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Structural Analysis and Optimization of Exhaust Muffler

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Abstract: Silencer is an integral part of the exhaust system. The silencer serves the function of noise and vibration reduction. The exhaust gases in the combustion chamber which are at temperatures of around 1200K are released to the atmosphere at around 323K. Temperature reduction takes place efficiently as the flue gases flow through the exhaust system. Apart from temperature loads, the exhaust system also carries pressure loads, acceleration load and load due to its self-weight. Modal analysis is carried out on the optimum geometry to determine its natural frequency to avoid the resonance. Fatigue analysis is further carried out on the geometry to determine the Life, Damage, Safety Factor, Equivalent alternating stress and Bi-axiality indication under extreme dynamic repetitive loading conditions.
Keywords: Silencer,ANSYS,Dynamic, Structural, Modal, Fatigue

I. INTRODUCTION

Silencer is an integral part of the exhaust system. The silencer serves the function of noise and vibration reduction. The exhaust gases in the combustion chamber which are at temperatures of around 1200K are released to the atmosphere at around 323K. Temperature reduction takes place efficiently as the flue gases flow through the exhaust system. Apart from temperature loads, the exhaust system also carries pressure loads, acceleration load and load due to its self-weight. Temperature reduction takes place efficiently as the flue gases flow progressively through the exhaust system. Sound travels in the form of pressure waves which are transverse in nature. So, to reduce the sound, these pressure waves can either be cancelled out or it can be absorbed. There are generally two main reasons for generation of noise in an engine.

A. Noise is generated as the result of the internal combustion
B. Rapid opening and closing of the inlet and exhaust valves generates pressure waves which also generates a lot of noise.

If a vehicle does not have a muffler or silencer than it creates noise due to the difference of frequencies of sound. This noise is undesirable. To reduce this noise arising out of the exhaust from internal combustion engine, mufflers are mandatory device to adopt with the stringent environmental regulations. Muffler contains more pressure hence sound is in the pressure waves.

The primary function of a silencer is to reduce the noise and vibrations and temperature of the exhaust gases flowing through it. Since sound is in the form of pressure waves, controlling the pressure will reduce the noise. So, there was need for a device which could disturb the flow of the exhaust gases flowing through it and also absorb the unwanted noise. As a result, the internal structure of a silencer consists of various baffles and inner pipes having perforations and certain sound absorbing materials so that effective damping and temperature reduction takes place. Based on the function various architectures of silencers are available which are as follows:

1) Absorption type
2) Reflection type
3) Absorption-reflection type
4) Wave cancellation type

II. LITERATURE REVIEW

R. Raguram [1] carried out an effective study of the analysis and failure of silencer and silencer mounting brackets. The study is carried out on Ashok Leyland passenger busses. The study depicts the common causes of silencer failures.P. Srinivas et.al [2] performed Design and Analysis of an Automobile Exhaust Muffler. Dynamic modal analyses were carried out to determine the mode shapes, stresses and deformations of exhaust muffler using CAE analysis.The muffler geometry was checked for its Pressure drop, Temperature drop and Velocity distribution and based on the study it was concluded that Double expansion chamber gives better results as compared to single expansion chamber.S. Rajadurai et.al [3] provided an extensive study for the materials used for the automotive exhaust systems. Functions of various exhaust components were studied and the extreme environmental conditions under which these components operate were identified.Prashant. A.Naik [4] carried out the Dynamic Analysis of Muffler using Finite Element Method and Experimental Method. The natural frequency and mode shapes of system were studied by using numerical and experimental method. SenthilnathanSubbiah et.al [5] studied the effect of muffler mounting brackets on durability. The failure analysis of muffler mounting brackets of 3-Wheelers were observed. Cracks at the weld locations in all vehicles were observed at an average distance of 10000 km. A FEM model was developed for the engine cradle assembly where the engine and the muffler were modeled as point masses. The entire assembly was tested for 4G acceleration. Various design modifications were carried out and its effect on durability was discussed and the new design passed the durability test of 100000 km.Dhirajkumar K. More et.al [6] carried out the Thermal Analysis of Two Wheeler Exhaust Silencer. The main objective was to assess the heat flow during the passage of hot gases and design the passage in such a way that it will minimize the destructive effects of hot-spots over the length of the silencer, especially at the front end mating with the exhaust manifold.Dr.Maruthi.B.H et.al [7] evaluated Structural Integrity of Passenger Car Exhaust System. The exhaust system under consideration was a Hyundai i10 model. Static analysis was performed by using MSc Nastran for exhaust components to determine the high stress region, and also the maximum displacement and reaction forces are observed at the bracket and hanger locations. Modal analysis was carried out to determine the structural behavior of the exhaust system. Somashekar G et.al [8] carried out Modal Analysis of Muffler of an Automobile by Experimental and Numerical Approach. Both experimental and numerical modal analysis was carried on a Tata Indica car. The experiment was conducted on thickness of existing muffler body by FFT analyzer and numerous iterations were carried out by changing the thickness of muffler body, perforation of baffle plates.Venugopala Reddy Kussam et.al [9] performed Structural Analysis of Passenger Car Exhaust System by Using Hypermesh. Modal analysis of the Hyundai i10 exhaust system was carried out to obtain the natural frequency. Stress analysis on the whole exhaust system model under 1G and road load condition was performed. Dynamic analysis was performed to check the displacement at different location of the exhaust system.Mr. N. Vasconcellos et.al [10] performed Structural Analysis of an Exhaust System for Heavy Trucks. A finite element model was generated including the complete vehicle and the exhaust system. The results obtained assured the structural integrity of the exhaust system and contribute to a better understanding of this system behavior and its structural strength.Mr.AmitMahadeoAsabe et.al [11] Postulated the design and modification of silencer in order to reduce the vibration. The experimental analysis was 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. Patekar [12] theoretically modeled the exhaust silencer of a two wheeler using Finite Element Method and experimentally validated using Fast Fourier Transform analyzer. It was noticed that the dynamic performance could be increased with increasing thickness of various parts.

III. PROBLEM STATEMENT

The main objective of this work is to design an exhaust muffler for a 4 Cylinder petrol engine. Natural frequency is checked for the silencer geometry to avoid resonance. Fatigue life is calculated for the silencer geometry subjected to fully reverse cyclic dynamic loading conditions. The silencer is to be designed for high cycle fatigue. Necessary specifications required for optimum silencer design are mentioned below.

Engine capacity: 1400cc.
Engine speed: 10000 Rpm.
Overall length of the muffler: 840mm
Thickness of the sheet metal used: -up to 1.5 mm
Fatigue life of the muffler: greater than 10^5 cycles
Nature of loading: Constant amplitude fully reversed load.
Method used: Stress life approach.

IV. 3D CAD MODEL

Initially design calculations are performed to find out the necessary dimensions of the silencer. Engine capacity and engine speed are taken as a reference for calculations. Based on the dimensions derived, 3D silencer geometry is created.

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Geometry 1 (Base geometry):

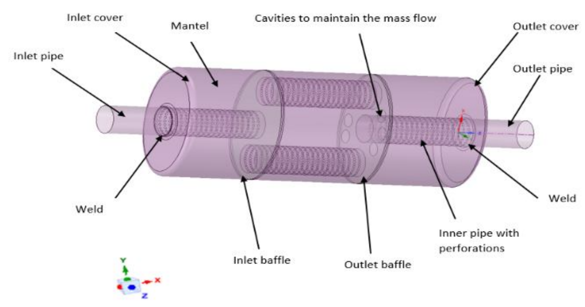


Fig.1 Geometry 1(Base geometry)

Geometry 2(Optimized geometry):

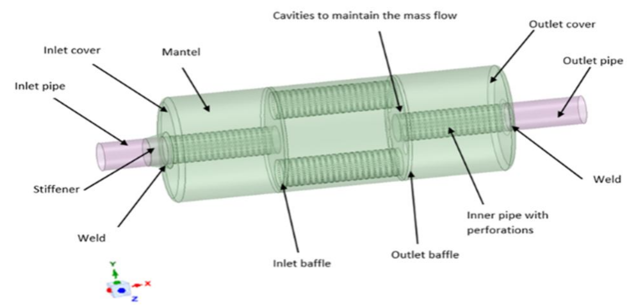


Fig.2 Geometry 2 (Optimized) stiffeners introduced

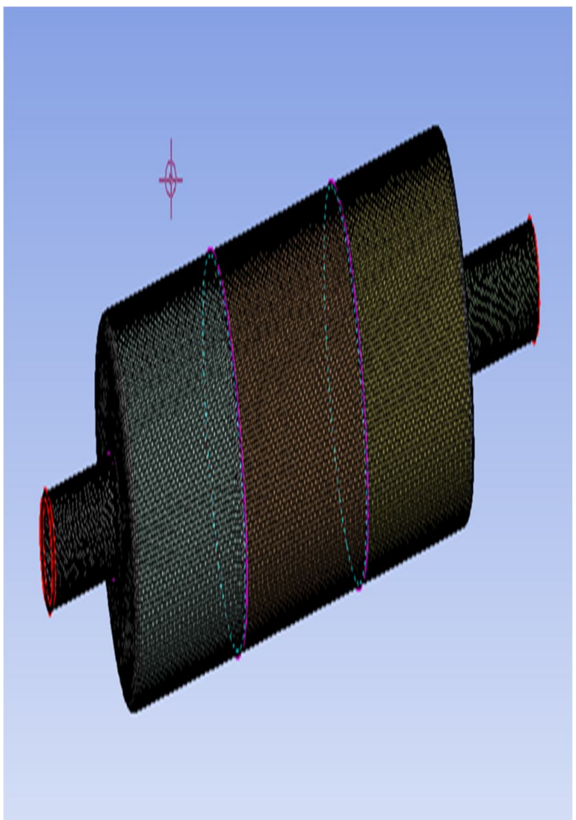
V. MESHING

A. Structural geometry

The entire geometry is made from sheet metal and the model is symmetric under plane stress condition, so the mid-surface is extracted to reduce the element count. The geometry is meshed using meshing module of ANSYS workbench 18.0.

B. Meshed geometry

Mid surface is extracted to reduce the element count.



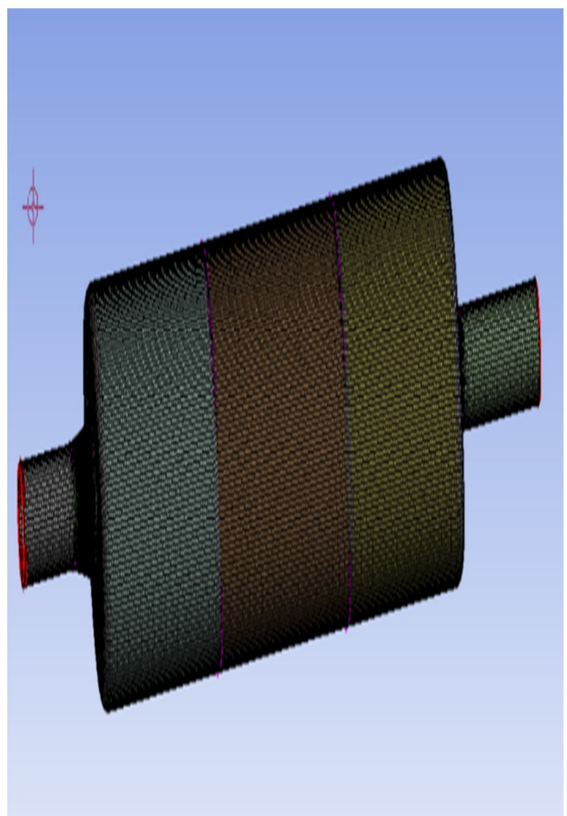


Fig.4 Meshed Geometry 2(optimized geometry).

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VI. MATERIAL

Steel of SS250 grade is used for the silencer geometry.

A. Steel SS250

Table no.1 Material properties of SS250

SS250 steel	
Density	7850 Kg/m^3
Coefficient of thermal expansion	1.2 x10^{-5}/C
Young's modulus	2 x 10^{5}Mpa
Poisons ratio	0.3
Bulk modulus	1.67x 10^{5}Mpa
Shear modulus	7.692 x10^{4}Mpa
Elongation coefficient	0.23

B. SN Curve for SS250

The SN curve for SS25 Grade steel for Fatigue calculations is as follows.

Table no.2 SN Curve SS250

Cycles	Alternating stress(Mpa)
10	3999
20	2827
50	1896
100	1413
200	1069
2000	441
10000	262
20000	214
100000	138
200000	114
1000000	86.2

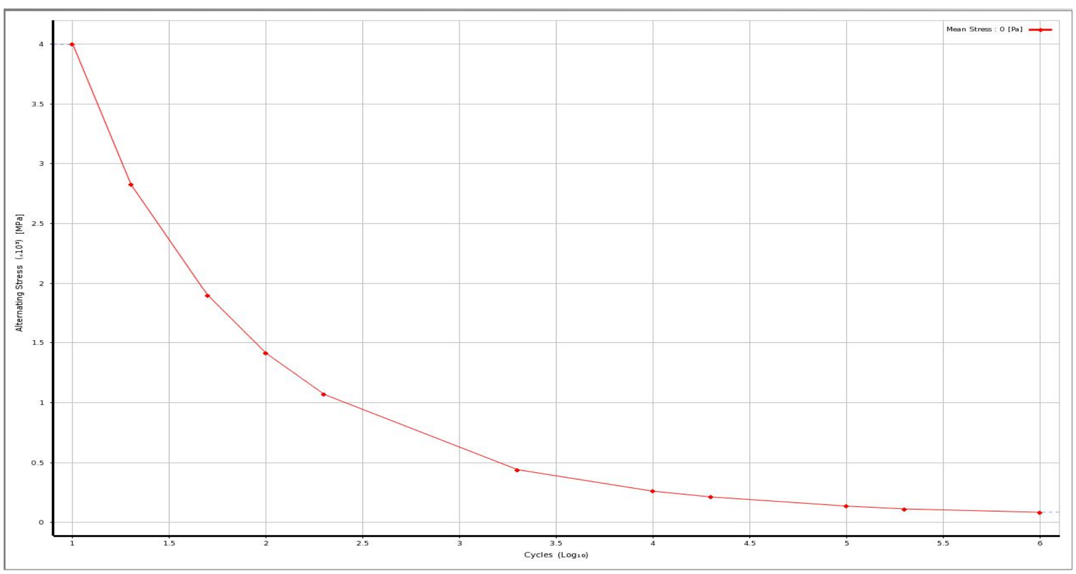

Fig .5 SN Curve SS250

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VII. BOUNDARY CONDITIONS

A. *For Structural Analysis*

Standard earth gravity (Self-weight) of the silencer geometry: 9810N /mm^2 (downward direction)
Two elastic supports are applied at the outer mantel just above the internal baffle locations
Foundation stiffness: 10N/mm.
A point mass of 8kg is applied remotely at a distance of 500 mm from the outer edge of the inlet pipe to compensate for the weight of the catalytic convertor and the connection pipes.

B. *Dynamic Loading*

Acceleration load:
X direction: -1.5 G (acceleration generated due to braking)
Y direction: +2G (acceleration generated due to bumping of the vehicle over a speed breaker)
Z direction: +0.5G (centrifugal acceleration generated due to sharp turning of the vehicle)

C. *Fatigue Analysis*

Type of loading: Constant amplitude fully reversed load.
Method: stress life approach

D. *Modal Analysis*

Two elastic supports are applied at the outer mantel just above the internal baffle locations
Foundation stiffness: 10N/mm.
Remote displacement at 500 mm from the outer edge of the inlet pipe
X displacement: 0mm
Y displacement: 0mm
Z displacement: 0mm
X rotation: 0 degree
Y rotation: 0 degree
Z rotation: 0 degree
Figure below shows the detailed boundary conditions.

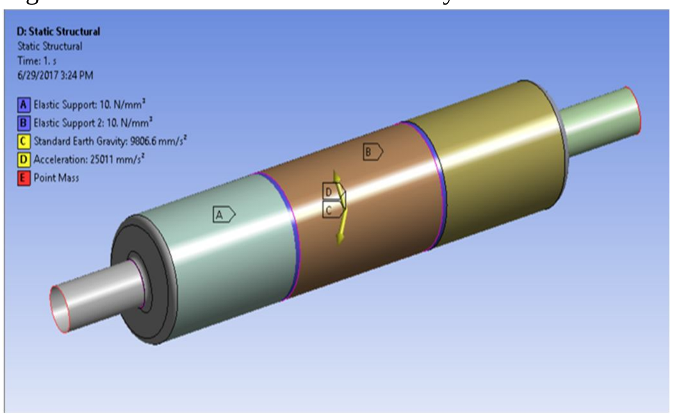

Fig.5 Boundary conditions for structural analysis

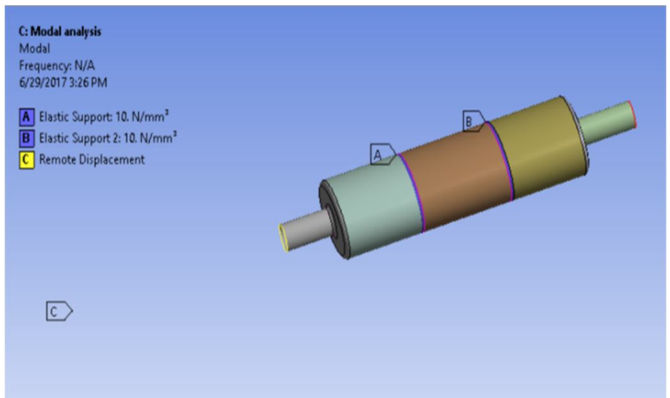

Fig.6 Boundary conditions for structural analysis

VIII. ANALYSIS AND RESULTS

A. *Structural Analysis*

Initially modal analysis is carried out on the given geometry to find its natural frequency to avoid the resonance.

B. *Modal Results*

Modal results are extracted for the first 6 modes are as tabulated below:

Table .3 Frequency at various modes

Mode	1st Mode	2nd Mode	3rd Mode	4th Mode	5th Mode	6th Mode
Frequency (Hz)	187.11	305.76	361.71	402.13	565.54	705.73

C. Structural Results Under Dynamic Loading

The structural results under dynamic loading are plotted below.

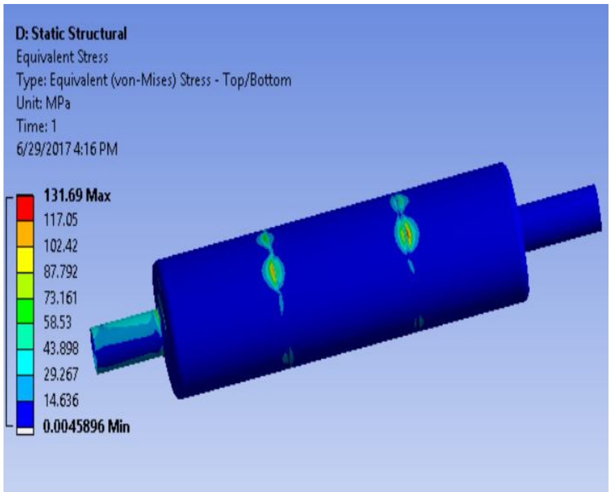

Fig no.7 Equivalent (Von-misses) stress

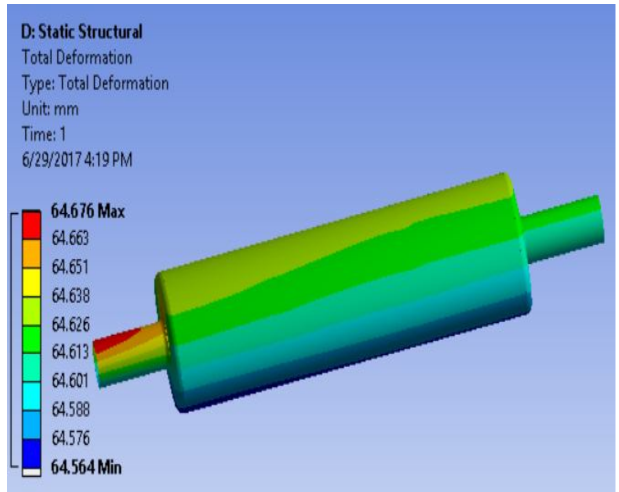

Fig no.8 Total deformation

The results of the structural analysis under dynamic loading conditions are as follows

Table.4 results of the structural analysis under dynamic loading condition

Equivalent (Von-misses) stress: Maximum: 131.69 Mpa Location: Outer Mantel above the internal baffles Minimum: $4.5x10^{-3}$Mpa Location: Outlet pipe	Total deformation: Maximum: 64.676 mm Location: Inlet pipe Minimum: 64.564 mm Location: Outer mantel

D. Fatigue Analysis Under Dynamic Loading

The fatigue results are as follows

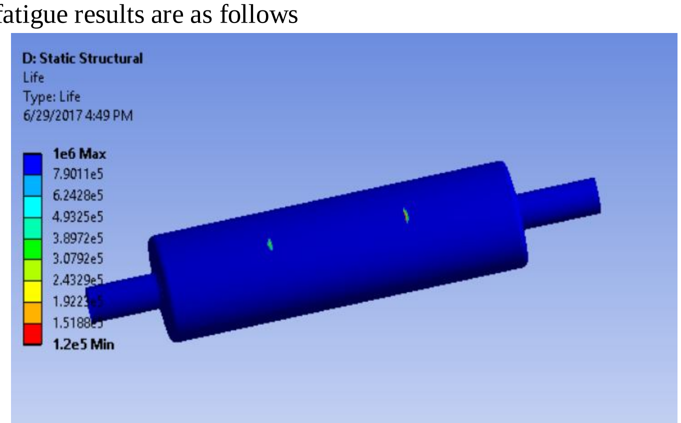

Fig no.9 Fatigue life of the silener

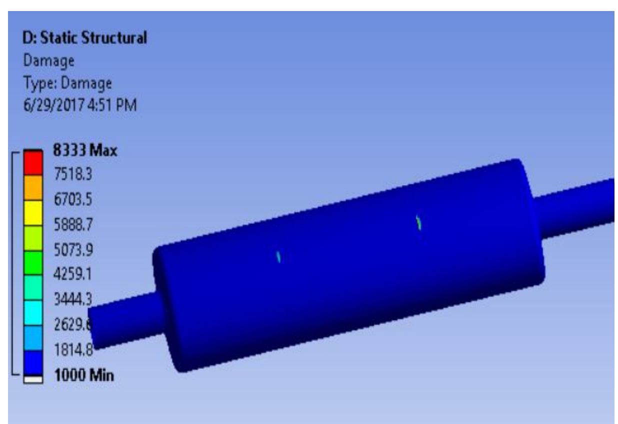

Fig no.10 Damage

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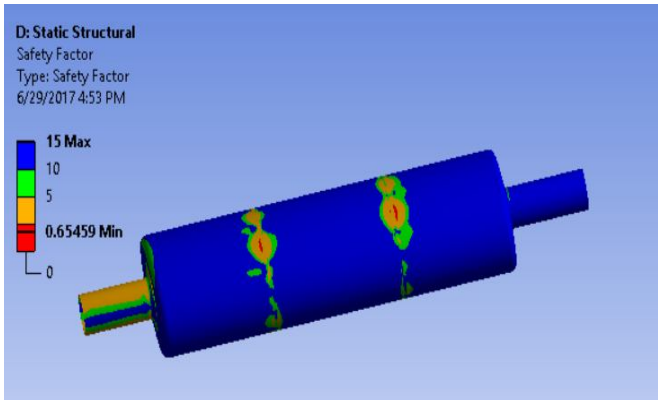

Fig no.11 Safety factor

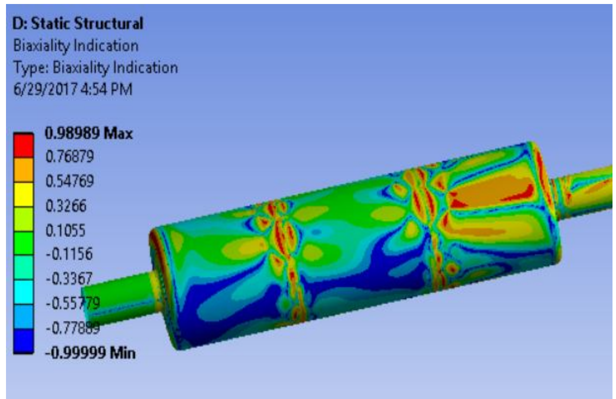

Fig no.12 Biaxiality indication

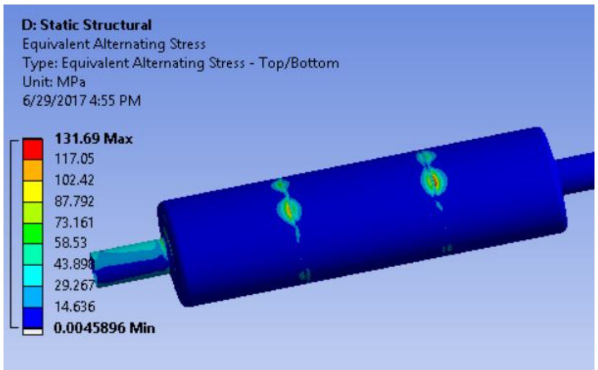

Fig no.13 Equivalent alternating stress

The results of the fatigue analysis for the silencer are given below.

Table.5 results of the fatigue analysis for the silencer

Fatigue life: Minimum:1.2x10 ^{5}Cycles Location: Outer mantel	Damage: Maximum:8333 cycles Location: Outer mantel	Safety factor: Minimum: 0.654 Location: Outer mantel	Biaxiality indication: Maximum:0.989 Location: Inlet baffle Minimum:-0.999 Location: Outlet inner pipe	Equivalent alternatingstress: Maximum:131.69 Mpa Location: Outer mantel Minimum:4.5x10^{-3}Mpa

E. Structural Results for the Modified Geometry

1) *Modal Results (Modified Geometry):* Modal results are extracted for the first 6 modes are as tabulated below:

Table.5 Frequency at various modes

Mode	1st Mode	2nd Mode	3rd Mode	4th Mode	5th Mode	6th Mode
Frequency (Hz)	172.34	280.53	341.74	367.38	555.3	671.69

F. Structural Results Under Dynamic Loading (Modified Geometry):

The structural results of modified geometry under dynamic loading are plotted below.

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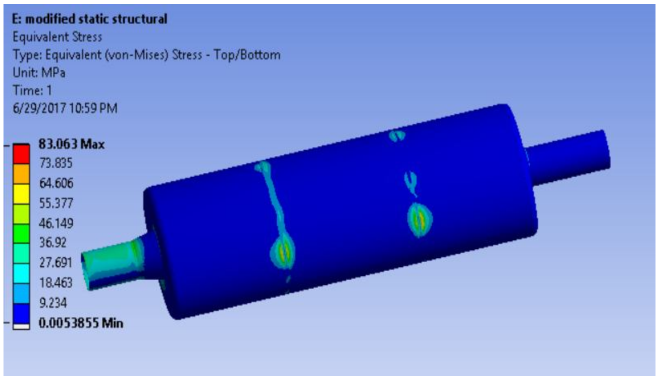

Fig no.14 Equivalent (Von-misses) stress

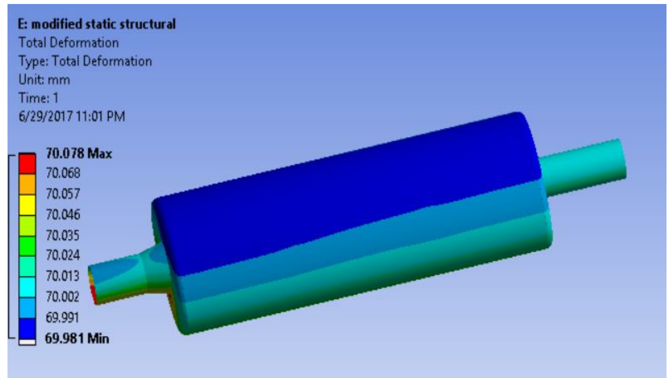

Fig no.15 Total deformation

The results of the structural analysis of the modified geometry under dynamic loading conditions are as follows:

Table no.6 results of the structural analysis of modified geometry under dynamic loading condition

Equivalent (Von-misses) stress:	Total deformation:
Maximum: 83.063 Mpa	Maximum: 70.078 mm
Location: Outer Mantel above the internal baffles	Location: Inlet pipe
Minimum: 5.3×10^{-3}Mpa	Minimum: 69.981 mm
Location: Outlet pipe	Location: Outer mantel

G. Fatigue Analysis Under Dynamic Loading (Modified Geometry)

The fatigue results of the modified geometry are as follows:

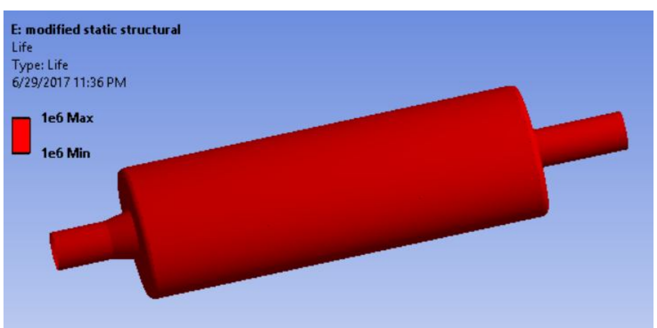

Fig no.16 Fatigue life of the silencer

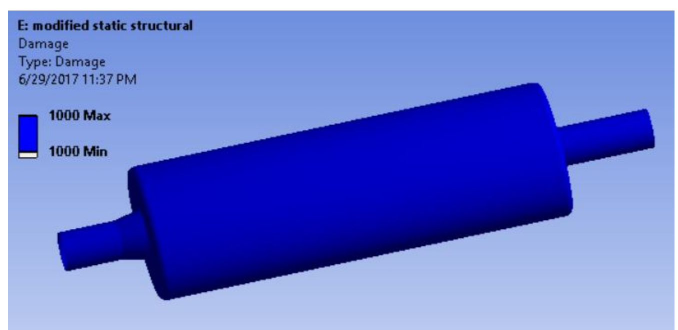

Fig no.17 Damage

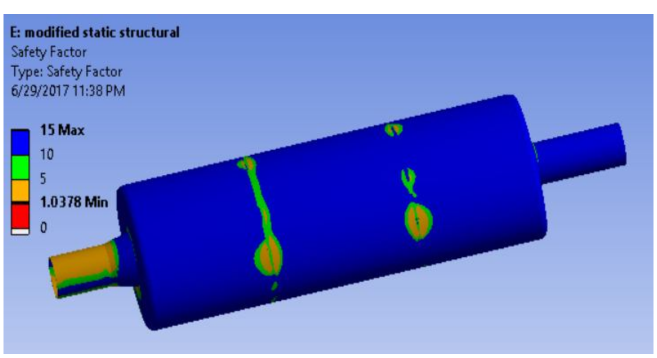

Fig no.18 Safety factor

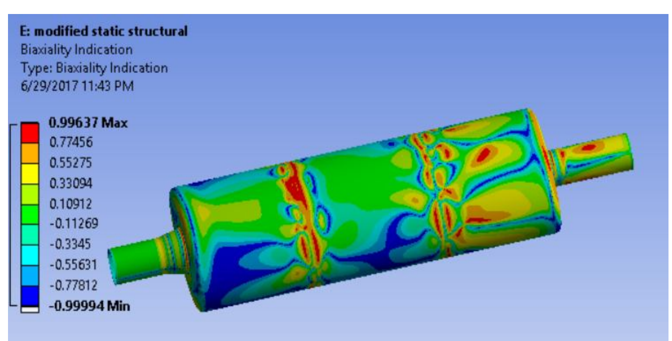

Fig no.19 Biaxiality indication

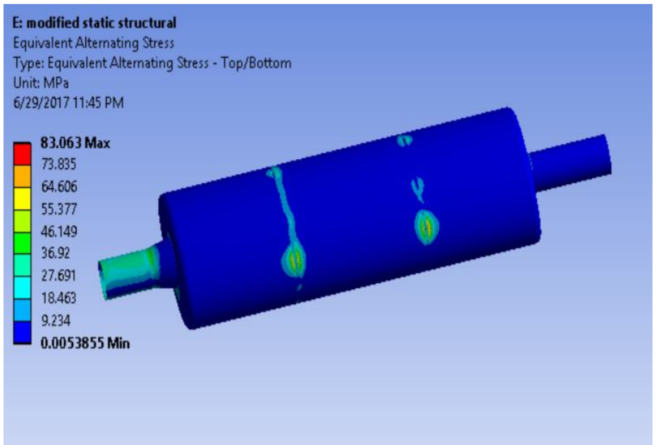

Fig no.20 Equivalent alternating stress

The results of the fatigue analysis for the silencer are given below.

Table no.7 results of the fatigue analysis for the silencer

Fatigue life: Minimum: 10^6 Cycles Location: Inlet pipe	Damage: Maximum:1000 cycles Location: Inlet pipe	Safety factor: Minimum: 1.0378 Location: Outer mantel	Biaxiality indication: Maximum:0.996 Location: Outer mantel Minimum:-0.999 Location: inner pipe	Equivalent alternatingstress: Maximum:83.063 Mpa Location: Outer mantel Minimum:5.38x10^{-3}Mpa Location: Outlet pipe

H. Summarized Results

1) Structural Analysis:

a) Modal Results:

Table .8 Summarized modal results

Original geometry						
Mode	1st Mode	2nd Mode	3rd Mode	4th Mode	5th Mode	6th Mode
Frequency (Hz)	187.11	305.76	361.71	402.13	565.54	705.73
Modified geometry						
Mode	1st Mode	2nd Mode	3rd Mode	4th Mode	5th Mode	6th Mode
Frequency (Hz)	172.34	280.53	341.74	367.38	555.3	671.69

I. Structural Results Under Dynamic Loading

Table.9 Summarized structural results

Original geometry	
Equivalent (Von-misses) stress: Maximum: 131.69 Mpa Location: Outer Mantel above the internal baffles Minimum: 4.5x10^{-3}Mpa Location: Outlet pipe	Total deformation: Maximum: 64.676 mm Location: Inlet pipe Minimum: 64.564 mm Location: Outer mantel
Modified geometry	

Equivalent (Von-misses) stress: Maximum: 83.063 Mpa Location: Outer Mantel above the internal baffles Minimum: $5.3\text{x}10^{-3}$Mpa Location: Outlet pipe	Total deformation: Maximum: 70.078 mm Location: Inlet pipe Minimum: 69.981 mm Location: Outer mantel

J. Fatigue Results Under Dynamic Loading

Table.10 Summarized fatigue results

Original geometry				
Fatigue life: Minimum:1.2x10 ^{5}Cycles Location: Outer mantel	Damage: Maximum:8333 cycles Location: Outer mantel	Safety factor: Minimum: 0.654 Location: Outer mantel	Biaxiality indication: Maximum:0.989 Location: Inlet baffle Minimum:-0.999 Location: Outlet inner pipe	Equivalent alternatingstress: Maximum:131.69 Mpa Location: Outer mantel Minimum:$4.5\text{x}10^{-3}$Mpa
Modified geometry				
Fatigue life: Minimum: 10^{6} Cycles Location: Inlet pipe	Damage: Maximum:1000 cycles Location: Inlet pipe	Safety factor: Minimum: 1.0378 Location: Outer mantel	Biaxiality indication: Maximum:0.996 Location: Outer mantel Minimum: -0.999 Location: inner pipe	Equivalent alternatingstress: Maximum:83.063 Mpa Location: Outer mantel Minimum:$5.38\text{x}10^{-3}$Mpa Location: Outlet pipe

IX. CONCLUSION

Modal analysis followed by structural analysis is carried out further for geometry 4 and the geometry is checked for its fatigue life under extreme environmental conditions. The natural frequency of the silencer is 187.11 Hz which changes to 172.34 Hz after modification. The Equivalent stress of the silencer are 131.69 Mpa which decreases to 83.063 Mpa after modification. Fatigue life of the silencer is 1.2x105 cycles which increases to 106 cycles (infinite life) after modification. The damage in the silencer decreases from 8333 to 1000 cycles after modification. The safety factor in the silencer increases from 0.654 to 1.0378 after modification. The alternating stresses in the silencer decreases from 131.69 Mpa to 83.063 Mpa after modification.

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