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Comparison between measurement and Boundary Element Modelization of subwoofers

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ABSTRACT

At very low frequency, even large anechoic chambers can not be used to measure subwoofers accurately. A solution consists in using the Field Separation Method. This technique allows subtracting the field reflected by the measurement room walls to the measured field, thus recovering the acoustic pressure that would have been radiated under free field conditions. In this work, the Field Separation Method is used to measure two subwoofer prototypes. Results are compared to the ones given by a boundary element modelization of the subwoofers. Input velocities required for the modeling are measured by using a laser Doppler vibrometer. Comparisons are performed on the following quantities: on-axis pressure and directivity. Discrepancies between results obtained by these two methods are discussed and explained when possible.

1. INTRODUCTION

Subwoofer measurement is very tricky to perform. The standardized measurement methods [1] require free field conditions which are very hard to achieve at very low frequencies. In fact, even the largest anechoic chambers exhibit wall reflections at such frequencies which modify the measured acoustical data. As an example, a 400 m³ anechoic room generally has a cut-off frequency of about 70 Hz which is too high for measuring subwoofers accurately. Over

the past years, several methods have been developed to overcome this problem. Outdoor measurement can give reliable results. However, quiet places with sufficiently distant buildings are not so easy to get. Moreover, excessive rain or wind can postpone measurements for a few days. For indoor techniques, one can use an apodization function to remove reflections from walls [2, 3, 4, 5]. However, this last method cannot be used with ordinary subwoofers whose impulse response can last up to 100 ms, thus

requiring walls distant from 17 m. An alternative solution consists in computing the source far field response from measurements performed in the near field [6, 7], on the loudspeaker membrane [8] or in the loudspeaker box [9]. The free field response can also be simulated from an electroacoustic model when an accurate identification of the system parameters has been performed [10]. These extrapolation techniques require the measurement or modelization of all potential acoustic sources (loudspeaker, passive radiator, vent) to get accurate results. Nevertheless, when the loudspeaker box radiates significant acoustic energy, they cannot take this effect into account unless performing vibration measurements over the full box. This step can be very long and tedious.

An interesting method, first proposed by Weinreich *et al.* [11] has been applied to the measurement of subwoofers [12] and seemed to have given reliable results. This technique called field separation method (FSM) allows the separation of the field radiated by the tested source from the field reflected by the walls of the testing room. It requires the measurement of both acoustic pressure and velocity fields on a closed surface surrounding the source. Results obtained by FSM agree well with the one given by a boundary element method (BEM) modeling of the subwoofer for the on-axis pressure [13]. For this BEM computing, the membrane velocity was supposed to be uniform and has been measured at the center of the dust cap by an accelerometer, while the loudspeaker box was supposed to be perfectly rigid. However some discrepancies (about 1 or 2 dB) occurred at the rear side of the subwoofer. These discrepancies may come from enclosure vibrations. The aim of the work presented here is to take enclosure behaviour into account. Thus, normal velocities on both loudspeaker's membrane and wooden box will be measured by a laser Doppler vibrometer (LDV). Then, a BEM modeling with the measured velocity will be performed and compared to results given by FSM. Finally, directivity curves will be shown.

2. FIELD SEPARATION METHOD

As the theory of FSM has already been described in Ref. [12, 13]; therefore only a brief summary will be given here. The measured pressure field p^m in the testing room is given by

$$p^m(\mathbf{r}) = p^f(\mathbf{r}) + p^i(\mathbf{r}) + p^s(\mathbf{r}) \quad (1)$$

where p^f is the pressure that the tested source would have been radiated under free field conditions, p^i is the ingoing field, i.e., the pressure reflected by the walls of testing room, and p^s is the ingoing then scattered pressure by the tested source. At very low frequency, the wavelength is much larger than the dimensions of the subwoofer, thus, p^s can be neglected. Let us suppose that the pressure $p(\mathbf{s})$ and the velocity $v(\mathbf{s})$ are measured on a closed surface S encompassing the source. By using an integral formulation of the problem with a $e^{-i\omega t}$ dependence convention, p^f is given for $r > s$ by

$$p^f(\mathbf{r}) = \int_S [p(\mathbf{s}) \partial_{\mathbf{n}_S} G(\mathbf{r}, \mathbf{s}) - i\omega \rho_0 v(\mathbf{s}) G(\mathbf{r}, \mathbf{s})] dS \quad (2)$$

where ω is the pulsation, $G(\mathbf{r}, \mathbf{s})$ is the free space Green's function and \mathbf{n}_S denotes the normal to the boundary S . When measured data are collected on a sphere surrounding the source, $p(\mathbf{s})$, $v(\mathbf{s})$ and $G(\mathbf{r}, \mathbf{s})$ can conveniently be expanded on normalized spherical harmonics. Then, the pressure field p_f is easily obtained by using orthogonal properties of spherical harmonics. For practical purpose, measurements are performed on a half-sphere which base plane is the ground of the testing room. Assuming perfectly rigid ground conditions, all expansions can be performed on even spherical harmonics only. Please note that considering a sphere of radius $a = 0.5$ m surrounding the source and its image relative to the ground, the minimum order N of the expansion for propagative waves is given by $ka \simeq N$ where k is the wave vector. To use FSM up to 500 Hz involves at least $N > 4.6 = 5$. Spherical harmonic expansions performed in this paper have been calculated with $N = 5$ which should be sufficient. Note that, with this set-up, the effect of the rigid ground will remain. To recover the free field response, the subwoofer has to be hanged and measurements have to be performed on a full sphere surrounding the tested source.

Two subwoofer prototypes have been tested: a closed box system (Sub1) and an active passive one (Sub2). A schematic view of the two prototypes is given in Fig. 1. A Peerless 269 SWR 51 XLS loudspeaker is mounted on each prototype. Sub2 has an additional 10" Passive Radiator. All enclosures are cubes with edge length of 0.395 m and are made of medium density fiberboard (MDF) wood. To high-

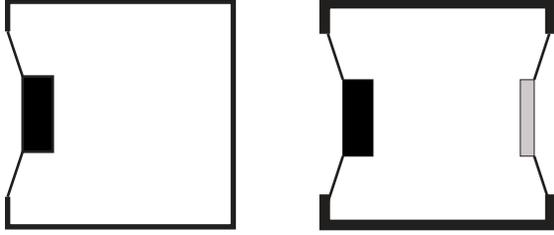


Fig. 1: Schematic view of the tested subwoofers: Sub1 (left) and Sub2 (right).

light the impact of the enclosure vibration, Sub1 has been built with thin walls (1 cm) while Sub2 has thick walls (3 cm). Electrical signals are not filtered, however, an additional 4.8Ω series resistance is inserted between the amplifier and the loudspeaker.

The tested subwoofer is put on the rigid ground of a semi anechoic chamber (Fig. 2). As the ground is not perfectly plane, the subwoofer stands on three thin pieces of carpet to obtain an isostatic weight distribution. The loudspeaker is driven by a band limited white noise (10 Hz-500 Hz) test signal. The amplifier used is a RANE MA 6S model. A p-p probe, calibrated in amplitude and phase, is moved on a hemispherical surface by an automatized positioning system. The distance between the two microphones is 10 cm. Acoustic data on the medium surface S can then be computed from the two adjacent measurement half-spheres using their mean value for the pressure and approximation of Euler's equation by finite difference for velocity. The mesh used in this experimentation is made of 36 points around the source on a half-sphere of radius 0.65 m. The points are located on the intersections of parallels and meridians, with $\Delta\varphi = \Delta\theta = 30^\circ$. Then, FSM is applied to calculate p^f .

Results are plotted in Fig. 3 for Sub1. One can see that the reflections from testing room walls are removed from the frequency response curves. Thus, curves obtained by FSM are smoother and have the typical shape of a closed box frequency response, especially for the rear position ($0.65, 15^\circ, 180^\circ$). However, several oscillations remains particularly at higher frequencies. The next part of this work will focus on the enclosure vibrations and will try to figure out if these vibrations can explain part of the

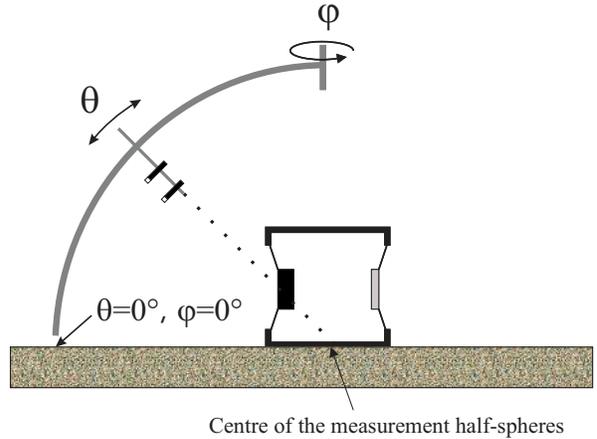


Fig. 2: Geometry of the measurement set-up.

remaining fluctuations.

3. BEM MODELING

3.1. Velocity measurement

The enclosure and membrane normal velocities have been measured using a scanning Laser Doppler Vibrometer (Polytec OFV 056 / OFV 3001 S) at each point of a 15×15 grid on each of the five visible subwoofer faces. A picture of the experimental set-up is given in Fig. 4. For each point, the frequency response function between the velocity and the loudspeaker voltage has been computed, taking approximately 22 minutes per subwoofer face to achieve the 1.25 Hz chosen frequency resolution. Since the vibrometer measures only the velocity component parallel to the laser beam, angular geometrical corrections have been applied to the measurement results to correct them, when it was not possible to align the vibrometer head with the mean normal of the subwoofer faces (especially for the upper face, see Fig. 4). Depending on the frequency band, two different behaviors are obtained:

- Rigid body motion up to about 200 Hz for Sub1 (see Fig. 5) and up to about 300 Hz for Sub2. The subwoofer remains rigid and moves on its three elastic holding pieces.
- At higher frequencies, elastic vibrations of enclosure walls appear (see Fig. 6).

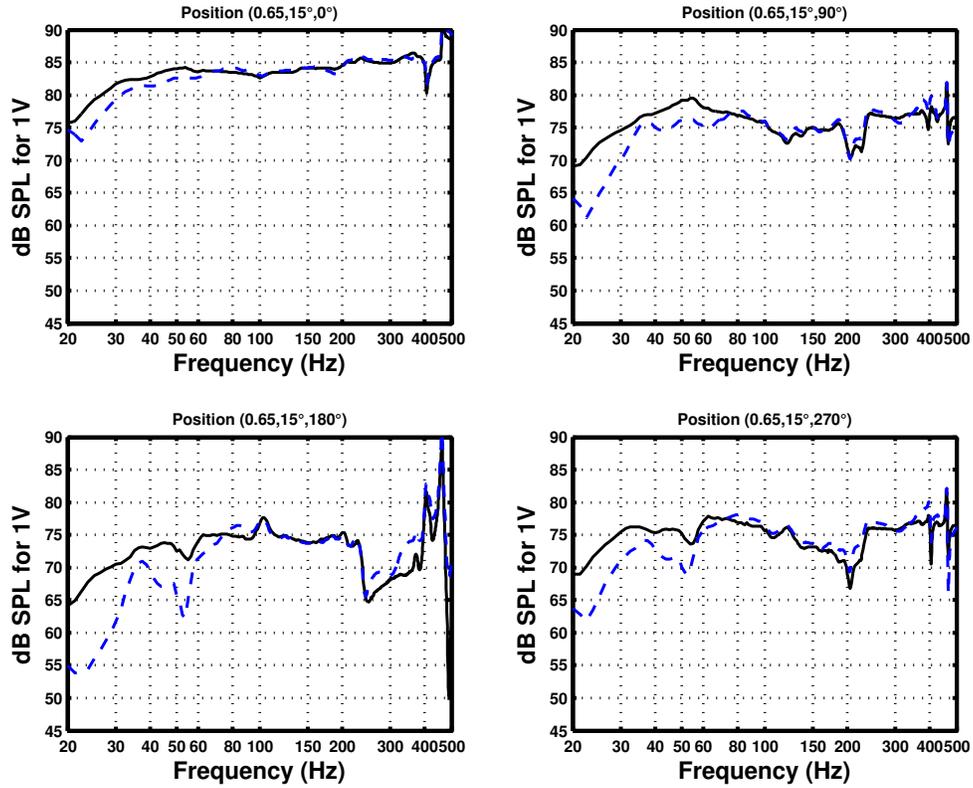


Fig. 3: Pressure levels (dB SPL) for 1 V at various positions $(0.65, \theta, \varphi)$ on S for Sub1. Dashed lines: p^m , solid lines: p^f .

Mean velocities for 1 V are shown in Fig. 7 for the loudspeaker face and the face opposite to it. Note that below 350 Hz, enclosure mean velocity is much lower than the membrane velocity. However, the enclosure surface over speaker surface ratio (26.5 dB) is in favor of the enclosure. At 465 Hz, mean velocities on these two sides are very close. This result shows that the impact of the enclosure vibrations can be very significant for Sub1. This point will be discussed in the next section.

3.2. Influence of enclosure vibrations

After velocities have been measured, a BEM modeling of the subwoofer has been performed. The mesh is made with 1009 points decomposed onto quadrangular elements with 4 nodes. Two cases have been processed, the first one with enclosure velocities equal to zero and the second one using all measured velocities. Results are plotted in Fig. 8. One

can see that the effect of the enclosure vibrations can be very strong especially for measurement points near the face opposite to the loudspeaker (microphones are very close from the enclosure and more distant from the membrane). On the other hand, on-axis results are almost unaffected by enclosure vibrations. Another interesting result is that rigid body motions are visible on the pressure level curves thus highlighting the importance of a good subwoofer stability which can be achieved with commonly used spikes or rubber pods.

Fig. 9 shows pressure level at $(0.65, 15^\circ, 180^\circ)$ for Sub2 which has thicker enclosure walls. One can see that the impact of enclosure vibrations is lower than for sub1. However, effect of rigid body motion is still visible in the 100-200 Hz frequency band and effect of enclosure walls modes is in evidence between 200 and 300 Hz.

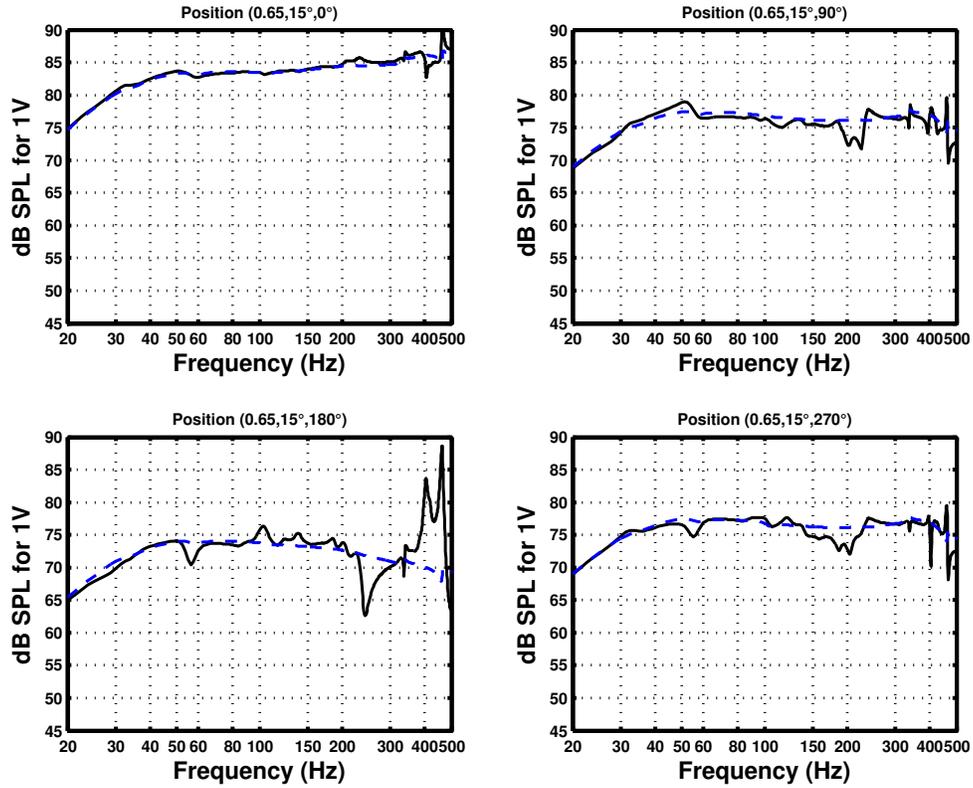


Fig. 8: Pressure levels (dB SPL) for 1 V at various positions $(0.65, \theta, \varphi)$ on S for Sub1. Dashed lines: processed from membrane velocities only, solid lines: processed from enclosure and membrane velocities.

4. COMPARISON OF BEM AND FSM RESULTS

4.1. Near-field results

Fig. 10 shows a comparison between FSM and BEM results for Sub1. One can see that curves with enclosure and membrane velocities have a better agreement with FSM results than curves with membrane velocities only. The agreement is very good when elastic modes of the enclosure are involved. However, for rigid enclosure motion, even if BEM curves have the tendency to come closer to FSM results, discrepancies in the range of 1 or 2 dB remains. This result may be partially explained by carpet support tamping during the 4 hours of the measurement process (about 3 hours for LDV and about 1 hour for FSM). This assumption should be verified by new measurements with subwoofer mounted on

spikes. Anyway, results are encouraging and shows that FSM can measure complex sources with several radiation surfaces.

4.2. Directivity

Sub1 directivity at 250 Hz and 10 m is plotted in Fig. 11. One can see that BEM and FSM (with membrane and enclosure velocities) results are very close. The effect of enclosure vibrations are very strong at 180° : pressure level is 5 dB lower than the value obtained with membrane velocity only. Figure 12 shows directivity curves at 10 m and at 450 Hz for Sub1. Here again, FSM and BEM with all velocities results agree well. An interesting outcome is that the radiated pressure at 180° is higher than the on-axis pressure. This result is not predicted if enclosure vibrations are not taken into account.

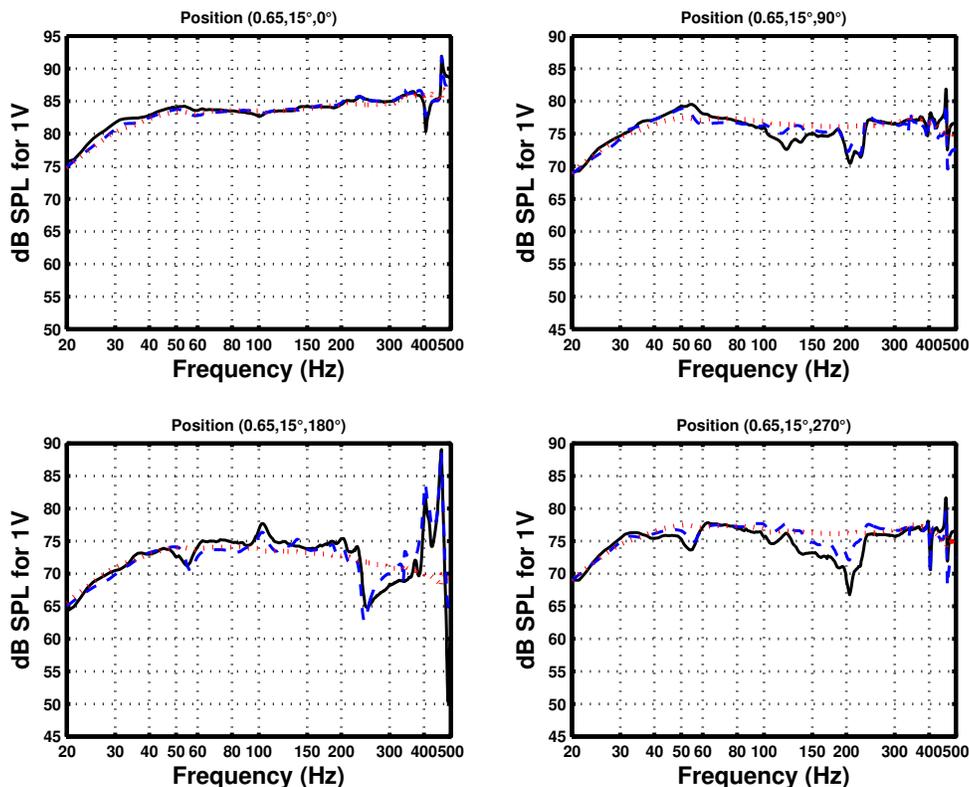


Fig. 10: Pressure levels (dB SPL) for 1 V at various positions $(0.65, \theta, \varphi)$ on S for Sub1. **Dotted lines:** BEM processed from membrane velocities only, **dashed lines:** BEM processed from membrane and enclosure velocities, **solid lines:** FSM.

5. CONCLUSION

In this paper, FSM has been applied to the measurement of two subwoofer prototypes. Results have been compared to the ones given by a BEM modeling. The agreement between these two methods is good when velocities of both loudspeaker membrane and enclosure are used. This work showed that FSM can be used to measure more complex sources than a subwoofer with rigid enclosure. Concerning the practical implementation of FSM, this result is encouraging, as measurements have been performed with relatively cheap microphones (Sennheiser KE4). Nevertheless, small discrepancies remains when rigid body motion of the subwoofer is involved. Future work will focus on this particular point. Result on directivities showed that, even at 10 m, effect of enclosure vibrations is still significant

when dealing with thin enclosures. Thus, special attention to the enclosure building should be payed to avoid such outcome.

6. ACKNOWLEDGEMENTS

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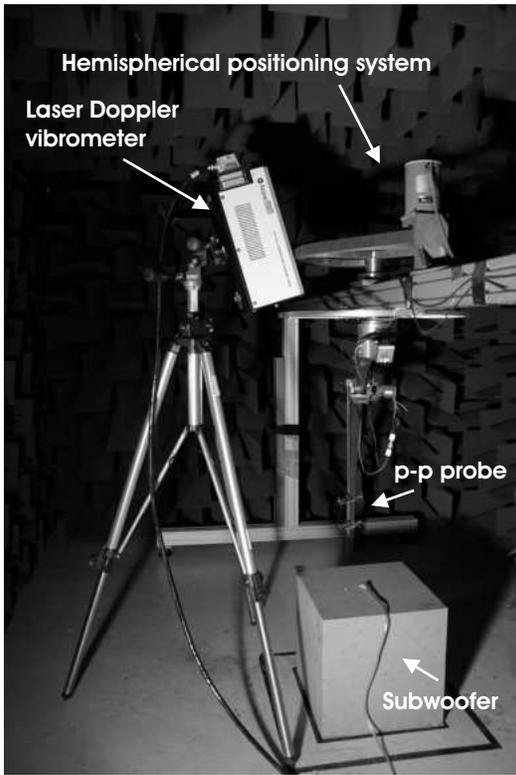


Fig. 4: Experimental set-up picture showing both Laser Doppler Vibrometer and acoustic measurement system.

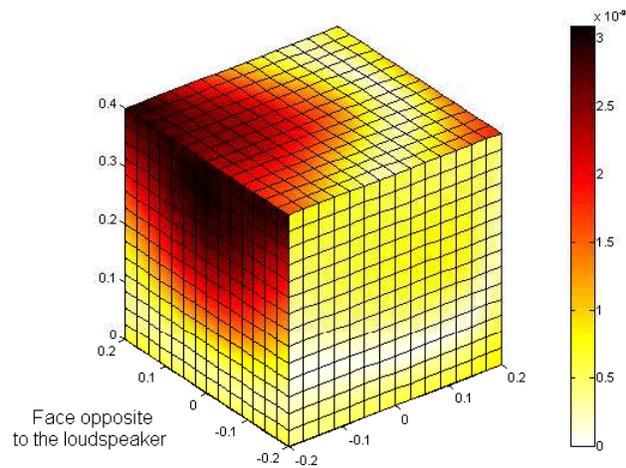


Fig. 5: Normal velocity map at 103 Hz for Sub1.

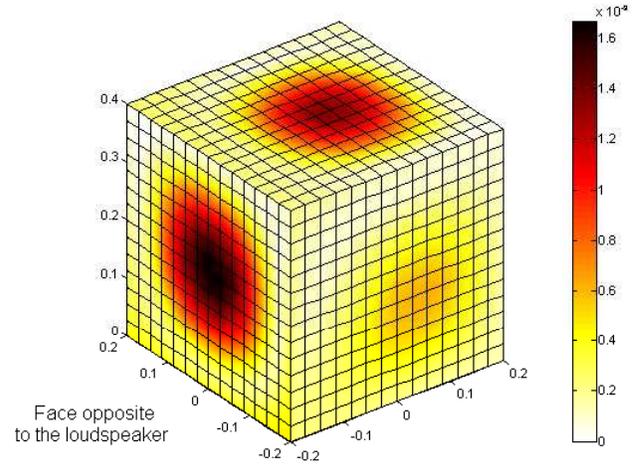


Fig. 6: Normal velocity map at 238 Hz for Sub1.

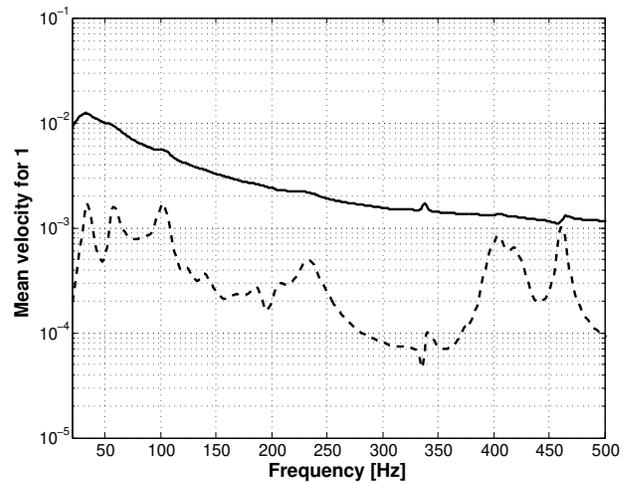


Fig. 7: Mean velocities for 1 V on the loudspeaker face (solid line) and on the face opposite it (Dashed line).

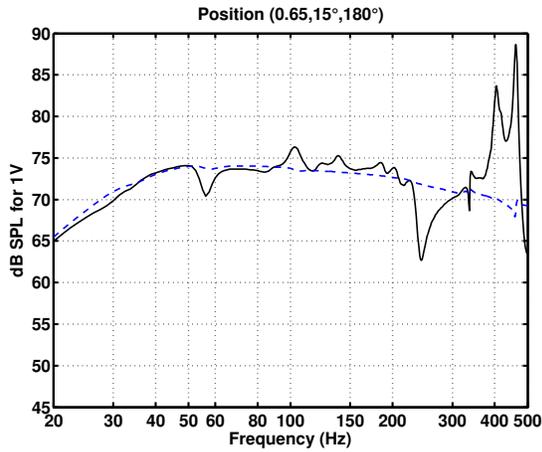


Fig. 9: Pressure levels (dB SPL) for 1 V at (0.65, 15°, 180°) for Sub2. **Dashed lines:** processed from membrane velocities only, **solid lines:** processed from enclosure and membrane velocities.

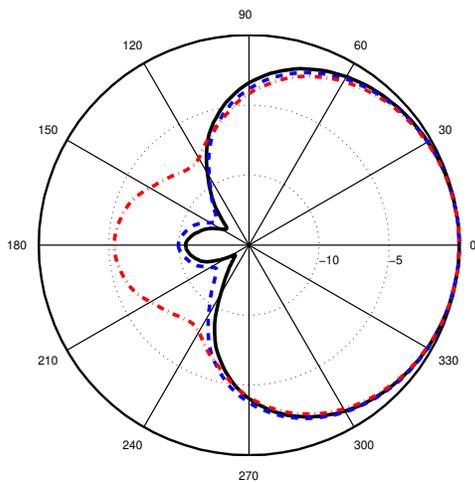


Fig. 11: Directivity at 10 m and at 250 Hz for Sub1 (dB). **Dashed/dotted lines:** BEM processed from membrane velocities only, **dashed lines:** BEM processed from membrane and enclosure velocities, **solid lines:** FSM.

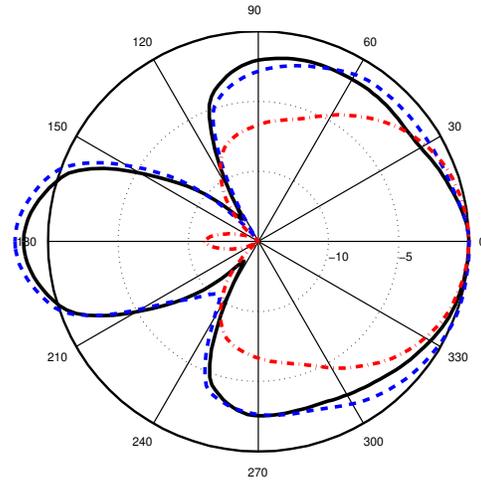


Fig. 12: Directivity at 10 m and at 450 Hz for Sub1 (dB). **Dashed/dotted lines:** BEM processed from membrane velocities only, **dashed lines:** BEM processed from membrane and enclosure velocities, **solid lines:** FSM.

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