Mitigating shock of operating HDDs using smart suspensions

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2010

Djati W. D. (2010). Mitigating shock of operating HDDs using smart suspensions. Master's thesis, Nanyang Technological University, Singapore.

<https://hdl.handle.net/10356/20853>

<https://doi.org/10.32657/10356/20853>

**NANYANG
TECHNOLOGICAL
UNIVERSITY**

**MITIGATING SHOCK OF OPERATING HDDs USING
SMART SUSPENSIONS**

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SCHOOL OF MECHANICAL AND AEROSPACE
ENGINEERING
2010**

MITIGATING SHOCK OF OPERATING HDDs USING SMART SUSPENSIONS

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A thesis submitted to the Nanyang Technological University
In partial fulfillment of the requirement for the degree of
Master of Engineering

2010

Acknowledgements

First and foremost, I would like to express my most sincere gratitude to my supervisor, **A/P Nader Vahdati**, for all of his guidance, and encouragement throughout my research work at Nanyang Technological University. His sincerity and generosity would be borne in my mind forever.

I sincerely thank **AUN/SEED-Net**, Japan for the financial support during my research work.

I thank Dr. Hendri Harmoko and Mr. Christopher J. Lim from Seagate for their guidance and discussion related to my research work.

I would like to express my sincere gratitude to my parents, my family, and imu for their support, understanding and encouragement throughout the difficult times.

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List of Abbreviations

CG	: Center of Gravity
DOF	: Degree of Freedom
EOM	: Equation of Motion
FBD	: Free Body Diagram
FEA	: Finite Element Analysis
FRF	: Frequency Response Function
GB	: Giga Byte
HAA	: Head Actuator Assembly
HCL	: High Capacity Laminate
HDD	: Hard Disk Drive
HSA	: Head Stack Assembly
LB	: Left Bottom
LT	: Left Top
RB	: Right Bottom
RT	: Right Top
SSD	: Solid State Disk
SS Eq	: State Space Equation
TVA	: Tuned Vibration Absorber

Nomenclatures

A	: mounting cross section (m^2)
α	: frequency ratio between absorber and system's natural frequency
β	: elastomer coefficient depending on rubber hardness
c	: damping coefficient (Ns/m)
Δx	: mounting deformation (m)
e	: eccentricity (m)
E	: Young's Modulus (GPa)
E_{eq}	: Equivalent Young's Modulus (GPa)
f_n	: natural frequency (Hz)
F	: force (N)
G	: modulus of rigidity (GPa)
H	: rubber hardness
I	: second moment of inertia (m^4)
k	: stiffness coefficient (N/m)
l	: cantilever beam length (mm)
L_0	: difference between cantilever beam length and micro HDD length (mm)
m	: mass (kg or gram)
M	: mass – for bigger size mass (kg)
M	: moment (Nm)
ϕ	: phase angle (rad)
ρ	: cantilever beam density (kg/m^3)
R	: radius of the edge of the head (m)
S	: shape factor of elastomer
τ_{max}	: maximum shear stress (N/m^2)
t	: cantilever beam thickness (mm)
μ	: mass ratio between absorber and system's mass
U	: normalized eigenvectors
v	: velocity (m/s)
ω	: angular velocity (rad/s)
ω_f	: forcing frequency (rad/s)

ω_n	: natural frequency (rad/s)
w	: cantilever beam width (mm)
x	: displacement in x direction (m)/ coordinate
$\dot{x}$	: derivative of x
X	: eigenvector
y	: displacement in y direction (m)/ coordinate
ζ	: damping ratio
z	: coordinate

Summary

Small form factor (1" HDD) and 2.5" hard disk drives are widely used in mobile consumer electronics. Due to their mobile applications, mobile HDDs are susceptible to an environmental shock/ vibration. The shock/ vibration can be very severe that can damage the HDDs and the data stored in it. In this thesis, we propose several different shock/ vibration isolation techniques to reduce the risk of head slap which can happen if the HDD is subjected to a very severe shock. These techniques are compared and documented in this thesis.

The core of this thesis is divided into two parts. The first part discusses the techniques to mitigate the shock in 1" HDD, in which we use Seagate ST1 as our case study. The first technique is to use a cantilever beam as an external shock isolator. The technique covers the study of the cantilever beam natural frequency, the cantilever beam damping ratio, and the mechanical properties of the HDD's mountings; which can reduce the shock input significantly. A Simulink model is first created to help develop the shock isolator. The model is created using mode superposition and reduced mode method. The final design which is created in Simulink – MATLAB® is then compared with a finite element model in ANSYS. Simulation results show that the cantilever beam shock isolator is able to protect the micro HDD from operational shock up to 1000 G's 0.5 ms and has increased the ST1 shock tolerance by 77%. As a comparison, we have designed another type of external shock isolator for the ST1 using elastomeric materials. The elastomer layers are added beneath the base plate of the ST1 and outside of the ST1 case. The analysis of the elastomeric shock isolator system is done in ANSYS. The results show that the cantilever beam performs better than the elastomeric system to reduce the shock to the HDD.

The second technique is to use a Tuned Vibration Absorber (TVA) as an internal shock isolator. The TVA is installed on the HAA (Head Actuator Assembly). The study of TVA covers the theoretical analysis to find the optimal TVA properties. For ST1 case, the TVA is only a theoretical model. The TVA created in the model is not meant to be built, so it is not practical. This is because we only intend to find if the TVA can reduce the risk of 'head slap' significantly.

The second part discusses the techniques to mitigate the shock in 2.5“ HDD, in which we use the 2.5” Seagate HDD as our case study. The first technique is to use TVA. Different from ST1, the TVA model created for 2.5” HDD is dimensionally practical, meaning its size, mass, and material are chosen in such a way that it can be used in the actual 2.5” HDD. With the use of TVA, the shock tolerance of 2.5” Seagate HDD is increased by more than 21%. The second technique is to add an elastomer to the base of pivot arm and disk spindle. With this technique, the shock tolerance can be increased by 75%. The last technique that we tried is to increase the arm stiffness. This technique only reduces the slider displacement by 10%.

In the end, we summarize all of the techniques we have tried to mitigate the shock experienced by HDDs and gave recommendations to Seagate.

Chapter I

INTRODUCTION

1.1 Background

A Micro Hard Disk Drive is a hard disk drive in a small form factor. If the common hard disk drive size used by a desktop computer is 3.5" and the common hard disk drive size used by a laptop is 2.5", a small form factor HDD size can be as small as 1" to 0.85". Fig 1.1 shows a 0.85" micro hard disk drive with 4GB storage capacity. The 0.85" refers to diameter of the recording disk whereas the external size of the micro drive is 1.61" x 0.66" x 3.3".

Fig 1.1 Small form factor Hard Disk Drive, *source*: [1]

The advantage of using small form factor HDD is its size which makes it possible to be placed in small electronic equipment. A small form factor HDD has an advantage compared to the flash drive (SSD or Solid State Disk); which is a HDD can last more cycles of write and erase than a flash drive [2]. It means the flash drives capability to sustain the number of write and erase cycles is lower than a HDD. These advantages make a small form factor HDD a better option to run a program or an application. Small form factor HDDs also has the larger memory capacity than the SSD and lower price for the same capacity. There are many electronic equipment manufacturers such as mobile media player manufacturer who integrated their products with small form factor HDDs. Below are the lists of consumer electronics using small form factor HDD as its data storage:

- [Apple iPod mini](#) - 4GB and 6GB Hitachi
- [Creative MuVo](#) - 4GB Hitachi
- [Creative ZEN Micro](#) - 5GB and 6GB Seagate

- [Creative ZEN MicroPhoto](#) - 8GB Seagate
- [Dell Digital Jukebox](#) - 5GB Seagate
- [Iriver](#) H10 - 5GB and 6GB Seagate
- [Nokia](#) N91 - 4GB Hitachi
- palmOne [LifeDrive](#) - 4GB Hitachi
- Rio Carbon - 5GB Seagate
- Sony NW-A1000 - 6GB Hitachi
- [TrekStor vibez](#) - 8GB and 12GB Cornice Dragon

Despite many HDD advantages, there is one disadvantage and that is the tolerance of SFF HDD's to shock and vibration. A HDD in general and particularly a small form factor HDD is a mechanical device. It has a disk (platter), head, arm, and a motor to drive the disk and the arm. The disadvantage of a HDD as a mechanical device is that it is very sensitive to physical shock while operating. A physical shock is possible to be experienced by a HDD if the device (the HDD or the electronic equipments integrated with the HDD) is dropped.

Fig 1.2 shows why a typical hard disk drive is very sensitive to physical shock in operating condition. The distance between the reading head and the disk is very small (less than 25 nm). A scratch made by the head when it collides with the disk's surface can destroy the disk entirely and make the disk unreadable.

Fig 1.2 A view of head and disk assembly while operating, *source:* [3]

A collision between the head and the disk is possible because the Head Actuator arm Assembly (HAA) is not a rigid structure. When the HDD body is subjected to a shock, the shock is transmitted to the HAA and the disk. It makes the disk and the arm/ HAA to vibrate and possibly collide. This is depicted in Fig 1.3.

Fig 1.3 Possible movements between the head and disk which can cause scratch on the disk

Most of the small form factor HDDs or portable hard disk drives have a certain ability to handle physical shock, but the ability is limited up to 300 G's¹ 0.5 ms for operating condition [4] and 1500 G's 1 ms for non-operating condition [5]. To reduce the risk of a HDD of being damaged or suffered from physical shock, the HDD must be equipped with external or internal shock isolators. The purpose of vibration or shock isolator is to limit the shock transmitted from the vibration source to the small form factor HDD and keep the vibration level below the G limit (the HDD shock tolerance). By external isolator we mean, the vibration isolators are placed outside of the HDD and in the case of internal isolators, the isolators are placed inside the HDD.

Fig 1.4 Laptop's HDD equipped with rubber mountings

For mobile applications, the size of the vibration isolator must be reasonable, smaller is better, or simply it must not exceed the size of the electronic equipment where the HDD is integrated. A simple vibration isolator or shock isolator can be a rubber or an elastomer placed at the edges of the HDD. Fig 1.4 shows an example of rubber mountings from a hard disk drive used by a laptop manufacturer. The HDD is equipped with a rubber at its

¹ 1000 G's is 1000 times of gravity acceleration. The information about micro HDD ability to handle physical shock is available in [6].

edges. The manufacturer claims that the rubber mountings can enhance the HDD's durability and reduce the shock from any directions.

Rubber or elastomeric vibration and shock isolators are better option than the metal spring and damper suspension system since rubber vibration and shock isolators can be molded to any size, stiffness, and damping whereas this is not so easily possible with a metal spring and a viscous damper. To cope with the size of the electronic equipments in which the HDDs are installed, the vibration and shock isolator size must be as small as possible. As an example; it is impossible to create a viscous damper within range of 10 mm.

Another option as an external vibration and shock isolator is a cantilever beam system. The cantilever beam has certain stiffness and damping coefficient depending on its size and its material properties. The cantilever beam can perform as a vibration and shock isolator for a small form factor HDD that satisfies both performances and size requirements if it is designed properly.

1.2 Motivation

Disk drives (HDDs) used in portable applications such as mobile phones, MP3 players and laptops (mobile HDDs) can often be dropped accidentally. When dropped, the portable device can experience large shocks, in the order of 600 to 1000 G in few milliseconds. Therefore, the HDDs need to be isolated by a vibration and shock isolator that can reduce the shock input to the HDDs, and then increasing the shock robustness (shock tolerance) of the HDDs. In this research project, our aim is to develop shock isolators useful for mobile HDDs and develop a dynamic model of a HDD with vibration isolators included in the model.

1.3 Objectives

The objectives of this research are:

- To develop a suspension system for small form factor HDDs (Seagate ST1, 1" form factor) exposed to harsh environment. The suspension system protects the HDD while operating;

- To develop an internal shock isolation system for The Seagate 2.5” form factor HDDs to increase the shock tolerance of the HDDs. The shock isolator protects the HDD while operating.

1.4 Research Methodology

There are many methods to reduce the shock input that comes to the HDD. Externally, we can install a shock isolator between the HDD and its environment. The shock isolator can be either elastomeric system installed beneath the base of the HDD or a special suspension system constructed from a cantilever beam combined with the elastomeric mountings. Internally, we can apply a TVA (Tuned Vibration Absorber) on the HAA, enhance the structure of the pivot axis and the spindle axis by adding an elastomer between the axis and the base, or we can apply a HCL (High Capacity Laminate) bearing on the pivot arm. The methods either external or internal intend to reduce the shock input and reduce the relative movement between the head and the disk, thus reducing the risk of ‘head slap’. These ideas are pictured in Fig 1.5.

Fig 1.5 Pictorial of the ideas on how to put the shock isolators on HDD

In this project, we perform ‘what if studies’ to find the best place to install the shock isolator. The ideas pictured in Fig 1.5 are attempted one by one and their performances are compared. The project is divided into two parts, according to the objectives. The first part of the project is to develop a suspension system for ST1. In the first part, the ideas will be attempted are the cantilever beam, elastomeric mounting, and the TVA. External shock isolator most likely is chosen for ST1, since internal shock isolator for a small form

factor HDDs is very difficult to be manufactured. External shock isolator is easier to manufacture and does not need any changing in HDDs' manufacturing process.

The second part of the project is to develop an internal shock isolation system for 2.5" HDDs. Since this form factor of HDD will be most likely used in laptops, there are almost no spaces to install an external shock isolator inside the laptop. So the shock isolator is developed inside the HDD. The ideas will be attempted are the TVA, the HCL bearing, the elastomer under the pivot arm and disk spindle, and also changing the HAA properties.

In the cantilever beam design as the shock isolator idea, a dynamic model of the cantilever beam system is made in the Simulink environment. By using this dynamic model, users are able to choose several properties of the shock isolator such as the beam length and the beam thickness (beam's natural frequency), beam damping ratio, and the elastomeric mountings mechanical properties. Users are also able to determine the magnitude of the shock and the shock duration. The simulations results appear as the small form factor HDD responses at its corners and its CG (Center of Gravity).

The design of elastomeric system under the base of the small form factor HDD is aided by finite element program package, ANSYS. The design is pre-analyzed using a single DOF system with base excitation problem. At the end of this report, the performances of the external shock isolator which have been studied are compared.

The preliminary design of the internal shock isolator by means of installing a TVA on the HAA of ST1 aims to determine whether the TVA has a potential to reduce the risk of head slap of ST1. The risk of head slap is reduced by means of reducing the head acceleration and the relative motion between the head and the disk. The analysis is first performed in MATLAB®, and then the results are verified in ANSYS using transient analysis.

The development of shock isolation system for 2.5" HDD involves a development of a simple model in MATLAB®. The simple model is derived from finite element model of the HAA of the drives. We can easily add the TVA model to the simple model, adding an elastomeric model also, and changing the HAA component properties such as increasing

the arm stiffness. The simple model is also useful to verify the performance of the external shock isolator developed for ST1. With the simple model, all of the ideas of the internal shock isolation system can be attempted in MATLAB® first. The simulations in MATLAB® are very fast simulation yet quite accurate. At least the simple model can tell us what happens at the slider when the drive is subjected to a shock disturbance. The final properties of the shock isolator are then applied to the finite element model of the HAA in ANSYS to verify the results.

1.5 Outline

This report consists of nine chapters. The first chapter is the Introduction chapter. It discusses the background, motivation, objectives, and the research methodology.

The second chapter is the Literature Review chapter. It discusses the HDD terminology, previous studies about small form factor hard disk drive shock isolators, and the overview of the ‘what if studies’ plan.

The third chapter discusses the analysis and design of the cantilever beam as shock isolator for the ST1. It discusses the modeling of the problem, design requirements, modal analysis, mode superposition, the result of the analysis, and design optimization.

The fourth chapter discusses another work which also aims to reduce the risk of shock input by installing elastomer layers under the base plate of the ST1. The shock analysis is done in ANSYS.

The fifth chapter discusses the idea to put the elastomer layers outside of the case of ST1. The shock analysis is done in MATLAB® and ANSYS.

The sixth chapter discusses use of TVA on the HAA (Head Actuator Assembly) of ST1 to reduce the risk of head slap. This chapter also discusses the theoretical study of the TVA; to find the optimal properties of the TVA. The analysis is done in MATLAB® and ANSYS.

The seventh chapter discusses the use of TVA on the HAA of a 2.5” Seagate HDD. The analysis is done in MATLAB® and ANSYS.

The eighth chapter discusses the idea of installing HCL to replace the tolerance ring in between the ball bearing and the HAA of 2.5” Seagate HDD. The analysis is continued with placing the elastomer under the pivot arm and under the disk spindle base. The last section of this chapter discusses the idea to change the HAA properties, by increasing the arm stiffness. The analysis is done in MATLAB® and ANSYS.

The last chapter discusses the conclusion and recommendations from this project.

Chapter II

LITERATURE REVIEW

2.1 Hard Disk Drive Geometry and Terminology

Hard disk drive (HDD) is a non-volatile device which stores digitally encoded data on rapidly rotating platters (disks) with magnetic surfaces [7]. Non-volatile is a kind of computer memory that can retain the stored information without being powered. With a number of platters rotating rapidly when it is operated to write and erase the digital data and the head that moves at a certain angle to perform the erase and write operation, HDD is a mechanical device.

The first HDD came to the market was on 1956, it was called IBM 305 RAMAC. The drives storage capacity was only 4.8 MB (five million characters); with fifty disk (platters), each having 24" diameter. The drives only have single head assembly, with two heads, to access all of the platters. The price at that time was US\$ 160,000. The production of the drives ended at 1969. In 1961 IBM introduced the IBM 1301 that has head for each data surface. In 1973, IBM introduced IBM 3340 "Winchester" disk drive. That was the first drives used low mass and low load heads, and has a lubricated media. The drive capacity was 35 MB and 70 MB.

In 1980, IBM introduced the IBM 3380, the world's first gigabyte (GB) capacity disk drive. The size was a size of a refrigerator, with 250 kg weight. In 1991, the first 2.5" 100 MB hard drive released. In 1992, the first 1.3" HDD introduced by Hewlet Packard (HP C3013A). In 1999, IBM introduced the micro hard disk drive (1" form factor) with brand *Microdrive* with 170 MB and 340 MB capacity. Since that year, IBM produced the same form factor and even smaller drives with the larger capacity.

Fig 2.1 shows the main parts of the modern HDD. The platters are driven by the spindle. Typically it rotates about 3600 rpm to 7200 rpm. To increase the amount of data that can be stored, many of HDD have more than one platter. The head role is to write, erase, or read the data. The head is placed on the actuator arm. The actuator arm moves in an arc with the actuator axis as the reference. Its movement is controlled by the actuator. The

actuator itself is usually a linear or a step motor. With this movement, the head is able to reach the entire platter surface.

Fig 2.1 HDD main parts, *source: [7]*

The gap between the head and the platter in the operating condition is less than 25 nm. The platters are rotating, made the air moves around the head and the platter. It becomes an air suspension/ air bearing between the head and the platter. The only 25 nm gap between the head and the platter become a reason HDD is susceptible to a physical shock. Under non-operating condition, the actuator arm rest on a parking space/ ramp. It prevents the head and the platters from colliding when a physical shock comes to the HDD. Therefore, the HDD is more invulnerable to a physical shock in the non-operating condition.

2.2 Review of Existing Studies Concerning HDD Failures

There are number of studies concerning HDD's head-disk failures. Although the HDD for desktop computers are all stationary when it is operated, there are many possibilities for the HDD to be failed mechanically. One example is when the HDD are on transporting processes. It is possible for the HDD to fall or handled carelessly such the HDD will experience external physical shock. If once the head scratch the disk surface, the HDD would be failing at its initial operation or maybe won't last longer in its operation. Therefore, since more than decades ago, people already studied about the head-disk failure mechanism. Such of the study about HDD is a useful record for a research to protect the HDD mechanically. This section will discuss the existing studies concerning

the mechanical failure at the head-disk of the large HDD, portable HDD, and some of the micro HDD (small form factor HDD), and also will mention some of the HDDs' development in industry through the decades.

In 1995, Kouhei et al. [8] studied the head-disk interface shock resistance. The HDD used was the 2.5" and 1.8" HDD. The relationship between the disk failure and slider dynamics was defined experimentally. Mainly, the shear stress appeared in the disk surface is estimated as:

$$\tau_{\max} = cE^{4/5}R^{-3/5}v^{2/5}m^{1/5} \quad (2.1)$$

where:

- m : mass of slider
- v : relative velocity between slider and disk
- τ : shear stress
- R : radius of slider's edge
- E : combined Young's modulus
- c : constant

Directly from the equation, the shear stress acted in the disk surface is greatly affected by the relative velocity between the disk and the slider. The rest of the variables are assumed constant, except the tilting angle of the slider that will affect the radius of the slider's edge. This is because of the slider's edge has unique shape. To reduce the relative velocity between the slider and the disk, they have proposed a structure called "jump stop". This structure has been experimentally proven to be able to withstand the acceleration up to 1800 G's while the system that has no "jump stop" can only hold below 400 G's.

In 1997, Dennis [9] performed a study about an external vibration isolator for a HDD application on boat in rough seas. Since there is so much space to put the HDD, the protection system for HDD on boat appears as large external vibration isolation rather than a protection system inside the HDD. The mounting system suggested by the author is a low frequency mounting system. The author suggested metal spring system or pneumatic bellows to support the small system of the HDD to produce low mounting system natural frequency.

In 1999, Kok-Kia Chew [10] studied a 4-D shock sensing for HDD. The idea was to shut off the writing process and park the actuator arm when a sensor senses a sufficient shock magnitude. The threshold of the shock magnitude is of course had been studied before, that can make the HDD in trouble. The 4-D shock sensing proposed by the author is basically an improvement from a similar idea. It has an improvement to distinguish which is the shock come from external load and the shock created by the drive seeking process (drive seeking processes shock can reach 50 G's). The 4-D term came from the three dimensional space and the servo chronological information.

At the same year, Edwards [11] performed a FEA (finite element analysis) of the shock response and head slap behavior of HDD. The HDD used was a 3.5" HDD. The result from the study is that the acceleration amplification factors are much higher than the values expected from a single DOF system response. This high acceleration amplification value is due to the combined effects of the multi modes of vibration from the HDD itself. Particularly, the HSA (head stack assembly) experienced the highest amplification value when the HDD is subjected to shock duration from 0.5 to 1 ms, the reason is that the HSA has natural frequency within the shock duration range. It was concluded that the HSA sensitivity to short shock duration is the key factor of head slap.

There were several studies conducted on 2000. Sung Jin Lee et al. [12] performed a study about the shock-resistance design of spindle/disk and HSA suspension that subjected to impulse excitation. The study was performed theoretically, numerically, and experimentally. The authors' mentioned that if the suspensions are designed properly, the relative velocity between the head and disk can be reduced, and it can reduce the possibility of head slap occurrence.

Haeng-Soo Lee et al. [13] performed a study to put a dynamic absorber on actuator arm inside a particular hard disk drive. The purpose is to control the displacement and acceleration of the actuator arm, to improve shock handling capability of the HDD both in operating and non-operating condition. The result of the experiments shows that the acceleration of the arm tip can be reduced by installing the dynamic absorber on the actuator arm.

Jin-Seung Sohn [14] performed an experimental analysis on HDD actuator to improve its shock resistance. The study was conducted on non-operating condition of the HDD. The result from this study was about the transmissibility and the beating phenomenon between the arm blade and the suspension. The low transmissibility is important to reduce the

acceleration transmitted to the actuator arm; this is done by selecting the proper actuator arm natural frequency. Though the transmissibility is low enough, the suspension frequency must not close with the arm blade natural frequency. If that is happen, it will induce the beating phenomenon that will only amplify the acceleration response of the arm blade. The first bending natural frequency of the arm blade must be adjusted to ensure the low transmissibility and avoid the beating phenomenon.

Harrison et al. [15] studied the flying height response of an actuator arm of 3.5" HDD to mechanical shock on operating condition. The shock input used was 60 G's with 2 ms duration. The authors' proposed a method to reduce the head slap risk by increasing disk stiffness, increasing disk clamping radius or decreasing the disk outer radius; increasing the effective air bearing stiffness; and decreasing the portion of the suspension that coupled with the motion of the slider.

2003 is a year that mentioned as the turning point of HDD. Vaughan-Nichols [16] in Computer magazine said that HDD will come in larger capacity in the next years and will be used much for home entertainment and small consumer electronics. This article mentioned about the new era of the HDD that will have much broader application. Mobile HDD in the next year from 2003 will be much more produced for small consumer electronics market. By this market demand, the HDD in the next years must be ready to be mobilized and handle various external possible loading (e.g. physical shock).

On the same year, Kemao Peng [17] performed a study to model and analyzes micro hard disk drives. Though this study was not concerned about mechanical shock resistance of the micro HDD, this research was a sign that micro HDD already considered as a trends in storage industry.

Jump to 2005, a data storage manufacturer released a 4GB micro HDD that equipped with protection from mechanical shock, mentioned by Stephan Ohr [18] in Building Blocks magazine. The protection system was called Crash Guard II. The system works to recognize whether the drive is falling or not. When it is falling, the system will react by latching the head away from the disk, thus prevent the head to scratch the disk. This micro HDD are already embedded in small consumer electronics such as mobile MP3 players. It seems that this protection system was derived from the similar research conducted by Kok-Kia Chew on 1999 about 4-D shock sensing for HDD. Such a system is needed by the mobile HDD to increase its reliability and lifetime usage. At that time,

the drives were sold for US\$199; while for the same capacity; other manufacturers sold it for US\$ 179 and US\$ 149.

At the same year, Sam Bhavnani [19] in Design Currents magazine mentioned about ruggedized laptop. The laptop was designed for professionals and workers who need to bring their laptop into the field. The laptop has a shock mounted and isolated HDD. These two articles at 2005 represent a high market development for a mobile HDD; that mentioned about the protection system needed by the drives to survive in the environment.

In 2006, there are many research conducted on mechanical consideration of HDD. F.F. Yap et al. [20] performed a study to design an external vibration isolation system for HDD. The suspension system is designed to meet the requirements for the mobile military applications. The design was verified using military standard MIL-STD-810E. The design basic was to achieve a low transmissibility through the three motion axes in the 5 to 500 Hz frequency range. The result was to put the natural frequency of the isolation system between 10 and 20 Hz. The damping ratio suggested in the design was more than 10%. Base on the experiments, the design can meet the vibration resistance requirements.

D.W. Shu et al. [21] conducted a drop test simulation of a head actuator assembly in HDD. The purpose of the study was to develop a method to predict the displacement and the failure of the head actuator assembly during a drop test. There was a corollary derived in the paper: when the characteristics frequencies of a group of acceleration shocks with different pulse shape are very close to the resonant frequency of the dynamic system, these acceleration shocks will have nearly equal acceleration powers at the resonant frequency; consequently, they will produce nearly equal shock responses. In other words, the transmissibility cannot be reduced.

B. Gu et al. [22] performed a finite element analysis (FEA) of linear and rotary drop test for small form factor HDD (1" form factor). The FEA result shows that the small form factor HDD is more vulnerable to a linear shock with the same properties (amplitude and shock duration) than to a rotary shock. Their analysis results are summarized in table 2.1. Table 2.1 shows the maximum lift-up height of the upper slider when the HDD is subjected to a shock input. It can be roughly estimated that for 1 ms shock duration; to make the same lift-up height of the upper slider, the rotary shock magnitude needed is two times of the linear shock magnitude. For 0.5 ms duration, we can only conclude that

it can be roughly estimated that for the same shock magnitude, the linear shock effect on lift-up height of the upper slider is about ten times of the rotary shock.

Table 2.1 Upper slider lift-up height of small form factor HDDs subjected to linear and rotary shock

Shock input	Linear shock upper slider lift-up height (mm)	Rotary shock upper slider lift-up height (mm)
300 G's 0.5 ms	0.23	0.025
300 G's 1 ms	0.37	0.05
600 G's 1 ms	1.28	0.45

Feng Gao et al. [23] performed a shock analysis of non-operating HDD using multi-body dynamics. The purpose of the study was to formulate the flexible multi-body dynamics of HDD. The flexible multi-body dynamics has advantage in time consumed to the FEA program package. It has been proven that the flexible multi-body dynamics is a reasonable method.

D.W. Shu et al. [24] performed a study of the pulse width effect on the shock response of HDD. It was found that there is a limit of input acceleration magnitude before the head slap behavior take places. It means if the transmissibility can be maintained to be low enough, the head slap occurrence possibility can be reduced. From the pulse width relationship to the resonant modes of the HSA, it was found that the largest acceleration response appears from the 0.3 ms pulse width shock input. It means, the HSA system is very sensitive to short shock duration.

In 2007, Bert Feliss et al. [25] performed a micro HDD operational and non-operational shock vibration testing. The results revealed that on operating condition, shock amplitude of 200 G's is enough to break the air bearing between the slider and the disk. This allows the slider to move vertically both in negative and positive direction, thus allow the head slap to be occurred. On non-operating condition, the micro HDD can handle up to 2000 G's shock amplitude very well.

In the same year, Bhargava and Boggy [4] performed a numerical simulation of operational shock in small form factor HDD (micro hard disk drives). The study revealed that the air bearing has different responses for upward and downward shocks. The shock input used in the simulation was 0.5 ms pulse width shock with -800 to 600 G's amplitude. The results show that the micro HDD is very robust to the downward shock. The air bearing was able to sustain relatively large loads up to 7 g (for -800 G's shock amplitude). On the

other hand, the air bearing was found to be failed when the micro hard disk drive was subjected to an upward shock more than 300 G's.

The last report on 2007 came from Rick Merrit [5] on Building Blocks. The author reported the heat of the tiny drives market. At that year, Hitachi came with 1" micro HDD that able to withstand 400 G's 2 ms operational shock; it was already improved from 200 G's 2 ms operational shock. Another product came from Hitachi is 1.8" micro HDD that able to withstand 600 G's 2 ms operational shock; it was already improved from 500 G's 2 ms. Without any external or internal reinforcement, the tiny drives can handle more than 400 G's 2 ms shock excitation.

The study concerning HDD mechanical failures has been started more than ten years ago. The study was become more intensive when the HDD are being used in mobile application. Most of the study discussed the failure mechanism, and only a few discussed on how to counter the failures. There is only a few of study about how to protect the mobile HDD from shock excitation. Particularly, there is still no study in which people created a general dynamic model of a HDD and its shock isolation system. This general dynamic model will allow people to conduct shock analysis of a HDD with its shock isolation system, thus designing the optimal shock isolation system for a particular HDD.

2.3 Possible Shock Isolation System For HDD

In this section, we will explore the ideas of installing shock isolator either outside or inside the HDD. We will first examine the possibility to install the shock isolator outside the HDD.

2.3.1 External Shock Isolator for HDDs

As earlier explained in Chapter I, section 1.4, we will use a small form factor HDD for our case study for external shock isolator. Particularly, the small form factor which will be used for our case study is Seagate ST1. The form factor of ST1 is 1" and has 8 GB of memory capacity. The external size of ST1 is explained in Fig 2.2.

To meet the size requirements of the shock isolator, the size of the micro HDD product (0.85" form factor) introduced the Chapter I, Fig 1.1, is used as the base size for scaling. The external case size is 41 mm x 16.7 mm x 84 mm. But actually the length inside for the HDD is about 55 mm, not 84 mm (84 mm is total length

if the hook is included). Thus the external size of the micro HDD product is scaled up by $\frac{1}{0.85}$, according to the form factor of the HDD. The scaled size of the external case is given in Fig 2.3.

Fig 2.2 Seagate ST1

Fig 2.3 Scaled up external size of the micro HDD's case

According to Fig 2.3, the space left for the external shock isolator is about 14.7 mm in vertical direction and 65 mm in horizontal direction.

One of the idea mentioned before in Chapter I is to use an elastomeric system as shock isolator. Typically, the stiffness of an elastomer is higher than typical steel coil spring [26]. If the stiffness is too high, the transmitted force will also be too high, resulting in higher acceleration response of the HDD. But elastomeric system has an advantage from typical coil spring in which it can be molded into

any size and shape and has various value of damping ratio. The idea to put an elastomer outside of the HDD is explained in Fig 2.4.

Fig 2.4 Elastomer placements as external shock isolator for ST1

The amount of shock reduced by using elastomeric system depends on how much deformation experienced by the elastomer. That is why the elastomer height in Fig 2.4 is thicker than the ST1, since it is possible that we will need very thick elastomer to meet the shock isolator performance. This idea will be examined further in Chapter IV and V about analysis of elastomeric shock isolation system for ST1.

Another idea is to use a highly damped cantilever beam as the shock isolator. This idea is explained in Fig 2.5.

Fig 2.5 Cantilever beam system as external shock isolator for ST1

One end of the cantilever beam is clamped to the wall of the external case. ST1 is mounted to the cantilever beam at four points (each corner). The mountings can be an elastomer or rubber-like materials. The cantilever beam size, its material, and the mounting mechanical properties will affect the shock isolator performance.

From Fig 2.5 we can see clearly the size limitations of the cantilever beam system. The beam length must not exceed 65 mm and the sway movement of the end of the beam must not exceed the vertical space of the external case (19.7 mm). The beam material will affect the natural frequency and the damping ratio of the system. Since the damping properties provided by the cantilever beam is structural damping, selection of material is very crucial. Table 2.2 shows the damping characteristics of several materials including metals, alloys, composite, and polymer.

Table 2.2 Values of damping and elastic characteristics of metal and alloys [27, 28, 34]

Material	Damping Ratio (%)	Ex10 ⁻⁵ , MPa
Magnesium (99.9%, cast)	1.9	0.45
Magnesium alloy (Mg, 0.5% Zr)	2.5	0.45
Sonoston (Mn, 36% Cu, 4.5% Al, 3% Fe, 2% Ni)	2	1.2
Silentaloy (Fe, 12% Cr, 3% Al)	1.8	2.0
Nitinol (50% Ni, 50% Ti, optimally heat treated and prestressed)	5%	0.4
Nickel (Ni, pure)	0.9%	2.0
Steel, low carbon	0.2%	2.0
Aluminum, 1100	0.02%	0.7
Iron (Fe, pure)	0.8%	2.0
Stainless steel, Austenitic	0.05%	2.0
Ti-6Al-4V	0.01%	1.1
AS Carbon/ Epoxy composites	40-50%	0.7
T300 Carbon/ Epoxy	40-50%	0.6-0.7
Ni-Mn-Ga Composite	30-50%	0.7-1
Greene Tweed Orthetek® WF Polyketone, Continuous Woven Carbon Fiber (Polyketone family)	30-50%	0.6

This idea will be investigated thoroughly in Chapter III. The next section will explain briefly the possibility to install an internal shock isolator for HDDs.

2.3.2 Internal Shock Isolator for HDDs

The idea of installing internal shock isolator will be applied to ST1 (TVA only) and the 2.5" Seagate HDD. The first idea is to apply a TVA (Tuned Vibration Absorber) on the HAA (Head Actuator Arm). The aim is to reduce the acceleration and the displacement response of the head/ slider directly by

canceling out the force or movement of the reading arm. The same idea has been conducted by Haeng Soo Lee et. al [13] by using dynamic absorber. Not like their idea, we will tune the TVA frequency close the first bending natural frequency of the HAA part where the TVA is installed. We will also examine the optimal location to put the TVA and set the mass of the TVA as small as possible so it will not affect the reading speed significantly.

The second idea is to apply a HCL (High Capacity Laminated) bearing for the pivot arm bearing. This idea is explained in Fig 2.6. The bearing has an ability to filter out the vibration caused by physical shock experienced by the outside case of the HDD.

Fig 2.6 HCL placements as pivot arm's bearing

The third idea is to enhance the pivot arm and the spindle structure so it can help to reduce a shock input from the external case of HDDs. The pivot axis and the spindle structure may be partly cut to put the shock isolator, as earlier explained in Chapter I, Fig 1.5. However, this idea may affect the flying height of the slider if we do not carefully design and manufacture the shock isolator height.

The fourth idea is to change the HAA component's mechanical properties, in example is to increase the arm stiffness by increasing the arm thickness. The idea of HCL, enhancing the pivot arm and the disk spindle structure, and changing the HAA component's mechanical properties will be done only for 2.5" Seagate HDD.

2.4 Concluding Remarks

The internal shock isolator such as TVA has been conducted by [13], but in our project we will apply the TVA in the different locations and see if we can achieve reduction of more than 10%. The study about external shock isolator for small form factor HDDs is never conducted before, particularly a study of an external shock isolator using a cantilever beam system. The idea of enhancing the pivot arm and disk spindle structure also has not been done by anyone. Thus, we conclude that our project has not been done by anyone before.

Chapter III

USE OF HIGHLY DAMPED CANTILEVER BEAM AS VIBRATION AND SHOCK ISOLATOR IN MOBILE APPLICATIONS

3.1 Case Study Overview and Design Requirements

As already mentioned in Chapter II section 2.3, a cantilever beam system can be fitted into relatively small space (see Fig 2.5). Thus, cantilever beam system can be a good option as an external shock isolator for small form factor HDDs particularly when there is little space available for shock isolator. The small form factor HDD that will be under investigation is the Seagate ST1. The maximum cantilever beam system size as external shock isolator is 65 mm for its length and 19.7 mm for maximum vertical space (overall thickness plus sway movement).

ST1 is the predecessor of ST1.3. ST1 can sustain an operational shock of 175 G's with 2 ms duration and non-operational shock of 1500 G's with 1 ms duration. The successor (ST1.3) came with more robust design that can sustain operational shock of 300 G's with 1 ms duration. According to [4], some of the latest small form factor HDDs can sustain operational shock of 300 G's with 0.5 ms duration in the upward direction. We know that when a HDD is dropped to a ground, the shock inputs to the HDD can be in the range of 600 to 1000 G's. Referring to many studies that have been performed, the shock input used in this project will be 1000 G's with 0.5 ms duration.

The model of cantilever beam – ST1 system is given in Fig 3.1. The beam width is the same as the ST1 width ($w=35$ mm). The beam length is defined as the addition of ST1 length plus a length called L_o (beam length = 43 mm + L_o). The ST1 is mounted to the beam at its four corners. The mounting thickness will be defined later; which indirectly determines the mounting stiffness. The beam thickness is denoted by t . The left end of the beam is constrained so it can move only in the z direction. Later in the next sections, the beam dimensions which will be studied are L_o and t , while the beam properties to be studied are the Young's modulus and damping ratio (ζ).

Fig 3.1 Cantilever beam – ST1 model

Below are the important assumptions of the MATLAB® model of the beam – HDD shock isolator:

- The HDD is a rigid body
- The beam surface at origin ($x, y, z = 0$) is constrained so it can move only in the z direction, thus the first mode of the beam is a rigid body motion;
- The CG location of the HDD is assumed at the center of its volume;
- There are only five nodes of the beam which will be used to construct the MATLAB® model. The nodes are: 1 at the wall (left end), two nodes coincide with the left mounting, and two nodes coincide with the right mounting. These five nodes are the nodes that connect the beam with its environment, which are wall and the HDD. With only five nodes used, the system matrix is highly reduced to save up simulation time. This is the same technique used in ANSYS (Reduced Mode analysis);
- The system's degree of freedom is only 5. This is because the eigenvectors, taken from the beam are only in the z direction. The x , y , z rotation, and y rotation vector do not much affect the vibration response. The x rotation vector contribution is considered small compared to z displacement vector in only to obtain the maximum acceleration and displacement response of the HDD. This is proven by comparing the result from MATLAB® analysis with Full Transient Analysis from ANSYS;
- According to the 3rd and the 4th assumptions, the mounting modeled in MATLAB® is a line comprising of a linear spring and a damper. The HDD

response from MATLAB® analysis will have greater error if the real mounting base area is too large;

- The damping properties assigned for beam is a modal damping that has uniform value of damping ratio at all modes ($\zeta_1=\zeta_2=\zeta_3=\dots=\zeta_n$). The reason is that the damping value of many materials such as metals, alloys, and composites has nearly the same value in kHz frequency range. Another reason is the dominant modes in the vibration response are the 2nd to the 3rd of the elastic modes. This is due to the frequency induced by the shock input is in range of 0 to 1000 Hz. Thus, the other modes participation in vibration response is very small. With this condition, the damping ratio at all modes can be set into uniform value;
- The simulation performed in MATLAB® may produce some errors. But the main purpose of MATLAB® simulation is to save up simulation time and keep the design process in the right direction. It is proven that the MATLAB® simulation can give us direction which shock isolator property that need to be modified (increase or decrease the value of the beam natural frequency, the beam damping ratio, and the mechanical properties of the mountings).

Before we discuss the simulation to perform a shock analysis, the next section will explain the entire procedure to perform the shock analysis and a short explanation about how to create the MATLAB® model.

3.2 Analysis Procedure

The procedure and the model explanation will be explained below in sub-part sequence:

3.2.1 Determining the Cantilever Beam 1st Natural Frequency

The main foundation of the shock isolator is the cantilever beam. Its 1st natural frequency (the 1st bending natural frequency) will affect the overall shock isolator performance. The 1st bending natural frequency of the beam must be such so the acceleration response and the displacement response of the HDD are both low. We call this natural frequency as ‘compromise’ natural frequency.

To obtain the ‘compromise’ natural frequency of the cantilever beam, we first simulate the shock response of single DOF system which resembles the 1st mode

of the beam (base excitation problem). The simulation is done in MATLAB®. The m-file used for this simulation is given in appendix A.

With this ‘compromise’ natural frequency, we can then determine the size of the beam and the beam material. This step will be discussed thoroughly in section 3.3.

3.2.2 Determining the Cantilever Beam Size and Material

The closed form solution for the 1st bending natural frequency of a cantilever beam is given by:

$$\omega_1 = 3.5160 \sqrt{\frac{EI}{mL^4}} \quad (3.1)$$

Inserting the second moment of inertia (I) and mass per length (m), equation (3.1) can be written as:

$$\omega_1 = 3.5160 \sqrt{\frac{Et^2}{12\rho L^4}} \quad (3.2)$$

Therefore the 1st bending natural frequency of the cantilever beam is strongly determined by its length (L) and thickness (t), followed by Young’s modulus and density. The beam width is determined the same as the HDD’s width. The Young’s modulus and density will follow the selection of highly damped beam materials. The beam thickness is preferred to be as thin as possible, so it is determined by judging the overall shock isolator thickness. After all of the properties are settled, the beam length can be calculated.

3.2.3 Extracting the Cantilever Beam Eigenvalues and Eigenvectors

The eigenvalues and eigenvectors of the cantilever beam are obtained using Modal Analysis in finite element program, ANSYS. To easily extract the eigenvalues and the eigenvectors of the needed nodes, we can use the GUI (Graphical User Interface) in the ‘Time History Postpro’ in the ANSYS main menu. According to the assumptions mentioned in section 3.1, we only need to extract the node’s displacement in the z direction. The GUI image is given in Fig 3.2. The eigenvalues and eigenvectors can be saved in a ‘csv’ file and can be processed further to create a state space equation. The example of ANSYS

command to create the cantilever beam model and extracting the eigenvalues and eigenvectors is given in Appendix B.

Fig 3.2 GUI to extract eigenvalues and eigenvectors

3.2.4 Constructing the State Space (SS) Equation of the Beam

To demonstrate on how to construct the state space equation of the cantilever beam, we will use a three DOF system in Fig 3.3. The spring is denoted by full line and the damper is denoted by dotted line. The wheel and the ground are assumed frictionless. The displacement of each mass is denoted by x_1 , x_2 , and x_3 respectively.

Fig 3.3 Three – DOF problems

The eigenvalues and eigenvectors of the system in Fig 3.3 are:

$$\omega^2 = \begin{bmatrix} 0 & & \\ & k/m & \\ & & 3k/m \end{bmatrix} \quad X = \begin{bmatrix} 1 & -1 & 1 \\ 1 & 0 & -2 \\ 1 & 1 & 1 \end{bmatrix} \quad (3.3)$$

The eigenvectors is then normalized to the mass by following equation below:

$$X_{ni}^T m X_{ni} = 1$$

$$X_{ni} = \frac{X_i}{\left(X_i^T m X_i\right)^{\frac{1}{2}}} \quad (3.4)$$

Where X_{ni} is the i^{th} normalized eigenvector.

The normalized eigenvectors is then:

$$X_{ni} = \begin{bmatrix} 1/\sqrt{3m} & -1/\sqrt{2m} & 1/\sqrt{6m} \\ 1/\sqrt{3m} & 0 & -2/\sqrt{6m} \\ 1/\sqrt{3m} & 1/\sqrt{2m} & 1/\sqrt{6m} \end{bmatrix} \quad (3.5)$$

The eigenvectors obtained from Modal Analysis in ANSYS are already normalized with respect to the mass, so we do not need to normalize it anymore.

Next, we write the governing equation of motion in matrix form:

$$[M] \ddot{X} + [C] \dot{X} + [K] X = 0$$

$$\begin{bmatrix} m_1 & & \\ & m_2 & \\ & & m_3 \end{bmatrix} \begin{Bmatrix} \ddot{x}_1 \\ \ddot{x}_2 \\ \ddot{x}_3 \end{Bmatrix} + \begin{bmatrix} c_1 & -c_1 & 0 \\ -c_1 & 2c_2 & -c_2 \\ 0 & -c_2 & c_2 \end{bmatrix} \begin{Bmatrix} \dot{x}_1 \\ \dot{x}_2 \\ \dot{x}_3 \end{Bmatrix} + \begin{bmatrix} k_1 & -k_1 & 0 \\ -k_1 & 2k_2 & -k_2 \\ 0 & -k_2 & k_2 \end{bmatrix} \begin{Bmatrix} x_1 \\ x_2 \\ x_3 \end{Bmatrix} = 0 \quad (3.6)$$

The mass, damping, and the stiffness matrix are then multiplied by the normalized eigenvectors X_n . By assuming:

$$m_1=m_2=m_3=m; c_1=c_2=c; k_1=k_2=k$$

Equation (3.6) is then written as:

$$X_n^T [M] X_n \ddot{X}(t) + X_n^T [C] X_n \dot{X}(t) + X_n^T [K] X_n X(t) = 0$$

$$\begin{bmatrix} 1 & & \\ & 1 & \\ & & 1 \end{bmatrix} \ddot{X} + \begin{bmatrix} 0 & & \\ c/m & & \\ & 3c/m & \end{bmatrix} \dot{X} + \begin{bmatrix} 0 & & \\ k/m & & \\ & 3k/m & \end{bmatrix} X = 0 \quad (3.7)$$

The C and K matrix are then can be written in the form of:

$$[C] = \begin{bmatrix} 2\zeta_1\omega_1 & & \\ & 2\zeta_2\omega_2 & \\ & & 2\zeta_3\omega_3 \end{bmatrix} \quad (3.8)$$

$$[K] = \begin{bmatrix} \omega_1^2 & & \\ & \omega_2^2 & \\ & & \omega_3^2 \end{bmatrix} \quad (3.8)$$

Where ζ is the damping ratio for the corresponding modes. With this new M, C, and K matrices, the governing equation of motion is uncoupled and easier to solve.

The state space equation is defined as:

$$\begin{aligned} \dot{x} &= Ax + Bu \\ y &= Cx + Du \end{aligned} \quad (3.9)$$

We will first derive A and B matrices, where Bu is the external forces that are assumed to be zero for now. The first row of the equation (3.7):

$$\ddot{x}_1 + 2\zeta_1\omega_1\dot{x}_1 + \omega_1^2x_1 = 0$$

Or in matrix form:

$$\begin{bmatrix} \dot{x}_1 \\ \ddot{x}_1 \end{bmatrix} = \begin{bmatrix} 0 & 1 \\ -\omega_1^2 & -2\zeta_1\omega_1 \end{bmatrix} \begin{bmatrix} x_1 \\ \dot{x}_1 \end{bmatrix} \quad (3.10)$$

Then a part of A matrix is:

$$\begin{bmatrix} 0 & 1 \\ -\omega_1^2 & -2\zeta_1\omega_1 \end{bmatrix} \quad (3.11)$$

Following the same steps for the rest of the row, we can obtain complete A matrix:

$$A = \begin{bmatrix} 0 & 1 & 0 & 0 & 0 & 0 \\ -\omega_1^2 & -2\zeta_1\omega_1 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 1 & 0 & 0 \\ 0 & 0 & -\omega_2^2 & -2\zeta_2\omega_2 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 1 \\ 0 & 0 & 0 & 0 & -\omega_3^2 & -2\zeta_3\omega_3 \end{bmatrix} \quad (3.12)$$

Previously, B matrix is:

$$B = \begin{bmatrix} 0 \\ F_1 \\ 0 \\ F_2 \\ 0 \\ F_3 \end{bmatrix} u \quad (3.13)$$

Where F_1 , F_2 , and F_3 are the constants of external forces applied to each mass m_1 , m_2 , and m_3 respectively. The value u is the multiplier for forces that has certain function, such as sine function. In our case, u is the shock (half sine) function. Because A matrix is obtained by multiplying the equation of motion by the normalized eigenvectors, the B matrix must also be multiplied by the normalized eigenvectors. The matrix multiplication is given below, assuming only m_1 is subjected to external force F_1 :

$$[M]\ddot{X} + [C]\dot{X} + [K]X = [F]$$

$$[F] = \begin{bmatrix} F_1 \\ 0 \\ 0 \end{bmatrix} \quad (3.14)$$

$$[F]_n = \begin{bmatrix} 1/\sqrt{3m} & 1/\sqrt{3m} & 1/\sqrt{3m} \\ 1/\sqrt{2m} & 0 & -1/\sqrt{2m} \\ 1/\sqrt{6m} & -1/\sqrt{3m} & 1/\sqrt{6m} \end{bmatrix} \begin{bmatrix} F_1 \\ 0 \\ 0 \end{bmatrix}$$

$$= \begin{bmatrix} F_1/\sqrt{3m} \\ 0 \\ 0 \end{bmatrix} \quad (3.15)$$

Finally, B matrix:

$$B = \begin{bmatrix} 0 \\ F_1/\sqrt{3m} \\ 0 \\ 0 \\ 0 \\ 0 \end{bmatrix} u \quad (3.16)$$

The matrices left are the C and D matrices. The D matrix will be left as zero matrixes. The C matrix is the state to obtain the solution or called as the output matrix. Previously, the C matrix would be in the form:

$$\begin{bmatrix} y_1 \\ \dot{y}_1 \\ y_2 \\ \dot{y}_2 \\ y_3 \\ \dot{y}_3 \end{bmatrix} = \begin{bmatrix} 1 & & & & & \\ & 1 & & & & \\ & & 1 & & & \\ & & & 1 & & \\ & & & & 1 & \\ & & & & & 1 \end{bmatrix} \begin{bmatrix} x_1 \\ \dot{x}_1 \\ x_2 \\ \dot{x}_2 \\ x_3 \\ \dot{x}_3 \end{bmatrix} \quad (3.17)$$

By the normalized eigenvectors, C matrix will be in the form:

$$\begin{bmatrix} y_1 \\ \dot{y}_1 \\ y_2 \\ \dot{y}_2 \\ y_3 \\ \dot{y}_3 \end{bmatrix} = \begin{bmatrix} 1/\sqrt{3m} & 0 & 1/\sqrt{3m} & 0 & 1/\sqrt{3m} & 0 \\ 0 & 1/\sqrt{3m} & 0 & 1/\sqrt{3m} & 0 & 1/\sqrt{3m} \\ 1/\sqrt{2m} & 0 & 0 & 0 & -1/\sqrt{2m} & 0 \\ 0 & 1/\sqrt{2m} & 0 & 0 & 0 & -1/\sqrt{2m} \\ 1/\sqrt{6m} & 0 & -1/\sqrt{3m} & 0 & 1/\sqrt{6m} & 0 \\ 0 & 1/\sqrt{6m} & 0 & -1/\sqrt{3m} & 0 & 1/\sqrt{6m} \end{bmatrix} \begin{bmatrix} x_1 \\ \dot{x}_1 \\ x_2 \\ \dot{x}_2 \\ x_3 \\ \dot{x}_3 \end{bmatrix} \quad (3.18)$$

Then, the output of displacement and velocity of each mass can be obtained. The same step is done with the eigenvalues and eigenvectors of the cantilever beam obtained from modal analysis in ANSYS. A routine in m-file is created to construct the state space equation. The routine can be found in appendix A. The beam's damping ratio is defined when constructing the state space equation (in matrix A). These procedures can be found in [30] in Chapter 5.

3.2.5 Constructing the Simulink Model using SimMechanics Toolbox

The idea of mode superposition in constructing the Simulink model of HDD – cantilever beam shock isolator is explained in Fig 3.4.

Fig 3.4 Main idea of Simulink model

Now we already have a state space equation of cantilever beam (explained in sub-part 3.2.3). The things left are to add the HDD model, mountings model, body sensor to measure HDD responses, apply the external force (shock input), and connect each model.

The HDD is a rigid body, and then the block used is Body block. Within this block, we can set the dimension of the HDD, set the CG (Center of Gravity) location, set HDD mass, set moment of inertia, and set the number of port (this port can be connected into any block; in this case the port is connected to mountings and Body sensor block to obtain its response). The HDD model and the dialog box to set entire HDD parameters can be seen in Fig 3.5.

Fig 3.5 HDD block

The mountings is modeled as a block having spring and damper, therefore the block used is Body spring and damper block. Within this block, we can set the spring and damper coefficients. The block can be seen in Fig 3.6.

The shock input is generated using several Simulink block. The blocks can be seen in Fig 3.7. The blue block is the embedded MATLAB® editor to create the routine. The routine is a function of pulse width (w) and delay time (delay time is introduced to see the effect without any external force to HDD). The half sine generated from the blue block is then amplified by a G's number (the triangle block). The entire parameters to generate the shock input are reconfigurable.

Users are able to change the shock duration (in ms) as well as the shock magnitude (in G's).

Fig 3.6 Body spring and damper block

Fig 3.7 Half sine generators

The body sensor is constructed by connecting several blocks. The body sensor composition can be seen in Fig 3.8. The joint sensor measures the acceleration, velocity, and displacement of the port connected with the body sensor. In our case, we have five body sensors to measure each HDD corner and CG responses.

Fig 3.8 Body sensor compositions

To connect cantilever beam with body spring and damper, we use a subsystem which converts the responses of cantilever beam (displacement and velocity) into forces. This subsystem also measures the reaction forces from mountings and sends the force to nodes in cantilever beam (as external forces from mountings). The subsystem compositions can be seen in Fig 3.9. The cantilever beam velocity responses are derived into acceleration, and then all of the responses (displacement, velocity, and acceleration) are converted into forces by 'joint actuator' block. The forces are sent to the HDD through body spring and damper. The reaction forces are then measured by joint sensor and sent back to cantilever beam nodes.

Fig 3.9 Subsystem compositions

With all of model components created and connected each other, the Simulink model of HDD – cantilever beam shock isolator is ready to be simulated. The entire Simulink model can be seen in Fig 3.10.

Fig 3.10 Simulink model of HDD – cantilever beam shock isolator

Within this section, the entire simulation procedures have been introduced. The next section will discuss the details of each step in the procedures, perform design optimization by simulating the problem using the Simulink model, and perform a FE analysis in ANSYS to compare the simulation result using Simulink model.

3.3 Shock Response of Single DOF System

As earlier explained in previous section, we will first study the response of single DOF system to a shock input. The main purpose of this study is to find the ‘compromise’ beam’s first bending natural frequency. The system which will be studied is the base excitation problem; which resembles the 1st mode of the beam – small form factor HDD configuration given in Fig 2.5. Fig 3.11 shows the single DOF system for base excitation problem, with m defined as 18 gram (ST1 mass).

Fig 3.11 Base excitation problem single DOF systems

The governing equation of motion of mass m using Newton’s law is given by:

$$c\dot{y}_2 + ky_2 = m\ddot{y}_1 + c\dot{y}_1 + ky_1 \quad (3.19)$$

The force input from the base is expressed in base displacement (y_2) and base velocity ($\dot{y}_2$). To construct the base displacement and base velocity referring to shock input that applied to the base, we will consider Fig 3.12 as the base subjected to the shock input $F(t)$.

Fig 3.12 Base subjected to a shock input $F(t)$

The governing equation of motion of the base is:

$$\begin{aligned}\sum F_y &= M\ddot{y}_2 \\ F(t) &= M\ddot{y}_2\end{aligned}\tag{3.20}$$

The shock input $F(t)$ is constructed by adding two similar sine waves. One of the sine waves has a time delay that is half of the sine wave period. These two sine waves are given in Fig 3.13.

Fig 3.13 Sine waves to construct the shock input

The first sine wave $F_1(t)$ is the wave shown with symbol 'x', while $F_2(t)$ is the solid line wave that has time delay of 0.05 s. If we add $F_1(t)$ to $F_2(t)$, the resulting wave is a shock input given in Fig 3.14. This figure shows shock input with shock duration 0.05 s and 130000 N amplitudes. These figures are just an example on how to construct the shock function as an input to the base.

Fig 3.14 Shock input

Referring to the method to construct the shock input, equation (3.20) is written as:

$$\begin{aligned}\sum F_y &= M\ddot{y}_2 \\ F_1(t) + F_2(t) &= M\ddot{y}_2\end{aligned}\quad (3.21)$$

Since the single DOF model is a linear system, the base response can be divided into two parts; response from $F_1(t)$ and from $F_2(t)$. The total response will be the total of the two responses.

Mass M is chosen large such as 1 kg mass. An m – file in MATLAB® is prepared to calculate the system response based on the above methods. By using 1000 G's shock input with 0.5 ms duration, the displacement, velocity, and acceleration of the base is given in Fig 3.15.

The base responses as displacement and velocity will be the input for equation (3.19) to obtain the response of mass m . A MATLAB® program was used to calculate the mass m response by setting the damping ratio of the system as 0% and varied the natural frequency of the system. This is to see the effect of the natural frequency to the response of the mass m . The acceleration response and the displacement response of mass m are given in Figs 3.16 and 3.17.

Fig 3.15 Base responses to a shock input

Fig 3.16 Mass m acceleration response at various natural frequencies

Fig 3.17 Mass m displacement response at various natural frequencies

It is clear from Figs 3.16 and 3.17 that the compromise system natural frequency is around 100 to 200 Hz. The acceleration response and the displacement response from the two natural frequencies are relatively low. This value of natural frequency is very useful

to determine the beam size and material to obtain the ‘compromise’ acceleration and displacement response of the HDD.

Since we will use a cantilever beam as the shock isolator, the second elastic natural frequency of the cantilever beam plus the HDD becomes important too. The second elastic natural frequency of the cantilever beam plus the HDD must be such that the input shock frequency is in between the first bending natural frequency and the second elastic natural frequency (torsional mode) of the cantilever beam plus the HDD.

3.4 Modal Analysis in ANSYS

Fig 3.18 Cantilever beam – HDD shock isolator with force at the tip

Consider a cantilever beam, length L , with small form factor HDD attached over it in Fig 3.18. The bottom cantilever beam is an equivalent model of the top; the HDD is now become a concentrated mass M . Therefore, the equivalent tip stiffness of the bottom model is:

$$k_{bending} = \frac{3EI}{L_1^3} \quad (3.22)$$

$$k_{torsion} = \frac{GJ}{L^1}$$

According to the results in the previous section, the 1st bending natural frequency of the cantilever beam plus ST1 is tuned between 100 Hz and 200 Hz. By using the formula for single degree of freedom system, we can estimate the 1st bending natural frequency of the system in Fig 3.18:

$$2\pi f_1 = \sqrt{\frac{k_{bending}}{m}}$$

where:

$$m = m_{HDD} + m_{beam}$$

$$m = 0.018kg + 0.23\rho_{beam} \cdot w \cdot L_1 \cdot t$$

(3.23)

Where **w** is the beam width, and **t** is the beam thickness.

As the size constraint, the beam's width is set the same as the ST1 width (35 mm). Concerning the shock isolator thickness, the beam's thickness is a crucial matter. It affects the shock isolator size, total mass, and the natural frequency of the system. By observing the available small form factor (micro) HDD product in the market, the shock isolator thickness is limited to 10 mm. For this purpose, the beam's thickness is determined as 1 mm. Then, Aluminum is selected as the initial material attempt for the beam. The minimum length of the beam is 43 mm (the same as the HDD's length), and the maximum length of the beam is 65 mm (design envelope). So for the initial modeling we will try three different beam lengths, which are 50 mm, 55 mm, and 60 mm. The resulting system's first bending natural frequency of the model in Fig 3.18, by using those beam lengths is given below:

$$2\pi f_1 = \sqrt{\frac{3EI}{mL_1^3}}$$

$$(2\pi f_1)^2 = \frac{3E \frac{1}{12} wt^3}{(0.018 + 0.23\rho_{beam} \cdot w \cdot L_1 \cdot t)L_1^3} \quad (3.25)$$

Inserting known variables, equation (3.25) becomes:

$$(2\pi f_1)^2 = \frac{\frac{1}{4} 70 \cdot 10^9 \cdot 0.035 \cdot 0.001^3}{(0.018 + 0.23 \cdot 2700 \cdot 0.035 \cdot 0.001 \cdot 0.0385) 0.0385^3} \Rightarrow f_n = 120Hz$$

$$(2\pi f_1)^2 = \frac{\frac{1}{4} 70 \cdot 10^9 \cdot 0.035 \cdot 0.001^3}{(0.018 + 0.23 \cdot 2700 \cdot 0.035 \cdot 0.001 \cdot 0.0335) 0.0335^3} \Rightarrow f_n = 148Hz$$

$$(2\pi f_1)^2 = \frac{\frac{1}{4} 70 \cdot 10^9 \cdot 0.035 \cdot 0.001^3}{(0.018 + 0.23 \cdot 2700 \cdot 0.035 \cdot 0.001 \cdot 0.0285) 0.0285^3} \Rightarrow f_n = 189Hz$$

To check whether the beam configurations satisfy the system's natural frequency in torsional mode, we perform a modal analysis in ANSYS using the above configurations.

The finite element model of the cantilever beam plus the HDD and its boundary conditions (BCs) is given in Fig 3.19. The HDD is mounted to the cantilever beam through four mountings. For this modal analysis, the mounting stiffness is very high. The left part of the beam is constrained so it can only move in the Z direction. The left part of the beam is also coupled with the shaker node in the Z direction. This is to simulate a shock analysis.

Fig 3.19 FE model of cantilever beam plus HDD and its BCs

The first four elastic mode shapes of the FE model of Fig 3.19 are given in Fig 3.20. And then, the eigenvalues for each beam lengths are given in Table 3.1.

Fig 3.20 The first four elastic mode shapes of the FE model of Fig 3.19

Table 3.1 Eigenvalues of the FE model of Fig 3.19 with three different beam lengths

Mode (elastic)	50 mm	55 mm	60 mm
1 st (Hz)	188	142	114
2 nd (Hz)	1888	1072	729
3 rd (Hz)	2410	1836	1319
4 th (Hz)	3460	2670	2586

The three beam lengths (50 mm, 55 mm, and 60 mm) are satisfying the first bending natural frequencies and design envelope, but only the beam with 50 mm length satisfy the second elastic natural frequency. From the first elastic natural frequency point of view, beam with 60 mm length is the best. In contrary, to separate the first elastic natural frequency and the second elastic natural frequency, beam with 50 mm length is the best (note that the shock input duration is 0.5 ms, which is equal to 1000 Hz input).

To keep the first elastic natural frequency low enough while make the second elastic natural frequency very high (more than 1000 Hz), we will use a composite as the beam material. A cantilever beam made from carbon fiber composite can be made such the bending stiffness is low enough while the torsion stiffness is very high. Moreover, the carbon fiber composite can be designed and built to have a very high damping ratio. Thus, carbon fiber composite will be used as the beam material.

In our simulation, we will use the same material as the above beam configurations, with beam length 60 mm (to obtain the lowest first elastic natural frequency), using Young's modulus and density of Aluminum, but we will increase the shear modulus (G) to increase the torsion stiffness. The shear modulus (G) is started from 70 GPa, and increased by two times factor (140 GPa, 280 GPa, 560 GPa, 1120 GPa, 2240 GPa, and 4480 GPa). The first four elastic natural frequencies of the beam with increased shear modulus are given in Table 3.2.

Table 3.2 Eigenvalues of the FE model in Fig 3.19 with increased beam shear modulus

Mode	70 GPa	140 GPa	280 GPa	560 GPa	1120 GPa	2240 GPa	4480 GPa
1 st (Hz)	119	121	122	123	124	125	126
2 nd (Hz)	926	1142	1458	1717	1783	1838	1890
3 rd (Hz)	1467	1555	1641	1894	2435	2937	3215

4 th (Hz)	2878	3040	3206	3369	3518	3647	3755
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From Table 3.2, the second elastic natural frequency that satisfies the requirement is achieved by using the beam with G of 280 GPa and above.

Summarizing the above results, beam properties for the initial modeling:

- beam width : 35 mm
- beam thickness : 1 mm
- beam length : 60 mm
- density : 2700 kg/m³
- Young's modulus (GPa) : 70
- shear modulus (GPa) : 280, 560, 1120, 2240, and 4480

Next we will extract the beam eigenvalues and eigenvectors to create the state space equation of the cantilever beam.

The beam finite element model in ANSYS is given in Fig 3.21. The FE model uses 20 noded *solid 95* elements. The element's aspect ratio is 1. To resemble the real assembly condition, the left end of the beam (the entire nodes of the left end surface) is coupled to a keypoint that has a large mass. This keypoint represent a big shaker (the big shaker is the wall); the keypoint mass is set to 1 ton. Along with the keypoint, all of the coupled nodes are constrained in all direction except in the Z direction.

Fig 3.21 Cantilever beam finite element model in ANSYS

As explained before in the assumptions of section 3.1, there are five nodes that will be extracted from the beam. The first node is the wall, the second and the third node are the beam nodes that coincide with the left mounting, and the last two nodes are the beam nodes that coincide with the right mounting (see Fig 3.2). The details are explained in Fig 3.22. LT and LB means Left Top and Left Bottom respectively, while RT and RB means Right Top and Right Bottom. The distance between the left and right nodes is exactly the same as the length of the ST1 (43 mm).

Fig 3.22 Extracted nodes location

The mode shapes, eigenvalues, and eigenvectors of beam for the respective nodes will be listed in the following section.

3.4.1 Mode Shapes of the Cantilever Beam

The boundary condition for modal analysis of the cantilever beam is explained in Fig 3.21. The left end of the beam (the entire nodes of the left end surface) is coupled to a keypoint that has a large mass. This keypoint represent a big shaker (the big shaker is the wall); the keypoint mass is set to 1 ton. Along with the keypoint, all of the coupled nodes are constrained in all direction except in the z direction. The shock input is in the Z direction, so uz needs to be free.

The mode shapes of the beam, shown as the deformed and un-deformed shape, are shown below in Fig 3.23 (chronologically from the first mode to the twelfth mode):

1st

2nd

3rd

4th

5th

6th

7th

8th

9th

Fig 3.23 Mode shapes of the cantilever beam

There are only twelve mode shapes kept to construct the state space equations. Based on the shock input duration (0.5 ms), the modes that will contribute the most are the modes with natural frequency below 1000 Hz. Using too few modes will reduce the accuracy, but using too many modes will waste computer resources when the accuracy is not increase anymore. A 0.5 ms shock input is equivalent to a 1000 Hz input. So keeping 12 modes (12th mode at 14000 Hz to 26000 Hz) is good enough to accurately capture the response of the beam to a 1000 Hz input.

3.4.2 Eigenvalues and Eigenvectors

This section lists the eigenvalues and eigenvectors of the beams extracted from the modal analysis results in ANSYS.

Table 3.3 Eigenvalues and Eigenvectors of 60 mm beam with G 280 GPa

	1 st (0 Hz)	2 nd (246 Hz)	3 rd (1531 Hz)	4 th (2492 Hz)	5 th (4320 Hz)	6 th (6752 Hz)
Wall	0.999997	-1.86E-03	1.03E-03	9.25E-14	-6.09E-04	-6.78E-05
RB	0.999997	841.3736	-803.89687	-1025.05	716.289138	1126.449
LB	0.999997	96.40891	427.29027	-384.402	616.133557	561.7181
RT	0.999997	841.3736	-803.89687	1025.049	716.289138	1126.449

LT	0.999997	96.40891	427.29027	-384.402	616.133557	561.7181
	7 th (7382 Hz)	8 th (7552 Hz)	9 th (8554 Hz)	10 th (12644 Hz)	11 th (14115 Hz)	12 th (14686 Hz)
Wall	2.11E-14	-4.14E-13	4.31E-04	3.15E-13	-3.35E-04	2.65E-12
RB	-980.964	-0.00019	927.20606	-979.411	714.272499	1152.057
LB	853.3511	1.625282	-406.94301	-824.489	75.705614	530.5391
RT	980.964	0.000186	927.20606	979.4109	714.272499	-1152.06
LT	853.3511	1.625282	-406.94301	-824.489	75.705614	530.5391

Table 3.4 Eigenvalues and Eigenvectors of 60 mm beam with G 560 GPa

	1 st (0 Hz)	2 nd (247 Hz)	3 rd (1543 Hz)	4 th (3431 Hz)	5 th (4352 Hz)	6 th (7623 Hz)
Wall	0.999997	-1.86E-03	1.03E-03	3.96E-14	-6.08E-04	7.50E-14
RB	0.999997	841.7429	-817.17922	-988.626	774.891056	0.000361
LB	0.999997	98.04678	420.16946	-385.752	624.765562	1.652372
RT	0.999997	841.7429	-817.17922	988.626	774.891056	-0.00036
LT	0.999997	98.04678	420.16946	385.7516	624.765563	-1.65237
	7 th (8205 Hz)	8 th (8655 Hz)	9 th (10043 Hz)	10 th (14188 Hz)	11 th (16358 Hz)	12 th (16964 Hz)
Wall	-0.00019	0.000389	-1.755E-15	-0.00034	4.8526E-16	-3.6E-14
RB	670.3662	1229.504	-924.5417	818.7198	1089.18365	-1027.71
LB	642.8896	-208.835	840.79865	1.067137	441.43264	-829.054
RT	670.3662	1229.504	924.5417	818.7198	-1089.1837	1027.711
LT	642.8896	-208.835	-840.79865	1.067137	-441.43264	829.054

Table 3.5 Eigenvalues and Eigenvectors of 60 mm beam with G 1120 GPa

	1 st (0 Hz)	2 nd (248 Hz)	3 rd (1554 Hz)	4 th (4377 Hz)	5 th (4732 Hz)	6 th (7662 Hz)
Wall	0.999997	-1.86E-03	1.03E-03	-6.07E-04	1.88E-14	1.62E-13
RB	0.999997	842.25	-827.59689	806.7414	-949.90198	0.001084
LB	0.999997	99.02625	414.3928	629.5155	-380.62009	1.666789
RT	0.999997	842.25	-827.59689	806.7414	949.901981	-0.00108
LT	0.999997	99.02625	414.3928	629.5155	380.620093	-1.66679
	7 th (8597 Hz)	8 th (10634 Hz)	9 th (13840 Hz)	10 th (14260 Hz)	11 th (19194 Hz)	12 th (21332 Hz)
Wall	4.32E-04	3.68E-05	3.41E-13	-3.37E-04	1.10E-11	2.76E-04
RB	742.4787	1116.477	889.62613	846.3818	1155.37522	796.2129
LB	-502.709	426.7401	-830.27875	2.237353	489.802264	468.7748
RT	742.4787	1116.477	-889.62613	846.3818	-1155.3752	796.2129
LT	-502.709	426.7401	830.27875	2.237353	-489.80226	468.7748

Table 3.6 Eigenvalues and Eigenvectors of 60 mm beam with G 2240 GPa

	1 st (0 Hz)	2 nd (250 Hz)	3 rd (1565 Hz)	4 th (4402 Hz)	5 th (6547 Hz)	6 th (7683 Hz)
Wall	0.999997	-1.86E-03	1.03E-03	-6.07E-04	-4.24E-13	1.07E-12
RB	0.999997	842.8377	-834.63407	823.9547	914.316882	0.001982
LB	0.999997	99.38558	409.97026	631.975	373.208669	1.674358
RT	0.999997	842.8377	-834.63407	823.9547	-914.31688	-0.00198
LT	0.999997	99.38558	409.97026	631.9751	-373.20867	-1.67436
	7 th (8651 Hz)	8 th (14024 Hz)	9 th (14375 Hz)	10 th (19250 Hz)	11 th (21493 Hz)	12 th (21693 Hz)
Wall	4.33E-04	1.04E-04	-3.22E-04	-2.69E-09	2.77E-04	-3.25E-08
RB	807.7842	746.746	1088.9902	-869.349	821.724885	-0.0007
LB	-481.562	408.7365	117.32693	823.1791	461.866583	-2.3022
RT	807.7842	746.746	1088.9902	869.3486	821.724867	-0.00068
LT	-481.562	408.7365	117.32693	-823.179	461.866581	-2.30219

Table 3.7 Eigenvalues and Eigenvectors of 60 mm beam with G 4480 GPa

	1 st (0 Hz)	2 nd (251 Hz)	3 rd (1575 Hz)	4 th (4430 Hz)	5 th (7695 Hz)	6 th (8714 Hz)
Wall	0.999997	-1.86E-03	1.03E-03	-6.07E-04	5.81E-13	4.34E-04
RB	0.999997	843.5683	-839.18956	833.7228	0.00310096	827.6668
LB	0.999997	99.20379	406.33591	633.1669	1.67819603	-478.888
RT	0.999997	843.5683	-839.18956	833.7228	-0.0031007	827.6668
LT	0.999997	99.20379	406.33591	633.1669	-1.6781973	-478.888
	7 th (9102 Hz)	8 th (14467 Hz)	9 th (18978 Hz)	10 th (21701 Hz)	11 th (21724 Hz)	12 th (26937 Hz)
Wall	-5.08E-13	-3.39E-04	-8.53E-06	9.50E-11	2.78E-04	1.37E-08
RB	886.7781	815.4896	963.67867	-0.00111	848.851237	857.6892
LB	366.9075	-6.80802	401.03895	-2.29992	461.535552	-818.988
RT	-886.778	815.4896	963.67867	-0.00112	848.851237	-857.689
LT	-366.908	-6.80802	401.03895	-2.29992	461.535552	818.9883

3.5 State Space Equation of Cantilever Beam

As explained earlier in section 3.2, section 3.2.4, the beam's damping ratio is determined when constructing the state space equation. Aluminum has damping ratio around 2% to 3%. In our MATLAB® (Simulink) simulation, the damping property of aluminum was assumed constant for all frequencies ($\zeta_1=\zeta_2=\zeta_3=\zeta_4=.....=\zeta_{12}$). The state space equation will be in the form (refer to section 3.2.4 and reference [30]):

$$\begin{aligned}\dot{x} &= Ax + Bu \\ y &= Cx + Du\end{aligned}\tag{3.26}$$

Where the A matrix is given as follows:

$$A = \begin{bmatrix} 0 & 1 & & & & \\ -\omega_1^2 & -2\zeta_1\omega_1 & & & & \\ & & \ddots & \ddots & & \\ & & & \ddots & \ddots & \\ & & & & 0 & 1 \\ & & & & -\omega_{12}^2 & -2\zeta_{12}\omega_{12} \end{bmatrix}\tag{3.27}$$

The size of A matrix will be 24 by 24, that is two times of the number of the modes kept.

The B matrix will be slightly different compared to B matrix mentioned in section 3.2.4. The cantilever beam will have 5 different external forces. They are wall forces, LT and LB forces, and RT and RB forces. The B matrix is then:

$$B = \begin{bmatrix} 0 & 0 & 0 & 0 & 0 \\ U_{1Wall} \cdot 5000 \cdot 9.81 & U_{1LT} & U_{1RT} & U_{1LB} & U_{1RB} \\ \cdot & \cdot & \cdot & \cdot & \cdot \\ \cdot & \cdot & \cdot & \cdot & \cdot \\ 0 & 0 & 0 & 0 & 0 \\ U_{12Wall} \cdot 5000 \cdot 9.81 & U_{12LT} & U_{12RT} & U_{12LB} & U_{12RB} \end{bmatrix} \quad (3.28)$$

The size of B matrix will be 24 rows and 5 columns. The 24 rows are from the number of the mode shapes kept, and the 5 columns are the number of the external forces. Particularly at the first column (external force from the wall) the normalized eigenvectors are multiplied by 49050 (5000 x 9.81). The number 5000 is the shaker (wall) mass defined in ANSYS, while 9.81 is the gravity acceleration. Later at Simulink environment, the shock magnitude will be defined in G's, not in m/s². Hence, the value of 9.81 m/s² must be included in the state space formulation. The rest of the columns will be the normalized eigenvectors at each point (LT, LB, RT, and RB). The external forces here are denoted by u₁ to u₅.

The C matrix is the output matrix. As explained in section 3.2.5, we want to obtain ten outputs from the state space equations. They are the wall, LT, LB, RT, and RB displacements and velocities. The C matrix is then divided into C matrix for displacement outputs and C matrix for velocity outputs. The C matrix for displacement outputs is:

$$C_{displacement} = \begin{bmatrix} U_{1Wall} & 0 & \cdot & \cdot & U_{12Wall} & 0 \\ U_{1LT} & 0 & \cdot & \cdot & U_{12LT} & 0 \\ U_{1RT} & 0 & \cdot & \cdot & U_{12RT} & 0 \\ U_{1LB} & 0 & \cdot & \cdot & U_{12LB} & 0 \\ U_{1RB} & 0 & \cdot & \cdot & U_{12RB} & 0 \end{bmatrix} \quad (3.29)$$

And the C matrix for velocity outputs is:

$$C_{velocity} = \begin{bmatrix} 0 & U_{1Wall} & \cdot & \cdot & 0 & U_{12Wall} \\ 0 & U_{1LT} & \cdot & \cdot & 0 & U_{12LT} \\ 0 & U_{1RT} & \cdot & \cdot & 0 & U_{12RT} \\ 0 & U_{1LB} & \cdot & \cdot & 0 & U_{12LB} \\ 0 & U_{1RB} & \cdot & \cdot & 0 & U_{12RB} \end{bmatrix} \quad (3.30)$$

The size of C matrix is 24 columns that represent the number of modes kept with 5 rows representing the five different outputs (wall, LT, LB, RT, and RB).

The D matrix is left as zero.

3.6 Beam Configuration and Simulation Result

The cantilever beam (Aluminum) damping ratio is set to 2%. Then the mounting stiffness is set to a very high stiffness, 1×10^6 N/m, to simulate a HDD bonded to the beam without any shock isolator. The shock input magnitude is 1000 G's with shock duration of 0.5 ms. The displacement and acceleration response at left corner, CG, and right corner of the HDD are given in Figs 3.24 to 3.24 (see Fig 3.1 for left corner, CG, and right corner).

Fig 3.24 Acceleration and displacement response at left corner of HDD

Fig 3.25 Acceleration and displacement response at HDD's CG

Fig 3.26 Acceleration and displacement response at right corner of HDD

To observe the results carefully, we list the peak acceleration response and displacement response of the HDD in tables 3.8 and 3.9. Refer to Fig 3.1 for Left corner, CG, and Right corner position.

Table 3.8 Peak of the acceleration responses of the HDD

	Left corner (G)	CG (G)	Right corner (G)
G 280 GPa	996	500	878
G 560 GPa	1037	439	841

G 1120 GPa	1047	437	840
G 2240 GPa	1051	434	842
G 4480 GPa	1054	435	844

Table 3.9 Peak sway of the displacement responses of the HDD

	Left corner (mm)	CG (mm)	Right corner (mm)
G 280 GPa	1.68	6.2	10.96
G 560 GPa	1.54	6.16	10.92
G 1120 GPa	1.54	6.15	10.9
G 2240 GPa	1.53	6.14	10.87
G 4480 GPa	1.52	6.12	10.84

In the next section, we will perform a parametric studies on the beam damping ratio by assuming that there is a material that has higher damping ratio (limited to 0.5, refer to Table 2.2) but has the Young's Modulus and the density the same as the Aluminum, and has the shear modulus (G) as listed in Table 3.8 and 3.10; this is to keep the same eigenvalues and eigenvectors discussed in this section.

3.7 Optimal Damping Ratio

In this simulation, the 60 mm beam length is used. With the same mounting's properties and the same shock input (1000 G's with 0.5 ms duration), the cantilever beam's damping ratio is varied and the acceleration responses are compared to find the optimum damping ratio. The damping ratio is varied from 0.1 to 0.5 with 0.1 increments. Tables 3.10 to 3.12 show the peak acceleration response of the HDD.

Table 3.10 Peak of the acceleration responses on the Left corner of the HDD (in G's)

	$\zeta = 10\%$	$\zeta = 20\%$	$\zeta = 30\%$	$\zeta = 40\%$	$\zeta = 50\%$
G 280 GPa	983	976	974	974	975
G 560 GPa	1024	1015	1010	1008	1007
G 1120 GPa	1031	1021	1015	1013	1011
G 2240 GPa	1036	1025	1019	1016	1015
G 4480 GPa	1040	1030	1025	1020	1018

Table 3.11 Peak of the acceleration responses on the HDD CG (in G's)

	$\zeta = 10\%$	$\zeta = 20\%$	$\zeta = 30\%$	$\zeta = 40\%$	$\zeta = 50\%$
G 280 GPa	408	333	308	314	320
G 560 GPa	353	295	287	292	297
G 1120 GPa	352	295	286	291	296
G 2240 GPa	350	294	285	290	294
G 4480 GPa	349	293	285	289	294

Table 3.12 Peak of the acceleration responses on the Right corner of the HDD (in G's)

	$\zeta = 10\%$	$\zeta = 20\%$	$\zeta = 30\%$	$\zeta = 40\%$	$\zeta = 50\%$
G 280 GPa	708	574	489	439	427
G 560 GPa	652	538	470	425	392
G 1120 GPa	651	536	469	424	392
G 2240 GPa	652	536	469	425	393
G 4480 GPa	652	537	470	426	394

The simulation results of Tables 3.10 to 3.12 show that the most appropriate shear modulus (G) for the beam material is 280 GPa. The acceleration responses are reduced at all point (Left corner, CG, and Right corner). The simulation results also show that the acceleration responses at the HDD corners are reduced significantly with the increase in the beam damping ratio, but the acceleration response at the HDD CG starts to increase when the damping ratio is greater than 0.3. However, the increasing in acceleration response at CG is not significant, while the acceleration responses at the HDD corners are reduced significantly. Therefore, we conclude that the most appropriate beam's damping ratio is 0.5.

Fig 3.27 shows the acceleration response of the HDD corners and HDD CG using the above configuration (G 280 GPa, E 70 GPa, and $\zeta = 50\%$).

The acceleration responses at HDD corners can be reduced by reducing the mounting stiffness, since the mounting stiffness used in this simulation is very high (1×10^6 N/m). The next step; which is to adjust the mounting stiffness to reduce acceleration response of the HDD, will be discussed.

Fig 3.27 Acceleration response of HDD

3.8 Adjusting The Mounting Stiffness and Damping Ratio

Most likely, the mountings will be made of rubber or plastics (polymers). Metal springs, as mounting, are difficult to use because the design envelope for the mounting is very small (less than 5 mm). Moreover, metal springs are lightly damped and a damping ratio of more than 5% is impossible. Previously, the mounting stiffnesses were set to a very high number, and the transmissibility ended up to be very high. To reduce the transmissibility, the mounting stiffnesses must be reduced. But there are some risks if we want to reduce the mounting stiffnesses:

- If the mounting stiffnesses are too low, there is a possibility that the HDD may collide with the beam at large shock input
- If the mounting stiffnesses are too low, the corner HDD displacements will be very large
- Lower mount stiffness means the mounting thickness must be increased to avoid HDD collision with the beam

Aware of the risks involved in adjusting the mounting stiffness, first we need to calculate the force generated in the mounts. By knowing each mount force, the stiffness can be adjusted as well as each mounting thickness.

Calculating analytically the force acts at the mounting is not convenient. So simulation is used to measure mount forces. A transient analysis in ANSYS is prepared by setting the same shock isolator configuration. The beam has 60 mm length, has 70 GPa of Young's modulus, 280 GPa of shear modulus, and 0.5 damping ratio. Fig 3.28 shows the cantilever beam - HDD shock isolator model in ANSYS, which is ready for shock response simulation. The mounting dimension in ANSYS is defined as 2 mm x 2 mm x 4 mm (cubic). To obtain the same stiffness as of 1×10^6 N/m, the Young's modulus of the mounting is set to 1 GPa; whereas the damping coefficient of the mounting is set as zero (0) to capture only the effect of its stiffness. The force acting at the mounting is obtained by measuring the mounting deflection. This deflection then multiplied by the mounting stiffness to obtain the force, according to formula:

$$F = E \frac{A}{L} \cdot \Delta x \quad (3.31)$$

Where E is mounting's Young's modulus, A is mounting cross section area, L is mounting initial length, and Δx is mounting deformation.

Fig 3.28 Cantilever beam - HDD shock isolator finite element model in ANSYS

Fig 3.29 shows the mounting nodes at left and right mounting to define its deformation. The deformation at each mounting is defined within three nodes: left, middle, and right node. The deformation at each node of left and right mounting is given in Fig 3.30.

Fig 3.29 Nodes mapping for left and right mounting

Fig 3.30 Left and right mounting deformation. Negative = compression, positive = tension

For the left mounting, we can consider the left and the middle nodes are the nodes with dominant deformation, whereas for the right mounting the dominant deformation is at the

middle and the right nodes. Thus, the maximum compression deformation at left mounting is 0.01824 mm; whereas the right mounting is 0.00487 mm. The maximum compressive forces acting on the left and right mounting are then:

$$\begin{aligned} F_{left} &= 0.01824 \text{ mm} \cdot 1000 \text{ MPa} \cdot \frac{4 \text{ mm}^2}{4 \text{ mm}} = 18.24 \text{ N} \\ F_{right} &= 0.00487 \text{ mm} \cdot 1000 \text{ MPa} \cdot \frac{4 \text{ mm}^2}{4 \text{ mm}} = 4.87 \text{ N} \end{aligned} \quad (3.32)$$

If we set the maximum deflection at the left and right mounting as 0.4 mm or 40%² of compressive strain [32], the lowest possible mounting stiffness for 1 mm mounting thickness are:

$$\begin{aligned} k_{left} &= \frac{18.24 \text{ N}}{0.4 \text{ mm}} = 45.6 \text{ kN/m} \\ k_{right} &= \frac{4.87 \text{ N}}{0.4 \text{ mm}} = 12.175 \text{ kN/m} \end{aligned} \quad (3.33)$$

It is still possible to reduce more the mounting stiffness by increasing the mounting thickness. With the previous maximum compressive force, we can estimate the lowest mounting stiffness with its corresponding mounting thickness. The lowest mounting stiffness for various mounting thicknesses are listed in table 3.13.

Table 3.13 Various mounting thicknesses with its lowest possible stiffness (roughly estimated)

Mounting thickness (mm)	Maximum deflection (mm)	Lowest left mounting stiffness (kN/m)	Lowest right mounting stiffness (kN/m)
1	0.4	45.6	12.175
2	0.8	22.8	6.0875
3	1.2	15.2	4.058
4	1.6	11.4	3.04

The force measured using ANSYS simulation is from the mounting deflection because the damping ratio of the mounting is set to zero. To reduce stiffness while retain the same deformation, the mounting damping ratio can be adjusted.

To investigate on how much damping ratios needed to reduce the mounting stiffness while retain the same deformation, a single DOF system (see base excitation problem in

² We assume that for 40% of strain, the mounting stiffness does not change too much

section 3.3) is used. The single DOF system cannot accurately predict the effect of mounting's damping ratio on cantilever beam - HDD shock isolator system, but it can roughly estimate how much the mounting stiffness can be reduced if we add more damping ratio to sustain 40% of maximum deformation. This is better and faster than performing a trial and error simulation to find the lowest possible mounting stiffnesses. For this purpose, we are only interested in left mounting, because it has higher deformation than the right mounting.

Fig 3.31 Modeling of mounting and a quarter HDD system

The single DOF model is given in Fig 3.31. The 850 G's shock input is retrieved from the previous ANSYS simulation result. This is the maximum acceleration given from the beam node to the left mounting node. The stiffness is set to 45.6 kN/m, 22.8 kN/m, 15.2 kN/m, and 11.4 kN/m (the lowest stiffness from table 3.9). The simulation result is then given in table 3.14.

Table 3.14 Maximum deformation for single DOF base excitation problem with various damping ratios

Mounting damping ratio (%)	Maximum deformation (mm) 45.6 kN/m stiffness	Maximum deformation (mm) 22.8 kN/m stiffness	Maximum deformation (mm) 15.2 kN/m stiffness	Maximum deformation (mm) 11.4 kN/m stiffness
0	0.7801	1.138	1.408	1.634
10	0.6731	0.9816	1.215	1.41
20	0.5901	0.8605	1.065	1.236
30	0.5241	0.7642	0.9456	1.097
40	0.4707	0.6862	0.849	0.9853
50	0.4265	0.6218	0.7692	0.8927

From the result of table 3.14, it can be roughly estimated that to retain the same deformation while reducing the system stiffness by 25%, the damping ratio needs to be 12%. To reduce the stiffness by 50%, the damping ratio needs to be 26%. Lastly, to reduce the stiffness by 75%, we need 45% of damping ratio. From this estimation, we choose the mounting damping ratio in range of 10% to 30%, because higher damping ratio is more difficult to be provided. Table 3.15 gives the lowest possible left mounting stiffness for various mounting thicknesses with damping ratio of 12% and 26%.

Table 3.15 Lowest possible left mounting stiffness for various mounting thickness with selected damping ratio (estimate values)

Mounting thickness (mm)	Mounting damping ratio (%)	Lowest left mounting stiffness (kN/m)
1	12	34.2
	26	22.8
2	12	17.1
	26	11.4
3	12	11.4
	26	7.6
4	12	8.55
	26	5.7

The next step is to adjust the stiffness at left and right mounting. Another problem is that the acceleration response at HDD corners and its CG is not at the same level. This is the second problem after the very high acceleration response at HDD corners. These two problems are recognized due to the high mounting stiffness and the difference in cantilever beam stiffness along its length. The mounting stiffness now can be reduced by using table 3.15 as reference. And then, the difference in beam stiffness can be solved by setting different mounting stiffnesses. This idea is explained in Fig 3.32.

Fig 3.32 Equivalent cantilever beam stiffness at left and right mountings

Using equation in Fig 3.32, the stiffness ratio (at L_1 and L_2) can be calculated:

$$\frac{k_{beam-L_1}}{k_{beam-L_2}} = \frac{\frac{3EI}{L_1^3}}{\frac{3EI}{L_2^3}} = \frac{L_2^3}{L_1^3} = \frac{60^3}{17^3} = 43.96 \approx 44 \quad (3.34)$$

The stiffness of the beam at $L_1=17$ mm is 44 times greater than the beam stiffness at $L_2=60$ mm. So, equation (3.34) implies that the right mounting stiffness should be 44 times greater than the left mounting stiffness. This is to “counter-balance” the overall cantilever beam stiffness. The next section will use this section results by modifying the mounting stiffness and damping ratio.

3.9 Enhanced Cantilever Beam - HDD Shock Isolator Simulation

We will now simulate the cantilever beam – ST1 system with enhanced mounting properties. The mounting properties are adjusted according to the previous section’s results. The simulation is performed in Simulink environment as previously done in section 3.6 and 3.7, but we set several configurations in different simulation to determine the optimum mounting’s stiffness and damping ratio. The shock input used is 1000 G’s with 0.5 ms duration. The beam’s length used is 60 mm (see section 3.5); its damping ratio is assumed as 50% (see section 3.7). The mounting used is varied in its thickness and stiffness (in different simulation), with damping ratio 12% or 26%; these damping ratios was selected based upon previous analysis (see section 3.8). The mounting configuration is given in table 3.16. The acceleration and displacement responses of each configuration are given in Figs 3.33 to 3.40.

Fig 3.33 Acceleration and displacement response of configuration 1

Fig 3.34 Acceleration and displacement response of configuration 2

Fig 3.35 Acceleration and displacement response of configuration 3

Fig 3.36 Acceleration and displacement response of configuration 4

Fig 3.37 Acceleration and displacement response of configuration 5

Fig 3.38 Acceleration and displacement response of configuration 6

Fig 3.39 Acceleration and displacement response of configuration 7

Fig 3.40 Acceleration and displacement response of configuration 8

Table 3.16 Mounting configuration

Configuration	Mounting thickness (mm)	Mounting damping ratio	Left stiffness (kN/m)	Right stiffness (kN/m)
1	1	0.12	34.2	1504.8
2	1	0.26	22.8	1003.2
3	2	0.12	17.1	752.4
4	2	0.26	11.4	501.6

5	3	0.12	11.4	501.6
6	3	0.26	7.6	334.4
7	4	0.12	8.55	376.2
8	4	0.26	5.7	250.8

From the above results, one can conclude that the acceleration response of the HDD can be reduced significantly by reducing the mounting stiffness, and setting a different stiffness between left and right mountings. With the optimal configuration (configuration 8), the acceleration response of the HDD (from this simulation) is all below 300 G's.

3.10 Transient Analysis in ANSYS

The simulation performed in Simulink environment is approximate since only five degrees of freedom have been kept. Simulation in ANSYS using transient analysis is then performed to compare it with the simulation result from Simulink. From the transient analysis we can also retrieve the acceleration response at more than three points; this is to measure the dominant acceleration response more accurately. The FE model of HDD – cantilever beam shock isolator is given in Fig 3.41. The left mounting and its dimensions are defined as follows:

- Width = length : 2 mm (square)
- Height : 4 mm

To follow the above dimensions and set the left mounting stiffness to 5.7 kN/m and right mounting stiffness to 250.8 kN/m, the mounting material is selected to be polypropylene. Polypropylene family has the range of Young's modulus from 0.005 GPa to 4 GPa. The left mounting Young's modulus needed is 0.006 GPa, while the right mounting Young's modulus needed is 0.25 GPa. Polypropylene with carbon black filler and chopped glass fiber will suffice the above requirements.

The cantilever beam used for transient analysis is a cantilever beam that has material properties as Aluminum but with shear modulus 280 GPa and 50% damping ratio. Referring to table 2.2, composites can reach nearly 50% of damping ratio. The bending stiffness and the torsional stiffness of a composite material can be adjusted so it can behave like the beam used in previous Simulink analysis. In this transient analysis, we will use the previous beam dimensions and properties used in Simulink analysis.

Figure 3.42 shows the overview to measure the space needed to install the shock isolator with the HDD. The lateral space needed is basically the shock isolator thickness plus the lateral movement at the end of shock isolator. This is because the end of shock isolator has the largest vertical movement (in the Z direction); whereas the X and Y direction movements are considered very small. Figure 3.42 also gives the node number from the left corner to the right corner. With 10 elements from left to right corner, we can retrieve 20 acceleration responses at different nodes (points).

Fig 3.41 Cantilever beam shock isolator (60 mm length) with 4 mm mounting thickness finite element model in ANSYS

Fig 3.42 Measuring the space needed and node numbering

The transient analysis in ANSYS are presented and compared with MATLAB®/ Simulink analysis in Figs 3.43 to 3.48; and summarized in Table 3.17. According to the

ANSYS simulation, the peak displacement of Z_1 (the right top-most node of the HDD model) in positive direction (up) is 3.34 mm, while the peak displacement of Z_2 (the right bottom-most node of the beam model) in negative direction is 5.63 mm. Therefore, the space needed is as follows:

$$\text{space} = 3.34\text{mm} + 5\text{mm} + 4\text{mm} + 1\text{mm} + 5.63\text{mm}$$

$$18.97\text{mm}$$

Fig 3.43 MATLAB – ANSYS; acceleration response at left corner of HDD

Fig 3.44 MATLAB – ANSYS; acceleration response at HDD CG

Fig 3.45 MATLAB – ANSYS; acceleration response at right corner of HDD

Fig 3.46 MATLAB – ANSYS; displacement response at left corner of HDD

Fig 3.47 MATLAB – ANSYS; displacement response at HDD CG

Fig 3.48 MATLAB – ANSYS; displacement response at right corner of HDD

Table 3.17 Comparison Result from ANSYS and MATLAB®

	ANSYS (transient analysis)	Simulink/ MATLAB®	Simulink/MATLAB® error (%)
Left mounting deformation (mm)	1.47	1.6	14.28
Peak left corner acceleration (G's)	291	271	-6.7

Peak CG acceleration (G's)	195	190	-2.5
Peak right corner acceleration (G's)	260	275	-5.7
Space needed (mm)	t=18.97; w=35; l=60	t=18.88; w=35; l=60	-0.47

Figure 3.49 shows the peak acceleration at each node from left to right corners of the HDD; the node numbers are explained in Fig 3.42. In Fig 3.43 the left corner of the HDD is used as the reference point, hence the 1st node distance is 0 mm (the left corner), the 11th node distance is 21.5 mm, and the 20th node distance is 43 mm (the right corner). The average of the acceleration peak is 236 G's.

Fig 3.49 Peak acceleration responses from left to the right corner of the HDD

Figure 3.49 shows that the maximum acceleration response of the HDD's body is actually below 300 G's. Another HDD's response that becomes our concern is the rotational acceleration about CG. The maximum rotational acceleration about CG experienced by the HDD is $1.5 \times 10^5 \text{ rad/s}^2$ (ANSYS result). The number of $1.5 \times 10^5 \text{ rad/s}^2$ is obtained by expression below:

$$\begin{aligned}
 Rot_{acc-CG} &= \frac{Left_{acc} - CG_{acc}}{r} + \frac{CG_{acc} - Right_{acc}}{r} \\
 &= \frac{(Left_{acc} - Right_{acc})}{r}
 \end{aligned} \tag{3.35}$$

r (21.5 mm) is the distance from each HDD corner to the HDD CG (half of the ST1 length). So then we can substitute this rotational acceleration about HDD CG by giving

the corners of the HDD (left corner and right corner) a linear acceleration or shock. This is depicted in Fig 3.50. The rotary shock which is given to the HDD body is 650 G's with duration of 5 ms.

Fig 3.50 Rotational acceleration of HDD about its CG and similarity with linear acceleration

According to B. Gu et al [20], a rotary shock is not as harmful as linear shock for small form factor HDD. Refer to table 2.1; we can roughly estimate that the small form factor HDD can sustain rotary shock magnitude more than 3 times of linear shock magnitude. But to really verify the shock isolation technique, we will perform other simulations using MATLAB® and ANSYS in the next section.

3.11 Shock Isolator Performance Verification

In this section, we will verify the shock isolator performance by simulating the problem in simpler model. The simple model is derived from the pictorial HDD shock isolator of Fig 1.5. The HDD cross section plus the shock isolator of Fig 1.5 is redrawn in Fig 3.51.

Fig 3.51 Cross section of HDD plus external shock isolator

Fig 3.52 Simple model of Fig 3.51

We assume the HDD structure from the HDD outer case to the HAA rigid block is rigid. So the HDD in Fig 3.51 can be modeled as the simple model of Fig 3.52.

The simple model of Fig 3.52 is also used in Chapters VII and VIII. The formulating process of the simple model of Fig 3.52 in MATLAB® is available in Chapter VII.

The ST1 HAA FE model is given in Fig 3.53. From the modal analysis performed in Chapter VI, the first natural frequency of the suspension is 297 Hz, and the first natural frequency of the arm is 4000 Hz. The mode shape of the first bending of the arm is given in Fig 3.54. The modes shapes of the ST1'sHAA are given in Chapter VI, Table 6.1.

Fig 3.53 FE model of ST1 HAA

Fig 3.54 First mode of the arm (4000 Hz)

From ANSYS, the suspension mass is 1.102E-2 grams, while the arm mass is 7.157E-2 grams. The HAA rigid block plus the HDD outer case is 18 grams, which is the ST1 mass. The arm stiffness and the suspension stiffness are adjusted to result the first suspension natural frequency of 297 Hz and the first arm natural frequency of 4000 Hz. The damping in the system is given only to the shock isolator which is 50% (the same damping ratio from the cantilever beam). The simple model properties in Fig 3.52 are given in Table 3.18.

Table 3.18 Simple model properties in Fig 3.52

	Mass (grams)	Stiffness – K (N/m)	Damping – C (Ns ² /m)
Moving base	5E6		
HAA rigid block + HDD outer case	18	6014.6	10.405
Arm	7.157E-2	45208	0
Suspension + slider	1.102E-2	38.3755	0

Using the model in Fig 3.52, we will simulate the ST1 slider movement; with and without external shock isolator. Without the external shock isolator (we set the shock isolator stiffness very high), we will give the moving base a linear shock input of 175 G's 2 ms. This shock input is the ST1 shock tolerance. We will keep the displacement response of the slider using this configuration. Then we will simulate the model in Fig 3.52 using external shock isolator, in which the natural frequency of the HDD plus the shock isolator is tuned to 92 Hz (the natural frequency of the cantilever beam plus ST1; it can be seen from the displacement and acceleration response). We will find out the highest shock input in which the displacement of the ST1 slider must not exceed the previous configuration displacement response. We choose to look only at the displacement since the head slap occur from the slider movements. The problem is simulated in MATLAB®.

From the simulation of the simple model in MATLAB®, using cantilever beam plus four mounts as a shock isolator for ST1, ST1 can sustain a shock input up to 1050 G's 0.5 ms, 540 G's 1 ms, and 310 G's 2 ms. So from the current ST1 shock tolerance, the external shock isolator of cantilever beam plus four mounts has increased the ST1 shock tolerance by 77% for shock duration 2 ms. The displacement response of the ST1 slider is given in Figs 3.55 to 3.57 for each shock duration.

Fig 3.55 Slider displacement with shock input 1050 G's 0.5 ms

Fig 3.56 Slider displacement with shock input 540 G's 1 ms

Fig 3.57 Slider displacement with shock input 310 G's 2 ms

To verify the shock tolerance prediction of MATLAB® – simple model in Fig 3.52, we set up a shock analysis in ANSYS using transient analysis. The HAA FE model in Fig 3.53 is added into the cantilever beam plus four mounts FE model in Fig 3.41. The FE model of HAA plus cantilever beam in ANSYS is given in Fig 3.58. Fig 3.58 shows the FE model of HAA + HDD + cantilever beam in two kinds of configurations, and also its boundary conditions. The difference between two configurations is the HDD placement orientation, as we can see in Fig 3.58. The end tip of the cantilever beam is constrained so it can only move in the Z direction. The end tip of the cantilever beam is also coupled with a shaker node. For the HAA boundary condition, the edge of the HAA where the bearing is placed is coupled in all directions with one node from the HDD model. The cantilever beam and the four mounts properties are the same as the last configurations in section 3.10. For the FE model of Fig 3.58, the shock input given to the shaker node is 1000 G's 0.5 ms and 310 G's 2 ms.

The HAA placement on the HDD model is explained in Fig 3.59. The HAA is placed according to the ST1 HAA placement. The HDD FE model thickness is drawn as 4 mm. Thus, the distance from the HDD base to the top most HAA is 5 mm.

Fig 3.58 HAA + HDD + cantilever beam FE model and its boundary conditions in ANSYS

We also set up another shock analysis of the HAA only in Fig 3.53, with the shock input given to the shaker node as 175 G's 2 ms (the current shock tolerance of the ST1). The boundary conditions for the FE model of HAA only are given in Fig 3.60. The edge nodes of the FE model of Fig 3.60 are constrained so it can only move in Z direction, and also coupled with the shaker node.

Fig 3.59 HAA placement on the HDD model

Fig 3.60 HAA boundary conditions in ANSYS

The slider displacements obtained from the above two shock analyses in ANSYS are compared; analogous with the method to find the shock tolerance improvement using simple model in MATLAB®. The slider nodes observed for its displacement is given in Fig 3.61. Nodes 1 and 3 will have the same responses; and the same for nodes 2 and 4, since the model is symmetric. Thus we only observe nodes 1 and 2 for its displacement.

Fig 3.61 Slider nodes to be observed

The slider displacement from the above shock analyses in ANSYS is given in Figs 3.62 and 3.63. From Figs 3.62 and 3.63, it is clear that the HDD placement of Cantilever I is better. The cantilever beam plus four mounts is able to protect the HDD up to 1050 G's 0.5 ms and 310 G's 2 ms. This is shown by the slider displacement not exceeding the slider displacement from the HAA only with shock input 175 G's 2 ms.

The G levels used in the ANSYS simulation (1050 G 0.5 ms and 310 G 2 ms) are derived from the MATLAB® simulation results of Figs 3.55 and 3.57. Thus, we conclude that the simple model of Fig 3.52 can be used to predict the shock tolerance improvement.

Fig 3.62 Node 1 displacement responses

Fig 3.63 Node 2 displacement responses

3.12 Short Conclusion

In this chapter, we have designed an external shock isolator for ST1. The cantilever beam plus four mounts as the shock isolator has been proven to protect the ST1 up to more than 1050 G's 0.5 ms, and increased the shock tolerance of ST1 by more than 77% (from 175 G's 2 ms to more than 310 G's 2 ms). The cantilever beam plus four mounts also satisfies the design envelopes.

Chapter IV

USE OF ELASTOMER LAYERS UNDER THE BASE OF THE ST1 AS A SHOCK ISOLATOR

4.1 Introduction

As already mentioned in Chapter II, elastomer layers can be applied as shock isolators for a HDD. This chapter will investigate elastomer performance as a shock isolator for the ST1. In this chapter, we will investigate if the elastomer layers should or should not be placed under the base plate of the ST1. The advantage is to minimize the total height of the ST1 plus elastomer, but the design envelope to put the elastomer is limited.

The parameters which will be investigated in this chapter are the location of the elastomer layers, the thickness of the elastomer layers, and mechanical properties of the elastomer (particularly its stiffness and damping ratio). The shock analysis is done in ANSYS using transient analysis.

4.2 Modeling the Problem

The HDD used in this chapter is the same as the used in Chapter III, ST1 8 GB. Only the base plate of the ST1 will be modeled using ANSYS. The reason for only FE modeling of the base plate and not the entire HDD is to keep the FE model small to reduce the simulation time. Since only the base plate is FE modeled, the base plate density needs to be increased to match the overall HDD mass. The ST1 base plate can be seen in Fig 4.1.

The elastomer layers will be placed at the bottom of the base plate of the ST1. The advantage of this placement is that the size of the overall shock isolator can be kept small. This advantage can be explained by looking at Fig 4.2. In Fig 4.2, the **original ST1**, the ST1 base plate sits on the ST1 body. In Fig 4.2, the elastomer placed under the base plate of ST1 is depicted as configuration A. So in configuration A, the elastomer layers are placed in between the ST1 base plate and ST1 body. Therefore, to put the elastomer under the base plate of the ST1, we need to change the PCB board design, as well as the ST1 body. The configuration B, where the elastomers are placed outside of the ST1, will be investigated in Chapter V.

We will first examine the possible locations where we can put the elastomer layers under the base plate. There are mainly four locations to put the elastomer layers. These locations are selected based on practical considerations. And these locations are explained in Fig 4.3.

Fig 4.1 ST1 base plate view from top with the ST1 Figure as comparison

Now that the locations of the elastomers under the base plate are known, shock analysis is preferred as seen in Fig 4.4. Later in ANSYS simulation, the support is modeled as a point (shaker) that has larger mass than ST1 mass ($\gg$ than ST1 mass). The half-sine forcing function magnitude and duration and base mass chosen such to create a clean 1000 G's with 0.5 ms duration shock input to the elastomer layers. The bottom most nodes of the elastomer layers are coupled to the shaker in z direction. Then the coupled nodes are constrained so it can only move in the z direction. Due to the boundary condition at the bottom most nodes and the shaker, and the direction of the shock input to the elastomer layers; the problem can be idealized into S-DOF problem. This idealization

provides convenience in predicting the optimum vertical natural frequency (in the Z direction) of elastomer layers – ST1 system for shock isolation purpose.

Fig 4.2 Comparison of elastomer layers' placement between under the base plate (A) and outside of the ST1 outer case envelope (B; B is actually the original ST1 plus elastomer). Example if using 5 mm thick of elastomer layers/ elastomeric mounts

Fig 4.3 Locations to put the elastomer layers

Fig 4.4 Modeling of shock analysis problem and idealization into S-DOF problem

4.3 Possible Elastomer Layers' Shape

Referring to Fig 4.3, there are four locations where the elastomer layers could be placed. For practicality, the elastomer layers will be placed as shown in Fig 4.5. The small plate width at location I, III, and IV is less than 1.2 mm (see Fig 4.5). To give the elastomer space to deform at horizontal direction (perpendicular to z direction) and to maximize the

elastomer width, we set the elastomer width at location I, III, and IV as 1 mm, and at location II as 1.8 mm.

The stiffness of the elastomer layers should be balanced to the ST1 CG (center of gravity). This could be done by balancing the amount of elastomer layers' area at each location to the ST1 CG. We assume that the ST1 CG location is at the center of the base plate volume. This assumed CG location (the CG location is calculated by ANSYS) can be seen in Fig 4.6. Fig 4.6 also gives the distance of the CG to the four locations of elastomer layers' location. By using area balance criteria, the ratio of each area can be obtained as follows:

$$I : II : III : IV = 8.6 : 44 : 10 : 7.8$$

Fig 4.5 Elastomer layers' shape of the four mounts

Fig 4.6 ST1 CG location and its distance to elastomer

4.4 Design Criteria

The shock isolation system for the ST1 aims to mitigate the shock input given by the support, so that the ST1 will not experience shock magnitudes higher than the shock input magnitude. However, since there are design envelopes for the shock isolators, the shock isolation system cannot be too thick.

Referring to Chapter 3, section 3.1, most of small form factor HDDs can sustain operational shock up to 300 G's with 0.5 ms duration. And according to section 4.2, the elastomer layers – ST1 system can be idealized into S-DOF problem. Thus, to reduce the shock input from 1000 G's to less than 300 G's; according to classical shock analysis of S-DOF problems (using MATLAB® which we have performed in Chapter 3, section 3.3), we need to set the elastomer layers – ST1 system's vertical natural frequency below 200 Hz (without damping effect). To be exact, according to the MATLAB® simulation, the system's (S-DOF system) natural frequency needed to reduce the shock input below 300 G's, is about 150 Hz. The MATLAB® simulation results on the S-DOF system shows that if the elastomer layers – ST1 system is considered linear, without any damping effect, and having vertical natural frequency of 150 Hz, the elastomer layers will deform about 3.295 mm (if subjected to a shock input of 1000 G's 0.5 ms); which means referring to [32], we will need an 8.25 mm thick of elastomer layers. From practicality

point of view, we think that an 8.25 mm thick elastomer layer is already unpractical. We suggest that for elastomer layers application under the base plate of ST1, the appropriate elastomer thickness is about 4 to 5 mm. The reason is this thickness will not affect the ST1 size too much, as well as not induce more problems such as column buckling. The elastomer layers may experience buckle if the ratio of the height (thickness) and the amount of area is too high. This suggested thickness implies that the resulting vertical natural frequency for elastomer layers – ST1 will be higher than 150 Hz.

We conclude that for elastomer layers' application under the base plate of ST1, we need to consider the elastomer layer's size before we think about the transmissibility.

4.5 Elastomer Dimension and Mechanical Properties

This section will explain how we can calculate the mechanical properties of the elastomer based on its shape and dimension.

The elastomer's basic shape at location I, III, and IV is explained in Fig 4.7. With this basic shape, we can calculate the total cross sectional area of elastomer at location I, III, and IV by formula:

$$\begin{aligned} A &= (L-1) \cdot 1 \cdot 2 + 1 \cdot 1 \\ &= (2L-1) \text{ mm}^2 \end{aligned} \quad (4.1)$$

Fig 4.7 Basic shape of elastomer at location I, III, and IV

Then the elastomer's basic shape in location II is given in Fig 4.8.

Fig 4.8 Basic shape of elastomer at location II

With this basic shape, we can calculate the total cross sectional area of elastomer at location II by formula:

$$A = \pi (R_2^2 - R_1^2) \frac{\alpha}{2\pi}$$

$$= \frac{\alpha}{2} (R_2^2 - R_1^2)$$
(4.2)

For our first simulation, we will set the elastomer thickness at 4 mm. Referring to Fig 4.2, the maximum allowable deformation is 2.2 mm if we use a 5 mm thick of elastomer (to avoid collision between the support/ body and the base plate). Thus, for 4 mm thick of elastomer, the maximum allowable deformation is 1.2 mm, whereas the maximum linear deformation is 1.6 mm.

Fig 4.9 Acceleration response of S-DOF base excitation problem at various damping ratios

Following the shock isolation theory [35], the damping ratio which gives the lowest transmissibility for S-DOF base excitation problem is around 0.25 (25%). We also prove this statement by performing shock analysis for S-DOF base excitation problem using MATLAB® simulation, the same simulation as the one used in Chapter 3, section 3.3. The MATLAB® simulation result is given in Fig 4.9. Therefore we will set the elastomer damping ratio in the transient analysis in ANSYS as 0.25.

Once again, the MATLAB® simulation of S-DOF system (as have been used in Chapter 3, section 3.3) is used. This time, the simulation is used to obtain the appropriate vertical natural frequency of the elastomer layers – ST1 system; if we set the damping ratio as 0.25 and only allow a 1.2 mm linear deformation. The simulation result in MATLAB® tells us that we need to set the 1st elastic elastomer layers' natural frequency to 290 Hz to satisfy the above condition.

For transient analysis in ANSYS, we will use a natural rubber (NR) as elastomer material. NR typically has shear modulus of 1 MPa and bulk modulus of 1000 MPa. Ideally, the relationship between the shear modulus and equivalent Young's modulus of elastomer is given by [28]:

$$E_{eq} = 3G(1 + \beta S^2) \quad (4.3)$$

Where β dependent on rubber hardness (H):

$$\begin{aligned} H : 30 - 55; \beta &= 2.68 - 0.025H \\ H : 60 - 75; \beta &= 1.49 - 0.006H \end{aligned} \quad (4.4)$$

and S is the shape factor:

$$S = \frac{ab}{h(a+b)} \quad (4.5)$$

h is the elastomer height (thickness), a and b are the width and length of the cross section. For the 1st elastic natural frequency of elastomer layers system, the equivalent stiffness can be calculated by formula:

$$k_{eq} = E_{eq} \frac{A}{L} \quad (4.6)$$

If we combine equations (4.3) and (4.4), and including equations (4.5) and (4.6), we will get:

$$\frac{kh}{A} = 3G(1 + \beta S^2) \quad (4.7)$$

L in equation (4.6) is substituted by h from equation (4.5). A from equation (4.6) is also substituted by ab from equation (4.5), considering rectangular cross section. Equation (4.7) can also be written as:

$$\frac{k_{eq}h}{ab} = 3G \left(1 + (2.68 - 0.025H) \left(\frac{ab}{h(a+b)} \right)^2 \right) \quad (4.8)$$

So with the required properties of elastomer system as:

- 4 mm thickness
- 290 Hz 1st elastic natural frequency

And using NR with 1 MPa of shear modulus; we also assume that NR in shore A has hardness (H) as 30. So using equation (4.8), we can estimate the elastomer dimension (referring to equations (4.1) and (4.20)) as:

- At location I, 7.4 mm²; $L = 4.2$ mm
- At location II, 37.8 mm²; $\alpha = 0.42\pi$
- At location III, 8.6 mm²; $L = 4.8$ mm
- At location IV, 6.7 mm²; $L = 3.85$ mm

These dimension and mechanical properties of elastomer layers will be used in our first simulation in transient analysis in ANSYS. The simulation will be discussed in the next section.

4.6 Simulation

The ANSYS FE model of the base plate of ST1 plus the elastomer layers is given in Fig 4.10. The base plate material has 190 GPa of Young's modulus. The base plate FE model uses a 10 noded *solid 92* elements; whereas the elastomer layers' FE model uses a 10 noded *solid 182* elements. The elastomer layers are modeled using Neo – Hookean material model. To capture nonlinearity of the elastomer layers, the elastomer layers mesh density is kept high.

As explained earlier in section 4.2, the most bottom nodes of the elastomer layers are coupled with a point (shaker) in z direction. And the nodes plus the shaker is constrained

so it can only move in z direction. We can see the boundary conditions (BCs) of the FE model in Fig 4.11.

Fig 4.10 Base plate and elastomer layers FE model in ANSYS (meshed)

Fig 4.11 Base plate and elastomer layers FE model BCs

The simulation time is set from 0 to 2.5 ms only, due to very long simulation time. 2.5 ms simulation time is just enough to obtain the peak acceleration response of the system.

Basically, a half-sine force input with a magnitude of 10kN and 0.5 ms duration is given to the shaker at z direction (the shaker mass is 1 kg). With very little mass of the ST1 (18 g), the shock input excited to the elastomer layers will be 1000 G's and 0.5 ms duration.

The force input is defined in 12 LS (Load Step) with ramped step. The half – sine force curve is shown below:

$$\begin{aligned} F(t) &= 10kN \cdot \sin(2\pi \cdot 1000 \cdot t) \rightarrow 0 \leq t \leq 0.5ms \\ F(t) &= 0 \rightarrow t > 5ms \end{aligned} \quad (4.9)$$

Where 1000 is the input frequency (from 2×0.5 ms), t is the time, and 10 kN is the force magnitude. The half – sine curve is then divided into 10 steps, with time interval of 0.05 ms. The list of the load steps can be seen in table 4.1. And the resulting acceleration excited by the shaker to the elastomer layers can be seen in Fig 4.12.

Fig 4.12 Shock input excited by the shaker (taken from ANSYS)

Table 4.1 List of load step given to the shaker

No of LS	Start time (s)	End time (s)	Force to shaker (kN)	Time increment (s)
1	0	5E-5	0	1E-5
2	5E-5	10E-5	3.09	1E-5
3	10E-5	15E-5	5.877	1E-5
4	15E-5	20E-5	8.090	1E-5
5	20E-5	25E-5	9.510	1E-5
6	25E-5	30E-5	10	1E-5
7	30E-5	35E-5	9.510	1E-5
8	35E-5	40E-5	8.090	1E-5
9	40E-5	45E-5	5.877	1E-5
10	45E-5	50E-5	3.09	1E-5
11	50E-5	55E-5	0	1E-5

12

55E-5

250E-5

0

1E-5

Then, the simulation results from transient analysis in ANSYS are given in Figs 4.13 to 4.15.

Fig 4.13 Acceleration responses of ST1 base plate at four elastomer bonded areas

Fig 4.14 Acceleration responses of ST1 base plate at HAA and Spindle base

Fig 4.15 Displacement response of ST1 base plate at four elastomer bonded areas

The ANSYS simulation results of Figs 4.13 to 4.15 mention two things:

- The shock input can only be reduced by 2.3 times; the maximum acceleration response at spindle base is 438 G's.
- The peak displacements at the four locations show that the support collides with the base plate; the peak displacement at location IV is 2.27 mm, while the allowed deformation is only 1.2 mm.

To avoid collision between the ground/ body and the base plate and also to reduce the base plate acceleration responses, the simulation was then continued with increasing the elastomer layers thickness to 5 mm. We will use the same elastomer dimension and mechanical properties as before. With 5 mm thickness, the natural frequency of the system is estimated as 250 Hz (by using equation (4.8)). The simulation results are given in Figs 4.16 and 4.17.

Fig 4.16 Acceleration responses of HAA and Spindle using 5 mm thick elastomer

Fig 4.17 Displacement responses of four locations using 5 mm thick elastomer

The ANSYS simulation results of Figs 4.16 and 4.17 show that the acceleration response at spindle and HAA base is lower than if we use 4 mm thick elastomer layers. But the peak displacement at location IV is still too high that the support and the base plate are still collides; the maximum allowable deformation is 2.2 mm while the peak displacement at location IV is 2.7 mm. This is due to the ST1 base plate is not rigid, so it is also deformed when the shock input is applied.

The collision between the support and the base plate can be diminished by adding more elastomer layers' thickness. In addition, adding more elastomer layers' thickness can reduce the transmissibility. But as explained before that adding too much thickness of elastomer layers to the base plate is not a practical solution, because it will affect the size of the HDD itself. So we did not continue the simulation.

4.7 Short Conclusion

Elastomer layers investigated in this Chapter failed to improve the ST1 shock tolerance and to protect the ST1 from a shock disturbance; the acceleration response at the HAA base and the spindle base are more than 300 G's. The reason is that the ST1 plus the shock isolator natural frequency is still too high. In addition, the ST1 base plate still collides with the ground/ body (even with 5 mm thick of elastomer layers). The HDD manufacturer also needs to change the PCB board and ST1 body design; which are not practical. We do not recommend this type of shock isolator configuration.

Chapter V

USE OF ELASTOMERIC MOUNTS OUTSIDE THE CASE OF THE ST1 AS A SHOCK ISOLATOR

5.1 Introduction

This chapter will investigate the use of elastomeric mounts placed outside of the ST1. With this kind of placement, the space to put the elastomeric mounts and the available space for sway movement are larger than the configuration discussed in Chapter IV (see Fig 4.2). We would like to find out if we can achieve more shock reduction with configuration B of Fig 4.2.

5.2 Modeling the Problem

Fig 5.1 shows the ST1 plus elastomeric mounts. The ST1 is modeled as a rigid block (red color), and the elastomeric mounts are placed at the ST1's corners (green color).

Fig 5.1 ST1 with elastomeric mounts as shock isolators

Since the shock is in the Z direction, the model in Fig 5.1 can be modeled as the model of Fig 4.4. So then the problem can be simplified to a single DOF base excitation problem. With the simple model, we can determine the proper vertical natural frequency of the system by performing a very fast simulation. Then the final results are then verified by performing a 3D FEA using ANSYS.

5.3 Design Constraints, Determining Natural Frequency of the System, and Elastomeric mounts' Properties

Following the configuration in Fig 2.4, we can draw the size constraints for elastomeric mounts placed outside of the ST1.

Fig 5.2 Elastomeric mounts size constraints

In Fig 5.2, the available height for elastomeric mounts height is about 10.5 mm. We estimate that the maximum compression deformation of the elastomeric mounts must not exceed 40% of its height, so that the stiffness of the mounts when it is compressed does not change too much. So then the space available for tension deformation is also set as 40% of 10.5 mm, which is 4.2 mm.

Using the simple model of single DOF base excitation problem, we can estimate the maximum compression deformation of the elastomeric mounts and also obtain the estimated system's natural frequency and the estimated transmissibility. We use the shock input duration of 0.5 ms. The estimated transmissibility, assuming 25% of damping ratio is given in table 5.1.

Table 5.1 Estimated transmissibility with various elastomeric mounts heights

Elastomer height (mm)	Maximum deformation (mm)	Estimated 1 st natural frequency (Hz)	Estimated transmissibility
4	1.6	220	0.353
5	2	175	0.282
6	2.4	150	0.241
7	2.8	130	0.21
8	3.2	111	0.18
9	3.6	100	0.162

10	4	89	0.144
10.5	4.2	84	0.136

The lowest transmissibility is about 0.136, and this is obtained by using 10.5 mm of elastomer height. According to [22], when the shock magnitude is below 100 G's for all shock duration, the head slap does not occur. Which means the maximum shock input that can be handled by the ST1 is now become 735 G's (100 G's/ 0.136) for any shock duration.

Using equation (4.8), we can calculate the elastomeric mount dimension needed to obtain the required system's natural frequency. The shape and dimension of the elastomeric mount are given in Fig 5.3. The cross section is square with length a , and the elastomeric mounts' height is denoted as h .

Fig 5.3 Elastomeric mounts dimension

Equation (4.8) is modified since the value b is equal to a :

$$\frac{k_{eq}h}{a^2} = 3G \left(1 + (2.68 - 0.025H) \left(\frac{a}{2h} \right)^2 \right) \quad (5.1)$$

The ST1 mass is 18 grams, so for elastomeric mounts at each corner the stiffness needed to obtain an 84 Hz vertical natural frequency is:

$$2\pi(84) = \sqrt{\frac{k}{4.5E-3}}$$

$$278560 = \frac{k}{4.5E-3}$$

$$k = 1253.52N / m$$

The elastomer used is NR with G equal to 0.6 MPa and hardness H of 30. The height of the elastomeric mounts' is 10.5 mm, so we can calculate a :

$$\frac{1253.52 \cdot 10.5E-3}{a^2} = 3(0.6E6) \left(1 + (2.68 - 0.025(30)) \left(\frac{a}{2(10E-3)} \right)^2 \right)$$

$$a = 2.7mm$$

With a of 2.7 mm and h of 10.5 mm, the slenderness ratio (SR) can be calculated by the following formula [36]:

$$SR = \frac{h}{\sqrt{I/A}} \quad (5.2)$$

With: I as second moment of inertia, A as cross section area, h as height. The SR of the above configuration is:

$$SR = \frac{h}{\sqrt{\left(\frac{1}{12}a^4\right)/a^2}}$$

$$SR = \frac{h}{\sqrt{\frac{1}{12}a^2}} = 3.47 \frac{h}{a} = 13.5$$

According to Shock and Vibration Handbook, for elastomer which has SR more than 1.6, we need to consider buckling.

When the shock is applied to the elastomeric mounts plus ST1 system, the elastomeric mounts will experience a compression load as:

$$P = 9.81 \cdot \frac{1}{4} m_{ST1} \cdot G \quad (5.3)$$

With: P as the compression load (in N), G as shock level in times of gravity acceleration, m as mass of the ST1.

While the critical load, using Euler formulation for the elastomeric mounts clamped at both side is:

$$P_{crit} = \frac{\pi^2 EI}{0.25h^2} \quad (5.4)$$

Then the critical load for h of 10.5 mm and a of 2.7 mm is:

$$\begin{aligned} P_{crit} &= \frac{\pi^2 EI}{0.25h^2} \\ &= \frac{\pi^2}{0.25h^2} \cdot \frac{kh}{a^2} \cdot \frac{a^4}{12} \\ &= \frac{\pi^2 ka^2}{3h} \\ &= \frac{\pi^2 \cdot 1253.52 \cdot (2.7E-3)^2}{3 \cdot 10.5E-3} = 2.863N \end{aligned}$$

So then the maximum G level which can be applied to the elastomeric mounts can be obtained by using equation (5.3):

$$\begin{aligned} P &= 9.81 \cdot \frac{1}{4} m_{ST1} \cdot G \\ G &= \frac{4 \cdot 2.863}{9.81 \cdot 18E-3} = 65 \end{aligned}$$

So then, to make the elastomeric mounts to be able to withstand 1000 G's of shock input, we need to increase the cross sectional area. Therefore we make a table consisting of elastomeric mounts dimension, in which we consider buckling and set the safety factor as 1.5 for 1000 G's shock input. The shock duration used is 0.5 ms. The elastomeric mount used is NR with G of 0.6 MPa, H as 30, and damping ratio of 25%. This data is given in Table 5.2. The data in Table 5.2 is made by performing simulation using the simple model (single DOF base excitation problem).

Table 5.2 Estimated transmissibility considering buckling (with safety factor of 1.5)

Elastomer height (mm)	Minimum a (mm)	Est. lowest nat. frequency (Hz)	Estimated transmissibility
4	3.69	220	0.353
5	3.841	195.98	0.3148
6	4.244	194.31	0.3122
7	4.614	193.06	0.3102

8	4.958	192.10	0.3087
9	5.28	191.30	0.3074
10	5.585	190.64	0.3064
10.5	5.73	190.35	0.3059

From Table 5.2, we can see that when we consider buckling, the transmissibility difference of 5 mm elastomer height and 10.5 mm elastomer height is very small (less than 13 G's).

Another problem for a slender elastomeric mounts is the lateral mode (shear load). Generally the elastomer stiffness at the lateral direction is lower than the compression direction. This is explained in Fig 5.4.

Fig 5.4 Elastomeric mounts load direction

The longer the elastomeric mounts' height, the lateral mode will be easier to excite. Thus, to minimize the lateral movement, prevent buckling, and also for practical reasons, we choose elastomeric mounts' height as 5 mm.

5.4 Transient Analysis in ANSYS

From the simple model, we obtain the transmissibility for elastomer height of 5 mm as 0.3148. This means, for 1000 G's 0.5 ms shock input, the HDD will experience 314.8 G's 2.5 ms shock. We will perform a transient analysis in ANSYS to verify the simple model simulation result.

The FE model of elastomeric mounts plus ST1 in ANSYS and its boundary conditions (BCs) is given in Fig 5.4. The bottom nodes of the elastomeric mounts are constrained so it can only move in the Z direction. The bottom nodes of the elastomeric mounts are also coupled with the shaker node in the Z direction.

The element used in ANSYS for elastomeric mounts is 20 noded *solid* 186 elements, and the element used for ST1 body is 20 noded *solid* 95 elements, since the general shape of the model is brick. The elastomeric mounts element size is 0.48 mm x 0.48 mm x 0.45 mm, so the aspect ratio is kept approximately 1. The elastomeric mounts element is using Hyperelastic – Neo Hookean table, to model the nonlinearities of the elastomer. The ST1 elements act as a rigid body, since the stiffness of ST1 body is far higher than the elastomeric mounts. The ST1 elements are using material as Aluminum.

Modal analysis is done in ANSYS to obtain the mode shapes of the FE model of Fig 5.5, and the first seven elastic mode shapes are given in Table 5.3. All of six rigid body natural frequencies of the FE model of elastomeric mounts plus ST1 are away from 1000 Hz.

Fig 5.5 FE model of elastomeric mounts plus ST1 and its BCs in ANSYS

Table 5.3 Mode shapes of FE model of Fig 5.4

Mode	Mode shapes
1 st (90.17 Hz)	<p>1 DISPLACEMENT STEP=1 SUB =1 FREQ=90.17 DMX =238.227</p>
2 nd (90.546 Hz)	<p>2 DISPLACEMENT STEP=1 SUB =2 FREQ=90.546 DMX =237.078</p>
3 rd (142.447 Hz)	<p>3 DISPLACEMENT STEP=1 SUB =3 FREQ=142.447 DMX =405.601</p>
4 th (200.438 Hz)	<p>4 DISPLACEMENT STEP=1 SUB =4 FREQ=200.438 DMX =235.167</p>

The simulation time in ANSYS is set only up to 2.5 ms to save simulation time. The simulation time is sufficient to obtain the maximum HDD body acceleration response.

The load is given by using the same method in Chapter IV, section 4.6. The shock magnitude is 1000 G's with shock duration of 0.5 ms. The shock is given in the Z direction. The simulation result is given in Fig 5.6. Fig 5.6 gives the acceleration response of the entire HDD body (since the HDD is assumed rigid) from ANSYS and MATLAB simulation. The MATLAB simple model cannot model the nonlinearities of the elastomer, but it can predict the maximum acceleration response of the HDD body.

Fig 5.6 Simulation result from ANSYS and MATLAB (simple model)

The peak acceleration from ANSYS is 318 G's, while from simple model in MATLAB is 316 G's. The shock reduction using configuration B of Fig 4.2 is then larger than the shock reduction using configuration A of Fig 4.2.

5.5 Shock Isolator Performance Verification and Practical Issues

Following Chapter III, the shock isolator in this chapter also needs to be verified. The model and the method used is the same as used in Chapter IV, but the shock isolator plus the ST1 natural frequency is tuned to 196 Hz using damping ratio of 25%. The simulation is performed in MATLAB. The simple model properties are given in Table 5.4.

Table 5.4 Simple model properties

	Mass (grams)	Stiffness – K (N/m)	Damping – C (Ns ² /m)
Moving base	5E6		
HAA rigid block + HDD outer case	18	27299	11.0835
Arm	7.157E-2	45208	0
Suspension + slider	1.102E-2	38.3755	0

From the simulation results of Figs 5.7 to 5.9, the shock isolator can protect the ST1 up to 440 G's 0.5 ms, 230 G's 1 ms, and only 140 G's 2 ms.

Fig 5.7 Slider displacement with shock input 440 G's 0.5 ms

Fig 5.8 Slider displacement with shock input 230 G's 1 ms

Fig 5.9 Slider displacement with shock input 140 G's 2 ms

Due to column buckling issue, the square shape mounts used in this chapter cannot improve the ST1 shock tolerance (particularly for 2 ms shock duration). There are Companies such as Lord Corporation, Barry Controls, or Paulstra who can provide such rubber mounts with stiffness lower than 1200 N/m, and it will not buckle. An example of product from Lord Corporation is shown in Fig 5.10. This product can provide stiffness as low as 500 N/m.

Fig 5.10 Multiplane mounts from Lord Corporation

Refer to Table 5.1 and section 5.3, with elastomeric mount having stiffness of 1254 N/m, we can achieve transmissibility of 0.136 (using 0.5 ms of shock duration). So with stiffness of 500 N/m, we can achieve transmissibility below 0.136.

If for example we use this stiffness (500 N/m) as the elastomeric mount stiffness, the transmissibility then becomes 0.0875 if we use shock duration of 0.5 ms. And if we redo the shock isolator performance verification in this section using this stiffness, the shock tolerance of ST1 becomes 680 G's 2 ms, 1300 G's 1 ms, and 2600 G's 0.5 ms. The ST1 shock tolerance is increased significantly by almost 400% (from 175 G's 2 ms to 680 G's 2 ms).

The problem is that the size of the rubber mounts available in the market is too large. The Multiplane mount of Fig 5.10 has a total height of more than 22 mm. So then it will violate the design envelope of the shock isolator (see Fig 5.2). The catalogue of the Multiplane mounts from Lord Corporation is given in Appendix C.

But if in the future, companies such as Lord Corporation able to make rubber mounts which satisfy the design envelope of the shock isolator with maximum height of 10.5 mm and stiffness as low as 500 N/m, the shock isolator will be able to increase the shock tolerance of ST1 significantly.

5.6 Short Conclusion

With 500 N/m of mount stiffness, the shock isolator can increase the shock tolerance of the ST1 by 400%. But the shock isolator must be designed carefully to make sure all six rigid body natural frequencies are away from the shock input frequency (see Table 5.4 as an example, all six rigid body natural frequencies are away from 1000 Hz).

Chapter VI

USE OF TUNED VIBRATION ABSORBER (TVA) ON THE HEAD ACTUATOR ASSEMBLY (HAA) OF THE ST1 (1" disk) AS A SHOCK ISOLATOR

6.1 Introduction

Many HDD companies prefer their shock isolator to be hidden so that their competitors cannot copy their ideas. Thus, we consider using a TVA as the shock isolator for the ST1, since TVA can be placed inside the HDD.

6.2 Previous Study of TVA for Shock Mitigation

Dynamic absorber or TVA application to mitigate operational shock on HDD has been introduced by [13]. The dynamic absorber is installed on the HSA (Head Stack Assembly), or so called as HAA (Head Actuator Assembly) of a particular type of HDD (the author did not mention the type of HDD). The purpose is to disperse the shock energy from the HAA to the dynamic absorber, so that the head/ slider will not experience severe shock impulse that may induce head slap. The dynamic absorber placement done by [13] is illustrated in Fig 6.1. In Fig 6.1, the green colored elements are the arm, whereas the purple colored elements are the suspension. The head and the slider are attached to the suspension.

Fig 6.1 Illustration of dynamic absorber on HAA

The dynamic absorber is installed on the relatively thicker part of the HAA (the arm) and in a form of a cantilever beam. Equations below are specified by [13] to find the optimum dynamic absorber properties (natural frequency, mass ratio, and damping ratio):

Frequency parameter (α), defined as:

$$\alpha = \frac{\omega_a}{\omega_o} = \frac{\omega_{absorber}}{\omega_{arm}} \quad (6.1)$$

With numerical analysis, the optimum frequency parameter and the optimum damping ratio of the absorber are defined as below:

$$\begin{aligned} \alpha_{opt} &= \frac{1}{1 + \mu} \\ \zeta_{opt} &= \sqrt{\frac{3\mu}{8(1 + \mu)^3}} \\ \mu &= \frac{m_a}{m} = \frac{m_{absorber}}{m_{arm}} \end{aligned} \quad (6.2)$$

From many references on vibration handbook, we know that the optimum dynamic absorber or TVA mass is usually $1/20^{\text{th}}$ to $1/10^{\text{th}}$ of the primary systems' mass. If for example we choose the absorber mass as $1/10^{\text{th}}$ of the arm's mass, then referring to equation (6.2) the value of μ will be 0.1. Thus, referring to equation (6.1) the value of α or the frequency ratio will be 0.909. The value of μ cannot be negative ($\mu > 0$). Therefore, it implies that the absorber frequency will always be lower than the arm's frequency, and its value depend on the mass ratio (μ), and vice versa.

The dynamic absorber introduced by [13] has been shown to increase the shock tolerance of a particular HDD by 5% - 6%, for shock duration of 0.5 ms (experimental result), which is not significant (see Fig 7 on [13]). Different from the experimental result of Fig 7, from the simulation result of Fig 5 in [13]; the shock tolerance of the HDD is increased by 15% (for shock duration 0.5 ms), which is quite significant. The Author on [13] did not mention the value of mass ratio (μ) they used in their configuration. But from Fig. 2 on [13], we can see that the size of their dynamic absorber is relatively large compared to the suspension's size. If the μ is too large, it will cause a problem to the actuator, for example reducing writing/ erasing speed and seeking time of the HDD.

There is another study to use dynamic absorbers for step motions and step disturbances [37]. The system studied by [37] is a spring – mass – damper system, with base excitation. The mass ratio (μ) used by [37] is 0.1, with frequency ratio (α) as 0.89 to 0.9. It is shown by [37] that the dynamic absorbers with the above configuration are able to reduce the peak overshoot by 9% and improve the settling time by 87% to 90%. The peak overshoot reduction is actually related to the dynamic absorber performance to increase the shock tolerance discussed by [13].

6.3 Determining The TVA Frequency; Theoretical and Simulation

In this Chapter we are interested in investigating if we can achieve higher shock tolerance of more than achieved by [13]. We will use the same technique as [13], meaning we install the TVA on the HAA of ST1, but we will place it at different locations. While [13] installed the TVA on the arm, we will install the TVA on the suspension and the slider of the ST1. This is due to the first arm bending natural frequency of the ST1 to be very high. So placing the TVA on the ST1 arm will not be effective.

Fig 6.2 Finite element model of HAA - ST1

The FE model of the ST1's HAA is shown in Fig 6.2. Number 1, the red colored elements (the arm) uses solid 92 elements. It has 190 GPa of Young's modulus and density of 7580 kg/m³. Number 2 (part of the suspension), the green colored elements uses shell 93 elements. It has 190 GPa of Young's modulus and density of 8079.3 kg/m³. Number 3 (the suspension), the purple colored elements uses shell 93 elements. It has 190 GPa of Young's modulus and density of 8079.3 kg/m³. Number 4 (structure holding the slider), the black colored elements uses solid 92 elements. It has 190 GPa of Young's modulus

and density of 8079.3 kg/m^3 . Number 5, which is the slider, uses solid 92 elements. It has 399 GPa of Young's modulus and density of 4232 kg/m^3 . This FE model of HAA and its material properties are obtained from Seagate.

Before we put the TVA on the suspension of ST1 and perform shock simulation in ANSYS, we will first perform a simple theoretical analysis of TVA for shock mitigation purpose. We would like to find out an analytical solution of the TVA properties such as TVA frequency ratio (α) and mass ratio (μ) and see if we can achieve reduction more than 10%.

First, we consider a two DOF system of Fig 6.3. The primary mass is denoted as m_a , and the absorber is denoted as m_t . To investigate the optimum absorber properties, we purposely neglect damping.

The mass matrix and the stiffness matrix of the system of Fig. 6.3 are given below:

$$M = \begin{bmatrix} m_a & 0 \\ 0 & m_t \end{bmatrix} \quad K = \begin{bmatrix} k_a + k_t & -k_t \\ -k_t & k_t \end{bmatrix}$$

Where the governing equation in matrix form is:

$$\begin{bmatrix} m_a & 0 \\ 0 & m_t \end{bmatrix} \begin{bmatrix} \ddot{x}_1 \\ \ddot{x}_2 \end{bmatrix} + \begin{bmatrix} k_a + k_t & -k_t \\ -k_t & k_t \end{bmatrix} \begin{bmatrix} x_1 \\ x_2 \end{bmatrix} = \begin{bmatrix} F(t) \\ 0 \end{bmatrix} \quad (6.3)$$

If $F(t)$ is a sinusoidal function:

$F(t) = F_0 \sin(\omega t)$; where ω is the excitation frequency, and F_0 is the force amplitude.

Fig. 6.3 Two DOF system; with TVA (m_t)

Then the steady state solution for x_1 and x_2 will tend to be sinusoidal [29]:

$$x(t)_{steady_state} = \begin{bmatrix} x_{1(t)} \\ x_{2(t)} \end{bmatrix} = \begin{bmatrix} X_1 \sin(\omega t) \\ X_2 \sin(\omega t) \end{bmatrix} \quad (6.4)$$

The amplitude of x_1 and x_2 are written as:

$$X_1 = \frac{k_t - m_t \omega^2}{(k_a + k_t - \omega^2 m_a)(k_t - \omega^2 m_t) - k_t^2} F_0 \quad (6.5)$$

$$X_2 = \frac{k_t}{(k_a + k_t - \omega^2 m_a)(k_t - \omega^2 m_t) - k_t^2} F_0 \quad (6.6)$$

Thus, according to equation (6.5), if we tune the absorber mass and stiffness so that

$k_t - m_t \omega^2$ is zero, or $\frac{k_t}{m_t} = \omega^2$, we will be able to eliminate the displacement of the

primary mass (m_a). But this solution is only for pure sinusoidal input and not a shock input, as we have here. The transient response of the primary mass will not be eliminated with the above absorber properties. Dealing with shock mitigation means that we are dealing with the transient response of the system too. This is because the very first shock response of the system has steady state response as well as transient response. Therefore, we will consider transient response of the primary mass too.

The transient response of the system, which is excited by initial conditions, is written as [29]:

$$x(t)_{transient} = \frac{1}{|U|} \left\{ \begin{bmatrix} (u_{22}x_{10} - u_{12}x_{20}) \cos \omega_1 t + \frac{u_{22}v_{10} - u_{12}v_{20}}{\omega_1} \sin \omega_1 t \\ (u_{11}x_{10} - u_{21}x_{20}) \cos \omega_2 t + \frac{u_{11}v_{10} - u_{21}v_{20}}{\omega_2} \sin \omega_2 t \end{bmatrix} \begin{bmatrix} u_1 \\ u_2 \end{bmatrix} \right\} \quad (6.7)$$

Where:

$$U = \begin{bmatrix} u_1 & u_2 \end{bmatrix} = \begin{bmatrix} u_{11} & u_{12} \\ u_{21} & u_{22} \end{bmatrix}; \text{ is the modal vector}$$

ω_1 and ω_2 are the first and the second natural frequency of the system

$$x_{(0)} = \begin{bmatrix} x_{10} \\ x_{20} \end{bmatrix} \text{ and } v_{(0)} = \begin{bmatrix} v_{10} \\ v_{20} \end{bmatrix} \text{ are the initial displacements and velocities of each mass}$$

So the complete response of the system will be:

$$x(t) = x(t)_{transient} + x(t)_{steady_state} \quad (6.8)$$

Thus, as long as the transient effect appears in the system's response, there will be two frequencies appearing in the response; which are the natural frequency of the system and the excitation frequency. The transient effect will disappear after some time if the system has a sufficient damping. To see this phenomenon, a program in MATLAB is created to plot the system's response in Fig 6.3 subjected to a sinusoidal function. The system properties are given below:

$$m_a = 5 \text{ grams}; m_t = 0.1 m_a$$

$$\frac{1}{2\pi} \sqrt{\frac{k_a}{m_a}} = 1500 \text{ Hz}; \quad \frac{1}{2\pi} \sqrt{\frac{k_t}{m_t}} = 1000 \text{ Hz}; \quad \frac{1}{2\pi} \omega = 250 \text{ Hz}$$

$$F_0 = 10 \text{ N}$$

The system's response without damping is given in Fig 6.4, and the system's response with sufficient amount of damping is given in Fig 6.5.

Fig. 6.4 Displacement response of each mass without damping

From Fig 6.4, we can observe that there are two frequencies appearing in each mass response. One is the natural frequency of the system, and the other is the excitation frequency (ω). If a sufficient amount of damping is added to the system, the transient response will disappear after some time (see Fig 6.5). We can observe from Fig 6.5 that in the first 0.003 s, the transient response is still affecting the total system's response. But after 0.003 s, the remaining frequency appear in the response is only the excitation frequency.

Fig. 6.5 Displacement response of each mass with sufficient amount of damping

Equation (6.5) is valid only when the system is in steady state condition. To prove it, we simulate the problem in Fig 6.3 in MATLAB and we change the system properties of m_t and k_t so that $\sqrt{\frac{k_t}{m_t}} = \omega$. We also change the excitation frequency to 1000 Hz, to see the reduction more clearly in 0.01 s time range. Fig. 6.6 show the displacement response of m_a , before and after TVA's placement (the m_t and k_t is considered as TVA), the system's damping is neglected.

Fig. 6.6 Displacement response of primary mass without damping

Since the system's response is still containing the transient response, in which the system's natural frequency appear in the response, the TVA did not work to eliminate the primary mass vibration. If we add a sufficient amount of damping to eliminate the transient response, we will see that the primary mass vibration is eliminated after some time, which is after the transient response is completely eliminated. Fig. 6.7 shows the response of the primary mass before and after TVA placement with sufficient amount of damping. It shows that after 0.06 s, the primary mass vibration is eliminated. For a continuous excitation function such as sinusoidal disturbance that appears in most of machinery problem, the result in Fig. 6.7 will work. This is because the usage of the TVA is in the steady state condition. But at the beginning of the excitation force start to excite the primary mass (the very first wave), we are facing the same problem with the use of TVA for shock mitigation to reduce the slider vibration in the HDD.

Fig. 6.7 Displacement response of primary mass with sufficient damping

To discuss the use of TVA for shock mitigation, we simulate the problem in Fig 6.3 in MATLAB, but we change the TVA properties into $\frac{1}{2\pi} \sqrt{\frac{k_t}{m_t}} = 1400\text{Hz}$; which is close to the resonant of the primary system. We use the same mass ratio and the same damping value with the previous simulation. Fig. 6.8 shows the response of the primary mass before and after TVA placement. We can observe from Fig. 6.8 that the TVA did not

eliminate the primary mass vibration. But if we observe the system's response at time 0 to 0.001 s, on the first peak of the response, we will see different results. A closer look at the primary mass response at the first peak (0 to 0.001 s) is given at Fig. 6.9.

Fig. 6.8 Displacement response of primary mass with sufficient amount of damping; TVA tuned close to primary mass resonance

Fig. 6.9 Comparison of displacement response of the primary mass using different TVA frequency; with sufficient amount of damping

Although the primary mass vibration at the rest of the response (at steady state condition) did not eliminated, the TVA with frequency close to the resonant of the system are able to reduce the first peak response of the primary mass better than the TVA with frequency

the same as the excitation frequency (see Fig 6.9). From the result of Fig. 6.9, we conclude that in the beginning of response (the first peak/ wave), the absorber frequency does not affect the response significantly. It has been shown with a close number of reductions between two different TVA frequencies. Another conclusion is that the transient response has a greater portion in the system's response at the very first wave. That is why the reduction is slightly larger when the TVA is tuned to a frequency close to the resonant frequency.

Now we will discuss the system in Fig. 6.3 with half sine input; $F(t) = F_0 [\sin(\omega t) + \sin(\omega(t-a))]$, a is the time delay with value of half of the sine wave period (the shock duration). Fig. 6.10 shows a half sine acceleration input of 25 G's with 2 ms of shock duration (equal to 250 Hz of input frequency).

Fig. 6.10 Half sine input

If the system in Fig. 6.3 is excited by a half sine input, it will have the same response with the system in Fig. 6.3 excited by a sinusoidal input at time $0 < t < a$ ($a = 2$ ms). After termination of the shock (at $t > a$), the system in Fig. 6.3 will have a response of free vibration:

$$x(t) = x(t)_{transient}; \quad \text{for } t > a$$

With initial condition as:

$$x_{(t=a)} = \begin{bmatrix} x_{1t=a} \\ x_{2t=a} \end{bmatrix} \text{ and } v_{(t=a)} = \begin{bmatrix} v_{1t=a} \\ v_{2t=a} \end{bmatrix} \quad (6.9)$$

So after termination of the shock, the system will vibrate at its own natural frequency (see equation (6.7)).

Since we are interested in the shock region response, we focus our attention to the system's response at time $0 < t < a$. Thus, in that time region, the system's response will be the same as the system's response excited by the sinusoidal input at time $0 < t < a$. Thus, the system's response is:

$$x(t) = x(t)_{\text{transient}} + x(t)_{\text{steady_state}}; \quad \text{for } 0 < t < a$$

$$x(t) = x_t(t) + x_{ss}(t)$$

Since the response of the system at $t = 0$ is 0, we can write the system response separately as:

$$x_{ss}(t + \tau) = \begin{bmatrix} x_{ss1(t+\tau)} \\ x_{ss2(t+\tau)} \end{bmatrix} = \begin{bmatrix} X_{ss1} \sin\left(\frac{\pi}{a}(t + \tau)\right) \\ X_{ss2} \sin\left(\frac{\pi}{a}(t + \tau)\right) \end{bmatrix} \quad (6.10)$$

$$x_t(t + \tau) = \frac{1}{|U|} \left\{ \begin{aligned} &\left[(u_{22}x_{1t} - u_{12}x_{2t}) \cos \omega_1 t + \frac{u_{22}v_{1t} - u_{12}v_{2t}}{\omega_1} \sin \omega_1 t \right] u_1 \\ &+ \left[(u_{11}x_{1t} - u_{21}x_{2t}) \cos \omega_2 t + \frac{u_{11}v_{1t} - u_{21}v_{2t}}{\omega_2} \sin \omega_2 t \right] u_2 \end{aligned} \right\} \quad (6.11)$$

τ is time interval

Where the initial condition for transient response is:

$$x_{(t)} = \begin{bmatrix} x_{1t} \\ x_{2t} \end{bmatrix} \text{ and } v_{(t)} = \begin{bmatrix} v_{1t} \\ v_{2t} \end{bmatrix}; \text{ so it means the combined response at time } t \text{ is the initial}$$

condition for the transient response at time $t + \tau$.

Refer to the previous conclusion that the transient response has a larger portion in the first wave response; we will concentrate on the transient response of the system. The transient response of the primary mass can be written as:

$$x_{1t}(t + \tau) = \frac{1}{|U|} \left\{ \begin{aligned} &\left[(u_{22}x_{1t} - u_{12}x_{2t}) \cos \omega_1 t + \frac{u_{22}v_{1t} - u_{12}v_{2t}}{\omega_1} \sin \omega_1 t \right] u_{11} \\ &+ \left[(u_{11}x_{1t} - u_{21}x_{2t}) \cos \omega_2 t + \frac{u_{11}v_{1t} - u_{21}v_{2t}}{\omega_2} \sin \omega_2 t \right] u_{21} \end{aligned} \right\} \quad (6.12)$$

So to eliminate the primary mass vibration in equation (6.12), the below condition must be fulfilled:

$$u_{22}x_{1t} - u_{12}x_{2t} = 0 \quad (6.13)$$

$$u_{22}v_{1t} - u_{12}v_{2t} = 0 \quad (6.14)$$

$$u_{11}x_{1t} - u_{21}x_{2t} = 0 \quad (6.15)$$

$$u_{11}v_{1t} - u_{21}v_{2t} = 0 \quad (6.16)$$

If these conditions are fulfilled, the transient effect in time range $0 < t < a$ will be eliminated.

The transient response will start to have a value greater than zero (0) at time $t + 2\tau$. The initial conditions for the transient response at time $t + 2\tau$ are:

$$\begin{bmatrix} x_{1(t+\tau)} \\ x_{2(t+\tau)} \end{bmatrix} = \begin{bmatrix} X_{ss1} \sin\left(\frac{\pi}{a}(t+\tau)\right) \\ X_{ss2} \sin\left(\frac{\pi}{a}(t+\tau)\right) \end{bmatrix} \quad (6.17)$$

$$\begin{bmatrix} v_{1(t+\tau)} \\ v_{2(t+\tau)} \end{bmatrix} = \begin{bmatrix} \frac{\pi}{a} X_{ss1} \cos\left(\frac{\pi}{a}(t+\tau)\right) \\ \frac{\pi}{a} X_{ss2} \cos\left(\frac{\pi}{a}(t+\tau)\right) \end{bmatrix} \quad (6.18)$$

Then, substituting equations (6.17) and (6.18) to equations (6.13) – (6.16):

$$\begin{aligned} u_{22}X_{ss1} \sin\left(\frac{\pi}{a}(t+\tau)\right) &= u_{12}X_{ss2} \sin\left(\frac{\pi}{a}(t+\tau)\right) \\ u_{22} \frac{k_t - m_t \frac{\pi^2}{a^2} \left(\sin\left(\frac{\pi}{a}(t+\tau)\right)\right)}{\left(k_a + k_t - \frac{\pi^2}{a^2} m_a\right) \left(k_t - \frac{\pi^2}{a^2} m_t\right) - k_t^2} &= u_{12} \frac{k_t \left(\sin\left(\frac{\pi}{a}(t+\tau)\right)\right)}{\left(k_a + k_t - \frac{\pi^2}{a^2} m_a\right) \left(k_t - \frac{\pi^2}{a^2} m_t\right) - k_t^2} \\ u_{22} \left(\left(k_t - m_t \frac{\pi^2}{a^2}\right) \left(\sin\left(\frac{\pi}{a}(t+\tau)\right)\right) \right) &= u_{12} \left(k_t \left(\sin\left(\frac{\pi}{a}(t+\tau)\right)\right) \right) \end{aligned} \quad (6.19)$$

$$\begin{aligned} u_{22} \frac{\pi}{a} X_{ss1} \cos\left(\frac{\pi}{a}(t+\tau)\right) &= u_{12} \frac{\pi}{a} X_{ss2} \cos\left(\frac{\pi}{a}(t+\tau)\right) \\ u_{22} \left(\left(k_t - m_t \frac{\pi^2}{a^2}\right) \left(\cos\left(\frac{\pi}{a}(t+\tau)\right)\right) \right) &= u_{12} \left(k_t \left(\cos\left(\frac{\pi}{a}(t+\tau)\right)\right) \right) \end{aligned} \quad (6.20)$$

$$\begin{aligned} u_{11}X_{ss1} \sin\left(\frac{\pi}{a}(t+\tau)\right) &= u_{21}X_{ss2} \sin\left(\frac{\pi}{a}(t+\tau)\right) \\ u_{11} \left(\left(k_t - m_t \frac{\pi^2}{a^2}\right) \left(\sin\left(\frac{\pi}{a}(t+\tau)\right)\right) \right) &= u_{21} \left(k_t \left(\sin\left(\frac{\pi}{a}(t+\tau)\right)\right) \right) \end{aligned} \quad (6.21)$$

$$\begin{aligned} u_{11} \frac{\pi}{a} X_{ss1} \cos\left(\frac{\pi}{a}(t+\tau)\right) &= u_{21} \frac{\pi}{a} X_{ss2} \cos\left(\frac{\pi}{a}(t+\tau)\right) \\ u_{11} \left(\left(k_t - m_t \frac{\pi^2}{a^2}\right) \left(\cos\left(\frac{\pi}{a}(t+\tau)\right)\right) \right) &= u_{21} \left(k_t \left(\cos\left(\frac{\pi}{a}(t+\tau)\right)\right) \right) \end{aligned} \quad (6.22)$$

For the system of Fig. 6.3, the modal vector is given by:

$$u_1 = \left[\begin{array}{c} \frac{-1}{2ma} \left(k_t m_t + m_a k_t + k_a m_t + \sqrt{k_t^2 m_t^2 + 2k_t^2 m_t m_a + 2k_t m_t^2 k_a + m_a^2 k_t^2 - 2m_a m_t k_a k_t + k_a^2 m_t^2} \right) + k_t \\ k_t \\ 1 \end{array} \right]$$

$$u_2 = \left[\begin{array}{c} \frac{-1}{2ma} \left(k_t m_t + m_a k_t + k_a m_t - \sqrt{k_t^2 m_t^2 + 2k_t^2 m_t m_a + 2k_t m_t^2 k_a + m_a^2 k_t^2 - 2m_a m_t k_a k_t + k_a^2 m_t^2} \right) + k_t \\ k_t \\ 1 \end{array} \right]$$

Writing the square root components of the modal vector as:

$$A = \sqrt{k_t^2 m_t^2 + 2k_t^2 m_t m_a + 2k_t m_t^2 k_a + m_a^2 k_t^2 - 2m_a m_t k_a k_t + k_a^2 m_t^2}$$

Substituting the modal vector into equation (6.19):

$$\frac{u_{22}}{u_{12}} = \frac{\left(k_t \left(\sin \left(\frac{\pi}{a} (t + \tau) \right) \right) \right)}{\left(\left(k_t - m_t \frac{\pi^2}{a} \right) \left(\sin \left(\frac{\pi}{a} (t + \tau) \right) \right) \right)}$$

$$\frac{1}{\frac{-1}{2ma} \left(k_t m_t + m_a k_t + k_a m_t - A \right) + k_t} = \frac{\left(k_t \left(\sin \left(\frac{\pi}{a} (t + \tau) \right) \right) \right)}{\left(\left(k_t - m_t \frac{\pi^2}{a} \right) \left(\sin \left(\frac{\pi}{a} (t + \tau) \right) \right) \right)} \quad (6.23)$$

$$\frac{k_t}{\frac{-1}{2ma} \left(k_t m_t + m_a k_t + k_a m_t - A \right) + k_t} = \frac{k_t}{\left(k_t - m_t \frac{\pi^2}{a} \right)}$$

If $t + \tau = \alpha$, equation (6.23) can be written as:

$$\frac{k_t}{\frac{-1}{2ma} \left(k_t m_t + m_a k_t + k_a m_t - A \right) + k_t} = \frac{\left(k_t \left(\sin \left(\frac{\pi}{a} \alpha \right) \right) \right)}{\left(\left(k_t - m_t \frac{\pi^2}{a} \right) \left(\sin \left(\frac{\pi}{a} \alpha \right) \right) \right)}$$

$$\frac{k_t}{\frac{-1}{2ma} \left(k_t m_t + m_a k_t + k_a m_t - A \right) + k_t} = \frac{k_t}{k_t - m_t \frac{\pi^2}{a}}$$

$$\frac{-1}{2ma} \left(k_t m_t + m_a k_t + k_a m_t - A \right) + k_t = k_t - m_t \frac{\pi^2}{a}$$

$$\frac{-1}{2ma} \left(k_t m_t + m_a k_t + k_a m_t - A \right) = -m_t \frac{\pi^2}{a}$$

$$k_t m_t + m_a k_t + k_a m_t - m_t \frac{\pi^2}{a} m_a = A$$

$$\left(k_t m_t + m_a k_t + k_a m_t - m_t \frac{\pi^2}{a} m_a \right)^2 = A^2$$

$$\begin{aligned}
 & \left(k_t m_t + m_a k_t + k_a m_t - m_t \frac{\pi^2}{a} m_a \right)^2 = A^2 \\
 & k_t^2 m_t^2 + 2k_t^2 m_t m_a + 2k_t k_a m_t^2 + m_a^2 k_t^2 + 2k_a m_a k_t m_t + k_a^2 m_t^2 + \left(\frac{\pi}{a} \right)^4 m_t^2 m_a^2 = A^2 \\
 & 2k_a m_a k_t m_t + \left(\frac{\pi}{a} \right)^4 m_t^2 m_a^2 = -2m_a m_t k_a k_t \\
 & 4k_a k_t + \left(\frac{\pi}{a} \right)^4 m_t m_a = 0 \\
 & \frac{4k_a k_t}{m_a m_t} + \left(\frac{\pi}{a} \right)^4 = 0 \\
 & 4\omega_a^2 \omega_t^2 + \omega^4 = 0
 \end{aligned} \tag{6.24}$$

One can know that equation (6.24) is impossible to solve.

We try another way to substitute the modal vector into equation (6.21):

$$\begin{aligned}
 & \frac{\frac{-1}{2ma} (k_t m_t + m_a k_t + k_a m_t + A) + k_t}{k_t} = \frac{\left(k_t \left(\sin \left(\frac{\pi}{a} (t + \tau) \right) \right) \right)}{\left(\left(k_t - m_t \frac{\pi^2}{a} \right) \left(\sin \left(\frac{\pi}{a} (t + \tau) \right) \right) \right)} \tag{6.25} \\
 & \frac{\frac{-1}{2ma} (k_t m_t + m_a k_t + k_a m_t + A) + k_t}{k_t} = \frac{k_t}{k_t - m_t \frac{\pi^2}{a}} \\
 & \left(\frac{-1}{2m_a} (k_t m_t + m_a k_t + k_a m_t + A) + k_t \right) \left(k_t - m_t \frac{\pi^2}{a} \right) = k_t^2 \\
 & \frac{-k_t^2 m_t + \frac{\pi^2}{a} k_t m_t^2 - k_t^2 m_a + \frac{\pi^2}{a} k_t m_t m_a - k_a m_t k_t + \frac{\pi^2}{a} k_a m_t^2 - A k_t + \frac{\pi^2}{a} m_t A}{2m_a} + k_t^2 - \frac{\pi^2}{a} k_t m_t = k_t^2 \\
 & -k_t^2 m_t + \frac{\pi^2}{a} k_t m_t^2 - k_t^2 m_a + \frac{\pi^2}{a} k_t m_t m_a - k_a m_t k_t + \frac{\pi^2}{a} k_a m_t^2 - A k_t + \frac{\pi^2}{a} m_t A = \frac{\pi^2}{a} k_t m_t 2m_a \\
 & -k_t^2 m_t + \frac{\pi^2}{a} k_t m_t^2 - k_t^2 m_a + \frac{\pi^2}{a} k_t m_t m_a - k_a m_t k_t + \frac{\pi^2}{a} k_a m_t^2 - \frac{\pi^2}{a} k_t m_t 2m_a = A \left(k_t - \frac{\pi^2}{a} m_t \right)
 \end{aligned}$$

After long calculations, equation (6.25) becomes:

$$\frac{\pi^2}{a} (k_t m_t + k_t m_a - m_t k_a) + k_t k_a = \frac{\pi^4}{a} \left(m_t^2 + \frac{1}{2} m_t m_a \right)$$

We can substitute secondary mass by setting the mass ratio $\mu = \frac{m_t}{m_a}$, and also the

secondary stiffness by setting the stiffness ratio $\delta = \frac{k_t}{k_a}$

$$\frac{\pi^2}{a} (\delta k_a \mu m_a + \delta k_a m_a - \mu m_a k_a) + \delta k_a k_a = \frac{\pi^4}{a} \left(\mu^2 m_a^2 + \frac{1}{2} \mu m_a^2 \right)$$

If we assume the product of mass ratio and stiffness ratio is small, equation (6.25) becomes:

$$\begin{aligned}
 \frac{\pi^2}{a} (\delta k_a m_a - \mu m_a k_a) + \delta k_a k_a &= \frac{\pi^4}{a} \left(\frac{1}{2} \mu m_a^2 \right) \\
 \delta \left(\frac{\pi^2}{a} k_a m_a + k_a^2 \right) &= \mu \left(\frac{1}{2} \frac{\pi^4}{a} m_a^2 + \frac{\pi^2}{a} m_a k_a \right) \\
 \delta \left(\frac{\pi^2}{a} \frac{k_a m_a}{m_a^2} + \frac{k_a^2}{m_a^2} \right) &= \mu \left(\frac{1}{2} \frac{\pi^4}{a} \frac{m_a^2}{m_a^2} + \frac{\pi^2}{a} \frac{m_a k_a}{m_a^2} \right) \\
 \frac{\delta}{\mu} &= \frac{\left(\frac{1}{2} \frac{\pi^4}{a} m_a^2 + \frac{\pi^2}{a} m_a k_a \right)}{\left(\frac{\pi^2}{a} \omega_a^2 + \omega_a^4 \right)} \\
 \frac{k_t / k_a}{m_t / m_a} &= \frac{\left(\frac{1}{2} \frac{\pi^4}{a} m_a^2 + \frac{\pi^2}{a} m_a k_a \right)}{\left(\frac{\pi^2}{a} \omega_a^2 + \omega_a^4 \right)} \\
 \omega_t^2 &= \omega_a^2 \frac{\left(\frac{1}{2} \omega_a^4 + \omega^2 \omega_a^2 \right)}{\left(\omega^2 \omega_a^2 + \omega_a^4 \right)}
 \end{aligned} \tag{6.26}$$

So then equation (6.26) gives us the optimum absorber (secondary) mass natural frequency ($\omega_t^2 = k_t / m_t$). But equation (6.26) is valid when the initial condition is pure from the steady state response (at time $t + 2\tau$). So if we check the result by simulation in MATLAB, the TVA does not work.

Since the above methods do not work, we will try a different approach. Generally, the primary mass displacement and velocity response before they reach the first peak (the primary mass peak) is always larger than the secondary mass response. After the first peak, the secondary mass will have larger response, and it have to be greater than the primary mass response to absorb the primary mass vibration. Thus, the transient response at time before the primary mass reaches its peak response can be written as:

$$x_{1t(t < peak)} = \frac{1}{|U|} \left\{ \left[\left(u_{22} X - u_{12} x \right) \cos \omega_1 (t < peak) + \frac{u_{22} V - u_{12} v}{\omega_1} \sin \omega_1 (t < peak) \right] u_{11} \right\} + \left\{ \left[\left(u_{11} X - u_{21} x \right) \cos \omega_2 (t < peak) + \frac{u_{11} V - u_{21} v}{\omega_2} \sin \omega_2 (t < peak) \right] u_{21} \right\} \tag{6.27}$$

Where:

$$\begin{matrix} X > x \\ V > v \end{matrix}, \text{ for convenience we set } \begin{matrix} x = \alpha X \\ v = \beta V \end{matrix}, \text{ with } \alpha, \beta < 1$$

Then we back to equation (6.13) to (6.16):

$$\begin{aligned} u_{22}X - u_{12}\alpha X &\approx 0 \\ u_{22}V - u_{12}\beta V &\approx 0 \\ u_{11}X - u_{21}\alpha X &\approx 0 \\ u_{11}V - u_{21}\beta V &\approx 0 \end{aligned} \quad (6.28)$$

We write again the condition in equation (33), the first line and the third line:

$$\begin{aligned} u_{22}X - u_{12}\alpha X &\approx 0 \\ X(u_{22} - u_{12}\alpha) &\approx 0 \end{aligned} \quad (6.29)$$

$$\begin{aligned} u_{11}X - u_{21}\alpha X &\approx 0 \\ X(u_{11} - u_{21}\alpha) &\approx 0 \end{aligned} \quad (6.30)$$

We will begin with equation (6.29):

$$\begin{aligned} X(u_{22} - u_{12}\alpha) &\approx 0 \\ u_{22} - u_{12}\alpha &\approx 0 \\ u_{22} &\approx u_{12}\alpha \end{aligned}$$

It means we want u_{22} to be as close as possible with $u_{12}\alpha$.

$$\begin{aligned} u_{22} &\approx u_{12}\alpha \\ 1 &\approx \frac{\frac{-1}{2ma} \left(k_t m_t + m_a k_t + k_a m_t - \sqrt{k_t^2 m_t^2 + 2k_t^2 m_t m_a + 2k_t m_t^2 k_a + m_a^2 k_t^2 - 2m_a m_t k_a k_t + k_a^2 m_t^2} \right) + k_t}{k_t} \alpha \\ \frac{k_t}{\alpha} &\approx \frac{-1}{2ma} (k_t m_t + m_a k_t + k_a m_t - A) + k_t \\ \frac{k_t}{\alpha} - k_t + \frac{k_t m_t + m_a k_t + k_a m_t}{2m_a} &\approx \frac{A}{2m_a} \\ \left(\frac{k_t}{\alpha} - k_t + \frac{k_t m_t + m_a k_t + k_a m_t}{2m_a} \right)^2 &\approx \left(\frac{A}{2m_a} \right)^2 \\ \frac{k_t^2}{\alpha^2} - \frac{2k_t^2}{\alpha} + \frac{2k_t^2 m_t + 2m_a k_t^2 + 2k_a k_t m_t}{\alpha 2m_a} + k_t^2 - \frac{2k_t^2 m_t + 2m_a k_t^2 + 2k_a k_t m_t}{2m_a} \\ &+ \frac{k_t^2 m_t^2 + 2k_t^2 m_a m_t + 2k_t k_a m_t^2 + k_t^2 m_a^2 + 2k_t k_a m_t m_a + k_a^2 m_t^2}{4m_a^2} \approx \frac{A^2}{4m_a^2} \end{aligned}$$

$$\frac{k_t^2}{\alpha^2} - \frac{2k_t^2}{\alpha} + \frac{2k_t^2 m_t + 2m_a k_t^2 + 2k_a k_t m_t}{\alpha 2m_a} + k_t^2 - \frac{2k_t^2 m_t + 2m_a k_t^2 + 2k_a k_t m_t}{2m_a} + \frac{2k_t k_a m_t m_a}{4m_a^2} \approx 0$$

Substituting secondary mass and stiffness with ratio $\mu = \frac{m_t}{m_a}$, and $\delta = \frac{k_t}{k_a}$, also

multiplying equation with α^2 :

$$\begin{aligned} \delta^2 k_a^2 - 2\alpha \delta^2 k_a^2 + \alpha \frac{2\delta^2 k_a^2 \mu m_a + 2m_a \delta^2 k_a^2 + 2k_a^2 \delta \mu m_a}{2m_a} + \alpha^2 \delta^2 k_a^2 \\ - \alpha^2 \frac{2\delta^2 k_a^2 \mu m_a + 2m_a \delta^2 k_a^2 + 2k_a^2 \delta \mu m_a}{2m_a} + \alpha^2 \frac{2k_a^2 \delta \mu m_a m_a}{4m_a^2} = 0 \\ \delta^2 k_a^2 - 2\alpha \delta^2 k_a^2 + \alpha (\delta^2 k_a^2 \mu + \delta^2 k_a^2 + k_a^2 \delta \mu) + \alpha^2 \delta^2 k_a^2 - \alpha^2 (\delta^2 k_a^2 \mu + \delta^2 k_a^2 + k_a^2 \delta \mu) + \frac{1}{2} \alpha^2 k_a^2 \delta \mu = 0 \end{aligned}$$

Grouping equation with μ and δ to find the parameter, also eliminating part which contains the highest degree:

$$\begin{aligned} \delta^2 k_a^2 - 2\alpha \delta^2 k_a^2 + \alpha (\delta^2 k_a^2 + k_a^2 \delta \mu) &\approx 0 \\ \delta (\delta k_a^2 - 2\alpha \delta k_a^2 + \alpha \delta k_a^2 + \alpha \mu k_a^2) &\approx 0 \\ \delta k_a^2 - 2\alpha \delta k_a^2 + \alpha \delta k_a^2 + \alpha \mu k_a^2 &\approx 0 \\ k_a^2 (\delta - \alpha \delta + \alpha \mu) &\approx 0 \\ \delta (\alpha - 1) &\approx \alpha \mu \\ \frac{\delta}{\mu} &\approx \frac{\alpha}{\alpha - 1} \\ \frac{k_t / k_a}{m_t / m_a} &\approx \frac{\alpha}{\alpha - 1} \\ \omega_t^2 &\approx \omega_a^2 \frac{\alpha}{\alpha - 1} \end{aligned} \tag{6.31}$$

The value of α is less than 1 when the primary mass has not reached its first peak. But we want the TVA to eliminate the primary mass vibration, so at all times the TVA response should be larger than the primary mass response. The TVA used in continuous forcing function like sinusoidal is moving greatly while the primary mass is not moving. So it implies that in equation (6.31), the value of α must be greater than 1. Equation (6.31) now can be written as:

$$\omega_t^2 \approx \omega_a^2 \frac{\alpha}{\alpha - 1}$$

$$\omega_t \approx \omega_a \sqrt{\frac{\alpha}{\alpha - 1}}$$

Since $\alpha \gg 1$

$$\omega_t \approx \omega_a \quad (6.32)$$

So equation (6.32) shows us that the optimum TVA parameter (TVA natural frequency) is close to the primary system's natural frequency. But this condition can only minimize the transient response, also minimizing the total response during the shock duration. The TVA cannot completely eliminate the primary mass response, particularly at the shock duration region.

From the analytical study we have done, we cannot find the exact TVA parameters which can completely eliminate the transient response or the shock response during the shock duration in closed form solution. The only result is that the TVA natural frequency should be close to the primary system's natural frequency (by simulations, the TVA natural frequency is found to be between 0.9 to 1 times of the main system's natural frequency). We did not look at the steady state response or eliminating the steady state response during the shock, because the system's response after shock termination is a transient response and the transient response portion during the shock duration is greater than the steady state response. Therefore, this analytical result matches with the study from [37], where they set the frequency ratio as 0.89 to 0.9 which is close to the primary system's natural frequency.

To find out how much reduction we can get by using TVA, we will use the same simulation in MATLAB for problem in Fig 6.3. We set the TVA natural frequency to be the same as the primary system frequency $\omega_t^2 = \omega_a^2$ or $k_t/m_t = k_a/m_a$, so then the value of mass and stiffness ratio will be the same. We perform a simulation in MATLAB, giving the primary mass in Fig. 1 a shock input with 2 ms duration, 25 G's of shock magnitude, with system properties as below:

$m_a = 5$ grams; mass ratio $= \mu = \delta = 0.1$

$$\frac{1}{2\pi} \sqrt{\frac{k_a}{m_a}} = 1500 \text{ Hz}; \quad \frac{1}{2\pi} \sqrt{\frac{k_t}{m_t}} = 1500 \text{ Hz}$$

Shock magnitude = 25 G's; shock duration = 2 ms

The primary mass displacement response before and after TVA placement is given in Fig. 6.11. The system's damping is neglected.

Fig. 6.11 Primary mass displacement response to a shock input

From the result of Fig 6.11, the reduction during the shock period is 16%, and the reduction after shock termination is 86%. The reduction after termination is greater since the response of the primary mass is only the transient response. We can also observe from Fig. 6.11 that during the shock period, the primary mass displacement after TVA placement shows only a small portion of transient effect, in which the natural frequency appearance is smaller than before. During the simulation, we change the value of frequency ratio and mass ratio by trial and error, and we achieved the best reduction for problem of Fig 6.3 with shock duration 2 ms by using the above configuration (mass ratio = 0.1; frequency ratio = 1). This simulation also has proven that the best reduction is obtained by tuning the TVA frequency close to the primary system's frequency before TVA placement. The best mass and stiffness ratio for this system is also found by simulation as 0.1.

From this study, we conclude that to isolate a shock input, the optimum TVA frequency needs to be close to the primary system's natural frequency before TVA placement.

Equation (6.32) only gives us direction that the optimum TVA frequency for shock isolation purpose is close to the first resonant of the primary system. But to obtain the exact optimum TVA frequency and also the optimum TVA mass portion of another system, we need to perform a simulation by comparing the reduction from different TVA properties (trial and error).

Since the problem of the HDD subjected to a shock input is a base excitation problem, the system in Fig 6.3 is now remodeled in Fig 6.12 to resemble base excitation problem. This time we only use simulation in MATLAB (trial and error) to find out the optimum TVA frequency. The model properties in Fig 6.12 are defined below:

Primary mass (m_a)	: 5 grams
Primary natural frequency	: 1500 Hz
Absorber mass (m_t)	: 10% of primary mass
Absorber frequency	: varied from 500 Hz to 2000 Hz
Shock input ($F(t)$)	: 25 G's 2 ms ³

Fig. 6.12 Two DOF system; with TVA (m_t) (base excitation)

The simulation results are given in Fig 6.13.

³The shock level is arbitrarily selected, since we are only interested in investigating how much acceleration and displacement reduction (in percentage) we can get with TVA placement. The shock level will not affect the percentage of reduction. On the other hand, the shock duration of 2 ms is selected from one of Seagate HDDs' shock tolerance specification, which are 2 ms, 1 ms, and 0.5 ms. In section 6.4, we use shock duration of 0.5 ms, which is the shock duration specified by Seagate for investigating the ST1 drive (1" drive).

Fig. 6.13 Primary mass displacement response to a shock input (base excitation)

So then we have proven that equation (6.32) can lead us to find the optimum TVA frequency. The displacement reduction using 0.93 (1400 Hz TVA frequency) of the primary natural frequency is about 9% in the first wave, and 80% in the second wave. However, the amount of reduction is varied; it depends on the shock input duration and

the primary system’s natural frequency. In this study, we did not perform an optimization study to find the optimum TVA properties.

6.4 ANSYS Simulation

First of all, the HAA mode shapes are extracted using modal analysis in ANSYS to find the natural modes of HAA (particularly the suspension modes). The HAA boundary conditions for modal analysis and shock analysis are given in Fig 6.14.

Fig 6.14 Boundary conditions of HAA

The HAA nodes that are connected to the bearing shaft are constrained so it can only move in the Z direction. A *keypoint* at the coordinate origin is created to resemble a shaker. The shaker mass (1 kg) is chosen as such to give a clean shock input magnitude. Then, the shaker is coupled with the HAA nodes that are connected to the pivot axis in the Z direction. This is done to resemble base excitation problem.

The elastic natural modes of the HAA are given in Table 6.1.

Table 6.1 Elastic natural modes of HAA

Mode	Mode shapes
1 st (290 Hz)	<div><div><div>1</div><div>DISPLACEMENT</div><div>STEP=1</div><div>SUB =2</div><div>FREQ=290.328</div><div>DMX =28320</div></div><div><div>ANSYS</div><div>JUL 7 2009</div><div>19:57:20</div><div>Arm_suspension_with_mass_compressed</div></div><div></div></div>

Next, we perform a shock analysis in ANSYS using transient analysis of the HAA without TVA. The analysis is performed to observe the HAA response and to prove that the HAA, particularly the suspension, is vibrating dominantly at its own natural frequency when subjected to a shock input. Fig 6.15 gives the location where the HAA acceleration responses to shock input will be plotted. The input shock to the HAA is created by using the same principle in Chapter IV, section 4.6 (see Fig 4.12 and table 4.1). The shock magnitude used is 500 G's, with shock duration 0.5 ms. To generate this shock, the force magnitude given to the shaker is 5 kN. The HAA responses at points A, B, C, and D can be seen in Fig 6.16.

Fig 6.15 HAA boundary condition, shock input location, and the location where the shock response of HAA are observed

Fig 6.16 HAA responses to a shock input

From the shock analysis result of Fig 6.16, it can be observed that the suspension (at points B, C, and D) are vibrating dominantly at 290 Hz and 580 Hz. 290 Hz and 580 Hz are the 1st and the 2nd elastic natural frequencies obtained from the modal analysis (see Table 6.1). Therefore, we have proven that the HAA is vibrating dominantly at its own natural frequency when subjected to a shock input.

We will try five different locations to put the TVA, and these locations can be seen in Fig 6.17. The acceleration responses of the HAA, where the TVA are applied, are given in Fig 6.18.

The TVA FE model in ANSYS of Fig 6.16 is only a theoretical model, meaning that the TVA's dimension, density, and Young's modulus are not real. The TVA's dimension is too small to be manufactured and added to the current ST1 product. Therefore, if TVAs work, we suggest changing the suspension or the slider holder designs in the new HDD model so that the TVA structure can be embedded on the HAA.

Fig 6.17 TVA locations

Fig 6.18 Acceleration response on TVA locations

From the result of Fig 6.17, the HAA locations where the TVA 2, 3 and 4 are applied are vibrating dominantly at 290 Hz; whereas the HAA locations where the TVA 1 and 5 are applied are vibrating at higher frequency (580 Hz and up).

The TVA frequency is then tuned to 260 Hz (0.9 of the 1st elastic natural frequency of the HAA) or 522 Hz (0.9 of the 2nd elastic natural frequency of the HAA). In this simulation, we did not include the air bearing due to modeling limitation. However, this condition is sufficient because we are only interested to investigate if the TVA has a potential to reduce the head/ slider displacement and acceleration. The TVA configurations and its properties used in simulations in ANSYS are given in Table 6.2. The first two configurations are using the TVA only at one location per configuration, whereas the rest are using more than one TVA per configuration. We simulated many possible configurations using five different locations shown in Fig 6.17. We conclude that using only one TVA is not enough, but using too many TVA is also not good. We show only eleven configurations in Table 6.1 that represent many possible configurations we have tried.

TVA 1, TVA 2, and TVA 5 are placed on the suspension (suspension is the green colored elements), then its mass ratio (μ) is measured relative to the total mass of the suspension. TVA 3 and TVA 4 are placed on the slider, so then its mass ratio (μ) is measured relative to the total mass of the slider.

Table 6.2 TVA configuration and TVA properties

Config.	TVA 1	TVA 2	TVA 3	TVA 4	TVA 5
I	$\mu = 2.3\%$ 260 Hz; $\zeta = 0$	-	-	-	-
II	-	$\mu = 2.3\%$ 260 Hz; $\zeta = 0$	-	-	-
III	-	-	$\mu = 2.3\%$ 260 Hz; $\zeta = 0$	-	$\mu = 2.3\%$ 260 Hz; $\zeta = 0$
IV	-	-	$\mu = 2.3\%$ 522 Hz; $\zeta = 0$	-	$\mu = 2.3\%$ 260 Hz; $\zeta = 0$
V	-	-	$\mu = 2.3\%$ 260 Hz; $\zeta = 0$	-	$\mu = 2.3\%$ 522 Hz; $\zeta = 0$
VI	-	-	$\mu = 10\%$	-	$\mu = 10\%$

			260 Hz; $\zeta = 0$		522 Hz; $\zeta = 0$
			$\mu = 10\%$		$\mu = 10\%$
VII	-	-	260 Hz; $\zeta = 0.1$	-	522 Hz; $\zeta = 0.1$
			$\mu = 10\%$		$\mu = 20\%$
VIII	-	-	260 Hz; $\zeta = 0$	-	522 Hz; $\zeta = 0$
			$\mu = 10\%$		$\mu = 40\%$
IX	-	-	260 Hz; $\zeta = 0$	-	522 Hz; $\zeta = 0$
			$\mu = 20\%$		$\mu = 20\%$
X	-	-	260 Hz; $\zeta = 0$	-	522 Hz; $\zeta = 0$
	$\mu = 1.15\%$	$\mu = 1.15\%$	$\mu = 1.15\%$	$\mu = 1.15\%$	$\mu = 1.15\%$
XI	522 Hz; $\zeta = 0$	260 Hz; $\zeta = 0$	260 Hz; $\zeta = 0$	260 Hz; $\zeta = 0$	522 Hz; $\zeta = 0$

Fig 6.19 Slider nodes to be observed

The slider nodes which will be monitored for its acceleration and displacement responses are given in Fig 6.19. Nodes 1 and 3 will have the same responses; and the same for nodes 2 and 4, since the model is symmetric. Thus, we only observe node 1 and node 2 for its acceleration and displacement response. The shock input given at the shaker node is 500 G's with 0.5 ms duration. The simulation results which are the acceleration and displacement reduction of nodes 1 and 2 are given in Table 6.3. The acceleration and displacement reduction of the slider are measured at the first peak of the response.

The maximum acceleration and displacement response is achieved by configuration VIII. The slider acceleration is reduced by 40% which is significant; and the slider displacement is reduced by only 8.75% (by averaging nodes 1 and 2 responses).

Table 6.3 List of acceleration and displacement reduction on slider nodes

TVA Configuration	Node 1 reduction (%)		Node 2 reduction (%)	
	Acceleration	Displacement	Acceleration	Displacement
I	18	1.6	-2	-8
II	24	1.1	2.7	-8.3
III	1.2	0.5	3	1.8
IV	13.7	-0.5	5.5	0.2
V	9	1.3	7.7	0.8
VI	24.5	6.11	39.5	8.6
VII	24.8	6.06	40.75	8.49
VIII	33.6	7.34	46.7	10.16
IX	40	-1.1	55	9.9
X	25.7	9.68	54.5	0.6
XI	6.1	-0.3	5.3	0.2

To predict the shock tolerance improvement, we simulate the HAA using TVA configuration VI; we increase the shock magnitude by 10% and observe the slider nodes acceleration and displacement. The slider nodes acceleration and displacement before and after TVA configuration VI placement are given in Figs 6.20 to 6.23.

From Figs 6.20 and 6.21, even after the shock input is increased by 10%, by using TVA configuration VI the acceleration response of slider nodes are still far below slider nodes acceleration without using TVA. From Figs 6.22 and 6.23, the displacement response of the slider nodes when the shock is increased by 10% and using TVA configuration VI, the first peak of slider nodes displacement are almost the same with the first peak of slider nodes displacement without using TVA. From these results, we can only predict from the displacement response, that the shock tolerance of the ST1 has been increased by 10%. If this prediction is correct, the improvement by using TVA on the ST1 is the same as achieved by [13]. Simulation must be done on operating HDD with the real air bearing model to find out the real shock tolerance improvement.

Fig 6.20 Node 1 acceleration response

Fig 6.21 Node 2 acceleration response

Fig 6.22 Node 1 displacement response

Fig 6.23 Node 2 displacement response

6.5 Short Conclusion

In this Chapter, we have investigated that the TVA works to reduce the slider displacement and acceleration of the ST1 when the HAA is subjected to a shock input. The acceleration of the slider is reduced significantly by 40%, but only 8.75% slider displacement reduction is achieved. From simulation in ANSYS, we predict that with our current TVA installment (TVA configuration VIII), the shock tolerance of the ST1 has been increased by 10% (for 0.5 ms shock duration).

Since we are dealing with multi degree of freedom system, it is not that easy to achieve 16% reduction which is achieved when we are dealing with two degree of freedom system (discussed in section 6.3). Where the TVAs are located, how many TVAs are used, what are the TVAs frequencies, and what are the TVA mass ratios are the parameters that we have to specify in multi degree of freedom system problem; so then the number of possible configurations are too many to mention. Different system's characteristic and shock duration also resulting in different TVA performance (see Figs 5 and 6 in [13]).

According to our study on two degree of freedom system in section 6.3, it is possible to achieve shock reduction of more than 10%. It is a matter that we have not found the optimum configurations for the TVA parameters for the ST1's HAA.

Future students can perform a classical optimization study on the TVA parameters for multi degree of freedom system to see if more than 10% of shock reduction can be achieved. The shock tolerance improvement verification using the model that has the real air bearing (HDD in operating condition) is also suggested for future students.

As our final conclusion on TVA study, we list several methods that will be useful for TVA study on shock mitigation:

- The best location for the TVA is on the location that has a relatively high acceleration response (when the system is subjected to a shock input); this is shown by our simulation that the best locations are on the TVA 3 and TVA 5 (see Fig 6.17 for acceleration responses at TVA location);
- For multi degree of freedom system, the TVA frequency can be determined by the dominant frequency of the acceleration response where the TVA will be located (when the system is subjected to a shock input); this is shown by our simulation that TVA 3 works best if it is tuned to 260 Hz, and TVA 5 works best if it is tuned to 520 Hz;
- The trial and error and optimization analyses are then performed to find the optimum mass ratio;

Chapter VII

USE OF TUNED VIBRATION ABSORBER (TVA) ON THE HEAD ACTUATOR ASSEMBLY (HAA) OF 2.5" SEAGATE HDD AS A SHOCK ISOLATOR

6.2 Introduction

In Chapter VI, we have investigated the use of TVA as a shock isolator for the ST1 HDD. We tried many possible TVA configurations and TVA locations on ST1's HAA (on the suspension and slider) but we only achieved an 8.75% reduction in displacement. In this Chapter, we are interested to use the TVA on the current product of 2.5" Seagate HDD. We learned from Chapter VI that using too many TVAs on the suspension is not good, so we want to try to put the TVA not only on the suspension of the 2.5" drive but to see if we can get displacement reduction more than 10%. Since the first arm bending natural frequency of 2.5" drive is lower than the ST1, we consider putting the TVA on the arm as well as on the suspension of the 2.5" Seagate HDD. Fig 7.1 shows the FE model of HAA of 2.5" drive. The FE model of the HAA is obtained from Seagate.

Fig 7.1 FE model of HAA of 2.5" Seagate HDD

The FE model of the arm uses 8 noded solid 45 elements, and the material is Aluminum. It has three arms holding four suspensions. The FE model of the suspension uses 4 noded shell 63 elements, and the material is steel. A closer look at the suspension is given in Fig 7.2.

Fig 7.2 Closer look of FE model of Fig 7.1 at the suspension

In this Chapter, we will first examine the natural frequencies and modes of the HAA. Then we will develop a simple model of HAA in MATLAB which resembles the FE model of HAA in ANSYS. With the simple model, we will simulate to use the TVA on the arm and on the suspension. After we obtain the optimum properties of the TVA using MATLAB, we will simulate the problem in ANSYS using transient analysis (to verify the simulation results in MATLAB).

6.3 Determining The TVA Frequency; Developing Simple Model and Simulation in MATLAB

First, we will conduct a modal analysis in ANSYS for the HAA of 2.5" Seagate HDD to find its natural frequencies and modes. The method used to extract the eigenvectors and eigenvalues is Block Lanczos. The natural modes are searched up to 2500 Hz only, since we know already from Seagate that the arm resonant is around 1500 to 1600 Hz. In addition, the modes higher than 2500 Hz will not be excited, since we will only use shock input duration of 2 ms to 0.5 ms (equal to 250 Hz to 1000 Hz input frequency).

We do not have the model of the real air bearing in the ANSYS FE model. But to prove that the air bearing is not a concern in investigating the TVA performance, we set two kind of slider configuration. The first configuration is the slider without linear spring, and the second one is the slider with a linear spring. We use 0.4 lbf/in for the spring constants (each spring). We will see in the results that the TVA still works for the two configurations of the slider. The configuration of the slider is given in Fig 7.3. Fig 7.3 also gives the nodes numbering. The node is numbered from 1 to 4. These nodes are the corner node of the slider in the model, so it represents the slider movement. These four nodes will be monitored for its acceleration and displacement (node 1 to node 4 in Fig 7.3) throughout our analyses.

Fig 7.3 Slider configuration and node numbering

The boundary condition for the modal analysis is given in Fig 7.4. The upper and bottom node of the pivot bearing are constrained in all directions. One end of the linear springs is held fixed in all directions.

Fig 7.4 Boundary conditions for modal analysis

The lists of HAA natural frequencies found from modal analysis are given in table 7.1. The mode shapes of the first suspension bending and the first arm bending are given in Fig 7.5.

Table 7.1 HAA natural frequencies

Mode	Natural Frequency (Hz)
1	397.48
2	398.91
3	879.78
4	881.46
5	940.07
6	940.08
7	940.09
8	940.16
9	1444.6
10	1527.8
11	1628.1
12	2209.1
13	2345.3

Fig 7.5 HAA mode shapes

The next step is to develop the simple model in MATLAB. Knowing the natural frequencies and modes of the HAA, we assume a sub – model to develop the simple MATLAB model. Fig 7.6 shows the sub – model of the HAA. The sub – model consists of the bearing + HAA rigid block, arm, and suspension + slider. The bearing + HAA rigid block is a sub – model that has a very high natural frequency, so we assume this sub – model as a rigid body. The arm has a natural frequency of 1527.8 Hz. The arm model consists of mass and spring. The suspension + slider has a natural frequency of 397.48 Hz

(no spring), and 879.78 Hz (with spring). The suspension + slider also consist of mass and spring.

Fig 7.6 Sub – model of HAA

Fig 7.7 Simple model of HAA

The simple model in MATLAB is given in Fig. 7.7. In this model, we add a moving base to act as the shaker (where we give a shock input). On top of the moving base is the

bearing + HAA rigid block. The bearing stiffness is and tunable in the model. The arm and the suspension + slider are placed on top of the bearing + HAA rigid block.

The bearing + HAA rigid block mass, arm mass, and the suspension + slider mass are matched with the mass obtained from the FE model of HAA. The stiffness in the simple model is tuned so that the eigenvalues of the simple MATLAB model of Fig 7.7 is match to the eigenvalues of the FE model of HAA in ANSYS.

The simple model in MATLAB is developed using Newton's law, and a MATLAB m – file is then created. The m – file script of the model of Fig 7.7 is available in appendix A.4. Free body diagram of Fig 7.7 is shown in Fig 7.8 by assuming $x_3 > x_2 > x_1 > y$

Fig 7.8 Free body diagram of the simple model

From the free body diagram (FBD) of Fig 7.8, we can develop the equation of the motion as below:

$$\begin{aligned} M_S \ddot{x}_3 + k_s (x_3 - x_2) &= 0 \\ M_A \ddot{x}_2 + k_A (x_2 - x_1) &= k_s (x_3 - x_2) \\ M_{RB} \ddot{x}_1 + k_B (x_1 - y) + c_B (\dot{x}_1 - \dot{y}) &= k_A (x_2 - x_1) \end{aligned} \quad (7.1)$$

Transforming equation (7.1) into Laplace domain with zero initial condition:

$$\begin{aligned} s^2 X_3(s) M_S + X_3(s) k_s &= X_2(s) k_s \\ s^2 X_2(s) M_A + X_2(s) (k_A + k_s) &= X_3(s) k_s + X_1(s) k_A \\ s^2 X_1(s) M_{RB} + s X_1(s) c_B + X_1(s) (k_B + k_A) &= X_2(s) k_A + Y(s) k_B + s Y(s) c_B \end{aligned} \quad (7.2)$$

Substituting the first equation to the second equation in equation (7.2) results:

$$\begin{aligned}
 &\rightarrow X_3(s) = X_2(s) \frac{k_s}{(s^2 M_s + k_s)} \\
 &\Rightarrow s^2 X_2(s) M_A + X_2(s) (k_A + k_s) = X_2(s) \frac{k_s}{(s^2 M_s + k_s)} k_s + X_1(s) k_A \\
 &\Leftrightarrow X_2(s) \left(s^2 M_A + (k_A + k_s) - \frac{k_s^2}{(s^2 M_s + k_s)} \right) = X_1(s) k_A \quad (7.3) \\
 &\Leftrightarrow X_2(s) = X_1(s) \frac{k_A}{\left(s^2 M_A + (k_A + k_s) - \frac{k_s^2}{(s^2 M_s + k_s)} \right)}
 \end{aligned}$$

Substituting the last equation in equation (7.3) into the last line in equation (7.2), we get:

$$\begin{aligned}
 X_1(s) (s^2 M_{RB} + s c_B + (k_B + k_A)) &= X_1(s) \frac{k_A}{\left(s^2 M_A + (k_A + k_s) - \frac{k_s^2}{(s^2 M_s + k_s)} \right)} k_A + Y(s) (s c_B + k_B) \\
 X_1(s) \left(s^2 M_{RB} + s c_B + (k_B + k_A) - \frac{k_A^2}{\left(s^2 M_A + (k_A + k_s) - \frac{k_s^2}{(s^2 M_s + k_s)} \right)} \right) &= Y(s) (s c_B + k_B) \\
 X_1(s) &= Y(s) \frac{(s c_B + k_B)}{\left(s^2 M_{RB} + s c_B + (k_B + k_A) - \frac{k_A^2}{\left(s^2 M_A + (k_A + k_s) - \frac{k_s^2}{(s^2 M_s + k_s)} \right)} \right)} \quad (7.4)
 \end{aligned}$$

So then with equations (7.3) and (7.4) we can find the relationship between the shock input (Y(s)) and three values of X₁(s), X₂(s), and X₃(s). In MATLAB, the above transfer functions are used to perform transient analysis.

Next we will simulate the simple model of Fig 7.7. The simulation is performed to find out which shock duration is the worst for the slider displacement and acceleration response (0.5 ms, 1 ms, or 2 ms), and also to check whether the simple model can resemble the ANSYS FE model responses.

The simple model properties and natural frequencies are given in table 7.2. Table 7.2 shows that the natural frequencies of the simple model resemble the FE model in ANSYS.

Table 7.2 Simple model properties and natural frequencies

	Mass (grams)	Stiffness (N/m)	Natural Frequencies
Moving base	5E6		
Bearing + HAA rigid block	2.5	2.4674 E8	1 st 396 (suspension)
Arm	0.65	5.9913 E4	2 nd 1531 (arm)
Suspension + slider	0.123	761.4742	3 rd 50000 (rigid body)

The shock input used in the simulation is 25 G's with duration of 0.5 ms, 1 ms, and 2 ms. The shock level of 25 G's is arbitrarily selected, since we are only interested in comparing slider acceleration and displacement response for different shock durations, but not shock level. The MATLAB simulation results are given in Figs 7.9 and 7.10.

From Figs 7.9 and 7.10, we learn that the longer the shock duration, worse the acceleration and displacement response magnitude. This is because the longer shock duration has higher energy content.

Fig 7.9 Acceleration response of slider (MATLAB results)

Fig 7.10 Displacement response of slider (MATLAB results)

To verify the simple model accuracy, we perform a transient analysis in ANSYS. The boundary conditions of the HAA are explained in Fig 7.11. A shaker node is created under the bearing nodes. The shaker mass is tuned as such to create a clean shock input magnitude of 25 G's. This shaker node is coupled with the center of the bearing nodes, and also coupled with the ground of the linear spring on the slider (see Fig 7.4). The shaker node, the center of the bearing nodes, and the base of the air bearing spring on the slider are constrained so it can only move in the Z direction. The shock input is given in the Z direction.

Fig 7.11 Boundary condition on HAA FE model for shock simulation in ANSYS

The shock input given to the shaker node is 25 G's with duration of 0.5 ms, 1 ms, and 2 ms. The method of the transient analysis is the same as the one explained in Chapter IV, section 4.6. The nodes observed are nodes 3 and 4 (without spring). The ANSYS simulation results are given in Figs 7.12 to 7.15.

Fig 7.12 Acceleration response of node 3 (ANSYS results)

Fig 7.13 Acceleration response of node 4 (ANSYS results)

Fig 7.14 Displacement response of node 3 (ANSYS results)

Fig 7.15 Displacement response of node 4 (ANSYS results)

The displacement responses from MATLAB agree very nicely with the displacement responses from ANSYS. The acceleration responses from MATLAB also agree very nicely with the acceleration responses from ANSYS, except for the 0.5 ms shock duration. This is due to the simple model only keeping two degrees of freedom. The results from simulation in MATLAB and ANSYS show that the longer duration is the worst for the slider movement. So then for the investigation on the TVA, we will only use

2 ms shock duration. This is also agreeing with the market demand on the 2.5" drives which uses 2 ms shock duration as the shock tolerance specification.

In the next section, we will add a TVA into the simple model, and then simulate the problem subject to a shock in MATLAB. We will try to find the optimum TVA properties.

6.4 Simple Model: TVA on the Arm

We assume the bearing and HAA rigid block as a rigid body. Therefore, we can modify the simple model of Fig 7.7 into another model with the TVA added on the arm. The simple model using TVA on the arm is shown in Fig 7.16.

Fig 7.16 Simple model TVA on the arm

Using Newton's law, we can derive the equation of the motion of the model of Fig 7.16. We will first build the free body diagram, assuming $x_3 > x_2 > x_1 > y$:

Fig 7.17 Free body diagram of model in Fig 7.16

From the free body diagram of Fig 7.17, we can develop the mathematical model as below:

$$\begin{aligned} M_T \ddot{x}_3 + k_T (x_3 - x_2) &= 0 \\ M_A \ddot{x}_2 + k_A (x_2 - y) + k_S (x_2 - x_1) &= k_T (x_3 - x_2) \\ M_S \ddot{x}_1 - k_S (x_2 - x_1) &= 0 \end{aligned} \quad (7.5)$$

Transforming equation (7.1) into Laplace domain with zero initial condition:

$$\begin{aligned} s^2 X_3(s) M_T + X_3(s) k_T &= X_2(s) k_T \\ s^2 X_2(s) M_A + X_2(s) (k_A + k_S + k_T) &= X_1(s) k_S + Y(s) k_A + X_3(s) k_T \\ s^2 X_1(s) M_S + X_1(s) k_S &= X_2(s) k_S \end{aligned} \quad (7.6)$$

Substituting the first line and the last line equation (7.6) to the second line equation (7.6):

$$\begin{aligned} s^2 X_3(s) M_T + X_3(s) k_T &= X_2(s) k_T \\ \rightarrow X_3(s) &= X_2(s) \frac{k_T}{(s^2 M_T + k_T)} \\ s^2 X_1(s) M_S + X_1(s) k_S &= X_2(s) k_S \\ \rightarrow X_1(s) &= X_2(s) \frac{k_S}{(s^2 M_S + k_S)} \\ s^2 X_2(s) M_A + X_2(s) (k_A + k_S + k_T) &= X_1(s) k_S + Y(s) k_A + X_3(s) k_T \\ \rightarrow s^2 X_2(s) M_A + X_2(s) (k_A + k_S + k_T) &= X_2(s) \frac{k_S k_S}{(s^2 M_S + k_S)} + Y(s) k_A + X_2(s) \frac{k_T k_T}{(s^2 M_T + k_T)} \\ \leftrightarrow X_2(s) \left(s^2 M_A + (k_A + k_S + k_T) - \frac{k_S k_S}{(s^2 M_S + k_S)} - \frac{k_T k_T}{(s^2 M_T + k_T)} \right) &= Y(s) k_A \end{aligned}$$

$$\leftrightarrow X_2(s) = Y(s) \frac{k_A}{\left(s^2 M_A + (k_A + k_S + k_T) - \frac{k_S k_S}{(s^2 M_S + k_S)} - \frac{k_T k_T}{(s^2 M_T + k_T)} \right)} \quad (7.7)$$

So then with equation (7.7) we found the relationship between the shock input ($Y(s)$) and the three value of $X_1(s)$, $X_2(s)$, and $X_3(s)$. The m – file script of the above model is available in appendix A.5. In MATLAB, the above transfer functions are used to perform transient analysis.

The properties of the simple model of Fig 7.16 are given in Table 7.3. The TVA mass is tuned to 6% of the arm mass, and there are two TVA frequencies selected as a comparison (1450 Hz and 84 Hz).

Table 7.3 Simple model properties of Fig 7.16

	Mass (grams)	Stiffness – K (N/m)	Damping – C (Ns ² /m)
Arm	0.65	5.7737 E4	-
Suspension + slider	0.123	765.3249	-
TVA	0.039	3238.1 (1450 Hz)	-
		10.8638 (84 Hz)	-

The shock input used is 25 G's 2 ms. The suspension + slider displacement and arm displacement are observed. The MATLAB simulation results are given in Figs 7.18 to 7.20.

Fig 7.18 Slider displacement; different TVA frequency (MATLAB results)

Fig 7.19 Arm displacement; TVA frequency 1450 Hz (MATLAB results)

Fig 7.20 Slider displacement TVA frequency 1450 Hz; suspension frequency 879.78 Hz (MATLAB results)

From Fig 7.18, we can see that the TVA has no effect on the slider displacement. But we can see the TVA's effect on the arm (Fig 7.19). The arm displacement was reduced about 3.6% at the negative peak, and 21% at the positive peak of the response. We conclude that because the suspension natural frequency is far from the arm natural frequency, the TVA placed on the arm cannot give a significant effect on the slider displacement. This

conclusion is supported by result in Fig 7.20 when the suspension frequency is changed to 879.78 Hz, which is closer to the arm resonance. The slider displacement is started to being reduced at the second wave, meaning that the TVA has a greater effect on the suspension with natural frequency that is closer to the arm resonance.

Generally, we conclude that the TVA is working; but since the TVA is placed on the arm, it almost has no effect on the slider displacement. In the next section we will modify the simple model by adding the TVA on the suspension.

6.5 Simple Model: TVA on the Suspension

The simple model for the case of placing TVA on the suspension is given in Fig 7.21. The development process is the same as the simple model development in section 7.2. The model in Fig 7.21 is actually the same as the simple model in Fig 7.7. So we will use the same m-file to simulate the problem. We only need to change the mass and the stiffness value in the model. The simple model properties of Fig 7.21 are given in Table 7.4.

Table 7.4 Simple model properties of Fig 7.21

	Mass (grams)	Stiffness – K (N/m)	Damping – C (Ns ² /m)
Arm	0.65	5.7737 E4	-
Suspension + slider	0.123	765.3249	-
TVA	0.0123	73.8574 (390 Hz)	-
		3.4262 (84 Hz)	-

The TVA mass is tuned to 10% of suspension + slider mass, and there are two different TVA frequencies selected as a comparison (390 Hz and 84 Hz).

Fig 7.21 Simple model TVA on the suspension

Fig 7.22 Slider displacement; TVA placed on the suspension (MATLAB results)

The slider displacement from MATLAB simulation results are given in Fig 7.22. Fig 7.22 also shows the slider displacement when the TVA is placed only on the arm as comparison (TVA on arm has frequency of 1450 Hz).

From the results of Fig 7.22, the best reduction of the slider displacement is achieved by placing the TVA on the suspension. The TVA has no effect at the first peak of the displacement response, but the displacement response of the slider is gradually decreased over time. The TVA frequency used is close to the resonant of the suspension. In the next

section, we will simulate in ANSYS to put the TVA on the arm, and then combine the TVA on the arm and on the suspension.

6.6 TVA on HAA Simulation in ANSYS; Transient Analysis

In this section, we will first try to put the TVA on the arm. To obtain the optimum reduction from the TVA, we must put the TVA on the tip of the arm.

We start from calculating the TVA mass needed. The arm FE model is shown in Fig 7.23. The mass of total three arms obtained from ANSYS is 0.655 grams. The mass of the upper and lower arm is 0.19 grams each, while the mass of the middle arm is 0.27 grams.

Fig 7.23 FE model of vibrating part of the three arms from HAA

We assume the arm as a cantilever beam, so the effective mass of the arm is 0.23 times of the arm mass. The upper and lower arm effective mass is 0.0437 grams each; whereas the middle arm effective mass is 0.0621 grams. For the TVA placed on the arm, we will use two kinds of TVA masses. First we use 6.4% of arm effective mass, and second we use 10.45% of arm effective mass. The TVA will be placed at the tip of the arm, as shown in Fig 7.24. The TVA length and width are limited so that the TVA will not interfere with the suspension and arm movements.

Fig 7.24 TVA placement on the arm and the size limit of the TVA

The TVA dimension is explained in Fig 7.25, while the TVA frequency can be roughly calculated using the following formula:

$$f_n = \frac{1}{2\pi} \sqrt{\frac{\left(\frac{3EI_1}{L_{eff}^3} \right)}{\rho(L_2W_2T_2 + 2 \cdot 0.23L_1W_1T_1)}} \quad (7.8)$$

$$\text{Where: } L_{eff} = L_1 + 0.48L_2 \quad (7.9)$$

Fig 7.25 TVA dimension

Referring to size constraints (design envelope) of Fig 7.24 and using equation (7.8) we can determine the TVA dimensions. We will tune the TVA frequency to 1500 Hz, so the frequency ratio is 0.98. We will only calculate the TVA for the upper and lower arms, since we are not interested in observing the sliders in the middle arm. The TVA material used is Aluminum, which is the same as the arm material. The TVA is meshed using 20 noded solid 95 elements. The TVA dimensions for each mass configuration are as follows:

6.4% of effective arm mass (2.8E-3 grams)

$$W_1 = 0.4318 \text{ mm}; \quad W_2 = 4.572 \text{ mm}$$

$$L_1 = 1.016 \text{ mm}; \quad L_2 = 1.143 \text{ mm}$$

$$T_1 = 0.04064 \text{ mm}; \quad T_2 = 0.2 \text{ mm}$$

10.45% of effective arm mass (4.57E-3 grams)

$$W_1 = 0.508 \text{ mm}; \quad W_2 = 5.08 \text{ mm}$$

$$L_1 = 0.635 \text{ mm}; \quad L_2 = 1.651 \text{ mm}$$

$$T_1 = 0.04064 \text{ mm}; \quad T_2 = 0.2 \text{ mm}$$

The boundary conditions for the transient analysis are the same as in Fig 7.11. The method used for the simulation is also the same as one used in section 7.2. The shock input given to the shaker is 25 G's 2 ms. The simulation results are given only in percentage of reduction in acceleration and displacement; meaning that the acceleration and displacement of the slider nodes using TVA on the arm are compared with the acceleration and displacement of the slider nodes without TVA. The ANSYS simulation results for the case of using TVA on the arm are given in tables 7.5 and 7.6.

Table 7.5 Percentage of acceleration reduction (in %)

	6.4% mass	10.45% mass
Node 1	6	6
Node 2	14	13
Node 3	2.8	1
Node 4	14	13

Table 7.6 Percentage of displacement reduction (in %)

	6.4% mass	10.45% mass
Node 1	17	16.5
Node 2	21	20
Node 3	8	7.8
Node 4	6.9	6.7

From tables 7.5 and 7.6, we can conclude that the smaller TVA mass (6.4% mass) performs better in reducing both acceleration and displacement of the slider than TVA with larger mass. The results in tables 7.5 and 7.6 also agree with the results in Figs 7.18 and 7.20; the slider with spring, in which the suspension natural frequency is higher, has the highest acceleration and displacement reduction. Since the TVA performance is measured by the slider without the linear spring, the maximum displacement reduction using the TVA on the arm is only 8%.

To crosscheck with the result of Fig 7.19, we also check the arm displacement before and after TVA placement. The comparison of the arm tip displacement is given in Fig 7.26. Fig 7.26 shows that the arm displacement reduction is larger than the slider displacement reduction. The arm displacement reduction reaches 11%.

Fig 7.26 Arm displacement with and without TVA

The slider acceleration and displacement reduction are still not significant by using the TVA on the arm. Thus, we will try to put the TVA on the suspension as well as on the arm, to see if we can achieve reduction more than 8%.

We will use the TVA on the arm with TVA mass equal to 6.4% of the effective arm mass. The TVA on the suspension is placed at the middle of the suspension. The suspension plus slider mass are 0.041 grams. So the effective suspension + slider mass are $9.43\text{E-}3$ grams. The TVA location on the suspension is shown in Fig 7.27.

Fig 7.27 TVA placement on suspension

The TVA mass is chosen as such to have a significant effect but does not reduce the suspension natural frequency too much. From trial and errors to draw and mesh the TVA, and perform a modal analysis and transient analysis in ANSYS, we choose the TVA mass

as 3% of the suspension effective mass. The TVA frequency is then calculated to be 395 Hz (using equation (7.8)). The TVA material used is steel, the same material as the suspension. The TVA is meshed using 20 noded solid 95 elements. The TVA size is determined so it does not exceed the suspension dimension, especially the width and the length of the TVA. The FE model of TVA as well as its dimensions is given in Fig 7.28.

Fig 7.28 FE model of TVA on suspension

Before we perform the shock analysis using transient analysis in ANSYS, we first check the first bending natural frequency of the suspension after TVA placement. With the above TVA placement on the suspension, the suspension natural frequency becomes 366.12 Hz (it degrades 7.8% from 397.48 Hz). The mode shape of the 1st bending of the suspension is given in Fig 7.29.

Fig 7.29 Mode shape of the suspension after TVA placement

The simulation in ANSYS is performed using the same technique as in simulating the TVA on the arm. The shock input used is 25 G's 2 ms. Since we are only interested in the displacement reduction of the slider without spring; the TVA is only placed on the upper suspension (the slider without springs). Figs 7.30 and 7.31 show the ANSYS simulation results. Simulation results show the comparison of TVA placement on the arm only and the TVA placement on the arm plus on the suspension.

Fig 7.30 Slider displacement (nodes 3 and 4) using TVA on the arm plus on the suspension (ANSYS results)

The displacement reduction on the slider (nodes 3 and 4) reach 21% at the first wave and more than 40% at the second wave. By placing the TVA on the suspension, the displacement of the slider is decaying over time quite fast too.

The slider acceleration is given in Fig 7.31. The acceleration reduction of the slider by using TVA on the arm plus on suspension and the arm only is quite similar; 1% on node 3 and 13% on node 4.

Fig 7.30 Slider acceleration (nodes 3 and 4) using TVA on the arm plus on the suspension (ANSYS results)

Since the displacement reductions on the slider reach 21%, we predict that the shock tolerance of the HDD has been increased by 21%; since the shock tolerance is improved greatly by displacement reduction. To prove this prediction, we simulate again the HAA with TVA on the arm and suspension in ANSYS, using the same configuration as above, and then we increase the shock magnitude by 21%. Thus, the shock input used is 30 G's 2 ms. We want to find out if the shock magnitude is increased, the slider displacement will

not exceed the slider displacement without TVA placement on the arm and suspension. The slider displacement when the shock magnitude increased by 21% is given in Fig 7.31.

Fig 7.31 Slider displacement (nodes 3 and 4) using TVA on the arm plus on the suspension with 121% shock magnitude (ANSYS results)

From the result of Fig 7.31, we find out that even with 121% shock input, the slider displacement is still below the slider displacement without TVA. Therefore, the shock tolerance improvement of the HDD can be achieved by more than 21% by using TVA on the arm as well as on the suspension.

6.7 Short Conclusion

In this chapter we have created a simple model in MATLAB which resembles the FE model of HAA in ANSYS. We have used the simple model in MATLAB to perform a pre

– simulation to find the best location and the optimum properties of the TVA. The TVA with the best location and the optimum properties were drawn and meshed in ANSYS. The FE model of HAA plus TVA were then simulated using transient analysis.

We conclude from our simulation, the optimum TVA location is to put the TVA on the suspension as well as on the arm. This conclusion also meets with the conclusion in Chapter VI; that if we are dealing with multi degree of freedom system, we need to use more than one TVA with different frequency that match with the resonant of the primary system. We found that the shock tolerance of the HDD is increased by more than 21% (predicted). It means if the current HDD shock tolerance is 350 G's 2 ms, the shock tolerance after TVA placement on the arm and the suspension is predicted to be more than 423.5 G's 2 ms.

According to Seagate, adding the TVA to the arm and as well as to the suspension of the current 2.5" HDD product is a great manufacturing effort since the TVA size is too small and they need to redesign the arm and the suspension to balance its mass (since new mass are added), while the shock tolerance improvement is only 21%. Thus, adding the TVA into the current 2.5" Seagate HDD is not a feasible solution. If Seagate makes the product from the scratch, it is possible to add the TVA to the arm and to the suspension. Therefore from this study, we suggest Seagate to embed the TVA to the arm and suspension of the new HDD designs.

Chapter VIII

IMPACT OF HCL (HIGH CAPACITY LAMINATE) BEARING and ARM STIFFNESS INCREASE ON HDD SHOCK TOLERANCE

8.1 Introduction

In this chapter we will try two methods to increase the shock tolerance of the 2.5" Seagate HDD. The first method is to replace the tolerance ring of HAA with a HCL bearing, to reduce the overall vertical bearing shaft stiffness. The second method is to increase the arm stiffness. We will use the simple model in MATLAB developed in Chapter VII to find the proper HCL ring properties and also to find the effect of increasing the arm stiffness. After we find the proper properties of the HCL ring and also the arm stiffness, we will perform the simulation in ANSYS to verify the simple model simulation results.

5.7 HCL Bearing: Modeling the Problem in MATLAB

We will first examine the HAA FE model in ANSYS to build the simple model in MATLAB. The HAA viewed from top is given in Fig 8.1.

Fig 8.1 HAA viewed from top

The tolerance ring is shown in Fig 8.1. The tolerance ring is a metal ring in between the bearing shaft and the HAA rigid block. We will replace the tolerance ring with HCL which is made from elastomer. With HCL replacing the tolerance ring, the vertical bearing stiffness can be reduced to reduce the overall transmissibility to the slider.

The simple model in MATLAB to simulate replacement of the tolerance ring with HCL is given in Fig 8.2.

Fig 8.2 Simple model to simulate HCL replacing tolerance ring

The moving base now consists of the bearing shaft plus the HDD base plate. Between the moving base and the HAA rigid block is the tolerance ring in which the stiffness can be tuned in the simple model. Since the mass of the tolerance ring is very small compared to the HAA rigid block, the tolerance ring mass can be neglected. So the simple model of Fig 8.2 is actually the same simple model of Fig 7.7. We only need to change the HAA rigid block mass and tune the tolerance ring stiffness. The arm and the suspension properties are the same as mentioned in Table 7.2.

The shock duration used is 2 ms. The shock magnitude can be selected arbitrarily, since we will compare the slider response before and after changing the tolerance ring stiffness (replacing the tolerance ring with HCL).

Before replacing the tolerance ring with HCL, the tolerance ring stiffness in the simple model is very high (using the same properties in table 7.2). The shock duration used is 2 ms, which is equal to 250 Hz of input frequency. According to section 3.3, Fig 3.16, the system natural frequency must be tuned to be lower than the input frequency. By tuning the system natural frequency into half of the input frequency, the transmissibility becomes 0.9. We can extend the result in Fig 3.3 by creating a transmissibility curve for single DOF system, base excitation problem. This transmissibility curve is given in Fig 8.3.

Fig 8.3 Single DOF base excitation problem transmissibility curve

Refer to Fig 8.3, by reducing the tolerance ring stiffness until the first natural frequency of the simple model of Fig 8.2 becomes 125 Hz (half of 250 Hz, the shock duration frequency), the transmissibility from the moving base to the HAA rigid block can be reduced by 10%. Although the transmissibility from the moving base to the HAA rigid block is reduced by only 10%, the slider acceleration is reduced significantly (see Fig 8.4). We will simulate the problem in MATLAB using shock input of 1000 G's 2 ms. The simple model properties after reducing the tolerance ring stiffness are given in Table 8.1. The slider acceleration and displacement responses before and after reducing the

tolerance ring stiffness is given in Fig 8.4. The first natural frequency of the simple model of Fig 8.2 is now become 125 Hz.

Table 8.1 Simple model properties of Fig 8.2

	Mass (grams)	Stiffness – K (N/m)	Damping – C (Ns ² /m)
Moving Base	5E6		
HAA rigid block + ring	2.5	1.5421 E3	0
Arm	0.65	5.7737 E4	-
Suspension + slider	0.123	765.3249	-

Fig 8.4 Slider/ suspension acceleration and displacement response; before and after reducing tolerance ring stiffness

The acceleration of the slider is reduced significantly by 33%, while the displacement of the slider relative to the moving base/ shaker is increased by 4 times. With the increase in the slider displacement, the slider and the arm may hit the disk, since we keep the disk spindle base as rigid.

To make this idea work (reducing the tolerance ring stiffness), then we also have to reduce the disk spindle base stiffness. This idea is according to Fig 1.5, and we redraw it again in Fig 8.5.

Fig 8.5 Elastomer under the pivot arm base and disk spindle base

By placing the elastomer at both locations so HAA rigid block and the disk spindle base have the same vertical natural frequency, the HAA and the disk will move together. Thus, the slider displacement can be referred to the HAA rigid block displacement.

To investigate the disk movement after placing the elastomer under the disk spindle base, we create another model which derived from the model of Fig 8.2 in Fig 8.6. The simple model properties of Fig 8.6 are given in Table 8.2.

Table 8.2 Simple model properties of Fig 8.2

	Mass (grams)	Stiffness – K (N/m)	Damping – C (Ns ² /m)
Moving Base	5E6		
HAA rigid block + ring/ elastomer	2.5	2.4674 E8 / 1.5421 E3	0/ 0.3927
Arm	0.65	5.7737 E4	-
Suspension + slider	0.123	765.3249	-
Spindle/ elastomer	2.5	2.4674 E8 / 1.5421 E3	0/ 0.3927
Disk	0.65	2.5661 E4	-

Note: the bolded stiffness value is the elastomer stiffness when the tolerance ring stiffness is very high; the bolded damping value is the elastomer damping for 10% damping ratio case

Fig 8.6 Simple model including the spindle and the disk

The spindle and the disk masses are tuned arbitrarily, since we only want to know the difference between before and after elastomer placement. But we set the disk natural frequency according to Seagate original data.

Fig 8.6 shows the simple model, the disk spindle base and the disk are added to the model, and also the additional spring model which is the elastomer. The disk spindle base model is actually the same model of the simple model of Fig 8.2, but the slider mass and the suspension spring model is eliminated. According to Seagate, the first natural frequency of the disk is 1000 Hz, while the first natural frequency of the arm is 1527.8 Hz. In the real application, the moving base of the disk and the HAA must be coupled so they will move together (same phase) whenever a shock or disturbance comes to the HDD.

First we will simulate the simple model in Fig 8.6 with very high elastomer stiffness, so the condition is before elastomer placement. The shock input used is 1000 G's 2 ms, given to the moving base. The arm displacement relative to the HAA rigid block and the disk displacement relative to the spindle base are given in Fig 8.7.

Fig 8.7 Arm and disk relative displacement before reducing elastomer stiffness

The elastomer stiffness in simple model of Fig 8.6 is then reduced until the first natural frequency of each model (HAA rigid block and spindle base) become 125 Hz. The simple model is simulated with the same shock input. The arm displacement relative to the HAA rigid block and the disk displacement relative to the spindle base after reducing the elastomer stiffness are given in Fig 8.8.

Fig 8.8 Arm and disk relative displacement after reducing elastomer stiffness

Previously in Fig 8.7, the largest displacement difference is 0.177 mm, while in Fig 8.8 the largest displacement different is 0.133 mm. By placing the elastomer at the base of the pivot arm and disk spindle, the largest relative displacement between the arm and the disk

is reduced. Thus, instead of colliding with each other, the relative displacement between the disk and the arm is reduced.

If we add a sufficient damping value to the elastomer, for example 10% of damping ratio, the arm displacement relative to the HAA rigid block and the disk displacement relative to the spindle base will be different. The simple model simulation result by adding damping to the elastomer is given in Fig 8.9. The arm and the disk relative displacement are reduced and also decayed over time.

Fig 8.9 Arm and disk relative displacement with elastomer damping ratio 10%

With this study, the slider displacement is referred to the HAA rigid block. Thus, the slider will not hit the disk like has been depicted in the bottom picture of Fig 8.4.

In the next section, we will repeat all the analyses done so far but using ANSYS.

5.8 HCL Bearing: Transient Analysis in ANSYS

Fig 8.10 Spring element in FE model of HAA in ANSYS

In the FE model of the HAA that was given to us by Seagate, the tolerance ring was modeled as bunch of very stiff spring, see Fig 8.10. To simulate the effect of an elastomer at the base of the pivot shaft, all we have to do is to reduce the spring element stiffness until the vertical natural frequency of the whole HAA reach near 125 Hz. To check the mode shape and the eigenvalue, we perform a modal analysis in ANSYS. The 1st mode shape of the HAA after reducing the spring element stiffness is given in Fig 8.11.

The first vertical natural frequency of the HAA is 124 Hz, so it is close to 125 Hz. The shock analysis is performed using transient analysis. The method used is the same as used in Chapter VII. The shock input given to the shaker node is 25 G's 2 ms. The slider acceleration and displacement before and after elastomer placement under the pivot base are given in Figs 8.11 and 8.12. The transient analysis is performed until 3 ms only. We know that from Fig 8.4, the peak acceleration and displacement of the slider is at 3 ms, so the simulation time is sufficient to find out the maximum acceleration and displacement responses of the slider.

Fig 8.10 1st mode shape of HAA after reducing spring stiffness

Fig 8.11 Slider acceleration response before and after elastomer placement under the pivot base

The slider acceleration are reduced greatly up to 50% (on node 4) and 36% (on node 3). The acceleration reduction on node 3 is quite close to the result obtained from the simple model in MATLAB (33% reduced).

By placing the elastomer under the pivot base and also under the spindle disk base, the slider displacement can be measured relative to the HAA rigid block, since the HAA rigid block and the spindle disk base are moving together. In Fig 8.12, we also show the displacement response using TVA placed at the suspension.

Fig 8.12 Slider displacement response before and after elastomer placement under the pivot base

The slider displacement reduction values by placing the elastomer under the pivot base are almost the same with the slider displacement reduction by placing the TVA on the arm and the suspension, which is around 20%.

Following the method used in sections 3.11 and 5.5; we can predict the shock tolerance improvement by comparing the slider displacement before and after elastomer placement under the pivot arm and disk spindle base. We will use the simple model of Fig 8.2. For the configuration before elastomer placement, the shock input given to the moving base is 350 G's 2 ms; which is the current shock tolerance of 2.5" Seagate HDD. Using the simple model, it is found that after elastomer placement, the shock tolerance is increased up to 610 G's 2 ms; the shock tolerance improvement reaches almost 75%. This improvement is almost the same with the improvement using external shock isolator (cantilever beam) in Chapter III. The slider displacement obtained from the simulation in MATLAB using simple model; before and after shock isolator placement; is given in Fig 8.13.

Fig 8.13 Slider displacement response before and after elastomer placement (MATLAB results)

Although the displacement reduction of the slider by using TVA on arm and suspension and using elastomer under the pivot base and disk spindle base is quite close, the shock tolerance improvement by using elastomer under the pivot base and disk spindle base is far greater than using TVA on arm and suspension. Acceleration of the slider by using elastomer under the pivot base and disk spindle base is reduced greatly by 50%, while by using the TVA on arm and suspension the acceleration reduction is only 13%. So the

prediction of shock tolerance improvement is affected by both acceleration and displacement reduction.

5.9 Increasing the Arm Stiffness: Modeling the Problem in MATLAB

To simulate the effect of increasing the arm stiffness on the slider acceleration and displacement, we will use the simple model of Fig 7.7. The simple model is redrawn in Fig 8.14.

Fig 8.14 Simple model to simulate increasing the arm stiffness

The shock duration given to the simple model of Fig 8.14 is 2 ms. In this simple model, the bearing stiffness is tuned to be very high. According to Fig 8.3, the transmissibility from the moving base to the HAA rigid block should be 1, since the HAA rigid block natural frequency is assumed very high. The current arm natural frequency is 1527.8 Hz. Thus, according to Fig 8.3 the transmissibility from the moving base to the arm is 1.15. If we can increase the arm stiffness, the transmissibility to the arm can be reduced, then also reducing the transmissibility to the slider.

We will try to increase the arm stiffness so the arm natural frequency becomes 3000 Hz (two times larger). The simple model properties of Fig 8.14 are given in Table 8.3.

Table 8.3 Simple model properties of Fig 8.2

	Mass (grams)	Stiffness – K (N/m)	Damping – C (Ns ² /m)
Moving Base	5E6		
HAA rigid block + ring	2.5	2.4674 E8	0
Arm	0.65	5.7737 E4 / 2.3095 E5	-
Suspension + slider	0.123	765.3249	-

Note: the bolded stiffness value is the arm stiffness before it is increased

The simulation result of this simple model is given in Fig 8.15. Fig 8.15 shows the arm acceleration and the slider acceleration before and after increasing the arm stiffness.

Fig 8.15 Arm and slider acceleration before and after increasing arm stiffness

The acceleration reduction on the slider is not significant (only 4.5%). This is due to the transmissibility of the arm being close to 1, so increasing the arm natural frequency by two times cannot reduce the transmissibility to the slider too much.

Fig 8.16 Arm and slider acceleration before and after increasing arm stiffness; using 0.5 ms shock duration

The result will be different if for example the shock duration used is 0.5 ms. We tried the shock duration of 0.5 ms and show the acceleration of the arm and the slider before and

after increasing the arm stiffness in Fig 8.16. The acceleration reduction of the slider reaches 15%.

We will verify the result from the simple model using transient analysis in ANSYS in the next section.

5.10 Increasing the Arm Stiffness: Transient Analysis in ANSYS

To increase the arm stiffness in the FE model of HAA, we will increase the Young's modulus of the arm. By using approximation formula of 1st natural frequency of cantilever beam:

$$\omega_n = \sqrt{\frac{3EI}{L^3 M_{beam}}} \quad (8.1)$$

Increasing the Young's modulus by 4 times can increase the arm natural frequency by 2 times. The arm natural frequency is then verified by performing modal analysis. The mode shape of the HAA obtained from modal analysis in ANSYS is given in Fig 8.17.

Fig 8.17 Arm mode shape after increasing the arm Young's modulus by 4 times

By modal analysis, the arm natural frequency now becomes 3016 Hz, which is almost two times of 1527.8 Hz. So then the transient analysis is performed for the HAA using the same method as used in the previous section (section 8.3). The shock input given to the shaker node is 25 G's 2 ms. The transient analysis is performed until 2.05 ms only, to save simulation time. 2.05 ms is sufficient to find out the maximum acceleration and displacement responses of the slider.

The slider acceleration and displacement before and after increasing the arm stiffness are given in Figs 8.18 and 8.19. The slider acceleration is reduced up to 18% on the node 4, and only 5.8% on the node 3. Same with the result in section 8.3, the response of node 3 from simple model simulation in MATLAB and transient analysis in ANSYS are quite close.

Fig 8.18 Acceleration responses of the slider before and after increasing the arm stiffness

Fig 8.19 Displacement responses of the slider before and after increasing the arm stiffness

The slider displacement reduction reaches 10.5% (on both nodes). This reduction is larger than if we use the TVA on arm only.

In reality, to increase the arm stiffness we have to increase the arm thickness. According to equation (8.1), we have to increase the arm thickness by 1.58 times. This is to increase the second moment of inertia (I) to become 4 times larger. Increasing the arm thickness by 1.58 times will increase the HAA mass by 13%. This is obtained by the calculation below:

* HAA mass without three arms : 2.245 grams

* Three arm mass : 0.655 grams

→ Arm thickness increased by 1.58 times, so the arm mass becomes 1.0349 grams

* The total HAA mass now : $2.245 + 1.0349 = 3.28$ grams

* The original HAA mass : 2.9 grams

* HAA mass increased by
$$: \frac{3.28 - 2.9}{2.9} \times 100\% = 13\%$$

5.11 Short Conclusion

Changing the tolerance ring by HCL causes decrease in HAA acceleration but increase in HAA displacement, which is not good. The elastomer must be placed under the pivot arm base as well as under the disk spindle base. With placing the elastomer under the pivot arm base and under the disk spindle base, the slider acceleration is reduced by 50% while the slider displacement is reduced by 20%. The shock tolerance improvement is simulated and it can reach 75%, which is very significant. Despite of its significant performance, this idea is not practical to be implemented to the existing 2.5" HDD, since there is no design envelope to put the elastomer and also give the HAA and the disk enough space to move.

The result of increasing the arm thickness was not as significant as using elastomer under pivot base and spindle base, except for shorter shock duration such as 0.5 ms. This idea has greater acceleration and displacement reduction than the idea of placing TVA on the arm only. This study on increasing the arm stiffness also concludes that for any HDD, the arm natural frequency should be higher than the shock input frequency. The arm natural frequency of 1527.8 Hz is good enough for a shock input frequency of 250 Hz (2 ms shock duration). We did not simulate the HAA in ANSYS to find out the shock tolerance improvement since the displacement reduction by increasing the arm stiffness is only 10.5%.

Chapter IX

CONCLUSIONS and RECOMMENDATIONS

9.1 Conclusions

In Chapter III, we created a dynamic model of an external shock isolation system for small form factor HDDs using a cantilever beam as the shock isolator. We found that for mobile applications the cantilever beam must be made from composite material. With composite material, the first bending natural frequency of the system (the HDD plus cantilever beam) can be made to be far away from the second elastic natural frequency which is the torsional modes. This is done so that the shock input frequency is in between the resonance of the system. By using the cantilever beam plus four mounts, the shock tolerance of the ST1 has been increased by 77% (predicted). Chapter III has yielded three things:

1. Dynamic model of a cantilever beam system as an external shock isolator for HDDs;
2. From the case study, we have yielded a design of cantilever beam system as a shock isolator for HDDs in mobile application (particularly 1" form factor);
3. Indirectly, the steps used in the analysis and simulation can be considered as a design guideline to design an external shock isolator for any instruments (not only for HDDs).

In Chapters IV and V, we have investigated the use of elastomer layers/ elastomeric mounts as an external shock isolator for ST1. Elastomer layers appear relatively simpler than cantilever beam system discussed in Chapter III, though it needs greater height to gain the same performance as the cantilever beam system to reduce the shock. Since the design envelope is very small, we conclude that the elastomer layers/ elastomeric mounts discussed in Chapters IV and V are not able to protect the ST1.

In Chapter VI, we have investigated the use of TVA on HAA of the ST1 as a shock isolator. We found that the TVA installed on the HAA works to reduce the risk of 'head slap'. The risk of 'head slap' is reduced due to the fact that the head/ slider acceleration response and displacement response are reduced with the TVA installment. The

acceleration of the slider is reduced by 40%, while only 8.75% displacement reduction is achieved.

In Chapter VII, we have investigated the use of TVA on HAA of 2.5" Seagate HDD. We found from our simulation that the best location of the TVA is on the suspension plus on the arm. The shock tolerance of the HDD can be increased by 20% (predicted) by installing the TVA on both locations. This idea is not practical to be applied to the existing 2.5" HDD product since the arm and the suspension design need to be changed to add a new mass, but on new HDD designs, TVAs can be implemented.

In Chapter VIII, we have investigated the use of elastomer to increase the shock tolerance of the 2.5" Seagate HDD. The elastomer is placed under the pivot arm and the disk spindle base. This idea can increase the shock tolerance of the HDD by 75%, but this idea is not practical to be implemented to the existing 2.5" HDD because there is no design envelope to put the elastomer inside the HDD. If it is possible to increase the thickness of the future HDD design, this idea can then be implemented.

Increasing the arm stiffness has also been investigated in Chapter VIII. By increasing the arm stiffness by 4 times, the shock tolerance of the HDD for shock duration 2 ms is increased by 10.5% (predicted). But then this idea is not practical since the arm needs to be redesigned.

To summarize the results, we made a table of various shock isolators that has been attempted to each form factor HDDs. Table 9.1 shows the shock isolator attempted for the ST1, and table 9.2 shows the shock isolator attempted for 2.5" Seagate HDD.

Table 9.1 Shock tolerance improvements for various methods attempted for ST1 (current shock tolerance in operating condition is 175 G's 2 ms)

	Maximum shock tolerance (predicted)	Shock tolerance improvement (%)
Cantilever beam + 4 mounts	1050 G's 0.5 ms 540 G's 1 ms 310 G's 2 ms	77
Elastomer layers under the base plate	n/a	null
Elastomeric mounts outside the ST1 case	2600 G's 0.5 ms	400 (<i>the elastomeric</i>

	1300 G's 1 ms 680 G's 2 ms	<i>mounts needed is currently not available)</i>
TVA placed on the suspension (HAA)	n/a	10 (only for 0.5 ms shock duration)

Table 9.2 Shock tolerance improvements for various methods attempted for 2.5" Seagate HDD
(current shock tolerance in operating condition is 350 G's 2 ms)

	Maximum shock tolerance (predicted)	Shock tolerance improvement (%)
TVA placed on the arm only	n/a	8 (d)
TVA placed on the arm plus on the suspension	423.5 G's 2 ms	21
Enhancing the pivot arm base and the disk spindle base using elastomer	610 G's 2 ms	75
Increasing the arm stiffness	n/a	10.5 (d)

Notes:

- (d) : displacement reduction only
n/a : data not available
null : no improvement

We have fulfilled all of our objectives. We have tried to put the shock isolator on the HDDs, either external or internal shock isolator, on several places on the HDD. We have designed several shock isolators that will be suggested to Seagate for their future HDD development.

9.2 Recommendations

From all of our study on shock isolator for HDD (1" and 2.5" form factor), the feasible/practical shock isolator for the existing HDD product is the external shock isolator. External shock isolator has a very high shock tolerance improvement and Seagate does not need to redesign the HDD. For the future HDD product, we recommend Seagate to change the HDD design. Below are our recommendations for future HDD design:

- add the TVA into the arm and the suspension design, so it embedded already in the design;

- increase the HAA structure's stiffness: arm stiffness and if possible the suspension stiffness, this is to reduce the overall transmissibility to the slider;
- if it is possible to increase the HDD thickness, add the elastomer under the pivot arm and disk spindle base to reduce transmissibility to the slider;

All of the shock tolerance improvements presented in this thesis is only prediction by observing the slider displacement. A thorough simulation using the real air bearing model (to model the operating condition of the HDD) is encouraged for future students to find the real shock tolerance improvement of the HDD.

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APPENDIX A

MATLAB COMMAND

A.1 M-file Routine to Simulate Shock Response of a Single DOF Base Excitation

Problem (Base Response)

%This program is created to obtain the velocity and displacement of the base as an input
%for the single DOF system.

```
clear all
clc
clf
% m=18E-3;
% fn=100; % (Hz)
% wn=2*pi*fn;
% k=wn^2*m;
% zeta=0.5;
% c=zeta*2*m*wn;
M=1;

t=0:0.00001:0.01;
Max_G=1000;
Fmax=Max_G*9.81*M;
pulse_width=5E-4;
period=2*pulse_width;
wf=2*pi/period;
Force_input=Fmax*sin(wf*t);

% num=[m c k];
% den1=[(m*M) (c*(M+m)) (k*(m+M)+c^2) (2*k*c-c) (k^2-k)];
% den2=[(m*M) (c*(M+m)) (k*(m+M)+c^2) (2*k*c-c) (k^2-k) 0];
num=[1]
den1=[M 0 0]
den2=[M 0]
den3=[M]
sys1=tf(num,den1)
sys2=tf(num,den1)
sys2.InputDelay=pulse_width;
ydisp1=lsim(sys1,Force_input,t);
ydisp2=lsim(sys2,Force_input,t);
ydisp=ydisp1+ydisp2;
plot(t,ydisp)
xlabel('Time(s)')
ylabel('Displacement (m)')

pause

sys3=tf(num,den2)
sys4=tf(num,den2)
sys4.InputDelay=pulse_width;
yvelo1=lsim(sys3,Force_input,t);
yvelo2=lsim(sys4,Force_input,t);
yvelo=yvelo1+yvelo2;
plot(t,yvelo)
```



```

xlabel('Time(s)')
ylabel('Velocity (m/s)')

pause

sys5=tf(num,den3)
sys6=tf(num,den3)
sys6.InputDelay=pulse_width;
yaccel1=lsim(sys5,Force_input,t);
yaccel2=lsim(sys6,Force_input,t);
yaccel=(yaccel1+yaccel2)/9.81;
plot(t,yaccel)
xlabel('Time(s)')
ylabel('Acceleration (g)')

save y_input ydisp yvelo yaccel

```

A.2 M-file Routine to Simulate Shock Response of a Single DOF Base Excitation

Problem (HDD Response)

```

%Single DOF System to attempt shock input connected to DOF_TRY3.m
clc
clf
clear all
%Define the system
m=18E-3; %mass of HDD
fn=1000; % (Hz) %set the natural freq of the system
wn=2*pi*fn;
k=wn^2*m;
critical_damping=2*m*wn;
zeta=0.0; %set the damping factor of the system
c=critical_damping*zeta;

t=0:0.00001:0.01;

%Load the input displacement and velocity
load y_input

%calculate the acceleration response due to displacement input
numdisp=[k 0 0];
dendisp=[m c k];
sys1=tf(numdisp,dendisp);
accel_1=lsim(sys1,ydisp,t);

%calculate the acceleration response due to velocity input
numvelo=[c 0 0];
denvelo=[m c k];
sys2=tf(numvelo,denvelo);
accel_2=lsim(sys2,yvelo,t);

accel=(accel_1+accel_2)/9.81;
%subplot(2,1,2),
plot(t,accel,t,yaccel)
xlabel('time [s]'),ylabel('Acceleration [g]'),title('Output Acceleration')

pause

%calculate the displacement response due to displacement input
numdisp_disp=[k];
dendisp_disp=[m c k];
sys1=tf(numdisp_disp,dendisp_disp);
disp_1=lsim(sys1,ydisp,t);

%calculate the displacement response due to velocity input
numvelo_disp=[c];

```

```
denvelo_disp=[m c k];
sys2=tf(numvelo_disp,denvelo_disp);
disp_2=lsim(sys2,yvelo,t);

disp=(disp_1+disp_2-ydisp)*1000;

%subplot(2,1,1),
plot(t,disp)
xlabel('time [s]'),ylabel('Displacement [mm]'),title('Output Displacement')

save accel_response_one_DOF accel %save the response to .mat file to find the exact maximum displacement and acceleration
load accel_response_one_DOF
```

A.3 M-file Routine to Construct State Space Equation of a Cantilever Beam

```
load mode_shapes_and_natural_freq_alu_1mm_60mm;

shaker_mass = 5; % ton from ANSYS run

% define the number of degrees of freedom and number of modes from size of modal matrix
[numdof,num_modes_total] = size(evr);

% define number of input and output;

num_input = 5; % shaker, front, middle, and behind node on the slider

num_output = numdof; % point of interest

% define rows for shaker node, front, middle, and behind node on the slider

input_node(1) = 1; %Fz to wall

input_node(2) = 2; %Fz ...

input_node(3) = 3; %Fz ...

input_node(4) = 4; %Fz ...

input_node(5) = 5; %Fz ...

xn = evr;

% calculate the dc amplitude of the displacement of each mode by
% multiplying the forcing function row of the eigenvector by the output row

omega2 = (2*pi*freqvec)'.^2; % convert to radians and square

num_modes_used = num_modes_total;

zeta = input('enter value for damping, .02 is 2% of critical (default) ... ');

if (isempty(zeta))
    zeta = .02;
end

% all modes included model, use original order

xnnew = xn(:,(1:num_modes_total));

freqnew = freqvec((1:num_modes_total));

% define variables for all modes included system matrix, a

w = freqnew*2*pi; % frequencies in rad/sec

w2 = w.^2;

zw = 2*zeta*w;

% define size of system matrix

asize = 2*num_modes_total;
```

```

disp(' ');
disp(' ');
disp(['size of system matrix a is ',num2str(aside)]);
%      setup all modes included "a" matrix, system matrix

a = zeros(aside);

for col = 2:2:aside

row = col-1;

a(row,col) = 1;

end

for col = 1:2:aside

row = col+1;

a(row,col) = -w2((col+1)/2);

end

for col = 2:2:aside

row = col;

a(row,col) = -zw(col/2);

end

%      setup input matrix b, state space forcing function in principal coordinates

for i=1:num_input

%      f_physical is the vector of physical force

%      zeros at each output DOF and input force at the input DOF

if (i==1)

f_physical = zeros(numdof,num_input); %      start out with zeros

%      b is the vector of forces in principal coordinates, state space form

b = zeros(2*num_modes_total,num_input);

f_physical(input_node(1),1) = shaker_mass*9.81*1000; %      input force at shaker, 1g

else

f_physical(input_node(i),i) = 1;

end

%      now setup the principal force vector

%      f_principal is the vector of forces in principal coordinates

f_principal(:,i) = xnnew'*f_physical(:,i);

%      b is the vector of forces in principal coordinates, state space form

for cnt = 1:num_modes_total

b(2*cnt,i) = f_principal(cnt,i);

end

end

%      setup cdisp and cvel, padded xn matrices to give the displacement and velocity
%      vectors in physical coordinates
%      cdisp and cvel each have numdof rows and alternating columns consisting of columns

```

```
% of xnnew and zeros to give total columns equal to the number of states

%      all modes included cdisp and cvel
for col = 1:2:2*length(freqnew)
    for row = 1:numdof
        cdisp(row,col) = xnnew(row,ceil(col/2));
        cvel(row,col) = 0;
    end
end

for col = 2:2:2*length(freqnew)
    for row = 1:numdof
        cdisp(row,col) = 0;
        cvel(row,col) = xnnew(row,col/2);
    end
end

for i=1:num_output
    c(i,:)=cdisp(i,:);
end

%      define output
d = [0]; %

%      define tip force state space system with the "ss" command
sysforce_beam = ss(a,b,c,d);

%      use "bode" command to generate magnitude/phase vectors
save state_space_of_beam sysforce_beam

clear all

load state_space_of_beam
```

A.4 M-file Routine to Simulate Four Degrees of Freedom System Base excitation Problem

```
clc
clear all

%a program to simulate four degrees of freedom system base excitation problem,

Fa=50000; %bearing frequency in Hz%
Fs=1500; %Arm frequency in Hz%
Ft2=397; %suspension frequency in Hz%

ma=2.5e-3; %HAA rigid block mass in kgrams%
ms=0.65e-3; %arm mass in kgrams%
mt2=0.123-3; %suspension mass in kg
M=5000; %base mass in kg

ka=((2*pi*Fa)*(2*pi*Fa))*ma %HCL bearing stiffness in N/m%
ks=((2*pi*Fs)*(2*pi*Fs))*ms %arm stiffness in N/m%
kt2=((2*pi*Ft2)*(2*pi*Ft2))*mt2%suspension stiffness in N/m
```

```
%eigenvalue extraction
MM=[ma 0 0;0 ms 0;0 0 mt2];
KK=[ka+ks -ks 0;-ks (kt2+ks) -kt2;0 -kt2 kt2];

E=inv(MM)*KK;
[u,w]=eig(E)
Freq=(sqrt(w))/(2*pi)

zeta_suspension=0.0;
zeta_HCL=0.3;
zeta_arm=0.0;

ct2=zeta_suspension*2*(sqrt(kt2*mt2)); %TVA on suspension damping value%
ca=zeta_HCL*2*(sqrt(ka*ma));
cs=zeta_arm*2*(sqrt(ks*ms));

%t=0:0.00000001:0.01;

%SHOCK INPUT

load base_value

Max_G=310; % in G
Fmax=Max_G*9.81*M;
Freq=250; % shock frequency
wf=2*pi*Freq;
period=1/Freq;
pulse_width=period/2;

Force_input=Fmax*sin(wf*t);

%num=[m c k];
%den1=[(m*M) (c*(M+m)) (k*(m+M)+c^2) (2*k*c-c) (k^2-k)];
%den2=[(m*M) (c*(M+m)) (k*(m+M)+c^2) (2*k*c-c) (k^2-k) 0];
num=[1];
den1=[M 0 0];
den2=[M 0];
den3=[M];
sys1=tf(num,den1);
sys2=tf(num,den1);
sys2.InputDelay=pulse_width;
ydisp1=lsim(sys1,Force_input,t);
ydisp2=lsim(sys2,Force_input,t);
ydisp=ydisp1+ydisp2;
%plot(t,ydisp1,t,ydisp2,t,ydisp,'k')
%xlabel('Time(s)')
%ylabel('Displacement (m)')

%pause

sys3=tf(num,den2);
sys4=tf(num,den2);
sys4.InputDelay=pulse_width;
yvelo1=lsim(sys3,Force_input,t);
yvelo2=lsim(sys4,Force_input,t);
yvelo=yvelo1+yvelo2;
%plot(t,yvelo,'k')
%xlabel('Time(s)')
%ylabel('Velocity (m/s)')

%pause
sys5=tf(num,den3);
sys6=tf(num,den3);
sys6.InputDelay=pulse_width;
yaccel1=lsim(sys5,Force_input,t);
yaccel2=lsim(sys6,Force_input,t);
yaccel1=(yaccel1+yaccel2)/9.81;
yaccel=(yaccel1+yaccel2)/9.81;
plot(t,yaccel,'k')
xlabel('Time(s)')
ylabel('Acceleration (G)')
title('Half Sine Input')

%END OF SHOCK INPUT
```

pause

```
%Q/R --> transfer function
Q_I=[(ms*mt2*ca) (ca*(ms*ct2+(mt2*(cs+ct2)))+(ka*ms*mt2)) (ca*(ms*kt2+cs*ct2+(mt2*(ks+kt2)))+(ka*(ms*ct2+mt2*(cs+ct2))))
(ca*ct2*ks+(ka*(ms*kt2+cs*ct2+(mt2*(ks+kt2)))) (ca*kt2*ks+ka*ct2*ks) (ka*kt2*ks)];
Q_III=[(ms*mt2*ca) (ca*(ms*ct2+(mt2*(cs+ct2)))+(ka*ms*mt2))
(ca*(ms*kt2+cs*ct2+(mt2*(ks+kt2)))+(ka*(ms*ct2+mt2*(cs+ct2)))) (ca*ct2*ks+(ka*(ms*kt2+cs*ct2+(mt2*(ks+kt2))))
(ca*kt2*ks+ka*ct2*ks) (ka*kt2*ks) 0 0];
R=[(ma*ms*mt2) (ma*ms*ct2+ma*mt2*(cs+ct2)+ms*mt2*(ca+cs))
(ma*ms*kt2+ma*cs*ct2+ma*mt2*(ks+kt2)+((ca+cs)*(ms*ct2+(mt2*(cs*ct2))))+ms*mt2*(ka+ks)-cs*cs*mt2)
(ma*ct2*ks+((ca+cs)*(ms*kt2+cs*ct2+(mt2*(ks+kt2))))+((ka+ks)*(ms*ct2+(mt2*(cs+ct2))))-cs*cs*ct2-2*cs*ks*mt2)
((ca+cs)*ct2*ks+ma*kt2*ks+((ka+ks)*(ms*kt2+cs*ct2+(mt2*(ks+kt2))))-cs*cs*kt2-2*cs*ct2*ks-ks*ks*mt2)
(ca*kt2*ks+ka*ks*ct2-cs*ks*kt2) (ka*kt2*ks)];
```

```
%solution for HCL bearing displacement
sys_arm_TVA2=tf(Q_I,R);
disp_arm_TVA2=lsim(sys_arm_TVA2,ydisp,t);
disp_HCL=(disp_arm_TVA2-ydisp)*1000;

plot(t,disp_HCL,'r',t,disp_HCL_base,'k');
xlabel('Time(s)')
ylabel('Displacement of HCL bearing (mm)')
title('HAA Rigid Block Displacement')
legend('Reduced ring stiffness','current configuration','Location','NorthEastOutside')
```

pause

```
solution for HCL bearing acceleration
sys_accel_HCL=tf(Q_III,R);
accel_HCL1=lsim(sys_accel_HCL,ydisp,t);
accel_HCL=(lsim(sys_accel_HCL,ydisp,t))/9.81;

plot(t,accel_HCL,'r',t,accel_HCL_base,'k');
xlabel('Time(s)')
ylabel('Acceleration of HAA rigid block (G)')
title('HAA Rigid Block Acceleration')
legend('Reduced ring stiffness','current configuration','Location','NorthEastOutside')
```

pause

```
%transfer function for arm displacement
S_I=[(mt2*cs) (cs*ct2+mt2*ks) (cs*kt2+ks*ct2) (ks*kt2)];
S_III=[(mt2*cs) (cs*ct2+mt2*ks) (cs*kt2+ks*ct2) (ks*kt2) 0 0];
TI=[(ms*mt2) (ms*ct2+(mt2*(cs+ct2))) (ms*kt2+cs*ct2+(mt2*(ks+kt2))) (ct2*ks) (kt2*ks)];

sys_suspension_TVA2=tf(S_I,TI);
disp_suspension_TVA2=lsim(sys_suspension_TVA2,disp_arm_TVA2,t);
disp_susp_TVA2=(disp_suspension_TVA2-ydisp)*1000;

plot(t,disp_susp_TVA2,'r',t,disp_arm_base,'k');
xlabel('Time(s)')
ylabel('Displacement of Arm tip (mm)')
title('Arm Displacement')
legend('Reduced ring stiffness','current configuration','Location','NorthEastOutside')
```

pause

```
sys_accel_arm=tf(S_I,TI);
accel_arm1=lsim(sys_accel_arm,accel_HCL1,t);
accel_arm=(lsim(sys_accel_arm,accel_HCL1,t))/9.81;

plot(t,accel_arm,'r',t,accel_arm_base,'k');
xlabel('Time(s)')
ylabel('Acceleration of Arm tip (G)')
title('Arm tip Acceleration')
legend('Reduced ring stiffness','current configuration','Location','NorthEastOutside')
```

pause

```
%transfer function for suspension displacement
U_I=[ct2 kt2];
U_III=[ct2 kt2 0 0];
VI=[mt2 ct2 kt2];
```

```

sys_disp_susp=tf(U_I,VI);
disp_susp=lsim(sys_disp_susp,disp_suspension_TVA2,t);
disp_suspension=(disp_susp-disp_arm_TVA2)*1000;

sys_accel_susp=tf(U_I,VI);
accel_susp=lsim(sys_accel_susp,accel_arm1,t);
accel_suspension=accel_susp/9.81;

plot(t,disp_suspension,'r',t,disp_suspension_base,'k')
xlabel('Time(s)')
ylabel('Displacement of Suspension (mm)')
title('Suspension Displacement')
legend('With shock isolator','Without shock isolator','Location','NorthEastOutside')

pause

plot(t,accel_suspension,'r',t,accel_suspension_base,'k')
xlabel('Time(s)')
ylabel('Acceleration of Suspension (G)')
title('Slider Acceleration')
legend('With shock isolator','Without shock isolator','Location','NorthEastOutside')

pause

% arm relative displacement
arm_relative=(disp_suspension_TVA2-disp_arm_TVA2)*1000;

plot(t,arm_relative,'r',t,disp_arm_base,'k')
xlabel('Time(s)')
ylabel('Displacement(mm)')
title('Displacement of Arm tip and Disc After using Elastomer')
legend('Arm relative displacement','Disc relative displacement','Location','NorthEastOutside')

```

A.5 M-file Routine to Simulate TVA placed on the Arm of HAA

```

clc
clear all

%a program to simulate TVA on Arm and TVA on suspension, base excitation problem

Fa=1500; %arm frequency in Hz%
Fs=397; %suspension frequency in Hz%
Ft=1560; %TVA frequency in Hz%

ma=5e-3; %arm mass in kgrams%
ms=0.2*ma; %suspension mass in kgrams%
mt=0.06*ma; %TVA on arm mass in kgrams%
mt2=0.01*ms; %TVA on suspension in kg
M=5000; %dummy mass in kg

ka=((2*pi*Fa)*(2*pi*Fa))*ma; %arm stiffness in N/m%
ks=((2*pi*Fs)*(2*pi*Fs))*ms; %suspension stiffness in N/m%

%ks=2;
%kt=3;

%Stiffness and Mass Matrix before TVA placement:
MM=[M 0 0;0 ma 0;0 0 ms];
KK=[ka -ka 0;-ka (ka+ks) -ks;0 -ks ks];

```

```

E=inv(MM)*KK;
[u,w]=eig(E)
Freq=(sqrt(w))/(2*pi)

Ft=0.86*(Freq(2,2))      %TVA on arm frequency in Hz%
kt=((2*pi*Ft)*(2*pi*Ft))*mt2; %TVA stiffness in N/m%

Ft2=0.21*Freq(3,3)
kt2=((2*pi*Ft2)*(2*pi*Ft2))*mt2;

zeta_TVA=0.000000; %TVA damping ratio%
zeta_TVA2=0.000000;
zeta_arm=0.02;
zeta_suspension=0.000;

ct=zeta_TVA*2*sqrt(kt*mt); %TVA on arm damping value%
ct2=zeta_TVA2*2*sqrt(kt2*mt2); %TVA on suspension damping value%
ca=zeta_arm*2*sqrt(ka*ma);
cs=zeta_suspension*2*sqrt(ks*ms);

t=0:0.000001:0.01;

%SHOCK INPUT

Max_G=25; %in G
Fmax=Max_G*9.81*M;
Freq=250; %shock frequency
wf=2*pi*Freq;
period=1/Freq;
pulse_width=period/2;

Force_input=Fmax*sin(wf*t);

%num=[m c k];
%den1=[(m*M) (c*(M+m)) (k*(m+M)+c^2) (2*k*c-c) (k^2-k)];
%den2=[(m*M) (c*(M+m)) (k*(m+M)+c^2) (2*k*c-c) (k^2-k) 0];
num=[1];
den1=[M 0 0];
den2=[M 0];
den3=[M];
sys1=tf(num,den1);
sys2=tf(num,den1);
sys2.InputDelay=pulse_width;
ydisp1=lsim(sys1,Force_input,t);
ydisp2=lsim(sys2,Force_input,t);
ydisp=ydisp1+ydisp2;
%plot(t,ydisp1,t,ydisp2,t,ydisp,'k')
xlabel('Time(s)')
ylabel('Displacement (m)')

```



```
%pause

sys3=tf(num,den2);
sys4=tf(num,den2);
sys4.InputDelay=pulse_width;
yvelo1=lsim(sys3,Force_input,t);
yvelo2=lsim(sys4,Force_input,t);
yvelo=yvelo1+yvelo2;
%plot(t,yvelo,'k')
xlabel('Time(s)')
ylabel('Velocity (m/s)')

%pause
sys5=tf(num,den3);
sys6=tf(num,den3);
sys6.InputDelay=pulse_width;
yaccel1=lsim(sys5,Force_input,t);
yaccel2=lsim(sys6,Force_input,t);
yaccel=(yaccel1+yaccel2)/9.81;
plot(t,yaccel,'k')
xlabel('Time(s)')
ylabel('Acceleration (G)')
title('Half Sine Input')

%END OF SHOCK INPUT

pause

%X1/Y = E/F --> transfer function of arm displacement%

E_disp=[(ms*mt*ka) ((ms*ct*ka)+(cs*mt*ka)) ((ms*kt*ka)+(cs*ct*ka)+(ks*mt*ka)) ((cs*kt*ka)+(ks*ct*ka)) (ks*kt*ka)];
E_velo=[(ms*mt*ca) ((ms*ct*ca)+(cs*mt*ca)) ((ms*kt*ca)+(cs*ct*ca)+(ks*mt*ca)) ((cs*kt*ca)+(ks*ct*ca)) (ks*kt*ca)];
EE_disp=[(ms*mt*ka) ((ms*ct*ka)+(cs*mt*ka)) ((ms*kt*ka)+(cs*ct*ka)+(ks*mt*ka)) ((cs*kt*ka)+(ks*ct*ka)) (ks*kt*ka) 0];
EE_velo=[(ms*mt*ca) ((ms*ct*ca)+(cs*mt*ca)) ((ms*kt*ca)+(cs*ct*ca)+(ks*mt*ca)) ((cs*kt*ca)+(ks*ct*ca)) (ks*kt*ca) 0];
F=[(ma*ms*mt) ((ma*ms*ct)+(ma*cs*mt)+((ca+cs+ct)*ms*mt))
((ma*ms*kt)+(ma*cs*ct)+((ca+cs+ct)*ms*ct)+(ma*ks*mt)+((ca+cs+ct)*cs*mt)+((ka+ks+kt)*ms*mt)-(cs*cs*mt)-
(ct*ct*ms)) ((ma*cs*kt)+((ca+cs+ct)*ms*kt)+(ma*ks*ct)+((ca+cs+ct)*cs*ct)+((ka+ks+kt)*ms*ct)-
(cs*cs*ct)+((ca+cs+ct)*ks*mt)+((ka+ks+kt)*cs*mt)-(2*cs*ks*mt)-(ct*ct*cs)-(2*kt*ct*ms))
((ma*ks*kt)+((ca+cs+ct)*cs*kt)+((ka+ks+kt)*ms*kt)-(cs*cs*kt)+((ca+cs+ct)*ks*ct)+((ka+ks+kt)*cs*ct)-
(2*cs*ks*ct)+((ka+ks+kt)*ks*mt)-(ks*ks*mt)-(ct*ct*ks)-(2*kt*ct*cs)-(kt*kt*ms))
(((ca+cs+ct)*ks*kt)+((ka+ks+kt)*cs*kt)-(2*cs*kt*ks)+((ka+ks+kt)*ks*ct)-(ks*ks*ct)-(2*kt*ct*ks)-(kt*kt*cs))
(((ka+ks+kt)*ks*kt)-(ks*ks*kt)-(kt*kt*ks))];

%solution for arm displacement
sys_arm_disp1=tf(E_disp,F);
disp_arm_dummy1=lsim(sys_arm_disp1,ydisp,t);
sys_arm_disp2=tf(E_velo,F);
disp_arm_dummy2=lsim(sys_arm_disp2,yvelo,t);
```

```
disp_arm=(disp_arm_dummy1+disp_arm_dummy2-ydisp)*1000;
disp_armx=disp_arm_dummy1+disp_arm_dummy2;

%plot(t,disp_arm)
%xlabel('Time(s)')
%ylabel('Displacement Arm with TVA (mm)')

pause

%solution for arm velocity
sys_arm_velo1=tf(EF,F);
velo_arm_dummy1=lsim(sys_arm_velo1,ydisp,t);
sys_arm_velo2=tf(EF,F);
velo_arm_dummy2=lsim(sys_arm_velo2,ydisp,t);
velo_arm=(velo_arm_dummy1+velo_arm_dummy2);

%sys_arm_accel=tf(EF,F);
%accel_arm=(lsim(sys_arm_accel,ydisp,t))/9.81;

%plot(t,velo_arm)
%xlabel('Time(s)')
%ylabel('Velocity of Arm with TVA')

% $X_2/X_1 = G/H$  --> transfer function of suspension displacement*

%pause

G_disp=[ks];
G_velo=[cs];
GG_disp=[ks 0];
GG_velo=[cs 0];
H=[ms cs ks];

sys_suspension_disp1=tf(G_disp,H);
disp_suspension_dummy1=lsim(sys_suspension_disp1,disp_armx,t);
sys_suspension_disp2=tf(G_velo,H);
disp_suspension_dummy2=lsim(sys_suspension_disp2,velo_arm,t);
disp_suspension=(disp_suspension_dummy1+disp_suspension_dummy2-ydisp)*1000;

%plot(t,disp_suspension)
%xlabel('Time(s)')
%ylabel('Displacement Suspension with TVA')

%pause

%sys_suspension_accel=tf(GG,H);
%accel_suspension=(lsim(sys_suspension_accel,disp_arm_dummy1))/9.81;

%plot(t,accel_suspension)
%xlabel('Time(s)')
```

```
%ylabel('Acceleration Suspension with TVA (G)')

%pause

%O/P --> transfer function for arm alone (no suspension nor TVA)
O_I=[ka];
O_II=[ca 0];
P=[ma ca ka];

sys_disp_arm_alone1=tf(O_I,P);
disp_arm_alone1=lsim(sys_disp_arm_alone1,ydisp,t);
sys_disp_arm_alone2=tf(O_II,P);
disp_arm_alone2=lsim(sys_disp_arm_alone2,ydisp,t);
disp_arm_alone=(disp_arm_alone1+disp_arm_alone2-ydisp)*1000;

%I/J --> transfer function for arm + suspension without TVA for displacement input%
I_disp=[(ms*ka) (cs*ka) (ks*ka)];
I_velo=[(ms*ca) (cs*ca) (ks*ca)];
II_disp=[(ms*ka) (cs*ka) (ks*ka) 0];
II_velo=[(ms*ca) (cs*ca) (ks*ca) 0];
J=[(ma*ms) ((ma*cs)+(ms*ca)+(ms*cs)) ((ma*ks)+(cs*ca)+(ms*ka)+(ms*ks)) ((ks*ca)+(cs*ka)) (ks*ka)];

%Solution for arm displacement
sys_disp_arm_nTVA1=tf(I_disp,J);
disp_arm_nTVA_dummy1=lsim(sys_disp_arm_nTVA1,ydisp,t);
sys_disp_arm_nTVA2=tf(I_velo,J);
disp_arm_nTVA_dummy2=lsim(sys_disp_arm_nTVA2,ydisp,t);
disp_arm_nTVAXx=(disp_arm_nTVA_dummy1+disp_arm_nTVA_dummy2-ydisp)*1000;
disp_arm_nTVAX=disp_arm_nTVA_dummy1+disp_arm_nTVA_dummy2;

plot(t,disp_arm,'g',t,disp_arm_nTVAXx,'r',t,disp_arm_alone,'k')
xlabel('Time(s)')
ylabel('Displacement of Arm (mm)')
title('Simple Model for TVA placed on the Suspension - Arm Displacement')
legend('arm + susp + TVA','arm + suspension','arm alone no suspension','Location','NorthEastOutside')

pause

%Solution for arm velocity
sys_velo_arm_nTVA1=tf(II_disp,J);
velo_arm_nTVA_dummy1=lsim(sys_velo_arm_nTVA1,ydisp,t);
sys_velo_arm_nTVA2=tf(II_velo,J);
velo_arm_nTVA_dummy2=lsim(sys_velo_arm_nTVA2,ydisp,t);
velo_arm_nTVA=velo_arm_nTVA_dummy1+velo_arm_nTVA_dummy2;

%plot(t,velo_arm_nTVA)
%xlabel('Time(s)')
```

```
%ylabel('Velocity Arm no TVA')

pause

%K/L --> transfer function for suspension without TVA%

L_disp=[ks];
LL_disp=[ks 0 0];
L_velo=[cs];
LL_velo=[cs 0 0];
N=[ms cs ks];

sys_disp_suspension_nTVA1=tf(L_disp,N);
disp_suspension_nTVA_dummy1=lsim(sys_disp_suspension_nTVA1,disp_arm_nTVAx,t);
sys_disp_suspension_nTVA2=tf(L_velo,N);
disp_suspension_nTVA_dummy2=lsim(sys_disp_suspension_nTVA2,velo_arm_nTVA,t);
disp_suspension_nTVA=(disp_suspension_nTVA_dummy1+disp_suspension_nTVA_dummy2-ydisp)*1000;

%Q/R --> transfer function for TVA on suspension
Q_I=[(ms*mt2*ca) (ca*(ms*ct2+(mt2*(cs+ct2)))+(ka*ms*mt2)) (ca*(ms*kt2+cs*ct2+(mt2*(ks+kt2)))+(ka*(ms*ct2+mt2*(cs+ct2))))
      (ca*ct2*ks+(ka*(ms*kt2+cs*ct2+(mt2*(ks+kt2)))) (ca*kt2*ks+ka*ct2*ks) (ka*kt2*ks)];
R=[(ma*ms*mt2) (ma*ms*ct2+ma*mt2*(cs+ct2)+ms*mt2*(ca+cs))
    (ma*ms*kt2+ma*cs*ct2+ma*mt2*(ks+kt2)+((ca+cs)*(ms*ct2+(mt2*(cs*ct2))))+ms*mt2*(ka+ks)-cs*cs*mt2)
    (ma*ct2*ks+((ca+cs)*(ms*kt2+cs*ct2+(mt2*(ks+kt2))))+((ka+ks)*(ms*ct2+(mt2*(cs+ct2))))-cs*cs*ct2-2*cs*ks*mt2)
    ((ca+cs)*ct2*ks+ma*kt2*ks+((ka+ks)*(ms*kt2+cs*ct2+(mt2*(ks+kt2))))-cs*cs*kt2-2*cs*ct2*ks-ks*ks*mt2)
    (ca*kt2*ks+ka*ks*ct2-cs*ks*kt2) (ka*kt2*ks)];

%solution for arm displacement
sys_arm_TVA2=tf(Q_I,R);
disp_arm_TVA2=lsim(sys_arm_TVA2,ydisp,t);

%solution for suspension displacement
S_I=[(mt2*cs) (cs*ct2+mt2*ks) (cs*kt2+ks*ct2) (ks*kt2)];
TI=[(ms*mt2) (ms*ct2+(mt2*(cs+ct2))) (ms*kt2+cs*ct2+(mt2*(ks+kt2))) (ct2*ks) (kt2*ks)];

sys_suspension_TVA2=tf(S_I,TI);
disp_suspension_TVA2=lsim(sys_suspension_TVA2,disp_arm_TVA2,t);
disp_susp_TVA2=(disp_suspension_TVA2-ydisp)*1000;

%,t,disp_susp_TVA2,'r'
plot(t,disp_suspension_nTVA,'k',t,disp_suspension,'g',t,disp_susp_TVA2,'r')
xlabel('Time(s)')
ylabel('Displacement of Slider (mm)')
title('Simple Model for TVA placed on the Arm and on the Suspension - Slider Displacement')
legend('no TVA','with TVA on arm','with TVA on suspension','Location','NorthEastOutside')

pause

Y=10*sin(2*pi*Fa*t); %arm frequency
```

```
X=10*sin(2*pi*Fs*t); %suspension frequency

%plot(t,Y,'k',t,X,'g')
%xlabel('Time(s)')
%ylabel('Amplitude')
%title('Comparison Arm Frequency (black) and Suspension Frequency (green)')
```

APPENDIX B

ANSYS COMMAND

B.1 ANSYS Command to Create a Cantilever Beam FE Model and Extract the Eigenvalues and Eigenvectors

```
/BATCH
/COM,ANSYS RELEASE 10.0  UP20050718  18:06:53  02/22/2008
/input,menust,tmp,",,,,,,,,,,,,,1
/GRA,POWER
/GST,ON
/PLO,INFO,3
/GRO,CURL,ON
/CPLANE,1
/REPLOT,RESIZE
WPSTYLE,,,,,,,,0
!*
/NOPR
/PMETH,OFF,0
KEYW,PR_SET,1
KEYW,PR_STRUC,1
KEYW,PR_THERM,0
KEYW,PR_FLUID,0
KEYW,PR_ELMAG,0
KEYW,MAGNOD,0
KEYW,MAGEDG,0
KEYW,MAGHFE,0
KEYW,MAGELC,0
KEYW,PR_MULTI,0
KEYW,PR_CFD,0
/GO
!*
/COM,
/COM,Preferences for GUI filtering have been set to display:
/COM, Structural
!*
/PREP7
!*
ET,1,SOLID95
!*
ET,2,MASS21
!*
!*
R,1,5,5,5, , , ,
!*
!*
MPTEMP,,,,,,,,
MPTEMP,1,0
MPDATA,EX,1,,40e3
MPDATA,PRXY,1,,0.33
MPTEMP,,,,,,,,
MPTEMP,1,0
MPDATA,DENS,1,,6.7e-9
```

```

BLC4,0,0,35,50
TYPE, 1
EXTOPT,ESIZE,0,0,
EXTOPT,ACLEAR,0
!*
EXTOPT,ATTR,0,0,0
MAT,1
REAL,1
ESYS,0
!*
!*
VOFFST,1,1,,
/USER, 1
/VIEW, 1, -0.486335600426 , -0.601243938179 , 0.634021616793
/ANG, 1, 21.2232979062
/REPLO
FLST,5,12,4,ORDE,2
FITEM,5,1
FITEM,5,-12
CM,_Y,LINE
LSEL,, , ,P51X
CM,_Y1,LINE
CMSEL,,_Y
!*
LESIZE,_Y1,1, , , , , ,1
!*
/UI,MESH,OFF
CM,_Y,VOLU
VSEL,, , , 1
CM,_Y1,VOLU
CMSEL,S,_Y
!*
CMSEL,S,_Y1
VATT, 1, 1, 1, 0
CMSEL,S,_Y
CMDELE,_Y
CMDELE,_Y1
!*
CM,_Y,VOLU
VSEL,, , , 1
CM,_Y1,VOLU
CHKMSH,'VOLU'
CMSEL,S,_Y
!*
MSHAPE,0,3d
MSHKEY,1
VMESH,_Y1
MSHKEY,0
!*
CMDELE,_Y

```



```

CMDELE,_Y1
CMDELE,_Y2
!*
/FOC, 1, 21.6509804687 , 18.8729806198 , -2.12619386900
/REPLO
/DIST,1,0.924021086472,1
/REP,FAST
/DIST,1,0.924021086472,1
/REP,FAST
/DIST,1,0.924021086472,1
/REP,FAST
/DIST,1,0.924021086472,1
/REP,FAST
/DIST,1,0.924021086472,1
/REP,FAST
/DIST,1,0.924021086472,1
/REP,FAST
/FOC, 1, 21.5866035180 , 14.3249254772 , -6.48850473459
/REPLO
/DIST,1,0.924021086472,1
/REP,FAST
/DIST,1,0.924021086472,1
/REP,FAST
/DIST,1,0.924021086472,1
/REP,FAST
/DIST,1,0.924021086472,1
/REP,FAST
/FOC, 1, 21.3091819178 , 12.7874096421 , -8.15933432168
/REPLO
/FOC, 1, 23.4548305776 , 11.6985257922 , -7.54607366172
/REPLO
FLST,2,2,1,ORDE,2
FITEM,2,1
FITEM,2,5423
NWPAVE,P51X
BLC5,0,0,6.2,16
FLST,2,2,3,ORDE,2
FITEM,2,9
FITEM,2,-10
KWPAVE,P51X
wpstyle,1,0.1,-1,1,0.003,0,2,,5
FLST,3,1,8
FITEM,3,17.5,-8,0.5
K, ,P51X
CM,_Y,KP
KSEL, , , , 13
CM,_Y1,KP
CMSEL,S,_Y

```

```

!*
CMSEL,S,_Y1
KATT, 1, 1, 2, 0
CMSEL,S,_Y
CMDELE,_Y
CMDELE,_Y1
!*
KMESH, 13
EPLOT
ADELE, 7, , ,1
EPLOT
/VIEW,1,-1
/ANG,1
/REP,FAST
/DIST,1,1.08222638492,1
/REP,FAST
/DIST,1,1.08222638492,1
/REP,FAST
/DIST,1,1.08222638492,1
/REP,FAST
/DIST,1,1.08222638492,1
/REP,FAST
/DIST,1,1.08222638492,1
/REP,FAST
/DIST,1,1.08222638492,1
/REP,FAST
/DIST,1,1.08222638492,1
/REP,FAST
/FOC, 1, 23.4548305776 , 7.22392351734 , -3.39624090681
/REPLO
/DIST,1,0.924021086472,1
/REP,FAST
/DIST,1,0.924021086472,1
/REP,FAST
/FOC, 1, 23.4548305776 , 4.57423411143 , 1.00963798906
/REPLO
SAVE
FINISH
/SOL
FINISH
/REP7
FLST,4,179,1,ORDE,7
FITEM,4,1
FITEM,4,-71
FITEM,4,5422
FITEM,4,-5492
FITEM,4,10843
FITEM,4,-10878
FITEM,4,12679
CP,1,UZ,P51X

```

```
FINISH
/SOL
!*
ANTYPE,2
!*
MSAVE,0
!*
MODEOPT,LANB,50
EQSLV,SPAR
MXPAND,50, , ,0
LUMPM,0
PSTRES,0
!*
MODEOPT,LANB,50,0,20000, ,OFF
!*
MSAVE,0
!*
MODEOPT,LANB,20
EQSLV,SPAR
MXPAND,20, , ,0
LUMPM,0
PSTRES,0
!*
MODEOPT,LANB,20,0,20000, ,OFF
FLST,2,179,1,ORDE,7
FITEM,2,1
FITEM,2,-71
FITEM,2,5422
FITEM,2,-5492
FITEM,2,10843
FITEM,2,-10878
FITEM,2,12679
!*
/GO
D,P51X, ,0, , , ,UX,UY,ROTX,ROTY,ROTZ,
/STATUS,SOLU
SOLVE
FINISH
/POST1
SET,LIST
/DIST,1,1.08222638492,1
/REP,FAST
/DIST,1,1.08222638492,1
/REP,FAST
/DIST,1,1.08222638492,1
/REP,FAST
/DIST,1,1.08222638492,1
/REP,FAST
/DIST,1,1.08222638492,1
/REP,FAST
```

/DIST,1,1.08222638492,1
/REP,FAST
/DIST,1,1.08222638492,1
/REP,FAST
/DIST,1,1.08222638492,1
/REP,FAST
/FOC, 1, 23.4548305776 , 13.5023985873 , 1.76331421105
/REPLO
/FOC, 1, 23.4548305776 , 15.9373525353 , 1.76331421105
/REPLO
/DIST,1,0.924021086472,1
/REP,FAST
/FOC, 1, 23.4548305776 , 17.5444588157 , 1.70974400170
/REPLO
/DIST,1,0.924021086472,1
/REP,FAST
/DIST,1,1.08222638492,1
/REP,FAST
/FOC, 1, 23.4548305776 , 18.8837140493 , 1.76331421105
/REPLO
/VIEW, 1, -0.557462644929 , -0.204422304077E-01, -0.829950308588
/ANG, 1, 0.812137071565
/REPLO
/VIEW, 1, -0.263353920361 , -0.391365096076 , -0.881747171361
/ANG, 1, 4.79347480288
/REPLO
/VIEW, 1, 0.305537756849 , -0.362787395174 , -0.880359009179
/ANG, 1, 18.8460804555
/REPLO
/VIEW, 1, -0.728207962897 , -0.396283802960E-01, -0.684209583570
/ANG, 1, -51.0629273152
/REPLO
/VIEW, 1, -0.413365013519 , -0.590576611913 , -0.693071880154
/ANG, 1, -46.2812688148
/REPLO
/VIEW, 1, -0.768330756348 , -0.315704626796 , -0.556775033087
/ANG, 1, -76.5532420482
/REPLO
/DIST,1,0.924021086472,1
/REP,FAST
/DIST,1,0.924021086472,1
/REP,FAST
/DIST,1,0.924021086472,1
/REP,FAST
/DIST,1,0.924021086472,1
/REP,FAST
/FOC, 1, 19.7838382123 , 25.4226022141 , 3.12145537743
/REPLO
/DIST,1,0.924021086472,1
/REP,FAST

```

/DIST,1,0.924021086472,1
/REP,FAST
FINISH
/POST26
FILE,'Nitinol 50mm','rst',''
/UI,COLL,1
NUMVAR,200
SOLU,191,NCMIT
STORE,MERGE
FILLDATA,191,,,1,1
REALVAR,191,191
/FOC, 1, 20.6093356853 , 20.9855887505 , 4.49818841365
/REPLO
!*
NSOL,2,12679,U,Z, UZ_2
STORE,MERGE
! Save time history variables to file Mode Shapes/Wall.csv
*CREATE,scratch,gui
*DEL,_P26_EXPORT
*DIM,_P26_EXPORT,TABLE,20,1
VGET,_P26_EXPORT(1,0),1
VGET,_P26_EXPORT(1,1),2
/OUTPUT,'Wall','csv','Mode Shapes'
*VWRITE,'TIME','UZ_2'
%C, %C
*VWRITE,_P26_EXPORT(1,0),_P26_EXPORT(1,1)
%G, %G
/OUTPUT,TERM
*END
/INPUT,scratch,gui
! End of time history save
/VIEW, 1, -0.545014909795 , -0.505919575807 , -0.668583675329
/ANG, 1, -70.2071948449
/REPLO
/VIEW, 1, -0.232313840108 , -0.750261465588 , -0.618981431829
/ANG, 1, -58.1987402739
/REPLO
/VIEW, 1, -0.631826381666E-01, -0.820800939737 , -0.567709231527
/ANG, 1, -46.8523054484
/REPLO
/VIEW, 1, -0.466504220340 , -0.601655489787 , -0.648370637841
/ANG, 1, -68.8065481254
/REPLO
/VIEW, 1, -0.743129582604 , -0.364229477542 , -0.561333511514
/ANG, 1, -77.3651927355
/REPLO
/FOC, 1, 18.5254214608 , 22.2101226409 , 6.46245198620
/REPLO
/VIEW, 1, -0.872208123155 , -0.237119664945E-01, 0.488559855644
/ANG, 1, -90.7464944425

```

```

/REPLO
/VIEW, 1, -0.106550387666 , 0.668645686019E-01, 0.992056522762
/ANG, 1, -88.5409796168
/REPLO
/VIEW, 1, 0.711215404681 , 0.890800586267E-01, 0.697307243114
/ANG, 1, -82.3323241842
/REPLO
/VIEW, 1, 0.988793532441 , -0.112757478063E-01, 0.148863050197
/ANG, 1, -80.0324834510
/REPLO
/FOC, 1, 19.3011251644 , 21.5223865581 , 1.25789959490
/REPLO
/VIEW, 1, 0.766527255472 , -0.327946772211 , 0.552165628426
/ANG, 1, -74.1173421306
/REPLO
!*
NSOL,3,5748,U,Z, UZ_3
STORE,MERGE
! Save time history variables to file Mode Shapes/LT.csv
*CREATE,scratch,gui
*DEL,_P26_EXPORT
*DIM,_P26_EXPORT,TABLE,20,1
VGET,_P26_EXPORT(1,0),1
VGET,_P26_EXPORT(1,1),3
/OUTPUT,'LT','csv','Mode Shapes'
*VWRITE,'TIME','UZ_3'
%C, %C
*VWRITE,_P26_EXPORT(1,0),_P26_EXPORT(1,1)
%G, %G
/OUTPUT,TERM
*END
/INPUT,scratch,gui
! End of time history save
/FOC, 1, 21.7592149404 , 21.3148260198 , -2.27774535973
/REPLO
!*
NSOL,4,5507,U,Z, UZ_4
STORE,MERGE
! Save time history variables to file Mode Shapes/LB.csv
*CREATE,scratch,gui
*DEL,_P26_EXPORT
*DIM,_P26_EXPORT,TABLE,20,1
VGET,_P26_EXPORT(1,0),1
VGET,_P26_EXPORT(1,1),4
/OUTPUT,'LB','csv','Mode Shapes'
*VWRITE,'TIME','UZ_4'
%C, %C
*VWRITE,_P26_EXPORT(1,0),_P26_EXPORT(1,1)
%G, %G
/OUTPUT,TERM

```

```
*END
/INPUT,scratch,gui
! End of time history save
!*
NSOL,5,5593,U,Z, UZ_5
STORE,MERGE
! Save time history variables to file Mode Shapes/RT.csv
*CREATE,scratch,gui
*DEL,_P26_EXPORT
*DIM,_P26_EXPORT,TABLE,20,1
VGET,_P26_EXPORT(1,0),1
VGET,_P26_EXPORT(1,1),5
/OUTPUT,'RT','csv','Mode Shapes'
*VWRITE,'TIME','UZ_5'
%C, %C
*VWRITE,_P26_EXPORT(1,0),_P26_EXPORT(1,1)
%G, %G
/OUTPUT,TERM
*END
/INPUT,scratch,gui
! End of time history save
/FOC, 1, 25.1282207895, 28.1883569288, -2.87228169992
/REPLO
!*
NSOL,6,5493,U,Z, UZ_6
STORE,MERGE
! Save time history variables to file Mode Shapes/RB.csv
*CREATE,scratch,gui
*DEL,_P26_EXPORT
*DIM,_P26_EXPORT,TABLE,20,1
VGET,_P26_EXPORT(1,0),1
VGET,_P26_EXPORT(1,1),6
/OUTPUT,'RB','csv','Mode Shapes'
*VWRITE,'TIME','UZ_6'
%C, %C
*VWRITE,_P26_EXPORT(1,0),_P26_EXPORT(1,1)
%G, %G
/OUTPUT,TERM
*END
/INPUT,scratch,gui
! End of time history save
FINISH
!/EXIT,NOSAV
```

APPENDIX C

MULTIPLANE MOUNTS CATALOGUE

Multiplane Mounts

Economical protection from lower frequency vibration

Standard stock Multiplane Mounts are recommended for the isolation of vibration. Lightweight and compact, they provide economical protection from lower frequency disturbances regardless of directions of the forces. They are not recommended where severe, frequently recurring shock is encountered.

These mounts are available in load ratings from 0.25 to 8 lbs. per unit. When loaded to their capacity, a system natural frequency of approximately 10 Hz results, providing effective isolation in applications where disturbing frequencies are above 20 Hz. The radial stiffness is the same as that in the axial direction.

Multiplane Mounts are easy to install. They are available in square or diamond configurations to suit a variety of design requirements.

The contour of the flexing element provides uniform stress distribution.

Snubbing washers provide an interlocking system of metal parts which act to prevent damage from overload or excessive shock impact.

106APL SERIES

(Metric values in parenthesis)

Load capacity: 0.25 to 2 lbs. (0.10 to 0.90 kg)

Materials:

Metal Parts — 2024-T3 or 2024-T4 aluminum alloy per QQ-A-225

Elastomer — Lord BTR® or BTR® II

Finish: Metal Parts — chromate treated per MIL-C-5541, Class 1A

Performance Characteristics

Part Number	Static rate		Nominal axial natural frequency (Hz)	Axial spring rate [†]	
	lbs	kg		lbs/in	N/mm
106APL*-A	1/4	.10	13	3	.5
106APL*-B	1/2	.20	13	5	.9
106APL*-C	3/4	.34	13	8	1.4
106APL*-1	1	.45	13	11	1.9
106APL*-1B	1 1/2	.70	13	16	2.8
106APL*-2	2	.90	13	20	3.5

[†]At .036 in. (.91 mm) D.A. input and rated load.

When ordering, use the following in place of the ():

Q = BTR II Elastomer

W = BTR Elastomer

Dimensions Under No Load

	A*	B	C	D +.008/- .005 +.20/- .12	E +.003/- .002 +.07/- .05	F	G*	I	Q	U*
in	1.00	1.25	1.000	.166	.141	.032	.53	.84	1.414	.38
mm	25.4	31.7	25.40	4.22	3.58	.81	13.4	21.3	35.92	9.6

*Reference dimensions

Snubbing Washer Dimensions

Part Number is	Outside Diameter	Inside Diameter	Thickness
J-2049-1D			
in	.88	.17	.03
mm	22.3	4.3	.8

Transmissibility vs. frequency

Load vs. deflection for 106APLW-2

106APDL SERIES

(Metric values in parenthesis)

Load capacity: 0.25 to 2 lbs. (0.10 to 0.90 kg)

Materials:

Metal Parts — 2024-T3 or 2024-T4 aluminum alloy per QQ-A-225

Elastomer — Lord BTR® or BTR® II

Finish: Metal Parts — chromate treated per MIL-C-5541, Class 1A

Performance Characteristics

Part Number	Static rate		Nominal axial natural frequency (Hz) [†]	Axial spring rate [†]	
	lbs	kg		lbs/in	N/mm
106APDL*-A	1/4	.10	13	3	.5
106APDL*-B	1/2	.20	13	5	.9
106APDL*-C	3/4	.34	13	8	1.4
106APDL*-1	1	.45	13	11	1.9
106APDL*-1B	1 1/2	.70	13	16	2.8
106APDL*-2	2	.90	13	20	3.5

[†]At .036 in. (.91 mm) D.A. input and rated load.

When ordering, use the following in place of the ():

Q = BTR II Elastomer

W = BTR Elastomer

Dimensions Under No Load

	A [*]	D	E	F	G [*]	I	Q	R	S	U [*]
	+.008/- .005 +.20/- .12		+.003/- .002 +.07/- .05							
in	1.00	1.66	.141	.032	.53	.84	1.414	.62	1.66	.38
mm	25.4	4.22	3.58	.81	13.4	21.3	35.92	15.7	42.2	9.6

♣Reference dimensions

Snubbing Washer Dimensions

Part Number is	Outside Diameter	Inside Diameter	Thickness
J-2049-1D			
in	.88	.17	.03
mm	22.3	4.3	.8

Transmissibility vs. frequency

Load vs. deflection for 106APDLW-2

156APL SERIES

(Metric values in parenthesis)

Load capacity: 3 to 8 lbs. (1.4 to 3.6 kg)

Materials:

Metal Parts — 2024-T3 or 2024-T4 aluminum alloy per QQ-A-225

Elastomer — Lord BTR® or BTR® II

Finish: Metal Parts — chromate treated per MIL-C-5541, Class 1A

Performance Characteristics

Part Number	Static rate		Nominal axial natural frequency (Hz)	Axial spring rate [†]	
	lbs	kg		lbs/in	N/mm
156APL*-3	3	1.40	13	30	5.2
156APL*-4B	4.5	2.00	13	45	7.8
156APL*-6B	6.5	2.95	13	65	11
156APL*-8	8	3.60	13	80	14

[†]At .036 in. (.91 mm) D.A. input and rated load.

When ordering, use the following in place of the ():

Q = BTR II Elastomer

W = BTR Elastomer

Dimensions Under No Load

	A*	B	C	D +.008/- .005 +.20/- .12	E +.003/- .002 +.07/- .05	F	G*	I	Q	U*
in	1.50	1.75	1.375	.257	.166	.050	.55	.97	1.945	.38
mm	38.1	44.4	34.92	6.53	4.22	1.27	13.9	24.6	49.40	9.6

*Reference dimensions

Snubbing Washer Dimensions

Part Number is	Outside Diameter	Inside Diameter	Thickness
J-2049-2D			
in	1.38	.26	.05
mm	35.0	6.6	1.3

Transmissibility vs. frequency

Load vs. deflection for 156APLW-3

156APDL SERIES

(Metric values in parenthesis)

Load capacity: 3 to 8 lbs. (1.4 to 3.6 kg)**Materials:**

Metal Parts — 2024-T3 or 2024-T4 aluminum alloy per QQ-A-225

Elastomer — Lord BTR® or BTR® II

Finish: Metal Parts — chromate treated per MIL-C-5541, Class 1A **Performance Characteristics**

Part Number	Static rate		Nominal axial natural frequency (Hz)	Axial spring rate [†]	
	lbs	kg		lbs/in	N/mm
156APDL*-3	3	1.40	13	30	5.2
156APDL*-4B	4.5	2.00	13	45	7.8
156APDL*-6B	6.5	2.95	13	65	11
156APDL*-8	8	3.60	13	80	14

[†]At .036 in. (.91 mm) D.A. input and rated load.

When ordering, use the following in place of the ():

Q = BTR II Elastomer

W = BTR Elastomer

Dimensions Under No Load

	A [♣]	D [♣] +.008/- .005 +.20/- .12	E [♣] +.003/- .002 +.07/- .05	F	G [♣]	I	Q	R	S	U [♣]
in	1.50	.257	.166	.050	.55	.97	1.945	.88	2.32	.38
mm	38.1	6.53	4.22	1.27	13.9	24.6	49.40	22.4	58.9	9.6

♣Reference dimensions

Snubbing Washer Dimensions

Part Number is	Outside Diameter	Inside Diameter	Thickness
J-2049-2D			
in	1.38	.26	.05
mm	35.0	6.6	1.3

Transmissibility vs. frequency **Load vs. deflection for 156APDLW-3**