Nonlinear bending vibration isolation of sandwich beams

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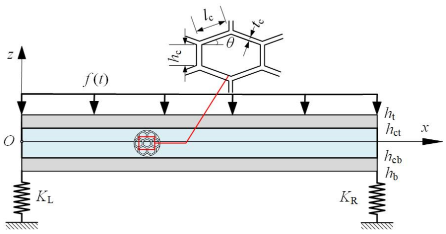
Abstract. Elastic laminated structure is one of the most common basic structures in the engineering applications. The geometric nonlinearity of the elastic beam cannot be ignored. Considering the geometric nonlinearity of beams is conducive to the accuracy of engineering analysis. Compared with the vibration transmissibility of the rigid boundary, the advantages of the elastic support boundary in vibration isolation performance are demonstrated. The transmission law of the parameters on the multi-mode resonant response on the elastic supported laminated beam model is analyzed.

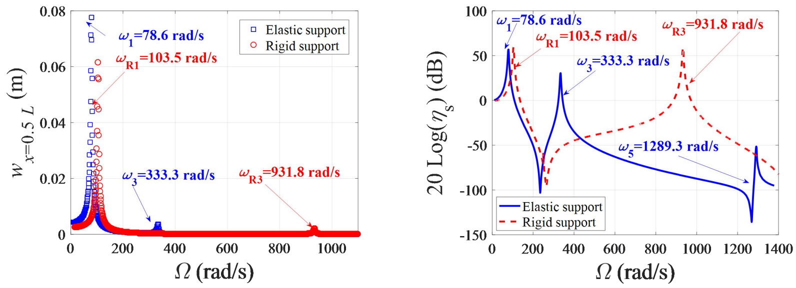
Introduction

Compared with the extensive research on the vibration characteristics, there are few studies on the passive vibration isolation of the honeycomb sandwich beam. The vibration isolation control scheme usually takes the continuum as the vibration isolator [1], rarely considers the geometric nonlinearity of the continuum structure itself. Engineers often use elastic supports to suppress vibration transmission. Research related to continuous vibration mainly focuses on the simple, fixed and free boundary, and the linear elasticity boundary . Ding et al. defined the vibration isolation transmission coefficient of elastic beam bending vibration [2]. Considering the geometric nonlinearity of the beam is conducive to the accuracy of the calculation results. The isolation control of its transverse multi-mode bending vibration is beneficial to the wide application of honeycomb sandwich beams.



(a)

Figure 1: Sandwich beam model supported by vertical springs at both ends.



(b) (c)

Figure 2: Influence of elastic support and rigid support on amplitude and transmissibility.

Results and discussion

Figure 1 shows the system schematic diagram. Mathematical model of the system as

$$\rho A\int_0^L \sum_{n=1}^{N}\phi_n(x)\ddot{q}_n(t)\varphi_m(x)\,\mathrm{d}x + EI\int_0^L \sum_{n=1}^{N}\phi_n''''(x)q_n(t)\varphi_m(x)\,\mathrm{d}x + \int_0^L F\cos(\Omega t)\varphi_m(x)\,\mathrm{d}x$$

$$+\alpha\int_0^L \sum_{n=1}^{N}\phi_n(x)\dot{q}_n(t)\varphi_m(x)\,\mathrm{d}x = \frac{1}{2}EA\int_0^L \sum_{n=1}^{N}\phi_n''(x)q_n(t)\varphi_m(x)\,\mathrm{d}x\int_0^L \left\{\sum_{n=1}^{N}\left[q_n(t)\varphi_n'(x)\right]\right\}^2 \mathrm{d}x,$$

E is the Young's modulus of elastic beam. A is the cross-sectional area, ρ is the total material density. For the energy dissipation of transverse vibration, external damping is introduced α. $\varphi(x)$ and $q(t)$ respectively represent the modal function of the beam and its corresponding generalized coordinates. Harmonic balance method and Runge-Kutta method are used to verify the correctness of the calculation results. Figure 2 shows the influence of elastic support and rigid support on the amplitude response and vibration transmissibility of the beam. For rigid supports, the vibration transmissibility at the third resonance is higher than that at the first resonance. The spring support on the boundary can effectively reduce the vibration transmissibility at the resonance of the laminated beam, especially in the high-frequency region. It should be noted that although the nonlinear characteristics in the case are not very obvious, geometric nonlinearity cannot be ignored to ensure the accuracy of the calculation results.

References

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