Optimization of Exhaust System Hangers for Reduction of Vehicle Cabin Vibrations

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Abstract

Excitations from the vehicle engine and the road surface cause vibrations in the exhaust system and the exhaust noise and vibrations are transmitted through the vehcile body and structure to the cabin, causing distractions and discomfort for the driver and passengers. In this article the method of average driving degrees of freedom displacement (ADDOFD) has been used to determine and optimize the location of suspended hanger points. Based on this approach, a model of car exhaust system is used using ANSYS software to optimize the hanger installation points for reducing vibration and to select the best positions for these points. The optimum hanger positions must have a relatively lower ADDOFD value compared to adjacent points. Then the static and dynamic analysis of the exhaust system is illustrated and finally on the basis of the above analyses, the position is chosen for the exhaust system hangers to reduce the transmission of noise and vibrations into the car cabin. Results indicate that optimization of the locations has resulted in a significant decrease in hanger loads, significantly reducing the vibrations transmitted to the vehicle cabin and increasing the life of the rubber hangers. This study has practical significance for reducing the vibration of automobile exhaust systems and the vehicle cabin.

Keywords: Exhaust system hangers, Exhaust tuning, Cabin vibration, Finite element method, ADDOFD method

1. Introduction

Vehicle exhaust system consists of the pipings, the fundamental purpose of which is to convey engine exhaust gases into the atmosphere and prevent them from entering the passenger compartment [1], along with assemblies and compartments designed to partly treat the exhaust gases and reduce the noise and vibration caused by the gases. The exhaust system extends from the exhaust manifold down to the tailpipe, as shown in Figure 1, consisting of components such as catalyst chamber, resonator, muffler, and piping material.

The exhaust manifold is connected directly to the engine and therefore, vibrations from the engine are transmitted to exhaust system which can in turn transmit the vibrations to the cabin structure through connecting hangers. It is important to understand the dynamic characteristics of the exhaust system in order to reduce the vibration transmitted from the engine to the cabin, through the exhaust system [2]. Figure 2 summarizes the most important design attributes of the exhaust system. These attributes can point out the significance and importance of the design of the exhaust system and its influence of vehicle performance and customer satisfaction.

Many researchers have studied various solutions in order to minimize emissions [3, 4]. However, the study of the exhuast system itself, can lead to significant improvements in the sense of vibrations and noise. The exhaust system can not be fixed directly to the vehicle body because it will lead to the transfer of vibrations to the vehicle cabin. Therefore, the exhaust system has to be connected through flexible hangers. One of the parameters that can be effective in reducing the noise and vibration of the exhaust system are the selection of the locations of hanger points. The layout of hanger points along the

length of the exhaust pipe can be crucial for increasing the life of the rubber hanger, determining the natural frequency of the exhaust system and influencing the forces transferred to the cabin from the exhaust system [5]. In this paper, we consider the optimized exhaust system hanger locations which are developed through the method of average driving degrees of freedom displacement (ADDOFD). Then with finite element method (FEM) static and modal analyses, the harmonic responses for the exhaust system can be extracted.

In order to reduce the impact of automotive exhaust systems on the vehicle vibration, finite element modal analyses of the exhaust system of a motor vehicle are conducted in its natural state, partially constraints state and fully constraints state. The natural frequencies and mode shapes of the exhaust system are obtained. The harmonic analysis is conducted on the basis of the modal analysis. The results has shown that the vibration amplitude of the exhaust system at low speed operating condition is relatively bigger than that of the exhaust system in high-speed conditions. The excitation frequency of the engine at the idle and economy speed can avoid the natural frequency of the exhaust system, so it can be stated that the system has good dynamic characteristics. Suspension points on the exhaust system are optimized and the optimal mounting points are located based on the method of the average drive DOF displacement (ADDOFD) [6]. The study has practical significance for reducing the vibration of automobile exhaust systems and the vehicle cabin.

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 [7].

To improve the performance of one exhaust system, in this article, an exhaust system in the market was conducted by the stress analysis and optimization of the hanger. First of all, the 3D model was simulated by CATIA and a FEM model was simulated by Hypermesh, and then the calculation of the free modes were achieved by ANSYS. Then, with the method of component mode synthesis and then through the analysis by the software Nastran, the best hanger position has found. Finally, the optimized system was conducted the restrained modes analysis, and the reaction force of hanger was calculated as well, by verification and comparison, the modal mode of vibration is more appropriate, and the peak of the force has decreased, in the low frequency section, the curve is more smooth, the optimization was proved to be effective to improve the system's NVH performance [8].

To improve the vehicle NVH performance and reduce the vibration of the exhaust system, average driving DOF displacement (ADDOFD) [9] and dynamic analysis are used to optimize hanger locations. Based on the finite element model and rigid-flexible coupling model, exhaust system analysis model was established. Using the finite element method, the exhaust system is supposed to release beam, and the position of the hanging point of the exhaust system is optimized by using the ADDOFD method. Furthermore, through the dynamics analysis, the force of each hanger to the body is calculated by the dynamic analysis, then verified the rationality of the hanging position. The combination of the two methods can effectively determine the better NVH performance of the exhaust system with hanger locations in the earlier vehicle development process [10].

2. Modeling and Methodology

According to the modal analysis theory of multi-degree system, the response function between the response point l and point p can be described by:

$$H_{lp}(\omega) = \sum_{r=1}^{N} \frac{\varphi_{lr}\varphi_{pr}}{M_r(\omega_r^2 - \omega^2 + j2\zeta_r\omega_r\omega)} \qquad (1)$$

where φlp is the pth mode shape system of the lth measuring point; Mr is the modal mass; ξr is damping ratio of the rth order modal. If the frequency of the exciting force is ωr, Equation (1) can be further approximated as

$$H_{lp}(\omega) \approx \frac{\varphi_{lr}\varphi_{pr}}{jM_r2\zeta_r\omega_r^2\omega} \qquad (3)$$
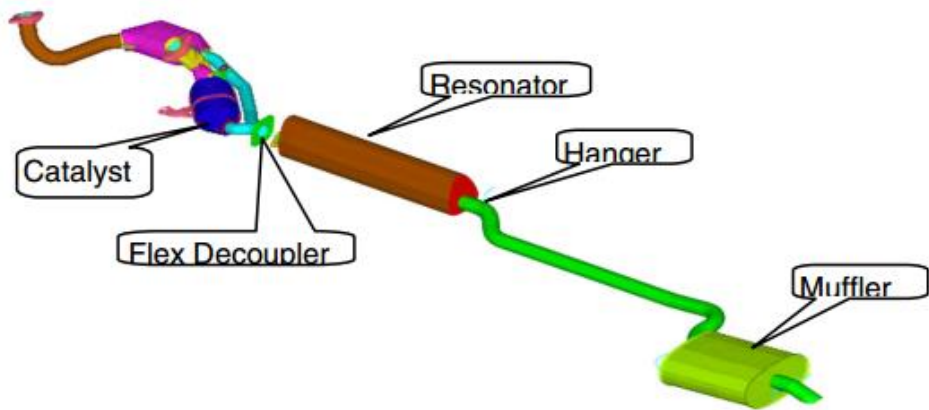
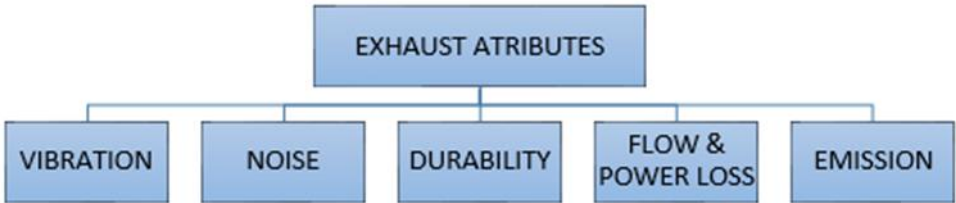**Fig1.**: Schematic representation of an exhaust system

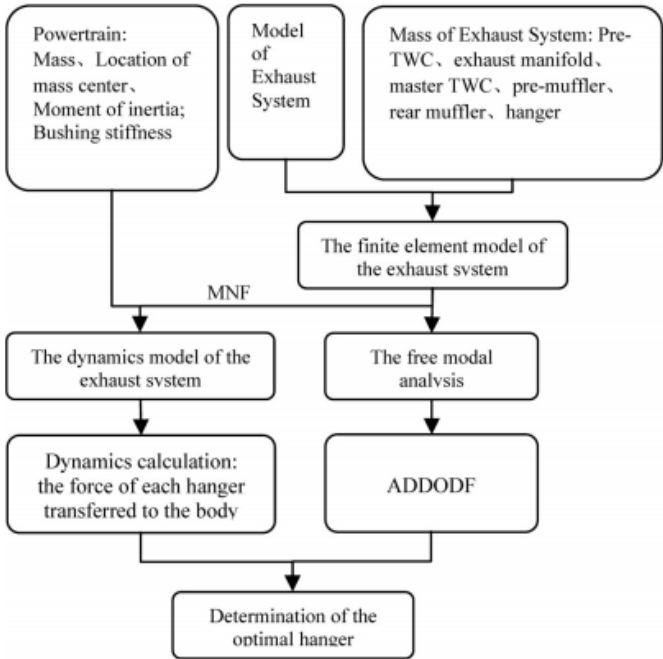**Fig2.** The main design attributes of the exhaust system

Fig3.: Analysis and optimization methodology [9]

For linear system, the amplitude of the displacement response is proportional to the amplitude of the frequency response function, namely:

$$X(\omega_r) \propto \frac{\varphi_{lr}\varphi_{pr}}{\omega_r{}^2} \qquad (3)$$

The value of the jth degree of freedom can be depicted with the following equation

$$ADDOFD\,(\,j\,) = \sum_{r=1}^{N} \frac{\varphi_{jr}^2}{\omega_r{}^2} \qquad (4)$$

Lower values of ADDODF represent lower displacement responses, and the position of the nodes with lowest ADDODF values is selected as the hanger position.

The analysis and optimization procedure is conducted according to Hailan and Yan [9]. In order to perform the analysis, first, a FEM model of the exhaust system is built. Then, FEM model of the rigid-flexible coupling of exhaust maifold and powertrain is constructed. This allows for the procedure of determining the optimal hanger locations based on ADDODF values to take place. The procedure has been summarized in Figure 3.

Optimization approaches incorporating finite element models allow for an efficient evaluation of designs using mathematical models [11]. Using the exhaust system CAD model, a finite element model was constructed using ANSYS software, as shown in Figure 4. The boundary conditions of the FEM model consist of a fixed boundary on the end attached to the manifold, and a free end at the other side. All of the hanger isolators are modeled by springs consisting of three spring elements (in local x, y and z directions) for each isolator. Each spring is modeled with one end connected to a hanger and the other end connected to the vehicle body, simulating an isolator bridging between a hanger and the chassis. A convergence test was performed on the mesh quality of the FE model by adjusting the relevance parameters for meshing [12]. The finite element model of the exhaust system comprises of two types of elements SOLID185 and SHELL91. Finally, the converged model has 20647 elements (mostly shell elements) and 21756 nodes. The material used for the FE model is structural steel with Young's modulus E =210GPa, Poisson's ratio ν =0.3 and density ρ =7800 kg/m.

Modal analysis

Modal analysis is performed on the exhaust system with clamped-free boundary conditions. According to previous studies, based on the excitation of the exhaust system, the most important modes which induce the vibration and need to be considered are between 20Hz and 200Hz [13]. Modal analysis is performed using ANSYS finite element software for the following system to identify the frequencies and mode shapes of the system. The first ten natural frequencies of the exhaust system are listed in Table 1. The first ten mode shapes of the automotive exhaust system in the free-constrained condition are also shown in Figure 5.

As shown in Figure 6, a total of 66 nodes are selected along the exhaust pipe and modal analysis is performed on the exhaust system to obtain the frequencies below 200Hz. The ADDOFD values of the selected nodes are calculated and shown in Figure 6.

Considering the actual structure of the exhaust system, under-body and chassis assembly space, the optimal hanger location is shown in Figure 6. After the hanger locations have been confirmed, the modal analysis of the exhaust system excited by the powertrain and road vibrations will be studied.

3. Static analysis

Next, static analysis of the exhaust system is performed using ANSYS. Again, the side of the exhaust system connected to the engine manifold is fixed, and the possible locations of the hangers are assumed to be simply-supported. Then, the whole finite element model of the system is loaded within a gravitational field. After several iterations, the locations of four hanger are selected based on the values of the loads exerted on the hangers and also the uniformity of load distribution [2]. These four hanger locations are shown in Figure 7, along with the maximum vertical force exerted on each hanger.

Figure 8 depicts the maximum displacement at the nodes of the exhaust system. According to design guidelines, the transmitted force must be in the range of less than 50 N and its displacement must be less than 5 mm [14].
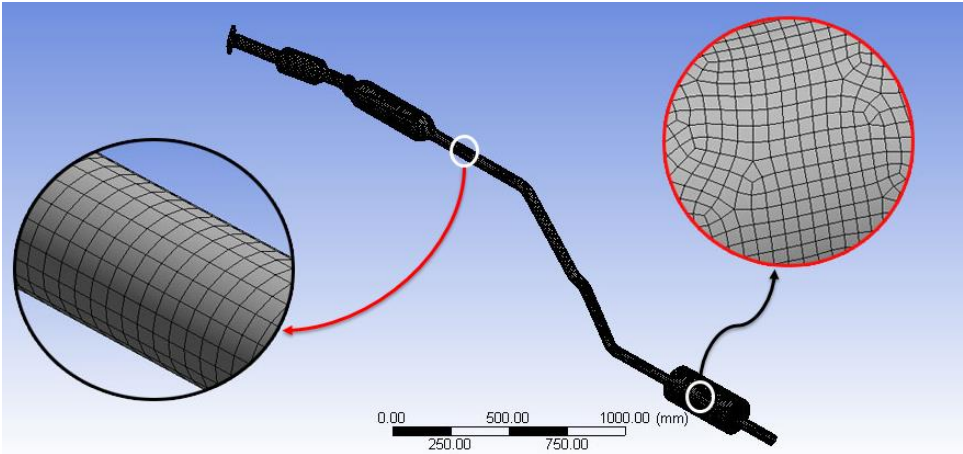
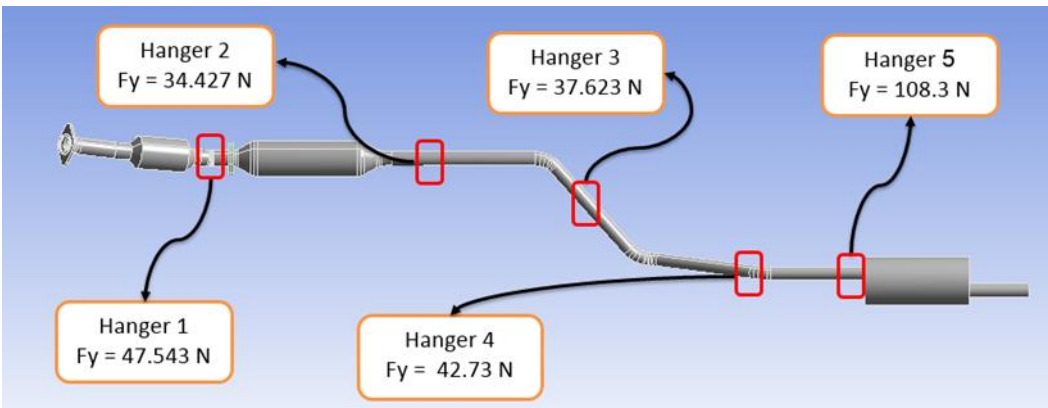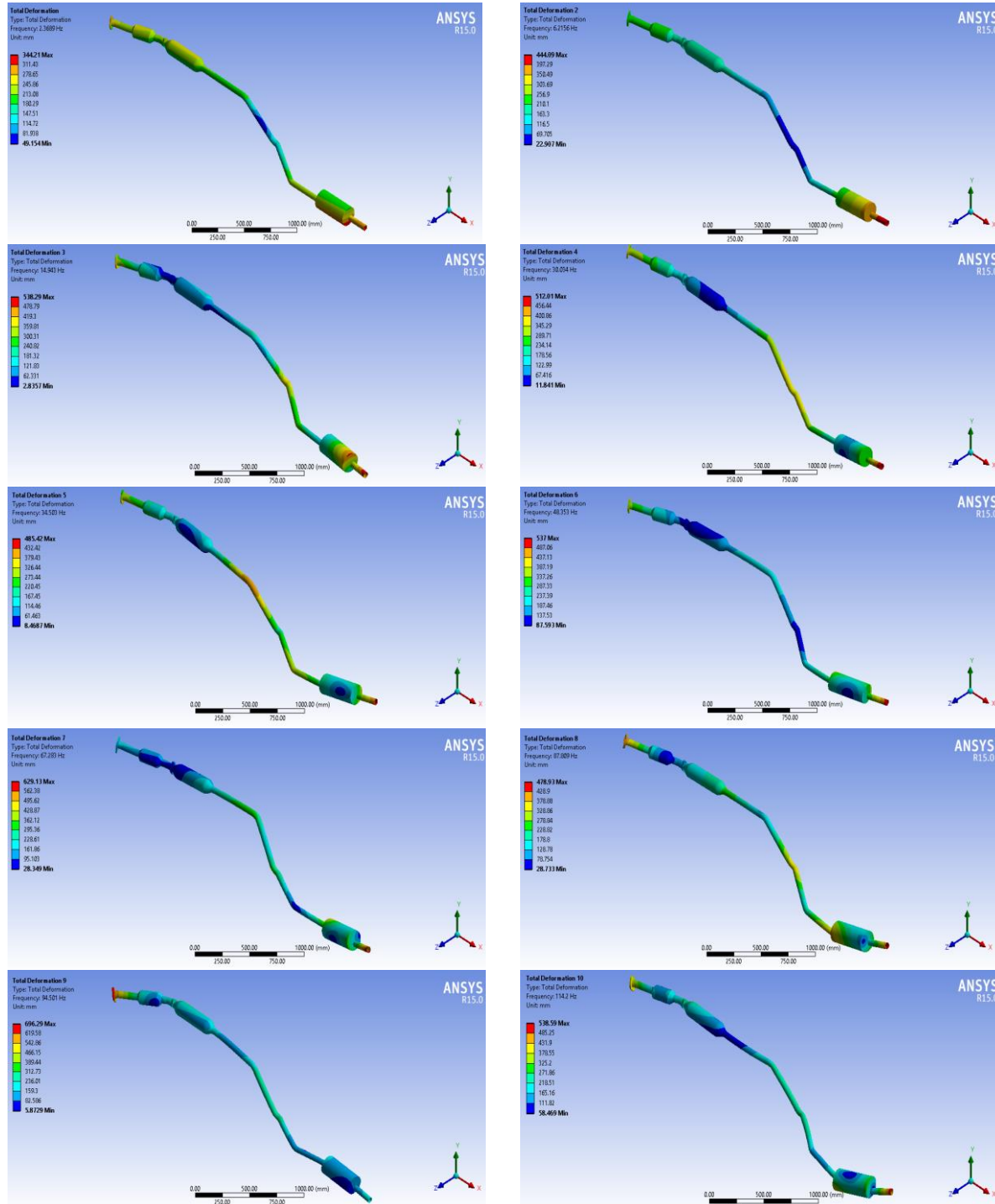|	Frequency [Hz]
1	2.37
2	6.22
3	14.94
4	30.03
5	34.50
6	48.35
7	67.28
8	87.81
9	94.50
10	114.20

Figure 7: Hanger locations and load value

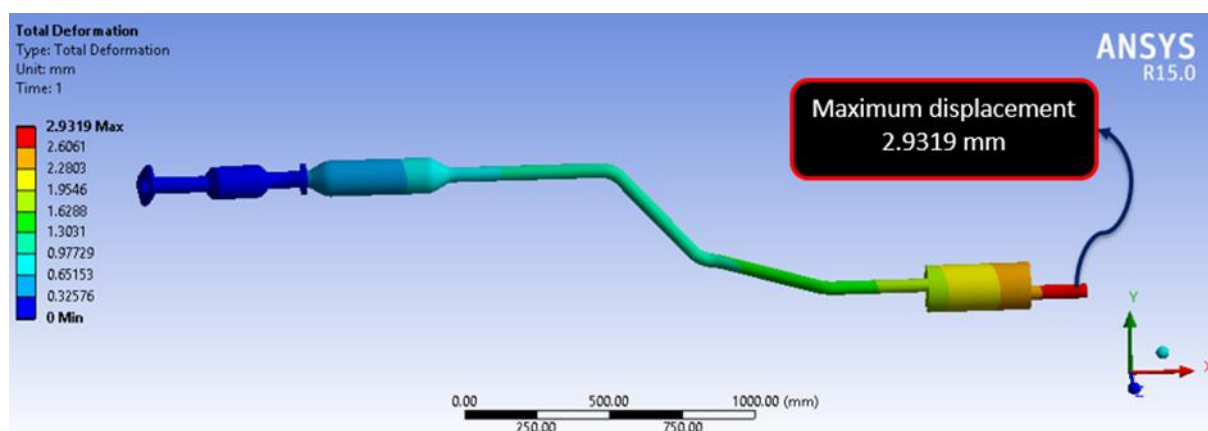**Fig5..** The first ten mode shapes of the exhaust system
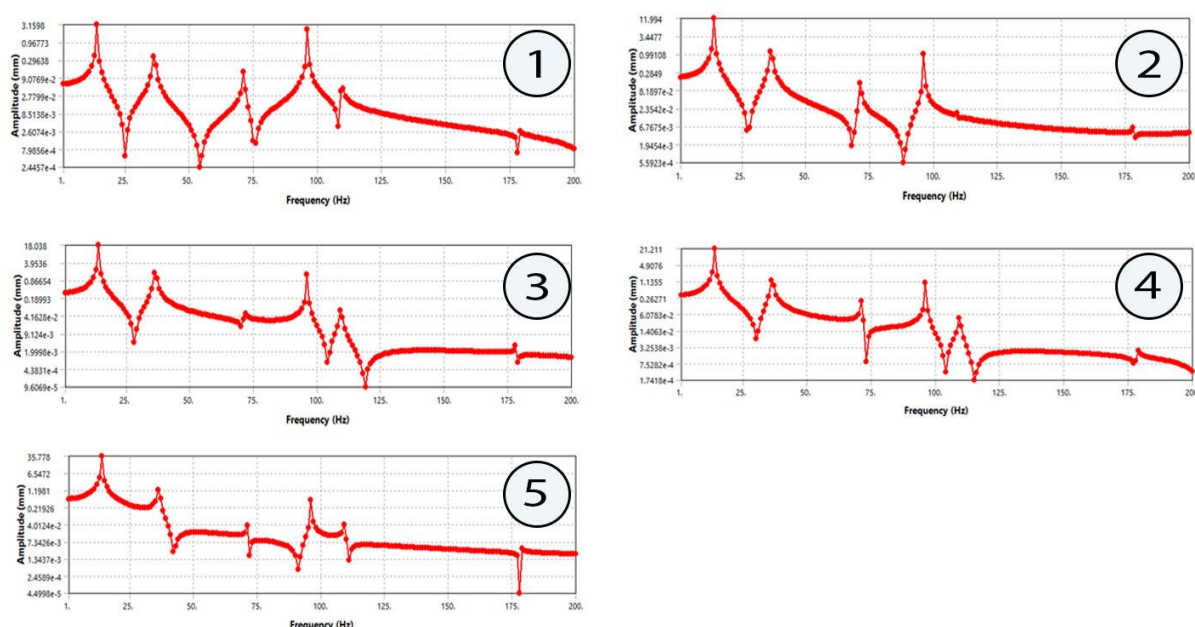
Figure 8: Maximum displacement of the exhaust system

Figure 9: Characteristic displacement-frequency response for suspension point 1 to 5

4. Dynamic analysis

Harmonic analyses are conducted to study the response of the exhaust structure to harmonic excitations woth different frequencies. The response is plotted in the form of displacement-frequency curve. The source of excitation of the exhaust system consists of structural vibrations of the engine along with random road vibration caused by road surface roughness. In this paper, vibration of the exhaust system subject to excitations from the engine and road is considered. Results of the frequency response analysis of suspension point 1 to 5 are shown in Figure 9, which shows that the vibration amplitudes of the exhaust system suspension point becomes smaller with increasing excitation frequencies.

It is worth noting that low frequency band, below 140Hz, is mainly considered when reducing the noise and vibration of the exhaust system. The dynamic analysis can be evaluated using the actual force and displacement of the body at the position of the hangers. The hanger locations should be selected such that the maximum force generated at the hanger is

less than 10 N [15]. During the dynamic analysis simulation, a certain amount of acceleration is applied to the mass. The resultant transmitted force has been plotted versus the frequency in the range of 10 to 100 Hz, as shown in Figure 10. As shown in Figure 10, the translational forces of hangers 4 and 5 have exeeded the maximum permissible value and their locations should be optimized

5. Optimization

In this section we investigate the optimal layout position of the hanger points. In the previous section, we had determined the natural frequencies and mode shapes using modal analysis. Nodal displacements and force were then obtained using harmonic analysis, under excitation from the engine and road. Since at a number of hangers, the standard requirement for the hanger forces are not satisfied, these points should be moved to optmized points in order to obtain the best position for the suspension points which reduce the vibrations. For this purpose, the following objective function equation is defined [15] and the problem is optimized using MATLAB Optimization Toolbox.

$$objective: Min \sum_{i=1}^{N} \left\{ \sqrt{\left(\sum_{j=1}^{3} x_j^2\right)} + \sqrt{\left(\sum_{j=4}^{6} x_j^2\right)} \right\}$$

where N=5 is the number of hangers, and x_j is the displacement at the jth degree of freedom.

constraints :
$$\mathbf{g_k(\vec{X})} \leq \mathbf{15} \ (\mathbf{mm}) , \mathbf{0.2} \ (\mathbf{rad})$$
K; number of frequency
$\vec{X} = \{x_1, \dots, x_{18}\}$; displacement at hanger position

$$\mathbf{g_k(\vec{X})} \leq \mathbf{1.0 \times 10^9} \ (\mathbf{N/m^2})$$

$\vec{X} = \{x_1, \dots, x_4\}$; total stress (bending and axial)

$$\mathbf{g_k(\vec{X})} \leq \frac{\mathbf{F_{axi}}}{\mathbf{F_{cr}}} - \mathbf{1.0} \leq \mathbf{0.0}$$

F_{axi} ; axial force
F_{cr} ; critical weight
subject to : $\mathbf{X_L \leq X_i \leq X_M}$
X; position of bellows
X_L , X_U; Min. and Max. of design value
$\mathbf{L_L \leq L \leq L_U}$
L; length of bellows
L_L, L_U; Min. and Max. of design value

After the hanger positions are achieved and optimizations of transmitted forces are performed, the obtained results are shown in Figure 12.
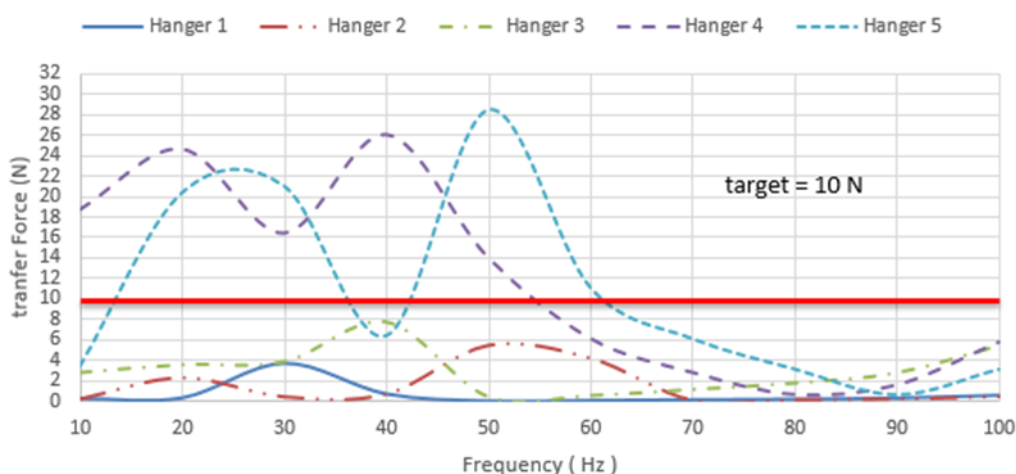

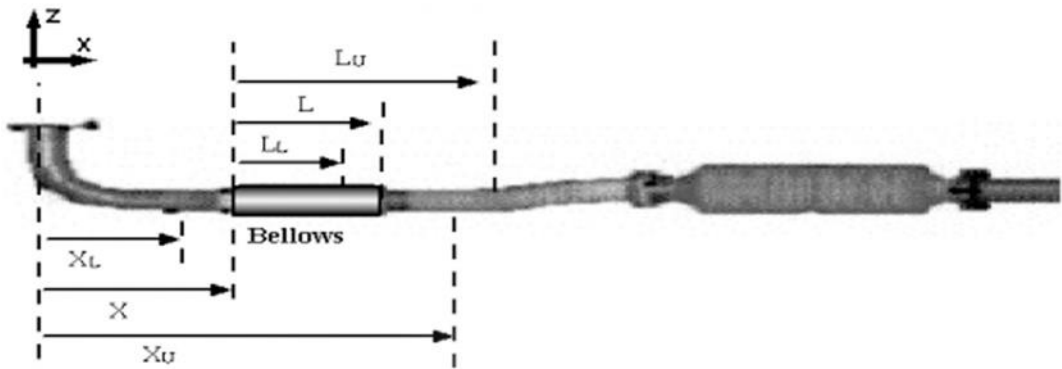**Figure 10:** Transmitted forces at the hanger location

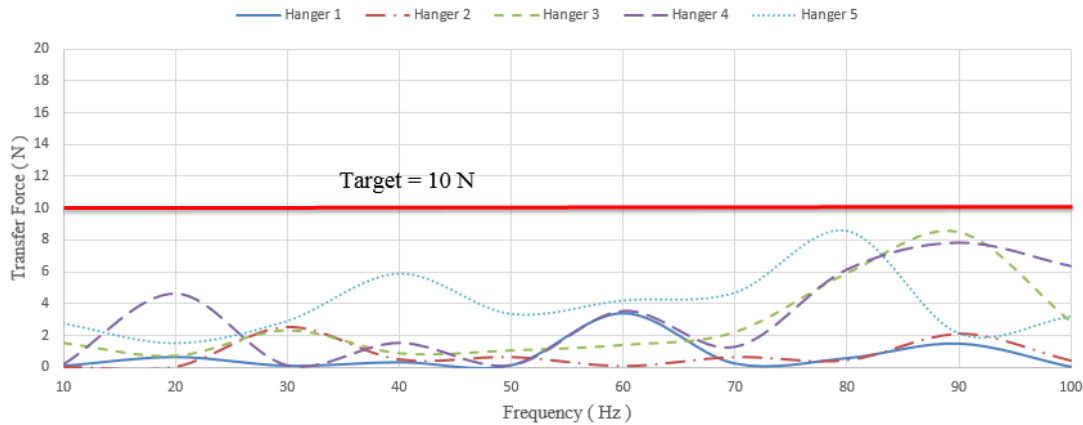**Figure 11:** Optimization of position and length of Bellows

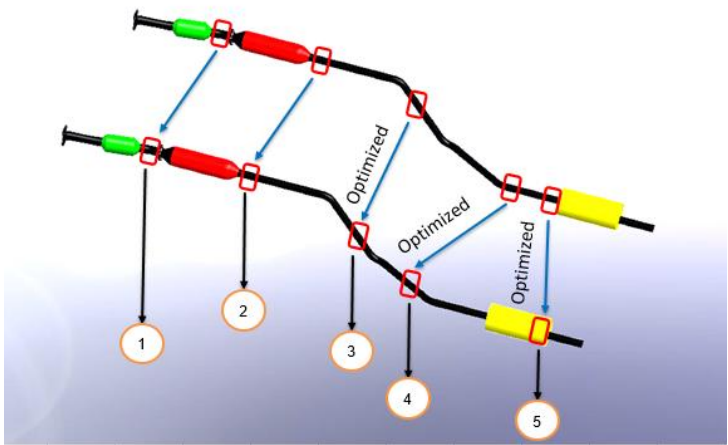**Figure 12:** Optimization of hanger positions

**Figure 13:** The position of the optimized suspension points

As shown in Figure 12, it can be noted that the maximum transitional forces in a state of engine excitation for the range of frequencies up to 100 Hz are less than 10 N which is in accordance with design requirements. Results of the optimization of exhaust system hanger locations have been depicted in Figure 13. It is visible that after optimization, the locations of points 3, 4 and 5 have changed which has resulted in a significant decrease in hanger loads, increasing the life of the rubber hangers and significantly reducing the vibrations transmitted to the vehicle cabin

Moreover, Figure 14.shows the displacements of the exhaust system hangers in the range frequency range of 0 to 200 Hz. This plot shows that the maximum displacement requirement of less than 15 mm has been achieved and the design of the exhaust system has been optimized to be in accordance with design requirements, by changing the hanger positions.

The values of forces and displacements in the hangers in their initial locations are summarized in Table 2, along with the results obtained from optimizing the exhaust system hanger positions.

Finally, a static analysis of the optimized exhaust system is illustrated in Figure 15, which shows that the maximum displacement is 2.07 mm, which is less than the design limit of 5 mm.
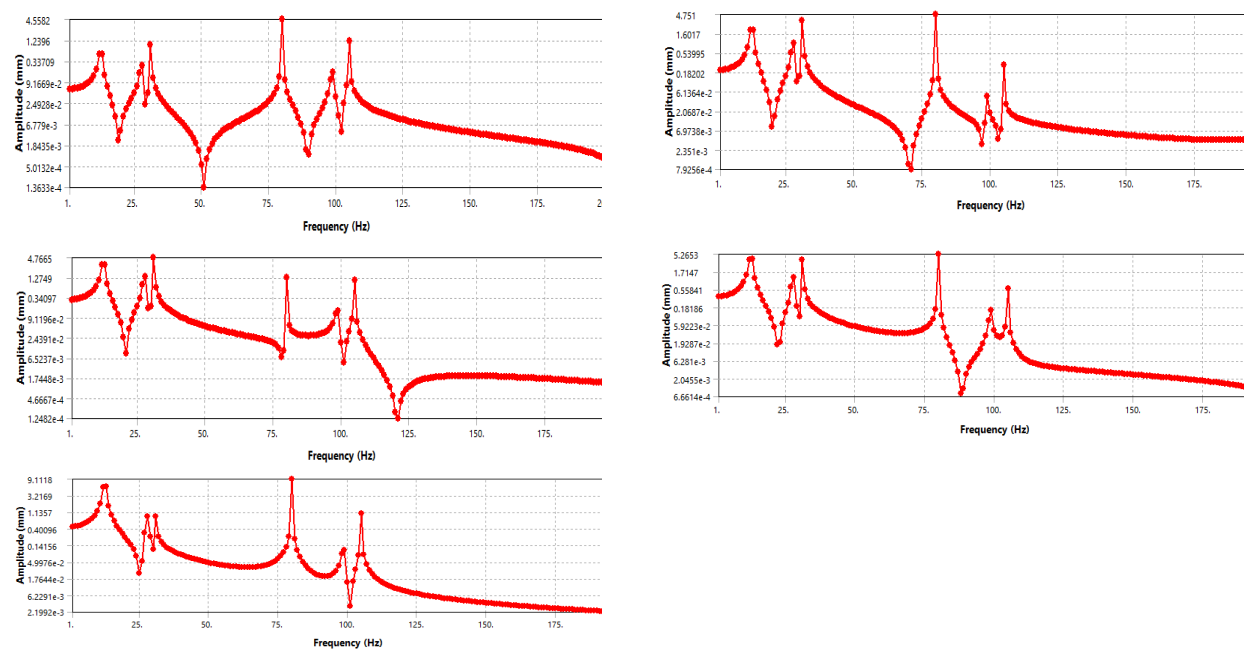
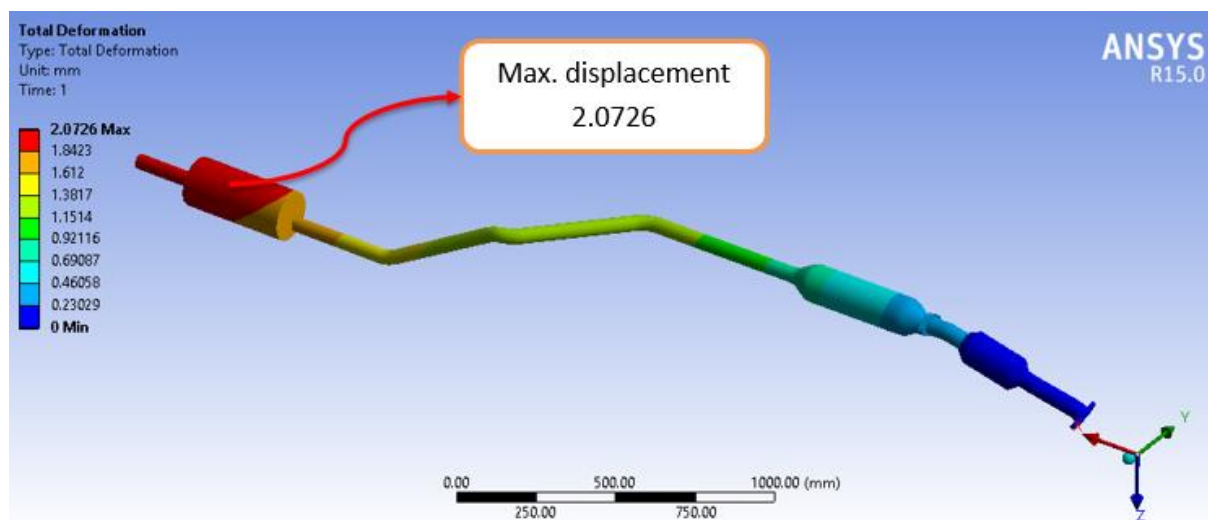|			Optimized	
	Force (N)	Displacement (mm)	Force (N)	Displacement (mm)
1	3.78	3.16	1.50	4.56
2	5.45	11.99	0.81	4.75
3	7.73	18.04	8.50	4.77
4	26.02	21.21	7.80	5.27
5	28.45	35.78	7.94	9.11

Figure 15: Static analysis of exhaust system, illustrating the maximum displacement

CONCLUSION

In this paper, simulation of exhaust system using finite element model is performed to optimize the locations of the hangers. The ADDOFD method is employed on the FEM model of the exhaust system to determine the hanger locations of the vehicle exhaust system. The value of the hanger force must be in accordance with the standard requirements of less than 50 N. Dynamic analyses are then performed to calclate the transmitted forces in a state of engine and road excitation forces, which shoulb be less than 10 N. Initial simulations revealed that the forces exceed the standard limits at a number of hanger locations and an optimization problem is defined with a suitable objective function to improve the problem. The results of this paper showed that the use of ADDOFD method and finite element modal analysis with the help of static, harmonic, and dynamic analyses can reduce the maximum transitional forces in the state of engine and road vibration excitation to less than 10 N to comply with standard requirements, in the frequency range of up to 200 Hz. This study can be of practical significance for reducing the vibration of automobile exhaust systems and the vehicle cabin.

6. References

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