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ACHIEVING DIRECT ACOUSTIC IMPEDANCE CONTROL WITH ONLY TWO MICROPHONES

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ABSTRACT

This work presents a control architecture of an active electroacoustic absorber that does not require an accurate model of the transducer thanks to a non-intrusive feedback system. This control could therefore be implemented on other transducers where the dynamics are not well known. Measurements were conducted to compare this architecture with a state-of-the-art feed-forward control scheme.

1. INTRODUCTION

Active electroacoustic absorption is an efficient way to reduce noise in cavities and ducts and is a promising technology for aircraft engine noise reduction [1]. Hybrid sensor-/shunt-based impedance control [2] has proved effective but requires an accurate model of the electroacoustic actuator.

Alternatively, by sensing the pressure on the electrodynamic absorber membrane simultaneously with its velocity, a target acoustic impedance can be achieved on electroacoustic actuators, without requiring a thorough estimation of the transducer model parameter [3–5]. In the proposed implementation, a microphone located inside the enclosure of the loudspeaker measures a signal proportional to the membrane displacement, which allows deriving the membrane velocity. The great advantage of this control scheme is that no model of the electrodynamic loudspeaker is required except the estimation of the back cavity specific compliance, which can easily be obtained by measuring the membrane velocity once.

2. CONTROL STRATEGY

A passive speaker is a second order resonator. Its specific impedance Z_{ss} is defined as the ratio between pressure and normal particle velocity on the membrane. It is expressed as

$$Z_{ss}(s)|_{i=0} = R_{ss} \frac{s^2 + s\omega_0/Q_{ms} + \omega_0^2}{s\omega_0/Q_{ms}} \quad (1)$$

where s is the Laplace variable, ω_0 the angular resonant frequency and Q_{ms} the mechanical quality factor. The specific resistance R_{ss} is defined as the mechanical

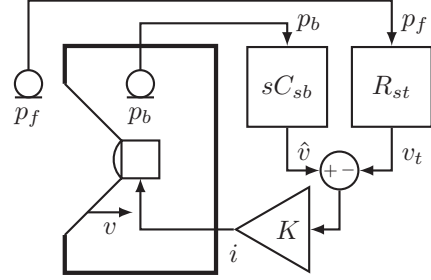


Figure 1. Control architecture

resistance R_{ms} divided by the speaker effective piston area S_d . Furthermore, when driven by a current i , a Lorentz force acts on the membrane. Its behavior is modeled by

$$v = \frac{p_f - i \cdot Bl/S_d}{Z_{ss}} \quad (2)$$

where p_f is the acoustic pressure in front of the membrane and Bl the coil force factor.

The active impedance control system presented in [3, 4] estimates the membrane velocity by integrating the signal generated by an accelerometer placed on the speaker cone. This approach is especially impractical for small absorbers where the accelerometer would be too big and an incompatible added moving mass. On the other hand, Meynial et al. [5] use a Wheatstone bridge to estimate the membrane velocity, which requires a fine tuning of resistors.

In this paper, the membrane velocity is estimated via a microphone placed in the cabinet of the speaker, as shown in figure 1. This signal p_b is numerically derivated and multiplied by the box specific compliance C_{sb} to obtain a velocity estimation \hat{v} . C_{sb} is linked to the box volume V_b by

$$C_{sb} = \frac{V_b}{\rho c^2 S_d} \quad (3)$$

where ρ the air mass density and c the speed of sound.

The target velocity v_t is obtained by dividing the acoustic front pressure p_f by the target specific resistance R_{st} . With these two velocities, a proportional controller with gain $K \geq 0$ is used ($K > -R_{ms}/(Bl)$ to be stable). The achieved specific impedance is

$$Z_{sa} = R_{st} \frac{Z_{ss} + K \cdot Bl/S_d}{R_{st} + K \cdot Bl/S_d} \quad (4)$$

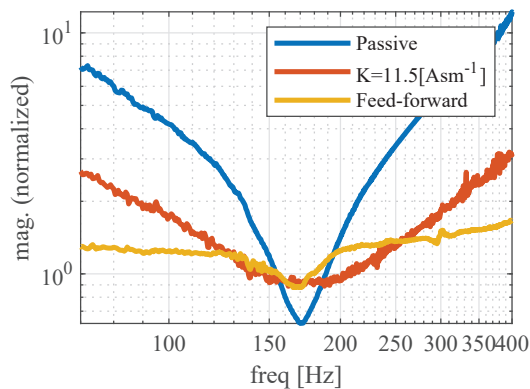


Figure 2. Achieved impedance with a target of ρc

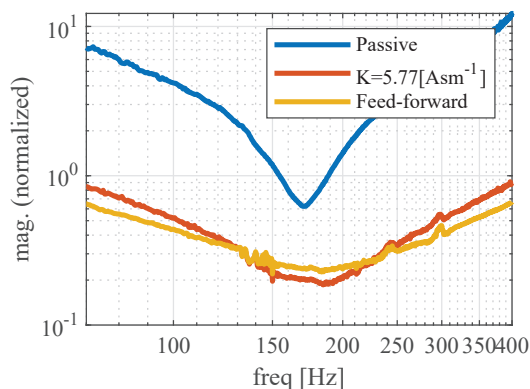


Figure 3. Achieved impedance with a target of $\rho c/5$

which tends to R_{st} when K tends to infinity. Even though the ratio Bl/S_d appears in (4), it has no real impact because it acts as a scaling on K . The only needed physical parameter is the enclosure acoustic compliance C_{sb} . It can easily be identified by knowing the enclosure volume or by measuring the ratio between membrane displacement and acoustic pressure in the cavity using the microphone and a vibrometer.

3. EXPERIMENTAL RESULTS

This control scheme was implemented on a Monacor SPX-30M transducer, whose identified parameters are reported in table 1. Note that only C_{sb} needs to be known. Two target impedances were considered: ρc (optimal normal incidence absorption) and $\rho c/5$. In both cases, we target a specific impedance corresponding to the widest bandwidth achievable (without instability) with the proposed control scheme and the feedforward strategy presented in [2].

When targeting a resistance close to the uncontrolled one, as in figure 2, the feed-forward control strategy presents a wider band, but the proposed control is more accurate around the resonance because it does not depend on an uncertain model. However, when the target resistance is much smaller, both control schemes performances are quite similar.

Parameter	Symbol	Value
Effective piston area	S_d	32.0 [cm ²]
Resonant frequency	ω_0	$2\pi \cdot 170$ [Hz]
Mechanical Q factor	Q_{ms}	5.97
Specific resistance	R_{ss}	263 [Pa s m ⁻¹]
Force factor	Bl	3.80 [N A ⁻¹]
Cabinet volume	V_b	1.6 [dm ³]
Density of air	ρ	1.2 [kg m ⁻³]
Speed of sound	c	343 [m s ⁻¹]

Table 1. Monacor SPX-30M parameters

4. DISCUSSION

Although it is still rudimentary, this control scheme has been proven working. Because it does not require a transducer model, it is more accurate around the resonance frequency than what is obtained by a feed-forward scheme. Indeed, the feed-forward architecture is sensitive to model inaccuracies and the controller digital delay. However, there still is room for improvement, especially in the feedback control. Here, a simple proportional controller was built, but more advanced control theory techniques could be applied. Furthermore, because this scheme is not dependent on a model, it is probably capable to control non-linear actuators, such as very low-profile speakers.

5. ACKNOWLEDGEMENTS

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