Research Optimization of Suspension Properties Based on Dynamic Model of Vehicle

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Abstract. Dynamic model of 4X4 vehicle was researched in this article, the mass,damping and stiffness matrices were deduced through lagrange theorem. On the other hand, model transfer function was got by fourier transform of differential equations, after that, random excitation coherence function of 4 wheels and the power spectrum matrix were researched for the model simulation. The performance evaluation system of vehicle suspension system was derived, and the change regulation of vibration acceleration of vehicle body with damping ratio was given. It provides theoretical basis for design of components and optimization of suspension properties.

Introduction

In order to analysis dynamic characteristics of vehicle, it must establishing dynamics model.Vehicle vibration systems are very complicated, and dynamic response results are influenced by numerous factors.In this article, the dynamics model is built up through parameters 4×4 off-road vehicle,which has seven freedom degrees. After that,simulation and optimization of suspension properties are completed, it is important to improve ride comfort and handling stability of the vehicle.

Dynamic model of the vehicle

Physical model

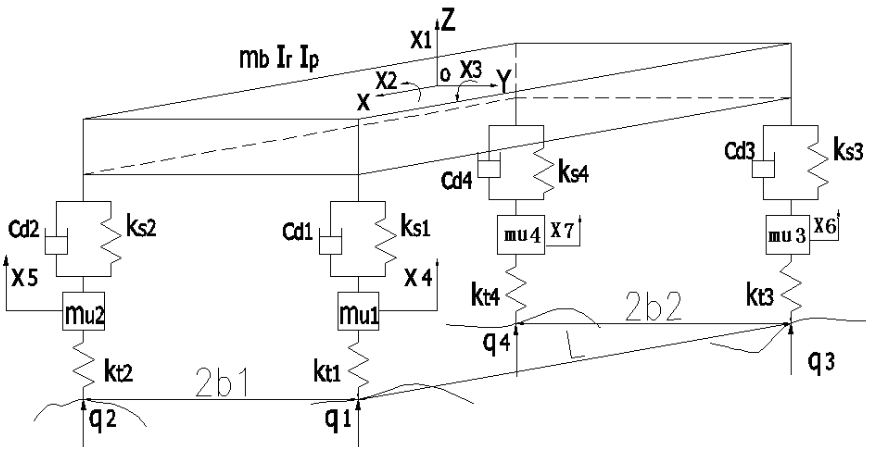

Fig.1 physical model of the vehicle

4×4 off-road vehicle dynamic model shown in fig.1,which has seven freedom degrees, point O is mass center of body, and the vehicle travels along the X direction.

Model deduction

Dynamic differential equation is :[1-2]

$$M\ddot{X}+C\dot{X}+KX=F=K_qQ \tag{1}$$

In which, $X = \begin{bmatrix} x_1 & x_2 & x_3 & x_4 & x_5 & x_6 & x_7 \end{bmatrix}^T$

$Q = \begin{bmatrix} q_1 & q_2 & q_3 & q_4 \end{bmatrix}^T$

Considering the symmetry of the vehicle:

$$k_{s1} = k_{s2} = k_{fs}; k_{s3} = k_{s4} = k_{rs}; c_{d1} = c_{d2} = c_{fd},$$
$$c_{d3} = c_{d4} = c_{rd}; k_{t1} = k_{t2} = k_{tf}; k_{t3} = k_{t4} = k_{tr}$$

The mass matrix is :

$$M = \begin{bmatrix} m_b & 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & I_r & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & I_p & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & m_{u1} & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & m_{u2} & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & m_{u3} & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & m_{u4} \end{bmatrix} = diag(m_b, I_r, I_p, m_{u1}, m_{u2}, m_{u3}, m_{u4}) \quad (2)$$

The damping matrix is :

$$C = \begin{bmatrix} 2(c_{fd}+c_{rd}) & 0 & -2(L_1 c_{fd} - L_2 c_{rd}) & -c_{fd} & -c_{fd} & -c_{rd} & -c_{rd} \\ 0 & 2(L_{d1}^2 c_{fd} + L_{d2}^2 c_{rd}) & 0 & -L_{d1} c_{fd} & L_{d1} c_{fd} & -L_{d2} c_{rd} & L_{d2} c_{rd} \\ -2(L_1 c_{fd} - L_2 c_{rd}) & 0 & 2(L_1^2 c_{fd} + L_2^2 c_{rd}) & L_1 c_{fd} & L_1 c_{fd} & -L_2 c_{rd} & -L_2 c_{rd} \\ -c_{fd} & -L_{d1} c_{fd} & L_1 c_{fd} & c_{fd} & 0 & 0 & 0 \\ -c_{fd} & L_{d1} c_{fd} & L_1 c_{fd} & 0 & c_{fd} & 0 & 0 \\ -c_{rd} & -c_{rd} L_{d2} & -L_2 c_{rd} & 0 & 0 & c_{rd} & 0 \\ -c_{rd} & L_{d2} c_{rd} & -L_2 c_{rd} & 0 & 0 & 0 & c_{rd} \end{bmatrix} \quad (3)$$

The stiffness matrix is:

$$K = \begin{bmatrix} 2(k_{fs}+k_{rs}) & 0 & -2(L_1 k_{fs} - L_2 k_{rs}) & -k_{fs} & -k_{fs} & -k_{rs} & -k_{rs} \\ 0 & 2(L_{s1}^2 k_{fs} + L_{s2}^2 k_{rs}) & 0 & -L_{s1} k_{fs} & L_{s1} k_{fs} & -L_{s2} k_{rs} & L_{s2} k_{rs} \\ -2(L_1 k_{fs} - L_2 k_{rs}) & 0 & 2(L_1^2 k_{fs} + L_2^2 k_{rs}) & L_1 k_{fs} & L_1 k_{fs} & -L_2 k_{rs} & -L_2 k_{rs} \\ -k_{fs} & -L_{s1} k_{fs} & L_1 k_{fs} & (k_{fs}+k_{tf}) & 0 & 0 & 0 \\ -k_{fs} & L_{s1} k_{fs} & L_1 k_{fs} & 0 & (k_{fs}+k_{tf}) & 0 & 0 \\ -k_{rs} & -L_{s2} k_{rs} & -L_2 k_{rs} & 0 & 0 & (k_{rs}+k_{rt}) & 0 \\ -k_{rs} & L_{s2} k_{rs} & -L_2 k_{rs} & 0 & 0 & 0 & (k_{rs}+k_{rt}) \end{bmatrix} \quad (4)$$

Force matrix is:

$$F = \begin{bmatrix} 0 \\ 0 \\ 0 \\ k_{tf} q_1 \\ k_{tf} q_2 \\ k_{rt} q_3 \\ k_{rt} q_4 \end{bmatrix} = \begin{bmatrix} 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 \\ k_{tf} & 0 & 0 & 0 \\ 0 & k_{tf} & 0 & 0 \\ 0 & 0 & k_{rt} & 0 \\ 0 & 0 & 0 & k_{rt} \end{bmatrix} \begin{bmatrix} q_1 \\ q_2 \\ q_3 \\ q_4 \end{bmatrix} = K_q Q \quad (5)$$

Fourier transform to the formula (1):[3-4]

$$-w^2 M X(w) + jwCX(w) + KX(w) = F(w) = K_q Q(w) \quad (6)$$

Random excitation of road

Pavement power spectral density is expressed:[5-6]

$$G_q(n) = G_q(n_0)(\frac{n}{n_0})^{-W} \qquad n_l < n < n_u \tag{7}$$

In which, n is time frequency, n_u and n_l are upper limit, lower limit of spatial frequency of road spectrum, n_0=0.1 m^{-1} is reference space frequency, W is frequency index.

The road spatial power spectrum density could be change to time power spectrum density:

$$G_q(w) = G_q(f) = \frac{1}{u} G_q(n) = (2\pi)^2 n_0^2 G_q(n_0) \frac{u}{w^2} \tag{8}$$

And power spectrum matrix of 4 wheels is:

$$\left[G_q(n)\right] = \begin{bmatrix} 1 & coh(n) & e^{-j2\pi nl} & coh(n)e^{-j2\pi nl} \\ coh(n) & 1 & coh(n)e^{-j2\pi nl} & e^{-j2\pi nl} \\ e^{j2\pi nl} & coh(n)e^{j2\pi nl} & 1 & coh(n) \\ coh(n)e^{j2\pi nl} & e^{j2\pi nl} & coh(n) & 1 \end{bmatrix} G_q(n) \tag{9}$$

In which, $coh_{xy}(n)$ is the coherence coefficient for the wheels, it can be expressed as:

$$coh_{xy}^2(n) = \begin{cases} (a-bn)^r & 0 \le |n| \le 0.1 \\ (a-0.1b)^r & 0.1 < |n| \le 2 \\ 0 & 2 < |n| \end{cases} \tag{10}$$

And $a = 1$； $b = (1-0.1^{1/r})/0.1$； $r = B/0.25$， B is the tread.

The evaluation index system of suspension performance

The suspension determined the vehicle ride comfort and stability, which performance can be evaluated through the following three basic parameters:

(1) The vibration acceleration response

It is expressed through root mean square of vertical body acceleration:

$$\sigma_{\ddot{x}_1} = \sqrt{\frac{1}{2\pi}\int_0^{+\infty} 2S_{\ddot{x}_1}(w)dw} = \sqrt{\frac{1}{2\pi}\int_0^{+\infty} 2w^4 G_{11} dw} \tag{11}$$

(2) Suspension stroke

Suspension stroke parameter is defined as the wheels and the body displacement difference RMS value, it can be used to describe the degree of relative displacement of the static state, have great influence on handling stability of vehicle.

The four suspension stroke is defined as:

$$\begin{aligned} f_{d1} &= x_1 + L_{s1}x_2 - L_1x_3 - x_4 \\ f_{d2} &= x_1 - L_{s1}x_2 - L_1x_3 - x_5 \\ f_{d3} &= x_1 + L_{s2}x_2 + L_2x_3 - x_6 \\ f_{d4} &= x_1 - L_{s2}x_2 + L_2x_3 - x_7 \end{aligned} \tag{12}$$

And the four suspension stroke RMS for:

$$\sigma_{f_{d1}} = \sqrt{\sigma_{x1}^2 + b_1^2\sigma_{x2}^2 + L_1^2\sigma_{x3}^2 + \sigma_{x4}^2} = \sqrt{\frac{1}{2\pi}\int_0^{+\infty} 2(G_{11} + b_1^2 G_{22} + L_1^2 G_{33} + G_{44})dw}$$

$$\sigma_{f_{d2}} = \sqrt{\sigma_{x1}^2 + b_1^2\sigma_{x2}^2 + L_1^2\sigma_{x3}^2 + \sigma_{x5}^2} = \sqrt{\frac{1}{2\pi}\int_0^{+\infty} 2(G_{11} + b_1^2 G_{22} + L_1^2 G_{33} + G_{55})dw}$$

$$\sigma_{f_{d3}}=\sqrt{\sigma_{x1}^{2}+b_{2}^{2}\sigma_{x2}^{2}+L_{2}^{2}\sigma_{x3}^{2}+\sigma_{x6}^{2}+b_{2}^{2}\sigma_{x7}^{2}}=\sqrt{\frac{1}{2\pi}\int_{0}^{+\infty}2(G_{11}+b_{2}^{2}G_{22}+L_{2}^{2}G_{33}+G_{66}+b_{2}^{2}G_{77})dw}$$

$$\sigma_{f_{d4}}=\sqrt{\sigma_{x1}^{2}+b_{2}^{2}\sigma_{x2}^{2}+L_{2}^{2}\sigma_{x3}^{2}+\sigma_{x6}^{2}+b_{2}^{2}\sigma_{x7}^{2}}=\sqrt{\frac{1}{2\pi}\int_{0}^{+\infty}2(G_{11}+b_{2}^{2}G_{22}+L_{2}^{2}G_{33}+G_{66}+b_{2}^{2}G_{77})dw} \quad (13)$$

Vehicle dynamics simulation

The vehicle dynamics model is calculated through matlab programm. Road and vibration acceleration simulation results are as follows. The vertical vibration acceleration responses of the vehicle body changing with different suspension damping characteristics lay the foundation for establishing control strategy of semi-active suspension.

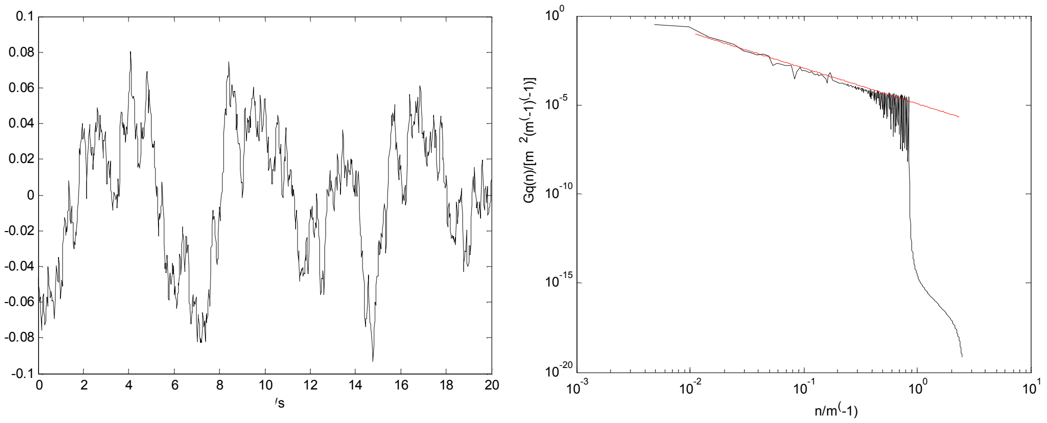

Fig.2 random excitation of the road

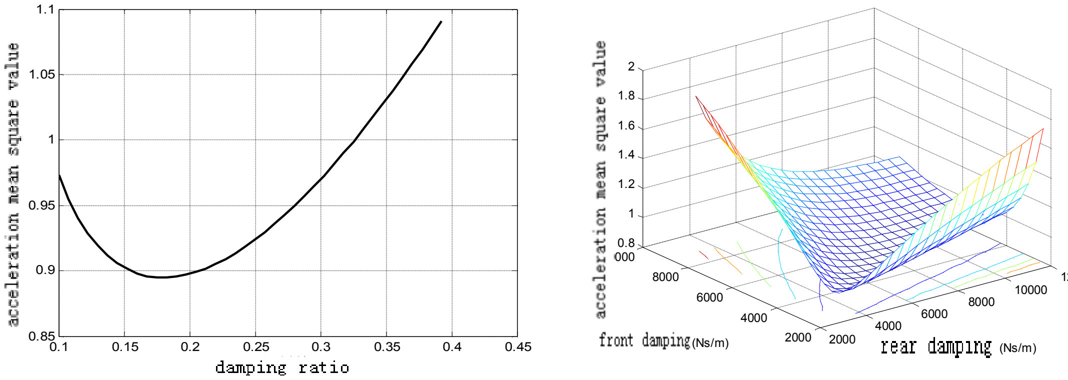

Fig.3 vertical vibration acceleration response of different damping

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