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Numerical study of the broadband vibro-acoustic response of an earmuff

K. Carillo[a,*], F. Sgard[b], O. Doutres[a]

[a] Department of Mechanical Engineering, École de Technologie Supérieure, 1100 rue Notre-Dame Ouest, Montréal, Québec H3C 1K3, Canada
[b] IRSST, Direction Scientifique, 505 Boulevard de Maisonneuve Ouest, Montréal, Québec H3A 3C2, Canada

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ABSTRACT

In this paper, the vibro-acoustic response of a passive earmuff in a broadband frequency range (100 Hz to 5 kHz) is investigated using a finite element analysis. Firstly, the study focuses on the vibro-acoustic response of the cup and the back-plate regardless of the comfort cushion contribution. Secondly, emphasis is put on the foam-filled comfort cushion which is the trickiest component to model because of its physical complexity. This multiphasic cushion is modeled in a simplified way as an equivalent solid, either isotropic or transverse isotropic in order to take into account the added transverse stiffness due to the bulging of the cushion polymeric sheath. The accuracy of these models is investigated by comparing the simulated insertion loss (IL) to measurement data. The IL predicted with the isotropic cushion model is highly underestimated between 500 Hz and 2.5 kHz due to the presence of an unrealistic mode of transverse deformation. It is found that (i) neglecting the acoustic excitation on the cushion' external flanks of the isotropic model or (ii) using the transverse isotropic cushion model significantly improves the simulated IL.

1. Introduction

Passive earmuffs are commonly used when the sound level cannot be reduced at the source [1]. Their performance is characterized by their sound attenuation. The earmuff sound attenuation can be predicted using analytical models such as the lumped parameters model (LPM) [2–5] in which the earmuff is considered as a mass-spring-damper system. The cup and the back-plate are considered to behave as a rigid mass while the comfort cushion and the internal air cavity constitute the spring-damper part. In addition, the cushion is assumed to be fixed on its face in contact with the flesh and the acoustic excitation is uniform over the surface of the cup. Therefore, the use of a LPM is restricted to low frequencies ($<$ 500 Hz). Alternatively, the broadband frequency vibro-acoustic response of an earmuff can be computed using numerical approaches such as the finite element (FE) method and/or the boundary element (BE) method [4,6–11]. Even though the trend of the estimated attenuation's frequency behavior provided by these models [4,7,8] is close to that obtained during experiments, level discrepancies up to 40 dB can be observed. These deviations are attributed either to the limits of the experimental set-up [7,8] or to a lack of knowledge of the material parameters used in the FE model [4,7], or to the modeling and characterization of the comfort cushion [4,8]. The latter is either modeled as a dashpot-spring system [6] or as an isotropic equivalent solid (ES) [4,8]. Currently, no broadband (100 Hz to 5 kHz) numerical model is available to predict accurately the vibro-acoustic response of an earmuff and very few details are given about the acoustic role of each earmuff components as a function of frequency.

In this work, a FE model of the broadband vibro-acoustic response of a commercial earmuff (EAR-MODEL-1000, 3M™E-A-R™, Indianapolis, USA) is proposed (Fig. 1). It is based on the one developed and validated at low frequencies ($<$ 500 Hz) by Boyer et al. [5]. The accuracy of the FE model is assessed by comparing the numerical results with existing experimental measurement data [12]. The numerical analysis is carried out in two steps. The first one focuses on the vibro-acoustic response of the cup and the back-plate regardless of the contribution of the comfort cushion. In this configuration, the comfort cushion is replaced by a motionless lead cushion. The second configuration accounts for the comfort cushion. Three alternative modeling of this one are investigated: isotropic ES whose external lateral walls are either (i) excited acoustically or (ii) not excited and (iii) transverse isotropic ES whose external lateral walls are excited acoustically. The transverse isotropic cushion model can be considered as an intermediate model between the isotropic ES model and a fully detailed model, not studied yet, where each component (sheath, foam, air) of the cushion is modeled. In all studied configurations, the sound absorbing foam usually located inside the cup is removed in order to focus on the structural components (cup, back-plate and cushion).

* Corresponding author.
E-mail address: kevin.carillo.1@ens.etsmtl.ca (K. Carillo).

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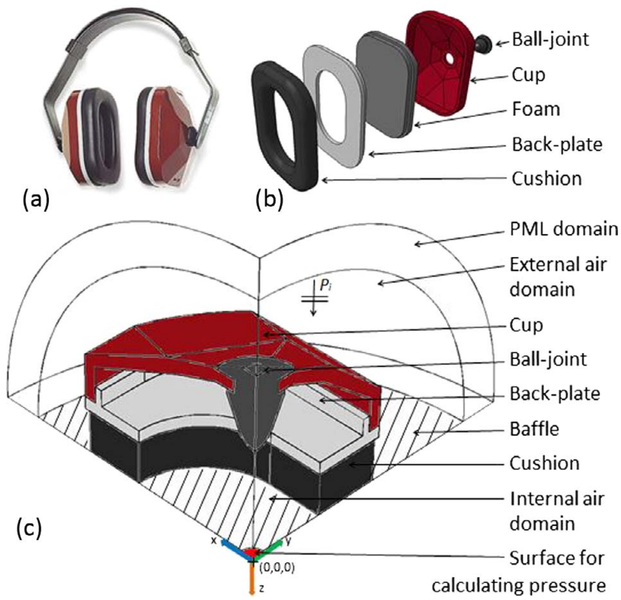

Fig. 1. (a) EAR-MODEL-1000 earmuff, (b) exploded view of one earcup and (c) view of the FE model ($x = 0$ and $y = 0$ are the symmetry planes, $z = 0$ corresponds to the baffle plane).

2. Experimental measurements

2.1. Experimental set-up

The measurements that are used for comparisons with simulations were carried out by Boyer et al. [12]. In their experimental set-up (Fig. 2), one single earcup of the EAR-MODEL-1000 earmuff was attached to a rigid baffle situated in a hemi-anechoic chamber. A microphone flush mounted in the baffle captured the acoustic pressure inside the earmuff. The acoustic excitation was ensured by a loudspeaker placed at 1 m from the baffle and generating a pink noise at an overall sound pressure level between 85 and 90 dB. The acoustic indicator used was the insertion loss (IL) defined as the difference between the sound pressure level measured without and with the earcup. Its calculation is detailed in Section 3.5.1 of the present paper. Contrary to the ANSI/ASA S12.42-2010 standard, IL measurements were therefore not performed using an acoustic test fixture (artificial head) but rather using a rigid plane in order to simplify the FE model.

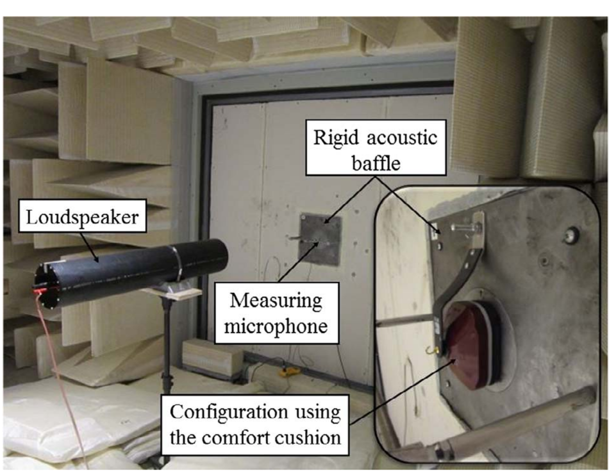

Fig. 2. Experimental set-up developed by Boyer et al. [12].

2.2. Configurations of interest

Among the various configurations considered by Boyer et al. in [12], two of them are selected in this work. In the first one, a lead cushion is used instead of the comfort cushion so that there is no pumping motion and no sound transmission through its lateral walls. The lead cushion is glued on one side to the back-plate and on the other side to the baffle with a synthetic butyl rubber sealant (MONO acoustical sealant) [12]. In the second configuration, the comfort cushion is glued to the back-plate using the adhesive surface provided by the manufacturer and is compressed onto the baffle by the headband. The NRR estimated from noise attenuation measurements of the later configuration is 17 dB (note that the missing data at high frequencies required for NRR calculations have been taken equal to the values at the maximum measured frequency). This NRR is very close to the one provided by the manufacturer (NRR = 20 dB) and the difference can be attributed to the fact that the foam pad is removed from the cup in the present work as mentioned previously.

3. FE model

3.1. General configuration

The two configurations presented in Section 2.2 are simulated using a FE model (Fig. 1, c) solved with COMSOL Multiphysics 5.2 (COMSOL®, Stockholm, Sweden). The headband is not accounted for in the FE model because preliminary simulations, not presented here for the sake of conciseness, have shown that the stiffness of the headband and its diffraction effect are negligible. The clamping force of the headband is taken into account through the static deformation of the comfort cushion which impact its mechanical properties [5]. The incoming sound field is assumed to be an incident plane wave which propagates toward the baffle in the normal direction. Thus, thanks to the symmetry for both geometry and loading, only one quarter of the earcup is modeled. The couplings between solid domains are considered as perfect (continuity of stresses and displacements) while the solid-fluid couplings reflect the continuity at the interface of the normal structural and acoustic displacements on the one hand and of the structural stress vector and acoustic force per unit area on the other hand. The cup and the back-plate are made of ABS (Acrylonitrile Butadiene Styrene) while the ball-joint is made of rubber. These components are modeled as isotropic elastic linear domains and their mechanical parameters are given in Table A.2 in Appendix A.1. The internal earmuff air cavity and the external air domains are modeled as compressible perfect gas domains and their physical properties are given in Table A.3 in Appendix A.1. In the internal air domain, the dissipation induced by visco-thermal effects at the boundaries is accounted for using a structural loss factor η_a.

3.2. Cushion models

The comfort cushion is a complex assembly of a foam surrounded in a non-homogeneous way by a thin polymer sheath (Fig. 6, b). It is modeled in a simplified way as an equivalent solid (ES). Its geometry is also simplified as done in [5]. Indeed, its lateral walls are assumed to be straight (Fig. 1, c) and not curved as observed in the case of the real compressed comfort cushion (Fig. 6, b). Its thickness of 10.9 mm corresponds to a static compression rate of 19.8% [5]. The lead cushion geometry is identical to the one of the comfort cushion but its thickness is 12.5 mm rather than 10.9 mm. All cushion models are fixed on their face in contact with the baffle ($x = 0$ plane).

The lead cushion is modeled as an isotropic linear elastic domain and its mechanical parameters are given in Table A.2 in Appendix A.1. Regarding the contact between the lead cushion and the back-plate, two cases are considered: (i) a perfect contact (PC) and (ii) a viscoelastic contact (VC). The VC model takes into account the glue layer (estimated