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Modelling nonlinear viscoelastic behaviours of loudspeaker suspensions-like structures

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ABSTRACT

Mechanical properties of an electrodynamic loudspeaker are mainly determined by its suspensions (surround and spider) that behave nonlinearly and typically exhibit frequency dependent viscoelastic properties such as creep effect. The paper aims at characterizing the mechanical behaviour of electrodynamic loudspeaker suspensions at low frequencies using nonlinear identification techniques developed in recent years. A Generalized Hammerstein based model can take into account both frequency dependency and nonlinear properties. As shown in the paper, the model generalizes existing nonlinear or viscoelastic models commonly used for loudspeaker modelling. It is further experimentally shown that a possible input-dependent law may play a key role in suspension characterization.

1. Introduction

Suspensions (surround and spider) play an important role in electrodynamic loudspeaker design and operation. Their role is twofold: first to centre and adjust the voice-coil in the magnetic air gap, allowing an axial motion of the diaphragm while preventing lateral motion or rocking; second to provide the restoring force. The materials used together with the assembly geometry usually result in a complex nonlinear viscoelastic behaviour, even at low amplitude of excitation.

On the one hand, there are many studies dealing with mechanical systems and their nonlinear dynamics including viscoelastic properties [1–3] that take into account many nonlinear phenomena, temperature dependence [4,5] or even time-variation [6] of the viscoelastic materials. On the other hand, in the today's most used classical model [7], the moving part of a loudspeaker is usually described by a simple mass-spring-damper linear system. Besides, more accurate linear descriptions have been proposed, taking into account the frequency dependence of damping and/or stiffness due to viscoelasticity [8,9]. Since there are many nonlinear phenomena including the viscoelastic effects of the suspensions [10], the classical linear model is not sufficient for describing the loudspeaker behaviour for larger amplitudes. In order to describe at least partly these phenomena, the stiffness of the mass-spring damper model is usually described as a nonlinear function of instantaneous displacement expressed in a polynomial way [10].

Even if the viscoelastic and nonlinear behaviours are known for decades [11], none of the existing models of loudspeaker suspensions take simultaneously both effects (frequency dependence of damping and/or stiffness and nonlinear effects) into account. In this paper, we propose a nonlinear model of the moving part of an electrodynamic loudspeaker taking into account

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both the nonlinear behaviour together with the complex viscoelastic phenomena.

The structure of the proposed model, derived from the basis of existing viscoelastic and nonlinear models in Section 2, is shown to be very similar to the structure of the so called Generalized Hammerstein model with polynomial inputs (Section 3) that has been successfully used to model nonlinearities in other fields of physics [12–14] and for which several measurement techniques have been developed in recent years [15–17]. To apply one of these techniques for the study of dynamical behaviour of loudspeaker suspensions, we propose an experimental bench (Section 4) in which the mechanical part of the loudspeaker is separated from the loudspeaker to be measured apart. The experimental results from the measurement on an off-the-shelf loudspeaker are presented in Section 6 and a discussion of the main important results is proposed in Section 6 with a concluding summary.

2. State of the art

In its simplest form, the behaviour of the moving part of an electrodynamic loudspeaker is described by a mass-spring-damper equation

$$M_{ms}\,\frac{\mathrm{d}^2x(t)}{\mathrm{d}t^2} + R_{ms}\,\frac{\mathrm{d}x(t)}{\mathrm{d}t} + K_{ms}\,x(t) = F(t), \tag{1}$$

with M_{ms} the mass of the diaphragm, R_{ms} and K_{ms} the damping (also called mechanical resistance) and the stiffness of the suspension, respectively, $F(t)$ the force created by the current passing through the voice-coil, and $x(t)$ the displacement of the moving part (considering only a piston motion). The relation (1) suffers from two drawbacks: first it is valid only for small displacements corresponding to linear behaviour and, second, it does not take into account the viscoelastic properties of the suspensions. Both drawbacks and existing solutions are briefly described below in the remaining of this section.

As shown in Refs. [10,18–21], expression (1) does not succeed in describing the nonlinear behaviour in the large signal domain. Indeed, in case of high input level, the stiffness factor K_{ms} can no longer be considered as constant-valued but varies with the displacement $x(t)$. This behaviour can be modelled using polynomial approximations [10,22–24]. Besides, the damping R_{ms} is usually considered independent of displacement or velocity. However, as suggested in Ref. [25] and later demonstrated in Ref. [26], the damping R_{ms} can also vary with instantaneous velocity and/or displacement [27,28].

Considering here both the damping R_{ms} and the stiffness K_{ms} nonlinearly depending on the instantaneous displacement $x(t)$, we can modify the expression (1) in the following manner

$$M_{ms}\,\frac{\mathrm{d}^2x(t)}{\mathrm{d}t^2} + \left(\sum_{n=1}^{N} r_n \cdot x^{n-1}(t)\right)\,\frac{\mathrm{d}x(t)}{\mathrm{d}t} + \left(\sum_{n=1}^{N} k_n \cdot x^{n-1}(t)\right)\,x(t) = F(t), \tag{2}$$

$N \in \mathbb{N}^*$ being the model order, and the parameters r_n and k_n being the coefficients of the polynomial expansion respectively for the damping and the stiffness.

As shown in Refs. [8,9,29,30], the linear mass-spring-damper Eq. (1) has a limited use even in small signal domain. Indeed, materials used for loudspeaker suspensions exhibit viscoelastic properties leading to behaviours that are far more complex than those described with a simple mass-spring-damper representation. In Ref. [8], several linear models considering frequency dependent parameters are described, the most accurate one being the so-called LOG (logarithmic) model. In Ref. [31], fractional derivatives are successfully used to model the viscoelastic behaviour. All these models show that either an extra parameter must be added to the mass-spring-damper system, or frequency dependent parameters K_{ms} and R_{ms} must be considered.

The time domain relation between force and displacement from Eq. (1) can be expressed in the frequency domain including the frequency dependent parameters $K_{ms}(f)$ and $R_{ms}(f)$ as (with $\mathrm{i} = \sqrt{-1}$)

$$-(2\pi f)^2\,M_{ms}\,X(f) + \mathrm{i}2\pi f\,R_{ms}(f)\,X(f) + K_{ms}(f)\,X(f) = F(f). \tag{3}$$

In order to consider both effects presented in this section (the instantaneous displacement dependence and the frequency dependence of both damping and stiffness) a so-called viscoelastic nonlinear model is proposed in the next section.

3. Visco-elastic nonlinear model

The model proposed in this paper takes into account the frequency dependence of stiffness and damping from Eq. (3) together with the nonlinear functions represented in time domain in Eq. (2), we either have to express the frequency dependence of parameters K_{ms} and R_{ms} in time domain using a convolution or to express the nonlinear laws in frequency domain. Both solutions being equivalent from the mathematical point of view, we choose to use the frequency domain for the sake of simplicity.

Considering

$$X^{(n)}(f) = \mathcal{F}\left\{x^n(t)\right\}, \tag{4}$$

and

$$\mathcal{F}\left\{x^{n-1}(t)\frac{\mathrm{d}x(t)}{\mathrm{d}t}\right\} = \mathcal{F}\left\{\frac{1}{n}\frac{\mathrm{d}x^n(t)}{\mathrm{d}t}\right\} = \mathrm{i}\frac{2\pi f}{n}X^{(n)}(f), \tag{5}$$