Journal of Information & Optimization Sciences
ISSN 0252-2667 (Print), ISSN 2169-0103 (Online)
Vol. 45 (2024), No. 4, pp. 1165–1175
DOI : 10.47974/JIOS-1700

Optimization for anti-vibration mount performance analysis

Pravin Hujare *
Dinesh Kamble †
Department of Mechanical Engineering
Vishwakarma Institute of Information Technology
Pune 411048
Maharashtra
India

Deepak Hujare @
Dr. Vishwanath Karad MIT World Peace University
School of Mechanical Engineering
Pune 411038
Maharashtra
India

Sandeep Kore §
Sonal Shamkuwar ‡
Department of Mechanical Engineering
Vishwakarma Institute of Information Technology
Pune 411048
Maharashtra
India

Abstract

The only substantial source of vibration and noise is the IC engine itself. Vibration causes system failures or damage, which is bad for the supporting system for the engine. It is necessary to ensure correct mounting in order to prevent engine vibration. Engine Mounting is one of the basic but important design characteristics in a vehicle. The engine mounts are designed with optimum stiffness and damping so as to reduce the system

* *E-mail:* **pravin.hujare@viit.ac.in (Corresponding Author)**
† *E-mail:* **dinesh.kamble@viit.ac.in**
@ *E-mail:* **deepak.hujare@mitwpu.edu.in**
§ *E-mail:* **sandeep.kore@viit.ac.in**
‡ *E-mail:* **sonal.shamkuwar@viit.ac.in**

© Taru Publications

vibration and avoid resonance phenomenon. This paper deals with the two characteristics namely damping coefficient and stiffness of the Anti-Vibration Mounts. The modal analysis and Harmonic analysis of different models of AVM has been performed. From the dynamic analysis the optimized characteristics of AVM are obtained. The model is optimized for minimum acceleration by varying two factors, i.e. stiffness and damping coefficient of AVM. Taguchi method has been used for optimization Vibration response of system using a commercial Minitab software. It is observed that the optimize values of stiffness is 450 N/mm and damping coefficient is 0.03 NS/mm. From Numerical Analysis, it is observed that the vibration response amplitude is decreases by 13.91% by using optimized AVM.

Subject Classification: *6AXX.*

Keywords: *Vibrations, Anti-vibration mount (AVM), Stiffness, Damping coefficient, Dynamic analysis.*

1. Introduction

The engine system has more number of moving and rotating parts so the engine is the most significant source of vibration and noise in the engine test bed system [1]. Hence Engine vibrations are one of the major problems for test bed manufactures. Engine runs for a long period of time, so engine as well as engine test bed system should have sufficient sustainable life for long working against vibrations, because of the engine and the engine test bed system are too much costly. If the operational frequencies of engine lie into the natural frequencies of engine test bed system, then the resonance will create and it may lead to failure of the engine components and engine test bed system. A finite element model of Engine and engine test bed system is created for this study. The finite element model was used to simulate and analyse the vibrational behaviors of the engine assembly. T. Ramachandran et al. observed that an IC engine's vibration behaviour is influenced by imbalanced rotating and reciprocating components, and the structural properties of the mounts [2]. S. Santhosh et al.'s investigation and comparison of three distinct rubber engine mounts' analytical results focused on the stiffness variation and vibrational properties, such as frequency and amplitude, in each mount. In contrast to the traditional mount made of natural rubber, the study suggested using a mix of fluorocarbon rubber mount and butyl rubber mount for improved vibration isolation [3]. The investigation was carried out by Lee et al. to lessen driver seat vibrations. To quantitatively forecast the vibration performance, CAE analysis was performed. The mount longitudinal stiffness is one of the most significant characteristics that impacts the engine start-stop vibration, as demonstrated by the author using sensitivity analysis. Vibration that is communicated to the vehicle is decreased by increasing the longitudinal rigidity of the mounting system [4]. The study by Yunhe Yu et al. is broken down into three sections: performance criteria; an overview and development of several engine mounts; and engine mount system optimization.It demonstrates that the engine mount's dynamic rigidity and damping should vary on frequency and amplitude. Because of this, the majority of research on the development of engine mounting systems has concentrated on enhancing features that are frequency and amplitude dependent[5]. J.S. Lee et al. researched engine mount rubber design strategies with the goal of enhancing performance by lowering weight, reducing maximum generated stress, and lengthening fatigue life. The Ogden-Roxburgh model was employed in the study to simulate the material non-linearities. The maximum induced stresses were reduced by 20.4% when compared to the original design, and the fatigue life was improved by 113.8%, all while just slightly increasing the weight by 0.4%[6]. The parameter optimization method was used by J. J. Kim et al.to build a bush type engine mount for passenger automobiles. To make the design more efficient, the geometry was parameterized. In order to get the appropriate rigidity of the mount, the parameter was tuned. This optimization was carried out using computer software with a finite element foundation. The virtually incompressible hyper elastic rubber material's behavior was chosen to be described by the Mooney-Rivlin model [7].

2. Anti-Vibrating Mounts(AVM)

Anti-vibration mounts are devices that are fastened to vibrating machinery in an effort to lessen the amount of vibration that is transmitted to the machine's main structure or foundation. Depending on the application, a variety of anti-vibration mounts are available, including conical mounts, machine feet mounts, wire rope mounts, etc. Additionally, these mounts are utilized in automobiles to join the engine to the chassis. A bolt holds the steel mount to the chassis on one side and the engine on the other. The rubber portion of the AVM gives the device flexibility and reduces the amount of vibration that is transferred when energy is dissipated. There should be low stiffness and damping at high frequency and high stiffness and damping at low frequency in the engine mount. [8]. The mounting system is represented by a mass-spring- damper system in Figure 1. It shows the system for a single mount [9].

The mass represents the engine of the system, and the ground represents the pallet of the system whereas the spring and damper represent the AVM. The equation of motion for mountings have combined characteristics like springs and damping is given as

$$m\frac{d^2y}{dx^2}+c\frac{dy}{dx}+kx=A\sin(\omega t) \tag{1}$$

Where, m = mass of engine, k =stiffness, c = damping coefficient, A=amplitude, ω=forced frequency [10]

2.1 *Engine Test Bed*

Figure. 2 shows the schematic diagram of the Engine Test Bed system with all the components. The anti-vibrating mounts are mounted on the ground floor on which the base frame is mounted which have fixed joint between them.

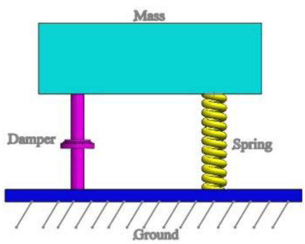


Figure 1

Spring-Mass-Damper System

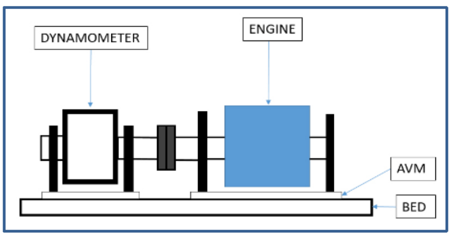


Figure 2

Schematic diagram of Engine Test Bed

The anti-vibrating mounts are providing vibration control as well as support to the system. The engine is connected to the dynamometer through rigid coupling. Dynamometer is used for loading to the system. For loading to the system, the electrical dynamometer is used. The setup is made to test engine at different speeds. The testrig contain engine, engine mounting, engine pallet, bed frame and anti-vibration mounts. Most of the vibration in a vehicle is induced by the engine. The vibration transfers towards the foundation of the system via engine mounting, engine pallet, bed frame and anti-vibration mounts. The engine is connected to the dynamometer through flexible coupling.

3. Numerical Analysis

3.1 *CAD Modelling*

The CAD model is prepared in a commercial CAD software. The CAD model consists of different components like engine test bed, base

Table 1

Details of Components

Description	Bed	Pallet	Mounting	Engine
Mass (Kg)	325	1900	650	1500

plate, pallet, mountings and engine. All components are connected with each other by fixed joint.The details of components are shown in Table. 1. The material of all components is considered to be structural steel.

3.2 *Boundary Conditions*

The Figure 3 shows the CAD model used for analysis. It consists of Engine block is placed on mountings. The mounting is placed on 8 Antivibration Mounts (AVM). The other end of AVM is connected to heavy pallet. The pallet is connected to Engine bed. The engine bed is rest on 4 fixed supports. All the contacts in the system have been considered to be bonded contacts. The boundary conditions for the AVM are mentioned in Table.2.

Table 2

Boundary Conditions for all AVM

Boundary Conditions	**Degree of freedom**
Displacement	Ux, Uy = 0
Moment	Mx, My, Mz = 0

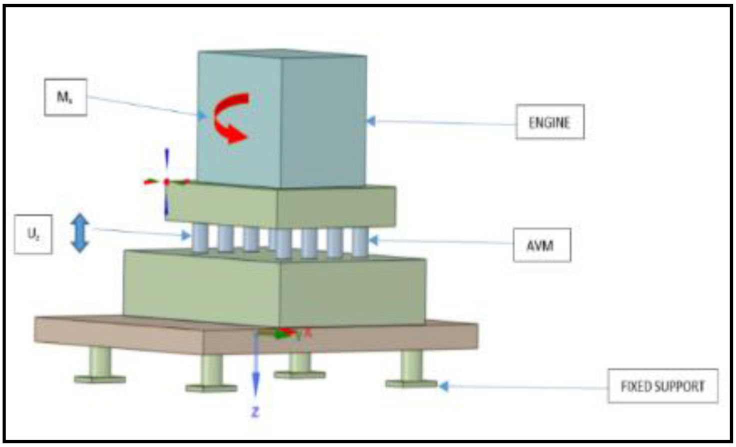


Figure 3

Boundary conditions for Engine Test Bed

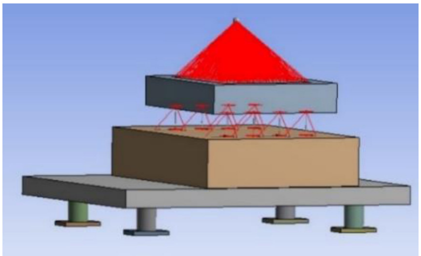


Figure 4

Engine Block Representation

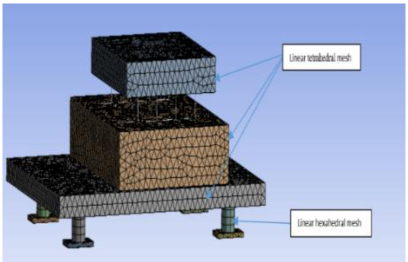


Figure 5

Mesh model of Engine Test Bed

3.3 *Meshing*

Meshing is the process of discretizing the modelled system into smaller elements to proceed with the FEA analysis. The CAD model is imported in ANSYS software for analysis. The Engine block is represented by point node element located at center of mass of engine as shown in Figure 4, Figure 5 shows mesh model with element size of 40 mm. This point element is connected to nodes of elements for top surface of mounting with the help of RB2 elements. The Linear tetrahedral element is used for mounting, pallet and engine bed. The linear hexahedral elements are used for fixed support. The antivibration mounts are represented by spring element, COMBIN14.The modal analysis and harmonic analysis of CAD model is performed in ANSYS software.

3.4 *Modal Analysis*

Modal analysis is used to calculate the various natural frequencies of the system. It is also used to find out mode shapes and mode vectors of a structure. Mode shapes or natural vibration shapes describe the deformation that the body in concern would show when vibrating at its natural frequencies. In reality, there are infinite number of natural frequencies and hence infinite mode shapes for any system. However, by reducing the degree of freedom of the system, a finite number of natural frequencies are taken into consideration. Figure 6 shows the mode shapes obtained for the first 4 natural frequencies of the engine bed.

Mode 4 shows the movement of the engine bed in the vertical axis. Hence the frequency corresponding mode 4 is considered for harmonic analysis of all models. The natural frequencies corresponding to the first 4 modes are shown in Table. 3

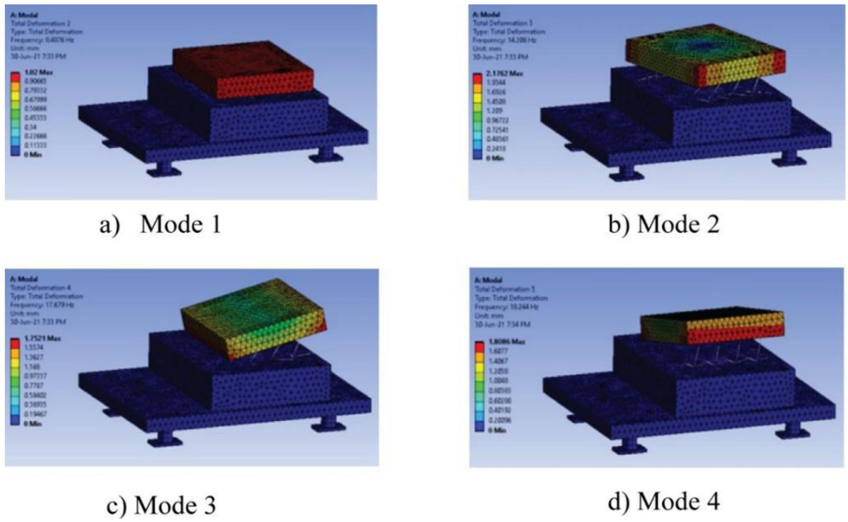


Figure 6

Mode shapes of Engine bed

Table 3

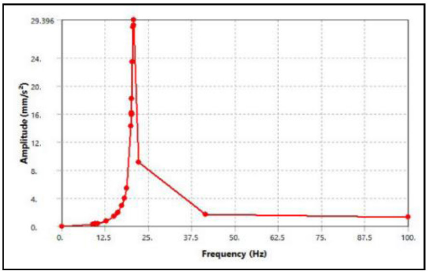
Modes and their natural frequencies

Mode No.	1	2	3	4
Frequency (Hz)	8.4876	14.206	17.679	18.244

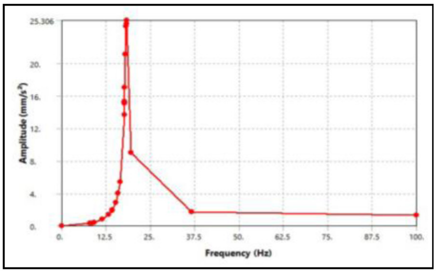
3.5 *Harmonic Analysis*

A harmonic analysis is used to determine the structure's response to a steady-state sinusoidal (harmonic) loading at a particular frequency. The effect of damping properties of the material are take into account in this analysis. For setting up the Harmonic analysis for this study, a moment of 300 N-m was applied on the engine about its principal axis, and bottom plates were fixed. The study was performed parametrically in which the stiffness of the anti-vibration mount was varied between 350 and 450 N/mm. Also, the damping co-efficient was varied between 0.01 N-s/mm and 0.03 N-s/mm. As shown in Figure 7 for case 7, the harmonic analysis yields a maximum amplitude of 29.396 mm/s^2 and for case 9 yields a maximum amplitude is 25.306mm/s^2.

Table 4 shows that as stiffness increases, the system's natural frequency increases. Also, varying the damping co-efficient does not have any effect on the natural frequency of the system. Table 4 shows vibration response amplitude for all 9 cases obtained by harmonic analysis.



Case 7:For K=450 & C=0.01 the maximum amplitude is 29.396 mm/s^2 peak at 20.691Hz



Case 9:For K=450 & C=0.03 the maximum amplitude is 25.306 mm/s^2 at 20.691 Hz.

Figure 7

Frequency Response Curve

Table 4

Vibration response amplitude for all 9 cases

Sr. No.	Stiffness (N/mm)	Damping Coefficient (N-S/mm)	Max. Amplitude (mm/s^2)	Freq. (Hz)
Case-1	350	0.01	29.130	18.25
Case-2	350	0.02	27.083	18.25
Case-3	350	0.03	25.913	18.25
Case-4	400	0.01	29.275	19.51
Case-5	400	0.02	27.334	19.51
Case-6	400	0.03	25.634	19.51
Case-7	450	0.01	29.396	20.69
Case-8	450	0.02	27.545	20.69
Case-9	450	0.03	25.306	20.69

4. Taguchi Analysis

The Taguchi optimization method is used to identify the optimized values of stiffness and damping. Once the range of factors are fixed then levels from Taguchi method are found. For finding levels each factor is varied while other factors are constant. With FEA simulation the vibrational acceleration is estimated. The parameters used for Taguchi analysis are shown in Table. 5

Table 5.

Parameters for Taguchi Analysis

Parameter	Values
Damping Co-efficient (c) ,N-S/mm	0.01, 0.02, 0.03
Stiffness (k), N/mm	350, 400, 450

4.1 *Taguchi Optimization*

Taguchi method is used for design optimization. Minitab software is a commercial statistics software which has been used to perform data. For the design optimization, two factors with three levels are considered, hence L9 orthogonal array method is used [11, 12] as shown in Table. 6.

Table 6

Orthogonal Array – L9

Experiment No.	1	2	3	4	5	6	7	8	9
Stiffness	350	350	350	400	400	400	450	450	450
Damping Coefficient	0.01	0.02	0.03	0.01	0.02	0.03	0.01	0.02	0.03

Our main objective is to minimize the vibration response, i.e., vibrational acceleration should be minimum. Hence signal to noise ratio-smaller the better is used for analysis.

The Equation (2) gives the relation of S/N ratio with experimental data and number of readings.

$$\frac{S}{N} = -10\log\frac{y_i^2}{n} \tag{2}$$

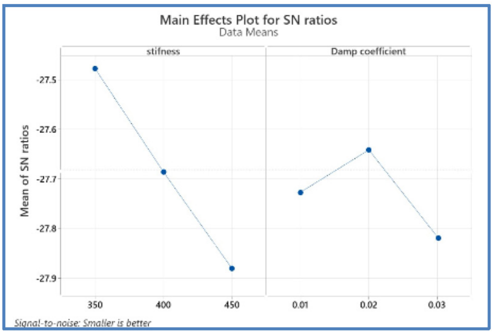


Figure 8

S/N ratio

Where, yi = Experimental data, n = Number of readings

According to Figure 8, the mean SN ratio shows a decreasing trend as stiffness increases. This leads us to consider stiffness values that are greater and damping coefficients that are greater as the optimal input parameters for further study.

5. Result and discussion

The performance of AVM is measured in terms of vibration response amplitude of Engine bed. The performance of AVM depends upon its two properties i.e. stiffness and damping coefficients. The Taguchi optimization method is used to determine optimization parameters of AVM. The optimization of AVM parameters is used to identify minimum vibration response of Engine bed at mode 4. Table 4 shows the optimize parameter of AVM. Stiffness of 450 N/mm and damping co-efficient of 0.03 N-s/mm were obtained as the optimum parameters.

6. Conclusion

In this study the method of parameter optimization to reduce the vibrations in an engine assembly was discussed. All the steps for performing such an analysis including setup in commercial software to result interpretation have been discussed. From Figure 8, it is observed that the optimize values of stiffness is 450 N/mm and damping coefficient is 0.03 NS/mm It is found that the minimum amplitude of vibration of Engine bed using optimized AVM is 25.306 mm/s^2. From Numerical analysis, it is concluded that the vibration response amplitude is decreases by 13.91% by using optimized AVM.

References

[1] A. J. Martyr, M. A. Plint, "Engine Testing Theory and Practice", Third Edition, Butterworth-Heinemann, pp. 21–46 (2007).

[2] T. Ramachandran, K.P. Padmanaban, *Int. J. Eng. Sci. Emerg. Technol.* 3, 63–73 (2012).

[3] S. Santhosh, V. Velmurugan, V. Paramasivam, S. Thanikaikarasan, Experimental investigation and comparative analysis of rubber engine mount vibration and noise characteristics. Materials Today: Proceedings, (2019)

[4] Y. Lee, Study on Transient vibration response characteristics of front transverse power train at key on/off, SAE Technical Paper Series (2007).

[5] Yunhe Yu, Saravanan M.Peelamedu, Nagi G.Naganathan, Rao V.Dukkipati, Automotive engine mounting systems- A survey. J.of dynamic systems, measurement and control, 123: 185-194 (2001).

[6] Altameem, Ayman, et al. "P-ROCK: a sustainable clustering algorithm for large categorical datasets." Intell. Autom. Soft Comput 35.1 : 553-566 (2023).

[7] J J Kim, H Y Kim, Shape design of an engine mount by a method of parametric shape optimization. Proceedings of Institutions of Mechanical Engineers, 211 (1997).

[8] N.S.Ahirrao, Dr.S.P.Bhosle, Dr.D.V.Nehete, Dynamics and Vibration Measurements in Engines, Procedia Manufacturing 20, 434–439 (2018).

[9] Hazim Nasir Ghafil, Vibration Analysis of a Generator Anti-Vibration Rubber Mounts, *Journal of Computer Science & Computational Mathematics*, Volume 6, Issue 4, December (2016).

[10] A. Carrella, Nonlinear identifications using transmissibility: Dynamic characterisation of Anti Vibration Mounts (AVMs) with standard approach and nonlinear analysis, *International Journal of Mechanical Sciences* 63, 74–85 (2012).

[11] Alexander Bervian, Eliane Coser, and et.all, Application of Taguchi Method to Study Morphological Evolution of TiO2 Nanotubes Obtained Via Anodization Process, Materials Research (2018).

[12] Abdulghani, Farah A., and Nada AZ Abdullah. "Hybrid deep learning model for Arabic text classification based on mutual information." *Journal of Information and Optimization Sciences* 43.8 : 1901-1908 (2022).