

# Study and Application of the Transfer Characteristics of the Mount under the Testing Correlation

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*Abstract:* -The vibration from different moving structure in car traveling is interactional, judging interactional relations among structures reasonably is premise to analyze the mount system characteristics. In paper, the transfer characteristics law with the acceleration being exciting are analyzed and summarized, the mount transfer characteristic of the car in driving is derived, and combining with experiment, calculation flow of the mount isolation characteristic is designed, experiments with independent power system excitation test, independent road excitation test, driving test in steady speed are proposed to obtain interference components, common frequency components and the corresponding proportion between vibration source; and finally, by the actual driving testing and data processing, the time-domain isolation characteristics of the power system mounts are obtained. The proposed method can effectively improve the evaluation precision of the mount system and transfer characteristics it is a certain engineering application value.

*Key-words:* -Vibration; Mount; Relevant; Isolation rate; Acceleration transfer characteristics

## 1 Introduction

Vehicle vibration is one of the crucial indicators that affect driver's ride comfort. The analysis of the vibration transmission path and the source of vibration is an essential prerequisite<sup>[1-3]</sup> solving such problems.

The connection of automotive structural parts mostly adopts mount parts to insulate or reduce vibration excitation. The input direction of mount parts should be reasonably judged, which is the key to analyze the structure performance. For example, a car in motion analyzes mount parts between power system and car frame through acquiring real vehicle data. Meanwhile, the engine, as a source of vibration, influences the frame that is also motivated by road and drive axle to cause vibration which affects the engine. If power system is purely considered as the vibration excitation input, the

vibration isolation rate computed will be obviously higher, therefore, the recognition of vibration source and the transitive property analysis of mount parts will be inevitably impacted.

Literature [4] filtrates the jamming of other vibration sources among response points with partial coherence analysis method. It only retains responses caused by the main vibration source, but is restricted by the sequencing relation of each source. Literature [5] conducts relevant vibration sources on average among response points. To some extent, problems of the sequencing are solved applying the method of partial coherence. In fact, however, the influence of the vibration source cannot briefly be averaged in a certain way, which lies in the reality that contributions of different vibrations and corresponding components are diverse in degree.

The thesis written is on the basis of transfer function, which summarizes the transfer rule of

response point taking acceleration as the stimulus, deduces the transfer characteristics of upper and lower acceleration of mount parts, as well as designs multiple experimental programs for the acquisition of interference frequency, dual-frequency components and corresponding proportion. Eventually, isolation characteristic of mount will be analyzed by virtue of real vehicle state in comparison with vibration isolation rate directly calculated by test signal so as to evaluate the isolation characteristic of mount.

## 2 Law summary of the acceleration transfer characteristic

Generally, reinforcing hammer or vibration exciter input force will be adopted in stimulating in the measurement of transfer characteristic, of which the transfer function is endowed with reciprocity in location. It is convenient when the installation location or excitation location of sensor is limited. While the acquired transfer function under the circumstance of static condition is not sufficient for completely representing the transfer rule of the operating structure, and when the real vehicle is at work, more of the input and output acceleration will be collected in the operating condition.

S1, S2 are constrains, E is incentives position, R1 and R2 are responsive position, the boundary status is considered in experimental calculation as shown in Table 1, each of the border includes a pulse + random noise, sine+ random noise, random signals etc. three incentive types (Figure.2). Inherent characteristics of the metal plate are shown in Table 2.

This section will design a simulation model based on the transient calculation of metal sheet mathematical simulation so that acceleration value of a structural component can be acquired. Then make a summary of acceleration as the transfer rule in input and output, so as to provide the following mount part vibration isolation and transfer characteristic with theoretical guidance. Diagram of the simulation

model is shown in figure.1, in which S1 and S2 are constraints, E is excitation location, and R1 and R2 are response location.

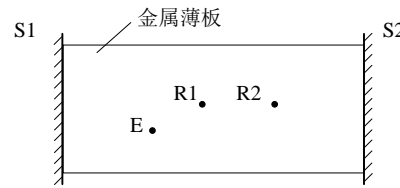


Figure.1 Transient calculation model of sheet metal

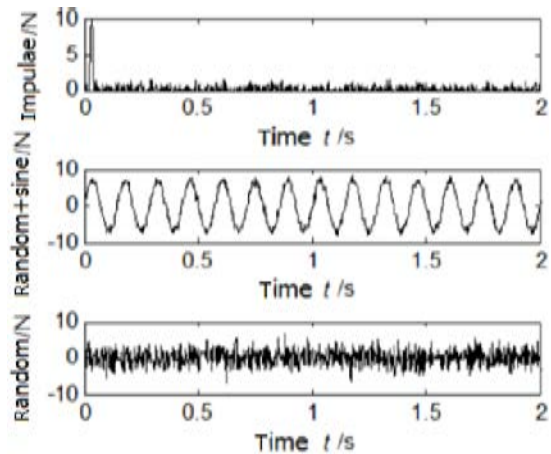


Figure.2 Different incentives forms of vibration sources

Table 1 Boundary constraints and response location description

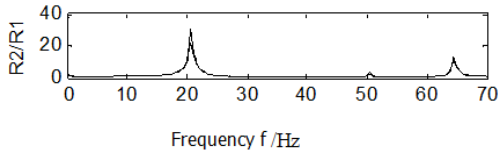
Boundary element	Ordinal position of Response Point R1, R2 and Incentive point E
status 1	E、R1、R2
status 2	R1、E、R2
status 3	R1、R2、E
status 4	R1(E)、R2
status 5	R1、R2(E)

Table 2 Natural frequency distribution of Metal plate /Hz

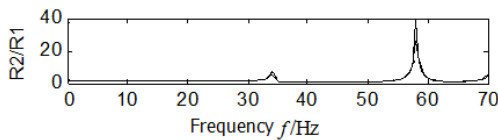
Order	1	2	3	4	5
Natural frequency	13.68	22.22	37.77	50.92	67.31

Figure.3 shows the response points' acceleration transfer characteristics of three setting incentives. It is indicated that after the position change of incentive points, there is no regularity of response points' transfer characteristics, nor can reflect on the natural frequencies of the metal plate; Any incentives forms' change on the same incentive point, the transfer characteristics is consistent when

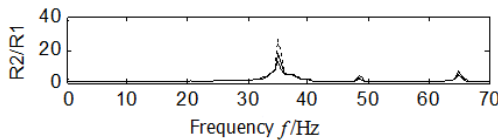
acceleration as the input and output calculation, namely when response measurement point position unchanged with different working conditions, acceleration transfer characteristics in any status calculation can be as structural transferring rules used in other working conditions.



(a) status 1

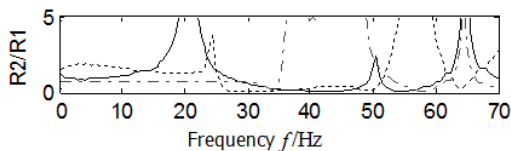


(b) status 2

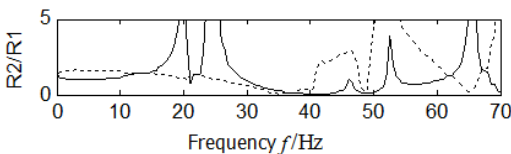


(c) status 5

Pulse + Random---- Sine + Random - Random  
Figure.3 Acceleration transfer characteristics of different incentive positions in different incentive forms.



-----status 1-----status 2-----status 3-----  
(a) Acceleration transfer characteristic contrast of status 1 ,status 2 and status 3



-----status 4-----status 5-----  
(b) Acceleration transfer characteristic contrast of status 4 and status 5

Figure.4 Transfer characteristics of same incentive in different input positions

Figure.4 shows Acceleration transfer characteristic of same incentive in different input positions, it is indicated that regardless the input

sequence of incentives, the transitive relation in natural frequency of response points R1 and R2 are in reciprocal forms. While there is no strict transitive relation at other locations, which are related to incentive location and damping.

Consider above, taking before and after acceleration of the structure as input and output to study its transfer characteristics, the attenuation law can be obtained at each frequency; Any changes of input and output role, its transfer characteristics is in reciprocal relationship at natural frequency location.

### 3 Theoretical Analysis of Vibration

#### Isolation Characteristics of Mount

#### Parts

#### 3.1 Theoretical Analysis of Isolation Characteristics of Mount Parts

It is assumed that the output vibration sources at power system and frame of a normal driving car are  $x'_{11} + m\Delta$  and  $x'_{22} + n\Delta$ , where  $\Delta$  are co-frequency components (There are some output incentive overlap at frame and power system when power system is in low speed and high gear),  $x_1$  and  $x_2$  are test signals, as shown in Figure 5.

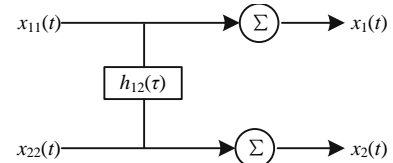


Figure .5 Interference system model of mount parts measurement

$$x_1 = (x'_{11} + m\Delta) + (x'_{22} + n\Delta)h_{21} \tag{1}$$

$$x_2 = (x'_{11} + m\Delta)h_{12} + (x'_{22} + n\Delta)$$

In the formula (1),  $h_{12}$  and  $h_{21}$  respectively are mounts parts acceleration attenuation characteristics of the power system to frame and frame to the power system,  $m$  and  $n$  respective are co-frequency coefficients.

Consider the irrelevance of frame and power systems in some frequency is, the mounts parts

frequency parts vibration isolation rate obtained by the actual test acceleration value, see the formula:

$$\frac{X_2}{X_1}(\omega) = H_{12}(\omega_1) + \frac{1}{H_{21}}(\omega_2) + \frac{mH_{12} + n}{m + nH_{21}}(\omega_3) \quad (2)$$

In the formula (2),  $\omega_1$  and  $\omega_2$  respectively are independent source frequency range for the power system and frame,  $\omega_3$  is co-frequency range. It shows that when there is mutual interference and co-frequency, the calculated vibration isolation rate is significantly greater than the theoretical vibration isolation characteristics.

### 3.2 Co-frequency component judgment of mounts parts' two ends vibration sources

It is assumed that the Independent sources forms of power system and frame is shown in the formula (3), expansion of Fourier series form is:

$$\begin{aligned} x_{11}'(t) + m\Delta &= \sum_{i=1}^{U_i} a_i \exp(j\omega_i t) + \sum_{p=1}^{V_p} m_p \exp(j\omega_p t) \\ x_{22}'(t) + n\Delta &= \sum_{k=1}^{U_k} b_k \exp(j\omega_k t) + \sum_{p=1}^{V_p} n_p \exp(j\omega_p t) \end{aligned} \quad (3)$$

In the formula,  $\omega_i$  and  $\omega_k$  is an independent component of two vibration sources,  $\omega_p$  is co-frequency component of sourcing signals;  $a_i$ ,  $b_k$ ,  $m_p$ ,  $n_p$  is the Fourier series expansion coefficients;  $U_i$  and  $U_k$  is non co-frequency numbers in vibration sources;  $V_p$  is co-frequency numbers.

Power spectral density function of mutual interference signals obtained by test is:

$$\begin{aligned} X_1(\omega) &= \sum_{i=1}^{U_i} a_i \delta(\omega - \omega_i) + \sum_{k=1}^{U_k} h_{2k} b_k \delta(\omega - \omega_k) \\ &\quad + \sum_{p=1}^{V_p} (m_p + h_{2p} n_p) \delta(\omega - \omega_p) \\ X_2(\omega) &= \sum_{i=1}^{U_i} h_{1i} a_i \delta(\omega - \omega_i) + \sum_{k=1}^{U_k} b_k \delta(\omega - \omega_k) \\ &\quad + \sum_{p=1}^{V_p} (n_p + h_{1p} m_p) \delta(\omega - \omega_p) \end{aligned}$$

In the formula,  $h_{1i}$ ,  $h_{1p}$ ,  $h_{2k}$  and  $h_{2p}$  is acceleration attenuation coefficient. For some frequency component  $\omega_i$ ,  $\omega_k$  and  $\omega_p$ , spectral amplitude ratio at any frequency can be obtained:

$$\lambda(\omega_i) = h_{1i}, \quad \lambda(\omega_k) = \frac{1}{h_{2k}}, \quad \lambda(\omega_p) = \frac{n_p + h_{1p} m_p}{m_p + h_{2p} n_p} \quad (4)$$

At the non-conjugated frequency, the ratio is independent of the source signal amplitude, but related to the acceleration attenuation coefficient. So the ratio is consistent with the acceleration transfer law for the different frequencies of the same source signal. Co-frequency attenuation rules and source signal amplitude are related to acceleration attenuation characteristics. Co-frequency and non-co-frequency component classification can be obtained by the two power spectrum ratio and attenuation characteristics.

### 4 Experimental design and calculation of mount performance test

The interaction of power system vibration source, frame vibration source and mounts is shown in Figure.6, labeled F1, F2, F3 stand for the vibration test locations of the frame, E1, E2, E3 are vibration test locations of corresponding power system, the actual test scenario shown in Figure.7. Design three kinds of tests to obtain parameters in Section 3. Test and signal processing methods shown in Figure.8.

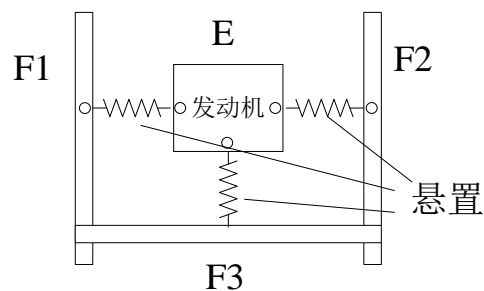


Figure.6 Diagram of connected power system and frame

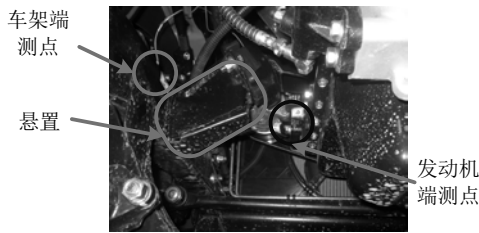


Figure.7 Actual test scenario of engine's left mount

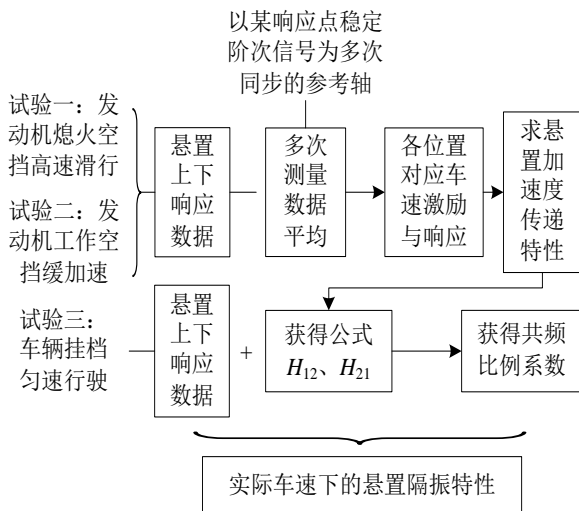


Figure.8 Vibration isolation characteristics process of actual driving speed in the test

Test 1: When in high speed, the attenuation coefficient  $H_{21}$  is obtained of frame to power system of neutral taxiing after turning off the engine, so that obtain  $h_{1i}$  and  $h_{1p}$  in formula (4). F1, F2, measuring point signals include wheels, axle, mount, frame, and car body etc vibration information caused by pavement that may force response parts, and then transfer the vibration into power system. Because the frame measurement points include the modal information of frame, and then the vibration of pavement to frame in different ways, the vibration energy reflection must be different in different measurement points. Meanwhile, the transferring vibration of power system and power-train mounting system may also arouse part of the modal characteristics

Test 2: Attenuation coefficient  $H_{12}$  of power system to frame is obtained of neutral engine operation, then get  $h_{2k}$  and  $h_{2p}$  in the formula (4). E1, E2, E3 is caused by the engine's incentive, including the engine and the power system's modal information, which transfers to the frame by three mounts. And

the three measuring points due to the overall effect of the frame, the test information includes the frame influence, which shows there coupling of the three measurements points vibration energy.

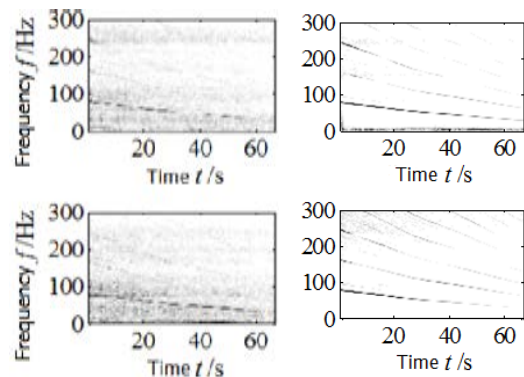
The above test, the mount parts are affected by structural constraints, where the constraints structure's modal information is included in the two ends' vibration signals. The calculated mount characteristics is the combined influence of real vehicle installation and operation status. It means that the study of mount's dynamic characteristics in real vehicle installation status is a rational research direction of mount improvement.

Test 3: Based on the actual vehicle running conditions, the co-frequency ratio is obtained by formula (4), meanwhile the interference sources influence is reduced from output response, and then get the time-domain isolation rate of actual mounts.

#### 4.1 Test Data Analysis

(1) Draw the time-domain and frequency-domain fitting picture in neutral taxiing. Describe the transferring rules from frame to power system of the mounts.

Figure.9 is acceleration spectra array chart of up and down of the mounts.



(a) Spectrum array chart of the mount's lower end at frame (b) Spectrum array chart of mount's lower end at engine

Figure.9 Left and right mounts' spectrum array chart of the power system in neutral taxiing

From figure.9, obvious order is showed at lower end of the mounts(frame), and with better consistence of left and right frame measurement

points. Base on the linear relation of frequency with slowing driving speed in slide, the order in figure.9 is chosen to replace driving speed, and then time axis is replaced by frequency axis.

Make alignment and average of multiple glide measurement by frequency axis to avoid the time axis inconsistencies in multiple glide. The glide data after multiple alignment is shown in Figure.10, the measurement data of multiple measurement has a better consistency.

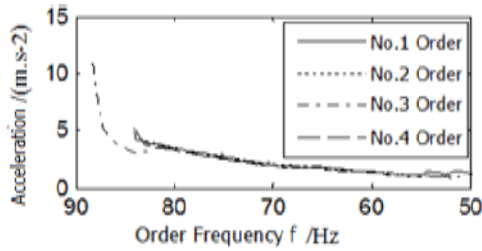
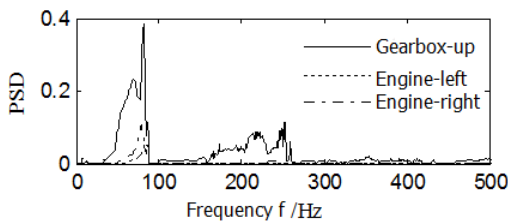
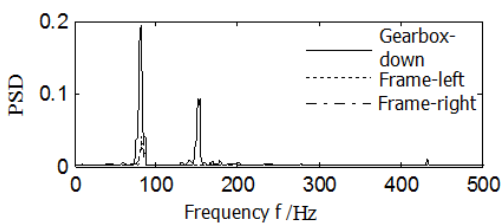


Figure.10 Left mount's vibration signals of power system-engine

Figure.11 is mount's up and down frequency domain contribution in neutral high-speed taxiing.



(a) mount upper end (engine)'s peak envelope of frequency domain response



(b) mount lower end (engine)'s peak envelope of frequency domain response

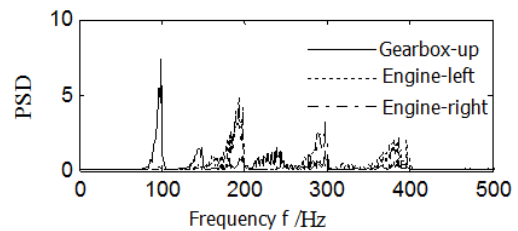
Figure.11 mount's up and down frequency domain contribution in neutral high-speed taxiing

The sliding speed is stable in 140km/h-60km/h of the test, Figure.11 shows that the frequency components of the frame and power system integration is concentrated on the former 300Hz, especially with the large upwards and downward vibration in gearbox. And the incentives

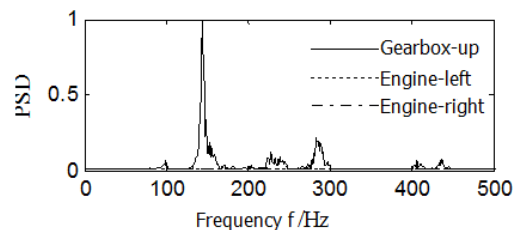
concentrate on 80 Hz and 150 Hz frequency range causes by pavement. The power-train's response is relatively larger than frame, which is inconsistent of frame as incentive in common sense. The possible reason of power-train as vibration system output in response: there are incentives transferred from transmission shaft etc. rear axle at power-train gearbox, and mounts plays an amplification role of some incentives. In fact, in actual driving, the influences on frame and power-train of pavement incentives follows the transferring rules.

(2) Transferring rules in power system neutral acceleration

Figure.11(a) shows that the pavement surface incentive energy is concentrated on former 100Hz and 200Hz frequency band, Figure.12(a) shows that the power system incentive energy is concentrated on 100Hz frequency band and corresponding frequency doubling. Consider above, there are obvious overlap of the producing incentives and response area. So, when in high speed of the actual operation, there are co-frequency between the power-train incentives and pavement incentives. Combined with its absolute amplitude contribution, the power system's contribution is the main component. which is consistent with the mount's traditional calculation thought that neglects pavement contribution.



(a) mount upper end (engine)'s peak envelope of frequency domain response



(b) mount lower end (engine)'s peak envelope of frequency domain response

Figure.12 mount's up and down frequency domain

contribution in neutral high-speed taxiing

(3) Vibration isolation rate calculation at constant speed.

Take 100km / h constant speed as a test evaluation conditions, Figures.13 and Figures.14 respectively are transferring characteristics comparison of power system response power spectral density curve at left mount frame and three test status left mounts in this condition.

Figure.13 shows that the response energy at frame is concentrated on frequency doubling of low frequency and engine running, that is, frame contribution concludes the incentive frequency from pavement and power system.

Combined with Figure.14, at low frequencies i.e. before 50Hz, constant 100km / h acceleration transferring characteristic is consistent with neutral slide curve. Major contribution except for 97Hz, is also consistent with the neutral acceleration curve. Based on the formula (4) and Figure 5's systematic approach, the frequency ratio can be obtained of the two status in 97Hz are:0.7:0.3, and the 50Hz pavement contribution is 4%, then the time domain vibration isolation rate in 100Km/h can be obtained of 4.28%. It improves 23.8% compared with the direction calculation of 5.62%. Other mounts vibration isolation rate comparison of different constant speed conditions is shown in Table 3.

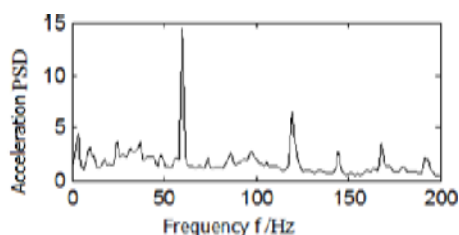


Figure.13 Response power spectral density curve of power system left mount at frame.

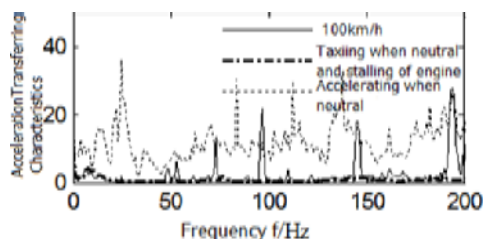


Figure.14 Transferring characteristics of left mount under three kinds of test conditions

Table 3 Time domain vibration isolation rate evaluation of gearing constant power system mount

Working condition	Left mount	Right mount	Gearbox mount
3 30	0.05	0.027	0.07
4 40	0.06	0.039	0.09
4 50	0.06	0.045	0.09
5 50	0.08	0.067	0.10
5 70	0.07	0.056	0.10
5 100	0.06	0.048	0.05
5 120	0.04	0.030	0.04
5 130	0.03	0.023	0.04
5 140	0.04	0.032	0.03

Table 3 shows 10-40% accuracy is improved of the time domain vibration isolation rate of the mounts anti-vibration performance after systematic method calculation, which is good for mount evaluations and further analysis.

### 5 Conclusion

This paper takes mount isolation efficiency as research objective, theoretically analyze the traditional computing error. The evaluation accuracy of can be greatly improved by designing a variety of test methods. The main conclusions are::

(1) When the actual transferring function cannot be got, this paper proposes taking acceleration transferring characteristics as computing basis, which show the mount's vibration-isolation characteristics.

(2)Theoretically analyzes the traditional calculation error of isolation-vibration characteristics, this papers takes high-speed neutral slide and setting-neutral acceleration etc. test to obtain mount's frequency domain acceleration isolation vibration characteristics, effectively correct time domain isolation rate calculation.

Test results show that the new method can effectively improve the evaluation accuracy of power system mounts vibration isolation rate, providing reliable data for the further improvement.

## Acknowledgment

This work is supported by the National Nature Science Foundation of China (51405221, 51305194). The authors also gratefully acknowledge the helpful comments and suggestions of the reviewers, which have improved the presentation.

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