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USE OF PARALLEL MICROPERFORATED PANEL SUBABSORBERS FOR NOISE CONTROL IN DUCTS

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Abstract

A sound absorber with two or three different microperforated panel subabsorbers connected in parallel, is used to achieve a broader absorption band than an ordinary microperforated panel absorber in the control of noise due to plane waves in a duct. The absorber works in a way that the poor absorptions of the subabsorbers outside their absorption bands due to their low acoustical resistances and large acoustical reactances, are compensated by an extra term of acoustical impedance. This impedance accounts for the interaction/coupling between the reflected sound fields of the individual subabsorbers. As a result, the overall resistance of the sound absorber is around twice of the characteristic impedance of air and the overall reactance is small over a large frequency range. When two different and properly tuned subabsorbers are used, the normal absorption coefficient of the sound absorber can exceed 0.5 over three to four octave bands. When three different subabsorbers are used, the absorption band with coefficient exceeding 0.5, can include up to between five and six octave bands. Unlike series subabsorbers that have been investigated, which are difficult to be tuned because their impedances are dependent on each other, each parallel subabsorber can be easily and independently tuned to achieve the best absorption of the sound absorber. This paper provides an insight into the behavior of parallel subabsorbers for noise control in ducts.

INTRODUCTION

The sound absorption theory for an ordinary microperforated panel absorber (a single panel absorber) without the effect of the panel vibration has been well developed [1]. When the absorber is optimized, its absorption band where the normal absorption coefficient exceeds 0.5 can fully cover two octaves and partially include two other octaves at both ends of the band [1]. The effects of vibration of the panel with [2] and without [3] bending stiffness have been studied, but in most cases, the panel vibration deteriorates the absorber performance when the absorption bandwidth is reduced. A sound absorber with two subabsorbers connected in series (one subabsorber is in front of the other) have also been proposed to broaden the absorption band [4]. For the absorption coefficient that is 0.5 and above, the band can fully cover three octaves and partially include two octaves at its two ends. However, the acoustical impedances of the series subabsorbers are dependent on each other [4] and changing one will alter the other. Thus, the tuning of the subabsorbers is difficult. Also, the relationship between the absorption properties of the sound absorber and the impedances of the subabsorbers [4] is too complicated for one to understand how those impedances affect the resistance, reactance and absorption coefficient of the absorber. In this paper, a sound absorber with two or three subabsorbers connected in parallel (all subabsorbers facing the incident sound field) is studied. It provides an insight into the behavior of the absorber in terms of the way the impedances of the subabsorbers in the parallel arrangement contribute to the absorption of the absorber and how the absorption band of the absorber is improved.

REVIEW OF SINGLE MICROPERFORATED PANEL ABSORBER

The specific acoustical resistance and reactance of the holes on a microperforated panel are given by [1–4]:

$$\zeta_h^r = 32\mu l_h\left(\sqrt{1+d^2\rho_0\omega/128\mu} + d^2\sqrt{2\rho_0\omega/\mu}\Big/16 l_h\right)\Big/\sigma\rho_0 c_0 d^2\,, \tag{1}$$

$$\zeta_h^i = \omega\left(l_h + 0.85d + l_h\Big/\sqrt{9+d^2\rho_0\omega/8\mu}\right)\Big/\sigma c_0\,. \tag{2}$$

ρ_0 and μ are the density and viscocity of air, c_0 is the speed of sound in air, ω is the excitation frequency, and l_h and d are the depth and diameter of the holes. The perforation ratio is given by $\sigma = N\pi d^2/4A$, where N is the number of holes and A is the panel surface area. If the air cavity behind the panel has a depth of D, then the absorber will have an extra impedance of $\zeta_c = -j\cot(\omega D/c_0)$. Considering that the panel is lightweight and very thin (<1 mm in typical microperforated panel sheets) where its bending stiffness is not significant, the specific acoustical impedance due to the panel vibration is $\zeta_p = j\omega m_p/\rho_0 c_0 A$ (m_p is the panel mass) [3,4]. By writing $\zeta_h = \zeta_h^r + j\zeta_h^i$, the specific acoustical impedance of the absorber is given by [3,4]

$$\zeta_A = \zeta_c + \zeta_h\zeta_p\big/(\zeta_h + \zeta_p). \tag{3}$$

By writing $\zeta_A = \zeta_A^r + j\zeta_A^i$, the normal incident sound absorption coefficient is

$$\alpha_A = 4\zeta_A^r\big/[(\zeta_A^r + 1)^2 + (\zeta_A^i)^2]. \tag{4}$$

The peak absorption coefficient and the ratio of upper to lower frequencies (f_u and f_l) of the half-absorption band are [1]

$$\alpha_{A,pk} = 4\zeta_A^r\big/(\zeta_A^r + 1)^2\,, \tag{5}$$

$$R_\alpha = f_u/f_l = [\pi\big/\cot^{-1}(\zeta_A^r + 1)] - 1. \tag{6}$$

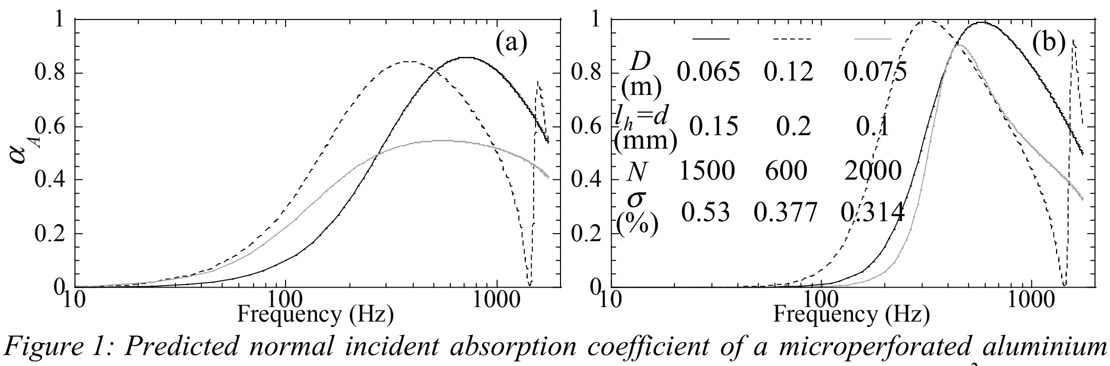


Figure 1: Predicted normal incident absorption coefficient of a microperforated aluminium panel absorber (a) without and (b) with the panel vibration. A=0.005 m².

From Eq. (5), $\alpha_{A,pk}$ increases with ζ_A^r when ζ_A^r<1, but decreases with ζ_A^r when ζ_A^r>1. Also, from Eq. (6), the absorption bandwidth (or R_α) can be shown to increase with ζ_A^r. Knowing these variations of R_α and $\alpha_{A,pk}$, the absorber can be optimized for ζ_A^r as in Ref. 1 to give α_A> 0.5 over two octaves in full and two octaves in partial when the panel vibration is excluded [e.g., the two α_A's in Fig. 1(a) where the absorption bands include the 125-1000 Hz and 250-2000 Hz octaves]. When the absorption band includes more octave bands [e.g., the 125-2000 Hz octaves as in Fig. 1(a)], α_A<0.5 at most frequencies in the band. The absorption bandwidth is also significantly decreased when the panel vibration is included [see Fig. 1(b)] because the vibration can reduce ζ_A^r to about one or lower around the absorption peak [3]. So, a different construction of the absorber is necessary to achieve a broader absorpion band with α_A>0.5.

PARALLEL MICROPERFORATED PANEL SUBABSORBERS

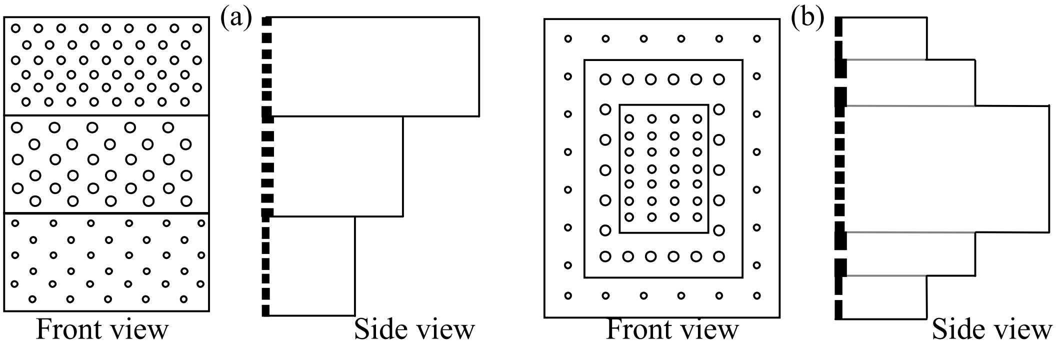


Figure 2: Some examples of configuration of a sound absorber with three parallel subabsorbers. (a) Staggered configuration and (b) concentric configuration.

Figure 2 shows some examples of sound absorber configuration with parallel microperforated panel subabsorbers (the case of three subabsorbers is illustrated). When an absorber with two subabsorbers is located at an end of a one-dimensional duct and is subject to a plane wave excitation, the parallel-circuit rule [3,4] gives $A_T/\zeta_T=A_1/\zeta_1+A_2/\zeta_2$ where $A_T=A_1+A_2$ and ζ_T is the specific acoustical impedance of the absorber. For $A_1=A_2=A_T/2$, $\zeta_T=2/(1/\zeta_1+1/\zeta_2)$. As the sound field in front of the absorber is a combination of the reflected sound fields of the two subabsorbers and the incident sound field, an interaction/coupling between the reflected sound fields exists (the impedances of the two reflected sound fields have a phase relationship). Hence, if $\zeta_1=\zeta_1^r+j\zeta_1^i$, $\zeta_2=\zeta_2^r+j\zeta_2^i$ and $\zeta_T=\zeta_T^r+j\zeta_T^i$, then $\zeta_T^r=\zeta_1^r+\zeta_2^r+\zeta_C^r$ and $\zeta_T^i=\zeta_1^i+\zeta_2^i+\zeta_C^i$ where $\zeta_C=\zeta_C^r+j\zeta_C^i$ is the specific acoustical impedance defined to represent the effects of the coupling:

$$\zeta_C^r=2\beta_{r2}\big/(\beta_{r2}^2+\beta_{i2}^2)-(\zeta_1^r+\zeta_2^r), \tag{7}$$

$$\zeta_C^i=2\beta_{i2}\big/(\beta_{r2}^2+\beta_{i2}^2)-(\zeta_1^i+\zeta_2^i), \tag{8}$$

$$\beta_{r2}=\zeta_1^r\big/|\zeta_1|^2+\zeta_2^r\big/|\zeta_2|^2, \tag{9}$$

$$\beta_{i2}=\zeta_1^i\big/|\zeta_1|^2+\zeta_2^i\big/|\zeta_2|^2. \tag{10}$$

When an absorber with three subabsorbers is used, the parallel-circuit rule [3,4] gives $A_T/\zeta_T=A_1/\zeta_1+A_2/\zeta_2+A_3/\zeta_3$ where $A_T=A_1+A_2+A_3$. For $A_1=A_2=A_3=A_T/3$, $\zeta_T=3/(1/\zeta_1+1/\zeta_2+1/\zeta_3)$. If $\zeta_3=\zeta_3^r+j\zeta_3^i$, then $\zeta_T^r=\zeta_1^r+\zeta_2^r+\zeta_3^r+\zeta_C^r$ and $\zeta_T^i=\zeta_1^i+\zeta_2^i+\zeta_3^i+\zeta_C^i$. Here, ζ_C is defined to represent the effects of the coupling between the three reflected sound fields of the individual subabsorbers, where ζ_C^r and ζ_C^i are given by

$$\zeta_C^r = 3\beta_{r3} \big/ (\beta_{r3}^2 + \beta_{i3}^2) - (\zeta_1^r + \zeta_2^r + \zeta_3^r), \tag{11}$$
$$\zeta_C^i = 3\beta_{i3} \big/ (\beta_{r3}^2 + \beta_{i3}^2) - (\zeta_1^i + \zeta_2^i + \zeta_3^i), \tag{12}$$
$$\beta_{r3} = \zeta_1^r \big/ |\zeta_1|^2 + \zeta_2^r \big/ |\zeta_2|^2 + \zeta_3^r \big/ |\zeta_3|^2, \tag{13}$$
$$\beta_{i3} = \zeta_1^i \big/ |\zeta_1|^2 + \zeta_2^i \big/ |\zeta_2|^2 + \zeta_3^i \big/ |\zeta_3|^2. \tag{14}$$

The normal absorption coefficient of the absorber with either two or three subabsorbers is

$$\alpha_T = 4\zeta_T^r \big/ [(\zeta_T^r + 1)^2 + (\zeta_T^i)^2]. \tag{15}$$

By substituting the above expressions of ζ_T^r and ζ_T^i for the case of two subabsorbers into Eq. (15), $\alpha_T = \alpha_{12} + \alpha_C$ where $\alpha_{12} = \alpha_1 A_1/A_T + \alpha_2 A_2/A_T$. α_1 and α_2 are the absorption coefficients of the subabsorbers which can be calculated by Eq. (4) given ζ_1 and ζ_2. α_{12} is the area-weighted average of α_1 and α_2 when the two reflected sound fields do not interfere with each other as when the subabsorbers act independently. α_C is the modification term that accounts for the coupling between the sound fields when the subabsorbers act simultaneously. It is given by

$$\begin{aligned}\alpha_C = {} & \frac{4A_1\zeta_1^r[2\zeta_1^r(1 - A_1/A_T) + 1 - |\zeta_1|^2 \ (2A_2\zeta_2^r / A_T \ |\zeta_2|^2 + G_{r2}^2 + G_{i2}^2)]}{A_T \ |\zeta_1|^2 \ (|\zeta_1|^2 + 2\zeta_1^r + 1)(1 + 2G_{r2} + G_{r2}^2 + G_{i2}^2)} + \\ & \frac{4A_2\zeta_2^r[2\zeta_2^r(1 - A_2/A_T) + 1 - |\zeta_2|^2 \ (2A_1\zeta_1^r / A_T \ |\zeta_1|^2 + G_{r2}^2 + G_{i2}^2)]}{A_T \ |\zeta_2|^2 \ (|\zeta_2|^2 + 2\zeta_2^r + 1)(1 + 2G_{r2} + G_{r2}^2 + G_{i2}^2)},\end{aligned} \tag{16}$$

where $G_{r2} = A_1\zeta_1^r / A_T \ |\zeta_1|^2 + A_2\zeta_2^r / A_T \ |\zeta_2|^2$ and $G_{i2} = A_1\zeta_1^i / A_T \ |\zeta_1|^2 + A_2\zeta_2^i / A_T \ |\zeta_2|^2$. For the case of three subabsorbers, α_T also has a term that accounts for the coupling between their reflected sound fields, but will not be presented here because its expression is too large.

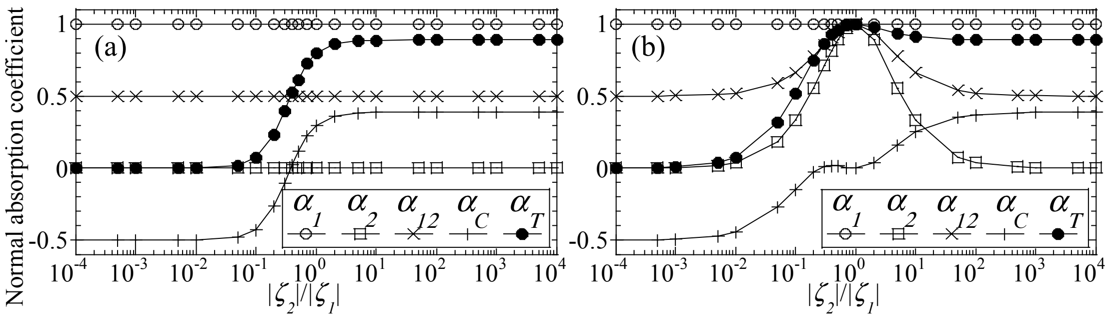


Figure 3: Predicted absorption coefficients when ζ_1=1.0 and ζ_2 is varied (a) from j0.0001 to j10000 and (b) from 0.0001 to 10000, for a sound absorber with two subabsorbers.

For the case of two subabsorbers, their reflection coefficients are $r_1=(\zeta_1-1)/(\zeta_1+1)$ and $r_2=(\zeta_2-1)/(\zeta_2+1)$. When $|\zeta_1|>1$ and $|\zeta_2|>1$, the real parts of r_1 and r_2 are positive, and the reflected sound fields of the subabsorbers couple in-phase, which generates an extra absorption ($\alpha_C>0$). When $|\zeta_2|<1$ but close to 1, the real part of r_2 is negative and has a small magnitude. So, the coupling is dominated by the reflected sound field of the first subabsorber but becomes out-of-phase. Thus, $\alpha_C>0$ but is less than before due to the slight cancellation effect between the reflected sound fields. This cancellation effect increases with the decrease of $|\zeta_2|$ until $\alpha_C=0$. When $|\zeta_2|<<1$, the real part of r_2 is close to -1 and the out-of-phase coupling is now dominated by the reflected sound field of the second subabsorber and an increased reflection ($\alpha_C<0$) is produced. As can be seen from Fig. 3 that shows an example where ζ_2 is varied for a fixed ζ_1, $\alpha_C\geq0$ when $|\zeta_2|/|\zeta_1|$ is close to 1 or greater, and $\alpha_C<0$ when $|\zeta_2|/|\zeta_1|<<1$. However, it can be shown that $|\zeta_2|<<1$ cannot be obtained from the properties of practical subabsorbers. So, there is always an extra absorption due to the coupling where $\alpha_C\geq0$ and $\alpha_T\geq\alpha_{12}$. Thus, by tuning the subabsorbers such that their absorption peaks are away but not far from each other, α_C can increase α_T at frequencies where the absorption coefficient of one subabsorber is high and the other is low. This explanation is also applicable to the case of three subabsorbers.

RESULTS AND DISCUSSION

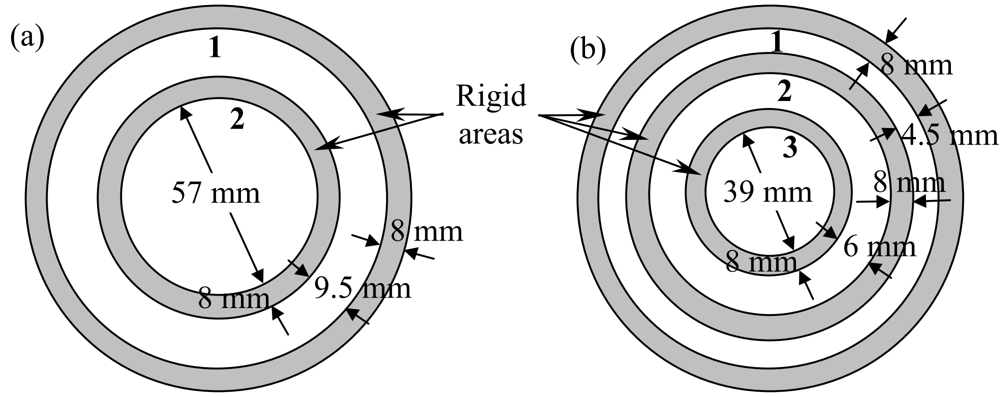
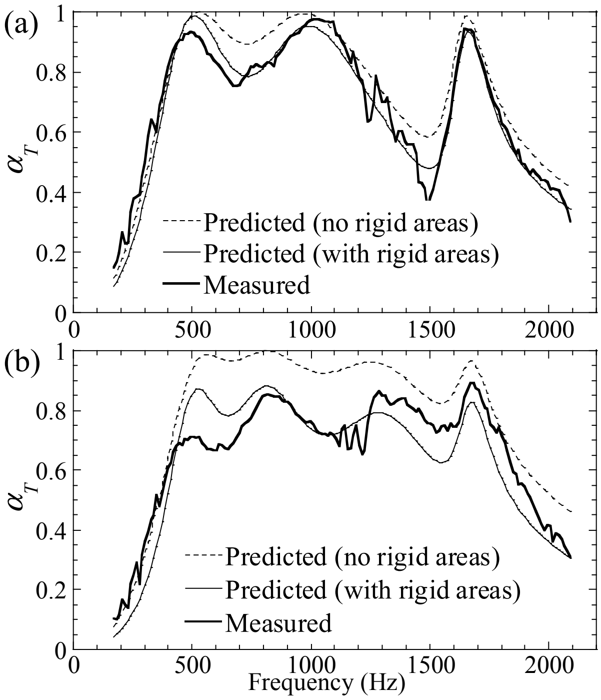


Figure 4: Dimensions of the steel sound absorbers with concentric parallel subabsorbers used in the experiment. (a) Two subabsorbers and (b) three subabsorbers.

Measurements of normal absorption coefficient are carried out for two sound absorbers with two and three microperforated panel subabsorbers, respectively, to first validate the above theoretical model. The experiment is conducted in a cylindrical impedance tube of a length of 1.9 m and diameter of 94 mm. The dimensions and configuration of the absorbers are shown in Fig. 4. An 8-mm steel used for the absorber is sufficiently thick that the cylindrical shells negligibly couple with the air cavities of the subabsorbers and the cavities do not couple with each other via the shells. A commercially available microperforated steel sheet of thickness, l_p=0.3 mm, is used for the facing of the subabsorbers. The holes have been indented that their depths are much smaller than 0.3 mm. Due to imperfections in manufacturing, the depths and diameters of the holes are slightly different, and their mean values are 0.08 mm and 0.31 mm. Other dimensions of the subabsorbers are summarized in Fig. 5.

Figure 5(a) shows that when the contribution of the rigid areas on the subabsorbers (see also Fig. 4) are not included in the theoretical model, α_T is overpredicted in the case of the two subabsorbers. The overprediction is larger in the case of the three subabsorbers [see Fig. 5(b)]. By including the rigid areas, $A_T=A_1+A_2+A_{r(2)}$ for the two subabsorbers and $A_T=A_1+A_2+A_3+A_{r(3)}$ for the three subabsorbers, where $A_{r(2)}$ and $A_{r(3)}$ are the total surface areas of all the rigid areas for the two cases, respectively. From the dimensions of the rigid areas in Fig. 4, $A_{r(2)}$ and $A_{r(3)}$ can be calculated. As the rigid areas have infinite impedance, ζ_C^r and ζ_C^i can be rederived and have a multiplication factor of 2.768 instead of 2 in Eqs. (7) and (8) for the two subabsorbers, and 5.811 instead of 3 in Eqs. (11) and (12) for the three subabsorbers (i.e., A_T is 2.768 and 5.811 times the panel surface area of each subabsorber). Hence, the rigid areas increase ζ_T by about 38% [i.e., (2.768−2)/2] for the two subabsorbers, and by about 94% [i.e., (5.811−3)/3] for the three subabsorbers, where α_T will decrease. This is the reason why the overprediction in the latter is larger than the former when the rigid areas are excluded in the prediction of α_T. It is obvious from Fig. 5 that the predicted and measured results agree fairly well after the contribution of the rigid areas is included in the theoretical model. So, the rigid areas are a major source of deterioration of the absorptivity of the sound absorbers although thick shells are required in this kind of absorber construction to isolate the couplings between the shells and the air cavities of the subabsorbers as well as between the air cavities.



Subabsorber:	1	2
D (m):	0.039	0.12
l_h (mm):	0.08	0.08
l_p (mm):	0.3	0.3
N:	200	206
d (mm):	0.31	0.31
A (m^2):	0.0025	0.0025
σ(%):	0.604	0.622

Subabsorber:	1	2	3
D (m):	0.029	0.063	0.12
l_h (mm):	0.08	0.08	0.08
l_p (mm):	0.3	0.3	0.3
N:	108	112	107
d (mm):	0.31	0.31	0.31
A (m^2):	0.0011	0.0011	0.0011
σ(%):	0.741	0.768	0.734

Figure 5: Predicted and experimentally measured normal incident absorption coefficients of the steel sound absorbers with (a) two subabsorbers and (b) three subabsorbers.

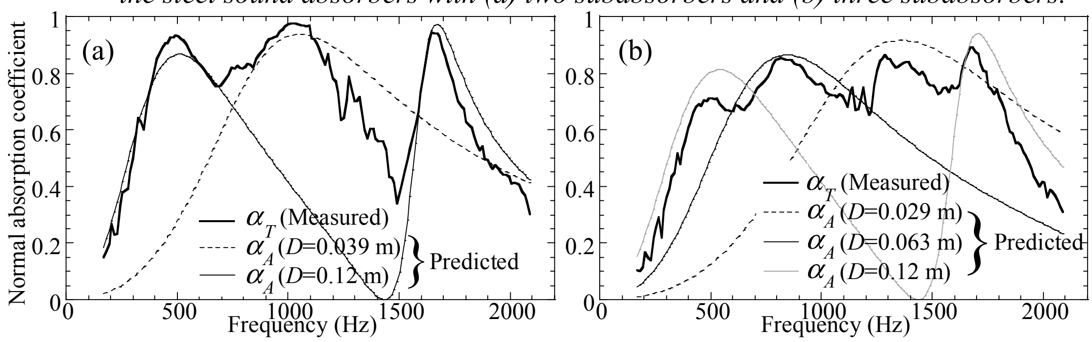


Figure 6: Experimentally measured absorption coefficient of the sound absorbers with (a) two subabsorbers and (b) three subabsorbers. Predicted absorption coefficient is for ordinary single absorbers with the same l_h, l_p and d, and (a) N=406(200+206), A=0.005 m^2, σ=0.613%, and (b) N=327(108+112+107), A=0.0033 m^2, σ=0.745%.

In Fig. 6, the measured α_T is compared to the predicted α_A of an ordinary single absorber for $A=A_T$, and the same l_h, l_p and d. The ordinary absorber is successively assigned the cavity depth of each subabsorber and is considered to have the total number of holes of the subabsorbers. The second peak for the ordinary absorber for D=0.12 m is due to the second acoustic mode in the air cavity [i.e., the $\cot(\omega D/c_0)$ term repeats itself], and it also exists in α_T. It can be seen that around the peaks, the envelopes of α_T are nearly similar to the envelopes of α_A. The absorptions of the two absorbers with the parallel subabsorbers can thus be considered as being equivalent to those when their subabsorbers or the ordinary absorbers act simultaneously. So, α_T is still high even at frequencies away from the absorption peaks where the subabsorbers have poor absorptions or away from the peaks of α_A where the ordinary absorbers have poor absorptions. As a result, it is obvious from Fig. 6 that the absorption bands of α_T for α_T>0.5 are much broader than those of α_A for α_A>0.5. It can be shown that these absorption bands of α_T are also much broader than those of the subabsorbers. This observation also implies that for the absorption band of an absorber with parallel subabsorbers to be broader than that of any of its subabsorbers or an ordinary single absorber with the same dimensions and physical properties of the facings, the absorption peaks of the subabsorbers have to be located away from each other. Given this requirement, each subabsorber can be easily and independently tuned in terms of D, l_h, N and d to obtain the desired α_T and absorption bandwidth since ζ_1 and ζ_2 (the case of two subabsorbers), or ζ_1, ζ_2 and ζ_3 (the case of three subabsorbers), do not affect each other.

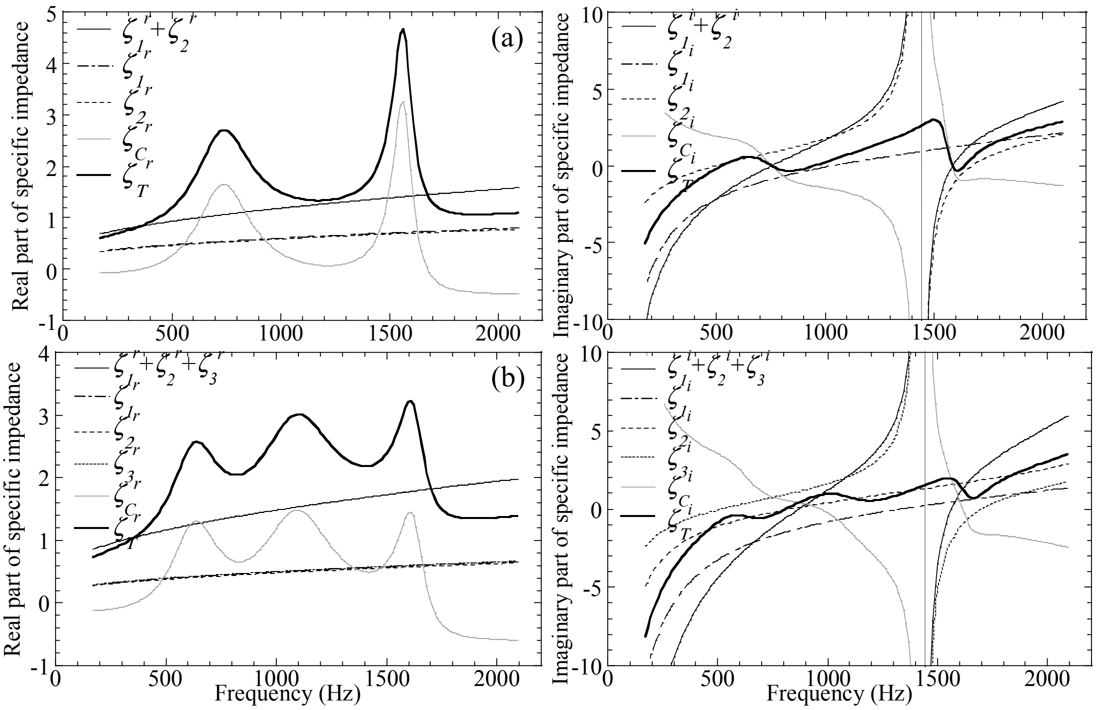
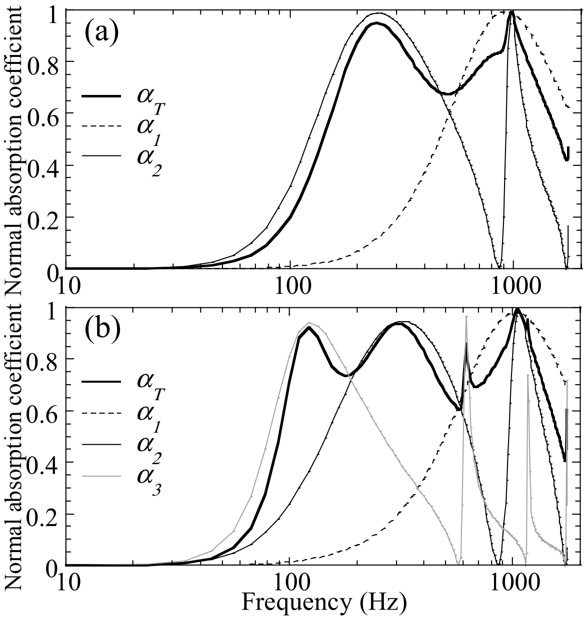


Figure 7: Predicted real and imaginary parts of the specific acoustical impedance of the steel sound absorbers with (a) two subabsorbers and (b) three subabsorbers. D, l_h, l_p, N, d, A and σ are given in Fig. 5.

The impedances of the subabsorbers and the impedance defined for the coupling between their reflected sound fields, are studied to show how the parallel arrangement improves the poor absorptions at frequencies away from the absorption peaks of the subabsorbers. From the preceding section, ζ_T is determined by two terms. For two subabsorbers, ζ_T^r depends on $(\zeta_1^r+\zeta_2^r)$ and ζ_C^r, and ζ_T^i depends on $(\zeta_1^i+\zeta_2^i)$ and ζ_C^i. For three subabsorbers, ζ_T^r depends on $(\zeta_1^r+\zeta_2^r+\zeta_3^r)$ and ζ_C^r, and ζ_T^i depends on $(\zeta_1^i+\zeta_2^i+\zeta_3^i)$ and ζ_C^i. Figure 7 shows that ζ_1^r, ζ_2^r and ζ_3^r are below one, and $|\zeta_1^i|$, $|\zeta_2^i|$ and $|\zeta_3^i|$ are large throughout the frequency range indicated. So, the subabsorbers have narrow absorption bands. Although $(\zeta_1^r+\zeta_2^r)$ [Fig. 7(a)] and $(\zeta_1^r+\zeta_2^r+\zeta_3^r)$ [Fig. 7(b)] are slightly above one, $(\zeta_1^i+\zeta_2^i)$ [Fig. 7(a)] and $(\zeta_1^i+\zeta_2^i+\zeta_3^i)$ [Fig. 7(b)] are still large. However, since ζ_C^r is only close to one or below, and ζ_C^i has a nearly opposite effect to $(\zeta_1^i+\zeta_2^i)$ [Fig. 7(a)] and $(\zeta_1^i+\zeta_2^i+\zeta_3^i)$ [Fig. 7(b)], the presence of ζ_C due to the parallel arrangement, increases slightly the resistance of the absorber but reduces significantly its reactance throughout the frequency range shown (correspond to α_C>0). So, ζ_T^r is around 2 but ζ_T^i is close to 0, which result in α_T>0.5 at most frequencies from Eq. (15) and thus, the broader absorption bands of α_T relative to those of the subabsorbers as in Fig. 6.



Subabsorber:	1	2
D (m):	0.04	0.2
l_h (mm):	0.2	0.3
N:	480	160
d (mm):	0.2	0.3
A (m^2):	0.0025	0.0025
σ (%):	0.603	0.452

Subabsorber:	1	2	3
D (m):	0.05	0.2	0.3
l_h (mm):	0.2	0.25	0.4
N:	580	300	20
d (mm):	0.2	0.25	0.4
A (m^2):	0.0017	0.0017	0.0017
σ (%):	1.09	0.884	0.151

Figure 8: Predicted absorption coefficient of a microperforated aluminium panel absorber with (a) two and (b) three subabsorbers, when the facing has no rigid areas.

Figure 8 shows an example of a sound absorber whose subabsorbers have been properly tuned in terms of D, l_h, N and d. It can be seen that when two subabsorbers are used, α_T>0.5 over three to four octaves (250 Hz, 500 Hz and 1000 Hz octaves in full, and 125 Hz and 2000 Hz octaves in partial). When three subabsorbers are used, the absorption band with α_T>0.5, can include up to between five and six octaves (125 Hz, 250 Hz, 500 Hz and 1000 Hz octaves in full, and 63 Hz and 2000 Hz octaves in partial).

CONCLUSIONS

The behavior of a sound absorber with parallel subabsorbers has been studied. The absorber works in a way that the poor absorptions of the subabsorbers outside their absorption bands, are compensated by an extra term of impedance that accounts for the coupling between the reflected sound fields of the individual subabsorbers. As a result, a broader absorption band than that of an ordinary absorber can be achieved. However, the rigid areas on the facing of the absorber are found to reduce its absorption coefficient although its absorption bandwidth is only slightly affected. This factor should be considered in future designs of such absorber.

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