Journal of Visualization, Vol. 7, No. 1 (2004) 93-101

Acoustic Characteristics inside Cylindrical Structure with End Plates Excited at Different Frequencies

Moriyama, H.*[1] and Tabei, Y.*[2]

*1 Faculty of Engineering, Tokai University, Kitakaname1117, Hiratsuka-shi, Kanagawa, 259-1292, Japan.
*2 Graduate School of Engineering, Tokai University, Kitakaname1117, Hiratsuka-shi, Kanagawa, 259-1292, Japan.

Received 23 June 2003
Revised 3 September 2003

Abstract : In order to study coupling between vibration of a structure and a sound field in contact with the structure, a cavity surrounded by a rigid cylinder having thin elastic plates at both ends is adopted as an analytical model. When excitation forces of different frequencies are applied to the respective plates, the plate vibrations and the sound field inside the cavity become aperiodic, because of the coupling between the systems. In the present investigation, distribution of the sound pressure level inside the cavity is studied in detail in order to clarify the coupling behavior. The results show that when the respective plates vibrate at the same circumferential order, the vibration modes of the plates, which effect the coupling of the plate vibrations and the sound field, cause the aperiodic nature of each system to develop. In this case, since the dominant mode exists in formation of the sound field, it significantly influences the aperiodic nature of the coupling systems. In the case of vibration modes where the plates vibrate at different circumferential orders, the behaviors of the three systems, whose coupling has been restrained, approximate a steady state. Consequently, the dominant mode does not appear in the sound field.

Keywords : Coupled vibration, Circular end plate, Cylindrical sound field, Distribution of sound pressure level, Modal shape

1. Introduction

Various vessels adopt the form of a cylindrical structure having a plate at one end or plates at both ends. In such a structure, periodically excited coupling occurs between the cylinder and the plate vibration, and also occurs between the structural vibration and any sound field that has been formed in the cylinder. Prediction of these coupling behaviors is important from the viewpoint of noise control, and under some circumstances may have to be controlled. Recent investigations with respect to coupling between structural vibration and an internal sound field have yielded interesting results, such as a suggested method of omitting the vibration modes of both the higher order and the lower orders in order to increase calculation efficiency (e.g., Iwatsubo et al., 1995), and estimation by application of the contribution ratio for each mode of the noncoupling systems to the natural mode of the coupling system (e.g., Minami et al., 2000). With regard to the cylindrical structure studied in the present investigation, the rough coupling behavior between the cylinder and the end plates was confirmed (e.g., Cheng et al., 1992), along with that between their enclosures and an internal sound field (e.g., Cheng et al., 1992). However, the above reports describe the results as steady-state coupling, because the vibrations are due to single-frequency excitation. If a certain structure were to be excited at different frequencies, structural vibration would be aperiodic. If the end plates of the cylindrical structure were to be excited with different frequencies, the plate vibrations, which influence each other via the sound field, would be predicted to be aperiodic, even if the plates are structurally separated. In this case, although the sound field should also be aperiodic, its nature would depend on the coupling intensity between the three systems. Furthermore, thus far, little investigation has been conducted on the coupling problem of an aperiodic nature. The present investigation adopts an analytical model in which the end plates of the above cylindrical structure are set to the same support condition. Point forces of different frequencies are applied to the respective plates so that they behave with different modes. Coupled vibration analysis is carried out between the plate vibrations and the sound field inside a cavity, and then the coupling phenomena are estimated by reference to distribution of sound pressure level. In particular, the behaviors of the aperiodic coupling systems are understood by means of presumption of acoustic modes contributing to formation of the sound field.

2. Analytical Method

2.1 Analytical Model

The analytical model considered is a cavity having two circular end plates, as shown in Fig. 1. The plates are supported by translational and rotational springs at distributed uniform intervals, and the supporting conditions are determined by spring stiffnesses K_1, K_2 (N/m²) and C_1, C_2 (Nm/m). The plates are assumed to be made of aluminum, having a Young's modulus E of 71 GPa and a Poisson's ratio ν of 0.33, and to have a radius a of 150 mm and a thickness h of 3 mm. The sound field is regarded a cylindrical sound field having the same radius as the plates and a length of 500 mm, and, other than the plate surfaces, the boundary conditions are structurally and acoustically rigid. Coordinates are radius r, angle θ within the planes of the plates and the cross-sectional plane of the cavity, and length z corresponding to cylindrical length. Point forces F_1 and F_2 are applied to the plates at $r_1 = r_2 = 60$ mm and $\theta_1 = \theta_2 = 0$ deg. , respectively.

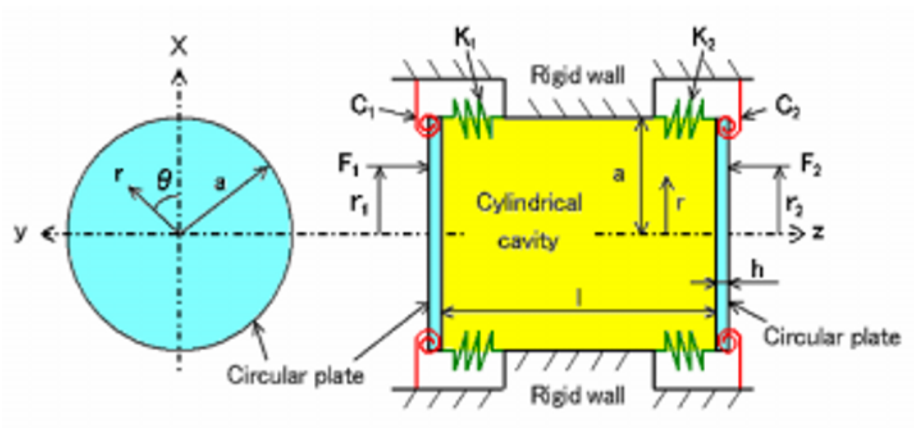


Fig. 1. Analytical model.

2.2 Coupling equation between plates and cavity

Flexural displacement w_1 and w_2 on the plates at the F_1 and F_2 sides are expressed by substituting Eq.(2) of the plate mode shape into Eqs.(1), and are expanded over two sets of suitable trial functions (e.g., Cheng et al., 1992) as

$$w_1 = \sum_{s=0}^{1}\sum_{n=0}^{\infty}\sum_{m=0}^{\infty} B_{1nm}^{s}\psi_{nm}^{s}\, e^{j(\omega_1 t+\Phi_1)},\quad w_2 = \sum_{s=0}^{1}\sum_{n=0}^{\infty}\sum_{m=0}^{\infty} B_{2nm}^{s}\psi_{nm}^{s}\, e^{j(\omega_2 t+\Phi_2)} \tag{1}$$

$$\psi_{nm}^{s} = \sin(n\theta + s\pi/2)(r/a)^{m} \tag{2}$$

where n, m, and s are a circumferential order, a radial order, and a symmetry index, respectively. B^{S}_{1nm} and B^{S}_{2nm} are coefficients to be determined, and suffixes *1* and *2* refer to the F_1 and F_2 sides, respectively. ω_1 and ω_2 are angular frequencies of the point forces on the respective side plates, and t is elapsed time. The equations of plate motion are obtained by finding an extremum of Hamilton's function in terms of Eqs.(1). The acoustic excitations, each being considered coupling between a plate and the cavity, the coupling equations are expressed as,

$$\left[\sum_{m'=0}^{\infty}\left\{R_{1nmm'}^{s}\left(1+j\eta_p\right)-\omega_1^2 M_{1nmm'}^{s}\right\}+\sum_{m'=0}^{\infty} aF_{sn}\left\{K_1+\left(\frac{m}{a}\right)\left(\frac{m'}{a}\right)C_1\right\}\right]B_{1nm'}^{s}e^{j(\omega_1 t+\Phi_1)}$$

$$= F_{1nm}^{s}(t)+\frac{\rho_c c^2 A^2}{V_c}\times\sum_{m'=0}^{\infty}\sum_{p=1}^{\infty}\sum_{q=0}^{\infty}\frac{L_1}{M^{s}_{npq}}\left\{\frac{\omega_1^{2}L_1 B^{s}_{1nm'}e^{j(\omega_1 t+\Phi_1)}}{\omega^2_{npq}+j\eta_c\omega_{npq}\omega_1-\omega_1^{2}}-\frac{\omega_2^{2}L_2 B_{2nm'}e^{j(\omega_2 t+\Phi_2)}}{\omega^2_{npq}+j\eta_c\omega_{npq}\omega_2-\omega_2^{2}}\right\} \tag{3}$$

$$\left[\sum_{m'=0}^{\infty}\left\{R_{2nmm'}^{s}\left(1+j\eta_p\right)-\omega_2^2 M_{2nmm'}^{s}\right\}+\sum_{m'=0}^{\infty} aF_{sn}\left\{K_2+\left(\frac{m}{a}\right)\left(\frac{m'}{a}\right)C_2\right\}\right]B^{s}_{2nm'}e^{j(\omega_2 t+\Phi_2)}$$

$$= -F_{2nm}^{s}-\frac{\rho_c c^2 A^2}{V_c}\times\sum_{m'=0}^{\infty}\sum_{p=1}^{\infty}\sum_{q=0}^{\infty}\frac{L_2}{M^{s}_{npq}}\left\{\frac{\omega_1^{2}L_1 B^{s}_{1nm'}e^{j(\omega_1 t+\Phi_1)}}{\omega^2_{npq}+j\eta_c\omega_{npq}\omega_1-\omega_1^{2}}-\frac{\omega_2^{2}L_2 B^{s}_{2nm'}e^{j(\omega_2 t+\Phi_2)}}{\omega^2_{npq}+j\eta_c\omega_{npq}\omega_2-\omega_2^{2}}\right\} \tag{4}$$

where $R^{S}_{1nmm'}$, $R^{S}_{2nmm'}$ and M^{S}_{1nmm} , M^{S}_{2nmm} are the stiffness and mass matrices of the respective side plates. The index m', being the radial order ($m = m'$), the matrices bring symmetrical. η_p is a structural damping factor, and F_{ns} is a coefficient to be determined by the indexes n and s. On the right sides of Eqs.(3) and (4), the first and second terms express the point force and the coupling between the plate and the cavity, respectively. P_c and c are, respectively, fluid density and the speed of sound in the cavity, A is the total surface area of the plates, and V_c is the volume occupied by the cavity. L_1 and L_2 are, respectively, spatial coupling coefficients between the respective side plates and the cavity, and η_v is the acoustical damping factor in the cavity. In the acoustic mode (n, p, q), represented by the indexes p and q to the radial and longitudinal orders, ω_{npq} and M^{S}_{npq} are, respectively, an angular natural frequency and a modal generalized mass of the nth cavity mode. Further, from B^{S}_{1nm} and B^{S}_{2nm}, having been obtained from Eqs.(3) and (4), the behaviors of the plate vibrations and the sound field under the coupling phenomena can be determined.

3. Results and Discussion

3.1 Fundamental Characteristics of Coupling Between the Plates and Cavity

The supporting conditions of the plates having a flexural rigidity D [$=Eh^3/\{12(1-\nu^2)\}$] are expressed by nondimensional stiffness parameters; for instance at the F_1 side, K_{b1} ($=K_1a^3/D$) and C_{b1} ($=C_1a/D$). In this study, the stiffness, which is set to be slightly higher for a translational support than for a slide support (for instance K_{b1}=0 and C_{b1}=10^8), are $K_{b1} = K_{b2} = 10^2$ and $C_{b1} = C_{b2} = 10^8$. Fig. 2 shows spectra of the power level expressions V_1, V_2, and L_{pv}, respectively, for both of average quadratic velocities of the each side plate and an average sound pressure inside the cavity with respect to the excitation frequency. An excitation force ratio F_R (= F_2/F_1) exciting only the F_1 side plate is 0, and vibration modes and modal shape of the plate are expressed by (n, m)

and brief illustrations at the inside and the outside of this figure. When peak levels in the region below the middle frequency are compared, V_1, V_2 and L_{pv} are nearly identical, whereas V_2 gradually decreases when frequency shifts beyond middle frequency to higher frequency. The plate vibrations and the sound field remain close in energy, indicating that the three systems are strongly coupled. In the region beyond the middle frequency, the coupling with respect to the systems depends on the vibration of the F_1 side plate and the sound field, and is hardly influenced by the vibration of the F_2 side plate. In consideration of each vibration mode of the F_1 side plate, coupling with the (1,0) mode at excitation frequency f_1 = 485 Hz is influenced by the nature of the three systems, whereas coupling with the (1,1) and (2,1) modes, respectively, at f_2 = 1087 Hz and f_2 = 1586 Hz, is influenced by the nature of the two systems other than vibration of the F_2 side plate.

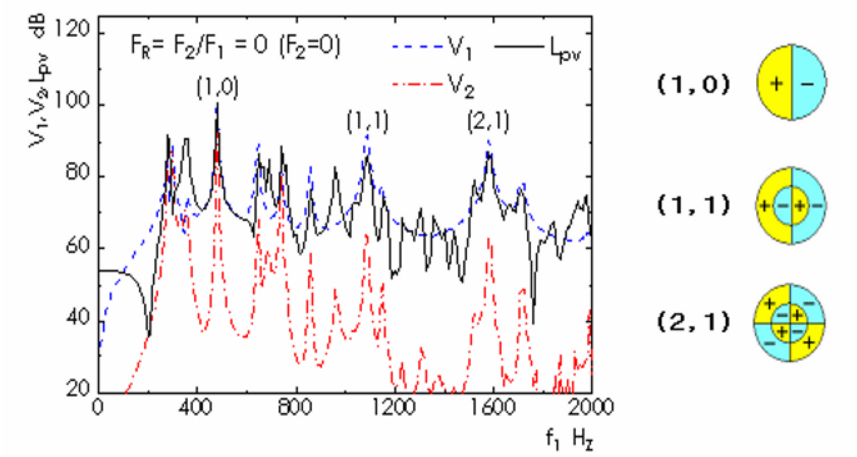
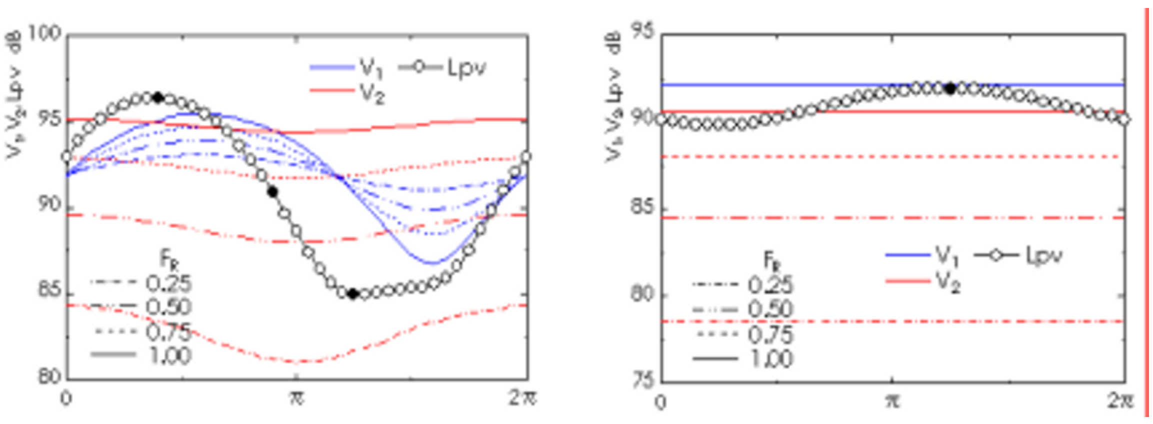


Fig. 2. Spectra of quadratic velocity averaged on plate and sound pressure level averaged inside cavity.

Fig. 3 (a) and (b) show time histories of V_1, V_2, and L_{pv}, whose excitation frequency f_2 is set at 1087 Hz and 1586 Hz at the F_2 side plate, respectively, with the excitation frequency f_1 of the F_1 side plate being fixed to 485 Hz. The f_1 and f_2 are determined by whether the vibration modes of the plates coincide in the circumferential order n or not, because the coupling phenomena are greatly influenced by the similarity between the vibration and acoustic modes (e.g., Moriyama, 2003). The plates and the cavity are excited by the different frequencies, and their behaviors, which are aperiodic, depend on frequency $f_2 - f_1$. The lateral axis ranges between 0 and 2π with $(\omega_2 - \omega_1)t$, and the excitation force ratio F_R is varied among 0.25, 0.50, 0.75, 1.0 by the shifting F_2. In the case of f_2 = 1087 Hz, V_2, increases with F_R, and ΔV_2 the difference between the maximum and minimum levels of V_2 shows the tendency to decrease. As compared with V_2, V_1 varies less with F_R, but ΔV_1 increases with F_R. The decrease in the level difference indicates approaching steady-state vibration, weakening the aperiodic nature, and therefore in this case F_R is closely related to aperiodic vibration. With regard to F_R =1.0, the aperiodic nature of L_{pv} is more distinguished than that of V_1 or V_2. Although the relative relations between V_1, V_2, and L_{pv} with f_2 = 1586 Hz are similar to the above relations, the changes with time are less restrained, and these characteristics approximate steady-state behavior. The above results point out that the coincidence about the circumferential orders of the plates contributes to the aperiodic nature. If the f_1 and f_2 were close with some conditions having such a circumferential order, the time histories would have the aperiodic nature including a beat phenomenon. In the first place, the acoustic characteristics of the cavity are studied with the maximum levels of the L_{pv} concerning both excitation frequencies, as follows. With regard to f_2 = 1087 Hz, the characteristics with the middle and minimum levels of L_{pv} are compared with the above results, and these data are represented by solid circles in Fig.3 (a).



(a) f_1=485Hz, f_2=1087Hz (b) f_1=485Hz, f_2=1586Hz

Fig. 3. Time histories of quadratic velocity averaged on plate and sound pressure level averaged inside cavity.

3.2 Acoustic Characteristics of Cavity with Coincident Circumferential Order on Plates

In the section, the exciting frequencies f_1 and f_2 are, respectively, set at 485 Hz and 1087 Hz at the F_1 and F_2 side plates in order to make the plates of the same circumferential order. Fig. 4 shows distributions of sound pressure level L_{pa} averaged on a lateral cross-sectional plane (x - y plane) of the cavity with respect to the z direction when the excitation force ratio F_R is varied within the range of 0 ~ 1.0. As the sound field with F_R = 0 is greatly influenced by the vibration of the F_1 side plate, its distribution is such that L_{pa} is relatively large in the range near the F_1 side plate. However, because of influence by the vibration of the F_2 side plate, L_{pa} gradually increases with F_R in the range beyond z/l = 0.5. In order to indicate the distribution more clearly, Fig. 5 (a) and (b) show three-dimensional distribution of sound pressure level L_p at each point in the cavity with F_R = 0 and F_R = 1.0. In these figures, the colors red, yellow, green, and dark blue represent 72.5, 86, 90.5, and 96 dB, respectively. When F_R = 0, decreasing with increasing distance from the F_1 side plate, L_p distribution on the x-y plane is observed with the fixed acoustic mode. When F_R = 1.0, showing decided changes in the range beyond z/l = 0.5, the distribution is maintained at n = 1 within the whole range. However, the acoustic mode in the vicinity of the F_2 side plate is caused by the (1,1) vibration mode of that plate.

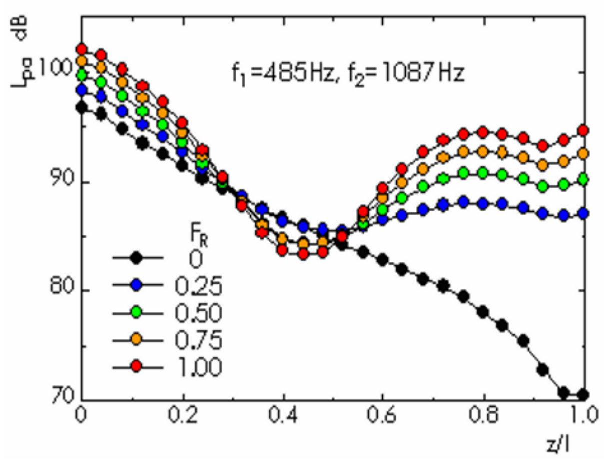
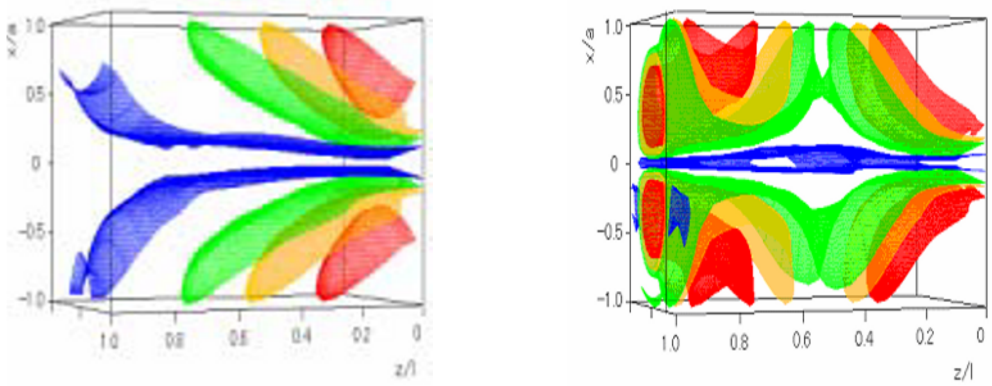


Fig. 4. Distributions of sound pressure level averaged on x-y plane.



(a) F_R=0 (b) F_R=1.0

Fig. 5. Distributions of sound pressure level inside cavity (f_1=485 Hz, f_2=1087 Hz).

The acoustic characteristics inside the cavity are related to not only the plate vibration mode, but also the internal acoustic mode. As indicated in Table 1, in which the acoustic modes are presented by comparison with resonance frequencies, the cavity exhibits the (0,0,1) mode in proximity to f_1 = 485 Hz, the (1,1,2), (1,1,3), (2,1,0), and (2,1,1) modes in proximity to f_2 = 1087 Hz, and the (0,0,2) and (1,1,0) modes in proximity to $f_2 - f_1$ = 602 Hz. In order to study the effect of such acoustic modes approaching the natural frequencies of the plates, Fig. 6 shows the distributions of the L_{pa} obtained by analysis, except for the (1,1,0), (1,1,2) and (1,1,3) modes with F_R = 1.0. Compared with the circle symbol presented in Fig. 4, the results clearly show what the acoustic characteristics greatly depend on the (1,1,0) and (1,1,2) modes with the n of the vibration modes. The acoustic modes outside those presented in Fig. 6 have been confirmed to have little effect on the sound field.

Table 1. Comparison of acoustic mode and natural frequency.

n,p,q	f_{npq} Hz	n,p,q	f_{npq} Hz	n,p,q	f_{npq} Hz
0,0,1	344	1,1,0	671	2,1,0	1114
0,0,2	687	1,1,1	754	2,1,1	1166
0,0,3	1031	1,1,2	961	2,1,2	1309
0,0,4	1375	1,1,3	1230	2,1,3	1519
0,0,5	1719	1,1,4	1530	2,1,4	1769
0,0,6	2062	1,1,5	1845	3,1,0	1532
0,1,0	1397	1,1,6	2169	3,1,1	1570
0,1,1	1439	1,2,0	1944	3,1,2	1679
0,1,2	1557	1,2,1	1974	3,1,3	1847
0,1,3	1737	1,2,2	2062	4,1,0	1939
0,1,4	1960	1,2,3	2201	4,1,1	1969

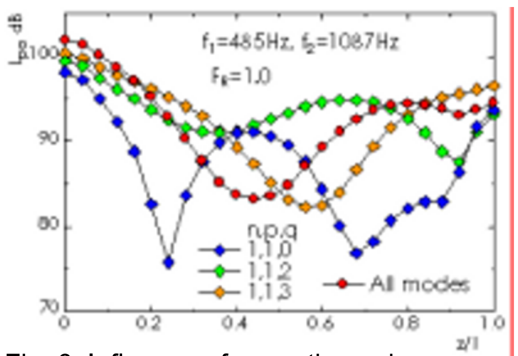


Fig. 6. Influence of acoustic mode on sound field.

The above results correspond that the coupling phenomena of the two systems having only one side plate (e.g., Cheng et al, 1992) and of the three systems (e.g., Moriyama, 2003) were intensified by the coincidence about the n of each system, and clarify that the behaviors of the three systems are aperiodic if the acoustic modes in proximity to $f_2 - f_1$ satisfy the above condition.

3.3 *Acoustic Characteristics of the Cavity where Vibrations of the Plates are of Different Circumferential Orders*

In this section, the excitation frequency f_2 is 1586 Hz so that the circumferential order n may be fixed to 1 and 2 on the F_1 and F_2 side plates, respectively. Fig. 7 shows the distributions of the averaged sound pressure level L_{pa} with respect to the z direction when the excitation force ratio F_R varied within the range of 0~1.0. Although the distribution is greatly influenced by the application of F_2 as f_2 = 1087 Hz, the determined changes with F_R are limited to the range beyond

z/l = 0.8. If the acoustic characteristics are represented by the three-dimensional expression shown Fig. 5, for instance, the distribution of L_p with F_R = 1.0 becomes that shown in Fig. 8. Exhibiting the acoustic modes corresponding with the vibration mode in the range in proximity to each plate, the distribution is distorted more greatly in the middle range than in the case where f_2 = 1087 Hz. The results confirm that the distortion change remarkably with increasing F_R. Fig.9 shows the results for the relation between occurrence of the sound field and the acoustic modes in same manner as in Fig. 6. In this case, the (1,1,2), (1,1,3), and (2,1,3) modes, selected from among the resonance modes in proximity to the excitation frequency, occupy a less important position in the sound field than in the case of f_2 = 1087 Hz.

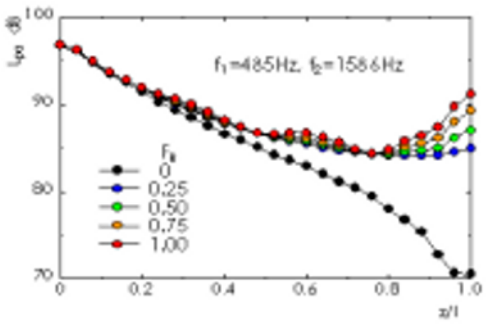


Fig. 7. Distributions of sound pressure level averaged on x-y plane.

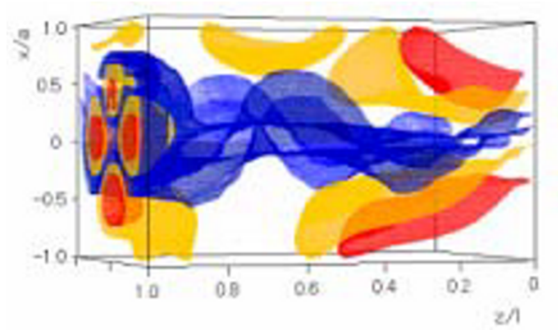


Fig. 8. Distributions of sound pressure level inside cavity (f_1=485 Hz, f_2=1586 Hz, F_R=1.0).

In order to estimate changes in the distribution of the sound pressure level L_p on the x-y plane in detail, we observe changes in acoustic modal shape S, which is similar to the spatial coupling coefficient, and is defined as follows (e.g., Moriyama, 2003);

$$S = \frac{1}{A_1}\int_{A_1}\left(\frac{w_1'}{w_{1\,\max}'}\right)\cdot\left(\frac{P_c'}{P_{c\,\max}'}\right)\,dA_1 \tag{5}$$

where w_1' and $w_1'_{max}$ are, respectively, the flexural displacement and its maximum value at each point on the F_1 side plate, and P_c' and $P_c'_{max}$ are, respectively, the sound pressure and its maximum value at each point on the x - y plane at z/l, making these parameters the products of complementary elements.

Fig. 10 shows the distributions of S with respect to z/l under the same condition as in Fig. 7. S decreases with increasing F_R, and the decrease is remarkable in the vicinities of z/l = 0, 0.32, 0.64, 1.0. Representing a similar degree between a modal shape of the F_1 side plate and a distribution shape of the sound pressure level on x - y plane, the value of S which shifts uniformly with F_R = 0 corresponds to the three-dimensional distribution occupied by n =1 in Fig. 5 (a). The difference between the S with F_R = 0 and that with varying F_R is caused by the distorted distribution of the sound pressure level on the x-y plane, and periodically occurs with respect to z/l. If the z direction order q =3 is indispensable to the above S distribution, the (0,0,3), (1,1,3), and (2,1,3) modes, which exist within close ranges of both excitation frequencies, are important for that sound field. The distribution of L_{pa} is somewhat influenced by the (1,1,3) and (2,1,3) modes, because these modes are coincident with the vibration modes of the respective plates about n . However, the distribution shape of the sound pressure level is distorted by mixing the different acoustic modes, and therefore these acoustic modes cannot become dominant over the sound field when f_2 = 1087 Hz. Although the mixing of different acoustic modes was predicted to occer in the steady-state (e.g., Moriyama, 2003), they could be confirmed by the distributions of sound pressure level estimated in the three-dimensional expression and the S distribution.

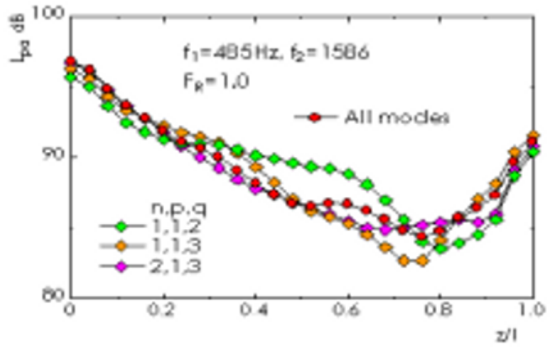


Fig. 9. Influence of acoustic mode on sound field.

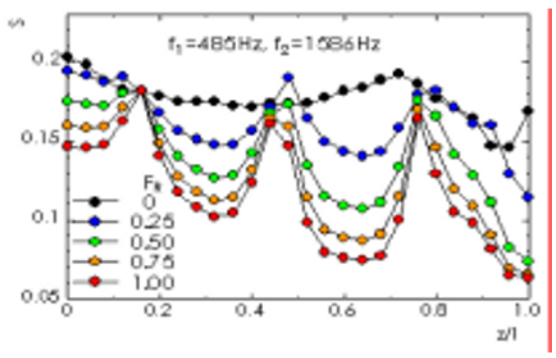
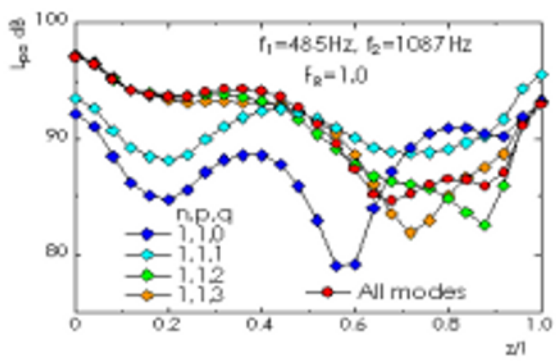


Fig. 10. Changes of acoustic modal shape on x-y plane.

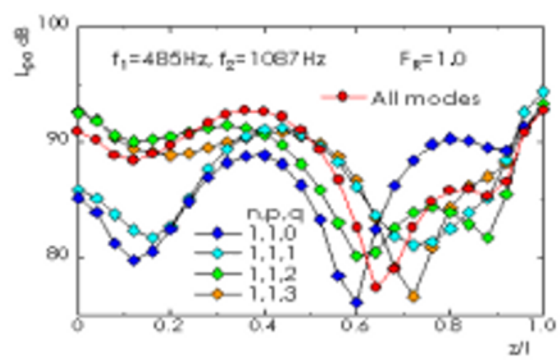
3.4 Shift of Dominant Acoustic Mode with Changes of Sound Pressure Level

Discussion of the coupling phenomena mentioned above is limited to the case where the average sound pressure level L_{pv} inside the cavity is at maximum level, and in this section the phenomena with changes in L_{pv} are studied. Specifically, a shift of the dominant mode over the sound field with f_2 = 1087 Hz is studied by means of considering the middle and minimum levels in Fig. 3 (a).

Fig. 11 (a) and (b) show the distributions toward the z axis of the sound pressure level L_{pa} averaged on the x-y plane with respect to the middle and minimum levels of L_{pv}, respectively. In order to compare the shift of the dominant acoustic mode, these figures show the distribution with all the modes and the distributions except the respective (1,1,0),(1,1,1),(1,1,2), and (1,1,3) modes. Both results indicate what the (1,1,0) mode is most important, and next in importance, the (1,1,1) mode contributes to formation of the sound field. As compared with the result concerning the maximum level of L_{pv}, the appearance of the (1,1,1) mode is a point of difference. Furthermore, the sound field, in which L_{pv} decreases, has a tendency to suppress the difference between the distribution with all the modes and the distributions not including the specific modes. In the case of the same phase with respect to the excitation forces at both plates, because the vibration strongly influences sound field when the longitudinal order q is an even number, the coupling between the three systems with the maximum level of the L_{pv} is intensified by the (1,1,2) mode. Consequently, contributing to the appearance of the acoustic mode with an odd order of the q, the decrease in L_{pv} makes production of the dominant mode difficult. When the sound field is aperiodic because of coupling with the vibrations of the plates excited by different frequencies, the sound pressure level clearly correlates with the intensity of coupling.



(a) L_{pv}=90.9dB



(b) L_{pv}=85.0dB

Fig. 11. Influence of acoustic mode on sound field.

4. Conclusion

With regard to the sound field inside the cavity with the circular plates excited at different frequencies at respective ends of the cylinder in the present study, the coupling between the plate vibrations and the sound field was studied from the viewpoint of the relation between the vibration modes and the dominant acoustic mode, then from the viewpoint of the distribution shape of the sound pressure level. The results show that when both vibration modes coincide in the circumferential order n, the aperiodic nature of the coupling phenomena develops, so that the average sound pressure level L_{pv} inside the cavity varies widely over the time history. In the formation of such a sound field, effected by several acoustic modes in the regions of the respective excitation frequencies and the difference in frequency between the excitation frequencies, the acoustic modes coincide with the vibration modes concerning the n. Furthermore, the appearance of the dominant acoustic mode clearly depends on the level of L_{pv}.

In contrast, when the vibration modes don't coincide in terms of n, changes in L_{pv} are considerably restrained in the time history, approximating the steady state. In this case, the respective vibrations easily couple with the corresponding acoustic mode, thereby weakening the coupling of the three systems included the vibrations and the sound field. Consequently, the dominant acoustic mode is less likely to appear under this condition than in the vibration modes with the same n.

If some systems, which are concerned in coupling phenomena, have the similar influence on one another, the above conclusions can be applied to the systems, even if those have other configurations.

References

Iwatsubo, T., Kawamura, S., Kamada, M. and Shiohata, K, Development of Structural-Acoustic Coupled Analytical Method, Trans. of JSME, (C), 61-585 (1995), 163.

Minami, H. and Di Mondo, Y., A Study on Vibro-Acoustic Coupling Between Body Structure and Interior Acoustic System in Railway Vehicles, J. Acoust. Soc. Jpn, 56-4 (2000), 225.

Cheng, L. and Nicolas, J., Free Vibration Analysis of a Cylindrical Shell-Circular Plate System with General Coupling and Various Boundary Conditions, J. Sound Vib., 155 (1992), 231.

Cheng, L. and Nicolas, J., Radiation of Sound Into a Cylindrical Enclosure From a Point-Driven End Plate with General Conditions, J. Acoust. Soc. Am., 91 (1992), 1504.

Moriyama, H., Acoustic Characteristics of Sound Field in Cylindrical Enclosure with Exciting End Plates, Trans. of JSME, (C), 69-679 (2003), 603.

Author Profiles

Hiroyuki Moriyama : He is an associate professor in Department of Prime Mover Engineering at Tokai University. He received his M.E. and Ph.D. in Mechanical Engineering from Tokai University in 1984 and 1993, respectively. His research has mainly focused on vibration and the acoustic radiation characteristics with respect to thin structures, particularly on the coupling phenomena between structural vibration and the sound field surrounding that structure. He has recently published his research results in ASJ and JSME, and is a member of VSJ, JSME, AJS, and JSAE.

Yasuhito Tabei : He was born in Gunma, Japan on June 6, 1978. He received a B.E degree in Mechanical Engineering from Tokai University, and then studied valve gear trains with high-speed engines. He is a member of the Japan Society of Mechanical Engineering, is studying Structure vibration and Acoustics.