Automobile Power-train—Coupling Vibration Analysis on Vehicle System

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ABSTRACT

Engine is one of the main vibration sources, and it has a big impact on the vibration characteristics of the car. Reasonable design of suspension system can obviously reduce the vibration of automobile powertrain and the body. Aiming at the vehicle vibration induced by automobile engine, the powertrain mounting system is put into the environment of vehicle to study its coupling vibration characteristics. A multi-body dynamic model including powertrain, vehicle, body and suspension system is established to carry out the simulation calculation, then analyze the vibration transmissibility and the coupling vibration between powertrain and vehicle. The research shows that there is difference in vibration characteristics between vehicle model and six-degree-of-freedom model of mounting system. After optimizing the parameters of mounting system, the vibration transmissibility has significantly reduced, thus the vibration reduction effect has been effectively improved.

Keywords: powertrain, vehicle, coupling vibration I-INCE Classification of Subjects Number(s): 38.3

1. INTRODUCTION

With the rapid development of automotive technology and manufacturing, the performance of Noise, Vibration and Harshness (NVH) is received more and more attention. The performance of NVH is an important indicator to measure the quality of automobile. How to effectively isolate the vibration transmission from engine to the body, and how to improve ride comfort have become a crucial content during car design.

Viewing from the full vehicle system, there are two major sources which cause the automobile vibration. One is the random road excitation in the process of driving; the other is the excitation engendered by reciprocating inertia force during engine working. As the improvement of road condition and the perfection of the automobile assemblies design, the effects of random road excitation on ride comfort gradually weaken. The design of modern vehicle is more and more emphasis on lightweight, while the mass of the engine is difficult to be reduced. This causes that the mass of the engine accounts for a rising proportion in the whole vehicle mass. At the same time, more and more cars use monolithic thin-walled structure body, which increases the body flexibility and vibration trend. Therefore, minimizing the transmission of vibration and noise generated by the engine to the body is the key of automobile vibration and noise reduction. And the powertrain mount as an important element of vibration transmission is of great significance for isolating the vibration of the engine.

Matthew M B and Michael M [1~2] designed the decoupling of powertrain mounting system by using the theory of impact center. Aiming at collocating inherent frequency of the system reasonably and realizing the vibration decoupling between various degrees of freedom, it optimizes the design of mounting system. The variables of optimization include mount rigidity, mount position and the ratio of mount vertical and lateral rigidity. Finally the system vibration coupling between the various degrees of freedom has been greatly reduced, and it ensured the reasonable distribution of system inherent frequency. Demic M A and Crowther A R [3~4] take the roll motion decoupling, reduction of inherent

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frequency of the roll modal, mount point response force and torque in response as its optimization goals, and it optimized the mount point location and mount characteristics. In order to make the engine mounting system vibration between various degrees of freedom decoupled, it also decorates the position and angle of mount components reasonably to make the elastic center locate in the center of mass or the principal inertia axis of engine mounting system. Shang G and others [5~6] came up with the mounting system optimization model built in torque axis coordinate system. Aiming at the system inherent frequency, it optimized the calculation which was restrained by each degree of freedom decoupling of the system and the first-order bending modal node. By adopting the method of matrix decoupling applied to the rigidity of engine mounting system, it realized the vibration decoupling and optimization design of the mounting system, as well as the improvement of vibration reduction effect. Taking the mounting system rigidity and location as the design variables, Shi W and Zhang J [7] optimized the vehicle, aiming at decoupling the rotational modal around x axis of the vehicle with the three mobile modal and collocating vertical vibration modal and rotational mode reasonably (increase the natural frequency of vertical vibration, reduce the natural frequency of the rotational modal).

Based on the above research, powertrain mounting system is in a vehicle which is a complicated system with many degrees of freedom. When modeling, we should take the body mass, damping of the suspension rigidity and the tire rigidity into account. The model should remarkably reflect the powertrain modal and the movement coupling under the environment of the vehicle. Therefore, only the complete vehicle model including body and suspension system can accurately analyze and control various influence of excitation on vehicle vibration. As a result, it can determine the position of mount components, the layout style and rigidity damping parameters and so on. It is of great importance for the design of appropriate mounting system, as well as the reduction of vehicle vibration and noise.

2. SYSTEM VIBRATATION TRANSMISSIBILITY

The most common indicator in evaluating the effect of vibration isolator is vibration transmissibility which is the ratio of vibration response of the object isolated and foundation vibration response, the transmissibility generally represents in the form of acceleration.

$$T_{dB} = 20\lg\left|\frac{a_p}{a_a}\right| \tag{1}$$

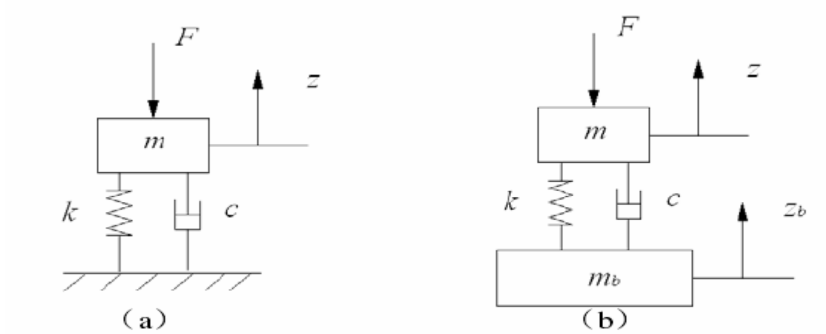
Where a_p is the acceleration amplitude of foundation vibration response; a_a is the acceleration amplitude of vibration response of the object isolated.

It is generally considered vibration transmissibility should be under 20dB which can show the vibration isolator is in a good state. If the vibration isolation rate is less than 20dB, the acceleration attenuation from the edge of active vibration(the powertrain) to the passive edge(the frame) is more than ten times, it means the vibration passed to the frame is no greater than one-tenth vibration of engine. Because the engine vibration source is related to the rotational speed and frequency, the vibration transmissibility of mounting is also associated with the rotational speed and frequency. All mount components should reach the standard of transmissibility mentioned above during the whole rotational speed in work.

3. ISOLATION VIBRATION MODEL

The existing powertrain mounting model is the single-degree-of-freedom liner vibration system based on the typical rigid-foundation. As shown in figure 1(a), mass m connects with the rigid foundation with infinite mass by a k rigidity spring and a c damping damper, the kinetic equation is

$$m\ddot{z} + c\dot{z} + kz = F \tag{2}$$



(a) Single-degree –of-freedom isolation vibration model
(b) Two-degree-of-freedom isolation vibration model

Figure 1 – This is a caption for the figure

The system inherent frequency is

$$\omega_n = \sqrt{k/m} \tag{3}$$

In the rigid-foundation model, it is generally recognized that the mass of supporting structure is much greater than the mass of the object isolated. So regardless of the movements of supporting structure, model's complexity is greatly simplified. In fact, the mass of car body is so lightweight that it is not far greater than the mass of powertrain at all, and the mass of the body which is the vibration isolation base is not regarded as the rigid-body with infinite mass. At the moment, the vibration isolation system should include three aspects, namely the object isolated (mass m), vibration isolator with k rigidity and c damping and the base (mass m_b) ， as shown in figure 1(b), the kinetic equation is

$$\begin{cases} m\ddot{z} + c(\dot{z} - \dot{z}_b) + k(z - z_b) = F \\ m_b\ddot{z}_b + c(\dot{z}_b - \dot{z}) + k(z_b - z) = 0 \end{cases} \tag{4}$$

Above equation can be rewritten as

$$\frac{mm_b}{m + m_b}\ddot{z}_p + c\dot{z}_p + kz_p = \frac{mm_b}{m + m_b}F \tag{5}$$

Where $z_p = z - z_b$, making $mm_b/(m + m_b) = m_p$ ·, so m_p can be called the “effective mass” of the system. And the system inherent frequency is

$$\omega_{np} = \sqrt{k/m_p} = \sqrt{\frac{m + m_b}{m_b}\frac{k}{m}} = \sqrt{\frac{m + m_b}{m_b}}\omega_n \tag{6}$$

Formula (6) shows that the mass of foundation has great influence on the calculation of system inherent frequency that the closer between the mass of object being isolated and the mass of the base, the greater effect on the system inherent frequency.

4. AUTOMOBILE POWERTRAIN—COUPLING VIBRATION ANALYSIS ON VEHICLE SYSTEM

This paper researches on the active vibration isolation of mounting system, when modeling, engine mounting system、car body、suspension and the rigidity and damping of tire are main concern, and it takes no account of influence caused by road surface disturbance.

4.1 Vehicle multi-body dynamic modeling

Powertrain's excitation force is mainly induced by unbalanced force and torque ripple when the engine works. When considering the vehicle modeling, it's able to simplify the excitation forces in other directions; And the vibration of body is mainly consider the up-and-down motion at z way that is perpendicular to the road surface, so the vehicle multi-body dynamic model with six-degree of freedom mounting system is established in the software of ADAMS, as shown in figure 2. Powertrain is regarded as rigid and the constraints in parts linked up by hinges and force, regardless of energy loss between hinges. Table 1 shows some parameters of the vehicle system. Defining the center of front axle as the origin of vehicle coordinate system, the coordinates of powertrain, body and unsprung mass can be obtained through coordinate transformation.

Table 1 – Some parameters of the vehicle system

Name	Value	Name	Value
Body mass m_b	880 kg	Front suspension rigidity k_{s1}、k_{s2}	17400 N/m
Front unsprung mass m_{u1}、m_{u2}	38 kg	Back suspension rigidity k_{s3}、k_{s4}	23500 N/m
Back unsprung mass m_{u3}、m_{u4}	30 kg	Tire rigidity k_{u1}~k_{u4}	200000 N/m
Body rotational inertia i_{bxx}	1702.8 kg·m^2	Front suspension damping c_{s1}、c_{s2}	1897 N·s/m
Body rotational inertia i_{byy}	258.9 kg·m^2	Back suspension damping c_{s3}、c_{s3}	2356 N·s/m

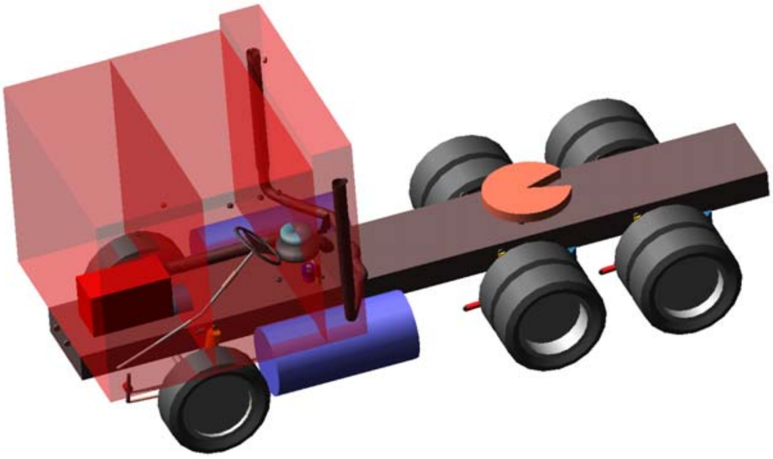


Figure 2 – Vehicle multi-body dynamic model

Definitions of 13 generalized variables in whole system are

$$\mathbf{x} = [\mathbf{x}_1 \quad \mathbf{x}_2 \quad \mathbf{x}_3]^T \tag{7}$$

Where $\mathbf{x}_1 = [x \quad y \quad z \quad \theta_x \quad \theta_y \quad \theta_z]^T$ is the generalized coordinate of powertrain; $\mathbf{x}_2 = [z_b \quad \theta_{bx} \quad \theta_{by}]^T$ is the generalized coordinate of sprung mass-body ; $\mathbf{x}_3 = [z_{u1} \quad z_{u2} \quad z_{u3} \quad z_{u4}]^T$ is the generalized coordinate of unsprung mass-wheel.

4.2 Simulation calculation and results analysis

4.2.1 Inherent frequencies comparison

Considering the multi-body dynamic model, energy method is used to carry out modal analysis and vibration excitation simulation in ADAMS/Vibration module, so the inherent frequency of vehicle system and energy distribution percentage based on the multi-body dynamic model can be obtained. As shown in Table 2, the main coupling vibration direction indicates the coupling vibration direction in which the energy is more than 10%. Because road surface excitation is not considered, the external force of system is still the excitation force of engine itself. For 4-cylinder engine, the main excitation forces include two-stage reciprocating inertial force of piston and reciprocating mass and the wave vector of crank torque.

Then a comparison analysis between the consequences and the inherent frequency analysis of the six-degree-of-freedom model of mounting system is made, as shown in Table 4.

Table 2 – Inherent frequency and decoupling ratio (energy distribution) of the vehicle system

Modal order	Inherent frequency (Hz)	Energy distribution direction	Energy distribution percentage	Main coupling direction
1	0.93	θ_{bx}	59.85	z
2	1.35	θby	83.70	—
3	1.48	zb	69.57	θ_{bx}
4	5.17	y	91.27	—
5	7.13	z	75.66	θ_x, z_b
6	8.08	x	71.13	θ_z
7	13.45	z_{u1}	53.14	z_{u2}
8	13.45	z_{u2}	53.14	z_{u1}
9	13.81	z_{u3}	70.86	z_{u4}
10	13.81	z_{u4}	70.86	z_{u3}
11	14.20	θ_y	65.18	θ_z, θ_{by}
12	16.71	θ_z	61.50	x, θ_y
13	19.25	θ_x	63.06	z, θ_{bx}

Table 3 – Inherent frequency and energy distribution (decoupling) in orders of each direction of six-degree-of-freedom model of mounting system

Modal order	Inherent frequency (Hz)	Energy distribution matrix（direction）(%)					
		x	y	z	θ_x	θ_y	θ_z
3	8.02	**79.33**	0.82	5.08	0.33	0.61	11.55
1	5.14	0.03	**99.05**	0.25	0.03	0.04	0.03
2	6.53	4.97	0.39	**78.14**	8.50	5.49	0.06
6	18.59	0.54	0.26	9.15	**73.16**	3.28	2.61
4	12.83	0.21	0.53	3.70	1.17	**71.81**	15.26
5	16.24	11.00	0.51	0.69	0.15	8.30	**71.64**

In Table 3, x, y, z respectively represents the translational vibrations of generalized coordinates x, y, z, θ_x, θ_y, θ_z respectively represents torsional vibrations around the coordinate axis x, y, z. The study shows that the system vibration decoupling ratio in the direction of y is relatively ideal, but the ones in other directions are all lower than 80%; In the second and sixth order modals, the coupling degree of z and θ_x is relatively high, while in the third order modal the coupling of x and θ_x is rather serious, and in the fifth order modal there is 3-degree-of-freedom coupling vibration phenomenon among x、θ_y and θ_z which indicates the system exists more serious vibration coupling so the vibration isolation effect is not good. From the point of view of frequency, the sixth order inherent frequency is 18.59 Hz. The inline 4-cylinder engine at idle state is given priority to two-stage torque excitation whose excitation source is same with the unbalance force of two-stage. Assuming that the original engine rotational speed at idle state is 780r/min and the excitation frequency is f = 2n/60 =26Hz. According to the principle of vibration isolation, the inherent frequency should be controlled under the $1/\sqrt{2}$ times of excitation frequency which can show the system is in good state, so the inherent frequency's maximum is 18Hz. Obviously, it is unfavourable to the vibration isolation of idle state when the frequency of highest order exceeds the inherent frequency's maximum. Besides, it is necessary to optimize the modal frequency of the mounting system if the range of resonance frequency is too wide and the frequency distribution is unreasonable.

Table 4 – System inherent frequencies comparison between the vehicle multi-body model and six-degree-of-freedom model

Main vibration direction		x	y	z	θ_x	θ_y	θ_z
Inherent frequency (Hz)	Six-degree-of-freedom model	8.02	5.14	6.53	18.59	12.83	16.24
	The vehicle multi-body model	**8.08**	**5.17**	**7.13**	**19.25**	**14.20**	**16.71**

As we know from Table 4, the powertrain inherent frequency between six-degree-of-freedom model and the vehicle model is rather different which mainly displays in the directions of z and θ_y. Taking the inherent frequency in z direction as an example, system equivalent mass has changed because powertrain and whole vehicle are both taken into account, based on formula (6), the ratio of two models' inherent frequencies in z direction is

$$\frac{f_{13DOF}}{f_{6DOF}} = \frac{\omega_{np}}{\omega_n} = \sqrt{\frac{m + m_b}{m_b}} \tag{8}$$

In terms of the vehicle, that the mounting system inherent frequency is controlled within a certain range as far away from both excitation frequency of system itself and other subsystems as possible is one of the goals of optimization. The example shows that the system inherent frequency got by the vehicle model can reflect the inherent frequency of powertrain subsystem, as well as the vibration coupling state between powertrain and the body. This is more comprehensive than six-degree-of-freedom model in the optimization matching of mounting system.

4.2.2 Vibration transmissibility comparison

1) Vertical vibration transmissibility comparison

Bring main rigidity values which are before and after optimization to the vehicle multi-body model, and work out the vibration transmissibility curves of mount components of the mounting system at a certain rotational speed range. Vibration transmissibility curves of the mount before and after optimization of the mount components rigidity parameters are shown in figure 3 and figure 4.

It can be seen from the figure 3, vertical vibration isolation effect of the original mounting system is not good, especially in the stage of idle speed, that vibration transmissibility reaches 40%.Transmissibilities in other rotational speed conditions are between 20% ~ 30%. So the vibration transmissibility of three mounts does not achieve ideal vibration isolation requirements basically. Therefore, it is necessary to optimize the mounting system. As shown in figure 4, after the main rigidity parameters optimization, three mounts all reach the ideal vibration isolation effect in some conditions, especially in the medium and high rotational speed period. The vibration isolation effect on three mount points of the system is good, and the vibration transmissibility is held on the range of 10% ~ 20%. Although transmissibility is big in the idle state, it decreased rapidly in low-speed stage. So it can satisfy the requirement of vibration isolation in engineering. From the vibration transmissibility change trend of each mount point, the fluctuation of transmissibility is not big, and the system is stable. These show that the optimization of spindle rigidity parameters of three mount components of the powertrain has a good effect.

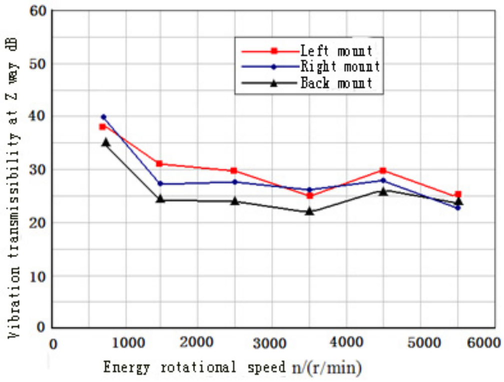


Figure 3 – Vibration transmissibility curves of the original system at Z way

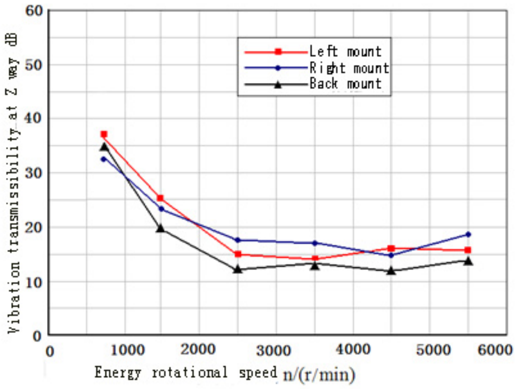


Figure 4 – Vibration transmissibility curves at Z way after optimization

2）Comparison of vibration transmissibility around crankshaft direction

In order to further illustrate the optimization effect, another indicator is used to evaluate the optimization results. Selecting the powertrain roll direction which is the direction around the crankshaft (namely the direction of θ_x) to analyze rotation torque vibration transmissibility, the computation expression is

$$J = \frac{M_0}{M_{\theta x}} \times 100\% \tag{9}$$

$$M_0 = \sum_{i=1}^{3} (F_{iy} z_i - F_{iz} y_i) \tag{10}$$

Where M_0 is the engine excitation torque amplitude; $M_{\theta x}$ is the reactive torque amplitude around x axis formed by dynamic resistance force of every mount; F_{iy}、 F_{iz} respectively represents the dynamic resistance force of the i_{th} mount in y and z directions; y_i and z_i respectively represents the displacements of i_{th} mount in y and z directions of generalized coordinate system.

From the perspective of active vibration isolation, it's necessary to try to reduce the transmission from the engine vibration to the body or frame to make the system vibration transmissibility minimal. From figure 5 ~ 6, before optimizing the main rigidity parameters of mount, powertrain mounting vibration transmissibility in the direction of θ_x is 27% when engine is working, while after optimization the maximum vibration transmissibility of powertrain mounting in the direction of θ_x has reduced to 21.5%.It signally reduces the vibration transmissibility of powertrain mounting system in the direction of θ_x and also improves the performance of vibration isolation of engine mounting system.

In conclusion, in accordance with the vibration simulation analysis of the vehicle model, the results show that the optimized vibration transmissibility for each mount point has improved obviously, as well as the vibration isolation effect of system.

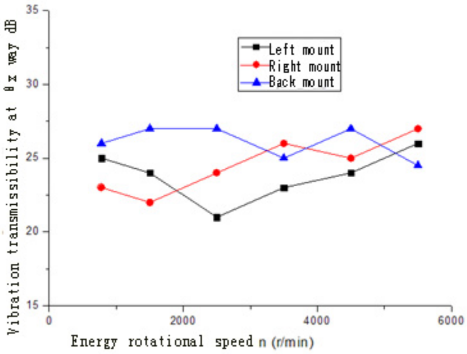


Figure 5 – Vibration transmissibility curves of the original system at θ_x way

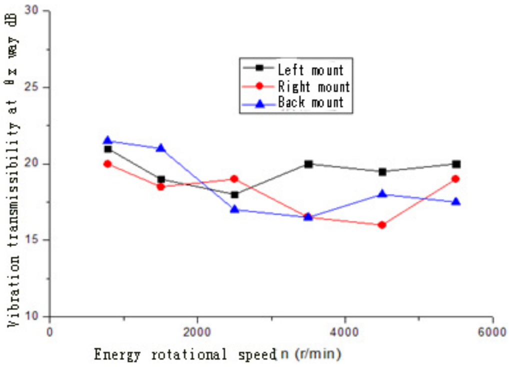


Figure 6 – Vibration transmissibility curves at θ_x way after optimization

5. CONCLUSIONS

In this paper, the powertrain mounting system is put into the environment of vehicle to study its coupling vibration characteristics. And based on the six-degree-of-freedom model of mounting system, a multi-body dynamic model including powertrain, vehicle, body and suspension system was established in the software of ADAMS. The research shows that there is difference in vibration characteristics between vehicle model and six-degree-of-freedom model of mounting system. After optimizing the main rigidity parameters of mounting system, the vibration reduction effect has been effectively improved.

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