Attenuating nonlinear effects of pendulum tuned mass damper by an isochronous spring

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Abstract. An isochronous spring is introduced in a pendulum tuned mass damper (PTMD), with the aim of attenuating the detrimental nonlinear effects of PTMDs on vibration control. By letting the third-order term of nonlinear resilience generated by the pendulum and the isochronous spring be opposite, the nonlinear pendulum contribution can be greatly reduced. The frequency response function (FRF) of the primary structure is obtained via the harmonic balance method (HBM) combined with arc-length continuation. The results show that with a proper isochronous spring design, the PTMD response can be tailored to be linear up to 0.85 radians (about 49 degrees), which is much larger than the "linear" oscillation amplitude of the PTDM without the isochronous spring, thus ensuring the PTMD control performance also at large angles.

Introduction

Pendulum tuned mass dampers have been widely used in high-rise buildings to attenuate wind/seismic-induced vibrations[1]. Usually, PTMDs are treated as linear assuming small oscillation amplitude, while their nonlinear behaviour may have detrimental effects on the control performance when the controlled primary structure is subjected to strong excitations[2]. Although several studies addressed the PTMD with its nonlinearity[3], the external excitations are uncertain thus posing a challenge when the design targets a specific excitation intensity. Therefore, in this paper we propose to cope with the nonlinear effects of PTMDs introducing an isochronous spring, referred to as IE-PTMD, which is properly designed to attenuate the PTMD nonlinear effects at large rotation angles.

Problem formulation

A single-degree-of-freedom (SDOF) primary structure is charactered by mass m_1, stiffness k_1 and damping coefficient c_1, as shown in Figure1(a). Additionally, a PTMD with tip mass m_θ and damping coefficient c_θ is suspended by a pendulum rod with length l. At the same time, an isochronous spring with original length l_0 and stiffness k_2 is hinged between the pendulum rod and the primary structure, and the distance between the lower hinged and suspended points is denoted by l_2. In addition, f, ω and t are the excitation amplitude, frequency, and time, respectively. The equations of motion can be obtained using the Lagrangian as:

$$m_1\ddot{q}_1 + c_1\dot{q}_1 + k_1 q_1 + m_\theta \ddot{q}_1 + m_\theta l(\ddot{\theta}\cos\theta - \dot{\theta}^2\sin\theta) = f\cos(\omega t)$$

$$m_\theta l^2\ddot{\theta} + m_\theta l\cos\theta\ddot{q}_1 + c_\theta l^2\dot{\theta} + \left[m_\theta g l\sin\theta + k_2 \frac{l_2\sin\theta\left(l_0 - \sqrt{[l_0 - l_2(1-\cos\theta)]^2 + (l_2\sin\theta)^2}\right)\left(l_0 - l_2\right)}{\sqrt{[l_0 - l_2(1-\cos\theta)]^2 + (l_2\sin\theta)^2}} \right] = 0 \quad (1)$$

The squared term in Eq. (1) is expanded in Taylor series, and the third-order term is set to 0, so as to eliminate the main nonlinear term on the system response. The HBM combined with arc-length continuation is utilized to solve Eq. (1) and obtain the FRFs of the primary structure and pendulum, as shown in Figure1(b) and 1(c), respectively. It is found that, with the increase of excitation amplitude, the peak in the FRF of the primary structure with the attached PTMD increases significantly without exhibiting the two equal peaks corresponding to the in- and out-of-phase modes. On the other hand, the IE-PTMD has approximately two equal peaks, indicating much better control performance and nonlinearity attenuation capability. The FRFs of the pendulum have the same trend as that of the primary structure, and the rotation angle can reach 0.85 radians for IE-PTMD while without strong nonlinearities.

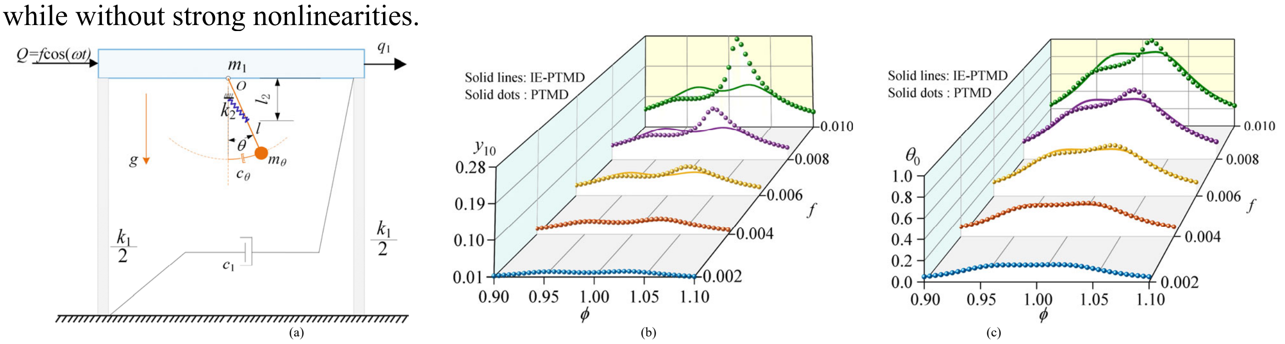

Figure 1: (a) Schematic of IE-PTMD attached to a SDOF primary structure, (b) FRF comparison of primary structure with/without isochronous spring and (c) FRF comparison of pendulum.

References

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