A comparative study of recorded and computed sounds radiated by vibrating plates

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This study concerns the auditory properties of sounds radiated by a fluid-loaded vibrating plate excited by a transient point force. It is related to the field of psychomechanics, which is a combination of vibroacoustics and psychoacoustics. The object of psychomechanics is to establish correlations between the mechanical parameters of a vibrating structure, the acoustic field generated by its vibrations and the auditory attributes of the corresponding sounds. The particular aim of the present study is to estimate the validity of a model designed to simulate the acoustical radiation of a vibrating plate. The model is based on the development of resonance modes. We consider the case of a clamped plate excited by an impact hammer. Vibroacoustic measurements were done, in an anechoic chamber (excitation, acceleration of the plate and acoustic pressure); they provide the entry data for our model. Next, dissimilarities between recorded and synthesized sounds were evaluated from a physical (temporal and spectral) point of view. Further dissimilarity tests were run to determine how many modes have to be taken into account in the calculations, an important issue in terms of computation time.

1 Aim of psychomechanics

Pollution is one of the dominant problems of the 21st century. Notably, the increase in road, air and rail traffic leads to the increase in noise pollution. One common way to fight against noise is to reduce sound-source level via passive or active control. Another solution is to modify the mechanical or geometrical properties of vibrating structure in order to reduce the loudness of these sounds. However, the modification of vibrating-structures properties can lead to an improvement of sound quality too, an increase of pleasantness. This aspect is taken into consideration by automobile industrials for sound quality improvement of door banging, for example.
To a certain extent, one of the aims of psychomechanics is to modify mechanical or geometrical properties of vibrating structures in order to improve sound quality. Globally, its aim is to establish the links between the parameters of a vibrating structure, the acoustic field radiated by its vibrations and the perceptual aspects of this sound. This discipline, combination of vibroacoustics and psychoacoustics, is relatively recent (about 15 years). Psychomechanic investigations mainly concern the effect of parameter variations of bars or plates. McAdams et al. [1] studied dissimilarities between synthesized sounds radiated by variable cross-section bars (xylophons). Bars varying in mass density and viscoelastic damping coefficient were stricked by a mallet. Canevet et al. [2] worked on synthesized sounds radiated by plates excited by a transient point force. They studied the effect of the variation of structural damping, duration and location of the impact on perceived dissimilarities and pleasantness. The case of white noise transmitted through clamped plates was studied by Faure and Marquis-Favre [3]. They examined dissimilarities and preferences between sounds radiated through plates varying in structural damping, thickness and Young's modulus. Stoelinga et al. [4] studied plates impacted by bouncing balls. On the one hand, they presented estimation of naturalness of sounds as a function of the "restitution coefficient" and on the other hand, they asked the listeners to judge the size of bouncing balls. Another aspect of psychomechanics is to test whether it is possible to simplify a complex model of vibrating structure by an other simpler that would be equivalent in terms of auditory perception. Demirdjian et al. [5] showed that it is possible to simplify a variable-thickness plate by a constant one which is equivalent from a perceptual point of view.
The object of the present paper is to compare recorded and synthesized sounds. These sounds are radiated by clamped vibrating plates impacted by an impact hammer. Thus, we would like 1) to identify physical dissimilarities between recorded and computed sounds radiated by the plate and 2) to know if these dissimilarities are perceived by listeners. In order to answer these questions, excitations and sounds were firstly recorded in an anechoic chamber. The experimental procedure is presented in part 1. Secondly, the respective sounds were calculated by the method of resonance modes. Then, spectral and temporal dissimilarities between both types of sounds are described in part 2. The last part finally presents the results of perceptual experiments. The aim of this last part is to know which frequency range should be taken into account for our model so that recorded and computed sounds are more similar.

2 Experimental procedure

2.1 Experiment

The study concerns the particular case of a baffled rectangular plate with the following dimensions $89.3 \times 75.5 \times 0.2\ cm$. The plate is elastic, clamped and made of steel. The density of the material $\rho_p = 7800\ kg.m^{-3}$. The Young's modulus E is equal to $2.10^{11}\ Pa$ and the Poisson's ratio is $\nu = 0.3$. These values are not known precisely. The plate was clamped in a rigid partition which separates two anechoic chambers. The plate was excited by an impact hammer with a rubber head. The head had a sensor which permitted to measure the impulse as a function of time. A microphon recorded sound pressure at one point of one of the anechoic chambers. The plate was excited at 3 different locations. One of the impact location is between the center and one corner of the plate (point 1), another is close to the center (point 2) and the last is close to one corner (point 3). Respective sounds will be called sound 1, sound 2 and sound 3.

2.2 Characteristics of the sounds

The recorded sounds had durations around 8 s. In order to reduce the duration of the auditory tests, only the first 5 seconds were used. In the time domain, sounds can be seperated in two parts : the first is the transient part (0-100ms) and the second one is the resonant part (100ms-5s). The transient part carries the most important energy of the signal as we can see in figure 1. The duration of the impact is around 4 ms so the main energy in the resonant part is concentrated below 1000 Hz. The first resonance frequency is around 25 Hz.

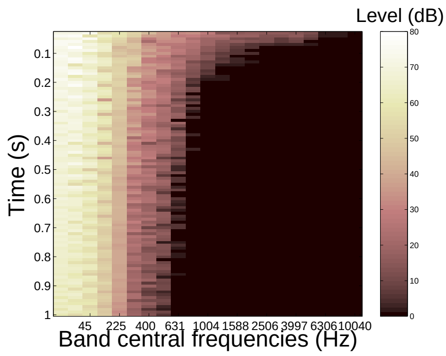

Figure 1: Time frequency representation of sound 2

3 Physical Analysis

The aim of the study is firstly to identify dissimilarities between recorded and computed sound signals and secondly to know if they are audible. This part presents physical differences between recorded and computed signals. In section 3.1, the exact values of the structural parameters will be determined comparing recorded to synthesized sounds. Indeed, structural parameters are not precisely known, the aim of this section is to go back to these parameters from the signal radiated by the plate. This identification permits us to calculate sounds which correspond to the recorded one. Temporal and spectral dissimilarities between both types of sounds are quantified in section 3.2.

3.1 Matching of structural parameters

3.1.1 Structural damping

For our model, structural damping η is introduced in the imaginary part of the Young's modulus of the plate. To define the value η that we should introduce in our numerical model, we studied the decrease of sound-pressure level as a function of time in each critical band. First, we calculated the slope α of the level decrease over the 3-5s interval of each sound signal in each critical band. Afterwards, we identified the frequency fc component which has the most important energy per critical band. The respective structural damping is given by expression 1 for each critical band.

$$\eta = 2.3 \times \alpha/(20\pi f_c) \tag{1}$$

We thus found that, the value of η varies from $0.6\ 10^{-3}$ to $1.8\ 10^{-3}$ for the whole sounds and the different critical bands. The values differ with a factor equal to 3. This variation is reasonable for our model of structural damping. For example, for sound 3, η is equal to $1.4\ 10^{-3}$ for the first critical band, $0.7\ 10^{-3}$ for the second, $0.7\ 10^{-3}$ for the third and $1.1\ 10^{-3}$ for the fourth. We determinated a global value of structural damping for sound 2 equal to $1\ 10^{-3}$. This last value is the mean of the four values. In this way, η is equal to $1\ 10^{-3}$ for sound 1 and 3.

3.1.2 Thickness

Thickness of plate is a relevant parameter (see [5] and [3]) to take into account from a perceptual point of view. Thickness of the real plate is not well defined because it is not exactly constant. The resonance frequencies of the plate, which are the different components of the sound spectrum, are linked to thickness. Thereby, a variation of 20% of the plate thickness implies a variation of 20% of the resonance frequencies. These resonance frequencies are linked to E, ρ_p, ν and the dimension of the plate. These last parameters are not defined accurately. We choose to match the resonance frequencies of the plate by modifying the thickness of the plate.
Sounds were calculated with the entry data defined in

part 2.1 and 3.1.1. The first comparison between recorded and computed acoustic signals on the first 100ms showed that computed signals have higher frequency components. We thus try to match the two spectrum by reducing the value of the plate thickness in the model. The best fit corresponds to a coincidence of the recorded frequency with the greater energy and the synthesized frequency with the greater energy. We identified an equivalent thickness of 1.92 mm for sound 3 and 1.86 mm for sounds 1 and 2. We identified two different values because the maximum of spectral energy of sound 3 is located at a frequency which is different from sound 1 and 2.

3.2 Physical differences

New sounds were computed with the parameters defined in part 3.1.1 and 3.1.2 and were normalized. The method based on the development of resonance modes was used to compute these sounds. Theoretical details of the method are described by Habault and Filippi [6]. Habault and Filippi [7] studied the case of a plate excited by an impact hammer. They worked on the comparison between numerical and experimental of sound pressure signals. They showed that the first 60 ms of recorded and computed signals are quite similar. Our study concerns 5 seconds of sounds signals. Temporal and spectral dissimilarities are presented.

3.2.1 Temporal differences

Sound were calculated with the structural damping and the thickness defined in section 3.1.1 and 3.1.2. Figure 2, 3 and 4 present temporal envelope of the three pairs of recorded (thick) and computed (thin) sounds. On the whole, the envelopes of sound pairs are similar. More precisely, temporal signals can be seperate into three parts : the transient part (0-100 ms), the intermediate part (100ms-3s) and the damped part (3-5s).

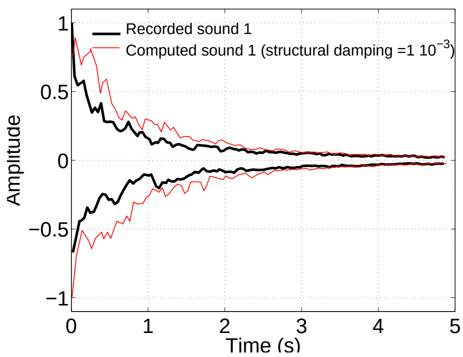

Figure 2: Envelope of normalized amplitude of sound 1 as a function of time

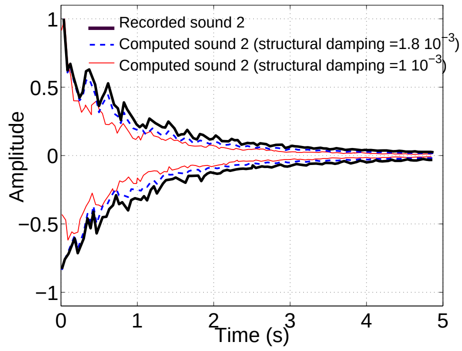

Figure 3: Envelope of normalized amplitude of sound 2 as a function of time

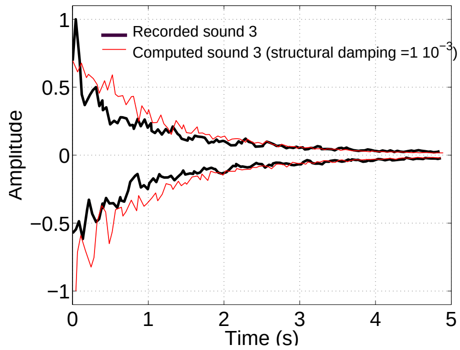

Figure 4: Envelope of normalized amplitude of sound 3 as a function of time

In the transient part, the maximum of amplitude is different for recorded and computed sounds 1 and 3 because of the normalization. In the intermediate part, modulations in amplitude seem differ for recorded and computed sounds. This is due to the fact that two resonances with strong energy produce fluctuation strength (caracteristics of fluctuation strength are presented by Fastl [8]). These fluctuations strength are different for recorded and computed sounds because the proximity of the two frequencies is different. In the resonant part, the decrease seems to be equivalent except for sound 2 where recorded sound is damped more quickly. In figure 3, the amplitude of synthesized sound 2 with a structural damping equal to $1.8\ 10^{-3}$ is presented (dashed). This last synthesized sound corresponds to a better decrease.

As a conclusion, in part 3.1.1 a value of structural damping equal to $1\ 10^{-3}$ was identified for each sound. Recorded sounds and sounds computed with this structural damping seem to have the same temporal decrease except for sound 2. The best matching for this last sound is satisfied for a structural damping equal to $1.8\ 10^{-3}$. It would be interresting to evaluate what value of structural damping ($1\ 10^{-3}$ or $1.8\ 10^{-3}$) would be more relevant from a perceptual point of view.

3.2.2 Frequency differences

In part 3.1.2, we have identified an equivalent thickness of plate equal to 1.86 mm for sound 1 and 2 and another equal to 1.92 mm for sound 3. Now, we study the difference between the measured frequency and the frequency computed by our model with a thickness equal to 1.92 mm. Figure 5 presents relative frequency differences (for each n^{th} resonance frequency) as a function of measured resonance frequency : $\Delta f/f = (f_n^{mes} - f_n^{synt})/f_n^{mes}$ versus f_n^{mes}; f_n^{mes} and f_n^{synt} are respectively the n^{th} measured and synthesized resonance frequencies.

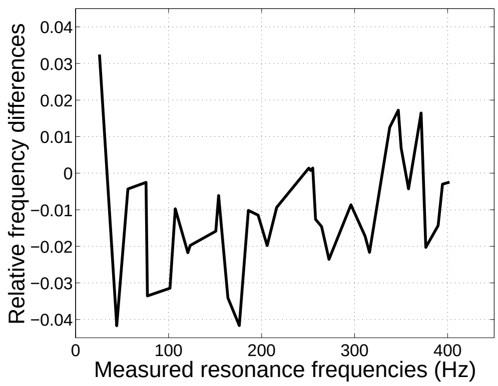

Figure 5: Relative frequency differences between measured and synthesized frequencies corresponding to sound 3 as a function of measured resonance frequencies

Firstly, the maximum of energy of recorded sound 3 is located at 75 Hz. That is the reason why in figure 5, we can see that $\Delta f/f$ is close to zero at 75 Hz. Secondly, $\Delta f/f$ varies from $4\ 10^{-2}$ (for the fundamental 25 Hz) to zero at 75 Hz. These values are greater than the auditory threshold for frequency discrimination which is approximately 10^{-2} to 10^{-3} in this frequency range. Consequently, these frequency differences will probably be slightly perceptible.

3.2.3 Spectral level differences

Frequency dissimilarities were quantified in part 3.2.2. Now, we will try to evaluate level dissimilarities between recorded and computed sounds of each excited resonance frequency, for the three different parts of the signals defined in the section 3.2.1. To do so, sound spectra were computed on these three parts for both recorded and computed sounds. In this regard, figure 6 shows level differences $\Delta N = N_n^{mes} - N_n^{synt}$ as a function of measured resonance frequencies for the transient part of the three sounds. N_n^{mes} and N_n^{synt} are respectively the levels of recorded and computed sound at each excited resonance frequency n. Figure 7 concerns the intermediate part and figure 8 the damped part of the three sounds.

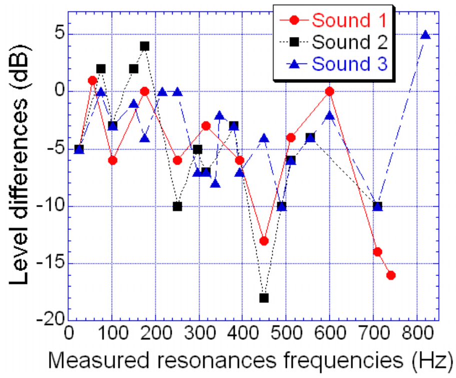

Figure 6: Spectrum level differences computed between 0 and 100 ms as a function of measured frequencies

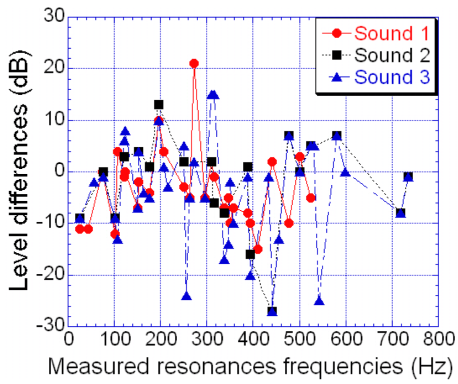

Figure 7: Spectrum level differences computed between 100 ms and 3 s as a function of measured frequencies

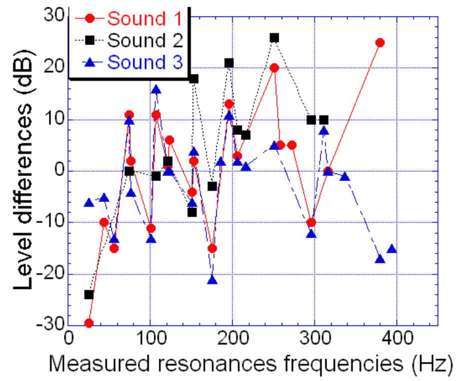

Figure 8: Spectrum level differences computed between 3 s and 5 s as a function of measured frequencies

Concerning the transient part (figure 6), $|\Delta N|$ is close to 5 dB from 0 to 200 Hz. ΔN is more important (-18<ΔN<18 dB) and globally negative in the frequency range of 200-800 Hz. In fact, at each resonance frequency, the levels of recorded sounds are smaller than the levels of synthesized sounds. The spectrum of excitation

is probably incorrectly defined in this frequency range.
In intermediate part (figure 7), ΔN globally are more important than in the transient part ($\Delta N_{intermediate} \simeq 2 \Delta N_{transient}$).$|\Delta N|$ is less important from 0 to 200 Hz than from 200 to 800 Hz for the same reason than in the transient part.
In the damped part of the signal (figure 8), ΔN varies from -30 to 30 dB. These values are very high. It is probably due to the structural damping. In fact, in our numerical model, we choose a global value of $1\ 10^{-3}$ (see section 3.1.1) whereas it varies in each Bark band. Another comment concerns the first resonance frequency (25 Hz) : $\Delta N = -30Hz$ in the case of sound 1 and 2. The energy of radiation (which is take into account in the imaginary parts of the resonance frequencies) of the first mode of the plate seem badly defined in our model.
To sum up, until 3s, spectrum levels of recorded and computed sounds are close below 200 Hz but they are further above 200 Hz. Nevertheless, the levels of sounds are lower from 300 to 800 Hz so that the differences will probably be less perceptible.

4 Psychoacoustic experiments

Our model is based on the development of resonance modes. The more mode we take into account, the longger the calculation time is. Then, the aim of this part is to know which frequency range should be taken into account for our model so that computed and recorded sounds are more similar. Knowing this frequency range, we can define the mode number to take into account, an important entry data in terms of computation time. Consequently, auditory experiments were run with low-pass filtered recorded sounds. The bandwidth is equivalent to the frequency range we would describe in our model. The larger the bandwidth is, the most resonance frequencies we should describe, the most resonance mode we should take into account.

4.1 Procedure

Sounds 1, 2 and 3 were tested. For each of them, dissimilarity tests were run between one recorded sound and this same recorded sound low-pass filtered with cut-off frequency set to 14 values between 300 and 8000 Hz. Only the first 500 ms of these sounds were tested since this part is the brightest. The overall loudness of all sounds had been set to approximately the same loudness level, on the order of 70 phons, as judged by the experimenters. The non-filtered sound was presented to itself so as to have a reference of what we could called "very similar". A method of paired comparison was used to evaluate the dissimilarity between the non-filtered sound and the 14 filtered sounds. Pairs were presented in a random order and after each presentation, the listeners were asked to evaluate how similar or dissimilar the signals of the pair were and then to quantify their jugment by locating a cursor on a line display on a screen. The two end points of the line were labelled very similar and very dissimilar and have the assigned values of 0 and 1 respectively. Twenty-one subjects took part in the experiments. The results of each subject were normalized so that the maximum of dissimilarity was equal to one.

4.2 Results

Figure 9 presents the results of the dissimilarities for each of the three sounds tested, as a function of the cut-off frequency. At first glance, the three curves are close. The point at the right bottom of the figure is the "very similar" reference. According to the values of dissimilarities, sounds filtered at 8000 Hz are judged very similar to the non-filtered sounds. In addition, from 8000 to 2000 Hz, the dissimilarities increases progressively. This is due to the presence of audible components in this band. Finally, the three curves present a stabilization of dissimilarities from 2000 to 400 Hz. In fact the three sounds have a frequency component around 300 Hz which is probably intens enough to mask the other components contained between 400 and 1500 Hz for sounds 1 and 2 and between 400 and 900 Hz for sound 3. This masking effect can be seen in figure 10. That shows the specific loudness of sound 2 over the first 100 ms because the most important energy of the sounds is contained in this time interval. In summary, the components up to 8000 Hz in the recorded sound are perceptible so that they should be taken into account for our synthesis. But added components between 300 and 900 Hz would not yield to different sounds in a perceptual point of view.

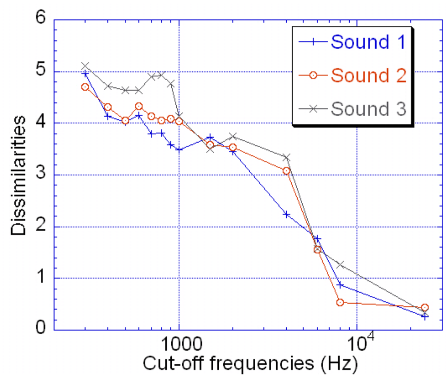

Figure 9: Dissimilarities versus cut-off frequencies

Lastly, it is interesting to correlate these dissimilarities with a physical parameter of these sounds. So that we calculated specific loudness from Bark 4 to 24 in the first 100 ms. Figure 11 presents dissimilarities of sound 2 as

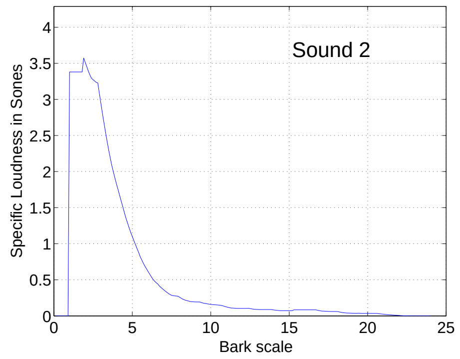

Figure 10: Specific Loudness of sound 2 over the first 100 ms

a function of its specific loudness. We choose this bark interval because this is where the filtering is effective. In this figure 11, the correlation between dissimilarities (perceptive parameter) and total loudness (physical parameters) is established.

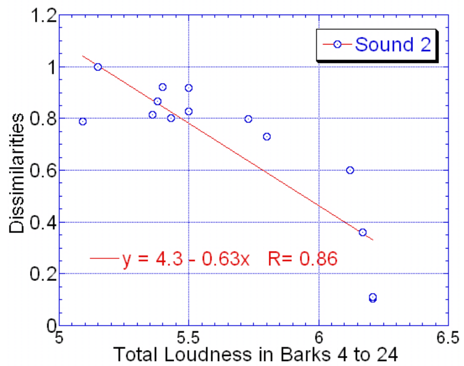

Figure 11: Dissimilarities as a function of total loudness calculated over the first 100 ms, in Barks 4 to 24

5 Conclusion

Firstly, a physical study of sounds was undertaken. We identified precisely the thickness of the plate so that recorded and computed sounds match. We choose a global structural damping by studying the decrease of sound-pressure level over the 3-5 s time interval and in each Bark band. We introduced these new entry data in our numerical model and we studied physical differences of both recorded and computed sounds. The temporal decreases of the amplitude of the two types of sounds seem globally similar. From a spectral point of view, levels of recorded and computed sounds are close below 200Hz. Nevertheless, level differences are notable above 200 Hz and particularly in the damped part of signals. This is due to the fact that real structural damping is not constant, whereas we used a constant damping in our model.

Secondly, a perceptual experiment shown that the frequency band 25-8000 Hz is to be taken into account for our numerical model to synthesize sounds that are perceptually equivalent to real sounds. Furthermore, this last experiment shown that it may exist masking effects in the frequency range contained between 400 and 900 Hz. Consequently, frequency differences between recorded and sythesized sounds are not audible in this interval.
Final auditory experiments would be able to run between recorded and these computed sounds. This psychoacoustic evaluation would permit to know if physical dissimilarities are perceptible or not.
From now on, more complex structures are studied in psychomechanics so as to satisfy the request of industrials. They mainly concern the coupling of plate and cavity excited by an acoustical or a mechanical force; this system being a simplified model of an automobile cell.

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