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Prototype test of a resistance control device for Helmholtz resonators

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ABSTRACT

The resistance of the resonator neck determines the sound dissipation and the sharpness of resonance of a resonator, while its cavity volume, sectional area and length of the resonator neck determine its resonant frequency. Out of these control parameters, the resonator neck resistance is the most problematic to adjust. It is desirable to able to adjust the sound dissipation without modifying the resonant frequency.

To realize a resistance controller having this property, a spiral spring was installed in a neck of circular cross-section. The gap width between the spring plates could be adjusted by twisting the spiral core shaft. The resistance of the neck aperture was expected to be inversely proportional to the gap width. Prototypes of spiral-spring mounted necks were produced and were tested by conducting experiments using the impedance-tube two-microphone position method.

Consequently, it was confirmed that each of the spiral-spring mounted necks had a variable resistance sufficient for sound dissipation control and had a resistance that was monotone decreasing function of the spring-plate gap width. Varying the gap width had little effect on the effective sectional area and the effective length of the neck and hence did not modify the resonant frequency much.

1 INTRODUCTION

Onsite tuning is mandatory to utilize Helmholtz resonators for air-duct networks. However no tuning system for general use is available. This is a reason why engineers have been avoided to use resonators in HVAC duct systems. The objective of this study is to develop a tuning system for resonators.

Resistance of a resonator neck determines its sound dissipation and resonance sharpness, while its cavity volume, the sectional area and length of the resonator neck determine its

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resonant frequency. Out of these control parameters, the resonator neck resistance is the most problematic to adjust.

Only a limited number of studies on resistance controllable apertures for Helmholtz resonators have been found. Slivinski [1] proposed a resistance controllable neck of variable-distance two-parallel-plates with orifices. However this neck varies the mass reactance significantly as the distance between the parallel plates is varied. It is desirable to able to adjust the sound dissipation without modifying the resonance frequency.

To realize a resistance controller having this property, a spiral spring is installed in a neck of circular cross-section. The gap width between the spring plates can be adjusted by twisting the spiral core shaft. The resistance of the neck aperture is expected to be monotone decreasing function of the gap width. Prototypes of spiral-spring mounted necks were produced and were tested by conducting experiments using the impedance-tube two-microphone position method.

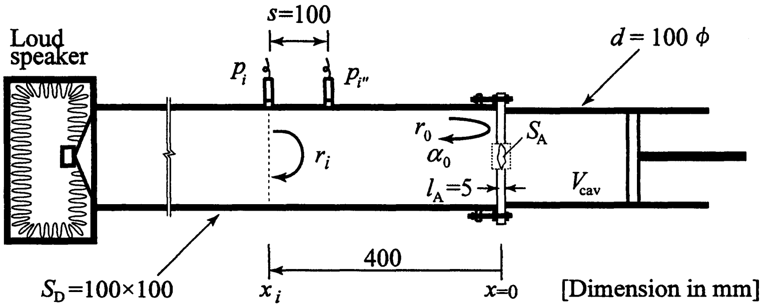


Figure 1: Impedance tube test rig for a resonator at its end.

2 CONTROL PARAMETERS OF A HELMHOLTZ RESONATOR

To control a sound field by a resonator, as shown in Figure 1, is to control the resonant frequency ω_{res} and the quality factor Q of the resonator.

The resonant frequency ω_{res} is governed by the volume of the cavity, V_{cav}, the cross-sectional open area of the neck, S_{A}, and the length of the neck, l_{A}, as

$$\omega_{\mathrm{res}} = c\sqrt{\frac{S_{\mathrm{A}}}{l_{\mathrm{e}}V_{\mathrm{cav}}}} \tag{1}$$

where c is the speed of sound, l_{e} is the effective neck length, which is represented by

$$l_{\mathrm{e}} = l_{\mathrm{A}} + \Delta l_{\mathrm{inn}} + \Delta l_{\mathrm{out}} \tag{2}$$

in which Δl_{inn} and Δl_{out} are the mass end-corrections of the inner side and the outside, respectively, of the neck [2, 3]. The mass end-corrections Δl_{inn} and Δl_{out} are dependent of the sectional areas of the cavity, S_{cav}, and the sectional areas of the duct, S_{D}, respectively.

On the other hand, the quality factor Q, which can be defined as resonant frequency divided by bandwidth between half-power points, is represented as

$$Q = \frac{\omega_{\mathrm{res}}\rho l_{\mathrm{e}} / S_{\mathrm{A}}}{R_{\mathrm{A}}} \tag{3}$$

where ρ is the density of air. The resistance R_A controls not only the quality factor Q but also the sound dissipation. The sound dissipation factor α_0 is a function of $R_A/(\rho c/S_D)$, where S_D is the cross-sectional area of the duct which the resonator is attached to.

As stated above, to tune a resonator is to control R_A and one of the parameters V_{cav}, S_A and l_A.

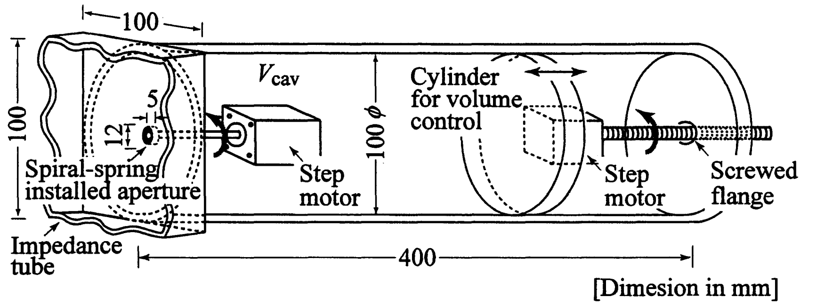


Figure 2: A prototype tuning device for Helmholtz resonators.

3 A SPIRAL-SPRING INSTALLED APERTURE AS A REGISTANCE CONTROLE DEVICE

We employ V_{cav} and R_A as control parameters, and S_A and l_A as fixed parameters. Figure 2 shows a prototype tuning-device for Helmholtz resonators. Both V_{cav} and R_A is controlled by their own step-motors. The step-motors are driven by respective signals supplied from a computer. To control the cavity volume V_{cav}, a cylinder with a step-motor with a screwed shaft was employed. On the other hand, to control the aperture resistance R_A a spiral-spring installed aperture was introduced.

Figure 3 shows a spiral-spring which is installed into the aperture. Winding the spring up and down, the width of the air-gap width between the spring plates can be varied as shown in Photo 1, and the airflow resistance through the gap can be controlled.

To turn the spring, the step-motor shaft was attached to the spiral-spring at its central end. The step-motors are driven by respective signals from a personal computer.

Five different widths ($l_A = 2.3$, 5.0, 7.5, 10 mm) of spiral-spring plates were used. The spring plates are of 0.1 mm thick stainless steel and 30 cm in length.

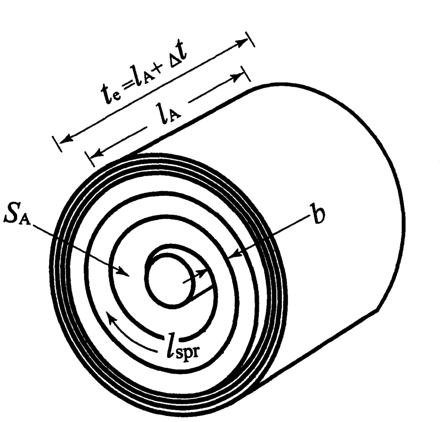
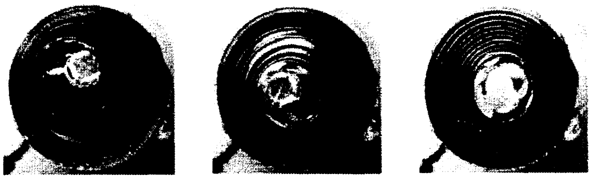


Figure 3: a spiral-spring installed aperture.



$b = 0.9$ mm $b = 0.5$ mm $b = 0.3$ mm

Photo 1: Control of equivalent hydraulic radius b of a spiral-spring installed aperture ($l_A = 5.0$ mm).

4 THEORETICAL FORMULATION FOR A SPIRAL-SPRING REGISTANCE

For theoretical formulation on the resistance R_A of the spiral-spring installed aperture, no literature has been found. We examined to apply the Poiseuille and Kirchhoff models for resistance of circular cylinders to a resistance model of the spiral-spring installed apertures.

4.1 Equivalent Hydraulic radius

To utilize the resistance expression for a circular tube resistance to that for a noncircular one, the concept of equivalent hydraulic radius is employed. The equivalent hydraulic radius is defined as the ratio of two-times sectional-area to perimeter length. In case of the spiral-spring as shown in Figure 3, the equivalent hydraulic radius, b, can be written as

$$b = S_A / l_{spr} \tag{4}$$

where l_{spr} is the length of the air gap between the spiral-spring plates. The radius b, in this case, coincides with the spring-plate gap width (the width of the air gap between the spiral-spring plates).

4.2 Poiseuille Model

The Poiseuille model for tube resistance may be appropriate, when the equivalent hydraulic radius b is nearly equal to or less than the thickness of the viscous boundary layer, d_v, which is represented as

$$d_v = \sqrt{2\mu / \rho\omega} \tag{5}$$

where μ is the viscosity. The resistance of the Poiseuille model, R_A^{Pois}, is given as

$$R_A^{Pois} = \frac{8\mu t_e}{S_A b^2} \tag{6}$$

where t_e is the effective spring-plate width for resistance. This effective width may be written as

$$t_e = l_A + \Delta t \tag{7}$$

in which Δt is a resistance end-correction to account for viscous effects just outside the entrances to the spring plates as shown in Figure 3 [2, 4]. The resistance end-correction, Δt, is to be determined experimentally.

4.3 Kirchhoff Model

The Kirchhoff wide-tube resistance model may be appropriate when the equivalent hydraulic radius of the spring open area, b, is far larger than the viscous boundary layer thickness d_{v}. The resistance of Kirchhoff model, $R_{\mathrm{A}}^{\mathrm{Kirch}}$, is given as

$$R_{\mathrm{A}}^{\mathrm{Kirch}} = \frac{\rho c k d_{\mathrm{v}} \theta t_{\mathrm{e}}}{S_{\mathrm{A}} b} \tag{8}$$

where

$$\theta = 1 + (\gamma - 1)\sqrt{\mathrm{Pr}} \tag{9}$$

in which γ is the specific-heat ratio, $\mathrm{Pr} = \mu c_p / \kappa$ is the Prandtl number, where c_p is the specific heat at constant pressure, and κ is the thermal conductivity.

4.4 A Suitable Combination of the Poiseuille and Kirchhoff Resistance Models

For the equivalent hydraulic radius b is not much larger than and not less than the viscous boundary layer thickness d_{v}, a suitable combination of the Poiseuille and Kirchhoff resistance models leads to a resistance expression R_{A} as [4, 5],

$$R_{\mathrm{A}} = \frac{8 \mu t_{\mathrm{e}}}{S_{\mathrm{A}} b^2} \sqrt{1 + \frac{\rho \omega b^2 \theta^2}{32 \mu}} \tag{10}$$

5 EXPERIMANTAL SETUP

Experiments were conducted for prototype resonator tuners with apertures in which four different spiral-spring plate-width of 2.3, 5.0, 7.5 and 10 mm are installed respectively. The diameter of the aperture and the cylinder are 12 mm and 100mm respectively.

Each resonator tuner was located at one end of a rectangular duct for impedance measurement as shown in Figure 1. The duct is of cross-sectional dimension 10 cm×10 cm and the cross-sectional area $S_{\mathrm{D}} = 0.01\ \mathrm{m}^2$. Employing the impedance tube transfer function method described in the standard ISO10534-2 [6], the reflection factor r_{i}, as a function of frequency, at one of the two microphone locations, x_{i}, was determined varying the spring-plate gap-width b or varying the cavity volume V_{cav}.

Assuming the plane wave propagation in the duct, the reflection factor r_0 at the duct section just in front of the resonator neck, x_0, was readily converted from the reflection factor r_{i}. From the reflection factor r_0, the impedance, the resonant frequency, the resistance and the sound dissipation factors (the acoustic absorption coefficients) $\alpha_0 = 1 - |r_0|^2$ at x_0, as a function of frequency at x_0, were also obtained readily.

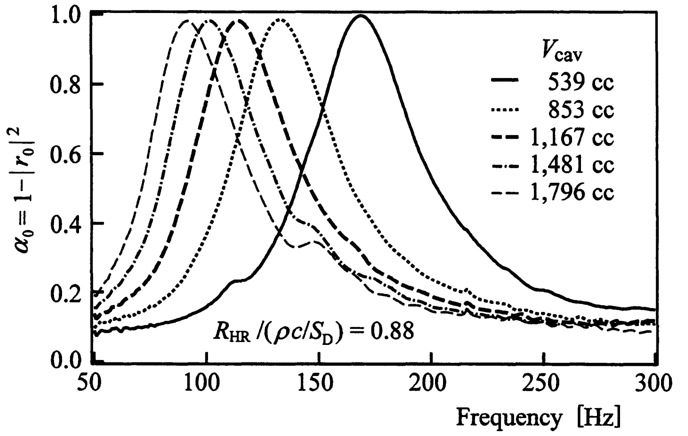


Figure 4: Control of resonance frequency by means of cylinder volume (l_A = 5.0 mm).

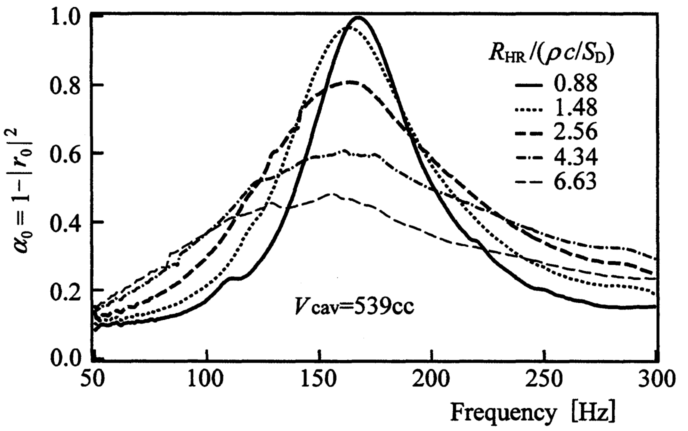


Figure 5: Control of resonance sharpness by means of spiral-spring resistance (l_A = 5.0 mm).

6 EXPRIMENTAL RESULTS

6.1 Resonant Frequency Control by Cylinder Volume

Figure 4 shows the sound dissipation factor as a function of frequency for several cylinder volumes for a spiral-spring plate width l_A = 5.0 mm as a typical.

The sound dissipation factor is defined by the ratio of the sound power dissipated by the resonator tuner to the sound power of incident traveling wave in the duct just in front of the resonator aperture. In this case, the sound dissipation factor coincides with the sound absorption coefficient α_0 for this resonator configuration. The prototype resonator tuner easily shifts its resonant frequency as wide as one octave band by means of the cavity volume control of the step-motor driven cylinder system.

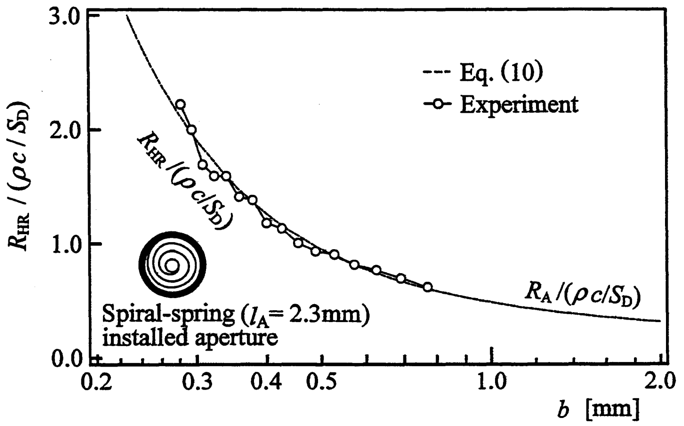
6.2 Quality Factor Control by Spring Plate Gap Width

Figure 5 shows the sound dissipation factor (the sound absorption coefficient) α_0 as a function of frequency for several spring-plate gap-width (the gap width between the spiral-spring plates), b, for a spiral-spring plate width l_A = 5.0 mm as a typical.

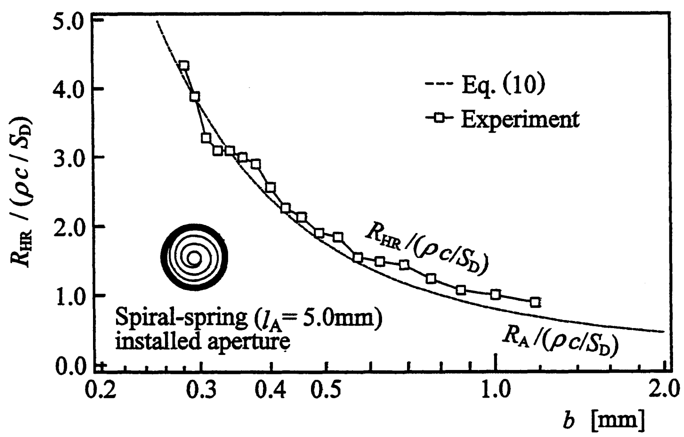
The sharpness of the resonance can be varied as wide as expected for 30 cm in length of the spring plate and for the open area $S_A = 0.005^2\pi$. The tuning range of the sharpness of the resonance (the quality factor Q) can be widen by increasing the spring plate length and open area S_A. The impact of the resistance change to the resonance frequency is marginal. This implies that the resistance tuning does not affect on the open area S_A and the mass end-corrections $\Delta l_{inn} + \Delta l_{out}$, and hence the resistance tuning is possible without affecting to the resonant frequency.

Figure 6 shows the resistance as a function of the spring-plate gap-width b for four spiral-spring plate-widths. It is confirmed that the resistance is a monotone decreasing function of the spring-plate gap-width b. This nature is favorable in trial and error tuning of the sharpness of resonance for a resonator.

Resistances given by expression (10) are also plotted in the Figure 6 for comparison. For the resistance end-correction, assuming $\Delta t = c_1 b + c_2 l_A$ expediently as a formulation, $\Delta t = 1.6b + 0.18 l_A$ was obtained as a best fit to the experimental data. Although farther examinations are required to figure out the resistance end-correction, the expression (10) for the spiral-spring plate resistance as a combination of the Poiseuille and Kirchhoff models accounts for the resistances obtained by the experiment.

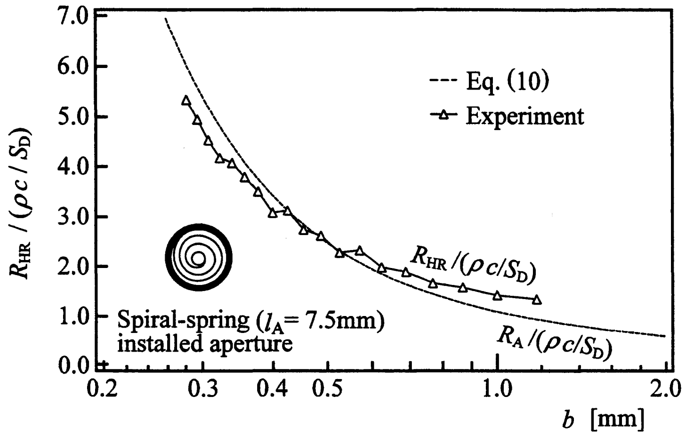


(a) $l_A = 2.3$ mm

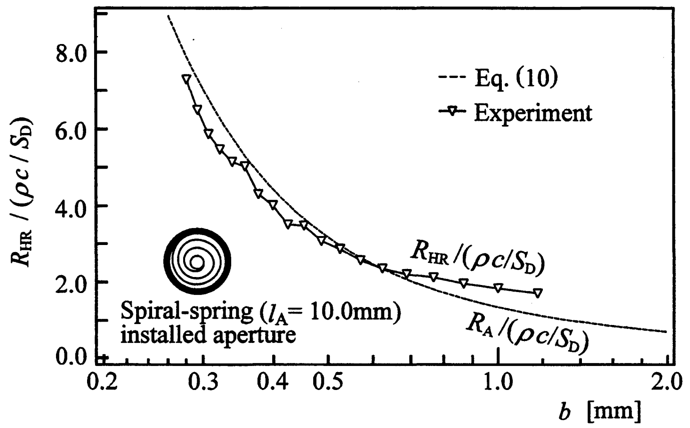


(b) $l_A = 5.0$ mm

Figure 6: Controllability of spiral-spring resistance by means of equivalent hydraulic radius.



(c) l_A = 7.5 mm



(d) l_A = 10 mm

Figure 6 (Continued.)

7 CONCLUSIONS

To realize a resistance tuner for Helmholtz resonators, prototypes of spiral-spring mounted necks of circular cross-section were produced. The gap width between the spring plates could be adjusted by twisting the spiral core shaft.

These prototypes were tested by conducting experiments using the impedance-tube transfer-function method. Consequently, it was confirmed that each of the spiral-spring mounted necks had a variable resistance sufficient for sound dissipation control and had a resistance that was monotone decrease function of the spring-plate gap-width. Varying the gap width had little effect on the effective sectional open-area and the effective length of the neck, and hence did not modify the resonant frequency much.

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