Available online at www.sciencedirect.com
SCIENCE DIRECT®
ELSEVIER
Journal of Sound and Vibration 271 (2004) 1158–1162
JOURNAL OF SOUND AND VIBRATION
www.elsevier.com/locate/jsvi

Author's reply ☆

S.W. Kang[a,*], J.M. Lee[b]

[a] *Department of Mechanical Systems Engineering, Hansung University, 389, 2-ga, Samsun-dong, Sungbuk-gu, Seoul 136-792, South Korea*

[b] *School of Mechanical and Aerospace Engineering, Seoul National University, San 56-1 Shinlim-dong, Kwanak-gu, Seoul 151-742, South Korea*

Received 23 June 2003

1. Introduction

The authors thank Dr. Fahy for his comments on our paper [1]. The comments may be condensed into three claims:

1. The air-gap system cannot be called a sound absorber.
2. The influence of the panel (partition sheet) damping on the performance of the air-gap system should be explored.
3. A comparison between theoretical results and experimental ones should be presented.

2. Technical consideration

Our opinions on the above three claims are as follows:

(1) *The air-gap system cannot be called a sound absorber*: Dr. Fahy indicates that the air-gap system should be termed 'dynamic neutrailizer' or 'detuner' instead of 'sound absorber', because no consideration is taken for the damping of the panels (partition sheets) and 'absorption' can be produced only by a dissipative process due to damping. We essentially agree with the above indication. However, the air-gap system may be called a 'sound absorber' because the dynamic absorber, composed of mass m_a, spring constant s_a and damping coefficient R_a in Fig. 1, is called a 'dynamic absorber' in both cases of $R_a = 0$ and $R_a \neq 0$; i.e., it is called an 'undamped dynamic absorber' for $R_a = 0$ or 'damped dynamic absorber' for $R_a \neq 0$ [2]. The reason that the terminology 'absorber' is commonly used for both cases is that the vibration energy of the main system composed of mass m and spring constant s in Fig. 1 is absorbed (moved) into the dynamic absorber when m_a and s_a are appropriately tuned.

☆ Reply to doi:10.1016/j.jsv.2003.08.005.

*Corresponding author. Tel.: +82-2-760-4228; fax: +82-2-760-4327.
E-mail address: swkang@hansung.ac.kr (S.W. Kang).

0022-460X/$ - see front matter
doi:10.1016/j.jsv.2003.08.017

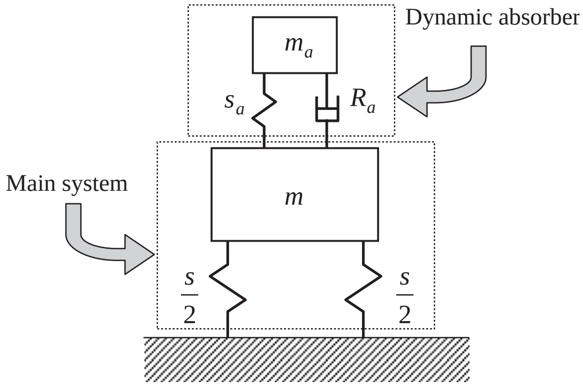

Fig. 1. Dynamic absorber of mass m_a, spring constant s_a and damping coefficient R_a.

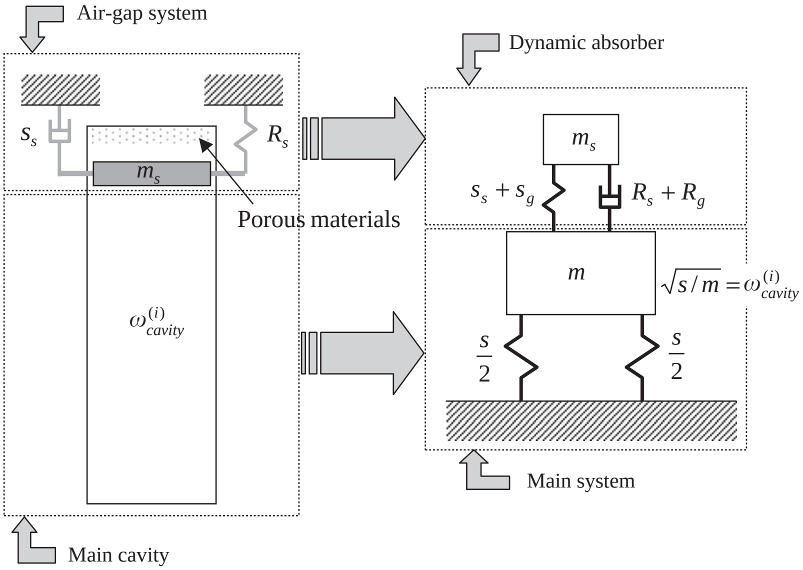

Fig. 2. Analogy of a cavity with an air-gap system to a vibratory system with a dynamic absorber.

Similarly, for the air-gap system, the acoustic energy of the main cavity indicated in Fig. 2 is absorbed (moved) into the gap when the gap thickness and the modal characteristics of the partition sheet are suitably tuned. From this fact, the air-gap system may also be called a 'sound absorber' regardless of the existence of the panel (partition sheet) damping. An 'absorption process' in the air-gap system means the movement of the acoustic energy to the gap, not the dissipative action of the acoustic energy.

(2) *The influence of the panel* (*partition sheet*) *damping on the performance of the air-gap system should be explored*: The so-called *damped* air-gap system, which considers both the damping of the partition sheet and the sound absorption of porous materials inside the gap, is being investigated by the authors. The current and future work on the damped air-gap system are concisely addressed here.

As shown in Fig. 2, to explore more analytically the effect of the damping elements in the air-gap system, the acoustic–structural coupled system composed of the main cavity and the air-gap system is replaced by the simple two-degree-of-freedom vibratory system. Mass m and spring constant s of the main system in the simple vibratory system are determined so that the natural frequency $\sqrt{s/m}$ of the main system can equal $\omega^{(i)}_{cavity}$, which denotes the ith acoustic natural frequency of the main cavity. Note that the target resonance that must be suppressed by the air-gap system is the resonance of the ith acoustic mode of the cavity in the current case ($\sqrt{s/m} = \omega^{(i)}_{cavity}$).

Next, in order to determine the mass and spring constant of the dynamic absorber that can simulate the air-gap system in Fig. 2, the non-dimensional resonance frequency of the air-gap system obtained in the previous study [1] is employed. In the previous work [1], the non-dimensional resonance frequency Ω_{reso} of the single gap system was extracted as

$$\Omega_{reso} = \sqrt{\Omega_s^2 + 1/(\pi^2 \mu_s \eta)}, \tag{1}$$

where Ω_s and μ_s denotes the dimensionless natural frequency and mass of the partition sheet, respectively; η represents the dimensionless gap thickness. For more detailed explanations on Eq. (1), refer to the previous paper [1]. By using $\Omega_{reso} = \omega_{reso}/\omega^{(1)}_{cavity}$, $\Omega_s = \omega_s/\omega^{(1)}_{cavity}$, $\mu_s = m_s/M_{cavity}$ and $\eta = \Delta L/L_{eq}$ expressed as the resonance frequency of the air-gap system ω_{reso}, the mass of air inside the cavity M_{cavity}, the gap thickness ΔL, and the equivalent length of the cavity $L_{eq} = \pi c/\omega_{target}$ (ω_{target}: the frequency (unit, rad/s) of a target resonance peak), Eq. (1) may be written in the dimensional form

$$\omega_{reso} = \sqrt{(s_s + s_g)/m_s}, \tag{2}$$

where s_s and m_s denotes the spring constant and mass of the partition sheet in Fig. 2; $s_g = \rho c^2 S/\Delta L$ denotes an added stiffness expressed by the density of air ρ, the speed of sound c, and the area of the partition sheet S. Note that the added stiffness is generated by sound pressure inside the gap, and that it is inversely proportional to the gap thickness ΔL.

From Eq. (2), the air-gap system may be considered as the dynamic absorber with mass m_s and spring constant $s_s + s_g$ as shown in Fig. 2. Note that R_s and R_g are, respectively, related to the damping of the partition sheet and the absorption of the porous materials inside the gap. Finally, the two-degree-of-freedom vibratory system as illustrated in Fig. 2 will be used to explore the influence of the damping ($R_s + R_g$) of the air-gap system on an acoustic frequency response at a position in the main cavity.

(3) *A comparison between theoretical results and experimental ones should be presented*: To verify the resonance frequencies of the double-gap system presented in the paper [1], experiments on the double-gap system have been carried out as shown in Fig. 3. Experimental results and discussions have been fully addressed in another paper that will be published soon. Furthermore, investigations into the influence of the damping in the design optimization of the air-gap system have also been performed, in a similar manner that was explored in Fahy's paper [3].

Here, one of many experimental results obtained is shown in Fig. 4, where the double-gap system needs less gap than the single-gap system does to suppress the target resonance peak. The single-gap system needs a 10.5 cm-gap but the double-gap system needs only a 6 cm-gap, which is

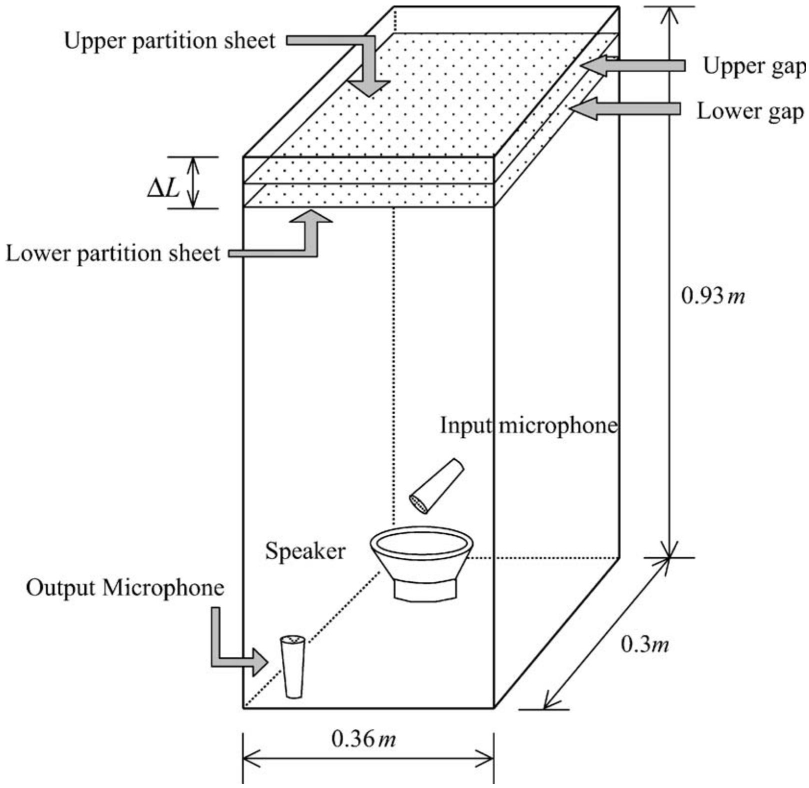

Fig. 3. Experimental set-up of a box-shaped cavity with the double-gap system.

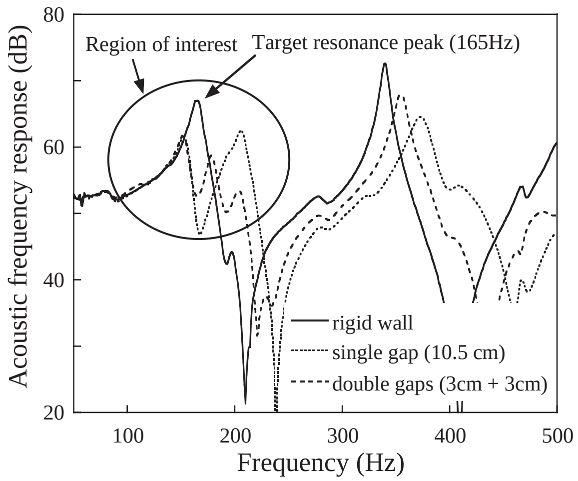

Fig. 4. Acoustic frequency responses measured inside the box-shaped cavity.

equally divided into the upper and lower gaps. This usefulness of the double-gap system has already been predicted from the theoretical model on the left hand side of Fig. 2 [1]. Note that the acoustic frequency responses change interestingly when the upper and lower gaps are not divided equally. The discussion on the changes in the acoustic frequency responses is given in another paper that will be published.

Acknowledgements

This research was financially supported by Hansung University in the year of 2002.

References

[1] S.W. Kang, J.M. Lee, Sound absorption characteristics of air-gap systems in enclosed cavities, *Journal of Sound and Vibration* 259 (1) (2003) 209–218.
[2] S.T. Francis, E.M. Ivan, T.H. Rolland, Mechanical Vibrations, 2nd Edition, Allyn and Bacon, Newton, MA, 1978, pp. 169–173.
[3] F.J. Fahy, C. Schofield, A note on the interaction between a Helmholtz resonator and acoustic mode of an enclosure, *Journal of Sound and Vibration* 72 (1980) 365–378.