Available online at www.sciencedirect.com

ScienceDirect

Procedia Engineering 144 (2016) 729 – 735

Procedia Engineering

www.elsevier.com/locate/procedia

12th International Conference on Vibration Problems, ICOVP 2015

Vibration Environment and the Rockets

P C Jain*

Scientist 'G', DRDL Hyderabad (TS)

Abstract

Rockets are subjected to vibrations in various operational phases. Source of vibration may be due to air turbulence, propeller noise, engine noise, stage separation, pyro ignition, release mechanism etc. Some of the sections of rocket houses electronic equipments for data acquisition, navigation purposes etc. Any Structure possessing mass and elasticity vibrate under external disturbances. Electronic equipment can be subjected to many different forms of vibration over a wide range of frequencies and acceleration levels. Vibration is usually considered to be an detrimental condition and can produce many different types of failures in electronic equipment. It is always an endeavour to alleviate or minimize the vibration amplitudes. Wide variety of structures and their components are susceptible to vibration as they are subjected to time dependent forces. If the vibration amplitude of structure escalates to the upper levels, the failure of structures takes place. Vibration suppression has greater significance for aero-space structures, which houses sophisticated electronic equipments, as the high vibration amplitudes hinder the performance of such equipments. In this study it is demonstrated that Tuned Mass Dampers (TMDs) placed with individual modal tuning shows satisfactory performance for the structure subjected to wide band harmonic excitation, transient or random excitations.

Peer-review under responsibility of the organizing committee of ICOVP 2015

Keywords: Vibrations; Tuned mass damper ; modes; finite element

1. Introduction

Many of the structural components at some point during their lifetime are subjected to mechanical vibrations. They must therefore be designed to withstand such conditions without damage. In this study Rocket structural components are considered. Rockets are subjected to vibration in various operational phases due to air turbulence, propeller noise, engine noise, stage separation, pyro ignition, release mechanisms etc.. Many **authors [1-4] have**

* Corresponding Author:
Email: jainpcj@yahoo.com

Peer-review under responsibility of the organizing committee of ICOVP 2015
doi:10.1016/j.proeng.2016.05.077

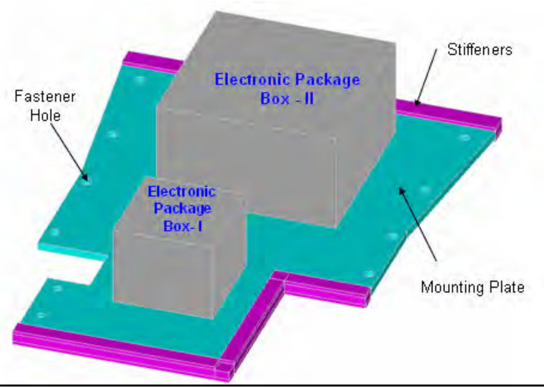
studied the tuned mass dampers for vibration reduction. The concept of tuned mass damper also called vibration TMD is extended to real life structural system. Modal tuning algorithm is implemented and the effectiveness of vibration TMDs for various type of input excitation is studied.

2. Concept of Tuned Mass Damper

Sinusoidal force acting on main spring-mass system causes the structure to go into resonance if the forcing frequency equals the natural frequency of the main mass. The resonance can cause severe problems for vibrating systems. When an absorbing mass-spring system is attached to the main mass and the resonance of the TMD is tuned to match that of the main mass, the motion of the main mass is reduced to zero at its resonance frequency. Thus, the energy of the main mass is apparently absorbed by the tuned mass damper. The motion of the tuned mass damper is finite at this resonance frequency, even though there is no damping in either oscillator. This is because the system has changed from a 1-DOF system to a 2-DOF system and now has two resonance frequencies, neither of which equals the original resonance frequency of the main mass (and also the tuned mass damper). The present study focuses the extending the concept of vibration TMD to real life structural systems.

3. Description of the Structural System

A plate mounted in longitudinal direction of the Rocket is used to house the on-board electronic package box. This plate is termed as mounting plate. The mounting plate with electronic package box is shown in Fig.-1. The basic dimensions and thickness of the mounting plate is shown in Fig.-2. The mass of the mounting plate with TMD is 3.5 kg. The weight of electronic package box is 15 kg.



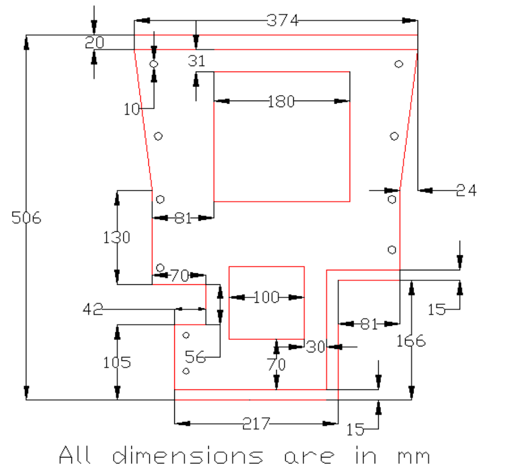


Fig.-1 Mounting plate with electronic package box

Fig.-2 Dimensioned mounting plat

4. Finite Element Modeling and Free Vibration Characteristics of Bare Plate

The finite element modelling has been carried out using Shell elements available in ANSYS. The Electronic package mass is lumped at 4 fastening locations. The mounting plate is fastened to Rocket through bracket. The fastener locations are considered fixed in finite element model. Fig.-3 shows the finite element mesh and boundary conditions of mounting plate. Free vibration analysis for plate under consideration carried out using finite element method. The first three modes of bare plate are shown in Fig. 4 to Fig.6

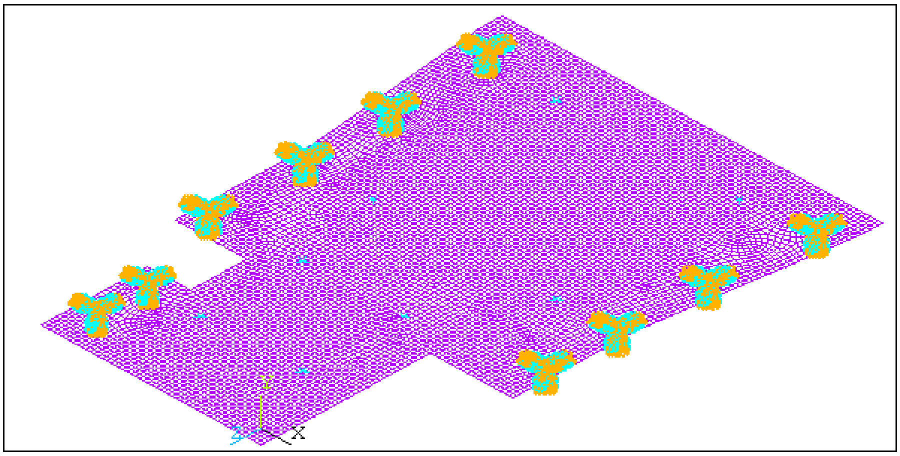


Fig. 3. Finite element mesh and boundary conditions

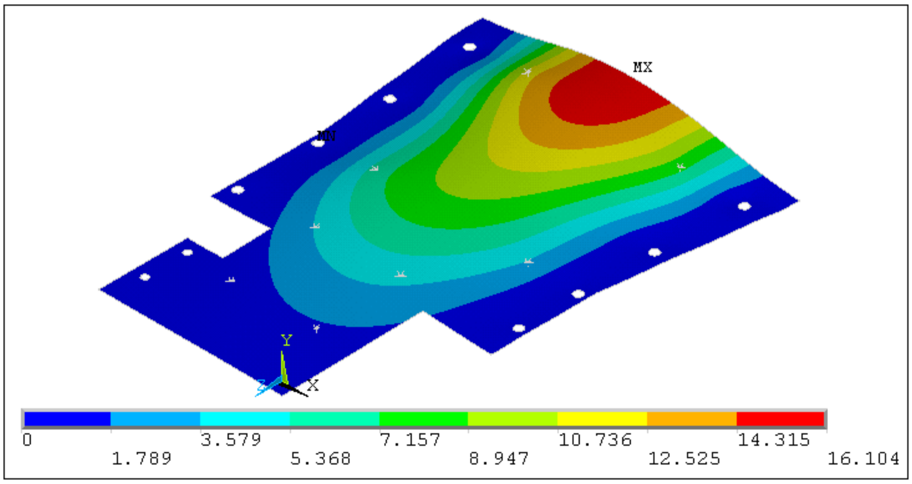


Fig. 4. Mode-I of bare plate (Frequency = 199.0 Hz)

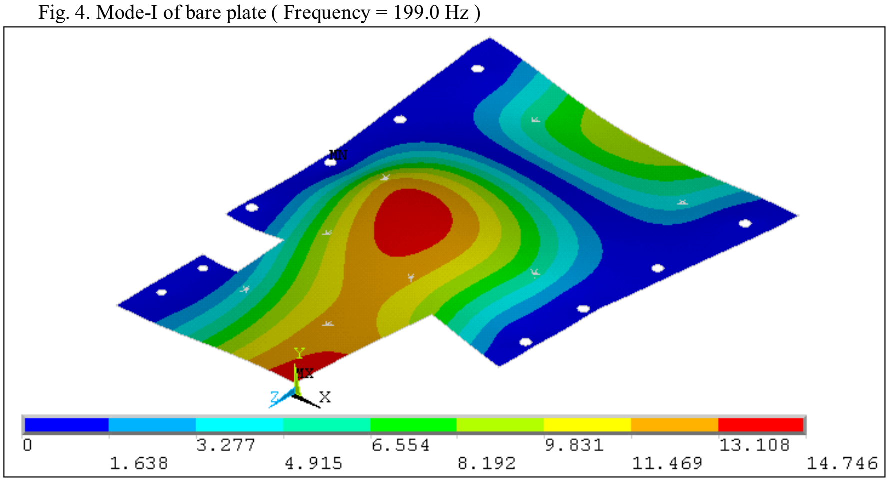


Fig.5 Mode-II of bare plate (Frequency = 231.3 Hz)

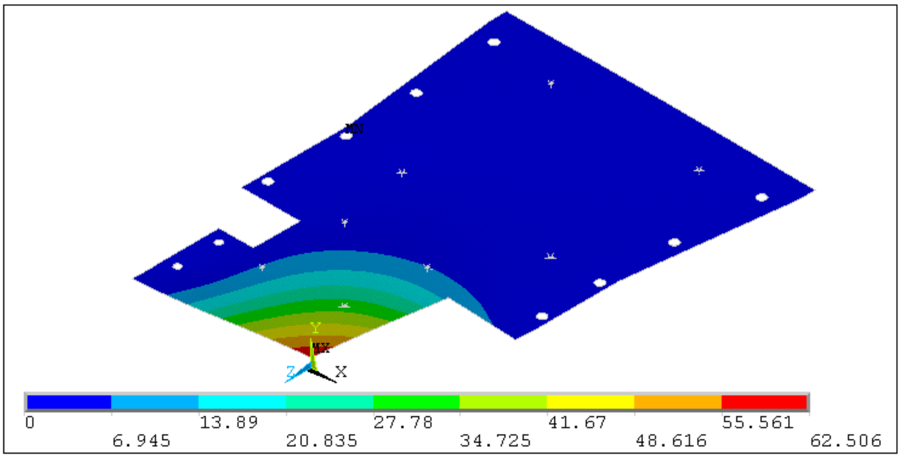


Fig. 6 Mode-III of bare plate (Frequency = 261.3 Hz)

5. Mode Wise Tuning

Modal analysis for the plate has been carried out. After retrieving the corresponding modal frequency, peak amplitude point is identified and the stiffness and mass of TMD are selected. The TMD is positioned on the identified peak amplitude point. Same steps are repeated for other modes. TMD mass is fixed from the criteria of not to exceed 10 % of mass of Plate with box , and accordingly the stiffness of TMD is selected so as to match with the first natural frequency of plate. With similar criteria the second & third TMD are fixed.

6. Harmonic Response

After first mode tuning, excitation is carried out at first modal peak point, and harmonic response is studied at various locations on the plate. Comparison between response of bare plate and that with TMD is shown in Fig.-7 for location A. After second mode tuning, the structure with two TMDs is excited at second mode peak point. Comparison between response of bare plate and that with TMD at point - A is shown in Fig.-8. After third mode tuning, the plate is excited at third mode peak point. Comparison between response of bare plate and that with TMD is shown in Fig.-9. Mode wise displacement comparison for three modes are given in Table-1. It is observed from the Table-1 that at point-A and C the % response reduction increases when the number of modes tuned are increased from one to three, however the point B and D which are away from excitation point have overall reduction. Gross reduction in the range of 10 – 80% is seen.

Table-1 Harmonic Response Analysis Results For Various Modes Tuning

Number of modes tuned	Peak Displacement in (mm)					
	Point – A			Point - B		
	Without TMD	With TMD	% Reduction in response	Without TMD	With TMD	% Reduction in response
1	2.01	1.653	17.8	0.85	0.24	71.7
2	0.415	0.136	67.2	0.365	0.32	14.1
3	0.0548	0.012	78.1	0.0966	0.0286	70.4

Number of modes tuned	Peak Displacement in (mm)					
	Point - C			Point – D		
	Without TMD	With TMD	% Reduction in Response	Without TMD	With TMD	% Reduction in Response
1	2.9	2.4	17.2	1.53	0.39	74.5
2	0.62	0.2	67.7	0.7216	0.6476	10.2
3	0.0834	0.0158	81.0	0.2418	0.0305	87.4

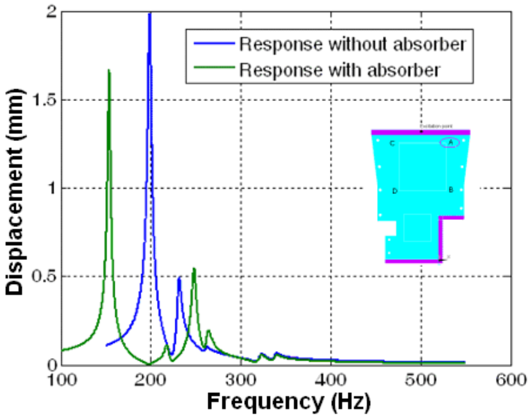


Fig. 7. Mode- I tuning (response location – A)

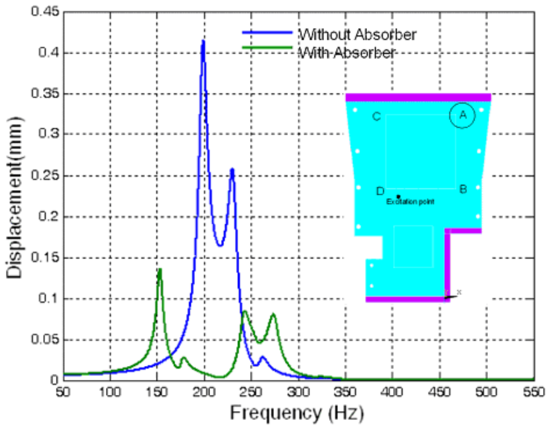


Fig. 8. Mode- I & II tuning (response location – A)

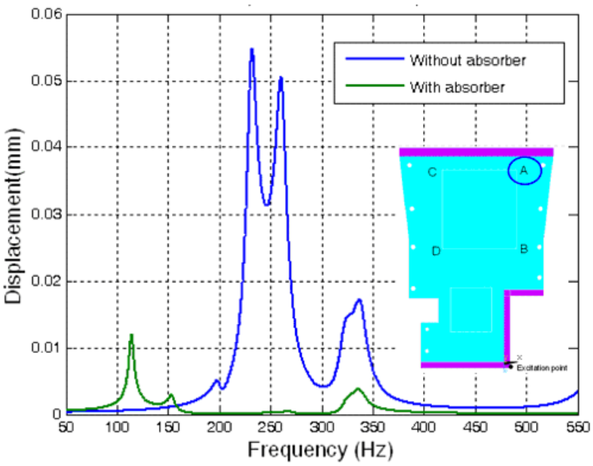


Fig. - 9 Mode - I , II & III tuning , excitation point -3 , response location – A

7. Time History Response

Harmonic loading as considered in the previous section is not always the case, instead time dependent transients are experienced on the structure, hence the measured time histories on the Rockets (rocket noise) have been considered as input to the plate, time domain analysis is carried out. The Newmark method is used for transient dynamic analysis. This method uses finite difference expansions in the time interval Δt. New mark integration parameters used in the analysis α, δ are 0.5 & 0.25 respectively. Time step used for integration is 0.0005 sec. A continuous time history in the time range of 0 sec to 0.5 sec have been considered. Transient dynamic analysis carried out for the time history shown in Fig. 10. The results for the input are given in Fig.-11. The tabular comparison of time history peak results are given in Table-2. It is observed from Table - 2 that there is reduction in peak response. The response peaks at various time instants are identified and the responses are compared without TMD and with TMD. The response reduction is observed from 13 % to 56 %.

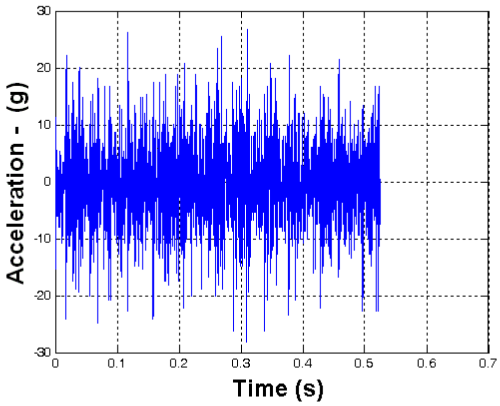


Fig. 10 Measured time history acceleration

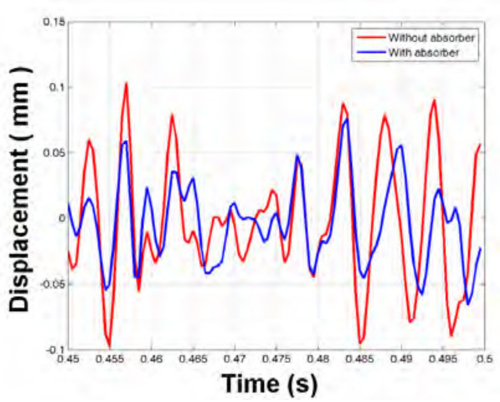


Fig. 11 Typical time history response

Table - 2 Comparison of time history response

Time (s)	Disp (mm) Without TMD	Disp (mm) With TMD	% Reduction in response
0.0025	0.0237	0.0143	39.66244726
0.02	0.0757	0.0656	13.34214003
0.041	0.1002	0.0438	56.28742515
0.047	0.1525	0.0715	53.1147541
0.1395	0.0516	0.0317	38.56589147
0.1515	0.0213	0.01058	50.3286385
0.1835	0.0909	0.0608	33.11331133
0.224	0.0723	0.0513	29.04564315

8. Power Spectral Density Analysis

Based on previous flight measured time history data, spectrums are developed. This input spectrum (Fig.- 12) is applied at the fixed points of plate and PSD analysis is carried out. The displacement response PSD is plotted for plate without TMD and with TMD at point A and shown in Fig.-13. Tabular comparison of PSD response at four mounting locations are given in Table-3. It is observed from Table-3 that there is peak response reduction at the four mounting points ranging from 23.8 to 48.5 %. As a measure to evaluate the effectiveness of TMDs the area under the curve is evaluated and the same is summarized in Table - 4 for the points A , B , C & D.

Table 3. PSD Response

Response point	Disp^2/Hz with TMD	Disp^2/Hz without TMD	% Reduction in response
Point A	0.0085	0.0165	48.48
Point B	2.10e-03	3.25e-03	35.38
Point C	0.02	0.0355	43.66
Point D	0.008	0.0105	23.80

Table 4. Area under the response curve

Location	Area integral (disp)^2		Ratio
	Without TMD	With TMD	
Point - A	0.1077E-6	0.6227E-7	1.729
Point - B	0.2689E-7	0.1873E-7	1.436
Point - C	0.2375E-6	0.1345E-6	1.766
Point - D	0.8538E-7	0.6299E-7	1.355

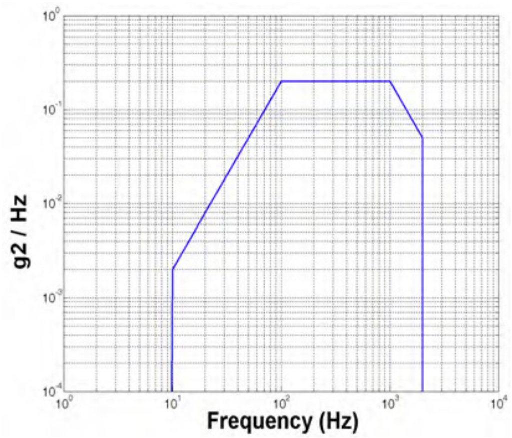


Fig. 12 Input PSD curve

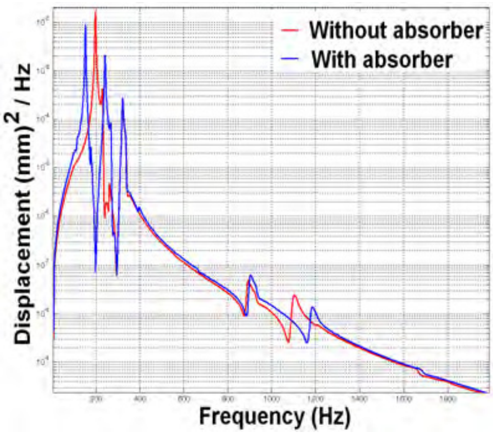


Fig. 13 Response PSD plot

9. Displacement Response of TMDs

Concept of vibration TMD is based on the fact that the energy of main mass is apparently absorbed by tuned mass damper , hence from design perspective it is important to study the response of TMD. The TMD's harmonic displacement response for TMD 1, 2 and 3 are given in Table - 5. It is observed from Table - 5 that TMD's response is less than the original plate's response.

Table 5 . Harmonic response of TMD

Location	Displacement (mm)
1	6.8
2	1.2
3	1.2

10. Conclusions

The mode wise tuning of the TMD for three modes of real life structure have been done. If the frequency of TMD is tuned to the individual modes than overall response of the plate is reduced. The response comparison is presented through graphs and table for harmonic, transient and power spectral density input. There is significant reduction in response for three types of loading. The magnitude of response reduction depends upon the location of point with respect to excitation point. TMDs placed with individual modal tuning shows satisfactory performance for the structure subjected to wide band harmonic excitation, transient or random excitations.

References

[1] T. Igusa and K. Xu, Vibration control using multiple tuned mass damper, Journal of Sound and Vibration 175(4) (1994) , pp. 491–503

[2] R. Rana and T. T. Soong, Parametric study and simplified design of tuned mass dampers, Engineering Structures, Vol. 20(3) (1998), pp. 193-204.

[3] Y. H. Chena and Y. H. Huangb, Timoshenko beam with tuned mass dampers and its design curves, Journal of Sound and Vibration 278 (2004) , pp. 873–888.

[4] Den Hartog J.P., Mechanical Vibrations,(Text Book) McGraw-Hill, New York.