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A System Approach to Dynamic Characteristics of Hanger Rod in Exhaust System

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Abstract

The first and foremost important prerequisite that a designer require to design a part is stiffness information. The dynamic characteristics of automotives exhaust hanger rods are studied for high frequencies at which the resonance vibration occurs, which is also an important transfer path for vibration to exhaust system. The point is to elaborate an approach which provides guidance towards optimized designs, mainly in early development stages. Theoretical and experimental modal analyses are used to suggest design parameters in terms of natural frequencies, which should be above 150 – 200Hz from the engine excitation frequency. Case studies are presented to show the methodology and validation of full system hanger rods for stiffness applications. The results acquired in this case study will highlight the potential applications of this approach, as well as the challenges associated with this method.

Keywords: *Hanger rod, Modal analysis, Design parameters, Experimental validation.*

1. Introduction

Effective and efficient product development is critical to corporate success on the increasingly competitive global market and simulation has proven to support this in many sectors. This translates in design 'first-time-right' philosophy, where the use of advanced numerical and experimental methods that account for the product environment is essential.

"What if" studies to understand the effects of changing geometrical parameters or to change a design parameter to avert failure and improve the product design is essentially known as design optimization. Design optimization provides a robust and systematic methodology by carefully studying the effects of various design variables and improves the design by varying the variables. Along with the load - deflection characteristics to understand the stiffness, the deformed shape of the component and the resulting stress-strain distribution can also be predicted.

Noise and vibrations are indiscernible to the occupants of the car. The main source of vibration related to exhaust system is engine. There are mainly two transfer paths namely; structure and airborne vibrations. The structure borne paths starts from the engine and transmission line mounts and it transfers to the exhaust system through the hanger rod. Therefore, it is of great importance that the hangers are designed so that their natural frequencies are higher than the frequencies of the exiting sources acting on the system.

Several papers dealing with the dynamic behavior of the whole exhaust system have been found [2-4], but no treatise of the hanger's dynamic behavior has been found. The purpose of this paper is to analyze the design parameters of hangers that have affect on natural frequencies. Theoretical and experimental modal analysis will be performed for some existing.

2.Limitations

The main restraint is that all natural frequencies of the hangers must depend on vehicle applications. The excitation sources are assumed to vibrate the exhaust system up to frequency range from 100-250Hz for passenger car vehicles. The vendors demand includes a safety margin of 150-200Hz, which should be able to withstand the loads such as mechanical loads, thermal loads and corrosion characteristics.

Design parameters are influenced by mass and geometry. The materials used in all case studies are stainless steel grades (SS409.S10C). Stiffness depends on material mechanical properties, geometrical design and connections between hanger and the exhaust system, i.e. weld. The geometrical limitation of hanger rod, i.e. it should be able to fit the standard rubber isolators and should also meet the packaging conditions of the system. As a secondary solution, influencing the damping ratio of the isolator in the structure is possible to decrease the effects of natural frequencies that cannot be increased above the desired level. So, damping effects of rubber isolator will not be discussed in this paper.

3.Modal analysis theory

Modal analysis provides details about mode shapes, natural frequencies and damping ratios for the investigated structure. The analyses can be performed both as theoretical calculations on a FE-model and experimental tests on the real structures [5-7]. The damping ratios can only be determined experimentally. Theory is common for both theoretical and experimental modal analyses are described briefly below.

3.1.Theoretical modal analysis

- The basic equation for typical un-damped modal analysis is classic Eigen value problem.
- According to mode theory, the structure will be typically seen as a system constituted by the mass point, rigid body, damper and discrete it as finite number of elastic coupling rigid bodies.
- Therefore, an infinite multi-degree of freedom system turns into limited multi-degrees of freedom system.
- When the linear time-invariant system requirements are met, the system's general motion mathematical model can be expressed as:

$$M\ddot{x} + C\dot{x} + Kx = f(t) \qquad (1)$$

Where,

M, C, K: The mass matrix, damping matrix and stiffness matrix

x: The exhaust pipe vibration displacement vector

f (t) : The exhaust pipe load vector

- Modal analysis method is to replace the physical coordinates of modal coordinates that each principal mode corresponded, so that the differential equation decoupling to be independent differential equations in order to obtain the system modal parameters.
- The vibration of the engine exhaust pipe is a random vibration, which basically belongs to linear time-invariant systems. It can be assumed that M is a constant matrix. The structural damping of exhaust pipe has little effect on the natural frequencies and therefore external load and damping are not considered.
- Thus equation shown above becomes :

$$K- \omega^2 M\Phi = 0 \qquad (2)$$

Where,

M- $\int_\Omega \rho N^T N d\Omega$ is the structure overall quality matrix

- When the order of matrix K and M is n, the ω^2 in formula shown above is the n times real coefficient equation and the system degree of freedom vibration characteristics (natural frequencies and mode shapes) problem is to solve the matrix eigen value ω.

3.2.Experimental modal analysis

The modal properties are estimated from the frequency response functions (FRFs) obtained from the test data. In the FRF, a peak of the magnitude marks every resonance frequency. Each resonance frequency can be associated with a certain mode shape that represents the deflection shape of the structure [8]. There are several methods available for estimation of the mode shapes, both single and multiple degree of freedom methods. The estimation techniques, also called curve-fitting methods, are used to generate an analytical function that approximates the measured FRFs.

Inertance is the ratio of acceleration like quantity to a force like quantity, when the arguments of the real or imaginary parts of quantities increase linearly with time (dB reference $1ms^{-2}/N$).

Receptance is defined as displacement per unit harmonic force. (dB reference 1m/N)

Average displacement value of receptance plot in the range of 200- 600Hz should be considered.

3.3.Correlation

Correlation is a process where data from the experiment are compared with theoretical results. There are several methods available, which are more or less complicated. Two graphical methods are “Graphical comparison of natural frequencies” and “Graphical comparison of mode shapes”. They are easy to use, but they are very time consuming for models with many nodes. Two numerical methods for comparison of mode shapes are the “Modal scale factor” and the “Modal assurance criterion”.

The methods are briefly explained below. For a more complete discussion the reader is referred to for example Maia [9].

4.Methodology

This paper includes full exhaust system hanger rods as subject, which was pre-processed using Hypermesh software, which was developed with the use of shell, solid, bush and rigid elements. Exhaust pipes, hanger rods, sub-resonator, main resonator and tail pipe are modeled with 4 noded shell elements. Inlet flange modeled with 8 noded Hexa elements. Flexible bellows and rubber isolators are modeled with bush elements and necessary stiffness values are assumed as per available references. Schematic representation of the system is shown below in the Figure 3. The natural frequency, static response and frequency response stress are determined from Normal modes, Static 1G and Durability analysis. These are performed using Msc Nastran and the results are posted using Hyperview. The results are compared and correlated with experimental data using NVH and Fatigue lab.

4.1.Analysis simulated for structural validity

- Stiffness analysis - To determine the natural stiffness of the hanger rods.
- Static 1G loading - System stability, hanger forces and hanger displacements.
- Resonance frequency analysis - Determining the mode shape behaviour of the system and checking for resonance.
- Engine Roll $\pm$ 4° analyses - To check the bellows relative displacement, bending moment at the inlet of the main resonator, inlet and outlet of the sub-resonator.

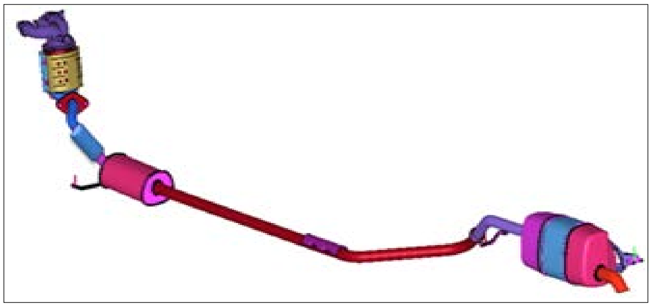

Figure 1. System overview

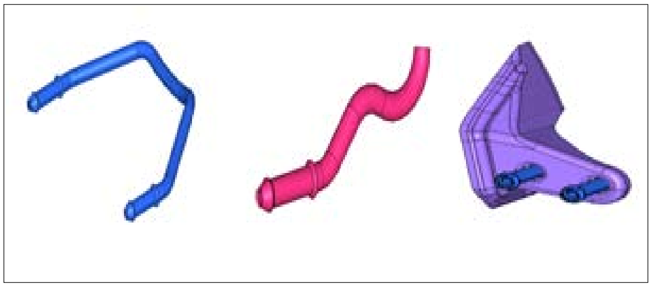

Figure 2. Hanger 1, 2 and 3

4.2.Input details

- No of Cylinders = 3
- Engine Rated Speed = 6000 rpm
- Natural Frequency from engine rated speed = (6000/60) * (3/2)

  = 150Hz

4.2.1.Criteria

From the engine excitation frequency with a safety margin of 150% which was mentioned above, criteria was set to 375 Hz for first mode frequency. But a constraint like diameter of the hanger rod is set by

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vendor. So, only the geometrical aspect of the design is analyzed and validated.

4.2.2.Materials property

Linear material properties in cold condition like Young's modulus, density, and Poisson ratio are considered.

Table 1. Material properties

Materials	Young modulus N/mm^2	Poisson ratio	Density Ton/mm^3
SS409	2.08e05	0.3	7.7e-09
S10C	2.0e05	0.29	7.84e-09

4.2.3. Analysis process flow

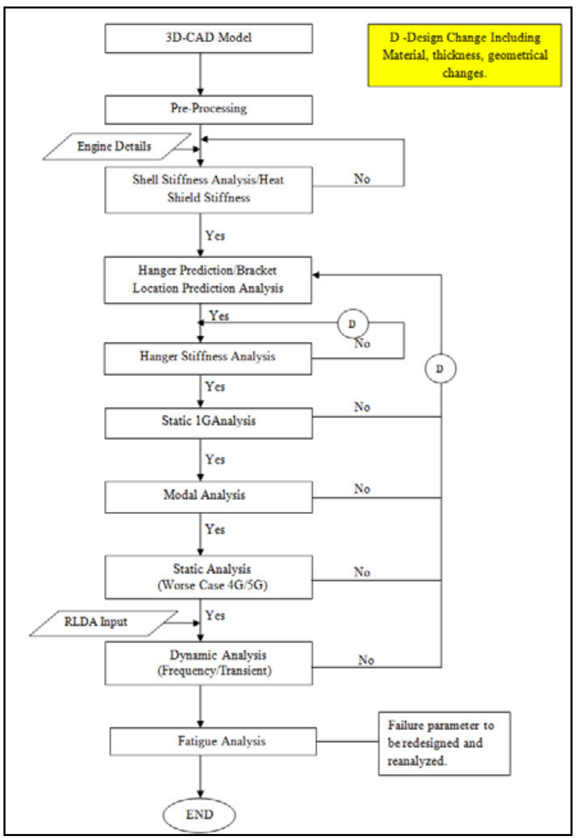

Figure 3. Process flow

4.2.4.Boundary conditions

Table 2.Weld length and hanger type

Hangers	Type	Materials	Weld length(mm)
Hanger 1	Solid	S10C	60*2
Hanger 2	Hollow	SS409	80*2
Hanger 3	Bracket with hanger	SS409	60 25

All weld nodes of hanger rod and bracket with hanger rod are constrained in all degrees of freedom, which are represented by the Figure 4 shown below

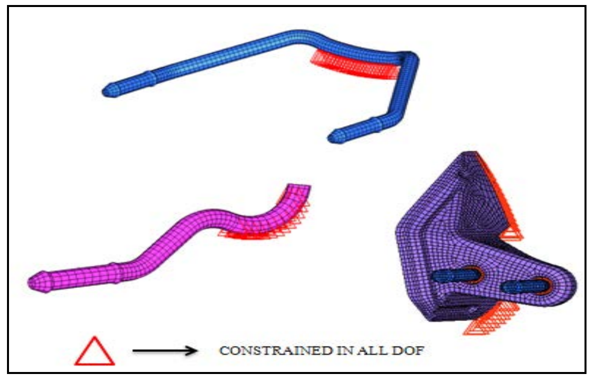

Figure 4. Weld nodes constrained in all d.o.f

4.3.Hanger stiffness analysis- virtual analysis plots

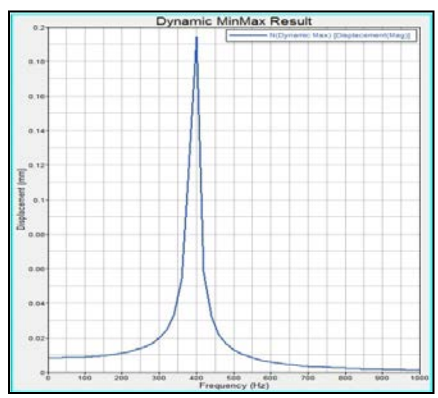

Figure 5. Hanger 1 min. resonance frequency 400Hz

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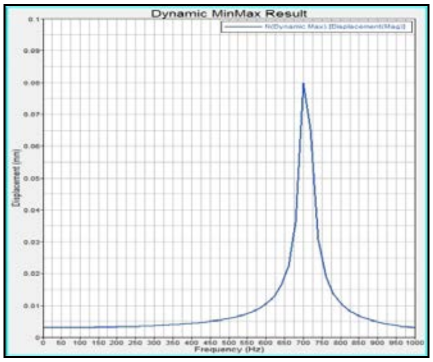

Figure 6. Hanger 2 min. resonance frequency 700Hz

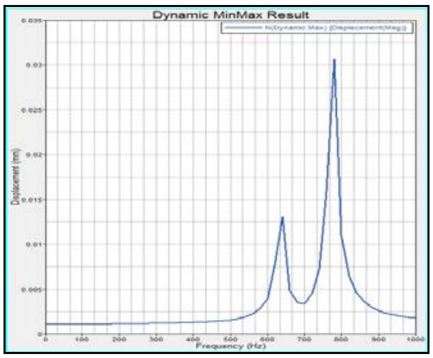

Figure 7. Hanger 3 min. resonance frequency 635Hz

4.3.1.Observations

- It has been observed that Hanger rod's stiffness is above the targeted natural frequency of 375 Hz.
- It has been concluded that Static analysis, Modal analysis have to be performed to check the resonance frequency, hanger forces, stresses and deformations for validating the hanger rod design.

4.4.Impact testing

To verify the resonance frequency of the hanger rod, say for example shown in the Figure 8, roving hammer impact test has been performed for all 3 hanger rods.

Figure 8. Tri-accelerometer mounted in the hanger 3 for impact testing

4.4.1.Hanger stiffness analysis-experimental analysis plots

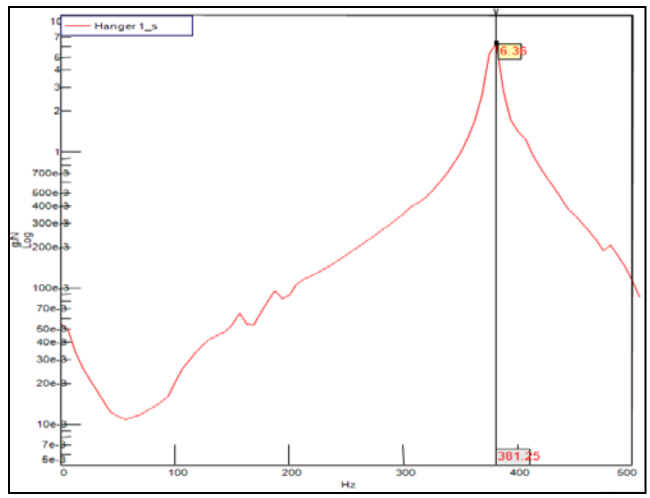

Figure 9. Hanger-1,frequency-381Hz

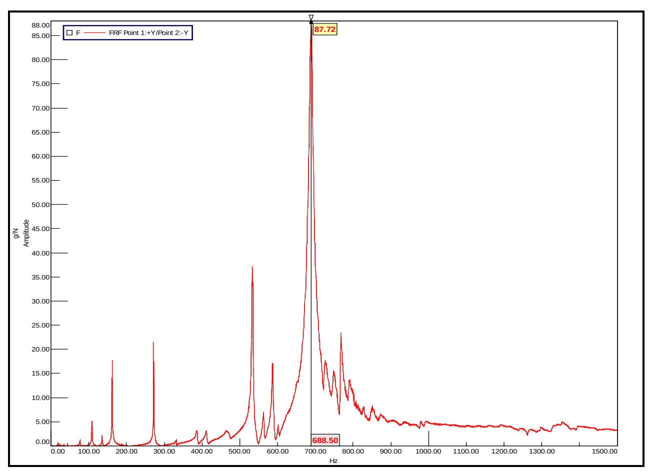

Figure 10. Hanger-2,frequency-688Hz

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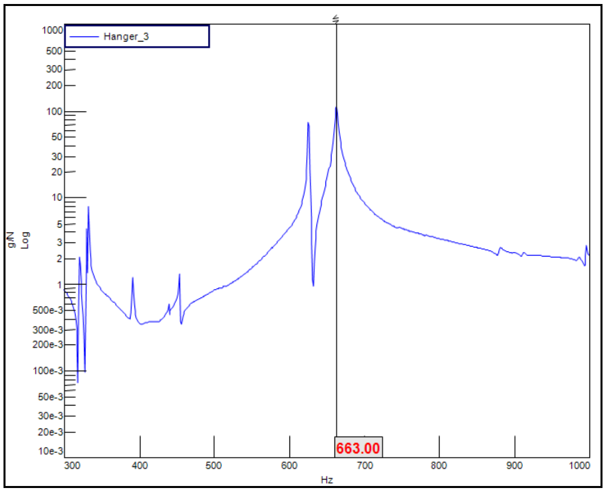

Figure 11. Hanger-3, frequency-663Hz

Table 3. Error calculation

Hanger rods	Frequency(Hz)		Error (%)
	Experimental	Impact test	
Hanger 1	400	381	4.75
Hanger 2	700	682	2.57
Hanger 3	635	663	4.4

4.4.2.Observation

It has been concluded that the error results observed in both the analysis and tests are less than 5%, and thus the CAE results are validated.

5.Static analysis

5.1.Boundary condition

For rubber isolator– static stiffness has been defined. Isolator is simulated as spring element – (cbush) connected to 3 rows of nodes of hanger rod (modeled with rigid element rbe2)

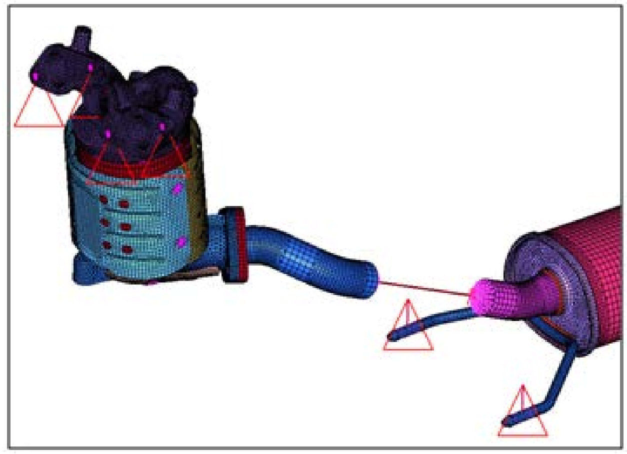
Figure 12. Bolt holes of manifold connecting to engine are constrained in all d.o.f (u_x, u_y, u_z, r_x, r_y, r_z= 0)

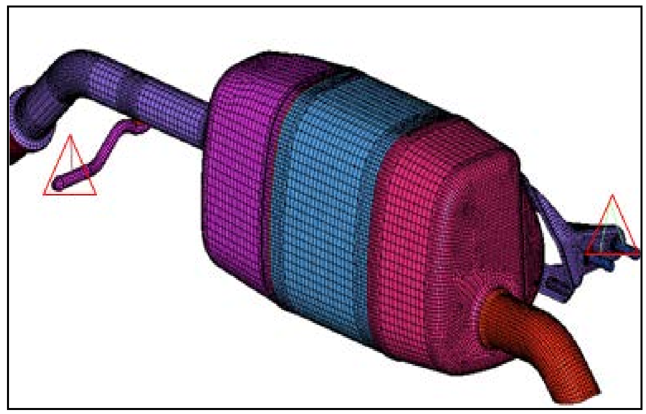
Figure 13. Top node of isolator is constrained in all d.o.f

5.2.Loading condition

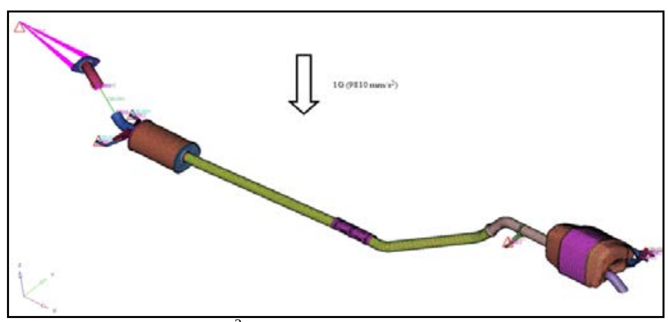
Figure 14. 1G (9810mm/s^2) is applied in vertical direction i.e. system self weight condition

5.3.Static plots

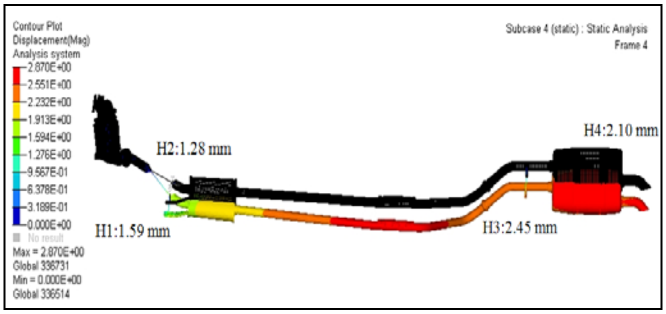

Figure 15.Max displacement of 2.87mm in the middle pipe

Table 4.Hanger forces and hanger displacements

Components		Load (N)	Displacement(mm)
Hanger 1	LH	11.56	1.59
	RH	9.27	1.28
Hanger 3		30.05	2.45
Hanger 4		23.67	2.10

5.4.Observations

- For 1G Static loading, maximum displacement of 2.87mm is observed at the middle of the centre pipe.
- The stresses obtained are within the stress limit for all the components when comparing with their material yield strength.
- Further to this analysis, system natural frequency has to be verified with engine excitation frequency to check for resonance, in-order to proceed with random vibration analysis.

6.Resonance frequency analysis

6.1.Boundary condition

For rubber isolator and flex bellow – dynamic stiffness are defined. Boundary condition is same as above mentioned in the Figure (12-13). Isolator is simulated as spring element – (cbush) connected to 3 rows of nodes of hanger rod (modeled with rigid element rbe2.

6.2.Mode shape plots

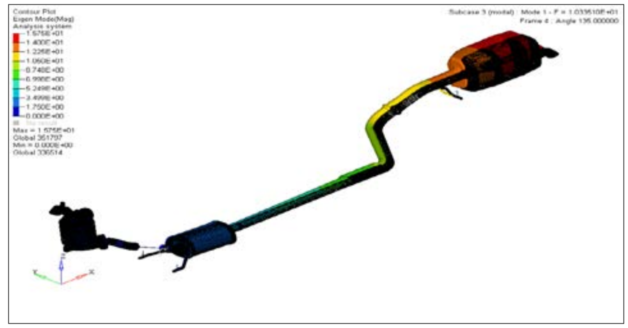

Figure 16. Mode1-Lateral at 10.33Hz

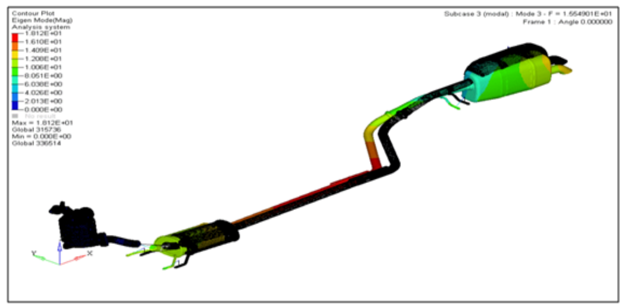

Figure 17. Mode3-Lateral mode at 15.54Hz

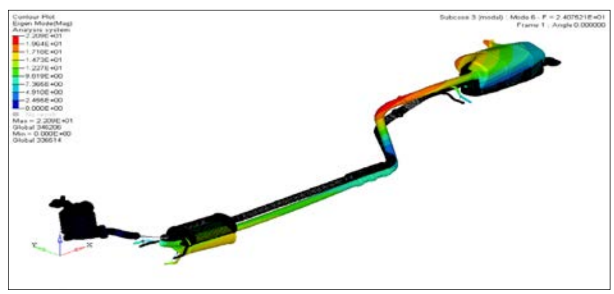

Figure 18. Mode 6-Vertical mode at 24.07Hz

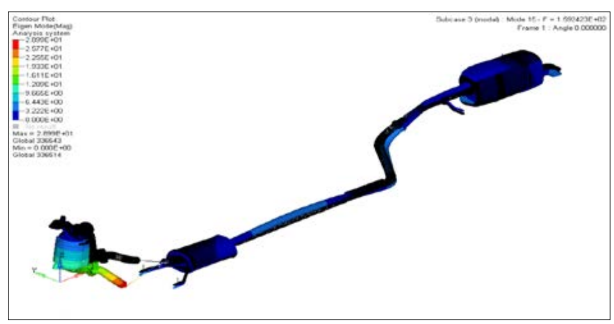

Figure 19. Mode 15-Vertical mode at 159.24Hz

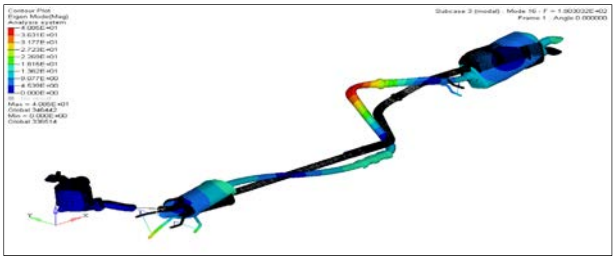

Figure 20. Mode 16-Twisting mode at 180.30Hz

Table 5. Mode shapes

Modes	Description	Frequency (Hz)
1st	Lateral of intermediate pipe	10.33
2nd	Lateral mode of sub resonator, CTR pipe and vertical bending mode of Main resonator assembly.	13.78
3rd	Lateral mode of sub resonator, CTR pipe and Main resonator.	**15.54**
4th	Vertical Bending mode of Hanger rods, Sub- resonator and Main resonator assembly.	17.05
6th	Vertical mode of sub resonator, CTR pipe and Main resonator.	24.07
8th	Vertical Bending mode of sub resonator, Twisting mode of CTR pipe and Main resonator	37.08
10th	Vertical Bending mode of Hanger rods, Sub- resonator and Main resonator assembly.	73.65
12th	Twisting mode of Hanger rods, Sub- resonator and Main resonator assembly.	114.35
15th	Vertical bending mode of Manifold and Converter assembly.	**159.24**
16th	Twisting mode of Sub- resonator, Vertical Bending mode of CTR Pipe and Main	180.30

	resonator assembly.	

6.3.Observation

- The natural frequency and mode shapes are determined from resonance frequency analysis and compared with excitation frequency.

- The natural frequency values do not match with excitation frequency(150Hz) values, so resonance will not occur.

Further to this analysis, dynamic analysis and fatigue analysis has to be performed on the finalized system with road load data to validate the design for optimization.

7.Dynamic analysis

For rubber isolator and flex bellow – dynamic stiffness are defined. Boundary condition is same as above mentioned in the Figure (12-13). Isolator is simulated as spring element – (cbush) connected to 3 rows of nodes of hanger rod (modeled with rigid element rbe2)

7.1.Loading Conditions

- Dynamic load considered as engine rocking load of $\pm 4^{o}$ varying with time i.e. (0 - 2sec).
- Deformation and stress plot values are to be checked for the applied load.
- Bending moment at Front resonator inlet and outlet and main resonator inlet should meet the target criteria.
- Stress in the hanger rod should be within the endurance strength limit.

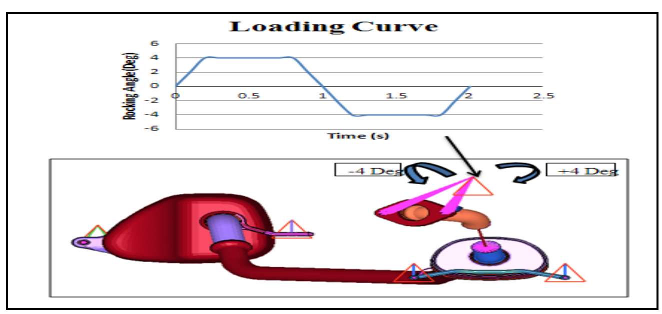

Figure 21. Applied load as enforced rotational displacement of +/- 4 degree at engine C.G

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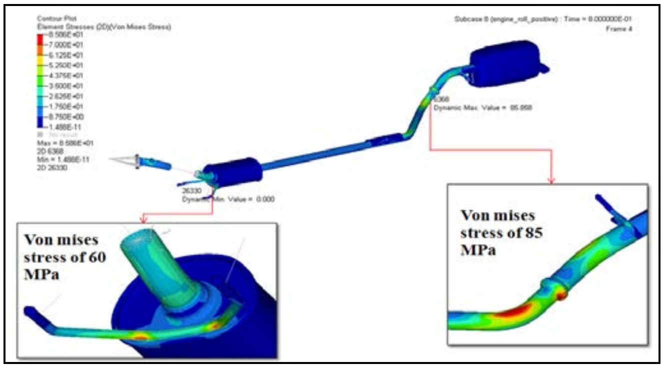

Figure 22. Stress plots

7.3.Observation

- Maximum Von-mises stresses of 85MPa observed in the bend region of the intermediate pipe. And von misses stress of 60MPa near the weld regions of hanger 1.
- It has been concluded that the system satisfies the durability requirements under engine roll loading condition.
- Dynamic stresses are compared with the acceptable limit of the materials. Since the Power-train translation and the engine vibration loading have a high number of cycles, the Von-mises stresses were judged against endurance limit of the material at the respective temperature for expected life. Empirical formulas were used to convert the ultimate strength to endurance limit. Number of cycles to failures observed in testing (life of the system) justifies the stress level predicted in the analysis.

8.Validation

The exhaust system models were validated by comparing the natural frequencies and modes of the FE model to the experimentally measured values. To calculate the natural frequencies, modal analysis was performed with the preloading effect to capture accurate behavior of system. The modal strain energy distribution was used to identify the critical locations. These critical locations were refined in the model to meet stress convergence criteria before full dynamic analysis. Table 3 shows theoretical and experimental modal analysis results for the exhaust system. It shows good agreement with the testing and CAE evaluation.

8.1.Durability test

- Maximum load that may act on the exhaust system during its service life may be derived from RLDA.
- RLDA input can be used for accelerated validation of exhaust system durability by component wise.
- RLDA is very important to derive whether the system will meet the end usage durability target or not.
- Usually, the load is acquired for Hanger rod is in the vertical direction (Fz).

8.2.Analysis and test setup

- To carry out the test, hanger rods are fixed in the same direction as it is mounted in the chassis.
- To interpret the same position, hanger rods was fixed as shown in the Figure (26-28) below. From damage comparison, the drive file generated by Belgian block was considered. Drive file was applied to vehicle's vertical (Z) direction.
- Uni-axial actuator is connected to the hanger rod in which the force Fz is applied.

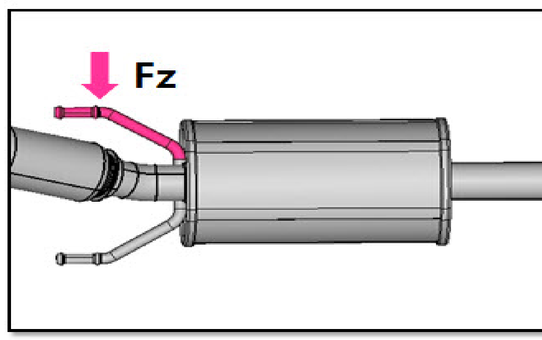

Figure 23. F_z applied-hanger 1

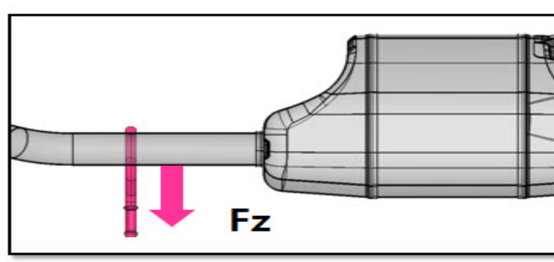

Figure 24. Fz applied-hanger 2

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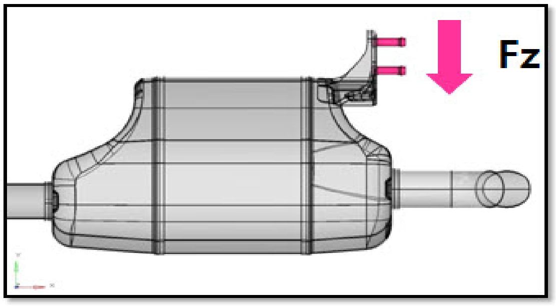

Figure 25. Fz applied-hanger 3

8.3.Test result

The test profile was repeated to pre-defined cycles to simulate the durability target. After the test, the hanger rods showed no defects.

Figure 26. Test setup-hanger 1

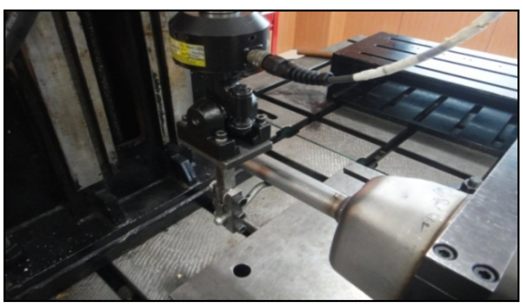

Figure 27. Test setup-hanger 2

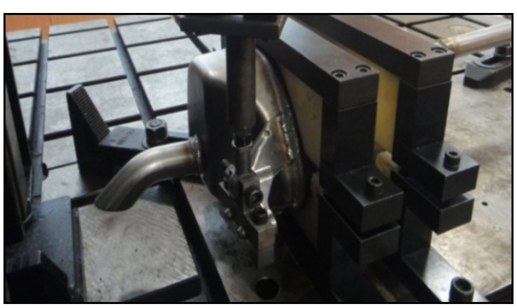

Figure 28. Test setup-hanger 3

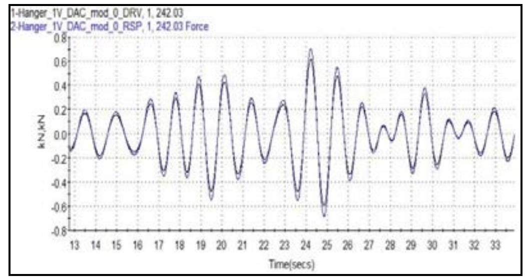

Figure 29. Drive file-hanger 1

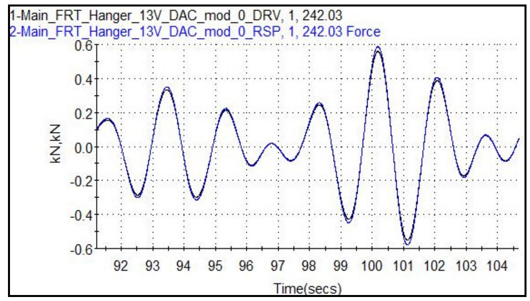

Figure 30. Drive file-hanger 2

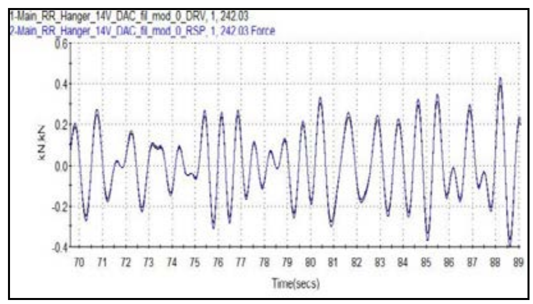

Figure 31. Drive file-hanger 3

9.Design improvement

In order to make the design optimization in the hanger rods 2 & 3, parameters like diameter, thickness and types can be altered. Stress results should be compared for initial design and modified design in same locations. Need to evaluate the dynamic analysis of initial design and improved design to predict the induced stresses which should be within acceptable limits of the material.

10.Conclusion

The detailed analysis approach developed in this study will help the engineers to predict the stiffness and durability performance of the exhaust system and develop a better exhaust system with quick turnaround time. Application of measured excitations, assembly loads and effects of manufacturing inaccuracies in dynamic analysis has shown enhancement in predicting the performance of exhaust system. Material properties change significantly with temperature; material stiffness is reduced by the increased temperature and has effect on the vibration characteristics.

It is possible to identify the failure locations and find out solution for the problem. The results obtained assure the structural integrity of the modified exhaust system when implemented on the vehicle. This methodology also contributes to a better understanding of system behavior and its structural strength, for future project applications. The same approach can also be extended to analysis of exhaust system resonator shell, baffles, end cap, heat shield and mounting brackets.

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