Vibration Analysis and Design Modification of Automobile Silencer

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Abstract - *In automobiles, exhaust system has great importance and silencer is used to minimize unwanted noise. The failure of silencer is due to thermal stresses and vibration of silencer. 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. In present work, the modal analysis of existing Silencer is carried out by finite element analysis (FEA) method using ANSYS 16.0 software and these results are compared with reading taken on experimental modal analysis by using FFT analyzer to validate the analytical results. Then, modification in silencer geometry has done to optimize natural frequency and compare the results of natural frequency of existing silencer and modified silencer.*

Key Words*: *Modal Analysis*, *Finite Element Analysis*, *FFT Analyzer*, *Silencer.

1.INTRODUCTION

The silencer is main part of exhaust system of internal combustion engine. The exhaust gases leave from the engine at very high speeds. During each cycle of combustion, the exhaust valves opens and closes simultaneously so that exhaust gas pressure changes continuously from high to low causing a vibration and noise. The basic function of silencer is to reduce noise and vibrations produced by exhaust gases. The silencer vibration contributes the total vibrations of two-wheeler. Due to vibrations it is subjected to several stresses therefore to study the behavior of silencer by analyzing the vibration modes and its respective natural frequency, help to minimize crack, improving life and efficiency of silencer. The effect of vibration on silencer and effect of different geometry on various parameters are studied by many scholars. Ahmed Elsayed et al. [1] presented a parametric study on the effect of baffle configuration on transmission loss. Results show that decreasing the baffle cut ratio tends to increase the transmission loss at intermediate frequencies by up to 45%. Decreasing the spacing between silencer plates was shown to enhance the silencer transmission loss by 40%. Sidharam Ambadas Basargi et al. [2] determined the modes which affect the performance of automobile adversely and suggested increasing the mass, increase the damping, or providing a negative stiffness to make the silencer more damped. Mukesh D. Bankar et al. [3] explained the effect of different material's on natural frequency of silencer and found out that the increase in natural frequencies for silencer of material SUS409L is more than other two silencers of materials SUS436j1L and SUS436L. Therefore, the silencer of material SUS409L has more stiffness as compared to other two silencers, so it has minimum vibrations. Mohit Juneja et al. [4] emphasizes the importance of the design methodology a practical approach from the concept design to proto manufacturing and validation of exhaust silencer. This approach serves the purpose of reducing the number of iterations, product development time and cost with Better design. H. Bartlett et al. [5] presented the modeling and analysis of variable geometry, exhaust gas systems. The modeling and analysis procedures for a variable geometry, exhaust gas system have been outlined in this paper.

2. METHODOLOGY

a) Selection of particular Engine and its following Silencer.
b) Experimental testing of Silencer using FFT analyzer to obtain natural frequencies at different modes
c) CAD modelling of existing Silencer using CATIA V5.
d) Modal Analysis of existing silencer to obtain natural frequency & mode shapes using finite element analysis software ANSYS16.
e) Compare the analytical results with experimental testing results.
f) Modification in silencer geometry and CAD modelling of modified silencer using CATIA V5.
g) Modal Analysis of Modified silencer to obtain natural frequency & mode shapes.
h) Comparative analysis existing silencer results and modified silencer.

3. EXPERIMENTAL TESTING OF SILENCER

The experimental modal analysis was conducted using FFT analyzer. The aim of this experimental work is to investigate the modal parameters that are natural Frequency and mode shape. In order to find out natural frequency and mode shape free-free analysis has done. For this, an experimental test rig was designed in order to measure the natural frequency of silencer as shown in fig. 2.1. The results obtained in experimental testing explained in results and discussion. The procedure of experimental testing of silencer as follows.

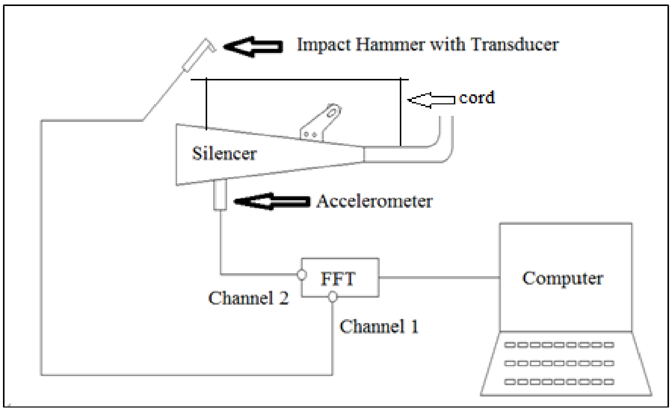

Fig.2.1 Experimental Test Set up for Silencer

3.1 Experimental Procedure

1. Silencer is hanged horizontally with the help of flexible cord. Arrangement should be such that it behave like free-free system.
2. Connection of FFT analyzer with accelerometer, impact hammer, computer is carried out. All the connections were made properly.
3. The silencer is divided into six section having equal distance between each section.
4. The 1-D accelerometer is attached at the extreme end of silencer.
5. To find natural frequency of silencer experimentally, excitation is given to silencer at suitable distance. The silencer is assumed to be free-free system.
6. The structure is impacted at each marked point that is from position 1 to 6 and simultaneously the corresponding natural frequency of silencer are recorded.
7. The experiments were repeated to check the repeatability of the experimentation.
8. All the data obtained from the vibrating silencer was recorded with the help of accelerometer attached to it.

4. MODAL ANALYSIS USING FEA

In order to create a CAD model in CATIA software measurement of dimensions of silencer carried out by using coordinate measuring machine (CMM). Silencer outer body dimensions which include main pipe and tail pipe are measured by coordinate measuring machine. Silencer internal body dimensions which include short pipe, long pipe, baffle plates and perforated tubes are measured by using Vernier caliper and CAD model created in CATIA V5 software. This CAD model imported in ANSYS16 software and analysis has done in MODAL module. The material of silencer is mild steel and its properties are length of silencer: 1120.9 mm, density of material: 7850 kg/m3, Modulus of elasticity: 2e^5 MPa, Poisson's ratio: 0.3. For free-free analysis there no fixed constrains applied. The results obtained by modal analysis are explained in results and discussion.

5. MODAL ANALYSIS OF MODIFIED SILENCER

The modification in silencer geometry is done as mass as main parameter. So, there are two cases of modal analysis of modified silencer which are modal analysis of increased mass of silencer and modal analysis of decreased mass of silencer. Following figure 5.1 shows modified design of silencer in which thickness of baffle plate is increased by 0.25 mm, it lead to increased mass from 6.971 kg to 7.21 kg and which is highlighted by green color.

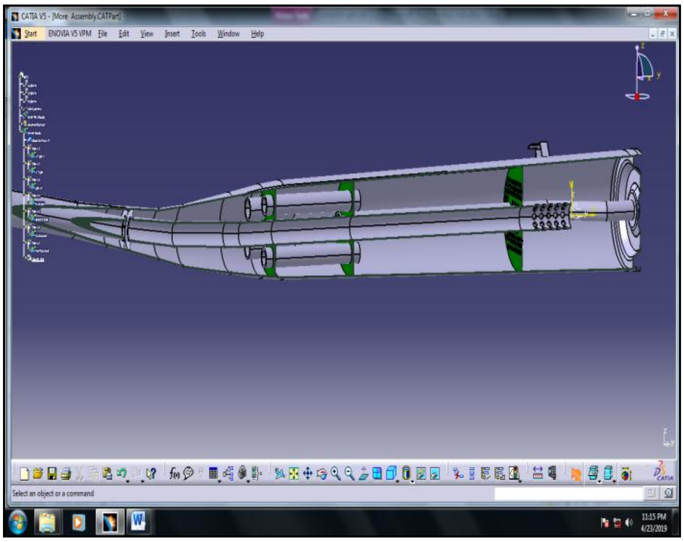

Fig.5.1 Cut Section of Mass Added Silencer in CATIAV5R20

Following figure 5.2 shows modified design of silencer in which thickness of baffle plate is decreased by 0.25 mm, it lead to decreased mass from 6.971 kg to 6.766 kg and which is highlighted by brown color.

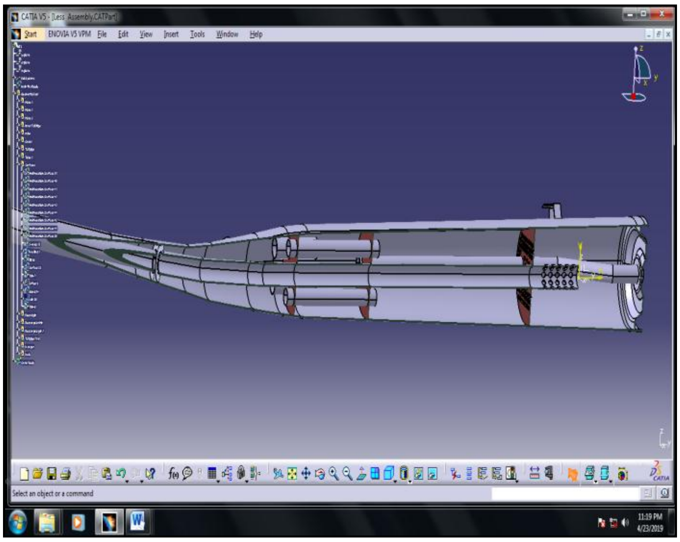

Fig. 5.2 Cut Section of Mass Reduced Silencer in CATIAV5R20

CAD model of modified geometry has created in CATIA V5 and imported in ANSYS16 software for modal analysis in free-free condition. The results obtained by modal analysis are explained in results and discussion.

6. RESULTS AND DISCUSSION

The results obtained for modal analysis of existing silencer by FEA and experimental testing has been given below.

6.1 Validation of Experimental Results with Analytical Results

Silencer has tested by using finite element analysis software that is ANSYS16. In ANSYS16, modal analysis module has used to find out natural frequency. The eighteen natural frequencies are obtained for eighteen mode number. For free-free analysis, there are no loads or constraints, there will be 6 rigid body modes, three translational and three rotational. This means the body will not undergo any internal deformation but will be able to move or rotate freely. The first six natural frequencies are zero because of the free-free boundary conditions. Silencer has two kind of modes that are rigid body and flexible mode. All structures can have up to six rigid body modes, three translational modes in X, Y, Z-direction and three rotational modes about X, Y, Z-axis. Each mode has zero frequency because of rigid body modes and rigid body modes do not involve any deformation of silencer so it is neglected. On other hand, flexible modes involve deformation of the silencer and has different value of natural frequency. Each flexible mode has explained as follows.

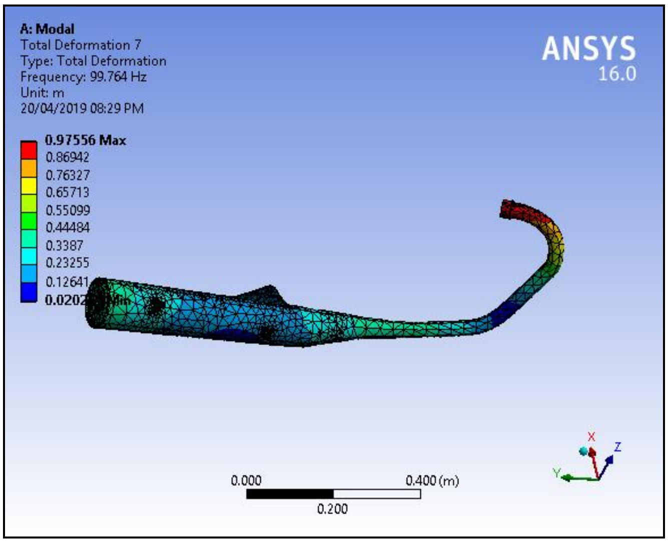

Fig.6.1 First Mode and Natural Frequency of Existing Silencer

The above Fig.6.1 shows that first mode of silencer whose natural frequency is 99.764 HZ. This mode shape shows that the deformation is more at extreme end of tail pipe of silencer. The maximum deformation is indicated by red color and minimum deformation in indicated by blue color. While other color shows the intermediate deformation. In this mode shape silencer bend at tail pipe.

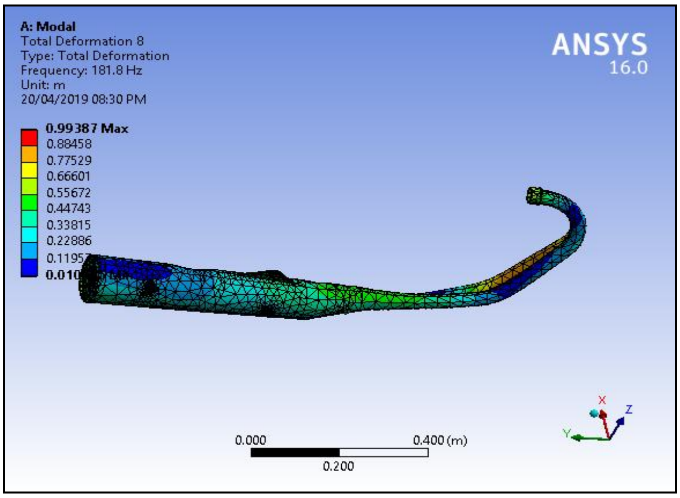

Fig. 6.2 Second Mode and Natural Frequency of Existing Silencer

The above Fig 6.2 shows second mode shape of silencer with deformed shape and undeformed shape whose natural frequency is 181.8 HZ. As compered to natural frequency of first mode shape, the natural frequency of second mode is more than the first mode shape. The increase in natural frequency causes deformation in middle of tail pipe is more which is indicated by orange color and yellow color. The minimum deformation in silencer at the end of main pipe shown by blue color. Silencer bend more at mid-section of main pipe in X-direction.

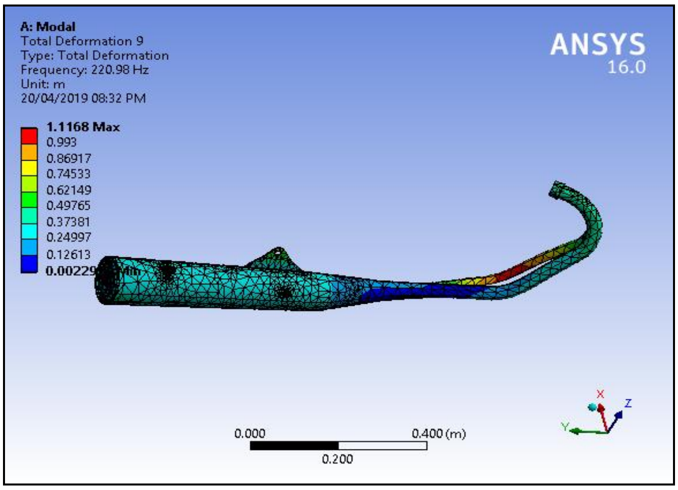

Fig. 6.3 Third Mode and Natural Frequency of Existing Silencer

The third mode with natural frequency 220.98 Hz shown in above Fig. 6.3. From the above three mode shapes, it is found out that as increase in natural frequency causes to increase in amplitude of maximum deformation. Also, in this mode shape deformation is more in mid-section of tail pipe which is shown by red color but amplitude of maximum deformation increased. In this mode shape mid-section of tail pipe bend more in X-direction. Similarly, for mode number 4 the deformation in tail pipe is more and tail pipe bend at mid-section in X-direction. For mode number 5 and 6 silencer bend at mid-section in Z-direction. As increase in mode number natural frequency increases and mode shape becomes complex. The results obtained in finite element (FEA) analysis are tabulated in table 6.1. To validate this result experimental testing has been done by using FFT analyzer. Roving impact hammering has been given twelve natural frequency for twelve mode shape along with time domain ODS and frequency domain ODS. Time domain ODS shows deformation of silencer at discrete time along with its corresponding amplitude and the fourier transform transforms a time domain signal into a frequency domain representation of that signal. Frequency domain ODS shows value of frequency along with its corrosponding amplitude of acceleration. The result obtained by using experimental analysis is tabulated in table 6.1.

Silencer has tested analytically to find out the natural frequency of existing silencer and to validate this result experimental testing of silencer has done by using Fast Fourier Transform (FFT) analyzer. Then, analytical results and experimental results are compared. By comparing both results found out that both results are approximately same. The variation in result are from 0.22 % to 13.23 %. So, such variation in results has accepted. The analytical results have validated by experimental testing.

To find out the forced frequency of silencer the experimental testing has been done while two-wheeler running at ideal speed. Again, for experimental testing FFT analyzer used for obtaining results of forced frequency along with time domain ODS and frequency domain ODS. The obtained forced frequency is 18.625 Hz.

Table 6.1 Comparison of Experimental Results with Analytical Results

Mode Number	Experimental Result (Hz)	Analytical Result (Hz)
1	100.72	99.764
2	155.83	181.8
3	162.45	220.98
4	433.19	270.74
5	439.66	290.09
6	457.69	461.26
7	508.05	477.41
8	622.66	624.09
9	719.57	653.11
10	758.96	822.54
11	763.24	885.71
12	780.26	971.83

6.2 Results of modified silencer

The modal analysis of modified silencer is carried out using finite element analysis software ANSYS16. The modification in silencer geometry is done as mass as main parameter. So, there are two cases of modal analysis of modified silencer which are modal analysis of increased mass of silencer and modal analysis of decreased mass of silencer. The modification of silencer is done by increasing thickness of baffle plates by 0.25 mm which leads to increase mass of silencer from 6.97 kg to 7.21 kg which has significant effect on natural frequency of silencer. As natural frequency is directly proportional to stiffness of silencer and inversely proportional to mass of silencer. So, increased in mass of silencer has to be reduced natural frequency. In below table 6.2, the result of Existing model and increased mass of silencer shows that natural frequency of silencer is decreased from 99.764 Hz to 97.959 Hz for mode number 1 also for other mode number natural frequency is decreased. Similarly, eighteen mode shapes are obtained for modal analysis of modified silencer like modal analysis of existing silencer. Out of which six are rigid body mode while remaining are flexible body modes. Rigid body modes have no deformation and has zero frequency so it is neglected. In following fig.6.4 shows first flexible body mode and its corresponding natural frequency. The below Fig.6.4 shows that first mode of silencer whose natural frequency is 97.959

Hz. This mode shape shows that the deformation is more at extreme end of tail pipe of silencer which is shown by red color. While minimum deformation is shown by blue and sky-blue color. The silencer bends at tail pipe section. By modifying silencer geometry, the amplitude of maximum deformation 0.97556 m has reduced to 0.957887 m.

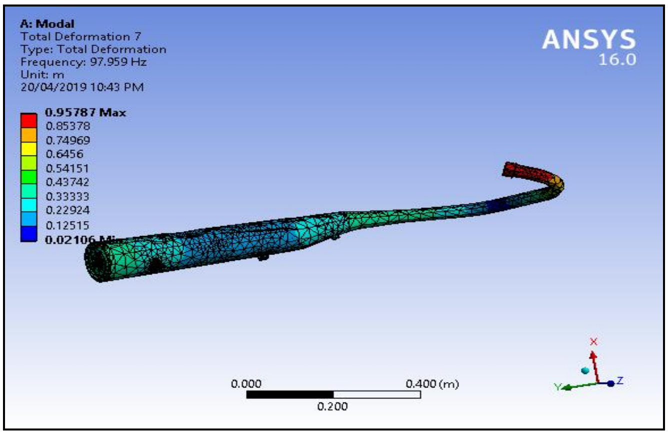

Fig. 6.4 First Mode and Natural Frequency of Modified Silencer

The following fig. 6.5 shows second flexible body mode and its corresponding natural frequency. The increase in natural frequency causes deformation in middle of tail pipe is more which is indicated by orange color and yellow color. The minimum deformation in silencer at the end of main pipe shown by blue color. Silencer bend more at mid-section of main pipe in X-direction.

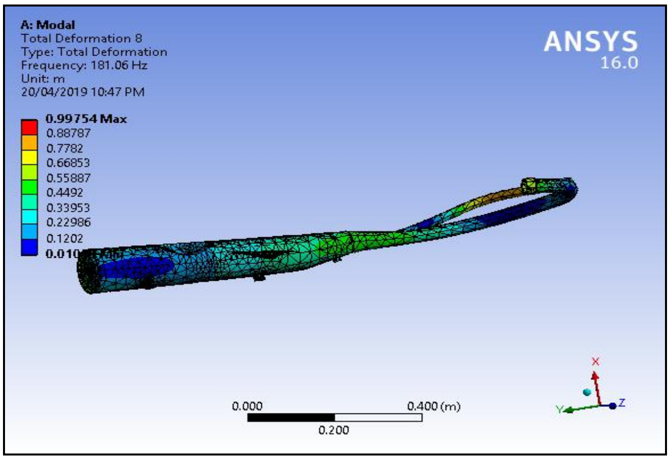

Fig.6.5 Second Mode and Natural Frequency of Modified Silencer

The third flexible body mode with natural frequency 223.62 Hz is shown in fig.6.6. The deformation in silencer body is more at tail pipe section which is shown by red and orange color and minimum deformation in silencer is at end of main pipe.

For mode number 4 deformation in silencer is same as third mode number but natural frequency and total deformation is different. As like mode shape of existing silencer, the mode shape of modified silencer for mode number 5 and 6 are bend at mid-section in Z-direction.

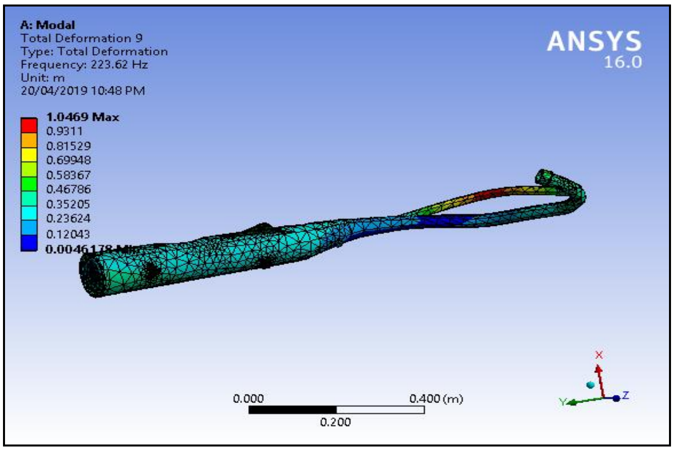

Fig. 6.6 Third Mode and Natural Frequency of Modified Silencer

The complexity of mode shape increases as increase in natural frequency. The natural frequency of corresponding mode number is shown in table 6.2. Similarly, in second case of modal analysis of decreased mass of silencer has been done by ANSYS16 software. The modification of silencer is done by decreasing thickness of baffle plates by 0.25 mm which leads to decrease mass of silencer from 6.97 kg to 6.71 kg. So, decreased in mass of silencer has to be increased in natural frequency and corresponding increase in natural frequency is shown in table. 6.2. The below Fig. 6.7 shows that first mode of silencer whose natural frequency is 100.97 Hz. This mode shape shows that the deformation is more at extreme end of tail pipe of silencer which is shown by red color. While minimum deformation is shown by blue and sky-blue color. The silencer bends transversely at tail pipe section. By modifying silencer geometry, the amplitude of maximum deformation 0.97556 m has increased to 1.0771 m. The characteristics of remaining modes are same as explained for increased mass of silencer.

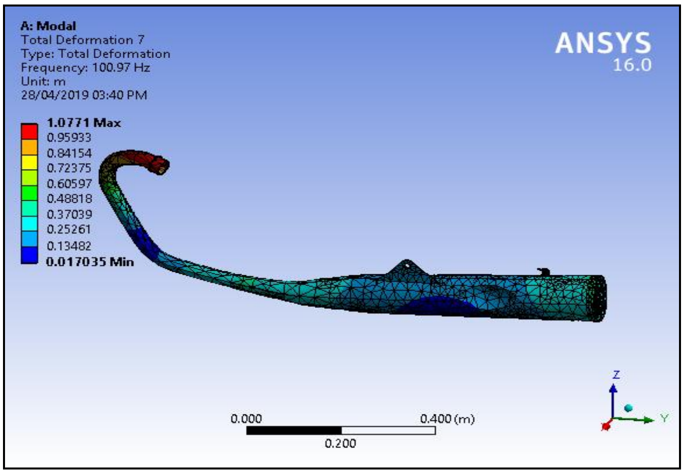

Fig. 6.7 First Mode and Natural Frequency of Modified Silencer

In Following Table. 6.2, the result of natural frequency for Existing model of silencer solved in ANSYS16 are compared with result of increased mass of silencer and decreased mass.

Table 6.2 Result for Comparisons

Mode Number	Natural frequency for Existing silencer (Hz)	Natural frequency for increased mass of silencer (Hz)	Natural frequency for decreased mass of silencer (Hz)
1	99.764	97.959	100.97
2	181.8	181.06	181.83
3	220.98	223.62	224.22
4	270.74	271.38	271.6
5	290.09	285.57	289.2
6	461.26	470.12	466.79
7	477.41	473.36	487.33
8	624.09	629.1	605.48
9	653.11	647.24	684.8
10	822.54	823.01	805.34
11	885.71	873.48	875.7
12	971.83	958.45	928.05

7. CONCLUSION

From design point of view, the modification of silencer is done by increasing thickness of baffle plates by 0.25 mm which leads to increase mass of silencer from 6.97 kg to 7.21 kg which has significant effect on natural frequency of silencer. As natural frequency is inversely proportional to mass of silencer. So, increased in mass of silencer has to be reduced the natural frequency. In above Table 6.2, the result of Existing model and increased mass of silencer shows that natural frequency of silencer is decreased from 99.764 Hz to 97.959 Hz and maximum deformation has reduced from 0.97556 m to 0.957887 m. Similarly, the modification of silencer is done by decreasing thickness of baffle plates by 0.25 mm which leads to decrease mass of silencer from 6.97 kg to 6.71 kg. So, decreased in mass of silencer has to be increased natural frequency of silencer. In above Table 6.2, the result of Existing model and decreased mass of silencer shows that natural frequency of silencer is increased from 99.764 Hz to 100.97 Hz and maximum deformation has increased 0.97556 m to 1.0771 m. For increased mass of silencer maximum deformation has reduced while for decreased mass of silencer maximum deformation has increased so out of two cases first case that is increased mass of silencer reduces vibration and optimizes the results. With reduction in frequency of vibration for silencer as vibration has direct effect on noise of silencer the noise of silencer can be reduced. As the there are no changes done in transmission path of silencer so other functional parameter of silencer like back pressure, insertion loss is not affected.

8. FUTURE SCOPE

The resonance condition of silencer can be avoided by knowing the natural frequency of silencer also for future work the modified silencer can be manufactured and it can be tested by using FFT Analyzer. This modified design of silencer will provide better solution and the increase in performance of silencer and improved life of silencer.

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