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Study of sound insulation properties of splitter silencer with chevron blade louvers

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Abstract: *In this paper are presented results from the study of sound insulation properties of splitter silencer with chevron shape louvers. The dissipative silencer is used to attenuate noise emanating from air moving devices such as fans and is with complicated splitter design, with parallel chevron blade louvers of absorbent materials. The acoustic performance of these silencers is investigated here using a finite element based numerical mode matching scheme. The insertion loss of the silencers is then calculated and compared with the measured results on site of a real case. A lot of numerical models have been developed to predict the insertion loss of these silencers; however, there is a little experimental data available in the literature that is suitable for validating these insertion loss predictions and so questions marks still remain regarding the accuracy of the models.*

Keywords: *insertion loss, splitter silencer, chevron blade, louvers*

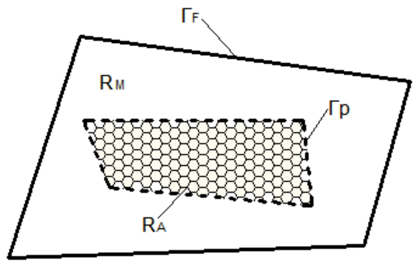
I. Introduction

Passive silencers with absorber fill are commonly used to reduce the fan noise in intake or exhaust from the fluid moving device of conventional air-handling systems. Normally at considerations of space the large reactive based silencers are inappropriate and so attention turns to the use of sound absorbing materials. Commonly, fibrous materials such as rock/glass wool or porous elastomers are used and these are arranged in parallel wings to give the so-called splitter silencer configuration. The use of only of porous materials to attenuate sound does mean that these splitter silencers generally perform well at medium to high frequencies but are less effective at lower frequencies. This article examines in more detail the design of chevron blade silencer and investigates the effect of the optimal angle between the wings and their optimal length in order to improve acoustic performance over a wider frequency range. Theoretical models for describing the wave propagation in splitter silencers as well as for predicting the acoustical insertion loss were presented by Scott [1], Morse [2] and Cremer [3]. Other prediction models are proposed by Mechel [4], Beranek [5], Ver [6] and Cummings [7] and all are based on the two models of Scott [1] and Morse [2]. The differences in the various procedures are due to the amount of complexity applied while solving the mathematical model and range from the simplistic procedures to the calculation of a large set of design curves. Other, more complete, models (e.g. Mechel [4], Sorrnaz [8] and Cummings [7], for finite length silencers without flow and Sorrnaz [8] for such silencers with flow) involve mode matching at the silencer’s terminations, and allow for a specified multimode incident sound field. Finite element analysis has also sometimes been applied to finite length silencers. Either mode matching or extended numerical schemes can require considerable computational effort, or there may be some difficulty in identifying and tracking modes in the former case. At high frequencies, where there are many incident modes, the problem may become complicated and then the need of a simpler approach such as ray acoustics can produce results of acceptable accuracy. However, these models do not take into account the scattering of sound from the inlet and outlet planes. It is only recently that these effects have been fully characterised for splitter silencers and here Lawrie and Kirby [9], and Kirby [10] used numerical and analytic methods to compute silencer insertion/transmission loss for those silencers typically found in air conditioning ducts. Here, the method of Kirby [10] is the most appropriate for studying the insertion loss of silencers with a more general design since the numerical approach facilitates the study of arbitrary cross-sectional geometries, whilst also accommodating perforated plates and fairings that are typically found in both the intake and exhaust of gas turbine silencers.

II. Theoretical model for Calculation of Insertion Loss

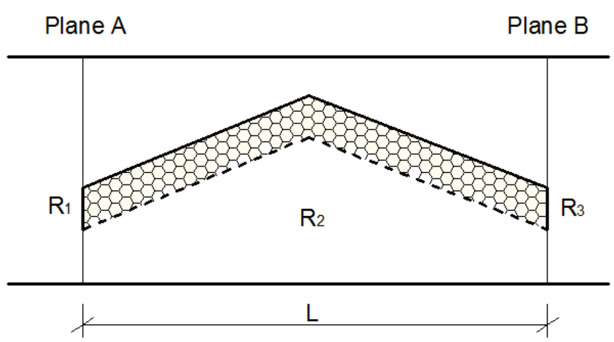
Kirby [10] presents theoretical model with two main regions and two transition regions. The main regions are fluid and absorber media respectively R_m and R_A and transition regions are denoted by Γp and $\Gamma_{F.}$ The length of absorber is considered to be uniform and the cross section is random. The initial model with schematic cross section is shown to Figure 1.

Figure 1 Plain view of the cross section of splitter silencer.



In the analysis is shown only one absorbent region in reason that the analysis is completely general and it is assumed that region R_M may be separated in future into additional regions of absorbent material in order to take on the geometry of a parallel chevron blades. Transition region Γp denotes a perforate separating the fluid from the porous material and Γ_F denotes the outer surface of the silencer, which is assumed to be a hard wall. The geometry in the axial direction is shown in Figure 2.

Figure 2 Vertical cross section of the chevron blade.



The direction of acoustic process is denoted by region of incident $R_{1,}$ grouped region $R_2 = R_M + R_A$ and the anechoic termination in region R3 with no reflected waves that are permitted in this region. The inlet plane A, and the outlet plane B, denote the location of the silencer fairings, which are present at the entrance and exit to the silencer. The acoustic wave equation for each region is given as:

$$\frac{1}{c_q^2}\frac{\partial^2 p_q}{\partial t^2} - \nabla^2 p_q = 0 \qquad (1)$$

In the equation (1) p_q is the acoustic pressure, and c is the speed of sound in region q, and t is time. First are calculated eigenmodes for the silencer cross section and then using point collocation to match continuity of pressure and acoustic particle velocity over planes A and B. The sound pressure in each region is first expanded as an infinite sum over the silencer eigenmodes, to give:

$$p_1(x,y,z) = \sum_{j=0}^{\infty} F_j \Phi_j(x,y) e^{-ik_0\gamma_j z} + \sum_{j=0}^{\infty} A_j \Phi_j(x,y) e^{+ik_0\gamma_j z} \qquad (2)$$

$$p_2(x,y,z) = \sum_{m=0}^{\infty} B_m \Psi_m(x,y) e^{-ik_0\lambda_m z} + \sum_{m=0}^{\infty} C_m \Psi_m(x,y) e^{+ik_0\lambda_m z} \qquad (3)$$

$$p_3(x,y,z') = \sum_{n=0}^{\infty} D_n \Phi_j(x,y) e^{-ik_0\gamma_n z'} \qquad (4)$$

In the previous equations A_j, B_m, C_m, D_n and F_j are modal amplitudes, λ_m, is the wave number in region R_2 and γ_j, is the wave number in the inlet/outlet section. The quantities $\Phi(x,y)$ and $\Psi(x,y)$ are the transverse duct eigenfunctions in the inlet/outlet region and the silencer section respectively. Other definitions are $i = \sqrt{-1}$ and $k_0 = \omega / c_0$.

On substituting the modal expansion for the each region into the governing wave equation, an eigenequation is obtained for regions R_1, R_2 and R_3. Here, regions R_1 and R_3 are assumed to be identical and an eigenequation may be found by implementing the hard wall boundary condition of $\nabla p \cdot n = 0$ over ΓF. This problem is solved using finite elements method and is not discussed further here. For the silencer section, one may arrive at an eigen equation by also enforcing continuity of velocity over the perforate, and for the pressure enforcing

$$p_m - p_F = \rho_0 c_0 \xi u_m \cdot n_m \tag{5}$$

Where, ζ is the (dimensionless) impedance of the perforate plane and *u* is the acoustic particle velocity. An eigenequation may then be constructed and it is solved here using the finite element method. This eigenproblem is solved using the finite element method. This eigenequation will contain the acoustic properties of the porous material, via the complex sound speed c_m and the complex density p_ω, which appears following the application of continuity of velocity between the fluid and the material. On obtaining the eigenvalues λ_m and γ_j, and the eigenvectors $\Phi(x, y)$ and $\Psi(x, y)$ the silencer insertion loss may then be computed. To do this it is necessary to enforce continuity of acoustic pressure and normal particle velocity over planes A and B. The modal amplitudes are found by enforcing the matching conditions over plane A,

$$p_1(x, y, 0) = p_2(x, y, 0) \quad \text{over } R_M \tag{6}$$

$$u_1(x, y, 0) = u_2(x, y, 0) \quad \text{over } R_M \tag{7}$$

$$u_1(x, y, 0) = 0 \quad \text{over } R_A \tag{8}$$

$$u_2(x, y, 0) = 0 \quad \text{over } R_A \tag{9}$$

And over plane B,

$$p_2(x, y, L) = p_3(x, y, 0) \quad \text{over } R_M \tag{10}$$

$$u_1(x, y, 0) = u_2(x, y, 0) \quad \text{over } R_M \tag{11}$$

$$u_2(x, y, L) = 0 \quad \text{over } R_A \tag{12}$$

$$u_3(x, y, 0) = 0 \quad \text{over } R_A \tag{13}$$

Equations (2)-(4) must be converted in vector form and after that substituting into equations (6) to (13) to obtain a complete set of $n_t = 2(n_1 + n_2)$ simultaneous equations. After setting $F_0 = 1$, and $F_j = 0$ for $j > 0$ (plane wave excitation) then equations (6) to (13) may be solved for the unknown modal amplitudes. The silencer insertion loss IL is then defined as:

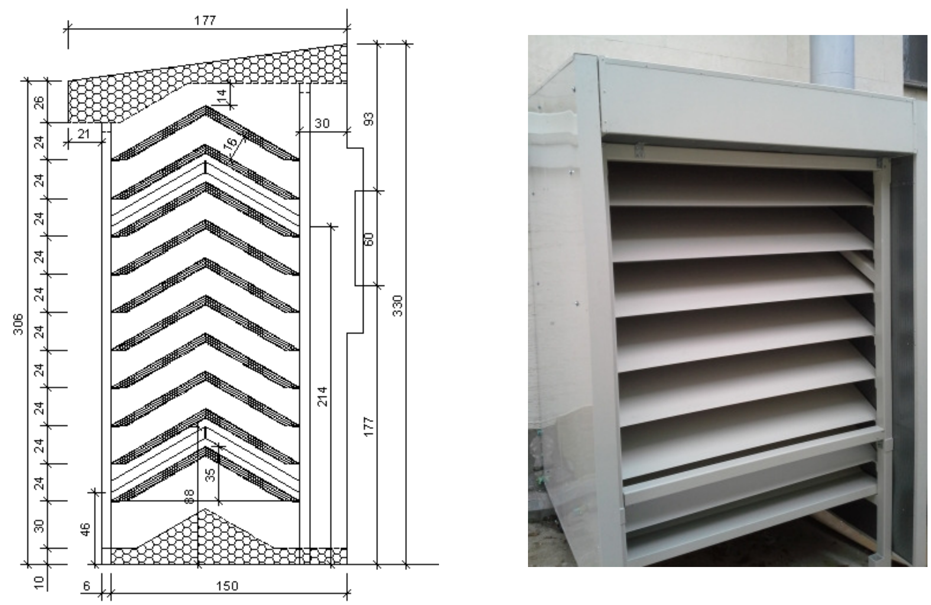
$$IL = -10 \log_{10} \mathrm{Re}\left[\sum_{n=o}^{n_1} \frac{\gamma_n I_n |D_n|^2}{I_0} \right] \tag{9}$$

$$I_n = \int_{R_1} |\Phi_n(x, y)|^2 \, dxdy \tag{10}$$

III. Design of splitter silencer and chevron blade louvers

For current case there is a limitation of the dimensions of the silencer due to surrounding space. The construction for the corpus is made by steel panels with outer sheets with thickness of 0.6 mm, core of 80 mm stone wool with density of 70 kg./m^3 and inner sheets of perforated steel with 25 % opening. Chevron blades are produced from the same metal parts with stone wool core with thickness of 50 mm. All the panels and blades are mounted on metal frame. Construction design of vertical cross section and picture from the onsite installation are shown at Figure 3.

Figure 3 Schematic view of the cross section and onsite picture of splitter silencer.

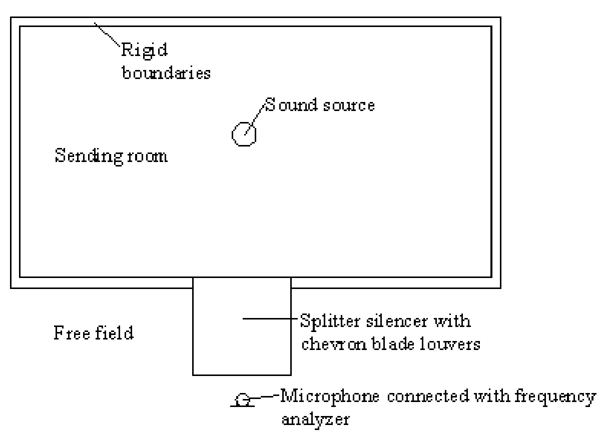


The stone wool applied here has a slightly larger average fibre diameter of 6-18 µm. For the analysis is demanded a knowledge of the bulk complex density ρ(ω), and the propagation constant P_G for the porous material. It is convenient to write ρ(ω) in terms of the (complex) characteristic impedance (z_a), where $\rho(\omega) = z_a P_G / i\omega$. The propagation constant and characteristic impedance are specified here by combining the empirical power law approach of Delany and Bazley [11] with theoretical low frequency corrections.

IV. Comparision between calculated insertion loss and measured results

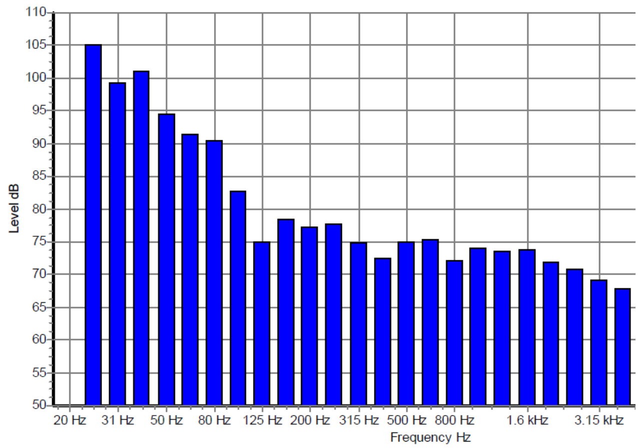
To verify the predictions is provided experimental measurement with sound source placed in a sending room with rigid boundaries. At one of the boundaries there is an opening on which is placed the splitter silencer. The sound pressure level is measured on the exit of the opening without take into account the reverberation time of the sending room. On Figure 4 is shown plane view of the experimental set.

Figure 4 Plan view of the experimental installation.



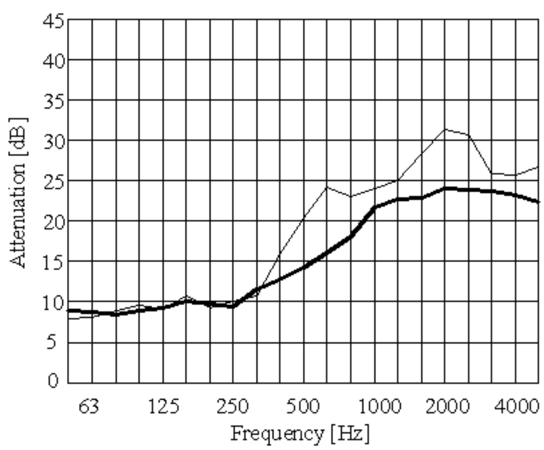
The sound source is generated from existing compressor machinery and is made six measurements on different points of the sound pressure level. The angle of the incident wave on the intake opening of the silencer is accepted to be random. The averaged sound pressure level is shown on Figure 5.

Figure 5 Sound pressure level generated from the sound source presented by frequencies.



On Figure 6 is presented comparison between calculated and measured results. For the low and middle frequency range the prediction model works relatively well below 315 Hz and predictions lie within approximately 2 dB of measured values. Up to these frequencies there is a deviations up to 8 dB. Larger discrepancies at higher frequencies are, however, likely to be caused by experimental error. Nevertheless, over the frequency range studied here, agreement between prediction and experiment is deemed to be acceptable and is at least comparable in accuracy to studies of dissipative silencers by other author.

Figure 6 Sound pressure level generated from the sound source presented by frequencies.



V. Conclusions

A finite length dissipative silencer of arbitrary, but uniform, cross section has been modeled by combining a finite element eigenvalue analysis with a point collocation matching scheme. The method is computationally efficient when compared to a three-dimensional finite element approach and avoids the question of modal orthogonality. A good correlation between prediction and experiment is observed both with and without mean flow, up to a frequency of 315 Hz for the silencers studied here, although in principle the method is applicable over a much wider frequency range. Furthermore the flexibility and robustness of the finite element method allows the technique to be applied to any cross sectional dissipative silencer geometry, such as rectangular air conditioning ducts, and, in principle, to include any number of duct discontinuities. Future work let on study of the geometry of the main silencer body and position of the chevron blades in the inner volume regarding axial axis, and of course optimal angle between the chevron wings in accordance of pressure drop and low frequency transmission.

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