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**Department of
Mechanical Engineering**

VIBRATION ISOLATION FOR SCANNING TUNNELING MICROSCOPE

Final Design Review

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Statement of Disclaimer

Abstract

Scanning Tunneling Microscopy (STM) works by scanning a fine metal wire over a conductive sample in order to image samples on the atomic level. The microscope uses a feedback loop with a PID controller to maintain a constant tunneling current between the sample and the tip. In order to obtain clear imaging at the atomic level, the microscope and the sample need to be isolated from external vibrations in a vacuum chamber. The scope for this project is to build and test a single-stage vibration isolation system that effectively attenuates the amplitude of external vibrations acting on the microscope. This document summarizes the preliminary design process, including the customer interviews we performed, the background research we collected about existing patents and designs, and the technical journal articles dealing with this subject that we read. In addition, this project defines the final design along with manufacturing and testing results to validate the design. The final design includes a final design description, design justification, manufacturing results, as well as the testing plan and results. These sections serve to overview the process for our final design selection and the results of the manufacturing and testing phases of the final design.

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1 Introduction

Scanning Tunneling Microscopes (STM) are used to obtain surface topography and have become a popular measurement device in laboratories because they can perform a non-destructive surface analysis [1]. A tunneling current occurs when the tunneling tip is brought within less than 1 nm to a sample surface. The tunneling current varies exponentially with gap distance, with a work function of 4eV, the tunneling current can increase by a factor of 10 when the gap distance decreases by just 1 Å [2]. A scanner, which is typically made from a piezo electric actuator, controls the tip-sample separation and must be controlled within an accuracy better than 0.05 Å [2]. Considering this resolution and current sensitivity, it is important to keep the microscope stage mechanically stable when ambient vibrations exceed this scale (which they often do). Therefore, a Vibration Isolation System (VIS) is a critical component in a Scanning Tunneling Microscope (STM), as it needs to attenuate the amplitude of ambient vibrations to within acceptable positional accuracy levels [3]. Although many designs have been used to eliminate the effect of ambient vibration on samples (which are detailed in the background section), we found the two main isolation systems used in STMs are: spring suspension with eddy current dampening, or stacked metal plates with Viton damping between each plate.

Our senior project was a continuation of a master's thesis, which could not be completed due to the COVID-19 pandemic. The design employed a single stage VIS, which consisted of a housing enclosure with Viton damping, a copper main stage suspended by stainless steel springs, and permanent magnets which damp the copper main stage. Design and analysis for all three systems had been completed, however our group found several opportunities to improve the design in all areas. Procurement as well as testing of the copper mass-spring system was completed. Finite Element vibration analysis were been conducted on the housing with Viton damping and a 6061-Aluminum tube has been purchased. Additionally, simulations had been conducted to find magnetic flux density for permanent magnets and damping ratio as a function of gap distance.

Our goal for this project was to design, build, test, and integrate all subassemblies in the vibration isolation system for use by the Cal Poly Chemistry Department's Scanning Tunneling Microscope. This includes the housing enclosure with Viton damping, copper mass-spring system, and magnetic damping system. This report outlines our design process from our preliminary research and interviews with the sponsor through our preliminary concept selection, final design, manufacturing and design verification.

2 Background

2.1 Sponsor Interviews

To better understand the needs of the sponsor and transfer knowledge from previous research and design work, several meetings were conducted with the sponsors, Mr. Le and Dr. Scott. From these meetings, we developed a list of the customers' desired outcomes for the project. We also obtained more details about the project requirements from Dr. Scott and CAD files from the VIS design in Mr. Le's master's thesis. To avoid hampering our ideation process, we will treat Mr. Le's design as a competing product in this SOW; however, a copper stage, Al 6061 - T6 hollow tube for machining into the housing, and set of stainless-steel extension springs have already been acquired and will need to be incorporated into our final design. A summary of the customers' needs and wants is provided in Table 1.

2.2 Existing Designs

Over the course of our research, we came across several different solutions to the problem of isolating sensitive equipment from ambient vibrations. These can be broadly divided into two categories: passive systems and active systems. Passive systems typically use some combination of mass, spring, and damper systems to filter out and dampen vibrations at various frequencies. Active systems work on a similar principle to noise-canceling headphones, generating vibrations out of phase with the ambient vibrations to cancel them out. Of the two, passive systems are generally simpler, far less expensive, and more reliable, although active systems can react better to changes in ambient frequency and are less constrained by the natural frequencies of materials. In industrial or highly funded research applications, a combination of active and passive systems is often used. Below is a summary of similar solutions from our research.

Minus K Technologies Negative Stiffness Isolator was a system developed by Minus K Technologies for use with UCLA's alternating current STM [4]. The system uses a negative stiffness system in combination with a spring to achieve a very low stiffness without having to resort to an overly soft spring. Using this method, they were able to achieve a natural frequency of 0.5 Hz and isolate frequencies above 0.7 Hz. While a system like this would not be able to integrate with our vacuum chamber, this design highlights the potential usefulness

Table 1: Customer needs and wants list.

Customer Needs	Customer Wants
Build, test, and integrate vibration isolation system (VIS) into STM by end of spring 2021. — Natural frequency of VIS as low as possible — Natural frequency of aluminum housing as high as possible — Built, test and validate system theoretically using FEA and experimentally using a shaker table — Ease of manipulating/ switching out samples — Ability to view sample from top of vacuum chamber — Place damping material in necessary locations and test to show effects of damped vs. undamped — Test damping ratio of magnetic damping system — Needs to fit inside vacuum chamber and function within high vacuum system (HVS)	— Harmonic frequency of mass-spring system below 2 Hz. — Harmonic frequency of structure over 200Hz — Stage vibrations die out after around 10s(to avoid coupling stage with structure). — Perform vibrational analysis of chemistry lab to identify target frequencies — Ease of inserting/removing VIS from vacuum chamber. — Focus on wobble mode of vibration of the frame as this has been the most difficult to isolate. — Mechanism to adjust distance between permanent magnets and conductive stage to change damping ratio

of negative stiffness systems in our design. A schematic of the negative stiffness isolation system is shown in Figure 1.

A thermally compensated tube scanner has been constructed and successfully tested [Variable-temperature scanning tunneling microscope] which employs two concentric piezoelectric tubes which account for thermal expansion/contraction and inertial sample translation. This design eliminates vibration isolation systems. Samples were taken in at room temperature and in a liquid-helium Dewar and notable images were obtained of graphite. Another design developed by the Center for Research and Advanced Studies of the National Polytechnic Institute incorporates a series of stacked metal plates [5]. Between each metal plate, the designers sandwiched a particular rubber material with desirable stiffness and damping values. By changing the stiffness and damping coefficient of each of the layers, the behavior of response of the entire system can be altered. However, as the number of stacked metal plates increases, the degrees of freedom of the system also increases, which makes finding the overall transfer function for the system difficult [5]. However, one of the primary advantages of this system is that the damping and stiffness of the system can be fine-tuned to produce consistent attenuation of external perturbations down to the subatomic range.

2.3 Technical Research

Scanning Tunneling Microscopes (STM) are not a commonly commercialized technology and thus there are a wide variety of types. Since STM's are highly dependent on the vibrations in their environment, there are many unique methods to isolate the microscope. However, a number of techniques are common in reliable, cost-effective STM's. These methods have been explored below and include mass-spring systems and damping mechanisms including magnetic eddy current damping and rigid housing structures.

2.3.1 Mass-spring System

Vibrations that impact a STM can generally be categorized as either high or low frequency. Low frequency vibrations fall in the range of 1 to 100 Hz and are caused by building natural frequencies, people walking, or doors being closed [6]. Mass-spring systems are a common way to isolate the microscope from these low frequencies when the microscope is used in an ultra-high vacuum (UHV). To avoid resonant frequencies with

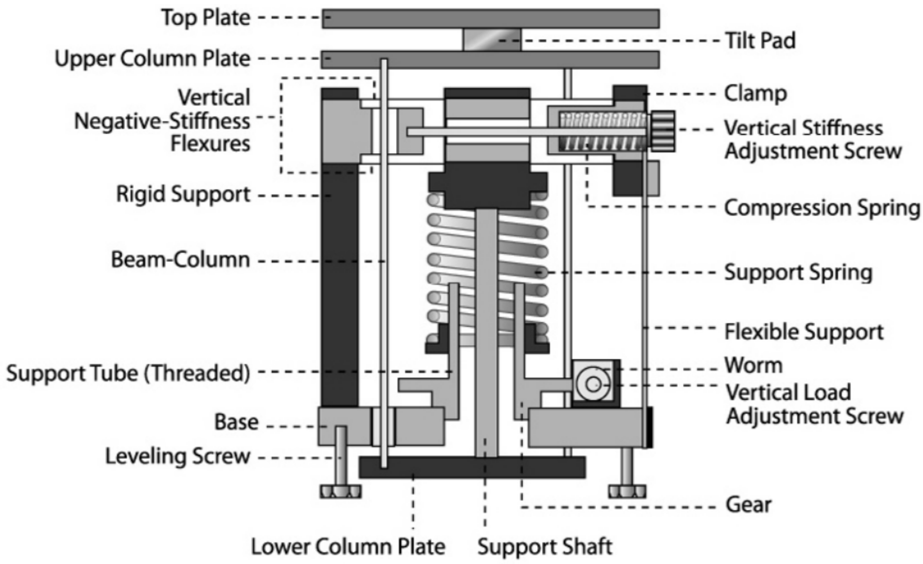

Figure 1: Schematic of a negative stiffness VIS. This design is a stage on top of which the vacuum chamber rests. Stiffness can be adjusted via a screw on the upper right [4]

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any of the low frequencies the springs are attempting to isolate, it is advisable to use springs with the lowest possible resonant frequency. However, the result of this is a low internal damping factor [7]. This relationship can be observed for a simple mass-spring damper system with the following equation relating natural frequency to damping coefficient.

$$\omega_n = \frac{c}{2\zeta m} \tag{1}$$

It can be observed that as the natural frequency (ω_n) decreases, the damping coefficient (c) also decreases. For this reason, when a mass-spring system is utilized, it is often paired with other elements to provide additional damping for the vibration isolation system [6].

2.3.2 Damping Mechanisms

In order to attenuate the amplitude of external vibrations acting on a mechanical system, damping must be used to dissipate energy. In general, damping can be broken up into two categories: active damping and passive damping. One common way to damp out vibrations is to use electromagnetic damping. Active electromagnetic damping devices measure vibrations in real time and react accordingly [8]. This requires a closed-loop feedback control system. One of the benefits to active damping is that is provides outstanding, customized performance. The main disadvantage to active damping is that it requires a sophisticated feedback control system to maintain a desired damping between the pole face and the reaction surface [9]. Additionally, active damping can increase the cost of the system and decrease the reliability.

Therefore, most of our research focused on passive damping, as it is generally less expensive, requires no control system, and is simpler and more reliable than active forms of damping [10]. If the main stage were not a conductive material (such as copper), a magnetically tuned damper [11] could be employed; this device, developed by CSA Engineering Inc., could allow freedom for the main stage material used. Additionally, many passive damping methods are used in high vacuum environments for scanning tunneling microscopes such as viscoelastic materials, magnetic damping, and damping using piezoelectric materials. A popular viscoelastic material is Viton [6], as it is effective at damping, relatively incompressible, and when compressed between two plates generate a resonant frequency between 10 Hz to 100 Hz [6].

Though viscous fluid damping and piezoelectric materials are used in various tunneling microscope vibration isolation systems, magnetic eddy current damping is a better alternative for vacuum compatibility and for the ability to vary the damping coefficient [12].

2.3.3 Vacuum Chamber Support Mechanism

In order for a STM to obtain desirable results, a vacuum chamber surrounding the microscope is needed to further isolate the system from external disturbances [13]. Reference [7] shows the impact of a concrete slab with air spring supports on the vacuum chamber frame. They used a seismic accelerometer with a sensitivity of 10 V/g and built in low pass filter of 100 Hz to measure the acceleration and displacement amplitude of the laboratory floor upon which the their STM sat. They measured the maximum acceleration and displacement amplitude of the laboratory floor to be $50 * 10^{-}6$ g (at 16Hz) and 50 nm, respectively. They then placed the STM vacuum chamber on a concrete slab and air springs and measured the acceleration and displacement amplitude of the vacuum chamber base to be $2.5 * 10^{-}8$ g (at 68 Hz) and 71 pm, respectively.

2.3.4 Vibration Isolation Structure

Another method of isolating and damping out vibrations is focused on the high-frequency vibrations. A rigid structure with a high natural frequency above 100 Hz [14] is often used which acts as a high-pass filter. If used in conjunction with a mass-spring system, it is important that the structure has a natural frequency as far away from the mass-spring system to avoid any resonance.

2.4 Existing Patents

Despite a thorough search for patents relevant to our design, we were unable to turn up similar vibration isolation systems. Many of the patents are for active control systems, and others are unlikely to yield the precision needed for our purposes. Nevertheless, the designs are interesting from an ideation standpoint and have been documented in Table 2.

One patent we considered was for an active damping system called a Stewart parallel platform, which uses a system of six linkages to isolate a platform from ambient vibrations [15]. While this design is not capable of isolating a sample in a vacuum chamber from vibrations produced by the chamber itself, it is of interest because of the configuration of spring-and-damper linkages that is able to isolate vibrations in wobble and yaw modes in addition to vertical. A schematic from the patent is shown in Figure 2.

Table 2: Results of patent research.

Patent Name	Patent No.	Description
A kind of six-degree-of-freedom isolation platform based on Stewart parallel institution	CN 103587724 B	A damping platform that uses six linkages to absorb vibrations in six degrees of freedom in response to input vibrations
Arrangement and method for damping vibrations during microscopic examinations	EP 2 903 709 B1	Describes a method for arranging and programming an active vibration isolation control feedback loop
Compact damping device and compact damping table	JP 2004-360784 A	Miniature damping table consisting of two stacked channels connected by a spring, with damping material at the interface
Passive eddy current nutation damper	US 3,806,062	Device that uses eddy current damping from permanent magnets to stabilize the axis of a rotating object
Vibration isolation coupling including an elastomer diaphragm for scanning microscopes and the like	KR 2003-0038473	Couplings made from a vibration damping material for use in connections within a large scanning tunneling microscope

Another of the patents was for an active control scheme for a vibration isolation system. The system has several different materials with different elasticity and damping properties in series that can be engaged separately to fine tune the system frequency and damping ratio [16]. While this system is regulated by a complex set of control loops, the idea of being able to select different materials for adjustable damping and elasticity may prove useful. The other patents found, while interesting, were of less use to us. One was for a miniature damping table that was able to be used with desktop devices such as optical microscopes [17]. Another described an eddy current damping system used to stabilize the spin of instruments such as gyroscopes [18]. is design is interesting as a case study in the implementation of permanent magnets for passive damping. Finally, we found a design for a vibration-reducing coupling between the sample chamber and test chamber of

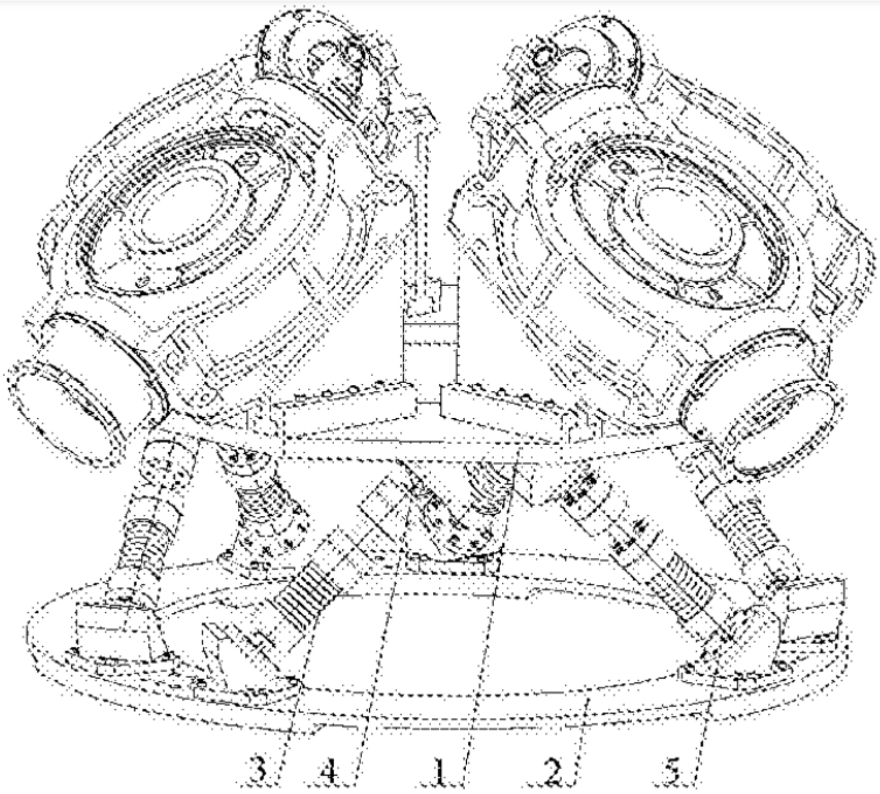


Figure 2: Schematic of Stewart platform. Note the way springs and dampers are integrated with the linkages. Additionally, the springs are oriented so that vibration can be isolated in multiple dimensions [**roscher**].

a large STM [19]. This design is of note because they used an elastomeric diaphragm made of Viton to damp vibrations.

2.5 Industry Codes, Standards, and Regulations

Given the research-focused nature of the application, there are not many codes and regulations for vibration isolation systems for scanning tunneling microscopes. However, one potential resource is the Institute of Environmental Sciences and Technology's Environmental Vibration Criteria (VC) curves. The VC curves give a conservative estimate for the allowable max and working vibration amplitudes, in $\mu m/s$ rms, for various instrument applications. By these criteria, scanning tunneling microscopes would fall under the VC-D curve and thus would require a maximum amplitude of 6 $\mu m/s$ rms to resolve a detail of 0.3 microns [20]. As Dr. Scott is looking to resolve detail on a scale of picometers, this would be difficult to extrapolate to a useful amplitude for our purposes.

3 Objectives

3.1 Problem Definition

Scanning Tunneling Microscopes are an extremely precise instrument that need the tip of the microscope to be less than a nanometer away from the sample to function properly. Vibrations from the surroundings can cause movement between the tip and sample, causing unreliable atomic pictures. The Cal Poly Chemistry Department needs a way to isolate their scanning tunneling microscope (STM) from ambient system and environmental vibrations as well as dampen these vibrations. This will help decrease the amplitude of vibrations so the department can consistently obtain accurate measurements on an atomic scale without excessive movement between the tip and sample.

The main customer need is to create a vibration isolation system that allows the STM to consistently obtain accurate atomic images. Subsequent needs are to validate the existing designs that have been proposed as well as create a simple design that is inexpensive, reliable and allows viewing of the STM in use. Proposed designs that require validation are a mass-spring system that connects the aluminum housing to the copper stage as well as a magnetic eddy current damping system using the copper stage as the conductor. The complete set of customer needs and wants are listed in the QFD found in Appendix A.

3.2 Boundary Diagram

One of the ways in which we defined our project scope was visually with a boundary diagram, which is shown in Figure3. The boundary of our system separates what we will be designing from what already exists. Therefore, we need to consider how our system will integrate with the supporting structure and vacuum chamber.

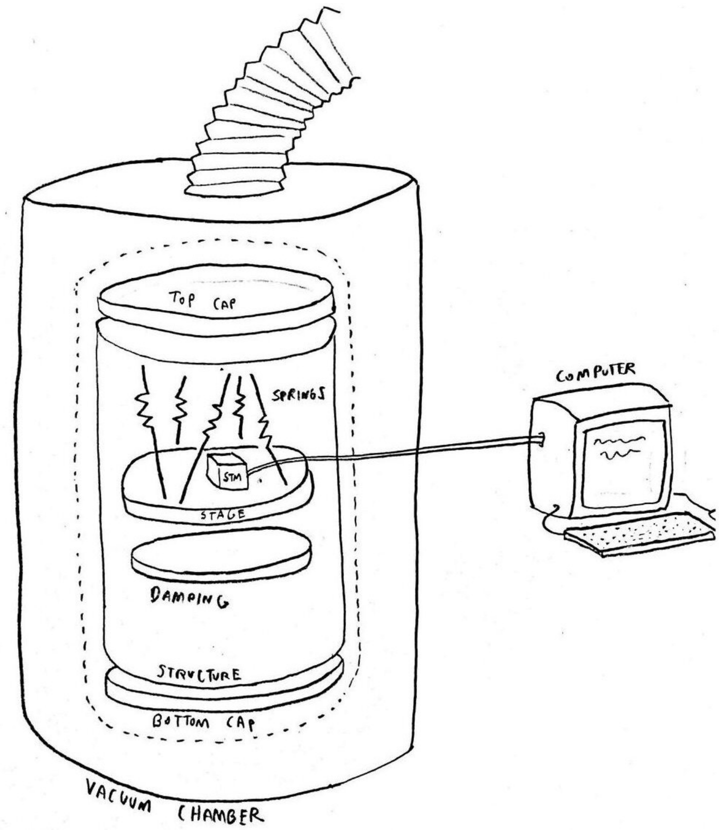


Figure 3: Vibration Isolation System Boundary Diagram. Our boundary diagram includes the aspects of our project that we will be working on as well as the parts which will directly interact with what we design. Inside of the boundary is the aluminum housing structure, including the top and bottom caps, the spring and stage system as well as some type of damping mechanism. We will be designing these parts to interface with the scanning tunneling microscope (STM) and vacuum chamber, however we will not actually be designing those elements.

3.3 Quality Function Deployment (QFD)

Our team used the method of quality function deployment (QFD) to translate the sponsors needs and wants into a list of engineering specifications, along with their appropriate targets and tolerances. QFD first starts by identifying customer needs and wants and weighting according to importance. These are then compared to existing products to understand where the shortcomings are in these products. Lastly, using the needs/wants, specifications are developed and related to the customers' needs/wants and target values are assigned. The QFD for our project is referenced in Appendix A.

3.4 Engineering Specifications

Using the House of Quality that we created using QFD, we created a set of engineering specifications that have been compiled below in Table 3. These specifications represent all the requirements that must be met for our project to be successful and meet the needs and wants of the sponsor. Below Table 3 we have listed how we plan to measure each of our specifications, whether it be by testing, analysis, inspection or a combination of them.

Table 3: Engineering specifications along with their associated risk and method of compliance.

Spec.	Description	Target (Units)	Tolerance	Risk*	Compliance**
1	Housing natural Frequency	200 Hz	Min.	M	A, T
2	Mass-spring system natural frequency	2 Hz	Max.	H	A, T
3	Magnetic damping coefficient	0.2-0.3 Ns/m (vertical and yaw modes)	±0.1	M	A
4	Elastomeric material damping coefficient	50-400 Ns/m	±20	M	A
5	Vibration amplitude	100 pm	±10	H	I
6	Time for vibrations to die out	10 sec	±1	M	I
7	Integration dimensions	17.78 cm (diameter), 34.82 cm (height)	Pass/Fail	L	I, T
8	Viewing area in aluminum housing	100 cm^2	Min	L	I
9	Sample swap time	2 min.	Max.	L	T
10	VIS swap time from vacuum chamber	30 min.	Max.	L	T
11	Vacuum-compatible materials	Pass/Fail	N/A	M	I
12	Cost	$500	Max.	L	A
13	Consistent atomic measurements	90% success rate	Min.	H	T

*Parameters are high (H), medium (M), and low (L) risk.

**Parameters are analysis (A), test (T), inspection (I), and similarity (S).

The method of measuring compliance for each specification is elaborated below for each engineering specification:

- Spec. 1: Natural frequency will be measured analytically using finite element analysis (FEA) and measured experimentally using a shaker table.
- Spec. 2: Natural frequency will be measured analytically using a MATLAB and measured experimentally using a shaker table.
- Spec. 3: Damping coefficient will be measured analytically using a simulation of the magnetic field.
- Spec. 4: Damping coefficient will be measured analytically using a simulation of the material under stress using FEA.
- Spec. 5: Vibration amplitude will be measured by inspecting the images the scanning tunneling microscope (STM) outputs with the completed vibration isolation system (VIS) in place.
- Spec. 6: Time for vibrations to die out will be measured by inspecting the images the STM outputs with the completed VIS in place.
- Spec. 7: Dimensions of the VIS will be measured by inspecting with a measuring tape and by testing that it fits and integrates with the vacuum chamber and microscope.

- Spec. 8: Dimensions of the aluminum housing viewing area can be measured by inspection using a measuring tape.
- Spec. 9: Sample swap time will be measured by timing how long it takes to swap a sample for a number of trials and averaging the time.
- Spec. 10: VIS swap time will be measured by timing how long it takes to put in/take out the VIS for a number of trials and averaging the time.
- Spec. 11: Only vacuum compatible materials will be chosen to fulfill this specification.
- Spec. 12: Cost of every item used will be tallied and appropriate budgeting will be followed to stay below cost limit.
- Spec. 13: Success rate of the atomic measurements will be tested by running the STM for a number of trials and finding the average success rate of atomic scale images.

4 Concept Design

To determine a preliminary concept design direction, our senior project team underwent a rigorous ideation design process. We defined the key functions of our system, brainstormed ideas and developed Pugh Matrices for each function, and then created a decision matrix in order to select our final design. By going through this process, we effectively developed a concept design for our system, which will help focus our analysis.

4.1 Idea Generation

The first step in our ideation process was identifying the key functions our design needed to address. This took the form of a functional decomposition tree which can be found in Appendix B. The functional decomposition identifies the primary functions our design must achieve as well as the sub-functions necessary for the primary functions to work correctly. We identified "Improve Microscope Function and Image Quality" as our primary function, because if our design fails to achieve this we will not have met our objective. Naming this as our primary function allowed us additional freedom and flexibility in the ideation process. To achieve our primary function, we identified five secondary functions which would need to be fulfilled:

- Integrate with Vacuum Chamber
- Damp Vibrations
- Hold Microscope
- Make Sample Visible from Outside Device
- Facilitate Manipulation of Samples.

Once these sub-functions had been identified, we began to generate ideas for each in a series of ideation sessions. We employed several techniques, including brainstorming, brain-dumping, and worst idea generation. We found the method of "Worst Possible Idea" particularly helpful. Using this method we chose a function, and then as a group ideated all of the worst ideas we could come up with. This method is detailed in Figure 4. We found it particularly helpful to increase our creativity and look at more out-of-the-box solutions to different functions.

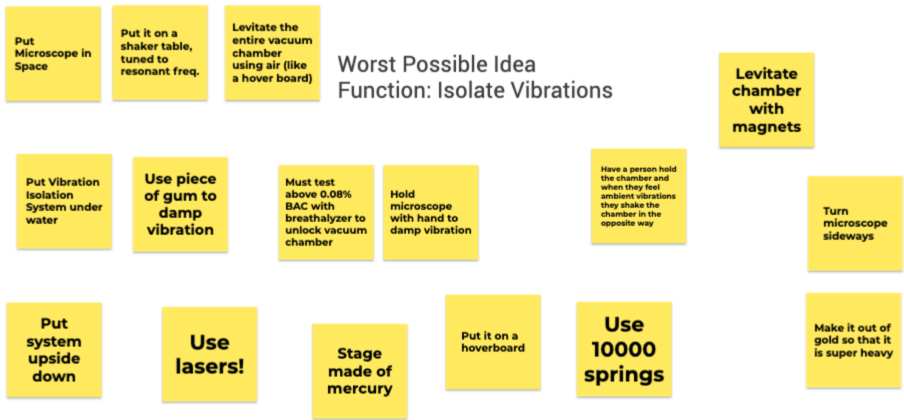


Figure 4: Worst Possible Idea. Using the brainstorm technique of worst possible idea, we ideated ridiculous solutions to the function of isolating vibrations.

After a number of different ideation sessions we spent time building concept models of some of our ideas. The purpose of this was both to test their feasibility, as well as continue challenging us to think of other ideas. Some of our models are shown below in Figures 5, 6, 7, and 8. Additionally, a complete list of our ideation ideas are documented in Appendix C.

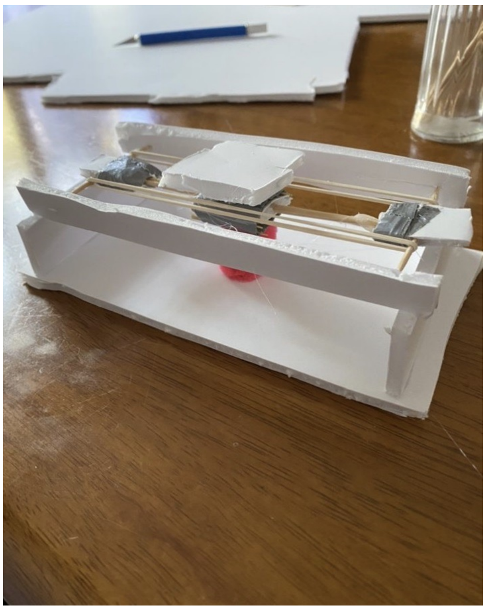


Figure 5: Ideation Model 1.

Figure 6: Ideation Model 2.

Figure 7: Ideation Model 3.

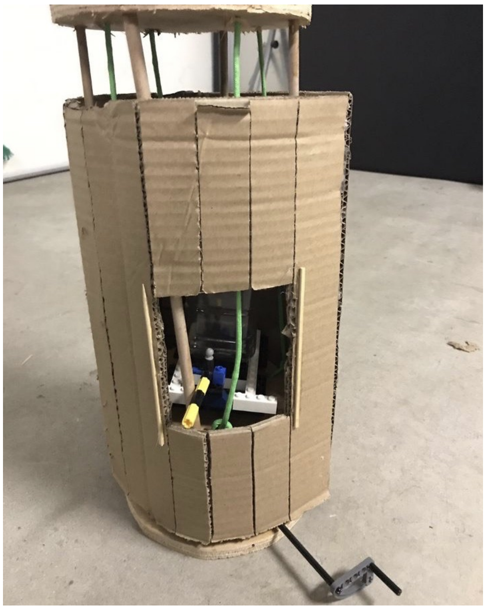


Figure 8: Ideation Model 4.

4.2 Idea Refinement and Selection

We next moved to start refining our ideas and choosing the best ones. We accomplished this first using Pugh matrices. For each Pugh matrix, we sketched our top five ideas and compared each idea to a datum design idea to determine the most effective idea using our previously selected criteria. This allowed us to narrow down our design ideas for each function. In most cases, the datum used was Mr. Le's design. There were several functions in which our original designs did not exceed the performance of the datum. In those instances, we concluded that we had validated Mr. Le's original design and will implement his concept in our final design for that function. The Pugh matrices used to compare our functions can be found in Appendix D.

We next generated a morphological matrix to combine our function-level designs into various system-level combinations of our top ideas. Using the morphological matrix, we were able to determine our top four system-level ideas which are presented below in Figures 9, 10, 11, and 12.

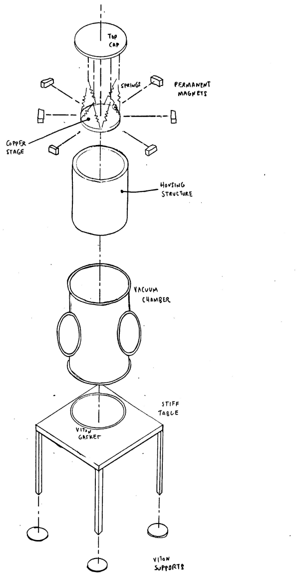


Figure 9: This design featured an aluminium housing, a ring of permanent magnets surrounding the copper stage, and six stainless steel springs for the mass-spring system. This design scored lower than our chosen concept due to a lack of adjustability of the magnetic damping mechanism.

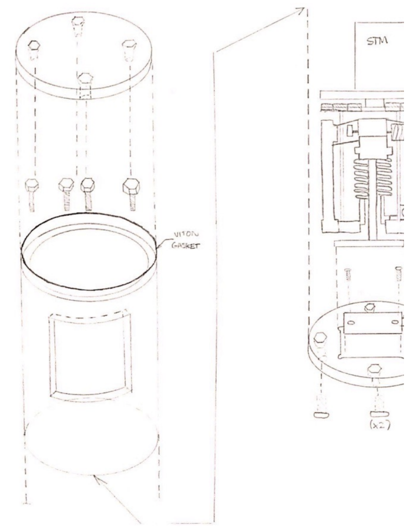


Figure 10: This design featured an titanium housing, a negative stiffness spring system, and an adjustable magnetic damping mechanism. This system scored lower than our top choice due to the added expense of a titanium housing as well as the added complexity of the negative stiffness spring system

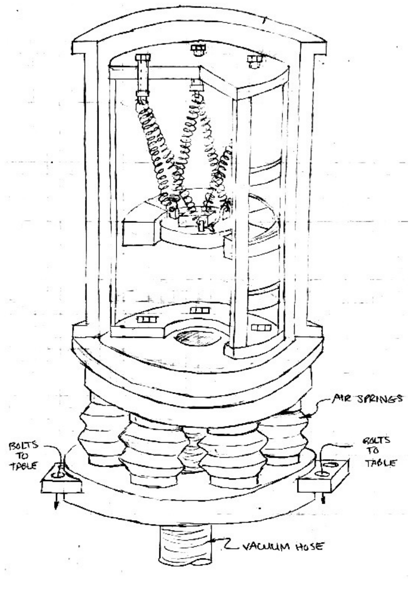


Figure 11: This design featured a large ring of magnets surrounding the stage to provide eddy-current damping and spring-mass system hanging on springs. The chamber is mounted on air springs to isolate from the environment.

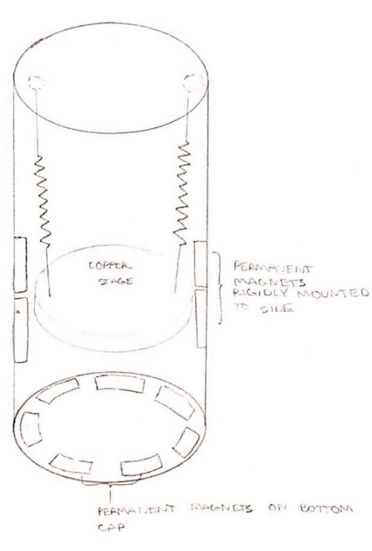


Figure 12: This design featured an titanium housing, a negative spring system, and an adjustable magnetic damping mechanism

The morphological matrix helped us generate top-level concepts which we then compared in our weighted decision matrix. From the morphological matrix, we were also able to conclude that each function of our system-level design is independent from the others and thus the best design could be attained by selecting the best idea for each function. Appendix E shows the morphological matrix that was used.

We then placed our top ideas into a weighted decision matrix to determine the best overall system design. We did this by listing all of our criteria from our House of Quality and determining the weight of each criteria. We then scored each of our system level designs to determine the best solution. Criteria that were rated as pass/fail were listed as constraints and only the designs which met all constraints were considered. The remaining criteria were weighted according to the importance to the user as discussed in the sponsor interviews. Our weighted decision matrix can be found in Appendix F. The highest-scoring system-level design can be seen below in Figure 13.

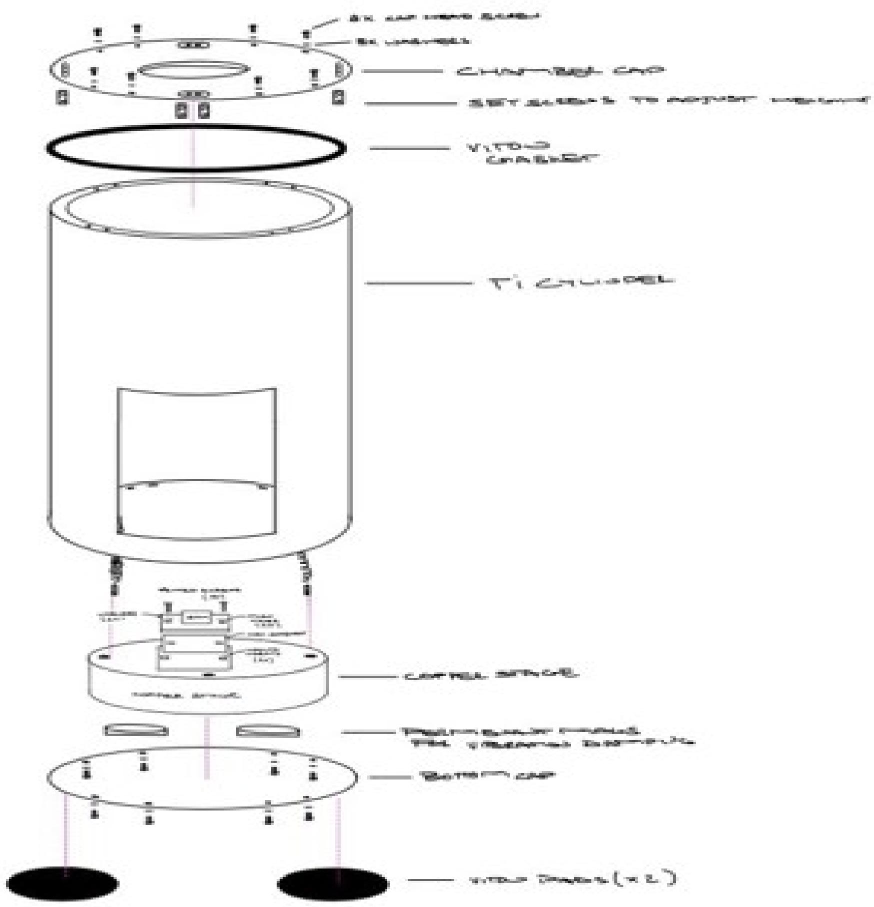


Figure 13: Our highest-scoring system-level concept. A single stage VIS consisting of an aluminum housing with cutouts for viewing the sample, a copper stage hung by springs from the cylinder cap, and an interposer plate and viton pad between the microscope and the stage.

We discussed this design with Dr. Scott and Mr. Le, and incorporated their feedback into our final design. They liked the viton damping between the aluminum beams of the cart from one of our other designs and suggested we improve the adjustability of the height of the permanent magnets in our eddy-current damping system. After further ideation and modeling to test our ideas, we arrived at the following concept design.

4.3 Selected Concept

The selected design concept for the scanning tunneling microscope vibration isolation system utilizes components of Mr. Le's concept design as well as additional design concepts created by our senior project team. The entire vibration isolation system sits inside of the high vacuum chamber which is mounted to an extruded aluminum frame This entire frame is supported by 4 castor wheels. A rough model of this system was made with Solidworks and is shown in Figure 14. Additionally, the cart is set on top of an elastomeric material (likely Viton) in order to dampen vibrations.

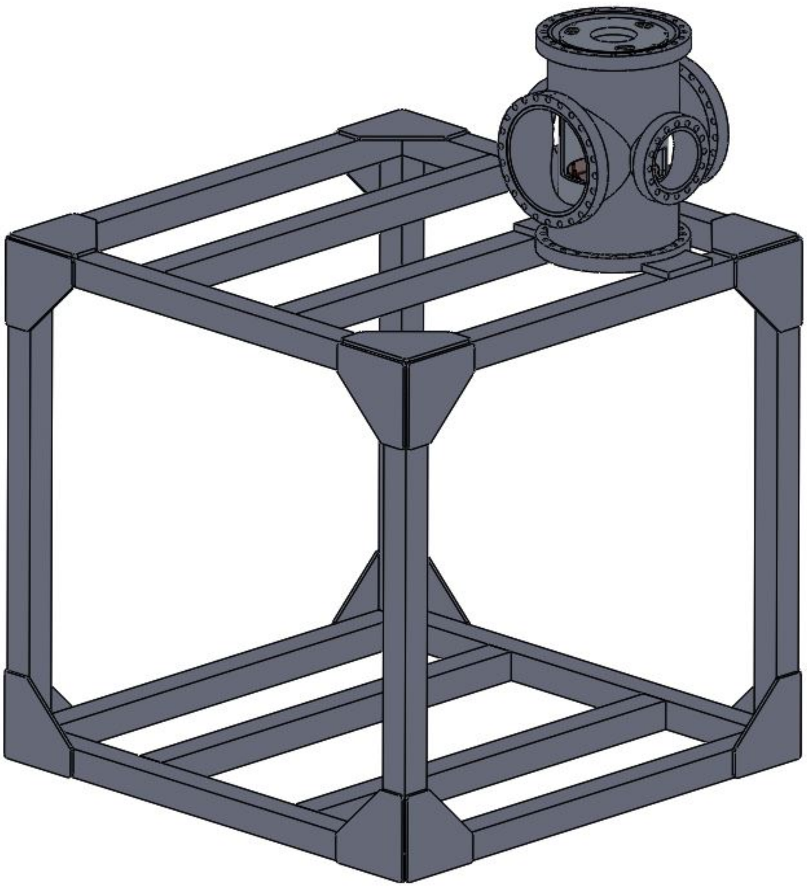


Figure 14: Preliminary CAD System Model. CAD model of the vacuum chamber and extruded aluminum frame. The vibration isolation system sits inside the main channel in the vacuum chamber. What is not shown is a rotary vane vacuum pump which is also mounted to the aluminum frame. A hose connects the inlet of the vacuum pump to a port in the bottom of the vacuum chamber.

The vibration isolation system sits within the vacuum chamber. It consists of a rigid exterior housing with cutouts made from aluminum which has a top and bottom cap, each of which are screwed into the housing. There is a Viton gasket between each cap and the housing to aid in damping any vibrations. The copper stage is suspended by 6 stainless steel springs, which hang from the top cap at angles. The reason for this is that by angling the springs, it helps to counteract any horizontal tipping or rotations. The springs are mounted using stud anchors, which are mounted in both the top cap and the stage. The STM is mounted on top of the copper stage and below the stage is a magnetic adjustment mechanism. This assembly can be seen in an exploded view in Figure 15. In Figure 16, the entire vibration isolation system is shown with multiple orientations and positions of each component. Not shown are the springs which connect the hanging copper stage with the top cap using stud anchors.

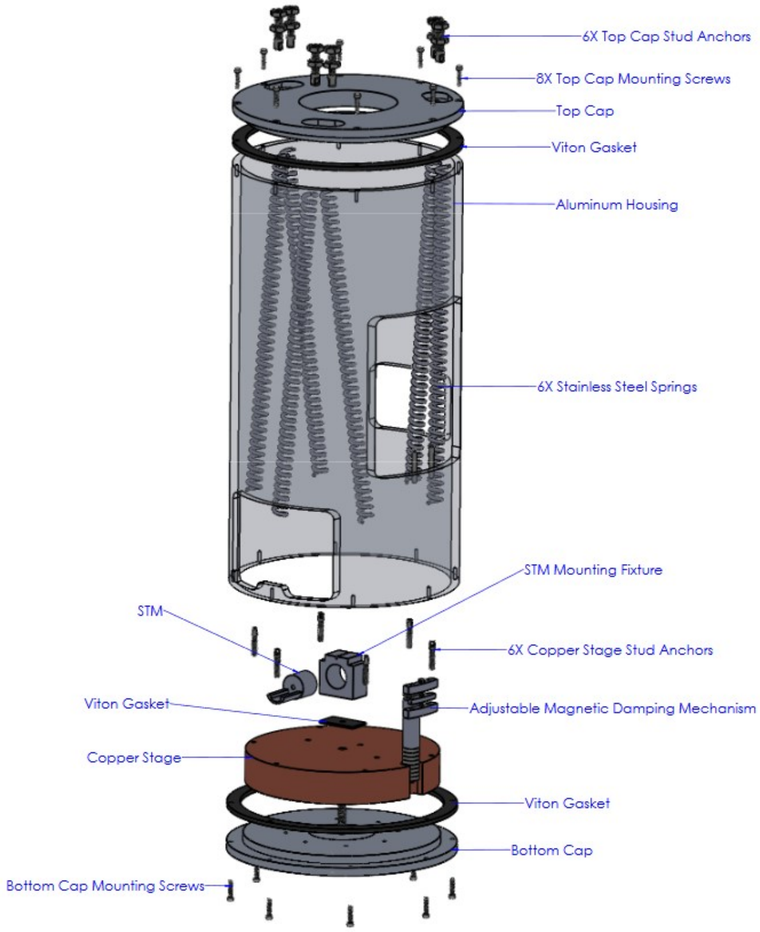


Figure 15: Vibration Isolation System Exploded View. Exploded view of the main components of the vibration isolation system. Main components are labeled, including the top cap, STM, copper stage, magnetic adjustment mechanism, aluminum housing, bottom cap, Viton Gaskets, stainless steel springs, mounting screws, and stud anchors. The aluminum housing with cutouts is transparent, so the springs are visable. Only one adjustable magnetic damping mechanism is shown, however the final assembly should have 2 mechanisms, separated by 180 about the copper stage. This entire assembly sits inside the vacuum chamber.

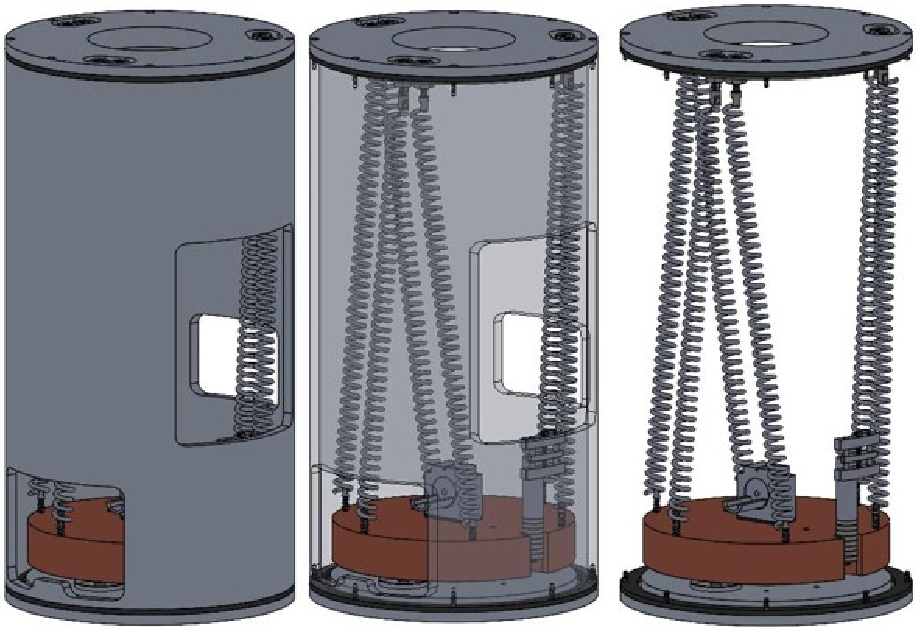


Figure 16: Vibration Isolation System CAD shown with increasing frame transparency. The top cap, bottom cap, frame, Viton gaskets, and copper stage were made by Toan Le. The socket screw used to adjust the height of the permanent magnet assembly is right below the main frame cut out. This means magnetic damping can be adjusted without the need to remove the entire frame from the vacuum chamber. Additionally, 6 stainless steel springs are shown, which are angled and suspend the copper stage. Two additional cutouts function to allow viewing and manipulation of the sample.

4.3.1 Preliminary Magnetic Adjustment Mechanism

The magnetic adjustment mechanism was designed to allow the operator of the STM to easily change the distance between the permanent magnets and copper stage. Allowing the operator to adjust this gives them the ability to easily change the damping coefficient and allow for fine tune adjustments to be performed in order to achieve the highest quality images from the STM. The magnetic adjustment mechanism is shown in Figure 17. The copper stage was located such that there was only 0.4" of clearance space beneath it and 0.3" on the side of it. This space constraint eliminated most concept designs and for this reason we needed to think outside the box. Many popular permanent magnet damping designs vary the damping coefficient by changing the distance between the permanent magnet and the conductive/non-magnetic surface, With our design, however, the damping coefficient can be varied by changing the number of "active magnets". "Active magnets" refers to the number of permanent magnets below the top face of the copper stage. This mechanism would require a slot to be cut out of the copper stage. With only a 0.25" long, 0.6" diameter slot cut into the side of the copper stage, this mechanism could be incorporated. Furthermore, the final design should have two mechanisms, each 180°from each other, to provide symmetrical damping forces about the center of the copper stage.

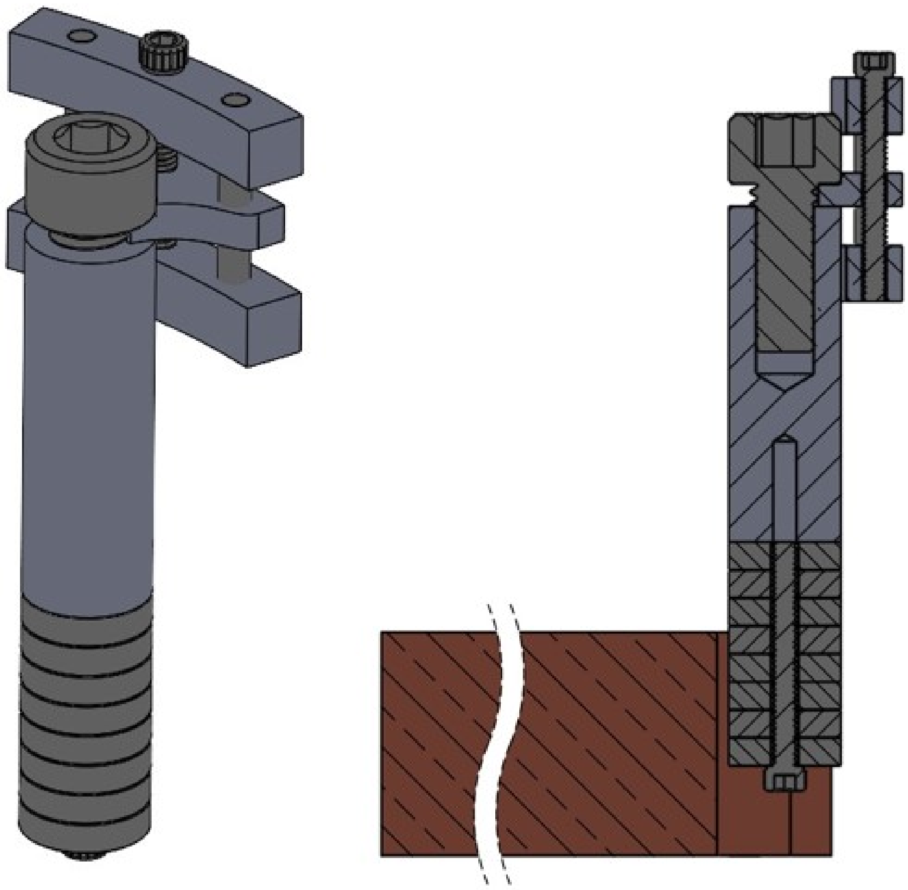


Figure 17: Isometric and section view of the adjustable passive magnetic damping mechanism. This assembly consists of an aluminum rod, to which a series of cylindrical permanent magnets are stacked. This rod-magnet assembly is attached to a center bracket which has 3 holes; the outermost holes have a clearance fit with 1/8" steel dowel pins. The center hole is internally threaded and mates with a silver plated socket head screw. The top and bottom bracket also have 3 holes; the two outermost holes are press fit with the steel dowel pins. The diameter of the center hole, however, allows for clearance between a silver plated socket head screw. As a result, when the silver plated screw rotates, the center bracket/rod-magnet assembly can be moved up and down.

4.3.2 Scanning Tunneling Microscope Stage Attachment

The next component of our selected concept is the attachment of the STM to the copper stage. An isometric exploded view of the microscope mounting mechanism is shown in Figure 18. Both the microscope and aluminum fixture in which the microscope is mounted are possessed by Dr. Gregory Scott. An isometric exploded view of the microscope mounting mechanism we propose is shown in Figure 18. This assembly is simple, and with a Viton gasket compressed between the aluminum fixture and copper stage, vibration will be damped at the interface.

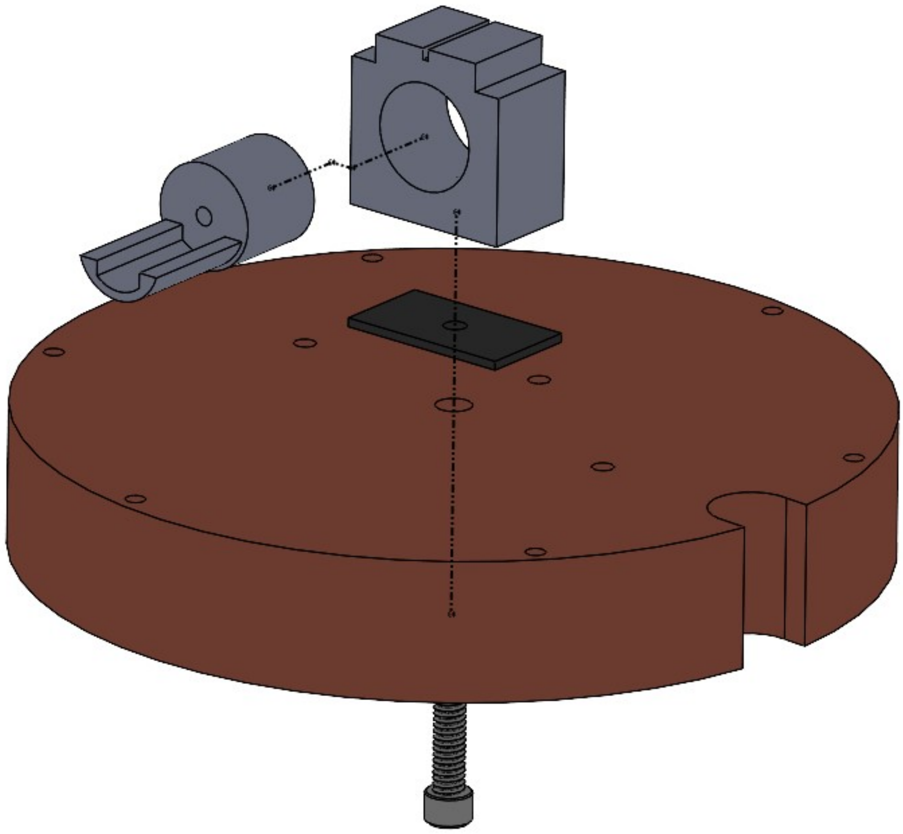


Figure 18: Isometric exploded view of the mechanism used to mount the STM to the copper stage. A thru hole is located in the middle of the copper stage with a diameter such that it fits with clearance with a screw. The aluminum fixture which holds the STM has an internally threaded hole which mates with the same screw. A Viton gasket is placed between the copper and aluminum interface. Finally, a silver plated and vented screw fastens the aluminum fixture to the copper stage and also compresses the Viton gasket.

4.3.3 Concept Prototype

The concept prototype for our project is picture below in Figures 19, 20, 21, and 22. We decided not to model the vacuum chamber and cart because these aspects of the system are already defined. The aluminum housing and copper housing are modeled using cardboard tubes. The STM was made using a 3D printer and is placed atop the copper stage which is hanging from the top cap using strings that are meant to represent the springs. The magnetic adjustment mechanism can be seen in Figures 21 and 22 and was also made using a 3D printer. This concept prototype proved to be an important process for our team, as it helped us understand, physically, the spacial constraint between the copper stage and the aluminum housing. The magnetic adjustment mechanism shown in Figure 22 was more than 2" tall. The required space between the bottom of the copper stage and bottom of the aluminum housing is approximately 0.5". For this reason, we pivoted from a damping mechanism located underneath the copper stage to one located on the side of it. This updated magnetic damping mechanism is shown in our updated CAD models.

Figure 19: Hanging Copper Stage.

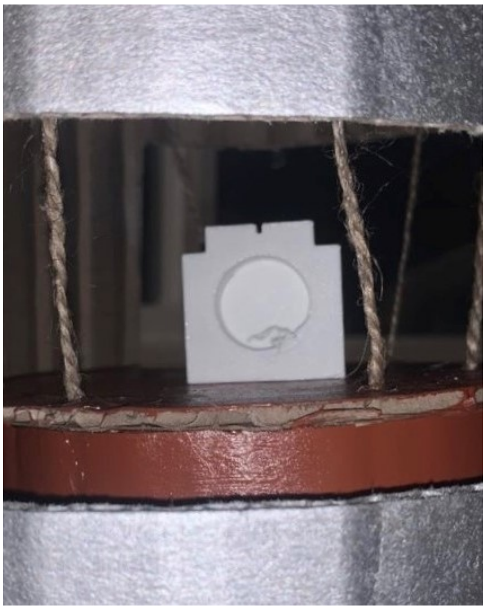


Figure 20: Close Up of Viewing Area through the side window.

Figure 21: Aluminum Housing with bottom window pictured.

Figure 22: Magnetic Adjustment Mechanism seen through bottom window.

4.4 Preliminary Finite Element Analysis

Analysis was performed on the spring mass system which was detailed in the concept prototype and explained in "Magnetic and Elastomeric Damping Effects on the Vibration Amplitude of a Vibration Isolation System" [14]. Our team utilized the MATLAB code generated in the quarter car simulation from ME 326 to create a tool which can input data and output the response of the VIS system. Once we have access to the lab, we hope to obtain data using an accelerometer, to understand the displacement of the laboratory floor. With this data we can use our modified quarter car simulation code to obtain a theoretical output for the microscope stage. Additionally, we have began to simulate the motion of simple cantilever beams with base excitations.

In order to justify the concept choices of Mr. Le concerning the aluminum housing, preliminary analysis in Abaqus was performed. The main goal of this analysis was to confirm the first three natural frequencies of the aluminum housing using finite element analysis (FEA). The results of the simulations can be seen in Figure 23.

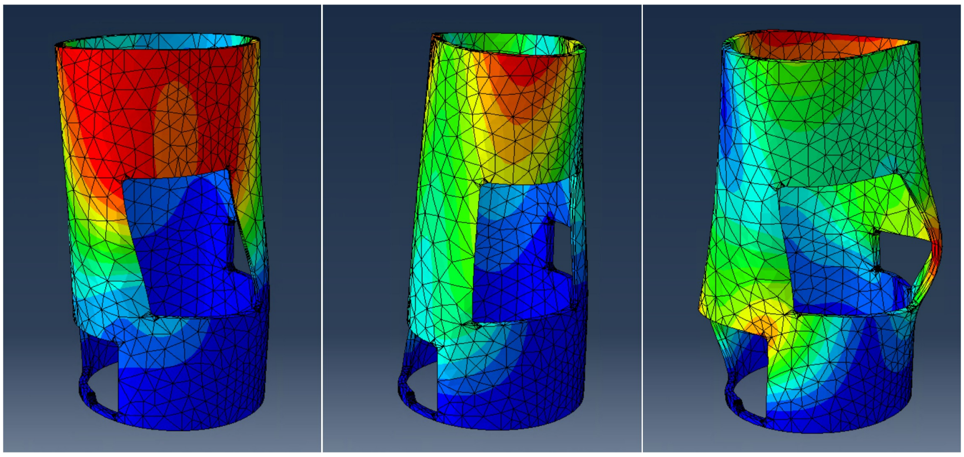


Figure 23: Aluminum Housing Natural Frequencies. The first three natural frequencies of the aluminum housing, modeled in Abaqus.

The results from the simulation validated the simulated results done by Mr. Le. While some parameters were chosen differently in the simulation, leading to minor differences in values for the first three natural frequencies, the results agree to within five percent. In Table 4 the values from our simulations, Mr. Le's simulation and the percent difference can be seen.

Table 4: Aluminum housing first three natural frequencies.

	Mode 1 (Hz)	Mode 2 (Hz)	Mode 3 (Hz)
F-41 Senior Project	492.6	639.3	1169.3
Toan Le	539.34	642.07	752.27
Percent Difference	8.67	0.43	35.67

For the simulation, a "tet" mesh pattern was chosen over the aluminum housing, with every node spaced by 1 in. We plan to run additional analysis, specifically of the entire housing structure. This will include the top cap, bottom cap and Viton gaskets. We believe the extra rigidity of the top and bottom caps will help to further raise the first natural frequency of the housing structure.

4.5 Risks, Challenges, and Unknowns

The ultra high vacuum environment poses potential hazards and dangers. Moving forward, we plan to cross reference every material used with LIGO's Vacuum Compatable Materials list [**ligo**]. This list excludes most plastics, including Delrin and Teflon. Additionally, all screws which mate in threaded holes which are not thru, need to be vented. We also need to make sure that any pre-treatment of materials is suitable for high vacuum. In addition, if pre-treatment of the system is important, such as baking, powder-free latex gloves must be used to prevent finger print greasing.

5 Final Design

5.1 Overall Design Description

Our final design incorporates most of the features of the concept design. The basic design of our vibration isolation system consists of mass-spring system contained within an aluminum housing that sits within Dr. Scott's vacuum chamber. Figure 24 depicts an overview of the overall assembly which will be broken up into three sub-assemblies. The first of these is the mass spring system, which consists of the STM itself, the copper stage, six springs, six each of two different sizes of stud anchors, the top cap, and twelve hex nuts which are tightened onto the stud anchors on either side of the top cap. The next assembly is the magnetic adjustment assembly, which is integrated into the top cap and contains circular neodymium magnets press-fit into an aluminum swash plate with hardware to raise and lower the swash plate to vary the distance between the magnets and stage and thus the magnitude of the damping ratio attained. That system is described in greater detail in Section 5.1.2 below. The housing assembly consists of both sub assemblies attached to either end of an aluminum cylinder by sixteen small machine screws, with viton gaskets at the interface between metal components.

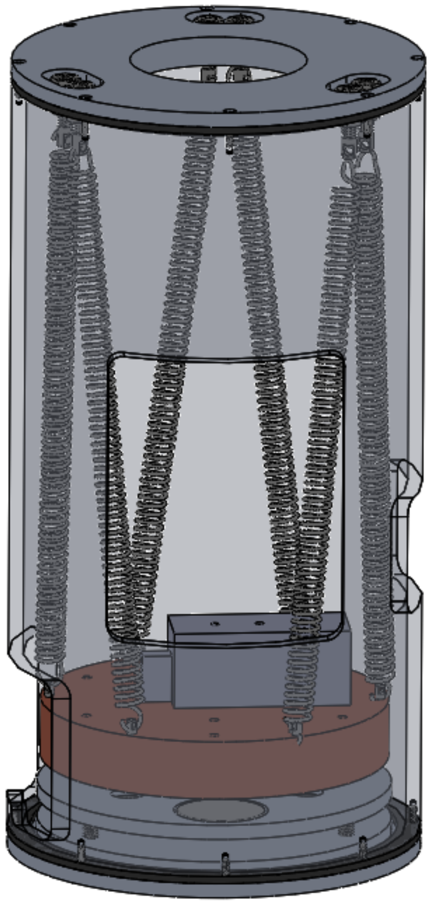


Figure 24: Assembly view of the final design of the vibration isolation system. Transparency of the aluminum cylinder has been changed so internal components can be seen.

The only significant changes we made from our concept design were to the magnetic adjustment system. After some modeling and research, we concluded it was better to return to Mr. Le's concept of placing the magnets below the copper stage, and designed a mechanism for adjusting the magnet height which can be accessed from the bottom of the housing without requiring the disassembly of the housing or the use of any specialized tools. The reasons for this change are three-fold: First, we wished to avoid the additional modifications to the copper stage, magnets, and housing that would have to be made prior to testing in order to accommodate the aluminum rods of our concept mechanism, which would make future changes to the mechanism either impossible or costly. Second, the ease of modelling and greater ability to fine tune the damping of flat magnets at a variable distance below the stage lends itself better to our plan of repeated testing and adjustment to fine-tune the damping ratio. Finally, with the help of CSM Equipment Technician Rob Brewster, we were able to come up with a design which would fit in the small vertical space between the stage and bottom cap yet still allow the variability of the magnet height from five to ten millimeters, the predicted range we would need. The new design solves this dilemma by placing the hardware necessary for the raising and lowering of the magnets in a cutout on the bottom face of the bottom cap, allowing us to utilize the entirety of the limited space below the stage for adjusting the height of the magnets.

The following sections describe in greater detail each major subsystem and component of our final design. A complete list of components can be found in the Indented Bill of Materials in G, and detailed drawings of each component we will be manufacturing can be found in H.

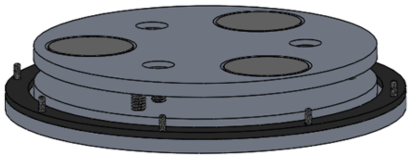
5.1.1 Mass-Spring Assembly

The mass-spring assembly has not been changed significantly since the concept design. The assembly can be seen in Figure 25. The scanning tunneling microscope and its fixture, provided by the sponsor, will be anchored to the copper stage with a single bolt. A footprint of the STM fixture will be cut from a sheet of viton and placed between the fixture and the stage to damp any vibration modes in which the stage and fixture move relative to each other. The six steel springs that were purchased by Mr. Le will be attached to the stage and the top cap with stud anchors in the angled configuration shown in the figure. In our preliminary testing, this configuration has proven to be best at restoring the stage to a purely vertical mode of vibration after a rotational or yaw vibration has been induced. This will allow our magnetic damping system to work most efficiently and predictably. The top cap will be machined from aluminum. The only major change to the components of this assembly is an increase in the spacing of the through holes for the stud anchors. In our original design, there was not enough space for a socket head to fit over one of the hex nuts when both nuts were in place. In manufacturing the part we may also change the configuration of the groove around the holes for the stud anchors, depending on our manufacturing capabilities.

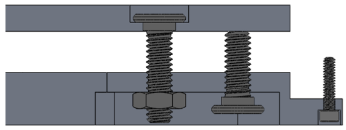
Figure 25: Assembly view of the mass-spring assembly.

5.1.2 Magnetic Adjustment Assembly

The new design for the magnetic adjustment assembly is comprised of the bottom cap, an aluminum swash plate to hold the magnets, and six bolts arranged in pairs as shown in Figure 26. The outer bolts screw into threaded holes in the bottom cap and can be tightened or loosened to raise or lower the swash plate. The inner bolts are longer and pass through drilled and counter-bored holes in the magnet swash plate and screw into threaded holes in the bottom plate. They will lock the swash plate in place by applying a downward force on the swash plate to oppose the upward force of the outer screws. A regular hex nut will be tightened onto the end of the inner screw sticking out of the bottom cap, and the hex nut and threaded hole in the bottom cap will act as jam nuts to prevent the bolts from loosening during regular operation of the STM. This mechanism can be better seen in the section view in Figure 26. The swash plate will be the same thickness as the magnets, to prevent added material into the space we need for adjustment, and the magnets will be press-fit into the swash plate. The number, size and strength of the magnets will be determined by testing of our structural prototype.



(a)



(b)

Figure 26: (a) Assembly view of the final design of the magnetic adjustment assembly. (b) Section view of magnetic adjustment assembly showing the adjustment mechanism. The right hand socket head screw raises and lowers the swash plate. The left hand screw, when tightened, applies counter-pressure to lock the height of the magnets.

5.1.3 Housing Assembly

The housing assembly will contain both sub-assemblies, as well as the aluminum cylinder, viton gaskets, and screws needed to connect the top and bottom caps to the cylinder. The aluminum cylinder will have windows cut into it to allow for the viewing and manipulation of the sample in the STM. The housing assembly has not been changed from that of our concept design.

5.2 Design Functionality

The goal of our design is to damp ambient vibrations at each step as they move from the environment to the STM. Incoming vibrations must pass through the housing, where viton gaskets between the cylinder and the two end caps provide visco-elastic damping of incoming vibrations. The cylinder's natural frequency is sufficiently above the frequency of incoming vibrations that the transmissibility ratio of the cylinder is pushed close to unity, minimizing the amplification of vibrations through the housing. Next, vibrations pass through the mass spring system, whose low spring constant and high mass cause its natural frequency to be very low (close to one Hz). Due to this low natural frequency, most incoming vibrations will fall into the system's isolation region, reducing their amplitude as they pass through the springs. Finally, more viton between the stage and the STM itself will damp the amplitude of any vibrations that have made it through the mass spring system.

5.3 Design Justification

In order to justify that our design will meet our engineering specifications, we decided to use a combination of empirical testing as well as simulation to develop theoretical models. These tests and analyses have provided us confidence in our design and will continue to give us better results as we continue to tune the models. There are three models that we have simulated in order to give us more insight into our three main system assemblies: the housing structure, the mass-spring system, and the permanent magnet damping system.

5.3.1 Housing Structure Analysis

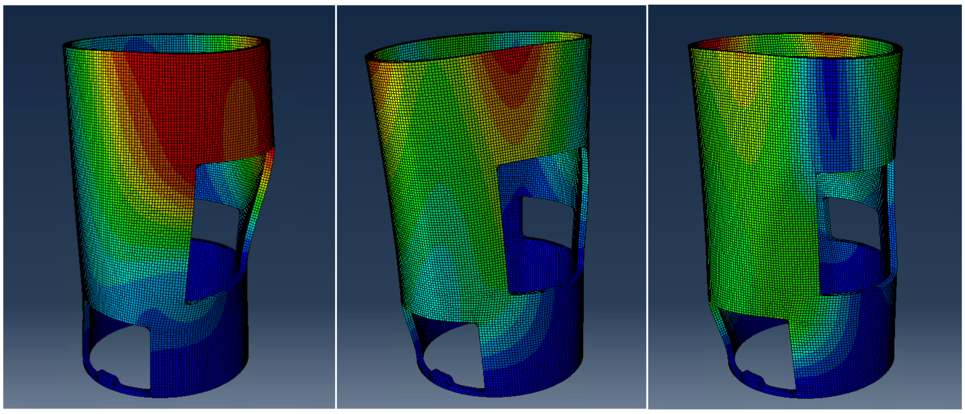


Figure 27: Aluminum Housing Natural Frequencies. The first three natural frequencies of the aluminum housing, modeled in Abaqus.

The housing structure was modeled in Abaqus in order to determine the first three natural frequencies. Our engineering specification for the housing requires it to have a minimum first mode of 400 Hz and as seen in Table 5, the modeled first natural frequency was 485.5 Hz which is within our specification. In our preliminary design review (PDR), we received feedback that we would obtain better results from our Abaqus model if we adjusted some of our parameters. The method used in the PDR was a "tet" mesh pattern, with a linear shape function and nodes spaced every 1.0 in. Since the PDR, we refined the model for more accurate results and adjusted some of the modeling techniques. Specifically, we changed the mesh to a "hex" pattern with a quadratic shape function and nodes spaced every 0.10 in. A complete list of the parameters and results from the Abaqus simulation can be found in Appendix J. We plan to continue justifying our design through testing of the tube on a vertical shaker table in order to verify our simulation results.

Table 5: Aluminum housing first three natural frequencies.

	Mode 1 (Hz)	Mode 2 (Hz)	Mode 3 (Hz)
PDR Model	492.6	639.3	1169.3
CDR Model	485.5	630.6	642.6
Percent Difference	1.46	1.36	58.14

5.3.2 Permanent Magnet Analysis

We began modeling the permanent magnet eddy-current damping in MATLAB. For this analysis, we modelled the permanent magnet flux densities in order to solve for the magnetic force and eventually the distance between the magnets and the copper stage necessary to achieve the required damping. We obtained a vector field of the magnetic flux densities which can be seen in Figure 28. The method of obtaining the magnetic flux densities was found using the current method [22]. Using the BrBz function shown in Appendix M, using Equations provided by Ebrahimi, we calculated the radial and longitudinal component of magnetic flux density for a cylindrical magnet [22]. This model equates the permanent magnet to a solenoid with only one turn. The equations presented were used to calculate the magnetic field at all points outside the solenoid. Based on first principals such as Lenz's law, Faraday's Law, and the Biot-Savart law, we then calculated the ratio of force to velocity due to eddy current damping and used it to numerically solve a non-linear, non-constant coefficient, 2nd order ODE to simulate the response of the stage with respect to time. The calculated magnetic flux density is three orders of magnitude lower than what is predicted from the Biot-Savart Law and therefore it can be seen in Appendix M that the associated response has a lower damping coefficient. We believe further investigation is needed to use this method for simulating the response of the stage.

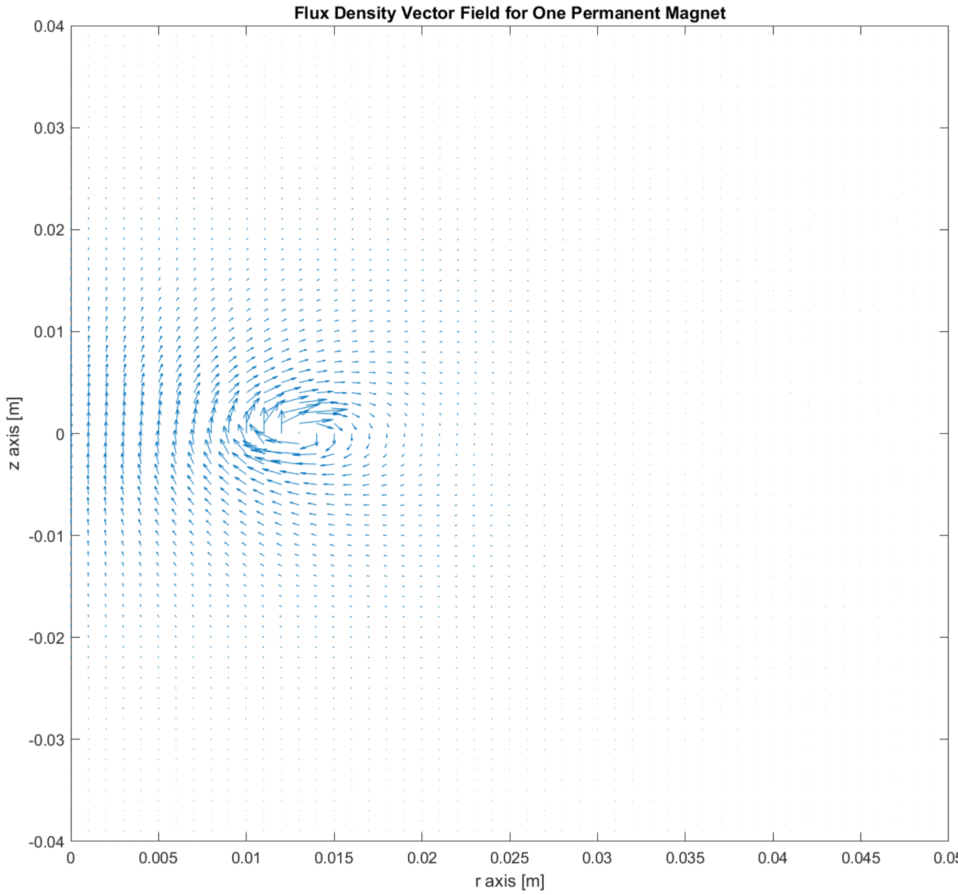


Figure 28: Flux Density Vector Field. The vector field of the flux densities in the r- and z-directions for one 0.25" thick, 1.5" diameter, N52 strength magnet.

We also conducted preliminary empirical testing of our magnetic damping. In order to do this we attached an accelerometer to the top of our copper stage and measured the output using an oscilloscope. We plan to continue this testing using different magnet sizes and strengths, as well as adjusting the distance between the magnets and copper stage in order to help us determine the optimal size, strength and number of permanent magnets. In Figure 29, the set up for our magnet testing is shown.

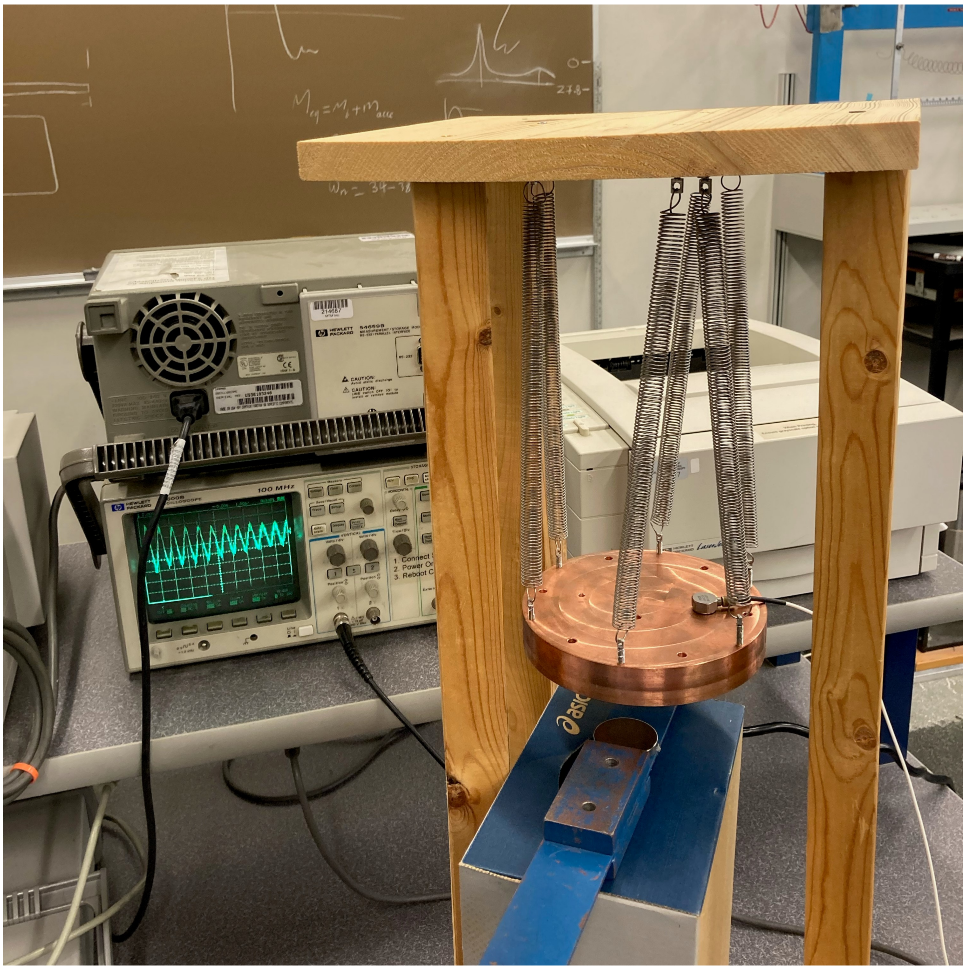


Figure 29: Permanent Magnet Empirical Testing. Setup of the permanent magnet empirical testing. The copper stage is mounted to the springs using stud anchors on both ends. An accelerometer is attached to the top of the stage and wired to the oscilloscope shown.

5.3.3 Mass-Spring System Analysis

Due to the complex behavior of the aluminum housing and the fact that the vibration input into the mass spring system depends on the behavior of the housing, we decided to analyze each system independently. Also, we will need to test each system independently in order to compare to our theoretical simulation results. In addition, we decided to assume sinusoidal vibration as opposed to random vibration. Although this does not accurately model the real environment that the VIS will be in, these assumptions allow us to simplify the analysis and testing. Therefore, we performed analysis for the mass spring system independent of the rest of our system using MATLAB and Simscape. In Figure 30, the Simscape model that was used to model the vertical mode of vibration is shown.

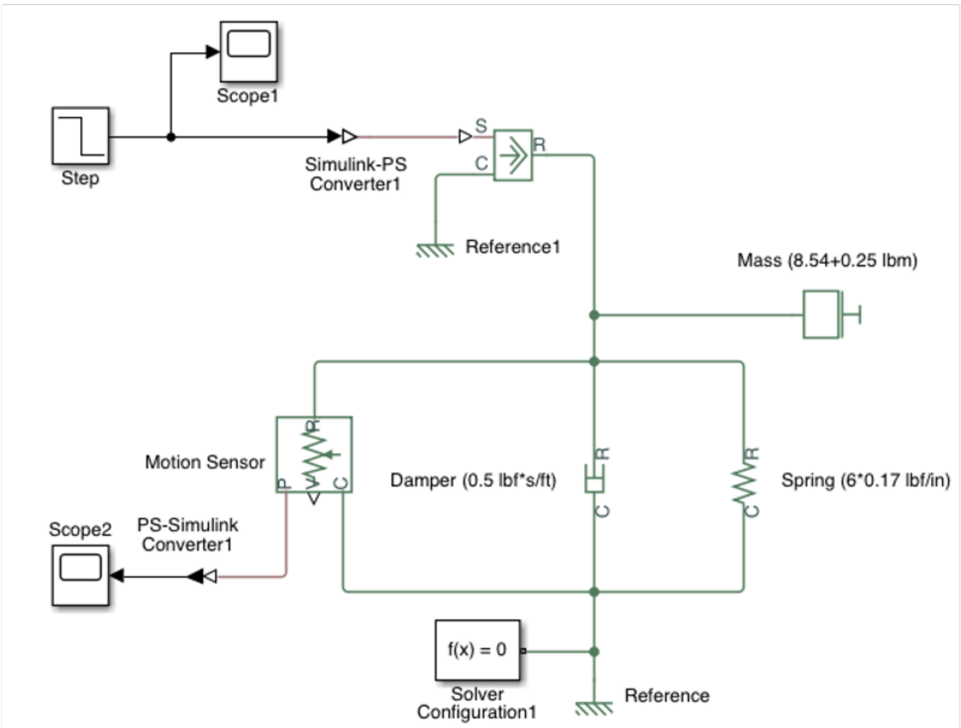


Figure 30: Simscape model for vertical mode of vibration. This model incorporates a linear translational spring with a spring constant of 1.02 lbf/in. Using the damping ratio of 0.21 found from the MATLAB analysis, the damped response was then plotted.

The purpose of the MATLAB analysis was to model the time response of the copper stage due to a sinusoidal excitation of the top plate where the springs integrate with the housing. Due to the fact that the six springs used to suspend the copper stage are mounted at angles, the mass springs system cannot be modelled as a linear, single degree-of-freedom system. However, to simplify the analysis, we began by assuming that the system was indeed linear as a first iteration. In general, the motion of the stage can be broken down into vertical translation and rotation. This neglects the yaw mode of vibration as this mode of vibration is not nearly as significant as the vertical and rotational modes. Because we already knew the mass of the copper stage and the stiffness and lengths of the springs, the goal of this analysis was to back solve for the necessary damping ratio based on a design maximum amplitude of the copper stage of 2 mm and settling time of 10 seconds. With our current design, we have six springs which each have a stiffness of 0.17 lbf/in and a copper stage with a mass of 8.54 lbm. As a first approximation, assuming that the springs are completely vertical and in parallel configuration with each other, the theoretical natural frequency for the vertical mode of vibration was found to be 1.22 Hz. This is below our design requirement of less than 2 Hz. Using this information, we found that a damping ratio of approximately 0.21 was needed to achieve the design requirement for the motion of the stage. Figure 31 shows the Simscape damped step response in the time domain. appen:MATLAB Analysis Results contains the both the undamped and damped theoretical step response in the time domain. In addition, appen:Preliminary Testing contains the results from our initial testing as well.

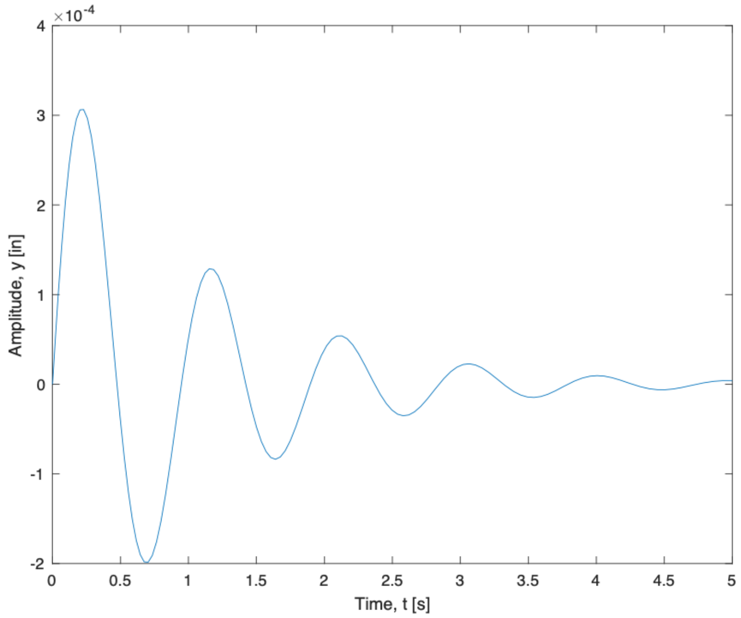


Figure 31: Simscape model damped time response based on a step input.

Finally, for simplicity, the rotational mode of vibration was also initially modelled as being linear by approximating the equivalent rotational stiffness due to the angle of the six translational springs. From this analysis and knowing the mass moment of inertia of the copper stage, we determined the rotational natural frequency to be approximately 1.52 Hz. After CDR, our goal is to significantly improve upon the vertical and rotational models by considering the non-linearity of the system due to the angle of the springs.

5.4 Safety, Maintenance and Repair

We anticipate few safety concerns for our final vibration isolation system assembly. The one concern we identified in our Design Hazard Checklist found in Appendix L was the possibilities of sharp edges on our assembly. In order to mitigate this concern we plan to finish each of our parts to rid it of any burrs or sharp edges. This finishing process will likely occur with a fine sand paper.

After analyzing failure modes in our Failure Mode & Effects analysis we have determined some very minor causes for concern. If there are any extreme vibrations that shock the system it has the potential to stress the springs past their elastic limit causing plastic deformation and rendering their properties different. However, we believe this would only be an issue when the vibration isolation system is being installed and is not a cause of great worry. Additional failure modes are most likely to occur when the VIS is outside of the vacuum chamber which should be very infrequently. These additional failure modes are described in more detail in Appendix N. For these reason, we anticipate maintenance of our device to be very minimal. There will be nearly no wear on any components and it will not be subject to any repeated force of any strength.

We designed the vibration isolation system to be as easy to repair as possible. Per our specifications, it should be easy to remove from the vacuum chamber, and when designing the overall system we kept simplicity in mind. This should make repair/replacement of any part easy because the entire system is easy to assemble and disassemble.

5.5 Cost Analysis

For this project, we have been given a total budget of $1000 for building and testing our VIS. This includes purchasing the necessary materials and for paying for any additional costs that might arise from manufacturing. Because this is a continuation of Mr Le's Master's Thesis, some of the major cost items have already been purchased including the aluminum housing, copper stage, springs, and stud anchors. This greatly reduces our expected cost, as the copper stage was over three hundred dollars alone. Currently, the only purchase we have made was for permanent magnets for preliminary testing. These magnets cost us a total of $114.02 not including tax. Therefore, we still have approximately $870. We still need to purchase the aluminum disks for the top and bottom caps as well as for the magnet fixture, the silver plated screws, various viton sheets and gaskets, additional magnets once the exact quantity has been determined, and various hardware for fixtures needed to secure the housing to the vertical shake table in the vibrations lab.

Finally, we need to be save room in the budget in case we need to hire shop techs to perform additional machining using the CNC machines in the Mustang 60 machine shop. A complete project budget can be found in Appendix I.

6 Manufacturing Plan

6.1 Procurement

As discussed previously, the aluminum cylinder, copper stage, springs, stud anchors, and hex nuts were already purchased when Mr. Le was working on his masters' thesis, and these components will be retained in our final design. For the bottom and top caps, we purchased 7" square aluminum stock with a thickness of 3/4" from McMaster. Additionally, we purchased viton sheets, a tight tolerance 6" 6061 aluminum disc, stainless steel hex nuts, ultra-low-profile hex screws for the magnetic adjustment assembly, and top cap screws for the top and bottom caps from McMaster. Some of the parts from McMaster that we purchased are not ultra-high-vacuum compatible and were procured for testing. We plan to buy all the final vacuum-compatible screws and nuts that are silver plated from a manufacturer called UC Components. We procured the final design magnets, which are 1.5"x0.25" N52 neodyminium magnets from K&J Magnetics.

6.2 Manufacturing

The following sequence of operations was used to manufacture the parts for our final design. Further information for each part can be found in the Appendix H.

6.2.1 Aluminum Cylinder

First, we trimmed the cylinder to 13.25" in length using a circular saw. Next, we transferred the cylinder to a large chuck on a lathe and faced one end of the cylinder until it was flat. We then flipped the cylinder in the chuck and faced the other end to 13" in length with a parallel tolerance of 0.005". Next, we used a plasma cutter to cut out the approximate window shapes in the aluminum housing. The aluminum cylinder after the plasma cutting procedure is pictured in Figure 32. To clean up the windows we transferred the cylinder to a mill and placed two V-shaped 3-D printed blocks beneath the each end for support, with clamps to hold the part down. Then, using a 3/8" end mill we cut the windows out and cleaned up the edges using a chamfer tool. Figure 33 shows the windows in the aluminum housing being finished using an end mill.

Figure 32: Aluminum cylinder after the plasma cutting procedure.

Figure 33: Aluminum housing windows being cleaned up on a manual mill using a 3/8" end mill.

Lastly, we placed the cylinder face end down on a drill press, and drilled and tapped the eight #4-40 by quarter-inch-deep holes, equally spaced around diameter, using the already CNCed top cap as a guide. We repeated the procedure on the opposite face to drill the holes for the bottom cap. The final housing part is pictured in Figure 34.

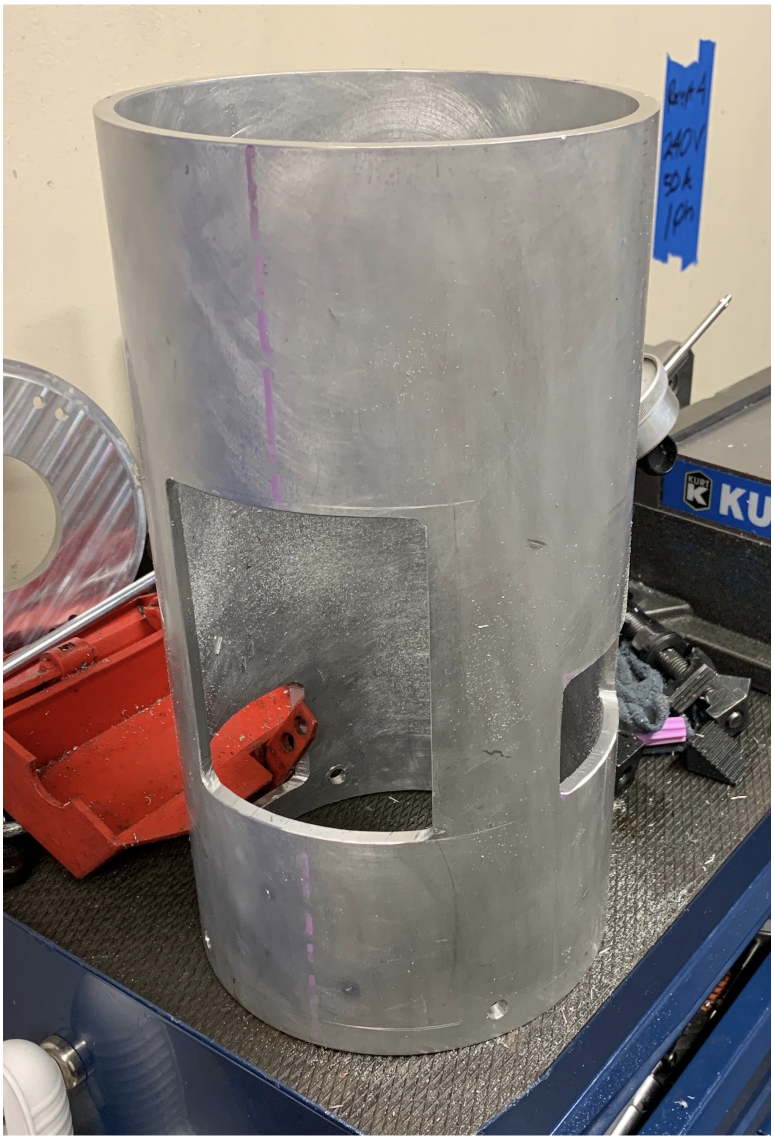


Figure 34: Aluminum housing after all manufacturing procedures have been performed.

6.2.2 Top Cap

The top cap was machined using a CNC machine. In order to prepare the part for machining, we the built in Autodesk Fusion 360 CAM software to determine the tool paths and fixtures necessary to machine the part. The machining process was a two part procedure. The first procedure had the entire part fixed in a vise and a 1/2" end mill was used to face and cut the top of the top cap. The part after the first procedure is pictured in Figure 35. The second procedure involved securing part in soft jaws and using a 1/4" end mill to face the bottom of the part, drill the holes and counterbore the three sections pictured in Figure 36 where the stud anchors holding the springs will be secured.

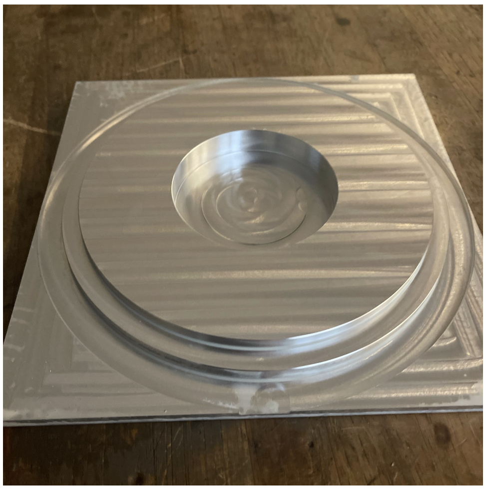


Figure 35: First procedure for the CNC of the top cap. This procedure faced and cut the features on the top of the top cap.

Figure 36: Final top cap part after the second procedure in the CNC. Additonal hand finishing such as deburring was done as well.

6.2.3 Bottom Cap

The bottom cap was machined using a CNC machine. First, the machines cutting and tooling paths were determined using the Autodesk Fusion CAM software. Then the process was done in two parts, with the first part performed in a clamp and taking off the material of the bottom face of the cap. The second operation was performed in soft jaws with the part flipped 180 degrees. This operation cut the contours of the top of the part and completed all the operations for this part.

6.2.4 Viton Sheet

The Viton gaskets were manufactured using scissors. Due to the low tolerances of the part, we simply outlined the necessary cuts using a paint pen, and used scissors to cut out the gaskets.

6.2.5 Magnet Swash Plate

The magnetic swash plate was cut using a manual mill and rotary table. First we purchased 6061 Aluminum Rod stock which was .25" thick and 6" in diameter. We then used an H sized drill bit and a 9/16 sized drill bit to cut the thru and counter bored holes through which the low profile screws will attach to the bottom cap. Because the permanent magnets were press fit into this plate, we used a boring bar to cut holes to precise tolerances of ± 0.001 inches.

6.3 Assembly

6.3.1 Mass Spring Assembly

First, attach the housing to the copper stage using the fixture and bolt provided by Dr. Scott, with the viton sheet footprint between the fixture and the stage. Next, screw the six 1/4"-20 stud anchors into the six tapped holes in the top cap, and the six #8-32 stud anchors into the six tapped holes in the copper stage. Hook one end of each springs into a stud anchor in the copper stage and the other end into a stud anchor in the top cap, arranging the springs in the configuration seen in the drawing.

6.3.2 Magnetic Adjustment Assembly

Using a punch press, press fit magnets into the one-inch holes in magnetic swash plate. Use as little force as possible to avoid breaking the brittle magnets. It may even be helpful to heat the aluminum swash plate before inserting the magnets, as long as as the temperature is kept below 175°F, as the magnets may be damaged above that temperature. Next, screw the three $\frac{1}{4}$-20x5/8" socket screws into the outer of the two rings of tapped holes in bottom cap. Insert the three $\frac{1}{4}$-20x7/8" socket screws into the counter-bored through holes in magnetic swash plate and screw them into the inner of the two rings of tapped holes in the bottom cap until the bottom of the swash plate presses against the outer three screws. To lock the height of the swash plate, crew hex nuts onto the threaded ends protruding from bottom of bottom cap. 37 and **??** show the top and side views of the magnetic adjustment assembly, respectively.

Figure 37: Top view of magnetic adjustment mechanism after final assembly.

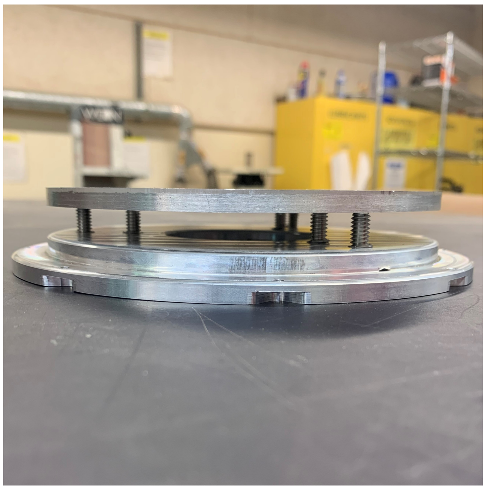


Figure 38: Side view of magnetic adjustment mechanism after final assembly.

6.3.3 Housing Assembly

Place a viton gasket into the outer lip of the top and bottom cap. Then, screw the top and bottom caps into place with sixteen #4-40 vented screws. Figure 39 shows the final assembly of the VIS.

Figure 39: Picture of housing assembly.

6.4 Manufacturing Challenges

There were many challenges that we faced in the manufacturing of our final design. First, we had little CNC machining experience, so it was necessary to find someone who could help us with the process who had access to a CNC machine. We were able to overcome this challenge with the help of Peter Brewster and Peter Moe-Lange of the Cal Poly ITP department who offered their services in helping to CAM and CNC both the top and bottom caps. Another challenge we faced was machining the windows in the aluminum housing. We initially thought a rotary table on a manual mill was the best solution, however it was complicated to set up and did not secure our part as well as possible. We were able to overcome this challenge by cutting out a majority of the windows using a plasma cutter and then cleaning them up on a manual mill. On the manual mill we were able to mount the housing using large V-blocks and two clamps on either end. While this set up was not the most secure, it allowed us to successfully clean the edges of the windows and keep tolerances within 0.050" which was enough for the windows. Overall, many of our challenges were helped by the aid of others, and did not present large issues for the manufacturing other than that they set back our timeline to complete the manufacturing of our design.

6.5 Recommendations for Future Production

It is unlikely that our STM design will be replicated because it is a very unique design for a specific situation. However, if someone does wish to use the same design there are a few recommendations we have the manufacturing of the assembly. The first, and most important, is to use a CNC for all of the parts. This is because the intricacy of the parts and the tight tolerances make it difficult to exact by hand, except by an experience machinist. Using a CNC for the aluminum housing would cut down on the lead times and work times greatly, and produce a higher tolerance part in the end.

7 Design Verification

We performed the following tests in order to verify that our design meets our engineering specifications. These tests helped us quantify various parameters listed in our specifications such as component natural frequencies, magnet damping ratios and vibration amplitudes. Testing was performed in the Mechanical Vibrations lab at California Polytechnic State University.

First, we experimentally verified the first natural frequency of aluminum housing chamber. Our modeling predicts a natural frequency of 485.5 Hz which is higher than our minimum goal of 400 Hz. In order to test this, we attached the tube to the vertical shaker table in the vibrations laboratory using L-brackets. With accelerometers attached to the top, we were able to determine the natural frequency and corroborate our Abaqus simulation. The natural frequency we measured was around 500 Hz. We planned additional shake table testing of the housing with viton if time allotted, but this testing was not completed. We would recommend future testing and analysis to determine precisely the effect of the Viton on the overall system and the differences between Viton of different grades, thicknesses, and levels of compression.

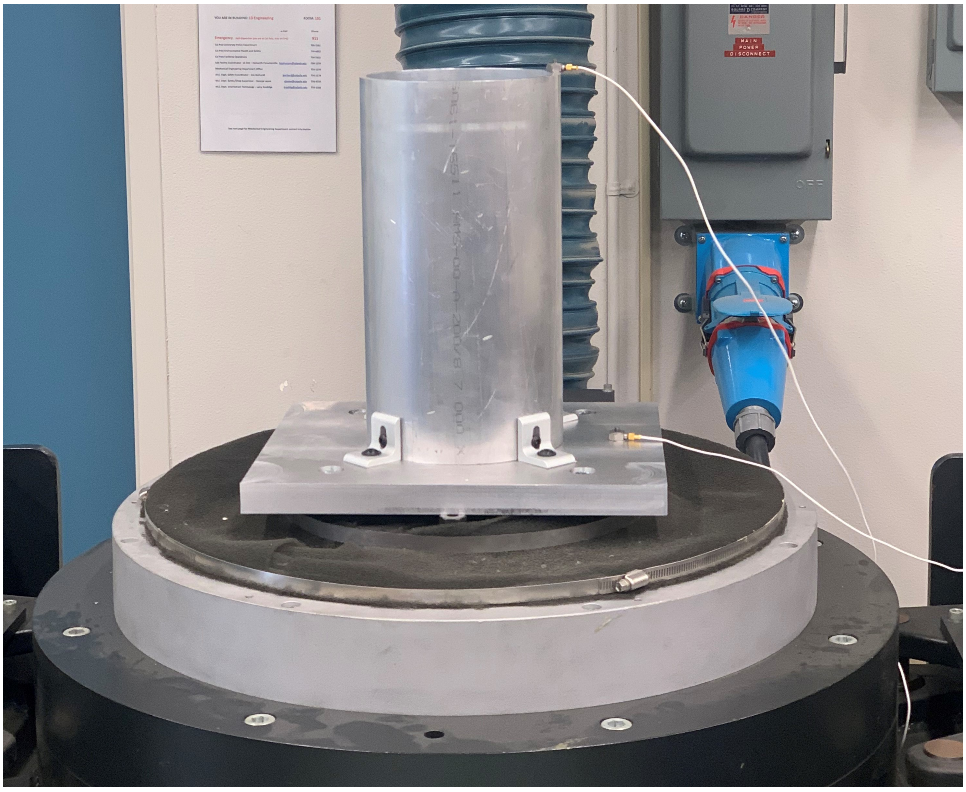


Figure 40: Setup for shaker table test of housing with accelerometers on plate and housing.

The following is a review of the experimental procedure, equipment and materials used in the testing of the passive magnetic damping mechanism. The purpose of this test was to confirm that magnetic eddy current damping exhibits viscous damping behavior by showing that the free vibration is characterized by exponential decay of the oscillation. In addition, the goal of this test was to verify the second specification for the natural frequency of the mass-spring system which needed to be below 2 Hz and the third specification we tested for was the damping ratio of the permanent magnet system as a function of gap distance. Figure 41 shows the testing apparatus for this experiment.

As shown in Figure 41, the test stand was constructed out of 2 inch by 4 inch wood studs and 1/2-inch plywood and had an overall height of approximately 1 m. The top plywood plate contained six tapped holes for which the six swivel stud anchors for the translational springs were threaded into. An additional six stud anchors, which were located on the bottom end of the springs, were threaded into the copper stage. The copper stage had a weight of approximately 3.88 kg and each translational spring had a spring constant of 29.8 N/m. The test apparatus was designed such that the copper stage hung about 10 cm above the middle plywood platform. This allowed for enough clearance to place the magnet holder below the stage. To adjust the distance between the magnet holder and the copper stage, 1/4-inch steel washers with a thickness of 1.5 mm were used as shown in Figure 41. Figure 42 shows a more detailed view of the magnet holder that was designed, and 3D printed to secure different types of magnets. Specifically, this mechanism supported nominal magnet diameters of either 1.0 inch or 1.5 inch. For both diameters of magnets, two thicknesses were tested (1/4 inch and 1/8 inch). Finally, both N52 and N42 grade magnets for each size were tested.

Table 6 shows the equipment list for this laboratory experiment and includes each type of magnet that was tested. Each magnet was a cylindrical magnet with axial magnetization. To capture data, a Siglent SDS 1104X-E digital storage oscilloscope was used which had a bandwidth of 100 MHz and a real-time sampling rate of 1 GSa/s along with a signal processor. The purpose of the signal processor was to amplify the analog signal from the accelerometer and to convert it to a digital signal which could be read by the oscilloscope. Finally, we used a PCB Sn 5657 single axis accelerometer with a sensitivity of 103 mV/g and a frequency range of 0.5 Hz to 13 kHz with a resonant frequency of 30 kHz.

To prepare for testing, the PCB single axis accelerometer was screwed into the copper stage by using a tapped hole on the top surface of the copper stage. Next, the cable from the accelerometer was screwed into the input for the signal processor box. The output from the signal processor was then plugged into the Siglent

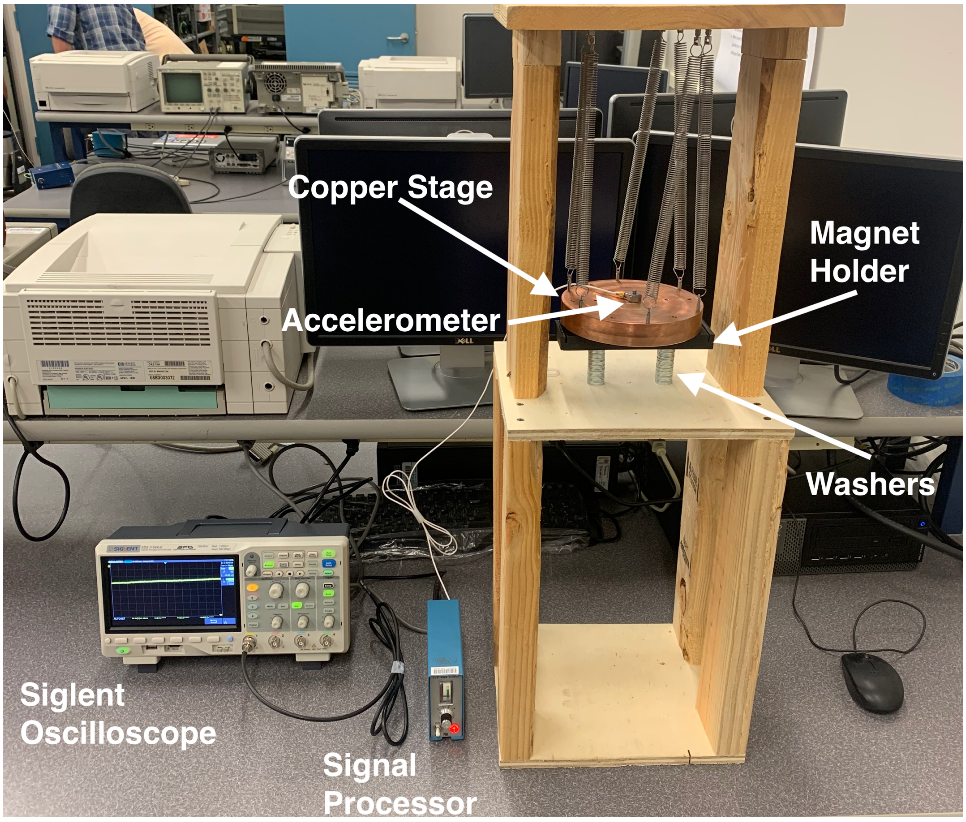


Figure 41: This figure shows the test apparatus for testing our magnetic damping mechanism. As shown in the figure, an SN 5657 single axis accelerometer was connected to the surface of the copper stage. We then connected the accelerometer to the signal processor which was then sent to the Siglent SDS 1104X-E oscilloscope.

SDS 1104X-E oscilloscope. Finally, a USB drive was plugged into the oscilloscope such that the experimental data could be directly imported to Excel after the appropriate data was collected. To being testing, the magnet holder was removed from below the copper stage. The stage was then displaced approximately 40 mm from its equilibrium position and then it was released and allowed to oscillate freely. Data from the free vibration of the stage was recorded for a span of approximately 10 seconds. The undamped natural frequency was recorded from the reading on the oscilloscope screen with its associated resolution uncertainty of $\pm$ 0.005 Hz. This procedure was then repeated an additional four times to make up a sample size of five trials for the purpose of calculating a statistical uncertainty.

The magnet holder was then installed using the two steel bolts with 25 washers (10 mm gap height) for each bolt to elevate the magnet holder approximately 40 mm above the plywood plate. At this height, each of the seven magnets shown in Table 6 were tested. Specifically, one N52 1.5 inch by 1/4-inch diameter magnet was placed in the center slot of the magnet holder and the stage was again displaced and the resulting damped vibration of the stage was recorded for approximately ten seconds. For each magnet, data from the oscilloscope was exported to Microsoft Excel and the log decrement method was used to calculate the damping ratio of the system. At this height, the entire process described for the N52 1.5 inch by 1/4-inch magnet was then repeated for the remaining magnet sizes as shown in Table 6. Finally, the same procedure was repeated for 30 washers (3 mm gap height) for all of the seven magnets.

Based on a mass of 3.88 kg for the copper stage and a spring stiffness of 29.8 N/m for each of the six springs, we calculated a theoretical natural frequency of 1.08 Hz. Experimentally, we recorded a natural frequency of 1.14 ± 0.01 Hz at 95% confidence for a sample size of five runs and a resolution uncertainty of $\pm$ 0.005 Hz. We conclude that one of the reasons why the experimental natural frequency does not agree with the theoretical natural frequency is that the real system is non-linear due to the angle of springs. Specifically, the nominal experimental value varies by 54% relative to the theoretical approximation. However, these values differ by less than 0.1 Hz. The primary purpose of recording this data was to understand how closely the real system could be approximated by a linear, SDOF system. From this analysis, we conclude that the SDOF model can be applied to our system as long as the amplitude of vibration is less than about 80 mm peak-to-peak

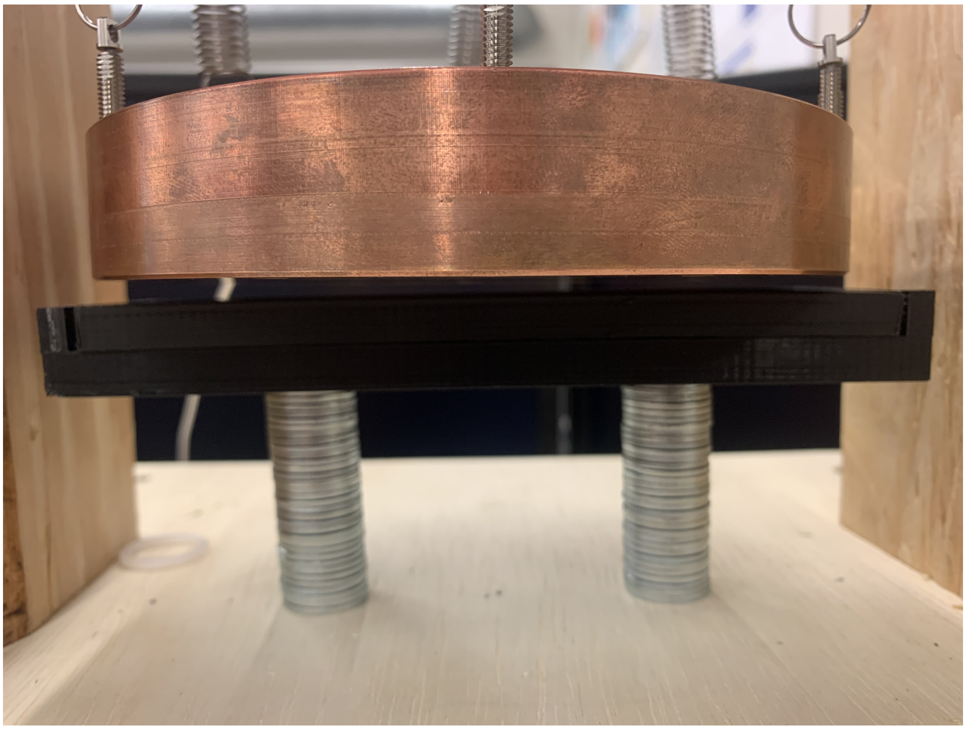


Figure 42: This figure shows a more detailed view of the magnet holder that was designed, and 3D printed to secure different types of magnets. Specifically, this mechanism supports nominal magnet diameters of either 1.0 inch or 1.5 inch. For both diameters of magnets, two thicknesses were tested (1/4 inch and 1/8 inch).

(amplitude of 40 mm).

Figure 43 shows a plot of the undamped response and damped response of the system for the free vibration of the stage. The damped oscillation of the stage was collected for the strongest magnet (N52 1.5 inch by 1/4 inch) at the smallest gap height (3 mm). In order to reduce the noise of the collected data for the damped response, a moving-average fiter was applied to the data in MATLAB in order to smooth out the curve. As shown in the figure, the damped natural frequency is smaller than the natural frequency of the system. This is expected because the damped frequency must always be lower than the natural frequency of the system. In addition, the plot shows that the amplitude of the vibration of the undamped system remains nearly constant for the duration of the data experiment as expected for no damping.

The next goal of testing the mass spring system was to understand how the use of magnets can be used to provide viscous damping. In order for damping to be considered viscous, the amplitude of vibration must decrease exponentially. Therefore, a smooth curve of the general form $V(t) = ae^{-bt}$ can be fit to the oscilloscope output data where V is in mV and t is in s. Figure **??** shows a plot of the time response of the free vibration of the system as a function of time, t [s]. As shown in the figure, an exponential curve was fit to data using data points for each peak of the oscillation. It is important to note that this data was collected for an N52 1.5 inch diameter by 1/4 inch thick magnet with a gap height of 3 mm. As shown in the plot, this exponential curve fit the data reasonably well with an R^2 value of 0.992. Therefore, we conclude that magnetic eddy current damping is indeed viscous damping due to the exponential decay of oscillation.

The final goal of this experiment was to calculate the damping ratio ζ for the seven different magnets that were used for the two gap heights of 25 and 30 washers (gap heights of 10 mm and 3 mm, respectively). To calculate the damping ratio for each magnet configuration, data from the oscilloscope was exported to a .csv file for a time domain of approximately 7 seconds with a data capture rate of 2000 data points per second. To calculate the damping ratio for each run, the log decrement method was used. Equation 2 shows the formula that was used for the log decrement method. As shown in the equation, x_1 represents the amplitude of the first peak and x_2 represents the amplitude of the second peak. It is important to note that ζ is a dimensionless parameter that indicates how damped a system is. If $\zeta = 0$, the system is said to be undamped and the system will oscillate at its natural frequency under free vibration. If $0 < \zeta < 1$ the system is said to be underdamped. Next, if $\zeta = 1$, the system is critically damped, and the response will not have any overshoot and will not oscillate. Finally, if $\zeta > 1$, the system is overdamped. For our system, we expect the response to

Table 6: Equipment & Magnet List

Test Setup & Magnets Equipment List			
Iten #	Description	Details	Quantity
1	Siglent SDS 1104X-E Digital Storage Oscilloscope	100 MHz Bandwidth	1
2	Signal Processor Box	Amplifier and Analog to Digital Converter	1
3	PCB Sn 5657 Accelerometer	Signle Axis Accelerometer	1
4	Wood Test Stand	Approximately 1 m Tall	1
5	Copper Stage	Mass of 3.88 kg	1
6	Translational Springs	29.8 N/m (Per Spring)	6
7	N52 (1.5"×1/4")	42.1 lbf Pull Strength	1
8	N52 (1.0"×1/4")	33.7 lbf Pull Strength	1
9	N52 (1.0"×1/8")	15.6 lbf Pull Strength	1
10	N42 (1.5"×1/4")	34.0 lbf Pull Strength	1
11	N42 (1.5"×1/8")	20.4 lbf Pull Strength	1
12	N42 (1.0"×1/4")	27.2 lbf Pull Strength	1
13	N42 (1.0"×1/8")	12.6 lbf Pull Strength	1

be significantly underdamped with ζ values less than approximately 0.3.

$$\zeta = \frac{\ln(x_1/x_2)}{\sqrt{(2\pi)^2 + \ln^2(x_1/x_2)}} \tag{2}$$

Table 7 summarizes the values of ζ for each magnet configuration at the two different gap distances. It is important to note that each value of ζ has a unique uncertainty because the uncertainty in the damping ratio was calculated by using propagated uncertainty methods with Equation 1. As shown in the table, as the damping ratio increases with a decrease in gap height, the percentage uncertainty decreases. This is expected because the associated uncertainties for the measurements used to calculate the damping ratio remain constant for each magnet and gap height as the nominal value of the damping ratio decreases. Therefore, the percentage uncertainty increases as the damping ratio decreases. In addition, the damping ratio is dependent on the various parameters of the magnets being testing including: magnet thickness, magnet diameter and magnet strength. Therefore, for the purpose of this paper, the primary goal was to compare each magnet independently at two different gap heights in order to better understand how significantly the damping ratio changes. As shown in the table, for the strongest magnet (N52 1.5 inch by 1/4 inch), the damping ratio was found to be 0.044 ± 3.45% and 0.093 ± 10.1% for gap heights of 10 mm and 3 mm, respectively.

Finally, based on research from Toan Le, it was determined that doubling the number of magnets from one magnet to two magnets approximately doubles the damping ratio [23]. With this testing complete, we finalized the design to use three 1.5 inch wide by 1/4 inch thick N52 magnets. This quantity, size and strength of magnet will give us the appropriate damping ratio to achieve an approximate settling time of 10 seconds. We also have a MATLAB analysis that corroborates our empirical results so that we have two means of justification in our final design decision for the permanent magnets.

Once we completed all the prior analysis and our finalized design was manufactured and assembled we mounted the entire assembly on the shaker table. We adjusted the swash plate such that there was a 12mm gap between the copper stage and permanent magnets. This gap height resulted in an approximate damping ratio of 0.2. To perform testing, we attached 3 accelerometers to the assembly. We attached one to the bracket used to mount the shake table to the housing. This accelerometer was used as a control, to set the target peak of the shake table. We placed another one on the top cap of the housing. Finally, we placed one in the center of the copper stage, where the microscope will sit. We set the test profile to sine sweep from 1 to 500 Hz. We then set the target peak to increase at a constant rate from 0.01 g at 1 Hz to 1.5 g at 10 Hz. From 10 Hz to

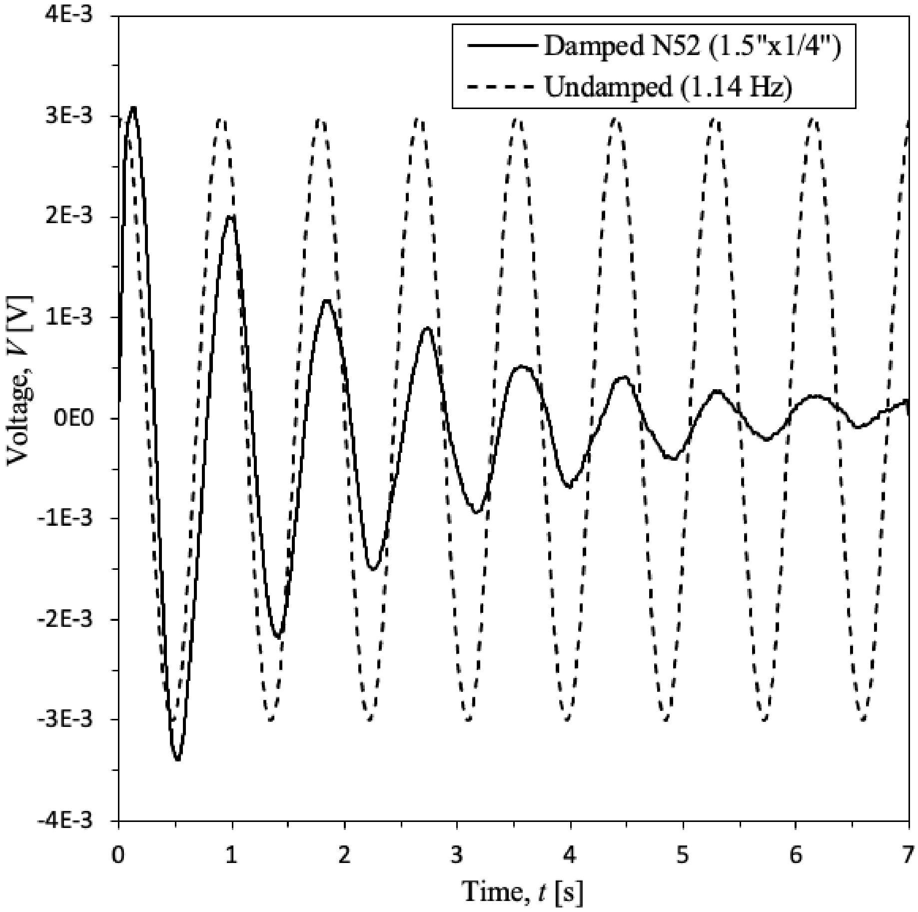


Figure 43: This figure shows a plot of the undamped response and damped response of the system for the free vibration of the stage. The damped oscillation of the stage was collected for the strongest magnet (N52 1.5 inch by 1/4 inch) at the smallest gap height (3 mm). As shown in the figure, the damped natural period is larger than the natural period of the system.

500 Hz, we kept the target peak constant at 1.5 g. Figure 45 shows the acceleration of all three accelerometers with respect to frequency with a sine sweep from 1 to 500Hz.

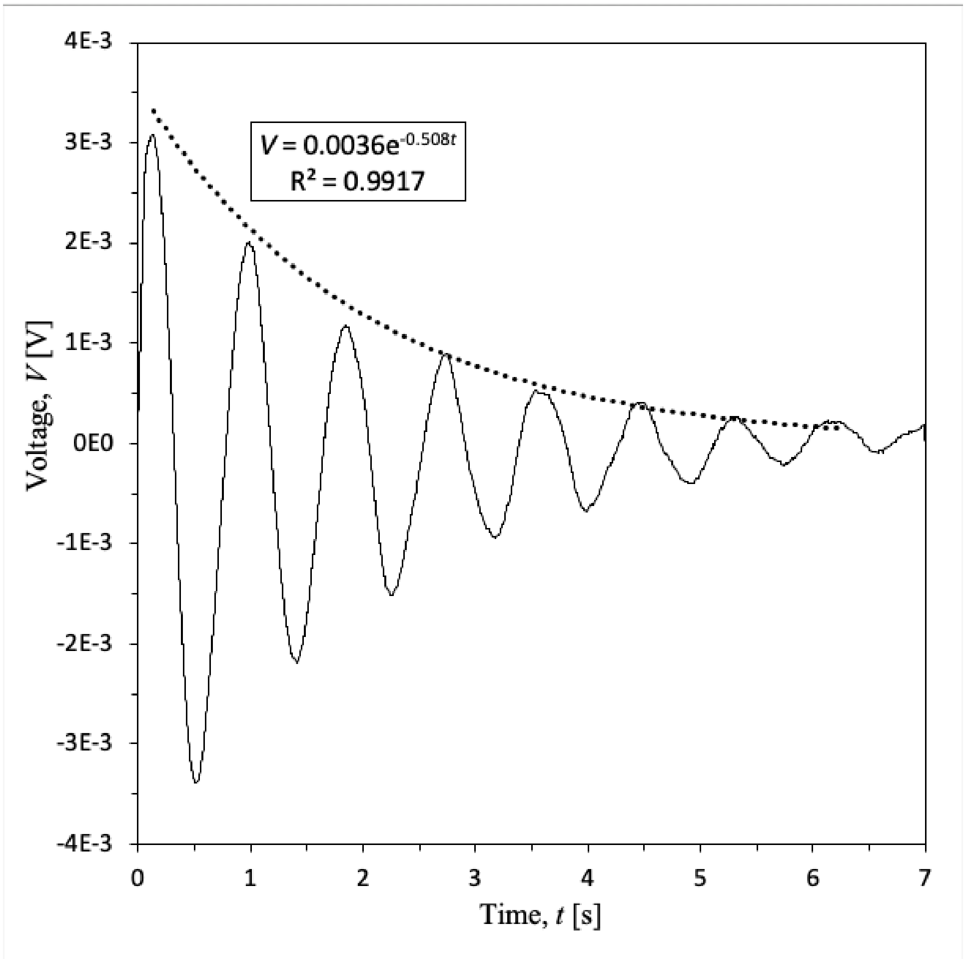


Figure 44: The plot shows data for an N52 1.5 inch diameter by 1/4 inch thick magnet with a gap height of 3 mm. An exponential curve fit was fit to data for the amplitude of each oscillation. As shown on the plot, this exponential curve fit the data reasonably well with an R^2 value of 0.992. Therefore, we conclude that magnetic eddy current damping is indeed viscous damping due to the exponential decay of oscillation.

Table 7: Experimental damping results for each magnet at gap heights of 3 mm and 10 mm.

MagnetType	Gap Height h [mm]	Damping Ratio ζ [ ]	Uncertainty U_ζ [± %]
N52 (1.5”×1/4”)	10	0.044	3.45
	3	0.093	10.1
N52 (1.0”×1/4”)	10	0.019	6.50
	3	0.039	5.82
N52 (1.0”×1/8”)	10	0.008	12.3
	3	0.031	9.56
N42 (1.5”×1/4”)	10	0.015	10.1
	3	0.090	1.23
N42 (1.5”×1/8”)	10	0.022	5.25
	3	0.042	6.28
N42 (1.0”×1/4”)	10	0.012	9.94
	3	0.024	11.0
N42 (1.0”×1/8”)	10	0.007	13.0
	3	0.016	10.3

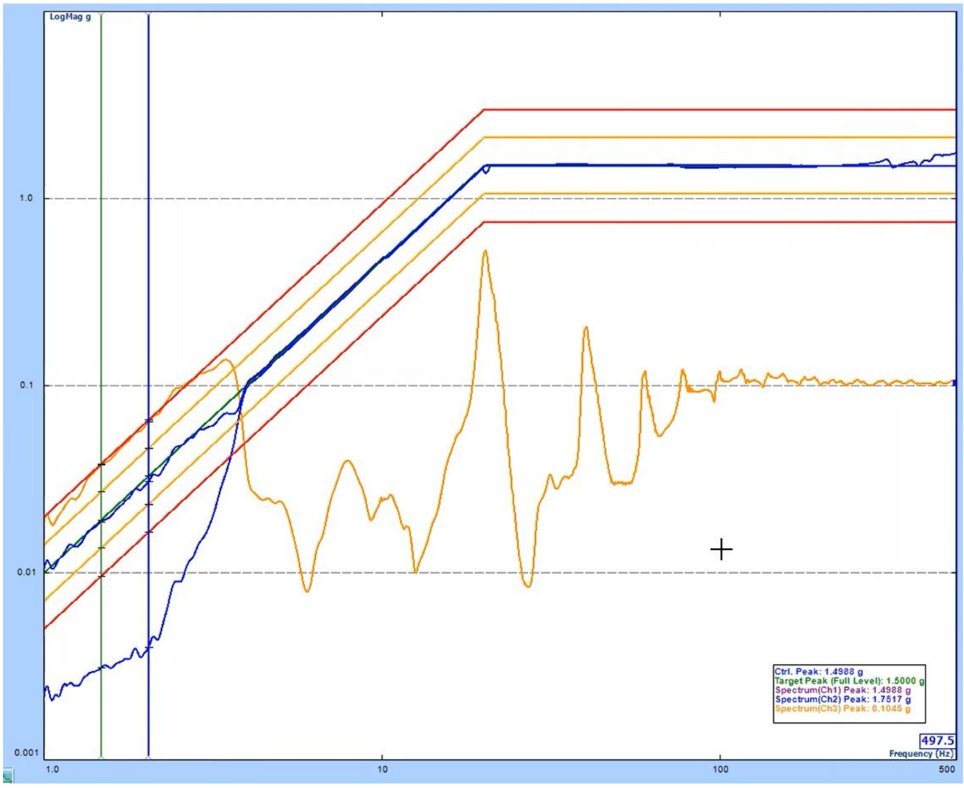


Figure 45: Acceleration vs frequency shown for the copper stage (orange), mounting bracket (blue) and top cap (blue) with 3 1.5” diameter, 1/4” thick N-52 magnets set to a gap height of 12mm. The accelerometer on the mounting bracket was used to set the target peak (solid green line), so these lines follow each other closely. The accelerometer of the top cap starts below the mounting bracket. The copper stage enters the isolation region at 2.82Hz.

We then took the raw data from our test and plotted the ratio of the stage acceleration to the ratio of the excitation acceleration with respect to frequency. Figure 46 shows our results. Our goal was to isolate vibrations from 1-100 Hz, and with the data shown you can see we were able to isolate from 2.82 Hz through 500 Hz. Therefor, we are confident that this system will isolate most of the ambient vibrations which will be encountered by the VIS in the chemistry lab. Furthermore, we made the system such that the damping ratio can be adjusted if the damping needs to be further tuned.

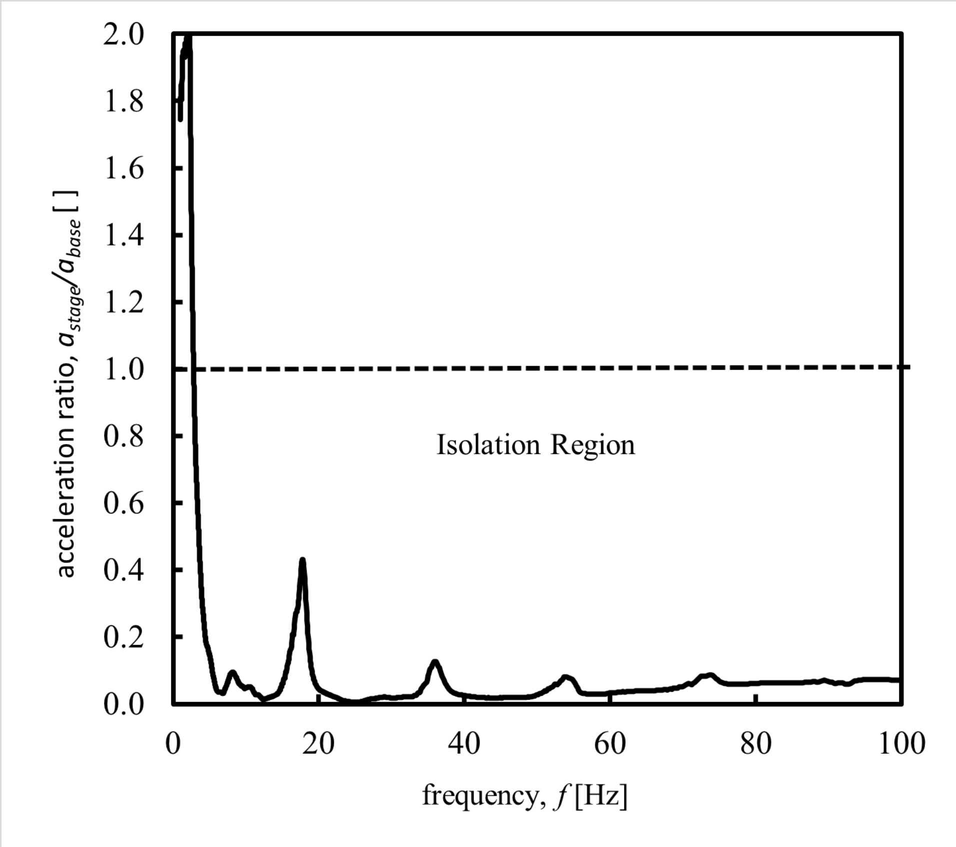


Figure 46: Ratio of the stage acceleration to the ratio of the excitation acceleration with respect to frequency from 1-100 Hz. At 2.82 Hz, the stage enters the isolation region. Within the isolation region, after 4.5 Hz, the largest measured ratio was .411 at 17.9 Hz. From 17.9 Hz to 100 Hz, the average ratio was measured to be 0.059.

Additionally, some specifications were confirmed via inspection of the final design. These include the integration with the vacuum chamber and the viewing areas of the VIS when it is mounted in the vacuum chamber. Our complete design verification plan can be found in Appendix Q.

We were not able to complete our planned user testing for sample swap time and ease of use due to time constraints and laboratory restrictions due to COVID-19. However, due to the research applications of our VIS, we do not anticipate these factors being nearly as critical as the system's ability to adequately damp vibrations. Additionally, we were not able to test the ability to take an atomic level image with the STM as planned because the STM itself was not ready for testing this quarter. While we would have like to see the results of this test, we are confident because of the results of our other testing that our product will perfom to the required specifications.

8 Project Management

To ensure that we solved the correct problem and met the customer's requirements, we went through a rigorous design process which will consist of several key components. Managing this design process was a crucial step to help us avoid building a prototype that is not wanted or does not meet the appropriate needs.

To begin this process, we conducted research about that problem that we were trying to solve. In addition, we met with our sponsor to better understand the customer needs. Using this information along with our QFD, we worked to define this project scope and statement of work document for our sponsors to help define the specifics of what we are going to do. From this point, we began the ideation phase of the project to brainstorm as many potential solutions as possible. Specifically, we brainstormed designs for the main structure as well as the overall layout of the entire prototype including the location of the permanent magnets and the integration with the vacuum chamber and microscope. This allowed us to compare a multitude of designs and utilize decision matrices to help us to select the most feasible option. In addition, we constructed simple prototypes of our top designs using low cost material to aid in the overall selection process. We then presented our findings as well as our preliminary analysis results to our sponsor in a Preliminary Design Review (PDR). After obtaining sponsor approval and peer feedback, we then began to turn our selected concept into a final design.

Moving forward towards our Critical Design Review (CDR), we performed a rigorous analysis process to optimize and determine the effectiveness of our design. Specifically, we analyzed the mass-spring system using MATLAB to determine the response of the copper stage due to input vibrations from the housing. Our goal from this analysis was to optimize the mass-spring system in order to minimize the amplitude of the response of the copper stage, and we did this by analyzing the response for varying system parameters including the stiffness of the springs, how many springs were used, how the springs were fixed to the aluminum housing (angle), and the weight of the copper stage. In addition, we performed additional analysis on the aluminum housing using Abaqus Finite Element Analysis (FEA) software in order to optimize the natural frequency of the housing. One of the challenges with this analysis was determining which modes of vibrations were of the biggest concern, as the housing is a multi-degree of freedom system. Our final analysis was concerned with the magnetic eddy current damping. Specifically, we wanted to optimize the damping ratio by inspecting the transmissiblity plot. The challenge with this analysis was developing theoretical relationships for how the damping ratio varies with the location of the permanent magnets relative to the copper stage. We used Abaqus electromagnetic analysis to simulate the damping ratio. In addition, we used MATLAB to develop a theoretical model which we could compare to the Abaqus simulation results as well as real test results. While imperftect, this analysis gave us confidence in our design as we moved into the building and testing phases.

The next phase of the project was to construct a prototype to test and evaluate the effectiveness of our design. We finalized our design for the housing and began to manufacture it concurrently with additional testing of the magnetic damping system to finalize our selection of magnets. The manufacturing process took us around two months due to shop accessibility during COVID-19 and our relative inexperience at manufacturing, and this required us to prioritize our most critical tests and leave out some of the minor ones.

Throughout this process, we relied on sponsor feedback as often as possible. We presented our findings and conclusions to our sponsor in a Final Design Review (FDR), as well as in the online Senior Project Expo. Table 4.1 outlines the major deliverables as well as the approximate deadline for each task.

Table 8: Major senior project deliverables with associated deadlines

Deliverable	Description	Deadline
Scope of Work (SOW)	Contract between sponsors and team for what will be completed.	10/13/20
Preliminary Design Review (PDR)	First review of our design choices and opportunity for feedback.	11/10/20
Critical Design Review (CDR)	Detailed review of our design and analysis results. Will include cost estimate and updates from our PDR.	2/6/21
Initial Test Plan	Complete plan for testing as well as outlining our potential safety hazards.	2/27/21
Manufacturing Test and Review	Complete plan for manufacturing the prototype including safety. Will also status of component manufacturing.	3/13/21
Operators Manual	Detailed operators' manual explaining safety concerns, how to use the product, and trouble shooting.	5/22/21
Final Design Review	Presentation of final prototype and how it meets our customer requirements.	5/31/21
Wrap up Paperwork	Required paperwork for handing off the final design and prototype to our sponsor	6/1/21

To help manage this year long process, our team created a Gantt Chart which is a display of the timeline for the year. We continually updated this document as our progress level changed and as new issues or added tasks arose. A full version of our Gantt chart is included in Appendix S.

9 Conclusion

Working together over the course of a year, we designed, manufactured, and tested a vibration isolation system based on the design and analysis from Toan Le's master's thesis. Our product was able to isolate vibrations from 2.8 to well over 100 Hz, and is integrated with a tunable magnetic eddy current damping system to match the damping with the environment in which our system is used. We also experimentally explored the effect of different magnets and gap heights on the damping ratio obtained, and while our theoretical model was not able to accurately predict the experimental results, we gained valuable insight into the physical relationships involved.

We were unable to model and test the effect of varying the grade, thickness, and compression of Viton on the overall damping and natural frequency of the system, and we believe this would be a valuable area for future research. We were also unable to test our system in the Chemistry lab and in the vacuum chamber because we were ultimately unable to obtain lab access due to restrictions resulting from the COVID-19 pandemic. Additionally, we could not obtain an image using our VIS because the STM was not ready for testing by the time of completion of this project. However, our product did meet our design specifications, which would suggest it should significantly improve STM imaging. Our VIS should be thoroughly cleaned to remove any residues each time it goes into the vacuum chamber. Additionally, we will be providing our sponsor with silver-plated versions of the screws after the completion of this project. Finally, we would suggest further research and testing into the damping effects of the Viton and permanent magnets.

10 References

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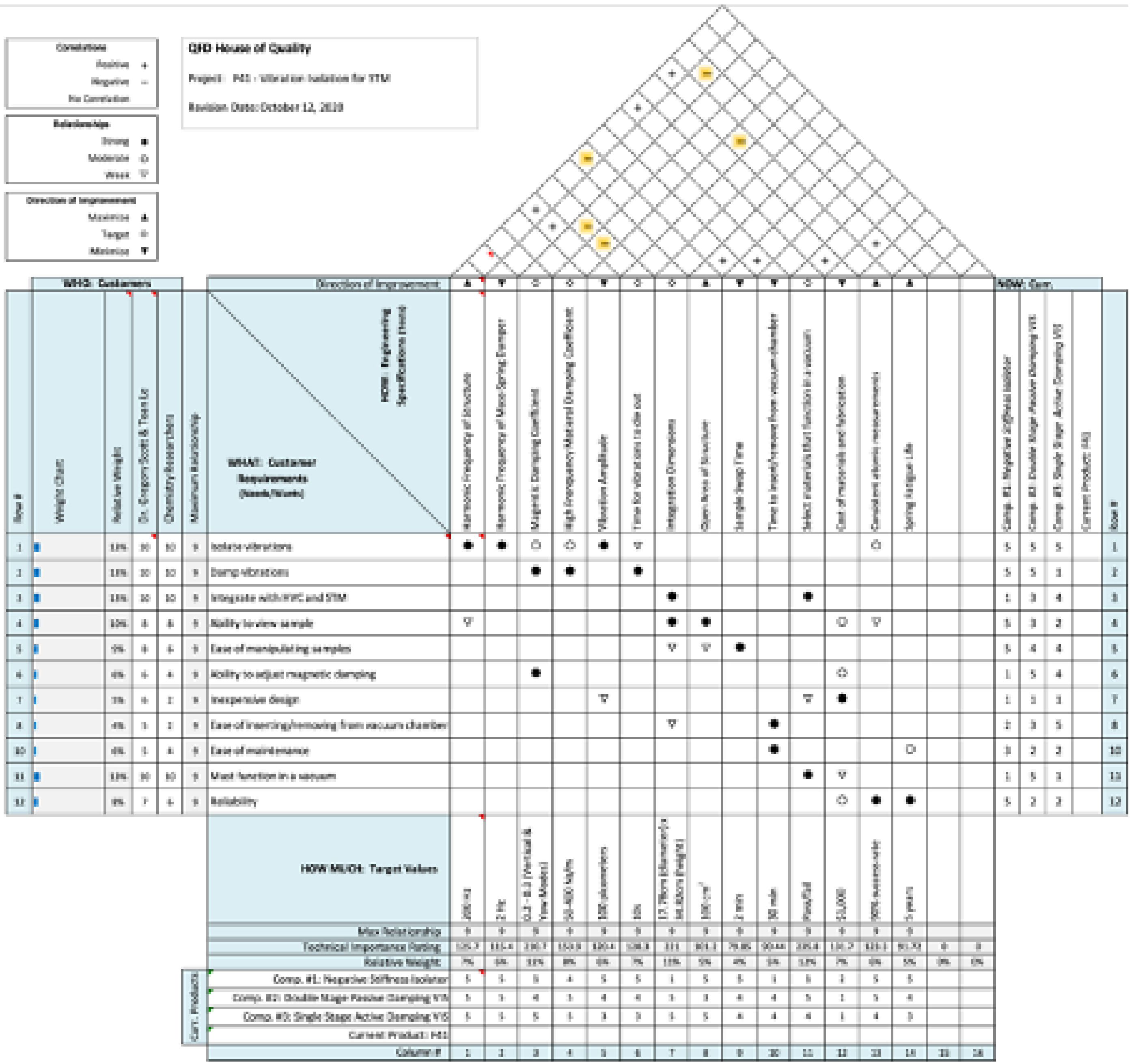
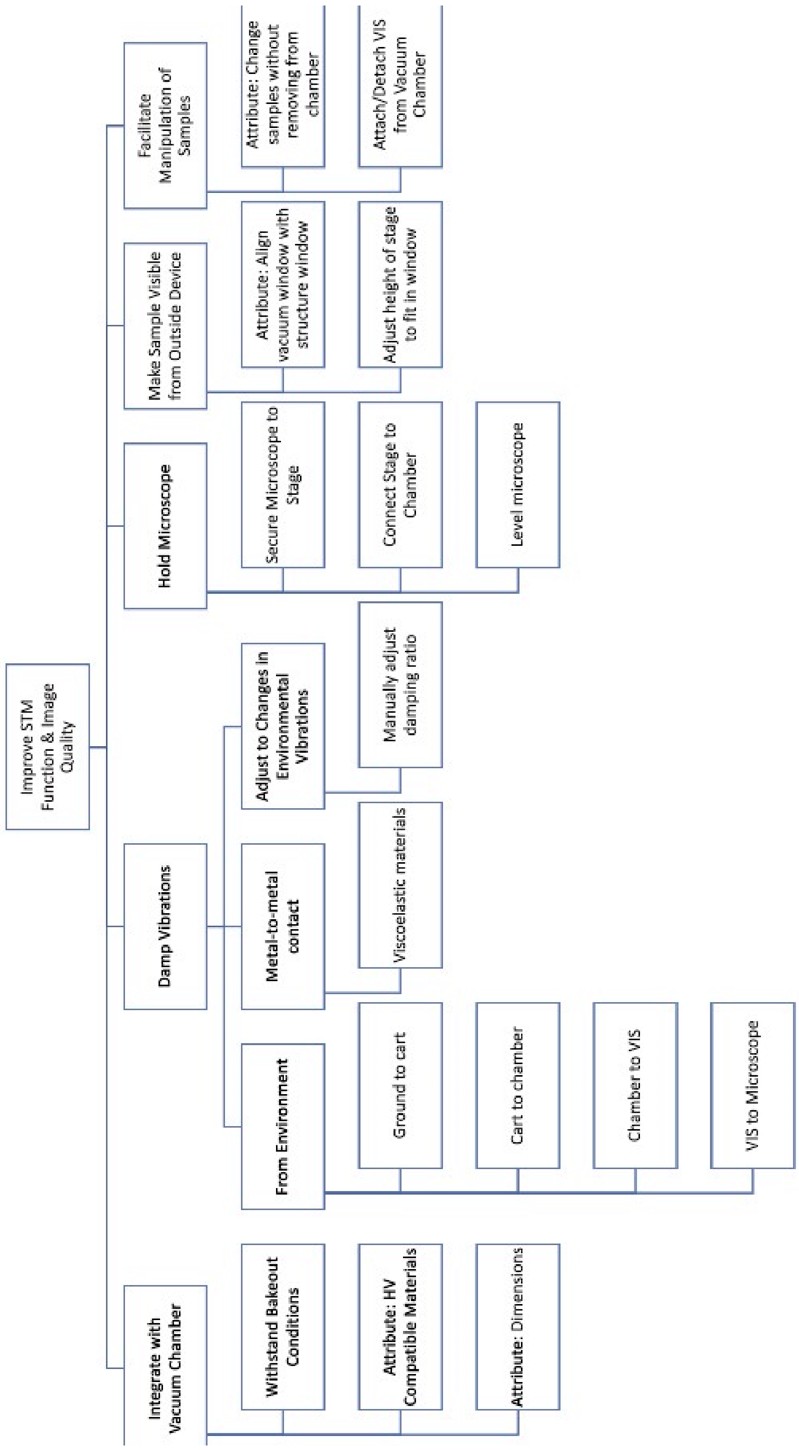
Appendices

A Quality Function Deployment (QFD): House of Quality

The following "House of Quality" chart outlines our design of engineering specifications to meet the customer's requirements. On the left in the "Who" column, each customer or group of customers is named, along with the requirements weighted by importance for each customer's use case. In the "What" column, the specific requirements are listed. The row at the top labeled "How" generates engineering specifications, which are correlated with the relevant customer requirements in the middle. In the upper triangle, the effect of each specification on the others is represented with a (+) sign if meeting one specification is beneficial in meeting the other and a (-) sign if meeting one specification is detrimental to the other. Finally, specifications are quantified in the "How Much" section at the bottom.


QFD House of Quality
Project: P41 - Vibration Isolation for STM
Revision Date: October 12, 2020
Correlations
Positive +
Negative −
No Correlation
Relationships
Strong
Moderate
Weak
Direction of Improvement
Maximize
Target
Minimize
WHO: Customers
Row #
Weight Chart
Relative Weight
Dr. Gregory Scott & Team Lu
Chemistry Researchers
Maximum Relationship
WHAT: Customer Requirements (Needs/Wants)
HOW: Engineering Specifications (Hows)
Direction of Improvement
Harmonic Frequency of Structure
Harmonic Frequency of Mass-Spring Damper
Magnetic Damping Coefficient
High Frequency Material Damping Coefficient
Vibration Amplitude
Time for vibrations to die out
Integration Dimensions
Open Area of Structure
Sample Swap Time
Time to Insert/remove from vacuum chamber
Select materials that function in a vacuum
Cost of materials and fabrication
Consistent atomic measurements
Spring Fatigue Life
NOW: Com.
Comp. #1: Negative Stiffness Isolator
Comp. #2: Double Stage Passive Damping VIS
Comp. #3: Single Stage Active Damping VIS
Current Product: P41
Isolate vibrations
Damp vibrations
Integrate with HVC and STM
Ability to view sample
Ease of manipulating samples
Ability to adjust magnetic damping
Inexpensive design
Ease of inserting/removing from vacuum chamber
Ease of maintenance
Must function in a vacuum
Reliability
HOW MUCH: Target Values
200 Hz
2 Hz
0.2 - 0.3 (Vertical & Yaw Modes)
50-400 Ns/m
100 picometers
10s
17.78cm (diameter) x 34.80cm (height)
100 cm²
2 min
30 min
Pass/Fail
$1,000
90% success rate
5 years
Max Relationship
Technical Importance Rating
Relative Weight
Curr. Products
Column #

B Functional Decomposition



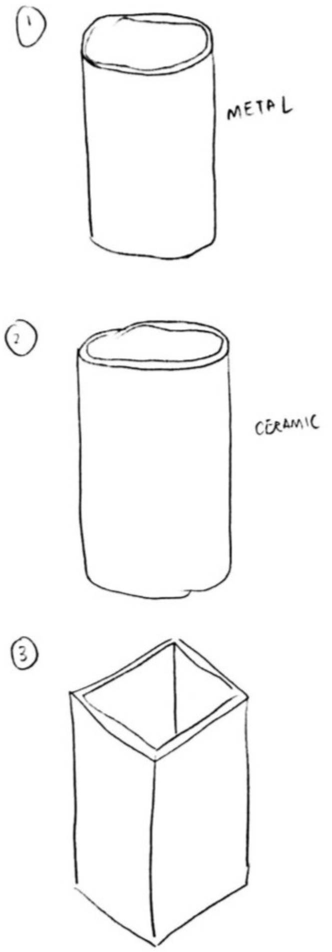
C Ideation and Model Building

C.1 List of Concept Ideas by Function

1. Integrate with Vacuum Chamber
   - Solid cylindrical housing
   - Cross-hatched housing
   - Vertical bars
   - Attach springs to top of vacuum chamber
   - Steel housing
   - Titanium Housing
   - Aluminum housing
   - Ceramic housing
2. Damp Vibrations
   - Original cart with viton added
   - Place vacuum chamber on air springs
   - Negative stiffness mechanism
   - Tripod table
   - Remove original cart wheels and place on viton
   - Magnetic eddy-current damping on stage
   - Elastomeric diaphragm to damp stage vibrations
   - Stage on piston with viscous damping
3. Hold Microscope
   - Microscope rests on stage
   - Press fit attachment
   - Design fixture to hold microscope and attach to stage
   - Interposer plate with viton gasket
   - Adjustable arm
   - Fluid layer
4. Make Sample Visible from Outside Device
   - Window in solid housing
   - Align gaps in bars or cross-hatched housing with vacuum chamber window
   - Mirrors inside housing
   - Change cap height with leveling screws
   - Multiple spring attachment points
5. Facilitate Manipulation of Samples
   - Housing open at top
   - User can reach through window to manipulate samples
   - Design tool that allows manipulation of samples
   - Mechanism to raise stage to window so microscope can be accessed.

D Pugh Matrices

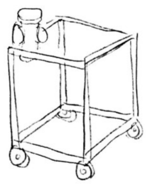
D.1 Function: Integrate with Vacuum Chamber



CONCEPT / CRITERIA	1	2	3
ISOLATE VIBRATIONS	+	+	S
INTEGRATE W/ HVC/STM	+	+	+
ABILITY TO VIEW SAMPLE	+	+	−
MUST FUNCTION IN VACUUM	+	+	+
WITHSTAND BAKEOUT	+	+	+
TOTAL	5	5	2

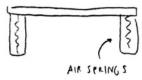
D.2 Function: Damp Vibrations From Environment

1



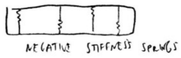
THE VACUUM CHAMBER ASSEMBLY IS CURRENTLY MOUNTED ON A CART, THIS IS OUR BASELINE AND CAN DEFINITELY BE IMPROVED UPON. THE WHEELS AND STIFF FRAME OF THE CART HAVE UNKNOWN EFFECTS ON VIBRATION ISOLATION.

2



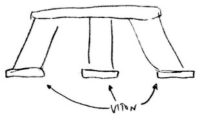
THE VACUUM COULD BE LOADED ON A TABLE ON AIRSPRINGS. THIS WOULD BE VERY EFFECTIVE AT ISOLATING THE VACUUM ASSEMBLY FROM VIBRATIONS.

3

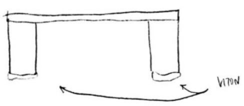


THE VACUUM COULD BE LOADED ON A TABLE ON NEGATIVE STIFFNESS SPRINGS. THIS HAS BEEN PROVEN TO BE VERY EFFECTIVE, BUT ALSO EXPENSIVE.

4



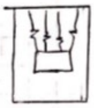
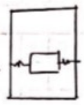
5



THE VACUUM COULD SIT ON A TABLE THAT IS LOADED ON VITON WHICH COULD BE EXTREMELY EFFECTIVE AT DAMPING VIBRATIONS.

CRITERIA \ CONCEPT	1	2	3	4	5
ISOLATE VIBRATIONS	−	+	+	S	S
DAMP VIBRATIONS	−	S	+	+	+
ABILITY TO ADJUST DAMPING	S	S	S	+	+
INEXPENSIVE DESIGN	+	−	−	S	S
EASE OF MAINTENANCE	+	−	−	+	+
RELIABILITY	−	−	−	S	S
TOTAL	-1	-2	-1	3	3

D.3 Function: Isolate from Environmental Vibrations

CONCEPT / CRITERIA	 1. TOAN'S DESIGN	 2. SINGLE SPRING RESTING	 3. SIX SPRINGS RESTING	 4. SIX SPRINGS HANGING	 5. SPRINGS AROUND
1. HARMONIC FREQUENCY	+	D	−	−	+
2. VIBRATION AMPLITUDE	+		+	+	
3. TIME TO DIE OUT	−	A	−	−	+
4. PRODUCTION COST	+	T	−	+	−
5. ADJUSTABLE HEIGHT	+			+	−
Σ+	4	U	1	3	2
Σ−	1		3	2	2
ΣS	3	M	-2	1	0

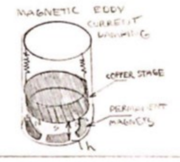
TOP IDEA SKETCH

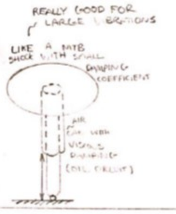
DISCUSSION:

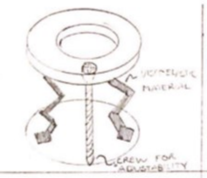
THE TOP IDEA IS THE HANGING MASS WITH ANGLED SPRINGS. THE HANGING MASS HAS BETTER HEIGHT ADJUSTABILITY THAN THE OTHER DESIGNS. IN ADDITION, SELECTING THIS DESIGN WILL LOWER COST BECAUSE THE SPRINGS HAVE ALREADY BEEN PURCHASED. WE BELIEVE THE ANGLED SPRINGS WILL ALLOW US TO ACHIEVE A LOWER STIFFNESS WHILE STILL SUPPORTING THE MASS. FURTHER FEA ANALYSIS IS NEEDED TO CONFIRM THE ASSUMPTIONS USED IN THIS MATRIX.

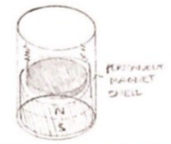
D.4 Function: Damp Stage Vibrations

PUGH MATRIX — DAMP VIBRATIONS



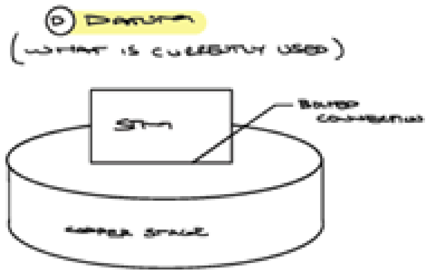


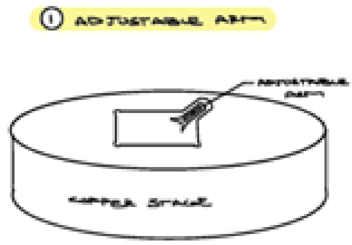


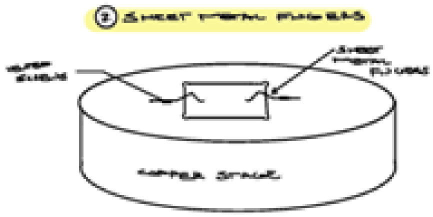


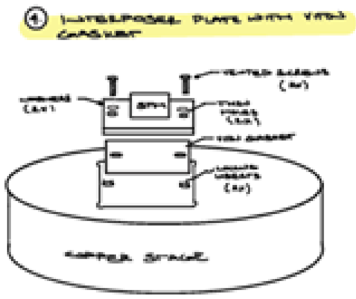
EFFECTIVENESS AT DAMPING VIBRATIONS IN VERTICAL DIRECTION	S	−	+	−
EFFECTIVENESS AT DAMPING VIBRATIONS IN OTHER DIRECTIONS.	S	−	S	+
COMPLEXITY	S	−	−	S
COST	S	−	S	−
MANUFACTURABILITY	S	−	−	−
ADJUSTABILITY OF DAMPING RATIO	S	+	+	S
DESCRIPTION:	THIS DESIGN USE PERMANENT MAGNETS ON THE BOTTOM CAP FOR EDDY CURRENT DAMPING. THE ARRANGEMENT OF THE MAGNETS DETERMINES HOW EFFECTIVELY THE VIBRATIONS WILL BE DAMPED OUT. IN ADDITION, THE DISTANCE BETWEEN THE MAGNETS AND THE COPPER STAGE CHANGES THE DAMPING RATIO. THEREFORE, WE ARE STILL DEVELOPING THE BEST METHOD TO ADJUST THE DAMPING RATIO.	THIS DESIGN FEATURES VISCOUS DAMPING LIKE THE SHOCKS THAT ARE USED ON HIGH END MOUNTAIN BIKES. THIS TYPE OF DAMPING WORKS REALLY WELL FOR LARGE IMPACTS. THEREFORE, THIS IS NOT THE BEST CHOICE FOR THIS APPLICATION. IN ADDITION, THE INTERNAL OIL CIRCUIT IS QUITE COMPLEX AND EXPENSIVE. HOWEVER, THIS ALLOWS FOR EASY ADJUSTABILITY.	THIS DESIGN USES VISCOELASTIC MATERIAL TO DAMP OUT VIBRATIONS. HOWEVER, WE ARE UNSURE OF HOW EFFECTIVE THIS METHOD OF DAMPING IS WITHOUT FURTHER TESTING. ONE BENEFIT OF THIS DESIGN IS THAT IT ALLOWS FOR QUICK AND EASY ADJUSTMENT THROUGH THE USE OF THE SCREW.	THIS DESIGN FEATURES A MAGNETIC SHELL WHICH SURROUNDS THE COPPER STAGE. THIS APPROACH WORKS VERY WELL FOR DAMPING OUT ROTATIONAL VIBRATIONS. HOWEVER, IT DOESN'T DAMP OUT VIBRATIONS VERY WELL IN THE VERTICAL DIRECTION. IN ADDITION, THIS DESIGN DOES NOT ALLOW FOR EASY ADJUSTING OF THE DAMPING RATIO.

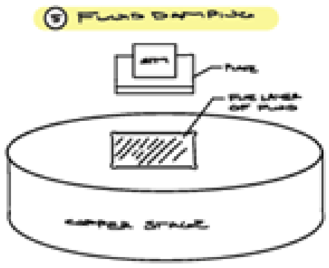
D.5 Function: Hold Microscope









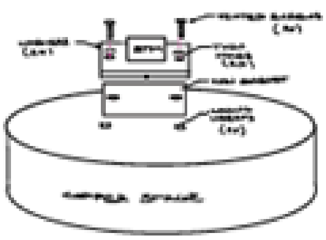


PUGH MATRIX:

criteria \ concept	D	1	2	3	4	5
isolate vibration	D	S	S	S	+	+
ease of manipulating samples	A	S	+	−	S	−
ergonomics	T	−	S	+	S	−
vacuum compatible	U	S	S	S	S	−
reliable	M	S	−	−	S	−
ease to view samples	S	S	−	S	S	S
ease to machine	S	−	S	S	S	S
adjustable	S	+	+	+	+	−
total	S	−1	0	0	+2 *	−5

FINAL DESIGN:

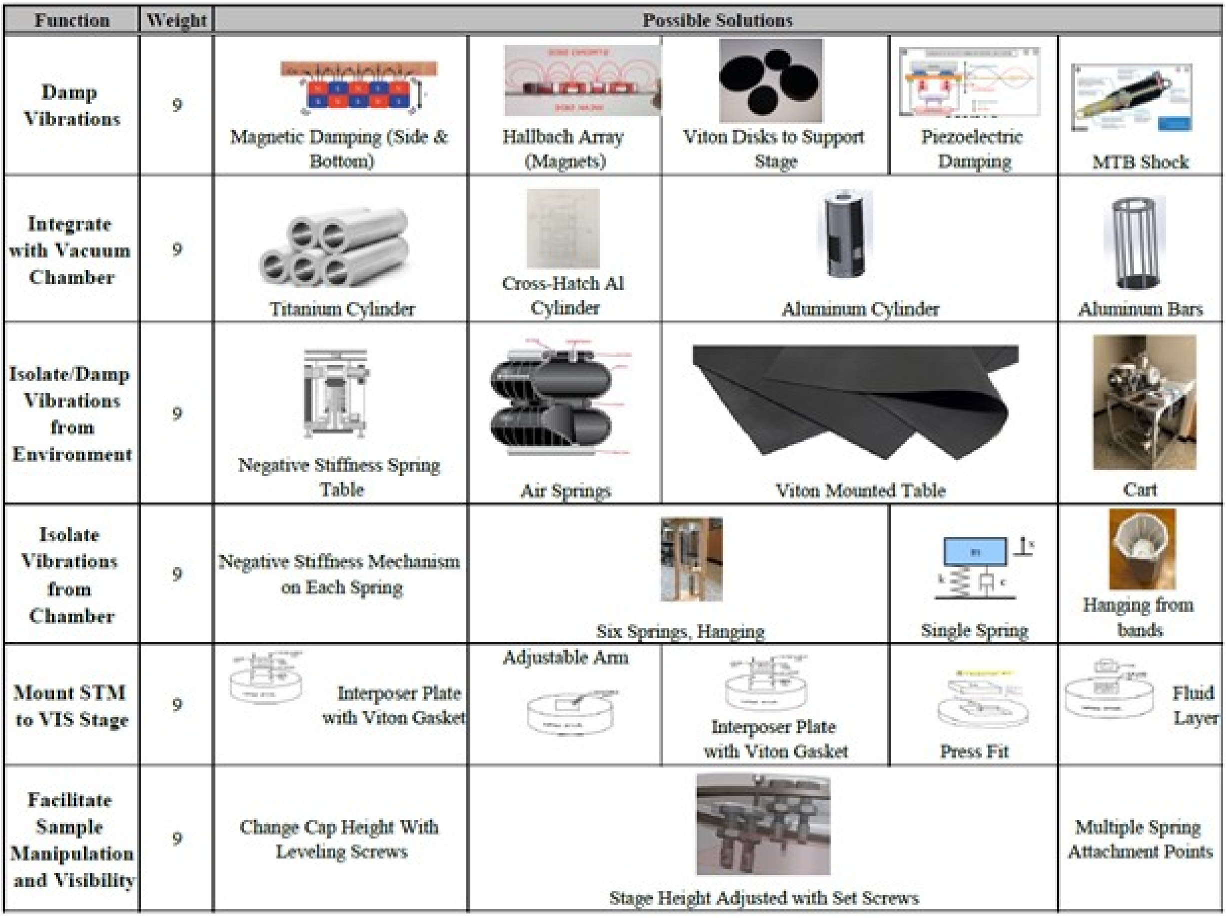
④ INTERFACE PLATE WITH VITON GASKET



DESCRIPTION:

THE VITON GASKET CAN ENABLE THE LEVEL OF THE INITIAL INTERFACE PLATE TO BE ADJUSTED BY TURNING THE TEFLON SCREWS. IN ADDITION, I REMOVED THE STM IMPRESSION ON THE COPPER STAGE. IF A NEW STM IS DESIGNED FOR USE, A NEW INTERFACE PLATE CAN BE MACHINED.

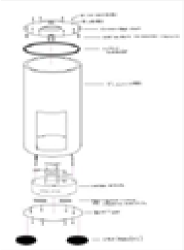
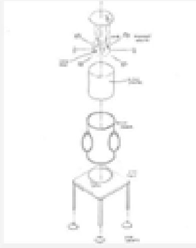
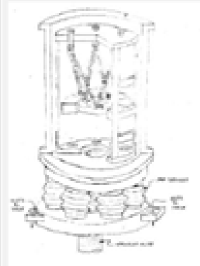
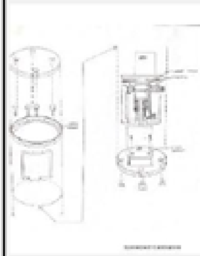
E Morphological Matrix



Function	Weight	Possible Solutions				
Damp Vibrations	9	Magnetic Damping (Side & Bottom)	Hallbach Array (Magnets)	Viton Disks to Support Stage	Piezoelectric Damping	MTB Shock
Integrate with Vacuum Chamber	9	Titanium Cylinder	Cross-Hatch Al Cylinder	Aluminum Cylinder		Aluminum Bars
Isolate/Damp Vibrations from Environment	9	Negative Stiffness Spring Table	Air Springs	Viton Mounted Table		Cart
Isolate Vibrations from Chamber	9	Negative Stiffness Mechanism on Each Spring	Six Springs, Hanging		Single Spring	Hanging from bands
Mount STM to VIS Stage	9	Interposer Plate with Viton Gasket	Adjustable Arm	Interposer Plate with Viton Gasket	Press Fit	Fluid Layer
Facilitate Sample Manipulation and Visibility	9	Change Cap Height With Leveling Screws	Stage Height Adjusted with Set Screws			Multiple Spring Attachment Points

F Weighted Decision Matrix

Design Ideas

		Design 1		Design 2		Design 3		Design 4	
									
		Aluminum housing with cutouts for sample viewing. The single stage VIS also consists of a copper cylinder hung by springs. The copper stage has an interposer plate with a viton pad used between the STM and copper stage.		Magnetic damping, copper stage hanging from soft springs, circular aluminum housing. Vacuum chamber mounted on table with viton supports. Microscope press fit into stage.		Large circular magnetic damping of stage, with spring-mass system of hanging stage. Entire structure is mounted on airsprings for additional vibration damping. Between the top and bottom cap are viton gaskets.		Negative stiffness spring system underneath copper stage. Permanent magnets located under copper stage on adjustable platform. The copper stage has an interposer plate with a viton pad used between the STM and copper stage.	
Constraint	Y/N	Y		Y		Y		Y	
Integration Dimensions	Y/N	Y		Y		Y		Y	
Vacuum Compatible Materials	Y/N	Y		Y		Y		Y	
Criteria	Weight	Score	W. Score	Score	W. Score	Score	W. Score	Score	W. Score
Housing Natural Frequency	5	8	40	8	40	8	40	9	45
Mass-Spring System Nat. Frequency	5	7	35	7	35	7	35	8	40
Magnetic Damping Coefficient	4	6	24	6	24	6	24	6	24
Elastomeric Material Damping	3	7	21	5	15	5	15	6	18
Vibration Amplitude	5	7	35	7	35	7	35	9	45
Time for Vibrations to Die Out	5	8	40	7	35	7	35	7	35
Viewing Area in Housing	4	9	36	4	16	9	36	5	20
Sample Swap Time	2	5	10	5	10	5	10	4	8
VIS Swap Time from Vacuum Chamber	2	7	14	7	14	7	14	4	8
Cost	4	7	28	6	24	4	16	3	12
Consistent Atomic Measurements	5	7	35	7	35	7	35	7	35
		Total =	318	Total =	283	Total =	295	Total =	290
		**Top Score							

G Indented Bill of Materials

F41 - Vibration Isolation System

Indented Bill of Material (iBOM)

Assembly Level	Part Number	Description (Lvl0 / Lvl1 / Lvl2 / Lvl3 / Lvl4)	Qty	Cost	Ttl Cost	Source	More Info
0	000	Housing Assembly				------	------
1	001	— Aluminum Cylinder	1	0.00	0.00	custom	sponsor provided
1	002	— Viton Ring	2	9.73	19.46	McMaster	P/N 9464K559
1	003	— #4-40 Vented Screw	16	2.11	33.76	McMaster	P/N 93235A110
1	100	— Mass-Spring Assembly				------	------
2	101	—— Top Cap	1	29.71	29.71	custom	machined aluminum
2	102	—— Stainless Steel Springs	6	0.00	0.00	McMaster	sponsor provided
2	103	—— Copper Stage	1	0.00	0.00	custom	sponsor provided
2	104	—— STM Fixture	1	0.00	0.00	custom	sponsor provided
2	105	—— Viton Sheet	1	5.60	5.60	Viton.com	P/N 1084N91
2	106	—— Long Stud Anchor	6	0.00	0.00	McMaster	sponsor provided
2	107	Short Stud Anchor	6	0.00	0.00	McMaster	sponsor provided
2	108	—— Thin Hex Nut (1/4-20)	12	0.04	0.48	McMaster	P/N 91847A029
1	200	— Magnetic Adjustment Assembly				------	------
2	201	—— Bottom Cap	1	29.71	29.71	custom	machined aluminum
2	202	—— Permanent Magnets	8	12.51	100.08	K&J Magnetics	P/N DX84
2	203	—— Magnet Swash Plate	1	16.78	16.78	McMaster	P/N 1610T56; make from
2	204	—— ULP Socket Screw (1/4-20) 5/8"	3	4.19	12.57	McMaster	P/N 91223A334
2	205	—— ULP Socket Screw (1/4-20) 7/8	3	5.12	15.36	McMaster	P/N 91223A335
2	206	—— Thin Hex Nut (1/4-20)	3	0.04	0.12	McMaster	P/N 91847A029
	Total Parts		**72**		**263.63**		

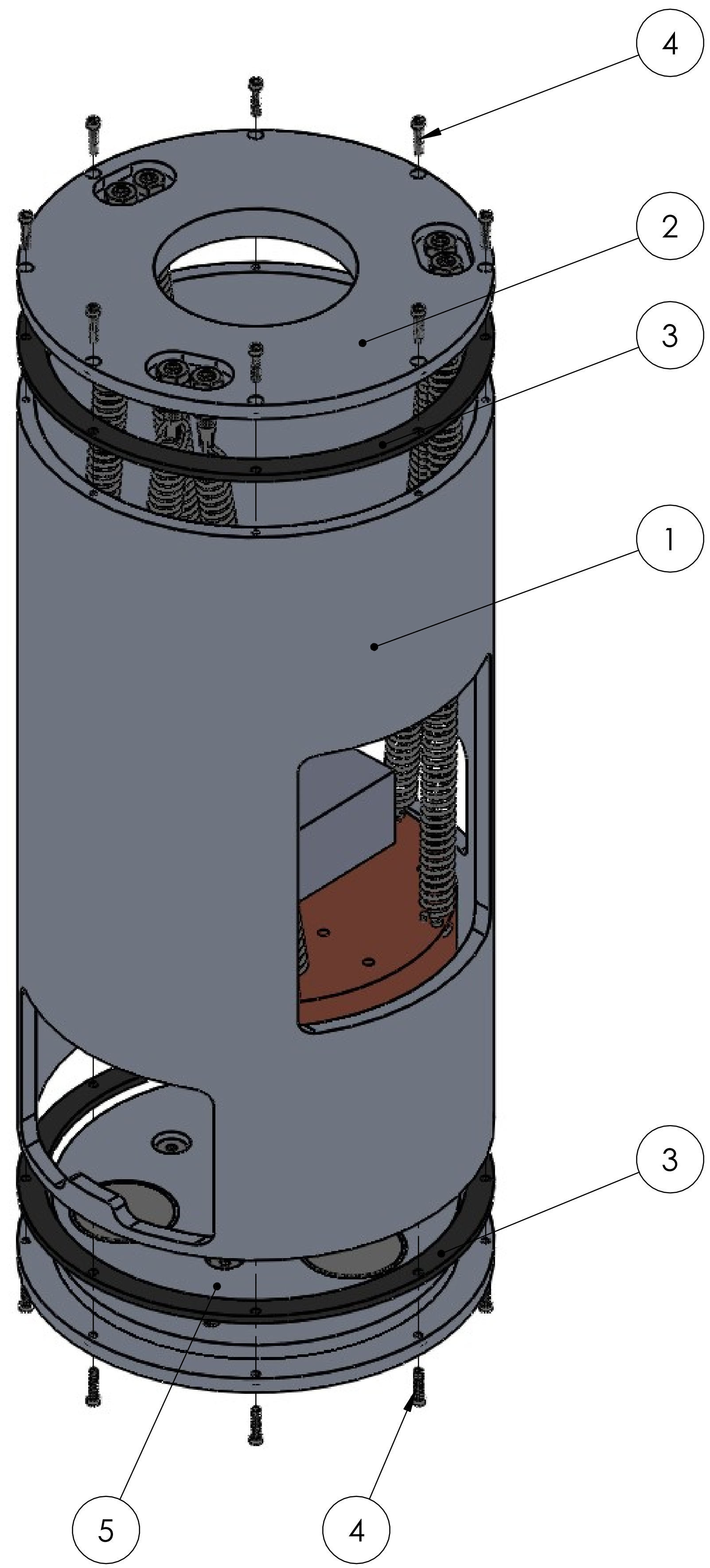
H Drawing Package

List of Drawings and Spec Sheets by Part Number:

000 Housing Assembly
- 001 Aluminum Cylinder
- 002 Viton Rings
- 003 #4-40 Vented Screws

100 Mass-Spring Assembly
- 101 Top Disk
- 102 Stainless Steel Springs
- 105 Viton Sheet
- 106 Long Stud Anchors
- 107 Short Stud Anchors
- 108/206 1/4"-20 Thin Hex Nuts

200 Magnetic Adjustment Assembly
- 201 Bottom Disk
- 202 Permanent Magnets
- 203 Magnetic Swash Plate
- 204 1/4"-20 x 5/8" Socket Screws
- 205 1/4"-20 x 7/8" Socket Screws



QTY.	ITEM NO.	CAGE CODE	DESCRIPTION	NOTES
1	5	39428	MAGNETIC ADJUSTMENT ASSEMBLY	
16	4	39428	#4-40 VENTED SOCKET SCREW	
2	3	39428	VITON RING	
1	2	39428	MASS-SPRING ASSEMBLY	
1	1	39428	ALUMINUM HOUSING	

UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN INCHES
TOLERANCES:
FRACTIONAL±
ANGULAR: MACH± BEND ±
TWO PLACE DECIMAL ±
THREE PLACE DECIMAL ±

INTERPRET GEOMETRIC TOLERANCING PER:

MATERIAL VARIES

FINISH VARIES

DO NOT SCALE DRAWING

NEXT ASSY | USED ON

APPLICATION

	NAME	DATE
DRAWN	S. D.	2-10-2021
CHECKED	J. B.	2-10-2021
ENG APPR.		
MFG APPR.		
Q.A.		

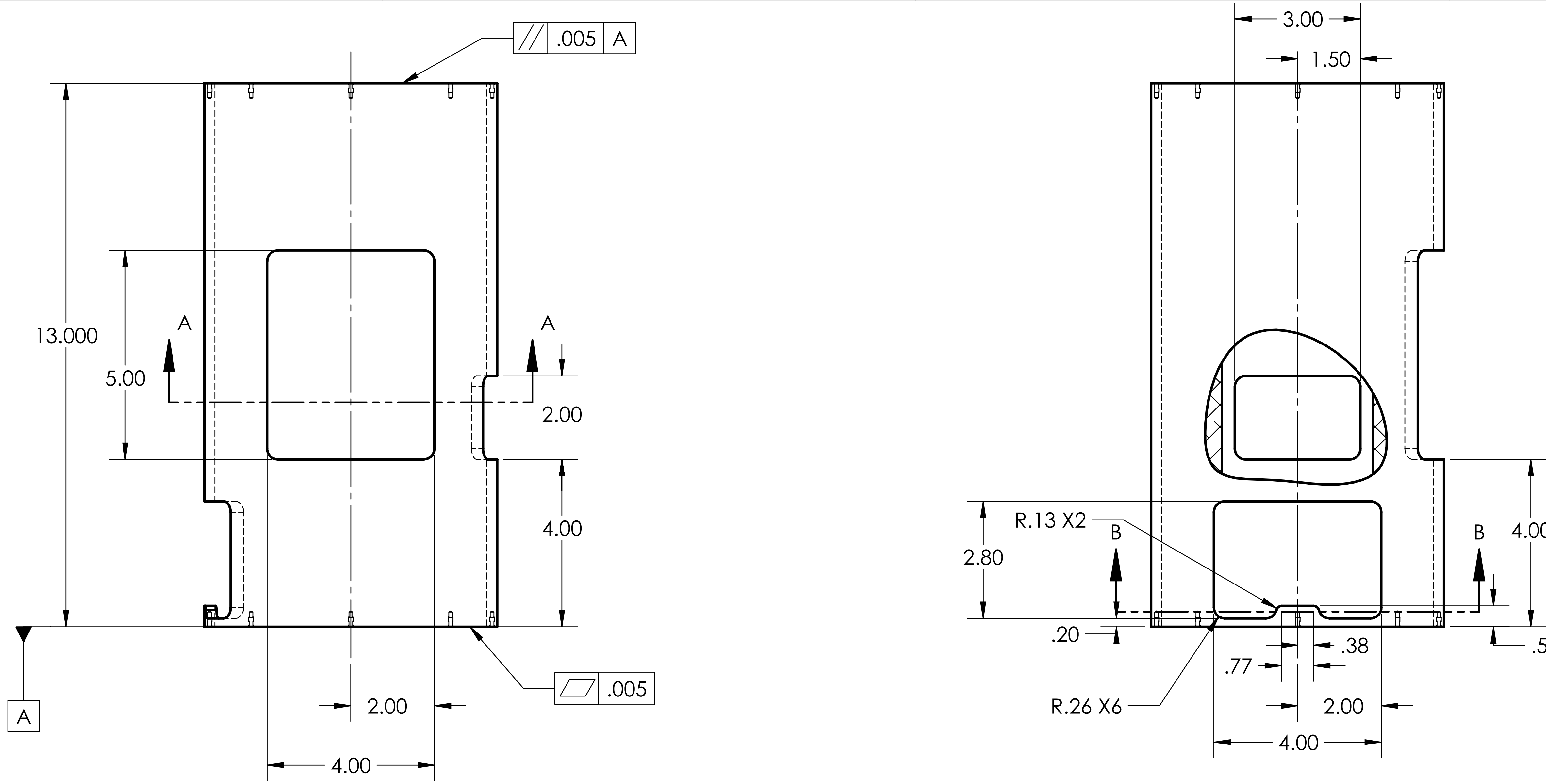
COMMENTS:

TITLE:

HOUSING ASSEMBLY

SIZE B | DWG. NO. 000 | REV

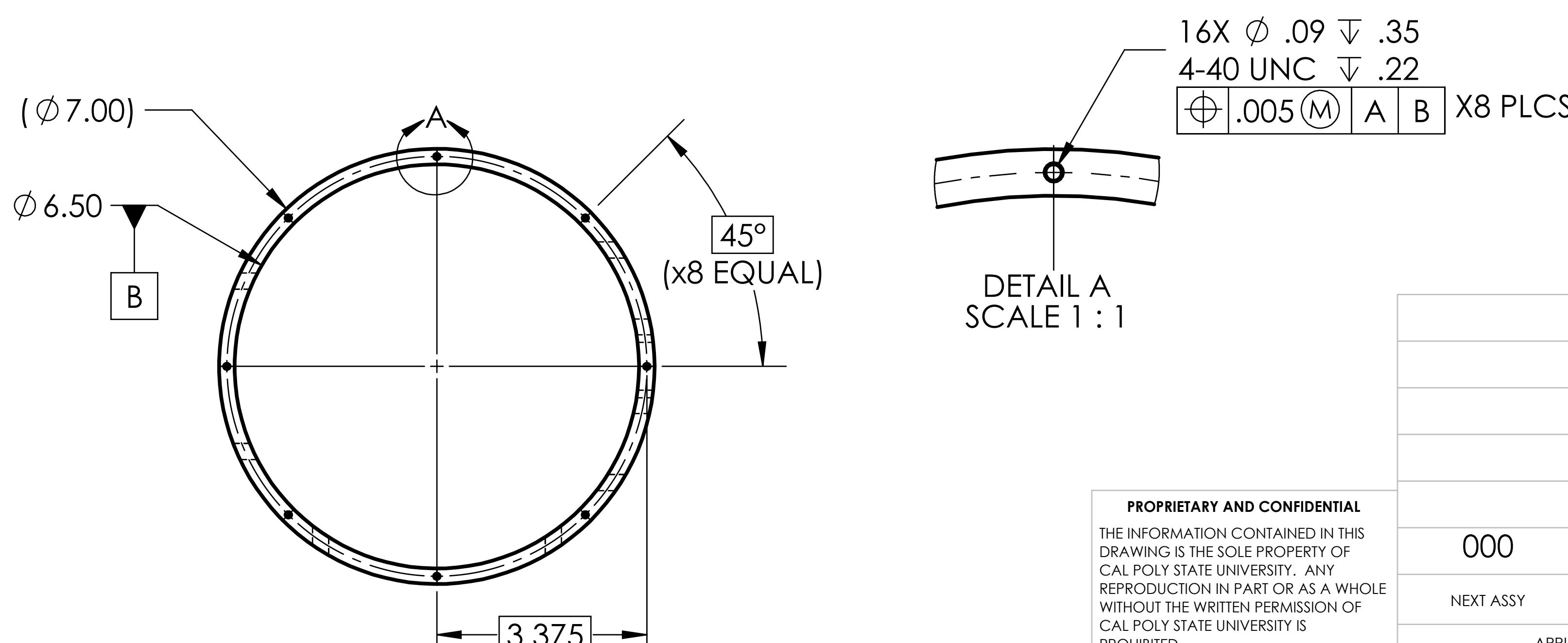
SCALE: 1:2 WEIGHT: SHEET 1 OF 1



Notes:

UNLESS OTHERWISE SPECIFIED

1. ALL DIMENSIONS ARE IN INCHES
2. ALL DIMENSIONS TO FEATURES WITH TRUE POSITION ARE IMPLIED BASIC
3. 32 FAO
4. BREAK SHARP EDGES 0.02 MAX



PROPRIETARY AND CONFIDENTIAL

NEXT ASSY	USED ON
000	
APPLICATION	

UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN INCHES
TOLERANCES:
FRACTIONAL ± N/A
ANGULAR: ±1/4°
TWO PLACE DECIMAL ± .01
THREE PLACE DECIMAL ±.005

INTERPRET GEOMETRIC TOLERANCING PER:

MATERIAL
6061 T-6 ALUMINUM

FINISH

DO NOT SCALE DRAWING

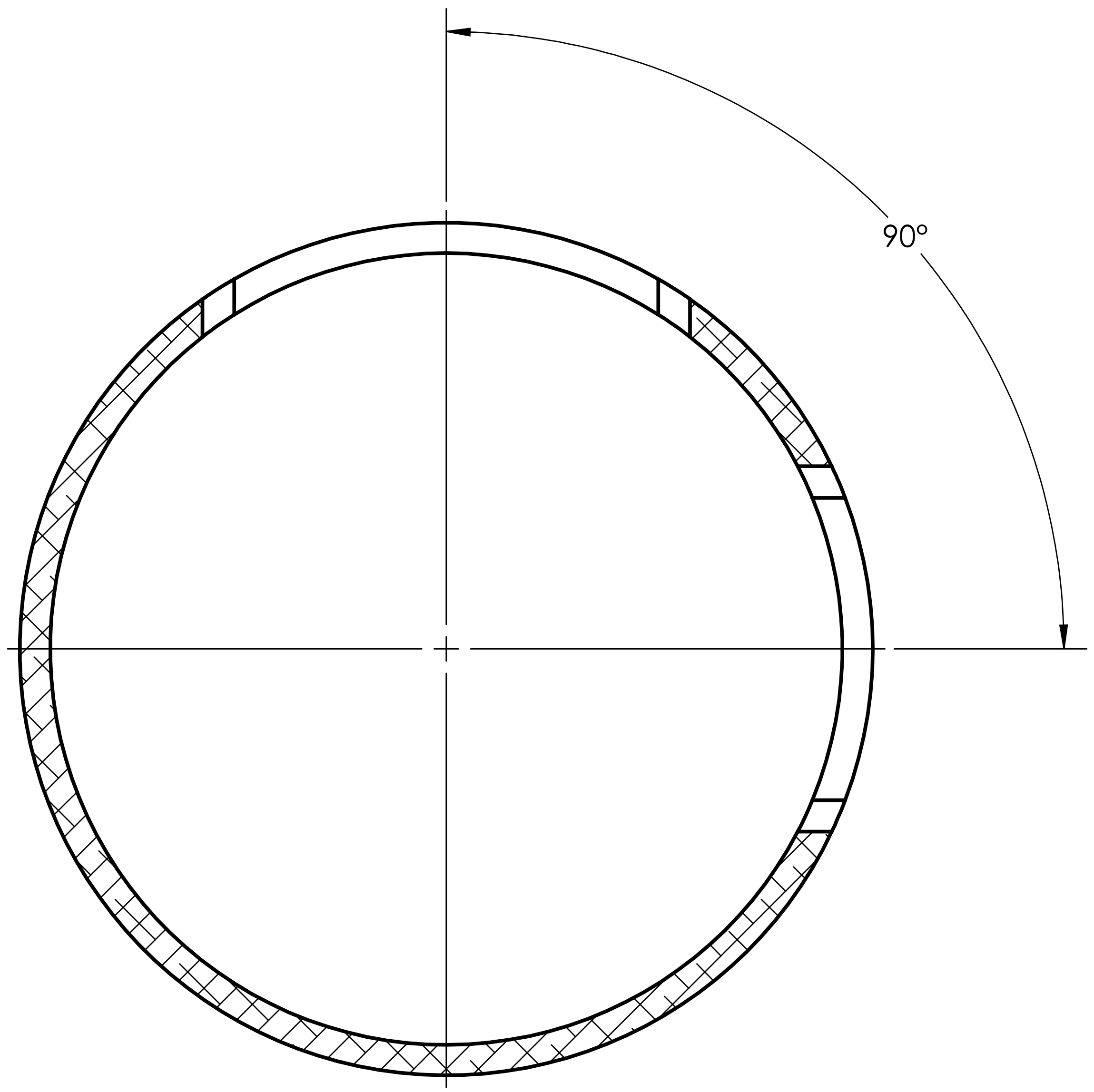
	NAME	DATE
DRAWN	JASON B	1/11/21
CHECKED	Stephen	2/10/21
ENG APPR.		
MFG APPR.		
Q.A.		
COMMENTS:		

TITLE:

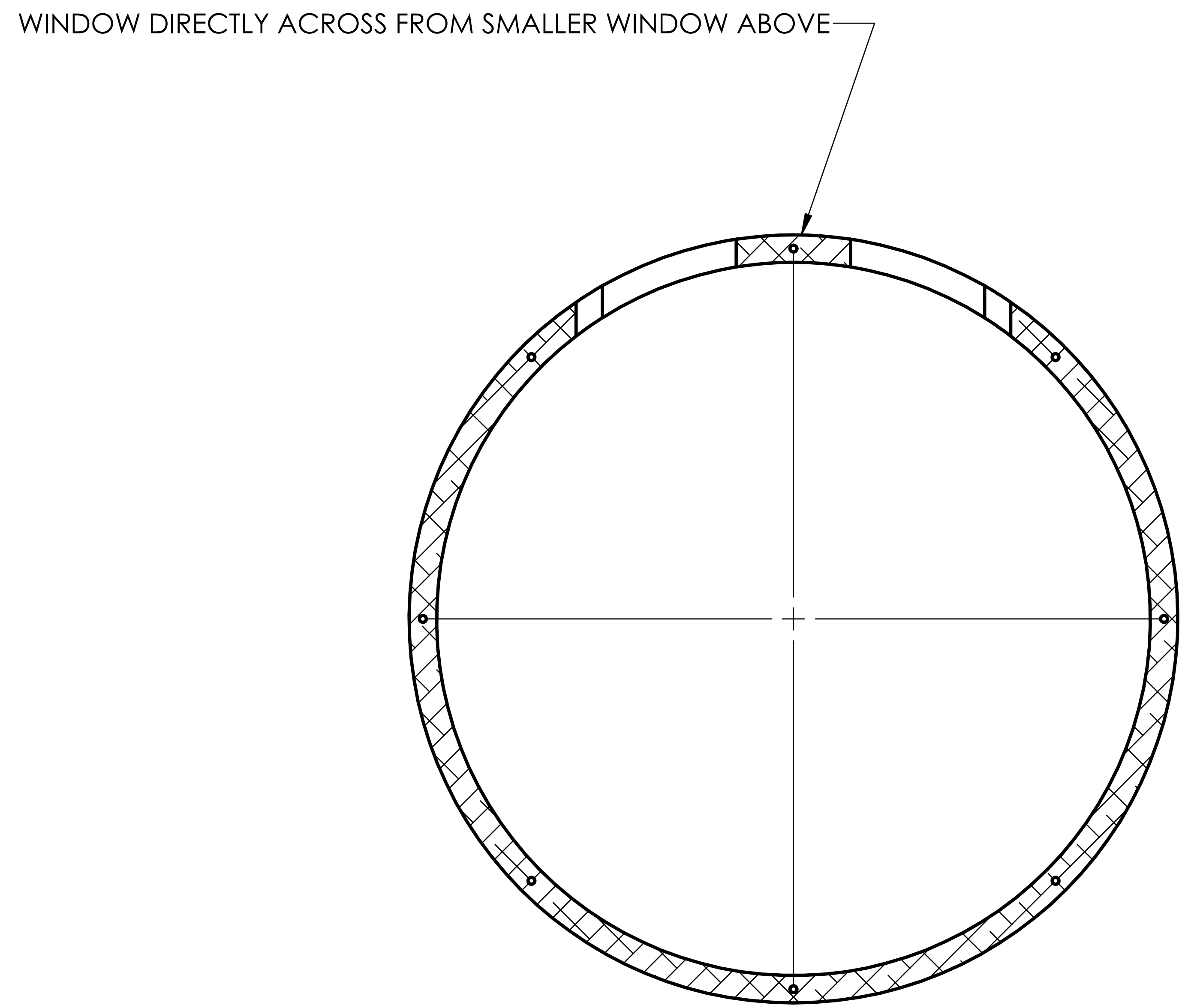
HOUSING DRAWING

SIZE	DWG. NO.	REV
B	001	3

SCALE: 1:3 WEIGHT: SHEET 1 OF 3



SECTION A-A
SCALE 2 : 3



SECTION B-B
SCALE 2 : 3

NEXT ASSY	USED ON
000	
APPLICATION	

UNLESS OTHERWISE SPECIFIED:

DIMENSIONS ARE IN INCHES
TOLERANCES:
FRACTIONAL± N/A
ANGULAR: ±1/4°
TWO PLACE DECIMAL ±.01
THREE PLACE DECIMAL ±.005

INTERPRET GEOMETRIC TOLERANCING PER:

MATERIAL
6061 T-6 ALUMINUM

FINISH

DO NOT SCALE DRAWING

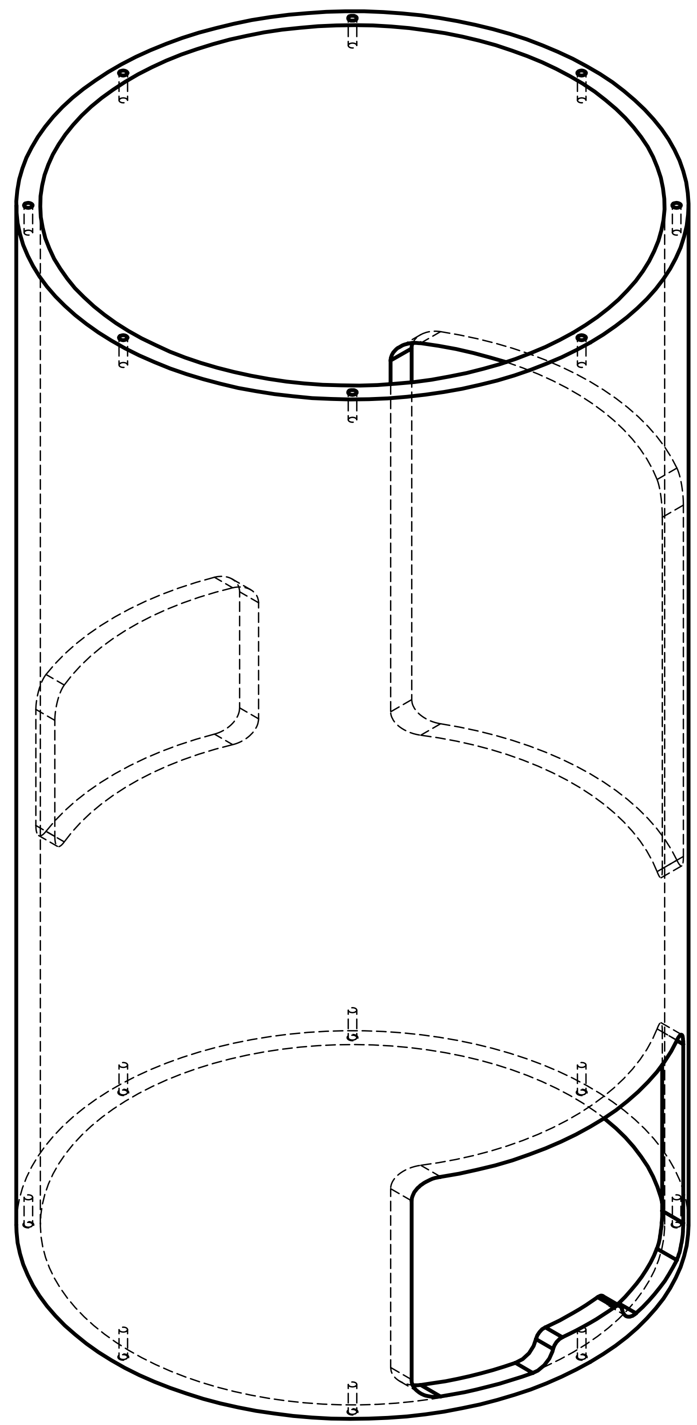
	NAME	DATE
DRAWN	JASON B	1/11/21
CHECKED	Stephen	2/10/21
ENG APPR.		
MFG APPR.		
Q.A.		

COMMENTS:

TITLE:
HOUSING WINDOW DETAILS

SIZE	DWG. NO.	REV
B	001	1

SCALE: 1:3 WEIGHT: SHEET 2 OF 3



ISOMETRIC VIEW

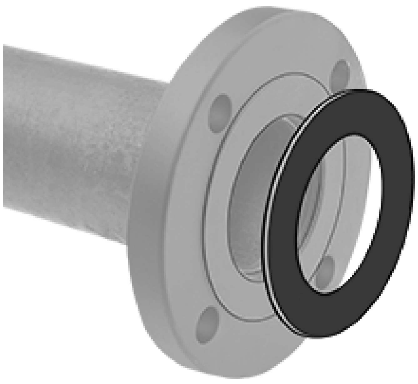
		UNLESS OTHERWISE SPECIFIED:		NAME	DATE	
		DIMENSIONS ARE IN INCHES TOLERANCES: FRACTIONAL± N/A ANGULAR: ±1/4° TWO PLACE DECIMAL ±.01 THREE PLACE DECIMAL ±.005	DRAWN	JASON B	1/11/21	
			CHECKED	Stephen	2/10/21	TITLE: ISOMETRIC VIEW
			ENG APPR.			
			MFG APPR.			
		INTERPRET GEOMETRIC TOLERANCING PER:	Q.A.			
000		MATERIAL 6061 T-6 ALUMINUM	COMMENTS:			SIZE B DWG. NO. 001 REV 1
NEXT ASSY	USED ON	FINISH				
APPLICATION		DO NOT SCALE DRAWING				SCALE: 1:2 WEIGHT: SHEET 3 OF 3

McMASTER-CARR®

Chemical-Resistant Viton® Fluoroelastomer Gasket

for 6 Pipe Size, ANSI Class 150

$19.38 Each
9473K643



Shape	Ring
For Pipe Size	6
ID	6 5/8"
OD	8 3/4"
For Flange ANSI Class	150
Maximum Pressure	Not Rated
Temperature Range	-20° to 450° F
Color	Black
Material	Viton® Fluoroelastomer Rubber
Backing Type	Plain
Thickness	1/8 in.
RoHS	RoHS 3 (2015/863/EU) Compliant
REACH	REACH (EC 1907/2006) (06/25/2020, 209 SVHC) Compliant
DFARS	Specialty Metals COTS-Exempt
Country of Origin	United States
USMCA Qualifying	No
Schedule B	392690.4500
ECCN	EAR99

Resistant to boric acid, citric acid, isopropyl alcohol, fuel, oil, and transmission fluid, these gaskets are also known as ring gaskets. They are for use on raised-surface pipe flanges; they fit the surface inside the bolt holes and do not interfere with the bolt connection.

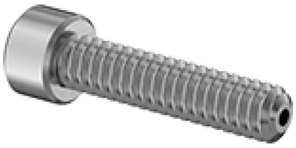
Gaskets for ANSI class flanges are sized so all gasket specifications meet ANSI standards. Gaskets for ANSI Class 150 flanges are compatible with low-pressure pipe flanges.

McMASTER-CARR®

Vented Socket Head Screw

4-40 Thread Size, 1/2" Long

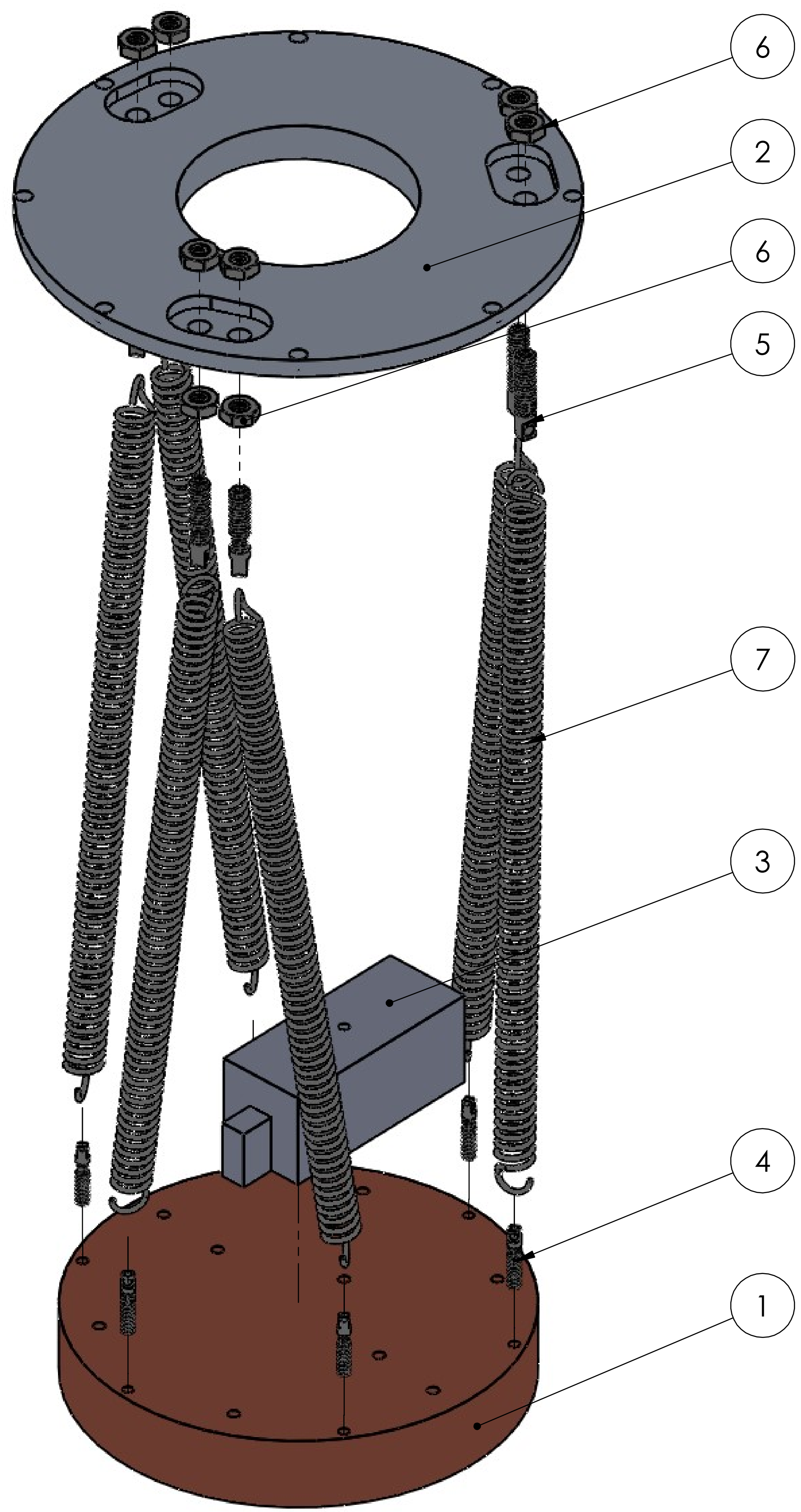
$10.53 per pack of 5
93235A110



Thread Size	4-40
Length	1/2"
Threading	Fully Threaded
Thread Spacing	Coarse
Head	
Diameter	0.183"
Height	0.112"
Drive Size	3/32"
Vent Diameter	0.035"
Material	18-8 Stainless Steel
Hardness	Rockwell B70
Tensile Strength	70,000 psi
Screw Size Decimal Equivalent	0.112"
Thread Type	UNC
Thread Fit	Class 3A
Thread Direction	Right Hand
Head Type	Socket
Socket Head Profile	Standard
Drive Style	Hex
Screw Features	Vented
System of Measurement	Inch
RoHS	RoHS 3 (2015/863/EU) Compliant
REACH	REACH (EC 1907/2006) (06/25/2020, 209 SVHC) Compliant
DFARS	Not Specialty Metals Compliant
Country of Origin	Peoples Republic Of China, Taiwan, or United States
USMCA Qualifying	No
Schedule B	731815.9000
ECCN	EAR99

Drilled through the head and shaft, these screws vent fluid and gases trapped below the screw, making them good for vacuum applications. Screws are 18-8 stainless steel; they have good chemical resistance and may be mildly magnetic. Length is measured from under the head.

Coarse threads are the industry standard; choose these screws if you don’t know the pitch or the threads per inch.



ITEM NO.	QTY.	CAGE CODE	DESCRIPTION	NOTES
7	6	39428	STAINLESS STEEL SPRINGS	
6	12	39428	THIN HEX NUT	
5	6	39428	LONG STUD ANCHOR	
4	6	39428	SHORT STUD ANCHOR	
3	1	-----	STM FIXTURE	
2	1	39428	TOP DISK	
1	1	-----	COPPER STAGE	

000 - HOUSING ASSY	
NEXT ASSY	USED ON
APPLICATION	

UNLESS OTHERWISE SPECIFIED:

DIMENSIONS ARE IN INCHES
TOLERANCES:
FRACTIONAL±
ANGULAR: MACH± BEND ±
TWO PLACE DECIMAL ±
THREE PLACE DECIMAL ±

INTERPRET GEOMETRIC TOLERANCING PER:

MATERIAL VARIOUS

FINISH VARIOUS

DO NOT SCALE DRAWING

	NAME	DATE
DRAWN	S.D.	2-10-2021
CHECKED	J.B.	2-10-2021
ENG APPR.		
MFG APPR.		
Q.A.		

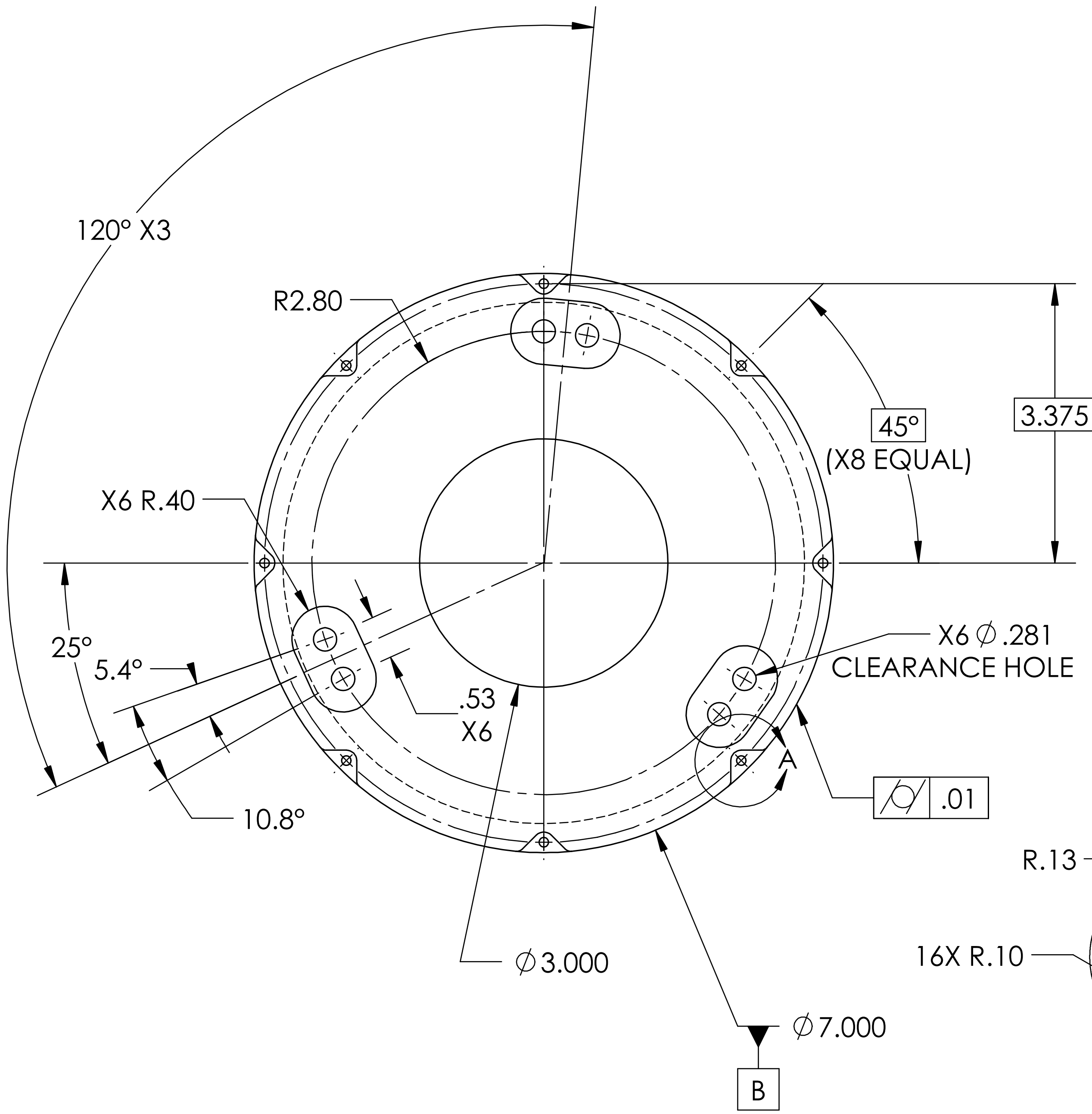
COMMENTS:

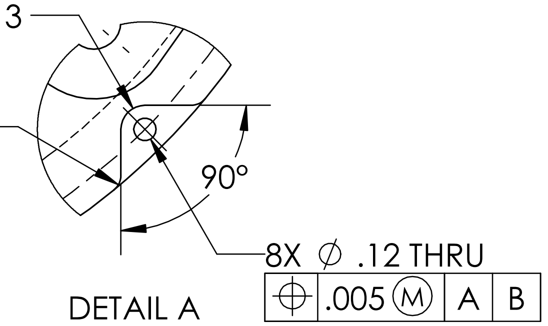
TITLE:

MASS SRING ASSEMBLY

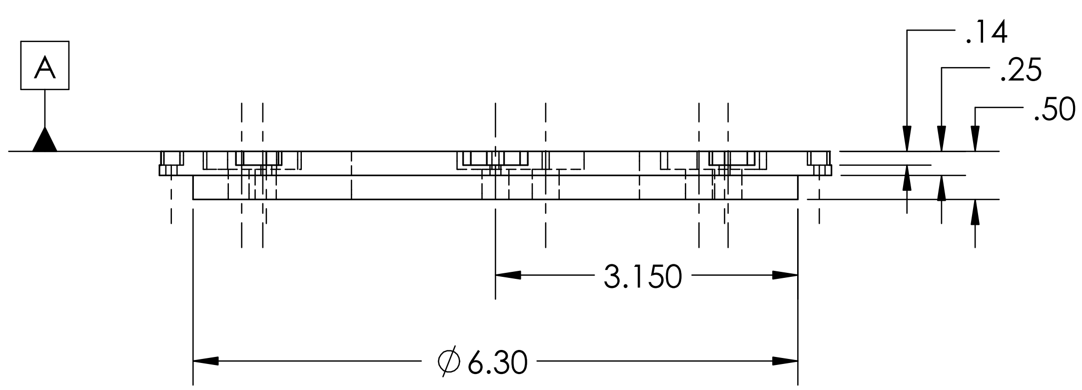
SIZE **B** DWG. NO. 100 REV

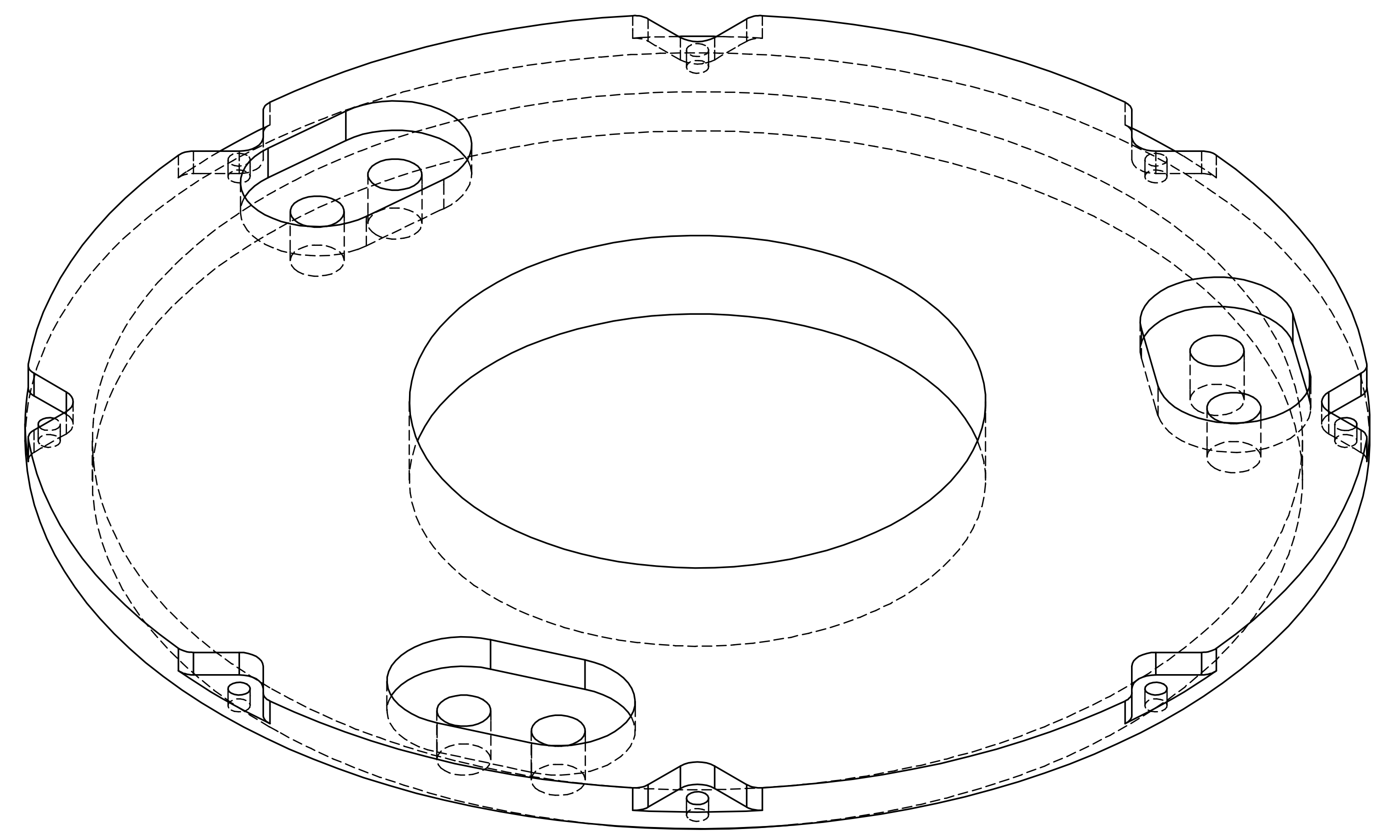
SCALE: 1:2 WEIGHT: SHEET 1 OF 1





DETAIL A
SCALE 1 : 1





SCALE: 1:1

Notes:

UNLESS OTHERWISE SPECIFIED

1. ALL DIMENSIONS ARE IN INCHES

2. ALL DIMENSIONS TO FEATURES WITH TRUE POSITION ARE IMPLIED BASIC

3. 32 FAO

4. BREAK SHARP EDGES 0.02 MAX

NEXT ASSY	USED ON
100	

APPLICATION

UNLESS OTHERWISE SPECIFIED:

DIMENSIONS ARE IN INCHES
TOLERANCES:
FRACTIONAL±
ANGULAR: ±1/4°
TWO PLACE DECIMAL ±.01
THREE PLACE DECIMAL ±.005

INTERPRET GEOMETRIC TOLERANCING PER:

MATERIAL
AL-6061 T6

FINISH

DO NOT SCALE DRAWING

	NAME	DATE
DRAWN	JASON B	1/11/21
CHECKED	STEPHEN D	6/4/21
ENG APPR.		
MFG APPR.		
Q.A.		

COMMENTS:

COUNTERBORES ON #4-40 HOLES CHANGED TO V-SLOTS FOR EASE OF MANUFACTURING.

TITLE:

TOP DISK DRAWING

SIZE	DWG. NO.	REV
B	101	4

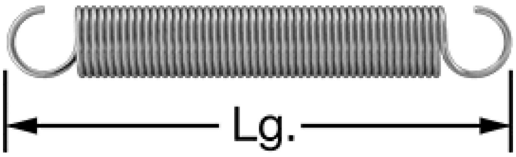
SCALE: 1:2 | WEIGHT: | SHEET 1 OF 1

McMASTER-CARR®

302 Stainless Steel Extension Spring with Hook Ends

5" Long, 0.75" OD, 0.075" Wire Diameter

$9.99 per pack of 3
9433K114



Spring Type	Extension
System of Measurement	Inch
Length	5"
OD	0.75"
Wire Diameter	0.075"
Extended Length @ Maximum Load	11.22"
Load, lbs.	
Min.	1.74
Maximum	18.48
Rate	2.7 lbs./in.
Material	302 Stainless Steel
End Type	Hook
OD Tolerance	-0.015" to 0.015"
Min. Load Tolerance	-0.26 to 0.26 lbs.
Rate Tolerance	-0.27 lbs./in. to 0.27 lbs./in.
RoHS	RoHS 3 (2015/863/EU) Compliant
REACH	REACH (EC 1907/2006) (06/25/2020, 209 SVHC) Compliant
DFARS	Specialty Metals COTS-Exempt
Country of Origin	United States
USMCA Qualifying	No
Schedule B	732020.5000
ECCN	EAR99

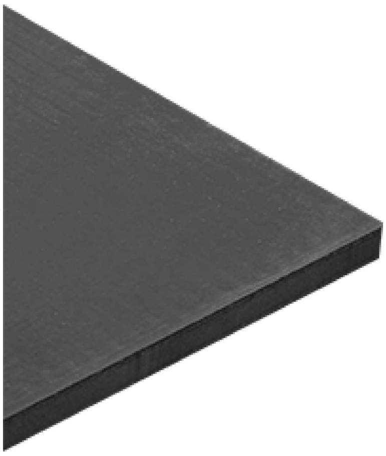
Made of stainless steel, these springs are more corrosion resistant than steel springs. They're also easier to extend than steel springs. As you stretch an extension spring, it gets harder to pull. Minimum load is the amount of force required to start to extend the spring. Maximum load is the amount of force required to fully extend the spring. Rate is the amount of force required for every inch of extension or, for metric springs, millimeter of compression.

302 stainless steel springs have good corrosion resistance.

Viton® Fluoroelastomer Rubber Sheet

Chemical-Resistant, 6" x 6", 1/8" Thick

$30.33 Each
86075K24



Material	Viton® Fluoroelastomer
Shape	Sheet and Bar
Texture	Smooth
Thickness	1/8"
Thickness Tolerance	-0.020" to +0.020"
Width	6"
Width Tolerance	+0.500"
Length	6"
Length Tolerance	+0.5"
Backing Type	Plain
For Use Outdoors	Yes
Temperature Range	-20° to 400° F
Tensile Strength	1,000 psi
Color	Black
Durometer	75A (Hard)
Durometer Tolerance	-5 to +5
RoHS	RoHS 3 (2015/863/EU) Compliant
REACH	REACH (EC 1907/2006) (06/25/2020, 209 SVHC) Compliant
DFARS	Specialty Metals COTS-Exempt
Country of Origin	United States
USMCA Qualifying	No
Schedule B	392099.0000
ECCN	EAR99

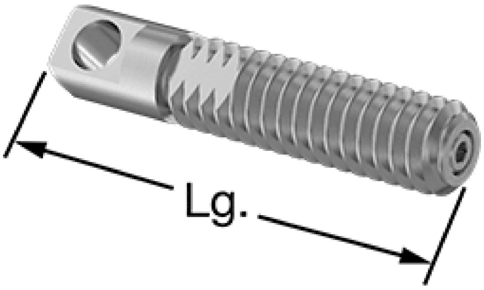
Viton® fluoroelastomer stands up to tough chemicals, such as nitric acid, ethylene glycol, and isopropyl alcohol. It is also known as FKM.

McMASTER-CARR®

18-8 Stainless Steel Swivel Extension Spring Stud Anchor

1/4"-20 Thread Size

$19.25 Each
96376A330



For Spring Type	Extension
Thread	
Size	1/4"-20
Type	UNC
Spacing	Coarse
Direction	Right Hand
Min. Length	7/8"
Length	1 1/4"
Diameter	1/4"
Hole Diameter	0.15"
Hex Size	5/64"
Material	18-8 Stainless Steel
Anchor Type	Stud
Features	360° Rotating Head, Wrench Flats
RoHS	RoHS 3 (2015/863/EU) Compliant
REACH	REACH (EC 1907/2006) (06/25/2020, 209 SVHC) Compliant
DFARS	Specialty Metals COTS-Exempt
Country of Origin	United States
USMCA Qualifying	Yes
Schedule B	731600.0000
ECCN	EAR99

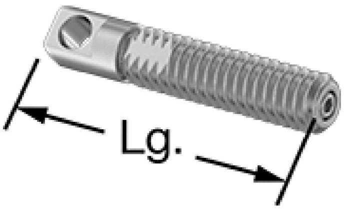
These stainless steel anchor studs offer excellent corrosion resistance. The head swivels 360° so the spring stays straight and properly coiled. Attach an extension spring to a stud and then thread the stud into your part for a secure connection. To make tightening easier, anchors have a hex socket on the bottom and wrench flats at the top of the threads.

McMASTER-CARR®

18-8 Stainless Steel Swivel Extension Spring Stud Anchor

8-32 Thread Size

$16.90 Each
96376A220



For Spring Type	Extension
Thread	
Size	8-32
Type	UNC
Spacing	Coarse
Direction	Right Hand
Min. Length	5/8"
Length	7/8"
Diameter	0.164"
Hole Diameter	0.1"
Hex Size	0.05"
Material	18-8 Stainless Steel
Anchor Type	Stud
Features	360° Rotating Head, Wrench Flats
RoHS	RoHS 3 (2015/863/EU) Compliant
REACH	REACH (EC 1907/2006) (06/25/2020, 209 SVHC) Compliant
DFARS	Specialty Metals COTS-Exempt
Country of Origin	United States
USMCA Qualifying	Yes
Schedule B	731600.0000
ECCN	EAR99

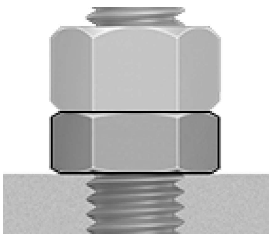
These stainless steel anchor studs offer excellent corrosion resistance. The head swivels 360° so the spring stays straight and properly coiled. Attach an extension spring to a stud and then thread the stud into your part for a secure connection. To make tightening easier, anchors have a hex socket on the bottom and wrench flats at the top of the threads.

McMASTER-CARR®

18-8 Stainless Steel Thin Hex Nut

1/4"-20 Thread Size

$4.16 per pack of 100
91847A029



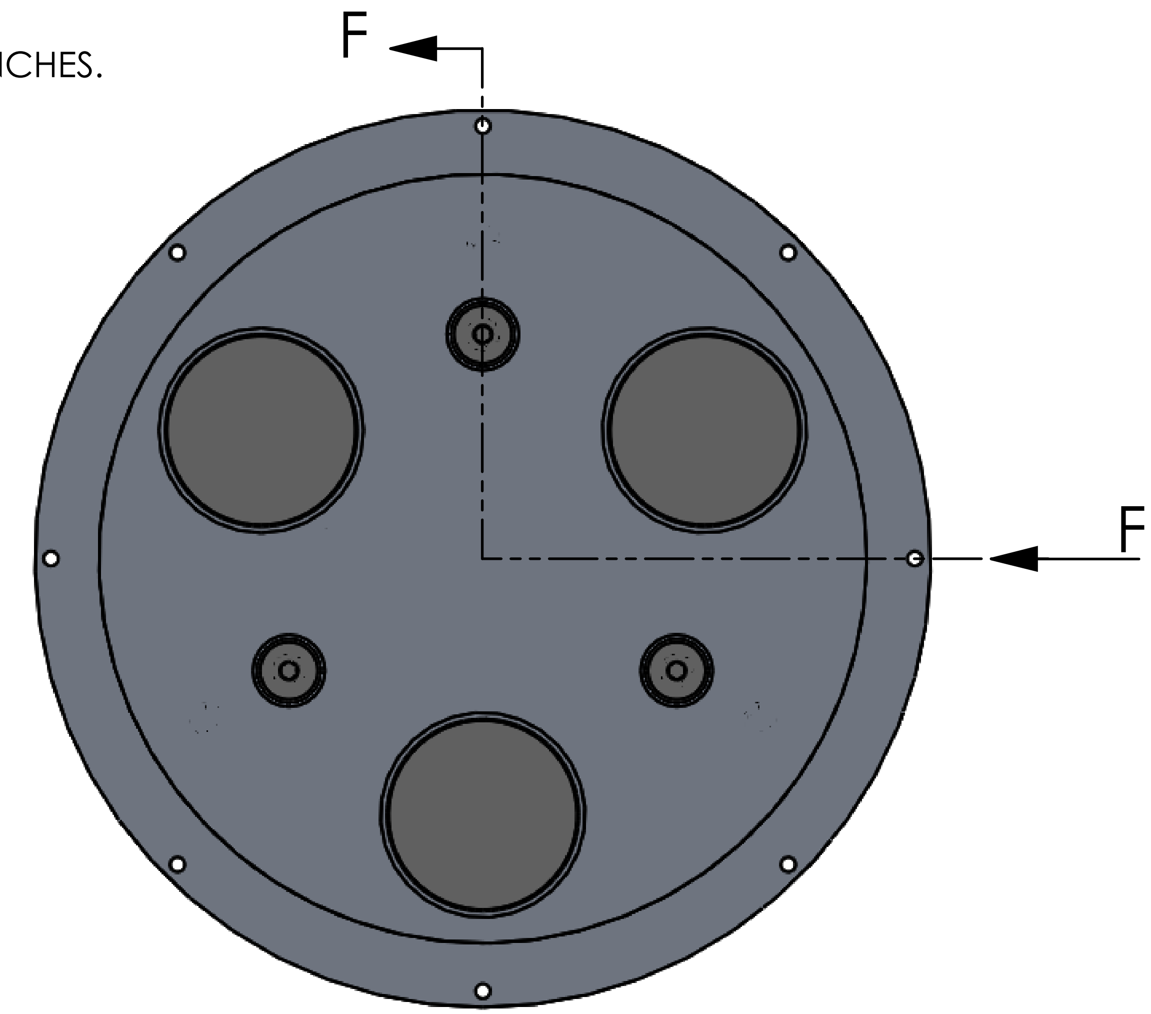
Material	18-8 Stainless Steel
Thread Size	1/4"-20
Thread Type	UNC
Thread Spacing	Coarse
Thread Fit	Class 2B
Thread Direction	Right Hand
Width	7/16"
Height	5/32"
Drive Style	External Hex
Nut Type	Hex
Hex Nut Profile	Thin
System of Measurement	Inch
RoHS	RoHS 3 (2015/863/EU) Compliant
REACH	REACH (EC 1907/2006) (06/25/2020, 209 SVHC) Compliant
DFARS	Not Specialty Metals Compliant
Country of Origin	Taiwan
Schedule B	731816.0000
ECCN	EAR99

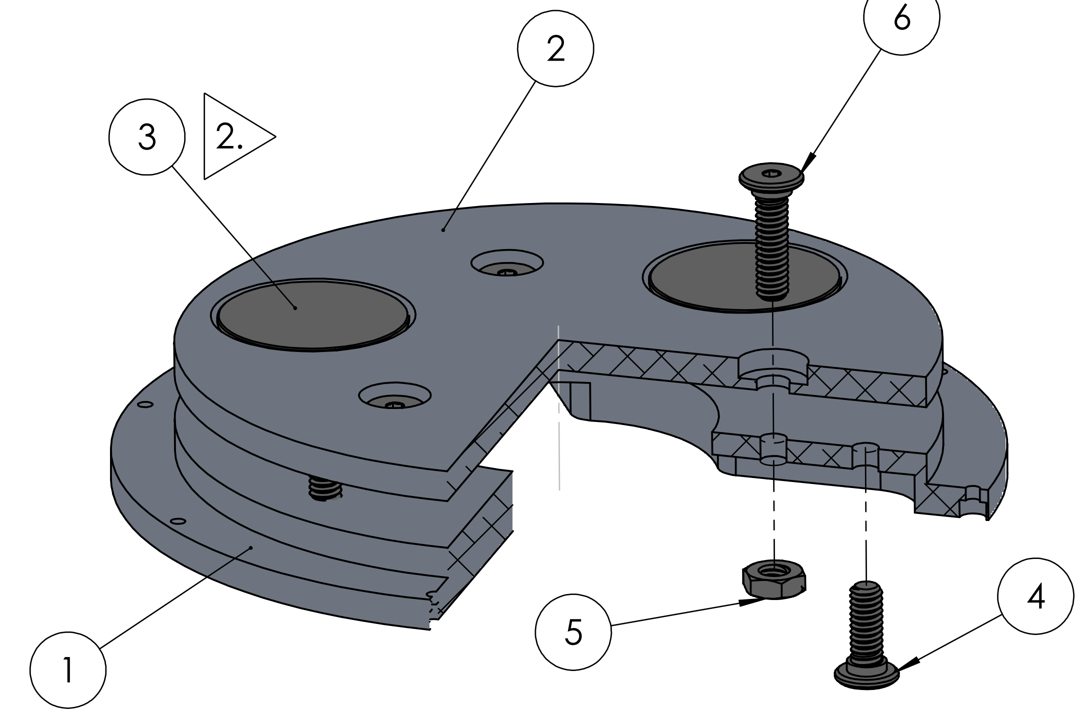
These nuts have good chemical resistance and may be mildly magnetic. Also known as jam nuts, they are about half the height of standard hex nuts. Use them in low-clearance applications or jam one against another nut to hold it in place.

NOTES:

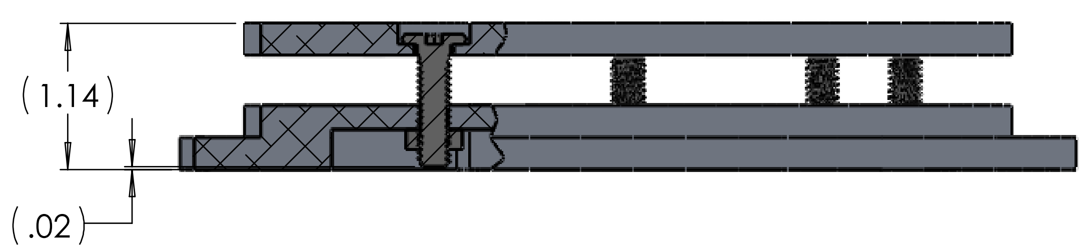
1. DIMENSIONS ARE IN INCHES.

2. PRESS FIT 3 INTO 2.





SECTION G-G
EXPLODED VIEW
REFERENCE ONLY
SCALE 2 : 3



QTY	ITEM	CAGE CODE	DESCRIPTION	NOTES
3	6	39428	316 STAINLESS STEEL ULP SOCKET HEAD SCREW, 1/4"-20, 7/8" LONG	
6	5	39428	THIN HEX NUT, GRADE 5, ZINC-PLATED, 1/4"-20 THREAD SIZE	
3	4	39428	316 STAINLESS STEEL ULP SOCKET HEAD SCREW, 1/4"-20, 5/8" LONG	
3	3	39428	PERMANENT MAGNETS 1.25" DIAMETER, 1/8" THICK	
1	2	39428	MAGNETIC SWASH PLATE	
1	1	39428	VIBRATION ISOLATION HOUSING BOTTOM CAP	
-1				

NEXT ASSY	USED ON
APPLICATION	

UNLESS OTHERWISE SPECIFIED:

DIMENSIONS ARE IN INCHES
TOLERANCES:
FRACTIONAL ±
ANGULAR: MACH ± BEND ±
TWO PLACE DECIMAL ±
THREE PLACE DECIMAL ±

INTERPRET GEOMETRIC TOLERANCING PER:

MATERIAL

FINISH

DO NOT SCALE DRAWING

	NAME	DATE
DRAWN	R. M.	2021-02-11
CHECKED	JASON B	2/10/21
ENG APPR.		
MFG APPR.		
Q.A.		

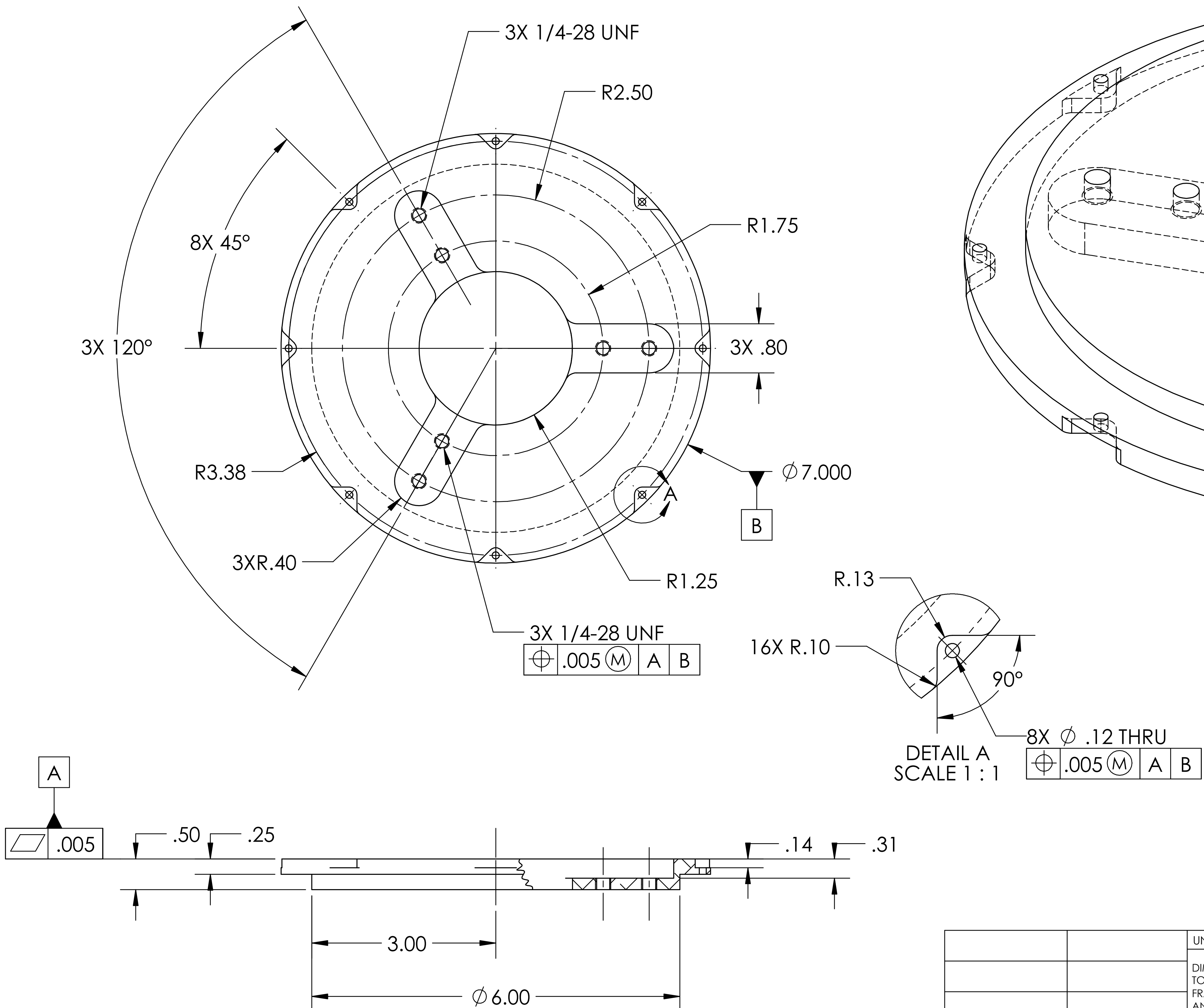
COMMENTS:

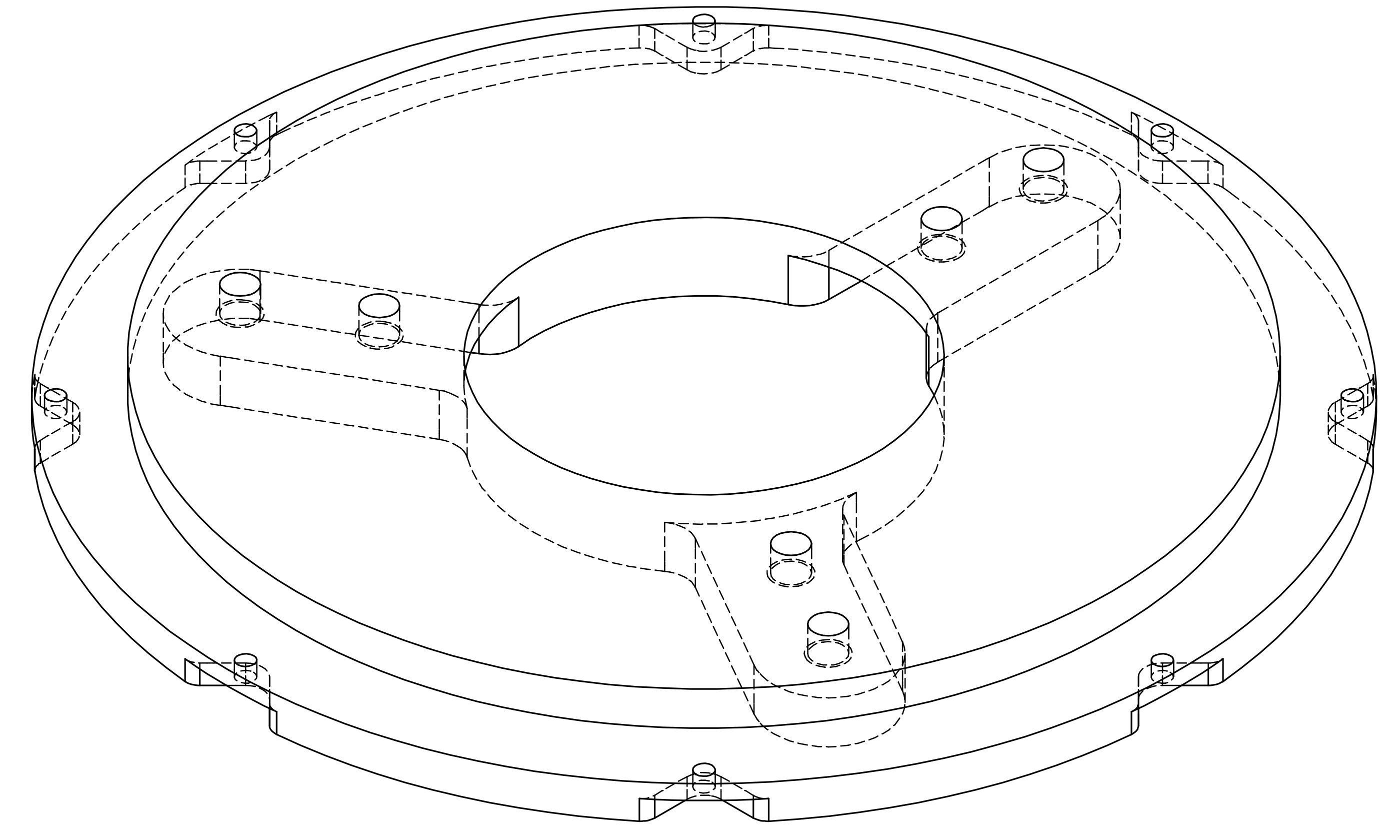
TITLE:

MAGNETIC ADJUSTMENT ASSEMBLY

SIZE	DWG. NO.	REV
B	200	X1

SCALE: 1:2 WEIGHT: SHEET 1 OF 1





NOTES:

UNLESS OTHERWISE SPECIFIED

1. DIMENSIONS ARE IN INCHES
2. ALL DIMENSIONS TO FEATURES WITH TRUE POSITION ARE IMPLIED BASIC.
3. 32√ FAO
4. MATERIAL: MAKE FROM P/N: ##### FROM MACMASTER-CARR.
5. BREAK SHARP EDGES 0.05 MAX

PROPRIETARY AND CONFIDENTIAL

NEXT ASSY	USED ON
200	

APPLICATION

UNLESS OTHERWISE SPECIFIED:

DIMENSIONS ARE IN INCHES
TOLERANCES:
FRACTIONAL±
ANGULAR:±1/4°
TWO PLACE DECIMAL ±.01
THREE PLACE DECIMAL ±.005

INTERPRET GEOMETRIC TOLERANCING PER:

MATERIAL: Al 6061 - T6

FINISH

DO NOT SCALE DRAWING

	NAME	DATE
DRAWN	STEPHEN	2/5/21
CHECKED	JASON	2/10/21
ENG APPR.		
MFG APPR.		
Q.A.		

COMMENTS:

TITLE:

BOTTOM DISK

SIZE	DWG. NO.	REV
B	201	2

SCALE: 1:2 WEIGHT: SHEET 1 OF 1

DX84 Specification Sheet

Product Specifications

Type:	DISC
Dimensions:	1.5 dia x 0.25 thk (in)
Tolerance:	All dimensions ± 0.004 in
Material:	NdFeB, Grade N42
Plating:	NiCuNi
Max Op Temp:	176ºF (80ºC)
Br max:	13,200 Gauss
BH max:	42 MGOe

Performance Specifications

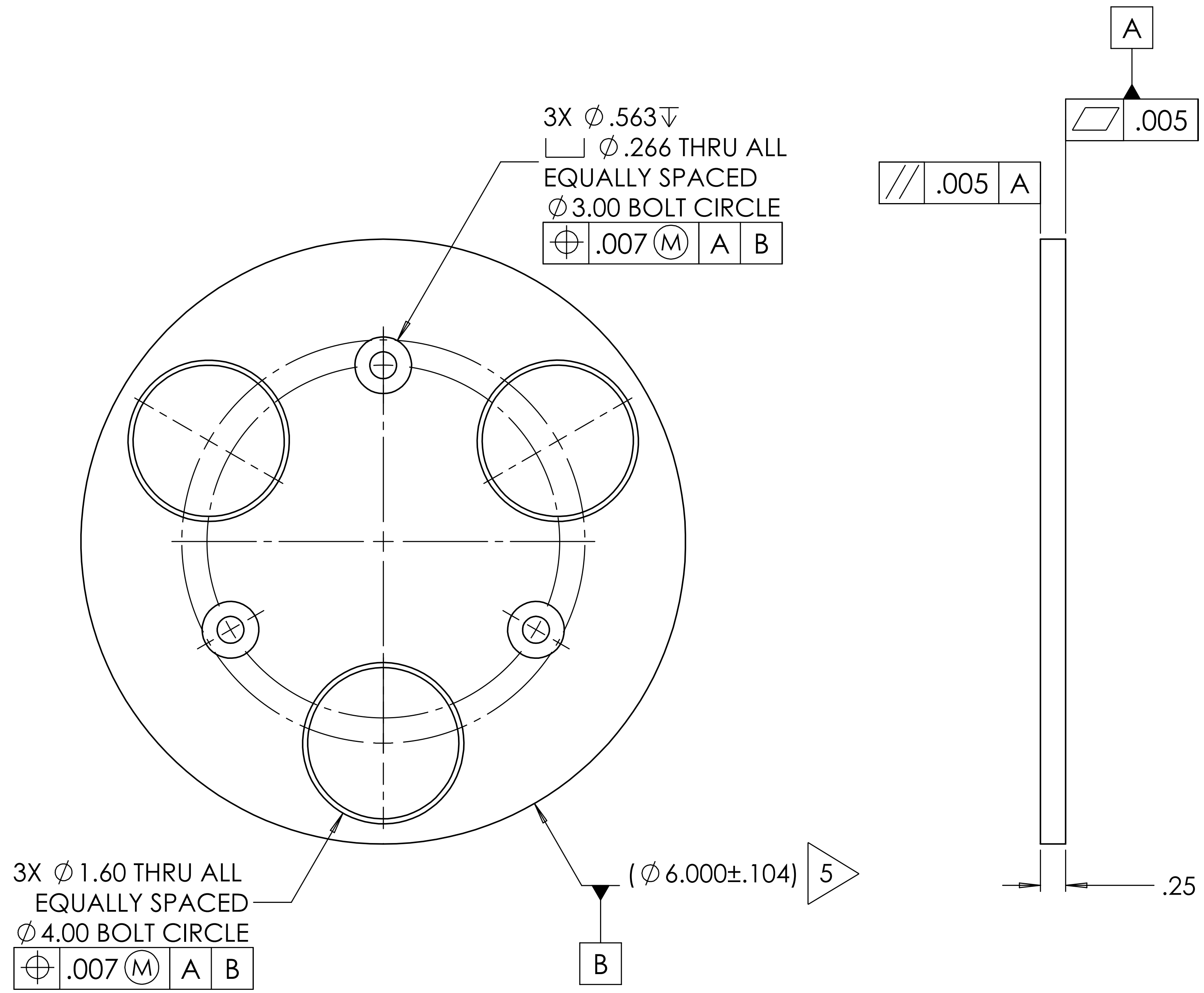
Pull Force, Case 1, Magnet to a Steel Plate:	34 lb

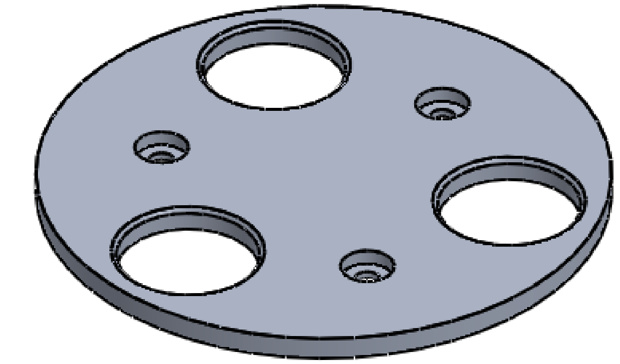
Surface Field values are derived from calculation and verification with experimental testing. These values are the field values at the surface of the magnet, centered on the axis of magnetization. Measurement of the B field with a magnetometer may yield varying results, depending on the geometry of your sensor. Pull Force values are based on extensive product testing in our laboratory. Different configurations of magnets and surrounding ferromagnetic materials may substantially alter your results.

NOTES:

UNLESS OTHERWISE SPECIFIED

1. DIMENSIONS ARE IN INCHES.
2. NOT ALL DIMENSIONS ARE SHOWN. REFER TO .STEP FILE FOR DIMENSIONS WHICH ARE NOT SPECIFIED. DRAWING DIMENSIONS AND NOTES SHALL TAKE PRECIDENT OVER DEFAULT DIMENSIONS IN THE STEP FILE.
3. ALL DIMENSIONS TO FEATURES WITH TRUE POSITION ARE IMPLIED BASIC.
4. 32 FAO
5. MATERIAL: MAKE FROM P/N: 8974K6, FROM MCMASTER-CARR.
6. BREAK SHARP EDGES 0.05 MAX.





ISOMETRIC VIEW
-1 CONFIGURATION
REFERENCE ONLY
SCALE 1:2

UNLESS OTHERWISE SPECIFIED:		NAME	DATE
DIMENSIONS ARE IN INCHES TOLERANCES: FRACTIONAL ± 0.100 ANGULAR: MACH ± 1° TWO PLACE DECIMAL ±0.010 THREE PLACE DECIMAL ±0.005	DRAWN	R.M.	2020-2-2
	CHECKED	JASON B	2/10/21
	ENG APPR.		
	MFG APPR.		
INTERPRET GEOMETRIC TOLERANCING PER: ASME Y14.5 2018	Q.A.		
MATERIAL SEE NOTES	COMMENTS:		
FINISH SEE NOTES			
DO NOT SCALE DRAWING			

300	
NEXT ASSY	USED ON
APPLICATION	

PROPRIETARY AND CONFIDENTIAL

TITLE: MAGNETIC SWASH PLATE

SIZE B | DWG. NO. 301 | REV X1

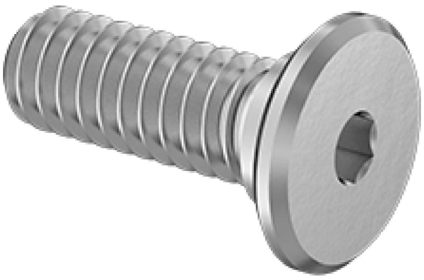
SCALE: 1:2 | WEIGHT: | SHEET 1 OF 1

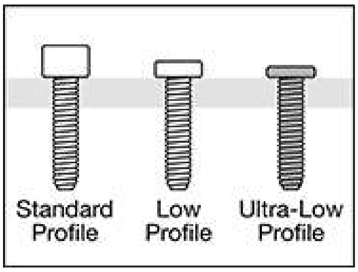
McMASTER-CARR®

316 Stainless Steel Ultra-Low-Profile Socket Head Screw

1/4"-20 Thread Size, 5/8" Long

$4.19 Each
91223A334





Thread Size	1/4"-20
Length	5/8"
Threading	Fully Threaded
Thread Spacing	Coarse
Head	
Diameter	1/2"
Height	5/64"
Shoulder	
Diameter	5/16"
Length	5/64"
Drive Size	1/8"
Material	316 Stainless Steel
Hardness	Rockwell B70
Tensile Strength	70,000 psi
Screw Size Decimal Equivalent	0.25"
Thread Type	UNC
Thread Fit	Class 2A
Thread Direction	Right Hand
Head Type	Socket
Socket Head Profile	Ultra-Low
Drive Style	Hex
System of Measurement	Inch
RoHS	RoHS 3 (2015/863/EU) Compliant
REACH	REACH (EC 1907/2006) (06/25/2020, 209 SVHC) Compliant
DFARS	Specialty Metals Compliant (252.225-7009)
Country of Origin	United States
USMCA Qualifying	No
Schedule B	731815.9000
ECCN	EAR99

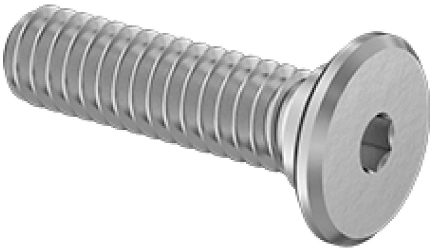
More corrosion resistant than 18-8 stainless steel screws, these 316 stainless steel screws have excellent resistance to chemicals and salt water. They may be mildly magnetic. To fit in the tightest spaces, they have a head that's about one-third the height of a standard socket head, but their low profile makes their head weaker. Not recommended for critical fastening applications. These screws have a shoulder that provides some support and stability to the head. Length is measured from under the shoulder.

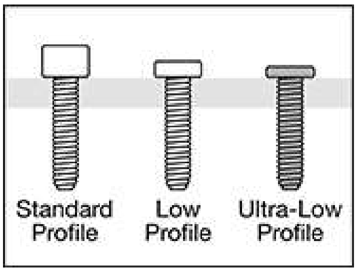
MCMASTER-CARR®

316 Stainless Steel Ultra-Low-Profile Socket Head Screw

1/4"-20 Thread Size, 7/8" Long

$5.12 Each
91223A335





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I Project Budget

Title of Senior Project:	F41- Vibration Isolation System for a Scanning Tunneling Microscope		
Team members:	Stephen DeHerrera, Seth Ewing, Jason Behrens, Ryan McMullen		
Designated Team Treasurer:	Seth Ewing		
Faculty Advisor:	Dr. Eltahry Elghandour		
Sponsor:	Dr. Gregory Scott, Toan Le		
Quarter and year project began:	Fall 2020 - Spring 2021		
Materials budget given for this project:	**Approximately $1000.00**		
Date purchased	**Vendor**	**Description of items purchased**	**Transaction amount**
Janurary 29, 2021	K&J Magnetics	Variety of neodymium magnets, ranging in diameter from 1 to 1.5", in thickness from 0.125 to 0.25", and strength from N42 to N52.	$114.02
Planned purchase.	McMaster	Viton ring with ID of 6.625" and OD of 8.75". Part number 9464K559.	$19.46
Planned purchase.	McMaster	#4-40 Vented Screw. Part number 93235A110.	$33.76
Planned purchase.	McMaster	Top cap, aluminum cylinder, 1" thick, 7" diameter.	$29.71
Planned purchase.	Viton.com	Sheet of viton, 0.25" thick. Part numbr 1084N91.	$5.60
Planned purchase.	McMaster	Bottom cap, aluminum cylinder, 1" thick, 7" diameter.	$29.71
Planned purchase.	McMaster	Magnetic swash plate, aluminum cylinder, 0.25" thick, 6" diameter. Part number 1610T56.	$16.78
Planned purchase.	McMaster	ULP Socket Screw (1/4-20), 0.625". Part number 91223A334.	$12.57
Planned purchase.	McMaster	ULP Socket Screw (1/4-20), 0.875". Part number 91223A335.	$15.36
Planned purchase.	McMaster	Hex Nut (1/4-20). Part number 92673A113	$0.12
		Total:	$277.09

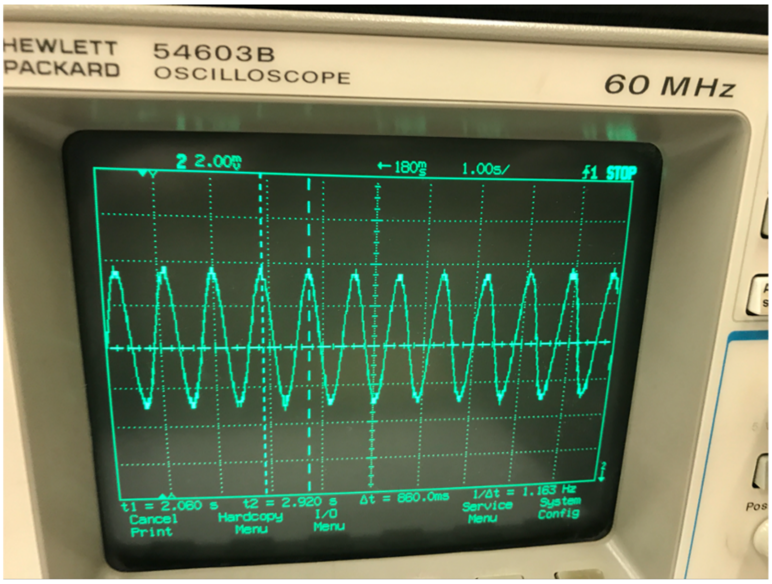
J Abaqus Parameters and Results

Abaqus Simulation Parameters		
Material Properties		
Property	Value	Unit
Material	Al 6061-T6	-
Density	2.52643E-04	lbf s^2/in^4
Modulus of Elasticity	9.9931E+06	lbf/in^2
Poisson's Ratio	0.33	-
Mesh Properties		
Property	Value	Unit
Element Shape	Hex	-
Element Technique	Sweep	-
Algorithm	Medial Axis	-
Global Size	0.1	inch
Element Library	Standard	-
Geometric Order	Quadratic	-
Hex Element Type	Reduced Integration	-
Boundary Condition Properties		
Property	Value	Unit
Boundary Type	Encastre	-

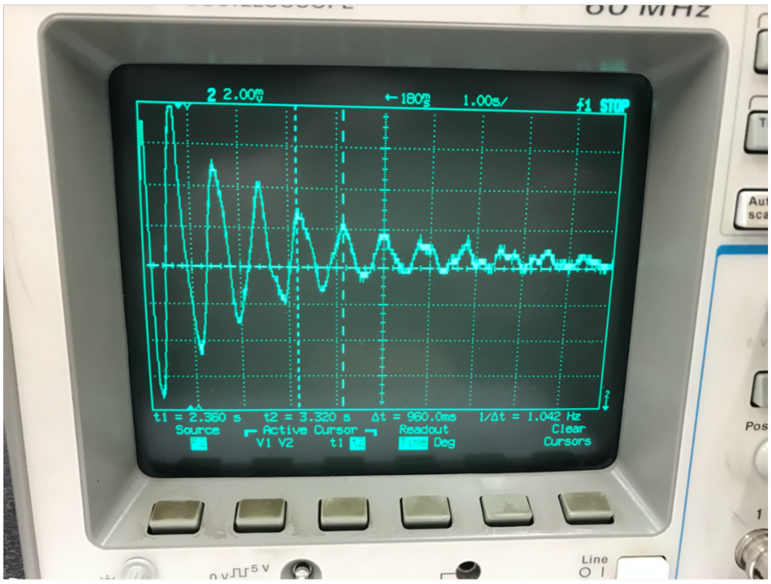
Abaqus Simulation Results	
Mode Number	Frequency [cycles/s]
1	485.48
2	630.59
3	642.64
4	708.97
5	1152.9
6	1494.7
7	1544.9
8	1625.9
9	1809.4
10	1866.6

K Preliminary Testing

The following images were taken during preliminary testing of the mass-spring-damper system. Image (a) shows the oscilloscope screen during a test of the undamped system with a natural frequency of 1.14 Hz displayed. Image (b) shows the oscilloscope screen during a test where the stage is damped with a single N52 magnet. The amplitude appears to decay exponentially, confirming our modeling of eddy current damping as viscous damping. The damped frequency of 1.04 H is displayed.



(a)



(b)

L Design Hazard Checklist

PDR Design Hazard Checklist **F41-Vibration Isolation System**

Y	N	
	N	1. Will any part of the design create hazardous revolving, reciprocating, running, shearing, punching, pressing, squeezing, drawing, cutting, rolling, mixing or similar action, including pinch points and sheer points?
	N	2. Can any part of the design undergo high accelerations/decelerations?
	N	3. Will the system have any large moving masses or large forces?
	N	4. Will the system produce a projectile?
	N	5. Would it be possible for the system to fall under gravity creating injury?
	N	6. Will a user be exposed to overhanging weights as part of the design?
Y		7. Will the system have any sharp edges?
	N	8. Will any part of the electrical systems not be grounded?
	N	9. Will there be any large batteries or electrical voltage in the system above 40 V?
	N	10. Will there be any stored energy in the system such as batteries, flywheels, hanging weights or pressurized fluids?
	N	11. Will there be any explosive or flammable liquids, gases, or dust fuel as part of the system?
	N	12. Will the user of the design be required to exert any abnormal effort or physical posture during the use of the design?
	N	13. Will there be any materials known to be hazardous to humans involved in either the design or the manufacturing of the design?
	N	14. Can the system generate high levels of noise?
	N	15. Will the device/system be exposed to extreme environmental conditions such as fog, humidity, cold, high temperatures, etc?
	N	16. Is it possible for the system to be used in an unsafe manner?
	N	17. Will there be any other potential hazards not listed above? If yes, please explain on reverse.

For any "Y" responses, on the reverse side add:
(1) a complete description of the hazard,
(2) the corrective action(s) you plan to take to protect the user, and
(3) a date by which the planned actions will be completed.

PDR Design Hazard Checklist **F41-Vibration Isolation System**

Description of Hazard	Planned Corrective Action	Planned Date	Actual Date
Sharp Edges	After manufacturing and building has been completed, we will carefully inspect all of the edges on the part and will sand down any edges that are too sharp.	End of Winter Quarter	

M MATLAB Code

The following MATLAB programs were developed based on the work of Sodano et. al. and Ebrahimi et.al. to model the magnetic damping. The program force_calc.m calculates and plots damping ratio as a function of gap distance. It uses the function Br_Bz.m to calculate the magnetic flux produced by a permanent magnet in the radial and vertical directions. Using this model, we were not able to obtain results that matched our experimentation, so further refinement of the model is recommended.

Contents

House Keeping

=============

```
clear
close all
format short
clc
```

Parametric Values

```
L = 3/16*(2.54/100);      % thickness of cylindrical permanent magnet [m]
R = .5*(2.54/100);        % radius of cylindrical permanent magnet [m]
Brmax = 13000;            % max magnetic flux density [gauss]
dz = 0.00005;            % incriment used for evaluating gap between magnet and stage [m]
dr = 0.001;              % incriment used for integration across radius of stage [m]
dzd = 0.001;             % incriment used for integration across thickness of stage [m]
hD = 1*2.54/100;         % thickness of conductor [m]
sigma = 5.96*10^7;       % conductivity of copper [S/m]
m = 3.95;                % mass of stage [kg]
keq = 6*29.8;            % equivalent stiffness of springs [N/m]
disp_init = 0.0075;       % initial displacement of the stage [m]
JJ = 1;
```

Experimental Response

N52 | d = 1.5" | t = 1/4"

```
syms t_exp1
exper = -100*(3.5069)/1000*exp(-0.5.*t_exp1).*cos(1.08*2*pi.*t_exp1);
acc2vel = int(exper);
vel2pos = int(acc2vel);
t_exp = linspace(0,10,1000);
vel2pos = subs(vel2pos,t_exp1,t_exp);
```

Main Code

```
for ZF = linspace(L/2,(L/2+disp_init),50)
```

```
    % for each location, ZF, we want to get the acting magnetic force by
    % integration (summation with "r" then with "zd") over the thickness
    % of the disk "hD" because the magnetic force will be due to the eddy
    % current generated in the disk.

    sum_z = 0; % accumulator used to integrate across thickness of stage
    J = 1; % counter used to index force at various gap heights
    Zd_i = linspace(ZF,(ZF+hD),20); % [m] range and step for integration across thickness of
stage
    r_i = linspace(0.0000000001,R,20); % [m] range and step for integration across radius of
stage
    [Br,Bz] = Br_Bz(R,L,r_i,Zd_i,Brmax); % call function which calculates magnetic flux, Br_B
z.m

    for Zd = Zd_i
        sum_r = 0;
        I = 1;

        for r = r_i
            Bz2 = real(Bz(I,J+1));
            Bz1 = real(Bz(I,J));
            Br1 = real(Br(I,J));
            Fmag = -pi*sigma*(Bz2-Bz1)/dzd*Br1*r^2*dr*dzd;
            I = I + 1;
            sum_r = sum_r + Fmag;
        end

        sum_z = sum_z + sum_r;
    end

    F(JJ) = real(sum_z);
    Z(JJ) = real(ZF - L/2);
    JJ = JJ + 1;
end
```

Polynomial Curve Fit and ODE Solution

fit a second order polynomial to the theoretical plot developed for force vs distance.

```
a = polyfit(Z,F,4);

% set x to symbolic variable
syms z(t)

% second order, non-linear, ODE which desribes the motion of the stage
ode_2nd =   diff(z,2) == -(1/m)*(a(1)*z^4+a(2)*z^3+a(3)*z^2+a(4)*z^1+a(5))*diff(z) - ...
                         (1/m)*keq*z;

% Convert the second order diff eq to a system of first order diff eq
ode_1st =   odeToVectorField(ode_2nd);

% generate a MATLAB function handle from ode_1st by using matlabFunction
M = matlabFunction(ode_1st,'vars',{'t','Y'});

% set the options on ode45 solver
```

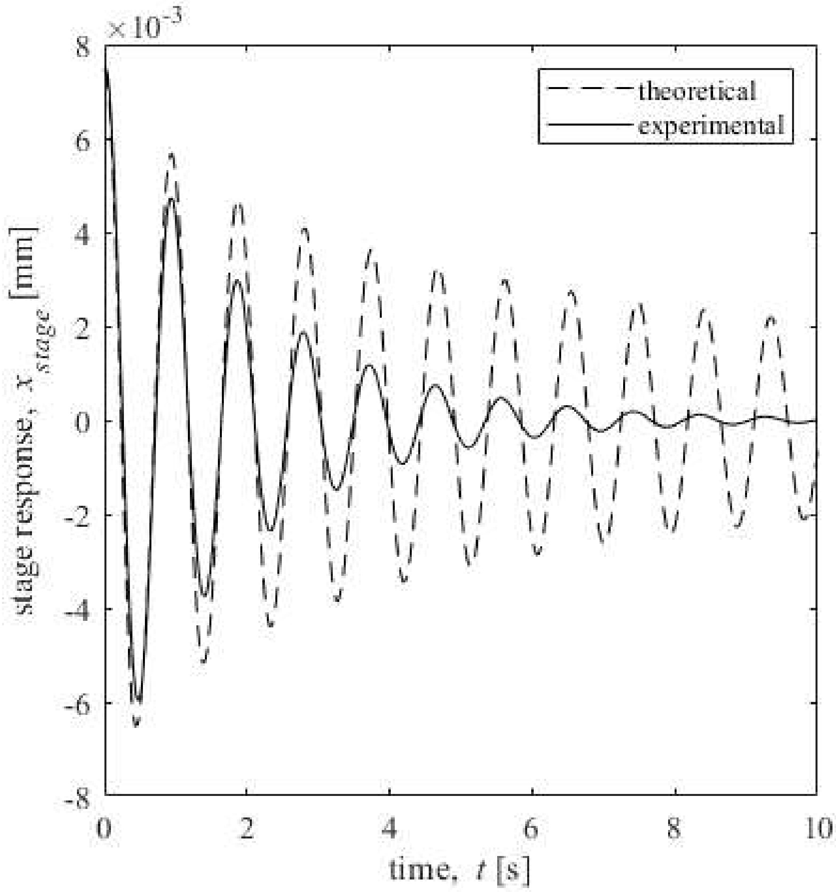
```
options = odeset('RelTol',1e-8,'AbsTol',1e-8);

% specifiy solution interval and initial conditions
sol = ode45(M,[0 10],[disp_init, 0], options);
```

```
Warning: Polynomial is badly conditioned. Add points with distinct X values,
reduce the degree of the polynomial, or try centering and scaling as described
in HELP POLYFIT.
```

Plot Experimental vs Theoretical Response

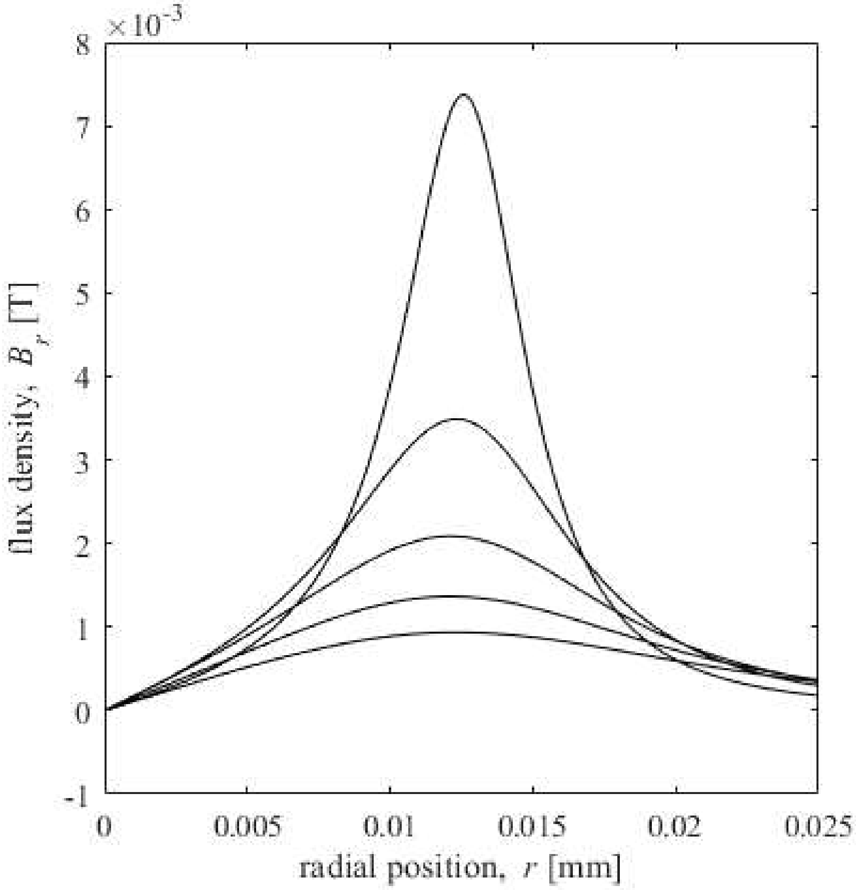
```
figurename = '1x_DX84_response'; % USER TO MODIFY
figure('Name',figurename,'NumberTitle','off',...
       'units','inches') % generate figure - set inch units for size
    pos = get(gcf,'pos'); % default MATLAB position of figure on screen
    set(gcf,'pos',[pos(1)/2 pos(2)/2 4.5 4.5]); % sets figure position and size
    set(gcf,'PaperPosition',[0 0 4.5 4.5]); % sets figure print size in inches
    set(gcf,'DefaultAxesFontName','Times','DefaultAxesFontSize',11);
    plot(sol.x,sol.y(1,:),'--k');
    hold on
    plot(t_exp, vel2pos,'-k');
    xlabel('time, {\it t} [s]');
    ylabel('stage response, {\it x_s_t_a_g_e} [mm]');
    legend('theoretical','experimental');
print(gcf,figurename,'-djpeg','-r600'); % resolution set to 600 dpi
```



Plot Flux Density Br Vs r

```
r_i = linspace(0.0000000001,.025,200);
z_i = linspace(0.005,0.015,5);
[Br,Bz] = Br_Bz(R,L,r_i,z_i,Brmax);

figurename = 'flux density'; % USER TO MODIFY
figure('Name',figurename,'NumberTitle','off',...
       'units','inches') % generate figure - set inch units for size
    pos = get(gcf,'pos'); % default MATLAB position of figure on screen
    set(gcf,'pos',[pos(1)/2 pos(2)/2 4.5 4.5]); % sets figure position and size
    set(gcf,'PaperPosition',[0 0 4.5 4.5]); % sets figure print size in inches
    set(gcf,'DefaultAxesFontName','Times','DefaultAxesFontSize',11);
    plot(r_i,Br,'-k');
    xlabel('radial position, {\it r} [mm]');
    ylabel('flux density, {\it B_r} [T]');
print(gcf,figurename,'-djpeg','-r600'); % resolution set to 600 dpi
```



$\times 10^{-3}$
8
7
6
5
4
3
2
1
0
-1
flux density, B_r [T]
0
0.005
0.01
0.015
0.02
0.025
radial position, r [mm]

```matlab
function [Br,Bz] = Br_Bz(R,L,ri,zi,Brmax)

% From Ebrahimi's publication "Design and Modeling of a magnetic shock
% absorber based on eddy current damping effect". Equation (2) from Craik's
% textbook is used to calculate the radial and longitudinal component of
% magnetic flux density for a cylindrical permanent magnent. This model
% equates the permanent magnet to a solenoid with only one turn.

% Inputs:
%   1. radius of the magnet, R [m]
%   2. thickness of the magnet, L [m]
%   3. radial points at which the flux is evaluated, ri [m]
%   4. longidudinal points at which the flux is evaluated, zi [m]
%   5. maximum flux density for magnet, Brmax [gauss] (typically found on
%   manufacturer website).
%
% Outputs:
%   1. radial component of flux density evaluated at ri and zi, Br [Tesla]
%   2. longidudinal component of flux density evaluated at ri and zi, Bz
%   [Tesla]

% Parametric Values
%  =================
u_o = 4*pi()*(10^(-7));          % permeability of free space [kg*(m/s^2)/A^2]
Brmax_T = Brmax*10^(-4);         % residual flux density [T]
M = Brmax_T/u_o;                 % magnetization [A/m]
I = M*L;                         % magnet equivalent current [A]
dzd = 0.0001;                    % step size for Br and Bz integral
i = 1;                           % counter for r index

for r = ri
    j = 1;
    for z = zi
        iB = 1;
        for zd = -L/2:dzd:L/2
            k_2 = (4*R*r)/((R+r)^2+(z-zd)^2);
            [KK,EK] = ellipke(k_2);
%           [KK,EK] = ellipt_int(R,r,z,zd,0.01);
            Bri(iB) = u_o*I/(2*pi*L)*((z-zd)/(r*sqrt((R+r)^2+(z-zd)^2)))*(-KK+((R^2+r^2+(z-zd
)^2)/((R-r)^2+(z-zd)^2))*EK)*dzd;
            Bzi(iB) = u_o*I/(2*pi*L)*(1/(((R+r)^2+(z-zd)^2)^(0.5)))*(KK+((R^2-r^2-(z-zd)^2)/(
(R-r)^2+(z-zd)^2))*EK)*dzd;
        end
        Br(i,j) = sum(Bri);
        Bz(i,j) = sum(Bzi);
        zz(j) = z;
        j = j + 1;
    end
    rr(i) = r;
    i = i + 1;
end

end
```

N Failure Mode & Effects Analysis (FMEA)

System / Function	Potential Failure Mode	Potential Effects of the Failure Mode	Severity	Potential Causes of the Failure Mode	Current Preventative Activities	Occurence	Current Detection Activities	Detection	Priority	Recommended Action(s)	Responsibility & Target Completion Date
VIS / Withstand bakeout conditions	a) Fracture of parts b) Explosion of parts	a) Damage the VIS b) Damage the STM	10	1) Materials used that cannot withstand high temperatures	1) Research material temperature limits	2	N/A	2	40	1) Research bakeout conditions	Seth Week 3 of Winter Quarter
VIS / UHV compatible materials	a) Outgassing	a) Insufficient vacuum b) Poor STM image quality	6	1) Non-UHV materials	1) LIGO vacuum compatible mat. list	3	N/A	7	126	1) Research materials that are UHV compatible	Ryan Week 3 of Winter Quarter
Housing / Integrate w/ vacuum chamber	a) VIS cannot fit	a) VIS has to be redesigned and remanufactured b) Cannot use VIS in vacuum chamber	8	1) Wrong dimensions	Design to chamber measurements	1	Check pics and SolidWorks dimensions	1	8	1) Double check all dimensions	Jason Week 3 of Winter Quarter
Permanent Magnetic Damping System / Damp vibrations	a.) Chamber base excitation is not damped to within acceptable range for STM b.) Magntic field interfers with STM measurement c.) Damping changes over time.	a) Poor STM imaging b) Doesn't meet requirement for maximum amplitude of vibration c) Doesn't meet requirement for time for vibrations to die out	5	1) Permanent magnets not proper strength 2) 3 bracket assembly changes height over time 3) Permanent magnets are not able to damp in stage rotational modes. 4) Permanent magnets are too close to STM 5) Permanent magnets are too far from copper stage	1) Permanent magnet magnetic field simulation, which will be used to analyze magnetic damping 2) Tight tolerances on adjustability mechanism	4	1) Quality of STM samples are poor 2) Visably notice height of permanent magnets has changed over time	2	40	1) Preliminary testing in the vibrations lab	Ryan By first or second week of Winter Quarter
Housing / Isolate Vibrations	a) Natural frequency too low b) Insufficient damping between caps and housing	a) Microscope image loses clarity b) Vibrations take too long to die out c) Vibration peaks are not reduced	5	1) Windows too large 2) Housing walls too thin 3) Viton gaskets too thin 4) Viton gaskets too weak	FEA of housing to determine natural frequency	4	Testing of housing on shake table to verify FEA	4	80	1) Abaqus FEA analysis to find simulated natural frequency	Seth Week 2 of Winter Quarter
Mass Spring System / Isolate Vibrations	a) Natural frequency too high	Microscope image loses clarity	5	1) Springs too stiff 2) Mass too heavy	Analysis in MATLAB to determine necessary pring constants and stage mass	5	Testing of system on shake table to verify modeling assumptions	4	100	Preliminary testing of angled springs to determine magnitude of non-linear effects	Stephen Week 3 of Winter Quarter
Mass Spring System / Hold Microscope	a) Set screws don't allow for enough adjustment b) STM fixture comes loose from stage c) Set Screws come loose d) Springs break	a) Microscope image loses clarity b) Housing is damaged c) STM is damaged	7	1) Set screws too short 2) Springs too weak 3) Screws not tightened properly	1) MATLAB analysis to determine necessarry strength and spring constant of springs 2) Determine necessary adjustment range prior ro purchasing springs	2	1) Check screws to ensure tightened properly 2) Place level on stage	1	14	1) Matlab analysis 2) Mass-spring system test	Stephen Week 3 of Winter Quarter
Housing / Facilitate Sample Manipulation and Visibility	a) Can't access STM from outside the vacuum chamber b) Too heavy to remove from vacuum chamber c) Sharp edges on chamber	a) user unable to align microscope tip with sample visually b) multiple users required to lift c) user is injured	9	1) Housing windows are too small 2) Housing windows not aligned with vacuum chamber windows/door	1) SolidWorks CAD model to ensure correct dimensions for windows on housing	2	1) CAD model for housing	1	18	1) Measure actual dimensions of vacuum chamber 2) Double check dimensions of CAD model	Jason Week 2 of Winter Quarter
Housing / Damp vibrations	a) Housing vibrations not attenuated to an acceptable range	a) Vibrations not damped in an appropriate time	5	1) Viton gaskets sized improperly 2) Viton not compressed enough 3) Wrong type of Viton chosen	1) Talk to sponsors about design decisions 2) Check every material choice on trusted website	5	1) Poor image quality	5	125	1) Preliminary testing in the vibrations lab	Seth Week 3 of Winter Quarter

O Risk Assessment

The following report was generated using the DesignSafe software provided through the Mechanical Engineering department. As can be seen in the report the risks are generally low and centered around sharp edges created in the manufacturing process. As a result, care was taken to remove any sharp edges that ocurred at each step throughout the manufacturing process.

designsafe Report

Application: Vibration Isolation
Description:
Product Identifier:
Assessment Type: Detailed
Limits:
Sources:
Risk Scoring System: ANSI B11.0 (TR3) Two Factor

Analyst Name(s):
Company:
Facility Location:

Guide sentence: When doing [task], the [user] could be injured by the [hazard] due to the [failure mode].

Item Id	User / Task	Hazard / Failure Mode	Initial Assessment Severity Probability	Risk Level	Risk Reduction Methods /Control System	Final Assessment Severity Probability	Risk Level	Status / Responsible /Comments /Reference
1-1	All Users Common Tasks	<None>						
1-2	All Users normal operation	<None>						
2-1	operator Common tasks	<None>						
2-2	operator normal operation	<None>						
2-3	operator basic trouble shooting / problem solving	<None>						
2-4-1	operator load / unload materials	mechanical : cutting / severing sharp edges on housing windows	Moderate Unlikely	Low	remove sharp edges with file/deburring tool /Not Applicable	Moderate Remote	Negligible	In-process Stephen
2-4-2	operator load / unload materials	mechanical : pinch point hand caught between housing and stage	Minor Unlikely	Negligible	provide warnings against putting fingers in pinch points /Not Applicable	Minor Unlikely	Negligible	TBD Jason

Item Id	User / Task	Hazard / Failure Mode	Initial Assessment Severity Probability	Risk Level	Risk Reduction Methods /Control System	Final Assessment Severity Probability	Risk Level	Status / Responsible /Comments /Reference
3-1	maintenance technician Common Tasks	<None>						
3-2-1	maintenance technician set-up or changeover	mechanical : crushing heavy stage/VIS	Moderate Unlikely	Low	provide tool for handling/maintenance of stage /Not Applicable	Moderate Unlikely	Low	TBD Ryan
3-2-2	maintenance technician set-up or changeover	mechanical : cutting / severing sharp edges on machined parts	Moderate Unlikely	Low	remove sharp edges with file/deburring tool /Not Applicable	Moderate Remote	Negligible	In-process Seth
3-2-3	maintenance technician set-up or changeover	mechanical : pinch point between housing and stage, stage and vacuum chamber	Minor Unlikely	Negligible	provide warnings against putting fingers in pinch points /Not Applicable	Minor Unlikely	Negligible	TBD Stephen
3-3-1	maintenance technician parts replacement	mechanical : crushing strong permanent magnets	Moderate Likely	Medium	provide warnings for handling of magnets /Not Applicable	Moderate Unlikely	Low	TBD Jason
3-3-2	maintenance technician parts replacement	mechanical : pinch point pinching between housing and stage	Minor Unlikely	Negligible	provide warnings against putting fingers in pinch points /Not Applicable	Minor Unlikely	Negligible	TBD Ryan
3-4-1	maintenance technician adjust controls / settings / alignment	mechanical : pinch point between adjustment screws and swash plate	Moderate Remote	Negligible	provide warnings against putting fingers in pinch points /Not Applicable	Moderate Remote	Negligible	TBD Seth
3-5-1	maintenance technician periodic maintenance	mechanical : crushing under heavy stage	Moderate Unlikely	Low	provide tool for handling/maintenance of stage /Not Applicable	Moderate Unlikely	Low	TBD Stephen
3-6-1	maintenance technician trouble-shooting / problem solving	mechanical : pinch point between stage and housing	Minor Unlikely	Negligible	provide warnings against putting fingers in pinch points /Not Applicable	Minor Unlikely	Negligible	TBD Jason

Item Id	User / Task	Hazard / Failure Mode	Initial Assessment Severity Probability	Risk Level	Risk Reduction Methods /Control System	Final Assessment Severity Probability	Risk Level	Status / Responsible /Comments /Reference
4-1	passer by / non-user work next to / near machinery	<None>						
4-2	passer by / non-user walk near machinery	<None>						

P User Guide

I. ASSEMBLY INSTRUCTIONS

1. Attach STM to copper stage using the screw provided with the STM fixture.
2. Connect the large stud anchors to the six threaded holes around the edge of the copper stage.
3. Connect the small stud anchors to the six threaded holes around the edge of the top cap.
4. Connect the copper stage to the top cap by threading the springs through the holes in the stud anchors in the angled formation shown.
5. Place the top cap on the rim of the aluminum housing with the Viton gasket in between. Screw the #4-40 vented screws into the eight holes around the edge of the top cap into the aluminum cylinder.
6. Place the 3 magnets into the corresponding holes in the magnetic swash plate.
7. Screw the long adjustment screws through the swash plate into the bottom cap.
8. Screw the short adjustment screws into the bottom cap through the bottom so they lift the magnetic swash plate up until it stops against the shoulder of the long adjustment screws.
9. Place the bottom cap on the rim of the aluminum housing with the Viton gasket in between. Screw the #4-40 vented screws into the eight holes around the edge of the bottom cap into the aluminum cylinder.

II. CALIBRATION INSTRUCTIONS

1. Begin with the magnetic swash plate as high as possible.
2. Place an accelerometer on the copper stage, as near to the center as possible.
3. Connect the accelerometer to an oscilloscope.
4. Displace the copper stage some distance and record the response.
5. Calculate the damping ratio using the measured damped natural frequency and the undamped natural frequency of 1.14 Hz or by using the log decrement method.
6. Lower the stage by loosening the short adjustment screws by a quarter turn.
7. Repeat until the desired damping ratio is reached. A damping ratio of 0.2 is desirable.

III. INSERTING/REMOVING VIS FROM VACUUM CHAMBER

1. Remove the top cap from the vacuum chamber.
2. Holding VIS from the hole in the top cap, center over the chamber and lower into place.
3. WARNING! Be wary of pinching hazards when lowering the VIS into the vacuum chamber. Do not lower with fingers through the windows, as this can severely injure your fingers!
4. Reattach top cap of vacuum chamber.
5. To remove from vacuum chamber, reverse these steps.

Q Design Verification Plan & Report (DVP&R)

The following Design Verification Plan & Report outlines our planned testing for each specification, along with measurements to be taken, expected and allowable results. For completed tests, results and notes are also shown. All critical specifications were verified, however some tests were not completed due to time constraints and restrictions to lab access resulting from the COVID-19 Pandemic. For detailed procedures for each of the listed tests, see Appendix R.

DVP&R - Design Verification Plan (& Report)

Project:	Vibration Isolation System for Scanning Tunneling Microscope	Sponsor:	Dr. Gregory Scott, Toan Le, Dr. Eltahry Elghandour	Edit Date:	4/28/2021

Test #	Specification	Test Description	Measurements	Acceptance Criteria	Required Facilities/Equipment	Parts Needed	Responsibility	TIMING Start date	TIMING Finish date	Numerical Results	Notes on Testing
	TEST PLAN									**TEST RESULTS**	
1	1 Natural frequency of housing structure and housing assembly.	Mount housing on shaker table with accelerometers attached.	First five modes of natural frequency in Hz.	First mode > 400 Hz	Vertical shaker table, vibrations laboratory, accelerometers.	Housing chamber, viton gaskets, top cap, bottom cap	Seth	2/21/2021	4/15/2021	First natural frequency around 500 Hz, which matches with the FEA model.	Testing looks good.
2	2 Natural frequency of mass-spring system.	Displace mass-spring system and measure the natural frequency using accelerometers.	First five modes of natural frequency in Hz.	First mode < 2 Hz	Vibrations laboratory, vertical shaker table, accelerometers.	Housing structure (or wood frame), springs, copper stage	Jason	2/15/2021	2/15/2021	Natural frequency of 1.14 Hz.	This was preliminary testing and needs to be corroborated with mulitple trials.
3	3, 6 Magnetic damping ratio and time for vibrations in stage to be damped.	Displace stage until it touches magnets and measure response using accelerometers.	Step response and damping ratio (from calculation).	0.15	Vibrations laboratory, accelerometers.	Magnets, copper stage, springs, wooden frame	Ryan	2/3/2021	3/10/2021	Damping ratio of different size magents, at different radial positions and different heights was tested. Data in spreadsheet.	Redoing some of the measurements for greater accuracy. This will be done using a better oscilloscope and exporting data in order to get the most accurate numbers.
4	3, 6 Magnetic damping ratio and time for vibrations in stage to be damped.	Shake spring and magnet assembly on shaker table with accelerometers attached.	Step response and damping ratio (from calculation).	0.15	Vibrations laboratory, vertical shaker table, accelerometers.	Magnets, copper stage, springs, housing structure (or wooden frame).	Stephen	5/10/2021	6/2/2021	Damping ratio of 0.09-0.3 could be obtained. Final damping ratio was calibrated to 0.2	A wide range of damping ratios are usable. May need further tuning when used with the STM for the first time.
5	4 Damping coefficient of vistoelastic materials.	Shake housing structure with top and bottom cap and viton gaskets in between. Compare to when shaken without the viton gaskets.	Damping coefficient and step response of system.	50-400 Ns/m	Vibrations laboratory, vertical shaker table, accelerometers.	Top cap, bottom cap, viton gaskets (2), housing structure.	Seth	5/15/2021	Not completed	-	-
6	5 Vibration amplitude seen by copper stage.	Shake entire housing structure and mass-spring system on shaker table.	Vibration amplitude on top of copper stage.	2 mm	Vibrations laboratory, vertical shaker table, accelerometers.	Top cap, bottom cap, viton gaskets (2), housing structure, springs, copper stage, magnets, magnet swash plate, screws.	Jason	5/20/2021	Not completed	-	-
7	9 Sample swap test.	Qualitative evaluation of how easy it is to swap samples.	User Feedback (Google survey).	Rating 1-5	User survey.	Entire assembly, fake sample.	Ryan	5/15/2021	Not completed	-	-
8	10 VIS swap time from vacuum chamber.	Timer set to remove VIS from vacuum chamber and replace it back in.	Time in minutes.	30 min	Chemistry department, vacuum chamber, timer.	Entire assembly, vacuum chamber.	Stephen	5/15/2021	Not completed	-	-
9	13 Consistent atomic measurements.	Mount VIS in vacuum chamber with live sample and take pictures.	Output pictures.	Atomic quality.	Chemistry department, vacuum chamber, STM.	Entire VIS assembly, sample, STM, vacuum chamber.	Seth	5/15/2021	Not completed	-	-

Print Date:

R Test Procedures

The following procedures were designed for all of the tests outlined in the DVP&R. Not all of the outlined tests were completed. For a full summary of the completed tests and results, see Appendix Q.

1. Housing Natural Frequency
2. Mass-Spring System Natural Frequency
3. Magnetic Damping Ratio
4. Viscoelastic Damping Coefficient
5. Copper Stage Vibration Amplitude
6. Sample Swap Test
7. VIS Install Time Test
8. Atomic Measurements Test

Test Procedure for Housing Natural Frequency

Test Name: Housing Natural Frequency

Purpose: To determine the natural frequency of the completed assembly.

Scope: We will verify the natural frequency of the completed assembly to ensure that incoming vibrations will be in the isolation region of our system.

Equipment: Housing chamber, viton gaskets, top cap, bottom cap, vertical shake table, shake table test plate and hardware, connected computer for data collection, accelerometer

Hazards: Vibrating objects at high frequencies/low amplitudes, compressed air, high voltage, small chance of flying objects, damage to shake table.

PPE Requirements: Safety glasses, masks, face shields (if in close proximity for >15 mins)

Facility: Vibrations lab

Procedure:

1. Attach aluminum housing to adapter on top of slip table (12x14" aluminum plate).
2. Attach aluminum plate adapter to slip table (if not already connected).
3. Attach accelerometer to slip table.
    a. Record accelerometer serial number and calibration constant.
    b. Attach BNC cable labeled "1" (located near exhaust fan switch) to channel 1 of power supply.
    c. Attach accelerometer cable to power supply in the channel 1 socket.
    d. Turn on and set knob to channel 1.
4. Attach accelerometer to top of aluminum housing.
    a. Record accelerometer serial number and calibration constant.
    b. Attach BNC cable labeled "2" (located near exhaust fan switch) to channel 2 of power supply.
    c. Attach accelerometer cable to power supply in the channel 2 socket.
    d. Turn on and set knob to channel 2.
5. Turn on slip table.
6. Open EDM-VibrationController on Desktop.
    a. **Note:** Be sure to press the black power button before you open the EDM-VibrationController (otherwise the system will not be able to connect to the slip table)
    b. **Note:** It may take a few minutes for the system to come online.
    c. **Note:** If the system is still not connected after a minute, close the EDM-VibrationController program and reopen it. The system should be displayed as connected then.
7. Open a new test.
    a. Select "Swept Sine" on the menu on the left.
    b. Name the file, give a description and select "Finish".
8. Click on the "Setup" on the menu on the top of the screen.
    a. Select "Engineering Units", and "Custom Engineering Units".

    b. Select “Set default English units”.
    c. Select “OK”.
9. Click on the purple button labeled “Config” on the right side of the screen.
    a. Select “Test Parameters” from the menu on the left.
        i. Set the sweep type to “logarithmic” and set sweep speed to “oct/min”.
    b. Select “Test Profile” from the menu on the left.
        i. Set the acceleration for the second and third point on the profile.
        ii. Set the “Displacement in (pk-pk)” for the first and second point on the profile.
        iii. Ensure that the frequency of the third point of the profile is at the final desired frequency, or higher...
    c. Select “Test Against Shaker”.
        i. Check the “Profile/shaker Limits” column of the table to ensure the Profile does not exceed the shaker limits (percentage should not exceed 100%).
    d. Select “Run Schedule” from the menu on the left.
        i. Double click on the test entry.
        ii. Edit the Sweep Entry Parameters.
            1. Set “Left Frequency (Hz)”.
            2. Set “Right Frequency (Hz)”.
            3. Set “Initial Sweep Direction”.
                a. Sweeping up will cause the sine sweep to move from the Start Frequency towards the Right Frequency. Sweeping down will cause the frequency to head towards the Left Frequency.
            4. Set the “Sweep Speed”.
        iii. Select “OK”.
    e. Select “Limit Channels” from the menu on the left.
        i. Ensure that the boxes corresponding to Channel 1 and Channel 2 under “Enabled” are checked.
        ii. Select the “Edit” button for Channel 2.
            1. Uncheck the boxes for all limits except for the abort limit.
            2. Set the acceleration for the abort limits to safest maximum acceleration the system can handle.
        iii. Repeat this process for Channel 1.
    f. Select “OK”.
10. Select the “Setup” drop-down menu on the top.

Results: Natural frequency of the housing assembly, target for first vertical mode >400 Hz,

Test Date(s):

Test Results:

Performed By:

Test Procedure for Mass-Spring System Natural Frequency

Test Name: Mass-Spring System Natural Frequency

Purpose: To determine the natural frequency of the mass-spring system

Scope: We will verify the natural frequency of the completed assembly to ensure that incoming vibrations will be in the isolation region of our system.

Equipment: Housing chamber, viton gaskets, top cap, bottom cap, springs, stud anchors, copper stage. vertical shake table, shake table test plate and hardware, connected computer for data collection, accelerometer

Hazards: Vibrating objects at high frequencies/low amplitudes, compressed air, high voltage, small chance of flying objects, damage to shake table.

PPE Requirements: Safety glasses, masks, face shields (if in close proximity for >15 mins)

Facility: Vibrations lab

Procedure:

1. Attach aluminum housing to adapter on top of slip table (12x14” aluminum plate).
2. Attach aluminum plate adapter to slip table (if not already connected).
3. Attach accelerometer to slip table.
    a. Record accelerometer serial number and calibration constant.
    b. Attach BNC cable labeled “1” (located near exhaust fan switch) to channel 1 of power supply.
    c. Attach accelerometer cable to power supply in the channel 1 socket.
    d. Turn on and set knob to channel 1.
4. Attach accelerometer to top of copper stage.
    a. Record accelerometer serial number and calibration constant.
    b. Attach BNC cable labeled “2” (located near exhaust fan switch) to channel 2 of power supply.
    c. Attach accelerometer cable to power supply in the channel 2 socket.
    d. Turn on and set knob to channel 2.
5. Turn on slip table.
6. Open EDM-VibrationController on Desktop.
    a. **Note:** Be sure to press the black power button before you open the EDM-VibrationController (otherwise the system will not be able to connect to the slip table)
    b. **Note:** It may take a few minutes for the system to come online.
    c. **Note:** If the system is still not connected after a minute, close the EDM-VibrationController program and reopen it. The system should be displayed as connected then.
7. Open a new test.
    a. Select “Swept Sine” on the menu on the left.
    b. Name the file, give a description and select “Finish”.
8. Click on the “Setup” on the menu on the top of the screen.

a. Select “Engineering Units”, and “Custom Engineering Units”.
b. Select “Set default English units”.
c. Select “OK”.

9. Click on the purple button labeled “Config” on the right side of the screen.
    a. Select “Test Parameters” from the menu on the left.
        i. Set the sweep type to “logarithmic” and set sweep speed to “oct/min”.
    b. Select “Test Profile” from the menu on the left.
        i. Set the acceleration for the second and third point on the profile.
        ii. Set the “Displacement in (pk-pk)” for the first and second point on the profile.
        iii. Ensure that the frequency of the third point of the profile is at the final desired frequency, or higher...
    c. Select “Test Against Shaker”.
        i. Check the “Profile/shaker Limits” column of the table to ensure the Profile does not exceed the shaker limits (percentage should not exceed 100%).
    d. Select “Run Schedule” from the menu on the left.
        i. Double click on the test entry.
        ii. Edit the Sweep Entry Parameters.
            1. Set “Left Frequency (Hz)”.
            2. Set “Right Frequency (Hz)”.
            3. Set “Initial Sweep Direction”.
                a. Sweeping up will cause the sine sweep to move from the Start Frequency towards the Right Frequency. Sweeping down will cause the frequency to head towards the Left Frequency.
            4. Set the “Sweep Speed”.
        iii. Select “OK”.
    e. Select “Limit Channels” from the menu on the left.
        i. Ensure that the boxes corresponding to Channel 1 and Channel 2 under “Enabled” are checked.
        ii. Select the “Edit” button for Channel 2.
            1. Uncheck the boxes for all limits except for the abort limit.
            2. Set the acceleration for the abort limits to safest maximum acceleration the system can handle.
        iii. Repeat this process for Channel 1.
    f. Select “OK”.
10. Select the “Setup” drop-down menu on the top.

Results: Natural frequency of the copper stage w.r.t shaker table, target natural frequency <2 Hz

Test Date(s):

Test Results:

Performed By:

Test Procedure for Magnetic Damping Ratio

Test Name: Magnetic Damping Ratio

Purpose: To examine the damping coefficient of the final mass-spring system.

Scope: We will test our magnetic damping mechanism once the design is finalized in order to verify the damping coefficient achieved.

Equipment: Completed mass-spring assembly, stud anchors, springs, copper stage, magnets, accelerometer, load cell, oscilloscope, data collecting unit, ruler, caliper, printer.

Hazards: Small chance of pinching under copper stage, between strong magnets. Small chance of cuts on magnet edges if broken.

PPE Requirements: Masks, face shields (if in close proximity for >15 mins)

Facility: Vibes lab.

Procedure:

1. Mount copper stage to top cap using 6 springs.
2. Mount accelerometer and load cell onto copper stage and connect to oscilloscope.
3. Place magnets into magnetic swash plate.
4. Assemble into housing.
5. Displace stage to the top of 3D holder and measure response using oscilloscope.
6. Repeat for a total of five runs.
7. Repeat steps 5-6 at varying swash plate heights

Results: Accelerometer calibration constant, Baseline voltage, Undamped Natural Frequency, Damped natural frequency, Peak voltage of the first five damped oscillations for log decrement method.

Test Date(s):

Test Results:

Performed By:

Test Procedure for Viscoelastic Damping Coefficient

Test Name: Viscoelastic Damping Coefficient

Purpose: To determine the damping coefficient of the viton gaskets in the housing assembly.

Scope: We will verify the damping coefficient of varying thicknesses of viton gaskets.

Equipment: Housing chamber, viton gaskets, top cap, bottom cap, vertical shake table, shake table test plate and hardware, connected computer for data collection, accelerometer

Hazards: Vibrating objects at high frequencies/low amplitudes, compressed air, high voltage, small chance of flying objects, damage to shake table.

PPE Requirements: Safety glasses, masks, face shields (if in close proximity for >15 mins)

Facility: Vibrations lab

Procedure:

1. Attach aluminum housing to adapter on top of slip table (12x14" aluminum plate).
2. Attach aluminum plate adapter to slip table (if not already connected).
3. Attach accelerometer to slip table.
    a. Record accelerometer serial number and calibration constant.
    b. Attach BNC cable labeled "1" (located near exhaust fan switch) to channel 1 of power supply.
    c. Attach accelerometer cable to power supply in the channel 1 socket.
    d. Turn on and set knob to channel 1.
4. Attach accelerometer to top of aluminum housing.
    a. Record accelerometer serial number and calibration constant.
    b. Attach BNC cable labeled "2" (located near exhaust fan switch) to channel 2 of power supply.
    c. Attach accelerometer cable to power supply in the channel 2 socket.
    d. Turn on and set knob to channel 2.
5. Turn on slip table.
6. Open EDM-VibrationController on Desktop.
    a. **Note:** Be sure to press the black power button before you open the EDM-VibrationController (otherwise the system will not be able to connect to the slip table)
    b. **Note:** It may take a few minutes for the system to come online.
    c. **Note:** If the system is still not connected after a minute, close the EDM-VibrationController program and reopen it. The system should be displayed as connected then.
7. Open a new test.
    a. Select "Swept Sine" on the menu on the left.
    b. Name the file, give a description and select "Finish".
8. Click on the "Setup" on the menu on the top of the screen.
    a. Select "Engineering Units", and "Custom Engineering Units".
    b. Select "Set default English units".

c. Select “OK”.

9. Click on the purple button labeled “Config” on the right side of the screen.
    a. Select “Test Parameters” from the menu on the left.
        i. Set the sweep type to “logarithmic” and set sweep speed to “oct/min”.
    b. Select “Test Profile” from the menu on the left.
        i. Set the acceleration for the second and third point on the profile.
        ii. Set the “Displacement in (pk-pk)” for the first and second point on the profile.
        iii. Ensure that the frequency of the third point of the profile is at the final desired frequency, or higher...
    c. Select “Test Against Shaker”.
        i. Check the “Profile/shaker Limits” column of the table to ensure the Profile does not exceed the shaker limits (percentage should not exceed 100%).
    d. Select “Run Schedule” from the menu on the left.
        i. Double click on the test entry.
        ii. Edit the Sweep Entry Parameters.
            1. Set “Left Frequency (Hz)”.
            2. Set “Right Frequency (Hz)”.
            3. Set “Initial Sweep Direction”.
                a. Sweeping up will cause the sine sweep to move from the Start Frequency towards the Right Frequency. Sweeping down will cause the frequency to head towards the Left Frequency.
            4. Set the “Sweep Speed”.
        iii. Select “OK”.
    e. Select “Limit Channels” from the menu on the left.
        i. Ensure that the boxes corresponding to Channel 1 and Channel 2 under “Enabled” are checked.
        ii. Select the “Edit” button for Channel 2.
            1. Uncheck the boxes for all limits except for the abort limit.
            2. Set the acceleration for the abort limits to safest maximum acceleration the system can handle.
        iii. Repeat this process for Channel 1.
    f. Select “OK”.
10. Select the “Setup” drop-down menu on the top.
11. Repeat test with gasket thicknesses of 1/16, 1/8, and 3/16 inches.

Results: Damping coefficient of viton gaskets, target range 50-400 Ns/m

Test Date(s):

Test Results:

Performed By:

Test Procedure for Copper Stage Vibration Amplitude

Test Name: Copper Stage Vibration Amplitude

Purpose: To determine the maximum vibration amplitude seen by the copper stage.

Scope: We will subject the entire assemble to a random noise test and measure the maximum amplitude seen by the copper stage.

Equipment: Housing chamber, viton gaskets, top cap, bottom cap, springs, stud anchors, copper stage. vertical shake table, shake table test plate and hardware, connected computer for data collection, accelerometer

Hazards: Vibrating objects at high frequencies/low amplitudes, compressed air, high voltage, small chance of flying objects, damage to shake table.

PPE Requirements: Safety glasses, masks, face shields (if in close proximity for >15 mins)

Facility: Vibrations lab

Procedure:

1. Attach aluminum housing to adapter on top of slip table (12x14" aluminum plate).
2. Attach aluminum plate adapter to slip table (if not already connected).
3. Attach accelerometer to slip table.
    a. Record accelerometer serial number and calibration constant.
    b. Attach BNC cable labeled "1" (located near exhaust fan switch) to channel 1 of power supply.
    c. Attach accelerometer cable to power supply in the channel 1 socket.
    d. Turn on and set knob to channel 1.
4. Attach accelerometer to top of copper stage.
    a. Record accelerometer serial number and calibration constant.
    b. Attach BNC cable labeled "2" (located near exhaust fan switch) to channel 2 of power supply.
    c. Attach accelerometer cable to power supply in the channel 2 socket.
    d. Turn on and set knob to channel 2.
5. Turn on slip table.
6. Open EDM-VibrationController on Desktop.
    a. **Note:** Be sure to press the black power button before you open the EDM-VibrationController (otherwise the system will not be able to connect to the slip table)
    b. **Note:** It may take a few minutes for the system to come online.
    c. **Note:** If the system is still not connected after a minute, close the EDM-VibrationController program and reopen it. The system should be displayed as connected then.
7. Open a new test.
    a. Select "Swept Sine" on the menu on the left.
    b. Name the file, give a description and select "Finish".
8. Click on the "Setup" on the menu on the top of the screen.

a. Select “Engineering Units”, and “Custom Engineering Units”.
b. Select “Set default English units”.
c. Select “OK”.

9. Click on the purple button labeled “Config” on the right side of the screen.
    a. Select “Test Parameters” from the menu on the left.
        i. Set the sweep type to “logarithmic” and set sweep speed to “oct/min”.
    b. Select “Test Profile” from the menu on the left.
        i. Set the acceleration for the second and third point on the profile.
        ii. Set the “Displacement in (pk-pk)” for the first and second point on the profile.
        iii. Ensure that the frequency of the third point of the profile is at the final desired frequency, or higher...
    c. Select “Test Against Shaker”.
        i. Check the “Profile/shaker Limits” column of the table to ensure the Profile does not exceed the shaker limits (percentage should not exceed 100%).
    d. Select “Run Schedule” from the menu on the left.
        i. Double click on the test entry.
        ii. Edit the Sweep Entry Parameters.
            1. Set “Left Frequency (Hz)”.
            2. Set “Right Frequency (Hz)”.
            3. Set “Initial Sweep Direction”.
                a. Sweeping up will cause the sine sweep to move from the Start Frequency towards the Right Frequency. Sweeping down will cause the frequency to head towards the Left Frequency.
            4. Set the “Sweep Speed”.
        iii. Select “OK”.
    e. Select “Limit Channels” from the menu on the left.
        i. Ensure that the boxes corresponding to Channel 1 and Channel 2 under “Enabled” are checked.
        ii. Select the “Edit” button for Channel 2.
            1. Uncheck the boxes for all limits except for the abort limit.
            2. Set the acceleration for the abort limits to safest maximum acceleration the system can handle.
        iii. Repeat this process for Channel 1.
    f. Select “OK”.
10. Select the “Setup” drop-down menu on the top.

Results: Maximum amplitude below 2 mm.

Test Date(s):

Test Results:

Performed By:

Test Procedure for Sample Swap Test

Test Name: Sample Swap Test

Purpose: To verify ease of swapping samples in the STM while VIS is installed in vacuum chamber.

Scope: We will ask Dr. Scott and several of his students to swap a sample from the VIS while in the vacuum chamber to test how easily samples can be swapped. Participant will evaluate various criteria on a scale form 1-5.

Equipment: VIS assembly, fake sample, vacuum chamber.

Hazards: Small chance of cuts from sharp edges on housing/vacuum chamber.

PPE Requirements: Masks, face shields (if in close proximity for >15 mins)

Facility: Chemistry lab.

Procedure:

1. Insert fake sample into VIS.
2. Install VIS in vacuum chamber.
3. Have test subjects remove and replace fake sample with VIS in vacuum chamber. Test subjects will be given freedom to install sample how they ordinarily would, as long as deemed safe by Dr. Scott.
4. Have test subjects fill out a survey about the experience.

Results: User feedback in various criteria, rated 1 (worst) to 5 (best).

Test Date(s):

Test Results:

Performed By:

Test Procedure for VIS Install Time Test

Test Name: VIS Install Time Test

Purpose: To verify VIS can be installed in vacuum chamber in a reasonable amount of time.

Scope: We will time Dr. Scott installing our VIS into the vacuum chamber.

Equipment: VIS assembly, vacuum chamber, stopwatch.

Hazards: Small chance of cuts from sharp edges on housing/vacuum chamber.

PPE Requirements: Masks, face shields (if in close proximity for >15 mins)

Facility: Chemistry lab.

Procedure:

1. Start stopwatch.
2. Have Dr. Scott install VIS into the vacuum chamber using his standard procedures or best practices.
3. Record time once completed.

Results: VIS should not take longer than 15 minutes to install.

Test Date(s):

Test Results:

Performed By:

Test Procedure for Atomic Measurements Test

Test Name: Atomic Measurements Test

Purpose: To verify VIS allows consistent atomic measurements with STM

Scope: This is the final test of our intended goal. Dependent on whether Dr. Scott has the STM ready to take atomic measurements before our project deadline, but would be the ultimate validation that our VIS has achieved its purpose.

Equipment: VIS assembly, vacuum chamber, STM, sample.

Hazards: Small chance of cuts from sharp edges on housing/vacuum chamber.

PPE Requirements: Masks, face shields (if in close proximity for >15 mins)

Facility: Chemistry lab.

Procedure:

1. Install VIS in vacuum chamber with STM inside.
2. Place sample in STM.
3. Take 5 pictures while varying the ambient conditions as much as possible (stomping around, running equipment, etc.).

Results: Output pictures. For a pass, at least 3 of the 5 pictures should obtain atomic level measurements.

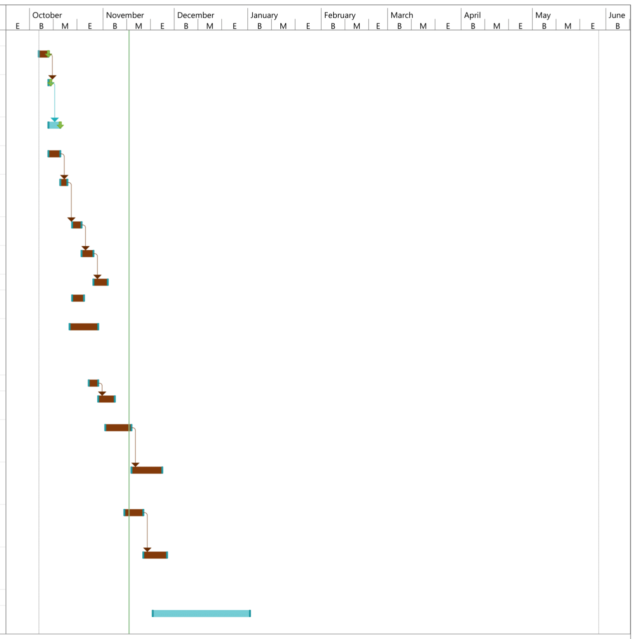
Test Date(s):

Test Results:

Performed By:

S Gantt Chart

ID	Task Mode	Task Name	Duration	Start	Finish	Predecessors	Responsibility
1		**Fall Quarter**					
2		Scope of Work (SOW) - Rough Draft	4 days	Mon 10/5/20	Thu 10/8/20		Ryan
3		Scope of Work (SOW) - Peer Review	1 day	Fri 10/9/20	Fri 10/9/20	2	Seth
4		Scope of Work (SOW) - Final Draft	3 days	Fri 10/9/20	Tue 10/13/20	3	Stephen
5		Prototype Functions - Ideation	3 days	Fri 10/9/20	Tue 10/13/20		Jason
6		Functional Decomposition Matrix	3 days	Wed 10/14/20	Fri 10/16/20	5	Seth
7		Build Ideation Models	4 days	Mon 10/19/20	Thu 10/22/20	6	Ryan
8		Decision Matrix for Design Choices	3 days	Fri 10/23/20	Tue 10/27/20	7	Ryan
9		Concept Selection	4 days	Wed 10/28/20	Mon 11/2/20	8	Jason
10		ANSYS FEA Analysis - Viton	5 days	Mon 10/19/20	Fri 10/23/20		Stephen
11		MATLAB & Simulink Analysis (Mass-Spring System)	10 days	Sun 10/18/20	Thu 10/29/20		Jason
12		Concept CAD	4 days	Mon 10/26/20	Thu 10/29/20		Seth
13		Concept Prototypes & Models	5 days	Fri 10/30/20	Thu 11/5/20	12	Ryan
14		Preliminary Design Review (PDR Report)	9 days	Mon 11/2/20	Thu 11/12/20		Jason
15		Preliminary Design Review (PDR Presentation)	9 days	Fri 11/13/20	Wed 11/25/20	14	Stephen
16		Failure Modes & Effects Analysis (FMEA)	6 days	Tue 11/10/20	Tue 11/17/20		Seth
17		Design for Manufacturing & Assembly (DFMA)	8 days	Wed 11/18/20	Fri 11/27/20	16	
18		**Winter Quarter**					
19		ANSYS FEA Analysis - Structure	31 days	Sun 11/22/20	Fri 1/1/21		Stephen



Project: Project1
Date: Thu 11/12/20

Task		Project Summary		Manual Task		Start-only		Deadline	
Split		Inactive Task		Duration-only		Finish-only		Progress	
Milestone		Inactive Milestone		Manual Summary Rollup		External Tasks		Manual Progress	
Summary		Inactive Summary		Manual Summary		External Milestone			

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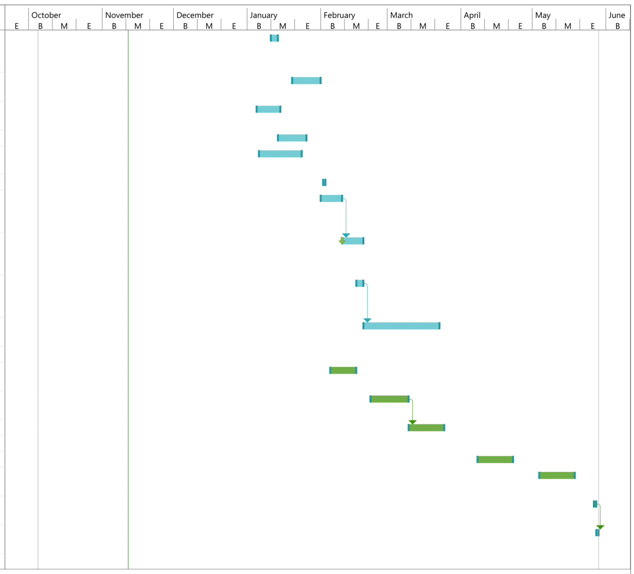
ID		Task Mode	Task Name	Duration	Start	Finish	Predecessors	Responsibility
20			Chem Lab Ambient Vibration Measurements	3 days	Mon 1/11/21	Wed 1/13/21		Ryan
21			Mass Spring System Test	9 days	Wed 1/20/21	Sun 1/31/21		Jason
22			Interim Design Review (IDR)	8 days	Tue 1/5/21	Thu 1/14/21		Ryan
23			Structural Prototype	8 days	Thu 1/14/21	Mon 1/25/21		Jason
24			Indented Bill of Material (BOM)	14 days	Wed 1/6/21	Sat 1/23/21		Stephen
25			Yellow Tag Tests	1 day	Tue 2/2/21	Tue 2/2/21		All
26			Critical Design Review (CDR Presentation)	7 days	Mon 2/1/21	Tue 2/9/21		Jason
27			Critical Design Review (CDR Report)	7 days	Wed 2/10/21	Thu 2/18/21	26	Seth
28			Risk Assessment/Safety Review	3 days	Tue 2/16/21	Thu 2/18/21		Ryan
29			Manufacturing & Test Review	22 days	Fri 2/19/21	Mon 3/22/21	28	Stephen
30			**Spring Quarter**					
31			Project Update Memo	7 days	Fri 2/5/21	Mon 2/15/21		Seth
32			Verification Prototype Sign-Off	12 days	Mon 2/22/21	Tue 3/9/21		Jason
33			Testing Sign-Off	11 days	Wed 3/10/21	Wed 3/24/21	32	Seth
34			ABET Activities					Ryan
35			Operators' Manual	11 days	Thu 4/8/21	Thu 4/22/21		Stephen
36			Verification Prototype (FDR)	11 days	Tue 5/4/21	Tue 5/18/21		Jason
37			Final Design Review (FDR REPORT)	1 day	Thu 5/27/21	Thu 5/27/21		Seth
38			Project Expo Poster Presentation	1 day	Fri 5/28/21	Fri 5/28/21	37	Stephen
39								



Project: Project1
Date: Thu 11/12/20

Task		Project Summary		Manual Task		Start-only		Deadline	
Split		Inactive Task		Duration-only		Finish-only		Progress	
Milestone		Inactive Milestone		Manual Summary Rollup		External Tasks		Manual Progress	
Summary		Inactive Summary		Manual Summary		External Milestone			

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