A STUDY ON THE ACOUSTIC PERFORMANCE OF A REACTIVE MUFFLER

Shemin Thomas Varkey[1], Lalu P. P[2], K. Balakrishnan[3]
[1]Post Graduate Student, Government Engineering College, Thrissur,
[2]Assistant Professor, Government Engineering College, Thrissur,
[3]Assistant Professor, Government Engineering College, Thrissur,
Email: shemin.thomas@gmail.com[1], lalujesus@yahoo.co.in[2], balangectcr@gmail.com[3]

Abstract

The tail pipe noise from an automotive muffler was studied experimentally and numerically. The transient acoustic characteristics of muffler were predicted using computational fluid dynamics. To validate the results of the simulation, the transient acoustic characteristics of the exhaust muffler were measured. It was found that the results of simulation are in very good agreement with the experimental results. According to these results, the differences were caused by the flow noises which was not considered in the simulation. Based on the theory of Computational Fluid Dynamics model, a simplified model that can provide an acceptable accuracy and save much execution time compared with the standard model was proposed for the optimization design to meet the demand of time to market.

Index Terms: Muffler, Transient Acoustic Performance, Tail Pipe Noise, Multi-Chamber.

I. INTRODUCTION

Exhaust mufflers are widely used to muffle the noise of an engine or the noises of other predominant sources in the vehicles. In general, the mufflers are designed to satisfy the following two requirements: (1) Very high noise attenuation performance, which is a very fundamental requirement of the muffler. An exhaust muffler should be able to muffle the frequency range of noise, especially the low frequency range, because it is very clear that most of the noise is limited to engine rotational frequency and its first few orders. (2) Minimum back pressure, the back pressure here represents the extra static pressure that is exerted by the muffler on the engine through the small restriction in the flow of the exhaust gases. This needs to be kept to a very minimum, because a high back pressure will results in the reduction of the volumetric efficiency and also the specific fuel consumption rate. These two design requirements are very important while designing a muffler and are often contradictory.

In the recent decades, the transfer matrix method, the finite element method (FEM), the boundary element method (BEM) or the computational fluid dynamics method (CFD) are usually employed to predict the noise of the tail pipe. The transfer matrix method (or four-pole theory) is the most common calculation method [1]. The four-pole theory method is based on the linear one dimensional wave propagation in the ducts and the four-pole theory formulation of the individual elements (such as: area discontinuities, ducts and the branches) in an analogy to the electric filter theory. However, this transfer matrix method has many disadvantages: (1) It should be used only under the assumption that it is a linear wave propagation; (2) It is also impossible to calculate the back pressure using this transfer matrix method; (3) This method cannot predict the transient acoustic characteristics of a muffler. In the literature [9–11], the boundary element method (BEM) was employed to calculate the acoustic attenuation performance of the reactive silencers and this approach is not confined to a plane wave treatment anymore. In the work done by Ji et al. [2-4], boundary element method predictions obtained 3-D analytical results for many muffler configurations which are in good agreement with the experiment values. In the work by Middelberg et al. [5], both the mean flow of noise and the acoustic performance of an expansion chamber muffler, along with various changes including extended inlet/outlet pipes and baffles, were successfully evaluated using 3-D computational fluid dynamics.

Combination of 3D computational fluid dynamics and 3D boundary element method were used in Yumex Corporation [6] to analyze the noise from a commercial muffler. Barbieri and Barbieri [12] used the finite element method to analyze the transient acoustic performance of a muffler. Even though the simulations of finite element method, boundary element method and computational fluid dynamics obtain very high accuracy results, they often limits the user to try other possible design alternates due to the large expense of time for each result.

In the psychoacoustics field [7, 8], loudness, fluctuation strength, sharpness and roughness are used to analyze the sound quality. These parameters along with the acceleration performance feeling [14, 15] should also be considered in the automotive industry. Therefore, it is very clear that the transient acoustic performance of a muffler should be evaluated in the design process. The purpose of the present paper is to study the transient acoustic performance of a typical muffler using CFD and experimental method, and then to develop a simplified model with an acceptable accuracy that meets the demand of the time to market in the optimization design.

2. FUNDAMENTAL THEORIES

2.1. Fundamental equations of the flow problem

The flow problem in the present research is analyzed and solved using 3D computational fluid dynamics method. The whole system is divided into many volumes. The scalar variables (pressure, internal energy, temperature, enthalpy, mass, density, species concentrations) are all assumed to be uniform over each of the volume. The vector variables (mass flux, mass fraction fluxes, and velocity) are calculated for each of the boundary. The flow model does involve the simultaneous solution of the continuity equation, energy equation and the momentum equation. These equations are solved only in one dimension. This means that all the quantities are averages across the flow direction.

Primary solution variables are density, mass flow, and total internal energy. The gas temperature in an exhaust system changes roughly with the distance from that of the engine. Therefore, the gas temperature in an exhaust system is not a constant. When we calculate the gas temperature, the heat exchange between that of flow components and gas and the heat exchange between that of the environment and the flow component should also be considered in the conservation of energy equation and in the enthalpy equation.

Continuity Equation:

$$\frac{dm}{dt} = \sum\nolimits_{\text{boundaries}} \dot{m} \qquad (1)$$

Energy Conservation Equation:

$$\frac{d(me)}{dt} = p\frac{dV}{dt} + \sum\nolimits_{boundaries} (\dot{m}H) - hA_s\,(T_{gas} - T_{wall}) \qquad (2)$$

Enthalpy:

$$\frac{d(\rho HV)}{dt} = V\frac{dp}{dt} + \sum\nolimits_{boundaries} (\dot{m}H) - hA_s\,(T_{gas} - T_{wall}) \qquad (3)$$

Momentum Conservation Equation:

$$\frac{d\dot{m}}{dt} = \frac{dpA + \sum boundaries\,(\dot{m}u) - 4C_f\frac{\rho u|u|}{2}\frac{dxA}{D} - C_p\frac{\rho u|u|}{2}A}{dx} \qquad (4)$$

where $\dot{m}$, m, V, p, ρ, A, A_s, e, H, h, T_{gas}, T_{wall}, u, C_f, C_p, D, dx, and dp are boundary mass flux into volume, mass of the volume, volume, pressure, gas density, flow area (cross sectional), heat transfer surface area, total internal energy (internal energy plus kinetic energy) per unit mass, total enthalpy, heat transfer coefficient, gas temperature, wall temperature, velocity at the boundary, skin friction coefficient, pressure loss coefficient, equivalent diameter, length of mass element in the flow direction (discretization length), pressure differential acting across dx, respectively.

2.2. Tail pipe noise

According to the equations (1)–(4), the back pressure and the gas flow velocity that at the exit of the tail pipe can be easily calculated. In the case of tail pipe noise, there is not an exact formula to calculate it. But in the case of a free field, the volume of the gas flow from that of the tail pipe is infinitesimal compared with that of the volume of the free field (infinite). The volume of the gas flow from that of the tail pipe is regarded as a variation in the tail pipe volume.

3. EXPERIMENTAL SETUP AND METHOD

Experiments were conducted separately for a simple muffler as well as for a multi-chamber muffler. Automobile noise frequency ranges from 700 Hz to 1300 Hz, mainly 1000 Hz. So, experiment and numerical simulation were done for this frequency range only. Simple muffler setup involved a single chamber with inlet and outlet pipes and microphones fixed inside both pipes for measuring the inlet sound as well as the outlet sound. Multiple chamber muffler setup involved a muffler with three chambers inside it along with a diffluent hole and perforated holes for maximum noise attenuation. Microphones were fixed on inlet pipe and outlet pipe as in the case of simple muffler setup. Decibel levels from the microphones were recorded and analyzed in both simple muffler and multi-chamber muffler setup.

4. NUMERICAL SETUP AND SIMULATION

In the present research, simulation of the tail pipe noise on simple and multi-chamber muffler are conducted using computational fluid dynamics in Ansys 16.0. A schematic diagram of the structure of multi-chamber muffler is shown in Fig. 1. The multi-chamber muffler consists of three pipes and three chambers which are separated by two set plates. To set up such a model of the multi-chamber muffler correctly, the function of each acoustic component of the multi-chamber muffler should be understood fully. There are several acoustic components in this multi-chamber muffler, including the diffluent hole, extended tube resonator, expansion chamber and Helmholtz Resonance chamber.

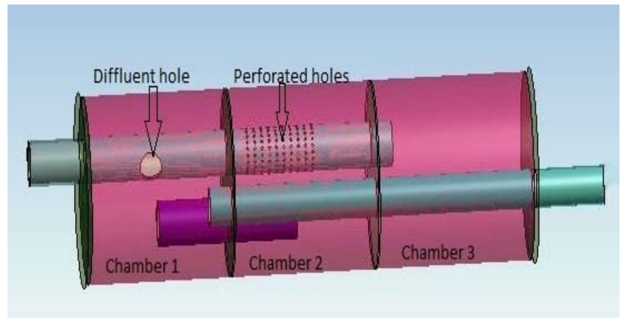

Fig 1: Schematic Diagram of structure of multi-chamber muffler

In Fig. 1, the diffluent hole which is a bigger hole on the inlet pipe that is located in the chamber 1 is designed in order to destroy the resonance condition of inlet pipe in its operating frequency range. The perforated holes on the inlet pipe are designed such as a flow stream guide in order to prevent the flow separation and the excessive turbulence that may occur at sharp discontinuities in the flow path, thus it is clear that the perforated pipe is used in order to reduce the generated flow noise and the pressure drops in the flow. The Helmholtz Resonance chamber is formed by a part of inlet pipe and the chamber 3 in order to reduce a specified frequency noise. This resonance chamber is acting as an effective acoustic filter and is designed in order to attenuate very low frequency noise. The Helmholtz Resonance chamber also reflect back the sound waves towards the sound source and this prevents sound waves from being transmitted through the pipe.

The expansion chamber in the chamber 2 is designed such a way in order to eliminate the booming noise. This expansion chamber reflects the sound waves by introducing a quick change in the cross-sectional area in the muffler region. The expansion chamber does not have the high attenuation as that of the Helmholtz resonator, but it does have a broadband frequency characteristic, that along with the pass bands when half of the sound wave length equals that of the cavity length. The expansion chamber also allows the sharp pressure pulses in the muffler to be smoothed out, and hence reducing the individual sound pulses at the outlet region.

The extended tube resonator that is being created when the area of the incoming sound waves expand or contract to a different size is designed in order to cancel the specified frequency noise in the muffler. In this multi-chamber muffler, the inner pipe along with the set plate forms two effective extended tube resonators. In this multi-chamber arrangement, incoming sound waves get reflected off by the set plate and also return in the opposite direction in order to effectively interfere with the incoming sound waves causing high attenuation. One of the important advantage of this particular component and this multi-chamber muffler is the good attenuation over a high frequency range. It is been noted that this setup performs little efficiently at low frequency range. The specifications of the simple muffler and multi-chamber muffler are listed in Table 1 and Table 2 respectively. It is assumed in the current simulation that the heat capacity of the environment is infinite, hence namely the temperature of the environment will not be subjected to any variation in the whole simulation setup.

Table 1: Specifications of Simple Muffler

Items	Value(mm)
Length of Inlet Pipe	378
Length of Muffler Body	623
Length of Outlet Pipe	500
Diameter of Inlet Pipe	47
Diameter of Outlet Pipe	40
Diameter of Muffler Body	200

Table 2: Specifications of Multi-Chamber Muffler

Items	Value(mm)
Length of Chamber 1	202
Length of Chamber 2	200
Length of Chamber 3	221
Length of Inlet Pipe	500
Diameter of Inlet Pipe	47
Length of Inner Pipe	190
Diameter of Inner Pipe	47
Length of Outlet Pipe	500
Diameter of Outlet Pipe	40
Number of Perforated holes	210
Diameter of Perforated holes	3
Length of Muffler Body	623
Diameter of Muffler Body	200
Diameter of Diffluent hole	30

5. RESULTS AND DISCUSSION

Experimental and simulation results of the simple muffler model are shown in Fig. 2. It can be found from the Fig. 2 that the experimental results are in good agreement with that of the simulation results of the simple muffler setup.

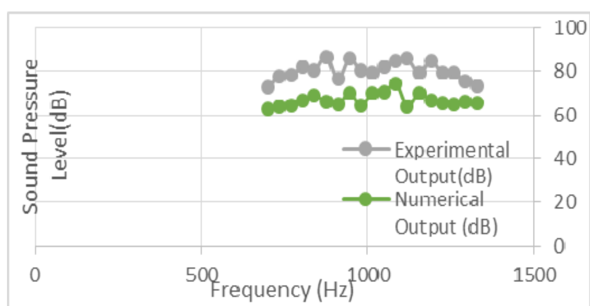

Fig. 2: Experimental results and simulation results of simple muffler setup

The sound reduction achieved by both experimental setup and numerical setup in simple muffler setup are shown in Fig. 3.

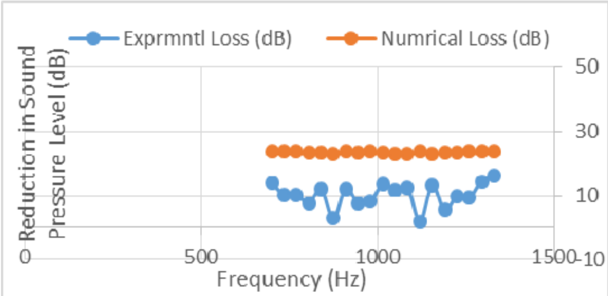

Fig. 3: Sound reduction achieved by both experimental and numerical setup in simple muffler setup

It is very clear from the results that the sound pressure level reduction obtained from numerical setup is higher in comparison to the experimental results of the simple muffler. Moreover, the decibel level loss in numerical setup in all frequency range shows only minimum variation comparing with experimental results. The experimental and numerical result differences are mainly caused by the flow noise. It should be noted that the flow noise was not considered in the numerical simulation.

Similar to the simple muffler setup, numerical simulation was done for multi-chamber setup and results are shown in Fig. 4 and Fig. 5.

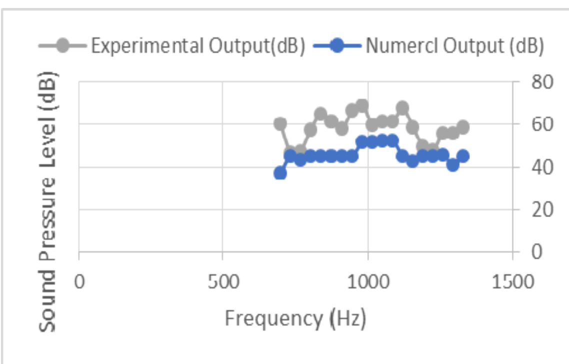

Fig. 4: Experimental and simulation results of multi-chamber muffler setup

In multi-chamber muffler setup, when the sound waves get passed through the succeeding silencing components

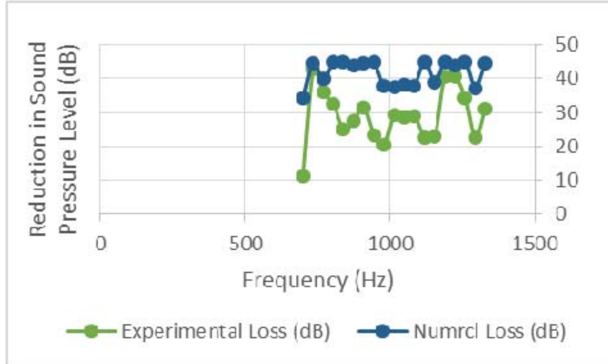

Fig. 5: Sound reduction achieved by experimental and numerical setup in multi-chamber muffler setup

and when finally exhausts as a relatively high velocity jet, a considerable level of turbulence is generated, which is the reason for broadband self-generated noise. Due to this, the silencer itself behaves like a noise source. The multiple Chambers and perforated reduce this self-generated noise to a good extent.

It is very clear from the results that the sound pressure level reduction obtained from numerical setup is higher in comparison to the experimental results of the multi-chamber muffler. The experimental and numerical result differences are mainly caused by the flow noise. It should be noted that the flow noise was not considered in the numerical simulation.

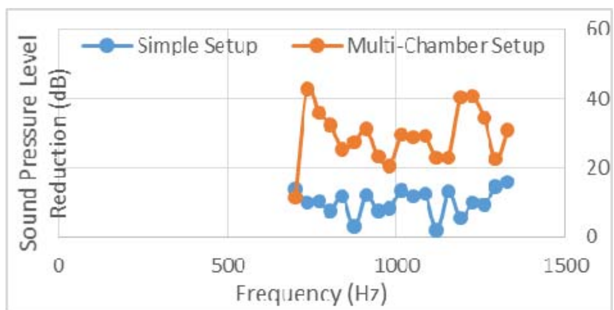

Fig. 6: Comparison of experimental values of Simple Muffler and Multi-Chamber Muffler

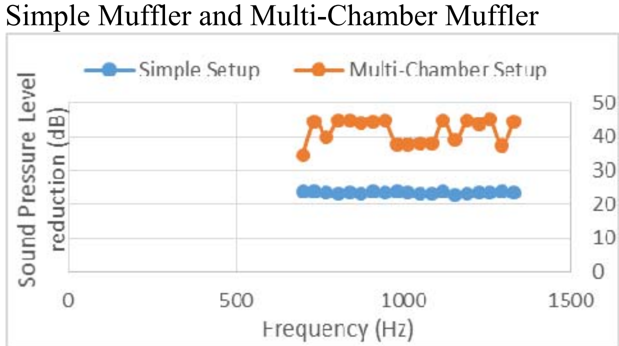

Fig. 7: Comparison of Numerical Simulation values of Simple Muffler and Multi-Chamber Muffler

Comparing the noise reduction achieved by both cases of simple muffler setup and multi-chamber muffler setup from Fig. 6 and Fig. 7, , it is very clear that there has been an effective noise reduction in all frequency range in multi-chamber muffler setup.

6. CONCLUSIONS

The tail pipe noise of an exhaust reactive muffler was obtained using experimental setup and its results were validated using 3D computational fluid dynamics model simulation. It was found that the results obtained from the experiments al setup is in good agreement with the values obtained from the simulation, although there is a slight difference between the values. These differences are caused due to the effect of flow noise which was not considered into effect in numerical simulation. Experiment and numerical analysis was done on a simple muffler setup for comparing with multi-chamber muffler and found out that multi-chamber muffler holds high effectiveness in all frequency range.

7. ACKNOWLEDGEMENTS

The authors would like to thank the Department of Mechanical Engineering of the Government Engineering College Trissur for providing the computational facilities for our work.

REFERENCES

[1] Munjal ML. Acoustics of ducts and mufflers. John Wiley & Sons; 1987.

[2] Selamet A, Ji ZL, Radavich PM. Acoustic attenuation performance of circular expansion chambers with offset inlet/outlet: II. Comparison with computational

[3] and experimental studies. J Sound Vib 1998;213:619–41.

[4] Selamet A, Ji ZL. Acoustic attenuation performance of circular expansion chambers with extended inlet/outlet. J Sound Vib 1999;223:197–212.

[5] Ji ZL, Selamet A. Boundary element analysis of three-pass perforated duct mufflers. Noise Control Eng J 2000;48:151–6.

[6] Middelberg JM, Barber TJ, Leong SS, Byrne KP, Leonard E. CFD analysis of the acoustic and mean flow performance of simple expansion chamber mufflers. In: Proceedings of the 2004 ASME international mechanical engineering congress and exposition, Anaheim, California, USA, 13–20 November 2004.

[7] Yasuda T, Nakagawa N, Wu Chaoqun. Computational simulations of automotive muffler radiation noise. In: Proceedings of inter-noise 2008, Shanghai, China, 26–29 October 2008.

[8] Fastl H, Zwicker E. Psychoacoustic: facts and models. Springer; 2006.

[9] Blauert J. Communication acoustic. Springer; 2005.

[10] Seybert AF, Cheng CYR. Application of the boundary element method to acoustic cavity response and muffler analysis. J Vib Acoust Stress Reliab Des 1987;109:15–21.

[11] Cheng CYR, Seybert AF. A multidomain boundary element solution for silencer and muffler performance prediction. J Sound Vib 1991;151:119–29.

[12] Ji ZL, Ma Q, Zhang ZH. Application of the boundary element method to predicting acoustic performance of expansion chamber mufflers with mean flow. J Sound Vib 1994;173:57–71.

[13] Barbieri R, Barbieri N. Finite element acoustic simulation based shape optimization of a muffler. Appl Acoust 2006;67:346–57.

[14] Crocker MJ. Handbook of noise and vibration control. Wiley; 2007.

[15] Fukuhara C, Kamura T, Suetomi T. Subjective evaluation of acceleration performance feeling with driving simulator. Trans Soc Automot Eng Jpn 2004;35(1):227–32.

[16] Fukuhara C, Kamura T, Suetomi T. A method of predicting acceleration performance feeling. Soc Automot Eng Jpn 2007;61(6):97–103.