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Dynamical Performance Research on a Novel MR Mount

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Abstract. The lumped parameter model of this novel MR Mount was established and nonlinear mathematical state equations of bond graph model for the MR mount is developed. Finally, its dynamical performance is analyzed and predicted. The results show that the dynamic stiffness (400N/mm) in the frequency range 0-20Hz and the dynamic stiffness (500-700n/mm) in the frequency range 0-25Hz are effective for the isolation of road and powertrain source excitation.

1. Introduction

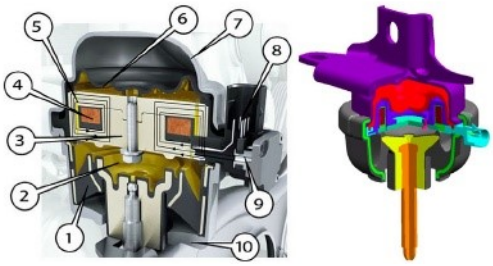
Along with the development of automobile technology, the NVH problem of automobile is getting more and more attention from enterprises and customers. NVH performance has become an important index of automobile comfort. The optimization and design of power source is an effective way to solve engine dynamic noise. At present, rubber suspension and hydraulic suspension are widely used. With the development of mount technology, magneto-rheological mount has been applied in advanced passenger vehicles in recent years [1-3]. However, the technology of magneto-rheological mount development and optimization parameter design is not very mature at present, so it is very necessary to study the electromagnetic coupling and dynamic characteristics of magneto-rheological mount.

As a vibration isolation system of power source, magneto-rheological mount plays an important role in reducing vibration and noise. In order to control the low speed fluctuation of power source, the vibration isolation system often needs to be designed. For controlling the fluctuation of low speed, the large damping characteristics of the mount system have significant effects, while for vibration excitation caused by high frequency unbalanced force, it is suggested that the suspension system has small damping characteristics. This paper presents a decoupled membrane magneto-rheological suspension system. The mount structure, state equation, performance parameters and parameter model simulation are studied [4-7].

2. Basic structure

The damping properties of magneto-rheological (M r) mount are variable, and the damping is correlated with the magnetic flux of the external magnetic field. When excited by external magnetic field, the suspended internal liquid changes from liquid to liquid and solid coupling state, presenting a controllable yield strength [8]. Considering the magneto-rheological characteristics of the mount system, the damped adjustable magneto-rheological mount was designed [9, 10]. The magneto-

rheological mount structure designed in this paper is shown in figure 1. The magneto-rheological mount is mainly composed of rubber main spring, liquid chamber, extrusion disc, coil, liquid channel, magneto-rheological liquid, sensor, decoupled film and other structures. Under low frequency excitation, the suspended upper liquid volume changes, the magneto-rheological fluid flows through the variable damping channel, and changes the current size according to the vibration of the suspension active terminal, so as to realize the change of magnetic flux. Adjusting the flow resistance of the liquid channel can adjust the damping characteristics of the suspension. When high frequency excitation occurs, fluid flow through the damping channel increases the liquid energy loss [11], thus controlling the high frequency vibration characteristics.



1- rubber main spring 2- liquid chamber 3- magnetic core 4- coil 5- liquid channel 6- Rubber soles mold 7- mounting plate 8- guide line 9- sensor 10- body parts

Fig. 1 Schematic diagram of magnetic rheological mount structure

3. Magneto-rheological model

3.1. Parameter model

The suspended parameter model is shown in figure 2. The parameters Kr constitute the stiffness parameters of the spring system, K and B represent the stiffness and damping system of the spring respectively. Parameter C is the volume stiffness of the suspension model, parameter A is variable area, working mode and principle is to control the flow of liquid by adjusting the aperture of the damping channel. The flow variation of the decoupled film and the throttling disc controls the area of the upper and lower Chambers respectively, thus controlling the flow rate to change the acting force. The control quantity in the mount model includes input excitation X (t) and suspension transmission force F (t).

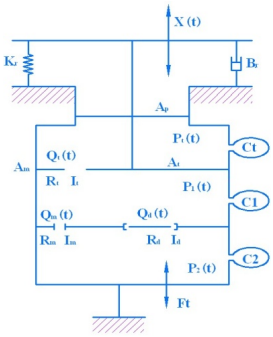


Fig. 2 mount parameter model

3.2. Bond graph model

The bond graph structure was proposed by professor Paynter H in 1950, and then Karnopp d.c. and Rosenberg R.C developed its theory based on the theory of statistical energy. It is a dynamic analysis method of complex subjects which can be quantified and modularized. Due to the intuitionistic nature

of this method, it has been widely used in the design of magneto-rheological (M r) mount. According to the bond graph structure, the bond graph model of magneto-rheological (M r) mount was obtained, as shown in figure 3.

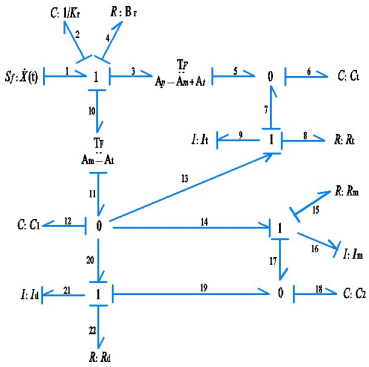


Fig. 3 Bond graph model

According to the bond graph structure theory, the nonlinearity of the volumetric stiffness of the liquid chamber can be simplified. In this paper, we consider only the nonlinearity of the magneto-

Rheological fluid yield limit and magnetic flux intensity. Where, the liquid resistance of the variable flux of magneto-rheological rheological is expressed as $R_m = R_{m0} + R_\tau(\tau_y(B))$. The dynamic equation of magneto-rheological suspension is derived as follows:

$$F(t) = K_r X + B_r \dot{X} + (A_p - A_m + A_t)P_t + (A_m - A_t)P_1 \tag{1}$$

$$C_t \dot{P}_t = Q_t - (A_m - A_p - A_t)\dot{X} \tag{2}$$

$$C_1 \dot{P}_1 = (A_m - A_t)\dot{X} - Q_t - Q_m - Q_d \tag{3}$$

$$C_2 \dot{P}_2 = Q_m + Q_d \tag{4}$$

$$P_1 - P_t = I_t \dot{Q}_t + R_t Q_t \tag{5}$$

$$P_1 - P_2 = I_m \dot{Q}_m + R_m Q_m \tag{6}$$

$$P_1 - P_2 = I_d \dot{Q}_d + R_d Q_d \tag{7}$$

The above mentioned kinds of Laplace transforms are derived and the magneto-rheological suspension stiffness is:

$$\begin{aligned} K_d^*(s) &= L(F(t))/L(X(t)) \\ &= K_r + B_r s + (A_p - A_m + A_t) f_1(s)/C_t \\ &+ (A_m - A_t) f_2(s)/C_1 \end{aligned} \tag{8}$$

Among them:

$$f_1(s)=\frac{a_0+a_1s+a_2s^2+a_3s^3+a_4s^4+a_5s^5}{b_0+b_1s+b_2s^2+b_3s^3+b_4s^4+b_5s^5}$$
$$f_2(s)=\frac{c_0+c_1s+c_2s^2+c_3s^3+c_4s^4+c_5s^5}{d_0+d_1s+d_2s^2+d_3s^3+d_4s^4+d_5s^5} \tag{9}$$

The correlation coefficient a_i,b_i,$c_i$and d_i(i=0,1,2,3,4,5)can be represented by set parameters. make $s=j\omega$,

$$K_d^*(j\omega)=K_1+jK_2 \tag{10}$$

Where: K1 is the energy storage stiffness of the system; K2 is the system loss stiffness.

4. Fluid resistance performance

In order to obtain the nonlinear liquid resistance R_m of magneto-rheological mount, this paper USES the electro-hydraulic coupling law of finite element method to identify. Under different current excitation, the magneto-rheological magnetic induction intensity and magnetic flux were calculated by finite element method, and then identified by liquid resistance formula. The dimension parameters of magnetic core structure of magneto-rheological mount are marked as shown in figure 4. The calculation formula of nonlinear liquid resistance R_m can be deduced as follows:

$$R_m=\frac{6\eta(2L+b)}{\pi R_1(R_2-R_1)^3}+\frac{6\tau_y(B)L}{(R_2-R_1)|Q_m|} \tag{11}$$

Where, Q_m represents the liquid flow rate of magneto-rheological mount. ρ is magneto-rheological mount liquid density; τ_y is the yield stress of magneto-rheological fluid.

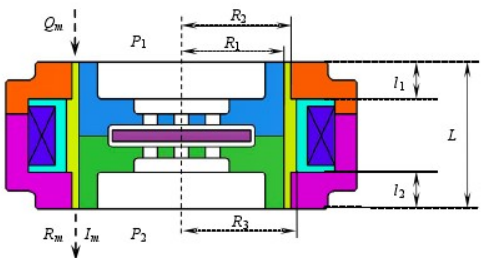


Fig. 4 Structure of the mount magnetic core component

The solid-liquid coupling model of suspended magnetic core component and magneto-rheological fluid was established by using finite element software. The magnetic core module uses 2-dimensional units, and the magneto-rheological liquid is simulated by one-dimensional units. The current density is loaded at the position of the magnetic coil of the model. Under the action of magnetic field, the magnetic flux and induction density of the magneto-rheological liquid can be obtained. When the allowable value of excitation current is 2.0A, the magnetic field intensity distribution of part of the magnetic core is shown in figure 5.

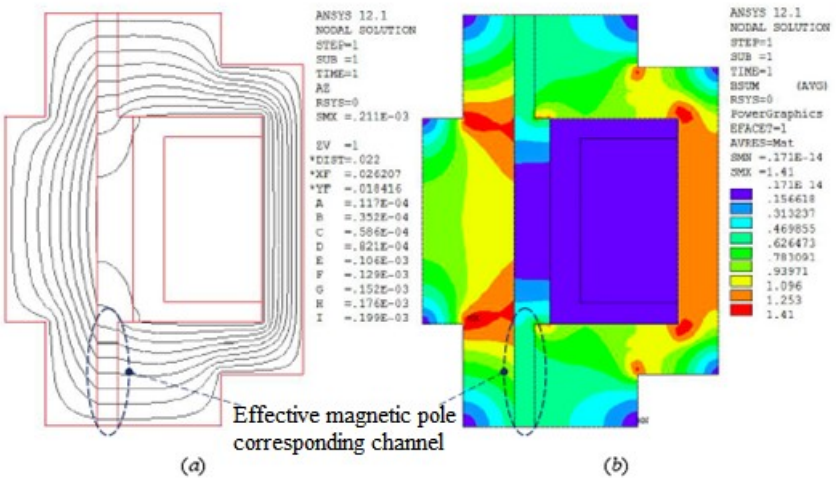


Fig. 5 magnetic field intensity distribution under excitation current of 2.0A

The remaining parameters in the magneto-rheological parameter model can be identified and optimized according to the derived formula (8, 9, 10) and the finite element method. The literature [4, 5] has an introduction. The above method is used to optimize the parameters of magneto-

Rheological (M r) suspension. The initial parameters without parameter identification optimization and the parameter pairs after optimization are shown in table 1.

Table 1. Mount related parameters

Related parameters	initial state	after parameter optimization
$K(N/mm)$	70	100
$C(m^5/N)$	2.1×10^6	3.1×10^6
$C_1(m^5/N)$	0.5×10^4	1.1×10^4
$C_2(m^5/N)$	2736	3118
$R(N/sm^5)$	967	1156
$I(kg/m^4)$	*5786*	6179

5. Stiffness and damping properties

5.1. Dynamic stiffness of road excitation isolation

Because road excitation is low-frequency, the analysis frequency of magneto-rheological mount is 0-50hz low-frequency, and the excitation amplitude is± 0.5mm. The dynamic stiffness of the mount in the initial state is shown in fig. 8. According to the curve results, the dynamic stiffness of the mount in the initial state is 600N/mm at 0-20Hz. The dynamic stiffness of frequency 20-50Hz is up to 1000N/mm, and the dynamic stiffness value is large, which has a poor effect on the excitation isolation caused by inhibiting the ground.

According to the curve results, the dynamic stiffness after parameter optimization is 400N/mm in the range of 0-20Hz, and the dynamic stiffness of frequency 20-50Hz is up to 600N/mm. The dynamic stiffness value is lower than the initial state, and the effect of excitation isolation on the ground is better. The dynamic stiffness distribution range is reasonable. The dynamic stiffness in the vertical direction of low frequency band mainly reflects small stiffness and large damping characteristics. This stiffness can effectively isolate the vibration of the ground through the tire and the suspension system, and after the suspension of the powertrain after the suspension of secondary vibration.

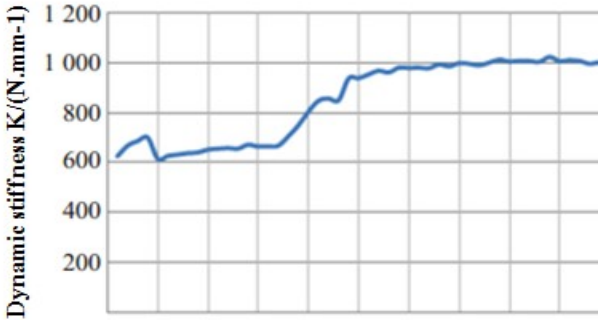


Fig. 6 vertical dynamic stiffness curve of initial state mount

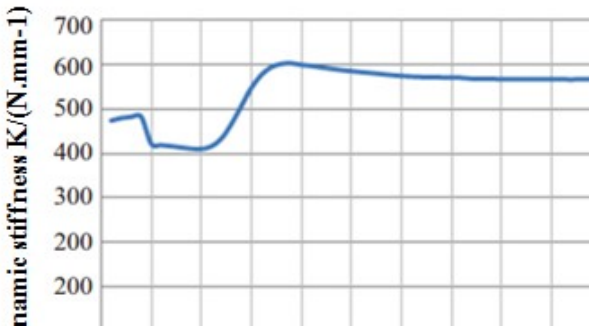


Fig. 7 dynamic stiffness curve of the vertical suspension after optimization

5.2. Damping performance

Damping property is an important index to evaluate the isolation effect of suspension system. The test curve of the suspension damping Angle of the initial state parameter is shown in figure 8, and the damping Angle shows a damping peak at 20Hz. Since the initial parameter of damping peak frequency of 20Hz is greater than the vertical rigid-body mode of power source, the effect of isolating power source is weak.

After parameter identification and parameter optimization, the test curve of overhanging damping Angle is shown in figure. 9. According to the design criteria, the damping peak frequency of the suspension vibration isolation of the power source is close to or overlapped with the vertical rigid-body mode of the power source. Therefore, the damping peak frequency of the magneto-rheological suspension structure after parameter optimization is 13Hz close to the rigid-body mode of the vertical direction of the power source (11Hz-15Hz), and the effect of isolating the power source is significant.

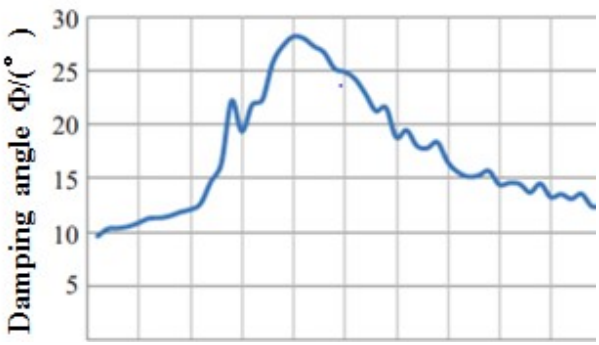


Fig. 8 Initial state vertical damping Angle curve

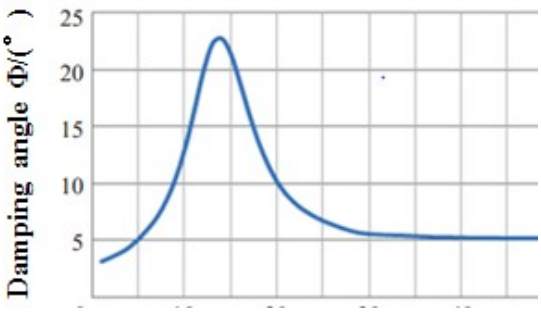


Fig. 9 Vertical damping Angle curve after parameter optimization

6. Conclusion

In this paper, the dynamic equation of the magneto-rheological system is deduced based on the theory of bond graph structure. Combined with the dynamic equation and the finite element electromagnetic coupling simulation results, the optimal identification of the suspension parameters is carried out. The following conclusions were obtained: (1) In terms of reducing road excitation: the dynamic stiffness after parameter identification reaches 400N/mm at 0-20Hz. The dynamic stiffness value is lower than the initial state, and the effect of excitation isolation on the ground is better.(2) In terms of reducing vibration peak: the damping peak frequency after parameter optimization is 13Hz, close to the vertical rigid-body mode of power source (11Hz-15Hz), and the effect of isolating power source is significant. It can reduce the vibration peak which is transmitted from the rigid body mode of the power source as the main transmission path.

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