Vibration Table for Environmental Chamber

ME 428/429/430 Senior Design Project 2014-2015

Presented by:

Random Noise Generators

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Statement of Disclaimer

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Chapter 1: Introduction

Slime is a company specializing in tire repair and maintenance of bicycles, cars, ATV's, and many other vehicles. They make a variety of products, including sealants, gauges, air compressors, and the Tire Repair Kit (TRK). The TRK quickly repairs flat tires by injecting a sealant to patch the hole and inflating the tire to pressure using a built in air compressor. This device eliminates the need for a spare tire in your car. Refer to Appendix A for contact information on Slime.

In order to determine that the TRK is road ready, vibration tests need to be conducted to ensure that it can withstand the conditions encountered in automobiles. Outsourcing this testing costs about two thousand dollars each run and takes about two weeks. Current testing equipment on the market is either insufficient or incredibly expensive. That is why we, Random Noise Generators, have tasked ourselves to design a vibration test system for the TRK.

The vibration test system will need to test the TRK over a multitude of acceleration profiles, frequencies, and orientations and be able to support up to three TRKs at a time. The TRK must be tested within an environmental chamber that varies in temperature to ensure that it can withstand thermal extremes it may experience. As the system will be inside an office building, it cannot transmit vibrations to the surroundings, and must be light enough and small enough to fit within the designated space on the second floor. The system will also need to test similar products made by competitors to ensure that Slime's equipment remains cutting edge. The TRK must be tested within an environmental chamber that varies in temperature to ensure that it can withstand thermal extremes.

Objectives

Before being installed in a vehicle,TRKs need to be tested to ensure that they can perform after being exposed to the vibrations and thermal extremes encountered in a vehicle. Outsourcing this testing is neither cost nor time efficient when multiple tests need to be performed, and testing equipment currently on the market is incredibly expensive. Slime Sealant Systems needs a device that can run vibration tests on the Tire Repair Kit specific to various OEMs based on the requirements and specifications listed below. Our goal is to meet all of these requirements as best we can in order to provide the most useful system possible.

In order to better define the requirements of our system, we came up with a list of customer requirements based on the information given to us by Slime. The list below mostly includes areas of concern that must be designed for. These gave us a starting point to identify our engineering specifications, which are explained below.

- Functional performance
- Reliability
- Low cost
- Minimal vibration transmission
- Size
- Weight
- Aesthetics
- Temperature range
- Programmability
- Versatility
- Ease of transportation
- 3 axes of testing
- Quick testing rate
- Manufacturing ease

We developed quantifiable engineering specifications by using the QFD House of Quality included in the Appendix B. We started this process by taking the various customer requirements and turning them into specifications that could be measured and designed for. As an example, one of our customer requirements was the size of the system. To turn this into a specification, we reworded this to “system dimensions” and gave it a quantifiable limitation of 3x3x7 feet based on the needs of Slime. For other specifications, the quantifiable requirement was less obvious like the vibrations transmission requirement. We had no specific values to base our requirement off of so our specification that is measurable is that the vibration transmitted by our system must be no more than that of the current lab equipment. We plan to test this by collecting data from an accelerometer put on the current equipment.

This process takes into account all customers that are affected by our design which includes Slime, the lab technician (equipment operator), and automobile companies. We ignored the end user of the TRK because we assumed their interests to be included in the automobile manufacturers needs. Each requirement is assigned an importance relative to each group in order to find the most important design criteria. We found that “functional performance” was the most important requirement which includes creating the correct vibration profiles and operating within the specified temperature range. The next most important requirement was reliability because this is important to all of the chosen customers. The House of Quality also shows relationships between different specifications. An example is how vibration transmission is positively related to max amplitude. The higher the amplitude, the more vibration will be transmitted. Another useful aspect of the QFD was benchmarking current designs to see how well each one meets requirements.

The final specifications can be seen in Table 1 below. The risk column assesses how difficult it will be to accomplish each given specification. The H, M, and L represent high, medium, and low respectively. The more test intensive and difficulty is design, the higher the risk. The compliance column indicates how we will measure if the requirement was met or not, using analysis, inspection, or testing (symbolized by A, I, T). Analysis includes calculations and modeling. Inspection will consist of verifying specifications of purchased components as well as observation of the system parameters. The final compliance method is testing. We will activate

the machine and run each of the specified tests to ensure each requirement is met. After the table, we summarize the information for each specification. in order.

Table 1: Vibration Table Formal Engineering Requirements

Spec #	Parameter Description	Requirement or Target	Tolerance	Risk	Compliance
1	Testing Temperature	-40□ to 70□	Min	M	A, T
2	Dimensions	3x7x7 feet	Max	M	A, I
3	Weight	2000 lbs	Max	M	I, A
4	Vibration transmission	Less than tire test machine	Max	H	I,T
5	Testing Frequency	8-1000Hz	Min	M	I,T
6	Shaker rating	10 lbs per TRK at 80.7 m/s^2	Min	M	I,T
7	# of TRKs to test simultaneously	1-3	Min	M	I
8	Testing Axes	3 axes	Min	M	I
9	Cost	$100,000	Max	L	I
10	Vibration profiles	All (from OEM)	± 3dB, ±5% RMS	M	A,T
11	Run time	12 hours	Min	M	I,T
12	Setup time	30 minutes	±10 min	M	T

*H=high, M=medium, L=low
*A=analysis, I=inspection, T=testing

The testing temperature range of -40□ to 70□ is based on the limitations of the environmental chamber that testing will occur in. We are designing to allow for the most extreme temperature conditions that could occur. The tolerance indicated is minimum because a larger range allows for a safety factor and future expansion. In order to verify that the components to be exposed to temperature changes will work, we will perform thermal analysis during the design stage. We will also monitor the thermal performance during the testing phase.

The dimensions of the system, 3’x7’x7’ were provided by Slime and indicate the maximum allowable space our system can occupy. The tolerance for this is a maximum, as we will aim to design a system smaller than these dimensions. We will perform analysis on the parts we

design as well as the assembled system and inspect the dimensions of all parts to be ordered in order to verify that the size restrictions are met.

The 2000 pound maximum weight of our system is limited by the structure of the building. The system will be assembled on the second floor of an existing office building, so we must find a system that is under this weight limit or Slime will be force to reinforce the building, which is costly and could cause delays. We will verify this is met by observing the weight of all components of our system as we design and order them. Some analysis may need to be done to reduce the weight of our design.

Vibration transmission is a high risk specification because if too much vibration is transmitted to the floor it will impair the ability for others to work in the building. In order to minimize vibration transmission, we need a system that can reduce the vibrations to the ground but still transmit them to the TRK being tested. This specification uses all compliance methods to see if it is accomplished. We will verify this requirement by testing the current vibration and comparing that to our system. If others in the building sense too much vibration then we have failed at meeting the need of our customer. We can operate the current equipment as well as our system to see if we are indeed transmitting less to the floor.

The shaker rating is typically given in pounds of force that the shaker can output. The force rating we need depends on, the number of TRKs we wish to test, the maximum acceleration required by a given test, and the weight of the fixturing device. Ten pounds at 80.7 m/s^2 is the minimum forcing capacity required to move one TRK (not including fixture weight). For our final system, once we have determined how many TRKs need to be tested and the weight of the fixture, we will select a shaker rated to provide enough force for our tests. We will verify this value by inspecting the shaker rating before purchasing as well as through testing to ensure we can run each company's' individual tests.

The number of TRKs able to be tested is provided by Slime and our goal is to maximize this number. The ability to test more devices at a time increases the speed at which they can be tested. We will have to consider the effect on shaker size and cost when we decide how many TRKs our system can support. This requirement will be evaluated by inspection.

The number of axes we must test on is dependent on the test specifications given by the OEMs (Appendix C). Because multiple of these require to test the TRK along three axes, we must design a system that can hold a TRK and vibrate it along each of the three major axes relative to the device. We can verify this by inspection.

The cost of our system depends on all of the other design requirements and must be kept under the allowable budget. This will be comparatively simple to accomplish because we simply have to keep track of our budget and ensure our system cost is within our limit. We can verify this by inspection.

The next specification is that our system must have the capability to perform all OEM specified vibration tests (Appendix C) to maximize the usefulness of our shaker. In order to evaluate this, we will perform simple calculations using the weight of TRKs to be tested and maximum acceleration values specified by all tests to ensure the shaker we select will be sufficient for all tests. We can verify this by inspection of the shaker we decide to use. Once we have the shaker in our hands, we can physically test the shaker by running each of the profile tests. An accelerometer attached to the table will show us whether or not the test specifications were met. The results need to be within ±3dB of the required test.

Run time refers to how the long the device must be able operate continuously. This time is based on the the longest individual test. This specification is assuming that for any test that must be performed in three axes there will be downtime in between each test due to tests finishing after hours or re-fixturing time. To verify this, we will ensure that any shaker we decide to use has adequate cooling and is designed for prolonged operation. We will also run the longest test in order to make sure it operates continuously without any issues.

The last specification is for the setup time of the fixture. This time needs to be short so that a technician is not wasting hours of time setting up and resetting the fixture in between tests. This will save time and allow Slime to transition more easily between tests. We will verify this testing time by testing how quickly we can set up the fixture ourselves.

Chapter 2: Background

In order to gain a better understanding of our project, we gathered information on existing vibration testing devices and the current testing procedure as well as the operating environment. This section summarizes our findings.

A vital part of our research was contacting Compliance Measurement Group (CMG)[1], the company Slime currently sends their TRKs to for vibration testing. By contacting them (Appendix A), we learned about the equipment they use and companies to contact about purchasing. The following points summarize our findings.

1. CMG uses a vibration table capable of providing three axes of motion by utilizing a shaker that can be used in both vertical motion as well as horizontal motion using a slip table.
2. In order to avoid resonance in the materials, we should avoid steel because it resonates at low frequencies. CMG recommends an Aluminum-Magnesium alloy. This material can be difficult to machine but will improve performance.
3. Dedicated vibration controllers are often used for testing. These controllers require knowledgeable operators and usually cost between $30-40,000.
4. They use a closed loop controller which has an accelerometer to create a feedback loop.
5. The testing procedure to verify that a TRK is functional is to power on before and after the vibration test and see if the device performs as required.[1]
6. We were recommended to look at Unholtz-Dickie and Data Physics for controllers and shakers currently on the market. These are U.S.manufacturers, so shipping costs and maintenance costs will be lower. Their contact information is included in Appendix A.

Further research gave us insight into the types of devices available and what will be best for our purposes. There are four types of electronically controlled shakers:

- Servo hydraulic
- Eccentric motor
- Piezoelectric
- Electrodynamic

Servo hydraulic motors are controlled by a fluid flowing through a chamber to cause motion, which can be seen in Figure 1. They typically are used for low frequency applications because the viscosity of the fluid begins to affect performance at higher frequencies. Since our application needs higher frequency profiles, the hydraulic would not be ideal. Another reason the hydraulic wouldn't work is because the fluid is affected by temperature change. If we put the servo hydraulic motor in an environmental chamber, the viscosity would change and affect the dynamics of the system.

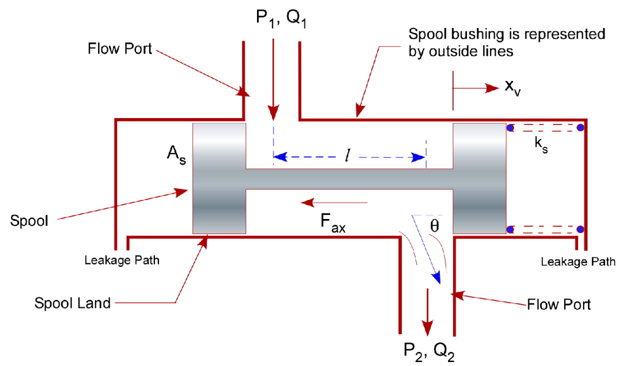

Figure 1: Diagram of servo hydraulic motor driving system.

The next type of shaking mechanism is an eccentric motor, seen in Figure 2. This motor is off balance in order to create vibrations. The only way to change the vibration profile is to change the rotational speed of the motor or where the center of mass on the motor shaft is placed. Due to slip within the motor caused by frequency, voltage, and load of the beam, it is difficult to reproduce a specific frequency profile with the motor. Also, since the eccentricity of the motor is unchanging, the amplitude of vibration at each frequency is unchanging, making it unusable for this application.

Figure 2: Eccentric motor where black mass spinning causes the vibration

The third shaker type used in vibration testing is the piezoelectric shaker, seen in Figure 3. They contain a piezoelectric core that expands and contracts as different levels of electricity are run through it.[2] When current is sent through the material, it expands and contracts rapidly, causing vibration. These kind of shakers are compact and inexpensive compared to their counterparts. The drawbacks, however, are that they are very low force, and only operate in very high frequency ranges. These shakers are more typically used in small component and acoustic testing.[3]

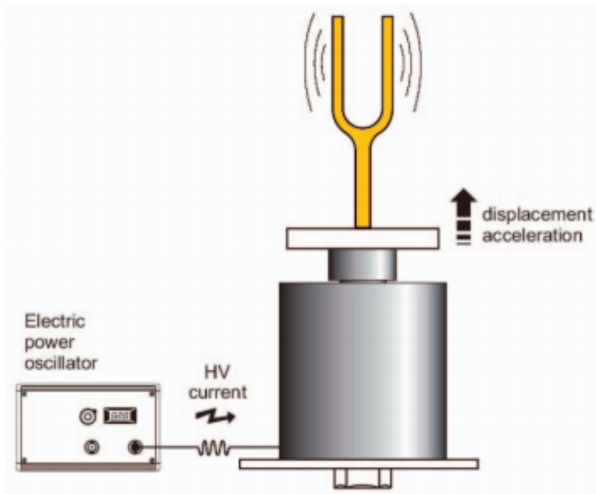

Figure 3: Piezoelectric shaker testing acoustic properties of a tuning fork.

The final shaker type is an electrodynamic shaker. These are more capable of providing the motion profiles we need, because they offer a wide frequency range and provide a large amount of force and displacement[4]. Figure 4 shows the internal working parts of an electrodynamic shaker. The platform is attached to an armature which has a coil around it and is suspended in a fixed magnetic field. As current is passed through the coil, an axial force is produced that drives the load on top. A downside of electrodynamic shakers is that they are somewhat affected by temperature, as a change in temperature affects the conductivity of the magnetic coil[5].

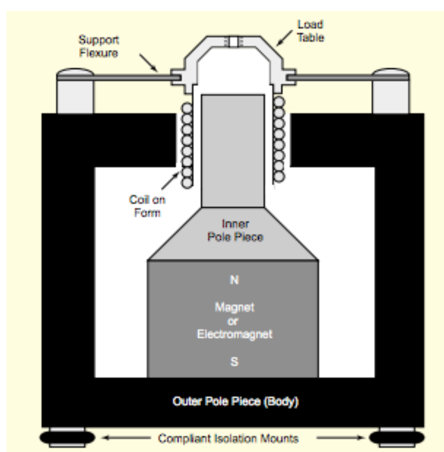

Figure 4: Electrodynamic shaker inner components

Although there are many standards for vibrational testing procedures, most automobile manufacturers have their own set of qualifications[6]. General guidelines for vibration testing are readily available from both the International Organization for Standardization (ISO) and the US Military. MIL-STD-810 can be used as a baseline for requirements of vibrational testing. However, this is just a standard for the testing procedure. Actual test specifications vary greatly

from one company to the next. Appendix C shows the required vibration specifications from each car company that our tests need to meet.

The vibration specifications found in Appendix C generally fall into three categories. Sine wave, sine sweep, and random noise. Sine wave is perhaps the easiest and most self explanatory, it is just a vibration at a specific frequency and amplitude. Sine sweep is a sine wave that starts at a low frequency, then logarithmically increases over a specified period of time to a higher frequency, then decreases to the original value over the same time interval[6]. Random noise tests are much more complicated, and are described by acceleration spectral density (ASD), RMS (root mean squared) or effective acceleration, and decibels of deviation. These specifications are explained in detail below.

Since random signals are non-reproducible and non-periodic, an acceleration spectral density is used as a way to approximate real world vibration. The ASD is simply a measure of the probability of an acceleration occurring at a specific frequency, and is given in units of acceleration squared per hertz[7]. A graph of the ASD is seen in Figure 5 on the right.

How the ASD is produced is a random signal in the time domain (shown in Figure 5 on the left) divided into very small sections to make a large number of samples, and each sample is run through a discrete fourier transform algorithm. Any repeated signal can be approximated as a series of sine waves at different frequencies, amplitudes, and phases using the normal fourier transform. the discrete fourier transform takes one of the short samples, treats it as if it repeats, and then runs fourier analysis to get the acceleration (amplitude) and phase of each frequency. The acceleration is then squared to make it in phase, and normalized depending on sampling rate, making it units of acceleration squared per hertz[8]. Then the squared acceleration is averaged across all the samples for each frequency and plotted on the ASD.

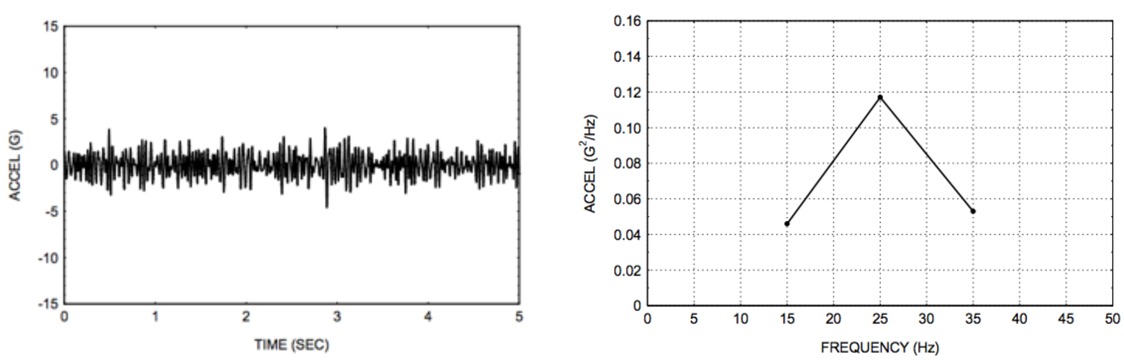

Figure 5: Graphs of acceleration over time and ASD.

Because a vibration always returns to a central “zero” value, taking the average would just cause the positive amplitudes to cancel all the negative amplitudes, resulting in an average amplitude of zero for any vibration signal. To avoid this, the amplitudes are squared to make them positive before the average is taken, then the square root is taken to return it to it’s original units. This is called the root mean square, or RMS (and occasionally referred to as the effective

acceleration). This value is used in industry to specify the overall power of the system across it's entire frequency band. The RMS can also be calculated from the ASD, by integrating it across its entire frequency band and taking the square root[7].

Since the ASD is a measure of the probability that an acceleration occurs at a specific frequency, there is occasional variation in the actual levels. Some variation is okay, but too much will invalidate the test. Manufacturers specify the amount of acceleration using decibels of deviance, and is a measure of the relative amplitude ratios of the actual values and the specified values. The equation for decibels of the ASD is

$$dB = 10 \times log_{10}(\frac{ASD}{ASD_0})$$

where ASD is the actual value, and ASD_0 is the value specified in the test. Most specifications often have a value of ±3 dB, meaning that the actual value can range from 50% to 200% of the specified value. however, the RMS value is unaffected by this.

It is critical that the vibration from the device is not transmitted into the building, but there are advantages and drawback to each method of vibration isolation. There appear to be three major methods of vibration isolation:

- Active vibration control
- Passive control with a large base mass
- Passive control with tuned damping

Active damping can target and eliminate most signals; however, it requires the addition of an entirely separate vibration control system to the system, and therefore drives the price up significantly. This type of damping basically counteracts whatever vibrations are produced. If there was a electrodynamic motor shaking the table, there would be another motor attached to the base to cancel the transferred vibrations.

Increasing the mass of the base on which the vibration actuator stands will also significantly reduce the transmission of vibration, especially at the higher frequencies in which the controller must operate. An example of what this would look like is in Figure 6. Having a mass that can absorb all the vibrations may not be feasibly due to the weight it adds. Since our device needs to operate on the second floor, weight is a critical factor that depends on the structural design of the building.

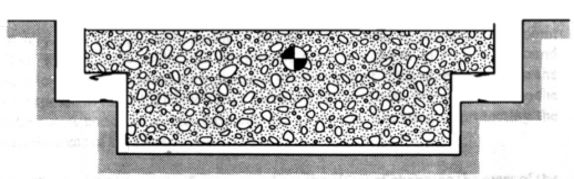

Figure 6: Example of mass used to absorb vibrations

The third option, tuned vibration isolation, can target specific frequency bandwidths and eliminate transmission based on resonance of the device and the building. Figure 7 shows some examples of these using rubber feet or rubber platforms that the system can sit on. However, by absorbing and eliminating the vibration, this type of isolation limits the maximum acceleration and frequency at which the system can operate[5].

Figure 7: Elastomers used as damping in system

Figure 8 below demonstrates how effective a damping system can be. In this figure, a damping system with a natural frequency ω undergoes excitation of varying frequencies and the response at each frequency is measured. The important thing to note is that for any input frequency above the natural frequency, the magnitude of vibration is reduced through the damping device.

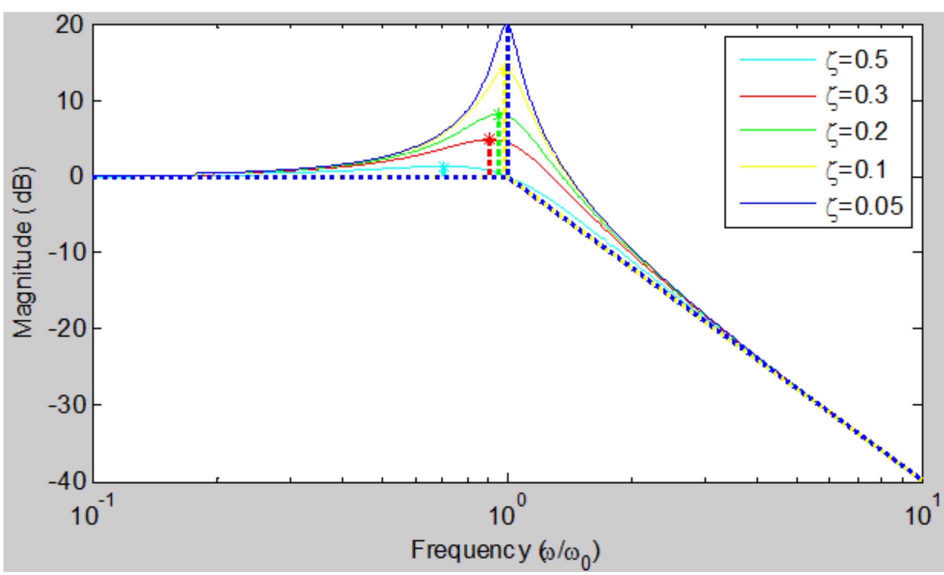

Figure 8: Frequency response across a spring

The drop-off rate for a spring system is -40 dB/decade, or -12 dB/octave. above its natural frequency; therefore, the equations governing the transmissibility are:

$$DOR \times N_{oct} = 10 \times log_{10}(\frac{P_T}{P_0})$$

$$N_{oct} = log_2(\frac{f_2}{f_1})$$

$$T = \frac{P_T}{P_0} \times 100\%$$

Where DOR is the drop-off rate, N_{oct} is the number of octaves, P_T is the power transmitted to the surroundings, P_0 is the power of the vibration signal at frequency f_2, f_1 is the natural frequency of the isolator, and T is the transmissibility in percent. So for an air spring with a natural frequency of 5 Hz, 10 Hz signals transmit less than 7% of the amplitude to the ground. At 8 Hz, the lowest frequency specified in the tests, less than 16% of the signal is transmitted.

From a conversation with Ray Fish of Technischer Überwachungsverein (TUV), we received useful information on the actual testing procedure. The test is run by affixing the test device to the shaker by clamping it to a fixture on the shaker armature. Simple methods may be used to clamp, such as using plywood held together by bolts. An accelerometer is attached to the fixture or to the text device itself, and reads off into a controller. The run-time of the test is entered into the controller, and different values are entered into the controller depending on the test type; for sine test, the amplitude and frequency, for sine-sweep, the amplitude, starting and ending frequencies, and the time of each sweep, and for random tests, the ASD values for each frequency are entered. The controller then produces a signal to be used and displays the RMS and crest-factor, which are used to determine the validity of random tests. If they are not acceptable, then extra values may need to be added to the ASD to raise/lower these values. Abort parameters can also be entered into the controller, allowing it to stop the test and inform the operator if anything goes wrong with the test (e.g. the accelerometer comes off of the fixture)[9]. Ray Fish's contact information is listed in Appendix A.

When testing in an environmental chamber, there are two possible ways to accomplish the temperature parameters. The entire shaker can be placed in a large chamber or just the armature and device being tested can be placed in a raised chamber. After talking with vendors of electrodynamic shakers, they said that putting the entire machine into a chamber is hard on the components and decreases performance. The standard way is attaching a thermal barrier to the top of the shaker and shaking only the device in the chamber seen in Figure 9. The thermal barrier is essentially a rubber gasket that seals off all the internal components from the chamber. It prevents any moisture getting into the shaker and keeps the shaker operating at an ideal temperature.

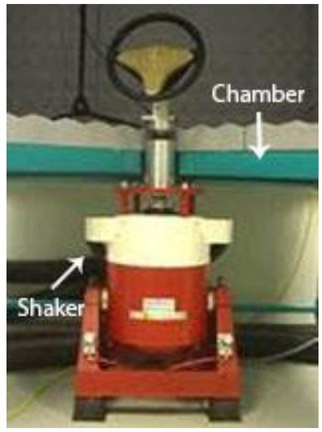

Figure 9. Interface between shaker and environmental chamber. The shaker extends through a hole in the bottom of the chamber.

After deciding that an electrodynamic shaker would be the best option to drive our system, we began to contact different companies in order to compare the products on the market. As more research was done, we determined that an integrated system from a single company is the best choice to ensure system compatibility. This is because each component is designed to work together rather than trying to adapt different products to work with each other. A fully integrated system would include a shaker, amplifier, and controller with software.

When buying electrodynamic shakers, there are two options for shaker orientation, vertical only or vertical and horizontal (Figure 10). The vertical shakers are typically smaller since they only need to shake in the z axis. They have an armature that oscillates up and down with threaded holes so a part can be mounted. The vertical and horizontal shakers have a pivot point in the system so the shaker can be in the vertical position or it can be flipped on its side. When it is on its side, the system needs a slip table in order to support the load on the armature. The armature attaches directly to the slip table which is on linear guide ways. The devices being tested can then be mounted to the table rather than the armature used in the vertical axis. The dual axis shakers are useful because of the large mounting surface but they are much heavier due to all the additional components. They are also significantly more expensive than only z-axis shakers.

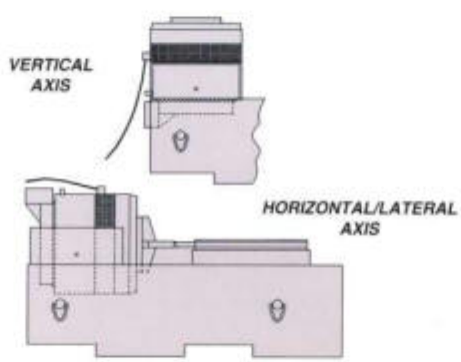

Figure 10. Electrodynamic shaker with slip table for vertical or horizontal vibration

Another thing we found when researching shakers on the market was that the armature is only a few inches in diameter and the mounting surface also has minimal bolt holes. This makes mounting things more difficult. In order to overcome this challenge, a custom fixture can be made to attach to the armature or you can use a head expander shown in Figure 11. This increases the mounting surface available but increases the mass being shaken which means that the shaker needs to be stronger in order to move at the vibration profiles. When a head expander is used, a larger shake table is needed as a rule of thumb.

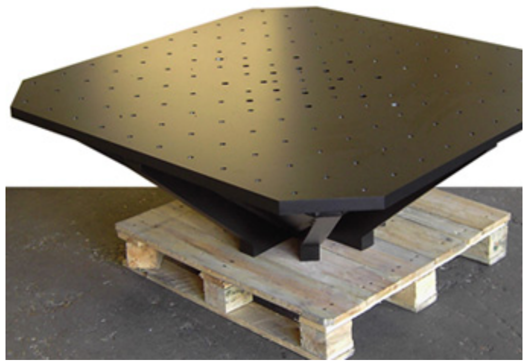

Figure 11: Head expander for vibration table

Chapter 3: Design Development

Concept Generation/Ideation

We spent a couple weeks doing ideation for the entire system. We began by identifying necessary components; a shaker, a controller, a method of attaching the TRK to the shaker, and an isolation system. We generated as many ideas as possible regardless of how impractical in order to have a wide range to choose from. We first began with the brainwriting method of idea generation for shaker, controller and fixture. A component was selected, and each team member wrote as many ideas as they could for that single component in five minutes. The papers were then switched among the team, and then new ideas would be generated by expanding on the ideas of the newly received paper. This was then repeated until every team member had written on every sheet of paper.

The next part of ideation was a brainstorming session. Each member would come up with a new idea, or expand on another idea and then write it on a sticky note and stick it to a board, as shown in Figure 12. By saying each concept aloud, ideas can bounce off each other and build in a conversational style. The sticky notes ensured that no ideas were being filtered by a single person writing down the ideas and made each idea easily visible so we could look back and expand on them. We also each engaged in individual concept generation outside of meetings. Once we had as many ideas we thought we could generate, we proceeded to narrow down each subsystem.

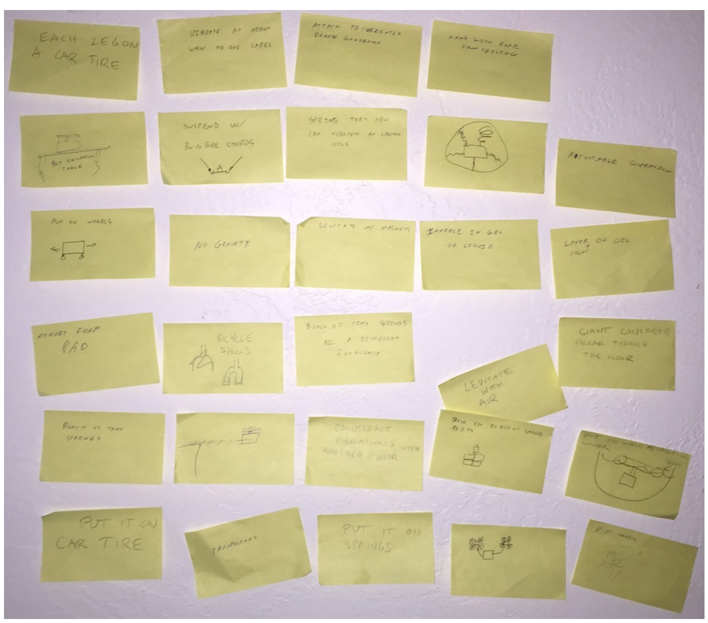

Figure 12: Sticky notes used for ideation

Idea Selection

Initially, we eliminated any ideas that were impractical, impossible, or dangerous. We then began using different methods for narrowing each individual component.

Shaker

The requirements precisely spelled out in the test specifications are critical to the functionality of the system, which immediately eliminated some of our ideas (like an unbalanced motor). One of the important specifications is the shaker needs to be be controlled in order to accurately produce the tests. Since the vibration profile needs to be controlled, the different motor types discussed in the background section like the eccentric and the servo-hydraulic will not work. Only the electrodynamic shaker can reach the needed specifications. Although it is heavy and expensive, it is the only idea that would conceivably be able to achieve the acceleration and frequency requirements of the tests. Using an electrodynamic shaker also necessitates the addition of a power amplifier to our system, which are typically sold in conjunction with the shaker. Due to the very specific need for an accurate device, we plan to purchase an electrodynamic shaker with accompanying amplifier. We did not use any formal decision making process for this choice as it was the only reasonable option available to us. We later go into detail on how we plan to select the appropriate shaker.

Controller

There were a few ideas on how to implement a controller that could produce a signal to fit the necessary specifications. Our main idea was to design and program a controller ourselves using MATLAB, which would greatly reduce the cost. This problem proved to be much more complex than we anticipated. Due to the nature of random signals, reproducing one requires an advanced level of expertise in spectral analysis and statistical analysis. Being able to do so inside of a program also requires familiarity with random number generators, fourier algorithms, and signal filters. Since none of us have adequate experience in these areas, designing a controller was unfeasible so we decided to purchase this component. In order to ensure system compatibility, we will be purchasing a controller from the same company that will provide us a shaker and amplifier.

Isolation

There were many ideas generated for the isolation system, which we began narrowing by eliminating any impractical designs. The remaining ideas were then entered into a Pugh matrix (Appendix E), narrowing our selection to using either visco-elastic feet or air springs. This Pugh matrix provides a way to quickly compare ideas to an industry standard (air springs) and determine positives and negatives of each alternative. However, after researching different shakers, we found that they come pre equipped with an air spring damping system. The natural frequency of this air spring is 5 hz, which means that above this frequency, the vibration transmitted through the air spring is reduced. Our system will have air spring damping because it is already included in any shaker we will purchase and adequately meets our needs.

Fixture

We came up with a large amount of ideas for the fixture, but had to eliminate a few based on certain criteria. We needed the fixture to be able to hold multiple test subjects, and be able to test them in three different axes. The remaining devices were further narrowed by eliminating designs that did not fit the space requirements, or would prove difficult to attach to the shaker armature. After this elimination, we were left with two possible fixture designs.

The first fixture design is called “The Box”, shown in Figure 13. The top of the box can hold one TRK above and one TRK below. Each side can hold one TRK. The TRKs on the sides can be rotated. With this combination, two TRKs can be tested in any axis at the same time.

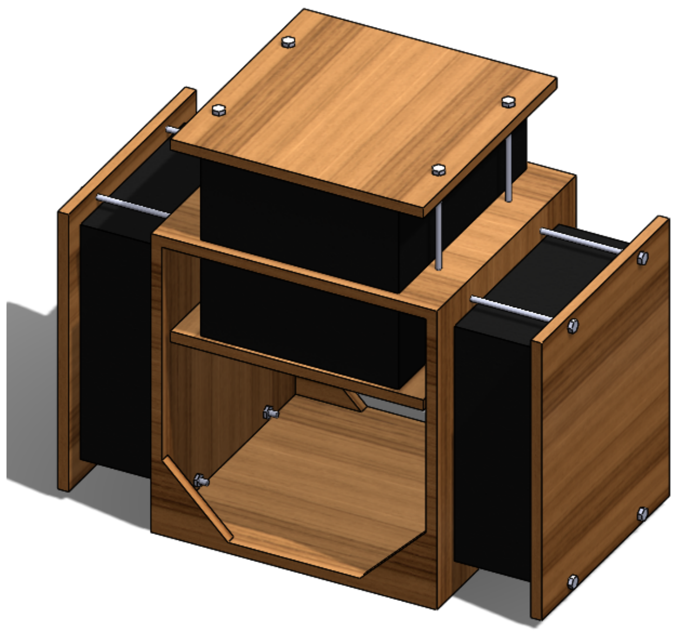

Figure 13: “The Box” fixture (wood) holding four TRKs (black)

The second idea for consideration is similar to “The Box”. It involves using two different kinds of fixtures, one designed for holding two TRKs vertically (Figure 14), and the second for holding two TRKs horizontally (figure 15) other.

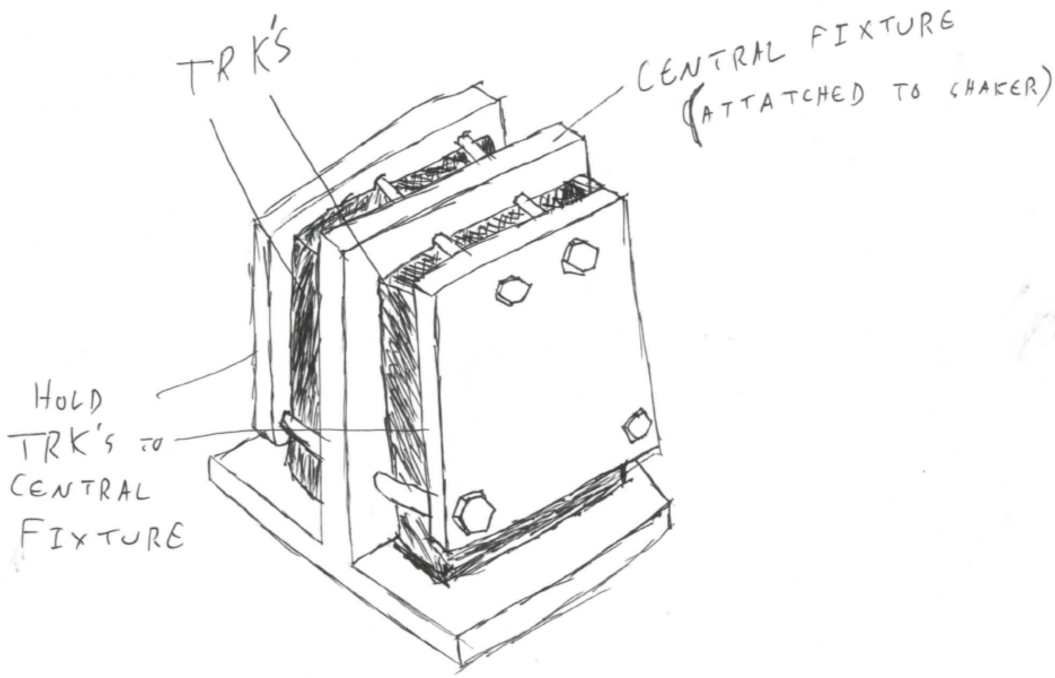

Figure 14: Fixture to hold TRKs vertically

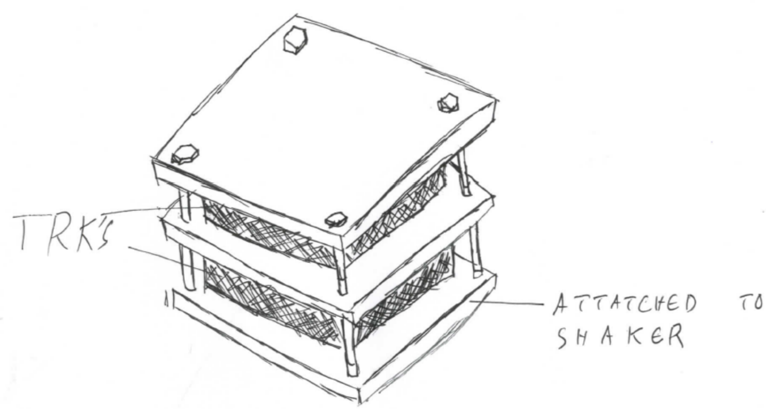

Figure 15: Fixture to hold TRK's horizontally

A simple fixture made of particleboard held together with bolts was constructed to ensure that the TRK could be held in the vertical orientation, shown in Figure 16. When tightened, the TRK could not be forced out of the fixture by shaking, pushing, or striking the TRK. This demonstrated that having it held between plates that are parallel to the axis of vibration can be done without the TRK sliding out of the fixture. We will still need to determine the minimum amount of tension in the bolts required to hold the TRK in the fixture.

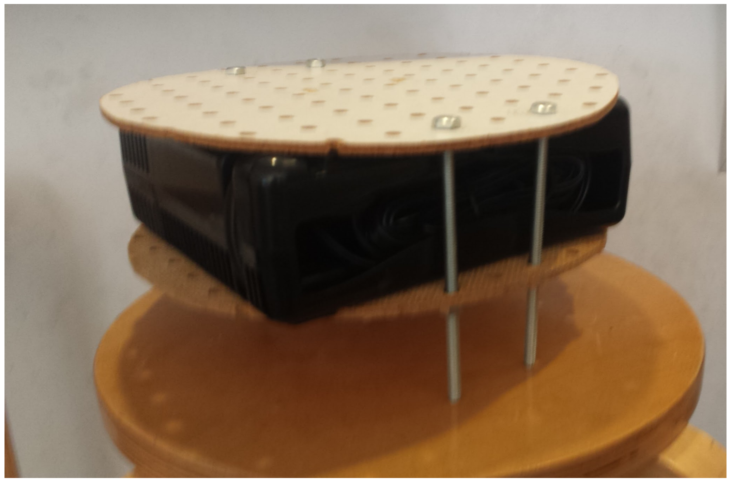

Figure 16: concept model of fixture

Several things will need to be further analyzed for these fixtures. The deflection of the fixture needs to be low enough that it does not result in any phase/amplitude changes between the TRKs and the shaker armature. This will result in inaccuracies in the testing. The fasteners holding the fixture also need to be minimized for deflection, and stress/fatigue analysis will need to be conducted to ensure that they don't break. We will also need to run analysis on the holding strength of the fasteners to ensure they do not come loose in the middle of testing. We need to ensure that the vertically oriented TRKs will remain in place, which will either require a minimum friction force imposed on the TRK or a way to physically block any vertical motion of the TRK. Weight needs to be minimized to make it easier for the shaker to handle, and size needs to be reduced to ensure it can fit within the thermal chamber.

Final Design Concepts

For the construction of the system, several options were considered for how the system would be oriented/installed. The one thing that was constant in each system was a purchased shaker, controller, and amplifier from a vendor which includes air springs as an isolation system. This will ensure as much system compatibility as possible. The main variance in each one of the system concepts is how the TRK is mounted to the shaker. We considered three main system concepts. For all of these concepts, a system design is included in Appendix B.

The first option was to use a slip table in combination with a vertical axis shaker as shown in Figure 10. As discussed in our background research, a slip table allows for a shaker to be rotated from the z-axis to test x-axis vibration. This is beneficial in that it provides a large area to mount TRKs to in order to test multiple at a time without designing a complicated fixture. It allows for devices to be mounted to the slip table while the shaker is being used in the z-axis or vice versa. The downside of this design alternative is that it drastically increases the weight and

size of our system, failing two of our specifications. It also requires a larger shaker to move the additional mass and this drives the cost of our system up.

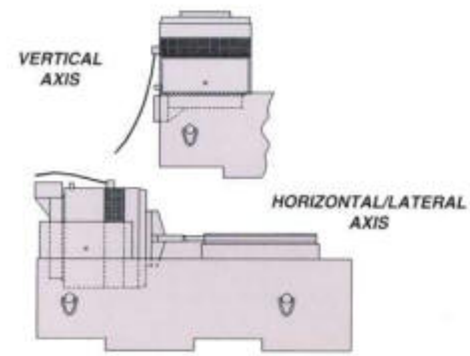

Figure 10. Electrodynamic shaker with slip table for vertical or horizontal vibration

The second option was to combine a head expander with a z-axis only shaker, as shown in Figure 11 in the background section, page 10. The TRKs could then be attached to the expander using simple fixtures and there would be different fixtures for different orientations. A benefit of this is having a large area that is convenient to attach TRKs to, however this adds a significant amount of weight. This means that the shaker needs to be larger and heavier in order to move this additional load. This is also contributes to an increase in cost because a larger shaker is more expensive.

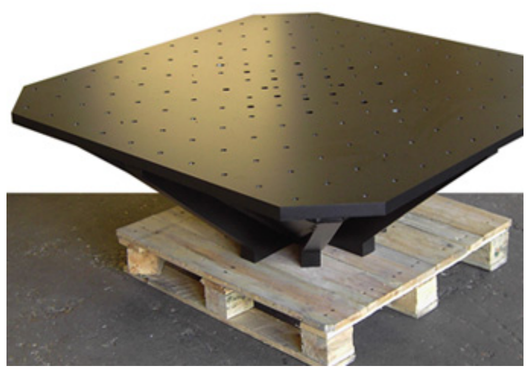

Figure 11: Head expander for vibration table

The third design we considered uses a custom designed fixture to optimize weight and cost. The fixture would be designed to minimize amount of extra weight to secure the TRKs to the shaker. Our concepts for these fixtures are discussed in the Fixture section above. The benefits of this design are that it allows us create a cheaper fixture alternative and design an optimized solution for our problem. This should also reduce the weight of our fixture.

These system configurations were put into a weighted decision matrix shown in table 2. This matrix only compares the fixture of each alternative since this is the feature that is changing.

Each fixture was given a relative value of 1-5 for each criteria depending on how well it fulfilled the requirement compared to the others. The weights of the decision matrix were determined by a pairwise table (table 3) that compared the relative importance of each criteria. If a criterion on the left was deemed more important than a criterion on the top, then the cell connecting them was assigned a value of 1, otherwise it was assigned a zero. The totals were found for each criteria and normalized, giving the weighting of each.

Table 2: Weighted decision matrix, top concepts highlighted in yellow

	Rigidity	Cost	Weight	# of TRKs	Configurability	Total
Slip Table	5	1	1	5	5	**2.87**
Head expander w/ single TRK fixtures	4.5	1.5	1.5	4	4	**2.73**
The Box	3.5	5	4	3	3	**3.87**
Separate Vertical and Horizontal fixtures	4	5	5	3	3	**4.20**
Weights	0.13	0.27	0.27	0.20	0.13	1

Table 3: Pairwise comparison matrix

	Rigidity	Cost	Weight	# of TRKs	Configurability	Total	Weight
Rigidity	0	0	0	0	1	**1**	**0.13**
Cost	1	0	0	1	1	**3**	**0.27**
Weight	1	1	0	1	0	**3**	**0.27**
# of TRKs	1	0	0	0	1	**2**	**0.20**
Configurability	0	0	1	0	0	**1**	**0.13**

The two rows highlighted in yellow represent the best options from our decision matrix. Based on these results, we plan to use one of these custom fixtures. At this stage in our project, we have not performed enough analysis to conclude which custom design will more appropriate. Our management plan which follows discusses our plans for future analysis to develop the details of our fixture.

Chapter 4: Final System Design

Our final design allows two TRK devices to be tested at the same time with a specific vibration profile. The shaker is connected to a head expander which increases the mounting area. The TRKs are then clamped in either the horizontal or vertical direction depending on what the test specifies.The horizontal position fixes the TRKs with a bar over the top and bolts into the head expander. The vertical positions are secured using an L bracket and a bar to clamp the TRK. After the TRKs are fixed in the appropriate position, the operator can input the desired vibration profiles into the controller software on a laptop. The controller generates the signals and passes it through the amplifier in order to drive the shaker. An accelerometer attached to the head expander creates a closed feedback loop so the controller can verify the correct profiles are being created. After the test is completed, the operator can inspect the TRK to see if it passed or failed and then move onto the next test orientation.

The following sections present our final design selection in detail, as well as the analysis we performed to verify our design decisions. Since we are designing a whole system, we will look at each subsystem in detail and then the complete system integration. Figure 17 shows the different components of the design:

- Shaker/Amplifier System
- Head Expander
- Fixturing

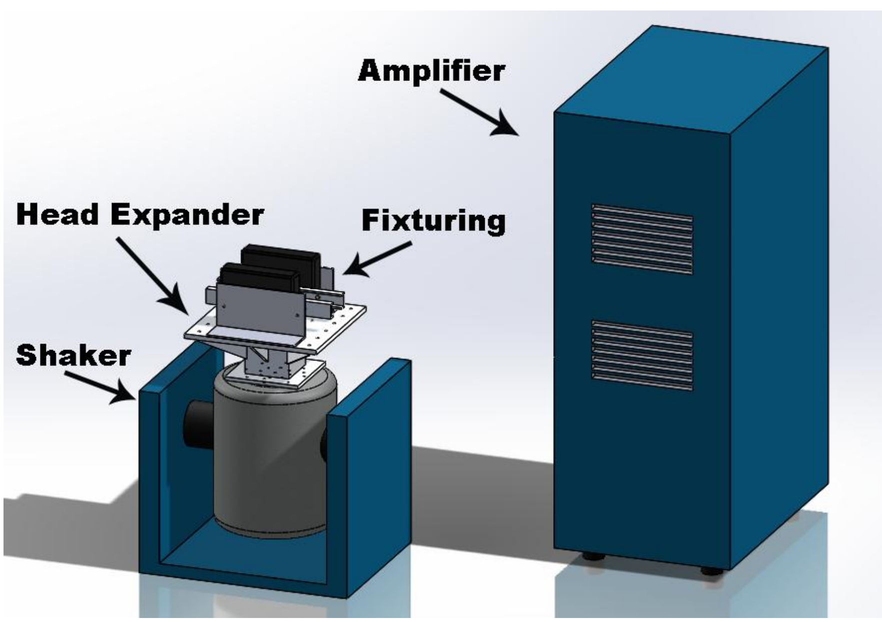

Figure 17. Integration of all system components

Detailed Design

Shaker/Amplifier System

The main part of the system is the electrodynamic shaker and amplifier. After talking with four different companies, we were able to get specifications and quotes for different systems. Table 4 shows a comparison of all the different shakers to see the differences in price and performance. The shaker we are purchasing for our design is the Spectral Dynamics SD-660-230M which comes with an amplifier and blower and can be seen in Figure 18. Refer to Appendix J for the SD-660 data sheet.. The main reason for this is the much lower cost compared to all the other companies which gives a larger return on investment. It also has a 9.1 inch armature diameter for mounting and meets the force requirements for both the sine and random noise tests. The bolt pattern on the armature can be seen in Figure 19 and consists of 17 M8 bolts which is equivalent to a 5/16”-18 bolt.

Table 4. Price breakdown of different shakers from different companies. Orange=low safety factor; Red=failed specification

		Bruel & Kjaer		Data Physics	Spectral Dynamics		Unholtz Dickie	
		V-650	V-830	V617	SD-440	SD-660	TA240D-S092	TA240-S202
Shaker	Armature Diameter (in)	6.14	7.28	6.9	5.9	9.1	7	13.3
	Sine Force (lbf)	495	1524	1500	440	660	900	1000
	Random Force (lbs)	346	1298	1120	440	660	700	700
	Acceleration Sine (g)	100	99	100	80	65	90	45
	Displacement (in)	1	2	2	1.5	1.5	1.5	2
	Load Support (lb)	110	350	200	220	264	-	600
	Weight (lb)	417	1358	1385	1280	1320	1100	1700
	LxWxH (in)	26.4x29.3x23	30.4x39.6x33	26x30x31.4	26x23x23.2	28.3x23.2x24.6	31.25x21.8x27.8	27.7x28.5x29.7
Amplifier	Weight (lb)	463	1087	-	-	-	-	-
	LxWxH (in)	21.1x32.5x39.4	21.1x33x74	-	-	24x35.4x57	-	-
Prices	Shaker	$46,000	$105,000	$61,025	$17,500	$21,000	$59,000	$60,000
	Amplifier	(included)	(included)	(included)	(included)	(included)	(included)	(included)
	Controller	$26,000	$26,000	$14,300	$8,500	$10,500	$22,000	$22,000
	Shipping	-	-	-	$3,000	$3,000	-	-
	Commission	(included)	(included)	(included)	$3,000	$3,000	(included)	(included)
	Accelerometer	$500	$500	$500	$400	$400	(included)	(included)
	Total	$72,500	$131,500	$75,825	$32,400	$37,900	$81,000	$82,000

Figure 18. Layout of shaker and all the included components. From left to right: controller, amplifier, shaker, blower

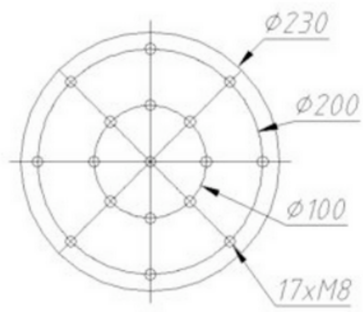

Figure 19. Bolt pattern for shaker armature

For the controller, we are going to use the Lynx model from Spectral Dynamics seen in Figure 20. This is an entry level controller for the system. Since the vibration profiles given by the original equipment manufacturers are fairly simple sine and random signals, not much computational power is needed. The Lynx system comes with a notebook computer and the software needed to generate the vibration profiles. It also has four input ports for accelerometers and one output channel to the amplifier.

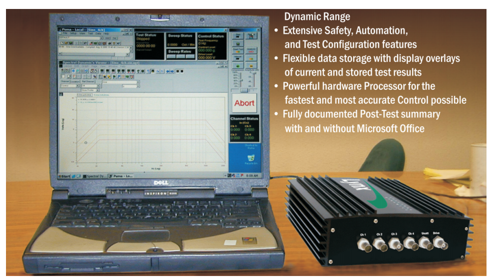

Figure 20. Lynx controller system

There are a couple accessories that need to be integrated with the shaker system. The first one is an accelerometer which is needed to verify that the profiles being generated are actually being transmitted to the test device. One accelerometer with a resolution of 100 mV/g will work for this application and is supplied by Spectral Dynamics. Figure 21 shows the sensitivity variation with temperature. Our range is -40 to 160 degrees Fahrenheit so the sensitivity deviation is very close to zero above 30 degrees. Below this temperature, we need to take into account the variation that can range to about -10%.

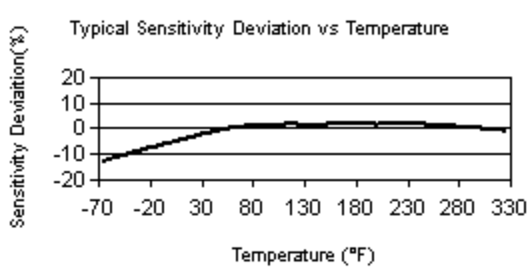

Figure 21. Accelerometer sensitivity with different temperatures

The next accessory is the thermal barrier which is needed because the shaker is being integrated with an environmental chamber. This prevents any moisture from getting into the shaker and ruining the electrical components. This is supplied by Spectral Dynamics and is attached in the factory.

Head Expander

In order to attach the TRKs to the table for testing, we are going to build a head expander. The head expander increases the mounting area on the shaker and can be seen in Figure 22. It is constructed of 6061 aluminum plates welded to a central square tube. We originally selected aluminum because we were recommended by the testing company CMG that this material has better vibrational properties. Bruel and Kjaer, a manufacturer of vibration systems, published the document “Fixtures for B&K Exciters” (included as Appendix E) which explains that the ratio of Young’s Modulus to density of a material is the determining factor for its natural frequency in construction. Interestingly, this ratio is very similar for steel, aluminum, and magnesium. Due to this fact, we selected aluminum for its lower weight and sufficient stiffness.The more mass we put on the shaker armature, the less acceleration it can produce. The aluminum head expander we designed will weigh less than twenty pounds and will still allow the shaker to reach the desired acceleration of up to 10 g. We chose 6061 aluminum because of its high weldability properties to make manufacturing easier.

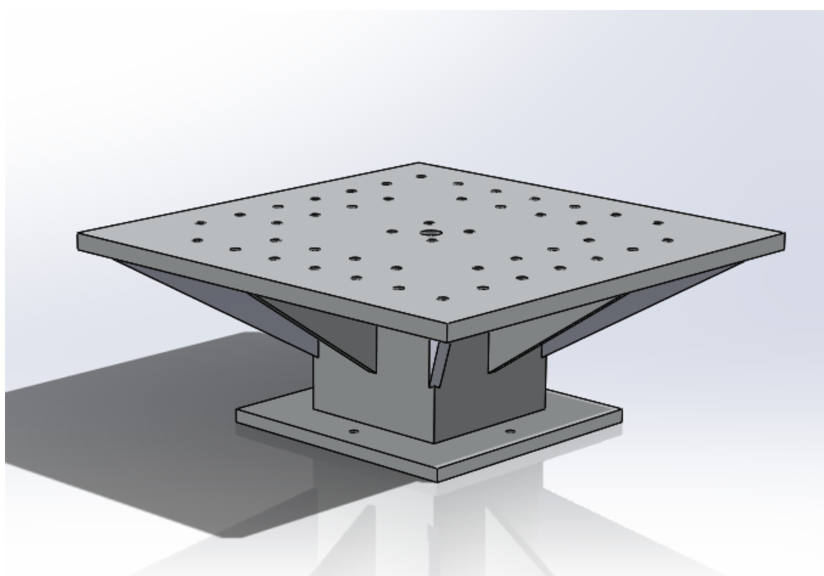

Figure 22. Head expander

The top plate on the head expander measures 18”x18”x5/8”. The eighteen inches on top gives a large mounting surface to put the TRKs on. We based this on the max length of two 9”x11” TRKs sitting next to each other in the horizontal orientation. If for some reason the TRKs are bigger than this, they can still hang off the edge of the head expander without affecting the results. The 5/8 inch thickness is based on the need to have natural frequency above our

testing range of 1000 Hz. The thicker the plate, the higher the natural frequency and the calculations for this are talked about in the following analysis section.

Another way to increase the natural frequency is to have support ribs to give the top plate more rigidity. The eight ribs attached to the head expander are made from ⅜ inch thick aluminum. They extend to the edge of the top plate for maximum support. The dimensions and drawings for these parts can be seen in Appendix F. For manufacturing, we plan to cut four rectangular pieces (2 different sizes), down the middle to reduce costs. The four rectangular pieces will turn into the eight triangular ribs. The bill of materials in the cost analysis section below shows the material we will order.

The base of the head expander is 9”x9”x1/2” with a four drilled holes in the same pattern as the armature. The armature on the shaker is 9.1” so this plate can be mounted directly to the armature with 5/16” bolts.The square tube that connects the base with the top plate is half inch thick and 6 inches cube. This dimension allows for the head expander to be integrated with an environmental chamber in the future.

The top of the head expander has drilled holes in a grid pattern. There are 32 holes spaced 3 inches apart so that the different fixturing orientations can be attached to the table. These holes are going to be drilled into the aluminum and then steel threaded inserts shown in Figure 23 are going to be press fit into the holes. We are using threaded inserts because if we tapped aluminum holes, there is a much higher chance that the threads could be stripped. If the threads strip, it decreases the configurability of the table by decreasing the number of mounting holes. The threaded inserts will allow for a much more rigid fixturing system with tighter clamping forces and more reliability.

Figure 23. Steel threaded inserts for head expander

The last part of the head expander will be welding it all together. We are taking our drawings in Appendix F and materials to a professional welder so that all the dimensions are correct. An important tolerance on the head expander is the flatness of the top plate and the parallelism between the top and bottom plate. This will allow for all the vibration profile to be transmitted axially and will eliminate any off center forces that can create resonance or out of plane loads.

Fixturing

When testing, the TRKs need to be rotated in different orientations so that they can be vibrated along each axis. Our design has two set up orientations, one for horizontal and one for vertical

testing. The horizontal fixturing is the simplest and can be seen in Figure 24. This consists of a U bar and bolts to hold the TRK to the table. The reason we are using a U bar instead of a flat beam is for the higher area moment of inertia which helps to prevent bending and deflection. For similar weights, a U bar is much more rigid in the clamping direction and will give a flatter mounting area when the bolts are tightened. The bolts will be 5/16”-18 and made of steel so that they can be threaded into the inserts on the head expander. The length of the bolts is four inches because the max thickness of a TRK is three inches. This will allow for the bolts to go through the U bar and still have at least a half inch of threads. The head of the bolts will be a hex pattern so that the bolts can be torqued down to the correct specification with a standard tool.

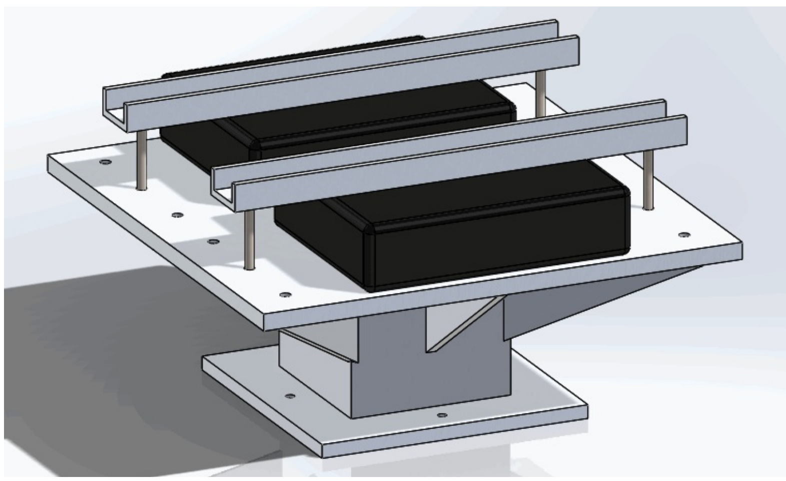

Figure 24. Horizontal mounting orientation to fixture TRK

The second fixturing orientation is in the vertical direction which can be seen in Figure 25. This orientation uses all the same components as the horizontal fixture with the addition of an L bracket. The L bracket is made out of 6061 aluminum and is bolted to the head expander table with two, one inch long 5/16” bolts. The length of the L bracket is a foot long so that it can accommodate different sized TRKs. The length of the leg attached to the table is four inches while the length of the vertically extended leg is six inches. This allows for a larger clamping surface on the TRK and is clamped in a similar way to the horizontal fixture with a U bracket on one side and an L bracket on the other. A four inch long 5/16” bolt will secure the two together and be threaded into a steel insert in the L bracket. After a TRK undergoes a test in this vertical direction, it can be rotated 90 degrees in order to test the final axis.

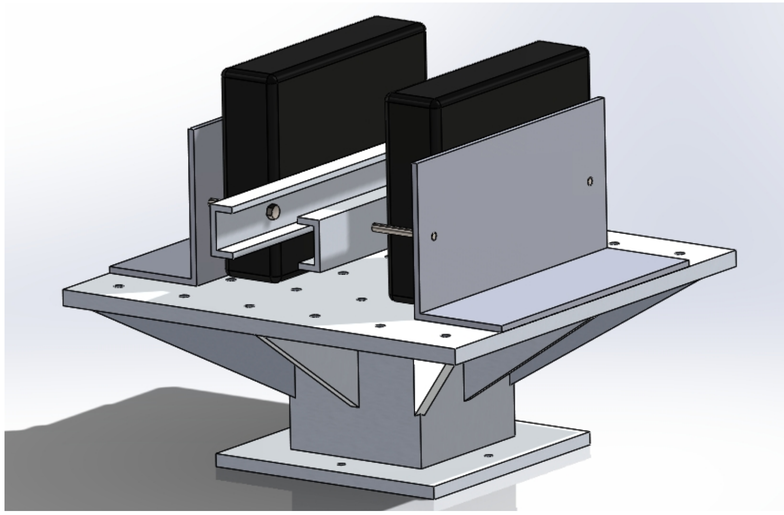

Figure 25. Vertical mounting orientation to fixture TRK

Analysis

This section details the analysis we performed to verify that our design will sufficiently meet all the needs of the sponsor. This section will only include a brief summary, and all of the supporting documentation will be included as Appendix G.

The first step in our analysis was designing the head expander for both strength and vibration properties. Since our tests range from 0-1000 Hz in frequency, we needed to design an expander with a natural frequency higher than 1000 Hz to avoid resonance in the part during testing, as this could affect the accuracy of the vibrations being passed to the TRK during testing. In order to the verify this, we calculated, based on the size of the mounting surface, what thickness of aluminum was needed to achieve the desired frequency. We also determined the thickness of the support ribs with a similar analysis. Using these initial dimensions, we selected nominal sizes of readily available materials to simplify our design. We then had to repeat our first calculation to reflect the marginally altered dimensions. Table 5 below summarizes these results, showing that each critical piece of our design meets the vibration criterion. This calculation follows the procedure outlined in Appendix E. You can also see drawings of these parts in Appendix F.

Table 5. Results of head expander frequency analysis

Top plate dimensions	18”x18”x5/8”
Short rib dimensions	6”x4”x3/8” triangle
Long rib dimensions	8”x4”x3/8” triangle
Top plate natural frequency	1767 Hz
Long rib natural frequency	1713 Hz
Short rib natural frequency	2015 Hz

To evaluate the properties of our head expander design, we ran a series of simulations in SolidWorks. These simulations included stress, deflection, frequency, and fatigue analysis. These simulations not only verified our frequency analysis, but also proved that our design was stiff and resistant to fatigue failure.

We ran each of our test cases and recorded the results, in order to verify that our design satisfied the frequency requirements while also being rigid and strong enough to withstand vibration. SolidWorks allows us to place loads in various location on the object and fix the geometry to simulate realistic load cases that may be experienced during testing. We ran several of these cases, and the results are summarized in Table 6 below. Note that the values listed are the maximums and only occur in the far edges of the head expander.

Table 6. Results of SolidWorks simulations

Case 1: Corner Loads			Case 2: Distributed Load		
Deflection	0.00035	in	Deflection	0.0001	in
Stress	2000000	psi	Stress	690000	psi
Safety factor	27.575	-	Safety factor	79.92754	-
Case 3: Frequency Analysis			Case 4: Fatigue Failure		
Mode 1	1033	Hz	Cycles to Failure	1.00E+08	Cycles
Mode 2	1127	Hz			

Case 1 considers two loads on opposite corners of the fixture of 1000 N each. These loads represent worst case loading conditions based on our maximum acceleration and the weight of three TRKs. For this (and all subsequent simulations), the base of the head expander was marked as fixed to simulate being bolted to the armature. Even in a drastically exaggerated loading scenario, we obtained a factor of safety from yield stress of 27. This factor of safety was determined using vonMises criteria based on the yield strength of aluminum (5.5 x 10^7 psi). For figures showing the simulation results in 3D, see Appendix G.

Case 2 considers a singular load of 1000 N distributed evenly across the top surface. This case assumes that the force of three TRKs being accelerated at 60 m/s^2 is being supported by the

entire head expander. As expected, this case produce significantly less stress and deflection, giving a very large factor of safety from reaching yield strength. Appendix G shows the simulation results.

Case 3 is a frequency analysis. This simulation fixes the bottom of the head expander, and determines the modes of vibration. The majority of these modes are useless because they are not in the direction of forced vibration the system will experience. Table 6 lists the modes that we are concerned about. For our purposes, we needed a fixture with natural frequencies above 1000 Hz, and we have achieved that in both of these modes. Appendix G shows these results.

Case 4 is a fatigue life analysis based on 1 million cycles of Case 1 in alternating load. This simulation did not provide specifically useful data because SolidWorks returned the maximum life of 100 million cycles. Note that this is 100 million cycles of 1 million alternating loads. This value in combination with our factor of safety for yield strength leads us to assume that fatigue failure is not a concern for our head expander.

Press fit

Since the top of the head expander is going to have threaded inserts press fit into the table, we did some calculations to make sure the fit was tight enough for the given forces. We originally looked at using 3/8” bolts but since the shaker armature has 5/16” bolts, we wanted to keep a standard size. Even though the analysis is for a different size, the end results are still the same. We initially calculated the interference fit between the aluminum hole and the steel threaded inserts. This gave us a radial pressure of 399.5 ksi. From this we calculated the hoop stress in the aluminum plate. This was found to be 399.5 ksi because the aluminum plate diameter essentially goes to infinity since it is much larger than the hole. The yield stress of aluminum is 40 ksi which would mean that the plate would fail. However, since we did calculations based on what interference the manufacturer recommended, we knew it shouldn’t fail. This led us to look at the geometry of the threaded inserts. As seen in Figure 23, there is a helical pattern on the outside with grooves. This leads us to believe that as the insert is pushed into the aluminum hole, it shears away the material for a tight fit. This will wedge the insert into the material so it will resist torque as well as any axial forces. As long as we follow the manufacturing suggested hole size, we are confident that the inserts will withstand any forces in the system. Appendix G shows these calculations.

U beam deflection

The U beam is used to clamp the TRK in both the vertical and horizontal positions so we calculated the deflection it would experience. The dimensions of the U bracket we analyzed are ½ inch thick, 2 inch base, and 1 inch leg height. The overall length was one foot. For the forces experienced on the beam, we put two point loads at the ends to represent the bolts and two forces that act on the corners of the TRK. This is because if the beam starts to bend, it will lift off the center of the TRK and be in contact at the two outside corners. The magnitude of the forces is based on the clamping force needed to hold a TRK in the vertical position at max acceleration. We calculated this to be about 135 lbf. This means that each bolt needs to exert a force of 67.5 lbf when fixtured. In our calculations we used a factor of safety and applied 100 lbf to each bolt. We also assumed a six inch TRK length which is much smaller than expected and increases the length of the overhanging beam. Using the area moment of inertia for the U bracket as well as the modulus of elasticity for aluminum, we found that the max deflection would be .0016 inches. As a comparison, we also calculated the deflection in a straight beam of 2"x.5" to be .029 inches. These are both small deflections but we want the smaller deflection to reduce any resonance. Appendix G shows these calculations.

Cost Analysis

Using the estimated cost of our shaker and head expander, we calculated a simple return on investment (ROI) using a present worth method. This method takes into account annual costs and initial costs and converts it to a single dollar amount based on the life span and an assumed interest rate. The data presented in Table 7 takes the upfront costs and operating costs of our proposed system and compares it to the operating costs of the currently outsourced testing. Note that the operating time is based on 10 testing iterations per year. Our analysis indicates that after only 2.9 years, our design would make up for the initial investment and start creating savings. Furthermore, we calculated that after 10 years, our system would create an ROI of $85,000 when compared to the current testing state. Our analysis does not take into account maintenance costs of the shaker or replacement costs for fixturing material.

Table 7. Cost comparison between outsourcing tests and purchasing a new system. The break even point for return on investment is about 2.9 years

Capital Costs	
Shaker/amp/blower/barrier	$ 26,400.00
18x18 head expander	$ 1,000.00
Controller	$ 10,500.00
Tax	$ 6,519.75
TOTAL	$ 44,419.75
Outsourcing costs	
Cost per test	$ 2,000.00

Constants		
Interest Rate	0.05	-
Life Span	2.9	years
Price of electricity	$ 0.1408	$/kWhr
Power consumption of shaker	15	kW
Tests per year	10	
Hours per test	150	

ROI	
P/A	2.638748503
Annual Cost of outsourcing	$ 20,000.00
Annual Cost of Proposed Design	$ 3,168.00
Present Worth of Outsourcing	$ 52,774.97
Present Worth of Proposed Solution	$ 52,779.31

The ROI analysis above is based on the total cost of our system. Table 8 below is the Bill of Materials for our system. It includes the price quoted to us by Spectral Dynamics as well as the

cost of materials need for the head expander and fixturing materials as quoted by McMaster-Carr. Appendix J contains data sheets for each piece we plan to purchase.

Table 8. Bill of Materials

Component	Item	Quantity	Unit Price	Total Price	Supplier
Shaker	SD-660-230M	1	$26,400	$26,400	Spectral Dynamics
	Lynx Controller	1	$10,500	$10,500	Spectral Dynamics
	Thermal Barrier	1	$540	$540	Spectral Dynamics
	Accelerometer	2	$400	$800	Spectral Dynamics
	Shipping	1	$3,000	$3,000	Spectral Dynamics
	Subtotal			$41,240	
Head Expander	18x18x5/8 aluminum plate	1	$204.81	$204.81	McMaster Carr
	4x6x3/8 aluminum plate	2	$62.85	$125.70	McMaster Carr
	4x8x3/8 aluminum plate	2	$69.58	$139.16	McMaster Carr
	6x6x1/2 aluminum plate	1	$15.41	$15.41	McMaster Carr
	6x6x6 1/2" aluminum tube	1	$69.36	$69.36	McMaster Carr
Fixturing	90 degree angle 4x6x3ft	1	$88.16	$88.16	McMaster Carr
	U channel 2x1x.5 5ft	1	$20.28	$20.28	McMaster Carr
	Press fit 5/16-18 (5)	5	$12.10	$60.50	McMaster Carr
	5/16-18 1" long bolts (50)	1	$10.07	$10.07	McMaster Carr
	5/16-18 4.5" long bolts (5)	2	$14.38	$28.76	McMaster Carr
	Shipping estimate	1	$200.00	$200.00	McMaster Carr
	Subtotal			$962.21	
Tax				$4,220.22	
Total				$46,422.43	

Chapter 5: Product Realization

After we received all of the materials from McMaster Carr, we began manufacturing the head head expander. The first part in building the head expander was cutting all the stock to the correct size and preparing the pieces that needed to be welded together. After this they were welded together and then finished with any final machining steps. These manufacturing processes were all completed in the machine shops on campus using various machines.

Ribs

The first manufacturing step was cutting the ribs that would give support to the head expander. We initially started with rectangular pieces of metal that would then be cut in half to form the triangular pieces. In order to get a straight cut on the vertical band saw, we created a jig out of wood that butted up against the flat edge parallel to the blade. This can be seen below in Figure 26. The jig allowed us to have a flat edge while cutting at the specified angle. The first rectangular piece of aluminum we cut was not straight because when we started the cut, the blade was trying to start on a sharp corner. This caused the blade to veer to one side and start the cut at the wrong place. To fix this, we ground the starting corner flat so the blade hit a flat plane. After this adjustment, the rest of the triangular ribs came out finished according to the drawings.

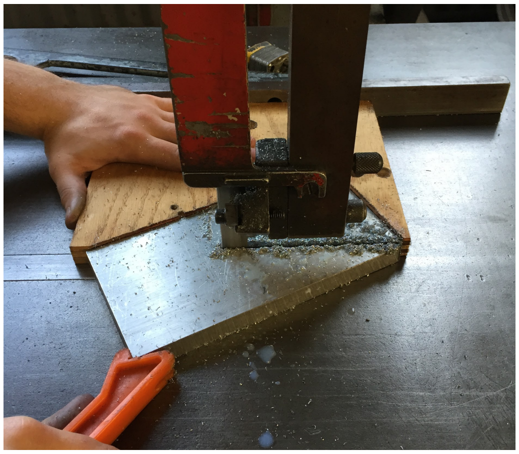

Figure 26. Cutting the triangular ribs on a vertical band saw

Bottom plate

The next step was drilling the holes of the bottom plate that would mount to the shaker's armature. This part is important because it will be mounting to a pre existing bolt pattern on the armature. If the holes are not in the correct place or slightly off, it would not bolt to the shaker. In order to drill these, we put the bottom plate on a mill to accurately find the center and corresponding bolt pattern. The setup can be seen in Figure 27. We started with a center drill to

start the hole and then stepped up to a quarter inch drill. We did not go to the very end size because we didn't know if there would be any warpage during the welding process.

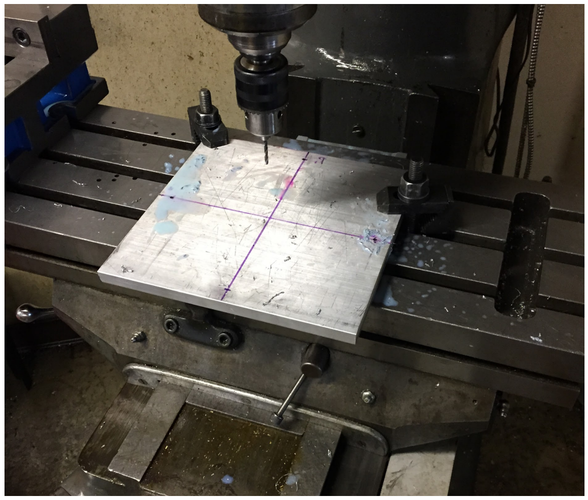

Figure 27. Drilling holes in bottom plate on mill

The next part of manufacturing came after talking to the head expander dealer. We found out that the armature retracts into the shaker so there couldn't be any overhanging parts bolted to the top. In order to fix this, we made the bottom plate circular and the same size as the armature. We first cut the corners off with a vertical bandsaw and then ground them down more precisely to the final size. This can be seen below in Figure 28.

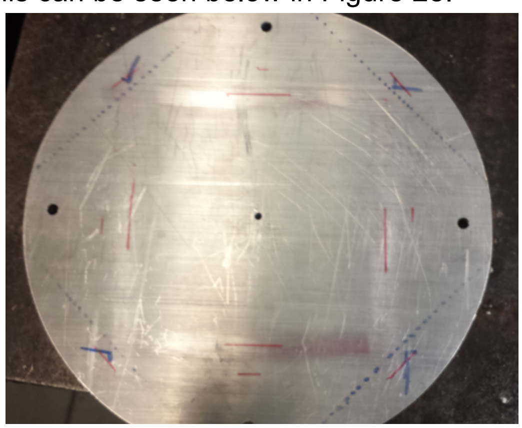

Figure 28. Final circular bottom plate

Center column

Manufacturing for the center column involved milling the corners so that the ribs could sit flat on the surface. This would make welding easier and improve the support that the ribs can give. Initially we secured the square tube to the table with a vice. Then we used a 45 degree tapered tool to make the edges flat. This was difficult to machine because there were a lot of step over calculations and the tool couldn't do the final cut in one pass. This led to the flats not being completely flat but a small tool mark path in the middle. To fix this we went with a different approach which can be seen in Figure 29.

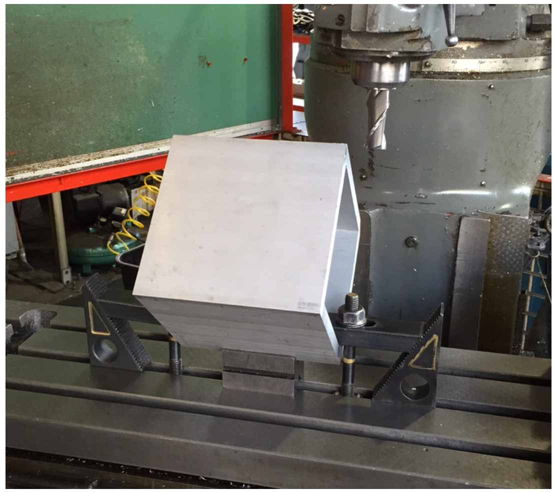

Figure 29. Second setup for milling corners flat

This new setup involved putting the square tube into a V-block and then clamping the edges to the table. Then we used a flat end mill to cut across the corner to the desired depth. This was much faster and gave a lot better finish. The final result can be seen in Figure 30.

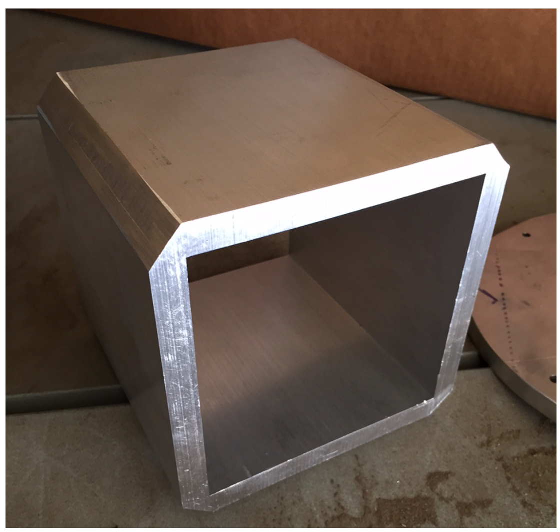

Figure 30. Final center column with flat edges

Top plate

The final manufacturing process we did before welding was drilling center holes in the top plate. The reason for this was because we didn't know if the plate would warp with all of the heat added to it during welding. We wanted the center holes to be drilled so we would have a starting point for drilling after the welding was completed. This way we would know where the holes were supposed to be. On challenge with this plate is that fact that it is so big. This made it hard to fit on any mills because none of them had enough travel to drill all of the holes at once. In order to fix this, we had to overhang the plate which can be seen in Figure 31. Then we drilled

half of the holes in the plate using the X axis with the majority of the travel. Then we flipped the plate over and drilled the other half of the holes.

Figure 31. Center holes drilled in top plate

L and U bracket

For the brackets that hold the TRK to the table, we had to drill holes to match the table pattern. We did these on a mill to ensure hole accuracy and used a P sized drill bit for clearance. One change we made on the L brackets was that we cut one inch off the side that mounts to the table using the vertical band saw. This was so that we could flip the vertical mounting orientation so the heavy mass is towards the center.

Welding

The welding process began on campus at Cal Poly. Kevin Williams in the Industrial & Manufacturing Engineering department, agreed to weld our project at no cost. Professor Williams was able to partially weld our project together, however due to the thickness of the material and the limitations of the welding equipment on campus, he was unable to complete the project after several weeks of attempting. To finish the welding, we took the head expander to Haener Metalworks in San Luis Obispo and they were able to complete the welding in just a couple of days at very low cost. The welding stage of our manufacturing delayed us several weeks, but thanks to Haener Metalworks, we were able to quickly move into testing of our design. Figure 32 shows the way the materials were layed up for welding. After welding was complete, it was noticed that the top plate had a slight downward curvature due to the heating and cooling involved in the process. This was not significant enough to affect the overall performance.

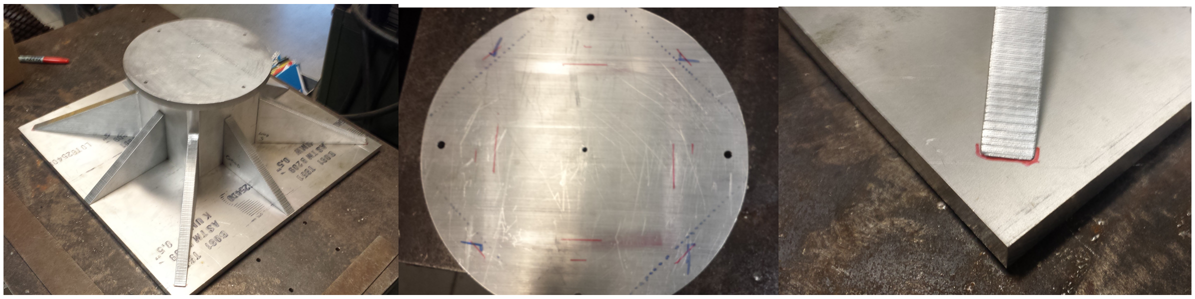

Figure 32. Welding assembly before delivering the parts to Kevin Williams. The center and right images show the bottom and top plates marked, respectively

Drilling Final Holes Sizes

Once welding was complete, we took the head expander back into the shop to drill the center holes on the top plate. Using a mill, we followed the same procedure that was used to drill the center holes which can be seen in Figure 33. We milled the holes to a 9/16" so that the press fits would fit properly. Before drilling this size, it was confirmed with a test piece to make sure it would work. The bolt holes for the bottom plate were also drilled to the correct size. We gradually stepped up to a size P drill to ensure that there was clearance for the bolt but not too much that it would slide around on the armature.

Figure 33. Final hole sizes in top plate

Press Fits

The final step for manufacturing was pressing the press fits into the holes drilled. This was challenging at first because the head expander is almost as large as the hydraulic press. To fix this, we pushed the press all the way to one side and rotated the head expander for each hole. This can be seen in Figure 34.We also put a bolt in the press fit so that it could be pressed flush with the table without any edges sticking out. Another issue was that when pressing, the head expander would tip over since the force is off center. To fix this we put supports under the table where the press was. This helped and allowed all the press fits to be pushed in successfully. Figure 35 shows the final product after all of the manufacturing.

Figure 34. Pushing press fits into top plate with hydraulic press

Figure 35 below shows the completed head expander after manufacturing. This does not include the fixture materials that bolt to the top plate.

Figure 35. Completed manufacturing of head expander

Chapter 6: Design Verification Plan

In order to verify our final design meets all the specifications of our sponsor, we need to test various aspects of our system. There a few sets of tests we need to run on the head expander. These are on the press fits, head expander, and the overall assembly and system.

Press Fits

The press fits needed to be fit that the hole sizing was correct, the forces could withstand testing, and there would be no corrosion.

Sizing

Before drilling the final hole sizes in the head expander, we wanted to verify that the hole size we drilled would work. The manufacturer suggests a 9/16" hole for the press fit so we tested this by drilling a hole in a spare piece of metal using the drill press. When we went to push the press fit in, it dropped through the whole because the hole was too big. Then we went down to a 17/32" to try again. This worked for the press fit but required a lot of force. After wondering why the manufacturers specification didn't work, we determined that the drill press chattered while we drilled which made the hole slightly larger than the drill size. To fix this we put the drill in a mill and this made it rigid enough so that the press fits worked in the 9/16" hole.

Forces

In order to make sure the press fits wouldn't fail during testing, we wanted to find the force required to remove them. After pressing a set of inserts into the aluminum, we put it under the hydraulic press to see the force as seen in Figure 36. There was a gage that read pressure that the cylinder was applying. The initial pressure required to push the insert out from the top was 2500 psi. Since the cylinder is 2.25 square inches, this pressure converts to 5625 lbs of force needed to remove. We also put the press fit back into that hole and tested a total of 5 more times. Even after it was removed and reinserted, the pressure only went down to 1500 psi or 3375 lbs. We also pushed from the bottom side to simulate pulling on the inserts. This yielded the same results as above which gave us the confidence that these inserts would not fail under testing.

Figure 36. Testing press fits on hydraulic press

Corrosion

In order to test corrosion between the aluminum and the steel inserts, we made a test sample. With this we put it in a bag with moisture and allowed it to sit in the sun to create a humid environment. After about a month of this corrosion indusive environment, we checked the final results. The test piece did not appear to have any differences between when we started and after this test which can be seen in Figure 37. Although this was not a fully intensive corrosion

test, we are confident that if the head expander is dried when not in use, corrosion will not be an issue.

Figure 37. Corrosion final results

Head Expander

The head expander needed to be tested to find the natural frequency of the design after manufacturing. Two different tests were run in order to verify that the natural frequency of our design was outside of the testing range of 0-1000 Hz. This was to ensure that the head expander did not invalidate the tests being run by adding to or amplifying the signal. Refer to Appendix K for plots from these tests.

Sine Sweep

The first test we ran was a sine sweep test using the hydraulic shaker in the vibrations lab on campus at Cal Poly. For this test, we created a wood plate to act as an intermediate between the hole pattern on the hydraulic shaker and the hole pattern on the bottom of our head expander. Figure 38 shows the assembly as we used it for testing. Using a sine sweep signal from a function generator input into the controller for the shaker, we generated a sine sweep from 0-1000 Hz and attached accelerometers to the top plate of the head expander to measure the response of the system.

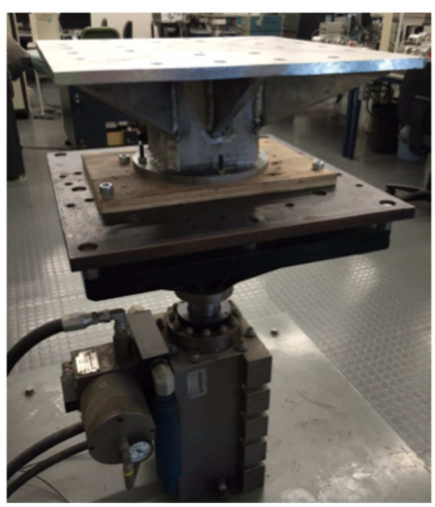

Figure 38. Testing the head expander with a sine sweep on the hydraulic shaker

Unfortunately, the results of this test were inconclusive. The inherent vibration modes of the hydraulic shaker's fixture as well as the vibration of the wood plate made it so any acceleration profiles we measured off of the top plate included the vibration of the shaker and the wood plate. This means that we could not accurately measure the response of our head expander with this setup. Due to a lack of time available to spend upgrading our testing rig, we decided to focus our time on another test.

Modal Hammer

The second test we performed was a modal hammer test. Using the modal hammer, accelerometer, and spectrum analyzer, we tested the free response of our head expander. To do this, we placed it on a concrete block so the setup of our test did not induce any additional vibrations. Then, we placed an accelerometer on the head expander to measure the frequency that the head expander would respond at. Then, to excite the head expander, we hit it with the modal hammer, pictured at right in Figure 39. The hammer has a force transducer that measures an input and triggers the spectrum analyzer to begin measuring on the accelerometer channel. With the viewing window set to show responses between 0 and 3200 Hz, we measure the response of our head expander by placing the accelerometer on the top plate in several positions and recording the frequency response. Measuring the response in several places ensures that the accelerometer is not place on a node, which would measure no acceleration in some modes.

The results we obtained from this test gave us a couple of things. The first result that we measured consistently between all tests was a very large acceleration peak in the range of 1100 Hz. This result reflected our SolidWorks simulation result that we expected and also satisfied our design requirement of no natural frequencies below 1000 Hz. This was the result we were hoping for and verifies our design and can be seen in Figure 41.

The second result we measured was less conclusive. Several of our tests show a natural frequency response in the range of 275-350 Hz. This peak only appeared in some of our tests and the magnitude of this peak was strongly dependent on the strength at which the hammer was used to excite the head expander. Stronger hits would cover the signal at the lower frequencies and the 1100 Hz response would dominate. This signal tells us that there may be some natural frequency modes that occur at lower frequencies. In our computer simulation, we also saw this mode around 300 Hz, but felt we could neglect it because it was not a vertical excitation, but a torsional excitation, about the central vertical axis of the head expander. This torsional mode cannot be ignored, and we talk about improvements that can be made in our conclusion.

Table 9 summarizes the values that were initially intended to be verified once the design was complete. Many of these values are listed as “N/A” because measuring them was not possible. The shaker system has not been ordered yet, so the only part of the design that could be measured is any part relating to the head expander.

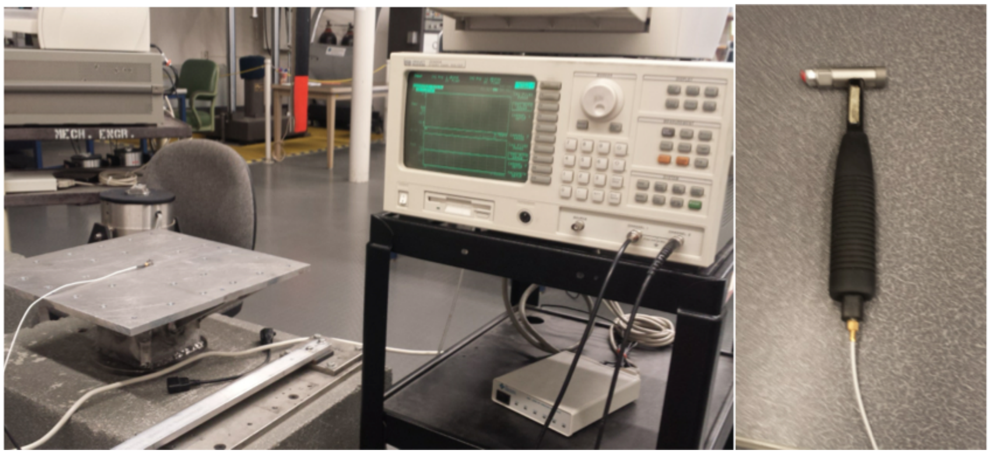

Figure 39. Setup for testing with the modal hammer

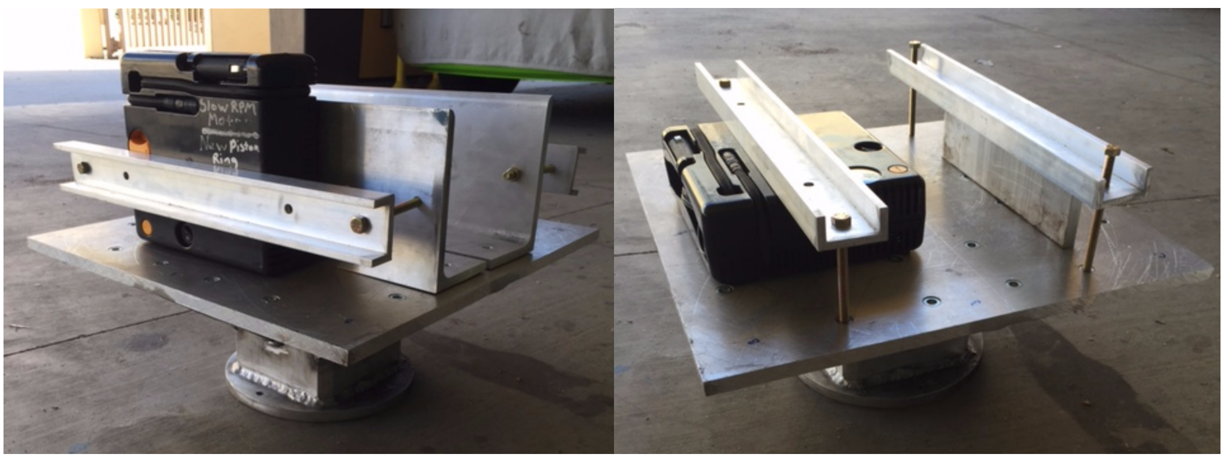
Figure 40. Final setup of head expander with both fixture orientations

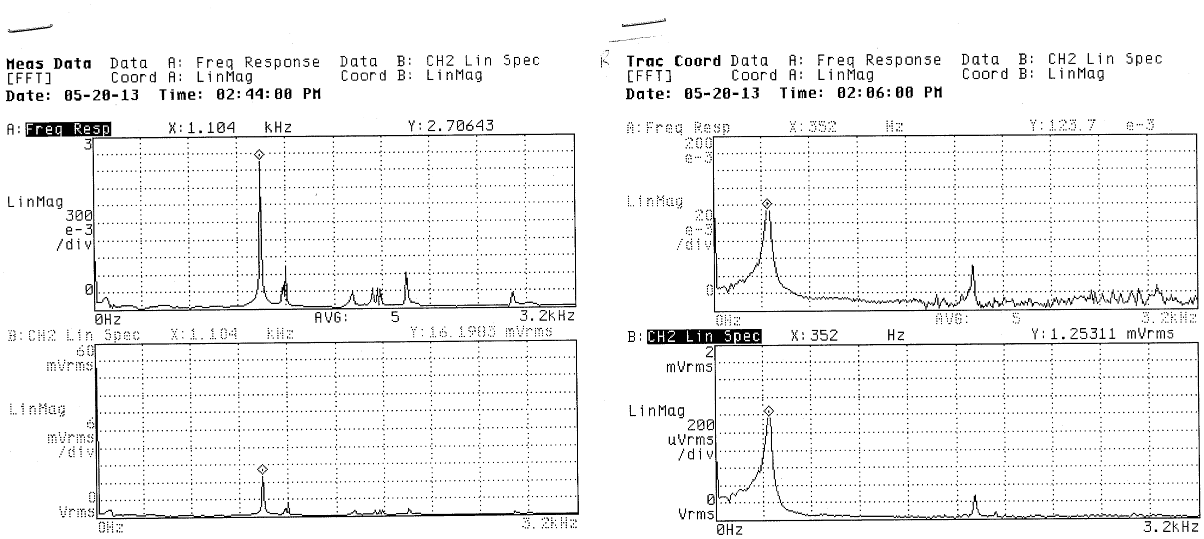

Figure 41. Frequency response from modal hammer. Left plot shows the desired 1100 Hz only response while the right shows noise at the lower frequencies.

Table 9. Specification Verification Checklist

Spec #	Parameter Description	Requirement or Target	Tolerance	Actual Value Tested
1	Testing Temperature	-40 ^{o}C to 70 ^{o}C	Min	N/A
2	Dimensions	3x7x7 feet	Max	N/A
3	Weight	2000 lbs	Max	1500 lbs
4	Vibration transmission	Less than tire test machine	Max	N/A
5	Testing Frequency	8-1000Hz	Min	1100 Hz
6	Shaker rating	10 lbs per TRK at 80.7 m/s^2	Min	N/A
7	# of TRKs to test simultaneously	1-3	Min	2
8	Testing Axes	3 axes	Min	3
9	Cost	$100,000	Max	$47,000
10	Vibration profiles	All (from OEM)	± 3dB, ±5% RMS	N/A
11	Run time	12 hours	Min	N/A
12	Setup time	30 minutes	±10 min	5 min

Chapter 7. Conclusions and Recommendations

The result of our analysis, manufacturing, and testing is that we have designed a complete system that can be used to test TRKs. The first part of our design is the Spectral Dynamic electrodynamic shaker which can provide the necessary forces and operate in the desired conditions. The other part is that we have designed a head expander than can withstand the vibrations that will be experienced in use. The system is also capable of handling up to two TRKs in any axis and can easily reconfigured quickly between tests. Through design verification, we established that there are possible major changes for our design that are described below.

Since there were some frequency responses in the lower range, we recommend a design modification to improve the performance of the frequency response. This can be done with supports between the ribs. We have talked with head expander manufacturers and they said that this is one step that can increase the natural frequency. Even though the vibration is only in the vertical direction, some off balance parts can cause vibrations in other planes to occur. In order to mediate this, the head expander needs to be stiffened in the torsional direction. This would entail adding material between between ribs to stiffen the design. This could be in the form of sheet metal or rods welded between the ribs.

Another significant change in the design is in the manufacturing process selected. In industry, head expanders are often cast as a single piece and not welded together from multiple plates. This process allows for a more complicated geometry that allows the designer to select a design more suited to controlling the natural frequency. For our project, casting a single head expander was unreasonable as welding is quicker and cheaper, but for the production of multiple, casting is preferred. If the head expander were to be improved in the future, contracting a cast version with an updated design to stiffen and reduce the overall weight.

This project has been beneficial in many to us as engineers. First off it has given a deeper insight into vibrational testing and all the aspects that need to be considered when testing. It has also given us the opportunity to step through the design process. From concept generation and ideation to manufacturing and testing, each step has given us first hand experience that we can apply in our future projects. We would like to thank Slime for the opportunity to work on this project and all the help from Professor Schuster and other faculty.

Appendices

Appendix A. Contacts

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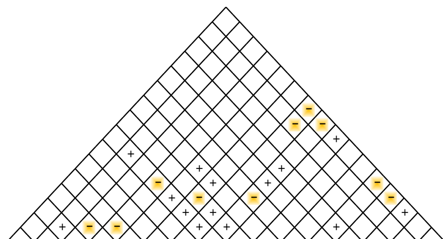
Appendix B. QFD

APPENDIX B QFD:
House of Quality

Correlations	
Positive	+
Negative	−
No Correlation	

Relationships	
Strong	●
Moderate	○
Weak	▽

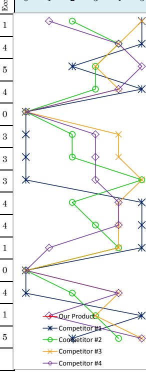
Direction of Improvement	
Maximize	▲
Target	◇
Minimize	▼



Column #	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16
Direction of Improvement	◇	▼	▼	▼	▲	▲	◇	▼	▲	◇	▼	▼	▼	◇	▼	▲

WHO: Customers — NOW: Current Product Assesment - Customer Requirements

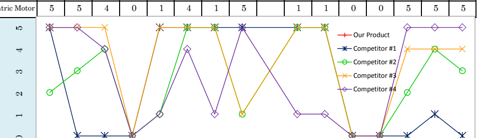
Row #	Weight Chart	Relative Weight	Slime	Our team	Lab Tech	Auto maker	Maximum Relationship	HOW: Engineering Specifications / WHAT: Customer Requirements (explicit & implicit)	Operates in temperature range	Dimensions	Weight	Vibration transmission	Frequency Range	Max Acceleration	Axes of opertation	Cost	Projected Life Span	Accepts any profile	Deviation from profile	Does it require custom parts?	# of manufacturing processes	Ability to be dissasembled	Setup time	Serviceability	Outsource testing	Servo-hydraulic	Electro-dynamic	Eccentric motor	Row #
1		13%	10	8	6	9	9	functional performance*	●			○	●	●	▽			▽	○						5	2	5	1	1
2		10%	8	6	5	8	9	reliability				▽					▽		●			▽		●	5	4	4	4	2
3		4%	7	2	0	4	9	cost		▽	▽	▽	▽	●	●	●	▽	○	○	○	○	▽		○	2	3	3	5	3
4		5%	7	4	3	0	9	longevity				▽		▽	○		●			▽				●	5	3	4	4	4
5		7%	7	5	7	0	9	vibration transmission			○	●	○	●	▽										0	0	0	0	5
6		4%	8	6	0	0	9	size		●	○			○	○										0	2	4	3	6
7		4%	8	7	0	0	9	weight		○	●	○		○	○										0	2	4	3	7
8		4%	5	4	3	0	1	aesthetics		▽					▽					▽	▽				0	5	5	3	8
9		9%	8	5	2	8	9	temperature range*	●																5	2	4	4	9
10		7%	4	6	8	0	9	programibility					○		▽			●							5	3	4	4	10
11		8%	5	3	5	7	9	versatility					●	○				●							5	4	4	1	11
12		4%	3	9	0	0	9	manufacturing ease		▽	▽	○	▽	▽	●		▽		▽	●	●	▽			0	0	0	0	12
13		3%	6	6	0	0	9	ease of transportation		○	●	▽			○							●			0	2	4	4	13
14		10%	7	3	7	7	9	3 axes of testing							●								▽		5	3	3	1	14
15		8%	9	3	3	7	9	Testing rate							●			▽	▽				●		2	4	5	5	15
16		0%																											16



	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16
HOW MUCH: Target	neg 40 to +70 C	3'x7'x7'	500 lb	less than current device	10-1000 Hz	80.7 m/s^2	3	$80,000	10 years	yes	3dB	no	minimize	yes	30 minutes	easy
Max Relationship	9	9	9	9	9	9	9	9	9	9	9	9	9	9	9	9
Technical Importance Rating	190.9	70.21	109.3	146.5	237.5	272.3	312.8	39.29	59.39	171.3	156.9	54.61	50.07	49.25	83.66	148.1
Relative Weight	9%	3%	5%	7%	11%	13%	15%	2%	3%	8%	7%	3%	2%	2%	4%	7%
Weight Chart																

Current Product Assesment - Engineering Specifications

	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16
Outsourcing	5	0	0	0	5	5	5	5		5	5	0	0	0	1	0
Servo-hydraulic	2	3	4	0	1	5	5	1		5	5	0	0	2	4	3
Electro-dynamic	5	5	5	0	5	5	5	1		5	5	0	0	4	4	4
Eccentric Motor	5	5	4	0	1	4	1	5		1	1	0	0	5	5	5



Column # 1 2 3 4 5 6 7 8 9 10 11 12 13 14 15 16

Template Revision: 0.9 Date: 4/23/2010
Christopher Battles

Appendix C. OEM Test Specifications

Company 1

Procedure:
Mount samples to a rigid fixture using motor attachment points & recommended fasteners to the lower tolerance of designed torque values. Mark fasteners & record torque values. Ensure the fixture does not have any resonance below 300 Hz. Between 5 & 500 Hz, there shall be no transmissibility less than 0.1 or greater than 10.
Install representative connector & cable harness, as necessary.
Subject unit to vibration test level III per DIN IEC 68-2-34 Test FD, equivalent to DIN 40 046 part 23 with the following exceptions:
- DIN level III (see DVO 14491) *DIN level II
- Form of Spectrum *Form of Spectrum w/ effective acceleration: 14.1 m/s2
10 Hz 9.68 (m/s2)2 / Hz 4.84 (m/s2)2 / Hz
300 Hz 0.326 (m/s2)2 / Hz 0.163 (m/s2)2 / Hz
1000 Hz 0.0296 (m/s2)2 / Hz 0.0148 (m/s2)2 / Hz
- The Spectral Density Profile (DVO 14491) shall be maintained within +/- 3 dB. The RMS of the profile shall be maintained within +/- 5%.
- Effective value of acceleration: 20.0 m/s2.
- Acceleration point: Interface between test sample & test fixture.
- Test is to be conducted in the three main axes (x,y,z).
- Duration of vibration in each axis: 8 h. Total test duration is 24 hours.

Acceptance Criteria:
Motor shall be free from mechanical or electrical damage. Damage is broadly defined as, but not limited to, any kind of failure ranging from visible cracks to obvious factures. Such damage includes faults that can induce either mechanical or electrical failure of the motor (i.e. loose magnets or end plates, noisy bearings, mounting flange movement, etc.)
Motor shall not trip circuit breaker during test.
All test samples must complete compressor's functional test.

Company 2

Noise & Vibration	All configurations and operating positions of the TSCK shall have an Instationary Zwicker Loudness ≤ 5 sones N10 when evaluated in the Vertical, Fore-Aft and Lateral directions using the input frequencies and amplitudes defined below					
	Vertical Acceleration		Fore/Aft Acceleration		Lateral Acceleration	
	Freq (Hertz)	Accel ((m/s2)2/Hz)	Freq (Hertz)	Accel ((m/s2)2/Hz)	Freq (Hertz)	Accel ((m/s2)2/Hz)
	8	0.91778	8	0.09397	8	0.2888
	12	2.24356	11	0.36561	12	1.80539
	15	1.73925	16	0.6406	19	0.65142
	20	0.22577	26	0.0459	24	0.05051
	25	0.25211	39	0.04682	48	0.08065
	48	0.08097	67	0.01669	76	0.0047
	65	0.09001	97	0.00186	100	0.00011
	82	0.00635	100	0.00057	m/s2 RMS	3.82
	100	0.00058	m/s2 RMS	2.6		
	m/s2 RMS	4.7				

Company 3

Test conditions:
- Actual bottle assy + sealant volume must be used during test
- Dispose the bottle horizontally on the jig, as illustrated in fig.2
- Vibration direction: along the longitudinal axis of the bottle (highest dimension, 180mm)
- Acceleration: 59,8m/ s² (Frequency: 11Hz)
- 1,000,000 cycles
- Temperature: -45°C, room temperature, 80°C

Total: 1 direction x 3 temperatures = 3 tests
After test completion, cool the bottle at room temperature, in same horizontal configuration, for 24hrs
Shall fulfill general durability criteria

Company 4

Test	Description	Standard or Test Condition
Vibration Durability	Frequency	10~200Hz 15 min sweep
	Acceleration	21.6 m/s^2 (2.2G)
	Vibration Duration	4 hours each for up and down, back and forth, and left and right directions

Company 5

Frequency [Hz]	Power spectral density (PSD) [(m/s2)2/Hz]	Power spectral density (PSD) [G2/Hz]
10	20.00	0.2080
55	6.50	0.0676
180	0.25	0.0026
300	0.25	0.0026
360	0.14	0.0015
1000	0.14	0.0015

RMS acceleration 27.78 m/s2 2.84 GRMS
12 Hours / Axis

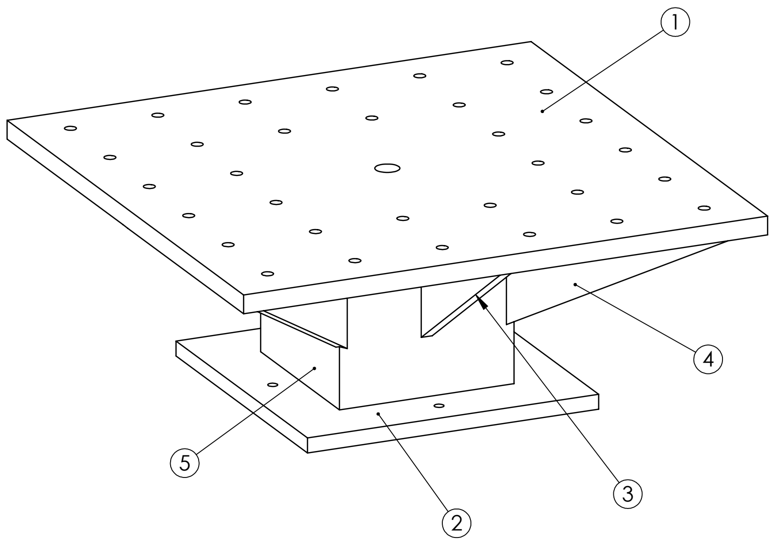
Acceptance Criteria
Performs as designed during and after the test. Meets all functional requirements before & after the test. No damage/BSR, no functional or performance degradation, no undesired or intermittent operations shall be allowed. Part shall not have any signs of physical deformation, burn marks, etc

Appendix E. Bruel & Kjaer head Expander Design

"Fixtures for Bruel and Kjaer Vibration Exciters." (n.d.): n. pag. Bruel and Kjaer. Web.
<http://www.bksv.com/doc/BO0179.pdf>

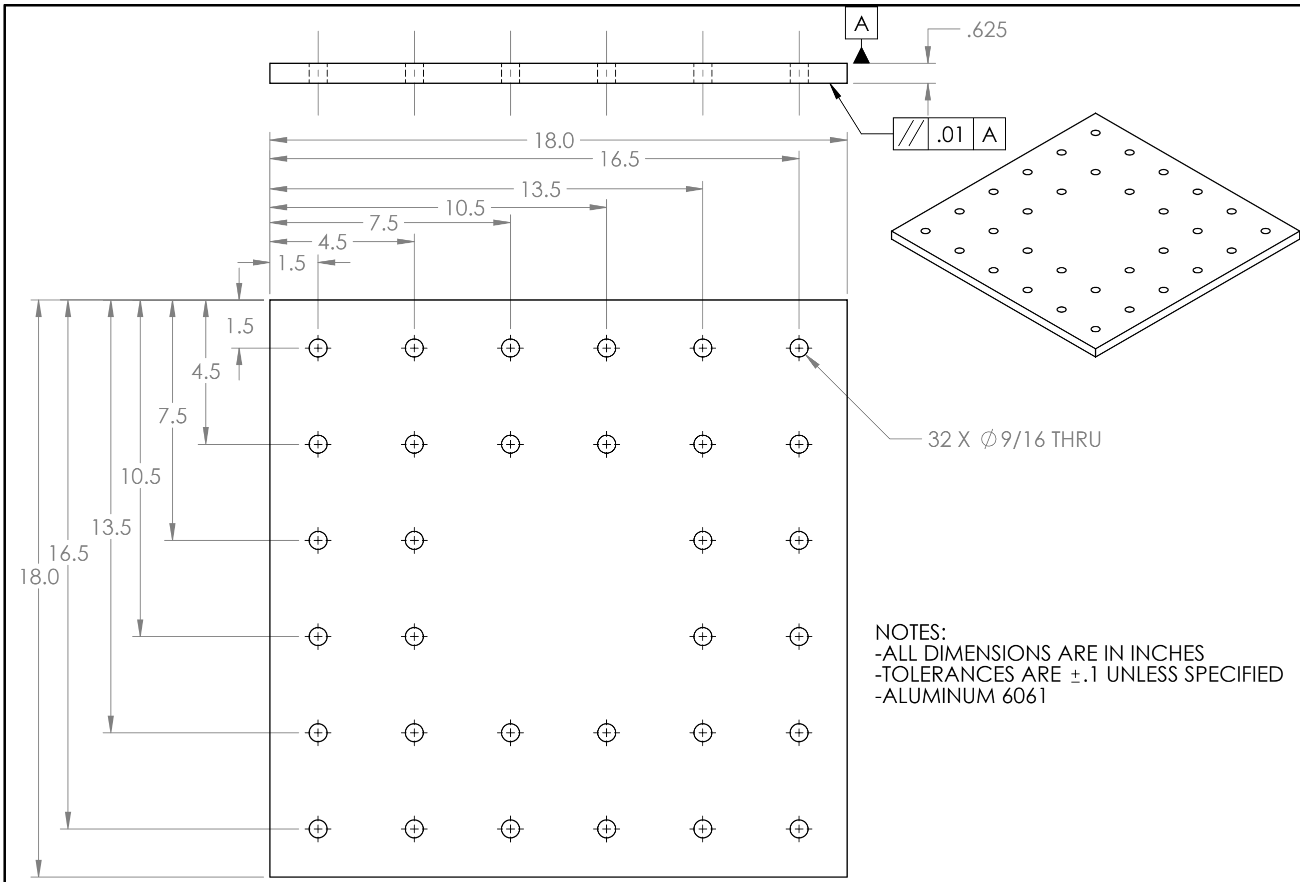
Appendix F: Part Drawings

ITEM NO.	PART NUMBER	DESCRIPTION	QTY.
1	SA001	TOP PLATE	1
2	SA002	BOTTOM PLATE	1
3	SA003	SIDE RIB	4
4	SA004	CORNER RIB	4
5	SA007	6X6X6 COLUMN	1



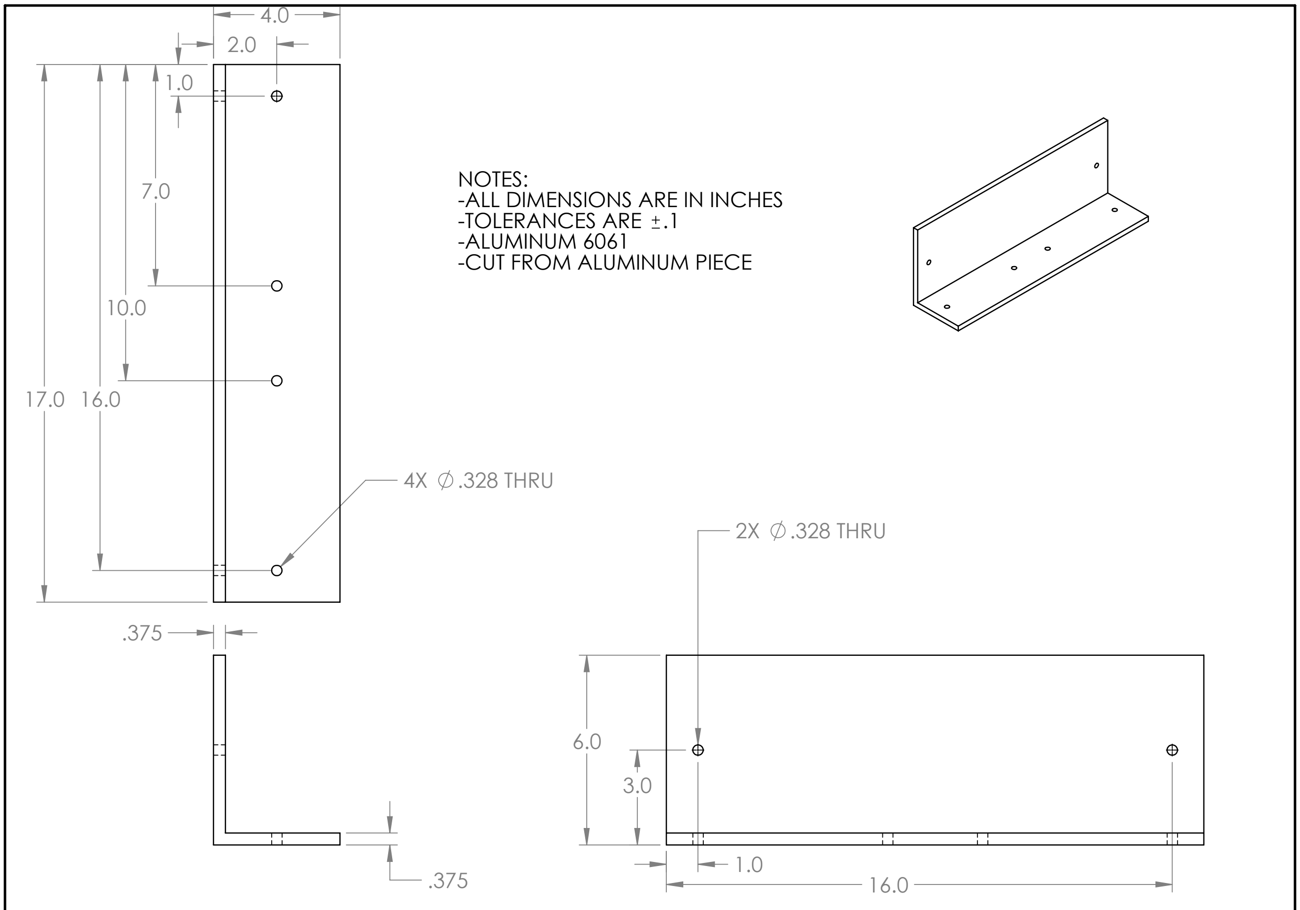
NOTES:
-ALL DIMENSIONS ARE IN INCHES
-TOLERANCES ARE ±.1
-ALUMINUM 6061

Cal Poly Mechanical Engineering	Quantity: 1	Inches	Title: HEAD EXPANDER SIDE RIB		Drwn. By: DYLAN DE CAUSSIN
RNG Senior Project	ASB. #:HE001	Nxt Asb:	Date:2/3/15	Scale: 1:2	Chkd. By:

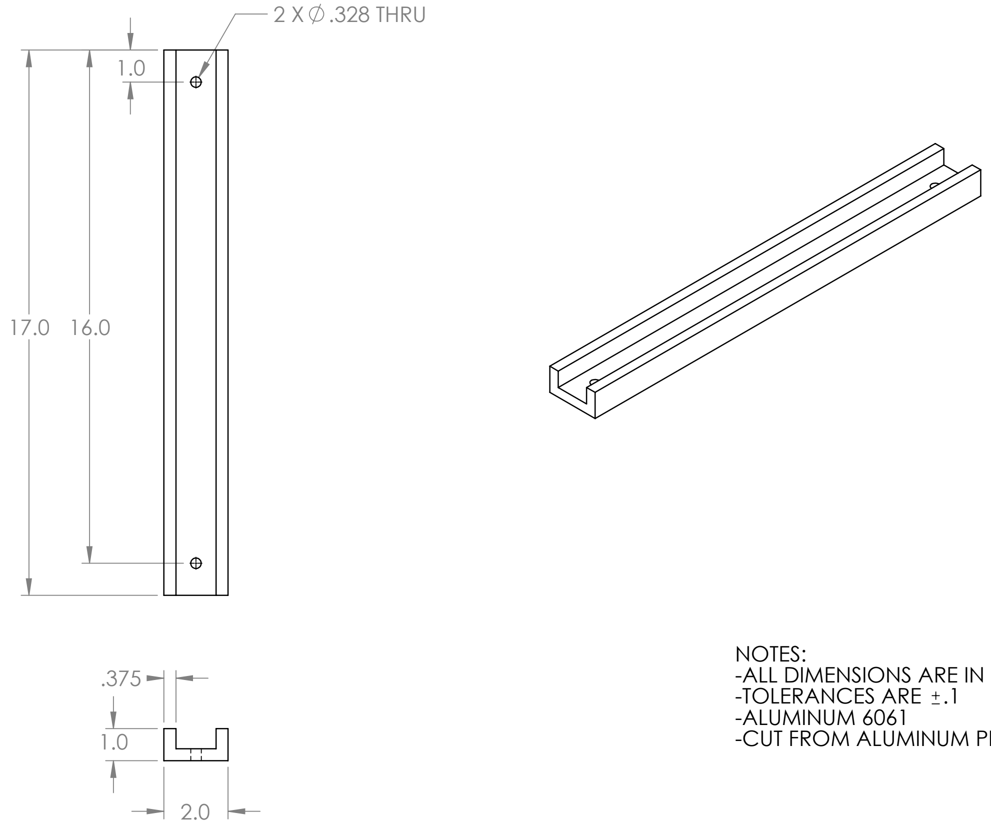


NOTES:
-ALL DIMENSIONS ARE IN INCHES
-TOLERANCES ARE ±.1 UNLESS SPECIFIED
-ALUMINUM 6061

Cal Poly Mechanical Engineering	Quantity: 1	Inches	Title: HEAD EXPANDER TOP PLATE		Drwn. By: DYLAN DE CAUSSIN
RNG Senior Project	Dwg. #: SA001	Nxt Asb: HE001	Date:2/3/15	Scale: 1:4	Chkd. By:

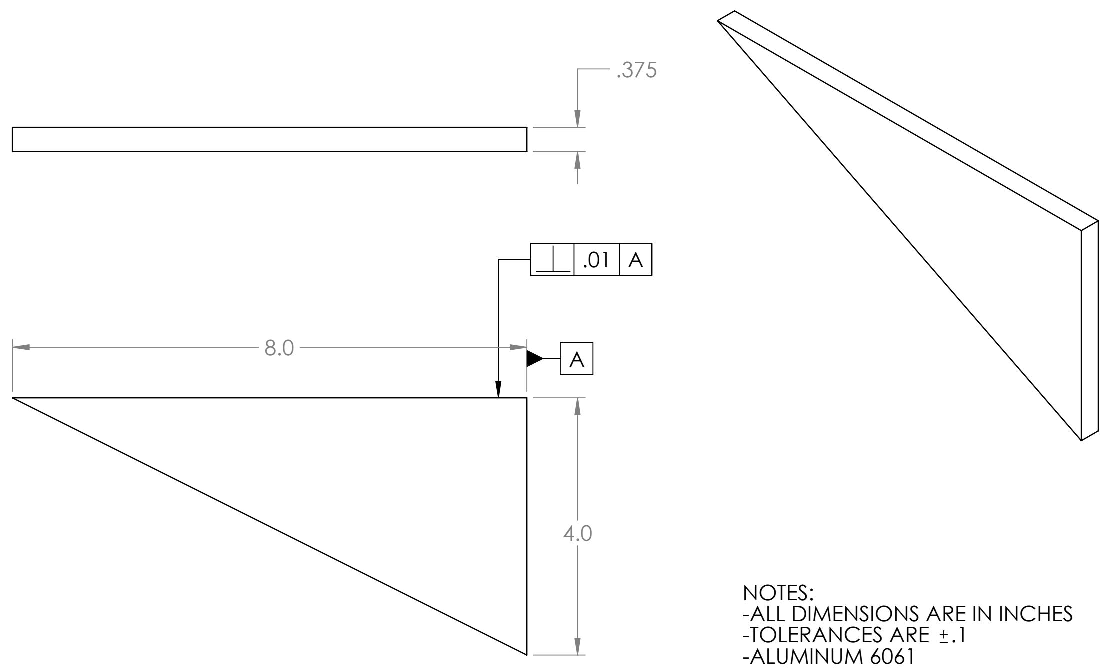


Cal Poly Mechanical Engineering	Quantity: 2	Inches	Title: L BRACKET		Drwn. By: DYLAN DE CAUSSIN
RNG Senior Project	Dwg. #: SA005	Nxt Asb: HE001	Date:2/3/15	Scale: 1:4	Chkd. By:



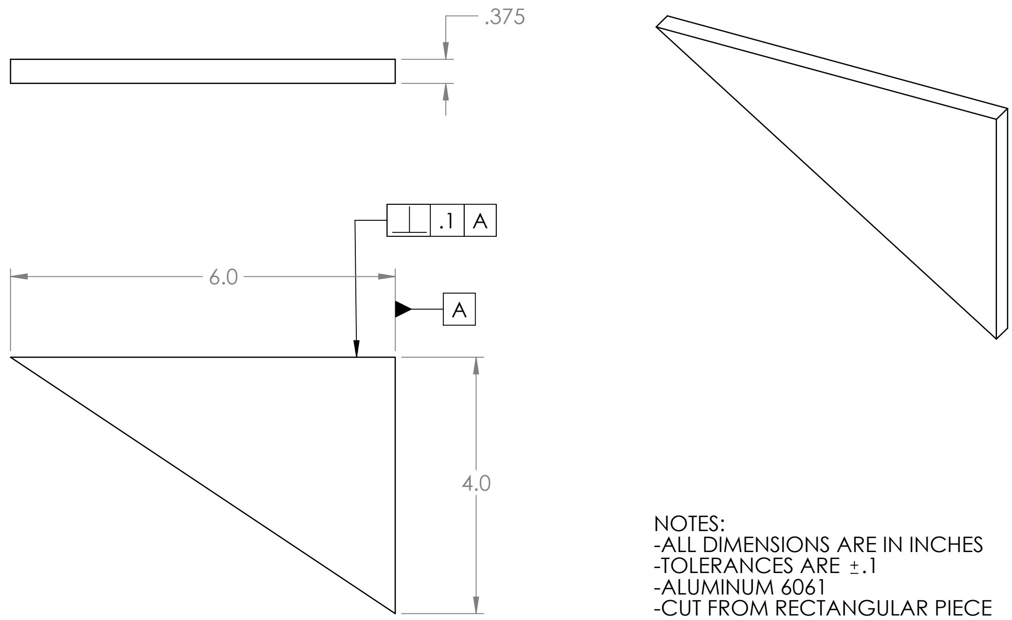
NOTES:
-ALL DIMENSIONS ARE IN INCHES
-TOLERANCES ARE ±.1
-ALUMINUM 6061
-CUT FROM ALUMINUM PIECE

Cal Poly Mechanical Engineering	Quantity: 2	Inches	Title: U BRACKET		Drwn. By: DYLAN DE CAUSSIN
RNG Senior Project	Dwg. #: SA006	Nxt Asb: HE001	Date:2/3/15	Scale: 1:4	Chkd. By:



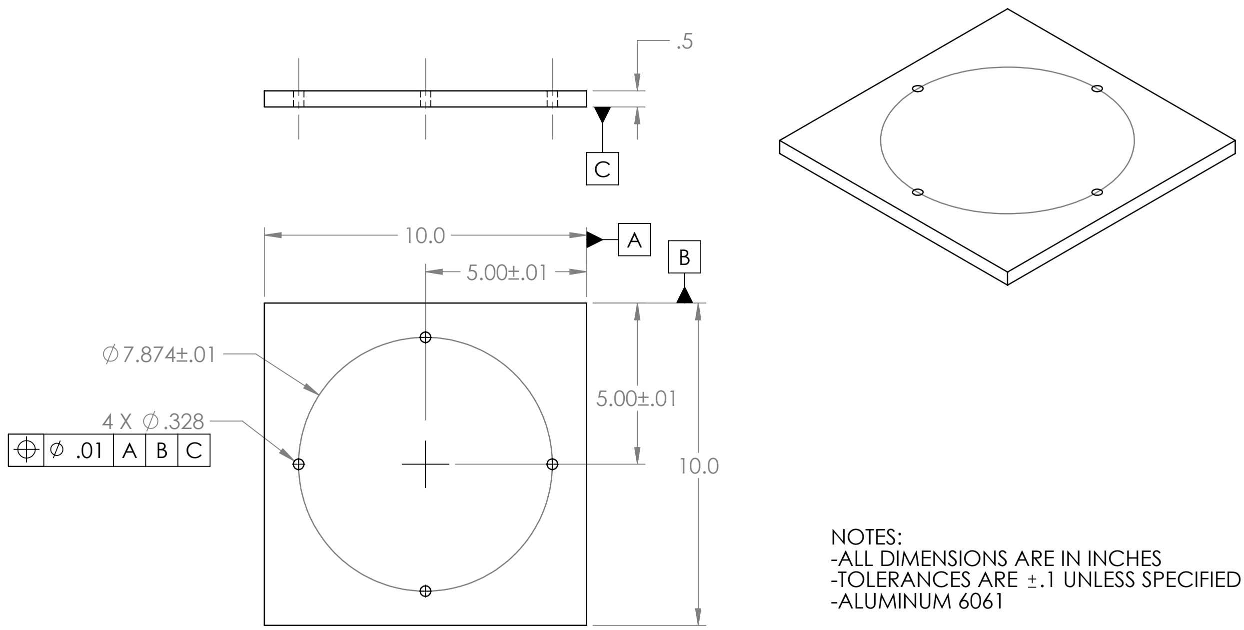
NOTES:
-ALL DIMENSIONS ARE IN INCHES
-TOLERANCES ARE ±.1
-ALUMINUM 6061
-CUT FROM RECTANGULAR PIECE 4X8

Cal Poly Mechanical Engineering RNG Senior Project	Quantity: 4	Inches	Title: HEAD EXPANDER CORNER RIB		Drwn. By: DYLAN DE CAUSSIN
	Dwg. #: SA002	Nxt Asb: HE001	Date:2/3/15	Scale: 1:2	Chkd. By:



NOTES:
-ALL DIMENSIONS ARE IN INCHES
-TOLERANCES ARE ±.1
-ALUMINUM 6061
-CUT FROM RECTANGULAR PIECE

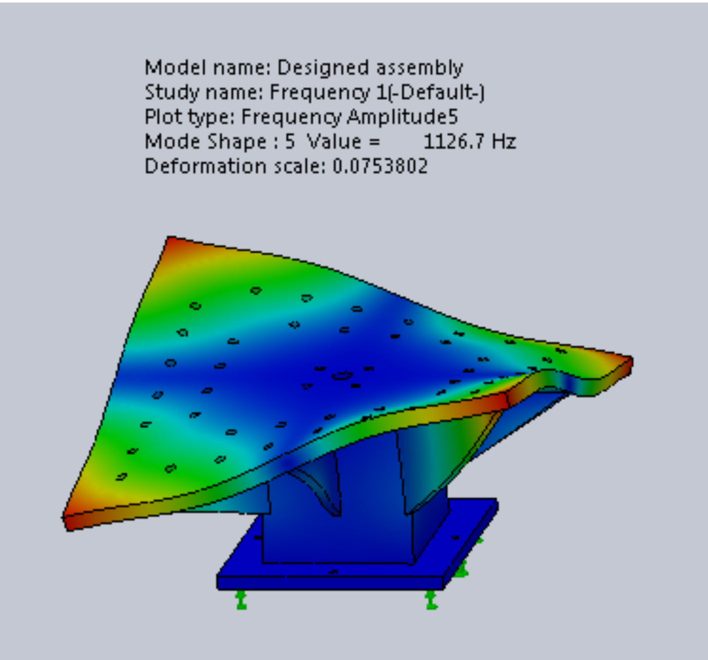
Cal Poly Mechanical Engineering	Quantity: 4	Inches	Title: HEAD EXPANDER SIDE RIB		Drwn. By: DYLAN DE CAUSSIN
RNG Senior Project	Dwg. #: SA003	Nxt Asb: HE001	Date:2/3/15	Scale: 1:2	Chkd. By:



NOTES:
-ALL DIMENSIONS ARE IN INCHES
-TOLERANCES ARE ±.1 UNLESS SPECIFIED
-ALUMINUM 6061

Cal Poly Mechanical Engineering	Quantity: 1	Inches	Title: HEAD EXP. BOTTOM PLATE		Drwn. By: DYLAN DE CAUSSIN
RNG Senior Project	Dwg. #: SA004	Nxt Asb: HE001	Date:2/3/15	Scale: 1:4	Chkd. By:

Appendix G. Analysis



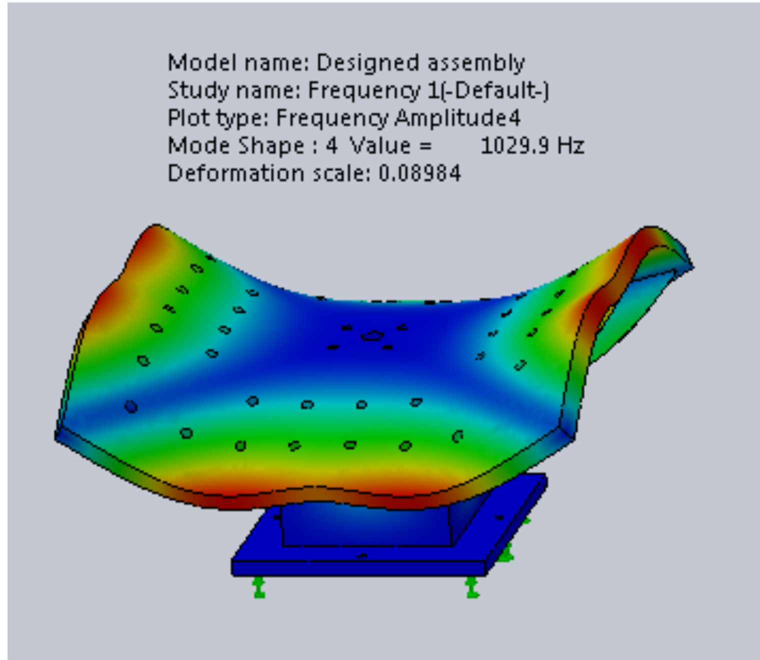


Figure G1. Results of SolidWorks frequency simulation

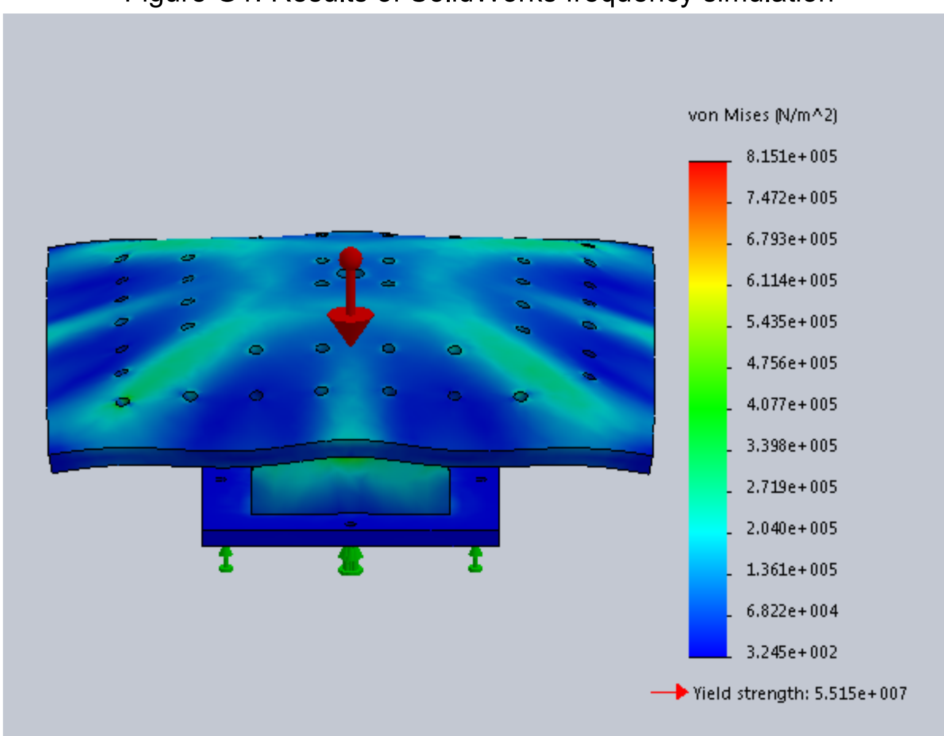


Figure G2. SolidWorks stress simulation results for a 1000 N distributed load showing a maximum stress of 8.15 x 10^5 Pascals

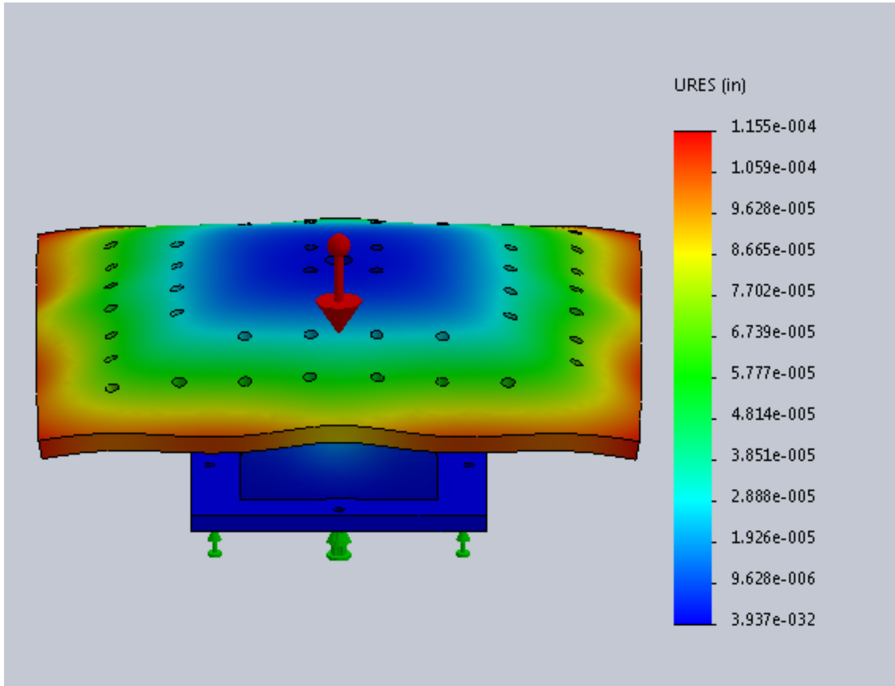


Figure G3. SolidWorks deflection results for a 1000 N distributed load showing a maximum deflection of 0.0001 inches

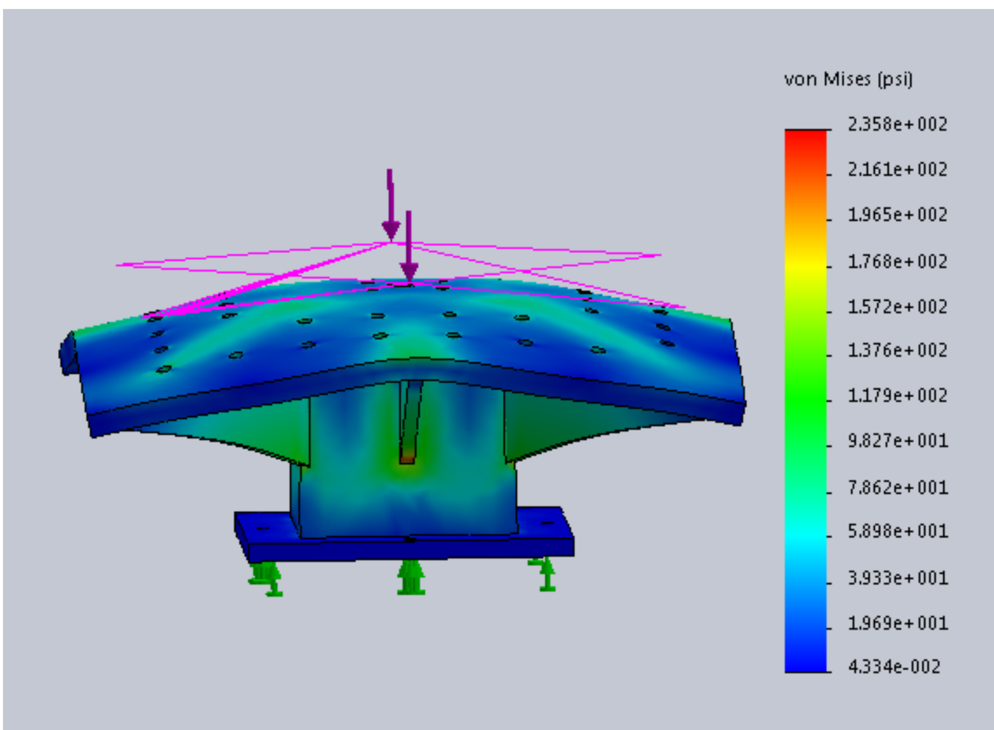


Figure G4. SolidWorks stress simulation results for two 1000 N point loads showing a maximum stress of 235 psi.

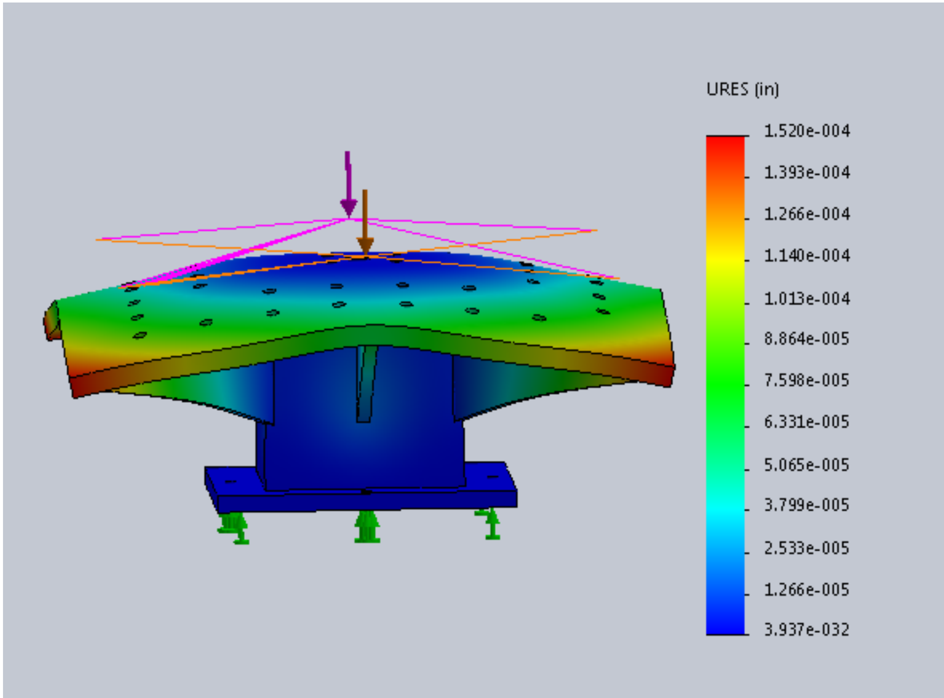
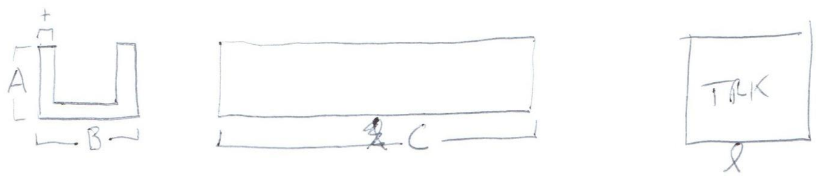
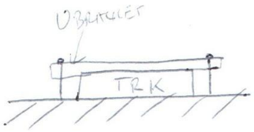


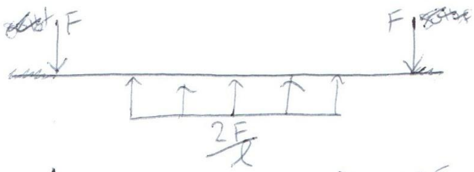
Figure G5. SolidWorks deflection simulation results for two 1000N point loads showing a maximum deflection of 0.0001 inches.

HAND CALCS FOR BEAM DEFLECTION

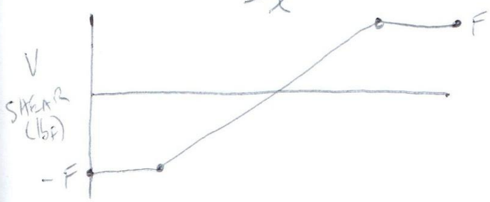


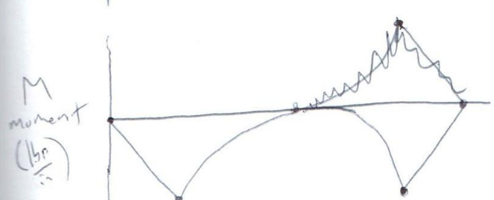
U BRACKET IS CLAMPING TRK TO TABLE





ASSUMING BOLT FORCES ARE AT END OF BEAM





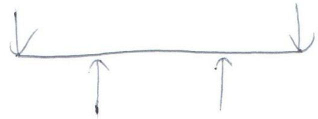
U BRACKET

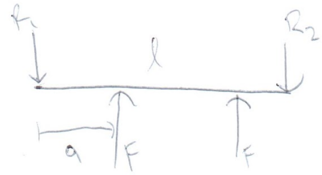
$$I = \frac{2sb^3 + ht^3}{3} - A(b-y)^2 = \frac{2tA^3 + (B-2t)t^3}{3} - (2At + Bt - 2t^2)(A - A)^2$$

E = 10.4 Mpsi (ALUMINUM)

A = 1 in
B = 2 in
t = .5 in
I = .375 in^4

ASSUMING SIMPLE SUPPORT W/ TWIN LOADS SINCE IT WILL CURVE





$Y_{max} =$ ~~[illegible]~~

$$Y_{at\ bolts} = \frac{Fa^2}{6EI}(2a+3l)$$

$$= \frac{F(3\,in)^2}{6(10.4\times10^6\,psi)(.375\,in^4)}\left(2(3\,in)+3(12\,in)\right)$$

$$= F\left(1.615\times10^{-5}\,\frac{in}{lb}\right)$$

ASSUMING
6 inch TRK (small)
$l = 12$ inch
$F = 1000$ lb (way over force)

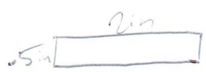
$$\underline{Y_{at\ bolts} = .016\,in \ @\ 1000\,lbs}$$

IF A FORCE OF 1000 lb is CREATED BY BOLTS, IT WILL DEFLECT .016 in.

2000 lb clamping

THIS IS FOR A ~~[illegible]~~ BEAM WITH OVERHANGING SUPPORTS AND TWO EQUAL SUPPORT LOADS

FOR STRAIGHT BEAM



$$I = \frac{bh^3}{12} = .0208\,in^4$$

EVERYTHING ELSE THE SAME

$$Y_{at\ bolt} = \left(2.91\times10^{-4}\,\frac{in}{lb}\right)F$$

A FORCE OF 1000 lb deflects .29 in
100 lb deflects .029 in

~~HOLE~~ PRESS FIT CALCS 1/27/15

FOR THREADED INSERT "YARDLEY #37516QSI-7"

PLATED STEEL

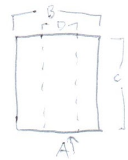
A THREAD: $\frac{3}{8}$-16

B OUTSIDE ⌀: $d_o = .580$ in $r_o = .290$ in

C LENGTH: $l = .380$ in

D INNER ⌀: $.391$ in $r_i = .1955$ in

STARTING HOLE SIZE: .531 in ≈ 13.5 mm = .5315 in



$R = \frac{.531\text{ in}}{2} = .2655$ in

$\delta = \frac{.580 - .531}{2} = .0245$ in USE DIAMETERS INSTEAD

$r_i = .1955$ in

$D = .531$ in

$\delta = .049$ in

$d_i = .391$ in

$d_o = \infty$ LARGE COMPARED TO OTHERS

aluminum $\nu_o = .333$

$\nu_i = .292$ ASSUMING CARBON STEEL (ZINC PLATING NEGLIGIBLE)

$E_o = 10.4 \times 10^6$ psi

$E_i = 30 \times 10^6$ psi

$$P = \frac{\delta}{D\left[\frac{1}{E_o}\left(\frac{d_o^2 + D^2}{d_o^2 - D^2} + \nu_o\right) + \frac{1}{E_i}\left(\left(\frac{D^2 + d_i^2}{D^2 - d_i^2}\right) - \nu_i\right)\right]} \quad (3\text{-}56)$$

$$= \frac{.049\text{ in}}{.531\text{ in}\left[\frac{1}{10.4\times10^6 \frac{lb}{in^2}}\left(1 + .333\right) + \frac{1}{30\times10^6 \frac{lb}{in^2}}\left(\left(\frac{.531^2 + .391^2}{.531^2 - .391^2}\right) - .292\right)\right]}$$

$$= \frac{.049\text{ in}}{.531\text{ in}\left[1.28\times10^{-7} + 1.03\times10^{-7}\right]\frac{in^2}{lb}}$$

$\boxed{P = 399.5\text{ Ksi}}$ PRESSURE BETWEEN FIT

TANGENTIAL STRESS IN INNER MEMBER (STEEL)

$$\sigma_t = -P\frac{R^2 + r_i^2}{R^2 - r_i^2} = -P\frac{D^2 + d_i^2}{D^2 - d_i^2} \quad (3\text{-}58)$$

3.368

$$= -399.5\text{ Ksi}\left(\frac{.531^2 + .391^2}{.531^2 - .391^2}\right)$$

$\boxed{\sigma_{t_i} = -1345.8\text{ Ksi}}$

TANGENTIAL STRESS IN OUTER MEMBER (ALUMINUM)

$$\sigma_t = p\,\frac{d_o^2 + D^2}{d_o^2 - D^2} \quad (3\text{-}59)$$

$$= 399.5\,\text{ksi}\left(\frac{\infty}{\infty}\right)$$

$$\boxed{\sigma_{t_o} = 399.5\,\text{ksi}}$$

RADIAL STRESS IN INNER MEMBER AND OUTER

$$\sigma_r = \frac{p_i r_i^2 - p_o r_o^2 + r_i^2 r_o^2 (p_o - p_i)/r^2}{r_o^2 - r_i^2}$$

Since $p_o = 0$

$$\sigma_r = \frac{r_i^2 p_i}{r_o^2 - r_i^2}\left(1 - \frac{r_o^2}{r^2}\right)$$

$$= \frac{(.1955\,\text{in}^2)(399.5\,\text{ksi})}{\infty^2 - .1955\,\text{in}^2}\left(1 - \frac{\infty^2}{.2655\,\text{in}^2}\right)$$

$$\sigma_r = \frac{r_i^2 p_i}{r_o^2 - r_i^2}\left(\frac{r^2 - r_o^2}{r^2}\right)$$

$$= \frac{r_i^2 p_i}{r^2}\left(\frac{r^2 - r_o^2}{r_o^2 - r_i^2}\right)\frac{1/r_o^2}{1/r_o^2} =$$

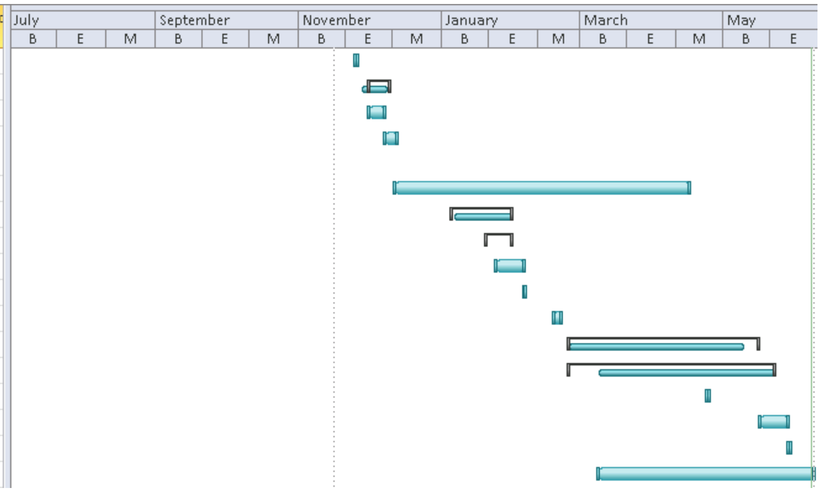
$$= \frac{r_i^2 p_i}{r^2}\left(\frac{\frac{r^2}{r_o^2} - 1}{1 - \frac{r_i^2}{r_o^2}}\right)$$

$$\boxed{\sigma_r = 399.5\,\text{ksi}}$$

CONCLUSION: THE TANGENTIAL STRESS IS MUCH HIGHER THAN THE YIELD OF ALUMINUM. THESE INSERTS MAY BE LESS PRESS FIT AND MORE SHEAR FORCES. SINCE THE INSERTS HAVE GROOVES, THEY SHEAR INTO THE METAL AND STICK IN THAT WAY.

Appendix H. Gantt Chart

	Task Mode	Task Name	Duration	Start	Finish	Pred
1		Preliminary Design Review	1 day	Tue 11/25/14	Tue 11/25/14	
2		**Detailed shaker analysis**	**7 days**	**Mon 12/1/14**	**Tue 12/9/14**	
5		Review vendor options	6 days	Mon 12/1/14	Sun 12/7/14	
6		Order shaker/controller/amplifier	5 days	Mon 12/8/14	Fri 12/12/14	
7		Shaker lead time	18 wks	Fri 12/12/14	Thu 4/16/15	
8		**Detailed analysis**	**20 days**	**Mon 1/5/15**	**Fri 1/30/15**	
17		**Preliminary Testing**	**9 days**	**Tue 1/20/15**	**Fri 1/30/15**	
20		Critical Design Report	9 days	Sat 1/24/15	Wed 2/4/15	
21		Critical Design Review	1 day	Thu 2/5/15	Thu 2/5/15	
22		Order fixture materials	3 days	Wed 2/18/15	Fri 2/20/15	
23		**Manufacturing**	**59 days**	**Tue 2/24/15**	**Fri 5/15/15**	
30		**Testing**	**64 days**	**Tue 2/24/15**	**Fri 5/22/15**	
35		Cal Poly safety review	1 day	Fri 4/24/15	Fri 4/24/15	
36		Prepare for expo	10 days	Sun 5/17/15	Thu 5/28/15	
37		Senior Design Expo	1 day	Fri 5/29/15	Fri 5/29/15	
38		Final design report	66 days	Mon 3/9/15	Mon 6/8/15	



Appendix I. Officials Purchasing Quotes

McMaster-Carr quote:

McMASTER-CARR

Date	February 2, 2015
Purchase order	head expander
Order created by	Dylan de Caussin (dylan@decaussin.org)

Ship to
Slime
2260 Loomis St.
San Luis Obispo, CA 93405

Shipping method
Ground

Send invoice to
dylan@decaussin.org

Payment method
Open Account

Line	Quantity	Product		Ships	Unit price	Total
1	1 each	89155K37	Oversized Multipurpose 6061 Aluminum, 5/8" Thick, 18" x 18"	in the morning	$204.81 each	204.81
2	2 each	9057K221	Multipurpose 6061 Aluminum with Certification, Precision-Ground Blank, 3/8" Thick, 4" x 6"	in the morning	$62.85 each	125.70
3	2 each	9057K222	Multipurpose 6061 Aluminum with Certification, Precision-Ground Blank, 3/8" Thick, 4" x 8"	in the morning	$69.58 each	139.16
4	1 each	9246K493	Multipurpose 6061 Aluminum, 1/2" Thick, 6" x 6"	in the morning	$15.41 each	15.41
5	1 pack	92620A583	High-Strength Grade 8 Steel Cap Screw, 5/16"-18 Fully Threaded, 1" Long, Zinc-Plated, packs of 50	in the morning	$10.07 pack	10.07
6	2 packs	92620A598	High-Strength Grade 8 Steel Cap Screw, 5/16"-18 Fully Threaded, 4-1/2" Long, Zinc-Plated, packs of 5	in the morning	$14.38 pack	28.76
7	1 each	1630T29	Multipurpose 6061 Aluminum, U-Channel, 2" Base x 1" Legs, 5' Length	in the morning	$20.28 each	20.28
8	5 packs	97191A250	Press-Fit Insert for Metal, Zinc-Plated Steel, 5/16"-18 Internal Thread, 5/8" Long, packs of 5	in the morning	$12.10 pack	60.50
9	1 each	8982K95	Multipurpose 6061 Aluminum, 90 Degree Angle, 3/8" Thick, 4" x 6" Legs, 3' Long	in the morning	$88.16 each	88.16
10	1 each	6546K89	Multipurpose 6061 Aluminum Rectangular Tube, 1/2" Wall Thick, 6" Height x 6" Width, 1/2' Long	in the morning	$69.36 each	69.36
					Merchandise total	$762.21

Applicable shipping charges and tax will be added.

Phone (630) 833-0300 **Fax** (630) 834-9427 **Internet** www.mcmaster.com **Email** chi.sales@mcmaster.com

Spectral Dynamics Quote:

1.0 Control and Shaker System Proposal

Date: January 30, 2015
Attention: Dylan de Caussin
C/O Slime
125 Venture Drive
Suite 210
San Luis Obispo, CA 93401

Phone/Fax: 714-628-2278; Fax: 714-628-2369
Email: shammi.hundal@itt.com
Quote #: 0130-GMR-2015-03
Reference: Shaker System Proposal

Item	Part Number	Description	Qty	Total Price
1	**SD-660-230M**	**SD-660-230M Shaker System**	1	
	SPA302	Amplifier Model SPA302, 3 kVa Switching Amplifier		
	ACU751	.75 kW Air Blower		
		Sine force (peak value) 660 lbf		
		Random force (RMS) 660 lbf		
		Shock force (peak value) 1,320 lbf		
		Frequency range (Hz) 5-2,500		
		Maximum acceleration 65 g		
		Maximum velocity 78.7 in/s		
		Maximum displacement 1.5 in		
		Maximum payload 264 lbs		
		Armature Diameter 230 mm (9.0 inches)		
		Weight of Shaker (Uncrated) 600kg (1,320 lbs)		
		Compressed air requirement 87 psi		
	SD-Install	*** Complete System Installation and Training**	1	
		Installation for Shaker System and Blower Electrical Power Checkout Blower Cooling System Checkout Check out of Amplifier, Shaker and Slip Table if present Check air pressure to shaker Power Up Amplifier and check alignment Checkout Amplifier Output for Distortion Connect Amplifier to Shaker and checkout Operation Connect Controller and commence Operation		
		Note: It is the customer's responsibility to remove the shaker system from the delivery truck and place the system in its end use position. This includes the shaker, amplifier and cooling blower. Facility power is to be connected to the amplifier, but not turned on until SD arrives for the installation, site ventilation duct installation should be in place.		
		FOB Factory Shipping*	est	$3000
	SD-delivery			
		System Delivery 8 to 10 weeks After Receipt of Order		
		One Year Parts and Labor Warranty		
		Shaker Delivery is 8 to 10 weeks ARO		
I1		**Investment for SD-660-230M Shaker and Amplifier System**		**$26,400**
		***Shipping from Long Beach Harbor billed at actual cost**		

2		**Four (4) Input Lynx Control System**		
	2307-9700	**LYNX Vibration Control System**	1	
		Includes: Four (4), 24 bit, 102.4 Ksample/sec input A/D channels with programmable attenuators, advanced anti-alias filters, selectable ICP power, and on board floating point Digital Signal Processing for advanced Digital Filters, one 24 bit D/A output with 48 bit attenuator and one unattenuated output, true digital anti-imaging filters, One unattenuated output channel. One Year Return to Factory Warranty.		
	2300-9501	**Computer System**	1	
		Bobcat/Lynx Notebook Computer-typically a 15.6" screen with dual core processor and Windows 7-64 OS		
		System Warranty		
		* *One Year, RTD Hardware warranty*		
		* *One Year Software Support Program*		
	2400-8900	* *Onsite installation and familiarization*		
		Hardware Subtotal		$8,500
		Application Software Included		
	2307-9413	LYNX Random Vibration Control LEVEL 1 includes: 2KHz frequency range, 400 lines of resolution, time and frequency domain displays	1	
	2307-9416	LYNX Sine Vibration Control LEVEL 1 includes: 1 Hz - 2KHz frequency range, 800 points per sweep, BB_rms channel processing	1	
		Software Subtotal		$2,000
I2		**Investment for Four (4) Channel Lynx Control System**		**$10,500**
I3		**Investment for 660lbf Shaker System with Lynx Control System (I1 and I2)**		**$36,900**
3	**Optional Items**			
	2307-9414	LYNX Random Vibration Control LEVEL 2 includes: 5KHz frequency range, 800 lines of resolution, time and frequency domain displays	1	$1,500
	2307-9417	LYNX Sine Vibration Control LEVEL 2 includes: 1 Hz - 5KHz frequency range, 800 points per sweep, BB_rms channel processing	1	$1,500
	2307-9419	LYNX Classical Shock Vibration Control LEVEL 1 - Half Sine and Trapazoid pulse shapes, Manual Mode Only.	1	$1,000
	2307-9420	LYNX CATS Classical Shock Vibration Control LEVEL 2 - Half Sine, Sawtooth, Terminal Peak Sawtooth, and Trapazoid pulse shapes, Reference Importing, Manual Mode Only.	1	$2,000
	2400-9441-P	Data Explorer Pro: Stand Alone Utility (Win7-64 compatible) Imports UFF files into a quick Graphing Package with full cursors, markers, tags, color and font selection, includes Graph Tool.	1	$1,750
	NPN-THB	Thermal Barrier (FR4-G10) for SD-660 shaker	1	$540
	NPN-XDCR	100mV/g ICP Accelerometer (Need at least one)	1	$400

Terms:

- *F.O.B. factory U.S.A.*
- *Quotation Valid for a period of 30 days*
- *Payment Terms (with approved credit) NET 30 days*
    - *SD reserves right to increase the sales price with change in Payment terms*
- *SD reserves the right to correct quotation errors*
- *A 15% cancellation charge applies for any order cancelled within 30 days prior to shipment*

American Express, MasterCard & VISA Accepted

Remit to address:

Spectral Dynamics, Inc.
2199 Zanker Road, San Jose, CA 95131-2109
Phone: (800) 778-8755 Fax: (408) 944-9403

- *Delivery is 15 – 30 days A.R.O.*
- *All prices are quoted in U.S. dollars*
- *Any order placed against this quotation is subject to acceptance by the manufacturer*

Note: 50% payment is required with the order for Shaker Systems

Please Fax/Mail/Email orders to:

Mark Remelman
Spectral Dynamics, Inc.
2199 Zanker Road
San Jose, CA 95131
800-778-8755 ext 55 Phone
408-944-9403 Fax
remelmanm@sd-star.com

Appendix J. Off-the-Shelf Data Sheets

Spectral Dynamics, Inc.

Low Force Vibration system Series

Model: SD-660-230M/SPA302/ACU751

Typical System Application

The **Model SD-660-230M** Series is a versatile wide bandwidth electrodynamics vibration test system. It is designed to test small to medium sized payloads such as electronic assemblies in the automotive, aviation, military, medical and electronic manufacturing industries.

The **SD-660-230M** is capable of a **Random RMS force of 660 lbf and Sine Vector force rating of 660 lbf** in the frequency of 5 Hz to 2,500 Hz under controlled conditions. The system consists of a model SD-660-230M shaker and is driven by a 3 KVA power amplifier and a 0.75 KW cooling blower. Optional items including slip tables, head expanders, accelerometers and a vibration controller can be added upon request.

How to select the suitable model

It is critical to consider the size and position of the test article and the total moving mass of the payload as well as the payload's inertial and overturning moments when selecting a system for your application. It is recommended the force selected should be 1.2 times the theoretical value, to insure appropriate safety margins. For assistance selecting the best system for your needs, please contact your Spectral Dynamics sales representative.

◎ High FRF & Wide UF

The new shaker design significantly raises the Fundamental Resonance Frequency and Useable Frequency of the long stroke systems and out performs similar products from other manufacturers.

◎ State-of-the-Art Armature

The unique reinforced armature structure design is state-of-the-art, providing increased reliability and unsurpassed performance. The proprietary armature structure has been re-designed to optimize rigidity and force transmissibility. Designed for continuous duty and ideal for research & development, production, stress screening and qualification testing, the ruggedized armatures can endure severe vibration and shock forces and extreme temperature conditions.

◎ Efficient Air Cooling

The SD-660-230M shaker system is totally air cooled for easy installation and economical operation.

◎ Cooling Blower Unit

The Cooling blower is the ACU751 as specified below.

◎ Air-Isolated Rotating Trunnion

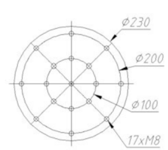
All shakers come standard in a rotating trunnion for easy 90° rotation between the horizontal and vertical test axes. The trunnion is pneumatically isolated providing high stability and allowing for direct mounting onto conventional industrial concrete floors. All shakers are optionally available with an integrated or stand-alone slip table assembly.

◎ D-Class Switching Amplifier

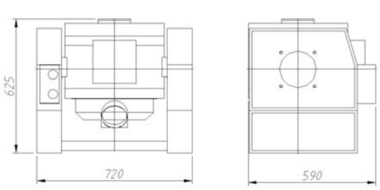
The state-of-art modular switching amplifiers are 100% air-cooled with redundant safety systems and system interlocks insuring performance that is reliable and stable. All amplifiers adopt IGBT power modules of high quality.

◎ Safety

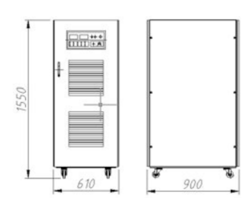
Products comply with European tests standards and ISO regulations.



Armature insert pattern



Shaker body



Amplifier

SD-660-230M/SPA202/ACU751
TECHNICAL SPECIFICATIONS

Shaker Specifications		***SD-660-230M***	
Sine (Pk)	660 lbf (300 kg)	*Table Diameter*	230 mm (9'')
Random (RMS)	660 lbf (300 kg)	*Load Attachment Points (Standard)*	17 Stainless Steel Inserts of M8 or 5/16 UNC (option). Bolt circles are 1@0; 8@100mm; 8@200mm.
Shock (Pk)	1,320 lbf (600 kg)		
Usable Frequency	5 to 2,500 Hz	*Degauss Coil*	Standard
Maximum Displacement (p-p)	1.5'' (38 mm)	*Stray Flux Density @6 inch (152 mm) above table*	< 10 gauss
Maximum Velocity	78.7 in/s (2 m/sec)	*Overall Dimensions*	28.3''L×23.2''D×24.6''H
Maximum Acceleration	65 g		
Fundamental Resonance Frequency (Bare table)	2,500 Hz (nom.) +/- 5%	*Vertical Load Support*	264 lbs (120 kg)
Body Suspension Natural Frequency (Thrust Axis)	Less than 3 Hz	*Weight of Shaker (Uncrated)*	1,320 lbs (660 kg)
Armature Effective Nominal Weight	10 lbs (4.5 kg)	*Compressed Air Requirement*	87 psi

Switching Power Amplifier Specifications	***SPA302***
Rated Output Capacity	3KVA
Signal to Noise Ratio	Greater than 65 dB
Amplifier Efficiency	Greater than 90%
Interlock Protection(to prevent the output devices from working outside their specified limits)	*●Input Over/Under Voltage ●Logic Fault ●Output Over Voltage/Current ●Control Power ●External ●Shaker Oil Pressure ●Module O/T ●Door Open ●Shaker Temp*

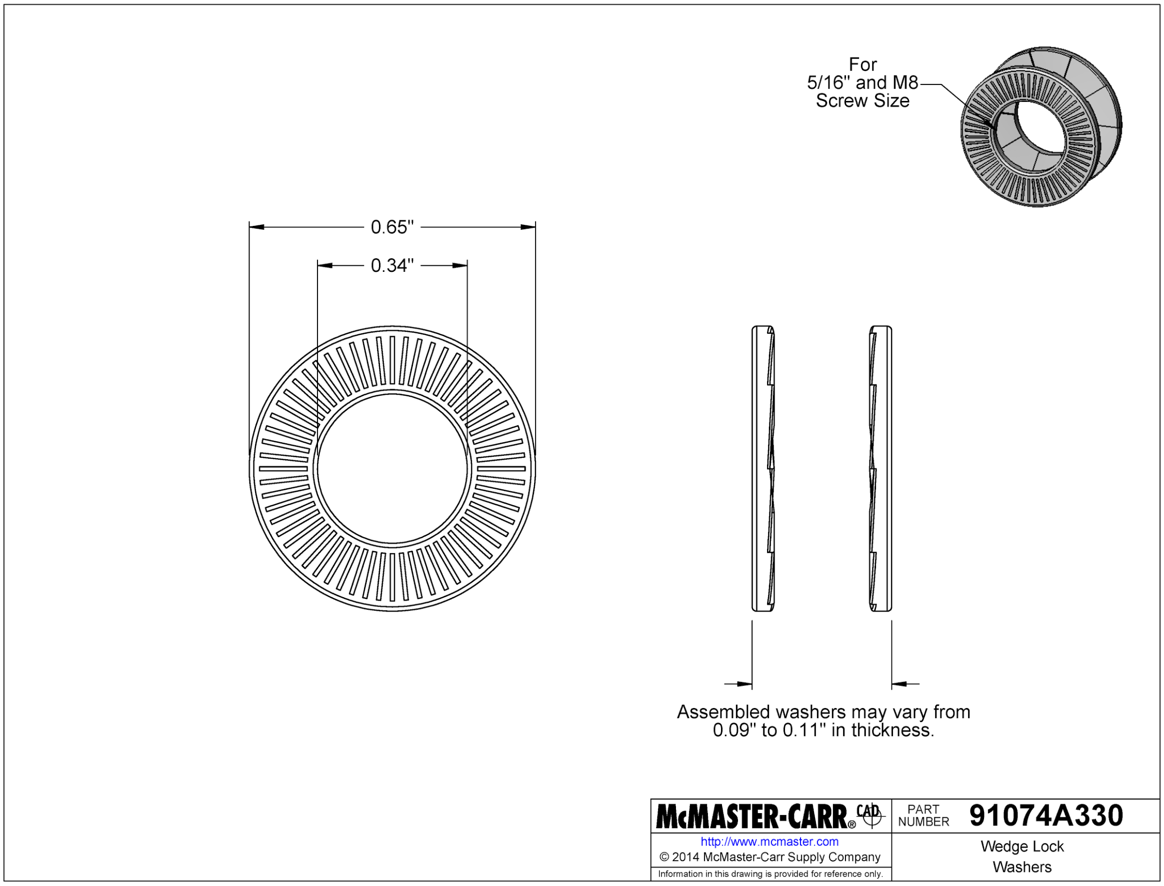
Air Cooling Blower	***ACU751***
Blower Power (Full Load)	0.75 kW (1 HP)
Air Flow Rate	Air Flow: 215CFM Air Pressure: 0.15psi

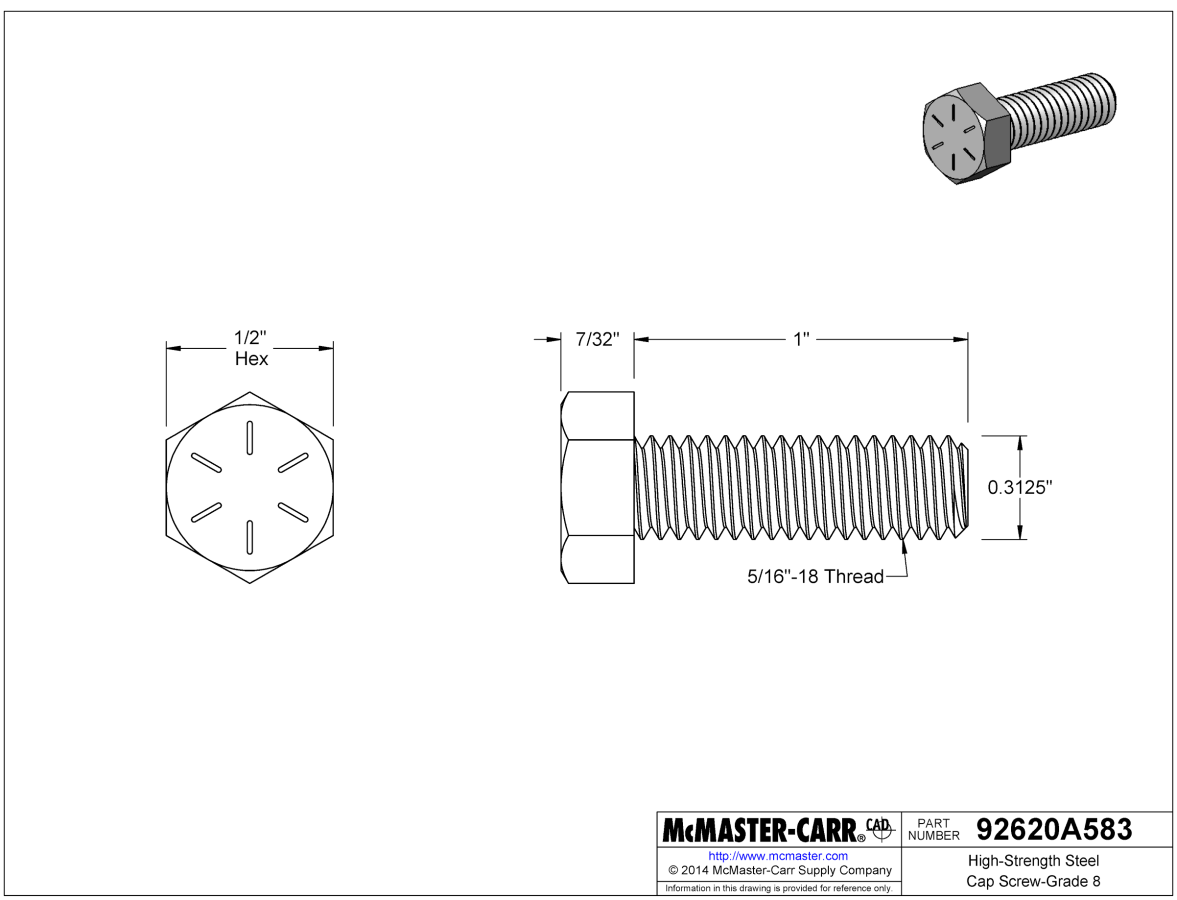
System Environmental Requirement	
Operating Room Temperature	0 to 40 degree C
Humidity	0 to 85%, non condensing
System Continuous Duty	not less than 7 hours at the full ratings
Amplifier Power Requirement, including blower motor	380/415/480 VAC, 50/60 Hz, 3Ph, 6 kVA

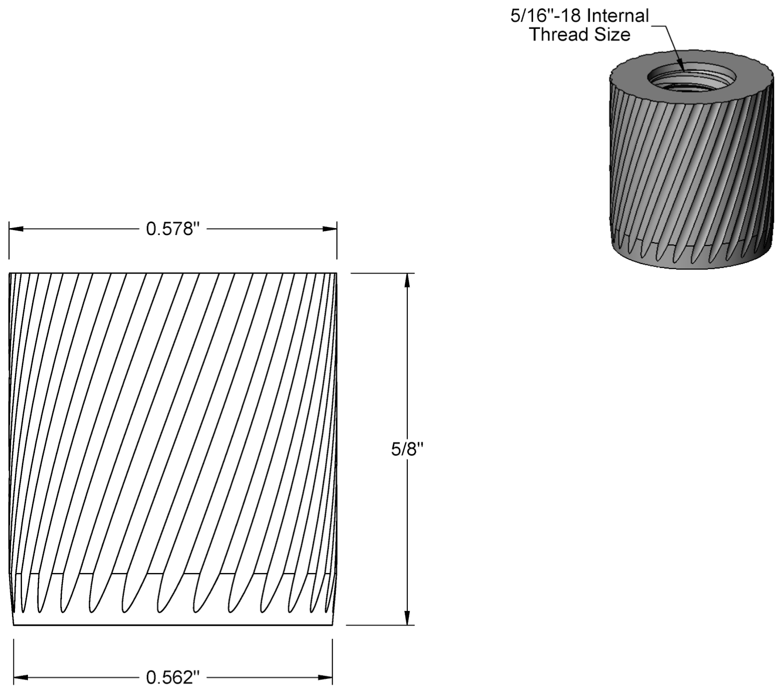
SYSTEM OPTIONS	
●Slip Table Configuration *●V-Groove Caster and Rail System* *●Remote Control* *●Head Expander*	*●Thermal Barrier* *●Load Support Air Compensator* *●Air Caster*

All the features make the SD-660-230M/SPA302/ACU751 a reliable and affordable system for your applications.

NOTE: In keeping with our commitment to continuous product improvement, the information herein is subject to change.


For
5/16" and M8
Screw Size
0.65"
0.34"
Assembled washers may vary from
0.09" to 0.11" in thickness.
McMASTER-CARR® CAD
PART NUMBER 91074A330
http://www.mcmaster.com
© 2014 McMaster-Carr Supply Company
Information in this drawing is provided for reference only.
Wedge Lock
Washers


1/2"
Hex
7/32"
1"
0.3125"
5/16"-18 Thread
McMASTER-CARR® CAD
PART NUMBER 92620A583
http://www.mcmaster.com
© 2014 McMaster-Carr Supply Company
Information in this drawing is provided for reference only.
High-Strength Steel
Cap Screw-Grade 8

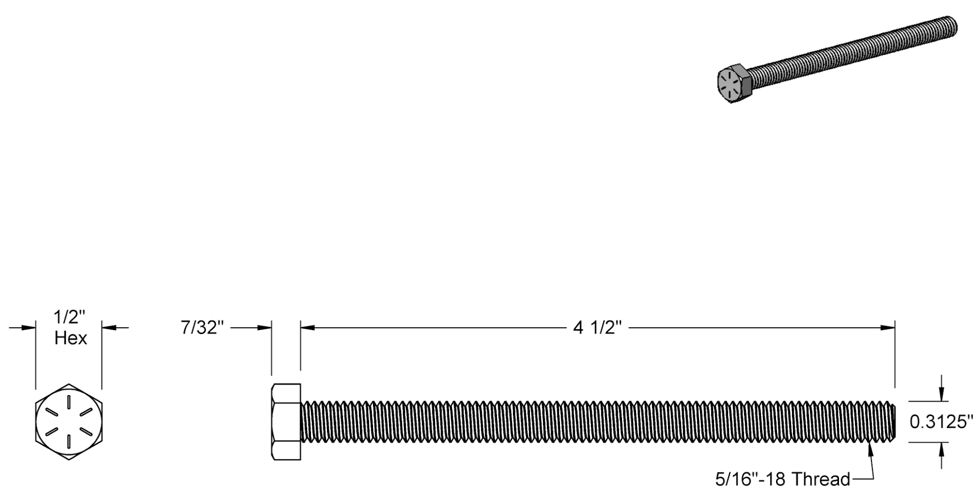


Suggested Drill Bit Size 0.564"

McMASTER-CARR CAD | PART NUMBER **97191A250**

http://www.mcmaster.com

Press-Fit
Insert for Metal



McMASTER-CARR CAD | PART NUMBER **92620A598**

http://www.mcmaster.com

High-Strength Steel
Cap Screw-Grade 8

Appendix K. Testing Data Sheets

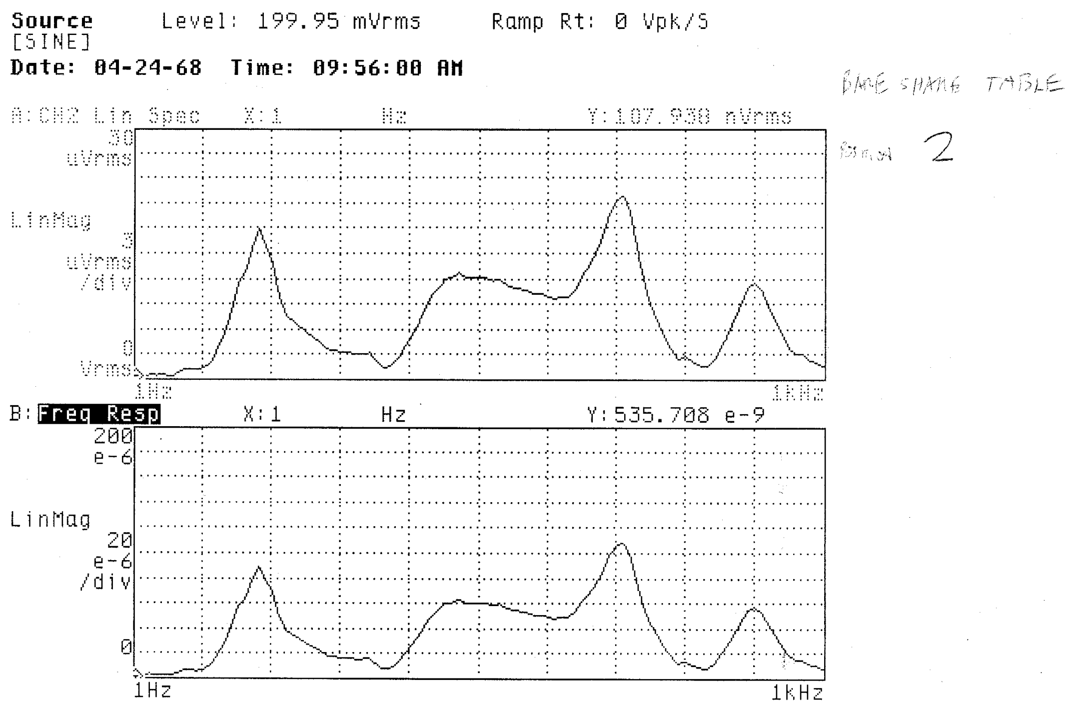
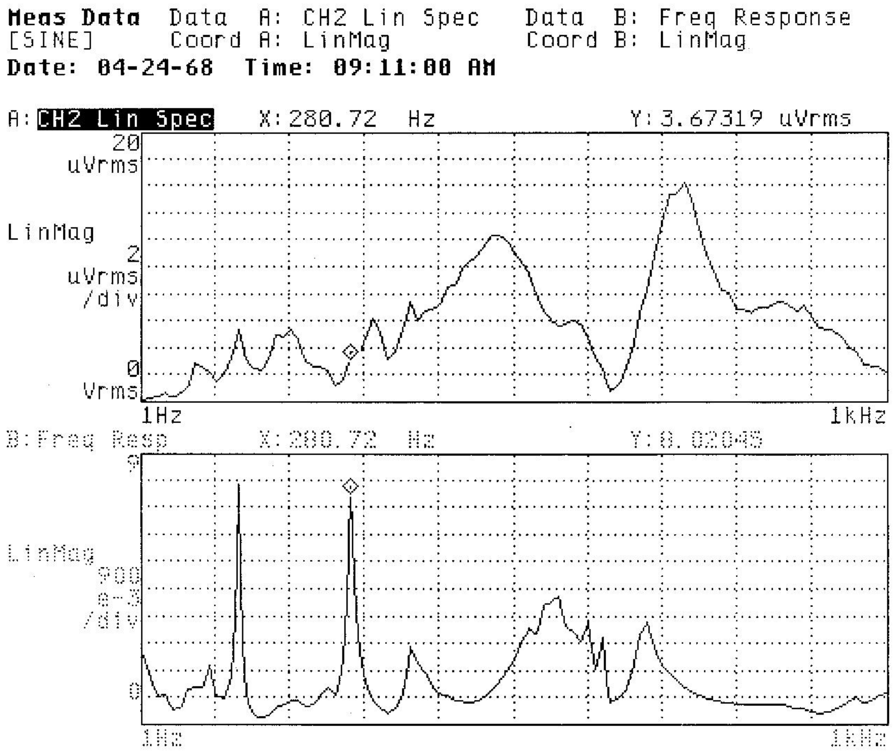


Figure K1: Both signal show frequency response of the hydraulic shaker table, giving evidence that the shaker we were using was not free from natural responses.



ON WOOD PLATE NEXT TO OTHER ACCELEROMETER

Figure K2: Frequency response of the shake table (top) and the wood interface plate (bottom) showing that the signal coming from the shake table is not being passed directly to the plate and ultimately to the head expander.

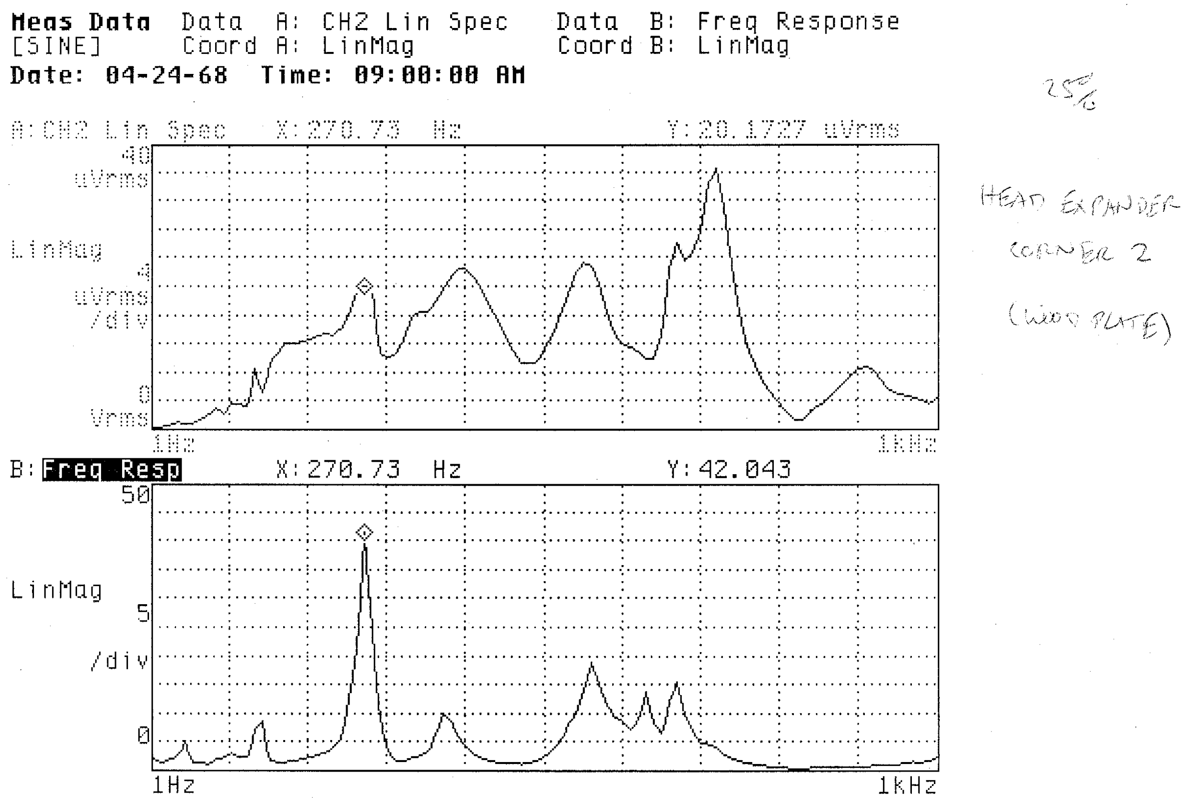


Figure K3: Frequency response of shake table (top) and the edge of the head expander (bottom). This response shows a peak in the profile at 270 Hz on the head expander. This data was rejected due to the complicated response of both the wood interface and the shaker table.

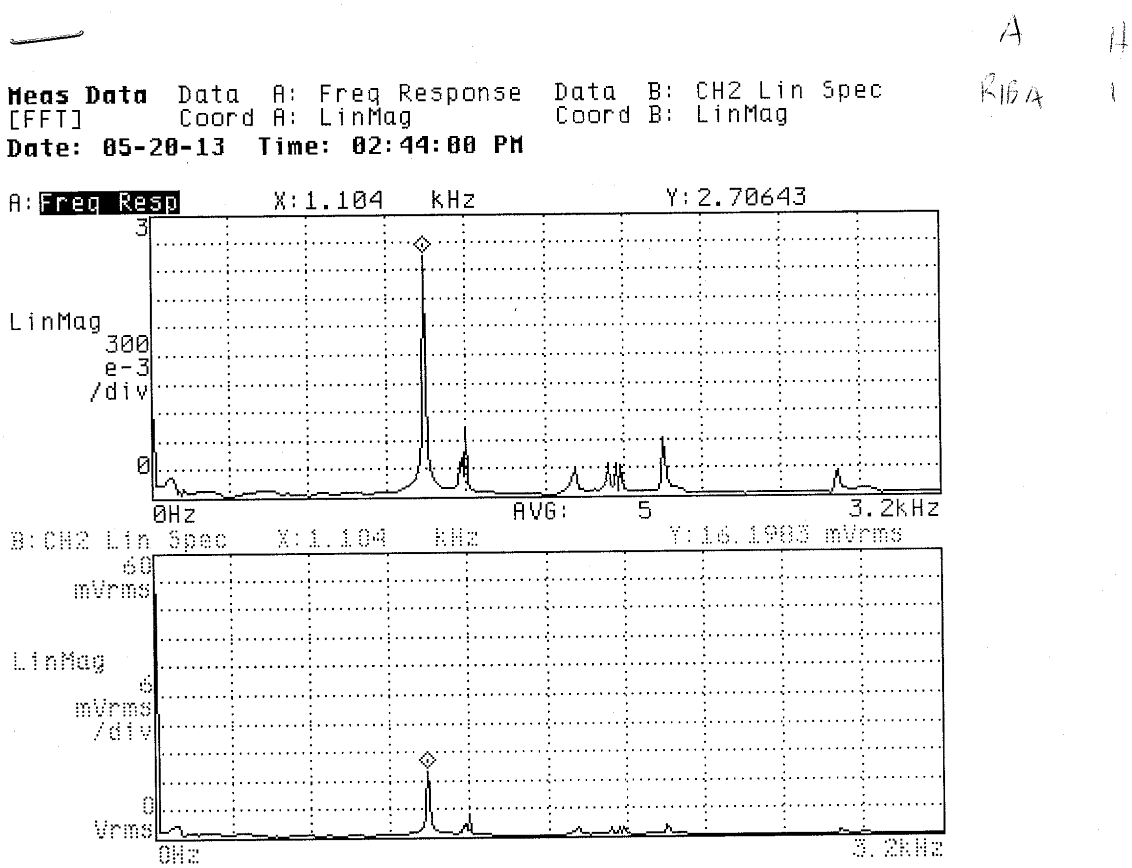


Figure K4: Modal hammer frequency response of the rib of the head expander. This is a very positive result, with the first peak of the response at 1100 Hz which is in the acceptable range.

A H
2 1

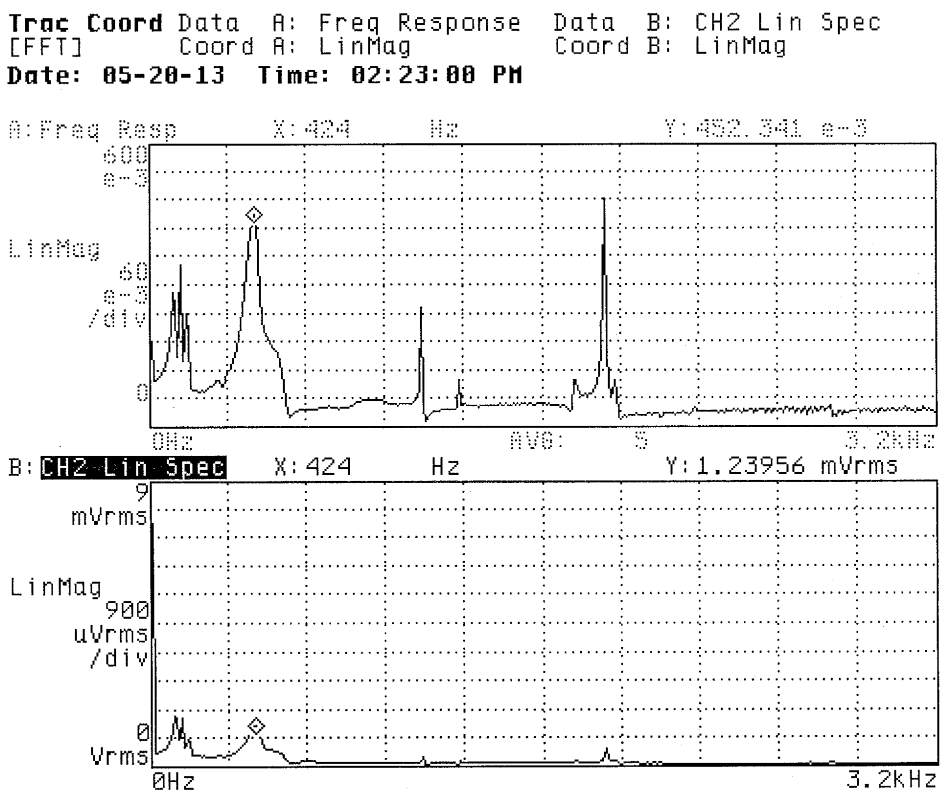


Figure K5: Modal hammer frequency response of the edge of the head expander. This response shows that the head expander has a lower response at 425 Hz, indicating that the design may not be sufficient.

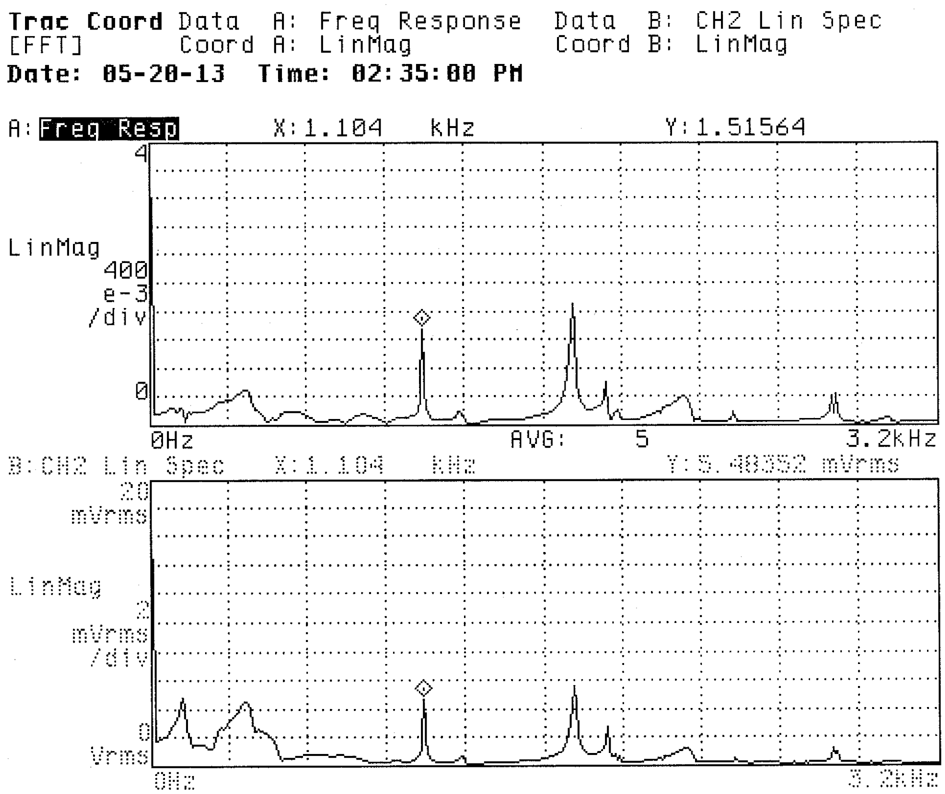


Figure K6: Modal hammer frequency response of the corner of the head expander. This response again indicates that the lowest natural frequency of the head expander is around 1100 Hz.

A H

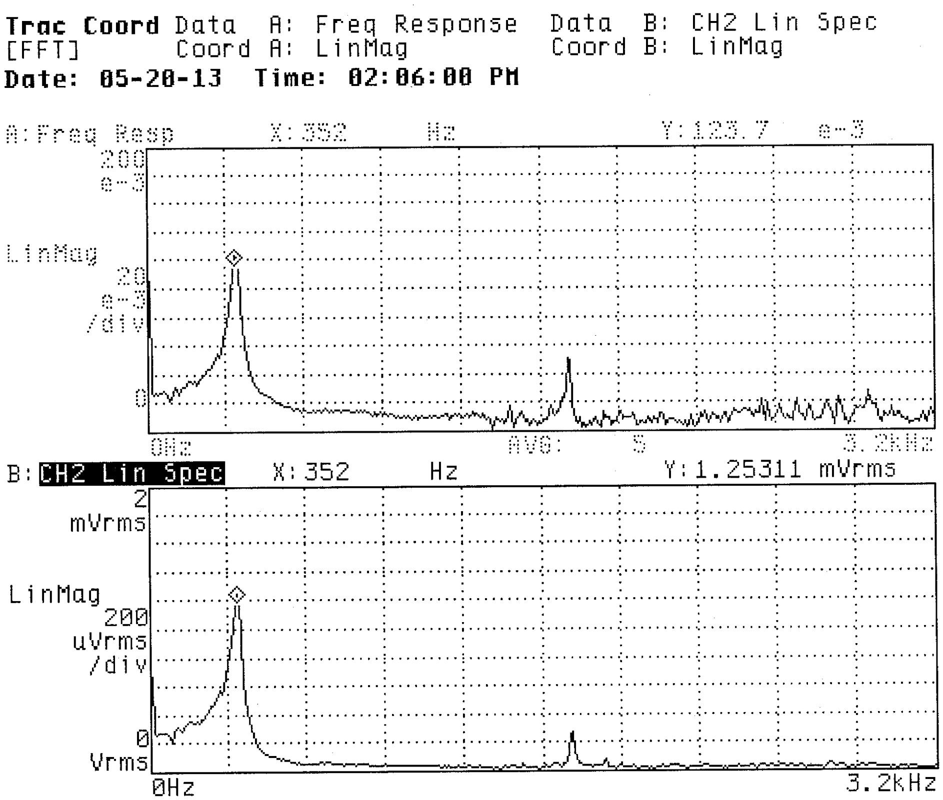


Figure K7: Modal hammer frequency response of the center of the head expander. This response shows a small peak at 350 Hz. The magnitude of this peak is very small, so it may not affect performance, but eliminating it would be preferred.

Appendix L. Operator's Manual

This guide outlines the tools and procedures necessary to operate the vibration testing system described in this report.

Parts

The tools needed to secure the fixture are:

- ½" Socket wrench
- ½" wrench

The parts of the fixture system are:

- Vibration table
- Head expander
- 2 L brackets
- 2 U brackets
- 12 - 1" long 5/16" bolts
- 4 - 4.5" long 5/16" bolts
- 4 - 5/16" nuts
- 20 wedge lock washers

Assembly

After all the above parts are located, the assembly can begin which includes the following steps.

Securing head expander to shaker:

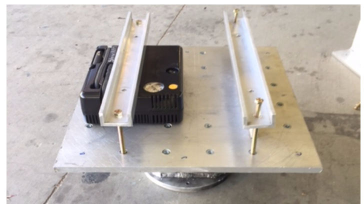
1. Place the head expander on the shaker's circular armature and line up the holes.
2. Place four 1" long, 5/16" bolts with washers into each hole.
3. Tighten the bolts to 7 ft lbs with the socket wrench so there is no movement.

For vertical testing orientation:

1. Place the two L brackets on the head expander with the shorter side on the table. The longer flat edge should be facing outwards as shown in the picture below.
2. Place four 1" long, 5/16" bolts with washers into each hole on each L bracket.
3. Tighten the bolts with the socket wrench so there is no movement.
4. Place a TRK flat against the L bracket in the desired test axis.
5. Put 2, 4.5" long 5/16" bolts with washers in the outer holes of the L bracket.
6. Slide a U bracket onto these bolts so that it sandwiches the TRK between the L bracket.
7. Place washers and nuts onto the bolts and tighten down with the wrench holding one side and a socket on the other. The force on the TRK should not deform the casing.
8. Repeat steps 4 through 7 with the other L bracket and TRK.

For horizontal testing orientation:

1. Place the two TRKs on the head expander flat in the correct axis.
2. Place a U bracket on each TRK to sandwich it to the table.
3. Put 2 4.5” long 5/16” bolts with washers in each U bracket’s outer holes.
4. Tighten the bolts with the socket wrench tight enough to secure it but not too tight to deform the TRK casing.



Testing

Once the fixture is all secure and assembled, the tests can be run according to the specifications needed. Read the user manual for the vibration table to understand the safety and operation steps. Also familiarize yourself with using the controller and putting inputs into the system. The operating manuals provided for the shaker and controller will ensure safe operation and testing.