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Geometric and number effect on damping capacity of Helmholtz resonators in a model chamber

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ABSTRACT

An acoustic cavity was selected as a stabilization device to control high-frequency combustion instabilities in gas turbines or liquid rocket engine combustors, and the acoustic damping capacity of the acoustic cavity was investigated for various geometric configurations under atmospheric non-reacting conditions. The tuning frequency of the acoustic cavity and the acoustic responses of a model chamber with a single acoustic cavity were studied first. Damping capacity was initially quantified through the frequency width of two split modes and the amplitude-damped ratio. The results showed that the cavity with the largest orifice area or the shortest orifice length was the most effective in acoustic damping of the harmful resonant mode. The effect of the number of cavities on acoustic damping capacity was also studied. Damping capacity was improved by increasing the number of cavities. For a better evaluation of acoustic damping capacity, two quantified parameters; the acoustic absorption, meaning the damping efficiency, and acoustic conductance, meaning the acoustic power loss, were introduced. The case was observed that has had insufficient loss of acoustic power in spite of having the highest absorption efficiency. As a result, fine geometric tuning for the acoustic cavity is required for the sufficient passive control. Also, the choice of the number of cavities is important to optimize the damping efficiency and absolute damping loss in consideration of the restriction of the cavity volume.

1. Introduction

High pressure and large scale combustors have been widely adopted to achieve a large amount of thrust for high performance liquid rocket engine systems [1,2]. In addition, lean premixed combustion conditions have been preferably used to reduce NO_x emissions in gas turbine combustors [3]. Therefore, these circumstances make them more susceptible to damaging combustion instabilities. Combustion instabilities of high-frequency characteristics, also known as acoustic instabilities, have been induced generally by the in-phase interaction of acoustic fields and unsteady heat release by combustion responses. These self-excited pressure oscillations can cause deterioration of propulsion performance and even cause severe physical damage to the injector faceplate and the chamber wall in propulsion devices [1–5].

In attempts to suppress combustion instabilities, both active and passive control methods have been used to break the strong coupling between acoustic waves and unsteady heat release [1–6]. The active control method uses a pressure

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detecting system and sequential dynamic actuators such as valves or loudspeakers which interrupt these couplings. Therefore, active controllers should have fast response characteristics to make the system stable especially in the high frequency regime. In a practical sense, the fuel modulation technique is widely used, but there is a bandwidth limit for fuel valves. Also, active control schemes often have the tendency to make the system more unstable due to the system complexity [3,7].

Passive control techniques generally used additional acoustic dampers for the combustors. These methods have the disadvantage of not being applicable to the transient operational periods, although there are less likely to make the system more unstable. Recently, an adaptive passive control method has been investigated to choose the advantages of both active and passive control methods [8–10]. As acoustic dampers, baffles and acoustic cavities which modify acoustic fields of a combustion chamber are widely used [1–6,11,12]. Baffles can be easily designed to install on the injection faceplate, but have accompanying problems such as cooling and structural concerns which would be severe in a high pressure range due to the increased thermal load on the baffle surfaces. To minimize these cooling problems and performance/thrust loss due to the installed baffles, optimization of baffle configuration such as length-shortening and arrangement with gaps is necessary to satisfy both damping capacity and thermal survivability [13,14].

On the contrary, acoustic cavities such as quarter-, half-wave resonators and Helmholtz resonators do not give rise to severe secondary problems compared with baffles, but they must be finely tuned to effectively eliminate combustion instabilities. For the fine tuning, the properties of the fluid in the cavities such as its molecular composition and temperature for the precise evaluation of sonic velocity are very important in the actual propulsion devices under combustion conditions. Also, the geometrical configuration effect of acoustic cavities such as orifice area (diameter), orifice length, and cavity volume should be scrutinized. Previous literature [6,15–18] provides useful information on design procedures of typical resonators, but quantitative information is not fully available and still depends on trial and error methods in actual situations [19,20]. In this regard, the evaluation of damping capacity as well as tuning frequency of the acoustic cavity is necessary. The reason is that the combustion system could be unstable even if the acoustic dampers are finely tuned, unless the control systems have enough damping capacity. Therefore, the objective of the present study is focused on the quantitative evaluation of the geometric and number effect on the damping capacity of acoustic cavities in Helmholtz type. Based on the experimental results, the present study provides practical design guidelines for optimal acoustic resonators with satisfactory damping capacity.

2. Experimental methods

2.1. Theoretical background

The nozzle throat of a combustion chamber is generally considered to be acoustically closed due to choking near the nozzle throat [15,21]. Therefore, if the chamber is assumed to be a cylindrical shape closed at both ends with uniform cross-sectional area, the theoretical solutions for resonant frequencies of the chamber can be determined from [21,22]:

$$f_{l,m,n} = \frac{c}{2\pi}\sqrt{\left(\frac{\lambda_{mn}}{R_c}\right)^2 + \left(\frac{\pi l}{L_c}\right)^2} \tag{1}$$

where c is the speed of sound, R_c is the radius of the chamber, L_c represents the idealized cylindrical length of the chamber, the subscripts l,m,n are indices for longitudinal, tangential, and radial mode, respectively; and λ_{mn} is the eigenvalue of transverse modes [15,21,22].

Acoustic resonators located along the circumference of a chamber are connected to this chamber through orifices. Pressure perturbations in the chamber can be suppressed by viscous dissipations which occur with vibrations of the gases in the orifice. Fig. 1 shows a schematic of a typical Helmholtz resonator which is one of the representative resonators. The undamped resonant frequency f_0 of the Helmholtz resonator can be obtained from [21,22]:

$$f_0 = \frac{c}{2\pi}\sqrt{\frac{S}{V(\ell+\Delta\ell)}} \tag{2}$$

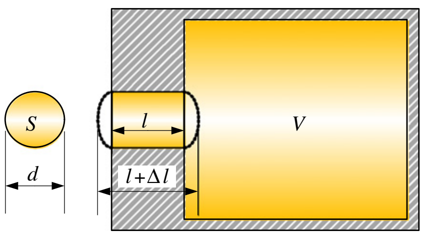

Fig. 1. Schematic of typical Helmholtz resonator.

where S is the orifice area, ℓ is the length of the orifice, V is the cavity volume, and $\Delta\ell$ is the correction factor for compensating the vibrating mass of gases in the orifice [21,22]. In the present study, $\Delta\ell = 0.85d$ is used.

2.2. Determination of a model chamber and acoustic cavities

In this study, we designed a model chamber which will be used to investigate the damping effects of acoustic cavities at its resonant frequency. The damping effect of an acoustic cavity is known to be irrelative to the direction of the resonant mode, so the model chamber is designed to have only 1L mode under 1000 Hz for the convenience of this study. Its dimensions are 324 mm in length and 100 mm in diameter. Table 1 presents the resonant frequencies of this model chamber.

The effects of geometry and number of acoustic cavities on the damping of the specific resonant frequency of the model chamber are investigated to gather important information on the design of an acoustic cavity. Table 2 shows the geometrical dimensions of acoustic cavities which are first tuned to the specific resonant mode (1L in the present study) in the model chamber. The volume of the cavities can be controlled and finely tuned by a piston as shown in Fig. 2. The effects of cavity numbers (up to 5) are analyzed by FRF (frequency response function) or a sinusoidal sweeping method. The positions of microphones (H1 ~ H5) and acoustic cavities (C1 ~ C3) can be changed as shown in Fig. 2.

2.3. Absorption coefficient using an impedance tube

An absorption coefficient, α is defined as the ratio of sound power entering the surface of the test object (without return) to the incident sound power for a plane wave at normal incidence, and it can be obtained by measurements in a

Table 1
Resonant frequencies of the model chamber.

Mode	1L	2L	3L	1T	1T1L
Frequency (Hz)	524	1048	1573	1992	2060

Table 2
Dimension of acoustic cavities and orifices.

Case	ℓ (mm)	d (mm)	Area ratio (%)	V (mm^3)
A	36.8	18	3.24	5.15×10^4
B	41.8	18	3.24	4.55×10^4
C	46.8	18	3.24	4.15×10^4
D	51.8	18	3.24	3.85×10^4
E	36.8	12	1.44	2.45×10^4
F	36.8	14	1.96	3.25×10^4

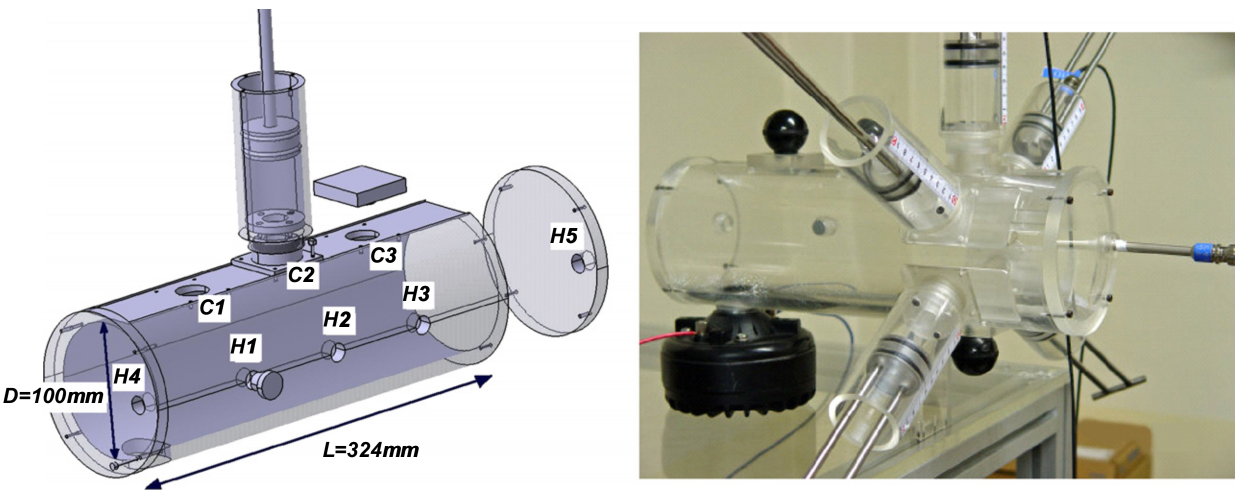

Fig. 2. Schematic of a model chamber and adopted acoustic cavities.

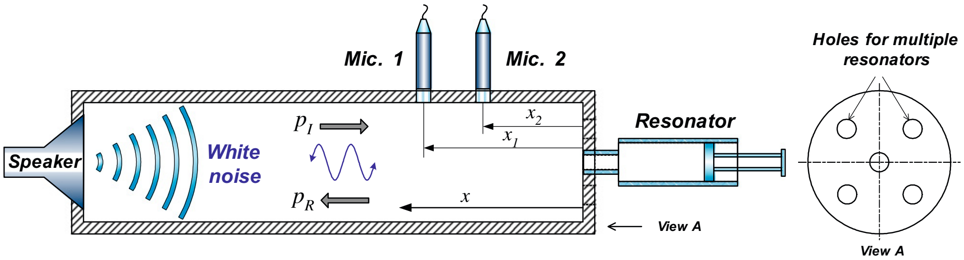

Fig. 3. Impedance tube for transfer function method.

reverberation room or the impedance tube method [22–25]. The reverberation room method uses a number of simplifying and approximate assumptions concerning the sound field and determines the sound absorption coefficient for random sound incidence. On the contrary, the impedance tube method gives relatively exact values by means of a standing wave tube and has simplicity of manufacturing test objects which are mounted at one end of a straight and rigid tube with a loudspeaker on the other end. These two methodologies are known as standing wave ratio [22,23] and transfer function method [22,24] to achieve the sound absorption coefficient of sound absorbers using an impedance tube, respectively. The method using standing wave ratio measures the minima and the maxima of sound pressure of the standing wave with a single microphone which can be moved along the tube, but the absorption coefficient should be obtained for each pure tone of interest. Therefore, it takes a much longer time to investigate the absorption coefficient in the range of frequencies of interest compared with the transfer function method. On the contrary, the transfer function method can collect absorption coefficients over a wide range of frequencies at one time with two microphones [24]. In this study, both of the methods are experimentally performed and compared to reveal the data accuracy, and then the data quantification of the experiments is mainly conducted by the latter transfer function method. Absorption coefficients in the transfer function method are obtained using the impedance tube with a sound source connected to one end and the resonator(a test sample) mounted at the other end. The sound source generates plane waves in the tube and two fixed microphones at the wall measure acoustic pressures of the incident wave p_I and the reflective wave p_R, as shown in Fig. 3.

First, the speed of sound, c in the tube is determined. While acoustic pressure is excited by the loudspeaker, acoustic amplitudes p_1 and p_2 are measured as a function of time with two microphones, Mic. 1 and Mic. 2, respectively. The acoustic signals are transformed by FFT. From FFT data, the transfer function, H_{12} is obtained by the equation,

$$H_{12} = \frac{P_2}{P_1} \tag{3}$$

where P_1 and P_2 denote the signals transformed from p_1 and p_2, respectively. The normal incidence reflection factor, r is calculated by the following equation:

$$r = \frac{H_{12} - \mathrm{e}^{-\mathrm{j}k(x_1 - x_2)}}{\mathrm{e}^{\mathrm{j}k(x_1 - x_2)} - H_{12}} \mathrm{e}^{2\mathrm{j}kx_1} \tag{4}$$

where k is the wavenumber calculated by $k = 2\pi f/c$, and x_1, x_2 are the distances between the resonator inlet and the first and the second microphone locations. Then, the sound absorption coefficient, α is finally calculated by the equation

$$\alpha = 1 - |r|^2 \tag{5}$$

2.4. Experimental apparatus

This study can be largely classified into two subjects; one is the tendencies of geometric and number effects on damping characteristics of acoustic cavities, and the other is the quantification of damping efficiency and the capacity of acoustic cavities by an absorption coefficient and acoustic conductance [22]. Fig. 4 shows the experimental apparatus for the study on tendencies in damping characteristics and the measurement of absorption coefficients of acoustic cavities, respectively. It consists of a test section, a signal generation system and a data acquisition system. The test section for damping characteristics consists of a model chamber and the acoustic cavities which are shown in Fig. 2. Another test section for absorption coefficient consists of an impedance tube (I.D. 150 mm, length 1480 mm), a loudspeaker, Helmholtz resonators, and two fixed microphones and a movable probe microphone. The signal generation systems can generate random noise or a sinusoidal signal by using a LabVIEW program and an A/D board (NI-6014), and it is connected to the speaker through an amplifier (R300 plus). The signals acquired by $\frac{1}{4}$ in microphones (M360, Microtech Gefell Gmbh) are collected by a high-speed A/D board (NI-4472) and analyzed by a LabVIEW program using power spectrum, FFT, auto-spectrum, cross-spectrum and FRF in the data acquisition system (DAQ). In particular, the test equipment for acquiring absorption coefficients is validated by a series of tests to exclude error sources and secure the minimum requirements [23,24,26].

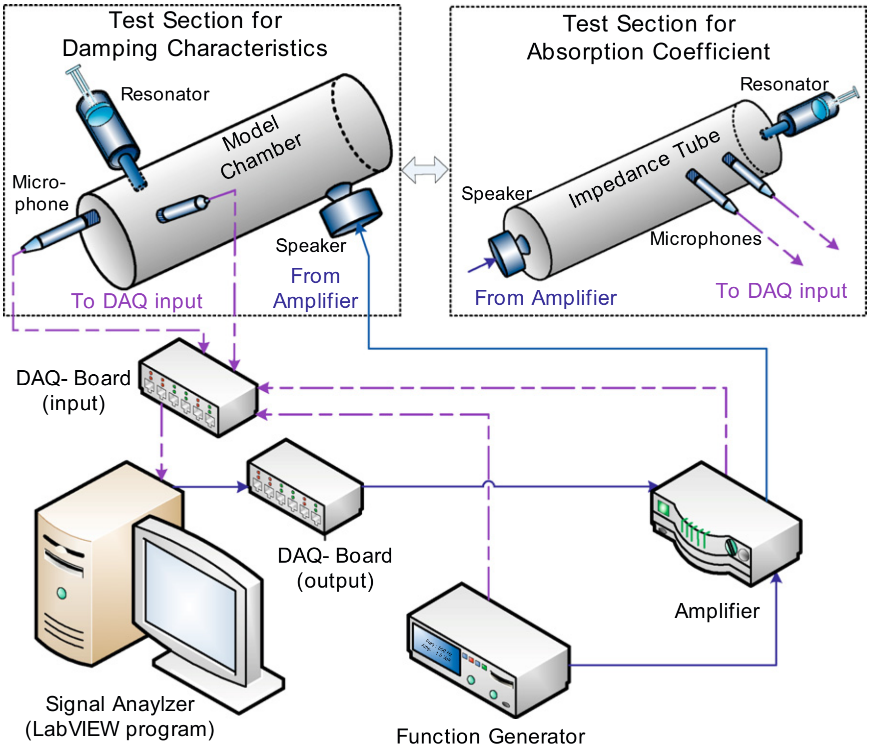

Fig. 4. Experimental apparatus for damping characteristics and absorption coefficients of acoustic cavities.

3. Geometrical and number effects of cavities

First, resonant frequencies of the present model chamber were experimentally investigated and compared with the analytic solutions obtained through Eq. (1) for validation. Fig. 5 shows the amplitude characteristics of acoustic pressure oscillations at two different axial locations; one is at a quarter of the axial length (H1) and the other is at the closed end (H4). In the frequency range of 200–1800 Hz, all pure longitudinal acoustic modes can be measured at the H4 point. But at the H1 point, the $2n$-th longitudinal modes cannot be detected because this point is near the nodal plane of these modes. The resonant frequency of the first longitudinal (1L) mode, 530 Hz, is quantitatively the same with the analytic value of 524 Hz, showing just about a 1 pecrcent difference. In addition, the resonant frequencies of other detectable acoustic modes showed satisfactory agreement with analytic values, as shown in Table 1. Therefore, the present data acquisition system is thought to be satisfactory to study the acoustic characteristics of 1L mode which is selected to be tuned by acoustic cavities.

In the model chamber with a single cavity of various cavity volumes with d=18 mm and $\ell = 36.8$ mm, acoustic pressure responses were measured at the position of H4 for various cavity volumes and are shown in Fig. 6. The original 1L mode in a model chamber without cavity was split into two separate modes with the cavity. For a cavity with small volume (V=38,955 mm^3), there was one lower mode of high amplitude and a similar resonant frequency with the original mode, and the another upper mode of low amplitude and higher resonant frequency. As the volume of the cavity increases, the former lower mode moves away from the original 1L mode in frequency and its amplitude decreases. On the contrary, the latter upper mode approaches the original mode in frequency and its amplitude increases with the volume of the cavity. These characteristics show that improper cavity volume cannot modify the acoustic mode sufficiently away from the original mode which is desired to be damped, and acoustically well-tuned cavities should be used to suppress combustion instabilities which would occur through the coupling of certain acoustic modes and combustion processes.

The cavity volume that had the smallest amplitude at the original resonant frequency (530 Hz) was V=51,522 mm^3, and this case was selected as an effective tuning geometry. Therefore, as in Table 2, we chose various combinations of orifice lengths, diameters and cavity volumes by Eq. (2) and applied this procedure to investigate the geometrical effects on the damping characteristics of cavities.

The tuning frequency of an acoustic cavity by geometric variations was measured through two different methods. One is the method where the responses of an isolated cavity detached from the model chamber are examined with respect to the perturbing sinewave at the orifice inlet. The other is through the amplitude ratio of the cavity to the chamber, p_{ac}/p_{ch}

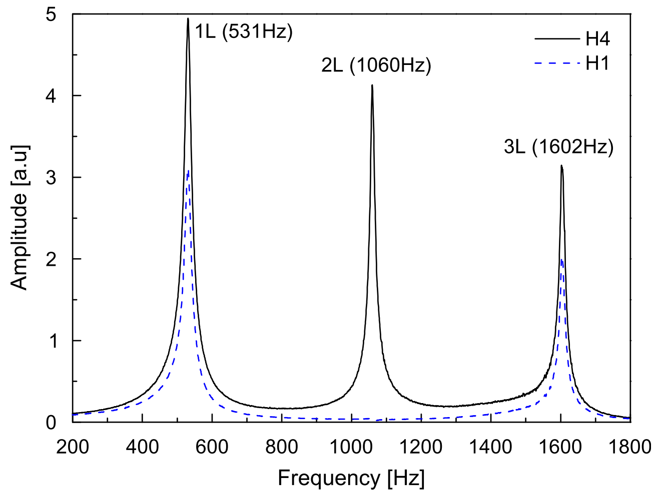

Fig. 5. Acoustic response characteristics of model chamber without cavity (uniform white noise).

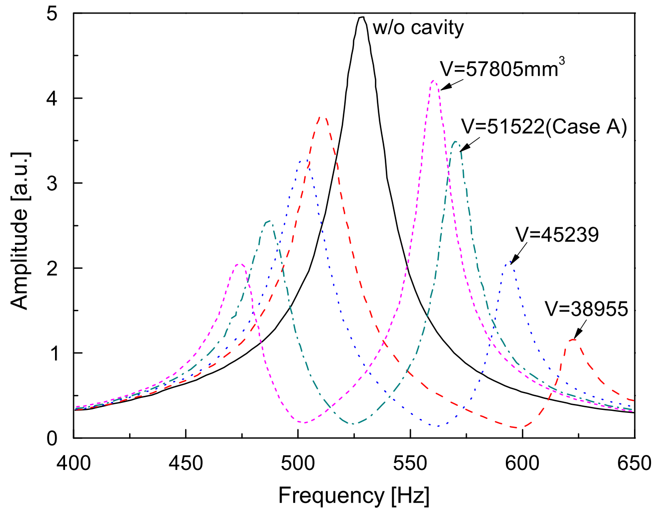

Fig. 6. Acoustic response characteristics of model chamber on variable volume of cavity with d=18 mm and $\ell = 36.8$ mm.

when the cavity was installed with the chamber [27,28]. The latter means that the amplitude of the acoustic pressure in a chamber, p_{ch} is damped out the most for the frequency having maximum amplitude ratio, p_{ac}/p_{ch}. Fig. 7 shows the measured tuning frequency of the cavity with d=18 mm and $\ell = 36.8$ mm. Very similar response characteristics have been obtained through these two methods, which show the cavity adopted in the present study has a resonance of about 530 Hz.

The geometric configuration of orifices in cavities such as orifice length or area is one of the main parameters determining the damping capacity of acoustic cavities. First, the effect of orifice length, ℓ on damping characteristics of acoustic cavities has been studied under a fixed orifice diameter, d, as shown in Fig. 8. For the Cases A–D in Table 2, the increase of orifice length results in the decrease of cavity volume to make constant the tuning frequency of the cavity, which can be inferred from Eq. (2). Although the damping capacity increases with the orifice length, the decrease of cavity volume for a constant tuning frequency decreases the damping capacity. Therefore, the overall resultant damping capacity shows minor change with respect to the orifice length.

The ratio of the magnitude of amplitude decrease at this resonant frequency with cavity to the pressure amplitude of a resonant mode without cavity is called the “amplitude-damped ratio” in this study, and is used to quantify how much the resonant mode could be damped by an acoustic cavity. In addition, the frequency difference between upper and lower modes, also known as the frequency width of mode split is thought to be an important parameter indicating how much the original mode can be modified and decoupled from the combustion processes by installing acoustic cavities. Fig. 9 shows these amplitude-damped ratios and the frequency widths of mode split with respect to the orifice length. These amplitude-damped ratios and the frequency widths of mode split all show a similar tendency of slightly decreasing with orifice length. The increase of orifice length improves viscous dissipation which is one of the important factors to suppress

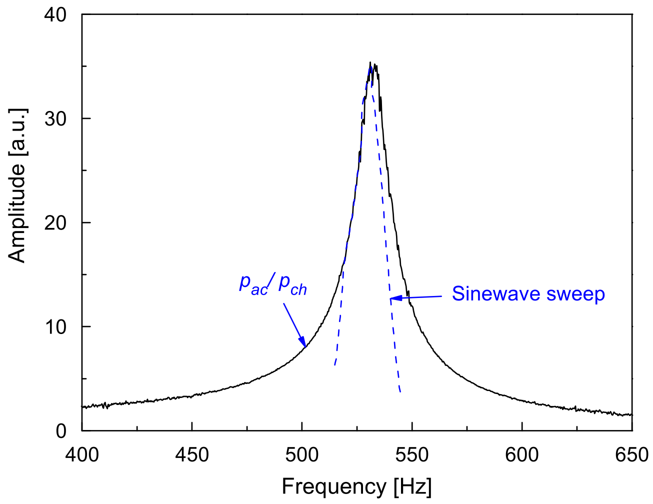

Fig. 7. Tuning frequency of acoustic cavity.

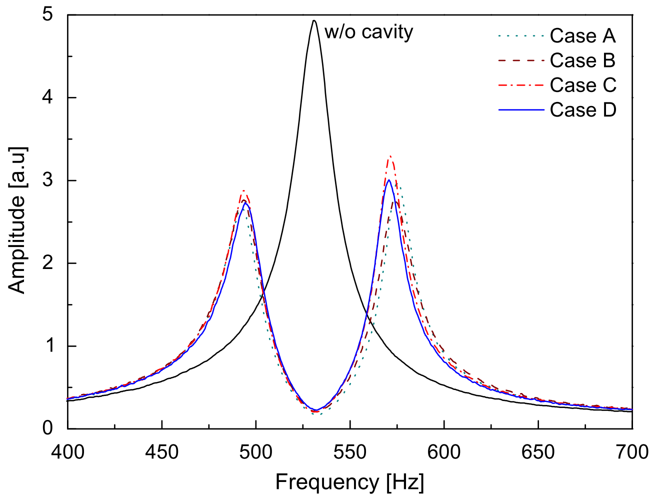

Fig. 8. Feature of acoustic damping in a model chamber with various orifice lengths of acoustic cavity.

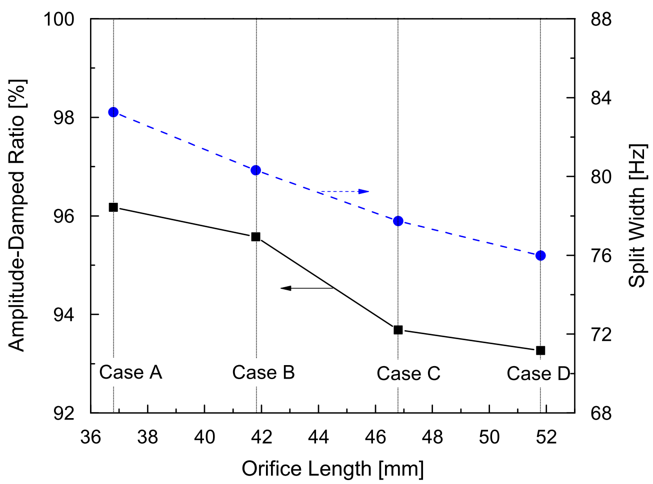

Fig. 9. Damping ratios of amplitude and frequency widths of mode split for various orifice lengths.

combustion instability, but the increase of viscous dissipation due to the increase of orifice length is slightly overwhelmed by the decrease of cavity volume to make the tuning frequency constant. It could be thought to imply that cavity volume is a more dominant factor than the orifice length, and conclusively the shorter orifice length is more effective in this case.

Like Cases A, E, and F in Table 2, the effect of orifice area, S has been investigated and shown in Fig. 10. As inferred from Eq. (2), the increase of orifice area, i.e. orifice diameter, results in the increase of cavity volume to make constant the tuning frequency of the cavity. The larger the orifice area and the cavity volume are, the more the damping capacity is. Therefore, overall amplitude-damped ratios and frequency widths of mode split showed fair increase with respect to the orifice area. Fig. 11 shows these characteristics in terms of open area ratio which can be defined as the ratio of orifice area, S and cross-sectional area of combustion chamber. As a result, it would be recommended to make the orifice area as large as possible to increase the damping capability.

Orifice area is generally known to be a significant parameter in designing acoustic cavities. As explained earlier, it has been suggested from experience that the total orifice area of all cavities should exceed at least a certain guideline value of the cross-sectional area of the combustion chamber [17,19,20]. But these guideline values are different from each other and 20 pecrcent of the cross-sectional area of combustion chamber, i.e. open area ratio =20 pecrcent, recommended by Christensen and Nesman [20], is thought to be too large to be practical. In general, there would be some inevitable geometric limitations because acoustic cavities are mostly installed on the periphery of the injection head or chamber wall. So, the optimization of orifice area should be necessary to guarantee sufficient combustion stabilization capacity.

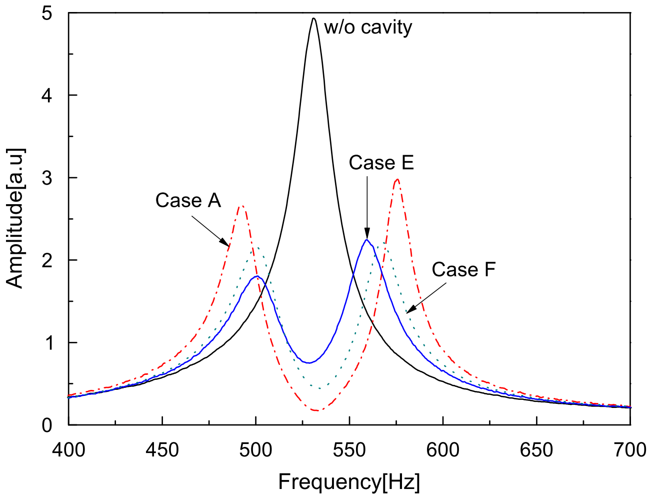

Fig. 10. Feature of acoustic damping in a model chamber with various orifice areas of acoustic cavity.

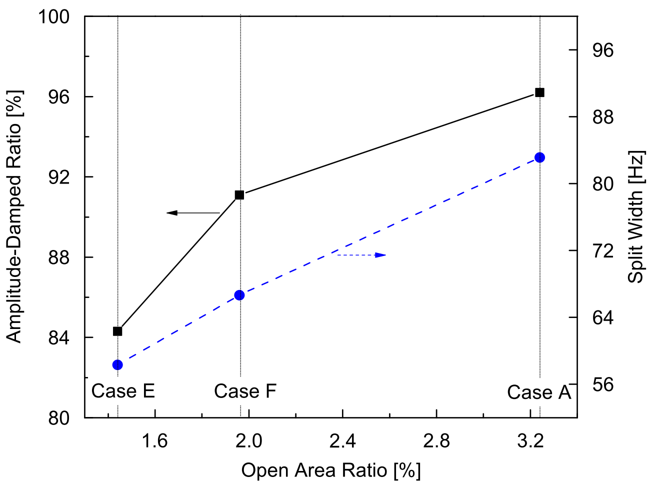

Fig. 11. Damping ratios of amplitude and frequency widths of mode split for various orifice areas.

It is possible to estimate the frequencies under actual conditions in combustion chambers by simply multiplying the ratio of sonic velocity between cold and hot-firing conditions, about 3.5 [15]. In this respect, the frequency width of mode split in Case A is estimated to be 290 Hz, which can be considered sufficient to decouple the harmful acoustic modes with the combustion processes.

In general, multiple cavities or resonators are installed to suppress combustion instabilities in gas turbines or liquid rocket combustors. In this respect, acoustic responses with several cavities(up to 5) have been investigated to quantify the effect of cavity numbers on the damping capabilities in Case A and Case E, where previously Case A was the most effectively damped one and Case E was the worst in the viewpoint of amplitude-damped ratio and frequency widths of mode split. Fig. 12 shows the acoustic responses with respect to the number of cavities in Cases A and E, respectively. The two cases both have a similar tendency that the amplitude-damped ratios and the frequency widths of mode split increase with the number of cavities. In Case A, the amplitude-damped ratio with only one cavity was 96.2 pecrcent which was sufficiently high. So the amplitude-damped ratio was not increased substantially from the viewpoint of cavity number.

On the contrary, Case E revealed some remarkable behavior that showed a steeper increase of amplitude-damped ratio with respect to the number of cavities. These characteristics are quantified through the amplitude-damped ratio for damping and the frequency widths with respect to the number of cavities and shown in Fig. 13. But parameters such as amplitude-damped ratio and the frequency widths of mode split are thought to be insufficient to fully explain the acoustic damping capacity quantitatively. Therefore, the measurement of acoustic absorption is adopted in the following section to evaluate the damping capability of acoustic cavities.

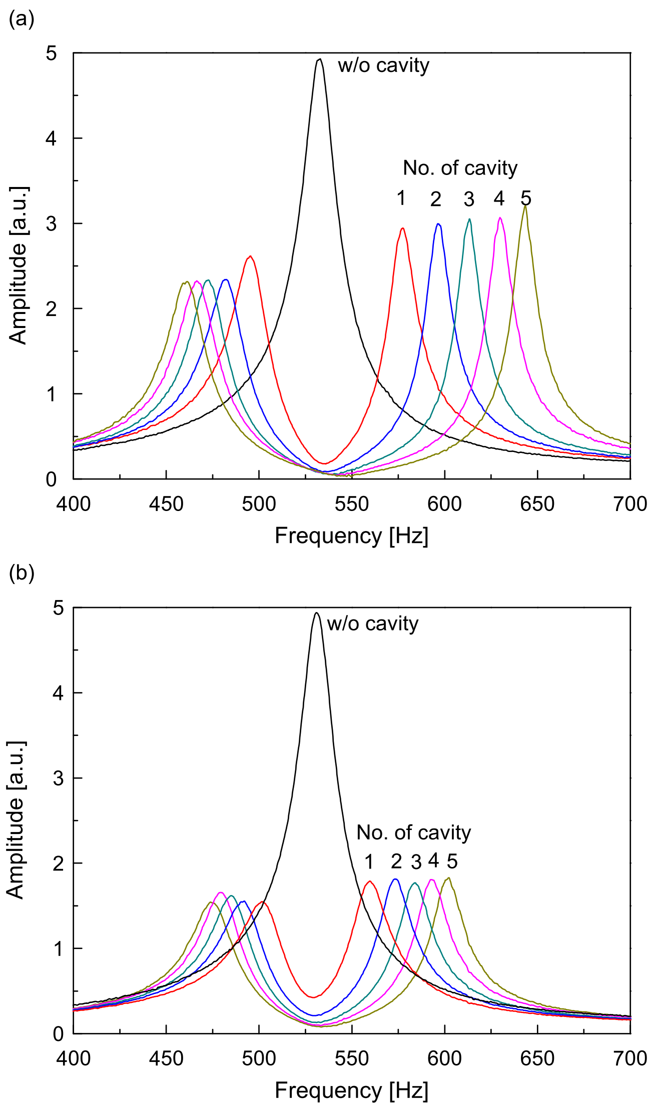

Fig. 12. Feature of acoustic attenuation for various number of cavities. (a) Case A and (b) Case E.

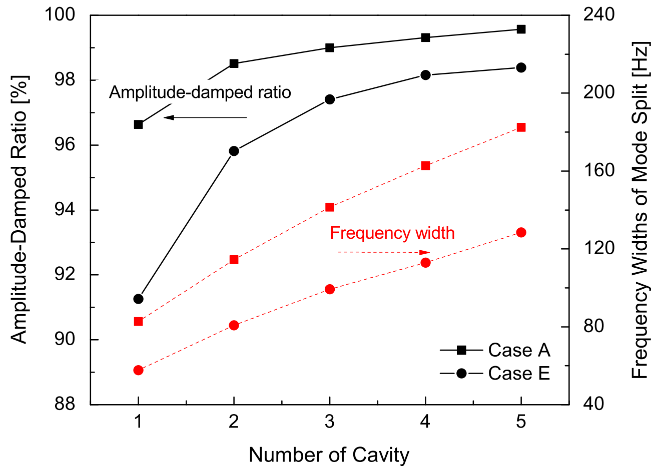

Fig. 13. Damping ratios of amplitude and frequency widths of mode split for various number of cavities.

4. Absorption coefficients of acoustic cavities

Two methods of absorption coefficient measurement using impedance tube, standing wave method [23,27] and transfer function method [22,24] are known to be essentially the same, although they are different in measuring technique and data processing. Fig. 14 shows the absorption coefficient of one acoustic cavity installed in a model chamber using these two methods. These results show satisfactory agreement with each other, especially in the vicinity of the tuning frequency, 530 Hz. As shown in Fig. 14, the standing wave method requires many more tests to obtain a reliable absorption coefficient with sufficient resolution compared to the transfer function method. On the contrary, the transfer function method is relatively fast and has good resolution, although it is more complex in experimental setup and data processing. Therefore, absorption coefficients of cavities in this paper are analyzed afterward by the transfer function method, because the setup and data processing were proved by comparing the two methods.

Fig. 15 shows the absorption coefficient of one cavity with respect to orifice length and orifice area by using the transfer function method. In Fig. 15(a), maximum absorption was obtained with the tuning frequency of 530 Hz with respect to the orifice length. Damping did not show much increase with respect to orifice length in the geometric variations of the present study, which was in satisfactory agreement with the previous result of Fig. 8. Variations of acoustic absorption with respect to orifice area are also shown in Fig. 15(b). The least absorption was observed at the smallest orifice area in Case E and it could be thought of as an under-damped condition. These behaviors show a similar tendency with Fig. 10 in the previous section.

Next, the effect of cavity number has been investigated through the schematic setup in Fig. 3 and results have been shown in Figs. 16 and 17. In Case A, the absorption coefficient decreased and its bandwidth increased with the number of acoustic cavities. Intuitively, an increase of bandwidth means that there would be much more possibility of suppressing combustion instabilities by preventing the coupling of acoustic and combustion fields. On the other hand, the absorption coefficient increased continuously from 1 to 3 cavities, and decreased above three cavities in Case E. But the variations of bandwidth showed monotonic increase with the number of acoustic cavities, as in Case A.

As shown in Fig. 16, the existence of maximum absorption with only one cavity in Case A means that this condition has enough damping efficiency, i.e. optimal damped condition. Therefore, 2 or more cavities showed over-damped condition. Although the absorption coefficient, in other words, the efficiency for damping decreases with the number of acoustic cavities, absolute quantity of energy damping increases with the number of cavities.

On the contrary, 1- and 2-cavity conditions were under-damped, 3-cavity condition was optimally damped, and four and more cavity conditions were over-damped in Case E, as shown in Fig. 17. These characteristics are qualitatively similar to those in the previous literature [15]. General guidelines to determine the number of cavities are known to achieve the maximum absorption coefficient close to 100 pecrcent, and then if impossible, to maximize the absolute damping quantity by increasing the number of cavities [15,17]. If we follow this rule, just one cavity is sufficient in Case A, and three cavities are necessary in Case E. But optimization of absorption efficiency and absolute quantity of acoustic damping should be considered thoroughly to achieve combustion stabilization by installing acoustic cavities. Obviously, this determining method should be verified through hot-firing tests which can confirm certain and reliable damping capability by coupling of acoustic and combustion fields.

Acoustic conductance, ξ, is known to be the acoustic power loss factor [15,17,21,22]. As the imaginary part of complex acoustic impedance is generally 0 at the tuning frequency, the absorption coefficient and acoustic conductance are all dependent only on the acoustic resistance which is the real part of the acoustic impedance. Therefore, the tuned acoustic

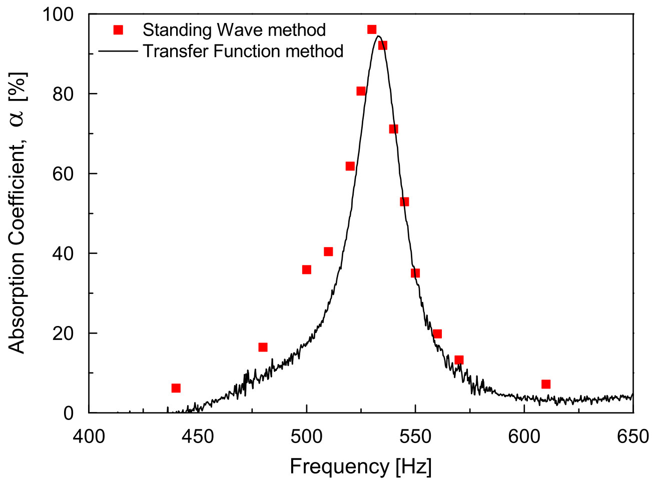

Fig. 14. Absorption coefficients from standing wave method and transfer function method.

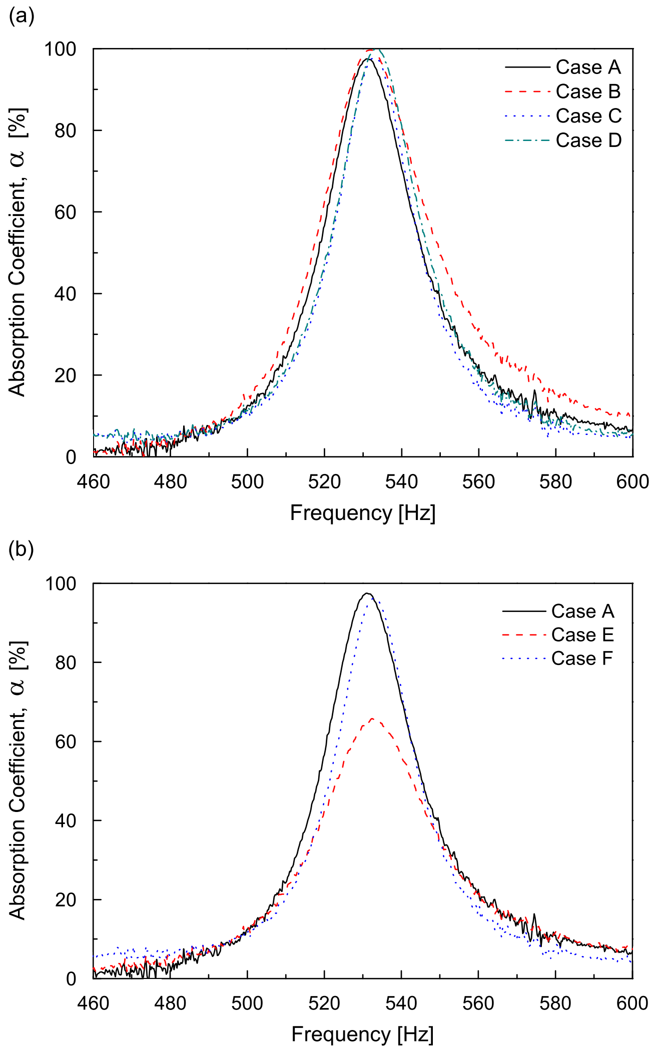

Fig. 15. Absorption coefficients for various geometric configurations. Various orifice (a) lengths and (b) areas.

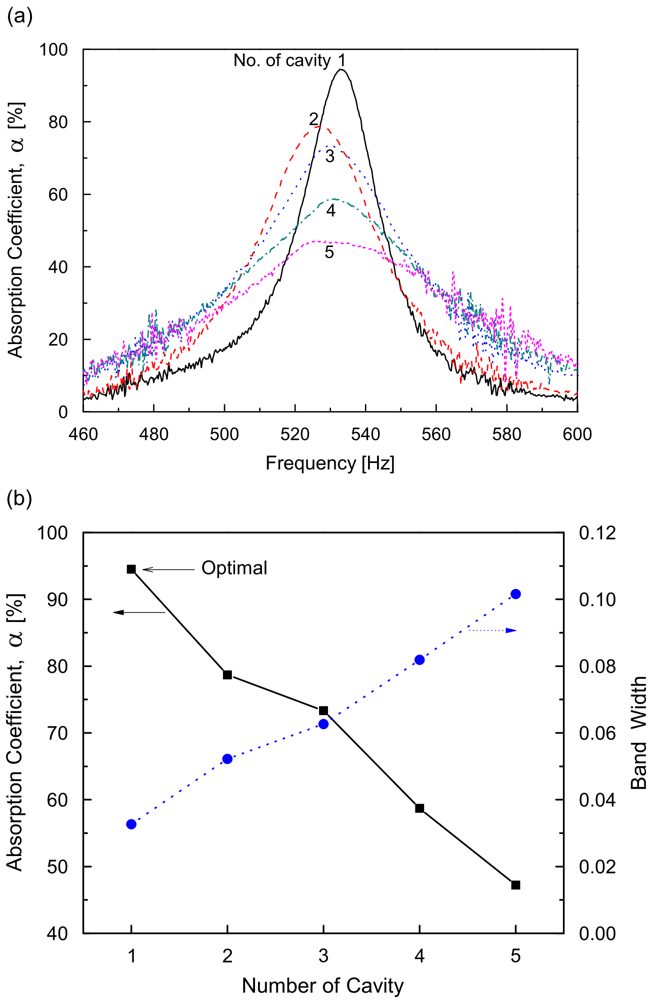

Fig. 16. Absorption coefficients for various number of cavities (Case A).

conductance can be evaluated from the acoustic resistance. Fig. 18 shows these acoustic conductance values in tuned frequency in Cases A and E with respect to the open area ratio. As can be seen in this figure, the acoustic conductance increased with the number of cavities. In particular, the acoustic conductance was monotonically proportional to the open area ratio.

The absolute acoustic conductance values of Case E were much smaller than those of Case A. This is due to the magnitude difference of orifice area which is thought to be an important parameter in designing the acoustic cavities. It should be taken into account that the acoustic power loss induced by cavities in Case E would not be sufficient although the absorption coefficient can be approximately 100 pecrcent by installing three cavities. On the contrary, sufficient damping characteristics are expected to be obtained in Case A which has relatively large orifice area compared with Case E. Recommended values of orifice area with respect to the chamber cross-sectional area, i.e. open area ratio can be understood in this sense [16,17,19,20]. But these values are different from each other and are case by case. Moreover the value of the area ratio used in FASTRAC combustor by Christensen and Nesman [20] was much below the recommended value (20 pecrcent). Therefore, they should be determined from these acoustic tests and further confirmed by hot-firing tests through logical procedures.

Besides the geometrical effects, the fluid properties inside the orifice and cavity are very important to determine the sonic velocity, because actual acoustic cavities operate generally under combustion condition. On the whole, the direct measurement of hot-gas properties inside the orifice and cavity is known to be very difficult, especially in case of aviation fuels and kerosene which are complex higher hydrocarbon fuels. Moreover, the broad tuning of cavities is often necessary to suppress combustion instabilities in the transient period of operation as well as steady operating condition, which may result in the change or shift of the resonant frequencies to harmful modes. Consequently the increase of acoustic power loss or bandwidth would be important in these cases. Also, the effects of acoustic cavities having different configurations for the stabilization of multimodes would be a very interesting topics and can be considered for future research.

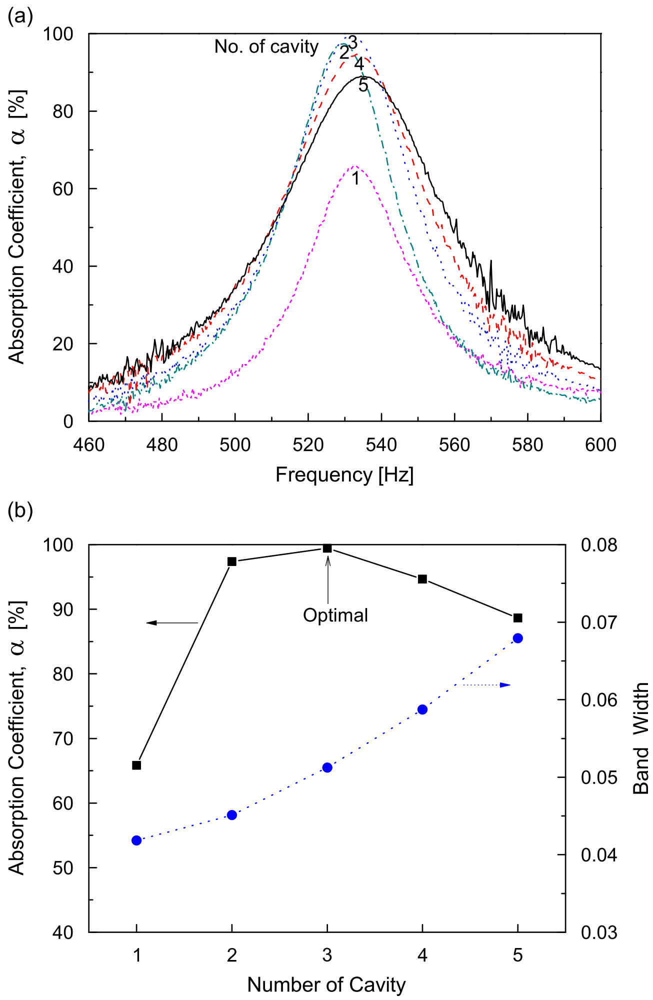

Fig. 17. Absorption coefficients for various number of cavities (Case E).

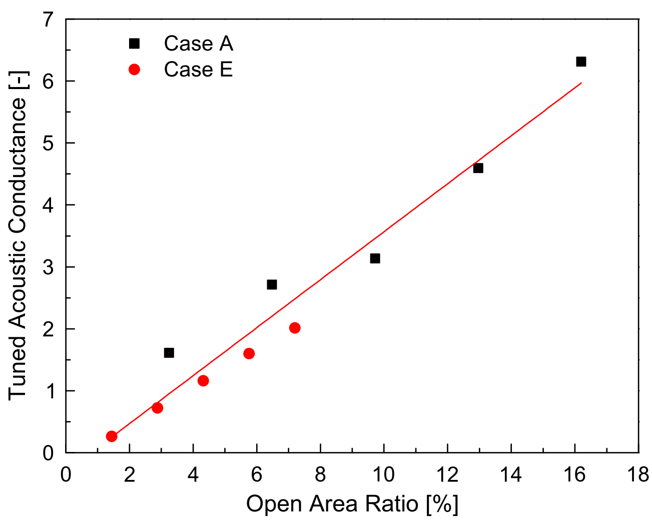

Fig. 18. Acoustic conductance showing damped acoustic power.

5. Conclusion

Damping characteristics of acoustic cavity were investigated according to geometries and number of acoustic cavities, for the use of a suppression device of high frequency combustion instabilities in liquid rocket engines or gas-turbine engines. Considering only geometric effects, the result showed that larger orifice area and shorter orifice length were desirable to increase damping capacity at the harmful resonant frequency, but it must be considered within the geometric limitations of cavity volumes that can be installed in propulsive combustors. The number effect of acoustic cavities on damping capacities showed that increasing the number of cavities improved the damping capacity as a matter of course, but the number had to be restrained by geometric limitations and possible multi-harmful resonant frequencies. Therefore, impedance tube methods were used to analyze damping efficiencies quantitatively and the results showed that there was an optimal number of cavities for a certain geometry of cavity considering damping efficiency. It was also shown by acoustic conductance which means acoustic power loss factor. Finally, an optimization procedure of the geometry and the number of acoustic cavities by an atmospheric experiment has been shown in this study. Therefore, the preliminary choice for optimal cavities could be obtained from this acoustic test procedure with additional information about hot gas compositions and temperature in the cavities.

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