Study on Test and Evaluation of Engine Mounting System Based on Transmission Force

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Abstract. The vibration isolation performance of the engine mounting system can be evaluated by the transmission force. The transmission force characteristics of engine mounting system are analyzed by simulation and test. The 6-DOF model of engine mounting system is established by ADAMS software. The results of modal parameters and transmission force of engine mounting system are obtained by simulation. The force sensor is made with resistance strain gauge. The sensor is calibrated by chassis dynamometer method. The transmission force of the engine mounting system is tested under the complete vehicle condition. The test results of transmission force and acceleration transmissibility are compared. It is proved that the transmission force is more suitable to evaluate the vibration isolation performance of the mounting system when the vehicle is running at medium and high speed.

Keywords: Engine mounting system; transmission force; vibration isolation evaluation; force sensor.

1. Introduction
Powertrain generally includes engine and its related accessories, transmission and clutch. It is the main source of vehicle vibration and noise. The vibration and noise level of powertrain can directly affect the vibration and noise level of the vehicle. The vibration transmitted from the powertrain to the body or chassis can be reduced by the engine mounting system effectively [1]. Therefore, the vibration isolation performance becomes the most important properties of the mounting system.

The vibration isolation performance evaluation of the engine mounting system is the key point for the test and optimization of the vehicle [2]. The most common method is using the acceleration transmissibility to evaluate the performance. This method has the characteristics of mature testing equipment, small size of sensor and easy arrangement [3]. However, the evaluation results are affected by road excitation in the road test. The evaluation results are more inaccurate when the vehicle speed is high [4].

The transmission force of the engine mounting system is the force that the powertrain transmits to the body or chassis through the engine mounting system. The vibration isolation performance of the mounting system can be directly evaluated by the transmission force. However, it is difficult to test the
transmission force of the engine mounting system. And few people evaluate the vibration isolation performance by the transmission force. The transmission force can be tested by the special sensor such as multi-axial load sensor or road spectrum collector [5]. However, the loads collected are all used for the part fatigue test, and few are used for the evaluation of the vibration isolation performance.

This paper explores the evaluation method vibration isolation performance of the engine mounting system by simulation and test. The force sensor is made in order to test the transmission force. The chassis dynamometer is used to calibrate the sensor. The evaluation method suitable for the vibration isolation performance of the mounting system is obtained by comparing the test results of the idle condition and the road driving condition.

2. Evaluation method of Mount vibration isolation performance

2.1. Vibration Isolation Principle of Engine Mounting System

Suppose the mass of the powertrain is m, the stiffness of the powertrain mounting system is k, the damping of the powertrain mounting system is c, the displacement of the powertrain is x, and the excitation of the engine is F₀sinωt. Then the force transmitted to the vehicle body is given as follows:

$$F_{T}(t) = kx + cx \& \approx F_{T} \sin(\omega t - \phi + \gamma)$$

where: γ is the phase angle between the transmission force $F_{T}$ and the displacement of the mounting system.

Because the phase between the force transmitted by the spring and the damping is 90°, the amplitude of $F_{T}$ is given as follows:

$$F_{T} = \sqrt{(kA)^2 + (c\omega A)^2} = kA\sqrt{1 + (2\xi\lambda)^2}$$

where: ζ is the viscous damping coefficient, λ is the frequency ratio, A is the vibration amplitude of the system.

The force transmissibility of mounting system can be expressed as follows:

$$\eta_{F} = \frac{F_{T}}{F_{0}} = \frac{\sqrt{1 + (2\xi\lambda)^2}}{\sqrt{(1 - \lambda^2)^2 + (2\xi\lambda)^2}}$$

2.2. Evaluation Method of Mounting System

Theoretical model evaluation: The theoretical model evaluation is used in the early design stage of the mounting system. The method includes the evaluation of the mounting part layout and modal parameters. The angle and distance between the elastic center line of the mounting parts and the torque axis are examined in the layout evaluation [6]. The modal distribution, frequency interval and modal energy decoupling rate of the mounting system are checked in the modal parameters evaluation [7].

Vibration acceleration of passive side of mounting system: The vibration acceleration of the passive side of the mounting system can reflect the actual vibration energy transferred from the powertrain to the body. In general, if the vibration acceleration of the passive side decreases, the vibration level of the cab decreases, too. This evaluation method often corresponds to the development target of the mounting system.

Evaluation of acceleration transmissibility: This method is based on the test results. By calculating the ratio of the mean square value of the acceleration of the active and passive sides of the mounting parts, the vibration isolation performance of the mounting system can be evaluated as follows:

$$T_a = \frac{a_p}{a_a} \times 100\%$$

or

$$T_{ab} = 20 \log \frac{a_p}{a_a}$$
where: asp is the root mean square value of the acceleration on the passive side, aa is the root mean square value of the acceleration on the active side.

Modal of mounting bracket: There will be local modal problem and complete vehicle vibration problem if the modal frequency of the mounting bracket is too low. It is generally required that the modal frequency of the mounting bracket should be above 480Hz, and the frequency interval should be more than 20%.

3. Modeling and Simulation of mounting system

The multi-body dynamic model of engine mounting system is established by ADAMS software, and the model is simulated and analyzed. The purpose is to analyze modal characteristics and the transmission force of the engine mounting system. The results can be used as a reference for the vehicle test.

3.1. Engine Excitation Analysis

The vibration excitation of the engine generally includes: reciprocating inertia force, pitching moment, roll moment, etc. For an in-line four-cylinder engine, the reciprocating inertia force can be expressed as follows:

$$F_p = 4m_t \omega^2 \lambda_p \cos 2\omega t$$  \hspace{1cm} (6)

where: $m_t$ is the equivalent mass of piston, $r$ is the radius of crank, $\omega$ is the angular velocity of crankshaft, $\lambda_p$ is the ratio of crank to connecting rod.

The pitching torque of engine can be expressed as follows:

$$M_p = M_g + M_y \approx \bar{M} (1 + 1.3 \sin 2\omega t)$$  \hspace{1cm} (7)

where: $M_g$ is the torque generated by combustion gas, $M_y$ is the reciprocating inertia torque, $\bar{M}$ is the average output torque of the engine.

The engine roll moment can be given as follows:

$$M_r = F_x \cdot e$$  \hspace{1cm} (8)

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. Modeling of Mounting System

Powertrain and vehicle body: Because the powertrain is connected to the body with a relatively soft rubber mount. The mass of the powertrain is smaller than that of the vehicle body. The vehicle body is temporarily replaced by an infinite mass earth element. The modal frequency of the elastic body of the powertrain is much higher than that of the rigid body of the mounting system. The powertrain can be simplified as a mass point [8]. The mass of the powertrain accessory parts, cooling water, lubricant are ignored. The mass of the powertrain is 165 kg and the moment of inertia is shown in Table 1.

| $I_{xx}$ (kg·m²) | $I_{yy}$ (kg·m²) | $I_{zz}$ (kg·m²) |
|-----------------|-----------------|-----------------|
| 9.2             | 3.7             | 8.6             |
| $I_{xy}$ (kg·m²) | $I_{xz}$ (kg·m²) | $I_{yz}$ (kg·m²) |
| 0.063           | 0.029           | 0.149           |

Mounting parts: Generally, rubber or hydraulic parts are selected as the mounting parts. The powertrain can be effectively supported in transverse, longitudinal and vertical directions by the mounting parts. In order to ensure the accuracy of dynamic simulation, the bushing elements are used to simulate the mounting parts. After the bushing elements are established, the stiffness and damping parameters of the mounting parts are set in the model. The parameters of mounting parts are shown in Table 2 and Table 3.
Table 2. Position parameters of Mounting parts

| Mounting position | X (mm) | Y (mm) | Z (mm) |
|-------------------|--------|--------|--------|
| Front mount       | 209    | 275    | -254   |
| Rear mount        | -356   | 313    | -229   |
| Left mount        | -41    | 579    | 111    |
| Right mount       | 24     | -349   | 206    |

Table 3. Stiffness parameters of Mounting parts

| Mounting position | Spindle stiffness (N/mm) |
|-------------------|--------------------------|
|                   | U | V | W |
| Front mount       | 192 | 50 | 78 |
| Rear mount        | 287 | 50 | 302 |
| Left mount        | 55  | 445 | 625 |
| Right mount       | 60  | 573 | 775 |

3.3. Modal Analysis of Mounting System

The simulation results are shown in Table 4. The results show that the modal frequency interval of the mounting system is more than 1.2 Hz, which is reasonable [9]. The idle speed of the engine is 800 rpm, and the corresponding second-order excitation frequency is 26.67 Hz. When the mounting system achieves the function of vibration isolation, the modal frequency of the mounting system must be lower than 18.86 Hz. The results show that the modal frequencies of each order are lower than 18.86 Hz, which can ensure the vibration isolation ability of the mounting system. The main problem of the mounting system is the energy distribution of the modes is not concentrated enough, except for the 14.45 Hz yaw mode. The energy distributions are less than 90%, which may lead to coupled vibration during the engine working process and reduce the vibration isolation performance of the mounting system.

Table 4. Results of Modal simulation

| Modal frequency (Hz) | Proportion of modal energy (%) |
|----------------------|--------------------------------|
|                      | X | Y | Z | Pitch | Roll | Yaw |
| 5.03                 | 80.26 | 0.08 | 0.74 | 0.2 | 12.49 | 7.25 |
| 6.84                 | 0.11 | 86.91 | 0.29 | 12.56 | 0.55 | 0.08 |
| 8.93                 | 0.91 | 3.27 | 84.5 | 10.52 | 1.76 | 0.05 |
| 11.34                | 0.31 | 9.63 | 13.63 | 75.89 | 0.54 | 0.02 |
| 12.53                | 15.02 | 0 | 0.85 | 0.85 | 84.38 | 1.24 |
| 14.45                | 6.95 | 0.14 | 0.02 | 0.02 | 0.27 | 95.01 |

3.4. Dynamic Simulation

The reciprocating inertia force, pitching moment and roll moment of the engine are added to the center of mass of the powertrain in the model. The simulation analysis is carried out, and the analysis conditions are shown in Table 5.

Table 5. Dynamic simulation condition

| Working condition | Speed (rpm) | Engine torque (N·m) |
|-------------------|-------------|---------------------|
| Idle              | 800         | 96                  |
| Maximum torque    | 5000        | 170                 |
| Maximum power     | 6500        | 155                 |
The simulation results are shown from Figure 1 to Figure 3.

![Graph](image1.png)

**Fig.1** Steady force of idle speed

![Graph](image2.png)

**Fig.2** Steady force of maximum torque

![Graph](image3.png)

**Fig.3** Steady force of maximum power

The steady transmission force amplitude of front mount is about 55 N at idle speed, 35 N at maximum torque and 30 N at maximum power. The amplitude of the force decreases with the increase of the engine speed. Thus, it can be proved that the idle condition is the most destructive case for the mounting parts. And it also shows that the rationality of idle condition is the first consideration in the mount design.

4. Design and calibration of force sensor

4.1. Design and Manufacture of Force Sensor

The vibration generated by the powertrain is transmitted to the body or subframe through the mounting parts. Therefore, the sensor must be placed in the force transmission path when testing the transmission force.

Generally, there are 2-4 connection points between the mount and the powertrain, the body or the subframe in order to ensure the connection strength and reliability of the mounting parts. Because there are too many connection points, the transmission force of each point is scattered. The position is not suitable for the arrangement of force sensors. In this paper, the resistance strain gauge is arranged on the bolt connecting the mounting rubber and mounting bracket on the body side. This position is the only way for the powertrain to transfer the force to the body or subframe. And there is only one connecting bolt at this position, so it is unique and suitable for the arrangement of force sensors [10].
Through milling the middle part of the bolt, arranging the resistance strain gauge and packaging the patch, the mounting force sensor is made, as shown in Figure 4.

![Figure 4: Force sensor](image)

**Fig.4 Force sensor**

### 4.2. Sensor Calibration

The test system of the transmission force includes: mounting transmission force test part (strain gauge, bridge, amplifier, filter), acceleration test part (acceleration sensor, amplifier, filter), data collector, analysis program, display, storage, as shown in Figure 5.

![Figure 5: Composition diagram of test system](image)

**Fig.5 Composition diagram of test system**

Because the signal collected by the test system is a voltage signal. In order to get the test result of force, it is necessary to obtain the numerical relationship between the transmission force and the test signal. The calibration process is shown in Figure 6.

![Figure 6: Calibration process of force sensor](image)

**Fig.6 Calibration process of force sensor**
Firstly, the vehicle is tested by chassis dynamometer. The vehicle is a gasoline vehicle with 1.5L in-line four-cylinder engine. The vehicle test speed is 30 km/h. The engine load is full load. The transmission gear is second gear, as shown in Figure 7.

The driving force curve of the driving wheel is obtained after the test. According to the relationship between engine output torque and driving force of the driving wheel as follow:

\[ F_i = \frac{T_q i \eta_i}{r} \]  

(9)

The engine output torque is calculated. According to the signal ratio of the mounting force sensor in the test results, the proportional relationship between the front and rear mounting transmission forces is obtained. Based on the balance relationship between the engine output torque and mounting counter torque, the mounting force curve is calculated. Finally, the scatter diagram is drawn with the amplitude of the transmission force as the ordinate and the amplitude of the test signal as the abscissa. And the linear fitting is carried out to obtain the proportional relationship between the transmission force and the test signal as shown in Figure 8 and Figure 9.

Fig.7 Sensor calibration test

Fig.8 Calibration curve of front mount

Fig.9 Calibration curve of rear mount
5. Vehicle test and evaluation
The actual data of the transmission force and acceleration of the mounting system can be obtained through the vehicle test, which can provide the basis for the vibration isolation performance evaluation of the mounting system.

The test equipment includes: force sensors, acceleration sensors, laptop, dynamic strain gauge, signal conditioning instrument, etc. The force sensors are arranged on the front and rear mount. The acceleration sensors are arranged on the active and passive sides of the front and rear mount.

The test conditions include idle condition and road driving condition. The test data of each condition are measured under the stable engine speed as shown in Table 6.

| Condition | Engine speed(rpm) |
|-----------|-------------------|
| Idle      | 1000 1500 2000 2500 3000 |
| 2\textsuperscript{nd} gear | 1000 1500 2000 2500 3000 |
| 3\textsuperscript{rd} gear | 1000 1500 2000 2500 3000 |

The test results of the front mount are shown is Figure 10. The test results of the rear mount are shown is Figure 11. The change trend of the transmission force of is similar to the acceleration transmissibility under idle condition. With the increase of engine speed, the vibration transmissibility of the mounting system decreases, the vibration isolation performance of the mounting system increases, and the corresponding mounting transmission force decreases, which is close to the trend of simulation results.

The change trend of transmission force and acceleration transmissibility is basically the same under the 2\textsuperscript{nd} gear constant speed driving condition. Under the 3\textsuperscript{rd} gear constant speed driving condition, the trend of transmission force and acceleration transmissibility is basically the same before 2000 rpm. However, the acceleration transmissibility increases obviously after 2000 rpm. This is due to the increase of vehicle speed. When the vehicle speed is high, the road excitation to the vehicle body increases obviously. And the vibration acceleration of the passive side of the engine mounting parts increases greatly. The acceleration transmissibility is reduced. But the influence on the transmission force is not very obvious.

The transmission force amplitude under road driving condition is larger than that under idle condition. The change trend of transmission force is basically the same. Under the 3\textsuperscript{rd} gear constant speed driving condition, the transmission force of the rear mount is affected by the increase of road excitation after 2500 rpm. The change trend of the acceleration transmissibility is relatively consistent under the idle condition and the 2\textsuperscript{nd} gear driving condition. However, the acceleration transmissibility increases significantly after 2000 rpm under the 3\textsuperscript{rd} gear constant speed driving condition. The results show that the disturbance of road excitation on the acceleration transmissibility is very obvious at medium and high speed.

Through the above analysis, it can be determined that the evaluation method of the engine mounting transmission force and acceleration transmissibility is more consistent when there is no external vibration interference or the interference is small. However, it is more reliable to evaluate the vibration isolation performance by the transmission force when there is a large external interference. The transmission force is more suitable to evaluate the vibration isolation performance of the engine mounting system under road driving conditions.
6. Conclusion
The model of engine mounting system is established by ADAMS software. The theoretical evaluation of the sample vehicle mounting system is carried out. The modal energy decoupling rate of sample vehicle engine mounting system is poor. The vibration coupling problem is easy to occur. The simulation results of mounting transmission force under three typical conditions are obtained. The results show that the amplitude of steady transmission force decreases with the increase of engine speed, which is consistent with the test results.

The mounting force sensor is made by arranging strain gauges on the mounting connecting bolts in order to test the transmission force of the mounting system. The proportional relationship between the transmission force of the signal amplitude of the sensor is calculated to realize the calibration of the sensor based on the test results of the chassis dynamometer.

The mounting transmission force and the acceleration transmissibility are obtained through the idle and road test. The evaluation results of mounting transmission force are basically consistent with the results of acceleration transmissibility under idle and low-speed driving conditions. The increase of the acceleration transmissibility of the front and rear mount is more than 90% compared with the test results at 1000rpm. While the increase of mounting transmission force is within 28%. This shows that the evaluation method of mounting transmission force is more stable than that of acceleration transmissibility when the vehicle is running at medium and high speed. It is more suitable for the vibration isolation performance evaluation of the engine mounting system under road driving conditions.
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References
[1] Prasad Mungi and Thanikaikarasan S., “Experimental investigation and material characterization of rubber engine mount for automobile application,” Materials Today: Proceedings, vol. 33(7), pp. 4139-4141, January 2020.
[2] TAO, J. S., LIU, G. R. and LAM, K. Y., “Design Optimization of Marine Engine-mount System,” Journal of Sound and Vibration, vol. 235, pp. 477-494, 2000.
[3] M Ravi Teja Reddy, V Jayakumar, and D Santhoshkumar, et al., “Vibration Testing of Novel Engine Mount: Technical Note,” International Journal of Vehicle Structures & Systems, vol. 10(6), pp. 417-419, January 2018.
[4] Jian-Da Wu, Fu-Cheng Su and Wen-Kung Tseng, “Vibration isolation for engine mount systems using an active hybrid robust controller,” Int. J. of Vehicle Noise and Vibration, vol. 1(3/4), pp. 251-264, June 2005.
[5] U.E. Ozturk and L. Ucar, “Road Load Identification and Accelerated Life Testing of Engine Mounts,” Experimental Techniques, vol.41(3), pp. 267-277, June 2017.
[6] Rahime Naseri, Abdolreza Ohadi, and Vahid Fakhari, et al., “Optimal characteristics determination of engine mounting system using TRA mode decoupling with emphasis on frequency responses,” Journal of Theoretical and Applied Vibration and Acoustics, vol. 3(2), pp. 111-126, July 2017.
[7] Bernhard Angrosch, Manfred Plöchl and Werner Reinalter, “Mode decoupling concepts of an engine mount system for practical application,” Proceedings of the Institution of Mechanical Engineers, Part K: Journal of Multi-body Dynamics, vol. 229(4), pp. 331-343, December 2015.
[8] Jianxin Xie, Xiaole Wang and Chao Liu, “Simulation and Optimization Design Based on Adams Engine Mounting System,” Applied Mechanics and Materials, vol. 3629, pp. 567-572, December 2014.
[9] Zhang Huijie, Hao Huirong and Guo Zhiping, “Analysis And Optimization Design of Six Degrees of Freedom Engine Mount System,” vol. 32(5), pp. 801-810, October 2019.
[10] Deng Zhaoxue, Yang Qinghua, and Cai Qiang, et al., “Design and Test of a Magneto-Rheological Mount Applied to Start/Stop Mode of Vehicle Powertrains,” vol. 55(1), pp.56-66, January 2021.