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Design and analysis of a compact acoustic filter for broad band noise absorption

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ABSTRACT

Acoustic filters have been widely used for reducing the tonal noise produced by several types of machinery such as compressors, automobiles, etc. There have been several changes in the designs of filters to increase the band of absorption such as the use of extended tube resonators, using perforates and others. However, many such filters are bulky, and due to space constraint, it is practically difficult to use them. This paper deals with the design of a compact acoustic filter having two or more cavities, which can attenuate noise in a wide frequency band by suitably varying cavity lengths. A modified theory of concentric hole cavity resonator has been used to analyze and design the filter, which is further validated experimentally by Two Load Method and then Finite Element Analysis for corroboration. Additionally, an experimental study has also been performed by varying the lengths of the cavity to get a suitable correction factor. It was observed from the results that the transmission loss peaks obtained due to different cavity lengths is capable of attenuating noise in a frequency band of user's interest. This absorption band could be altered by changing the lengths of slits, without any change in volume and size of the filter.

1. Introduction

The tonal noise produced in the ducts by various sources such as turbofans, automobiles, compressors, etc. is a cause of annoyance and noise pollution. A typical unmuffled SPL Contour plot of a turbocharged vehicle can be seen in Fig. 1.

The yellow line shown in the Fig. 1 represents a varying tonal noise of a turbocharged vehicle, which is in the frequency range of 1000 to 2000 Hz. Keeping this in mind,an acoustic filter may be developed to attenuate the noise as per the prescribed environmental norms.

Several analysis related to acoustic filters have been proposed with the passage of time to enhance the attenuation performance in broadband frequency range. Ouedraogo et al. used a perforated lining backed by a cavity to reduce broadband noise [1] and have shown that cavities could act as expansion chambers and absorb noise up to 3 kHz. Another use of perforated plates for attenuating noise was done by Cherrier [2] using a panel of resonators made by two perforated plates superimposed on each other and displaced the movable plate to tune the resonators. However, perforated plates and linings may not be applicable in every application because of relatively large volume and size. Koopman et al. [3] used a little expensive option of a secondary sound source to cancel the noise from the centrifugal fans actively, but it is also not feasible in all cases to place it due to limited space; also it has to be synchronized with the rotation of impeller by an external signal. Chen et al. [4] used metallic foams to absorb the noise, but it can't be used in the places where there are space constraints and where the gases may choke the pores of foam for example in the exhaust of automobiles.

Some filters were also developed, in due course of time, which were economical and lighter, such as, Sakamoto et al. developed a filter using a sub-muffler [5], which he proposed to install in front of the main silencer in exhaust pipe, Yasuda et al. improved inner structure of the same [6]. Takashi et al. [7] developed a new muffler by interconnecting the tailpipe with muffler using a hole. Some researchers also used Helmholtz resonator to filter the noise, for example, Bedout et al. [8] used it to attenuate the noise by tuning its filter using a feedback control law. Griffin [9] coupled two Helmholtz resonators together and gave a theory for prediction of sound absorption up to 400 Hz, but all of the above ([5–9]) had one thing in common, i.e., they could attenuate noise of low frequency only.

Considering the paucity of space in various applications, a need arises for an acoustic filter which is compact, economical, and capable of reducing noise in a wide-band. One solution for this can be the use of extended inlet and outlet ducts into the filter also known as Extended Tube Resonators (ETR). Selamet and Ji [10] investigated the effect of extended outlet and inlet on TL (Transmission Loss) and found good

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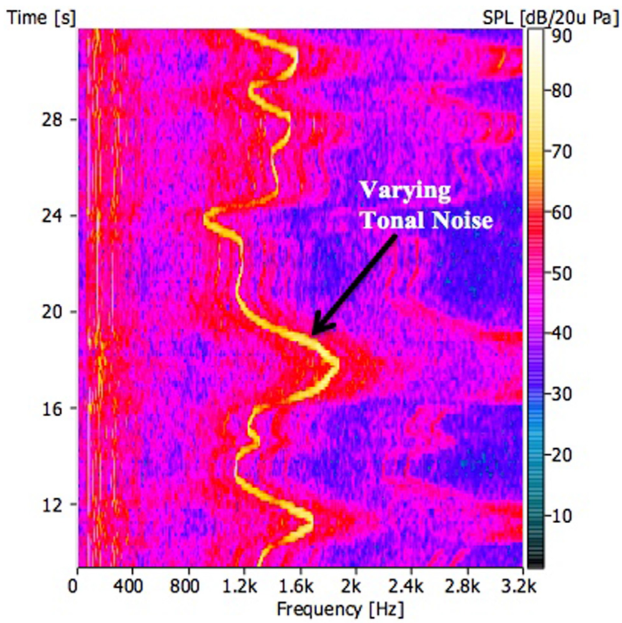

Fig. 1. Typical unmuffled SPL curve of a Turbocharger.

results. Munjal and Gowri [11] observed that, by providing quarter and half chamber lengths at inlet and outlet, 3/4th of troughs could be lifted and wideband attenuation can be achieved. But extended inlet and outlet ducts suffer from back pressure problems, and an end correction term needs to be added in the 1D model of filter to incorporate the effect of higher order evanescent modes that occur at sudden area discontinuities([12,13]). Also, to reduce noise in low frequencies up to 2000 Hz, the size of extended tubes has to be large which in turn affects the whole aim of developing a compact muffler. Ramya and Munjal [14] studied the effect of extended concentric resonators and tuned their result by giving a correlation considering several parameters like geometric and environmental factors. But the filter they proposed was having a perforated bridge between the extended inlet and outlet which is relatively large in size and volume.

To minimize these disadvantages, a new kind of compact acoustic filter has been proposed in this investigation based on the theory of modified Concentric hole Cavity Resonator (CCR). To achieve this, an extended outlet type acoustic filter has been fabricated out of mild steel having a partition in the chamber as shown in Fig. 3. Analytical modelling has been done for an open cavity which is assumed to be a concentric hole cavity resonator. A Transfer Matrix Method (TMM) [12] assuming 1-D wave propagation theory is used for prediction of TL in the broadband frequency range of 500–3000 Hz. The assumptions involved in the formulation are cavities with rigid wall with no mean flow and without any acoustic source inside the filter.

The proposed work is organized as: in Section 2, a general TMM for obtaining TL has been described which is based on 1-D wave propagation theory [12]. In Section 3, experimental validation has been shown for a fabricated prototype to validate the proposed theoretical model with experimental results. Moreover, a study of cavity dimensions has also been done, and the results are discussed to strengthen the claim further.

2. Analytical modelling

Practically for any specific broadband absorption, we need to have two or more peaks with a proper frequency gap in the TL plot. There are two ways to accomplish this as described in [12]:

- Using Extended inlet and outlet type resonator (ETR)
- Use of Concentric hole cavity resonator (CCR)

For analytical modelling of our specimen filter, we applied the theory of ETR and then CCR one by one:

For ETR:

$$Z = -jY\cot(kl) \tag{1}$$

where:

k = wave number, l = extended length of resonator pipe, Y = Characteristic Impedance;

For CCR:

The branch impedance Z of the concentric hole cavity resonator is [12]:

$$Z = \frac{1}{n}\left(j\omega\frac{l_{eq}}{S_h} + \frac{\omega^2}{\pi c}\right) + \frac{(-jY_c\cot k_0 l_a)(-jY_c\cot k_0 l_b)}{(-jY_c\cot k_0 l_a) + (-jY_c\cot k_0 l_b)} \tag{2}$$

$$Z = \frac{1}{n}\left(j\omega\frac{l_{eq}}{S_h} + \frac{\omega^2}{\pi c}\right) - jY_c\frac{1}{(\tan k_0 l_a) + (\tan k_0 l_b)} \tag{3}$$

where: sub scripts h and c stand for hole and cavity and

$S_h = \pi\frac{(d_h)^2}{4}$, d_h = diameter of hole;

$Y_c = \frac{c}{S_c}$, S_c = Annular cavity area; c= Speed of sound; $l_{eq} = t_w + k'd_h$;t_w= wall thickness; n_h = number of holes in a row; k' (correction factor) = 0.85.

In frequency domain analysis, acoustical properties of such an element can be obtained by the use of TMM and it's analogous electrical circuit. In various exhaust systems and applications, wave travelling distance is longer than the diameter of the ducts (Necessary condition for plane wave propagation and settling of higher order evanescent modes). Therefore, TMM is widely used for design of acoustic filters.

Also, as compared to other methods TMM takes less time for calculation because it is based on linear 1-D wave propagation model [12] in ducts. The relation between upstream and downstream velocity and pressure in TMM is given as-

$$\begin{bmatrix} P_u \\ V_u \end{bmatrix} = \begin{bmatrix} T_{11} & T_{12} \\ T_{21} & T_{22} \end{bmatrix} \cdot \begin{bmatrix} P_d \\ V_d \end{bmatrix} \tag{4}$$

For series elements or simple pipes the transfer matrix is-

$$\begin{bmatrix} \cos(k_0 l) & jY\sin(k_0 l) \\ \frac{j}{Y}\sin(k_0 l) & \cos(k_0 l) \end{bmatrix} \tag{5}$$

For shunt elements the transfer matrix is-

$$\begin{bmatrix} 1 & 1 \\ \frac{1}{Z} & 0 \end{bmatrix} \tag{6}$$

Now, we multiply the matrices of all the elements shown in Fig. 3 and the resultant 2 × 2 matrix is given as-

$$\begin{aligned}\begin{bmatrix} P_u \\ V_u \end{bmatrix} = &\begin{bmatrix} \cos(k_0 l_6) & jY\sin(k_0 l_6) \\ \frac{j}{Y}\sin(k_0 l_6) & \cos(k_0 l_6) \end{bmatrix} \cdot \begin{bmatrix} 1 & 1 \\ \frac{1}{Z_5} & 0 \end{bmatrix} \cdot \begin{bmatrix} \cos(k_0 l_4) & jY\sin(k_0 l_4) \\ \frac{j}{Y}\sin(k_0 l_4) & \cos(k_0 l_4) \end{bmatrix} \cdot \\ &\begin{bmatrix} 1 & 1 \\ \frac{1}{Z_3} & 0 \end{bmatrix} \cdot \begin{bmatrix} \cos(k_0 l_2) & jY\sin(k_0 l_2) \\ \frac{j}{Y}\sin(k_0 l_2) & \cos(k_0 l_2) \end{bmatrix} \cdot \begin{bmatrix} \cos(k_0 l_1) & jY\sin(k_0 l_1) \\ \frac{j}{Y}\sin(k_0 l_1) & \cos(k_0 l_1) \end{bmatrix} \cdot \begin{bmatrix} P_d \\ V_d \end{bmatrix}\end{aligned} \tag{7}$$

The transmission loss is given as [12] -

$$TL = 20\log\left(\left(\frac{Y_1}{Y_6}\right)^{\frac{1}{2}} \left|\frac{T_{11} + \frac{T_{12}}{Y_1} + Y_6 T_{21} + T_{22}}{2}\right|\right) \tag{8}$$

where:

$Y = c/S$;

S = Cross sectional Area of Resonator;

c = speed of sound in air.