

Analysis of Absorptive Muffler for Inductive & Resistive Damping

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Abstract : *The analysis of mufflers has always been a great challenge. In this paper Glass wool as a absorptive material with different density has been used to analyze mufflers. This is used to evaluate the transmission loss. Results from the present approach have been validated through comparison with the finite element method (FEM) results using COMSOL 5.0. Two different muffler configurations are considered, representing the effects adding lining with different densities desired broadband attenuation performance can be achieved, from computational analysis it is observed that liner with different densities, to tackle different frequency regions, a desired attenuation performance can be achieved, thus offering wider scope in silencer design.*

Keywords – *Transmission loss, Acoustic liners,*

I. INTRODUCTION

Silencers are extensively used for the attenuation of exhaust noise in an automobile. The attenuation of broadband noise has always been a challenging topic in the acoustical design of engine exhausts, automotive mufflers. An increasing demand for their efficient performance has led to the development of various acoustic elements like area discontinuities, adding absorptive liner with changing their densities, perforated surfaces, baffles and so on. However, these elements also introduce substantial pressure drop in the flow. To overcome this drawback, absorptive liner with various combination of area discontinuities are used in muffler design.

II. LITERATURE REVIEW

The acoustic behavior of a circular dual-chamber muffler is investigated in detail by D. Denia [1] 2D axisymmetric analytical approach based on the mode-matching technique for concentric configurations. A number of effects is studied, including rigid baffle presence in the chamber, its radius & position along axial direction with and extended inlet/outlet. Some of these effects are shown to modify the acoustic behavior drastically. S. Bilawchuk [2] et. al. The increased use of large industrial machinery and the increase in design silencers for specific applications is increasing. Due to the size and expense of these silencers, it would be beneficial to have means to predict the insertion loss (IL) or transmission loss (TL) characteristics at the design stage. The derivation is based on the mode matching technique and is verified by measurements done on prototypes. R. Glav [3] presented for 2-port matrix for a folded quarter-wave side-branch resonator including higher order modes but neglecting flow interaction effects. The improved four parameters method is used for TL evaluations and the Helmholtz's equation is solved numerically with the finite element method (FEM). Renato Barbieri [4] et. al. Combines finite element analysis and Zoutendijk's feasible directions method for mufflers shape design. The quadratic triangular finite element meshes are adequately constructed to control the pollution error and the optimization problem is solved using the Zoutendijk's feasible directions method due to robustness and efficiency for problems with nonlinear constraints Numeric experiments performed with circular expansion chambers with extended inlet and outlet show results for constrained and unconstrained shape optimization. Min-Chie Chiu [5] worked on shape optimization of multi-chamber plug-inlet mufflers along with work on maximal back pressure has been sorely neglected. Key Fonseca de Lima,[6-7] use of Parametric optimization to evaluate the appropriate size of the inlet and outlet ducts. The results showed the optimization efficiency of the inlet ducts profile for acting in specific frequency ranges with gains up to 20 dB with respect to silencers without shape optimization. The numerical analyses agree well with experimental results. The commercial automotive mufflers are generally of a complicated shape with multiply connected parts and complex acoustic elements. an Integrated Transfer Matrix method has been developed to analyze complex mufflers. Results from the present approach have been validated through comparisons with the available experimental and three-dimensional finite element method (FEM) based results. The results show good agreement with both measurements and FEM analysis up to the cut-off frequency. The broadband sound attenuation characteristics of expansion chamber silencers can be altered by their internal configuration. Xiang Yu, Li Cheng [8] In order to tackle the system complexity, to investigate the effects of several typical silencer configurations and provide

guidelines for possible system optimization. Xiang Yu [9-12] et. al. A theoretical basis is presented for a description of the overall transmission loss (TL) of the silencer, using the TLs of each of the cascade-connected multiple sub-chambers and the interactions between them. It is demonstrated that, by connecting sub-chambers with optimized TLs to tackle different frequency regions, a desired broadband attenuation performance can be achieved. V. Ovidiu [10] investigated acoustic performance of a reactive silencer for two special cases using numerical and experimental techniques. The method is applied for each silencer configuration and the numerical predictions are compared with the results obtained by means of an experimental setup. Jin Woo Lee [11] proposed and validated experimentally a topology-optimization-based muffler design method for a reactive muffler. Numerical simulation results show that the proposed formulation requires a smaller volume of partition than the previous topology-optimization-based formulation.

III. TYPES OF MUFFLERS

3.1 Reactive Type Muffler

The reactive or reflective mufflers use the phenomena of destructive interference to reduce noise. For complete destructive interference to occur a reflected pressure wave of equal amplitude and 180 degree out of phase needs to collide with transmitted pressure wave. Reflections occur when there is a change in geometry or an area discontinuity. This is achieved with the help of perforations in the pipes inside the muffler. The inlet and outlet tubes are generally offset and have perforations that allow sound pulses to scatter out in numerous directions inside a chamber resulting destructive interference. They have the ability to reduce noise at various frequencies due to the numerous chambers and changes in geometry that the exhaust gasses are forced to pass through. Reflective mufflers, also called as reactive mufflers are non-dissipative. A reflective muffler consists of a number of tubular elements of different transverse dimensions joined together so as to cause, at every junction, impedance mismatch and hence reflection of a substantial part of the incident acoustic energy back to the source.

3.2 Absorptive Type Muffler

Sound waves are reduced as their energy is converted in the heat in the absorptive material. A typical absorptive muffler consists of a straight, circular and perforated pipe that is encased in a larger steel housing. Between the perforated pipe and the casing is a layer of sound absorptive material that absorbs the sound of the pressure pulses. Dissipative mufflers consist of ducts lined on the inside with an acoustically absorptive material. Absorptive mufflers create less back pressure than reactive mufflers. Generally reactive mufflers use resonating chambers that target specific frequencies to control noise whereas an absorptive muffler reduces noise considerably over the entire spectrum and more so at higher frequencies.

IV. THEROTICAL ANALYSIS OF MODEL

This model solves the problem in the frequency domain using the pressure acoustics, frequency domain interface. The model equation is a slightly modified version of the Helmholtz equation for the acoustic pressure p .

$$\nabla \cdot \left(-\frac{\nabla p}{\rho} \right) - \frac{\omega^2 p}{c^2 \rho} = 0 \dots\dots\dots [1]$$

Where ρ is the density, c is the speed of sound, and ω is the angular frequency. In the absorbing glass wool, modeled as a Poroacoustics domain, the damping enters the equation as a complex speed of sound, $c_c = \omega / kc$, and a complex density, $\rho_c = k_c Z_c / \omega$, where k_c is the complex wave number and Z_c equals the complex impedance. This is a so-called equivalent fluid model for the porous domain where the losses are modeled in a homogenized way. For a highly porous material with a rigid skeleton, the well-known model of Delany and Bazley estimates these parameters as functions of frequency and flow resistivity. Using the original coefficients of Delany and Bazley the expressions are

$$k_c = k_a \left(1 + 0.098 \times \left(\frac{\rho_a f}{R_f} \right)^{-0.7} - i \times 0.189 \times \left(\frac{\rho_a f}{R_f} \right)^{-0.595} \right) \dots\dots\dots [2]$$

$$Z_c = Z_a \left(1 + 0.057 \times \left(\frac{\rho_a f}{R_f} \right)^{0.374} - i \times 0.189 \times \left(\frac{\rho_a f}{R_f} \right)^{-0.732} \right) \dots\dots\dots [3]$$

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Where R_f is the flow resistivity, and where $k_a = \omega / c_a$ and $Z_a = \rho_a c_a$ are the free-space wave number and characteristic impedance of air, respectively. This is the default selected porous model in the Poroacoustics Domain feature. Several porous models can be selected here depending on the situation at hand.

An important parameter for a muffler is the transmission loss or attenuation. It is defined as the ratio between the incoming and outgoing acoustic energy. The attenuation or transmission loss TL (dB) of the acoustic energy is defined by

$$TL = 10 \text{Log} \left(\frac{P_{in}}{P_{out}} \right) \dots\dots\dots [4]$$

Here P_{in} and P_{out} denote the incoming power at the inlet and the outgoing power at the outlet, respectively. You can calculate each of these quantities as an integral over the corresponding inlet and outlet surface. Assuming plane wave propagation the expressions are

$$P_{in} = \int_{\Omega} \frac{P_0^2}{2\rho c} dA \quad \text{and} \quad P_{out} = \int_{\Omega} \frac{|p|^2}{2\rho c} dA$$

4.1 Cut-Off Frequency

It can be defined as the frequency above which all the frequencies are suppressed by the muffler. This is achieved by using a Low pass filter or a High pass filter in case of mufflers. Cutoff frequency, corner frequency, or break frequencies a boundary in a system's frequency response at which energy flowing through the system begins to be reduced (attenuated or reflected) rather than passing through.

V. COMPUTATIONAL ANALYSIS & STEPS INVOLVED

5.1 Comsol Multiphysics Acoustic Module

This module solves problems in the general areas of acoustics, acoustic-structure interaction, aero acoustics, thermo acoustics, pressure and elastic waves in porous materials, and vibrations. The physics interfaces included are fully Multiphysics enabled, making it possible to couple them to any other physics interface in COMSOL Multiphysics 5.0

5.2 Pressure Acoustics Frequency Domain

Frequency Domain Study Wave propagation is modeled by equations from linearized fluid dynamics (pressure waves) and structural dynamics (elastic waves). The full equations are time dependent, but noting that a harmonic excitation of the field u has a time dependence of the form

$$u = ue^{i\omega t}$$

Gives rise to an equally harmonic response with the same frequency; the time can be eliminated completely from the equations. Instead the angular frequency $\omega = 2\pi f$, enters as a parameter where f is the frequency.

5.2 Parameter Required for Simulation

Table 1 List of Parameters required for simulation

Name	Value	Description
p0	1[Pa]	Amplitude of incoming pressure wave
rho_ap	24[kg/m ³]	Apparent density of glass wool
d_av	10[um]	Mean fiber diameter
R_f	4112.9 kg/(m ³ .s)	Flow resistivity
L	600[mm]	length
A	100[mm]	Minor axis length
B	200[mm]	Major axis length
L_io	150[mm]	Inlet and outlet length
R_io	50[mm]	Inlet and outlet radius
D	15[mm]	Liner thickness

5.3 Boundary Conditions

The boundary conditions are of three types.

- The model uses hard wall boundary conditions; their velocity at the boundary is zero.
- The boundary condition at the inlet involves a combination of incoming and outgoing plane waves.
- At the outlet boundary, the model specifies a radiation condition for an outgoing plane wave.

5.4 Meshing the Model

The waves are characterized by a wavelength λ in space, whose value depends on the frequency and speed of sound c in the medium according to $\lambda = c / f$. This wavelength has to be resolved by the mesh. In reality, the lower limit for a fully reliable solution lies at about ten to twelve degrees of freedom per wavelength. Because the direction of propagation is generally not known beforehand, it is good practice to aim for an isotropic mesh with about twelve DOFs per wavelength on average, independently of the direction.

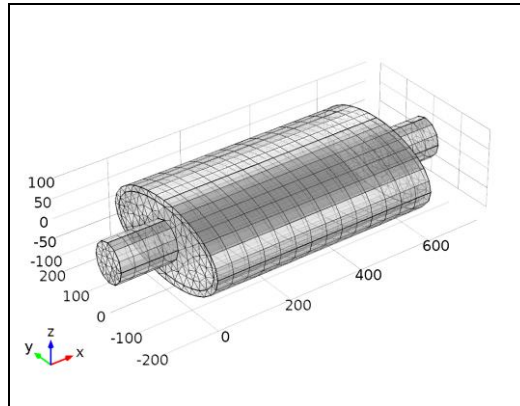


Fig. 1 Meshed model

VI. RESULT AND DISCUSSIONS

The pressure distribution in the absorptive muffler without the lining material is shown in Fig.2 a) & b) for the frequency $f = 950$ Hz. From the figure it is seen that at this frequency not only longitudinal standing waves exist but also transverse modes are present. At frequencies higher than approximately 950 Hz, the plot's behavior is more complicated and there is generally less damping. This is because, for such frequencies, the tube supports not only longitudinal resonances but also cross-sectional propagation modes. Not very far above this frequency a whole range of modes that are combinations of this propagation mode and the longitudinal modes participate, making the damping properties increasingly unpredictable. The glass-wool lining improves attenuation at the resonance frequencies as well as at higher frequencies. The flow of energy in the muffler is shown in Fig. 4 at 950 Hz without the liner. The plot represents the intensity field depicted as streamlines.

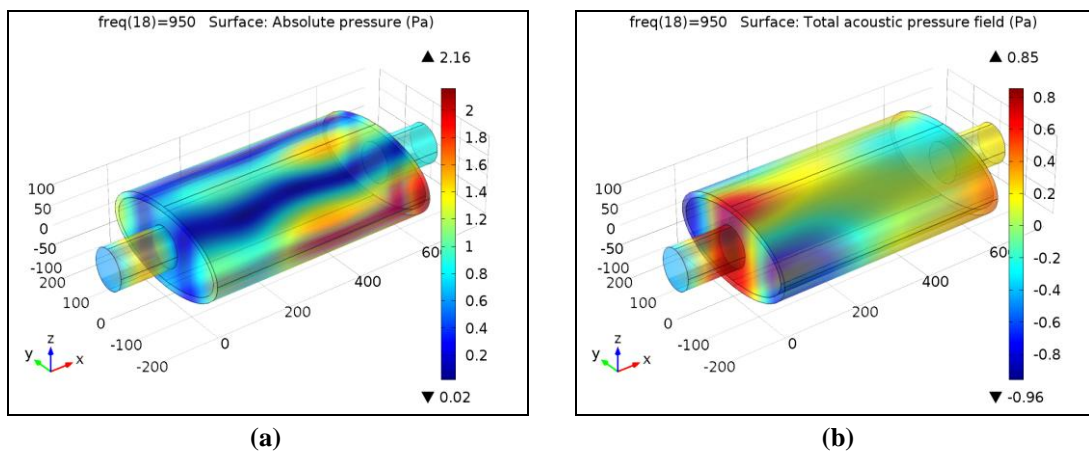


Fig. 2 Absolute pressure level a) without liner b) With Absorptive liner

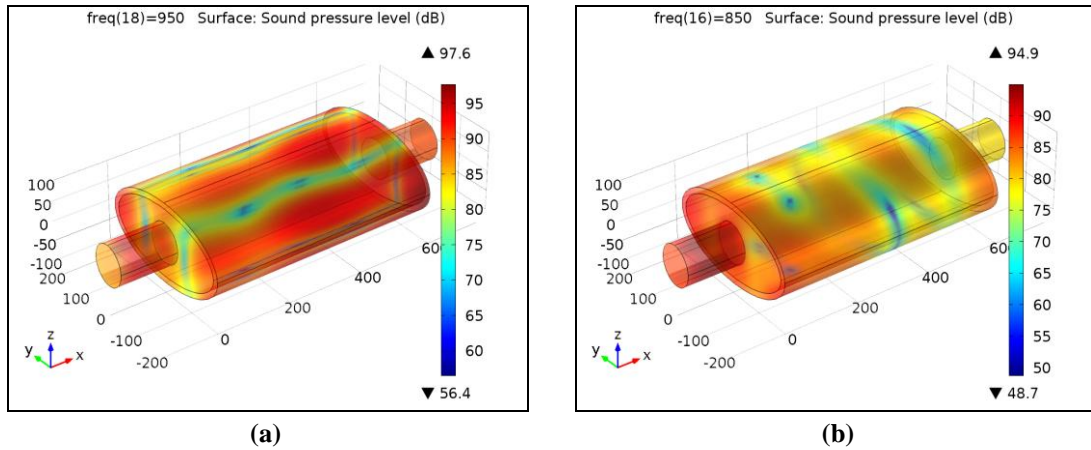


Fig. 3 Sound pressure level a) without liner b) With Absorptive liner

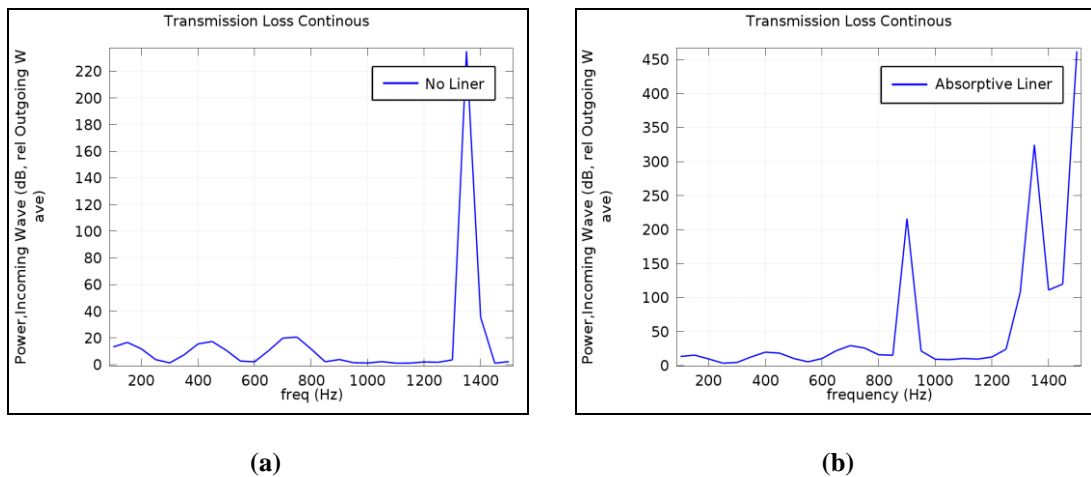


Fig. 4 Transmission loss a) without liner b) With Absorptive liner

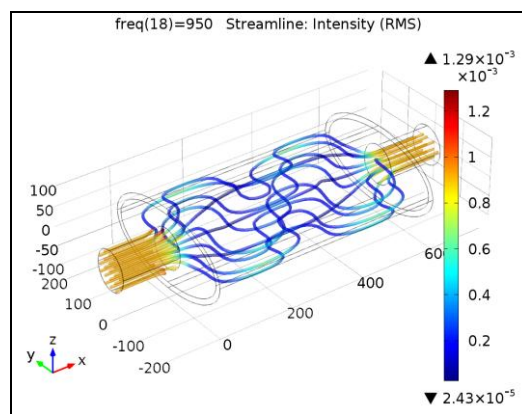


Fig. 5 Stream line Intensity (RMS)

VII. CONCLUSION

The Fig.3 (a & b) is the exported 1-D plot data from the COMSOL Multiphysics. This plot shows the Transmission loss [dB] on y-axis and Frequency [Hz] on the x-axis. The dips in the graph are due to longitudinal resonance and the dips reduce due to the absorptive lining of the silencer. The Fig.3 shows the values of Transmission loss obtained from a frequency range of 0 Hz to 1500Hz. These graphs tell us about the transmission losses across a range of frequency mentioned above. It is observed that on a frequency value of 950Hz we get maximum attenuation of 97.6dB in case 1) without liner in case 2) frequency value of 950Hz we

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get maximum attenuation of 94.9dB, from both value we can clearly see that by using Adsorptive liner the value of is decreased, similarly by use of lower density material we can maximize transmission loss, which provide wider scope in different combination of material for better design.

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