Numerical Investigation of Back Pressure and Acoustic Attenuation Performance of Two and Three Chamber Exhaust Muffler

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Abstract. Two and Three chamber expansion muffler are commonly used in light category vehicle for meeting the noise level legislation requirement. The Helmholtz resonator and expansion chamber are used in exhaust muffler to attenuate the noise in low and middle frequency range. In this work, the effect of expansion chamber size on back pressure is determined using CFD and the same geometry were used to study the acoustic attenuation performance using FEA in the frequency range of 0 to 3000 Hz. Regarding the size of individual expansion chamber there is about 17% increase and decrease in size from base model geometry were considered for performance comparison in both two and three chamber type of muffler. It is found that minimum back pressure of about 241 Pa is observed for 2 chamber base model (2C-B) with slightly higher turbulent kinetic energy value of 77.53 J/Kg. The sound pressure level (SPL) at outlet pipe is less for 3 chamber mufflers and the obtained value is less than 120 dB for most of the frequency range of 0 Hz to 3000 Hz. Similarly transmission loss (TL) is higher for 3 chamber mufflers and the value remains same for 0 to 1600 Hz frequency range and peak fluctuation occurs at higher frequency. The obtained results shows that 3C-A model is having higher noise attenuation performance with back pressure value within acceptable limit.

1. Introduction
The engine exhaust noise is one of the main source of noise from a vehicle. The exhaust system uses muffler to attenuate the noise level within the required legislative limit [1]. The back pressure effect due to muffler design is another important parameter to be considered along with noise attenuation to improve the engine performance [2], [3]. The muffler design was modified in 1) increase the volume of muffler from 5 to 12 times the engine displacement [4] 2) optimizing the muffler internal geometry [5] 3) using absorbing material for attenuating higher frequency noises [6], [7]. Mufflers are differentiated as dissipative or reactive according to suppression of acoustic energy by dissipation through absorbing materials or reflection of waves through expansion chamber or Helmholtz resonator. Expansion chamber muffler has been used widely in exhaust silencer and air-conditioning HVAC system for attenuating low and mid-range frequency [8], [9].

The muffler geometry consist of inlet pipe from exhaust manifold, expansion chamber or Helmholtz resonator chamber, absorbing material and tail pipe. The inlet pipe was perforated and extended inside the muffler for reducing the flow discontinuity and turbulent kinetic energy generation, which decreases the flow induced noise and back pressure [3], [4]. Helmholtz resonator and expansion chamber uses destructive interference to attenuate the noise. Helmholtz resonator is used to reduce the low frequency
noise from engine and it has higher attenuation characteristics [10]. Expansion chamber is used to reduce booming noise [11] and it has broadband frequency characteristics with pass band, when cavity length is half of the acoustic wavelength [4]. The sudden change in expansion chamber cross sectional area creates reflective waves and its attenuation characteristics is less than Helmholtz resonator. The pressure drop reduces for interconnected multiple expansion chamber mufflers compared to single expansion chamber muffler [12], [13]. Presence of baffles in muffler have improved the transmission loss effect by 50% [14], [15]. Reducing the baffles shifts the peak transmission loss to high frequency region. At higher baffle size the sound noise is reduced for low and intermediate frequency range and for smaller baffle size higher frequency sound waves was attenuated effectively by increasing the reflection count [12]. The absorbing components uses absorbing materials to convert sound energy to heat energy [16]. It can be used to attenuate high frequency noises and further it is used to reduce flow noise, radiation noise and exhaust crackle noise [4].

Sound pressure level (SPL) at inlet and outlet, Insertion loss (IL) & Transmission loss (TL) are some of the important parameter for analysing the acoustic attenuation performance of muffler. The performance parameters of muffler are generally determined using Transfer matrix method, Boundary element method (BEM) and Finite element method (FEM) [10], [13], [17]. Transfer matrix method is based on the assumption of linear 1D plane wave propagation. The effect of multi-dimensional wave inside the muffler is excluded in plane wave method and the analytical result matches well with experimental results for low frequency and deviates for the higher frequency range [18]. BEM and FEM can be used to analyse the attenuation performance of complicated silencer having multi-dimensional wave characteristics with results closely matching to experimental results [4], [18]. The restriction in computational capacity restrict the application of BEM and FEM on using more complicated models in acoustic analysis [17].

The function of muffler is to reduce the exhaust noise without increasing the backpressure effect. Computational fluid dynamics (CFD) tool is used to optimize the muffler geometry for reducing the backpressure and flow noises. This method developed as an alternative for experimental analysis by researchers due to its improved accuracy and adaptability with reduced operating cost [3]. Numerous work has been carried out in past to determine the flow behaviour inside different modified muffler using CFD to determine the back pressure and turbulent kinetic energy generation [3], [9], [13]. Flow induced noise was created due to flow discontinuity and turbulence flow inside muffler and it was more in high frequency range. The SPL (dB) obtained using FEM matches well with experimental results for low frequency and it deviates at higher frequency due to not considering flow noise in the analysis [1].

In this work, noise attenuation performance and backpressure effect of double and triple chamber muffler is compared using FEM and CFD analysis. The expansion chamber spacing is modified with respect to base model dimension for both muffler and the performance parameters are compared. The flow behaviour is simulated using Ansys Fluent 16.2 CFD package and TL is obtained separately without considering the mean flow effect using Harmonic Response Acoustic module of Ansys Workbench 16.2.

![Figure 1. (a) 2 chamber and (b) 3 chamber wire frame CAD model](image-url)
2. Models

2.1. CAD Model
In this work, dimensions of standard two and three chamber muffler used is taken as base design [9]. The 17% increase and decrease in expansion chamber volume is taken as modification for comparative study. 3D wire frame model of two and three chamber base model geometry is shown in figure 1. The dimensions of the model considered for the analysis are given in table 1 and table 2.

| SI. No | Model       | Chamber dimension in mm |
|--------|-------------|-------------------------|
|        |             | P1 | P2     |
| 1      | 2C-A        | 150| 70    |
| 2      | 2C-B (Base model) | 160| 60    |
| 3      | 2C-C        | 170| 50    |

| SI. No | Model       | Chamber dimension in mm |
|--------|-------------|-------------------------|
|        |             | Q1 | Q2 | Q3 |
| 3      | 3C-A        | 95 | 43 | 180 |
| 2      | 3C-B (Base model) | 105| 43 | 170 |
| 3      | 3C-C        | 115| 43 | 160 |

2.2. Meshing
Separate meshing is used for CFD and FEM analysis. Tetrahedral unstructured mesh is considered for discretizing the geometry due to flexibility and accurate for complex geometry. The element size of 4 mm and 8 mm are used for CFD and FEM acoustic analysis [13]. In order to capture the wall effect and turbulent kinetic energy generation, mesh near walls are made denser to capture the effects accurately. The generated mesh model is shown in figure 2. The mesh for CFD analysis is denser than acoustic FEM analysis to obtain accurate result. The general condition to be meet by acoustic mesh is shown in equation (1).

\[ L \leq \frac{c}{6f_{\text{max}}} \]  

Where c is the speed of sound and \( f_{\text{max}} \) is maximum frequency of interest. The maximum frequency in most conditions will be 3000 Hz, hence the accuracy of acoustic computation can be ensured.

Figure 2. Mesh model of (a) 2 chamber and (b) 3 chamber muffler
2.3. Numerical Models

The effect of backpressure due to muffler is studied numerically using CFD. The governing equation used in this work are continuity and Reynolds Average Navier-Stokes equation for fluid flow analysis. The flow is considered incompressible and the effect of gravity and other body forces are neglected. The flow medium is consider as air with density and viscosity taken as 0.5508 kg/m³ and 3.821e-5 (pa.s) [9].

Pressure based solver with simple pressure-velocity coupling is used [18]. Turbulent flow is generated inside muffler due to its geometrical complexity and large number of turbulent models are available with respect to flow behaviour and application. Here realizable $k – \epsilon$ turbulent model is used due to its ability to capture turbulence kinetic energy for flows involving rotation, boundary layers under strong adverse pressure gradients, separation, and recirculation [19], [20]. The Continuity equation (2), Momentum equation (3) and Turbulence Transport equation (4) & (5) used to obtain fluid flow parameters in considering above conditions are given below [19].

$$\frac{\partial (\rho u)}{\partial x} + \frac{\partial (\rho v)}{\partial y} + \frac{\partial (\rho w)}{\partial z} = 0$$  \hspace{1cm} (2)

$$\frac{\partial (\rho u u_j)}{\partial x} = -\frac{\partial p}{\partial x_i} + \frac{\partial}{\partial x_j} \left( \mu \left( \frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} \right) \right) - \frac{\partial (\rho u_i u_j)}{\partial x_j}$$  \hspace{1cm} (3)

$$\frac{\partial (\rho k u_j)}{\partial x_j} = \frac{\partial}{\partial x_j} \left( \mu + \frac{\mu_t}{\sigma_k} \frac{\partial k}{\partial x_j} \right) + G_k + G_b - \rho \epsilon - Y_M + S_k$$  \hspace{1cm} (4)

$$\frac{\partial (\rho \epsilon u_j)}{\partial x_j} = \frac{\partial}{\partial x_j} \left( \mu + \frac{\mu_t}{\sigma_\epsilon} \frac{\partial \epsilon}{\partial x_j} \right) + \rho C_1 \epsilon \rho - \rho C_2 \frac{\epsilon^2}{k + \sqrt{\kappa \epsilon}} + C_1 \frac{\epsilon}{k} C_3 \epsilon G_b + S_\epsilon$$  \hspace{1cm} (5)

The turbulent kinetic energy $G_k$ due to velocity gradient is used to determine the turbulent generation in the flow due to geometrical discontinuity. The $Y_M$ represents the dissipration rate of turbulent eddies. $S_k$ and $S_\epsilon$ indicates the source term defined externally [19].

The sound pressure level (dB) at silencer outlet and Transmission Loss (TL), which requires pressure at inlet and outlet of silencer is determined using Helmholtz equation (6) [19]. FEM numerical procedure is used to discretise the equation (6) to obtain the pressure values.

$$\nabla \left( \frac{1}{\rho_o} \nabla p \right) + \frac{k^2 p}{\rho_o} = 0$$  \hspace{1cm} (6)

Where $p$ is the acoustic pressure, $k = 2\pi f / C_o$ is the wave number, $\rho_o$ is the density and $C_o$ is the speed of sound. The flow properties used for CFD analysis is repeated here in FEA analysis and speed of sound is taken as 340 m/s.

The Galerkin’s weighted residual method is used to obtain the finite element formulation of equation (6).

$$([M] - k^2 [P]) \{p\} = -j \rho \omega \{F\}$$  \hspace{1cm} (7)

Where $[M]$ and $[P]$ are inertia and stiffness matrix of element, $\{p\}$ is pressure vector at the nodes, $\rho$ is the density of flowing medium and $\{F\}$ is force vector at the nodes.

The sound pressure calculated in equation (7) is inserted in equation (8) to determine TL of silencer.

$$TL = 20 \log \left( \frac{p_{inc}}{p_{tra}} \right)$$  \hspace{1cm} (8)
Figure 3. Pressure point for TL calculation

$P_{\text{inc}}$ and $P_{\text{tra}}$ is the acoustic pressure of incident wave and transmitted wave at the inlet and outlet of the silencer. The FEM analysis results gives the pressure at inlet $P_{\text{in}}$ and outlet $P_{\text{out}}$, which contains $P_{\text{ref}}$ to be eliminated for obtaining the required pressure values. By assuming plane wave conditions the incident wave pressure can be obtained by equation (9).

$$P_{\text{inc}} = \frac{P_{\text{in}} + \rho c}{2}$$  \hspace{1cm} (9)

The transmitted wave pressure is obtained by assuming anechoic end conditions i.e. $P_{\text{ref}} = 0$, hence

$$P_{\text{tra}} = P_{\text{out}}$$  \hspace{1cm} (10)

Figure 4. Velocity contour of (a) 2C-A (b) 2C-B (c) 2C-C (d) 3C-A (e) 3C-B (f) 3C-C model muffler
3. Result and discussion

3.1. Effect of geometry on velocity, back pressure and turbulent kinetic energy generation

CFD analysis is performed for the muffler geometries given in table 1 & table 2. The velocity contour for 2 and 3 chamber muffler is shown in figure 4. The velocity is more at inlet and outlet pipe of both type of muffler compared to chamber velocity to satisfy mass conservation. In all 2 chamber muffler model velocity peak is obtained in the inlet pipe orifice hole. Higher velocity in orifice hole is obtained for 2C-A model compared to 2C-B & 2C-C due to higher pressure at the end of inlet pipe which forces the gas to flow through orifice at higher velocity.

The velocity at the entrance of exit pipe (tail pipe) is more for 2C-B & 2C-C model to overcome the velocity loss inside the expansion chamber. In 3 chamber muffler the velocity distribution remains same for all the models with peak value obtained for 3C-B base model. Disturbed flow pattern is found in Q3 chamber of 3C-B model, which increases the flow loss in the system.

![Figure 4. Turbulent Kinetic Energy contour for (a) 2C-A (b) 2C-B (c) 2C-C (d) 3C-A (e) 3C-B (f) 3C-C model muffler](image)

The turbulent kinetic energy generation is more at orifice hole for all 2 chamber models and P2 expansion chamber of 2C-A model as shown in figure 5, which increases the flow induced noise [2]. The minimum value of 77.53 J/kg occurs in 2C-B model and for 3 chamber muffler the turbulent kinetic energy generation is more at Q1 and Q3 chamber of 3C-A & 3C-C model and least value of 66.93 J/kg is obtained for 3C-B base model with little disturbance occurs at Q2 chamber as shown in figure 5 (e).

In 2 chamber muffler, back pressure created at inlet is changing with respect to P2 expansion chamber volume. Minimum back pressure at inlet is obtained for 2C-B model as shown in figure 6 (c) and...
maximum is obtained for 2C-A model due to the formation of recirculating flow region in P2 chamber. In 3 chamber muffler least back pressure occurs for 3C-B base model as shown in figure 6 (e) and second least back pressure value occurs for 3C-C model with 8% increase in value compared to 3C-B base model.

![Figure 6](image_url)

**Figure 6.** Pressure contour for (a) 2C-A (b) 2C-B (c) 2C-C (d) 3C-A (e) 3C-B (f) 3C-C model muffler

### 3.2. Effect of geometry on noise attenuation

The noise attenuation performance of the different model is analysed using FEA analysis with inlet velocity taken as 13 m/s. Sound pressure level (SPL) in dB at inlet and outlet for 2 and 3 chamber muffler is shown in figure 7. The SPL at inlet varies according to the pressure developed in the muffler inlet pipe and in outlet pipe it depends on the attenuation performance of the geometry. The SPL pattern at inlet for 2 chamber muffler is approximately same for 2C-B and 2C-C model and for 2C-A model the change in SPL pattern at the inlet occurs due to development of higher static pressure in the inlet pipe.

The SPL at the outlet for 2C-A model is reduced at 400 to 900 Hz frequency compared to SPL inlet and nearly equal to SPL inlet for other frequency range. SPL at outlet for 2C-B and 2C-C model is less at most of low, middle and higher frequency compared to SPL inlet, hence higher attenuation is obtained for these models. The less noise is obtained for 2C-C model at higher frequency.

In 3 chamber muffler the highest value SPL value is same for all the models and SPL distribution pattern with respect to frequency is different according to the change in Q1 and Q3 chamber volume. SPL at outlet is same till 1600 Hz for all the 3 chamber models and fluctuation in SPL occurs at higher frequencies. Minimum SPL at outlet occurs for 3C-A model at higher frequency range. The noise at outlet is within the required regulation limit of 120 dB in most of the frequency range [21].
The Transmission Loss (TL) for the 2 and 3 chamber muffler is shown in figure 8. In the frequency range of 0 Hz to 2000 Hz the TL curve remains same for 2C-A, 2C-B and 2C-C model and highest attenuation of 50 dB is obtained for 2C-C model at 920 Hz and 53 dB at 2760 Hz. The TL curve shift towards higher frequency for the change in chamber volume. Fluctuation in TL occurs at higher frequency which can be reduced by using absorbing materials. Resonance in 2 chamber muffler occurs at the frequency of 30 Hz, 1200 Hz and 1800 Hz.

![Figure 7](image)

**Figure 7.** Sound pressure level at inlet and outlet for (a) 2C-A (b) 2C-B (c) 2C-C (d) 3C-A (e) 3C-B (f) 3C-C model muffler

TL curve for all the 3 chamber muffler is remains same for the frequency range of 0 Hz to 1600 Hz and peak attenuation of 70 dB is obtained at the frequency of 1300 Hz. The maximum range of 60 - 70 dB is maintained at the frequency range of 900 to 1500 Hz which is not there in 2 chamber muffler. In 3
chamber muffler resonance in TL occurs at the frequency of 30 Hz, 150 Hz and 2040 Hz. In higher frequency range the fluctuated TL is obtained for 3 chamber muffler same as 2 chamber muffler and peak value of 99 dB is obtained for 3C-A model at 2700 Hz frequency.

![Graph](image_url)

**Figure 8.** Transmission Loss (TL) for (a) 2 chamber models (b) 3 chamber models

4. **Conclusion**

The back pressure effect and acoustic attenuation performance is compared for 2 and 3 chamber muffler using CFD and FEM tools. Three different 2 and 3 chamber muffler 2C-A, 2C-B & 2C-C and 3C-A, 3C-B & 3C-C with dissimilar expansion chamber volume and by maintaining overall volume same is considered for analysis. The flow velocity, turbulent kinetic energy and back pressure differs according to the flow behaviour in the chambers. In average the backpressure of 2 and 3 chamber muffler is well below the required back pressure limit. In comparison, back pressure at inlet pipe is low for 2 chamber 2C-B model compared to 3 chamber models and turbulent kinetic energy is low for 3C-B model.

In considering acoustic attenuation performance (SPL outlet) 3 chamber muffler is having 25% less outlet pipe noise level than 2 chamber muffler and in most of the frequency range the SPL at outlet is well below the regulatory noise limit. The transmission loss for 3 chamber muffler is higher compared to 2 chamber muffler with gradual increase in low and middle frequency range whereas in 2 chamber sudden peak in transmission loss occurs at certain frequencies.

The noise level for 2 chamber muffler for the considered engine analysis is largely higher than the regulatory limit, but the less back pressure effect can be useful in racing vehicle were the noise level is
not a mandatory condition. In higher frequency range fluctuated transmission loss is obtained for both 2 and 3 chamber which can be reduced by using aborning materials at the outlet pipe. In overall it can be concluded that 3C-A model is having better acoustic attenuation characteristic with back pressure and turbulent kinetic energy within acceptable limit.

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