VIBRATION RESPONSE OF SPACECRAFT UNDER FILL EFFECT

S. Ando(1), *Q. Shi* (2), *M. Tsuchihashi* (2)
Space Structures Engineering Group, Institute of Aerospace Technology (1),
Environmental and Structural Test Center, Office of Flight Operations(2),
Japan Aerospace Exploration Agency
2-1-1 Sengen, Tsukuba, Ibaraki, 305-8505 Japan

OVERVIEW

Fill effect or fill factor is the phenomenon, which was recognized in 1990's, that the sound pressure level (SPL) of interior fairing will increase up to 5-10dB than the empty fairing particularly at narrow gap between the walls of fairing and surface of satellite facing the fairing wall, when spacecraft is filled into. There have been few reports to discuss the vibration response of the structures under fill effect, which is the original motivation of acoustic test of spacecraft. This paper discusses the mechanisms of fill effect by using analysis approaches of Statistical energy analysis (SEA), Finite element method (FEM), and Boundary element method (BEM) and vibration response due to the fill effect is analyzed theoretically and acoustic test is applied to verify the analysis results.

1. INTRODUCTION

Launch vehicles, payloads and their components experience severe high-level random acoustic loads during launch phase. Acoustic loads to the spacecraft and payload are particularly severe and important when the vehicle is lift-off and at the transonic phase. Prediction and analysis of such severe random acoustic loads applying on payloads and their components is of significance to the specification assignment, design and verification of spacecraft. Besides, in acoustical environment, fill effect is one of the important issues which will surpass the definition of sound pressure level based on unfilled or empty faring. Fill effect is the phenomenon, indicated a few decades ago, that the sound pressure level (SPL) will be increased in the narrow gap between spacecraft and launch vehicle's fairing wall [1-4]. Fill effect also occurs inside the satellite itself, for example, in a narrow gap of solar array paddle and primal structure [5]. However, there have been few reports to discuss the fill effect induced vibration response, which is fundamental issue of vibroacoustic environment of spacecraft design criteria. If the fill effect would only affect the sound pressure level increase and no effect on the acoustic induced vibration response, the fill effect would not be worth being considered in the definition of vibroacoustic environment. The ground acoustic testing may be reduced to the acoustical environment level defined for the empty fairing.

Researches on this phenomenon dates back to the early 1980's. Many researches have been conducted by NASA to analyze and predict the SPL increase of payload filled interior fairing in comparison to the unfilled environment (fill effect) for the spacecraft. These researches modelled the phenomenon based on the statistical point of view, by SEA (Statistical Energy Analysis). Empirical approaches for fill effects prediction were reported [6]. Analytical prediction of the SPL increase by using the theory of SEA is proposed [7-8]. However, although the prediction equation can obtain the result that almost agrees with an experiment value in the high frequency, unfortunately, it may produce the error 15dB or more in the low frequency, which is dominated by specific low modes. The phenomenon of SPL increase in low frequency was discussed by using FEM (Finite Element Method) and BEM (Boundary Element Method) [5,9].

This paper discusses and clarifies the mechanism of fill effect and vibration response of the structure backed by the acoustic cavity, in which the sound pressure level increases due to fill effect, by the vibroacoustic coupling approaches. Acoustic tests of full-scale fairing/spacecraft and spacecraft panels are conducted to verify the results of the vibration analysis.

2. REASON OF FILL EFFECT

Fill effect is a phenomenon that the sound pressure level (SPL) in the narrow cavity between spacecraft and launch vehicle's fairing wall, is greater than that of wide cavity (see FIG 1). This SPL in the narrow cavity is greater than that of unoccupied (empty) fairing cavity, which SPL is equal to wide cavity of occupied fairing. Fill effect may be caused by [3],

Mainly, surface area increase of acoustic cavity due to the spacecraft's fill into the fairing, which results in the increased number of acoustic cavity modes. These acoustic cavity are an enlarged energy reservoir to store more sound energy. In the frequency where acoustic modes are dense, the energy of the cavity is increasing. In the low frequency range, specific low order resonant mode dominates the sound pressure level in resonant frequency.

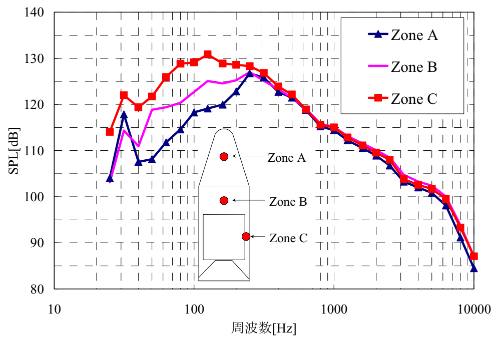

FIG 1. Sound pressure level at different location interior fairing.

Therefore, the author's idea about Fill Effect is that the resonant of the acoustic cavity is, in the end, the phenomenon of fill effect. The resonant frequency of the enclosed or unenclosed cavity may be solved by FEM (finite element method) or BEM (boundary element method). For simple rectangular enclosed cavity, the analytical mode frequencies can be solved by following Eq. (1).

$$f_{m,n} = \frac{c_0}{2}\sqrt{\left(\frac{m}{L_x}\right)^2 + \left(\frac{n}{L_y}\right)^2}\quad (m,n = 1,2,\ldots) \qquad (1)$$

Where L_x, L_y are the surface dimension, c_0 is the speed of sound of air, and m,n is the mode number integers. For example, the rectangular cavity with surface dimension 1m×1m, the frequency of (m=1, n=1) acoustic mode is about 240Hz.

In practice, boundary of cavity is not a rigid wall. Boundaries are open air at the air gap boundary and elastic at cavity-plate surface. The approximation of rigid wall boundary approximates the resonant mode of acoustic cavity and offers a rough estimate of resonant frequency.

The analysis model is shown in FIG 2. The two panels were modelled by FEM, and cavity between two panels and reverberant acoustic excitation were modelled by BEM. FIG 3 shows the SPL difference between those of cavity and reverberant acoustic field (excitation) in the case of air gap 200mm [5]. It is clear that the resonant frequency of the acoustic cavity (air gap) is 200Hz, a bit small than the approximation by Eq. (1), due to the boundary difference. The analysis of pressure increase agrees well with the test result. The pressure distribution at resonant frequency on the surface is shown in FIG 4, which is close to the (m=1, n=1) mode shape, a half sinusoid distribution which has peak value at the centre of surface, and lowest pressure at the edge. Furthermore, the test result FIG 5 shows the resonant frequency and peak pressure are gradually going down, as the gap of the air cavity becomes larger.

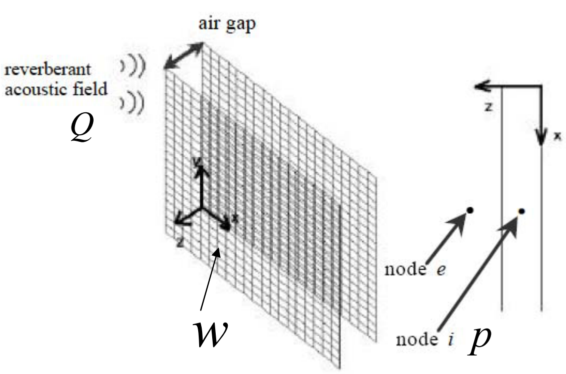

FIG 2. Test and analysis model for rectangular cavity

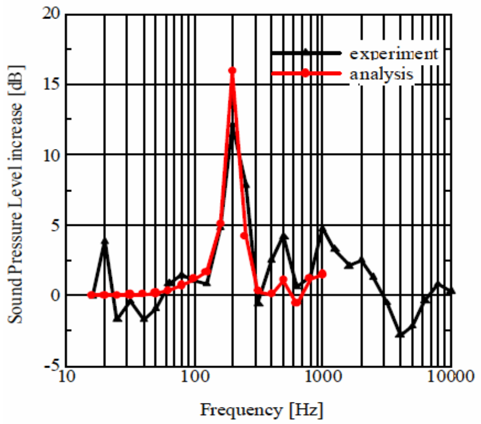

FIG 3. Comparison between analysis and experiment results (gap 200[mm])

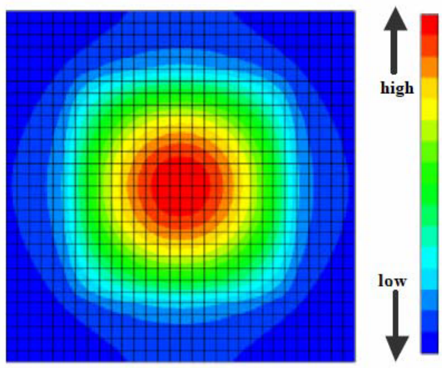

FIG 4. Pressure distribution of cavity at resonant frequency

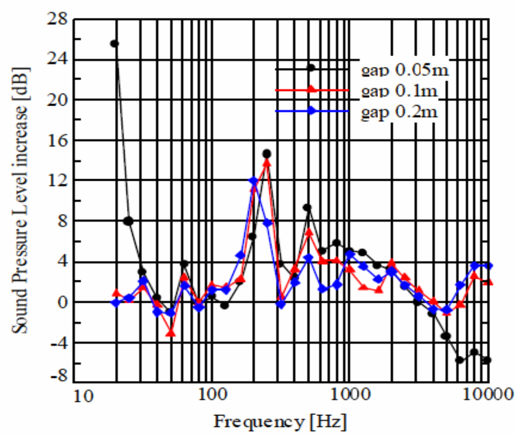

FIG 5. Measurement results of SPL increase of air gap for these three different testing configurations

3. STRUCTURE VIBRATION RESPONSE ANALYSIS UNDER FILL EFFECT

As discussed in section 2, Fill effect is mainly caused by the increase of the acoustic modal density. The term 'modal density' is usually employed in high frequency range problem, in which Statistical Energy Analysis may be a good approach. In low frequency range where specific low order acoustic mode dominates the SPL, individual structural-acoustic modal coupling may be considered. These two approaches are applied in the following.

3.1. Structural-acoustic Individual Mode Coupling (Low frequency model)

The coupling of structural and acoustic modes when excited by acoustic volume velocity Q in FIG 2 may be simplified by two modes (oscillators): pth structural mode and nth acoustic mode. This model ignores the influence of remote natural modes, so that becomes appropriate when both structural and acoustic modes are sparsely distributed in frequency domain.

$$\ddot{w}_p + 2\xi_p\omega_p\,\dot{w}_p + \omega_p^2\,w_p = -\frac{\rho_0 S}{\Lambda_p}C_{np}\dot{\Phi}_n + \frac{F_p}{\Lambda_p} \tag{2a}$$

$$\ddot{\Phi}_n + 2\xi_n\omega_n\,\dot{\Phi}_n + \omega_n^2\,\Phi_n = \frac{c_0^2 S}{\Lambda_n}C_{np}\dot{w}_p - \frac{c_0^2 Q_n}{\Lambda_n} \tag{2b}$$

where, w_p, Φ_n are pth vibration mode displacement and nth velocity potential which is related to nth mode pressure p_n as $p_n = -\rho\dot{\Phi}_n$, ω_p, ω_n are the structural and acoustical mode circular frequency, ξ_p, ξ_n are damping ratio of individual modes, S is the surface area of panel structure, ρ_0, c_0 are density and speed of air, F_p is the model force acting on the structure, and C_{np} is the structure and acoustical coupling coefficient defined by the average value of product of structure and acoustic mode shape. Individual modal mass Λ_p, Λ_n can be expressed as $\Lambda_p = M/4, \Lambda_n = V/8$ where M, V are mass of the panel and volume of the cavity, when both strucural and acoustic mode shapes are sinusoidal.

Eq. (2) indicates the two modes is coupling via $\dot{\Phi}_n$ and $\dot{w}_p$, which is called 'gyrostatic coupling' by Rayleigh. Energy exchange between the two modes is involved but energy dissipation is not involved in the coupling mechanism. The acoustic mode may acts as a dynamic absorber on the structure, and vice versa.

In the following, we consider structure panel backed by rectangular cavity is excited by diffuse sound field using Eq. (2). Modal force F_p and modal volume velocity Q_n can be roughly written as,

$$F_p = \int_S P(\mathbf{r})\phi_p(\mathbf{r})dS \approx \sqrt{\int_S P^2(\mathbf{r})dS}\cdot\sqrt{\int_S \phi_p^2(\mathbf{r})dS} = \frac{PL_xL_y}{2} \tag{3a}$$

$$Q_n = \frac{2\zeta_n\omega_n\Lambda_n P\times 10^{\Delta L/20}}{\rho_0 c^2} \tag{3b}$$

Where, $\phi_p(\mathbf{r})$ is structural mode shape, $P(\mathbf{r})$ is pressure of diffuse sound field, ΔL is pressure increase in dB due to Fill Effect for the uncoupled cavity. Eq. (3b) is readily derived considering resonant frequency of uncoupled acoustic mode. The set of Eq. (3) gives a *rough* estimate for the acting force on two modes, but will suffices for *qualitative* understanding of the coupling.

Uncoupled strucure response using only Eq. (2a) setting $C_{np} = 0$ can now be compared to the coupled response using Eq. (2a, 2b). Table 1 shows the parameters for the coupling calculation. ΔL is set as 15dB for conservative analysis referring to Fill Effect data[8]. C_{np} is reduced to 0.025, since maximum value of coupling tem $C_{np} = 0.25$ is very unrealistic[10].

TAB 1. Parametes for coupling calculation

Variable	Value
L_x	2m
L_y	1m
M	10kg
ω_p	650rad/s(=105Hz)
ω_n	628rad/s(=100Hz)
ξ_p, ξ_n	0.025
c_0	340m/s^2
ρ_0	1.2kg/m^3
P	1Pa
ΔL	15dB
C_{np}	0.025

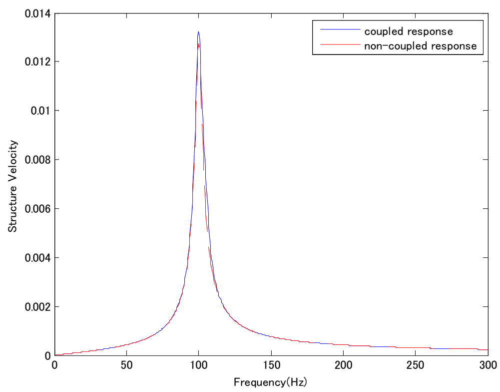

FIG 6a Panel velocity (gap=1m)

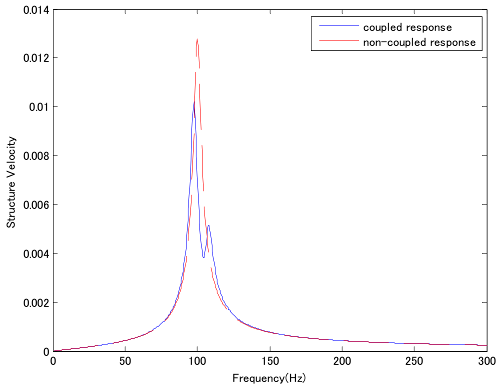

FIG 6b Panel velocity (gap=0.2m)

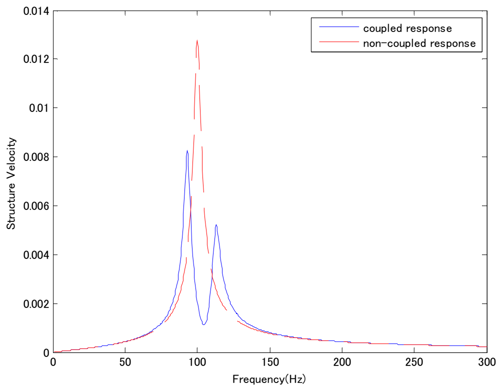

FIG 6c Panel velocity (gap=0.04m)

The structure panel velocity results for the case of cavity gap(depth)=1m, 0.2m, 0.04m are shown in FIG 6a-6c. As the cavity gap becomes narrower, the coupling is stronger and velocity peak split into two peaks. As discusssed earlier, velocity valley is found in 100Hz caused by the dynamic absorber effect of acoustic mode (FIG 6b and 6c). Also, it is observed that the peak velocity tends to be attenuated for stronger coupling case. It is found by the analysis that even if strong coupling between structure and acoustics occurs, the structure response peak will not increase. However, the further investigation for more general conclusion will be necessary.

3.2. SEA model (High frequency model)

Statistical Energy Analysis (SEA) deals with coupling of cavity and structure in broad frequency band. The fundamental concept of SEA does not treat a specific mode, but modal average in the frequency band of interest. There are N equations, corresponding to N subsystems, with NxN unknown parameters need to be solved simultaneously. The energy balance equation of SEA in case of 2 subsystems can be expressed as Eq. (4),

$$\omega\begin{pmatrix}\eta_{11}+\eta_{12} & -\eta_{21}\\ -\eta_{12} & \eta_{22}+\eta_{21}\end{pmatrix}\begin{pmatrix}E_1\\ E_2\end{pmatrix}=\begin{pmatrix}\pi_1\\ \pi_2\end{pmatrix} \quad (4)$$

Where E_1 is the energy of subsystem i, π_1, π_2 are the power injected to subsystem 1and subsystem 2, damping loss factor η_{11}, η_{22} and coupling loss factor η_{21}, η_{12} are unknown parameters.

The SEA model of fairing with and without spacecraft was modelled as shown in FIG 7. There were 21 Subsystems: 18 structures, 3 cavities. The details of the modelling are shown in ref. [11]. The difference of *spacecraft* vibrational susceptibility to acoustics may be expressed by the frequency response function (FRF) change in dB as defined by Eq. (5),

$$\Delta ALSPL = 20\log_{10}\left(\frac{a_{w/o\ filleffect}}{p_{w/o\ filleffect}}\right) - 20\log_{10}\left(\frac{a_{with\ filleffect}}{p_{with\ filleffect}}\right) \quad (5)$$

where $a_{w/o\ filleffect}$ and $a_{with\ filleffect}$ are the acceleration response in reverberant acoustic chamber without (spacecraft only) and with fill effect (spacecraft with fairing). Also, $p_{with\ filleffect}$ and $p_{w/o\ filleffect}$ are the sound pressure in reverberant acoustic chamber and air gap facing spacecraft panel of interest. The larger $\Delta ALSPL$, the vibration response of spacecraft in reverberant chamber (spacecraft only) is higher than that of the spacecraft with fairing (fill effected).

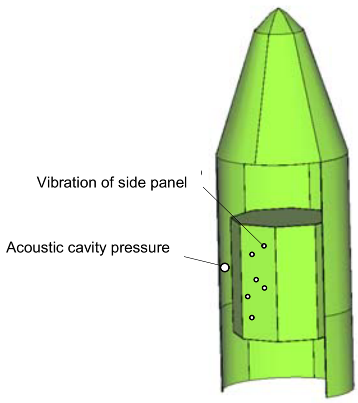

FIG 7 SEA model of fairing w/ and w/o spacecraft

4. ACOUSTIC TEST

Acoustic tests using test item of honeycomb panels listed in TAB. 2 were carried out. The test configuration at acoustic test facility in JAXA Tsukuba Space Center is shown in FIG 8. The test case without Fill Effect is achieved by hanging the panel in the center of acoustic chamber as shown in FIG 8a. The test panel supported by 5cm thick soft mount shown in FIG 8b is the Fill Effect test configuration.

The average pressure power spectrum density (PSD) of 6 control microphones in the chamber and microphone in the narrow gap are shown in FIG 9a. The narrow gap pressure is up to 10dB larger than the control microphone in the frequency range 200 to 300(Hz), which indicates fill effect. Acceleration responses of configurations shown in FIG 8a, 8b are compared in FIG 9b. Acceleration response of FIG 8b configuration which involves fill effect in the narrow gap is somewhat less than that without fill effect. This result is seen to be consistent with the result obtained in FIG 6a, 6b. The acceleration response is the average of 6 accelerometers vertically mounted on the panel. The detailed investigation on the shift of resonant frequency and peak value is illustrated in FIG 10. Acceleration at resonant frequency keeps nearly constant while the resonant frequency shifts down slightly especially in low resonant mode. Meanwhile, 'Change of FRF' and 'SPL increase' is almost symmetric with respect to 0dB line, which indicates that the acceleration induced by unit sound pressure at the narrow gap decreases to the same degree as the SPL increase in the narrow gap. It is clear from the test result that fill effect is caused by the acoustic modes in the narrow gap, and these acoustic modes act as a dynamic absorber to the structure modes, to shift down the resonant modes slightly rather than as an driver to the structure modes.

TAB 2 panels used for verification test

	panel1	panel2	panel3	panel4	panel5
skin material	aluminum	CFRP	aluminum	aluminum	aluminum
core material	aluminum	aluminum	aluminum	aluminum	aluminum
thickness of skin [m]	0.0003	0.0003	0.0003	0.0003	0.0003
thickness of core [m]	0.025	0.025	0.025	0.1	0.025
Young's modulus of skin [MPa]	7.16E+10	1.04E+11	7.16E+10	7.16E+10	7.16E+10
modules of rigidity of core [MPa]	1.38E+08	1.38E+08	1.38E+08	1.38E+08	1.38E+08
Poisson ratio of skin	0.33	0.33	0.33	0.33	0.33
Length [m]	0.91	0.91	0.91	0.91	0.91
width [m]	1.82	1.82	1.82	1.82	1.82
mass of panel [kg]	7.6	4	6.2	10.6	7.6
mass mounted [kg]	0	0	0	0	20

(a) (b)

FIG 8 Acoustic test configuration

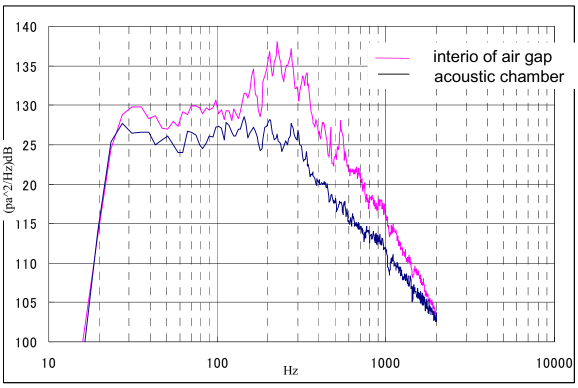

FIG 9a Pressure PSD of acoustic chamber and narrow gap

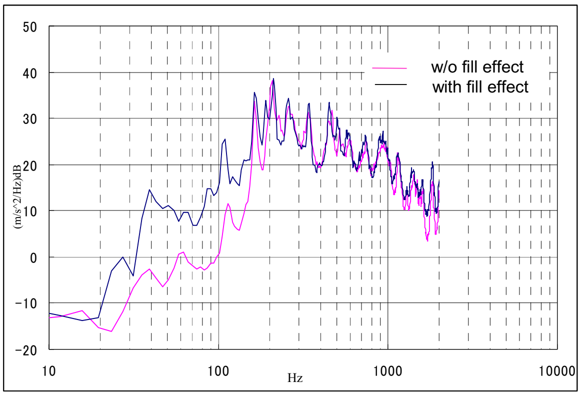

FIG 9b Vibration response w/ and w/o fill effect

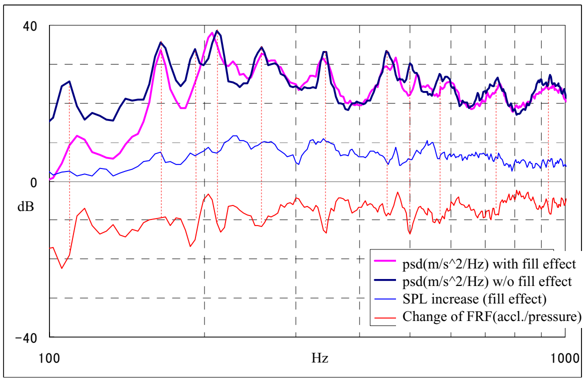

FIG 10 Detail investigation of acceleration, SPL and FRF change

In addition to the low frequency verification described above, high frequency verification as to SEA model (FIG 7) was executed.

Acoustic test of full scale fairing with dummy spacecraft was carried out as shown in FIG 11. In order to obtain $\Delta ALSPL$ of spacecraft, the measurement of spacecraft vibration (6 accelerometers in each panel) and SPL of the fairing interior (3 microshpnes in each cavity) facing toward the spacecraft panel were made. $\Delta ALSPL$ of SEA result and acoustic test measurement are compared in FIG 12. Measured fill effect at the cavity adjacent to the side panel of the dummy spacecraft (see FIG 7) is also shown in FIG 13. Result from FIG 12 and FIG 13 indicates that $\Delta ALSPL$ increases to the same degree as fill effect measured at the narrow gap. Also, the $\Delta ALSPL$ obtained by SEA technique is in good agreement with the measurement. The result shows that the fill effect have the spacecraft vibration less susceptible to acoustics ($\Delta ALSPL > 0$), in other words, that fill effect may not increase the vibration response.

FIG 11 Acoustic test of full scale fairing and dummy spacecraft

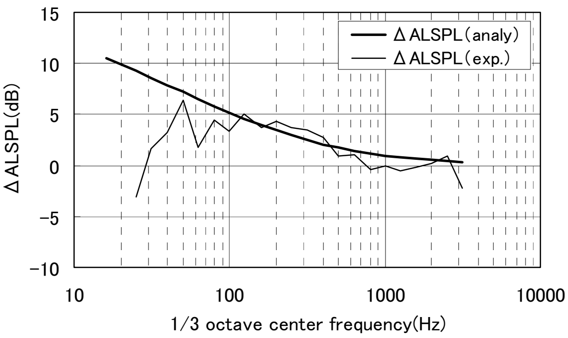

FIG 12 Comparison of $\Delta ALSPL$

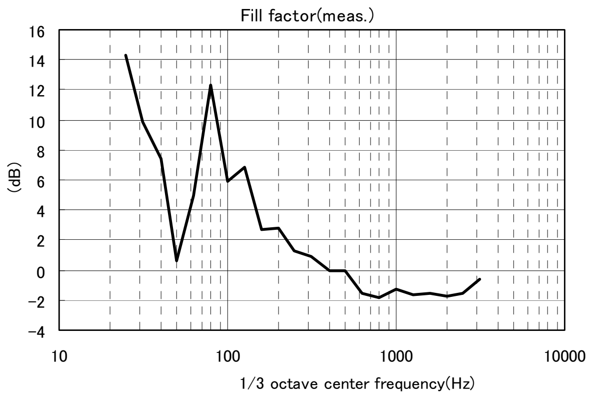

FIG 13 Fill effect measured at air gap between the side panel and fairing wall

5. CONCLUSION AND FUTURE WORK

The conclusion of this paper is the following:

1) Fill effect is one of the important issues in the definition of acoustic environment for the spacecraft design.
2) The main reason of the fill effect is the domination of low cavity resonant mode (1,1) rather than structure mode. The higher the order of cavity acoustic mode, the lower SPL increase occurs. The frequency of local SPL increase mainly depends on the surface dimension, but air gap thickness of panel will slightly cause the frequency shift.
3) The acceleration of the structure backed by a cavity will be effected by the mode coupling of cavity and structure modes. The cavity resonant modes to the structure resonant modes may act as “dynamic absorber”, which slightly shifts the resonant frequency of structure, which will reduce, rather than increase the vibration response.
4) At the frequency of non resonant structure mode, fill effect will increase the vibration response. It is normally not to surpass the definition of vibration environment.

The further investigation on the individual structural and acoustic coupling in lower frequency will be a future work.

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