Finite-difference time-domain simulation of floor-impact sound excited by one-dimensional contact model and its application to auralization of floor-impact sound

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Abstract: To develop a numerical method of determining the time and frequency characteristics of the external forces excited by various impact sources, the collision between the free-fall mass and the elastic platelike structure are modeled by the one-degree-of-freedom contact model structured on the discrete wave-based numerical analysis of bending vibration with the finite-difference time-domain (FDTD) method. Moreover, the calculated impact sounds were auralized and applied to the evaluation experiment of the subjective impression of loudness and annoyance. As the basic study, the excitation characteristics of generally used devices that is, the impact ball, bang machine, and tapping machine and human trotting are simulated and used for a numerical simulation targeting the prediction of the impact sound pressure levels inside a wall-type concrete structure. After that, the proposed method was applied to a subjective evaluation experiment of the loudness and annoyance influenced by a floor-impact sound on various types of floor slab. The results have shown that the loudness and annoyance induced by the floor-impact sound indicated a high correlation with the maximum A-weighted sound pressure level.

Keywords: Finite-difference time-domain method, Impact vibration, Voigt model, Floor-impact sound

1. INTRODUCTION
The characteristics of floor-impact sound transmission through floor structures are of considerable importance from the viewpoint of sound environment quality. The sound insulation performance of a floor structure is conventionally evaluated on the basis of sound pressure levels, which are measured by exciting the surface of the floor with various types of devices suited to each purpose of sound insulation evaluation. To determine the general vibroacoustic characteristics of a floor structure, impact hammers are used, whereas the sound insulation performances of heavyweight and lightweight impact sounds are measured and evaluated by the method described in JIS 1418-1 [1] and 1418-2 [2], respectively. In the latter case, the bang machine, impact ball, and tapping machine are utilized as the excitation devices. Then, the results measured and evaluated by these excitation methods greatly depend on the time and frequency characteristics of the excitation force itself.

On the other hand, structure-borne sound transmission characteristics can also be numerically predicted by statistical energy analysis (SEA) [3–5], finite-element method (FEM) [6], and finite-difference time-domain (FDTD) method [7–13]. Particularly in the prediction by the FEM and FDTD methods, the floor-impact sound is predicted on the basis of discrete numerical schemes. In most cases, the excitation characteristics caused by various types of impactor are modeled using mathematical functions such as the Gaussian function [13]. On the other hand, some research studies for modeling the excitation characteristics of the excitation devices for heavyweight floor-impact sound measurement have been conducted [14–16]. In the usual cases of floor impact sound occurring in residential spaces, the radiated impact sound varies largely from case to case depending on the exciting impactors and their excitation conditions, where the floor-impact sound performance is basically evaluated using the regulated impactors such as the bang machine, impact ball, and tapping machine. Hence, to accurately predict the floor-impact sound transmission characteristics of floors excited by various impactors with optimal physical properties under various excitation conditions of, for example, drop heights and initial dropping velocity, it will be highly advantageous if the floor impact sound can be predicted and evaluated without any technical regulation related to the detailed conditions of the impactors in the simulation.
This advantage is lost when one uses the conventional numerical method with the measured excitation characteristics as input to the numerical calculation. However, the above-mentioned numerical schemes of the excitation characteristics require further investigation before they can be utilized as a versatile simulation method to freely calculate the radiated sound regardless of the excitation conditions.

While numerical results can be used for the quantitative evaluation of floor impact sound, they can also be applied to the auralization of impact sounds if such a time-transient numerical simulation is available. Not only the loudness but also the annoyance, and other impressions of the radiated sounds can be evaluated by considering subjective evaluation of the human auditory perception. The auralization techniques have been applied in architectural acoustics [17–25]. Vorländner investigated an auralization system for the evaluation of the sound insulation of buildings, employing a convolution technique using filtering treatment considering the sound insulation characteristics of building walls [18,19]. Rindel and Holger proposed an auralization method for airborne sound insulation using ODEON [20]. On the other hand, wave-based numerical methods are often used for acoustic auralization [22–25], because they are suitable for the simulation of the sound propagation inside vibroacoustic fields of arbitrary shapes. Among the wave-based methods, the FDTD method has been frequently used for auralization, since it can directly provide time-domain results, such as the impulse responses inside sound fields. The impulse responses can also be obtained between separated sound fields that sandwich sound-insulation elements such as wall structures. The reference [25] provides an auralization method in which the measured vehicle sounds and sound insulation characteristics obtained by a wave-based vibroacoustic numerical method are combined. In this case, the calculated impulse responses are indirectly auralized, whereas the impact sounds caused by the excitation of floor structures can be directly auralized on the basis of results of vibroacoustic FDTD simulation. Although Yokota et al. have investigated the application of the calculated impulse responses in subjective evaluation experiments of room-acoustic reverberation [22], there have not yet been any case studies on the investigation of the subjective evaluation of structure-borne sounds, including floor-impact sounds, employing numerically obtained impact sounds.

In this study, for the purpose of developing a numerical method of simulating the time and frequency characteristics of the external forces excited by various impact methods, the collisions between the free-fall mass and the elastic platelike structure are modeled by the one-degree-of-freedom contact model structured on the basis of the discrete wave-based numerical analysis of bending vibration with the FDTD method. In this work, as a basic study, the excitation characteristics of the generally used devices, namely, the impact ball and bang machine, are simulated and applied to a numerical case study for the prediction of impact sound pressure levels inside a wall-type concrete structure. After that, the subjective impression of the loudness and annoyance perceived from the calculated impact sounds caused by the above-mentioned sources were experimentally evaluated through a subjective evaluation test, and the results are discussed by comparison with the physical indicators.

2. NUMERICAL METHOD

2.1. Basic Theory of FDTD

In this study, we carried out the time-domain calculation for bending vibration analysis of a thin plate by the FDTD method, which simulates vibration propagation by sequentially performing time integration on 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 [26,27], in which the plate is excited by the collision force F caused upon contact between the present elastic plate and the dropped free-fall mass, was used.

\[
\begin{split}
\left\{ D \left( 1 + \xi \frac{\partial}{\partial t} \right) \left( \frac{\partial^2}{\partial x^2} + \frac{\partial^2}{\partial y^2} \right) - \frac{\rho_{\text{plate}} h^3}{12} \frac{\partial^2 w}{\partial t^2} \right\} \\
\times \left\{ \left( \frac{\partial^2}{\partial x^2} + \frac{\partial^2}{\partial y^2} \right) - \frac{\rho_{\text{plate}} h}{k G h} \frac{\partial^2 w}{\partial t^2} \right\} w + \frac{\rho_{\text{plate}} h}{k G h} \left( \mu + \frac{\partial}{\partial t} \right) w \\
= F \left\{ 1 - \frac{D}{k G h} \left( \frac{\partial^2}{\partial x^2} + \frac{\partial^2}{\partial y^2} \right) - \frac{1}{12} \frac{\rho_{\text{plate}} h^3}{k G h} \frac{\partial^2 w}{\partial t^2} \right\}
\end{split}
\]

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

2.2. Modeling of Collision between Mass and Plate

Dropped materials such as the impact ball, bang machine, and hammer of the tapping machine are modeled as a one-degree-of-freedom mass model, as shown in
The contact force between the dropped mass and the plate is calculated using the finite-difference scheme. The contact force is generated, which is accelerated by gravity. When the mass collides with the elastic plate, a contact force is generated, which is expressed as follows, on the basis of the Voigt model with damping and spring elements.

\[ F = k \cdot \delta + c \cdot \dot{\delta} \]  
\[ \delta = w_m - w_{i,j} - r_{th} \]  
\[ \dot{\delta} = \dot{w}_m - \dot{w}_{i,j} \]

where \( \delta \) and \( \dot{\delta} \) respectively are the relative displacement and velocity between the mass and the plate, \( k \) is the spring constant, and \( c \) is the viscosity constant. Here, \( \delta \) is calculated 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 \( \dot{w}_m \) and plate \( \dot{w}_{i,j} \). On the other hand, the contact force \( F \) in the case of the tapping machine with a higher Young’s modulus is expressed as follows, on the basis of the Voigt model with a nonlinear Hertz spring, which is commonly used in investigations of the discrete element method [28].

\[ F = k \cdot \delta^{3/2} + c \cdot \delta^{1/4} \cdot \dot{\delta} \]  

Here \( E \) is Young’s modulus and \( r \) is the radius of the mass. The subscripts \( m \) and \( p \) refer to the mass and plate, respectively. In this case, \( \delta \) and \( \dot{\delta} \) are calculated in the same way using Eqs. (3) and (4), respectively, and the threshold distance \( r_{th} \) is set to zero, because the surface of the bottom of the tapping hammer has a curvature radius of 500 mm, which is almost flat. To simulate the mutual effect of the collision between the mass and the plate, the contact force \( F \) is 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 shown in Table 1. The spring constants of the impact ball and bang machine are difficult to define because they have complex structures composed of rubber and air layers. In addition, the elastic properties of these materials may considerably change [29] owing to the inflation air pressure. For these reasons, in this study, the spring constants \( k \) were simply estimated as \( k = (2\pi)^2 m \cdot f_i^2 \) by using the first mode frequencies \( f_i \) and the weights \( m \) of the impact ball and bang machine [30]; \( f_i \) is reported as 24 and 25 Hz, respectively. On the other hand, the viscosity constant \( c \) is calculated as

\[ c = \alpha \sqrt{m \cdot k}, \]  
\[ \alpha = 2 \sqrt{(\ln R)^2/((\ln R)^2 + \pi^2)} \]

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

3. NUMERICAL STUDY

3.1. Simulated Model

Firstly, the spatially discretized vibroacoustic model of a structure consisting of walls and a floor slab of 150 mm thickness, as shown in Fig. 2(a), was simulated as composites of a 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). On the other hand, the other setting conditions, in which the positions of the source and receiver points are as shown in Fig. 2(c) and the thickness of the slab is 200 mm; this setup is called Arrangement-2, which is used only for the validation of the numerical results by comparison with the results shown in [31]. The coupling scheme between the sound and vibration field is taken from [11]. Then, the excitation characteristics affected 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 the 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 the FDTD, the discrete spatial and time intervals of the two-dimensional plate elements are set to \( \Delta x = \Delta y = 0.02 \) m and \( \Delta t = 0.0048 \) 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 \( \Delta t \) as above.

The sound pressures defined at the five receivers from Rs1 to Rs5 shown in Figs. 2(a), 2(b) and 2(c) were calculated by using five excitation point 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 of 16, 31.5, 63, 125, 250, and 500 Hz. The statistical-incidence absorption coefficient of 0.06 corresponds to a normal-incidence absorption coefficients of 0.032, which is simulated by setting the normal-incidence acoustic impedance of 50,000 Ns/m². Lastly, the contact force between the one-DOF mass model of each impactor and the elastic plate was calculated for the distance between the centerpoint of the mass and the plate smaller than the threshold distance \( r_{th} \).

### Table 1 Numerical parameters set in the FDTD simulation.

|                     | Ball     | Car-tire | Heel       | Body      | Tapp. hammer | Concrete |
|---------------------|----------|----------|------------|-----------|--------------|----------|
| Spring const., \( k \) [N/m] | 6.17 \cdot 10^4 | 1.66 \cdot 10^5 | 1.06 \cdot 10^6 | 1.391 \cdot 10^4 | 2.20 \cdot 10^{10} | —        |
| Damping coef. \( \alpha \) [-]     | 0.141    | 0.141    | 0.141      | 0.0       | 0.103        | —        |
| Viscosity const., \( c \) [N·s/m]  | 55.7     | 156.0    | 1,130.1    | 0.0       | 1.09 \cdot 10^4 | —        |
| Young’s modulus [N/m²]               | —        | —        | —          | —         | 2.0 \cdot 10^{11} | 2.4 \cdot 10^{10} |
| Poisson’s ratio [-]                  | —        | —        | —          | —         | 0.3          | 0.3      |
| Drop height [m]                     | 1.00     | 0.85     | 0.05       | 0.05      | 0.04         | —        |
| Coef. rest., \( R \) [-]            | 0.80     | 0.80     | 0.80       | —         | 0.85         | —        |
| Weight of the human, \( M \) [kg]   | —        | —        | \( M \)    | \( M \)   | —            | —        |
| Weight, \( m \) [kg]                | 2.5      | 7.3      | 0.08\( M \) | 0.92\( M \) | 0.5          | —        |
| Density [kg/m³]                     | —        | —        | —          | —         | 2,400        | —        |
| Radius, \( r \) [cm]                | 4.5      | 24.0     | 0.0        | 0.0       | 50           | —        |
| Threshold distance, \( r_{th} \) [cm] | 2.25     | 12.0     | 0.0        | 0.0       | 0.0          | —        |

3.2. Modeling of Impact Force of Human Trotting

3.2.1. Measurement scheme

The impact forces caused by human trotting were measured using a force sensor (SS-FP40UD, Sports Sensing Co., Ltd.). The measurement setup is shown in Fig. 3. Each force plate has the dimensions of 400 mm × 400 mm, and, as shown in Fig. 3(e), the plates are arranged at an interval of 200 mm. The step lengths of males and females at an ordinary walking speed have been reported as 0.899 ± 0.068 and 0.814 ± 0.095 m, respectively [32]. To ensure that all subjects are not forced to walk with a larger step length than usual, the step length in the measurement setting was set to 0.6 m. As can be seen in Fig. 3(a), the subjects trotted on the six plates from P1 to P6, and the impact force was measured on force plate P5, whereas the dummy plates P1, P2, P3, P4, and P6 were set for ease of simulation of usual trotting. The measurement was performed for 15 subjects comprising 9 males and 6 females in their twenties. The averaged body weights is 59.3 kg (maximum: 85.0 kg, minimum: 36.1 kg, standard deviation: 12.8 kg). To avoid variation of the measured forces owing to the physical features of the surface of the sole of footwear, all the subjects wore the same commercially available sneaker, as shown in Fig. 3.

The example of the measured time-transient characteristics of the impact force is shown in Fig. 4. Note, in this figure, that the numerical results of the impact force obtained by the scheme described in the following section are also indicated. Figures 4(a), 4(b), and 4(c) show the results for of a 85.0 kg male, a 70.0 kg male, and a 36.1 kg female, respectively. These waveforms have curved shapes with long durations of 300 ms up to 600 ms, whereas, particularly under the conditions of (a) and (b), some
discontinuous points can also be seen at around 100 ms. In addition, as shown in Fig. 4(d), the reproducibility of the impact forces for subject (a) is considered to be sufficient to compare them with the calculation results. The impact forces of the other subjects also indicated sufficient reproducibility. Therefore, in this study, one set of data among the three trials was extracted and used for the comparison with the calculation. These time-transient characteristics will be modeled in the next section.

3.2.2. Modeling of excitation by human trotting

The modeling scheme of the excitation force caused by human trotting is described in this section. Firstly, the contact condition between the trotting human and the floor includes two types of excitation phase: one is the contact between the heel of the trotting human and the floor (b-1), and the other is that between the entire surface of the sole of the human (b-2). Clark et al. also adopted this excitation model [33], and they concluded that the weight of the foot (Mass 1, in Fig. 5(a)) contributes to the excitation by the heel of the human (b-1), whereas the weight of the entire body excluding the foot (Mass 2, in Fig. 5(a)) contributes to the excitation by the entire body of the human (b-2). In such a case, the time-transient characteristics of the impact force is modeled as shown in Fig. 5(c). In the modeled excitation, the total force $f_t(t)$ is modeled as the sum of the force of Mass 1 $f_1(t)$ and that of Mass 2 $f_2(t)$ as

$$f_t(t) = f_1(t) + f_2(t).$$

In this study, the impact force of human trotting was modeled using the above scheme. Note that two peaks of the impact forces are reported in many papers [34,35]. In our measured results, the sharp impact peak at the beginning of the excitation by human trotting can be observed as discontinuous points in Figs. 4(a) and 4(b), whereas there are no sharp peaks in Fig. 4(c). These results suggest that the tendencies of the first peak depend on the individual’s way of trotting. Specifically, it was clarified that subject (c), who tended to trot stealthily, landed on the
force plate first on her toes followed by her heel. In contrast, it was also clear that subjects (a) and (b), who did not care if their footsteps were loud, fist landed on the force plate on their heels. In [36], it was mentioned that the first sharp peak can also be seen in the case of landing on the heel, whereas peaks were nominal in the case of landing on the toes or the whole surface of the sole. Yokoyama and Matsunaga [37] reported the time-transient characteristics of the impact force for four subjects. Relatively sharp peaks were observed for two subjects, while no peaks were observed for the other two subjects. Nakamori and Yoshimura [35] indicated impact forces for two subjects, where the appearances of the first sharp peak were different from each other. Because of these results, it was suggested that the impact force largely depends on the manner of the first contact between the sole and floor surfaces. Specifically, the impact force is much more reduced in the case of landing on the toes than on the heel. The following reason may be considered. When a human lands on the floor on only the toes, the ankle joint is used as a spring in order for the heels to not strongly touch the surface of the floor. Consequently, the collision velocity of the heel with the surface of the floor may be greatly reduced after the toes receive the body weight with the ankle joint acting as a spring, and the amplitude of the first sharp peak may also be reduced. In addition, the impact force was also calculated under the conditions that the collision velocity between the heel and the floor for subject (a) was reduced by 10 and 50% of the original one. The results are discussed in the next section.

Then, the impulsive excitations caused by the heel contact (b-1) and the foot contact (b-2) were modeled using the estimated masses Masses 1 and 2, and the spring coefficients between the surface of the heel and the floor and that of the sole and the floor. Firstly, the Masses 1 and 2 were set as 8 and 92% of the entire weight of the whole body as in [33]. Secondly, the spring coefficient, $k_{\text{heel}}$, of the Mass 1 (heel part of the sneaker) was estimated as $k_{\text{heel}} = \frac{EA}{L}$ [13], where $E$ is Young’s modulus, $A$ is the contact area of the heel, and $L$ is the thickness of the rubber of the sneaker sole. Then, $k_{\text{heel}}$ was set to $1.06 \times 10^6$ N/m by using the following parameter values: Young’s modulus was set to $4.6 \times 10^2$ N/m$^2$ assuming the hardness of the rubber of the heel to be 50 deg., $A$ was set to $46.4$ cm$^2$ assuming the surface area of the heel of the sneaker to be 20% of the entire area of the rubber sole, and $L$ was 0.02 m. On the other hand, the leg stiffness of 13.91 kN/m obtained by Coleman et al. [34] was adopted to model the excitation force by Mass 2. Each of the physical parameters are shown in Table 1. As indicated in the table, the drop height of both of the heel and the body are set to 0.05 m. This drop height was determined as follows using a video camera that
recorded the movements of trotting. The maximum height of each subject’s sole was approximately 10 cm; however, at that time, the entire surface of the other foot was still touching the surface of the floor. Thus, the height of the moving sole surface at the time when the other foot was almost leaving the floor was measured from the video recorded to be approximately 4 to 6 cm depending on the subject. For this reason, the height was set as 5 cm in the simulation.

Although $C$ of the heel made of rubber was obtained from Eq. (7) by substituting $R$ of 0.8, suitable data for the damping characteristics of a human leg could not be obtained. However, in this study, the estimated spring coefficient, which includes all the mechanical effects caused by the reaction forces of the elastic action performed by a human [38] as well as the viscoelastic action such as dampers, is adopted. For this reason, $C$ was set to zero only when modeling the elastic effect by Mass 2.

### 3.3. Results and Discussion

#### 3.3.1. Accuracy of simulated forces

First, the impact forces of the impact ball, bang machine, and tapping hammer, dropped from each drop height listed in Table 1 are shown in Fig. 6. The measurement results are taken from [2,14], as shown in the figure. Note that the calculated velocities of the ball, car tire, and tapping hammer at the exact time of collision were consistent with the theoretically obtained collision speed $v = 2\sqrt{gh}$, where $g$ and $h$ are the gravity and the drop height, respectively. The duration and maximum force of the transient force characteristics of the (a) bang machine and (b) impact ball obtained by the FDTD method generally show good agreement with the reference data. In the case of the bang machine, the change in the maximum excitation force with the drop height is also well simulated. Compared with these heavyweight impactors, the lightweight impactor of the tapping hammer shows relatively short duration time and has higher frequency components. Figure 7 shows the impact force exposure levels of these impactors along with the measurement results from [39,40]. In these results, the frequency characteristics of these impactors also show agreement with the reference data. However, the numerical results also show slight differences in higher-frequency ranges, particularly at 125 and 250 Hz for the bang machine and 250 and 500 Hz for the tapping machine. 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 real phenomena are conspicuous in the high frequency range.

Next, the impact forces caused by human trotting were simulated by the following procedure. First, the time-transient characteristics of Masses 1 and 2 caused by dropping a one-dimensional mass from the heights listed in Table 1 were calculated as $f_1(t)$ and $f_2(t)$, respectively. Then, the total force $f(t)$ was calculated using Eq. (9). The results shown in Fig. 4 generally agree well with the measured results. Note that similarly to the case of subject (a), the results of the additionally investigated cases, where the collision velocity between the heel of subject (a) and the floor was reduced by 10 and 50% of the
original one, are also indicated in Fig. 4(a). In these results, as the collision velocity becomes lower, the amplitude of the first sharp peaks indicated in the expanded figure decreases. When the velocity is decreased by 10%, the amplitude of the first peak becomes quite small and almost invisible. Then, the frequency characteristics of these time-transient characteristics of the same three subjects as in the case of Fig. 4 are obtained through FFT and are shown in Fig. 8. The frequency components over 89 Hz are not indicated because of the lack of the sound-to-noise ratio in the measurement. In the cases of (a) and (b), the calculated results slightly differ from the measurement at frequencies lower than 16 Hz, but the general characteristics of the measured results, particularly those above 16 Hz, indicate similar trends to those of the calculated results. As shown in Fig. 8(d), the reproducibility of the impact forces in the frequency domain for subject (a) is considered to be satisfactory as well as the time-domain results in Fig. 4(d). On the other hand, when the velocity of collision between the heel and the floor in the case of subject (a) is reduced, the frequency components of the force, particularly at higher frequencies, is increased, whereas that at much lower frequencies indicate almost the same level as in the original case. In addition, the calculated impact force exposure levels, which are converted from the frequency characteristics in Fig. 8, agree with the measurement for of the three subjects, as shown in Fig. 9, while the calculated levels, particularly in the 63 Hz band of subject (a) and (c), are slightly higher than the measured ones. However, the additionally calculated case of the collision velocities reduced by 50% indicates almost the same level as the measured one. The measured and calculated results originally agreed in the case of subject (b), who has a more prominent first sharp peak than the other two subjects. These results of the additionally calculated cases suggest that the collision velocity in the first contact between the subject’s heel and the floor affects the impact forces particularly in the 63 Hz band, and that the frequency components of the impact force in this band is slightly dependent on the manner of trotting of each subject. Note that the reproducibility shown in Fig. 9(d) is
also considered to be satisfactory. The measured and calculated impact force exposure levels of all the subjects are shown in Figs. 10(a) and 10(b), respectively. Although the frequency characteristics of the lines differ slightly between the measurement and calculation, both of these impact force exposure levels fall in the range between approximately 20 and 30 dB.

### 3.3.2. Accuracy of simulated floor-impact sound

The impact sound pressure level was basically calculated by following JIS A 1418-1 [1] and 1418-2 [2]. To obtain the heavyweight impact sound pressure level, the maximum sound pressure level \( L_{\text{max}} \) in each frequency band was first estimated on the basis of the obtained transient responses filtered to each frequency band. The energetically averaged level among the calculated maximum sound pressure levels at all receiving points Rs1 through Rs5 in Arrangement-1 (Fig. 2(b)) was obtained by setting the excitation points at each of S1 through S5. Then, the impact sound pressure level in each frequency band was obtained by arithmetically averaging the energetically averaged levels for each excitation point. Note that the rigid wall in Arrangement-1 models the plasterboards used for partitioning the receiving room in [31].

On the other hand, the normalized impact sound pressure level \( L_a' \), which was calculated as \( L_a' = L_a + 10 \cdot \log_{10}(A/A_0) \) using the total absorption area \( A \) standardized by \( A_0 = 10 \text{ [m}^2\text{]} \), was determined to evaluate the lightweight floor-impact sound. The lightweight impact sound level should be obtained in steady-state sound field. To simulate such a sound, the impact sound excited by the tapping hammer was overlapped at a time interval of 100 ms [1] as shown in Fig. 11. Then, the values of the equivalent continuous sound pressure level of such an artificially synthesized waveform were used to calculate the lightweight impact sound pressure level. Herein, the total absorption area \( A \) was estimated as \( A = 0.16 \cdot V/T_{30} \) using the reverberation time \( T_{30} \) and the volume of the receiving room \( V \). The reverberation time \( T_{30} \) in each octave band was calculated by using the integrated impulse response method over a 30-dB decay range in the reverberation curve from \(-5\) to \(-35\) dB below the starting level and are shown in Table 2. In the table, the theoretically obtained reverberation times using Sabine’s and Eyring’s theories are also indicated.

Table 2 Results of the averaged reverberation times and their standard deviations with the reference to the theoretically obtained reverberation times based on Sabine’s and Eyring’s theories.

| Frequency [Hz] | 31.5 | 63 | 125 | 250 | 500 |
|---------------|------|----|-----|-----|-----|
| Ave. [s]      | 2.52 | 2.19 | 1.98 | 1.80 | 2.07 |
| S. D. [s]     | 0.02 | 0.14 | 0.09 | 0.06 | 0.06 |
| Theory [s]    | 1.79 s (Sabine), 1.73 s (Eyring) |

The impact sound pressure levels excited by the impactors of the bang machine and tapping hammer are calculated by the FDTD method and shown in Fig. 12. First, the calculated results of the impact sound pressure levels of the bang machine in Fig. 12 show almost the same characteristics as the measured ones. Next, the calculated result of the tapping machine is indicated with Ref. [41] in the same figure. Note that the reference results were obtained by a floor-impact sound test in a two-layered wall-type concrete structure including a 150-mm-thick slab floor with the dimensions of 3,600 mm \( \times \) 5,400 mm. The height of the ceilings of the excitation and receiving rooms were 2,200 mm and 3,500 mm, respectively. In [41], six kinds of normalized impact sound pressure levels were measured by using different tapping machines, all of which were compliant with JIS 1418-1 [1]. The calculated result shows relatively similar characteristics to those indicated by multiple green lines. These comparisons between the calculation and measurement validated our proposed method.
4. APPLICATION TO AURALIZATION OF FLOOR-IMPACT SOUND

The subjective investigation described in this section was performed to confirm the correspondence between the subjective impression of the loudness and annoyance caused by the numerically calculated impact sounds and the maximum A-weighted sound pressure level $L_{A_{\text{max}}}$, which has been experimentally confirmed to agree well with the subjective impression of loudness [42]. In addition, the correspondence between the subjective impression and the physically evaluated results obtained using the $L$-number was investigated as well as $L_{A_{\text{max}}}$. The details of the investigation are described below.

4.1. Numerical Models

The subjective impression of the auralized floor-impact sounds calculated by the proposed method was experimentally investigated. In this investigation, three vibration sources, two floor thickness, and two sizes of the floor were used in the calculation as numerical parameters. Thus, a total of 12 sets of conditions were investigated. Note that the floor dimensions correspond to the room dimensions shown in Fig. 2(d). The source and receiving points were set at S and Rs in Fig. 2(d), respectively. To investigate the sound impression of the floor impact sound inside typical rooms, the statistical-incidence absorption coefficient of 0.1 in the entire frequency range between 16 and 500 Hz bands was set for the surface inside the room. The statistical-incidence absorption coefficient of 0.1 corresponds to a normal-incidence absorption coefficient of 0.055, which is simulated by setting the normal-incidence acoustic impedance of 29,000 Ns/m$^2$. The results of the fast weighted peak levels in each octave band are shown in Fig. 13. Note that each of the condition numbers indicated in Table 1 is additionally shown in the figure. The results for the floor structure with the dimensions of $3.6 \times 3.6\text{ m}$ indicated by the black lines show characteristics with slight peaks compared with those of $3.6 \times 3.6\text{ m}$ indicated by the gray lines. In the figure, frequency characteristics obtained under many conditions have the first mode frequency of the slab in the 63 Hz band, whereas in the specific cases with the 300-mm-thick floor with the dimensions of $3.6 \times 3.6\text{ m}$, the first resonant frequency of the floor was much higher than those under other conditions. Such uncommon conditions were also added to obtain fundamental knowledge of the human response to floor-impact sounds, whereas no such peaks were observed in the floor-impact sound characteristics obtained in the box-type measurement room commonly used for the floor-impact sound experiment.

4.2. Experimental Model

Differences in the subjective impressions of loudness and annoyance regarding the presented sound stimuli were evaluated by Scheffe’s paired [43] comparison method modified by Nakaya [44]. Nakaya’s variation was adopted in this work because all stimuli were subjectively evaluated by each subject. In addition, by using Nakaya’s variation we can disregard the subjective effect of order of presentation of paired stimuli.

In this experiment, 12 types of calculated floor-impact sounds as shown in Table 3, were presented to the subject via headphones (SONY, MDR-1A). Here, the same monaural signals were simultaneously reproduced in the left and right channels of the headphone. The experimental measure of the binaural reproduction of the impact sound can also be applied to increase the presence of the sound; however, in such a case, the following numerical problems...
will arise. The computational cost becomes large for precisely modeling the human. In addition, it becomes difficult to purely evaluate the correspondence between the subjective response and the physical indicators, because the subjective results may also include the effect of sound localization caused by the numerical problems related to the numerical modeling on the FDTD mesh, particularly for the ear parts, such problems have not yet been clarified. Furthermore, it is considered that the binaural impact sound including the prominent frequency components just under the 125 Hz band may not indicate clear localization when taking the directional band of the binaural sounds over 1 kHz into account.

Ten subjects comprising five males and five females between the ages of 20 and 50 participated. On the basis of Nakaya’s variation of Scheffe’s paired comparison method, the relative impressions of one condition compared with the other one for a total of 66 pairs were evaluated as follows. For the loudness evaluation, if the former impact sound is louder than the latter one, the score is 1; if the former is much louder, the score is 2. On the other hand, if the latter impact sound is louder than the former, the score is −1, and if the latter impact sound is much louder than the former, the score is −2. If the two of them are the same, the score is 0. For annoyance, the same scheme of judgement was adopted. Note, for the annoyance evaluation, the subjects were instructed to judge the annoyance by supposing that they are focusing on some activity of daily living such as working on a PC or reading a book. The order of presentation of each pair was randomized. The subjects were allowed to hear and compare the paired stimuli many times, because it is difficult to evaluate the difference in the subjective impressions of loudness and annoyance. Note that, in the experiment, the floor impact sound was reproduced three times for the sounds under each condition with a time interval of 1.0 s. To familiarize the subjects with the experiment, four randomly chosen conditions were additionally evaluated by the subjects as a trial, but the results of the trial evaluation were excluded from the analysis.

The sound reproduction level was set by controlling the output volume from the PC so that the waveform of a pure tone with an amplitude of 1 Pa at 1 kHz would be reproduced at 94 dB on the sound level meter. In addition, to make the frequency characteristics of the sound reproduction by the headphone flat, the impulse response was first measured by reproducing the time-stretched pulse signal (time duration: 2^{17} samples) with flat characteristics. By using the frequency characteristics of the measured impulse response, we made the inverse filter, which flattens the frequency characteristics of the output signal from the headphone, and we convolved with all floor-impact sound data. Then, to check the reproducibility of the floor-impact sound data, the frequency characteristics of the reproduced impact sound measured by setting the headphone on the ears of the head and torso simulator (ACO, Type 8328A) were compared with those of the original signal input to the headphone. As can be seen in Fig. 14, the frequency characteristics are almost the same although the frequency components below 16 Hz cannot be reproduced well because of the low reproduction capability of the headphone in the very low frequency range.

### 4.3. Results and Discussion

Next, the relationship between the obtained evaluation results of the loudness and annoyance and the single-number ratings (maximum A-weighted sound pressure level $L_{A\text{max}}$ and $L$-number) are shown in Figs. 15 and 16, respectively. Note that the frequency components between the octave bands of 8 and 250 Hz were used to obtain $L_{A\text{max}}$.
sounds recorded by in situ measurement shows high loudness experimentally obtained using the floor-impact sound. This is supported by the fact that the subjective impression of loudness indicated lower $L_{\text{Amax}}$ under 50 dB, whereas the impact sound caused by the bang machine and impact ball indicated higher values than that by trotting. Therefore, the mechanism behind the subjective response to impact sounds caused by trotting and by the other two standard impact sources may be different owing to the difference in sound pressure levels. Such a difference in the mechanism of subjective evaluation should be investigated in future work by increasing the variety in the investigated conditions of the impact sound.

By using the numerical results obtained from the vibroacoustic FDTD simulation, we have confirmed that the effect of the excitation characteristics caused by the bang machine, the impact ball, the tapping hammer, and human trotting can be evaluated not only by quantitative methods such as the measurement of impact sound pressure levels, but also by testing the subjective impression of the floor-impact sound with the auralized sounds.

5. CONCLUSIONS

A method of numerical simulation, based on the FDTD method, of the impact forces of a free-fall mass acting on elastic plates employing the one-dimensional contact model is proposed. The impact forces caused by the conventional impactors such as a bang machine, impact ball, and tapping hammer, as well as a more practical vibration of human trotting were modeled by the proposed method. The comparison of the calculated impact forces and the impact sound pressure levels calculated using the impact forces with the measurement results showed a general validity of the proposed method. In addition, the

and $L_{\text{Amax}}$. To reflect the A-weighted filter, the attenuation by the A-weighted characteristics was employed in each 1/3 octave band following [45].

Firstly, the estimated scores of loudness for each condition show good correlation with $L_{\text{Amax}}$ when the coefficient of determination $R^2 = 0.97$, as seen in Fig. 15(a), whereas the correlation with the $L$-number is lower when $R^2 = 0.66$, as seen in Fig. 15(b). This result is supported by the fact that the subjective impression of loudness experimentally obtained using the floor-impact sounds recorded by in situ measurement shows high correlation coefficients of 0.91 to 0.94 with $L_{\text{Amax}}$ [42]. The lower correlation of the subjective impression with the rating of the $L$-number can be attributed to the two data was surrounded by the broken line, which have peaks in the frequency characteristics, as shown by red and green symbols in Fig. 13. By excluding these two data values, we found that $R^2$ is increased to 0.93. On the other hand, the estimated scores of annoyance for each condition show also a good correlation with $L_{\text{Amax}}$ with $R^2 = 0.96$, as seen in Fig. 16(a), whereas their correlation with the $L$-number is good when $R^2 = 0.56$ as seen in Fig. 16(b). In this study, annoyance was evaluated by each subject by a method similar to that used for loudness, except for the specific supposition of imaging themselves performing daily activities such as reading and PC work during the annoyance evaluation. This may be one of the reasons why the loudness and annoyance in Figs. 15 and 16 indicated slightly similar tendencies. Further evaluation of the annoyance with a more realistic environmental setup and longer impact sound exposures of the subjects than in this study should be the target of future work. In these results for both loudness and annoyance, the trotting sounds indicated lower $L_{\text{Amax}}$ under 50 dB, whereas the impact sound caused by the bang machine and impact ball indicated higher values than that by trotting. Therefore, the mechanism behind the subjective response to impact sounds caused by trotting and by the other two standard impact sources may be different owing to the difference in sound pressure levels. Such a difference in the mechanism of subjective evaluation should be investigated in future work by increasing the variety in the investigated conditions of the impact sound.

Fig. 15 Relationship between experimental results of estimated relative score for loudness, and single-number ratings of (a) $L_{\text{Amax}}$ and (b) $L$-number.

Fig. 16 Relationship between experimental results of estimated relative score for annoyance, and single-number ratings of (a) $L_{\text{Amax}}$ and (b) $L$-number.
proposed method was applied to a subjective evaluation experiment of the loudness and annoyance perceived by humans when the numerically obtained floor impact sounds are auralized and heard by the subjects. The results of the subjective evaluation experiment related to loudness and annoyance showed a high correlation with the single-number rating of the maximum A-weighted sound pressure level, and the L-number. In our future work, the proposed scheme will be applied to environmental evaluation inside buildings in more practical cases.

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