DOI: 10.61782/fa.2025.0771

FORUM ACUSTICUM EURONOISE 2025

A SILENCER DESIGN FOR A MECHANICAL HEAT RECOVERY UNIT

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ABSTRACT

Modern homes require energy efficient airtight designs using controlled ventilation systems to retain the heat energy. Mechanical Ventilation Heat Recovery (MVHR) units help to maintain air circulation between the building and the outdoors, while not compromising energy efficiency. A significant challenge to implementation and continuous use of these devices is the noise emission. Noise from MVHRs can cause indoor noise levels to exceed the recommended levels, with respect to the Noise Rating (NR) guidelines. The noise has a frequency content below 500 Hz, which many conventional sound absorbers struggle to attenuate without large space requirements. This research develops a compact silencer suitable for a domestic MVHR unit with a large duct diameter. The design consists of stacked Helmholtz resonators with embedded necks and relatively large cavity volumes. The proposed design achieved >8dB transmission loss for frequencies 250 Hz, 300 Hz and 350 Hz. These were compared with data from an MVHR and the NR guideline and was found to reduce the MVHR SPL to below NR35 for the octave bands 250 Hz and 500 Hz.

Keywords: Transmission Loss, MVHR, Helmholtz Resonators, 3D Printing

1. INTRODUCTION

Mechanical Ventilation and Heat Recovery systems (MVHR) are devices that ventilate rooms and buildings, while recovering heat energy from outgoing air. In modern homes that are highly insulated and airtight, this technology is essential to maintain good indoor air quality [1].

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The main challenge to further adoption and continuous use of MVHR are the noise emissions from the fans. It has been reported that a significant portion of MVHR users either rarely or never use their systems and cite noise as the cause [2, 3]. The noise emissions typically exist below 500 Hz [2, 4]. Some manufacturers provide noise emission data on their systems, for example the unit considered in this work is the Air Cycle 1.3+, which has a reported sound power level of 45.4 dB and 46.8 dB at 250 Hz and 500 Hz respectively [5]. Data were taken at the maximum possible flow rate for the unit. With regards to guidelines, the Noise Rating standard for dwellings should be within NR35. Other guidelines suggest a lower SPL for dwellings [6]. In residential buildings, research indicates that due to relatively lower background noise, noise from ventilation systems is more noticeable [2, 3].

There are many ways to address the noise emissions of MVHRs, either at the source (fan blades, DC motor) or applying an acoustic treatment downstream from the source. Suitable designs from literature used to achieve low frequency attenuation in ventilated systems include space-coiled channels [7], singular or double Helmholtz resonators [8, 9] and adaptive resonators [10]. MVHRs available on the market typically have duct diameters >5 inches (127 mm) [5], which is much larger than the experiment setups seen in the cited works. Further more, while research has been done to study the effects of flow on transmission loss [11], no work has been found on application in real MVHR systems.

In this paper, an acoustic treatment for a large circular duct is proposed. The design utilizes Helmholtz resonators with embedded necks, and can be manufactured with a desktop 3D printer at a low cost. The objective is to achieve high attenuate at specific frequencies with minimal resources. Hence, a basic analytical approach is taken for the design rather than a detailed FEM combined with an optimisation scheme.

2. THEORY

The goal of the design is to attenuate the tonal noise of the fan at specified frequencies, which are characteristic of the blade pass frequency. The resonance frequency of a Helmholtz resonator is expressed as:

$$f_0 = \frac{c_0}{2\pi}\sqrt{\frac{S_n}{V(l_n+\delta)}} \tag{1}$$

Where c_0 is the speed of sound in air, S_n is the area of the neck cross section, V is the volume of the cavity, l_n is the neck length and δ is the end correction length. The many different formulations of the δ exist in the literature, mainly for resonators with cylindrical cavities. For a cylindrical neck and cuboid cavity, Catapane *et al* derived an expression for the end correction length [12]. Given in Eq 2 , where $\xi = 2r_n/d_c$, d_c is the cavity width.

$$\delta = (-5.98\xi^3 + 14.8\xi^2 - 9.75\xi + 3.60)l_n\xi \tag{2}$$

Solving Eq 1 for a resonance at 300 Hz and a cavity volume of 125000 mm^3 yields different combinations of r_n and l_n (neck radius and length). The chosen values are list in Table 1. Although there are more sophisticated methods to maximize the magnitude and bandwidth of the transmission loss, the low cost of this design approach may be more relevant for industrial settings. The drawback to this analytical approach is that while target frequencies can be easily attained there, the expected transmission loss is not estimated. A parametric investigation of 3D printed structures allows for empirical refinement of the design at low cost.

3. EXPERIMENTAL SETUP

The transmission loss experiment setup is shown in Fig 1 in accordance with ASTM E2611-09, using the two-load method. The loudspeaker has a frequency range of 10 Hz - 1400 Hz, and is enclosed in a foam filled box. Four GRAS 40PL-10 microphones were used, they have a frequency range of 50 Hz - 5000 Hz ($\pm$1.5dB). The microphones were connected to a National Instruments USB-4431 analogue I/O device.

The setup is designed for large samples with an inner radius of 63.5 mm, with mounting screws 106 mm from the centre. This leaves 42.5 mm of width for the acoustic treatment. For every sample and configuration, the tests were performed 3 times to test the repeatability of the results. The frequency resolution of the calculated spectra was 2.4 Hz, with a sampling rate of 40 kHz. The measurement duration was 30 seconds, then the termination is changed, and another 30 second measurement was taken. Each of the resonator rings described in Section 4 were tested alone before being stacked together for broadband attenuation.

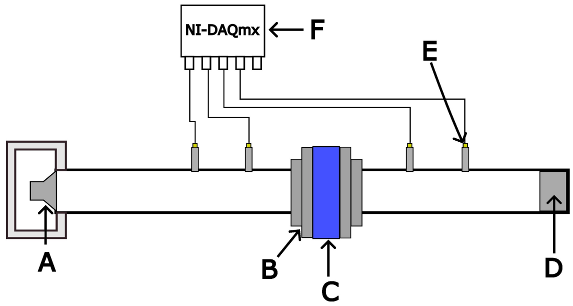

Figure 1. Experiment Setup. (A) Enclosed Loudspeaker, (B) Gaskets, (C) Sample, (D) Anechoic Termination, (E) Microphones, (F) Data Acquisition System

4. DESIGN AND MANUFACTURING

The goal of the resonator is to attenuate transmission noise while being compact and sub wavelength. Since the operating frequencies are between 250 Hz - 350 Hz, the subwavelength constraint is easily satisfied. For compactness, the overall thickness was arbitrarily constrained to 40 mm which was considered representative of a real system. Eq 1 was solved for the specified frequencies, returning a set of radius and length values, given in Table 1. The length L is determined by the volume of the embedded neck in Eq 3, where W is the width of the resonator, S is the arc length and V_n is the volume of the neck, with the wall included. $R_1 = 63.5$mm and $R_2 = 100$mm

$$L = \frac{V_c + V_n}{WS} \tag{3}$$

The Helmholtz Resonator rings were manufactured with a Prusa i3 MK3 printer, using PLA filament and a 0.4 mm nozzle. The printer settings were as follows: 0.2 mm layer height, 205°C and 65°C nozzle and bed temperature respectively. The lower nozzle temperature helped to eliminate stringing while maintaining layer adhesion.

The resonators are asymmetric, this deviation from the model, where the neck and cavity are assumed to be

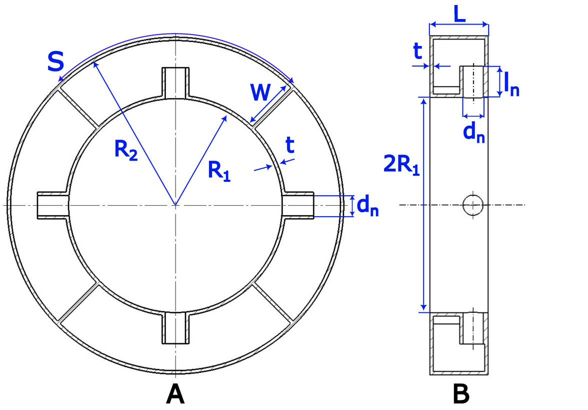

Figure 2. Cross section view of the 300 Hz ring, with dimensions provided in Table 1.

Frequency (Hz)	r_n	l_n	W	L	t
250	5.5	25.5	32.5	36.5	2
300	6	18.8	32.5	35	2
350	5.5	14	32.5	31.5	2

Table 1. Dimensions of resonator rings, dimensions are all in millimeters

concentric, was made to improve the manufacturability. In this configuration, there is no overhang underneath the neck (see Fig 2B), hence no internal supports are required. From the literature, the effect of the asymmetry on the transmission loss should be negligible [11]. The print time for each ring was 14h 30m, using 194 g of PLA, with an estimated cost of €3.8 per unit (estimated cost of PLA is €20/kg).

5. RESULTS AND DISCUSSION

Figure 3 shows the transmission loss of individual resonator rings, and the broadband attenuation when they are all stacked together in the transmission loss tube. The designs achieve transmission loss values at the target frequencies between 8dB and 11 dB. The transmission loss of the stacked configuration provides an increased transmission loss at 358 Hz, from 11 dB to 13 dB. The largest deviation between the calculated and measured resonance frequency is 4%, which is acceptable.

Applying this design to a real MVHR system will be the subject of future work. However the expected per-

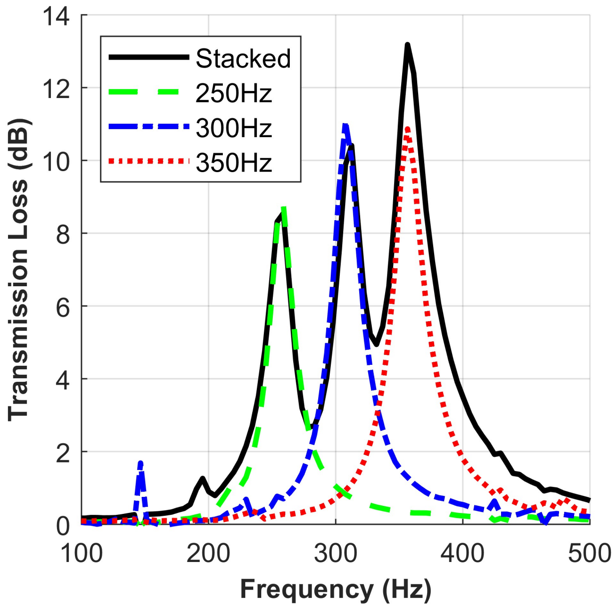

Figure 3. Broadband and tonal transmission loss with 3 resonator rings stacked together. Dashed lines represent individual performance

formance can be evaluated with a known MVHR and a comparison with the Noise Rating Curves. Provided that the tonal noise emission of the fan aligns with the chosen frequency values this design provides approximately 9 dB attenuation in each of the 250 Hz and 500 Hz octave bands. Figure 4 shows the curve for NR25 and NR35. The data available for the Air Cycle 1.3+ is used to calculate the total sound pressure level, direct plus reverberant, in a domestic bedroom. Reverberation time measurements were made in a domestic bedroom to accurately estimate the sound field in the room for a MVHR unit installed in one wall far from a corner or ceiling. It shows that without acoustic treatment, the SPL in the room would be unacceptable for a private dwelling. The noise reduction from the resonator ring is applied, using the experimentally determined values of 9 dB at 250 Hz and 500 Hz. The SPL reduces from 42 dB to 33 dB at 250 Hz. For 500 Hz, the SPL reduces from 44 dB to 35 dB. This means that the MVHR unit will not exceed the NR35 curve for the bedroom which may be deemed acceptable. The figure also shows that for further improvements additional attenuation will be required for the 1 kHz octave band. It is suggested that acoustic foam could be applied in combination with the designed resonator to provide this attenuation.

6. CONCLUSION

In this work, a 3D printed silencer for a circular duct MVHR was designed and tested. The design approach used basic analytical expressions for a Helmholtz resonator, to define the geometries. Three rings with four Helmholtz resonators with embedded necks were manufactured and tested for transmission loss in a 127 mm tube. The individual rings successfully achieved tonal transmission loss ranging from 8 to 13 dB. The maximum error between the target and measured resonance frequency was 4%, which is acceptable for this application. Applying the noise reduction to a MVHR unit in a domestic bedroom room would reduce the SPL below the NR35 curve, which is deemed acceptable. Future work will include applying the design to a real MVHR, by attenuating the measured noise from the system.

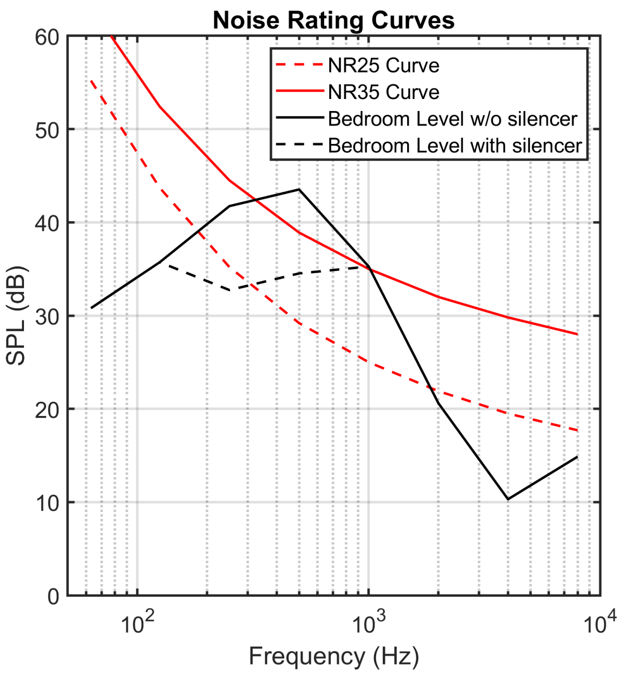

Figure 4. Noise Rating curves for bedroom with a AirCycle 1.3+ present

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