Design of a Vibration Isolator for Launch Vehicle Applications

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Abstract - *A launch vehicle experiences severe vibration environments during ascent phase due to aerodynamics excitation, solid motor thrust oscillations, liquid engine cutoff transients, stage separation etc. These will produce large vibrations which are detrimental to electronic packages/modules and payloads during launch. The vibration isolators are designed to reduce the transmitted structure-borne vibrations. Isolators are used to protect electronic packages/modules and payloads from the large vibration at the time of launch and during flight. Additional damping material is also introduced in the isolator to improve the damping. Silicon rubber is usually used because of its good environmental resistance, excellent chemical resistance, easy compound modification and good ageing characteristics. This paper deals with the design and analysis of a metallic vibration isolator with a layer of silicon rubber.*

Key Words: Launch Vehicle, Vibration Isolator, Elastomer, Transmissibility

1. INTRODUCTION

In general vibration isolators are used to reduce the transmitted force from a running machine to the foundation or to reduce the transmitted response from vibrating base to the component. A launch vehicle experiences severe vibration environments during ascent phase due to aerodynamics excitation, solid motor thrust oscillations, liquid engine cutoff transients, stage separation etc. Vibration isolators are flexible elements introduced at the base of critical components so that the vibration transmission is reduced for concerned frequency range. The objective of the work is to design a metallic isolator for payloads in Launch vehicles. Frequency of isolator is specified as 20Hz with a load rating of 50kg. Elastomeric material is required to limit the response at resonance of the system. Multiple numbers of such isolators have to be used at the base of the payload in order to get the desired vibration isolation and the number of isolators depends on the total mass of the payload. Silicon rubber based isolators are commonly used in ISRO launch vehicles, a metallic design can improve the load rating and hence it is advantageous for application with heavy assemblies and satellites

2. LITERATURE REVIEW

Conor et al.(2018) in their paper discusses a resilient box shaped satellite isolator as a combination of metal and elastomer. The paper describes two isolation systems namely SoftRide and ShockRing systems with different isolation frequency specification [1].

Yewei et al. (2013) conducted an experimental study on the whole rocket vibration isolation which helps improve dynamic environment during payload launch. Using analytical expressions, of Whole-Spacecraft Vibration Isolation (WSVI) technology is discussed in this paper [2].

Xibin et al. (2018) designed Metal Rubber (MR) structures to absorb vibrations for cameras and other high-precision payloads. Here a simple isolator is configured in which elastomers are provided parallel to the loaded metallic element. Dynamical behaviors of the integrated system were analyzed to ensure that isolator flexibility would not cause the payloads to collide with the fairing [3].

Heine et al. (2013) in the paper discusses about rubber suspension of X-ray imaging spectrometer which has a very stringent heat transfer requirement from the support [4].

Valeev et al.(2016), analyzed the influence of dynamic force of machines on its foundation. Special attention is paid for vibration isolators with quasi-zero stiffness that have high efficiency. Specially designed research equipment for measurement of dynamic load has been constructed to measure the efficiency of vibration isolator very accurately.

3. PRINCIPLE OF ISOLATION

Vibration Isolator is a resilient member introduced in the load path between support and equipment in order to attenuate the vibration transferred from support to the equipment. Effectiveness of isolator is expressed in terms of motion transmissibility in the present case and it is defined as the ratio of motion transmitted to the equipment to motion of the support. For a single degree of freedom system transmissibility is less than one for the excitation frequencies more than $2^{1/2}$ times the natural frequency. So basic purpose of the resilient mount is to reduce the frequency of the system to meet this condition. With the present design of 20Hz isolator, effective isolation can be expected for excitations more than 28.3Hz.

3.1 Elastomers used for Vibration Isolators

Elastomeric materials are used to improve the structural damping so that the response at resonance can be limited. These materials have mechanical properties similar to natural rubber. This can be manufactured in desired shapes, having different stiffness values from low to high, having more damping behavior than metals, fulfilling mass and volume requirements, binding to metals effectively are important properties of elastomers for being used in vibration isolators. Elastomeric vibration isolators can sustain large deformations and can recover to their original state without any permanent deformation. Here silicon rubber is used due to its good damping properties, excellent resistance to chemicals and environment, good aging characteristics and easy compound modification. It shows minor change in transmissibility and its dynamic absorption characteristics with ageing. The excellent ability of silicone rubber to withstand extremes of temperature is one of the main reasons of choice. So in order to incorporate these properties the concept of Mooney Rivlin model for hyper elastic material is adopted in modeling.

4. DESIGN OF VIBRATION ISOLATOR

A box shaped configuration was conceived with M8 holes at middle of top and bottom sections for bolt interfaces as shown in figure 1. Beam bending of top and bottom sections with center load provides the flexibility. An isolator design has to meet strength and stiffness requirements which largely depend on yield strength and modulus of the selected material respectively. Among the common aerospace metals, Aluminum alloy, Steel and Titanium alloy, it was found that Titanium alloy is the suitable choice for our design. Load rating for the isolator is 50kg with frequency spec of 20Hz. Expected transmissibility is less than 6, which depends on the elastomer performance and has to be derived from the isolator characterization tests. Design axial load on the isolator is 5000N which corresponds to the amplification of 10 (base input is 1g).

A basic design was carried out by assuming the horizontal portions as a simply supported beam with concentrated load of at the center. Width (b) is taken as 30mm constant. From the beam bending equation the relation of length (L), thickness (d) of the section with the peak bending stress can be written as:

$$25\times10^4\times\frac{L}{d^2} = \sigma_b \text{ .... } (1)$$

Required stiffness to get 20Hz with 50kg is 788.7kN/m which gives: $\frac{d}{L} = 0.0488$.................. (2)

Elastic modulus of Ti6Al4V considered in this paper is 113GPa. Considering d=6mm, obtained value of L=123mm and peak bending stress, σ_b = 853.8MPa.

5. MATERIAL AND MODELING

Initially Aluminum alloy was considered but since the stress on the isolator exceeded the yield stress. Steel is better than Aluminum alloy in terms of strength but less flexible. With Titanium alloy (Ti6Al4V), stress on the isolator for the given dimensions is within the yield stress of 920MPa.

Above calculations showed the feasibility of the box shaped configuration for the isolator and further analysis were carried out to get frequencies and stresses. A preliminary model was made using shell elements (CQUAD4) using MSC Nastran as in figure 1. Total length of the horizontal portion in the mid-plane model is 127mm, width is 30mm and height 30mm. Axial mode of a single isolator with shell thickness 5mm and with 50kg lumped mass at fore end interface is shown in figure 2. This frequency is highly sensitivity to the thickness and variation of frequency with thickness is shown in Table 1. Other low frequency modes are discussed in the coming sections along with final design results.

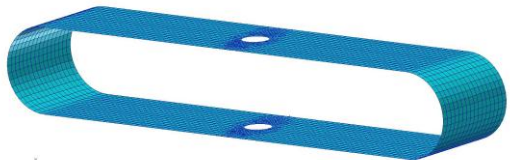


Fig -1: Shell model of the Isolator

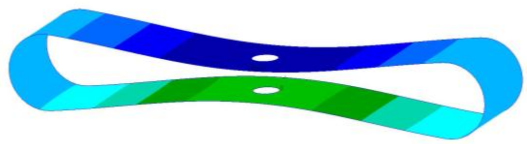


Fig -2: First mode (f= 20.9Hz)

Table -1: Shell thickness vs Frequency

Sl. no	Thickness(mm)	Frequency (Hz)
1	4.5	17.95
2	5.0	20.90
3	5.5	24.03

A case considered where isolators are used for a payload mass of 1800kg with 1.2m center of gravity (C.G) offset and interface diameter of 1.4m. Number of isolators required to carry this mass is 36nos. and isolators are arranged in circular pattern at the interface equi-spaced. Mass is connected to the isolator interfaces using rigid links (RBE2 elements) and isolators are base fixed. First mode seen is lateral with frequency of 15.7Hz as shown in figure 3 which is lower than the single isolator frequency. It can be seen from the mode shape that the axial flexibility of the individual isolator only contributing to this mode and this mode depends on Payload CG offset, number of isolators and its arrangement. However, the axial mode is same as that of a single isolator because of the symmetric mass loading as shown in figure 4. Low frequency lateral modes will cause large lateral deflection that will reduce the available dynamic envelope. Also low frequency modes will affect the whole vehicle structural dynamics since the payload mass is heavy. So we have to study these implications case by case and these aspects are not covered in this paper as it is beyond the scope.

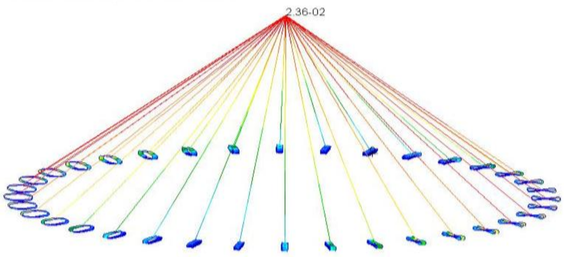


Fig -3: Lateral mode, f= 15.7Hz

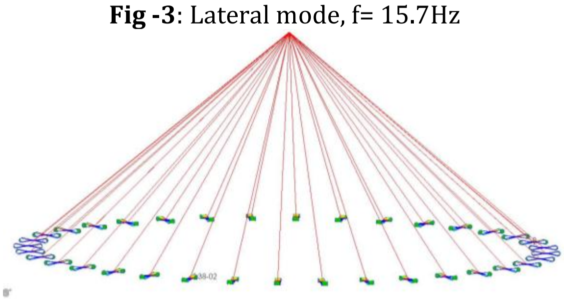


Fig -4: Axial mode, f= 20.9Hz

Design was reconfigured based on the analysis results from the shell model, by varying thickness to reduce the stress. Thickness is increased at the center and at ends to 6mm and other location it is 4mm. A solid model was made using eight node CHEXA solid elements as shown in figure 5. A silicon rubber layer of 4mm thickness is bonded on inside surfaces. A constraint layer of 1mm thickness and made of same metallic material is bonded to the surface of silicone rubber to improve damping. Nonlinear material model of silicon rubber was entered using Mooney Rivilin material model. Mooney developed a theory based on large elastic deformation of rubber and rubber-like materials in which relation for strain energy density function is obtained in terms of three strain invariants and Mooney-Rivlin constants. The three test data input to NASTRAN run (ie; uniaxial tension test, biaxial tension test and pure shear) which are required to derive the constants are shown in chart 1 to 3.

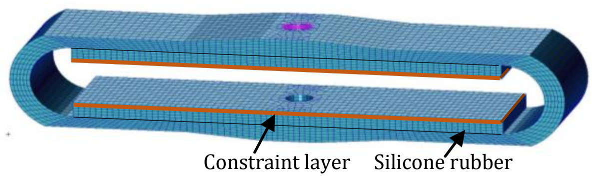


Fig -5: FE model of metallic Isolator bonded with elastomer/constraint layers

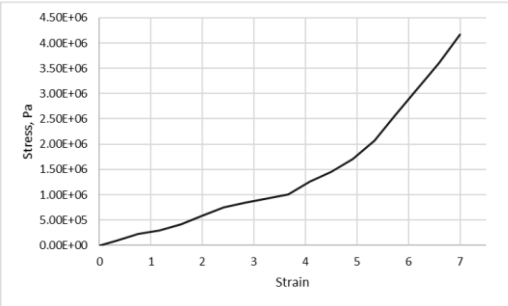


Chart -1: Uniaxial tension/compression test data of silicone rubber

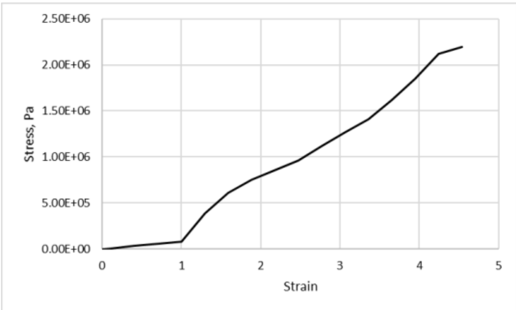


Chart -2: Biaxial tension test data of silicone rubber

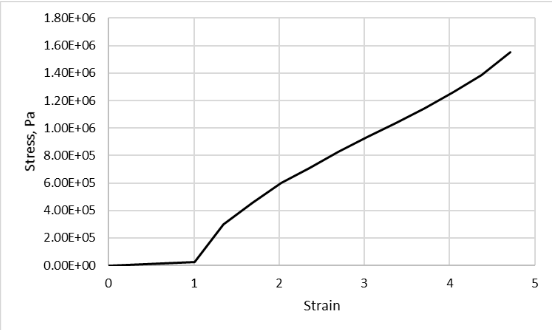
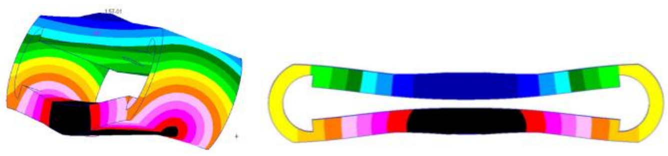
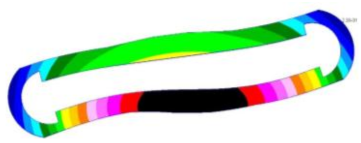


Chart -3: Pure shear test data of silicone rubber

Normal mode analysis using direct method and nonlinear static analysis were carried using the solid model. In normal mode analysis, 50kg mass was lumped at fore end interface and aft end was fixed. Mode shapes in all three directions are shown in figure 6. Axial mode is 21.6Hz as designed and all three frequencies (f) are closer as per the requirement.



Lateral-1, f=21.3Hz Axial, f= 21.6Hz



Lateral-2, f= 25.5Hz

Fig -6: Modes of a single isolator with rated mass

Axial load of 5000N corresponds to a transmissibility of 10 (for 1g excitation at base) was considered for static analysis. von Mises stress on the metallic parts are shown in figure 7. High stresses are seen on edges very locally; otherwise values are well within the yield of the material. Stress on the constraint layer is very less (154MPa max) and is not shown here. Maximum axial deflection is 5.29mm against the available gap of 15mm between the flanges. Stress on the elastomer is very small as shown in figure 8. Actual damping provided by the elastomer can be estimated by doing a dynamic characterisation test.

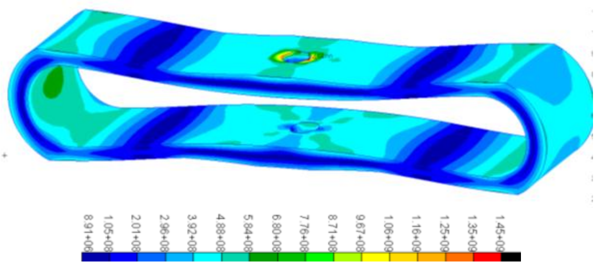


Fig -7: von Mises stress on metal part

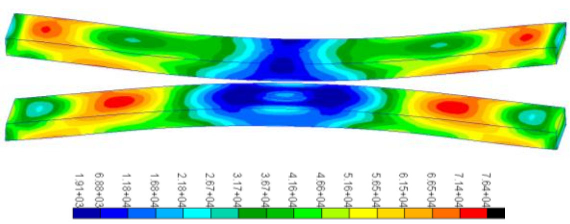


Fig -8: von Mises stress on elastomer

6. FREQUENCY RESPONSE STUDY

Direct frequency response analysis was carried for response characteristics of the isolator with payload. A payload of mass 1800kg is considered and 36 isolators are arranged at the interface between the payload adapter and the payload mass as shown in figure 9. Each isolator is simplified as CBUSH elements with simulated stiffness in axial direction 951kN/m. Damping ratio is assumed as 6% for each isolator. Here only the axial dynamics is studied hence the payload mass is lumped without any CG offset and connected using rigid links.

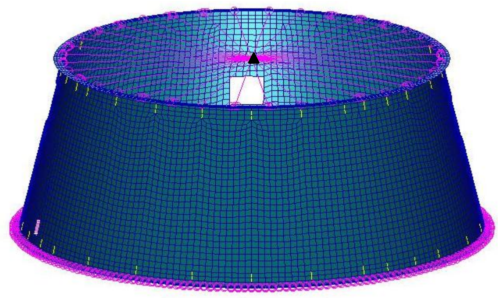


Fig -9: Finite element model of isolator with payload

An enforced input acceleration with respect to frequency is applied at the base in axial direction. Mode shape is shown in figure 10.

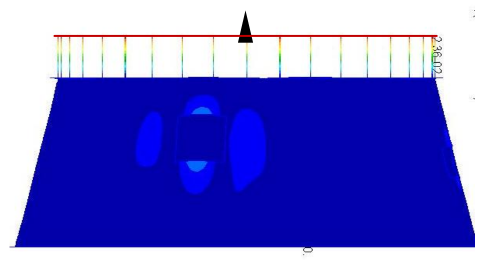


Fig -10: Axial mode shape

Input acceleration is $10 m/s^2$ and the peak response, at natural frequency, is $73.4 m/s^2$. Acceleration with respect to frequency upto 100Hz is given in chart-4. Transmissibility can be estimated as;

$$\text{Transmissibility} = \frac{A_{out}}{A_{in}} = 7.34$$

Amplification is less than 1.0 after 31Hz on the obtained transmissibility curve. So the vibration isolation is effective after 31Hz for the considered case.

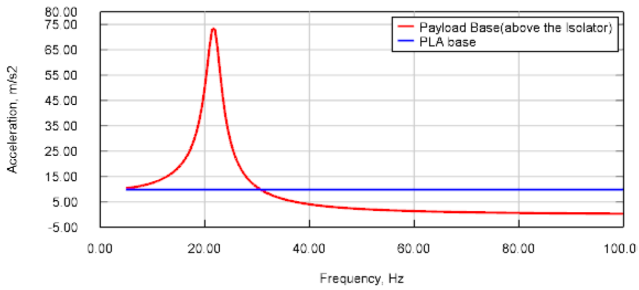


Chart -4: Acceleration plot at the base (input) and above the isolators

7. CONCLUSION

Various types of elastomeric isolators with different rated load are currently used in Launch vehicle subassemblies. This paper describes design of new isolator for Launch Vehicle applications, with metal as the main load bearing member. A box shape configuration is adopted and Ti alloy was found to be suitable material for this because of the high strength requirement. Silicone rubber material with thin metallic constrained layer is bonded to the metallic surface to obtain the damping property. Analysis results show that the isolator is effective against axial excitation after 31Hz.

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