Keywords: MRS, large ring-to-diameter ratio, parameter identification, porosity

Abstract
The demand for high-temperature-resistant metal rubber seals (MRSs) with a large ring-to-diameter ratio is increasing. In this work, an O-type MRS with a large ring-to-diameter ratio was developed by embedding a spiral network metal rubber into a stainless-steel ring with a special preparation process. The effects of the frequency, porosity, and amplitude on the dynamic performance of this O-type MRS were studied in detail. The mechanical properties of the MRS were characterized through dynamic tests, and the damping sensitivity was analyzed using orthogonal tests. The results show that the MRS has a better stability under different vibration frequencies. The energy consumption and loss factor of the sample increase with increasing porosity. With an increase in the loading amplitude, the energy consumption and loss factor of the test samples with the same porosity increase, whereas the dynamic average stiffness of the specimen gradually decreases. Furthermore, the range analysis of the orthogonal experiment shows that the factors affecting the damping performance of the seal are in the following order: porosity > amplitude > frequency. This study presents the dynamic mechanical properties of O-shaped MRSs with a large ring-to-diameter ratio and provides a foundation for their engineering application.

1. Introduction
The sealing technology has been widely used in the aerospace and pipeline transportation industries. Being one of the most basic components of any equipment, this technology plays a vital role in ensuring the stable and reliable operation of equipment. In recent years, the national defense equipment has been developed, and the utilization environment of aerospace products, weapons, and equipment is becoming increasingly complex; thus, the sealing problem under special working conditions needs to be solved urgently. Studying a more reliable sealing technology and achieving a more effective sealing are urgently required to ensure the sustainable development of modern industry [1, 2]. At present, the sealing products mainly consist of rubber and other polymer materials [3, 4]. However, as the sealing conditions are moving toward high-speed and high-temperature environments and modern equipment has stricter restrictions on the reliability and service life of seals, the traditional seals cannot meet the requirements of special equipment.

Metal rubber (MR) is a new type of elastic porous material and is known for its elasticity. It can be generally made from a specific quality of helically coiled wires that are entangled or knitted together, and compression forming is then used to obtain a desired shape. The unique forming process causes MR to have special mechanical properties, including temperature resistance, anti-aging properties, non-volatility, high sound absorption coefficient, and an ability to be prepared into various complex shapes [5–7]. As a consequence, MR has gradually become an excellent candidate for use in ordinary rubber products [8–10]. Additionally, extensive research has been conducted on the preparation of MR. More specifically, Bai et al [11, 12] investigated various...
winding and laying methods, which provide a reference for the mechanized and large-scale production of Mr.
Ren et al. [13] employed the Ansys software to virtually prepare the MR and addressed the complex problem of
the MR process simulation, which is of great importance for fabrication. This MR preparation technology
constitutes a prerequisite for MRS research.

The first application of MRs was to fabricate a liquid nitrogen end contact seal for the HK-89 gas turbine,
and positive sealing results were achieved [14]. Subsequently, Yan et al. [15] applied the MR technology to the
high-temperature sealing of an aero-engine combustion chamber, and they analyzed the calculation method
and failure mechanism of the minimum interference of the seal. Wang [16] reported the stress–strain
relationship of MR samples with different relative densities, clarified the mechanical properties (such as the
elastic modulus and the Poisson’s ratio of the MR materials), and established a simplified mechanical model of
the MRS system. Zhang [17] developed two types of MR seals with cladding layers and derived the equation for
the leakage rate of the seals based on the Navier–Stokes equation and the differential pressure flow model
established by Abouel-Kasem [18]. Liu [19] improved the preparation process of MR seals; based on the
Persson’s contact theory and the small gap percolation theory [20], they explored the variation of the contact
area of the two substrates at different magnifications and derived the expression for the theoretical leakage rate of
MR seals. Jiang [21] prepared four types of MRS materials with different wire diameter characteristics and
analyzed the influence of these different materials on the mechanical properties of the MRSs using both
experiments and theory.

The aims of the present work on O-type MRSs are to investigate the development of small-size seals, simulate
the mechanical properties of the seals, provide a theoretical derivation, and analyze the sealing mechanism.
However, for a sealing system that has more strict requirements, it is still necessary to further explore and research
O-type MRSs that have a large ring-to-diameter ratio. In this work, a porous metal-damping material was
prepared by winding and pressing the wires. A large-ring-ratio O-type MRS was then prepared by placing an MR
elastic inner core in the cladding layer via a three-step method. Furthermore, the influence of the amplitude,
frequency, and porosity on the dynamic mechanical properties of the samples was studied from an energy
perspective. In addition, a new MRS mechanical model was established, and its parameters were identified. Finally,
the mechanical model was compared with the test data to verify the accuracy of the model. This work provides a
reference for the engineering design and application of O-type MRSs with a large ring-to-diameter ratio.

2. Materials and methods

2.1. Test preparation

Figure 1(a) illustrates the preparation process of Mr In this study, austenitic stainless-steel Grade 304
(06Cr19Ni10) with a wire diameter of 0.1 mm was selected as the raw material of the Mr. The metal wire was then
wound into a spiral coil using a CNC-05 wire winding machine, and it was made into a blank using a wire guide
mechanism. Finally, it was formed using cold stamping into a ring-shaped MR with an outer diameter of
283 mm, an inner diameter of 276 mm, and a cross-sectional diameter of 2.5 mm. The metal wire is mixed with
dust and oil during the wire winding and blank stamping processes, and the prepared MR must be cleaned via
ultrasonic cleaning after cold stamping. As shown in figure 1(b), the preparation process of the coating samples
that were fabricated in a three-step process are: (i) The coating material was a hollow stainless-steel pipe with a
thickness of 0.5 mm to improve the service life of the MRS ring. To place the MR elastic inner core in the
stainless-steel cladding, a slot with a width of 2 ± 0.1 mm was required on one side of the stainless-steel
cladding. (ii) The slotted steel pipe was placed in a special bending mold for bending. (iii) A laser welding
machine was used to weld the joint of the bent steel pipe, and the cladding layer was rounded. Finally, an MRS
with a ring-to-diameter ratio of 552:7 was prepared, as shown in figure 2; this corresponds to an O-shaped MRS
with a large ring-to-diameter ratio. Clearly, the inner core of the MRS was made of a metal wire. As can be seen
from the scanning electron microscopy (SEM) image, the MR exhibits a porous structure of mutual hooks and
extrusions between the metal wires. The energy dissipation characteristics of the MRSs were mainly dominated
by contact friction and sliding deformation between neighboring spiral wires.

MR is a porous damping material prepared from wires through a series of preparation processes. Therefore,
the porosity of MR is an important factor that affects its damping and stiffness characteristics. In this work, a
wire of a specific quality was selected, and an adequate forming pressure was chosen in order for the sample to
reach the required diameter. The relationship between the porosity and the mass of the wires can be expressed as follows:

\[ \hat{\rho} = \frac{m}{V \cdot \rho} \]  (1)

\[ P = (1 - \hat{\rho}) \times 100\% \]  (2)
Here, \( m \) is the mass of the MR elastic inner core, \( V \) is the volume of the MR elastic inner core, and \( \rho \) is the density of the stainless-steel wire.

In order to analyze the mechanical properties of the MRS, the quasi-static mechanical properties were investigated using the virtual preparation simulation technology \([22]\). It was found that the meso contact pressure of the MRS is affected by the joint action of the compression condition, inner core support, and cladding failure. Moreover, MR has a high elasticity and good comprehensive mechanical properties to support it as the inner core of seals. Furthermore, dynamic tests were conducted to explore the mechanical properties of the MRS under actual loading.

A series of dynamic loading tests for the MRS specimens were carried out using an SDS–200 material testing machine, as illustrated in figure 3. The maximum loading stroke was 200 kN, ±50 mm, and the loading frequency range was 0.01–50 Hz. The testing system is shown in figure 3(a). The cross-sectional size of the MRS...
that was studied in this work was small, but the test range of the dynamic machine was large. To better test and analyze the mechanical properties of the sample, it was necessary to level the holding platform with a horizontal ruler before the experiment. Before the test, the MRS was evenly arranged in the holding platform. As shown in figure 3(b), the upper clamping chuck was adjusted to move the upper press plate into the cavity of the holding platform, and a certain amount of pre-tightening was applied to the sample piece to complete the tool debugging.

2.2. Calculation method of the damping test parameters

MR exhibits nonlinear hysteresis characteristics associated with the force–displacement curve and the energy dissipation under dynamic loading conditions. Therefore, it is impossible to use several direct measurement methods based on linear principles (such as the phase method, free attenuation method, and half-power method) [23]. Thus, Hou et al [24] proposed a new damping capacity measurement method of MR by means of the decomposition of the hysteresis curve. Furthermore, they proposed a parameter calculation method based on energy consumption and the maximum elastic potential energy. The energy consumption $\Delta W$ and the maximum elastic potential energy $W$ within a period are illustrated in figure 4.

The displacement excitation in the loading process can be expressed as:

$$x = x_0 \cos (\omega t + \alpha)$$

where $x_0$ is the displacement amplitude, $\omega$ is the loading period, and $\alpha$ is the initial loading phase. Then, the displacement value of equation (1) can be discretely expressed as:

$$x_i = x_0 \cos \left(\frac{2\pi i}{N} + \alpha\right), \quad i = 1, 2, \ldots, N$$

where $N$ is the sampling number of a vibration period. Moreover, $N = f_0/f$, where $f_0 = 2500$ Hz is the sampling frequency, and $f$ is the loading frequency.

The area of the hysteresis curve $\Delta W$ can be expressed as follows:

$$\Delta W = \oint F dx = \oint F d[x_0 \cos (\omega t + \alpha)]$$

$$= -\frac{2\pi x_0}{N} \sum_{i=1}^{N} F_i \sin \left(\frac{2\pi i}{N} + \alpha\right)$$

The maximum elastic potential energy $W$ stored by the material is:

$$W = \frac{1}{2} k x_0^2 = \frac{1}{2} F_0 x_0 = \frac{1}{4} (F_{\text{max}} - F_{\text{min}}) x_0$$

Here, $F_{\text{max}}$, $F_{\text{min}}$ are the maximum and minimum values of the restoring force in all of the data for the collection points, respectively, and $K = \frac{F_{\text{max}} - F_{\text{min}}}{2x_0}$ represents the dynamic average stiffness.
The equivalent loss factor can be calculated from the above equation as follows:

\[
\eta = \frac{\Delta W}{2\pi W} = -\frac{4f}{f_0(f_{\text{max}} - f_{\text{min}})} \sum_{i=1}^{N} F_i \sin \left( \frac{2\pi i}{N} + \alpha \right) \tag{7}
\]

3. Results and discussion

3.1. Dynamic mechanical properties and deformation mechanism of the MRS

In this study, the restoring force signal of the MRS under sinusoidal excitation was recorded. Moreover, the test initially runs for 100 cycles at each frequency so as to stabilize the specimen and measure reliable data only afterwards. The original obtained dynamic hysteresis loop has a large fluctuation because of the special geometry of the MRS. Therefore, the original data was filtered using the moving average method after stable loading cycles. The moving average filtering method is a signal smoothing method in the time domain. The idea of the algorithm is to average the sampling points near it as the smooth value of the point. The comparison between the original data and the processed data is shown in figure 5. As seen from the figure, the dynamic test results of the MRS show that the hysteresis area is narrow and the slope is large; this is caused by the large diameter and the relatively small cross section of the prepared MRS.

When MR bears a dynamic load, the spiral metal wires intertwined inside the materials have three possible types of interactions: (a) noncontact, (b) slip, and (c) stick, as illustrated in figure 6 [25, 26]. The MRS is composed of two parts (stainless-steel cladding and an MR inner core). When the test sample is subjected to a sinusoidal excitation (vertical load), the cladding has a high rigidity and is mainly subjected to most of the excitation load; the MR elastic core has a good resilience for providing the elastic force. The interaction between the cladding and the core enables the test specimen to be restored to its original position after a given amount of the excitation load. The MRS has a large rigidity because the cladding is a rigid material, and the inner core section of the MR is small and results in weak energy consumption and absorption; this makes the proposed MRS different from other Mr.

3.2. Single factor–controlled experiment and analysis of the results

An O-type MRS with a large ring-to-diameter ratio was selected and prepared using the abovementioned method, and the tests were conducted at room temperature. The influence of the frequency, porosity, and amplitude on the mechanical properties of the damped structures is discussed in this section. The specific parameters for the three sets of tests are listed in table 1.
3.2.1. Influence of the frequency on the damping capability

To better distinguish the influence of different porosities on the frequency, MRS samples with the largest and smallest porosity (69.7%, 51.5%) were selected to study the impact of the excitation frequency on the damping characteristics of the samples. The test results are shown in figure 7.

The size of a hysteresis loop indicates the damping capacity of a material. When the hysteresis loop is larger, the energy dissipation capacity of the material is higher. Figures 5 and 7 show that the hysteresis loop of the MRS in the sinusoidal excitation test has the shape of a thin 'willow leaf'. The main reasons for this phenomenon can be explained as follows: Firstly, the MRS has a large outer diameter, but its cross-sectional diameter is small; thus, the MR elastic inner core contains fewer spiral wires per unit volume. This leads the sample to have a smaller load capacity and damping capacity. Secondly, the stainless-steel coating of the MRS is a rigid material, and this also reduces the energy consumption characteristics. Moreover, it can also be seen that the hysteresis loops that correspond to different excitation frequencies are basically coincident with the same porosity. This means that the excitation frequency has little effect on the dynamic response of the test sample and that the MRS has better stability under vibration with different frequencies. To better quantitatively study the influence of the frequency on the MRS, the MATLAB software was used to program and calculate the test values of the loss factor $\eta$.
consumption $\Delta W$, and dynamic average stiffness $K$ of the test samples under different excitation frequencies. The calculation results are listed in table 2 and figure 8.

### Table 2. Effect of the frequency on the parameters of the dynamic properties.

| Porosity (%) | Frequency (Hz) | 1   | 2   | 3   | 4   |
|--------------|----------------|-----|-----|-----|-----|
| 69.7         | $\Delta W$ (N$ \cdot$ m) | 2.9711 | 2.9191 | 2.8638 | 2.794 |
|              | $\eta$ (N$ \cdot$ m$^{-1}$)  | 0.0352 | 0.0743 | 0.1248 | 0.1776 |
| 51.5         | $\Delta W$ (N$ \cdot$ m) | 2.4311 | 2.3761 | 2.2842 | 2.2502 |
|              | $\eta$ (N$ \cdot$ m$^{-1}$)  | 0.0379 | 0.0832 | 0.144  | 0.2053 |
|              | $K$ (N/m)               | 48.5071 | 48.487  | 48.9693 | 48.9127 |

#### 3.2.2. Influence of the porosity on the damping capability

Compared with solid metallic materials, the porosity of MRSs can be varied by controlling the mass of the metal wire for an identical volume. As a key factor affecting the material properties, it may play a considerable role on the damping capability. Therefore, four samples with different porosities (51.5%, 57.6%, 63.7%, 69.7%) were selected to study the effect of the porosity on the dynamic mechanical properties. The constant loading-unloading frequency was set as 1 Hz, and the target value of the amplitude was 0.15 mm. The corresponding hysteresis curves (figure 9(a).) and the calculated results (figure 9(b).) of the samples are shown in figure 9.
As seen in figure 9(a), with an increase in the porosity, the area of the hysteresis loop decreases gradually, and the hysteresis loop tends to be horizontal and smooth. Figure 9(b) shows that the energy consumption and loss factor of the test sample increase continuously with the porosity, but the dynamic average stiffness decreases. Additionally, the loss factor of the sample can be obtained from the dynamic test results on the energy dissipation, average stiffness, and equation (7). The above results on the relationship between the damping characteristics and the porosity can be explained as follows: With an increase in the porosity of the test samples, more pores are contained per unit volume of the sample so that the metal wire inside has a larger extension space and is more likely to slip. Additionally, the friction and sliding distance between the inner wires also increase, and this results in a higher absolute energy consumption capacity. However, with an increase in the porosity, the mass of the MR decreases, and this reduces the load-bearing capacity and the average stiffness of the MRS.

3.2.3. Influence of the amplitude on the damping capability
To further study the effect of the loading amplitude on the performance of the MRS, test samples with a porosity of 69.7% were selected, and dynamic tests with different amplitudes were carried out under an excitation frequency of 1 Hz. The obtained dynamic hysteresis loop was processed, and the result is shown in figure 10(a); the performance calculation results of the test samples are illustrated in table 3 and figure 10(b).

For test samples that have the same porosity, the hysteresis loop shifted in the direction shown in figure 10(a) with an increase in the amplitude, and the hysteresis area also increased gradually. Figure 10(b) shows that within a certain amplitude range, the energy consumption and loss factor of the test samples increased with an

| Amplitude (mm) | 0.1 | 0.15 | 0.2 | 0.25 | 0.3 | 0.35 |
|----------------|-----|------|-----|------|-----|------|
| Energy consumption (ΔW) | 0.2994 | 0.6385 | 1.0837 | 1.5986 | 2.2299 | 2.9711 |
| Loss factor (η) | 0.0302 | 0.0328 | 0.0327 | 0.0355 | 0.0369 | 0.0352 |
| Dynamic average stiffness (K) | 54.319 | 53.1516 | 51.5732 | 50.747 | 50.2215 | 48.5071 |

As seen in figure 9(a), with an increase in the porosity, the area of the hysteresis loop decreases gradually, and the hysteresis loop tends to be horizontal and smooth. Figure 9(b) shows that the energy consumption and loss factor of the test sample increase continuously with the porosity, but the dynamic average stiffness decreases. Additionally, the loss factor of the sample can be obtained from the dynamic test results on the energy dissipation, average stiffness, and equation (7). The above results on the relationship between the damping characteristics and the porosity can be explained as follows: With an increase in the porosity of the test samples, more pores are contained per unit volume of the sample so that the metal wire inside has a larger extension space and is more likely to slip. Additionally, the friction and sliding distance between the inner wires also increase, and this results in a higher absolute energy consumption capacity. However, with an increase in the porosity, the mass of the MR decreases, and this reduces the load-bearing capacity and the average stiffness of the MRS.
increase in the amplitude. When the loading amplitude was 0.35 mm, the loss factor decreased slightly. The dynamic average stiffness of the sample had an opposite trend to the amplitude. The reasons for this phenomenon are as follows: The increased applied amplitude allows the sample wires to slide to a greater distance and increases the friction probability; this results in the densification of the MR and an increase in the energy consumption. Furthermore, the state of the MRS in the compression process can be simplified, as shown in figure 11. The average dynamic stiffness results from the superimposition of the stiffness of the metal cladding layer and the MR elastic core. It is known that the stiffness of the metal cladding layer is basically unchanged when the linear elastic displacement is small (when the loading displacement is less than 0.2 mm (figure 11(b))). Because of the softening of the elastic stiffness and the enhanced damping of the MR inner core layer [23], the average stiffness degrades with an increase in the amplitude, and the decreasing trend becomes more pronounced [27, 28]. When the amplitude increases, the stiffness of the metal cladding layer increases, so the decreasing trend of the overall average dynamic stiffness is reduced. As the amplitude continues to increase, the stiffness of the MR core layer decreases significantly, so the average dynamic stiffness decreases. Moreover, the dynamic average stiffness gradually decreases with an increase in the amplitude, whereas the energy consumption increases more. Thus, the loss factor increases, but the increase is smaller, and this corresponds to equation (5).

3.3. Orthogonal test and analysis of the results for various parameters

The previous sections investigated the impact of various factors on the damping characteristics of the MRS. In this section, an orthogonal test method is described to further study the sensitivity to various factors of the damping energy consumption. According to the principle of the orthogonal test planning, the (a) porosity, (b) frequency, and (c) amplitude were selected as the influencing factors for this test, and three levels were selected for each factor. The orthogonal planning parameters are listed in table 4. The loss factor was used as the experimental index. Therefore, orthogonal arrays were chosen to carry out the orthogonal experiments. The specific factors are illustrated in table 5.

As seen in table 5, the comparison of the extreme values for the levels of each parameter shows that the relationship between the degree of influence of each factor is A > C > D > B. This means that the order of the degree of influence is porosity > amplitude > frequency. According to the calculation results shown in the table, the porosity has the greatest influence on the loss factor of the test sample followed by the loading amplitude; furthermore, the test sample is the least sensitive to the excitation frequency. Based on these results, the influence of the sample porosity on the loss performance should be prioritized in the design and selection of MRS components.
4. Establishment of a mathematical model and identification method for MRSs

4.1. Decomposition of the hysteresis curves and mechanical model

The dynamic force–displacement curve of the MRS has typical hysteresis characteristics. Additionally, the hysteresis loop can be decomposed to obtain the analytical diagram, as shown in figure 12.

According to the dynamic test data, the restoring force of the MRS is composed of an elastic force and a damping force. Thus, the restoring force hysteresis loop of the MRS \( g_y(y(t), \dot{y}(t), t) \) can be decomposed into a nonlinear elastic restoring force curve \( F_k(y) \) and nonlinear viscous damping force curve \( F_C(\dot{y}) \) (as illustrated in figure 12). Then, the nonlinear elastic restoring force can be further decomposed into a linear elastic force and a third-order nonlinear elastic force. This can be expressed as follows:

\[
\begin{align*}
g_y(y(t), \dot{y}(t), t) &= F_k(y(t)) + F_C(\dot{y}(t)) \\
&= k_0(P)y(t) + k_3(P)y^3(t) \\
&\quad + c(P, f) |\dot{y}(t)|^\alpha(P, f) \text{sgn}(\dot{y}(t))
\end{align*}
\]

In this formula, \( k_0, k_3 \) are the first-order linear stiffness coefficient and the third-order nonlinear stiffness coefficient, respectively; \( y \) is the deformation of the MR sample; \( C(P, f) \) is the damping coefficient; \( \alpha(P, f) \) is the damping component factor; \( \rho \) is the porosity.

4.2. Parameter identification method

4.2.1. Identification method of the nonlinear elastic restoring force

From the sinusoidal displacement loading data of the MRS, the displacement of one cycle and the corresponding restoration force sampling data are obtained as \( \{ y_i, g_y(t), i = 1, 2, ..., N + 1 \} \); then, the average value of the

\[
\begin{array}{|c|c|c|c|c|c|}
\hline
\text{Factor} & \text{Porosity} & \text{Frequency (Hz)} & \text{Amplitude (mm)} & \text{Empty column} & \text{Loss factor} \\
\hline
A & B & C & D & \\
1 & 1 & 1 & 1 & 1 & 0.0328 \\
2 & 1 & 2 & 2 & 2 & 0.0396 \\
3 & 1 & 3 & 3 & 3 & 0.0473 \\
4 & 2 & 1 & 2 & 3 & 0.0665 \\
5 & 2 & 2 & 3 & 1 & 0.0702 \\
6 & 2 & 3 & 1 & 2 & 0.0468 \\
7 & 3 & 1 & 3 & 2 & 0.0523 \\
8 & 3 & 2 & 1 & 3 & 0.0434 \\
9 & 3 & 3 & 2 & 1 & 0.0459 \\
\hline
\end{array}
\]

Figure 12. Nonlinear hysteretic damping model of MRS.
The hysteresis loop of the restoration force $F_{jk}(j)$ can be expressed as follows:

$$F_{jk}(j) = \bar{g}_{jk}(j) = \frac{1}{2}\{g_{jk}(j) + g_{jk}(2N + 2 - j)\},$$

$$\times j = 1, 2,...,N + 1$$

(9)

At the same time,

$$F(j) = k_{1}y(j) + k_{3}y^3(j), j = 1, 2,...,N + 1$$

(10)

### 4.2.2. Identification method of the nonlinear damping force

According to equation (8):

$$F_{i}\{i\} = g_{jk}\{i\} - F_{k}\{i\}$$

(11)

Additionally,

$$F_{i}\{i\} = c |\dot{y}(t)|^\alpha \text{sgn} (\dot{y}(t)), i = 1, 2, ..., N + 1$$

(12)

The estimated values of the damping coefficient $c$ and the damping component factor $\alpha$ can be obtained by solving equation (12) using the nonlinear least squares parameter identification algorithm.

### 4.3. Parameter identification results

#### 4.3.1. First-order and third-order stiffness coefficients $k_1$ and $k_3$

The values of the stiffness coefficients $k_1$, $k_3$ under each test condition were obtained, and the results are shown in table 6. By analyzing the influence of the porosity on the test samples, an effective and reasonable expression for the fitting function was determined. The stiffness coefficient of the test sample as a function of the porosity is illustrated in figure 13.

According to the stiffness versus porosity curve (figure 13), the power-level function polynomials (13) and (14) were selected for the fit:

$$k_1 = \sum_{i}^{3} a_ip^i = a_0 + a_1p + a_2p^2 + a_3p^3$$

(13)
The linear least squares method was used to identify the polynomial coefficients in equations (13) and (14):

\[
 k_i = \sum_{i=0}^{3} b_i p^i = b_0 + b_1 p + b_2 p^2 + b_3 p^3
\]

(14)

The linear least squares method was used to identify the polynomial coefficients in equations (13) and (14):

\[
 k_1 = 1.3857 \times 10^4 - 718.2442 P + 12.4153 P^2 \\
+ 0.00714 P^3
\]

(15)

\[
 k_3 = -7.9734 \times 10^4 + 4.0647 \times 10^9 P - 6.8713 \times 10^7 P^2 + 3.8432 \times 10^5 P^3
\]

(16)

4.3.2. Damping coefficient \(c\) and damping component factor \(a\)

The relationship between the damping coefficient and the porosity and frequency can be obtained by calculating the data for test samples with different porosity values and under different frequency conditions. To establish a reasonable and high-precision functional relationship between the damping coefficient, porosity, and frequency, a power function was used to perform the polynomial fitting.

It was found that a function formed via the combination of an \(n\)-degree polynomial and a power function can better describe the changes in the damping coefficient \(c(P, f)\) with respect to the frequency and amplitude. The fitting effect can be expressed in terms of the sum variance SSE, where SSE is the square sum of the errors of the corresponding points of the fitting data and the original data. When SSE is closer to 0, the data prediction is more successful, and the model selection has a better fit. Here, \(SSE = 0.083\). The least squares method is used to identify the parameters of the calculated data, and the polynomial function is:

\[
 c(P, f) = 2.3131 \times 10^9 - 1.5394 \times 10^8 P \\
+ 2.3102 \times 10^8 f + 3.7793 \times 10^6 P^2 \\
- 8.8981 \times 10^6 P f - 106765 \times 10^7 f^2 \\
- 4.0234 \times 10^4 P^3 + 1.4303 \times 10^6 P f^2 \\
+ 7.2661 \times 10^4 P^2 f - 6.9000 \times 10^6 f^3 \\
- 30.0027 P^3 f - 1.1207 \times 10^6 P f^2 \\
- 250.1505 P^3 f + 156.9452 P^4 + 5.7614 \times 10^5 f^4
\]

(17)

The same method that was used for \(c(P, f)\) was used to identify the parameter \(\alpha(P, f)\) with respect to the frequency and porosity. Here, \(SSE = 0.092\), and the following polynomial function can be obtained:

\[
 \alpha(P, f) = 2.2508 \times 10^5 - 1.5041 \times 10^6 P + 81.0503 f + 3.7530 \times 10^6 P^2 \\
- 5.5085 P f + 15.4037 f^2 - 4.1443 \times 10^4 P^3 - 0.2327 P f^2 + 0.1054 P^2 f \\
- 2.0790 P^3 f - 6.5317 \times 10^{-4} P^4 f + 0.0017 P f^2 + 0.0043 P f^3 \\
+ 170.9000 P^4 + 0.1457 P^4
\]

(18)

4.4. Model validation

To verify the reliability and accuracy of the mechanical model of the MRS as well as the accuracy of the parameter identification algorithm, the dynamic nonlinear mechanical model of the O-type MRS can be obtained by substituting the above identified parameter expressions (15)–(18) into equation (8). This gives the following:

\[
 g_n(y(t), \dot{y}(t), t) = 1.3857 \times 10^4 - 718.2442 P + 12.4153 P^2 + 0.0714 P^3 \\
- 7.9734 \times 10^4 + 4.0647 \times 10^9 P - 6.8713 \times 10^7 P^2 + 3.8432 \times 10^5 P^3 \\
+ \{2.3131 \times 10^9 - 1.5394 \times 10^8 P + 2.3102 \times 10^8 f + 3.7793 \times 10^6 P^2 \\
- 8.8981 \times 10^6 P f - 106765 \times 10^7 f^2 \\
+ 7.2661 \times 10^4 P^2 f - 6.9000 \times 10^6 f^3 - 30.0027 P^3 f - 1.1207 \times 10^6 P f^2 \\
- 250.1505 P^3 f + 156.9452 P^4 + 5.7614 \times 10^5 f^4\} |\dot{y}(t)| \alpha(P, f) \operatorname{sgn}(\dot{y}(t))
\]

(19)

The dynamic nonlinear mechanical model can be used to predict the hysteresis loop of MRSs with different porosities and frequencies. A comparison of the measured hysteresis loops is shown in figure 14. As seen from the figure, the theoretical prediction and the measured results are in good agreement. Furthermore, the established dynamic nonlinear mechanical model of the MRS can be used to better predict the nonlinear restoring force at a given porosity and loading frequency.
5. Conclusions

In the present work, a novel O-type MRS with a large ring-to-diameter ratio was fabricated using a special preparation process with the aim of investigating the mechanical performance of the material. The influence of various test factors on the mechanical properties of the MRS were studied in detail, and then a new dynamic model of the MRS was established. The main conclusions are as follows:

(1) According to the single-factor control test results, this type of seal presents a hysteresis loop with a unique ‘willow leaf’ shape, and this is different from other MR materials. The MRS has a larger dynamic average stiffness and a smaller hysteresis area; that is, its energy consumption capacity is small. For samples with the same porosity but different loading frequencies in the range of 1–5, the energy consumption and dynamic average stiffness of the MRS change little; this shows that the sample has a good stability under vibrations at different frequencies. Porosity has a great influence on the dynamic stiffness, energy consumption, and loss factor of the sample. Furthermore, for samples that have the same porosity, the energy consumption and loss factor increased simultaneously with an increase in the amplitude, whereas the dynamic stiffness of the specimen decreased.

(2) The influence of different factors on the damping properties of the MRS were analyzed by performing an orthogonal test, and the obtained results show that of the degree of influence follows the order porosity > amplitude > frequency. The MRS is rather sensitive to the porosity followed by the amplitude. However, frequency variations have little effect on the damping characteristics of the MRS.

(3) A new mathematical model for the restoring force of O-type MRSs with a large ring-to-diameter ratio was established on the basis of the theoretical analysis of the dynamic test data. The accuracy of the model was verified by comparing the model with the measured data.

Acknowledgments

We thank the National Natural Science Foundation of China (Grant No. 52175162, 51805086 and 51975123), the Fujian Provincial Natural Science Foundation (Grant No. 2019J01210), and the Fujian Province health education joint project (Grant No. 2019-WJ-01).

Data availability statement

The data generated and/or analysed during the current study are not publicly available for legal/ethical reasons but are available from the corresponding author on reasonable request.

Figure 14. Comparison between the theoretical prediction and the measured hysteresis curves ($P = 69.7\%$, $f = 3$ Hz).
Conflicts of interest

The authors declare no conflict of interest. The authors declare that they have no conflict of interest concerning the publication of this manuscript.

Author contributions

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