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Attenuation analysis and acoustic pressure levels for double expansion chamber reactive muffler: Part 2

Mahesh V Kulkarni[1] and Ravindra B Ingle[2]

Abstract

The major source responsible for noise pollution is internal combustion engine. These engines are used for various purposes such as in automobiles, locomotives, and in various manufacturing machineries. In an engine, the exhaust noise and the noise produced due to friction of various parts of the engine share maximum contribution to noise pollution. Muffler is a device used to reduce noise within the exhaust system. It is arranged along the exhaust pipe for the purpose of noise attenuation. The paper describes the propagation of pressure wave in a double expansion chamber reactive muffler. The approach is useful in analysis of damping for propagation of harmonic pressure waves. The purpose of paper is to describe the finite element analysis of double expansion chamber reactive muffler using pressure acoustics and to validate it with experimental evaluation using two-load method.

Keywords

Reactive muffler; transmission loss; finite element analysis; acoustic pressure levels; two-load method

Introduction

Noise pollution produced by engines becomes a major concern when used in residential areas or areas where noise creates hazard. The exhaust noise is the most harmful. Noise level greater than 80 dB is injurious for human being. Various types of muffler are used to attenuate this noise. The reduction in the level of exhaust noise depends upon the construction and the working procedure of mufflers. Therefore, design of muffler plays an important role as it affects the noise characteristics and fuel efficiency of the engine. The exhaust muffler is characterized by numerous parameters like insertion loss (IL), transmission loss (TL). TL is one of the most frequently used criteria of muffler performance because it can be predicted very easily from the known physical parameters of the muffler. The TL could be achieved by analytical, numerical, and experimental method. Analytical methods are cumbersome as the associated algebra is complicated; therefore, many times it is impossible to solve such problems by analytical methods.[1] The numerical methods are general and allow the analysis of all types of mufflers and therefore used for optimization of model of complicated shapes and cost involved is less than experimental methods. In this article, the double expansion chamber reactive muffler is examined using finite element method. The detailed design procedure is available in the literature.[2,3]

Geometry definition

The muffler depicted in Figure 1 consists of two resonator chambers. The exhaust pipe is attached at center at both ends. The length of input output tube is 95 mm and diameter is 44 mm. The resonator chamber is having circular cross section with diameter of 110 mm and length of 95 mm. The connecting tube for both chambers is having length of 95 mm and diameter of 44 mm.

The input–output tube diameter is taken same as that of engine exhaust pipe. The exhaust fumes enters through the left pipe called as exhaust pipe and exits through right pipe called as tail pipe.

Domain equation

The muffler model solves the problem in the frequency domain.[4] It uses the time harmonic pressure acoustics

[1]Department of Mechanical Engineering, MAEER's Maharashtra Institute of Technology, Pune, India
[2]Department of Mechanical Engineering, MKSSS's Cummins College of Engineering for Women, Pune, India

Corresponding author:
Mahesh V Kulkarni, Department of Mechanical Engineering, MAEER's Maharashtra Institute of Technology, Paud Road, Kothrud, Pune 411038, Maharashtra, India.
Email: mahesh.kulkarni@mitpune.edu.in

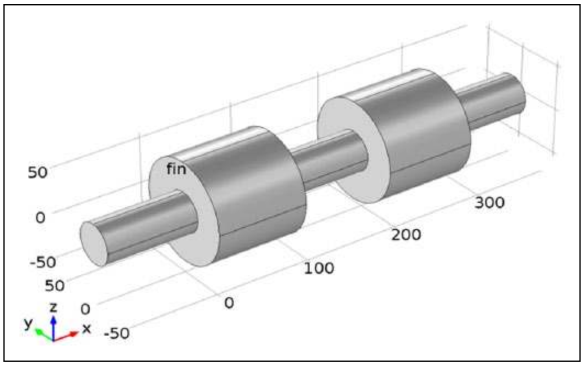

Figure 1. Double expansion chamber reactive muffler.

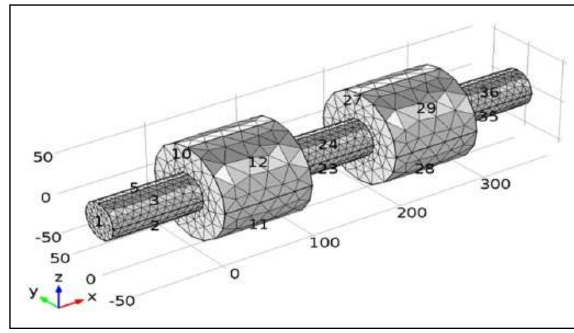

Figure 2. Meshed model of double expansion chamber reactive muffler.

application mode. The equation for the model is given by

$$\nabla\left(-\frac{\nabla p}{\rho}\right)-\frac{\omega^2 p}{c_s{}^2\rho}=0 \tag{1}$$

where ρ is density, C_s is the sound speed, and ω is angular frequency.

Boundary conditions

The boundary conditions are of three types

1. The model uses sound hard (wall) boundary conditions at the solid boundaries, in this case the outer walls of resonator chamber and pipes

$$\left(-\frac{\nabla p}{\rho}\right)\cdot n=0 \tag{2}$$

2. The boundary condition at the inlet involves the combination of incoming and outgoing plane waves

$$n\cdot\frac{1}{\rho_0}\nabla p+ik\frac{p}{p_0}+\frac{i}{2k}\Delta_T\cdot p=\left\{\frac{i}{2k}\Delta_T\cdot p_0+\left[1-(k\cdot n)\right]ik\frac{p_0}{\rho_0}\right\}e^{-ik(k\cdot r)} \tag{3}$$

where Δ_T is boundary Laplace operator, i is the imaginary unit, and p_0 represents the applied outer pressure. As long as the frequency is kept below the cut-off frequency for the second propagating mode in the muffler, this boundary condition is valid.

3. The model specifies the outgoing plane wave at the outlet boundary

$$n\cdot\frac{1}{\rho_0}\nabla p+i\frac{k}{\rho_0}p+\frac{i}{2k}\Delta_T\cdot p=0 \tag{4}$$

Meshing

The required boundary conditions are applied to the model and then meshing is performed. The option of physics controlled mesh is chosen and finer size is allocated for meshing. The maximum and minimum element size for meshing is 26.1 and 1.9 mm, respectively. Figure 2 shows the three-dimensional (3D) meshed model for double expansion chamber reactive muffler.

The equation for analyzing the attenuation (dB) of acoustic energy (also known as TL) is given by[4–6]

$$d_w=10\log\frac{w_0}{w_i} \tag{5}$$

where w_0 denotes outgoing power at outlet and w_i denotes incoming power at inlet. These quantities can be calculated by taking integral over corresponding surface as follows

$$w_i=\int_{\partial\Omega}\frac{p_0{}^2}{2\rho c_s}dA \tag{6}$$

and

$$w_o=\int_{\partial\Omega}\frac{|p|^2}{2\rho c_s}dA \tag{7}$$

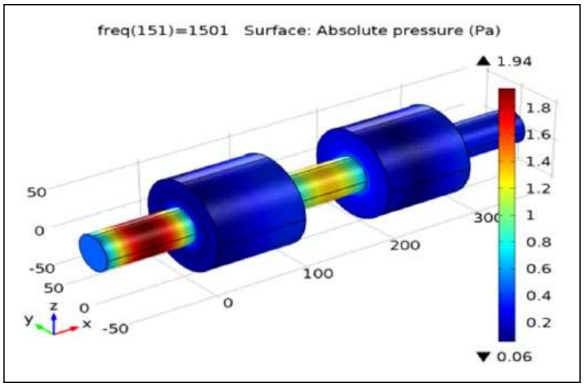

Figure 3. Acoustic pressure level (Pa) at 1501 Hz.

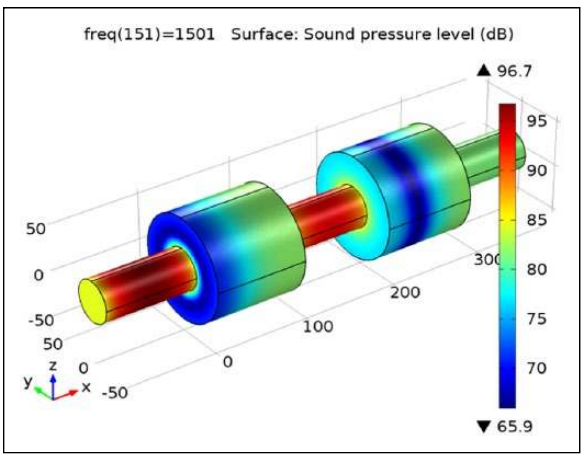

Figure 4. Sound pressure levels (dB) at 1501 Hz.

Post processing of numerical analysis

Figure 3 shows the acoustic pressure levels, absolute pressure, for the double expansion chamber reactive muffler and Figure 4 shows sound pressure level in the muffler. The variation in color indicates change in pressure which is useful in determining the presence of resonance inside the muffler.

Figure 5 shows the plot of total acoustic pressure field for the muffler. Figures 3–5 shows plots corresponding to frequency of 1501 Hz. A 3D analog of an isoline is called as isosurface. This surface represents points of constant value within volume of space of muffler (in present study, it is pressure); in other words, it is a level set of a continuous function whose domain is 3D space. For the representation as in Figure 5 ,the model was set to a total of 10 levels.

Figure 6 shows the result of attenuation of the muffler, a parametric frequency study for the case of an empty muffler without any absorbing material. The plot depicts that for most low frequencies, the damping works rather well, with the exception of a few distinct dips where the muffler chamber displays resonances. The maximum TL occurs between the frequency range of 800–900 Hz.

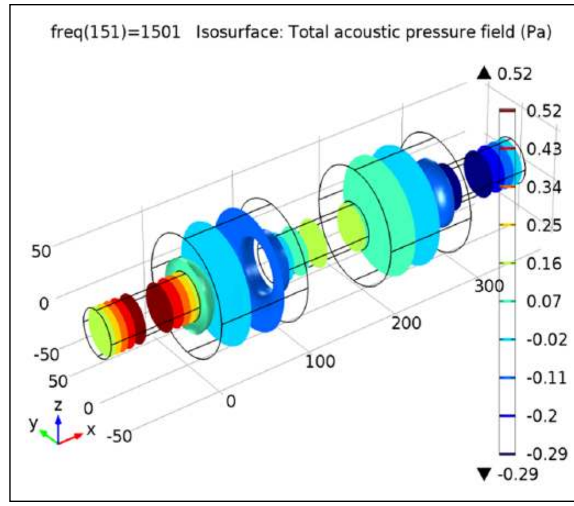

Figure 5. ISO surface-total acoustic pressure field.

At frequencies higher than approximately 1501 Hz, the plot's behavior is more complicated and there is generally less damping.

Experimental analysis

The experimental analysis is carried out using the two-load method. The two-load method is based on the transfer matrix approach.[6] Using the transfer matrix method, one can readily obtain TL of any muffler using four-pole equations from the four positions of microphones. The setup for experimental analysis consists of (1) system for noise generation, (2) system for noise propagation, and (3) system for noise measurement. The main components of setup are sound source, amplifier, FFT analyzer, and impedance tube [ref Part 1]. The experiment is performed for frequency range of 1–2000 Hz. A signal generated in analyzer is random noise signal, which is directed to the speaker through amplifier. The speaker converts the signal into sound wave which travels through impedance tube and muffler. The microphones collect the sound pressure signals and after amplifying them, transmit to FFT analyzer. In the experimental analysis, two microphones are used to apply random excitation technique; the transfer functions which are obtained are used directly to four-pole elements to calculate the TL. The TL is calculated experimentally using equation

$$\mathrm{TL} = 20\log_{10}\left[\frac{1}{2}\left(\left|\mathrm{A}_{14} + \frac{\mathrm{B}_{14}}{\rho c} + \rho c\,\mathrm{C}_{14} + \mathrm{D}_{14}\right|\right)\right] \qquad (8)$$

Results and conclusion

Figure 7 shows the TL curve obtained for the experimental analysis. The curve in red line shows TL obtained through

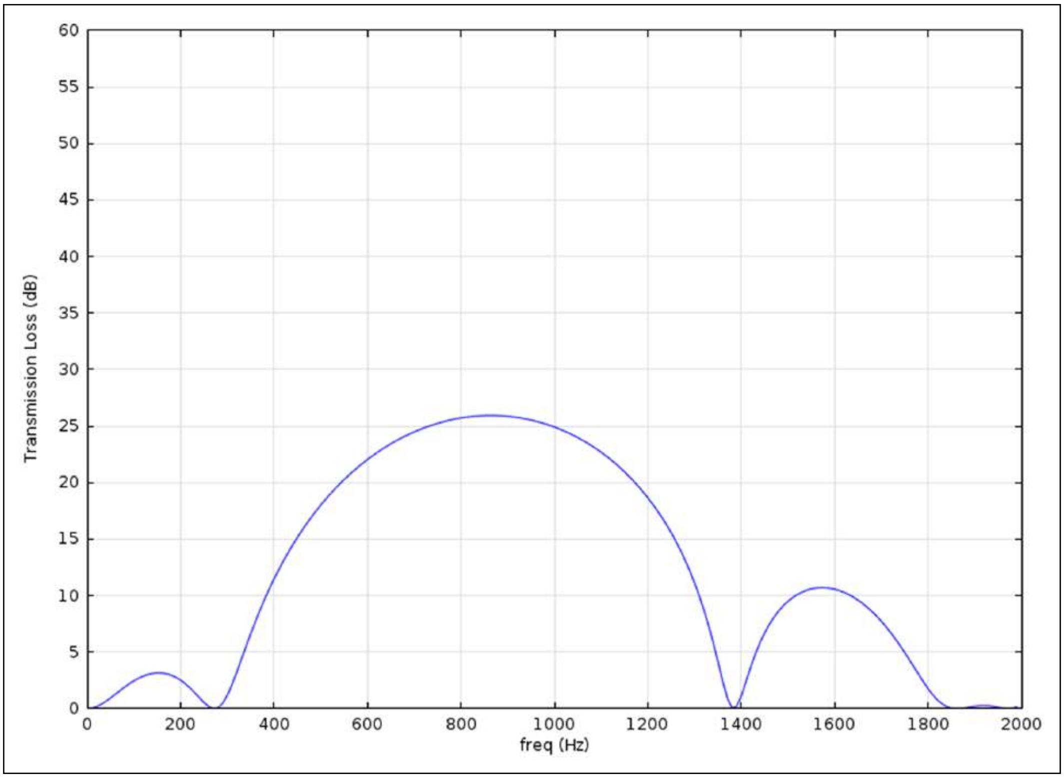

Figure 6. Attenuation of muffler as a function of frequency.

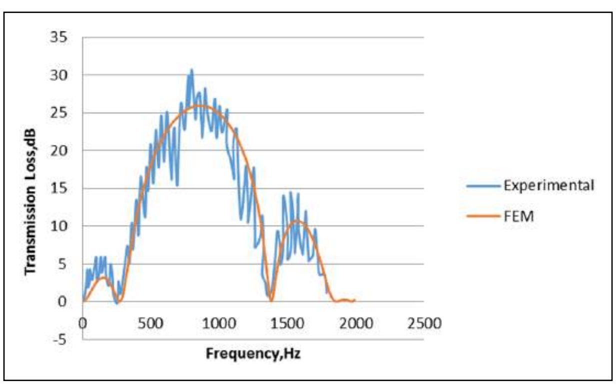

Figure 7. TL curve for experimental analysis.

numerical analysis, while the curve in blue line shows the TL obtained through experimental analysis. The TL at 800 Hz (design frequency) is 30.55 dB, while the TL calculated using numerical analysis is 25.73 dB. The percentage error calculated is 15.77% which shows that the experimental results calculated are in good agreement with the numerical analysis.

In this research article, the double expansion chamber reactive muffler is analyzed with finite element analysis using pressure acoustics and the result is verified with experimental analysis using two-load method. The results obtained through experimental analysis agreed well with numerical analysis. The small deviation in experimental result from that of numerical result may be attributed to leakage of sound from impedance tube to the surrounding, problems in generating white noise from FFT, inaccurate surface finish quality of impedance tube. From the result, it is concluded that for most low frequencies, the damping works rather well, with the exception of a few distinct dips where the muffler chamber displays resonances.

At frequencies higher than approximately 1501 Hz, the plot's behavior is more complicated and there is generally less damping. This is due to reason that for such frequencies, the muffler supports longitudinal resonances as well as cross-sectional propagation modes.

Declaration of conflicting interests

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References

1. Narayana TSS and Munjal ML. Prediction and measurement of the four pole parameters of a muffler including higher order mode effects. *Noise Control Eng J* 2005; 53(6): 240–247.

2. Rahman M, Sharmin T, Hassan AFME, et al. Design and construction of a muffler for engine exhaust noise reduction. In: *Proceedings of the international conference on mechanical engineering*, Dhaka, Bangladesh, 28–30 December 2005.
3. Munjal ML. *Acoustics of ducts and mufflers*. New York: John Wiley & Sons, p. 328, 1987.
4. COMSOL AB. *COMSOL multiphysics: user's manual*, Burlington, MA: COMSOL AB, 2008.
5. Beranek L. *Noise and vibration control*. Cambridge, MA: McGraw-Hill, 1988.
6. Tao Z and Seybert AF. *A review of current techniques for measuring muffler transmission loss* (03NVC-38). Warrendale, PA: Society of Automotive Engineers, Inc, 2003.