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ACOUSTIC MODELING OF AN AIR CLEANER FILTER IN THE ENGINE INTAKE SYSTEM

Jeong-Guon Ih and Jang-Hoon Kang

Center for Noise and Vibration Control (NOVIC), Department of Mechanical Engineering,
Korea Advanced Institute of Science and Technology (KAIST), Taejon 305-701, Korea
E-mail: J.G.Ih@kaist.ac.kr

Abstract

The air filter in engine intake system has a function of removing the dirt in the scavenging air as well as attenuating the noise. The noise attenuation within the air cleaner filter, however, has been regarded as negligible by the field engineers. In this paper, for the analysis of the acoustical performance of air filter, an acoustical model was suggested and the characteristics of filter system were investigated. Fibrous structure of the filter element was modeled as a micro-perforated panel using the flow resistivity and porosity. The pleated geometry of the filter element was modeled as two coupled ducts that have permeable walls, in which each duct area was assumed being constant. Using such simplified geometry, a mathematical model was developed for the sound propagation within a narrow duct system. Visco-thermal effect was considered in modeling the sound propagation through such tubes; the filter box was modeled as a rigid rectangular box. By combining two models, a four-pole transfer matrix was derived. For the validation purpose, transmission loss was measured for a plastic rectangular box containing an air filter. A noticeable effect of the air filter element was observed by including the filter into the box. Comparing the predicted and measured data, we found that the predicted TL agrees well with experimental results, in particular, in magnitude and frequency at TL troughs.

INTRODUCTION

Intake noise becomes very important as one of major sources of car interior noise. In general, intake noise includes all patterns such as noise radiated from the snorkel opening and noise transmitted through pipe and air cleaner box, but this study was confined to the duct-borne noise only. The noise generated from the on-off motion of inlet valves of an engine propagates through the intake system and it is eventually radiated from the snorkel opening. During propagation inside the intake system, noise is affected by various compartments of the intake system that comprises the circular ducts with different lengths, areas, and materials, the manifold, the resonators, and the air cleaner box. In the air cleaner system, dirt and moisture in the breathing air is filtrated in the filter that is supported by the air cleaner box. The noise attenuation within the air cleaner filter has been regarded as negligible by the field engineers. However, as the noises from other sources, which were traditionally stronger than intake noise before, are being tamed by the effort of NVH engineers, such sources which were considered as negligible contributors to the overall noise become important nowadays. Also, we cannot disregard the change of overall sound quality due to even a small sound, in particular related to a tonal component.

In this study, we suggested an acoustical model for the analysis of acoustical characteristics of the air cleaner system. Acoustical behavior of the air cleaner box may be easily predicted because the box geometry is mostly a parallelepiped or a circular cylinder. However, due to the fibrous nature of material and the pleated overall structure, the air cleaner filter could not be analyzed in a simple way. From this reason, we focused on the acoustical modeling of the air cleaner filter.

Fibrous structure of a filter element was modeled as a micro-perforated panel. The pleated geometry was modeled as an assemblage of many elements comprised of two coupled ducts that have permeable walls. The cross-sectional area of a duct was assumed to be constant. Due to the lack of previous works, as far as we investigated, a theory that was used for the analysis of concentric resonators or diesel particulate filters was adopted in the development of our mathematical model [1,2]. Visco-thermal effect was considered in modeling the sound propagation through the tubes [3,4]. In addition, the parallelepiped box was used for modeling the air cleaner box. By combining the acoustical models for filter and box, a transfer matrix for the whole system could be obtained. Using the derived four-pole parameters, the acoustic performance of an air cleaner system could be predicted.

ACOUSTICAL MODEL

The air cleaner filter consists of many folded pleats, usually made of paper. Each pleat has a relatively wide width compared to the narrow gap made with the adjacent pleat layer. Based on this geometrical characteristic, the air cleaner filter was modeled as a bundle of narrow ducts that are coupled through permeable walls. However, each duct has a closed end, either in upstream or downstream side; the closed end of each narrow duct can be regarded as an acoustically rigid boundary due to high compression during the manufacturing process [5]. Figure 1 shows the model geometry and notations of a filter element.

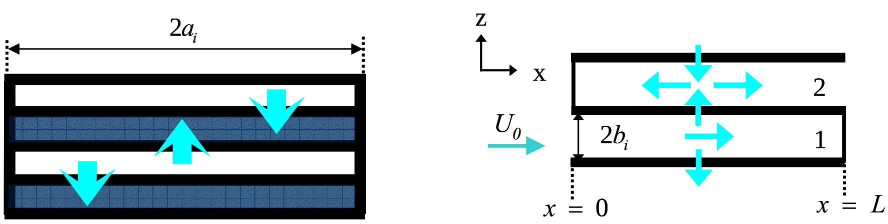

Figure 1 – A geometrical model for describing the pleated geometry of an air cleaner filter.

For the acoustic analysis of a filter element having such geometrical condition, a theory that was used for the analysis of concentric resonators or diesel particulate filters was adopted in the development of our mathematical model [1,2]. For the mathematical model, the acoustic impedance is required. By assuming that the porosity of fibrous filter material can be replaced by a group of straight tubes that have the same volume, a thin filter media can be modeled as a micro-perforated panel. Using the acoustic model of porous materials [6], the acoustic impedance of filter media can be also estimated. Although the structural factor in the porous material is also important in sound propagation in general, it may be neglected here due to thin thickness. The permeable wall was represented by the acoustic impedance as

$$p_1 - p_2 = Z_m \cdot u_w \, , \tag{1}$$

where Z_m is the acoustic impedance, p_i denotes the sound pressure in the ith duct, and u_w is the averaged velocity in the media. The impedance Z_m can be also expressed as [6]

$$Z_m = \frac{j\omega\rho_0 t}{\sigma}\left[1 - \frac{2}{s\sqrt{-j}}\frac{J_1\left(s\sqrt{-j}\right)}{J_0\left(s\sqrt{-j}\right)}\right]^{-1}, \quad s = \sqrt{\frac{8\omega\rho_0}{R_m\sigma}} \, . \tag{2}$$

Here, J_0 is the zeroth order Bessel function of the 1st kind, η the dynamic viscosity, ρ_0 the density of the air, t the thickness, and R_m, σ denote the flow resistivity and porosity of the filter media, respectively. Flow resistivity and porosity of the filter material should be determined from measurement.

Because the distance between pleats is very short, the visco-thermal effect is an important factor in the analysis of the sound propagation. For developing a mathematical model considering the visco-thermal effect, governing equations used in References [2-4] were employed with assuming linear oscillations, uniform mean flow velocity profile across the cross-sectional area of the duct, and harmonic time dependence. The boundary conditions which are characteristic to the air cleaner filter were applied to these governing equations. Equations of motion, continuity equation, energy equation, and state equation are as follows [2]:

$$\rho_{0i}\left(-j\omega + U_{0i}\frac{\partial}{\partial x}\right)u_{xi} = -\frac{\partial p_i}{\partial x} + \mu\nabla_s^{\,2}u_{xi} \, , \quad \left(-j\omega + U_{0i}\frac{\partial}{\partial x}\right)\rho_i + \rho_{0i}\nabla\cdot\mathbf{u} = 0 \, , \tag{3,4}$$

$$\rho_{0i}C_p\left(-j\omega + U_{0i}\frac{\partial}{\partial x}\right)T_i = \left(-j\omega + U_{0i}\frac{\partial}{\partial x}\right)p_i + k_{th}\nabla_s^{\,2}T_i \, , \quad \rho_i = \left(\frac{p_i}{RT_{0i}}\right) - \left(\frac{\rho_{0i}T_i}{T_{0i}}\right). \tag{5,6}$$

Here, ρ_{0i}, $U_{0\mathrm{i}}$, and $T_{0\mathrm{i}}$ are mean density, flow velocity, and temperature in each ith duct, respectively, p_i, ρ_i, T_i are perturbed pressure, density, and temperature, respectively, u is the particle velocity that comprises u_x, u_y, and u_z, for each direction, and μ, k_th, C_p, and R denote the dynamic viscosity coefficient, the thermal conductivity of air, the specific heat at constant pressure, and the gas constant, respectively. The pressure, temperature, and particle velocity in the x-direction of the ith duct can be represented as [2-4]

$$p_i = A_i \exp(j\Gamma kx),\quad u_{xi} = H_i(y,z)\,p_i,\quad T_i = F_i(y,z)\,p_i, \tag{7,8,9}$$

where Γ is the propagation constant and k is the wave number. If H_i and F_i are known, the solution of particle velocity in x-direction, temperature, and density can be obtained, and, thus, the propagation constants, Γ, can be determined by

$$\begin{bmatrix} K_2 - \dfrac{K_1}{(1-M_0\Gamma)} & \dfrac{K_1}{(1-M_0\Gamma)} \\ \dfrac{K_1}{(1-M_0\Gamma)} & K_2 - \dfrac{K_1}{(1-M_0\Gamma)} \end{bmatrix} \begin{Bmatrix} p_1 \\ p_2 \end{Bmatrix} = \begin{Bmatrix} 0 \\ 0 \end{Bmatrix}, \tag{10}$$

where

$$(K_1)_i = \frac{\gamma R T_{0i}\rho_{0i}}{j\omega Z_m b_i (1-M_i\Gamma)},\quad (K_2)_i = \left(\gamma + J(\beta_i\sigma a_i)(\gamma-1)\right) + \frac{\Gamma^2}{(1-M_i\Gamma)^2} I(\beta_i a_i), \tag{11a,b}$$

$$(\beta_i a_i)^2 = j(1-M_i\Gamma)s_i^2,\quad s_i = a_i\sqrt{\rho_{0i}\omega/\mu}, \tag{11c,d}$$

$$I(\xi) = J(\xi) = -\frac{64}{\pi^4}\sum_{m,n}\left(\frac{1}{m^2n^2\alpha_{mn}(\xi)}\right),\quad \alpha_{mn}(\xi) = 1 - \frac{\pi^2}{4\xi^2}\left(m^2 + n^2\frac{a_i^2}{b_i^2}\right). \tag{11e,f}$$

The relation between pressure and particle velocity in each duct can be obtained in matrix from as

$$\begin{Bmatrix} p_1 \\ V_{x1} \\ p_2 \\ V_{x2} \end{Bmatrix}_{x=0} = T_{4\times4} \begin{Bmatrix} p_1 \\ V_{x1} \\ p_2 \\ V_{x2} \end{Bmatrix}_{x=L}. \tag{12}$$

Because the closed end of a duct is assumed as a rigid one, the following boundary conditions can be imposed:

$$V_2 = 0 \quad \text{at} \quad x = 0,\quad V_1 = 0 \quad \text{at} \quad x = L. \tag{13a,b}$$

The 4x4 matrix in Eq. (13) can be reduced to a 2x2 transfer matrix using the boundary conditions. Because the number of pleats is N, the overall transfer matrix of an air cleaner filter can be expressed as

$$T_m = \begin{bmatrix} 1 & 0 \\ 0 & 1/N \end{bmatrix} T_{2\times2} \begin{bmatrix} 1 & 0 \\ 0 & N \end{bmatrix}. \tag{14}$$

For the derivation of transfer matrix for whole system, transfer matrix for each section should be defined [7]. By combining the transfer matrices of air cleaner filter and sections comprising air cleaner box, a transfer matrix for the whole system can be obtained (Fig. 2). Using the four pole parameters of the overall transfer matrix, the performance, e.g., transmission loss, of air cleaner system can be predicted easily.

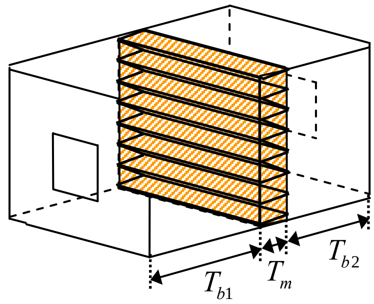

Figure 2 - The air cleaner system in total.

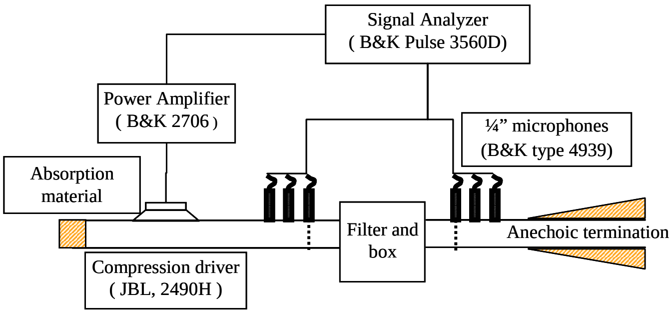

Figure 3 - Experimental set-up for the measurement of transmission loss.

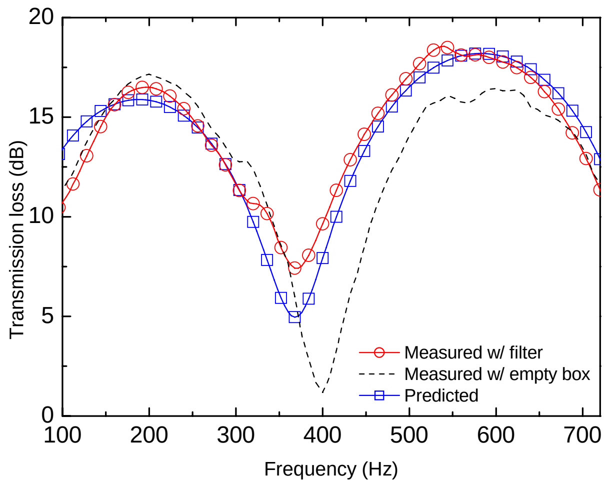

Figure 4 – A comparison of predicted and measured transmission losses.

MEASUREMENT OF TRANSMISSION LOSS

For the validation of proposed method, predicted and measured transmission losses were compared. The measurement set-up is shown in Fig. 3. A rectangular box, made from hard plastic, and an air cleaner filter, in the intake system of a passenger car, were used in the experiment. The filter had 77 pleats and the overall size was 0.14 m × 0.24 m. Measured flow resistivity and porosity of the filter media were 5.49×10^5

rayls/m and 0.85, respectively. Transmission loss was calculated using the measured reflection coefficients [8] by using the 3-microphone technique [9]. A comparison of measured and predicted results are shown in Fig. 4. The plane wave limit inside of the box was 714 Hz.

By comparing the TL of empty box and box with filter, one can observe that the acoustical effect of the air cleaner filter is significant near the trough of the TL curve, which is contrary to the general opinion. Predicted results agree well with measured data, in particular, in magnitude and frequency of troughs. One can find that, owing to the introduction of the filter, the trough frequency was shifted to low frequency region by more than 40 Hz; the amount of level change at trough was as much as 5 dB.

CONCLUSIONS

Acoustical effect of the air cleaner filter was investigated by using the narrow tube model including the permeable wall effect. By using the derived transfer matrix for the air cleaner system, the transmission loss was predicted. Predicted results agreed well with measured data, in particular, for magnitude and frequency at TL trough. It was observed that the filter influences greatly on the level change and frequency shift at the TL trough. As a further study, in order to extend the effective frequency range for prediction, high-order modes should be considered in the modeling.

ACKNOWLEDGEMENTS

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