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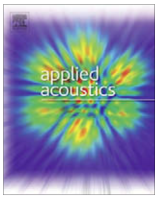

Research Paper

Attenuation of flexural vibration for floating floor and floating box induced by ground vibration

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ABSTRACT

This paper investigates the vibration isolation performance of floating floor and floating box structures to control rail vibration transmission. Simple theoretical and experimental methods are developed to analyze the effects of stiffener beam, mass and arrangement of isolator on the fundamental natural frequency of the flexural vibration of floating floor and box structure.

The vibration reduction performances of floating floor and box structure are found to be degraded by flexural vibration of the floor or supporting stiffener beam. From the results of vibration measurements; stiffener beams increase the fundamental natural frequency of flexural vibration of floating floor and enhance vibration isolation. Also they can further alleviate the effect of flexural vibration using optimum isolator arrangement effectively. The proposed floating box design achieved a vibration reduction of 15–30 dB in frequency region of critical rail vibration (30–200 Hz).

1. Introduction

When a noise sensitive receiver is built after the establishment of a rail track, the buildings have to be isolated. Floating floor and box-within-box structure are the common vibration isolation designs to decrease the energy transmission from the base to the floating floor. The selection of isolator material, size of floating floor and position of isolators should be less restrictive than those for floating slab on rail track, which are designed with stringent limitation, maximum allowable static track deflection and size of floating slab.

The vibration spectrum generated by rail traffic is dominated by a low frequency range below 200 Hz with the peaks coming at around 40–80 Hz [1–3]. The fundamental natural frequency of the mass–spring system of an isolated building design for rail vibration control is usually lower than 14 Hz [4]. In practice, however, Saurenman et al. [5] point out that other than the first natural frequency of the mass-spring system, vibration isolation also depends on the wave motion of slab support, flexural vibration of floating slabs and other secondary effects to prevent reduction higher than 20 dB. Furthermore, based on the predication by Southward and Cooper [6], the rumbling noise radiation is mainly contributed by the local modes of the vibrating box structure.

The sound reduction requirements for test rooms are usually recommended to be above 100 Hz according to ISO 140 series [7], thus most designs in floating floor have insufficient vibration attenuation below 100 Hz. Recent researches on floating floor [8,9] have more concern on vibration control on low frequency range below 100 Hz. According to ISO 2631 [10], the vibration below 80 Hz can result in adverse effects on human health, comfort and perception. Mirowska [11] confirms that the increase in depression, anxiety, fatigue, insomnia, and heat ailment are due to the low frequency structure-borne noise.

The critical design parameters of floating floors include stiffness, dimension, mass of the floating floor (Jutulstand [12]; Baron [13] and Jeon and Yoo [14]), and the position of isolators (Mead [15] and Yan and Xie [16]). Jeon and Yoo [14] attached fiberglass reinforced plastic beams beneath the slab to investigate the effects of structural stiffness of concrete slabs on the floor's low frequency vibration due to impact. They noted that the installation of those beams can decrease impact noise and vibration. Wilson [4] suggests a new construction and design method for an isolated theater located at the fifth level in an existing relatively lightweight steel frame structure. One of the design requirements is that the natural frequency of the flexural vibration of the theater floor slab should be higher than 15 Hz to reduce coupling flexural vibration modes of the theater floor slab and the supporting structures. The control of structure parameters can change the natural frequency of flexural vibration that could be significant for structure-radiated sound.

The position of isolator are generally at the corners of the floating floor, there is little optimization of isolator position to reduce the critical flexural vibration mode of the floating floor. The important design feature of isolator position is rarely mentioned in previous study. Hui and Ng [17] suggests that soft isolators

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should be placed at nodal points of the first flexural vibration mode of floating panel. This method was applied in small square isolated stiff and light weight honeycomb panel and vibration reduction was up to 40 dB in 100–500 Hz. The nodalization concept, however, have not been applied in common building structures of large rectangular floor and isolated box to reduce rail vibration critical in low frequency range, which will be discussed in this paper.

In recent years, a further development has been to isolate a box building to alleviate vibration transmission from train. For example, the two-storey isolated box design is constructed underneath a rail viaduct of ballast track. The designs of stiffener beam and isolator arrangement are investigated in order to alleviate the vibration transmission.

2. Transmissibility of isolation system

In this paper, the dynamic behaviours of floating floor and isolated box structures will be examined. The analytical formulae of structure behaviour, finite plate and box structures are assumed to be linear. Vibration isolation performance of floating floor and isolated box can be calculated with multi-degree of freedom equation based on modal analysis.

2.1. Basic modal analysis

For a general MDOF system, with N degrees of freedom, the governing equation of motions in matrix form is shown in Eq. (1a).

$$[M]\{\ddot{x}\} + [C]\{\dot{x}\} + [K]\{x\} = \{f(t)\} \tag{1a}$$

where [M] is a mass matrix; $[K]$ is a stiffness matrix; and $[C]$ is a viscous damping matrix; $\{x\}$ is displacement vector; $\{f(t)\}$ is force vector various with time.

Mode shape matrix precedes the normalization process to simplify the mass, damping to stiffness matrix to be diagonal matrix, and $\{x\} = [\Psi]\{q\}$, $\{q\}$ is modal displacement.

A simpler matrix $[\diagdown m_r \diagdown]\{\ddot{q}\} + [\diagdown c_r \diagdown]\{\dot{q}\} + [\diagdown k_r \diagdown]\{q\} = [\Psi]^T\{f\}$ can be obtained and the rth individual equation is: $m_r\ddot{q}_r + c_r\dot{q}_r + k_r q_r = f_k\psi_{kr}$

$$q_r = \frac{f_k\psi_{kr}}{(k_r - \omega^2 m_r) + j(\omega c_r)} \quad \text{and} \quad \psi_{ir}q_r = x_i \tag{1b}$$

Then $\omega_r^2 = \frac{k_r}{m_r}$ and $\zeta_r = \frac{c_r}{2\sqrt{k_r m_r}}$ can be derived, ζ_r is the rth damping ratio. In this paper, structural damping η is modelled as equalivant viscous damping using the approximation of $\zeta_r = 0.5\eta$.

The individual general receptance frequency response function (FRF) in terms of mass, damping and stiffness elements can be established as shown in Eq. (2):

$$\alpha_{ik}(\omega) = \sum_{r=1}^{N} \frac{(\psi_{ir})(\psi_{kr})}{(k_r - \omega^2 m_r) + j(\omega c_r)} \tag{2}$$

0where $\alpha_{ik} = (\frac{x_i}{f_k}) = \alpha_{ki} = (\frac{x_k}{f_i})$, ψ_{ir} is the ith element of the rth eigenvector $\{\psi\}_r$, and ψ_{kr} is the kth element of the rth eigenvector.

Simplification of the Eq. (2) is given in Eq. (3) in Section 2.2, which is used for simple structure of beam and plate.

2.2. Theoretical analysis of nodalization

A general isolation system were initially derived as a rigid body motion, the matrix approach was then developed on the basis of the theory of modal analysis. The system was simplified as the problem of a rigid body supported with isolators which consist of spring and damping elements.

According to modal analysis theory used to identify the ratio of vibration velocity on the foundation with the isolator to that without isolation (ref. Fig. 1). The equation for vibration transmissibility has been derived in Eq. (3) for the effects of resonance frequency of flexure vibration. This equation can be applied for mass-spring system on a rigid foundation.

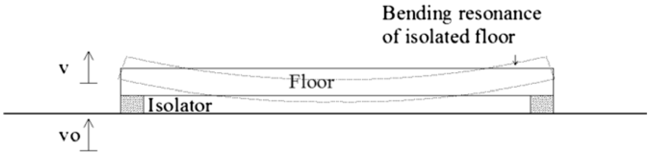

Fig. 1. Schematic of vibration transmission.

$$\text{Vibration transmissibility (Tr)} = \frac{V}{\text{Vo}} = \left|\frac{Z_m}{Z_i} + 1\right|^{-1} \tag{3}$$

where, $Z_i = \frac{k_o}{j\omega} + c_O, Z_m = \left(\frac{1}{m_o j\omega} + \sum_{r=1}^{N} \frac{R_r}{\frac{k_r}{j\omega} + m_r j\omega + c_r}\right)^{-1}$

V	the velocity of the floating slab,
Vo	the velocity of the base structure,
Z_m	the impedance of the floating slab,
Z_i	the impedance of the isolator,
k_o	the stiffness of isolator,
m_o	the mass of slab,
m_r	the modal mass of rth resonance frequency of flexure vibration,
c_o	the damping of isolator,
c_r	the modal damping of rth resonance frequency of flexure vibration, and
R_r	the mode shape factor of rth resonance frequency of flexure vibration.

On the basis of Eq. (3), the magnitudes of transmissibility at the resonance of isolator and at the resonance of first flexural vibration of the slab are shown in Eqs. (4) and (5).

$$\text{At the resonance of isolator: } \mathrm{Tr}\ (\omega_o) = \frac{1}{2\zeta_o} \tag{4}$$

where, ζ_o is the damping of isolator

At the first symmetric flexure vibration of slab :

$$\mathrm{Tr}(\omega_1) = B_1 \frac{R_1}{\zeta_1\gamma_1^2} \tag{5}$$

where, $\gamma_1 = \frac{\omega_1}{\omega_o}$ and $\mathrm{B}_1 = \frac{m_1}{m_o}$ is dependent on the modal mass.

The effect of resonance frequency of flexure vibration depends on the mode shape factor and the frequency ratio as given by Eq. (5). Thus, to alleviate the effect of the first symmetric resonance of flexural vibration, efficient design should be achieved by decreasing mode shape factor and increasing the frequency ratio of resonance of flexural vibration and rigid body resonance.

The effects of resonances on vibration isolation are considered and listed in Table 1. Fig. 2b shows the theoretical transmissibility using Eq. (3) when the isolators are placed at the corners and the nodal points of the first symmetric flexural vibration mode of a 1.6 m (L) × 0.8 m (W) × 0.02 m (H) wooden panel as illustrated in Fig. 2a. When the isolators are placed at four corners, the resonance peak of flexural vibration occurs at 42 Hz. As the transmissibility at 42 Hz depends on the mode shape factor R_1 of first symmetric flexural vibration mode (ref to Eq. (5)), it should be minimized (assumed to be zero) when the isolators are placed at the nodal points. Thus, the peak is attenuated significantly when the isolators are placed at the nodal points of the first flexural vibration mode.

3. Performance of conventional floating floor

Vibration analysis was performed in an existing floating floor. The vibration transmissibility levels and mode shapes from site test were verified with computer Finite Element Model (FEM).