In order to get the excitation forces of the vehicle powertrain, a six-degree-of-freedom model of the powertrain mounting system is established. Two different identification methods are presented. Through the test of mount dynamic response, the powertrain excitations identified through these two methods are compared. The results show that the powertrain excitations identified through two methods are basically the same, since the mount accelerations at the body side are really small. The identified excitation is verified by comparing the deformation of torque strut. The robustness analysis of the mount acceleration phase affecting the identification results is also proposed. Improving the accuracy of phase in test input is helpful to improve the accuracy of identification results.

1. Introduction

The automotive NVH (Noise Vibration & Harshness) attracts much attention and becomes a problem that needs to be overcome in the vehicle design. To solve the NVH problem, the source of the vibration should be identified first. The powertrain suspension system consists of a powertrain composed of an engine and a transmission and several mounts [1]. Mount is a typical type of rubber vibration isolator, including pure rubber mount, hydraulic mount, magnetorheological mount, and other types. The main functions of the mount include supporting the powertrain, isolating the vibration of the powertrain to the vehicle, and reducing the vibration of the powertrain caused by toughness of road [2]. The mount materials used in this work are all filled rubber. Rubber has significant variability in high ductility, damping, and stiffness. It is widely used in vibration isolation in the industry, and the effect is obvious. In order to obtain sufficient compression stiffness, shear strength, and good fatigue resistance, it is often necessary to use additives such as carbon black to fill the rubber. The powertrain excitation force is one of the main factors causing vibration and structural noise in the vehicle and is also one of the important input parameters for vehicle NVH calculation [1]. Therefore, no matter in the engineering applications or the improvement and development of basic theoretical research, the research on the identification method of vehicle powertrain excitation force has its necessity and urgency.

Since the force sensor is affected by factors such as tooling, sensor arrangement, and engine compartment space, it is not easy to test the amplitudes of excitation forces directly in practical engineering applications. In a vehicle design process, the theoretical excitation force is usually calculated based on engine dynamics. Taking the conventional internal combustion engine as an example, the theoretical values of powertrain excitation forces are calculated from the engine structural parameters, the combustion pressure, the transmission ratio, etc., or the theoretical forces could be simulated by the simulation software. However, due to the actual factors such as the manufacturing error of the engine components, the pressure change in the cylinder, and the gear transmission efficiency of the transmission,
there will be a large error between the theoretical value and the actual excitation force value.

To obtain the actual powertrain excitation force value, it needs to be identified by the indirect test method, that is, by measuring the dynamic response of each powertrain mount (such as acceleration) and the mount stiffness of the system. In the current research, there are various methods for identifying the excitation forces or dynamic loads. Each method has its applicable range. The methods used to identify the excitation forces of the powertrain can be divided into the following categories: the direct method, the regular method, and the acceleration phase correction method.

1.1. The Direct Method. The direct method refers to solving inverse problems only through mathematical or physical models without considering the nonlinearity of the system, using filters or other correction techniques for test data. The direct method was performed in the time domain or the frequency domain alone. The common methods were the frequency domain inverse Fourier transform method, the time domain inverse modal analysis method, and the time domain subsystem matrix method [3]. It was also possible to indirectly identify the forces by calculating the displacement transmissibility of the vibration system. This identification method was based on the relationship between the displacement transmissibility and the force transmissibility established in the frequency domain [4–6]. The differential equation of motion in the frequency domain was transformed into integral equation in the time domain by combining the time domain with the frequency domain. Then, the excitations in both time domain and frequency domains were identified [7].

In this method, it does not consider the influence of the calculation process error, input parameter error, system model error, and other factors on identification results. The accuracy of identification results is not so high. However, at present, the direct method is still widely used in industry since it has a clear physical meaning and a simple calculation process.

1.2. The Regularization Method. The singular value decomposition [8] was used in the regularization method. Tikhonov regularization method or least square method is used to reduce the error of the calculation process or the input parameter error during measurement.

In order to solve the error caused by the matrix inversion and the ill-conditioned matrix during the identification and calculation process, Tikhonov proposed the Tikhonov regularization method which can effectively solve this problem, and it is also a regularization method widely used at present [9, 10]. In addition, other regularization methods such as the least squares method, truncated SVD-based LS method, and Tikhonov filter-based LS method were used; each method has different application scopes [11].

In order to treat the noise error in the test data, the coherence value of the test data with noise and other interference factors were used to judge the accuracy of the test. The lower the coherence is, the more reliable the result is. The noise was reduced by the multiple averaging method or the singular value decomposition technique. Singular value decomposition [9] (SVD) was a commonly used method. According to the different criteria for discarding singular values, it can be divided into the following categories: discarding the minimum singular value method, determining the singular value numbers according to condition numbers, and the maximum singular value percentage standard method. However, the abandoned singular value contains part of the information of the original signal. The contribution of the abandoned singular value should be considered before making a trade-off. The unreasonable method of abandoning the singular value may lead to errors in the calculation results.

1.3. The Acceleration Phase Correction Method. The acceleration phase correction method is a method for performing signal correction processing on the error (the maximum error is up to 180 degrees) generated during the A/D conversion process of the acceleration phase in the test.

Tao et al. proposed a method to treat the phase error [12]. The phase of the excitation forces was set to zero, and then phases of the measured points were reconstructed through the test data of phase difference. But in fact, the phases of each excitation force are not zero, and the true phase of one of the excitation forces was needed to know when reconstructing the actual phase. The true phase needs to be obtained through signal correction processing.

In recent years, most of the methods to correct the tested acceleration phase were based on signal processing technology, which had some defects, such as complex derivation, large amount of calculation, and optimization of only one frequency at one time. Therefore, the acceleration phase correction method has some limitations in practical engineering.

In summary, at present, the direct method is widely used in engineering practice, with high calculation efficiency and simple calculation process. However, its calculation accuracy is limited. The regularization method has higher calculation accuracy, but only some discrete frequency points can be calculated. The calculation efficiency is low when calculating the full frequency range. The acceleration phase correction method still needs to obtain the true phase of a certain test point, so the excitation identification error caused by the phase error cannot be avoided. The two calculation methods presented in this paper are both direct method.

In the previous research work, the influence of the mount acceleration phase on excitation identification results is rarely discussed. In this work, a detailed calculation and analysis of this research was proposed in the robustness analysis.

The main content of this paper consists of two parts:

1) The vibration accelerations of the mount connecting points at the engine side and at the body side were tested. The excitation forces and moments of the powertrain under steady and acceleration conditions were identified and compared. The identification
methods were carried out under two conditions; considering the mount accelerations at the body side or not. Using the identified excitation forces and moments, the deformation of the torque strut was calculated. Compared with the measured torque strut deformation value, the calculation error of the identification method was analyzed.

(2) The influence of mount acceleration phases on excitation force identification results was analyzed. To improve the identification accuracy of excitation forces, the acceleration phase should be controlled strictly.

2. Two Kinds of Excitation Force Identification Methods

A six-degree-of-freedom dynamic model of the powertrain mounting system is established. As shown in Figure 1, the vibration accelerations of the connecting points of the mount at the engine side and at the body side are measured. According to whether the acceleration response of the mount at the body side is considered, the identification methods can be divided into two kinds: direct method and mount deformation method.

2.1. Direct Method. The two sides of the engine mount are connected to the engine and vehicle body, respectively. If the body is considered as a rigid body, the body is equivalent to connected to the static ground. Then, the mount acceleration at the body side is zero. If the mount acceleration at the engine side is \(a^e_i(f)\) \((i = 1, 2, 3)\), the acceleration of the powertrain CG (center of gravity) can be obtained by the equation as follows:

\[
a_{cg}(f) = E_i^{-1} \cdot a^e_i(f),
\]

where \(E_i\) is the transformation matrix of the powertrain CG to the mount point. The expression of \(E_i\) can refer to reference [12].

Since the position matrix \(E_i\) is a \(3 \times 6\) matrix, the inverse matrix cannot be directly obtained. Therefore, a pseudo-inverse calculation formula is needed. In order to reduce the calculation error, a \(9 \times 6\) matrix is constructed (the number of rows is greater than or equal to the number of columns), its inverse matrix can be written as follows:

\[
E_i^{-1} = \left(\begin{bmatrix} E_1 & E_2 & E_3 \end{bmatrix} \cdot \begin{bmatrix} E_1 & E_2 & E_3 \end{bmatrix}^T\right)^{-1} \begin{bmatrix} E_1 & E_2 & E_3 \end{bmatrix}.
\]

From \((\omega^2M + K) \cdot a_{cg}(f) = F_{cg}(f)\), the excitation forces acting on the centroid of the powertrain can be obtained as follows:

\[
F_{cg}(f) = \left(M - \frac{1}{\omega^2}K\right) \cdot a_{cg}(f), \quad \omega = 2\pi f,
\]

where \(M\) and \(K\) are the mass and stiffness matrices of the powertrain mounting system, respectively. The specific expressions are given in reference [11]. \(\omega\) is the intrinsic circular frequency.

2.2. Mount Deformation Method. In this method, the elasticity of the body is taken into consideration, and then the acceleration of the mount at the body side is not zero. The two ends of the mount are connected to the engine and the body, respectively. Through the acceleration of the mount at the body side and at the engine side, the mount deformation in the frequency domain is given by

\[
X_i(f) = \frac{a_i^b(f) - a_i^e(f)}{-\omega^2},
\]

where \(a_i^b(f)\) is the acceleration of the mount \(i\) at the body side.

Through the mount deformation, the dynamic reaction force of each mount in their local coordinate is given by

\[
F_i(f) = k_i \cdot X_i(f),
\]

where \(k_i\) is the stiffness matrix of the mount \(i\) in its local coordinate system [13].

According to the position matrix \(E_i\), the dynamic reaction forces in the local coordinate system of mounts can be converted to the powertrain CG. The inertial force at the powertrain CG should also be considered. The excitation forces \(F_{cg}(f)\) at the powertrain CG in the PCS are as follows:

\[
F_{cg}(f) = \sum_{i=1}^{3} E_i^T \cdot F_i(f) + (-M) \cdot a_{cg}(f).
\]

2.3. Identification Process. The direct method needs to test the mount acceleration at the engine side. In addition to the mount acceleration at the engine side, the mount deformation method requires the mount acceleration at the body side as input. The implementation processes of the two methods are shown in Figure 2.
3. **Identification Result of Excitation Forces**

3.1. **Input Parameters.** Based on the principle of the moment of inertia test [14], through the moment of inertia test bench (in Figure 3) and after multiple tests, the average value of the powertrain is 172.43 kg. The moment of inertia parameters of the powertrain in the powertrain CG coordinate system (PCS) is shown in Table 1. The origin of the PCS is located at the X powertrain CG, the positive Z direction points to the rear of the vehicle, and the direction follows the right-hand rule.

Through the inertia test bench, based on the principle of the moment of inertia test [13] and after multiple tests, the average value of the powertrain is 172.43 kg. The rotation parameters of the powertrain in the centroid coordinate system are shown in Table 1.

The left and right mounts are pure rubber mounts, and the rear mount is a torque strut. Both ends of the torque strut are made of pure rubber bushes. The physical diagram of the torque strut is shown in Figure 4(b). The mount is fixed to the MTS elastomeric dynamic performance test bench (in Figure 5) by design tooling. The excitation amplitude is ±1.0 mm at a test frequency of 1–50 Hz and ±0.1 mm at a test frequency of 50–500 Hz. When testing the X and Y direction stiffness, the Z direction is preloaded. The quasistatic forces and dynamic forces are applied to the mount, respectively, the deformations of the mount are tested, and the force-displacement curve is obtained. The static stiffness values of each mount in its local coordinate system are fitted as shown in Table 2.

The coordinates of the installation position of each mount, the powertrain CG, and the midpoint of crankshaft spindle neck in the automotive coordinate system (ACS) are shown in Table 3.

During the test, a LMS test equipment, six three-direction acceleration sensors, and a laser displacement sensor...
the body side are smaller than those at the engine side. The acceleration amplitudes increase as the rotation speed increases. The maximum acceleration at the body side is 0.80 m/s² which is in the Y direction of the right mount at 4000 rpm.

At 3000 rpm and 4000 rpm, the maximum acceleration amplitude of the mount at the body side is 0.33 m/s² and 0.80 m/s². The reason is that the high speed rotation of the axles in the transmission causes a greater impact on the meshing of gears. Then, a large excitation moment around the X direction produces a large force in the Y direction.

Table 5 shows the transmissibility ratio between the body side and the engine side at the 2nd order. From the table, the maximum transmissibility ratio is 24%. Except the Z direction of the left mount in the 1000 rpm and the X direction of the right mount in the 1000 rpm, the vibration isolation ratio in all other directions exceeds 90%. The vibration isolation performance of mounts is good.

The acceleration amplitudes of the mount at the body side and at the engine side and the deformation of the torque strut are measured under the condition of Wide Open Throttle in 2nd gear. Because of the fluctuation of engine speed in the WOT condition, the signal collected at this time does not satisfy the requirement of Fourier transform for signal stationary and is a nonstationary signal which is not suitable for the conventional FFT (fast Fourier transform) spectrum analysis method. Using the order tracking method, the acceleration signal is resampled at equal angular intervals in the angular domain, and the unsteady signal in the time domain is transformed into the steady signal in the angular domain. The curve of the amplitudes of the second-order accelerations of the mount with the speed is obtained, as shown in Figure 6.

It can be seen from the figure that the acceleration amplitude of the mount at the engine side increases with the increase of rotational speed, while that of the left mount at the body side is smaller, and that of the right mount is approximately zero.

3.2. Excitation Force Identification Result under Steady-State Conditions. The second-order excitation force and moment acting on the center of mass at each stable speed condition are calculated by direct method and mounting
deformation method, respectively, as shown in Tables 6 and 7. By comparing these two tables, it can be found that considering the acceleration of the mount at the body side or not has little effect on the identification results of excitation forces. The reason is that the accelerations of the mount at the body side are small and can be ignored. At the same time, it shows that the vibration isolation performance of each mount is good. The magnitude of the

| Engine speed (rpm) | Position  | Left mount | Right mount | Torque strut |
|--------------------|-----------|------------|-------------|--------------|
|                    |           | X  | Y  | Z  | X  | Y  | Z  | X  | Y  | Z  |
| 1000               | Engine side | 0.54 | 0.22 | 0.25 | 0.20 | 0.49 | 1.60 | 0.52 | 0.37 | 1.02 |
|                    | Body side  | 0.01 | 0.01 | 0.06 | 0.03 | 0.03 | 0.03 | 0.01 | 0.02 | 0.02 |
| 2000               | Engine side | 0.70 | 0.72 | 1.01 | 0.59 | 1.92 | 5.47 | 1.01 | 1.08 | 3.10 |
|                    | Body side  | 0.02 | 0.05 | 0.06 | 0.04 | 0.06 | 0.08 | 0.01 | 0.02 | 0.06 |
| 3000               | Engine side | 1.57 | 1.27 | 2.86 | 2.19 | 4.92 | 13.36 | 2.66 | 2.50 | 6.40 |
|                    | Body side  | 0.02 | 0.12 | 0.20 | 0.01 | 0.33 | 0.01 | 0.00 | 0.02 | 0.03 |
| 4000               | Engine side | 2.81 | 1.79 | 10.33 | 2.95 | 10.96 | 23.98 | 2.39 | 2.53 | 11.45 |
|                    | Body side  | 0.09 | 0.08 | 0.26 | 0.01 | 0.80 | 0.02 | 0.02 | 0.10 | 0.11 |

Table 5: The second-order transmissibility ratio (%).

| Engine speed (rpm) | Left mount | Right mount | Torque strut |
|-------------------|------------|-------------|--------------|
|                   | X  | Y  | Z  | X  | Y  | Z  | X  | Y  | Z  |
| 1000              | 1.85 | 4.55 | 24 | 15 | 6.12 | 1.88 | 1.92 | 5.4 | 1.96 |
| 2000              | 2.86 | 6.94 | 5.94 | 6.78 | 3.13 | 1.46 | 0.99 | 1.85 | 1.94 |
| 3000              | 1.27 | 9.45 | 6.99 | 0.46 | 6.70 | 0.07 | 0 | 0.80 | 0.47 |
| 4000              | 3.20 | 4.47 | 2.52 | 0.34 | 7.30 | 0.08 | 0.84 | 3.95 | 0.96 |

Figure 6: The acceleration amplitude of each mount at the body side and the engine side under the second gear WOT condition. (a) Left mount. (b) Right mount.

Table 6: Second-order excitation forces identified by the direct method (in the powertrain centroid coordinate system).

| Engine speed (rpm) | Fx (N) | Fy (N) | Fz (N) | Mx (Nm) | My (Nm) | Mz (Nm) |
|--------------------|--------|--------|--------|---------|---------|---------|
| 1000               | 8.56   | 19.25  | 160.10 | 13.41   | 10.71   | 1.22    |
| 2000               | 30.27  | 83.83  | 585.58 | 55.38   | 22.86   | 4.82    |
| 3000               | 45.84  | 195.18 | 1412.51| 140.41  | 79.27   | 21.08   |
| 4000               | 208.17 | 466.16 | 2840.66| 201.51  | 95.36   | 10.35   |

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second-order excitation forces and moments increases with the increase of rotational speed.

Compared with the excitation force along the $Z$ direction of the PCS, the excitation forces in the $X$ and $Y$ directions are smaller because the balance mechanisms of the four-cylinder engine are well designed. In theory, after balance, the forces acting on the engine cylinder block by crank-connecting rod mechanism in the $X$ and $Y$ directions are zero, and the identified excitation forces in the $X$ and $Y$ directions are mainly from the impact of helical gears in transmission.

Table 7: Second-order excitation forces identified by the mount deformation method (in the powertrain centroid coordinate system).

| Engine speed (rpm) | $F_x$ (N) | $F_y$ (N) | $F_z$ (N) | $M_x$ (Nm) | $M_y$ (Nm) | $M_z$ (Nm) |
|-------------------|-----------|-----------|-----------|------------|------------|------------|
| 1000              | 8.68      | 19.21     | 159.76    | 13.73      | 10.79      | 1.20       |
| 2000              | 30.35     | 83.71     | 585.35    | 55.60      | 22.89      | 4.85       |
| 3000              | 45.87     | 195.00    | 1412.10   | 140.52     | 79.38      | 21.12      |
| 4000              | 208.21    | 465.74    | 2839.62   | 201.75     | 95.50      | 10.34      |

Figure 7: Excitation forces at the center of mass of powertrain identified under stable rotational speed (in the PCS). (a). Excitation forces in the time domain (1000 rpm). (b). Excitation moments in the time domain (1000 rpm). (c) Excitation forces in the time domain (4000 rpm). (d) Excitation moments in the time domain (4000 rpm).
during meshing. The Z direction force is larger, and the main excitation comes from the second-order reciprocating inertia force produced by the reciprocating inertia mass of crank-connecting rod mechanism moving along the cylinder. The excitation of powertrain around the Y direction (around the crankshaft direction) comes from inertia force of crank-connecting rod mechanism and combustion pressure of gas in the cylinder. The excitation in X direction comes from the reciprocating motion of crank-connecting rod mechanism and the impact of gear meshing in transmission. Theoretically, the excitation moments of the powertrain around the Z-axis are zero, and the moments identified by 1000–4000 rpm in the table are relatively small, which corresponds to the theoretical calculation values.

The dynamic amplitudes of the excitation forces and moments acting on the centroid of the powertrain along the direction of the centroid coordinate system of the powertrain are obtained by superposing the orders of the excitation forces in the time domain at the stable speed. The curves are shown in Figure 7.

It can be seen from Figures 6(a) and 6(c) that the amplitudes of X and Y directions are smaller than that of Z direction, which is similar to the amplitudes of the second-order excitation forces. Because of the inertia force of the crankshaft, the amplitude of the excitation forces along the X, Y, and Z directions increases with the increase of the rotational speed. It can be seen from Figures 6(b) and 6(d) that the moment around the Y direction is bigger at 1000 rpm and the moment around the X direction is bigger at 4000 rpm. It shows that the moment around the X direction increases faster than the moment around the Y direction with the increase of rotational speed and the inertia force produced by the powertrain in the process of crankshaft rotation has a greater impact on the amplitude of the excitation moment around the X-axis. Because the 0th-order frequency is zero, the mounting deformation cannot be calculated by the acceleration of the moment at the body side and at the engine side. The identification of 0th-order excitation moments requires the deformation of each mount measured by the displacement sensor. Therefore, the excitation forces and moment identified in this chapter do not contain 0th-order excitation forces and moments.

3.3. Excitation Force Identification Results under the Second Gear WOT Condition. According to the acceleration of each mount in Figure 6, the curves of the excitation forces and moment acting on the centroid of the powertrain with the rotational speed calculated by the mount deformation method and the direct method are shown in Figure 8. As can be seen from Figure 8, the excitation forces and moments identified by the two methods are basically the same. The identified excitation force along the Z direction in PCS is obviously greater than that in X and Y directions. At low speed, the moment around the Y direction is greater than that around X direction. At high speed, the moment around the X direction is larger, which is consistent with the identification result under steady-state condition. This is due to the increase of engine speed and inertia force of each mechanism. Under the WOT condition, the acceleration amplitudes of the mount at the body side are small, which has little effect on the identification results of excitation forces. It also shows that the vibration isolation performance of each mount is good.

Through the deformation method, the forces and moments of the powertrain CG are identified. Translating these forces and moments from the PCS to the midpoint of the main shaft of the engine crankshaft, that is, the midpoint of the second and third cranks, and obtaining the torque acting on the midpoint of the crankshaft main shaft with the change of the rotational speed, as shown in Figure 9. It can be seen from the figure that the moment around the Y...
direction (that is, the direction around the crankshaft axis) is obviously larger than the moment around the other two directions. This is because the output torque of the engine is provided by the moment around the crankshaft. It can be seen from the figure that the moments around the crankshaft are smaller at 3500 rpm and 4600 rpm because the 0th-order excitation forces cannot be identified, resulting in the smaller amplitude of the moment than the actual value. Since the magnitude of the forces does not vary with the action point, the force acting on the middle point of the crankshaft main journal is the same as the excitation force acting on the centroid of the powertrain.

The second-order displacements of the torque strut calculated by the second-order excitation forces under the second-order WOT condition identified by the deformation method are shown in Figure 10. Compared with the measured value, the displacement of the torque strut is in good agreement at high speed. At low speed, the maximum displacement of the torque strut is 0.057 mm, the maximum calculated value is 0.048 mm, and the relative error is 16%.

4. Robustness Analysis of Identification Results of Excitation Forces

In the process of testing, the absolute phase errors of the accelerations are very large, and the maximum error can reach 90 degrees. In order to analyze the influence of acceleration phase of the mount at the engine side on the identification of excitation forces, taking 2000 rpm as an example, eight groups of experiments are carried out to determine the acceleration phases of three mounts in all directions at \((-\pi, \pi)\). The identification results compared with the nominal value (the excitation forces identified by direct method under steady state condition) are shown in
Table 8: The identified forces under the random values of the mount acceleration phase at the engine side.

| Number | $F_x$ (N) | $F_y$ (N) | $F_z$ (N) | $M_x$ (Nm) | $M_y$ (Nm) | $M_z$ (Nm) |
|--------|-----------|-----------|-----------|------------|------------|------------|
| 1      | 39.52     | 146.82    | 540.16    | 44.18      | 11.82      | 6.57       |
| 2      | 108.60    | 156.10    | 553.80    | 49.02      | 26.59      | 8.82       |
| 3      | 83.65     | 199.03    | 468.38    | 52.44      | 35.92      | 17.07      |
| 4      | 64.00     | 182.43    | 488.85    | 52.49      | 26.62      | 20.99      |
| 5      | 30.89     | 143.92    | 287.49    | 53.65      | 42.27      | 2.63       |
| 6      | 73.82     | 109.58    | 422.27    | 49.95      | 23.99      | 16.01      |
| 7      | 102.96    | 156.59    | 479.99    | 59.74      | 19.73      | 4.73       |
| 8      | 16.82     | 153.77    | 396.98    | 67.77      | 46.51      | 10.42      |
| Nominal value | 30.27     | 83.83     | 585.58    | 55.38      | 22.86      | 4.82       |
| Minimum value | 16.82     | 109.58    | 287.49    | 44.18      | 11.82      | 2.63       |
| Maximum value | 108.60    | 199.03    | 553.80    | 67.77      | 46.51      | 20.99      |
| Minimum percentage (%) | −44.45   | 30.72     | −50.91    | −20.22     | −48.31     | −45.51     |
| Maximum percentage (%) | 258.78   | 137.42    | 5.43      | 22.37      | 103.47     | 335.38     |

Table 8. It can be seen from the table that the influence of the phase change of acceleration on the excitation forces and moments is more than 15%, and the maximum is 335%. Therefore, improving the accuracy of phase in test input is helpful to improve the accuracy of identification results of excitation forces.

5. Conclusions

In this paper, the linear dynamics analysis model of the powertrain mounting system is established. The powertrain excitation forces are identified and calculated by the method based on the mount deformation and the method based on the accelerations of the mount at the engine side. The identification results of the two methods are compared and verified. Finally, the robustness analysis of the influence of the mount acceleration phase at the engine side on the identification results is carried out. The study shows that

(1) Because the acceleration amplitude of the mount at the body side is small, the excitation forces and moments of the powertrain identified by the direct method and the mount deformation method are basically the same, which means that the vibration isolation performance of the mounting system is good.

(2) The mount acceleration phase has a great influence on the identification results. Therefore, improving the accuracy of phase in test input is helpful to improve the accuracy of identification results of excitation forces.

The research on the excitation forces identification helps to improve the NVH performance of various types of mechanical vibration systems and has broad application prospects in two-cylinder and three-cylinder downsized engine vehicles and new energy vehicles.

Data Availability

The test data used to support the findings of this study are available from the corresponding author upon request.

Conflicts of Interest

All the authors do not have any possible conflicts of interest.

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