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## MEASUREMENT AND EVALUATION OF DAMPING PROPERTIES OF DAMPING MATERIAL

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**Abstract** – This paper presents a testing method and process of damping properties of a kind of damping material. A damping system was regarded as the experimental subject composed of an oblate solid cylinder specimen of the damping material and an additional mass. The specimen is excited by an electromagnetic shaker in the vertical direction, and the specimen excites the additional mass vibrating. Not only the traditional damping properties, such as the damping ratio and vibration transfer rate, but also the nonlinear properties are measured to evaluate the damping specimen. Firstly the magnitude-frequency characteristics of vibration transfer rate are measured under constant stable excitation over a frequency range of 10 Hz to 320 Hz which includes the resonant frequency of the fundamental mode. Based on the magnitude-frequency characteristic, the half band power method is used to measure the damping ratio. In order to investigate the effects of vibration magnitude on the damping properties, the characteristic of the vibration transfer rate versus excitation magnitude at 210 Hz is measured. The experiment results reveal the fact that linear and nonlinear damping properties of the damping system change as the exciting magnitude increases.

**Keywords:** damping material, damping ratio, nonlinear vibration

### 1. INTRODUCTION

The damping material is widely used in different kinds of industrial products. In this paper, the testing subject is a specimen of a kind of damping material which is used to protect optical equipments and instrumentations from environmental shock and vibration, as shown in figure 1.

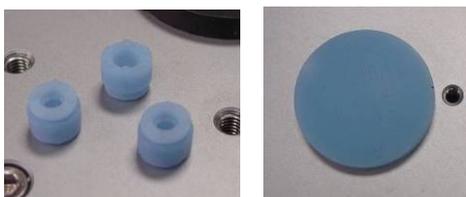


Fig. 1. The damper sample (left) and tested specimen (right).

Many methods have been developed to measure the material damping properties. R. F Gibson and R. Plunkett presented a forced-vibration technique for testing fiber-

reinforced composite materials as early as 1977 [1]. An ASTM (the American Society for Testing and Materials) standard E756-05 “Standard Test Method for Measuring Vibration-Damping Properties of Materials” proposed a series of test methods to measure the vibration damping properties of materials over a frequency range of 50 to 5000 Hz [2]. But the two methods specified cantilever specimens. In this paper, in order to reflect the operating conditions of the specimen in practice, a damping system composed of an oblate solid cylinder specimen clamped on a vertical shaker and an additional mass clamped on the specimen is taken as the test subject which is used to simulate the devices protected. The resonant frequency and the damping ratio are the basic properties of damping materials, so they are taken as main linear testing parameters which are tested by measuring the frequency characteristic of the vibration transfer rate. The vibration transfer rate is defined as the ratio of the vibrating magnitude of the additional mass to the vibrating magnitude of the shaker. Normally the half power method or its improved method is utilized to calculate the damping ratio [3].

The frequency characteristic of the vibration transfer rate was measured under constant magnitude vibrating state. Considering that the spring rate is nonlinear when the vibrating magnitude of the specimen is too large [4], the characteristic of the vibration transfer rate versus excitation magnitude is measured to determine the thresholds, above which different nonlinear properties of the damping system will appear.

To evaluate the damping performance, the resonant frequency and the damping ratio under constant exciting state are selected as two basic linear testing parameters. The nonlinear vibration of the damping system will produce some harmonic components. If the frequencies of the harmonic components are equal or close to the resonant frequencies of the protected devices, it is possible to damage them or cause some problems, such as lowering the measuring precision of instrumentations, etc. So it is necessary to evaluate the nonlinear vibrating characteristics of the damping system in the form of the main harmonic components.

### 2. TESTING SYSTEM

The testing system consists of seven major parts, as illustrated in figure 2. The damping specimen is the testing

subject. The additional mass and the damping specimen compose a damping system to simulate the practical operating conditions. The damping specimen is excited by the electromagnetic shaker, and the exciting vibration magnitude is measured by the standard accelerometer. The vibration signal is damped by the damping system to excite the additional mass vibrating. To avoid the interference from a contact measure process and reduce the measure uncertainty, the vibrating signal of the additional mass is measured by a non-contact measuring method with a laser vibrometer. The power amplifier is used to drive the shaker.

The vibration control of shaker and the acquisition and analysis of the vibration signals are processed by the testing control and analysis system. The vibration transfer rate is measured by calculating the ratio of the vibrating magnitude of the additional mass to that of the shaker at a stable sinusoidal vibration state. So the frequency characteristic of the vibration transfer rate can be measured over the frequency range concerned.

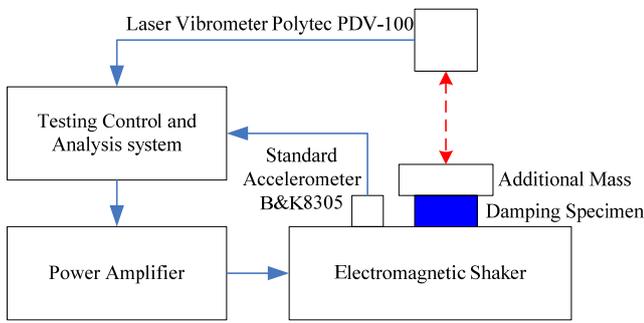


Fig. 2. The testing system of the damping properties.

### 3. MEASURING OF DAMPING RATIO

When the exciting magnitude is stable at 10 m/s<sup>2</sup>, and the additional mass is 167 g, the frequency characteristic of the vibration transfer from 10 Hz to 320 Hz is measured, as shown in figure 3.

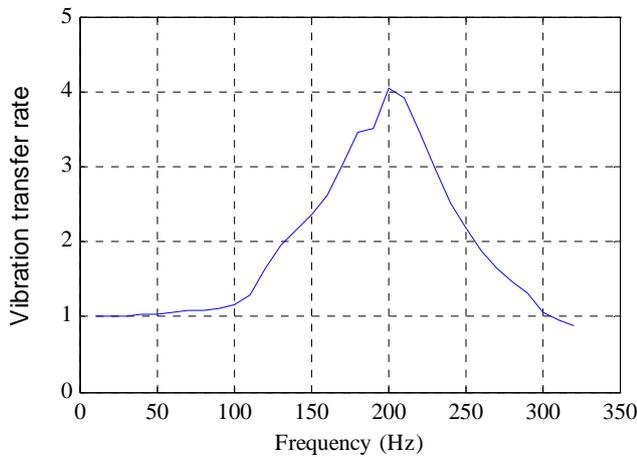


Fig.3. The frequency characteristic of the vibration transfer rate from 10 Hz to 320 Hz.

The resonant frequency of the damping system is approximately  $f_0 = 200.0$  Hz. The damping ratio is calculated by the half power method. The two half power frequency points respectively are measured:  $f_1 = 166.0$  Hz,  $f_2 = 232.6$  Hz. The loss factor is

$$\eta = \frac{|f_2 - f_1|}{f_0} = 0.333 \tag{1}$$

The damping ratio is estimated by

$$\zeta \approx \frac{1}{2Q} = \frac{|f_2 - f_1|}{2f_0} = 0.167 \tag{2}$$

where  $Q$  is the quality factor.

The damping ratio is far less than 1. It proved that the damping system is a weak damping system. When the damping system is excited to vibrate stably at the resonant frequency, the vibration magnitude vibration characteristic is illustrated in figure 4. The main vibration frequency is 200 Hz, at which the vibration magnitude is 0.03238 m/s. And the second harmonic component is 400 Hz, at which the vibration magnitude is 0.0008486 m/s. The ratio of the vibration of second harmonic to that of resonant frequency is 0.0262. And at other frequency range the distortion is also very low. So when the exciting magnitude is below 10 m/s<sup>2</sup>, the damping system can be regarded as a linear system.

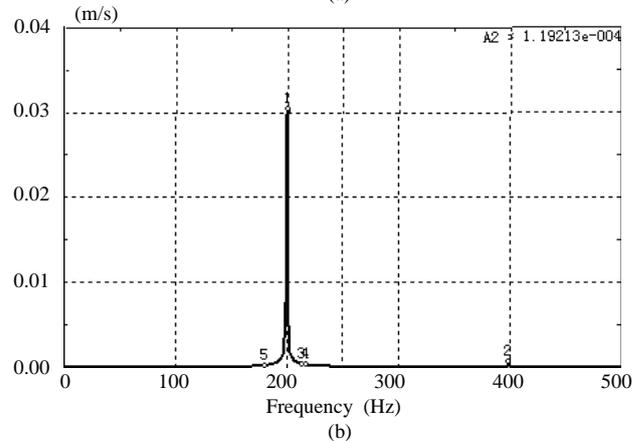
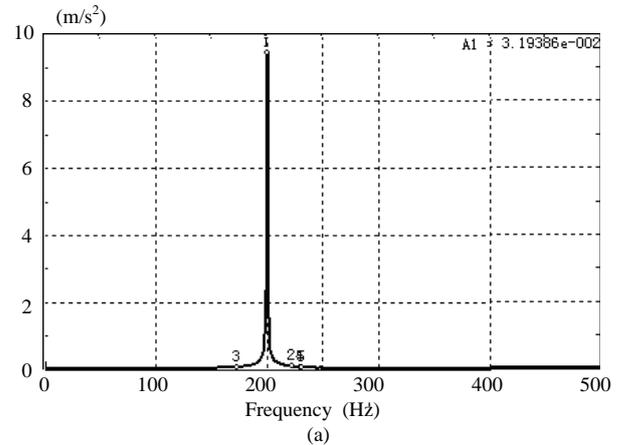


Fig. 4. The magnitude frequency characteristic of the exciting vibration (a), and the magnitude frequency characteristic of the responding signal of the additional mass (b).

#### 4. MEASURING OF NONLINEAR VIBRATION

To evaluate the vibration transfer characteristics with higher exciting vibration magnitude, a series of experiments are processed with different exciting vibration magnitudes at 210 Hz. As illustrated in figure 5, the vibration transfer rate from shaker to the additional mass reduced as the exciting vibration magnitude increased.

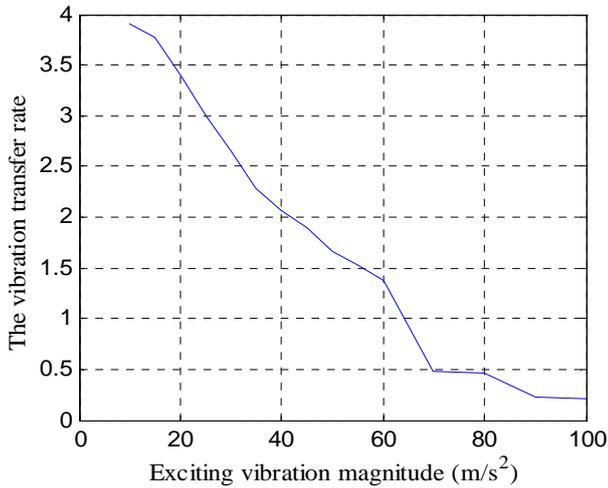


Fig. 5. The vibration transfer characteristics as increasing of the exciting vibration magnitude.

Ostensibly the damping capability changes for the better as the exciting vibration magnitude increases. In order to investigate the principle, the magnitude frequency characteristics are measured under different exciting magnitudes. To quantify the harmonic distortion, the statistics of the harmonic components of the additional mass with different exciting magnitudes are shown in Table 1.

When the exciting magnitude is 10 m/s<sup>2</sup>, the harmonic components of the responding signal of the additional mass are very small. So the vibration transfer from the shaker to the additional mass is practically linear.

When the exciting magnitude is more than 20 m/s<sup>2</sup>, the second harmonic components are more and more obvious. According to the data in table 1, the magnitude ratio of the second harmonic component to the fundamental component is shown in figure 6 as the exciting magnitude increases. Overall the magnitude ratio of the second harmonic component to the fundamental component increases as the exciting magnitude increases. However when the exciting magnitude is higher than 80 m/s<sup>2</sup>, the level of the nonlinear vibration gets deeper, not only the higher order harmonic components are produced, but also the half order harmonic component at 105 Hz is generated.

In summary, actually as the exciting vibration magnitude increases, the nonlinear level of the vibration transfer gets deeper. And the form of expression changes when the exciting magnitude is different. There are two thresholds: a low threshold and an upper threshold. When the exciting magnitude is below the low threshold, the vibration transfer can be regard as being linear. When the exciting magnitude

is between the low and the upper thresholds, the vibration transfer is nonlinear as appears as higher order harmonic vibration components. And the magnitude ratio of the second harmonic component to the fundamental component nearly increases as the exciting magnitude increases. While the exciting magnitude is more than the upper threshold, the vibration transfer is nonlinear in the form of production of higher order harmonic vibration components as well as half harmonic vibration component.

Table 1. The harmonic components of the damping system.

Exciting magnitude (m/s <sup>2</sup> )	The harmonic magnitudes of the additional mass with different exciting magnitudes.			
	Magnitude at 210 Hz (cm/s)	Magnitude at 420 Hz (cm/s)	Magnitude at 630 Hz (cm/s)	Magnitude at 105 Hz (cm/s)
10	3.004	0.061	/	/
15	4.296	0.122	/	/
20	5.163	0.175	/	/
25	5.661	0.213	/	/
30	6.037	0.278	/	/
35	6.089	0.260	/	/
40	6.290	0.318	/	/
45	6.456	0.355	0.074	/
50	6.298	0.320	0.078	/
55	6.393	0.358	/	/
60	6.257	0.381	0.091	/
70	2.531	0.285	0.059	/
80	2.768	0.268	/	/
90	2.350	0.248		3.200
100	2.693	0.302		2.990

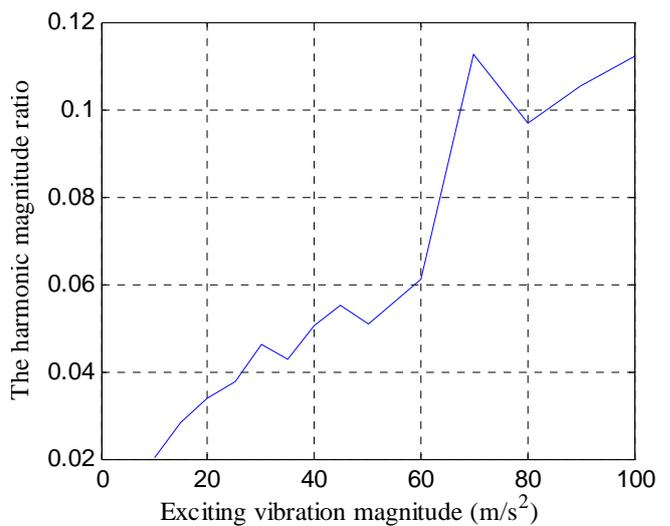


Fig. 6. The magnitude ratio of the second harmonic component to the fundamental component.

## 5. CONCLUSIONS

To evaluate the vibration damping capability of the damping material, a vibration testing system is built, based on which a systematic experiment investigation was conducted to test a damping system composed of an oblate solid cylinder specimen and a block of 167 grams of additional mass in the vertical direction. Two thresholds of the exciting vibration magnitude can be obtained by a series of experiments: a low threshold and an upper threshold. When the exciting vibration magnitude is below the low threshold, the vibration transfer of the damping system is practically regarded as a linear process. In this case, normally the resonant frequency and damping ratio are proposed to evaluate the damping capability of damping materials which can be calculated by measuring the magnitude frequency characteristics of the vibration transfer from the shaker to the additional mass. When the exciting vibration magnitude is above the low threshold, the vibration of the damping system should get into nonlinear vibrating state. When the exciting vibration magnitude is between the low and the upper thresholds, the vibration of the damping system is nonlinear as appears as higher order harmonic vibration components. And the magnitude ratio of the second harmonic component to the fundamental component increases as the exciting magnitude increases. While the exciting magnitude is above the upper threshold, the vibration of the damping system is nonlinear in the form of production of higher order harmonic vibration components as well as half harmonic vibration component.

Considering the practical operating situation of the damping material, it is possible that the damping system is

excited by large environmental vibration and shock to damage the protected devices by damping materials. The environmental vibration and shock are possibly of relatively high amplitude, such as the vibration in cars which excites the protected instrumentations. So the traditional testing method of the damping material is not sufficient to evaluate the damping capability with the linear damping properties. The nonlinear damping properties are necessary to be taken into consideration and tested to evaluate the damping capability of the damping material.

Because of the complexity of nonlinear vibration, in this paper, only some nonlinear vibration characteristics are investigated by means of experiments. As planned, the study and experiments of the nonlinear vibration properties of the damping system composed of the damping material and the protected devices are to be investigated further.

## REFERENCES

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