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Energy Dissipation Characteristics and Dynamic Modeling of the Coated Damping Structure for Metal Rubber of Bellows

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Received: 23 June 2018; Accepted: 19 July 2018; Published: 23 July 2018

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Abstract: A novel coated damping structure for metal rubber (MR) of bellows is designed based on large-size metal rubber sheets. This structure is dynamically tested in the bending direction at normal temperature. According to the test results, a model of the nonlinear elastic restoring force is set up, which describes the dynamic characteristics of the coated damping structure for metal rubber of bellows, and identifies the parameters of the model. The results show that the coated damping structure for metal rubber of bellows has a strong damping energy dissipation ability, its dynamic vibration characteristics are related to the vibration amplitude and frequency, and it is a complex nonlinear hysteretic system with multiple damping components. After identification of the parameters, the model of nonlinear elastic restoring force shows highly accurate results.

Keywords: bellows; metal rubber; coated damping structure; model of nonlinear elastic restoring force; parameter identification

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1. Introduction

A flexible connection can properly insulate from the noise produced by vibration of the pipeline system and structures of ships. Metal bellows are often used for connecting the pipeline system. They have the structure form of an expansion joint, and serve as vibration absorbers and noise reducers in the pipeline system [1,2]. However, bellows in use will mutually couple with fluid because of their special structure, leading to vibration of the pipeline and causing poor operation of the system [3,4].

To achieve more vibration reduction and energy dissipation of bellows, a common viscoelastic damping material: rubber, is often used to improve the vibration and noise reduction performance of bellows based on viscoelastic damping technologies. Ma et al. reported that the loss factor of bellows can be increased by using a rubber material with a large thickness and an elastic modulus of 200 MPa through the finite element model of bellows [5]. Liu et al. researched the stiffness and damping of the sandwich damping metal bellows through simulation and experiments. Their results provided a basis for the design of damping bellows [6]. Zhao et al. took the nominal diameter 219 mm (DN219) laminated damping bellows as the research object of a vibration test. They pointed out that the viscoelastic damping material can effectively reduce the high frequency vibration of the bellows [7,8]. The composite structure arising from the combination of bellows and viscoelastic damping materials with high damping can reduce the transfer rate of vibration in the pipeline system. In addition, it has a strong energy dissipating ability and can dissipate the energy produced by the pipeline vibration in the form of heat energy.

However, high temperature fluid often flows in the pipeline system of ships, and the dynamic mechanical property of rubber is greatly influenced by the temperature and vibration frequency. The damping performance of rubber is poor at normal temperature and low frequency, and the applicable band range is narrow, greatly limiting its application in the engineering field [9,10]. Metal rubber is a type of elastic porous material with a nonlinear stiffness and high damping. Its internal structure is a spatial network in which metal wires are staggered and interwound, providing such advantages as radiation resistance, corrosion resistance, high/low temperature resistance, long fatigue life, nonvolatility in a vacuum, and stable performance. A novel coated damping structure for metal rubber of bellows is designed by replacing a viscoelastic damping material with metal rubber [11]. Ma et al. used thermal experiments to analyze the relationship between the thermal expansion properties of metal rubber and the density of metal rubber and temperature. The experimental results provide a strong theoretical basis for the application of MR in the field of high-temperature insulation materials [12]. Hu et al. performed a damping performance test on the metal rubber damper in the temperature range of −70 °C to 300 °C, and pointed out that the damping performance of the metal rubber is far less dependent on the temperature than the viscoelastic damping material [13]. Lin et al. put the metal rubber into the simulated marine environment and corroded it. Then the damping energy dissipation test was carried out on the corroded metal rubber. They reported that the damping loss factor of the metal rubber in the simulated marine environment is more stable and has a certain resistance to ocean corrosion [14].

This paper describes the preparation of large-size metal rubber sheets and design of the coated damping structure for metal rubber of bellows. The structure is dynamically tested for the bending direction at normal temperature. Studies are carried out for the change rule of dynamic characteristics of the coated damping structure for metal rubber of bellows when the vibration amplitude and frequency excited by sinusoidal displacement have changed. To study the effect of sinusoidal displacement excitation on the coated damping structure at normal temperature, a model of nonlinear elastic restoring force is set up, which describes the dynamic characteristics of the coated damping structure for metal rubber of bellows and identifies the parameters of the model.

2. Design of the Coated Damping Structure for Metal Rubber of Bellows

To enhance the damping energy dissipation ability of metal bellows, the coated damping structure for metal rubber of bellows was designed as shown in Figure 1. The coated damping structure used large-size metal rubber sheets as the main energy dissipation elements, with wire rope clamps playing the role of fixing the metal rubber and providing preload together with the metal rubber. To facilitate the forming and installation of metal rubber, two pieces of metal rubber of the same size were used to cover the bellows. Compared with the structure of multiple metal rubber damping tapes, a large-size metal rubber sheet was more holistic and fit more closely with the bellows. During use, the large-size metal rubber sheet did not detach from the nodes of the bellows.

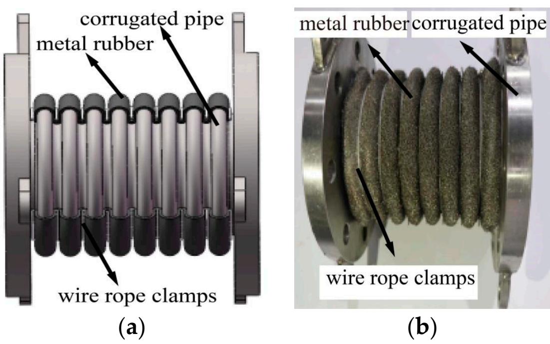

Figure 1. (**a**) Schematic diagram of the coated damping structure for metal rubber of bellows; and (**b**) physical picture of coated damping structure for metal rubber of bellows.

When the metal bellows deformed due to expansion, bending, and torsion, the metal wires in the metal rubber deformed accordingly. Upon deformation, the metal wires produced friction, which dissipated the energy produced by vibration of the metal bellows in the form of heat energy, thus enhancing the damping energy dissipation ability and structural stability of the metal bellows.

3. Test Specimen and Test Design

3.1. Preparation of Test Specimen and Design of Device

For the specimen of metal rubber in the test, as shown in Figure 2, 304 (06Cr19Ni10) austenitic stainless steel was used (Fiyta, Dongguan, China). The diameter of metal wire was 0.15 mm and outer diameter of metal spiral coil was 1.5 mm. A computerized numerical control (CNC) winding machine was used to stretch the spiral coil by a constant pitch, and then the stretched spiral coil was wound by a certain trajectory. The forming pressure of the metal rubber was 2520 kN. The length of the formed metal rubber was 281 mm, with a width of 173 mm, thickness of 2.5 mm, and density of 1.8 g/cm^3.

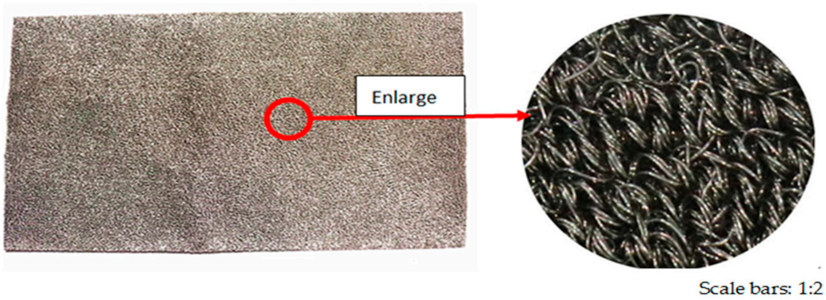

Figure 2. Large-size metal rubber sheets.

The design of the test device is shown in Figure 3. The flange at one end of the coated damping structure for metal rubber of bellows was fixed with the mounting bracket by bolts. Then the vibration exciter was connected to the base by a rigid connection. To study the dynamic characteristics in the bending direction of the coated damping structure for metal rubber of bellows, the vibrating head of the vibration exciter was tightly topped to the lower-middle position of the flange at the other end. The eddy current displacement sensor was mainly used to measure the longitudinal displacement of the coated damping structure for metal rubber of bellows.

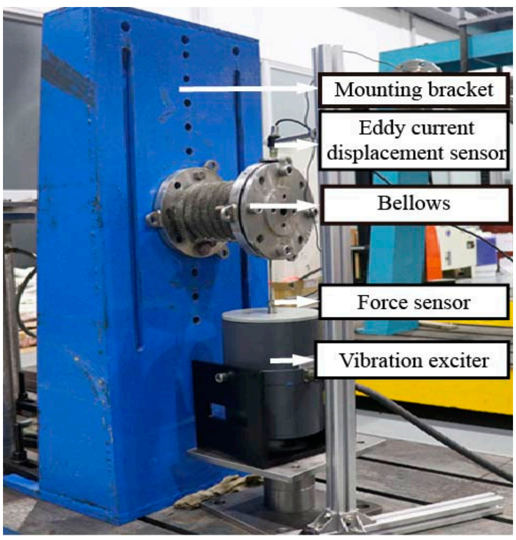

Figure 3. The test device.

3.2. Test System and Equipment

The dynamic test system of the coated damping structure for metal rubber of bellows was mainly composed of a sinusoidal signal generator, a power amplifier, a vibration exciter, coated damping structure for metal rubber of bellows, and a data acquisition system. The test principle is shown in Figure 4. The signal generator generated a sweep signal or a sinusoidal excitation signal, amplified the signal through a power amplifier, and then transmitted the amplified signal to the exciter. The metal rubber-coated damping structure of bellows received dynamical excitation from the exciter. Finally, the data acquisition system acquired displacement signals and force signals reflecting the dynamic characteristics of the metal rubber-coated damping structure of the bellows and sent them to a computer.

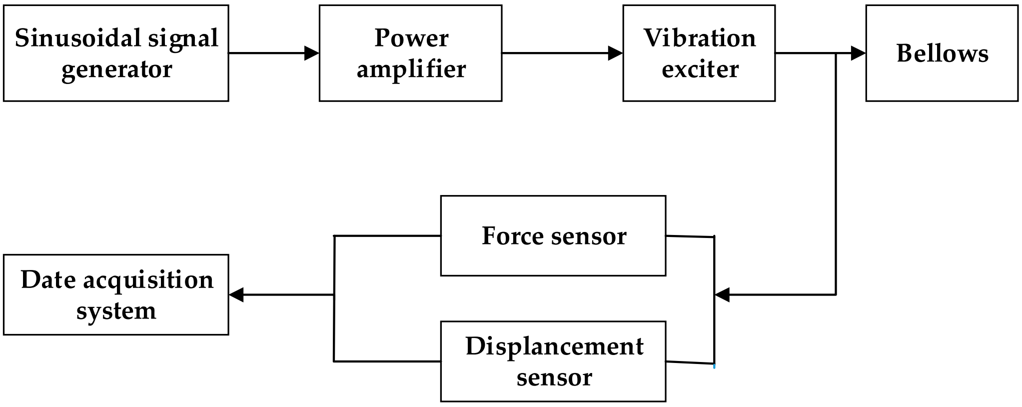

Figure 4. Principle of dynamic test for coated damping structure for metal rubber of bellows.

Technical specifications of the main equipment for the test system:

(1) Signal generation system:

VT-900X vibration controller manufactured by ECON (Hangzhou, China) was used to generate a sinusoidal sweeping signal of constant thrust in the frequency sweeping test. A 33512B waveform generator manufactured by Agilent Technologies (California, CA, USA) was used to generate a sinusoidal excitation signal in the damping energy dissipation test.

(2) Excitation system:

A vibration exciter JZK-50 provided with ECON power amplifier. The maximum exciting force was 500 N and the maximum displacement was 7.5 mm.

(3) Data acquisition system:

The data acquisition and analysis software provided for the VT-900X vibration controller manufactured by ECON was used in the frequency sweeping test. The DH5923N (Donghua Test, Jiangsu, China) data acquisition unit with data acquisition and analysis software (Version 8.1) was used in the damping energy dissipation test.

(4) Force measurement:

A YD-303 piezoelectric type quartz force sensor (Wuxi, China). Its resonant frequency was more than 60 KHz and had a charge sensitivity of 3 Pc/N.

(5) Displacement measurement:

A KD9004 eddy current displacement sensor (Dalian, China). Its sensitivity was 4 mv/μm, a measuring range is 4 mm, and a resolution of 1 μm.

4. Dynamic Test

4.1. Frequency Sweeping Test

4.1.1. Test Method

A sinusoidal frequency sweeping test was carried out for the coated damping structure for metal rubber of bellows. Data acquisition system was used to collect data on the elastic restoring force of the coated damping structure for metal rubber of bellows and longitudinal displacement during bending of the bellows. The test was carried out in two groups.

Group 1: The exciting force of the control vibration exciter was 1 N. A frequency of ≈5–100 Hz was recorded as forward sweeping and ≈100–5 Hz was recorded as backward sweeping to determine the frequency sweeping direction of the control vibration exciter.

Group 2: The frequency sweeping direction of the control vibration exciter was forward sweeping and the exciting force of the vibration exciter was 3 N.

4.1.2. Test Results and Analysis

The curve for sinusoidal frequency sweeping test results is shown in Figure 5. Based on the analysis of Figure 5, the following conclusions are drawn.

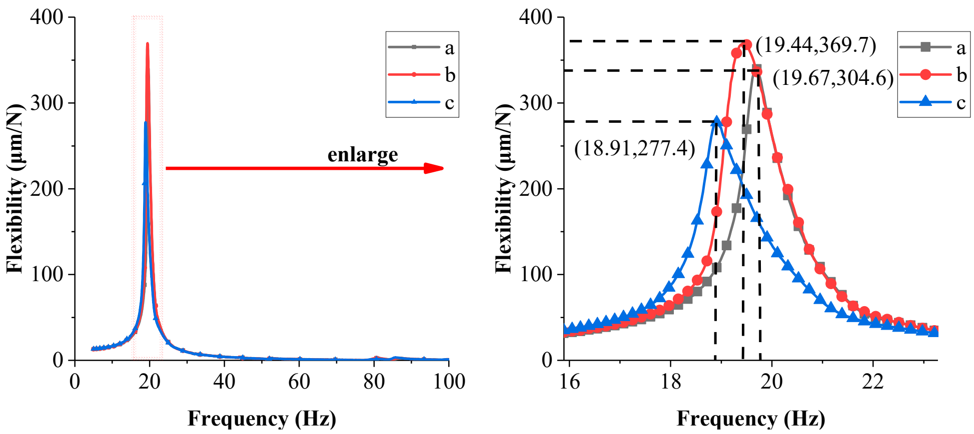

Figure 5. Frequency sweeping test results. Line a: Frequency sweeping curve for ≈5–100 Hz (exciting force = 1 N); Line b: Frequency sweeping curve for ≈100–5 Hz (exciting force = 1 N); Line c: Frequency sweeping curve for ≈5–100 Hz (exciting force = 3 N).

(1) When the exciting force was 1 N, the resonant frequency in the forward frequency sweeping test (Figure 5 (Line a)) was 19.67 Hz, and in the backward frequency sweeping test (Figure 5 (Line b)) it was 19.44Hz. It can be seen that the resonant frequency in forward frequency sweeping was higher than that in backward frequency sweeping, indicating that the frequency response curve of the coated damping structure for metal rubber of bellows bent and jumped. The bending and jumping of the frequency response curve indicate that the coated damping structure for metal rubber of bellows contained a high-order nonlinear elastic restoring force.

(2) When the exciting force was 3 N, the resonant frequency in the forward frequency sweeping test (Figure 5 (Line c)) was 18.91 Hz, and the flexibility was 277.4 μm/N. When the exciting force was 1 N, the resonant frequency in the forward frequency sweeping test (Figure 5 (Line a)) was 19.67 Hz, and the flexibility was 340.6 μm/N. It was found that the resonance peak of the sweeping curve drifted towards low frequencies, and the height of resonance peak was reduced at a high exciting force. This indicated that the increase of exciting force order led to a stiffness reduction and damping energy dissipation enhancement of the coated damping structure for metal rubber of bellows.

By carrying out forward and backward sinusoidal frequency sweeping tests for the coated damping structure for metal rubber of bellows, it was found that a nonlinear phenomenon exists for the coated damping structure for metal rubber of bellows. The test proved that the coated damping structure for metal rubber of bellows contained high-order nonlinear elastic restoring force and had the characteristics of variable stiffness and damping.

4.2. Damping Energy Dissipation Test

4.2.1. Test Method

A fixed-frequency excitation test was carried out for the coated damping structure for metal rubber of bellows. A data acquisition system was used to collect data on the elastic restoring force and longitudinal displacement during bending of the corrugated pipe. The test was carried out in two groups.

Group 1: The displacement amplitude of the coated damping structure for metal rubber of bellows was controlled at 0.5 mm, and the vibration excitation test was carried out for the coated damping structure for metal rubber of bellows at the exciting frequencies of 5 Hz, 10 Hz, 12 Hz, and 15 Hz, respectively.

Group 2: The exciting frequency of the vibration exciter was controlled at 10 Hz, and the vibration amplitude of the coated damping structure for metal rubber of bellows was changed.

4.2.2. Test Results and Analysis

Hysteresis loops are drawn in accordance with the data on force and deformation collected by the data acquisition system, as shown in Figures 6 and 7.

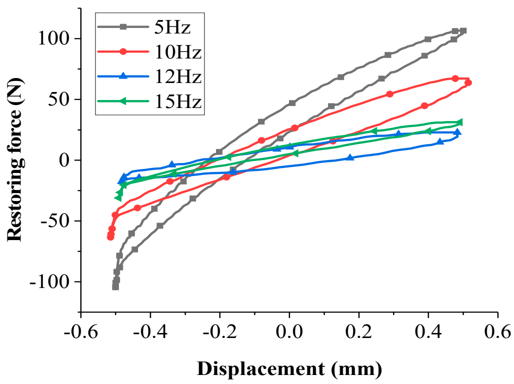

Figure 6. Hysteresis loop at different frequencies f (A = 0.5 mm).

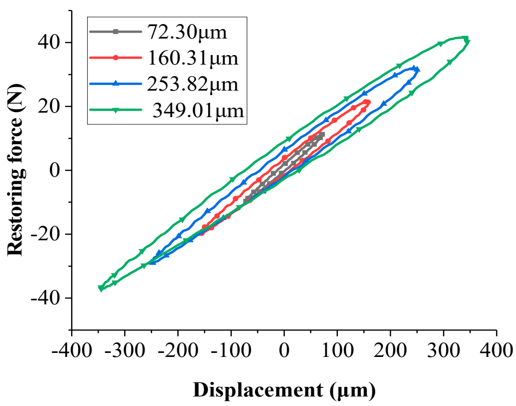

Figure 7. Hysteresis loop at different amplitudes A (f = 10 Hz).

During the test, a sinusoidal excitation was applied for the coated damping structure for metal rubber of bellows. As the coated damping structure for metal rubber of bellows contained a high-order nonlinear restoring force, the hysteresis response could not be measured directly. However, the hysteresis loops formed by its elastic restoring force and displacement represented the basic characteristics of damping energy dissipation in the coated damping structure for metal rubber of bellows. The area surrounded by the hysteresis loops was the energy dissipated by metal rubber in one vibration period.

(1) Damping energy dissipation ΔW:

A sinusoidal exciting force was applied on the coated damping structure for metal rubber of bellows, given by $F(t) = F_0 \sin \omega t$, the measured displacement response function was set as $x = x_m \sin(\omega t + \phi)$, and the sampling frequency was set as f. The number of samples in one vibration period was then $N = 2\pi f/\omega$, i.e., one vibration period was a dispersed period into N phase (or time) intervals, and the area of the measured hysteresis loops (energy dissipation) ΔW could be calculated using the following equation:

$$\Delta W = \int F(t_i)dx. \tag{1}$$

where ω was the angular frequency, F_0 was the amplitude of elastic restoring force, t was the time when elastic restoring force changes, x was the vibration displacement, x_m was the amplitude of vibration displacement, ϕ was the initial phase angle of displacement change, t_i was the time of displacement change.

(2) Dynamic average stiffness k was given as:

$$k = \frac{F_{\max} - F_{\min}}{2x_0} \tag{2}$$

where $F_{\max}$ was the maximum elastic restoring force, $F_{\min}$ was the minimum elastic restoring force, and x_0 was the vibration amplitude.

(3) Maximum elastic potential energy W:

The maximum elastic potential energy was calculated by taking half of the product between the average stiffness and square of vibration amplitude.

$$W = \frac{1}{2}kx_0^2 \tag{3}$$

(4) Structural loss factor η:

$$\eta = \frac{\Delta W}{2\pi W} \tag{4}$$

The test data was processed in accordance with the above equations for dynamic average stiffness, energy dissipation, and structural loss factor. The results are shown in Figures 8 and 9.

As shown in Figure 8, when the vibration amplitude was the same, with the increase of vibration frequency, the dynamic average stiffness of the coated damping structure for metal rubber of bellows reduced first and then increased slightly, the energy dissipation of the coated damping structure for metal rubber of bellows gradually reduced. Furthermore, the structural loss factor of the coated damping structure for metal rubber of bellows increased first and then reduced. In the average stiffness reduction stage, according to Equation (3), when the vibration amplitude was the same, the maximum elastic potential energy also decreased. As shown in Figure 10, the reduction rate of the average stiffness was higher than the reduction rate of the energy dissipation in the average stiffness reduction stage. According to Equation (4), the structural loss factor of the system should increase. In the average stiffness increasing stage, according to Equation (3), when the vibration amplitude was the same, the maximum elastic potential of the system also increased, and the energy dissipation reduced. According to Equation (4), the structural loss factor of the system should reduce. As shown in Figure 9, when the vibration frequency was the same, with the increase of the vibration amplitude, the damping energy dissipation of the coated damping structure for metal rubber of bellows gradually increased, the dynamic average stiffness of the coated damping structure for metal rubber of bellows gradually reduced, and the structural loss factor of the coated damping structure for metal rubber of bellows gradually increased. As shown in Figures 8 and 9, under all test conditions, the structural loss factor of the coated damping structure for metal rubber of bellows was greater than 0.1.

To sum up, the metal damping bellows were highly capable of damping dissipation; during vibration, the elastic restoring force of the coated damping structure for metal rubber of bellows was related to the vibration frequency and amplitude, and the coated damping structure for metal rubber of bellows had the characteristics of nonlinear hysteresis of variable damping and variable stiffness.

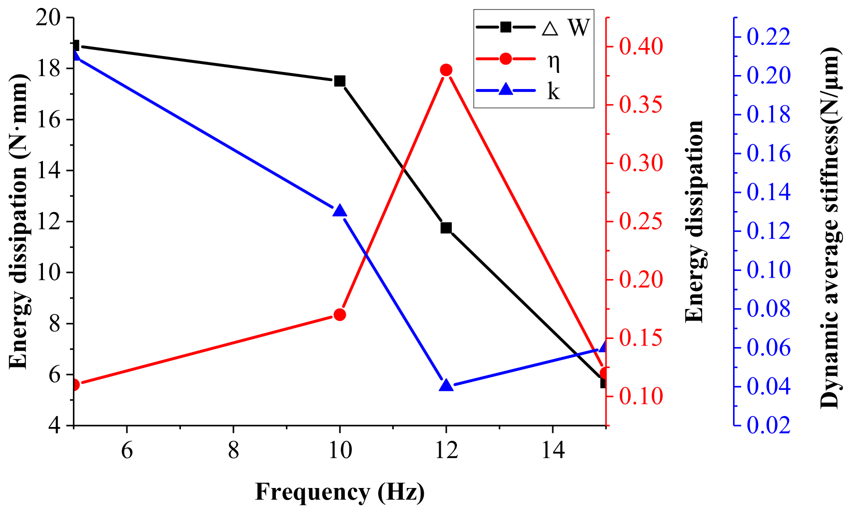

Figure 8. Test data processing results of group 1 (A = 0.5 mm).

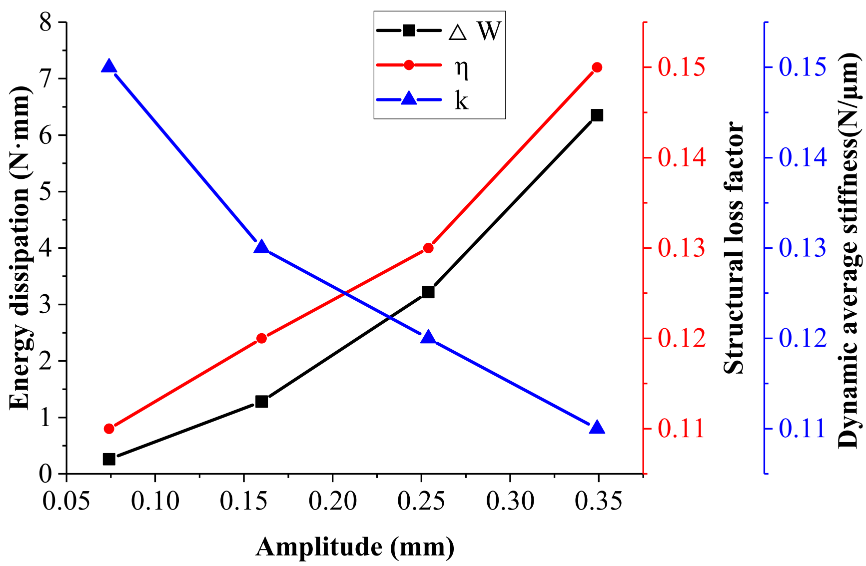

Figure 9. Test data processing results of group 2 (f = 10 Hz).

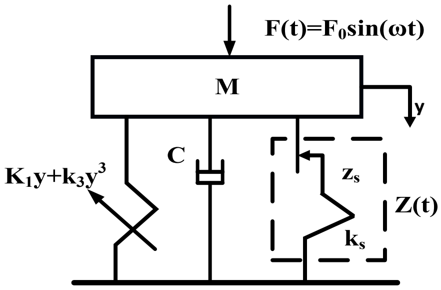

Figure 10. Hysteretic oscillator model.

5. Modelling and Parameter Identification

5.1. Modelling

For the construction of the cladding structure model, Rongong et al. has solved problems by applying the constrained layer damping (CLD) concept. This has been proved to be very instructive for the author to understand the coated damping structure for metal rubber of bellows. In the coated damping structure for metal rubber of bellows, the bellows is equivalent to the host structure, the metal rubber is equivalent to the viscoelastic damping layer, and the wire rope clamp is equivalent to the constraining layer [15]. Paimushin et al. proposed a method for identifying the elastic and damping properties of soft and hard materials containing rigid (substrate) materials, and using a quadratic function containing experimental and calculated internal damping parameters. Some effective multi-coating configurations, giving a relevant increase of the damping estimators of the coated structure with respect to the same uncoated structure, are obtained from the model simulation, and the results are critically assessed [16]. Catania et al. proposed a dynamic model based on simple beam geometry, while considering the previously introduced local dissipative mechanism and distributed viscoelastic constraints are proposed [17].

As shown in the frequency sweeping test, when one end of the coated damping structure for metal rubber of bellows was fixed and the other end was subjected only to dynamic bending force, the flexibility–frequency curve had only one obvious resonance peak. Therefore, the coated damping structure for metal rubber of bellows could be simplified into a single-degree-of-freedom system. The coated damping structure system for metal rubber of bellows had the nonlinear hysteresis characteristics of variable damping and variable stiffness, and its elastic restoring force contained multiple components. When the coated damping structure system for metal rubber of bellows was subject to the sinusoidal excitation of $F(t) = F_0 \sin(\omega t)$, the hysteretic oscillator model, as shown in Figure 10, could be used to describe the nonlinear elastic restoring force system. The elastic restoring force involved two aspects: non-memory and memory.

As shown in Figure 10, M was the total mass of the coated damping structure system for metal rubber of bellows; y was displacement response of the system; k was the primary linear stiffness coefficient; k_3 was the tertiary nonlinear stiffness coefficient; z_s was the restoring force during sliding of the memory; k_s was the linear stiffness before sliding; and *z(t)* was the memory restoring force.

For non-memory, if the high-order nonlinear elastic restoring forces that were above tertiary force were ignored, the model of elastic restoring force of the coated damping structure system for metal rubber of bellows could be described by Equation (5).

$$F(y(t), \dot{y}(t), t) = k_1 y(t) + k_3 y^3(t) + c\dot{y}(t) + z(t) \tag{5}$$

where $y(t)$ was the vibration displacement.

The incremental constitutive relation of the memory restoring force *z(t)* could be expressed by Equation (6).

$$dz(t) = \frac{k_s}{2}[1 + \mathrm{sgn}\{z_s - |z(t)|\}]dy(t) \tag{6}$$

$$k_s = \frac{z_s}{y_s} \tag{7}$$

where y_s was the elastic deformation limit of two solid contact faces upon macro sliding.

5.2. Parameter Identification

To check whether the nonlinear elastic restoring force model was applicable to the vibration system of the coated damping structure for metal rubber of bellows, the parameters were identified for the coated damping structure for metal rubber of bellows at the working conditions of vibration amplitude = 0.125 mm, frequency = 5 Hz; vibration amplitude = 0.172 mm, frequency = 5 Hz; and vibration amplitude = 0.350 mm, frequency = 10 Hz in accordance with the parameter identification method specified in references [18,19]. The results are shown in Tables 1–3.

After the identified parameters were put into the equations of the model elastic restoring force, the comparison diagrams were drawn between the measured curve and estimated curve (Figures 11–13). It was found that that the hysteresis loops drawn in accordance with the measured data matched well with the hysteresis loops drawn by simulation after parameter identification, indicating that the accuracy of parameter identification was high, and the model of nonlinear elastic restoring force built for the coated damping structure system for metal rubber of bellows could properly describe the dynamic characteristics of the system.

Table 1. Identification result (vibration amplitude = 0.125 mm, frequency = 5 Hz).

k_1/(N/mm)	k_3/(N/mm)	c/(N·s/mm)	z_s/N	y_s/mm
217.0547	93.7498	1.4049	0.2140	0.0006

Table 2. Identification result (vibration amplitude = 0.172 mm, frequency = 5 Hz).

k_1/(N/mm)	k_3/(N/mm)	c/(N·s/mm)	z_s/N	y_s/mm
207.5159	126.6203	1.6700	0.2571	0.0008

Table 3. Identification result (vibration amplitude = 0.350 mm, frequency = 10 Hz).

k_1/(N/mm)	k_3/(N/mm)	c/(N·s/mm)	z_s/N	y_s/mm
121.4934	20.2165	0.4099	0.1627	0.0016

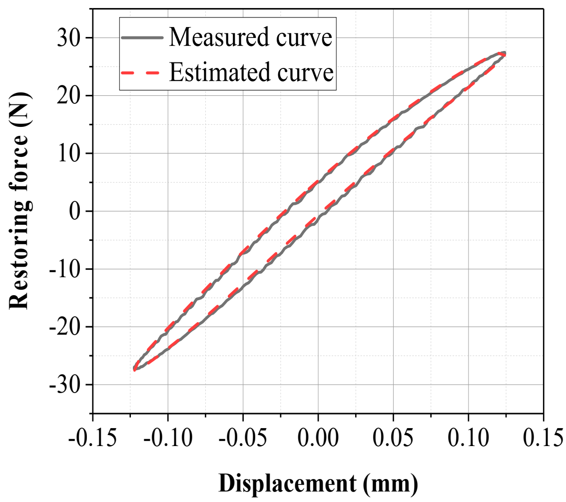

Figure 11. Comparison between the measured curve and estimated curve (vibration amplitude = 0.125 mm, frequency = 5 Hz).

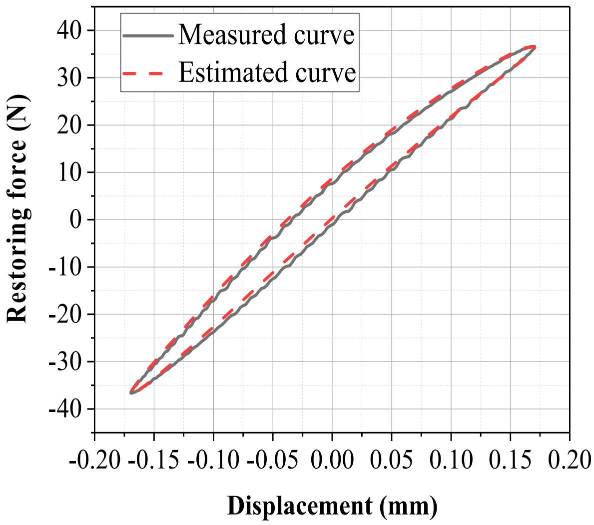

Figure 12. Comparison between the measured curve and estimated curve (vibration amplitude = 0.172 mm, frequency = 5 Hz).

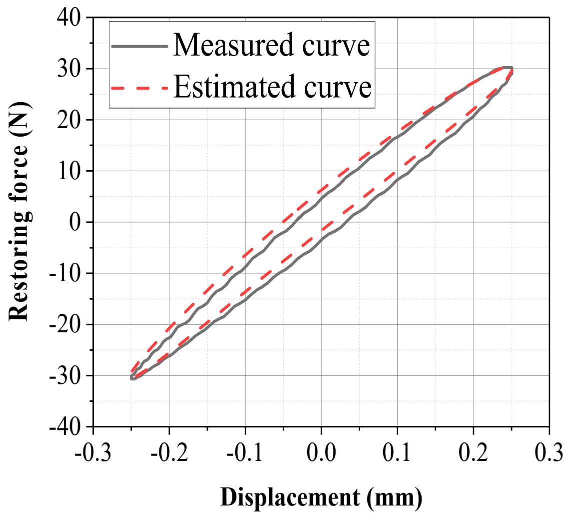

Figure 13. Comparison between the measured curve and estimated curve (vibration amplitude = 0.350 mm, frequency = 10 Hz).

6. Conclusions

In the present paper, a coated damping structure for metal rubber of bellows was put forward. The structure was dynamically tested for the bending direction at normal temperature. A model of nonlinear elastic restoring force was set up, describing the dynamic characteristics of the coated damping structure for metal rubber of bellows, and identified the parameters of the model.

(1) Large-size metal rubber sheets were prepared, and a novel coated damping structure for metal rubber of bellows was designed. In the dynamic experiment, it was found that when the bellows expands, bends, and torsionally deforms, the metal rubber and the bellows were closely attached, and the metal rubber followed the bellows to have a good deformation effect.

(2) Under all the test conditions, the structural loss factor of the coated damping structure for metal rubber of bellows was greater than 0.1, which proved that the coated damping structure for metal rubber of bellows was highly capable for damping energy dissipation.

(3) It also proved that the coated damping structure for metal rubber of bellows was a nonlinear hysteresis system of variable damping and variable stiffness.
(4) A model of non-linear elastic restoring force was set up for the coated damping structure for metal rubber of bellows. The model was proved to properly describe the dynamic characteristics of the coated damping structure for metal rubber of bellows by parameter identification and simulation.

Author Contributions: K.W., H.B., and X.X. conceived and designed the coated damping structure for metal rubber of bellows; K.W., T.L., and M.L. performed the experiments; K.W. and H.B. analyzed the data; K.W., H.B., and X.X. wrote the main paper.

Funding: Weapon Equipment "Thirteenth Five-Year Plan" Advance Research Project of China (99502040604); National Natural Science Foundation of China (No. 51705080); Natural Science Foundation of Fujian Province (No. 2018J01764).

Acknowledgments: The authors wish to thank Engineering Research Center for Metal and Rubber, Fuzhou University, for the assistance of all the experiments and funding support. The project supports from Weapon Equipment "Thirteenth Five-Year Plan" Advance Research Project of China (99502040604), National Natural Science Foundation of China (No. 51705080), and Natural Science Foundation of Fujian Province (No. 2018J01764).

Conflicts of Interest: The authors declare no conflict of interest.

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