Audio Engineering Society

Convention Paper

Presented at the 127th Convention
2009 October 9–12 New York, NY, USA

Target modes in moving assemblies of a pleated loudspeaker

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ABSTRACT

In this work we present the process followed for the adjustment of a numerical model in finite elements of the mechanical behavior of a pleated loudspeaker, based on the AMT technology (Air Motion Transformer). In this type of transducers, the diaphragm is formed by longitudinal folds. In the internal face of each one of these folds is printed a conductive ribbon. We have obtained firstly the participation factors and the generalized mass from the results of a natural vibration modal analysis. Next, an analysis is realized taking into account the loss factors of the materials, followed by a forced vibration modal analysis. Finally, a method is described for the characterization of the materials (Young Modulus and Loss Factor), by using modal analysis techniques.

1. INTRODUCTION

As it is known, the response of a transducer is a function of the interaction of electrical, mechanical and acoustical parameters. In particular, a mechanical FEM model for moving assemblies in pleated loudspeakers has been performed. In this work we present the process followed for the adjustment of this numerical model. The paper deals how find the important modes in the moving assembly of this type of transducers. Results of this work are applicable to loudspeakers based on this technology.

As it is well known, those modes, which have high generalized mass, are modes that are highly excited and have relevant importance in the system response.
The standard modal analysis gives a huge number of modes. This is very unpleasant for the staff engineers. Due to the pleated moving assembly specificity, it is very convenient to know which modes determine the mechanical response which is the cause of the high quality of sound reproduction in these transducers.

The work is divided in two basic items:
First, due to the importance of the information of entry in the simulations, there is described, likewise, the process followed for the characterization of the materials (Young Modulus and Loss Factor), by using modal analysis techniques. The available data for some materials, such as fabric or polymer, are spread and sometimes are erroneous. For this purpose, we have performed an experimental setup and method which is a modification and an improvement of the standardized method proposed by the American Society of Testing and Materials (ASTM).
Experimental results will be presented on the paper. Several tests have been performed with all kind of materials, such as paper, polyimide, and others. These results are in good agreement with relevant manufacturers' information.
Secondly, the paper presents results made by the FEM technique. The presented models are based on the Ansys Code. This software provides elements which are suitable for laminated materials as the used in real pleated moving assemblies (copper and foil). Taking the problem step by step we have approached the topic at first stage, which is only mechanical, removing the final acoustic part. Software has quadrilateral elements which are appropriate for the pleated moving assembly. Due to circumstance of the specific periodic geometry of the diaphragm, we have found sizes of the quadrilaterals elements which give high spatial resolution and, obviously, uniformity along and across the complete pleated moving assembly.
The type of elements used is "shell", with six degrees of freedom for each node, (three displacements 3D and three drafts 3D), since the thickness is much smaller that the other two dimensions. The size of the elements has been chosen according to the criterion normally used in FEM, reaching about 80 000 nodes each of the contemplated geometries.
In order to guarantee accurate results, the size of elements was progressively reduced until convergent eigenvalues were obtained.
From this basic numerical model we have studied the influence of small variations of the geometric parameters on the mechanical response, which causes the final acoustic response.
First of all, we have obtained the participation factors and generalized mass from results of a natural vibration modal analysis.
Next, in order to come closer to the real world, a dynamic analysis is realized taking into account the loss factors of the materials.
Damping has been introduced with a constant loss factor for each material, independently of the frequency. Finally, a forced vibration modal analysis is performed, trying to simulate the real behaviour of this type of transducers.

2. CONCEPTS

Target modes are those mode shapes that are determined to be dynamically important using some definition [1]. This concept has been widely used in various fields as the aeronautics, space engineering and in seismic engineering as well. In these fields it is very common to have modal analysis of structures with many eigenvectors and it is necessary to select only the most important modes. These modes are the target modes.

The main tool to find important and target modes is the value of the generalized mass. The definition of this concept can be found in books as [2] and papers as the ones in [1], [3] and [4]. This concept, and the participation factor, has been widely used in the field of seismic engineering. This is, because, as in space engineering, the target modes must be enhanced respect to the many found in modal analysis of large bodies.

Other aspects of interest of the use of the generalized mass are: a) to divide large model systems in subsystems [1], and b) separation of modes which have a certain overlap in spectrum [3].

If we have a system defined by:

$$M\ddot{x} + Kx = F \qquad (1)$$

Being: M: the mass matrix, K: the stiffness matrix, F: the forcing function, and being x and $\ddot{x}$ the displacement and acceleration vector respectively.

The solution of the system (1) is found in terms of eigenvalues and eigenvectors. Being, as was said before, the eigenvectors the vibration modeshapes. Let Φ be the eigenvector matrix.

The system's generalized mass matrix $\hat{m}$ is given by:

$$\hat{m} = \Phi^T M \Phi \qquad (2)$$

Generally, those modes which have high generalized mass are modes that are highly excited and they have relevant importance in the system response.

The generalized mass concept is very important in dynamics because it can be associated to the coherence concept in acoustics and vibrations. Notice that the generalized mass will be maximal when the mass matrix will "fit" with the mode shape, see equation (2). Notice in this equation that the mode shape is represented by Φ and by Φ^T as well. Observe that in dynamics, we may have a motion of a certain point of a system in a specific direction because the action of inertia force(s) acting in other part(s) of the body and not necessarily in the same direction. The coherence is high if the motion of each part of a system is due to inertia force acting on the deformed point and applied in the deformation direction and sense. Simply speaking we can say that the coherence is high when the responsible of the deformation of a system part is the inertia force of each own deformed part. This is the cause of the high coherence of a pleated tweeter. In these transducers the acting forces deform the pleated moving assembly at the same points of deformation and at the direction of the needed mode-shape. Thus in these transducers the generalized mass is high for many modes. Simply speaking, notice that a high generalized mass for a specific mode which has a modeshape or eigenvector Φ_n, must have the amplitudes of all nodes as high as possible in order to increase the products of equation (2). For example, in the main mode of a speaker all node masses of the moving assembly move with the same amplitude which equals 1. In this case, for this specific mode, the generalized mass, equals the moving assembly mass. This is the cause of the high excitation of the main mode in a speaker. This assert seems obvious, but as we will see, there are other interesting aspects of this topic.

The participation factor is another important parameter that has full application in the electroacoustic transducers field. In loudspeakers it is very important to target the modes which excite masses only in the excitation direction, and the participation factor takes it into account. Translational components of the participation factors are defined as:

$$r_{iv} = \frac{\sum_{j=1}^{N} \Phi_{ij} M_{ij}}{\{\Phi\}_v (M) \{\Phi\}_v^T} \qquad (3)$$

Where:

Symbols are the previously defined and the rest are as follows:

i: is the component identification (the degree of freedom) (1 to 6)

j: is the node identification (1 to N nodes the model has)

v: is the mode identification (1 to V modes). V is the total number of modes found by the model extractor

N: is the total number of modes.

Thus, the participation factor r_{iv} for the mode v at the i direction equals the summation of all displacements j of the eigenvector Φ, times the associated masses divided (normalized) by the generalized mass of this specific mode. Observe that the participation factor is an interesting concept because it takes into account the global mass associated to one specific direction and the coherence of this mass in motion due to the phase or the motion sense of all masses. Notice that at high frequencies (where the second target mode is activated) modes are localized, which implies loss of generalized mass. Thus, as the participation factor is normalized to the generalized mass, any localized mode will have a loss of the participation factor as well.

A mode will be target if the generalized mass is high, and it is convenient that the axial participation factor must is also high. With this concept in mind it is easy to find which modes are target when design a moving assembly.

3. PLEATED LOUDSPEAKER

Air Motion Transformer (AMT) is a loudspeaker mechanism, or audio transducer, invented by Dr. Oskar Heil. It operates on a different principle than both electrodynamic and electrostatic speaker drivers. The AMT moves air in an augmented, semi-perpendicular motion using a folded sheet structured around a series of struts positioned in a high intensity magnetic field. The diaphragm pushes back and forward from itself in a similar physical motion pattern to what is observed when an accordion is squeezed in and out to pump air though the reed chambers, albeit over an exceedingly smaller motion range.
A phenomenological description of such transducers can be found in [5].

The experimental basis we have taken as reference in our study is that the speaker TPL-150,[6] , in figure 1, produced by Acústica Beyma.

Figure 1 TPL-150 loudspeaker

It should be noted that the dimensions of the models described and implemented in the next section are similar but not identical to those of the speaker subjects.

The typical frequency response of a loudspeaker of this type, measured in free field, is shown in figure 2. As it can be seen, its passband begins slightly above 1 kHz. From the principle of operation of such transducers, it is expected small changes in the geometry of the loudspeaker have some influence on the response curve.

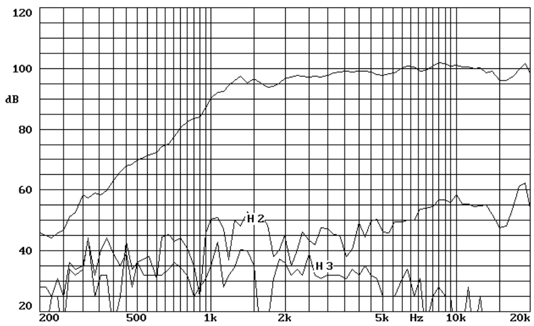


Figure 2 Frequency response and THD of TPL-150

To characterize this type of speakers, there has been undertaken near field measurements (figure 3):

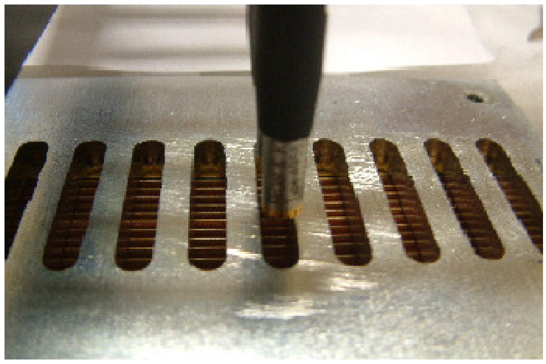
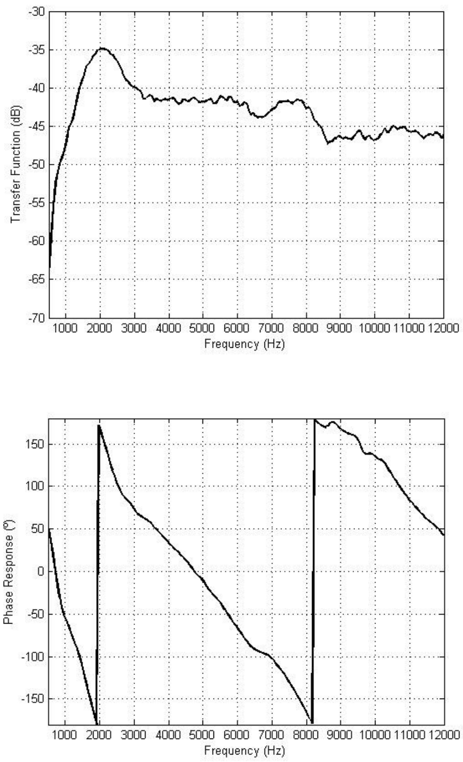


Figure 3 Near field measurements

In figures 4a and 4b, module and phase of the transfer function when using a random noise test signal are shown:



Figures 4a and 4b a) Module. b) Phase of the transfer function (Near Field)

Measures have also been conducted in near-field at different positions of the grid (figure 5) that allow us to compare the radiation from the loudspeaker by location. Using the notation *ij* to denote rows and columns:

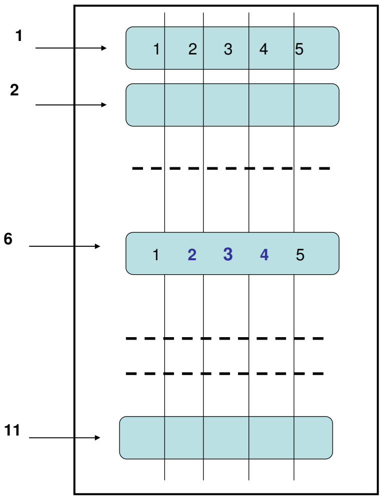


Figure 5

In figure 6a it is compared the response corresponding to point 6 - 3 with 6 - 2 and in 6b the 6 - 3 with 1 – 2.

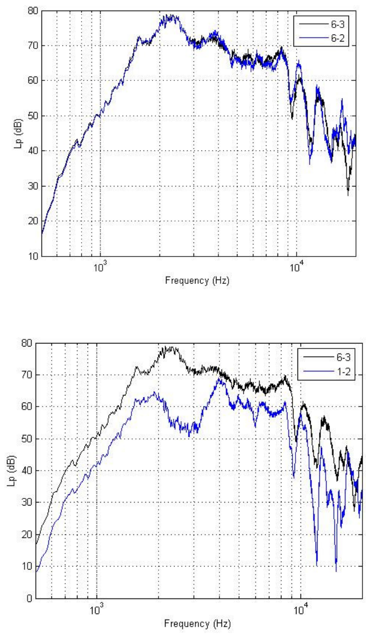


Figure 6a and 6b Sound pressure level comparison between a) 6-3 and 6-2. b) 6-3 and 1-2

4. NUMERICAL MODEL

The finite element model has been implemented with the computer program *Ansys*. The type of elements employed is shell, with six degrees of freedom per node (three 3D displacements and three 3D gyres), being that the thickness is much smaller than the other two dimensions. The number of elements of the implemented models has been increased to check that the modal basis does not change significantly. It has been applied the approach that there are two elements in cross direction to the loudspeaker plane in each copper band, having each element two dimensions similar length, resulting about 80.000 nodes and elements in each one of the models implemented. In this way, the wave length for the maximum analysis frequency is much higher than ten times the elements dimensions, that is why the length of the elements employed is suitable in accordance with the Finite Element Models criterion. Although it has been carried out an analysis in the 500-1200 Hz range, only the results corresponding to the 500-2000 Hz range are shown.

Some details of the geometry under consideration is shown in figures 7a and 7b (in the latter one, the two different material layers can be appreciated):

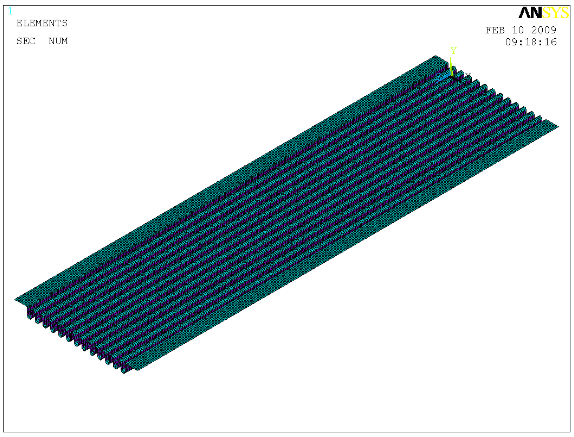


Figure 7a Complete model front side view.

We will refer to the models implemented as A, B, B2,E and F. The dimensions and parameters of each one of the models are shown in figure 8 and table 1. The thickness of the base material and the Cu bands is 17 microns.

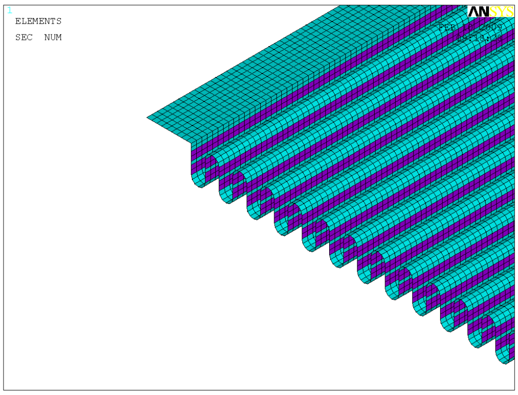


Figure 7b Model detailed front side view

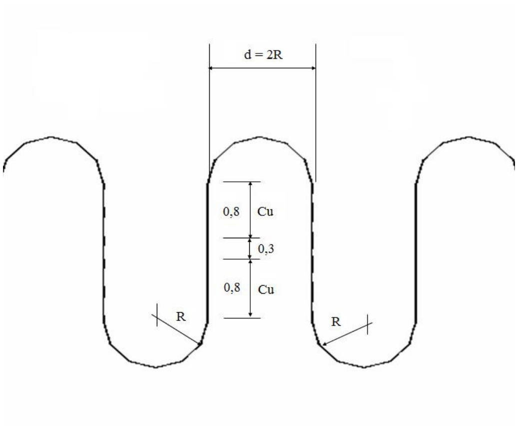


Figure 8 Model geometry detail

Loudspeaker	Length (mm)	R (mm)
A	140	0.65
B	145	0.65
B2	160	0.65
E	140	0.78
F	140	0.52

Table 1 Models dimensions

4.1. Materials characterization

It is not necessary to highlight the importance of the input data in simulations. In our case, besides geometric data, mechanical features must be considered, specifically, Young modulus and loss factor. Metallic materials such as *Cu* offer different reliable data and method for the characterization in references. However, there are no methods within reach of non-specialized material characterization laboratories for the other type of materials employed. This fact has led to devise a measurement method (**patent pending)**, based on a variant of modal analysis, as it is seen in figures 9a and 9b.

The material to be characterized (1), previously cut into sheets such as those shown in figure 9b, is fixed at the top. On the bottom is fixed to a support of mass m, which is supportive to a coil that is placed within the magnetic field. When the coil is excited with a random noise, it produces a force which causes compression and depletion of the foil in the longitudinal sense that will be captured by a sensor located in the support. In our case, we have used two accelerometers, one in each corner of the support so that it would be possible to detect transversal vibrations in the case the system is not completely balanced. If we call m the mass of the stand, including the accelerometers, the system can be modeled as a first approximation as a system of single degree of freedom and it can be shown that, if it is verified that the support mass is much greater than the mass of the material sheet that is intended to characterize, the frequency corresponding to the first mode is given by the equation:

$$f_0 = \frac{1}{2\pi}\sqrt{\frac{E*A}{L*m}} \qquad (4)$$

Then:

$$E = \frac{4\pi f_0^2 Lm}{A} \qquad (5)$$

where E is the Young Modulus, L is the length and A is the transversal area of the sample under test.

In our case, the dimensions of the layers have been 15x3 cm, the weight of the support including accelerometers was around 320 g. There have been characterized two different materials: Polyetheryimide of 100 and 250 microns thickness, respectively, and 130 micron thickness Polyester. The results match those supplied by the manufacturers.

The great advantage of this method, besides its simplicity, is that the transducer used to activate the movement does not involve a contact (non-contact shaker), with all the advantages that it entails.

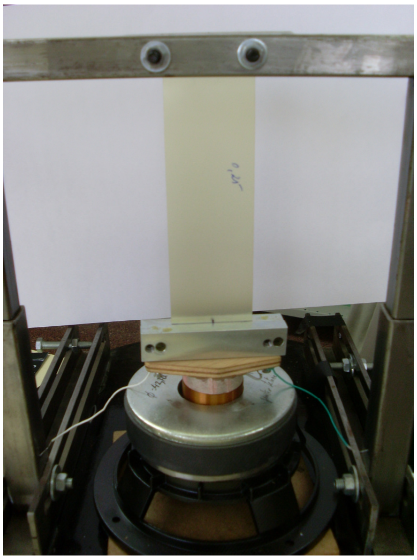


Figure 9a Experimental Setup

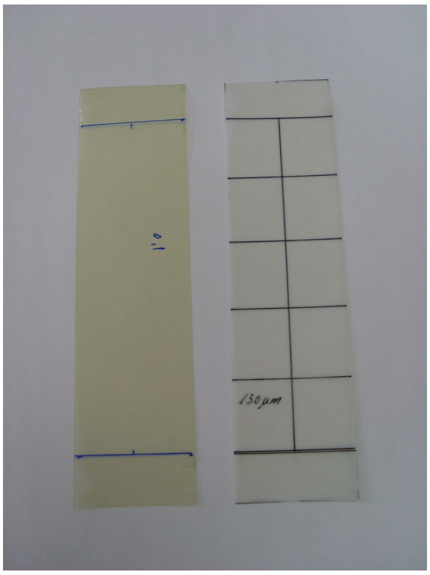


Figure 9b Samples of materials under test

4.2. Results

Ansys code allows determining Mass Matrices with two options:

a) Consistent formulation method (CMM Method), which is element dependent.

b) Lumped Mass approximation (LMM).

First of them is default formulation and it is recommended for most applications. However, for some problems involving "skinny" structures such as slender beams or very thin shells, the lumped mass approximation often yields better results. Also, the lumped mass approximation can result in a shorter run time and lower memory requirements. As we will prove next, the results obtained with both methods in the problem we are dealing with bring about similar conclusions.

It has been justified previously the importance of the determination of vibration modes and their participation factors, since they are determinant to explain the loudspeaker acoustical behavior. Somehow, they quantify the easiness or predisposition of modes to vibrate in a particular way.

A first step to face the problem is to determine these participation factors for a particular case: when the loudspeaker is excited with the inertia forces correspondent to a constant drag acceleration in all the loudspeaker points in a ***t*** instant (same complex drag acceleration vector for each one of the points in a harmonic analysis), as it is shown in the next figure:

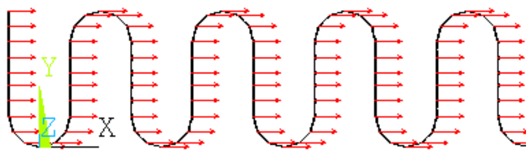


Figure 10 Drag acceleration.

In this case, since the whole loudspeaker with its inertia forces corresponding to drag acceleration constitutes a symmetrical structure respect to the middle plane parallel to YZ subjected to an anti-symmetric load system, participation factors will be then null for all the symmetric modes.

From these participation factors, moving mass distribution is obtained, as it is shown in figures 11a (CMM) and 11b (LMM). It can be seen the similarity between the results obtained from each method:

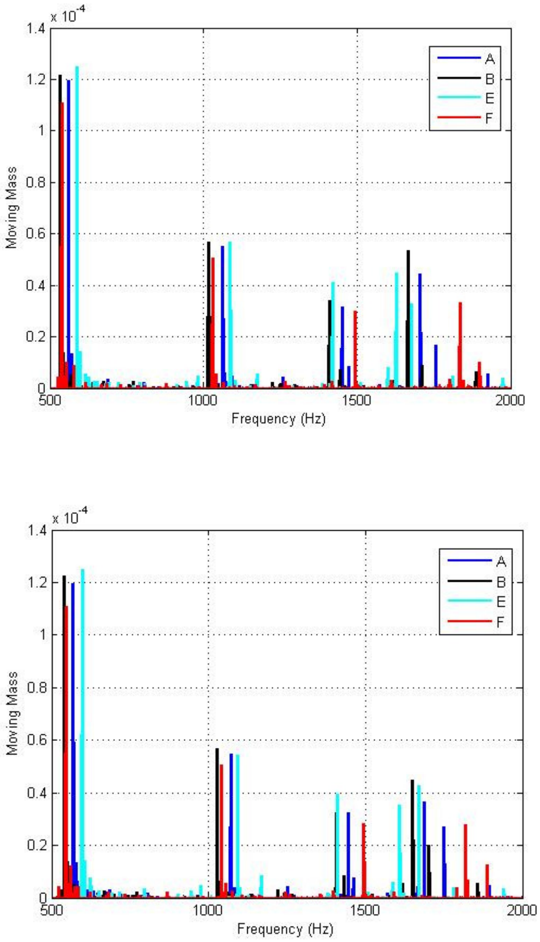


Figure 11 Moving Mass obtained employing
a) CMM method. b) LMM method

As a second step and aiming to reproduce the real operation of the loudspeaker, supposing that the magnetic field is uniform and that the mechanical force direction is not altered significantly by nonlinear effects, the loudspeaker will be excited with a force distribution as it is shown in the next figure (uniform force module):

In order to produce a forced excitation to simulate the effect of the current sense, there have been introduced forces in the nodes corresponding to those elements where the copper bands are with the same sense that a given fold and opposite in the next fold.

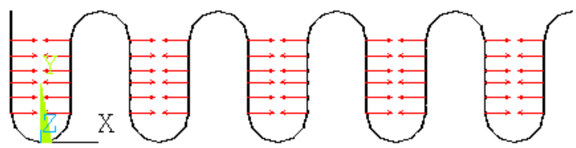


Figure 12 Forced analysis force distribution

Since the force analysis corresponds to a structure symmetrical respect to the middle plane parallel to YZ subjected to a symmetric load system, the modes excited will be then symmetrical, but participation factors in a modal analysis when the structure is symmetrical are null. Then, to use them comparatively, the whole loudspeaker model should not be used, but one corresponding to half loudspeaker (actually, the simulations can be carried out with only a fourth of the model because the middle plane parallel to XZ is a second plane of symmetry).

It has been introduced the damping with a constant loss factor for each one of the materials, regardless of frequency, according to the damping matrix expression [C]:

$$\sum_{j=1}^{2} \frac{\eta_j}{\omega}[K_j] \qquad (6)$$

where:

j is the material identifier (j=1 or j=2)

ηj is the j material loss factor

[Kj] is the stiffness Matrix corresponding to the j material

ω is the circular frequency (rad/s).

In the figure 13a and 13b, the moving mass obtained from the methods mentioned above is shown. As before, the results are similar.

In figures 14 and 15 the dominant mode (because of its high moving mass) of loudspeakers A and E is shown.

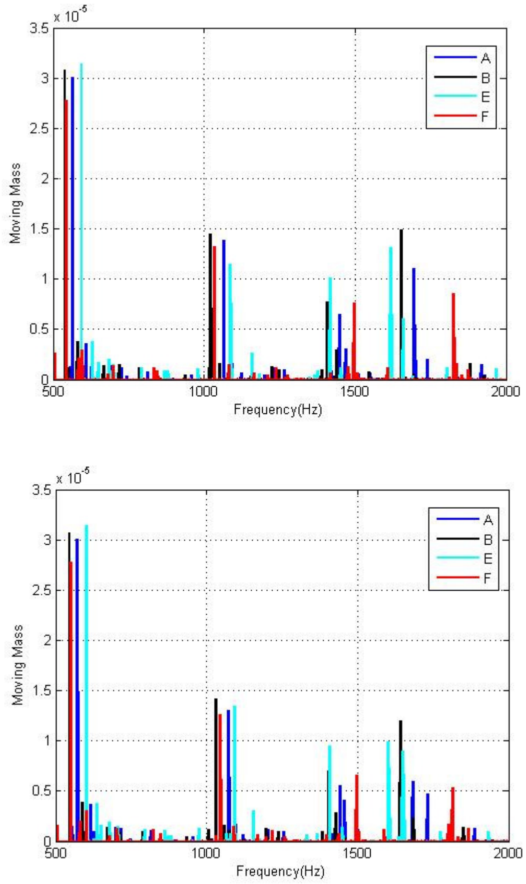


Figure 13 Moving Mass obtained from a Harmonic Analysis employing a) CMM method. b) LMM method.

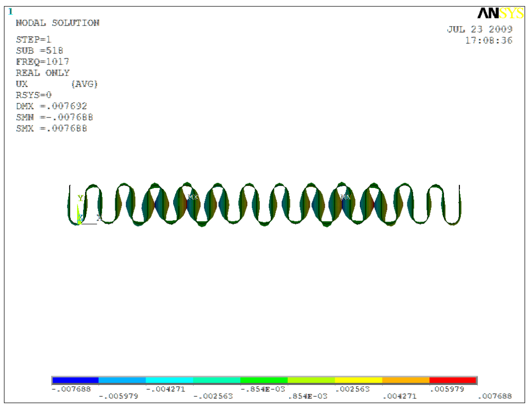


Figure 14a Dominant mode in loudspeaker A (cross view).

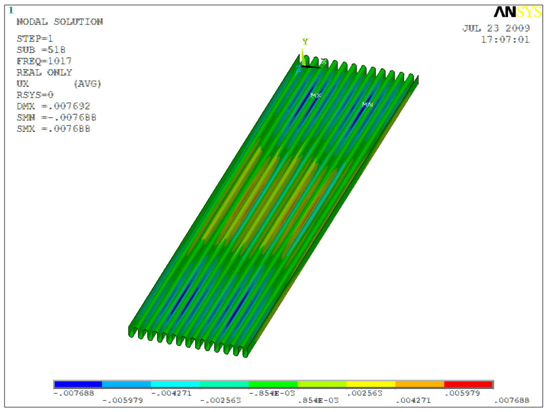


Figure 14b Dominant mode in loudspeaker A (front side view)

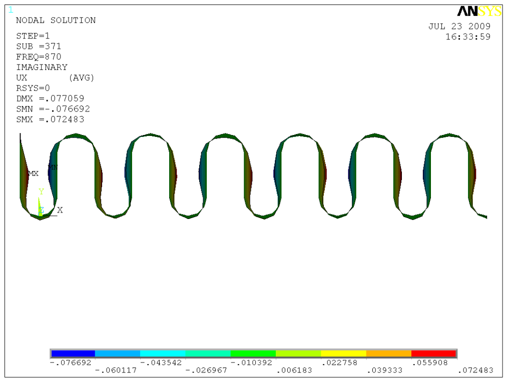


Figure 15a Dominant mode in loudspeaker E (cross view).

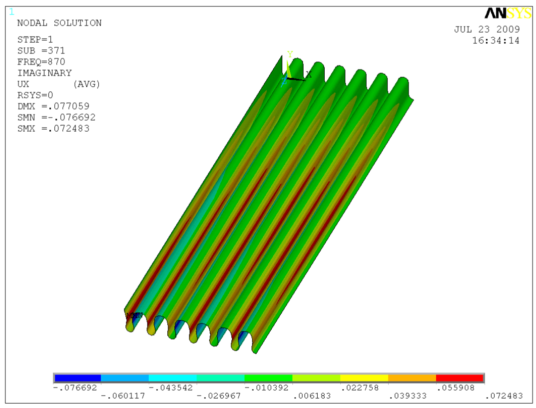


Figure 15 b Dominant mode in loudspeaker E(front side view)

Finally, it is observed that the kinetic energy of the structure obtained in the forced analysis reaches a resonance frequency very close to a eigenfrequency of a significant participation factor not damped vibration mode. This vibration mode is still the best according to the mechanical forces caused by the loudspeaker operation.
In figure 16, normalized kinetic energy associated to the x direction velocity is shown for each frequency, according to equation:

$$E_{CX} = \frac{1}{2}\int v_x^2 dm \cong \frac{1}{2}\sum_{i=1}^{N} v_{xi}^2 m_i = \\ = \frac{1}{2}\sum_{i=1}^{N} \omega^2 U_{xi}^2 m_i \qquad (6)$$

where:
N is the number of nodes, Vxi is the amplitude of velocity (x direction), m_i the mass of node *i*, ω =2*pi*f and Uxi es amplitude of displacement (x direction) for frequency considered.

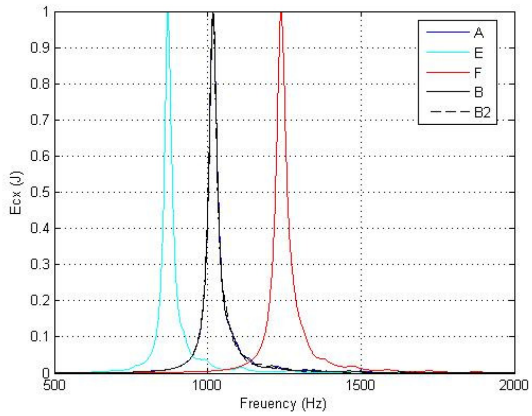


Figure 16

5. CONCLUSIONS

It has been implemented a FEM model that can describe the mechanical behaviour of moving assemblies in pleated loudspeakers and allows to analyze the influence of small modifications in the geometry on the modal basis and the moving mass.
The next step is to incorporate the acoustical component in the problem. It can be done in different ways. At present we are developing a Finite Differences Time Domain (FDTD) model that works from data supplied by the mechanical model.

6. ACKNOWLEDGEMENTS

The authors would like to thank J. Carbajo San Martín for her essential help in data processing.

7. REFERENCES

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