Damping Energy Dissipation and Parameter Identification of the Bellows Structure Covered with Elastic-Porous Metal Rubber

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In order to improve damping energy dissipation of a U-shaped bellows structure, elastic-porous metal rubber as a cover layer was adopted and the corresponding vibration parameters were identified. First, the evolution of energy dissipation characteristics with respect to the changes of amplitude and frequency was investigated through a dynamic experimental test in the bending direction of the covered bellows structure. Second, the conspicuous hysteresis loop characteristics were described while the nonlinear constitutive relation was analytically modelled based on the exact decomposition method. Third, the corresponding parameters on dynamic properties of the covered bellows structure were determined by generalized least-squares estimation. Finally, the prediction results were compared with the measured displacement-restoring force curves to verify the accuracy of the developed dynamic model. The results indicate that the proposed dynamic model associated with the nonlinear constitutive relation for the covered bellows structure can well describe the evolution of the restoring force in terms of amplitude and frequency.

1. Introduction

The flexible metal bellows are commonly used for the connector of the pipeline system to realize the displacement compensation and reduce the vibration problem in a limited adjustment space [1]. Generally, the traditional polymer rubbers or foam with excellent damping performance can be used to achieve the purpose of vibration and noise reduction and have been widely used in engineering practice [2–4]. Recently, there has been growing interest in the usage of metallic material dampers (e.g., elastic-porous metal rubber or entangled wire mesh) to improve the environmental service performance of the pipe structures [5]. However, few studies are conducted for the dynamic model and its vibration analysis of the bellows structure covered with damping elements.

In the past decade, many scholars have conducted a series of studies on the mechanical properties and damping characteristics of pure metal bellows [6–8]. The previous efforts demonstrated that the improvement of damping performance of the pure metal bellows is clearly limited. It is vital to further investigate the bellows structure with an enhanced damping element. Zhao et al. [9] used the loss factor as the evaluation index to study the influence of geometric parameters such as wave height, wave distance, diameter, and wall thickness on the loss factor of multilayer bellows with interlayer damping. Lee et al. [10] evaluated on-orbit and launch environment isolation performance of a vibration isolator using bellows and viscous fluid. However, the stability of the structure cannot be guaranteed since the fluid-filled U-shaped bellows has high requirements for liquid sealing. The sandwich structure with a damping layer can suppress vibration by changing the natural frequency [11, 12]. In view of the damping structure annexed in bellows, damping vibration is generally applied to the surface of the bellows or covered with viscoelastic high damping elements due to the internal friction-induced energy dissipation. Thus, the vibration of the bellows structure in the pipeline system is attenuated by the damping effect. Since traditional damping materials such as viscoelastic polymers deteriorate or become ineffective in extreme environments such as high/low temperature, the metallic
damping materials (e.g., elastic-porous metal rubber) have been developed and applied in terms of individual case [13, 14].

Elastic-porous metal rubber is known as a type of macrostructural porous materials with a nonlinear stiffness and a high damping performance. Its internal structure of entangled metal rubber is a spatial network in which the coiled metal wires are staggered and interwound. This unique metallic damper element is capable to replace the traditional polymer damping materials because of its intrinsic advantages such as radiation resistance, corrosion resistance, high/low temperature resistance, long fatigue life, nonvolatility in vacuum, and stable performance [15]. In recent years, many scholars have studied the static mechanical properties of metal rubber. Courtois et al. [16] linked the microstructure evolution of the sample to its mechanical behaviour through X-ray tomography, thereby monitoring the number of contacts per unit volume and changes in density distribution during the compression test. Rodney et al. [17] reported that entangled single-wire materials show a unique strain-dependent Poisson function, reflecting reversible dilatancy in both compression and tension. Due to the nonlinear damping characteristics of metal rubber, of particular interest and complexity are the dynamic model and the corresponding parameter identification of elastic-porous metal rubber as a vibration damper. For the study of nonlinearity, experimental or finite element methods are mainly used [18, 19]. Ma et al. [20] studied the feasibility of using SMA-MR materials for active vibration control at different temperatures of the rotor dynamics system. Zhou et al. [21] studied porous twisted wire material with a porosity of 46–70% from two aspects: porous structure and Charpy impact behaviour. Macroscopically integral failure morphologies of the porous twisted wire material present mixed ductile brittle failure mechanisms, but microscopic impact deformation and failure mechanisms mainly show the ductile failure and fracture of pore skeletons and show complex energy absorption mechanisms. Yang et al. [22] investigated the vibration reliability characterization and damping performance of annular periodic metal rubber subjected to cyclic dynamic loading, provided an alternative method for the design of a periodic metal rubber damper in terms of individual application. The abovementioned efforts provide a trigger for the further investigation on parameter identification of the bellows structure covered with elastic-porous metal rubber.

The present work aims to investigate the energy dissipation characteristics of the covered bellows structure with elastic-porous metal rubber and the parameter identification of a nonlinear hysteresis loop. First, the bellows structure covered with a particular metal rubber damper is designed as well as the the dynamic test is introduced. Second, the damping performances including damping energy dissipation, dynamic average stiffness, and structural loss factor are characterized through the measured hysteresis loop, which can be obtained by means of a dynamic test in the bending direction of the covered bellows structure. Third, the nonlinear constitutive relation with four parameters is established based on the exact decomposition method. Finally, the comparison of predicted displacement-restoring force curves by the proposed model and experimentally measured results is performed and analysed in depth.

2. Descriptions of the Bellows Structure Covered with Elastic-Porous Metal Rubber

2.1. The Covered Bellows Structure. In this paper, a metallic bellows (DN108) made of stainless steel 304 with the thickness of 1.0 mm is adopted. Its corrugation part is a thin-walled tube with a U-shaped single-layer structure, and the flanges in both ends have several fixed holes for the connection of the external pipeline system. The dimensions of the bellows are shown in Figure 1(a). The covered bellows structure contains a U-shaped bellows, two pieces of elastic-porous metal rubber sheets (half structure), and several wire ropes on the bottom side of the bellows, as shown in Figure 1(b). The half-structure metal rubber sheets are covered on the outer wall of the bellows by pretightening connection of wire ropes.

2.2. The Fabrication of Elastic-Porous Metal Rubber. The fabrication process of the metal rubber damper can be summarized as follows: (i) the wire selection of raw material. (ii) Encircling and twisting the wire into a tight helix with a particular diameter which is generally controlled to be 5–15 times the wire diameter. (iii) The pretension of the helix into a spiral coil with a particular pitch, which is approximately identical to the diameter of the helix to ensure effective meshing. (iv) Weaving the coil in a crisscross pattern for a rough porous base material by a numerical control blank entangled device. (v) Molding or compression formation to obtain a designed or expected shape of metal rubber. (vi) Ultrasonic cleaning. Sometimes, the heating treatment of metal rubber can be used to improve the properties of metal rubber. The compression loading is tailored to provide a specific relative density for the specimens under a relatively stationary forming configuration [23, 24]. The main fabrication parameters of elastic-porous metal rubber are listed in Table 1. Figure 2 illustrates the compression forming tools and the fabricated metal rubber layer.

2.3. Experimental Description of the Dynamic Test. In order to investigate the evolution of energy dissipation characteristics and dynamic stiffness with the change of amplitude and frequency, an experimental platform of dynamic test was developed as shown in Figure 3. The exciter and the bracket are fixed on the ground, and one side flange of the bellows structure covered with a metal rubber damper is bolted to the bracket. The experimental dynamic system is mainly composed of a sinusoidal signal generator, power amplifier, vibration exciter, bellows covered with a metal rubber damper, and data acquisition system. The experimental schematic diagram is shown in Figure 4. A certain pretightening force can be applied on the pole of the vibration exciter and transmitted to the other flexible side of the bellows covered with a metal rubber damper in the
pretest process. The pretightening force ensures the exciter not disengages from the bellows covered with elastic-porous metal rubber during the dynamic test.

3. Dynamic Experimental Results and Model Establishment

3.1. Dynamic Testing Results. The fixed-frequency excitation tests ($f = 2 \text{ Hz}$) with the change of vibration amplitude (group 1) as well as the fixed-amplitude excitation test ($A = 150 \mu \text{m}$) with the change of vibration frequency (group 2) were carried out. In group 1, the hysteresis loops formed by its elastic restoring force and displacement represent the basic characteristics of damping energy dissipation in the bellows structure covered with elastic-porous metal rubber. The area surrounded by the hysteresis loops is the energy dissipation in one vibration period. From Figure 5, it can be obviously seen that the hysteresis loops expand gradually as the deformation amplitude increases, and this means that there is a stable positive relationship between the energy dissipation and the excitation amplitude. It indicates that the distortion of the hysteresis loop has almost no change in this situation.

On the other side, the distortion angle of the hysteresis loop with respect to the horizontal axis of the coordinate axis gradually decreases with the increase of the deformation frequency, as shown in Figure 6. This means that the stiffness of the bellows covered with the metal rubber damper is significantly affected by the frequency. However, energy
Figure 3: Dynamic experimental test device for the bellows structure covered with an elastic-porous metal rubber damper.

Figure 4: Schematic diagram of the dynamic test system.

Figure 5: The expansion of hysteresis loops with respect to the change of amplitude ($f = 2$ Hz).

Figure 6: The distortion of hysteresis loops with respect to the change of frequency ($A = 150 \mu$m).
dissipation does not respond significantly to the change of frequency.

In order to further study energy dissipation characteristics of the bellows structure covered with a metal rubber damper and describe the evolution of the hysteresis loop with amplitude and frequency in more detail, the damping energy $\Delta W$, the dynamic average stiffness $K$, and the structural loss factor $\eta$ were used to characterize the hysteresis loop. The detailed definitions on the abovementioned dynamic parameters can be also referred in some other publications [25, 26].

1. Damping energy dissipation $\Delta W$:
   The damping energy dissipation $\Delta W$ of the bellows coated with a metal rubber damper in one vibration period is the area of the hysteresis loop:
   \[
   \Delta W = \int F_d dx,
   \tag{1}
   \]
   where $F_d$ is the elastic restoring force and $x$ is the measured displacement.

2. Dynamic average stiffness $K$ is given as
   \[
   K = \frac{F_{\text{max}} - F_{\text{min}}}{2x_0},
   \tag{2}
   \]
   where $F_{\text{max}}$ is the maximum elastic restoring force, $F_{\text{min}}$ is the minimum elastic restoring force, and $x_0$ is the vibration amplitude.

3. Maximum elastic potential energy $W$:
   The maximum elastic potential energy is calculated by taking half of the product between the average stiffness and square of vibration amplitude:
   \[
   W = \frac{1}{2} K x_0^2.
   \tag{3}
   \]

4. Structural loss factor $\eta$:
   \[
   \eta = \frac{\Delta W}{2 \pi W}.
   \tag{4}
   \]

Figure 7 shows that the structural loss factor $\eta$ and the damping energy dissipation $\Delta W$ of the bellows covered with a metal rubber damper increase with the increase of the amplitude at the same excitation frequency. The interpretation for this evolution may be the number of contact points between the wires inside the metal rubber increases with the increase of the amplitude, which leads to the increase of energy dissipation. Despite the positive correlation between the amplitude and energy dissipation for other nonlinear dynamic systems [27], the bellows structure covered with elastic-porous metal rubber in the pipeline system has never been performed and authenticated. Besides the energy dissipation, the dynamic average stiffness $K$ of the bellows covered with a metal rubber damper decreases with the increase of amplitude. This means that the bellows covered with a metal rubber damper has stiffness softening characteristics. In a word, the amplitude has a great influence on the energy dissipation and the stiffness of the bellows covered with a metal rubber damper. It can be seen from Figure 8 that all the damping performances including the structural loss factor $\eta$, the damping energy dissipation $\Delta W$, and the dynamic average stiffness $K$ structure decrease as the frequency increases at the same excitation amplitude. This may mean that, with the increase of frequency, the wires inside the metal rubber are relatively loose, and the contact points between the wires are reduced, which leads to the reduction of damping energy dissipation $\Delta W$ and the decrease of structural loss factor $\eta$.

It should be noted that the structural loss factor of the bellows covered with a metal rubber damper is about 0.1, and this indicates that it has a strong damping dissipation capacity of the studied structure. The variations of the energy dissipation and the stiffness with respect to frequency and amplitude also imply that the bellows covered with a metal rubber damper has a nonlinear hysteresis characteristic.

3.2. Model Establishment. In order to better understand the nonlinear behavior, the dynamic characteristics of the bellows covered with a metal rubber damper can be theoretically modelled. According to the abovementioned analyses of experimental dynamic test, the hysteresis loop can be divided into upper and lower half branches. Thus, the upper half of the hysteresis loop whose speed is greater than zero can be expressed as a polynomial function:

\[
Q_h(t) = \sum_{i=0}^{N} a_i y(t)^i, \dot{y}(t) > 0.
\tag{5}
\]

According to the antisymmetric relationship, the lower half of the hysteresis loop can also be expressed by the polynomial fit as follows:

\[
Q_l(t) = \sum_{i=0}^{N} (-1)^{i+1} a_i y(t)^i, \dot{y}(t) < 0,
\tag{6}
\]

where $Q_h, Q_l$ are the upper and lower half curves of the hysteresis loop, respectively; $y(t)$ is the deformation; and $a_i$ is the polynomial coefficient. Among them, the number of items (odd number) taken by the $N$ polynomial is chosen according to the fitting precision.

Separating the odd and even terms of the polynomial in equations (5) and (6), they can be further expressed as

\[
Q_h(t) = \sum_{i=1}^{(N-1)/2} a_{2i-1} y(t)^{2i-1} + \sum_{i=0}^{(N-1)/2} a_{2i} y(t)^{2i}, \dot{y}(t) > 0,
\]

\[
Q_l(t) = \sum_{i=1}^{(N-1)/2} a_{2i-1} y(t)^{2i-1} - \sum_{i=0}^{(N-1)/2} a_{2i} y(t)^{2i}, \dot{y}(t) < 0.
\tag{7}
\]

Therefore, the total restoring force of the bellows covered with a metal rubber damper is given by
After the abovementioned processing, the dynamic hysteresis loop can be decomposed into two parts. Among them, \( F_k[y(t)] \) is a single-valued nonlinear function and \( F_e[\dot{y}(t), \ddot{y}(t)] \) is a two-valued nonlinear closed curve. \( F_k[y(t)] \) represents the part of the nonlinear elastic restoring force, and \( F_e[\dot{y}(t), \ddot{y}(t)] \) denotes the part of the nonlinear damping force in the hysteretic restoring force. The dynamic test can be simplified to a single-degree-of-freedom vibration
model, and the restoring force–displacement curve can be accurately decomposed, as shown in Figure 9.

It can be seen from the dynamic testing results that the nonlinear elastic restoring force for the total restoring force of the bellows covered with a metal rubber damper is not particularly strong; thereby, the two-stage expansion of the first term of formula (5) can be adopted. Therefore, the nonlinear elastic restoring force \( F_k \{ y(t) \} \) is constructed as follows:

\[
F_k \{ y(t) \} = k_1 y(t) + k_3 y^3(t). \tag{9}
\]

The dynamic results indicate that many factors make a notable effect on energy dissipation of the bellows covered with a metal rubber damper, due to the many damping components in the restoring force. If only one or two kinds of damping are used, it is far from the actual situation. Therefore, the damping component factor is introduced in this work to reflect the damping component and the nonlinear damping force. It can be given by

\[
F_c \{ \dot{y}(t) \} \{ \dot{y}(t) \} = c | \dot{y}(t) |^n \text{sgn} (\dot{y}(t)), \tag{10}
\]

where \( c \) is the damping coefficient and \( n \) is the damping component factor. It can be seen from the abovementioned formula that the larger the \( n \), the more the sensitivity of the damping force with respect to the change of the speed. On the contrary, the change of the damping force to the speed is relatively slow. When \( n = 1 \), equation (10) is actually simplified to linear viscoelastic damping force; when \( n = 0 \), the damping force is only related to the sign of velocity, indicating dry friction field results in the damping performance; and when \( n \) in the range of (0–1), it is expressed as a hybrid damper having both dry friction damping characteristics and viscoelastic damping characteristics. It can be seen that the damping component factor \( n \) can well describe the various complex damping conditions in the coated bellows structure with metal rubber damper.

In summary, a nonlinear dynamic model that can reasonably describe the dynamic performance of the bellows covered structure with a metal rubber damper can be obtained:

\[
g_n \{ y(t), \dot{y}(t), t \} = F_k \{ y(t) \} + F_c \{ \dot{y}(t), \dot{y}(t) \} = k_1 y(t) + k_3 y^3(t) + c | \dot{y}(t) |^n \text{sgn} (\dot{y}(t)). \tag{11}
\]

Considering the variation of elastic restoring force and the damping force with the deformation amplitude and frequency (see Figures 5 and 6), the nonlinear functions of the covered bellows structure can be accurately decomposed, and then, the constitutive relationship is described as

\[
g_n \{ y(t), \dot{y}(t), t \} = F_k \{ y(t) \} + F_c \{ \dot{y}(t), \dot{y}(t) \},
\]

\[
= k_1 (A, f) y(t) + k_3 (A, f) y^3(t) + c(A, f) | \dot{y}(t) |^n(A, f) \text{sgn} (\dot{y}(t)).
\]

4. Parameter Identification

4.1. Identification of Nonlinear Elastic Restoring Force

4.1.1. Stiffness Coefficient. The displacement under a certain working condition and the sampling data on the corresponding restoring force can be obtained from the dynamic test \((y_i, g_n(i), i = 1, 2, \ldots, N)\), and the polynomial can be fit in terms of the hysteresis loop by the least-squares method:

\[
g_n \{ y(t), \dot{y}(t), t \} = a_0 + a_1 y + a_2 y^2 + a_3 y^3. \tag{13}
\]

The odd-numbered coefficient in the formula is the coefficient of nonlinear elastic restoring force stiffness, which can be represented as

\[
\tilde{k}_1 = a_1, \\
\tilde{k}_3 = a_3.
\]

4.1.2. Damping Coefficient and Damping Component Factor. The nonlinear elastic restoring force can be reconstructed by using the identified \( \tilde{k}_1, \tilde{k}_3 \) and the measured displacement sampling signal \( y_i (i = 1, 2, 3, \ldots, N) \); then,

\[
F_k (y_i) = \tilde{k}_1 y_i + \tilde{k}_3 y^3_i.
\]

Subtracting \( F_k (y_i) \) from the total hysteresis restoring force sampling data \( g_n(i) \), the nonlinear damping force \( F_c (y_i, \dot{y}_i) \) of each sampling point is obtained:
\[ F_c(y_1, y_2) = g_n(i) - \tilde{k}_1 y_1 - \tilde{k}_3 y_1^3. \] (16)

According to the nonlinear damping force model \( c(A, f)\dot{y}(t)\), the damping coefficient \( \tilde{c} \) and the damping component factor \( \tilde{n} \) can be also identified by the nonlinear least-squares method.

4.2. Results of Parameter Identification. By using the abovementioned identification method, the values of the stiffness coefficients \( (k_1, k_3) \), the damping coefficient \( c \) and the damping component factor \( n \) under various working conditions can be obtained. Integrated with the analyses of the evolution of damping characteristics with frequency and amplitude, the corresponding constitutive relationship can be determined.

4.2.1. Parameter Identification of the Stiffness Coefficient. Figure 10 shows the space surface of the first-order stiffness coefficient \( k_1(A, f) \) varies with amplitude and frequency. It can be found that the stiffness coefficient \( k_1(A, f) \) decreases with increasing frequency. This can explain the occurrence of the distortion phenomenon of the hysteresis curve with frequency shown in Figures 5 and 7. The nonlinear least-squares method can be used for nonlinear fitting of the surfaces. According to the series of fitting results, it is found that the polynomial form is able to well describe the evolution of the stiffness coefficient \( k_1(A, f) \) with amplitude and frequency. Also, the function expression of\( k_1(A, f) \) can be acquired, as listed in Table 2, in which \( a_i \) denotes the coefficient of the function expression.

\[ k_1 = a_{00} + a_{10}A + a_{01}f + a_{20}A^2 + a_{11}Af + a_{02}f^2 + a_{30}A^3 + a_{21}A^2f \times 10^{-5}. \] (17)

As shown in Figure 11, the third-order stiffness coefficient \( k_3(A, f) \) varies with amplitude and frequency. It decreases rapidly at the beginning and then gradually stabilizes with the decrease of frequency. This evolution is consistent with the change of dynamic average stiffness \( K \) as in Figures 6 and 8. The polynomial order can be added to improve the fitting precision based on the first-order stiffness coefficient fitting. Table 3 lists the corresponding parameter coefficients of \( k_3 \).

\[ k_3 = \left( a_{00} + a_{10}A + a_{01}f + a_{20}A^2 + a_{11}Af + a_{02}f^2 + a_{30}A^3 + a_{21}A^2f \right) \times 10^{-5}. \] (18)

4.2.2. Damping Coefficient and Damping Component Factor. As shown in Figure 12, along with the change of amplitude and frequency, the damping coefficient \( c(A, f) \) exhibits a significant negative correlation with the frequency. This evolution is consistent with the change of damping energy dissipation \( \Delta W \) in Figure 9. The mathematical expression of the damping coefficient \( c = a_{00}f^{n_0}A^{n_0} \) can be calculated by power function fitting, and the corresponding parameters are listed in Table 4.

\[ n = a_{00} + a_{10}A + a_{01}f + a_{20}A^2 + a_{11}Af + a_{02}f^2 + a_{30}A^3 + a_{21}A^2f + a_{12}Af^2 + a_{03}f^3 + a_{04}A^4 + a_{31}A^3f + a_{22}A^2f^2 + a_{13}Af^3. \] (19)

4.3. Model Verification. According to the abovementioned parameter identification of the nonlinear elastic restoring force and the nonlinear damping force, the hysteresis loop of recovery force for the bellows covered with metal rubber damper can be reconstructed. In order to verify the developed model accuracy of the parameter identification algorithm, the comparison of hysteresis curve between estimated curves fitted by the nonlinear functional
The comparison results of the hysteresis curve associated with restoring force and displacement under different amplitudes and frequencies was performed. Figure 14 shows the comparison results of the hysteresis curve associated with restoring force and displacement under different amplitudes and frequencies.

It can be found that the fitting curves when the restoring force is greater than zero are better than those when the restoring force is less than zero. All the maximum errors occur at the lowest point of the hysteresis loops. This may be resulted from the pretightening force given by the vibration exciter on the bellows covered with a metal rubber damper during the dynamic test. The hysteresis loop seems to be asymmetric. When the positive displacement and the negative displacement are equal, the elastic restoring force of the positive displacement is greater than that of the negative displacement elastic restoring force. Due to the nonlinear stiffness characteristics of the bellows covered with a metal rubber damper, the deviation may increase with increasing amplitude, and the maximum deviation is up to 17.78%, as shown in Figure 14(c). A certain increase in frequency can result in an increase of deviation, as shown in Figures 14(a) and 14(d). This may be because the friction frequency of the wire inside the metal rubber cannot keep up with the excitation frequency, resulting in a decrease in the restoring force and an increase of deviation.

Comparison of the measured and estimated curves manifests a high consistency. The alternative mathematical model and parameter identification method are effective, which can well reflect the evolution of elastic restoring force. The presented work provides a theoretical basis for the dynamic design optimization of the metal rubber damper.
5. Conclusions

In this work, the evolution of the energy dissipation characteristics respected to the changes of amplitude and frequency was studied through experimental dynamic tests in the bending direction of the covered bellows structure. The nonlinear constitutive relation was modelled based on the exact decomposition method. The main conclusions of this study are drawn as follows:

- The measured loss factor of the covered bellows structure with metal rubber damper is about 0.1. This indicates that the proposed structure has a relatively strong damping dissipation capacity.
- The bellows covered with metal rubber damper has nonlinear hysteresis characteristics with variable damping behavior and variable stiffness. At the same frequency, the damping energy dissipation of the covered bellows structure with a metal rubber damper
increases with the increase of amplitude, while the dynamic average stiffness $k$ decreases with the increase of amplitude.

Based on the accurate decomposition of the hysteresis loop, considering the influence of amplitude and frequency on stiffness and energy dissipation, the mathematical model of the bellows structure covered with metal rubber damper restoring force is identified. Comparison of the measured and estimated curves manifests a high consistency. The results show that the developed model associated with the nonlinear constitutive relation of the covered bellows structure can well describe the evolution of the restoring force with amplitude and frequency.

**Data Availability**

The data used to support the findings of this study are included within the article.

**Conflicts of Interest**

The authors declare no conflicts of interest regarding the publication of this article.

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