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MODELLING THE EFFECT OF VIBRATION ON THE SOUND ABSORPTION PERFORMANCE OF GREEN SOUND ABSORBER USING WAVE PROPAGATION TECHNIQUE

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Abstract— Natural fibers (NF) have been known to have good acoustic performance and can potentially substitute the classical synthetic absorber materials which give negative impact for the environment. Commonly, the sound absorption model available for porous acoustic panel ignores the vibration of the panel. In this paper, the effect of the vibration on the sound absorption of the NF panel backed with a solid panel and air layer is modeled using wave propagation method. It is shown that the model can be adjusted into the case of an absorber panel backed with air layer and rigid surface by increasing the impedance of the solid panel.

Keywords—Microperforated Panel, Absorption Coefficient, Wave Propagation, Natural Waste Material, Fiber.

I. INTRODUCTION

A good acoustic performance particularly for the speech intelligibility is important in a building such as an auditorium. To increase the sound absorption, the room walls and ceilings are covered by absorptive layers using glass wools or acoustic foams made from synthetic chemical substance. These type materials not only have negative impacts concerning their risk on pollution, health and fire hazard, but also contribute to CO₂ pollution in their fabrication that triggers global warming.

A. Green sound absorbers

Until recently, many studies focused in investigating organic or natural fiber (NF) as sound absorber materials are still progressing. Natural materials give more advantages than synthetic ones as they are renewable and available in abundance amount in certain countries. Previous researchers have investigated the potential of the NF as sound absorbers. Ersoy and Kucuk [1] investigated the industrial waste of tea leaf fiber as an acoustic material. It is found that this gave promising result for significant frequency range. Zulkifli *et al.* [2] worked on coir and oil palm fibers on their acoustic properties. Both are compressed under pressure to form a fiber sheet. It is shown that both natural wastes have good sound absorption particularly at high frequency. Lindawati *et al.* [3] measured sound absorption of *Arenga Pinnata* fiber. The result indicated that it has good absorption from mid to high frequency. Abdullah *et al.* [4] investigated natural waste fibers from dried paddy straw. Good acoustic performance is found at frequency above 2000 Hz.

Apart from finding potential natural fibers to produce environmentally friendly sound absorber, the newest method is the employment of a microperforated panel

(MPP) as a non-fibrous sound absorber. Its theory and design were pioneered by Dah You Maa [5] in 1975. Typically, the system consists of a panel with submillimetric holes backed by an air cavity and a rigid wall. The diameter of the hole should be less than 1 mm and the perforation ratio is between 0.5 – 1.5%.

B. Existing models for sound absorption coefficient

The existing conventional model for MPP and also porous absorber is for the case of the panel backed by an air layer and a rigid wall. The sound absorption coefficient is given by

$$\alpha = \frac{4 \operatorname{Re}[Z_{tot}]}{(1 + \operatorname{Re}[Z_{tot}])^2 + \operatorname{Im}[Z_{tot}]^2} \quad (1)$$

where the $\operatorname{Re}[Z_{tot}]$ and $\operatorname{Im}[Z_{tot}]$ denotes the real and imaginary part of the surface acoustics impedance of the panel.

Eq. (1) assumes the panel as a rigid frame, i.e. ignoring the effect of panel vibration due to impinging sound pressure.

Sakagami *et al.* [6] investigates the effect of induced vibration for double panels of MPP using the Helmholtz-Kirchhoff integral. The results show that the vibration improves the sound absorption of the MPP only at the resonance.

The effect of flexural vibration on a porous absorber panel was proposed by Takahashi and Tanaka [7] employing the Helmholtz integral formula and Green's function to numerically calculate the acoustics pressure acting on the panel. It is also found that the vibration increases the sound absorption.

This paper proposes a simpler analytical model to calculate the absorption coefficient of a thin porous absorber by including the effect the panel vibration using the wave propagation method.

II. MATHEMATICAL MODEL

Fig. 1 shows the diagram of the NF-solid panel system separated by distance D excited by sound pressure. The incident, reflected and transmitted of sound pressure at the system are, respectively given by

$$p_i = \tilde{A}e^{-jkx} \quad (2)$$

$$p_r = \tilde{B}e^{jkx} \quad (3)$$

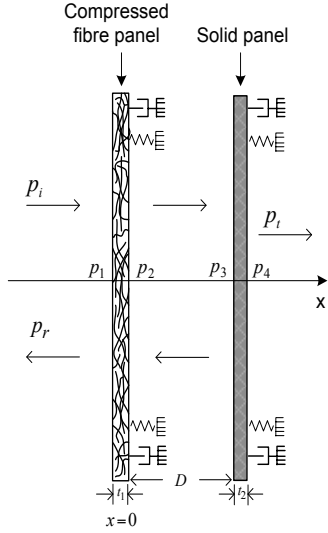


Figure 1. Mechanical system of a NF – solid panel

$$p_i = \tilde{C}e^{-jkx} \quad (4)$$

where $k = \omega/c$ is the acoustic wavenumber, ω and c are angular frequency and the speed of sound respectively.

The acoustic pressure acting on the porous panel surface p_1 and p_2 at $x = 0$ and on the solid panel surface p_3 and p_4 at $x = D$ are given by

$$p_1 = p_i + p_r = \tilde{A}_1 e^{-jkx} + \tilde{B}_1 e^{jkx} = \tilde{A}_1 + \tilde{B}_1 \quad (5)$$

$$p_2 = \tilde{A}_2 e^{-jkx} + \tilde{B}_2 e^{jkx} = \tilde{A}_2 + \tilde{B}_2 \quad (6)$$

$$p_3 = \tilde{A}_2 e^{-jkD} + \tilde{B}_2 e^{jkD} \quad (7)$$

$$p_4 = \rho c v_{p_2} \quad (8)$$

where v_{p_2} is the velocity of the solid panel and \tilde{A} and \tilde{B} are the complex pressure amplitudes.

Using Euler equation $\tilde{v} = -1/j\rho\omega(\partial p/\partial x)$ at $x = 0$ for p_1 and p_2 , the sound particle velocity at the surface of the panels can be written as

$$\tilde{v}_1 = \frac{1}{\rho c} (\tilde{A}_1 - \tilde{B}_1) \quad (9)$$

$$\tilde{v}_2 = \frac{1}{\rho c} (\tilde{A}_2 - \tilde{B}_2) \quad (10)$$

On the porous elastic plate surface the particle velocity is given by [6]

$$\tilde{v} = \xi_I v_p - \frac{1}{\gamma Z_c} \frac{\partial p}{\partial x} \sigma \quad (11)$$

where $\xi_I = 1 - [\gamma Z_c]_{react} \sigma / \gamma Z_c$ with γ is the propagation constant, Z_c is the characteristic impedance and σ is the porosity. For quantities γ and Z_c empirical models developed by Allard [8] are used.

Substituting Eqs. (9) and (10) into Eq. (11) yields mean the sound particle velocity at $x = 0^+$ and $x = 0^-$, respectively

$$\tilde{v}_1 = \frac{\xi_I v_{p_1}}{\left(1 - \frac{jk\rho c \sigma}{\gamma Z_c}\right)}, \quad \tilde{v}_2 = \frac{\xi_I v_{p_1}}{\left(1 - \frac{jk\rho c \sigma}{\gamma Z_c}\right)} \quad (12)$$

Using Eq. (6), Eq. (7) can be written as

$$p_3 = p_2 \cos kD - j\rho c v_2 \sin kD \quad (13)$$

Using Euler equation for Eq. (7) then substituting Eqs. (6) and (10) yields

$$p_2 = \frac{\rho c \tilde{v}_2 \cos kD - \rho c v_{p_2}}{j \sin kD} \quad (14)$$

Substituting Eq. (14) into Eq. (13) gives

$$p_3 = \frac{\rho c (\tilde{v}_2 - v_{p_2} \cos kD)}{j \sin kD} \quad (15)$$

The equation of motion of the solid panel is given by

$$Z_{p_2} v_{p_2} = (p_3 - p_4)(1 - \sigma) \quad (16)$$

where Z_{p_2} is the impedance of the solid panel.

Substituting Eqs. (8), (12) and (15) into Eq. (16) and after mathematical manipulation

$$v_{p_2} = \frac{\tilde{v}(1 - \sigma)}{\Theta} \quad (17)$$

where $\Theta = jZ_{p_2} \sin kD / \rho c + (1 - \sigma)e^{jkD}$ is the non-dimensional term.

The equation of motion of the NF panel is given by

$$Z_{p_1} v_{p_1} = (p_1 - p_2)(1 - \sigma) \quad (18)$$

where Z_{p_1} and v_{p_1} are the impedance and the velocity of the NF panel, respectively.

Substituting Eq. (5) and (14) into Eq. (18) yields the surface velocity at the solid panel

$$\tilde{v} = \frac{j2\tilde{A}_1(1 - \sigma)\xi_I \Theta \sin kD}{jZ_{p_1} \Theta \Omega \sin kD + \rho c \xi_I (1 - \sigma) [\Theta e^{jkD} - (1 - \sigma)]} \quad (19)$$

where $\Omega = (1 - jk\rho c \sigma / \gamma Z_c)$ is the non-dimensional term.

The reflection coefficient R at $x = 0$ is given by

$$R = \frac{p_r}{p_i} \quad (20)$$

Rearrange Eq. (9) then substituting into Eq. (3). After that substituting Eq. (2), (3), and (19) into Eq. (20) and after mathematical manipulation, the reflection coefficient of NF-solid panel system is given by

$$R = \frac{jZ_{p1} \Theta \Omega \sin kD + \rho c \xi_l (1 - \sigma) \left[\Theta e^{-jkD} - (1 - \sigma) \right]}{jZ_{p1} \Theta \Omega \sin kD + \rho c \xi_l (1 - \sigma) \left[\Theta e^{jkD} - (1 - \sigma) \right]} \quad (21)$$

The absorption coefficient of NF-solid panel can be calculated as

$$\alpha = 1 - |R|^2 \quad (22)$$

Note that absorption coefficient defined here includes the transmission loss of the system.

III. RESULT AND DISCUSSION

Fig. 2 shows the theoretical results of absorption coefficient of NF back with solid panel and air layer. The calculation was done by varying the porosity and by fixing the flow resistivity R_f . It can be seen that absorption coefficient at low frequency below 100 Hz decreases as the porosity increases. This corresponds to the transmission loss of the system. For frequency range 200 – 1000 Hz, the sound absorption increases as the porosity increases. This relates the absorption of

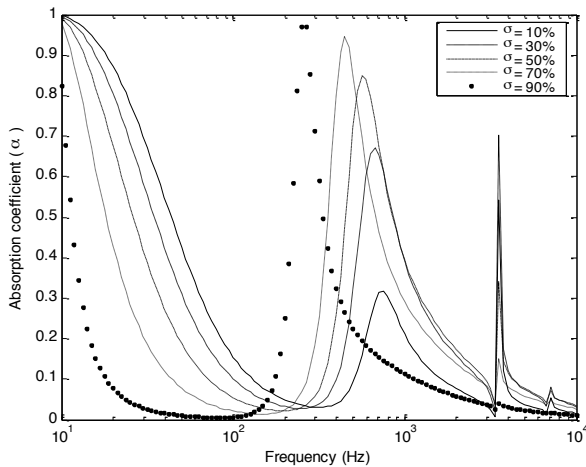


Figure 2. Absorption coefficient of a NF – solid panel system ($t_{1,2} = 1 \text{ mm}$, $D = 50 \text{ mm}$ and $R_f = 1.1 \times 10^4 \text{ N.s.m}^{-4}$)

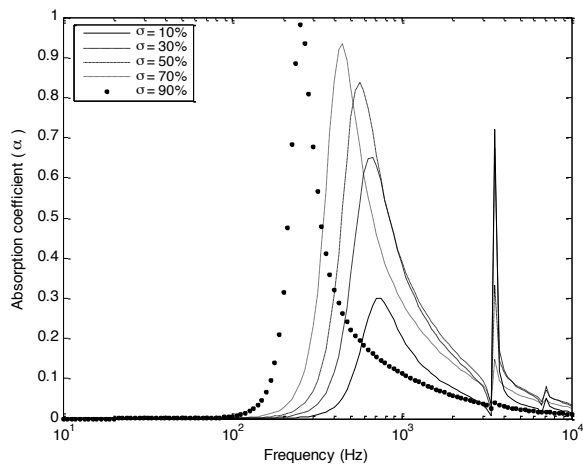


Figure 3. Absorption coefficient of a NF – rigid backing system ($t_{1,2} = 1 \text{ mm}$, $D = 50 \text{ mm}$ and $R_f = 1.1 \times 10^4 \text{ N.s.m}^{-4}$)

the sound energy by the air layer between the panels.

The case of an NF panel backed by air layer and an infinite rigid surface can be obtained by setting large value for the solid panel impedance Z_p in Eq. (21). The results are shown in Fig. 3 where the absorption coefficient below 100 Hz is now suppressed into zero as the sound transmitted behind the solid panel has been blocked.

The Fig. 4 plots the results for different air layer thickness. The thickness of the air layer backing the NF corresponds to the maximum peak of the sound absorption coefficient. The corresponding frequency of this peak shifts to lower frequency if the air layer thickness is increased.

Fig. 5 presents for different flow resistivity of the NF panel which indicates that absorption increases as the flow resistivity reduces.

Fig. 6 plots the results for different thicknesses of the porous panel of NF-rigid backing system. It can be seen that the thinner panel not only shift the performance to higher frequency but also widen the frequency range. This is due to the effect of vibration on the NF panel which increases the loss of sound energy.

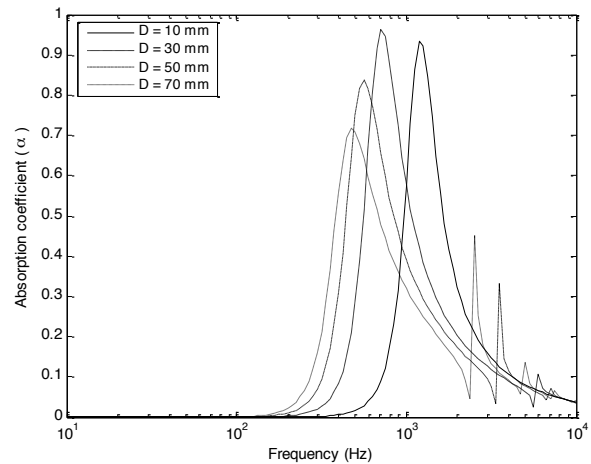


Figure 4. Absorption coefficient of a NF – rigid backing system ($t_{1,2} = 1 \text{ mm}$, $\sigma = 50\%$ and $R_f = 1.1 \times 10^4 \text{ N.s.m}^{-4}$)

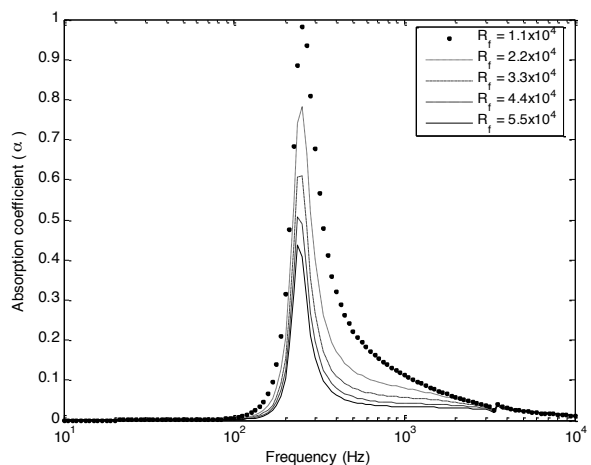


Figure 5. Absorption coefficient of a NF – rigid backing system ($t_{1,2} = 1 \text{ mm}$, $\sigma = 90\%$, $D = 50 \text{ mm}$)

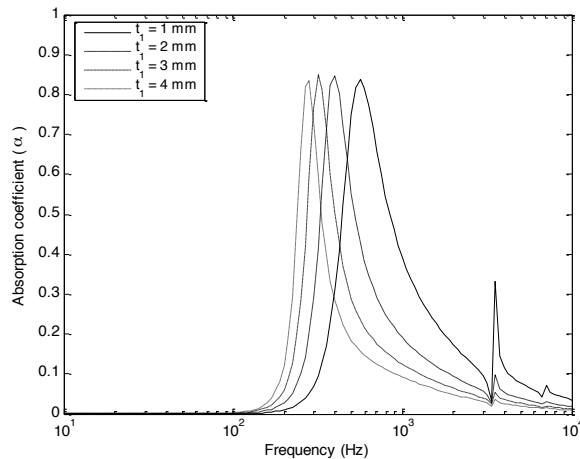


Figure 6. Absorption coefficient of NF – rigid backing system
 ($t_2 = 1 \text{ mm}$, $\sigma = 50\%$, $D = 50 \text{ mm}$, $R_f = 1.1 \times 10^4 \text{ N.s/m}^4$)

IV. CONCLUSION

The sound absorption coefficient of a NF-solid panel system has been formulated by wave propagation method for normal incident sound by considering the vibration of the panel. The model is attractive as the solid panel can be adjusted to a rigid wall by increasing the impedance of the solid panel into infinity (very large value). It is found that vibration of the NF panel widens the frequency range of the absorption coefficient. The model can also be extended to the case of oblique and diffuse field incidence which consider the bending stiffness of the NF panel.

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