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VIBRATIONAL SOURCE STRENGTH CHARACTERIZATION OF MINIATURE LOUDSPEAKERS USED IN HEARING AIDS

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A variety of specialized miniature loudspeakers are used in hearing aids. The sound from such a loudspeaker is typically let through a tube system from the loudspeaker to an earmould placed in the ear canal of the user. Unfortunately some loudspeakers vibrate heavily during operation and this generates an unwanted transmission of vibrations and sound back to the microphones resulting in feedback. To minimize feedback it is important to know both the *acoustical* and the *vibrational source strength* of the loudspeaker. Methods for characterizing the *acoustical* source strength have been known for more than twenty years. Nonetheless, similar practical methods considering the *vibrational* source strength are still absent in the open literature. The present paper therefore presents a practical “black box” method for characterizing and determining the vibrational source strength. Traditional methods for characterizing the vibrations of machinery generally use the free vibration velocities of the source as a measure of the internal excitation forces. In miniature loudspeakers, however, the internal excitation forces are strongly dependent on the acoustical load impedance experienced at the spout. Nevertheless, by expressing the free vibration velocities as a function of the acoustical load impedance, acoustical and vibrational source strengths can be separated. In this paper simulations of the vibrations of a loudspeaker typically used in hearing aids is compared to measurements. A finite element model of the loudspeaker, which includes coupled vibrations and acoustics, further substantiates the results.

1. Introduction

Modern hearing aids are met with several demanding aesthetic requirements such as minimal physical size and visibility. This has led to designs where the hearing aid casing is very small and where the loudspeaker and the microphones are placed closely together. As a consequence, problems with transmission of sound and vibrations from the loudspeaker to the microphones easily occur during operation and this causes feedback at certain critical gain levels. The vibroacoustic transmission often constitutes the limiting factor for the maximum obtainable amplification in the hearing aid and it therefore represents a critical design problem[1].

Feedback problems in hearing aids are caused by the high sound pressure and strong vibrations generated by the loudspeaker. It is therefore important to be able to characterize and determine both the *acoustical* and the *vibrational* source strengths of these miniature loudspeakers. A so-called “black box” method for characterizing and determining the acoustical properties of hearing aid transducers was presented by Egolf and co-workers[2-4] more than twenty years ago. Nonetheless, similar practical methods considering the vibrational source strength, which is stongly coupled to the acoustics, are still absent in the open literature. This paper therefore presents an extension of the "black box" method for characterizing and determining the vibrational source strength. The method is formulated in the frequency domain for time harmonic forcing.

Hearing aid companies use a variety of specially designed miniature loudspeakers of different sizes and acoustic performance. The principle of all these loudspeakers, however, is generally the same and is also known the "balanced armature" construction. Figure 1 shows a sketch of the particular balanced armature loudspeaker considered in this paper, which is a type 26A03 from the manufacturer Pulse. The loudspeaker consists of an outer housing with spout containing the armature, magnets, coil, diaphragm, drive rod and membrane. Furthermore, it has a length of approximately 6 mm and a total weight of 0.2 g. When voltage is applied to the terminals of the loudspeaker, a current is introduced in the coil. The coil surrounding the armature hereby generates a magnetic field polarizing the armature. At the same time the tip of the armature is placed between the magnets and magnetic forces are therefore formed. These forces produce displacements of the upper part of the armature which are transmitted to the membrane through the drive rod. Finally, the motion of the membrane produces acoustic pressure both in the cavity below and above the membrane. While the pressure waves in the upper cavity are transmitted through the spout and to the succeeding tube system, the small amount of air in the lower cavity imposes a spring-like effect on the membrane.

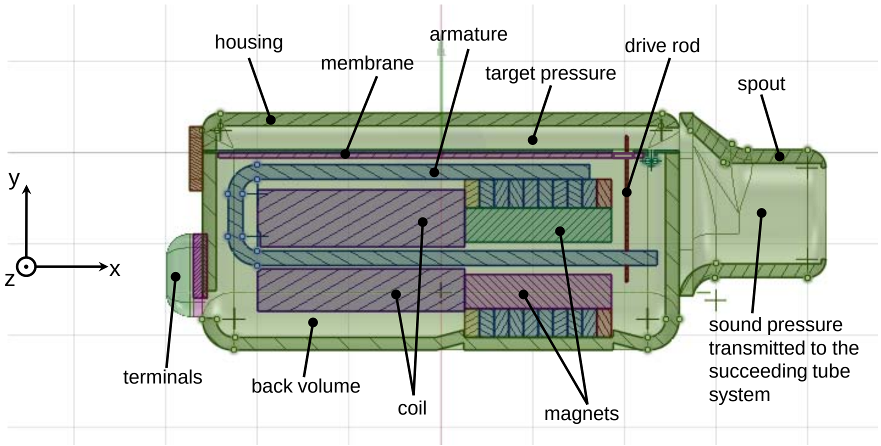

Figure 1. Type 26A03 miniature loudspeaker from Pulse.

The tube system, which transmits the sound to the earcanal of the user, acts as a significant acoustic load on the loudspeaker and therefore affects the inner driving forces of the loudspeaker[5]. This also implies that the pressure and vibrations generated by the loudspeaker depend on the dimensions of the tube system. Depending on the type of the hearing aid the tube system may vary in length between about 10 mm and 80 mm. Surprisingly to many, however, the internal forces and hereby the acoustics are practically independent of outer mechanical loading of the loudspeaker housing. This special feature, which has been observed from experiments, may be utilized for "separating" the acoustical and vibrational source strengths of the loudspeakers. Traditional methods for characterizing the vibrations of machinery[6] such as engines, gearboxes etc. generally use the mechanically free vibration velocities of the source as a measure of the internal excitation forces. By expressing the loudspeaker's free vibration velocities as a function of the acoustic load from the tube system, the use of known methods becomes possible.

In favour of the reader, the method of characterizing the acoustical source strength of miniature loudspeakers will be introduced briefly in the next section. Hereafter follows a presentation of the vibrational source strength method developed by the authors. In Section 3 the method is examined through a virtual experiment considering a finite element model of the considered loudspeaker. The model both includes the electrical, acoustical and vibrational properties of the Pulse 26A03 loudspeaker. Measurements and simulations of the acoustics and vibrations of the real loudspeaker are then finally compared in Section 4.

2. Method development

2.1 The acoustical source strength

The main acoustic system of a hearing aid typically consists of the loudspeaker, different tubes, ear canal and ear drum[1], and the inside acoustics of this system can be modelled by using transmission line theory and more specifically "two-port networks"[2-4]. This method is a way to account for the impedance loading between components in any linear system. Each component participating is regarded as a "black box" and only the input and output parameters are considered. As an example, the source strength of any linear miniature loudspeaker are described in terms of the four "two-port parameters", A_{loud}, B_{loud}, C_{loud} and D_{loud} relating the output pressure p_{loud} and volume velocity Q_{loud} produced at the spout to the input driving voltage U and current I as

$$\begin{Bmatrix} U \\ I \end{Bmatrix} = \begin{bmatrix} A_{loud} & B_{loud} \\ C_{loud} & D_{loud} \end{bmatrix} \begin{Bmatrix} p_{loud} \\ Q_{loud} \end{Bmatrix}. \tag{1}$$

Assume that the loudspeaker is connected to a tube system terminated by a so-called coupler or artificial ear representing the acoustic properties of the average adult ear. The two-port parameters of the whole system relating the pressure p_{coup} and volume velocity Q_{coup} in the coupler to U and I are found by multiplying the individual transmission matrices of the three components as

$$\begin{Bmatrix} U \\ I \end{Bmatrix} = \begin{bmatrix} A_{loud} & B_{loud} \\ C_{loud} & D_{loud} \end{bmatrix} \begin{bmatrix} A_{tubes} & B_{tubes} \\ C_{tubes} & D_{tubes} \end{bmatrix} \begin{bmatrix} A_{coup} & B_{coup} \\ C_{coup} & D_{coup} \end{bmatrix} \begin{Bmatrix} p_{coup} \\ Q_{coup} \end{Bmatrix}. \tag{2}$$

By using these two-port parameters it is possible to calculate the pressure anywhere in the tube system for a given voltage U driving the loudspeaker. Also the current I, which is highly dependent on the acoustical load represented by the tube system, can be predicted.

Closed form expressions for the two-port parameters of narrow tubes including viscous-thermal losses were presented in final form by Egolf and co-workers nearly twenty years ago[2-4]. Moreover, the two-port parameters of the loudspeaker and the coupler are typically supplied by the manufacturer in the form of *equivalent networks*. An alternative and more precise way of determining the two-port parameters of the loudspeaker is to use the experimental "two-load method"[2-4]: Using this method the loudspeaker is loaded with two different acoustical loads with known properties and the pressure generated in a coupler and the loudspeaker current is measured for a known driving voltage. This yields four equations with the four two-port parameters as unknowns. The vibration method considered in the present paper, may be regarded as an extension of the two-load method to include vibrations.

2.2 The vibrational source strength

2.2.1 The mechanically free vibration velocities

To enable the use of traditional methods of vibrational source characterization on miniature loudspeakers, it is necessary to separate vibrations from acoustics. The authors therefore suggest that the internal driving forces and hereby also the free mechanical vibration velocities are given as a function of the loudspeaker driving voltage and current. This assumption, which will be substantiated in the following sections, is convenient as the driving voltage and the current are accessible during experiments. Moreover, these two frequency dependent quantities are a result of the full interplay between the loudspeaker and the specific acoustic tube system connected to the spout. An experimental setup enabling measurements on a miniature loudspeaker under the conditions acoustically loaded and mechanically free will be presented in Section 4.

Consider the free mechanical velocity of the loudspeaker $v_{free,n}$ at the "terminal" n. The term terminal here refers to an arbitrary motion degree-of-freedom (translational motion and rotational motion etc.) at an arbitrary position on the loudspeaker housing. It is assumed that this velocity is related to the driving voltage U and current I of the loudspeaker as

$$v_{free,n} = A_n U + B_n I \,, \tag{3}$$

where A_n and B_n are the frequency dependent vibrational two-port parameters for terminal n. By analogy to the acoustic two-port parameters of the loudspeaker, the vibrational two-port parameters for terminal n can be determined by measuring the driving voltage, current and free velocity for two different acoustical load cases 1 and 2. By using the expression in Eq. (3) we then get the following two equations with the two-port parameters as unknowns:

$$v_{1,free,n} = A_n U_1 + B_n I_1 \quad \text{and} \quad v_{2,free,n} = A_n U_2 + B_n I_2 \,. \tag{4),(5}$$

Here index 1 and 2 refers to the acoustic load cases and by combination of the two equations, the two-port parameters yield

$$A_n = \frac{v_{2,free,n} - v_{1,free,n}(I_{n,2}/I_{n,1})}{U_{n,2} - U_{n,1}(I_{n,2}/I_{n,1})} \quad \text{and} \quad B_n = \frac{v_{2,free,n} - v_{1,free,n}(U_{n,2}/U_{n,1})}{I_{n,2} - I_{n,1}(U_{n,2}/U_{n,1})} \,. \tag{6),(7}$$

By using these two-port parameters and Eq. (3) the free velocity for terminal n can be simulated for any acoustical load. In order to determine the current for a given driving voltage it is necessary to know the acoustic two-port parameters. It is therefore an advantage to measure the pressure in a coupler during the vibration experiment. This allows for the determination of the acoustic two-port right away.

2.2.2 Mechanical coupling to a receiving structure

When a miniature loudspeaker enters into the construction of a hearing aid, it is connected mechanically at the spout to the following tube system and often also in other positions. Naturally, these mechanical connections load the loudspeaker, and its velocities are altered. Let us consider a miniature loudspeaker that is connected to a hearing aid construction through N connection terminals. As mentioned before, these terminals might have different motion-degrees-of-freedom and refer to one or more positions. When the loudspeaker is mechanically free it has the vibration velocities at the connection terminals denoted by $v_{free,n}$ where $n = 1, 2, \ldots N$ These free velocities are determined from Eq. (3) and are ordered in the column vector $\{v_{free}\}$. The free velocities are generated by unknown internal forces. The excitation of the internal forces is therefore modeled by N *equivalent* internal forces $F_{int,m}$ acting at the terminals and ordered in the column vector $\{F_{int}\}$. As the equivalent internal forces results in the free velocities $\{v_{free}\}$ it applies that[6]

$$\{v_{free}\} = [C]\{F_{int}\} \,, \tag{8}$$

where $[C]$ is the N by N element mechanical mobility matrix of the loudspeaker containing the terminal mobilities

$$C_{n,m} = v_{free,n} / F_{int,m} \quad \text{where} \quad n = 1, 2, \ldots N \ \text{and} \ m = 1, 2, \ldots N \,. \tag{9}$$

Moreover, it is assumed that the properties of the receiving structure are known and that they are given in terms of the N by N element mechanical mobility matrix $[D]$ defined by

$$D_{n,m} = v_{rec,n} / F_{rec,m} \quad \text{where} \quad n = 1, 2, \ldots N \ \text{and} \ m = 1, 2, \ldots N \,. \tag{10}$$

Using mobility synthesis[6,7] the N terminal velocities $\{v\}$ for the loudspeaker connected to the receiving structure become

$$\begin{aligned}\{v\} &= [C]\{F_{int}\} - [C]\{F\} \\ &= v_{free} - [C]\{F\}\end{aligned}, \tag{11}$$

where $\{F\}$ are the N unknown coupling forces at the terminals. Due to continuity it also applies that

$$\{v\} = [D]\{F\} \tag{12}$$

and by combination with Eq. (11) the coupled velocities at the terminals $\{v\}$ becomes

$$\{v\} = \left([\mathrm{I}] + [C][D]^{-1}\right)^{-1}\{v_{free}(U,I)\}, \tag{13}$$

where $[\mathrm{I}]$ is the N by N identity matrix and superscript -1 denotes a matrix inversion. Also the coupling forces can be determined by combination of Eqs. (10) and (11). The exact terminal mobilitiy matrix of the loudspeaker $[C]$ is typically difficult to measure. Due to the small size of the loudspeakers, however, their mechanical properties can often be replaced by those of a solid mass with the same outer dimensions and weight. In Section 4, it will be shown that this is a good approximation within the frequency range of interest.

3. Numerical simulations

3.1 The simulation model

As a virtual experiment a simplified finite element (FE) model of the Pulse 26A03 loudspeaker shown in Fig. 1 was developed using the Comsol Multiphysics software (ver. 3.5a). Using such a model it is possible to 1) study the crucial assumption given in Eq. (3) under ideal conditions 2), examine the effects of coupling the loudspeaker to a receiving structure and 3) investigate the influence of different mechanical and acoustical couplings.

In the simulation model, linear elastic solids are coupled to lossless acoustics. Solutions thus include the displacements in the solid in the directions of the coordinate system shown in Fig. 1 as well as the acoustic pressure in the fluid domain. The equations are formulated in the frequency domain with time harmonic forcing. An expression for the magnetic force exciting the membrane, which is a function of the driving voltage and the membrane displacement, has been extracted from equivalent networks supplied by Pulse for the specific loudspeaker considered. Finally, an acoustic impedance boundary condition at the spout of the loudspeaker was used to model the load of different succeeding tube systems.

3.2 Simulation results

Two different acoustical loads (load cases 1 and 2) were used to determine both the acoustical and vibrational two-port parameters. The calculated two-port parameters were utilized for predicting the pressure and vibration velocities for a third load case (load case 3). All three acoustic loads are a three-piece tube system terminated by a 711 coupler[5]. The different dimensions of the included tubes are given in Table 1 and they correspond to the experimental setup used in Section 4. Note that tube 1 is connected to the spout of the loudspeaker and that tube 3 is the spout of the coupler. In figure 2, simulation results from the FE model and the method using two-port parameters are compared. From Fig. 2a it is seen that the two-port prediction of the coupler sound pressure for load case 3 is practically identical to the FE result. The pressure simulated by the FE model for load case 1, however, only show two resonances. This is because of the short length of the tube system, which is only 13.8 mm. In Fig. 2b it is also seen that the two-port prediction of the vibration velocity at the spout is coinciding with the FE-results. This agreement proofs that Eq. (3) is a reasonable assumption. Again the vibration velocity for load case 1, which is used for predicting the vibrational two-port parameters, is shown as a reference. Not surprisingly, the vibration velocities in the two load cases show resonances at the same frequencies as the corresponding pressure responses.

Table 1. Dimensions of the tube systems in mm used as acoustical loads.

		Load case 1		Load case 2		Load case 3	
		length	diameter	length	diameter	length	diameter
Tube 1	resilient tube	12.8	1.6	12.8	1.6	12.8	1.6
Tube 2	plastic tube	0	0	48.5	0.7	23.6	0.7
Tube 3	coupler spout	1	1	1	1	1	1

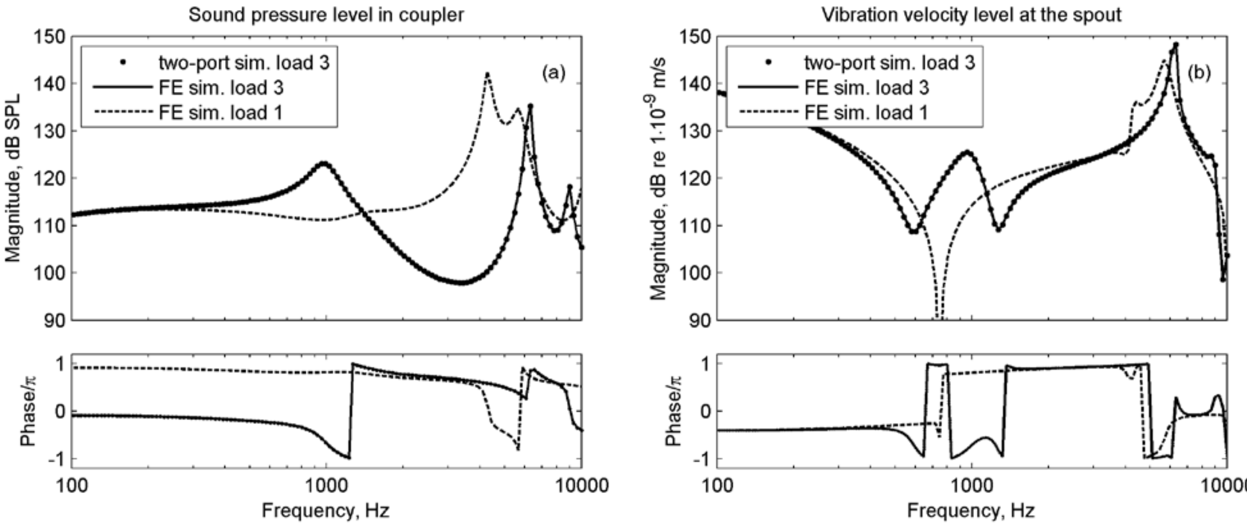

Figure 2. Simulation of (a) the sound pressure level in the coupler and (b) the free vibration velocity at the spout. All results are relative to 1 V driving the loudspeaker.

Next the loudspeaker was coupled mechanically to a receiving structure at the spout. The particular receiving structure considered is a plastic tube with the length 50 mm and outer and inner diameters of 2 and 1 mm, respectively. The material properties are Youngs modulus 3 GPa, density 1050 kg/m^3 and structural loss factor 0.01. The tube is considered to be clamped at the far end and its mobility properties are given in ref. 6. The vibration velocities at the spout when the loudspeaker is mechanically free and coupled to the plastic tube are shown in Fig. 3. Again load case 3 has been chosen as acoustical load. The velocities have been predicted by using the determined two-port parameters and Eq. (13). The loudspeaker has two terminals at the spout, which relate to the coordinate system in Fig. 1 and these are translational motion in the *y*-direction and rotational motion around the *z*-axis. Due to symmetry there is no translational motion in the *z*-direction. The coupled translational velocity in the *y*-direction at the spout has been predicted using the actual mobilities of the loudspeaker (FE simulation) as well as the properties of a solid mass with same dimensions and weight. From the velocities shown in Fig. 3 it is clearly seen that the simple mass approximation of the loudspeaker performs very well. As expected the

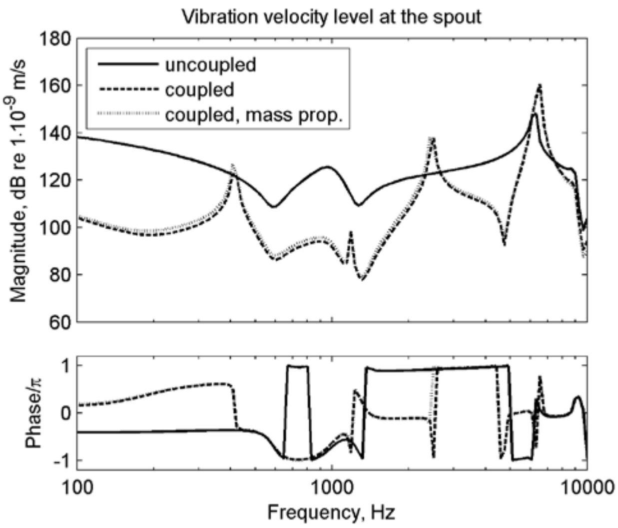

Figure 3. Two-port simulations of the uncoupled and coupled velocities at the spout.

coupled velocities are considerably lower than the uncoupled velocities at low frequencies and since the whole system is spring-controlled the phase is shifted. It is also seen that the coupled velocity level becomes higher than the uncoupled velocity level at certain frequencies since the loudspeaker is subjected to the resonant properties of the receiving structure.

4. Pressure and vibration measurements

4.1 Experimental setup

The virtual experiment using finite element simulation presented in the previous section showed that the method using vibrational two-port parameters works in theory. More practical issues, however, are still to be investigated.. For that particular reason, an experiment on an actual loudspeaker of type 26A03 was performed. This experiment involved measurements of the free terminal velocities using a laser vibrometer, the sound pressure in a coupler, the input driving voltage and the current. All these quantities were measured for the three load cases previously described in Table 1 using a Pulse spectral analyzer from B&K. The experimental setup used is shown in Fig. 4. The miniature loudspeaker is attached at the spout to a very resilient rubber tube extended by a transparent plastic tube. The plastic tube is terminated by a coupler, which has a small spout. As seen in Fig. 4 the resilient rubber tube is barely capable of carrying the small weight of the loudspeaker of approximately 0.2 g. In practice, this implies that the loudspeaker can be regarded as “mechanically free in space” in the frequency range of interest. Acoustically, however, the loudspeaker is significantly loaded by the whole tube system. From measurements it has been confirmed that the sound pressure inside the soft-walled tube system is practically the same as for a completely stiff-walled tube system.

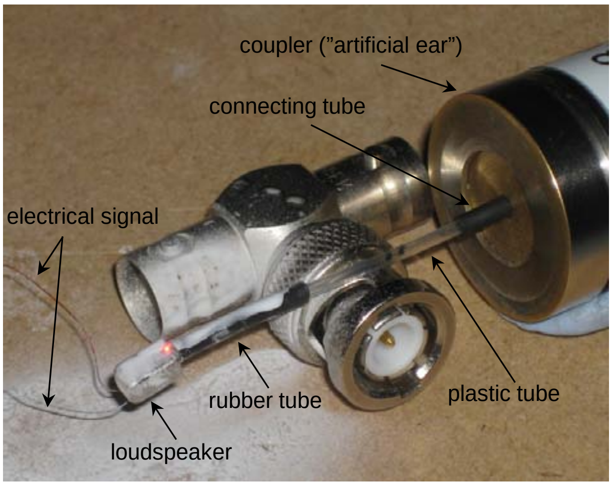

Figure 4. Experimental setup where the loudspeaker is mechanically free and acoustically loaded.

4.2 Comparison of simulations and measurements

The acoustic and vibrational two-load parameters of the miniature loudspeaker from Pulse were determined using the methods outlined in Sections 2.1 and 2.2.1, respectively. Again load cases 1 and 2 have been used to determine the two-port parameters. Using these parameters the sound pressure in the coupler and the vibration velocities at the spout were predicted for load case 3. Note that the loudspeaker has the same two connecting terminals as in the simulations. Fig. 5 shows comparisons of the predicted and measured pressure and free translational velocity for load cases 1 and 3. The simulated and measured pressure responses in Fig. 5a show very good agreement at nearly all frequencies. Only smaller discrepancies of less than 5 dB are seen around two of the four displayed resonances. As a reference, the measured pressure in load case 1 is showed. By comparison of the pressure responses in Fig. 5a with those simulated by the simplified FE-model in Fig. 3a, some similarities are seen. Especially for load case 1, the two pressure curves have the same pattern of resonances. The simulated resonance peaks, however, are significantly higher and are shifted somewhat up in frequency. The discrepancy in magnitude is most likely caused by the loss-less acoustics used in the simulation model of the loudspeaker. Finally, despite of smaller differences Fig. 5b shows clear agreement between the simulated and measured free velocities at the spout for load case 3. It can therefore be concluded that the assumption in Eq. (3) also works in practice. Moreover, as expected, the reference velocity in load case 1 exhibits a much different pattern of resonances. Unfortunately, by comparing the measured vibration velocities in Fig. 5b with the simulated velocities in Fig. 2b no clear resemblance is found. This demonstrates that a FE model needs to be very detailed in order to model actual vibrations of a loudspeaker. Also, it further confirms that “black box” methods as the one presented in this paper are very robust and therefore can be very helpful in the effort to minimize feedback in hearing aids.

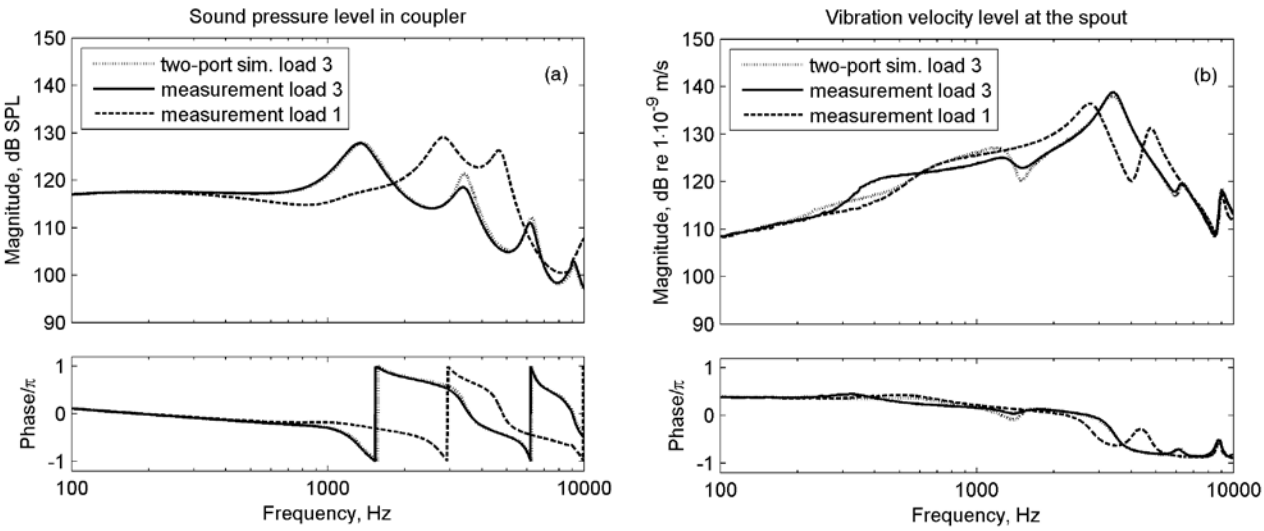

Figure 5. Simulations and measurements of (a) the sound pressure level in the coupler and (b) the free vibration velocity at the spout. All results are relative to 1 V driving the loudspeaker.

5. Concluding remarks

In the present paper, a special “black box” method of describing and determining the acoustical source strength of miniature loudspeakers has been extended to include vibrations. It has been shown that the vibrational source strength can be isolated from the acoustics by expressing the free vibration velocities in terms of the acoustic load experienced by the loudspeaker. Moreover, this separation enables the use of traditional methods of vibrational source characterization on miniature loudspeakers. Both simulations and measurements verify that the suggested method applies in practice and that all assumptions taken are reasonable.

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