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Structural analysis for exhaust gas flow through an elliptical chamber muffler under static and dynamic loading condition

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ABSTRACT

High pressure and temperature exhaust gases coming out from automobile engine are made to pass through muffler for reduction of sound resulting from propagation of these pressure waves. The mufflers may be of reactive, dissipative and resonating type. The present paper deals with an automotive muffler that is modeled based on practical dimensions of a 4-stroke 2-cylinder MAHINDRA MAXIMO PLUS C.I. engine in CATIA V5 software. The geometry adopted is elliptical in nature. Comparative static structural analysis for stress, strain and deformation along with modal analysis for deformation under dynamic loading has been performed for perforated and non-perforated design of the muffler using ANSYS Workbench 14.5. The effect of incorporation of perforation is studied on the corresponding static and dynamic behavior of the muffler.

1. INTRODUCTION

The stringent of environmental regulations that are being put out have led to an increasing need for effective design and use of muffler in automotive vehicles sound attenuation. The high pressure and temperature exhaust gases released from engine cylinder lead to excessive vibration and noise. These gases are transmitted along the exhaust system to the muffler where noise reduction takes place by means of expansion and resonating chambers, baffle plates, perforated pipes, etc. Determination of structural failure in automotive vehicles mostly involves ground tests or actual road load tests. As such it slows down the progress of design and development of automobile engines. This is where virtual calculation of structural failure results using mathematical modeling techniques or modern CAE tools finds its indispensability. These tools not only lead to an optimum design with increased effectiveness but are also cost effective and time efficient. Due to space constraints in the chassis of an automobile body elliptical geometry is apparently seen to be more preferred over circular cross section.

Munjal [1] classified mufflers primarily into two types – Reflective or reactive mufflers, which works on the principle of impedance mismatch and Dissipative mufflers which works on the principle conversion of sound energy to heat. The author has worked in frequency and time domain one dimensional analysis of acoustic filters, three dimensional numerical analysis of mufflers, aero-acoustic measurements and analysis of HVAC (heating, ventilation and air-conditioning) ducts for breakout noise.

Acoustic behavior of elliptical chamber mufflers is studied taking into account the effect of chamber length, location of inlet and outlet ports and eccentricity of the ellipse by Denia et al. [2]. The authors have expressed the solution of the wave equation in elliptical coordinates in terms of Mathieu functions to obtain the natural frequency and mode shapes.

Gerges et al. [3] stated that the transfer matrix method could be used for prediction of transmission loss of mufflers in the low frequency range. Several muffler configurations were measured experimentally for transmission loss and the results were found to be in good agreement with those obtained numerically from transfer matrix method.

Sohei et al. [4] derived the four pole parameters of an elliptical chamber muffler with a perforated pipe by solving the governing equations of these four pole parameters. A detailed examination of one such parameter, without considering the effect of mean flow velocity, yielded similar results as that of experimental verification.

Shah et al. [5] exemplifies the use of modern CAE tools for optimum design and development of automotive muffler so as to obtain a short product development cycle time based on parameters like noise and back pressure. The practical approach adopted by them involves prototype development and validation of an exhaust muffler. It filters out the best design in the beginning itself.

Mimani et al. [6] analyzed short elliptical chambers used in automobile exhaust system using the plane wave propagation model. Differential equation governing this model is solved to obtain the Frobenius solution in terms of transmission loss which is seen to agree very closely with the Matrizant solution obtained earlier by the authors [7].

Dhaiban et al. [8] predicted the transmission loss of an elliptical chamber muffler using 3-D finite element analysis. Results obtained were then seen to have an excellent agreement with experimental results obtained earlier by Gerges et al. [3].

Mimani et al. [9] further analyzed the behavior of an elliptical chamber muffler having an end inlet and side outlet port (Ref. Fig. A). Studies were performed on a 3-D analytical piston driven model and the results thus obtained were compared with 3-D FEA results performed for various muffler configurations. The effect of chamber length and location of end and side ports on transmission loss of the muffler configurations was discussed and the results were obtained in very less time as opposed to the 3-D FEA results.

Wankhade et al. [10] presented an effective elliptical silencing model which is simple in construction and provided efficient sound attenuation. Sound level at the tail pipe was analyzed by acoustic analysis. Transmission loss was found to increase in an elliptical muffler model having an extra divided inlet tube along with extended inlet and outlet (Ref. Fig. B).

Patekar et al. [11] gave a method for distinguishing working frequency from natural frequency so that resonance can be avoided. For the exhaust system of an automobile six natural frequencies are determined by using FEM and the results are compared against those obtained from FFT analyzer. Both are found to be nearly similar.

2. PROBLEM STATEMENT AND BOUNDARY CONDITIONS

2.1 Problem statement

An elliptical chamber muffler having a single inlet and single outlet is modeled. The dimensions of the muffler are taken from a MAHINDRA MAXIMO PLUS 2-cylinder 4-stroke C.I. engine. An existing design of the muffler consisting of non-perforated inlet and outlet pipes and the resonating chamber divided into three parts by baffle plates is modeled. The design proposed in this paper incorporates perforations in the pipes and the objective is to study the behavior of the mufflers under static as well as dynamic loading condition.

2.2 Boundary conditions

For structural analysis, two fixed supports are provided as constraint at two ends of the muffler, the position of which are in accordance to real time clamping of the muffler in the engine. The inlet and outlet pressure and temperature boundary condition are obtained as maximized values from real time operating conditions of the engine. These are imported to the structural module of ANSYS Workbench 14.5.

Similar fixed supports are applied for modal analysis whereas pressure loads are applied along with fixed supports in harmonic analysis

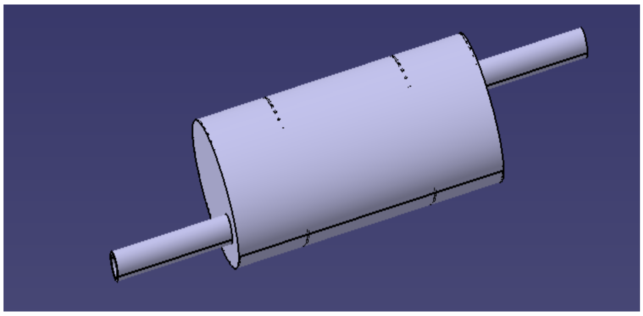
3. GEOMETRIC MODELING

3.1 Three-dimensional CATIA model

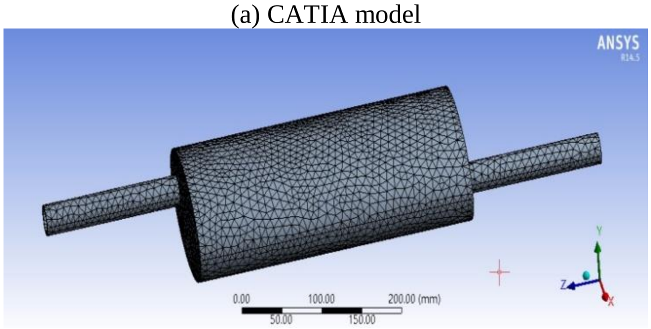
Dimensions for both the muffler models are enumerated as follows:

Major axis diameter of the resonating chamber = 180 mm
Minor axis diameter of the resonating chamber = 110 mm
Length of the resonating chamber = 350 mm
Inner diameter of the inlet and outlet pipe = 24 mm
Outer diameter of the inlet and outlet pipe = 30 mm
Length of the inlet and outlet pipe = 150 mm

Alter preparation of the initial geometric modeling of the elliptical chamber muffler in CATIA v5, the model has meshed in structural module of ANSYS Workbench 14.5. Figure 1 (a) and (b) illustrates the 3-D CATIA model and ANSYS meshed model.



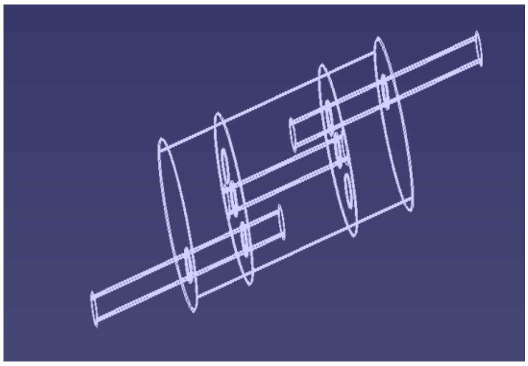
(a) CATIA model



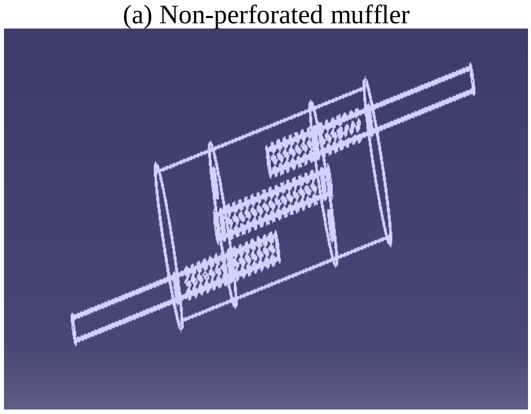
(b) ANSYS model after meshing

Figure 1. 3-D model of the elliptical chamber muffler

Wireframe model for the non-perforated and perforated muffler for static structural analysis is shown in Figure 2 (a) and (b) respectively.



(a) Non-perforated muffler



(b) Perforated muffler

Figure 2. Wireframe models

3.2 Material property

The material applied for analysis is structural steel which is most commonly used for construction of muffler due to its high strength and high corrosion resistance properties. Table1 depicts the material properties of structural steel.

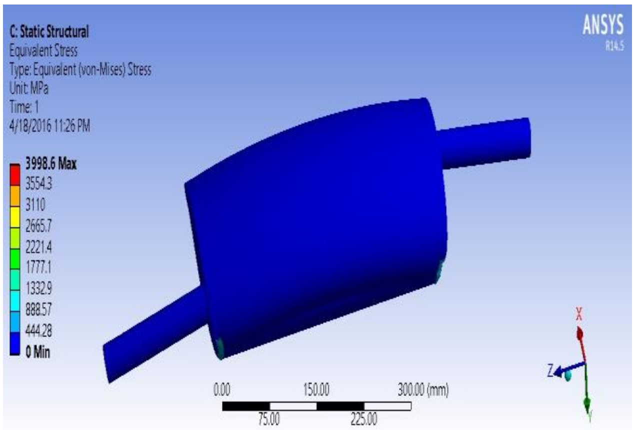
Table 1. Material properties of structural steel

Material property	Value
Young's Modulus	2×10^{11} N/m^2
Poisson's ratio	0.266
Density	7860 kg/m^3
Coefficient of thermal expansion	1.17×10^{-5} /K
Yield strength	2.5×10^8 N/m^2
Specific heat	0.49 KJ/ kg/ K
Thermal conductivity	6.05×10^{-2} W/mm/K
Ultimate tensile strength	4.6×10^8 N/m^2
Bulk modulus	1.6667×10^5 MPa
Shear modulus	76923 MPa

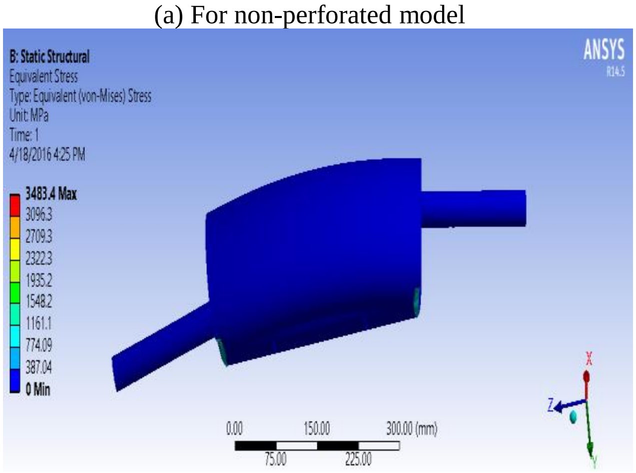
4. RESULTS AND DISCUSSION

4.1 Structural analysis

Structural analysis deals with the determination of effect of loads on structural components and parts like automobile parts, machinery parts and bridges. It is used to calculate total and directional deformation, equivalent elastic strain and stress, normal and support reaction, velocity, acceleration, etc.



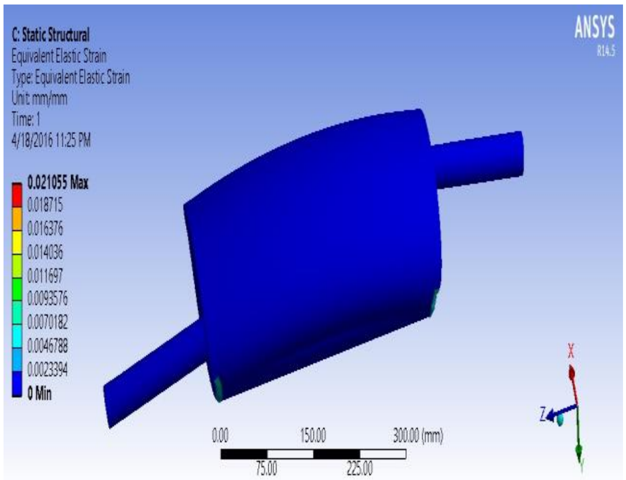
(a) For non-perforated model



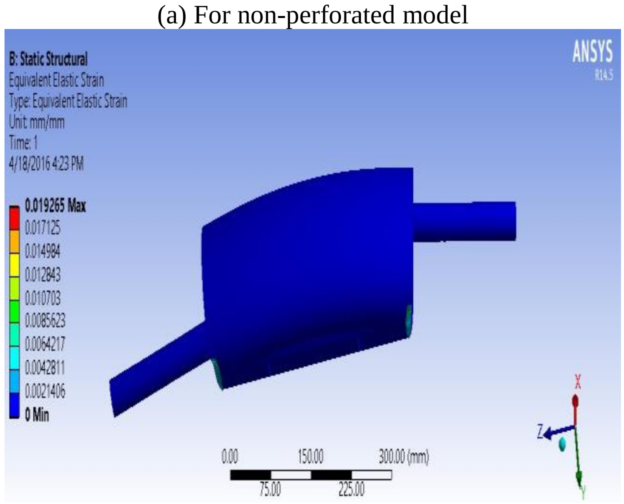
(b) For perforated model

Figure 3. Equivalent von-Mises stress contours

Present work deals with determination of stress, strain and deformation under static loading condition. The results for equivalent von-Mises stress, equivalent elastic strain and total deformation are analysed for both the designs of muffler, i.e, non-perforated and perforated using the simulation results obtained in ANSYS 14.5.



(a) For non-perforated model

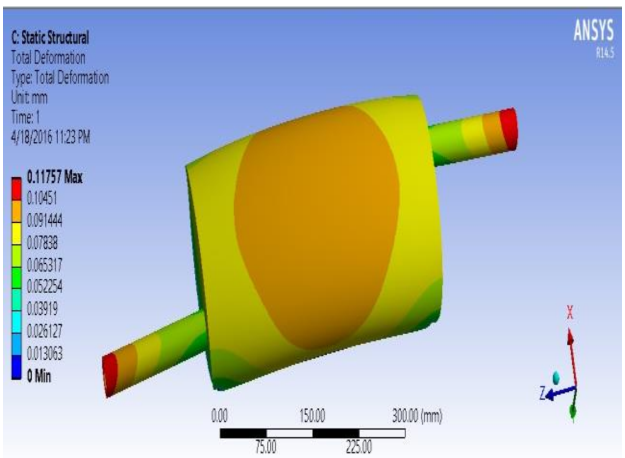


(b) For perforated model

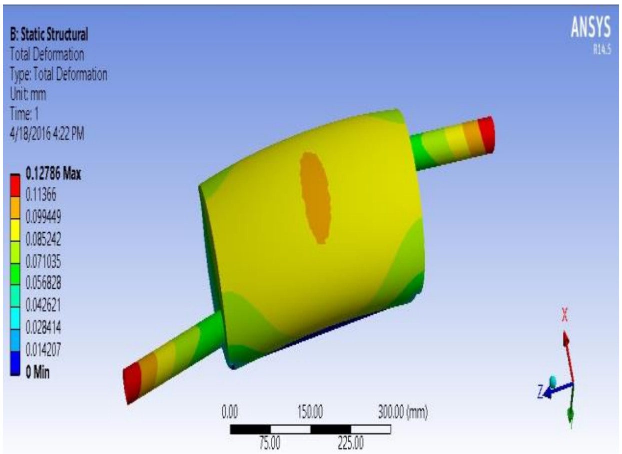
Figure 4. Equivalent elastic strain contours

Figure 3 presents the equivalent von-Mises stress contour and adopts maximum values of 3998.6 and 3483.4 MPa for the non-perforated and perforated muffler models respectively.

Equivalent elastic strain and total deformation is seen to vary nominally in both the cases from Figure 4 and 5. Maximum values of equivalent elastic strain and total deformation is found to be 0.021 mm/mm and 0.11757 mm respectively for the non-perforated muffler. On the other hand these values are found to be 0.019 mm/mm and 0.12786 mm in case of perforated model.



(a) For non-perforated model



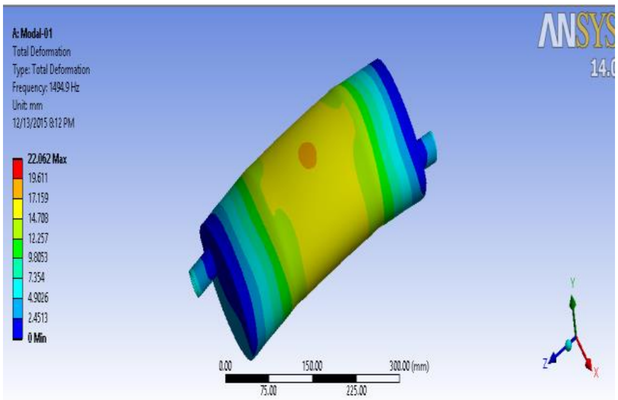
(b) For perforated model

Figure 5. Total deformation contours

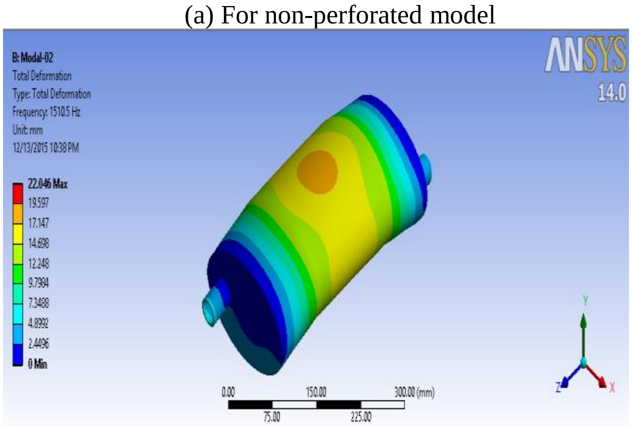
4.2 Modal analysis

Modal analysis is performed to determine the vibration characteristics like natural frequency and mode shapes of a structure under dynamic loading condition. It serves as the most fundamental type of dynamic analysis and is generally a pre-requisite for most other dynamic analyses types. The significance of this modal analysis is determination of mode shape or pattern of vibration of the structure in free vibration state at its natural frequency. It is used for standardization of a finite element model under study by determination of its natural frequency and mode shapes.

In this work six natural frequencies and mode shapes have been shown for both the muffler designs.

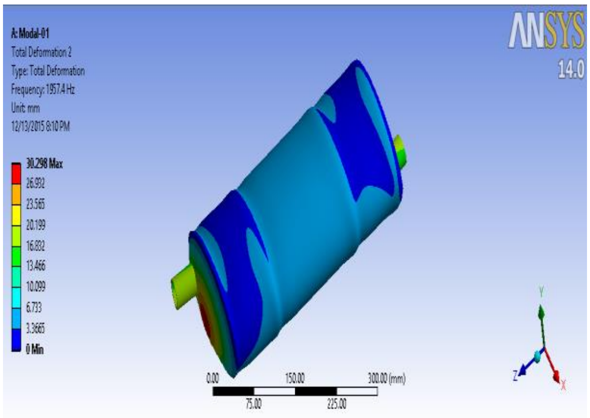


(a) For non-perforated model

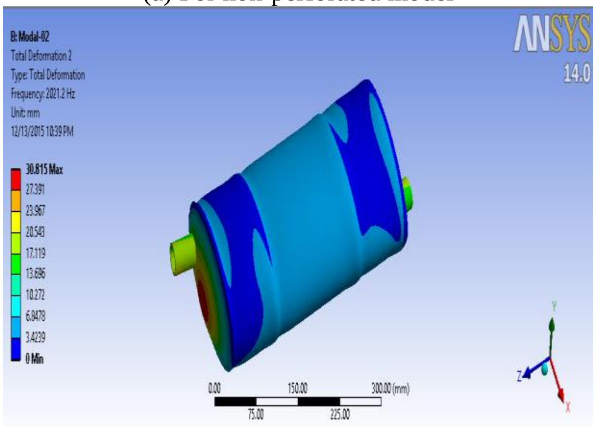


(b) For perforated model

Figure 6. First mode shape result

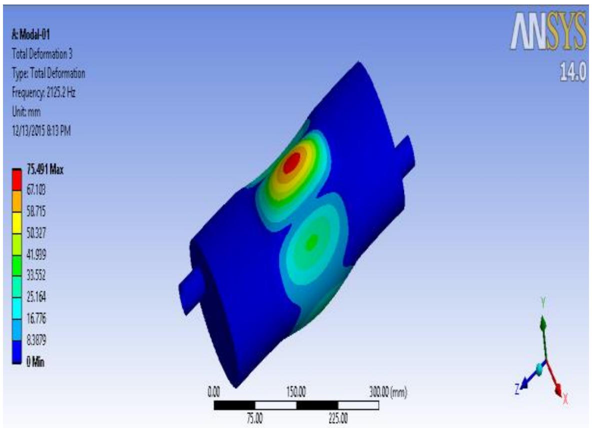


(a) For non-perforated model

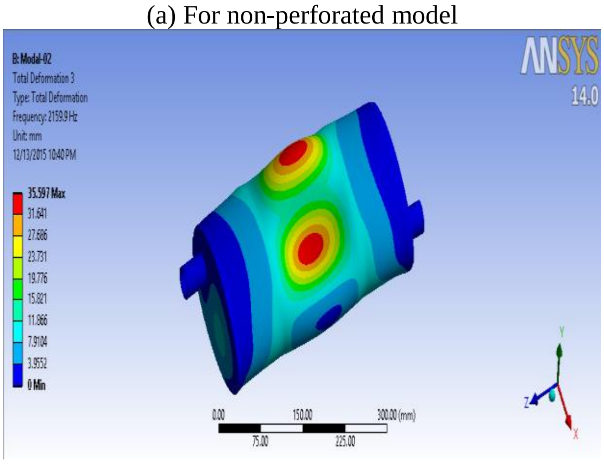


(b) For perforated model

Figure 7. Second mode shape result



(a) For non-perforated model



(b) For perforated model

Figure 8. Third mode shape result

Figure 6 shows the natural frequency for first mode shape as 1494.9 and 1510.5 Hz for the non-perforated and perforated model respectively.

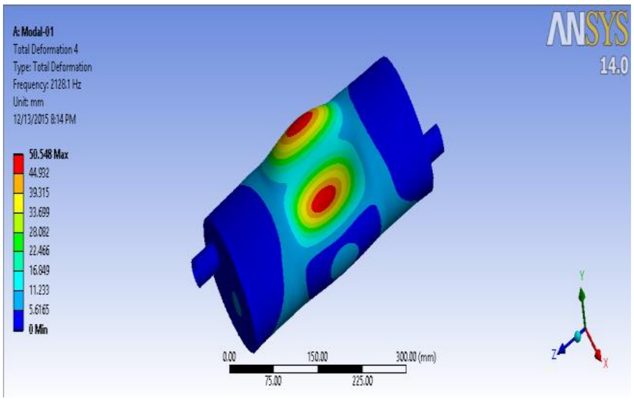
Figure 7 shows the natural frequency for second mode shape as 1957.4 and 2021.2 Hz for the non-perforated and perforated elliptical chamber muffler models respectively.

Figure 8 shows the natural frequency for third mode shape as 2125.2 and 2159.9 Hz for the non-perforated and perforated model respectively.

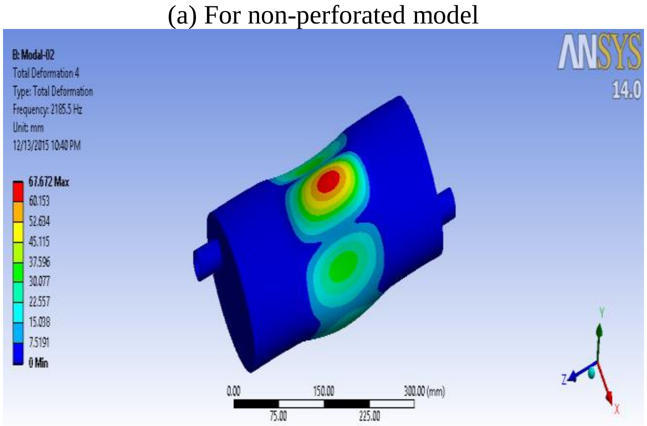
Figure 9 shows the natural frequency for fourth mode shape as 2128.1 and 2185.5 Hz for the non-perforated and perforated model respectively.

Similarly, Figure 10 shows the natural frequency for fifth mode shape as 2153.0 and 2209.2 Hz for the non-perforated and perforated model respectively.

Finally, Figure 11 shows the highest natural frequency in sixth mode shape as 2162.2 and 2259.8 Hz for the non-perforated and perforated model respectively.

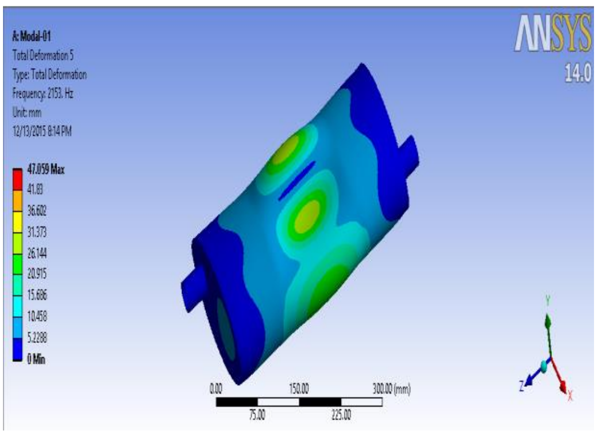


(a) For non-perforated model

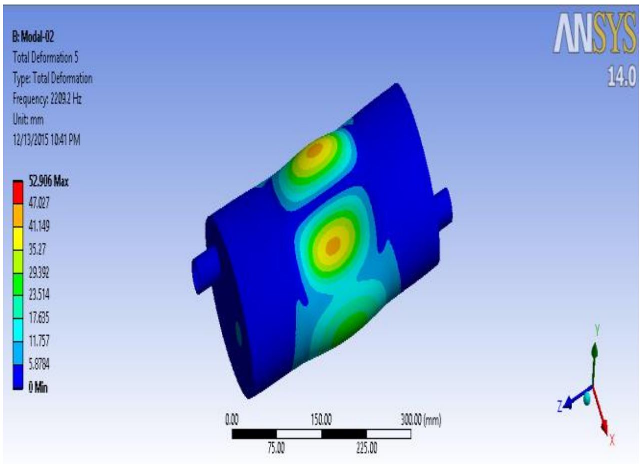


(b) For perforated model

Figure 9. Fourth mode shape result

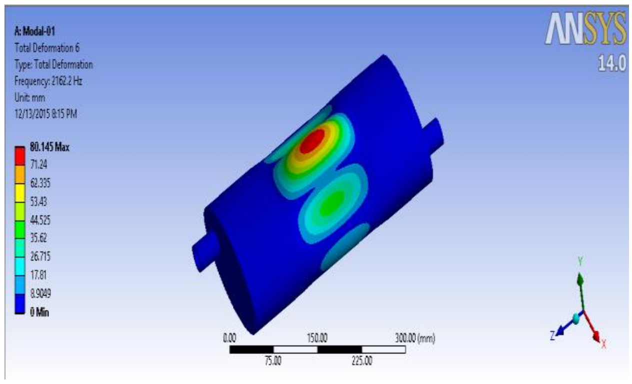


(a) For non-perforated model

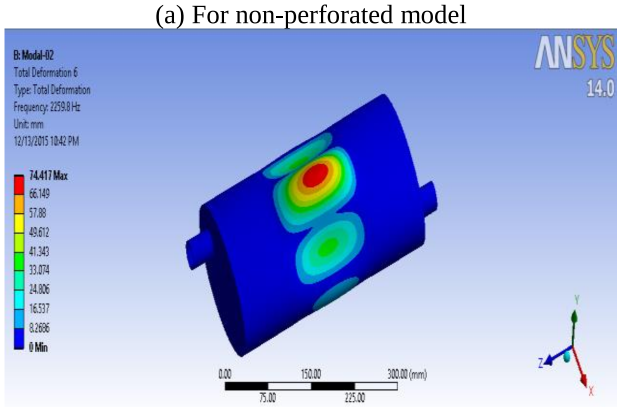


(b) For perforated model

Figure 10. Fifth mode shape result



(a) For non-perforated model



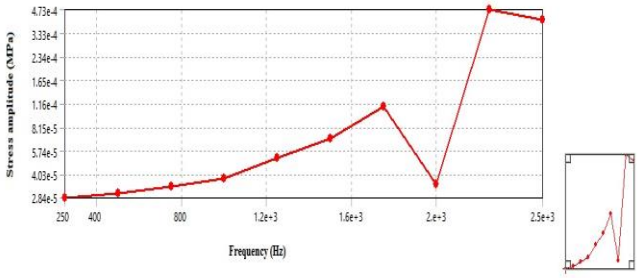
(b) For perforated model

Figure 11. Sixth mode shape result

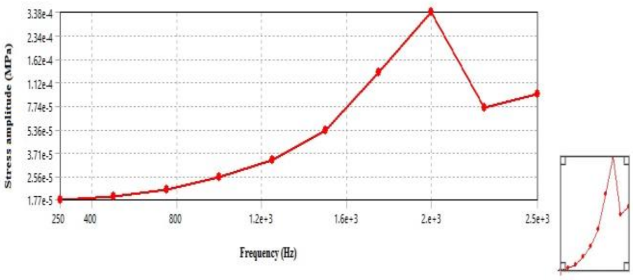
4.3 Harmonic analysis

Harmonic analysis is performed to determine the response of a structure under steady-state sinusoidal load at a particular frequency. A free vibration analysis in the form of modal analysis is run prior to harmonic analysis to obtain a better understanding of the dynamic characteristics of the structure.

Figure 12 shows the variation of stress amplitude with frequency for both the muffler designs. Cut-off frequency of the elliptical chamber is calculated as 2031 Hz [3]. As seen from the figure above, the non-perforated muffler model shows a non-linear variation of stress amplitude up to cut-off frequency whereas the perforated model shows a more or less linear variation up to cut-off frequency.



(a) For non-perforated muffler



(b) For perforated muffler

Figure 12. Stress amplitude vs. frequency for (a) Non-perforated and (b) Perforated elliptical chamber muffler

4.4 Quantitative discussion of results

Structural analysis carried out to determine the muffler behavior under static loading condition produces total deformation, equivalent von-Mises stress and equivalent elastic strain results. These are depicted in Table 2 accompanied by the percentage change in their values for both the non-perforated and perforated elliptical chamber muffler.

Table 2. Structural analysis results

Parameter of analysis	Non-perforated muffler model	Perforated muffler model	% change in perforated muffler
Total deformation (mm)	0.11757	0.12786	8.8% increase
Equivalent (von-Mises) stress (MPa)	3998.6	3483.4	12.9% decrease
Equivalent strain (mm/mm)	0.021	0.019	9.5% decrease

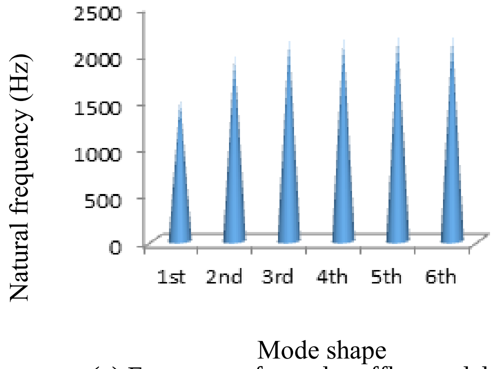
Under static loading condition, results of structural analysis show that the equivalent stress and strain are higher for the non-perforated muffler design as compared to the proposed perforated model whereas total deformation become greater for the perforated model. This might be attributed to the fact that the presence of perforation releases stress and prevents its accumulation which might be the case for a non-perforated muffler under static loading condition.

Modal analysis has been carried out to calculate up to six mode shapes and natural frequencies for both the non-perforated and perforated muffler model. Comparative results of natural frequencies for both the muffler models have been demonstrated in Table 3.

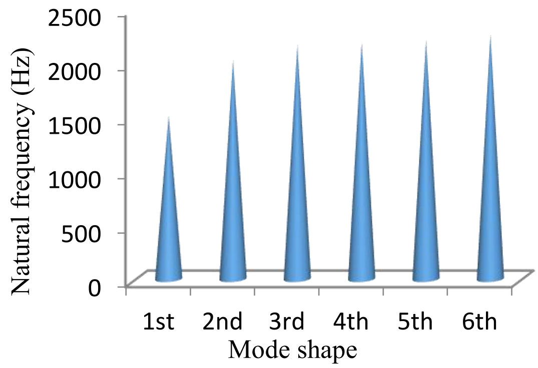
Table 3. Modal analysis results

Mode shape	Natural frequency for non-perforated muffler (Hz)	Natural frequency for perforated muffler (Hz)
1	1494.9	1510.5
2	1957.4	2021.2
3	2125.2	2159.9
4	2128.1	2185.5
5	2153.0	2209.2
6	2162.2	2259.8

The modal analysis results portray a higher set of fundamental frequencies for the perforated muffler model at every mode shape. Convergence is seen to occur after the third mode shape, beyond which, there is no further increase in frequency for both the muffler models. Figure 13 (a) and (b) shows the convergence graphs for the non-perforated and perforated muffler model respectively.



(a) For non-perforated muffler model



(b) For perforated muffler model

Figure 13. Convergence graph

5. VALIDATION OF RESULTS

The results of natural frequencies for six mode shapes from the Modal analysis of the muffler models is plotted along with experimental results obtained by Patekar et al. [11]. The authors have calculated the natural frequencies using Fast Fourier Transform (FFT) analyzer which is a broadband spectrum analyzer that works at a very fast speed. The natural frequencies and mode shapes obtained from modal analysis are seen to follow nearly the same pattern of vibration as that of the experimental frequencies as shown in Figure 14. Determination of these natural frequencies serve as an important consideration in design of silencer so as to avoid resonance.

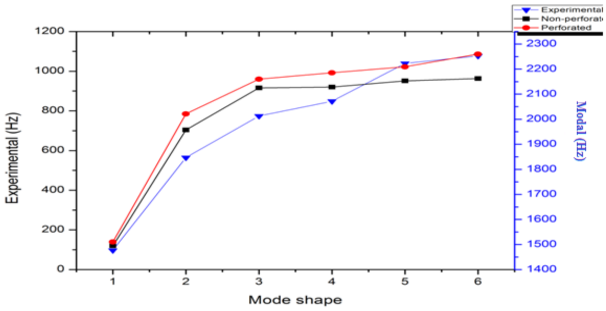


Figure 14. Mode shape versus frequency graph for modal and experimental results

6. CONCLUSIONS

Comparison of results of structural analysis renders a decreased equivalent elastic strain and equivalent von-Mises stress for the proposed design and increased total deformation. Deformation has been found to increase by 8.8 % whereas strain and stress values decreased by 9.5% and 12.9 % in the modified design. This implies that under static loading condition, the proposed perforated muffler model has more strength than the non-perforated design from equivalent von-Mises stress values. As such the perforated elliptical chamber muffler model gives efficient noise reduction without any loss of structural strength.

Modal analysis results for both the muffler models follow nearly the same pattern of vibration at their respective natural frequencies. These natural frequencies must be kept in mind while designing the muffler so as to avoid resonance. It can be observed that the maximum value of total deformation under dynamic loading condition occurs at the sixth mode shape. Thus at an increased natural frequency the deformation in the muffler is seen to increase non-linearly as observed from the study of the six mode shapes. The maximum total deformation is comparatively higher for the non-perforated muffler model at a frequency of 2162.2 Hz.

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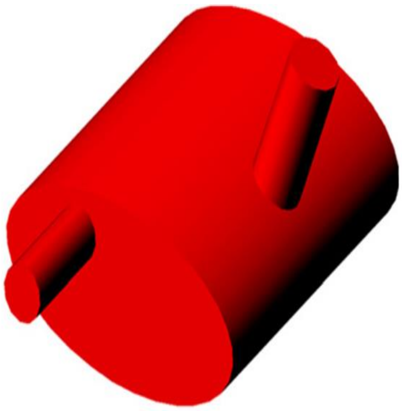
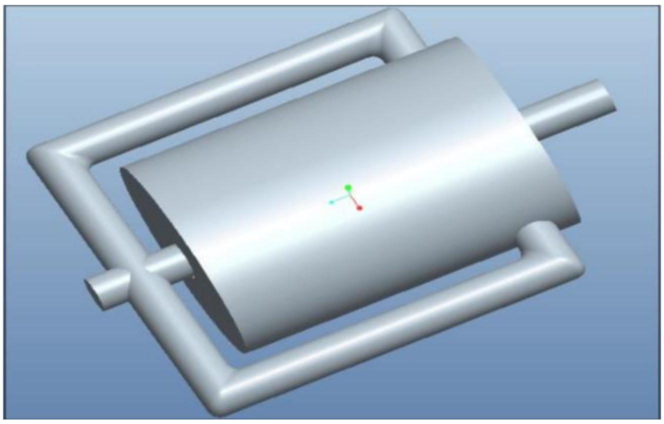
APPENDIX



Ref. Fig. A: CAD model of elliptical chamber muffler having an end inlet and side outlet port



Ref. Fig. B: CAD model of extra divided inlet tube along with extended inlet and outlet