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Research on automobile power plant active mount based on model reference adaptive fuzzy control system

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ABSTRACT

A 3-point mount system is designed which aiming at one vehicle model in our company, and some correlative parameters are optimized. For improving decreasing vibration's capability when the engine is in high-frequency vibration state, we designed an active mount system and proved the scheme's feasibility by emulation of software. This research will help to opening up the future active control mount system.

KEYWORDS

Mount system; High-frequency vibration; Vibration decreasing; Active control.

INTRODUCTION

At present, the power assembly mounting system based on the mode of action can be divided into passive suspension, semi-active suspension and active control mounts. Passive suspension (rubber and hydraulic mount) has become increasingly difficult to meet the requirements for ride comfort[1]; Study on semi- active suspension systems in many of the engine, but a major change in the performance of the system in the low frequency range and high frequency vibration isolation effect is not ideal[2,3]; The active control suspension has the characteristic of low frequency, high frequency of hard and soft and it can better isolate the Power assembly vibration transmission in various conditions and it is the development trend of the Powertrain vibration isolation technique (such as the use of active control of fuzzy control method[4]).

Total suspension system is a complex nonlinear time-varying system; the traditional control method can not meet the need of complex conditions; and the model reference adaptive fuzzy control method has good control effect and robustness to nonlinear system and it has good control stability under external excitation.

This paper first determines vehicle suspension system parameters. A fuzzy controller based on piezoelectric actuators for active hydraulic control components of the suspension has been designed on this basis and the damping performance of the computer simulation and comparison. The results show that the suspension has good ability of active vibration control.

DETERMING THE PARAMETERS OF SUSPENSION SYSTEM

The model selection of three type suspension system, namely the left, right, rear suspension (Of which: Left, rear suspension are the rubber mounts and right is the hydraulic mount). System layout is as follows:

Figure 1 : Schematic diagram of the suspension system layout scheme

According to the given Power assembly Parameters (see TABLE 1), by using Powertrain-TRA-Computer program calculate the position of Powertrain TRA in the space and according to the result of the calculation, the ultimately determine suspension elastic center position is as shown in TABLE 2 (the reference vehicle coordinate system):

TABLE 1 : A type of power assembly part parameters

Powertrain centroid(the vehicle coordinate system)	Power assembly mounting Angle(DEG)	The power assembly quality (Kg)	Rated power (KW)	The maximum torque (N·m)	Idle speed (r/min)	Maximum speed (r/min)
X:-213.041 Y:28.000 Z:175.088	α: 0.000 β:2.500 γ:0.000	163.7	89	146	805±50	6021

TABLE 2 : Each suspension elastic center position

Mount	Location (Vehicle Cord.)		
	X(mm)	Y(mm)	Z(mm)
LHM	-117.5	-388.68	27.431
RHM	-216	471.059	361.24
Rear rod	26.083	-65.616	-90.202
Rear sub	153.588	-65	-98.813

After optimization of the suspension stiffness scheme, natural frequency and energy decoupling rate distribution respectively see TABLE 3, TABLE 4:

TABLE 3 : After optimization of the stiffness scheme

MOUNT	Static Stiff. (Vehicle Cord)		
	X(N/mm)	Y(N/mm)	Z(N/mm)
LHM	150	45	160
RHM	150	45	160
REAR_ROD	2000	350	2000
REAR_SUB	180	250	60

TABLE 4 : Each order natural frequency and energy decoupling rate

MODE	Freq.(Hz)	X	Y	Z	RXX	RYY	RZZ
LATERAL	4.50614	0	99.58	0.02	0.31	0.03	0.01
BOUNCE	8.58334	0.15	0.03	99.24	0.5	0.04	0
FORE/AFT	10.0303	95.8	0	0.14	0.01	0.02	3.83
PITCH	12.2582	0.02	0.04	0.01	4.04	83.76	1.49
ROLL	15.8984	0.03	0.25	0.57	94.29	1.32	0.33
YAW	17.8845	3.07	0	0	0.59	2.02	86.4

For transverse engine cars, the decoupling rate of "Bounce-Z" and "Pitch-RYY" has the maximum effect on the vibration isolation. We can see from TABLE 3, in addition to the decoupling rate of "Pitch-RYY and "Yaw-RZZ" are a bit poor with the rate of 83.76% and 86.4%; the other direction decoupling rate is around 95% and it has better decoupling effect. But after the actual loading verification, the suspension scheme has a lower effect of vibration in high frequency vibration than the expected goal.

Because the vibration isolation effect is not ideal while the passive suspension system is in the high frequency vibration; using piezoelectric actuator and passive (hydraulic) suspension composed of active suspension system to improve the vibration isolation performance of suspension system.

ACTIVE SUSPENSION SYSTEM STRUCTURE AND SYSTEM MODELING

Active suspension system structure

In view of the passive suspension is difficult to meet the multiple needs of the isolation; semi-active control hydraulic mount has a general effect in high frequency and small amplitude vibration isolation and due to the vibration of powertrain engine side in the system of maximum, comprehensive cost perspective, the use of active suspension replacement engine side (right side) of the hydraulic mount. Among them, we choose the piezoelectric actuator for high speed response ability as the better actuator (Figure 2); In order to overcome the disadvantages of the piezoelectric actuator of displacement is very small (maximum output of less than 50μm). The use of hydraulic displacement in the output displacement of piezoelectric pile end amplification mechanism, which comprises a closed hydraulic room and a large, small piston. When the piezoelectric stack of longitudinal deformation, promoting the movement of the piston; the liquid driving a small movement of the piston while the small piston connected with the vehicle body so that the deformation and displacement of such piezoelectric stack according to size piston area ratio is enlarged in order to meet the requirements of the output displacement[5]. Piezoelectric active control structure and mechanical type suspension model as shown in Figure 3:

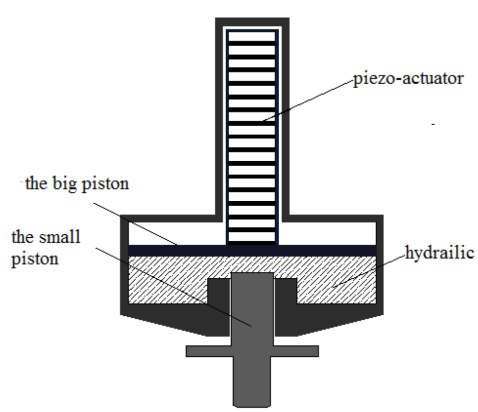

Figure 2 : The structure of the piezoelectric actuator mount

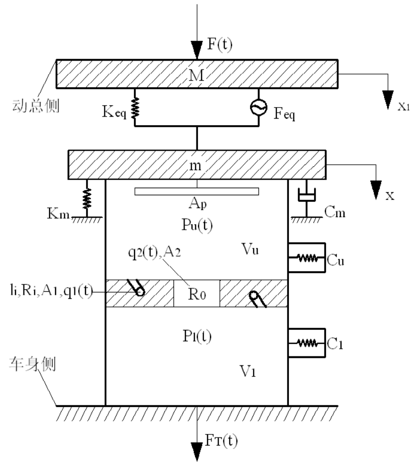

Figure 3 : The mechanical model of the piezoelectric actuator mount

The definitions of the parameters in Figure 3 in the model are as follows:

$F(t)$, $F_T(t)$: Respectively for the input force of active suspension and suspension force is transferred to the body;

M : Power active suspensions bear the assembly quality;

K_{eq}, F_{eq}: Respectively for the dynamic of piezoelectric actuator for equivalent stiffness and its produce;

m: Piezoelectric actuator is equivalent quality

K_m, C_m : They are the main rubber spring in-phase dynamic stiffness and equivalent viscous damping coefficient;

A_p : It is the main rubber spring equivalent piston area;

$P_u(t)$, $P_l(t)$: Respectively for a moment under the pressure and fluid chamber;

V_u, V_l : Respectively for a moment, a lower liquid chamber volume expansion volume;

I_i, R_i : Respectively for the fluid inertia effect and fluid resistance effect of the liquid in the flow inertia channel shown;

A_1, A_2 : Respectively for the inertia channel and orifice cross sectional area;

$q_1(t)$, $q_2(t)$: Respectively for a moment (orifice opening) by the fluid flow on the cavity through the inertia channel and orifice into the inferior vena cava;

C_u, C_l : Respectively for the upper and the lower liquid chamber volume stiffness;

$R_{0:}$ Fluid resistance effect of liquid in the flow orifice is shown;

x_1, x: Respectively for the displacement variable a moment of powertrain and rubber spring;

Active suspension system modeling and parameter determination

The working principle of the control system is comparison of output system and input signal generated by the reference model and the actual model; putting the error and its rate of change transferred to the fuzzy control system; by the control system of judgment error and change rate scope and outputting the control signal and ultimately it makes the control object of the error and the change rate of close to zero, such that the system is stable. Active suspension is formed by a passive suspension and a piezoelectric actuator; the principle of vibration isolation is that the actuator effect produced by the reverse forcing on the suspended; so as to offset the power assembly via passive suspension is transferred to the body force and ultimately it achieves the purpose of reducing vibration. Therefore, the piezoelectric actuator power is generated as the control object and it will be transferred to the mechanical model of passive suspension body as the reference model; according to the force of reference model output and the error and the change rate of the power as input signal control system to realize the control.

According to the stress model and state equation is obtained in the input force under the action of $F_{(t)}$ and through the active suspension system transferred to the body of the dynamic force of $F_{T(t)}$[8] for:

$$F_T(t) = K_m X(t) + C_m \dot{X}(t) + m\ddot{X}(t) + \left(A_p - A_1 - A_2\right)\frac{V_u}{C_u} + (A_1 + A_2)\frac{V_1}{C_1} - A_p P_u(0) \quad \textbf{(1)}$$

Actuator output force device:

$$F_a(t) = (M + m)\ddot{X}(t) \quad \textbf{(2)}$$

In Figure 3, there are some important parameters of hydraulic mount and were determined by the experiment and calculation as following:

TABLE 5 : The main parameters of the system list

The parameter name	$M(kg)$	$m(kg)$	$A_p(mm^2)$	$A_1(mm^2)$
The parameter value	79.3	4.2	2500	210
The parameter name	$A_2(mm^2)$	$\rho(Kg/m^3)$	$K_m(N/mm)$	$C_m(N/mm)$
The parameter value	660	1040	150	2.9718

DESIGN METHODS FOR CONTROLLING IMMUNE

The basic principle of immune particle swarm algorithm

Immune particle swarm algorithm is the basic on framework of particle swarm algorithm, the and take the principle of immune in the life sciences into the particle swarm algorithm, on one hand a use of immune memory and self regulatory mechanism to keep the fitness levels of particles at a certain concentration, so as to ensure the diversity of population; On the other hand, introduced the operation of vaccine inoculation were purposely and selectively to guide the evolutionary process of the algorithm, improves the search efficiency of the algorithm[6].

This paper introduces the immune particle swarm optimization algorithm to overcome the particle swarm algorithm for convergence and loss of diversity issues, and this method not only retains the characteristics of particle swarm algorithm is simple and easy, but also overcomes the premature phenomenon in process of optimization, the algorithm of global search ability and higher convergence speed.

Design of PID controller based on immune algorithm

PID algorithm is a classical control theory at present in the process of industrial production applications more, but often because of the complex system model as well as the operator's lack of experience and other reasons, it is difficult to determine the accurately control various signal process and the amount of evaluation index. Using the immune particle swarm algorithm for tuning of PID' three parameters controller can quickly and accurately to the appropriate parameters optimized[7].

Servo press hydraulic system immune particle swarm optimization algorithm based on the control block diagram is shown in Figure 4.

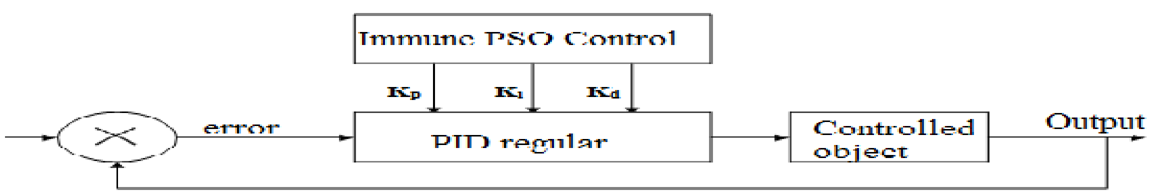

Figure 4 : Immune particle swarm optimization control system

In order to process to obtain the satisfactory excessive dynamic characteristics, the minimum objective function using the IAE performance index of parameter selection as the minimum objection function. In order to prevent the control energy is too large, the square of control input to join in the objective function, selection of optimal index type as parameter selection:

$$J = \int_0^\infty \left(\omega_1 \left| e(t) \right| + \omega_2 u^2(t) \right) dt + \omega_3 t_u \quad \textbf{(3)}$$

In the formula: t_u is rise time; e(t) is System error; u(t) is Immune particle swarm controller output; ω_1, ω_2, ω_3 are all weights.

In order to avoid the overshoot phenomenon, namely the controlled output system is less than the previous output, using penalty function and making the weighted of the absolute value of the difference between before and after the system output as an optimal index, which can effectively solve the problem of optimal index for overshoot:

if ey(t) <0, $$J = \int_0^\infty \left(\omega_1 \left| e(t) \right| + \omega_2 u^2(t) + \omega_4 \left| ey(t) \right| \right) dt + \omega_3 t_u \quad \textbf{(4)}$$

In the formula: : ey(t)=y(t)-y(t-1),y(t) are the output of the controlled object; ω_4 is weight; ω_4 is greater than ω_1.

Because of the bigger and the better fitness of the particle swarm algorithm, so we select the fitness function of $f = \frac{1}{J}$ and J is more small, the greater degree of adaptation.

Parameter optimization process as shown in Figure 5:

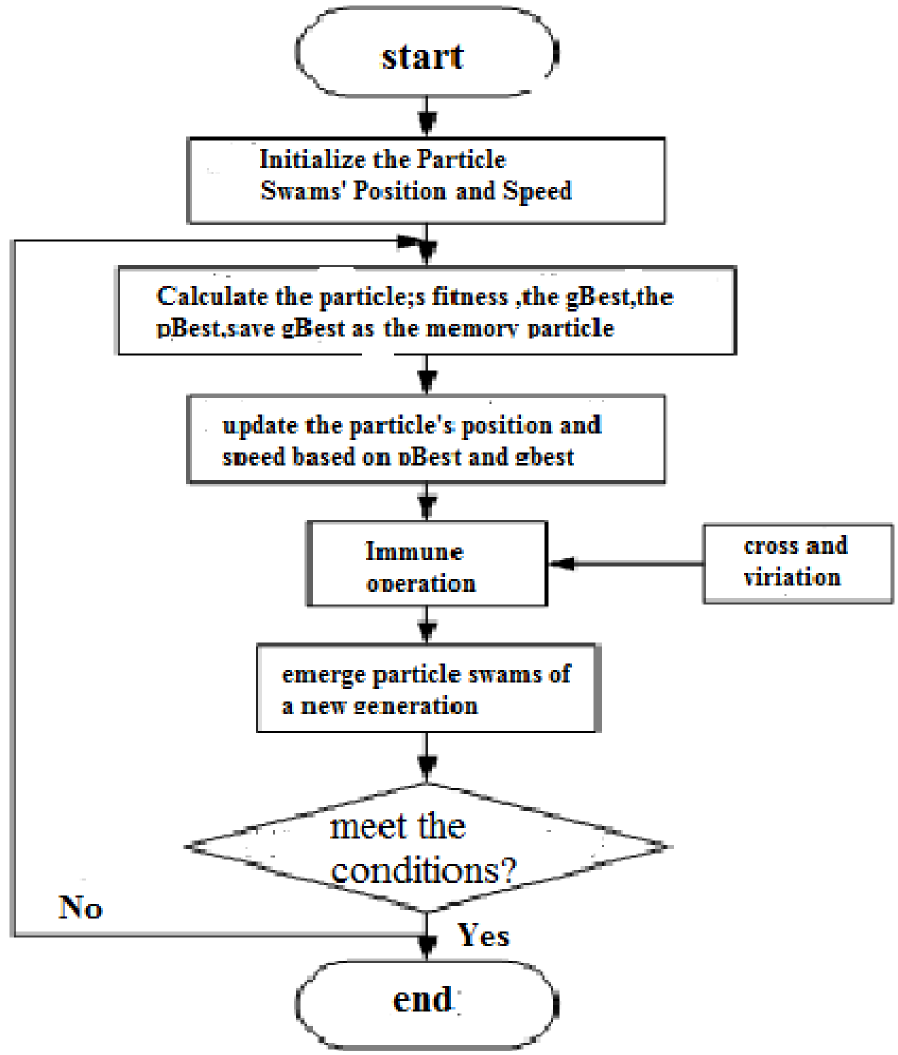

Figure 5 : The optimization process map

THE COMBINED ESTABLISHMENT OF CONTROL SYSTEMS

In order to obtain more accurate power assembly vibration via passive suspension is transferred to the body force and in this paper, by using the method of combined ADAMS with SIMULINK to co simulate on the simulation analysis of active suspension. First, established the virtual prototype of passive hydraulic mount in ADAMS and defined the input and output (input for powertrain mount system transfer to force F(T) and the output is the body acceleration), then exported it into working directory of MATLAB ; finally established the control system and simulated it in SIMULINK.

In this paper, the main research object is the in idle speed and the high frequency vibration by the powertrain transferred to the body force in the simulation when the powertrain is look as rigid body. The speed range of the four cylinder engine is 0-6000r/min and the vibration source is mainly two order of mechanical vibration. According to the performance of the engine at idle speed, the vibration frequency is 14.2Hz (850r/min) and the high frequency vibration frequency is 58.3Hz (3500r/min). Simulated by the input engine of sinusoidal signal simulation vibration (without considering the road excitation) and selected the vehicle body acceleration as the output of the simulation system.

Finally, established the simulation model in SIMULINK as shown below:

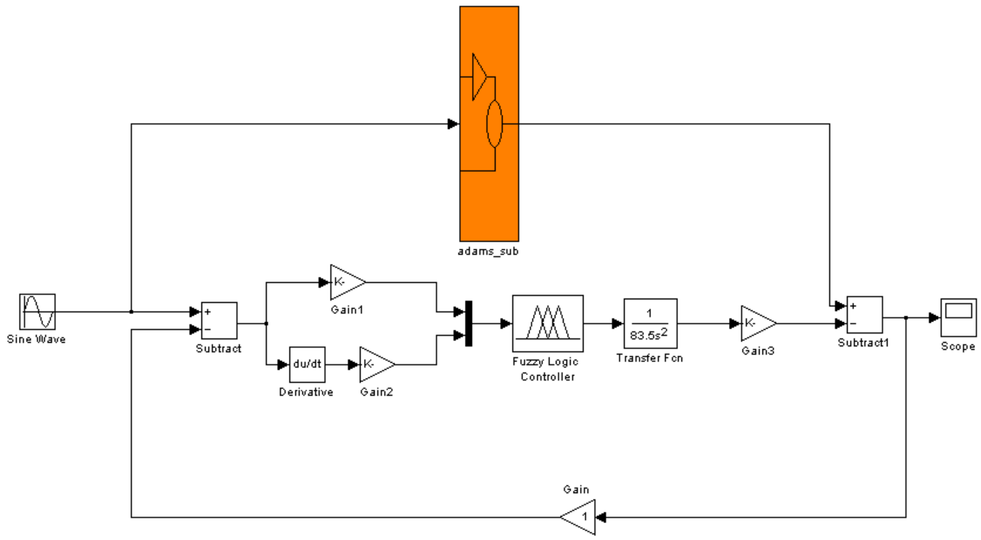

Figure 6 : Structure diagram of combined control system

By the control system of Figure 6, taking the simulation time is 0-5s; respectively, to the low frequency and high frequency vibration of main and passive suspension was simulated, we obtained the curve under condition of the body acceleration as shown in Figure 7-10:

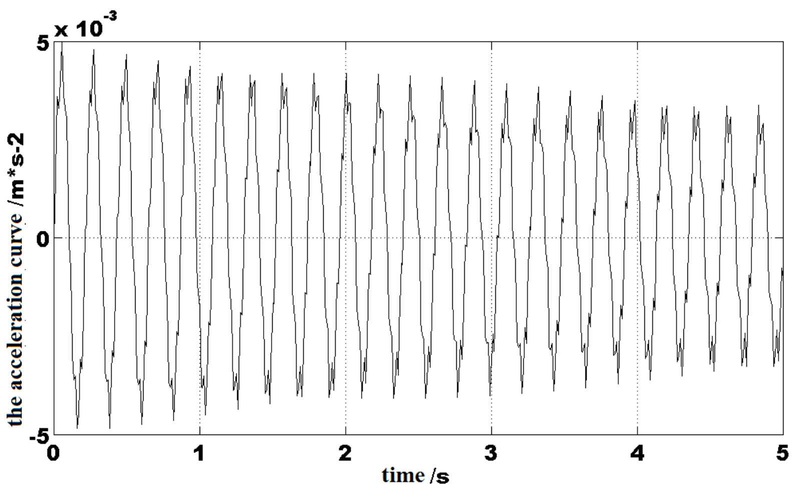

Figure 7 : The acceleration curve of passive suspension of the vehicle body (low frequency)

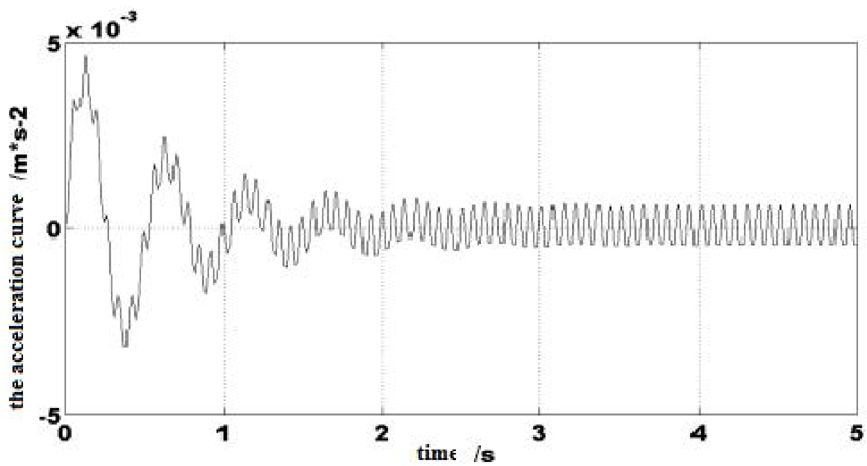

Figure 8 : The acceleration curve of active suspension of the vehicle body (low frequency)

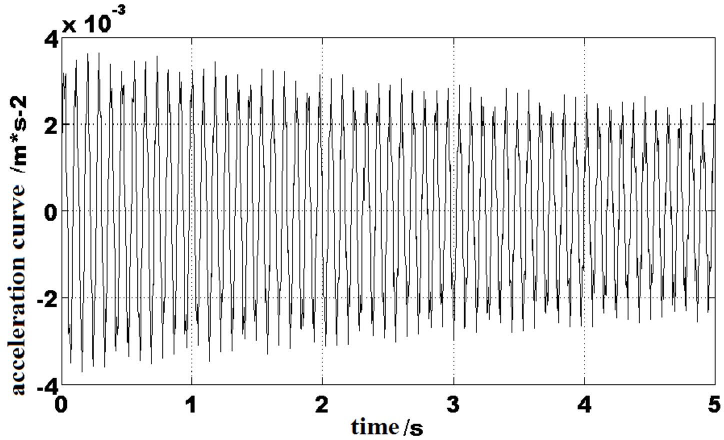

Figure 9 : The acceleration curve of passive suspension of the vehicle body (high frequency)

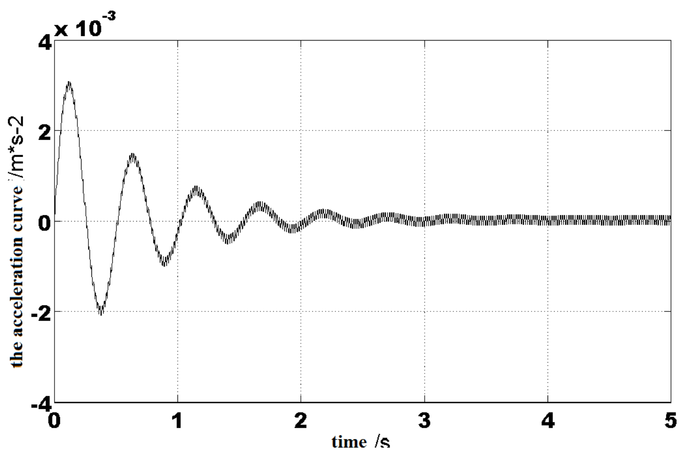

Figure 10 : The acceleration curve of active suspension of the vehicle body (high frequency)

Figures 7 and 8 are the respectively for the curves when the passive suspension and the active suspension are in the engine idle speed (frequency). We can see from Figure 8: the effect of passive suspension damping is general when using passive suspension and the body acceleration amplitude did not change significantly in the range of zero to five seconds, namely the passive suspension has a general effect of damping in low frequency. While using active suspension, the acceleration amplitude is significantly reduced after the system restart 0.5s; the system tends stable around 2 seconds and the acceleration amplitude of the vehicle boby is decayed about 90% after the stability of the vehicle body.

Figures 9 and Figures 10 are respectively for the boby acceleration curves when the passive suspension and the active suspension are in the engine speed (high frequency) during the operation. Using the vehicle acceleration amplitude of active suspension reached a stable value at about 2.5s and amplitude attenuation is about 95%; basically isolated the powertrain dynamic force transferred to the body; and the use of passive suspension of vehicle body acceleration change a little in the simulation time, the effect of the damping is general.

CONCLUSION

By ADAMS combined with SIMULINK simulation results show that: Based on the immune control of active suspension system in the low frequency and high frequency excitation conditions, compared to the passive suspension and it has a good damping effect is good. Which proves the feasibility of the scheme of active suspension.

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