Applied Acoustics 93 (2015) 9–14

Contents lists available at ScienceDirect

Applied Acoustics

journal homepage: www.elsevier.com/locate/apacoust

Technical Note

Influence of enclosure wall vibration on the frequency response of miniature loudspeakers

Xuan Zhong *, Qunli Wu, Xiaodong Li

Institute of Acoustics, Chinese Academy of Sciences, 21 Bei-Si-huang-Xi Road, Beijing 100190, China

ARTICLE INFO

Article history:
Received 18 April 2014
Received in revised form 6 January 2015
Accepted 14 January 2015
Available online 4 February 2015

Keywords:
Miniature loudspeaker
Loudspeaker back cavity
Frequency response
Loudspeaker directivity
Structural vibration

ABSTRACT

The miniature loudspeaker is widely used in consumer electronic products. The unwanted vibration of the enclosure wall of the loudspeaker could add to the overall acoustic output and cause distortion of the frequency response. An experimental miniature loudspeaker model with a low-damping enclosure wall was constructed. The vibration of the enclosure wall plate was simulated with an acoustical analogous circuit, in which the wall plate was modeled as a separate branch in parallel with the back cavity air volume. The acoustic frequency response of the enclosure wall was simulated with combined finite element method and boundary element method (FEM–BEM). The vibration and acoustic measurements validated the effectiveness of the simulation methods. Finally, the frequency response of a production type miniature loudspeaker was measured before and after modification. Distortion up to ±15 dB on the frequency response curve was observed around 7.8 kHz. With damping material applied to the enclosure wall, the distortion was largely suppressed.

1. Introduction

The miniature loudspeaker is widely used in consumer electronic products, often as a sub-system in applications such as hands-off telephone calls and music playing [1]. A smooth and flat frequency response is commonly required by telecommunication standards and is favorable for perceived sound quality [2]. The acoustic part of a loudspeaker system consists of the loudspeaker unit and the back cavity. When the loudspeaker unit plays sounds, the enclosure walls also vibrate and may distort the frequency response [3]. However, it is common engineering practice to simply generalize the whole back cavity as a volume of air [4], and the enclosure wall vibration problem was seldom documented in the context of miniature loudspeaker designs.

The thickness of a miniature loudspeaker unit is typically 2–3 mm, and almost always under 10 mm. Its diameter is typically 10 mm, and almost always under 50 mm. To save space and lower manufacturing complexity, the two suspension parts on a normal-sized loudspeaker unit (the roll surround and the spider) are combined into a single one, i.e. the outer part of the membrane. The structure of miniature loudspeaker units and systems has been discussed in greater detail in [4]. The miniature loudspeaker enclosures are often made of low-damping materials such as plastic or metal, which cannot easily suppress the resonance of plates compared to the commonly used wood materials for the normal-sized loudspeaker [3]. Thus, some higher modes of enclosure wall resonance may also be excited to such an extent to affect the loudspeaker's frequency response. In another aspect, the miniature loudspeaker systems in handheld/desktop/car audio applications do not always point to the user's listening position with their main axis [5].

If the problems of enclosure wall vibration could be simulated in an early phase of the acoustic design process, they could be fixed at a lower cost. A straightforward way of modeling is the analogous circuit method. Tappan [3] modeled the enclosure wall with an analogous circuit and observed distortion on the frequency response curve of the wooden-box loudspeaker systems. The distortion had a shape in the form of a combination of a peak and a valley on the frequency response curve. It was concluded that only the first structural resonant mode of the loudspeaker wall with the largest dimension should be controlled. Iverson [6] discussed the resonance of loudspeaker cabinet boards in general. Such lumped parameter methods could predict the wall vibration up to the first resonance mode.

Another simulation option is to use computational models. The task of simulating the frequency response from loudspeaker enclosure walls consists of two parts, a structural part and an acoustic part. The structural part is to accurately calculate the enclosure

* Corresponding author at: 975 S Myrtle Ave., Lattie F Coor Hall 2211, Tempe, AZ 85287, United States.

E-mail address: xuan.zhong06@gmail.com (X. Zhong).

http://dx.doi.org/10.1016/j.apacoust.2015.01.012
0003-682X/

wall vibration based on the parameters of the loudspeaker unit, the back cavity and the wall structures. Historically, with the finite element method (FEM), Karjalainen et al. [7] measured and simulated, the vibration of the loudspeaker enclosure walls, but did not calculate the frequency response. So the problem of the second acoustic part is, given the vibration pattern, how to calculate the acoustic response at a certain point in the sound field. Bastyr and Capone [8] measured the enclosure wall vibration with a laser vibrometer. Based on the measurement, they predicted the acoustic radiation from loudspeaker enclosure walls with the boundary element method (BEM) with success. But they did not attempted modeling the back cavity and the enclosure wall itself. The structural and acoustic simulations were seldom discussed together in the simulation of enclosure walls, so a more complete study based on an integrated model is needed.

The current study investigated the influence of loudspeaker wall vibration on the frequency response of the miniature loudspeaker systems. Both an analogous circuit model and a FEM–BEM model were used to calculate the enclosure wall vibration and the acoustic frequency response. Simulations were validated with the vibration measurement and the acoustic frequency response measurement. Finally, a real design model of a production type loudspeaker was measured before and after modifications of the enclosure wall to show how the distortion could be reduced.

2. Theory and calculation

An experimental loudspeaker model was constructed as shown in Fig. 1. A miniature loudspeaker unit was mounted at the front of the model. The front and side enclosure walls were made of 5 mm thick steel which can be considered as a rigid boundary in the model. A 5 mm deep back cavity was left open at the back side with clamps on all the edges. The back enclosure wall was a 31 mm $*$ 31 mm $*$ 0.34 mm aluminum plate mounted with the clamps. The boundary condition could be regarded as clamped on all the four sides of the plate. The acoustic signal measured at a certain point in the space would be the addition of the sound from the loudspeaker unit and that from the back enclosure wall.

The main design consideration of the experimental model was to demonstrate the worst case of enclosure wall vibration. In real acoustic engineering cases, a square shaped back cavity is often avoided to reduce the combined modes in the back cavity. However, the outcomes of the research on the simplified model could still be suggestive of the considerations in actual designs.

2.1. The analogous circuit modeling

The acoustical analogous circuit method combines the electrical, the mechanical and the acoustical parameters of the loudspeaker system in a unified model [9]. The first resonance of the enclosure wall plate and vibration velocity frequency response of the plate were simulated. The effect of the back plate vibration on the vibration of the loudspeaker membrane could also be derived from the model, as suggested by Tappan [3].

In the model, the electrical and mechanical domains were reflected to the acoustic domain. As shown in Fig. 2, several groups of the components were used to represent the three parts of the experimental model, (a) the loudspeaker unit, (b) the back cavity, and (c) the enclosure back wall plate.

For the loudspeaker unit branch, the total acoustical impedance was given by:

$$Z_{AS} = R_{AT} + j\omega M_{AS} + \frac{1}{j\omega C_{AS}} \qquad (1)$$

where

Z_{AS} – total acoustical impedance of the loudspeaker unit.
R_{AT} – total acoustical resistance of the loudspeaker, including voice coil resistance and damping of the membrane suspension.
M_{AS} – acoustical mass of the loudspeaker unit.
C_{AS} – acoustical compliance of the loudspeaker unit.

The loudspeaker unit parameters were derived from T–S parameter measurement [10]. The loudspeaker unit was type Ra miniature loudspeaker from the former Philips Sound Solutions (presently a part of Knowles) and its size was 15 mm $*$ 11 mm $*$ 3 mm. The parameters of the unit were: the effective radiation area $S_D = 2\ \mathrm{cm}^2$, force factor $Bl = 0.7$ T m, $C_{AS} = 1.025 * 10^{-10}\ \mathrm{m^5/N}$, $M_{AS} = 247.4\ \mathrm{g/m^4}$, and $R_{AT} = 47{,}430\ \mathrm{kg\ s^3/m^4}$.

For the back cavity branch:

$$Z_{AB} = R_{AB} + \frac{1}{j\omega C_{AB}} \qquad (2)$$

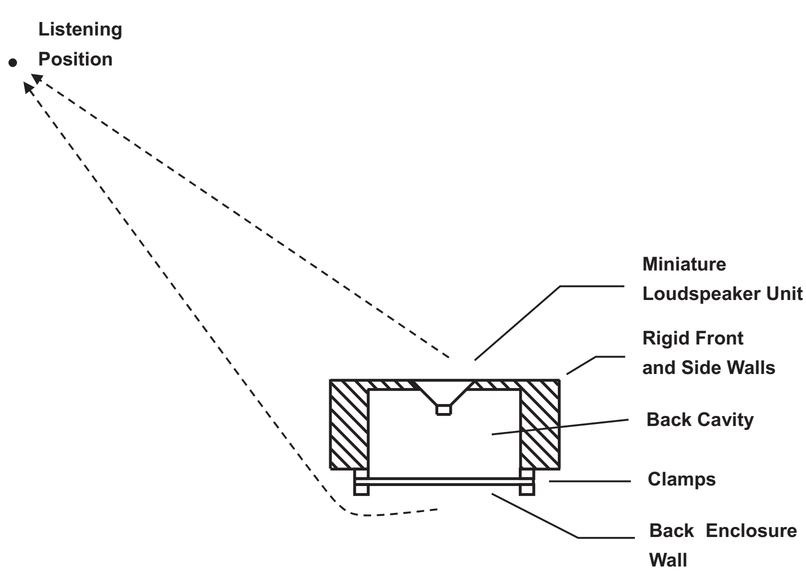

Fig. 1. The miniature loudspeaker experimental model under test. The size of the cavity was 31 mm $*$ 31 mm $*$ 5 mm.

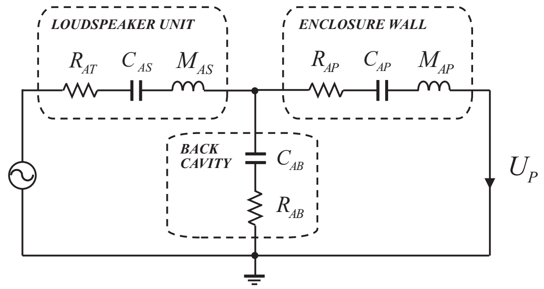

Fig. 2. The analogous circuit of a loudspeaker with vibrating enclosure wall plate (impedance analogy in the acoustic domain). Please refer to Section 2.1 for definitions of all components.

where

Z_{AB} – total acoustical impedance of the loudspeaker back cavity.
R_{AB} – acoustical resistance in the cabinet.
C_{AB} – acoustical compliance of the air in back cavity.

Here C_{AB} was determined by the back cavity volume, for the current model $C_{AB} = 2.682 * 10^{-11}$ m^5/N. And R_{AB} was determined by the air and the sound absorptive materials in the cavity [9], in the present simulation its effect was omitted.

The non-rigid back cavity wall was a plate clamped at all sides. Past investigations described this branch as follows [11]:

$$Z_{AP} = R_{AP} + j\omega M_{AP} + \frac{1}{j\omega C_{AP}} \tag{3}$$

where

Z_{AP} – total acoustical impedance of the enclosure wall plate.
R_{AP} – equivalent acoustical resistance of the enclosure wall plate.
M_{AP} – equivalent acoustical mass of the enclosure wall plate.
C_{AP} – equivalent acoustical compliance of the enclosure wall plate.

For simple shapes, such as square, rectangular and circular plates, M_{AP} and C_{AP} could be obtained by analytical methods. According to the calculation methods described in [12], $M_{AP} = 5053$ g/m^4 and $C_{AP} = 8054 * 10^{-13}$ m^5/N. In a pilot study R_{AP} was found to be very small compared to R_{AT}, so in the current study it was omitted.

Based on the model, the velocity of the enclosure wall plate was given by:

$$U_P = \frac{1}{Z_{AP} + Z_{AS}\left(1 + \frac{Z_{AP}}{Z_{AB}}\right)} \cdot P_0 \tag{4}$$

The resonance of the plate happened when Z_{AP} is at its minimum, which was decided mostly by M_{AP} and C_{AP}. Once the parameters were known, the peak response frequency could be determined. The normalized transfer function, in its full form, could be represented as:

$$G(j\omega) = \frac{\dfrac{j\omega M_{AP} + \frac{1}{j\omega C_{AP}}}{1 + j\omega M_{AP}\left(j\omega M_{AP} + \frac{1}{j\omega C_{AP}}\right)}}{R_{AT} + j\omega M_{AS} + \frac{1}{j\omega C_{AS}} + \dfrac{j\omega M_{AP} + \frac{1}{j\omega C_{AP}}}{1 + j\omega M_{AP}\left(j\omega M_{AP} + \frac{1}{j\omega C_{AP}}\right)}} \cdot \frac{1}{j\omega M_{AP} + \frac{1}{j\omega C_{AP}}} \tag{5}$$

The analogous circuit model was effective up to the frequency at which any component in the circuit started to show break-up modes. In this study the frequency range was under the first resonant mode of the enclosure wall. A shortcoming of this method was that lumped parameters of the enclosure walls were not available for irregular shapes.

2.2. The BEM–FEM modeling

FEM is commonly used in the prediction of steady-state pressure filed in a confined space volume, whereas BEM is used to calculate sound fields in unbounded spaces based on known vibration velocities at the boundaries. In contrary to prior work which only addressed a part of the whole question [7,8], the current study used a combined FEM–BEM method to simulate the enclosure wall vibration as well as the acoustic frequency response. The efforts consisted of two sections:

(1) Structural simulation (FEM): loudspeaker parameters ⇒ enclosure wall vibration.
(2) Acoustic simulation (BEM): enclosure wall vibration ⇒ acoustic response.

First, the steady state dynamic response of the coupled enclosure wall and back cavity was calculated without considering the external sound field. The coupling between the air in the back cavity and the enclosure wall was considered. For this purpose, a coupling matrix ensured that the normal fluid displacements in the back cavity equaled that of the plate of enclosure wall. The vibration pattern of the enclosure wall was calculated. The result was saved in the form of normal velocity vectors distributed along the surface of the structural model.

Secondly, the structural model was exported to the BEM simulation environment. The normal velocity distribution on the boundaries was applied on the BEM model and Green's functions were calculated. Finally, the frequency response at any given spatial point of interest could be calculated.

The three dimensional model of the loudspeaker system was drawn with ANSYS software (version 11.0) and so was the meshing. Both the FEM and BEM simulations used LMS Sysnoise (version 5.6). All the modeling and simulation were carried out on a workstation located in Institute of Acoustics, Chinese Academy of Sciences.

3. Experiments

Acoustical and mechanical measurements demonstrated the extent of the influence that the enclosure wall vibration has on the frequency response of miniature loudspeakers [13]. The effectiveness of analogous circuit method and FEM–BEM modeling was validated by comparing measured data with simulation results.

3.1. Vibration measurement

The velocity of enclosure wall vibration was measured with a Metrolaser ViroMet 500V laser vibrometer, which emitted a laser beam targeted at the measurement point on the enclosure wall, and computed the instantaneous velocity based on the Doppler Effect. A dot was drawn on the back enclosure wall to facilitate focusing of the laser beam. The laser vibrometer and the loudspeaker system under test were 30 cm apart. During a test, a frequency sweep signal was generated and played from a B&K PULSE system. The measured signal from the laser vibrometer were fed back into the PULSE system and analyzed. A more detailed explanation of the setup and measurement of the vibrating plates can be found in [13].

3.2. Acoustic frequency response measurement

The half-space frequency response of a miniature loudspeaker was measured to study how much sound radiated from the back enclosure wall alone. The test room was a half-space anechoic chamber located in Institute of Acoustics, Chinese Academy of

Sciences (Beijing). The loudspeaker system was mounted on a 100 cm ∗ 100 cm ∗ 5 cm wooden board on the ground of the anechoic chamber as shown in Fig. 3. The front side of loudspeaker system faced down toward a pit (70 cm ∗ 70 cm ∗ 80 cm) under the ground of the anechoic chamber. The pit was filled with sound absorptive materials. The back enclosure wall faced toward the measurement microphone positioned at 30 cm away. In this way, the acoustic radiation from the loudspeaker unit and that from the back enclosure wall were separated.

The measurement transducer was the B&K 4189 pre-polarized sound field microphone. A B&K PULSE analyzer platform generated a frequency-sweep signal over the audio frequency range. The electric signal was amplified with type B&K 2716 power amplifier to drive the loudspeaker system under test with a constant 1 V voltage. The acoustic signals received by the microphone were put back to the PULSE analyzer platform.

4. Results

4.1. Enclosure wall vibration velocity

The vibration velocity of the enclosure wall at low frequencies was predicted with the analog circuit method as shown in the dotted line in Fig. 4. At around 3 kHz a peak due to enclosure wall vibration was expected. The simulation generally agrees with the measurement results of laser vibrometer.

4.2. Frequency response

The half-space acoustic frequency response measurement showed two peaks at 3 kHz and 10.5 kHz. The simulation of the loudspeaker enclosure wall were shown together with measurements in Fig. 5. The FEM–BEM model predicted that the (1, 1) mode was at 3039 Hz (1.3% error) and the combined (1, 3) (3, 1) modes at 11344 Hz (8.0% error). The trend of the frequency response measurement generally agreed with the FEM–BEM simulation.

5. A case study

Although significant distortion was observed in the aforementioned experimental model due to enclosure wall vibrations, a natural question that arises was whether the same effect was also affecting loudspeaker enclosures of irregular shapes, especially those in production. To answer this question, an off-the-shelf production type loudspeaker module was measured before and after modification to the enclosure wall. Loudspeaker modules are combinations of loudspeaker units and enclosures, and are increasingly widely used in consumer electronics. In acoustic modeling they can be treated as complete loudspeaker systems.

The dimensions of the loudspeaker module under test was 47 mm ∗ 12 mm ∗ 4 mm (Fig. 6). The loudspeaker unit was sealed in one end of the bar-shaped enclosure. The only sound port was on the side of the enclosure. The most common listening position, however, was at a direction directly facing the biggest enclosure wall instead of the sound port.

The shape of the enclosure was close to a rectangular block. The analogous circuit method was used for simplicity. It is highly likely that the effects of the smaller walls, if there were any, was negligible. So only the enclosure wall of the largest dimension (about 47 mm ∗ 10 mm ∗ 0.5 mm) was considered. For a rectangular plate like this, lumped mass and compliance parameters can be calculated easily [12]. Based on the material properties and dimensions, the first resonant frequency of the biggest enclosure wall was estimated to be 7500 Hz. A thin layer of high-damping material was added to the enclosure wall to suppress the vibrations. The frequency responses were measured before and after this modification.

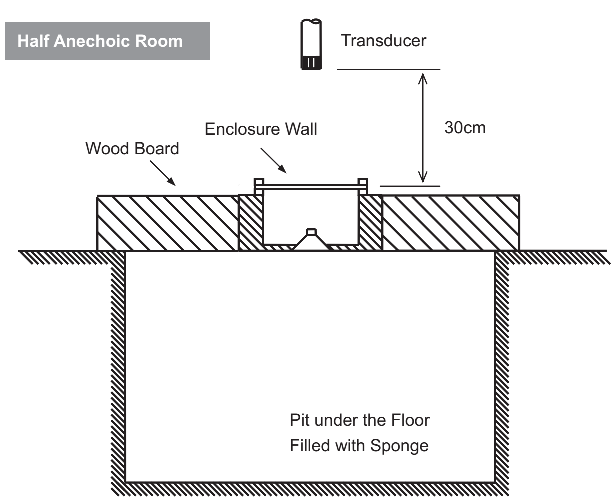

Fig. 3. Measurement of the acoustic radiation from the back enclosure wall. The test room was a half-space anechoic chamber. Only the acoustic radiation from the back side of the loudspeaker system was measured by the microphone. The wood board was 100 cm ∗ 100 cm ∗ 5 cm. The pit was 70 cm ∗ 70 cm ∗ 80 cm. The figure is disproportionate and only to demonstrate the test setup.

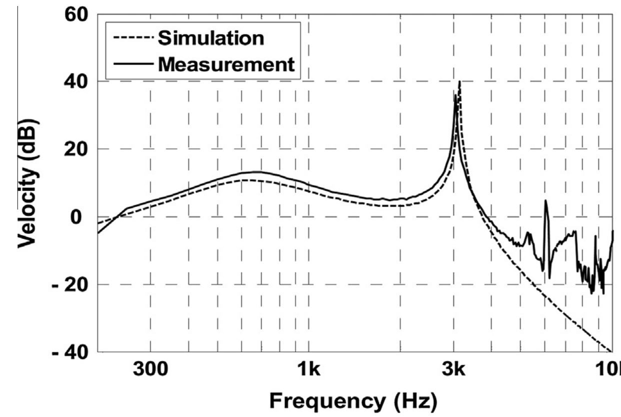

Fig. 4. Vibration velocity amplitude (in dB relative to 5×10^{-8} m/s) of enclosure wall, simulated with the analogous circuit method (dotted) and measured with laser vibrometer.

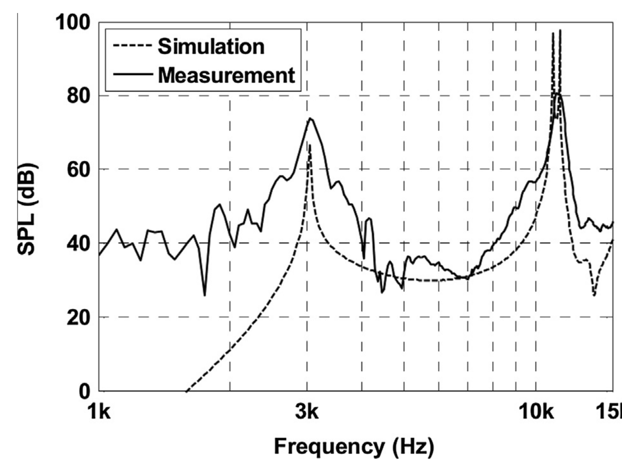

Fig. 5. Acoustic radiation into the half-space at the backside of the loudspeaker, as simulated with FEM–BEM model and measured in half-anechoic room.

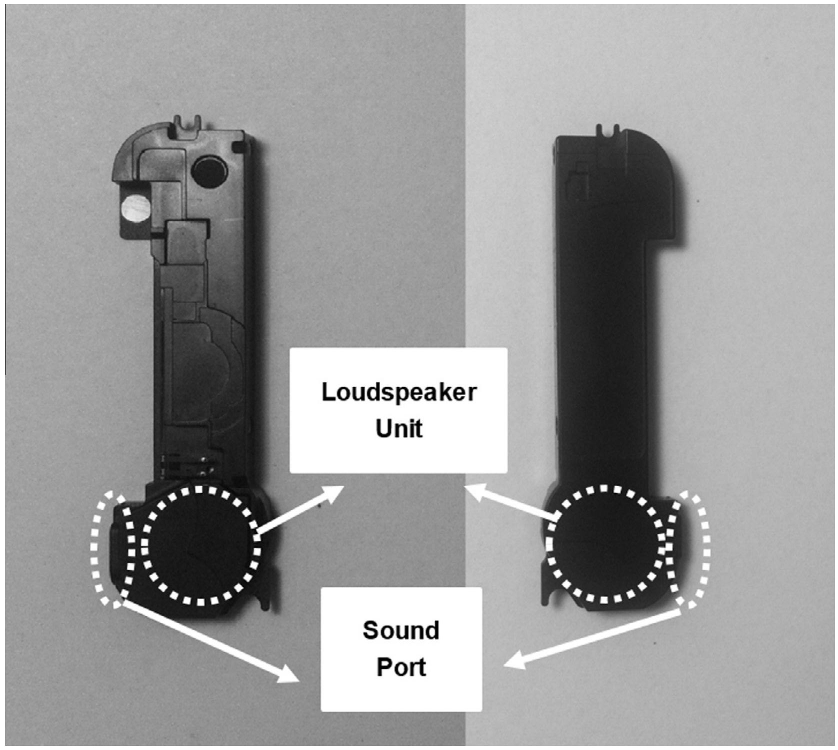

Fig. 6. The front and back views of the loudspeaker module under test. The dimension of the loudspeaker module was 47 mm $*$ 12 mm $*$ 4 mm. The miniature loudspeaker unit was at one end of the enclosure case. The only sound port was on the side of the module.

The loudspeaker module was tested following procedures that were close to prior experiments. Without the damping material, i.e. in the original condition of the loudspeaker module, a typical distortion of a peak followed by a dip was observed around 7800 Hz (Fig. 7), which largely agrees with the simulation (3.8% error). The amplitude of the frequency response distortion was ±15 dB. With damping material applied, the distortion was largely gone. In casual listening tests, the difference between the original and modified module was clearly audible.

In engineering practice, such a module would eventually be put into a product, such as a mobile phone case. Some reduction of the distortion can be expected, but there is no guarantee that the distortion could be prevented altogether.

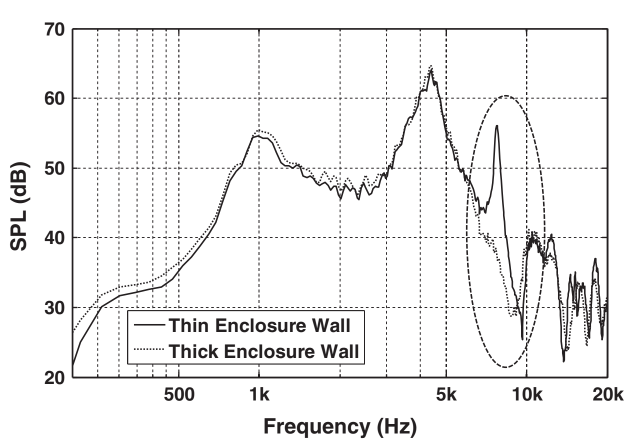

Fig. 7. The frequency response of the miniature loudspeaker module with original enclosure wall (real line) and damped enclosure wall (dotted line). A significant difference was observed around 7800 Hz and was audible.

6. Discussion

In this study, the simulation methods were found to be effective in the prediction of vibration and acoustic radiation from the enclosure walls. The analogous circuit method was straightforward to implement and did not require special software, but could only predict the vibration velocity of the enclosure wall up to its first resonant frequency, and was subject to availability of lumped parameters of the enclosure walls. The combined FEM–BEM modeling, on the other hand, required special software and was more time-consuming, but it could predict the frequency response in a larger frequency range, and could easily compute complex shapes and boundary conditions.

It has been suggested that the higher odd-odd numbered modes above the first resonance (1,1) could not alter the frequency response in a noticeable manner [3]. However, the current study showed that, in a miniature loudspeaker design, the next highest odd-odd numbered mode (1,3) and (3,1) may also distort the frequency response.

To reduce the influence from the enclosure wall vibration, adding to the thickness and damping of the material was straightforward and effective [3,13]. Changing the materials to laminated plates would also increase damping and thus reduce the peak vibration [14]. Adding irregular ribs would increase the resonant frequency to a region where the acoustic radiation is less effective, so it may also be a feasible solution [15]. In the case study in Section 5, a production type loudspeaker module was modified with a thin layer of damping material, and the resonant mode at 7.8 kHz was effectively suppressed.

The real design cases of miniature loudspeakers differ greatly in material and dimensions, so the range of the resonance peaks was hard to define. However, in both the experimental and the production models, enclosure wall vibrations caused peaks in the frequency response curve within the audible range. So when reducing the size or the enclosure wall thickness of miniature loudspeakers, the influence of enclosure wall vibration is an engineering problem that should be addressed carefully.

7. Conclusion

This technical note showed that the frequency response of miniature loudspeaker system could be distorted by the vibration of enclosure walls. The analogous circuit simulation could predict the first resonant peak of the enclosure wall for simple shapes and boundary conditions. The FEM–BEM model, on the other hand, provided an estimation of all the structural modes that might affect the frequency response and the extent of the influence.

Acknowledgment

The authors are indebted to Dr. William Yost for proofreading the manuscript.

References

[1] Huang JH, Her HC, Shiah YC, Shin SJ. Electroacoustic simulation and experiment on a miniature loudspeaker for cellular phones. J Appl Phys 2008;103(3):033502.
[2] Bahne A. Perceived sound quality of small original and optimized loudspeaker systems. J Audio Eng Soc 2012;60(1/2):29–37.
[3] Tappan PW. Loudspeaker enclosure walls. J Audio Eng Soc 1962;10(3):224–31.
[4] Bai MR, Liao J. Acoustic analysis and design of miniature loudspeakers for mobile phones. J Audio Eng Soc 2005;53(11):1061–76.
[5] Schwarz BA, Brinkerhoff DE. Some observations on reproduced sound in an automobile. J Audio Eng Soc 1958;6(1):58–63.
[6] Iverson JK. The theory of loudspeaker cabinet resonances. J Audio Eng Soc 1973;21(3):177–80.
[7] Karjalainen M, Ikonen V, Antsalo P, Maijala P, Savioja L, Suutala A, et al. Comparison of numerical simulation models and measured low-frequency behavior of loudspeaker enclosures. J Audio Eng Soc 2001;49(12):1148–66.
[8] Bastyr KJ, Capone DE. On the acoustic radiation from a loudspeaker's cabinet. J Audio Eng Soc 2003;51(4):234–43.
[9] Beranek LL, Mellow T. Acoustics: sound fields and transducers. Academic Press; 2012.
[10] Small RH. Direct radiator loudspeaker system analysis. J Audio Eng Soc 1972;20(5):383–95.
[11] Frost AD. Analog circuit representation for wall panels. J Acoust Soc Am 2005;28(6):1285–91.
[12] Leissa AW. The free vibration of rectangular plates. J Sound Vib 1973;31(3):257–93.
[13] Zhong X, Wu Q, Li X. Improving acoustical performances of miniature loudspeaker systems with a laser vibrometer. Audio Eng 2008;32(8):22–5 (in Chinese).
[14] Sun CT, Sankar BV, Rao VS. Damping and vibration control of unidirectional composite laminates using add-on viscoelastic materials. J Sound Vib 1990;139(2):277–87.
[15] Photiadis DM. The effect of irregularity on the scattering of acoustic waves from a ribbed plate. J Acoust Soc Am 1992;91(4):1897–903.