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Modal Analysis of Exhaust System to Optimize Mounting Hanger Location

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***Abstract*— An exhaust system with a superior performance becomes unserviceable if its durability is insufficient, for example, due to excessive level vibrations. This excessive level of vibration caused by various excitation forces from engine and road surfaces are transferred on to the exhaust hangers which plays a vital role in clamping the exhaust systems in proper place, thus damaging the hangers much before its service life. Hence it becomes obligatory for NVH engineers to optimize the hanger location so that it undergoes minimum damage, thus increasing its durability. This paper presents a modal analysis approach for optimizing the hanger location using FEA and comparing the results with experimental modal analysis. For FEA technique, Hypermesh was used as a pre and post processor whereas Nastran was used as a solver. The methodology adopted here was to determine node and antinode points on the exhaust system so that the mounting hangers can be shifted to node points. Hence after the identification of critical frequencies, its mode shapes were analyzed to identify optimum hanger location. Further the results were compared by performing experimental modal analysis using LMS Data Acquisition System.**

Keywords*— *NVH; Exhaust system; Modal Analysis; Natural Frequency; Mode Shape; MAC Diagram

I. INTRODUCTION

Exhaust systems are subjected to many dynamic input loads, the most important one coming from the engine and road surface. The induced vibrations are spread along the exhaust system, and forces are transmitted to the car body through the attached points, mainly holding hangers. Due to this, these holding hangers undergoes maximum damage. To tackle this problem, a NVH engineer can use modal analysis technique. Modal analysis has turned into a real option to give an accommodating commitment in understanding control of numerous vibration phenomena which are experienced in everyday practice. Deciding the nature and degree of vibration response levels and confirming theoretical models and prediction are both significant targets that can be accomplished with experimental modal testing.

II. FEA TECHNIQUE

First, a FEA technique was used to perform modal analysis. In this approach, starting from the structural geometry, the boundary conditions and material characteristics i.e. mass, stiffness and damping distribution of the structure is expressed in terms of respective matrices. Theses contain sufficient information to determine the system modal parameters. Nowadays advanced software packages such as Hypermesh and Nastran are available.

Hypermesh is a general purpose finite element modelling package whereas NASTRAN for numerically solving a wide variety of mechanical problems. These problems include: Static/Dynamic Structural Analysis (both linear and nonlinear), Heat Transfer and Fluid Problems, as well as Acoustic and Electro-Magnetic problems. The combination of these two software packages is effectively used for solving modal analysis problem in this work.

A. Description of the elements used

Shells are essentially 2-D elements that represent 3-D space, thus the term 2.5-D is also used. Shells are excellent elements for thin 3-D structures, such as body panels, sheet metal, injection moulded plastic or any part that can be described as having a thickness that is small relative to its global dimensions. Deflections are given at the nodes, but stresses can be found at the upper and lower surfaces as well as at the midplane. This gives the analyst the ability to extract membrane effects versus bending effects in the results.

The welds and bolted connections are simulated using rigid and Beam elements respectively. Outer surface of the muffler was meshed with shell elements of appropriate thickness. Suitable material properties were assigned to the shell elements of outer surface of muffler to match the mass of the muffler. Flex pipe is modelled with CBUSH element and its mass is represented by point mass. Contact between straps and muffler is modelled with CBUSH elements. In order to account for the weight of glass wool and other substrate materials, NSM (Non Structural Mass) element was used.

B. Boundary Conditions

Much attention is required in specifying boundary conditions. Improper specification of the boundary conditions may cause various problems. In this case to extract all natural frequency at the system level, free-free boundary condition is used. Fig. 1 shows boundary conditions for a conventional exhaust.

C. Material Properties

The material properties such as Young's Modulus, poisons ratio and density are required as an input values for performing modal analysis. For the flex pipe present in the system, the stiffness values in X, Y and Z directions are also required. The mechanical properties of the constituents of these considered exhaust systems are listed in Table I. and the stiffness values of flex pipe are listed in Table II.

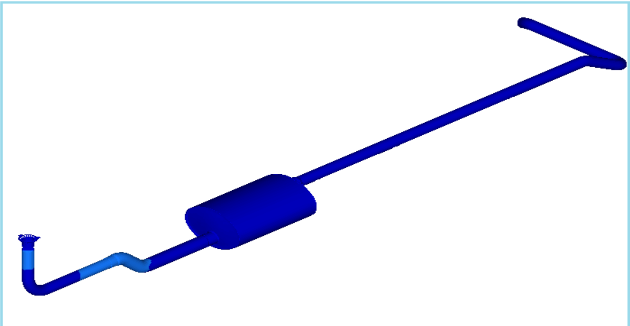

Fig. 1. Free-free boundary condition for conventional exhaust system

TABLE I. MATERIAL PROPERTIES OF EXHAUST SYSTEM

Material Properties	Conventional exhaust system
Material and Grade	IS 3074 ERW 1
Young's Modulus	210000 N/mm2
Poisson's Ratio	0.29
Density	7.8 X 10-9 T /mm3
Yield Strength	240 Mpa

TABLE II. STIFNESS VALUE OF FLEX PIPE

Direction	Stiffness value (N/mm)
In Compression (Kx)	50
In extension (Ky)	50
In free bending (Kz)	107.91

D. Assumptions

i. Holes on tubes in internal parts of muffler are not considered.

ii. Heat shields are not considered.

iii. The analysis is carried out for free-free condition.

iv. Glass wool and other substrate materials are modelled with NSM

E. Procedure for Mode Extraction

The middle surfaces are extracted from the geometry. The surfaces are meshed with shell elements. The welds and bolted connections are simulated using rigid and Beam elements respectively. Outer layer of the muffler has been meshed and has been assigned with suitable material density to match the actual weight of the muffler. The meshed model is then checked for quality checks such as wrap angle, aspect ratio, jacobian etc. By giving required loading conditions and choosing desired output, this FE model is submitted to solver (MSC Nastran) for extracting modal frequencies and mode shapes. Modal frequencies obtained are listed in Table III.

TABLE III. MODAL FREQUENCIES OBTAINED

Mode No.	Frequency
Mode 1	12.7
Mode 2	22.4
Mode 3	26.1
Mode 4	37.9
Mode 5	40.9
Mode 6	44.3
Mode 7	66.1
Mode 8	71.6
Mode 9	83.6
Mode 10	88.8

Mode shapes of Conventional Exhaust System at few critical frequencies are shown in Fig. 2, Fig. 3 and Fig. 4

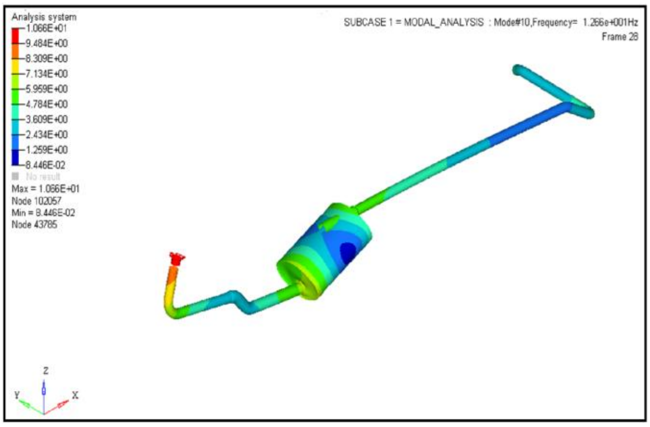

Fig. 2. 1st Mode of conventional exhaust system

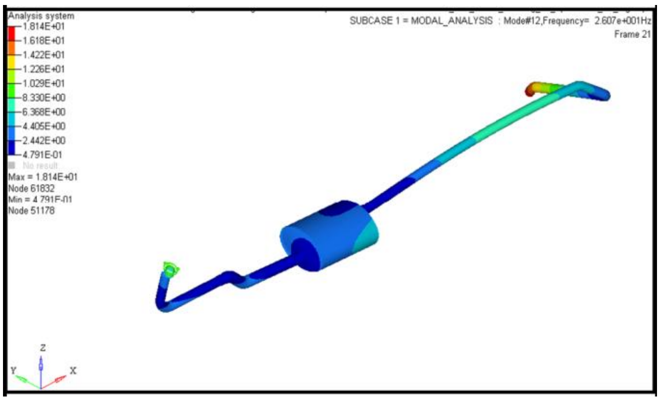

Fig. 3. 2nd Mode of conventional exhaust system

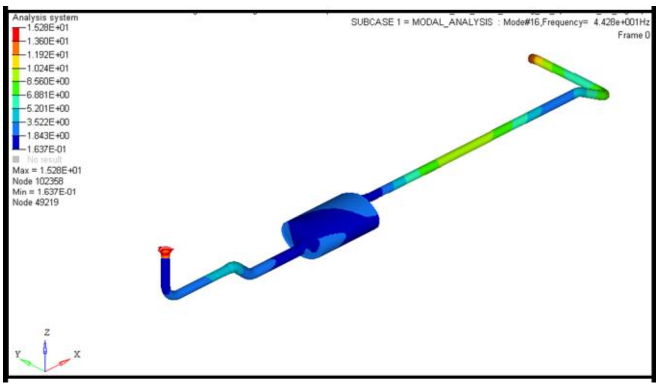

Fig. 4. 6th Mode of conventional exhaust system

III. Pre Test

In order to take measurements on physical structures, the modal information of preliminary Finite Element (FE) model is used to define the optimal measurement set-up for physical testing. In modal test setup, measuring points where accelerometer will be placed and excitation points are identified. LMS Virtual.Lab Correlation provides tools to carry out this pre-test analysis.

The objective is:

- To excite the structure at the right Degrees of Freedom (DOFs) as to excite the structures dynamics (modes) as good as possible.
- To measure the vibrations at the right DOFs to capture the requested real-life structural dynamic information.

Following are the steps involved in pre-test.

A. Import and visualization of FE modal data

The first step consists importing an FE model, including mode shapes, and visualizing the modes of the structure. The imported model is first investigated for displacement mode shapes. This will provide information on where to put the measurement points. Fig. 5 shows the imported FE model of conventional exhaust system on which geometry is created.

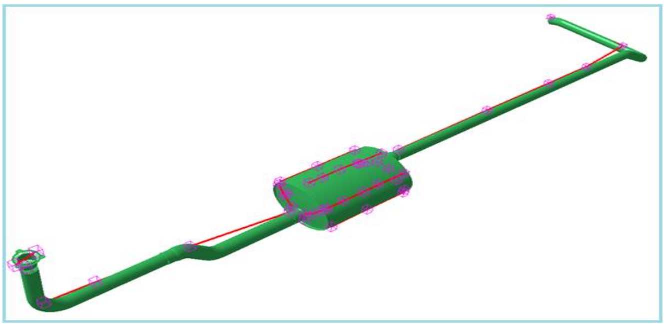

Fig. 5. A test wireframe for conventional exhaust system

B. Creation of the Test Geometry

The nodes of the wireframe, which will contain the measuring points of structure, need to capture the relevant mode shapes of the FE model. If the wireframe has poor spatial resolution, it cannot distinguish between different mode shapes, typically in the higher frequency range. The goal is to capture all relevant modal information, with a limited set of measuring points. This reduces test effort, cost and time. Typically, start off with a somewhat coarse wireframe. After having carried out a first pre-test analysis, the points can be easily added and/or moved until good pre-test result is obtained. LMS Virtual.Lab Correlation provides the tools to create the wireframe interactively and productively.

C. Evaluation and Refinement of the Test Geometry

To evaluate if the set of measuring DOFs is capable of distinguishing between all relevant mode shapes, the MAC (Modal Assurance Criterion) is used. It provides a quantitative measure to express how well/poor the mode shapes of two different mode sets are correlating. This guarantees that, if FRFs are measured in a physical test in these DOFs (as response DOFs), the test data will contain enough information to estimate all mode shapes and clearly distinguish between them. Fig. 6 shows MAC diagram for Conventional Exhaust System.

Once the evaluation and refinement of the wireframe model is completed, the points used to create wireframe can be used as locations for mounting accelerometer in experimental modal analysis. In this case, 43 points were used to create wireframe which are listed in Table IV.

TABLE IV. Coordinates To Locate Accelerometer On Conventional Exhaust System

Points	Node Id	Coordinates		
		X	Y	Z
1	38906	2171.862	-179.502	-599.702
2	39356	2182.011	24.4384	-674.164
3	39415	2172.206	-156.479	-434.396
4	39673	2180.332	207.4181	-599.519
5	39721	2171.609	29.8922	-379.206
6	39896	2158.239	222.512	-472.94
7	40942	1623.007	-181.517	-597.624
8	41399	1647.988	211.5442	-595.11
9	41447	1647.982	45.3584	-380.484
10	42602	1894.445	-175.279	-603.707
11	43715	1902.955	19.7621	-674.298
12	43917	1860.403	-156.998	-434.777
13	46261	1911.466	211.5543	-595.098
14	46314	1868.914	45.3136	-380.479
15	46912	1868.914	221.766	-471.725
16	47952	1605.402	11.1579	-674.345
17	47993	1596.55	-156.805	-434.635
18	48282	1596.342	217.2566	-465.084
19	48477	2188.748	197.4347	-601.146
20	48797	2188.75	-171.074	-589.634
21	49124	2198.759	68.4494	-499.644
22	49346	2194.703	-96.2882	-554.314
23	49364	2200.053	-18.3106	-504.213
24	49774	2193.564	140.5062	-549.384
25	50127	1570.736	-23.9196	-568.446
26	50199	1573.622	84.9391	-582.652
27	50266	1580.75	207.7272	-474.223
28	50395	1580.75	-147.04	-443.314
29	50679	1576.531	151.975	-531.826
30	50931	1574.042	-91.8684	-513.714
31	51698	2339.058	50.0374	-442.304
32	54671	3120.539	45.6596	-431.438
33	57548	4176.326	46.4321	-432.784
34	58168	3736.425	38.8433	-423.137
35	59919	4990.747	243.4751	-617.75
36	61831	5055.413	970.9751	-680.527
37	62108	1531.084	55.7097	-590.484
38	63975	884.929	56.6156	-497.677
39	66068	357.027	45.934	-434.001
40	66571	102.057	49.8433	-421.743
41	102078	108.824	29.7582	0
42	102085	0	0	0
43	102623	57.253	50.7631	-103.51

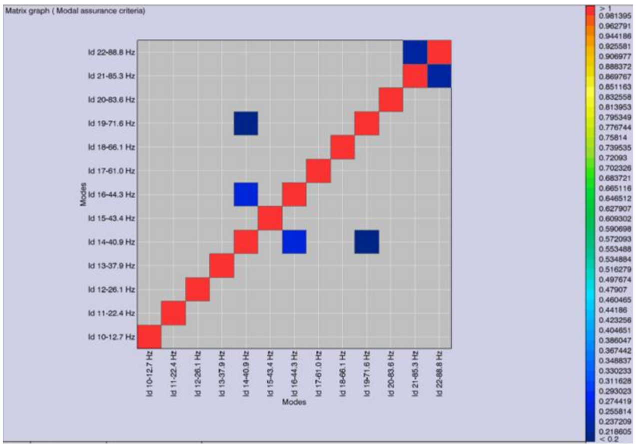

Fig. 6. MAC diagram for Conventional Exhaust System

IV. EXPERIMENTAL MODAL ANALYSIS

The experimental approach starts from the measurement of dynamic input forces and output response of the structure of interest. These measurements are often transformed into frequency response functions. They can be expressed in terms of modal parameters. The FFT analyzer is a batch processing device i.e. in samples the input signal for specific time interval collecting the sample in a buffer, after which it performs the FFT. A calculation on that batch and displays the resulting spectrum. The FFT analyzer used for this experiment is of LMS Test.Lab 12A.

A. Test Procedure

Using the accelerometer mounting locations from Table IV, wireframe geometry was created using LMS Test Lab. Fig. 7 shows the wireframe geometry for Conventional Exhaust System.

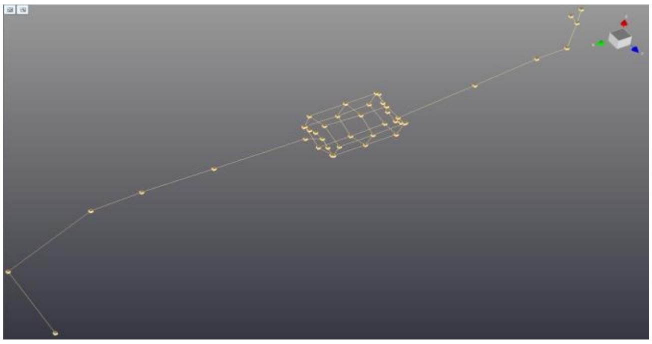

Fig. 7. Wireframe geometry for Conventional Exhaust System

For preparing the structure to perform experimental modal analysis, the exhaust systems are supported using bungee cords in order to simulate the free-free boundary condition which is shown in Fig. 8. All the points where accelerometers are to be mounted were marked with the help of measuring tape and marker. Also based on the structure, the exciting points are located and marked.

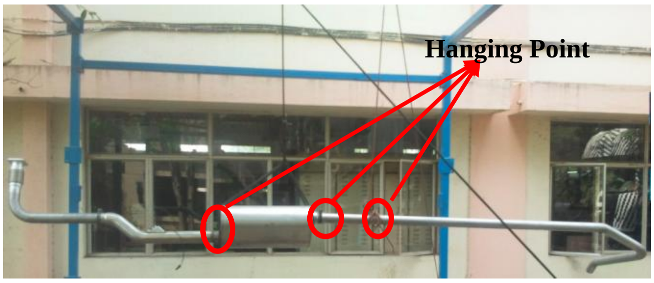

Fig. 8. Free-free hanging of Conventional Exhaust System

The excitation points are generally chosen with comparatively higher stiffness so as to excite whole structure. Fig. 9 shows the selected excitation point for this exhaust system.

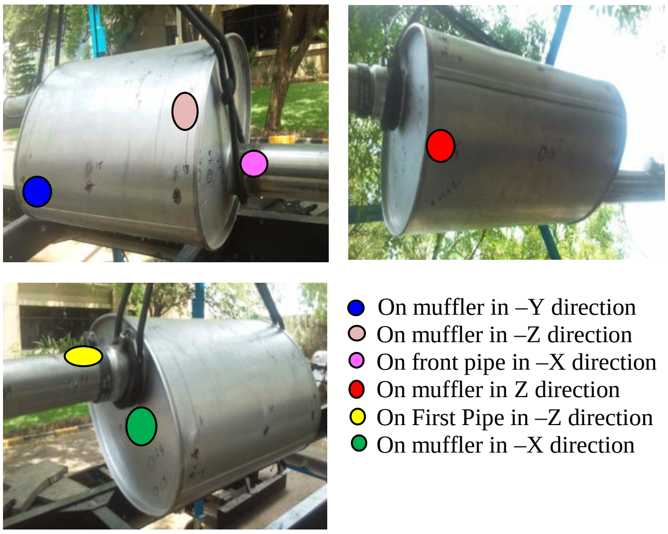

Fig. 9. Excitation points for Conventional Exhaust System

For acquiring the data some initial setup like channel set up, impact scope, trigger and windowing are done in LMS Data Acquisition System. After this initial setup the data is acquired through various accelerometers for different hitting locations. Once the data is acquired, it is post processed using LMS Test.Lab.

B. Post Processing

Post processing refers to activities that are involved after the completion of experimental modal analysis to extract results from the test. These include plotting the frequency response curves, coherence graphs, identifying the modes and extracting modal frequencies and shape.

1) FRF curve to obtain Model Frequency

Fig. 10 shows the summed FRF for Conventional Exhaust System. The summed FRF is a summation of FRF of all the points considered on the structure. The peak in sum FRF indicates the mode at particular frequency. The series of 's' in red color indicates that the mode is stable. The peaks with the stable mode denoted by 's' has to be accepted and one with the unstable mode has to be rejected. In this manner all the possible modes are selected which are displayed in left side of LMS Test.Lab window.

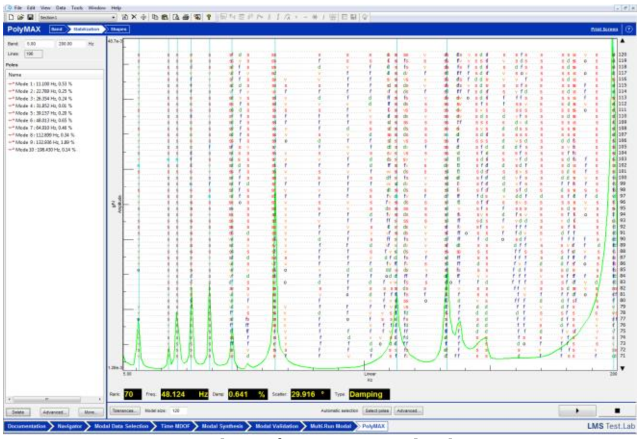

Fig. 10. Summed FRF for Conventional Exhaust System

Table V. shows the model frequencies obtained.

TABLE V. EXPERIMENTAL MODEL FREQUENCIES

Mode No.	Frequency
Mode 1	11.069
Mode 2	22.786
Mode 3	25.355
Mode 4	31.843
Mode 5	39.155
Mode 6	47.753
Mode 7	64.809
Mode 8	112.606

2) *Modal Validation*

The Modal Assurance Criterion (MAC) is a statistical indicator that is most sensitive to large differences and relatively insensitive to small differences in the mode shapes. This yields a decent measurement marker and a level of consistency between mode shapes. The MAC considers just modal shapes which imply that a different frequency comparison must be utilized as a part of conjunction with the MAC qualities to focus the corresponded mode pairs.

All the modes that were selected in the previous step are analyzed and validated. The undesirable ones are rejected after determining them from the MAC diagram. The modes which contribute to a greater extent (above 20%) on other mode are determined and rejected in the stabilization window. Fig. 11 shows the MAC diagram for Conventional Exhaust System.

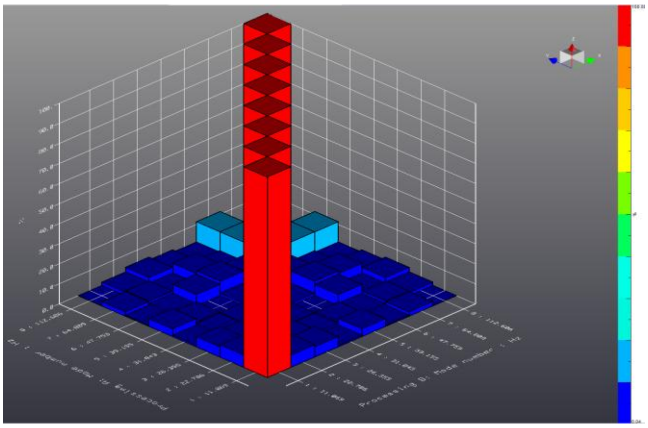

Fig. 11. MAC diagram for Conventional Exhaust System

3) *Mode Shapes*

After validating the modes in MAC, the mode shapes are extracted to visualize the behaviour of the structure. Mode shapes of Conventional Exhaust System at few critical frequencies are shown in Fig. 12, Fig. 13, and Fig. 14.

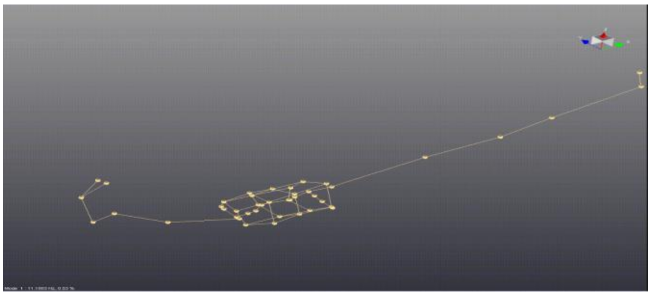

Fig. 12. 1st Mode of conventional exhaust system

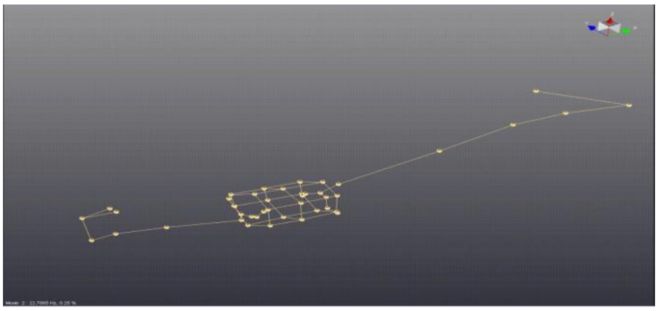

Fig. 13. 2nd Mode of conventional exhaust system

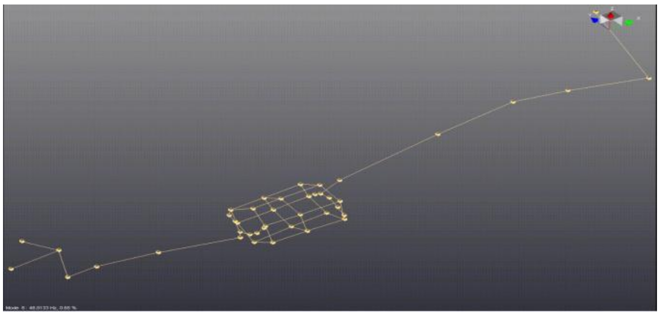

Fig. 14. 6th Mode of conventional exhaust system

V. RESULTS AND DISCUSSION

For the purpose of optimizing hanger location, modal analysis was performed. Table VI. shows the comparison of modal frequencies obtained for two exhaust systems using FEA and Experimental modal analysis approach.

TABLE VI. COMPARISON OF MODEL FREQUENCIES

Mode No.	Modal Frequency (Hz)	
	FEA	Experimental
Mode 1	12.7	11.069
Mode 2	22.4	22.786
Mode 3	26.1	26.355
Mode 4	37.9	31.843
Mode 5	40.9	38.155
Mode 6	44.3	47.753
Mode 7	66.1	64.809
Mode 8	71.6	----------
Mode 9	83.6	----------
Mode 10	88.8	112.606

i. Marginal variations between FEA and Experimental modal frequency values are observed. This is due to the assumptions considered while performing modal analysis through FEA.

ii. In mode shapes, nodal points are the point with minimum displacements from mean position and anti-nodal points are the points with maximum displacements from mean position. Nodal and anti-nodal points are identified in mode shape for location of mounting clamps. In Fig. 15, E1 and E2 are the existing locations for mounting clamps and P1 and P2 are the new proposed location for conventional exhaust system. Fig. 16 and Fig. 17 show the FRF comparison between existing and new proposed location. Prepare Your Paper Before Styling

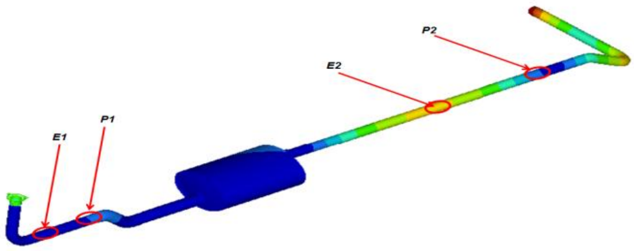

Fig. 15. Proposed mounting location for Conventional Exhaust System

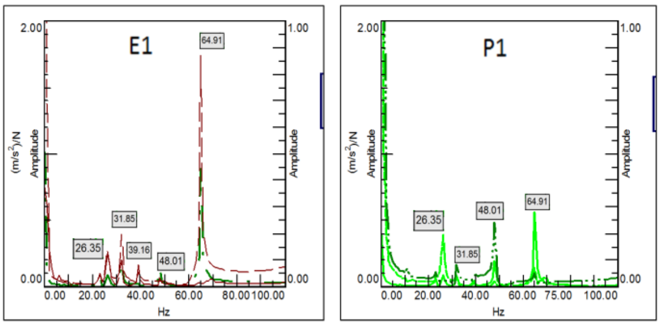

Fig. 16. FRF comparison of point E1 v/s P1

The peak amplitude in the existing location (E1) has a magnitude of 1.74 $(m/s^2)/N$ at 64.91Hz whereas the magnitude of amplitude at 64.91Hz in proposed location (P1) is 0.56 $(m/s^2)/N$.

% Reduction in amplitude = 100 - (0.56/1.74)*100

= 67.82%

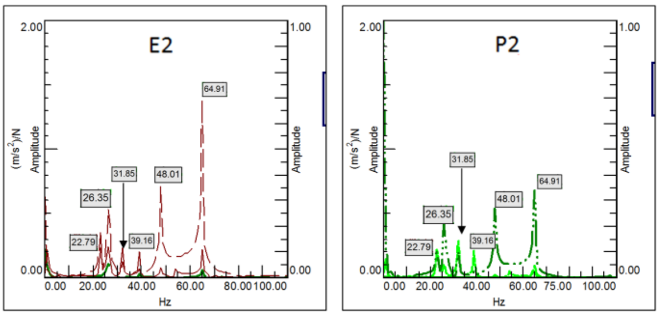

Fig. 17. FRF comparison of point E2 v/s P2

The peak amplitude in the existing location (E2) has a magnitude of 1.38 $(m/s^2)/N$ at 64.91Hz whereas the magnitude of amplitude at 64.91Hz in proposed location (P2) is 0.68 $(m/s^2)/N$.

% Reduction in amplitude = 100 - (0.68/1.38)*100

= 50.72%

From the above comparisons it can be seen that amplitudes of FRFs at a point of current locations are high. Ample amount of percentage reduction in the amplitudes is observed in case of proposed location. Thus durability of mounting brackets can be improved by shifting them to new proposed location.

VI. Conclusion

The purpose of this work was to optimize the hanger location using FEA technique and comparing the results with experimental modal analysis. For this purpose modal analysis was performed. New locations for mounting clamps have been suggested to increase its durability by shifting it to nodal locations. Obtained % reductions in amplitude of FRF in case of new proposed mounting location are 67.82% and 50.72%. Fatigue Life calculation can be done for new proposed mounting location to check the actual increased life and durability of the mounting clamps.

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