Infrasound transmission of non porous windscreens

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Outdoor measurement of infrasound requires the use of windscreens to reduce the noise due to aerodynamic perturbations. Classical methods are based on a spatial averaging of the pressure using a pipe array or a wind barrier. Their size is of the order of several meters, which is not compatible with portative systems. To overcome these limitations, windscreens made out of non porous polymer foam of size less than 1 meter have been studied. However the link between prediction and measurement of infrasound transmission has not been clearly established. In this paper, an analytical vibroacoustic model using the long wavelength hypothesis is derived. The considered geometry is an elastic non porous plate coupled with a cavity connected to the pressure sensor. A good agreement with finite element modeling and experimental data is found. Finally, rules to design an optimized system, related to geometry and material properties, are given.

1 Introduction

Monitoring of natural or human activities may be performed via outdoor infrasound measurement [1]. Reduction of noise coming from atmospheric disturbances requires the use of windscreens. Usual methods take benefit of the long wavelength of infrasound to average the pressure with a pipe array. However, such an array will induce a high-frequency limitation due to acoustical resonances inside the pipes [3]. Wind barriers may also help to reduce the size of the turbulences and may increase the efficiency of the array as observed by Hedlin et al. [2]. Alternative systems, which reduced size makes them portative, have been proposed by Shams et al. [4]. The so-called portative windscreens are hollow cylinders of size less than 1 meter, made out of a non porous materials. Noise reduction and infrasound transmission are tested for several geometries and materials. More recently, Zuckerwar [5] investigates theoretically the performance of such systems by determining the infrasound transmission for several geometries. The models are based on a wave propagation approach within the non porous material, but no comparison with previous experimental data [4] are presented.

The present paper aims at showing the effect of the geometry and the material properties to the infrasound transmission of such a non porous windscreen.

2 Model

The non porous windscreen is an acoustic cubic cavity whose one face is an elastic plate. Its size is less than 1 meter. The wavelength of infrasonic waves is several times the windscreen size. This allows to consider that the sound pressure on both sides of the plate p^- and p^+ is uniform. Moreover, only the first mode of the plate is considered : it is the most coupled with a uniform pressure [6]. The windscreen may be schematized by a single degree of freedom system (fig. 1), with M the mass of the plate, K the stiffness of the plate and K_a the stiffness of the cavity.

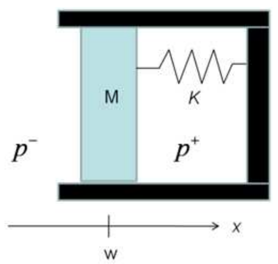

Figure 1: Schematic representation of the analytical model.

Assuming a harmonic time dependence, the second Newton law writes:

$$(K - \omega^2 M)w = S p^- - S p^+ \tag{1}$$

with $\omega = 2\pi f$ the circular frequency, S the area of the plate and p^+ the acoustic pressure inside the cavity. For an adiabatic process within the cavity, the internal pressure is related to the cavity length L by

$$p^+ = \frac{\gamma P_0}{L} w \tag{2}$$

with γ=1.4 the ratio of specific heat and P_0 the atmospheric pressure.

The transmission coefficient T_a, ratio between internal and external pressures, writes

$$T_a = \frac{p^+}{p^-} = \frac{K_a}{K_a + K - \omega^2 M} \tag{3}$$

where $K_a = S\gamma P_0/L$ is the adiabatic stiffness of the cavity. The low frequency asymptotic expression writes

$$T_a(\omega \to 0) = \frac{K_a}{K_a + K}. \tag{4}$$

Figure 2 shows a typical transmission coefficient as function of frequency. Below the resonance frequency, occuring at $f_1 = \frac{1}{2\pi}\sqrt{\frac{K+K_a}{M}}$, the response is flat.

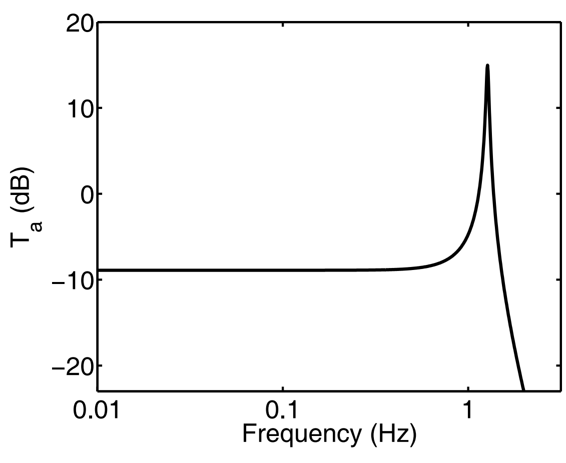

Figure 2: Magnitude of the transmission coefficient.

2.1 Mass and stiffness of the plate

Let consider a simply supported square plate of side length a and thickness h. Its mass by unit surface is $m = \rho h$ with ρ the mass density. The bending stiffness of the plate is given by $D = \frac{Eh^3}{12(1-\nu^2)}$, with E the Young's modulus and ν the Poisson ratio. Its first natural frequency is [7]

$$f_1 = \frac{\pi}{a^2}\sqrt{\frac{D}{m}}, \quad (5)$$

and its modal shape is

$$\phi_1(x,y) = 2sin\left(\frac{\pi x}{a}\right)sin\left(\frac{\pi y}{a}\right). \quad (6)$$

The modal mass M_1 is then

$$M_1(x,y) = m\int_a^0\int_a^0 \phi_1^2(x,y)dxdy = ma^2. \quad (7)$$

Considering the equivalent single degree of freedom system of natural frequency, $f_1 = \frac{\pi}{a^2}\sqrt{\frac{K_1}{M_1}}$,the modal stiffness may writes

$$K_1 = \frac{4\pi^4}{a^2}D. \quad (8)$$

For a clamped plate, considering the natural frequency of the first mode

$$f_1 = \frac{35.99}{a^2}\sqrt{\frac{D}{m}}, \quad (9)$$

and its more complex mode shape [7], the equivalent cavity stiffness is

$$K_c = \beta\frac{S\gamma P_0}{L}, \quad (10)$$

with $\beta = 0.4766$.

2.2 Transmission coefficient for a square plate

To get the transmission coefficient, the response of the plate must be derived first:

$$w(x,y) = \frac{(F_1^- + F_1^+)\phi_1(x,y)}{K_1 - \omega^2 M_1}, \quad (11)$$

with the modale force F_1^- corresponding to a uniform external pressure

$$F_1^- = p^-\int_a^0\int_a^0 \phi_1(x,y)dxdy = p^-\frac{8a^2}{\pi^2}, \quad (12)$$

and $F_1^+ = -p^+\frac{8a^2}{\pi^2}$ for the uniform internal pressure.

The mean displacement of the plate is given by

$$\bar{w} = \frac{1}{a^2}\int_a^0\int_a^0 w(x,y)dxdy = \frac{p^- - p^+}{K_1 - \omega^2 M_1}\frac{64}{\pi^4}. \quad (13)$$

Introducing $\bar{w} = p^+\frac{L}{\gamma P_0}$ (eq. 2) in equation 13, the transmission coefficient writes

$$T_a = \frac{p^+}{p^-} = \frac{K_s}{K_s + K_1 - \omega^2 M_1}, \quad (14)$$

with the equivalent cavity stiffness acting on the simply supported plate

$$K_s = \frac{64}{\pi^4}\frac{S\gamma P_0}{L}. \quad (15)$$

3 Validation

The studied configuration (figure 3) is a non optimized cubic windscreen with one elastic wall and five concrete rigid walls. The elastic wall is made up with a closed-cell polymer foam (ρ= 132 kg.m^{-3}, E= 57 MPa and ν=0.35) plate of 12.5 mm thickness and 300 mm side length. The plate edges are bonded to the rigid wall on a narrow surface. The free surface dimension are 295 mm by 295 mm. A nozzle is added in order to connect the sensor to the inner volume of 16 dm^3.

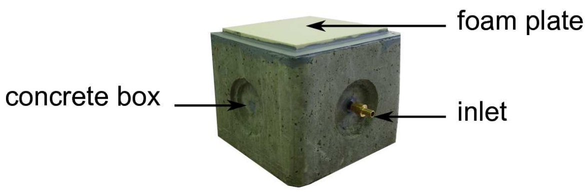

Figure 3: Studied system.

3.1 Numerical validation

In order to verify the validity of the analytical model, a comparison with a finite element model (FEM) has been performed with Rayon® code now implemented in VA One™ software. 2500 quadrangular linear elements were used for the plate mesh. The acoustic domain mesh was composed of elements whose size was approximately 0.01 m. The load was applied on the nodes corresponding to the free surface and was defined by an uniform pressure distribution. To obtain the response of the coupled system, only the first 15 modes were accounted for. Figure 4 shows an excellent agreement between analytical and FEM results for the two boundary conditions of the plate.

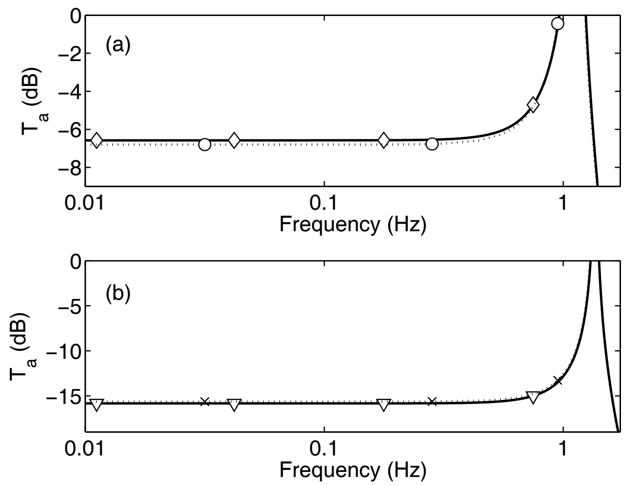

Figure 4: Magnitude of the transmission coefficient simulated by FEM and analytical model for two load cases: (top) pinned plate and (bottom) clamped plate. (circle marker) analytical pinned plate, (diamond) FEM pinned plate, (triangle) analytical clamped plate, (cross) FEM clamped plate.

3.2 Experimental validation

The measurements were performed at the laboratory of the French Department DASE of CEA. The infrasound waves were generated using a loudspeaker in the range 3 Hz to 10 Hz. Pressure is monitored by three microbarometers (MB2005 Martec): MB1 is connected to the windscreen, MB2 is linked to a gridded inlet and MB3 is sealed in order to know the instrumental and seismic noise. In this configuration, the incident pressure was measured by MB2 whereas the transmitted pressure was measured by MB1.

Transmission coefficient is obtained from the difference of pressure level measured between the outside and inside of the windscreen, at frequencies corresponding to the emergence of the acoustical signal. The results are presented figure 5 and compared with analytical predictions. It is seen that the experimental results are not frequency dependant. The values fall between analytical curves, but are closer from the model assuming clamped boundary conditions. Indeed, the plate have been bonded on a significant surface over its entire periphery, avoiding a free rotation.

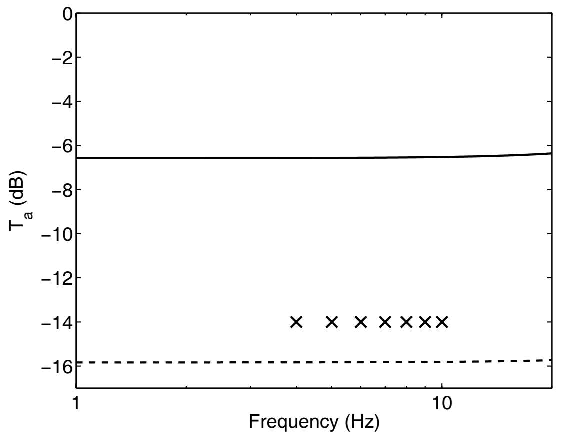

Figure 5: Magnitude of the transmission coefficient: experimental results (cross), analytical predictions with pinned plate assumption (solid line) and clamped plate assumption (dashed line).

4 Design rules

Table 1 shows the influence of geometrical parameters and material properties on the transmission coefficient in the low frequency asymptotic regime. To increase the transmission, the stiffness of the plate has to be lowered: this is achieved by increasing the surface or the density, by decreasing the thickness or the Young's modulus. It is also possible to reduce the thickness of the air cavity.

Table 1: Magnitude of the simulated transmission coefficient at low frequency for several configurations.

Configuration	Transmission coefficient T_a (dB)
Reference	-6.5
Double sized	-0.5
Double air gap thickness	-10.0
Double wall thickness	-20.0
Double Young's modulus	-10.0
Double density	-6.5

5 Conclusion

This paper examined the acoustic transmission through a non porous windscreen. In this case, the acoustic transmission mechanism is the ability of the incident pressure to deform the non porous wall and modify the volume of the air gap confined in the windscreen. An analytical model, based on the assumption that the pressure is uniform inside and outside the windscreen, has been derived. In the considered low frequency range, it is shown that the coupled wall-cavity system may be modeled by a single degree of freedom system. The model has been succesfully compared with FEM simulations and measurements performed on one system. The effects of system dimensions and materials have been enlighten and design rules given. Further works will focus on the efficiency of the windscreen to reduce noise due to atmospheric turbulences.

Acknowledgments

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