International Journal of Engineering Research and Development
***e**-ISSN: 2278-067X, **p**-ISSN: 2278-800X, www.ijerd.com*
Volume 4, Issue 5 (October 2012), PP. 39-48

Application of Design of Experiments (DOE) Technique to Vibro-Acoustic Transfer Path Analysis

Sahil P.Pandare[1], S.B.Wadkar[2]
[1,2]Department of Mechanical Engineering, SCOE, Pune University, Pune,

Abstract:—Road noise inside a vehicle's passenger compartment is caused by many sources. The most common is tyre rumble generated by contact of the tyres with the road surface. Other causes of low frequency noise are the vehicle's engine and power train and structural vibration (caused by vibration of the suspension or vehicle body). Road noise is transmitted through the vehicle into the vehicle cabin via two distinct paths: through the air (air-borne noise) and through the vehicle's bodywork (structure-borne noise). These paths are primarily governed by the frequency of the noise.Transfer path analysis is widely used terminology in noise, vibration and harshness of automobiles. It is a technique for identifying the vibro-acoustic transfer paths from the source to receiver through physical connections such as engine mounts or bushes. This paper describes a methodology of component sensitivity which is used for finding the critical transfer paths by using DOE (Design of Experiment) technique. Component sensitivity is a measure of change in response due to change in any component of the system i.e. change in stiffness of mounts/bushes. By performing design of experiments, critical transfer path is identified. The sound pressure level is reduced by varying the stiffness of bushes.

Keywords:—Transfer path analysis, sound pressure level,stiffness of bushes, design of experiments

I. INTRODUCTION

The noise in a vehicle is a result of the contribution of various sources, reaching the occupants via various transmission paths. The main sources are the engine and transmission, the intake and exhaust, the tyres and roads. The main transmission paths are the engine and wheel suspension, the exhaust hangers and the airborne transmission paths.

Vehicle components such as the engine, transmission, drivetrain, tyre and exhaust system transmit noise through the structural connection to the body which dominate the interior cabin noise. The excitations generated within these dynamic components induce structural vibrations. The vibrations are subsequently transmitted across bearings, casing and mounts to the body panels. Interior noise is radiated from these vibrating panels.

The structure-borne noise component tends to dominate overall noise levels except on the smoothest of road surfaces. The structure borne element become clear when a vehicle is operated on a rolling road with the body jacked on air jacks & the suspension link disconnected. When the vehicle moves over the rough road patch, vibrations are produced in tyre due to irregularities of the road surface. These vibrations are further transmitted to body structure through the suspension mounts and activate the acoustic mode in the cabin. If the body panels are not stiff enough, the effect of this structure borne noise will be higher.

Transfer path analysis is a test based procedure which allows to trace the flow of vibro-acoustic energy from a source to a given receiver location through a set of known structure-borne and air-borne paths. The aim is to find out the vector contribution of each path of energy from the source to receiver.

In complicated structures which involve many sub-assemblies, the vibro-acoustic sensations experienced by an observer at the receiver location are caused by vibration source away from the receiver location. Energy from a source in a car is transmitted into the passenger cavity by a number of different routes; from the engine mountings, the exhaust system connection points, via the drive shafts and the wheel suspensions.

The purpose of a Transfer Path Analysis is to determine which paths are dominant in transmitting vibrations or noise from one or several sources to one or more receivers. In this work the Transfer Path Analysis will be limited to structural transmitted vibrations, frequency less than 200 Hz.

The basis of the transfer path analysis process is to quantify the total energy response in terms of individual transmission paths and sources contributions into the vehicle. This approach allows breaking down each of the partial load contributions (force and volume velocity) and frequency response functions (FRF) between the load locations and the necessary target responses. As an example, the interior sound pressure at a particular position can be expressed as:

$$\mathrm{yk}(\omega) = \sum_{i=1}^{n} FRF_{ik}(\omega) * \mathrm{Fi}(\omega) + \sum_{\mathrm{j}=1}^{\mathrm{p}} FRF_{jk}(\omega) * \mathrm{Qj}(\omega)$$

$y_k(\omega)$ is target response at a certain point k
ω is the frequency
n are the number of paths
$F_i(\omega)_{(i = 1,\ldots,n)}$ are the structural loads or forces
$Q_j(\omega)_{(j = 1,\ldots,p)}$ are the acoustic loads, typically volume accelerations

$FRF_{ik}(\omega)$ and $FRF_{jk}(\omega)$ are the system response functions from the input loads to the target.
Transfer path analysis creates a physical mathematical model of the structure under test that allows:

- The quantification of source strengths
- The ranking of contributions of the different sources along the different paths for every target receiver positions

II. METHODOLOGY

Finite element modeling of sample model is carried out. The meshing and application of boundary conditions and constraints is carried out in Altair Hypermesh v11.Acoustic cavity is prepared inside the model. Modal analysis is performed to find the natural frequencies, mode shapes and checking connectivity between all components.Bushes are added to BIW at the connecting locations of chassis and cabin.

Modal analysis in free-free condition was performed using MSC/NASTRAN in order to calculate natural frequencies and mode shapes. Vibro-acoustic response analysis has been carried out to calculate noise transfer function NTF i.e. sound pressure level at given response location. Peak sound pressure levels were identified among all load cases.
Noise transfer function analysis is done again to get the critical transfer path of noise. This is done by varying bush stiffness and getting contribution of each bush with the help of DOE technique. Stiffness of bushes which are contributing more to sound pressure level is decreased to get the necessary change in SPL.

III. FINITE ELEMENT MODEL DEVELOPMENT

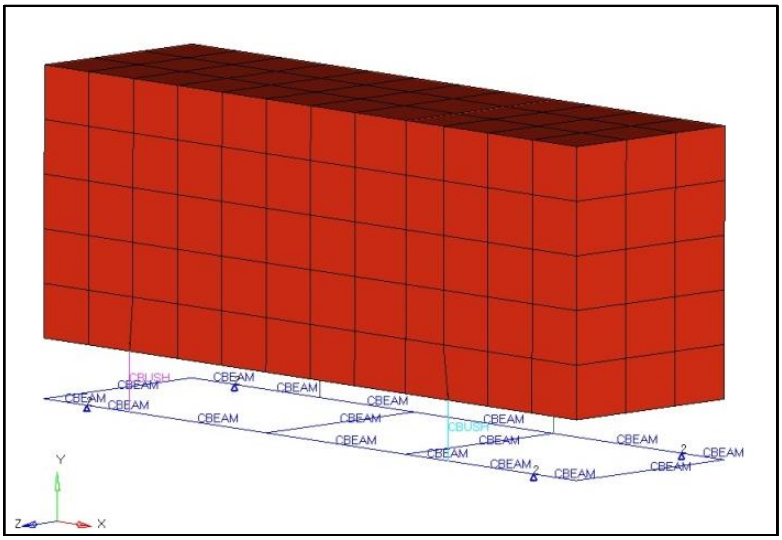

Fig. 1*:* FE Model of Structure

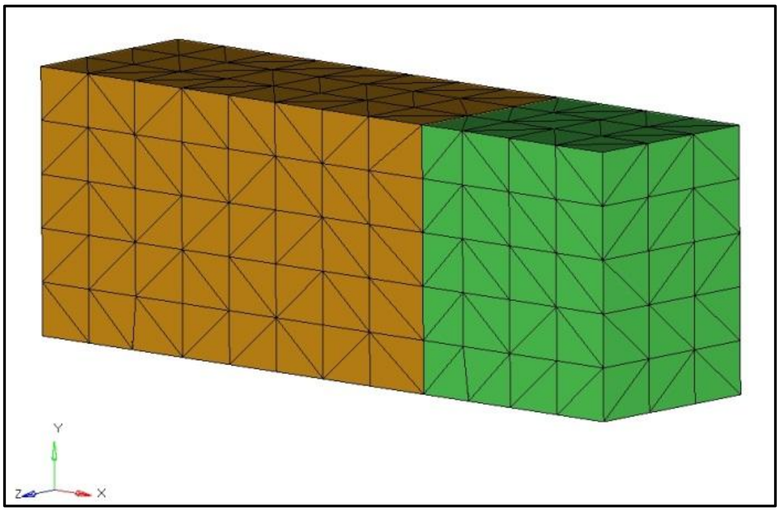

Fig.2*:* Acoustic Cavity Model

The primary structure is modeled with beam elements. There are four load application points on this frame and there are four elastic isolators or bushes between primary structure and the secondary structure. Unit force is applied on all the load application nodes. The load is applied at a frequency from 1 to 250 Hz. Here the bushes which contribute to the noise path are A, B, C and D.

Each of these bush elements isolates only the vertical degree of freedom. The secondary structure is modeled as shell elements. Material used for this structure is steel. Properties are as follows:

- Young's Modulus, E= 2.1*e^5 N/mm2
- Poisson Ratio, υ =0.3
- Density, ρ = 7.89e^{-9}tonne/mm^3
- MAT1 card image is assigned for the material.
- Type is 2D and card image for the property is PSHELL.

The secondary structure is divided into parts: front part represents cabin space and rear part represents passenger compartment. The acoustic cavity is modeled by using solid elements i.e. tetra elements as shown in Figure 2. Properties of air aregiven to it. Material type for air is FLUID and card image assigned is MAT10.
Properties for air:

- Type is 3D and card image is PSOLID.
- Bulk= 0.117MPa
- Density= 1.2e^{-9}tonne/mm^3
- Speed of sound in air= 3.4 e^5 mm/s.

A point inside the cavity is chosen as the response location.

IV. MODAL ANALYSIS

Modes are inherent properties of a structure and are determined by the material properties (mass, damping, and stiffness) and boundary conditions of the structure. Each mode is defined by a natural (modal or resonant) frequency, modal damping and mode shape (i.e. “modal parameters”). If either the material properties or the boundary conditions of a structure change, the modes will change. For instance, if mass is added to a structure, it will vibrate differently.

Modal analysis of structure and acoustic cavity has been carried out to extract frequencies and mode shapes. This frequency domain up to 200Hz was considered for the analysis. Modal analysis of the model is carried out for free free condition. The connection between the bushings and the main structure is checked for different frequencies.
Figure 3a, 3b and 3c show the three vibration modes of the structure.

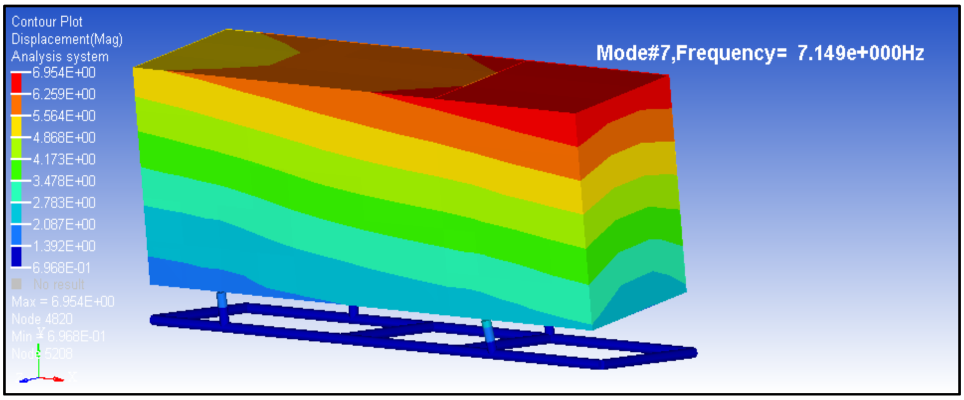

Fig. 3a: First Mode Shape, Frequency= 7.14 Hz

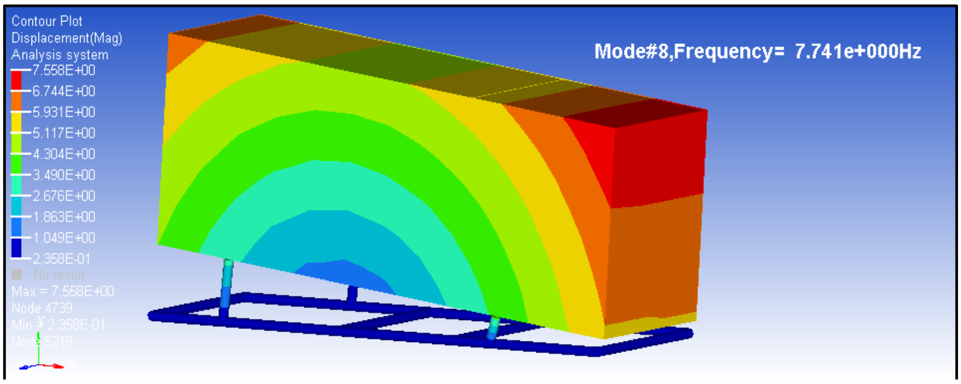

Fig. 3b: Second Mode Shape, Frequency= 7.74 Hz

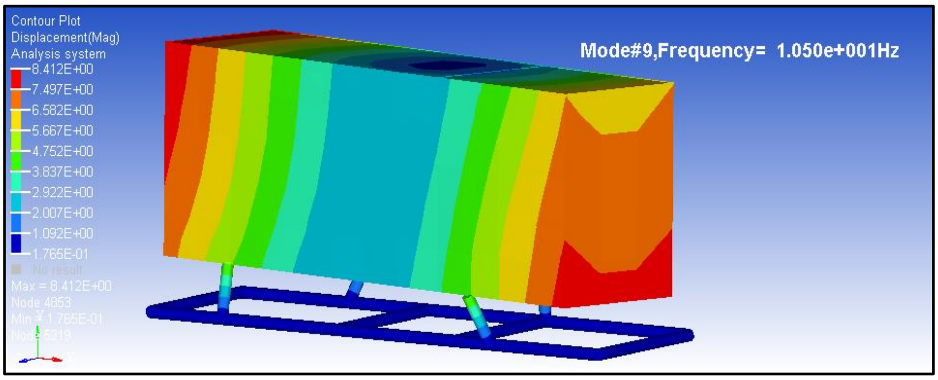

Fig. 3c: Third Mode Shape, Frequency= 10.5Hz

V. NOISE TRANSFER FUNCTION ANALYSIS

Frequency response analysis is a method used to compute structural response to steady-state oscillatory excitation.In frequency response analysis the excitation is explicitly defined in the frequency domain. All of the applied forces are known at each forcing frequency. Forces can be in the form of applied forces or enforced motions (displacements, velocities, or accelerations)

Unit forces are applied simultaneously at all the 4 nodes on the primary structure (frame). The forces are applied in translational Y – directions only. The sound pressure response is measured at a point taken in the front part (cabin side) of the cavity. SOL 111 (Modal Frequency Response analysis) is performed using MSC Nastran to measure the response. The following graph shows the output measured at node no. 5156. The abscissa represents the frequency in Hertz and the ordinate represents the pressure in Decibels.From the following graph, it is observed that the sound pressure is around 85 dB.

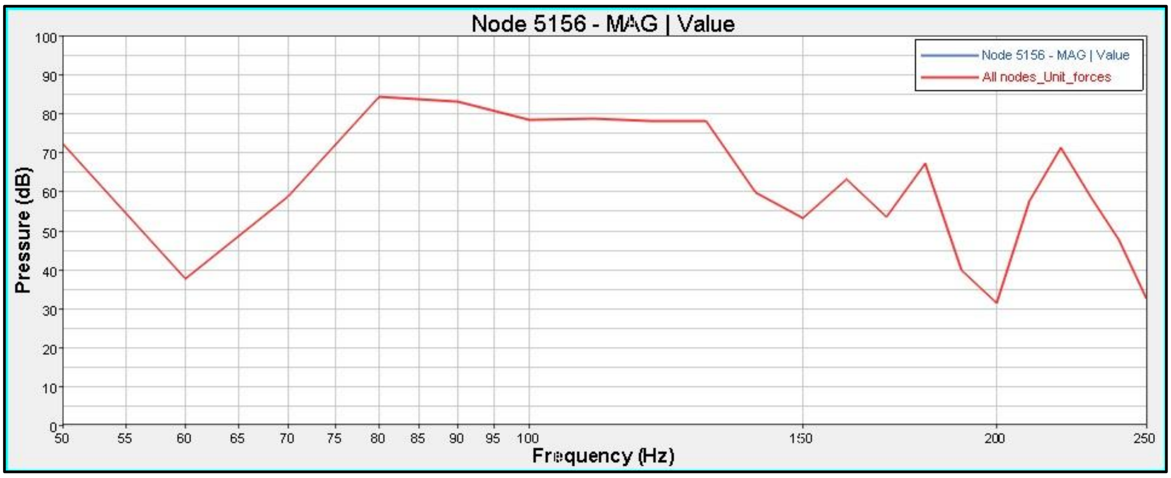

Fig. 4: Sound Pressure vs frequency for unit forces at all four nodes

VI. DESIGN OF EXPERIMENTS (DOE)

The design and analysis of experiments involves understanding the effects of different variables on other variable(s). In mathematical language, the objective is to establish a cause-and-effect relationship between a number of independent variables and a dependent variable of interest. The dependent variable, in the context of DOE, is called the response and the independent variables are called factors. Experiments are run at different factor values called levels. Each run of an experiment involves a combination of the levels of the investigated factors. Each of the combinations is referred to as a treatment. In a single factor experiment, each level of the factor is referred to as a treatment. In experiments with many factors, each combination of the levels of the factors is referred to as a treatment. When the same number of response observations is taken for each of the treatments of an experiment, the design of the experiment is said to be balanced. Repeated observations at a given treatment are called replicates. The number of treatments of an experiment is determined on the basis of the number of factor levels being investigated in the experiment. For example, if an experiment involving two factors is to be performed, with the first factor having x levels and the second factor having z levels, then x*z treatment combinations can possibly be run, and the experiment is an x z factorial design. If all xz combinations are run, the experiment is a full factorial. If only some of the xz treatment combinations are run, the experiment is a fractional factorial. In full factorial experiments, all of the factors and their interactions are investigated, whereas in fractional factorial experiments, all interactions are not considered because not all treatment combinations are run.

Designs of experiments (DOE) capabilities provide a method for simultaneously investigating the effects of multiple variables on an output variable (response). These experiments consist of a series of runs, or tests, in which purposeful changes are made to input variables or factors, and data is collected at each run. Quality professionals use DOE to identify the process conditions and product components that influence quality and then determine the input variable (factor) settings that maximize results.

The results of DOE are displayed on a pareto chart. Pareto chart is a type of chart that contains both bars and a line graph, where individual values are represented in descending order by bars, and the cumulative total is represented by the line.The purpose of the Pareto chart is to highlight the most important among a (typically large) set of factors.

Pareto charts are very useful because they can be used to identify those factors that have the greatest cumulative effect on the system, and thus screen out the less significant factors in the analysis. Ideally, this allows the user to focus attention on a few important factors in the process.

In this case, there are 4 bushes namely A, B, C and D. All these bushes are attached to the secondary structure. The stiffness of the bushes is kept as 50 and 100 N/mm in translational Y direction and $1*e^6$ N/mm. in rotational x, y and z directions. Type used for bush element is Springs/gaps and card image is PBUSH.By performing fractional DOE, taking 4 factors (4 bushes) and 1 treatment, levels are taken as 2, the formula for finding the different combinations is:

2^{k-1},k =No. of factors.

Hence, the total numbers of combination runs required are 8.The bush stiffness of each combination is varying. The response is measured for each combination run by performing FRF analysis. This will result in finding the most contributing bush to the noise level.The following graphs show the different stiffness combinations of bushes contributing to sound pressure.

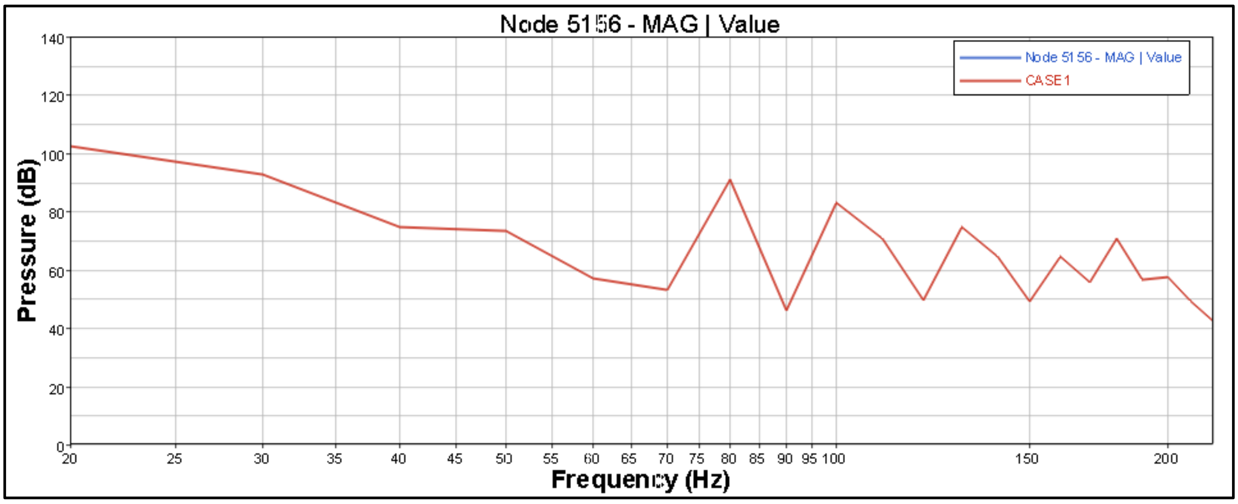

Fig. 5: Sound pressure vs frequency for First stiffness combination of bushes

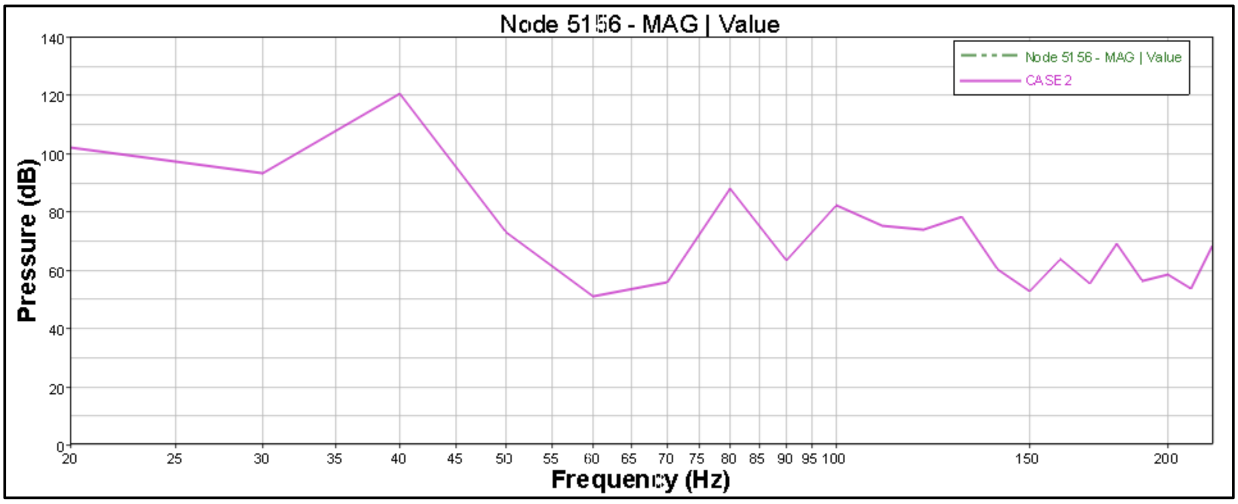

Fig. 6: Sound pressure vs frequency for Secondstiffnesscombination of bushes

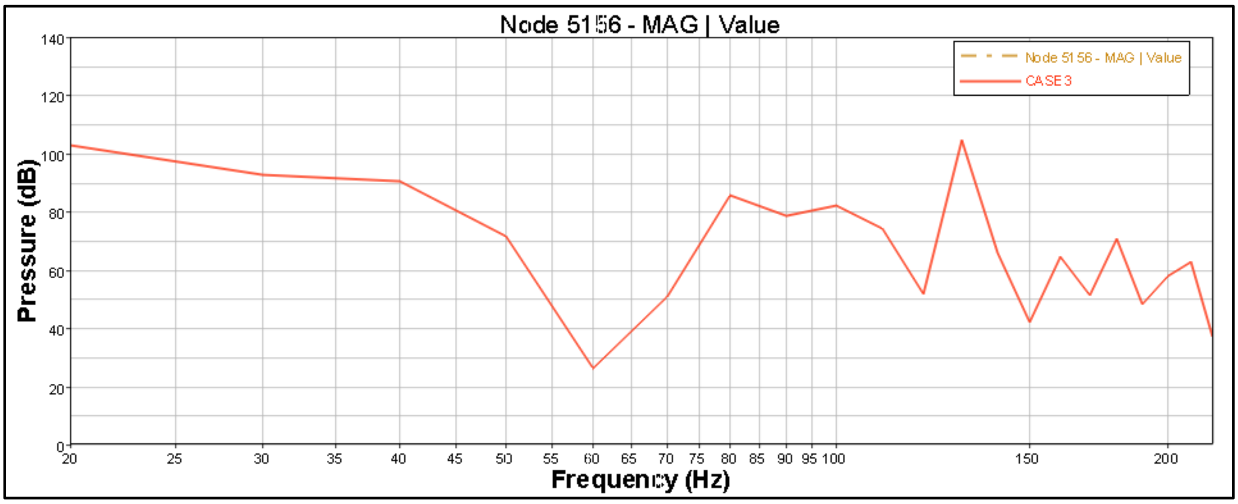

Fig. 7: Sound Pressure vs frequency for Third stiffness combination of bushes

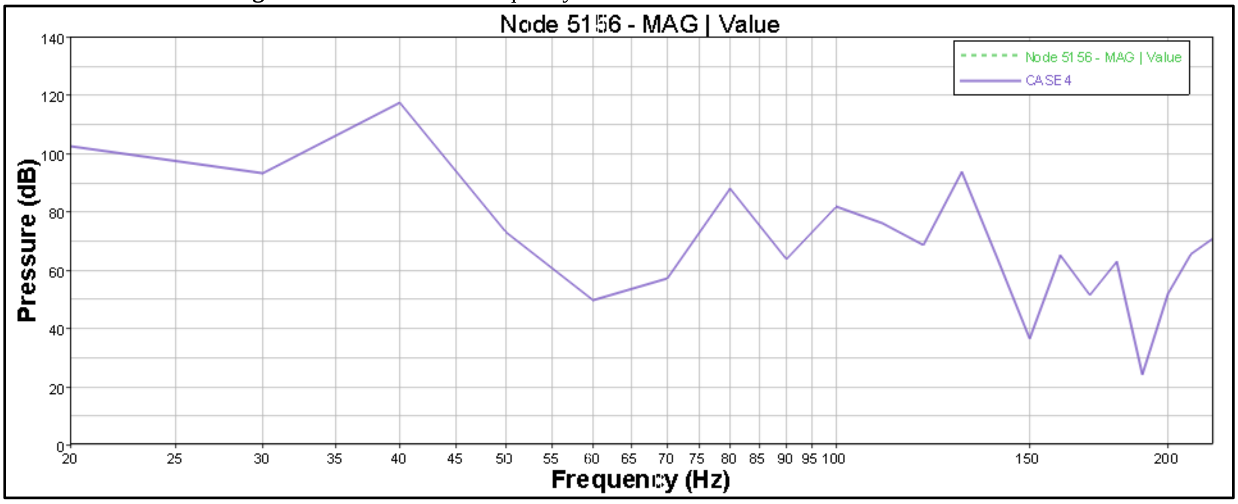

Fig. 8: Sound Pressure vs frequency for Fourth stiffness combination of bushes

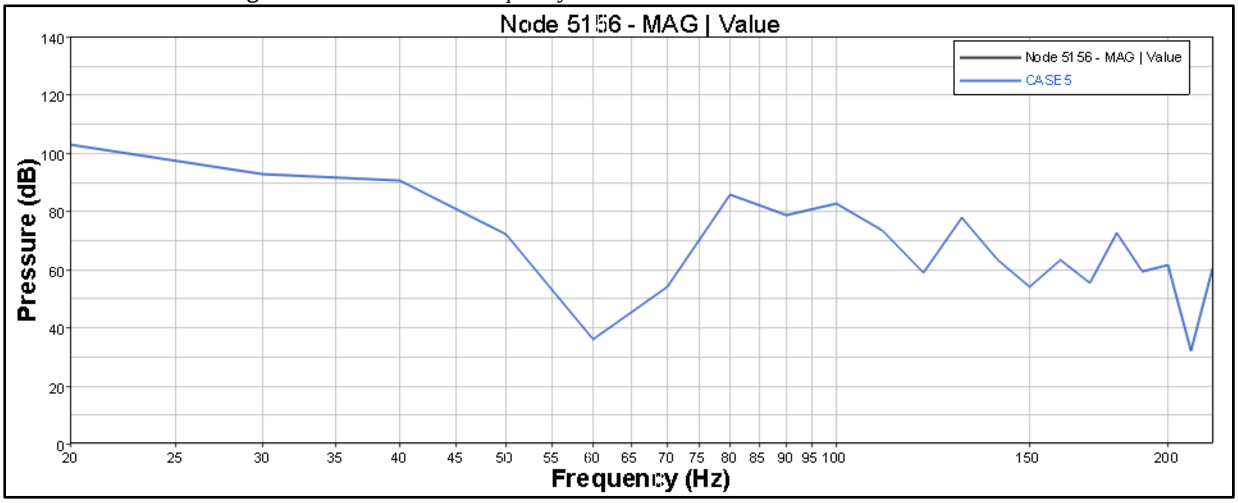

Fig. 9: Sound Pressure vs frequency for Fifth stiffness combination of bushes

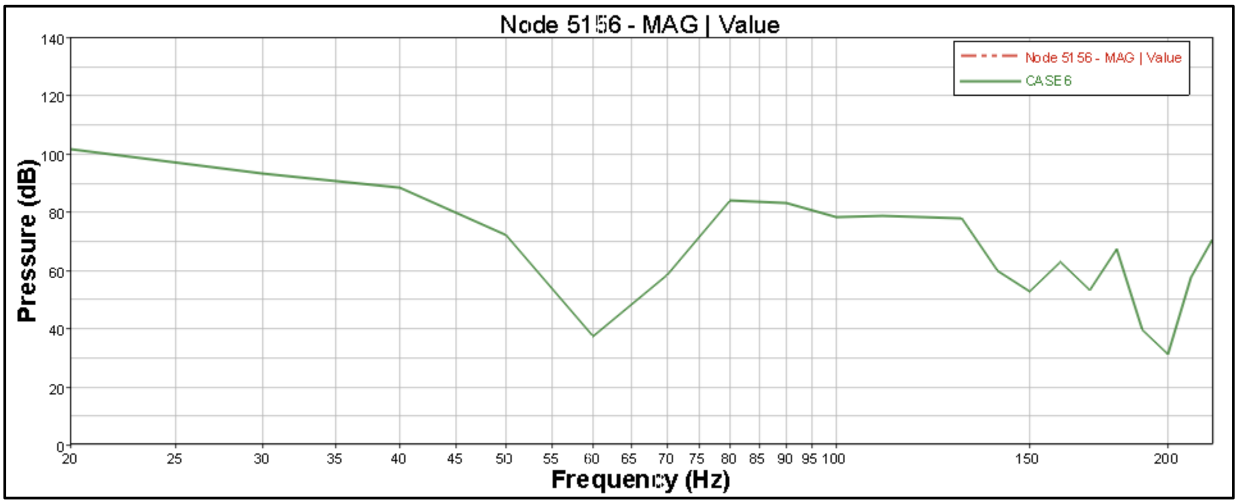

Fig. 10: Sound Pressure vs frequency for Sixth stiffness combination of bushes

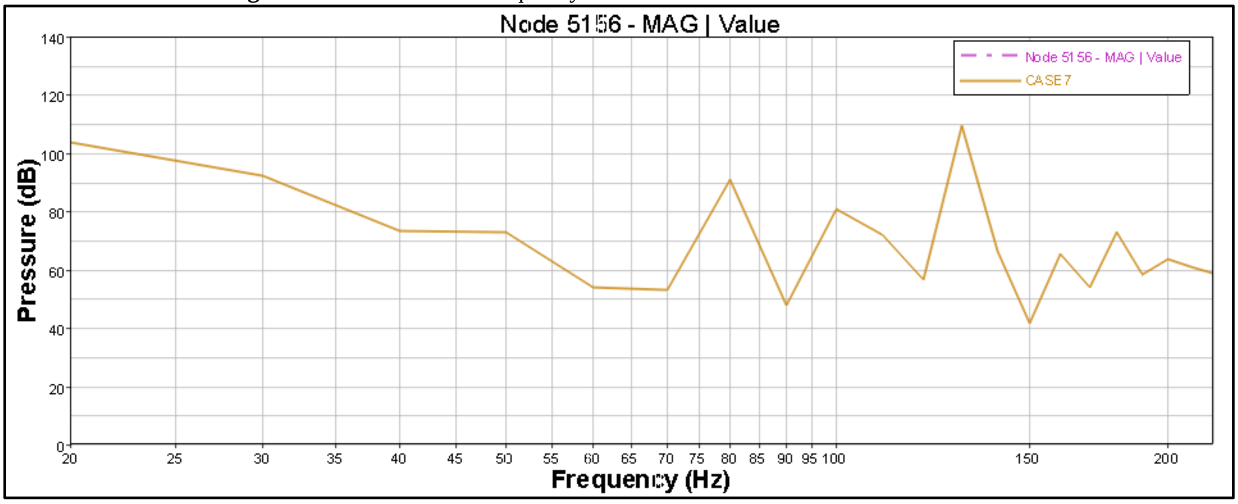

Fig. 11: Sound Pressure vs frequency for Seventh stiffness combination of bushes

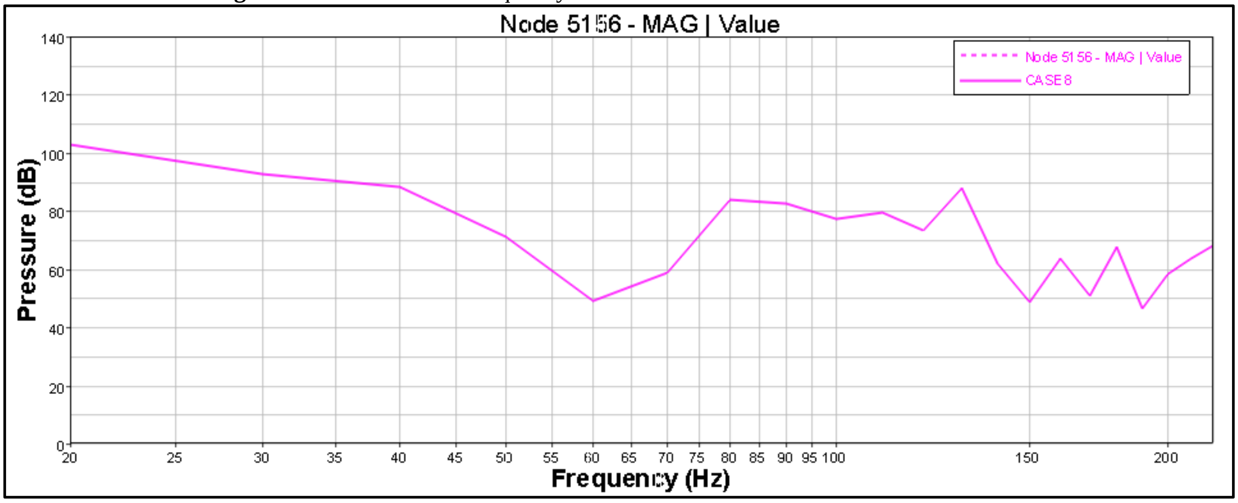

Fig. 12: Sound Pressure vs frequency for Eighth stiffness combination of bushes

Table 1: Sound pressure response for different bush stiffness combinations

Sr. No.	Stiffness of Bush A (N/mm)	Stiffness of Bush B (N/mm)	Stiffness of Bush C (N/mm)	Stiffness of Bush D (N/mm)	Sound Pressure Level (dB)
1	100	50	50	100	103.9
2	50	50	100	100	121.2
3	100	100	50	50	107.8
4	50	100	50	100	118.2
5	100	50	100	50	104.3
6	50	50	50	50	103.0
7	100	100	100	100	111.1
8	50	100	100	50	104.3

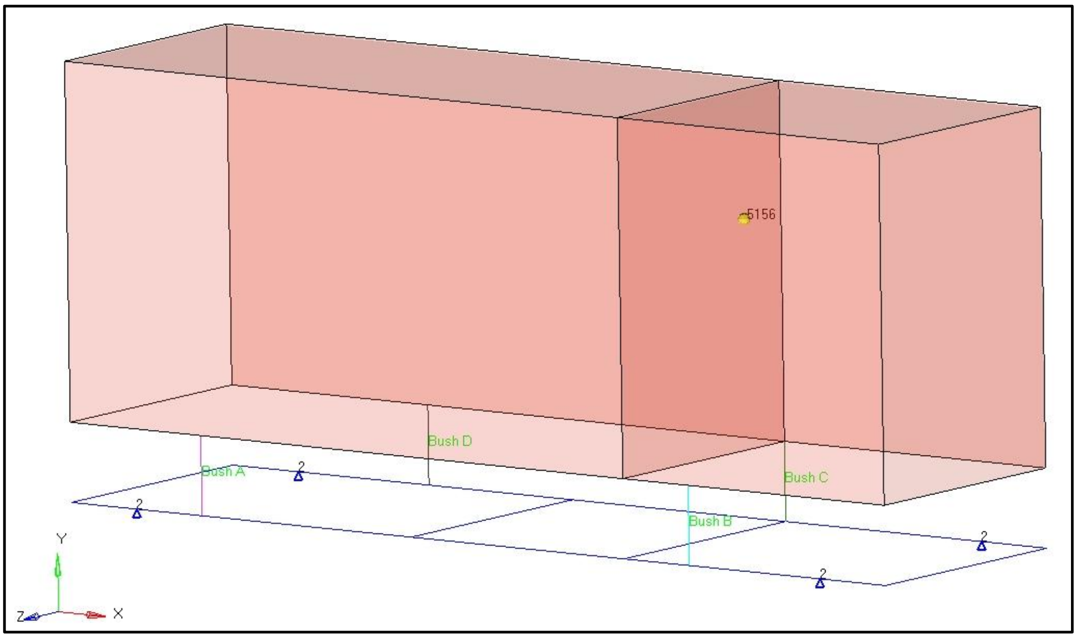

Fig. 13:Location of all four bushes on primary and secondary structure and response node location

The above table shows the sound pressure level for each combination of bushes. This is observed from the graphs as in Figure 5 to Figure 12. From these results, contribution of each bush is calculated. The DOE for finding the paretochart is performed in Minitab Software. The location of all the buses is shown in Figure 13. The following screenshot shows the calculations in Minitab software.

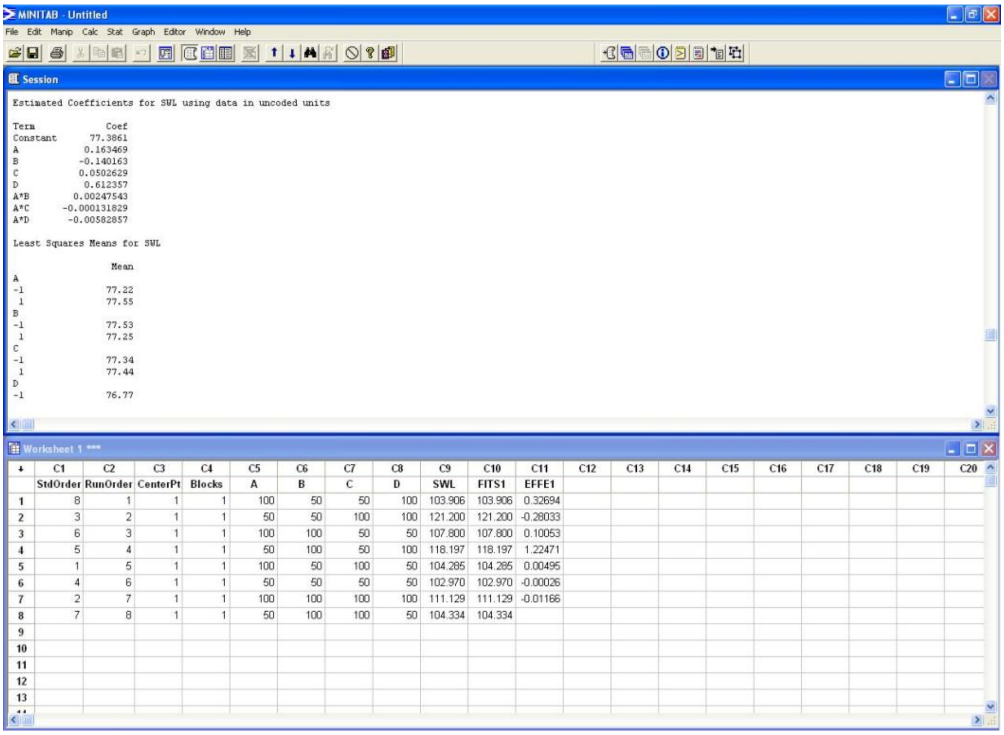

Fig.14: Screenshot of Minitab Software for performing DOE

The pareto chart of the bushes contributing to sound pressure is seen in Figure 15. Maximum sound transfer from the primary structure to the point inside the cavity is through Bush D, followed by Bush A. So, the critical noise transfer path is through bush D.

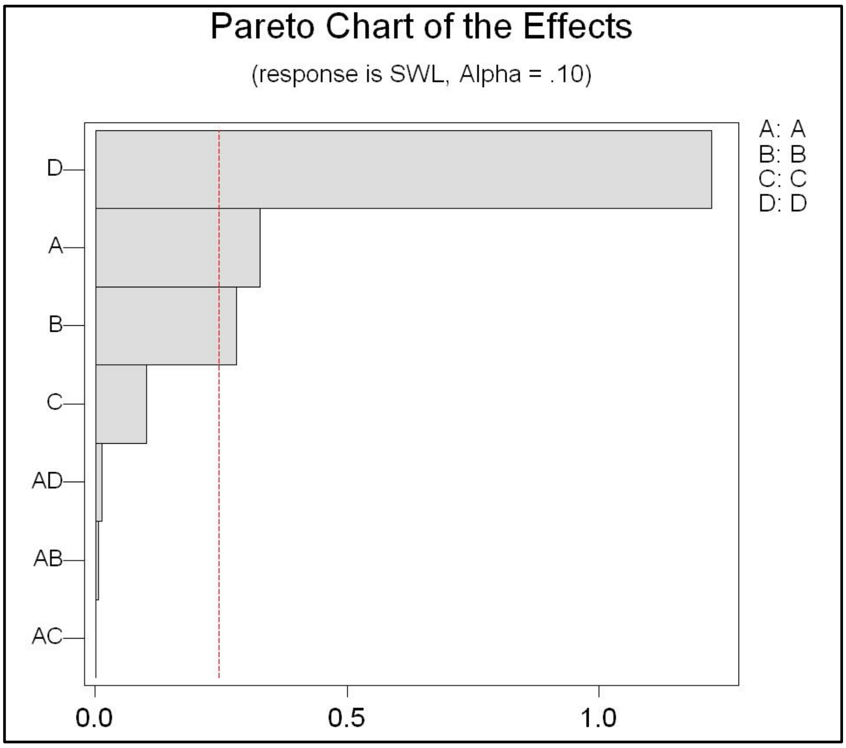

Fig. 15: Pareto chart for contribution of each bush to sound pressure level

The stiffness of Bush D is modified from 100 N/mm to 50 N/mm for Case 1 combination. The sound pressure level is reduced from 103.9 dB to 95.3 dB. The following graph shows the reduction in sound pressure level. Hence the sound pressure level is decreased by 8.6 dB as the stiffness of most contributing bush is reduced.

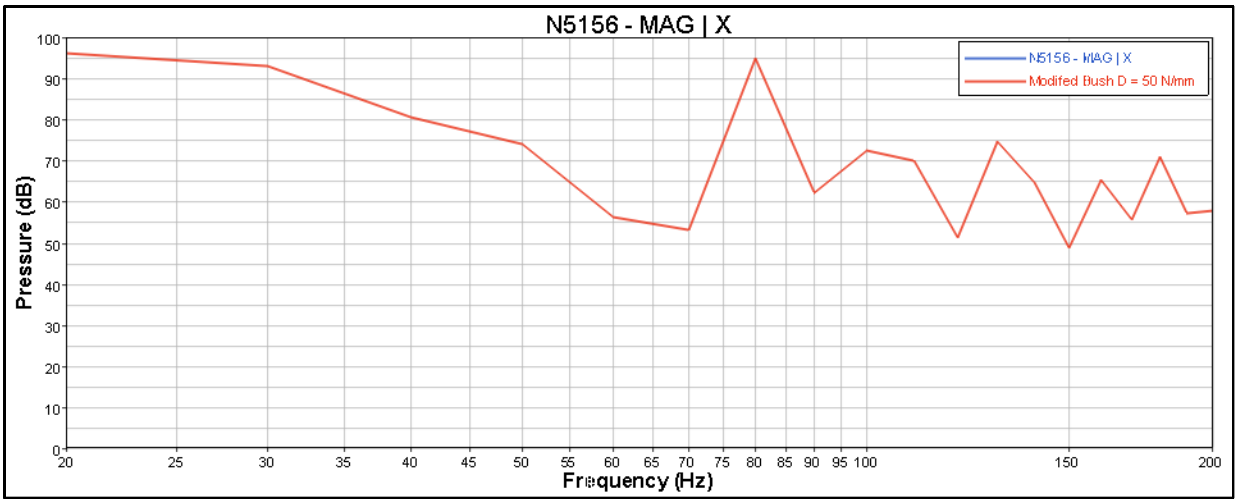

Fig.16: Graph of sound pressure vs frequency for Modified bush D stiffness

VII. CONCLUSION

Transfer path analysis has been performed for selected vibro-acoustic responses. In Transfer path analysis, the design of experiment approach has been considered to identify critical noise transfer path. For DOE, two levels (Bushing stiffness variation) and four factors (Bushes A, B, C & D) are considered for the analysis.

From DOE approach, it has been found that major noise is transferred from bush D to the receiver location inside the acoustic cavity. By refining the bush stiffness value of bush D, the noise is reduced by 8.6 dB.Hence, the noise is seen to be decreasing as the stiffness of this bush is reduced.

From this analysis, it is possible to calculate the partial contributions of all bushes affecting the sound pressure level. This is helpful for using TPA as a tool to understand the effects on NVH and ride/handling by specifying different bush stiffness combinations.

REFERENCES

[1]. Farshid Haste and AnbuNachimuthu, “Calculating Partial Contribution Using Component Sensitivity Values: A Different Approach to Transfer Path Analysis”, SAE paper no. 1999-01-1693.

[2]. LMS International, “Transfer Path Analysis: The Qualification and Quantification of Vibro-acoustic Transfer Paths”, www.lmsintl.com, 1997, pp. 1-20.

[3]. Nitin S. Gokhale, Sanjay S. Deshpande, Sanjeev V. Bedekar, Anand N. Thite, “Practical Finite Element Analysis”, 1st Edition, Finite to Infinite Publication, Pune, 2008, pp. 219, 352.

[4]. Designing an Experiment, Minitab Software Manual