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"Performance Enhancement of Automotive Silencer Using Finite Element Analysis"

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ABSTRACT : *This paper postulates the design and modification of silencer in order to reduce the vibration which is secondary source of noise generation, by considering the specified material properties and FEM package. The experimental analysis is carried out with the help of FFT analyzer to evaluate the natural frequency and to distinguish it from the working frequency to avoid resonating condition. The dimensions of the existing model of the silencer are reffered as benchmarking dimensions to create modified model. Frequency responce analysis is carried out to study behaviour of silencer at different frequencies and free free analysis is done with the help of NASTRAN.*

KEYWORDS : *CAD, FFT, FRA, Modal analysis, NASTRAN*

I. INTRODUCTION

The most important objective of silencer is to reduce the vibration and noise coming from engine. When the natural frequency of any object matches to the operating frequency of the same object then resonance, and resonance is necessarily to be avoided. Resonance leads to catastrophic failures. Therefore every machine or equipment should be properly addressed for overcoming vibration problems before installing. It is therefore necessary to study the behavior of silencer by analyzing the vibration modes and the response of vibrations by its sources. Modal analysis will be done for existing model on the basis of modal analysis, we can suggest weight optimization if natural frequencies are higher than the engine frequencies which is basically considered up to ~70 Hz followed by Frequency response analysis.

NEED FOR ANALYSIS & INDUSTRIAL RELEVANCE

The durability of that part of the system is therefore crucial customer demands for the comfort and long product kind guarantee also for the exhaust system as a whole are additional reasons for the increasing importance for design engineers to be able to predict ,describe and access the dynamics of various system design proposals during product develop. The Automobile silencer under study belongs to a popular 4-Wheeler manufacturer in India with the rated HP of the engine up to @ 51kw; 69HP .The exhaust gases coming out from engine are at very high speed and temperature. Silencer has to reduce noise, vibrations. While doing so it is subjected to thermal, vibration and fatigue failures which cause cracks. So it is necessary to analyze the vibrations which would further help to pursue future projects to minimize cracks, improving life and efficiency of silencer.

RELATED THEORIES AND PRACTICES

The best method to describe the natural characteristic such as frequency, damping, model shapes and its dynamic properties is Model analysis. It involves process of determining the modal parameters of a structure in order to construct a modal model of the response. Both the techniques like theoretical and experimental are different technologies for solving noise and vibration problem. In this experiment Modal analysis will be done for existing model on the basis of modal analysis, we can suggest weight optimization if natural frequencies are higher than the engine frequencies which is basically considered up to ~70 Hz followed by Frequency response analysis. If natural frequencies are not within the acceptable limit then we have to shift the natural frequencies out of concerned zone by suggesting some modifications (Change in geometry or mass or boundary conditions) and then frequency response analysis will be done at first resonance frequency to check the stress levels, stress criterion should also satisfy.

II. EXPERIMENTAL VALIDATION-

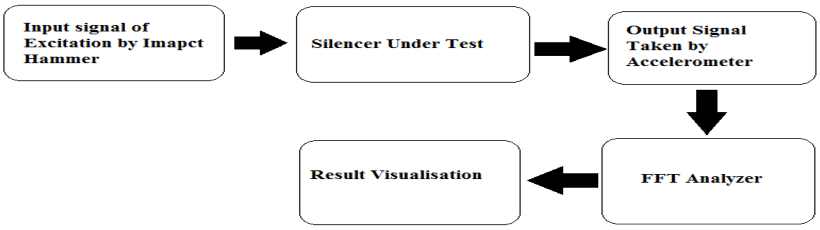
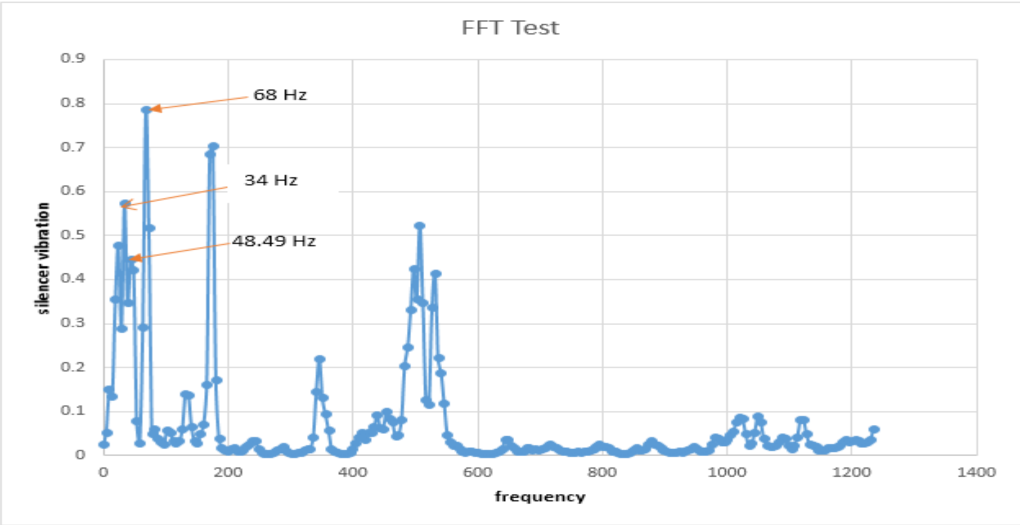


Fig1- Block diagram for experimental setup.

Fig2- Position of Accelerometer

Fast Fourier Transform (FFT) analyzer is used to do the experimental validation. FFT analyzer validates the input signal, computes the magnitude of its sine and cosine components and displays the spectrum of the measured frequency components. This method carries advantage of being fast and accurate. The method is faster than traditional analog spectrum analyzers as faster than traditional analog spectrum analyzers. The experimental validation is carried out at authorized company the result of the same is,



Graph1- FFT test report.

III. ANALYTICAL APPROACH

Design study of existing silencer
Geometry modeling of existing silencer
Modal Analysis of existing silencer
Modification in the geometry to avoid resonance & hence vibration
Frequency Response Analysis of both Silencer

Element Size	10mm
Material	Steel
Mesh type	Solid mesh
Density	7850 kg/m^3
Thickness of plate	2 mm
Material Endurance limit	170 N/mm^2

Table 1- Material Properties

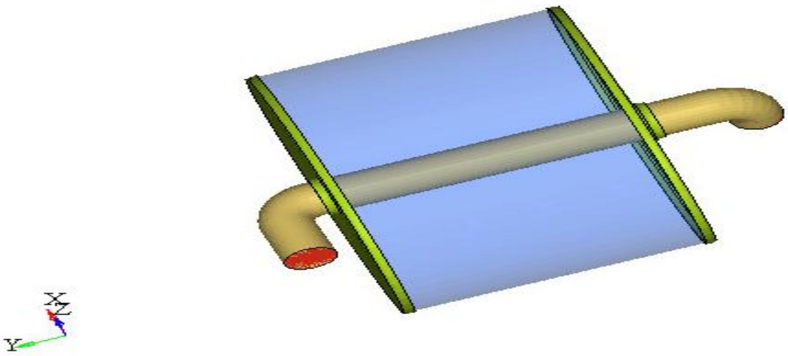


Fig3.-Silencer Existing Model

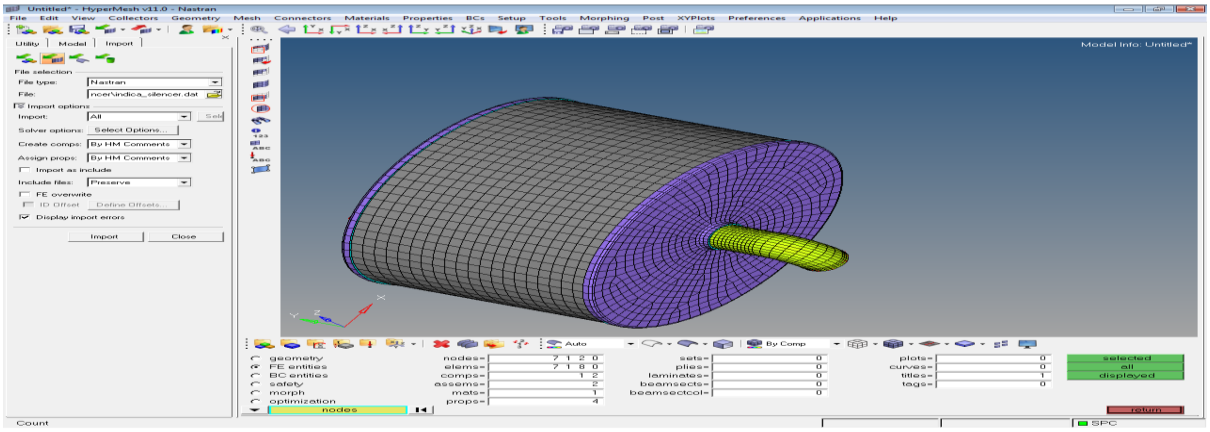


Fig4 - Meshing of silencer model.

After generating the model of silencer then analysis is done by NASTRON. The result obtained by modal analysis for first four natural frequencies are determined and tabulated as follow:

Mode	1st	2nd	3rd	4th
Frequency (Hz)	35	43	52	100

Table2 -Natural Frequency at first 4 modes for Existing silencer.

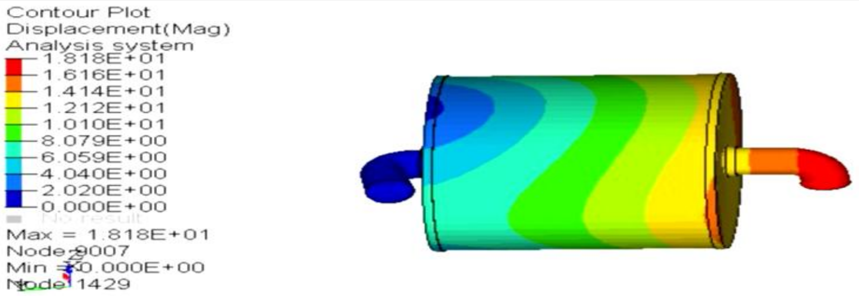


Fig5- 1st mode of frequency for 35Hz

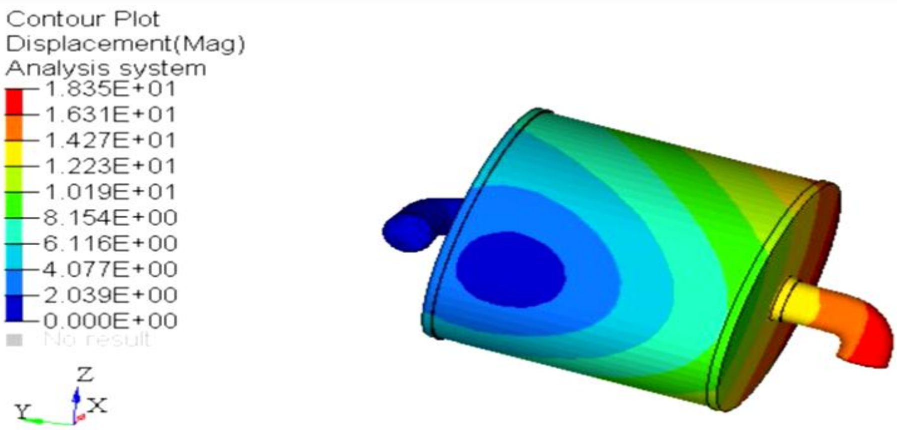


Fig6- 2nd mode of frequency 43Hz

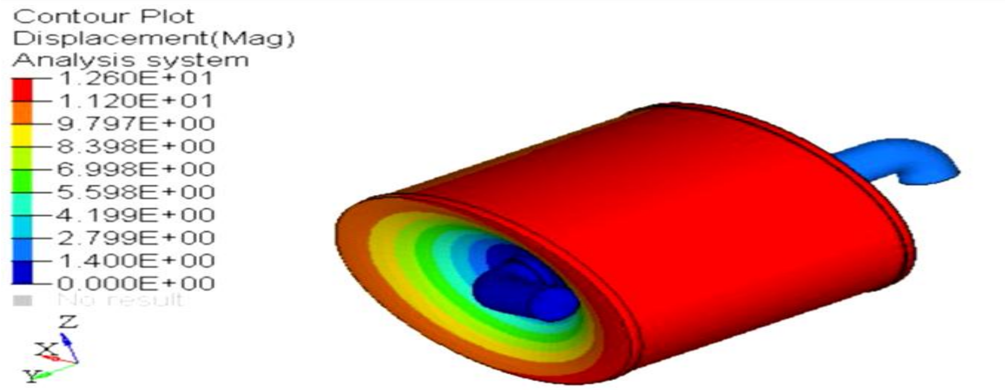


Fig7- 3rd mode of frequency 52Hz

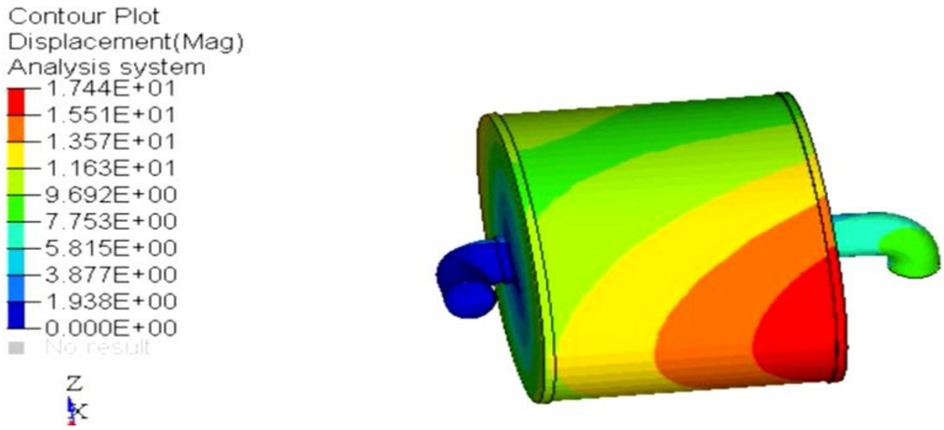


Fig8-4th Mode of Existing Silencer 100Hz

IV NEED OF MODIFICATION OF EXISTING SILENCER & RESULTS

According to JIS D 1601 Vibration Testing for Automobile Silencer the damageable frequencies are 33Hz and 67Hz which are necessarily to be reduced as these causes more vibration hence noise in the exhaust system. To reduce the vibration and to shift the frequency the stiffener is added in a bead pattern. After generating the modified model of silencer then analysis is done by NASTRON. The result obtained by modal analysis for first three natural frequencies are determined and tabulated as follow:

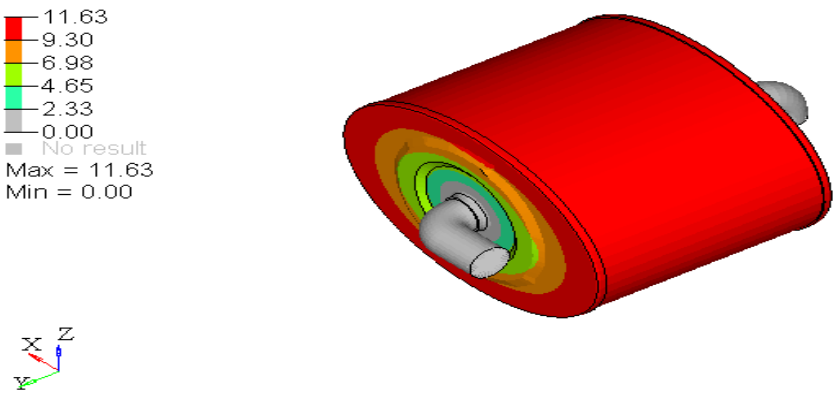


Fig9- 1st mode of modified Silencer

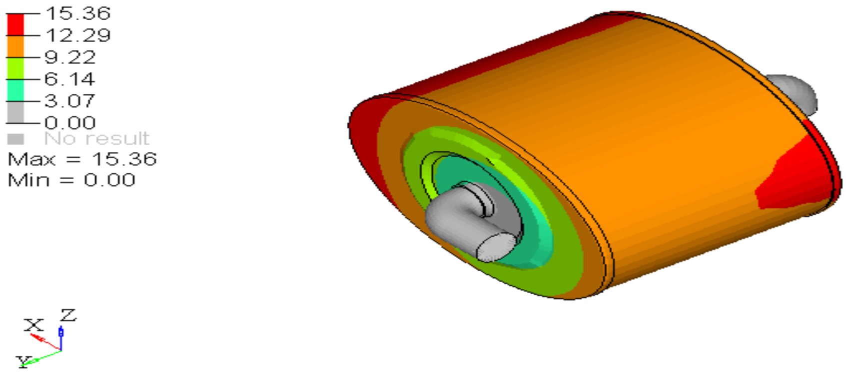


Fig 10- 2nd mode of modified Silencer

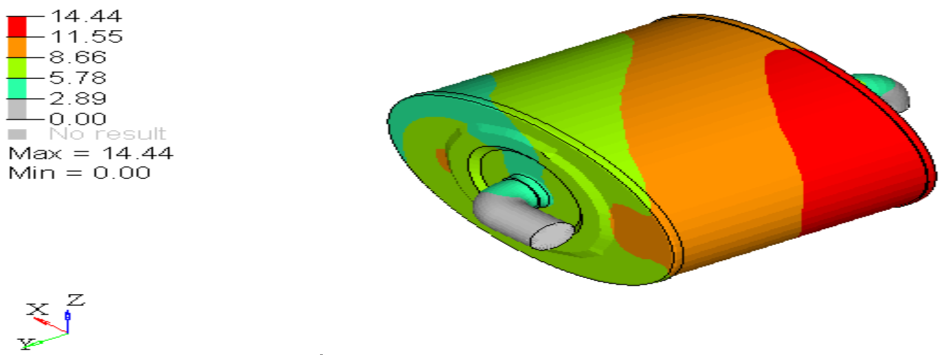


Fig 11- 3rd mode of modified Silencer

Mode	1st	2nd	3rd
Frequency (Hz)	103	140	380

Table 3-Natural Frequency at first 3 modes for modified silencer.

V. COMPARISON OF FREQUENCY RESPONSE ANALYSIS OF EXISTING AND MODIFIED MODEL

According to JIS D 1601 Vibration Testing for Automobile Silencer the damageable frequencies are **33Hz and 67Hz** so it's necessary to do FRA for both frequency and for both Existing and Modified models.

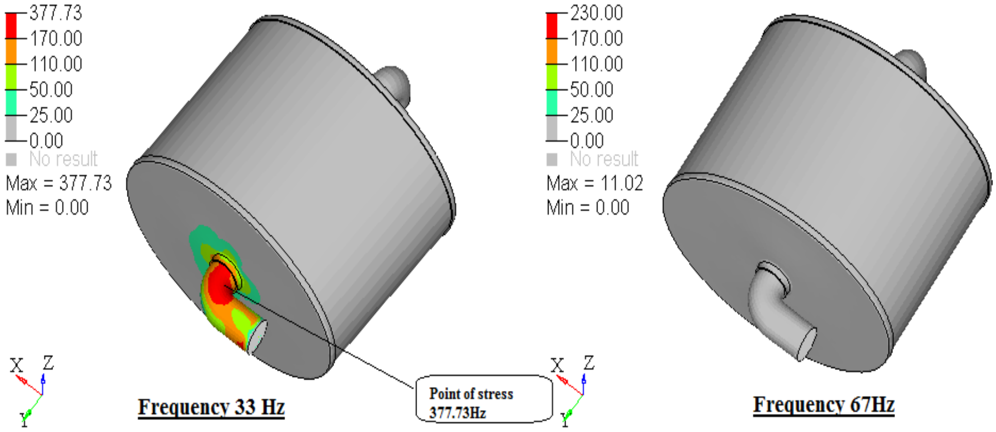


Fig-12 -FRA(X- Axis) Existing at 33Hz & 67Hz

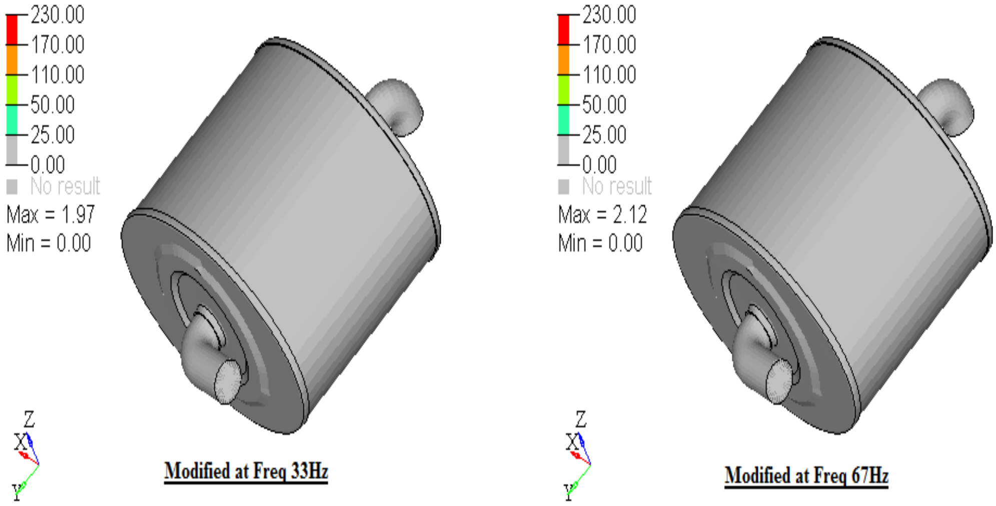
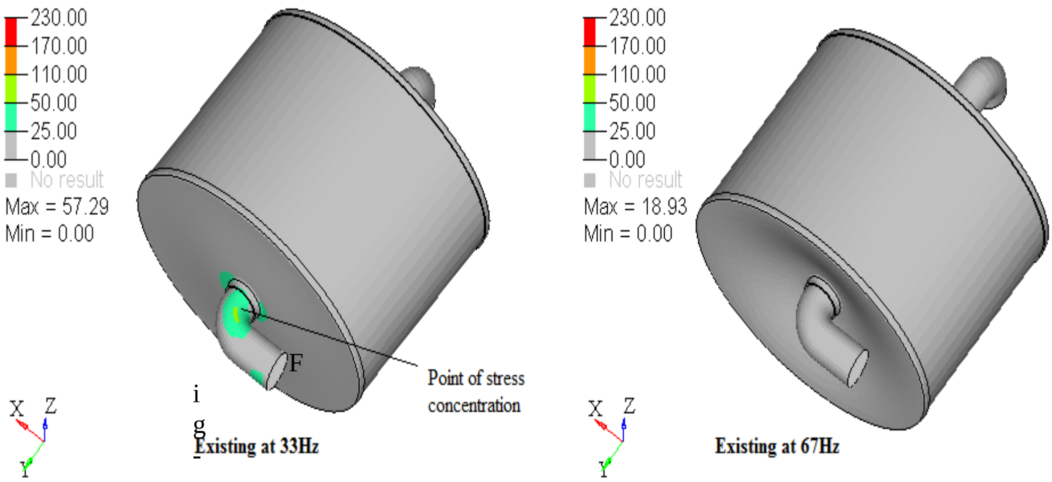


Fig 13-FRA(X- Axis) Modified at 33Hz & 67Hz



14 - FRA (Y-Axis) Existing at 33Hz & 67Hz

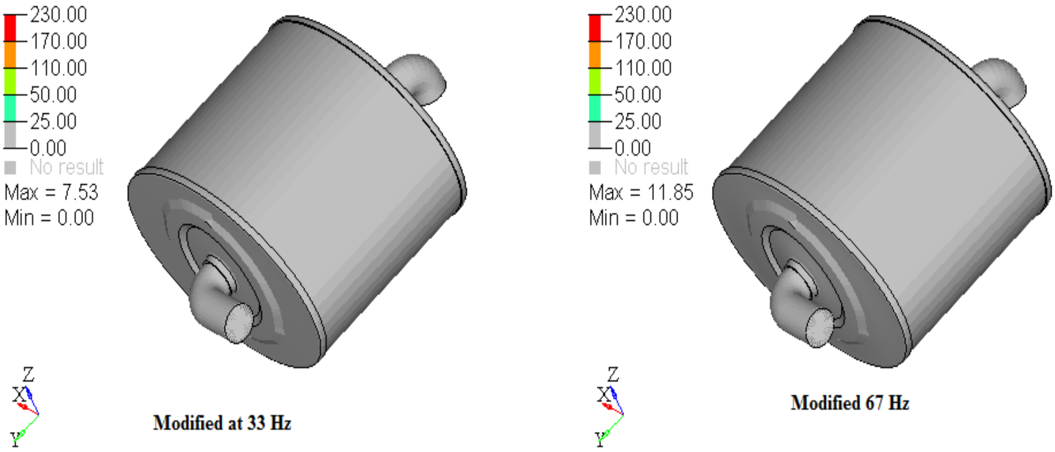
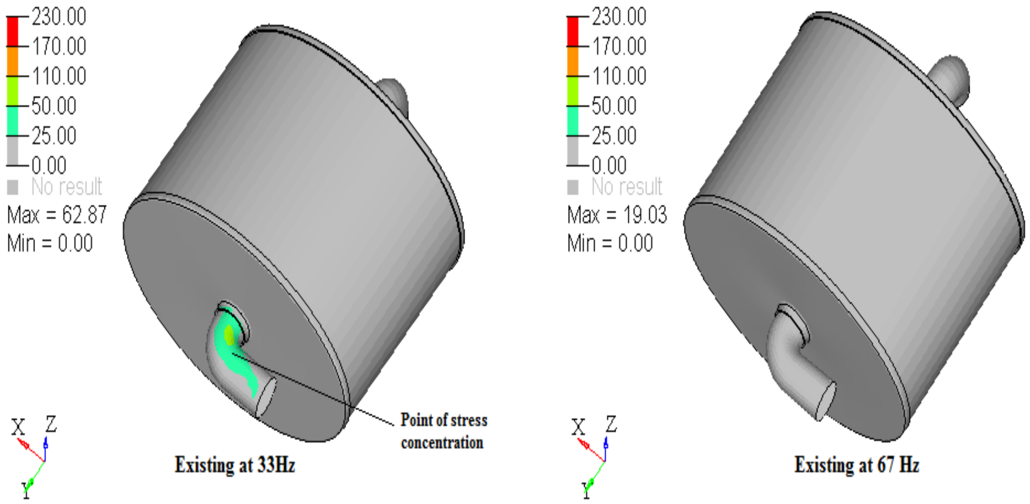


Fig 15- FRA (Y-Axis) Modified at 33Hz & 67Hz



At 33Hz Stress in N/mm^2	At 67 Hz Stress in N/mm^2

Fig- 16-FRA (Z-Axis) Existing at 33Hz & 67Hz

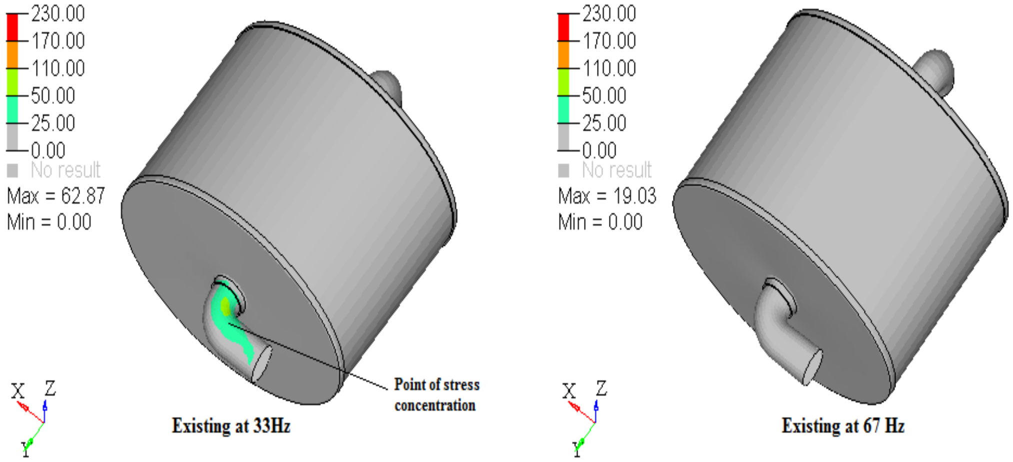


Fig 17- FRA(Z-Axis) Modified at 33Hz & 67Hz

IV. RESULTS AND DISCUSSION

We know as per the design the maximum allowable stress on silencer is around 170 N/mm ^{2}. The stresses in the silencer during FRA of existing silencer at 33Hz and 67 Hz are tabulated below-

At 33Hz Stress in N/mm^2	At 67 Hz Stress in N/mm^2
377.73	11.07
57.29	18.93
62.87	19.03

Table 3- The stresses in existing Silencer

The stresses in the silencer during FRA of modified silencer at 33Hz and 67 Hz are tabulated below-
The table shows that the stresses generated in modified silencer are within the limit
Table 4- The stresses in modified Silencer

1.97	2.12
7.53	11.98
2.29	2.47

VI. CONCLUSION

The silencer natural frequencies have been calculated by using the NASTRAN and by FFT analyzer. The dynamic performance is increased by changing design i.e. by adding stiffener in the form of bead in the modified silencer. The difference between experimental and analytical method is 2.94%.The stresses induced in the modified silencer are less than permissible yield strength of material i.e 170 N/mm^2

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