Hybrid Passive Noise Control on a Plate Silencer with Micro-Perforations

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Abstract
A theoretical study has been carried out for investigating the use of micro-perforation in the plate silencer. A plate silencer consists of two side-branch, rectangular cavities covered by plates. Previous study showed that if properly selecting the parameters of the plate, there would be three resonant peaks in the low to medium frequency range and the transmission loss (TL) between adjacent peaks remained above 10 dB. Modal analysis pointed out that the first two vibration modes of plate *in vacuo* play the important roles in reflecting sound, and the peaks of TL are contributed by different combinations of the first two modes. Besides, an optimal performance requires a high bending stiffness of the plate which helps lifting the dip between 2nd and 3rd peak of the TL to achieve a broad bandwidth. This study introduces absorption by adding micro-perforation into the plate. By combining the reflection and absorption, this research aims at achieving a wider stopband. Theoretical model has been established and the vibro-acoustic coupling mechanism between the micro-perforation plate, the duct and the cavity has been investigated. It is found that the micro-perforation is beneficial for releasing the bending stiffness and widening the stop band by 20%.

Key words: Plate silencer, Micro-perforation, Modal analysis

1. Introduction

Low frequency noise is difficult to tackle by traditional passive methods and it is still a technical challenge. Active control method performs very well for controlling low frequency noise. However, it requires additional sensors and actuators, therefore the system is sophisticated. Also there are issues of cost and reliability. Passive noise control is a more reliable and attractive method for users. For example, a layer of porous material lined on the duct wall which is so called duct-lining is widely used to deal with the medium to high frequency noise(1) in the air conditioning and ventilation system. However, it cannot perform well at low frequencies, also fibrous materials cause the environmental issues such as accumulation of dusts in the pores of the porous material and bacteria breeding.

Aiming for a passive method to control broadband noise from low to medium frequencies effectively, the concept of using flexible membrane to reflect low-frequency noise was introduced(2),(3). This device is so called drum-like silencer(3). It is composed of two tensioned membranes covered with two rectangular cavities. Through the theoretical investigation of the coupling between structure and acoustic system, desirable transmission loss with wide stopband can be achieved on condition that the membrane should be very light and applied moderately high tension. It is because the first two modal vibrations of the membrane are very effective in reflecting sounds. Practically, a tensile machine to stretch the membrane is required to be installed on the wall of duct to apply the tension. However, it is difficult to implement in reality on the site such as fine tuning the tension and the membrane relaxation. Therefore Huang replaced the membrane by a simply supported plate(4). Theoretical investigation has shown that the proposed plate silencer can achieve a wider bandwidth than the drumlike silencer of the same cavity geometry owning to acoustic interference between odd and even *in vacuo* vibration modes of the plate. The plate's natural bending moment is used as the sole structural restoring force. Therefore, the installation of the plate silencer is easier.

Furthermore, the plate silencer with clamped boundary condition was studied(5), which is easier to implement in practice. To achieve desirable performance, the plate should have high bending stiffness and low mass ratio. However, it is difficult to find the plate with this physical property from the existing materials. Wang(6) tried to realize it by making the sandwich structure which is composed of a relatively thick and light foam core adhering to two high-rigidity thin sheets. Wang also proposed the hybrid silencer which used traditional duct lining in the plate silencer(7), and simulation results show that the sound absorption materials lining in the cavities are only effective in high frequency range.

On the other hand, Micro-perforated panel (MPP) is another fiber-free absorber satisfying the requirement for a robust sound absorber used in severe environment. Due to the micro-perforations, they themselves provide enough acoustic resistance and relatively low acoustic mass reactance which are necessary for a wide-band absorber(8). Equivalent circuit was used to describe the performance of the MPP absorber. The vibration of the plate was also excluded in many studies(9), and also the cavity was assumed infinite that the cavity modes were not considered(10).

This project aims to study the performance of the plate silencer with the effect of micro-perforations. Grazing incidence induces the plate to vibrate. Part of the sound is reflected and part of it is absorbed due to the air oscillation in the orifice of the plate with the existence of the pressure difference between two sides of the plate. The main difference from the hybrid silencer introduced by Wang(9) is that the absorption here is provided by MPP instead of inserting sound absorption material inside the cavity. And the perforation parameters can be tuned to achieve the optimal performance. Sec. 2 outlines the theoretical model for the plate silencer with micro-perforations. Modal expansion is used to solve the fully coupled system among the plate, the cavity, the duct and the micro-perforated orifices. The performance will be given in Sec. 3. In Sec. 4, the mechanism of vibration and absorption will be analyzed and optimization process is to be investigated for searching each optimal physical parameter so that best performance will be obtained.

2. Theoretical Analysis

Fig.1 shows the two-dimensional configuration of a plate silencer with the main duct height h^*, and two identical cavities of length L^*. Two micro-perforated plates with the length L^* cover the cavity.

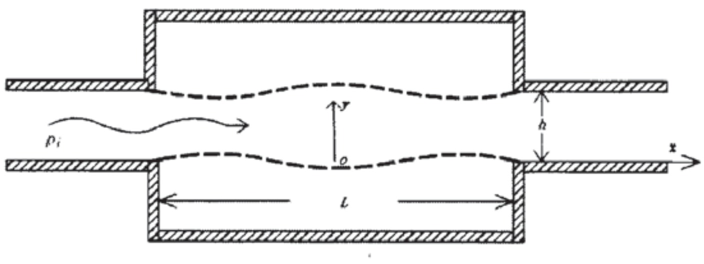

Fig. 1 The configuration of the plate silencer with perforation.

The asterisks denote dimensional variables. In the following presentation, all parameters are normalized by three basic quantities: speed of sound, c_0^*, air density ρ_0^*, and the duct height, h^*. The normalized method for the parameters is given as below:

$$x=\frac{x^*}{h^*}, f=\frac{f^*h^*}{c_0^*}, m=\frac{m^*}{\rho_0^* h^*}, B=\frac{B^*}{(h^*)^3\rho_0^*(c_0^*)^2}, p=\frac{p^*}{\rho_0^*(c_0^*)^2} \quad (1)$$

For a plate silencer with micro-perforation, if the homogenous plate with bending stiffness, B and mass ratio, m , the dynamics of the plate is governed by the following equation:

$$B\frac{\partial^4\eta}{\partial x^4}+m\frac{\partial^2\eta}{\partial t^2}+(p_i+p_{duct}-p_{cav})=0 \quad (2)$$

where p_i is the incident wave, p_{duct} and p_{cav} are the radiation pressure caused by the plate vibration. p_{duct} is the radiation pressure in the duct, p_{cav} is the radiation pressure in the cavity and η denotes the displacement of the plate.

When there are perforations on the plate, the sound pressure acted on the two sides of the MPP causes the air in the orifices to vibrate. Assuming the velocity of the air in one hole is v_o , an average velocity $\overline{v_o}$ is obtained by averaging the discrete air particle velocity over each orifice across the adjacent un-perforated region. The acoustic impedance due to micro-perforation is also averaged over the whole panel so the relative acoustic impedance of the MPP is found as [8]:

$$z=z_{resist}+z_{react}$$

$$\frac{32\mu}{c_0}\frac{t}{\sigma d^2}[\sqrt{1+\frac{x^2}{32}}+\frac{\sqrt{2}xd}{32\cdot t}]+\frac{i\omega\cdot t}{\sigma c_0}[1+\frac{1}{\sqrt{9+\frac{x^2}{2}}}+0.85\frac{d}{t}] \quad (3)$$

t is the thickness of the plate, d is the diameter, σ is the perforation ratio, μ is coefficient of viscosity, and $x=\frac{d}{2}\sqrt{\rho_0\omega/\mu}$.

The fluid loading on the orifices is divided into three parts. One is the grazing incidence and the other two are the radiation pressure of p_{duct} and p_{cav}. The acoustic boundary condition on the MPP[10]:

$$z_{resist}(\overline{v}_o-v_p)+z_{react}\overline{v}_o+p_i+p_{duct}-p_{cav}=0 \quad (4)$$

where v_p is the velocity of the plate, and the viscous force at the air-structure interface in the hole depends on the relative velocity $(\overline{v}_o-v_p)$.

The full coupling between the plate and acoustic waves can be dealt with by combining Eq. (2) and Eq. (4). For a harmonic motion of the plate, $v_p=\frac{\partial\eta}{\partial t}=i\omega\eta$ and the overall velocity of the plate $\overline{v}(x)$ can be described as[8]

$$\overline{v}(x)=v_p(x)+(v_o(x)-v_p(x))\sigma=(1-\sigma)v_p(x)+\overline{v}_o(x) \quad (5)$$

Following the standard Galerkin procedure, $\overline{v}(x)$ is expanded based on the mode shape of a clamped plate, $\varphi_j(\xi)$ [5].

$$\overline{v}(x)=\sum_{j=1}^{\infty}\overline{V}_j\varphi_j(\xi), \quad \xi=\frac{x}{L}+\frac{1}{2} \quad (6)$$

The radiation pressure inside the duct p_{duct} can be expressed as

$$p_{duct}=\sum_{j=0}^{\infty}p_{duct,j}\overline{V}_j \quad (7)$$

Similarly, the cavity pressure p_{cav} becomes

$$p_{cav}=\sum_{j=0}^{\infty}p_{cav,j}\overline{V}_j \quad (8)$$

Where p_{duct} and p_{cav} are referred to Ref.3. Also $p_{duct,j}$ and $p_{cav,j}$ are given there.

By defining

$$Z_{duct} = \int_0^1 2\varphi(\xi) p_{duct} d\xi$$
$$Z_{cav} = \int_0^1 2\varphi(\xi) p_{cav} d\xi \quad (9)$$
$$I_j = \int_0^1 2\varphi(\xi) p_i d\xi$$

Finally, these two equations are turned into the form as:

$$L_j V_{p,j} + (Z_{duct} - Z_{cav})[(1-\sigma)V_{p,j} + \overline{V_{o,j}}] = -I_j$$
$$z_{resist}(\overline{V_{o,j}} - V_{p,j}) + z_{react}\overline{V_{o,j}} + (Z_{duct} - Z_{cav})(V_{p,j} + \overline{V_{o,j}}) = -I_j \quad (10)$$

For simplicity, $L_j = mi\omega + \frac{B}{i\omega}(\frac{j\pi}{L})^4$. $\overline{V_{o,j}}$ and $V_{p,j}$ can be solved through matrix inversion.

The transmitted sound pressure and the transmission loss respectively are

$$p_t = p_{duct}|_{r=0, x\to+\infty} + p_i \quad (11)$$

and

$$TL = -20\log_{10}|p_t| \quad (12)$$

3. Performance

The performance of the plate silencer with and without micro-perforations is shown in Fig. 2. The dotted line shows the optimal performance of the plate silencer with B=0.0698 in case of no perforation. When the bending stiffness of the plate is reduced to 0.057, there is a serious dip in the mid-frequencies so the working bandwidth is reduced (the dashed line). According to the previous parametric study by Huang, bandwidth of transmission loss can be assessed by the stopband which is defined as the frequency $f \in [f_1, f_2]$ in which TL≧10 dB and the cost function is set as the ratio of the band limits f_2/f_1. Fig. 2 also shows the performance of the plate silencer with B=0.057 in case of perforations (solid line). It is found that the stopband is highly improved and the original dip is raised up so that the bandwidth over 10 dB is wider.

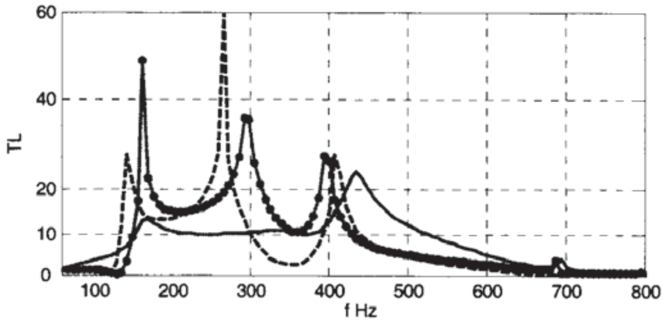

Fig. 2 Transmission loss of the silencer with different B;
Dotted line (-o-): optimal performance without perforation B=0.0698;
Dashed line (- -): performance without perforation B=0.057;
Solid line: performance with perforation, B=0.057, t=4mm,d=0.7mm.

From the three curves, it is obvious that adding perforation brings some changes to the performance of the silencer. For the optimal performance of the plate silencer (B=0.0698) without holes, there are three obvious peaks. The peaks are contributed by the dominance of first two vibration modes of the plate. When the bending stiffness is reduced (B=0.057), although the performance in the low frequencies is improved as the first peak of TL is shifted to the lower frequency, a dip appears in the middle range. Based on this configuration, the perforated plate performs much better as the dip is greatly improved. Meanwhile, the amplitude of the peaks is relatively mild but stopband is generally wider than that found in the optimal performance of the plate without perforations. Generally speaking, perforated plate can enhance the performance in middle frequency range compared with the optimal one with higher bending stiffness and the plate with the supplement of perforations can achieve wider bandwidth by 21% with the reduction of bending stiffness by 18%.

4. Analysis of the performance and Optimization

The micro-perforation brings absorption into the system, and changes vibration of the plate. As a result, the performance of the plate silencer has been changed. This section analyzes the influence to these three aspects. The bending stiffness will be keep unchanged at B=0.057. The mechanisms behind the phenomenon of such fully coupling between the perforated plate and acoustic waves are to be explained in the following section.

4.1 Vibration

The amplitude of the peaks is greatly reduced when there are perforations. This can be explained by the modal amplitude.

$$\overline{V}=\sum_{j=1}^{\infty}\overline{V}_j\varphi_j(\xi) \qquad \overline{V}_j=\int_0^1\overline{V}(\xi)\varphi_j(\xi)d\xi$$
$$p_r=p_{+rad}\big|_{n=0,x\to-\infty}=\frac{L}{2}c_0\int_0^1\overline{V}(x')e^{-ik_0x'}d\xi'=\sum_{j=0}^{N}\overline{V}_jR_j \qquad (13)$$
$$R_j=\int_{-\frac{L}{2}}^{\frac{L}{2}}\varphi_j(\xi')e^{-ik_0\xi'}d\xi'$$

R_j is the complex amplitude of the reflected sound by the induced vibration of the jth mode with unit amplitude. It shows the reflection ability of each mode. Similar to the case of the plate without perforations, the first and the second mode play the most important role in reflection. The first peak of the transmission loss spectrum is mainly contributed by the second mode of the plate vibration while the second peak is due to the first mode. And the third peak is attributed to the combination of the first and second modes of the plate. The peaks are reduced when adding perforation is due to the decrease of the response of the first and second mode. This can be observed in Fig. 3.

By combining Eq.(4) and Eq.(5), $\bar{v}$ and v_p can be connected using the perforation parameters. Δp is the pressure worked on the plate. z_0 and z_{react0} is the impedance of single hole of Maa's approximation.

$$\bar{v}=v_p\left(1-\frac{z_{react0}}{z_0}\cdot\sigma\right)+\frac{\Delta p}{z_0}\cdot\sigma \qquad (14)$$

When there are holes on the plate, the pressure difference on the plate is reduced to some extent due to the air flow in the holes. As a result, the modal velocity is reduced and thus the peaks are smoothed. From Eq.(14), Δp and v_p is reduced, thus $\bar{v}$ is smaller correspondingly.

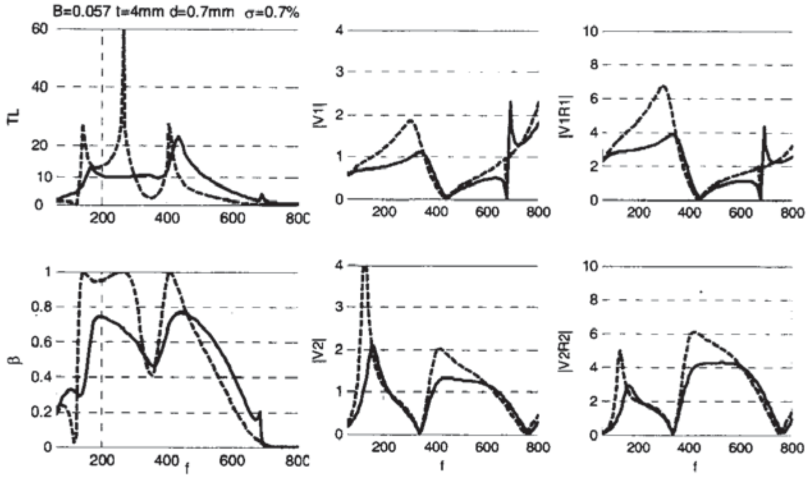

Fig.3 Comparison of the non-perforated plate with the perforated plate.
The dashed line is the non-perforated plate silencer and the solid one is the perforated plate.
The first column is the transmission loss and the reflection coefficient. The second column is the modal velocity of first 3 modes and the third column is for the modal reflection

4.2 Absorption

The optimal performance of the plate silencer without perforations can be obtained when B is high enough, seen from Fig.2 (dotted line). When B is reduced, a serious dip appears (dashed line).And by perforation, the dips can be lifted and stopband is wider (dotted line). Fig.4 shows the absorption coefficient and there are two peaks which are located at 143 Hz and 355 Hz. In the frequency range around 350 Hz, large amount of energy is absorbed (48%) with some of them is reflected (50%). So the dip disappears and the transmission loss is above 10 dB. By ignoring the radiation pressure in the duct, the plate and the cavity form a coupled system. The response of the plate can be studied more simply to explain the effect of perforations by considering backing cavity as a stiffness matrix of the plate [11].

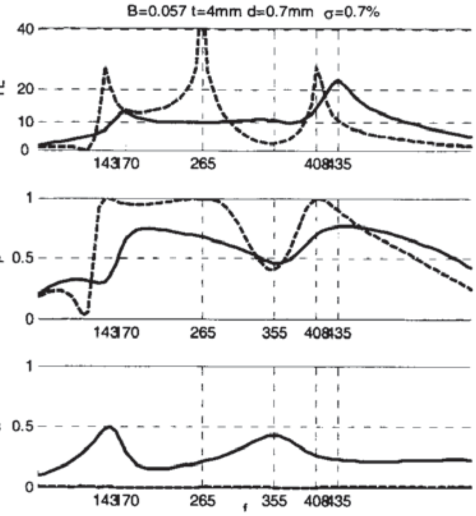

Fig.4 The transmission loss, reflection coefficient and absorption coefficient.
Dashed line: without perforation; Solid line: with perforation.

Table 1. The resonance frequencies of several systems

Mode Number	1	2	3	4
Rigid cavity (dimensionless)	0.1	0.2	0.3	0.4
Beam-cavity coupling system (dimensionless)	0.0467	0.1057	0.1586	0.1629
Dimensional frequencies (Hz)	159	360	539	553

Table 1 gives the resonance frequencies of cavity and the coupled system. By comparing the absorption peaks and the resonance frequencies, the first two resonance frequencies of the plate-cavity coupled system are close to the peaks of absorption. Explanation is that when resonances occur, friction between air flow and the holes is fiercer, so there is more energy transferred into heat. Therefore by making using of the system resonances, micro-perforation introduces absorption into the system. The dip in the middle frequency range occurs due to the reduction of bending stiffness, but is lifted up by the absorption peak.

4.3 Optimization

With the fixed bending stiffness, the diameter of the hole and the perforation ratio are varied in order to obtain the optimal result which has the widest stopband f_2/f_1. Fig.6 shows the variation of the optimal stopband f_2/f_1 with the corresponding diameter and perforation ratio. When the diameter is 0.2 *mm*, the optimal bandwidth appears at σ=0.9%. As shown in Fig.6, when the diameter of the holes is decreased, the perforation ratio required is larger and vice versa. Therefore there will be the compromise between d and σ. It shows that the highest optimal stopband is 3.08 on the condition that d=0.4 *mm* and σ=0.5% when B=0.069. So the search procedure can be repeated like this for each bending stiffness, as a result the optimal result can be gotten when B=0.057 with d=0.7 *mm* and σ=0.7% as shown with solid line in Fig.2.

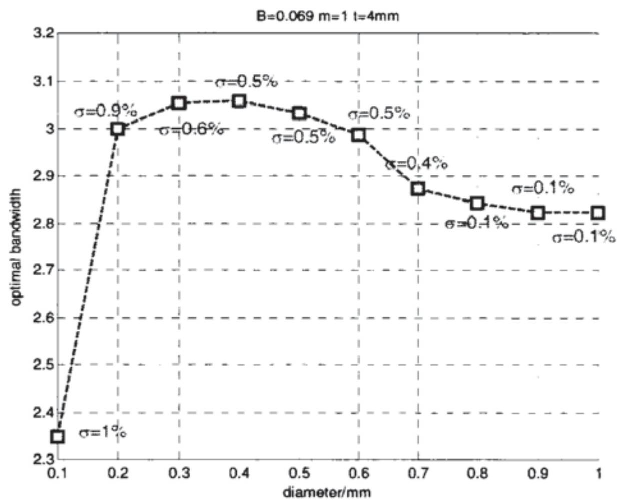

Fig.6 Searching new optimal setting for the new system with perforation.

5. Conclusion

A numerical study has been carried out to investigate the performance of the plate silencer with the effect of the micro-perforations. The micro-perforations can improve the performance of the silencer. This silencer can be regarded as the hybrid with the combination of sound reflection and sound absorption. The modal response of the plate due to the perforations and energy distributions has been used to explain the change of the performance. And optimization process has also been conducted to search for the appropriate diameter and perforation ratio of the plate. Several conclusions are drawn as follows:

1) The first two vibration modes remain very important roles to reflect sound. Perforation will reduce the response of the first two modes and thus the peaks of transmission loss are reduced. Nevertheless, the sound reflection attributed these two first modes are still strong to maintain about 10dB transmission loss.

2) Peaks of the absorption coefficient occur due to the resonance of the system. When

the system is at the resonant frequencies, the air in the holes moves more fiercely than that in other frequencies, so more frictions occur.

3) By combining the sound reflection and absorption, even through the peaks of TL are smoothed, a noise device is gained with a wider working bandwidth which is also the original intension of this study. Also the effect of micro-perforation on the plate can release the bending stiffness; therefore it can release the harsh requirement for the material.

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