DESIGN AND DEVELOPMENT OF METALLIC VIBRATION ISOLATORS FOR AIR BORNE VEHICLES

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by

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DEDICATION

I wish to dedicate this dissertation to my beloved father (Late) Shri Pillarisetty Raghava Rao (Baburao) and my beloved mother Smt. P Ahalya (Choti) and offer this work at the lotus feet of the Universal Spirit.

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Date: 2018 **P Anil Kumar**

ABSTRACT

Electronic packages meant for executing key roles like control, guidance, navigation, etc. in air borne vehicles will experience random vibration during flight. This vibration gets further transmitted to internal subsystems like Printed Circuit Boards (PCBs) with amplification. If the amplified vibration response on PCB exceeds the qualification limit, then the electronic components mounted on PCB will either malfunction or lead to catastrophic mission failure.

Mounting configuration of PCBs in chassis greatly influence the amplification factor. Stacked mounting and wedge guide mounting are two basic configurations of electronic packages. During qualification testing of packages in both the configurations, very high vibration responses are observed. Thorough analysis of this problem conveyed a message that the rigid mounting condition of PCBs with respect to chassis is the chief contributing factor to the high vibration response.

It is understood that the solution for reducing this high vibration response is to relax the rigidity between PCBs and chassis using vibration isolators. Commercially available vibration isolators which are made of rubber are not fit for electronic packages as shelf life of rubber is not on par with anticipated storage time of air borne vehicles and also due to the environmental sensitivity of rubber material.

Keeping these issues into consideration, an attempt is made in this research work to design, develop and experimentally evaluate metalic spring isolators for electronic packages in stacked mounting configuration and metalic C-isolators for electronic packages in wedge guide mounting configuration.

To begin with design criteria for effective isolation is evolved considering methodology identified from literature as a reference. Design constraints are identified and the associated governing relations are brought out. Subsequently isolators are designed and their dimensions are worked out.

Further Graphical Use Interface (GUI) based software is developed in MATLAB for design of spring isolator. The necessary formulation, which is derived as part of design, is converted in form of code. The intention behind developing the software is to enable the designer

to use it as a hand calculator, which just takes the inputs from the user and gives the output quickly.

Design is validated by comparing the stiffness values obtained using calculations with that of Finite Element Analysis (FEA). Subsequently isolators are manufactured with dimensions evolved from the design.

In the later stage these isolators are experimentally evaluated in random vibration environment and isolation effectiveness is quantified. Both the isolators are found to be very effective in reducing vibration levels in their respective electronic packaging configurations. Efficacy of the isolators thus developed is methodically tested in many real life packages and there by consistency in the methodology is established.

Keywords: Mounting configuration, Isolators, Experimental evaluation.

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NOMENCLATURE

B	Magnetic flux density
C	Spring index
c	Damping coefficient
c_c	Critical damping coefficient
C_l	Factor depending on ratio between static deflection and free length and ratio
D	Mean coil diameter of spring isolator
d	Size of wire
E	Young’s modulus
F	Force
f_n	Natural frequency
G	Shear modulus
h_s	Working height of spring isolator
I	Moment of inertia
i	current
K	Stiffness
K_E	Endurance factor
K_{lat}	Stiffness in lateral direction-X
K_{S1}	Minimum stress factor
K_{S2}	Maximum stress factor
K_U	Ultimate strength factor
K_{vert}	Stiffness in axial (Vertical-Y) direction
l_o	Free length
M	Bending moment
m	Mass
n^1	Total number of turns
R	Radius of curvature
r_x	X-component of position vector from a point on the section to the point of force application
r_y	Y-component of position vector from a point on the section to the point of force application
r_z	Z-component of position vector from a point on the section to the point of force Application

t	Thickness
w	Width
x	Vibration response
$\dot{x}$	Response vibration velocity
$\ddot{x}$	Response vibration acceleration
y	Vibration input
$\dot{y}$	Input vibration velocity
$\ddot{y}$	Input vibration acceleration
	between free length and mean coil diameter of spring isolator
σ_{Von}	Von Misses stress
θ	Angle
ω	Circular natural frequency of excitation
ω_n	Desired circular natural frequency of isolator
δ_{st}	Static deflection of spring
ρ	Density
τ_s	Nominal shear stress
τ_m	Mean shear stress
τ_v	Variable shear stress
τ_y	Yield strength
τ_e	Endurance limit
τ_U	Minimum ultimate tensile strength
τ_{min}	Minimum shear stress
μ	Poisson’s ratio
ζ	Damping factor

CHAPTER 1

INTRODUCTION

1.1 MOTIVATION

Air borne vehicles consist of different electronic packages for serving various functions like guidance, control, navigation, etc. During flight these packages experience aerodynamically induced loads like random vibration. These loads, gets further transmitted to internal subsystems like Printed Circuit Boards (PCBs). In general, vibration gets amplified while it is getting transmitted from chassis (Supporting structure for PCBs) to PCB. If the amplified vibration response on PCB exceeds the qualification limit, then the electronic components mounted on PCB will either malfunction or lead to catastrophic mission failure. Mounting configuration of PCBs in chassis greatly influence the amplification factor. Stacked mounting is one such configuration with which electronic package designers are comfortable due to ease of assembly and disassembly. In this configuration PCBs are mounted one over another and desired gap between any two adjacent PCBs and also between bottom most PCB and chassis is maintained with the help of special screws. During qualification testing of these packages with stacked mounting configuration, very high vibration responses are observed when vibration input is given in lateral direction i.e. in the plane of PCBs. Thorough analysis of this problem conveyed a message that the rigid mounting condition of PCBs with respect to chassis is resulting in a vibration mode (Like the typical first bending mode of a cantilever) which is responsible to the high vibration response. It is understood that, the solution for reducing this high vibration response lies in relaxing the rigidity between the stacked PCBs and the chassis by replacing screws in the bottom most row with vibration isolators [1]. Commercially available vibration isolators which are made of rubber are not fit for electronic packages as the shelf life of rubber is not on par with anticipated storage time of air borne vehicles and also due to the environmental sensitivity of rubber material. Keeping these constraints into consideration, it is proposed in this research project to design, develop and experimentally evaluate metallic spring isolators with an aim to replace the screws between bottom most PCB and the chassis in the present configuration. Wedge guide mounting is another configuration in which PCBs are inserted from the front side of the package along the wedge guides, provided integral to side walls of chassis. Wedge locks provided integral to PCB ensure the rigidity with respect to chassis wedge guides. During random vibration testing of the packages in wedge guide configuration, high vibration responses are observed at locations on PCB away from the wedge locks, when vibration input is given in vertical direction i.e. normal to the plane of PCBs. Accordingly, the entire space on PCB is not utilized effectively while populating with electronic components. Hence the study is extended for evolving another type of metallic isolator i.e. C-isolator for package in wedge guide configuration to replace wedge locks.

1.2 OBJECTIVE AND SCOPE

The objective of the present research work is to design, develop and experimentally evaluate metallic isolators for electronic packages for air borne vehicles with the aid of conventional experimental tools.

a) The first step to achieve this objective is to understand the behaviour of different packaging configurations considered.

b) The second step is to take the methodology identified from literature as a reference so as to evolve the design philosophy of vibration isolators bringing out the design criteria for effective isolation. Design constraints (If any) are to be identified and the associated governing relations are to be brought out which will aid as a tool for design check at final stage. Further, to carry out experimental evaluation of an electronic package with stacked mounting configuration in real time random vibration environment to capture frequency information from measured vibration response spectrum. Taking frequency as input, spring isolator is to be designed. To carry out design evaluation with the aid of Finite Element Analysis (FEA).

c) Fabrication of spring isolator and experimental evaluation of its isolation effectiveness.

d) Further, to support the work done, consistency studies are also to be carried out to gain confidence in the obtained results.

e) Repeating the same exercise for evolving other type of metallic isolator i.e. C-isolator for electronic package in wedge guide configuration. Confirmations with the consistency checks helps in developing a reasonable amount of confidence in the applicability of the design of the established isolator. Finally establishing consistencies with the database and the principles developed from ‘a’ to‘d’.

Thus, the scope of this research is to:

Design, develop and experimentally evaluate metallic isolators for two different types of electronic packaging configurations.

1.3 ORGANISATION OF THESIS

The thesis is organized as follows.

Chapter-1: Deals with introduction to problem statement, motivation and objectives.

Chapter-2: Deals with the literature survey on vibration isolators and information with regard to configuration aspects of various types of isolators.

Chapter-3: Deals with the constituents of an electronic package and problem definition.

Chapter-4: Deals with design of metallic spring isolator for packages in stacked configuration, details of software developed in MATLAB followed by design evaluation with the aid of FE analysis.

Chapter-5: Deals with experimental evaluation of isolation effectiveness of spring isolators. It also deals with real life packages as a case study for testing the efficacy of the developed isolator.

Chapter-6: Deals with design of metallic C-isolator for packages in wedge guide configuration followed by design evaluation with the aid of FE analysis.

Chapter-7: Deals with experimental evaluation of isolation effectiveness of C-isolators and consistency studies.

Chapter-8: Deals with summary and conclusions followed by future scope of work.

CHAPTER 2

LITERATURE SURVEY

2.1 INTRODUCTION

Electronic packages of air borne vehicles experiences high levels of random vibration during flight. Vibration gets further transmitted to internal subsystems like Printed Circuit Boards (PCBs) generally, with amplification. If the amplified vibration response on PCB exceeds the qualification limit, then the electronic components mounted on PCB will either malfunction or lead to catastrophic mission failure. Mounting configuration of PCBs in chassis greatly influence the amplification factor. Stacked mounting and wedge guide mounting are two basic configurations of electronic packages. During qualification testing of the packages in both the configurations, very high vibration responses are observed. Thorough analysis of this problem conveyed a message that the rigid mounting condition of PCBs with respect to chassis is the chief contributing factor to the high vibration response. It is understood that the solution for reducing this high vibration response is to relax the rigidity between PCBs and chassis using vibration isolators. Commercially available vibration isolators which are made of rubber are not fit for electronic packages as shelf life of rubber is limited and not suitable for the anticipated storage of air borne vehicles. The environmental sensitivity of rubber material is also a negative factor for rubber isolators. Keeping these constraints into consideration, it is proposed in this research project to design, develop and experimentally evaluate two different types of metallic isolators for electronic packages in stacked mounting and wedge guide mounting configurations. In this chapter, the published literature in the area of vibration isolators and the research work carried out by various investigators on design, development and experimental evaluation of vibration isolators is presented. The learning and their limitations are also brought out.

2.2 LITERATURE

Published literature on the various aspects of vibration isolators are presented below.

Lead-rubber isolators are recommended by **Zhao Wei (1992)** for isolation of structures against low frequency and large amplitude excitation. Benefit gained in terms of high damping by combining lead with rubber is mentioned. Cost effectiveness of the lead-rubber isolators over rubber isolators is brought out. Acceleration, relative displacement and strain are measured to quantify the isolation effectiveness of the proposed lead-rubber isolators. Analytical method to predict vibration response is proposed. Further, DOF of the mounted subsystem is also reduced to a greater extent in the analytical method. Addressing environmental sensitivity of rubber material (i.e. its stiffness and damping changes with change in ambient temperature) would have made the research work more effective. Quantifying outcome of research work in terms of isolation effectiveness obtained either from analytical work or experimental evaluation would have enabled better appreciation of the proposed concept. Same is the case with validation of analytical method with that of measurements.

Joseph Giaime et al (1996) brought out four stage vibration isolation system by stacking masses and springs alternatively. Masses are made of stainless steel and springs are made of elastomeric material. Vibration modes of stacking system is damped with the aid of viscoelastic characteristics of springs. Transmissibility is predicted with the help of 3D finite element models and verified with that of measurements. Proposed configuration enables compact masses to be embedded in the system avoiding subsystems with coupled vibration modes. As the intended system which needs to be isolated operates in vacuum conditions, care must be taken while selecting material for springs to avoid outgassing. FEM analysis is done in ABAQUS software in order to test for static stability and also to find out response of the system against dynamic excitation. This software enables to consider frequency dependent modulus of elasticity so as to predict the elastomeric behavior.

Reid J. R. et al (1998) developed a vibration isolation system for reducing the vibration sensitivity of surface resonators. The isolation system consists of a support structure for mounting the resonator, four support arms, and a support rim. Size of the isolation system 8 mm x 9 mm x 0.4 mm excluding resonator. The system acts as a passive vibration isolation system, attenuating the magnitude of high frequency vibrations i.e. > 2 KHz. Finite element analysis is used to analyze the acceleration response of the mounted resonator. The isolation system is modeled as a conventional damped mass-spring system and the isolation effectiveness is calculated. Multiplying the acceleration response of the resonator by the transmissibility results in the expected system vibration response. The isolation systems are fabricated using silicon wafers. Vibration response measurements are taken for excitation with frequencies ranging from 100 Hz to 5 kHz.

Mark G. Malowicki (2000) designed an active vibration isolator with smart materials. Hardware of same is realized and experimentally evaluated. Sensitivity of a human being for vibration is elaborated. Methodology for derivation of test specifications from properties of roughness of road to evaluate passenger seat is discussed. A 4DOF model is built in MATLAB to estimate actuator requirements. Configuration of active isolation system with incorporating piezo ceramic material is illustrated. Subsequently experimental results of the same for evaluating the performance are discussed. A vibration isolation system ~~for~~ test setup is realized with the weight of passengers and an actuator.

Sang-Myeong Kim et al (2001) discussed about active based four mount vibration isolation system. Theoretical aspects along with experimental evaluation of the same is mentioned. Electromagnetic based actuators are positioned in parallel with each other between mounted system and the base. Velocity feedback control is implemented, where autonomous operation of each actuator is executed by providing feed back of the velocity of associated equipment at the same location. Study of this actuator hardware and its practical application are the motivation of this work. Isolation of low-frequency vibration is taken into account where the mounted system can be modeled as a rigid body and the mounts as lumped-parameter system.

Clemens A.J. Beijers et al (2003) modelled cylindrical vibration isolator using Finite Element Method (FEM) in ABAQUS software package. Analysis is done in two steps. To Non-linear analysis is done with preload taken into account followed by linear harmonic analysis. Material model introduced by Yeoh is considered for analysis. Rubber portion of isolator is discretized with linear solid elements and steel portion is meshed with linear shell elements. The role of preloading in computing axial dynamic stiffness is discussed. Further it is stated that material behavior also gets influenced as a result of stiffening. It is shown that axial dynamic stiffness increases as a function of preload at low frequencies which ~~is~~ results due to increased cross section of the isolator. It is mentioned that for lower frequencies, dynamic stiffness is almost constant and on par with static stiffness. With the increase in frequency of excitation, effect of mass behavior of isolator also increases. Increase in dynamic stiffness is noticed when excitation frequency is in the close proximity of anti-resonance. Whereas dynamic stiffness becomes least when excitation frequency is in the vicinity of resonance. Within the frequency band of 200 to 300 Hz dynamic stiffness in lateral direction exceeds dynamic stiffness in axial direction. Material behavior of rubber makes its dynamic displacement complex. It is concluded that dynamic stiffness matrices can be used to estimate the performance of isolation system.

Mustafa E. Levent et al (2003) stated that design optimization of elastomeric vibration isolators is tedious due to dependence of their properties on frequency and temperature. It is mentioned that the experimental methods are very well available for estimation of properties of vibration isolators. But these methods need costlier test equipment like hydraulic actuators with controllers and transducers. To overcome this problem, two theoretical methods are presented for estimation of properties of isolators without need for expensive hardware. But temperature effects are ignored as all the measurements are taken at room temperature. However frequency effect is taken into account. Grommets (Miniature isolators) are considered for measurements. Limitation of the approach adopted in this work is that non linearity associated with amplitude cannot be quantified. Logarithmic decrement method is adopted for quantifying modal parameters (Natural frequency and damping). Properties thus estimated using approximate methods proposed in this work are compared with that of direct methods recommended by standards. It is observed that the proposed approximate methods yield exaggerated estimation of properties of isolators.

Complexity associated with design of isolator with respect to its stiffness is brought out by **Eugene I. Rivin (2006)**. In his work he mentions that stiffness of the isolator should be less for providing better isolation which is contradicted by need to have higher stiffness to support the mounted subsystem effectively. It is also mentioned that active based vibration isolators can overcome this problem. Mathematical model of an uni-axial vibration isolation is presented. Concept of Constant Natural Frequency (CNF) isolators is introduced. Four varieties of systems which needs vibration isolation are mentioned. However precision machine tools which need vibration isolation alone are dealt. Adverse effects of rocking motion which results due to less stiffness are discussed. Fast decay that occurs in floor vibration particularly in high frequency zone is brought out. Information from standards on anticipated vibration levels in precision machine tools are considered, rather than measuring actual vibration levels. But precise information is needed with respect to frequency and vibration level for designing an isolator effectively which is possible only through measurements. Sensitiveness of precision machinery against vibration in horizontal direction is highlighted. It is also mentioned that severity of vibration will be 10 times more in horizontal direction than that of vertical direction. Influence of CG mount in avoiding dynamic coupling is discussed. It is also mentioned that same benefit can be gained with inclined isolators when it is not practical to adopt CG mount.

Bruno de Marneffe (2007) brought out basic elements of active vibration isolation techniques. An alternative solution to these complex elements in the form of shunt circuits is proposed to be coupled to the sensor. Further, methods to ensure the stability of said shunted structure by inclusion of passive elements like resistors and inductors are suggested. Performance of these shunt devices is compared with that of conventional active vibration isolation systems. Vibration isolator made of electromagnetic sensor is presented. The effect of combining inductor with resistor in shunt is analyzed. Finally the details of six axis isolation system based on Stewart platform is discussed.

Collette et al (2010) provided compilation of various active vibration isolation methodologies to isolate precision machinery. Aim of this work is to evolve tools for identification of best approach for a given requirement. Various approaches are compared with the aid of single DOF models. However the study is limited to compilation of techniques but not application of same.

J. Byron Davis (2010) compiled failures in vibration isolators due to poor design. Need for thorough examination of hardware installation on isolators is highlighted. It is also mentioned that misalignment between the axes of the mounted system, isolator and eccentric loading can lead to amplification in vibration levels rather than isolate. Vibration isolation aspects are dealt with in the domain of force transmissibility in rotating machinery. Need for separation of frequency of isolator with frequency of mounted system during design is brought out. Primary reasons behind failures are stated to be improper design, wrong selection of isolator, deviations in manufacturing the isolator as per design and improper installation of hardware on isolators. Importance of avoiding contact between any accessory of the mounted system like piping and floor on isolation effectiveness is highlighted. It is also mentioned that improper weight distribution of mounted system on all isolators can also yield to failure. Issues with respect to misalignment associated with nested isolators is brought out. Influence of debris of the mounted system (If any) on fouling isolators is discussed. It is mentioned that unexpected stiffness addition in the isolation scheme can be revealed with the aid of testing. Case studies demonstrating failures are shown. However all the case studies are confined to rotating machinery.

Kuinian Li (2010) has developed three types of isolators namely lever damper isolator, lever spring isolator and lever spring mass isolator by adding lever mechanism to conventional vibration isolator. Transmissibility of these isolators is evaluated through analytical means. Performance of these isolators is studied with the aid of numerical simulation. Outcome of the research work has shown that performance of isolator with lever mechanism is better than that of conventional one. It is mentioned that stability and load carrying capacity of the isolator limits its stiffness towards lower side. Solutions to overcome this problem are discussed. The requirement of of high damping at resonance and a smaller damping at higher frequencies is discussed. To meet this details of a skyhook vibration isolator is mentioned. However skyhook vibration isolator is not practical in nature due to the requirement of a viscous damper. Mathematical models of the proposed isolators are presented from which transmissibility is estimated. Further effect of lever ratio on isolation effectiveness is discussed. With the addition of lever mechanism to conventional vibration isolator, it is shown that limitations associated with design of isolator are overcome.

Taksehi Mizuno (2010) proposed a design of vibration isolator with negative stiffness. It is mentioned that isolation effectiveness in case of base excitation would be high if the stiffness of isolator is less and zero stiffness would be ideal. But for the case of direct excitation i.e. mass excitation, isolator having high stiffness would be more apt. Concept of yielding high effective stiffness by combining one spring having normal stiffness with other spring having negative stiffness is presented. It is stated that negative stiffness is made possible with the aid of an actuator. But for experimental evaluation of negative stiffness isolators demands ~~for~~ use of displacement sensors which will be heavy when compared to accelerometers. However high power actuators would be needed to suspend if the weight of mounted system goes high which will have cost implication. Controller design aspects needed to achieve negative stiffness isolator are mentioned. Idea of pushing the suspended mass in a direction opposite to that of applied force using magnetic means is discussed. Hardware is realized and its functionality is demonstrated. In the later stage this concept is extended to six axis vibration isolation system. Another type of isolator is evolved by replacing electromagnet with pneumatic actuator for providing negative stiffness. However external power supply is needed to use this isolator.

Yann Frizenschaf et al (2011) has presented a conceptual design for a levitating magnetic vibration isolation device, objective of which is to yield a load-independent resonance frequency across a wide range of masses of mounted subsystems with the aid of inclined magnetic springs. Static and dynamic analyses are performed using FEM for validating the assumptions. Challenges pertaining to the contradictory design requirements of stability, high damping and force independent resonance frequencies are answered. Test set up for testing the prototype model is presented. Analytical modelling brought out that achieving load independent resonance frequency is possible for a vibration isolator consisting of inclined cuboid permanent magnet springs.

D. Kamesh et al (2012) designed a low frequency isolator consisting of folded continuous beams for isolation of vibration in space crafts. As per the design criteria, isolator has to support a payload of 5.5–6.5 kg with isolation frequency within 15 Hz. Both analytical and experimental studies are carried out to evaluate the isolation effectiveness. Simulations are carried out using MATLAB to find out the optimum geometry of the proposed low frequency isolator. Measurements are carried out at different wheel speeds in order to figure out the isolation possible from the reaction wheel borne vibration. Measurements are extended further to other inversions of the proposed design so as to identify the best isolation scheme against given excitation. Analysis of measured data indicates that proposed isolator is very effective for isolation of vibration in space crafts.

Isolator configuration based on magnetic levitation is discussed by **Tao Zhu (2013)**. Influence of trade-off that exists between load carrying capacity and isolation effectiveness is mentioned. Variation in stiffness in all six Degrees of Freedom (DOF) is brought out. To represent the model of the mounted system, the stiffness is enhanced in all DOF's except along the axis of the isolator. The said isolator is configured based on active vibration isolation method. Proposed magnetic levitation system enables the isolator to facilitate quasi-zero stiffness in axial direction and almost zero stiffness in the rest five directions. It is stated that static forces and moments which normally arise in conventional isolators will be balanced with the aid of permanent magnets. The author has derived numerical models for estimating forces and torques. These models are used to analyze cross-coupling effects that exists between any two DOF which are orthogonal. A mechanism is proposed to use the unwanted cross-coupling effects for balancing the loads. Apart from this, the design aspects of the said isolator is discussed. Proposed isolator is realized and also the active control system necessary to drive the isolator is ~~also~~ developed. Laser based sensors are used in instrumentation scheme. Noise issues associated with laser sensors are investigated. However solutions are not implemented to avoid two resonant modes which are claimed to degrade the desired functionality of the mounted system. Some issues which are unaddressed are however, the presence of so many un-damped peaks in the response spectrum with proposed isolator brings out the need to improve damping. Transmissibility is used as means to quantify the isolation effectiveness which is more suitable to harmonic excitation. But real life aerospace involves, broad band random excitation exists which calls for the need of quantifying absolute acceleration response in terms of Power Spectral Density (PSD).

Lazutkin G.V et al (2014) compared load carrying capability, damping, and shelf life, dimensional and frequency properties of vibration isolators made of two different metallic materials (MR and spring cushion). Load carrying capability and damping of earlier one are superior to later one. From the study, shelf life of both the materials are found to be identical. Both of these isolators are made of wire pressed materials. It is stated that manufacturing of MR material is laborious than that of spring cushion.

R. Boonen et al (2014) has developed a vibration isolator with coulomb friction which is configured with shaped spring wire whose ends will slip in their respective mountings. By locating these elements at an angle of 90^0 around the isolation element, the isolation element works in all six degrees of freedom. The isolator is modelled as a combination of a spring with a dry friction damper, which in turn is oriented in parallel with a second spring. This model is analysed with the help of Hamilton's equations of motion to study the non-linear behavior. A test setup is developed using which isolator is validated with the help of impact hammer excitation. Proposed vibration isolator is claimed to be environmentally insensitive. Spring constants are estimated using FEM.

Xiuting Sun (2014) has studied the influence of dry friction and nonlinearity in a vibration isolation system. Main focus is on design and analysis of nonlinear parameters like stiffness, frictional force and damping factor to attain an effective vibration isolation. Dry friction exists in various joints due to elements like springs, connecting rods, etc. Performance of vibration isolation system is studied to check the influence of various structural parameters on frictional forces. The intended system, takes advantage of nonlinear stiffness and damping parameters in reducing vibration. It is stated that sensitive frequency range of vibration for a human being seated in automobile vehicle is 4 Hz to 9 Hz. It is also mentioned that improving isolation effectiveness by varying stiffness and damping is more practical as they can be varied by changing the structural parameters.

A novel design of vibration isolator using voice coil motor actuator is proposed by **Dong-Pyo Hong (2015)**. Proposed isolator was configured to generate an adequate force for isolation of vibration. Magnetic flux is modeled and the force is estimated. Merits of hybrid active passive isolator over conventional isolator are discussed. Simulation is carried out with the aid of Finite

Element Method (FEM) and the results thus obtained are verified with that of test results. Further FEM is used in tandem with MATLAB for simulation. Proposed isolator is specifically targeted for isolation of precision measuring equipment. However, proposed isolator may not be effective against acoustic noise. Magnetic flux density is of the order of 0.14 Tesla. Flow of electric current in the motor is also obtained. Performance of the isolator is estimated in terms of transfer function between input voltage and vibration response. Velocity is considered as a parameter for quantifying vibration response however, acceleration is a better parameter for evaluation of isolation effectiveness. Frequency band width considered for this study is limited to 200 Hz only. Sine test is conducted to study the performance of the isolator whereas, random vibration would have simulated real life conditions.

Robbert Voorhoeve et al (2015) has brought out an active vibration isolator for industrial applications. It is made of an aluminum table suspended on pneumatic mounts. It is aided with eight actuators, two for each corner, and six sensors, also two for each corner away from one. These actuators are identified by linear relation between smeared current and the output force. Geophones are used as sensors which are basically inertial sensors and configured to measure the absolute velocity of the system with reference to an inertial frame. Measurements are carried out with both sine and white noise excitations.

Run-pu Li (2015) introduced magneto rheological based elastomeric vibration isolator. Stiffness and damping of this isolator can be varied by varying magnetic field. Further a hybrid design is proposed by combining this isolator with negative stiffness isolator to increase the isolation effectiveness significantly. Relation between negative stiffness and effectiveness is mathematically established. Vibration response of the system is studied for simple harmonic disturbance with less amplitude along with the influence of composition of iron particles on isolation characteristics. The study concluded that lesser the composition of iron particles better is the performance of isolator. Mechanical properties like Young’s modulus, shear modulus and damping factor can be altered unlike for conventional isolators. But the variation needs to be quantified before realizing the hardware. Stiffness of the isolator is function of magnetic flux density of the magneto rheological fluid. It is stated that composition of iron particles influences the frequency of isolator. It is also mentioned that with increase in magnetic strength, frequency of isolator reduces and damping factor increases. Studies are carried out up to frequency of 30 Hz only. Extending the study beyond this frequency makes the proposed isolator to suit to practical applications.

Junyi Lee (2016) has identified that phononic crystals and elastic meta-materials are more effective in vibration isolation among various techniques. It is mentioned that these materials have regular variations in geometry and material properties and has the capability to isolate vibration levels over a wide band of frequencies. These materials isolate vibration through the band gap scheme which inhibits propagation of elastic waves. Apart from this stiffness to weight ratio and strength to density ratio of these regular structures are very high which enables them to yield light weight structures. Objective of the work carried out is to evolve techniques to encourage their utility. A test is carried out on a particular design to demonstrate the vibration isolation characteristics of this material. Work carried out lead to—design of vibration isolation system with light weight materials.

2.3 SUMMARY

Extensive literature survey carried out in this particular context brought out that most of the research carried out so far is confined to active vibration isolation systems only. But active isolation systems are not fit for air borne applications as they demand electrical power. Moreover these systems are effective for isolation of low frequency vibration only. In order to isolate high frequency vibration, reaction time of feedback sensor and controller needs to be very less which cannot be met by active vibration isolation systems in use. Further, no literature brings out information on development of metallic isolators for electronic packages and hence designers are left with no choice except using conventional rubber isolators, with its inherent limitations.

CHAPTER 3

ELECTRONIC PACKAGES AND ASSOCIATED VIBRATION ISSUES

3.1 INTRODUCTION

Electronic Packaging is “the technology of packaging electronic equipment”, which includes the interconnection of electronic components into Printed Circuit Boards (PCBs) and the interconnection of PCBs to electronic assemblies. Because of the increased use of computers and electronics in all aspects of flight vehicle development programmes, increasing performance of electronic packaging configurations without increasing the cost is becoming a major thrust for the electronic packaging designers. This increased use of electronics in conjunction with reduced size and weight and the designer’s need to introduce products rapidly has led to increased use of analysis instead of build and test to determine if equipment can meet the appropriate requirements. Electronic packages are considered to be mission critical in any air borne vehicle. During their course of application these packages are subjected to various harsh environments like random vibration. If the vibration levels experienced by these packages exceed their qualification limits functional failure may happen. An isolator helps in reducing the vibration levels which is the topic of this research work. However, if we need to get rid of problems associated with commercially available rubber isolators, it is essential to use metallic isolators to ensure proper functioning of the packages for a successful mission.

3.2 PROBLEM DEFINITION

A typical electronic package consists of chassis which houses PCBs on which critical electronic components like ICs are mounted as shown in Figure 3.1.

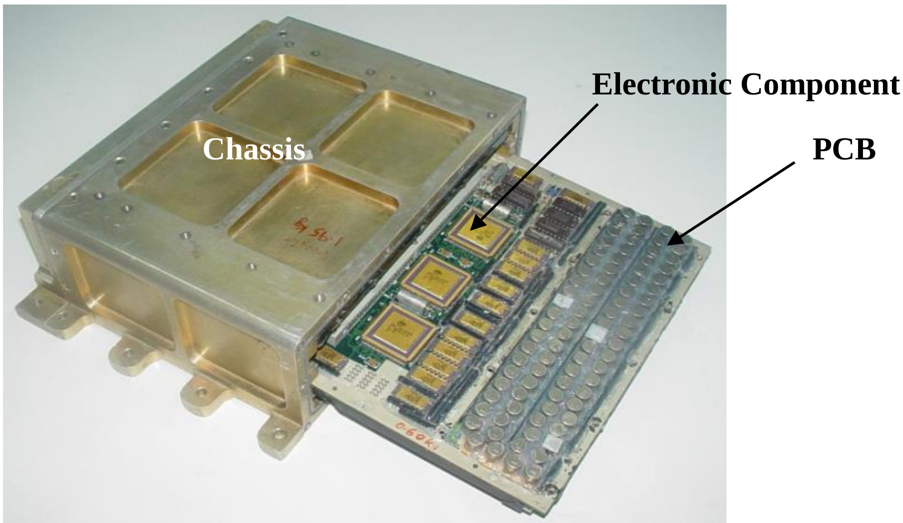


Figure 3.1 Typical electronic package

The mounting configuration of PCBs with respect to chassis is one factor which governs the vibration levels that are experienced by the electronic components mounted on PCBs. Associated standard for electronic packaging is MIL-M-38510E.

In general two types of mounting configurations are being used for PCBs. They are shown in Figure 3.2.

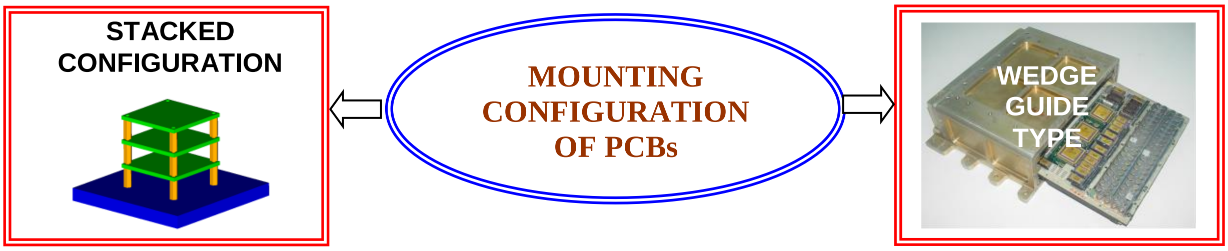


Figure 3.2 Two types of mounting configurations

3.2.1 STACKED CONFIGURATION

In this configuration PCBs are mounted one over another in stacked fashion and screws are used to connect adjacent PCBs and bottom PCB with chassis as shown in Figure 3.3.

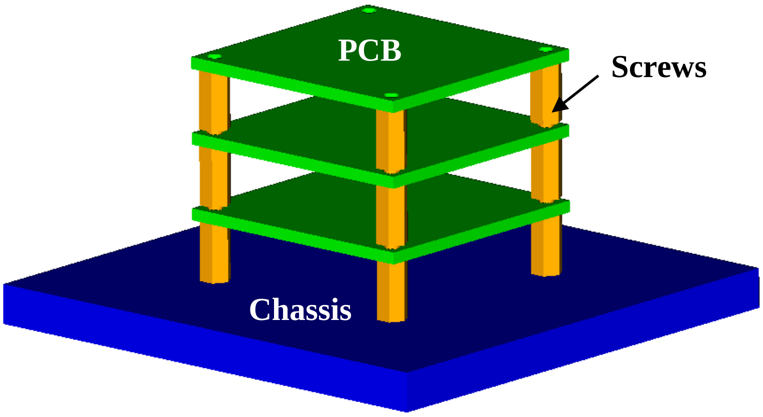


Figure 3.3 Stacked configuration

It is not recommended to mount PCBs in stacked configuration as it imparts cantilever (One end fixed and other end free) type bending modes which causes high vibration levels. Typical cantilever mode which commonly appears in stacked configuration PCBs [1] is shown in Figure 3.4.

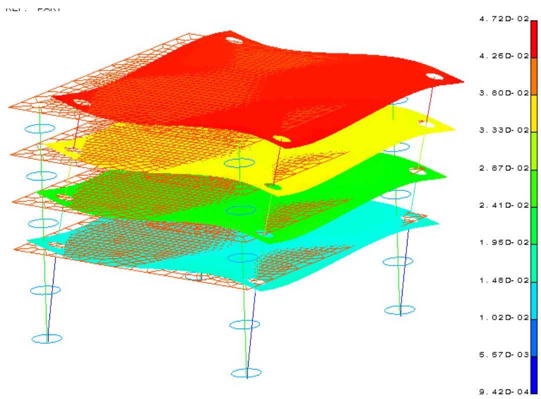


Figure 3.4 Typical cantilever mode in stacked configuration

This mode further results in high vibration response as shown in Figure 3.5.

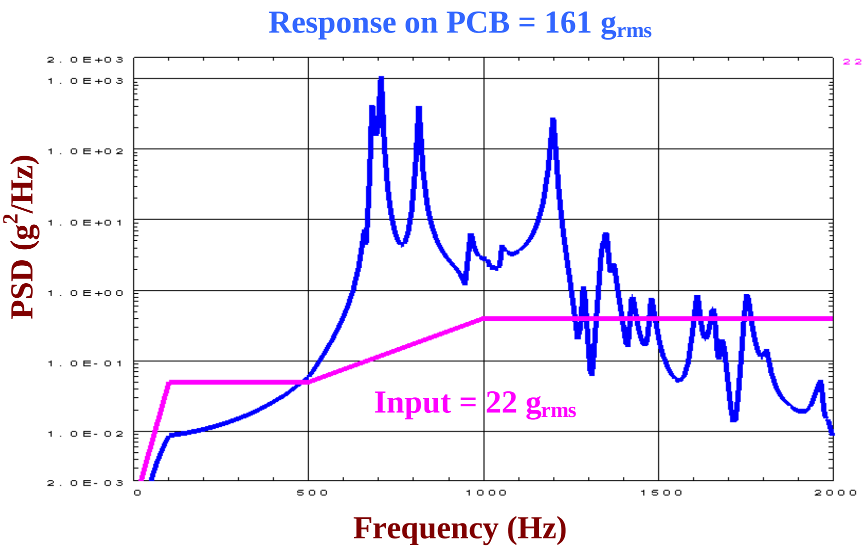


Figure 3.5 Vibration response with stacked configuration

- Due to this high vibration response i.e. 161 g_{rms}, components mounted on PCBs either malfunction or lead to catastrophic failure of mission (Acceptable limit is 30 g_{rms}).

3.2.2 WEDGE GUIDE CONFIGURATION

In this configuration rectangular shaped guides known as wedge guides are provided integral to the side walls of the chassis as in Figure 3.6.

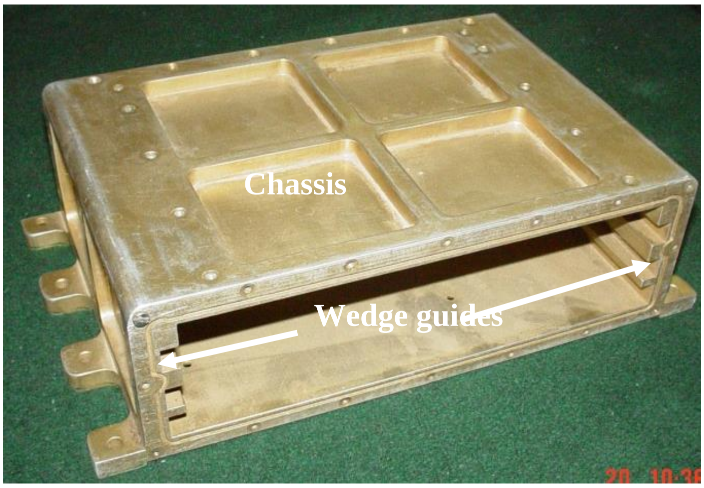


Figure 3.6 Wedge guide configuration

PCBs having wedge locks provided integral to them are inserted from front side along the chassis wedge guides. Once the PCB is fully inserted wedge lock is tightened with which part of the wedge lock remains in contact with chassis wedge guides and rest remains in contact with PCB. PCBs having male connectors are connected to female connectors provided on mother board located in rear side of the chassis. Typical wedge lock is shown in Figure 3.7.

Figure 3.7 Typical wedge lock

Top side of PCB (Components side) is shown in Figure 3.8.

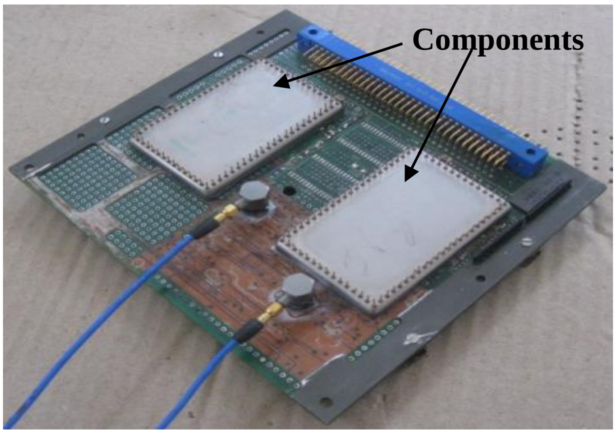


Figure 3.8 Top side of PCB (Components side)

Further bottom side of PCB (Wedge lock side) is shown in Figure 3.9.

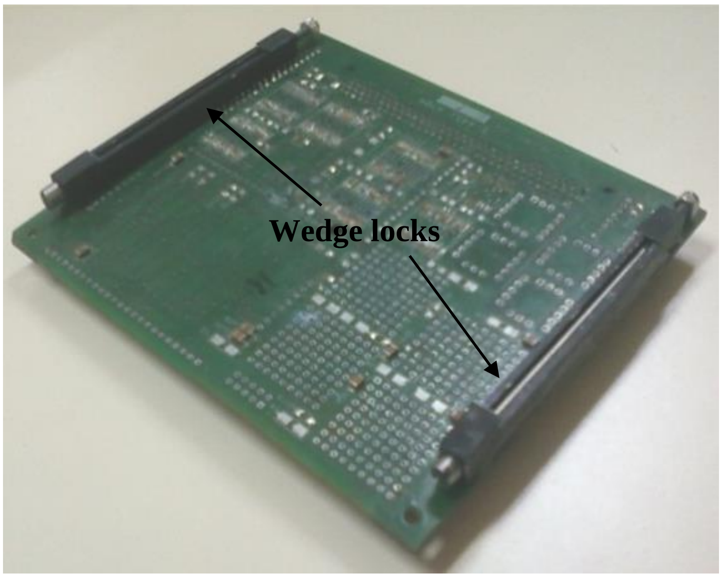


Figure 3.9 Bottom side of PCB (Wedge lock side)

During vibration, three parts of wedge lock are set in wedging action which results in energy dissipation (Damping) across the wedge locks. Hence vibration levels measured on the PCB within the vicinity of wedge locks are within the allowable limits (i.e. < 30 g_{rms}). However vibration levels measured on PCB away from wedge locks are very high. Hence very little space is permitted for mounting components on a wedge lock mounted PCB which is not in favour of the designer. Ideal choice for the designer would be utilization of entire space on PCB for mounting components which seeks for design modification.

Hence it is proposed to design, develop and experimentally evaluate metallic isolators for two different types of electronic packaging configurations.

3.3 VIBRATION ISOLATOR

Vibration isolator is an isolation mount that reduce the transmission of energy from one body to another by providing a resilient connection between them. The concept of vibration isolation is illustrated by consideration of the Single Degree of Freedom (SDOF) system in Figure 3.10.

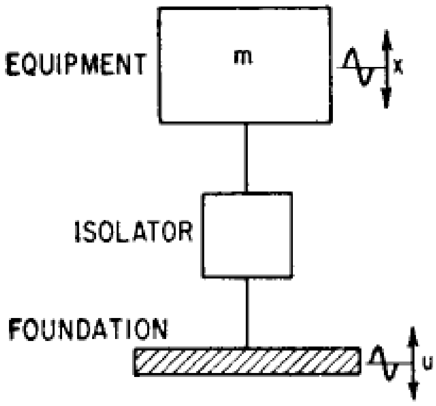


Figure 3.10 Isolator represented by SDOF system

This system consists of an equipment (Rigid body) mounted on a moving foundation through isolator. If the isolator is configured to have a specific frequency relation (In turn stiffness) with the mounted system it attenuates the motion that gets transmitted from foundation. Incorrect mount may reduce the high frequency vibration, but resonant conditions can actually amplify the induced vibration. Or else introducing an improper isolation mount may worsen the situation. Apart from stiffness, damping of an isolator also will have significant influence on its isolation characteristics. Adding damping to a resilient mount greatly improves its performance. Damping enables isolator to reduce the amplitude of resonant vibration by converting a portion of the energy into low-grade heat. Damping also dissipates shock energy during impact.

As the frequency continues to increase above the crossover frequency, the level of isolation, or the isolation efficiency increases as shown in Figure 3.11.

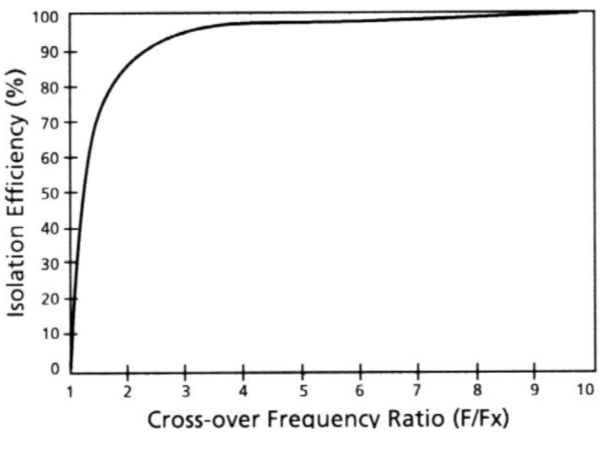


Figure 3.11 Variation of isolation efficiency with frequency ratio

The designer must know the isolation efficiency of the mounting system when transferred energy must be below a specified level.

3.4 FACTORS TO BE CONSIDERED WHILE SELECTING ISOLATORS

3.4.1 PHYSICAL DATA

- Equipment weight
- C.G. location relative to mounting points
- Sway space
- Maximum mounting size
- Equipment and support structure resonance frequencies
- Moment of inertia through C.G. for major axes
- Fail-safe installation required

3.4.2 DYNAMIC DATA

- Vibration requirements
    - Sinusoidal – Sweep rate, duration and magnitude
    - Random input – Duration and magnitude
- Resonance dwell – input and duration
- Shock requirements
    - Pulse shape, pulse period, amplitude, number of shocks per axis
- Sustained acceleration – Magnitude and direction
- Vibration fragility envelope (Maximum vibration level vs frequency preferred)
    - Or desired natural frequency and maximum. transmissibility
- Matched mount required

3.4.3 ENVIRONMENTAL DATA

- Temperature – Operating and non-operating
- Salt spray, humidity, Sand and dust, mould growth, Oil and/or gas
- Special finishes on components
- For critical applications – Gyros, optics and radars the requirements for control of angular motion of the isolated equipment are requested
- C.G. plane mounting to decouple translation and rotation
- Mounting the isolators at 45^0 to horizontal if C.G. plane mounting is not possible
- Dynamic matching of isolators to avoid angular errors

Typical schematic of isolator along with some of above mentioned parameters is shown in Figure 3.12.

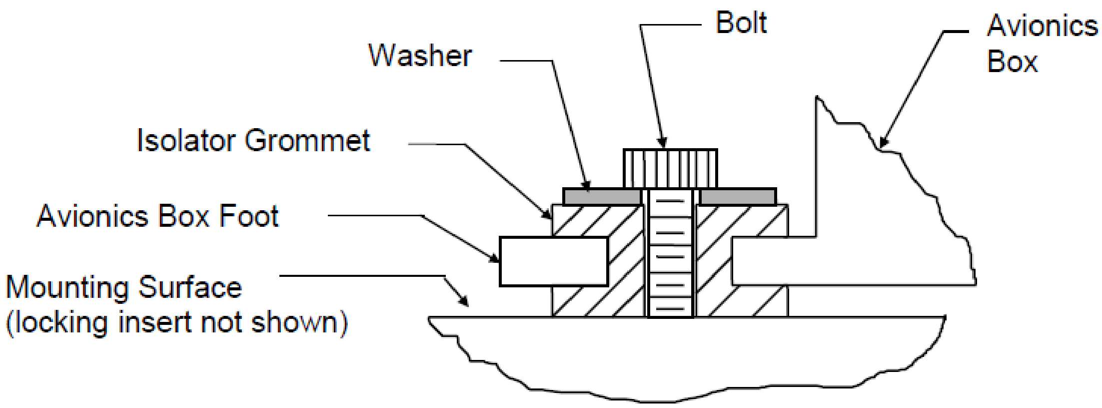


Figure 3.12 Typical schematic of isolator

3.5 TYPES OF ISOLATORS

3.5.1 ELASTOMERIC VIBRATION ISOLATOR

An elastomeric vibration isolator [7] is shown in Figure 3.13.

Figure 3.13 Elastomeric vibration isolator

With this type of isolator, by varying the configuration of the molded product, and by properly locating rigid steel components, an infinite range of stiffness can be achieved in every direction. Different formulations can provide varying amounts of damping, which can affect vibration isolation. There are numerous families of elastomer types which can be used based on the intended use and environment. Operating temperature and fluid exposure are two critical environmental parameters which should be considered in selecting the elastomer types.

3.5.2 COILING SPRING MOUNT

Typical coiling spring mount [7] is shown in Figure 3.14.

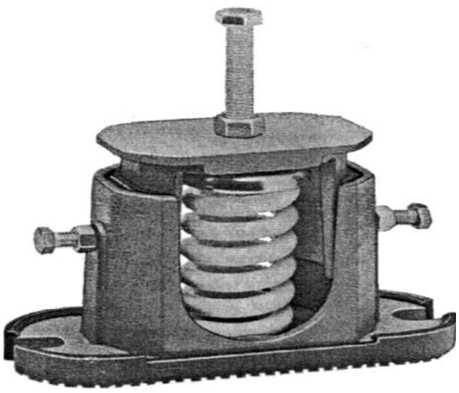


Figure 3.14 Typical coiling spring mount

For applications requiring a lower natural frequency than can be obtained with elastomer isolators, a frequent isolation choice is a coil spring mount. Features required in a coil spring mount are

- To minimize the effect of surge frequency, and to isolate high frequency vibrations, elastomer pad should be placed between the spring and the support structure
- If there are significant uplift forces the springs should be housed with an integral restraint mechanism

However springs have very little inherent damping. If damping is desired, standard designs are available which incorporate friction damping, hysteretic damping or viscous damping.

3.5.3 PNEUMATIC ISOLATOR

Typical pneumatic isolator [7] is shown in Figure 3.15.

Figure 3.15 Pneumatic isolator

Item which is sensitive to low frequencies, the logical choice is a pneumatic isolation system which can provide natural frequencies as low as 0.5 Hz. Pneumatic isolation system uses a gas, such as air, to provide vibration isolation. Pneumatic isolators do not require a large static deflection, because the gas can be compressed to support the load while providing the low stiffness necessary to provide vibration isolation. One advantage of the pneumatic isolation system is the large load carrying capacity for a given low natural frequency. Softer systems are obtained by varying the volume of the chamber and the effective area of the diaphragm.

3.5.4 WIRE ROPE ISOLATOR

Typical wire rope isolator [7] is shown in Figure 3.16.

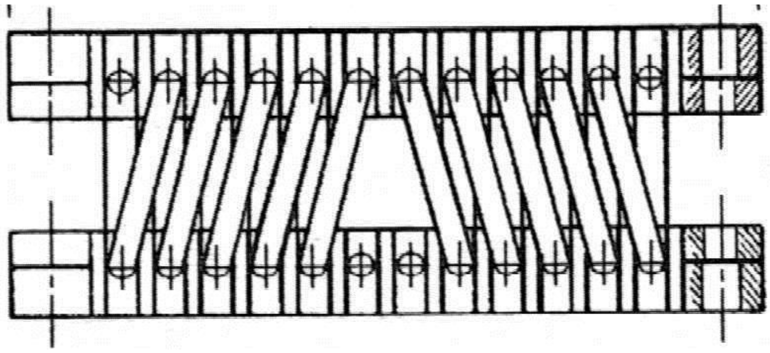


Figure 3.16 Wire rope isolator

Advantages of wire rope isolator are given below.

- Large deformation capability for shock absorption purposes and the unusually good vibration damping caused by the friction between the individual wires when the cables deform.
- The usable deflection is very large relative to the geometrical dimensions, hence small space requirements.
- Low natural frequencies down to about 2.5 Hz are possible.
- Can be loaded in all attitudes.
- Outstanding resistance to environmental effects.
- Long life due to the high aging resistance.
- No maintenance required.
- The characteristics unaltered at extreme temperatures.

- No adverse effects on the function and damping properties due to oil or pollution or varying temperatures.

3.6 GOVERNING FACTORS FOR ISOLATOR PERFORMANCE

3.6.1 SUPPORT STRUCTURE STIFFNESS

- For an isolator to perform its function, it must be able to deflect.
- If the support structure, and/or the supported structure are relatively soft, they will deflect rather than the isolators, and the isolators will not function as intended.
- Besides obtaining inadequate isolation, this may result in fatigue of the structure.
- The common rule of thumb is that the structures should have a stiffness ten times that of the isolators. This will ensure that the isolators are providing 90% of the required deflection.

3.6.2 STRUCTURAL RESONANCES

- Every structure has numerous frequencies at which it will resonate.
- The frequencies at which the parts of the structure vibrate are referred to as structural resonances, and are function of the material, dimensions, shape and end conditions (Method of support).
- On a transmissibility curve, they could appear as individual sharp peaks or as a broad region of numerous peaks, resulting in transmissibility higher than would be predicted based on theory.
- Stiffening the structure may help, but more often the selective application of damping materials seems more effective

3.6.3 ROCKING MODES

- For horizontal excitation of an isolation system, the possible modes of vibration are a axial mode and a pitch mode.

- These modes are said to be coupled when a vibrating force at the frequency of one mode causes vibrations to occur at the frequency in the other mode.
- Rocking modes decrease the efficiency of an isolation system, and hence to improve isolation, the system must be decoupled.
- This can be accomplished by locating isolators on the same horizontal plane as the center of gravity of the isolated mass.
- A second method of decoupling is to localize the isolation system to project the axis of isolator to the axis of center of gravity.

3.7 OBSERVABLE PHENOMENA DURING VIBRATION TESTING

3.7.1 WAVE EFFECTS

Wave effects (Shown in Figure 3.17) may be observed at high frequencies when the mount dimensions become comparable with multiples of the half wavelength of the elastic wave passing through the mounting.

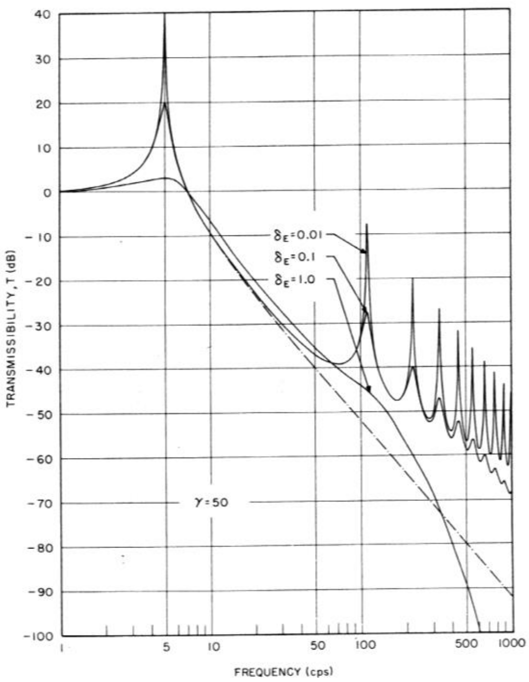


Figure 3.17 Wave effects of isolators

It is improbable, however, that wave resonances will appear in the transmissibility curves of mounts that are heavily damped. The geometry of rubber components of anti-vibration mountings is frequently complex which makes precise theoretical calculations of transmissibility difficult at high frequencies. However, by examining the transmissibility of mountings that obey simple wave equation for the longitudinal vibration of rod of uniform cross section, the characteristics of wave effects in anti-vibration mountings may be obtained. The vibration mounts of significant lateral dimensions will be determined from a corrected wave equation given by **Love**. In **Love** theory the radial motion of the plane cross sections of the mount caused by axial compression and extension is some measure accounted for.

3.7.2 JUMP PHENOMENA

For system with hardening type characteristics as the frequency is gradually increased from zero, the amplitude response also increases, until at the point where the response curve has a vertical tangent and the amplitude jumps to a lower value as shown in Figure 3.18.

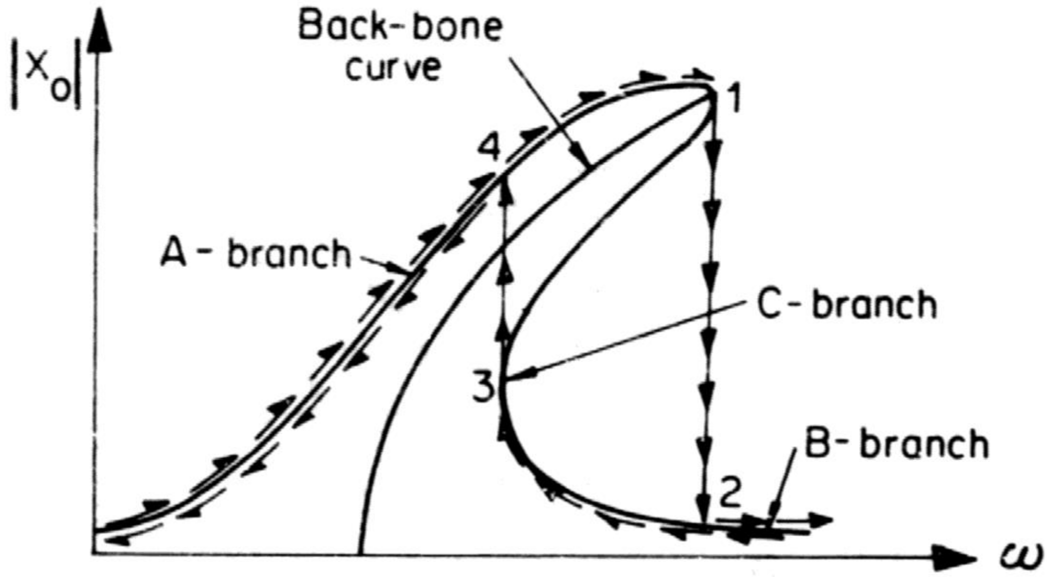


Figure 3.18 Jump phenomena of isolators

This jump in amplitude is accompanied by a phase jump through 180°. For further increase in frequency, the absolute value of the amplitude decreases as in the linear system. If the frequency is gradually decreased through values below ω_2, the amplitude gradually increases until at a point such as 3, the amplitude jumps to a higher value, while the motion changes from anti-phase to in-phase and the amplitude decreases when the frequency is decreased below ω_3. For systems with softening type spring characteristics similar jump occurs shown, but the direction of jump will be just the reverse of that for hardening type characteristics.

3.7.3 PARAMETRIC RESONANCES

By periodically changing the parameter defining the natural frequency the amplitude of vibration increases and is known as parametric resonance. Example for this is the length of a pendulum can be varied periodically, increasing the amplitude at the extreme end and decreasing it at the equilibrium position. The force of tension in the string decreases at its extreme ends and is greater at the equilibrium position when its velocity is highest. The work done by the pendulum at its extreme ends is smaller than the work done on the pendulum at its equilibrium position. There is a net transfer of energy to the pendulum due to which amplitude increases gradually.

3.8 SUMMARY

Details of two basic configurations of electronic packages are brought out along with associated vibration problems. Further, problem definition pertaining to the present research work is also discussed. Configuration details of various types of isolators are presented. Factors governing the isolator performance are mentioned. Further observable phenomena during vibration testing are discussed.

CHAPTER 4

DESIGN OF METALLIC SPRING ISOLATOR FOR PACKAGES IN STACKED CONFIGURATION

4.1 INTRODUCTION

In the previous chapter, two standard mounting configurations of electronic packages along with associated vibration problems are discussed elaborately. This chapter brings out the solution for vibration problem in stacked mounting configuration in form of spring isolator. Design philosophy stating the criteria for effective vibration isolation is discussed. Steps in design are elaborated in detail and outcome of design is summarized. Further Graphical User Interface (GUI) based software is developed in MATLAB for performing design calculations. Design is validated with the aid of Finite Element Analysis (FEA) by comparing stiffness obtained using design with that of FEA. This chapter also brings out the FE modelling aspects along with analysis results.

4.2 DESIGN CONFIGURATION

In the stacked configuration (Figure 4.1), screws in the bottom row i.e. between bottom most Printed Circuit Board (PCB) and chassis are replaced with spring isolators in proposed configuration (Figure 4.2). However it is planned to configure the mounting arrangement of spring isolator identical to that of screws. Accordingly ease of assembly and disassembly associated with stacked configuration is retained.

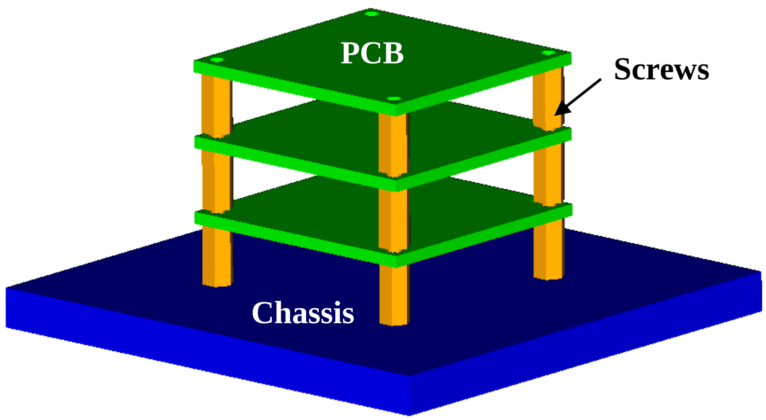


Figure 4.1 Configuration with rigid screws

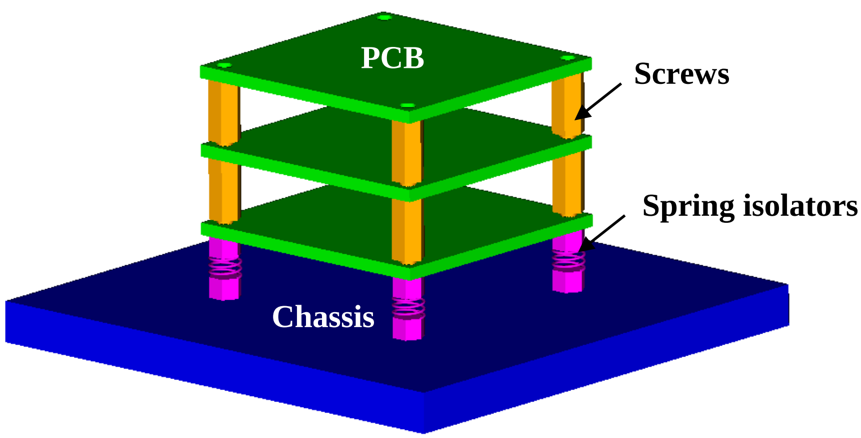


Figure 4.2 Configuration with spring isolators

Proposed configuration with spring isolators aims at reducing higher vibration levels experienced in lateral direction.

4.3 DESIGN PHILOSOPHY

Mathematical model of a system mounted on isolator [27] is shown in Figure 4.3.

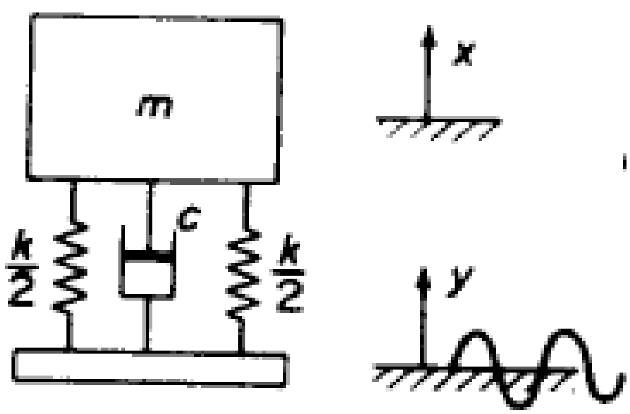


Figure 4.3 Mathematical model of a system mounted on isolator

In which

m: Mass representing the system mounted on isolator

k: Stiffness of isolator

c: Damping coefficient of isolator

y: Vibration input

x: Vibration response

The spring and damper are connected in parallel with one end of each connected to the mass and other end of each connected to a movable support. Application of Newton's law to this system yields.

$$k(x - y) + c(\dot{x} - \dot{y}) + m\ddot{x} = 0 \quad (4.1)$$

Where

$\dot{x}$: Response vibration velocity

$\ddot{x}$: Response vibration acceleration

$\dot{y}$: Input vibration velocity

$\ddot{y}$: Input vibration acceleration

Solving which gives expression for motion Transmissibility as follows

$$\text{Transmissibility, T} = \frac{X}{Y} = \sqrt{\frac{1 + \left(2\zeta\left(\frac{\omega}{\omega_n}\right)\right)^2}{\left(1 - \left(\frac{\omega}{\omega_n}\right)^2\right)^2 + \left(2\zeta\left(\frac{\omega}{\omega_n}\right)\right)^2}} \qquad (4.2)$$

Where

$$\text{Frequency ratio} = \frac{\text{Frequency of excitation}}{\text{Desired frequency of isolator}} = \frac{\omega}{\omega_n}$$

$$\zeta : \text{Damping factor} = \frac{c}{c_c}$$

In which

c_c: Critical damping coefficient

Frequency of excitation is same as that of first natural frequency of PCBs in stacked configuration.

Equation 4.2 is represented graphically [7] in Figure 4.4.

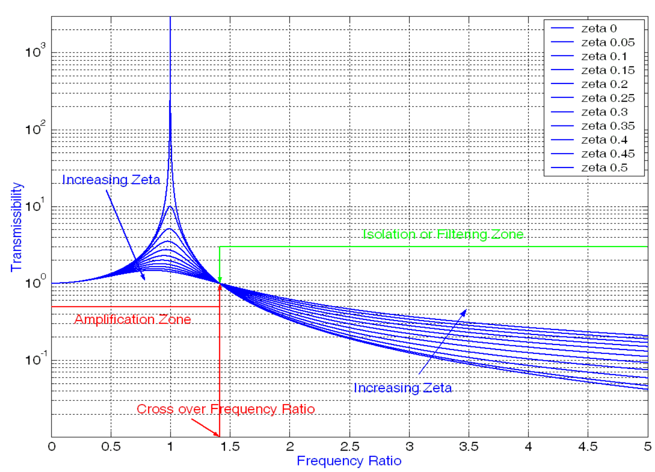


Figure 4.4 Transmissibility curve

From the above figure it is clearly evident that

- When the frequency ratio equals square root of two, Transmissibility drops to 1. This is known a cross over frequency.
- Area below the crossover frequency is known as amplification region
- Above this frequency lies the isolation region, where transmissibility is less than 1.

In other words

$$\text{For } X < Y \Rightarrow \frac{\omega}{\omega_n} \geq \sqrt{2}$$

Alternatively

$$\omega_n \leq \frac{\omega}{\sqrt{2}}$$

Hence information about excitation frequency i.e. first natural frequency of PCBs in stacked configuration is required to estimate the desired frequency of isolator.

A package (Package-1) consisting of PCBs mounted in stacked configuration using screws considered for study is shown in Figure 4.5.

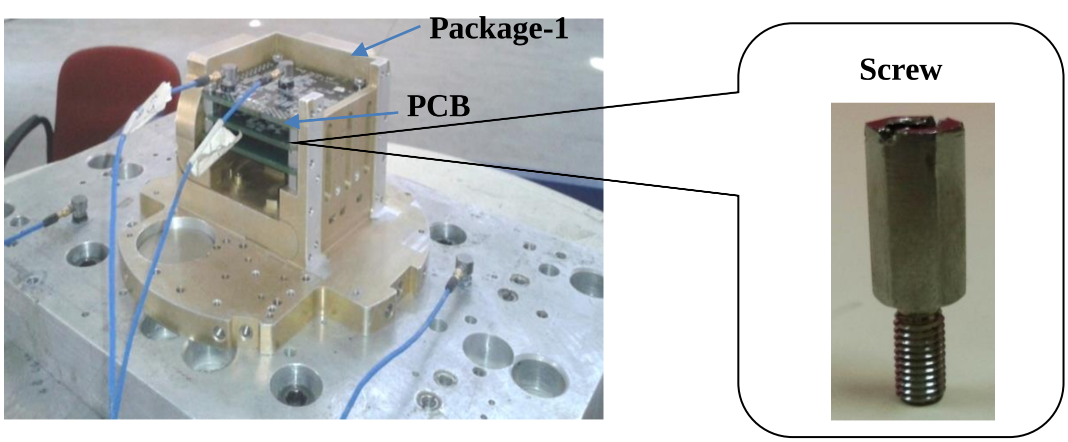


Figure 4.5 Package-1 in stacked configuration

Details of PCBs and screws are given below.

- Number of PCBs = 3
- Size of PCB is 65 mm x 65 mm x 2.4 mm
- Size of screw is 8 mm
- Height of screw is 15 mm

- Number of screws = 4 (For each row)

4.4 RANDOM VIBRATION TEST ON PACKAGE-1 IN STACKED CONFIGURATION WITH SCREWS

Random vibration test is done on package-1 and details of key elements of test set up are given below.

- Electro-dynamic shaker
- Accelerometers
- Fixture

4.4.1 ELECTRO-DYNAMIC SHAKER

In principle, the electrodynamic shaker operates like a loudspeaker. The voice coil of a loudspeaker pushes and pulls a cone in and out causing sound pressure waves. In a shaker it is the armature coil that moves in and out, causing vibration. When an electric current passes in a coil it produces a magnetic field around it. This is the basic principle of electromagnetism. English physicist John Fleming devised the left hand rule to recall the relative directions of the magnetic field, current, and motion in an electric generator or motor. The three directions are represented by the thumb (For thrust or motion), forefinger (For field), and second finger (For current direction), all held at right angles to each other. The Armature force in the present shaker is directly proportional to the current in the coil as given below.

$$F = B \times I \times L$$

Where

F is the force in Newtons

B is the magnetic flux density in Tesla

I is the current in Amperes

L is the coil length in meters

The magnetic flux density can be thought of as the concentration of field lines. The force can be increased by increasing any of the terms within the equation. In a shaker the armature coil responds to the output of controller signal which has been amplified. In a small shaker there is a permanent static magnet that will attract or repel the magnetic field of the coil and cause movement by pulling or pushing. If the two magnetic fields are lined up then south to north attracts and north to north repels. The size of the armature will affect the frequency range of testing. Constructional details of an electrodynamic shaker [29] are shown in Figure 4.6.

Newton's third law of motion states; "Every action has an equal and opposite reaction" Therefore when vibration occurs vertically, the amount of thrust to move the test item will react against the building floor. To prevent damage and vibration to the surrounding area the vibrator must have isolation. One method is to construct a seismic reaction mass below the shaker installation point. This mass is at least 10 times the force rating of the system. Many electrodynamic vibration systems already have a form of isolation. The body is mounted on a spring system, typically air bags that hold the body in a mid-position using adjustable air pressure. As the body reacts to the vibration test there will be some displacement related to the mass ratios between body and payload.

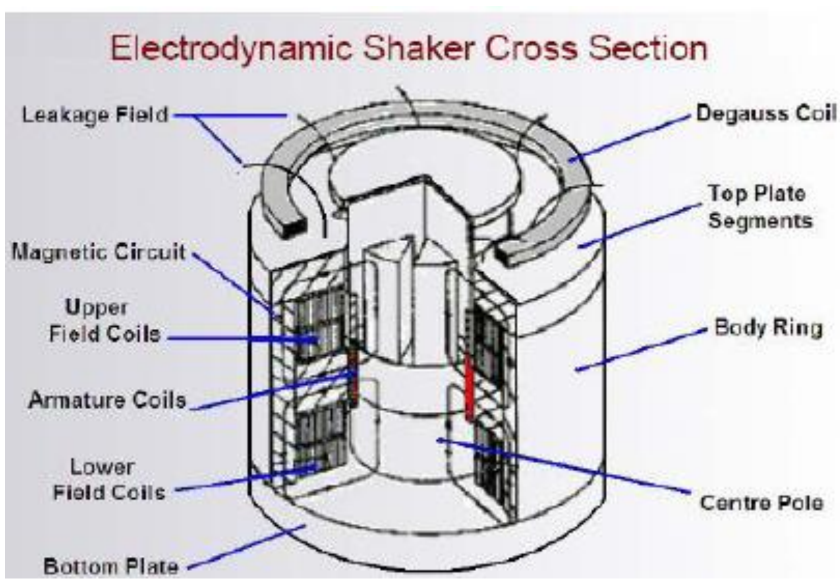


Figure 4.6 Constructional details of an electrodynamic shaker

The amount of vibrator body movement can be calculated by knowing the displacement of the test, the total moving mass of the setup and the body mass. The stiffness of the isolation system tends to give a natural resonant frequency at low frequencies. Normally this will be in between 2.5 Hz and 5 Hz. Suspension system of shaker [29] is shown in Figure 4.7.

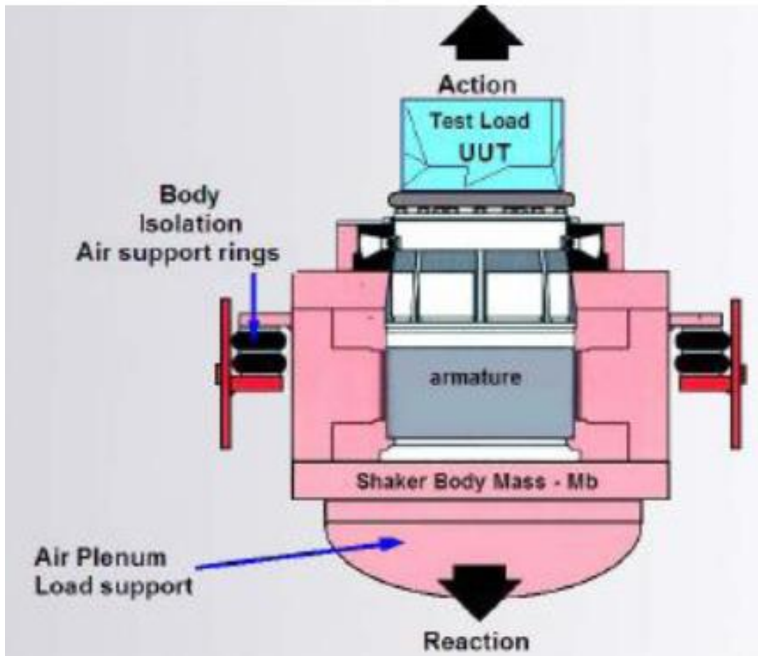


Figure 4.7 Suspension system of shaker

The electro-dynamic shaker used for carrying out present research work is shown in Figure 4.8.

Figure 4.8 Shaker used in the present research work

The detailed specifications of the shaker are clearly described in a tabular format as shown in table 4.1.

Table 4.1 Specifications of the shaker used in the present research work

Parameter	**Specifications**
Model	SEV 200
Rated Force	2000 kgf peak SINE 2000 kgf rms RANDOM 4000 kgf, Peak SHOCK
Maximum Acceleration	Above 60 ‘g’ Peak
Useful frequency range	5 Hz to 2000 Hz
Pay Load Capacity (with Pneumatic ILS)	400 kg

4.4.2 FIXTURE

Vibration testing requires a test fixture to interface the specimen and vibration shaker. Vibration test fixtures simulates mounting interface for both the specimen and vibration shaker on either side. The purpose of a test fixture is to couple mechanical energy from a shaker into a test specimen. During the vibration testing, the effects caused by the fixture are important. Vibration test fixtures exhibit resonant frequencies in the test frequency range due to the mass/stiffness characteristics. The resonant frequencies causes significant problem when conducting vibration tests. However, vibration controller controls the shaker input through a feedback accelerometer and controls the level of vibration input into the test specimen. But, the control accelerometer controls the level of vibration where the control accelerometer is bonded. This indicates that the resonant behavior of the vibration fixture is dependent on the fixture itself.

As a vibration test fixture designer, the following inputs of test article and test equipment are to be considered prior to designing a fixture.

- Details of the shaker table such as pattern of the attachment holes, bolt size and thread sizes are to be known to which the fixture attaches.
- Size and configuration of the test article.
- Weight and C.G. of test article.
- Details of the test specimen such as the mounting interface along with its dynamic characteristics.
- Details of the dynamic test specifications.
- Test axis for which fixture is designed.
- Mass that can be attached to the shaker table without possibly causing damage.
- Available shaker rating.
- Necessary pre load between the table and fixture.
- Awareness of the possibilities of the shaker resonances.
- Anticipated/repeated usage of the fixture.

Test article size and configuration are preliminary information required to size and configure the fixture. Weight of the test item and its C.G and an estimate of its location are required to calculate the combined C.G of the test specimen and the fixture to fall as close as possible to the center line of the shaker. The axis of motion such as X, Y or Z should be defined relative to the test item or to the assembly into which the item is attached in normal service. The test intensities must be known for two reasons,

- The inertia force acting on the test item F must be with stood by the bolts or other fasteners connecting the test item to the shaker
- There is a limit to fixture weight that can be allowed without exceeding the force rating of the shaker.

Mechanical impedance of fixture is also an important parameter. Simple definition of mechanical impedance is that it is the force required to produce a desired motion to the specimen. Launch vehicle equipment is exposed to random vibration excitations during launch and is functionally designed to survive random vibration testing. In this testing, the random vibration design levels are applied at the vibration shaker interface. For light weight aerospace structures, the mechanical impedance of the equipment and of the mounting structure are typically comparable, so that the vibration of the combined structure and the load involves modest interface forces and responses. Whereas the complex space launch vehicle test fixtures will exhibit resonant and anti-resonant frequencies in the testing frequency band. At anti-resonant frequencies, force exceeding the capability of vibration equipment is necessitated to maintain the required test level. The extra force requirement leads to infinite mechanical impedance. So, the test fixture has to be designed in such a way that it should not induce mechanical impedance to the specimen during testing.

Another criterion is “transmissibility” i.e. the fixture must be as stiff as possible so that it is not deflected by the load and transfers motion with high fidelity. This quality is called transmissibility, which is a comparison of the output to the input. At a transmissibility of 1.0, the output faithfully follows input. Ideally, a dynamic test fixture couples the motion from the vibration shaker table to the specimen with zero distortion at all amplitudes and frequencies. This ideal situation is approachable, if the test frequency range is narrow or if the test specimen is small. Practically, the ideal situation cannot be met and the limitation of the fixture must be known. The basic fixture shortcoming is insufficient stiffness. In theory, with an infinitely stiff fixture, the natural frequency of the fixture-specimen system can be made as high as necessary to prevent resonance in the testing frequency band and to provide a transmissibility of 1.0. Thus, the most important factor in fixture design is that its natural frequency should be above the frequency range of interest. However, weight and cost factors require some design tradeoffs. Weight is a drawback as it infringes the capability of the vibrator to deliver enough acceleration to the specimen. Cost considerations influence whether several single-purpose fixtures or one general purpose fixture will be built. Weight and cost considerations almost always compromise fixture performance. The material of the fixture should be as low in mass as possible without detriment to the stiffness. As F=m x a, the material mass will have an influence on the shakers thrust ability. Often a fixture will be sculptured and material removed to reduce the mass. The dynamics of the fixture remain unchanged but the stiffness and strength may be compromised and should be considered. If the first resonant frequency can be kept above the test frequency range the quality of test will be improved. The fixture must be stiff enough not to influence the test or change during a test. The fixture will see high stress levels from the applied vibration or the specimen response and must be strong enough to transmit the forces and survive the tests. A fixture needs to be significantly stronger than the specimen so that it does not fatigue during use. Avoid any thin brackets, small bolts, sharp corners, overhanging areas and weak areas that will fatigue quickly to improve the quality of the fixture.

Typical fixture is shown in Figure 4.9.

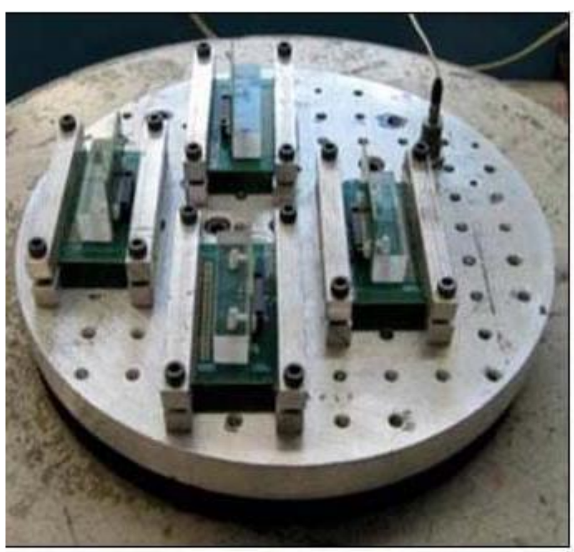


Figure 4.9 Fixture used for the present research work

4.4.3 RANDOM VIBRATION TESTING

Package-1 is mounted on shaker and excited with a broad band random vibration input. Test set up is shown in Figure 4.10.

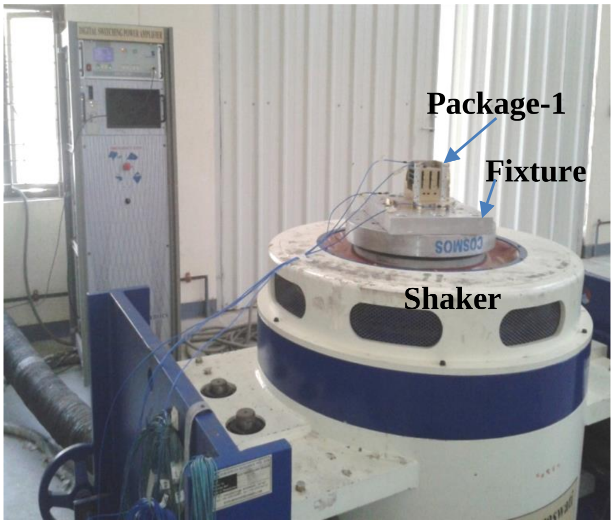


Figure 4.10 Test set up

Figure 4.11 shows accelerometers mounted on top PCB for measuring its vibration response.

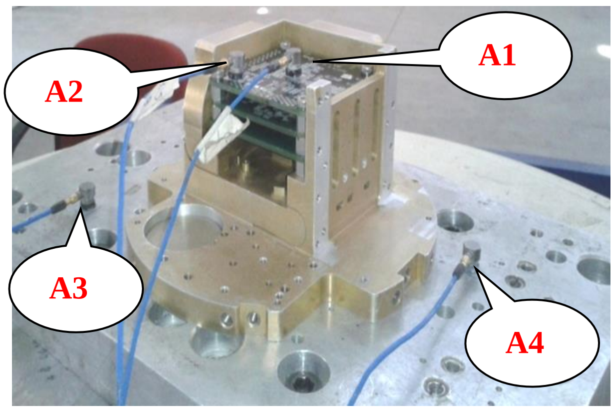


Figure 4.11 Accelerometers mounted on PCB

Where:

A1 (Accelerometer-1): To measure the response at that specific location

A2 (Accelerometer-2): To measure the response at that specific location

A3 (Accelerometer-3): Input to the package

A4 (Accelerometer-4): Input to the package

The vibration input (12.5g_{rms}) between 20- 2000 Hz given in Table 4.2 is considered for the test in a direction normal to the surface of the PCB (Vertical) as well as tangential to the PCB (Lateral).

Table 4.2 Vibration input

FREQUENCY, Hz	**PSD, g^2/ Hz**
20	0.0044
100	0.11
1000	0.11
2000	0.0275

Vibration input mentioned in the above table is fed to the vibration shaker and the same is shown in form of graph in Figure 4.12.

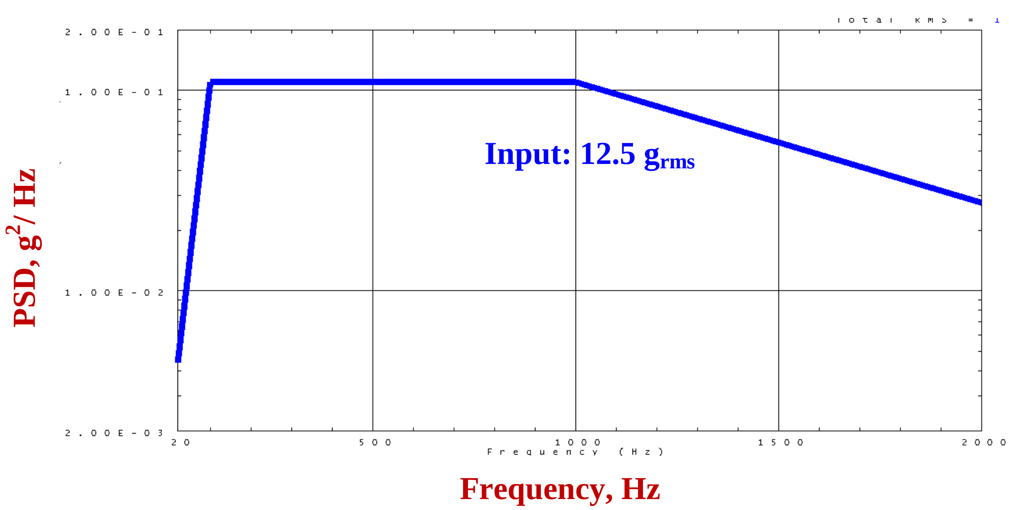
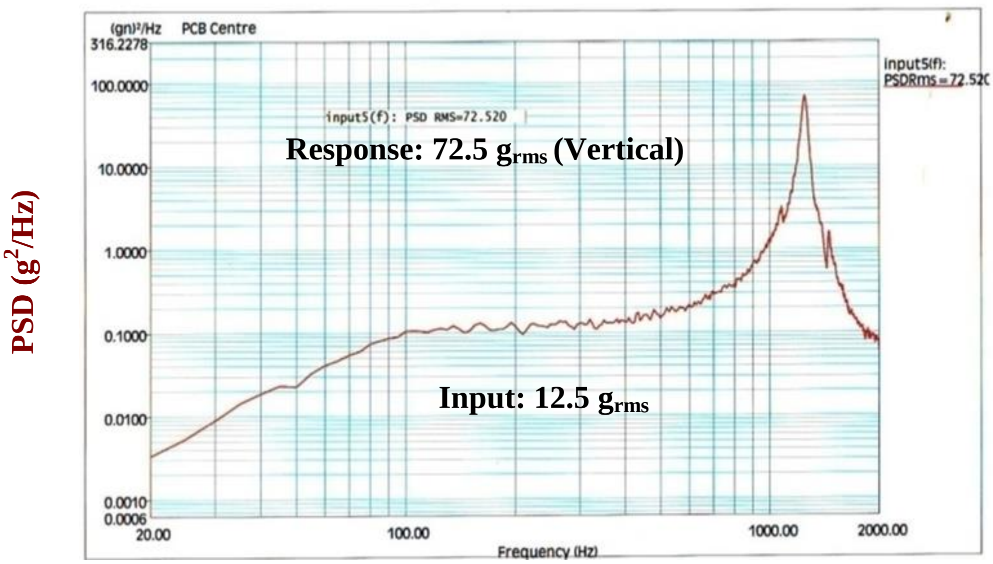


Figure 4.12 Vibration input spectrum

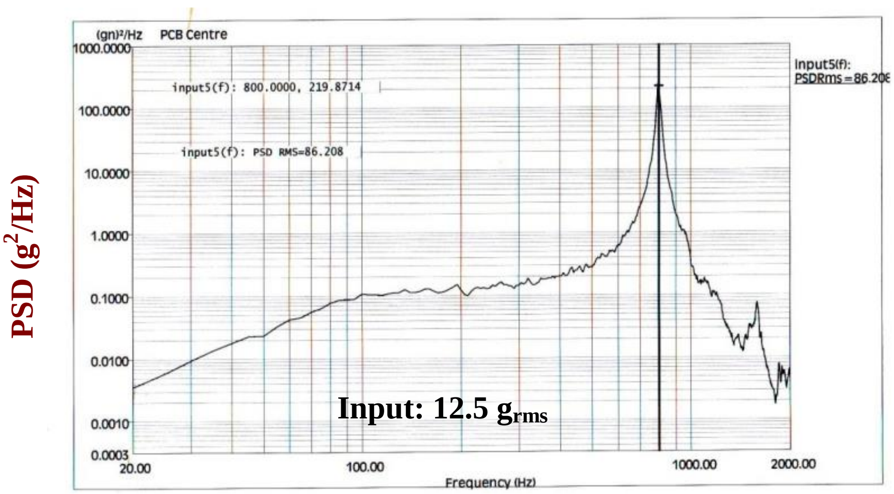
Vibration response spectra of package-1 for both vertical and lateral direction are shown in Figure 4.13 and Figure 4.14 respectively.



Frequency (Hz)

Figure 4.13 Vibration response spectra of package-1 – Vertical

Response: 86.2 g_{rms} (Lateral)



Frequency (Hz)

Figure 4.14 Vibration response spectra of package-1 - Lateral

Vibration test results are summarized in Table 4.3.

Table 4.3 Vibration test results for package-1 in stacked configuration with screws

DIRECTION OF EXCITATION	VIBRATION RESPONSE (INPUT: 12.5 g_{rms})	FREQUENCY
Vertical	72.5 g_{rms}	1200 Hz
Lateral	86.2 g_{rms}	800 Hz

4.5 DESIGN OF SPRING ISOLATOR

4.5.1 DESIGN INPUTS

Following inputs are considered for design.

- Number of PCBs = 3
- Density of PCB material i.e. FR4 = 1975 kg/m^3
- Size of PCB = 65 mm x 65 mm x 2.4 mm
- Mass of component mounted on PCB = 8.5 grams
- Size of screw = 8 mm
- Height of screw = 15 mm
- Number of screws = 4 (For each row)
- Number of isolators = 4 (Isolators are planned to replace screws in bottom row only)
- Density of screw material i.e. steel = 7820 kg/m^3

4.5.2 DESIGN CONSTRAINTS

4.5.2.1 Dimensional

Proposed spring isolator should replace screw one to one without seeking for changes in space allocated for (Which means outer dimensions of spring isolator must be identical to that of screw i.e. height is 15 mm and outer diameter is 8 mm).

4.5.2.2 Stability

As the intended spring isolator is meant for low frequency vibration for which natural frequency and consequently the stiffness of the mounting system must be low. With the springs of low stiffness deflected as shown in Figure 4.15, the lateral stiffness of the springs may be insufficient to restore the mounted equipment to its initial position [7].

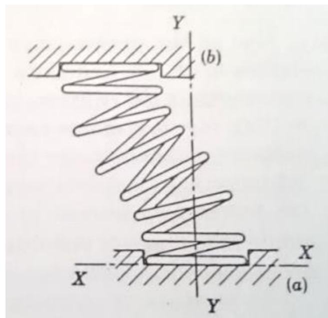


Figure 4.15 Lateral deflection of a typical helical spring

The mounted equipment will then fall sideways as a result of instability of the springs. The necessary conditions to assure stability in such an installation are as follows.

The stability of a system is determined by the energy relations existing within the system. Stable equilibrium exists when a system is in a condition of minimum potential energy. The stability of the system now being discussed is, therefore, analysed by writing the equation for the change of potential energy as a function of horizontal deflection, and noting the conditions for which the energy increases and decreases as horizontal movement of the mounted equipment takes place. During horizontal deflection of the spring shown in Figure 4.15, the vertical stiffness decreases as a result of the eccentric load. The mounted equipment is therefore lowered, and its potential energy is reduced. If the stability is to be maintained, this loss of potential energy must be at least balanced by an increase in strain energy in the spring. Increased strain energy in the spring results from its horizontal deflection and its increased vertical deflection. From this energy relation, it is found that the system is stable when [7]

$$\frac{K_{lat}}{K_{vert}} \geq 1.20\left(\frac{\delta_{\mathrm{st}}}{\mathrm{h_s}}\right) \qquad (4.3)$$

Where

K_{lat}: Stiffness in lateral direction-X

K_{vert}: Stiffness in axial (Vertical-Y) direction

δ_{st}: Static deflection of spring

h_s: Working height of spring isolator = l_o - δ_{st}

In which

l_o: Free length

4.5.2.3 Stiffness

Based on the design criteria for effective isolation stated earlier

$$\textit{Frequency} \text{ of isolator} \leq \frac{800}{\sqrt{2}} \leq 565.7\ \text{Hz}$$

4.5.2.4 Buckling

Design of any helical spring needs to be checked for possibility of buckling to occur. Spring will not buckle if the following condition is satisfied [14].

$$\frac{l_o}{D} < 4$$

Where

D: Mean coil diameter of spring isolator

4.5.3 DESIGN OF SPRING ISOLATOR

Mass of PCB = Volume x Density

$$\text{Mass of screws} = \frac{\pi}{4} d_s^2\ H_s\ \rho\ n_s$$

Where

d_s = Size of screw

H_s = Height of screw

ρ = Density of screw material

n_s = Total number of screws

Total Mass = Mass of PCBs + Mass of component + Mass of screws

$$\text{Mass on each isolator, } m_{isoaltor} = \frac{\text{Total mass}}{\text{Number of isolators}}$$

Dimensions of the spring isolator are worked out while meeting above mentioned design constraints. Outer diameter of the spring isolator is kept at its limiting value i.e. 8 mm. However to meet above mentioned stability criteria height of the spring isolator has to be less than outer diameter while retaining total height as 15 mm. Accordingly height of the spring portion is considered as 5 mm while sharing rest 10 mm equally to top nut and bottom bolt portions. Size of wire is taken as 0.6 mm and number of turns is considered as 2.

The end connections for compression helical springs are suitably formed in order to apply the load. Various forms of end connections are shown in Figure 4.16 [14].

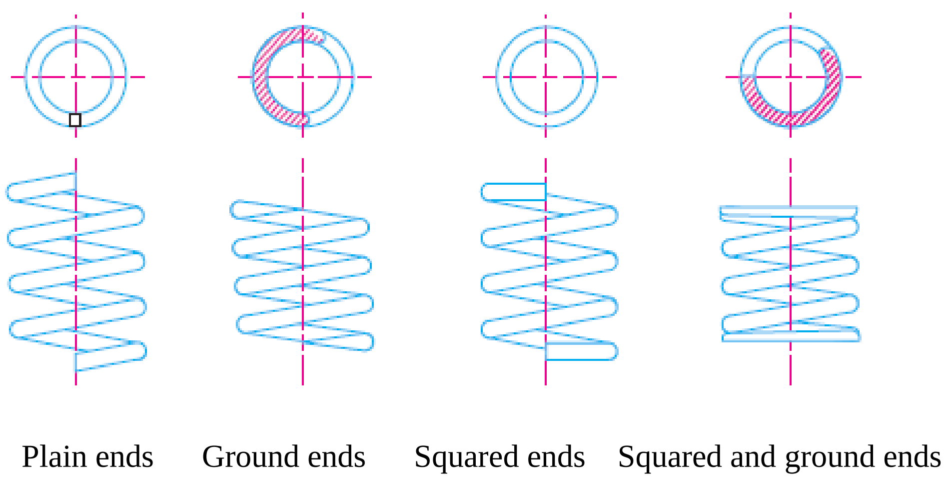


Figure 4.16 End connections for compression helical spring

In all springs, the end coils produce an eccentric application of the load, increasing the stress on one side of the spring. Under certain conditions, especially where the number of coils is small, this effect must be taken into account. The nearest approach to an axial load is secured by squared and ground ends, where the end turns are squared and then ground perpendicular to the helix axis. Based on this squared and ground ends are planned for the proposed spring isolator.

Further it may be noted that part of the coil which is in contact with the seat does not contribute to spring action and hence are termed as inactive coils. The turns which impart spring action are known as active turns. As the load increases, the number of inactive coils also increases due to seating of the end coils and the amount of increase varies from 0.5 to 1 turn at the usual working loads.

Criteria for selection of material

- Spring factor (Specific strain energy) should be high

Spring factor for different materials is compared in Table 4.4.

Table 4.4 Spring factor for different materials

Sl. No.	**Material**	**Spring factor**
1.	Spring steel	0.4246
2.	BASF A3WG7	0.4206
3.	BASF B3ZG6	0.3541
4.	BASF A3WG10	0.3341
5.	DuPont Delrin 100	0.3057
6.	BASF N2200G53	0.2875

From Table 4.5 spring steel is considered as it has high spring factor. Additional merits of spring steel are

- High yield stress
- High Young's modulus
- High damping

For squared and ground ends

Total number of turns, $n^1 = n + 2$

In which

n: Number of active turns

Pitch of spring isolator is expressed as follows

$$p = \frac{h_s}{n^1 - 1} \tag{4.4}$$

Axial stiffness of the spring isolator is expressed as follows

$$K_{vert} = \frac{\mathrm{G\,d}^4}{8\,\mathrm{D}^3 n} \tag{4.5}$$

Where

G: Shear modulus of spring material = 80 GPa

d: Size of wire

Static deflection of the spring is expressed as follows

$$\delta_{st} = \frac{\mathrm{W_{isolator}}}{K_{vert}} \tag{4.6}$$

Where

$W_{isolator}$ = Load on each isolator = $m_{isolator}$ x 9.81

Natural frequency of spring isolator in axial (Vertical) direction is expressed as follows

$$f_n = \frac{1}{2\pi}\sqrt{\frac{K_{vert}}{m_{isolator}}} \tag{4.7}$$

Lateral deflection of the spring isolator (As shown in Figure 4.15) embodies strain of the wire in both torsion and flexure [7]. Analysis of lateral deflection becomes rather complicated for applications where the ends of the spring are constrained to remain parallel during deflection. The results of the analysis may be simplified by introducing limitations that the spring be made of steel and that the cross section of the spring wire be circular. These limitations make possible the substitution of specific numerical values for the moduli of steel and known relations for the cross section of the wire. It is assumed for the moment that the axial force is zero. The following expression is then obtained for the stiffness in lateral direction [28].

$$K_{lat} = \frac{10^6\,\mathrm{d}^4}{C_l\, n\,\mathrm{D}\left(0.204\,\mathrm{h_s^2} + 0.256\,\mathrm{D}^2\right)} \tag{4.8}$$

In which

C_l: A factor depending on ratio between static deflection and free length and ratio between free length and mean coil diameter of spring isolator

Values of which may be taken from the Figure 4.17 [28].

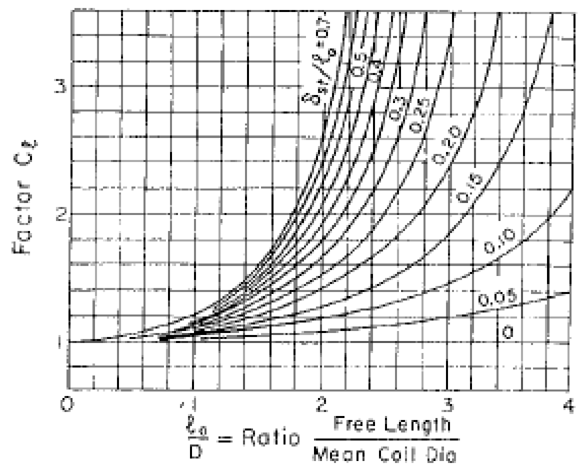


Figure 4.17 Variation of C_l with ratio between static deflection and free length and ratio between free length and mean diameter of spring isolator

For the spring isolator

$$\frac{l_o}{D} = 0.67 \text{ and } \frac{\delta_{st}}{l_o} = 0.036$$

For the above values from Figure 4.9, $C_l = 1.0$

Equation 4.8 gives an expression for the lateral stiffness of a helical spring that neglects the influence of an axial load. The effect of an axial load is to increase the lateral deflection i.e. to decrease the lateral stiffness. Corrected value of lateral stiffness can be obtained from Figure 4.18 [28].

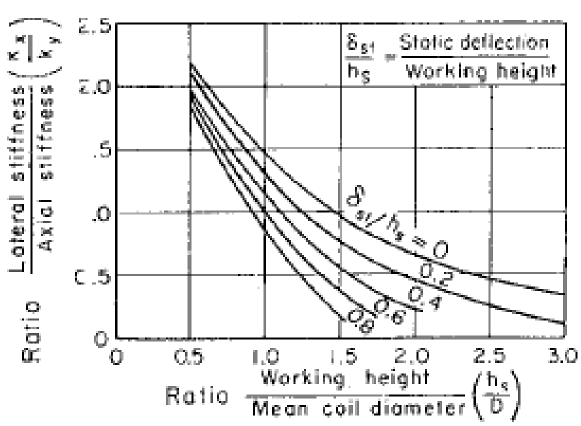


Figure 4.18 Corrected value of lateral stiffness

For the spring isolator

$$\frac{h_s}{D} = 0.65 \text{ and } \frac{\delta_{st}}{h_s} = 0.0375$$

For the above values from figure 4.20,

$$\frac{K_{lat}}{K_{vert}} = 2$$

After substituting axial stiffness in the above expression corrected value of lateral stiffness is obtained.

Gap between adjacent turns= p – d

Spring index is expressed as follows

$$C = \frac{D}{d} \qquad (4.9)$$

Wahl's stress factor is expressed as

$$K = \frac{4C-1}{4C-4} + \frac{0.615}{C} \qquad (4.10)$$

Nominal shear stress experienced by the spring isolator is expressed as follows.

$$\tau_s = \mathrm{K}\frac{8W_{isolator}D}{\pi d^3} \qquad (4.11)$$

The helical springs subjected to fatigue loading are designed by using the Soderberg line method [14]. The spring materials are usually tested for torsional endurance strength under a repeated stress that varies from zero to a maximum. Since the springs are ordinarily loaded in one direction only (The load in springs is never reversed in nature), therefore a modified Soderberg diagram is used for springs, as shown in Figure 4.19.

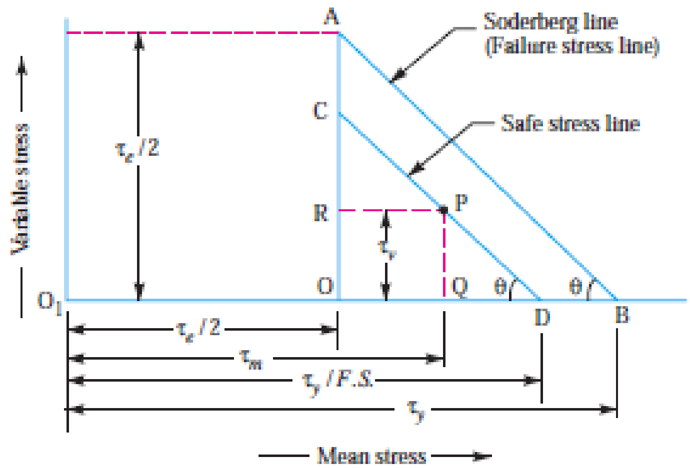


Figure 4.19 Modified Soderberg method

In Figure 4.19, the endurance limit for reversed loading is shown at point A where the mean shear stress is equal to τ_e / 2 and the variable shear stress is also equal to τ_e / 2. A line drawn from A to B (The yield point in shear, τ_y) gives the Soderberg's failure stress line. If a suitable factor of safety (FOS) is applied to the yield strength (τ_y), a safe stress line CD may be drawn parallel to the line AB as shown in the same figure. Consider a design point P on the line CD. Now the value of factor of safety may be obtained as given below

$$\frac{1}{FOS} = \frac{\tau_m - \tau_v}{\tau_y} + 2\frac{\tau_v}{\tau_e} \tag{4.12}$$

In which

$$\text{Mean shear stress}, \tau_m = K\frac{8\,W_{isolator-mean}\,D}{\pi\,d^3} \tag{4.13}$$

Where

$$W_{isolator-mean} = \frac{W_{isolator-min} + W_{isolator-max}}{2} \tag{4.14}$$

In which

$W_{isoaltor-min}$: Minimum anticipated load per isolator which is taken as 90% of rated load per isolator

$W_{isoaltor-max}$: Maximum anticipated load per isolator which is taken as 110% of rated load per isolator

$$\text{Variable shear stress}, \tau_v = K \frac{8\, W_{isolator-variable}\, D}{\pi\, d^3} \tag{4.15}$$

Where

$$W_{isolator-variable} = \frac{W_{isolator-max} - W_{isolator-min}}{2} \tag{4.16}$$

Further

Yield strength, τ_y = 300 MPa

Endurance limit, τ_e = 2000 MPa

Fatigue life is expressed as follows [8]

$$Life = e^{\left(\frac{1}{Y}\log_e\left(\frac{K_E}{C_E}\right)\right)} \tag{4.17}$$

Where

Y: Constant = -0.537

$$\text{Endurance factor}, K_E = \frac{K_U\,(K_{s2} - K_{s1})}{2\,K_U - (K_{s2} + K_{s1})} \tag{4.18}$$

In which

K_U: Ultimate strength factor = 0.7

$$\text{Maximum stress factor}, K_{s2} = \frac{\tau_{max}}{\tau_U} \tag{4.19}$$

In which

$$\text{Maximum shear stress}, \tau_{max} = K \frac{8\, W_{isolator-max}\, D}{\pi\, d^3} \tag{4.20}$$

τ_U: Minimum ultimate tensile strength = 465 MPa

Further

$$\text{Minimum stress factor}, K_{s1} = \frac{\tau_{min}}{\tau_U} \tag{4.21}$$

In which

$$\text{Minimum shear stress}, \tau_{min} = K \frac{8\, W_{isolator-min}\, D}{\pi\, d^3} \tag{4.22}$$

Further

C_E: Constant = 0.5758

Outcome of design calculations are summarized in Table 4.5.

Table 4.5 Design parameters of spring isolator

Sl. No.	**Design parameter**	**Value**
1.	Mean coil diameter, D	7.4 mm
2.	Size of wire, d	0.6 mm
3.	Number of turns, n	2
4.	Axial stiffness, K_{vert}	1599 N/m
5.	Natural frequency, f_n	37.4 Hz
6.	Corrected value of lateral stiffness, K_{lat}	3198 N/m
7.	Spring index, C	12.3
8.	Nominal shear stress, τ_s	27 MPa
9.	Factor of safety, FOS	11.67
10.	Life	1.8×10^{36} cycles

Design checks

$$\frac{l_o}{D} = 0.6757 \ < 4, \text{Hence spring isolator will not buckle}$$

$$\frac{K_{lat}}{K_{vert}} = 2 \ \geq 1.20\left(\frac{\delta_{st}}{h_s}\right) = 0.0426, \text{Hence spring isolator is stable}$$

4.6 GRAPHICAL USER INTERFACE (GUI) BASED SOFTWARE IN MATLAB FOR DESIGN OF SPRING ISOLATOR

It is planned to develop a Graphical Use Interface (GUI) based software in MATLAB for design of proposed spring isolator. The necessary formulation, which is derived earlier, is converted in form of code. The intention behind developing the software is to enable the user to use this software as a hand calculator, which just takes the inputs from the user and within no time gives the output there by avoiding getting the desired output through usage of formulae. To meet this task the software should have GUI features and those features should be understood clearly before proceeding for developing the software and the same are given below.

4.6.1 GRAPHICAL USER INTERFACE (GUI)

A Graphical User Interface (GUI) is a graphical display that contains devices, or components, that enable a user to perform interactive tasks. To perform these tasks, the user of the GUI does not have to create a script or type commands at the command line. Often the user does not have to know the details of the task at hand. Each component, and the GUI itself, is associated with one or more user-written routines known as callbacks. The execution of each callback is triggered by a particular user action such as a button push, mouse click, selection of a menu item, or the cursor passing over a component. User, as the creator of the GUI, provides these callbacks. In the GUI the user selects a data set from the pop-up menu, then clicks one of the plot type buttons. Clicking the button triggers the execution of a callback that plots the selected data in the axes. This kind of programming is often referred to as event-driven programming. The event in the example is a button click. In event-driven programming, callback execution is asynchronous, controlled by events external to the software. In the case of MATLAB GUIs, these events usually take the form of user interactions with the GUI. The writer of a callback has no control over the sequence of events that leads to its execution or, when the callback does execute, what other callbacks might be running simultaneously.

4.6.2 DEVELOPMENT OF GUI

The formulation stated in earlier sections is converted in form of MATLAB GUI and the same is shown in Figure 4.20.

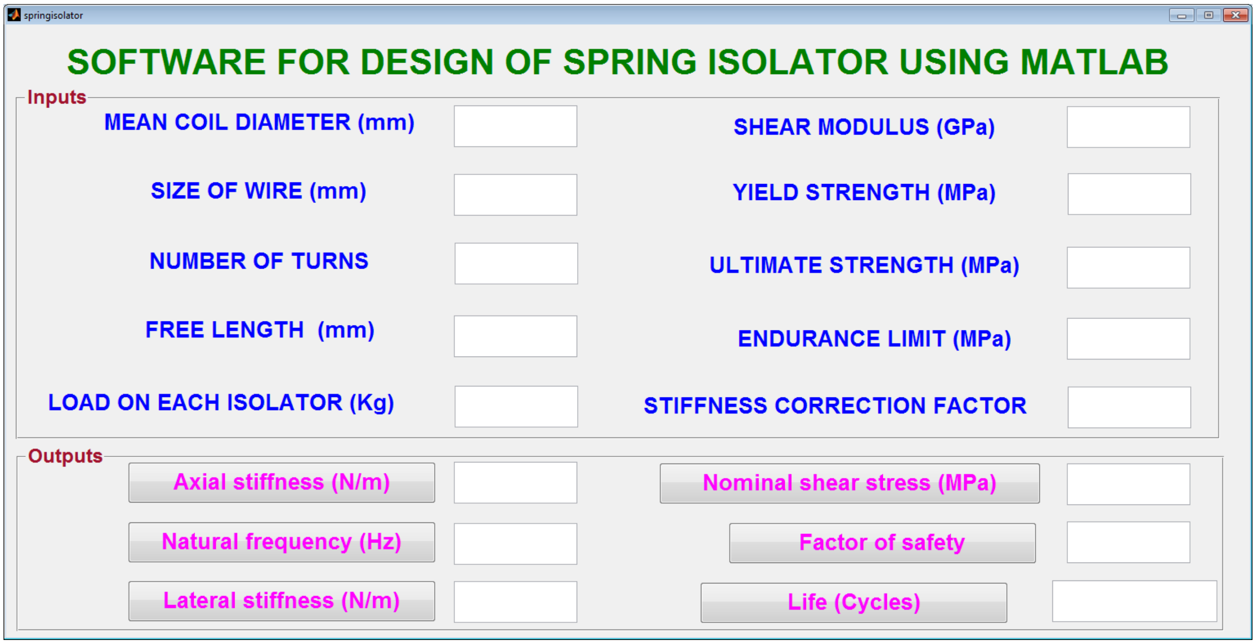


Figure 4.20 Front end of GUI based software

Inputs to GUI are as follows

- Mean coil diameter (mm)
- Size of wire (mm)
- Number of turns (mm)
- Free length (mm)
- Load on each isolator (Kg)
- Shear modulus (GPa)
- Yield strength (MPa)
- Ultimate strength (MPa)
- Endurance limit (MPa)
- Stiffness correction factor

Outputs from GUI are as follows

- Axial stiffness (N/m)
- Natural frequency (Hz)
- Lateral stiffness (N/m)
- Nominal shear stress (MPa)
- Factor of safety
- Life (Cycles)

Procedure to use this software

- Enter into MATLAB
- Type the command GUIDE
- Open the GUI front end file
- Run the GUI with Green colored arrow located on top center
- Type the values in the text boxes corresponding to inputs
- Press the output buttons

Front end of GUI after entering inputs and execution is shown in Figure 4.21.

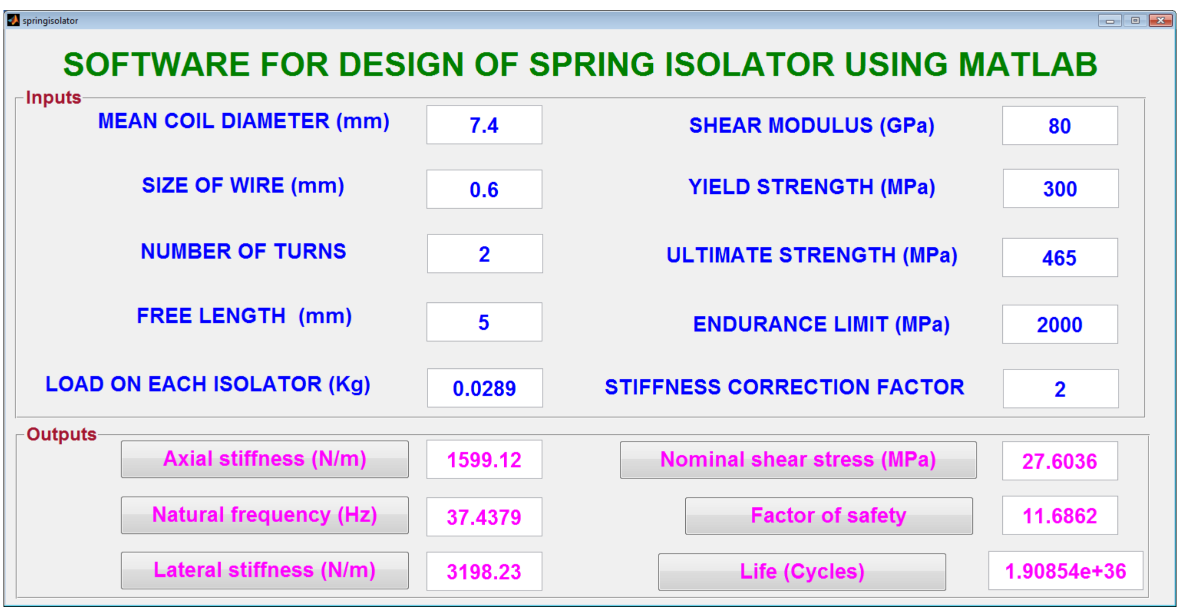


Figure 4.21 Front end of GUI after execution

4.7 VALIDATION OF DESIGN THROUGH FINITE ELEMENT ANALYSIS (FEA)

The design calculations are validated by carrying out Finite Element Analysis (FEA) in commercial software package NX-IDEAS. Static analysis is carried out to obtain displacement in both axial and lateral directions independently against unit load of 1 N. Accordingly stiffness values in both axial and lateral directions are calculated from displacements thus obtained from analysis. Objective of FE analysis is to validate the design by comparing the stiffness values obtained using design calculations with that of FEA. Geometry of isolator is modeled (1D model) and then discretized with linear two nodded beam elements. Beam element has six Degrees of Freedom (DOF) per node and it can take load in all 3 directions.

4.7.1 MATERIAL PROPERTIES

Properties of spring steel (Material of spring isolator) given in Table 4.6 are considered for analysis.

Table 4.6 Material properties

Sl. No.	**Property**	**Value**
1.	Young’s modulus, E	210 GPa
2.	Poisson’s ratio, μ	0.3
3.	Shear modulus	80 GPa

4.7.2 BOUNDARY CONDITIONS

As bottom portion (Threaded portion) of spring isolator will be fastened to chassis, corresponding bottom node in the FE model is constrained for all DOF in the FE model.

4.7.3 LOAD

A load of 1 N is applied on the top node of FE model of spring isolator. FE model of spring isolator is shown in Figure 4.22.

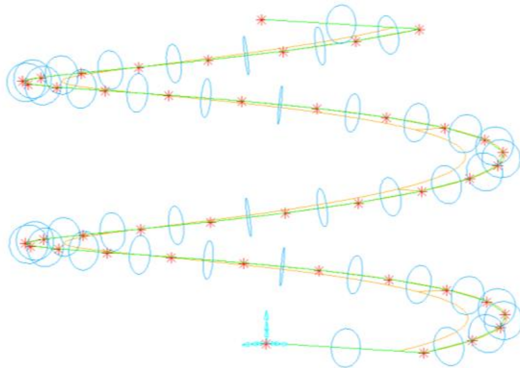


Figure 4.22 FE model

4.7.4 STATIC ANALYSIS

Static analysis is carried out independently for both axial and lateral directions to obtain displacement. Displacement plots for axial and lateral directions are shown in Figure 4.23 and Figure 4.24 respectively.

Maximum displacement in axial direction = 0.622 mm

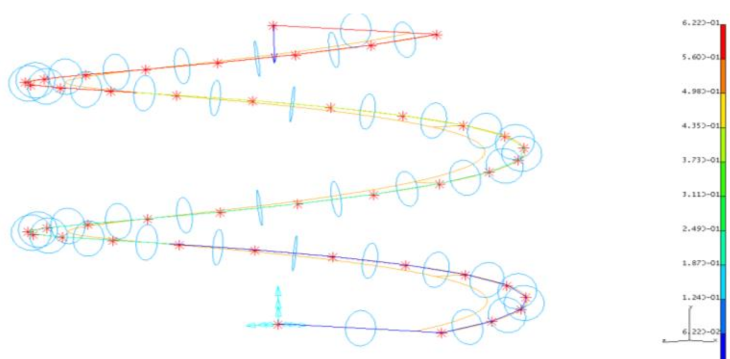


Figure 4.23 Displacement plot – Axial direction

Maximum displacement in lateral direction = 0.312 mm

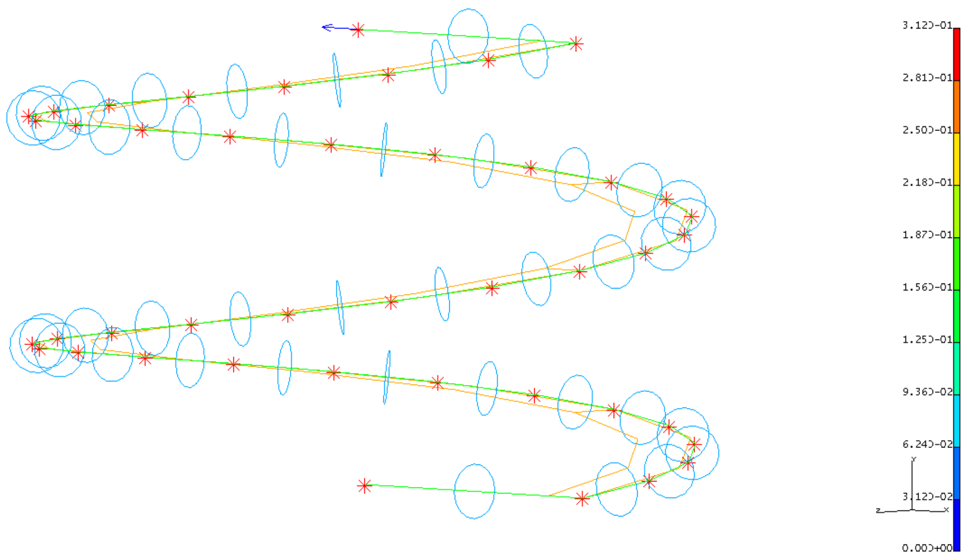


Figure 4.24 Displacement plot – Lateral direction

- From the analysis maximum displacement in axial direction is 0.622 mm.
- Whereas maximum displacement in lateral direction is 0.312 mm.

From displacement, stiffness is calculated as follows.

$$\text{Axial stiffness} = \frac{\text{Load}}{\text{Displacement}} = \frac{1}{0.622} = 1.607 \text{ N/mm} = 1607 \text{ N/m}$$

$$\text{Lateral stiffness} = \frac{\text{Load}}{\text{Displacement}} = \frac{1}{0.312} = 3.205 \text{ N/mm} = 3205 \text{ N/m}$$

Stiffness values obtained from design calculation are compared with that of FEA in table 4.7.

Table 4.7 Comparison of stiffness between design calculation and FEA

Method	**Axial Stiffness**	**Lateral Stiffness**
Design calculation	1599 N/m	3198 N/m
FE analysis	1607 N/m	3205 N/m

- Stiffness values obtained from design calculations are in good agreement with that of FEA validating the design.

4.8 SUMMARY

Spring isolators are designed which are meant for reducing vibration levels in packages with stacked configuration. Design procedure is transformed in form of a GUI based software using MATLAB which can aid as a hand calculator for designing spring isolator. Design is validated with the aid of FEA indicating the poise associated with the design procedure.

CHAPTER 5

DEVELOPMENT AND EXPERIMENTAL CHARACTERIZATON OF METALLIC SPRING ISOLATORS

5.1 INTRODUCTION

Metallic spring isolators are designed for packages in stacked configuration and the design parameters are summarized in earlier chapter. Subsequently spring isolator is developed with dimensions evolved as an outcome of design. This chapter elaborates on manufacturing process of spring isolator. Experimental evaluation of the intended spring isolator in random vibration environment is discussed. Isolation effectiveness of the said isolators is quantified and consistency of the same is studied for other packages in stacked configuration.

5.2 DEVELOPMENT OF METALLIC SPRING ISOLATOR

Spring isolator is manufactured with dimensions evolved as an outcome of design. It is made in three parts namely top nut portion, bottom bolt portion and spring portion and then joined using brazing operation to form spring isolator. Top nut portion and bottom bolt portion are manufactured using conventional methods.

5.2.1 Manufacturing process for spring portion

Steps involved in manufacturing process of spring portion are given below.

- Cold winding
- Heat treatment
- Grinding
- Shot peening
- Setting
- Electroplating

5.2.1.1 Cold winding

Springs having wire diameter up to 18 mm can be manufactured through cold winding process. As the proposed spring isolator is having wire diameter of 0.6 mm it is manufactured using cold winding process at room temperature. To start with wire made of spring steel with size of 0.6 mm is wound around a mandrel. Cold winding is done on a spring winding machine which is operated by hand cranking. Guiding mechanism is adopted for aligning the wire into the desired pitch as it wraps around the mandrel.

5.2.1.2 Heat treatment

Then the spring is heat treated in order to relieve the internal stresses within the material and also to sustain basic resilience property. Spring is tempered by heat treating it in an oven by maintaining the desired temperature of 250^0 C for a time duration of one hour and then allowed to cool in air.

5.2.1.3 Grinding

Ends of spring are grounded with the help of grinding machine to achieve squared and grounded ends as per the design. Spring is positioned in a jig to maintain the right orientation during grinding operation. The spring is held in contact with a rotating abrasive grinding wheel and thus the desired flatness is achieved. The spring is mounted in a jig to ensure the correct orientation during grinding, and it is held against a rotating abrasive wheel until the desired degree of flatness is obtained first with a coarse wheel and later with finer wheel. During grinding operation oil-based lubricating fluid is used to remove the heat produced and also to drive away the debris resulted.

5.2.1.4 Shot peening

During its course of application intended spring isolator will experience material fatigue and also cracking due to cyclic loading. Shot peening is done for spring to impart resistance against the said fatigue and cracking. In shot peening operation small steel balls are bombarded on spring covering its whole surface.

5.2.1.5 Setting

Setting operation is done on the spring to freeze its desired height and pitch perpetually. In this operation the spring is compressed to the fullest extent forcing adjacent coils to come in contact each other. This operation is repeated till the desired height and pitch are obtained.

5.2.1.6 Electroplating

Electroplating is done to impart corrosion resistance properties to the spring. The spring is dipped in the appropriate electrically conducting fluid which will corrode the electroplating metal rather than the spring metal. A positive electrical charge is smeared to the plating metal which is also dipped in the fluid in which the spring is dipped. Simultaneously a negative electrical charge is smeared to spring metal. Positive charged molecules will be discharged from plating metal as it dissolves in the fluid. These molecules being positively charged will establish chemical bondage with spring due to its negative charge. Then the spring is baked at a temperature of 180^0 C for three hours to eliminate the brittleness which is resulted due to electroplating.

Spring isolator thus fabricated is shown in Figure 5.1.

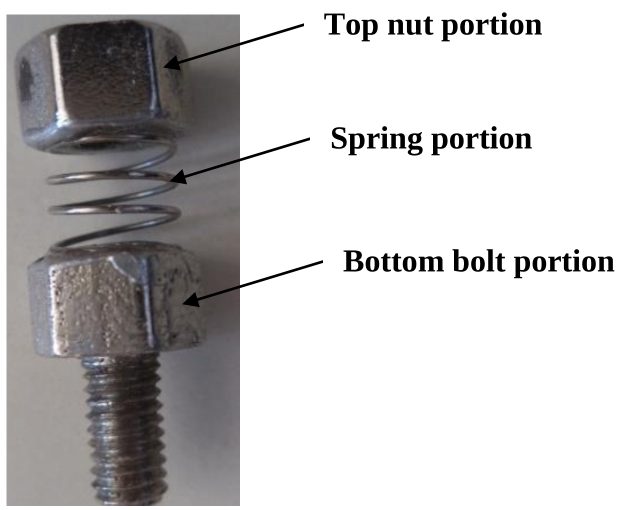


Figure 5.1 Spring isolator

5.3 RANDOM VIBRATION TEST ON PACKAGE-1 IN STACKED CONFIGURATION WITH SPRING ISOLATORS

Screws in the bottom row (Between bottom PCB and chassis) of package-1 are replaced with spring isolators. Then random vibration test is carried out on package-1 with spring isolators in order to evaluate the isolation effectiveness. Package-1 with spring isolators is mounted on shaker and excited with a broad band random vibration input. Test set up is shown in Figure 5.2.

Figure 5.2 Test set up

Figure 5.3 shows accelerometers mounted on top PCB for measuring its vibration response.

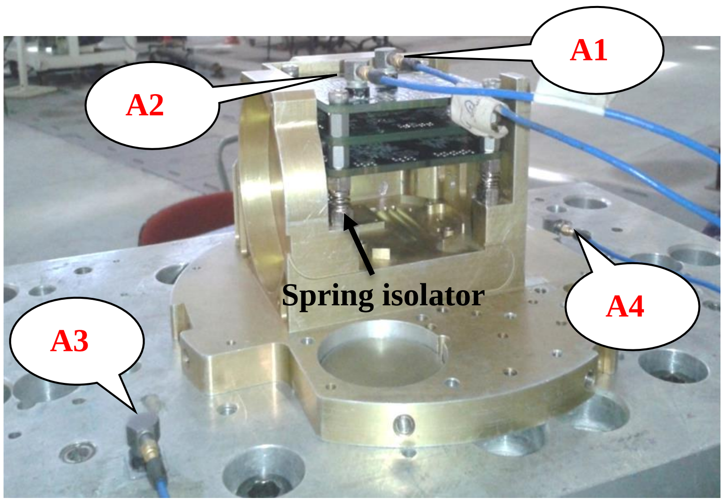


Figure 5.3 Accelerometers mounted on PCB

Where:

A1 (Accelerometer-1): To measure the response at that specific location

A2 (Accelerometer-2): To measure the response at that specific location

A3 (Accelerometer-3): Input to the package

A4 (Accelerometer-4): Input to the package

Details of accelerometer are given below.

- Make: PCB
- Model number: 340A15
- Sensitivity: 10 mV/g
- Measurement range: $\pm$ 500 g
- Frequency range: 1 Hz to 12000 Hz
- Type: Shear
- Weight: 2 grams

Accelerometer details are common for all the packages.

The vibration input ($12.5g_{rms}$) between 20- 2000 Hz given in Table 5.1 is considered for the test in a direction normal to the surface of the PCB (Vertical) as well as tangential to the PCB (Lateral).

Table 5.1 Vibration input

FREQUENCY, Hz	**PSD, g^2/ Hz**
20	0.0044
100	0.11
1000	0.11
2000	0.0275

Vibration input mentioned in the above table is fed to the vibration shaker and the same is shown in form of graph in Figure 5.4.

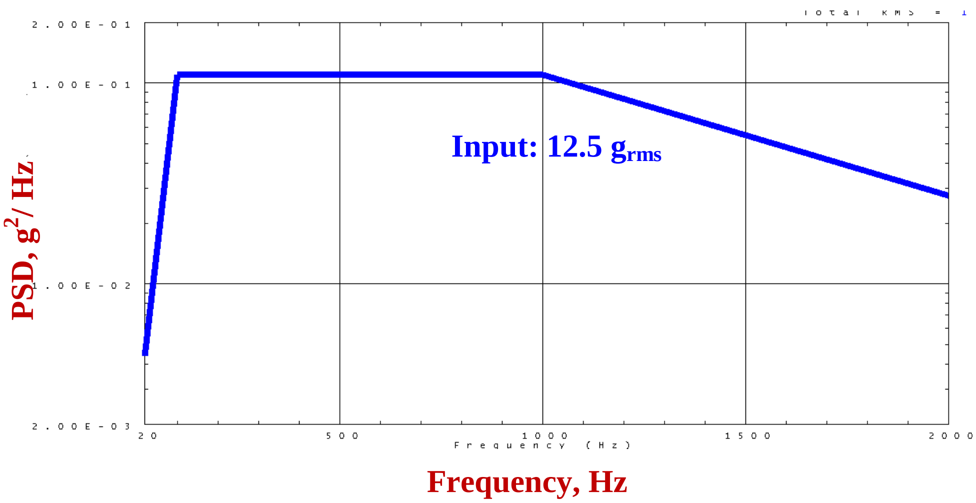


Figure 5.4 Vibration input spectrum

Vibration response spectra of package-1 with spring isolator for vertical and lateral directions are shown in Figure 5.5 and Figure 5.6 respectively.

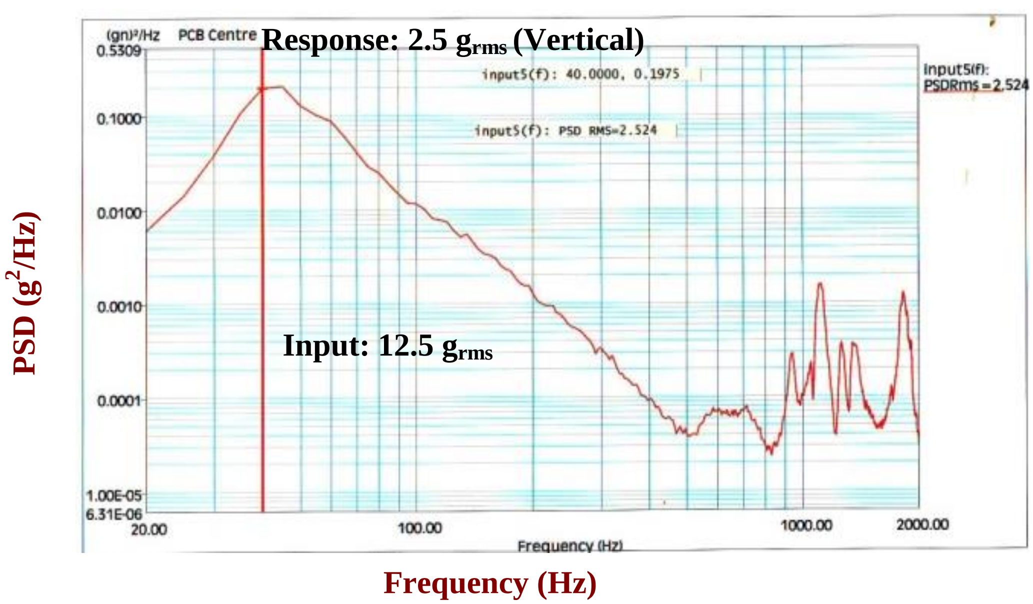


Figure 5.5 Vibration response spectrum of package-1 with spring isolators - Vertical

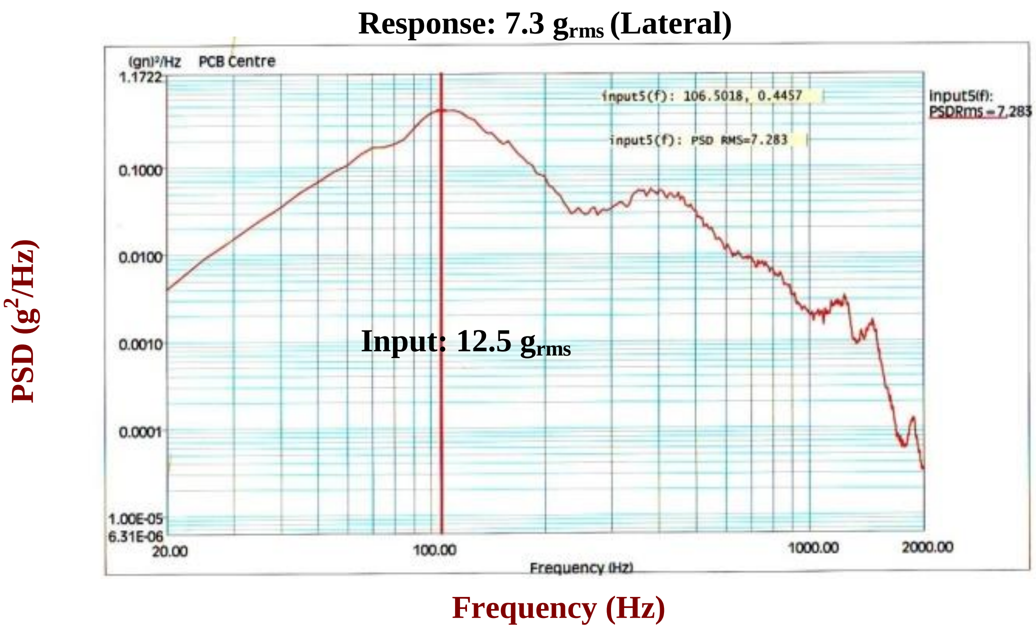


Figure 5.6 Vibration response spectrum of package-1 with spring isolators - Lateral

Vibration levels obtained for package-1 using proposed spring isolators are compared with that of screws in Table 5.2.

Table 5.2 Comparison of vibration levels for package-1

DIRECTION OF EXCITATION	**VIBRATION RESPONSE (INPUT: 12.5 g_{rms})**	
	With screws	**With spring isolators**
Vertical	72.5 g_{rms}	2.5 g_{rms}
Lateral	86.2 g_{rms}	7.3 g_{rms}

- With the proposed spring isolators maximum vibration response got reduced from 72.5 g_{rms} to 2.5 g_{rms} in vertical direction

- Whereas maximum vibration response got reduced from 86.2 g_{rms} to 7.3 g_{rms} in lateral direction.
- First natural frequency of PCBs arranged in stacked configuration with spring isolators is 40 Hz.
- From the results it is observed that the spring isolators are effective in reducing vibration response on PCB for package-1.

5.4 PACKAGE-2 IN STACKED CONFIGURATION WITH SPRING ISOLATORS

In order to establish the consistency associated with the performance of the proposed spring isolator, another package in stacked configuration (Package-2 shown in Figure 5.7) is considered for the study. Package-2 consists of two PCBs arranged in stacked configuration. As electronic components mounted on top PCB (Vibration levels on bottom PCB are well within limits) are experiencing excessive vibration levels, screws between top and bottom PCBs are replaced with spring isolators.

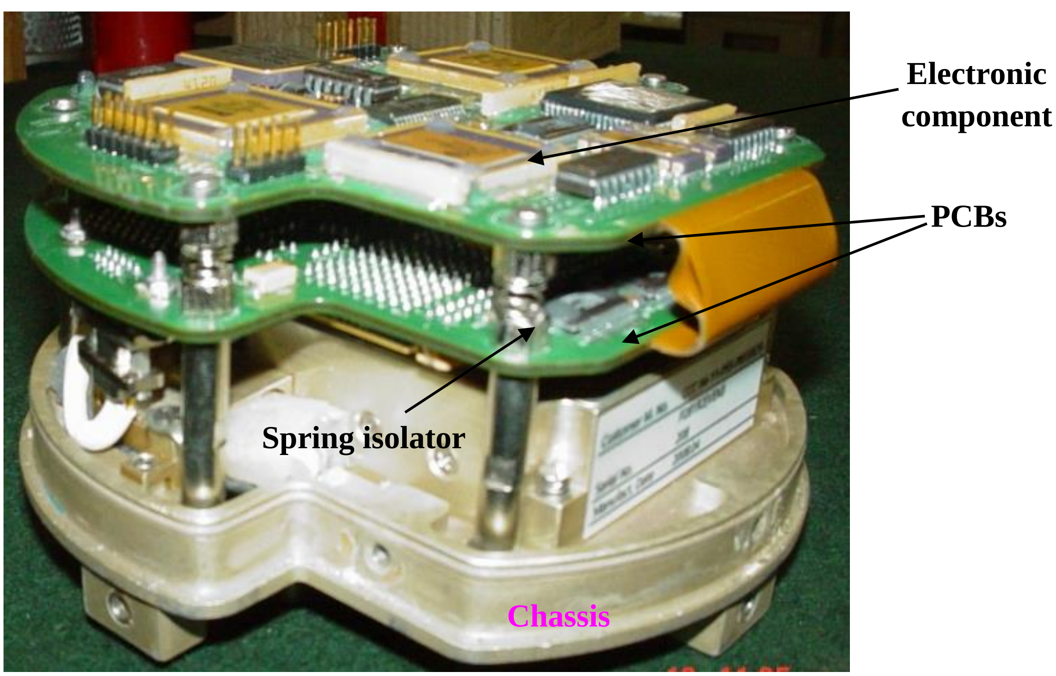


Figure 5.7 Package-2

Figure 5.8 shows the response location.

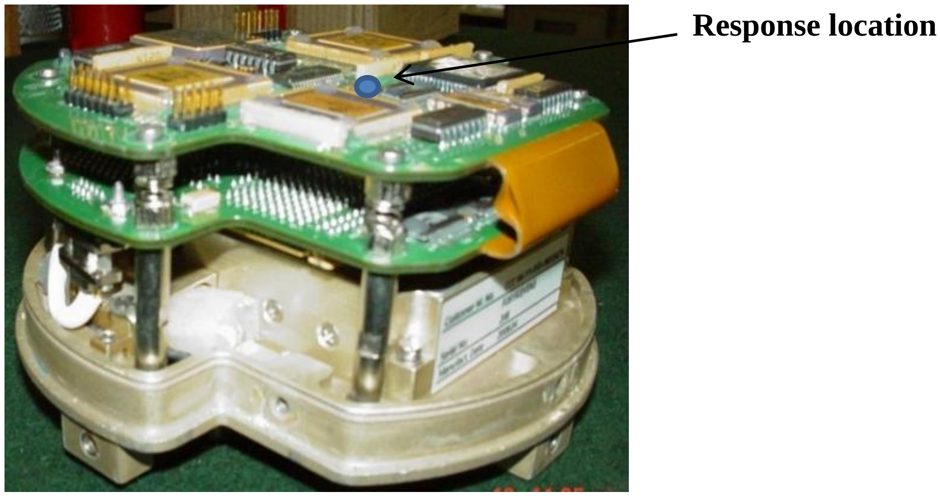


Figure 5.8 Response location

In order to ascertain the applicability of the proposed spring isolators, vibration levels obtained using spring isolators are compared with that of screws in Table 5.3.

Table 5.3 Comparison of vibration responses (Package-2)

VIBRATION RESPONSE (INPUT: 5.3 g_{rms})	
With screws	**With spring isolators**
47.7 g_{rms}	1.5 g_{rms}

5.5 PACKAGE-3 IN STACKED CONFIGURATION WITH SPRING ISOLATORS

Package-3 shown in Figure 5.9 consists of three PCBs arranged in stacked configuration. In this package screws between bottom PCB and chassis are replaced with spring isolators.

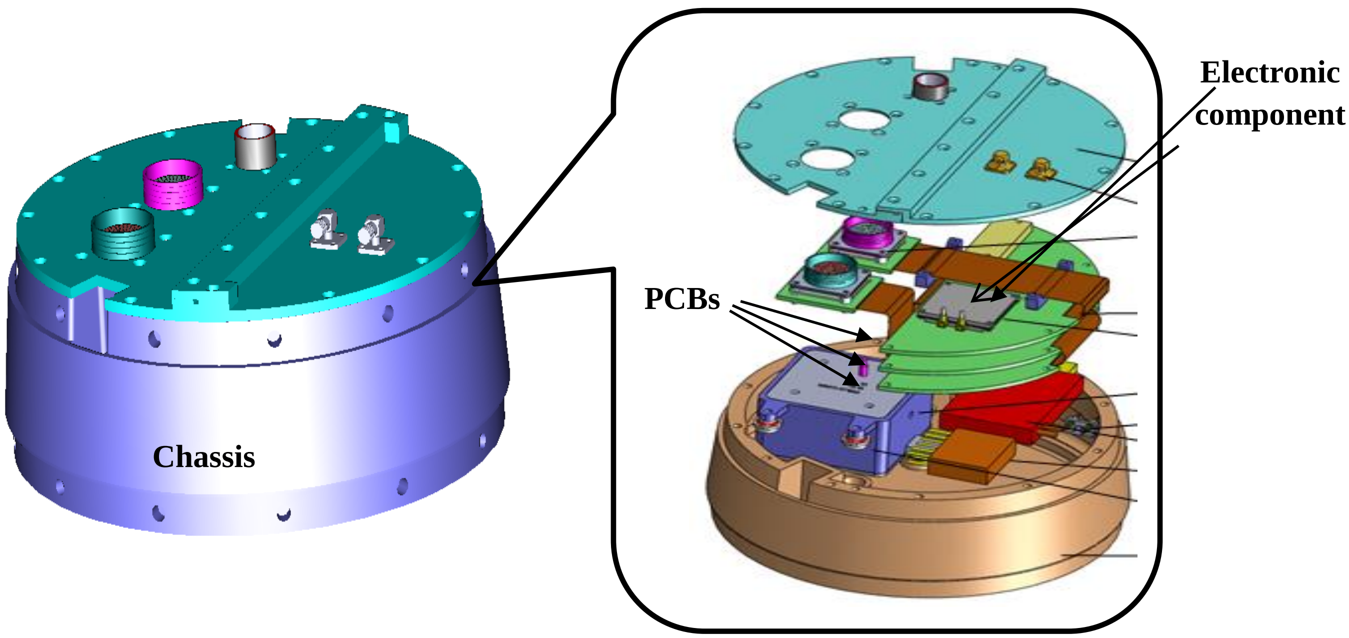


Figure 5.9 Package-3

Figure 5.10 shows the response location.

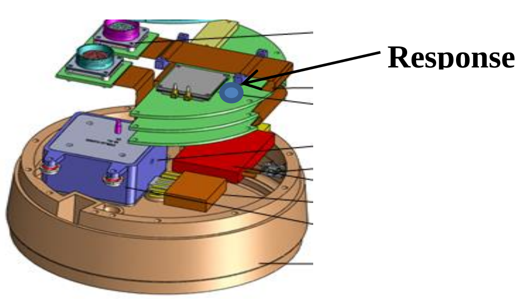


Figure 5.10 Response location

Vibration levels obtained using spring isolators are compared with that of screws in Table 5.4.

Table 5.4 Comparison of vibration responses (Package-3)

VIBRATION RESPONSE (INPUT: 6 g_{rms})	
With screws	**With spring isolators**
63.2 g_{rms}	4.1 g_{rms}

5.6 PACKAGE-4 IN STACKED CONFIGURATION WITH SPRING ISOLATORS

Similar attempt to verify the applicability of the proposed approach has been made on package-4 (Shown in Figure 5.11) consists of two PCBs arranged in stacked configuration. In this package screws between bottom PCB and chassis are replaced with spring isolators.

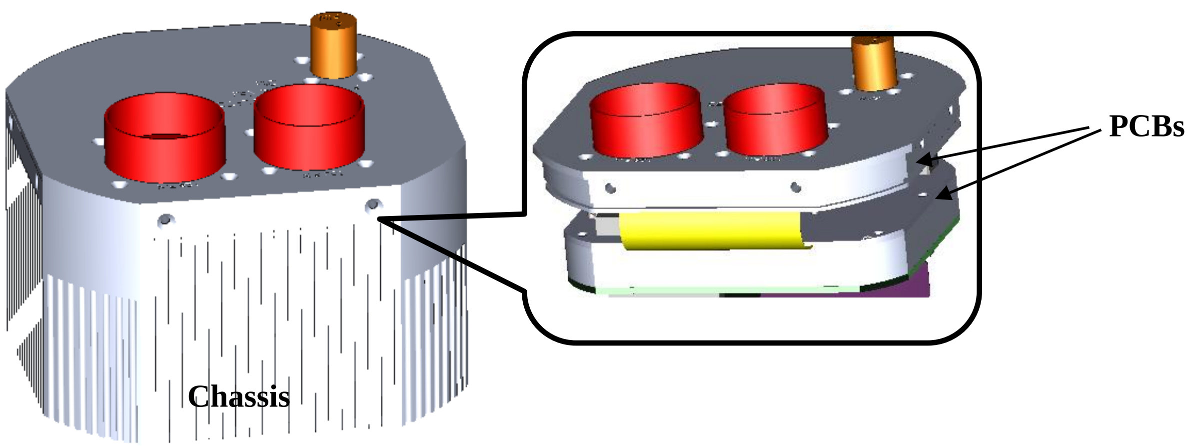


Figure 5.11 Package-4

Figure 5.12 shows the response location.

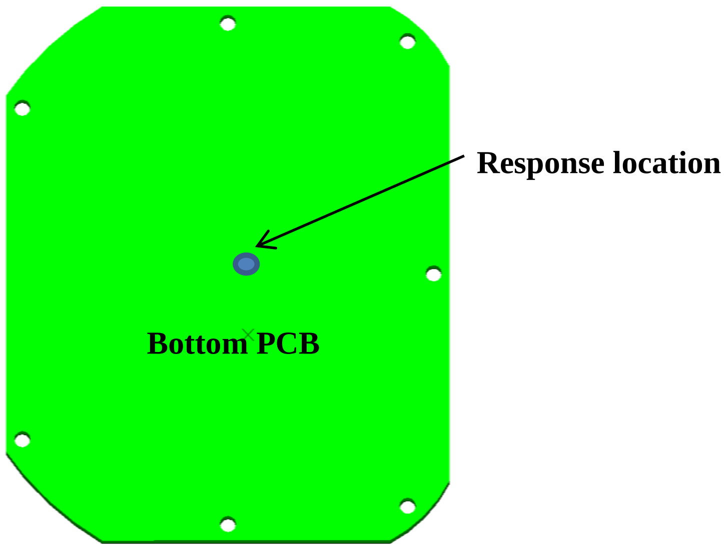


Figure 5.12 Response location

Vibration levels obtained using spring isolators are compared with that of screws in Table 5.5.

Table 5.5 Comparison of vibration responses (Package-4)

VIBRATION RESPONSE (INPUT: 8.1 g_{rms})	
With screws	**With spring isolators**
83.1 g_{rms}	4.1 g_{rms}

- Isolation effectiveness of proposed spring isolators is consistent in various packages having PCBs mounted in stacked configuration.

5.7 SUMMARY

- Spring isolators are developed and experimental evaluated in random vibration environment.
- These isolators are found to be very effective in reducing vibration responses in packages with stacked configuration.
- Care is exercised to ascertain the consistency associated with the proposed spring isolators.

CHAPTER 6

DESIGN OF METALLIC C-ISOLATOR FOR PACKAGES IN WEDGE GUIDE CONFIGURATION

6.1 INTRODUCTION

Design, development and experimental evaluation of metallic spring isolators for packages in stacked configuration is discussed in earlier chapters. This chapter brings out design of metallic C-isolator with an aim to reduce the vibration levels in packages of wedge guide configuration. Criteria for effective vibration isolation in spring isolator is commonly applicable for C-isolator also. Curved beam theory is adopted in designing the intended isolator. Design is validated with the aid of Finite Element Analysis (FEA) by comparing stiffness obtained using design with that of FEA. Further this chapter brings out the FE modelling aspects along with analysis results.

6.2 DESIGN CONFIGURATION

In the proposed configuration (Figure 6.1), wedge locks provided integral to Printed Circuit Board (PCB) are replaced with C-isolators (Figure 6.2). However it is planned to configure the mounting arrangement of C-isolator identical to that of wedge locks. Accordingly basic wedge guide configuration is retained.

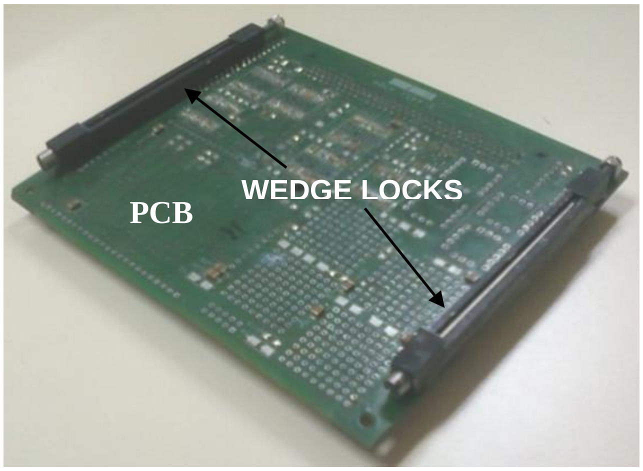


Figure 6.1 Configuration with wedge locks

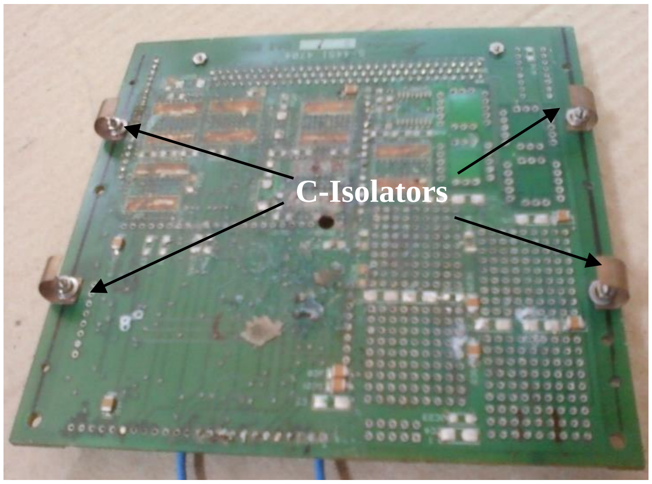


Figure 6.2 Configuration with C-isolators

Proposed configuration with C-isolators aims at reducing higher vibration levels.

6.3 DESIGN PHILOSOPHY

Mathematical model of a system mounted on isolator along with its motion Transmissibility (Expressed in section 4.3) information about excitation frequency i.e. first natural frequency of PCBs in wedge guide configuration is required to estimate the desired frequency of isolator.

A package (Package-5) consisting of PCBs mounted in wedge guide configuration considered for study is shown in Figure 6.3.

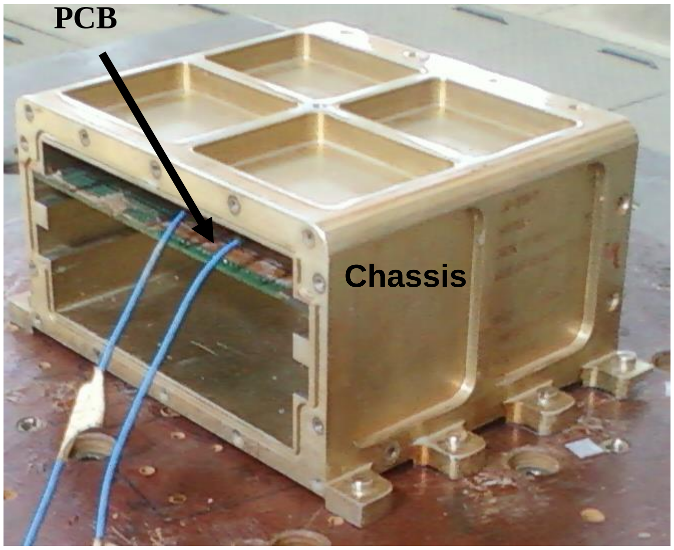


Figure 6.3 Package-5 in wedge guide configuration

Top view and bottom views of PCB with electronic components are shown in Figure 6.4 and Figure 6.5 respectively.

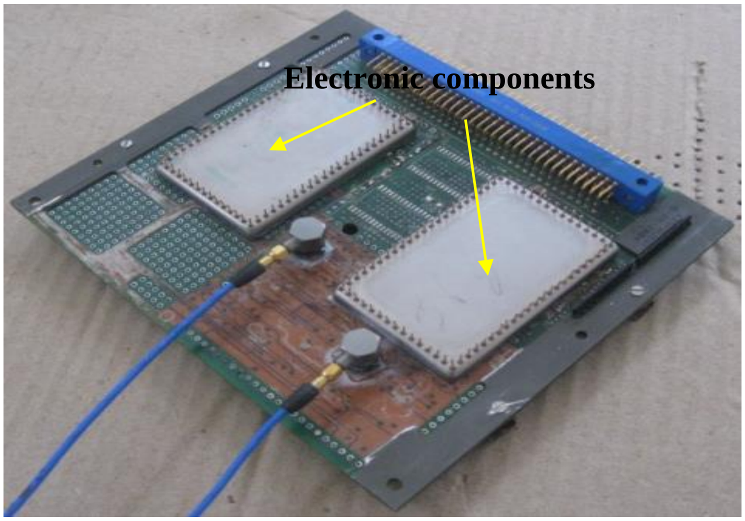


Figure 6.4 Top view of PCB – Component side

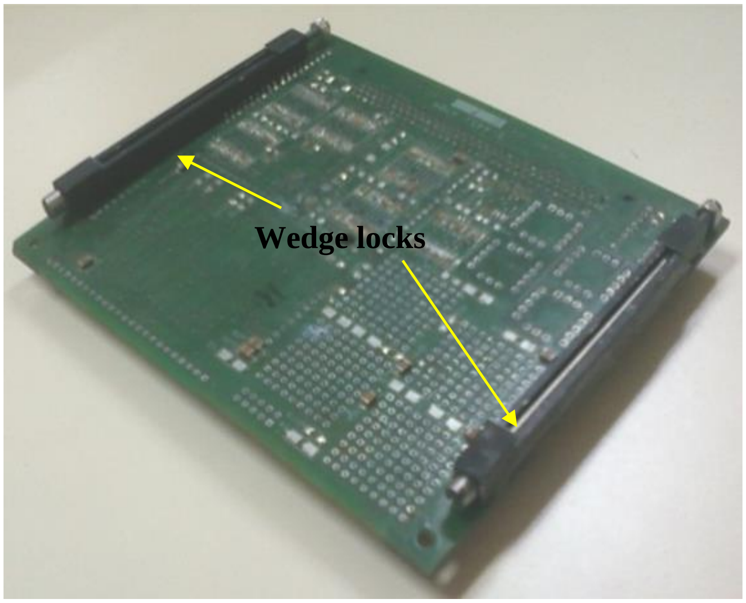


Figure 6.5 Bottom view of PCB – Wedge lock side

6.4 RANDOM VIBRATION TEST ON PACKAGE-5 IN WEDGE GUIDE CONFIGURATION WITH WEDGE LOCKS

Random vibration test is done on package-5 with a broad band random vibration input ($12.5g_{rms}$) between 20- 2000 Hz in a direction normal to the surface of the PCB. Test set up is shown in Figure 6.6.

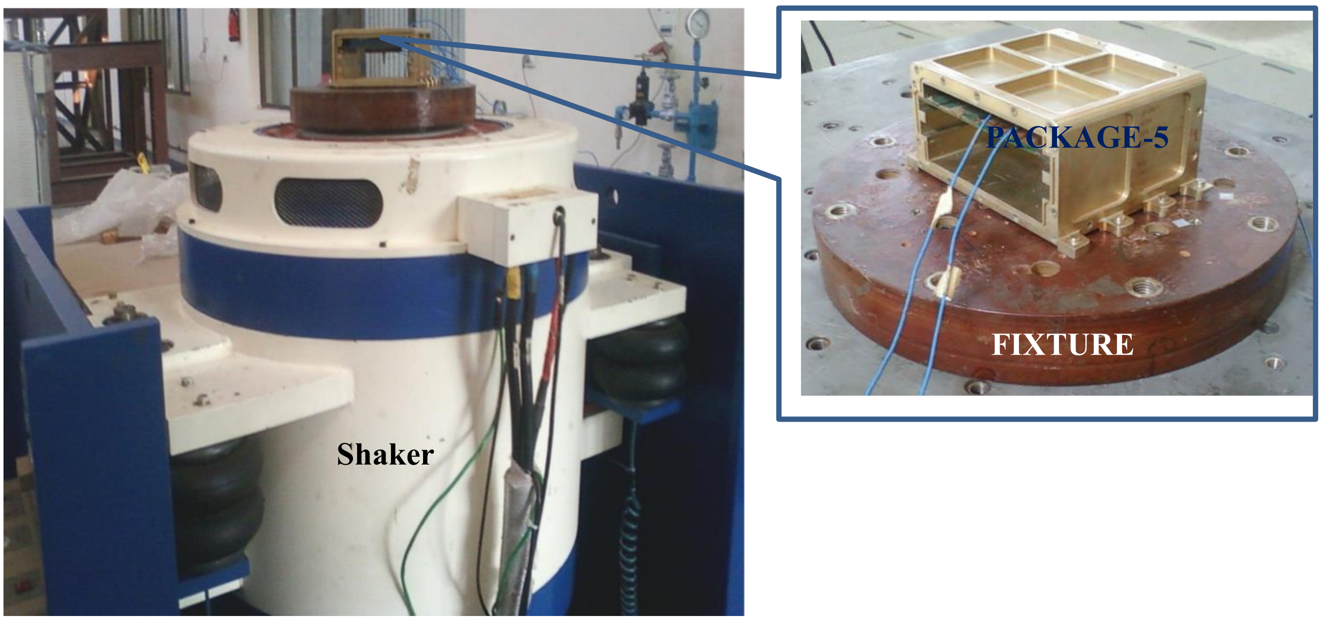


Figure 6.6 Test set up

Figure 6.7 shows accelerometer mounted on top PCB for measuring its vibration response.

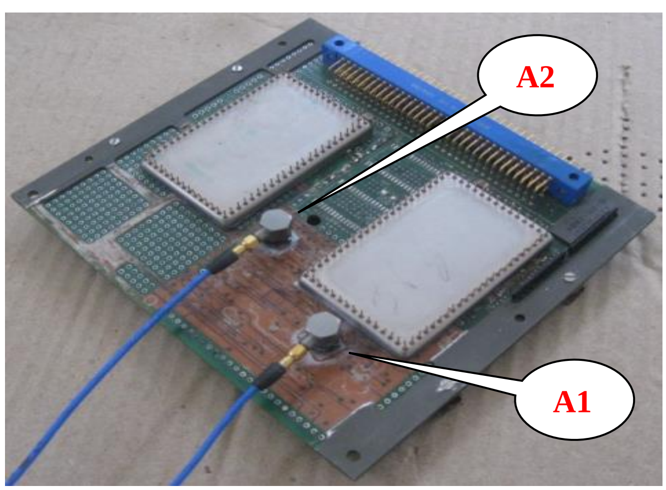


Figure 6.7 Accelerometer mounted on top PCB

Where:

A1 (Accelerometer-1): To measure the response at that specific location

Details of accelerometer are given below.

- Make: PCB
- Model number: 340A15
- Sensitivity: 10 mV/g
- Measurement range: + 500 g
- Frequency range: 1 Hz to 12000 Hz
- Type: Shear
- Weight: 2 grams

The vibration input (12.5g_{rms}) between 20- 2000 Hz given in Table 6.1 is considered for the test in a direction normal to the surface of the PCB (Vertical).

Table 6.1 Vibration input

FREQUENCY, Hz	**PSD, g^2/ Hz**
20	0.0044
100	0.11
1000	0.11
2000	0.0275

Vibration input mentioned in the above table is fed to the vibration shaker and the same is shown in form of graph in Figure 6.8.

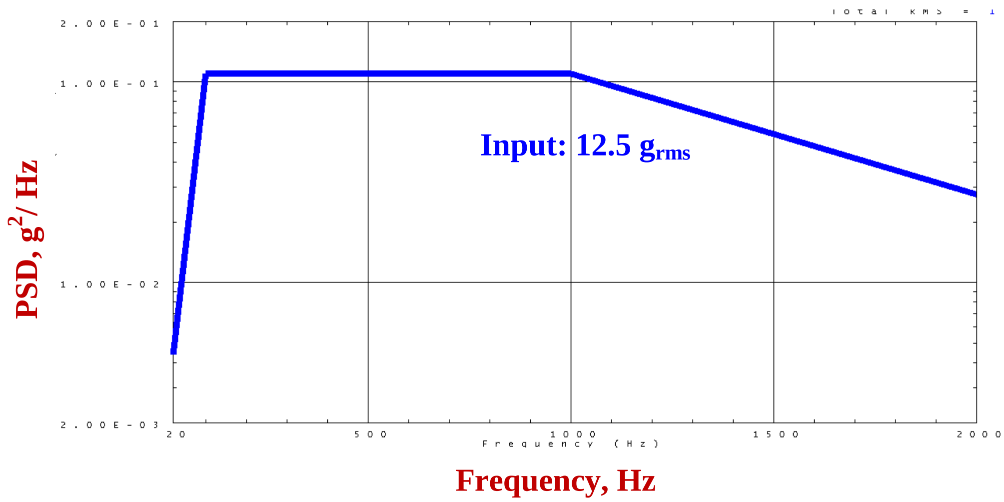


Figure 6.8 Vibration input spectrum

The vibration response spectrum of the PCB is shown in Figure 6.9.

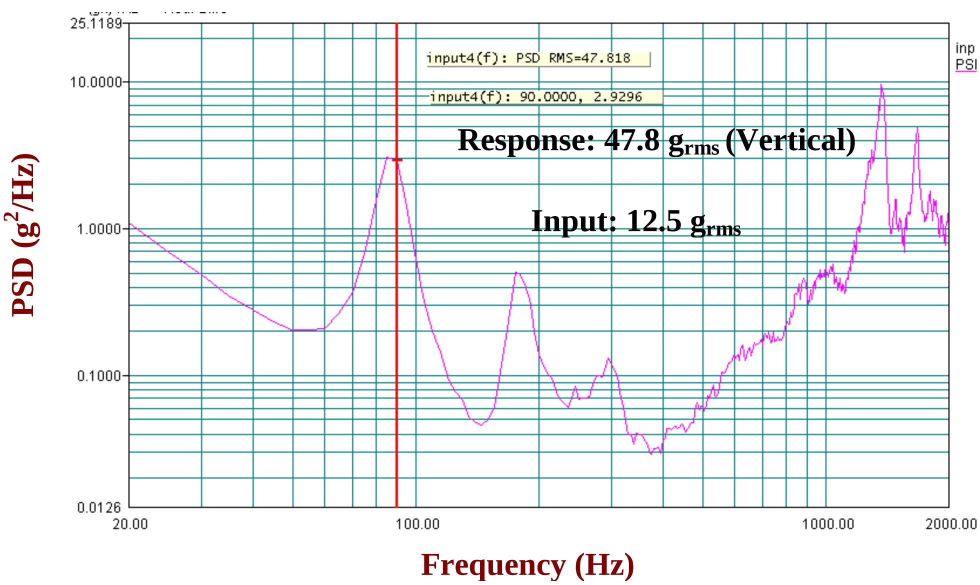


Figure 6.9 Vibration response spectra of package-5 in wedge guide configuration

- Maximum vibration response is 47.8 g_{rms} for the input of 12.5 g_{rms}.

- First natural frequency of PCB is 90 Hz.

6.5 DESIGN OF C-ISOLATOR

6.5.1 DESIGN INPUTS

Following inputs are considered for design.

- Weight of PCB = 0.365 Kg
- Width of the wedge lock = 7 mm
- Height of the wedge lock = 7 mm
- Cross section of the wedge lock = Square
- Number of wedge locks = 2 (For 1 PCB)

6.5.2 DESIGN CONSTRAINT

- Proposed C-isolator should replace wedge lock one to one with respect to size.

6.5.3 DESIGN OF C-ISOLATOR

Size of C- isolator (Width and height) is considered identical to that of wedge lock to meet the above mentioned design constraint. Beryllium copper is considered to be the material for proposed C-isolator as it possesses high damping and also resistance to fatigue. From the design philosophy, it is understood that lesser the natural frequency of isolator more will be the isolation. It is known that

$$\text{Circular natural frequency}, \omega_n = 2\pi f_n = \frac{K}{m} \tag{6.2}$$

Where

K: Stiffness of isolator (N/m)

m: Mass on each isolator (kg)

From this relation to achieve higher isolation the stiffness of isolator should be lowest possible. So it is thought of replacing existing cross-section of wedge lock i.e square with C-section for isolator.

From the frequency relation stated in design philosophy

$$\text{Desired natrual frequency of the isolator, } f_n < \frac{f}{\sqrt{2}} < \frac{90}{\sqrt{2}} < 63\,\text{Hz}$$

From the above natural frequency of the isolator is considered as 45 Hz. It is planned to mount four such isolators beneath the PCB. Desired stiffness of isolator is obtained by substituting frequency and mass on each isolator in equation 6.2. As the cross section of the proposed isolator is identical to that of a curved beam, deflection theory of curved beams is considered for design. A thin curved beam is characterized by a beam depth which is small compared to the radius of curvature as shown in Figure 6.10.

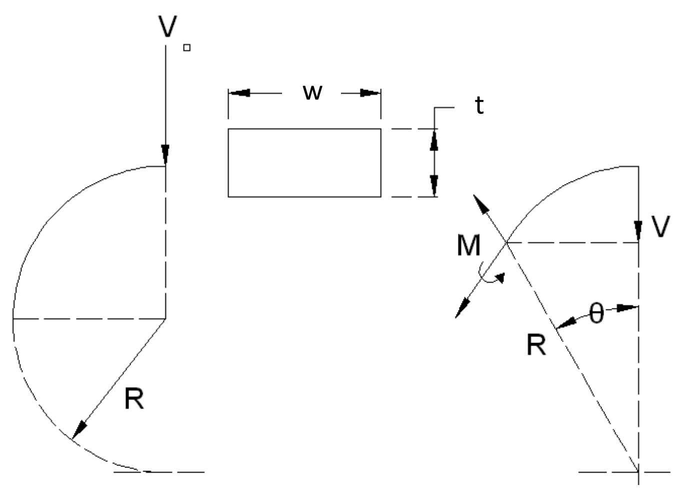


Figure 6.10 Thin curved beam

A thin curved beam is typified by the quadrantal cantilever as illustrated in Figure 6.11. Its radius is R and it is subjected to end load, ‘V’. Due to symmetry half part (quadrantal cantilever) is considered for the analysis. Castigliano's theorem is applied directly, the angle θ is usually a more convenient independent variable than the distance ‘s’ when it comes to curved beams. Thus in the sketch, at the general free body's cut end defined by θ, the bending moment required for equilibrium is:

$$M = VR \sin\theta.$$

and taking $s = R\theta$

For constant EI, deflection component in loading direction can be given as

$$y = \frac{1}{EI}\int_0^L M \frac{\partial M}{\partial V} dS = \frac{1}{EI}\int_0^{\frac{\pi}{2}} (V\,R\,sin\theta)\,R\,sin\theta\,R\,d\theta = \frac{\pi V R^3}{4EI} \qquad (6.3)$$

Then deflection for the half circle can be given as

$$2y = \frac{\pi V R^3}{2EI} \qquad (6.4)$$

Where

V = Weight of the unit

From which stiffness of the curved beam is expressed as follows

$$K = \frac{\text{Load}}{\text{Deflection}} = \frac{2EI}{\pi R^3} \qquad (6.5)$$

In which

E: Young's modulus of isolator material (Beryllium copper) = 2.237×10^{11} Pa

$$I: \text{Moment of inertia} = \frac{w\,t^3}{12} \qquad (6.6)$$

W: Width of the isolator

t: Thickness of the isolator

R: Radius of curvature

From the above formulae radius of curvature and thickness of isolator are evaluated while meeting the desired value of stiffness.

Further Stress analysis and deflection analysis are carried out for the C-isolator to determine its Factor of Safety (FOS) for different loading conditions. Load diagram of C-isolator is shown in Figure 6.11.

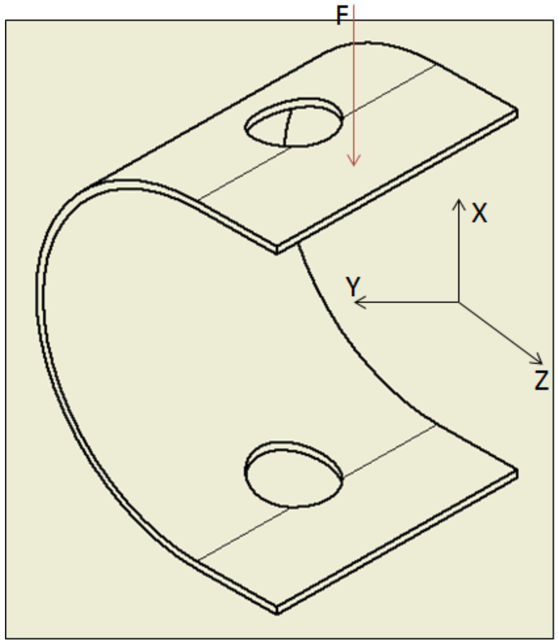


Figure 6.11 Load diagram of C-isolator

The following assumptions are taken.

- Force acting on the isolator is considered to be uniformly distributed over its flat face and hence, can be assumed to be completely acting on its centroid (Shown in Figure 6.12).
- The lower flat face of the isolator is rigidly held against a flat surface.
- Maximum local stress acting over a cross-section is considered to be uniformly distributed over that section.
- Neutral axis is assumed to pass through the centroid of the cross-section and the stress distribution at the cross-section is assumed to be linear as in a straight beam in view of thin-curved beam approximation.

Centroid of flat faces is located as given below.

The hole in the component for inserting a fastener overlaps with its flat surface (Shown in Figure 6.12). The overlapping segment subtends an angle of 91.15^{o} at the center of the hole (Diameter 2mm).

Distance of the centroid of this segment from the edge where the curvature begins = 0.122 mm

Distance of the centroid of flat face (without considering the overlap) from the edge where the curvature begins = 1.150 mm

Area of overlapping segment = 0.295 mm^2

Area of flat face (Without considering the overlap) = 16.1 mm^2

Thus, distance of overlapping segment from the edge where the curvature begins

$$= \frac{(16.1)(1.150) - (0.295)(0.122)}{16.1 - 0.295} = 1.169\, mm$$

Second moment of area about the neutral axis orthogonal to section centroid (Shown in Figure 6.12) is calculated as follows.

By perpendicular axis theorem, $I = \frac{(7)(0.2)^3}{12} + \frac{(7)^3(0.2)}{12} = 5.721\, mm^4$

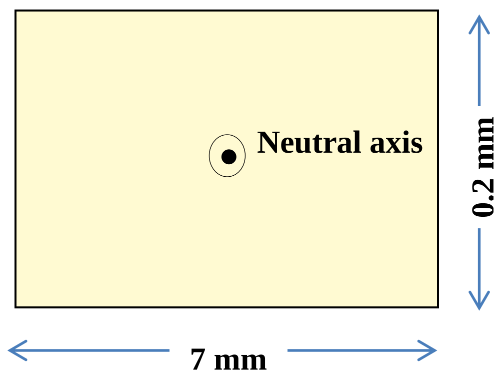


Figure 6.12 Component cross-section

Consider the bending moment due to the applied force on an arbitrary section on the circular arc. The value of the moment can be obtained as follows [18]

$$M = \begin{vmatrix} \hat{\imath} & \hat{\jmath} & \hat{k} \\ r_x & r_y & r_z \\ -F & 0 & 0 \end{vmatrix} = -(r_z)(F)\hat{\jmath} - (r_y)(F)\hat{k} \tag{6.7}$$

Where

r_x: x-component of position vector from a point on the section to the point of force application

r_y: y-component of position vector from a point on the section to the point of force application

r_z: z-component of position vector from a point on the section to the point of force application

Hence, the magnitude of the y-component of the moment would be maximized when calculated along the edge of a section lying in YZ-plane (Shown in Figure 6.13) and the magnitude of the z-component of the moment would be maximized when calculated along the outermost edge (Shown in Figure 6.14) for the section under consideration.

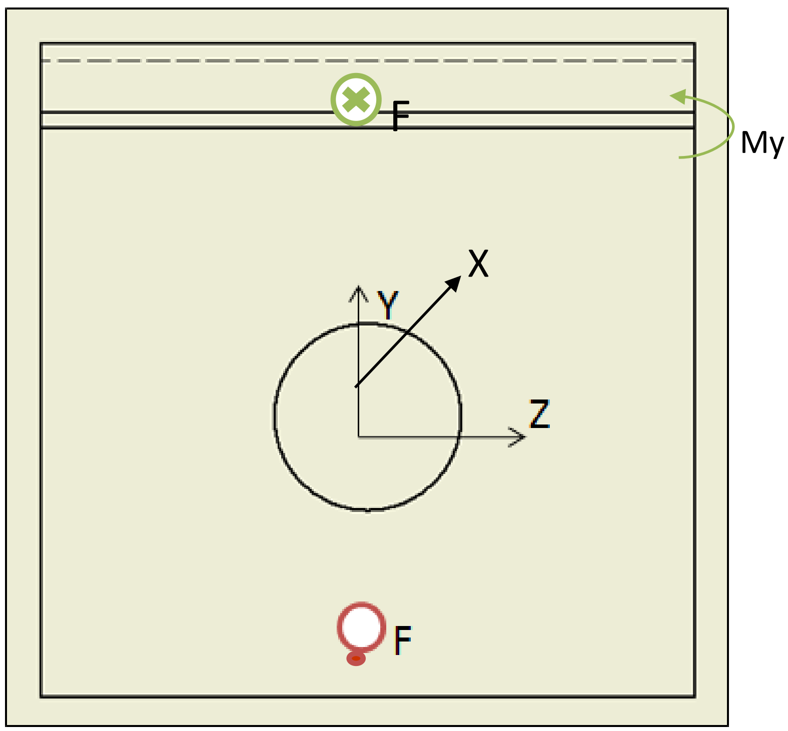


Figure 6.13 Projection of component section cut at an angle θ in YZ-plane

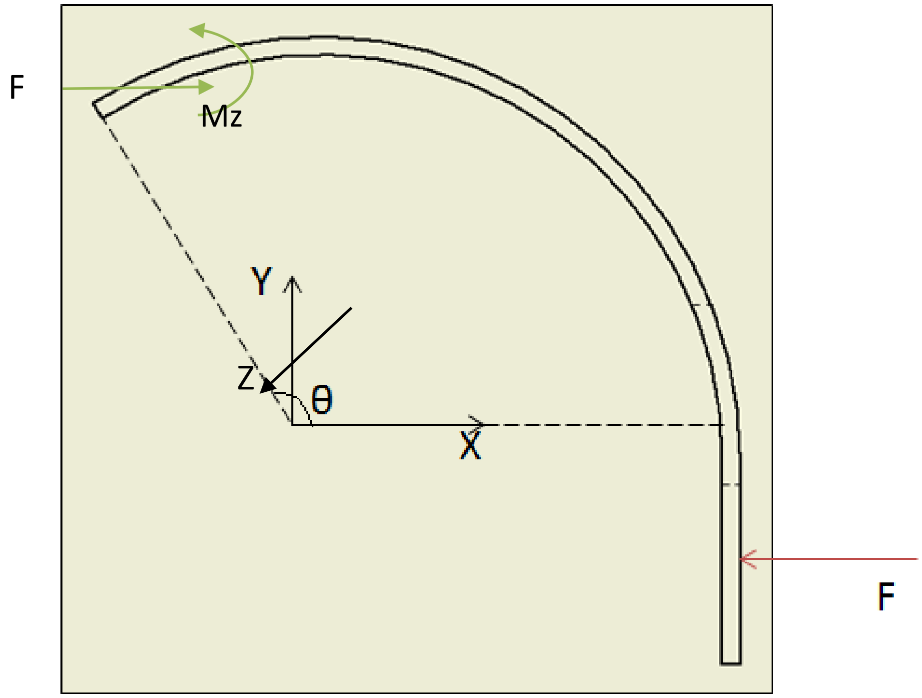


Figure 6.14 Projection of component section cut at an angle θ in XY-plane

These two bending moments would result in normal stresses acting over that section. Further, the internal force at a section would contribute to the shear and normal stresses at that section. The net normal stress on the section would be a vector sum of the aforementioned normal stresses. It is further assumed that the magnitude of the net normal stress to be the sum of the magnitudes of the individual stresses which is greater than the magnitude of the vector sum of the stresses. Hence, normal stress acting on a section at angle, θ is given as follows [18]

$$\sigma = \frac{(F)(\frac{w}{2})(\frac{t}{2})}{I} + \frac{(F)[(R+t)cos\theta+C](\frac{w}{2})}{I} + \frac{Fsin\theta}{w*t} \qquad (6.8)$$

Where

w: width of the section (7mm)

R: inner radius of curvature (4.5mm)

t: thickness of component (0.2mm)

C: distance of centroid of the flat face from the edge where the curvature begins (1.169mm)

F: applied force

Shear stress acting on a section at angle, θ is given as follows [18]

$$\tau = \frac{F\cos\theta}{w*t} \qquad (6.9)$$

Given the maximum and minimum values for F, maximum and the minimum normal and shear stresses acting at a section are calculated.

Further, maximum and the minimum Von Misses stress acting on that section are calculated. This would give the mean and the variable Von Misses stress acting on that section.

The Von Misses stress is expressed as follows:

$$\sigma_{von,max} = \sqrt{{\sigma_{max}}^2 + 3{\tau_{max}}^2} \qquad (6.10)$$

Where

σ_{max}: normal stress corresponding to $F=F_{max}$

τ_{max}: shear stress corresponding to $F=F_{max}$

$$\sigma_{von,min} = \sqrt{{\sigma_{min}}^2 + 3{\tau_{min}}^2} \qquad (6.11)$$

Where

σ_{min}: normal stress corresponding to $F=F_{min}$

τ_{min}: shear stress corresponding to $F=F_{min}$

$$\sigma_{von,mean} = \frac{\sigma_{von,max}+\sigma_{von,min}}{2} \qquad (6.12)$$

$$\sigma_{von,variable} = \frac{\sigma_{von,max}-\sigma_{von,min}}{2} \qquad (6.13)$$

Now, Soderberg's criterion is implemented to find the Factor of Safety (FOS) for given loading and material at that section.

$$FOS = \frac{1}{\frac{\sigma_{von,mean}}{S_y}+\frac{\sigma_{von,variable}}{S_f}} \qquad (6.14)$$

Where

S_y: Tensile yield strength

S_f: Fatigue strength for 10^8 cycles

FOS values for the sections on the circular arc vary with θ. For points on the flat faces as seen by the moment and stress calculations, the values obtained would be lesser than those on the circular arc resulting in higher FOS values. Hence, the variation of FOS with respect to θ is plotted to see that the critical section appears at 90^o for the following loading conditions:

Actual Loading

Load varies from

$$-0.15 * 12.5 * 9.81 * \frac{\sqrt{2}}{4} = -6.503\ N \text{ to } -0.3 * 12.5 * 9.81 * \frac{\sqrt{2}}{4} = -13.006\ N$$

The negative sign denotes the compressive nature of the applied forces.

Critical section is at $\theta = 90^o$ & FOS$= 5.072$.

Variation of FOS with 'θ' for actual loading case is given in Table 6.2.

Table 6.2 Variation of FOS with 'θ' for actual loading case

Angle, θ (degree)	**Factor of safety (FOS)**
0	15.160
15	10.633
30	7.949
45	6.459
60	5.629
75	5.203
105	5.203
120	5.629
135	6.459
150	7.949
165	10.633
180	15.160

Repeated Loading

Load varies from

$$0 \text{ to } -0.3 * 12.5 * 9.81 * \frac{\sqrt{2}}{4} = -13.006\ N$$

Critical section is at $\theta = 90^o$ & $FOS = 4.423$.

Variation of FOS with 'θ' for repeated loading case is given in Table 6.3.

Table 6.3 Variation of FOS with ‘θ’ for repeated loading case

Angle, θ (degree)	**Factor of safety (FOS)**
0	13.221
15	9.273
30	6.933
45	5.633
60	4.909
75	4.538
105	4.538
120	4.909
135	5.633
150	6.933
165	9.273
180	13.221

Completely Reversible Loading

Load varies from

$$0.3 * 12.5 * 9.81 * \frac{\sqrt{2}}{4} = 13.006\ N \text{ to } -0.3 * 12.5 * 9.81 * \frac{\sqrt{2}}{4} = -13.006\ N$$

Critical section is at $\theta = 90^o$ & FOS= 3.522.

Variation of FOS with ‘θ’ for completely reversible loading case is given in Table 6.4.

Table 6.4 Variation of FOS with ‘θ’ for completely reversible loading case

Angle, θ (degree)	**Factor of safety (FOS)**
0	10.528
15	7.384
30	5.520

45	4.486
60	3.909
75	3.613
105	3.613
120	3.909
135	4.486
150	5.520
165	7.384
180	10.528

Variation of FOS with ‘θ’ for all three loading cases are shown in Figure 6.15.

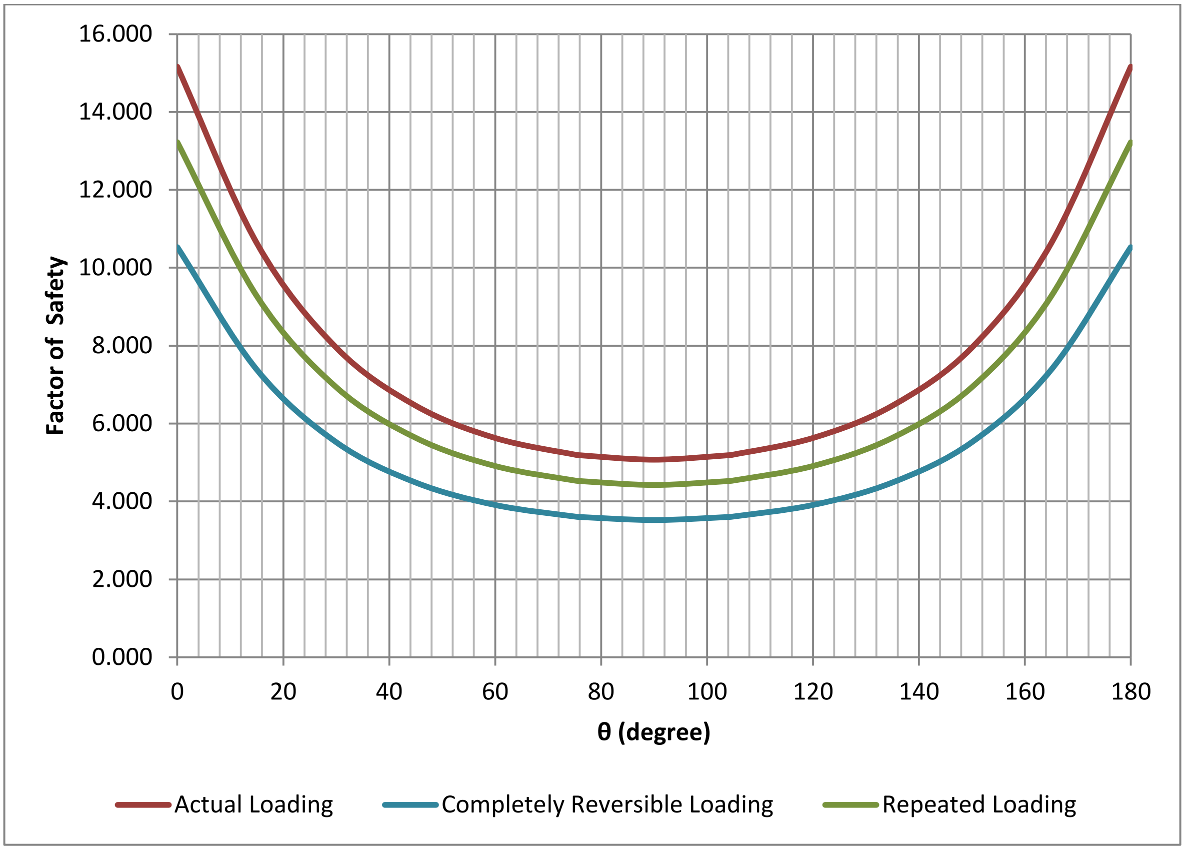


Figure 6.15 Variation of FOS with ‘θ’ for all three loading cases

- From the above exercise it is evident that the design is safe to take the expected loading for the designated number of cycles ($<10^8$).

Outcome of design calculations are summarized in Table 6.5.

Table 6.5 Design parameters of C-isolator

Sl. No.	**Design parameter**	**Value**
1.	Width	7 mm
2.	Height	7 mm
3.	Cross section	C
4.	Radius of curvature	4.5 mm
5.	Thickness	0.2 mm
6.	Stiffness	7294 N/m

A 3D CAD model of the C-isolator is realized in SOLDWORKS software from the above design parameters and the same is shown in Figure 6.16.

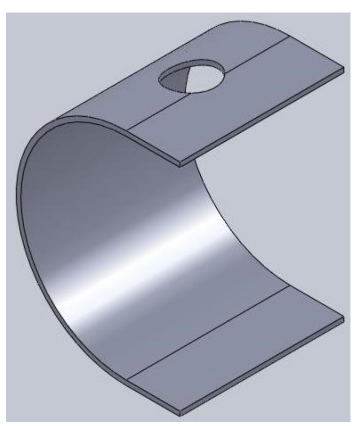


Figure 6.16 3D CAD model of C-isolator

6.6 VALIDATION OF DESIGN THROUGH FINITE ELEMENT ANALYSIS (FEA)

The design calculations are validated by carrying out Finite Element Analysis (FEA) in commercial software package ANSYS. Basic approach adopted for carrying out FE analysis is shown in Figure 6.17.

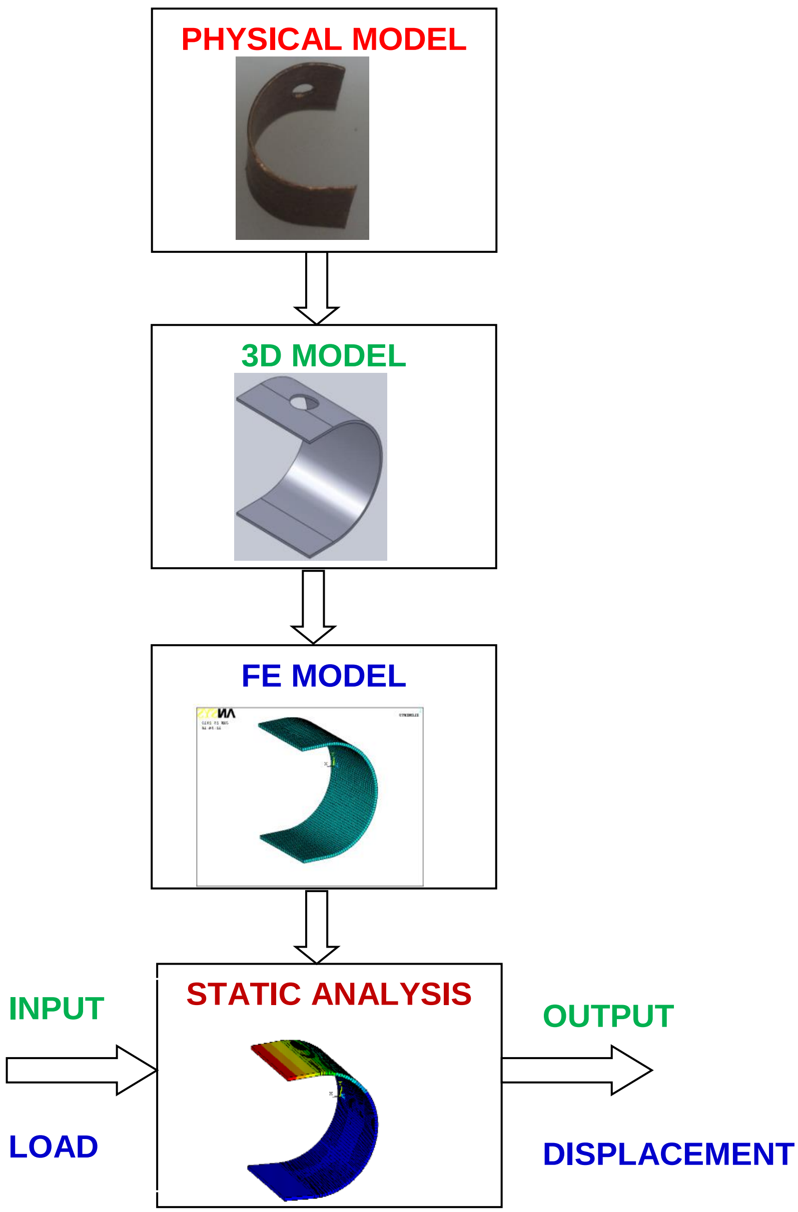


Figure 6.17 Approach for FE analysis

Static analysis is carried out to obtain displacement against unit load of 1 N. Accordingly stiffness value is calculated from displacement thus obtained from analysis. Objective of FE analysis is to validate the design by comparing the stiffness values obtained using design calculations with that of FEA. Geometry of isolator is modeled (2-D model) and then discretized with linear four nodded quadrilateral shell elements (SHELL63). SHELL63 element has six Degrees of Freedom (DOF) per node and it can take in-plane and bending loads.

6.6.1 MATERIAL PROPERTIES

Properties of beryllium copper (Material of C-isolator) given in Table 6.6 are considered for analysis.

Table 6.6 Material properties

Sl. No.	**Property**	**Value**
1.	Young's modulus, E	223 GPa
2.	Poisson's ratio, μ	0.3
3.	Shear modulus	80 GPa

6.6.2 BOUNDARY CONDITIONS

- Bottom side of the isolator where it touches the chassis is constrained for all DOF in the FE model.
- In addition to this middle portion (Line) where it touches the side wall of the chassis is also constrained for all DOF.

6.6.3 LOAD

A load of 1 N is applied on the nodes which are in the area of isolator which comes in contact with chassis on top side. FE model with boundary conditions and load is shown in Figure 6.18.

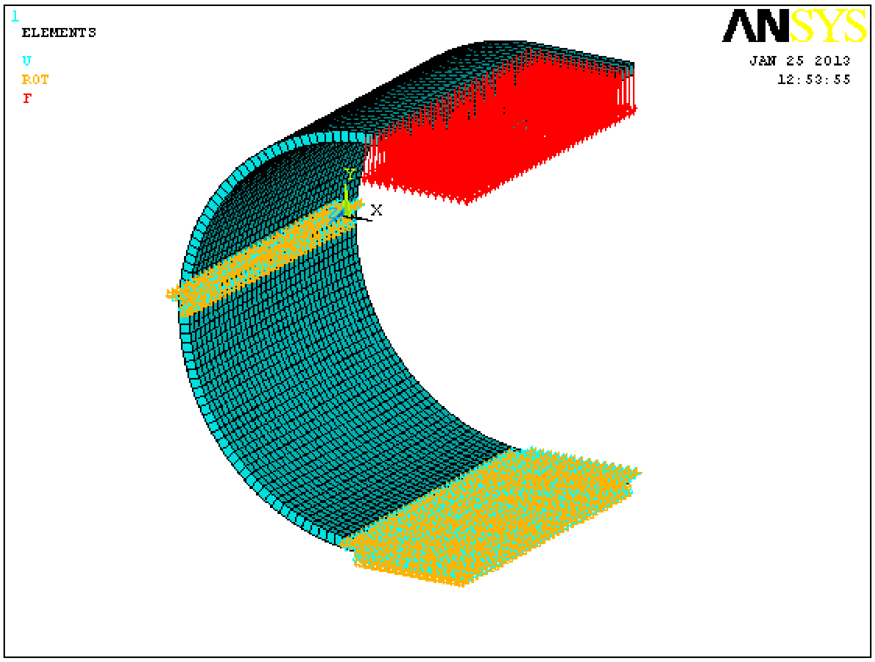


Figure 6.18 FE model

6.6.4 STATIC ANALYSIS

Static analysis is carried out in ANSYS software to obtain displacement. Displacement plot is shown in Figure 6.19.

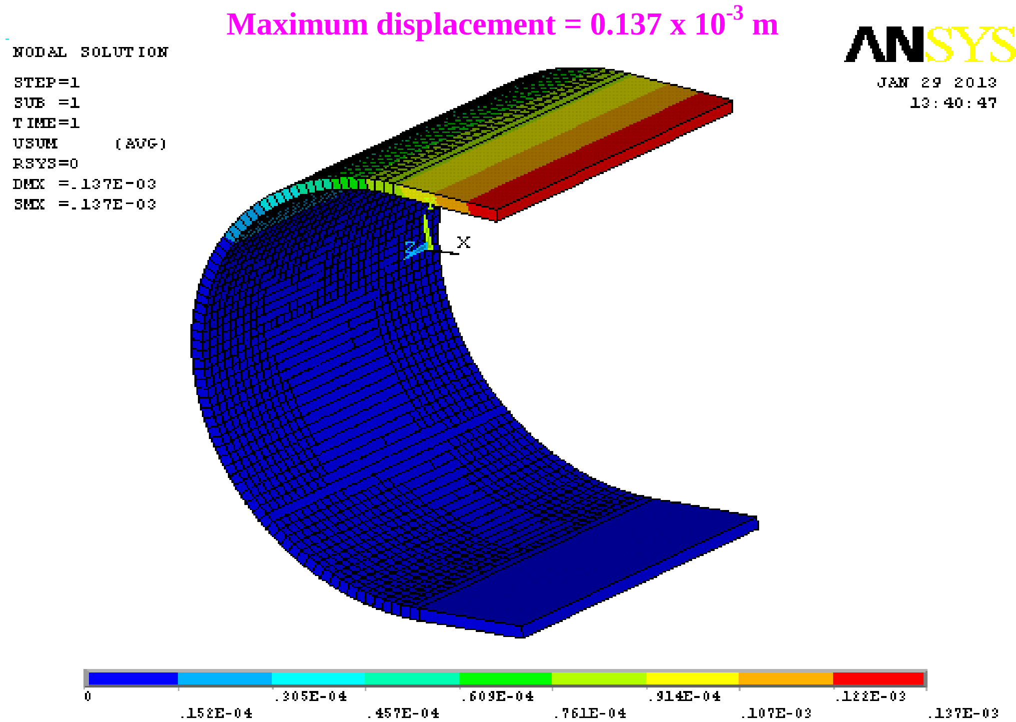


Figure 6.19 Displacement plot

From the analysis maximum displacement is observed to be 0.137 x 10^{-3} m. From displacement, stiffness is calculated as follows.

$$\text{Stiffness} = \frac{\text{Load}}{\text{Displacement}} = \frac{1}{0.137 \times 10^{-3}} = 7299 \text{ N/m}$$

Stiffness obtained from design calculation is compared with that of FEM in Table 6.7.

Table 6.7 Comparison of stiffness between design calculation and FEA

Method	**Stiffness**
Design	7294 N/m
FE analysis	7299 N/m

- Stiffness obtained from design calculation is observed to be in good agreement with that of FEM validating design.

6.7 SUMMARY

- C-isolators are designed which are meant for reducing vibration levels in packages with wedge guide configuration.
- Design is validated with the aid of Finite Element Analysis (FEA).

CHAPTER 7

DEVELOPMENT AND EXPERIMENTAL CHARACTERIZATON OF METALLIC C-ISOLATORS

7.1 INTRODUCTION

Metallic C-isolators are designed for packages in wedge guide configuration and the design parameters are summarized in earlier chapter. Once after ascertaining the design requirement, C-isolator is developed as per the dimensions from beryllium copper sheet. This chapter deals with experimental evaluation of the intended C-isolator in random vibration environment. Isolation effectiveness of the said isolators is quantified and consistency of the same is studied for packages in wedge guide configuration.

7.2 DEVELOPMENT OF C-ISOLATOR

C-isolator is manufactured with dimensions evolved as an outcome of design. Beryllium copper sheet is cut to the desired dimensions. From this sheet C-isolator is evolved through metal forming process.

C-isolator thus fabricated is shown in Figure 7.1.

Figure 7.1 C-isolator

7.3 RANDOM VIBRATION TEST ON PACKAGE-5 IN WEDGE GUIDE CONFIGURATION WITH C-ISOLATORS

Random vibration test is carried out in order to quantify the reduction of vibration with C-Isolators developed. Four such isolators are attached to the bottom side of PCB in place of wedge lock. Package-5 with C-isolators is mounted on shaker and excited with a broad band random vibration input. Test set up is shown in Figure 7.2.

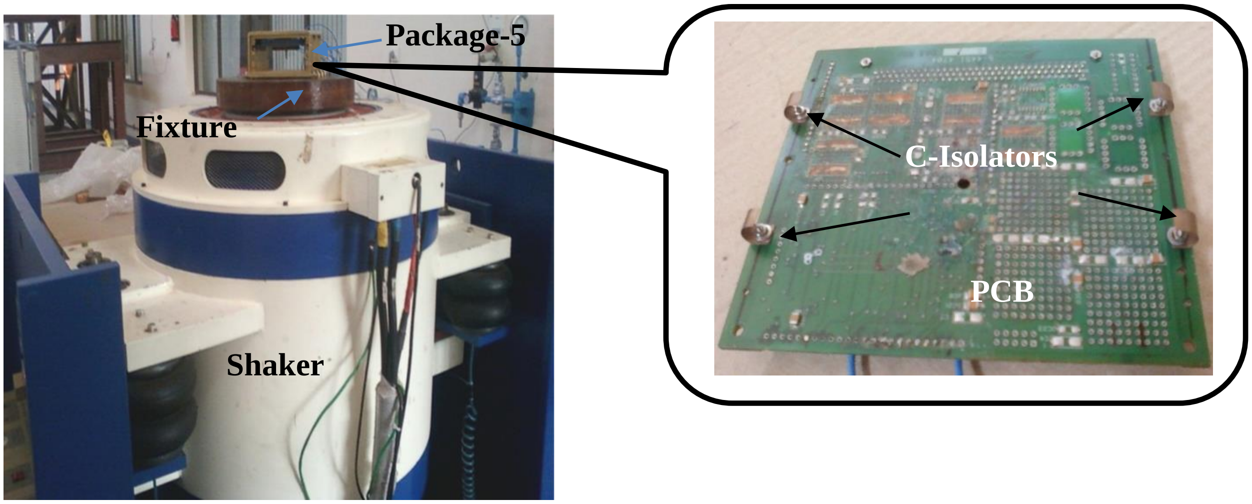


Figure 7.2 Test set up

Figure 7.3 shows accelerometer mounted on top PCB for measuring its vibration response.

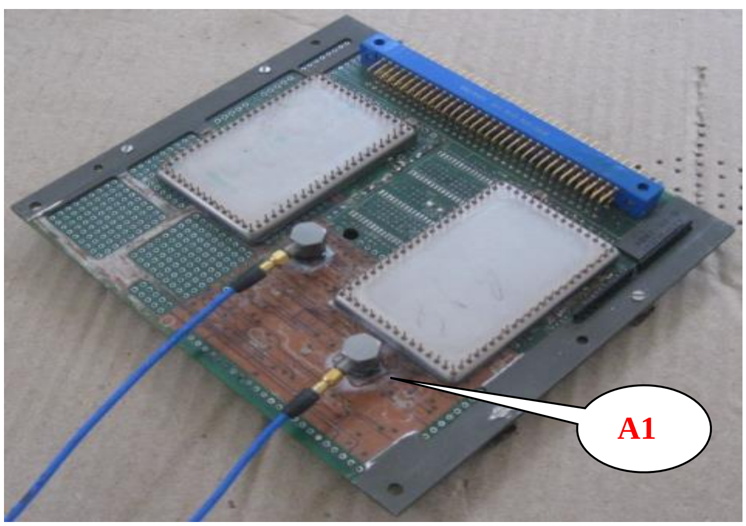


Figure 7.3 Accelerometers mounted on PCB

Where:

A1 (Accelerometer-1): To measure the response at that specific location

Details of accelerometer are given below.

- Make: PCB

- Model number: 340A15
- Sensitivity: 10 mV/g
- Measurement range: + 500 g
- Frequency range: 1 Hz to 12000 Hz
- Type: Shear
- Weight: 2 grams

Accelerometer details are common for all the packages.

The vibration input ($12.5 g_{rms}$) between 20- 2000 Hz given in Table 7.1 is considered for the test in a direction normal to the surface of the PCB (Vertical).

Table 7.1 Vibration input

FREQUENCY, Hz	**PSD, g^2/ Hz**
20	0.0044
100	0.11
1000	0.11
2000	0.0275

Vibration input mentioned in the above table is fed to the vibration shaker and the same is shown in form of graph in Figure 7.4.

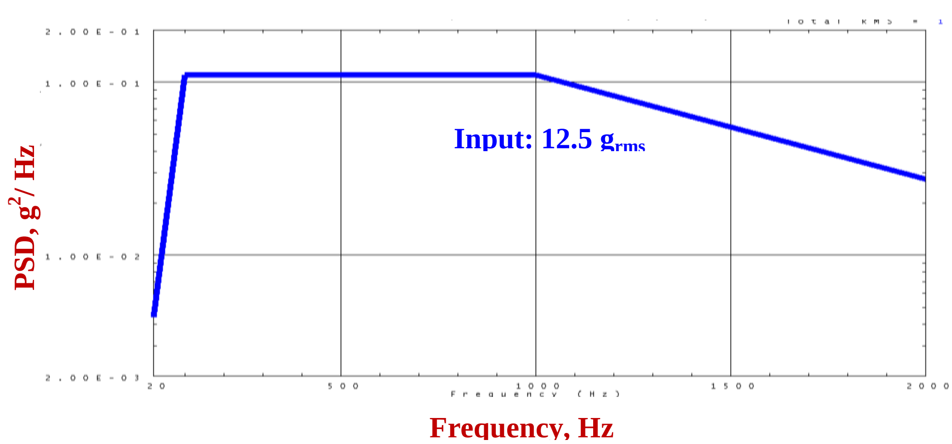


Figure 7.4 Vibration input spectrum

Vibration response spectrum of package-5 with C-isolators is shown in Figure 7.5.

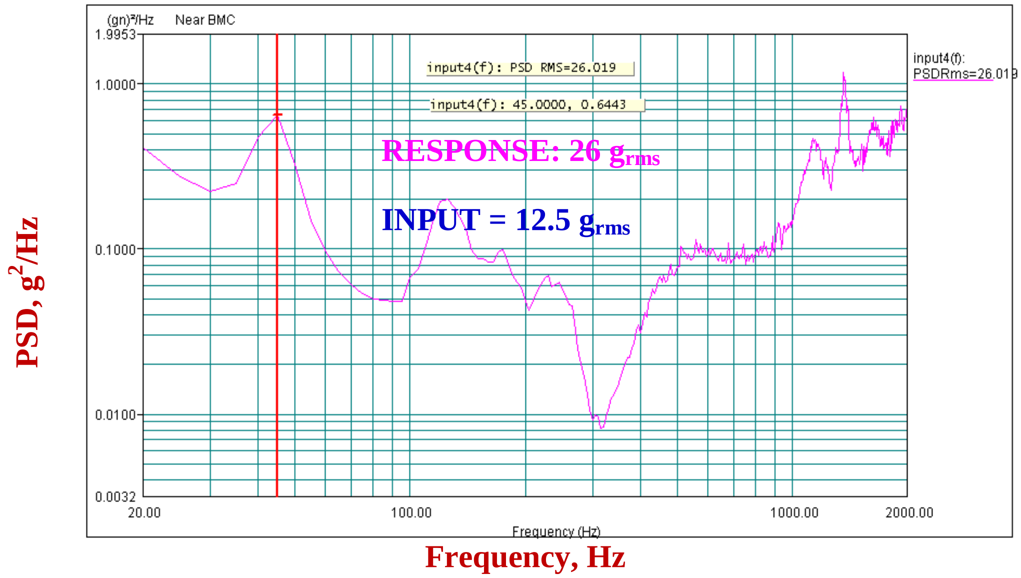


Figure 7.5 Vibration response spectrum of package-5 with C-isolators

- Maximum vibration response in package-5 is observed to be 26 g_{rms}.
- First natural frequency of the isolator is observed to be 45 Hz.
- However isolation is not found to be effective due to high frequency content predominant in the response as it can be seen from Figure 7.5.

7.4 IMPROVED DESIGN OF C-ISOLATORS

Reason for high frequency predominance in the vibration response is attributed to metal to metal contact between chassis side wall and the curved portion of the isolator. Hence it is proposed to modify the design of isolator accordingly and the modified configuration thus evolved is shown in Figure 7.6. In the improved design base portion is widened so as to impart contact between isolator and chassis only at the base portion of the isolator. Further improved design enabled to dispense with the contact between chassis side wall and the curved portion of the isolator in total. However material is scooped from the curved portion in order to compensate for the additional stiffness resulted due to increased width at base.

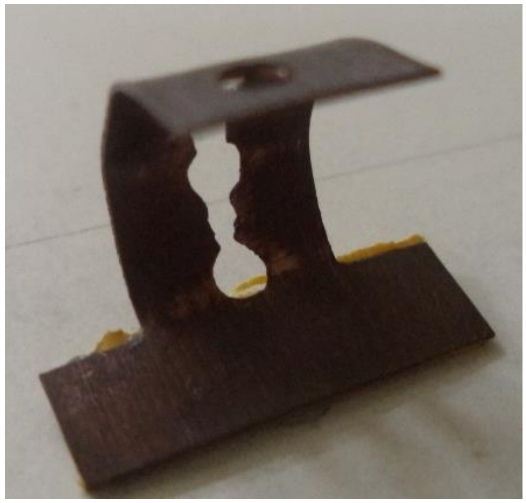


Figure 7.6 Improved C-Isolator

Random vibration test is carried out on package-5 in order to quantify the reduction of vibration with improved design of C-Isolator. Maximum vibration response observed on package-5 from test is shown in Figure 7.7.

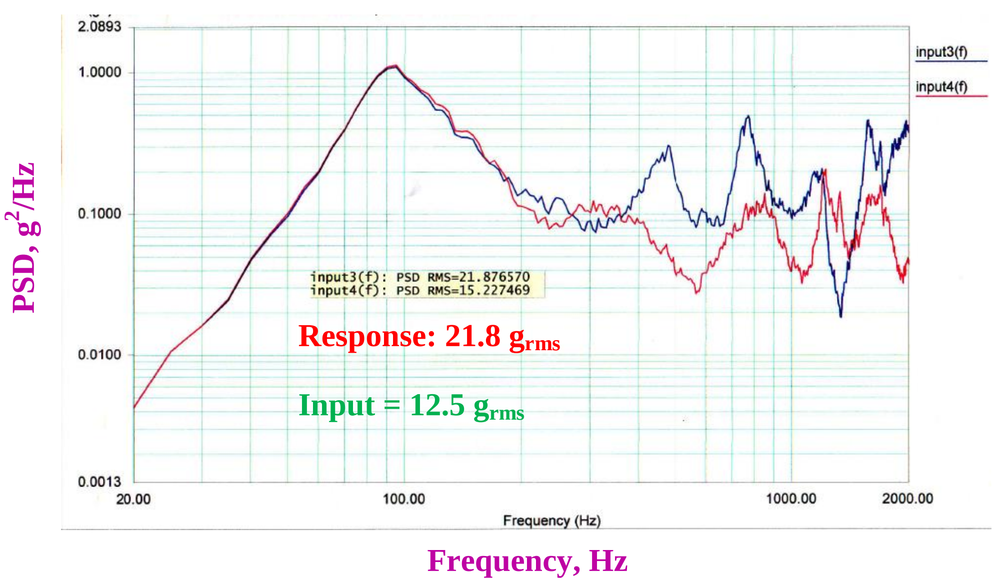


Figure 7.7 Vibration response spectrum of package-5 with improved C-isolator

Maximum vibration response obtained for package-5 with improved C-isolators is compared with that of wedge locks in Table 7.2.

Table 7.2 Comparison of vibration responses (Package-5)

	Test configuration		**Acceptable limit**
	Package-5 with wedge locks	**Package-5 with improved C-isolators**	
Maximum vibration response	47.8 g_{rms}	21.8 g_{rms}	30 g_{rms}

- With the proposed metallic C-isolators maximum vibration response got reduced from 47.8 g_{rms} to 21.8 g_{rms}.
- High frequency predominance noticed in earlier design got reduced.
- Maximum vibration response obtained with C-Isolators is less than the acceptable limit which satisfies the design goal.

7.5 PACKAGE-6 IN WEDGE GUIDE CONFIGURATION WITH C-ISOLATORS

In order to establish the consistency associated with the performance of the proposed C-isolator, another package in wedge guide configuration (Package-6 shown in Figure 7.8) is considered for the study. Package-6 consists of two PCBs arranged in wedge guide configuration. As electronic components mounted on top PCB (Vibration levels on bottom PCB are well within limits) are experiencing excessive vibration levels, wedge locks beneath the top PCB are replaced with C-isolators.

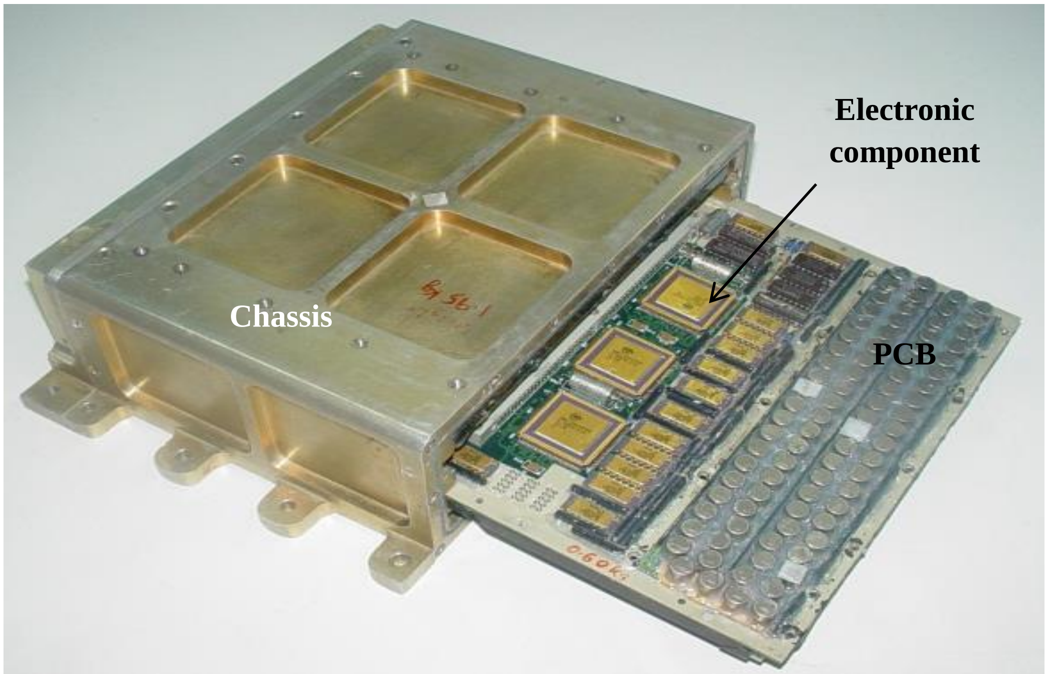


Figure 7.8 Package-6

Figure 7.9 shows the response location.

Figure 7.9 Response location

In order to ascertain the applicability of the proposed C-isolators, vibration levels obtained using C-isolators are compared with that of wedge locks in Table 7.3.

Table 7.3 Comparison of vibration responses (Package-6)

VIBRATION RESPONSE (INPUT: 8.9 g_{rms})	
With wedge locks	**With spring isolators**
45 g_{rms}	3.1 g_{rms}

7.6 PACKAGE-7 IN WEDGE GUIDE CONFIGURATION WITH C-ISOLATORS

Package-7 shown in Figure 7.10 consists of three PCBs arranged in wedge guide configuration. As electronic component mounted on one of the PCBs (Shown in Figure 7.10) are experiencing excessive vibration levels, wedge locks beneath that PCB are replaced with C-isolators.

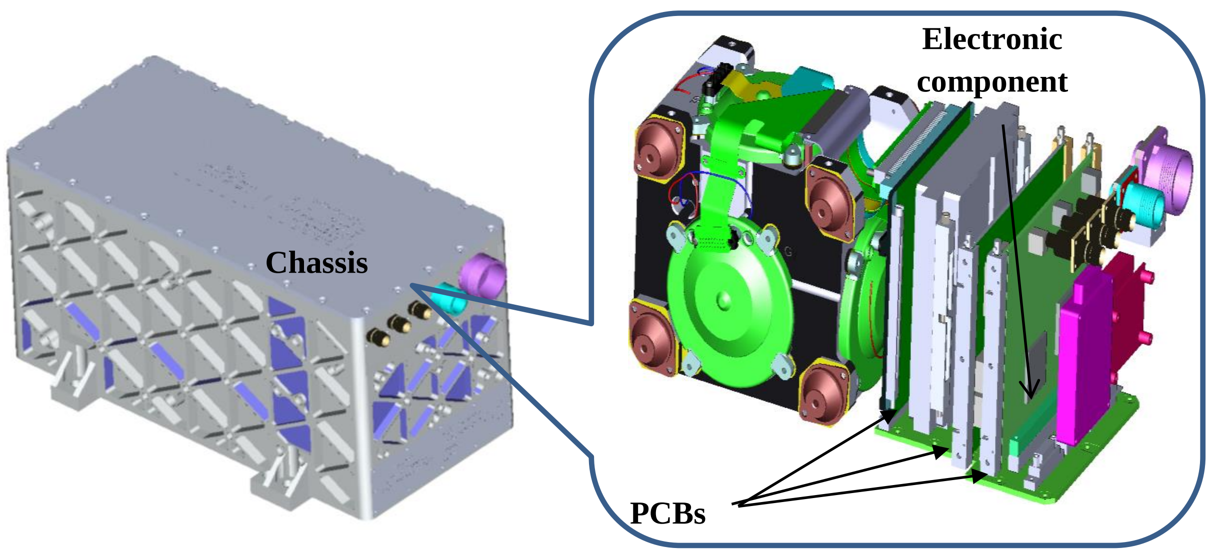


Figure 7.10 Package-7

Figure 7.11 shows the response location.

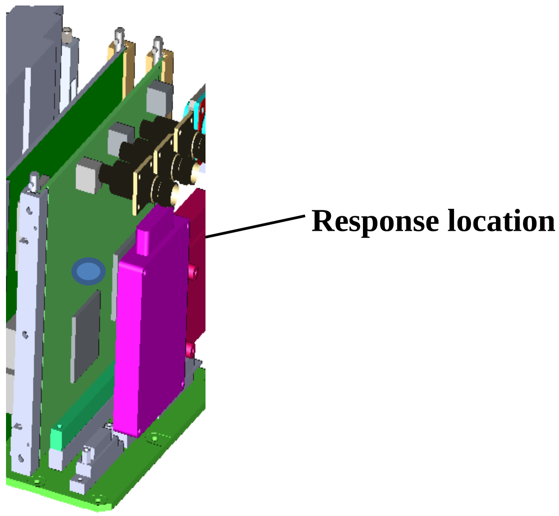


Figure 7.11 Response location

Vibration levels obtained using C-isolators are compared with that of wedge locks for package-7 in Table 7.4.

Table 7.4 Comparison of vibration responses (Package-7)

VIBRATION RESPONSE (INPUT: 8 g_{rms})	
With screws	**With spring isolators**
62.6 g_{rms}	2.6 g_{rms}

7.7 PACKAGE-8 IN WEDGE GUIDE CONFIGURATION WITH C-ISOLATORS

Similar attempt to verify the applicability of the proposed approach is made on package-8 shown in Figure 7.12 consists of seven PCBs arranged in wedge guide configuration. In this package wedge locks beneath one PCB on which mounted component is experiencing excessive vibration level are replaced with C-isolators.

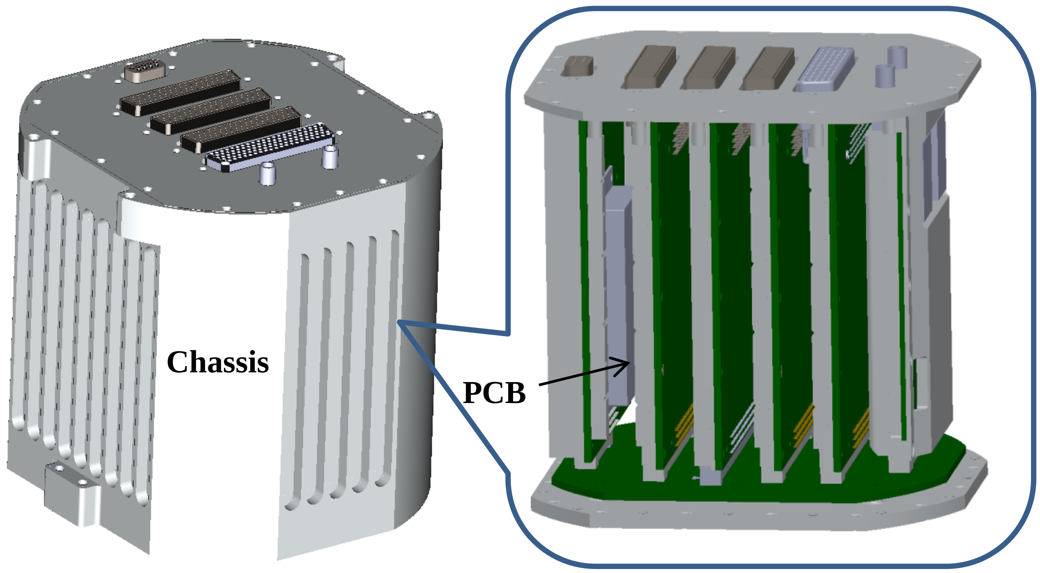


Figure 7.12 Package-8

Vibration levels obtained using C-isolators are compared with that of wedge locks for package-8 in Table 7.5.

Table 7.5 Comparison of vibration responses (Package-8)

VIBRATION RESPONSE (INPUT: 8.1 g_{rms})	
With screws	**With spring isolators**
71.5 g_{rms}	3.2 g_{rms}

- Isolation effectiveness of proposed C-isolators is consistent in various packages in Wedge guide configuration.

7.8 SUMMARY

C-isolators are developed and experimental evaluation of the same is carried out in random vibration environment. These isolators are found to be very effective in reducing vibration responses in packages in wedge guide configuration. Care is exercised to ascertain the consistency associated with the proposed C-isolators.

CHAPTER 8

SUMMARY AND CONCLUSIONS

8.1 SUMMARY

Primary scope of the current research study is to design, develop and experimentally evaluate metallic isolators for electronic packages for air borne vehicles with the aid of conventional experimental tools.

To begin with design criteria for effective isolation is evolved considering methodology identified from literature as a reference. Design constraints are identified and the associated governing relations are brought out. Subsequently isolators are designed and their dimensions are worked out. Further Graphical Use Interface (GUI) based software is developed in MATLAB for design of spring isolator. The necessary formulation, which is derived as part of design, is converted in form of code. The intention behind developing the software is to enable the designer to use it as a hand calculator, which just takes the inputs from the user and gives the output quickly. Design is validated by comparing the stiffness values obtained using calculations with that of Finite Element Analysis (FEA). Subsequently isolators are manufactured with dimensions evolved from design. In the later stage these isolators are experimentally evaluated in random vibration environment and isolation effectiveness is quantified. Both the isolators are found to be very effective in reducing vibration levels in their respective electronic packaging configurations. Efficacy of the isolators thus developed is methodically tested in real life packages (For both stacked configuration and wedge guide configuration) and there by established consistency.

8.2 CONCLUSIONS

The following are the conclusions drawn from the present work:

- To begin with spring isolators are designed which are meant for reducing vibration levels in packages with stacked configuration.
- Design procedure is transformed in form of a GUI based software using MATLAB which can aid as a hand calculator for designing spring isolator.
- Design is validated with the aid of FEA indicating the poise associated with the design procedure.
- Subsequently spring isolators are manufactured and experimentally evaluated in random vibration environment. These isolators are found to be very effective in reducing vibration responses in packages with stacked configuration.
- Care is exercised to ascertain the consistency associated with the proposed spring isolators.
- In later stage C-isolators are designed which are meant for reducing vibration levels in packages in wedge guide configuration.
- Curved beam theory is adopted to work out dimensions as part of design. From the calculations it is evident that the design is safe to take the expected loading for the designated number of cycles ($<10^8$). Design is validated with the aid of Finite Element Analysis (FEA).
- C-isolators are developed and experimental evaluation of the same is carried out in random vibration environment. These isolators are found to be very effective in reducing vibration responses in packages in wedge guide configuration. Care is exercised to ascertain the consistency associated with the proposed C-isolators.
- Metallic isolators evolved as an outcome of the present research work will serve as a protecting shield for two standard electronic packaging configurations against random vibration environment. These isolators stand as a promising alternative to problem prone commercial rubber isolators. Implementing C-isolators in wedge guide electronic packaging configuration, designer can make

use of entire space on Printed Circuit Boards (PCBs) very effectively for populating electronic components. Similarly with the induction of spring isolators in to the electronic package, designer can opt for stacked configuration as and when situation demands without having the fear of vibration problems.

8.3 SCOPE FOR FUTURE WORK

The following future work is proposed

- Application of major outcome of the present research i.e. metallic isolators for reducing vibration levels at electronic package level between package and sections of air borne vehicles.
- Evolution of metallic isolators for other environments like pyro shock having acceleration magnitudes of the order of 10,000g.
- Formulation of design procedure for configuring wire rope isolators for electronic packages.

REFERENCES

[1] Anil Kumar, P. and Jagadisan Krishnamoorthy, (2010), "'General guide lines for electronic packaging design'', Departmental report, pp. 33-34.

[2] Beijers, C. A. J. and De Boer, A., (2003), "Numerical modelling of rubber vibration isolators", 10th International Conference on Sound and Vibration, pp. 805-812.

[3] Boonen, R. and Sas, P., (2014), "Development of a vibration isolator with dry friction damping", Proceedings of ISMA, pp. 545-553.

[4] Bruno de Marne, (2007), "Active and Passive Vibration Isolation and Damping via Shunted Transducers", Ph.D thesis, Active Structures Laboratory, Department of Mechanical Engineering and Robotics.

[5] Byron Davis, J. and Ahmad Bayat, (2010), "Small Deviations and Big Failures in Vibration and Noise Isolation", Journal of sound and vibration, pp. 14-17.

[6] Collette C., Janssens S., Artoos K. and Hauviller C., (2010), "Active vibration isolation of high precision machines", Proceedings of Diamond Light Source, Vol. (1), pp. 1-5.

[7] Crede C.E., (1965), Vibration and Shock Isolation, 6th ed., John Wiley, New York.

[8] Dhiraj, V. S. and Niranjan, D.K., (2016), "Analysis of helical compression spring for estimation of fatigue life", Imperial journal of interdisciplinary research (IJIR), Vol. 2, pp. 2088-2093.

[9] Giaime, J., Saha, P., Shoemaker, D. and Sievers, L., (1996), "A passive vibration isolation stack for LIGO: Design, modeling, and testing", Review of Scientific Instruments, pp. 208 - 214.

[10] Hodges, C. H., (1983), "Vibration isolation from irregularity in a nearly periodic structure: Theory and measurements", The Journal of the Acoustical Society of America, Vol. 74, pp. 894-905.

[11] Hong, D., Hwang, H., Moo-Yeon, Yang-Keun, Duc-Thuan and Dong-Yeon., (2015), "New Electro-Magnetic Actuator for Active Vibration Isolators", International Journal of Precision Engineering and Manufacturing, pp. 209-212.

[12] Junyi Lee., (2016), "Vibration isolation with periodic structures", Ph. D. thesis, Imperial College London, Department of Mechanical Engineering.

[13] Kamesh, D., Pandiyan R. and Ashitava Ghosal., (2012), "Passive vibration isolation of reaction wheel disturbances using a low frequency flexible space platform", Journal of

Sound and Vibration, Vol. 331(6), pp. 1310–1330.

[14] Khurmi, R.S. and Gupta, J.K., (2005), A text book of machine design, 14th ed., Eurasia.

[15] Kim, S.M., Elliott S. J. and Brennan M. J., (2001), "Decentralized control for multichannel active vibration isolation", IEEE Transactions on Control Systems Technology, Vol. 9 (1), pp. 93-100.

[16] Lazutkin G.V., Ulanov A.M. and Boyarov K.V., (2014), “Comparison of mechanical characteristics of vibration isolators made of wire pressed materials”, International Journal of Engineering and Technology, Vol. 6, pp. 2201-2208.

[17] Li, Ku Nnian and Mitchell, Gohnert., (2010), “Lever mechanism for vibration isolation”, Applied Technologies and Innovations.

[18] Lukens, W. R., (1983), “Probabilistic fatigue life predictions of structural components in high cycle fatigue regime”, MS thesis, Naval post graduate school.

[19] Malowicki Mark G., (2000), “Active vibration isolation using an induced strain actuator with application to automotive seat suspensions”, MS thesis, Virginia Polytechnic institute and state university, Department of Mechanical Engineering.

[20] Mustafa, E. Levent and Kenan, Y. Sanliturk., (2003), “Characterisation of vibration isolators using vibration test data”, 21st International Congress on Sound and Vibration, pp.1-8.

[21] Pobedin, A., Kosov, O. and Shekhovtsov, K., (2011), “Vibration isolators test bench”, Journal of KONES Powertrain and Transport, Vol. 18 (1).

[22] Reid, J. R. , Bright, V. M. and Kosinski, J. A., (1998), "A micromachined vibration isolation system for reducing the vibration sensitivity of surface transverse wave resonators”, IEEE Transactions on Ultrasonics, Ferroelectrics, and Frequency Control, Vol. 45 (2), pp. 528-534.

[23] Rivin, E.I., (2006), “Vibration Isolation of Precision Objects”, 12th International Congress on Sound and Vibration, pp. 12-20.

[24] Robbert Voorhoeve, Annemiek van Rietschoten, Egon Geerardyn and Tom Oomen, (2015), “Identification of High-Tech Motion Systems: An Active Vibration Isolation Benchmark”, 17th IFAC Symposium on System Identification, pp. 1250-1255.

[25] Run-pu Li, Cheng-bin Du, Fei Guo, Guo-jun Yu and Xiao-guo Lin, (2015), “Performance of Variable Negative Stiffness MRE Vibration Isolation System”, Advances in Materials

Science and Engineering.

[26] Taksehi, M., (2010), “Vibration Isolation System Using Negative Stiffness”, Vibration Control, pp. 380.

[27] Thomson, W.T., (1993), Theory of Vibration with Applications, 4th ed., Prentice-Hall, Englewood Cliffs.

[28] Wahl, A.M., (1963), Mechanical springs, 2nd ed., McGraw-Hill, New York.

[29] Wayne Tustin (2005), Random vibration and shock testing, 4th ed., Equipment reliability institute.

[30] Xiuting Sun, Xingjian Jing, Li Cheng and Jian Xu, (2014), “Effect of coulomb friction on nonlinear vibration isolation system”, 21st International Congress on Sound and Vibration, pp. 1-8.

[31] Yann Frizenschaf, Siobhan Giles, Jack Miller, Thomas Pitman, Christopher Stapleton, Benjamin Cazzolato and Will Robertson, (2011), “Development of a magnetic levitation vibration isolator using inclined permanent magnet springs”, Proceedings of Acoustics, pp. 1-7.

[32] Zhao Wei, Qi Kai, Zhang Weihua and Fu Zhengxin, (1992), “Test and analysis of bridge vibration”, Earthquake engineering, Tenth world conference.

[33] Zhu, Tao., Cazzolato, Ben., Robertson, William. And Zander, Anthony., (2013), “Vibration isolation using six degree-of-freedom quasi-zero stiffness magnetic levitation”, Journal of Sound and Vibration.

OUTCOME OF THE PRESENT WORK

Part of the work is published as the following patent:

1. Anil Kumar P, "A vibration isolator, method of mounting and manufacturing the same", Application number: 2794/DEL/2014, Date of filing application: 30/09/2014, Publication date: 01/04/2016, Journal No. 14/2016.

Part of the work is published as the following papers:

1. Anil Kumar P, and Bangarubabu P., (2016), "Development and experimental characterization of metallic spring isolators for electronic packages of air borne vehicles," **Journal of Experimental & Applied Mechanics,** Vol. 7 (2), pp. 9-18.

2. Anil Kumar P, and Bangarubabu P, (2016), "Design and Development of Metallic Isolators for Wedge Lock Mounted Printed Circuit Boards (PCBs)," **Journal of Embedded System & Applications,** Vol. 4 (3), pp. 27-34.

Biographical Sketch of the Author

The author P Anil Kumar was born on August 10, 1975 as eldest son to (Late) Shri Pillarisetty Raghava Rao and Smt. Pillarisetty Ahalya at Pedana, Andhra Pradesh. He has got married to Smt. Mukkamalla Viswa Jyothi on February 26, 2010 and blessed with one daughter Chi. Pillarisetty Kamaneeya on November 07, 2017.

He has acquired his B.Tech (Mechanical Engineering) degree from JNTU, Hyderabad with first class with distinction in the year 1998. He joined in Defence R&D Organization (DRDO) in the year 2000.

He started his career as Scientist 'B' in the area of structural dynamic analysis of sub systems and systems of air borne vehicles using finite element method. At present he is Scientist 'F' and Head for structural dynamics and acoustic testing (SDAT) group.

He has provided solutions for vibration problems in various electronic packages used in air borne vehicles. Unique solution provided by him for vibration problem in one of the vehicle interface unit gave sigh of relief for 150 units. From the wide spectrum of solutions provided he has formulated guidelines for structural design of electronic packages and circulated among electronic designers of his laboratory. He has vast experience in conducting ground resonance testing of full scale air borne vehicle, structural load testing and modal testing.

He has received lab technology group award in the year 2008, lab scientist of the year award for the year 2010, Agni group award for excellence in self-reliance for the year 2014, Agni group award for excellence in self-reliance for the year 2016 and DRDO group award for performance excellence for the year 2016.

He has 15 publications out of which two are related to the present work and one patent related to the present work.