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Automotive Vehicle Powertrain Mounting System Optimum Design and Simulation Analysis

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Abstract

Automotive powertrain is a main excitation source of vehicle vibration and has a great impact on vehicle ride comfort. Rational design of a vehicle powertrain mounting system could significantly reduce vehicle vibration and noise, hence improve vehicle ride comfort. Based on vibration theory, the mechanical and mathematical models of the powertrain mounting system have been built in this paper. The rational natural frequency allocation of the powertrain mounting system and system energy decoupling are optimized in the design. A specific vehicle mounting system has been designed using the optimization programs developed in MATLAB. At the same time, to verify the optimization results, a powertrain mounting system dynamic model has been established and adopted for a modal simulation analysis in MSC.ADAMS/VIEW. The results show that the frequency distribution of the optimized system is even more significant than its prototype and the degree of decoupling has been greatly improved. Therefore, the developed optimization programs can serve as a valuable tool for powertrain mount design.

Introduction

For an internal combustion engine, there exist two basic dynamic disturbances: the firing pulse due to fuel explosion in the cylinder; the inertia force and torque caused by rotating and reciprocating parts (piston, connecting rod and crank). The major function of engine mounts is to isolate an unbalanced engine disturbance force from the vehicle structure. [1]

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Elastomeric (or rubber) mounts have been used to isolate the vehicle structure from engine vibration since the 1930s. Since then, much significant advancement has been made to improve the performance of the elastomeric mounts. Elastomeric mounts can be designed for the necessary elastic stiffness rate characteristics in all directions for proper vibration isolation [1]. Johnson and Subhedar proposed a design method, which tunes all of the system natural frequencies to 6±16 Hz and decouples each mode of vibration through dynamic analysis and optimization. The relatively low vertical, pitch and yaw natural frequency ranges are based on consideration of isolating engine disturbance during a normal operation range [2]. Geck and Patton proposed a lower roll natural frequency for consideration of torque isolation and a relatively higher vertical natural frequency for consideration of avoiding shock excitation. This is because the requirements for shock prevention and vibration isolation are conflicting ones as discussed earlier [3]. Anon and Timpner made used of the center of percussion theory to achieve vibration decoupling. Powertrain and chassis were taken as rigid bodies and rubber was simplified to three directions perpendicular springs. Powertrain inertia spindle characteristics and impact of theories were used to choose the mount stiffness and location parameters [4,5]. Taking the location coordinates and the stiffness of mount as design variables, Arai and Kabozuka took the modal decoupling of powertrain systems around the crankshaft direction (RXX), the RX, RY, RZ three-phase modal, the rational allocation of the vertical vibration mode (RZ), and rolling mode (RZZ) as the optimization goal. After optimization, the vertical mount dynamic reaction force was declined in the idle state. It also demonstrated that the layout of the mount position has a greatest impact on the isolation performance of the mounting system [6]. Soloman discussed a technique to focus the mounts to uncouple the dynamic matrix as much as possible within design constraints, while also maintaining control of the powertrain rigid body natural frequencies [7]. Cho formulated a configuration and sizing design optimization problem for powertrain mounting systems. The objective function was to decouple and maximize the component modal kinetic energy of the powertrain subsystem. Natural frequencies and collinear roll vector condition were used as the constraints. Cartesian coordinates and spring rates of the mounts were selected as the design parameters. An automated design optimization procedure was developed using an optimization code based on the method of feasible direction (MFD) and MSC/NASTRAN [8]. Based on the discussion of three uncoupling schemes for the powertrain mounting system under three different coordinates, Lu and Fan analyzed uncoupling characteristics of the powertrain mounting system, and predicted the best elastically uncoupling performance of the system with V-formed mounting pairs at the front and rear ends of the powertrain [9].

This paper carries out the methods and tools for automotive vehicle powertrain mounting system optimum design and simulation analysis. It establishes the mechanical and mathematical models of the powertrain mounting system, and develops optimization programs in MATLAB. To verify the programs, it builds a powertrain mounting system dynamics model in MSC.ADAMS/VIEW and conducts a modal simulation analysis.

1 Isolation Vibration Design and Analysis Method of Powertrain Mounting System

1.1 Powertrain Mounting System Mechanics Model

It is necessary to establish the mechanical and mathematical models of the powertrain mounting system for dynamic analysis and optimized design. Powertrain mounting system natural frequency is usually 6-30Hz [10]. In the frequency range, powertrain vibration is in its rigid body mode so it can be simplified to a rigid body. Powertrain mount uses a rubber mount. Due to the distances between the mounts are shorter, the torsional elasticity can be ignored. Low frequency vibration isolation is mainly concerned. Since the amplitude of vibration is small, the damping effect can be ignored as well. Therefore, the mount can be simplified to three vertical springs.

The vehicle engine mounting system generally consists of an engine and several mounts connected to a vehicle structure. In this paper, a simplified mechanical model of a vehicle powertrain mounting system is established as shown in Figure 1. It has six degrees of freedom. The mount location can be determined by the three rectangular coordinates x, y, z originated at the centroid, and the rotation angles about three moving axes that bypassing the fixed center of mass and paralleling to the fixed axes. The system has six generalized displacement vectors, $\{X\} = \{x \;\; y \;\; z \;\; \alpha \; \beta \;\; \gamma\}^T$.

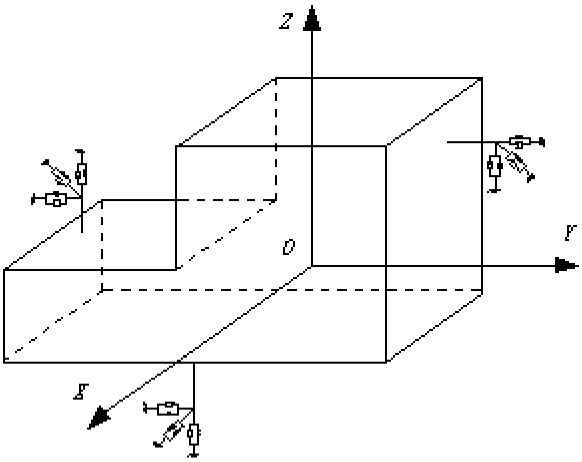

Figure 1: The mechanical model of powertrain mounting system

1.2 Powertrain Mounting System Mathematical odel

From an energy point of view, Lagrange method establishes a scalar relationship between the system kinetic energy, potential energy and power. Considering that the powertrain mounting system is a complex multi-freedom vibration systems, its mathematical model can be established by Lagrange method [11]. The general form of Lagrange's equation can be expressed as:

$$\frac{d}{dt}\left(\frac{\partial T_k}{\partial \dot{X}_i}\right)-\frac{\partial T_k}{\partial X_i}+\frac{\partial D}{\partial \dot{X}_i}+\frac{\partial T_p}{\partial X_i}=F_i \qquad (1)$$

$$i=1\sim n$$

where,

X_i- system six generalized coordinates, $\{X_i\}=\{\mathrm{x}\ \ \mathrm{y}\ \ \mathrm{z}\ \ \alpha\ \beta\ \gamma\}^T$

$\dot{X}_i$ - system six generalized speeds, $\{\dot{X}_i\}=\{\dot{\mathrm{x}}\ \ \dot{\mathrm{y}}\ \ \dot{\mathrm{z}}\ \ \dot{\alpha}\ \dot{\beta}\ \dot{\gamma}\}^T$

T_k - the system kinetic energy;

T_p - the system potential energy;

D- the system dissipated energy (as opposed to viscous damping);

F_i - 6 generalized coordinates corresponding to the system generalized force

Neglecting the damping characteristics of the system and considering that the system is a conservative system, namely, free vibration system, the system differential equations are expressed as follows:

$$[M]\{\ddot{X}\}+[K]\{X\}=0 \qquad (2)$$

where

$\{X\}$ - 6 generalized displacement vector, $\{X\}=\{\mathrm{x}\ \ \mathrm{y}\ \ \mathrm{z}\ \ \alpha\ \beta\ \gamma\}^T$

$\{\ddot{X}\}$ - 6 generalized acceleration $\{\ddot{X}\}=\{\ddot{\mathrm{x}}\ \ \ddot{\mathrm{y}}\ \ \ddot{\mathrm{z}}\ \ddot{\alpha}\ \ \ddot{\beta}\ \ \ddot{\gamma}\}^T$;

$[M]$- system mass matrix given by

$$[M]=\begin{bmatrix} m & 0 & 0 & 0 & 0 & 0 \\ 0 & m & 0 & 0 & 0 & 0 \\ 0 & 0 & m & 0 & 0 & 0 \\ 0 & 0 & 0 & J_x & -J_{xy} & -J_{xz} \\ 0 & 0 & 0 & -J_{xy} & J_y & -J_{yz} \\ 0 & 0 & 0 & -J_{xz} & -J_{yz} & J_z \end{bmatrix}$$

$[K]$- system stiffness matrix

1.3 Powertrain M-matrix and Stiffness Matrix K in the Torsion Axis Coordinate System

The design of a powertrain mounting system is generally carried out in the torsion axis coordinate system, so it is necessary to translate the powertrain moment of inertia matrix in the three-dimensional Cartesian coordinate system into a moment of inertia matrix in the torsion axis coordinate system.

In general, the measured moment of inertia matrix is not a principal inertia matrix in the principal inertia axis coordinate system, so the mass matrix M is non-diagonalizable. With a matrix congruent transformation, the powertrain moment of inertia matrix and mass matrix M in the torsion axis coordinate system can be calculated according to the moment of inertia matrix in the principal inertia axis coordinate system. In the torsion axis coordinate system, the powertrain stiffness matrix K can be calculated by Equation (3).

$$[K] = \sum_{i=1}^{n} F_i^T T_i^T K_i T_i F_i \quad (3)$$

where,

$$F_i = \begin{bmatrix} 1 & & & 0 & Z_i & -Y_i \\ & 1 & & -Z_i & 0 & X_i \\ & & 1 & Y_i & -X_i & 0 \end{bmatrix}$$

$(X_i Y_i Z_i)^T$ -the first i mount flexible center coordinate in the torsion coordinate system;

K_i -the first i mount stiffness matrix in itself elastic principal axis coordinate system;

T_i- the direction cosine matrix of the powertrain torsion axis coordinate system in the first i mount flexible principal axis coordinate.

T_i can be represented by Euler angle, such as Equation (4):

$$T_i = \begin{bmatrix} \cos\psi_i & -\sin\psi_i & 0 \\ \sin\psi_i & \cos\psi_i & 0 \\ 0 & 0 & 1 \end{bmatrix} * \begin{bmatrix} 1 & 0 & 0 \\ 0 & \cos\theta_i & -\sin\theta_i \\ 0 & \sin\theta_i & \cos\theta_i \end{bmatrix} * \begin{bmatrix} \cos\varphi_i & -\sin\varphi_i & 0 \\ \sin\varphi_i & \cos\varphi_i & 0 \\ 0 & 0 & 1 \end{bmatrix} \quad (4)$$

where, $(\psi_i, \theta_i, \varphi_i)$ are the attitude coordinates of the powertrain torsion coordinate system in the first i mount flexible principal axis coordinate [12].

2 Powertrain Mounting System Parameter Optimization

Tables 1- 3 show the parameters of three mounts of a vehicle. Since the vehicle layout had been determined, the mount fixing locations and angles can not be treated as the design variables. To improve the powertrain performance, it is necessary to optimize the mount stiffness.

Table 1: Mount coordinates in torsion axis coordinate system

Mount	Mount coordinates in torsion axis coordinate system		
	X	Y	Z
1	-122.41	609.17	192.42
2	-37.41	-365.83	192.42
3	186.59	39.62	-234.48

Table 2: Mount attitude coordinates

Euler angles / Mount	ψ, Precession angle,	θ, Nutation angle,	φ, Angle of rotation,
1	0°	0°	0°
2	90°	90°	0°
3	0°	0°	0°

Table 3: Mount stiffness

Coordinate / Mount	X, (N/mm)	Y, (N/mm)	Z, (N/mm)
1	165	165	160
2	140	60	165
3	185	10	10

2.1 Optimization Model Established by the Natural Frequency Rational Allocation

2.1.1The Objective Function

Powertrain has two types of vibration. One is caused by uneven road surface, usually low-frequency random vibration below 2.5Hz. The other is caused by the engine vibration. The natural frequency of the powertrain mounting system is greater than the minimum frequency of 2.5Hz, typically 6 ~ 8 Hz. The engine vibration frequency range can be calculated using Equation (5):

$$f_n = \frac{N \cdot n_{\min}}{30C} \sim \frac{n_{\max}}{60} \quad (Hz) \tag{5}$$

Therefore, the natural frequency of the powertrain mounting system is less than the maximum frequency $\frac{1}{\sqrt{2}} \frac{N \cdot n_{\min}}{30C}$ (Hz). Commonly, it is lower than 30 Hz. In different directions, the design requirements are different so their natural frequencies are different too. Therefore, according to the design requirements, our objective functions are listed below:

$$
\begin{aligned}
f(1) &= abs(f_1 - f_x) \\
f(2) &= abs(f_2 - f_y) \\
f(3) &= abs(f_3 - f_z) \\
f(4) &= abs(f_4 - f_{\theta x}) \\
f(5) &= abs(f_5 - f_{\theta y}) \\
f(6) &= abs(f_6 - f_{\theta z})
\end{aligned} \quad (6)
$$

2.1.2 Constraints

By setting all of the mount stiffness as the optimization design variables and assuming the stiffness of each mount is within reasonable range, the constraints can be represented as following expressions:

$$
\begin{aligned}
k_{p\min} &\le k_{pi} \le k_{p\max} \\
k_{q\min} &\le k_{qi} \le k_{q\max} \\
k_{r\min} &\le k_{ri} \le k_{r\max}
\end{aligned} \quad (7)
$$

where, i = 1,2, ... n, on behalf of each mount.

2.1.3 Optimization Methods and Results

Our design is a typical multi-objective programming problem. In the MATLAB optimization toolbox, the function “fgoalattain” can be used to solve a multi-objective programming problem. The procedure for solution can be listed as follows:

```
goal=[0,0];
weight=[10^-5,10^-9];
x0=1000*[165 165 160 140 60 165 185 10 10];
A=eye(9);
b=100000*[7;7;7;7;7;7;7;7;7];
lb=zeros(9,1);
options=optimset('display','iter','MaxFunEvals',30000,'MaxIter',2000)
[x,fval,attainfactor,exitflag,output] = fgoalattain (@wobjfun, x0,goal, weight, A,b,
[],[], lb,[], [], options)
```

After 244 iterations with 3232 operations, the objective function converges to its optimal solution. Each resultant mount stiffness is listed in Table 4.

Table 4: Mount stiffness optimization results

Coordinate mount	X, (N/mm)	Y, (N/mm)	Z, (N/mm)
1	257.41	177.53	35.09
2	141.40	140.40	396.63
3	698.80	242.64	699.43

2.2 Optimization Model Based on Energy Decoupling

It is not sufficient to optimize a mounting system by only considering the rational allocation of the mounting system natural frequency**.** In addition, the mounting system is also optimized by an energy decoupling method.

2.2.1 The Objective Function

We define the mounting system mode shape matrix as $[X_i]=[\chi_1,\chi_2,\chi_3,\chi_4,\chi_5,\chi_6]$

(where χ_i is the modal shape vector, i = 1 ~ 6) and the corresponding modal frequency as

ω_i. According to the powertrain mass matrix $[M]$ and modal shape matrix $[X_i]$, the

vibration energy distribution can be obtained and written in a matrix form $[T]$. For the

first i-order modal vibration, the total kinetic energy of the mounting system is given by Equation (8):

$$[T_{kl}]=\frac{1}{2}\omega_i^2[X_i]^T[M][X_i] \tag{8}$$

Expanding (8) leads to

$$[T_{kl}]=\frac{1}{2}\omega_i^2\sum_{l=1}^{6}\sum_{k=1}^{6}(\chi_i)_l(\chi_i)_k m_{kl} \tag{9}$$

where

ω_i - mounting system first i-order natural frequency;

m_{ki} - powertrain mass matrix; $[M]$: the first k row, first column element of l;

χ_i -the first i-mode shape matrix $[X_i]$: column vector.

Equation (9) contains 36 matrices in total. For the first k diagonal elements of line, the vibration energy is allocated directly to the first k-generalized coordinates. Non-diagonal elements of the first element k and l in a generalized coordinate coupled with each other caused by energy exchange, is expressed as $[T_{kl}]_i$. The mounting system total energy of all the elements of the matrix is expressed as $\left[T_{ki}\right]$. Thus, when the mounting system vibrates at the first i order natural frequency, the energy percentage of the first k-generalized coordinates can be calculated as follows:

$$T_p = \frac{\left[T_{kl}\right]_i}{\left[T_{kl}\right]} = \frac{\sum_{l=1}^{6} (\chi_i)_l (\chi_i)_k m_{kl}}{\sum_{l=1}^{6} \sum_{k=1}^{6} (\chi_i)_l (\chi_i)_k m_{kl}} \times 100\% \qquad (10)$$

Note that T_p indicates the mounting system decoupling in the k direction. If T_p = 100% for the first i-order modal vibration then the mounting system energy is concentrated in the coordinates k. Thus the vibration enegy of the remaining generalized coordinates is 0. To improve the powertrain mounting system decoupling in a direction, such as along the z axis and around the x-axis vibration, it is required to change the mount position and inclination and stiffness so that a gradual increase in the percentage of energy is close to 100% to make vibration to be concentrated in the z direction and θx.
We apply the objective functions expressed in Equation (11):

$$\begin{cases} f(1) = 1 - T_p(1,1) \\ f(2) = 1 - T_p(2,2) \\ f(3) = 1 - T_p(3,3) \\ f(4) = 1 - T_p(4,4) \\ f(5) = 1 - T_p(5,5) \\ f(6) = 1 - T_p(6,6) \end{cases} \qquad (11)$$

2.2.2 Constraints

Similarly, we select the stiffness parameters of the mount as the optimization design variables. The stiffness of the mount must be reasonable and the constraints are expressed in Equation (7). At the same time, the frequency constraints should be considered and expressed as Equation (12).

$$f_{j\min} \le f_j \le f_{j\max} \qquad (12)$$

where, j = 1,2, ..., 6, on behalf of six degrees of freedom.

2.2.3 Optimization Method and Optimization Results

The optimization is still a multi-objective programming problem. Hence the function "fgoalattain" can still be used. The initial values are obtained from Section 2.1.3 so that the natural frequency of the system can be in the constraint range. The design procedure is depicted as follows:

```
goal=[0,0];
weight=[10^-12,10^-12];
x0=1000*[ 257.41 177.53 35.09 141.40 140.40 396.63 698.80 242.64 699.43];
A=eye(9);
b=100000*[7;7;7;7;7;7;7;7;7]
lb=8000*[1;1;1;1;1;1;1;1;1];
options=optimset('display','iter','MaxFunEvals',50000,'MaxIter',2000)
[x,fval,attainfactor,exitflag,output] = fgoalattain(@twoobjfun,x0,goal,
weight,A,b,[],[],lb,[], @confun,options)
```

After 1900 iterations with 20045 floating-point operations, the objective function converges to its optimal solution. Optimization results are showed in Table 5.

Table 5: Mount stiffness optimization results

Coordinate mount	X, (N/mm)	Y, (N/mm)	Z, (N/mm)
1	250.51	89.81	71.12
2	147.89	148.00	481.13
3	643.46	187.14	190.37

3 Powertrain Mounting System Simulation and Analysis Based on MSC.ADAMS

3.1 Dynamics Model

According to the geometric position parameters of the powertrain mounting system (such as centroid and mounting positions), the dynamics model has been established for multibody dynamics software MSC.ADAMS. In the dynamics model, the mount is replaced by bushing and shown as Figure 2.

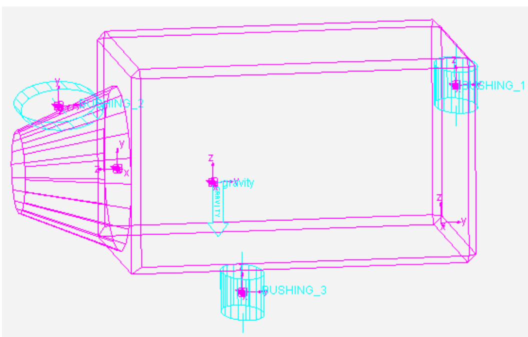

Figure 2: Powertrain dynamics model

3.2 Modal analysis and decoupling authentication

After assigning each component of the model and applying vibration analysis using MSC.ADAMS solver (Vibration), we can easily solve the six-order system, the frequency and the order frequency of energy distribution.

3.2.1 Initial Conditions

For the initial conditions, the distribution of the order of powertrain mount system energy is shown in Table 6.

Table 6: Initial energy distribution of the order

Mode	Total	X	Y	Z	RXX	RYY	RZZ	RXY	RXZ	RYZ
1	99.78	0.07	11.67	**83.19**	2.88	1.95	0.03	0.00	0.00	0.00
2	100.40	5.65	**47.09**	2.83	1.26	43.52	0.05	0.00	0.00	0.00
3	100.26	4.59	38.99	11.67	0.15	**43.91**	0.95	0.00	0.00	0.00
4	100.33	**84.75**	0.18	0.01	0.17	8.93	6.29	0.00	0.00	0.00
5	99.83	4.93	0.77	0.00	0.00	1.46	**92.66**	0.00	0.00	0.00
6	102.20	0.01	1.30	2.30	**96.94**	0.34	1.32	0.00	0.00	0.00

From the first-order mode and the energy distribution, it can be seen that along the Y direction and around Y axis are seriously coupled with the other directions.

3.2.2 Frequency Optimization Design Verification

In Section 2.1, the optimization design has been completed for the rational allocation of the powertrain mount system frequency. The energy distribution from the optimized system is shown in Table 7.

Table 7: Energy distribution of the optimization system

Mode	Total	X	Y	Z	RXX	RYY	RZZ	RXY	RXZ	RYZ
1	99.85	0.70	14.51	**55.91**	0.27	28.34	0.11	0.00	0.00	0.00
2	100.01	0.19	**85.26**	9.71	0.00	4.72	0.12	0.00	0.00	0.00
3	99.99	**81.67**	0.01	9.21	0.00	7.87	1.22	0.00	0.00	0.00
4	99.72	14.46	0.03	24.77	3.14	**54.42**	2.90	0.00	0.00	0.00
5	111.77	2.38	0.17	0.38	51.51	4.35	**52.98**	0.00	0.00	0.00
6	91.03	0.59	0.02	0.01	**46.25**	0.40	43.75	0.00	0.00	0.00

From the first-order mode and the energy distribution, it can be seen that along the Z direction, around the X,Y and Z directions, there are serious coupling problems. It can be observed that considering only the rational allocation of natural frequencies is not sufficient for the system decoupling.

3.2.3 Decoupling of Energy Optimal Design Verification

In section 2.2, the powertrain mount system can be optimized based on energy decoupling. After optimal design, the first-order modal natural frequency and the first-order distribution of energy are shown in Tables 8 and 9.

The first-order mode and the energy distribution can be seen that the frequency bands of the optimized system are within a reasonable range. The maximum frequency is approximately 17 Hz and the lowest frequency is about 7 Hz. Therefore, a high degree decoupling of the system is achieved.

Table 8: The first-order natural frequency of mode

MODE NUMBER	UNDAMPED NATURAL FREQUENCY	DAMPING RATIO	REAL	IMAGINARY
1	6.79747	0	0	+/- 6.79747
2	7.00131	0	0	+/- 7.00131
3	10.6145	0	0	+/- 10.6145
4	12.6423	0	0	+/- 12.6423
5	16.8894	0	0	+/- 16.8894
6	17.5701	0	0	+/- 17.5701

Table 9: Energy distribution of the order

Mode	Total	X	Y	Z	RXX	RYY	RZZ	RXY	RXZ	RYZ
1	99.99	0.00	0.12	**97.88**	0.01	1.97	0.01	0.00	0.00	0.00
2	100.01	0.00	**99.74**	0.13	0.01	0.00	0.13	0.00	0.00	0.00
3	99.97	0.36	0.02	1.98	0.00	**96.73**	0.89	0.00	0.00	0.00
4	100.00	**99.63**	0.00	0.00	0.00	0.34	0.02	0.00	0.00	0.00
5	101.24	0.01	0.12	0.00	0.30	0.99	**99.81**	0.00	0.00	0.00
6	101.27	0.00	0.01	0.01	**100.92**	0.08	0.26	0.00	0.00	0.00

4 Conclusion

Based on vibration theory, a mechanical model of the powertrain mounting system has been established and transferred into a mathematical model according to Lagrangian theory. By using the optimal design theory, the rational allocation of natural frequency and energy decoupling are optimized for designing the powertrain mounting system. Optimization programs have been developed in MATLAB. To validate the optimization design results, a specific powertrain mounting system dynamics model has been built and carried out for a modal simulation analysis in MSC.ADAMS/VIEW. The results suggest that a more reasonable design can be obtained by first achieving the rational allocation of the system natural frequency and then obtaining a systematic energy decoupling. With our developed optimal design approaches, the degree of decoupling has been greatly improved.

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Biography

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