Study of an Electroacoustic Absorber

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Abstract: In this paper, the underlying concept of electroacoustic absorbers is studied with the help of Comsol Multiphysics® Acoustics Module. An electroacoustic absorber is a loudspeaker which acoustic impedance can be varied by electrical means, be it passive or active [1]. Among the different ways to obtain variable acoustic properties on an electroacoustic transducer's voicing face, there is the shunting of the transducer's electrical input. With such shunt devices, the acoustic impedance that the transducer's membrane presents to the acoustic field takes account of an acoustic equivalent of the electrical load that can take many values within a specified range. It has been chosen to develop a multiphysics model of a passive shunt strategy, using a resistor.

This presentation aims at describing and assessing the performances of this concept as an acoustic damper of the modal behavior in an acoustic waveguide. Measurements in laboratory conditions (impedance tube) will be presented and compared to simulations before discussing on possible means of enhancement.

Keywords: shunt loudspeaker, finite element model, acoustic performances assessment, electroacoustic absorber design.

1 Introduction

Today, noise reduction is an increasing field of studies, fostered by environmental trends in the society. In this purpose, this paper focuses on the design of electroacoustic absorbers. Two approaches are commonly used for simulating their dynamics and for predicting their acoustic performances: lumped element model and finite element model. Although lumped element models are efficient for modeling the linear dynamics in the low-frequency range, they are not sufficient to provide a realistic simulation of the system behavior in a larger frequency range. In this context, we have considered the finite element method for studying and assessing the performances of an electroacoustic absorber.

In this paper, the underlying concept of electroacoustic absorbers is briefly introduced before developing a numerical model with the help of Comsol Multiphysics®. A comparison with experimental measurements will lastly be presented to show the validity of this model.

2 System dynamics modeling

2.1 Electroacoustic absorber dynamics

In the following, we have chosen to develop a model upon the electrodynamic moving-coil loudspeaker, for ease of understanding. The same applies for other type of actuation.

A schematic diagram of a typical moving-coil loudspeaker is shown in Figure 1.

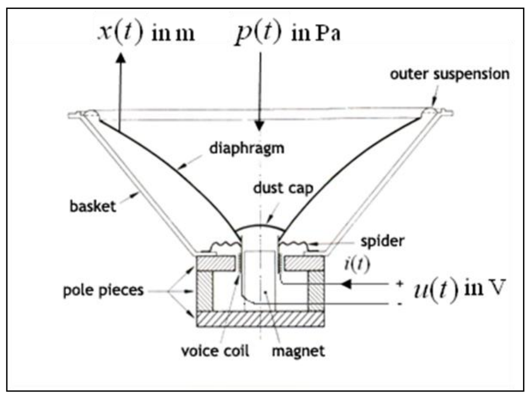

Figure 1: Schematic diagram of a typical moving-coil loudspeaker

Basically, a moving-coil loudspeaker is a physical system that can be described with linear differential equations derived from the Newton's law of motion, and the Kirchhoff's equation [2]:

$$\begin{cases} (Bl)i(t) - S_d\, p(t) = R_{ms}\ddot{x}(t) + M_{ms}\dot{x}(t) + \dfrac{1}{C_{ms}}x(t) \\ u(t) = R_e i(t) + L_e\dfrac{di(t)}{dt} + (Bl)\dot{x}(t) \end{cases} \tag{1}$$

where i is the induced current in the coil (in A), p the acoustic pressure (in Pa), u the applied voltage (in V), v the velocity of the cone (in m/s) and x the displacement of the cone (in m).

The parameters of Table 1 are intrinsically tied to the loudspeaker.

Name	Symbol	Units
Diaphragm area	S_d	m^2
Mechanical resistance	R_{ms}	$N.s.m^{-1}$
Moving mass	M_{ms}	kg
Mechanical compliance	C_{ms}	$m.N^{-1}$
Electrical resistance	R_e	Ω
Electrical inductance	L_e	H
Force factor	Bl	$N.A^{-1}$

Table 1: Constituting parameters of a moving-coil loudspeaker

By using phasor arithmetic, these linear equations can be reduced to algebraic ones:

$$\begin{cases}(Bl)\underline{I} - S_d\,\underline{p} = \underline{Z}_m\,\underline{v} \\ \underline{U} = \underline{Z}_e\,\underline{I} + (Bl)\,\underline{v}\end{cases} \tag{2}$$

where $\underline{Z}_m = R_{ms} + j\omega M_{ms} + 1/(j\omega C_{ms})$ is the free mechanical impedance in N.m/s (when $I = 0$), and $\underline{Z}_e = R_e + j\omega L_e$ is the blocked electrical impedance in Ω (when $v = 0$).

Then, by taking account of the closed-box environment, by considering the moving-coil loudspeaker in reverse as an electroacoustic absorber (that is to say with no applied voltage at its terminals) and by shunting the speaker's electric terminals we obtain the following characteristics equations:

$$\begin{cases}(Bl)\underline{I} - S_d\,\underline{p} = (\underline{Z}_m + S_d^{\,2}\,\underline{Z}_{ab})\underline{v} \\ 0 = \underline{Z}_e\underline{I} + (Bl)\underline{v} + \underline{Z}_{sh}\underline{I}\end{cases} \tag{3}$$

where $\underline{Z}_{ab}$ represents the acoustical impedance due to the closed-box environment, and $\underline{Z}_{sh}$ is the shunt electrical impedance. The current through the voice coil becomes:

$$I = -\frac{Bl\;\;v}{Z_e + Z_{sh}} \tag{4}$$

2.2 Acoustic waveguide

For simplicity in developing the finite element model and simulating the acoustic response of an enclosed sound field, a rigid-walled enclosure with low-frequency response dominated by a single dimension was selected for the purpose of this study.

A schematic diagram of the acoustic domain coupled with the electroacoustic absorber is depicted in Figure 2.

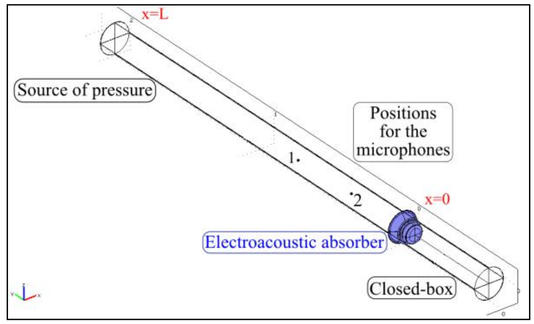

Figure 2: Acoustic domain coupled with the electroacoustic absorber

The governing equations relevant to describe the acoustic dynamics are the fundamental equations of fluid mechanics: mass conservation, equation of state and Euler's equation of motion. By combining them and solving for the acoustic pressure in the frequency domain, we obtain the Helmholtz equation for the spatial variable *x*:

$$\frac{\partial^2 p}{\partial x^2}(x) + k^2 p(x) = 0 \tag{5}$$

for $x \in \; 0;L$, which is subjected to the following closed-end boundary conditions:

$$\begin{cases}p(L) = p_L \\ \dfrac{\partial p}{\partial x}(0) + jk\beta_0\,p(0) = 0\end{cases} \tag{6}$$

where p_L is the imposed sound pressure at $x = L$, $k = \omega/c_0$, and β_0 is the normalized specific admittance at $x = 0$, which depends on the electroacoustic absorber dynamics.

The harmonic solution in the $0;L$ domain depends on the imposed sound pressure p_L and the dynamics of the absorber β_0. It is given by:

$$p(x) = p_L\,\frac{(\beta_0+1)e^{-jkx} - (\beta_0-1)e^{jkx}}{(\beta_0+1)e^{-jkL} - (\beta_0-1)e^{jkL}} \tag{7}$$

3 Numerical model

3.1 Loudspeaker structural properties

For modeling the woofer of our study (Visaton® AL-170), we used the lousdspeaker driver model from Comsol Multiphysics® as a starting point [4]. The model starts out with the small signal analysis from the AC/DC to compute the driving force and the blocked voice coil electrical impedance. 2D axi-symmetry acoustics is also selected to develop the model.

Geometric parameters are adjusted in order to fit the loudspeaker technical data. For instance, the radius of the coil was reduced; the remanent flux density in the magnet B_0 and the number of turns in coil N were adjusted to obtain the desired force factor Bl.

About material properties, the Young's modulus E and the Poisson's ratio ν are configured as real materials. Thus, the dust cap and the diaphragm are in aluminum, the spider and the outer suspension in rubber, the voice coil support in polyester and the coil in copper. In order to obtain the desired mass for each element, the density ρ is adjusted while respecting the proportions of the speaker.

Table 2 summarizes the material properties used within the model.

	E **(Pa)**	$\boldsymbol{\nu}$	$\boldsymbol{\rho}$ **(kg/m³)**	***m*** **(g)**
Dust cap	7e10	0.33	2700	1.1
Diaphragm	7e10	0.33	140	0.8
Spider	1e7	0.45	215	0.9
Outer suspension	1e7	0.45	405	1.2
Coil support	3.8e9	0.37	1500	0.6
Voice coil	1.1e11	0.30	8700	7.5

Table 2: Materials properties of Visaton® AL-170

In most loudspeaker specifications, the suspension is characterized by a mechanical compliance C_{ms} and resistance R_{ms}. In order to keep R_{ms} constant over a range of frequencies, the material needs to have a damping factor that increase linearly with frequency.

Consequently, we used a Rayleigh damping ζ [5] which relates to the mechanical quality factor Q_{ms} as expressed in the following relation:

$$\zeta = \frac{1}{2}\left(\frac{\alpha_R}{\omega} + \beta_R \omega\right) = \frac{1}{2Q_{ms}} \tag{8}$$

where we assume the coefficient $\alpha_R = 0$ (in s^{-1}).

As a result, the coefficient β_R (in s) is given by:

$$\beta_R = \frac{1}{\omega Q_{ms}} = \frac{\eta_0}{\omega_0} \tag{9}$$

where η_0 = 0.26 is the loss factor obtained from the mechanical quality factor at the resonance of the driver ω_0.

3.2 Acoustic performances assessment

When assessing the acoustic performances in this model, the electroacoustic absorber is set up in an empty cylindrical enclosure, as illustrated in Figure 2. The acoustic domain is a tube configured with rigid walls (that is to say sound hard walls), whose dimensions are equal to those of the experimental impedance tube. The medium is filled with air whose density is ρ_0 and celerity of sound c_0. The source is modeled with an imposed pressure condition p_L placed at one-end of the tube. Please refer to nomenclature for more details about constants and parameters.

For computing the sound absorption coefficient, we have used an approach based on impedance tube assessment, after ISO 10534-2 standard, using the two-microphone transfer function method [3]. The reflection coefficient r, ratio of the complex amplitudes of the incident and reflected acoustic waves, is calculated by assuming there is no attenuation within the tube:

$$r = \frac{H_{12} - e^{-jks}}{e^{jks} - H_{12}} e^{2jkx_1} \tag{10}$$

where the transfer function H_{12} is the ratio of the Fourier transforms of pressure signals p_1 and p_2 (see Equation (7)), computed in the tube at positions x_1 and x_2 in Figure 2, which are separated of an s distance. The sound absorption coefficient is derived from r according to the following relation:

$$\alpha = 1 - |r|^2 \tag{11}$$

Numerical computations were performed using the parametric direct UMFPACK linear system solver and a triangular mesh of about 67000 elements between 40 Hz and 400 Hz per step 2 Hz. Thus, due to the positions of the microphones and the dimensions of the tube, the hypothesis of plane wave is borne out. The results are presented in the following section.

4 Result and discussion

4.1 Thiele and Small parameters

The Thiele and Small parameters can be extracted from the numerical model. A comparison between the model parameters and those from the Visaton® AL-170 data sheet is given in Table 3.

Symbol	Unit	Technical data	Comsol model
a	cm	7.4	7.5
S_d	cm^2	133	137
f_s	Hz	38	37*
R_e	Ω	5.6	5.6
L_e	mH	0.9	3.2*
Bl	N/A	6.9	6.7*
R_{ms}	Ns/m	0.8	0.78*
M_{ms}	g	13	12.1
C_{ms}	mm/N	1.35	1.4**
V_{as}	L	34	39**
Q_{ms}		3.88	3.9**
Q_{es}		0.43	0.38**
Q_{ts}		0.39	0.35**
$R_{sh,opt}$	Ω	4.1	3.2

Table 3: Thiele and Small parameters (*values extracted from Comsol Multiphysics®, **values calculated with formulae given in annex)

4.2 Electrical impedance

In order to check the validity of the Visaton® AL-170 model, we have first computed the total electrical impedance which is very characteristic of loudspeaker driver.

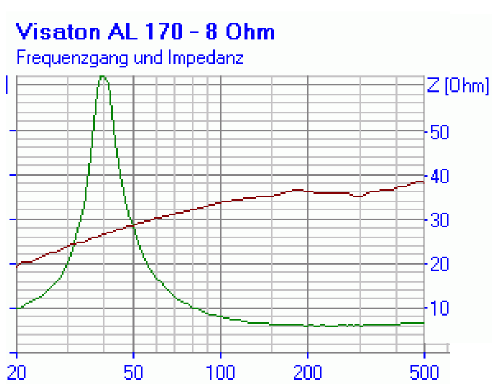

Figure 3: Frequency and impedance response from Visaton® data sheet

As depicted in Figure 4, the computed curve is compared to the reference curve of the loudspeaker data sheet given in Figure 3.

As expected, the computed electrical impedance curve is very close to that from the Visaton® AL-170 data sheet. The peaks at the mechanical resonance appear at the same frequency and they have almost the same magnitudes.

Due to the good agreement between numerical results and technical data, we can validate the loudspeaker model and use it for the computation of the absorption coefficient.

4.3 Absorption factor

In order to evaluate computed acoustic performances with experimental measurements, various configurations have been investigated by varying the shunt resistance. For comparing with the electroacoustic absorber in open circuit we entered $R_{sh} = 10^6\ \Omega$ in the model; for the closed circuit configuration, we entered $R_{sh} = 0\,\Omega$. Some intermediate values within these bounds have also been tested, including the optimal shunt resistance as regards loudspeaker's design which can be found as:

$$R_{sh,opt} = \frac{(Bl)^2}{(\rho_0 c_0 S_d - R_{ms})} - R_e \qquad (12)$$

The Figure 5 and Figure 6 illustrate the computed and measured normalized specific impedances and absorption coefficients in open circuit and closed circuit configurations.

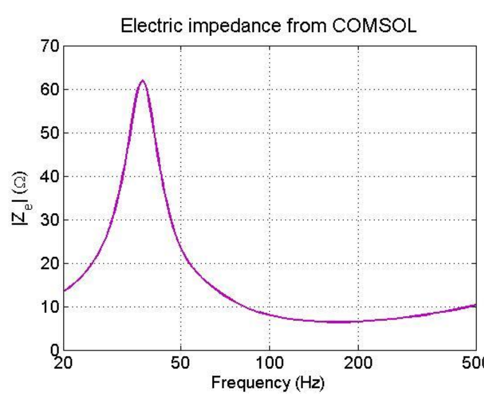

Figure 4: Total electrical impedance from Comsol Multiphysics®

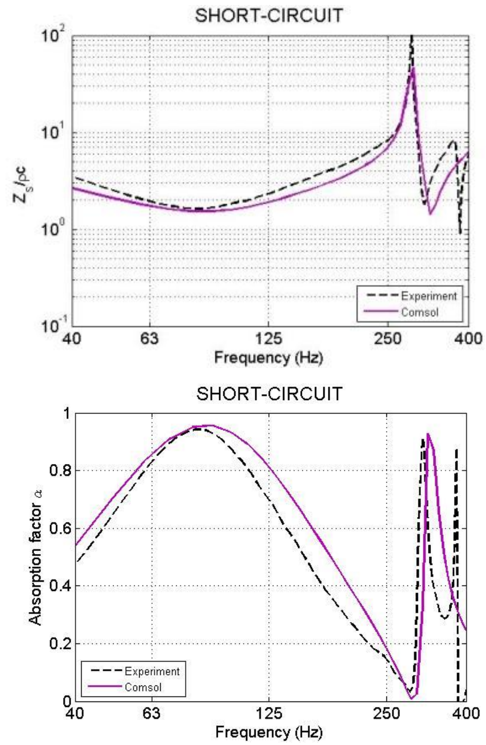

Figure 5: Measured and computed acoustic performances of the absorber in closed circuit: (top) specific normalized impedance, (bottom) absorption coefficient

As clearly shown in these figures, both computed and measured curves match rather well. These comparisons allow us to validate the model for studying and assessing the performances of an electroacoustic absorber.

However, some enhancements can be envisaged to get more accurate results. For instance, the slight shift in frequency between measured and computed results may be reduced by adjusting the resonance frequency of the electroacoustic absorber model so that it fits the measured one perfectly. Moreover, the parameterization of the software can also be optimized. Indeed, an optimization of the computation time can be envisaged by improving the mesh. Because this was not done at the beginning, it raises the problem of the accuracy of our results. In this purpose, various meshes have been tested and enable us to validate our model. Indeed, the same absorption coefficients have been found in our frequency range. However, in the prospect of higher frequencies studies, it would be interesting to take a look more carefully to optimize computation time.

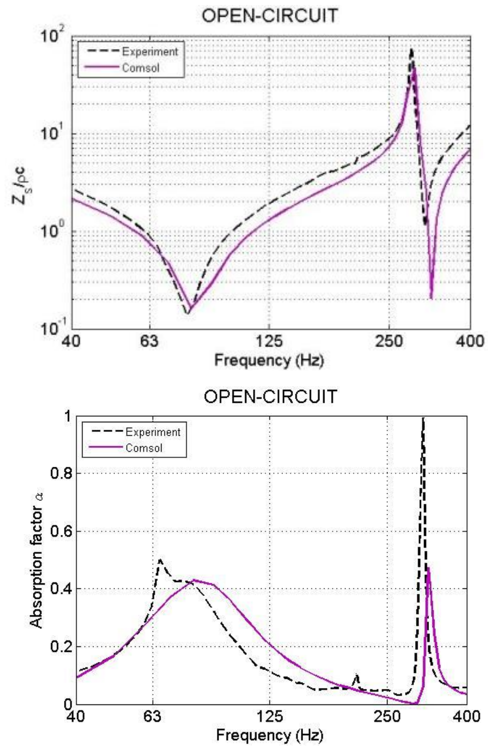

Figure 6: Measured and computed acoustic performances of the absorber in open circuit: (top) specific normalized impedance, (bottom) absorption coefficient

The Figure 7 illustrates the computed absorption coefficient for various shunt resistance configurations. As illustrated in Figure 7, shunting the loudspeaker allows to vary the quality factor, and therefore, to provide a full absorption at the resonance. As expected, it can be observed that the predicted optimal shunt resistance provides the best absorption at resonance.

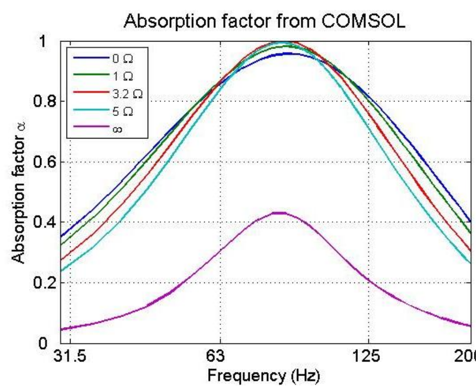

Figure 7: Computed absorption coefficient for various shunt resistances

5 Conclusion

In this paper, a finite element model of an electroacoustic absorber has been presented and demonstrates good agreement with standard impedance tube measurements. Starting from the finite element model, the effect of including a resistive load at the terminals of the loudspeaker has been investigated. As expected, numerical results clearly show that such a passive component is able to vary the dynamics of the electroacoustic absorber at resonance within certain bounds. Numerical results have been faced up to experimental measurements, after ISO 10534-2 standard, in order to validate the model. The predictions so far have dealt only with incident sound wave normal to the surface of the absorber, as it is a useful case that closely matches to measurement in an impedance tube.

Further models will be done to optimize the computation time and enhance the accuracy. Modeling will thus be used for improving the concept of electroacoustic absorbers.

6 References

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7 Nomenclature

Model parameters:

a	loudspeaker radius (m)
E	Young's modulus (Pa)
f_s	resonance frequency (Hz)
k	wave number
m	mass (kg)
ν	Poisson's ratio
Q_{es}	electrical quality factor
Q_{ms}	mechanical quality factor
Q_{ts}	total quality factor
ρ	density (kg/m^3)
R_{sh}	shunt resistance (Ω)
V_{as}	equivalent volume (m^3)
ω	angular frequency (rad/s)

Constants:

B_0=1.1 T	Remanent flux density in magnet
c_0=343 m/s	Celerity of sound
D_{cb}=0.15 m	Back enclosure inner diameter
D_t=0.15 m	Main tube diameter
L_{cb}=0.6 m	Back enclosure depth
L_t=2 m	Main tube length
N=125	Number of turns in coil
ρ_0=1.25 kg/m^3	Density of air
s=0.45 m	Space between microphones 1 and 2
p_L=1 Pa	Imposed pressure condition at $x = L$
x_1=0.8 m	Position of microphone 1

8 Annex

Thiele and Small parameters calculation:

According to Table 2, the dynamically moved mass is $M_{ms} = 12.1g$

Consequently: $M'_{ms} = M_{ms} + M_{ar} = 13.1g$ where $M_{ar} = 8/3\rho a^3 = 1g$ is the acoustic mass of radiation.

Then, we can calculate the mechanical compliance $C_{ms} = 1/((2\pi f_s)^2 M'_{ms})$ and the equivalent volume $V_{as} = \rho_0 c_0{}^2 S_d{}^2 C_{ms}$

The mechanical, electrical and total quality factors are given by:

$Q_{ms} = \omega_s M'_{ms} / R_{ms}$,

$Q_{es} = \omega_s R_e M'_{ms} / (Bl)^2$ and

$Q_{ts} = Q_{es} Q_{ms} / (Q_{es} + Q_{ms})$, respectively.