Study on Simulation and Optimization of Three-point Powertrain Mounting System

Kaiyan Wang¹, *, Zhicong Liu¹ and Xiaoxue Ren²

¹School of Automobile and Transportation, Shenyang Ligong University, Shenyang, China
²Department of Logistics Management, Liaoning Provincial College of Communications, Shenyang, China

*Corresponding author: wangky@sylu.edu.cn

Abstract. The characteristics of the three-point powertrain mounting system are studied by multi-body dynamics method. The 6-DOF model of the powertrain mounting system is established in ADAMS software. The natural frequency and the decoupling rate of modal energy are obtained through the simulation analysis. The 13-DOF model of the complete vehicle is established based on 6-DOF model, and the dynamic simulation analysis is carried out. The accuracy of the vehicle model is verified by comparing with the vehicle test data. The model is optimized by taking the stiffness and position parameters as the optimization variables, taking the transmission force and modal coupling rate as the optimization objectives. The decoupling rate of each mode is greatly improved, and the transmission force of the mounting system is effectively reduced after optimization.

Keywords: Powertrain mounting system, modal analysis, 13-DOF model, optimization.

1. Introduction

Powertrain mounting system is mainly composed of engine, transmission, clutch and mounting elements. The powertrain is usually connected with the frame through elastic elements, which can improve the NVH (Noise, Vibration and Harshness) performance of the vehicle. The vibration of powertrain can be effectively reduced by high quality mounting elements [1].

The theoretical analysis of the vibration isolation performance of powertrain mounting system includes: impact center theory, mechanical impedance method, energy decoupling method, vibration isolation test method, mounting design method [2, 3, 4], etc. The optimal design of the mounting system usually takes the layout angle, layout position or stiffness as the optimization variables, takes the modal energy decoupling rate or mounting transmission force as the optimization objectives [5, 6]. The optimal design of the mounting system is realized by single objective or multi-objective optimization method [7].

In this paper, the vibration isolation performance of the three-point powertrain mounting system is analyzed. ADAMS software is used to establish the model of the mounting system and the complete vehicle. The modal characteristics and dynamic response characteristics are obtained through simulation
analysis. The simulation results are compared with the test results to verify the accuracy of the model. The mounting system is optimized through the co-simulation method of ADAMS and ISIGHT software.

2. 6-DOF model of powertrain mounting system

2.1. Modelling of powertrain mounting system

2.1.1. Simplification of Powertrain. There are many parts in powertrain mounting system, so it is necessary to simplify the powertrain before modelling. The modal frequency of the elastic body of the powertrain is far greater than the rigid natural frequency of the mounting system. Therefore, the powertrain is regarded as a rigid body. The mass of the pipe and liquid is ignored. The powertrain is simulated by the mass point in the software.

2.1.2. Simplification of mounting elements. The mounting element is simplified as an elastic element with three-dimensional stiffness and damping. It is simulated by the bushing element in the software. The powertrain is connected to the ground by the bushing elements in ADAMS. The stiffness and damping parameters are set in the software.

2.2. Simulation and analysis
The modal analysis of the mounting system model is carried out, and the results are shown in Table 1. The simulation results show that modal energy decoupling rate of $Y$, $Z$ and pitch mode is good, reaching 97.54%, 96.32% and 96.71%. The decoupling rate of $X$, roll and yaw mode is poor. The decoupling rate of yaw mode is less than 60%. The mounting system is prone to vibration coupling problems under these three modes. Therefore, it is necessary to improve the energy decoupling rate of these three modes.

| Modal frequency (Hz) | X  | Y   | Z  | Pitch | Roll | Yaw |
|----------------------|----|-----|----|-------|------|-----|
| 5.28                 | 0.08 | 97.54 | 0.13 | 2.20   | 0.24 | 0.24|
| 6.91                 | 0.63 | 0.12 | 96.32 | 0.13 | 2.53 | 0.61|
| 8.15                 | 0.24 | 0.22 | 1.10 | 0.06 | 1.63 | 8.65|
| 11.83                | 5.31 | 0.15 | 1.92 | 0.86 | 86.86 | 34.46|
| 13.68                | 3.52 | 0.27 | 0.35 | 5.48 | 14.17 | 58.57|
| 14.32                | 0.16 | 1.70 | 0.18 | 96.71 | 6.23 | 3.74|

3. 13-DOF model of complete vehicle
The 13-DOF (Degree of Freedom) model is established based on the powertrain mounting system model. The body, wheels and suspensions are added in the model. There are 3-DOF for the body (roll, pitch and vertical). There is only vertical degree of freedom for the four wheels. So, there are 13-DOF in the vehicle model with 6-DOF of the powertrain mounting system model.

3.1. Analysis of engine excitation
The engine excitation includes: pitching moment, roll moment and reciprocating inertia force. The engine analyzed in the paper is a four-cylinder and four stroke engines. The reciprocating inertia force can be given as follows:

$$F_z = 4m_r r \omega^2 \lambda_p \cos(2\omega t)$$  \hspace{1cm} (1)

Where: $m_r$ is the equivalent mass of the piston, $r$ is the radius of the crank, $\omega$ is the angular velocity of the crankshaft, $\lambda_p$ is the ratio of crank to connecting rod.
The pitching torque of engine can be given as follows:

\[ M_y = M_y + M_g \approx M_{eo}(1 + 1.3\sin(2\omega t)) \] (2)

Where: \( M_g \) is the torque generated by combustion gas, \( M_f \) is the reciprocating inertia torque, \( M_{eo} \) is the average output torque of the engine.

The engine roll moment can be expressed as follows:

\[ M_X = F_z \cdot e \] (3)

Where: \( e \) is the distance between the center of mass and the center line of 2nd and 3rd cylinder in the direction of crankshaft axis.

3.2. Modelling of complete vehicle model

The mass of each system (except for the powertrain and the wheels) is integrated into the vehicle body mass. The vehicle body and the wheels are regarded as rigid bodies and replaced by the mass points. The spring and absorber of the suspension are replaced by the spring elements in the model. The stiffness of tire is also simulated by the spring elements, as shown in Figure 1. The engine excitations are added at the mass centre of the powertrain.

![Figure 1. 13-DOF model of vehicle.](image1.png)

3.3. Complete vehicle test

The equipment used in the test includes: LMS data acquisition instrument, American PCB acceleration sensor, laptop. The acceleration sensor is placed at the passive and active side of the mounts [8], as shown in Figure 2. The condition of the test is idle, and the speed of the engine is 750 rpm.

![Figure 2. Acceleration sensors layout.](image2.png)

3.4. Results of simulation and test

The acceleration in vertical direction of the test and the simulation on the active side of the mounts are compared. The frequency response of the data can be obtained by Fourier transform. The peak frequency
of the test and the simulation is 25.96 Hz and 25Hz. The amplitude of the peck frequency is shown in Table 2. It can be found that the results of the test and simulation are more consistent, which can prove the correctness of the model. The simulation errors are mainly caused by the simplification of the system in the process of establishing the model. The errors can be reduced if the elastic vibration is taken into the model.

### Table 2. Acceleration amplitude of peak frequency

| Type       | Left mount (mm/s²) | Right mount (mm/s²) | Rear mount (mm/s²) |
|------------|--------------------|---------------------|--------------------|
| Simulation | 1024.92            | 960.63              | 1768.04            |
| Test       | 1131.14            | 886.63              | 1604.57            |

4. **Optimization of the powertrain mounting system**

4.1. **Optimization objective**

The optimization models of the mounting system are different according to different design objectives. In this paper, the optimization objective is to minimize the modal coupling rate and the transmission force of each mount [9]. The objective function is given as follows:

\[
f(x) = \min \left\{ \sum_{i=1}^{6} a_i \left[ 1 - \max (T_{pi}) \right] \right\}
\]

Where: \( i \) is the response factor, \( \max (T_{pi}) \) is the decoupling rate of each main mode, \( a_i \) is the weighting factor.

\[
F = \sum_{i=1}^{n} \sqrt{F_{ix}^2 + F_{iy}^2 + F_{iz}^2}
\]

Where: \( n \) is the number of mounting elements, \( F \) is the transmission force of each mount.

4.2. **Design variables and constraint condition**

The layout position and stiffness in X, Y and Z directions of mounting elements are selected as the design variables. Therefor there are 18 variables. The range of the design variables is 20%.

The modal frequency range and interval are used as the constraints. The modal frequency interval should be more than 1 Hz. The frequency range should be 5Hz to 21.2Hz.

4.3. **Optimization results**

The model is optimized through the co-simulation method of ADAMS and ISIGHT software. DOE optimization method is used to optimize the model. 400 groups of variable combination data are generated by the software. The optimization results of the decoupling rate are shown in Table 3. The decoupling rate of each mode is improved, especially in the yaw mode. The modal energy decoupling rate of yaw mode is promoted from 53.88% to 99.71%. The modal energy decoupling rate of each mode is above 97%. The system is decoupled and meets the optimization requirements.

### Table 3. Comparison of modal energy decoupling rate

| Date type          | Decoupling rate (%) |
|--------------------|---------------------|
|                    | X   | Y   | Z   | Pitch | Roll | Yaw  |
| Original value     | 91.92 | 97.73 | 96.93 | 92.60 | 87.12 | 53.88 |
| Optimization value | 98.06 | 98.73 | 97.71 | 99.97 | 99.08 | 99.71 |
The transmission force after optimization is shown in Table 4. The vertical transmission force of each mount is reduced after the optimization and the life of each mounting element can be prolonged.

| Date type         | Left mount (N) | Right mount (N) | Rear mount (N) |
|-------------------|----------------|-----------------|----------------|
| Original value    | 692.53         | 910.25          | 69.24          |
| Optimization value| 665.65         | 905.59          | 62.46          |

5. Conclusion

The 6-DOF model of powertrain mounting system is established. The modal analysis is carried out. The modal decoupling rate of X, pitch and yaw mode is poor. The mounting system needs to be optimized.

The 13-DOF vehicle model is established based on the powertrain mounting system model. The simulation results are compared with the vehicle test results. The peak frequency and the amplitude of the simulation results are closed to test results. The errors are within 3.84% of the peak frequency and 11% of the amplitude. The correctness of the model is verified.

The model is optimized by DOE method. The optimization objective is to minimize the modal coupling rate and the vertical transmission force. The layout position and stiffness in X, Y and Z directions are taken as the design variables. The modal energy decoupling rate of each mode is above 97% after optimization. The vertical transmission force of each mounting element is also reduced. The vibration isolation performance of the powertrain mounting system is improved effectively.

Acknowledgments

This work is financially supported by Natural Science Foundation of Liaoning Province, China (20180550710).

References

[1] Courteille, E., and F. Mortier, “Idle shake vibration optimization of an engine mounting system through the practical application of transfer path analysis,” Noise Control Engineering Journal. 68.6 (2020): 459 - 469.

[2] Yu, Y., N. G. Naganathan, and V. D. Rao, “Literature review of automotive vehicle engine mounting systems,” Mechanism and Machine Theory. 36.1 (2001): 123 - 142.

[3] Yan, H. A., and S. Xu, “Energy Method of Decoupling and Computer Optimization of Engine Mounting Systems,” Automotive Engineering. (1993).

[4] Guo Rong and Zhou Zi wei, “Non-Linear Modeling and Parameter Identification of Semi-Active Engine Mounts With Air Spring,” Journal of Vibration and Acoustics. 142.1 (2020).

[5] Fan Ranglin, Fei Zhenman, and Zhou Bangyu, and Gong Huabing et al., “Two-step dynamics of a semiactive hydraulic engine mount with four-chamber and three-fluid-channel,” Journal of Sound and Vibration. (2020).

[6] Juhee Lim, Yong Sok Jang, Hong Suk Chang, and Jong Chan Park et al., “Multi-objective genetic algorithm in reliability-based design optimization with sequential statistical modeling: an application to design of engine mounting,” Structural and Multidisciplinary Optimization. 61.3 (2020): 1253 - 1271.

[7] Alejandro Alvarado-Iniesta, Luis Gonzalo Guillen-Anaya, Luis Alberto Rodriguez-Picón, and Raul Ñeco-Caberta, “Multi-objective optimization of an engine mount design by means of memetic genetic programming and a local exploration approach,” Journal of Intelligent Manufacturing. 31.3 (2020): 19 - 32.

[8] S. Santhosh, V. Velmurugan, V. Paramasivam, and S. Thanikaikarasan, “Experimental investigation and comparative analysis of rubber engine mount vibration and noise characteristics,” Materials Today: Proceedings. 21 (2020): 638 - 642.

[9] M. Asadi Garmaroudi and J. Mosayebi, “Design and Optimization of Engine Mount,” International Journal on Energy Conversion IRECON, 5.4 (2017): 112 - 121.