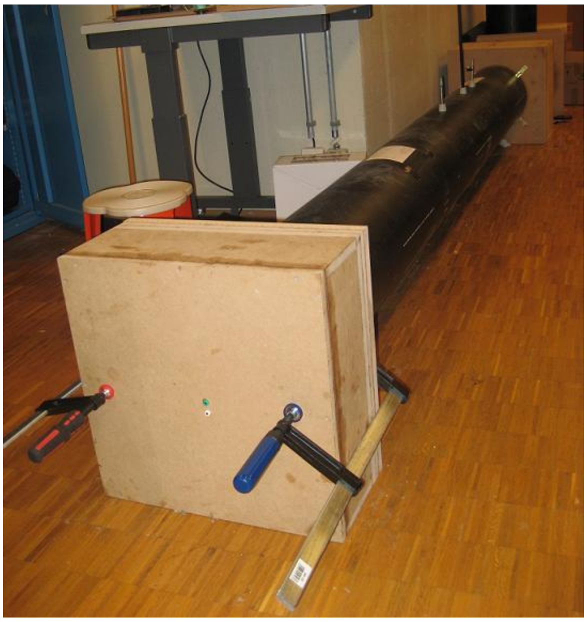
Study Of An Electroacoustic Absorber



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1. Preamble

1.1 Justification of the work and objectives

The problem of low-frequency acoustic noise attenuation still remains in spite of the fact that it has been widely studied. The techniques for absorbing high-frequency noise (greater than 500 Hz) such as the use of porous materials, Helmholtz resonators and mufflers not offer acceptable results at lower frequencies. Porous damping materials rely on the viscous damping of fluid low over a surface. And particle velocity is proportional to frequency, so for trying to absorb frequencies under 500 Hz, it would be necessary to use impractical volumes of material. The same problem appears with the Helmholtz resonators. They provide excellent attenuation of highly resonant acoustic modes but require restrictively large volumes for low frequencies.

This inadequacy of the traditional passive damping treatments has motivated the development of new techniques for absorbing low-frequency reverberant noise with loudspeakers and active control materials.

In this work we develop the design of a passive shunting loudspeaker, based on the use of a resistor as the energy absorber. The optimization of the loudspeaker parameters and the value of the shunting resistor are tested.

1.2 State of actual studies

According to the Niederberg's *Smart dumping materials using shunt control*, we can grope the non-traditional acoustic noise control treatment into 5 categories: 1) Passive baffles and compliant panels, 2) Feedforward noise control, 3) Feedback noise control, 4) Impedance Based, and 5) Electrical shunt control.

1.2.1 Passive absorbers

Helmholtz resonators can provide greater than 20 dB attenuation of lightly damped acoustic modes, and have been extensively used for these activities. A secondary acoustic cavity can be tuned for connecting to the primary enclosure and to damp a single acoustic mode. As many resonators are required as there are modes to control, and they require big volumes and experimental tuning.

Compliant panels can alter the mechanical boundary conditions of a wall of an enclosure by mounting a flexible membrane or panel. They are simple to construct, but they are unsuitable for DC pressure loads, and are difficult to optimize both theoretically and experimentally.

1.2.2 Feedforward

Techniques categorized as feedforward use a reference signal, filtered and applied « downstream » to arrest disturbance propagation. This performance is extremely sensitive to plant or filter variation. No model of the acoustic system is required, which is significant as reverberant acoustic system are difficult to identify and they can vary dramatically throughout service due to the movement of internal objects or changes in the boundary conditions.

Adaptive filters are simple in concept, but they require significant processing resources to provide adequate real time performance, and practical implementation requires also the use of additional filters.

1.2.3 Feedback

This techniques are motivated for eliminate the need for a feedforward reference sensor, but they have been criticized for their impracticality. They consider the regulation of pressure at a single microphone location, and majority of studies neglect the spatially distributed nature of acoustic enclosures, so consequently, offer extremely poor or detrimental performance at small distance from the error microphone.

1.2.4 Impedance based

Impedance based control consist in the active manipulation of acoustic impedance within an acoustic enclosure. The motivations are reduced complexity, robustness, intuitive acoustic design, and global performance. The main disadvantage of active impedance control is the difficulty in achieving tight velocity or pressure feedback control of the loudspeaker.

1.2.5 Electrical shunt control

A simple electrical impedance can be designed and tuned experimentally or adaptively, that when connected to a speaker coil improves the dissipation of acoustic energy. This can be done by identifying the interaction between sound field, mechanical speaker and electromagnetic transducer. The goal of this work is then to optimize the mechanical parameters of the loudspeaker and the value of the impedance for the case in which this impedance can be simplified to a passive resistor.

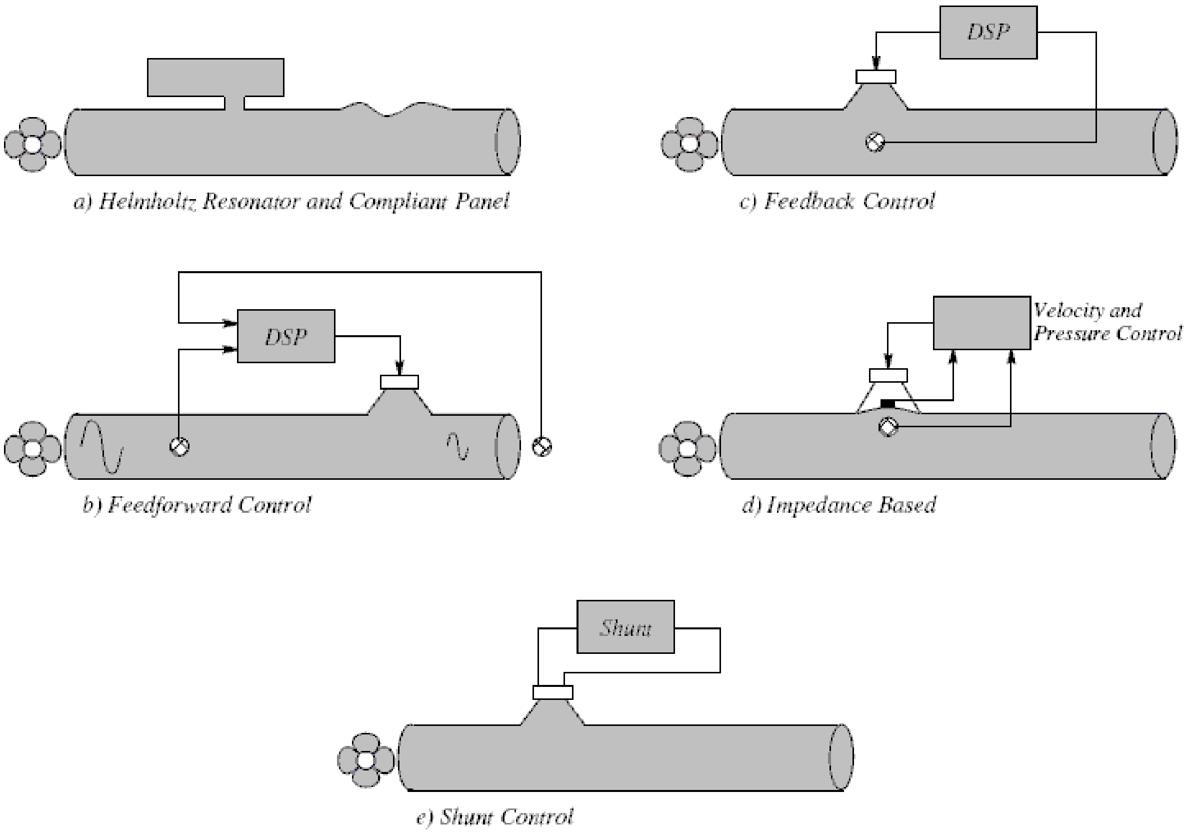


Figure 1: noise control treatment methods

2. Theoretical background: The loudspeaker

2.1 Introduction

The loudspeaker is an electro-acoustic transducer, where the transduction follows a double procedure: electrical-mechanic-acoustic.

At first, it turns the electrical waves into mechanical energy, and in the second place it turns the mechanical energy into acoustic energy. It is therefore the door where the sound goes out on the outside from the devices that made his amplification.

There are several ways of electroacoustic transduction but much of these systems are constituted by dynamic loudspeakers.

The performance in a dynamic loudspeaker is based on the existence of a magnetic field created by a magnet, in which is included a coil. When the electrical signal proceeding from the amplifier or from any other equipment is applied to the coil, a magnetic field that changes in agreement with the above mentioned signal is created. In the air gap between the magnets, there is placed a cylindrical coil that is joined to the diaphragm. The coil generates an electrical current that causes that the magnet produces a magnetic flux that makes vibrate the membrane.

As the membrane vibrates, it moves the air that is placed opposite to it, generating variations of pressure in the air, or that is the same thing, sonorous waves.

Depending on the variations of voltage of entry, the cone vibrates and generates equivalent disturbances in the air.

2. 2 Elements of a dynamic loudspeaker

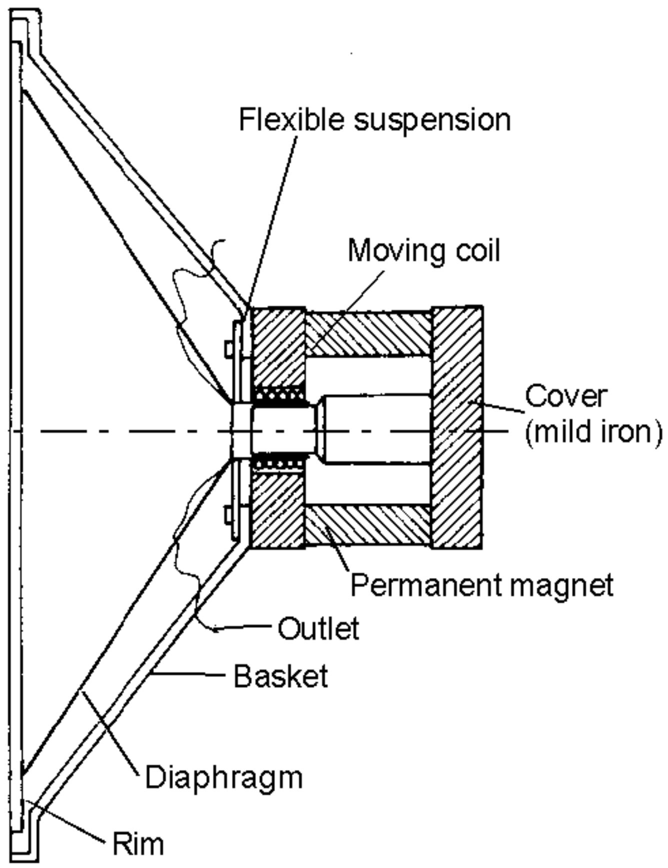


Figure 2.1: Parts of the loudspeaker

2. 2. 1 Diaphragm

It constitutes the mechanic acoustic transducer. It is the part that faces the fluid environment, the air, and which, with its vibrations, produces acoustic waves of pressure and wealth. The diaphragm is one of the most important parts of a loudspeaker, because much properties of the loudspeaker depend on it.

Characteristics

Size

The size of the diaphragm influences the efficiency, directivity and the frequency response.

The frequency response of a dynamic loudspeaker, in its useful part, is dominated by the mass. What is more, its pressure response is inversely proportional to the mass. Also, the efficiency and the performance are inversely proportional to this mass and therefore, to the diaphragm size. That's because loudspeakers with a bigger diaphragm have to have a bigger force factor which normally means a bigger magnet and magnetic circuit.

The bandwidth is also determined by the size of the diaphragm and it's mass. It's known that the lower limit frequency in the frequency response of a dynamic loudspeaker is determined by the mechanical resonance frequency. That's because the heaviest diaphragms have a lower resonance frequency and they offer more response in low frequencies. Nevertheless, it's possible to get a good radiation in low frequencies with a small diaphragm compensating the decrease of mass of the diaphragm with a bigger mechanical compliance, that is to say, with a « softer » or elastic diaphragm.

The directivity of a loudspeaker is much related to the size of its diaphragm. Concretely, the directivity of a radiator is proportional to the factor ka, being $k = w/c$ the wave number, and a the ratio of the diaphragm. Hence, a loudspeaker will be more directive as the frequency increases. On the other hand, at the same frequency, the biggest loudspeaker will be the most directive.

Materials

A light diaphragm turns out to be favorable to increase the efficiency and sensibility of the loudspeaker in the useful range of reproduced frequencies, since these characteristics are inversely proportional to the mass. Nevertheless, a light diaphragm has, normally, a minor inflexibility, since its thickness is smaller, and therefore a bigger trend to present own manners of transverse vibration. Because of it, at the moment of design diaphragms for loudspeakers, it is necessary rely on materials that they should be simultaneously light and rigid.

The inflexibility to the flexion of a material is determined by its module of Young (E). Rigid materials have a high E. On the other hand, is interesting that the specific module of Young (E/ρ) would be high, being ρ the diaphragm material density. With this we get the double condition of inflexibility and lightness, moreover if there is born in mind that the velocity of transverse transmission c_{mat} of the sound in a diaphragm is not more that the square root of the specific module of young, it’s clear that materials with high c_{mat} are requiered.

It suits to consider also the mechanical factor of quality, or factor of overcharge Q, in proper transverse manners. When the Q is very high, the material will have few mechanical losses relative to the flexion, and therefore when it exists a transverse resonance (proper mode) will produce to itself a vibration of high extent, which is not desirable.

Transverse proper modes.

The resonance transverse proper modes appear because there is a stationary wave of the transverse vibration in the loudspeaker. This produces that the diaphragm flexes according to few nodose lines, in such way that certain zones vibrate in a sense while others do it in the opposite direction. The own proper modes of the diaphragm produce coloration in the response in frequency, loss of response of the loudspeaker, and distortion due to the alineality inherent in the process of flexion of the diaphragm.

There are two kinds of proper modes, radial proper modes and circumferential or nodose proper modes. In the radial proper modes, the nodose lines are ratios of a circular diaphragm. In the circumferential proper modes, the nodose lines are circumferences of the diaphragm.

Radial proper modes are produced when a stationary wave crosses the diaphragm perimeter, and the circumferential proper modes are produced when the stationary wave crosses the profile of the loudspeaker. It is to say, in the radial modes the circumference of the diaphragm is deformed, and in the circumferential ones the profile does it. Since the circumference of the loudspeaker is major that the radius, the proper radial modes happen normally to minor frequency, and for it they are the most dangerous.

Shape of the diaphragm

The shape of the diaphragm is important if we want to radiate efficiently by sufficient inflexibility, avoiding the appearance of proper transverse modes, and with a low directivity, especially for the units of high frequency.

Dome

We can use diaphragms with dome shape especially for high frequencies due to the high inflexibility provided by its curved shape.

The usual version of a dome is the convex shape, but they exist also concave shapes. Convex ones are less directive and less efficient than concave ones.

Cone

The loudspeakers of cone are more frequently used at the moment of constructing a dynamic loudspeaker. This configuration allows a great surface of radiation with a driving surface relatively small (the diameter of the cylinder of the coil). In addition, we obtain a bigger subordination of the mobile part of the loudspeaker, avoiding the off-centre one of the coil, for what it is the configuration that is more used in low frequency, where the excursions of the diaphragm are very big, and where it is necessary a big mass to reduce the frequency of mechanical resonance, which depends inversely on this parameter.

On the other hand, the principal disadvantages of the diaphragms of type cone are the proper transverse modes (radial and circumferential), seen previously, and derivatives of the lack of inflexibility of this structure. To avoid this disadvantage, there are adopted curved or staggered profiles that send these proper modes towards the high frequency. The curved profile provides, over the first resonance of the loudspeaker, fewer acoustic signal that the straight profile, nevertheless it supports a response in pressure softer than the diaphragms of straight profile.

Flat diaphragms

Though they are less used, also they exist designs of loudspeakers with flat diaphragm. A dynamic loudspeaker of moving coil adapts very badly to move a flat diaphragm. The flat form is much less rigid than the shape of cone or dome. An only one coil would push a great surface supporting a limited area, and there would be produced easily proper transverse modes in low frequency if the diaphragm is big. They exist designs with several coils, but in the practice they are not very used because of their complexity.

2.2.2 Moving coil.

In a dynamic loudspeaker, the moving coil is the driver entrusted to transform the variations of electrical tension into his ends, into magnetic force and therefore into mechanical vibration and acoustic radiation.

The materials with which the conductive thread is made are fundamentally the copper and the aluminum, mechanized in enameled threads to avoid the electrical contact between the different helices of the coil.

The shape of the thread can be cylindrical or square. This thread is wound in 2, 4 or 6 caps, depending on the necessary electric impedance and the required sensibility and electrical power. With a bigger number of caps, though the total length of the thread contained in the coil is bigger (and therefore the force factor *Bl* is bigger), also the electrical impedance of entry will be bigger, diminishing the magnetic induction B. There exists an ideal size of the length of the thread of the coil in loudspeakers of current dimensions, which it takes to electrical impedances of the order of dozens of ohms (2, 4 and 8 Ω are the most frequent values).

The support that holds the coil is a hollow cylinder made of carton, aluminum or plastic, called former. The electrical power of the loudspeaker and the dissipation of the heat are related to the dimensions and nature of the mobile coil and of the former.

2.2.3 Magnet and magnet circuit.

The permanent magnet is the element entrusted to produce the magnetic field represented by the induction B.

The magnets are usually metallic or ceramic, being the last ones the most frequent. The shape can be cylindrical or, more habitually, with ring shape, that produces fewer losses of magnetic field in the bobbin and is easy to mechanize with the ceramic magnets. It's interesting to have a magnet as big as possible, with big circumference and surface, for producing a big force factor Bl.

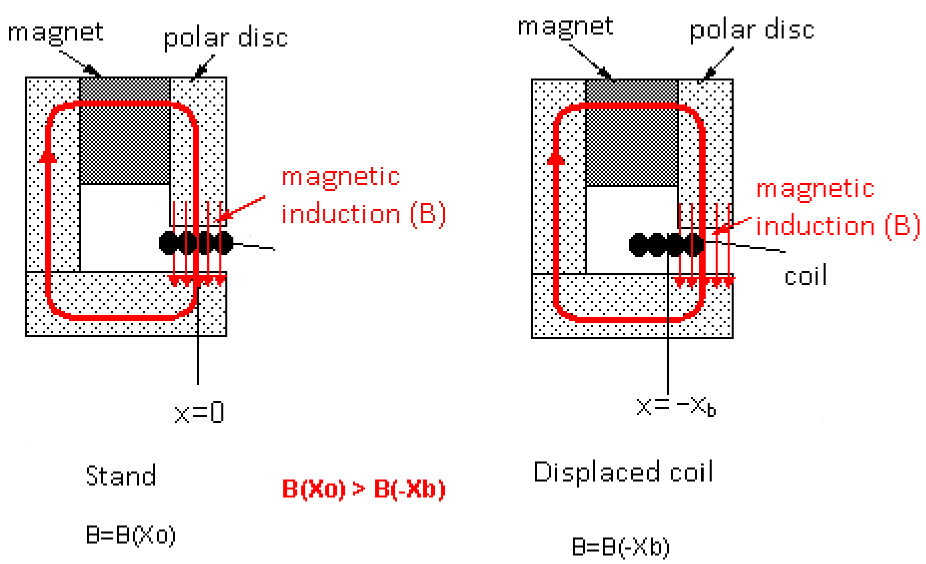


Figure 2.2: Performance of the magnetic circuit of a dynamic loudspeaker

2.2.4 Flexible suspension

The primary suspension of a loudspeaker is the principal responsible element of giving elasticity to the movement of the diaphragm and therefore this is directly related with the mechanical compliance of the diaphragm C_{ms}.

The quality and the suitable configuration of the primary suspension are basic if we want low distortion at low frequencies. Under the resonance frequency of a loudspeaker, the diaphragm movement verifies the Hooke law:

$$x_d(t) = C_{md} f_d(t)$$

(Eq.1)

Being $x_d(t)$ the movement of the diaphragm and $f_d(t)$ the force applied by the electro-mechanic transducer. This is a linear law, and if it is not fulfilled it will originate distortion in low frequency. But in loudspeaker this is only fulfilled for small movements, because is impossible to stretch indefinitely the suspension keeping its elastic properties.

If the loudspeaker is inside an hermetic box, the mechanical compliance of the loudspeaker C_{md} (or its acoustic compliance C_{ad}) is associated to the acoustic compliance C_{ab} of the air inside the box, and the diaphragm is hold by a whole compliance of:

$$C_{at} = \frac{C_{ad} C_{ab}}{C_{ad} + C_{ab}}$$

(Eq.2)

2.2.5 Other elements:

Rim

It is entrusted to hold the top part of the diaphragm to the structure of the loudspeaker. It does not take part excessively in the value of the mechanical compliance of the diaphragm.

Basket

Is the structure that holds the loudspeaker. It has to be inflexible and heavy, and made of magnetic materials that don't deform the internal magnetic field of the loudspeaker.

Cover

The back cover of the loudspeaker, made of ferromagnetic materials and entrusted to close the magnetic field.

For quantifying the mechanical characteristics of these elements of the loudspeaker, three parameters are defined:

- **Mechanical mass (M_{ms}):** represents the total weight of the coil, the diaphragm and the air mass. It's measured in kg. Is related to the performance and the frequency response of the loudspeaker, mostly in high frequencies.
- **Mechanical compliance (C_{ms}):** represents the elasticity of the suspension, it's meassured in µm/N. A high compliance means a very flexible suspension.
- **Mechanical resistance (R_{ms}):** represents the mechanical loss caused by the suspension by limiting the movement of the membrane. Its units are Kg/s.

One of the characteristic parameters of the loudspeaker is the resonance frequency, which depends directly on the values of elasticity and mechanical mass. The physical phenomenon of the resonance frequency is observed as a speed maximum of the mobile set in the above mentioned frequency, so that delivering the same power to the loudspeaker than for the rest of frequencies the mobile set moves more rapidly.

The fact that we have a speed maximum does not imply that the displacement has to be a maximum. These parameters depend on the frequency due to the limitations imposed by the elasticity and the mass of the mobile set. We can recognize two areas in the frequency :

- **Under the resonance frequency:** at low frequencies, the inflexibility of the suspension acts as a brake for rapid movements of the diaphragm due to the big movements required at this frequencies,
- **Over the resonance frequency:** As the frequency increases and the amplitude of the displacement decresses, the effect of brake of the suspensions dismishes. Nevertheless, frequencies are demanded increasingly rapid, and the mobile set cannot move its mass to such a rapid pace. Thus, the velocity starts diminishing with the frequency and helps to reduce the displacement increasingly.

If we compare the velocity curve of two mobile sets of different mechanical characteristics we see that the explained till now is fulfilled: in the first zone the most elastic set moves more rapidly, whereas in the second zone, the most light set is the one that has a bigger speed. Therefore, the resonance frequency represents the transition between these two zones: the first zone controlled by the elasticity and another second zone regulated by the mobile mass.

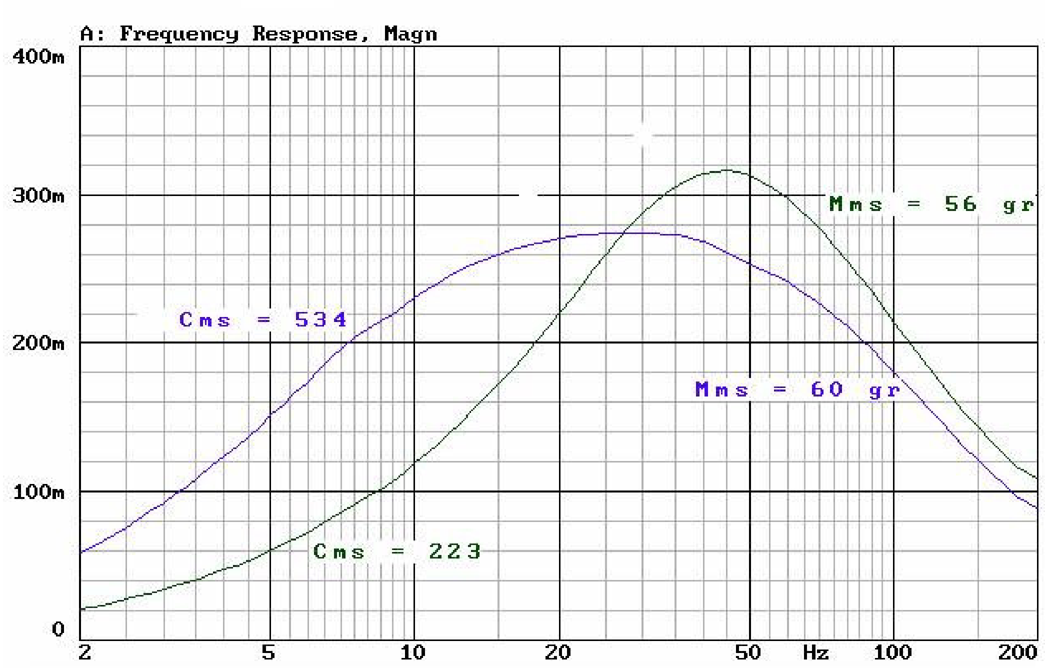


Figure 2.3: Comparison between the velocity curves of two mobile sets: green curve-12''BR70, blue curve-12''POWER

The dependence of the value of the resonance frequency with the mechanical parameters is produced in the following way:

- With more weight of the mobile set we obtain a frequency of lower resonance, and vice versa.
- With more elasticity of the mobile set we obtain a frequency of lower resonance, and vice versa.

2.3 General characteristics

2.3.1 Impedance

The mechanical elements C_{ms}, M_{ms} and R_{ms} already seen produce a mechanical impedance in the Laplace domain of:

$$Z_{ms} = R_{ms} + sM_{ms} - \frac{1}{sC_{ms}}$$

(Eq.3)

Which represent the opposition to the movement of the mobile set.

The electric part of the loudspeaker is composed by a resistance R_e and an inductance L_e due tu the coil, which presents complex impedance determined by:

$$Z_L = 2\pi f L$$

(Eq.4)

Where f is the frequency and L is the inductance

For expressing the total impedance sum of the mechanical and electric parts of the loudspeaker we have to write both impedances in the same analogy. For moving the mechanical impedance to the electric area we have to take in account the induced current which appears due to the movement of the membrane, which opposites the current applied to the loudspeaker. This represents the effect of the mechanical impedance over the electric part. Can be called movement impedance and its value is:

$$Z_{mov} = \frac{(Bl)^2}{Z_{ms}}$$

(Eq.5)

There is also another factor in the impedance of a loudspeaker added to the mentioned before known as R_{ed} or impedance due to the Focault currents. This consists in inducted currents of opposite sign that apperas because of the movement of the coil in the magnetic field.

Therefore, once defined all the factors that intervene in the impedance of the loudspeaker, we can conclude that the impedance measured in points of the loudspeaker comes given for:

$$Z_{total} = R_e + R_{ed} + Z_L + Z_{mov}$$

(Eq.6)

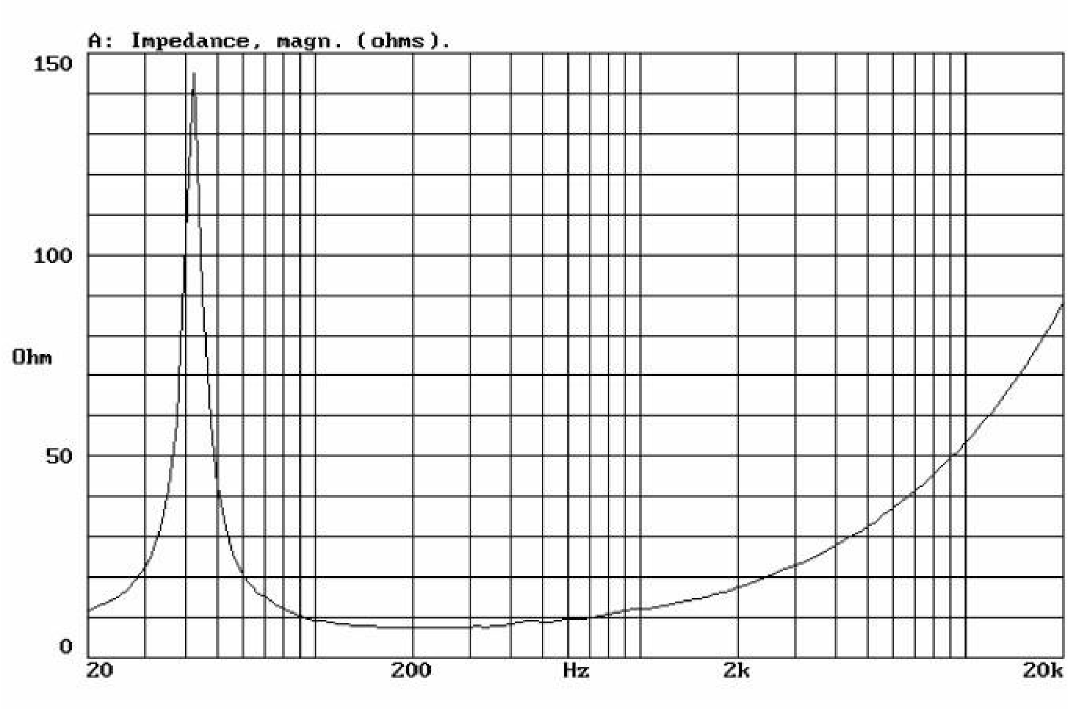


Figure 2.4: Impedance curve of the loudspeaker 12 ''LX60

We can see in the figure a peak corresponding to the resonance frequency. This peak is produced because at this frequency we have a minimum of mechanical impedance, because the part determined by the mass and the elasticity is annulled. This minimum makes the mobile set move with the maximum velocity and we obtain a maximum in the impedance movement and hence a maximum in the impedance of the loudspeaker.

2.3.2 Frequency response

The frequency response defines the useful working band of any transducer. In the case of the loudspeaker, is a measurement of how it reproduces a certain range of frequencies or bandwidth. Usually is given as a tone seep, in which it is specified the level given for the loudspeaker, for each frequency, at 1 meter of distance and with 1W of electric excitation.

For obtaining the specific relation between the radiate pressure and the applied voltage we can use the transfer function, which is defined by:

$$H(r)[dB] = 20log\frac{P(r)}{E}$$

(Eq.7)

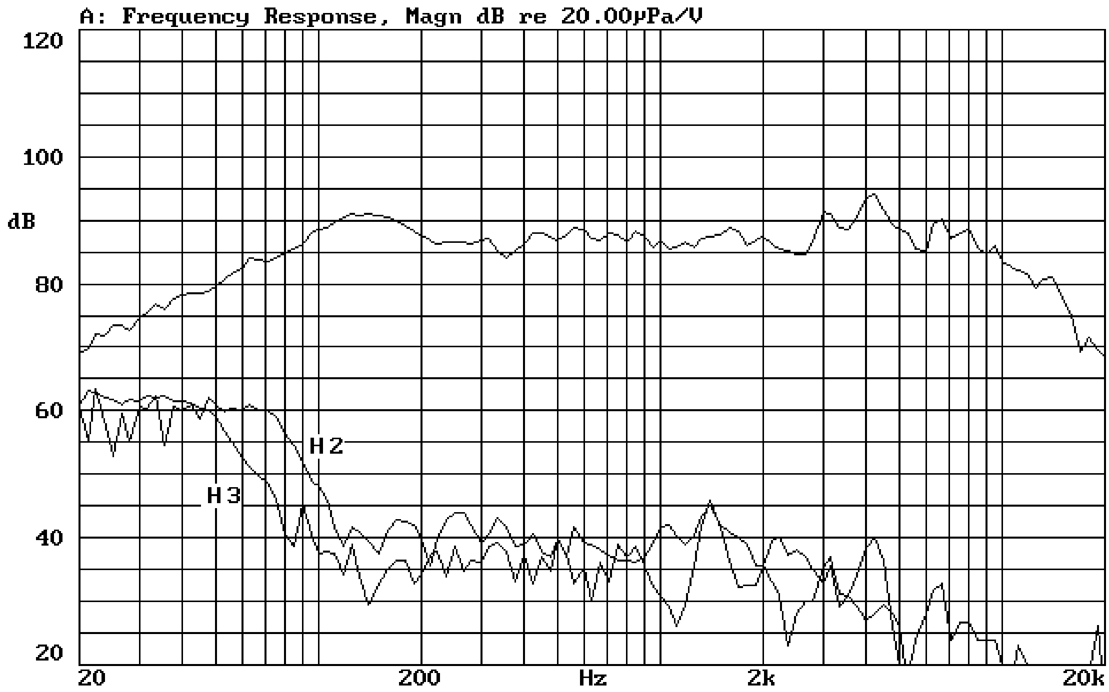


Figure 2.5: Frequency response

2.3.3 Other parameters

Rated power

Nominal (or even continuous) power, and peak (or maximum short-term) power a loudspeaker can handle, i.e., maximum input power before thermally destroying the loudspeaker. It is never the sound output the loudspeaker produces, in fact if a loudspeaker work with a power near the rated power, generally it will distort very much. Rated power is then, a security indication.

Sensibility

Is the sound pressure level produced by a loudspeaker in a non-reverberant environment, usually specified in dB, and measured at 1 meter with an input of 1 watt or 2.83 volts, typically at one or more specified frequencies. This rating is often inflated by manufacturers.

Directivity

Directivity can be given as a polar plot which reflects the relative level to the maximum radiation in function of the deviation angle of the axis of the loudspeaker. It is given for each frequency, and normally is expressed in dB.

3. Design of an electric shunting loudspeaker

3.1 Introduction

In this section, the design of a shunting loudspeaker according to some specific characteristics is described. We will focus in the case of the low-frequencies, and therefore in the case that the shunting is performed by a resistor. The program used for the design and the essays has been the *MATLAB.*

Our final aim is to get a reflection coefficient equal to 0 for one specific range of frequencies. We can define the reflection coefficient as:

$$r = \frac{p_r}{p_i} \tag{Eq.8}$$

but it can be determined also as:

$$r = \frac{Z_a - Z_c}{Z_a + Z_c} \tag{Eq.9}$$

Where Z_c is the characteristic impedance of the medium (the air in this case) determinated by

$$Z_c = \rho c$$

And Z_a is the acoustic impedance of the loudspeaker, which is the relationship between the pressure in the contact area of the surface and the total volume velocity that this surface moves. It can be expressed as:

$$Z_a = \frac{p}{v} \tag{Eq.10}$$

being p the acoustic pressure and v the volume velocity.

So we can equal the value of the intern impedance of the loudspeaker to the characteristic impedance of the air for reduce as much as possible the value of r.

For introduce the mechanical dynamics of the loudspeaker, the law that we have to satisfy is that the acoustic pressure originated by a radiant surface is in related to the mechanical force that impulses it by the expression:

$$p(t) = \frac{f(t)}{S}$$

(Eq.11)

Where S is the surface of the radiant element, and also by:

$$u(t) = \frac{v(t)}{S}$$

(Eq.12)

If we use the same analogy for both acoustic and mechanical dynamics, we can represent the mechanic-acoustic transduction by a T transformer.

As we have seen before, the diaphragm of the loudspeaker is represented by a mechanical mass M_{MS}, a mechanical resistance R_{MS} ,and a mechanical compliance C_{MS}, and all this elements are moving with the same velocity. That's because the variable velocity has to be common to all of they.

The movement of the diaphragm is related with the intensity of current in the electric part of the loudspeaker by:

$$V(t) = Blv$$

(Eq.13)

Where $V(t)$ is the inducted voltage produced by the movement of the diaphragm, B is the magnetic flux density, and l is the conductor lenght. So we get the electric circuit composed by a resistance R_E , and an inductance L_E .

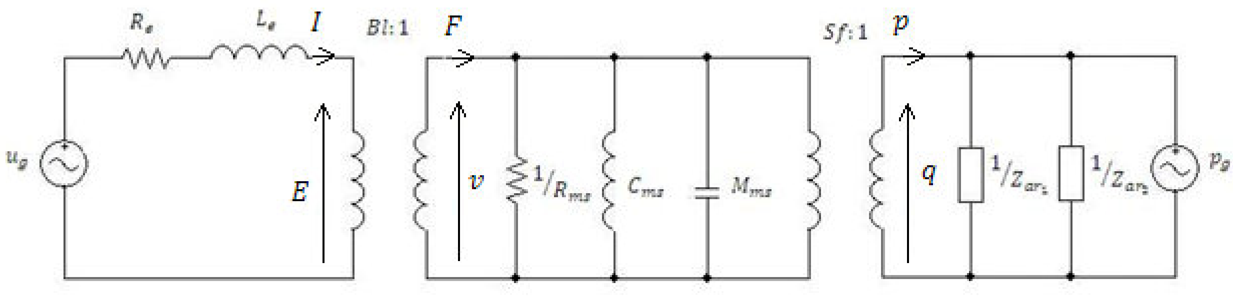


Figure 3.1: loudspeaker circuit, mobility analogy

In the figure, Z_{ar_1} and $Z_{ar_2}$are the radiation impedance, and represents the opposition that presents the air ahead and behind to the diaphragm, to the movement of itself.

If the loudspeaker is short-circuited after the electric elements, at frequencies below the cutoff of the filter 1/(sLe+Re), the coil acts as an electrodynamic brake, and physical damping is added to the mechanical speaker dynamics. An appropriate electrical resistive impedance $Z(s)$ can be connected to the terminals of the speaker coil to optimize the damping and passive acoustic mitigation.

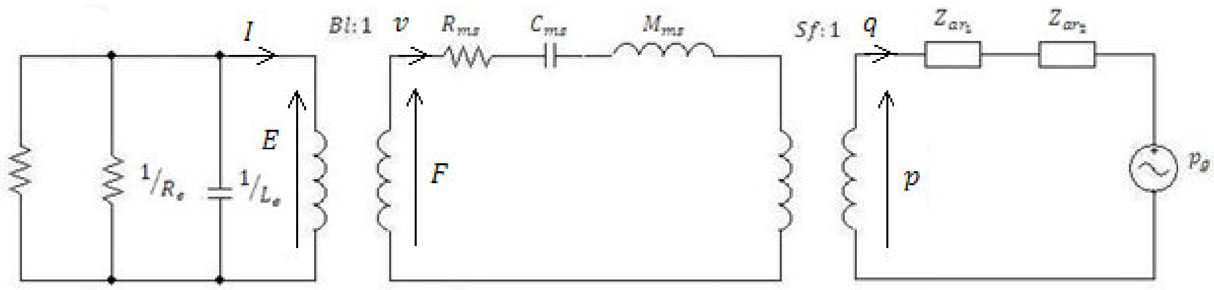


Figure 3.2 : loudspeaker circuit, impedance analogy

Now we can simplify the whole circuit transforming mechanical and electrical elements and variables into acoustic elements, considering the relations of the transformers, and we can calculate the value of the acoustic impedance of the system.

For getting total dumping of acoustic energy, Z_a has to be equal to Z_c. And we have to search the optimal resistance added to the loudspeaker for this.

3.2 Passive shunt circuit design

A network connected to the terminals of a loudspeaker can be designed to moderate the response to a coupled acoustic enclosure. In this section, the design of passive shunt is discussed.

An enclosed speaker emulates the acoustic response of a Helmholtz resonator. If the properties of this virtual Helmholtz resonator can be adjusted, the speaker can be employed to attenuate highly resonant acoustic mode in the same fashion as a physical Helmholtz resonator.

At low frequencies where w << Re/Le, the influence of the inductor Le can be neglected. In this case, damping can be increased by connecting a resistor R to the terminals of the speaker. The resistor R in addition to the coil resistor Re, can emulate the acoustic response of a Helmholtz resonator. The mechanical damping d of the resonator is represented in the circuit of a loudspeaker as a mechanical impedance. The shunting electric resistor is placed in parallel to this damping, so the total damping of the virtual Helmholtz resonator is

$$d_{tot} = d + \frac{1}{R + R_e}$$

(Eq.14)

Note that the total damping is restricted in range between d and d+1/Rs. A speaker with low d and Re will provide the greatest tuning range.

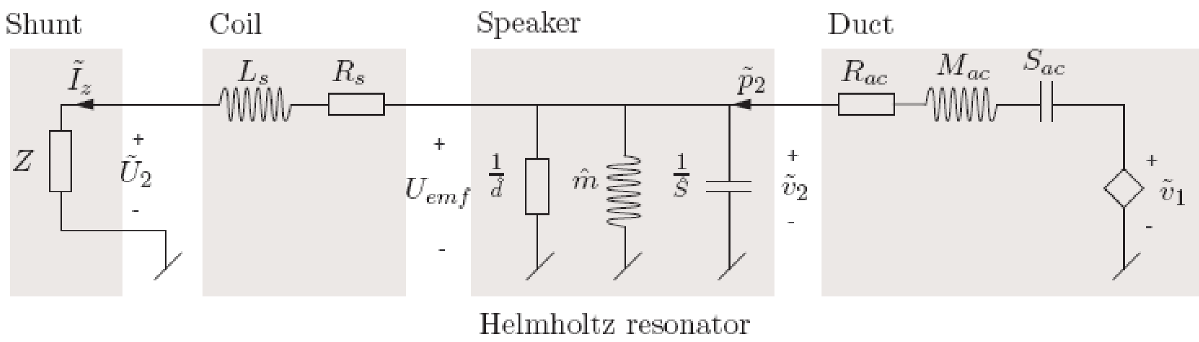


Figure 3.3: electric equivalent for the Helmholtz resonator

3.2.1 Design of an electric shunting resistor at 125 Hz

Now, we are going to approach the problematic of the absorption at low frequencies. Basing on the previous criterion, we can design an electric equivalent of a Helmholtz resonator for the concrete case of 125 Hz.

Simplifying the diagram of the shunting loudspeaker, in the electric part we have a resistance result of adding Re and Rc:

$$R'_e = \frac{(Bl)^2}{S(R_e + R_c)}$$

(Eq.15)

and an inductance represented by a capacitor of value :

$$L'_e = \frac{(Bl)^2}{Sf^2 L_e}$$

(Eq.16)

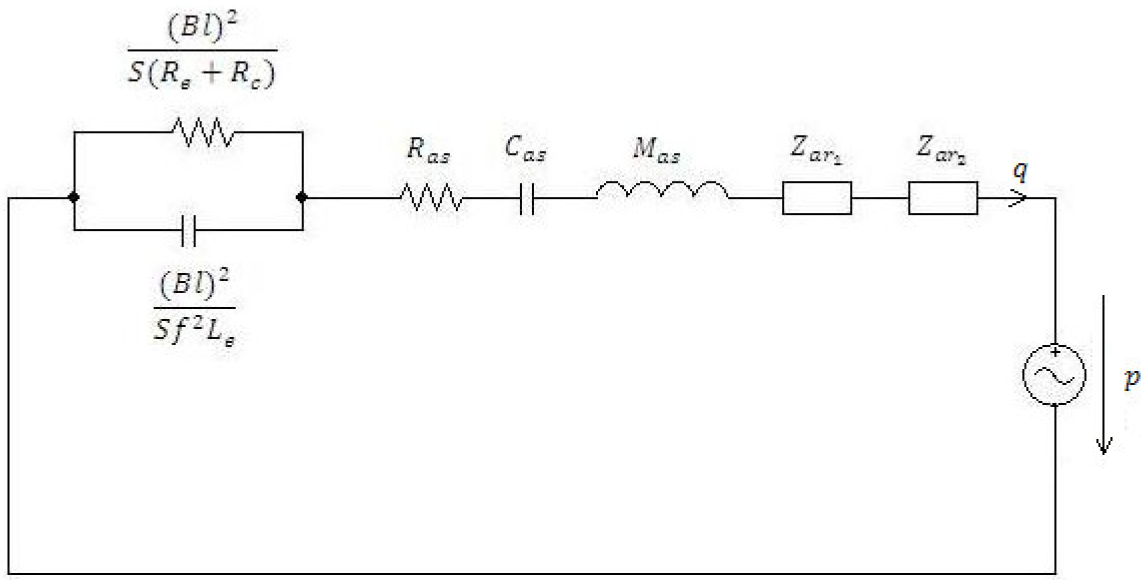


Figure 3.4: Shunting circuit

As we are performing in low frequencies, we have $\omega \ll \omega_c$ and we can despreciate the value of the capacitor and eliminate it. We can also sum the acoustic impedances obtaining an only impedance Z_{ar}.

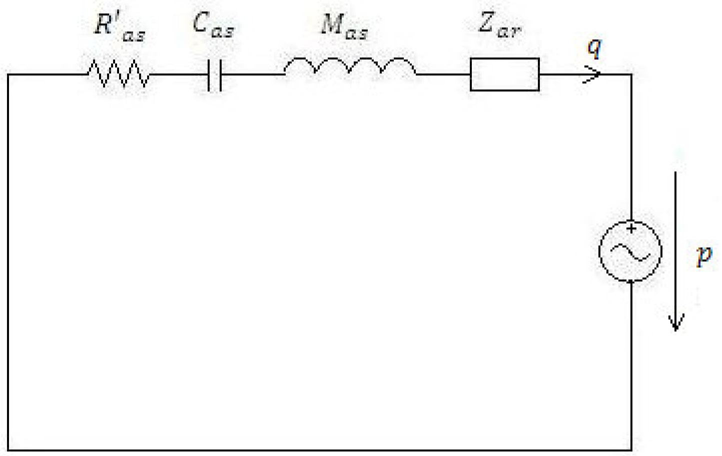


Figure 3.5 : simplified shunting circuit

Where

$$R'_{as} = R_{as} + R_e'$$

(Eq.17)

Is the sum of the acoustic resistance R_{as} and the electric resulting resistance including the shunt resistance R_c.

Then the criterion is to equal the equivalent resistance to the characteristic impedance of the air, Z_c :

$$R_{as} + \frac{(Bl)^2}{S(R_e + R_c)} = \rho c$$

(Eq.18)

And hence, to obtain the optimal value of the shunting resistance R_c is simple:

$$R_c = \frac{(Bl)^2}{\rho c S - R_{ms}} - R_e$$

(Eq.19)

Now the equation in the time domain for the simplified circuit is :

$$p(t) = R'_{as}(v(t)) + M_{as}\frac{\partial v(t)}{\partial t} + C_{as}\int v(t)\,dt + M_{ar}\frac{\partial v(t)}{\partial t} + R_{ar}$$

(Eq.20)

Where the acoustic radiation impedance has been divided in a mass and a resistor.

And in the Laplace domain :

$$P(s) = R'_{as}\,V(s) + sM_{as}V(s)\frac{1}{sC_{as}}V(s) + sM_{ar}V(s) + R_{ar}V(s)$$

(Eq.21)

So now we can get the acoustic impedance of the system, Z_a , rewriting the ecuation as :

$$Z(S) = \frac{P(s)}{V(s)} = R'_{as} + sM_{as}\frac{1}{sC_{as}} + sM_{ar} + R_{ar}$$

(Eq.22)

3.2.3 Parameters of the loudspeaker

If we choose a diaphragm of a diameter of 0.1 m, then the area is:

$$Sf = \pi 0.05^2$$

(Eq.23)

And the mass of the diaphragm is :

$$M_{md} = \rho_{md}Sf$$

(Eq.24)

Where ρ_{md} is the surfacic mass of the diaphragm, which is equal to 1

Then we choose the characteristics of the coil. As we have seen, we need a very light coil:

The rounds of the cable are $N = 1$; diameter of wire : $d_w = 1\exp(-3)$; Magnetic field of $B = 1$; The coil lenght is $L = (2\pi 0.01)N = 0.0628\, m$; The surface of the section of wire is : $s = (\pi d/2)^2 = 7.854\exp(-7)$; So finally we obtain a mass of the coil equal to $M_{mc} = \rho_{mc} L s = 4.4216$; Where ρ_{mc}= 8960 is the density of copper , and a force factor $Bl = 0.0628$

With this information and fixing the value on 125 Hz for the resonance frequency, we can get the value of the mechanical parameters of the loudspeaker :

$$M_{ms=}M_{mc} + M_{md} = 0.0083\, Kg$$

$$C_{ms} = \frac{1}{M_{ms}(2\pi f s)^2} = 1.9541\exp(-4)$$

$$R_{ms} = 1$$

And the electric parameters:

$$\rho_e = 1.7\exp(-8)$$

$$R_e = \frac{\rho_e L}{s} = 0.0014$$

$$L_e = 2\exp(-3)$$

Acoustic equivalent parameters:

$$R_{as} = 127.3240$$

$$M_{as} = 1.0563$$

$$C_{as} = 1.5347e(-6)$$

And in the end we get the values of the sum of resistances R'_{as} , and the optimal resistance R_c for the electric shunting at 125 Hz

Now with these values for the equation we obtain the curve of the acoustic impedance of the loudspeaker Z_a :

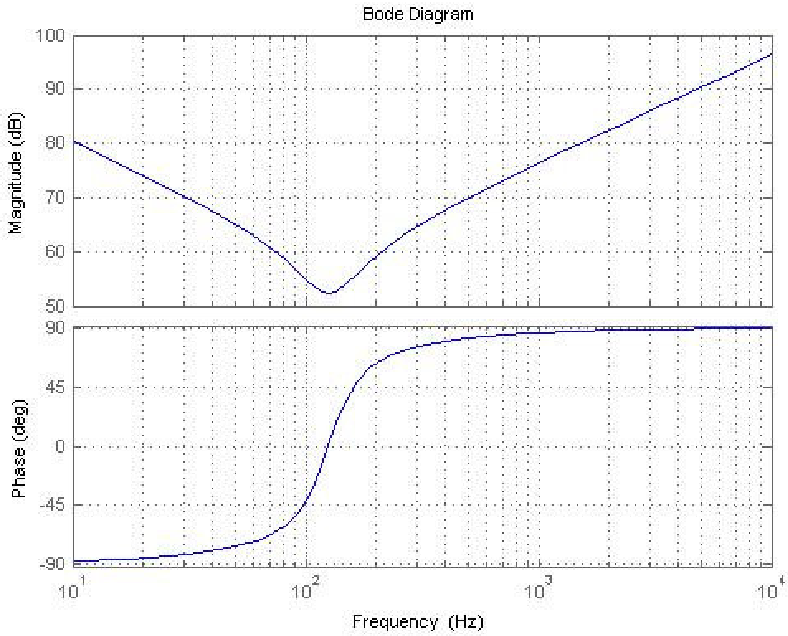


Figure 3.6: Curve impedance of the 125 Hz shunting loudspeaker

And the acoustic admittance Z_c / Z_a :

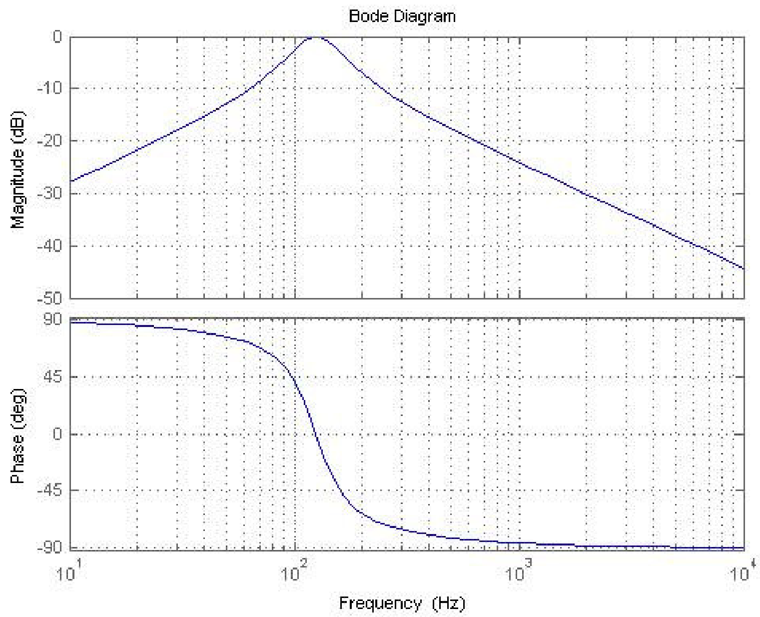


Figure 3.7: Curve of admittance of the 125 Hz shunting loudspeaker.

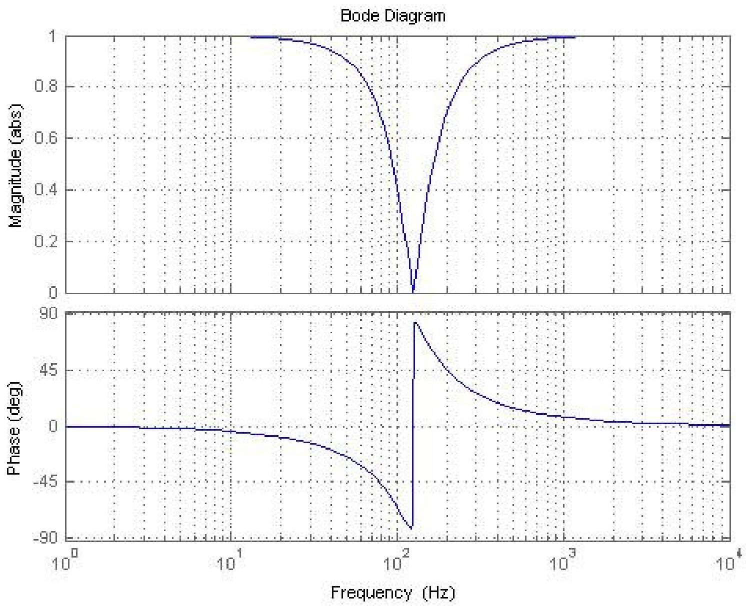


Figure 3.8: Reflection coefficient of the 125 Hz shunting loudspeaker

The results of the short circuited electric circuit present a not complete absorption at the resonance frequency:

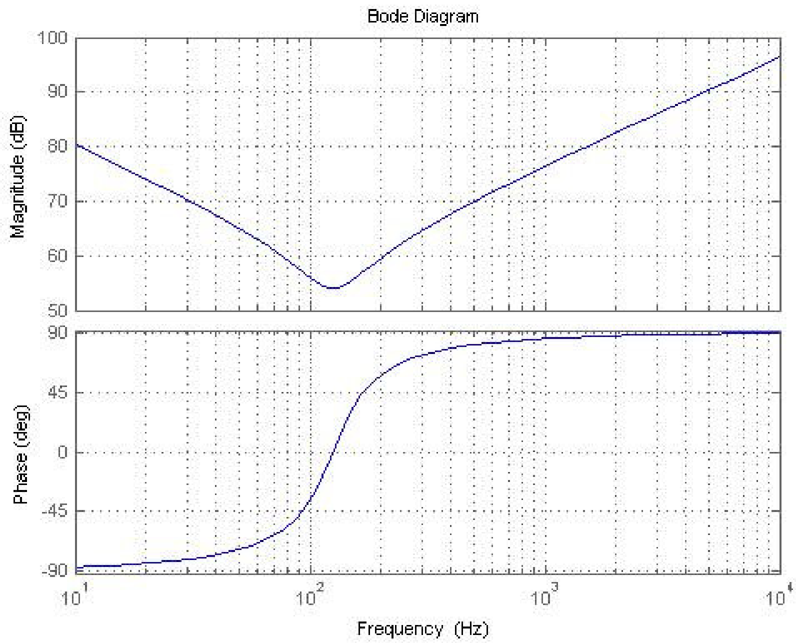


Figure 3.9: Impedance of the short circuited 125 Hz shunting loudspeaker.

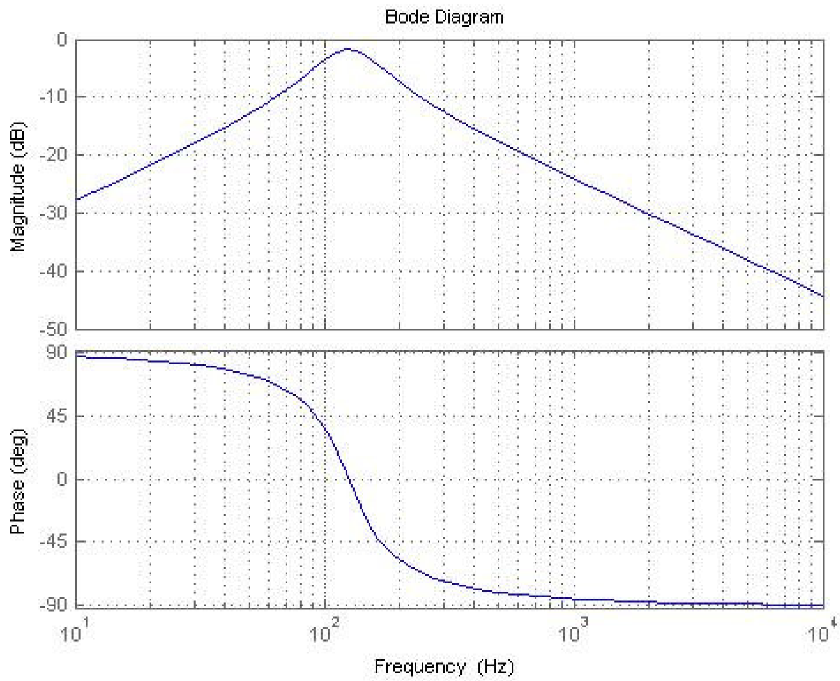


Figure 3.10: Admittance of the short circuited 125 Hz shunting loudspeaker.

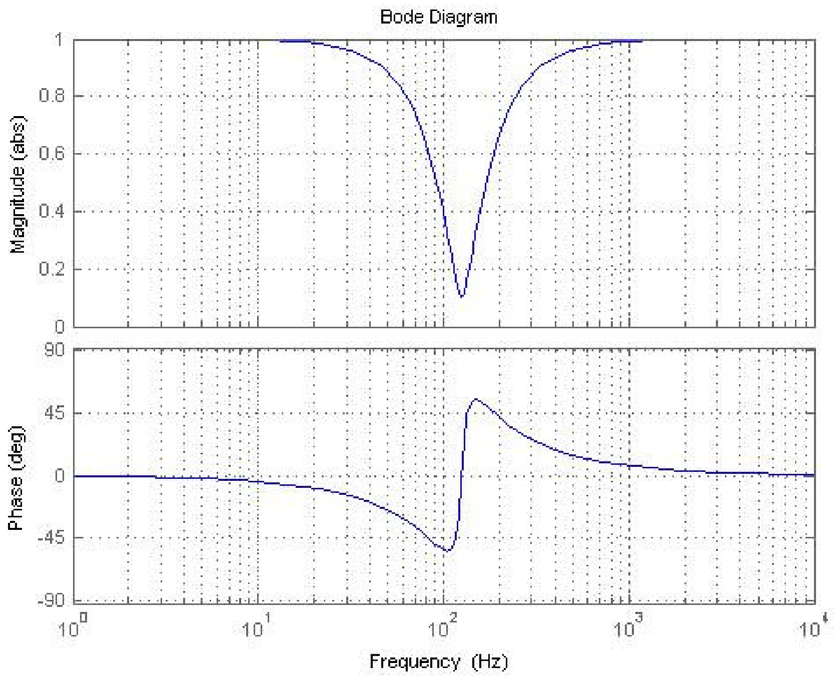


Figure 3.11: Reflection coefficient of the short circuited 125 Hz shunting loudspeaker.

As we can see in the figure, we can get a damping loudspeaker with total shunting at 125 Hz with the concrete values of the resistance, mass and compliance. But, due to the small of these parameters, we get a very narrow shunting around the resonance frequency. We can try experimentally with different combinations of parameters, knowing that a bigger compliance will enlarge the shunting range, but will make decrease the resonance frequency.

If we choose a bigger diameter of the diaphragm ($d = 0.2m$) we get a heavier loudspeaker, so we don't get total absorption at 125 Hz but we expand the absorption area.

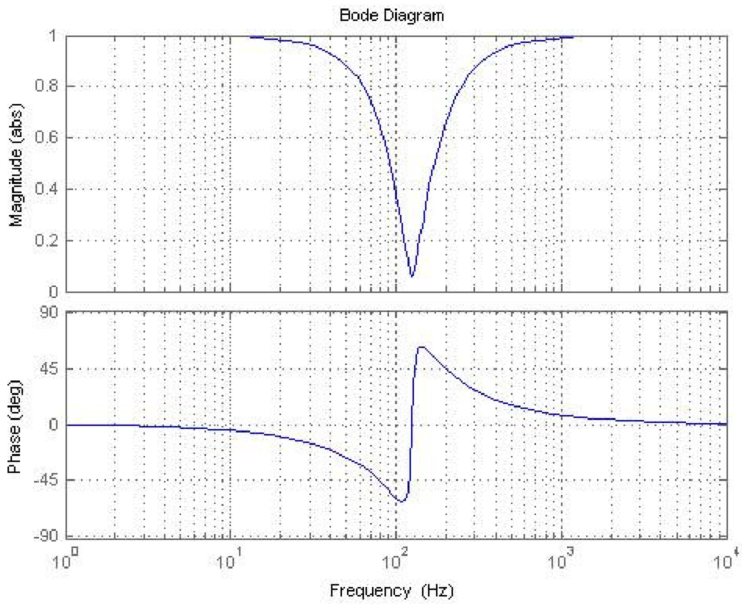


Figure 3.12: Reflection coefficient with a diameter of the diaphragm of d = 0.2m

Acting directly on the value of the compliance of the loudspeaker we can vary significantly the shape of the reflection curve, and the value of the resonance frequency

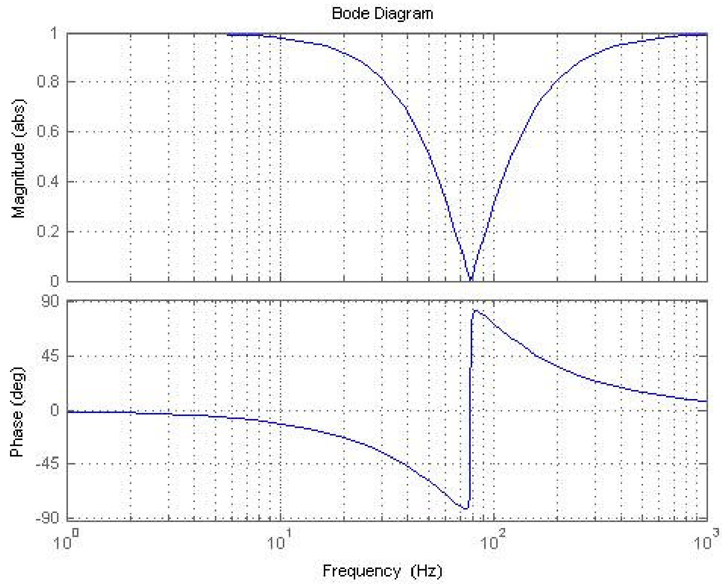


Figure 3.13: Reflection coefficient with a compliance of Cms=0.5exp(-3)

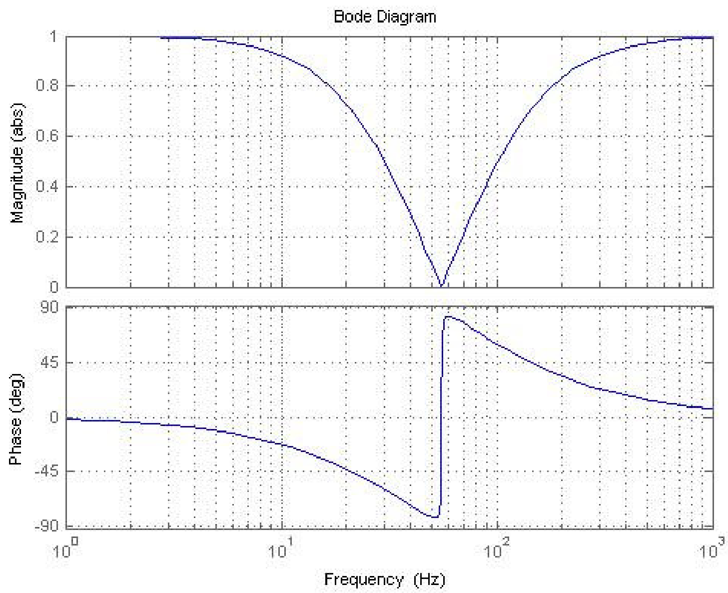


Figure 3.14: Reflection coefficient with a compliance of Cms=1exp(-3)

Acting on the value of the mechanic mass we can also change the shape of the curve. With lighter elements we could expand the range of absorption. In the next figure is shown the curve of impedance of the loudspeaker with an ideal value of the mechanical mass M_{ms} :

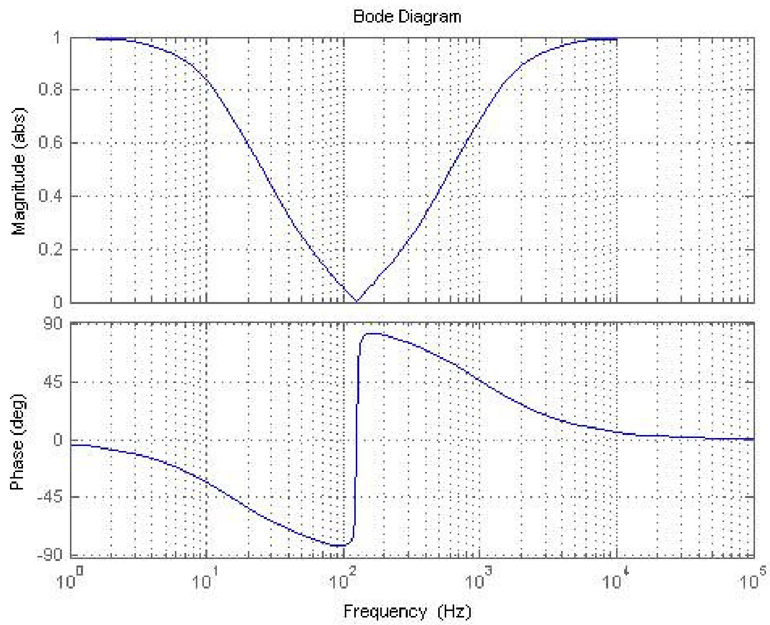


Figure 3.15: Reflection coefficient of the loudspeaker with a mechanical mass Mms=0.001 Kg

4. Experimental testing of a shunting loudspeaker

4.1. Description of the used method: the impedance tube

4.1.1 Introduction

We can determinate the absorption coefficient of a material using an impedance tube according to the norm ISO 10534-2. This method consists in the use of an impedance tube, two microphones, and a numeric frequency analyzer system. Then we can also determinate the acoustic impedance or admittance in the surface of the absorber material.

The sound source is placed in one of the ends of the tube, and the sample under test (the loudspeaker) is placed in the other side. The plane waves are generated inside the tube for the noise source, and the decomposition of the interference field is done by the measurement of acoustic pressures in two fixed places using the microphones over the walls or one microphone cross the pipe.

The measurements are determined in function of the frequency with a resolution determined from the frequency of sampling and the registered length of the numeric frequency of the analysis system used for the measurements. The usable frequency domain depends on the length of the tube and the spacing between the positions of the microphones. A bigger frequency range can be obtained from the combination of measures by different lengths and spacings.

4.1.1.1 Predecessor of the impedance tube: Kundt's tube

Kundt's tube is an experimental apparatus invented in 1866 by German physicist August Kundt for the measurement of the speed of sound in a gas or a solid rod. It is used today only for demonstrating standing waves.

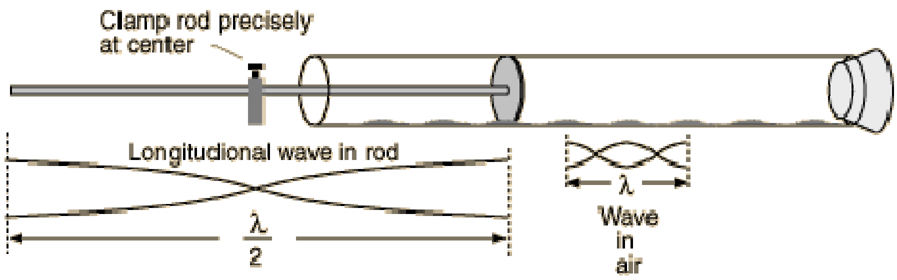


Figure 4.1: Kundt's tube

Sound in air is propagated only by means of longitudinal waves; waves in which the particle's motion consists of oscillations back and forth in the direction of propagation. In a solid such as a metal rod, sound can be transmitted either by longitudinal or transverse waves. In Kundt's experiments, longitudinal sound waves were produced in a metal rod and an air column. Using the properties of wave motion, the frequency of the sound and the speed of sound in the rod could be determined.

The apparatus consists of a glass tube supported on a metal base. A clamp at one end of the base holds a metal rod which has a metal disk attached to one end. The rod and disk extend inside the glass tube, whose position can be adjusted to centre the tube about the disk. It is important to make sure the disk is not touching the glass because the vibrations set up in the rod will break the glass. A stopper closes the other end of the tube. Modern demonstrations usually use a loudspeaker attached to a signal generator, and it's the other end of the tube which is blocked by a moveable piston to adjust the length of the tube. The velocity of any wave is given by:

$$v = \lambda f$$

(Eq.25)

where f is the frequency and λ is the wavelength. When the rod is properly stroked and set into vibration, standing waves are set up in the vibrating rod. Since the rod is clamped at its centre point, this point is a node (zero amplitude of motion) and the ends which are free to vibrate are antinodes (maximum amplitude of particles' motion along the direction of the rod).

When the rod is vibrating in this manner it is vibrating with its fundamental frequency and the wavelength of the standing wave is twice the length of the rod.

The vibrations of the rod are transmitted by the disk to the air in the glass tube closed at one end. The waves set up in the air in the glass tube have the same frequency as those in the rod. The waves are reflected at the closed end of the tube and the air in the tube is thus acted upon by two similar sets of waves travelling in opposite directions. When the length of the air column is some multiple of half wavelengths, the two oppositely travelling waves produce standing waves. The standing waves are characterized by alternate points of maximum and minimum disturbance called respectively antinodes and nodes. These nodes and antinodes may be detected by cork dust placed in the tube, the cork dust places showing characteristic striated vibration patterns at the antinodes. The distance between two successive patterns is therefore one-half the wave length of the sound in the air.

The wave length of the sound in air is thus determined. From tables the velocity of sound in air at the temperature of the room is obtained and using equation (Eq.25) the frequency may be computed. Since this is the frequency of the sound wave in the metal rod and since the wave length is twice the length of the rod, we can compute the velocity of the sound waves in the metal rod using equation (Eq.25).

Measure of absorption and reflection coefficients using the Kundt's tube and a mobile microphone

Before the two microphones method for measuring the absorption or reflection coefficients was developed, these parameters were measured using the Kundt´s tube and a mobile microphone.

The basis of this method is: A plane longitudinal wave, with amplitude A, is generated by a loudspeaker in one end of the tube. This wave comes into contact with the sample of the tested material, which is placed in the other end of the pipe. When the waves collide with the sample, one part of the energy is absorbed by the material, and the other part is reflected, returning along the tube with other amplitude of pressure B. Thus, as a result of the interference between the incident and the reflected waves, a standing wave is formed, whose study brings the information needed to calculate the absorption and reflection coefficients.

A thin rod is set up in a microphone car and moves along the tube going through the loudspeaker as is showed in the figure. This rod sends the pressure levels of the standing wave to a frequency analyser (see Figure 4.3). The position of the rod can be read in the scale parallel to the rails of the microphone car. The maximums (A + B) of pressure of two consecutive antinodes, are separated by a half wavelength, and between them, a node (A – B) is placed. These values are achieved moving the rod along the tube and reading the position in the scale.

With the relationship between maximums and minimums:

$$n = \frac{A + B}{A - B} = \frac{P_{max}}{P_{min}}$$

(Eq.26)

And considering that energy is proportional to the square of the amplitude, we can define, for the absorption coefficient:

$$Q = 1 - \frac{B^2}{A^2} = 1 - (\frac{n-1}{n+1})^2 = \frac{4}{n + \frac{1}{n} + 2}$$

(Eq.27)

which is defined as the relationship between the incident and reflected energies in the sample. This allows us to calibrate the scale of the frequency spectrometer, which gives directly the value of the absorption coefficient.

There is also a logarithmic expression which relates the absorption coefficient with the difference L in decibels between the amplitudes of the maximum and the minimum of pressure in the standing wave:

$$a = 1 - [\frac{log^{-1}\left(\frac{L}{20}\right) - 1}{log^{-1}\left(\frac{L}{20}\right) + 1}]^2$$

(Eq.28)

The relationship between a and L that allows the determination of the absorption coefficient is showed in the figure:

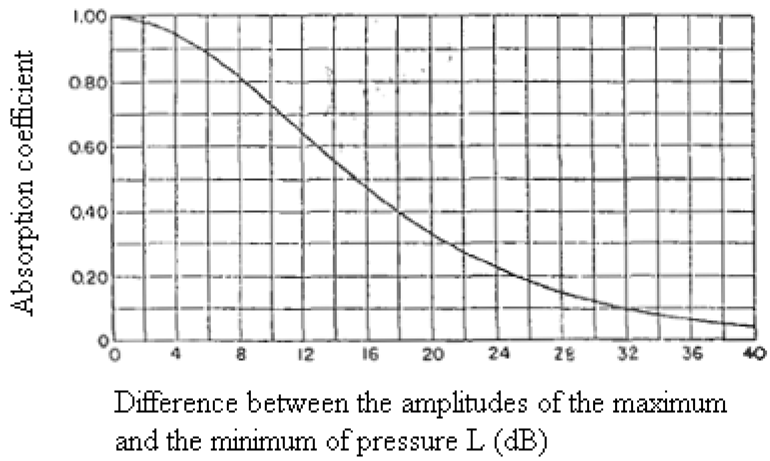


Figure 4.2: relationship curve between a and L

The use of the apparatus is restricted to small samples. This is determined by the diameter D of the tube, which has to be approximately a half wavelength of the standing wave. The exactly relationship is:

$$\lambda > 1{,}7\ D$$

That's because samples were usually tested with different sizes in tubes of different diameters.

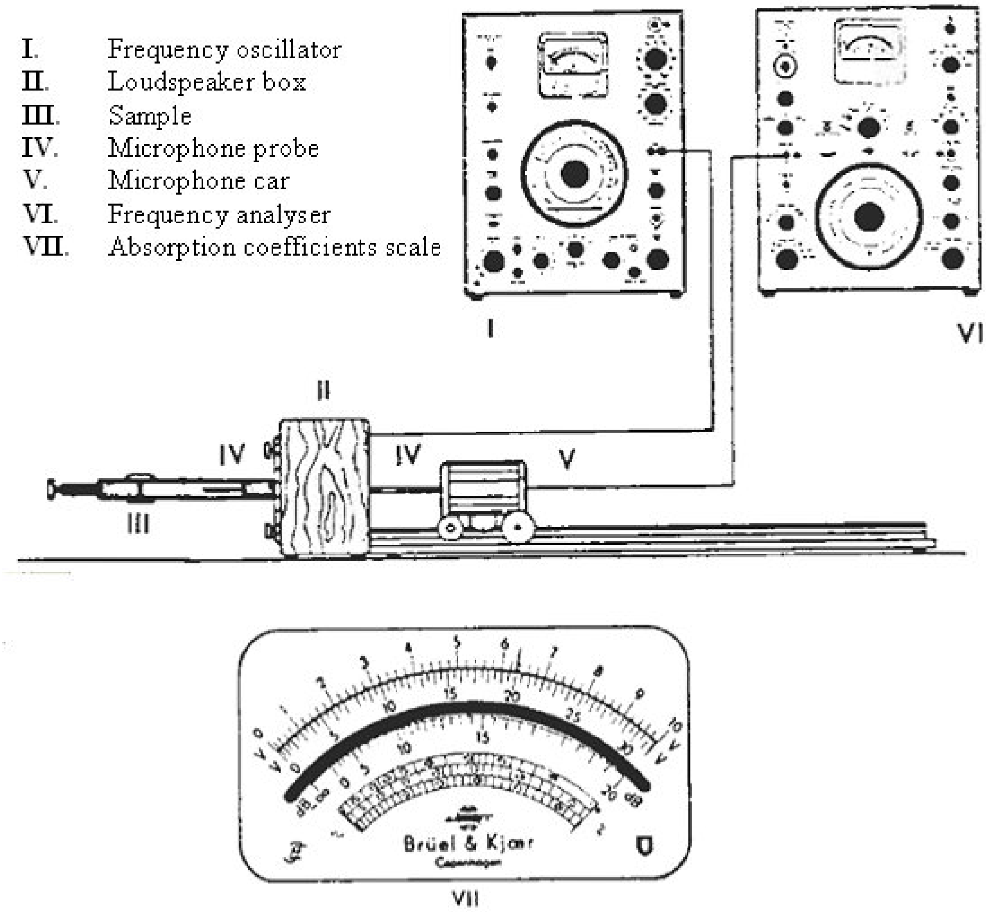


Figure 4.3: Diagram of the Kundt's tube set-up for measuring absorption coefficients

Acoustic impedance

The method of the mobile microphone in Kundt's tube was also used for measuring the acoustic impedance.

For determinate its value, it's necessary to know the value of L and the values of D_1 (distance between the sample and the first pressure minimum) and D_2 (distance between the first and second minimums). These two distances are read in the calibrated scale parallel to the microphone car.

Using these values in the expression:

$$\frac{Z}{\rho c} = \coth\left(A + jB\right)$$

(Eq.29)

With:

$$A = coth^{-1}(log^{-1}\left(\frac{L}{\rho}\right))$$

$$B = \pi(\frac{1}{2} - \frac{D_1}{D_2})$$

(Eq.30)

The values of $r/\rho c$ and $x/\rho c$ can be obtained with L, D_1 and, D_2 using the Smith Chart:

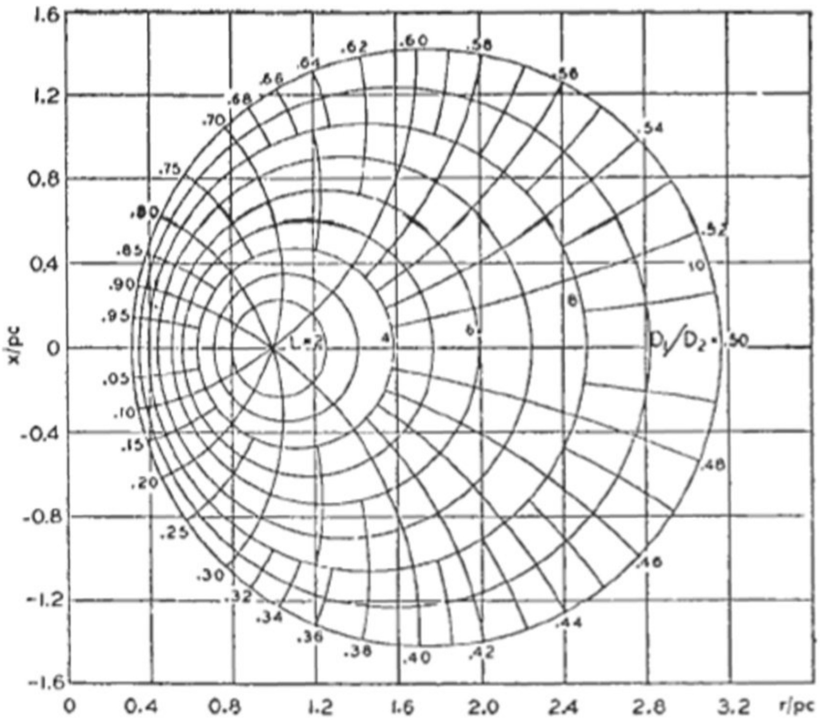


Figure 4.3: Smith Chart

4.1.2 The impedance tube

The tube should be massive and sufficiently rigid to avoid transmission of noise into the tube from outside and vibration excitation by the sound source or from background sources (e.g., doors closing). The standard ISO 10534-2 recommend the wall thickness has to be 5 percent of the tube diameter, but 10 percent is preferred. The standards do not specify the tube material, but a material such as brass is preferred because its damping is several times that of lightly damped materials such as aluminum. Brass is also three times as dense as aluminum.

The tube must be sufficiently long to present a stable plane-wave sound field to the sample under test. All sound sources produce spherical waves that decay into plane waves over distance inside a tube. However, this distance can be quite long for large diameter tubes. The standard recommends a tube length of at least three diameters, but this is marginal; a length of at least 10 - 15 diameters is preferred.

4.1.3 Microphones

Two identical microphones must be mounted flush with the inside wall of the tube and isolated from the tube (to minimize sensitivity to vibration). The diameter of the microphones must be small in comparison with c_0/f_u being c_0 the speed of sound and f_u the upper useful frequency of the tube. Also is recommended that the diameter should be lower than 20% of the distance between them. For measure with microphones mounted in the wall of the tube, are recommended pressure microphones.

4.1.4 The loudspeaker

The surface of the diaphragm must fill at least 2/3 of the section area of the tube. It must be placed inside an isolated box for avoiding any indirect transmission to the microphones.

4.1.5 Signal generator

The signal generator must be able to produce a stationary signal of plane spectral density in a specific frequency range.

4.1.6 Thermometer and barometer

The temperature inside the tube must be constant during the measurement, with a maximum variation of ± 1 K. The temperature transducer must have an exactitude of at least ± 0.5 K

The atmospheric pressure must be measured with a tolerance of ± 0.5 kPa

4.1.7 Signal analysis equipment

It is constituted by an amplifier and a measurement system with two channels of the Fast Fourier Transform (FFT). This is required for measuring the acoustic pressure in two places of the microphones and for calculate the transfer function H_{12} between them.

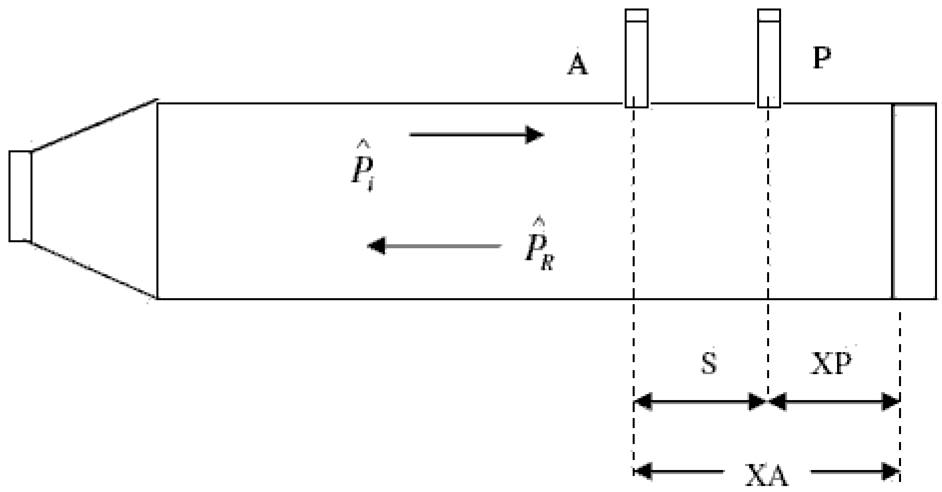


Figure 3.16: Mounting of the impedance tube and microphones

4.2. Procedure of measure

4.2.1 Correction of the maladjustment of the microphones.

This is effectuated by the change of the channels for each measure on a test tube.

First we place the microphones in the configuration 1 (normalized configuration), and we get the transfer function $H^{I}{}_{12}$, then we change the microphones A and B for obtaining the transfer function $H^{II}{}_{12}$.

Now we can calculate the calibration factor as:

$$H_c = (H^{I}{}_{12}/H^{II}{}_{21})^{1/2} = |H_c|e^{j\theta_c}$$

(Eq.25)

Or, as the analyzer can only measure the transfer function in one direction, we can use:

$$H_c = (H^{I}{}_{12} \cdot H^{II}{}_{12})^{1/2} = |H_c| e^{j\theta_c}$$

(Eq.26)

Then, we take the frequency response of the microphones in the configuration I:

$$H'_{12} = |H'_{12}| e^{j\theta'} = H'_r + H'_j$$

(Eq.27)

Where

θ' is the not corrected phase angle;

H'_r is the real part of H'_{12};

H'_j is the imaginary part of H'_{12} ;

And finally we correct the maladjustment of the microphone responses with:

$$H_{12} = |H_{12}| e^{j\theta} = \frac{H'_{12}}{H_c}$$

(Eq.28)

4.2.2 Determination of the reflection coefficient

From the transfer function H_{12}, the pressure reflection coefficient R of the material (the loudspeaker in our case) is determined from the following equation:

$$r = |r| e^{j\theta_r} = r_r + jr_i = \frac{H_{12} - H_I}{H_R - H_{12}} e^{2jk_0 x_1}$$

(Eq.29)

Where

r_r is the real component;

r_i is the imaginary component;

x_1 is the distance between the material and the further microphone

θ_r is the phase angle of the relection coefficient at normal incidence

H_I and H_R ar the transfer function of the incident wave, which are defined as:

$$H_I = e^{-jk_0 s}$$

$$H_R = e^{jk_0 s}$$

Being $s = x_1 - x_2$ the separation between both microphones.

4.2.3 Determination of the absorption coefficient

The absorption coefficient at normal incidence is:

$$\alpha = |\mathrm{r}|^2 = 1 - \mathrm{r}_\mathrm{r}^2 - \mathrm{r}_\mathrm{i}^2$$

(Eq.30)

4.2.4 Determination of the acoustic impedance

We can calculate the acoustic impedance as:

$$Z = (1 + r)/(1 - r)$$

(Eq.31)

And hence the acoustic admittance:

$$G = \frac{1}{Z}$$

(Eq.32)

4.3 Results

For the following measures, the equipment used has been an impedance tube ended in a loudspeaker SPH-300TC (see appendix C) mounted in a box of 50x50x34, which acts as a compliance of value:

$$C_{as} = \frac{V_{as}}{1.4\exp{(5)} \cdot Sf}$$

Where V_{as} is the volume of air in the box, and 1.4exp(5) is the adiabatic compression coefficient of the air.

Mounted in the wall of the tube, two microphones Norsonic 1201 (see appendix D) were connected to the signal analysis system PULSE.

For the short circuited loudspeaker, absorption coefficient and acoustic admittance values for each frequency are shown in figures 3.17 and 3.18, and for the loudspeaker ended in a resistor these curves are shown in figures 3.19 and 3.20.

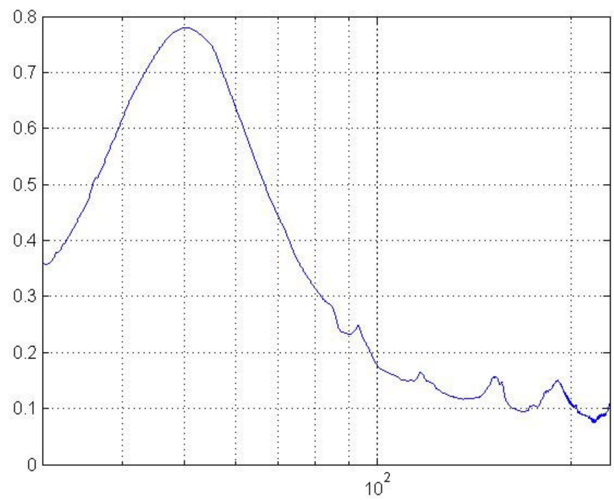


Figure 3.17: absorption coefficient of the short circuited SPH-300TC loudspeaker

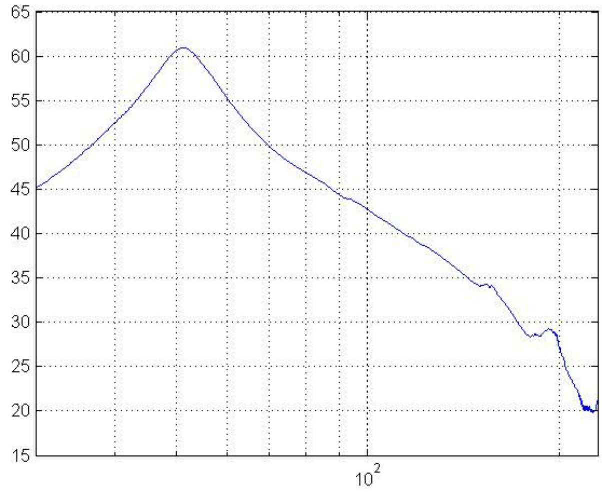


Figure 3.18: acoustic admittance of the short circuited SPH-300TC loudspeaker

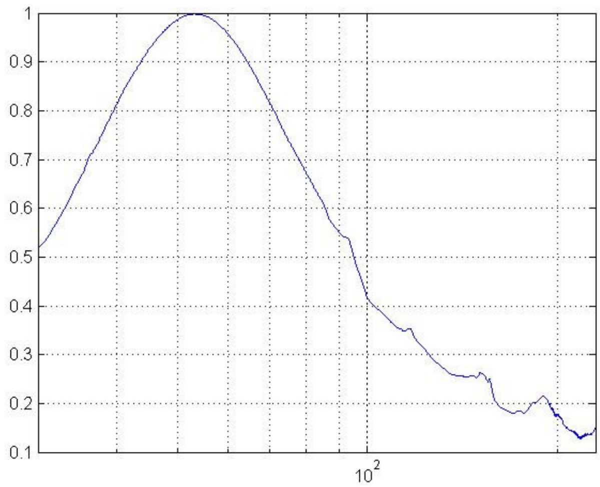


Figure 3.19: absorption coefficient of the SPH-300TC loudspeaker ended in a resistor

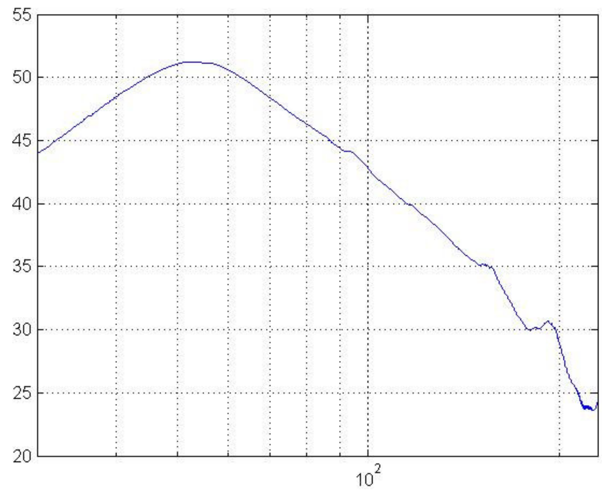


Figure 3.20: acoustic admittance of the SPH-300TC loudspeaker ended in a resistor

5. Conclusions

The main objective of this work was to optimize the parameters of a loudspeaker for achieving total dumping at an specific range of low frequencies. In this case, the most useful loudspeaker is a passive one, ended in a resistor, so that the performing is similar to a Helmholtz resonator.

The frequency of resonance is where a total absorption can be achieved. Moving this frequency through the spectrum, and changing its shape, is possible varying the value of the mechanical mass and compliance.

With the MATLAB designs, it's seen that is possible to achieve total damping at low frequencies, thought it needs the use of very light materials for the construction of the loudspeaker.

The experimental testing with the impedance tube has demonstrated that the prototype works, so once achieved the optimal conditions of the loudspeaker, it's possible to build panels of these loudspeakers for controlling the reflection of an enclosure.

6. References

1. Dominik niederberger, Smart damping materials using shunt control.

2. Dr. Hervé Lissek, Pr. Xavier Meynial, *Active impedance control for room acoustics*

3. José Luis Sánchez bote, Emilio Álvarez Fernández, *Transductores Electroacústicos*

4. José Luis Sánchez Bote, Altavoces: *características, filtros de cruce y bobinas*

5. Florian Sandoz, Project de Master, *Conception et optimisation de filtres num'eriques pour matériaux actifs à propiétés acoustiques variables*

Web sites

1. http://www.beyma.com

2. http://www.bmm-electronics.nl

3. http://softwarepractice.org

Appendix A : MATLAB codes

125 Hz loudspeaker absorber

```
%Javier Rodríguez De Antonio
%Modeling of a shunting loudspeaker


%bobine parameters
rhomc=8960;          % density of copper [kg/m^3]
N = 1;               % rounds of the cable
d = 1e-3;            % Diameter of wire [m]
B = 1;               % magnetic field [tesla]
L =(2*pi*0.01)*N;    % coil length [m] (0.01)
s=pi*(d/2)^2;        % surface of the section of wire [m^2]
Mmc=rhomc*L*s;       % mass of the coil [kg]
disp(B*L)

% loudspeaker parameters
fs = 125;                    % resonance frecuency [Hz]
rhomd= 1;                    % surfacic mass of diaphragm [kg/m2]
a = 0.5e-1;
Sf=pi*a^2;                   % diaphragm area and surface [m], [m^2]
Mmd = rhomd*Sf;              % diaphragm mass[kg]
Mms =Mmd+Mmc                 % total mass [kg]
Cms = 1/(Mms*(2*pi*fs)^2)    % mechanic compliance
Rms = 1;                     % mechanic resistance [N/m/s]
rho=1.18;                    % density of air [kg/m^3]
c=345;                       % velocity of air [m/s]
Zc=rho*c;                    % air specific impedance

%electric resistance
rhoe=1.7e-8;
Re=rhoe*L/s;
Le=2e-3;

%acoustic parameters
Ras = Rms/Sf;
Cas = Cms*Sf;
Mas = Mms/Sf;
Rar = Zc*Sf;
Mar = (rho*Sf*a)/(3*pi/8);
Zad = rho*c;
Rar = Zc*Sf;
Mar = 0;

%optimal value of the resistance
Rc=(B*L)^2/(Zc*Sf-Rms)-Re;

%value of total resistance
Ras2 = Ras + ((B*L)^2/(Sf*(Re+Rc)));

%acoustic impedance
Za = TF([(Mas+Mar)*Cas (Rar+Ras2)*Cas 1],[0 Cas 0]);
bode(Za)
%reflection coefficient
r=(Za-Zc)/(Za+Zc);
bode(r);
```

```
margin(r)
 %acoustic admitance
bode(Zc/Za);
```

SPH-300TC loudspeaker simulation

```
BL =10.3;


% loudspeaker
a = 0.153; Sf=pi*a^2;        % diaphragm area and surface [m], [m^2]
Mms= 0.068;                   % g              % total mass [kg]
Cms = 0.85;                  %  mm/N   mechanic compliance
Rms = 3.2;                    % kg/s              % mechanic resistance
[N/m/s]
rho=1.18;                     % density of air [kg/m^3]
c=345;                        % velocity of air [m/s]
Zc=rho*c;                     % air specific impedance

% box that includes the loudspeaker
Vas = (0.5*0.5*0.34);
Cas2 = Vas/(Sf*1.4e5);


%electric parameters
Re=6.3;
Le=1e-3;

%acoustic parameters
Ras = Rms/Sf;
Cas = Cms*Sf;
Mas = Mms/Sf;
Rar = Zc*Sf;
Mar = (rho*Sf*a)/(3*pi/8);

%optimal value of the resistance
Rc=(BL)^2/(Zc*Sf-Rms)-Re;
%Rc = 0;
%value of total resistance
Ras2 = Ras + ((BL)^2/(Sf*(Re+Rc)));

%acoustic impedance
 Za = TF([(Mas+Mar)*Cas*Cas2 (Rar+Ras2)*Cas*Cas2 Cas+Cas2 0],[0
Cas*Cas2 0 0])
bode(Za)


%reflection coefficient
r=(Za-Zc)/(Za+Zc);
bode(r);
margin(r) ;
```

Appendix B: PULSE codes

1. Corrected transfer function

```
transfer function corrected


H121 = IMPORT "C:\Documents and
Settings\Pulse\USERs\Sound_barrier\H12_tube300_30Hz_230Hz.txt"
H122 = IMPORT "C:\Documents and
Settings\Pulse\USERs\Sound_barrier\H21_tube300_30Hz_230Hz.txt"

Hc = sqrt(H121*H122)

H12={Function Group}:{Frequency Response H1(Micro 2,Micro 1) - Input}

RESULT = H12/Hc
```

2. Normalized admittance

```
H121 = IMPORT "C:\Documents and
Settings\Pulse\USERs\Sound_barrier\H12_tube300_30Hz_230Hz.txt"
H122 = IMPORT "C:\Documents and
Settings\Pulse\USERs\Sound_barrier\H21_tube300_30Hz_230Hz.txt"
Hc = sqrt(H121*H122)

H12={Function Group}:{Frequency Response H1(Micro 2,Micro 1) - Input}
H12c=H12/Hc

S=0.6
x1=1.3
pi=3.141592654
T=273+21.2
c=343.2*sqrt(T/293)

f=xvector(H12)
k0=2*pi*f/c

j=(H12-real(H12))/imag(H12)

HI=exp(-j*k0*S)
HR=exp(j*k0*S)

R = (H12c-HI) / (HR-H12c) * exp(j*2*k0*x1)

Z = (vector1(R)+R)/(vector1(R)-R)

RESULT = vector1(Z)/Z
```

3. Reflection coefficient

```
H121 = IMPORT "C:\Documents and Settings\Pulse\USERs\Sound_barrier\H12_tube300_30Hz_230Hz.txt"
H122 = IMPORT "C:\Documents and Settings\Pulse\USERs\Sound_barrier\H21_tube300_30Hz_230Hz.txt"
Hc = sqrt(H121*H122)

H12={Function Group}:{Frequency Response H1(Micro 2,Micro 1) - Input}
H12c=H12/Hc

S=0.6
x1=1.3
pi=3.141592654
T=273+21.2
c=343.2*sqrt(T/293)

f=xvector(H12)
k0=2*pi*f/c

j=(H12-real(H12))/imag(H12)

HI=exp(-j*k0*S)
HR=exp(j*k0*S)

R = (H12c-HI) / (HR-H12c) * exp(j*2*k0*x1)

RESULT = R
```

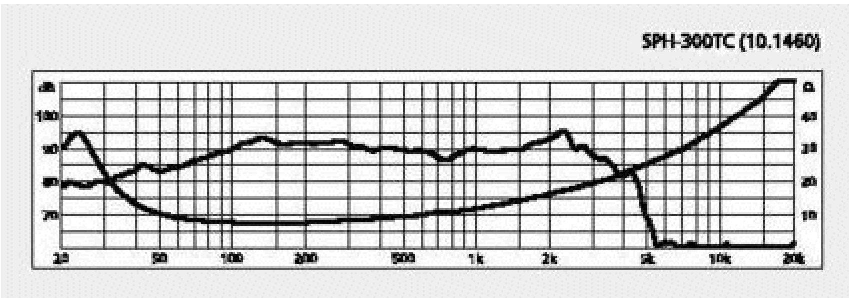
4. absorption coefficient

```
H121 = IMPORT "C:\Documents and Settings\Pulse\USERs\Sound_barrier\H12_tube300_30Hz_230Hz.txt"
H122 = IMPORT "C:\Documents and Settings\Pulse\USERs\Sound_barrier\H21_tube300_30Hz_230Hz.txt"

Hc = sqrt(H121*H122)

H12={Function Group}:{Frequency Response H1(Micro 2,Micro 1) - Input}
H12c=H12/Hc

S=0.6
x1=1.3
pi=3.141592654
T=273+21.2
c=343.2*sqrt(T/293)

f=xvector(H12)
k0=2*pi*f/c
```

```
j=(H12-real(H12))/imag(H12)

HI=exp(-j*k0*S)
HR=exp(j*k0*S)

R = (H12c-HI) / (HR-H12c) * exp(j*2*k0*x1)

alpha = vector1(f)-abs(R)^2

RESULT = alpha
```

Appendix C : SPH-300TC data sheet.

Technical Facts Speakers	
Impedance (Z)	2 x 8Ω
Resonant frequency (fs)	23Hz
Max. frequency range	f3-2,000Hz
Rec. crossover freq. (fmax.)	1,000Hz
Music power	2 x 250W_{MAX}
Power rating (P)	2 x 120W_{RMS}
SPL (1W/1m)	91dB
Suspension compl. (Cms)	0.85mm/N
Moving mass (Mms)	68g
Mech. resistance (Rms)	3.2kg/s
Mech. Q factor (Qms)	3.03
Electr. Q factor (Qes)	0.26
Total Q factor (Qts)	0.24
Equivalent volume (Vas)	290 l
DC resistance (Re)	2 x 6.3Ω/3.2Ω
Voice coil induct. (Le)	2 x 1.0mH/1.0mH
Voice coil diameter	50mm
Voice coil former	aluminium
Voice coil winding height	16mm
Air gap height	8mm
Linear excursion (X_{MAX})	±4mm
Eff. cone area (Sd)	495cm^2
Volume displacement (Vd)	198cm^3
Force factor (BxL)	10.3Tm
Reference efficiency (No)	1.3%
Magnet diameter	156mm
Magnet weight	49.8oz.
Mounting cutout	Ø 280mm
Mounting depth	140mm
Dimensions	Ø 306mm
Weight	4.0kg
Rec. net cabinet volume	
Closed	38 l
Bass-reflex	72 l



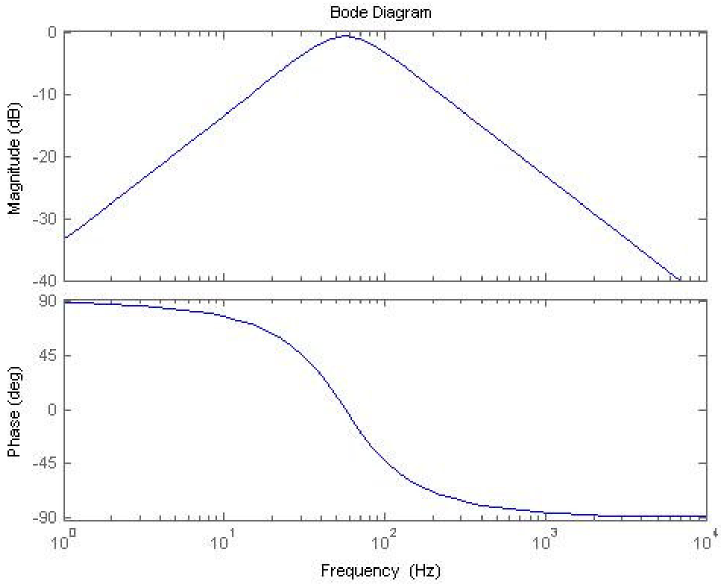
Appendix D: Norsonic 1201 data sheet

Input :	ISO specification for 1/2" measurement microphones
Connector	Integral Lemo 1B type
Frequency range	1 Hz to 1 MHz ± 0.5 dB
Input impedance	20 GΩ, 0.2 pF
Output impedance	55 Ω @ 1 kHz, C_s : 20 pF
Noise, A-weighted	C_s : 20 pF : < 3 µV
Noise, Linear	C_s : 20 pF : < 6 µV
Gain, typical	- 0.15 dB
Temperature, operating	- 40 to + 60 °C
Humidity, operating	0 to 95 %RH
Current consumption	120 V or ± 60 V : 2.5 mA
Current consumption	28 V or ± 15 V : 0.7 mA
Weight	33 grams
Dimensions	12.7 mm (Ø), 77.5 mm long

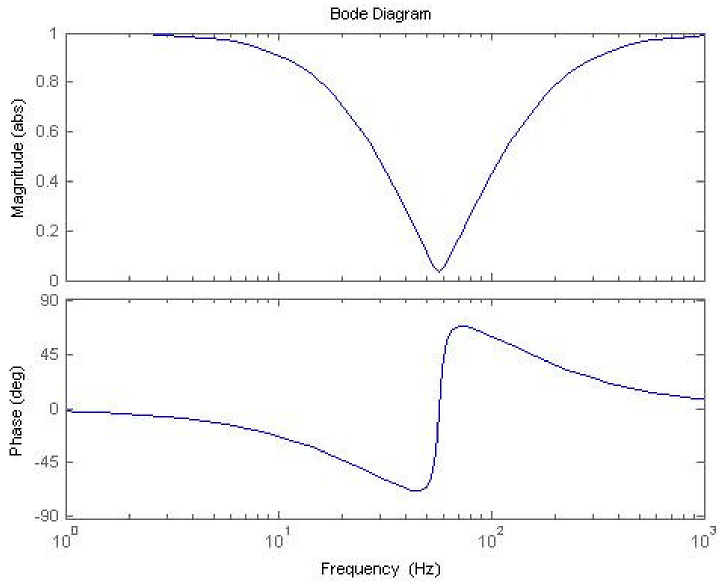
Upper Frequency vs. Cable Length :
THD < 1% : supply voltage 120 V and assuming 50 mV/Pa sensitivity

SPL	10 m	20 m	50 m	100 m
150 dB	8 kHz	5 kHz	2 kHz	1 kHz
140 dB	35 kHz	20 kHz	12 kHz	5 kHz
130 dB	> 100 kHz	70 kHz	40 kHz	15 kHz
120 dB	> 100 kHz	> 100 kHz	> 100 kHz	60 kHz
110 dB	> 100 kHz	> 100 kHz	> 100 kHz	> 100 kHz

Appendix E : MATLAB simulation of SPH-300TC results

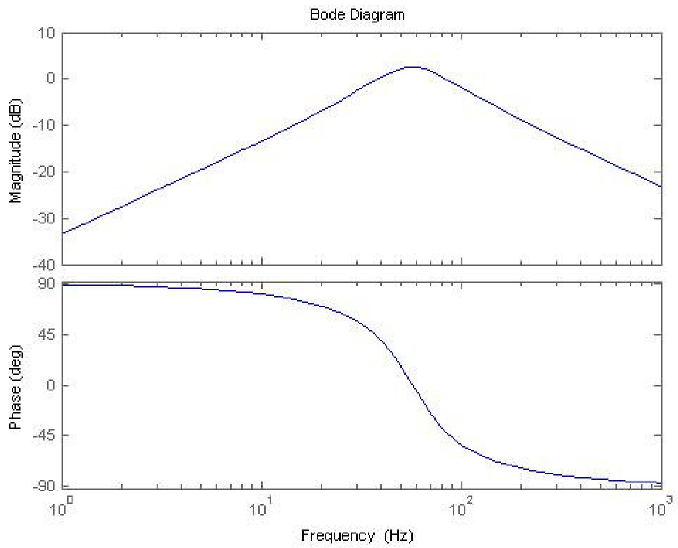


a)

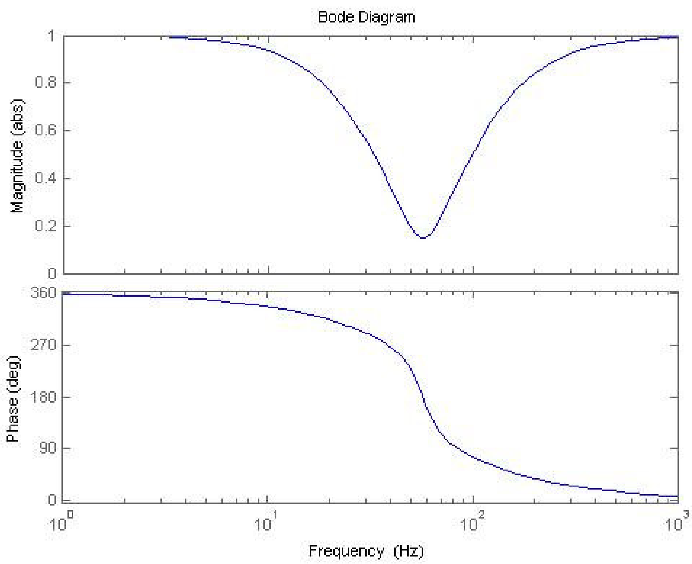


b)

Figure E.1: a) acoustic admittance and b) reflection coefficient of the MATLAB simulation of SPH-300TC with the optimal resitance Rc



a)



b)

Figure E.2: a) acoustic admittance and b) reflection coefficient of the MATLAB simulation of SPH-300TC short circuited

Appendix F : Set – up photos

Figure F1: SPH-300TC loudspeaker mounted in a box

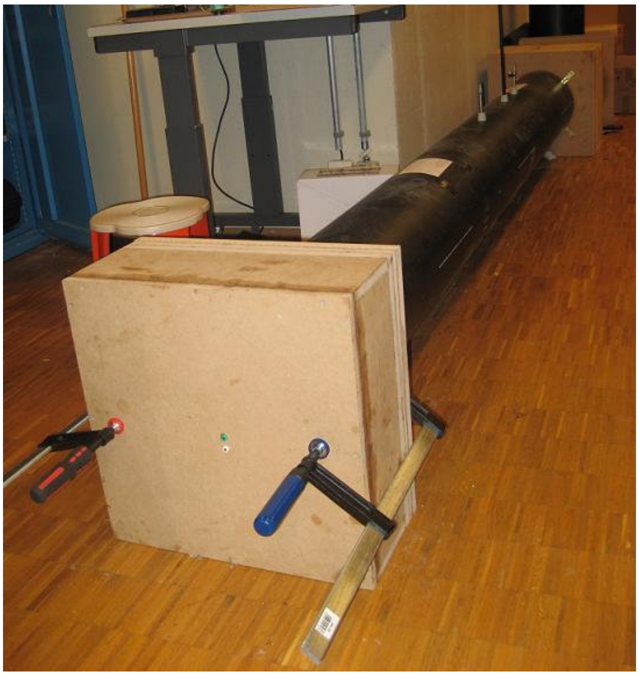


Figure F2: impedance tube ended in the SPH-300TC

Figure F3: Pulse hardware and signal amplifier