SUBJECTIVE ASSESSMENTS OF THE ACOUSTIC RADIATION FROM STEEL STRUCTURES: Some effects of a few parametric variations

Pacs Reference : 43.66 Ba or 43.40 Yq

N.HAMZAOUI*, C. SANDIER*, E. PARIZET*, P. WETTA**, C. BESSEYRIAS***
Laboratoire Vibration Acoustique, INSA Lyon bâtiment Antoine de Saint-Exupéry , 25 bis, avenue Jean Capelle Villeurbanne 69100 France
Tel : 04 72 43 80 81
Fax : 04 72 43 87 12
E-mail : nacer@lva.insa-lyon.fr
** IRSID Voie Romaine BP 30320 57283 Maizières-les-Metz Cedex France
Tel : 04 72 43 88 25
Fax : 04 72 43 88 25
E-mail :patrick.wetta@irsid.usinor.com
*** USINOR R&D Sollac Lorraine LEDEPP BP70011 57191 Florange cedex France
Tel : 03 82 51 65 63
E-mail :corinne.besseyria@sollac.usinor.com

ABSTRACT

Nowadays, manufactories have realized the real importance of an approach dealing with the quality of the noise emitted by their device, so as to determine the implications and the effects of what is called the pleasant and unpleasant components. As a matter of fact, even if all the annoying frequency components of a noise cannot be filtered out, more and more interest has been taken in making it as pleasant as possible. The subjective analysis of a few physical parameters concerning the acoustic radiation of simple structures is a first approach of the following general issue : can the sound quality of a structure submitted to any excitation be predicted ?

I/ INTRODUCTION

This study consists in a vibroacoustic analysis coupled to a sound perception analysis which has been based on an estimation of preference between synthesized pairs of sounds, varying several structural parameters at the same time. Sounds were generated by a temporal vibroacoustic calculation of a steel plate in two types of boundary conditions : clamped and simply supported. Both the physical model which has been used and the results of a psycho-acoustic test submitted to about thirty listeners will be presented. C. Marquis-Favre and J. Faure's works are particularly mentioned [3] in this type of approach because they have emphasized the importance of both damping and the thickness of a steel plate (baffled and submitted to a plane wave) on sound perception. They have used an assessment by pairs of the preference and dissimilarity between signals of computed acoustic pressure.
An experimental study has been realized from plane and plate pressed structures, submitted to a real excitation like the one of a lawn mower engine. A quantitative vibroacoustic analysis (vibratory energies and acoustic power measurement, damping……), coupled to a qualitative analysis (binaural measurement and listening through an headphone, paired comparison tests,…..), have allowed to extract correlated parameters with preference scores.
This study can be enlarged to other materials and could allow to identify the parameters having to be defined with accuracy or not concerning an input of a vibroacoustic computation for a better predictive accuracy.
This paper will briefly present the theoretical approach used for thr computation and the results of the subjective analysis. The experimental part enlarged to both pressed plates and the use of

different materials has also been presented in a spirit of being complementary with one another and in a spirit of assessment with regard to the theoretical part, because they have been made in the frame of 2 different contractual contexts.

II/ OBJECTIVE VIBROACOUSTIC CHARACTERIZATIONS

II.1 Vibroacoustic modeling of a plate in the temporal field

The main purpose of this modeling is the constitution of sound synthetic files represented by the radiated acoustic pressure of a vibrating plate with the variation of a few geometric and mechanical parameters.

The mere flexural equation of a plate submitted to an excitation (F(x,y,t)), neglecting the rotational inertia is given by :

$$\rho h\frac{\partial^2 W(x,y,t)}{\partial t^2}+\lambda\frac{\partial W(x,y,t)}{\partial t}+D\Delta^2 W(x,y,t)=F(x,y,t) \qquad (1)$$

with ρ the volume mass, h the thickness of the plate, λ damping, W(x,y,t) the transversal motion and $D=\frac{Eh^3}{12(1-\nu^2)}$ the flexural stiffness where E is Young's modulus and ν is Poisson's coefficient.

$\Delta^2=\frac{\partial^4}{\partial x^4}+2\frac{\partial^4}{\partial x^2\partial y^2}+\frac{\partial^4}{\partial y^4}$ is Laplace's operator.

The vibratory response has been decomposed in the basis of natural modes corresponding to the boundary conditions. A modal differential equation obtained and solved using Newmark's method. Two types of boundary conditions have been taken into account :clamped or simply supported. In the case of the frame, the modal schema has been obtained using Bolotin's edge effect method [4].

Different transfer functions (vibratory answer/ exciter force) could be built using an impulsional excitation. So, vibratory computations could be validated for both types of boundary conditions, comparing them to results found from the reference [5], and to a computation established with finite elements.

The radiated acoustic pressure can be computed in any point M of the environment (with the hypothesis of a light fluid) using Rayleigh's integral in the temporal field and knowing the normal vibratory speed V_n in every M_i point of the plate inserted into a rigid baffle :

$$P(M,t)=\frac{j\rho_0\omega}{2\pi}\sum_{i=1}^{N_e}\frac{1}{r_i}V_n\left(M_i,t-\frac{r_i}{c}\right)\Delta S_i \qquad (2)$$

with ρ_0 the volume mass of the air, c the sound celerity, r_i the distance between point M and the source points of the vibrating plate : M_i. The surface of the plate is discretized in N_e elements of the ΔSi elementary surface.

The validity of this type of computation mostly depends on the fineness of the grid of the vibrating surface which increases the time of computation with the increase of the maximal frequency of analysis. In this study, computation were limited in the frequency range [0-2000Hz].

The acoustic pressure radiated by a vibrating plate submitted to a punctual mechanic excitation depends on several parameters linked to mechanical characteristics (stiffness, mass, damping, Young's module, modal density, volume mass,….), geometrical elements of the structure and the sound environment (fluid, position,…..). Unfortunately, the effect of all the present parameters in the modeling cannot be analyzed, but parameters representing most a structure radiation issue will have to be chosen. The chosen mechanical excitation comes from a temporal recording of a force introduced by a lawn mower engine.

II.2 <u>Experimental determination of physical parameters on both plane plates and embossed structures</u>

An experimental study has been realized from plate and embossed structure pressed structures, submitted to a real excitation like the one of a lawn mower engine.

Structures have been fixed in an aperture (600mm*400mm for plates and 310mm*410mm for embossed structures) of a semi-anechoic chamber, and the dynamic excitation stemmed from a lawn mower engine (located in an adjoining chamber) has been transmitted by a beam stiffly linked to structures.
The vibroacoustic quantitative analysis consists in determining both vibratory energy and the radiated acoustic power associated with each structure in an experimental way : a scanning laser vibrometer (figure ?) has been used for the calculation of vibratory energy, and a robotised intensimetric acoustic probe has been used for the estimation of the acoustic power.
In addition, a technique of experimental modal analysis has been used for plates, so as to evaluate modal frequencies in an accurate way, modal damping coefficients, and modal shapes of the first modes of structures.

Characteristics of both structures and parameters of the measurement are listed in the table below:

	Plates	embossed structures
Materials (Thickness)	Aluminium (1mm)	Aluminium (1mm)
	Composite1 (1.5mm)	Composite1 (1.5mm)
	Composite2 (1mm)	Composite2 (1mm)
	Steel + asphalt (0.7+2.1mm)	Steel 1* (0.7mm)
	Steel (0.7mm)	Steel 2* (0.7mm)
Speed of rotation of the engine	25Hz and 21.5Hz	25Hz

*: These structures are similar.

Vibratory energy has been computed from the v/F (m/s/N) mobility. The table below presents results of plates. Vibratory energy values were computed on the whole frequency range of the spectrum. Damping correspond to the average of damping on the first two modes and the specific mass of the structures has also been determined.

<u>Plates</u>	Vibratory energy (dB ref 5.10^{-8} $(m/s/N)^2$)		η (%)	ρ_S (kg/m^2)	1st modal frequency (Hz)
	21.5Hz	25Hz			
Steel	67.3	63.7	1.12	5.35	37.79
Composite1	55.1	54.8	2.69	4.8	76.22
Aluminium	54.2	52.7	1.47	2.67	48.73
Composite2	45.1	45.8	8.87	8.01	44.2
Steel + asphalt	40	40.4	23.38	8.57	38.12

Whatever the two speeds of rotation are for plates, the same classification for vibratory energy is obtained. The classification of vibratory energy is different from the one of plates for embossed structures. Results of radiated acoustic power for embossed structures are presented in figure 2.

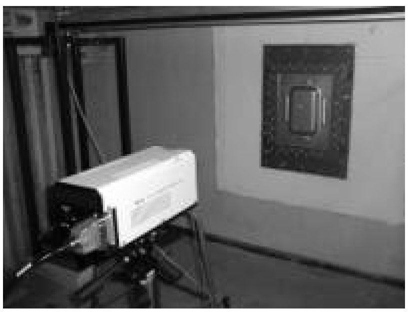

Figure 1 : Scanning laser vibrometer

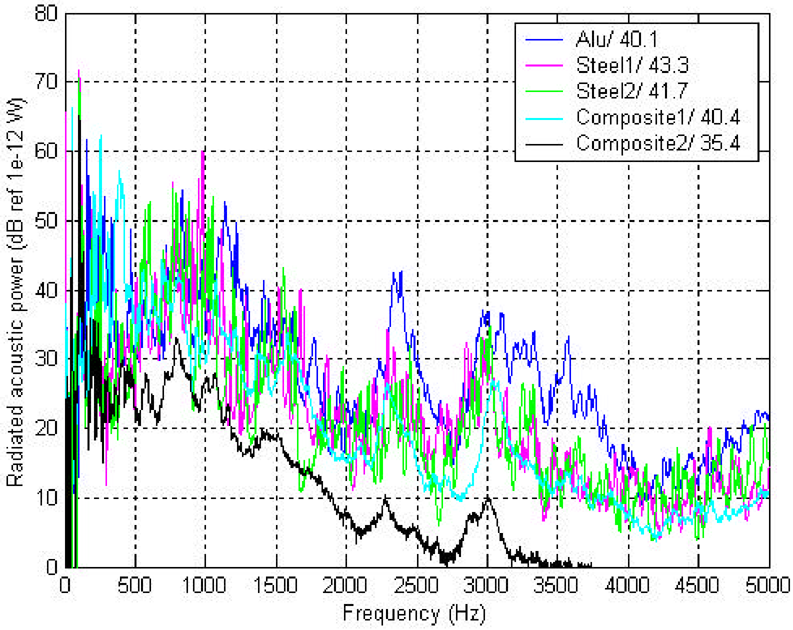

Figure 2 : Radiated acoustic power of embossed structures

III/ SOUND PERCEPTION ANALYSIS

III.1 Subjective assessment from the theoretical model

An analysis of the sound preference is presented in this part. It depends on the parameters linked to mechanical and geometrical characteristics of the vibrating plate. It concerns structural damping, the thickness of the plate and boundary conditions. The most influent parameters on the sound preference are to be extracted, while proceeding to a method of comparison by pairs. A limit of 12 sound samples corresponding to the combination of the following cases has been taken into account so that the length of subjective tests can be reasonable :

Damping	0.1%	1%	
Thickness	1mm	1.5mm	2mm
Boundary conditions	Supported end	Clamped end	

About thirty listeners have participated to the test and the obtained results have allowed to extract that damping seems to be the most determining parameter for preference: the most damped sounds seem to be the most preferred ones. The importance of the thickness is real, with regard to the preference, but boundary conditions appear in second order.

A research of a 2 parameters preference model has allowed to confirm the preponderance of both damping and thickness (figure 3).

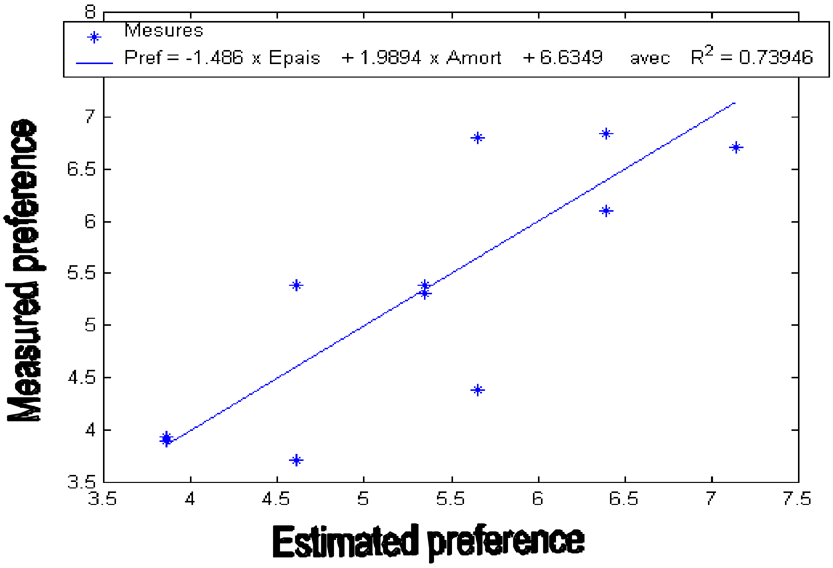

Figure 3 : Model including all the results of the preference

The same analysis, made for both classes of listeners, has allowed to extract the same type of results with a better coefficient of correlation.

The method of the plans of experience has also been applied in parallel so as to quantitatively and qualitatively draw the effect of parameters on the preference. The different methods of analysis of the results (graphic analysis and variance) have allowed to draw the most influent factors from the preference. Actually, damping remains the preponderant factor having a direct influence being very important. Then, thickness and interactions between thickness and boundary conditions appear on the one hand and thickness between damping and boundary conditions appear on the other hand. It can thus be seen how the modal density, depending on both the thickness and the type of boundary conditions can have an influence on the preference.

III.2 Subjective analysis from the experimental part

Sounds radiated by each structure have been recorded with a binaural artificial head. They have then been analyzed through the MTS Sound Quality software so as to calculate various physical and psycho-acoustical parameters. In addition, preference and similarity listening tests have been conducted.
The classification of preference for plates is (from highest to lowest merit score) : steel+asphalt, Composite2, Steel, Composite1 and Aluminium. This classification is a little different for embossed structures : Composite2, Aluminium, Composite1, Steel1and Steel2.
Metrics associated to loudness (Zwicker's Loudness, dB(A)) presented the best correlation with preference scores (R^2=0.96) and with the first principal component computed from the similarity results (R^2=0.9).
Analysis have not allowed to establish any correlation between subjective results and vibratory energy. Furthermore, no definitive conclusions can be observed on correlation with the radiated acoustic power, because too important variations of excitation forces appear for these measurements.
The first principal component was also correlated to the specific mass of plates (R^2=0.8). Moreover, the damping coefficient was well correlated to the preference (figure?), preferred sounds corresponding to most damped materials.

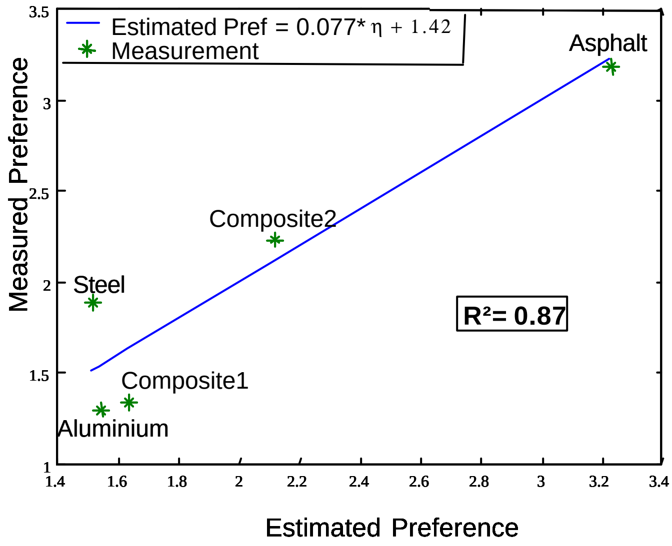

Figure 4 : Correlation between damping and preference for plates

IV/ CONCLUSIONS

Both the importance of structural damping and plate thickness on the subjective assessment of the acoustic preference of a steel plate, and the type of boundary conditions (leaned or built in) could be drawn from these two theoretical and experimental analysis made in an independent way. They only had an influence in the second order on the two previous parameters. The experimental part obviously shows that preferred sounds correspond to most damped materials while using different materials.

A research of a perceptive space allowed to define the surface mass as a preponderant dimension using dissimilarity results for the experimental part. Vibratory energy does not seem to be an influencing parameter in the subjective assessment, meanwhile the sound level (dBA or loudness) is correlated to both preference and dissimilarity.

Such results would finally allow to identify the parameters having to be accurately defined or not with the input of vibroacoustic computations for a better accuracy in predictive computations.

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