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TWO-DIMENSIONAL MODEL OF THE VIBRO-ACOUSTIC FEEDBACK IN A HEARING AID

Lars Friis*[1], Mogens Ohlrich[2]

[1]Widex A/S, Ny Vestergårdsvej 25, DK-3500 Værløse, Denmark, and
Acoustic Technology, Ørsted•DTU, Technical University of Denmark
Building 352, DK-2800 Kgs. Lyngby, Denmark
[2]Acoustic Technology, Ørsted•DTU, Technical University of Denmark,
Building 352, DK-2800 Kgs. Lyngby, Denmark
l.friis@widex.com

Abstract

In this paper we investigate the vibration patterns of a “behind the ear” hearing aid model. Due to the minimal structural design of hearing aids, problems with vibro-acoustic transmission from the loudspeaker to the microphones often arise. Vibrations and sound pressure are picked up by the microphones and an unwanted electrical/vibro-acoustical loop is formed. This phenomenon is also known as feedback. The vibratory part of the vibro-acoustic transmission from the loudspeaker to the hearing aid shells is examined for a simple mathematical vibration model of the hearing aid. This simple model includes the main parts of the hearing aid such as the loudspeaker, hook, shells, battery and resilient connections. By employing mobility synthesis, these components are modelled as lumped masses, springs and beam components, which are connected to one another. Results from the vibration model reveal a complicated pattern of resonances governed by the various components and their interaction with one another. Furthermore, the vibration isolation effect of the loudspeaker suspension is investigated.

INTRODUCTION

There are many aesthetics and structural design requirements to modern hearing aids. These have resulted in designs where the loudspeaker and microphones are placed very close to one another. As a consequence of this, problems with vibro-acoustic transmission from the loudspeaker to the microphones often arise. Vibrations and sound pressure are “picked up” by the microphones and an unwanted electrical/vibro-acoustical loop is formed. The phenomenon is called feedback and occurs at certain critical gain levels in the hearing aid where it produces an uncomfortable howling sound.

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This paper investigates the vibratory part of the vibro-acoustic transmission from the loudspeaker to the shells in a so-called "behind the ear" hearing aid. This is placed behind the user's ear with the curved *hook* around the upper part of the ear. It has two microphones that monitor the sound pressure and converts this into an electrical signal, which is amplified in the hearing aid and is fed to the loudspeaker. The sound pressure from the loudspeaker is led through a canal in the hook and into a soft plastic tube, which is firmly connected to a moulded ear plug. In order to obtain good vibration isolation, the loudspeaker and the microphones are mounted resiliently in soft rubber suspensions. More general information about hearing aids is found in Reference 1.

A full three-dimensional analysis of the vibration transmission is very difficult due to the interactions between structural components of complex shapes and because of mechanical properties and connections that are often uncertain. However, a fundamental understanding of certain overall transmission phenomena may be obtained from studies of less complicated models. Herein we consider only a two-dimensional hearing aid model, consisting of masses, springs and beam components. In a first attempt wide simple beams are chosen to represent approximately modal behaviour and certain elastic properties similar to those of the complex shells. The simple mechanical model considered in this paper is based on the hearing aid shown in Figure 1.

The simple vibration model considered includes the main parts of the hearing aid such as the loudspeaker, resilient suspensions, hook, shells, and battery. By employing mobility synthesis [2], these components are modelled as lumped masses and springs, and elastic beam components, which are connected to one another. Altogether the model has a total weight of about three grams.

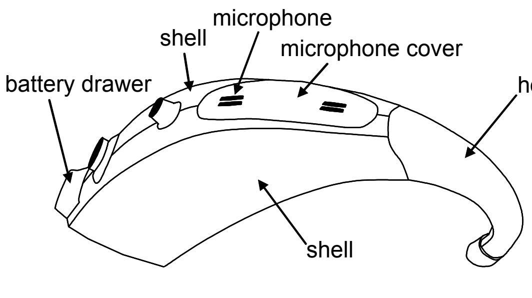

Figure 1 – "Behind the ear" hearing aid.

Examples of the use of mobility synthesis can be found in Ref. [2] where the method is described in details. Moreover, Gardonio [3] has examined a multi-degree-of-freedom system consisting of a source and receiver separated by vibration isolating mounts by using mobility synthesis.

HEARING AID MODEL AND OUTLINE OF THEORY

Figure 2 shows sketches of the "behind the ear" hearing aid and a two-dimensional model of masses, springs, and beams. The vibration response of this system is governed by three motion degrees-of-freedom, comprising velocities v_x, v_y, and υ_z in the *x*-direction, *y*-direction, and the rotational direction around *z*, respectively. Components are denoted by a capital letter, and the junctions where the components are connected are denoted with a number in a circle. Furthermore, springs are denoted by a capital S with a number as subscript and each spring represents a stiffness in all three motion coordinates.

Component A represents the loudspeaker which is the vibration source of the hearing aid. The driving mechanism and loudspeaker membrane are primarily vibrating in the *y*-direction and the loudspeaker is modelled as a mass which is driven by a harmonic force $F_{exc,y}e^{i\omega t}$ in this direction at its centre of gravity. The air pipe, B, is modelled as an equivalent Bernoulli-Euler beam, which is connected to the loudspeaker through one part of the loudspeaker suspension represented by spring S_1. Sound pressure from the loudspeaker is let through the air pipe and into the hook C, which is modelled as a beam with varying cross-section as shown in Figure 2. The hook is coupled to the air pipe and to the upper and lower shells D and E through the somewhat indefinable connection springs S_2, S_3 and S_6. These shells are represented by two simple beams with approximately the same dimensions as the actual shells. In the left-most end, the two shells are connected to one another through connection spring S_9. Vibration isolation of the loudspeaker from the shells is obtained by the loudspeaker suspension represented by springs S_4 and S_7. Finally the battery G, which is modelled as a mass, is held into place by two springs S_5 and S_8 connected to the upper and lower shell, respectively.

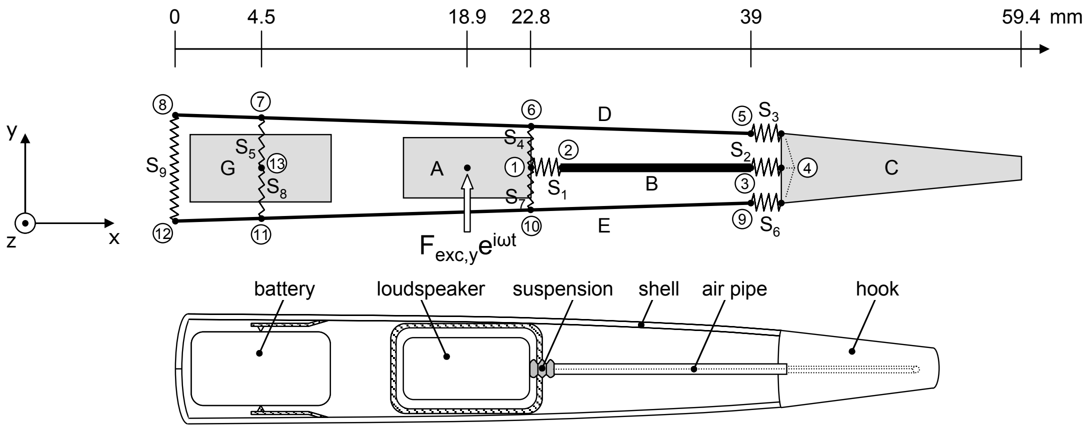

Figure 2 – Vertical section of hearing aid and model consisting of masses, springs and beam components.

Weight, dimensions and elastic properties of each component are given in Table 1-3.

Table 1 – Properties of beam components.

Symbol	Component	E-module [MPa]	Weight [g]	Loss factor	Length [mm]	Height [mm]	Width [mm]
B	Air pipe	210	0.104	0.003	14.0	1.76	0.582
C	Hook	1.9	0.262	0.061	20.4	4.9-2.3	8.1-2.3
D, E	Shells	2.3	0.575	0.05	39.0	1.3	10.8

Table 2 - Mass properties.

Symbol	Component	Weight [g]	Length x [mm]	Length y [mm]
A	Loudspeaker	0.770	8.15	4.45
G	Battery	0.812	7.8	5.3

Table 3 - Properties of resilient elements.

Symbol	Component	Stiffness			Loss factor
		s_x [N/m]	s_y [N/m]	s_z [Nm]	
S1	Loudspeaker suspension	180	180	$7.47\cdot10^{-4}$	0.1
S2, S3, S6, S9	Contact springs	∞	∞	3.33	0.01
S4, S7	Loudspeaker suspension	180	180	$7.47\cdot10^{-4}$	0.1
S5, S8	Battery springs	∞	2475	∞	0.001

The vibration model is developed by using mobility synthesis and each component is described in terms of its complex velocities and forces symbolized by the column vectors $\overline{v}_i$ and $\overline{F}_j$, respectively. By assuming harmonic motion at angular frequency ω, the complex velocities $\overline{v}_i$ at position i generated by the forces $\overline{F}_j$ at position j can be related through a 3×3 mobility matrix $\overline{\overline{Y}}_{ij}$ as $\overline{v}_i e^{i\omega t} = \overline{\overline{Y}}_{ij}\overline{F}_j e^{i\omega t}$. With the time dependence suppressed this is given by

$$\begin{bmatrix} v_x \\ v_y \\ \upsilon_{z'} \end{bmatrix}_i = \begin{bmatrix} Y_{xx} & Y_{xy} & Y_{xz'} \\ Y_{yx} & Y_{yy} & Y_{yz'} \\ Y_{z'x} & Y_{z'y} & Y_{z'z'} \end{bmatrix}_{ij} \begin{bmatrix} F_x \\ F_y \\ M_{z'} \end{bmatrix}_j . \tag{1}$$

Here subscript x, y, and z' refers to the x-direction, y-direction and the rotational direction around z, respectively. By superposition of velocity contributions from forces in all positions the total complex velocities of each component can be described. The mobilities of the masses, mass-less springs and beams can all be found in [2] and the dynamic properties of the non-uniform hook, is obtained by using transmission theory according to Reference [4].

Now, the loudspeaker is excited at its centre by the harmonic force $F_{y,exc}e^{i\omega t}$, and this results in a mixed force and moment excitation at junction 1. The excitation vector, $\overline{F}_{exc}$, acting at junction 1 is thus given as

$$\overline{F}_{exc} = \begin{bmatrix} F_{x.exc} \\ F_{y,exc} \\ M_{z',exc} \end{bmatrix} = \begin{bmatrix} 0 \\ F_{y,exc} \\ -(l_{A,x}/2)F_{y,exc} \end{bmatrix}, \tag{2}$$

where $l_{A,x}$ is the length of the loudspeaker in the x-direction. In the following the harmonic responses for each component are derived by using mobility synthesis. Continuity in all junctions is already included in the equations. Further, it is assumed that velocities in the left end of the hook, denoted junction 4, are the same at all three spring connections. Spring forces are defined as positive when the springs are compressed. Altogether, the equations of motion include 13 unknown velocity vectors and 9 unknown spring force vectors. In these equations the first subscript of the mobilities refers to the component, whereas the second and the third subscript refer to the position of the velocity vector and the force vector, respectively. The derived harmonic responses for the components or positions in Figure 2 are:

Loudspeaker: $$\bar{v}_1 = \overline{\overline{Y}}_A(\overline{F}_{exc} - \overline{F}_{S_1} - \overline{F}_{S_4} + \overline{F}_{S_7}) \qquad (3)$$

Pipe: $$\bar{v}_2 = \overline{\overline{Y}}_{B,2,2}\overline{F}_{S_1} - \overline{\overline{Y}}_{B,2,3}\overline{F}_{S_2} \qquad \bar{v}_3 = \overline{\overline{Y}}_{B,3,2}\overline{F}_{S_1} - \overline{\overline{Y}}_{B,3,3}\overline{F}_{S_2} \qquad (4\text{-}5)$$

Hook: $$\bar{v}_4 = \overline{\overline{Y}}_C(\overline{F}_{S_2} + \overline{F}_{S_3} + \overline{F}_{S_6}) \qquad (6)$$

Upper shell: $$\bar{v}_5 = -\overline{\overline{Y}}_{D,5,5}\overline{F}_{S_3} + \overline{\overline{Y}}_{D,5,6}\overline{F}_{S_4} + \overline{\overline{Y}}_{D,5,7}\overline{F}_{S_5} + \overline{\overline{Y}}_{D,5,8}\overline{F}_{S_9} \qquad (7)$$

$$\bar{v}_6 = -\overline{\overline{Y}}_{D,6,5}\overline{F}_{S_3} + \overline{\overline{Y}}_{D,6,6}\overline{F}_{S_4} + \overline{\overline{Y}}_{D,6,7}\overline{F}_{S_5} + \overline{\overline{Y}}_{D,6,8}\overline{F}_{S_9} \qquad (8)$$

$$\bar{v}_7 = -\overline{\overline{Y}}_{D,7,5}\overline{F}_{S_3} + \overline{\overline{Y}}_{D,7,6}\overline{F}_{S_4} + \overline{\overline{Y}}_{D,7,7}\overline{F}_{S_5} + \overline{\overline{Y}}_{D,7,8}\overline{F}_{S_9} \qquad (9)$$

$$\bar{v}_8 = -\overline{\overline{Y}}_{D,8,5}\overline{F}_{S_3} + \overline{\overline{Y}}_{D,8,6}\overline{F}_{S_4} + \overline{\overline{Y}}_{D,8,7}\overline{F}_{S_5} + \overline{\overline{Y}}_{D,8,8}\overline{F}_{S_9} \qquad (10)$$

Lower shell: $$\bar{v}_9 = -\overline{\overline{Y}}_{E,9,9}\overline{F}_{S_6} - \overline{\overline{Y}}_{E,9,10}\overline{F}_{S_7} - \overline{\overline{Y}}_{E,9,11}\overline{F}_{S_8} - \overline{\overline{Y}}_{E,9,12}\overline{F}_{S_9} \qquad (11)$$

$$\bar{v}_{10} = -\overline{\overline{Y}}_{E,10,9}\overline{F}_{S_6} - \overline{\overline{Y}}_{E,10,10}\overline{F}_{S_7} - \overline{\overline{Y}}_{E,10,11}\overline{F}_{S_8} - \overline{\overline{Y}}_{E,10,12}\overline{F}_{S_9} \qquad (12)$$

$$\bar{v}_{11} = -\overline{\overline{Y}}_{E,11,9}\overline{F}_{S_6} - \overline{\overline{Y}}_{E,11,10}\overline{F}_{S_7} - \overline{\overline{Y}}_{E,11,11}\overline{F}_{S_8} - \overline{\overline{Y}}_{E,11,12}\overline{F}_{S_9} \qquad (13)$$

$$\bar{v}_{12} = -\overline{\overline{Y}}_{E,12,9}\overline{F}_{S_6} - \overline{\overline{Y}}_{E,12,10}\overline{F}_{S_7} - \overline{\overline{Y}}_{E,12,11}\overline{F}_{S_8} - \overline{\overline{Y}}_{E,12,12}\overline{F}_{S_9} \qquad (14)$$

Battery: $$\bar{v}_{13} = \overline{\overline{Y}}_G(\overline{F}_{S_8} - \overline{F}_{S_5}) \qquad (15)$$

Springs: $$\bar{v}_1 - \bar{v}_2 = \overline{\overline{Y}}_{S_1}\overline{F}_{S_1} \qquad \bar{v}_3 - \bar{v}_4 = \overline{\overline{Y}}_{S_2}\overline{F}_{S_2} \qquad \bar{v}_5 - \bar{v}_4 = \overline{\overline{Y}}_{S_3}\overline{F}_{S_3} \qquad (16\text{-}18)$$

$$\bar{v}_1 - \bar{v}_6 = \overline{\overline{Y}}_{S_4}\overline{F}_{S_4} \qquad \bar{v}_{13} - \bar{v}_7 = \overline{\overline{Y}}_{S_5}\overline{F}_{S_5} \qquad \bar{v}_9 - \bar{v}_4 = \overline{\overline{Y}}_{S_6}\overline{F}_{S_6} \qquad (19\text{-}21)$$

$$\bar{v}_{10} - \bar{v}_1 = \overline{\overline{Y}}_{S_7}\overline{F}_{S_7} \qquad \bar{v}_{11} - \bar{v}_{13} = \overline{\overline{Y}}_{S_8}\overline{F}_{S_8} \qquad \bar{v}_{12} - \bar{v}_8 = \overline{\overline{Y}}_{S_9}\overline{F}_{S_9}\,. \qquad (22\text{-}24)$$

It should be kept in mind that all the unknown vectors include three motion coordinates and the matrix equations (3)-(24) therefore comprise of a system of 66 equations with 66 unknown velocities and spring forces. Now, if all terms in the equations involving velocities and spring forces are collected on the left-hand side of the equality signs, and thus isolating the terms involving the excitation forces $\overline{F}_{exc}$ on the right-hnd side, then the system of equations can be expressed as one matrix equation as

$$\overline{\overline{H}}_X\overline{X} = \overline{\overline{H}}_F\overline{F}_{exc} \quad \Leftrightarrow \quad \begin{bmatrix} \quad 66 \text{ x } 66 \quad \end{bmatrix}\begin{bmatrix} 66 \end{bmatrix} = \begin{bmatrix} 66 \text{ x } 3 \end{bmatrix}\begin{bmatrix} 3 \end{bmatrix}, \qquad (25)$$

where $\overline{X}$ is a column vector with 66 elements containing all velocities and spring forces, $\overline{\overline{H}}_X$ is a 66×66 element matrix containing the terms in front of the velocities (zeros and ones) and the terms in front of the spring forces (mobilities), and $\overline{\overline{H}}_F$ is a 66×3 element matrix containing the terms in front of the excitation forces (mobilities). Finally, the system of equations can be solved numerically for $\overline{X}$ for one frequency at a time by multiplying with the inverted matrix $\overline{\overline{H}}_X^{-1}$ on both sides of eq. (25), which yields

$$\overline{X} = \overline{\overline{H}}_X^{-1} \overline{\overline{H}}_F \overline{F}_{exc} . \tag{26}$$

Damping in the beam components and in the springs are included by introducing complex bending stiffness and complex spring constants, respectively, with corresponding damping loss factors. Furthermore, the loss factor of the hook, and the stiffness of the loudspeaker mounts have been determined experimentally. Other spring stiffnesses have been estimated theoretically.

NUMERICAL RESULTS

In the following numerical results from the vibration model are presented. In this preliminary study only two types of motion coordinates have been used. These are the velocities in the y-direction and in the rotational direction around z. The vibration of the hearing aid is here characterized in terms of the squared velocity ratio $< v^2 > / | v_{free} |^2$ where $< v^2 >$ is the spatially averaged mean-square velocity of the shells and $| v_{free} |^2$ is the squared magnitude of the free velocity of the loudspeaker in the y-direction. This free velocity is a convenient reference since it can be determined experimentally. The loudspeaker is modeled as a pure mass M_A and when this is excited by $F_{y,exc}$ its free velocity becomes

$$v_{free} = \frac{F_{y,exc}}{i\omega M_A} . \tag{27}$$

Figure 3 shows the normalized mean-square velocity of the shells for three different values of the spring stiffnesses of S_2, S_3, and S_6. It is seen that several resonances occur within the frequency range considered. And although the complex system vibrates as a whole, it is possible to identify the main cause of most of these resonances by performing a parameter study. For the dimensions and elastic properties chosen each resonance peak can be assigned to an individual component mode as indicated in Figure 3. The occurrence of the two lowest resonances at 58 Hz and 264 Hz are primarily controlled by the soft loudspeaker suspension e.g. springs

S_1, S_4, and S_7, because the hearing aid vibrates as a two-degree-of-freedom mass-spring-mass system with the loudspeaker as one mass, the suspension as the spring and the rest of the hearing aid as the second mass. In the frequency range above the mass-spring-mass modes the suspension works efficiently as a vibration isolator. The overall decrease in vibration level of the shells is about 40 dB per decade.

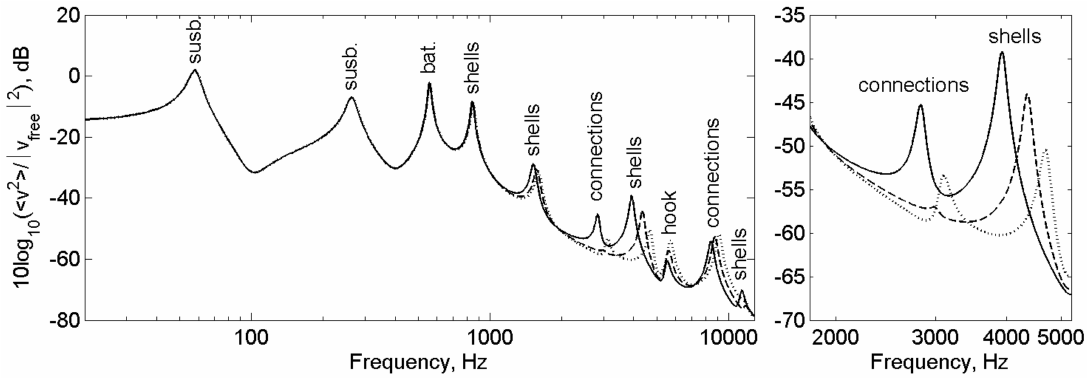

Figure 3. Frequency variation of normalized mean-square velocity of shells. For different values of stiffness $s_{z'}$ of the contact springs S_2, S_3, and S_6: ——, $s_{z'}$=3.33 Nm; - - -, $s_{z'}$=6.66 Nm; ·····, $s_{z'}$=10.0 Nm.

Especially the shells produce many resonances in the frequency range from 840 Hz to 11.4 kHz but also the resiliently connected battery, the hook and the more or less rigid connections (S_2, S_3, and S_6) may contribute to feedback problems. The connections in particular are difficult to describe dynamically and in order to investigate how sensitive the model is to small changes the mean-square velocity of the shells has been plotted for three different cases of the connection springs. Focusing on the frequency range from about 2000 Hz to 5000 Hz it is seen from the "zoom" in Figure 3 that a doubling of the connection spring stiffness from $s_z = 3.33\,\text{Nm}$ to $6.66\,\text{Nm}$ causes the resonance peak at 2823 Hz with a vibration level of -45.3 dB to almost vanish. Further, by increasing the stiffnesses to $10.0\,\text{Nm}$, the resonance peak moves up to 3102 Hz and becomes significant with a vibration level of -53.4 dB. Additionally, a large drop of more than 10 dB in vibration level of the resonant peaks caused by the shells can be observed. This parameter study reveals that small changes may remove or produce troublesome resonance peaks.

Figure 4 shows the normalized mean-square velocity of the shells for four different values of the loudspeaker suspension stiffnesses S_4 and S_7. Also shown in Figure 4 is a case where the loudspeaker is rigidly connected to the shells. Vibration at this condition forms the basic or reference for evaluating the isolation effectiveness of the resilient suspensions. Naturally, when the stiffness of the springs S_4 and S_7 is increased then the previously mentioned two mass-spring-mass modes moves towards higher frequencies. The frequency range where the suspension works as a vibration isolator is accordingly moved upwards and gives a poorer isolation in the whole frequency range. An increase of the stiffness values of three and ten times thus result in an increase of the overall vibration level of about 9 and 20 dB, respectively.

Especially in the frequency range from about 400 Hz to 1000 Hz the vibration isolation is very poor due to modes caused primarily by the battery and the shells. Nevertheless, at 10 kHz the vibration level is significantly lower than the reference even in the poorest performing case.

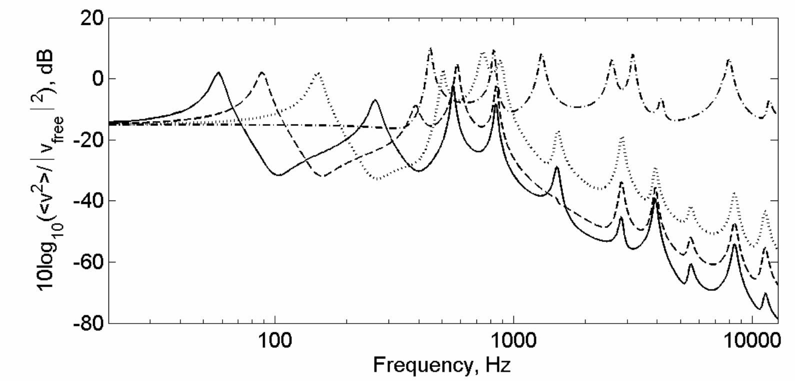

Figure 4. Frequency variation of the normalized mean-square velocity of the shells for different values of springs stiffnesses S_4 and S_7: ——, nominal values as in Table 1.3 ; - - -, three times larger; ·····, ten times larger; –·–, infinitely stiff.

CONCLUSIONS

The vibration patterns of a so-called "behind the ear" hearing aid has been examined through a simple mathematical vibration model developed by using mobility synthesis. In a first attempt wide, simple beams were chosen to represent approximately the dynamic characteristics of the complex shells. For this choice of components it was revealed that several structural resonances occur in the frequency range from 58 Hz and up to 10 kHz and these are caused by the loudspeaker suspension and individual component modes. Further, it was shown that little changes in the structural parameters may have a large effect on the vibrations of the simulated shells. Finally, the effect of the loudspeaker suspension was investigated, and this showed that appreciable vibration isolation is obtained in the frequency range above 1000 Hz.

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