angular velocity; r is the yaw rate; u is the longitudinal velocity of the body; F_{xi} is the tire longitudinal force [12].

According to the equation to build out the body model shown in **Figure 5**.

5. Tire Model

In this paper, the “magic formula” tire model to carry out dynamic simulation analysis. Because the tire model can accurately predict the tire performance, it is widely used, it is a combination of triangular function to fit the tire test data, obtained a set of the same form and can simultaneously express longitudinal force, lateral force and back Positive torque of the tire model [13], in the form of the following:

$$y = D \sin \{ C \arctan [Bx - E (Bx - \arctan Bx)] \} \tag{15}$$

$$Y(x) = G(x) y(x) + S_y \tag{16}$$

$$x = X + S_h \tag{17}$$

where, y is the positive moment; Lateral force or lateral force; C represents

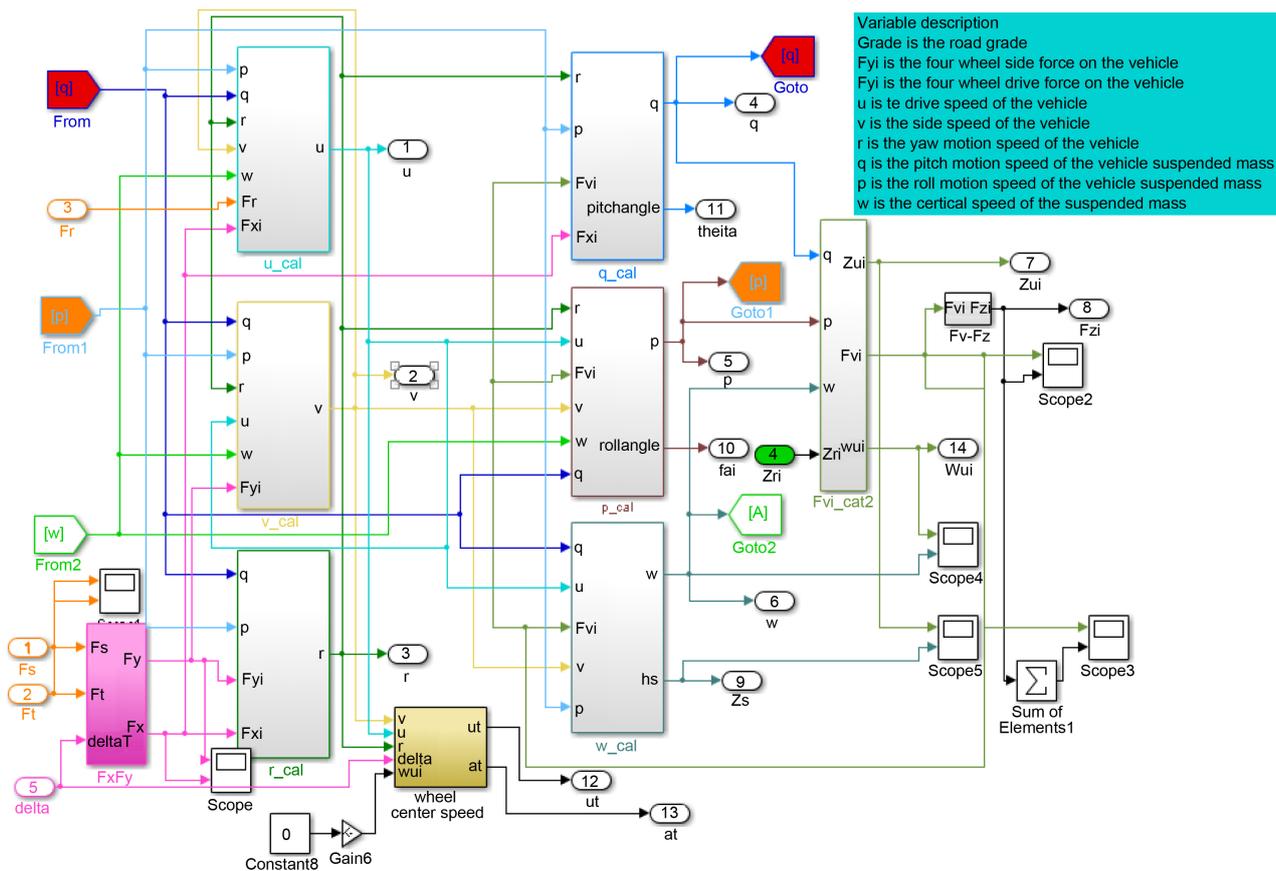


Figure 5. Vehicle body dynamics model.

the shape factor; $G(x)$ is the weight function of the slip and side-side compound conditions; B represents the stiffness factor; D represents the peak coefficient; x can be expressed as the tire side angle or the longitudinal slip rate, respectively; S_y represents the vertical offset; E represents the curvature factor; S_h represents the horizontal offset. The magic formula is illustrated by **Figure 6**, and the curve shown in the figure may be a positive moment, a lateral force or a longitudinal force curve, where, when determined, the product of the coefficient, which is the slope of the curve, that is, the shape of the curve, which determines the shape of the curve. The value can be determined by the peak and steady-state values [14]. The coefficient is used to control the curvature at the peak of the curve,; the amount

$$E = \left\{ Bx_p - \tan \left[\frac{\pi}{(2C)} \right] \right\} / \left[Bx_p - \arctan (Bx_p) \right] \tag{18}$$

6. Motor Model

In this paper, the four-wheel hub motor is a permanent magnet brushless DC motor. This kind of motor has the advantages of long service life, strong overload capacity, high power density, strong reliability and large starting torque compared with other kinds of motors, with a wide range of speed and good mechanical properties of the advantages [15].

The maximum braking torque of the hub motor has a great influence on the regenerative braking distribution strategy of the wheel motor-driven electric vehicle, which in turn affects the effect of braking energy recovery [16]. So the choice of wheel motor to meet the requirements of the electric car, the following is the selected motor, the motor parameters shown in **Table 1**.

Figure 7 for the Matlab/Simulink platform to build the motor model.

Four wheel hub motor drive electric vehicle motor speed and speed between the following relationship:

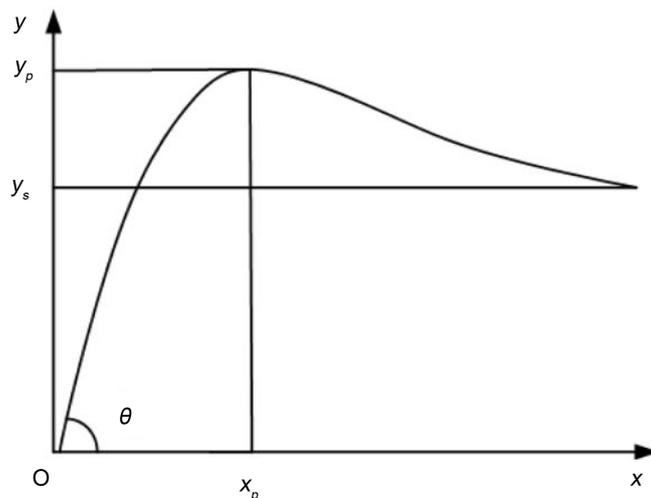


Figure 6. Illustrates the “magic formula” in the parameters of the tire characteristic curve.

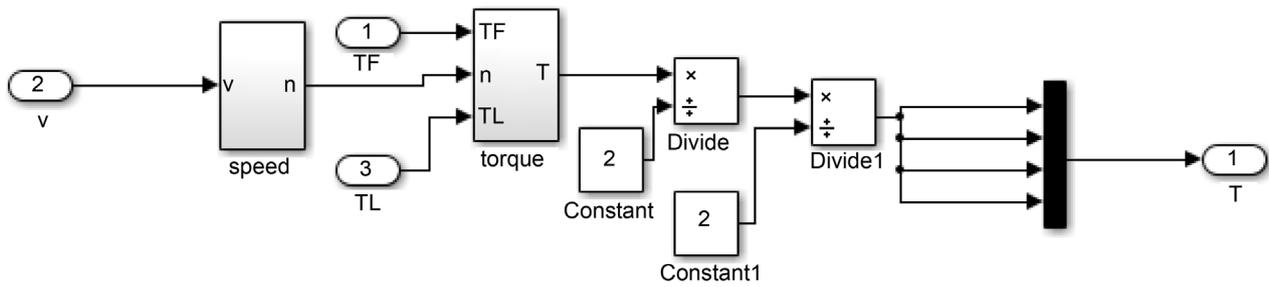


Figure 7. Motor model.

Table 1. Motor parameter data.

parameter	value	unit
Rated speed	4000	rpm
Peak speed	11,500	rpm
rated power	42	kW
Maximum power	90	kW
Rated torque	100	nm
Peak torque	210	nm
Motor efficiency	94%	

$$n = \frac{v}{0.377r} \tag{19}$$

where, r is the wheel rolling radius; n is the motor speed, (r/min); v is the driving speed, km/h.

The relationship between the wheel brake force and the wheel motor braking torque is:

$$T_m = F_m g r_w \tag{20}$$

where, T_m is the wheel motor braking torque; r_w is the wheel rolling radius; F_m is the hub motor braking force.

Since the hub motor has its own peak torque limit, the required braking force for the braking force distribution by fuzzy identification cannot be dispatched to the hub motor unconditionally [17]. In order to extend the service life of the hub motor, the generator torque should be two the smaller the braking torque of the brake. The peak torque of the hub motor at any speed can be measured by the test of the motor peak torque data set [14]. Figure 8 is the motor torque calculation of the module diagram

7. Suspension Model

The suspension of the car suspension system is non-linear and is multi-degree free. As shown in Figure 9. The degree of freedom will be more difficult to establish the model, so this article on the car suspension system two degree of freedom 1/4 model for simplified analysis [18] [19] [20]. To get a simplified model, the following assumptions were made: 1) body and frame are rigid, and in the

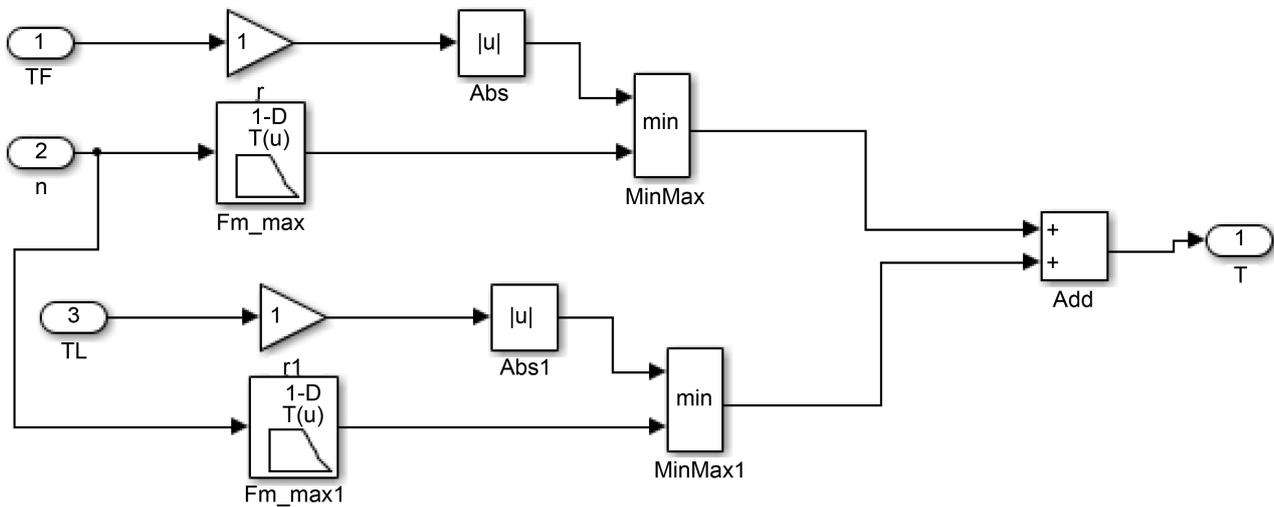


Figure 8. Module diagram of motor torque calculation.

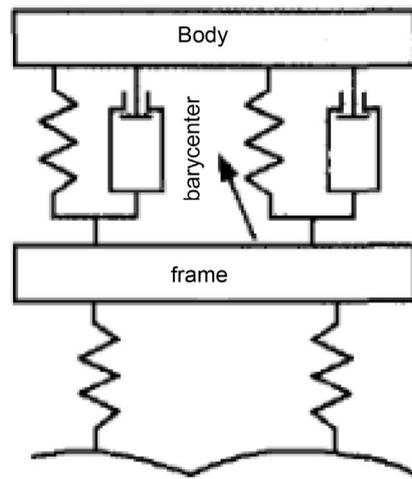


Figure 9. Suspension system model.

course of the study to ignore the body suspension of the elasticity and damping; 2) the car on the road evenly and always with its contact, no impact occurred; 3) the stiffness of the suspension and the stiffness of the tire are linear functions of displacement, and ignore the tire damping; 4) The input function for road displacement can be assumed to act at the center of the tire and ground contact point.

Figure 10 m_2 is the quality of the body, B is a damper, m_1 is the frame, k_1 represents the elastic element in the tire system.

By the Newton's second law we can see that the kinematic equation of the suspension system is

$$m_1 \ddot{x} = B(\dot{x}_0 - \dot{x}) + k_2(x_0 - x) + k_1(x_1 - x) \tag{21}$$

$$m_2 \dot{x}_0 = -B(x_0 - \dot{x}) - k_2(x_0 - x) \tag{22}$$

By introducing some variables y_1, y_2, y_3, y_4 , where $y_1 = x, y_2 = x_0,$

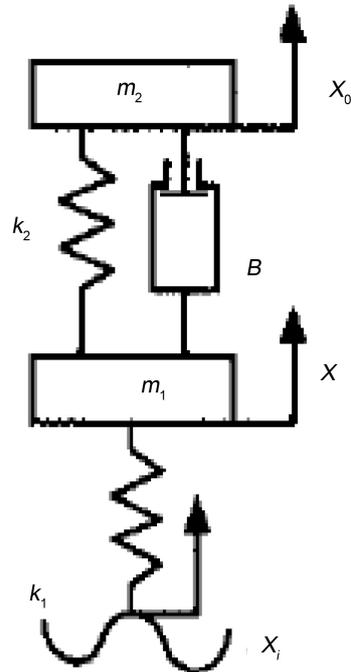


Figure 10. Simplify the system model.

$$y_3 = \dot{x}, \quad y_4 = \dot{x}_0, \quad \dot{y}_1 = y_3, \quad \dot{y}_2 = y_4.$$

$$\ddot{x} = \dot{y}_3 = -\frac{k_1+k_2}{m_1}y_1 + \frac{k_2}{m_1}y_2 - \frac{B}{m_1}y_3 + \frac{B}{m_1}y_4 + \frac{k_1}{m_1}x_1 \tag{23}$$

$$\ddot{x}_0 = \dot{y}_4 = \frac{k_2}{m_2}y_1 - \frac{k_2}{m_2}y_2 + \frac{B}{m_2}y_3 - \frac{B}{m_2}y_4 \tag{24}$$

$$\begin{bmatrix} \dot{y}_1 \\ \dot{y}_2 \\ \dot{y}_3 \\ \dot{y}_4 \end{bmatrix} = \begin{bmatrix} 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 1 \\ -\frac{k_1+k_2}{m_1} & \frac{k_2}{m_1} & \frac{B}{m_1} & \frac{B}{m_1} \\ \frac{k_2}{m_2} & -\frac{k_2}{m_2} & \frac{B}{m_2} & -\frac{B}{m_2} \end{bmatrix} \begin{bmatrix} y_1 \\ y_2 \\ y_3 \\ y_4 \end{bmatrix} + \begin{bmatrix} 0 \\ 0 \\ \frac{k_1}{m_1} \\ 0 \end{bmatrix} x_1 \tag{25}$$

where, $m_1 = 320, m_2 = 2500, k_1 = 100000, k_2 = 10000, B = 140000$.

Build the above constructed model as shown in Figure 11.

8. Conclusion

With MATLAB/Simulink software, you can easily and easily build a variety of automotive dynamics models, reduce the time in the programming, and quickly build a variety of automotive system dynamics models; the same can reduce the amount of work. In order to study the various performance indicators of the car, it provides a strong simulation result to ensure that arch, which can effectively reduce the car's R & D cycle, and improve the efficiency of automotive research. Through the simulation of the acceleration and power of the vehicle body, it can evaluate the structural performance of the automobile, which is of great

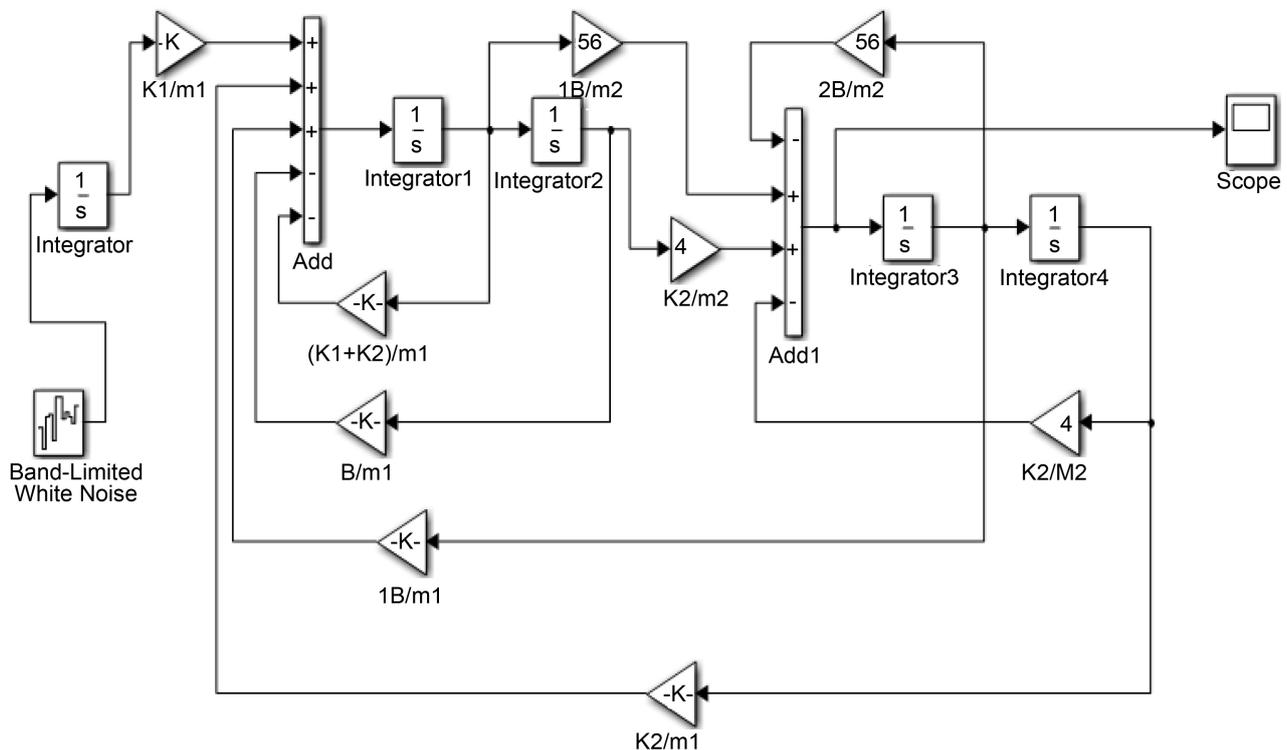


Figure 11. Wiring diagram.

significance to the optimization of the vehicle's ride comfort, handling comfort, safety and comfort.

References

- [1] Yu, Z.S. (2009) *Automotive Theory*. Mechanical Industry Press, Beijing.
- [2] Zhang, L. *Analysis and Optimization of Handling Stability and Smoothness of Electric Wheels*. Wuhan University of Technology, Wuhan.
- [3] Peng, Z.Z. (2012) *Modeling, Simulation and Ride Comfort of Semi-Active Suspension of Electric Vehicle*. Wuhan Polytechnic University, Wuhan.
- [4] Pan, L. (2004) *Modeling and Simulation of Vehicle Ride Comfort Based on Three Directional Vibration of Human Chair System*. Zhejiang Industry University, Hangzhou.
- [5] Xue, D.Y. and Chen, Y.Q. (2002) *System Simulation Technology and Application Based on MATLAB/Simulink*. Tsinghua University Press, Beijing.
- [6] Liu, W.J. and Song, C.X. *Impact on the Vehicle Ride Comfort Caused by the Changes of Electric Vehicle Unsprung Mass*.
- [7] Zhang, Y.L. and Zhong, Y.F. (2004) Time Domain Model of Road Undulation Excitation to Vehicles. *Transactions of the Chinese Society of Agricultural Machinery*, **35**, 9-12.
- [8] Tang, D. and Luo, Y.T. (2013) *Dynamics and Structure Optimization of the In-Wheel Motor System with Rubber Bushing*. South China University of Technology, Guangdong.
- [9] Wu, B.F. (2006) *Integrated Control and Optimization Design of Vehicle Ass and EPS Based on Model Predictive Control*. Hefei University of Technology, Hefei.

-
- [10] Luo, J.W. (2006) Research of Switched Reluctance Drive System for In-Wheel Driving Electric Vehicle. Huazhong University of Science and Technology, Wuhan.
- [11] Xu, X.J. (2007) Study of Vehicle Dynamics Modeling for Driving Simulator. Wuhan University of Technology, Wuhan.
- [12] Wang, Q.N. and Zhang, H.H. (2007) Torque Co-Ordinated Control of Four-Wheel Independent Drive Electric Vehicles in Cornering. *Journal of Jilin University*.
- [13] Liu, W., Shi, W.K. and Fang, D.G. (2011) A Research on Suspension System Control for Vehicle Handling and Stability. *Automotive Engineering*.
- [14] Ding, Y.K. (2009) Study on Vehicle Chassis Control Integration. Wuhan University of Technology, Wuhan.
- [15] Hou, Y. (2012) Study on Possible Battery Energy Storage Configuration of New Energy Applications in ShangHai.
- [16] Ma, Q.Z. (2010) Study on Regenerative Brake Control Alogrithm Based on Braking Intention Identification. Jilin University, Changchun.
- [17] Tong, W. and Hou, Z.C. (2013) Research on the Vertical Performance and Motor Vibration of Electrical Vehicle Driven by In-wheel Motors. Tsinghua University, Beijing.
- [18] Zhang, D.M. (2007) Control and Simulation Analysis of Electric Vehicle with Hub-Motors. Liaoning Institute of Technology, Jinzhou.
- [19] Guo, D.Y., Wang, F. and Zhao, H.M. (2016) Based on MATLAB/Simulink Simulation of Vehicle Suspension System. *Mechanical Engineering Automation*.
- [20] Liu, J. (2011) Based on MATLAB/Simulink Vehicle Suspension Modeling and Simulation. *Highway and Qiyun*.