Extrapolation Technique for Acoustically Induced Random Vibration of Honeycomb Panel

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Summary
This paper applies extrapolation technique to the prediction of acoustically induced random vibration of honeycomb panel. Statistical Energy Analysis (SEA) has been employed to predict vibroacoustic problem for space systems. SEA modelling includes the definision of subsystem and its parameter, therefore, it is very time-consuming work. Extrapolation technique, on the other hand, can predict the response based on the old experimental data acquired in the past and can give the result expeditiously. In this paper, in order to evaluate the accuracy of the extrapolation technique, the acoustically induced random vibration of satellite honeycomb panel under diffused sound field is predicted. The result of the prediction is compared with SEA result and acoustic experiment shows that extrapolation technique gives as satisfactory result as SEA.
Keywords: *Extrapolation technique, random vibrations, acoustics*

1. Introduction

Satellite structure inside a launch vehicle shroud is exposed to intense (more than 130dB) vibroacoustic environment during launch. In order to confirm that satellite structure and its equipment withstand the launch environment, ground acoustic test is conducted in high intensity diffused sound field in the process of design verification. Random vibration specification at the interface point of each satellite equipment needs to be determined for its design prior to acoustic test[1].

Since vibration response to acoustic test environment is of high frequency in random, extrapolation technique based on experimental data [2,3] and Statistical Energy Analysis (SEA) [4] have been employed for the prediction of the response. In SEA, structure and acoustic cavity of interest are divided into the SEA subsystems, and SEA parameters (damping/coupling loss factors) are incorporated into the SEA model. SEA modelling is a time-consuming work, because a large number of SEA parameters will be necessary to model a whole satellite composed of many honeycomb panels and equipments. On the other hand, extrapolation technique uses the baseline (old) data empirically / experimentally acquired and predicts the new system based on the scaling technique[2]. Therefore, extrapolation technique does not require any modelling work, and the response of the new system will be obtained expeditiously.
In this paper, in order to verify the accuracy of extrapolation technique, the vibration response of satellite honeycomb panels under diffused sound field is predicted by extrapolation, and the prediction result is compared with the acoustic experiment result and SEA.

2. Extrapolation technique

Extrapolation technique is an approach based on scaling of physical parameter of the system considered[2]. The technique was developed as empirical methodology for the vibroacoustic prediction using the flight and test results. Extrapolation technique does not require any models and its prediction accuracy depends on the baseline data (old data obtained in the past).

The principle of scaling is that "all physical systems can be expressed in nondimensional parameter." As Figure 1 shows, it provides the prediction of the analysis object (referred to as "New") by extrapolation based on baseline data (referred to as "Old").

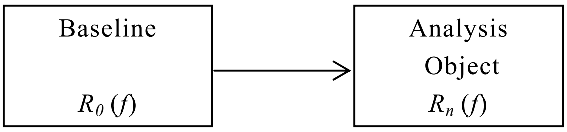

Figure 1 Prediction Model

The vibration response $R_n(f)$ of analysis object is described in equation (1) with baseline vibration response $R_0(f)$. R is the square of velocity or acceleration. Attention should be paid to the frequency that may be subjected to change as discussed later.

$$R_n(f) = R_0(f) \times C_m \times C_{ml} \times C_d \times C_e \tag{1}$$

where C_m is the density ratio, C_{ml} is the equipment mass ratio, C_d is the loss factor ratio, and C_e is the excitation force ratio.

The energy of the structure is assumed to be invariant with and without equipment. In this case, the structural energy E is represented by equation (2).

$$E = M\,v^2 = (M + M_c)\,v_c^2 \tag{2}$$

where M is the structural mass without equipment, M_c is the mass of equipment, v is velocity without equipment and v_c is the velocity with equipment. Hence, when structure density is ρ, thickness is H, and area is A, the term C_{ml} is obtained by equation (3).

$$C_{ml} = \left(\frac{\rho HA}{\rho HA + M_c}\right)_{New} \times \left(\frac{\rho HA + M_c}{\rho HA}\right)_{Old} \tag{3}$$

Since vibration response to acoustic excitation is linear, the excitation term C_e is expressed by equation (4).

$$C_e = \frac{p_{New}^2}{p_{Old}^2} \tag{4}$$

where p^2 is the square pressure. In general, C_m and C_d are represented by equation (6), because the analytical expression of modal analysis for the acoustic excitation problem is obtained in the form of equation (5).

$$w = \frac{1}{\rho^2 \eta} \times (Modal\ Terms) \tag{5}$$

$$C_d = \eta_{Old} / \eta_{New}\ ,\ C_m = (\rho_{Old} / \rho_{New})^2 \tag{6}$$

where η is damping loss factor. Finally, frequency shift is considered. Strouhal number S_t of the nondimensional parameter will be invariant for dynamically identical systems, and it is shown as;

$$S_t = \frac{f \times L}{V} = \frac{f_{New} \times L_{New}}{V_{New}} = \frac{f_{Old} \times L_{Old}}{V_{Old}} \tag{7}$$

where f is the frequency, L is the representative length of the system and V is the velocity of the structure wave. When the representative length L

is defined by square root of the area and the V is the phase velocity of longitudinal wave that is identical for same materials, the following equation is derived.

$$f_{New} = f_{Old} \times \sqrt{\frac{A_{old}}{A_{New}}} \times \frac{V_{New}}{V_{Old}} \tag{8}$$

It is found that the larger the analytical subject, the lower the frequency band shifts.

3. Application of Extrapolative Technique to Satellite Honeycomb Sandwich Panel

Extrapolation technique described in chapter 2 is applied to satellite honeycomb sandwich panels. The honeycomb panels dedicated to the analysis are shown in Table 1. The extrapolation technique are applied to Panel 2 to 11 based on the experiment result of Panel 1 as the baseline.

Since honeycomb panel is a non-homogeneous structure, it is necessary to derive the density of the equivalent panel for extrapolation. We find the panel that have the same significant parameters as honeycomb panel. The flexural rigidity D_1, modal density n, surface density ρh and critical frequency ω_c are significant parameters for the vibroacoustic prediction. These parameters are given by the following equation.

$$D_1 = \frac{E_s t_s (t_s + t_c)^2}{2(1-\nu_s^2)},\ n = \frac{A}{4\pi}\sqrt{\frac{\rho h}{D_1}},\quad \omega_c = c_0^2 \sqrt{\frac{\rho h}{D_1(1-\nu_s^2)}} \tag{9}$$

It can be seen from equation (9) that three parameters in equation (9) are conserved when the Young's modulus and Poisson's ratio of the equivalent plate is the same as those of honeycomb skin and the flexural rigidity and surface density are retained. Therefore, the equivalent thickness and equivalent density are presented as follows.

$$t_{eff} = \sqrt[3]{6t_s(t_S + t_c)^2},\ \rho_{eff} = \frac{2\rho_s t_s + \rho_c t_c}{t_{eff}} \tag{10}$$

Table 2 shows the C_m and C_{ml} values for Panel 2 to 11 and the distribution plot of values is illustrated in Figure 2. The density term ranges from 0.32 to 2.01 and mass term from 0.56 to 3.02, and they seem to be independet each other.

Table 1 Satellite honeycomb panels

Name	Comment	L	W	Mass	Skin	Skin Thickness	Honeycomb Core	Core Thickness	Coincidence Frequency	Fundamental Frequency
–	–	m	m	kg	–	mm	–	mm	Hz	Hz
Panel1	–	1.82	0.91	6.20	Aluminum	0.3	Aluminum	25	440	132
Panel2	–	1.82	0.66	4.55	Aluminum	0.3	Aluminum	25	442	236
Panel3	–	1.82	0.25	2.05	Aluminum	0.3	Aluminum	25	482	1497
Panel4	with Insert	1.82	0.91	7.60	Aluminum	0.3	Aluminum	25	487	111
Panel5	Φ300mm hole	1.82	0.91	6.00	Aluminum	0.3	Aluminum	25	441	127
Panel6	CFRP Skin	1.82	0.91	4.00	CFRP	0.3	Aluminum	25	277	198
Panel7	with Heat Pipe	1.82	0.91	8.70	Aluminum	0.3	Aluminum	25	522	110
Panel8	Core100mm	1.82	0.91	10.60	Aluminum	0.3	Aluminum	100	145	391
Panel9	Core50mm	1.82	0.91	7.10	Aluminum	0.3	Aluminum	50	236	240
Panel10	Skin1mm	1.82	0.91	11.10	Aluminum	1.0	Aluminum	25	314	182
Panel11	CFRP Skin	2.00	0.40	2.15	CFRP	0.18	Aluminum	15	635	148

Table 2 Mass and density ratios for extrapolation (Baseline 'Old' is Panel 1)

Panel	Density Term C_m	Mass Term C_{ml}
Panel 2	1.00	1.36
Panel 3	1.00	3.02
Panel 4	1.00	0.82
Panel 5	1.00	1.03
Panel 6	2.01	1.55
Panel 7	1.00	0.71
Panel 8	1.86	0.58
Panel 9	1.53	0.87
Panel 10	0.32	0.56
Panel 11	2.01	2.88

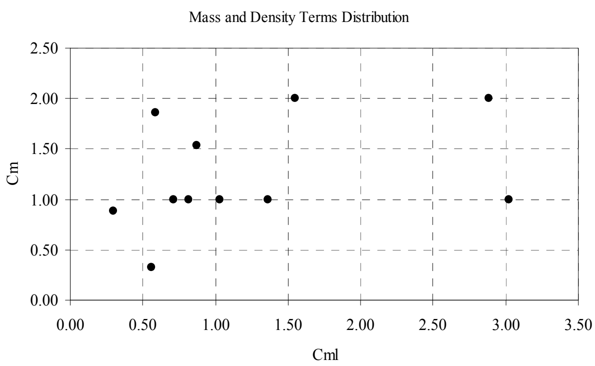

Figure 2 Mass and Density Term Distribution

Experimentally measured damping loss factors for each panel are applied to the damping loss factor term C_d . The frequency shift calculated by equation (8) is applied when the value of frequency shift is greater than each 1/3 octave bandwidth. The panel subjected to frequency shift is Panel 3 (shifted two higher 1/3 octave band) and Panel 11(shifted one higher 1/3 octave band).

4. Result of Extrapolation Technique and Its Comaprison with SEA and Experiment

Figure 3 to 10 show the extrapolation prediction results of the honeycomb panels along with SEA result and acoustic excitation experiment results. The extrapolation results show favorable accuracy, except for Panel 2 and 3 that are seen to be close to the beam, and for Panel 8 and 10 where the assumption as a flat plate is violated due to the thickness of the core and skin. The deep valleys at 315Hz observed in Panel 5 and 7 is due to the valley of the measured damping loss factor term C_d . In frequencies lower than 100Hz, the extrapolation predictions yield smaller prediction errors compared to SEA results which are likely to overpredict in low frequency (low modal density). Howerver, this frequency range is not so important concern in terms of satellite structure design. Extrapolation technique as well as SEA can be applied to the frequency band in which the vibration response is dominated by several modes rather than specific individual mode. Furthermore, extrapolation technique is not applicable to the case where inter-structural coupling is more dominant than acoustic excitation path.

The result of the comparison indicates that the extrapolation prediction is not always more accurate than SEA, but it provides as satisfactory result as SEA. Further investigation seems to be required to make a comparison of the extrapolation with SEA for structures to which SEA is not applicable such as antenna, beam and so on.

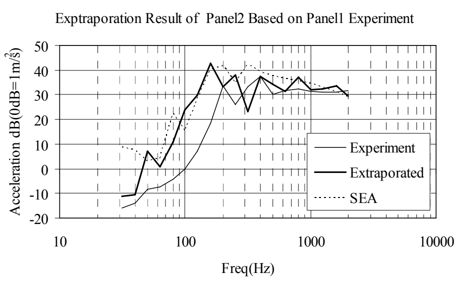

Figure 3 Prediction Result for Panel 2

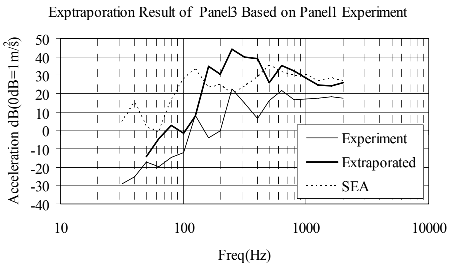

Figure 4 Prediction Result for Panel 3

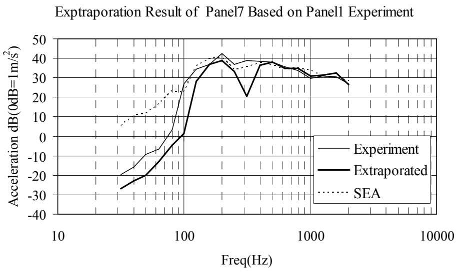

Figure 8 Prediction Result for Panel 7

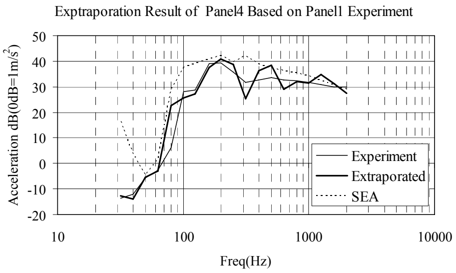

Figure 5 Prediction Result for Panel 4

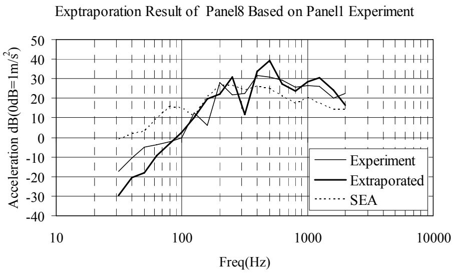

Figure 9 Prediction Result for Panel 8

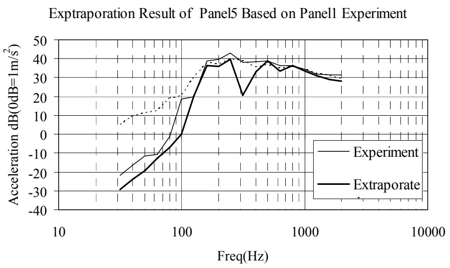

Figure 6 Prediction Result for Panel 5

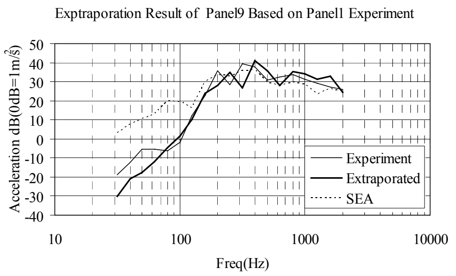

Figure 10 Prediction Result for Panel 9

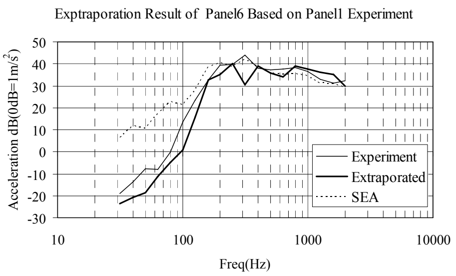

Figure 7 Prediction Result for Panel 6

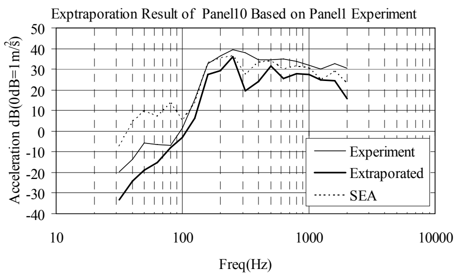

Figure 6.4-10 Prediction Result for Panel 10

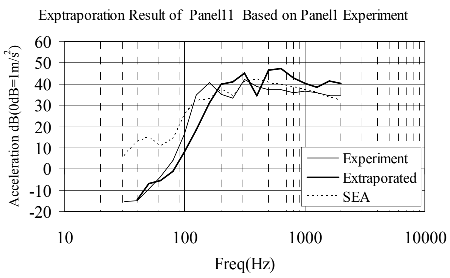

Figure 6.4-11 Prediction Result for Panel 11

5. Conclusion

In this paper, the prediction of acoustically induced random vibration of honeycomb panel has been conducted by extrapolation technique. The results of extrapolation technique for eleven honeycomb plates have been compared with the acoustic experiment results and Statistical Energy Analysis. The result from the comparison shows that extrapolation technique provides good prediction result and is more accurate than SEA in low frequencies below 100Hz. However, extrapolation technique does not lend itself to the case where inter-structural coupling is dominant to structural-acoustic coupling. Extrapolation technique may be the best approach to structures to which SEA cannot be applied, and this issue will be a future work.

Acknowledgment

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