



NAVAL POSTGRADUATE SCHOOL

MONTEREY, CALIFORNIA

**DEVELOPMENT OF A PARTICLE DAMPING
SOLUTION FOR PC-BOARDS IN CUBESATS**

by

Nicolas van Kevelaer

August 2016

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ABSTRACT

CubeSats are mounted as a secondary payload to unique locations on launch vehicles that are primarily used to bring bigger satellites to space. They are subject to high frequency vibration excitations which causes CubeSat developers to concentrate on survivability instead of improvement by equipping with sensitive state-of-the-art components. Commercial isolation systems only exist for larger primary payloads. To protect the CubeSats sensitive parts as printed circuit boards, a new low cost, low weight method is developed. A particle damping solution is designed to damp printed circuit boards individually to improve their ability to withstand the harsh launch environment. The oscillation is reduced significantly by the particle dampers. An optimized solution is found that reduces the weight and space needed further so that it is better usable in CubeSats. The results of this thesis improve the possibilities for use of highly sensitive printed circuit boards to increase the possible space research possibilities feasible with CubeSats. Particle Dampers can be adopted for larger space applications as well, to decrease the required weight and cost for useful damping.

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List of Acronyms and Abbreviations

ABC	Aft Bulkhead Carrier
AP	Auxiliary Payload
ASD	Acceleration Spectral Density
Cal Poly	California Polytechnic State University, San Luis Obispo
CG	Center of Gravity
CSD	Canisterized Satellite Dispenser
CubeSat	Small standardized cubic satellite (10x10x10cm, 1.33kg)
DD	Damping Device
DEM	Discrete Element Method
ELV	Expendable Launch Vehicle
EMS	Electro-Magnetic Shaker
EOM	Equations of Motion
FEA	Finite-Element-Analysis
FEM	Finite-Element-Method
FF	Fill-Factor
GEVS	General Environment Verification Standard
LAR	Last Absolute Residuals
LEO	Low Earth Orbit
LOX	Liquid Oxygen
LV	Launch Vehicle

MDOF	Multiple Degree of Freedom
MID	Mass Impact Damper
NASA	National Air and Space Administration
NPS	Naval Postgraduate School
NPSAT1	Naval Postgraduate School Satellite 1
NPSCuL	Naval Postgraduate School CubeSat Launcher
P-POD	Poly-Picosatellite Orbital Deployer
PC	Polycarbonate
PC/104	Standard Printed Circuit Board used in CubeSats
PCB	Printed Circuit Board
PP	Primary Payload
PSD	Power Spectral Density
RMS	Root-Mean-Square
SDOF	Single Degree of Freedom
SSAG	Space Systems Academic Group
SSME	Space Shuttle Main Engine
VD	Viscous Damper
VEM	Viscoelastic Material

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CHAPTER 1:

Introduction

1.1 Motivation

Small satellites and especially CubeSats are subject to harsh launch environments. They are often mounted in unique locations on a launch vehicle (LV) that are susceptible to unpredictable random vibratory excitations caused by shock and air flow. Poly-Picosatellite-Orbital-Deployers (P-PODs) are used to contain CubeSats on their way to orbit. These P-PODs may be mounted in the NPSCuL system designed by the Naval Postgraduate School's Space Systems Academic Group (SSAG) in Monterey, California. The NPSCuL has been attached several times, for example, to the Aft Bulkhead Carrier (ABC) of the Atlas V launch vehicle.

Lan discussed in [1] that NPSCuL amplifies the response from $7.6 G_{\text{RMS}}$ at the ABC by as much as a factor of two. Additional amplification caused by the P-POD assembly has to be considered. This high excitation can lead to damage to the sensitive components of a CubeSat and therefore makes it difficult to use highly sensitive parts and connections for further space exploration. It is of great importance to protect the sensitive components mounted to printed circuit boards (PCBs) to ensure successful launch and thus improve the success of CubeSat deployments.

To protect the CubeSats from being damaged during launch it is necessary to find a way to reduce the influence of the launch environment on the sensitive components of the satellite system. Additionally, this damping device has to perform well in the space environment and during launch. There are several methods of reducing the influence of the launch environment that are compatible with space applications. Passive damping devices are the most desirable due to space and energy storage limitations. The small amount of electrical energy that can be stored in a CubeSat is needed for the payload. Viscoelastic materials (VEMs) are the most commonly used in passive damping devices. Because most VEMs are affected by changes in tem-

perature, their use is limited for further space exploration. The launch environment, especially for CubeSats, entails too wide a temperature range for VEMs.

After careful examination of the passive damping possibilities in previous research discussed in [2], the aim of this thesis is to find an inexpensive, lightweight, damping mechanism for CubeSat PCBs. Therefore, the damping mechanism selected is a particle damping (PD) system. Particle damping consists of the addition of a small container, henceforth called cavity, to a system or to drill holes in specific locations of the component [3]. This cavity is then filled with particles, which dissipate some of the energy of the vibrating system as heat. Additionally, the particles transfer the most energy back to the system when they are 180° out of phase, which results in a decreased vibration response of the main system.

As a first step in this research to find a working damper setup for CubeSat PCBs, the vibrational behavior of a printed circuit board or, in this case, an aluminum model of a printed circuit board of type PC/104 is analyzed. Then, an introduction to the state-of-the-art in particle damping is provided. Furthermore, the design of a particle damping device is performed which is followed by an experimental investigation of the vibration attenuation achieved by different types of particle dampers for sinusoidal and random excitations. Lastly, an optimized solution is found that meets the low space and weight constraints inside the CubeSat but provides the system with high damping. This leads to the result that a CubeSat is capable of delivering better and more precise scientific results through the use of more sensitive parts like modern computer chips.

1.2 Background

1.2.1 CubeSats

CubeSats are a type of miniaturized spacecraft. They were invented in 1999 by professors Puig-Suari (Cal Poly) and Twiggs (Stanford University) to give graduate students the ability to design their own small satellites for educational purposes [1]. The standard has two primary elements: the first one is the definition of a stan-

standardized geometry for satellites, small enough to reduce the development time and to reduce cost, but large enough to allocate a useful payload volume. At the same time, the surface area must be big enough to carry solar power arrays to provide enough energy for the satellite instruments and the payload [4]. The second primary element is a standard deployment mechanism that allows a harmless occupation of unused volume on a launch vehicle by CubeSats; this objective was met by the Poly-Picosatellite Orbital Deployer (P-POD) [4].

P-PODs house up to three 1U CubeSats [4] and open the door to release them after being triggered by the launch vehicle sequencer. As soon as the door is open, a spring system deploys the CubeSats. The main task of a P-POD is to protect the primary payload of the launch vehicle. CubeSats started out as relatively simple and robust systems [1], but they were adopted by several companies and governments for research and small space applications and are nowadays equipped with state-of-the-art technology which results in more sensitivity of the built-in components to expand their overall capabilities. CubeSats provide low cost space exploration, which motivated its use on the commercial space sector. Since the CubeSat standard was established in 1999, over 600 CubeSats have been deployed in orbit. At the same time, the participating students gained valuable experience in satellite development, which is scalable to the development of large spacecraft.

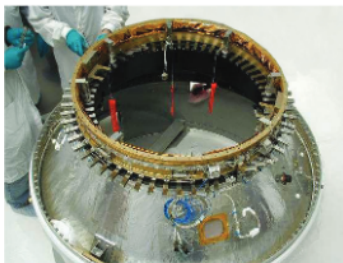
CubeSat sizes are described by units “U”. One unit (or 1U) occupies a volume of a 10cm edge dimension cube. As long as the cross section of 10 cm x 10 cm is maintained, an expansion of this geometry up to 3U is possible using a P-POD. Bigger CubeSat-based satellites up to 27U are deployed by a different deployment container called the Canisterized Satellite Dispenser (CSD) [5]. The standard size of the CubeSat makes it possible to collaborate with external manufacturers or sources because everyone is aware of the interfaces and size constraints. A quickly developing research area for the CubeSat community is the formation of CubeSat networks which would make sharing of computational, sensor and communication resources possible, thus achieving the functionality of larger satellites [4]. Furthermore, it is easy to achieve a redundancy of CubeSats on orbit, based on simpler launch requirements, making it less fatal if one of the CubeSats fails. [6] defines constraints for the design of CubeSats

and the attached parts. The driving requirements for the design of a damping device for a CubeSat are as follows; the dimensions refer to a 1U CubeSat:

1. All parts shall remain attached to the CubeSat
2. No hazardous materials, low outgassing
3. 100.0+-0.1 mm wide & thick
4. 100.0+-0.1 mm tall
5. 6.5mm max normal component height from the outside surface
6. Rails have a minimum width of 8.5 mm
7. At least 75% of rails have to be in contact with P-POD rails
8. Maximum mass of a 1U CubeSat shall not exceed 1.33 kg
9. Center of gravity within a sphere of 2cm from geometric center
10. Aluminum 7075 or 6061 used for structural parts of CubeSat
11. Total stored chemical energy will not exceed 100 Watt-Hours
12. Limited magnetic field strength
13. Capable of withstanding vibration frequencies between 20-2000 Hz
14. Temperature bandwidth during launch -50-100 °F

1.2.2 Launch Environment

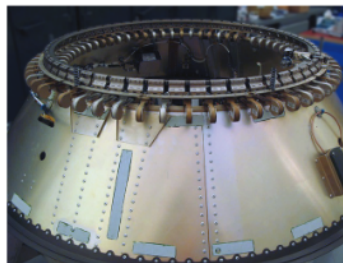
Currently, small-satellites are mostly launched as a secondary payload on launch vehicles used to launch bigger satellites or manned spacecraft as a primary payload (PP), although there are attempts to launch them separately. The primary payload is taken care of by reducing vibrations and damping the movements through technology designed by, for example, Moog CSA Engineering as seen in Figure 1.1.



SoftRide UniFlex

(US Patent 6,199,801)

- Designed to attenuate predominantly axial and bending loads
- Reduces vibration, shock and structure-borne acoustics



SoftRide OmniFlex

(US Patent 7,249,756, EP Patent Pending)

- Designed to attenuate axial, bending and lateral loads
- Reduces vibration, shock and structure-borne acoustics



SoftRide ShockRing

(US Patent 6,202,961, EP 1 268 277)

- Designed to attenuate shock loads
- Reduces shock and structure-borne acoustics

Figure 1.1: SoftRide Primary Payload Damping System, Source: Moog CSA Engineering, Available: http://www.csaengineering.com/literature/Space_Defense/Vibration_Control/MoogCSA_SoftRide_brochure.pdf, accessed: 6/22/2016

These damping devices are tuned to the resonance frequency of the primary payload and cost often more than \$50,000 per device. This is not affordable for secondary payloads such as CubeSats or their deployers, for example, the P-PODs. Furthermore, all parts that are supposed to be sent to orbit must satisfy the constraints previously mentioned in 1.2.1. Additionally, accelerations acting on the system are very high during launch, such as those due to shock during stage separation. This input can be further amplified by the CubeSat structure, especially if thin plates are used. These constraints make the use of different damping techniques besides particle damping very difficult; this prompts the development of particle damping for small satellites, especially for sensitive printed circuit boards such as the Pumpkin CubeSat board shown in Figure 1.2.

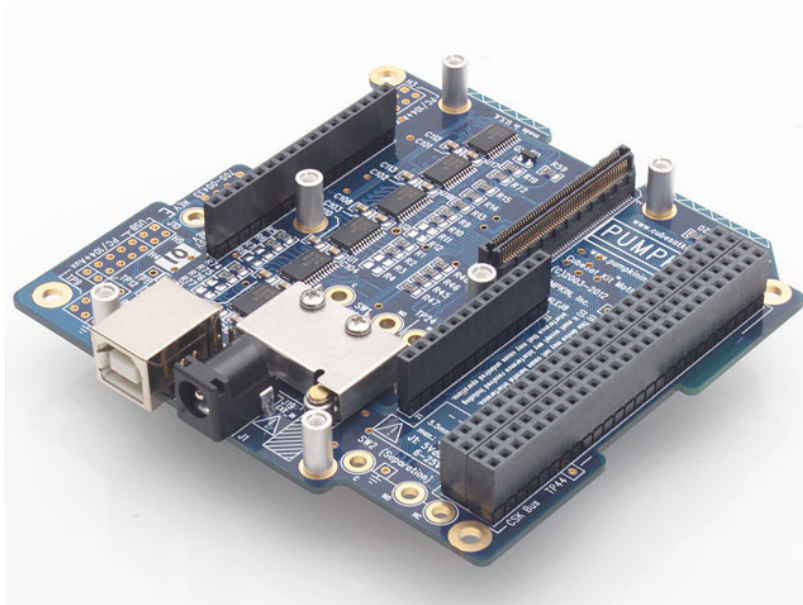


Figure 1.2: Pumpkin CubeSat PC board, Source: Pumpkin, Inc., cubesatkit.com, Available: <http://www.cubesatkit.com/content/news-2012.html>, accessed: 6/22/2016

CHAPTER 2:

Vibrational Analysis of Plates

This chapter gives a short introduction to calculation of vibrational movement of plates to find the most significant points where to attach a particle damper. The vibration of plates has been mathematically described by several authors including Kirchhoff, Chakraverty and Mindlin [7]. This thesis uses the Kirchhoff-Love plate theory due to the fact that a thin plate, as a model for a PC/104-board, is used to investigate different influences of particle damping parameters.

2.1 Mathematical Basics

The Kirchhoff plate theory, or classical plate theory, is used for most practical applications [8]. It is based on the Euler-Bernoulli beam theory and uses several assumptions [7]–[9]:

1. The height of the plate is small compared to length and width
2. Normal stresses in the transverse direction of the plate are negligibly small
3. Negligible effect of the rotatory inertia
4. Plate thickness is uniform (constant thickness)
5. No significant developments of extension of the plate's mid surface
6. Small strains
7. Material is homogenous and isotropic

These assumptions lead to the differential equation of the Kirchhoff-Love plate theory:

$$\frac{\partial^4 w}{\partial x^4} + 2 * \frac{\partial^4 w}{\partial x^2 \partial y^2} + \frac{\partial^4 w}{\partial y^4} = \frac{p}{K}$$

with p = load, w = deflection and K = bending stiffness ($K = \frac{Eh^3}{12(1-\mu^2)}$) with E = modulus of elasticity, μ = poisson ratio and h = plate's thickness. The differential equation is adapted to the actual boundary conditions of the system. Fixed, free and elastic boundaries differ. An analytical calculation has to be performed to obtain exact results in calculation. Alternatively, close solutions can be found by numerical

solutions of the Kirchhoff-Love equations, which can deliver results much faster, as well as for bigger and more complex systems.

Solving the Kirchhoff-Love equation by hand is difficult and very time consuming but the advent of fast computers leads to greatly increasing the data generation for calculating vibrational behavior of plates. The partial differential equation can be solved by separating into regular differential equations. Maple© or Matlab© may be used for solving such equations either analytically or numerically. Furthermore, finite element analysis (FEA) can be used for calculating the natural frequencies and mode shapes of a thin plate like a PCB.

2.2 Modal Analysis Using Finite Element Modeling

The desired outcome of a modal analysis is to find the natural mode shapes and their frequencies during free vibration. Through this investigation, it is possible to analyze the system's behavior during resonance excitation and the associated displacements and accelerations. The system design engineer has to study the system's modal behavior while designing a new system to make sure that no excitation leads to resonance excitation to protect the system from failure and damage. Resonance excitation often leads to damage and destruction of the system [10] as exhibited by the Tacoma-Narrows-Bridge's collapse on the morning of November 7, 1940. The natural frequency of the bridge was excited by 40-mile-per-hour winds, which resulted in a vertical movement of the bridge's deck that was too big for the structure to handle.

2.2.1 PC/104 Board

The PC/104 printed circuit board is the standard board for CubeSats. It is a small board which can be stacked using bolts and spacers. PC/104 boards mechanically and electrically support the assemblies and data processing units in a CubeSat [11]. The board perfectly fits on the inside of a CubeSat and can be mounted so that it does not move in x and y directions. A drawing of a PC/104 board with its dimensions and adapters is posted in Figure 2.1. The board has outer dimensions of 95.885 mm x 90.17 mm and is 1.5748 mm thick [12].

In this thesis, an aluminum plate is used to simplify calculations and to test the behavior of a CubeSat PCB of type PC/104 due to the fact that a printed circuit board is difficult to model. Printed circuit boards often differ in their layer structure and thus in their density, elasticity and stress resistance. Furthermore, the aluminum model is assumed as isotropic.

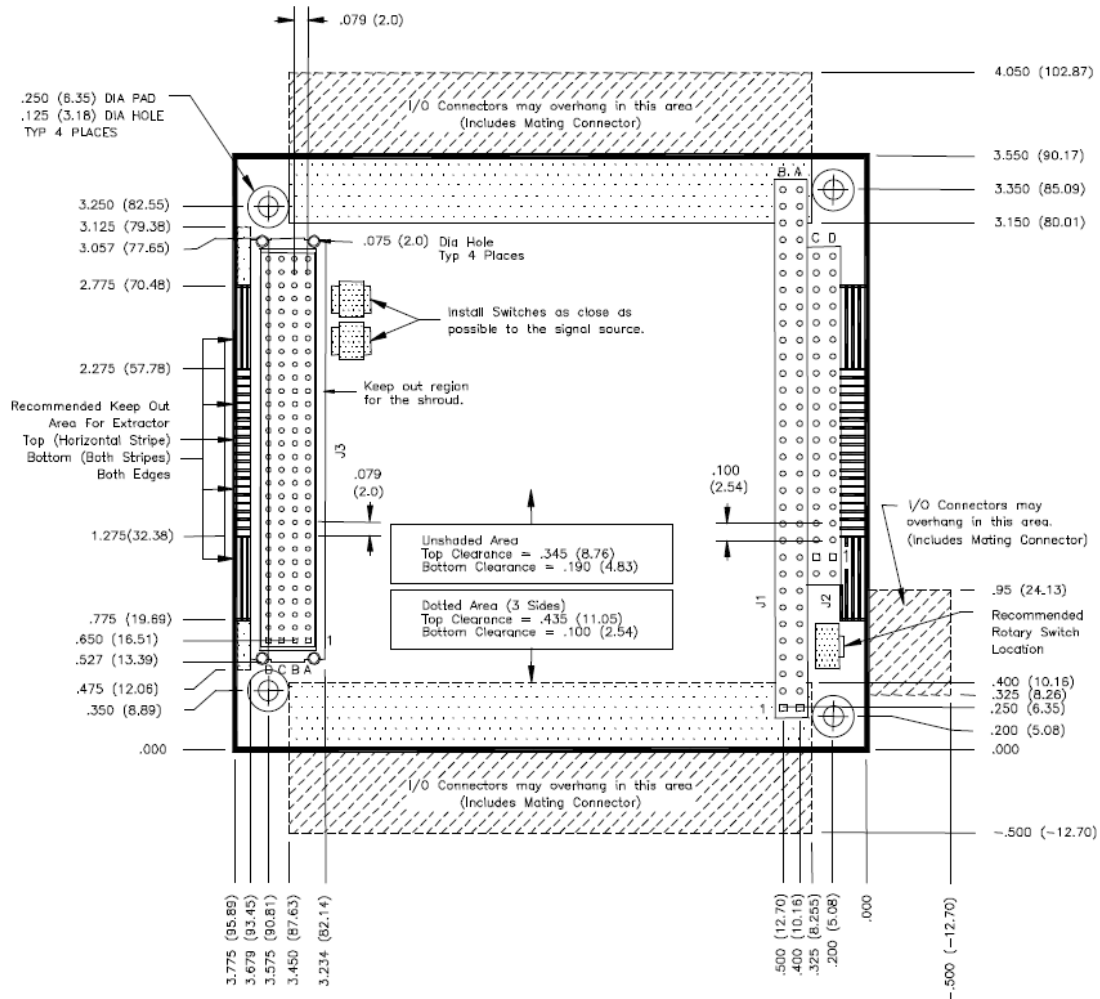


Figure 2.1: PC/104 drawing, board dimensions in mm in brackets,Source: PC/104 Consortium, Available: [12]

2.2.2 Finite Element Analysis

A first important step for designing particle dampers is to find the characteristics of the

system [13]. For this purpose, a finite element analysis (FEA) is executed to find the natural frequencies and mode shapes. Additionally, experimental verification is used to analyze the important modes of the system. Through this analysis, it is possible to find the most sensitive excitation regions and thus to protect these regions by developing a damping mechanism that reduces the oscillation at these points. Using FEA, the system is divided into a mesh of elements connected by nodes. The finer the mesh, the more nodes and elements are generated in the model which leads to more precise results. The FEA algorithm calculates stress, displacement and other properties for each node by solving the system of differential equations. Furthermore, a modal participating mass study is performed to find the modes, valuable enough to be damped to reduce the stress in the PCB-model. After modeling the PCB in Siemens NX10©, a structural analysis is performed for frequencies from 20 to 2000 Hz. The NASTRAN solver is used for this purpose. Afterwards, the excitation with NASA's General Environment Verification Standard (GEVS) random excitation requirements is simulated. Through this simulation, it is possible to evaluate the displacement that has to be attenuated. Another aim of the FEA is to evaluate the acceleration response of the plate to its input.

2.2.3 Results

The PC/104 aluminum model has four modes in the frequency bandwidth from 20 to 2000 Hz. The corresponding values can be seen in Table 2.1.

Table 2.1: Modes for PC/104 Model in Frequency Bandwidth 20 to 2000 Hz

Mode Number	Frequency [Hz]	Mass Participation Factor [%]
1	656.8	75.3
2	1139.6	0.003
3	1268.3	0.01
4	1396.1	0.3

The high mass participation factor for the first mode leads to the result that the

first mode has to be damped properly to prevent the board from high displacements that would damage electrical and mechanical parts attached to the board. The mode shapes displayed by the model are shown in Figure 2.2.

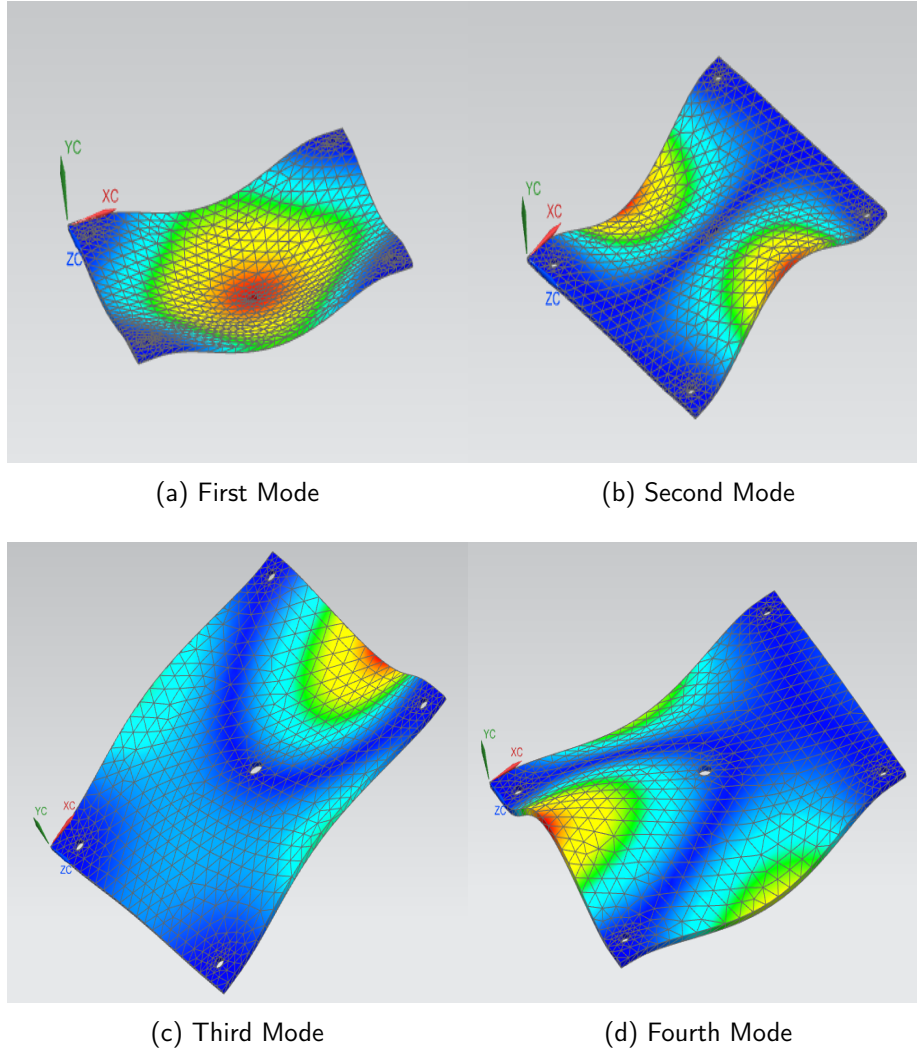


Figure 2.2: Mode shapes of PC/104-Aluminum-Model, red areas show high displacement while blue regions show areas of low relative displacement

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CHAPTER 3:

Particle Damping

Several damping techniques exist for mitigating resonant vibrations that are excited in a structure [14]. For a precise introduction to the different damping techniques, the interested reader is referred to literature or research papers such as [2]. Mass impact damping (MID) is a passive damping method used in several applications throughout many research fields; examples include vibration isolation of buildings in [15] and for use in space systems by NASA in [16]. In MID, a container is fixed to a vibrating structure, and a single mass is placed inside the container. Through the impact of the mass on the inner surface of the container, the vibrational movement of the primary system is reduced.

Particle damping (PD) is a newly developed derivative of the previously mentioned MID [17] which was first proposed by H. V. Panossian in 1989 [18]. In this method, a cavity is filled with many small masses (particles) [13]. Particle damping is proven to be very effective in damping vibrations and is applied in structural damping [19]. The cavity may be attached to a vibrating structure or cavities may be introduced in the structure itself and filled with particles [20] [19]. Several types of particles, such as powders and even liquid particles can be used [19]. Powders can contain metallic and non-metallic particles. The use of small particles, instead of heavier and bigger single masses, eliminates excessive noise and "potential damage to the interior wall of the containing hole" [21]. Furthermore, it reduces the high sensitivity of the MID to the size of the container and the input excitation [22]. A tuning to specific frequencies is not as sensitive as for a mass impact damper. Particle dampers work over a wide frequency bandwidth. A question to be answered is whether the usage of a particle damper is appropriate. Due to the fact that PD is preferable for well-separated modes [13], it is feasible to use PDs for the modes described in Section 2.2.3.

When the structure starts vibrating, the particles begin their movement and collide with the walls of the cavity and with each other. This process dissipates energy from the vibrating structure through highly nonlinear (i.e. input amplitude dependent) loss

mechanisms, including friction and the exchange of momentum along with the transfer of the kinetic energy to heat, which are dependent on the variables of the system itself [22], [23]. These loss mechanisms play different roles in different environments as discussed in [18]. When the cavity is accelerated by the systems displacement, the particles are pressed against one wall of the cavity. When the movement of the cavity is reversed by the oscillation of the system, the particles reach the opposite wall and their energy transfers in the opposite direction of the cavity's energy. This results in a lower displacement of the system where the cavity is attached [24].

The performance of particle dampers varies due to several factors including dependence on the amplitude of the input vibration, i.e. it is highly nonlinear [22], and orientation with respect to gravity [24].

The advantages of particle damping devices are that they are low cost and easy to design. Furthermore, they are of low weight and cover a wide range of temperatures [13] and frequencies [18] and are suitable for harsh environments. They work properly at cryogenic and high temperatures, where viscous dampers (VDs) and viscoelastic materials (VEMs) [22] fail regularly or are not suitable [18]. They have a low rate of degradation [22] and can work in high vacuum and corrosive environments. Additionally, high values of damping can be reached by very low "weight penalties" [22]. Weight penalties are defined by

$$\beta = \frac{\text{mass of the particles}}{\text{mass of the un damped system}} = \frac{m_p}{M} \quad (3.1)$$

The effective mass of the system, M_{eff} , at a specific vibrational movement dependent on the gap size, i.e. the space between top of the particles and the upper wall, is defined in Figure 3.3.

Particle damping techniques have been used, or at least designed, for small spacecraft, to reduce the vibration of a whole CubeSat [24], in "dead-blow" hammers (hammers that do not bounce back) [24] and for passively mitigating launch vibrations in cryocoolers. Research is carried out concerning the use of particle damping based solutions for vibration isolation in turbojet engines caused by compressor blade

vibrations [22], [24], which was evaluated by Pratt & Whitney, Rocketdyne [25] and NASA. Additionally, PD can be used for minimizing the vibration in a Space Shuttle Main Engine (SSME) liquid oxygen (LOX) inlet tee as in [19]. Further applications of PD can be found in [26], [27] and [28].

3.1 Mode of Operation

Particle dampers may show a completely different behavior when used in low gravity environments because mechanisms of energy dissipation that play an unimportant role in regular gravity environments may become critical in low gravity environments and vice versa [4]. This has not yet been examined. The California Polytechnic University in San Luis Obispo is developing a CubeSat called CP7 [24] for research in this field. It has three cantilever beams with a cavity at each of the free ends. These cavities are partially or fully filled with particles [24]. An example for a PD can be seen in Figure 3.1.



Figure 3.1: Particle damper with steel particles

The CP7 experiment is set up to determine whether the low-gravity environment on orbit changes the behavior of particle dampers or if it stays comparable. The doubts about the comparability are caused by the behavior of granular particles, which show three different phases, characterized by their rates of energy loss, for different velocity

amplitudes. These phases are solid, convective and gas-like [29]. The best damping behavior is shown in the solid phase (at least in gravity environments) while the damping forces are lower in the convective phase, which shows fluid-like behavior. This results in a minimum of damping at the transition from liquid behavior to gas-like behavior. In gas-like behavior, the damping forces rise again. Due to the fact that gravity influences become unimportant for high acceleration, it is assumed that particle dampers show gas-like behavior in low-gravity environments [4]. Salve[U+FFFD] al. show the behavior of the particles and the related damping in [29]. The trend can be seen in Figure 3.2. The vertical lines represent the edges of the phases named above. The solid phase is on the left while the rapid convective phase (liquid phase) is in the middle and the gas-like state is on the right.

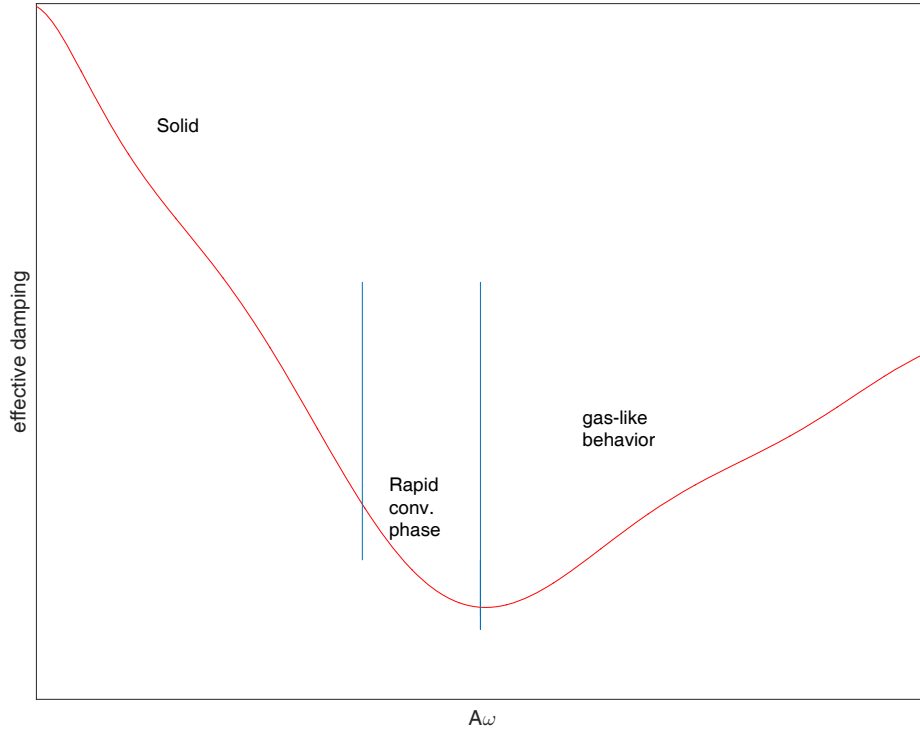


Figure 3.2: Damping by Particle Movement. Effective damping parameter vs. amplitude of velocity of vibration $A\omega$. Solid-state on the left side, liquid behavior in the middle until minimum is reached, then gas-like behavior. Trend is shown based on the investigation by Salueña et al in source: [29]

Due to the fact that the length of the gap L is important for the amount of impacts in a vibrational movement, equation 3.2 used this value as well as the gravity acceleration to calculate the minimum damping [29]. The relating velocity of vibration, $A\omega$ is the characteristic velocity at which the gas-state is first shown by the system.

$$\frac{L}{A\omega} = \frac{A\omega}{g} \quad (3.2)$$

Or by solving the equation for $A\omega$:

$$A\omega = \sqrt{L * g} \quad (3.3)$$

Since in a zero-gravity environment g becomes 0, the $A\omega$ value of the minimum damping equals 0 as well. This would result in a gas-like behavior at any time. Further research is needed to show whether particle dampers are useful for space applications, although they might just exhibit gas-like rather than solid behavior.

Simulation of PD behavior in a zero-gravity environment in [30] has shown that the effectiveness in a zero-gravity environment can be improved through fitting a blade in the center of the PD to force the particles to not form a floating cluster, which would reduce the damping effect.

After the Cal Poly satellite CP7 is launched and research results are published, the influence of the no-gravity environment parameters introduced here may be defined.

The mode of operation varies with the influence of gravity and the frequency at which the cavity is oscillating.

3.2 Design Parameters

The significant design parameters are categorized into those that can be adjusted and those that cannot. The parameters that can be adjusted are the shape, size and material of the cavity; the material and therefore the friction, whose coefficient determines the main part of the friction energy dissipation together with the "normal press force" [18]; the density, as well as the coefficient of restitution, which is the main factor to "influence the impact energy dissipation" [18], and the influence of magnetism of the particles, the size of the particles and the amount of particles (and therefore the fill-factor and weight).

Martín Sánchez and Luis A. Pagnaloni investigated the influence of the additional weight due to the addition of particles in 2011 in [31]. They studied the response of single-degree-of-freedom (SDOF) mechanical systems with an attached particle damper. After Yang [32] found that the system "may display masses above $M(\text{system's mass}) + m_p(\text{particles mass})$ " [31] they continued performing simulations using the Discrete Elements Method (DEM) to verify Yang's results and to characterize the influ-

ence of specific parameters relating to the effective mass of the system. They found that the gap size affects the displayed mass in vibratory motion. The effective mass as a function of the gap size is displayed in Figure 3.3. [23] shows that “there are times when the particles move while in contact with the system while at other times they move separately”.

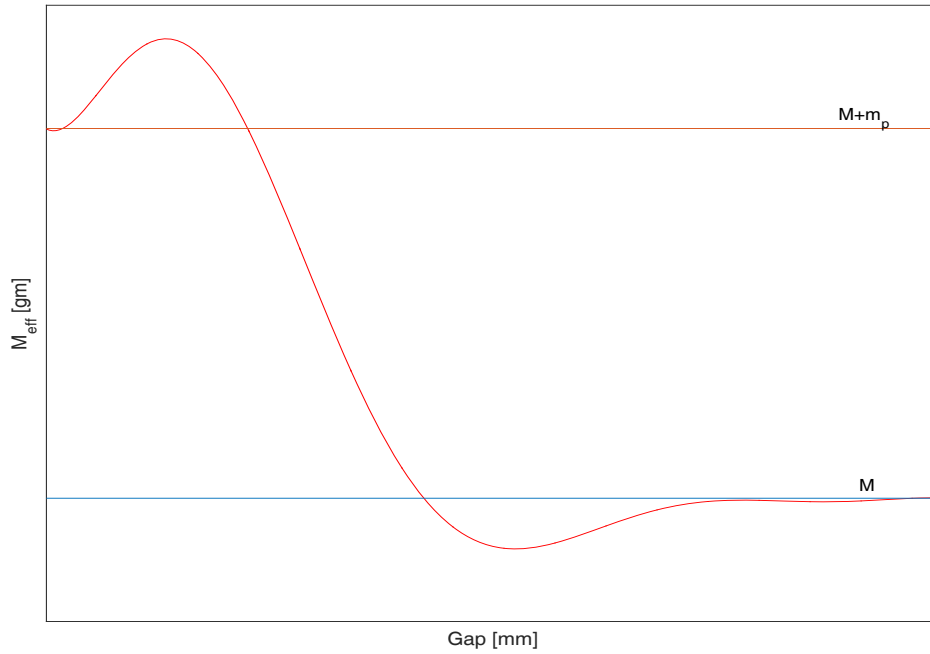


Figure 3.3: Effective mass (M_{eff}) vs. gap size. The upper red line represents the total weight of the system including the weight of the particles while the lower blue line represents only the systems mass. Based on results from Sánchez and Pugnaroni, 2011, Source: [31]

Fixed parameters include the influence of gravity (due to the fact that the gravity changes during launch with distance to earth until it equals zero) and the excitation (which is a result of shock or random vibration).

3.3 Simulation Techniques

3.3.1 Discrete Element Method

The Discrete Element Method (DEM) is a numerical method based on the Hertz theory of rigid particles [33]. “Each particle is modelled as a rigid body while the whole PD system is treated as an assembly” [18]. The algorithm calculates the position (through displacements) and the stresses of each single particle for a specified time step and focuses on the relative motions in particle-particle and particle-wall interactions [18]. During impact, the DEM takes into account the shear stiffness of the collision as well as the normal stiffness of the collision. The calculation effort of the DEM goes to infinity for higher amounts of particles, which introduces the necessity of using flow-theory-based finite element simulation programs such as COMSOL©Multiphysics.

3.3.2 Molecular dynamics

The dissipation of energy in an oscillating system containing a cavity is based on two aspects: first, the power that is transmitted to the walls of the cavity and second, the dissipative work through collisions between single particles. Salueña et al. used a modified soft-particle model by Cundall and Strack [29] in which they simulate the interaction between cavity and particle as well as particle with particle. The molecular dynamics method is a computer simulation method used to understand the behavior of molecular flow and impact, particularly in areas that cannot be tested in experiments or to minimize the effort needed to verify analytical models. Related to the topic of this thesis, the particles are seen as a fluid (flow theory), which interact with themselves and the cavity walls. The simulation of molecular dynamics consists of a step-by-step solution of the well-known, classical equations of motion (EOM). “At a specific time, the forces on a particle in the system are only determined by the interaction with directly contacting particles” [14]. The constraints of this method are given by restrictions in computational power. The time needed for a calculation increases by taking more particles and bigger systems into account. It is even possible to include the effects of bonding by welding and fraction of particles through impact with each other as discussed in [34].

CHAPTER 4:

Development of Particle Damping Application

This chapter describes the design of a particle damping application for the purpose of damping the vibration in CubeSat PCBs.

4.1 Requirements/Parameters

A Particle Damper (PD) typically consists of two different parts. The first part is the cavity (a small container or hole), which contains the second part, the particles. An example for these components can be seen in Figure 4.1 where the white outer material is the cavity and the dark material on the inside is the particles.



Figure 4.1: Cylindrical Particle Damper, white material is cavity, dark material is particles

The cavity is defined by the shape, the length and its diameter. The shape may have different forms such as cylindrical, rectangular, hexagonal, spherical and some less common special forms. Length and diameter (or base area) define the capacity of the damper.

The particles are defined by their size, density, coefficient of restitution, friction, hardness and damping properties. Furthermore, the amount of particles is of great

influence because it defines the total particle mass and the resulting gap size. The fill-factor (FF) is dependent on the particle properties as well.

$$FF = \frac{h_{particlesatrest}}{h_{cavity}} \quad (4.1)$$

All these factors can be varied to change the PD behavior and thus to maximize the outcome of the damping process.

4.2 Construction of Cavity

The chosen material for the cavities is polycarbonate. It has a working temperature range of -40 °C (-40 °F) [35] to 132 °C (270 °F) [36] that fits the needs for a stable damping device in the mentioned predicted environmental conditions during launch defined in Section 1.2.1. Furthermore, polycarbonate (PC) can be used in a 3D-printer to rapidly produce new cavities and cavity forms.

Six different cavity sizes are produced. The inner cavity size is of a diameter of 15.24 mm while the length is varied from 5 mm over 10 mm and 15 mm as well as 20 mm and 25 mm to 25.4 mm. The maximum height of 25.4 mm is chosen based on the cavity size in [23]. The other sizes were chosen to investigate smaller cavities which result in lower weight penalties β . The resulting volume of the inner cavity is 912, 1824, 2736, 3648, 4560 and 4633 mm³. The cavity is formed in the shape of a cylinder due to the fact that this leads to better damping in the x and y directions, which span the footprint, independent of the excitation direction [37]. Rectangular cavities only damp properly if the excitation is normal to a side-wall. The cavities consist of a main body and a cap glued on top of the cavity. The weight of the cavities varies from 1.85 gm to 3.15 gm for the smallest to largest inner space, respectively. Additional weight is added to the cavity due to the needed space to thread a cavity to the board. Gluing the cavities onto the board has been evaluated but could result in delamination and subsequent loss of the connection caused by the bending modes of the plate. Cavities that lose their connection could damage the whole CubeSat system, especially the sensitive components. Screwed connections maintain a consistent boundary condition when a torque wrench is used.

4.3 Introducing Different Fill Levels

Each designed cavity size is tested with different fill levels of particles. The particles used are steel shots of granularity S110 and glass bead particles of size 60-120. The steel shots measure approximately 0.5 mm in diameter and are chosen because of their high density and thus the high weight inside the cavities. Additionally, this material was readily available. According to [23], the influence of the chosen material is only interesting for small numbers of particles. As soon as the number of particles increases over a specific, unspecified, value the influence of the material properties decreases. The density of the particle material is important because it affects the mass of the particles m_p . Higher density is preferable to achieve smaller particle damping cavities. The fill-factors are varied in steps of 30 % from 0 % to 90 % in the first round of testing and then adjusted for a higher resolution in the most interesting areas. According to Sánchez and Pugnaroni in [31], a fill-factor of 100% would result in a solid body that behaves as a simple mass and therefore would lose its damping abilities. The fill-factors are evaluated by weighing the filled particles and measuring the distance from the top of the particles to the top of the cavity. The higher the fill, the higher the weight of the whole particle damper. According to the work of Sánchez and Pugnaroni in [31], increasing the fill-factor will not shift the resonance frequency if the damper's gap is tuned correctly. It is important that the resulting resonance frequencies, in case of a mistune of parameters, do not excite other critical frequencies in the CubeSat system or that the damping has to be high enough to protect the system from higher oscillations in different regions. This importance makes a testing of the whole satellite with the attached PDs necessary. The resulting gaps and weights, m_p of the particle damper assemblies are shown in Tables 4.1, 4.2, 4.3, 4.4, 4.5 and 4.6.

One aim is to minimize the weight penalties β while maximizing the damping ability. A lightly damped primary system is able to achieve a “high response reduction with a small weight penalty” [37].

Table 4.1: Weight and gap sizes of 25.4 mm-long particle damper design filled with S110 steel shot with weight penalty

Cavityheight [mm]	fill-factor [%]	m _p [gm]	gap [mm]	β [%]
25.4	100	20.05	0	50.1
	95	19.05	1.27	47.6
	90	18.05	2.54	45.1
	85	17.04	3.81	42.6
	80	16.04	5.08	40.1
	75	15.04	6.35	37.6
	60	12.03	10.16	30.1
	30	6.015	17.78	15.0

Table 4.2: Weight and gap sizes of 25 mm-long particle damper design filled with S110 steel shot with weight penalty

Cavityheight [mm]	fill-factor [%]	m _p [gm]	gap [mm]	β [%]
25	100	19.5	0	48.8
	95	18.53	1.25	46.3
	90	17.55	2.5	43.9
	85	16.58	3.75	41.4
	80	15.6	5	39
	75	14.63	6.25	36.6
	60	11.7	10	29.3
	30	5.85	17.5	14.6

Table 4.3: Weight and gap sizes of 20 mm-long particle damper design filled with S110 steel shot with weight penalty

Cavityheight [mm]	fill-factor [%]	m _p [gm]	gap [mm]	β [%]
20	100	15.75	0	39.4
	95	14.9625	1	37.4
	90	14.18	2	35.5
	85	13.39	3	33.5
	80	12.6	4	31.5
	75	11.81	5	29.5
	60	9.45	8	23.6
	30	4.72	14	11.8

Table 4.4: Weight and gap sizes of 15 mm-long particle damper design filled with S110 steel shot with weight penalty

Cavityheight [mm]	fill-factor [%]	m _p [gm]	gap [mm]	β [%]
15	100	11.89	0	29.7
	95	11.30	0.75	28.3
	90	10.70	1.5	26.8
	85	10.11	2.25	25.3
	80	9.51	3	23.8
	75	8.92	3.75	22.3
	60	7.13	6	17.8
	30	3.57	10.5	8.9

Table 4.5: Weight and gap sizes of 10 mm-long particle damper design filled with S110 steel shot with weight penalty

Cavityheight [mm]	fill-factor [%]	m _p [gm]	gap [mm]	β [%]
10	100	8	0	20.0
	95	7.6	0.5	19.0
	90	7.2	1	18.0
	85	6.8	1.5	17.0
	80	6.4	2	16.0
	75	6	2.5	15.0
	60	4.8	4	12.0
	30	2.4	7	6.0

Table 4.6: Weight and gap sizes of 5 mm-long particle damper design filled with S110 steel shot with weight penalty

Cavityheight [mm]	fill-factor [%]	m _p [gm]	gap [mm]	β [%]
5	100	4.03	0	10.1
	95	3.83	0.25	9.6
	90	3.63	0.5	9.1
	85	3.43	0.75	8.6
	80	3.22	1	8.05
	75	3.02	1.25	7.55
	60	2.42	2	6.1
	30	1.21	3.5	3.0

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CHAPTER 5:

Experimental Testing

5.1 Expectations

Based on prior research, expectations are generated. The particle weight in the cavity should be at least as high as 10-15% of the weight of the total participating modal mass of the oscillating system at the modal frequency [24] whose amplitude should be attenuated. Brown additionally states in [24] that a further addition of mass has little effect on the system and the particle damper's damping ratio (DR). Sánchez, Carlevoro and Pugnaroni state in [34] that the optimal particle damper does not change the resonance frequency. Furthermore, they state that a (large) shift in the resonance frequency only occurs when the gap size of the cavity is so small that movement of the particles is impeded, which is unlikely for very small, dusty particles. The unlikeliness leads to making an evaluation of fully filled cavities necessary. The gap should not be smaller than three times the amplitude of the system's vibration [34]. The gap influence has further been investigated for harmonic excitation, where an equation was found which states that the gap should be as wide as $A_{opt} * \omega t = L_g$ [38], where L_g is the gap (the space between the top of the particles and the cavity cap) and A_{opt} is the amplitude at the participating mode while ω is the excitation frequency.

Furthermore, the granularity, which means the diameter, of the particles should not have any influence on the damping behavior, nor should the fragmentation (separation of one particle into various) of the particles [31]. The only influencing parameters are weight, gap size and the coefficient of restitution which, according to [34], should not affect the particle damper's response if the number of particle layers is high enough. Simultaneously, friction at lower frequencies is of great influence for dissipating the energy and therefore reducing the vibration of the system but becomes less important at very high frequencies where impacts between particles and ceiling dominate the damping effect. Some sources prove that other parameters influence the damping as well. Tianning et al. state in [18] that the damping ratio of a damper filled with 85% shall be better than the damping ratio of a 75% filled damper.

5.2 Test-Setup

For simplification, a PC/104 model made out of aluminum is used for its availability and isotropy. The aluminum board is mounted onto an adapter plate using #4-40 screws (the same as in a typical CubeSat, torqued to 60 in-oz) and 12.7 mm standoffs. The adapter plate is mounted to a 11000 lb electro-magnetic-shaker (EMS). The PD cavity is mounted in the center of the mounting holes used to attach the PC/104 model to the adapter plate. A #8-32 screw is used to mount the damper on the plate, torqued to 86 in-oz. The specified position is chosen based on the fact that placing the PD at the anti-node maximizes the energy transferred into the damping device [39]. The cavity is made out of polycarbonate with dimensions mentioned above. An aluminum version was machined as well; these results will be discussed in chapter 6. One PC/104 testing assembly is mounted to the shaker at the same time. For each test, the assemblies are filled with different amounts of particles, more precise: S110 steel shots or glass beads as seen in Figure 5.1 using fill-factors from 0 to 100%. The 0% experiment setup is used for comparison purposes as a control baseline.

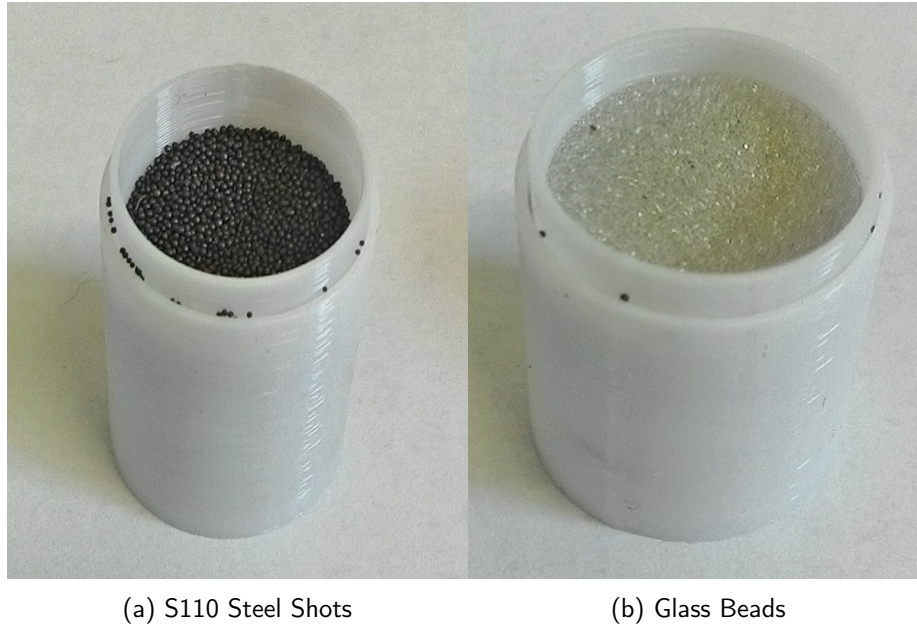


Figure 5.1: Different Particle Materials in Polycarbonate Cavities

The setup consists of an 11000 lb electro-magnetic shaker (EMS) connected to an amplifier and controlled by a personal computer, which runs the control software. The shaker can be seen in Figure 5.2. The selected excitation input is controlled by two control channel accelerometers. The acceleration of the plate is measured by a single accelerometer close to the point of the highest amplitude of displacement, where the highest acceleration at resonance frequencies is expected based on the FEA results of the PC/104 plate.



Figure 5.2: Electro Magnetic Shaker in SSAG's vibration laboratory

The control accelerometers are fastened to the adapter plate while the measuring accelerometer is connected to the specimen using Petro wax. For safety reasons, the cables of the accelerometers are fixed using Kapton tape. In each test run, a differently sized or differently filled cavity is attached to the assembly. Four different fill-factors for the 5mm sized cavity can be seen in Figure 5.3.

5.3 Testing of Different Particle Damper Alternatives

5.3.1 Fill-Factor

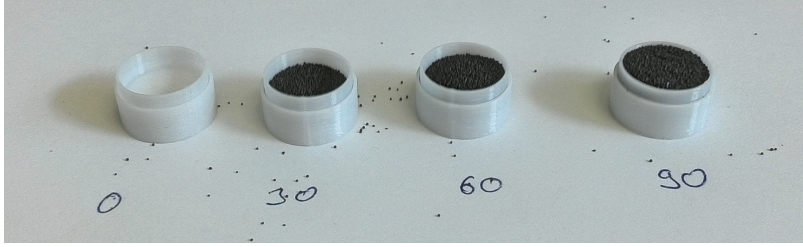


Figure 5.3: Cavity Fill Levels in % for 5mm cavity

The fill-factor is defined by the weight of the particles related to a specific cavity. First, the weight of a fully filled cavity is measured and then tested at specific fill levels and their related particle weight. The following table lists the different weights at different fill-levels of differently sized cavities. The shown fill-levels are 0, 30, 60 and 90%. For other fill-levels compare Tables 4.1, 4.2, 4.3, 4.4, 4.5 and 4.6. The used accelerometer mounted to the aluminum board weighs 1 gm while the #8-32 screw used for attachment of the PD to the board weighs 1.2 gm.

Table 5.1: Fill levels of particle dampers including weight and weights of fixing parts and measurement devices for FF of 30, 60, 90 and 100%

Cavity size [mm]	m_p [gm]				Cavity weight [gm]
Fill level [%]	100	90	60	30	
Steel shots					
25.4	20.05	18.05	12.03	6.01	3.15
25	19.5	17.55	11.7	5.85	3.2
20	15.75	14.18	9.45	4.72	2.85
15	11.89	10.70	7.13	3.57	2.5
10	8	7.2	4.8	2.4	2.20
5	4.03	3.63	2.42	1.21	1.85
Glass beads					
25.4	6.85	6.165	4.11	2.05	3.15

A high precision analogue scale (Figure 5.4) is used to perform the measurement of the filled particles' weight.

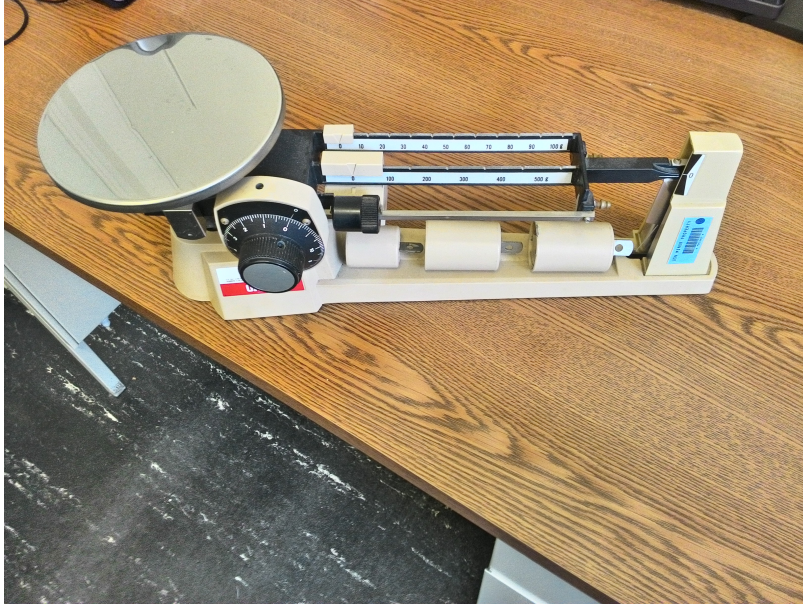


Figure 5.4: High precision scale used for measuring the fill-factor of particle dampers

5.3.2 Excitation

Each test run for the fill-factors mentioned in Table 5.1 is performed using a sine sweep excitation and a random excitation in a frequency range of 20-2000 Hz. The sine sweep applies a 1g acceleration while the random excitation applies 10 G_{RMS} . The ASD values of the random excitation defined in the NASA General Environmental Verification Standard (GEVS) are shown in the following Figure 5.5; this standard is used for spaceflight programs and projects [40].

Frequency (Hz)	ASD Level (g ² /Hz)	
	Qualification	Acceptance
20	0.026	0.013
20-50	+6 dB/oct	+6 dB/oct
50-800	0.16	0.08
800-2000	-6 dB/oct	-6 dB/oct
2000	0.026	0.013
Overall	14.1 G _{rms}	10.0 G _{rms}

Figure 5.5: GEVS Generalized Random Vibration Test Levels Components, Input for Random Vibration Testing for Components of 22.7 kg or less, NASA, Source: [40]

5.4 Damping Calculation

The half-power bandwidth method is utilized to calculate the damping ratio of a system. Although it is applicable only to SDOF systems with a small damping coefficient, it is frequently used for multiple-degree-of-freedom (MDOF) systems showing well-separated modes [41]. Utilizing the half-power bandwidth method the first step is to find the peak amplitude and the corresponding resonance frequency, f_{res} . Next, the peak amplitude is divided by $\sqrt{2}$ to determine the frequencies at the half-power points, f_1 and f_2 . It is shown in textbooks (e.g. [42]) that the damping ratio, b , for small damping is approximately

$$b = \frac{f_2 - f_1}{2 * f_{res}} \quad (5.1)$$

If the damping is not assumed small, the equation becomes [41]

$$b = \sqrt{0.5 - \sqrt{0.25 - 0.0625 \left(\frac{f_2 - f_1}{f_{res}} \right)^2 \left(\frac{f_2 + f_1}{f_{res}} \right)^2}} \quad (5.2)$$

The damping in this case is assumed to be small when lower than 5%. This leads to the use of Equation 5.1 for the cavities with large gaps and low fill-factors (FF) and Equation 5.2 for cavities with a small gap and high FF and thus high weight.

5.5 Experimental Results - Sine Sweep

5.5.1 Baseline - Control Setup

The first experiment studies the behavior of the plate without an attached cavity to investigate the system's parameters. The excitation used is a sine sweep with an amplitude of 1 g. An up sweep (from 20 Hz to 2000 Hz) and a down sweep (from 2000 Hz to 20 Hz) are performed. The down sweep is shown in Figure 5.6. The resonance peak is found at a frequency of 697.5 Hz with an amplitude of 208 g. A second resonance can be found at around 1450 Hz. This shows that the actual system's parameters are slightly different from the parameters used for the simulation, which shifts the resonance frequencies up. This can be caused by differences in the homogeneity of the plate's material or by differences in the modeled boundary conditions such as the mounting of the plate. This difference does not significantly change the modal mass participation. It is still the first mode that has to be attenuated to reduce the stresses on the PC/104 board. Another very small peak can be seen around 1350Hz. The aim of this thesis is to reduce the amplitude at the first mode because of the higher mass participation factor of approximately 77%; therefore, the amplitude at the second and third modes is negligible and any damping of these resonances would only be a positive side effect. It has to be observed whether the addition of both heavier cavities and higher particle masses shifts the resonance frequency to areas where the high acceleration excites other system's components resonances. According to [34], the addition of a properly tuned particle damper does not change the resonant frequency significantly but does attenuate the oscillation of the device to which it is attached. The baseline is used to compare the damping behavior of the particle dampers attached in further experiments to the undamped control setup. Therefore, it is easier to observe the PD effects on the undamped system. Furthermore, this baseline identifies the internal damping of the PC/104 board model to make sure that it is not too high, because particle damping works best for structures with low internal damping.

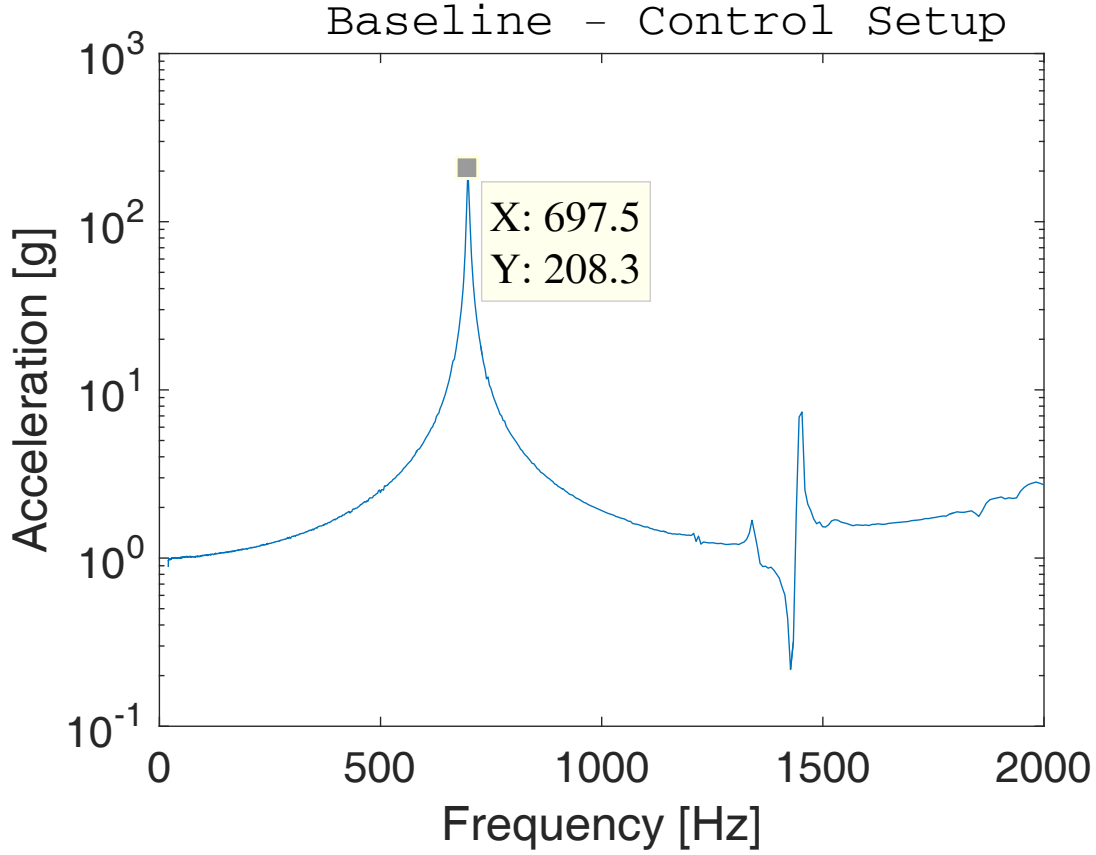


Figure 5.6: Down Sweep without Cavity using a 1g input

The resulting damping for the baseline without an attached damper is 0.67%. This is considered light damping; therefore, particle dampers are expected to be very effective [37].

5.5.2 Sine Sweep Results

This section points out significant results obtained by testing of different particle damping alternatives. A full overview of all the experimental results and the corresponding acceleration-frequency plots can be found in Appendix ??.

The addition of unfilled cavities to the plate slightly reduces the resonance frequency due to the added weight. With the addition of small particle masses, the damping ratio increases negligibly or even decreases. It was found that a minimum amount of

particles has to be added to the cavity to achieve notable damping.

The lowest weight setup that first achieves major results is the 90%-filled cavity with a length of 10 mm. This setup reduces the acceleration amplitude to 32.02 g at a frequency of 615 Hz. This results in a reduction of 84.6% with an addition of only 7.2 gm of steel-shot particles. Figure 5.7 shows the acceleration-frequency plot of this setup. Rattling occurs at a frequency bandwidth of 400-500 Hz. This rattling is attributed to slight adjustments of the damper parameters, such as gap size, and occurs because the particles' impact on ceilings is not perfectly out of phase with the movement of the plate. For optimal damping, the two systems (plate and particles) have to collide in out of phase because their movements cancel each other through the transfer of momentum. According to [31], in-phase motion can result in an amplification of the peak resonance. Rattling occurs in almost all damping setups investigated with this experiment setup. Mostly, rattling occurs in a frequency range of approximately 200-800 Hz.

Furthermore, the resonant peak around 1450 Hz is damped as well. The weight penalty is 18% while the damping is 0.9%.

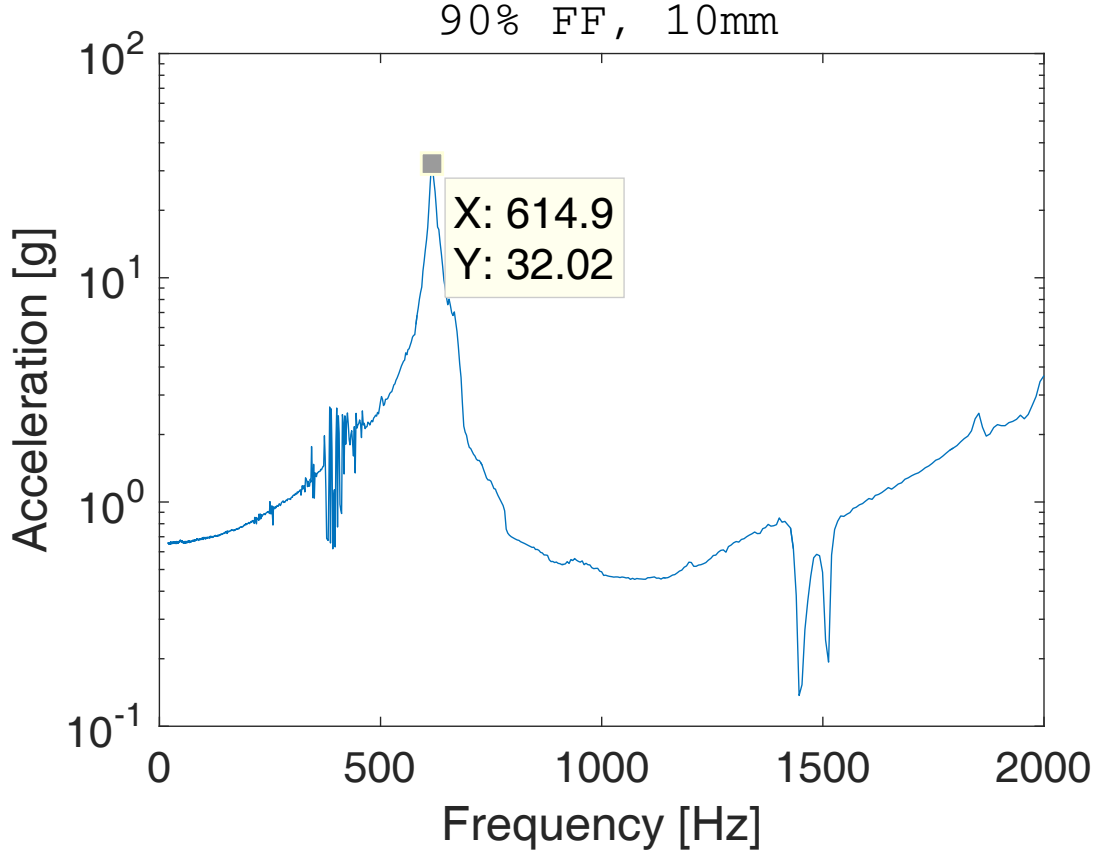


Figure 5.7: PD Response Due to Sine Sweep, 10 mm-long Cavity filled with 90% steel shots

The first particle damper setup that has a damping of higher than 5% is the 90%-filled, 15 mm-long cavity whose response plot is shown in Figure 5.8. The amplitude is reduced by 98% to 4.422 g at a frequency of 604 Hz, which is a major improvement. Compared to the unfilled 15 mm-long cavity (see Appendix ??) the resonance frequency does not shift at all. Consistent with [31], this leads to the assumption that the damper is designed properly. The other resonances in the range from 20 to 2000 Hz are damped to an amplitude lower than the excitation amplitude of 1g of the sine sweep. The total particle mass that has been added to the system is 10.7 gm which results in a weight penalty of 26.8%. The unchanged resonance frequency of the first mode can be investigated by using Figure 3.3. The gap size of the cavity for this specific application seems to be big enough to not affect the effective mass (M_{eff})

of the system. Due to the fact that the gap size is relatively small compared to the size of the damper, it is assumed that the gap size in this application corresponds to the point in Figure 3.3 where the M_{eff} line first crosses the line of the systems mass, M .

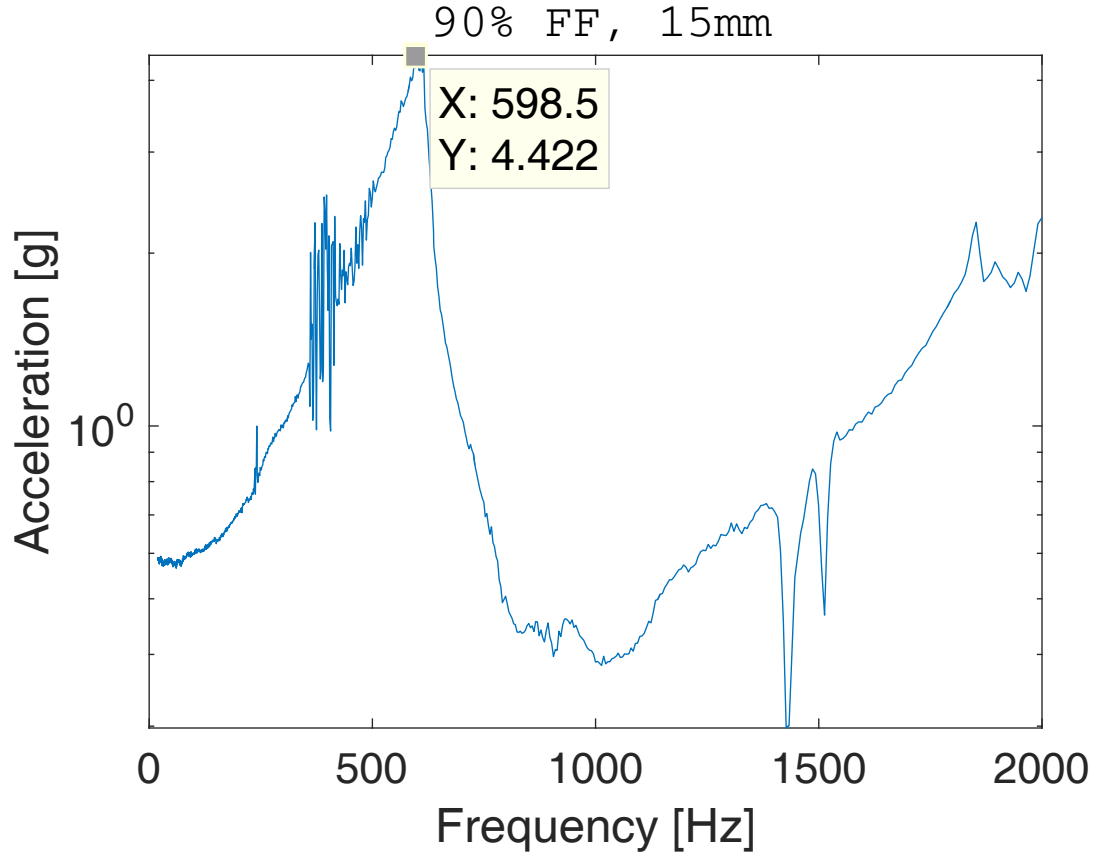


Figure 5.8: PD Response Due to Sine Sweep, 15 mm-long Cavity filled with 90% steel shots

It was confirmed that further increases in cavity size and weight lead to higher damping coefficients and lower acceleration amplitudes. An increase in cavity size up to 25.4 mm and weight up to 18.045 gm decreases the amplitude to 3.2 g at a frequency of 482 Hz. The shift in frequency indicates that the gap size is mistuned. According to Figure 3.3 and the findings in [31], the resonance frequency should not change compared to the resonance frequency of the undamped system if a properly tuned particle damper is added. The major downshift of more than 120 Hz could affect

other system components by exciting their resonance frequencies. Furthermore, the weight penalty increases to 45.1% which is a very high value. Although the damping increases to 8.6% which is considered high damping, the size of the cavity is long compared to other cavity shapes. This attribute decreases the benefit of the PD due to the fact that the space on a printed circuit board is very limited and the damper of this length could exceed space limitations.

Another significant finding is that the amplitude of the resonance at 1500 Hz exceeds the 1 g acceleration input which indicates that this resonance is amplified by the damping device. Further inspection of this trend shows that this amplification increase could damage the system. The plot for the 25.4 mm-long cavity application is shown in Figure 5.9.

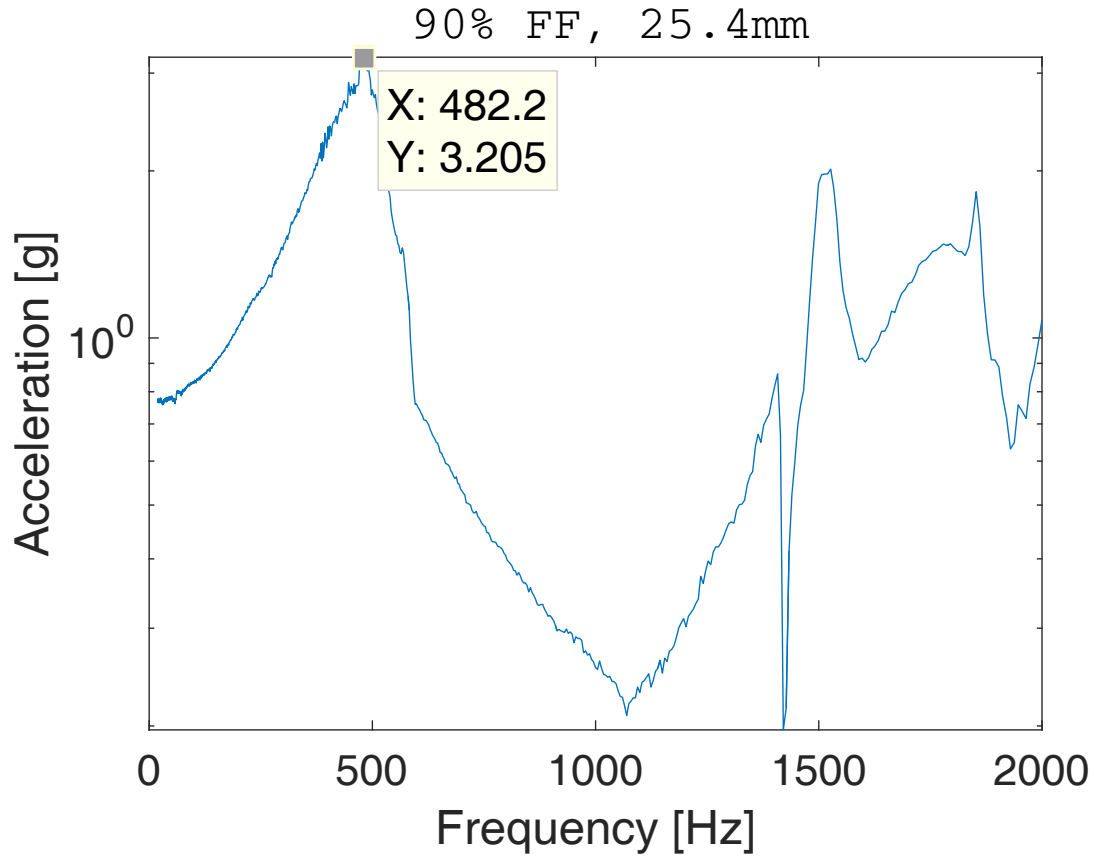


Figure 5.9: PD Response Due to Sine Sweep, 25.4 mm-long Cavity filled with 90% steel shots

The amplification of the higher mode around 1500 Hz is visibly apparent for a 100% filled 25.4 mm-long cavity. Here, the amplitude of the higher mode is higher than the first mode which can be seen in Figure 5.10. By adding more weight, the peak amplitude could increase more. This leads to the assertion that a bigger cavity is not useful for future investigations of PCB damping in CubeSats using particle damping.

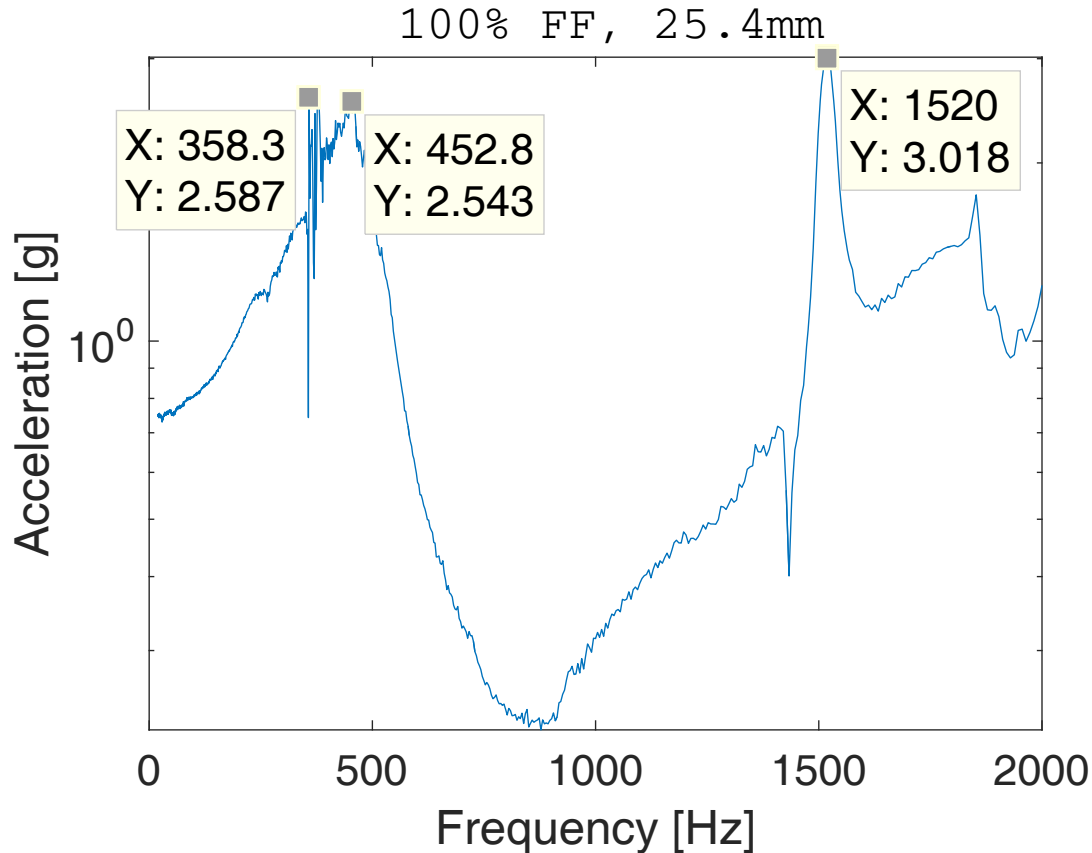


Figure 5.10: PD Response Due to Sine Sweep, 25.4 mm-long Cavity filled with 100% steel shots

The results of the sine excitation experiment highlight that a minimum amount of weight is needed for the PD to be effective. Furthermore, a maximum gap should not be exceeded. The minimum weight that is found to add significant damping to the first mode around 650 Hz is approximately 10 gm for a single particle damper. The overall weight of the damper is around 13 gm in that case. Further investigation needs to be done to find out whether significant attenuation of the system or damping

ratios can be achieved with less added weight or smaller cavity heights.

5.5.3 Glass-Bead-Filled Cavity Sine Excitation

Due to the fact that the shaker is electro-magnetic, the magnetic field has to be neutralized on top of the shaker so that it does not affect the performance of the damper (when using steel shots). To verify that the magnetic field created by the shaker does not influence the results of the experiment, glass beads are used to verify that the damping also works properly for both magnetic and non-magnetic particles. The result of the glass bead test show that the magnetic field of the shaker has no influence on the damping. The only glass-bead-filled cavity is 25.4 mm-long filled with 30, 60 and 90% FF. The particle weight along with the resulting peak amplitudes and damping, are shown in Table 5.2:

Table 5.2: Damping of glass-bead-filled PD

Fill-Factor[%]	m_p [gm]	β [%]	Amplitude[g]	Frequency[Hz]	Damping[%]
90	6.2	15.4	11.0	603.9	4.00
60	4.1	10.3	29.1	601.2	1.60
30	2.1	5.1	68.8	598.5	0.90
0	0	0	161.1	590.5	0.01

A very interesting finding is that a damping ratio of around 4% is achieved by just adding 6.165 gm of weight which results in a weight-penalty β of 15.4%. This supports the assumption that weight is not the only factor of interest in the design of particle damper applications. It is possible to achieve the same results as with steel shots but with lower weight penalties. The coefficient of friction and the gap size play an important role as well.

5.6 Experimental Results Random Excitation

After the sine sweeps are performed, a test using random excitations is executed. This test uses the GEVS random input as shown in Figure 5.5. The input is stepped up in 3 dB increments from -9 to 0 dB. The 0 dB results are discussed in this section. Random excitations are evaluated for fill-factors from 0 to 90% in steps of 30.

5.6.1 Baseline - Control Setup

The response of the system without an attached damper is first measured at all four levels from -9 dB to 0 dB to confirm that the undamped system shows the expected results generated by the system's input. The system's input is verified to work correctly, because all plots have the same shape and are only different in amplitude. The result is shown in Figure 5.11 and serves as a baseline for evaluating test results for the system with added damping. This comparison is done for peak acceleration spectral density (ASD) and root-mean-square acceleration (G_{RMS}) values. Calculating the G_{RMS} of different setups allows the evaluation of the effect of the damping device over the full frequency bandwidth of interest. The method of calculating G_{RMS} values is explained later in this section.

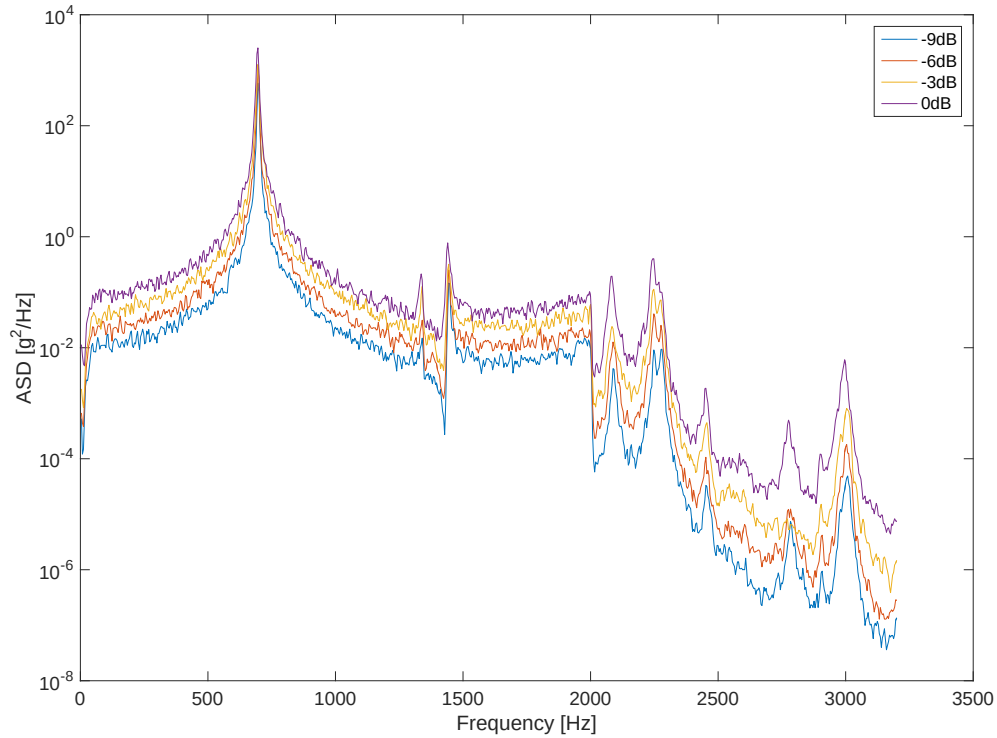


Figure 5.11: Response to random excitation in levels of -9 dB to 0 dB for the system (plate) without an attached damper over the whole tested frequency bandwidth, Acceleration Spectral Density vs. Frequency

The associated G_{RMS} values are shown in Table 5.3. Magnifying the plots shows the predicted differences in amplitude: the higher the input, the higher the response, as depicted in Figure 5.12. The interesting part is between 20 and 2000 Hz as extracted from the CubeSat constraint definition in Section 1.2.1.

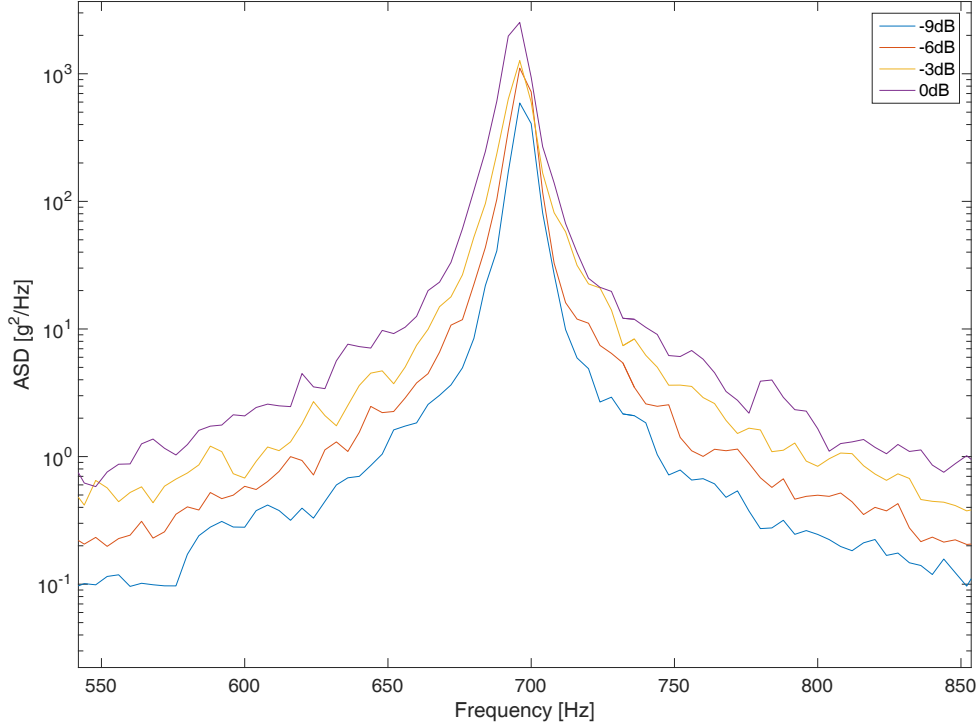


Figure 5.12: Response to random excitation in levels of -9 dB to 0 dB for the system (plate) without an attached damper, zoomed into the area where the first resonance peak occurs, 550-850 Hz

To calculate the G_{RMS} of the response, the following scheme from the NASA FEMCI Book [43] is used. The area below the ASD plot is divided into small bandwidths (here in bandwidths of $\Delta f = 4$). Now, the number of octaves is calculated by the equation

$$\#Octaves = \frac{\log(\frac{F_H}{F_L})}{\log(2)} \quad (5.3)$$

where F_L is the lower frequency of the current bandwidth and F_R is the upper one.

Afterwards, the decibel (dB) value is calculated by

$$dB = 10 * \log \left(\frac{ASD_H}{ASD_L} \right) \quad (5.4)$$

where ASD_H is the ASD value that belongs to the upper frequency while ASD_L is the ASD value that belongs to the lower frequency.

Now, the slope is calculated by dividing the number of dB by the number of octaves.

$$m = \frac{dB}{\#Octaves} \quad (5.5)$$

Thereafter, the area under the curve is calculated in the specified frequency bandwidth by

$$A = 10 * \log(2) * \frac{ASD_H}{10 * \log(2) + m} * \left[F_H - (F_L) * \left(\frac{F_L}{F_H} \right)^{\frac{m}{10 * \log(2)}} \right] \quad (5.6)$$

This equation becomes invalid for values of $m = -10 * \log(2)$ because this would result in a division by zero. In this case, the following equation is used for calculating the area under the curve

$$A = ASD_L * F_L * \ln \left(\frac{F_H}{F_L} \right) \quad (5.7)$$

This calculation is performed for all Δf steps. All single results are summed for calculating the area below the whole curve in the considered range.

$$A_{overall} = A_1 + A_2 + A_3 + \dots + A_n \quad (5.8)$$

while n is the number of areas of bandwidth Δf .

For calculating the G_{RMS} value, the square root of $A_{overall}$ is extracted.

$$G_{RMS} = \sqrt{A_{overall}} \quad (5.9)$$

The G_{RMS} values of the plate without an attached damper are shown in Table 5.3.

Table 5.3: G_{RMS} -Values of undamped system for different excitations

Excitation	-9 dB	-6 dB	-3 dB	0 dB
No Damper	72.63	99.53	115.52	167.96

The following discussion of the G_{RMS} results for different cavity sizes are compared to the undamped system. The G_{RMS} tables for the different cavity fillings are shown in the Appendix B.

5.6.2 Random Excitation Results

The results discussed in this section are based on the NASA GEVS random vibration excitation at 0 dB of 10 G_{RMS} . Further experiment results for other input levels can be found in Appendix B. The G_{RMS} value typically is reduced by adding more damping to the system, i.e. using a higher weight and fill-factor for the cavities.

The analysis of the G_{RMS} values reveals unexpected results. The addition of the 90%-filled, 5 mm-long cavity with a particle weight of 3.6 gm reduces the G_{RMS} value to 58.6 which is only 34.9% of the baseline systems' response. The bigger cavities with high fill-factors only add negligible improvements to the reduction of the G_{RMS} values while significantly increasing the weight penalty. A comparison of the 90%-filled 5 mm-long cavity and the 90%-filled 25.4 mm-long cavity can be found in Table 5.4. The slight decrease in G_{RMS} is not worth the high increase in weight penalty.

Table 5.4: Comparison of G_{RMS} -Values and weight influences of 5 mm-long and 25.4 mm-long cavities during random excitation

Cavity Size [mm]	G_{RMS}	% of undamped	m_p [gm]	β [%]
5	58.61	34.9	3.627	9.1
25.4	51.02	30.4	18.045	45.1

Comparing results between these two cavity sizes is of interest as well. First, the attachment of unfilled cavities improves the G_{RMS} value by at least 20% in all tested cases. Furthermore, the addition of particles up to a fill-factor of 30% does not add a significant improvement to the systems' damping. The addition of particles up

to a fill-factor of 60% improves the G_{RMS} by at least 10%; the only exception is the 5mm-long cavity where the addition from 30 to 60% only improves the G_{RMS} by 0.1%. This is due to changes in the system’s dynamic properties. First of all, the added cavity without any particles increases the stiffness of the system. By adding more particles to the system to achieve a specific FF, the weight of the damper increases, which causes the displacement to increase through the effect of the momentum. At the transition from a FF of 30 to 60%, both mechanisms neutralize each other which results in the described behavior.

Another significant finding is that the 90%-filled 5 mm-long cavity achieves a lower G_{RMS} response than the 90%-filled 10 mm-long cavity. This can be attributed to the additional weight in the 10 mm-long cavity that leads to increased amplitudes at higher modes and to inaccuracy in measuring and calculation techniques through the measuring resolution of the sample rate. The first mode’s amplitude should be lower for the heavier cavity as can be seen in the sine sweep results of Section 5.5.

The ability of the 25.4 mm-long cavity to reduce the G_{RMS} by almost 70% is significant because a reduction to 30.4% of the baseline system response means that the stresses the printed circuit board has to withstand in the overall frequency bandwidth of the launch environment are reduced effectively. This reduction leads to a higher probability of survival for the CubeSat PCBs during launch. Significant attenuation of the overall G_{RMS} can be achieved with as low as 3.6 gm of particle weight in the 5 mm-long cavity. Although the first mode is not damped as expected in sine sweep excitation, this low amount of weight makes a huge difference during random excitation by reducing the G_{RMS} value to 34.9% of the response of the undamped system.

5.7 Experimental Results Summary

The damping effect on the first resonant mode seems to increase with the total particle mass. Furthermore, the increase in the particle mass leads to a shift in the resonance frequency [13] for specific cases in which according to [31], the damping parameters are wrongly adjusted. Additionally, the gap size has a significant influence

on the damping behavior as well, as seen from the test of the glass-bead filled cavity in Section 5.5.3. If the gap becomes smaller, the damping ratio increases. A larger gap decreases the damping ratio because the particles do not interact enough with the ceiling of the cavity.

If the gap is misadjusted for a specific frequency range, rattling (or chaotic behavior) occurs due to the fact that some damper parameters are slightly mistuned [13]. This rattling leads to an increase in the acceleration response at frequencies where lower excitation is expected. The reason for this phenomenon is that the particles impact the ceiling of the cavity with a phase shift that supports the movement of the plate. Friend and Kinra analyzed this phase related behavior in [22]. A very good damping result is achieved for the 90%-filled, 25.4 mm-long cavity. The damping for this case is 8.66% while the random excitation response is reduced by 70%. However, the size of the cavity is very large compared to the narrow space inside of a CubeSat PCB assembly, and the weight penalty is 45.1%.

Reasonable damping results are achieved for a 15 mm-long cavity with a FF of 90%. The related damping is 5.57% while the weight penalty is only 26.8%. Furthermore, the random excitation response is reduced by up to 62.6%, which should allow the printed circuit board to better withstand the launch environment.

The collected data is next used to find an optimized solution to improve the damping ratio while simultaneously reducing the weight and size of the cavity. The addition of particles of only 10-15% of the system's fundamental modal mass as proposed by Brown in [24] is invalid for the specific case of attenuating the vibration of a PCB model of a PC/104 board used in CubeSats. By the addition of only 10-15% of the system's mass as particles, which results in an additional 3 to 4.62 gm, the particle damper would not reduce the vibration of the system as effectively as it is achieved in this experiment.

CHAPTER 6:

Particle Damper Cavity Parameter Optimization

The objective of this optimization effort is to find the best practical damping device to reduce the acceleration response for the first mode at approximately 650 Hz. Additionally, the reduction of the overall energy in the whole frequency bandwidth, i.e. the G_{RMS} , is of great importance and is evaluated using random excitations. It is desired to reduce the weight (for example, by using a different material that has comparable damping properties, such as glass beads), the size and footprint of the cavity to make it possible to use this device in the small space that may be available on a printed circuit board.

6.1 Using Curve Fitting To Obtain Trends In Particle Damping

A curve fit is used to find a trend where to find the optimum particle damper setup for damping a PC-board of type PC/104 in a CubeSat using a single particle damper cavity. The ‘poly53’ fit option in Matlab is used for creating a surface of damping over a mesh of weight and cavity length as well as weight and gap size, which turn out to be the most important influence factors when using very small particles. The damping ratio is calculated out of the sine-sweep (down sweep) results using the half-power bandwidth method as mentioned in Section 5.4.

For very low gap sizes (close to 0 mm), the particles are fixed in their positions, so they cannot dissipate enough energy to add sufficient damping to the system [31]. Almost no impact between the particles and the cavity occurs, so almost no momentum is transferred from the system to the particles. This leads to the assumption that a fully filled cavity (100%) adds no damping to the system and the only remaining damping is the system’s internal damping [31]. During the experiment performed for this thesis, it is found out that a fully filled cavity dissipates more energy than a partially filled one. The damping coefficient is highest for large cavities with very high fill-factors.

The ‘poly53’ fit creates the following equation that fits the data points by finding different weight parameters p that fit a surface over the weight (x) and length (y) mesh for the equation

$$\begin{aligned}
 damping(x, y) = & p00 + p10 * x + p01 * y + p20 * x^2 + p11 * x * y + p02 * y^2 + p30 * x^3 \\
 & + p21 * x^2 * y + p12 * x * y^2 + p03 * y^3 + p40 * x^4 + p31 * x^3 * y + p22 * x^2 * y^2 \\
 & + p13 * x * y^3 + p04 * y^4 + p50 * x^5 + p41 * x^4 * y \\
 & + p32 * x^3 * y^2 + p23 * x^2 * y^3
 \end{aligned}
 \tag{6.1}$$

The resulting surface is plotted in Figure [6.1](#).

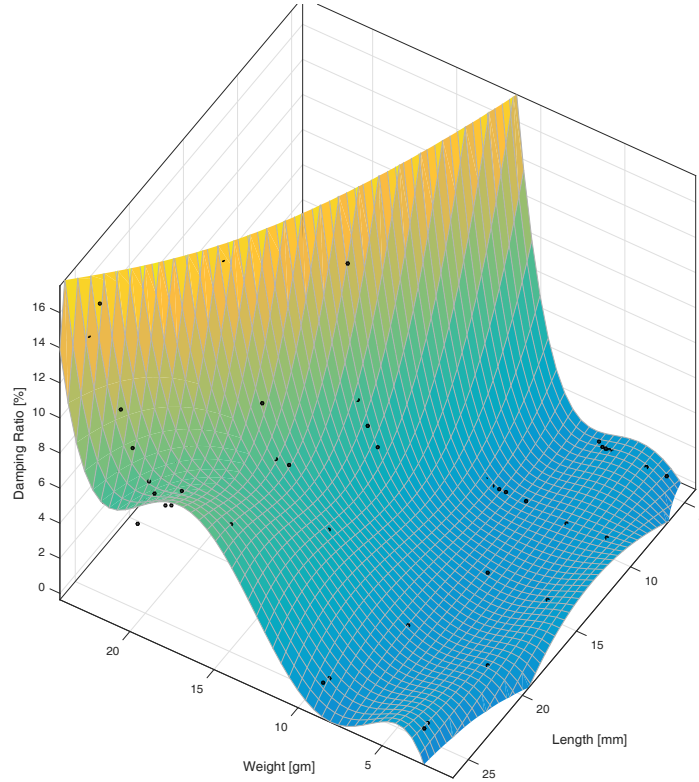


Figure 6.1: Curve fit Damping Ratio over cavity length and total cavity mass using 'poly53' polynomial fit, blue color represents a low damping ratio while orange represents a high damping value. Damping ratio increases for high length - and weight applications. The fill-factor increases for a fixed cavity length when adding particles.

The option of least absolute residuals (LAR) robustness ensures that all data points are equally important. This is accomplished by minimizing the sum of the absolute errors. This method can only be used if very few outliers are expected, which is the case for this data set.

The data set shows that the damping ratio increases with the fill-factor (FF) of the damper cavity. Contrary to the findings presented in [31], the damping ratio does not decrease above a certain fill-factor until it becomes the same value as the internal damping without an attached damper. The difference in behavior could be due to that the full 100% FF is not reached because of the air spaces between the particles,

that the polycarbonate rapid prototyping material is different from the material investigated by Sánchez, or both. The flexible polycarbonate cavities might contribute to the damping ratio because of its more compliant material properties. For example, the Young's modulus of PC is 2.0 - 2.4 GPa, which is around 96% lower than that of aluminum. The 100%-filled, 25.4 mm-long PC cavity has a peak acceleration response amplitude of 2.543 g at a frequency of 452.8 Hz, whereas a cavity made out of aluminum has a peak amplitude of 5.397 g at a resonance frequency of 378.2 Hz. For the aluminum cavity, no rattling occurs in frequency bandwidths of around 200-800 Hz which leads to the conclusion that the PC material could be a cause for the rattling behavior at frequencies between 200-800 Hz. The results for the aluminum cavity can be seen in Figure 6.2.

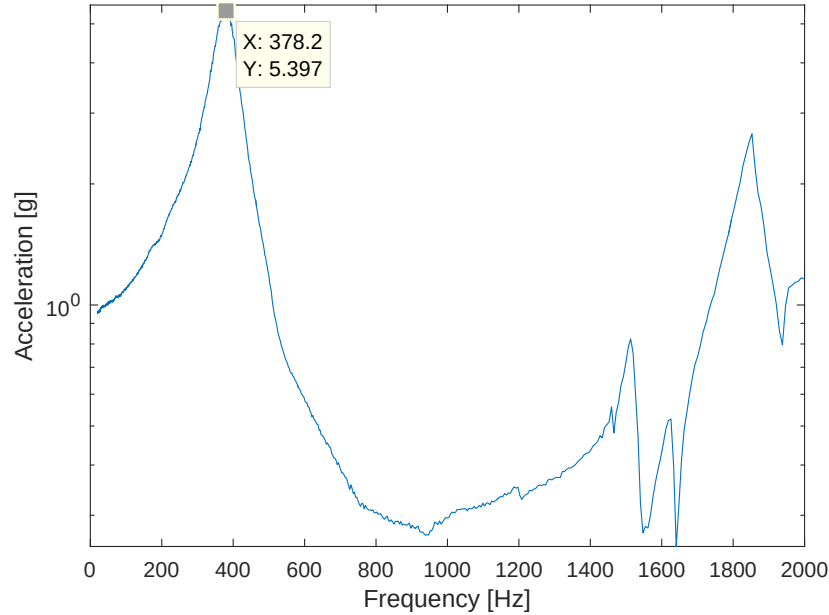


Figure 6.2: Acceleration-Frequency Plot Aluminum Cavity 100% steel shot

The significant downshift of the first modal frequency for the aluminum cavity is caused by the high added weight and the mistuned gap according to Figure 3.3. Additionally, the resonance around 1800-1900 Hz is amplified significantly. This leads to the conclusion that PC can improve the damping behavior with its material prop-

erties and therefore is a very suitable material for particle damping devices.

However, it has to be further investigated whether the fatigue cycling on the polycarbonate cavity decreases its damping abilities or leads to defective cavities by carrying out a long-duration test of PC particle dampers.

Another method that is investigated for finding an appropriate damping solution (i.e. all parameter combinations to reach a desired damping ratio) is to use a contour plot that shows the x-y-area (weight-length-area), which fulfills the requirement of a specific damping ratio. This can be done using the MATLAB © Curvefit Toolbox.

Afterwards the peak amplitude has to be investigated. As seen in the sine sweep results, the system with the attached 15 mm-long cavity with a fill factor of 90% has a damping of 5.57%, which results in a peak acceleration amplitude of only 4.68 g. The big advantage of this damper is that it only has a weight of 13.2gm while 4.68 g is only 2.1% of the undamped system's peak acceleration amplitude. To reduce the weight further, the 20 mm-long cavity with 60% fill can be used. It has a slightly lower damping ability with damping of 4.08%, which results in a peak acceleration amplitude of 6.191g. The advantage is the reduced weight of only 12.8 g. 6.191 g is just 2.8% of the undamped system's peak acceleration amplitude which is a reduction by 97.2%. If a certain maximum amplitude is chosen, a surface can be plotted using the contour plot that defines all possibilities of connection between weight and gap that fit, so that a specific amplitude, that could lead to damage to the system, is avoided. The basic plot for this purpose is shown in Figure 6.3. The amplitude, which needs to be reduced, decreases with an increasing weight and a decreasing gap size. With low weight and a large gap (and therefore a low fill-factor(FF)), no high damping can be obtained and the oscillation is not attenuated enough to fit the requirements of the damping device (DD).

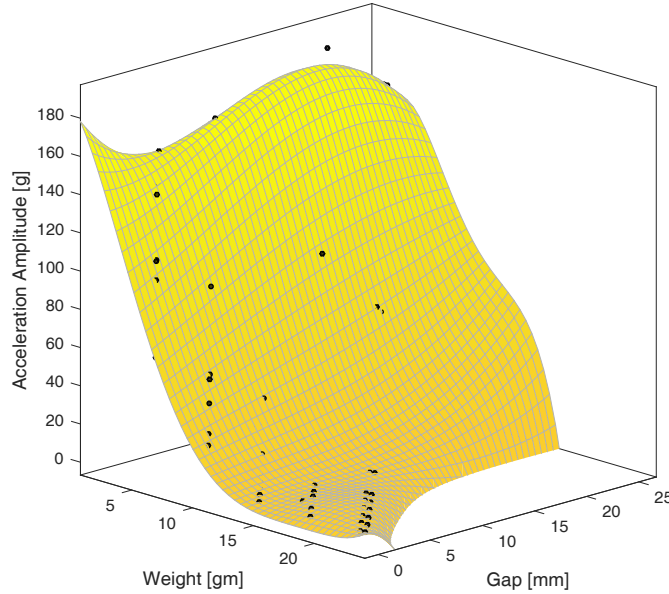


Figure 6.3: Curve Fit of Amplitude over Weight and Gap; Acceleration Amplitude decreases with higher weight and smaller gap

A result of the curve fit in Figure 6.1 and Figure 6.3 is that the most significant damping occurs with fill-factors (FF) of around 60-100%. Therefore a higher data-set resolution with FF of 75, 80, 85, 90, 95 and 100% is used in this specific area. The 100% FF results are used to verify the results of other papers that mention that a 100% filled cavity does not add any significant damping to the system and behaves only like an added mass.

The results of this experiment are contrary to those found in the paper published by Sánchez et al. in 2011 [31]. Another important plot that underlines the significance of a small gap size (independent of total cavity size) is shown in Figure 6.4. For this surface fit, a ‘poly53’ fit is used again. It is noticeable that the damping ratio increases for a lower gap and a higher total weight. The task is then to decrease the

size of the cavity in dimensions that are in conflict with the space on a PCB in a CubeSat and use other configurations to increase the weight capacity of the damping device.

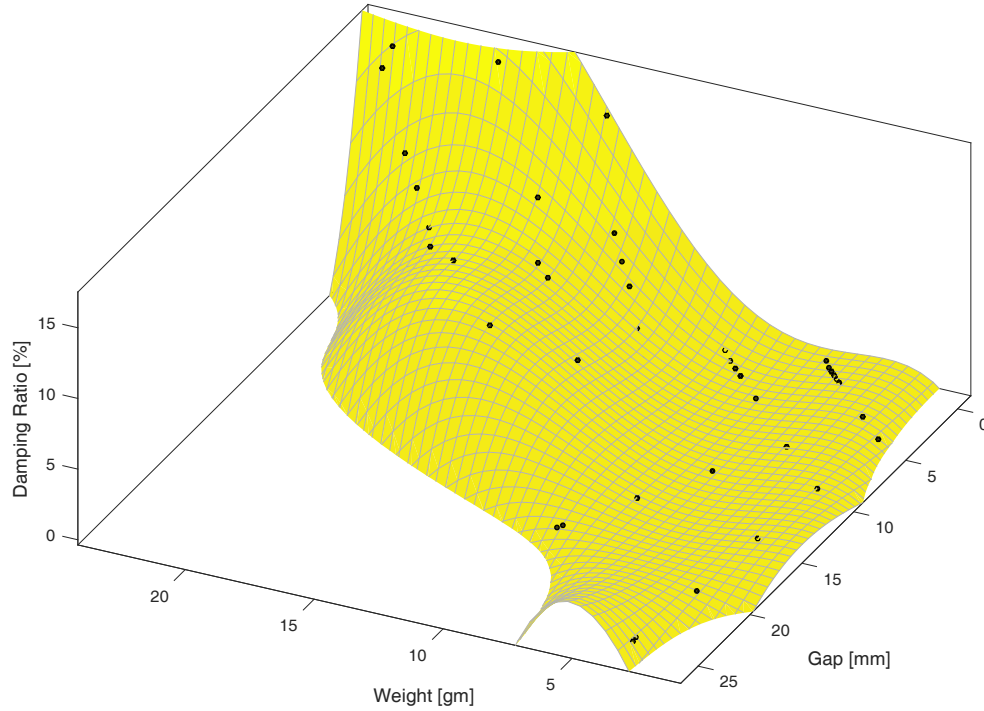


Figure 6.4: Curve Fit of Damping Ratio vs. Weight, Gap using ‘poly53’ polynomial fit; smaller gap and higher weight increase the damping ratio

6.2 Using damper pattern

As mentioned in [28], the use of a “multiple cell enclosure” can improve the damping capabilities of a particle damper by approximately 18 to 24%. Additionally, it is possible to reduce the single cavity’s size (especially height, which is of great relevance) by using a pattern of many small particle dampers. The reduction in height is desirable to satisfy the volume constraint in CubeSats. The pattern is designed to mitigate the displacement of the first mode of the system (close to the center of the PC/104 board). For this purpose a circular pattern concentrated at the middle

of the plate (surrounding the first mounting point from previous testings) is designed and machined; this is shown in Figure 6.5. Furthermore, this kind of damping is able to reduce, at least partially, the amplitude of the other resonant modes due to the fact that the dampers are somewhat located in areas which are affected by the higher modes as depicted in Figure 2.2. The plate's resonant frequency for the first mode is at 691.3 Hz with an acceleration response amplitude of 203 g. This is comparable to the response of the plate with only one hole in the middle.

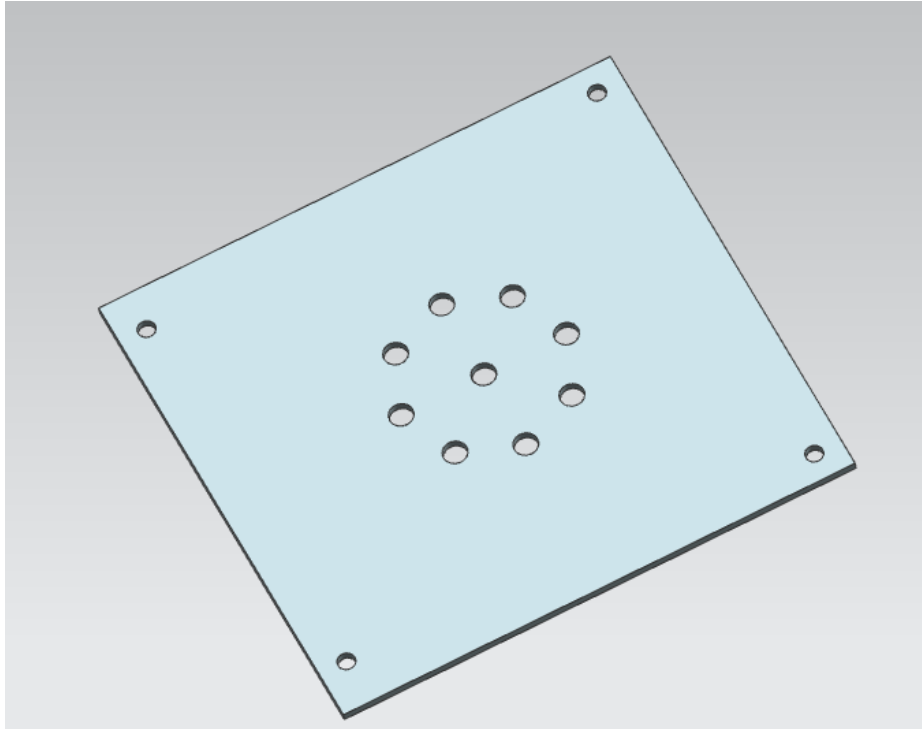


Figure 6.5: PC/104 model with drilled mounting holes for mounting of damper pattern

Additionally, [28] mentions the possibility of reducing the necessary weight to gain the same or even better damping results. The cavities designed for this purpose have an inner total volume of 2052 mm³. This results in a maximum weight at 90% filling of approximately 9 gm. They are again attached to the plate by using #8-32 screws. The aim is to reduce the weight to under 13.2 gm as needed for 5% damping with a single 15 mm-long cavity as mentioned in Section 5.5.2. Simultaneously, the height of the cavity is reduced by 10 mm. For this purpose smaller dampers are designed; The

cross sectional view is shown in Figure 6.6. The green area shows the solid material while the upper grey area shows the cavity with a length of 5 mm. The lower grey area shows the hole for the #8-32 screw to connect the PD to the board. The overall length of the damper, including the cap, is 12mm.

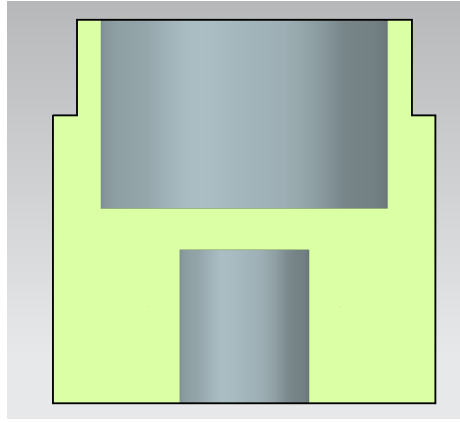


Figure 6.6: Shape of Cavity for Pattern of Particle Dampers to Reduce the Height of Single Dampers and to increase the performance

The result of this experiment shows that the acceleration amplitude is reduced to 43.43 g by adding 9 gm of weight to the system. Simultaneously, the resonance frequency shifts from 691.3 Hz down to 467.3 Hz as shown in Figure 6.7.

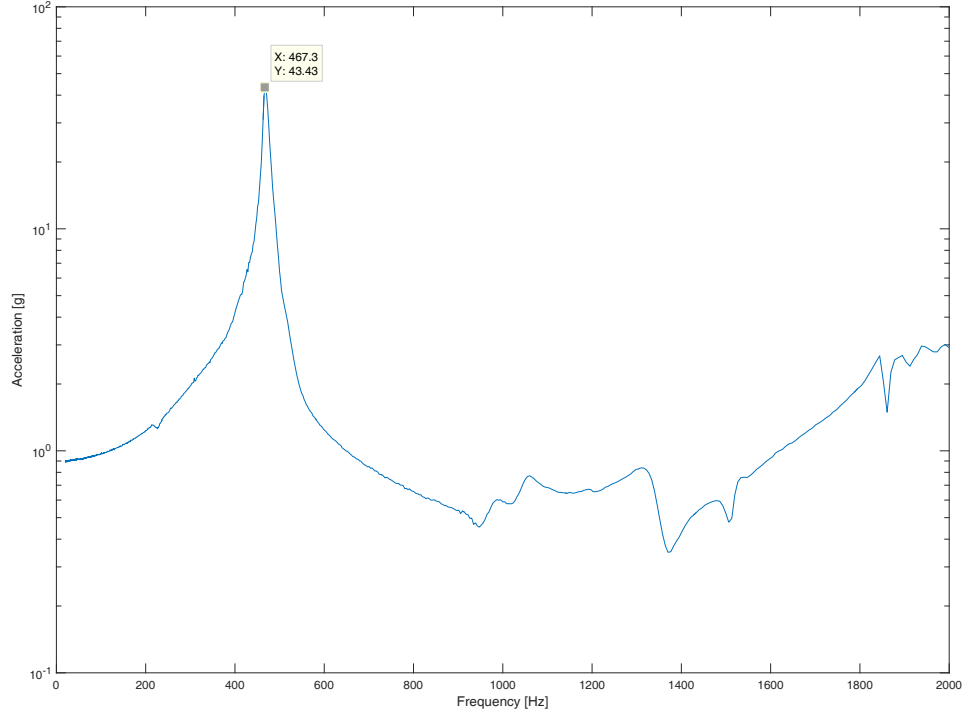


Figure 6.7: Acceleration-Frequency-Plot for Damper Pattern

Compared to the damping device where a single cavity with a particle mass of around 9 gm is added (length 20 mm, FF 60%), the acceleration is not attenuated as efficiently. The damping is 1.13%, which is much lower than the 4.08% of the comparable weight damper. The higher modes are not better damped either. The only advantage of the pattern for a sine excitation is the reduced cavity size in length. The predicted improvement of the damping capabilities by [28] were proven not to hold true for this application and excitation. The occupied area on the plate is larger than for the single cavity application.

However, the response due to the 10 G_{RMS} random excitation is 34.3461, which is 20.44% of the undamped system's value. This configuration produced the best results although the damping ratio is lower.

6.3 Dampers designed with standoff

Another design that is investigated to attach the damping cavities is to incorporate standoffs to minimize the area needed on the PC/104-board either external or internal to the damper. The cross-section view of a design for a damper that fits the needs of a reduced footprint is shown in Figure 6.8. This design decreases the needed space on the printed circuit board while simultaneously decreasing the length of the cavity but still adding enough weight with a specific gap as found out in this experiment by using a standoff.

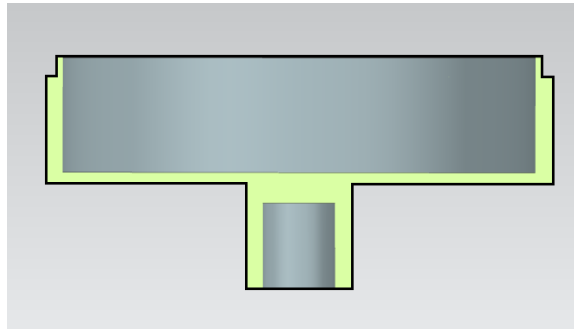


Figure 6.8: Damper designed with standoff

This damper has a volume of 2200 mm^3 , resulting in a 90%-filled cavity with a total particle weight of 15.94 gm. The overall length of the damper, including the cap, is only 13 mm, which is small compared to the dampers used before and therefore more suitable for the mentioned application. The cavity itself has a length of 5.5mm while the radius is 11.6mm. The shift in CG away from the PCB could lead to undesirable stress on the board, but this may be negated by the additional damping benefits achieved with the PD. This cavity is attached at the center of the plate. The cavity's total weight, including the cap, is 1.2 gm. The fill factor is about 72% which translates to a particle weight m_p of 7.4 gm. The damping of this setup is 14.6% as calculated using the Half-Power Bandwidth Method. The weight penalty β is only 18.5%.

As mentioned in Section 5.5.2, a higher minimum weight is needed to achieve proper damping ratios in other damping configurations. The damping in this case is significant considering the small amount of added mass. The acceleration response in the

frequency domain can be seen in Figure 6.9.

An addition of weight would cause the resonance frequency to shift down while a higher stiffness would result in it shifting up. A higher stiffness is not expected through the smaller base area of the cavity contacting the oscillating plate. However, the first resonance frequency shifts up to 846.5 Hz which is significantly higher than the resonance frequency of the undamped system. As described in Figure 3.3, there is an area where the effective mass displayed by the system is much lower than the mass of the primary system [31], which would cause a shift of the resonance frequency to higher values.

It is also noted that in this configuration, the acceleration amplitude at the resonance frequency is reduced to 4.762 g, using only half of the weight of the 20 mm-long cavity filled with 75% of steel shot, weighing around 15 gm.

The overall CubeSat benefits from the increase in the first modal frequency. The other resonance frequencies are not amplified by this setup and it does not seem to make any difference whether a standoff is used for stability reasons.

The movement of the center of gravity may cause problems if the excitation is not in the axial (z) direction. A disadvantage of this setup is the high amount of rattling in the lower frequencies (around 20-600 Hz). This effect needs to be investigated further and to be attenuated if possible.

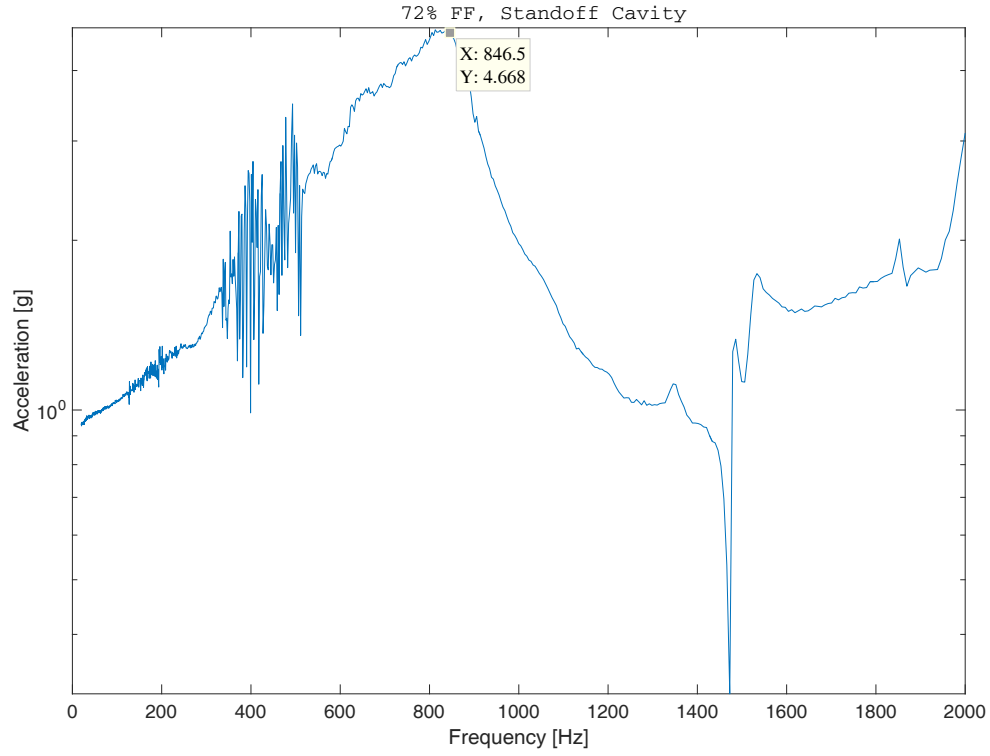


Figure 6.9: Acceleration-Frequency-Plot for Damper with Standoff

The random excitation with an input of $10 \text{ G}_{\text{RMS}}$ causes a response of $50.6 \text{ G}_{\text{RMS}}$, which is only 30.11% of the undamped system's response. This is the best result obtained for all configurations of single particle damper that focus on the anti-node of the plate.

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CHAPTER 7:

Conclusion

CubeSat PCBs are adversely affected by the vibration of the launch vehicle during flight to orbit, resulting in possible damage to the sensitive electronic parts and connections.

The aim of this thesis was to investigate the behavior of PCBs, represented by an aluminum model of a PC/104 printed circuit board, and to find a damping solution that reduces the stresses and vibration responses on the board.

The particle damping method was investigated for the use on CubeSat printed circuit boards. Damping is affected by several parameters, predominantly the weight of the particles inside the device and the size of the gap between the top of the particle bed when at rest and the cap of the cavity. Damping increases as the weight rises and the gap narrows. Different optimization efforts were developed and introduced improving the behavior of particle dampers even further, making them more suitable for the environment of a printed circuit board.

A reduction of up to 98.8% for the peak amplitude value in a sine sweep excitation and a reduction of up to 69.6% for random excitation in the overall frequency bandwidth is achieved. These significant reductions were obtained using a 25.4 mm-long cavity and a fill-factor of 90%. Other damper setups can achieve good results as well while reducing the damper's mass and size, but the standoff cavity has the best performance; it provides a high damping ratio using a very low weight and a very small footprint compared to other damping designs. The smaller footprint insures adequate room for electric components on a PCB inside the CubeSat. A pattern of dampers achieves reductions up to 79.55% for random excitation, but does not attenuate the sine excitation response properly and needs more room on the PCB.

The results improve the capabilities of a CubeSat developer to make sure that the designed printed-circuit-boards can withstand the environmental conditions a CubeSat

has to handle during flight to orbit. The application of particle dampers for printed-circuit boards specifically allows the use of highly sensitive electrical components by improving the likelihood that they will withstand the CubeSat launch environment. This gives CubeSat the potential to be high-end research devices used, for example, in large constellations. Additionally, this damping mechanism costs approximately \$10 for a polycarbonate cavity filled with S110 steel shot and is, therefore, only 0.02% of the price of an existing whole spacecraft isolation system. Furthermore, it is easier to mount than most other damping devices because only a single screw is needed to affix the cavity on the printed circuit board.

It is shown that polycarbonate particle dampers fit perfectly for damping of CubeSat PCBs and that they are the lowest cost option currently available. In addition, their design is simple and robust and they can withstand most challenging environments. Furthermore, they can meet the CubeSat design requirements.

7.1 Recommendations for Future Work

Many of the ideas presented in this thesis need to be investigated further. It is of great importance to evaluate whether different particles, in material and size, can improve the damping behavior while reducing the weight of the whole system or reducing the occupied volume in a CubeSat while maintaining the same weight, which would necessitate the use of denser particles.

Another idea to reduce the overall weight of the damping device is to use smaller screws to attach the dampers to the plate, which is feasible for very small cavities, like the ones used for the damper pattern, through the reduced weight.

Additionally, the option to use a damper with a standoff has to be investigated further by performing testing in the lateral excitation directions to determine whether the movement of the CG is fatal to the system's response in these cases or whether it leads to damaged cavities caused by the momentum associated with the PD.

By adjusting the cavity dimensions on a standoff an optimized shape of a stand-

off PD is to be found that increases the attenuation ability further but does not affect the electrical components on the PCB. Different cavity shapes should be investigated to maximize the damping. For this purpose, a bigger diameter could be an improvement because it contains a higher particle mass although it does not need a lot of space in the critical dimensions of overall length and standoff base diameter.

Furthermore, there has to be an investigation that forces the verification (including an evaluation of the PC-materials properties) of flight safety for the particle dampers attached to the printed circuit boards. It has to be further tested in low gravity environments due to the findings that particle damping could work totally different in space. The next step is to instrument these devices during launch and on orbit to measure the excitation and the response of the most sensitive printed circuit boards with the attached dampers; this would allow further investigation on whether particle damping works properly even in low gravity environments.

Additionally, a simulation model could improve the development of particle damping applications for specific printed circuit boards for fine tuning to specific builds. The PCBs that hold the gyros would most likely profit from different PD applications than a regular PCB with only processors mounted to it. The simulation could include an adjustable PCB model for tuning the parameters for different types of PCB and then to analyze the behavior of the board with an attached damper that can be changed by adjusting the FF and the shape of the cavity.

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APPENDIX A:

Sinesweep-Results

This chapter shows the figures of the sine sweeps for all tested cavities with different fill factors. The headline includes the fill-factor and the weight of the total particle mass contained by the cavity. The acceleration-amplitude is plotted vs. the frequency. The down sweep figures are used. The data-tips show the peak amplitudes values in acceleration and frequency of the first resonant mode of the particle damping assembly which is the mode to be tackled by the use of particle dampers due to its' high mass participation factor of around 77%.

A.1 Damper with 5mm cavity length

A.1.1 Unfilled

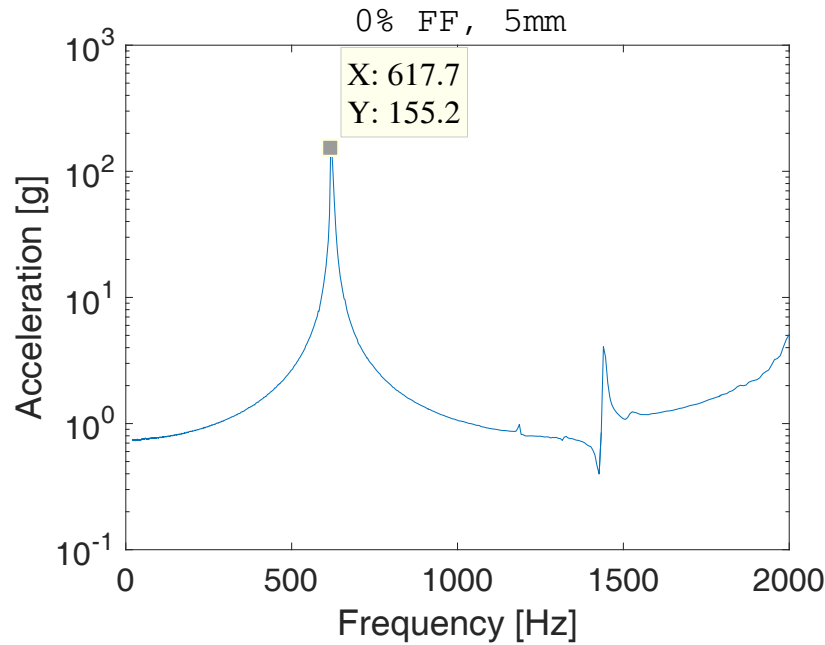


Figure A.1: 5mm Cavity down sweep

A.1.2 30% filling - 1.209gm particle mass

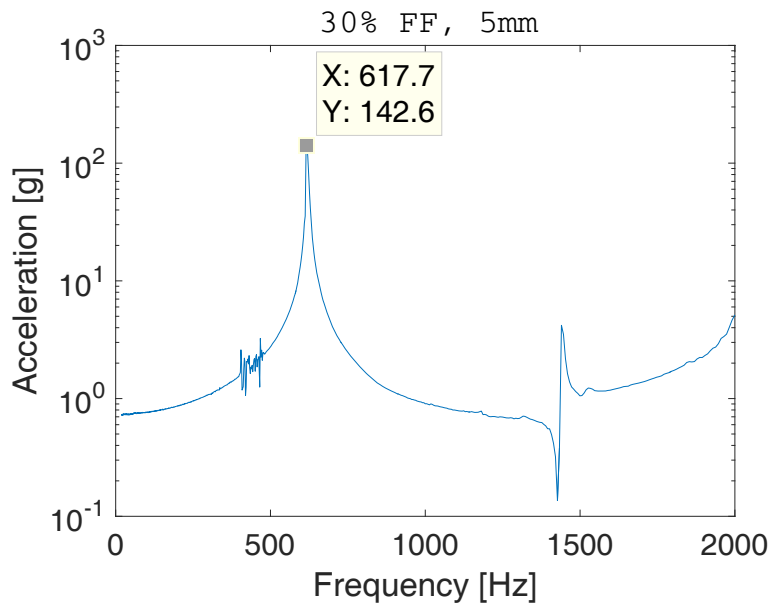


Figure A.2: 5mm Cavity filled with 30% steel shots

A.1.3 60% filling - 2.418gm particle mass

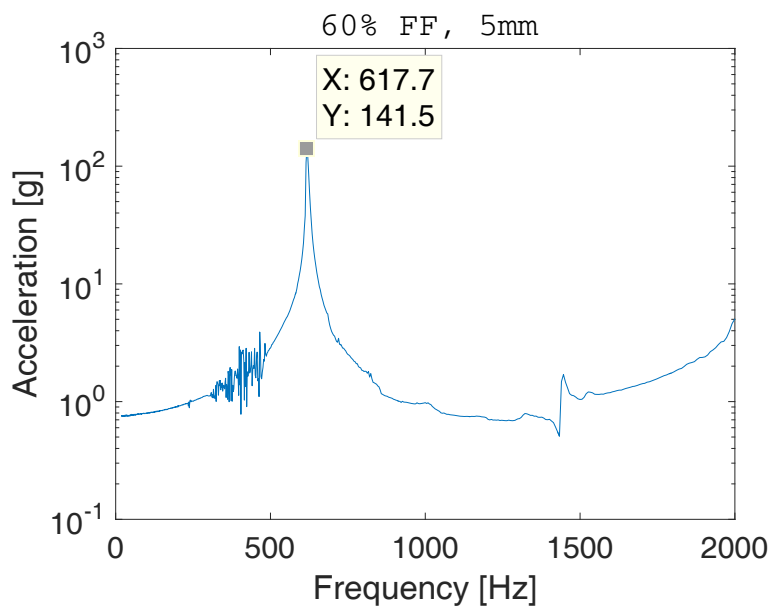


Figure A.3: 5mm Cavity filled with 60% steel shots

A.1.4 75% filling - 3.0225gm particle mass

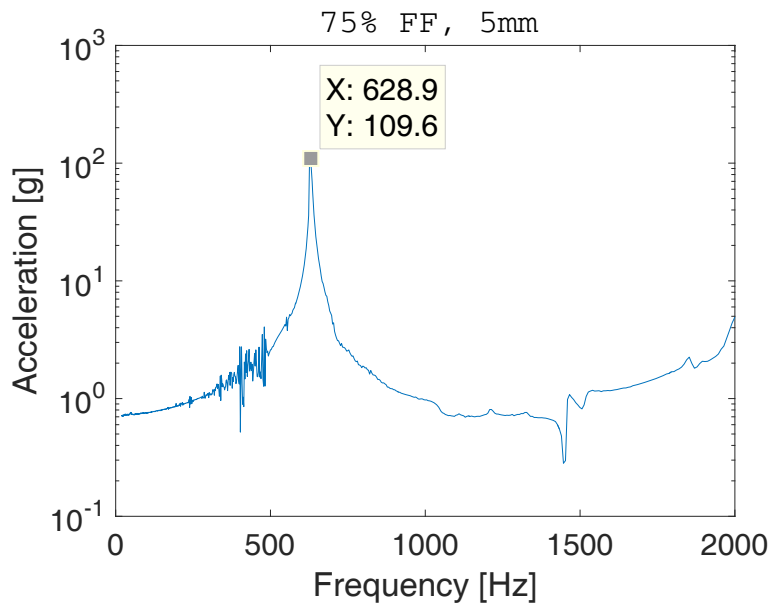


Figure A.4: 5mm Cavity filled with 75% steel shots

A.1.5 80% filling - 3.224gm particle mass

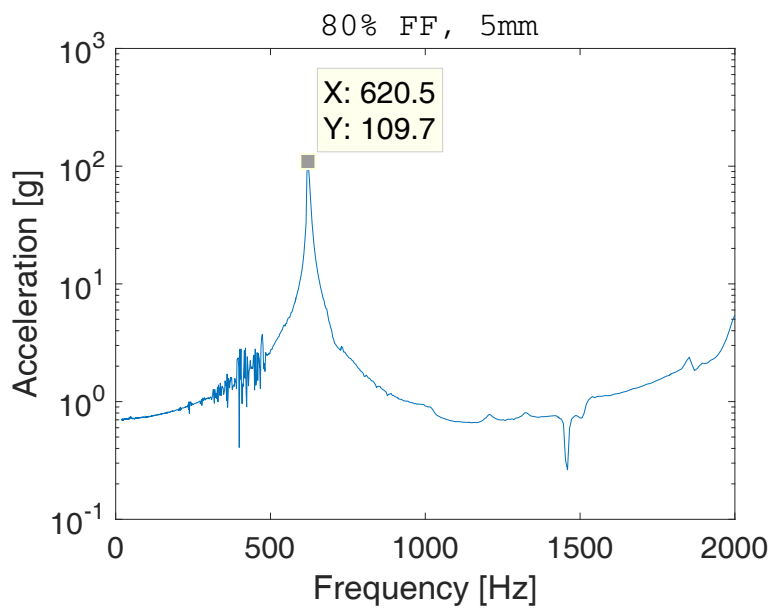


Figure A.5: 5mm Cavity filled with 80% steel shots

A.1.6 85% filling - 3.4255gm particle mass

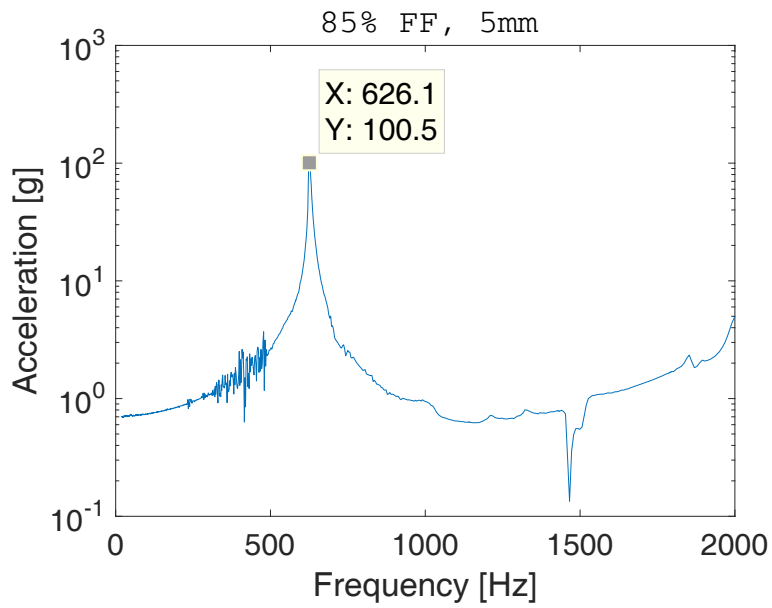


Figure A.6: 5mm Cavity filled with 85% steel shots

A.1.7 90% filling - 3.627gm particle mass

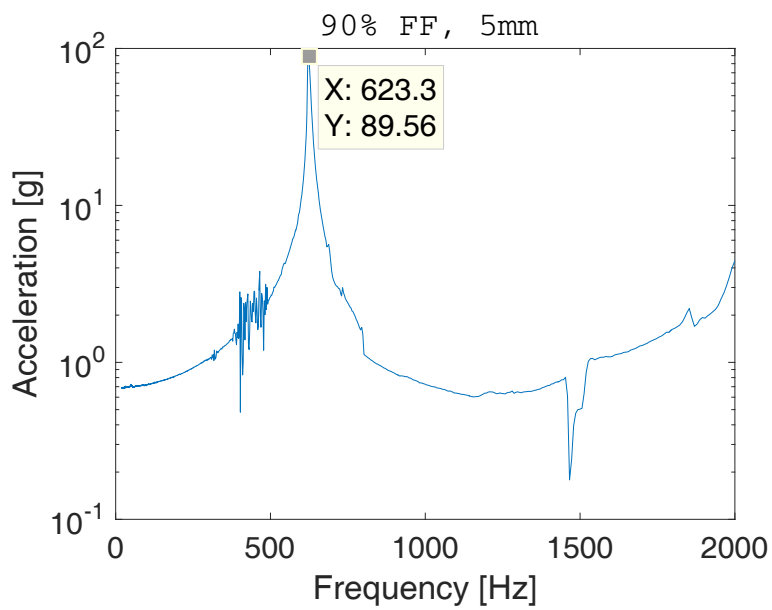


Figure A.7: 5mm Cavity filled with 90% steel shots

A.1.8 95% filling - 3.8285gm particle mass

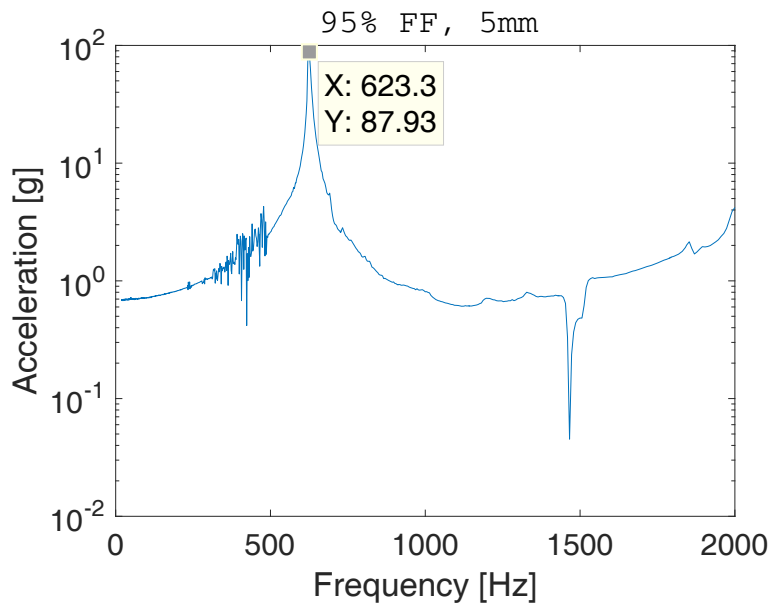


Figure A.8: 5mm Cavity filled with 95% steel shots

A.1.9 100% filling - 4.03gm particle mass

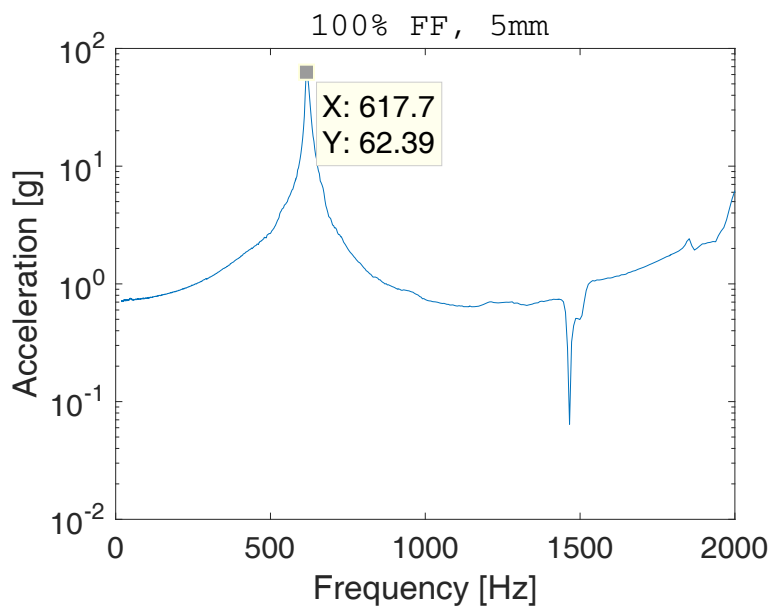


Figure A.9: 5mm Cavity filled with 100% steel shots

A.2 Damper with 10mm cavity length

A.2.1 Unfilled

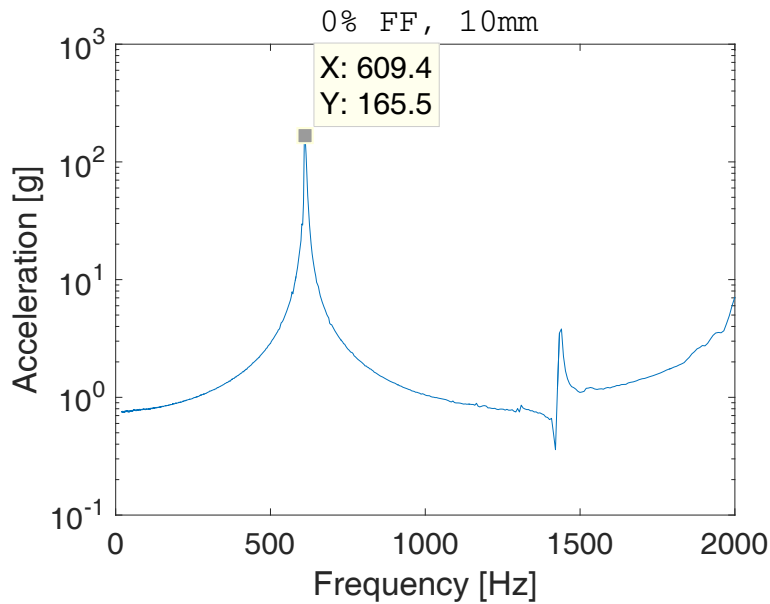


Figure A.10: 10mm Cavity filled with 0% steel shots

A.2.2 30% filling - 2.4gm particle mass

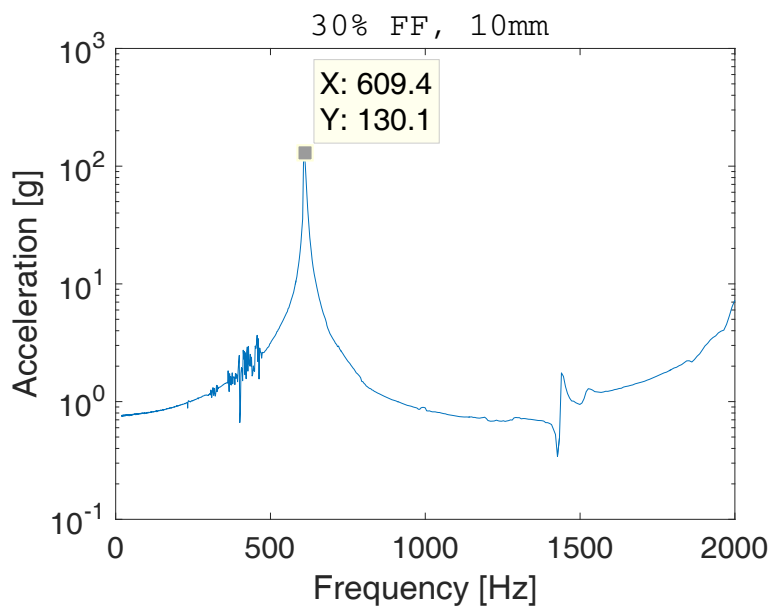


Figure A.11: 10mm Cavity filled with 30% steel shots

A.2.3 60% filling - 4.8gm particle mass

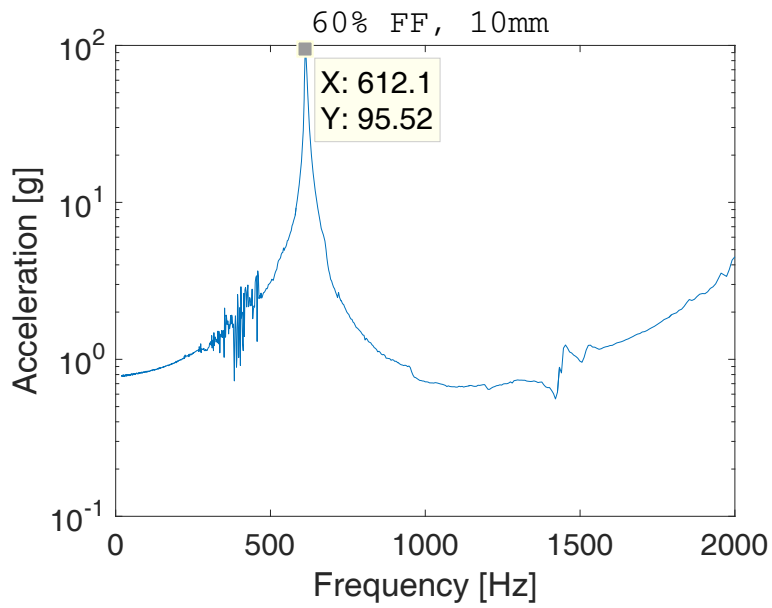


Figure A.12: 10mm Cavity filled with 60% steel shots

A.2.4 75% filling - 6gm particle mass

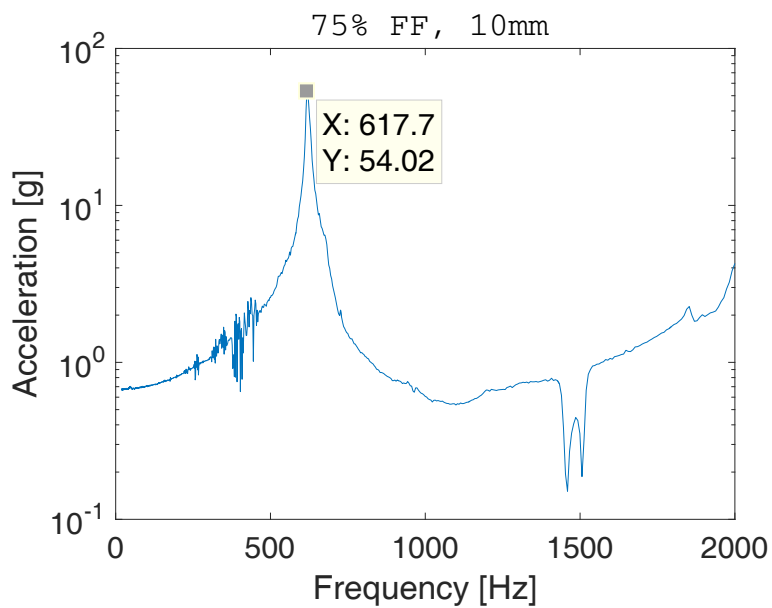


Figure A.13: 10mm Cavity filled with 75% steel shots

A.2.5 80% filling - 6.4gm particle mass

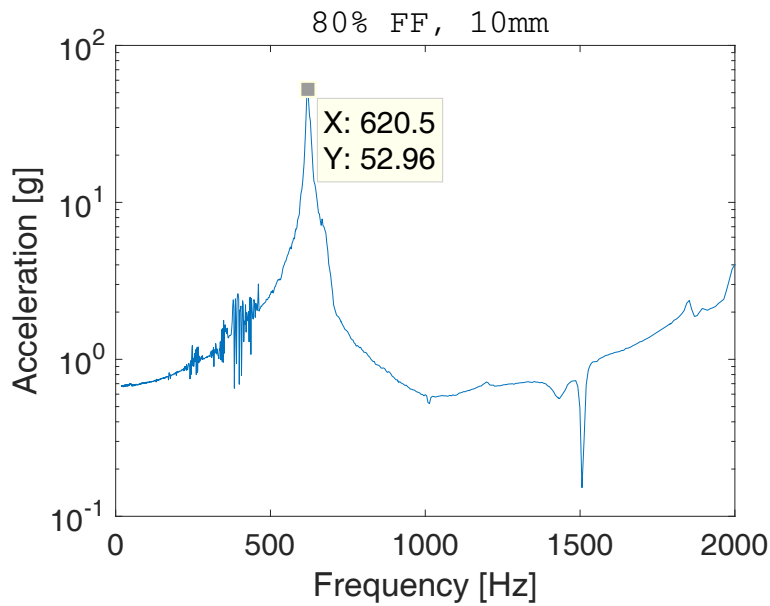


Figure A.14: 10mm Cavity filled with 80% steel shots

A.2.6 85% filling - 6.8gm particle mass

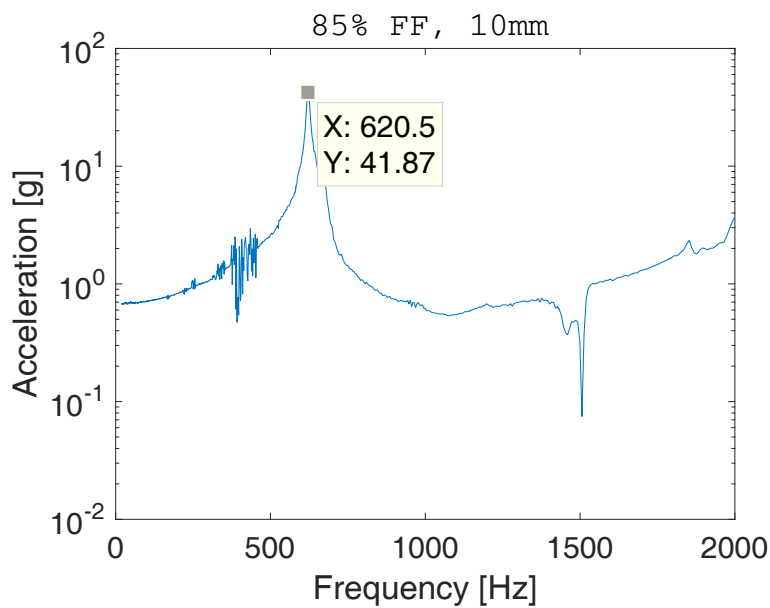


Figure A.15: 10mm Cavity filled with 85% steel shots

A.2.7 90% filling - 7.2gm particle mass

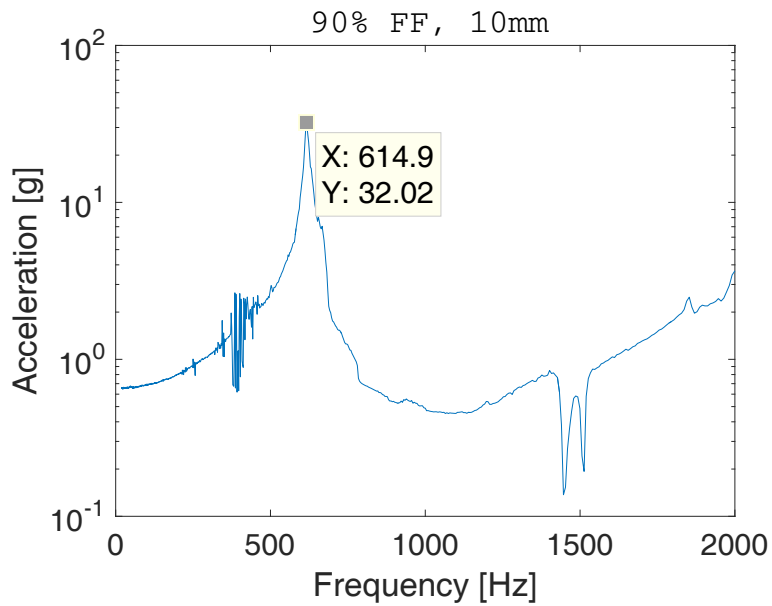


Figure A.16: 10mm Cavity filled with 90% steel shots

A.2.8 95% filling - 7.6gm particle mass

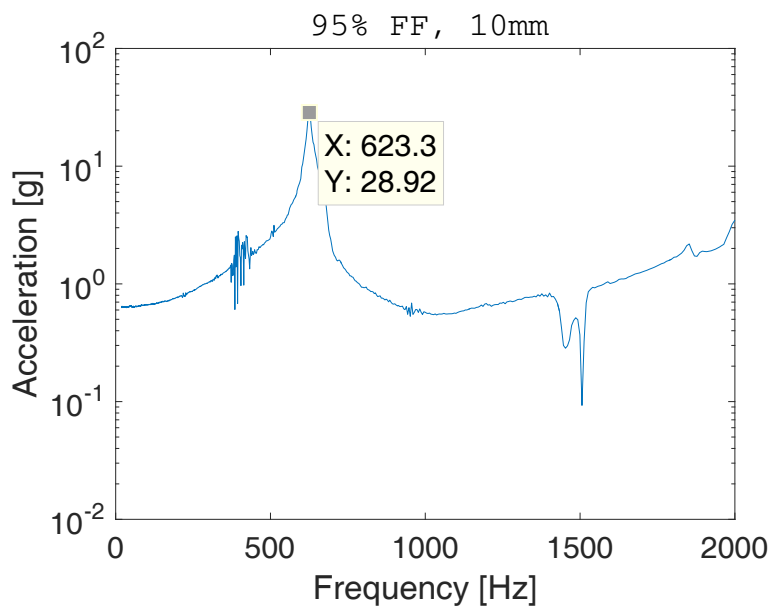


Figure A.17: 10mm Cavity filled with 95% steel shots

A.2.9 100% filling - 8gm particle mass

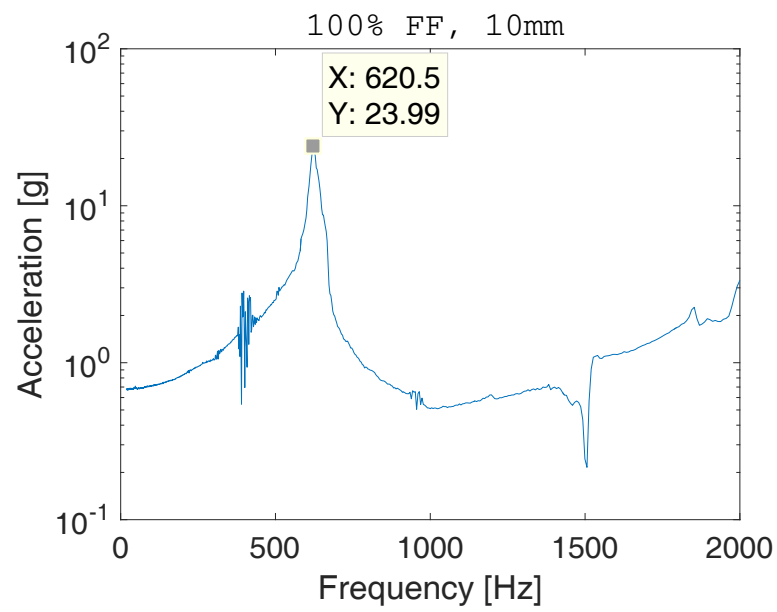


Figure A.18: 10mm Cavity filled with 100% steel shots

A.3 Damper with 15mm cavity length

A.3.1 Unfilled

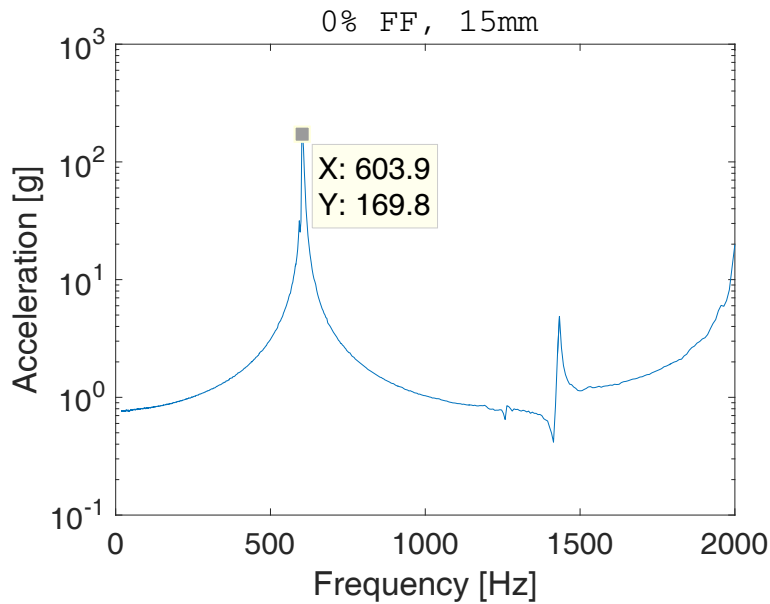


Figure A.19: 15mm Cavity filled with 0% steel shots

A.3.2 30% filling - 3.567gm particle mass

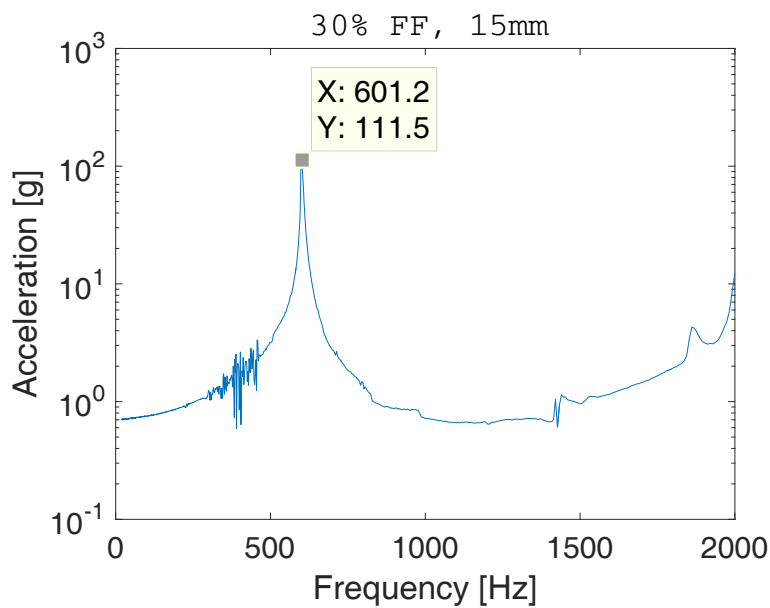


Figure A.20: 15mm Cavity filled with 30% steel shots

A.3.3 60% filling - 7.134gm particle mass

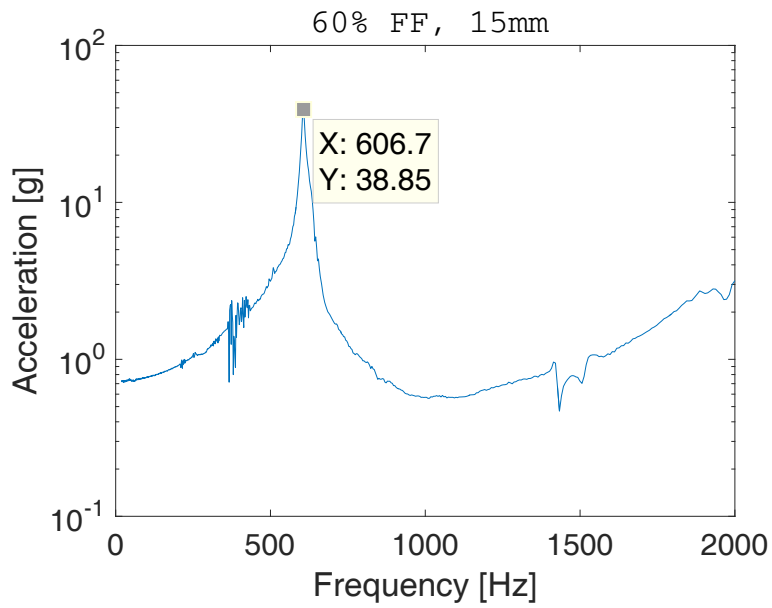


Figure A.21: 15mm Cavity filled with 60% steel shots

A.3.4 75% filling - 8.9175gm particle mass

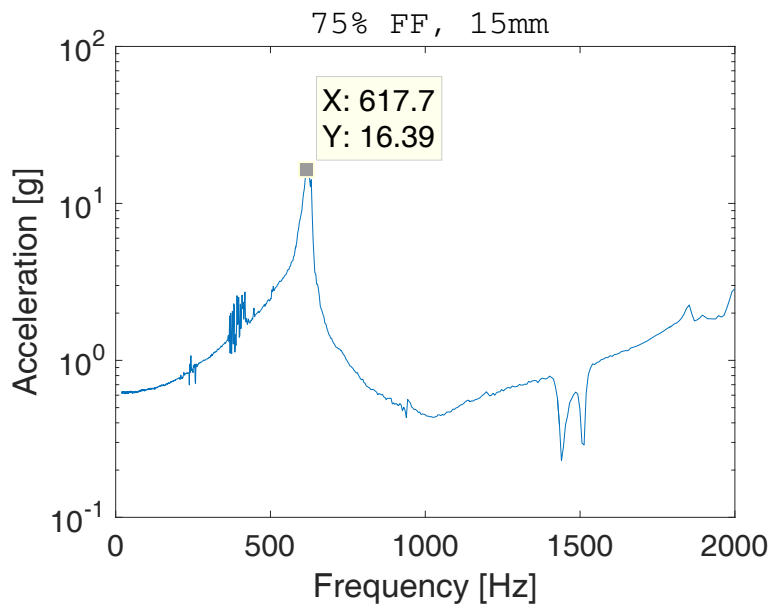


Figure A.22: 15mm Cavity filled with 75% steel shots

A.3.5 80% filling - 9.512gm particle mass

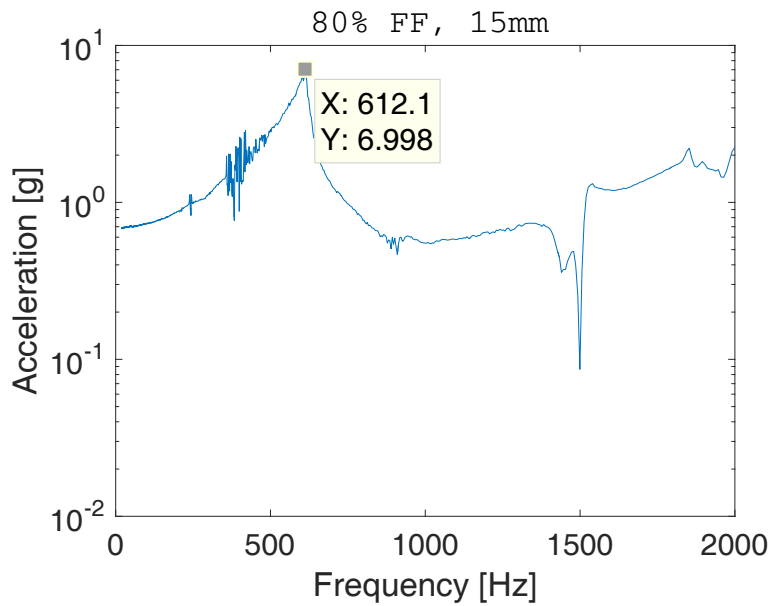


Figure A.23: 15mm Cavity filled with 80% steel shots

A.3.6 85% filling - 10.1065gm particle mass

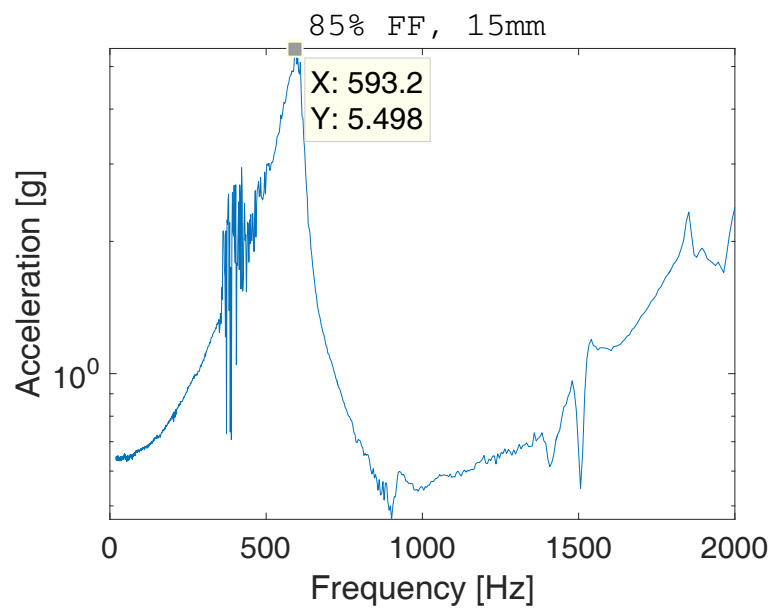


Figure A.24: 15mm Cavity filled with 85% steel shots

A.3.7 90% filling - 10.701gm particle mass

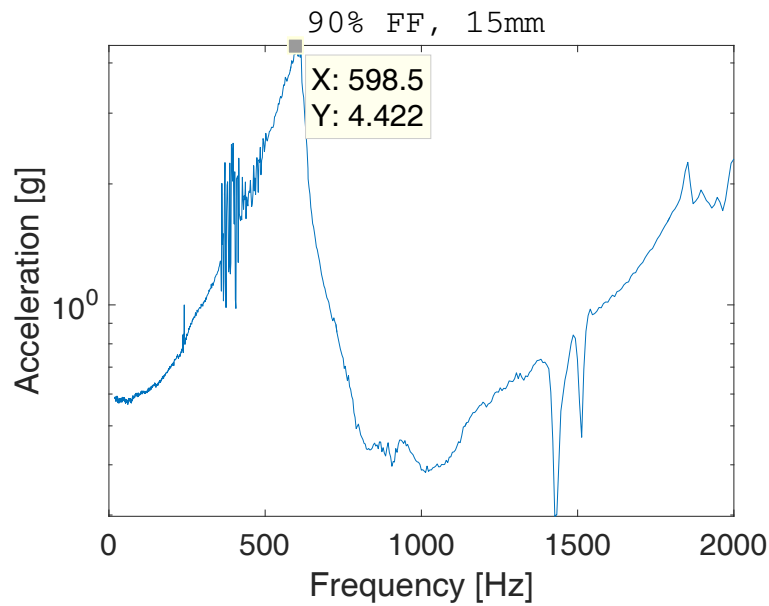


Figure A.25: 15mm Cavity filled with 90% steel shots

A.3.8 95% filling - 11.2955gm particle mass

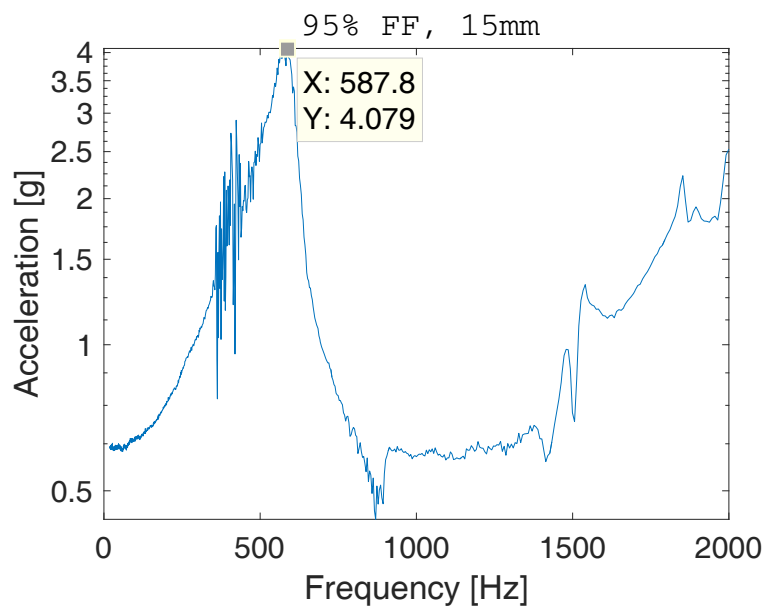


Figure A.26: 15mm Cavity filled with 95% steel shots

A.3.9 100% filling - 11.89gm particle mass

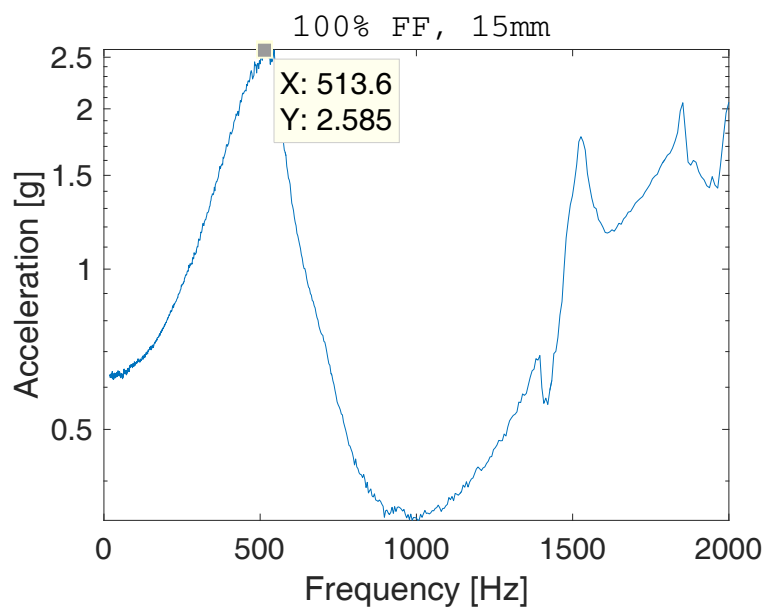


Figure A.27: 15mm Cavity filled with 100% steel shots

A.4 Damper with 20mm cavity length

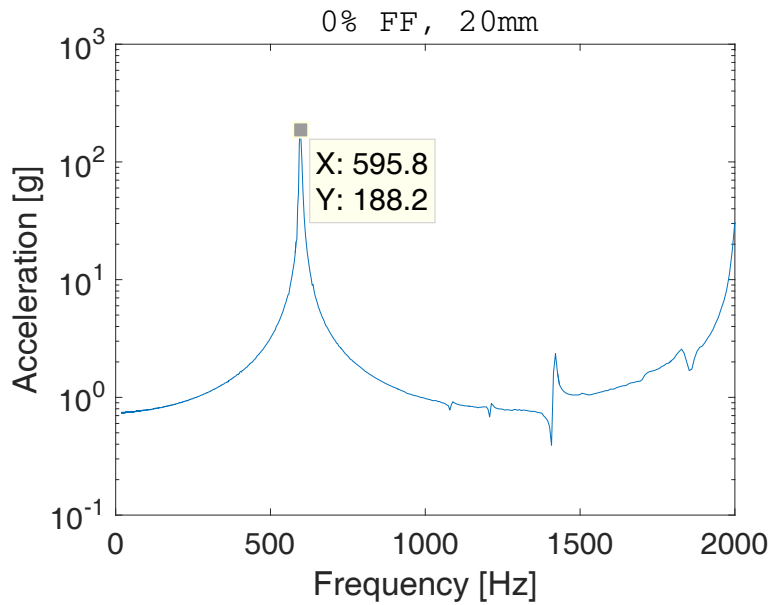


Figure A.28: 20mm Cavity filled with 0% steel shots

A.4.1 30% filling - 4.725gm particle mass

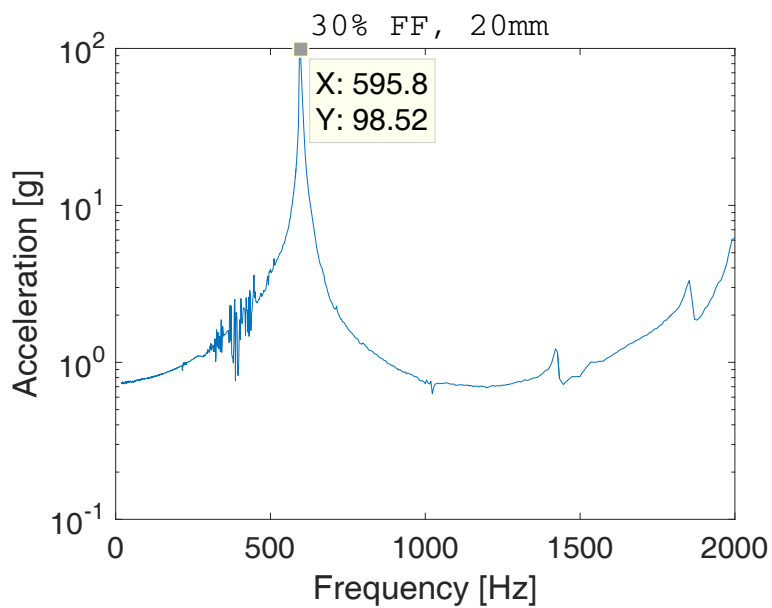


Figure A.29: 20mm Cavity filled with 30% steel shots

A.4.2 60% filling - 9.45gm particle mass

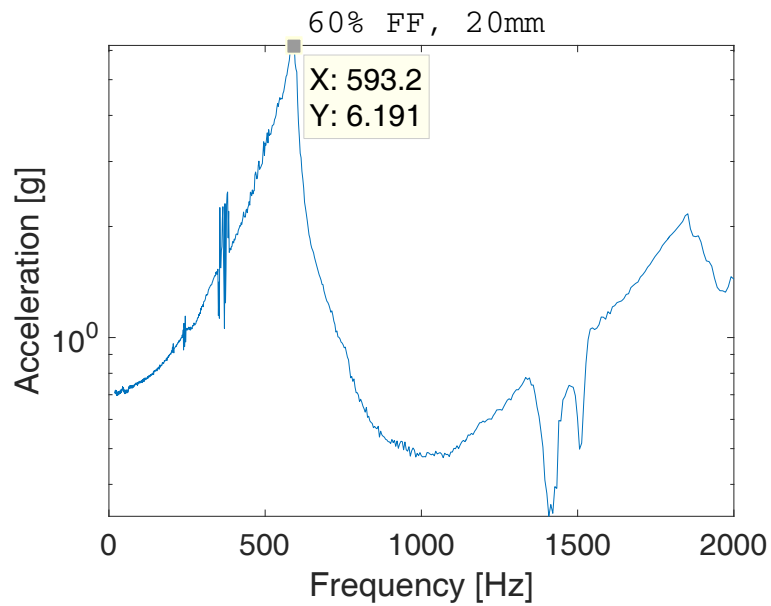


Figure A.30: 20mm Cavity filled with 60% steel shots

A.4.3 75% filling - 11.8125gm particle mass

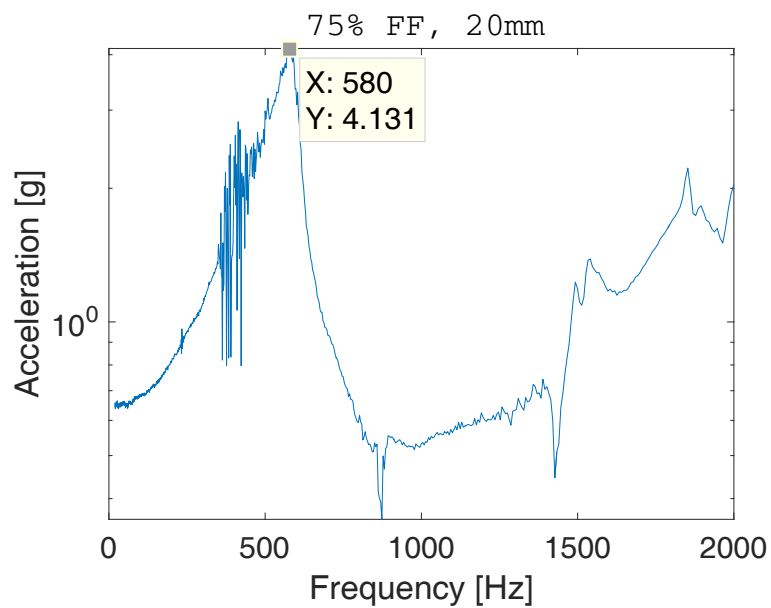


Figure A.31: 20mm Cavity filled with 75% steel shots

A.4.4 80% filling - 12.6gm particle mass

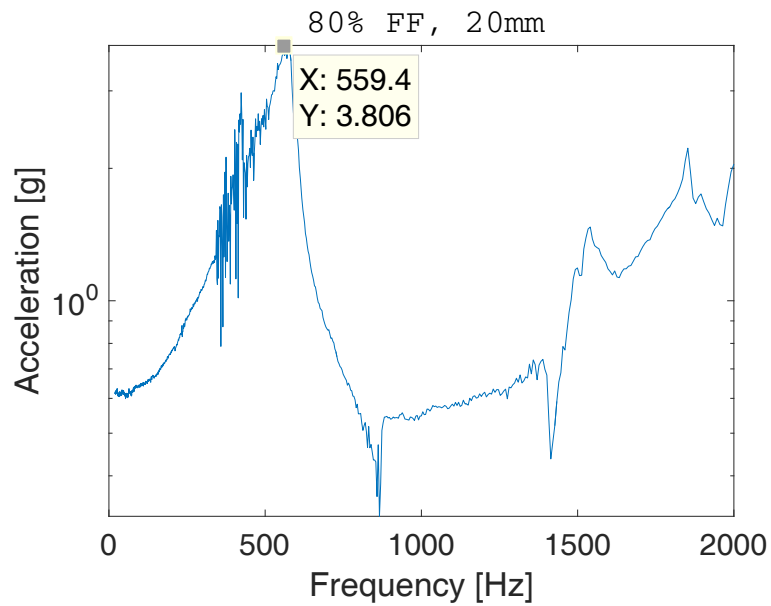


Figure A.32: 20mm Cavity filled with 80% steel shots

A.4.5 85% filling - 13.3875gm particle mass

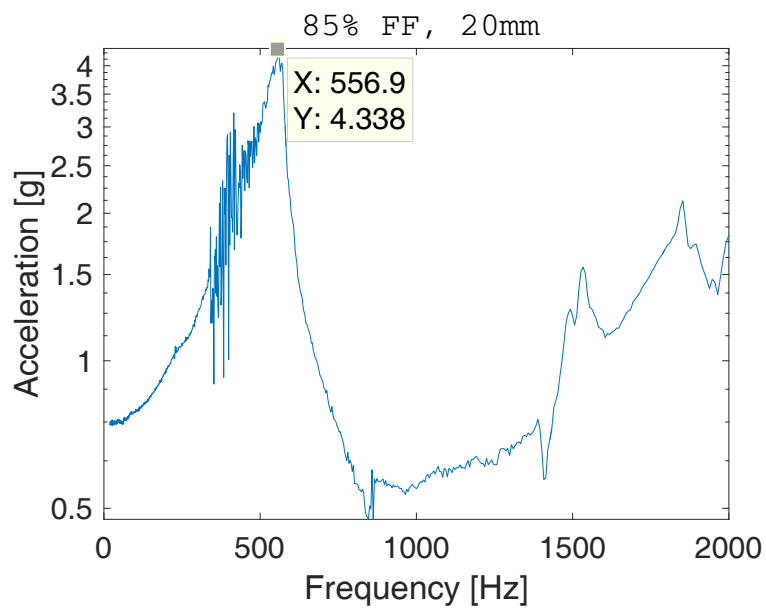


Figure A.33: 20mm Cavity filled with 85% steel shots

A.4.6 90% filling - 15.75gm particle mass

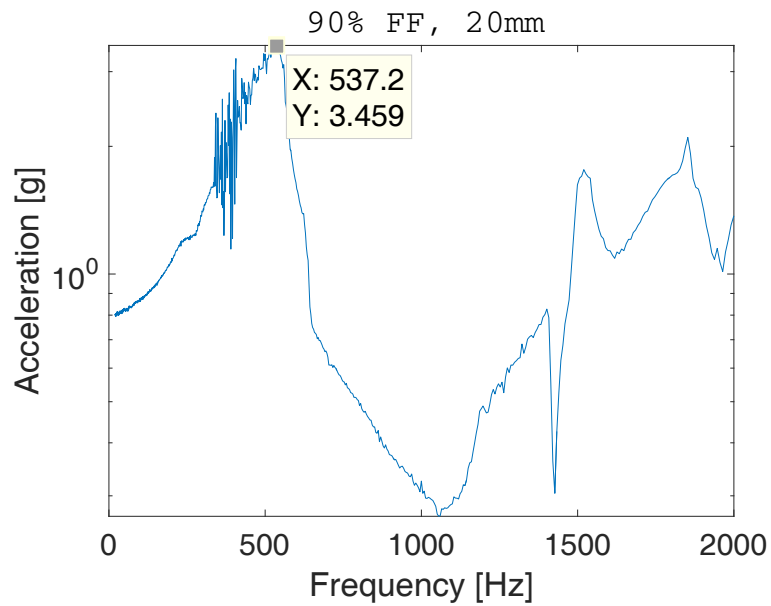


Figure A.34: 20mm Cavity filled with 90% steel shots

A.4.7 95% filling - 14.9625gm particle mass

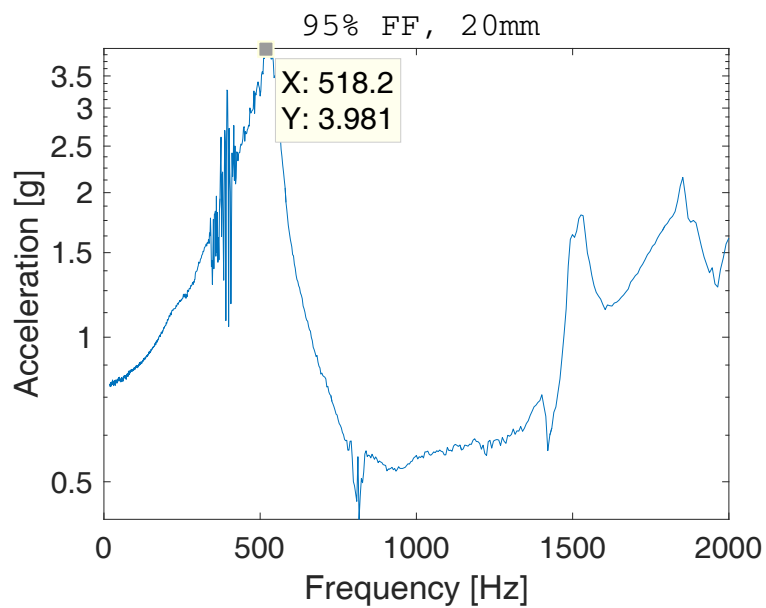


Figure A.35: 20mm Cavity filled with 95% steel shots

A.4.8 100% filling - 15.75gm particle mass

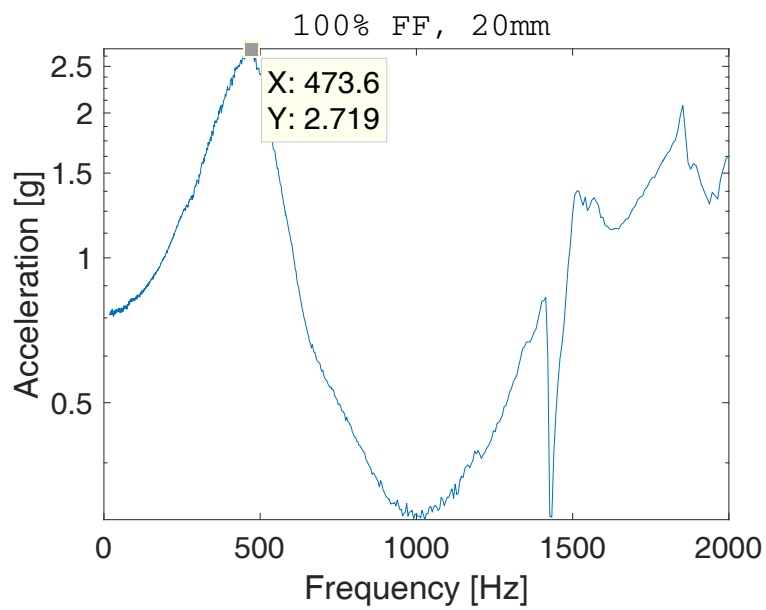


Figure A.36: 20mm Cavity filled with 100% steel shots

A.5 Damper with 25mm cavity length

A.5.1 Unfilled

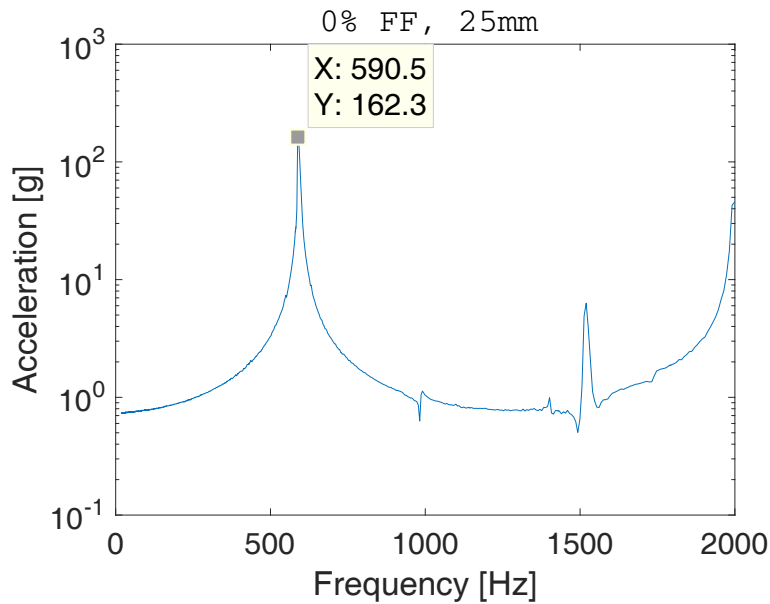


Figure A.37: 25mm Cavity filled with 0% steel shots

A.5.2 30% filling - 5.85gm particle mass

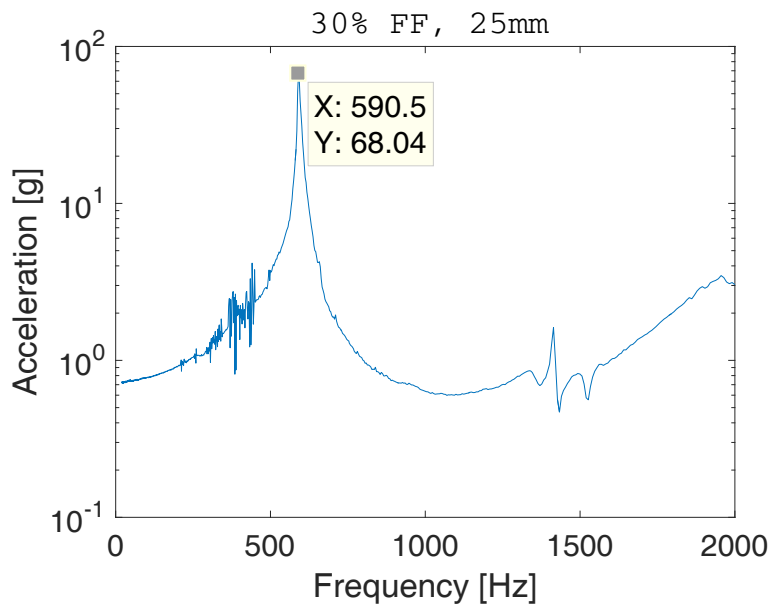


Figure A.38: 25mm Cavity filled with 30% steel shots

A.5.3 60% filling - 11.7gm particle mass

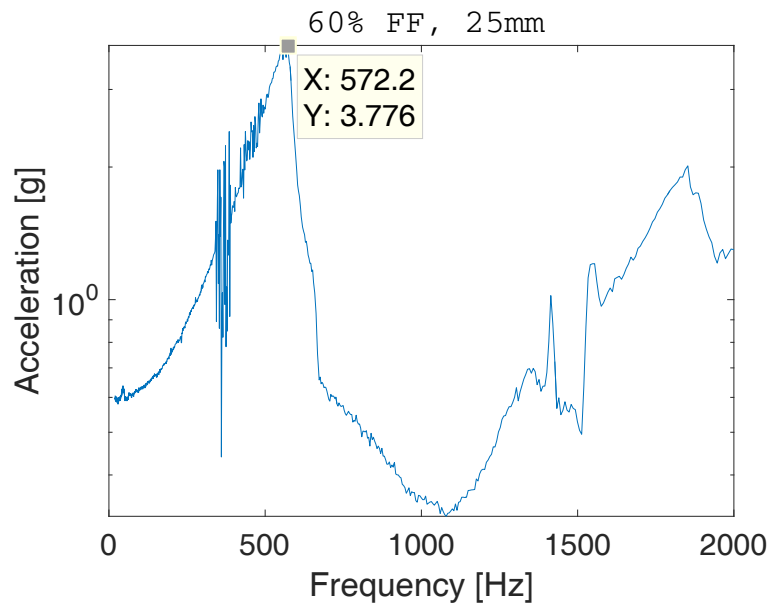


Figure A.39: 25mm Cavity filled with 60% steel shots

A.5.4 75% filling - 14.625gm particle mass

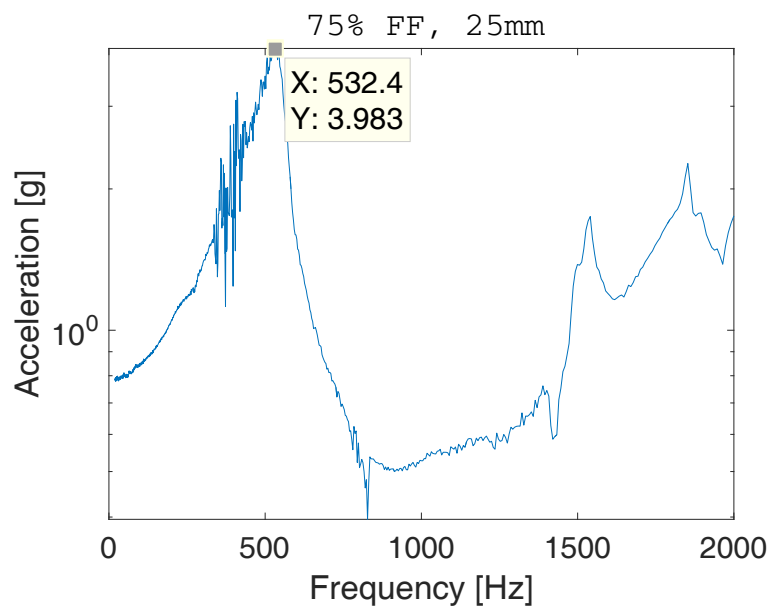


Figure A.40: 25mm Cavity filled with 75% steel shots

A.5.5 80% filling - 15.6gm particle mass

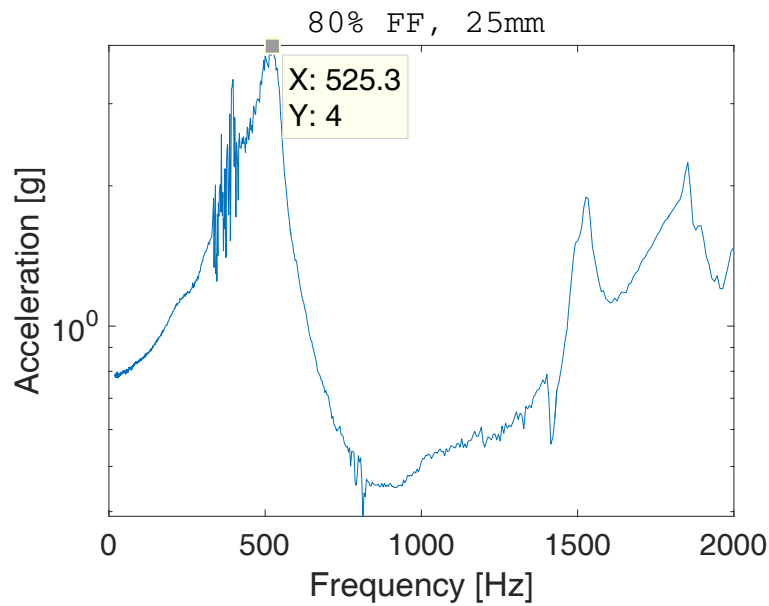


Figure A.41: 25mm Cavity filled with 80% steel shots

A.5.6 85% filling - 16.575gm particle mass

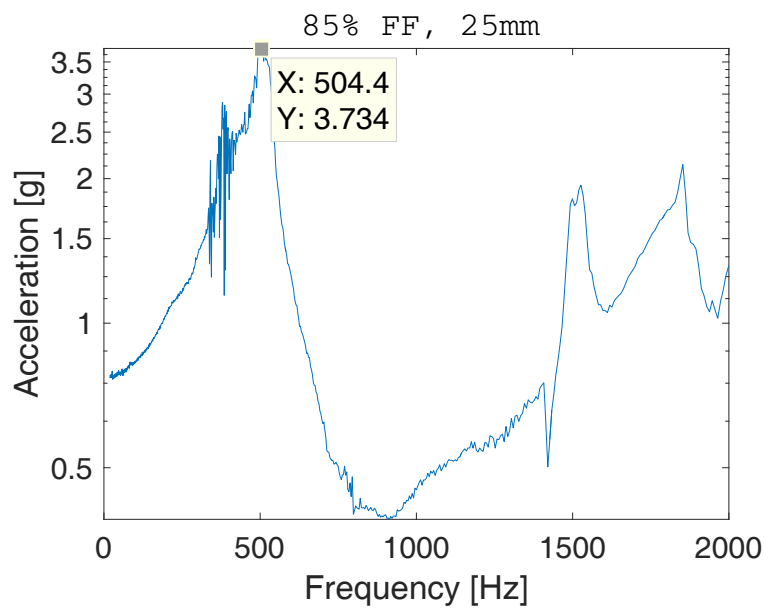


Figure A.42: 25mm Cavity filled with 85% steel shots

A.5.7 90% filling - 17.55gm particle weight

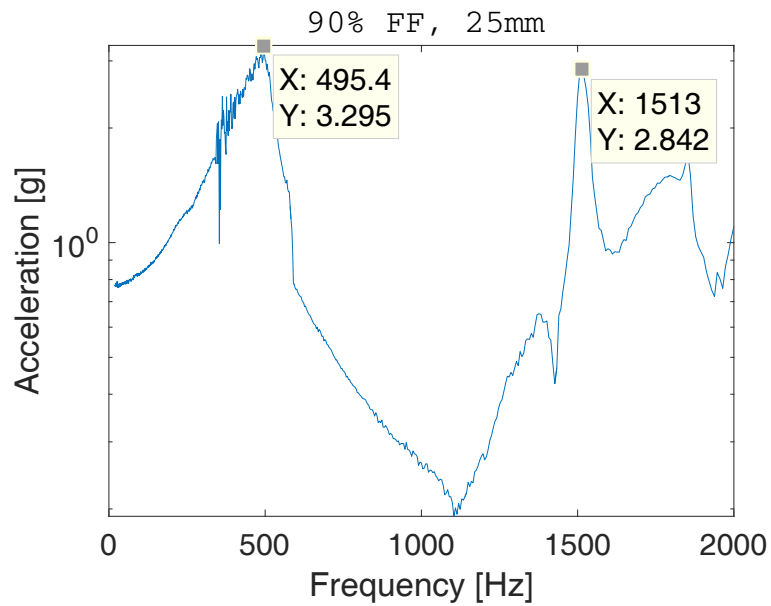


Figure A.43: 25mm Cavity filled with 90% steel shots

A.5.8 95% filling - 18.525gm particle mass

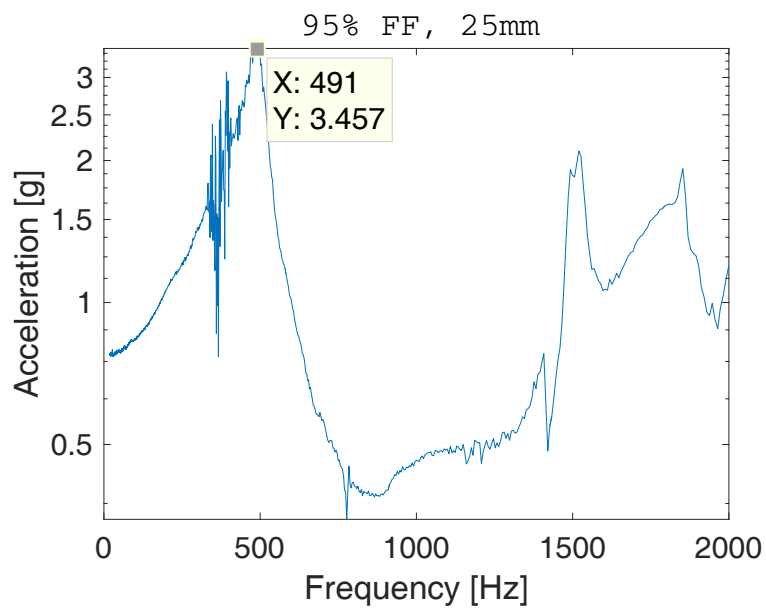


Figure A.44: 25mm Cavity filled with 95% steel shots

A.5.9 100% filling - 19.5gm particle mass

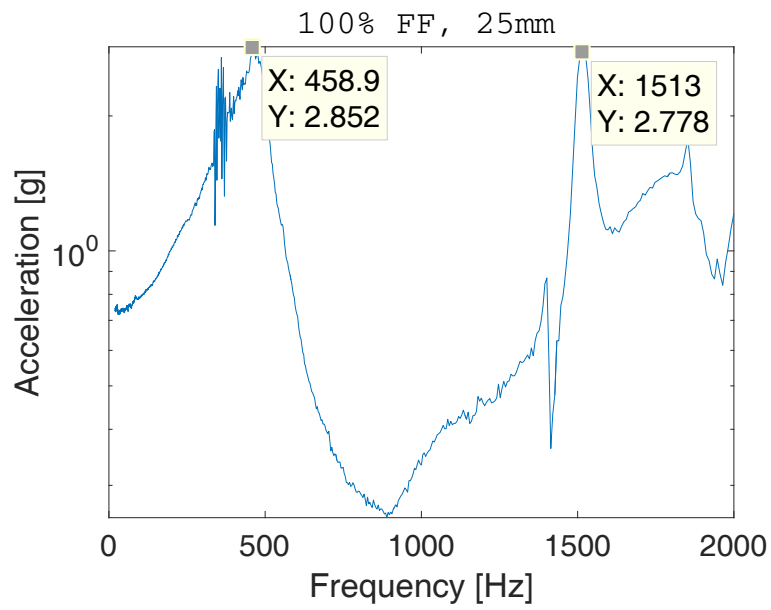


Figure A.45: 25mm Cavity filled with 100% steel shots

A.6 Damper with 25.4mm cavity length

A.6.1 Unfilled

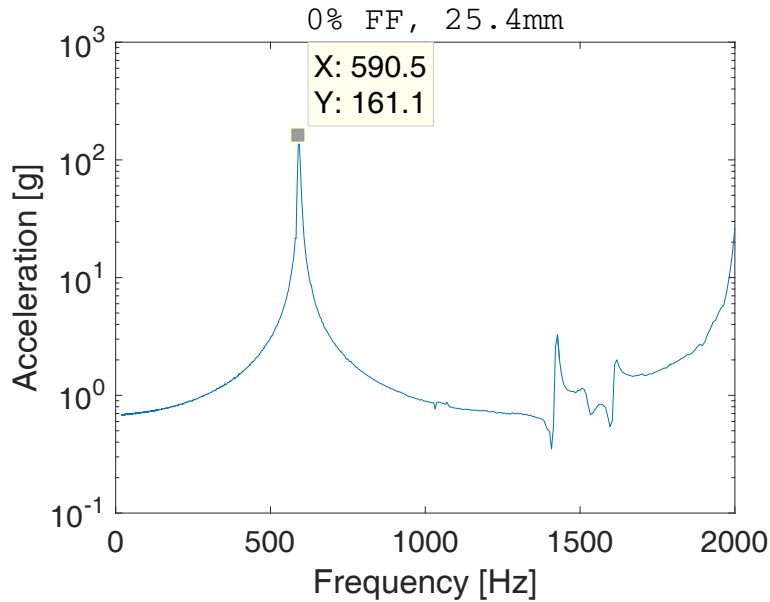


Figure A.46: 25.4mm Cavity filled with 0% steel shots

A.6.2 30% filling - 6.015gm particle weight

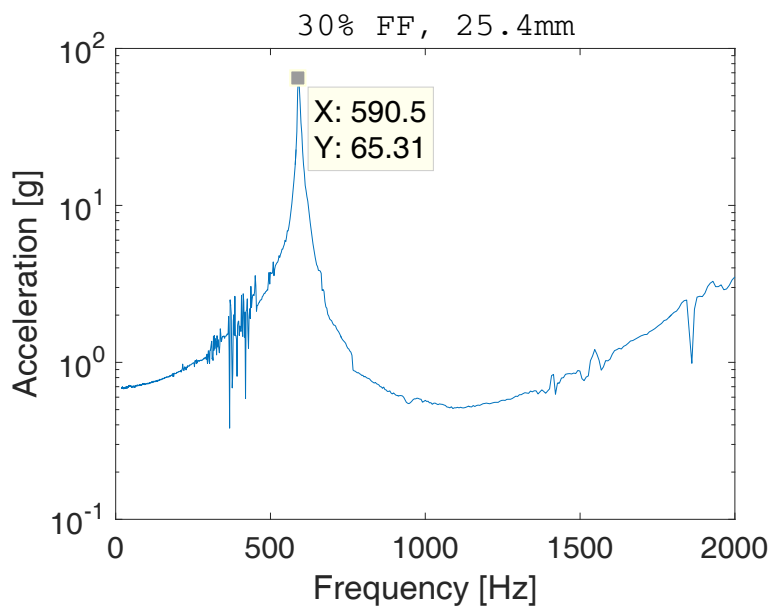


Figure A.47: 25.4mm Cavity filled with 30% steel shots

A.6.3 60% filling - 12.03gm particle weight

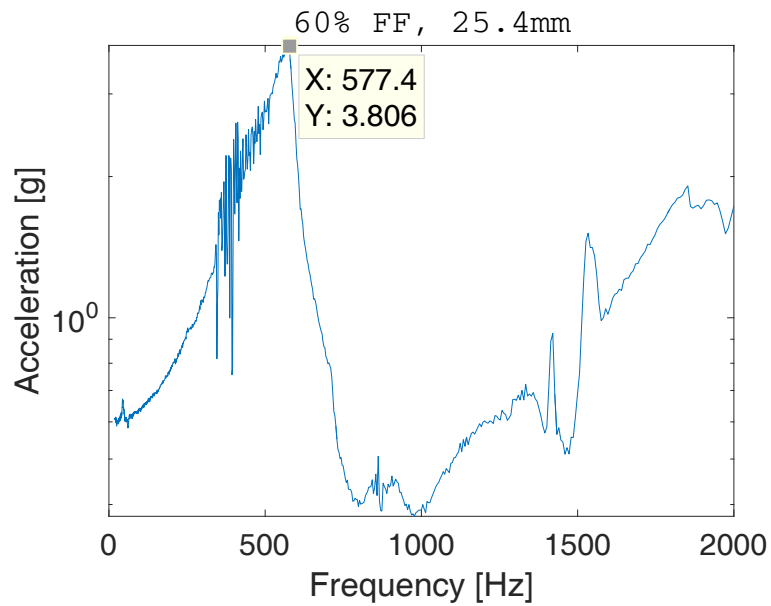


Figure A.48: 25.4mm Cavity filled with 60% steel shots

A.6.4 75% filling - 15.0375gm particle weight

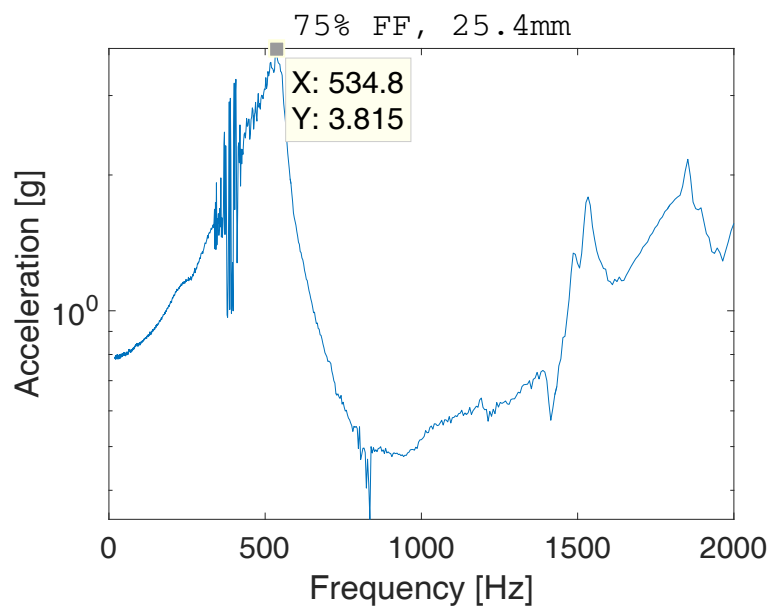


Figure A.49: 25.4mm Cavity filled with 75% steel shots

A.6.5 80% filling - 16.04gm particle weight

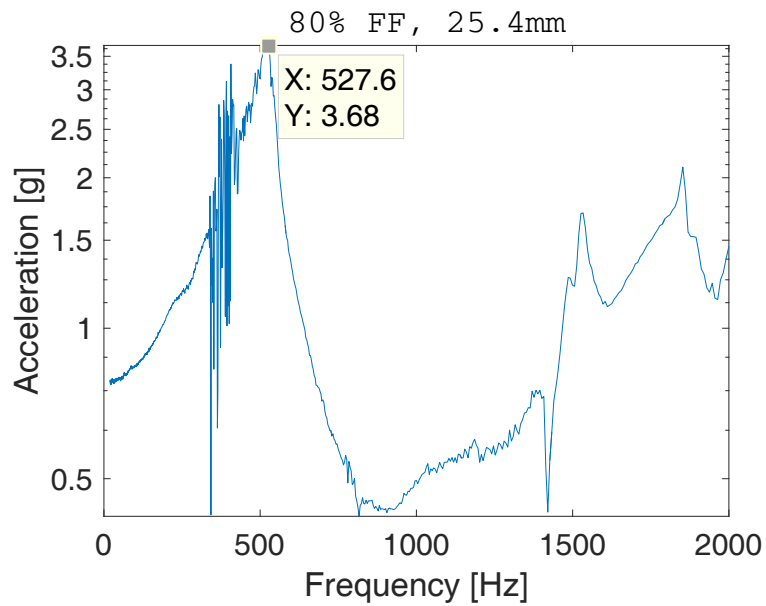


Figure A.50: 25.4mm Cavity filled with 80% steel shots

A.6.6 85% filling - 17.0425gm particle weight

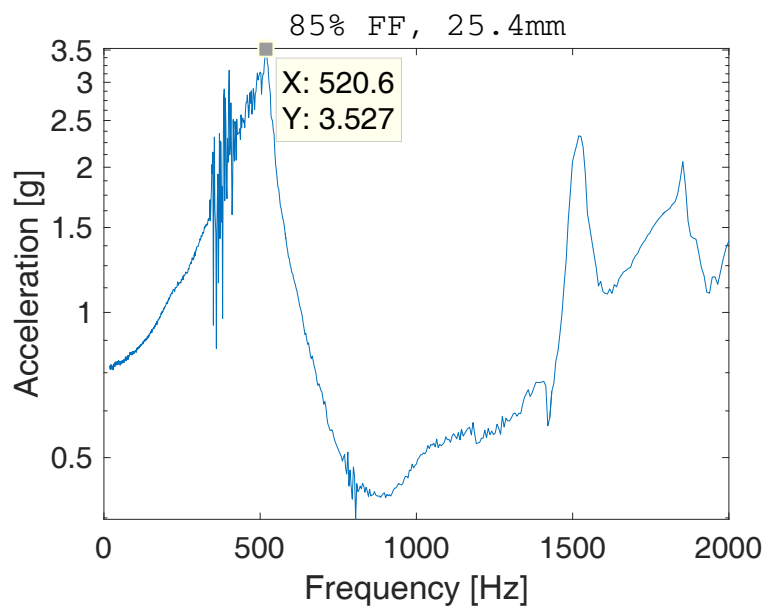


Figure A.51: 25.4mm Cavity filled with 85% steel shots

A.6.7 90% filling - 18.045gm particle mass

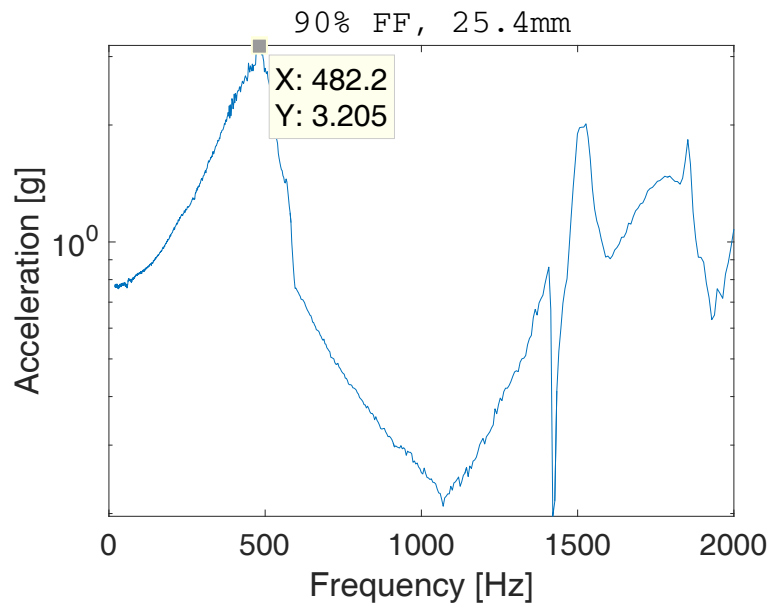


Figure A.52: 25.4mm Cavity filled with 90% steel shots

A.6.8 95% filling - 19.0475gm particle weight

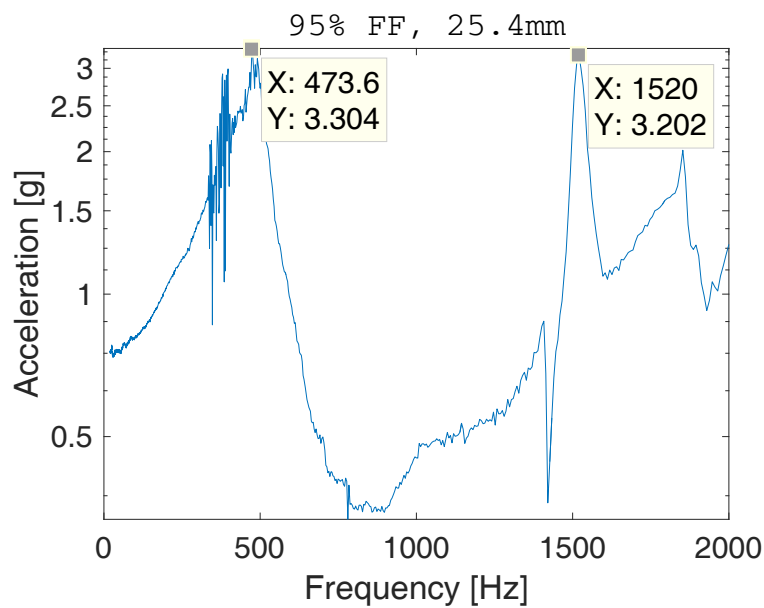


Figure A.53: 25.4mm Cavity filled with 95% steel shots

A.6.9 100% filling - 20.05gm particle weight

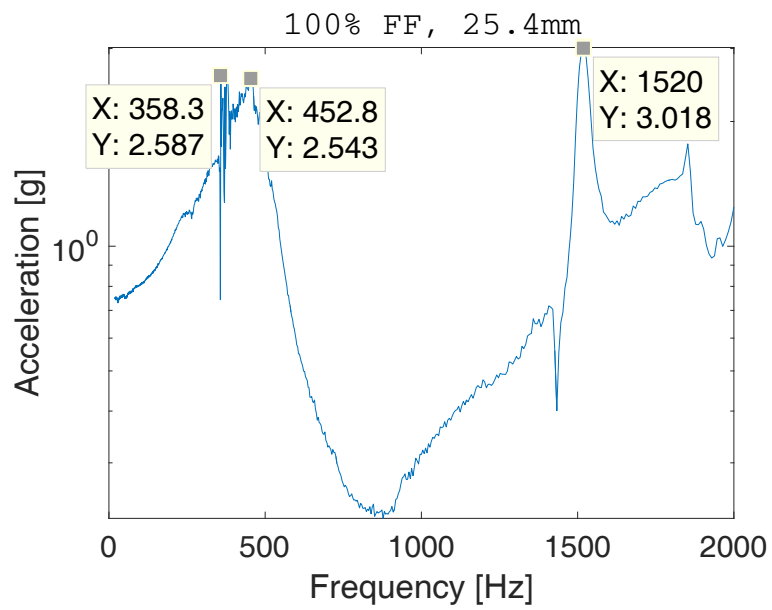


Figure A.54: 25.4mm Cavity filled with 100% steel shots

APPENDIX B:

G_{RMS} -Results

B.1 Damper with 5mm cavity length

Table B.1: G_{RMS} of undamped system and 5mm cavity with different fill-factors for different excitations including percentage of original system's value

Excitation	-9dB	-6dB	-3dB	0dB
No Damper	72.63	99.53	115.52	167.96
5mm unfilled	58.73(80.9%)	72.88(73.2%)	90.75(78.6%)	125.32(74.6%)
5mm 30%	43.41(59.8%)	60.94(61.2%)	84.12(72.8%)	107.36(63.9%)
5mm 60%	36.17(49.8%)	54.25(54.5%)	79.44(68.8%)	107.08(63.8%)
5mm 90%	29.15(40.1%)	36.57(36.7%)	47.05(40.7%)	58.61(34.9%)

B.2 Damper with 10mm cavity length

Table B.2: G_{RMS} of undamped system and 10mm cavity with different fill-factors for different excitations including percentage of original system's value

Excitation	-9dB	-6dB	-3dB	0dB
No Damper	72.63	99.53	115.52	167.96
10mm unfilled	62.24(85.7%)	75.89(76.2%)	93.64(81.1%)	133.57(79.5%)
10mm 30%	38.23(52.6%)	49.33(49.6%)	77.29(66.9%)	122.58(73.0%)
10mm 60%	28.76(39.6%)	48.33(48.6%)	70.23(60.8%)	99.67(59.3%)
10mm 90%	22.72(31.3%)	39.42(39.6%)	54.82(47.5%)	66.53(39.6%)

B.3 Damper with 15mm cavity length

Table B.3: G_{RMS} of undamped system and 15mm cavity with different fill-factors for different excitations including percentage of original system's value

Excitation	-9dB	-6dB	-3dB	0dB
No Damper	72.63	99.53	115.52	167.96
15mm unfilled	56.83(78.2%)	80.76(81.1%)	98.23(85.0%)	131.36(78.2%)
15mm 30%	30.05(41.4%)	51.37(51.6%)	69.81(60.4%)	112.64(67.1%)
15mm 60%	21.38(29.4%)	34.74(34.9%)	53.61(46.4%)	88.67(52.8%)
15mm 90%	15.46(21.3%)	24.19(24.3%)	45.50(39.4%)	62.86(37.4%)

B.4 Damper with 20mm cavity length

Table B.4: G_{RMS} of undamped system and 20mm cavity with different fill-factors for different excitations including percentage of original system's value

Excitation	-9dB	-6dB	-3dB	0dB
No Damper	72.63	99.53	115.52	167.96
20mm unfilled	52.86(72.8%)	77.28(77.6%)	97.20(84.1%)	121.06(72.1%)
20mm 30%	27.51(37.9%)	42.60(42.8%)	73.86(63.9%)	98.16(58.4%)
20mm 60%	17.12(23.6%)	27.49(27.6%)	56.55(49.0%)	74.78(44.5%)
20mm 90%	14.09(19.4%)	22.12(22.2%)	40.15(34.8%)	63.47(37.8%)

B.5 Damper with 25mm cavity length

Table B.5: G_{RMS} of undamped system and 25mm cavity with different fill-factors for different excitations including percentage of original system's value

Excitation	-9dB	-6dB	-3dB	0dB
No Damper	72.63	99.53	115.52	167.96
25mm unfilled	55.39(76.3%)	77.83(78.2%)	96.32(83.4%)	125.12(74.5%)
25mm 30%	25.48(35.1%)	40.46(40.7%)	62.30(53.9%)	99.28(59.1%)
25mm 60%	13.89(19.1%)	24.55(24.7%)	43.16(37.4%)	69.21(41.2%)
25mm 90%	11.32(15.6%)	18.68(18.8%)	30.74(26.6%)	51.85(30.9%)

B.6 Damper with an inch cavity length

Table B.6: G_{RMS} of undamped system and an inch cavity with different fill-factors for different excitations including percentage of original system's value

Excitation	-9dB	-6dB	-3dB	0dB
No Damper	72.63	99.53	115.52	167.96
inch unfilled	51.38(70.7%)	73.41(73.8%)	94.79(82.1%)	114.96(68.4%)
inch 30%	24.06(33.1%)	39.17(39.4%)	63.07(54.6%)	94.29(56.1%)
inch 60%	14.41(19.8%)	25.91(26.0%)	40.64(35.2%)	61.26(36.5%)
inch 90%	10.76(14.8%)	17.10(17.2%)	31.34(27.1%)	51.02(30.4%)

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