Investigation of vibration induced by moving cranes in high-tech factories

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Abstract
A finite element model was developed to simulate the crane induced vibration on the floor of high-tech factories, in which the mesh include beam, plate, spring-damper, and moving wheel elements. The finite element results were first compared with the experimental measurements in good agreement. The parametric studies were then performed to study the vibration behavior of high-tech factories due to the effects of rail irregularities, slab depth, and crane speed. The rail irregularities induce the vibration of the crane and slab at their natural frequencies, and both rail irregularities and the crane acceleration induce the crane rotation in its natural frequency, so that smoothing the wheel and rail should be the first priority to decrease slab vibration. The crane speed is another important issue to influence slab vibration, which decreases with the reduction of the crane speed clearly from the parametric study. Thus, decreasing the crane speed to reduce slab vibration is an alternative, and experiments are caused to find the optimal crane speed and acceleration. The crane induced vibration is the relatively largest and smallest at the column location and beam center, respectively. Therefore, increasing the slab and beam depth to decrease the slab vibration induced by the moving crane is an additional option.

Keywords
Ambient, crane, finite element method, high-tech factory, moving wheel element, one-third octave band, rail irregularity, vibration

Introduction
High-tech equipment used for the production of semiconductors and optical microscopes requires strict micro-vibration criteria. Floor vibration induced by moving vehicles, such as cranes inside the building and moving trucks nearby the building, is the major source of environmental loads that affect the operation in high-tech factories. There are a number of investigations in the literature to study this topic. Park et al. studied the motion of a Bernoulli-Euler cantilever beam clamped on a moving cart, and the results matched well with physical phenomena. Ngo and Hong discussed a sliding-mode control for a crane, and the proposed mechanism can suppress lateral sway. Wang et al. investigated the seismic behavior of the automation stocker system, and found that the speed type viscous dampers can reduce stocker acceleration and displacement to avoid cassette sliding and collision. Lee et al. proposed a sub-structural model to explore the floor micro-vibration induced by automated guided vehicles. They indicated that energy dissipation devices can be effective in vibration mitigations. Lee et al. explored vertical floor micro-vibration induced by the automated guided vehicles. They indicated that the energy-dissipation device is effective in vibration suppression.

Some papers studied the platform improvement to reduce vibration for high-tech buildings. Yang and Xu studied an active platform to keep high tech equipment free from micro-vibration induced by traffic disturbance,
where a platform was installed on a building floor through an active control actuator and a passive mount to stop floor vibration to transfer to the platform. Xu et al.\textsuperscript{7} performed experiments to demonstrate a hybrid platform to reduce the vibration induced by the nearby traffic, and the hybrid platform was mounted on the building floor through leaf springs and oil dampers and controlled actively by an actuator. Yang et al.\textsuperscript{8} established an analytical model for hybrid control of the building-platform system subject to ground motion. The hybrid control is effective in reducing both velocity response and drift of the high-tech equipment. Xu et al.\textsuperscript{9} used a control platform to isolate building vibration induced by nearby traffic ground motion, and they found that the actively controlled platform is superior to the passively controlled platform and passive base isolators. Choi et al.\textsuperscript{10} investigated an active vibration control of a translating tensioned steel strip in the zinc galvanizing line, while the effectiveness of the control laws proposed was demonstrated via simulations. Guo et al.\textsuperscript{11} used hybrid control platform to mitigate the vertical micro-vibration of sensitive equipment installed in high-tech building subjected to vehicle-induced ground motion. Results showed that the performance of hybrid control platform is superior to that of passive control platform. Guo et al.\textsuperscript{12} investigated a hybrid control platform to isolate micro-vibration due to vehicle-induced ground motions. They indicated that the hybrid control platform can effectively reduce both horizontal and vertical micro-vibration. Wang and Liu\textsuperscript{13} investigated the use of a micro-vibration hybrid control system, and they indicated that the platform with the hybrid control system for micro-vibration is better than that with passive control system. Jang et al.\textsuperscript{14} proposed a hybrid mount system with air springs and piezo-stack actuators that can deliver excellent performance in the micro-vibration control of high-tech facilities. Utku et al.\textsuperscript{15} presented a numerical and experimental study on an active vibration control system. Results showed that this method can suppress the vibration amplitudes at multiple modes of the structure.

A number of studies used isolators or dampers to reduce vibration for high-tech buildings.\textsuperscript{16–23} Hwang et al.\textsuperscript{16} studied the seismic protective systems for high-tech structures, and found that the incorporation of viscous dampers not only enhance the seismic safety but also minimize the micro-vibration of the structure. Lin et al.\textsuperscript{17} proposed a micro vibration mitigation system using viscous dampers for high-tech buildings, and results presented that the vibration can be effectively captured by the viscous damper and converted to lower frequency-content tremors. Shih et al.\textsuperscript{18} studied the effect of different damper for high-tech buildings, and found the Velocity and Displacement Hydraulic Damper has superior vibration-reduction capability in reducing vibration. Turnip and Hong\textsuperscript{19} studied a sequential quadratic programming method for determining the optimal damping coefficients of a semi-active suspension system. Simulation results showed that the system significantly improves the ride comfort and reduces the noise and harshness. Alqado et al.\textsuperscript{20} presented an approach on the control of structures with magnetorheological dampers, and the effectiveness of the proposed scheme was compared with those achieved by classical and well-established alternative control schemes. Shah et al.\textsuperscript{21} studied a residual vibration control problem of the refueling machine that transports fuel rods in water of the nuclear reactor. The simulation and experimental results showed that the proposed method can effectively suppress the vibrations of the flexible rod operating in water. Zamani et al.\textsuperscript{22} designed a fractional order PID controller to adjust the contact force of piezoelectric friction dampers for base-isolated structures during earthquakes, while they indicated that the controller is superior to several well-known control techniques. Mishra et al.\textsuperscript{23} developed a scheme based on the envelope analysis to extract the fault-related symptoms from noisy vibration signatures of defective ball bearings, and this scheme was tested using the experimental data collected from a machine fault simulator system.

Some papers investigated the vibration problems of high-tech buildings due to nearby traffic.\textsuperscript{24–28} Hu and Xiong\textsuperscript{24} analyzed the traffic-induced ground motion for structures, and evaluated the micro-vibration behavior in structures. Xu and Hong\textsuperscript{25} presented a framework for quantifying traffic-induced building vibration in a stochastic way, and the results showed that traffic-induced ground vibration impedes the normal operation of the high-tech facility. Ismail et al.\textsuperscript{26} presented a rolling-based seismic isolation bearing for motion-sensitive equipment protection. The numerical results reveal that the proposed isolator can attenuate seismic responses under different ground motion excitations. Arnaud et al.\textsuperscript{27} reviewed the improvement of the quality of buildings regarding sound insulation and reducing noise exposure, and this review presented a state-of-the-art of annoyance due to a combined exposure to vibrations and noise. Wang et al.\textsuperscript{28} developed a traffic noise propagation calculation method to simulate noise attenuation, and results of experiment and case study showed the error to be no more than 0.3 dB when the building data were appropriately set.

Rail irregularities or road roughness are the major vibration source of moving vehicles in high-tech buildings. This topic was well studied in the field of moving train problems,\textsuperscript{29–32} but loads moving on the high-tech floor without rail irregularities were often used in the numerical method to simulate the crane induced vibration in the literature. This study thus used finite element analyses with the rail irregularity effect to perform parametric study.
for micro-vibration induced by moving cranes. Moreover, experiments were used to validate the numerical result, and several schemes were also proposed to reduce the crane-induced vibration in high-tech factories.

**Illustration of the structure for high-tech factories**

**Vibration standard in high-tech factories using the one-third octave band**

The first step for the standard is selecting a velocity record, \( y(t) \), to analyze by the Fast Fourier Transform (FFT), and calculate the power spectrum density function, \( S_y(f) \)

\[
S_y(f) = \frac{2|Y(f)|^2}{T} \tag{1}
\]

where \(|Y(f)|\) is the FFT amplitude, \( T \) is the period of \( y(t) \), and \( f \) is the frequency (Hz). Then, calculate the root mean square velocity level (\( L_y(f_c) \)) presented in decibel (dB)

\[
L_y(f_c) = 20 \log_{10} \left( \frac{\int_{f_l}^{f_u} S_y(f) df}{\sigma_0} \right) \tag{2}
\]

where \( f_l, f_u \) and \( f_c \) are lower band, upper band and center frequencies, respectively, and the referred velocity \( \sigma_0 = 10^{-6} \) in/s (2.54×10^{-8} m/s). Equation (2) is frequency dependent, and one can collect \( L_y(f_c) \) to obtain the frequency-independent vibration total dB

\[
\text{dB}_{\text{total}} = 10 \times \log_{10} \left( \sum_{f_c}^{100 \text{Hz}} 10^{\frac{1}{10} \times L_y(f_c)} \right) \tag{3}
\]

Table 1 shows the recommended vibration guidelines. In this study, the particle velocities on the working floor of the high-tech building were obtained from the finite element analysis or experiment. At the selected locations, we obtained the time period (\( T \) in equation (1)) equal to 8 s to find the 1/3 octave band result using equation (2), and shifted 1 s to select another 8 s to find it again until the last time period is not enough to 8 s. Finally, the maximum dB of each center frequency was obtained.

**Structural types for high-tech factories**

High-tech factories require a large space for facilities, so long span steel truss systems are often used for the superstructure, and dense reinforced concrete (RC) columns with thick slabs are constructed to reduce

| Table 1. Recommended vibration guidelines \(^{33}\) |
|-----------------|--------------|----------------------------------|
| **Description of use** | **Max. level of vibration** | **Description** |
| Workshop        | 90 dB        | Distinctly perceptible vibration. Appropriate for workshops and non-sensitive areas |
| Office          | 84 dB        | Perceptible vibration. Appropriate for offices and non-sensitive areas |
| Residential day | 78 dB        | Barely perceptible vibration. Appropriate for sleeping areas in most instances |
| Operation room  | 72 dB        | Vibration not perceptible. Adequate in most instance for optical microscopes to 100× |
| VC-A standard in semi-conductor industries | 66 dB | Adequate in most instances for optical microscopes to 400×, microbalances, optical balances, proximity and projection aligners, etc. |
| VC-D standard in semi-conductor industries | 48 dB | Adequate in most instances for optical microscopes (TEM and SEM), microbalances, optical balances, proximity and projection aligners, etc. |

VC: Vibration criterion.
micro-vibration. In this study, the three-story factory for photovoltaic panels was analyzed, where the first story is the RC structure with dense columns and other two are the steel structure with large span truss frames. Figure 1 shows two typical frames in the X and Y directions, where the RC columns spans in the X and Y directions are 6 m and 5.2 m, respectively, and those of steel columns are 32 m and 12 m, respectively. The size of the square RC column is 1.5 m for those connected with steel columns and 0.6 m for others. The size (width by depth) of the rectangular RC beam is 0.95 m by 0.725 m for those connected with steel columns, and that for others is 0.55 m by 0.725 m. The RC slab has the depth of 0.625, 0.55, and 0.45 m for the first to third floor, respectively, with the 0.4 m diameter circular holes, and the interval between two hole centers is 0.7 m. The steel sections of the second and third stories are listed in Table 2.

**Finite element model of a high-tech factory with moving cranes**

**Model of the rail system and moving cranes**

The moving crane shown in Figure 2 contains four wheels, a loading frame, and a product support. It was modeled as the combination of four moving wheel elements with lumped mass for wheels, four three-direction spring-damper elements for the loading frame, and a lumped mass for the product support. The finite element model for those elements can be found in the literature, and this section will briefly explain the elements. For the moving wheel element, the three-node element stiffness for the nodal displacements \( (d_1, \theta_1, d_2, \theta_3) \) is

\[
S = T_N^T \begin{bmatrix} k_r & -k_r \\ -k_r & k_r \end{bmatrix} T_N,
\]

where \( d_1, \theta_1, d_3 \) and \( \theta_3 \) are the translations and rotations at target nodes 1 and 3, \( d_2 \) is the translation of the wheel node, \( N_i \) \( (i = 1, \ldots, 4) \) is the cubic Hermitian interpolation functions, and \( k_r \) is the stiffness between the rail and
wheel. The internal force vector of the wheel element is

\[ \begin{bmatrix} f_1 & m_1 & f_2 & f_3 & m_3 \end{bmatrix}^T = S \begin{bmatrix} d_1 & \theta_1 & d_2 & d_3 & \theta_3 \end{bmatrix}^T - \begin{bmatrix} N_1 & N_2 & -1 & N_3 & N_4 \end{bmatrix} k_r r_v(X) \]  

(5)

where \((f_1, m_1, f_2, f_3, m_3)\) are internal forces and moments at nodes 1, 2 and 3, and the nodal forces should exclude the terms of rail irregularities \(r_v(X)\). For the contact force \(f_2\), if it is negative, the wheel and rail are in contact together. Otherwise, they are separated, and \(k_r\) is set at zero for the next Newton–Raphson iteration. For a spring or damper connected to two master nodes, the stiffness and damping matrices are

\[ S = sBB^T \quad \text{and} \quad D = cBB^T \]  

(6)

where \(s\) is the spring constant for stiffness matrix \(S\), \(c\) is the damping constant for damping matrix \(D\), and \(B\) is a vector generated from the coordinate difference between the master and spring-damper nodes.

For the simulation of the crane acceleration, we assume that all the displacements, velocities, and accelerations at vehicles in the motion direction are relative to current averaged displacement, velocity, and acceleration. Thus, the forces at mass locations due to the averaged vehicle acceleration in the motion direction can be calculated explicitly as the inertia forces in the negative acceleration direction. The total net mass of the crane is 5.1 t, and the maximum carried mass is 0.5 t. Since the variation of the crane carried mass does not cause a large change of the micro-vibration, we performed all finite element analyses using the maximum crane load mass. The top of each spring-damper element are set to the slave node controlled by the master node at the support center with the mass of \(M_c = 2\) t, and each wheel center has a lumped mass of \(M_s = 0.9\) t. The spring constants of \(K_r = 0.4E5\) kN/m and \(K_s = 1.2E5\) kN/m are approximated from the wheel and column sections, respectively. In this study, the crane moves on the railway system with a distance of 30.5 m. As shown in Figure 2, the crane stops at the rightmost side of the rail for 2 s at beginning, and it then moves in the left direction with an acceleration of 1 m/s² to reach the final speed. After it moves a certain distance with the constant speed, the crane begins to decelerate with the deceleration of 1 m/s². Finally, it stops, and the total moved distance is 30.5 m. The backward route of the crane is then the reverse of the above procedures.

**Finite element model of rails and rail irregularities**

A sample function \(r_v(X)\) of the rail irregularity is used in this study.\(^{32}\)

\[ r_v(X) = \sum_{k=1}^{n} a_k \cos(\omega_k X + \phi_k) \]  

(7)
where \(a_k\) is the amplitude, \(\omega_k\) is a frequency (rad/s) within the upper and lower limits of the frequency \([\omega_l, \omega_u]\), \(\phi_k\) is a random phase angle in the interval \([0, 2\pi]\), \(X\) is the global coordinate in the rail direction, and \(n\) is the total number of terms. The parameters \(a_k\) and \(\omega_k\) are computed by

\[
\begin{align*}
a_k &= 2\sqrt{G_{rr}(\omega_k)}\Delta\omega, \\
\omega_k &= \omega_l + (k - 1/2)\Delta\omega \\
\text{and } \Delta\omega &= (\omega_u - \omega_l)/N, \quad k = 1, 2, \ldots, n
\end{align*}
\]

(8)

\[
G_{rr}(\omega) = \frac{A_r\omega_k^2(\omega^2 + \omega_1^2)}{\omega^4(\omega^2 + \omega_2^2)}
\]

(9)

where \(G_{rr}(\omega)\) is a power spectral density (PSD) function, \(A_r\) is the roughness coefficient, and \(\omega_1\) and \(\omega_2\) are frequencies that change the shape of \(G_{rr}(\omega)\). The rail irregularity had an amplitude of 1.5 mm per 10 m of the rail in the vertical and transverse directions of the rail, and the irregularity parameters are shown in Table 3 for the both directions.

**Finite element model of high-tech factories**

Figure 3 shows the finite element mesh including the high-tech factory, rail system, crane, and warrior slabs, where the total number of degrees of freedom is 1,605,894. The beams, columns, and trusses are modeled using the 2-node 3D beam element with the shear deformation effect, and the end released function is used to simulate the truss member. The foundation at each column bottom is simulated using a three-direction spring with the stiffness of 2E6 kN/m for that connected with a steel column and 1E6 kN/m for others, and those stiffness values were approximately estimated from the averaged soil stiffness of 3E4 kN/m². The waffle slab with the hole diameter of 0.6 m and hole interval of 0.7 m is simulated using the 2-node 3D beam element with 0.4 m width and 0.18 m rigid zone at two ends of the element. The slabs at the bottom levels of the trusses on the second and third stories are modeled using the 4-node plate element. The rail system, as shown in Figure 3, contains two steel rails with the properties of \(I_x = 0.19E-4 \text{ m}^4\), \(I_y = 0.6E-4 \text{ m}^4\), and the area of 0.17E-2 \text{ m}². The 2-node 3D beam element is used to simulate rails, and the spring-damper element with the stiffness of 4.8E5 kN/m and the damping of 10 kN-s/m is used to simulate the support between rails and slabs, where the interval of the supports is 1.3 m. Two slave nodes (node S in Figure 3) controlled by the master node at the beam center at each support section are set, and a slave node at the rail bottom of each support section is set for the target route of the moving wheel element. Then, the crane model can move on the rail system connected to the high-tech structure. The average acceleration Newmark method, Newton–Raphson method, and the consistent mass scheme are used to solve this nonlinear problem, where the time step length is 0.005 seconds, with 10,000 time steps being simulated.

**Experimental validation**

There are two stations with the distances of 5 and 10 m, respectively, from the rail centerline in the second level with RC beams and columns of the building, as shown in Figure 1. Each station contained three velocity sensors with the sensitivity of 10 V/(2.54 cm/s) to measure X, Y, and Z vibration velocities with a sample rate of 512 Hz. A crane moved the distance of 30.5 m from point A to point B, as shown in Figure 3, with the speed of 2, 2.5 or 3.5 m/s (120, 150, and 210 m/min), respectively. Computer software recorded the vibration continuously, and we picked up three-group 40-s data for the crane passing over this region to find the one-third octave band vibration due to the field experiment. Since finite element analysis only contains moving crane loading without the ambient vibration effect, the 600 s ambient vibration data were also obtained before the experiment to calculate the power spectrum density function, which were then added into equation (6) of the finite element result to find the vibration dB. Figure 4 shows the Z-direction ambient and the vibration in X, Y and Z directions for the 5 m station under the crane speed of 3.5 m/s. This figure indicates that the vibration in the vertical (Z) direction is much larger than those in the in-plane (X and Y) directions, so we will only focus on the vertical vibration for the

| \(Ar\) (m² rad/m) | \(\omega_1\) (rad/m) | \(\omega_2\) (rad/m) | \(\omega_l\) (rad/m) | \(\omega_u\) (rad/m) | \(N\) |
|------------------|------------------|------------------|------------------|------------------|-----|
| \(Ar_0 = 0.23E-7\) | 0.0             | 2.1             | 0.08             | 150              | 2000 |

| \(Ar_0\) | 0.23E-7 |
|---------|--------|

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subsequent sections. The experimental and finite element results of the vertical vibration are shown in Figures 4 and 5 for the 5 m and 10 m stations, respectively. The comparisons between experimental and finite element results are acceptably accurate. The major different is that the vibration of the finite element results near 40 Hz are underestimated, and the possible reason should be the rail irregularity data used in Table 3 are not accurate enough. It is noted that the rail irregularities with random angles shown in equation (1) are not exact but statistical, and parametric study due to rail irregularities will be further discussed in Section 5.3.

**Parametric finite element study**

The vibration effects of rail irregularities, crane velocities, and slab thickness are investigated in the following three sections. Figure 6 shows the particle velocity vibration dB changing with the distance from the rail centerline. This figure indicates that the trend of the vibration decreases with the increase of the distance from the centerline, and the vibration oscillates with the column locations. The vibration is the relatively largest and smallest at the column location and beam center, respectively. Without the ambient vibration, the decrease rate will approximately approach to a steady state condition. However, with the ambient vibration, the decrease rate at far field will approach to zero, since the major slab vibration is come from the ambient vibration at the far field. The ambient vibration may mislead the analysis result, so it was not considered in the subsequent finite element analysis.

**Effect of rail irregularities**

The crane only contains two wheel axes, so the periodic effect due to the wheel axes should be minor. Other sources of the crane-induced vibration include the effect of rail irregularities, the slab displacement due to the crane weight, and the resonance between the crane and slab. The first slab natural frequency is 20.5 Hz, and the first Y-rotation and Z-translation natural frequencies of the crane are 14.8 and 26.3 Hz, respectively, so that the resonance between the crane and slab does not occur. Figure 7 shows the particle velocity vibration dB in equation (2) changing with vibration frequencies for the location at 5 m from the rail centerline, and the crane speed is 3.5 m/s under five different rail irregularity parameter $A_r$. This figure indicates that the frequency range of

![Figure 3. Finite element model including the high-tech factory, rails, crane, and slabs.](image)
the crane-induced vibration is located between 15 and 35 Hz, and the vibration of other frequencies is considerably small, and the reason should be the natural frequencies of the slab and crane being within this frequency range. The 25 Hz vibration is very sensitive to rail irregularities, because the rail irregularities induce the vibration of the crane and slab at their vertical natural frequencies. The 15 Hz vibration is major induced from the crane rotation mode at its natural frequency of 14.8 Hz. The crane rotation mode is induced not only from rail

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**Figure 4.** Particle velocity vibration dB (equation (2)) changing with vibration frequencies for the location at 5 m from the rail centerline under the crane speed of 3.5 m/s and ambient vibration.

**Figure 5.** Comparisons of particle velocity vibration dB (equation (2)) changing with vibration frequencies for the locations at 5 and 10 m from the rail centerline under the crane speeds of 3.5, 2.5 and 2 m/s using experiments and finite element analyses.
irregularities but also from the acceleration and deceleration of the crane. Figure 8 shows the Particle velocity vibration total dB (equation (3)) changing with the ratio of rail irregularities for the location at 5 m from the rail centerline under the crane speed of 3.5 and 2 m/s. This figure indicates that the slab vibration increases with the rise of rail irregularities. It is difficult to find a theoretical relationship between the slab vibration and rail irregularities; however, smoothing the wheel and rail should be the first priority to decrease slab vibration in high-tech factories. This condition can be clearly seen in Figure 9, where the vibration due to the moving crane without rail irregularities is considerably smaller than that with the rail irregularities as shown in Table 3.

**Effect of crane speed**

Other than rail irregularities, the acceleration, deceleration, and speed of crane are other important factors to influence slab vibration. Figure 10 shows Particle velocity vibration total dB changing with the distance from the rail centerline under various crane speeds, and this figure indicates that the slab vibration decreases with the reduction of the train speed clearly. Two factors, changing crane speed and the speed magnitude, will affect the slab vibration, and they are discussed as follows:

1. The acceleration and deceleration produce a horizontal inertia force at the mass center of the crane, and this inertia force effect will be enlarged if the change of the crane acceleration and deceleration is frequent, since only a constant force does not produce large dynamic effect of the structure. This inertia horizontal force may cause the crane to rotate with the natural frequency of its rotation mode, so that the slab also shows the vibration with this frequency. Since the crane acceleration and deceleration are often controlled at some certain
locations, the slab vibration will not be large if the location is far away from the region of the crane changing speed.

2. The rail irregularity at a certain frequency is highly dependent on the crane speed, and this condition can be explained using equation (7). First let the distance $X$ in equation (7) change to $Vt$ as follows, where $V$ is the crane speed and $t$ is the time.

**Figure 8.** Particle velocity vibration total dB changing with the ratio of rail irregularities for the location at 5 m from the rail centerline.

**Figure 9.** Comparison between the slab vibration near the moving crane with (Table 3) and without rail irregularities under the crane speed of 3.5 m/s (Mutinying factor of the deformation $= 2E5$. The deformation at the crane is not shown due to too large value.).

**Figure 10.** Particle velocity vibration total dB (equation (3)) changing with the distance from the rail centerline under various crane speeds.
rvðXÞ¼XNk¼1akcosðxkVt+kÞ¼XNk¼1akcosðxvkttÞð10Þ

where \(xvk\) is a rail irregularity frequency. Thus, for a certain rail irregularity frequency \(xvk\), its amplitude is

\[a_k = a_k(\omega_k) = a_k(\omega_{vk}/V)\]

or

\[a_k = \sqrt{G_{irr}} = \left(\frac{A_j(\omega^2(\omega_{vk}/V)^2 + \omega^2)}{(\omega_{vk}/V)^4((\omega_{vk}/V)^2 + \omega^2)}\right)^{1/2}\]

(11)

If \(\omega_1\) and \(\omega_2\) are much smaller than \(\omega_{vk}/V\) (for example, \(\omega_1 = 0\), \(\omega_2 = 2.1\) rad/m, \(V = 3\) m/s, and \(\omega_{vk}/V > 2\pi\) (10 Hz)/(3 m/s) = 21 rad/m), equation (11) is approximated to

\[a_k \approx \frac{\sqrt{A_j\omega^2V^2}}{\omega_{vk}^2}\]

(12)

Thus, for a certain frequency of rail irregularities, its amplitude is linearly propositional to the square of the crane speed approximately.

3. Figure 11 shows the particle velocity vibration dB changing with vibration frequencies for the location at 5 m from the rail centerline under the crane speed of 2 and 3.5 m/s, where rail irregularities as shown in Table 3 are included or not included. This figure indicates that the difference of the 15 Hz slab vibration between the crane speed of 2 and 3.5 m/s is 4.5 and 6.4 dB, respectively, for the case with and without rail irregularities. This means that the 15 Hz slab vibration is not sensitive to the rail irregularities, because the 15 Hz slab vibration is mainly caused by the crane rotation mode activating the slab vertical mode (20.5 Hz) under the acceleration or deceleration of the crane.

4. Figure 11 also indicates that the difference of the 25 Hz slab vibration between the crane speed of 2 and 3.5 m/s is 0.6 and 6.5 dB, respectively, for the case with and without rail irregularities. This means that the 25 Hz slab vibration is not sensitive to the crane speed for the case without rail irregularities, but it will be sensitive to the crane speed while rail irregularities are included. This condition can be well explained using equation (12), since this equation represents that rail irregularity amplitude at a certain frequency is highly dependent on the crane speed, and this rail irregularity near the crane vertical natural frequency will activate the vibration of that frequency.

**Effect of slab depth**

This section studies the vibration reduction effect of increasing slab and beam depth. We changed the depth of all the slab and beam on the working platform from 0.425 to 1.025 m, and those on the other levels were not changed. Figures 12 and 13 show the particle velocity total vibration dB changing with the distance from the rail centerline...
under the crane speed of 2 and 3.5 m/s, respectively. The two figures are similar, and they indicate the following features:

1. The increase of the slab and beam depth can decrease the slab vibration. When a large depth is used, such as 0.825 m in this study, the oscillation of the vibration magnitude along the columns will become unobvious, so that the reduction of the vibration from the distance of the rail centerline will be efficient. However, far away from the rail centerline, the oscillation of the vibration magnitude along columns becomes obvious even for the deep slab and beam, but the vibration often smaller than the ambient will not cause a serious problem.

2. When the slab and beam depth is smaller, the vibration difference between the beam center and the column becomes larger. Since the stiffness of the slab and beam is much smaller than that of the column, the vibration wave will be difficult to transform to the column. Thus, the vibration at the column locations with smaller depth of the slab and beam may be smaller than that of larger depth. Nevertheless, increasing the slab and beam depth to decrease the slab vibration induced by the moving crane is still an efficient way.

**Conclusion**

This study developed a finite element model to simulate the crane induced vibration on the floor of high-tech factories, in which the model contains 3D beam elements to simulate the high-tech structure and rails, spring-damper elements to simulate pads and supports between rails and slab, and moving wheel elements with the lumped mass to simulate the crane. The finite element results were validated using the experimental measurements in good agreement. Then, the parametric study including the effects of rail irregularities, slab depth, and crane speed was performed to investigate the vibration behavior of high-tech factories.
The frequency range of the crane-induced vibration is obvious between 15 and 35 Hz, which is the range of the natural frequencies of the slab and crane, so the rail irregularities and acceleration of the crane activate vibration within this frequency range. The rail irregularities induce the vibration of the crane and slab at their vertical natural frequencies. Then, both rail irregularities and the acceleration of the crane induce the crane rotation mode in its natural frequency, which also causes the slab vibrates in that frequency. Thus, smoothing the wheel and rail should be the first priority to decrease slab vibration in high-tech factories.

The crane speed is another important issue to influence slab vibration, which decreases with the reduction of the train speed clearly, as shown in this study. Two factors, crane speed and acceleration will affect the slab vibration. The acceleration produces a horizontal inertia force of the crane, which causes the crane to rotate with the natural frequency of its rotation mode. For the crane speed effect according to the formulation of rail irregularities, this study indicates that the rail irregularity at a certain frequency is linearly propositional to the square of the crane speed approximately, so increasing the crane speed produces larger rail irregularities at the frequency of the crane vertical mode, which then activates the slab vibration. To reduce the slab vibration, decreasing the crane speed is an alternative, and experiments are suggested to find the optimal crane speed and acceleration.

Increasing slab and beam depth to reduce crane-induced vibration is a straightforward way. The crane-induced vibration is the relatively largest and smallest at the column location and beam center, respectively. When a large slab and beam depth is used, such as over 0.825 m, the oscillation of the vibration magnitude along the columns will become unobvious, so that the reduction of the vibration from the distance of the rail centerline will be efficient. Thus, increasing the slab and beam depth to decrease the slab vibration induced by the moving crane is an efficient way.

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