PSYCHOMECHANICAL STUDY OF SOUNDS RADIATED BY FLUID-LOADED VIBRATING PLATES

Pacs : 43.40 R, 43.66 J, 43.66 L, 43.66 W

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ABSTRACT The sound radiated by a fluid-loaded vibrating plate is analysed, as a function of its mechanical and geometrical parameters, and for a variety of excitations. A method of resonance modes in the time domain is used to calculate the acoustic field. The sound is then analysed both from a physical and psychoacoustical point of view. Psychoacoustical evaluation is done through dissimilarity tests and MDS analysis, and by direct estimation of pleasantness. The analysis aims at identifying the auditory attributes and associated physical parameters that are responsible for the quality of the sounds generated by the vibration.

INTRODUCTION

An increasing number of studies can be found that associate physical and psychoacoustical approaches to sound generation or control. In this latter field, an original proposition is made by Sommerfeld and Samuels (2001) to improve noise-control algorithms by taking into account the human auditory response to sounds. Instead of restricting the active reduction to the sound pressure level only, they suggest to add its loudness level as a criterion for minimization. In a somewhat similar attempt to evaluate the perceived quality of noise attenuation through walls, Marquis-Favre and her colleagues (2001) used auditory testing to analyse the influence of various physical parameters of a steel plate on the transmitted sound. They drew some useful conclusions about the role of density, Young modulus, thickness and damping, both in terms of transmission loss and of preference as judged by a group of listeners.

A related issue is the relationship between the mechanical properties of vibrating structures and the perceptual attributes of the sounds generated by the corresponding vibrations. Independently of any preference judgment, it is of great interest to understand how a specific mechanical parameter of a structure stimulates the auditory system, and to describe the perceptual attribute that it generates. One of the pioneer work was published by Freed (1990). His study was concerned with perceived mallet hardness. From recordings of metal objects being struck with various percussion mallets, he extracted four acoustical parameters and evaluated them as possible predictors of perceived hardness. Another thorough investigation was done by McAdams (2000) and his group (Roussarie, 1999; Houix et al., 1999). Using a model originally created by Chaigne and Lambourg (2001), they examined the various perceptual attributes of synthesized waveforms simulating sounds radiated by struck bars and plates. One of the aims was to identify the possible links between the mechanical behaviour of the structures and the corresponding perceptual images. Bars with constant or variable cross-sectional geometry were tested, with different densities and damping factors. Through a multidimensional scaling of dissimilarity ratings, it was possible to visualize the perceptual space in which the sound events were located relative to one another, and the dimensions of this space corresponded closely to the mechanical properties under study. Finally, one should mention the contribution by Lutfi and Oh (1997) and by Lutfi (2001), respectively on auditory discrimination of synthesized sounds simulating material changes in a struck-clamped bar, and on auditory detection of hollowness of freely vibrating bars.

The study reported in the present paper is a continuation of a preliminary work on auditory attributes and quality of sounds radiated by vibrating structures (Meunier et al., 2001). We examine the case of brief sounds created by transient excitation of a thin baffled plate. A new method is used to model the vibrations created in the plate and to calculate the acoustic field. A series of sound files are synthesized, that contain the radiated sound pressure as a function of time. These files serve as test sounds in a psychoacoustic investigation, on the intrinsic quality of the sounds and on their dissimilarities. A classical multidimensional analysis is then run, to determine the perceptual space in which the sounds are distributed. From this distribution, the main dimensions of the space can be determined, and the relevant perceptual dimensions identified. The final step is to correlate these perceptual dimensions and the sound quality with the mechanical characteristics of the plate.

ACOUSTIC AND VIBRATORY RESPONSE OF A FLUID-LOADED PLATE

This section briefly presents some of the main features of a theoretical and an experimental study of the response of a fluid-loaded plate. The theoretical part describes the method used to compute the sound signals which are listened to in the hearing tests. The experimental study shows the effect of the excitation on the radiated sound pressure. A comparison between these experimental results and numerical calculations, in another series of hearing tests, will be published in a forthcoming paper.

From a theoretical point of view, because of the coupling between the fluid and the plate, there are two unknown functions: the displacement on the surface of the structure and the sound pressure radiated in the fluid. To compute the sound signals radiated by the plate, we have chosen a method based on the resonance modes of the fluid-plate coupled system. It gives explicit expressions of the displacement and the sound pressure as functions of time (Filippi et al. 2001). The resonance modes w_n take into account the coupling between the fluid and the structure. They are the non-zero solutions of an eigenvalue equation which is nonlinear. For each mode, the corresponding resonance frequency has an imaginary part which is different from zero and describes the damping of each mode. These damping factors are proportional to the radiation of the modes in the fluid. This method has been developed for plates and shells (Filippi et al., 2001). In the case of thin, baffled plates, the expressions are quite simplified.

The displacement U(M,t) on the plate can be written as:

$$U(M,t) = -i\sum_{n=1}^{\infty}\left[\boldsymbol{a}_n w_n(M)\exp(-i\boldsymbol{w}_n t) + \overline{\boldsymbol{a}}_n \overline{w}_n(M)\exp(+i\boldsymbol{w}_n t)\right]\exp(-\boldsymbol{t}_n t)$$

where $\Omega_n = \boldsymbol{w}_n - i\boldsymbol{t}_n$ are the resonance frequencies. $\overline{w}$ represents the complex conjugate of w. In the particular case of a light fluid (air), the resonance modes w_n can be approximated at first order by the resonance modes of the *in vacuo* plate w_n^0. In the case of a thin plate, the coefficients $\boldsymbol{a}_n$ of the previous series can also be approximated by:

$$\boldsymbol{a}_n = \frac{w_n}{2}\frac{<\widehat{F},w_n^0>}{a(w_n^0,w_n^0)}$$

where $\widehat{F}$ is the Fourier transform of F, the mechanical excitation. The scalar product <f,g> of two real functions f and g is defined as the integral of their product on the surface of the plate. In the case of a baffled plate and if the displacement and its derivative are continuous functions, the sound pressure can be expressed as a function of the displacement by a Rayleigh integral on the surface of the plate:

$$p(Q,t) = \frac{m}{p}\int_{\Sigma} \ddot{U}(M, t - R/c)\frac{1}{R}\,ds\,(M)$$

Ü is the second derivative of U with respect to time t. m is the density of the fluid and R=R(Q,M) is the distance between two points Q and M. By using these formulas, it is possible to predict the effect of the geometrical and mechanical parameters of the structure and the excitation.

In the study presented here, the sound signals were computed for a thin baffled steel plate, clamped on its boundaries, with dimensions Lx=0,35m, Ly=0,50m and h=5mm. Each signal is computed with a sampling frequency of 44100Hz for a duration of 1,5s. The upper frequency of the spectra was set to 8000Hz; the first mode of the plate is around 274Hz. The sound pressure is computed at 1m from the plate. In order to point out the perceptual effect of the variation of the physical parameters, we have chosen three parameters: the location of the excitation point, the amount of structural damping and the duration of the impact. From the set of signals computed, sixteen were selected for hearing tests. The values of the parameters for these sixteen signals are shown in Table 1; in the second column, the origin is taken at one corner of the plate. Let us remark that some of the values chosen for the structural damping are quite high; they were chosen to point out more easily a perceptual effect, if any. The results of the hearing tests and their analysis are presented in the next section.

Signal number	Excitation point	Structural damping	Duration of the excitation
1	(Lx/2,Ly/2)	0.0001	0.1 s
2	(Lx/3,Ly/3)	0.0001	0.1
3	(Lx/4,Ly/4)	0.0001	0.1
4	(Lx/5,Ly/5)	0.0001	0.1
5	(Lx/3,Ly/2)	0.0001	0.1
6	(Lx/5,Ly/5)	0.0008	0.1
7	(Lx/5,Ly/5)	0.01	0.1
8	(Lx/5,Ly/5)	0.005	0.1
9	(Lx/5,Ly/5)	0.001	0.1
10	(Lx/5,Ly/5)	0.003	0.1
11	(Lx/5,Ly/5)	0.0003	0.1
12	(Lx/2,Ly/2)	0.0008	0.1
13	(Lx/5,Ly/5)	0.0001	0.5
14	(Lx/5,Ly/5)	0.0001	1
15	(Lx/5,Ly/5)	0.0001	2
16	(Lx/5,Ly/5)	0.0001	5

Table 1. Physical characteristics of the signals.

In parallel, an important point is also to check how the synthesized waveforms compare, from a perceptual point of view, with sounds taken from an experiment. For this purpose, we have measured in an anechoic room the response of a thin baffled steel plate, clamped along its boundaries. Excitation was obtained by hitting the plate with small bullets of various dimensions and shapes (small and bigger spheres, ellipsoid) and various materials (lead, wood, polyurethane,...). Acceleration was measured at a point near the corner of the plate and the radiated sound pressure was recorded at two points in front of the plate.

Figures 1 and 2 present two examples of results obtained for an excitation of the plate by a polyurethane sphere and a lead ellipsoid. They represent, in relative amplitudes, sound pressure as a function of time (Fig.1) and sound level as a function of frequency (Fig. 2). In both spectra, the peaks correspond to the resonance frequencies of the plate. The influence of impact appears in the general shape : For the polyurethane sphere, only the very first modes of the plate are excited. For the lead ellipsoid, the spectrum extends over a much broader range, with a notch around 4kHz.

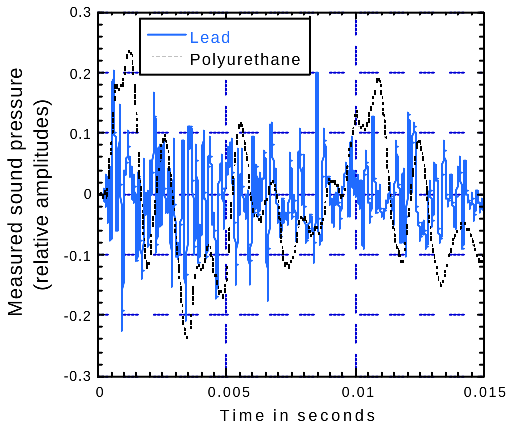

Figure 1: Measured sound pressure as a function of time.

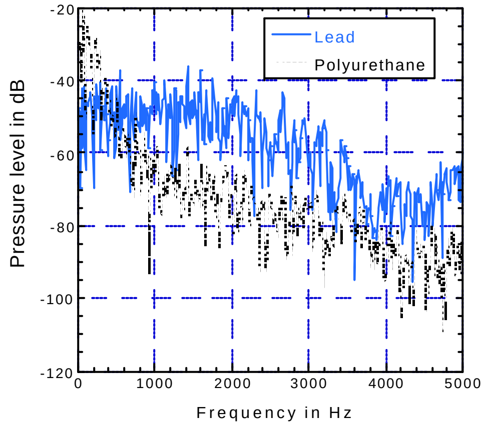

Figure 2: Measured spectra.

SIMILARITY AND PLEASANTNESS

A measure of similarity and pleasantness was run on these sounds. The aim of the similarity test was first to place the sounds in a perceptual space, then to determine the perceptual meanings of the principal dimensions of that space, and finally to infer the influence of the mechanical and/or acoustical parameters on the perception of the radiated sound signals. The pleasantness of the sounds was also tested to find out whether sound quality could be related to the three mechanical parameters described in Table 1. Seventeen subjects took part in the experiments.

<u>Similarity test</u>

Sounds were presented in pairs, in random order, to the listeners. These were asked to judge the similarity between the signals of each pair by locating a cursor on a line displayed on their response terminal, the two end points of the line being labelled "very similar" and "very dissimilar". A multidimensional analysis (MDS) was run on the dissimilarity matrix, built from similarity data, using Statistica sofware. The MDS computed by Statistica is a non-metric unweighted MDS, based on Kruskal's model (1967). A three-dimensional solution was found to be most appropriate, as a result of the analysis of stress-scree elbow and interpretability. The relative location of the sounds in the plane of the first two dimensions is shown on figure 3.

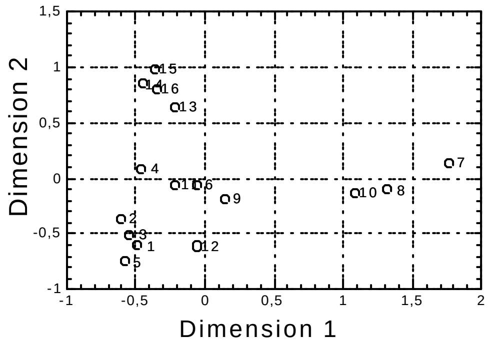

Figure 3: Spatial distribution of sounds 1 to 16 (unfilled circles) along dimensions 1 and 2 as revealed by MDS. The sound numbers are indicated next to the symbols.

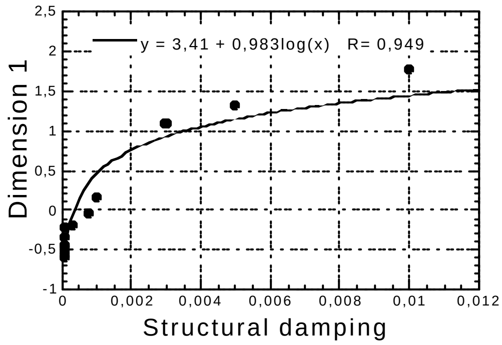

Figure 4: Dimension 1 as a function of structural damping. Curve fit equation and correlation coefficient R are given at the top of the figure.

Dimension 1 appears to relate to structural damping. On figure 3, one can observe a clear separation between sounds with higher structural damping (sounds 7, 8 and 10) on the right, and sounds with lower structural damping on the left. A plot of dimension 1 versus structural damping reveals a relationship between the two variables that can be modelled by a logarithmic function, as displayed on figure 4. Conversely, this figure shows how structural damping should be set to create perceptually-equidistant sounds.

Dimension 2 would appear to relate to the timbre of the sounds, that is to the spectrum. Sharpness was calculated for all signals. As they were non-steady sounds, sharpness was calculated on 100-ms samples. Averages of the first two samples, first three, etc... have been calculated, and it appears that dimension 2 is well correlated with the average of sharpness on the first two samples, that is on the first 200 ms (figure 5). Thus, subjects do not use the whole signal to make their similarity judgments on the basis of sharpness : the first two hundreds of milliseconds are enough.

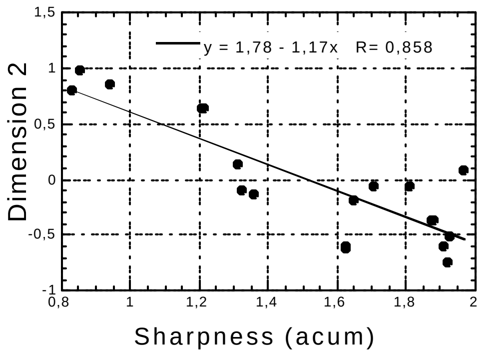

Figure 5: Dimension 2 as a function of sharpness, averaged over the first two samples of 100 ms.

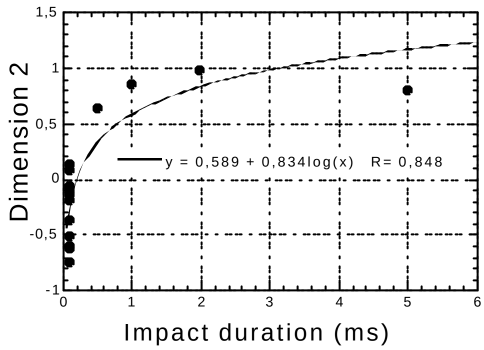

Figure 6: Dimension 2 as a function of impact duration.

The variation of the sound spectra is linked to the variation of the duration of impact (sounds 13 to 16 correspond to increasing impact durations) and to the point of excitation (this point varies, for sounds 1 to 5, from the centre to a corner of the plate). Indeed, the sounds with short impact duration have high-frequency components. Sounds 1, 5 and 12 correspond to an excitation at the centre of the plate, for which less modes are excited, and therefore they have less partials in their spectrum. Thus, dimension 2 could be related to these mechanical parameters. Figure 6 shows the relationship between dimension 2 and impact duration.

A 1/3rd-octave wavelet analysis of the signals was used to identify the acoustic attribute(s) of dimension 3. This time-frequency representation shows a notch around 400 Hz for some signals, this is due to the location of the impact. Indeed, dimension 3 seems to relate to this energy suppression in this part of the spectrum. For example, sounds 5, 1 and 12, which have the lowest coordinates along dimension 3, have no energy around 400 Hz, in contrast with sounds 2, 3, and 13 (which have the highest coordinates) for which the energy is fairly high. However, this relationship does not hold for all sounds, and we still have to find the complete signification of dimension 3.

Pleasantness test

In the second experiment, sound quality of these signals was evaluated. The values of the mechanical parameters used in the simulation produce non harmonic signals, that sound like bells. The aim was to find the mechanical and/or acoustical parameters influent in the resulting sound quality. The method used to assess sound quality was direct magnitude estimation of pleasantness without reference (Stevens, 1975). It was observed that pleasantness is correlated with dimension 1, and therefore with structural damping. This is shown in figure 7, where the geometric means of pleasantness are plotted as a function of structural damping.

Pleasantness is neither correlated with dimension 2 nor with dimension 3. Thus, changing the spectrum shape of the sounds (at least within the frequency range of our study) has no effect on their quality, neither does the energy notch around 400 Hz. Thus, the impact duration and the position of the excitation are parameters of the timbre of the signal but not of its quality.

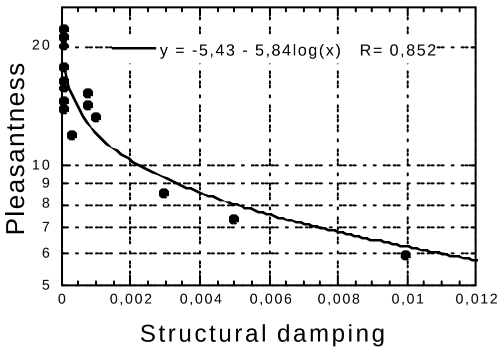

Figure 7: Pleasantness as a function of structural damping.

CONCLUSION

In this paper, we have shown, in the case of a fluid-loaded plate, how the mechanical and geometrical characteristics can be related to perceptual parameters of the sounds radiated by the plate. The whole study is based on three specific tools: A theoretical model to compute the sound pressure radiated by the plate, a selection of hearing tests and a statistical analysis of the results. It is demonstrated that structural damping is a prominent factor for both dissimilarity and preference. The point of excitation, and the impact duration, have an effect only on the dissimilarity because they only affect the timbre (or spectral content). The next step will be to develop a comparison between experimental and computed signals, using both physical and perceptual criteria.

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