ANALYSIS ON THE VIBRO-ACOUSTICAL CHARACTERISTICS OF A PANEL-CAVITY COUPLED SYSTEM

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Abstract

Theoretical analysis is carried out to identify the modal coupling effect between some particular acoustic modes of a vehicle compartment cavity and vibration modes of body panels like side doors, roof or floor. A simplified panel-cavity coupled model is investigated on the coupled resonance frequencies, modes and frequency response characteristics. Through parametric study, it is possible to explain how the acoustic response of a coupled system will be determined by the vibration and acoustic property of the individual panel and cavity system. Full coupled system shows some interesting features different from those of the semi-coupled system in frequency, mode and acoustic response.

Key Words : Panel-cavity coupled system, Coupled resonance frequency, Frequency response characteristics, Input mechanical impedance

Introduction

In the low frequency range below 250Hz, severe noise problem like booming can occur when a compartment cavity and car body panels are strongly coupled and they are well excited. It is very difficult to improve the acoustic characteristics once the problem occurs, because the modification of the car body structure or the vehicle compartment is not a simple problem. Therefore, detail analysis on the body-compartment system is required in the design stage and sufficient test should be performed. Wolf(1982), Nefske and Sung(1984), Yashiro(1985) have investigated this kind of low frequency noise problem by using MSC/NASTRAN. Kim and Lee(1998) have introduced an analytical model and have developed a special purpose program package(ACSTAP), which was efficient for the structural-acoustic semi-coupled problem. In those studies, car body vibration and compartment cavity acoustics were individually treated, i.e. in-vacuo structural modes of the car body and acoustic modes of the compartment having rigid boundary were independently solved, and then the results were synthesized in the acoustic response analysis stage. This approach seems to be practical, considering the situation that full coupling analysis on a real car is too complicated and too exhaustive. However, authors(1995) have proposed that the full coupling could sometimes generate a quite different acoustic response from that of the cavity having rigid boundary. Actually, measured resonance frequency of the compartment cavity sometimes shows strange characteristics

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compared with the calculated result, which is not likely to result from the only calculation error. In this study, full coupling effect is investigated on the panels-cavity system to understand the problem. A simplified one dimensional model is considered for the coupling analysis between panel vibration and cavity acoustics. This analysis model can explain the modal coupling effect between some particular compartment acoustic modes and panel vibration modes. Theoretical analysis focuses on the identification of the relation between the coupled acoustic response and the individual modal parameters of the panels and the cavity. The result will be applied to estimate the limitation and the accuracy of semi-coupled analysis performed in the structural-acoustic coupled system such as a vehicle compartment.

Theoretical Considerations

In a vehicle compartment system, low frequency acoustic response is dominated by the acoustic property of the compartment cavity and the vibration property of the surrounding panels, such as dash panel, roof, floor, side doors and windows. Compartment and panels are coupled each other, therefore, the modal parameters of the coupled system have different characteristics from the individual panel vibration and cavity acoustics. Coupled mechanism in a general system is too complicated to theoretically analyse. In this study, a simplified model, is analytically investigated to understand the acoustic charateristics in the acoustic-structural fully coupled system. Fig.1 shows the coupling between a compartment cavity and door panels or roof/floor panels. Fig.2 shows typical acoustic modes of a vehicle compartment cavity which can be strongly coupled with the panel vibration. In the figure, dark area represents the pressure nodal plane which has zero pressure. Fig.2 (a) shows a lateral acoustic mode which has a vertical nodal plane and lateral particle motion, therefore, well coupled with the vibration of side door panels. Fig.2 (b) shows an up/down acoustic mode. It is well coupled with the roof and floor vibration mode, because it has a horizontal nodal plane and has up/down particle motion. For those modal coupling, one dimensional panel-cavity coupled system as shown in Fig.3, can be used to analytically investigate the acoustic characteristics. In spite of the simple model, the analysis result is available to understand the modal coupling phenomena in a real vehicle compartment system.

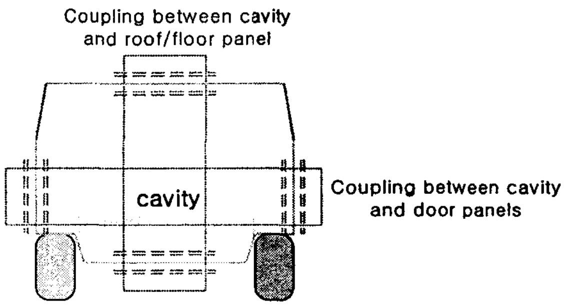
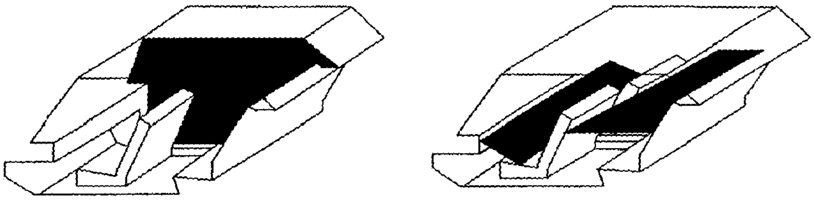


Fig.1 Structural-acoustic coupling in a vehicle compartment



(a) lateral cavity mode (b) up-down cavity mode

Fig.2 Acoustic mode of a vehicle compartment

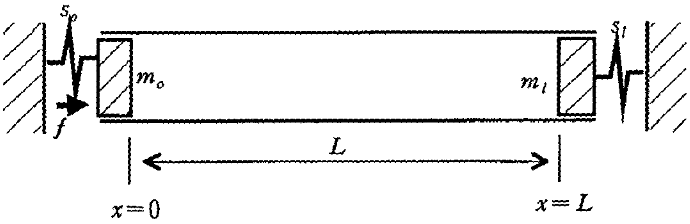


Fig.3 Analysis model of panel-cavity coupled system

Pressure in an acoustic cavity is generally represented as the sum of the waves travelling in positive and negative directions.

$$p(x,t) = Ae^{j(\omega t+kx)} + Be^{j(\omega t-kx)} \tag{425}$$

Where, ω and k mean angular frequency and wave number. Considering the boundary condition on the interior surface of the left panel,

$$(A+B)e^{j\omega t} = p(0,t) \tag{2}$$

Right panel motion is coupled with the particle motion in the cavity at $x=L$, as follows.

$$m_l \frac{d^2\xi}{dt^2} + s_l \xi = Sp(l,t) \tag{3}$$

Where, ξ means panel displacement at $x=L$.

Particle velocity at the right end is same as the panel velocity, i.e. $\dot{\xi} = u(l,t)$and the panel mechanical impedance is represented as follows.

$$\frac{Sp(l,t)}{u(l,t)} = j(m_l\omega - \frac{s_l}{\omega}) \tag{4}$$

A and B can be determined, using equation (2), equation (4) and Euler's equation $u(l,t) = -\frac{1}{\rho}\int_t (\frac{\partial p}{\partial x})_{x=l}\,dt$. Substituting the result into equation (1), interior pressure can be represented as follows.

$$p(x,t)=p(0,t)\frac{\cos k(l-x)+A_l\sin k(l-x)}{\cos kl+A_l\sin kl} \tag{5}$$

Where,

$$A_l=\frac{\rho cS}{\omega m_l-s_l/\omega} \tag{6}$$

S is the section area of the cavity and ρc is specific acoustic impedance of the plane wave in the cavity. $p(0,t)$ in equation (5) can be related to the excitation force by considering the left panel motion as follows.

$$m_o\frac{d^2\zeta}{dt^2}+s_o\zeta+Sp(0,t)=f \tag{7}$$

Particle velocity at the left end is also related to the pressure by Euler's equation as follows.

$$u(0,t)=\frac{-1}{\rho}\int_t\left(\frac{\partial p}{\partial x}\right)_{x=0}dt=j\frac{p(0,t)}{\rho c}\frac{\sin kl-A_l\cos kl}{\cos kl+A_l\sin kl} \tag{8}$$

Considering the relation $\dot{\zeta}(t)=u(0,t)$ and equation (5) to equation (8), interior pressure is finally obtained as follows.

$$p(x,t)=-\frac{f}{S}A_0[\cos k(l-x)+A_l\sin k(l-x)]/[(\sin kl-A_l\cos kl)-A_0(\cos kl+A_l\sin kl)] \tag{9}$$

Using Euler's equation, particle velocity is expressed from equation (9) as follows.

$$u(x,t)=\frac{f}{j\omega\rho S}kA_0[\sin k(l-x)-A_l\cos k(l-x)]/[(\sin kl-A_l\cos kl)-A_0(\cos kl+A_l\sin kl)] \tag{10}$$

Where,

$$A_0=\frac{\rho cS}{\omega m_0-s_0/\omega} \tag{11}$$

Also, total input mechanical impedance of the coupled system is described as follows.

$$Z_{mt}=\frac{f}{u(0,t)}=j[m_o\omega-\frac{s_o}{\omega}-\rho cS\frac{\cos kl+A_l\sin kl}{\sin kl-A_l\cos kl}] \tag{12}$$

Natural frequencies of the coupled system can be determined by making the imaginary part of equation (12) zero(Kinsler,1982), i.e. as the solution of the following characteristic equation.

$$\tan kl=\frac{A_0+A_l}{1-A_0A_l} \tag{13}$$

In equation (12), the input mechanical impedance of the coupled system has only reactance term, since the damping element is not considered here. Damping makes the input impedance complex and can generate complex frequencies and modes. However, small damping does not have an effect on the coupled resonance frequency and mode, except smoothening the sharpness of the resonance peaks, while the complex mode analysis is rather complicated.

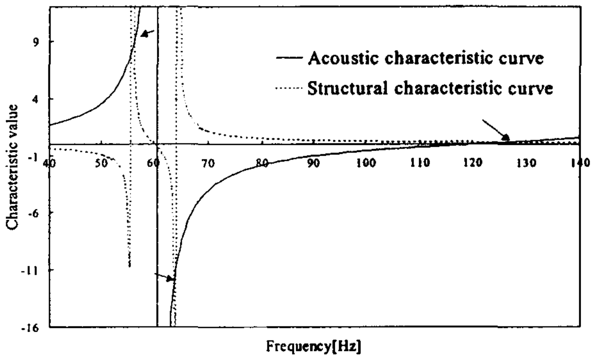
Finally, pressure response of the coupled system can be obtained by considering the difference of the pressure level between the excitation side and cavity interior.

$$PL_x = 20\log[\frac{p(x)}{F/S}] = 20\log[\frac{A_0\cos k(l-x) + A_l\sin k(l-x)]}{\sin kl - A_l\cos kl - A_0(\cos kl + A_l\sin kl)}] \quad (14)$$

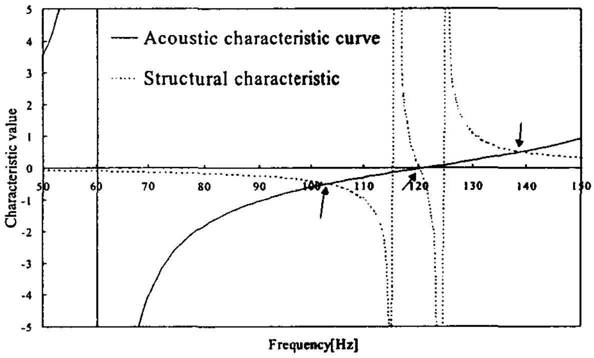
Vibro-Acoustical Characteristics of the Coupled system

Coupled Resonance Frequency

Coupling effect is investigated on the lateral acoustic mode(Fig.2-(a)) and door panel vibration as shown in Fig.1. Door panels are considered to vibrate in a ring mode having a natural frequency. Considering the width of the compartment of a small size car, cavity length is given 1.4m and the corresponding first acoustic natural frequency is 121.4Hz. In the calculation, cross section of 10cm by 10cm is considered for the cavity and steel panels. Spring constant is determined by the given panel mass at each panel frequency. In the previous studies by the authors(1998), it has been revealed that the closer the acoustic frequency and structural frequency were, the larger the coupling effect was. At first, the effect of the panel frequency on the coupled resonance frequency is examined since the panel frequency is one of the most important factors which can change the acoustic property of the coupled system. Two typical cases are investigated here. Case 1 considers when the panel frequency is far from the cavity frequency, and case 2 when the two frequencies are adjacent. Coupled frequencies are numerically determined as the solution of the characteristic equation (13). Fig.4 explains how the solutions are obtained. In each figure, the solutions are the frequencies corresponding to the crossing points(marked by arrows) of the acoustic characteristic curve(solid line) and the structural characteristic curve(dotted line). Fig.4 (a) shows the characteristic curve for the case 1 when the panel frequency(60Hz) is far from the original cavity frequency(121.4Hz). Panel frequency is splitted into two coupled frequencies(56Hz,64Hz) and the cavity frequency shifts a little as shown in the figure and Table. 1. Fig.4 (b) shows the coupling effect in case 2, in which panel frequency(120Hz) is very close to the cavity frequency(121.4Hz). Table. 2 shows the values calculated by theory and finite element analysis. Significant shift in the frequencies is observed, which is graphically confirmed in Fig.4 (b). In all cases, theoretical results show almost the same as those of the finite element analysis by ANSYS since the calculation is performed on a rather simplified model. Fig.4 (a) and (b) indicate that the amount of frequency shift largely depends on the relative position of the structural and acoustic characteristic curves, i.e. the relation between panel frequency and cavity frequency. Fig.5 shows how the coupled frequencies vary according to the panel frequency under the constant cavity frequency(121.4Hz). Horizontal solid line means the constant uncoupled cavity frequency and dotted straight line shows the uncoupled panel frequency. Three coupled frequencies are compared with the two uncoupled frequencies. When the panel frequency is far from the cavity frequency, one coupled frequency is close to the original cavity frequency(121.4Hz) and the corresponding mode is called cavity controlled mode. Other two frequencies locate near the panel frequency. They come from the uncoupled panel frequency, therefore, corresponding modes are called panel controlled modes. Large shift is observed when the panel frequency and cavity frequency are adjacent. Coupled cavity frequency shows a little higher value than the uncoupled one, when the panel frequency is much lower than the cavity frequency. This means that panels act like an added spring in this frequency range. However, panels act as an added mass, as panel frequency comes to be larger than the cavity frequency. Panel mass is also another important factor that influences on the amount of frequency shift. Fig. 6 shows panel mass effect on the coupled frequency when both the panel and the cavity have 120Hz as their natural frequencies i.e. strong coupling and large frequency shift occur. Horizontal axis represents the ratio of the panel mass to the cavity mass. Frequency shift seems to be very sensitive to the panel mass below the ratio of 3, i.e. when the panel is lighter than 3 times of the cavity weight. Frequency shift rate rapidly decreases as the ratio increases above 10. In a real car, panel mass is much larger than the cavity mass, therefore, panel frequency and cavity frequency will be more influential factors than the mass ratio.



(a) case 1



(b) case 2

Fig.4 Characteristic curves of a coupled system

Table. 1 Coupled natural frequencies in case 1

mode	uncoupled frequency	coupled frequency	
		theory	FEA
panel	60.00	56.33	56.33
		64.03	64.03
cavity	121.43	127.99	128.06

Table. 2 Coupled natural frequencies in case 2

mode	uncoupled frequency	coupled frequency	
		theory	FEA
panel	120.00	103.42 120.07 139.35	103.43 120.08 139.41
cavity	121.43		

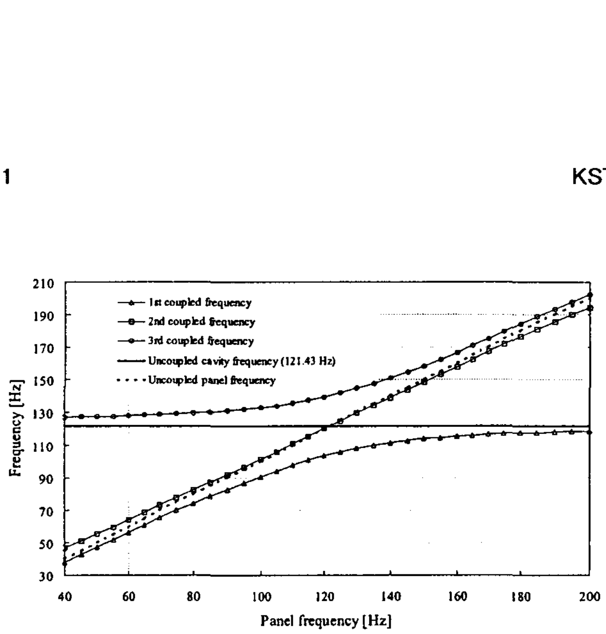


Fig.5 Coupled frequency v.s. panel frequency(cavity frequency : 120Hz)

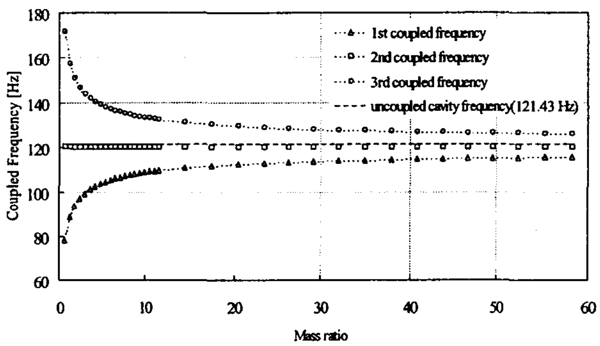


Fig.6 Coupled frequency v.s. mass ratio (panel and cavity frequency : 120Hz)

Coupled Mode

Equation (10) gives velocity mode of the panel-cavity coupled system. Fig.7 shows the velocity modes in case 1 under the same harmonic excitation force on the left panel. The 1st mode shows in-phase motion of panels and cavity particle, therefore, cavity makes a role of an added mass on the panels, causing the coupled frequency to decrease a little from the original panel frequency. Particle velocity has maximum value at the center, while the pressure mode has minimum at this point. This will be confirmed in Fig.9. The 2nd mode shows out of phase motion of panels. Therefore, cavity acts like an added spring in this mode, causing the panel frequency to increase as calculated in Table. 1. The 3rd mode is similar to the 1st mode in view of the in-phase motion, except that it has higher maximum velocity at the cavity center than in the 1st mode. The 1st and the 2nd modes are panel controlled modes since the coupled frequency are close to the original panel frequency. The 3rd mode is cavity controlled mode in the same reason. Fig. 8 shows the modes in case 2, in which the 2nd and 3rd modes are magnified in (b). The 1st mode is similar to the 1st mode in case1 since the panels and cavity particle have the same phase of motion. The 2nd mode is similar to the 2nd mode of case 1 which shows out of phase panel motion. The 3rd mode is almost same as the 3rd mode of case1.

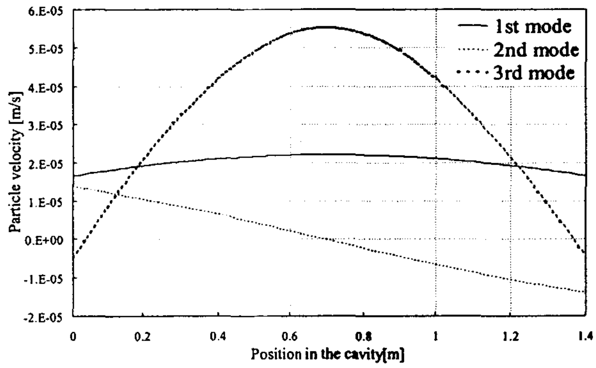
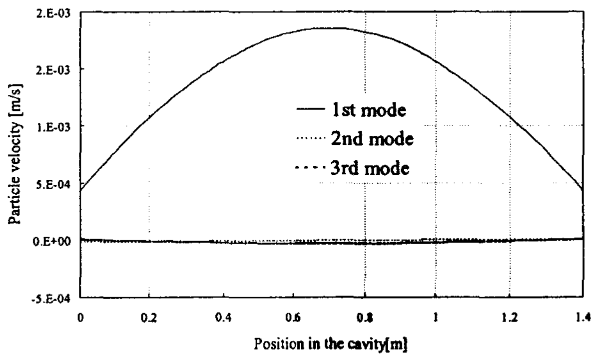
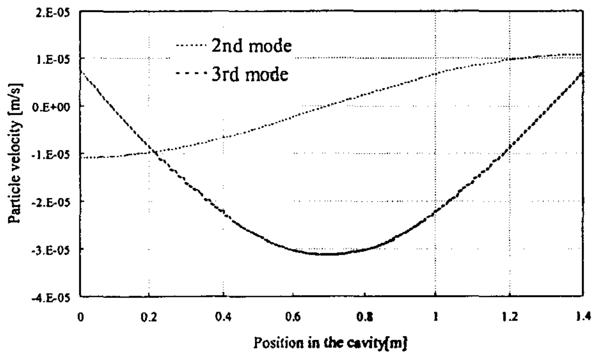


Fig.7 Velocity modes in case 1



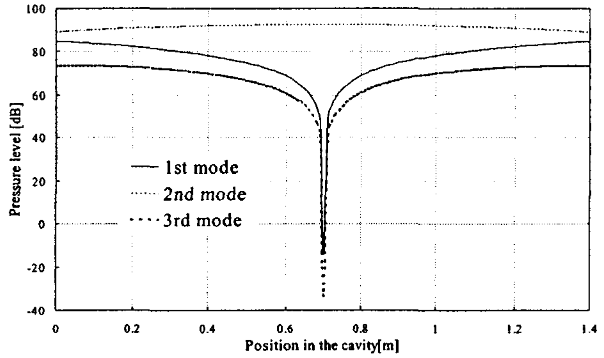
(a) velocity modes



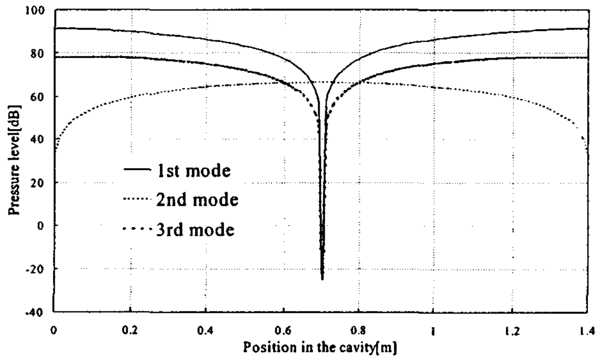
(b) 2nd and 3rd velocity modes magnified

Fig.8 Velocity modes of case 2

In most of the noise problems, pressure mode is more important acoustic property than the velocity mode because the acoustic response in the low frequency range can be represented by the sum of pressure modes. Fig.9 (a) shows the pressure distribution of the coupled natural frequencies in case1. Pressure level is calculated by using the reference value of 20μPa. As described above, the 1st and the 3rd pressure modes, which have maximum velocity at center, show pressure nodes at center position and almost the same pressure distribution over the cavity length. This means that very similar pressure distribution can be observed at two different resonance frequencies due to the coupling. This phenomenon sometimes occurs in the measurement of vehicle compartment acoustic modes. In the 2nd mode, pressure uniformly distributes over the cavity, which is same as the zero frequency mode in the cavity having rigid boundary. These are interesting features which can not be observed in the uncoupled cavity system. Fig.9 (b) shows the pressure distribution at three coupled frequencies in case 2. Panels and cavity are strongly coupled since the frequencies are very close each other. The effect on the acoustic response has been explained by using the structural-acoustic modal coupling coefficient in the semi-coupled analysis by the authors(1998). The 1st and the 3rd coupled frequencies showed large shift from the uncoupled frequencies as shown in Table. 2. However, each mode shows very similar pressure distribution to the couterpart of case 1. The 2nd mode shows uniform pressure distribution like that of case 1, but very low pressure at both ends. Consequently, analysis results say that the coupled system has different modal characteristics from the acoustic cavity model having rigid boundary, which is usually used in the analysis of vehicle passenger compartment.



(a) case 1

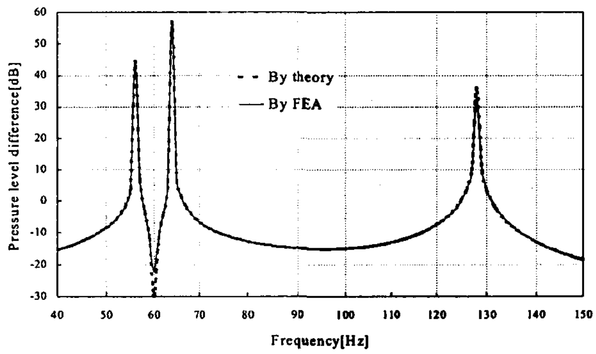


(b) case 2

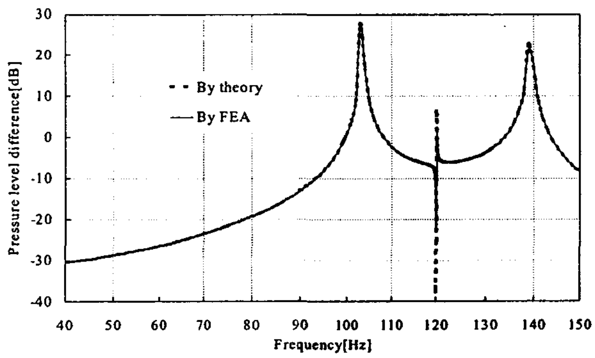
Fig.9 Pressure modes at coupled natural frequency

Frequency Response Characteristics

Finally, acoustic response characteristics of the coupled system is investigated. Pressure level difference versus frequency is determined by equation (14), in which the external pressure on the left panel is used as the reference value(-20dB means that interior pressure is one tenth of the external pressure). Therefore, the level difference on the vertical axis also means noise transmission chracteristics of the coupled system. Fig. 10 shows the pressure level difference between external pressure and the pressure at x=L in the cavity which is not the nodal point of any mode. In the figure, pressure level difference shows negative value over wide frequency range except near resonance peaks, which means that interior pressure is much lower than the external pressure. Around the coupled natural frequencies, however, high pressure level is observed due to the resonance of the panel-cavity system. This means that small external pressure or force can generate very high interior pressure at the coupled resonance frequencies, which explains the mechanism of air-borne boom in a vehicle compartment cavity. Peak level may be much reduced by the damping effect in real situation. In the semi-coupling analysis, noise peaks occur at the individual cavity frequency and panel frequency, which is different acoustic property from the full coupled system. Finite element analysis using ANSYS is performed on the acoustic frequency response characteristics of the coupled system to compare the results and to estimate the reliability of the theoretical model. In Fig.10, the results show good agreements with the theoretical values in both case.



(a) case 1



(b) case 2

Fig.10 Acoustic frequency response of the coupled system

Although above investigation is carried out for the coupling between a lateral acoustic mode and door panel mode, similar analysis is available for other modal coupling. In a real vehicle system, coupled acoustic response is so complicated that the full coupling effect cannot be distinguished from the various errors in measurement or calculation. This is due to the complexity of the modal characteristics of the car body structures. However, as investigated above, full coupling effect seems to be negligible as long as frequencies of the compartment and the car body are not adjacent each other.

Conclusions

A simplified panel-cavity model is analytically considered to understand the acoustic response of a vehicle compartment having vibrating panels, in the low frequency range. Coupled resonance frequency largely shifts from the uncoupled value as the panel frequency approaches to the cavity frequency and the mass of panel decreases. By the coupling effect, almost the same pressure modes are observed at two different coupled frequencies and also, zero frequency cavity mode(uniform pressure mode) occurs at non-zero frequency. Around the coupled resonance frequency, interior pressure shows much higher level than the external pressure, which means high sound transmissibility. The results of this theoretical study are useful to understand the full coupling phenomenon in the vehicle compartment acoustics.

Acknowledgement

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