A multi-degree of freedom acoustic lining for noise suppression in broad air ducts

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Abstract. A combination multi-degree of freedom acoustic lining is designed. A given space volume ($30 \times 37 \times 1000 \text{ cm}^3$) is properly distributed in different levels. As a result 28-acoustic resonators of the Helmholtz type and two similar triangular chambers are obtained. These resonators are tuned to different frequencies, so that they cover a frequency range from 65 Hz to about 450 Hz.

The measured third octave band insertion loss showed a wide band attenuation in the frequency range from 80 Hz to about 600 Hz. When the triangular chambers are filled with loose fibre (jute) as absorbent material, the attenuation of the lining extends to higher frequencies, up to about 2000 Hz. An average insertion loss of more than 20 dB is measured in a wide frequency range.

Keywords. Multi-degree of freedom acoustic lining; noise suppression; broad air ducts; acoustic resonators; wide band attenuation; insertion loss.

1. Introduction

To control noise emission from large ducts, designers often choose some variation of parallel baffles filled with fibrous material encased in perforated metal skin (Dyer 1956). With proper filler material and baffle geometry, this type of attenuator can be an excellent sound absorber. In severe environments, absorbed baffles are susceptible to clogging, erosion and settling (Mechel 1975; Lemmerman 1953).

In practice, such as in inducted ventilating and air conditioning systems as well as in turbofan engines, aerodynamic noise is produced, which contains both low frequency components, requiring deep linings, and high frequency components, requiring shallow linings, for attenuation purposes. In this case noise reduction, by lining the side walls of a duct with some porous materials, depends on the thickness and the properties of the lining (Beranek 1942).

Because of the above reasons, the interest in geometric means for sound attenuation has always been high. Therefore, the present work aims to offer some guidelines in choosing the proper geometric distribution of a given space in a great

number of acoustic resonators (Helmholtz type) in order to cover a wide frequency band of attenuation needed to suppress a given noise problem in broad air ducts. We also wish to study to what extent the effective attenuation can be extended to lower frequency bands by means of a combined multi-degree of freedom acoustic lining designed in a given limited volume space.

2. Absorption mechanism and criteria of Helmholtz resonator

Actually the use of Helmholtz resonators is not new. They were used very early to shorten the extremely long, low-frequency reverberation time. Brl (1949) built resonators like those used in the ancient open-air theatres, and concluded that a reverberation time of 0.2 to 0.5 seconds could be achieved.

A Helmholtz resonator, figure 1a, consists of a cavity V and an open neck of actual length t and cross-section area S_H . When sound waves fall on a wall containing a resonator, they force the mass M of air in the orifice of the resonator into vibration and a restoring force is provided by the volume of air in the cavity. Some of the energy which strikes the resonator is reflected, some is radiated as sound waves, and some is dissipated in the resonator in the form of heat.

The basic relations that govern the behaviour of a Helmholtz resonator can be determined from the analogy between acoustical and mechanical systems. The mass M combined with the stiffness ($1/C$) comprise a single-degree of freedom

Figure 1. The principal construction of the designed lining. (a) Helmholtz resonator. (b) Silencer without cover. (c) Cover of the silencer (silencer's face).

oscillator, whose natural frequency f_n is given by

$$f_n = (1/2\pi)(1/MC)^{1/2} = (c_0/2\pi)(S_H/Vt')^{1/2}, \quad (1)$$

where t' is the corrected length of the neck $= t + \Delta t$. The end correction Δt was calculated by several authors. Rayleigh (1945, pp. 177–180) considered a piston radiating into half space and found that the end correction (classical), $\Delta t = \Delta_0 + \Delta_i = 2\Delta_0 = 2(8a/3\pi)$, where Δ_0 and Δ_i are the outside and inside end corrections respectively, and a is the radius of the orifice. Ingard (1953) proposed, that the inside end correction, Δ_i , should model a piston radiating into a tube (cavity) of radius R and gave the following expression,

$$\Delta_i \approx (8a/3\pi) (1 - 1.24 a/R), \text{ for } a/R < 0.4.$$

For $a/R > 0.4$, Δ_i can be extrapolated from a curve, which is given by Ingard (1953). Panton and Miller (1975) carried out experiments and found that the classical end correction on both sides of the orifice would not predict the measured natural frequency f_n . They concluded, that the classical theory was valid only for resonators of lengths longer than $\lambda_n/16$, where λ_n is the wave length at f_n . Accordingly, the end correction Δ_t can be considered to be

$$\begin{aligned} \Delta_t &= \Delta_0(\text{Rayleigh}) + \Delta_i(\text{Ingard}) \\ &= (8a/3\pi) (2 - 1.24 a/R). \end{aligned} \quad (2)$$

To calculate R , one can consider the cavity volume V of the resonator to be equal to that of a cylinder of radius $R = (V/\pi L)^{1/2}$, where L is the depth of the cavity.

A resonator has an internal frictional resistance, which is determined by the shape of the resonator neck and the presence of sound absorptive material in the neck or the cavity. Material located against the cavity wall opposite to the orifice, where the particle velocity is small, is less effective than material placed near or in the resonator neck. A second damping mechanism arises from the acoustic energy radiated by the air slug vibrating in the resonator neck. A maximum absorption (peak absorption) at f_n can be achieved, when the frictional resistance is equal to the radiation resistance (i.e. impedance matching condition). At other conditions, the absorption is reduced. Junger (1975) gave a quality factor Q of the resonance peak,

$$Q = (\pi V)^{1/2} (t'/S_H)^{3/2}. \quad (3)$$

A large Q means a low frictional resistance and this gives a sharp resonant peak suitable for designing resonators to absorb isolated and time-invariant frequencies. For broadband absorption, a sharp peak is undesirable and accordingly resonators with low Q -values must be designed by using a large neck cross-section or small neck length, depending on application.

The cavity shape also affects the silencer performance. Kurze (1965) and Lansing & Zowmski (1973) showed, that the attenuation of the lined duct was improved by cavities with variable depth. The resulting impedance change creates a reactive effect on the sound that complements the dissipative action of the liner. Soderman (1981) examined the attenuation of triangular cavities and found that the attenuation curves of triangular cavities are much broader than would be expected from constant-depth and rectangular resonant-cavities.

3. Lining description

To start the design process of an acoustic attenuator one needs, as a first step, some ideas about the baffle size and spacing. In the present work, a combined multi-degree of freedom acoustic lining is designed. It consists of Helmholtz resonators tuned to attenuate different frequencies. The given volume is divided in such a manner as to allow a great number of resonators with the desired frequency band. These resonators are distributed in the given sizes. The larger volume resonators provide principally reactive attenuation for low frequency sound. If cavity damping is necessary, thin layers of sound absorptive material could be installed in the cavity, depending on the required application, to broaden the attenuation spectrum. The smaller volume resonators are tuned to attenuate the higher frequency sound components. For normal use, when there are no severe environments such as high speed nonuniform flow and high temperature or corrosive gases, chambers filled with fibrous material could be considered in the designed lining.

As an example, a silencer of this type is designed (figures 1 and 2). It consists of a rectangular pipe, of breadth 300 mm, depth 340 mm and length 940 mm, measured on the inside. The outside walls and the bottom of this pipe are made of 30 mm thick wood panels and the two sides (940 mm) are removable for experimental purposes. This pipe is divided into four groups of cavity resonators (each consists of seven resonators) and two separated triangular chambers. All the resonators have trapezoid shapes and the same breadth ($b = 120$ mm), except four resonators (two at each end) which have triangular shapes and breadth $b = 160$ mm. The inside walls of the resonators are made of 10 mm thick wood panel. The whole device represents the cover of a part of a rectangular duct of the same total length 1000 mm and of the same inner width 300 mm. The resonators are constructed and grouped as follows.

Two upper groups: Each group is at one side along the silencer length and they are separated by the triangular chambers. The orifices of these resonators are open to the surface of the silencer and have a common thickness t of 30 mm, which represents the cover thickness of the silencer. The two groups have different depths. One of them, the so-called right group, has a common depth d of 230 mm and the other, the so-called left group, has a common depth of 50 mm.

Two lower groups: They are present under the upper groups and their orifices open on the inside of the triangular chambers. These resonators have a common t of 10 mm, which represents the thickness of the inside common wall of the triangular chambers. These two groups are at different levels and, hence, have different depths. The lower right group has a common depth of 70 mm, whereas the lower left group has a common depth of 250 mm.

The official cross-sectional area of the resonators in the four different groups is chosen so as to be suitable for the desired natural frequency of each one. In certain applications, the orifice thickness t can be varied by using small tubes of suitable lengths and suitable cross-sections to cover a desired frequency band of attenuation.

Figure 2. The designed lining: **a.** vertical plane (the cover and the two long sides are removed); **b.** side view at the plane 'D-D'.

Two triangular chambers: Two triangular chambers are on the inside. They are of 940 mm length, 140 mm maximum breadth and 340 mm depth. They are separated from each other by a wooden wall of 10 mm thickness. They are also covered with a common perforated thin plate with a perforation ratio of more than 35%.

Table 1 gives the dimensions, the predicted natural frequency, f_n (Hz), and the quality factor, Q , of the resonators. As can be seen from the table, the 28 resonators cover a wide frequency range from 65 Hz to about 430 Hz. It also can be seen that the Q -factor is high, specially for resonators of low f_n -values. This means, that these resonators have sharp resonance peaks, which sometimes are undesirable if broadband attenuation is needed. In this case, thin layers of sound absorptive material may be used in the resonator cavity near its orifice. The predicted f_n -values were calculated using (2) for the end correction Δt . It was noticed that these values are less than those calculated using the classical end

Table 1. Dimensions, predicted natural frequencies and quality factor of the resonators.*Upper group resonators:*Common orifice thickness $t = 30$ mm; radius of the orifice = a ; cavity volume = V .

Right side group

Common depth $d = 230$ mm.

Left side group

Common depth $d = 50$ mm.

a (cm)	a/R	V (cm ³)	f_n (Hz)	Q	a (cm)	a/R	V (cm ³)	f_n (Hz)	Q
1.0	0.142	3588	71	328	1.0*	0.252	100	431	31.4
1.25	0.193	3036	93	171	1.0	0.284	195	309	72.7
1.5	0.256	2484	120	97	1.5	0.343	300	351	32.4
1.75	0.338	1932	156	57	1.5	0.290	420	294	39.3
1.25	0.286	1380	140	111	1.25	0.213	540	222	71.5
1.25	0.355	897	175	87	1.75	0.270	660	264	46.0
1.0*	0.396	460	204	107	1.75	0.248	780	242	50.3

*Lower group resonators:*Common orifice thickness $t = 10$ mm; radius of the orifice = a ; cavity volume = V .

Right side group

Common depth $d = 70$ mm.

Left side group

Common depth $d = 250$ mm.

a (cm)	a/R	V (cm ³)	f_n (Hz)	Q	a (cm)	a/R	V (cm ³)	f_n (Hz)	Q
1.0	0.142	1092	172	104	0.75*	0.297	500	213	87
1.25	0.193	924	220	43	1.0	0.284	975	187	65.5
1.5	0.256	756	279	26	0.75	0.172	1500	120	162
1.5	0.290	588	319	22	1.25	0.242	2100	148	62
1.25	0.286	420	334	27	0.5	0.085	2700	65	568
0.5	0.142	273	206	176	1.0	0.154	3300	99	131
0.5*	0.198	140	290	123	1.75	0.248	3900	136	43.3

* Triangular resonators.

correction by an amount of about 5 Hz for large volume resonators and of about 10 Hz for resonators with small volumes.

4. Test arrangement and measurements

The arrangement used for testing the designed silencer consists of three rectangular tubes with the same internal cross-section of 20×30 cm² (figure 3). The three tubes are the source tube (length, 1200 mm), the middle tube (length, 1000 mm), and the receiver tube (length, 1400 mm). They are made of 30 mm thick wood panels to increase the sound insulation of the three tubes against background noise. A loudspeaker in a separate cabinet is used as a sound source and is connected to the source tube through a thin rubber layer to prevent mechanical coupling between the source and source tube. The loudspeaker is fed with an amplified pure tone or white noise. The receiver tube ends with an absorbent wedge to provide an anechoic termination in a frequency range from 80 Hz to about 600 Hz. The sound insulation of the receiver tube can be raised by placing it inside a larger one and the

Figure 3. Block diagram of the test arrangement.

space in-between filled with 10 cm thick spongy rubber. This arrangement helps to minimize the background noise level in the receiver tube, thus allowing a true measurement of transmission loss to be measured even when the silencer has a high transmission loss. The middle tube is connected to both the source and receiver tubes through a 2 mm thick soft rubber layer in order to prevent the propagation of mechanical vibration from the source tube to the receiver one. The sound leakage is minimized by means of some elastic kind of clay. The cover of the middle tube, which is one of its large cross-dimensions, is removable and can be replaced by the designed lining for testing it.

Two similar and calibrated 1/2"-condenser microphones are used to measure the sound pressure levels in both the source and receiver tubes. The two microphones are connected via a two-channel microphone selector to an audio frequency spectrometer or frequency analyzer and level recorder. All these measuring equipment are Brüel and Kjær measuring systems.

At frequencies below 570 Hz, the sound wavelength is larger than twice the largest cross-dimension of the tube (300 mm) and the plane wave is the only one that can propagate. Therefore the sound pressure can be considered to be constant across the tube cross-section. In this case, the sound transmission loss ΔL_t due to the lining can be directly determined at the measuring frequency f (Hz) by the following equation,

$$\Delta L_t(f) = L_1(f) - L_2(f), \text{ (dB)} \quad (4)$$

where L_1 and L_2 are the sound pressure levels in the source and receiver tubes, respectively.

At frequencies above 570 Hz, the sound pressure fluctuates across the tube cross-section due to the propagation of high modes. In this case, a measuring technique was successfully used by the author (Shenoda & Fadl 1983). The microphones are moved slowly in the diagonal directions of the cross-sections of both the source and receiver tubes at every measuring frequency to record the variations in sound levels. The recorded chart can be used to calculate the mean

sound level $L_m(f)$ from the equation:

$$L_m(f) = 10 \log (1/n \cdot \sum_{v=1}^n 10^{L_v/10}), \quad (\text{dB}) \quad (5)$$

where n is the number of the different sound levels L_v included in the recorded chart. Accordingly, the sound transmission loss due to the lining is given by

$$\Delta L_t(f) = L_{m1}(f) - L_{m2}(f), \quad (\text{dB}) \quad (6)$$

where L_{m1} and L_{m2} are the mean sound levels in both the source and receiver tubes, respectively.

Also, the effectiveness of the lining can be measured by measuring its insertion loss ΔL_I . This can be done by measuring the sound pressure level L_2 (or L_{m2}) in the receiving tube once without the lining and then with it. Accordingly, the insertion loss due to the lining is given by

$$\Delta L_I = L_2(\text{without}) - L_2(\text{with}). \quad (7)$$

5. Results and discussion

To test the attenuation effectiveness of the designed acoustic lining, the cover of the middle tube was removed and replaced by the lining. The measuring microphones were put exactly at the central point of the tube cross-section. The sound pressure level as a function of frequency was automatically recorded, once when the triangular chambers were empty i.e. without the lining of absorbent material, $L_1(f)$ (figure 4a), and also when they were lined, $L_2(f)$ (figure 4b). The absorbent material, which is loose jute fibre of density 0.03 g/cm^3 , was homogeneously distributed in the chambers.

Figure 4a shows, that the level difference $L_1(f) - L_2(f)$ gives a picture about the attenuation of the acoustic lining. It has a wide band attenuation characteristic, specially in the very important frequency range from 80 Hz to about 600 Hz. Because the resonance frequencies of the different resonators are close to each other, a very small number of resonant peaks can be seen in the recorded level $L_2(f)$. Also a maximum transmission loss can be noticed at 250 Hz in agreement with the $\lambda/4$ -resonance of the triangular chamber, i.e., when $\lambda/4$ is = to its depth d . Owing to the propagation of higher modes, it is very difficult to measure the transmission loss due to the lining at frequencies above 600 Hz.

Figure 4b shows the recorded levels when the triangular chambers were filled with the absorbent material. A very slight change in the value of the transmission loss can be identified. The small number of the peaks in $L_2(f)$ have disappeared and the $\lambda/4$ -peak at 250 Hz is slightly reduced.

Figure 5 shows the third octave band sound insertion loss ΔL_I due to the acoustic lining when the triangular chambers are empty and also when they are filled with the absorbent material. There is a slight difference between the two insertion loss curves in the frequency range up to about 600 Hz. Above this frequency, some peaks disappear and the effectiveness of the acoustic lining is extended to higher frequencies. As a result, an insertion loss of more than 20 dB was measured over a wide frequency range.

Figure 4. The sound level as a function of frequency, in the source tube $L_1(f)$ and in the receiver tube $L_2(f)$. (a) Silencer without absorbent material. (b) Silencer with absorbent material.

Figure 5. Sound insertion loss ΔL_I (dB) due to the designed lining.

6. Conclusion

From the present work, it can be concluded that a wide band effective attenuation for broad ducts can be realized if a given volume space is properly distributed in accordance with its geometry. Also, proper choice of the parameters affecting the performance of a resonator lead to a significant low frequency attenuation.

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