



SOUND ABSORPTION BY AN ACTIVE RESONATOR IN A TUBE AND IN AN ENCLOSURE

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ABSTRACT

Sound absorption is one of the most efficient ways of noise canceling. At low frequencies significant sound absorption is usually provided by resonant systems. A single resonator (e.g. Helmholtz resonator) with the optimal friction has an absorption cross-section proportional to the squared wavelength. However the resonator absorbs effectively sound power only in a narrow frequency band. We used the remarkable property of resonators to design a local absorber working in wide frequency band. We propose to regulate the impedance of the ordinary resonator by means of active control. The practical construction of the active resonator is described as well. The active resonator has been experimentally tested in two applications.

INTRODUCTION

One of the most efficient sound absorbers is an acoustical resonator. The absorption cross section of the resonator can be much greater its geometrical size. For example, the absorption cross-section of the Helmholtz resonator is equal to $\lambda^2/4\pi$ (where λ is the wavelength) in free space at its eigenfrequency and does not depend on its size. This result is correct when the friction coefficient of resonator is equal to its radiation impedance. This means that the resonator behaves like a perfect passive absorber of area $\lambda^2/4\pi$. Another example of resonators is a gas bubble in liquid. Everybody knows that a glass with champagne does not clink. This fact can be explained by strong sound absorption provided by gas bubbles.

In practice resonators are often used for noise control in enclosures and indoors [1]. Moreover resonators are used for noise absorption in narrow pipes. But the main disadvantage of resonators is the narrow frequency band, in which high noise reduction is reached. Maximal noise absorption is provided only at resonator's eigenfrequency. The common condition of maximal absorption by resonator is given by

$$\operatorname{Re} Z = \operatorname{Re} Z_r, \quad \operatorname{Im}(Z + Z_r) = 0. \quad (1)$$

where Z is the mechanical impedance of resonator, Z_r is the radiation impedance. The mechanical impedance is actually characterized by the mass m , the elasticity κ and the friction coefficient γ

$$Z = -i\omega m + \gamma + \frac{\kappa}{-i\omega}, \quad (2)$$

where ω is the frequency. The radiation impedance Z_r depends on the type of the resonator and the radiation conditions.

The conditions (1) can be accomplished only at resonant frequency $\omega_0 = \sqrt{m/\kappa}$. Furthermore the friction coefficient of resonator has to have the certain value. Moreover the conditions (1) may be considered as the general conditions for maximal absorption provided by arbitrary body which size is much smaller in comparison with a wavelength and usually they are named “resonant conditions”.

In order to use the significant feature of resonator we propose to develop the active control strategy basing on realization of resonant condition for a secondary source. If the impedance of the secondary source satisfies the condition (1) in wide frequency band the secondary source will absorb the maximal value of sound power.

ACTIVE RESONATOR

As a secondary sound source let's consider a loudspeaker contained in the enclosure which size is much smaller a wavelength. The loudspeaker is modeled in the ordinary way as a mass-spring-dashpot system driven by a coil of electrical resistance R and length l in a magnet gap with induction B . The dashpot is characterized by the friction coefficient γ . The piston has a mass m and an area σ . The mechanical stiffness κ includes the suspension stiffness and the effective spring constant of the gas in the enclosure. The equation of motion of the loudspeaker placed in harmonic sound field with the frequency ω and the amplitude P_0 at the position of loudspeaker has the form

$$m\ddot{\xi} = -\gamma\dot{\xi} - \kappa\xi - Z_r\dot{\xi} - P_0\sigma + F_u, \quad (3)$$

where ξ is the displacement of the piston, $F_u = (U - Bl\dot{\xi})Bl/R$ is the force acting on the piston, U is the complex amplitude of input voltage. We suppose the input signal is harmonic with the frequency ω . From (3) we can find the velocity of the piston $\dot{\xi}$ in the following form

$$\dot{\xi} = \frac{Bl}{R(Z + Z_r)}U - \frac{\sigma P_0}{Z + Z_r}. \quad (4)$$

As we can see from (4) the velocity of the piston depends on the sound pressure in the vicinity of the loudspeaker and the input signal. It is obvious that the velocity of the piston can be adjusted by variation of the amplitude and the phase of the input signal. In such a way the impedance of the loudspeaker is adjusted as well. To form the input signal we place a microphone near the loudspeaker (see Fig.1). The microphone measures a sum of the pressure P_0 of the incident wave and the pressure $P_1 = Z_r\dot{\xi}/\sigma$ of the wave radiated by the loudspeaker. The microphone is connected to an amplifier with a transduction coefficient $K = K(\omega)$ depending on the frequency. The signal from the output of the amplifier is the input signal for the loudspeaker U , which can be expressed as

$$U = (P_0 + P_1)K = \left(P_0 + \frac{Z_r}{\sigma}\dot{\xi} \right) K. \quad (5)$$

Substituting this expression into (4) that allows writing the loudspeaker impedance as the ratio of the total force F_1 acting on the piston to its velocity in the form as

$$Z_1 = \frac{-F_1}{\dot{\xi}} = \frac{P_0\sigma + Z_r\dot{\xi}}{\dot{\xi}} = Z \left(1 - \frac{Bl}{R} \frac{K}{\sigma} \right)^{-1}. \quad (6)$$

The impedance Z_1 is a “new” impedance of the loudspeaker obtained due to active control, which is realized by the microphone and amplifier. The mechanical impedance Z depends only on the mechanical parameters of the loudspeaker and $Z_1 = Z$ when active control is off, i.e. $K = 0$.

Thus the described device consisting of the loudspeaker, transducer amplifier (active controller) and microphone is a feedback system that operates to regulate the impedance of the loudspeaker. At that the feedback coefficient is a main parameter of this active system. In order to set required impedance of the loudspeaker we have to find the feedback coefficient. If we want the loudspeaker absorbs maximal sound power then its impedance Z_1 has to satisfy the condition (1). In this case two equations (1) are equal to the equation $Z_1 = Z_r^*$. Substituting this into (6) we find the feedback coefficient

$$K = \sigma \frac{R}{Bl} \frac{Z - Z_r^*}{Z_r^*} . \quad (7)$$

where the superscript $*$ denotes complex conjugation. If the condition (7) is satisfied at some frequency band then the loudspeaker impedance is resonant at this band. So the proposed device has the resonant impedance in wide frequency band and not at one frequency as the ordinary resonator does.

The device presented in Fig.1 behaves with respect to the incident sound wave as the resonator at its eigenfrequency does. It is naturally to name this device “active resonator”. The word “resonator” characterized the acoustical properties of the device and the word “active” explains how these properties are provided.

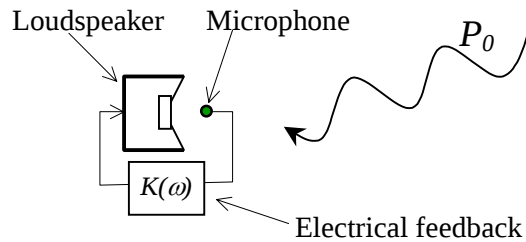


Figure 1. The scheme of the active resonator

The idea of using a secondary source with a nearby sensor to active noise control is not new. Olson and May [2] proposed the electronic sound absorber consisting of a microphone, amplifier and loudspeaker connected so that, for an incident sound, wave and sound pressure at the microphone was reduced. In this system sound pressure is minimized in the vicinity of the loudspeaker whereas in the active resonator the impedance of the loudspeaker is optimized and sound pressure can rise near the active resonator. The suggested method of active control can be referred to impedance matching technique [3,4], which is one of the most popular approaches to active noise and vibration control.

A simple construction of the active resonator permits to realize the proposed method of active control. We will show the application of active resonators by two examples.

SOUND ABSORPTION IN A NARROW PIPE

We assume the active resonator is set in a narrow pipe so it is shown in Fig. 2a. The incident sound wave (denoted by the red arrow) runs in the pipe toward the active resonator. The problem is to absorb the incident wave in wide frequency band. In other words we have to

cancel the wave reflected from the active resonator (denoted by the blue arrow). It is interesting to note that this problem setting is often used for testing of new active control methods [5,6]. The mechanical impedance of the loudspeaker is given by (2). The radiation impedance of the piston in the pipe is equal to $Z_r = \rho c S$, where ρ is the density of the medium, c is the sound velocity, S is the pipe cross section. In this case $\sigma = S$. From (7) the feedback coefficient can be expressed as

$$K = \frac{R}{\rho c B l} (Z - \rho c S), \quad (8)$$

In experimental study we use a circular pipe 10 cm in diameter and 1 m in length, which is shown in Fig. 2a. A primary source is mounted on one end of the pipe and a broadband signal is supplied to the source. The active resonator is placed at the other end. The efficiency of sound absorption is determined by the reduction coefficient, which is the ratio of the amplitude of the reflected wave to the amplitude of the incident wave. The reflection coefficient V is measured by two test microphones by means of standard procedure [7]. The results of measurements are presented in Fig. 2b. The line marked OFF corresponds to turned-off feedback filter. In this case the reflection coefficient is close to 0 dB and, therefore, the incident wave is not absorbed. After adjusting of the active resonator (the line marked ON) significant sound absorption is provided in the wide frequency band. Namely the absorption coefficient $\alpha = 1 - |V|^2$ is more than 99% in the band from 170 Hz till 710 Hz.

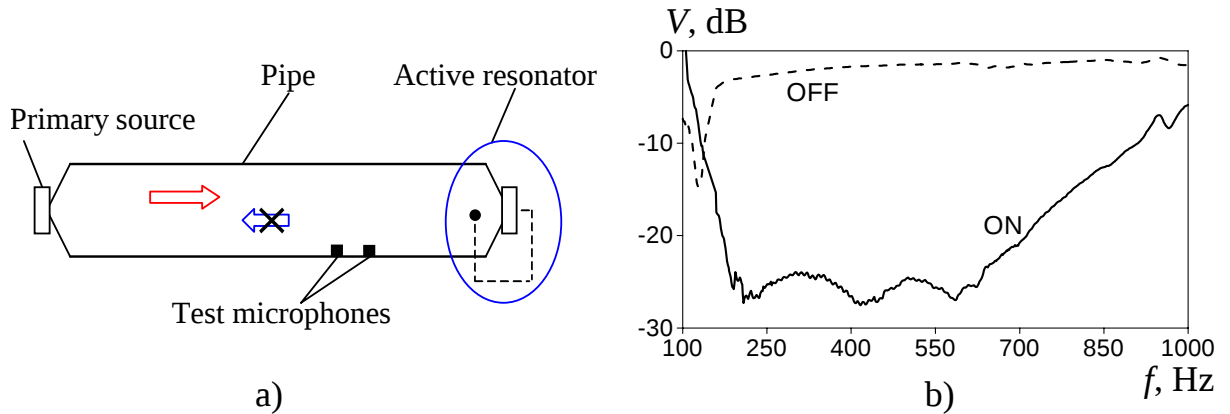


Figure 2. The scheme of the experimental setup (a) and the results of measurements (b)

SOUND ABSORPTION IN AN ENCLOSURE

The active resonator absorbing sound power can suppress the natural resonances of the enclosure. To investigate this effect the active resonator is placed in the rectangular enclosure (Fig. 3). Absorption efficiency may be determined by two means.

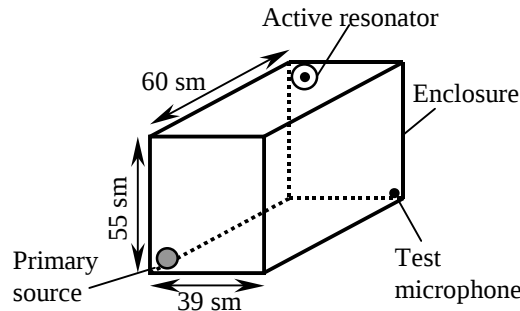


Figure 3. Active resonator in the enclosure

In the first case when the primary source radiates wideband noise the active resonator reduces resonant peaks in spectrum of pressure measured by test microphone. The reduction of the peaks is about 2-5 dB at frequency band from 550 Hz till 900 Hz (Fig. 4a). The first natural frequencies of the enclosure are 310, 340 and 450 Hz. The last frequency is far from the others that allows to estimate sound absorption by reduction of pressure in time domain at the test microphone. The primary source radiates sound wave with the frequency 450 Hz and is turned off at the moment $t = 20$ ms. The record of decreasing signal is represented in Fig. 4b. We can see that the active resonator reduces characteristic time of decreasing by a factor of 1.8. These results permit to suggest using of the active resonators for low frequency absorption in rooms and halls. But a required quantity of active resonators increases with the volume of the room.

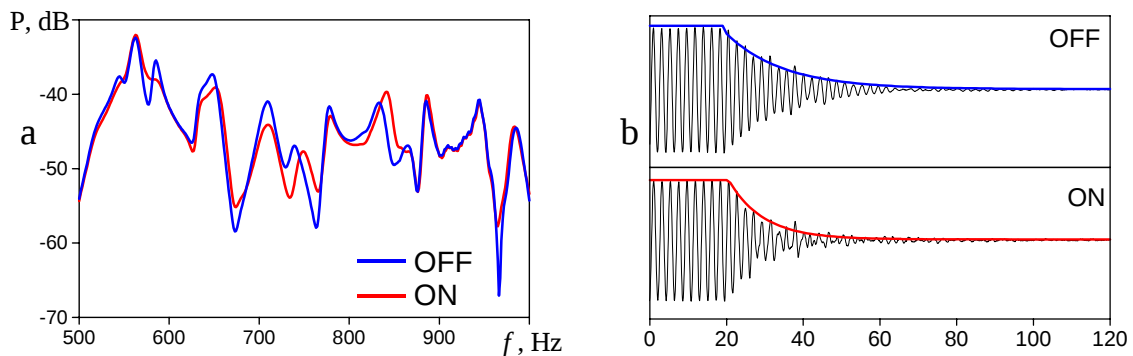


Figure 4. Sound pressure at the test microphone (a); reduction of pressure after turning off the primary source (at the moment $t = 20$ ms) at 450 Hz (b).

SUMMARY

The active resonator proposed in this paper proves to have chances to be applied in different noise control problems due to the simplicity of strategy and design. Examples demonstrated above reveal an opportunity of applying the active resonators to architectural acoustics and noise control in narrow pipes. Note that the active resonator has already been used with success for reduction of noise radiated from the open tube [8].

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