Vibration patterns of the front panel of the loudspeaker system : Measurement conditions and results

Ewa B. Skrodzka and Aleksander P. Sęk

*Institute of Acoustics, A. Mickiewicz University,
Matejki 48/49, 60-679 Poznań, Poland*

(Received 28 February 1997)

This paper is concerned with an experimental modal analysis of a loudspeaker system. Based on Frequency Response Functions (FRFs) measured in about 500 points uniformly spaced on the front panel of the loudspeaker system several vibrations modes of the loudspeaker membranes have been found. Then, the vibrations of the loudspeaker membranes were compared with vibrations of the cabinet of the loudspeaker system. It has been shown that the vibrations of the loudspeaker case were very small relative to the vibration of the loudspeaker membranes. Thus, the loudspeaker cabinet, which may be considered as a set of vibrating plates, is rather a source of a very weak sound and may be neglected. It does not influence significantly on the acoustics field radiated by the loudspeaker system.

Keywords: Modal analysis, Vibration patterns

PACS number: 43.40.At

1. INTRODUCTION

1.1 What is a Modal Analysis?

A term *modal analysis* refers to a process of characterising dynamic response of a structure by describing its vibrational motion by means of a set of mathematical relationships, generally referred to as modal properties.¹⁻³⁾ Vibrational modes of a structure can be obtained by means of one of two different approaches: mathematical models or experimental analysis.⁴⁾

Mathematical models generally discretize a structure by dividing it into a number of masses and springs, either by assuming a lumped mass and spring approach or by using the Finite Element Method (FEM) where each element is considered as a mass-spring system. A computer program, then solves the eigenvalue extraction problem to obtain the mass, frequency, damping and mode shapes of each eigenmode of the system. Once the modal properties are determined the system is entirely

described.

Experimental modal analysis on the other hand starts from a set of measured Frequency Response Functions (FRFs) of a structure. The FRF is defined as a ratio of a response signal spectrum and an excitation signal spectrum. Based on these functions the modal properties (*i.e.* modal frequencies, modal damping and mode shapes) are calculated without any specific assumptions concerned with the distribution of mass and stiffness of the structure. The modal model, that consists of the modal properties, is obtained by analytical curve fitting one or more FRFs. The experimental modal analysis, based on FRFs measurements, may be used to describe dynamic behaviour of any vibrating object including a loudspeaker system.

In terms of modal analysis, the FRF for the response X_i , at the point i , due to an excitation force F_k applied at the point k ,^{2,4-6)} has a form:

$$\frac{X_i}{F_k} = H_{ik}(s) = \sum_{r=1}^n \left(\frac{r_{ikr}}{s - p_r} + \frac{r_{ikr}^*}{s - p_r^*} \right), \quad (1)$$

where :

- r_{ikr} : residue for r th mode,
- s : Laplace variable,
- $p_r = \sigma_r + j\omega_r$, for $k = 1, \dots$,
- r : eigenvalue or the pole of the FRF,
- σ_r : modal damping of the r th mode,
- ω_r : modal frequency of the r th mode.

If all points i and k are taken into consideration, then Eq. (1) takes form :

$$H(\omega) = \sum_{r=1}^n \left(\frac{\{u_r\}\{u_r\}^t}{j\omega - p_r} + \frac{\{u_r^*\}\{u_r^*\}^t}{j\omega - p_r^*} \right) \quad (2)$$

The Laplace variable s in Eq. (1) has been replaced by the frequency ω along the frequency axis; r_{ikr} and $\{u_r\}$ are complex quantities. Thus, each mode of vibration is defined by a pair of complex conjugate poles (p_r, p_r^*) and a pair of complex conjugate mode shapes ($\{u_r\}, \{u_r^*\}$); $\{u_r\}^t$ and $\{u_r^*\}^t$ are transpositions of $\{u_r\}$ and $\{u_r^*\}$ respectively. A set of FRF with indices i and k is usually arranged in a matrix called a modal matrix.

1.2 Assumptions

Modal analysis imposes following assumptions on the investigated object :

(1) The system is linear. It means that a response of the system is proportional to an excitation. This assumptions has three following implications for FRFs measurements :

(1.1) Superposition. A measured FRF is not dependent on the type of excitation waveform used.

(1.2) Homogeneity. A measured FRF is independent of the excitation level.

(1.3) Reciprocity. The FRF measured between any two points of the structure is independent of which of them is used for excitation or response. Therefore, only one row or column of the modal matrix needs to be measured in order to identify the modal parameters of the structure. The only exceptions to the above statement are rows and columns corresponding to components of the modal vector which are zero. These are known as the nodal points of the mode shape.

(2) The structure is causal. It does not start to vibrate before it is excited.

(3) The structure is stable. The vibrations die out when the excitation is removed.

(4) The structure is time-invariant. The dynamic characteristics do not change during the measurements.

(5) Modes are defined in a global sense. Mode shapes are defined for all degrees of freedom of the model. Frequency and damping of the mode do not vary significantly from one part of the structure to another.

(6) Only one mode exists at each modal frequency.

These assumptions can be satisfied in a large number of test situations. However, accuracy of the resulting modal parameters depends largely on how closely the analysed system follows them.

In many cases, to simplify calculations, it is necessary to assume that loudspeaker and loudspeaker system are approximately linear systems. However, this assumption is only reasonable true for small displacements of the structure. In this case loudspeaker systems can be treated as linear systems in the whole explored frequency range. The other conditions which modal analysis imposes on the investigated system, as causality, stability and time-invariantness, are undoubtedly met by the loudspeaker system.

1.3 Aim of the Experiment

In classic experimental modal testing a structure is excited at one or more points and the response is determined at one or more points. It was the reason that at the beginning the modal analysis was used for dynamic investigation of large objects,⁷⁾ like cars, ships, aircraft, buildings, mill elements, vibrating machines, *etc.* It was relatively easy to excite the structure by a shaker or by an impact hammer with a force transducer and to measure the response signal by means of an accelerometer attached at some key points.⁸⁾ In this classic form experimental modal analysis is successfully used in acoustics in the investigations of the dynamic behaviour of music instruments like violins, pianos, guitars, handbells, *etc.*⁷⁾

However, application of modal analysis to investigation of objects which mass was comparable to mass of an accelerometer, like commonly encountered in electroacoustics loudspeaker membranes, was almost impossible. It is possible to attach the accelerometer to the loudspeaker membrane, but the response signal determined by the accelerometer will not be the response of the membrane itself, but the response of the membrane loaded by the accelerometer. Such a signal is rather useless in modal investigations. This serious drawback has

disappeared when a non-contact method of detection and measuring the structure response signal has appeared. A laser velocimeter enables successful measurements of the loudspeaker membrane response. It is not necessary to cover the membrane by a reflective material, so the problem of unwanted mass attached to the membrane does not exist.

Then, the use of the laser velocimeter is the only chance of successful modal analysis of loudspeaker membranes and complete loudspeaker systems. However, some more modifications of the classic experimental modal analysis technique have to be made, too (see next paragraph for details.).

In this paper the modal analysis was used to analyse of vibrations of the complete loudspeaker system. The main purpose was to find, visualise and compare all vibrating areas of the loudspeaker system including both the speakers and the front panel of the cabinet of the system. It is well known that the loudspeaker system cabinet can influence the reproduced sound. The effect may be simply due to an interference of reproduced sound from loudspeakers with unwanted sound produced by thrilling surfaces. If areas with unwanted significant vibrations are well known, then it will be possible to eliminate them or, at least, to change their frequency by attaching an additional mass, a damping element or a stiffening element.

2. MEASURING SETUP AND MEASURING CONDITIONS

The object under investigations was a typical ZgP-40-8-85 "Tonsil" loudspeaker system taken by chance from the manufacture. The system consisted of one woofer (GDN 20/40/12) and two tweeters (GDW 9/15/6 and GDW 5/40/9). The dimensions of the system were: $256 \times 456 \times 210$ mm. The loudspeaker system was mounted on a motor-step table controlled by a computer, enabling its positioning. The system was excited by a constant voltage (RMS), flat spectrum, pseudo-random noise. The frequency range of the signal was 0–4 kHz. To preserve a natural working condition of the system the signal was delivered to the loudspeaker's voice-coils. The signal was also delivered to one channel of the dual channel spectrum analyser (ONO SOKKI CF 6400) which was the main part of the measuring equipment.

The response signal from the vibrating body, *i.e.* the changes in the velocity of a vibrating point of the

Fig.1 FRF measured on a point of the woofer's membrane (left column) and corresponding coherence function (right column) after a) 2 averages, b) 10 averages, c) 25 averages.

system, was measured in a non-contact way by means of a laser velocimeter based on a Doppler effect. Velocity was measured in the parallel direction to the main axes of symmetry of the loudspeakers. The position of the velocimeter was fixed and did not change during experiment. The signal from the velocimeter was fed to the second channel of the analyser.

The spectrum of the excitation evoked by the stimulating pseudo-random noise was broadband and contained discrete components whose frequencies corresponded to the frequencies of analysis. A repeatable nature of the stimulating waveform enabled successful application of time averaging procedure for FRFs. So, the signal-to-noise ratio was significantly improved. 25 time averages were used. In Fig. 1 and in Fig. 2 examples of FRFs and corresponding coherence functions obtained after 2, 10 and 25 time averages are shown. It is clear that increasing number of averages improves the quality of FRFs.

The rectangular window, as the most appropriate for pseudo-random noise and transient signals, was used in both measuring channels. The analyser calculated FRFs based on the Eq.(1). Then, FRFs were transferred via GPIB card to a computer and stored for further analyse by means of the SMS *STARModal*® software.

Curve fitting, or parameter estimation, is a numer-

Fig. 2 FRF measured on a point of the middle range tweeter's membrane (left column) and corresponding coherence function (right column) after a) 2 averages, b) 10 averages, c) 25 averages.

ical process that is usually used to represent a set of experimentally measured data points by an assumed analytical function. The results of the curve fitting process are the coefficients, or parameters, that define an analytical function being the best approximation of the data points. With regard to the Frequency Response Function (FRF), the parameters that are calculated are modal parameters. The curve fitting process can also be thought of as a data compression process since a large number of experimental values can be represented by a much smaller number of modal parameters.

In the presented work so-called Advanced Curve Fitting method of modal parameters estimation was used.⁹⁾ This global method is based on both types, time and frequency domain, curve fitting algorithms. In the first step modal frequencies and damping are estimated in the time domain by means of the poly-reference time domain technique.¹⁰⁾ The advantage of this technique is the compact, recursive and extremely stable solution algorithm. In the second phase mode shapes are calculated in the frequency domain using the rational fraction polynomial method.¹¹⁾ This method enables taking into account the contribution of the out-of-band modes. The rational fraction polynomial method, by applying of orthogonal polynomials,¹²⁾ significantly reduces the dimension of the numerical problem making the technique suitable for implementation

Fig. 3 Measurement geometry of the front panel of the "Tonsil" loudspeaker system.

in a mini-computer environment.

Experimental modal analysis of an object is based on a set of measured Frequency Response Functions (FRFs). A distribution and a number of the measuring points applied to the object are very important for the accuracy and the resolution of the analyse. They must reflect the geometry of the investigated object in an optimal and the best possible way. In the presented study the loudspeaker system was characterised by means of 493 points uniformly spaced on the front panel of the system. The distance between two adjacent measuring points was approximately equal 1.5 cm. The distribution of the measuring points, plotted in Cartesian coordinates is given in Fig. 3.

In the carried out experiments it was necessary to adopt the classic modal analysis to measurements performed directly on a loudspeaker.^{13,14)} The classic modal analysis assumes that the excitation signal is a force signal. However, in the presented experiments the membranes were driven by the voice-coil. Thus, it was impossible to fulfil this requirement because the value of exciting force was unknown. The voice-coil voltage was taken as the input variable instead of the force.¹⁵⁾ High values of the input voltage could disturb proportional relation between the voltage and related force. However, this is not the case in here because values of input voltages used in the experiment were rather low (about 1V).

The classic modal analysis assumes also that the stimulating force has a point character. In the case

of our loudspeaker system the woofer's diameter was 156 mm and the diameter of its voice-coil was 25 mm. Thus, for the woofer the ratio of above diameters was about 0.16. The middle range tweeter's diameter was 53 mm and the diameter of its voice-coil was 13.5 mm. Thus, for the middle range tweeter the ratio of above diameters was about 0.25. For the tweeter the ratio was about 0.33. The central point of the woofer's membrane was chosen as the stimulation point in all calculation since the woofer is the main vibrating element in the investigated frequency range. The only way of measuring vibrations of the loudspeaker, that reflects a normal condition of its action, is to excite the loudspeaker by means of its voice-coil. The assumption might be a source of a certain error which may be estimated by means of the multi-point excitation version of the modal analysis.

3. RESULTS

3.1 Results of Modal Analysis

Based on the frequency response functions measured in all 493 points the modal parameters (*i.e.* modal frequencies, modal damping and mode shapes) were calculated by means of the *STAR-Modal*[®] program package. The results of modal analysis are presented in Table 1. Each cell of the Table shows modal frequency from the range of 0–4 kHz, the mode shape and the percentage of critical damping. The percentage of critical damping is defined as ⁵⁾:

$$\zeta_r = \frac{\sigma_r}{\sqrt{(\sigma_r^2 + \omega_r^2)}} \cdot 100\%, \quad (3)$$

were:

σ_r : damping value is s^{-1} for r -th mode,

ω_r : damped natural angular frequency is s^{-1} for r -th mode.

The shape of the mode is an envelope of the inherent deformation of the structure expressed in a linear scale, related to a particular modal frequency. In our case (as a stimulating spectrum is flat) it is a sort of distribution of the velocity across the front panel of the system. The basic structure and a character of each of presented modes is roughly constant. However, the shape of the plotted surfaces may somewhat vary as phase changes from 0 to 360 degrees.

The frequency range of the measurements was from 0 to 4 kHz and the frequency resolution was equal to 10 Hz (400 frequency values equally spaced

in the frequency range). Spatial resolution was roughly equal to 1.5 cm as all 493 measuring points were uniformly distributed in Cartesian coordinate system. Therefore, positions of characteristic nodal circles and lines on the loudspeaker membranes are difficult to identify. However, from the sequence of mode shapes shown in the Table 1 it can be seen that at frequencies f_0 to f_2 only the woofer is working and its vibrations relative to the vibrations of the rest of the cabinet are significant. At the frequency $f_0 = 90$ Hz the basic rigid-body resonance of the woofer is visible; all woofer points move in phase and this mode of the woofer is described as (0,1). The first mode of woofer's vibration is observed at the frequency $f_1 = 580$ Hz, (0,2). This mode has one nodal circle situated near the outer suspension. This is a typical axisymmetric mode. The motion of membrane's points inside the circle is characterised by a lower amplitude than the motion of the points outside of the nodal circle. The phase of the movement inside the nodal circle is opposite to the phase of the outer part of the membrane. At the frequency $f_2 = 650$ Hz a high-amplitude motion of woofer's outer suspension can be noticed. All remaining modes from the Table 1 can be analysed and described in a similar way. However, for higher modal frequencies identification of modes is more difficult and is based of some speculations. For example, the frequency $f_3 = 1.16$ kHz corresponds probably to the first mode of the mid-tweeter (1,1); the frequency $f_5 = 1.80$ kHz probably fits in the second mode of the woofer (2,2) and one of the first resonances of the middle range tweeter and so on. Modes of single loudspeakers only were found in separate experiments.¹⁶⁾ In this case cylindrical coordinates were used because they reflect much better axisymmetric nature of the loudspeakers. However, in the case of the loudspeaker system we used Cartesian coordinates, because before measurements we had not known that the cabinet was calm.^{13,16)} The higher frequency the more complicated mode shape and it is rather difficult to distinguish nodal lines and circles on the vibrating loudspeaker and to describe its vibrational patterns. The mode shapes of the complete loudspeaker system show that "something is going on" but give no details. Therefore, to obtain clear mode shapes, it is probably necessary to investigate the single loudspeaker placed in the system.¹⁶⁾

When values of modal damping are examined it

can be stated that all values of damping are greater than these encountered for “normal” mechanical systems. Especially, damping of the basic resonance of the woofer, $\xi_0 = 39.0\%$, is great in comparison with damping of other modes, which is equal or less than 7%. This is probably caused by resonances of the inner and outer suspension of the woofer. Vibrations of the membrane are reduced by those vibrations of the suspension which are in

antiphase to vibrations of the membrane. The basic resonance of the woofer is not totally cancelled by the vibrations of the suspensions as their frequencies are not the same. Vibrational modes of woofer suspensions occur at frequencies slightly higher than the membrane basic resonance and lower than subsequent woofer modes because the suspensions are the less stiff elements of the woofer (Young modulus of elasticity of the outer suspen-

Table 1 Results of modal analysis for the complete loudspeaker system.

$f_0 = 90 \text{ Hz}$ $\xi_0 = 39.0 \%$	$f_1 = 580 \text{ Hz}$ $\xi_1 = 6.0 \%$
$f_2 = 650 \text{ Hz}$ $\xi_2 = 5.4 \%$	$f_3 = 1.16 \text{ kHz}$ $\xi_3 = 7.0 \%$
$f_4 = 1.64 \text{ kHz}$ $\xi_4 = 4.3 \%$	$f_5 = 1.80 \text{ kHz}$ $\xi_5 = 3.7 \%$
$f_6 = 2.30 \text{ kHz}$ $\xi_6 = 1.9 \%$	$f_7 = 2.55 \text{ kHz}$ $\xi_7 = 3.3 \%$
$f_8 = 2.85 \text{ kHz}$ $\xi_8 = 3.6 \%$	$f_9 = 2.97 \text{ kHz}$ $\xi_9 = 4.7 \%$
$f_{10} = 3.20 \text{ kHz}$ $\xi_{10} = 5.0 \%$	$f_{11} = 3.92 \text{ kHz}$ $\xi_{11} = 3.8 \%$

sion is $E = 1.4 \times 10^7 \text{ N/m}^2$, for the inner suspension $E = 9.3 \times 10^7 \text{ N/m}^2$ and for the woofer's membrane and the dust cup $E = 3.35 \times 10^9 \text{ N/m}^2$.¹⁷⁾ Generally, mode shapes are complex in their character. It means that phases of motion of adjacent areas are not equal to 0 or 180 degrees, but they can have any value from the range of 0 to 360 degrees. Moreover the modes are not so "clear" as they are in mechanics. This is the result of the point excitation assumption, which in reality is rather multi-point excitation. However, our STAR version enables to perform only a single excitation version of modal analysis.

The shapes of the modes presented in the Table 1 express the motion of all points in the parallel direction to the axes of the loudspeaker system. Generally, the highest motion is connected with the points situated on the woofer membrane, which is rather expected. However, non zero velocities can be also observed at areas corresponding to the tweeters, particularly for higher frequencies. At some frequencies motion of the middle range tweeter was in phase while in others it was in antiphase to the motion of the woofer. The amplitude of the tweeter's movement was only slightly bigger than the motion of the cabinet.

3.2 Results of Spectrum Analysis

All active points situated on the front panel of the system correspond to the vibration of the speakers. The points beyond the membranes (excluding some points in the adjacent area to the woofer) may be characterised by a near zero velocity. It means that cabinet must be relatively rigid. It may transmit some vibrations but it behaves quite calm. Mode shapes presented in the Table 1 are expressed in a linear scale. So it is difficult to assess the dynamic range differences between vibrations of the cabinet relative to the vibrations of the membranes. To assess this differences some additional measurements were carried out. These measurement were concerned with determination of the power spectrum of the background noise and the power spectrum of the signal in all measuring points. First, the power spectrum of the velocity was determined while exciting pseudo-random noise was delivered to the loudspeaker system. Then, the signal was turned off and the power spectrum of the velocity was determined again. The velocity spectrum measured in the later case corresponded to the background noise

vibrations. Thus, the power spectrum of the background noise was determined.

Examples of the power spectra are presented in Figs. 4a-g (the data presented in these figures were collected in points which are shown in Fig. 3 by means of black dots). Both curves are plotted in a logarithmic scale ($20 \log(v/v_0)$, $v_0 = 1 \text{ m/s}$; in dB) and are scaled on 20 dB units. Frequency is given in a linear scale on abscissa. The upper curve in each panel of the figure shows the magnitude of response spectrum measured for the pseudo-random noise excitation. Each bottom curve shows the power spectrum of background noise.

The level of the vibration connected with the stimulating signal is generally higher than the level corresponding to the background noise particularly for the points placed on the membranes and for all points situated no further than 3 cm from a vibrating membrane (see Fig. 4c and 4d). The largest difference, equal approximately 60–75 dB in whole frequency range, was observed in point no. 4 (the membrane of the woofer) and in point no. 3 (the rim of the woofer). Next, the measuring point (no. 2) placed on the plastic ring of the woofer showed also significant vibrations, about 30 dB above the background noise (Fig. 4b). Significant vibrations were also observed in point no. 6 situated on the mid-tweeter membrane (Fig. 4f).

The data presented in Fig. 4e suggest that point no. 5 situated between woofer and middle range tweeter also vibrated significantly. These vibrations were about 20 dB above the level of the background noise in the frequency range up to 3 kHz. Notice that in this case, and for frequencies below 3 kHz, at least two speakers, *i.e.* the woofer and the middle range tweeter, were electrically stimulated, since crossover frequency between woofer and middle range tweeter is equal 3.5 kHz and the crossover network slope is 6 dB/octave. Thus, the vibration observed in the point no. 5 must have come from vibrations of the woofer and the mid-tweeter.

It is worthwhile to say that this type of results were also gathered for all points placed close (no more than 2–3 cm) from the woofer. So the data presented in Fig. 4e are representative for whole area around the woofer. They suggest that in the real conditions this area vibrates somewhat. However, the observed vibrations were approximately 40–50 dB below the vibration of the membrane of the woofer.

Fig. 4 Response signal : upper curve ; noise signal : bottom curve for the loudspeaker system in the different points placed on front panel of the system (see also Fig. 3).

Area above the tweeter (point no.7) is excited in the very limited range, because crossover frequency between the mid-tweeter and the tweeter is 10.5 kHz and the crossover network slope is 6 dB/octave. Thus, the tweeter might be excited electrically although the excitation is very weak. What is more, the mass of the woofer, the main vibrating speaker, is 0.95 kg while the mass of the less electrically stimulated speaker, the tweeter, is 0.09 kg.

Therefore, inertia of the woofer is much greater than inertia of the tweeter and the woofer excites some area around it while the tweeter—does not.

Almost no vibrations were observed in points no. 1 and 7 (see Figs. 4a and 4g) and in all other points situated further than 3 cm from the woofer. There is small difference between power spectra of the signal and background noise in the low frequency region determined in these points. There was no

measuring point that this difference was higher than 5 dB in the whole frequency range, *i.e.* in 0–4 kHz range. On the other hand, it is necessary to say that there is a huge difference between vibrations of the membrane and the vibration of these points. This difference is at least 60 dB in whole frequency range. Thus, assuming that the front panel of the loudspeaker system is a kind of vibrating plate, its vibrations are 1,000 times smaller than the vibrations of the membrane of the woofer. It means that the front panel is almost calm, its vibrations are very small and their contribution to the total vibration pattern in the majority of cases can be neglected.

4. CONCLUSIONS

(1) Vibration modes of single loudspeakers were found in the set of the system vibrational patterns. However, since the spatial resolution of measurements performed on the front panel of the loudspeaker system was rather rough, it was only possible to depict areas of the greatest vibration. Description of modes of separate loudspeakers was not our task; we wanted to investigate vibrations of the loudspeaker system as the whole and to find the most vibrating areas.

(2) The strongest vibrations measured on the front panel of the loudspeaker system are related mainly to the vibrations of the membranes of the speakers. Vibrations of the membranes of the speakers are at least 60 dB higher than vibrations observed in the most points of the front panel of the cabinet.

(3) Vibrations of the front panel of the loudspeaker (excluding the areas of the membranes) are very weak related to the vibrations of the membranes. They are no higher than 5 dB above the background noise. The only exception is the area close to the woofer where the vibrations were about 20 dB bigger than the level of the background noise.

(4) The contribution of the vibrations of the front panel of the loudspeaker system (excluding the membranes) in the sound field radiated by the loudspeaker system is rather small and may be neglected.

ACKNOWLEDGEMENTS

We thank an anonymous reviewer for helpful comments on earlier versions of this paper.

REFERENCES

- 1) D. J. Ewins, "On predicting point mobility plots from measurements of other mobility parameters," *J. Sound Vib.* **70**, 69–75 (1980).
- 2) K. Zaveri, *Modal Analysis of Large Structures — Multiple Exciter Systems* (Brüel & Kjaer, Naerum, 1985).
- 3) H. G. Natke, *Einfuehrung in Theorie und Praxis der Zeitreihen- und Modalanalyse* (Vieweg, Berlin, 1992).
- 4) K. D. Marshall, "Modal analysis of a violin," *J. Acoust. Soc. Am.* **77**, 695–709 (1985).
- 5) STAR, "The STAR system version 3.02D. Theory and application," in *User's Manual*, SMS, Ed. (1990).
- 6) T. Uhl, W. Lisowski, and L. Miękinia, "Wprowadzenie do analizy modalnej. Część I: Podstawy teoretyczne," *Energetistic Methods in Vibroacoustics, II School* (Krynica, 1993), pp. 95–114.
- 7) N. H. Fletcher and T. D. Rossing, *The Physics of Musical Instruments* (Springer-Verlag, New York, 1991).
- 8) E. B. Skrodzka and E. Hojan, "Initial modal testing of a cantilever beam," *Arch. Acoust.* **19**, 355–365 (1993).
- 9) STAR, "The STAR system. Advanced curve fitting," *Reference Manual*, SMS, Ed. (1992).
- 10) F. Deblauwe, D. I. Brown, and R. J. Allemang, "The polyreference time domain technique," 5th Int. Modal Analysis Conf., London, England, 832–845 (1987).
- 11) M. H. Richardson and D. L. Formenti, "Global curve fitting of frequency response measurements using the rational fraction polynomials method," 3rd Int. Modal Analysis Conf., Orlando, FL, 390–397 (1985).
- 12) H. Van der Auweraer and J. Lueridan, "Multiple input orthogonal polynomial parameter estimation," 5th Int. Modal Analysis Conf., London, England, 986–995 (1987).
- 13) E. B. Skrodzka and H. Fleischer, "Drgania strukturalne głośników," XLI Open Seminar on Acoustics, Wrocław-Szklarska Poręba, 415–418 (1994).
- 14) E. B. Skrodzka, H. Fleischer, T. Dillmann, and H. Winter, "Vibroacoustic investigation of a loudspeaker," DAGA, Dresden, 553–556 (1994).
- 15) M. R. Gander, "Dynamic linearity and power compression in moving-coil loudspeakers," *J. Audio Eng. Soc.*, **34**, 627–646 (1986).
- 16) E. B. Skrodzka, "Modelling of the sound field of loudspeakers based on their dynamic behaviour," Dissertation, A. Mickiewicz Univ., 1–124 (1995).
- 17) A. Dobrucki, "Numerical modelling of the moving system of electodynamic loudspeaker," *Wyd. Politechniki Wrocławskiej*, 1–170 (1992).