Experimental determination of air influence on loudspeaker cone vibrations by Scanning Laser Doppler Vibrometry

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ABSTRACT

At present the standard procedure of loudspeaker design involves a time consuming path consisting of the production of a limited series of prototypes that are then tested and evaluated in order to check their conformity to prescribed features. If significant discrepancies are found between designated and actual behavior, this procedure is iteratively repeated till an optimal result is reached. Doing so design time may become unacceptably long and makes difficult a systematic approach to loudspeaker development. For these reasons loudspeaker companies are implementing software packages that simulate loudspeaker response and completely eliminate the need of going through the above mentioned methodology. The approach agreed by FAITAL and the Dept. of Mechanics to model loudspeakers diaphragms is based on the simultaneous use of Finite Element Modeling (FEM) software packages and validation measurements by Scanning Laser Doppler Vibrometry (SLDV). Due to the structural complexity of loudspeakers, FEM models may require an excessive number of nodes and calculation time would rise to unpractical limits. To ease simulation requirements, it has been decided to eliminate air contribution from computer simulations. In this work we will experimentally determine which is the contribution of air on a loudspeaker cone vibrations, defining the percentage deviation from the normal functioning situation.

Keywords: FEM modeling, loudspeakers, vibrations, laser vibrometer.

1. INTRODUCTION

The loudspeaker history spans more than one hundred years, in fact Ernst W. Siemens was the first to describe the "dynamic" or moving-coil transducer in 1874[1], with a circular coil of wire in a magnetic field and supported so that it could move axially. However, he did not use his device for audible transmission, as did Alexander G. Bell who patented the telephone in 1876. Only four years later the German researcher patented the first horn, but it took more than twenty years to see the production of the first paper conical diaphragm terminating with a corrugated rim, patented by John Stroh in 1901. The centering device of a loudspeaker, the so called “spider”, was invented in 1908 and finally in 1925 Chester W. Rice and Edward W. Kellogg at General Electric wrote the research paper[2] that established the basic principle of the direct-radiator loudspeaker with a small coil and driven mass controlled diaphragm in a baffle with a broad midfrequency range of uniform response. Other important steps in loudspeaker history briefly followed, with Thuras defining the bass reflex enclosure in 1930 and the creation of the first two way system in 1931 at Bell Labs.
In the 50’s Edgar Villchur at Acoustic Research developed the acoustic suspension principle that led to the presentation of the small AR-1 bookshelf loudspeaker in 1954, see Figure 1. Seminal papers of Thiele and Small that defined the most diffused loudspeaker model and modern practice of loudspeaker box design appeared in 1971 and 1973[3-6].

In parallel with loudspeaker structural and acoustical improvements, researchers worried about mechanical behavior of loudspeaker components, with the cone and surround receiving most attention. Beranek[7] described cone modes, while Small indicated the upper frequency limit for the so called “piston” range of operation of a loudspeaker, where it can be considered acting as a perfect vibrating surface with all points moving in phase. Loudspeaker response modeling for

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frequencies beyond this limit and for non linear long excursion behavior has been studied by many authors, with papers describing an analytical approach (Frankort[8], Barlow[9], and Klippel[10]), a FEM approach (Shindo[11], Suzuki[12], and Kaizer[13]), and a mixed one (Shepherd[14]). The simplification of not including air load has been adopted in all three papers using the FEM approach, but none of them presents a theoretical or experimental justification of this assumption. More recent works (Beltran[15], and Rausch[16]) consider radiation load in their simulations, at least on one side of the loudspeaker, but avoid 3D studies of cone vibration, which is instead one of the goals of the ongoing research at the Dept. of Mechanics. Other authors have already mentioned the use of Laser Doppler Vibrometry for loudspeaker characterization (Bregant[17], and Rossi[18]) but they did not implemented any model of the vibrations they observed. In 1997 Shively[19] presented a study where FEM, vibrometry and even Boundary Element Modeling (BEM) have been employed and, at the knowledge of the authors, it is the only work that presents an approach similar to the one proposed in this article.

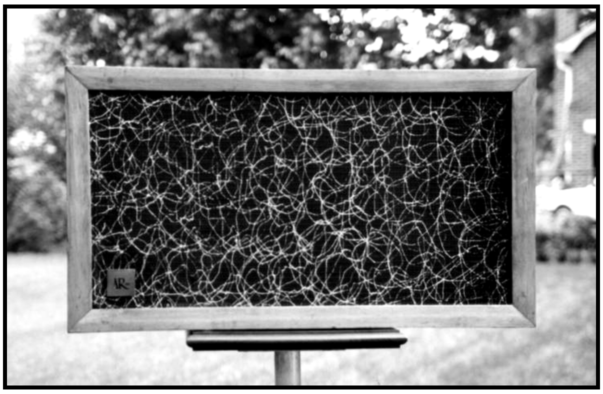

Figure 1. AR-1 system.

2. LOUDSPEAKER OPERATION

In this paragraph we will briefly present basic explanation of how a loudspeaker works and more relevant deviations from linear operation.

2.1 Basic working principle

The loudspeaker is an electro-mechanical-acoustical transducer whose behavior is dependant on the interactions among its components, both electrical and mechanical ones that are schematically described in Figure 2. Signal current coming from power amplifier pass through the leads and reaches the voice coil. The interaction between the current and the static magnetic field, due to the permanent magnet, generates a force on the voice coil itself whose value is expressed as:

$$F = B*l*i \qquad (1)$$

where

i = current (A)
B = flux density in magnetic gap (T);
l = lenght of voice coil thread immersed in magnetic flux (m).

Voice coil motion is transferred to the cone that will move ideally axially following the signal current variations. Consequently cone movements generate a modification to ambient static pressure that constitutes the sound we hear.

The simplest mechanical equivalent of a loudspeaker is a second order system formed by a mass, a spring and a damper. The spring action, or compliance C_{ms}, is due to the spider/suspension combination, while the mass is usually identified with the term "moving mass", M_{ms}, indicating that only a percentage of the actual mass of moving parts is taken into consideration. Finally, energy loss is mainly due to internal friction and usually referred as a resistive term, R_{ms}.

As all second order systems, also loudspeaker posses a resonance frequency, f_s, and a correspondent mechanical quality factor, Q_{ms}.

We may wrote:

$$f_s = 1/(M_{ms}*C_{ms})^{1/2} \tag{2}$$

$$Q_{ms} = 2\pi f_s*M_{ms}/R_{ms} \tag{3}$$

Taking into account also the electrical components of the loudspeaker it is possible to define a general Q:

$$Q_{ts} = 2\pi f_s*M_{ms}*R_e/((B*l)^2 + R_{ms} + R_e) \tag{4}$$

Re = DC voice coil resistance (Ω)

The acoustic output may be quantified by the mid band level, the so called SPL, Sound Pressure Level, defined as the pressure level measured at a distance of 1 meter with 1 watt of input electrical power. Usually the SPL is expressed in dB, using 20 μPa as the reference value:

$$SPL_{dB} = 20\log_{10}[(e_g*Bl*S_d*\rho_o/(2\pi*Mms*(R_e+R_g))/20*10^{-6}] \tag{5}$$

e_g = input voltage (V);
S_d = equivalent loudspeaker surface (m^2);
ρ_o = air density (kg/m^3);
R_g = amplifier output resistance (Ω), usually ≈ 0 for most common amplifier.

Knowing SPL, fs and Q_{ts} it is possible to draw an approximate frequency response of the loudspeaker employing the following relation:

$$SPL_{dB(fs)} = SPL_{dB} + 20\log(Q_{ts}) \tag{6}$$

As an example in Figure 3 we show the simplified response of a loudspeaker with f_s = 58 Hz, SPL = 87 dB and Q_{ts} = 0.4 mounted in a 10 liter closed box. For comparison we also present curves for Q_{ts} = 0.7 and 1.5.

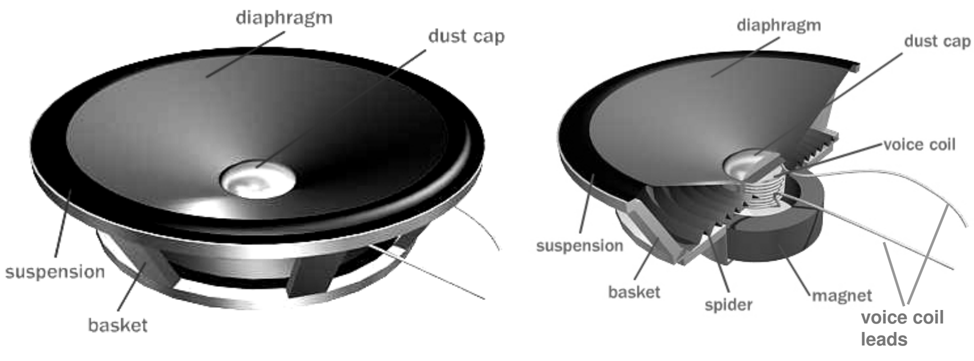

Figure 2. Loudspeaker schematic.

2.2 Limits of simple theory

Relations presented in paragraph 2.1 hold their validity for a certain frequency range, the already defined "piston" range, where the loudspeaker may be correctly represented by a lumped element circuit. This frequency range is limited by the value $f_{max} \leq c/2\pi a$, where c is the speed of sound (344 m/s in air) and a is the equivalent radius of the loudspeaker. For example, a 160 mm woofer will have an f_{max} of about 1053 Hz. Fundamental studies that founded modern loudspeaker systems design techniques by Thiele and Small dealt with this low frequency region and excellence of results still demonstrates the usefulness of this simplified approach, but beyond f_{max}, the loudspeaker cone does not behave anymore like a rigid piston and structural modes does appear[20]. Discrepancies from the prescribed behavior will also show up if the frequency is below f_{max} but high input levels and consequent long excursions drive the loudspeaker out of its linear range, inducing many sorts of non linear distortions in loudspeaker response[21]. As an example we show in Figure 4 an actual measurement of a 160 mm loudspeaker stiffness when cone is displaced of about ±5 mm. It sis evident that even for moderate displacements, less than 1 mm, the value of C_{ms} cannot be considered constant, making preceding relationships strongly dependent on signal input.

As we have already seen, other authors have tackled this difficulties by implementing sophisticated models but there not seem to be present a practical, efficient modeling approach capable of resolving industrial problems, especially when dealing with industrial mass production.

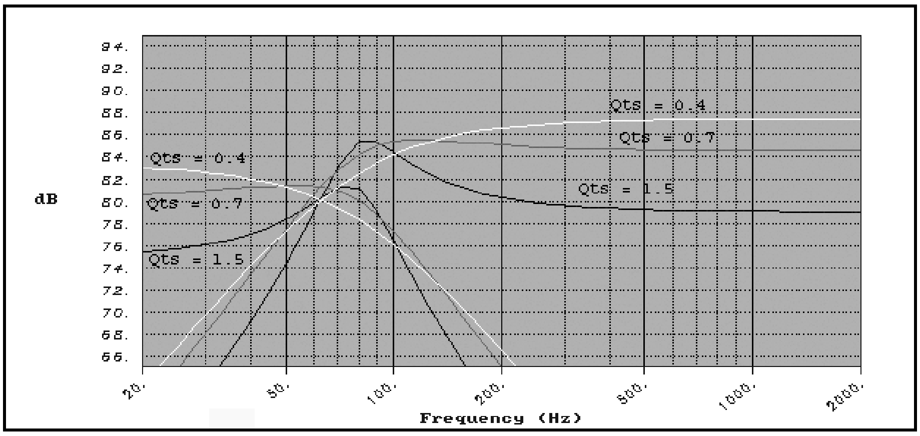

Figure 3. Frequency response of loudspeaker.

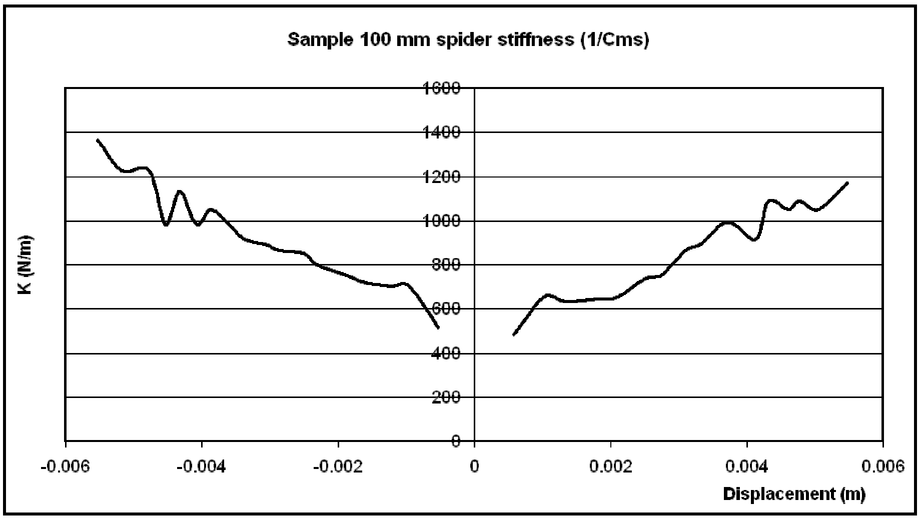

Figure 4. Sample spider static stiffness.

3. FINITE ELEMENT MODELLING

3.1 Program description.
The finite element method program utilized in this work is ANSYS release 5.5, produced by Swanson Analysis Systems, based in Houston, Pennsylvania.
Typical features of this program are manifold and powerful: it allows to perform analysis in structural mechanics, dynamics, heat exchange and fluid flows problems. In the mechanical field it is possible to deal with linear elastic, non-linear, plastic, temperature dependent materials. The dynamical analysis section calculates the natural frequencies, modes of structures and their frequency response function when bodies undergo vibrations caused by sinusoidal forces or displacements. Moreover program allows to perform steady state and transient analyses and has also a section dedicated to electrostatics and electromagnetic problems.

The wide choice of element type gives the best way to solve the most different problems related with engineering field.
In structural problems boundary conditions can be different: displacements, forces, moments, pressures, temperatures, velocities or accelerations.
This program allows the user to choice among different calculation algorithms: for example frontal solver, JCG (Jacobi Conjugate Gradient), ICCG (Incomplete Cholesky Conjugate Gradient), PCCG (Preconditioned Cholesky Conjugate Gradient.

The three main phases in an analysis are:

1. *Preprocessing* (model construction and setting): during this phase user defines elements, real constants associated with them, defines material properties, draws geometry of the bodies and fill geometric entities with finite elements. This operation is called meshing.
2. *Solution* (boundary conditions definition for model and calculation): in this phase boundary conditions are set, such as loads and displacements constraints. Numerical solution method is also chosen in this phase.
3. *Postprocessing* (results plotting, listing, processing): in this phase all results are presented in form of list or chromatic plot.

Every phase is a different routine, but the most important feature is that ANSYS does not require to use multiple modules, but allows the user to work without exiting the program remaining in the same environment. Moreover ANSYS allows the user to write routines with a simple language and build parametric models controlling all geometric and physical variables in the instructions given by a text file.

For loudspeaker analysis purposes, modal analysis and harmonic response analysis were performed to fix the finite element model, comparing numerical results with measurement data. Large deformation effects were used to compare the results obtained with a model of a loudspeaker spider with static stiffness measures carried out in the Department laboratory.

3.2 Problems related to introduction of air in the model
The effect of the air could be considered in general as an unknown added mass, spring and a damping related to the motion of the loudspeaker cone in a viscous fluid. Unfortunately it is very hard to say how much air has to be considered, which is the modification in the damping factor used in harmonic analysis, and which is the law of variation of these two parameters with frequency. For these reasons in the beginning of our experience in loudspeaker modeling we chose not to take in consideration this effect: only the measures carried out with and without air could establish if there is an appreciable difference and how much it is. Moreover it is not easy to introduce in ANSYS modal and harmonic analysis any effect that is non linear.
One more consideration has to be made: our first target is to understand and reproduce the behavior of the structure that constitutes the loudspeaker in the working conditions. For this purpose we investigated first the material features analysing each component separately in order to reproduce the mechanical behavior. Only if the problems related with this aspect are solved it is conceivable to add another effect more complicated like that of the air.

4. MEASUREMENTS ON SAMPLE LOUDSPEAKER

4.1 Preliminary acoustic mesurements

A 160 mm woofer for automotive use has been employed in our experimental study. This loudspeaker is formed by a plastic basket, paper cone and rubber surround with the spider formed in impregnated fabric. Loudspeaker has been mounted on a large wood panel and its response has been measured in the Department semi-anechoic chamber. Employed equipment consist of a Type 1027 Bruel & Kjaer generator delivering pink noise as the input signal, a Bruel & Kjaer PULSE multi channel acquisition system set at 1/3 octave analysis and a Bruel & Kjaer Falcon 4188 pre-polarized capacitor microphone. Loudspeaker frequency spectrum, see Figure 5, shows a peak at about 4 kHz, a moderate well controlled hump at about 100 Hz and a flat response elsewhere. Also from Figure 5 we may see that the frequency band of interest maybe considered as limited to 80-5000 Hz

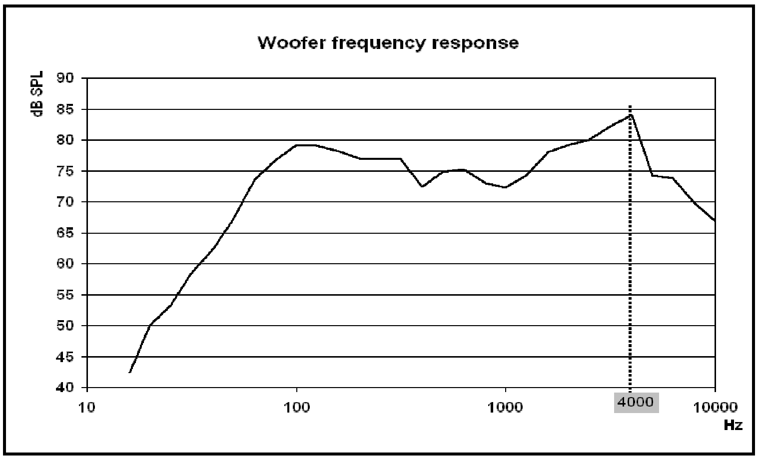

Figure 5. Sample woofer frequency response.

4.2 Equipment and set-up for vibration measurements

The scheme of the set-up for the measurement of cone vibrations is reported in Figure 6(a). Signal generator is included in the SLDV system, an Ometron VPI 4000 (for details on SLDVs see Castellini[22]), while a 30 watt Gearing&Watson amplifier provided signal amplification. The air tight cylindrical box has been realized in two versions, namely a plastic and a steel one, see Figure 6(b), both closed on top by a removable 8 mm thick Plexiglass lid; in the latter case there was also a steel counter flange on top of the lid. Box dimensions are 300 mm (ϕ) x 275 mm (h) and the internal lateral surface has been covered with sound absorbing material.

Measurement procedure consists of the acquisition of the loudspeaker frequency response using white noise in three different fixed points of the cone, one in the center of the dust cap, the second in the middle of the cone, and the last one close to the surround. Measurements have been repeated for six different conditions:

1. Loudspeaker out of the box
2. Loudspeaker into the box at ambient pressure with lid closed
3. Loudspeaker into the box at absolute pressures of 0.8, 0.6, 0.4, and 0.2 bar.

First two acquisitions have been done to check if the box was modifying the loudspeaker response but it has never been the case.

Finally, from the frequency spectra a certain number of resonance frequencies have been selected and complete scans of the cone and surround surfaces have been conducted to compare deformation shapes.

Due to results not showing significant differences among different acquisition points, only measurements regarding the dust cap will be presented.

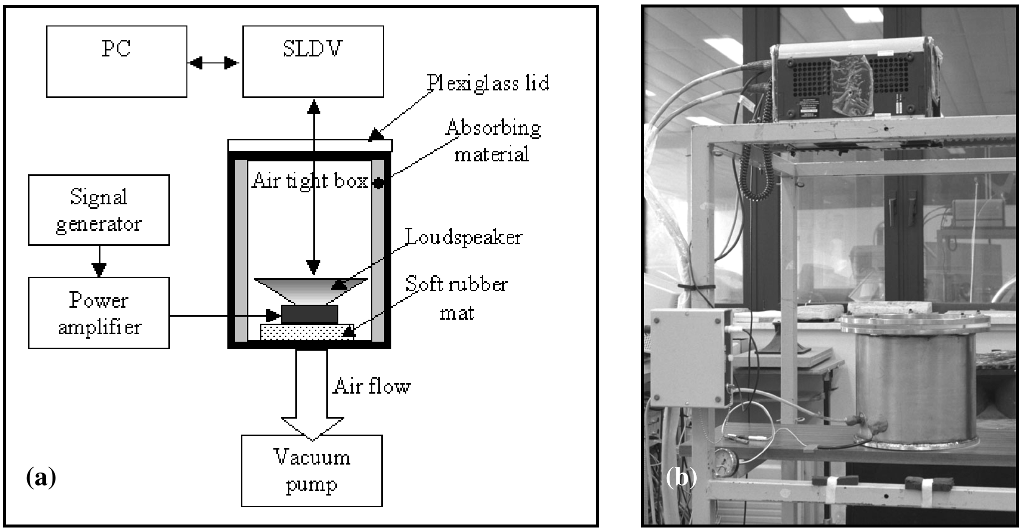

Figure 6. (a) Scheme of measurement set-up (b) Photo of set up with steel box.

4.3 Vibration measurements results

4.3.1 Single point acquisitions on dust cap

In Figure 7(a) we show a first result obtained with the plastic box at ambient pressure and 0.2 bar. Measurement repeatability is low and data interpretation is not easy, so this box has been rejected and the steel one has always been used here after. In Figure 7(b) we report a similar measurement inside the steel box. Apart from an evident response raise in the frequency range around 4 kHz, it is not possible to derive other observations. For this reason we have divided the frequency response in smaller regions and investigated them separately.

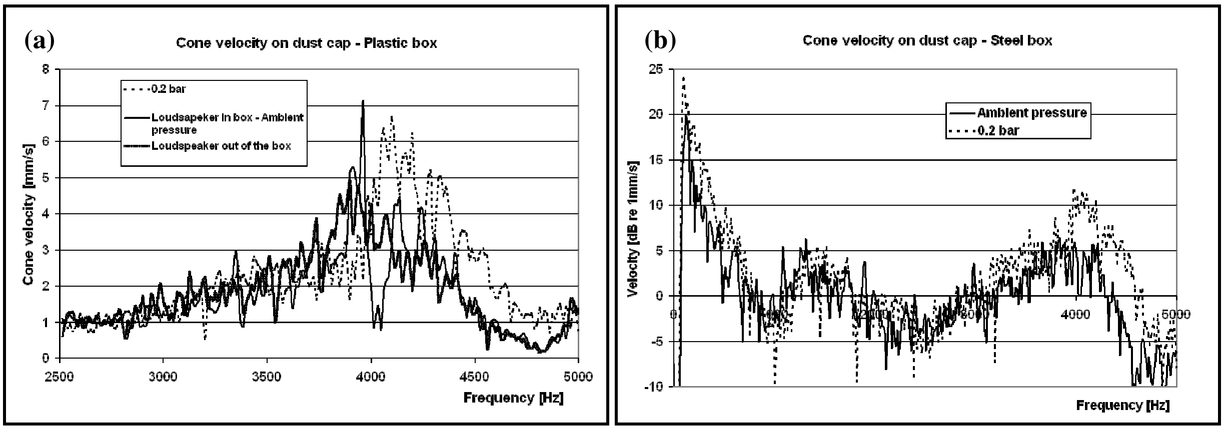

Figure 7. Frequency spectra of vibration velocity of dust cap. (a) Plastic box (b) Steel box.

Figure 8 illustrates the investigation performed between 2750 and 5000 Hz with decreasing values of internal pressure. Loudspeaker behavior does not change for the first two steps, 0.8 and 0.6 bar, while shows a significant deviation at 0.4 bar; decreasing the pressure to the lowest measured value, 0.2 bar, does not introduce any other significant change in spectrum shape, but we note a relevant increase in vibration level at about 4 kHz. In Figure 9(a) we have downshifted the spectra at 0.2 by 100 Hz and now we have a good coincidence of frequency peaks in comparison with the measurement at ambient pressure. Figure 9(b) illustrates the behavior of loudspeaker between 0-500 Hz and we note that in this frequency range the shift is of about 12.5 Hz. For the mid frequency band we present the result obtained in the mid measurement point, and in this case the frequency shift is of about 40 Hz (see Figure 10).

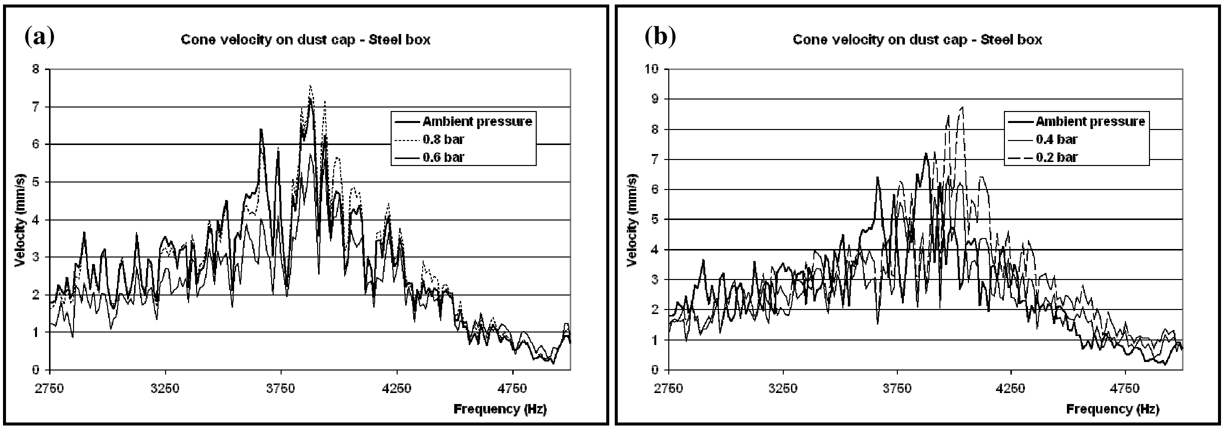

Figure 8. Frequency spectra of vibration velocity of dust cap. (a) Internal pressures of 1 bar (ambient), 0.8 bar, and 0.6 bar (b) Internal pressures of 1 bar (ambient), 0.4 bar, and 0.2 bar.

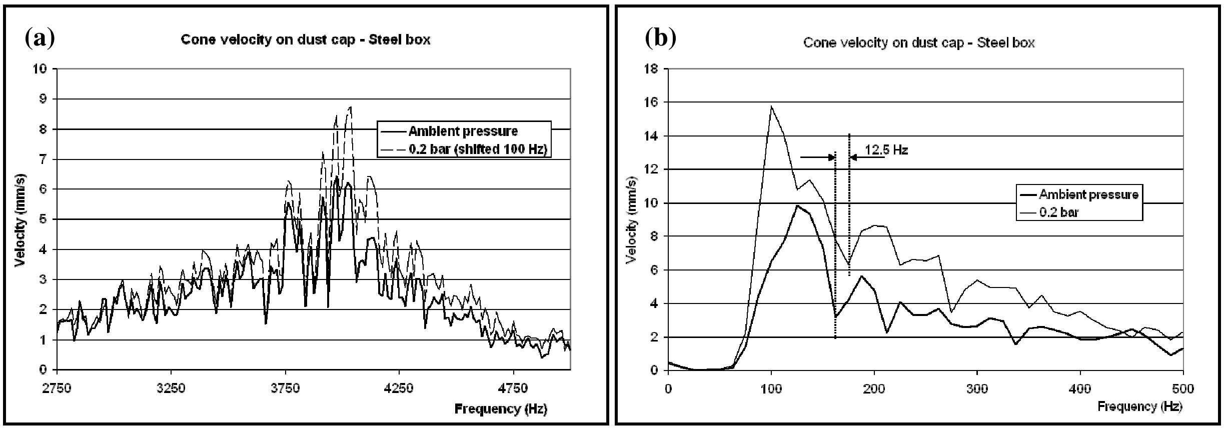

Figure 9. Frequency spectra of vibration velocity of dust cap. (a) Internal pressures of 1 bar, 0.2 bar down shifted of 100 Hz (b) Frequency range 0-500 Hz, internal pressures of 1 bar, 0.2 bar.

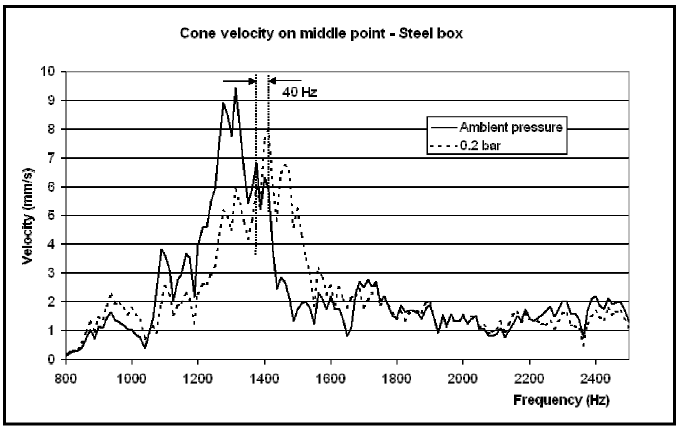

Figure 10. Frequency spectra of vibration velocity of middle point of cone for pressures of 1 bar (ambient) and 0.2 bar.

4.3.2 Vibration modes of loudspeaker

A second measurement session was dedicated to the acquisition of vibration velocity values on all the loudspeaker diaphragm surface. For this scope we employed sinusoidal excitation and performed two different measurements, the first one inside the box at ambient pressure, the second one at 0.2 bar; amplitude and phase values for each point have been recorded. In this work we present as a representative example the results obtained at 3875 Hz (see Figure 11), in correspondence with a high peak of the frequency response. Due to observed 100 Hz shift, map at 0.2 bar has been measured at 3975 Hz (see Figure 12) and we note that there is no relevant difference between the two set of results when the appropriate frequency shift is considered.

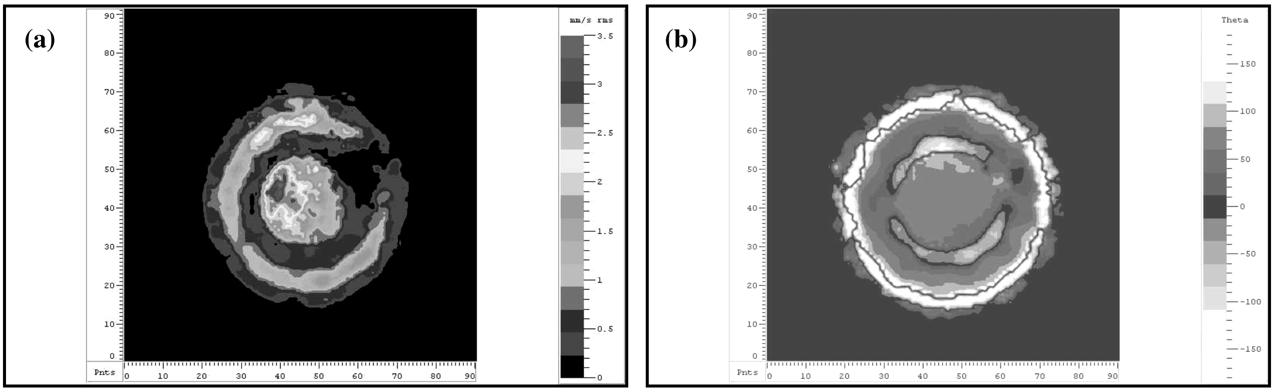

Figure 11. Vibration maps, amplitude (a) and phase (b) of loudspeaker at 3875 Hz. Ambient pressure, inside steel box.

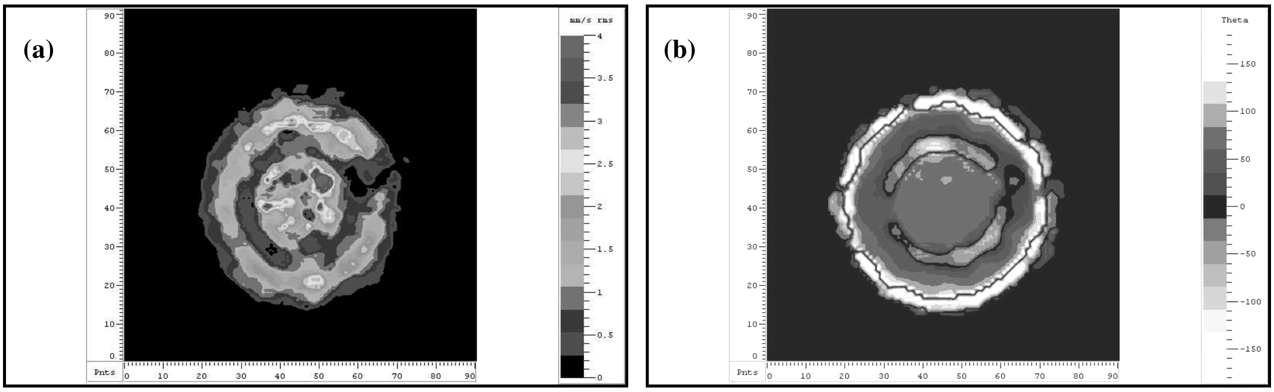

Figure 12. Vibration maps, amplitude (a) and phase (b) of loudspeaker at 3975 Hz. 0.2 bar pressure, inside steel box.

CONCLUSIONS

Aim of this work was to quantify the deviations introduced in a FEM model of a loudspeaker cone vibrations when the air load was not considered. We have divided our tests in three frequency bands, roughly defined as 80-500 Hz, 500-2500 Hz, and 2500-5000 Hz and examined five different values of pressure, namely ambient pressure, 0.8 bar, 0.6 bar, 0.4 bar, and 0.2 bar. For pressure levels greater than 0.4 bar the frequency spectra of the loudspeaker vibrations does not show any relevant modifications; for pressure levels lower than 0.4 bar we have observed an upward frequency shift of measured spectra. Absolute values of this shift depend on frequency band and may be indicated as about 12.5 Hz, 40 Hz, and 100 Hz for the above indicated regions, with corresponding percentage deviations of about 7%, 3% and 2.5% respectively. An upward frequency shift of resonance peaks had to be expected due to the lowering of mass load and we also have a practically constant percentage value from 500 Hz to 5000 Hz. Air pressure reduction also means that we are reducing the acoustic load and the value of the radiation resistance; this fact lower modal damping and contributes to increase the value of the resonance frequencies and of the amplitudes of resonance peaks, as we have observed in all our measurements on the dust cap. An higher percentage shift at low frequencies is probably due to the greater inertial influence of the box internal volume on loudspeaker vibrations but we must further investigate this frequency band.

Anyhow percentage deviations remains quite small when compared with other sources of error in the definition of the FEM model; for example cone material density and thickness are often quoted with tolerances in the order 20÷30 % and in general loudspeaker specifications are always accompanied by tolerances of not less than 20% even for fundamental parameters as f_s.
Vibration maps also showed a constant behaviour with decreasing pressure and this is quite important because it allows us to consider valid our modelling procedure even in the absence of air and small induced errors are rightly justified by important reduction in model complexity and calculation time.

Future work will be dedicated to the improvement of the measurement set up and condition to increase experimental repeatability that has not always been satisfactory, because measuring cone vibrations has not been easy due to the high damping of modes, as can be easily seen in the essentially flat curve of Figure 5. We will also investigate the discrete nature of deviations that seem to happen abruptly at certain pressure values and does not follow a constant smooth trend. In this respect we will try to investigate the effects of lower residual pressures but this will require the implementation of safety measures. Another interesting experiment will be conducted increasing the box internal pressure to see if the observed effects are reversible.

ACKNOWLEDGMENTS

We wish to thank a lot Massimo Rotatori for his precious contribution to this work that owes him many hours of his time. We also want to thank Claudio Santolini for his words of advice while performing the experiments in our laboratory.

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