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STRUCTURAL DYNAMIC ANALYSIS OF A BTE HEARING AID CASE

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Abstract: *The purpose of this paper is to present the structural dynamic analysis of a hearing aid case. The object under study is a behind-the-ear (BTE) hearing aid, whose case is made of ABS plastic material. It is initially performed a material characterization of the case, through experimental tests. In the sequence, an experimental modal analysis, with the aid of a laser vibrometer, is performed. The obtained material characteristics are fed to a Finite Element Method (FEM) commercial software so that a numerical modal analysis is also obtained. The numerical resulting model is updated in order to get a better match between experimental and numerical results. Finally, Frequency Response Functions (FRF´s), both experimental and numerical, are compared to confirm the close dynamic performances of the models.*

Key words: hearing aid case, dynamic analysis, structural vibrations

1. INTRODUCTION

A BTE hearing aid device is composed of the following basic components: microphone, loudspeaker, amplifier, case and battery (see Fig. (1)). It is connected to the ear mold by a tube. The mold is placed at the entrance of the ear canal and usually has a hole, to avoid the occlusion effect. As the microphone and the loudspeaker are close together, feedback commonly occurs, both externally and internally (DILLON, 2001). The external feedback happens when there is a sound leakage through the mold hole that reaches the microphone. The internal feedback is caused by structural vibrations produced by the loudspeaker. The loudspeaker mechanical suspension is not perfect, so that its vibrations reach the case, thus producing internal radiated sound.

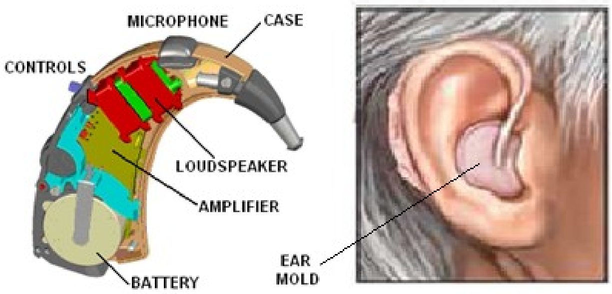

Figure 1. Behind-the-ear (BTE) hearing aid.

The ultimate objective of this study is to analyze the internal feedback phenomenon, although it will not be achieved in this paper. The way to perform such study begins with the construction of a good numerical model of the case, making it possible to simulate its dynamical behavior with good accuracy, avoiding the construction of many different physical prototypes. So, it will be here presented the development of such a numerical model.

2. MATERIAL PROPERTIES DETERMINATION AND VALIDATION

The initial process was the determination of the properties of the ABS plastic material. Samples of the case were provided by Amplivox, a brazilian company. Density and elastic properties of the material were then determined. The measuring procedures are here presented and the results are shown and discussed below.

The pycnometry procedure was used to measure the density (ρ), alcohol being the used fluid.

The values of the Young´s modulus (E) and the loss factor (η) were determined according to the standard *ASTM E0756-05*. This standard specifies the exact shape the sample must present, which was not really achieved, since only a few cases of the hearing aid were obtained for the tests. Rectangular plane samples, as big as possible, were cut from the cases. These stand alone beams were rigidly clamped at one end, leaving the other end free. The clamped-free thin beams were then hit by an impact hammer at mid-length, and their responses were measured by a laser vibrometer near the free end. The measuring setup is shown in Fig. (2).

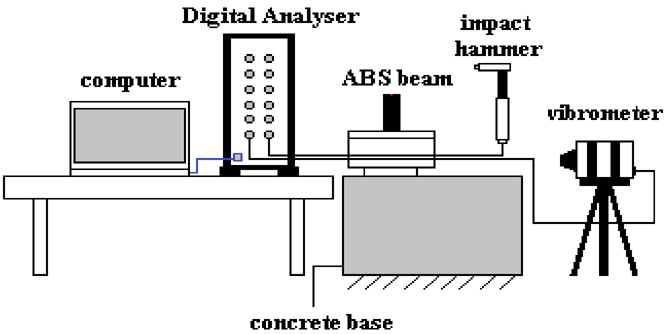

Figure 2. Dynamic properties measuring setup.

According to the ASTM standard, the sample under test should have a shape like the one shown in Fig. (3), with length L and thickness H. Due to the original small thickness, it was not possible to apply to the sample a cross section reduction, so that the used samples had constant thicknesses. The effective length, then, is measured from the final clamping point to the end of the beam. The width of the beam is not important, since both mass and stiffness of the beam are equally proportional to this measure, so that its effect is cancelled out.

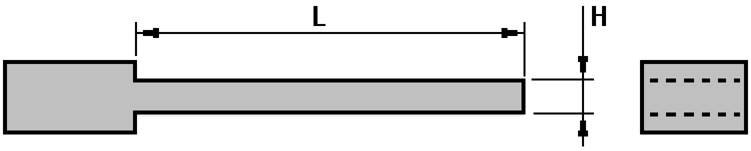

Figure 3. Standard shape and dimensions of the sample beam.

In this application, the signals coming from the hammer and from the laser vibrometer were digitally processed, producing a Frequency Response Function (FRF) in terms of receptance (displacement/force). This is a complex function and must be plotted in absolute values, to reveal many peaks, corresponding to the various natural frequencies (f_n) of the beam. Then, the Young´s modulus (E) of the material may be estimated, for each peak, by the equation:

$$E = \frac{12\ \rho\ L^4\ f_n^2}{H^2\ C_n^2} \quad , \tag{1}$$

where C_n is a particular constant for each n-th natural frequency. For instance: $C_1 = 0.55959$, $C_2 = 3.5069$, $C_3 = 9.8194$, etc.

The loss factor is obtained, equally, for each peak, by the use of the half power bandwidth method. A horizontal line, 3 dB below the peak value (half power line), is traced. This horizontal line intersects the absolute mobility in two points near the peak, one at a frequency below f_n and another above it. The frequency difference between this two points defines the half power bandwidth Δf_n. So, the loss factor is then easily obtained from:

$$\eta = \frac{\Delta f_n}{f_n} \quad . \tag{2}$$

A numerical model of the tested beam was created, using ANSYS software. It was used the SOLID45 element type, which presents 8 nodes and three displacements per node. The obtained mesh is shown in Fig. 4. It was applied a harmonic point force at mid-length and it was registered the displacement near the free end, reproducing the experimental procedure. The locations of the force and of the measuring point are also shown in Fig. 4. The material properties had to be slightly adjusted, to obtain a better match between experimental and numerical results.

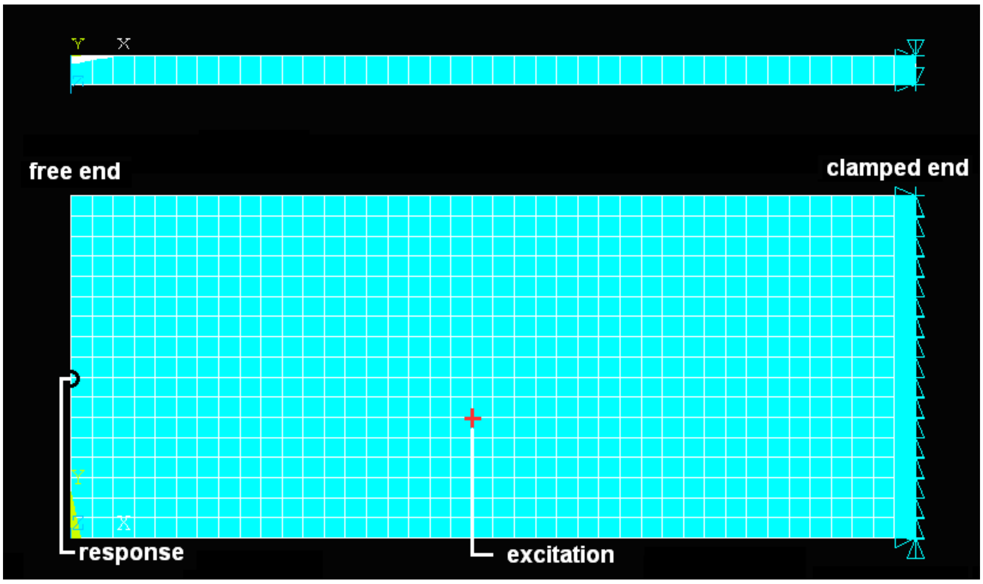

Figure 4. FEM mesh of the sample beam.

3. EXPERIMENTAL AND NUMERICAL MODAL ANALYSES

In order to confirm either the quality of the numerical model, both experimental and numerical modal analyses were performed. More than checking natural frequencies and loss factors, this procedure allows a comparison of modal shapes.

The experimental modal analysis determines the following parameters for each mode: natural frequency, loss factor and shape of vibration (JORDAN, 2002). The number of modes depends on the frequency range that is being analyzed. In the frequency domain, which is the case, a set of FRF´s is necessary to obtain the parameters identification. So, in both sides of the case, right and left, 44 points were marked, as can be seen in Fig. (5). Points were located on the almost flat region of the two parts, to easy the laser measurements. Then, for each part, 44 FRF´s were obtained and analyzed.

Figure 5. Measurement points on both parts.

Each part of the case was glued to a bolt on the top of the shaker table, as shown in Fig. (6). An accelerometer was attached to this table, and the laser vibrometer was used to measure the velocities on various points of the half-case. As a consequence, the resulting FRF´s present ratios of the velocities in the points by the acceleration at shaker table. Signals were processed by PULSE Labshop analyzer. Figure (6) also shows the measuring setup.

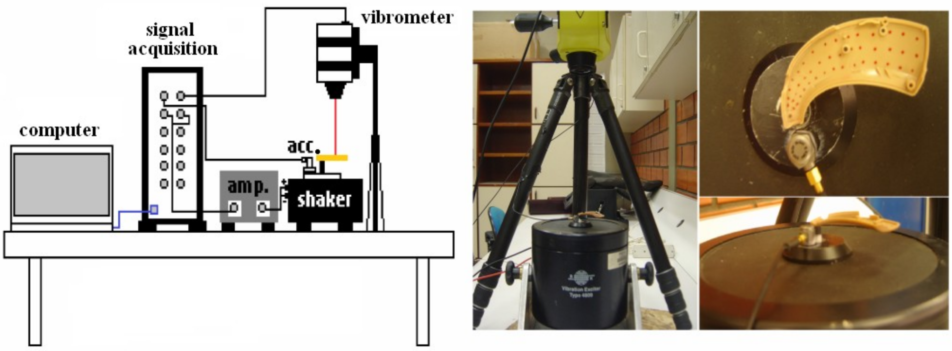

Figure 6. Experimental setup and details of FRF´s measurement.

The resulting FRF´s were imported by Test.Lab software and, by use of the Modal Analysis module, the modal parameters were determined. This experimental modal analysis was then applied to the whole case, after bolting both parts together. Figure (7) shows details of this new measurement.

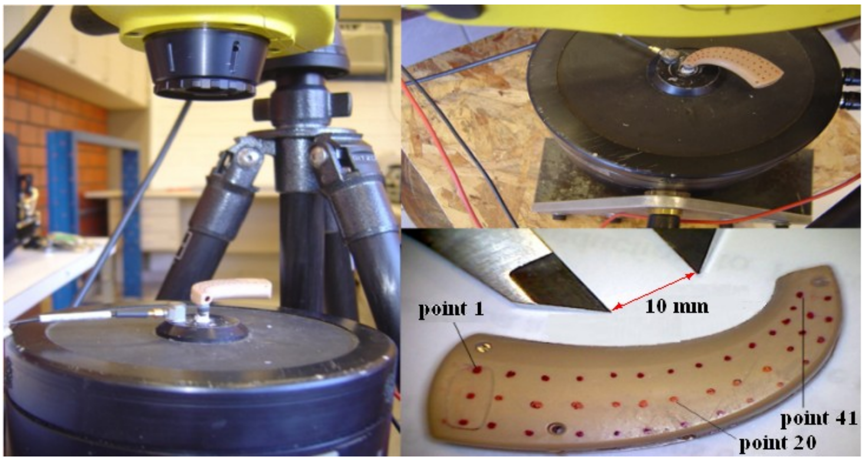

Figure 7. Details of whole case FRF´s measurement.

Numerical validation was again conducted with the help of ANSYS software, using the SOLID92 element type, which presents 10 nodes and 3 displacements per node. Unitary displacements were imposed at the same location the case was experimentally excited and responses were obtained at some points of the same case. Resulting FRF´s, in this case, represent the ratios of two displacements. The mesh used in this analysis is shown in Fig. (8).

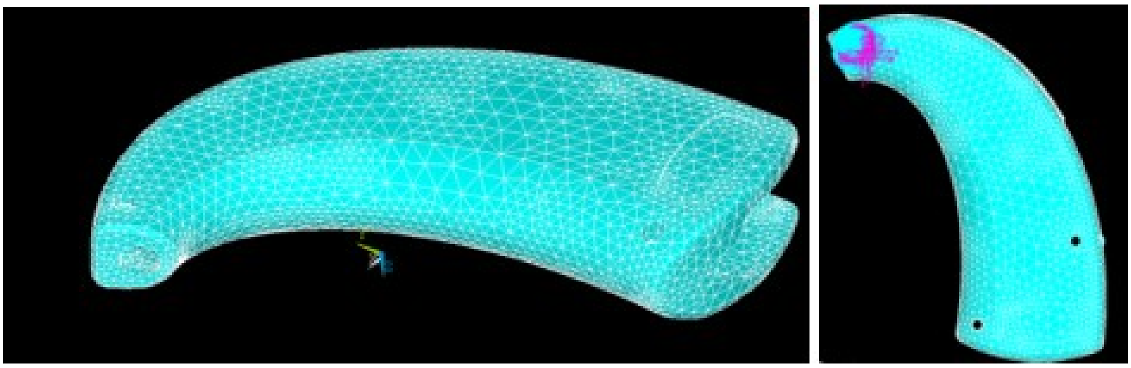

Figure 8. FEM whole case mesh.

4. RESULTS AND DISCUSSIONS

The density of the material was measured with five samples, and the obtained average value is1190 kg/m^3, which is in accordance with the type of material of the case.

Values of the Young´s modulus (E) varied between 0.92 to 1.58 GPa, considering the various samples and different modes of vibration. Calculated loss factor values have fallen in the 0.007 to 0.01 range. Thickness non-uniformity is probably the main source of error, since the samples were cut from the prototypes. The obtained experimental mobilities can were integrated, by post-processing, to furnish receptances (displacement/force).

The clamped-free simple beam was numerically simulated by FEM, taking into account the properties: E = 1,46 GPa, ρ = 1190 kg/m^3, ν = 0,42 (Poisson´s ratio) and η = 0,015. The final values of Young´s modulus and loss factor were chosen so as to guarantee a good match between the position and the height of the peaks in the FRF´s. As can be seen, the adopted loss factor value is slightly higher than the measured ones. The Poisson´s ratio value was obtained from bibliography, and numerical tests have shown that its influence on natural frequencies is very small. Receptance curves, both experimental and numerical, are compared in Fig. (9). The numerical receptances were determined by two procedures, both provided by the FEM software: direct calculation and modal approach. The difference in results is negligible. In the same figure, the lower curve presents the experimental coherence between the measured signals. Values of coherence were close to unity, denoting a quite good measurement quality. Mobility curves show good agreement on main flexural peaks. Unfortunately, a torsion mode, just above 1 kHz, presents a reasonable difference between the experimental and numerical natural frequencies.

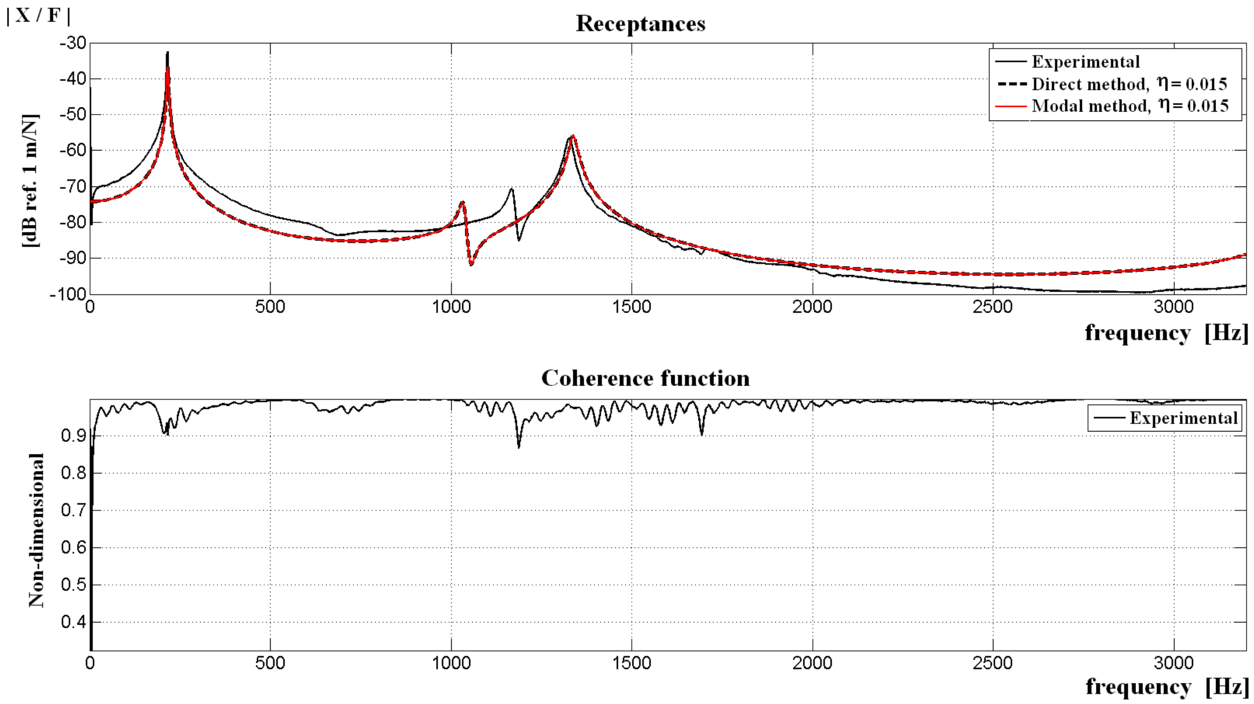

Figure 9. Receptances comparison (upper) and experimental coherence function (lower).

When performing the experimental modal analysis, with the help of the shaker, FRF´s were presented as velocity/acceleration ratios. These curves can be post-processed to obtain displacement/displacement ratios. In this case, such curves, in absolute values, may be called "transmissibility curves" (T_r). It is then possible to import them by Test.Lab software and to perform the experimental modal analysis. As an example, Fig. (10) shows the "sum" curve of all FRF´s obtained this way for the whole case. FRF´s were measured up to 6.4 kHz. Many peaks, corresponding to natural frequencies, can be seen in Fig. (10).

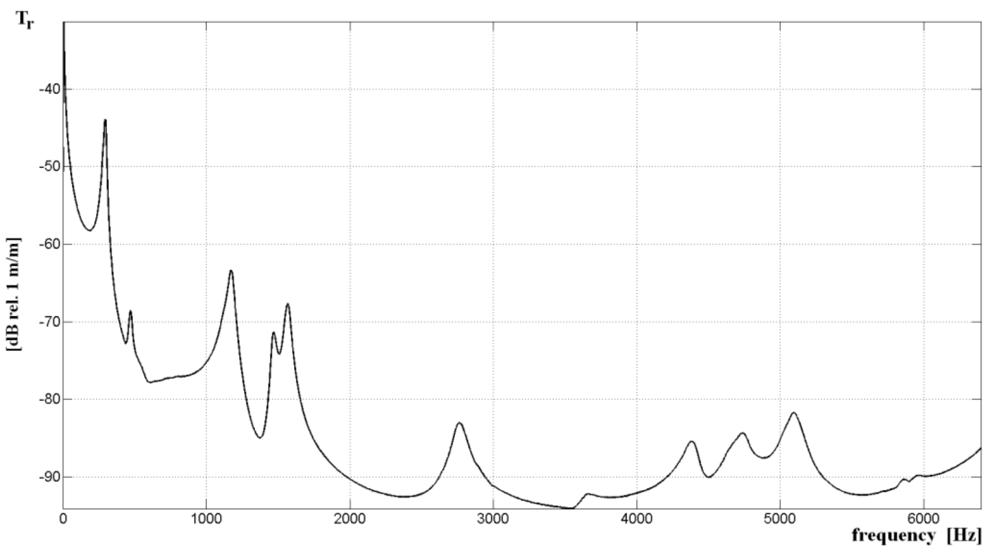

Figure 10. Sum of transmissibility curves.

A numerical modal analysis was performed. The applied material properties were: E = 2,13 GPa, ρ = 1183 kg/m^3, ν = 0,42 e η = 0,04. These values were obtained by an optimization process, based on a genetic algorithm. The aim of the process was the minimization of differences between experimental and numerical parameters, like natural frequencies and loss factors. Table (1) shows a comparison between the results of the two modal analyses that were performed.

Table 1. Experimental and numerical modal parameters.

Mode	Experimental f_n [Hz]	η[%]	Numerical f_n [Hz]	Δf_n [%]
1	289.801	1.40	274.660	-5.2246
2	475.299	3.65	483.340	1.6918
3	1172.886	1.73	1143.700	-2.4884
4	1470.091	1.16	1459.500	-0.7204
5	1563.011	1.40	1489.300	-4.7160
6	2757.015	1.68	2704.200	-1.9157
7	3635.092	1.46	3572.300	-1.7274
8	4389.276	1.49	4341.500	-1.0885
9	5095.677	1.35	4709.800	-7.5726
10	5901.474	1.34	5529.500	-6.3031

Analyzing Tab. (1) it is possible to obtain an average loss factor (η) equal to 1.67%. Furthermore, frequency differences do not exceed 8%, showing again a good numerical model adjustment.

Figure (11) shows a comparison of a few modal shapes. Again, there is good agreement between experimental and numerical results. The two second modes do not look alike, but there is a reason. The main vibrations of this mode occur in a plane that is orthogonal to the direction of the displacements imposed by the shaker, which is the same direction of laser measurements. So the main vibrations are simply lost, and they are not represented.

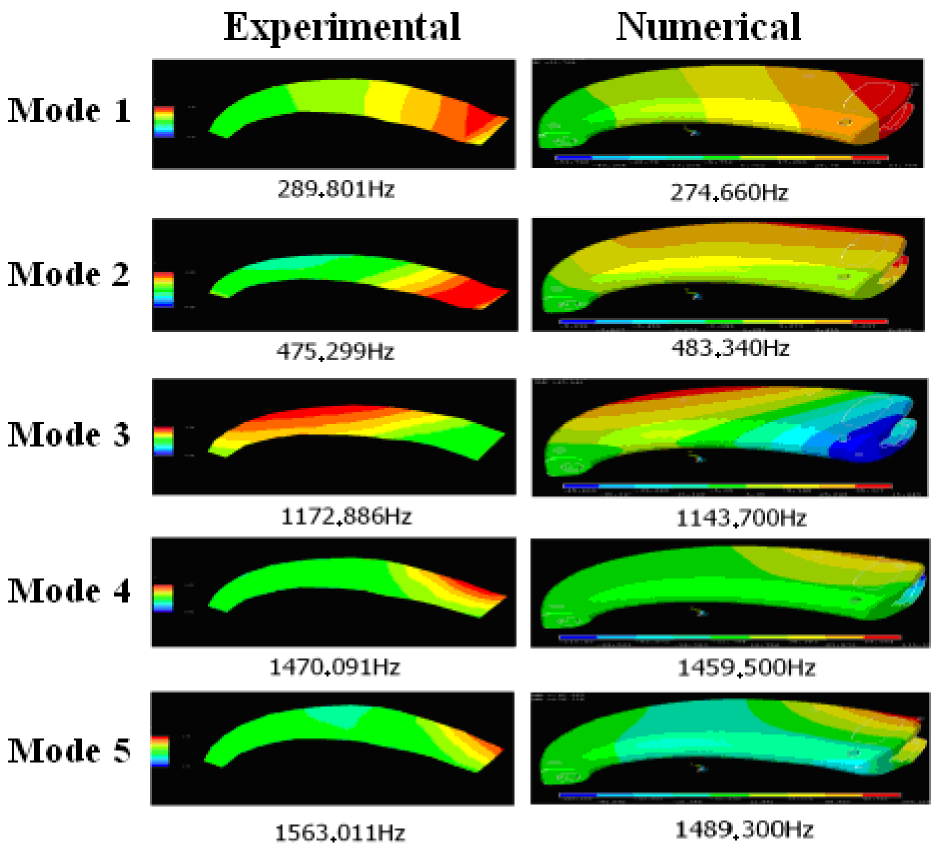

Figure 11. Modal shapes.

A final comparison is made between two transmissibility curves, experimental and numerical, both regarding Point 1 (see Fig. (7)). These curves are presented in Fig. (12). The average difference between the curves is approximately equal to 3 dB. Main resonance peaks are very alike. Results at high frequencies are poorer and could be enhanced by reduction of element sizes, which unfortunately would bring a reasonable and unwanted increase in the computational effort.

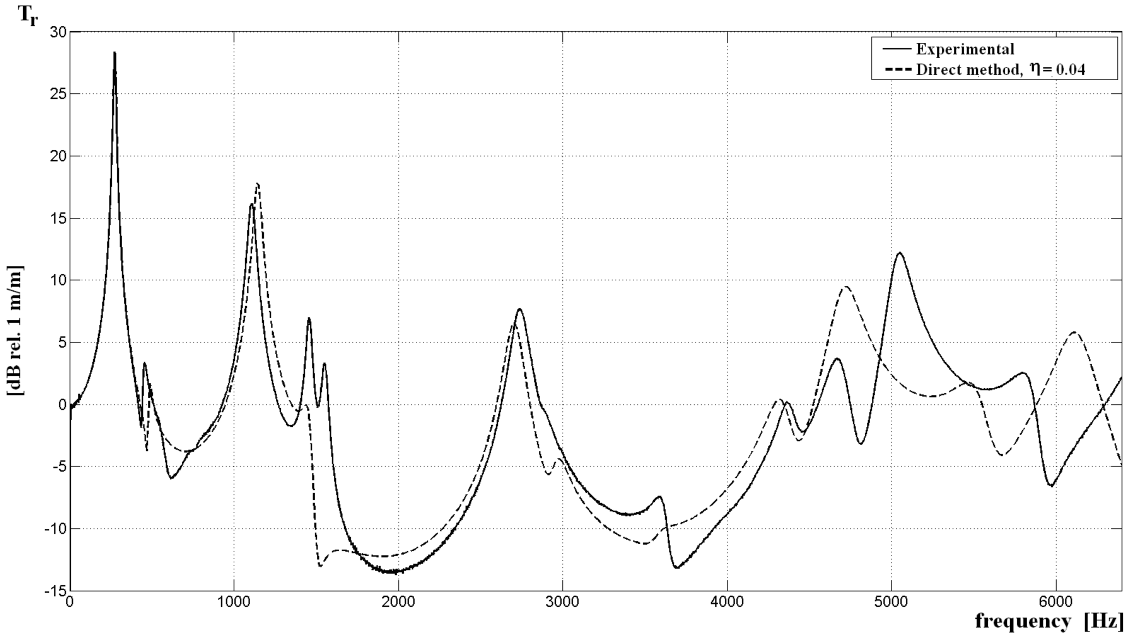

Figure 12. Comparison of transmissibilities at Point 1.

5. CONCLUSIONS

ASTM Standard was efficient in providing initial estimates of material properties. Unfortunately, it was not possible to produce samples according to this standard, because the available amount of material was too little. Non-uniformity in sample thickness and difficulties in cutting the small samples are probably the main reasons for low accuracy. Properties had then to be adjusted, so that experimental and numerical FRF´s came close each other.

Modal analysis gave a good insight on how the hearing aid case physically vibrates. All modes in the analyzed frequency range were investigated. Good agreement between experimental and numerical results was obtained.

Once the numerical model was updated by the experimental data, good match between experimental and numerical FRF´s was obtained, as can be seen in Fig. (12). So it is possible to say that the development of the numerical model was successful.

The proposed study will go on, this good model being the necessary investigation tool. It is now possible to virtually modify existing prototypes or to design new ones, in order to minimize the feedback between loudspeaker and microphone that occurs via structural path.

6. ACKNOLEDGMENTS

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