NUMERICAL SIMULATION OF FLOOR IMPACT SOUND USING VIBROACOUSTIC FINITE-DIFFERENCE TIME-DOMAION METHOD

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Abstract

The floor-impact sound transmission characteristics through floor structures are of great importance from the viewpoint of quality of sound environment. The sound insulation performance of the floor structure is commonly evaluated based on the sound pressure levels, which are measured by exciting the surface of the floor with various kinds of devices suitable for each purpose of the sound insulation evaluation. To obtain the sound insulation performances of heavy- and light-weight impact sound, the bang machine, impact ball, and tapping machine are utilized as the devices for excitation. Then, the measured and evaluated results by using these excitation methods greatly depend on the time and frequency characteristics of the excitation force itself. The structure-borne sound transmission characteristics can also be numerically predicted by using the SEA, FEM, and FDTD. However, in the present situation, the numerical modelling of the excitation characteristics should require more investigation to be incorporated into such a discrete wave-based numerical prediction of vibration characteristics. To investigate the simulation method of the time and frequency characteristics of the vibroacoustic transmission between the floor structures, the collision between the free-fall mass and the elastic plate-like structure are modelled by the one degree-of-freedom model structured on the discrete wave-based numerical analysis of bending vibration with finite-difference time-domain (FDTD) method. In this paper, as the basic study, the excitation characteristics of the generally-used devices of the impact ball, bang machine, and tapping machine are simulated, and are applied to a numerical case study targeting at prediction of the impact sound pressure levels inside a wall-type concrete structure.

Keywords: structure-borne sound, vibroacoustic finite-difference time-domain method.

1 Introduction

The floor-impact sound transmission characteristics through floor structures are of great importance from the viewpoint of quality of sound environment. The sound insulation performance of the floor structure is ordinarily evaluated based on the sound pressure levels, which are measured by exciting the surface of the floor with various kinds of devices being suitable for each purpose of the sound insulation evaluation. In order to obtain general vibroacoustic characteristics of the floor structure, the impact hammers are used, while the sound insulation performances of heavy- and light-weight impact sound are measured and evaluated based on the method described in the JIS 1418-1 [1] and -2 [2], respectively. In the latter case, the bang machine, impact ball, and tapping machine are utilized as the devices for excitation. Then, the measured and evaluated results by using these excitation methods greatly depend on the time and frequency characteristics of the excitation force itself.

On the other hand, the structure-borne sound transmission characteristics can also be numerically predicted by using the statistical energy analysis (SEA) [3-5], the finite-element method (FEM) [6], and the finite-difference time-domain (FDTD) [7-13]. Especially in the prediction by FEM and FDTD, the floor impact sound is predicted based on the discrete numerical schemes. In most cases, the excitation characteristics caused by various kinds of impactor are modeled by some mathematical functions such as the Gaussian's [13]. On the other hand, some researches for modeling the excitation characteristics of the excitation devices for heavy-weight floor impact sound measurement are conducted [14-16], however, in the present situation, the numerical modeling of the excitation characteristics should require more investigation to be incorporated into such a discrete wave-based numerical prediction of vibration characteristics.

In the present study, for the purpose of development of the numerical method of the time and frequency characteristics of the external forces excited by various kinds of impact method, the collision between the free-fall mass and the elastic plate-like structure are modeled by the one degree-of-freedom model structured on the discrete wave-based numerical analysis of bending vibration with FDTD. In the present paper, as the basic study, the excitation characteristics of the generally-used devices of the impact ball, bang machine, and tapping machine are simulated, and are applied to a numerical case study targeting at prediction of the impact sound pressure levels inside a wall-type concrete structure.

2 Numerical method

2.1 Basic theory of FDTD

The present study investigates the time-domain calculation for bending vibration analysis of a thin plate by using the FDTD method, which simulates the vibration propagation by sequentially performing time integration on the orthogonal meshes. Vibration analysis of a planar structure is performed by discretizing the bending wave on a thin plate in the *x-y* plane by finite-differential approximation in time and space. For the bending wave simulation, the following vibration equation based on the Mindlin-Reissner thick-plate theory [17, 18], which is excited by the collision force F caused in the contact between the present elastic plate and the dropped free-fall mass, was used:

$$\begin{aligned}&\left\{D\left(1+\xi\frac{\partial}{\partial t}\right)\left(\frac{\partial^2}{\partial x^2}+\frac{\partial^2}{\partial y^2}\right)-\frac{\rho_{\mathrm{plate}}h^3}{12}\frac{\partial^2}{\partial t^2}\right\}\left\{\left(\frac{\partial^2}{\partial x^2}+\frac{\partial^2}{\partial y^2}\right)-\frac{\rho_{\mathrm{plate}}h}{kGh}\frac{\partial^2}{\partial t^2}\right\}w+\rho_{\mathrm{plate}}h\frac{\partial}{\partial t}\left(\mu+\frac{\partial}{\partial t}\right)w\\&=F\left\{1-\frac{D}{kGh}\left(\frac{\partial^2}{\partial x^2}+\frac{\partial^2}{\partial y^2}\right)-\frac{1}{kGh}\frac{\rho_{\mathrm{plate}}h^3}{12}\frac{\partial^2}{\partial t^2}\right\}\end{aligned} \tag{1}$$

where w is the displacement of the out-of-plane deformation of the plate; ξ and μ are coefficients for modeling the damping characteristics of the bending deformation; D is the flexural rigidity ($D = Eh^3/12(1-\nu^2)$); and $\rho_{\mathrm{plate}}h^3/12$ is the rotational inertia. The other coefficients, E, ρ_{plate}, h, ν, k, and G are Young's modulus, the density of the plate, the thickness of the plate, Poisson's ratio, Timoshenko's shear coefficient, and elastic shear modulus, respectively. Finally, the discrete equation, which is obtained by applying a finite-difference approximation to the above mentioned basic equation, is solved in every time step. The updating method based on Eq. 1 is described in detail in the previous study [11].

2.2 Modeling of Collision between mass and plate

The dropped materials such as the impact ball, bang machine and the hammer of the tapping machine are modeled as a one degree-of-freedom mass model of Fig. 1, in which the mass falls on the elastic plate accelerated by the gravity. When the mass collides with the elastic plate, the contact force is caused, which is calculated using the following contact model between the dropped material and the elastic plate. Then, the acceleration, velocity, and displacement of the mass and each part of the elastic plate are calculated from the obtained contact force using the finite-difference scheme.

In the present study, firstly, the free-fall of the mass is calculated by considering the gravity as shown in Fig. 1(a). Then, the contact force between the dropped mass and the plate, defined in the normal directions of Fig. 1, was calculated. The contact forces F in case of the impactors of the impact ball and bang machine are expressed as follows based on the linear Voigt model:

$$F = k \cdot \delta + c \cdot \dot{\delta} \tag{2}$$

$$\delta = w_{\mathrm{m}} - w_{i,j} - r_{\mathrm{th}} \tag{3}$$

$$\dot{\delta} = \dot{w}_{\mathrm{m}} - \dot{w}_{i,j} \tag{4}$$

where δ and $\dot{\delta}$ are the relative displacement and velocity between the mass and the plate, k is the spring constant, and c is the viscosity constant. Here, δ is calculated by using the displacement of the mass w_{m}, that of the elastic plate at the contacting grid $w_{i,j}$, and the threshold distance r_{th} which is set to be equal to the radius of each impactor, whereas $\dot{\delta}$ is calculated as the difference between the velocities of the mass $\dot{w}_{\mathrm{m}}$ and plate $\dot{w}_{i,j}$. On the other hand, the contact force F in case of the tapping machine with relatively higher Young's modulus is expressed as follows based on the Voigt model with a nonlinear Hertz spring, which is commonly used in the researches of the discrete element method [19]:

$$F = k \cdot \delta^{3/2} + c \cdot \delta^{1/4} \cdot \dot{\delta} \tag{5}$$

$$K = \frac{4 \cdot E_m E_p \sqrt{r}}{3\left(E_m\left(1-\nu_p^2\right) + E_p\left(1-\nu_m^2\right)\right)} \tag{6}$$

where E is the Young's modulus, and r is the radius of the mass. The subscripts m and p mean the mass and plate, respectively. In this case, δ and $\dot{\delta}$ are calculated as the same way using Eqs. (2) and (3), and the threshold distance r_{th} was set to zero, because the surface of the bottom of the tapping hammer has curvature radius of 500 mm and that is almost flat. In order to simulate the mutual effect caused by the collision between the mass and the plate, the contact force F was treated as the external force as shown in Eq. (1), whereas the mass is accelerated by the acted force of F.

The numerical parameters used in the simulation are indicated in Table 1. The spring constants of the impact ball and bang machine are difficult to define because they have complex structures composed of the rubber and air layers. In addition, the elastic properties of these materials can be greatly changed [20] due to the inflation air pressure. For these reasons, in this study, the spring constants k were simply estimated as $k = (2\pi)^2 m \cdot f_r^2$ by using the first mode frequencies f_{r} and the weight m of the impact ball and bang machine [21], which were reported as 24 and 25 Hz, respectively. On the other hand, the viscosity constant c is calculated as follows.

$$c = \alpha\sqrt{m \cdot k} \tag{7}$$

$$\alpha = 2\sqrt{(\ln R)^2 \Big/ \left((\ln R)^2 + \pi^2\right)} \tag{8}$$

As can be seen in Eq. (8), the damping coefficient of the Voigt model α can be simply obtained from the coefficient of restitution R. Herein, for simplicity, the restitution coefficients of the impact ball and bang machine were assigned as the values indicated in Table 1, which are defined in the JIS 1418-2 [2].

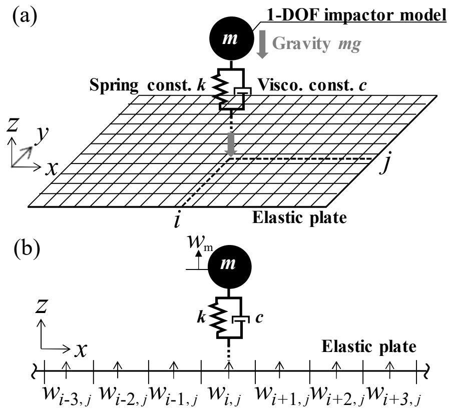

Figure 1 – Schematic of the numerical model of the collision between the elastic plate and the free-fall mass, and (b) the arrangement of the displacements on each of the discrete grid on the 2-dimensional plate model.

Table 1 – Numerical parameters set in the FDTD simulation.

	Ball	Car-tire	Tapp. hammer	Concrete
Spring const., k [N/m]	$6.17 \cdot 10^4$	$1.66 \cdot 10^5$	$2.20 \cdot 10^{10}$	—
Damping coef. α [-]	0.141	0.141	0.103	
Viscosity const., c [N・s/m]	55.7	156.0	$1.09 \cdot 10^4$	—
Young's modulus [N/m^2]	—	—	$2.0 \cdot 10^{11}$	$2.4 \cdot 10^{10}$
Poisson's ratio [-]	—	—	0.3	0.3
Drop height [m]	1.00	0.85	0.04	—
Coef. rest., R [-]	0.80	0.80	0.85	—
Weight, m [kg] or density [kg/m^3]*	2.5	7.3	0.5	2,400*
Radius, r [cm]	4.5	24.0	50	—
Threshold distance, r_{th} [cm]	2.25	12.0	0.0	—

3 Numerical study

3.1 Simulation model

Firstly, the spatially discretized vibroacoustic model of the structure consisting the walls and floor slab with thickness of 150 mm shown in Fig. 2(a) was modeled as composites of the two-dimensional plate and three-dimensional acoustic cubic elements, and the radiated sounds from the floor slab and wall structures are simulated by setting the source and receivers as shown in Fig. 2(b). The coupling scheme between the sound and vibration field should be referred to [11]. Then, the excitation characteristics acted by the impact ball, bang machine, and the hammer installed in the tapping machine are predicted. In the one DOF mass model, the spring and viscosity constants, required for simulation of the contact forces between the mass and plate, can be calculated by setting the parameters indicated in Table 1. In the updating calculation of FDTD, the discrete spatial and time intervals of the two-dimensional plate elements are set to $\Delta x = \Delta y = 0.02$ m and

Δt = 1/48,000 s, whereas those of the three-dimensional acoustic element are set to $\Delta x = \Delta y = \Delta z$ = 0.02 m and the same interval of Δt as the above. The sound pressures defined at the five receivers from Rs1 to Rs5 shown in Figs. 2 (a), (b) and (c) were calculated by changing the excitation point into the five from S1 to S5. The statistical-incidence absorption coefficient of 0.06 for the surface of the concrete walls, including the target floor slab was adopted for all the frequency bands among 16, 31.5, 63, 125, 250 and 500 Hz. The statistical-incidence absorption coefficient of 0.06 corresponds to normal-incidence absorption coefficients of 0.032 which is simulated by setting the normal-incidence acoustic impedance of 50,000 Ns/m^3. Lastly, the contact force between the one DOF mass model of each impactor and the elastic plate was calculated, when the distance between the center point of the mass and the plate becomes smaller than the threshold distance r_{th}.

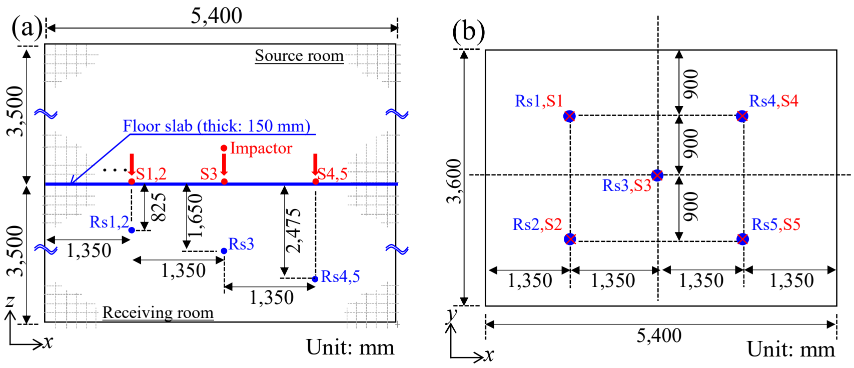

Figure 2 – Three-dimensional numerical model and (b) the arrangements of the source and receivers on the numerical model.

3.2 Results and discussion

The impact forces of the impact ball, bang machine, and the tapping hammer, when they are dropped from each of the drop height shown in Table 1, are shown in Fig. 3. The measurement results are referred from [2, 14] as shown in the figure. It should be noted that the calculated velocities of the ball, car-tire, and tapping hammer at the just time of the collision were consistent with the theoretically obtained collision speed v, which are obtained as $v = 2\sqrt{gh}$ where g and h are the gravity and the drop height, respectively. The time duration and the maximum force of the transient force characteristics of (a) bang machine and (b) impact ball obtained by FDTD show generally good agreement with the reference data. In case of the bang machine, the change of the maximum excitation force due to the dropping height is also well simulated. Compared to these heavy-weight impactors, the light-weight impactor of the tapping hammer shows relatively short duration time and has relatively higher frequency components. Figure 4 shows the impact force exposure level of these impactors with the referred measurement results [23, 24]. In these results, the frequency characteristics of these impactors also show agreement with the reference data. However, the numerical results also show slight difference in the higher frequency ranges, especially in the 125 and 250 Hz for the bang machine, and 250 and 500 Hz for the tapping machine, respectively. It may be considered that the numerical errors due to the deviation between the determined numerical parameters for the one-dimensional modeling of the collision and those in the real phenomena appeared especially in the higher frequency range.

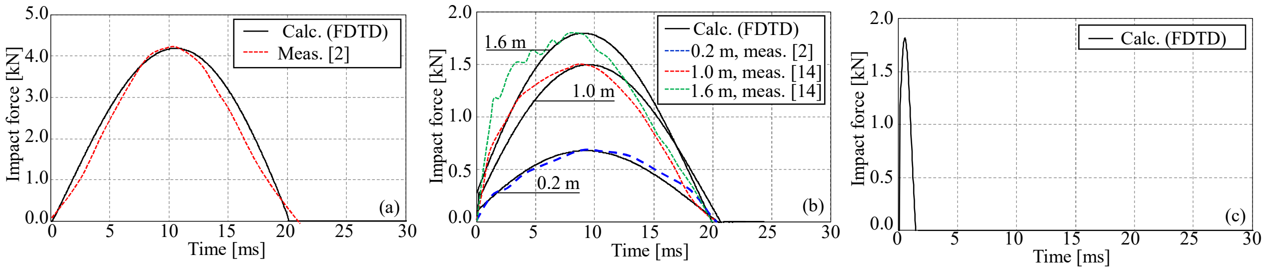

Figure 3 – Numerical results of the excitation forces in case of (a) bang machine, (b) impact ball, and (c) tapping hammer, and the reference data [2, 14].

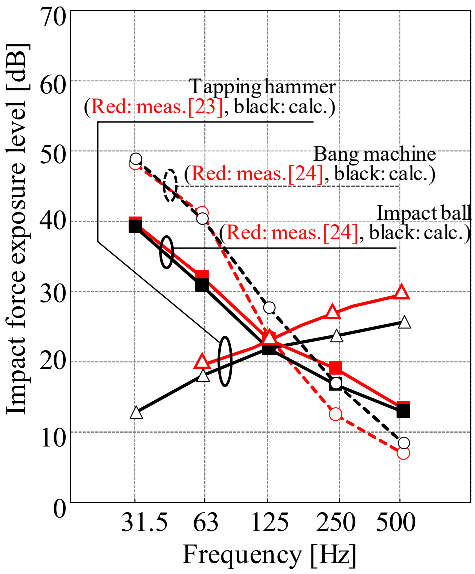

Figure 4 – Numerical results of the impact force exposure levels, and the experimental data referred to the references [23, 24].

4 Conclusions

A numerical simulation method of the impact forces of free-fall mass acted to elastic plates is proposed based on the FDTD method. The comparison of the calculated impact forces and the impact sound pressure levels calculated by using the impact forces with the measurement results has shown a general validity of the proposed method.

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