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Low-Frequency Performance of a Woofer-Driven Flat-Panel Loudspeaker (Part 2: Numerical System Optimization and Large Signal Analysis)

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ABSTRACT

The paneled woofer design is a new concept to improve the low-frequency behavior of flat-panel loudspeakers. This paper focuses on the numerical optimization of the transducer resources, the panel properties, and the system construction. The low-frequency optimization is based on an electro-mechanical network model, by which the modal depending air compliance is introduced. However, various parameters are modified and validated with measurements. Furthermore, the maximum sound pressure level is considered and the limiting distortions are differentiated from the panel and the woofer itself. The results are compared with the large-signal behavior of an exciter mounted on the same panel. Summarized, additional advice for the optimization of the frequency response is given and examples for unsuitable constructions are presented.

1 Introduction

Customers of audio devices expect that the audio signal can be reproduced at sufficient amplitude and quality [1]. To compete with existing speaker systems, flat-panel loudspeakers need to produce high sound pressure output with acceptable signal distortion. Nearly all available drivers for flat-panel loudspeakers are electro-mechanical and electro-acoustical transducers producing substantial distortion at high signal amplitudes. The accurate modeling of distortions and other nonlinear symptoms of loudspeakers have been investigated for a long time, and substantial progress has been made [2, 3, 4]. The results are more accurate models, which are reliable to predict the nonlinear behavior and investigating the impact on the subjective listening impression[5, 6]. How far those nonlinearities affect the large-signal behavior of a paneled woofer, will be clarified in this papers.

This paper presents the second part of this paper series, focusing on prospects for system optimization, characterizing the large-signal behavior, and determining the maximum sound pressure level for a given THD value. In the first part of this paper series [7] the paneled woofer design is introduced, and a significantly improved bass response could be reached. This refers to a driver being coupled to the sound radiating panel via a paneled volume. This woofer driven excitation

has been modeled with a simulation tool and validated regarding its mechanical and acoustical properties with experimental data. This model is able to predict the sound radiation. However, this system design has an improved low-frequency performance, which is comparable to conventional systems in a sealed enclosure. Whether this comparability also exists at high levels, is examined in the following paper.

This paper describes the optimization of the introduced paneled woofer design and analyzes the large-signal behavior. In section 2, an electro-mechanical network is developed, which is representing the simulation model for identifying optimization potentials. Furthermore, section 2 introduces the modal related air compliance. This parameter is important to understand the mode shift of individual modes and to justify the flatness of the transfer function. In the following section 3 and 4, woofer and panel related design considerations are developed to improve the sensitivity and the flatness of the pressure response. In section 5 a large signal identification is conducted. The influence of the panel is tested by mounting the woofer in several panel systems, and an increase of the linear range of the displacement-dependent stiffness could be determined. In section 6, the maximum SPL is determined for a conventional woofer, a paneled woofer, and a panel with the exciter. There, an approach considering the produced harmonic distortions is utilized, which helps to identify the physical cause of the distortion itself. In the last section 7, panel limits due to the panel size and its displacement are investigated, leading to design considerations for improving the maximum SPL. This results in the main research goals, which are summarized as follows:

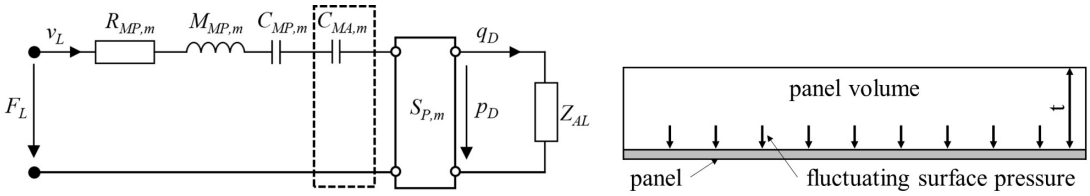
- How can the acoustical performance be improved using optimized panel and woofer related properties?
- How does the panel volume affect the individual modes and the radiation characteristics of the speaker?
- Is it possible to achieve the same maximum SPL with the paneled woofer design as a conventional system?
- What are the limiting factors of the maximum SPL regarding a woofer driven flat-panel loudspeaker?
- How does the panel influences the SPL max?
- Is there a difference in the SPL max between the paneled woofer and the panel with exciter?
- How to get smooth responses, with optimal usage of driver resources and low distortion?

2 Development of an Electro-Mechanical Network

Conventional loudspeakers can be modeled accurately, in the small signal domain, by the equivalent circuits assuming constant parameters of the lumped elements introduced by the research work of Thiele [8] and Small [9]. If the loudspeaker system becomes more complex and modal and strongly frequency-dependent characteristics are existing, the lumped parameter model is no longer sufficiently accurate. Flat-panel loudspeakers are a highly modal driven system, which needs to be modeled with distributed parameters. The distributed parameter model includes e.g. the individual modes, the complex effective radiation area, and the modal depending stiffness. The following electro-mechanical circuit illustrates how the mechanical modes couple to the electrical and acoustical domain. A fundamental model for the modal behavior of the loudspeaker diaphragm was developed by Bright et al. [10]. However, due to the size of the panel, the significant impact of the air stiffness is not negligible in this application. Therefore, the common model will be extended in the following section.

2.1 Model of the Modal Related Air Compliance

In the first part of this paper [7] it could be shown that mode 1 is used over a very wide frequency range, and consequently, a flatter frequency response is achieved. This effect is caused by the additional air volume, which is coupled to the modal driven panel. Therefore, it will be referred to as panel volume. For conventional pistonic speakers, the air volume acts as a mechanical spring because of finite compressibility [8, 9]. The air spring causes a change in the loudspeaker's fundamental resonance frequency, depending on the stiffness. An increased fundamental frequency leads to an increased lower cut-off frequency. In the case of a flat-panel loudspeaker, the diaphragm is not stiff at all and behaves strongly frequency-dependent. A simplified FE-simulation model is built in the following section for a detailed analysis of the modal stiffening by the air volume. This model represents the electro-mechanical network shown in Figure 1 and illustrates the modal related compliance $C_{MA,m}$. Each mode is represented as an analogous loop in the electrical circuit in the mechanical modal model. The modal behavior is represented by an LCR resonant circuit, in which $M_{MP,m}$ represents the mass, $C_{MP,m}$ the compliance, and $R_{MP,m}$



(a) Electro-mechanical model for a single mode (b) Abstracted FE-simulation model

Fig. 1: a) Electro-mechanical network model for a single mode, b) the abstracted FE-simulation tool to calculate the resulting influence of the panel volume.

the damping of mode m. Each mode has an individual effective acoustic radiating surface $S_{P,m}$.
The simplified FE-model is based on an FE-subsystem coupled to an FEA-subsystem, whereby the FE-subsystem represents the panel and the FEA-subsystem represents the panel volume. For all cases, the panel properties have not changed. The FEA-subsystem volume is varied by the thickness of t in the range of 5 mm to 1280 mm. The case of 1280 mm is more of a scientific case to visualize a low additional stiffening. Furthermore, no additional acoustical modes are calculated, only the constant pressure mode. Additionally, the panel modes one, three, or eight are calculated individually, representing their individual electro-mechanical circuit, without superposition with other modes. The mode shape, the natural eigenfrequency, and the effective acoustic radiating area of mode 1, 2, 3, and 8 are plotted in Figure 2. The applied force is a fluctuating surface pressure with a total load of 1 N. The results of the space averaged displacement of the panel surface, are presented in Figure 3. This Figure visualizes the modal related stiffening through the air compliance. It can be seen that the natural frequency of mode 1 is shifted due to the additional air compliance. The natural frequency of mode 1 is 50.1 Hz and shifts with the smallest panel volume close to 285 Hz. The displacement far below the resonance frequency is determined by the total stiffness of the modal compliance

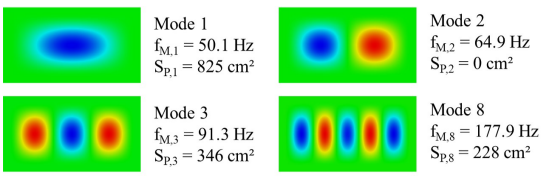


Fig. 2: Results of the eigenmode solver for mode 1, 2, 3, and 8. The eigenfrequency $f_{M,n}$ and the effective acoustic radiating surface $S_{P,n}$ are plotted.

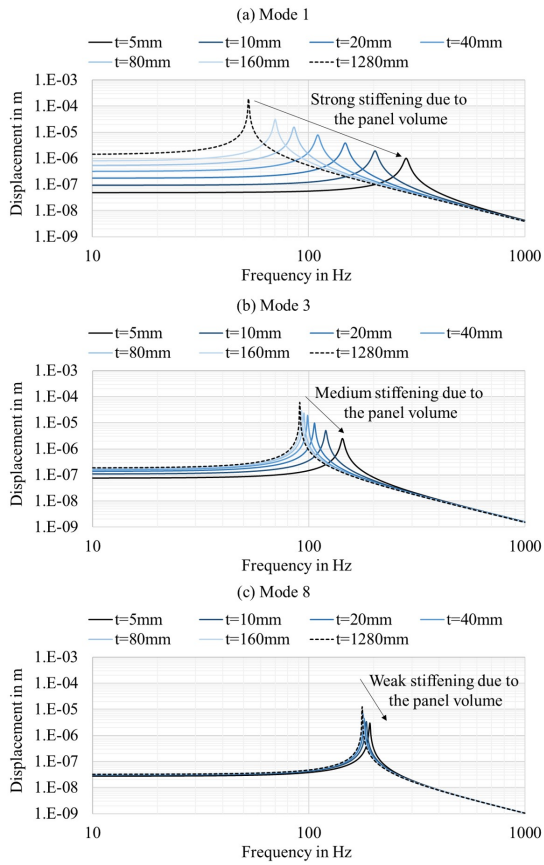


Fig. 3: Space averaged panel displacement of three individual modes (a) mode 1, (b) mode 3, and (c) mode 8, whereby the air stiffness related influence is caused.

and the air compliance of the panel volume. For small volumes, where the air stiffness is significantly higher than the modal compliance, the stiffness is essentially determined only by the panel volume. This shows that if the volume is doubled, the displacement is almost twice as high. If the panel volume is larger, then the additional stiffness becomes smaller. In this case, the overall stiffness is more dependent on modal compliance.

Considering the displacement of mode 3 and mode 8, it becomes obvious that the additional stiffening of the panel volume is much lower. At mode 8, only a small impact is detectable. It becomes evident that the additional air stiffening is related to the mode shape and effective radiating surface of the individual panel modes. Mode 1, having the simplest mode shape, shows the strongest influence, whereas mode 8, having the most complex mode shape of the three modes, shows the weakest influence.

Furthermore, it should be mentioned that the additional air compliance allows a change of the mode order. For small panel volumes, in this case for $t < 10\,mm$, the resonance frequency of mode 1 can be higher than that of mode 3 or mode 8 caused by the additional air stiffness. Mode 3 and mode 8 could not change the modal order, but the difference of the natural frequencies could be significantly reduced.

2.2 The Math of the Modal Related Air Complicance

The modal related air compliance is strongly dependent on the mode shape and the effective radiating surface, as shown in Figure 3. The effective modal specific acoustic radiating surface can be calculated analogously to Klippel and Schlechter [11]. In this case, it is calculated based on Equation 1 using the results of the natural frequency analysis as an integral of the modal displacement $X_m(\vec{r}_P)$ over the panel surface S_P. Additionally, the modal displacement $X_m(\vec{r}_P)$ is normalized by the maximum of the modal displacement of each mode $max(X_m(\vec{r}_P))$. The effective radiation area cannot be negative. Therefore, the absolute value needs to be taken. The index m represents the mode number.

$$S_{P,m} = \left| \int_{S_P} \frac{X_m(\vec{r}_P)}{max(X_m(\vec{r}_P))} dS_P \right| \tag{1}$$

To describe the modal dependent stiffness of the air, the fundamental considerations analogous to Beer et al. [12] are extended to the frequency dependent effective radiating surface. Figure 4 defines the essential relationships and variables for the following subsection. If the effective modal specific acoustic radiating surface is large, e.g. for mode 1, the force of the airspring $dF_{A,m}$ is high. For modes with no effective modal specific acoustic radiating surface, e.g. mode 2, the force of the airspring $dF_{A,m}$ is negligible. The balance of forces is described in the Equation 2:

$$dF_L, m = dp\, S_{P,m} = -K_{M,A}\, dx = dF_{A,m}. \tag{2}$$

The force $dF_{L,m}$, which is acting on the panel, is equal to the force of the air spring $dF_{A,m}$, which is acting contrary and results in the following relationship.

$$-K_{M,A} dx = S_{P,m} dp \quad \rightarrow \quad K_{M,A} = S_{P,m} \frac{dp}{dx} \tag{3}$$

The term $\frac{dp}{dx}$ can be solved with $dV_P = S_P\, dx$. Furthermore, the following relationships in Equation 4 also apply, in which κ represents the isentropic exponent.

$$\frac{dV}{dx} = S_{P,m} \qquad \frac{dp}{dV} = -\frac{\kappa\, p}{V} \tag{4}$$

If these formulas are applied to (3), the equation for the stiffness of the air spring is derived. The relationship between the panel volume, radiating area, and the resulting stiffness is shown in the following Equation 5.

$$K_{MA,m} = \frac{\kappa p {S_{P,m}}^2}{V_{panel}} = \frac{1}{C_{MA,m}} \tag{5}$$

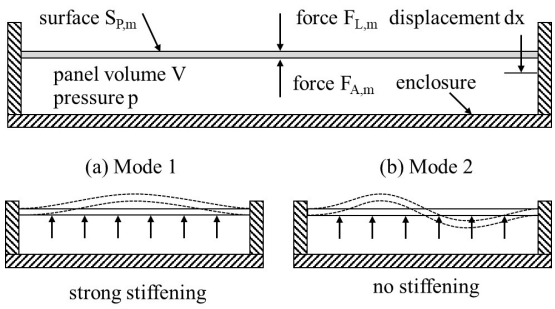


Fig. 4: Model for the derivation of the air spring description due to the panel volume, in which the effect for two modes is visualized.

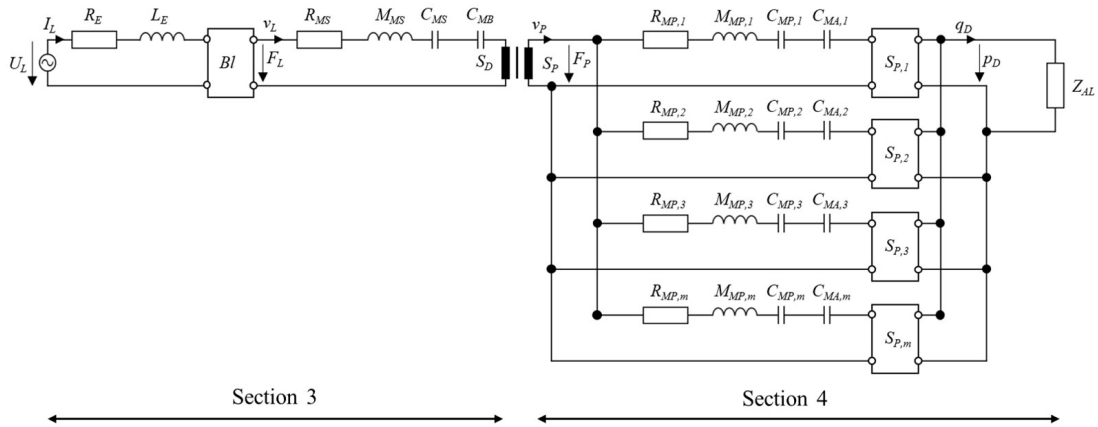


Fig. 5: Reduced electro-mechanical network of the sealed paneled woofer design with modal implementation.

The panel volume describes the volume that is directly coupled to the panel and is linked to the modal properties, in which ρ_0 describes the density of air, c the speed of sound in air, and $S_{P,m}$ the effective acoustic radiating surface of each mode m. The stiffness is the inverse of the compliance as shown in Equation 6.

$$C_{MA,m} = \frac{V_{panel}}{\kappa p {S_{P,m}}^2} = \frac{V_{panel}}{\rho c^2 {S_{P,m}}^2} \tag{6}$$

2.3 Lumped Parameter Model for a Woofer Driven Flat-Panel Loudspeaker

The following sections present the lumped parameter mechanical model, which is equivalent to the modal model when the paneled woofer system is driven with a conventional woofer. Figure 5 shows the electro-mechanical analogy of a modal model with four modes. The radiating surface S_D of the conventional driver is coupled to the total panel surface S_P via a transformator. This transformation represents a coupling from the mechanical to the acoustical domain of the conventional driver and backward from the acoustical domain to the mechanical domain of the panel. For lower frequencies, it is almost lossless.

Furthermore, this model is extended for a flat-panel loudspeaker. Each mode is represented as an analogous loop in the electrical circuit in the mechanical modal model. An LCR resonant circuit represents the modal behavior and each mode has an individual effective acoustic radiating surface $S_{P,m}$.

Additionally, the panel volume is coupled to the modal related panel. This allows an individual stiffening of each mode, as mentioned in the last subsection. The complex contributions of the individual eigenfunctions are summed up to the complex movement. However, if the natural frequencies are far apart, each mode is excited separately. If the modes have similar eigenfrequencies, new coupled modes will be generated.

Mode number 1 has the largest radiating surface and has a very large contribution to the total radiation. Furthermore, the stiffening by the panel volume is greatest in mode 1. This additional stiffening is represented by the compliance $C_{MA,1}$. By comparing mode number 2, it becomes obvious that this mode has no radiating surface and no contribution to the total radiation.

2.4 Conclusions

A summary of the essential conclusions of this section is given below:

(1) Flat-panel loudspeakers have to be modeled with a distributed parameter model.

(2) The contributions of each mode can be represented as an analogous electrical circuit, which is summed up to the complex movement of the whole panel.

(3) The panel volume has a strong influence due to the effective acoustic radiating surface of each mode.

(4) Modes with a large effective acoustic radiating surface will be increasingly stiffened compared to modes with a smaller effective acoustic radiating surface.

3 Numerical Optimization of Driver Resources

The woofer driven flat-panel loudspeaker is excited by an electromechanical transducer. Consequently, the sensitivity of the overall system directly depends on the driver's performance. A transducer can be described via an electro-mechanical equivalent circuit. Hence, lumped parameter modeling is sufficient for analyzing the speaker's behavior.
The optimization is performed with the software tool wave6 [13] and the implemented electro-mechanial network presented in [7].
According to Klippel [14], the SPL calculation based on lumped parameters can be divided into three main frequency bands - passband, fundamental resonance, and low frequencies. The voltage sensitivity for a transducer mounted in a large baffle provides at low frequencies f < fs:

$$SPL_{U_{ref},r_{ref}}(f < f_s) = 20lg\left(\frac{\rho_0 Bl S_D 2\pi f^2}{|r_{ref}| R_E K_{MS} p_0}\right) \quad (7)$$

The transducers SPL(f) in the passband with $2f_s < f < 5f_s$ can be calculated as follows:

$$SPL_{PB,U_{ref},r_{ref}} = 20lg\left(\frac{\rho_0 Bl S_D u_{ref}}{2\pi |r_{ref}| R_E M_{MS} p_0}\right) \quad (8)$$

At the fundamental resonance frequency the voltage sensitivity can be calculated using:

$$SPL_{U_{ref},r_{ref}}(f_s) = 20lg\left(\frac{\rho_0 f_s S_D u_{ref}}{|r_{ref}| Bl p_0}\right) \quad (9)$$

This flat-panel loudspeaker has a low resonance frequency. Therefore, the predominant scope is to improve the sensitivity at the resonance frequency and in the passband, while an optimization below the woofer's resonance is not productive. Considering the Equations 8 and 9, it becomes obvious that especially the moving mass M_{MS}, the stiffness of the woofers suspension K_{MS}, and the force factor Bl affect the sensitivity of the transducer. These factors are separately investigated in the following sections. All other parameters e.g., R_E, S_D were not changed.

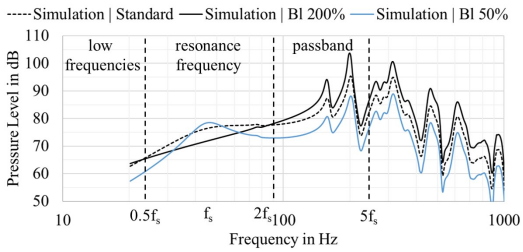


Fig. 6: Pressure response versus frequency f in 1 m distance with 2.83 V input of the paneled woofer design, using three different values of the force factor Bl(x=0) at the rest position of the woofer.

3.1 Force Factor

The force factor Bl varies in the first step, which is implemented by varying the transfer impedance in the FE-software. In Figure 6, the simulated frequency responses for three different force factors are illustrated. It can be derived that reducing the force factor leads to minor electrical damping of the resonance frequency, so the SPL is increased around the resonance according to Equation 9. However, in the passband, the frequency response is reduced by -6 dB, considering Equation 8. In contrary to that, a higher force factor leads to a decreased SPL at the resonance frequency, due to the stronger electrical damping. However, the passband is constantly raised by +6 dB.

3.2 Ratio of Mms and Kms

Likewise, the moving mass and the woofer's suspension stiffness are varied next to preserve the same ratio, and, respectively, the same resonance frequency. According to Figure 7, a lighter panel and weaker suspension clearly lead to a significant improvement in the passband, as well as a minor improvement below the resonance frequency. This can be explained considering the equations based on lumped parameters, as the low-frequency range depends on K_{MS} and the passband on M_{MS}, while the SPL at the resonance frequency is independent of K_{MS} and M_{MS}. It must be emphasized that the woofer's mass has a considerable influence on the frequency response, although the sound radiating panel is with 1.84 kg significantly heavier.

3.3 Example of an Optimized Driver

Based on the conclusions of the previous sections, a virtually optimized woofer is designed according to Table

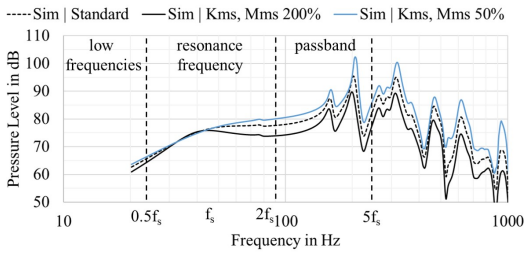


Fig. 7: Pressure response versus frequency f at 1 m distance with 2.83 V input of the paneled woofer design, using three different values of the ratio of K_{ms} and M_{ms} of the woofer.

1. These improved TSP lead to an optimized frequency response, which is illustrated in Figure 8. Due to the reduction of the woofer's mass and the suspension stiffness, the SPL of the passband is increased by 2-3 dB. Furthermore, the adjusted Bl guarantees no SPL loss around the resonance frequency. This case considers the optimization of the sensitivity. The Force Factor Bl could be further reduced, resulting in a level increase in the frequency range of the resonance frequency f_s.

Table 1: TSP of the woofer Dayton DA 175-8 (Standard) and an optimized woofer for this configuration

Parameter	Standard	Optimized	Change
M_{ms} in g	20.68	10.34	-50%
R_{ms} in kg/s	2.07	2.07	0%
K_{ms} in N/mm	0.89	0.45	-50%
Bl in N/A	6.28	5.02	-20%
f_s in Hz	32.9	32.9	0%
R_e in Ω	5.99	5.99	0%
L_e in mH	0.512	0.512	0%
R_{e2} in Ω	3.46	3.46	0%
L_{e2} in mH	0.635	0.635	0%

3.4 Conclusion

If the additional stiffness of the panel does not influence the behavior of the driver too much, the same optimization criteria as those for conventional woofers in a sealed box apply.
By using a softer suspension, an active stabilization of the voice coil rest position is recommended, as mentioned by Klippel [14]. In combination with a reliable protection system and a less progressive suspension, it is possible to reach more peak displacement and better bass performance.

4 Numerical Optimization of the Paneled Woofer Construction

In the following section, the FE-simulation model is used to optimize the panel parameters. Therefore, an electro-mechanical network representing the flat-panel loudspeaker model is developed for identifying optimization potentials. The influence of the panel's material properties, the panel size, and the panel volume on the sound radiation are investigated. Finally, design considerations for an improved flat-panel loudspeaker are evolved based on the detailed examination.

4.1 Introduction

In part one of this two-part paper [7], a numerical simulation model is developed and validated regarding its mechanical and acoustical behavior. It could be proven that the simulation is able to predict the actually measured sound radiation very well. In a further step, the simulation model is optimized by selectively adjusting geometrical and material parameters. Therefore, both simulations and measurements of different configurations are conducted to verify the simulation model's prediction capabilities. These simulations and measurements also serve to expand the variety of possible set-ups that can be analyzed.

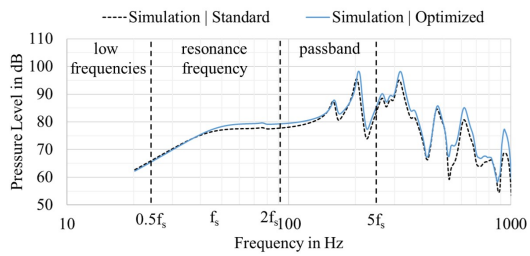


Fig. 8: Pressure response versus frequency f at 1 m distance with 2.83 V input of the paneled woofer in comparison of a standard design and an optimized woofer.

Table 2: Material parameters of the chosen panels to compare the resulting coincidence frequency

Properties		Mineral	Acrylic	HPL
Thickness [mm]	h	2.9	2.05	3.05
Young's mod. [GPa]	E	9.3	4.82	16.4
Density [kg/m^3]	ρ	1595	1250	1480
Poissons ratio	ν	0.32	0.32	0.3
Damping loss factor	η	0.05	0.02	0.07
Bending stiff. [Nm]	B	21.13	3.86	42.61
Area density [kg/m^2]	m''	4.63	2.56	4.51
Coincidence [kHz]	f_c	8.76	15.27	6.09

4.2 Optimization of Panel Properties

By adjusting panel properties, the mode distribution, and the panel's radiation impedance can be optimized with a focus on the acoustic performance. A vibration system is characterized by its mass-, damping-, and stiffness properties (M_{MM}, R_{MM}, C_{MM} in Figure 5). However, in this work's scope, the materials density and elastic modulus, respectively, Young's modulus only.
Table 2 illustrates the panel's material properties, which are investigated experimentally as well as numerically [15]. Considering the geometrical and material properties, the panels have significant distinctions regarding their mass and stiffness behavior. This can be derived from their different bending stiffness and area density.

Figure 9 illustrates the measured (a) and simulated (b) SPL frequency responses on-axis at a distance of 1 m of different panel materials. A comparison of the experimental and numerical data shows that the courses coincide very well, hence, the simulation model is able to predict the influence of different panel materials. In general, the radiated sound pressure decreases considerably below the first eigenfrequency, as the panel displacement is limited, according to the panel's elastic properties. The electro-mechanical network also represents this effect, because the panel stiffness of mode m is depicted as the capacitor $C_{MP,m}$, which has an increasing impedance towards low frequencies. A lager bending stiffness is equivalent to a higher first eigenfrequency from which the SPL begins to decline. In consequence, the panel made out of the acrylic glass with a comparatively low bending stiffness has a much

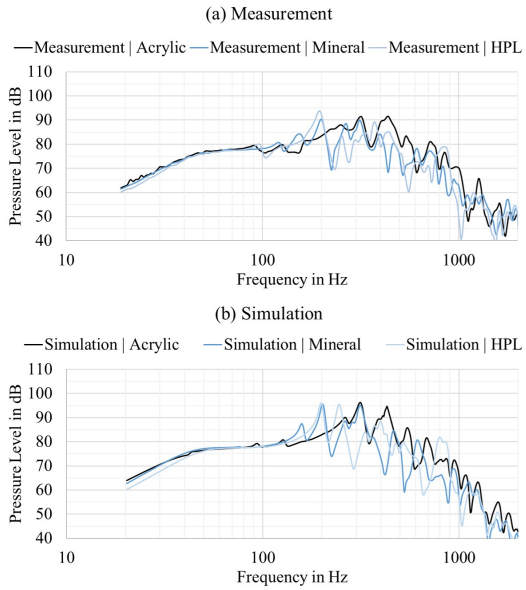


Fig. 9: Comparison of the influence of the panel material a) measurement, and b) simulation evaluated at 1 m distance with 2.83 Vrms input.

better acoustical performance at low frequencies than the significantly stiffer High-Pressure-Laminate (HPL) panel. Moreover, the panel made out of acrylic glass has the flattest transfer function, although this panel is connected to the lowest eigenfrequencies, due to the low Young's modulus. The reason for that phenomenon is the influence of the paneled volume, which affects materials with a low bending stiffness more intensely. Consequently, the first eigenmode driving the acoustical radiation will be further shifted towards higher frequencies if the panel material is more flexible. In contrary to the bending stiffness, the area density predominantly affects the mechanical impedance at higher frequencies as the panel's modal mass is represented as the inductor $M_{MP,i}$ in the electro-mechanical analogy. In consequence, higher densities are equivalent to a raised inertia. Hence, the frequency response drops significantly earlier. In Figure 9, it can be seen that the transfer function of the mineral panel has a lower SPL than the panel made out of the acrylic glass from approximately 200 Hz, due to its higher mass.

4.3 Influence of the Damping Loss Factor

The damping loss factor η describes the dissipation of kinetic and elastic energy into other energy forms. This can be determined using the bandwidth for 3dB decay, described with the following Equation 10, in which f_{m+} describes the upper -3 dB and f_{m-} the lower -3 dB limit, according to Carfagni et al. [16].

$$\eta_m = \frac{f_{m+} - f_{m-}}{f_m} = \frac{1}{Q} \tag{10}$$

The paneled woofer design is simulated with a damping loss factor η of 0.002, 0.02, and 0.2 to illustrate the impact of the damping. The results are shown in Figure 10. It is obvious that an increased damping loss factor will result in a significant flatter transfer function. Peaks, as well as, dips are minimized in the amount of the individual deviation from the mean value. In the frequency range (f < 150 Hz), in which no modes occur, the damping has no influence. The transfer function with a small amount of damping (η = 0.002) highlights the exact position of the individual modes through its peaks. Thereby, the eigenfrequencies of mode 1, 3, and 8 becoming very obvious, which are shifted to higher frequencies by the additional compliance through the panel volume, as mentioned in the previous section.
However, in cases with a wider frequency range, the panel should provide a reasonable amount of damping. Materials with low dampings, such as steel or aluminum, would cause a ringing. This is caused by low damped modes, which will reverberate for a fairly long time. Furthermore, these narrow peaks are difficult to compensate with DSP filtering. Sufficient practical results were achieved with a damping loss factor of $\eta = 0.05$.

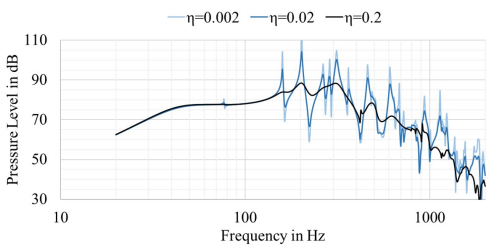


Fig. 10: Simulated frequency response of the paneled woofer using a mineral panel with different damping loss factors η.

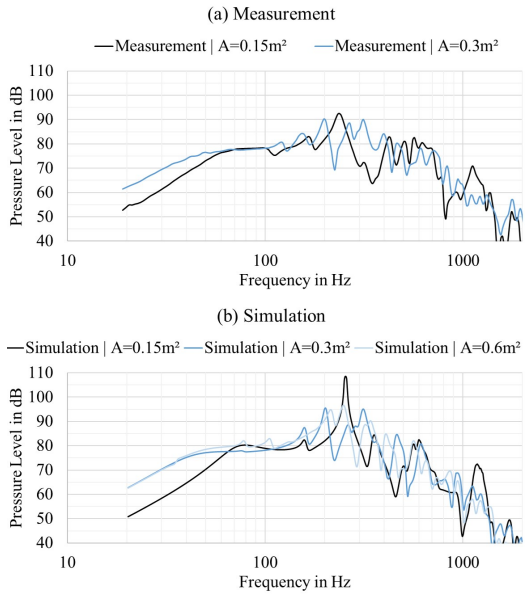


Fig. 11: Comparison of the influence of the panel size a) measurement and b) simulation.

4.4 Influence of Panel Size

Next to the elastic properties of the material, the geometric dimensions affect the modal panel stiffness ($C_{MP,m}$) as well. According to Mitchell and Hazell [17], larger panels enable lower eigenfrequencies, and in return, a smaller panel increases the stiffness.
Figure 11 depicts the measured (a) and simulated (b) frequency responses on-axis at 1 m distance of differently sized panels. Two panels, with a size of 0.15 m^2 and 0.3 m^2, are investigated experimentally. It can be derived that a smaller panel leads to a significant loss of the SPL at lower frequencies, as a reduced panel size adds additional stiffness and, thus, increases the eigenfrequencies. As a consequence, the frequency response is limited due to the geometrical rigidity. Above 60 Hz, there is no considerable difference regarding the SPL of the transfer functions. However, the resonances are shifted because of the different natural frequencies.
By comparing the larger panel size with 0.6 m^2, a minor improvement to the 0.3 m^2 sized panel could be reached. If the panel is too soft, the frequency response is limited to the response of the driver without panel. No additional improvements can be reached. Further, it can be concluded that the simulation can predict the impact of the panel dimensions on the transfer function.

4.5 Influence of the Panel Volume

A further adjustable variable acting in this system can be found as acoustical compliance $C_{MA,m}$ in the electromechanical network, which is caused by the paneled volume. The paneled volume will be adjusted by the thickness t of the air-gap. The influence of paneled volume to the eigenfrequencies of the panel is described in a previous subsection.
Figure 12 shows the measured (a) and simulated (b) transfer functions of a flat-panel speaker with different panel volumes. However, prototypes with an air gap height t=4 mm and t=10 mm were used. The experimental data of these configurations are validated by comparing it with the numerical simulation. As only small deviations occur, predominantly caused by incorrect damping coefficients and enclosure vibrations, it can be derived that the simulation model is able to predict the air gap impact on the sound radiation. In consequence, another case with an air gap height of t=20 mm is examined utilizing the numerical model.
A smaller panel volume leads to a much flatter transfer function and the modal break up is shifted towards higher frequencies. In consequence, reducing the air gap height enables a better acoustical coupling between woofer and panel. A small panel volume stiffens the first eigenmode predominantly to higher frequencies and hence acts in a significant broadened frequency range.
Moreover, a larger paneled volume is connected to lower chamber modes, leading to a less effective coupling between woofer and panel. As a consequence, these effects occur: the transfer function breaks up earlier, respectively, the systems upper-frequency limit is reduced, and the acoustical performance at low frequencies decreases. This can bee seen in Figure 12(b). The frequency response of a panel volume with an air gap height of 20 mm presents the lowest SPL at higher frequencies upon 250 Hz, and at low frequencies below 50 Hz. Summarized, the panel volume is an important construction parameter, which will improve the frequency response without additional costs.

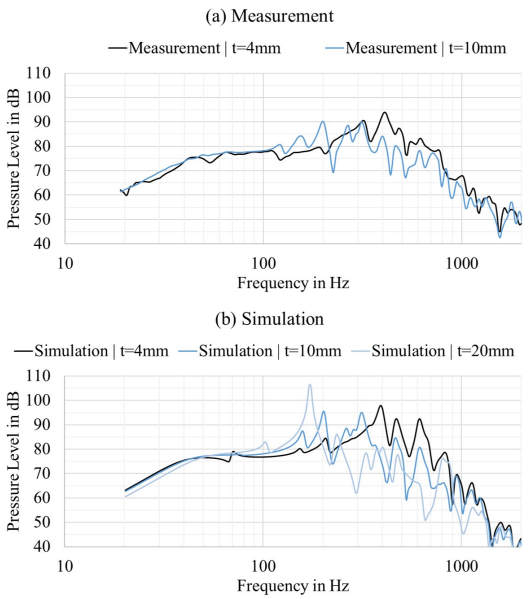


Fig. 12: Comparison of the influence of the panel volume a) measurement and b) simulation.

4.6 Design Considerations

Finally, the investigation is summarized below:

(1) The low-frequency performance is limited to the woofer properties and can't be higher than that.

(2) Additional sensitivity can be generated by the panel modes in the passband.

(3) Reducing the panel volume increases the frequency shift of mode 1, which results in a flatter frequency response in the passband.

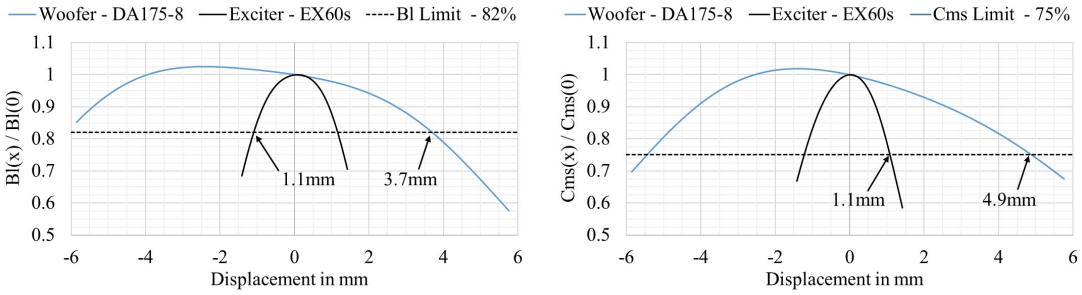
(4) The size of the panel is important. A panel that is too small tends to a higher cut-off frequency at low frequencies, caused by the additional panel stiffness. A larger panel does improve the low-frequency performance.

(5) The panel material has a weak influence. Lighter materials increase the sensitivity at higher frequencies but do not give any improvement for the sensitivity at lower frequencies.

(6) Increasing the damping loss factor will result in a significant flatter transfer function.

(7) It is recommended to use a panel with a certain degree of damping. Otherwise, the modes can occur strong deviations in the frequency response that cannot be compensated with standard filters.

(8) The numerical simulation provides a sufficient estimation for the optimization of the paneled woofer system.



(a) Normalized large signal behavior $Bl_N(x)$

(b) Normalized large signal behavior $Cms_N(x)$

Fig. 13: Normalized force factor a) $Bl(x)$ and compliance b) $C_{ms}(x)$ of woofer DA175-8 and exciter EX60s and the displacement limits X_C and X_{Bl}.

5 Large Signal Identification

The work of Thiele and Small marks a milestone in systematic loudspeaker design [8, 9]. Based on these results, nonlinearities in woofers have been investigated for a long time and large-signal models have been developed, which describe the generation of distortion and other nonlinear symptoms at sufficient accuracy [5, 18]. Since the displacement is high, the variation of force factor $Bl(x)$, stiffness $Kms(x)$ and inductance $Le(x)$ are the dominant nonlinearities in most woofers [19]. Understanding the causes and effects of nonlinearities is a crucial aspect to improve any loudspeaker system.

In the following section, the large-signal behavior of a woofer is compared to an exciter. Furthermore, the woofer mounted in different constructions to visualize the impact of the additional panel stiffness due to the large signal behavior of the displacement dependent compliance.

5.1 Large Signal Identification of Drivers

The transfer function $|H(j\omega)|$ of a loudspeaker driver in an infinite baffle is the product of three partial aspects of the acoustic radiation behavior $|H_{ac}(j\omega)|$, the mechanical structure $|H_{mech}(j\omega)|$, and the electrical transmission system $|H_{electr}(j\omega)|$:

$$|H(j\omega)| = |H_{ac}(j\omega)| * |H_{mech}(j\omega)| * |H_{electr}(j\omega)| \tag{11}$$

The important part in this context is the mechanical transfer function $|H_{mech}(j\omega)|$ to which applies

$$|H_{mech}(j\omega)| = \frac{Bl}{\sqrt{(\frac{1}{\omega C_{ms}} + \omega M_{ms})^2 + {R_{ms}}^2}} \tag{12}$$

The causes of nonlinear distortions by the displacement dependent changes of the respective parameters are described by Klippel [20]. The primary non-linearities are a variation of the compliance $C_{ms}(x)$ and the force factor $Bl(x)$. The effect of Bl is obviously present throughout the entire band. In contrast, $C_{ms}(x)$ is limited to frequencies below $2f_s$. Klippel [20] describes high harmonic distortion in sound pressure for both nonlinearities, $Bl(x)$ and $Cms(x)$, in the frequency range of $f < 2f_s$. Distortions generated by the moving mass M_{ms} and the modal damping R_{ms} do not need to be considered.

Figure 13 shows the displacement related compliance of the Dayton DA175-8 woofer and a Visaton EX 60 S exciter. Both curves are presented normalized to their small signal value. The displacement limit X_C is reached when the compliance has dropped to 75 % of the initial value $C_{ms}(0)$. Further, an asymmetrical behavior of the woofer's suspension and symmetrical behavior of the exciter's suspension can be seen. However, the woofer allows a much larger displacement limit with 4.9 mm compared to the exciter with 1.1 mm.

Similar results are achieved by comparing $X_{max}(Bl)$. The small-signal value of Bl is limited to 82 % in order to achieve THD of 10 % [21]. Figure 13 visualizes similar results for $Bl(x)$. The woofer has a much larger displacement limit of 3.7 mm compared to the exciter with 1.1 mm.

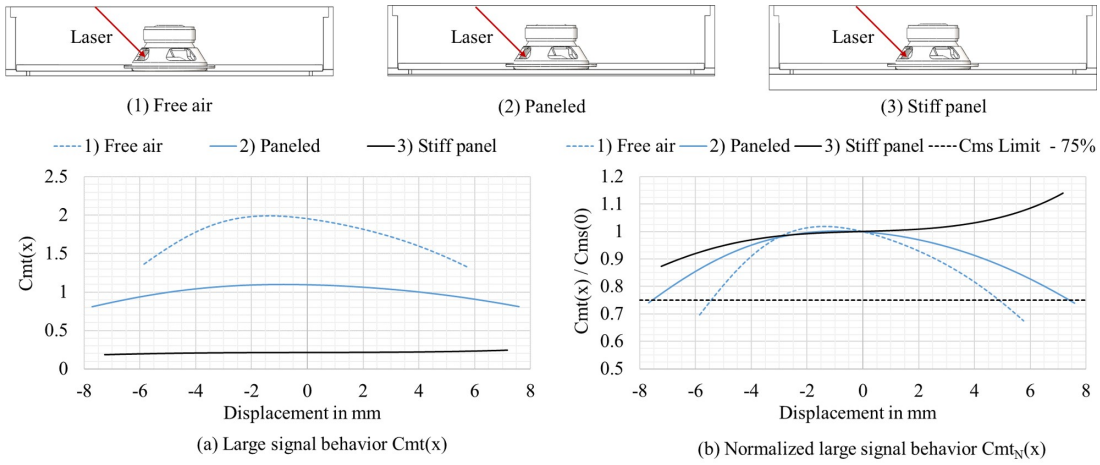


Fig. 14: (a) Total compliance of a free woofer, combined with mineral panel and combined with a fixed panel (b) Normalized compliance.

5.2 Large Signal Identification of a Woofer Mounted in Several Systems

In order to understand the driver's behavior with and without an attached panel, the LSI analysis of the driver was performed under different conditions. By removing the rear side of the prototype, the driver displacement could be measured from behind via a laser, even if a panel was attached at the front. The deviation resulting from the necessary angular (shown in Figure 14 top) has already been corrected in the following considerations. Since the back volume is not closed at the backside, the compliance C_{MB} of the box volume does not apply here accordingly.
Case 1 in Figure 14 shows the compliance of the Dayton DA175-8 in free configuration, as seen before, compared to a combination of woofer and mineral panel coupled by a small panel volume. This chamber adds further stiffness to the driver. Figure 14 also shows the characteristics of a special configuration, in which the woofer drives against a rigid panel, allowing a displacement only by a compression of the sealed air volume. The graph b) in Figure 14 visualizes that the added stiffness increases the displacement limit X_C of the woofer. By placing a mineral panel such as in case 2 - or, in the extreme variant, the stiff panel (case 3) - in front of the driver, the driver gets additional compliance, which shifts the $C_{MS}(x)$-curve downwards. It follows that the critical value $X_{MAX}(C_{MS})$ is always shifted to a larger displacement of x by adding an additional stiffening.

5.3 Conclusions

Following, the results of the large-signal analysis of the exciter and the woofer for the different configurations are given below:

(1) Conventional drivers tend to have higher displacement limits than exciters. This is due to the larger installation space for a larger magnet and the higher stiffness of the exciter spider to stabilize the magnet's large moving mass.

(2) The nonlinearity of the air compliance is low.

(3) The additional panel decreases the maximum displacement of the driver caused by the additional compliance.

(4) A nonlinear effect of the panel could not be detected. Rather, the panel tends to increase the nonlinear compliance limits of the conventional woofer.

(5) A driver with a softer and a more nonlinear K_{ms} can be chosen. The stiffness can be linearized by additional panel stiffness.

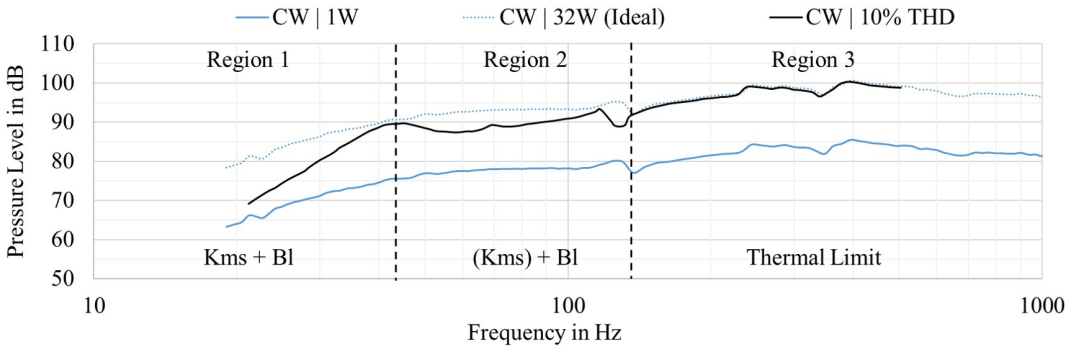


Fig. 15: Maximum SPL for 10 % THD measured at 1 m distance for the conventional woofer design (CW) with a Dayton DA175-8 driver.

6 Maximum SPL for given THD

In order to determine the maximum level of a speaker system, many approaches can be followed, which are specified in AES2-2012 and ANSI/CEA-2010-B [22, 23]. These methods evaluate all nonlinear distortions, harmonic and intermodulation distortions by using a multi-tone signal. However, it is known by using a single tone as a stimulus, it is not sufficient to identify all loudspeaker nonlinearities. This paper focuses primarily on the low-frequency performance, in which the dominant distortions are harmonic distortions. Intermodulation distortions only occur in the combination of a low and high tone, which are more important for mid- and higher frequencies. The used method to calculate the maximum SPL for a given 10 % THD value, visualizes as a worst-case scenario of the SPL limit of each frequency in the frequency range from 20 Hz to 500 Hz. This method extrapolates the 1 W measurement to the thermal limit of each driver. This value is reached if the THD is below 10 %, otherwise the value is lower. This method illustrates the amount of harmonic distortion to the SPL max limit.

6.1 Maximum SPL for a Conventional Woofer

Figure 15 shows the determined maximum level under the assumption that 10 % THD is not exceeded. It also shows the characteristic curve at 1 W input power, measured at a distance of 1 m (sensitivity). This curve was also extrapolated to 32 W and, thus, represented the ideal course of the loudspeaker when no distortion would occur. It is known that the driver has a thermal limit of 50 W. In order to protect the system and the small influence due to the results, the maximum power was limited to 32 W. It can be seen that the maximum achievable output of the driver only reaches the thermal limit starting at 140 Hz (region 3). Below this limit enormous differences can be observed. As already discussed in chapter 5.1, the influence of the stiffness K_{ms} is greatest at the lower end of the transmission band (region 1) and less in the mid-low range (region 2). In contrast, the force factor has the same effect in both regions.

6.2 Maximum SPL for a Paneled Woofer

The comparison of the maximum SPL of the conventional woofer with that of the paneled woofer is presented in Figure 17b. This Figure presents only a small difference at low frequencies. Therefore, it is concluded that the panel does not determine the maximum level in the low-frequency range. Furthermore, the woofer's nonlinearities and the resulting distortions are the limiting factors. Obviously, the panel works in its linear range and produces practically no additional distortions.

6.3 Maximum SPL for a Panel with Exciter

Figure 16 shows the maximum measured level of the design consisting of panel and exciter. Starting at a frequency similar to that observed in the closed woofer design, this level follows the thermal limit of the transducer from about 115 Hz (region 3). Region 2 shows an ambivalent picture: On the one hand, the maximum level could be reached in the frequency band from

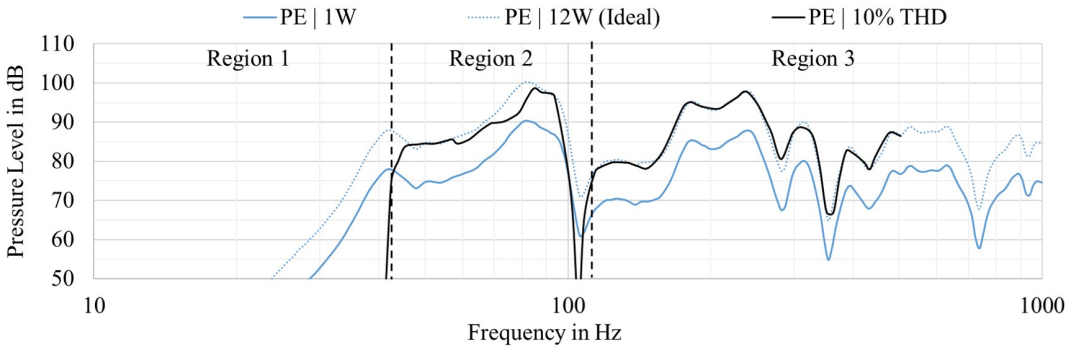


Fig. 16: Maximum SPL for 10 % THD of a panel with exciter design (PE) with a Visaton EX 60 S measured in 1 m distance.

50 Hz to 90 Hz. On the other hand, there is a large dip at 105 Hz, which limits the maximum SPL due to the strong THD. Below the resonance frequency of the exciter, the maximum level drops steeply (region 1). The magnet has a high movement, without coupling to the front. This results in high distortions and a low fundamental. The resulting distortions, however, are transferred to the panel and radiate more efficiently compared to the fundamental, which increases the THD and massively reduces the maximum achievable level. An in-depth illustration of the displacement of the exciter magnet and the panel can be found in part 1 of this paper series [7].

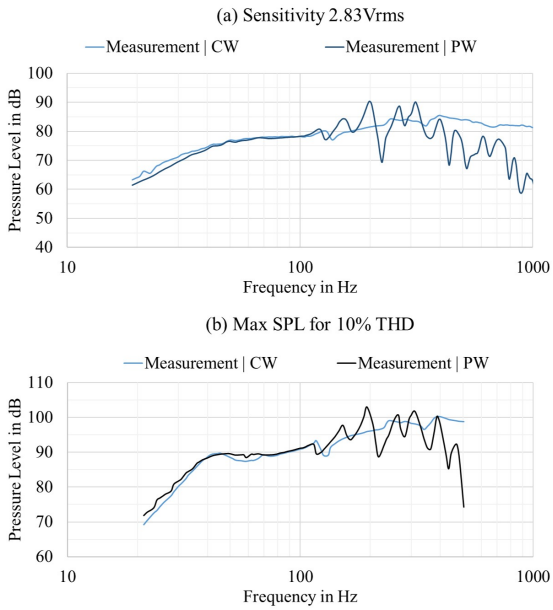


Fig. 17: Sensitivity and maximum SPL of paneled woofer (PW) with 0.3 m^2 radiating surface compared with a conventional woofer (CW).

6.4 Conclusions

Following a summary of the large-signal results in the lower frequency range is given:

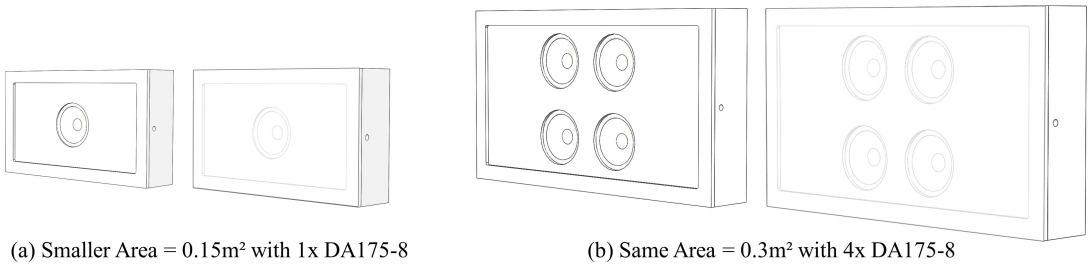
(1) The maximum SPL of a paneled woofer is limited due to the nonlinear behavior of the woofer itself.

(2) The exciter generates a lot of distortion below the resonance frequency. This is caused by the low fundamental and large magnet movement. To avoid low-frequency distortions, the magnet needs to be fixed.

(3) The panel mode increases the sensitivity and the maximum SPL in the frequency band of the mode.

(4) Modes can be a disadvantage for other frequency bands, which can be a part of the 2nd or 3rd order harmonics. Thus, distortions are radiating more efficiently, compared to the fundamental. This amplifies the distortion components and reduce the overall level.

(5) Dips tend to cause a lot of distortions. The driver or the exciter generates the same distortions, by the same displacement. The low pressure level of the fundamental increases the pressure level of the distortion indirectly, those a radiating more efficient. Therefore, it is not recommended to correct dips with a DSP. Dips need to be corrected by the speaker design.



(a) Smaller Area = 0.15m² with 1x DA175-8 (b) Same Area = 0.3m² with 4x DA175-8

Fig. 18: Visualization of both special cases: a) Smaller panel area with an area of 0.15 m^2 and b) with four woofers radiating on an area of 0.3 m^2.

7 Special Cases

This chapter presents two special of the presented prototype design. It will be investigated how the system behaves if several drivers provoke a higher panel displacement. Furthermore, the effect of an increased panel stiffness is presented. In this case, a smaller prototype was manufactured. The smaller radiating surface leads to a higher panel stiffness. Both special cases are illustrated in Figure 18.

7.1 Panel Limit trough Panel Size

Concluding from the above explained considerations, based on the 0.3 m^2 prototypes, it could be assumed that any level limitations are imposed by the large signal properties of the driver. To investigate the effects of a massively increased panel stiffness, the measurements were repeated on another prototype with 0.15 m^2 panel surface. Based on the assumption that the increased stiffness also allows for a larger driver displacement, it is surprising that the maximum level of this configuration is reduced. In contrast to the large prototype shown in Figure 17 (a), the 0.15 m^2 version has a reduced output in the low frequency range. Above 75 Hz the paneled woofer design has the same level as the conventional woofer design (Figure 19), and no additional stiffness effect occurs. This implicates that distortions caused by the driver itself are transmitted more effectively since the resulting overtones tend to be reproduced better than the corresponding fundamental frequency. The fundamental is blocked by the high panel stiffness, and the produced driver's THD can radiate more efficient. This increases the THD and lowers the maximum level. In that sense, this prototype configuration is of poor design.

7.2 Panel Limit trough Displacement

By loading the large prototype with four Dayton DA175-8 the behavior at larger displacement of the panel could be investigated. Figure 20 again shows the sensitivity and the maximum level in comparison. Here, it is particularly clear that the paneled woofer's sensitivity is reduced while the maximum level is identical. Obviously, by increasing the effective stiffness of the

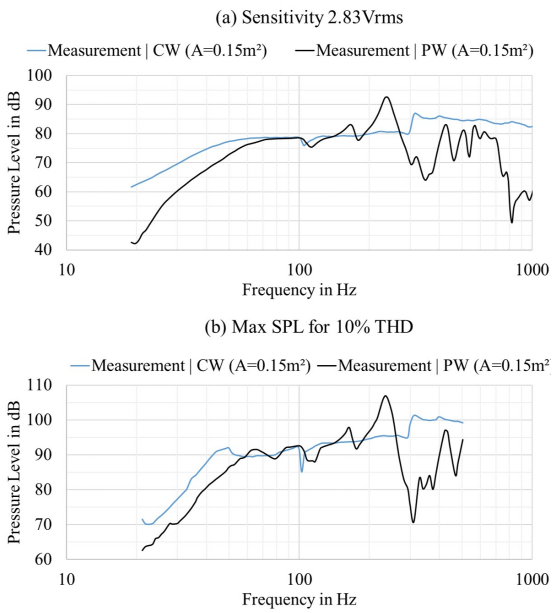


Fig. 19: Comparison of the result of the conventional woofer (CW) system and the paneled woofer (PW) system and a panel area of 0.15 m^2. Figure a) visualizes the voltage sensitivity for 2.83 Vrms at 1m distance and b) the maximum SPL for 10 % THD at 1 m distance.

driver's suspension due to the panel, a higher displacement limit can be achieved before 10 % THD is exceeded. This is already discussed in section 5.2. Since the sensitivity is reduced over a wide range (except the resonance frequency range), distortion products of the driver are not overemphasized, as in the case of the reduced prototype format in chapter 7.1. Consequently, it can be concluded that the panel does not create any additional distortions at these levels of displacements. Rather, the stiffening by the panel and the panel volume increases the usable displacement limits $X_{max}(C_{MS})$ of the distortion components due to the compliance.

7.3 Design Considerations

From the current state of research, it can be assumed that the panel itself generates no nonlinear distortions. If the panel is too stiff, it will reduce the sensitivity in the bass range. This increases the THD level and decreases the maximum level because the harmonics will radiate more efficiently compared to the fundamental. However, a softer panel allows a similar sensitivity at low frequencies as a conventional design, but will not increase the total nonlinear compliance $C_{MT}(x)$. Furthermore, the nonlinear behavior of $Bl(x)$ cannot be counteracted, which is why it is advisable to invest in high-quality drivers in this context. The most important conclusions are presented below:

(1) A nonlinear panel displacement limit could not be detected. Even with high displacements, the driver's nonlinearities are causing the distortion.

(2) The panel can limit the maximum SPL indirectly. A stiff panel blocks the fundamental, but the distortions radiate efficiently. Therefore, it should be considered that the additional panel stiffness needs to be low compared to the conventional system.

(3) In the case of a too stiff panel, even a better driver would not provide sufficient results. At this point, it is mandatory to redesign the construction.

(4) Use a driver with high displacement limits. For the paneled woofer design, a more linear $Bl(x)$ curve is more important compared to a linear $C_{MS}(x)$ curve, which gets compensated by the additional panel stiffness.

(5) The panel volume provides additional stiffness, which increases the nonlinear displacement limits of C_{MS}.

(6) No improvement of the conventional system could be achieved for all variants. However, this paneled woofer design provides the possibility to obtain a similar low-frequency response of flat-panel loudspeakers to that of a conventional system.

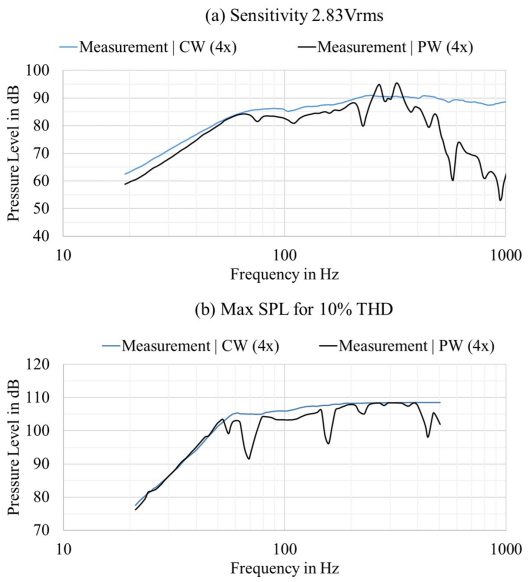


Fig. 20: Comparison of the results of the conventional woofer (CW) system and the paneled woofer (PW) system with four woofers and a panel area of 0.3 m^2. Figure a) visualizes the voltage sensitivity for 2.83 Vrms at 1m distance and b) the maximum SPL for 10 % THD at 1 m distance.

8 Summary

This paper presents a new woofer design for flat-panel loudspeakers. The paneled woofer design makes it possible to achieve similar low-frequency performance as a conventional system.

The development of an electro-mechanical equivalent circuit of a woofer-driven flat-panel loudspeaker presented in this paper is the basis for selective optimization of such a system's performance. The focus is to increase the sensitivity in the low-frequency band. By using a numerical simulation tool, the influence of certain parameters on such a loudspeaker's linear operation could be worked out very precisely. The necessary improvements for better sensitivity of a woofer driven paneled loudspeaker are not very different from the conventional loudspeaker optimization. Suggestions were made in this respect.

In order to optimize the performance of a sound system based on DML technology, investigations of the maximum level under different conditions were carried out. By analyzing the large-signal behavior of a driver with and without an additional panel, it has been proven that an increase of stiffness can counteract the decrease of sensitivity up to some point in terms of maximum possible level. However, an important prerequisite for this is a sufficiently flexible panel, as shown in this paper. If the panel is too stiff, this will affect the fundamental, while the harmonics will be radiated more efficiently. This reduces the maximum sound pressure level. This paper provides an overview of design guidelines to improve the maximum possible SPL. The current research state offers much potential for further investigations. The causes and effects of intermodulation distortions have not been investigated yet.

The paneled woofer design gives no improvements from the acoustical point of view, compared to the conventional woofer design. However, the paneled woofer design provides the possibility to obtain a similar low-frequency response of flat-panel loudspeakers to that of a conventional system. Even if the space requirements of the paneled woofer design are increased, the flat-panel loudspeaker can be installed in locations where it does not disturb the customer. The paneled woofer design offers the possibility of a sufficient low-frequency performance combined with invisible integration.

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