TECHNICAL UNIVERSITY OF CLUJ-NAPOCA

ACTA TECHNICA NAPOCENSIS

Series: Applied Mathematics and Mechanics
Vol. 56, Issue II, June, 2013

CONSIDERATIONS ON THE ACOUSTIC PANEL ABSORBER

Iulian LUPEA, Florina-Anca STREMȚAN

Abstract: *Sound absorption is of continuous interest considering nowadays the increasing environmental sound pollution or the indoor proper conditions for activities like audition or sound recording. The panel absorber, a resonant acoustic device, is under observation in this paper. The associated differential equation of the acoustic system, followed by the acoustic impedance and the sound absorption coefficient are derived. Experimental validations are performed on a plastic suction pump, a plywood panel absorber and a box with aluminum front panel connected to the box rim by a thin latex rubber membrane. The analytical results and the experiments are in good agreement. Better results are expected by using the experimental modal analysis for the panel vibration.*
Keywords: *acoustic panel absorber, resonant frequency, absorption coefficient, frequency response function.*

1. INTRODUCTION

At low frequencies the sound absorption can be accomplished based on the phenomenon of the resonance of the absorbing devices. Two candidates of this category are currently used, one is based on the Helmholtz resonator and the second is the panel or membrane absorber. Resonant absorbers are working efficiently at the boundaries of the rooms, the corners or the walls, and are focusing on the low acoustic modes effect reduction. A second category, namely the porous absorbers, is efficient and recommended from mid to high frequencies, in general over 400-500Hz. For this porous materials (mineral wool, fiberglass, open cell foam, fibrous materials etc.) the absorption is based on friction. For the first category the frequency band of sound absorption is narrow, being more selective. On the contrary the porous absorbers are able to absorb sound energy for a wide frequency band. The membrane absorber is designed based on a thin membrane which is displaced and elastic deformed by the acoustic wave. This is working properly in general less than 350 Hz.

Resonant absorbers can be reduced to the vibration of a mass against a spring. For the Helmholtz resonator the mass is the air filling the neck of the absorbing device of length L and surface S and the spring is the larger amount of air filling the cavity of volume V (Fig. 1) [7]. On the panel absorber case, the vibrating mass is the panel or the membrane mass M and the spring is the air captured into the cavity between the membrane and a backing wall of volume V (Fig. 2). The impinging sound wave is moving the mass of the mentioned two resonant absorbers. The two absorbers by changing the amount of the vibrating mass or the spring features (changing the volume of the air captured) can be tuned based on the needed absorbing frequency. The resonant absorbers are placed at the room boundary because at the walls the pressure of

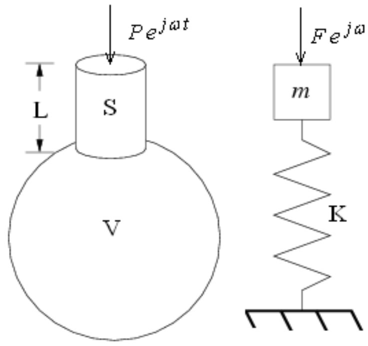

Fig. 1 Helmholtz resonator

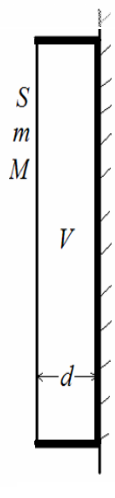

Fig. 2

the sound acoustic modes is at the maximum, hence being able to put into motion the vibrating mass. On the contrary the porous absorbers are efficient in the areas of the room where the acoustic modes are recording the minimal pressure and the air particles are at the maximum displacement.

The dissipated energy during the compression of the elastic medium is partially recovered during the rarefaction phase. The absorption of the resonators can be increased by adding a porous layer in the areas where the displacement of the air particles is at the maximum, more exactly in the air cavity close to the neck of the Helmholtz resonator and close to the vibrating membrane on the panel absorber case.

In the sequel a parallelepiped cavity is considered, with one elastic side of surface S and mass M (Fig. 2). The rest of the walls are rigid. The membrane is in contact with the air located in the cavity and the air outside of the cavity. In some cases the elastic panel and the backing wall are not parallel and therefore the air layer in of variable depth. Membrane absorbers are used mainly in broadcasting and recording studios to diminish the effect of the low order acoustic modes of the closed spaces.

2. THE MEMBRANE ABSORBER

The membrane absorber is a mass-spring-damper similar system. The vibrating mass is the flexible plate, the spring is the sealed air found between the wall, margins and the membrane and the damper is eventually the porous absorbent placed into the box.

2.1. The absorber impedance

A simplified assumption is considering the membrane translating like a piston of mass m and section S, the bending stiffness of the membrane being neglected. This displacement assumption is close to the displacement of a membrane in the first mode of vibration. The oscillating piston is alternatively compressing and rarefying the air filling the enclosure. Upon the membrane observed like a piston, the elastic (restoring) force from the captive air, a dissipative force and the active force from outside are acting.

The mass of the air is: $m_a = V\rho_0$ where V is the volume of the air and ρ_0 is the initial air density. Taking the log of each member, followed by the derivative (the mass variation is zero), results:

$$\frac{dm_a}{m_a} = \frac{dV}{V} + \frac{d\rho}{\rho_0} = 0 \qquad (1)$$

The equation of state of a gas for the adiabatic compression states: $c^2 = dp/d\rho$. The pressure variation results as: $dp = c^2 d\rho$. By taking the density variation from (1) results:

$$dp = -c^2\rho_0\frac{dV}{V} \qquad (2)$$

The volume of the captive air is less as a result of the membrane translation in positive direction: $dV = -Sx$. The force acting like a spring on the membrane or plate from the captive air side is:

$$F_e = Sdp \quad \text{or} \quad F_e = \frac{c^2\rho_0 S^2}{V}x \qquad (3)$$

where ρ_0 is the air density and c the sound velocity in air.

The resistive losses proportional to the piston velocity and due to the mounting or attaching to the support of the membrane is:

$$F_d = R_M\dot{x} \qquad (4)$$

The active excitation force acting harmonically from outside of the cavity on the membrane surface S and produced by an uniformly distributed pressure, is:

$$F_e = SPe^{j\omega t} \qquad (5)$$

By applying the Newton's law, results:

$$M\ddot{x} + R_M\dot{x} + S^2\rho_0 c^2/V \cdot x = SPe^{j\omega t} \qquad (6)$$

A harmonic solution $x(t) = ae^{j(\omega t-\varphi)}$ is proposed and plugged into the differential equation:

$$-M\omega^2 x(t) + R_M j\omega x(t) + S^2\rho c^2/V \cdot x(t) = SPe^{j\omega t}$$

The solution results of the form:

$$x(t) = \frac{1}{-M\omega^2 + R_M j\omega + S^2\rho c^2/V}\cdot SPe^{j\omega t} \qquad (7)$$

The solution is the product of the system complex frequency response function (compliance) and the excitation force. The compliance is function of the system characteristics and the frequency.

In order to get the system impedance which is important when considering the system as a sound absorber, we will differentiate the solution with respect to time:

$$\dot{x}(t) = \frac{j\omega}{-M\omega^2 + R_M j\omega + S^2 \rho c^2 / V} \cdot SPe^{j\omega t} \quad (8)$$

By dividing with $j\omega$ (1/j=-j), we get:

$$\dot{x}(t) = \frac{1}{R_M + j[M\omega - S^2 \rho_0 c^2 / (V\omega)]} \cdot SPe^{j\omega t} \quad (9)$$

The mechanical impedance is the ratio of the system excitation and the velocity response:

$$Z_M = R_M + j(M\omega - \frac{S^2 \rho_0 c^2}{V\omega}) \quad (10)$$

The expression of the specific impedance is:

$$Z_S = R_M / S + j(\frac{M\omega}{S} - \frac{S\rho_0 c^2}{V\omega}) \quad (11)$$

where the complex specific acoustic resistance and the specific acoustic reactance can be mentioned.

From the initial differential equation (6) or by setting the imaginary part of (11), the system resonant frequency is:

$$\omega_0 = \sqrt{\frac{S^2 \rho_0 c^2 / V}{M}} = c\sqrt{\frac{\rho_0}{md}} \quad \text{where}$$

$$M = mS, \quad V = Sd$$

or:

$$f_0 = \frac{60}{md} \quad \text{(for c=343m/s, } \rho_0\text{=1.21kg/m}^3\text{)} \quad (12)$$

where m is the mass of the plate or piston per unit area and d is the thickness of the air layer. This value is often inaccurate or within a 10% error. The resonant frequency approximation is for an empty cavity without absorber inside and for a membrane oscillating like a piston. An important source of error is coming from the capability to model properly the boundary conditions of the plate. For this case all the points on the plate should vibrate in phase which is the case for the first mode of vibration of the membrane cantilevered or pinned all around.

2.2. The absorption coefficient

Supposing an acoustic wave impinging normal to the membrane absorber, let us derive the absorption acoustic coefficient for the system.

The specific acoustic impedance of the air,

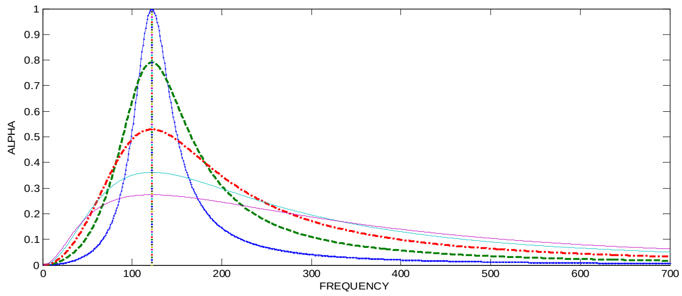

Fig.3.Sound absorption coefficient with frequency

the medium of propagation of the acoustic wave before to reach the membrane, is $\rho_0 c$. Knowing the specific impedance Z_S of the membrane absorbing system, the reflection factor can be found:

$$R = \frac{Z_S - \rho_0 c}{Z_S + \rho_0 c} \quad (13)$$

The acoustic absorption coefficient α of the system is:

$$\alpha = 1 - R^2$$

$$\text{or:} \quad \alpha = 1 - \left| \frac{Z_S - \rho_0 c}{Z_S + \rho_0 c} \right|^2 \quad (14)$$

For a membrane absorber with a plywood plate of 0.7x0.7 m^2, the density per meter square of 2.4242 and the air depth of 0.09 m, the absorption coefficient has been found. The following parameters are calculated: M=2.4242*0.7^2, the air spring stiffness $k = S^2 \rho_0 c^2 / V$ or $k = S\rho_0 c^2 / d$ and the critical absorption coefficient $b_0 = 2\sqrt{km}$.

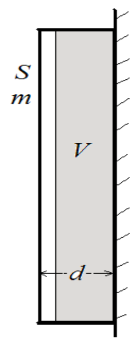

Fig. 4

Based on the relation (6) and for several proposed damping ratios $\zeta = b/b_0 = b/(2\sqrt{km})$ of 0.1, 0.3, 0.6, 1 and 1.4, the absorption coefficient variation with frequency has been derived and depicted in Figure 3. The viscous damping coefficient becomes $b = \zeta\, b_0 = R_M$. R_M may include the radiation impedance and a viscous damping coming from a porous absorbing material (like the fiberglass) placed into the cavity (Fig. 4). This material is in general attached to the back of the system.

The selectivity of the membrane absorber in terms of absorption is reduced in comparison to the selectivity of the Helmholtz resonator.

3. EXPERIMENTAL STUDIES

3.1. A plastic suction pump

A suction pump with the blocked nozzle has been employed for the study. A mini-accelerometer attached to the handle, like in Figure 5, has been used to measure the power spectrum of the recorded signal. The recorded signal is the response of the piston and handle assemble to the impulsive perturbation applied to the handle rivet by using the operator hand.

Fig. 5. Suction pump

Table 1

	Analytical		Experiment	Err.
	m =30.66 [Kg/m2]			
No	d [m]	f_0 [Hz]	f_0 [Hz]	%
1	d_max =0.19	$f_{0d\,\min}$ =24.88	26	4.3
2	d_min =0.07	$f_{0d\,\min}$ =40.9	37	10

A frequency peak of 26Hz resulted for the maximum length of the air column in the pump cylinder. For the minimum length of the air column a peak of 36 Hz has been recorded. A continuously increasing frequency peak has been observed as long as the air column gets shorter. The mass of the piston-handle assemble is M=0.077 kg (piston mass only 33.9e-3 kg). The piston area equals the air column section S=0.00256 m^2. The mass per square meter of the piston-handle assemble is: m=30.66 kg/ m^2. The theoretical resonant frequency of the system by using relation (12), is $f_{0d\,\max} = 24.88 Hz$.

The results from experimental and from analytical approaches are presented in Table 1.

For the air column maximum length the error is 4.3% and for the shorter air column the error of 10% is larger because of the fragility of the system and the air leak between the piston and cylinder.

3.2. A plywood absorber

A parallelepiped cavity with the base of 0.7x0.7 [m^2] made of plywood has been tested. The cavity is protected from air leak from or to the exterior. The mass per square meter of the plywood membrane is measured resulting m_pl=2.4242 kg/ m^2.

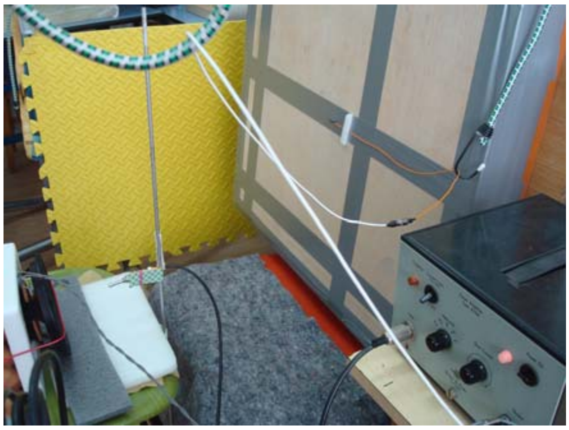

Fig. 6. Plywood membrane set-up

By using the relation (12) the resonant frequency of the sound absorbing device is:

$$\omega_0 = 343\sqrt{\frac{1.21}{2.4242 \cdot 0.1}} = 766\,[s^{-1}],$$

$$f_0 = 121.9 Hz \qquad (15)$$

For the experimental approach the plywood membrane is excited by using an impact hammer with a soft tip and the response is recorded with an accelerometer placed at the middle of the membrane. The frequency response function, inertance type, is recorded [4] and the frequency peaks are observed. From the multitude of the frequency peaks is not so easy to pick the right one hence the method is considered not enough selective for the identification of the proper peak. A similar set-up is used in which the impulsive excitation force is replaced by the sinus swept sound wave generated by a subwoofer (Fig. 6). The frequency of the swept tone is in the frequency of interest. This is an attempt to excite the membrane with the plane wave not locally like in the hammer case, but on a larger area and to excite the first mode of vibration of the system. Both methods are not enough precise to be able to select the proper frequency peak from the measured frequency response functions.

3.3. A plastic box with aluminum front panel

A plastic reinforced box with an aluminum plate connected to the box through an elastic thin latex membrane has been tested (Fig. 7). For the aluminum density of 2600kg/m^3, the plate thickness of 0.002m and the air cavity depth of d=0.1m, the resonant frequency based on relation (12) is $f_0 = 82.9 Hz$. The mass per meter square is the density multiplied by the plate thickness: m=5.2kg/m^2.

The modal frequencies of the cavity in function of the wave numbers are obtained by using the relation [3], [6]:

$$f_{n_x,n_y,n_z} = \frac{c_0}{2}[(\frac{n_x}{l_x})^2 + (\frac{n_y}{l_y})^2 + (\frac{n_z}{l_z})^2]^{1/2} \qquad (16)$$

Being interested for the modal frequencies associated to the modes perpendicular to the membrane, the particular simplified expression observed is $f_{0,0,n_z} = \frac{c_0}{2}\frac{n_z}{l_z}$ (nx=ny=0), resulting: $f_{0,0,1} = \frac{343}{2}\frac{1}{0.1} = 1715\, Hz$. Hence, the first acoustic mode frequency is much higher than the frequency band of interest and can not be excited under low incident sound frequencies. This plastic box exhibits a different resonant frequency compared to that predicted by the relation (12), because of the restoring force caused by the elastic membrane.

4. CONCLUSION

The membrane acoustic absorber from some theoretical and experimental perspectives has been considered. The differential equation of the simplified system, when the membrane is assumed to oscillate like a piston and its bending stiffness is ignored, has been derived.

Fig. 7. Plastic box set-up

The specific acoustic impedance and the coefficient of absorption of the membrane have been observed. Some experiments on a couple panel absorber systems have been performed and the best results in terms of the formula (12) validation has been obtained on the suction pump because of the free to translate boundary of the piston and of the restoring force caused mainly by the air compliance. For the plate absorbers the modes of vibration of the membrane are numerous. The higher order modes do not provide to much damping in comparison to the first mode of vibration of the plate with the specific boundary conditions. For the first mode of vibration of the plate all the points are vibrating in phase. For a higher mode some regions are moving in phase and the others are moving out of phase being not so easy to be excited by a plane acoustic wave. In order to validate with higher precision the first mode of vibration, the resonant damping and to select this mode from other modes of the system considering the boundary conditions hard to be modeled, the experimental modal analysis with excitations applied on the plate is recommended.

5. REFERENCES

[1]. Bădărău, E., Grumăzescu, M., Bazele acusticii moderne, Editura Academiei RPR, 1961.

[2]. Cox,T., D'Antonio, P., Acoustic Absorbers and Diffusers - theory, design and application, Spon press, 2004.

[3]. Everest F., Pohlmann, K., *Master handbook of acoustics*, McGraw Hill, 2009.

[4]. Lupea, I., *Vibration and noise measurement by using Labview programming*, Casa Cărţii de Ştiinţă, Cluj-Napoca, 2005.

[5]. Lupea, I., Szathmari, R., *Vibroacoustic frequency response on a passenger compartment*, J. of Vibroengineering, Kaunas, Lithuania vol. **12** (4), Dec. 2010.

[6]. Lupea, I., Tibrea, R., *Acoustic normal modes simulation of a classroom,* Acta Technica Napocensis, Applied Mathematics and Mechanics 54, vol. IV, 2011.

[7]. Lupea, I., *Considerations on the Helmholtz resonator - simulation and experiment*, Proceedings of the Romanian Academy, series A, Volume 13, Number 2/2012, pp. 118–124.

[8]. Sakagami, K., Morimoto, M., Sound absorption characteristics of a panel absorber with an air-back cavity of non-uniform depth, Kobe University Repository, 2007.

[9]. Stremtan, Florina-Anca, Garai, M., Lupea, I., *Micro-perforated panels and sound absorption*, Acta Technica Napocensis, Series: Applied Mathematics and Mechanics, Issue. III, Vol. 55, 2012, pp.549-554.

[10].** Labview 2010, National Instruments, Austin, Texas, USA, 2010.

[11].** TestExpress and TestLab, LMS International, Leuven, Belgium, 2008.

Consideraţii asupra absorbitorului acustic cu membrană

Rezumat: În lucrare sunt prezente consideraţii teoretice şi experimentale asupra absorbitorului acustic cu membrană, mecanismul de absorbţie fiind bazat pe fenomenul de rezonanţă. Plecând de la ecuaţia diferenţială asociată sistemului se pune în evidenţă impedanţa şi coeficientul de absorbţie al sistemului. Validarea experimentală a frecvenţei de rezonanţă se bazează pe măsurători asupra unei pompe de aer considerând o plajă de valori pentru înălţimea coloanei de aer. Studii experimentale s-au făcut de asemenea asupra unui absorbitor acustic cu membrana confecţionată din placaj şi asupra unei cutii cu membrana din aluminiu fixată pe contur printr-o membrană de latex. Pentru o mai bună concordanţă şi precizie de identificare prin experiment a frecvenţei de rezonanţă se propune analiza modală experimentală a plăcii sau membranei absorbitorului.

Iulian LUPEA, Professor Ph.D., Technical University of Cluj-Napoca, Department of Mechanical Systems Engineering, iulian.lupea@mep.utcluj.ro, 0040 264401691;

Florina-Anca STREMŢAN, PhD student, Technical University of Cluj-Napoca, Department of Mechanical Systems Engineering, 103-105 Muncii Blvd, 400641 Cluj-Napoca, e-mail: anca.stremtan@yahoo.com.