ANALYSIS OF VIBRO-ACOUSTIC EFFECT OF MICRO-PERFORATED PANEL SOUND ABSORBER

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ANALYSIS OF VIBRO-ACOUSTIC EFFECT OF MICRO-PERFORATED PANEL SOUND ABSORBER

by

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LIST OF SYMBOLS

$\boldsymbol{A}$	Amplitude of incident sound wave (Pa)
$\boldsymbol{A}_{mn}$	Modal amplitude of mode (m,n) (m)
$\boldsymbol{B}$	Amplitude of reflected sound wave (Pa) / Boltzmann constant
$\boldsymbol{B}_{mn}$	Modal velocity amplitude of mode (m,n) (m/s)
$\boldsymbol{CV}$	Coefficient of root mean square error
$\boldsymbol{D}$	Thickness of the air cavity (m)
$\boldsymbol{D}_e$	Expansion chamber diameter (m)
$\boldsymbol{D}_p$	Plate bending stiffness (N/m)
$\boldsymbol{E}$	Amplitude of incident sound wave at inlet tube (Pa) / Young's modulus (Pa)
$\boldsymbol{E}_1$	Amplitude of incident sound wave at inlet tube with rigid termination (Pa)
$\boldsymbol{E}_2$	Amplitude of incident sound wave at inlet tube with anechoic termination (Pa)
$\boldsymbol{F}$	Amplitude of reflected sound wave at inlet tube (Pa)
Ft_{final}	Function tolerance
$\boldsymbol{G}$	Amplitude of incident sound wave at outlet tube (Pa)

$\boldsymbol{G}_1$	Amplitude of incident sound wave at outlet tube with rigid termination (Pa)
$\boldsymbol{G}_2$	Amplitude of incident sound wave at outlet tube with anechoic termination (Pa)
$\boldsymbol{H}$	Amplitude of reflected sound wave at outlet tube (Pa)
H_{12}, H_{21}	Transfer function between two microphones
$\boldsymbol{H}_1$	Amplitude of reflected sound wave at outlet tube with rigid termination (Pa)
$\boldsymbol{H}_2$	Amplitude of reflected sound wave at outlet tube with anechoic termination (Pa)
$\boldsymbol{L}$	Length (m)
$\boldsymbol{M}$	Specific acoustic reactance of the MPP (kg/m^{2}s) / Number of structural modes for x direction
$\boldsymbol{N}$	Number of structural modes for y direction
$\boldsymbol{OBJ}_1$	Objective function of single layer MPP sound absorber
$\boldsymbol{OBJ}_2$	Objective function of dual layer MPP sound absorber
$\boldsymbol{P}$	Sound pressure (Pa)
$\boldsymbol{P}_D$	Sound pressure at location (*x,y,D*) (Pa)
$\overline{P}_D$	Average sound pressure at location (*x,y,D*) (Pa)
$\boldsymbol{P}_i$	Incident sound pressure (Pa)

$\boldsymbol{P_r}$	Reflected sound pressure (Pa)
$\boldsymbol{P_t}$	Transmitted sound pressure (Pa)
$\boldsymbol{P_1, P_2, P_3, P_4}$	Sound pressure for the microphones (Pa)
$\boldsymbol{R}$	Specific acoustic resistance of the MPP (kg/m^{2}s)
R^2	Correlation coefficient
$\boldsymbol{S}$	Surface area (m^2) / Plate displacement (m)
$\boldsymbol{S_i}$	Silencer inlet tube cross area (m^2)
$\boldsymbol{S_t}$	Silencer outlet tube cross area (m^2)
$\boldsymbol{T}$	Current temperature (K)
$\boldsymbol{T_0}$	Initial temperature (K)
$\boldsymbol{T_{new}}$	New temperature (K)
$\boldsymbol{T_{old}}$	Old temperature (K)
$\boldsymbol{TL_{exp}}$	Transmission loss of simple expansion chamber silencer (dB)
$\boldsymbol{TL_s}$	Transmission loss of a silencer (dB)
$\boldsymbol{U}$	Number of acoustic modes for x direction
$\boldsymbol{V}$	Particle velocity (m/s)
$\overline{V}$	Mean particle velocity (m/s)
$\boldsymbol{V_i}$	Incident particle velocity (m/s)
$\boldsymbol{V_r}$	Reflected particle velocity (m/s)

$\boldsymbol{V_t}$	Transmitted particle velocity (m/s)
$\boldsymbol{W}$	Number of acoustic modes for y direction
$\boldsymbol{W_i}$	Incident sound power (W)
$\boldsymbol{W_t}$	Transmitted sound power (W)
X_l	Random initial solution
X_l'	New random solution
$\boldsymbol{X_m(x)}$	Normal mode shape of (*m,n*) for x direction
$\boldsymbol{Y_n(y)}$	Normal mode shape of (*m,n*) for y direction
$\boldsymbol{Z}$	Specific acoustic impedance (kg/m^{2}s)
$\boldsymbol{Z_1}$	Specific acoustic impedance medium 1 (kg/m^{2}s)
$\boldsymbol{Z_2}$	Specific acoustic impedance medium 2 (kg/m^{2}s)
$\boldsymbol{Z_c}$	Specific acoustic impedance of the air cavity (kg/m^{2}s)
$\boldsymbol{Z_{DMPP}}$	Specific acoustic impedance of dual layer MPP (kg/m^{2}s)
$\boldsymbol{Z_{mpa}}$	Specific acoustic impedance of an MPP sound absorber (kg/m^{2}s)
$\boldsymbol{Z_{MPP}}$	Specific acoustic impedance of the MPP (kg/m^{2}s)
$\boldsymbol{Z_p}$	Impedance of the plate (kg/m^{2}s)
$\boldsymbol{Z_{Total}}$	Total specific acoustic impedance of MPP and air cavity (kg/m^{2}s)
$\boldsymbol{a}$	Length of rectangular cavity (m)
$\boldsymbol{b}$	Width of rectangular cavity (m)

$\boldsymbol{c}$	Speed of sound in the air (m/s)
$\boldsymbol{d}$	Perforation diameter (m)
$\boldsymbol{d}_t$	Inlet or outlet tube diameter (m)
$\boldsymbol{j}$	Unit imaginary number ($\sqrt{-1}$)
$\boldsymbol{k}$	Acoustic wavenumber (rad/m)
$\boldsymbol{l}$	Length of silencer (m)
$\boldsymbol{l}_{12}$	Distance between microphone 1 and microphone 2 (m)
$\boldsymbol{l}_{13}$	Distance between microphone 1 and microphone 3 (m)
$\boldsymbol{m,n}$	Integer number
$\boldsymbol{p}$	Distance between perforations (m)
$\boldsymbol{r}$	Random number
$\boldsymbol{r}_p$	Sound pressure reflection coefficient
$\boldsymbol{t}$	Thickness of panel (m) / Time (s)
$\boldsymbol{v}$	Cavity volume (m^3)
$\boldsymbol{v}_0$	Air particle velocity (m/s)
$\boldsymbol{v}_p$	Plate velocity (m/s)
$\overline{v}_0$	Average air particle velocity (m/s)
$\overline{v}_p$	Average plate velocity (m/s)
$\boldsymbol{x}$	Perforation constant

$\boldsymbol{x}_1,\boldsymbol{x}_2,\boldsymbol{x}_3,\boldsymbol{x}_4$	Distance between microphone and specimen (m)
z_0	Specific acoustic impedance of perforation on the plate (Kg/m^{2}s)
$\bar{z}_0$	Average specific acoustic impedance of perforation on the plate (Kg/m^{2}s)
z_0^{uw}	Specific acoustic impedance of air cavity for (*u,w*) acoustic modes (Kg/m^{2}s)
z_{mn}	Modal impedance of mode (*m,n*) of the plate (kg/m^{2}s)
α	Sound absorption coefficient
α'	Sound absorption coefficient of dual layer MPP sound absorber
$\overline{\alpha}'$	Average sound absorption coefficient of dual layer MPP sound absorber
τ	Transmission loss coefficient
ρ_0	Density of air under standard condition (kg/m^3)
ρ_p	Surface density (kg/m^2)
ω	Angular frequency (rad/s)
ω_{mn}	Resonant frequency at mode (*m,n*) (rad/s)
σ	Perforation ratio
μ	Kinematic viscosity constant of the air (m^2/s)
ξ_{mn}	Modal damping ratio
γ	Cooling rate

ϕ	Acoustic velocity potential function

LIST OF ABBREVIATIONS

ASTM American Society for Testing and Materials

BEM Boundary element method

CAD Computer-aided design

CFD Computational fluid dynamics

DAQ Data acquisition unit

CNC Computer numerical control

FEA Finite element analysis

FEM Finite element method

GA Genetic algorithm

IL Insertion loss

Im(..) Imaginary part of complex number

ISO International Organization for Standardization

MPP Micro-perforated panel

MRI Magnetic resonance imaging

NR Noise reduction

PC Personal computer

Re(..) Real part of complex number

RMSE Root mean square error

SA Simulated annealing

TMM Transfer matrix method

TL Transmission loss

WHO World Health Organization

ANALISIS KESAN GETARAN-AKUSTIK TERHADAP PENYERAP BUNYI PLAT BERLUBANG MIKRO

ABSTRAK

Kajian ini mengenai penggunaan plat berlubang mikro (MPP) dalam penyenyap dan pengoptimuman penyerap bunyi jenis dua lapisan plat berlubang mikro. Plat berlubang mikro sesuai digunakan di dalam penyenyap bagi mencapai nilai kehilangan penghantaran bunyi yang tinggi disebabkan prestasi penyerapan bunyinya yang tinggi. Kaedah unsur sempadan (BEM) dalam simulasi akustik digunakan bagi menilai prestasi kehilangan penghantaran bunyi penyenyap dengan lampiran plat berlubang mikro, dan keputusannya adalah sebanding dengan ukuran kehilangan penghantaran bunyi yang diperolehi dengan kaedah dua-beban serta pekali sekaitan yang tinggi di antara 0.77 sehingga 0.89. Ini telah mengesahkan kejituan model kaedah unsur sempadan di dalam analisis prestasi kehilangan penghantaran bunyi penyenyap. Bagi penyenyap jenis pengembangan mudah, penggunaan plat berlubang mikro di dalam model kaedah unsur sempadan menunjukkan peningkatan yang tinggi bagi kehilangan penghantaran bunyi dan lebih lebar jalur frekuensinya berbanding dengan penyenyap jenis pengembangan mudah yang tidak menggunakan plat berlubang mikro. Model yang telah disahkan ini kemudiannya digunakan untuk menilai prestasi akustik penyenyap jenis pemantulan dan tiub berlubang yang dipasang dengan plat berlubang mikro. Pengoptimuman bagi penyerap bunyi jenis dua lapisan plat berlubang mikro dijalankan di dalam bahagian kedua kajian ini. Kesan getaran-akustik, di mana persamaan klasik

plat digandingkan kepada persamaan gelombang akustik dan digunakan bagi pengoptimuman penyerap bunyi jenis dua lapisan plat berlubang mikro. Ini bagi meningkatkan prestasi penyerapan bunyi bagi penyerap bunyi plat berlubang mikro dengan mengambil kira kesan getaran plat berlubang mikro. Penyerap bunyi jenis lapisan tunggal plat berlubang mikro menunjukkan keputusan yang baik berbanding dengan keputusan yang telah diterbitkan apabila menggunakan model getaran-akustik. Oleh itu, model tersebut digunakan dalam pembangunan litar elektro-akustik setara bagi penyerap bunyi jenis dua lapisan plat berlubang mikro. Pengoptimuman menggunakan keadah simulasi penyepuhlindapan ke atas parameter reka bentuk penyerap bunyi untuk memperolehi nilai purata pekali penyerapan bunyi yang maksimum bagi jalur frekuensi di antara 200 Hz kepada 1000 Hz. Penyerap bunyi jenis dua lapisan plat berlubang mikro yang dioptimumkan telah dapat meningkatkan prestasi pekali penyerapan bunyi 8.88% hingga 12.52% berbanding dengan model yang mengabaikan kesan getaran-akustik. Mod (1,1) didapati mendominasi prestasi penyerapan bunyi berbanding dengan sembilan mod lain yang dipertimbangkan di dalam pengoptimuman ini. Ciri-ciri yang diambilkira ialah konfigurasi, ketebalan, nisbah redaman, dan keadaan sempadan plat berlubang mikro mempengaruhi pekali penyerapan bunyi. Secara keseluruhannya, peningkatan kehilangan penghantaran bunyi penyenyap telah dapat dicapai dengan lampiran plat berlubang mikro dan pekali penyerapan bunyi penyerap dengan mengambil kira kesan getaran plat berlubang mikro.

ANALYSIS OF VIBRO-ACOUSTIC EFFECT OF MICRO-PERFORATED PANEL SOUND ABSORBER

ABSTRACT

The application of micro-perforated panel (MPP) in the silencers and optimization of dual layer MPP sound absorber are evaluated in this study. MPP is a good candidate to be used in the silencer for achieving high transmission loss as it delivers high sound absorption performance. Boundary element method (BEM) in acoustical simulation has been used to assess the transmission loss performance of silencer attached with MPP, and when compared with the measurement using two-load method shows good agreement with the strong correlation coefficient, which is between 0.77 to 0.89. This verifies the BEM model of the silencer transmission loss analysis. For simple expansion chamber silencer, the application of MPP in the BEM model shows improvement in the transmission loss results with broader frequency bandwidth compared to the simple expansion chamber silencer without the MPP. The simulation work and analysis continues with the reflective system silencer and perforated tube silencer with MPP. The second part of this study is the optimization of the dual layer MPP sound absorber with consideration of the vibro-acoustic effect in which the classical plate equation is coupled to the acoustic wave equation. This is enhancing the sound absorption performance of an MPP sound absorber by taking the advantage of MPP plate vibration effect. The results of the vibro-acoustic model when applied to the single layer MPP, were found to be in good agreement with other published results. Thus, the model was used to develop the equivalent electro-acoustical circuit of a dual layer MPP sound absorber. The simulated annealing (SA) optimization technique was used to obtain the optimal design parameters of the MPP sound absorber in order to achieve the maximal averaged sound absorption coefficient for the frequency band of 200 Hz to 1000 Hz. The optimized dual layer MPP sound absorber improved the sound absorption coefficient by 8.88% to 12.52% when compared with the performance of the model designed without considering the vibro-acoustic effect. Mode (1,1) was found to dominate the sound absorption performance when compared with the other nine mode shapes that were considered in the present study. The effects of configuration, thickness, damping ratio, and boundary conditions of the MPP on the sound absorption coefficient were also investigated. The improvement of silencer transmission loss is achieved by attaching MPP and sound absorption coefficient of MPP sound absorber which is enhanced when MPP plate vibration effect is taken into consideration in the optimization.

CHAPTER 1

INTRODUCTION

This chapter discusses the background and the general flow of ideas of the study. The problem statement, scope, and objectives of this research study are listed. The final section points out the thesis outline and overview.

1.1 Introduction

Noise is generally defined as an unwanted sound which may come from the industries, traffic, public work, construction sites, and the electrical appliances in the residential area. The increase of noise pollution is hazardous to human health. It is reported that human will face health problems due to extended exposure to the noisy environment. In the same time, the noise affect the daily life in the area of speech interference, sleep disturbance, hearing impairment, cardiovascular, and physiological effect (Ising and Kruppa, 2004; Goines and Hagler, 2007). Designing a low noise level product is important not only to meet the legal requirement but also for marketability and competitiveness. It is a challenge for engineers and designers to design product with low noise level.

Normally, noise problem can be described in terms of a source, a transmission path, and a receiver (human). Noise control effort can be achieved based on the source-path-receiver concept where the noise control approaches are: modify the noise source to reduce noise output, control the transmission path as to reduce noise level reaching to recipients, and provide receivers with personal protective devices. One of the popular ways to reduce the undesirable or harmful noise is by applying the passive acoustic treatment materials on the transmission path. For instance, acoustic barriers are usually made from the dense or open celled absorptive materials to block sound energy from passing through (Delany and Bazley, 1970; Lu et al., 1999). Porous materials or foams are popular materials for the sound absorption and noise insulation solutions. These kinds of materials are easy to obtain, cost effective, and performed well. For the porous materials, the dissipation of sound energy is caused by the viscous losses of pores when sound waves travel through those materials.

In the recent years, numerous studies related to the MPP sound absorption have been performed by researchers since it's first introduction by Maa (1975). MPP can perform very well as a sound absorption panel when coupled with an air cavity depth at behind. The sound absorption mechanism of MPP is caused by the dissipation of sound energy when sound wave causes oscillatory movement through the small holes in the perforated panel which results in viscous dissipation. In order to obtain substantial sound absorption, the number of perforations must be significant. Perforation size of MPP plays a dominant role on the absorption performance, where the perforation diameter should be less than 1 mm (Maa, 1975).

Typically, porous or foam type materials are in the form of limp, soft and low stiffness. Support is needed to mount it on the wall or on the compartment of vehicles. The acoustic treatments using the porous materials result in extra installation cost and time. Since MPP is made from the stiffer metal panel, such as aluminum or stainless steel, mounting is not a problem.

Another advantage of MPP over the porous materials is the weather resistance. Porous materials tend to absorb not only the sound but also the water due to rain when applied outdoor or in humid condition. Weatherproof materials are required to cover the porous materials to resist the moist or oil absorption. It is impractical to apply the porous materials for the outdoor sound insulation purpose. Moreover, porous materials will deteriorate with time. The dust from the fibrous materials can affect the air quality especially when used for the sound absorption treatment in the ventilation system. These kind of absorbing materials have limitations, particularly when faced with industrial hygiene demanding application such as in hospital or pharmaceutical industry. By applying MPP coupled with an air cavity, the listed problems above can be solved and overcome the problem of the air polluted by fibers from foam type sound absorptive materials.

In general, perforated treatment has been applied in the muffler or silencer system, where the holes are created by drilling in the steel pipes, panels, or chambers. However, these perforations sizes are large and their acoustic resistances induced by the viscous dissipation of perforations are generally inadequate. This will cause the transmission loss performance of silencer to be relatively low. In this study, the MPP with perforations diameter of 0.9 mm are applied at the inlet and outlet of simple expansion chamber silencer. This is because the MPP with perforations size less than 1 mm will deliver high acoustic impedance (Maa, 1975, 1998) and improves the transmission loss of silencer. To date, there has been no reported study involving application of MPP in the silencer and the influence of MPP on the transmission loss.

In this study, except for the studying of application MPP in the silencer, the effect of structural movement of the MPP should not be neglected. In general, for the MPP of finite size, the transverse and flexural resonances of the solid part of the MPP can occur and will alter the sound absorption performance for the MPP sound absorber. This effect is used in conjunction with impermeable membranes to create low frequency panel absorbers. For panel absorber with impermeable membranes, high sound absorption is obtained at the resonance frequencies which are controlled by the bending stiffness or tension applied to the panel (Everest, 2001; Barron, 2003). These properties are adjusted to create the sound absorption at the desired frequency. However, for the flexible MPP, the basic sound absorption property is due to the oscillatory motion of air through the perforations. The sound absorption performance of MPP sound absorber might be substantially augmented as a result of flexural resonances of the panel. Therefore, the flexural resonances of MPP will constitute a second mechanism that can be used to enhance the sound absorption capabilities of the MPP sound absorber.

Optimization process is considered an essential part of design activity in all major disciplines. Designers and acoustic engineers are required to come up with optimum design of MPP sound absorber to meet the design constraints for their respective design. In order to obtain an optimum designed MPP sound absorber, the design parameters of the sound absorber should be well tuned. When the configuration of MPP sound absorber becomes complicated, the number of design parameter needed to be considered also increases simultaneously. At the same time, other issues such as materials, tooling time, production cost, and fabrication dimensions are also the factors that have major bearings on the final decision. Robust optimization technique is required to generate the optimized design parameters with the constraints consideration for the design. In this study, the simulated annealing (SA) optimization model with consideration of vibration effect for an MPP sound absorber is adopted to maximize the sound absorption performance in the configuration design of the MPP sound absorber, where the thickness, damping ratio, and edge clamping boundary condition are considered.

1.2 Problem Statement

In general, the task of defining the MPP using boundary element method (BEM) by geometrically following the actual dimension of the perforations requires the formation of submillimeter elements in the BEM model. This approach is computationally intensive and impractical for a complete silencer system. In this work, the approach adopted is by developing the acoustic impedance of the MPP based on the equation (Maa, 1998) and to use the acoustic impedance value in the BEM model for the calculation of various types silencer transmission loss when attached to the MPP.

The MPP vibration effect on the sound absorption coefficient of an MPP sound absorber has been analyzed using simplified model where the MPP vibration behavior is modeled as mass reactance or mechanical loss (Sakagami et al., 2005, 2009; Kang and Fuchs, 1999; Min et al., 2013). This simplified approach ignores the higher-order vibration modes of the plate. This work will consider the higher-order modes of the plate based on modal analysis of the MPP plate using the finite element method.

For the previous optimization studies, the sound absorption mechanism of the sound absorber depends only on the perforation and air cavity resonance, and the effect of plate vibration was ignored (Chiu et al., 2007; Ruiz et al., 2011). In this work, the optimization of the dual layer MPP is carried out to demonstrate the optimum design parameters of the MPP sound absorber when the vibration modes of the MPP plate are taken into consideration for optimization using the simulated annealing technique.

1.3 Scope of Study

In this study, the first criterion to be investigated is acoustic impedance of the MPP sound absorber. It is determined using classical Maa's equation (Maa, 1998) by assuming the MPP in high rigidity and the vibration effect of MPP is ignored. With the acoustic impedance obtained from the classical Maa's equation, the sound absorption coefficient for the MPP sound absorber also can be determined. The Maa's equation calculated sound absorption coefficient is used to verify with experimental measurement based on ISO-105342 standard.

Since MPP can be designed for the thinner and light-weight sound absorber, the application of MPP in the simple expansion chamber silencer will be studied here. The transmission loss of the silencer before and after the application of the MPP is measured using the impedance tube test rig system. In the same time, the simulation of the silencer is carried out using the boundary element method (BEM). The transmission loss of silencer obtained from the measurement is used to verify the accuracy of transmission loss obtained from the BEM simulation. The validated BEM simulation analysis model is also applied for (1) expansion chamber with extended tube silencer, (2) reflective system silencer, and (3) perforated tube silencer.

The optimization of MPP sound absorber is carried out using the simulated annealing (SA) algorithm as it is suitable for the global optimization in the noise control industries. In order to make the MPP sound absorber becomes more robust and covers the wider frequency range of sound absorption coefficient, a dual layer MPP sound absorber configuration is chosen and its design parameters are optimized tuned. The sound absorption coefficient of an optimized dual layer MPP sound absorber will be compared and studied in term of MPP vibration effect, configuration design of MPP sound absorber, thickness, damping ratio, and edge clamping boundary condition.

1.4 Objective

The objectives of this study are as follow:

1. To develop a boundary element method (BEM) acoustical simulation framework for the application of MPP in the silencers.

2. To determine the effect of MPP on the transmission loss of silencer for three different designs.

3. To optimize the MPP sound absorber for the maximum mean sound absorption coefficient with consideration of the vibration effect of the MPP plate.

1.5 Thesis Overview

This thesis is presented in six primary chapters which commenced with the introduction and followed by the literature review, theoretical development, methodology, results and discussion, and conclusion. The first chapter introduces the brief overview of the study, addresses the current issues of MPP sound absorber sound absorption analysis, and the availability application in the silencers.

Chapter two deals with the types of sound absorber, literature on MPP, its standard measurement techniques, and the existing applications. The reviews of silencer design and silencer analysis approaches are also summarized. The literature on the optimization of MPP sound absorber are included at end of this chapter.

The theoretical development, which is used for the analysis and optimization evaluation is included in chapter three. The vibro-acoustic model of MPP sound absorber is also presented, which is critical for answering the question of how plate vibration effect to enhance the sound absorption coefficient of MPP sound absorber.

In chapter four, the flow and ideas conducting the study is presented. The experimental procedures, simulation model built up, and optimization technique to achieve the objectives of the study are listed down.

In chapter five, the outcomes from the experiments, simulation analysis, and optimization are reported. The evaluation and discussion of the results are explained and presented also included in this chapter.

This thesis ends with the conclusion for the conducted study, recommendations for future works, and the contributions gained from the study, in Chapter six.

CHAPTER 2

LITERATURE REVIEW

In this chapter, different types of sound absorber are listed and their sound absorption performance are discussed. Then, the theoretical background and acoustic performance (sound absorption) of the micro-perforated panel (MPP) with back air cavity, which forms the basic MPP sound absorber are presented. The standard measurement processes to evaluate the acoustic performance of the MPP sound absorber are also included. The ideas of the MPP application and the various silencer designs are also highlighted. Subsequently, the standard measurement and analysis approaches for the silencer transmission loss performance are presented. The discussion of optimization on the MPP sound absorber is included at the end of this chapter.

2.1 Types of Sound Absorber

In the field of acoustic, sound absorber is considered as one of the tools for reducing the noise level (sound pressure level) to the acceptable level. According to World Health Organization,WHO (1999), long-term exposure to high noise level up to 70 dB(A) can cause hearing impairment. Sound absorber on the wall of an enclosed space can reduce the internal reflection of sound wave. Thus, the echo effect caused by the sound wave reflection is attenuated, and the condition of poor speech intelligibility can be avoided (Maekawa and Lord, 1994). In general, the sound absorber can be categorized into three major types, which are: (a) porous absorber, (b) membrane absorber, and (c) resonator absorber.

The porous absorber is considered the most commonly used for the sound absorption purposes. Normally, this type of sound absorber is applied as acoustic treatment in the room, hall, auditorium, and studio (Schwind, 1997). For the porous absorber, the sound absorption mechanism relies on the damping of the oscillation of air particles by viscous dissipation and friction, where the impinged sound wave (energy) is converted to the heat (Barron, 2003). The sound absorption ability of porous absorber mainly depends on several parameters, such as thickness, density, airspace, and surface treatment. As such, it has been reported that the poor sound absorption of porous absorber at the low frequency range can be enhanced by applying the thicker porous material (Crocker, 2007; Cox and D'Antonio, 2009). Typically, the sound absorption of porous absorber performs well for the higher frequency range compared with membrane and resonator absorber as shown in Figure 2.1 (Jacobsen et al., 2007). A similar observation was reported by Everest (2001).

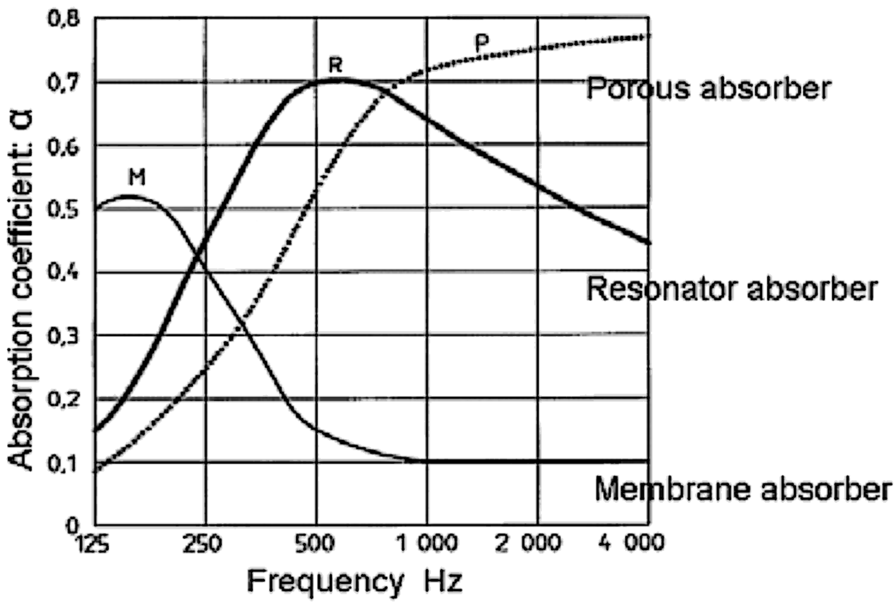


Figure 2.1: The sound absorption coefficient comparison for porous, resonator, and membrane absorbers respectively (Jacobsen et al., 2007)

Membrane absorber is non-rigid, non-porous, and thin membrane or sheet placed at a certain distance from a hard backing (rigid wall). Air cavity is located between both of them as shown in Figure 2.2. The working mechanism of the membrane absorber is based on the fact that the membrane vibrates in a flexural mode in response to the sound wave (energy). Thus, the sound wave will become weaker, and the energy of sound is converted to the movement of membrane and part of it dissipates in the form of heat. Based on Figure 2.1, it can be seen that the sound absorption coefficient of membrane absorber is most efficient at the low frequency range as compared with resonator and porous absorber (Jacobsen et al., 2007). The sound absorption performance of membrane absorber can be tuned by the technique of fixing membrane and its rigidity of membrane as reported in the study by Oldfield (2006). The airspace of membrane absorber filled with the damping materials, such as fiberglass, foam, and glass wool also enhances and widens the sound absorption bandwidth (Crocker, 2007).

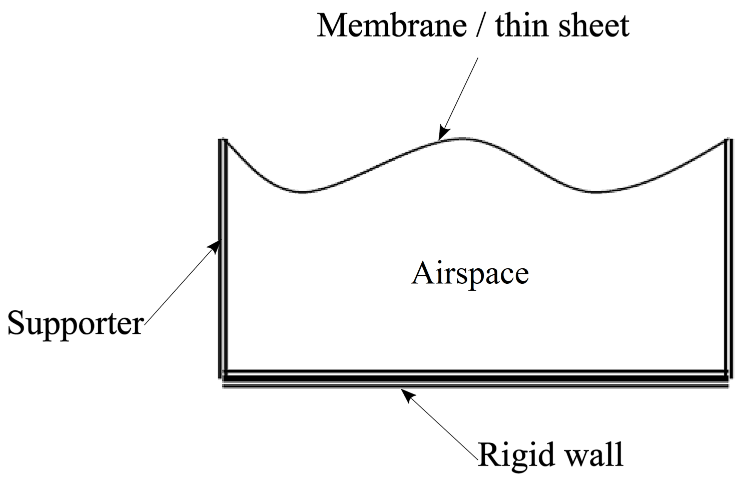


Figure 2.2: Configuration of membrane absorber

From Figure 2.1, resonator absorber typically acts to absorb noise in a relatively narrow frequency range as compared with the other two sound absorbers. Resonator absorber includes the perforated materials or the materials that have openings (holes and slots). The classic example of a resonator absorber is the Helmholtz resonator absorber as shown in Figure 2.3. Basically, the concept of Helmholtz resonator absorber can be referred as an enclosed air cavity with a small opening connected to exterior. The sound absorption mechanism of Helmholtz resonator absorber is caused by the resonant frequency and governed by the size of the opening area (S), the length of the neck (L), and the volume of air trapped in the cavity (v). These parameters should be tuned when targeting specific single frequency sound absorption (Adam, 2004).

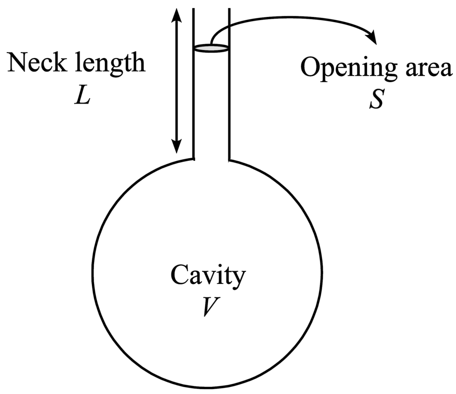


Figure 2.3: Configuration of Helmholtz resonator absorber

The perforated panel absorber is considered as another example of resonator absorber, where it shares similar working concept with the Helmholtz resonator absorber by assuming the perforations on the panel and a backed rigid wall are a collection of a large number of individual Helmholtz resonator absorber. According to Barron (2003), the sound absorption bandwidth for resonator absorber normally is narrow and sharp, and the special treatments such as applying the porous materials in the cavity, changing the size of perforation and cavity are needed to be taken in order to achieve a broader sound absorption bandwidth.

By comparing these three types of sound absorber, it is found that each type of sound absorber has limitations on the sound absorption performance and applications. For instance, the membrane absorber performs well in the low frequency range, with narrow sound absorption bandwidth of the resonator absorber. Porous absorber seems is the preferable choice among these sound absorbers, but there is hygiene issue as the glass fiber or foam can generate grime. Ackermann et al. (1988) applied the splitter combined with cubical honeycomb in the air cavity of the membrane absorber to improve the sound absorption performance at the low and middle frequency range (150 Hz to 500 Hz). Application of the porous materials inside the airspace of resonator absorber can widen the sound absorption bandwidth and improve the sound absorption performance (Barron, 2003; Crocker, 2007; Everest, 2001).

MPP sound absorber has been accepted as a reliable, noncombustible, and environmental friendly sound absorber (Herrin et al., 2011; Corin and Wester, 2005). The sound absorption performance of MPP sound absorber bridges the gap between the resonator absorber and porous absorber, which delivers high sound absorption performance with a relatively wide bandwidth by taking advantage of the submillimeter size of the perforations. This has been considered as one of the most likely alternatives for the next generation sound absorber due to its sound absorption performance, reliability, cleanable, and fiber free (Herrin et al., 2011).

2.2 Micro-perforated Panel (MPP)

A micro-perforated panel (MPP) is a sheet panel with many small diameter perforations (usually less than 1 mm) are punched and distributed over its surface. A micro-perforated panel placed before a rigid surface with an air gap makes an MPP sound absorber as shown in Figure 2.4(a) (Maa, 1975). Unlike ordinary perforated panels where the perforations are in millimeters or even centimeters, and of little acoustic resistance, the submillimeter diameter size of micro-perforations in MPP provides significant acoustic resistance and low acoustic mass reactance, both are necessary for the wide band sound absorbers, without the need for additional fibrous or porous materials applied in the air cavity of MPP sound absorber (Maa, 1998).

Maa (1975) established the approximate solution and theoretical basic for the MPP sound absorber using electro-acoustical equivalent circuit as depicted in Figure 2.4(b). According to this model, the sound absorption coefficient of an MPP sound absorber is determined by the diameter of perforations, the perforation ratio, the plate thickness, and the air cavity depth. Typically, an MPP can be made from any materials from cardboard, plastic, and plywood to sheet metal, with aesthetic finishing or decoration to suit the purposes (Fuchs and Zha, 1995).

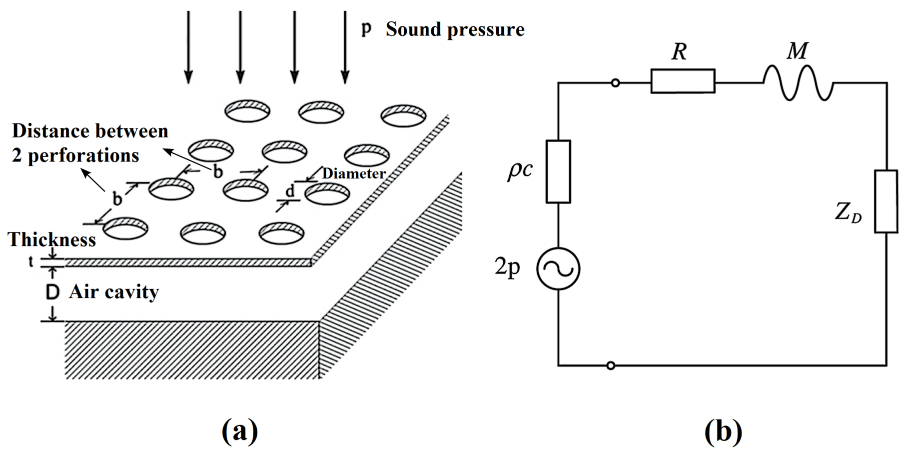


Figure 2.4: (a) Schematic diagrams of the MPP sound absorber, and (b) electro-acoustical equivalent circuit (Maa, 1975)

In recent years, the MPP sound absorber has been extensively studied on its sound absorption performance by carrying out different modifications on the design of MPP. Normally, the parameters considered for the modification are perforation size or shape, air cavity, and the configuration of MPP sound absorber. For instance, the idea of multi-size perforations modification on the MPP sound absorber had been proposed by Miasa et al. (2007), and followed by Randeberg (2000), who modified the traditional circular perforations to horn-shaped (with larger outer radius and small inner radius). By applying multi-size perforations on the MPP, it is found that the sound absorption performance of MPP sound absorber can achieve high sound absorption coefficient (more than 0.8) for wider bandwidth (800 Hz to 1500 Hz) (Miasa et al., 2007). According to Randeberg (2000), the MPP sound absorber delivers a relatively high sound absorption coefficient over large bandwidth by changing the circular perforations to the horn-shaped (with large outer radius and small inner radius). There is maximum sound absorption coefficient – 0.8 for the frequency range of 50 Hz to 500 Hz.

Another parameter to be considered is the back air cavity, which can give a relatively broader frequency band of sound absorption coefficient peak. This observation was made by Sakagami and Morimoto (2007) in the application of non-uniform air cavity size for the membrane absorber. This idea encourages Wang et al. (2010) to consider the irregular-shaped air cavity applied in the MPP sound absorber as shown in Figure 2.5. The irregular-shaped backing air cavity was shown to significantly alter the sound absorption mechanisms and the effective frequency range of the overall sound absorption coefficients. Application of the honeycomb structure for MPP sound absorber can improve the sound absorption performance, and stiffens the construction of MPP sound absorber which is typical built from thin MPP sheet (Pan et al., 2005;

Sakagami, Nakajima, Morimoto, Yairi and Minemura, 2006; Sakagami et al., 2007; Sakagami, Yamashita, Yairi and Morimoto, 2010).

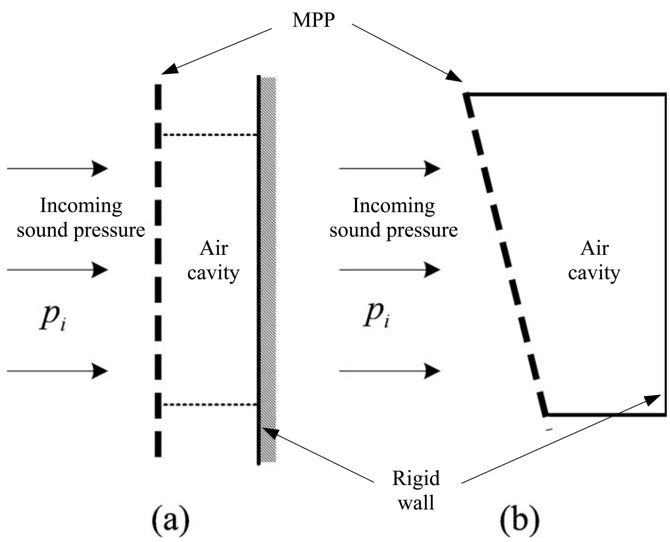


Figure 2.5: Schematic of the MPP sound absorbers, (a) conventional configuration in which an MPP is backed by a rigid wall or a rigid rectangular cavity, (b) irregular-shaped configuration in which an MPP is backed by a rigid trapezoidal cavity (Wang et al., 2010)

In order to make the sound absorption coefficient of MPP sound absorber to cover a broader bandwidth, two layers MPP design with two air cavities is developed as shown in Figure 2.6(a), in which it is treated as an integrated two resonators sound absorption system. This type of MPP sound absorber provides a relatively wider sound absorption bandwidth compared with the single layer MPP sound absorber (Maa, 1987). The backing rigid wall is considered as an essential part for this design. However, a new design of dual layer MPP sound absorber without the backing rigid wall was proposed by Sakagami, Morimoto and Koike (2006) as shown in Figure 2.6(b), where this new dual layer MPP sound absorber provided fairly high sound absorption coefficient for the low frequencies where the conventional dual layer MPP sound absorber (with backing rigid wall) was unable to achieve.

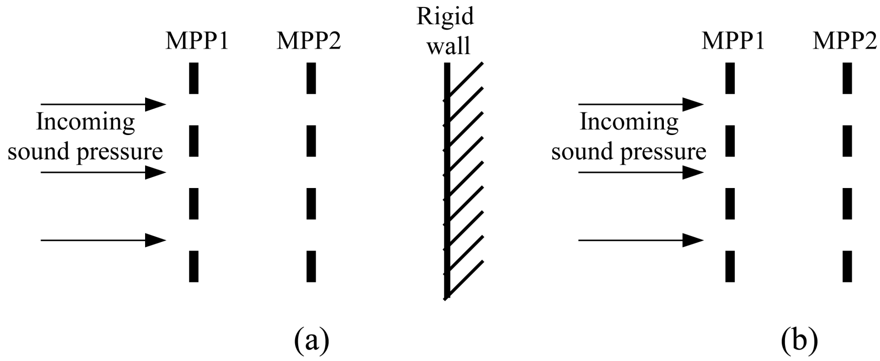


Figure 2.6: Configuration of dual layer MPP sound absorber, (a) with backing rigid wall (Maa, 1987), (b) without backing rigid wall (Sakagami, Morimoto and Koike, 2006)

A broader sound absorption bandwidth seem can be obtained with more layers MPP applied in the MPP sound absorber, and this hypothesis leads to Miasa and Okuma (2007) study of the multi-leaf MPP sound absorber without including backing rigid wall with reasonably high sound absorption coefficient at the low frequency range comparable with the performance of the dual layer MPP sound absorber. Thus, it can be said that the MPP sound absorber with two layers and above MPPs shares the similar sound absorption performance. In order to save the manufacturing cost and a lightweight multi-leaf MPP sound absorber, Sakagami, Yairi and Morimoto (2010) replaced one of the MPPs with the permeable membrane in the multi-leaf sound absorber and studied the sound absorption performance. Although, the permeable membrane had been replaced in the multi-leaf MPP sound absorber, it still maintains a similar sound absorption performance as the conventional multi-leaf MPP sound absorber.

For the cases discussed above, the analysis or predictions for the sound absorption coefficient of the MPP sound absorber were done by assuming the MPP is rigid (and thus, the vibration of the MPP can be neglected (Maa, 1975)). However, this assumption does not accurately describe the membrane or limp-type MPP, since this type of panel suffers from high flexibility and vibration. Therefore, the vibration effect must be considered when determining the sound absorption coefficient of MPP sound absorber (Lee, Sun and Guo, 2005; Yoo, 2009; Bravo et al., 2012). There are some researchers who considered the vibration effect of MPP, in which it may affect the sound absorption coefficient of MPP sound absorber (Sakagami et al., 2005, 2009; Kang and Fuchs, 1999; Min et al., 2013) where they showed that the vibration effect of MPP contributes and enhances the sound absorption coefficient. However, these proposed approaches which considered the vibration effect has ignored the higher-order vibration modes of the MPP that cannot fully represent the actual situation of the MPP vibration mechanism. For instance, when an MPP of considerable size is mounted or lined on a wall, the MPP will be more flexible and will exhibit more vibration modes than a smaller corresponding MPP specimen mounted in an impedance tube during testing (Leissa, 1993). In such a case, the size of the MPP and the mounting method will affect the vibration of the MPP as well as the sound absorption coefficient. These important criterions had not been considered by the researchers above.

In order to improve the weakness of the MPP vibration effect modeling, a more accurate and detail approach had been proposed by Lee, Lee and Ng (2005) by including the bending stiffness effect in the analysis of a flexible MPP, for which the classical plate equation is coupled with the acoustic wave equation. The results indicate that the panel vibration effect can dissipate more energy, and the peak sound absorption coefficient band can be widened by appropriately selecting the design parameters of the MPP sound absorber. The study by Lee, Lee and Ng (2005) is important because it shows how the vibration effect can be included in the sound absorption coefficient calculation of the MPP by including the vibration modes of the plate.

2.2.1 Measurement for MPP Sound Absorber

Basically, the sound absorption performance of an MPP sound absorber is determined based on the sound absorption coefficient, α. In order to make the sound absorption coefficient of an MPP sound absorber comparable, the standard procedures are used for the sound absorption coefficient measurement. There are two standards which were published by International Organization for Standardization (ISO) and American Society for Testing and Materials (ASTM) for the guideline on the sound absorption coefficient measurement (ISO 10534-1, 1996; ISO 10534-2, 1998; ASTM E1050-98, 1998). Although these two standards were drawn up by different international body, the majority of measurement procedures and setup are identical to each other.

There are two different methods used for these two versions of standard. For ISO 10534-1, the standing wave ratio method is used, and transfer function method is applied for the ISO 10534-2. Both different version standards measurement are conducted using the impedance tube as shown in Figure 2.7. Typically, the version of ISO 10534-2 is preferable as the transfer function method is relatively fast and accurate method to determine the sound absorption coefficient, and applicable over a wide frequency range (Seybert, 2003; Suhanek, Jambrosic and Horvat, 2008).

2.2.2 Existing Application of MPP Sound Absorber

Fuchs (2001) reported the motivation to use the grime free sound absorber, such as membrane absorber, Helmholtz resonator absorber, and MPP sound absorber since these are clean, non-deteriorate, and non-combustible as they are made from the metal (Herrin et al., 2011). One of the major reason for choosing MPP sound absorber is

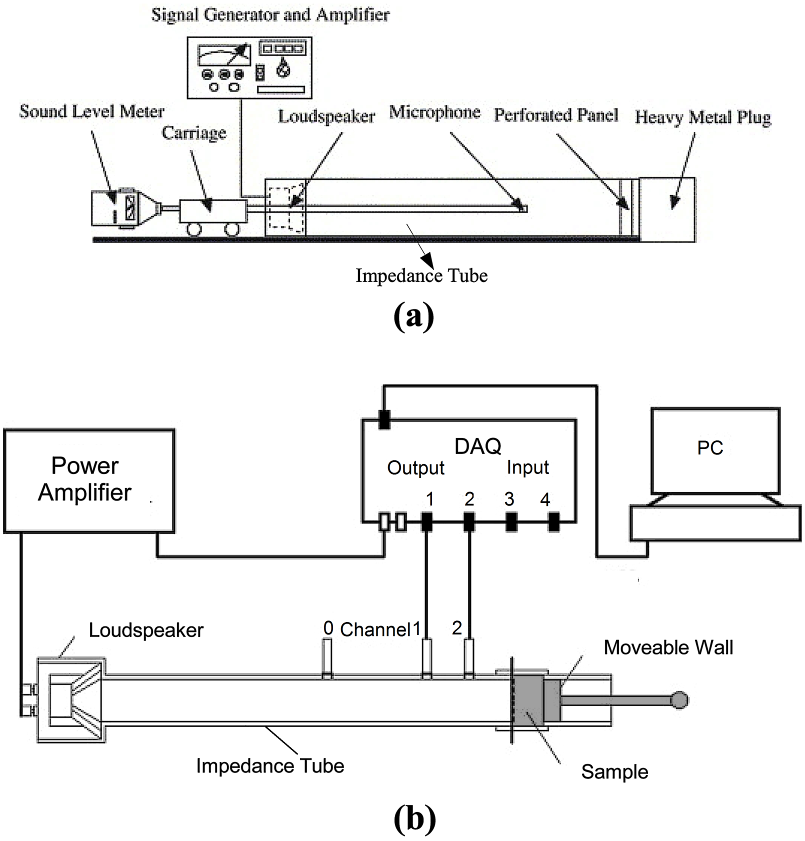


Figure 2.7: Schematic diagram of sound absorption coefficient measurement based on International Standard Organization, (a) standing wave ratio method (ISO 10534-1, 1996), and (b) transfer function method (ISO 10534-2, 1998)

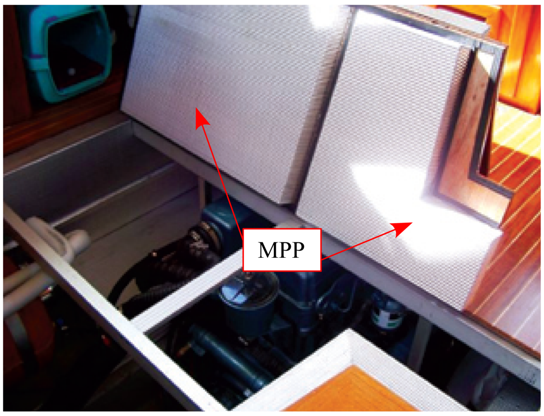
the frequency tunable capability for sound absorption performance and efficient at the lower frequency range making it the most suitable for the high speech intelligibility environment (Fuchs, 2001).

Kang and Brocklesby (2005) used the transparent type MPP for retaining the level of visibility and ventilation comfort whilst maintaining the high sound absorptivity with fairly good noise reduction, improved ventilation and light ingress. Application of the MPP sound absorber on the ceiling of an amphitheater and gymnasium showed the reduction of reverberation time, sound pressure level, and more pleasant aesthetical design of the building (Lu et al., 2007; Boe and Danoline, 2008). Sakagami, Morimoto and Yairi (2008) added the honeycomb on the MPP sound absorber to increase the stiffness of the MPP panel and to improve the sound absorption performance. Liu and Herrin (2010) conducted further research work by partitioning the backing air cavity in the MPP sound absorber to increase the physical strength and the sound absorption performance.

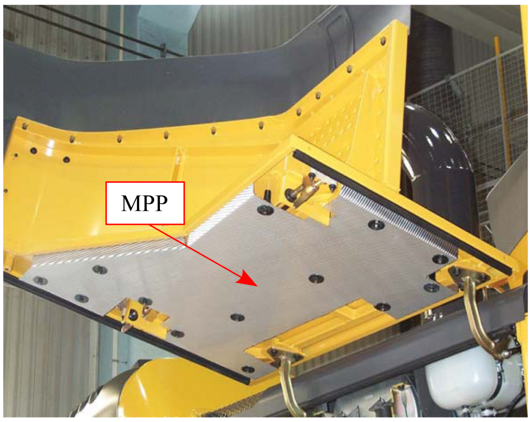
Wu (1997) applied the MPP in the rectangular ducting which form the MPP silencer to predict its sound attenuation capability and showed that the performance of MPP was affected by the thickness, backing air cavity depth, perforation porosity, and perforation diameter. The perforation diameter is the critical parameter to widen the MPP silencer attenuation peak frequency range.

In the automotive industry, the MPP sound absorber has also been applied for reducing the noise level in the study of Corin and Wester (2005). Normally, the application of MPP sound absorber for the vehicle is used to absorb the noise emitted from the engine combustion, tire, and wind. The metallic MPP plate is considered reliable and durable for the high temperature environment of the engine compartment has made Volvo and Scania to apply the MPP sound absorber in the engine compartment since 1997. Figure 2.8 shows the examples of MPP sound absorber in the bonnet near the engine compartment.

For the hygiene consideration, the MPP sound absorber is also used in the medical devices. Li and Mechefske (2010) applied the MPP sound absorber in the cylindrically shaped magnetic resonance imaging (MRI) scanner in the effort to reduce the generated noise during the scanning process. This study is important because of the high noise level in the scanner has become a major concern in clinical practice in



(a)



(b)

Figure 2.8: The application of MPP sound absorber for the engine compartment of, a) boot (Herrin et al., 2011), b) wheeled loader (Corin and Wester, 2005)

terms of patient comfort and patient hearing safety. The MPP sound absorber consists of multiple and wider sound absorption frequency bands at higher frequency ranges (1100 Hz and above) when used in the MRI scanner.

2.3 Typical Silencer Designs

Silencer is a device to attenuate the noise level of the system. The application of silencer is mostly by direct attachment to the components with high noise level. There are basically two major types of silencer design commonly used, namely absorptive and reflective. Both silencer types are based on the different working mechanism, which make the performances varies in different manner (Munjal, 1987).

For reflective silencer, it consists of a series of interconnect tubes inside the chamber of silencer in order to create the reflective effect for the sound attenuation. Referring to Figure 2.9, the silencer consists of two chambers separated by a baffle with two resonator tubes in it. The baffle will force the incoming sound wave to reflect in the chamber of silencer. These sound waves interfere and will cancel each other inside the silencer. The complete destructive interference (ideal case) can only occur when the reflected sound wave is the same amplitude and 180 degree out of the phase with the incoming (transmitted) sound wave. By changing the silencer geometry and chamber size, the destructive interference can be tuned in order to suit for the desired frequency of noise to be attenuated (Potente, 2005).

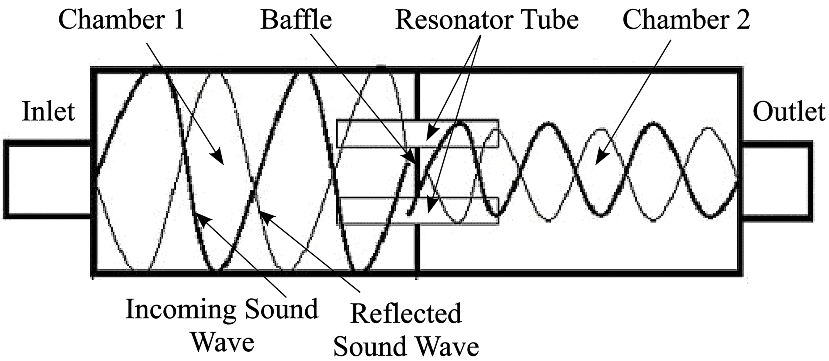


Figure 2.9: The sound wave pattern in a reflective silencer

Figure 2.10 shows the actual reflective silencer, which consists of a series of resonator chambers, baffles, and perforated tubes. It used the destructive interference working concept to achieve the noise level attenuation by designing the arrangement and sizes of components inside the reflective silencer. For instance, the perforated tube allow the sound waves to scatter out in different directions and the baffles force the sound waves reflected inside the chamber of the silencer to create the phenomenon of destructive interference (Elden, 2010b). In general, the reflective silencer performs well for the low frequency range as compared with the absorptive silencer. It is normally used for the fixed speed machinery or system which producing the pure tones.

Among the reflective silencers, the most basic design of reflective silencer is the simple expansion chamber silencer, where the inlet tube is connected to the expansion

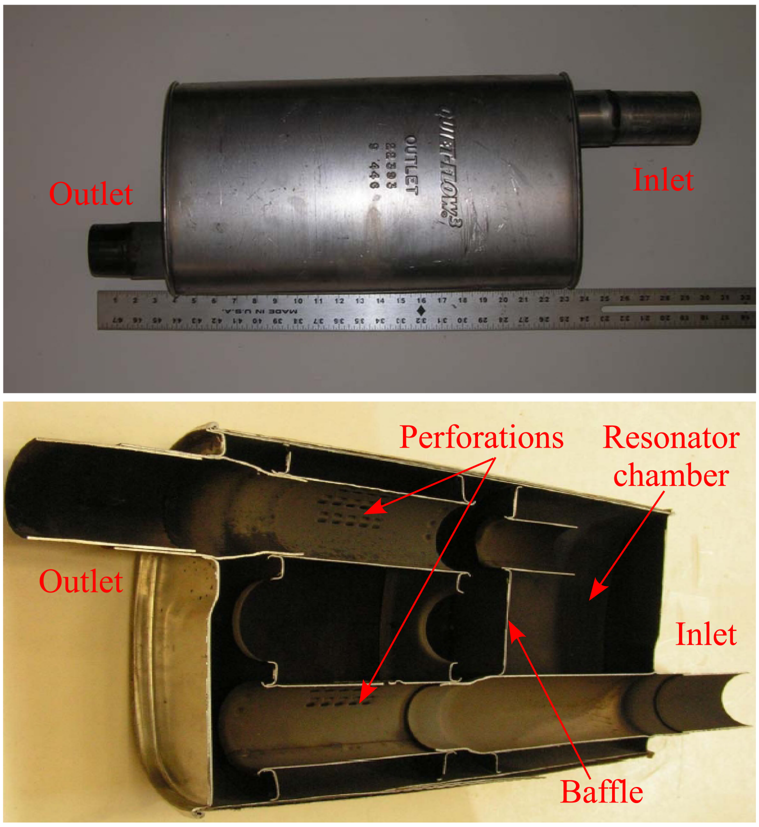


Figure 2.10: Typical reflective silencer (photo taken from http://www.paraglidingteam.nl/PPGTechnics/soundandnoise/Mufflers/1155795969.pdf)

volume of chamber, and the other side of outlet tube is connected to the end of it as shown in Figure 2.11. By this, it creates an abrupt change in the cross-sectional area at each end of the chamber volume. The simple expansion chamber silencer performs most efficiently in the applications involving discrete frequencies rather than broad band noise (Wu and Wang, 2011). The size of the chamber is adjusted to tune for the required frequencies of sound attenuation. The transmission loss of simple expansion chamber silencer (TL_{exp}) can be estimated by the equation as below (Bhattacharya et al., 2008):

$$TL_{exp} = 10\log_{10}\left(1+\left[0.25\times\left(\frac{D_e}{d_t}-\frac{d_t}{D_e}\right)^2\times\sin^2(kl)\right]\right), \tag{2.1}$$