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Assessment of tool platform micro vibrations induced by moving vehicles in hi-tech factories

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Abstract. This study explores the micro vibrations of floors and tool platforms induced by internal moving automated guided vehicles (AGVs) in hi-tech factories. The equation of motion of a simplified multi-span floor (or beam) system installed with a tool platform under AGV moving forces generated by a modified Kanai-Tajimi power spectral density (MKT-PSD) function is derived. Dynamic time history analyses of the continuous beam model travelled by different AGV weights are performed. The corresponding root-mean-square (RMS) floor and platform vibration spectra are obtained by using a one-third octave band spectrum analysis and in turn used to compare with the micro vibration criterion. Simulated results indicated that the floor and platform vibrations increase with the AGV weight. For the maximum AGV weight considered, the floor vibration exceeds VC-A, while under the minimum AGV weight, the floor vibration reaches VC-B which is required by the vibration-sensitive tools. Moreover, the platform vibrations are far more than VC-A regardless of the AGV weight. Therefore, the introduction of a proper vibration control scheme is recommended to suppress the excessive micro vibrations of the floors and taller tool platforms.

1. Introduction

The fabrication process of silicon wafers and glass panels (or liquid crystal displays, LCDs) with micro or nanometer feature size is susceptible to micro vibrations induced by a variety of disturbance sources. Vibrations directly caused by automated guided vehicles (AGVs) [1] or moving cranes [2] on the floors in hi-tech factories have rarely been studied when designing facilities. Consequently, most of the LCD factories encounter AGV-induced floor and platform vibrations that exceed the desired micro vibration criterion [3]. The LCD factory with thicker RC “cheese” slabs supported by a long span mega truss considered in this study is illustrated in Figure 1. A simplified continuous floor (or beam) model consisting three spans installed with a tool (or equipment) platform at the mid span will be used to quickly simulate the floor-platform system [1].

This study intends to assess the floor and platform micro vibrations caused by internal AGVs in hi-tech LCD factories. The AGV moving force time histories of five different AGV weights with a speed of 2.0 m/s were simulated by using a modified Kanai-Tajimi power spectral density (MKT-PSD) function [4]. Dynamic time history analyses of a simplified three span continuous floor system installed with a platform under AGV moving forces were first carried out by using the state space procedure (SSP) [5]. The corresponding root-mean-square (RMS) micro vibration spectra of the floor and platform were then obtained by performing a one-third octave band analysis [4]. Finally, the vibration levels of floor and platform under different AGV moving conditions are assessed and discussed.

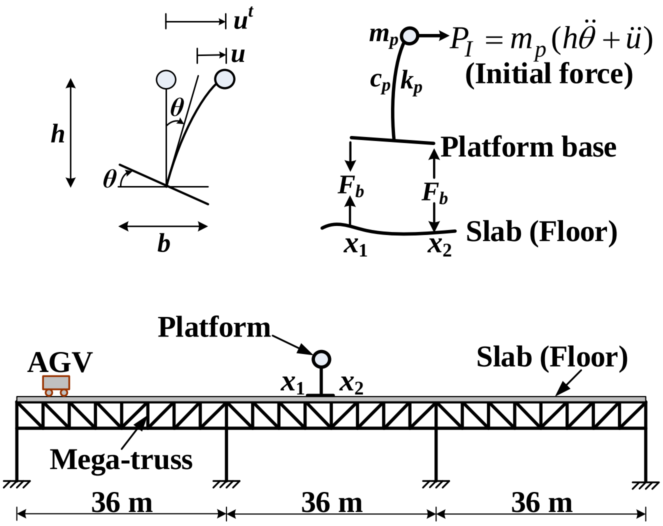

Figure 1. Floor system of an LCD factory and numerical model of the platform.

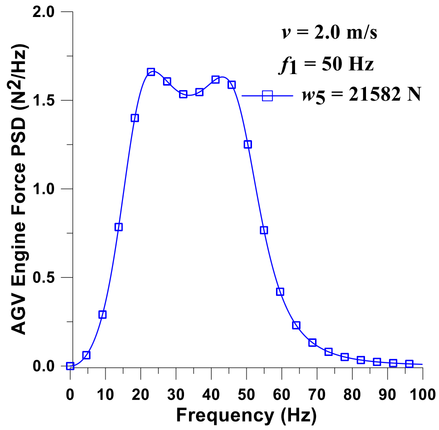

Figure 2. Dynamic engine force spectrum.

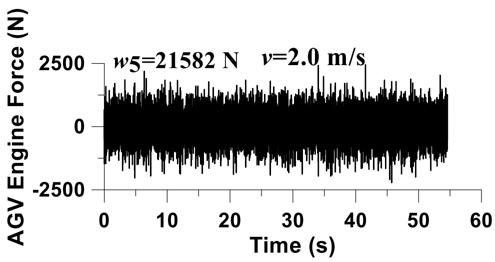

Figure 3. Dynamic engine force time history.

2. Numerical model of the floor system and platform

The vertical displacement, $w(x,t)$, of a continuous beam (or floor) system that considers the lowest n vibrational modes can be calculated by using the classical mode superposition method [6] as follows:

$$w(x,t) = \sum_{i=1}^{n} q_i(x)\phi_i(x) \tag{1}$$

in which $q_i(t)$ is the i-th modal coordinate and $\phi_i(x)$ denotes the corresponding i-th mode shape function [7]. Moreover, the equation of motion of the multi-span continuous beam travelled by N moving axle loads can be represented as [1, 4]:

$$\mathbf{M}\ddot{\mathbf{q}}(t) + \mathbf{C}\dot{\mathbf{q}}(t) + \mathbf{K}\mathbf{q}(t) = \mathbf{E}\mathbf{w}(t) \tag{2}$$

in which $\mathbf{M} = [m_{ij}]$ and $\mathbf{K} = [k_{ij}]$ are respectively the $n \times n$ mass and stiffness system matrices with coefficients calculated as [7]:

$$m_{ij} = \int_0^L \rho A \phi_i(x)\phi_j(x)dx \qquad k_{ij} = \int_0^L EI\phi_i''(x)\phi_j''(x)dx \tag{3}$$

in which ρ is the density, E is the Young's modulus, A is the cross-sectional area, I is the moment of inertia of the cross-section, $\phi_i''(x)$ represents the curvature function of the beam and $\mathbf{C}$ is the $n \times n$ damping matrix. Furthermore, $\mathbf{q}(t) = \begin{bmatrix} q_1(t) & q_2(t) & \cdots & q_n(t) \end{bmatrix}^{\mathrm{T}}$ is the $n \times 1$ modal displacement vector, $\mathbf{w}(t) = \begin{bmatrix} F_1(t) & F_2(t) & \cdots & F_N(t) \end{bmatrix}^{\mathrm{T}}$ is the $N \times 1$ moving load vector, and

$$\mathbf{E} = \begin{bmatrix} \phi_1[x_{F1}(t)] & \phi_1[x_{F2}(t)] & \cdots & \phi_1[x_{FN}(t)] \\ \phi_2[x_{F1}(t)] & \phi_2[x_{F2}(t)] & \cdots & \phi_2[x_{FN}(t)] \\ \vdots & \vdots & \vdots & \vdots \\ \phi_n[x_{F1}(t)] & \phi_n[x_{F2}(t)] & \cdots & \phi_n[x_{FN}(t)] \end{bmatrix} \tag{4}$$

is the $n \times N$ influence matrix of moving axle loads that vary with the locations of the vehicles.

If a platform with the width of b and height of h is mounted on the floor of the factory within the locations of x_1 and x_2 (Figure 1), its equation of motion along the horizontal direction under the rotational base excitation ($\theta(t)$) can be represented as [6]:

$$m_p \ddot{u}(t) + c_p \dot{u}(t) + k_p u(t) = -m_p h \ddot{\theta}(t) \tag{5}$$

in which m_p, c_p and k_p are respectively the mass, damping and stiffness parameters of the platform simulated as a single-degree-of-freedom (SDOF) system in this study. The angular base acceleration is further represented by using the relative vertical floor acceleration located at x_1 and x_2 as:

$$\ddot{\theta}(t) = [\ddot{w}(x_2,t) - \ddot{w}(x_1,t)]/b = (\mathbf{\Phi}_1^{\mathrm{T}} - \mathbf{\Phi}_2^{\mathrm{T}})\ddot{\mathbf{q}}(t)/b \tag{6}$$

in which $\mathbf{\Phi}_1^{\mathrm{T}} = \begin{bmatrix} \phi_1(x_1) & \phi_2(x_1) & \cdots & \phi_n(x_1) \end{bmatrix}$ and $\mathbf{\Phi}_2^{\mathrm{T}} = \begin{bmatrix} \phi_1(x_2) & \phi_2(x_2) & \cdots & \phi_n(x_2) \end{bmatrix}$ are the row vectors that contain the mode shape values of each mode at x_1 and x_2 respectively. Moreover, the resisting coupling force ($F_b(t)$) between the floor and the base of platform induced by the initial force of the platform can be derived with equation (6) as:

$$F_b(t) = m_p \ddot{u}^t(t)h/b = m_p[h\ddot{\theta}(t) + \ddot{u}(t)]h/b = \begin{bmatrix} \dfrac{m_p h^2}{b^2}(\mathbf{\Phi}_2^{\mathrm{T}} - \mathbf{\Phi}_1^{\mathrm{T}}) & \dfrac{m_p h}{b} \end{bmatrix} \begin{bmatrix} \ddot{\mathbf{q}}(t) \\ \ddot{u}(t) \end{bmatrix} \tag{7}$$

The modal coupling force ($F_b^*(t)$) exerted on the floor system at x_1 and x_2 can be represented as:

$$F_b^*(t) = (-\mathbf{\Phi}_1 + \mathbf{\Phi}_2)F_b(t) = \left\{ (\mathbf{\Phi}_2 - \mathbf{\Phi}_1) \begin{bmatrix} \dfrac{m_p h^2}{b^2}(\mathbf{\Phi}_2^{\mathrm{T}} - \mathbf{\Phi}_1^{\mathrm{T}}) & \dfrac{m_p h}{b} \end{bmatrix} \right\} \begin{bmatrix} \ddot{\mathbf{q}}(t) \\ \ddot{u}(t) \end{bmatrix} = \left\{ \bar{\mathbf{M}} \right\} \begin{bmatrix} \ddot{\mathbf{q}}(t) \\ \ddot{u}(t) \end{bmatrix} \tag{8}$$

The equation of motion of the floor equipped with a platform under the AGV moving loads can be further represented by using equations (2), (5), (6) and (8) as:

$$\left(\begin{bmatrix} \mathbf{M} & \mathbf{0} \\ \mathbf{0} & m_p \end{bmatrix} - \begin{bmatrix} \multicolumn{2}{c}{\bar{\mathbf{M}}} \\ -m_p \dfrac{h}{b}(\mathbf{\Phi}_2^{\mathrm{T}} - \mathbf{\Phi}_1^{\mathrm{T}}) & 0 \end{bmatrix} \right) \begin{bmatrix} \ddot{\mathbf{q}}(t) \\ \ddot{u}(t) \end{bmatrix} + \begin{bmatrix} \mathbf{C} & \mathbf{0} \\ \mathbf{0} & c_p \end{bmatrix} \begin{bmatrix} \dot{\mathbf{q}}(t) \\ \dot{u}(t) \end{bmatrix} + \begin{bmatrix} \mathbf{K} & \mathbf{0} \\ \mathbf{0} & k_p \end{bmatrix} \begin{bmatrix} \mathbf{q}(t) \\ u(t) \end{bmatrix} = \begin{bmatrix} \mathbf{E} \\ \mathbf{0} \end{bmatrix} \mathbf{w}(t) \tag{9}$$

The numerical solution for equation (9) will be recursively obtained at each time step by using a state space integration procedure [5].

3. Simulation results

The system parameters of a real LCD factory adopted in this study are $\rho A = 9.1505\times10^3$ kg/m and $EI = 2.4608\times10^{11}$ N-m^2 which result in a vertical fundamental frequency of 6.285 Hz [8]. All the equivalent damping ratios of the lowest 12 modes of the floor system are assumed to be 5%. Moreover, the system parameters of the platform mounted at x=54 m with a mass of 2500 kg, the natural vibration frequency of 25 Hz and the equivalent damping ratio of 2% are given in this numerical example. The major dimensions of the platform are b=2.0 m and h=2.6 m.

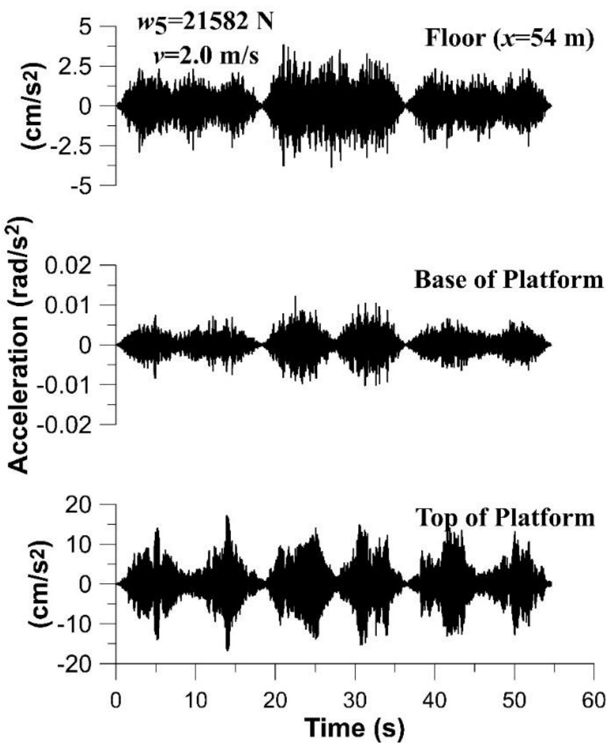

Figure 4. Simulated dynamic time history responses of floor system and platform.

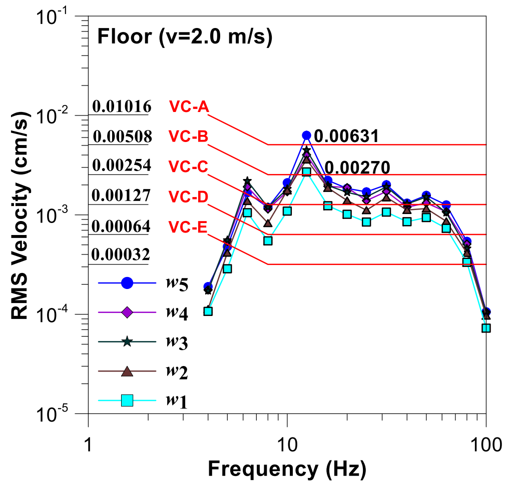

Figure 5. RMS floor vibration spectra.

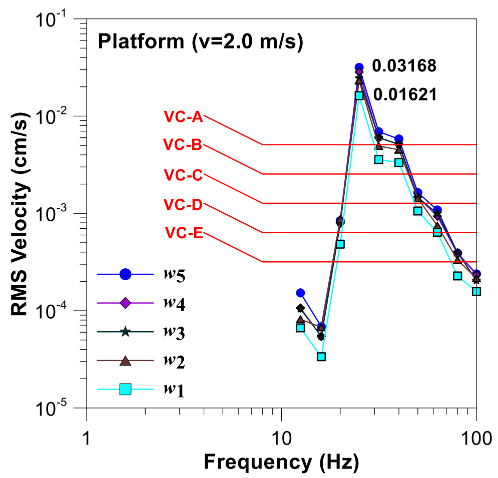

Figure 6. RMS platform vibration spectra.

In order to perform the assessment of the floor and platform micro vibrations under a single AGV (N=2, i.e. two axle loads) with different weights, five AGV weights with $w_5 = 1.0W$ = 21582 N, $w_4 = 0.8W$, $w_3 = 0.6W$, $w_2 = 0.4W$ and $w_1 = 0.2W$ are considered in this study. Due to the length limitation of the paper, only the simulated dynamic engine force (moving force) MKT-PSD spectrum and its corresponding dynamic engine force time history for an upper bound frequency of f_1=50 Hz and the maximum AGV weight (w_5) with v=2.0 m/s are shown in Figures 2 and 3 respectively [4].

Figure 4 shows the simulated vertical acceleration of the floor system (at x=54 m), the angular base acceleration of the platform and the absolute horizontal acceleration of the platform under the maximum AGV weight (w_5). The corresponding RMS floor and platform vibration spectra under different AGV weights are shown in Figures 5 and 6 respectively. The floor vibrations, in general, increase with the weight of the AGV. For the maximum AGV weight (w_5), the floor vibration exceeds VC-A, while under the minimum AGV weight (w_1), the floor vibration reaches VC-B. Moreover, the platform vibrations are far more than VC-A regardless of the AGV weight, and the vibration amplification ratios defined as the ratio of the maximum RMS platform vibration to the maximum RMS floor vibration may reach as high as 5.0 to 7.0 (at around 25 Hz). Therefore, the vibrations of the floors and taller tool platforms adopted by the LCD factory for the large-sized glass panels usually exceed the desired vibration level (VC-B) and the vibration isolation or control devices (dampers) may be introduced to suppress the excessive micro vibrations.

4. Concluding remarks

This study has explored AGV induced micro vibrations of tool platforms installed on a three span continuous beam model simplified from a real LCD factory. Time history analyses of the beam model subjected to a pair of moving forces simulated by the MKT-PSD function considering five different AGV weights with v=2.0 m/s are performed. The corresponding RMS floor and platform vibration spectra are obtained by using a one-third octave band analysis and further used to compare with the micro vibration criterion. Numerical simulations indicated that the floor and platform vibrations increase with the AGV weight. The floor vibration exceeds VC-A under the maximum AGV weight, while it reaches VC-B (desired vibration level) under the minimum AGV weight. Moreover, the platform vibrations are far more than VC-A regardless of the AGV weight. Therefore, the feasibility of using the vibration control schemes to suppress the excessive micro vibrations of the floors and taller platforms under different AGV weights and vehicle speeds may be evaluated in the future study.

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