A silencer design and analysis of the effect of silencer perforation towards resonant frequency and insertion loss in a duct

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Abstract. A silencer model was designed for application to HVAC systems, specifically for noise control in ducts. The goal was to see how the perforation of the silencer affected the resonant frequency and Insertion Loss (IL), as well as analyse physical phenomena that occurred in system. The materials used were 1/2-inch PVC pipes 5 cm in length and acrylic with a thickness of 2 mm used as the silencer frame. The varied parameters were the amount of PVC pipes (percentage of silencer perforation) from 0 holes (no perforation) to 10 holes. It is concluded that perforation affects the resonant frequency through changes in acoustic mass and affects the IL through acoustic resistance. It is inconclusive whether the resonant frequencies generated affected IL. It is assumed that negative valued IL observed at lower frequencies occurred due to resonance between the sound and duct (structure-borne sound) which increased sound pressure level, or that these may be the actual resonant frequencies generated. Further research is needed to study the generation of resonant frequencies and structure-borne sound in the system, study airflow and thermal performance of a HVAC system when the silencer is applied, the transmission loss (TL) of the silencer and optimisation of the design to improve low-frequency sound attenuation.

1. Introduction
Noise is one of the main problems handled in acoustics, especially the field of noise control, as it can cause discomfort as well health problems to listeners. Noise control is carried out through research as well as the implementation of regulations to regulate the level of noise allowed in the environment [1]. Heating, Ventilation, and Air Conditioning (HVAC) is an important topic in noise control as it relates to noise in buildings and rooms. The sound generated by a HVAC system not only travels through air (air-borne noise) but also system components and structures (structure-borne noise). Some examples of noise sources in a HVAC system are mechanical noise from fans, pumps, VAV boxes, air turbulence, and outlets such as diffusers [2][3]. Previous research on noise control in ducts was able to significantly reduce noise [4][5], however, we experimented with a silencer design, which was inspired by previous research titled A Novel Noise Reduction Technique for an Air-Cooled Case [6].

2. Sound in HVAC Systems
In a HVAC system, each component produces sound in a certain frequency range. Figure 1 shows the frequency range of major components in a HVAC system. Low frequencies are dominated by structure-borne vibration and air turbulence, while high frequencies are dominated by moving parts and outlets (diffusers) [2].
Fan instability, Air, turbulence rumble and structureborne vibration

Fan and Pump Noise

VAV Unit Noise

Reciprocating, Centrifugal, and Screw Chillers

Diffuser Noise

Fan instability, Air, turbulence rumble and structureborne vibration

Fan and Pump Noise

VAV Unit Noise

Reciprocating, Centrifugal, and Screw Chillers

Diffuser Noise

Octave MidBand Frequency, Hz

8 16 31.5 63 125 250 500 1000 2000 4000 8000

Figure 1. Frequency of HVAC Components [2]

Change in the cross-sectional area of a duct affects the sound waves that pass through. One contributing factor is the restriction of airflow. Figure 2 shows the change in cross-sectional area in a silencer system.

Figure 2. Change cross-section of the duct [7]

Base on figure 2, the ratio of change in cross-section affects IL. $S_1$ and $S_2$ refer to the cross-sectional diameter before and after the change. IL is lowest when the ratio is 1, meaning there is no change in a cross-sectional area [7].

Figure 3. Effect of Change in Cross-sectional Area towards Insertion Loss [7]
3. Measurement Parameters

3.1. Perforation Percentage (Porosity)

The perforation percentage was the ratio between the surface area of the object, in this case, the acrylic frame, and the surface area of the perforation, in this case, the PVC pipes. The equation for calculating perforation percentage is given as

\[
\% \text{ of Perforation} = \frac{\text{Total surface area}}{\text{Perforation surface area}} \times 100\%
\]  

(1)

3.2. Acoustic Components

Figure 4 shows the acoustic system of the study and the equivalent electro-acoustic circuit. These components explain and affect the acoustic phenomena in the system.

![Acoustic system diagram]

Figure 4. (a) Acoustic system of experiment. (b) cross-section perforation. (c) Equivalent electro-acoustic circuit

3.2.1. Acoustic Mass \( (M_A) \)

Difference in the pressure of a medium containing acoustic waves enables sound to travel. Acoustic mass \( (\text{in kg/m}^4) \) is the mass of the medium (air) which, is accelerated due to pressure change. The electro-acoustic equivalent of this is an inductor. The equation for acoustic mass is

\[
M_A = \frac{\rho_0 L_E}{\pi a^2}
\]

(2)

where \( \rho_0 \) is the medium's density, \( L_E \) is the equivalent length of the pipe, and \( a \) is the radius of the pipe. For identical acoustical masses arranged parallel (in this case 2 identical parallel pipes), the acoustic masses become a parallel circuit such as in Figure 3b. The equivalent length of the pipe considers a correction where additional acoustic mass is displaced at the ends of the pipe. There are two cases, a flanged end, and a free end. Each correction is calculated through the following equations:

\[
\Delta L_1 = \frac{8a}{3\pi}
\]

(3)

\[
\Delta L_2 = 0.613a
\]

(4)

The total equivalent length of the pipe, considering the end corrections of the pipe is

\[
L_E = L + \Delta L_1 + \Delta L_2
\]

(5)

Acoustic Compliance \( (C_A) \). If the acoustic medium flows into a fixed volume at a constant rate, it will undergo expansion and contraction. This occurs because the medium fills and empties the volume.
at a constant rate, much like the charge-discharge phenomena of a capacitor. This phenomenon is known as acoustic compliance (in m$^5$/N). The equation used is

$$C_A = \frac{V}{\rho_0 c^2} \quad (6)$$

where $V$ is the volume of the medium and $c$ is the speed of sound for an ideal gas [8].

3.2.2. Acoustic Resistance ($R_A$).

Acoustic resistance (in Pa·s/m$^3$ or Ns/m$^5$) occurs due to resistance or damping of the acoustic medium. The electro-acoustic equivalent of this component is the resistor. However, unlike a resistor, acoustic resistance is a function of frequency as well. The acoustic resistance of a pipe with dimensions smaller than the wavelength of sound is given as

$$R_A = \frac{8\eta l}{\pi a^4} \quad (7)$$

where $\eta$ is the dynamic viscosity of the fluid and $l$ is the length of the pipe. Dynamic viscosity is a characteristic of a flowing fluid. This characteristic creates resistance in the flow due to friction between the fluid and the pipe wall. The equation above resembles Poiseuille’s law of fluid flow. This equation can be rewritten as

$$\Delta p = Z_A U \quad (8)$$

which describes the relationship between acoustic impedance ($Z_A$), change in pressure ($\Delta p$), and the volumetric flow rate ($U$) [8].

4. Experimental Methodology

The experiment was carried out to measure sound pressure level (SPL) with and without the insertion of the silencer as well as the measurement of the dimensions of the impedance tube. Insertion Loss and resonant frequency were calculated from the sound pressure level measurements, and the acoustic components stated above ($M_A$, $C_A$, and $R_A$) are calculated from the impedance tube dimensions. Insertion Loss is calculated through the equation

$$IL = 10\log \left( \frac{Lw_{silencer}}{Lw_{silencer}} \right) = SPL_{silencer} - SPL_{silencer} \quad (9)$$

where SPL is measured in decibels (dB) [1]. The resonant frequency is a function of the acoustic mass and compliance generated by the acoustic system. It is calculated through the following equation:

$$f_0 = \frac{1}{2\pi (M_A C_A)^{1/2}} \quad (10)$$

The harmonic frequencies are calculated as multiples of the fundamental resonant frequency [8].

The main device used for measurement is an impedance tube, analogous to a duct, manufactured by a former student at the Laboratory of Acoustics and Building Physics as part of his bachelor’s thesis. Figure 4 describes the test facility used in the experiment. The measurement devices include impedance tube with a built-in speaker (length 1 m and diameter 10 cm), microphones DBX RTA-M, TASCAM US-800 audio interface, microphone calibrator Brue&Kjaer type 4231, Amplifier, and laptop with Yoshimasa Electronic DSSF3.
The experiment analysed the sound pressure level (dB) using Yoshimasa Electronics Realtime Analyzer with third-octave band analysis for frequencies between 20 to 20000 Hz. The sound pressure level measurements were performed on ten silencers, each varying in perforation from 0 holes (no perforation) up to 10 holes as shown in figure 6. The acoustic frame is 10 cm in diameter and the perforations are 2.25 cm in diameter. The perforations are spaced 2.4 cm apart, measure from the center of each hole. The silencers were designed so the perforations would be symmetrical and as close to the center of the frame as possible.

![Silencer Perforation Design](image)

The sound pressure level of the speaker was calibrated at 100 dB at 1000 Hz. Calibration was done using the microphone at the end of the pipe, located 1.2 m away from the speaker. Sound pressure level for each frequency was measured for 10 seconds and was recorded continuously from 20 Hz to 20000 Hz.

5. Results and Discussions

Table 1, 2, and 3 represents a data measurement and calculations obtained for perforation percentage, acoustic resistance, and resonant frequencies of each silencer variation. Acoustic compliance and mass are constant for each variation because they are only affected by the dimensions of the impedance tube.
Table 1. Perforation percentage of each silencer variation

| Variation (holes) | Perforation Percentage (%) |
|-------------------|-----------------------------|
| 1                 | 3.8                         |
| 2                 | 7.7                         |
| 3                 | 11.5                        |
| 4                 | 15.3                        |
| 5                 | 19.1                        |
| 6                 | 23.0                        |
| 7                 | 26.8                        |
| 8                 | 30.6                        |
| 9                 | 34.5                        |
| 10                | 38.3                        |

Table 2. Acoustic resistance of each silencer variation

| Variation (holes) | Acoustic Resistance (N.s/m²) |
|-------------------|-----------------------------|
| 1                 | 353.2                       |
| 2                 | 176.6                       |
| 3                 | 117.7                       |
| 4                 | 88.3                        |
| 5                 | 70.6                        |
| 6                 | 58.9                        |
| 7                 | 50.5                        |
| 8                 | 44.1                        |
| 9                 | 39.2                        |
| 10                | 35.3                        |

Table 3. Resonant frequencies of each silencer variation

| Variation | Resonant frequency (Hz) | Second Harmonic (Hz) | Third Harmonic (Hz) | Fourth Harmonic (Hz) |
|-----------|-------------------------|----------------------|---------------------|----------------------|
| 1         | 39.8                    | 79.6                 | 119.4               | 159.2                |
| 2         | 56.3                    | 112.6                | 168.9               | 225.2                |
| 3         | 68.9                    | 137.9                | 206.8               | 275.8                |
| 4         | 79.6                    | 159.2                | 238.8               | 318.4                |
| 5         | 89.0                    | 178.0                | 267.0               | 356.0                |
| 6         | 97.5                    | 195.0                | 292.5               | 390.0                |
| 7         | 105.3                   | 210.6                | 315.9               | 421.3                |
| 8         | 112.6                   | 225.2                | 337.8               | 450.3                |
| 9         | 119.4                   | 238.8                | 358.2               | 477.7                |
| 10        | 125.9                   | 251.7                | 377.6               | 503.5                |
Figure 7. Insertion Loss spectrum of each silencer variation

The silencer mounted in the duct creates acoustic mass and acoustic resistance. Acoustic resistance was generated by the perforations of the silencer, and the acoustic mass results from the thickness of the perforation (length of PVC pipes). Acoustic resistance generates resistance due to the restriction of sound flow through the pipe. Acoustic mass generates resistance in the form of inertia, due to the displacement of the mass of air in the PVC pipes. In addition, acoustic compliance is produced due to the compression and expansion of air in the volume between the speaker and silencer. This phenomenon causes the air to oscillate like a spring, and resistance is generated by the elasticity of the air.

As a result of the acoustic mass and compliance, resonance occurs in the pipe at certain frequencies. For each variation of silencer, there were an increase in the fundamental resonant frequency and subsequent harmonic frequencies due to a decrease in acoustic mass as perforation was increased. Theoretically, at the resonant frequencies, the sound should be amplified. At several frequencies (50, 125, 200, 250, 315, 1000, 8000, and 12500 Hz), the IL showed negative values, indicating resonance occurred. The silencer without perforation only showed negative IL at 400 Hz at -2.02 dB, and other frequencies tended to produce high IL. Based on comparison between the overall IL spectrum in figure 7 and the resonant frequency calculations, we find it inconclusive that resonant frequency has effect on the IL produced, due to mixed results at those frequencies. Furthermore, as the resonant frequency changes depending on silencer variation, each variation should show a different IL spectrum, however this was not the case in this experiment. There are also several assumptions regarding the negative IL produced at low frequencies. First, it was assumed that resonance between the sound and pipe structure produces structure-borne vibration generating additional sound in the pipe. The second assumption was that these are the actual resonant frequencies generated due to the insertion of the silencer, implying that there are other factors that affect resonance in the system besides the theory above. To prove these assumptions, further research is needed.
Figure 8. Average Insertion Loss of each silencer variation

Average IL produced by each silencer variation is shown in figure 8. It was shown that IL decreases as the perforation percentage increases. The 1-hole variation produces the largest IL at 19.9 dB and the 10-hole variation produces the smallest at 5.8 dB. The 6-hole and 9-hole variations are outliers in this graph, showing an increase in IL rather than a decrease. This may be due to error in the measurement of these variations.

The acoustic components of the system produce impedance in the form of acoustic resistance. Table 2 above shows acoustic resistance decreasing as perforation increases. It’s due to the design of the silencer where the PVC pipes are parallel to each other much like parallel resistors. Therefore, perforation and acoustic resistance are inversely proportional. Comparing acoustic resistance to average IL, it was shown that acoustic resistance and IL are directly proportional.

Silencer perforation restricts sound flow through the silencer. In agreement with the theories stated above, the silencers with less perforation percentage generate higher IL than the silencers with higher perforation percentage. This is due to the change in cross-sectional area being more drastic for lower than higher perforation percentages [7].

6. Conclusions
Silencer perforation is inversely related to IL due to change in acoustic resistance. Furthermore, the silencer generates acoustic mass and resistance, which contributes to the generation of resonant frequency in the system. It is inconclusive however that the resonant frequency generated affects the insertion loss of the silencer. Further experiments are needed to study the resonant frequency further. Because this research is a preliminary study in noise control for HVAC systems, there are several suggestions for future experiments: testing the thermal and airflow performance of the silencer to provide a better understanding of its performance in HVAC systems, measurements on transmission loss (TL), and optimization of the design to improve low-frequency sound attenuation.

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