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Nonlinear passive damping of the X-shaped structure

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Abstract

Nonlinear damping can demonstrate excellent performance in vibration control. It is known that the ideal nonlinear damping could be performed as that the damping effect is as high as possible around resonance frequencies but as low as possible at other frequencies. However, how to realize such a nonlinear damping in real engineering practice with a convenient and passive manner would be an open problem in the literature. To this aim, this study investigates the nonlinear damping effect of an X-shaped structure which is inspired from biological limb skeleton systems. It is shown that, (a) the equivalent damping characteristic of the X-shaped structure is a nonlinear function of the vibration displacement and varies at different frequency; (b) the equivalent damping is adjustable with several structural parameters, and can be very high at around resonance frequencies but very low at other frequencies, showing an ideal passive and nonlinear damping property; (c) the linear horizontally-placed damper and joint friction can both produce the desired equivalent nonlinear damping in the vertical direction. The results of this study demonstrate an innovative and passive solution for designing desired nonlinear damping characteristics in various engineering practices by employing the X-shaped structure.

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Keywords: X-shaped structures; Vibration isolation; Displacement-dependent damping; Nonlinear damping.

1. Introduction

Here Damping is important in suppressing vibration within a dynamic system or structure. Generally, for vibration isolation or suppression, an ideal damping is usually expected to be as high as possible around resonant

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frequencies but as low as possible at other frequencies. For example, for vibration transducers, where the damping is often preferred to be high to protect the device when vibration amplitude is large, but should be reduced when vibration level is relatively low to enhance the measurement performance [1-5]. All these practical cases indicate that a nonlinear damping property is expected which should be dependent on vibration displacement and/or vibration frequencies.

In [4-7], nonlinear stiffness and damping are highlighted as emerging technologies for advantageous vibration isolation/control. It is theoretically shown in [7-11] and the references therein that nonlinear damping characteristics can be used to achieve much better vibration isolation/suppression, i.e., effectively suppressing resonance peak without increasing high-frequency transmissibility. The benefits of nonlinear damping properties in vibration control lead to many research activities considering the applications of nonlinear dampers in vibration control. There are several ways could achieve a nonlinear damping in engineering applications, for example, using active-controlled magneto-rheological (MR) fluid dampers [12-14], and/or smart materials [15] etc.

However, the main shortcoming of active control methods is the additional complexity and cost due to the additional sensors, controllers and power supplies involved. Therefore, it is more relevant to develop passive methods for the realization of desired nonlinear damping within a given system. Actually in [12], a passive MR damper is studied with nonlinear displacement-dependent damping characteristics. A similar passive damping system is also studied in [13-14] using also permanent magnet and coils in the MR dampers. Material or structural damping is usually nonlinear but not easy for free design for a desired nonlinear characteristic in practice [15-20]. Noticeably, the design of nonlinear damping by utilizing geometric nonlinearity of structures show many advantages in engineering applications. In [2] an adaptable damping via a bi-stable oscillator is presented which can increase the power dissipation and provide passive damping adaptable to excitation amplitude and frequency. The authors in [21-24] investigate several geometric or displacement-dependent nonlinear damping in vibration isolation.

Natural animals and plants are always endowed with various advantageous features in maintaining stability and robustness in harsh environments which provide an inspiring source for technical development and improvement [25-28]. A bio-inspired limb-like structure (LLS) or X-shaped structure is recently studied in [26-28]. It is shown that the geometric nonlinearity within the X-shaped structure can produce nonlinear stiffness characteristics which are very beneficial for vibration isolation or suppression and easy to design and implement in practice.

To explore fully and understand more the X-shaped structure, this study is to report some recent understanding on the nonlinear damping characteristics of the X-shaped structures. In this study, the X-shaped structures is installed with only a linear damper horizontally placed within the structure, and all damping effects come from this installed passive linear damper and passive joint friction involved. It is shown that the equivalent damping characteristic of the X-shaped structure is a nonlinear function of the vibration displacement and varies at different frequency; the equivalent damping is adjustable with several structural parameters, and can be very high at around resonance frequencies but very low at other frequencies, showing an ideal passive and nonlinear damping property; the linear horizontally-placed damper and joint friction can both produce the desired equivalent nonlinear damping in the vertical direction.

The results reveal an innovative and purely passive solution for designing desired nonlinear damping characteristics by employing the X-shaped structure, which is adjustable, flexible and cheaper to implement, and can be applied in various engineering practices.

2. The X-shaped structure with a passive linear damper installed horizontally

Fig. 1 shows the model of the X-shaped structure with a passive linear damper installed horizontally. The X-shaped structure has asymmetric rod length L_1 and L_2. θ_1 and θ_2 are the initial angles at equilibrium. Rods are connected by joints using bearings. There are $n_x = 6n+2$ rotational joints in an n-layer X-shaped structure. The horizontal damper is installed between two joints of any one layer. φ_1 and φ_2 are the angular displacement regarding to L_1 and L_2 respectively. x_1 and x_2 are the horizontal displacement variables of the joints. The displacement in the vertical direction is denoted by y. The stiffness of the vertical linear spring is given by k. M is the mass of the upper load. c_1 is the air damping coefficient. c_2 is the rotational damping coefficient of each joint and c_3 is the damping coefficient of a horizontal damper. z is the base excitation and $y_1 = y - z$ is the relative motion of the mass in the vertical direction. When $y_1 = 0$ the structure is in the balanced position. ω_0 is the excitation frequency, ω_1 is the natural frequency of the X-shaped structure and $\Omega = \omega_0 / \omega_1$ is a dimensionless parameter. The main purpose of this study is to investigate the nonlinear damping characteristics.

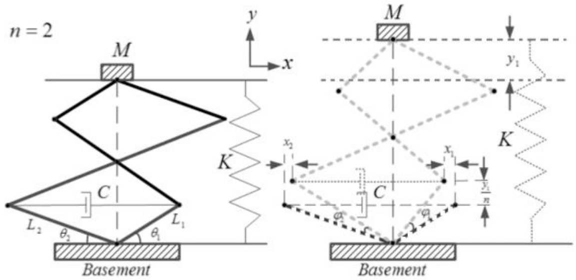

Fig. 1. The X-shaped structure with a passive horizontal damper

2.1. Modeling

With the Lagrange principle in (1), the equation of motion is obtained in (2):

$$\frac{d}{dt}\left(\frac{\partial T}{\partial \dot{y}}\right) - \frac{\partial T}{\partial y} + \frac{\partial V}{\partial y} = Q \tag{1}$$

$$\ddot{y}_1 + \frac{k}{M} y_1 + \left(\frac{c_1}{M} + \frac{c_2 n_x}{M}\left(\frac{\partial \varphi}{\partial y_1}\right)^2 + \frac{c_3}{M}\left(\frac{\partial x}{\partial y_1}\right)^2\right)\dot{y}_1 + \frac{z}{M}\cos(\omega_0 t) = 0 \tag{2}$$

where,

$$\varphi = \varphi_1 + \varphi_2 ,\ \varphi_1 = \arctan\left(\frac{L_1 \sin(\theta_1) + \frac{y_1}{2n}}{L_1 \cos(\theta_2) - x_1}\right) - \theta_1 ,\ \varphi_2 = \arctan\left(\frac{L_2 \sin(\theta_1) + \frac{y_1}{2n}}{L_2 \cos(\theta_2) - x_2}\right) - \theta_2 ,\ x = x_1 + x_2 \tag{3,4,5,6}$$

$$x_1 = L_1(\cos(\theta_1) - \cos(\theta_1 + \varphi_1)) = L_1 \cos(\theta_1) - \sqrt{L_1^{\,2} - (L_1 \sin(\theta_1) + \frac{y_1}{2n})^2} \tag{7}$$

$$x_2 = L_2(\cos(\theta_2) - \cos(\theta_2 + \varphi_2)) = L_2 \cos(\theta_2) - \sqrt{L_2^{\,2} - (L_2 \sin(\theta_2) + \frac{y_1}{2n})^2} \tag{8}$$

Define $S_2 = (\frac{d\varphi}{dy_1})^2$ and $S_3 = (\frac{dx}{dy_1})^2$ then

$$S_2 = \left(\frac{d\left(\arctan\left(\frac{L_1 \sin(\theta_1) + \frac{y_1}{2n}}{L_1 \cos(\theta_2) - x_1}\right) - \theta_1 + \arctan\left(\frac{L_2 \sin(\theta_1) + \frac{y_1}{2n}}{L_2 \cos(\theta_2) - x_2}\right) - \theta_2\right)}{dy_1}\right)^2 \tag{9}$$

$$S_3 = \left(\frac{d\left(L_1 \cos(\theta_1) - \sqrt{L_1^{\,2} - (L_1 \sin(\theta_1) + \frac{y_1}{2n})^2} + L_2 \cos(\theta_2) - \sqrt{L_2^{\,2} - (L_2 \sin(\theta_2) + \frac{y_1}{2n})^2}\right)}{dy_1}\right)^2 \tag{10}$$

The dynamic equation (2) can be written as:

$$\ddot{y}_1 + \frac{k}{M} y_1 + \left(\frac{c_1}{M} + \frac{c_2 n_x}{M} S_2 + \frac{c_3}{M} S_3\right)\dot{y}_1 + \frac{z}{M}\cos(\omega_0 t) = 0 \tag{11}$$

Using the Taylor series expansion up to the 3th order for S_2 and S_3, s_2 and s_3 can be obtained as

$$s_2 = \xi_{21} + \xi_{22} y_1 + \xi_{23} y_1^{\,2} \ \ \&\ \ s_3 = \xi_{31} + \xi_{32} y_1 + \xi_{33} y_1^{\,2} \tag{12,13}$$

Define , $F_1 = c_1 / M, F_2 = c_2 n_x s_2 / M, F_3 = c_3 s_3 / M$, $z_0 = z / M$ then Eq. (11) can be written as:

$$\ddot{y}_1 + \frac{k}{M} y_1 + (F_1 + F_2 + F_3)\dot{y}_1 + z_0 \cos(\omega_0 t) = 0 \tag{14}$$

The detailed Taylor series expansion coefficients ξ_{ij} are omitted. From the Taylor expression of s_2 and s_3 it can be seen that both rotational and horizontal damping characteristics are dependent on the relative displacement y_1. This means that the damping of the system is adaptable to the vibration displacement. Moreover, all coefficients are functions of structural parameters, the desired damping coefficients can be obtained and designed by adjusting the structure parameters. The dimensionless dynamic equation can be written as

$$y_1'' + y_1 + \xi y_1' + z_0 \cos(\Omega \tau) = 0 \tag{15}$$

where $(\cdot)' = d(\cdot)/\mathrm{d}\tau$ and $\tau = \omega_1 t$, with other parameters as

$$\omega_1 = \sqrt{\frac{k}{M}}, \Omega = \frac{\omega_0}{\omega_1}, \xi = f_1 + f_2 + f_3, f_1 = \frac{F_1}{\omega_1}, f_2 = \frac{F_2}{\omega_1}, f_2 = \frac{F_2}{\omega_1} \tag{16}$$

3. The nonlinear damping with respect to structural parameters

Ideally, the damping should be adaptable to vibration amplitude as discussed before. That is, it is usually expected to be high at resonant frequencies but low at those frequencies of low vibration amplitude.

It is shown in Fig. 2(a, b) that, the damping terms f_2, f_3 and ξ increase significantly at around the resonant frequency where the vibration displacement amplitude is larger, but can be very small at other frequencies where the vibration displacement is small. This is exactly what is expected for an ideal damping effect. While in Fig. 2(c, d), the excitation amplitude z_0 increases to 0.1m. It is clear as the amplitude increases the equivalent damping coefficient increases as well in the resonant range.

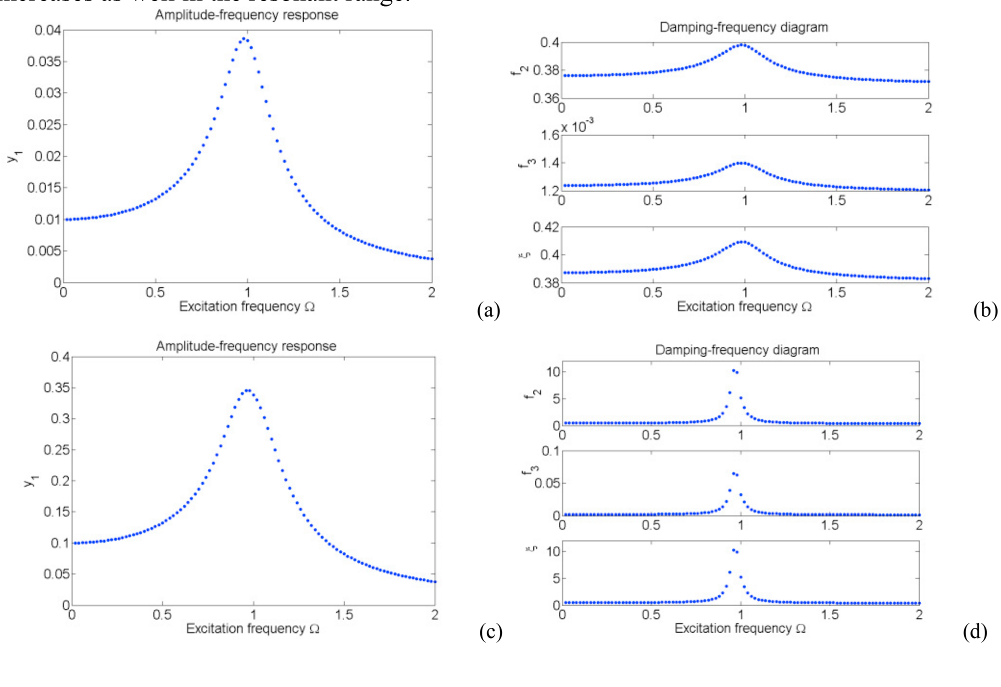

Fig. 2. Relationship between damping coefficients and excitation frequency with different z_0=0.01(a, b) and z_0=0.1(c, d) when θ1=π/4, n=2, L1=0.3 and L2=0.6; the amplitude-frequency response of Mass (a, c), damping-frequency diagram (b, d)

4. Vibration transmissibility with the nonlinear damping

This section is to demonstrate the transmissibility curves of a typical passive vibration isolation system by using the X-shaped structure as a nonlinear damping system to show its advantageous nonlinear benefits ad discussed above. With the standard HBM, the displacement transmissibility can be obtained. The structure can be installed with a horizontal linear damper with a coefficient c_3. Although it is a linear damping property, the equivalent damping via the structure will be completely nonlinear as discussed before. The comparison between a linear damping system and the nonlinear X-shaped structured damping is demonstrated in Fig. 3. The damping coefficient f_1 of the linear system is used with the same value in the X-shaped structure and all other parameters are also the same. From the transmissibility curves it is clear that the vibration isolation performance of the X-shaped structure is much better than that of a linear damping system. In the resonant frequency range, the peak of the vibration amplitude has been suppressed greatly with the structure compared with the linear damping system.

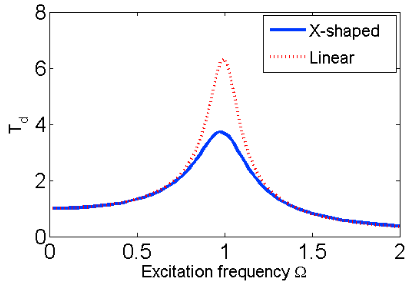

Fig. 3. Transmissibility curves with a linear damping system and the structure damping

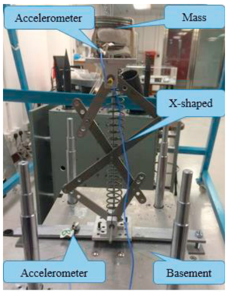

Fig. 4. Experimental setup

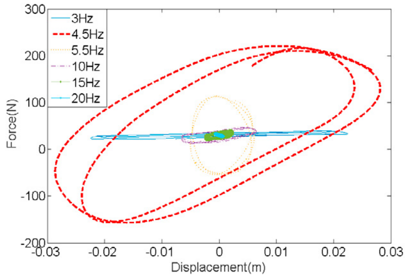

Fig. 5 Total force-displacement circles with different excitation frequencies

5. Experimental analysis

5.1 The experimental system

The experimental platform is shown in Fig. 4 where the X-shaped structure is fixed on a vibration platform connected with a shaker. The X-shaped structure damping system is constructed with aluminium links connected with ball bearings. The whole structure has two layers with rod length $L_1 = 0.1, L_2 = 0.075m$, and the assembly angle $\theta_1 = \pi/4$. The top mass M is 0.6kg. A spring with stiffness coefficient k=500 is installed in parallel within the structure. Theoretically, the system will have a resonance frequency around 4.6 Hz, which is verified by experiments.

5.2 Damping effect at different frequency

In steady-state forced vibration, the loss of energy is balanced by the energy that is supplied by the excitation. The force-displacement curve will enclose an area, referred to as the hysteresis loop, that is proportional to the energy lost per cycle. The energy dissipated per cycle is given by the area enclosed by an ellipse. By combing the damping force F_d and the restoring force ky of the loss-less spring, the hysteresis loop can be presented in Fig.5.

The frequency of the harmonic excitation is 3Hz, 4.5Hz, 5.5HZ, 10Hz, 15Hz, 20Hz, respectively. The amplitude of the excitation from the shaker is 0.5g in all the cases.

It can be clearly seen from the Fig. 5 that the area of the hysteresis curve at the resonant frequency is the biggest, and before or after the resonant frequency, the area of the hysteresis curves decreases obviously, which means the X-shaped structure has much higher energy dissipation capacity in the vicinity of the resonant frequency but lower at other frequencies, which is exactly the ideal nonlinear damping effect needed in practice, as discussed before.

6. Conclusions

Theoretical and experimental investigation into the nonlinear damping characteristics of the bio-inspired X-shaped structure is conducted in this paper. The results show that the damping of the X-shaped structure is nonlinear, and passively and beneficially adaptable to vibration displacement and frequency. It is unveiled that, the equivalent damping characteristic is a nonlinear function of the vibration displacement and varies at different excitation frequency (high damping at resonant frequency and low damping at high frequency range), showing an ideal, passive, and nonlinear damping property exactly needed in various vibration control practice. These results potentially for the first time reveal systematically such a beneficial nonlinear damping characteristic which can be realized with a simple limb-like structure in a pure passive manner, and will definitely bring benefits to a broad spectrum of engineering practice.

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