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PREDICTION OF NOISE LEVEL FOR TRACTOR CABIN USING STATISTICAL ENERGY ANALYSIS

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ABSTRACT
The purpose of this paper is to establish a method of predicting the noise and vibration of tractor cabins in the engine-idling state by using Statistical Energy Analysis (SEA) [1]. At first, an analytical model of a tractor cabin is constructed, and power flow equations [2] are formulated for the tractor cabin. To solve these equations, SEA parameters are estimated experimentally and analytically. These parameters are the modal density, loss factor, coupling loss factor, and input power. With these parameters, the noise and vibration responses of the tractor cabin are calculated. Good agreements are found between the analytical and experimental data.

1 - INTRODUCTION
Recent years have shown this era to view lower noise and vibration levels as important, and there have been demands for reduced noise and vibration levels to better the working environments of workers, even those who drive vehicles for agricultural use such as tractors. Because of this, noise and vibration estimation has been carried out, and there has been a growing awareness of the need to establish measures to reduce noise and vibration levels.

In this research, we adopt a tractor cabin as the object of the research, and we estimate the noise and vibration responses of the tractor cabin by using Statistical Energy Analysis (SEA) as the method of estimation. This tractor cabin has a complex shape constructed from components that use various materials and that have various shapes. Moreover, the inside of the cabin is a closed space. With the SEA method, we consider structural components as a set of equivalent vibrating elements, and moreover, we evaluate the vibration condition of the components as a macroscopic statistical average quantity versus the frequency band and space (by using the energy), by assuming that certain vibration modes within some frequency bands are uniformly distributed and are excited to the same degree.

2 - MODELING OF THE TRACTOR CABIN
In this research, we have constructed a complex structure (i.e., a tractor cabin) composed of components that use various materials and that have various shapes, and have estimated the noise and vibration within the closed-space tractor cabin by using the SEA method. This research has assumed the engine-idling state (for the tractor cabin), and has established conditions for noise and vibration. This is because our aim has been to determine the reductions in noise and vibration levels during the idling state. If an estimation method for noise and vibration levels under the idling state were to be established, then it would also be applicable under other conditions.

Length	mm	3050
Width	mm	1350
Height	mm	1990
Weight	kg	1290
Engine Type		4-cycle diesel engine
Total Displacement	cc	1499

Table 1: Main specification of the tractor.

Table 1 shows the main specifications of the tractor used in this research. Here, because we have assumed analysis with the SEA method, we have assumed that the tractor cabin is a composition of 18 components, and have constructed the SEA model accordingly. The component number of each of the individual components of the tractor cabin in the SEA model is as shown in Fig. 1.

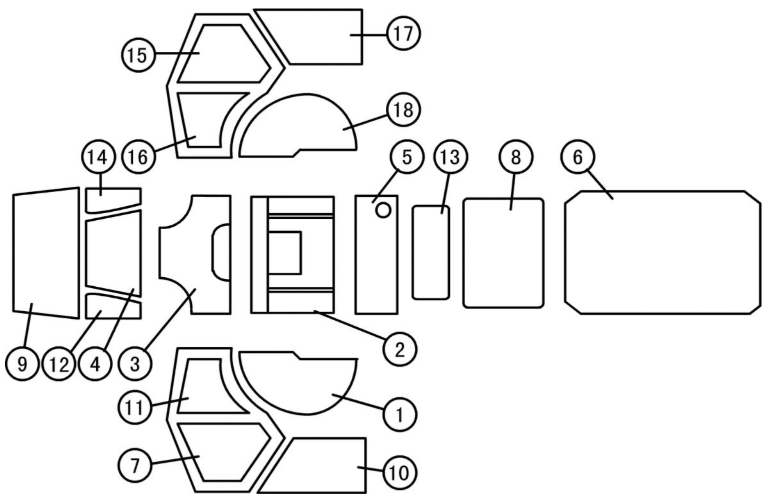

Figure 1: Modeling of the tractor cabin.

3 - SEA PARAMETERS

In order to solve the power flow equations, it is necessary to determine the SEA parameters, i.e., the modal density, loss factor, coupling loss factor, and input power. Below, an explanation is given on a method of calculating SEA parameters for tractor cabins.

Modal density $n(f_o)$:

The modal densities of the structural components are first determined by using a test equation that determines the modal densities by vibration excitation experiments (proposed by Clarkson et al. [3]).

$$n(f_o) = \int_{f_1}^{f_2} 4M\mathrm{Re}\left\{\frac{-iA(f)}{2\pi f F(f)}\right\} df \tag{1}$$

Here, f_o is the center frequency, M is the mass of the component, $F(f)$ is the drive point vibration-added force, and $A(f)$ is the vibratory acceleration. In addition, the modal density of the sound field component is determined theoretically, since it is difficult to measure the modal density.

Loss factor η:

Clarkson et al. [3] proposed the following equation as a test equation to determine the loss factor of structural components in vibration excitation experiments.

$$\eta = \frac{\displaystyle\int_{f_1}^{f_2} \mathrm{Re}(Y)\, F^2 df}{\omega_o M \left\langle \displaystyle\int_{f_1}^{f_2} v^2 df \right\rangle} \tag{2}$$

Here, Y is the drive point mobility in the range of f_1 to f_2, F^2 is the power spectrum of the added vibratory force, and v^2 is the power spectrum of the response speed. In addition, $\langle\rangle$ indicates the space average. The loss factor of the sound field component is determined theoretically, since it is difficult to measure the loss factor.

Coupling loss factor η_{ij} :

Because it is difficult to measure the coupling loss factor between components in a tractor cabin, here, we use the following equations which show the coupling loss factor between monotonous flat components, and the coupling loss factor between a monotonous flat component and sound field component respectively [1].

$$\eta_{ij} = \frac{c_{gi} L_c \tau}{\pi \omega S_i} \tag{3}$$

$$\eta_{ij} = \frac{Z_o S_c \sigma}{\omega M_i} \tag{4}$$

Here, c_{gi} is the group speed of the bending waves, S_i is the surface area, M_i is the mass, τ is the energy transmissibility, σ is the radiation efficiency, L_c is the coupled length, and S_c is the coupled surface area. The energy transmissibility differs depending on the coupled shape, e.g., L-shape, T-shape, or so on [4].

Input power:

The origin of the input power for a tractor cabin is the engine. We use the term "vibrational input power" for the input power when the engine vibration propagates through the frame, the cabin mounts and the cabin. We use the term "acoustical input power" for the input power when the engine noise propagates through the air and to each component.

As an example, Fig. 2 shows the input power of component No. 4. In Fig. 2, the vibrational input power is 10 to 30 dB larger than the acoustical input power. Even for the other components, the same tendency can be observed. In short, it can be understood that the influence of the vibrational input power is large when engine vibration is the vibrational force.

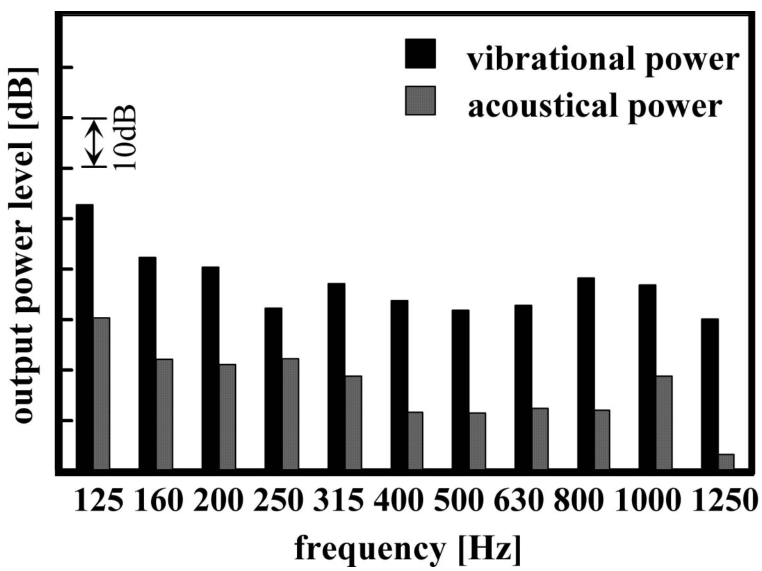

Figure 2: Input power of component No. 4.

4 - ESTIMATION BY THE SEA

4.1 - Acceleration response estimation

We formulated power flow equations by using the power flow relationships for the tractor cabin in Figure 1, and determined the acceleration response of each component by introducing the SEA parameters. The obtained results are shown in Fig. 3. Here, a 1/3 octave analysis was performed assuming the frequency range for the estimation to be from 125 Hz to 1250 Hz (where noise and vibration become problematic). In Fig. 3, results are also shown in which acceleration responses were measured by attaching acceleration pick-ups to each component while the engine was idling. In order to obtain the space averages of the measured values, the acceleration pick-ups were attached at 20 different locations for each component, and the average was taken.

From the above results, we arranged practical limits on the SEA method for the tractor cabin used in this research. For example, when the practical limits were arranged to obtain differences of 3 dB or less, the mode count within the frequency band for the estimation was 5 or more with components of a monotonous flat shape and was 15 or more with components of a complex shape. It is therefore possible to perform a better estimation with good accuracy as the frequency or component surface area is increased.

4.2 - Sound pressure estimation

Next, we determined the sound pressure levels of the components in the space within the cabin by using SEA method. Fig. 4 shows the results obtained. Then, we measured the sound pressure levels within the cabin by considering the influence of calculation involving transmitted sounds by employing the mass

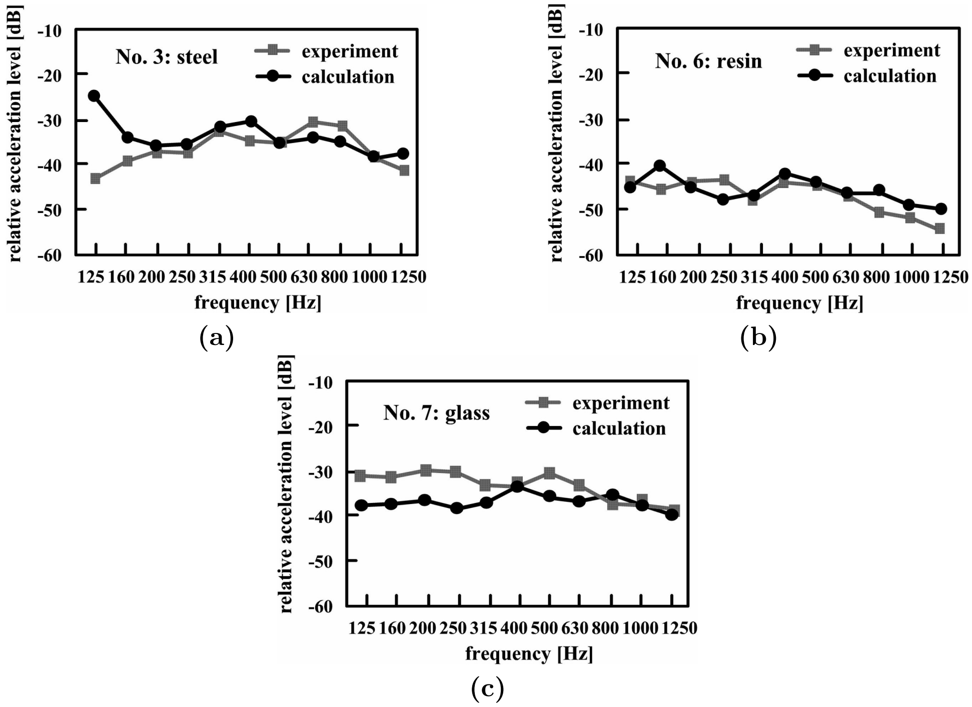

Figure 3: Estimation result of the acceleration response of individual component.

law. Fig. 4 shows the result mentioned above. In the Fig. 4, the result of the experiment and SEA (with air-borne sound) agree quite well. From the above results, we could clarify the possibility of noise and vibration response estimation by considering the influence of transmitted sounds in the SEA method.

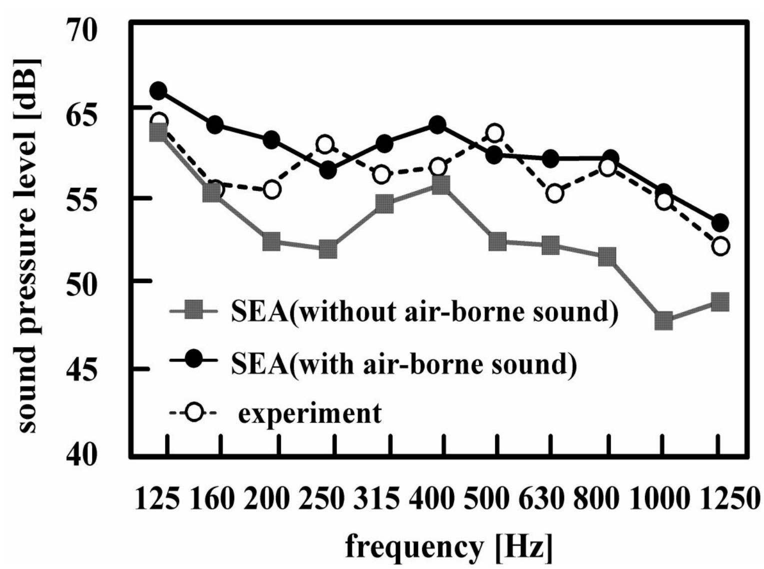

Figure 4: Result of estimating the sound pressure level in a cabin.

Predicted sound power level for each cabin part are shown in Fig. 5. It is shown from this results, No. 2, 3 and 6 components affect the cabin noise. Therefor, assuming that the sound absorbing material of sound absorption coefficient $\alpha = 0.3$ is used to these components, sound pressure level is estimated. From these results shown in Fig. 6, it is obtained the most effective sound absorbing component to reduce the cabin noise is No. 6 which is roof panel.

5 - CONCLUSION

In this research, the following points have been clarified:

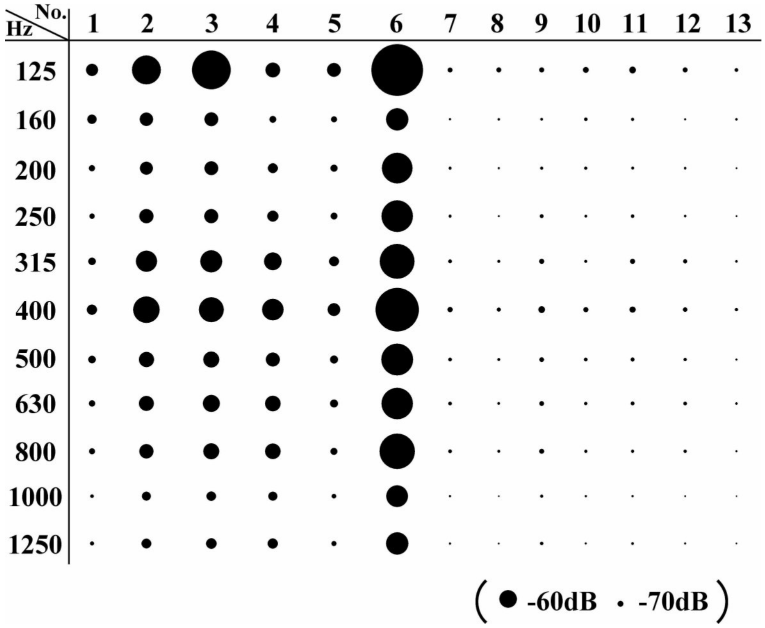

Figure 5: Sound power level of cabin parts.

- By determining the SEA parameters (i.e., modal density, loss factor, coupling loss factor, and input power) of each component of a tractor cabin, it becomes possible to carry out acceleration response estimation by using SEA.
- Because the SEA method considers only solid propagating sounds for the sound pressure within a cabin, there is a difference of 7 to 8 dB compared with actual measured values. However, when one considers the influence of transmitted sounds, it is possible to obtain a sound pressure estimation within a cabin with good accuracy.

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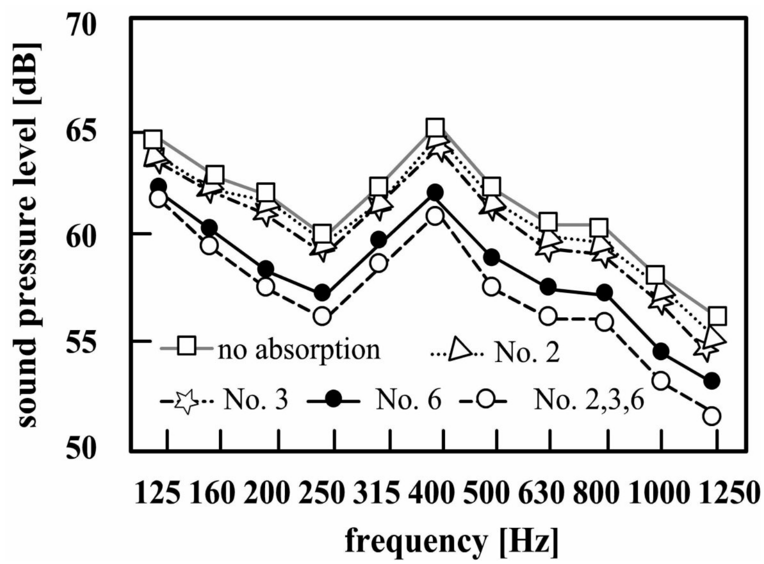

Figure 6: Estimated sound pressure level using sound absorbing material.