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Multi-source low frequency room simulation using finite difference time domain approximations

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ABSTRACT

Sound level distribution generated by loudspeakers placed in a room can be simulated using numerical methods. The purpose of this paper is to present an application based on finite-difference time-domain approximations (FDTD) for the study of low frequencies in audio reproduction such as ordinary stereo to multi-channel surround setups. A rectangular room is simulated by using a discrete model in time and space. This technique has been used extensively and gives good performance at low frequencies. The impulse response can be obtained in addition to the sound level distribution. Simulation of multiple loudspeakers in a room can be achieved to evaluate and visualize their coupling with the room. A high frequency resolution can be obtained for auralization purpose.

1. INTRODUCTION

When a loudspeaker is placed in a rectangular room a number of problems arise. Modification of the response of the loudspeaker at the listening position occurs due to the strong influence of the room and the position of the loudspeaker. The combination loudspeaker-room acts as a coupled system where the room typically dominate by its distinct normal modes. When placing more than one loudspeaker in the room some of the attenuation produced by the room is less sever but still the room has a strong influence at the different listening positions.

In order to deal with this problem equalization techniques have been investigated by several authors for example in [1] or [4], [5]. In connection to that a robust tool to simulate the low frequency behavior of multiple sound sources placed in rectangular rooms is needed. During the last years two main approximations using numerical methods has been developed which are Ray tracing and Image source method [2] [3]. Such methods are no longer adequate to simulate frequencies below 100Hz because they are based on geometrical acoustic approximations where the wavelength is smaller than the room dimensions.

The finite-difference time-domain (FDTD) method has been used with great success to model electromagnetic problems. In acoustics FDTD has shown good performance to approximate the low frequency room behavior. This method has been well described in early studies by Botteldooren [6]. By using this approximations the calculations are made directly in the time domain. The sound wave equation is discretized both in time and space and the inclusion of multiple sound sources in the room is possible since the finite time and space is always available.

Recently other methods as finite-element method (FEM) and boundary-element method (BEM) have been used extensively to simulate enclosures. This approximations work only in the frequency domain, direct translation to a direct time domain is rather difficult, some of this work can be seen in [8].

In this paper the FDTD method has been chosen to simulate a rectangular room exited by multiple sound sources. The estimation of the sound level distribution at low frequencies is calculated. The inclusion of loudspeakers assuming omni-directional compact sources is implemented inside the room. Since the particle velocity is always available, sound power and intensity can be estimated for equalization purposes. Moreover a band limited impulse response of the room can be derived.

2. SIMULATION OF SOUND SOURCES IN A ROOM USING FDTD

In this section a description of FDTD method is presented as well as important aspects for the simulation process which are stability, the boundary conditions and the modelling of the sound sources.

2.1. Method

Typically the FDTD method utilizes two coupled first order differential equations. Since this method works in the time domain it computes the derivative and linearized form of these two equations in the time domain. This is done by means of central finite difference [9].

2.1.1. Discretization of The Wave Equation

The first equation is the linear inviscid force equation valid for acoustic processes of small amplitude where the sound pressure and the particle velocity are related as:

$$\nabla p = -\rho \frac{\partial \vec{u}}{\partial t} \tag{1}$$

where ρ is the density of the transmission media in $\mathrm{kg/m^3}$.

The second equation is the linear continuity equation

$$\nabla \cdot \vec{u} = -\frac{1}{c^2 \rho} \frac{\partial p}{\partial t} \tag{2}$$

where c is the wave propagation speed in the media [10].

The typical formulation of FDTD approximation uses a Cartesian staggered grid [11] in which pressure and particle velocity are the unknown quantities. The acoustical pressure is determined at the grid points $(x\delta x, y\delta y, z\delta z)$ at time $t = \delta t$ in this paper $\delta x = \delta y = \delta z = h$ that is the spatial discretization step and $\delta t = k$ that is the time step.

Both equations can be sampled in time and space using the sampling rates $\frac{1}{k}$ Hz and $\frac{1}{h} m^{-1}$. The discretization is done by means of finding the central point between two neighbour time/space points [7].

After the derivation in time and space of the force equation, Eq. (1) the three components of the particle velocity are determined at positions:

$$\begin{array}{l} u^x_{(x \pm \frac{h}{2}, yh, zh)} \\ u^y_{(xh, y \pm \frac{h}{2}, zh)} \\ u^z_{(xh, yh, z \pm \frac{h}{2})} \end{array} \tag{3}$$

and at intermediate time $t = (t + \frac{1}{2})k$ by the following equations:

$$
\begin{aligned}
u^x_{x+\frac{h}{2},y,z}(t+\frac{k}{2}) &= u^x_{x+\frac{h}{2},y,z}(t-\frac{k}{2}) - \frac{k}{h\rho} \\
&\quad \times \Big[p_{x+h,y,z}(t) - p_{x,y,z}(t)\Big], \\
u^y_{x,y+\frac{h}{2},z}(t+\frac{k}{2}) &= u^y_{x,y+\frac{h}{2},z}(t-\frac{k}{2}) - \frac{k}{h\rho} \\
&\quad \times \Big[p_{x,y+h,z}(t) - p_{x,y,z}(t)\Big], \\
u^z_{x,y,z+\frac{h}{2}}(t+\frac{k}{2}) &= u^z_{x,y,z+\frac{h}{2}}(t-\frac{k}{2}) - \frac{k}{h\rho} \\
&\quad \times \Big[p_{x,y,z+h}(t) - p_{x,y,z}(t)\Big],
\end{aligned} \tag{4}
$$

Similarly from the continuity equation, Eq. (2) the acoustic pressure can be derived in time an space by:

$$
\begin{aligned}
p_{x,y,z}(t+k) &= p_{x,y,z}(t) \\
&- \frac{c^2\rho k}{h}\Big[u^x_{x+\frac{h}{2},y,z}(t+\frac{k}{2}) - u^x_{x-\frac{h}{2},y,z}(t+\frac{k}{2})\Big] \\
&- \frac{c^2\rho k}{h}\Big[u^y_{x,y+\frac{h}{2},z}(t+\frac{k}{2}) - u^y_{x,y-\frac{h}{2},z}(t+\frac{k}{2})\Big] \\
&- \frac{c^2\rho k}{h}\Big[u^z_{x,y,z+\frac{h}{2}}(t+\frac{k}{2}) - u^z_{x,y,z-\frac{h}{2}}(t+\frac{k}{2})\Big],
\end{aligned} \tag{5}
$$

This are the set of equations that are used to calculate particle velocity and sound pressure in an alternate manner.

In Fig.1 an example of an enclosure can be seen where the layout of the grid for the calculation of the components of the particle velocity and acoustic pressure points in two dimensions is shown. The circular points represent sound pressure while squares are particle velocity component points in x direction and stars are particle velocity components in y direction. As it can be observed there are no pressure points or particle velocity points at the boundaries. In this manner the components of the particle velocity in for example x direction are calculated at intermediate pressure points as well as at intermediate time steps.

The advantage of using this grid is that is easy to define the boundaries and it only requires that two values of the acoustic pressure and particle velocity are stored in each grid cell. It has to be mention that it only works for rectangular objects.

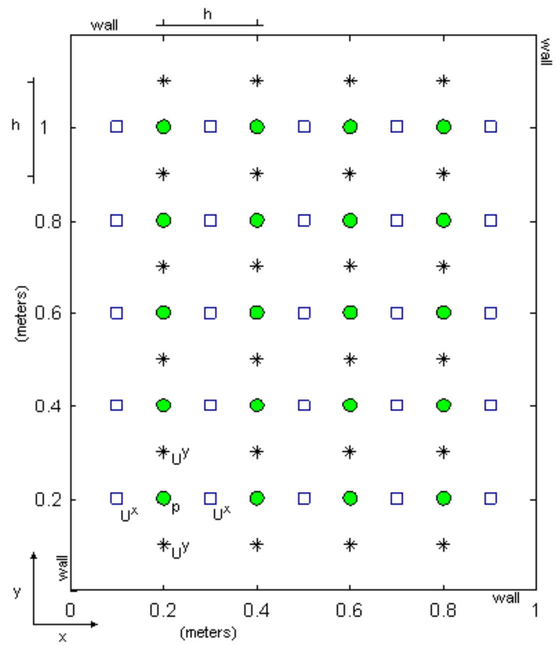

Fig. 1: Example of a calculation grid in a 1 x 1.20m enclosure.

2.1.2. Cell Size

A fundamental constraint for the simulation method is the choice of the size cell. The frequency range of interest before aliasing and the accurate grid dispersion is given by the cell size. The cell size must be much less than the smallest wave length for which accurate results are needed. Reasonable results can be achieved by using from five to ten cells per wavelength [9]. In this paper a cell size of 10 cm has been chosen since this cell size corresponds to $\frac{1}{5}\lambda$ it is expected to have an accurate result below 600Hz.

2.1.3. Stability

After the cell size have been chosen, the time step has to be set. In order to have an accurate wave propagation and to minimized grid dispersion errors the relation expressed in equation 6 has to be held more generally for a three dimensional rectangular grid [9].

$$c\delta t \leq 1/\sqrt{\frac{1}{\delta x^2}+\frac{1}{\delta y^2}+\frac{1}{\delta z^2}} \tag{6}$$

In this paper the sampling frequency was decided to be $fs = 8$kHz, and the time step $k = 1/fs$. Nevertheless the simulator can be set to find the minimum time step before it gets unstable.

2.2. Boundary Conditions

Taking the example of Fig.1 and assuming a right hand rigid wall at the boundary of the room the component of the particle velocity in the x direction cant be calculated with Eq. (4) because the term $p_{x+h,y,z}(t)$ is unknown. To solve this problem an asymmetric finite-difference approximation for the space derivative is implemented [11].

Since the component of the particle velocity in the x direction u^x represents the perpendicular part of the particle velocity to the wall, it is assumed that the acoustic pressure p at the wall can be expressed by the product of the component of the particle velocity in the x direction u^x and the characteristic impedance Z of that wall [7]. This manner an absorption coefficient of the walls can be introduced to calculate Z:

$$Z = \rho c \frac{1+\sqrt{1-\alpha}}{1-\sqrt{1-\alpha}} \tag{7}$$

where α is the absorption coefficient of the wall.
After this assumptions the new version of (4) for the nearest components of the particle velocity to the walls is introduced and for $u^x_{0.8+\frac{h}{2},y,z}$ in the example in Fig.1 the boundary equation is defined as:

$$\begin{aligned} u^x_{0.8+\frac{h}{2},y,z}(t+\frac{k}{2}) = & \frac{\frac{\rho h}{k}-Z}{\frac{\rho h}{k}+Z} u^x_{0.8+\frac{h}{2},y,z} \\ & -\frac{2}{\frac{\rho h}{k}+Z} p_{x,y,z}(t) \end{aligned} \tag{8}$$

2.3. Sound Source Model

A typical closed-box loudspeaker can be modelled as a point source with volume velocity function of time occupying, one point inside the room. Since for low frequencies where the wavelength of sound in the air is much longer than the physical dimensions of the loudspeaker it propagates the sound in spherical waves radiating outwards uniformly in all directions [12].

When the loudspeaker is driven by a sinusoidal signal it is modelled according to

$$p(r,t) = \frac{\rho}{4\pi r} A \cdot sin(\omega(t) - kr) \tag{9}$$

$$A = S_D \omega u \tag{10}$$

where A is the volume acceleration, u is the particle velocity, S_D is the effective area of the radiating piston, r is the distance from the sound source and ω is the angular frequency.

In this manner one or more than one point sources can be included in the model either in a pressure point or particle velocity using the volume velocity of the desired loudspeaker. It can be mention that the loudspeaker can be modelled as a membrane moving in a desire direction using some of the points of the components of the particle velocity.

3. EVALUATION OF THE SOUND FIELD IN A ROOM

3.0.1. The Test Room

For the purpose of this paper the standard listening room of the Department of Acoustics at Aalborg University in Denmark has been chosen to be simulated since this room has been well studied. The room has the following dimensions, length 7.80 m; width 4.12 m; height 2.77 m; the mean reverberation time T60 is 0.47 s. The floor is wooden and the walls are covered with special panels that can be removed or moved to different positions. The ceiling is curved in the corners covered with special plaster panels see Fig.9 from [13]. The panels from the walls have been removed as well as the carpet that normally covers most of the floor. The test room was measured and simulated early by Cherek and Langvad [14] as well as by Karup [7].

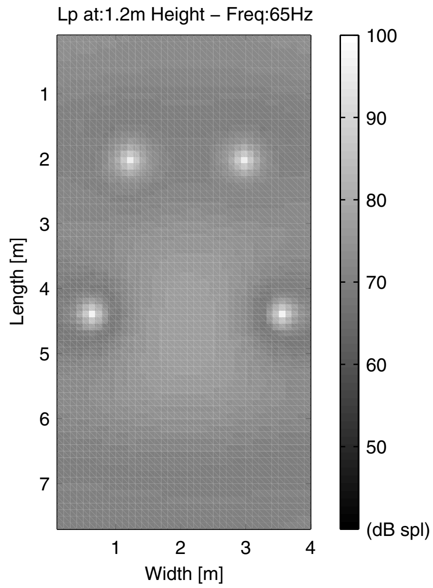

Fig. 2: Sound pressure level distribution resulting form the simulation of four loudspeakers reproducing a sinusoidal frequency of 65 Hz.

An horizontal layer of the room at a height of 1.2 m has been chosen to calculate the sound pressure distribution (SPL). It was decided to set the simulation time to 1 second taking in to account that the reverberation time of the test room simulated is less than 1 second.

In case of a simulation of a more reverberant room the simulation time should be grater than the mean reverberation time.

3.1. Sound Pressure Level Distribution

Since the root mean square (RMS) value of any signal is proportional to its energy content and therefore is one of the most important and most often used measures of amplitude it has been decided to calculate this value over the area of interest in the room. As it is mention in section 2.1.1 two time steps of the pressure points and particle velocity of the whole room are needed to determine the acoustic pressure in the grid positions. Equation (11) is used to calculate the RMS value of the pressure over the area of interest where T is the relevant period over which the averaging takes place and p is the instantaneous pressure [15].

$$p_{rms} = \sqrt{\frac{1}{T}\int_0^T p^2(t)\cdot dt} \tag{11}$$

For this calculation an extra matrix is loaded to be used as an accumulator of the result of the squared summation of the sound pressures for each time step.

From this pressure matrix the SPL distribution over the chosen layer in the room can be obtained, see Fig.2. The SPL distribution is calculated in the room according to:

$$Lp = 20Log_{10}\frac{p}{p_o} \tag{12}$$

where p is the sound pressure being computed and p_o is the reference sound pressure being 20 μ Pa.

3.2. Optimization of Used Memory

A huge amount of memory is needed if one wants to keep the time history of the instantaneous sound pressure all over the room [16]. To optimize the use of memory only two time steps of sound pressure and particle velocity all over the room are stored together with one extra matrix for the average sound pressure level. The averaged sound pressure level will only be calculated in the horizontal plane of interest. Nevertheless some other (virtual) microphones can be set up all over the room in order to calculate the time responses at any desired position in the enclosure.

3.3. Visualization in Time Domain

A very useful advantage of FDTD is the visualization aspect. Since it runs in the time domain the instantaneous sound pressure in the desired area of the room can be observed at any time.

An animation composed by indexed images of the instantaneous sound pressure along the simulation time can be set in the simulation program. In Fig.3 a sequence of images from the simulation of a sound source reproducing a Gaussian pulse can be seen. In Fig.3 the reflections from the walls are very clear but also some absorption can be observed.

3.4. Acquisition of Impulse Response

Two methods are considered in order to obtain a band limited impulse response of the room exited

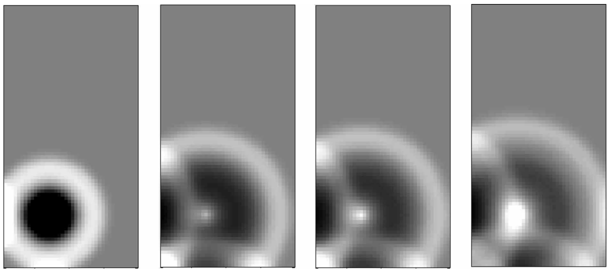

Fig. 3: Sequence of images of an spherical wave coming from a sound source close to the walls of the test room.

by a number of sound sources. The first one is performed by reproducing a finite length Gaussian pulse and record the impulse in the room at the desired position.

The second method is implemented by reproducing a maximum length sequence (MLS) and record the signal at the desired position in the room. In the next two sections both methods will be described.

3.4.1. Gaussian Pulse

The Gaussian pulse used in the simulation is defined by Eq. (13) and it is shown in Fig.4. This kind of pulse has the characteristic of having a limited flat frequency response. The cut off frequency is defined by σ in Eq. (14) where ω is the angular frequency. In this case ω represents the -3 dB frequency limit response of the sound source. Indeed it is an omni directional point source.

$$p_{x,y,z}(t) = \frac{A}{\sigma^2} sin(t - t_0) e^{\frac{-(t-t_0)^2}{\sigma^2}} \tag{13}$$

$$\sigma = \frac{2}{\omega} \tag{14}$$

In Fig.5 the recorded impulse response at position 1 together with the frequency response is presented. The test room was exited by one source at position (1.2m,2.0m,1.2m) refer to Fig.9 to see the microphone and loudspeaker position in the test room.

As it can be observed the influence of the room is severe and some of the room modes are revealed.

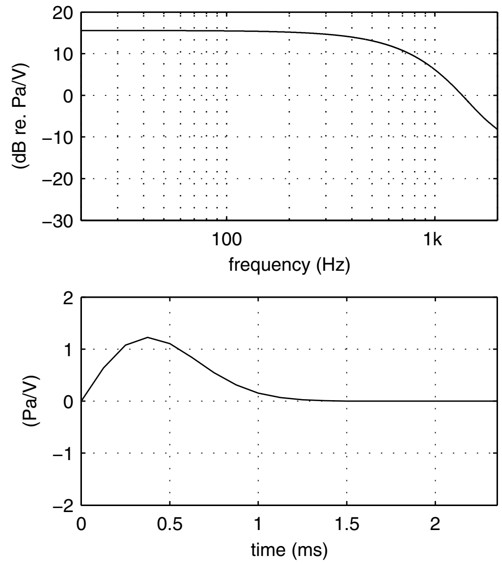

Fig. 4: Upper plot shows the frequency response of the Gaussian pulse, lower plot shows the Gaussian pulse in the time domain. The cut off frequency has been set to 600Hz.

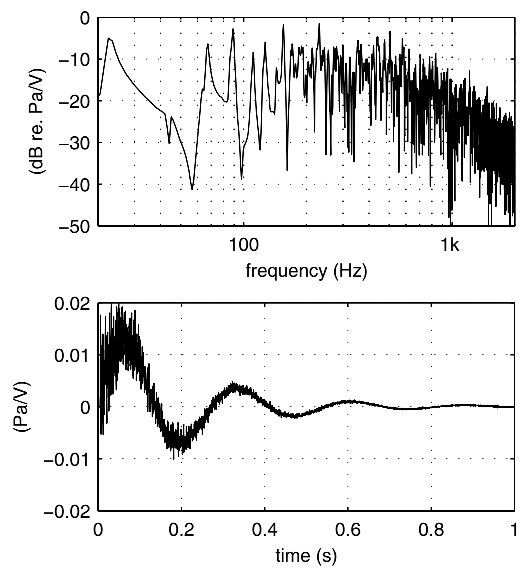

Fig. 5: Upper curve is the transfer function of the room plus the loudspeaker calculated at microphone position 1, lower plot is the impulse response, method: Gaussian pulse.

3.4.2. MLS method

As well as the Gaussian pulse method an MLS sequence is reproduced by the sound source. An MLS signal is very useful because it has high energy content and it is very suitable for different impulse lengths. It generates uniform probability density, its spectrum is absolutely flat, and the most important is that its periodic auto correlation is a unit sample sequence. The motivation to use the MLS method is that the MLS method unlike the Gaussian excitation has high energy content in all frequencies therefore the room can be excited properly.

The signal is implemented by generation of pseudo random numbers. The sampling frequency is set to 8 kHz. The length of the impulse response was set to $2^N - 1$ being chosen according to how reverberant is the room to simulate. The length of the input signal is two times the length of the MLS signal in order to stabilized the filter. The input signal is low pass filtered to avoid aliasing from the simulation itself. The cut off frequency is chosen according to the frequency range of interest. The MLS method has been extensively studied, details of the theory behind it can be found in [17] and [19] and other authors.

An anti aliasing filter at 2 kHz is implemented to filter the recorded signal. The cross correlation between the MLS input signal and the recorded sequence is calculated in order to obtain the impulse response. In Fig.6 the recorded impulse response and frequency response of the simulated room is shown. The room was exited by one sound source located at coordinates (1.2 m, 2.0 m, 1.2 m) see Fig.9.

3.4.3. Including a real loudspeaker impulse response

In order to get a more accurate result in the simulation the transfer function of a real loudspeaker can be introduced in the model. This procedure is valid just when the frequency range of interest is below 500Hz since a loudspeaker behaves almost omni directional within that range. The test loudspeaker (A) is a closed-box type with a volume of 12 litre and 35 cm height 23 cm width and 23.5 cm depth it has a 16.5 cm diameter woofer driver unit and a 1.9 cm, dome tweeter.

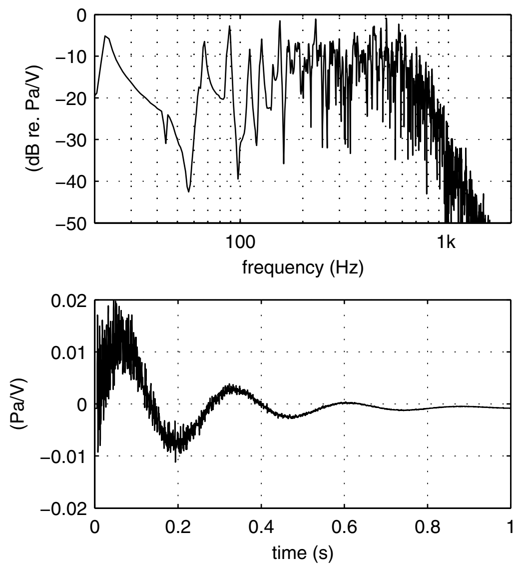

Fig. 6: Upper curve is the transfer function of the room plus loudspeaker calculated at microphone position 1, lower plot is the impulse response, method: MLS.

The test loudspeaker (A) has been measured in anechoic conditions in order to obtain two impulse responses to be tested in the simulator.

For the first measurement it has been decided to measure the near field impulse response at the membrane of the test loudspeaker see Fig.7, more details about near field measurements can be found in [18].

The second impulse response is an average in the frequency domain of measurements of the response of the loudspeaker in the horizontal plane and vertical plane at 1 m from the membrane with a resolution of 30 degrees. The magnitude of the averaged impulse response has been normalized with the near field measurement in the frequency domain in order to have the same gain as the near field measurement see Fig.7.

After obtaining the transfer functions of the loudspeaker they have been convolved with the MLS input signal, then low pass filtered and reproduced at the sound source position. The same procedure explained in section 3.4.2 has been applied to obtain the impulse response with an MLS signal, see Fig.10.

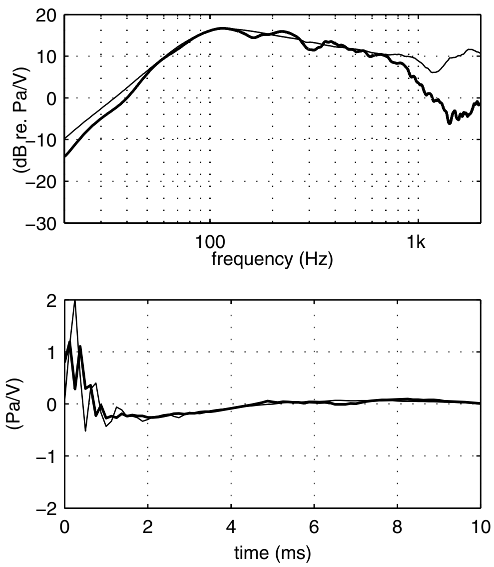

Fig. 7: Frequency response and time response of test loudspeaker (A), thin line is the near field measurement, thick line averaged in horizontal and vertical planes.

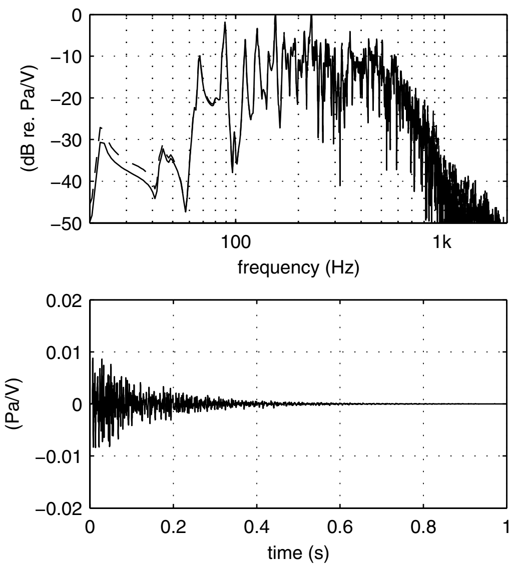

Fig. 8: Transfer function of the room exited by loudspeaker (A) calculated at position 1, continuous line near field impulse response is included, dashed line the averaged impulse response of the real loudspeaker is included. In the time response only the averaged impulse response is included, method: MLS.

In the lower plot of Fig.8 the impulse response recorded at position 1 is shown. The test room is exited by one sound source located at position (1.2m,2.0m,1.2m) including the loudspeaker transfer function of the near field measurement, in the upper plot in the same figure the frequency response is presented, in the same plot the doted line includes the averaged transfer function of the loudspeaker instead of the near field measurement.

3.4.4. Implementation of a fine grid for auralization

As it is mentioned in 2.1.2 by reducing the size of the cell the frequency range of interest is increased. In order to obtain a fairly frequency range for auralization purpose a fine grid has been implemented. In the simulation program. A grid size of 4 cm has been tested. In the sound source position a finite length signal in the time domain of music (speech) has been used as an input signal. The signal has been recorded at two microphone positions spaced by 16 cm. Since the computation is done in the time domain it takes so much time to compute even one minute of music. Instead auralization can be implemented using direct convolution with the obtained impulse response from the simulation program.

4. RESULTS AND VALIDATION

4.1. Measurements

In order to validate the results of the simulations two set of measurements have been carried out in the test room.

4.1.1. Sound Pressure Distribution Measurement

To validate the sound pressure distribution a rectangular area has been delimited in the test room. A rectangular grid of 10x9 points separated each other by 20 cm at a height of 1.20 m in the test room has been set up see Fig.9. The panels that cover the walls and the carpet from the floor were removed.

The test loudspeaker (A) was set in the room at position (1.2m,2.0m,1.2m). The sound pressure level has

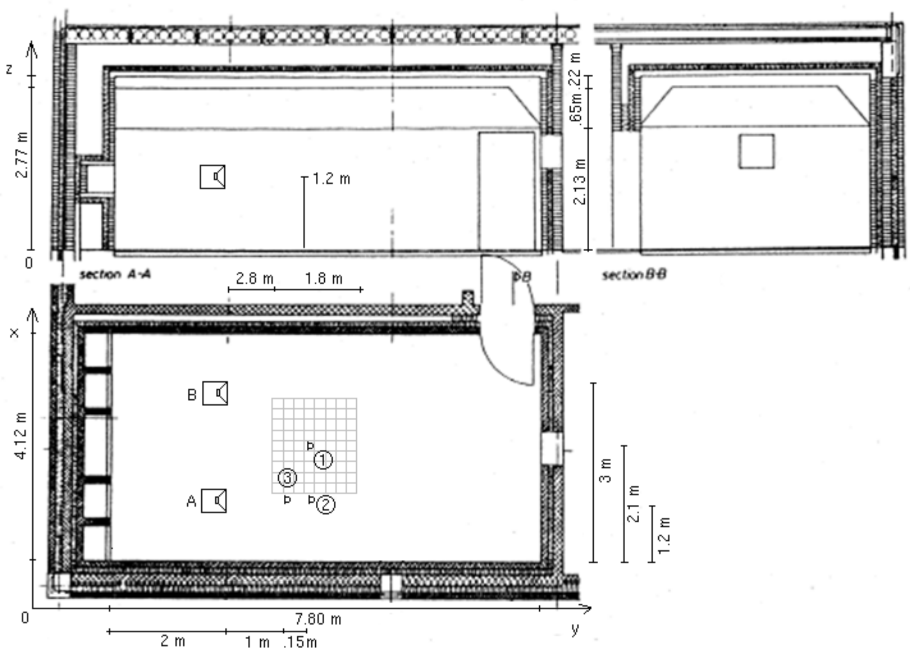

Fig. 9: Listening test room, dimensions shape and measurement set up, adapted from [14].

been measured at each grid point. The measurement was carried out driving the test loudspeaker with 65 Hz by a sine generator with 1.0 V RMS amplified by a reference stereo amplifier. As mention before the test loudspeaker is a closed box type with a volume of 12 litre and 35 cm height, 23 cm width and 23.5 cm depth it has a 16.5 cm diameter bass driver unit and a 1.9 cm, polyamide dome tweeter. The test loudspeaker (A) was pointing to the grid area as seen in Fig.9.

At each point in the grid a pressure microphone connected to a pre-amplifier was located with a precision of ± 1.5 cm. The output from the pre-amplifier was connected to a measuring amplifier. The average time from the measuring amplifier was set to 1 sec to obtain an RMS voltage value. The system was calibrated by a piston-phone to 124 dB SPL at 250 Hz.

4.1.2. Impulse Response Measurement

To validate the acquisition of the impulse response by the MLS method the impulse response at three microphone positions in the room were measured. The microphones were located at 1.20 m height. The room was exited first by the test loudspeaker (A) an afterwards by both test loudspeaker (A) and test loudspeaker (B), the test loudspeaker (B) was also a closed box the same type, model and dimensions. In Fig.9 the three microphone positions and the loudspeaker positions are shown.

At each microphone position a pressure microphone connected to a pre-amplifier was placed. The output from the pre-amplifier was connected to a measuring amplifier and sent to a MLS PC measuring system board. The system was calibrated with a pistonphone producing a sound pressure level of 124 dB at 250 Hz.

It was decided to band-pass filter the excitation signal since the result of the simulation is a band limited impulse response. From the MLS measuring system the output was connected to a analog band pass filter set to 10 Hz and 600 Hz as cut off frequencies. From the band pass filter the signal was sent to a reference stereo amplifier and from there to the test loudspeaker.

The length of the impulse response was set to 8191 samples with a sampling frequency of 8 kHz. The

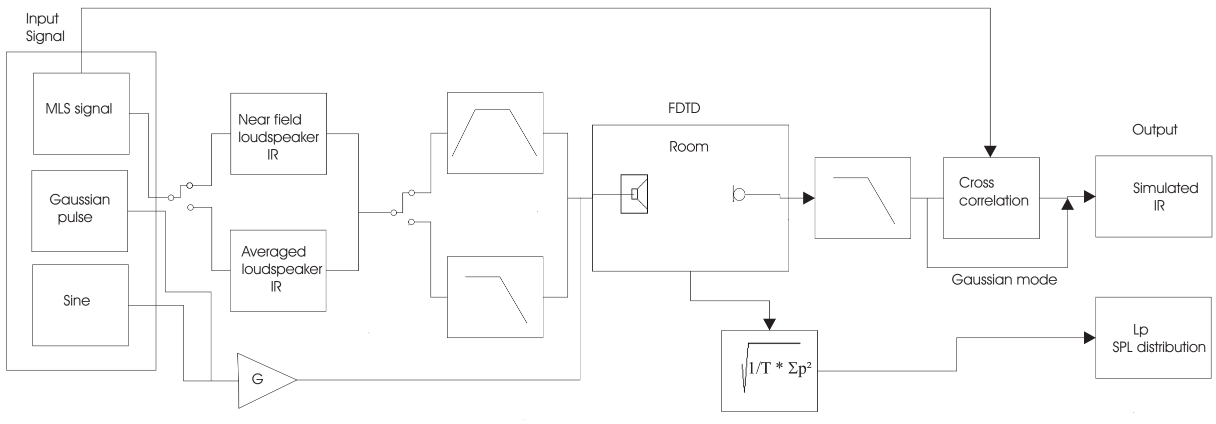

Fig. 10: Block diagram of the simulation program.

bandwidth was set to 2 kHz with a Butterworth 8th order low pass filter as a anti aliasing filter.

The first measurement was done measuring at the three microphone positions placing test loudspeaker (A) and the second one was measuring again the three microphone positions with test loudspeaker (A) and test loudspeaker (B) included both. The excitation signal was the same for both loudspeakers.

4.2. Simulation

Two main scenarios were simulated following the measurements. The sound pressure level distribution over the selected grid area and the computation of the impulse response of the room by the excitation of first test loudspeaker (A) and secondly adding test loudspeaker (B). The averaged impulse response of the real loudspeaker was included in the model. In both cases the simulation time was 1 sec at a sampling frequency of 8kHz.

The space grid was set to 10 cm for both cases. For the impulse response simulation the impulse response of the analog band pass filter and the anti aliasing filter from the MLS system were included in the model.

The boundary conditions for the walls were set as follows. It was decided to use the characteristic impedance as a parameter for the boundary condition. Not often one can find information about the absorption coefficient of materials at very low frequencies. Since it was possible to find on tables the density in $\frac{Kg}{m^3}$ and the speed of sound in the boundary materials the characteristic impedance of each wall was used as follows

$$Z = \rho c \qquad (15)$$

For example the floor that is wooden has an approximate density of 450 $\frac{Kg}{m^3}$ and the speed of sound in wood is 3500 $\frac{m}{s}$ therefore the characteristic impedance is, $\rho c = 1.575\text{x}10^6 \ \frac{kg}{m^2 s}$.

4.3. Comparison of simulations and real measurements

In Fig.11 the sound pressure distribution can be observed after simulation at a height of 1.2 m. It is noticeable the influence of the room forming the nodes and antinodes by the stationary waves.

In Fig.12 two surface plots are shown, these graphs represent the sound pressure level simulated and secondly measured along the chosen surface area, in Fig.11 the same simulation is shown along the complete surface layer of the room.

In Fig.13 the frequency response from the measured and simulated response are shown. In the left column the room was exited by test loudspeaker A while in the right column both test loudspeakers were used. The excitation signal was the same for both.

In Fig.14 the measured impulse responses derived by the simulation program are shown. It can be

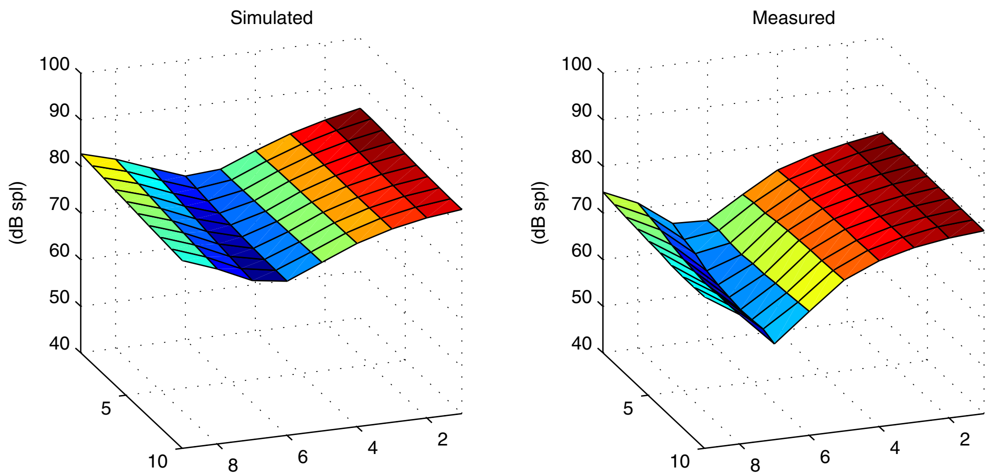

Fig. 12: Comparison of sound pressure level distribution in the rectangular grid area and measurements, the grid has 9x11 points separated by 20 cm form each other.

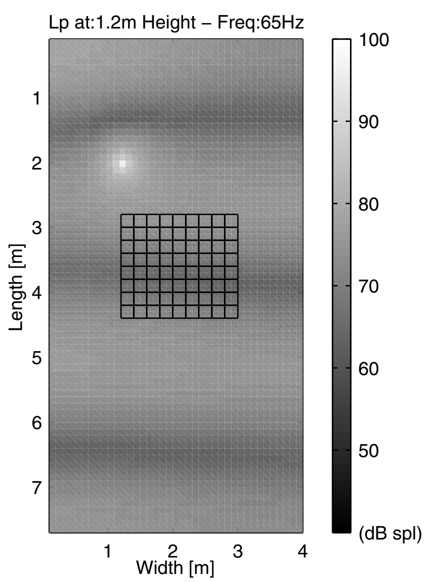

Fig. 11: Sound pressure level distribution resulting from the simulation of one loudspeaker reproducing a sinusoidal frequency of 65 Hz, the measuring grid points can be observed.

observed that the impulse responses calculated in the simulation have more energy content than the real ones. Nevertheless as it is shown in Fig.13 they do not differ so much in the frequency domain.

5. DISCUSSION

As it is shown in Fig.11 and in Fig.12 the simulation program present good agreement to what happen in the reality. The main room resonances are revealed by the simulation. It can be said that at very low frequency there is some divergence. Nevertheless the simulation program can be used as a predictor tool in order to know beforehand what would happen when a loudspeaker is placed in a rectangular room. It also has to be added to the discussion that the ceiling of the room is quite complex to model since it is not regular and some parts are covered with very absorptive material like rock wool but in some other areas the ceiling is very reflective. In connection to that it was quite difficult to find absorption coefficients for the boundary materials at very low frequencies. It was a good approximation to model the boundary condition using the characteristic impedance of the materials.

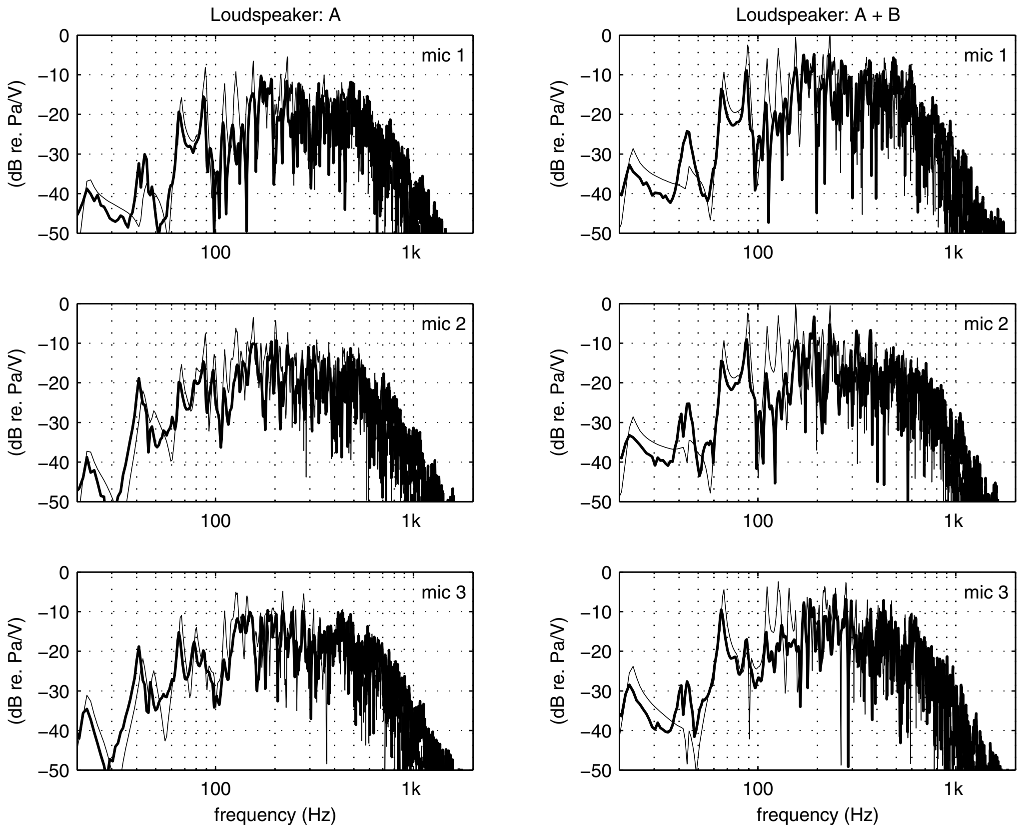

Fig. 13: Thick lines are frequency responses from measured impulse responses at microphone positions 1, 2 and 3; Thin lines are the simulations. Left column only loudspeaker (A) is included while in right column test loudspeaker (B) is also included.

6. CONCLUSION

A simulation tool has been developed. Finite-difference time-domain FDTD method has been used to approximate the sound pressure and particle velocity produced by multiple loudspeakers in a rectangular room. The simulation program has been tested with good results according to real measurements. The developed application can be used as a reliable tool for equalization purposes on multi-channel sound reproduction systems in conjunction with other approximations as Ray tracing or Image source. The solution gives the possibility to evaluate and visualize as an animation of moving pictures the interaction of multiple loudspeakers in a room. Moreover the possibility of direct auralization of multi-channel signals is possible by convolution of the calculated impulse response and anechoic recordings.

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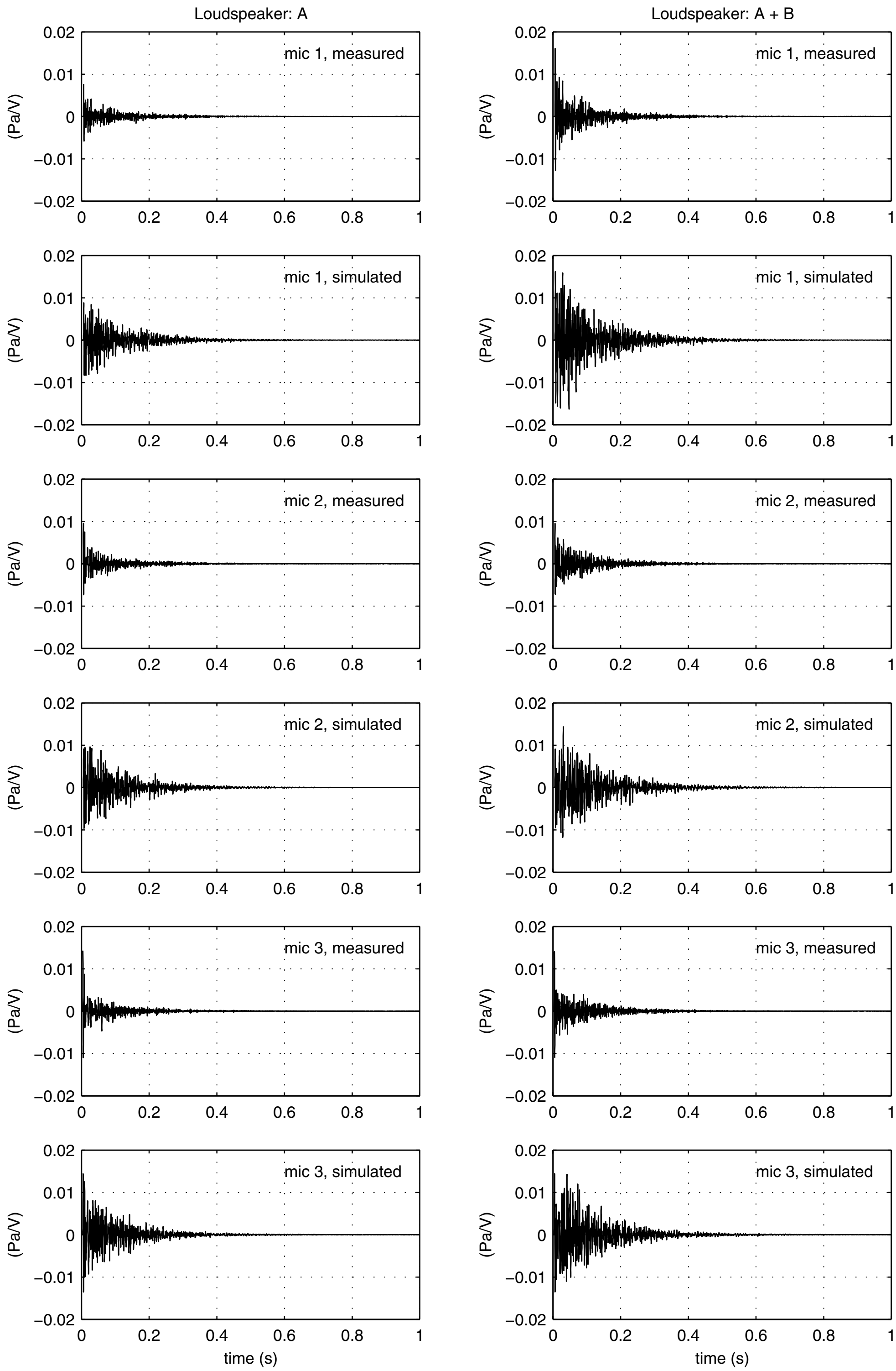

Fig. 14: Impulse responses measured and simulated by FDTD correspondent to the three microphone positions in the test room.

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