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Design, Analysis & Optimization of Muffler for Four Stroke Petrol Engine Motorcycle

Anjan Pal, Mr. R. R. Kulkarni, Dr. P. A. Makasare, Dr. L .V. Kamble
Department of Mechanical Engineering, Siddhant College of Engineering, Sudumbre, Pune – 412109

Abstract: Objective: The exhaust pipe is subjected to several stresses, most of which are due to vibration. Particular attention should be given to gas forces which will induce vibration. These vibrations will then induce a fatigue life to the system. It is therefore necessary to study the fatigue behavior of the exhaust pipe by analyzing the vibration modes and the response of vibrations by its sources.
Methods: The vibrations of silencer are affecting the performance of silencer and it is uncomfortable to operators. So, it is necessary to analyze the vibrations which would further help to minimize cracks, improving life and efficiency of silencer. the main goal of this project will be to design a new automobile exhaust piper muffler (silencer) is to increase the durability of its life. Decrease the weight, and reduce the manufacturing cost with efficient working condition. design, analyze the model using ANSYS workbench static structural and model analysis for vibration study. If muffler or silencer part impacts high vibrations, or stress then topology will be conducted to solve the high stress concentration and vibrational impacts.
Keywords: Automobile exhaust system; muffler; noise; vibration and modal analysis.

I. INTRODUCTION

One of the objectives when designing a new automobile exhaust pipe is to lengthen its durability period, which can be measured in terms of its life span and mileage. The exhaust pipe is subjected to several stresses, most of which are due to vibration. Particular attention should be given to gas forces which will induce vibration. These vibrations will then induce a fatigue life to the system. It is therefore necessary to study the fatigue behavior of the exhaust pipe by analyzing the vibration modes and the response of vibrations by its sources.
The vibration of a system involves the transfer of its potential energy to kinetic energy and of kinetic energy to potential energy, alternately. If the system is damped, some energy is dissipated in each cycle of vibration and must be replaced by an external source if a state of steady vibration is to be maintained.

A. Overview

The main goal of this project is to study the vibrational impact caused due to the vehicle moving on irregular road surface and also due to exhaust gas pressure variations in two-wheeler muffler. Mufflers is an important component in vehicle, without muffler or silencer one can't assume an engine.
After expansion in the engine exhaust gas produced containing harmful gases is exhausted in a long hollow pipe called muffler at back side of the vehicle.
For time being these mufflers have been optimized in different shape, size, and material etc. the main goal of this project will be to design a new automobile exhaust piper muffler (silencer) is to increase the durability of its life. Decrease the weight, and reduce the manufacturing cost with efficient working condition.
In this project I'm going to design, analyze the model using ANSYS workbench static structural and model analysis for vibration study. If muffler or silencer part impacts high vibrations, or stress then topology will be conducted to solve the high stress concentration and vibrational impacts.
Vibrations are measured in 3 different phases which are classified as: Frequency, amplitude and Phase are the three major characteristics which are used to describe a oscillation in the part or component (or vibrations).

There are 3 types of Vibration:
- Free or Natural.
- Forced and.
- Damped Vibration.

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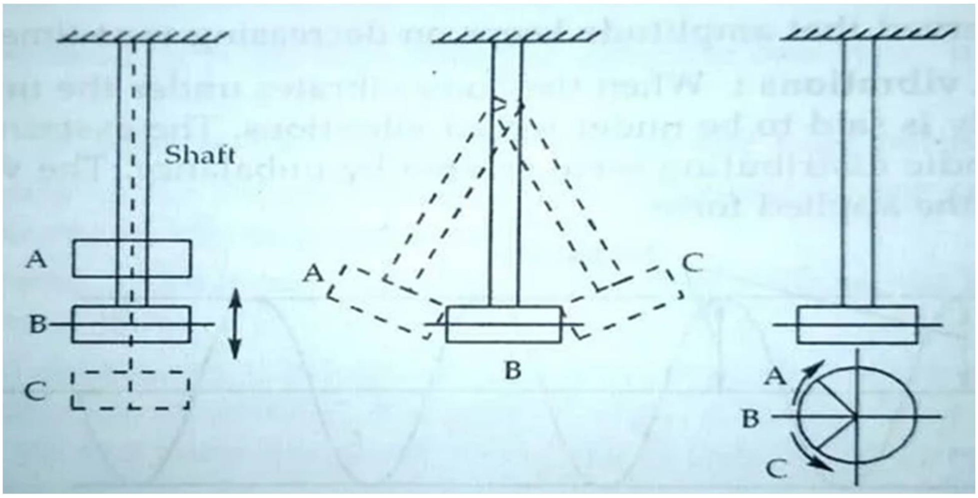


1) *Longitudinal Vibrations:* In this, the particles of the shaft or disc move parallel to the axis of the shaft as shown in the above diagram. In this case, the shaft is elongated and shortened alternately thus executing the tensile and compressive stresses alternately on the shaft.
2) *Transverse Vibrations:* In this, the particles of the shaft or disc move perpendicular to the axis of the shaft as shown in the above diagram. Here the shaft is straight and bent alternatively and hence bending stresses are induced in the shaft.
3) *Torsional Vibrations:* In this, the particles of the shaft or disc move in a circle about axis of the shaft as shown in the above diagram. Here the shaft is twisted and untwisted alternatively and hence torsional shear stress is induced in the shaft.

B. Need for Analysis
The Automobile silencer under study belongs to a popular 2-Wheeler manufacturer in India with the rated HP of the engine. 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.

C. Problem Statement
Vibrations are the real problem of machines they encounter almost in all components connected to dynamic constraint. Whenever a part or component undergoes or subjected to a periodic motion the part impacts to a vibration. Following are the causes of when vibration induces.
1) Unbalance: whenever vibration induces in a certain part, it disrupts the balance of it because of its higher tendency of frequency.
2) Resonance: it is an effect of collision or noise.
3) It effects in loosing of jointed parts, like nut & bolt loosening
4) Bearing damage: vibrations can damage the bearings by misaligning it.

For future esthetic condition high speed vehicles play a definable role, while a vehicle moves at a top speed on an irregular surface it under goes into vibrational impact caused due shock in road. This in term can result a muffler to fail due to high temperature exhaust has moving inside the silencer, also due to the road terrain. Muffler may undergo to failure or cracks or loosening from the engine manifold. To prevent this one must investigate the silencer under natural forces.

II. PROPOSED SYSTEM

A. Formulations
1) Equilibrium Method
It is based on the principle that whenever a vibratory system is in equilibrium, the algebraic sum of forces and moments acting on it is zero recordings to D'Alembert's Principle that the sum of inertia forces and external forces on a body in equilibrium must be zero.

Let, Δ = static deflection, k = Stiffness of the spring

Inertial force = ma (upwards, a = acceleration)

Spring force = kx (upwards)

So the equation becomes

$ma + kx = 0$

$\Rightarrow \omega n = \sqrt{(k/m)}$

Linear frequency $fn = (1/2\pi)\sqrt{(k/m)}$

Time period $T = 1/fn = 2\pi\sqrt{(m/k)}$

2) Energy Method

In a conservative system (system with no damping) the total mechanical energy i.e. the sum of the kinetic and the potential energies remains constant

d/dt (K.E+ P.E.) = 0

3) Rayleigh's Method

In this method, the maximum kinetic energy at the mean position is made equal to the maximum potential energy(or strain energy) of the extreme position.

The displacement of the mass 'm' from the mean position at any instant is given by

$a+\omega n^2 x = 0$

$x = A \sin\omega n t + B \cos\omega n t$

Let $A = X \cos \varphi$; $B = X \sin \varphi$

$x = X \sin(\omega n t + \varphi)$

Velocity, $V = X\omega n \sin[\pi/2 + (\omega n t + \varphi)]$

Acceleration, $f = X\omega n^2 \sin[\pi + (\omega n t + \varphi)]$

These relationships indicate that the velocity vector leads the displacement vector by $\pi/2$.

Acceleration vector leads the displacement vector by π.

Consider, 'm' = man of the spring wire per unit length

l = total length of the spring wire m1 = m'l

KE of the spring = 1/3 * KE of a mass equal to that of the spring moving with the same velocity as the free end.

$fn = (1/2\pi) \sqrt{(s/(m+(m1/m)))}$

$fn = (1/2\pi) \sqrt{g/\Delta}$

B. Damped Vibrations

When an elastic body is set in vibratory motion, the vibrations die out after some time due to the internal molecular friction of the mass of the body and the friction of the medium in which it vibrates. The diminishing of the vibrations with time is called damping.

Shock absorbers, fitted in the suspension system of a motor vehicle, reduce the movement of the springs, when there is a sudden shock.

It is usual to assume that the damping force is proportional to the velocity of vibration at lower values of speed and proportional to the square of velocity at high speeds.

$F\propto V$ at a lower speed and $F\propto V2$ at a higher speed

1) Damped

C = damping coefficient (damping force per unit velocity)

ωn = frequency of natural un-damped vibrations

$a + (c/m)v + (k/m)x = 0$

$\alpha 1,2 = -(c/2m) \pm \sqrt{[(c/2m)^2-(k/m)]}$

Degree of dampness:

$=(c/2m)^2/(k/m)$

Damping factor:

$\xi = c/(2\sqrt{km})$

2) Damping coefficient

$c = 2\xi\sqrt{km} = 2\xi m\omega n = 2\xi k/\omega n$

When $\xi = 1$, damping is critical, thus under critical damping conditions

$\xi = 2\sqrt{km} = 2m\omega n = 2k/\omega n$

$\xi = c/cc$ = Actual damping coefficient / Critical damping coefficient

$\xi > 1$; the system is over damped

$\xi < 1$; the system is under damped

$\omega d = \omega n\sqrt{(1-\xi^2)}$

In a critically damped system, the displaced mass return to the position of rest in the shortest possible time without oscillation. Due to this reason large guns are critically damped so that they return to their original positions in minimum possible time. An un-damped system ($\xi = 0$) vibrates at its natural frequency which depends upon the static deflection under the weight of its mass.

At critical damping ($\xi = 1$); $\omega d = 0$ and $Td = \infty$. The system does not vibrate and the mass 'm' moves back slowly to the equilibrium position. For over damped system ($\xi > 1$) the system behaves in the same manner as for critical damping

C. Present Theories and Practices

- Modal analysis is method to describe a structure in terms of its natural characteristics which are frequency, damping and Modal shapes and its dynamics properties.
- Modal analysis involves process of determining the modal parameters of a structure to construct a modal model of the response.
- Theoretical and Experimental Modal Analysis (EMA) have been very separate engineering technologies aimed for solving noise and vibration problems.
- The modal parameters may be determined by analytical means, such as finite element analysis and one of the common reasons for experimental modal analysis is the verification/correction of the results of the software approach (model updating).

1) Materials

Some of the basic materials used for manufacturing mufflers are Cast iron, stainless steel, mild steel / carbon steel. Recent trends towards light weight concepts, to increase the engine efficiency weight reduction is mandatory, cost reduction and better performance, designers are progressing towards sheet metals.

- steel,
- aluminum,
- titanium
- carbon fiber
- Glass fiber etc.

The above shown materials are the base use of muffler, which is made out of.

2) Properties of Materials

Grey cast density = 7.20 g/cm^3
Stainless steel = 7.65-8.03 g/cm^3
Steel = 7.86 g/cm^3
Titanium = 4500 g/cm^3

VOCADO PE-1 Hero CBZ Slip-on Exhaust System (Mild Steel)

General

Brand	: VOCADO
Model Number	: CBZBYKEXTSYSTM5819
Material	: Mild Steel
Type	: Slip-on Exhaust System
Vehicle Brand	: Hero
Vehicle Model Name	: CBZ
Vehicle Model Year	: 2018
Series	: PE-1
Model Name	: Bike Exhaust System for Hero CBZ
Weight	: 2 kg

Figure 2.1 Muffler VOCADO PE-1 Hero CBZ Slip-on Exhaust System (Mild Steel)

D. Hero Splendor Plus Dimensions

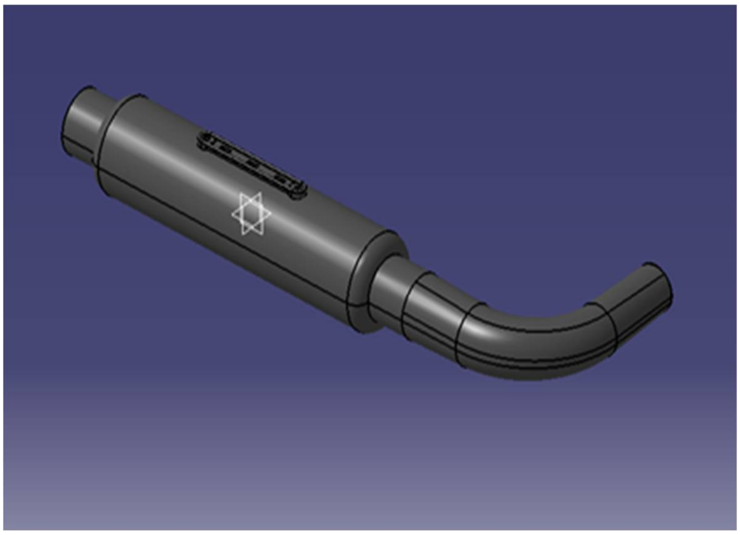
The Hero Splendor Plus Dimensions 1970 mm in length, 720 mm in width and 1040 mm in height with a wheelbase of 1230 mm. By knowing the Hero Splendor Plus dimension, you can be clear about the minimum space, which you require to park the bike in your garage.

Dimensions	**mm**	**cm**	**inches**	**feet**
Length	1970	197	77.56	6.46
Width	720	72	28.35	2.36
Height	1040	104	40.94	3.41
Wheelbase	1230	123	48.43	4.04
Ground Clearance	159	15.9	6.26	0.52

III. DESIGN AND ANALYSIS

A. Design Procedure

1) CATIA v5 Software is used to generate the CAD model of the Muffler
2) In CATIA v5 software, generative shape design with plane-based sketch has been prepared and converted to 3D using sketch-based tools.
3) The dimensions of 3D muffler for the designed model has been taken from the research paper.
4) Then the final 3D model is converted to IGS or STP for ANSYS importation.
5) Finally drafting of the 3D product is extracted from the drafting option using the conversion method.



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B. Analysis of ASTM A36 Mild Steel

1) Iteration 1

- Unit System : Metric (mm, kg, N, s, mV, mA) Degrees rad/s Celsius
- Angle : Degrees
- Rotational Velocity : rad/s
- Temperature : Celsius

ASTM A36 Structural Mild Steel material has been selected for the design consideration. This plate comes in several thickness of sizes varying from 1- 10 mm or more.

2) Geometry and Definition

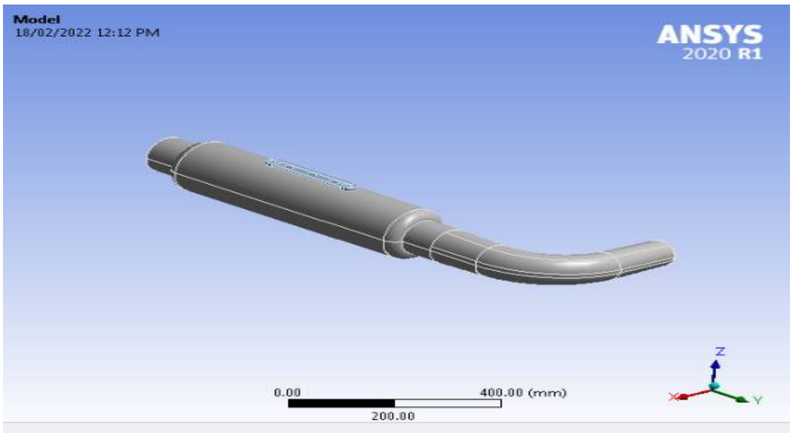

Figure 3.2 Muffler ANSYS IMPORT

Object Name	silencer| Part Body	clamper| PartBody
State	Meshed	
Graphics Properties		
Visible	Yes	
Transparency	1	
Definition		
Suppressed	No	
Stiffness Behavior	Flexible	
Coordinate System	Default Coordinate System	
Reference Temperature	By Environment	
Treatment	None	
Material		
Assignment	Structural Steel	
Nonlinear Effects	Yes	
Thermal Strain Effects	Yes	
Bounding Box		

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Length X	376.21 mm	30. mm
Length Y	1058.1 mm	230.41 mm
Length Z	152.4 mm	13.5 mm
Properties		
Volume	9.4325e+005 mm³	40007 mm³
Mass	7.4045 kg	0.31406 kg
Centroid X	-23.443 mm	-2.8303e-004 mm
Centroid Y	159.71 mm	1.9717e-003 mm
Centroid Z	4.1136e-003 mm	80.835 mm
Moment of Inertia Ip1	7.4624e+005 kg·mm²	1660. kg·mm²
Moment of Inertia Ip2	49606 kg·mm²	28.351 kg·mm²
Moment of Inertia Ip3	7.6496e+005 kg·mm²	1681.9 kg·mm²
Statistics		
Nodes	62304	3036
Elements	31092	1357
Mesh Metric	None	

Table 3.1 Geometry Definition

3) Mesh Generation

Object Name	Patch Conforming Method	Body Sizing
State	Fully Defined	
Scope		
Scoping Method	Geometry Selection	
Geometry	2 Bodies	
Definition		
Suppressed	No	
Method	Tetrahedrons	
Algorithm	Patch Conforming	
Element Order	Use Global Setting	
Type		Element Size
Element Size		10.0 mm
Advanced		
Defeature Size		Default
Behavior		Soft
Statistics		
Nodes	65340	
Elements	32449	

Table 3.2 Mesh configuration

Figure 3.3 Mesh creation

Model (D5)

Object Name	Fixed Support
State	Fully Defined
Scope	
Scoping Method	Geometry Selection
Geometry	6 Faces
Definition	
Type	Fixed Support
Suppressed	No

Table 3.3 Boundary condition

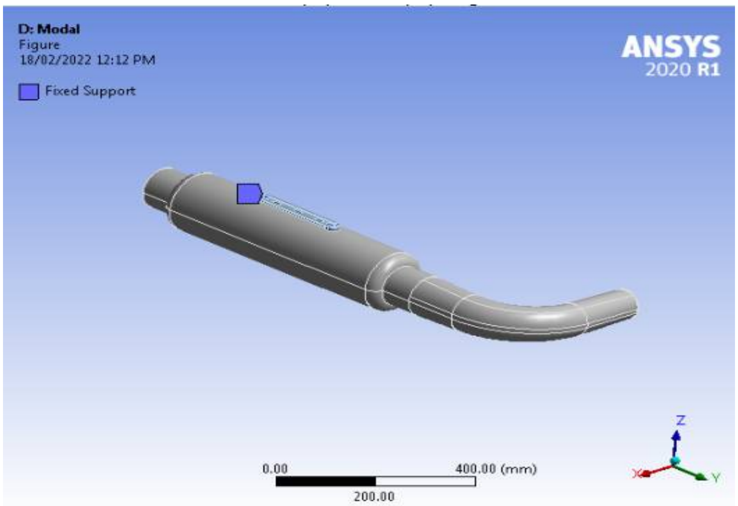
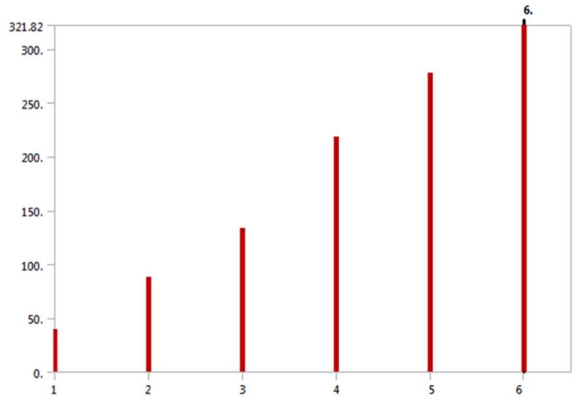


Figure 3.4 Fixed support

4) Solution (D6)



Graph 3.1 Result frequency model

Mode	Frequency [Hz]
1.	39.264
2.	87.423
3.	133.06
4.	218.32
5.	277.07
6.	321.82

Table 3.4 Frequency modulation with respect to modes and deformation

Object Name	Total Deformation
State	Solved
Scope	
Scoping Method	Geometry Selection
Geometry	All Bodies
Definition	
Type	Total Deformation
Mode	1.
Identifier	
Suppressed	No
Results	
Minimum	0. mm
Maximum	31.021 mm
Average	9.6413 mm
Minimum Occurs On	Clamper | Part Body
Maximum Occurs On	Silencer |Part Body
Information	
Frequency	39.264 Hz

Table 3.5 Deformational result for different modes

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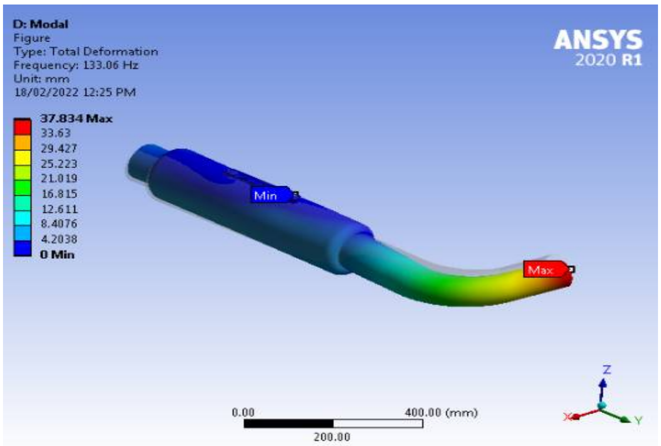
Figure 3.5 Total deformation at 39.264Hz

Figure 3.6 Total deformation at 87.423 Hz



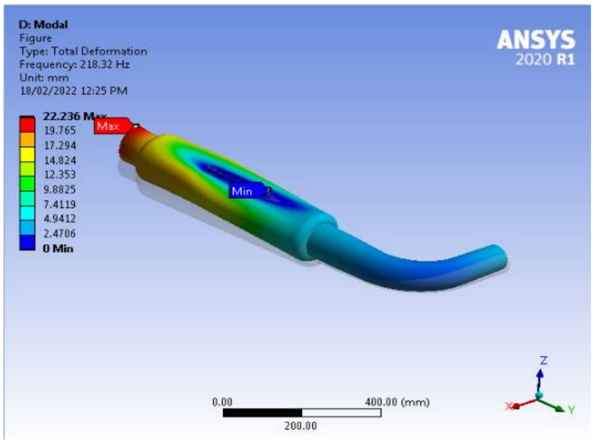
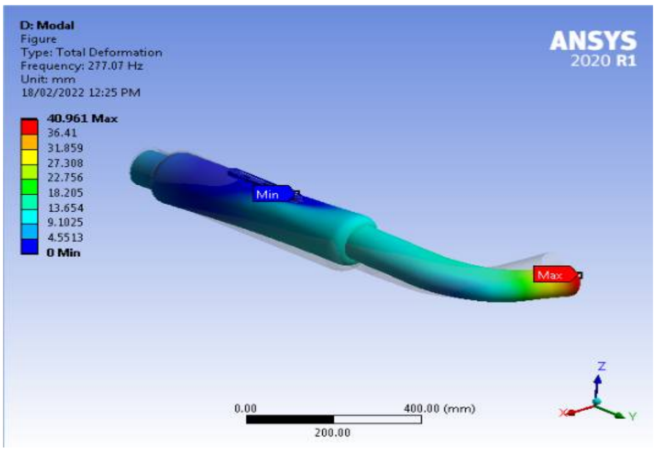


Figure 3.7 Total deformation at 133.08 Hz

Figure 3.8 Total deformation at 218.32 Hz



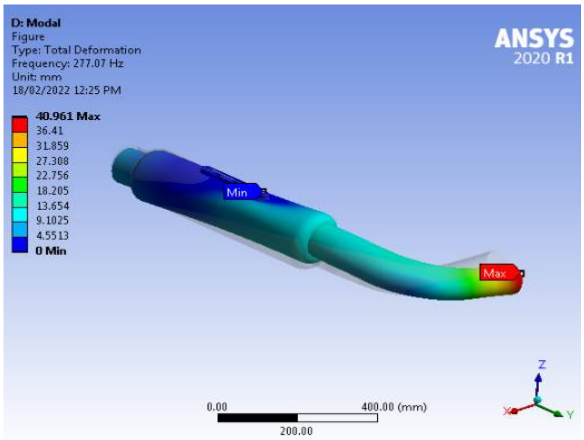


Figure 3.9 Total deformation at 277.07 Hz

Figure 3.10 Total deformation at 321.82 Hz

Object Name	Total Deformation 2	Total Deformation 3	Total Deformation 4	Total Deformation 5	Total Deformation 6	Total Deformation 7

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State	Solved					
Scope						
Scoping Method	Geometry Selection					
Geometry	All Bodies					
Definition						
Type	Total Deformation					
Mode	1.	2.	3.	4.	5.	6.
Identifier						
Suppressed	No					
Results						
Minimum	0. mm					
Maximum	31.021 mm	23.267 mm	37.834 mm	22.236 mm	40.961 mm	38.036 mm
Average	9.6413 mm	9.4923 mm	8.1692 mm	9.1515 mm	9.0555 mm	9.6036 mm
Minimum Occurs On	clamper| Part Body					
Maximum Occurs On	Silencer |Part Body					
Information						
Frequency	39.264 Hz	87.423 Hz	133.06 Hz	218.32 Hz	277.07 Hz	321.82 Hz

Table 3.6 Overall result of the model analysis with Structural steel

Strength Coefficient MPa	Strength Exponent	Ductility Coefficient	Ductility Exponent	Cyclic Strength Coefficient MPa	Cyclic Strain Hardening Exponent
920	-0.106	0.213	-0.47	1000	0.2

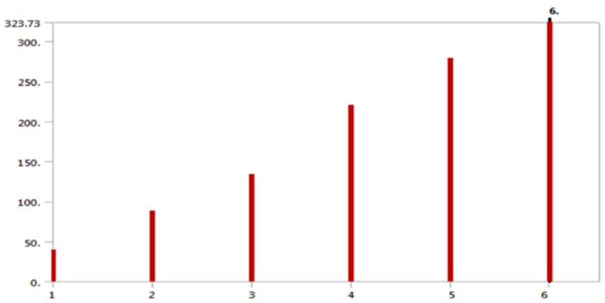
Table 3.7 Material properties

5) Discussion of Material ASTM A36 Mild Steel

Simulation of Model analysis for splendor muffler has been conducted using ANSYS workbench. The above result has been obtained & the result obtained are minimum with respect to the modulation of deformation and modes of frequency. In this iteration structural steel material has been taken into consideration. In next iteration material will be changed and compared, among the both of them which has minimum frequencies.

C. Analysis of Aluminum and Titanium

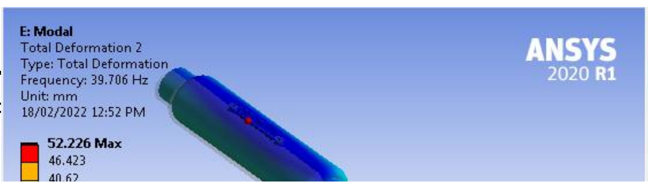
Same boundary condition as discussed above has been considered and results are as below

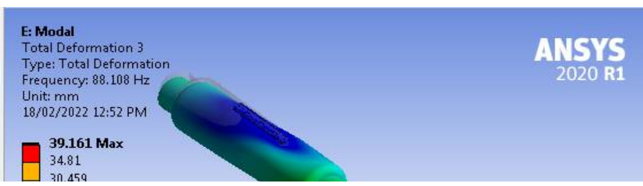


Graph 3.2 Frequency at different modes

Mode	Frequency [Hz]
1.	39.706
2.	88.108
3.	133.92
4.	220.
5.	278.9
6.	323.73

Table 3.8 frequency at different modes





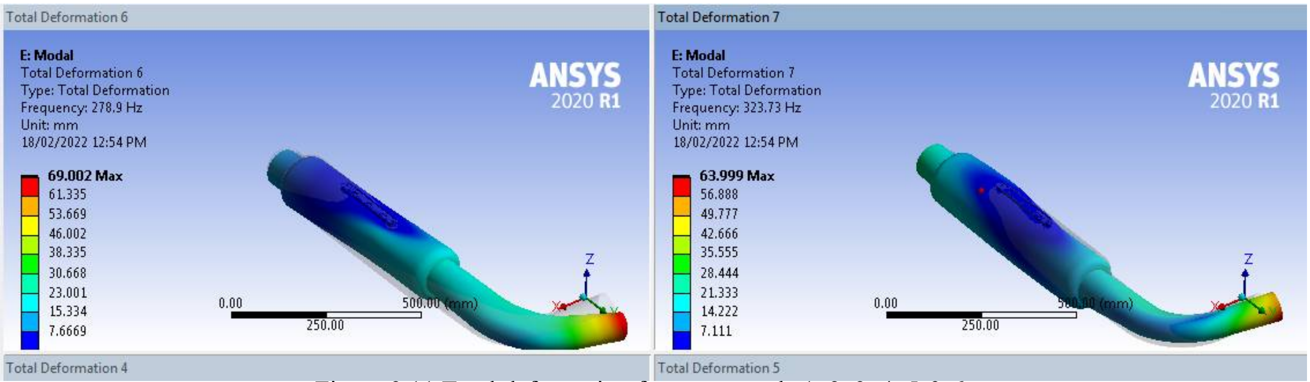
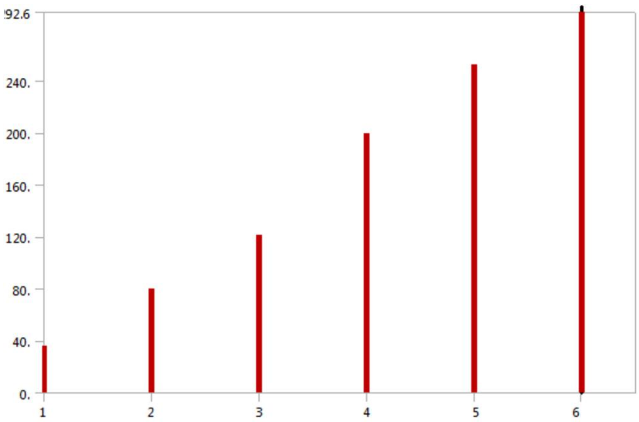


Figure 3.11 Total deformation factors at mode 1, 2, 3, 4, 5 & 6

D. *Analysis of Titanium*



Mode	Frequency [Hz]
1.	36.114
2.	79.82
3.	121.12
4.	199.24
5.	252.27
6.	292.6

Graph 3.3 Modes of frequency

Table 3.9 Modes of frequency for titanium material

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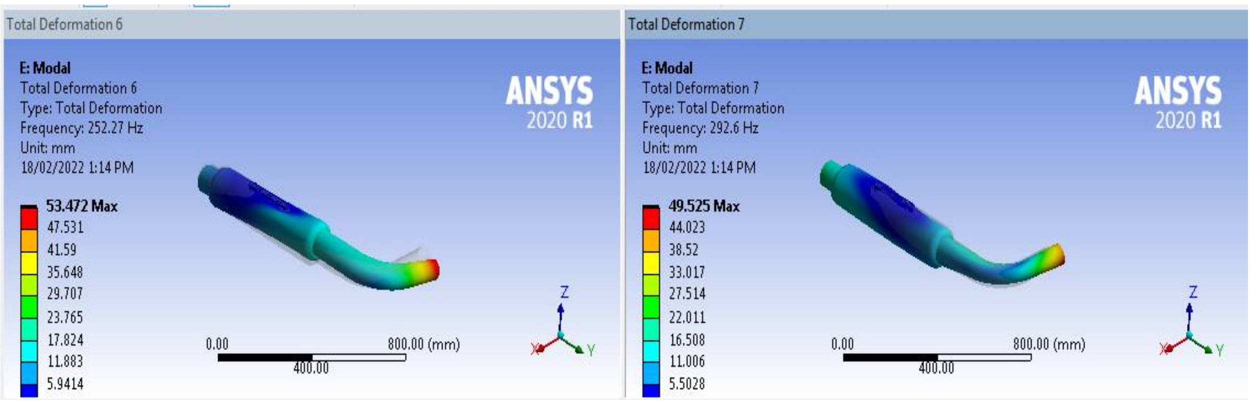

Figure 4.15 Total deformation factors at mode 1, 2, 3, 4, 5 & 6

E. Discussion of Material 2 & 3 with 1

Aluminum

Bounding Box
Length X 376.21 mm
Length Y 1058.1 mm
Length Z 164.2 mm

Properties
Volume 9.8325e+005 mm³
Mass 2.7236 kg

Titanium

Bounding Box
Length X 376.21 mm
Length Y 1058.1 mm
Length Z 164.2 mm

Properties
Volume 9.8325e+005 mm³
Mass 4.5426 kg

Mode	Frequency [Hz]

Mode	Frequency [Hz]
1.	39.706
2.	88.108
3.	133.92
4.	220.
5.	278.9
6.	323.73

Table 3.11 Structural steel material modes

1.	36.114
2.	79.82
3.	121.12
4.	199.24
5.	252.27
6.	292.6

Table 3.12 Titanium material modes

In this iteration aluminum & material has been considered and the results are forecasted. Among the structural steel, aluminum & titanium, aluminum is the best suited material because the vibrational frequency modes are near to the steel & titanium material, and also the weight consideration for aluminum, titanium and steel has much different variations but cost varies. Hence for the suited weight and modes of frequency aluminum is the best choice.

IV. CONCLUSION

In this project the 1st iteration of the analysis has been done to check the vibrational frequency of the muffler using structural steel material and the maximum frequency obtained at mode 6 is 321.82 Hz. In 2nd iteration same boundary condition is used and simulation is solved, here the material has been changed from SS to aluminum & titanium, the maximum frequency developed was 323 Hz which is 1-2 Hz more compared to steel but due to its less density the aluminum has less weight. And for titanium the maximum frequency developed is 292.6 HZ which is lesser than all steel as well as aluminum. But the cost for titanium is more compared to another material. The optimization of the muffler is carried in order to bring the natural frequency of the aluminum lesser then the steel.

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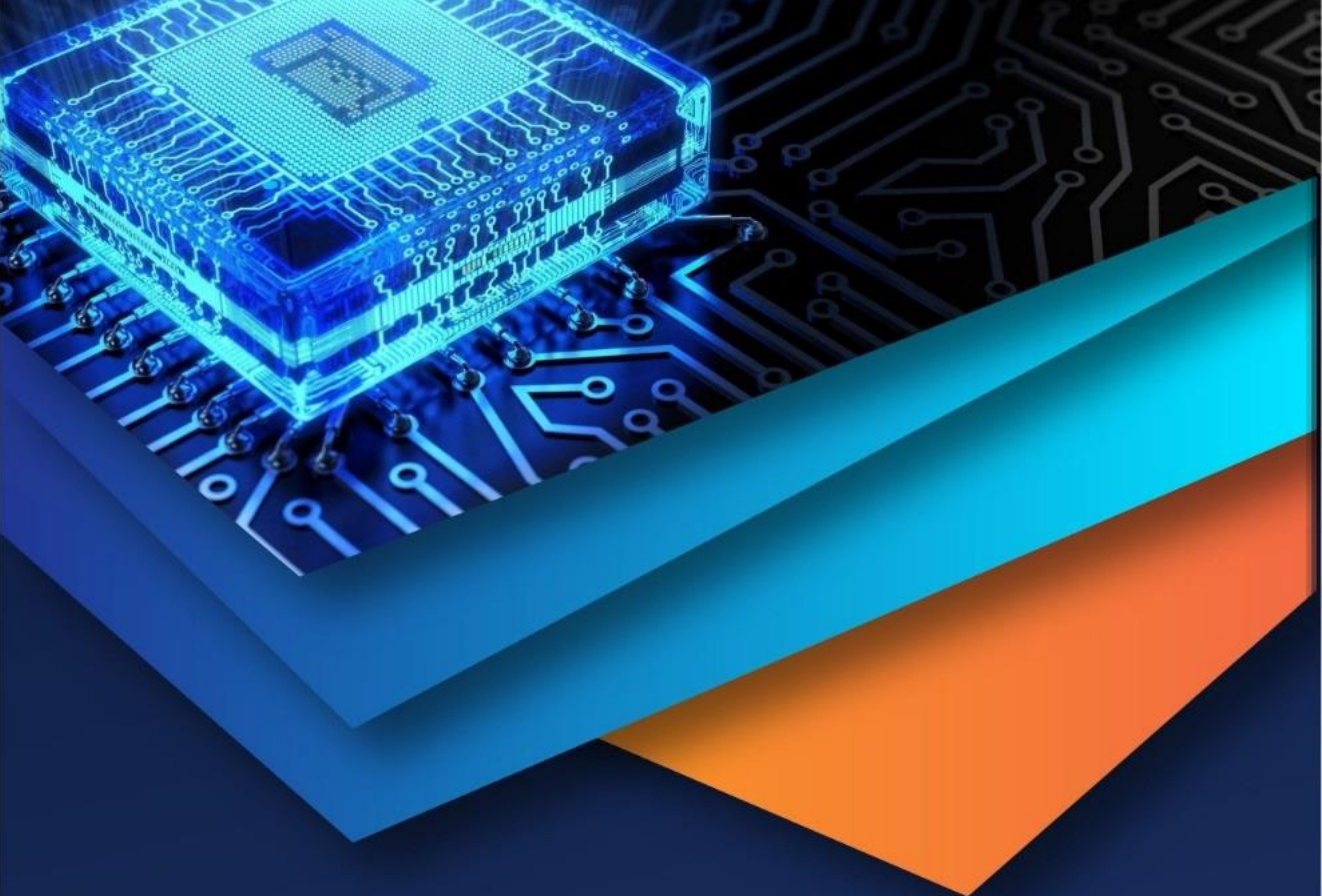

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