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Modeling of a Compression Driver using Lumped Elements

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ABSTRACT

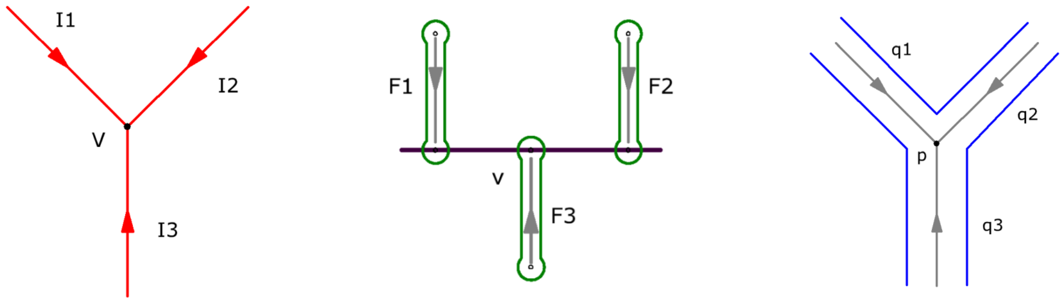
Modeling linear response functions of a typical electro-dynamic compression driver with the help of lumped elements should be an easy task. However, it turns out that acoustic channels and slits inside such a device can go into strong coupling with the fundamental mechanical resonance. This paper demonstrates an extended electro-mechano-acoustical network and compares its response functions to measurement.

Keywords: Electro-acoustic, Lumped Element, Compression Driver

INTRODUCTION

Modeling is a calculation technique, which in our context yields transfer functions in the frequency domain such as the sound pressure response or the electrical driving point impedance. Modeling with lumped parameters involves using building blocks or symbols, which represent dominant properties of the device, such as the mass and stiffness of a fundamental resonator.

Lumped elements result from forming mean-values of parameters such as potentials and flows by means of spatial integration. For example, the lumped pressure inside a waveguide can be regarded as a mean-value of the cross-section. In the frequency range, where the wavelength is much larger than the device, the lumped parameter features an approximation of this mean-value. In the mechanic domain lumped parameters refer to the rigid body motion in a certain direction.



Electrical domain — Mechanical domain — Acoustical domain

Figure 1 - Lumped network parameters

A lumped parameter is either a potential or a flow. There are the voltage V, the current I, the velocity v, the force F, the sound-pressure p and the volume-velocity q. The lumped parameters are selected in such a way that the product yields power: $P_e = 1/2 \cdot V \cdot \bar{I}$, $P_m = 1/2 \cdot v \cdot \bar{F}$, $P_a = 1/2 \cdot p \cdot \bar{q}$ or in more general terms: $P = 1/2 \cdot potential \cdot \overline{flow}$. P_e, P_m and P_a are the electric, mechanic and acoustic power. Figure 1 graphs the Kirchhoff rule, which says that the sum of flows should be zero. In between the electrical, mechanical and acoustical domains there are transducers.

A compression driver is an electro-acoustic transducer. It converts electric energy into an acoustic one. The electric terminals are typically connected to the output stage of an amplifier or a filter-network of low impedance. The acoustic outlet drives a waveguide, which can be a large horn or a beam-forming device. A symbolic network is shown in figure 2. This paper reports on the process of modeling with the help of mainly lumped elements. We want to write down the string of thoughts, which leads to particular design steps of the model.

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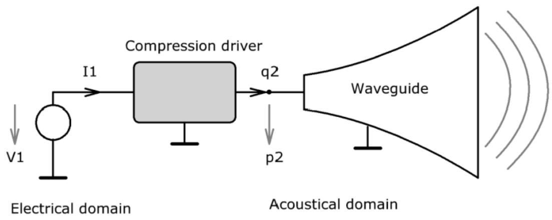

Figure 2 - Electro-acoustic transducer

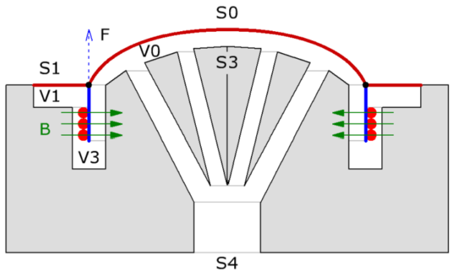

Figure 3 - Compression driver cut-view

Modeling Compression Driver

The driver we want to model has an outlet of 1.4 in, which is approx 36 mm. The diameter of the membrane is 86 mm. It has the shape of a dome. Figure 3 displays the circular symmetric cut-view. The motion of the membrane is in axial direction, which produces sound waves. The inner area of the dome S_0 is opposite a range of radial slits which form the entrance to a system of concentric ducts, which is called *phase-plug*. Between the membrane and the phase-plug there is a spacing, which should be large enough to allow for membrane excursion. The spacing together with the area of the membrane S_0 forms the volume V_0 of the so-called *compression chamber*. The design of the phase-plug and the size of V_0 should be designed carefully in order to provide a large bandwidth of the sound pressure response at high frequency. Because the effective area into the phase-plug S_3 is typically smaller than the area of the membrane S_0 there is a transformation of flux and pressure according to the *compression ratio* = S_0/S_3. The outer area of the dome is not used and typically shielded by an enclosure (not shown in figure 1). S_1 stands for a membrane area, which encircles the main-membrane and functions as suspension of the vibrating assembly. Between the suspension and the main-membrane there is attached the *voice coil former*. The voice coil consists of a number of wire-turns arranged in the flux of a magnetic field B in such a way that a force F drives the voice coil former in axial direction. Between the voice coil former and the iron of the magnetic device a small path of air is formed. This so-called *voice coil slit* should be as narrow as possible for reasons of motor-efficiency. Underneath the suspension there is a small chamber of volume V_1. It is important to note that volume V_1 and the compression chamber V_0 are acoustically coupled via the voice coil slit (see also [1,2]). Later, we will see that the response exhibits a high sensitivity to this acoustic arrangement.

One can argue that the driver should be analyzed when loaded with a *plane-wave tube*, so that the test is close to operation of horn-load. The plane-wave-tube would then provide an acoustic load of $Z_a = \rho \cdot c \cdot S_4^{-1}$ with radiation impedance $Z_a = p/q$, density of air ρ, speed of sound c and radiation-area S_4. Of advantage for analysis is that this value would be real and frequency independent. Hence, the plane-wave-tube-load would be an interesting alternative, and it will be investigated in a further work.

This paper investigates the behavior of the driver under free-radiation condition, i. e. there is no waveguide attached at the outlet. The advantage of this particular radiation condition is that there is little damping due to radiation. This in turn will yield to strong resonance among reactive components within the driver, which are easy to observe.

In Vacuum

Under no-air condition we can neglect all acoustic components and concentrate on the electrical and mechanical domains. In a *mechanical mobility network* the potential is the velocity in a particular direction. The flow is the force. We can apply standard network analysis [3] with impedance and admittance components:

$$Y_{mob} = \frac{1}{Z_{mob}} = \frac{flow}{potential} = \frac{F}{v} \tag{1}$$

$$\text{Mass} \quad Y_m = j\omega \cdot M \qquad \text{Dashpot} \quad Y_r = R \qquad \text{Stiffness} \quad Y_k = K / j\omega \tag{2}$$

The mobility type network facilitates the identification of the symbols. Typically, one looks first for the velocities, i. e. one may ask which components share a particular motion. In our simple example there is only one velocity, which is shared by the mass, the suspension and the dashpot. M symbolizes the total mass of the vibrating assembly assuming infinite stiffness. It includes the mass of the membrane, the suspension and the voice coil former without acoustic load. K is the stiffness of the suspension in vacuum assuming a linear spring without a mass. R stands for a velocity related mechanic loss-factor typically caused within the suspension. The schematic is displayed in figure 4. The mechanic mass is always grounded. The suspension is attached to a very large mass, so large that we can connect the spring and the dashpot also to ground. The mass and the spring form a resonance at $\omega_s = \sqrt{K/M}$. In loudspeaker jargon we call this the *fundamental resonance*.

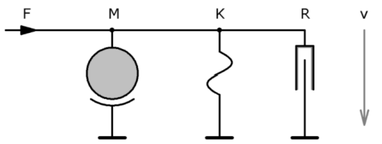

Figure 4 - Mechanical mobility network

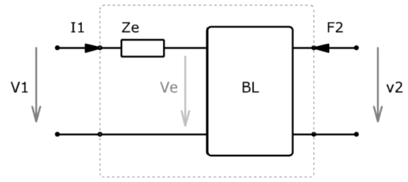

Figure 5 - Electro dynamic motor

Our compression driver features an electro-dynamic motor with *Lorentz Force* $F = BL \cdot I$, induction $V_e = BL \cdot v$ and voice coil impedance Z_e. The schematic is displayed in figure 5. At very low frequency Z_e is equivalent to a resistance R_e. At higher frequencies the value of R_e increases because of losses caused by eddy currents. The reactance of Z_e behaves similar to an inductance [4,5]. For network analysis it is useful to model the motor as an electro-mechanical four-pole with *admittance transfer matrix*, which couples the electrical impedance network to the mechanical mobility network.

$$[Y] = \frac{1}{Z_e} \cdot \begin{bmatrix} 1 & -BL \\ BL & -BL^2 \end{bmatrix} \qquad \begin{bmatrix} I_1 \\ F_2 \end{bmatrix} = [Y] \cdot \begin{bmatrix} V_1 \\ v_2 \end{bmatrix} \tag{3}$$

The force of the motor drives the mechanic structure. Its response is a velocity, which in turn induces a voltage at the electric terminals. This coupling can be symbolized by combining the model of the motor-four-pole with the mechanic circuit shown in figure 6.

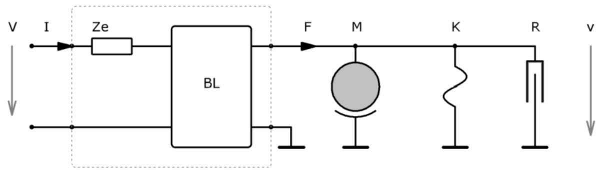

Figure 6 - Electro-mechanical network

This network is ready for inspection of transfer-functions. An easy to measure and to calculate parameter is for example, the electrical driving point function $Z_0 = V/I$. In order to measure this function the example-driver was placed into a vacuum chamber. Measurements can be used to verify the model and to identify the electro-mechanical parameters of the components. Figure 7 displays an overlay of the simulation and measurement curves. The fundamental resonance is at $f_s = 625$ Hz.

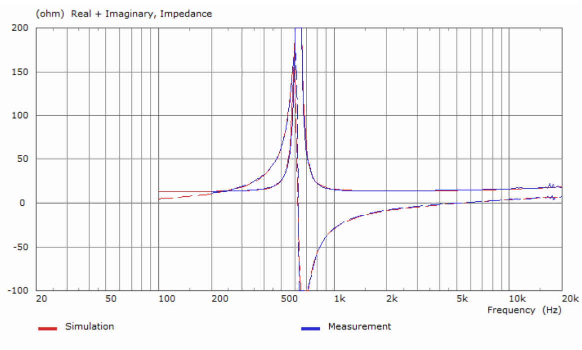

Figure 7 - Electrical driving point function of driver in vacuum

Acoustic Coupling - Simple Model

The mechanical domain is coupled to the acoustical domain with the help of membranes of area S, which are assumed to vibrate in axial rigid body motion. In the *acoustical impedance network* the potential is the sound-pressure p. The flow is the volume-velocity q in a certain direction. As before we can apply standard network analysis [3] with impedance and admittance components, where

$$Y_{imp} = \frac{1}{Z_{imp}} = \frac{flow}{potential} = \frac{q}{p} \quad (4)$$

The three basic acoustic elements are:

$$\text{Mass} \quad Y_{ma} = 1/(j\omega \cdot M_a) \qquad \text{Resistance} \quad Y_{ra} = 1/R_a$$
$$\text{Compliance} \quad Y_{ca} = j\omega \cdot C_a \quad (5)$$

In the simplified model of the compression driver we take into account only the two sides of the main membrane. The particular shape of the membrane is ignored. Each side of the membrane can be symbolized as a gyrator, which transforms the mechanical mobility network into the acoustical impedance network as shown in figure 8.

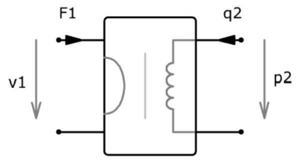

Figure 8 - Mechano-acoustical coupler

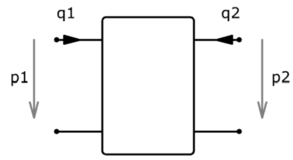

Figure 9 - Acoustic four-pole

Because the sound pressure is p = F/S and the volume-velocity is q = v·S with effective area S [6], the admittance matrix yields:

$$[Y] = \begin{bmatrix} 0 & S \\ -S & 0 \end{bmatrix} \qquad \begin{bmatrix} F_1 \\ q_2 \end{bmatrix} = [Y] \cdot \begin{bmatrix} v_1 \\ p_2 \end{bmatrix} \quad (6)$$

Between the membrane and the input to the phase-plug there is the compression chamber. The actual shape of this chamber is notoriously complicated, because typically there is curvature and a distribution of slits as sketched in Figure 3. However, at frequencies where the wavelength is much larger than these details we can allow for some approximation. For example, the compression chamber can be modeled with the help of a simple acoustic compliance

$$C_a = V/(\rho \cdot c^2) \quad (7)$$

with volume V, medium density ρ and speed of sound c. The acoustic compliance is always grounded.

Because of low frequency approximation we ignore the complex structure of concentric ducts of annular cross-section of the phase-plug and simply calculate with the cross-section areas. The symbolic representation of this waveguide is a four-pole as shown in figure 9.

The admittance matrix for a duct of linear rising cross-section area is (following ideas of [7]):

$$[Y] = y_0 \cdot \begin{bmatrix} y_{11} & y_{12} \\ y_{21} & y_{22} \end{bmatrix} \qquad \begin{bmatrix} q_1 \\ q_2 \end{bmatrix} = [Y] \cdot \begin{bmatrix} p_1 \\ p_2 \end{bmatrix} \quad (8)$$

$$y_{11} = y_{0L}^2 \cdot (\cos(kL) + y_{20} \cdot sinc(kL)) \qquad y_{12} = -y_{0L}$$

$$y_{21} = -y_{0L} \qquad y_{22} = \cos(kL) - y_{2L} \cdot sinc(kL)$$

$$y_0 = \frac{S_{mo}}{j \cdot \rho c \cdot \sin(kL)} \qquad y_{20} = \sqrt{(S_{mo}/S_{th})} - 1$$

$$y_{0L} = 1/(y_{20} + 1)$$

$$y_{2L} = y_{20} \cdot y_{0L}$$

$$sinc(x) = \sin(x)/x$$

With wave-number $k = \omega/c$, length of waveguide L, cross-section areas at the throat S_{th} and at the mouth S_{mo}.

The output of the phase-plug would normally be connected to a horn or similar device. However, here, in a test-case, we arrange for free radiation, this means there is no further waveguide attached to the outlet S_4 of the driver. Also, the rear of the membrane radiates free because we do not want any influence from the rear-enclosure. This condition was found to be optimum for measurement as well as for the simulation, and it provides low damping of the resonances of the driver.

The assembly of the acoustic network follows the same recipe as in the mechanical or electrical domains. As a first step we try to identify zones of equal potential which is the sound-pressure. We assume a large wavelength for approximation. For example, we assume identical sound pressure at the membrane, the compression chamber and at the inlet of the phase-plug. Hence, all associated terminals of the acoustic lumped elements are connected.

The acoustic of radiation is modeled with the help of the *boundary element method* (BEM), which is coupled to the lumped element part. As a result we should get excellent values for the radiation impedance at the outlet and at the rear membrane. The radiation impedance can be calculated from the mean-value of pressure distribution over the radiator-area. Details are explained in a dedicated paper [8]. Further the BEM also helps in predicting radiation to the exterior space because it takes into account most of the diffraction and reflection conditions of the measurement setup. A sketch of the boundary conditions and an example field is shown in figure 10.

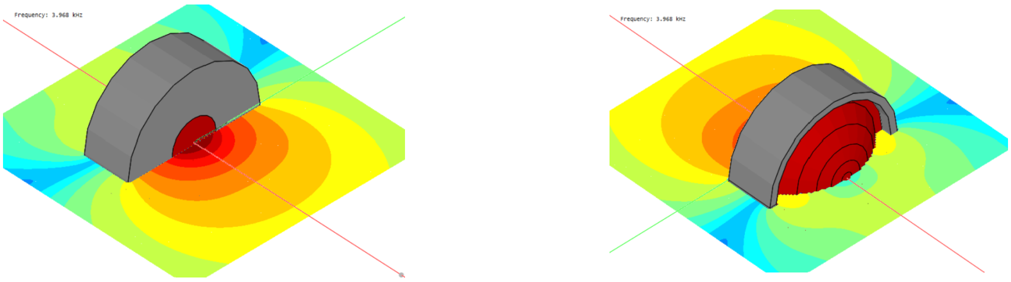

Figure 10 - Free radiation. Example of pressure field at 3900 Hz (range 50 dB).

At this stage we have all ingredients available in order to model the whole electro-mechano-acoustical network. Going from left to right we have a voltage source S_1, which feeds the electro-dynamic-motor via a generator-resistance R_g. The motor makes the membrane, voice-coil former and the suspension system vibrate axially with velocity v.

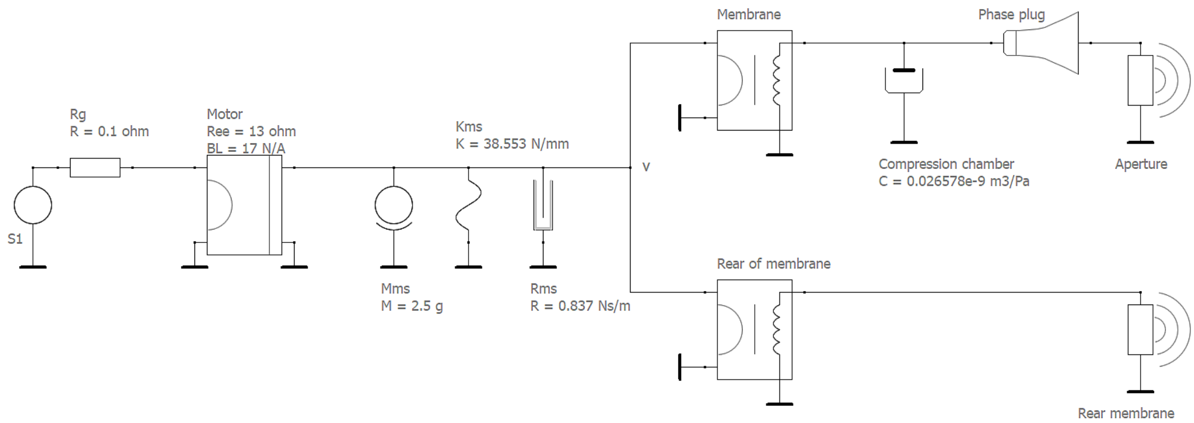

Figure 11 - Electro-mechano-acoustical network - simple modeling

Its mass and stiffness go into resonance. However, because of acoustic loading there can be multiple resonances. One side of the membrane radiates into the phase-plug and from there into the exterior domain. The rear of the membrane radiates directly into the exterior domain. There is also mutual coupling between the radiators in the exterior domain. We can place a microphone and measure the sound pressure response. Results are shown in figure 12.

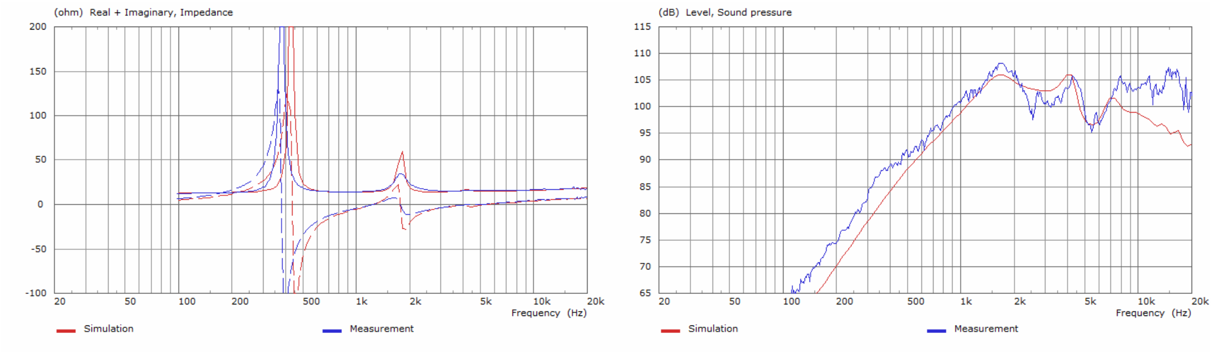

Figure 12 - Simple model: Electric driving point impedance and
sound pressure level at 180 mm on-axis in front of aperture

Comparing simulation to measurement shows good agreement between our model and the device. We can say this with confidence because we obtain a good match at two distinct observation parameters.

Clearly visible are peaks and troughs, which indicate coupled resonances. The peak at 380 Hz most likely belongs to the resonance, which was at $f_s = 625$ Hz in vacuum (figure 7). The strong alteration to low frequency is an indication of mass-load. The additional mass is caused by acoustic elements such as the phase-plug and the radiation impedance. The deviation at very high frequency of the sound-pressure curves seems to be caused by bending-vibration of the membrane, which is not taken into account (see below).

Acoustic Coupling - Refined Model

However, there is still a difference between the simulated and the measured curves. It seems that we need even more mass to take into account for our model. Candidates for acoustic mass are structures were air is moved, preferably around bends and corners. An additional mass can be found at the voice-coil-path which connects the acoustic field generated by the suspension and the one generated by the main membrane. The refined model is shown in figure 13.

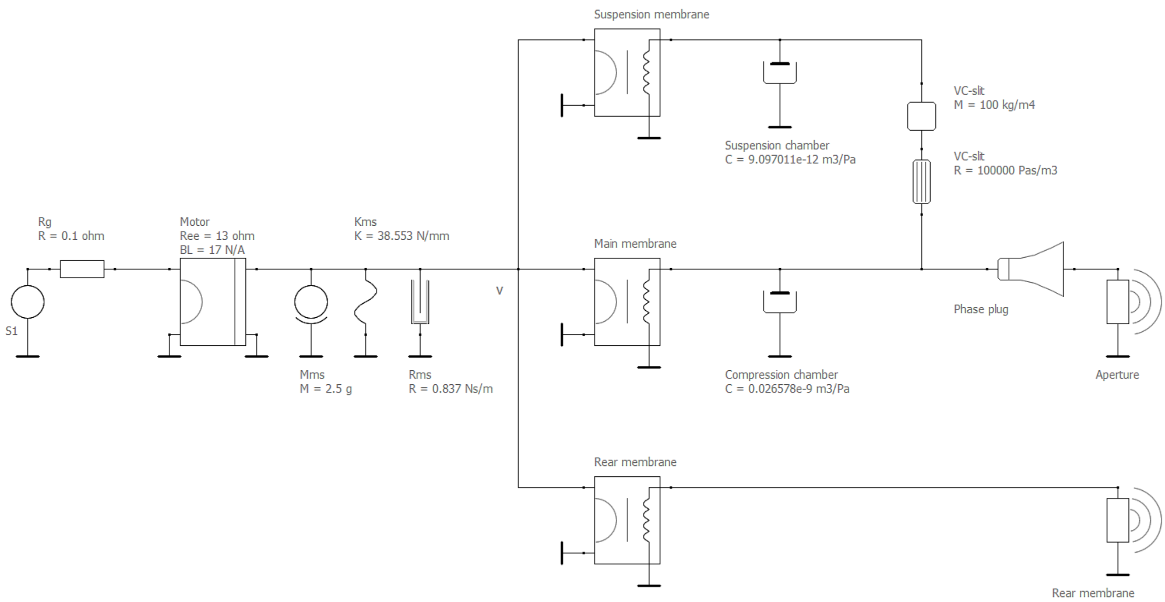

Figure 13 - Electro-mechano-acoustical network - refined model

In the refined model of the compression driver we model also the radiating area of the suspension. It is excited by the same velocity as the main membrane because the suspension is attached to the voice-coil-former, too. On the acoustic side of the suspension-membrane we can identify a spatial point were components share the same pressure-potential. This point is underneath the suspension. To this potential there is another acoustic compliance attached, which is caused by the suspension chamber with volume V_1 as shown in figure 3. This compliance can be calculated with the help of equation 7.

The voice coil-path was initially modeled with the help of another waveguide with transfer-functions given by equation 8. However, experiments yield the best curve-fit by using a series-connection of a simple acoustic mass and resistance component.

The output of the voice-coil slit is at the side of the compression chamber. Noting that the wavelength is larger than the radius of the membrane we assume the same sound pressure as inside the chamber. This is, of course, an approximation and may need further investigation. Simulation results are displayed in figure 14.

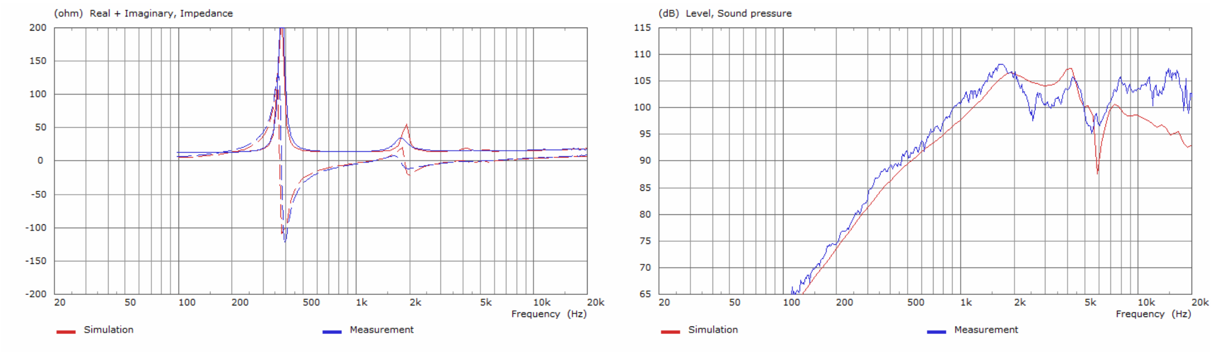

Figure 14 - Refined model: Electric driving point impedance and sound pressure level at 180 mm on-axis in front of aperture

The resonance peak is now properly aligned as the impedance curves of figure 14-left demonstrate. The curves of the acoustic observation are close. Why there are still deviations between curves of simulation and measurement is unclear at this stage.

Figure 12 and figure 14 both display a measured sound pressure curve of high level at the upper frequency band between 10 kHz and 20 kHz. This would be impossible to achieve with only the rigid body motion of the membrane. We suspect bending vibration at high frequency. In order to check this the velocity distribution was measured with the help of a Doppler laser. Figure 15 shows measurements taken at various points along the membrane.

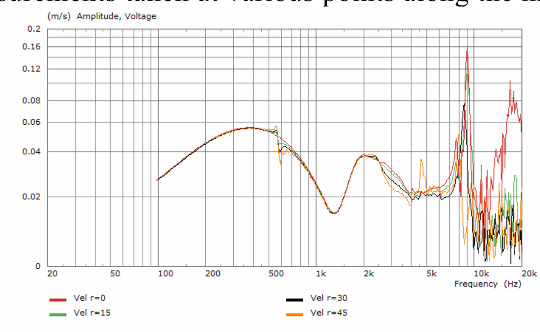

Figure 15 - Velocity distribution of main membrane Center at r = 0. Voice-coil-former at r = 45 mm

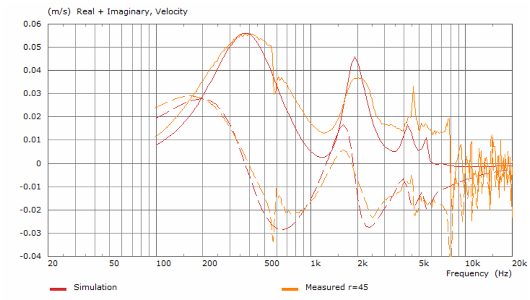

Figure 16 - Refined model: Mechanic velocity close to the voice-coil former

Bending-vibration starts at 7-8 kHz as indicated by fine ripples. The amplitude of the velocity of the rigid body mode should decrease by 6 dB/oct in this frequency range. Superimposed is the bending vibration. The highest amplitude was measured at the center (red curve).

The curve which we measured at the voice-coil former (r = 45 mm) should be close to the rigid body velocity. Hence, we can compare this measurement to the mechanical velocity v of our model as shown in figure 16.

The simulation shows resonances at peaks of the measurement curve. Deviations are likely to be caused by missing damping in the model. At high frequencies, residuals of bending vibration are clearly visible in the measurement curve.

We can normalize all velocity measurements to the curve taken at the voice coil former. This would give us a regulating filter which allows to weight the driving-source. Figure 17 shows the regulated sound pressure response and the applied amplitude of the filter (green).

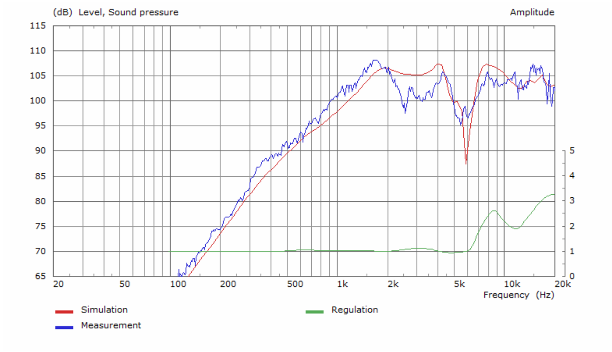

Figure 17 - Refine model. Weighted by normalized velocity distribution
Sound pressure level at 180 mm on-axis in front of aperture

CONCLUSIONS

This paper reports on the process of setting up a model with the help of lumped elements for the interior acoustics and boundary element method for external radiation. The device to be modeled is a typical compression driver. The modeling demonstrates the complexity of the device, especially at the acoustical domain. The advantage of lumped element modeling is the speed of calculation and à priory knowledge applied. The disadvantage is the reduced accuracy due to the limitation to rigid body motion and to simple wave mechanics.

The paper provides all ingredients in order to set up the lumped element analysis, using for example open *SciLab* or *Octave* calculation software. Alternatively, one may use dedicated software packages such as *Akabak*.

Further investigations, which study different radiation loads and specialized lumped elements, are under way.

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