Noise Transmission Loss Maximization in Absorptive Muffler with Shells

Maryam Alinaghi

Submitted to the
Institute of Graduate Studies and Research
in partial fulfillment of the requirements for the Degree of

Master of Science
in
Mechanical Engineering

Eastern Mediterranean University
June 2015
Gazimağusa, North Cyprus

Approval of the Institute of Graduate Studies and Research

Prof. Dr. Serhan Çiftçioğlu
Acting Director

I certify that this thesis satisfies the requirements as a thesis for the degree of Master of Science in Mechanical Engineering.

Prof. Dr. Uğur Atikol
Chair, Department of Mechanical Engineering

We certify that we have read this thesis and that in our opinion it is fully adequate in scope and quality as a thesis for the degree of Master of Science in Mechanical Engineering.

Asst. Prof. Dr. Mostafa Ranjbar
Supervisor

Examining Committee

1. Assoc. Prof. Dr. Hasan Hacışevki
2. Asst. Prof. Dr. Mostafa Ranjbar
3. Asst. Prof. Dr. Neriman Ozada

ABSTRACT

The reduction of the emitted noise pollution from the exhaust system of engines is a real challenge for various industries. At this regard, mufflers have been used to reduce the transmitted noise from the engine of vehicles into the surrounding environment. Mufflers are designed to reflect the sound waves produced by the engine in such a way that they partially cancel themselves out. Noise transmission loss performance in muffler depends on its geometry. Therefore, maximization of noise transmission loss in mufflers using shape modification concept is an important research area. In this M.Sc. study, maximization of noise transmission in mufflers is studied and investigated. A model is developed to present the absorptive muffler with shell. The muffler structure and its sound absorbing layer are modeled using shells elements. This model analyzes the muffler structure which has effects on the transmission loss (TL). The results are compared to a model without any absorbing layer. It indicates that the thickness and material type of absorbing layer have distinctive effects on the amount of noise transmission loss of muffler over a wide frequency range**.**

Keywords: Absorptive muffler, noise transmission loss, sound absorbing material, shell

ÖZ

Motorların egzoz sisteminden yayılan gürültü kirliliğinin azaltılması, çeşitli endüstriler için gerçek bir meydan okumadır. Bu bağlamda içinde susturucular çevredeki ortama araç motoru iletilen gürültüyü azaltmak için kullanılmıştır. Susturucuları kısmen kendini iptal şekilde motor tarafından üretilen ses dalgalarının yansıtacak şekilde tasarlanmıştır. Susturucuda gürültü iletim kaybı performansı kendi geometrisine bağlıdır. Bu nedenle, şekil değiştirme kavramını kullanarak susturucu gürültü iletim kaybı maksimizasyonu önemli bir araştırma alanıdır. Bu Yüksek Lisans içinde Tez araştırması , susturucular gürültü iletim maksimizasyonu okudu ve incelenmiştir. Bir model kabuğu ile emme susturucu sunmak için geliştirilmiştir. Susturucu yapısı ve ses emici katmanı kabukları elemanları kullanılarak modellenmiştir. Bu model iletim kaybı (TL) üzerinde etkileri vardır susturucu yapısını analiz eder. Sonuç, her türlü emici tabakası olmayan bir model ile karşılaştırılır. Bu katman emici kalınlığı ve malzeme tipi geniş bir frekans aralığında susturucu gürültü iletim kaybının miktarına ayırt edici etkilere sahip olduğunu göstermektedir.

Anahtar Kelimeler: Emici susturucu , gürültü iletim kaybı , ses emici malzeme , kabuki.

To my family

ACKNOWLEDGMENT

I would like to thank Asst. Prof. Dr. Mostafa Ranjbar for his invaluable supervision and guidance in the preparation of this study. Without his continuous support, all my efforts could have been short sighted. My parents, the most precious people in my life, were all the time supporting me and without their energy, I could not be able to achieve this success. I would also want to thank from jury members for their invested time regarding the evaluation of my thesis. And the last but not the least, I want to thank all my dear friends who were besides me all through this way. I'm very much pleased to have you all. Finally, I am solely accepting the full responsibly over the accuracy, the content and every issues of this thesis.

TABEL OF CONTENTS

LIST OF TABELS

LIST OF FIGURES

Chapter 1

INTRODUCTION

Mufflers are devices which attenuate the transmitted noise via them. They have designed to reduce the produced noise in their inlets by various methods, e.g. passive or active approaches. Actually, active mufflers are still not ready for mass production. Therefore, the attention of industry is focused on passive mufflers which use either reflection or absorption methods to reduce the energy of transmitted noise.

Internal combustion engine is a major source of noise pollution. These engines are used for various purposes such as, in power plants, automobiles, locomotives, and in various manufacturing machineries. Noise pollution created by engines becomes a vital concern when used in residential areas or areas where noise creates hazard.

Generally, noise level of more than 80 dB is injurious for human being. The main sources of noise in an engine are the exhaust noise and the noise produced due to friction of various parts of the engine. The exhaust noise is the most dominant. To reduce this noise, various kinds of mufflers are usually used. The level of exhaust noise reduction depends upon the construction and the working procedure of mufflers.

Engine makers have been making mufflers for more than 100 years. As the name implies, the primary purpose of the muffler is to reduce or muffle the noise emitted by the internal combustion engine.

Muffler technology has not changed very much over the past 100 years. The exhaust is passed through a series of chambers in reactive type mufflers or straight through a perforated pipe wrapped with sound deadening material in an absorptive type muffler. Both types have strengths and weaknesses.

Figure 1 shows the application of muffler in an automobile for the reduction of produced noise by the engine of car.

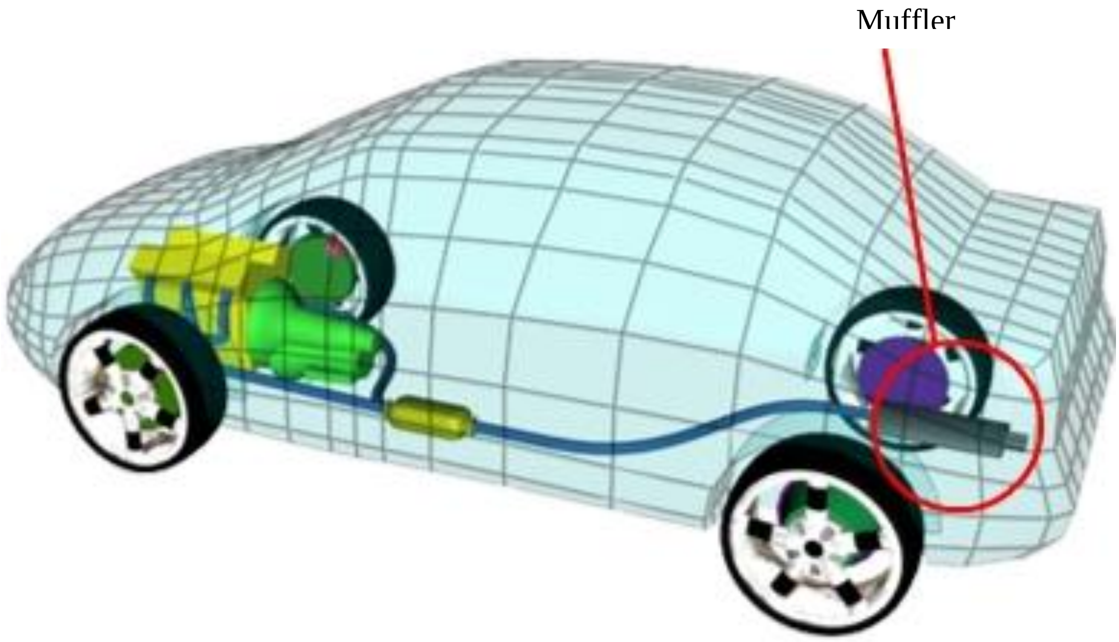

Figure 1. Application of muffler in an automobile for the reduction of produced noise by the engine of car [1]

Generally, reflective mufflers are attenuating the noise by reflecting and interfering inside a chamber while absorptive mufflers are dissipating the acoustic energy into heat through use of porous or absorptive materials.

Reactive mufflers can be mainly used for low frequency ranges while absorptive mufflers should be used for mid-range to high frequency ranges, i.e. more than 500 Hz, with little back pressure [2].

The reactive type muffler is usually restrictive and prevents even the good engine sounds from coming through, but does a good job of reducing noise. On the other hand, most absorptive type mufflers are less restrictive, but allow too much engine noise to come through. Regardless of the packing material, absorptive type mufflers tend to get noisier with age. [3]

Figure 2 shows a reactive (dissipative) muffler for automotive application. Located inside the muffler is a set of tubes. These tubes are designed to create reflected waves that interfere with each other or cancel each other out. Take a look at the inside of this muffler: The exhaust gases and the sound waves enter through the center tube. They bounce off the back wall of the muffler and are reflected through a hole into the main body of the muffler. They pass through a set of holes into another chamber, where they turn and go out the last pipe and leave the muffler. In this process, the intake energy of sound is dissipated.

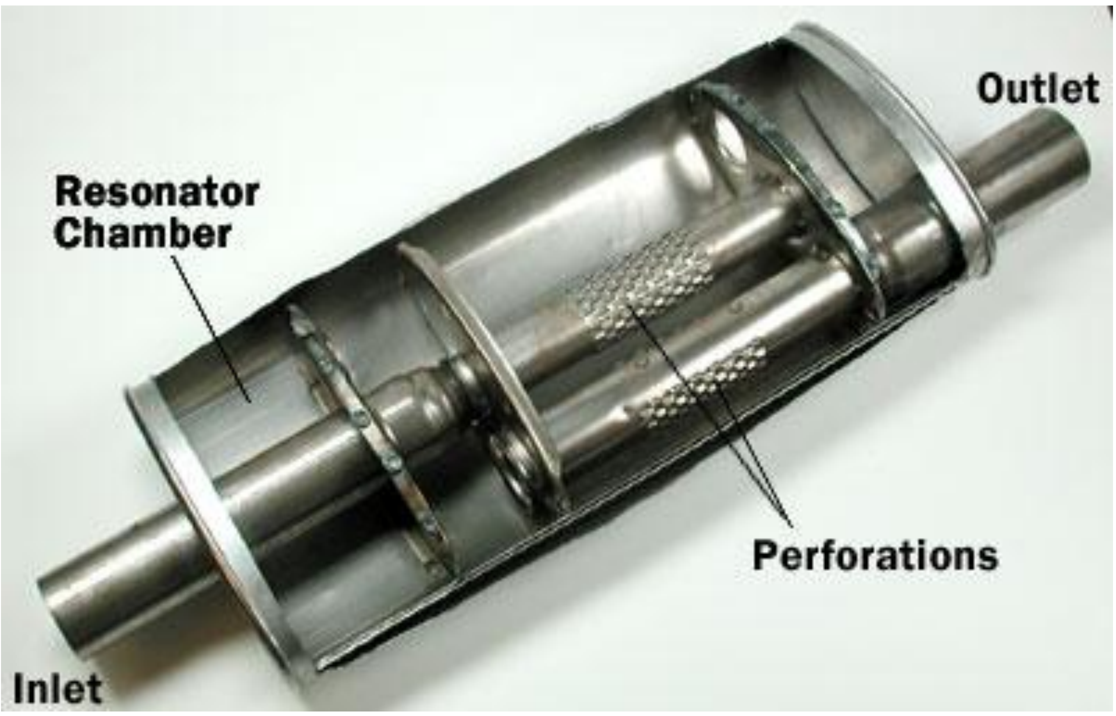

Figure 2. A typical reactive muffler for automotive application [1]

The absorptive muffler is the classic dissipative design, deriving its noise control properties from the basic fact that noise energy is effectively “absorbed” by various types of fibrous packing materials. That is, as the sound waves pass through the spaces between the tightly packed, small diameter fibers of the absorptive material, the resulting viscous friction dissipates the sound energy as small amounts of heat.

Figure 3 shows an absorptive muffler for automotive applications. This represents the most typical absorptive mufflers for cars. The engine exhaust gases are passing through perforated interface to the absorbing layer; hence its energy is being reduced.

Figure 3. A typical muffler with absorptive layer for automotive application [1]

Absorptive mufflers are highly effective on high-frequency noise (1000-8000 Hz) [3]. At frequencies above and below this range, attenuation performance progressively diminishes with common absorptive materials unless special design considerations are implemented.

Since noise is absorbed by the acoustic packing media, absorptive mufflers generally employ straight-through or annular internal designs, which impose very little restrictions on air flow.

Typically, the greater the ratio of packing surface area to flow area, the greater is attenuation capability of the silencer. Many different packing materials can be used in absorptive silencers and are chosen for use based on varying absorptive performance, price, temperature and corrosion characteristics.

The effect of the thickness of absorptive material and spacing play an important role in sound attenuation. The attenuation increases sharply at high frequencies as the spacing is narrowed. Better performance at lower frequency is obtained as the thickness of the absorbing material is increased.

In order to attenuate high frequency noise, a metal tube surrounded by acoustical-quality glass wool inside the muffler outer containment shell has been used here. The sides of the tube are perforated that permit sound waves impinge on the absorbing materials.

This thesis is investigating the performance of absorptive mufflers for low-frequency range applications up to 4000 Hz. Various types of absorptive materials will be considered. The level of reduction in the transmitted noise will be investigated and reported.

In the following sections, the literature review, modelling of noise propagation in circular ducts and calculation of noise in mufflers will be presented. Finally, the conclusion and some comments for the future work will be presented.

Chapter 2

LITERATURE REVIEW

Mufflers have been developed over the last century based on electro- acoustic analogies and experimental trial and error. Many years ago Stewart used electro – acoustic analogies in deriving the basic theory and design of acoustic filters [4].

Later Davis et al. published results of a systematic study on mufflers [5]. They used travelling wave solutions of the one-dimensional wave equation and the assumption that the acoustic pressure and acoustic volume velocity are continuous at changes in cross sectional area.

An important step forward in the analysis of the acoustical performance of mufflers is the application of two- port network theory with use of four –pole parameters. Igarashi and his colleagues calculated the transmission characteristics of mufflers using equivalent electrical circuits [6].

Parrot later published results for the certain basic elements such as area expansions and contractions. Sreenath and Dr. Munjal gave expression for the attenuation of mufflers using the transfer matrix approach [7].

The expression they developed was based on the velocity ration concept. Later, Dr. Mujal modified this approach to include the convective effects due to flow [8].

Young and Crocker used the finite element method to predict four-pole parameters and then the transmission loss of complex shaped mufflers for the case of no flow [9].

A generalized scheme for analysis of multifarious commercially used mufflers was proposed by Panigrahi et al. [10]. They explained that the commercial automotive mufflers are often too complex to be broken into a cascade of one dimensional element with predetermined transfer matrices. The one-dimensional (1-D) scheme presented in that paper was based on an algorithm that uses user-friendly visual volume elements along with the theory of transfer matrix based muffler analysis. That work attempted to exploit the speed of the one-dimensional analysis with the flexibility, generality and user-friendliness of three-dimensional analysis using geometric modeling. A code based on the developed algorithm was employed to demonstrate the generality of the proposed method in analyzing commercial mufflers by considering three very diverse classes of mufflers with different kinds of combinations of reactive, perforated and absorptive elements. Though the examples used were not very complex for they were meant to be just representative cases of certain classes of mufflers, yet the algorithm could handle a large domain of commercial mufflers of high degree of complexity. Results from the present algorithm have been validated through comparisons with both the analytical (plane wave based) and the more general, three-dimensional FEM based results. The forte of the proposed method was its power to construct the system matrix consistent with the boundary conditions from the geometrical model to evaluate the four-pole parameters of the entire muffler and thence its transmission loss, etc. Thus, the algorithm could be used in conjunction with the transfer matrix based muffler programs to analyze the entire exhaust system of an automobile.

Boundary element analysis of packed silencers with protective cloth and embedded thin surfaces was presented by Wu et al. [11].

Bulk-reacting porous materials are often used as absorptive lining in packed silencers to reduce broadband noise. Modelling the entire silencer domain with a bulk-reacting material will inevitably involve two different acoustic media, air and the bulk-reacting material. A so-called direct mixed-body boundary element method (BEM) has recently been developed to model the two-medium problem in a single-domain fashion. Wu et al. [11] presented an extension of the direct mixed-body BEM to include protective cloth and embedded rigid surfaces. Protective cloth, an absorptive material itself with a higher flow resistivity than the primary lining material, is usually sandwiched between a perforated metal surface and the lining to protect the lining material from any abrasive effect of the grazing flow. Two different approaches were taken to model the protective cloth. One was to approximate sound pressure as a linear function across the cloth thickness and then used the bulk-reacting material properties of the cloth to obtain the transfer impedance. The other was to measure the transfer impedance of the cloth directly by an experimental set-up similar to the two-cavity method. As for an embedded thin surface, it was a rigid thin surface sandwiched between two bulk-reacting linings. Numerical modelling of an embedded thin surface was similar to the modelling of a rigid thin surface in air. Several test cases were given and the BEM results for transmission loss (TL) are verified by experimental TL measurements.

The sound attenuation performance of micro-perforated panels (MPP) with adjoining air cavity was investigated for a plenum in Ref. [12]. The sound field inside of a plenum was compared for two cases. In the first case, the plenum was treated with an MPP and adjoining air cavity without any partitioning. For the second case, the adjoining air cavity was partitioned into a number of sub-cavities. The resulting sound pressure fields indicated that partitioning the adjoining air cavity increased the overall sound attenuation due to the MPP by approximately 4 dB. The explanation for this phenomenon was investigated by measuring the sound pressure level on planes in front of the MPP. Additionally, boundary element analyses were conducted to simulate the effect of the MPP and adjoining cavity with and without partitioning on the sound field in the plenum. It was demonstrated that a MPP can be modeled as transfer impedance and that partitioning the adjoining cavity enhances attenuation to acoustic modes that propagate transverse to the MPP.

Application of absorptive mufflers in automotive industry was investigated by Yasuda et al. [13]. The tail pipe noise from a commercial automotive muffler was studied experimentally and numerically under the condition of wide open throttle acceleration in the present research. The engine was accelerated from 1000 to 6000 rpm in 30 s at the warm up condition. The transient acoustic characteristics of its exhaust muffler were predicted using one dimensional computational fluid dynamics. To validate the results of the simulation, the transient acoustic characteristics of the exhaust muffler were measured in an anechoic chamber according to the Japanese Standard. It was found that the results of simulation were in good agreement with experimental results at the 2nd order of the engine rotational frequency. At the high order of engine speed, differences between the computational and experimental results existed in the high revolution range (from 5000 to 6000 rpm at the 4th order, and from 4200 to 6000 rpm at the 6th order). According to their results, the differences were caused by the flow noise which was not considered in the simulation. Based on the theory of one dimensional CFD model, a simplified model which can provide an acceptable accuracy and save more than 90% of execution time compared with the standard model was proposed for the optimization design to meet the demand of time to market.

The effect of liner for the acoustic energy absorption was studied by Herrin et al. [14]. They indicated that, if the dimensions of a silencer or muffler component are small compared to an acoustic wavelength, plane wave propagation can be assumed. This is not the case for HVAC (heating, ventilation, and air conditioning) duct systems, and large diesel engine mufflers commonly used in ship and generator sets. For such applications, the wave behavior in the inlet and outlet ducts is three-dimensional. In their paper, the finite element method was utilized to simulate large duct systems with an aim to predict the insertion loss. The boundary condition on the source side was a diffuse field applied by determining a suitable cross-spectral force matrix of the excitation. At the termination, the radiation impedance was calculated utilizing a wavelet algorithm. Simulation results were compared to published measurement results for HVAC plenums and demonstrate good agreement.

The proper use of plane wave models for muffler design was introduced by Herrin et al. [15]. In many industries, muffler and silencer design is primarily accomplished via trial and error. Prototypes were developed and tested, or numerical simulation (finite or boundary element analysis) was used to assess the performance. While these approaches reliably determined the transmission loss, designers often do not understand why their changes improve or degrade the muffler performance. Analyses are time consuming and models cannot be changed without some effort. It was first demonstrated that plane wave models can reliably determine the transmission loss for complicated mufflers below the cutoff frequency. Moreover, it is shown that plane wave models used correctly help designers develop intuition and a better understanding of the effect of their design changes.

In this thesis, the effect of absorptive layer (liner) on the noise attenuation in muffler is investigated. At this regard, following chapters will present the modelling, theory, calculations and results. Final conclusions and recommendations for the future work will also be presented at the last part of this work.

Chapter 3

METHODOLOGY

3.1 Plain wave propagation theory

An absorptive muffler is shown in Figure 4. It uses absorption to reduce the sound energy. Sound waves are reduced as their energy converted into 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 made from shell layers [16]. Between the perforated pipe and the casing is a layer of sound absorptive material that absorbs some of the pressure pulses.

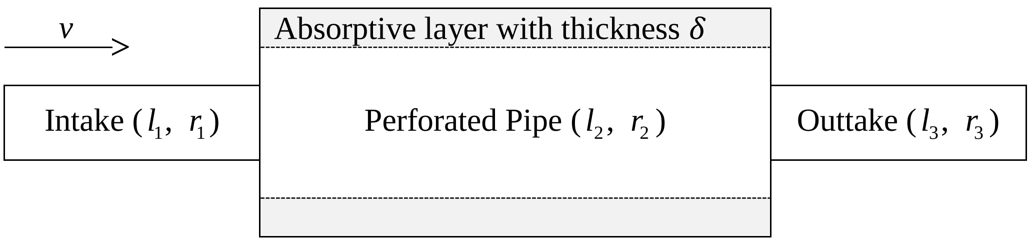

Figure 4. An absorptive muffler

For scattering the plane wave in a direct tube with length of *l*, constant cross-sectional area, and the velocity of the mean flow *v* (figure 4), the acoustic pressure *p* and the fluid speed *v* across the tube section represent the summation of input and output waves. Utilizing the impedance similarity, the sound pressure *p* and volume constancy *v* at the input and the output of each duct can be stated by:

$$\mathrm{p}_1 = \mathrm{Ap}_2 + \mathrm{Bv}_2 \tag{1}$$

$$v_1 = Cp_2 + Dv_2 \tag{2}$$

where A, B, C, and D are usually called the four-pole variables. They are frequency-conditional compound values manifesting the acoustical characteristics of the tube [17].

The values of A, B, C and D for non-viscous medium are:

$$\mathrm{A} = \exp(-\mathrm{jMk_cL}) \cos k_c L \tag{3}$$
$$B = j\left(\frac{\rho c}{S}\right) \exp(-jMk_cL) \sin k_c L \tag{4}$$
$$C = j\left(\frac{S}{\rho c}\right) \exp(-jMk_cL) \sin k_c L \tag{5}$$
$$D = \exp(-jMk_cL) \cos k_c L \tag{6}$$

Where $M = v/c$ is the Mach number of mean current flow which is less than 0.2, c is the speed of sound v (m/s), k_c is the thermally conductive wavenumber (rad/m) which is a function of Mach number ($k_c = k/(1 - M^2)$), k is the acoustic wavenumber (rad/m) ($k = \omega/c$), ω is the circular speed (rad/s), ρ is the fluid density (kg/m^3), and j is the complex value. The value of Mach number is considered to be zero for stationary medium. Actually, in this case it is assumed that the air has no speed and only sound wave front are moving one-dimensionally across the duct.

Then the sound pressure and speed at the outtake of each part of muffler with respect to the sound pressure and velocity of intake can be generally written in the way of matrix form [18] as

$$T_1 = \begin{bmatrix} A & B \\ C & D \end{bmatrix} \tag{7}$$

This approach computes the TL of muffler by using transfer matrix approach [19]. A linear acoustic four-pole transfer matrix is

$$\begin{bmatrix} p_1(x) \\ v_1(x) \end{bmatrix} = \begin{bmatrix} A & B \\ C & D \end{bmatrix} \begin{bmatrix} p_2(x) \\ -v_2(x) \end{bmatrix} \tag{8}$$

where the p_1 and v_1 are the pressure of sound and velocity of normal particle at the inlet, respectively. Also, p_2 and v_2 are similar values at the outlet. There is negative sign on v_2 is added because the vector at the outlet on the BEM model is against the normal vector at the inlet. To obtain the matrix, imagine a simple rectangular duct with (v_1, p_1) and (v_2, p_2) parameters as inlet and outlet one. The governed pressure equation is:

$$p(x) = A \cos kx + B \sin kx \tag{9}$$

By taking derivation this equation with respect to location (x), we have

$$\frac{dp}{dx} = -kA \sin kx + kB \cos kx \tag{10}$$

And the equation of velocity is

$$v(x) = \frac{i}{\rho\omega}\frac{dp}{dx} = \frac{i}{\rho c}(-A \sin kx + B \cos kx) \tag{11}$$

With considering the existed boundary conditions on the equations of the pressure and velocity respectively, you will find the unknown parameters. Finally, the matrix of simple duct will be derived:

$$\begin{bmatrix} p_1(x) \\ v_1(x) \end{bmatrix} = \begin{bmatrix} \cos kl & i\rho c \sin kl \\ \frac{i}{\rho c} \sin kl & \cos kl \end{bmatrix} \begin{bmatrix} p_2(x) \\ -v_2(x) \end{bmatrix} \tag{12}$$

In the above sentence, the coefficients matrix is called Four-pole transfer matrix and is shown by [T] for a straight duct. In practice, it is more convenient to use volume velocity instead of the particle velocity v in [T]:

$$\begin{bmatrix} p_1 \\ s_1 u_1 \end{bmatrix} = \begin{bmatrix} \cos kl & \frac{i\rho c}{s_2} \sin kl \\ \frac{i s_1}{\rho c} \sin kl & \frac{s_1}{s_2} \cos kl \end{bmatrix} \begin{bmatrix} p_2 \\ s_2 u_2 \end{bmatrix} \tag{13}$$

where $s_1 v_1$ and $s_2 v_2$ are volume velocity at inlet and outlet, respectively.

3.2 Helmholtz equation

The equations that explain the propagation of sound in the fluid type mediums can be calculated from the equations of fluid flow. These equations are the fundamental equations of continuum mechanics which describes the conservation of mass, the conservation of momentum that is often referred as the Navier-Stokes equation and explain the energy conservation in a medium, and the equation of state that describes the relation between thermodynamic variables [19].

In most classical acoustic cases, the flow assumed lossless, viscous effects are neglected, and a linearized type of equation of state is used. Under these assumptions, the acoustic field can be described by one variable, i.e. pressure, and is governed by the wave equation as

$$\frac{1}{\rho_0 c^2}\frac{\partial^2 p}{\partial t^2}+\nabla.\left(-\frac{1}{\rho_0}(\nabla p-q)\right)=Q \tag{14}$$

Where t is time in second, ρ_0 is the density of fluid in (kg/m^3), and q and Q are possible acoustic sources in (N/m^3). In the homogenous case, when there are no acoustic sources q and Q, one simple solution to Helmholtz equation is the plane wave as

$$p=P_0 e^{i(\omega t-k\cdot x)} \tag{15}$$

where P_0 is the wave amplitude and it is moving in the k direction with angular frequency ω and wave number $k=|k|$.

3.3 Calculation of noise transmission loss in absorptive mufflers

In this thesis, an absoptive muffler which has been shown in Fig. 4, is considered. It has a layer of absorptive material in its scilencer. This abroptive layer is named as liner. Several materials are used in the absorptive liner.

In the absorbing glass wool, the damping enters the equation as a complex speed of sound $c_c = \omega / k_c$, and a complex density $\rho_c = k_c Z_c / \omega$, where k_c is the complex wave number, and Z_c is the complex impedance.

For a highly porous material with a rigid skeleton, Delany and Bazley [20] presented a model which estimates these parameters as a function of frequency and flow resistivity by

$$k_c = k_a(1 + 0.098 \cdot (\rho_a f / R_f))^{-0.7} - i \cdot 0.189 \cdot (\rho_a f / R_f)^{-0.595}) \tag{16}$$

and

$$Z_c = Z_a(1 + 0.057 \cdot (\rho_a f / R_f))^{0.734} - i \cdot 0.087 \cdot (\rho_a f / R_f)^{-0.732}) \tag{17}$$

where R_f is the flow resistivity, and k_a and Z_a are the free-space wave number and impedance of air, respectively. For glass wool-like materials, Bies and Hansen [21] give an empirical correlation:

$$R_f = \frac{3.18 \cdot 10^{-19} \cdot \rho_{ap}^{1.53}}{d_{av}^2} \tag{18}$$

Where ρ_{ap} is the material's apparent density and d_{av} is the mean fiber diameter. This model uses a lightweight glass wool with density of 12 kg/m^3 and mean fiber diameter of 10 micro-meter.

At the solid boundaries, which are the outer walls of the resonator chamber and the pipes, the model uses sound hard (wall) boundary conditions. The condition imposes that the normal velocity at the boundary is zero.

The boundary condition at the inlet involves a combination of an incoming imposed plane wave and an outgoing radiating plane wave.

An educational version of MAP software [22] is used to calculate the noise transmission loss in absorptive muffle.

The root mean square of calculated noise transmission loss (RMSL) in [dB] is considered as

$$RMSL = \sqrt{\frac{\int_{f_{\min}}^{f_{\max}} TL^2(f)\, df}{f_{\max} - f_{\min}}} \tag{19}$$

In the next sections, the results of simulations with MAP software are presented.

Chapter 4

RESULTS AND DISCUSSION

4.1 Model description

Figure 5 shows the geometrical description of absorptive muffler when a full rotational symmetry around the centerline of model is considered. The radius of cylendrical inlet and outlet, i.e. r_1, is considered to be same. The radius of cylendrical scilencer part, i.e. r_2, is considered from the centerline to the beginning of absorbtive layer. The thickness of absorptive layer is δ.

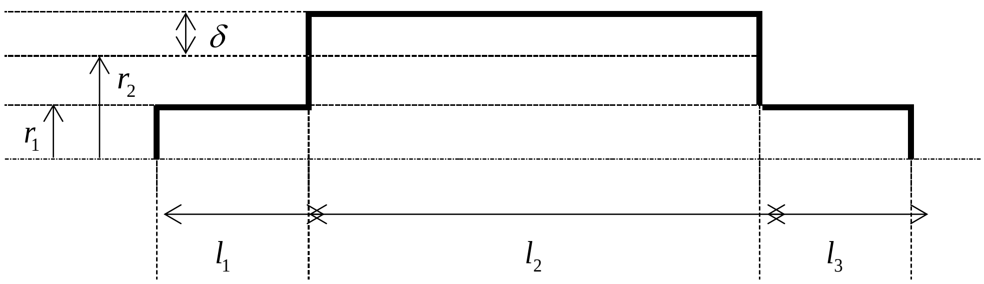

Figure 5. Model of absorptive muffler considering rotational symmetry

The initial geometry and dimension values for the absorptive muffler is given in table 1. The radius of inlet and outlet is set to be 0.0254 m. The radius of scilencer is set to be 0.0762 m. The thickness of absoptive layer is considered as 0.0254 m. The lenghts of inlet, scilencer and outlet are considered to be 0.1524 m, 0.4572 m and 0.1524 m, respectively.

If no liner is considered, then the radius of scilencer will be 0.1016 m.

Table 1. Initial dimensions of specification of muffler

Muffler part name	Dimension value in [m]
Radius of inlet r_1	0.0254
Radius of outlet r_1	0.0254
Radius of silencer r_2	0.0762
Thickness of absorptive layer δ	0.0254
Length of inlet l_1	0.1524
Length of outlet l_2	0.1524
Length of silencer l_3	0.4572

Various material specifications for the absorptive muffler are described in table 2. Several types of absorptive materials as basalt wool, polyester, needle fiber and cell foam are considered. The considered absorptive materials in this thesis are being commonly used as absorptive materials for producing absorptive mufflers. They absorb the energy of exhaust noise from the engine and convert it to heat. Fluid Mach number is zero.

Table 2. Material specification of absorptive material

Parameter	Value
Fluid type	Air
Temperature in [$^{\circ}c$]	400
Fluid density in [g/cm^3]	0.0005
Speed of sound in [*m/s*]	514.1
Density of Basalt wool in [g/cm^3]	2.7
Density of Polyester in [g/cm^3]	1.37
Density of Needle fiber in [g/cm^3]	0.18
Density of Cell foam in [g/cm^3]	0.3

4.2 Simulation results

An educational version of MAP software provided by the vibroacoustic consortium of university of Kentucky in USA is used for the simulation. MAP is the acronym for Muffler Analysis Program. It is a based on the direct mixed-body boundary element method (BEM) developed at the University of Kentucky [23].

MAP includes the four-pole method for evaluating the transmission loss (TL). The fundamental of four-pole methods and calculation of TL has been already discussed in the previous section. Hence, the one-dimensional wave theory is considered.

4.2.1 Transmission loss of muffler without absorptive liner

At first, the transmission loss of muffler without any absorptive layer should be evaluated. At this regard, the same dimension of muffler as given in table 1 is considered. Figure 6 shows that calculated transmission loss (TL) in decibel (dB) of such muffler over a wide frequency range from 0 to 4000 Hz. It indicates the TL is reduced after the frequency of 2000 Hz. The maximum TL peak is 23.1 dB at 2000 Hz. It represents that the muffler without absorptive liner cannot work good to attenuate the sound. Also, the RMSL is 10.7 dB over the wide frequency range.

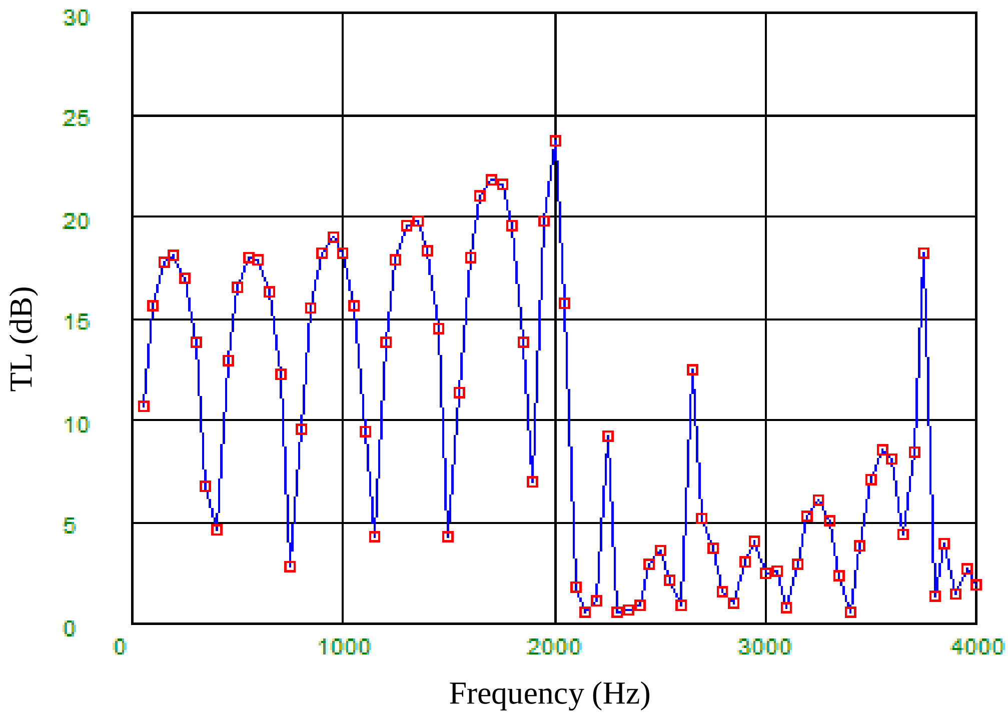

Figure 6. Transmission loss of muffler without liner, same dimension as given in table 1 but with silencer radius of 0.1016 m.

4.2.2 Transmission loss of muffler with absorptive liner

In this part, the transmission loss of muffler with absorptive layer with the geometry shown in figure 5 is presented. Figure 7 shows the TL for the case when Needle fiber is considered for the absorptive layer. The highest peak of TL curve in this case is 47.1 dB has appeared at the frequency of 2720 Hz. Here the RMSL value is 27.5 dB which is 16 dB more than the original case without absorptive layer. So, it shows significantly the effect of adding absorptive layer to muffler, especially for the high frequency ranges more than 2000 Hz.

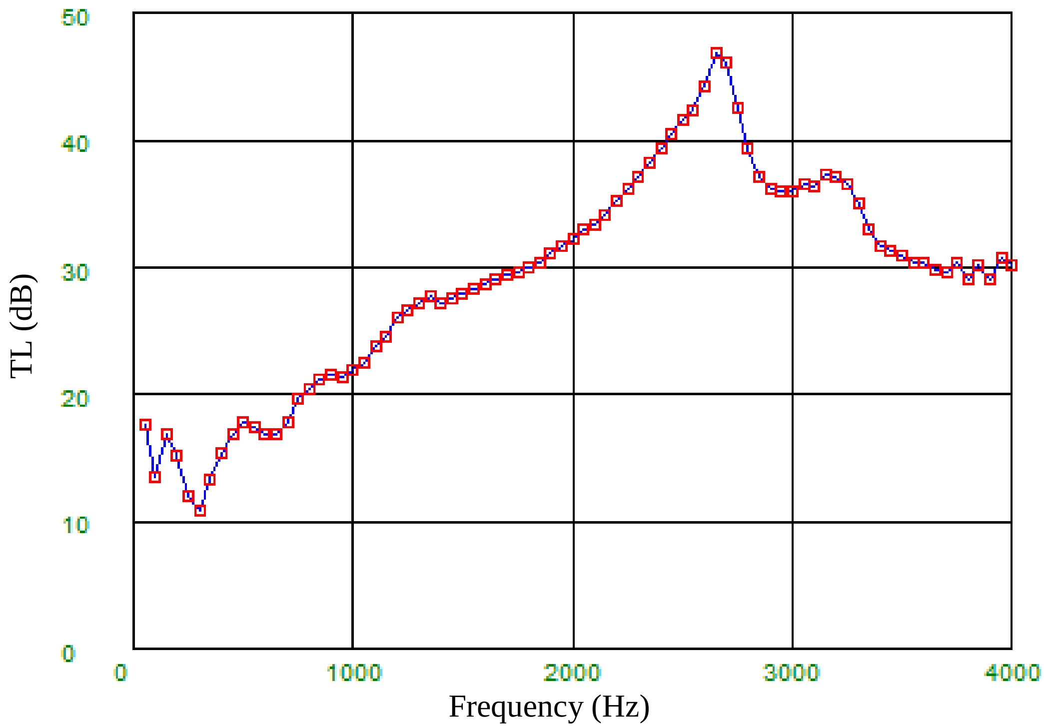

Figure 7. Transmission loss of muffler with Needle fiber absorptive layer around silencer

Figure 8 shows the TL for the case when Polyester is considered for the absorptive layer. The highest peak of TL curve in this case is 62.6 dB has appeared at the frequency of 2290 Hz. Here the RMSL value is 31.6 dB which is 20.9 dB more than the original case without absorptive layer. So, it shows significantly the effect of adding denser absorptive layer to muffler, especially for the mid-frequency range.

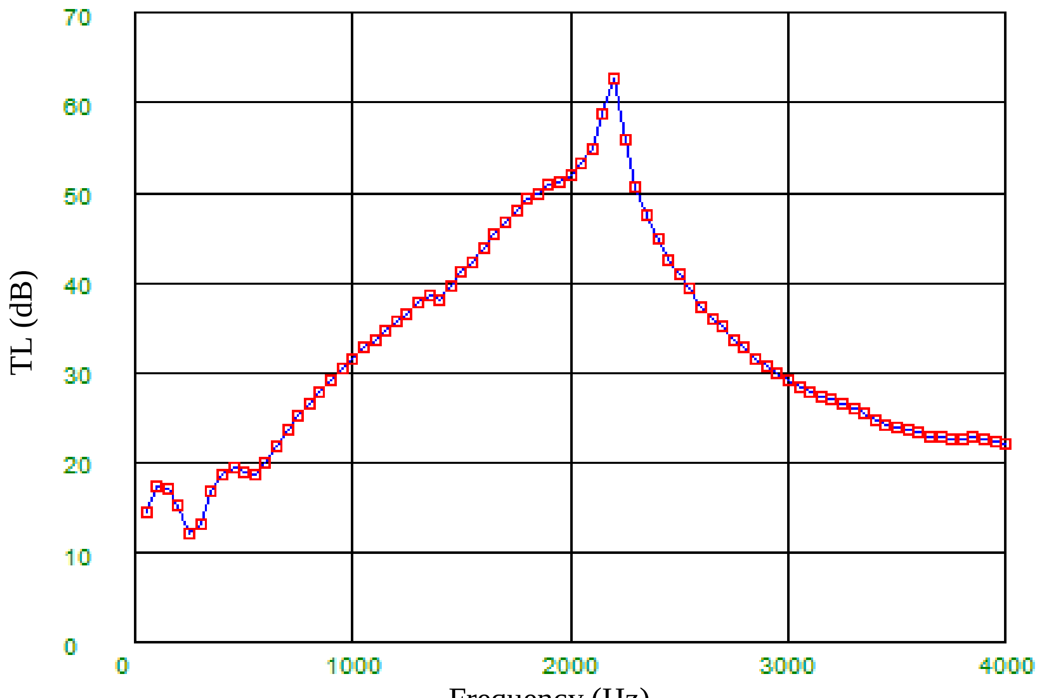

Figure 8. Transmission loss of muffler with Polyester absorptive layer around silencer

Figure 9 shows the TL for the case when Basalt wool is considered for the absorptive layer. The highest peak of TL curve in this case is 71.5 dB has appeared at the frequency of 2570 Hz. Here the RMSL value is 38.4 dB which is 27.9 dB more than the original case without absorptive layer. So, it shows significantly the effect of adding denser absorptive layer to muffler, especially for the mid-frequency range. This indicates that denser absorptive layer is absorbing more energy from the fluid stream inside the muffler. However, it causes to add the weight of muffler.

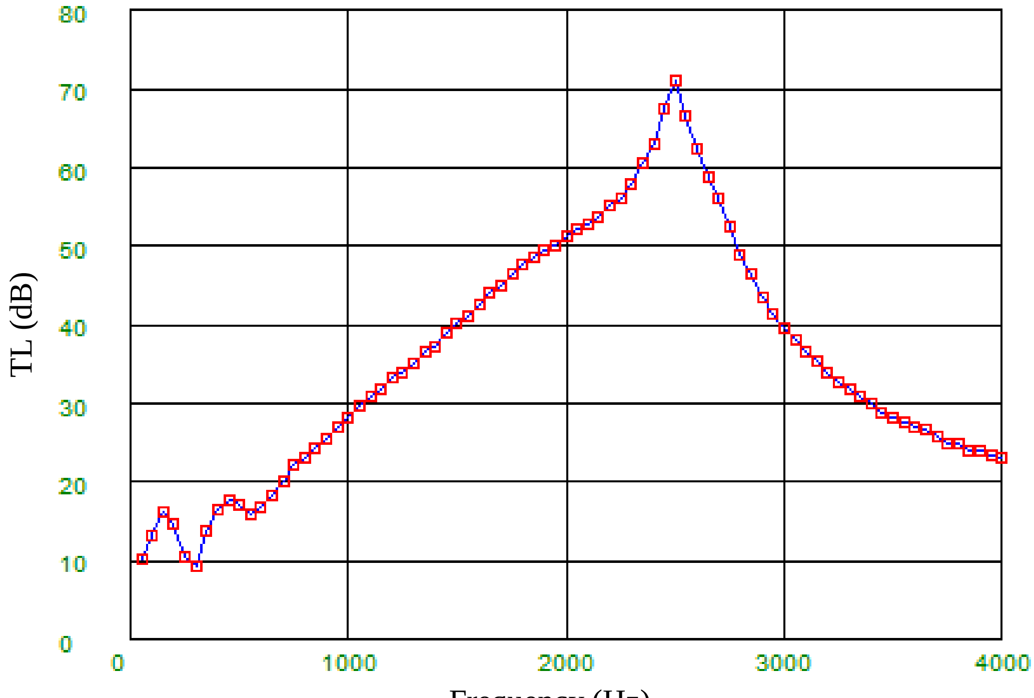

Figure 9. Transmission loss of muffler with Basalt wool absorptive layer around Silencer

Figure 10 shows the TL for the case when Cell foam is considered for the absorptive layer. The highest peak of TL curve in this case is 55.4 dB has appeared at the frequency of 2430 Hz. Here the RMSL value is 30.4 dB which is 19.9 dB more than the original case without absorptive layer. So, it shows significantly the effect of adding denser absorptive layer to muffler, especially for the mid-frequency range. This indicates that denser absorptive layer is absorbing more energy from the fluid stream inside the muffler. However, it causes to add the weight of muffler.

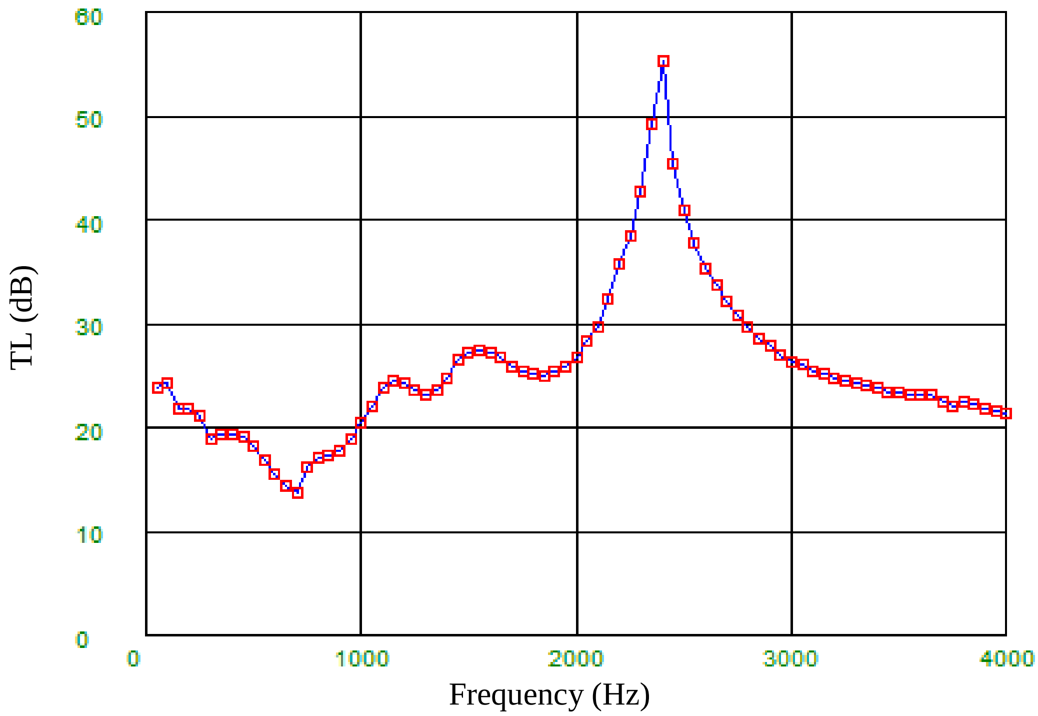

Figure 10. Transmission loss of muffler with Cell foam absorptive layer around silencer

In figure 11, a comparative study on the effect of various materials for absorptive layer is presented. In fact, it is a summary of previous cases. As it is shown, the muffler with no absorptive layer has the lowest level of sound transmission loss.

The situation even gets worse for the frequencies more than 2000 Hz. However, if absorptive layer is being added the structure of muffler, then the value of TL and RMSL is increasing. This fact is clear the figure 10. Moreover, it is understandable that with using denser absorbing materials as liner around the silencer, the value of noise transmission loss both over the wide frequency range and at picks are increased.

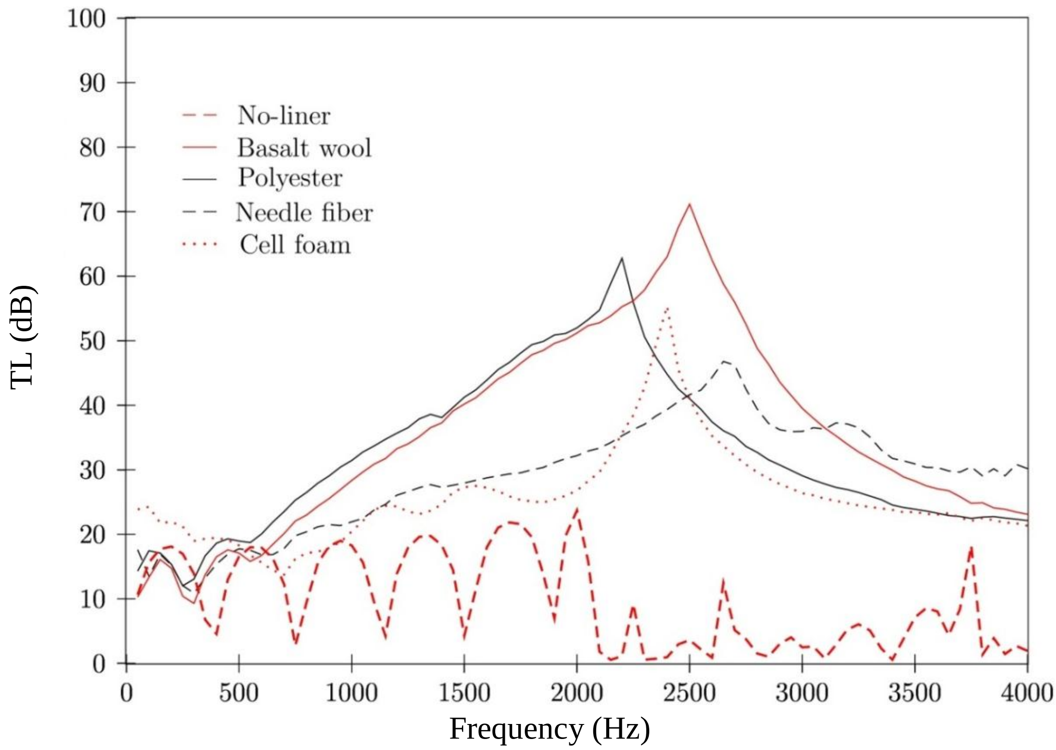

Figure 11. Summary of sound transmission loss for various absorbing materials of liner in absorptive muffler

4.3 Study on the geometry of absorbing layer and noise transmission loss

A sensitivity analysis is done in this section to understand the effect of various thickness layers for absorbing liner of muffler. Also, various geometries for the structure of muffler is considered to have more general understanding about the effect the geometry on the level of noise transmission loss in an absorptive muffler.

Table 3 shows the effect of glass wool liner thickness δ on TL and RMSL of absorptive muffler. The density of glass wool is considered to be 0.48 g/m^3 . Seven various cases are considered. In all cases, the geometry of muffler has not been changed; only the thickness of absorbing layer is modified.

Table 3. Effect of glass wool liner thickness δ on TL and RMSL of absorptive muffler

Parameter	Case 1	Case 2	Case 3	Case 4	Case 5	Case 6	Case 7
Inlet and outlet lengths l_1, l_3 (m)	0.15	0.15	0.15	0.15	0.15	0.15	0.15
Inlet and outlet radius r_1 (m)	0.04	0.04	0.04	0.04	0.04	0.04	0.04
Silencer length l_2 (m)	0.6	0.6	0.6	0.6	0.6	0.6	0.6
Silencer radius r_2 (m)	0.2	0.2	0.2	0.2	0.2	0.2	0.2
Liner thickness δ in (m)	0.0	0.005	0.01	0.015	0.02	0.03	0.04
Maximum TL (dB) at frequency (Hz)	22.6 (1225)	16.5 (1201)	16.1 (1201)	23.8 (1270)	39.1 (1280)	31.8 (1269)	30.2 (1230)
RMSL (dB)	6.4	6.8	7.3	9.7	14.2	16.5	19.1

Table 3 indicates that with increment of liner thickness, the level of noise transmission loss is increased. This confirms the previous results shown in section 4.2.2.

4.4 Modal analysis of muffler structure made from shell

Figure 12 shows the structure of muffler. It has been built by forming of thin shells with very low thickness, e.g. 1 mm. Furthermore, Figure 13 shows the meshed muffler.

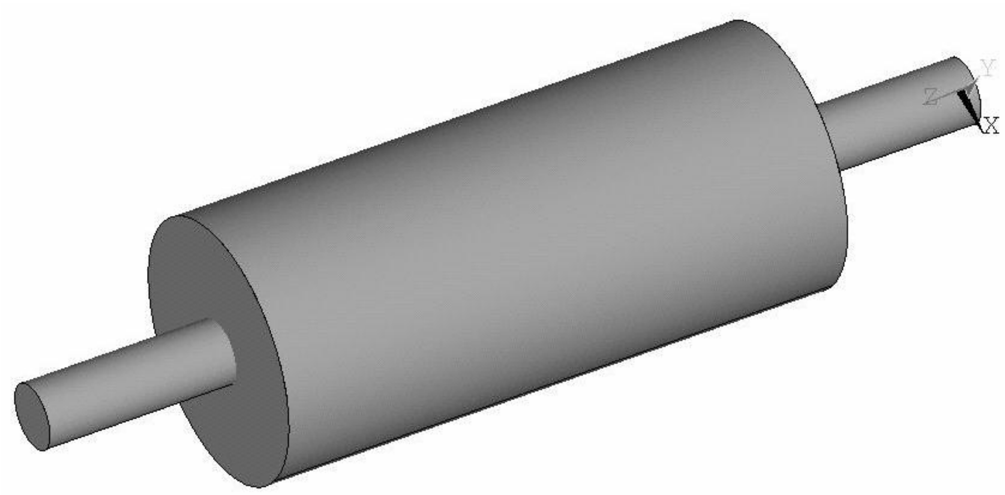

Figure 12. Structure of muffler made from shell

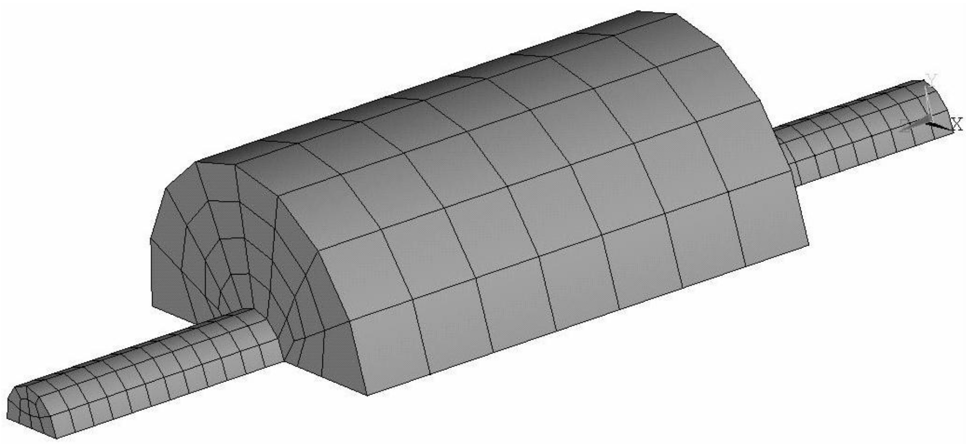

Figure 13. Meshed structure of muffler– half model based on the symmetry of structure

Both ends of muffler are considered to have simply supported condition. Lanczos modal analysis method is used. It is observed that that the structure of muffler made from shell has a lot of modes and natural frequencies on the frequency range of 0 to 5000 Hz. So, it can cause to resonance. This matter can reduce the ability of noise reduction of muffler. Figures 14 to 22 show the structural mode shapes of muffler made from shell at various frequencies.

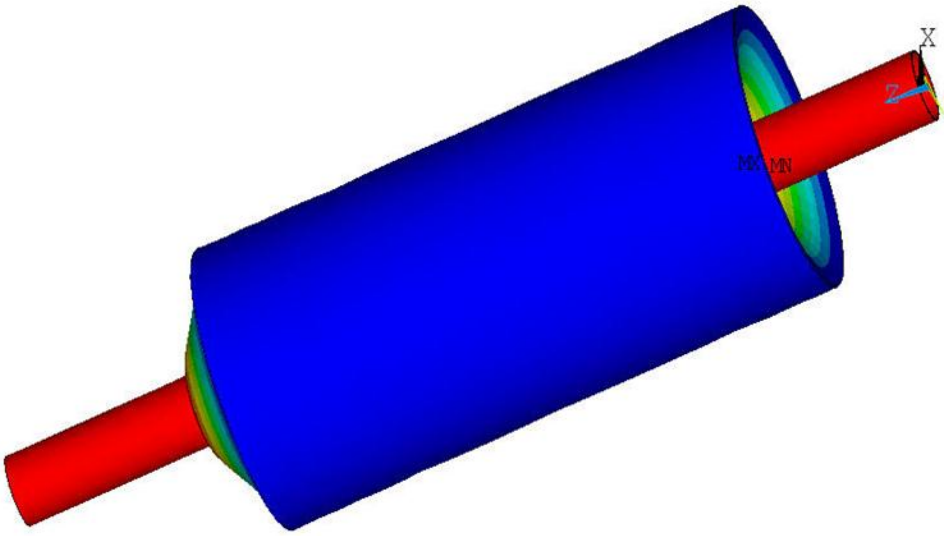

Figure 14. First mode shape of muffler made from shell at frequency of 57.6 Hz

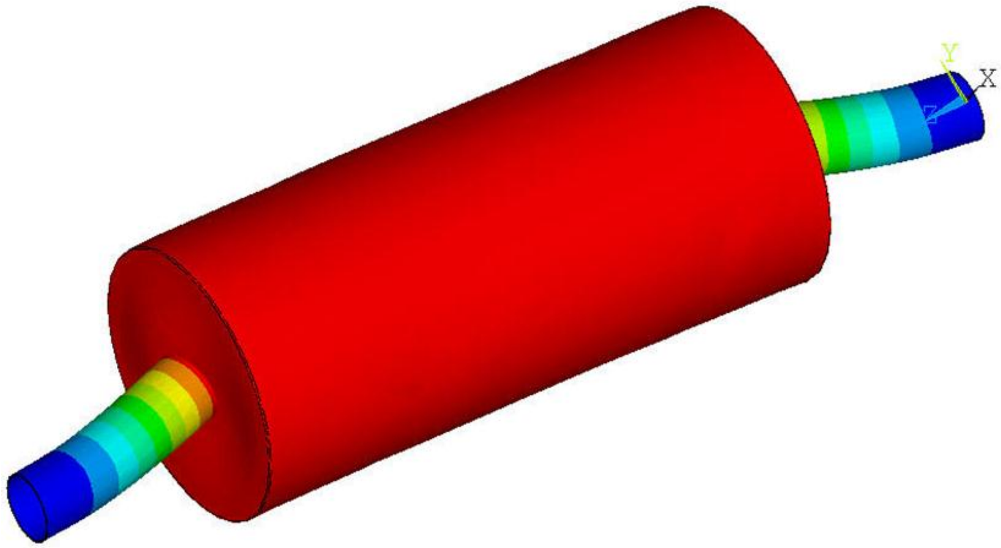

Figure 15. Second mode shape of muffler made from shell at frequency of 211.4 Hz

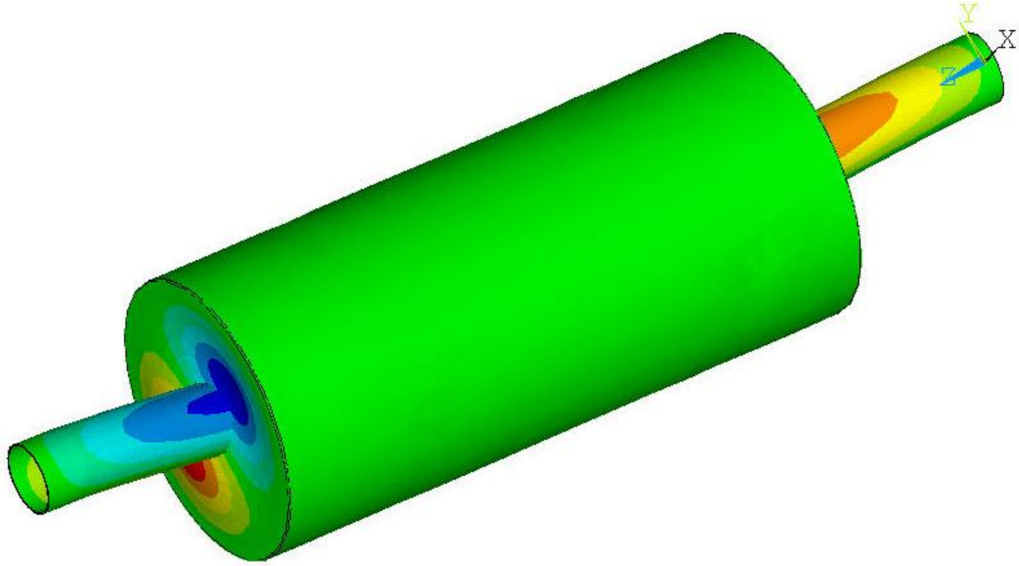

Figure 16. Third mode shape of muffler made from shell at frequency of 337.4 Hz

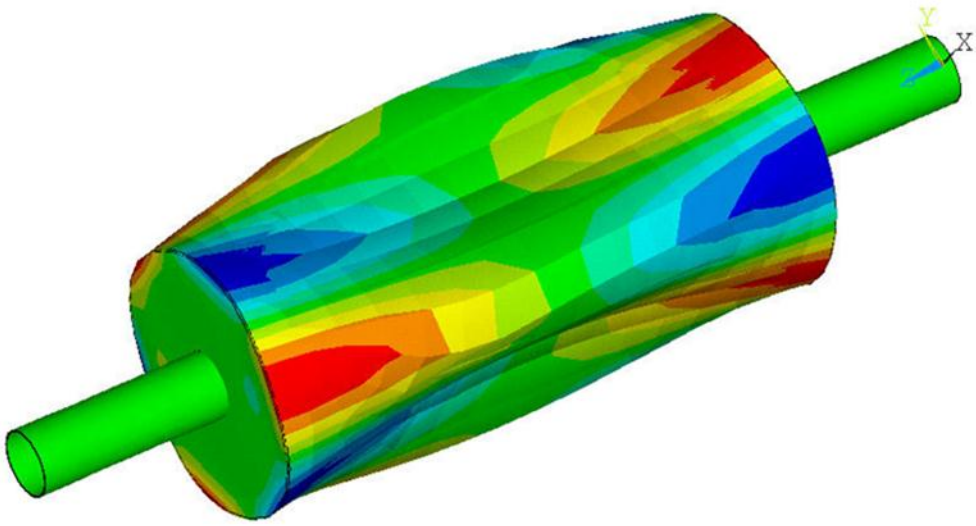

Figure 17. Fourth mode shape of muffler made from shell at frequency of 512.6 Hz

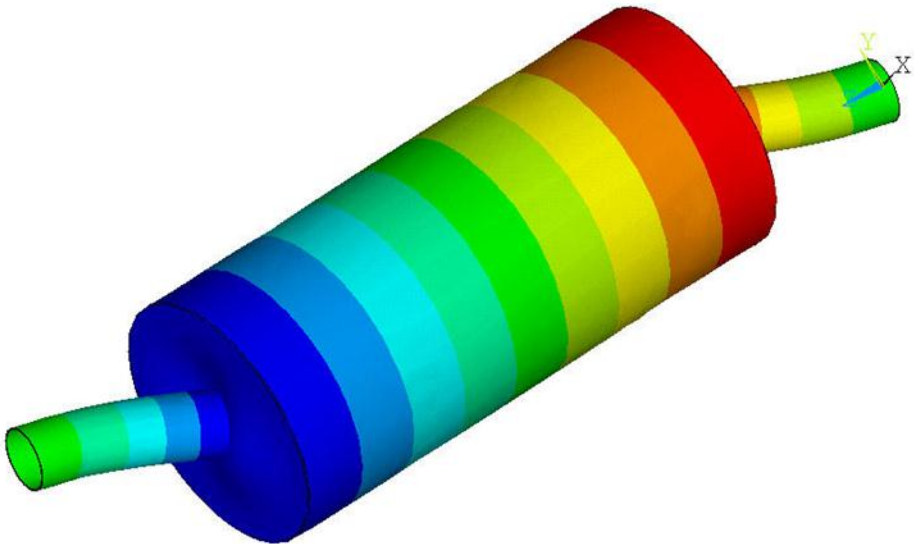

Figure 18. Fifth mode shape of muffler made from shell at the frequency of 734.8 Hz

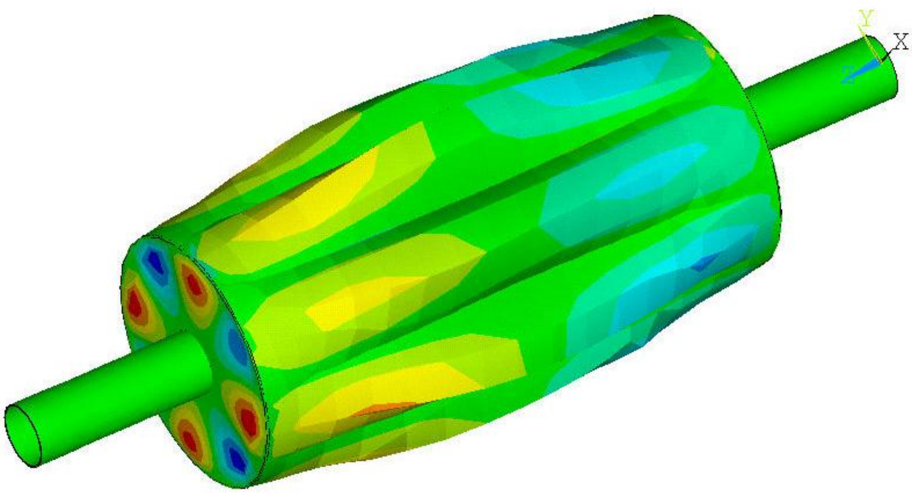

Figure 19. Sixth mode shape of muffler made from shell at the frequency of 880.1 Hz

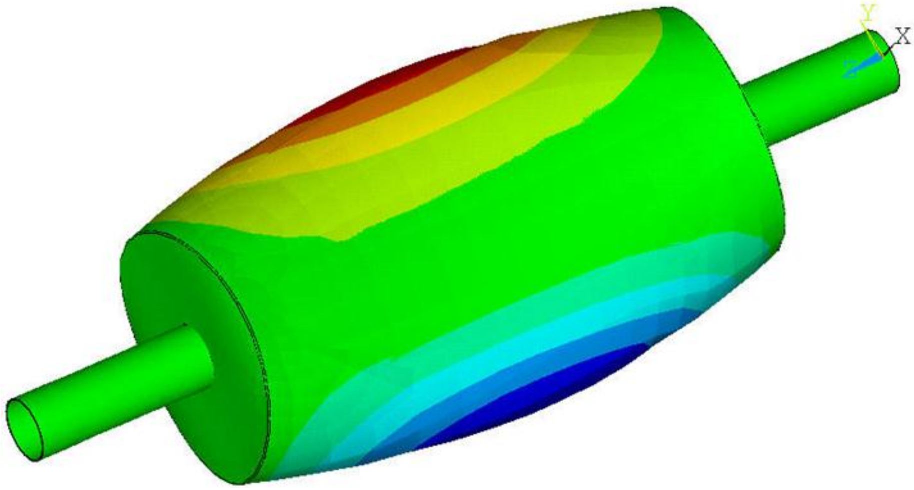

Figure 20. Seventh mode shape of muffler made from shell at frequency of 970.4 Hz

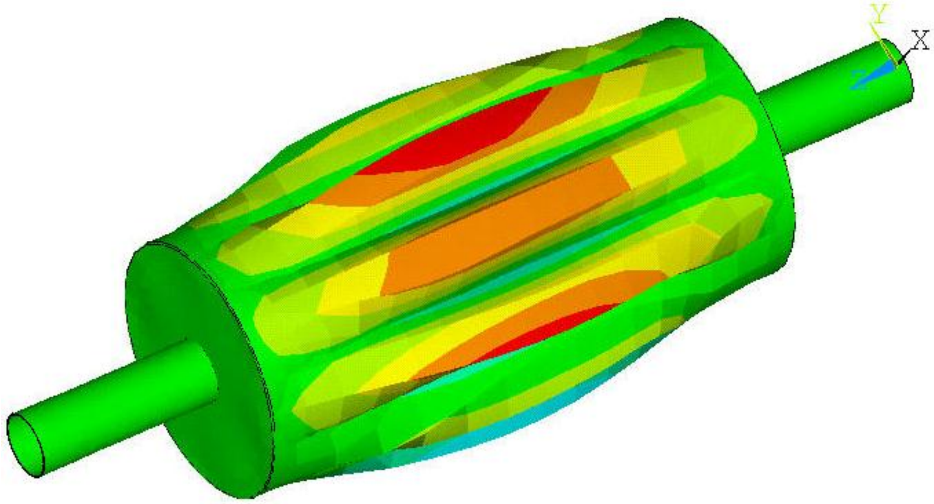

Figure 21. Eight mode shape of muffler made from shell at frequency of 1150.3 Hz

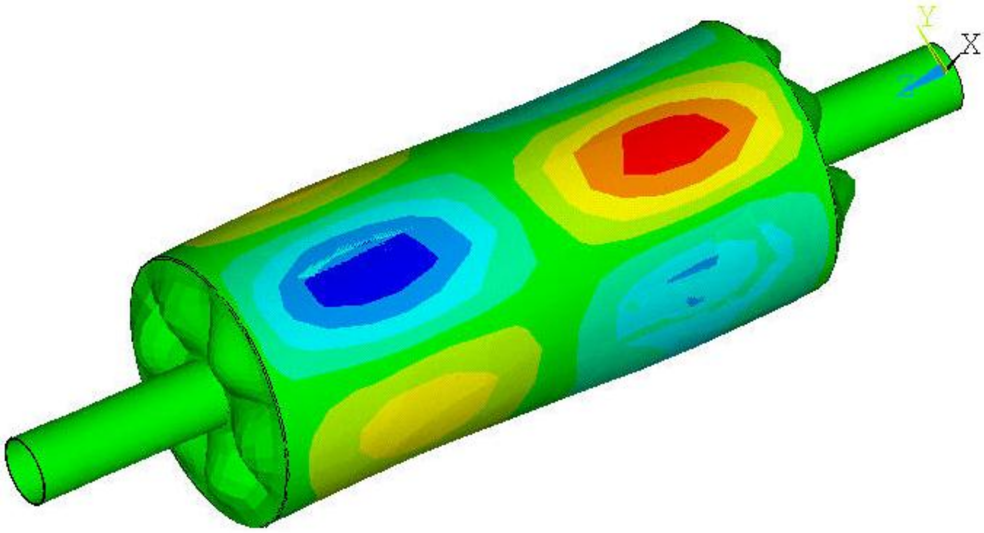

Figure 22. Ninth mode shape of muffler made from shell at frequency of 1263.7 Hz

These structural mode shapes can affect the motion of fluid flow inside of duct. Probably, one can say that even it can affect to validity of one-dimensional motion of wave front inside of muffler.

To have more impression about the movement of fluid inside of muffler, it is necessary to simulate the fluid flow separately. Figure 23 shows the 3-dimensional meshed model of acoustic fluid medium inside of muffler. Here, there is no interaction between the structure and the fluid. Hence, a pure acoustic model for the fluid is considered.

Impedance boundary condition, i.e. $z = \rho c$, is considered at the inlet port. The outlet port is assumed to be ambient condition. The temperature at the inlet is supposed to be $400^{\circ}C$. The first acoustic pressure distribution is shown in figure 24.

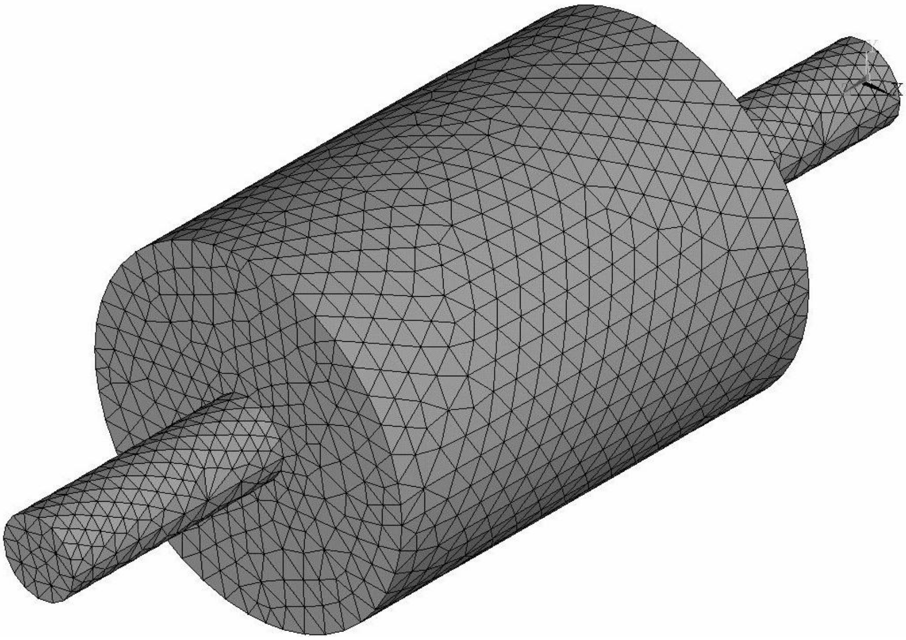

Figure 23. Fluid mesh of muffler

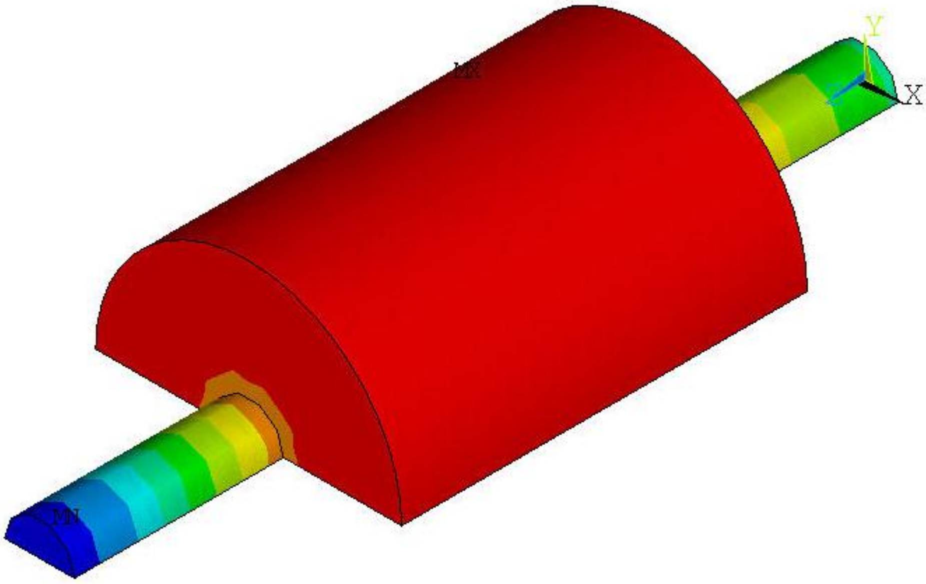

Figure 24. General acoustic pressure distribution in duct at the frequency of 50 Hz
(Red: maximum pressure, Blue: minimum pressure)

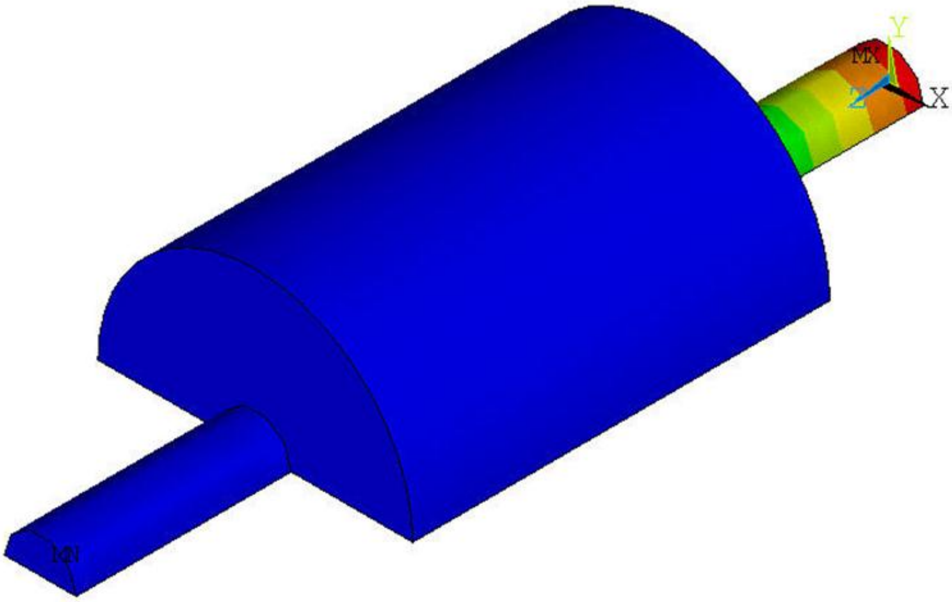

Figure 25. General acoustic pressure distribution in duct at the frequency of 350 Hz

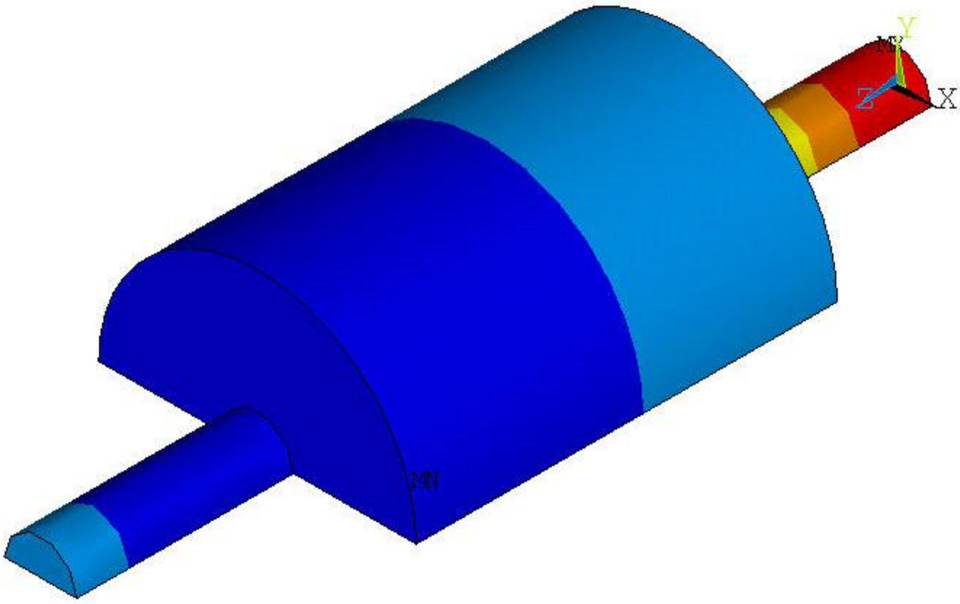

Figure 26. General acoustic pressure distribution in duct at the frequency of 650 Hz

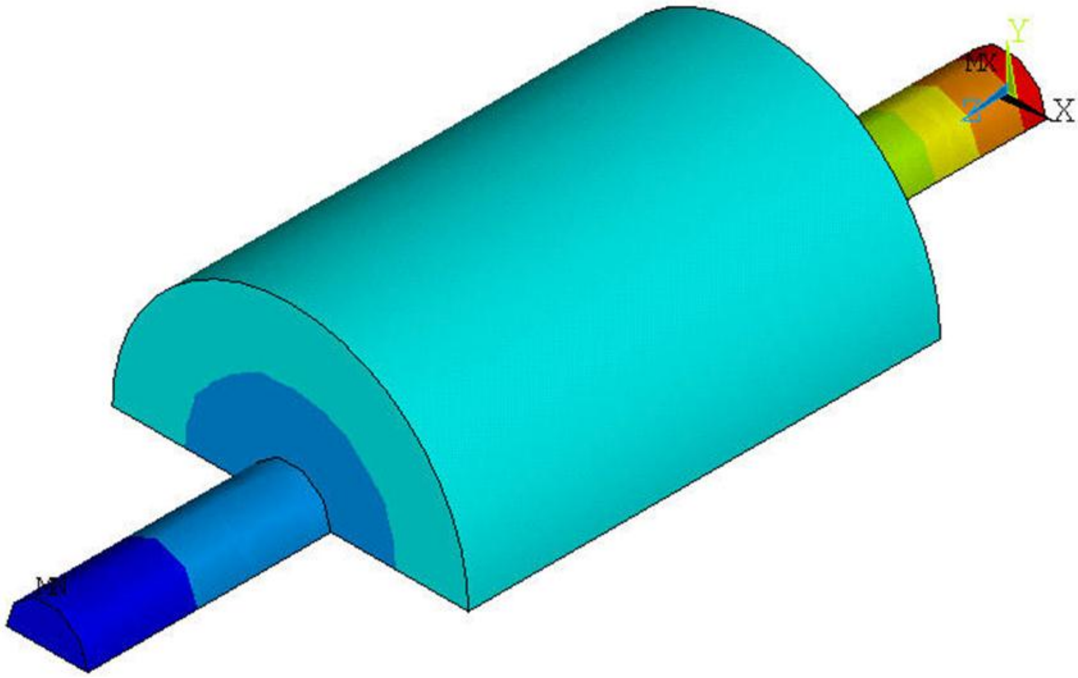

Figure 27. General acoustic pressure distribution in duct at the frequency of 950 Hz

Figures 14 and 27 indicate that the mode shapes of structure and fluid medium are very similar to each other. This confirms that for thin muffler structures made from shell, fluid-structure coupling and interaction plays an important role. With changing of muffler geometry during modal shapes, it can cause to affect the modal shape of fluid. Finally, as Fig. 28 shows, the calculated transmission loss by the written code in MAP is very similar with the TL calculated by ANSYS which shows the accuracy of calculations.

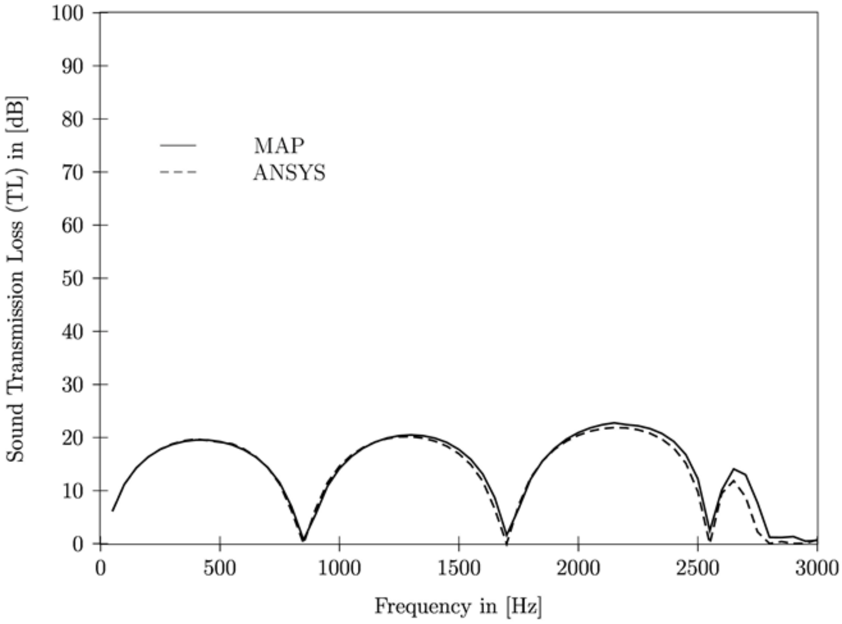

Figure 28. Calculation of TL by MAP software and ANSYS

The TL result which is shown in figure 28 is for the case when no liner is used. Hence, the TL curves are for no-absorptive mufflers. It is clear that from the frequency of 2700 Hz, the TL value is decreased.

Chapter 5

CONCLUSION AND FUTURE WORK

Typically, the greater the ratio of packing surface area to flow area, the greater is attenuation capability of the silencer. Many different packing materials can be used in absorptive silencers and are chosen for use based on varying absorptive performance, price, temperature and corrosion characteristics.

The effect of the thickness of absorptive material and spacing play an important role in sound attenuation. The attenuation increases sharply at high frequencies as the spacing is narrowed.

Better performance at lower frequency is obtained as the thickness of the absorbing material is increased. In order to attenuate high frequency noise, a metal tube surrounded by acoustical-quality glass wool inside the muffler outer containment shell has been used here. The sides of the tube are perforated that permit sound waves impinge on the absorbing materials.

Also, the density of absorptive materials plays an important role in the level of transmission loss. In fact denser absorptive materials can increase the TL more than the absorptive materials with lower density.

The structural mode shapes in mufflers made from thin shell are very close acoustic mode shapes. Therefore, it indicates that full fluid-structure interaction should be considered.

Generally, absorptive mufflers produce better performance for the maximization of noise transmission loss at high frequencies. This matter is clearly shown in this thesis. Moreover, with increment of thickness of absorptive layer in muffler, the value of TL at higher frequencies and even RMSL over the whole frequency range are increased.

However, usage of denser absorptive materials will result to heavier structure for the muffler. Furthermore, it will cause to reduce the natural frequencies of structure of muffler; hence resonance in muffler can be more seen.

For the future work, it is recommended to consider full fluid-structure coupling between the structure and acoustic medium. However, it increases the duration of calculation and complexity of problem. More environmental friendly absorptive materials should be considered for the reduction of air pollution impact of absorptive mufflers.

Novel shapes of absorptive muffler with well-designed absorptive layer shapes should be considered.

REFERENCES

[1] http://auto.howstuffworks.com/muffler3.htm, retrieved on 15/01/2015

[2] Jebasinski, R. (2000). Absorption mufflers in exhaust systems, J. Eberspaecher GmbH and Co.

[3] Rahman, M., Sharmin, T., Hassan, A., Al Nur, A. (2005). Design and Construction of a Muffler for Engine Exhaust Noise Reduction, Proceedings of the International Conference on Mechanical Engineering (ICME2005), 28- 30 December 2005, Dhaka, Bangladesh.

[4] Stewart, G. W. (1922). *Physics Review*, 20, 528-551 Acoustic waves filters.

[5] Divis, D. D., Stokes, Jr. G.M., Morse, D., Stevens, G.L. (1954). Theoretical and Experimental Investigation of Muffler with Comments on Engine- Exhaust Muffler Design. NASA Report No. 1192.

[6] Igarashi, j., Toyama, M. (1958). Fundamental of acoustical silencers. Aeronautical Research Institute, University of Tokyo, Report No.339, 223-241.

[7] Munjal M. L., Sreenath, A.V., Narasimhan, M. V. (1970). Velocity ratio in the analysis of linear dynamical system, *Journal of sound and Vibration*, 26, 173-191.

[8] Munjal, M. L. (1975). Velocity ratio cum transfer matrix method for the evaluation of muffler with neon flow, *Journal of sound and Vibration*, 39, 105-119.

[9] Young, C., Crocker, M. (1975). Prediction to transmission loss in mufflers by finite element method, *Journal of Acoustical society of America*, 57, 144-148.

[10] Panigrahi, S., Munjal, M. L. (2007). A generalized scheme for analysis of multifarious commercially used mufflers, *Applied Acoustics*, 68, 660–681.

[11] Wu, T., Cheng, C., Tao, Z. (2003). Boundary element analysis of packed silencers with protective cloth and embedded thin surfaces, *Journal of Sound and Vibration*, 261(1), 1–15.

[12] Liu, J., Herrin, D. W. (2010). Enhancing micro-perforated panel attenuation by partitioning the adjoining cavity, *Applied Acoustics*, 71, 120–127.

[13] Yasuda, T., Wu, C., Nakagawa, N., Nagamura, K. (2010). Predictions and experimental studies of the tail pipe noise of an automotive muffler using a one dimensional CFD model, *Applied Acoustics*, 71, 701–707.

[14] Herrin, D. W., Ramalingam, S., Cui, Z., Liu, J. (2012). Predicting insertion loss of large duct systems above the plane wave cutoff frequency, *Applied Acoustics*, 73, 37–42.

[15] Herrin, D. W., Hua, X., Zhang, Y., Elnady, T. (2014). The Proper Use of Plane Wave Models for Muffler Design, *SAE Int. J. Passeng. Cars - Mech. Syst.*, 7(3), 927-932.

[16] Potente, D. (2005). General Design Principles for an Automotive Muffler, Proceedings of Acoustics2005, 9-11 November, Busselton, Western Australia.

[17] Pierce, A.(1981). Acoustics: An Introduction to its Physical Principles and Applications, Mc Graw – Hill Series in Mechanical Engineering, 337- 357.

[18] Munjal, M. L. (1997). Plane Wave Analysis of Side Inlet/Outlet Chamber Mufflers with Mean Flow, *Applied Acoustics*, 52 (2), 165–175.

[19] Munjal, M. L. (1987). Acoustics of Ducts and Mufflers. 1st Ed., John Wiley and Sons, New York, 328 pages.

[20] Delany, M., Bazley, E. (1970). Acoustical properties of fibrous absorbent materials, *Applied Acoustics*, 3(2), 105–116.

[21] Bies, D., Hansen, C. (1980). Flow Resistance Information for Acoustical Design, *Applied Acoustics*., 14, 357–391.

[22] Wu, T. (2015). Muffler design software (MAP), University of Kentucky, USA.

[23] Wu, T. (1995). A Direct Boundary Element Method for Acoustic Radiation and Scattering from Mixed Regular and Thin Bodies, *J. Acoust. Soc. Am.*, 97, 84-91.

APPENDICES

Appendix A: Computer code for the modelling of an absorptive muffler by MAP software without liner (absorptive layer)

```
## Version 094, Inputfile

Title=Silencer with absorptive lining

NINT=6,8

ENPL=8,8

FBegin=50,FEnd=5000,Increment=100

UNIT=SI

TempScale=Celsius,Temp=400,Mach=0

Symmetry=Rotational

Block=1

NSub=1

Subsymm=0

NSubDiv=1
```

Xid=2

TLFile=Simplemuffler.tl

FPFile=fourpole.dat

Geometry,Grid=Auto

Geom=Inlet-C,Z=0,R1=0,R2=0.04,NU=0,NV=0,T1=0,T2=0,T3=0,RT=3,SB=1

ID=

Geom=Extube-C,R=0.04,Z1=0,Z2=0.15,NU=0,NV=0,T1=0,T2=0,T3=0,RT=3,SB=1

ID=

Geom=Extube-
C,R=0.1,Z1=0.15,Z2=0.75,NU=0,NV=0,File=Cellfoam,T1=0,T2=0,T3=0,RT=3,TH
K=0.015,SB=1,P=50,T=0.001,D=

ID=

Geom=Extube-
C,R=0.04,Z1=0.75,Z2=0.9,NU=0,NV=0,T1=0,T2=0,T3=0,RT=3,SB=1

ID=

Geom=Outlet-C,Z=0.9,R1=0,R2=0.04,NU=0,NV=0,T1=0,T2=0,T3=0,RT=3,SB=1

ID=

End

TLMethod,None

Method=FourPole,Mode=Auto,0,0

Ain=0.00502654816,Aout=0.00502654816

End

EndInput

Appendix B: Computer code for the modelling of an absorptive muffler by MAP software with liner (absorptive layer)

```
## Version 094, Inputfile

Title=Silencer 2x8x18 with 1/2 in ployester lining

NINT=6,8

ENPL=8,8

FBegin=50,FEnd=4000,Increment=50

UNIT=SI

TempScale=Celsius,Temp=20,Mach=0

Symmetry=Rotational

Block=24

NSub=1

Subsymm=0

NSubDiv=0
```

Xid=2

TLFile=basstudy.tl

FPFile=fourpole.dat

Geometry,Grid=Auto

Geom=Inlet-C,Z=0,R1=0,R2=0.0254,NU=2,NV=1,T1=0,T2=0,T3=0,RT=3,SB=1

ID=

Geom=Extube-
C,R=0.0254,Z1=0,Z2=0.1524,NU=5,NV=1,T1=0,T2=0,T3=0,RT=3,SB=1

ID=

Geom=Inflange-
C,Z=0.1524,R1=0.0254,R2=0.0762,NU=3,NV=1,T1=0,T2=0,T3=0,RT=3,SB=1

ID=

Geom=Extube-
C,R=0.1016,Z1=0.1524,Z2=0.6096,NU=15,NV=1,B,T1=0,T2=0,T3=0,RT=3,SB=1,
Bulk1=Cellfoam,resistivity=0

ID=bulk

Geom=Outflange-

C,Z=0.6096,R1=0.0254,R2=0.0762,NU=3,NV=1,T1=0,T2=0,T3=0,RT=3,SB=1

ID=

Geom=Extube-

C,R=0.0254,Z1=0.6096,Z2=0.762,NU=5,NV=1,T1=0,T2=0,T3=0,RT=3,SB=1

ID=

Geom=Inflange-

C,Z=0.1524,R1=0.0762,R2=0.1016,NU=1,NV=1,B,T1=0,T2=0,T3=0,RT=3,SB=1,Bulk1=Cellfoam,resistivity=0

ID=bulk

Geom=Outflange-

C,Z=0.6096,R1=0.0762,R2=0.1016,NU=1,NV=1,B,T1=0,T2=0,T3=0,RT=3,SB=1,Bulk1=Cellfoam,resistivity=0

ID=bulk

Geom=Tube-

C,Type=I,N,R=0.0762,Z1=0.1524,Z2=0.6096,NU=15,NV=1,T1=0,T2=0,T3=0,RT=3,SB=1,Bulk1=Basaltwool,resistivity=0

ID=interface

Geom=Outlet-

C,Z=0.762,R1=0,R2=0.0254,NU=2,NV=1,T1=0,T2=0,T3=0,RT=3,SB=1

ID=

End

TLMethod,None

Method=FourPole,Mode=Auto,0,0

Ain=0.002026829881816,Aout=0.002026829881816, End, EndInput

Appendix C: Computer code for the modelling of muffler

```
/PREP7

! define element and materials

et,1,shell181

r,1,0.001

mp,ex,1,200E09

mp,nuxy,1,.3

mp,dens,1,7850

! create the model

rapipe=0.0254

lpipe=0.1524

rchamb=0.1016

lchamb=0.4572

cylind,0.0244,rapipe,0,0.1524,0,360
```

```
cylind,0.1006,0.1016,0.1524,0.6096,0,360

cylind,0.0244,0.0254,0.6096,0.762,0,360

cylind,0.0254,0.1016,0.1524,0.1534,0,360

cylind,0.0254,0.1016,0.6086,0.6096,0,360

asel,all

nummrg,all

amesh,all

! define excitation and boundary conditions on inlet and outlet port

nsel,s,loc,z,0 ! nodes on inlet

nsel,a,loc,z,0.762 ! nodes on outlet

d,all,ux,,,,,uy,uz

alls

fini

! perform solutions
```

```
/solu
antype,modal,new
 modopt,lanb,500,0,7000
  mxpand,500
 solve
  fini
/POST1
SET,LIST,2
SET,FIRST
PLDISP,0
ANMODE,10,.5E-1
SET,NEXT
PLDISP,0
ANMODE,10,.5E-1
```

```
SET,NEXT

PLDISP,0

ANMODE,10,.5E-1

SET,NEXT

PLDISP,0

ANMODE,10,.5E-1

SET,NEXT

PLDISP,0

ANMODE,10,.5E-1

FINISH

/CLEAR,NOSTART

/eof
```