Rebounder Fatigue Test Machine

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Sponsored by:
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2013

Statement of Disclaimer

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List of Nomenclature

Life cycles - Number of cycles before components fail by yielding, fracturing, buckling, or fatiguing

Jump Rate - The cycles per minute that a person jumps expected to jump on a fitness trampoline

Force Applied - The force that the impactor sees from depressing the trampoline

Surrounding Structure - The structure that supports the trampoline and the mechanism that depresses the trampoline

Compliance - How each design requirement is to be met

Executive Summary

JumpSport, a company that designs and sells trampolines and trampoline accessories, has sponsored this senior project team to design, build, and test a trampoline fatigue test machine. The machine must simulate a person jumping on the trampoline to test the life of JumpSport's fitness trampolines and kids' trampolines. Partway through the design process, the objectives were altered and this senior project team was tasked with merging with another Cal Poly senior project group to create an all-inclusive test machine to accommodate both full-trampoline testing and individual bungee cord testing.

The final design is centered on a slider crank linkage driven by a servomotor. A load cell is bolted to the end of the slider crank for force measurements. An impactor subassembly is in turn bolted to the load cell. The impactor acts as the interface between the power system and the trampoline mat and is designed with running shoes attached to more closely mimic a person jumping. The linkage is supported by a gantry spanning the width of the supporting base plate. The base is mounted on leveling casters for transportation purposes. Some linkage and structure components are adjustable to account for varying strokes.

This report documents the Phase 1 design process including conceptual designs, research and analysis, the final design, the manufacturing process, and testing. This report focuses heavily on the processes leading up to choosing the slider crank linkage as the final design, and the analysis focuses on the impactor. Analysis of the linkage, gantry and frame, and motor will be included in the other senior project group's report upon completion in Fall 2013 (students Chris D'Elia, Andrew Brock, and Ryan Murphy). Additional objectives of Phase 2 will be adding individual bungee cord testing functionality and applying the power system.

Chapter 1: Introduction

Sponsor Background and Needs

JumpSport, Inc, located in San Jose, CA, develops and manufactures trampolines and safety enclosures. Since 1997, JumpSport has been dedicated to developing quality trampolines and accessories with the goal of keeping people safe. In order to design quality products, time and effort must be invested in testing prototypes. JumpSport has designed and built multiple machines that test spring life, trampoline mat impacts, and trampoline safety enclosure impacts. However, despite this current testing ability, JumpSport has expressed the need for a test machine capable of fatigue testing the entire trampoline system. A fatigue test device will greatly benefit the design and production of quality trampolines that will outlast competing designs and, most importantly, keep jumpers safe.

JumpSport presented the project to the Mechanical Engineering Department at California Polytechnic State University, San Luis Obispo, and the department chose this senior project team to take on designing the trampoline fatigue test machine. The goal of this project is to design, build, and test a machine that will test the system fatigue life of the JumpSport Fitness Trampoline and the iBounce Kids Trampoline.

To better understand the design challenge, the senior project team visited JumpSport's headquarters in San Jose, California to speak with JumpSport employees. The team was informed that two competitor's children's trampolines were recalled due to the handlebars unexpectedly breaking off. The handlebars of these trampolines break off during use due to fatigue. The trampolines were recalled by the U.S. Consumer Product Safety Commission. The recent recalls in competitor's trampolines have added to the interest of having a full-system trampoline fatigue testing machine. JumpSport would like to ensure the quality of their designs and have fatigue data to compare with competitor's trampoline data. JumpSport supplied this senior project team with the JumpSport Fitness Trampoline and the iBounce Kids Trampoline to use them as a basis for design.

Formal Problem Definition

The objective of the Bounce Test Trio team is to design and build an apparatus that will mimic human jumping in order to fatigue test both the JumpSport Fitness Trampoline series and the iBounce Kids Trampoline.

JumpSport wants a machine that will impact the trampoline mat and handlebars to simulate a person jumping during exercise. Bounce Test Trio is committed to the completion of the vertical

mat impact testing by the end of Spring quarter 2013. The horizontal handlebar impactor will be added at a later time.

Objectives and Specification Development

Customer needs are of utmost importance and provide the framework for the project. The house of quality, a quality function deployment (QFD) tool, was used to translate customer needs into measurable engineering requirements (Appendix A). These requirements were then ranked based on the customer's priorities. Table 1 shows target values, tolerances, risk, and method for compliance for each engineering requirement. They are listed in their hierarchal order of importance as determined by the QFD.

The top three most important engineering requirements based on the QFD are life cycles, jump rate, and force applied to the trampoline. The life cycles requirement is the most important as it has a strong correlation between three different customer requirements: variable cycles per minute, long endurance, and monitoring data. The trampoline is expected to endure millions of cycles during fatigue testing before a trampoline component fails such as the springs, mat, or frame; therefore, it is pertinent that the testing apparatus does not fail before the trampoline components fail. It is understood that some maintenance may be necessary on the testing apparatus for minor components. However, the major components should last for hundreds of millions of cycles over many tests. The next most important engineering requirement is the jump rate which strongly correlates to JumpSport specifications of operating with variable cycles per minute and mimicking human jumping. This will imitate a human jumper as it accounts for the various rates at which people jump on the trampoline. The next requirement of subsequent importance is the force applied. This engineering specification is also important in mimicking human jumping as it accounts for the loads seen by the trampoline.

Table 1. Specification and Compliance Matrix

Spec #	Parameter Description	Requirement or Target (units)	Tolerance	Compliance
1	Life Cycles	800 million	Min. for major components	A, S
2	Minimum Jump Rate	120 cycles per minute	Min.	A, T
2	Maximum Jump Rate	150 cycles per minute	Max.	A, T
3	Force Applied (Adult)	250 lbs (static)	Min.	A,T
3	Force Applied (Kid)	75 lbs (static) 6g's (dynamic)	Min.	A, T
4	Height of Contact	6-13	+ 2 (Max side) -2 (min side)	I
5	Surrounding Structure Width	52 inches	Min.	T, I
6	Applies Force to Both Mat and Handle Bars	Priority: Mat Secondary: Handle Bars	n/a	I
7	Auto Shut-off switch	yes	n/a	I
8	Low Noise	51dBA	Max.	T, S
9	Low Cost	Rough estimate $5000	Max.	A, I
10	Width of Impactor	1 square foot	Max.	I

Compliance Key: (**A**) Analysis (**T**) Test (**S**) Similarity to Existing Designs (**I**) Inspection

Project Management

Initial responsibilities were delegated to team members: Ethan as liaison to JumpSport, Will as liaison to industry, and Caroline as transcriber. During the conceptual design phase, Ethan was responsible for the linear motor concept, Will for the linkage concept, and Caroline for the dribbler concept.

During the design trimester, it was discovered that the chosen design concept was not able to meet both frequency and stroke requirements. As a result, all team members dedicated full attention to solidifying the power system.

When the senior project groups merged, both groups focuses on completing Phase 1 by the end of spring quarter. This senior project group’s responsibilities focused on impactor design and analysis as well as manufacturing machine components whenever the Cal Poly student machine shop was open.

Chapter 2: Background

Existing Products

To get familiarized with the current testing equipment that JumpSport had created, the Bounce Test Trio visited the testing warehouse in San Jose. The two existing testing apparatuses were very large in size. Both were approximately 20 feet tall and 30 feet wide. These machines were capable of testing the largest diameter models of trampolines that JumpSport manufactures. One of the machines simulated a single vertical mat impact (Figure 1) and the other simulated a single horizontal safety net impact. The impactor on the vertical mat impact tester (Figure 2) was lifted to a desired height by a hand-cranked winch. The desired data acquisition equipment is attached to the impactor and/or the mat surface and once the data acquisition process is started, the impactor is released and gravity pulls it into the trampoline. The computer software records data associated with the impact.

Figure 1. Vertical Test Machine

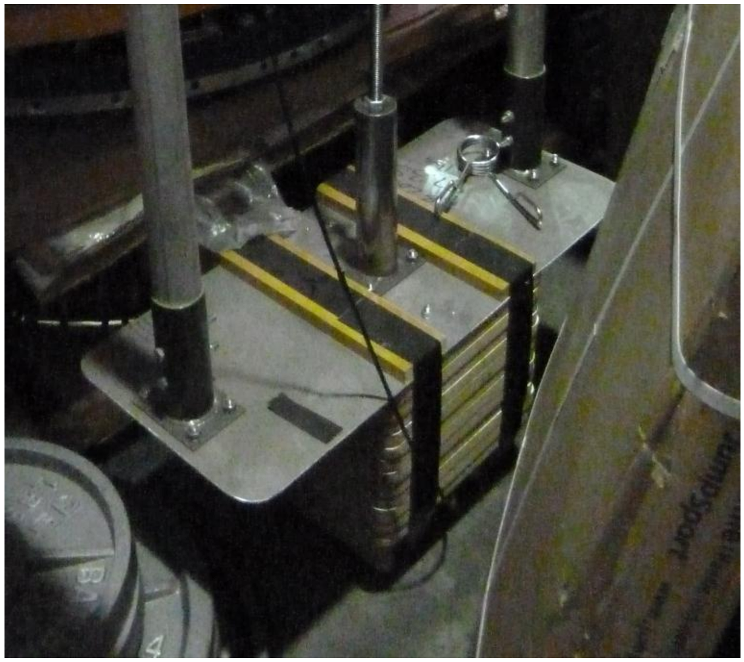

Figure 2. Vertical Test Impactor

The horizontal safety net impact tester (Figure 3) displaces a punching bag vertically and horizontally to a desired position. According to American Society for Testing and Materials (ASTM) standard, the bag is designed to resemble a human body's actual weight distribution. With testing equipment attached, the punching bag is released from rest and swings downward. When it reaches a predetermined point, the bag is completely released into free fall and flies horizontally into the net. This test simulates an individual bouncing off the trampoline into the safety net.

Figure 3. Safety Net Impact Test Machine

Other fatigue testing equipment and trampoline testing machines were studied to understand how fatigue testing and trampoline testing has been achieved successfully. Satra Technology is a global research and development testing company that has developed trampoline testing equipment that tests for different toy safety standards. Satra has equipment that tests trampoline material, strength, elasticity, and frame padding. All these tests meet the EN 13219 and EN 913 toy safety standards, but none of these standards are fatigue related.

A mattress testing machine achieves similar impacting that the trampoline fatigue test machine needed to accomplish. The machine is located at the Original Brand Mattress testing facility in Cleveland, Ohio. The machine also complies with ASTM testing standards. The Cornel Type Mattress Tester simulates 10 years of use on a mattress in only 10-11 hours (Figure 4). The machine uses a 230 pound ram that oscillates vertically while moving horizontally across the mattress. The machine impacts the mattress at 160 strokes per minute.

Figure 4. Cornel Type Mattress Tester

Specific Technical Data

JumpSport has conducted numerous tests on their trampoline systems in order to better understand the physical response of the trampoline mat and bungee cords to displacement. The results of these tests were independently verified using two jumpers, an accelerometer, and an oscilloscope. This separate validation test will be discussed in detail in the Supporting

Preliminary Analysis section of this report. For reference, JumpSport provided the test results in tabular format; it is included in this report in Appendix F.

The first three rows of the table correspond to the three different knot positions for one of JumpSport's regular rebounders using the standard 8 mm bungee. The next three rows are the three different knot positions for an extra firm 9 mm bungee cord. The last three rows are the three different knot positions for a rebounder with 36 extra firm 9 mm bungee cords. Each configuration had weights dropped on them from a height of 15 inches above the bed of the trampoline.

The results of these tests, in conjunction with the validation tests, were used to determine the maximum force that would need to be imparted upon the mat.

Applicable Standards

Some aspects of JumpSport's testing equipment meet ASTM standards. ASTM F2276-10 Standard Specification for Fitness Equipment and ASTM F2571-09 Standard Test Methods for Evaluating Design and Performance Characteristics of Fitness Equipment were applicable to JumpSport's existing testing equipment. The Cornel Type Mattress Tester also claims to meet ASTM standards.

Chapter 3: Design Development

Conceptual Design Development

The team synthesized many different ideas for ways to fatigue test trampolines, and, after refinement and development, converged on three concepts. Each concept emphasizes a different approach to the problem and a unique way of solving it. The goal of this chapter is to present those concepts in detail, offering the strengths and weaknesses of each concept in comparison to one another.

The first concept was a system that takes rotational motion output from an electric motor and uses a crankshaft-connecting rod configuration to translate it to the linear motion of the trampoline mat impactor. This idea came from internal combustion engine design, where the impactor is analogous to a piston in a cylinder. Impactor speed is controlled by changing the speed of the electric motor driving the crankshaft.

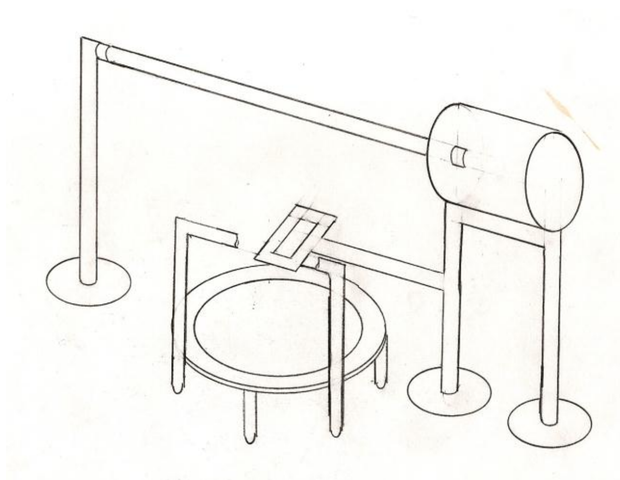

Figure 5. Rotational Motor Concept

The second concept generated came from the idea of dribbling a basketball. In this system, a weighted impactor is dropped onto the trampoline and allowed to rebound, with a linear motor supplying the extra energy needed to keep the system operating at a steady cycle. The goal behind this design is to utilize the elasticity of the trampoline to lower the necessary energy input and more closely mimic a human jumper.

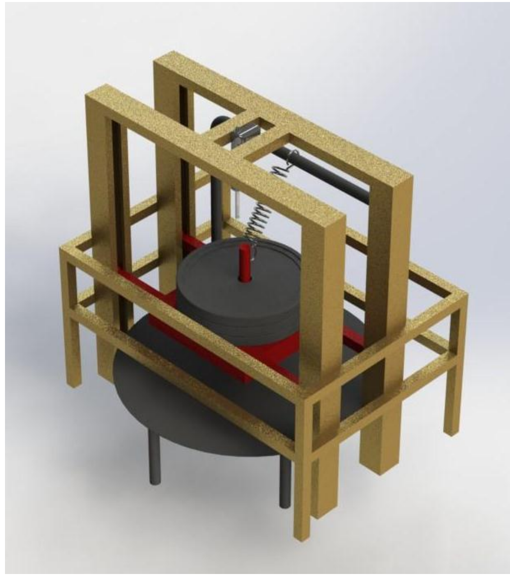

Figure 6. Dribbler Concept

The final concept being considered uses linear motors exclusively. By using linear motors for both the mat impactor and handlebar, intermediate stages are eliminated, mechanically simplifying the structure. A separate linear motor is used for each impactor. Each impactor is controlled and synchronized through software.

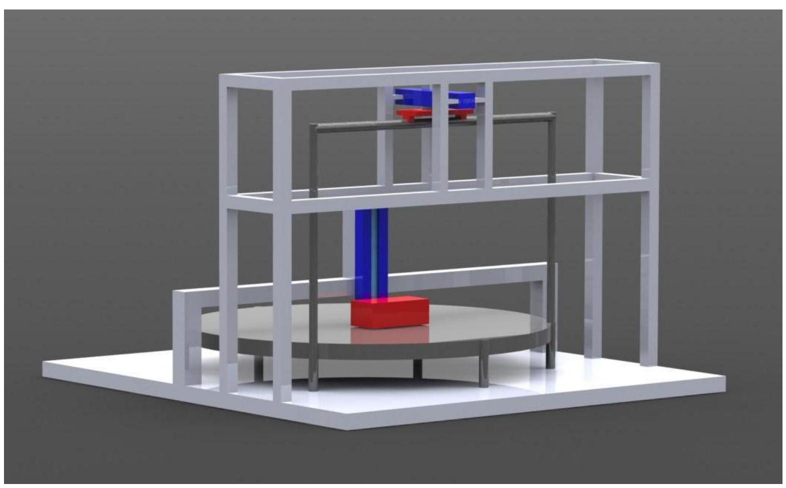

Figure 7. Linear Motor Concept

Rotational Motor Concept

The rotational motor concept stemmed from the spring fatigue testing machine that JumpSport developed and from the first intuition to use an electric motor to power the testing system. The rotational motor concept also led to a study of mechanisms and ways of translating rotational to

linear motion. The rotational motor concept is composed of an electric motor that is plugged into any home outlet. The motor rotates a coupled shaft. The shaft either uses cranks or mechanisms to convert the rotational motion into linear motion. The basic orientation for the motor and support assembly is shown in Figure 7. The motor is supported by the frame and has a long shaft coupled to it that will also be supported by the frame. There is an impactor guide that maintains the linear motion of the mechanism attached to the shaft.

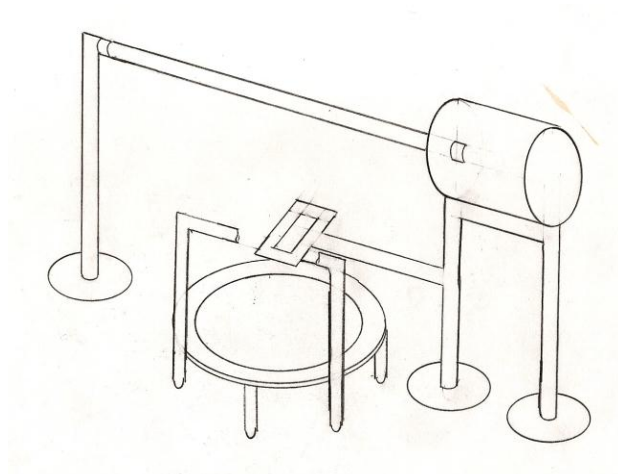

Figure 7. Rotational Motor Concept Layout

Three linkage concepts were considered to work with the basic rotational motor-frame assembly. The first concept is the triple crank concept, referring to the shaft design (Figure 8). The shaft has three cranks; the two outer cranks are smaller than the middle crank. The outer cranks have rods with hooks on the end connected to them. These hooks attach to the handlebars and resemble hands grasping the bar. The middle crank has a single piston crank mechanism with an impactor at the end. While the electric motor rotates the shaft, the impactor impacts the trampoline while the handlebar connecting rods deflect the handlebar assembly.

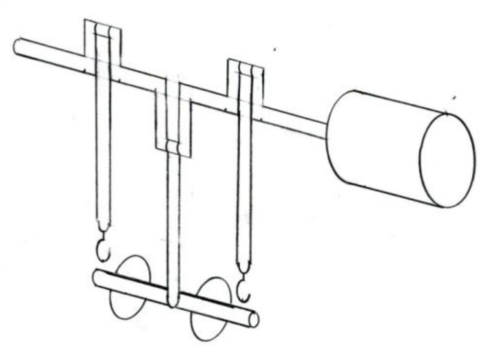

Figure 8. Triple Crank Concept

The second shaft concept is similar to the first but has a single crank design (Figure 9). The single crank powers the single cylinder impact mechanism. The handlebar connecting rod connects to the impactor instead of smaller cranks. The handlebar connecting rod has a spring in the middle.

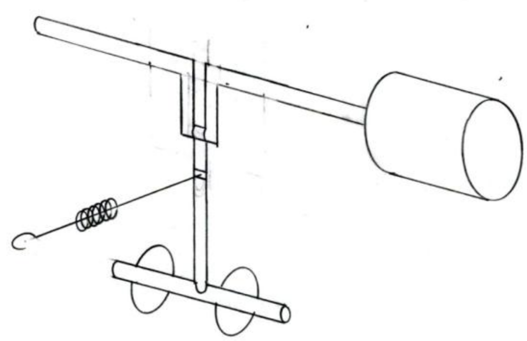

Figure 9. Single Crank Concept

Another idea within the rotational motor concept relates to the motor location (Figure 10). With the motor mounted on the ground rather than at the top of the frame, is reduced. The power is transferred from the electric motor to the shaft using belts and pulleys or chains and sprockets.

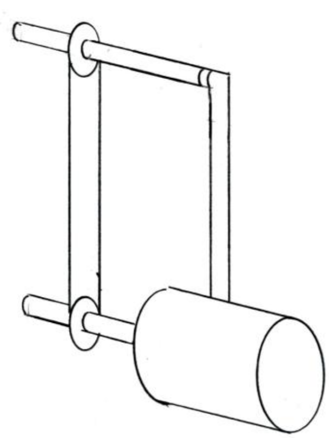

Figure 10. Lower Motor Mount Concept

Another option utilizes Hoeken's linkage, converting rotational motion to mostly linear motion with a portion of rounded off motion (Figure 11).

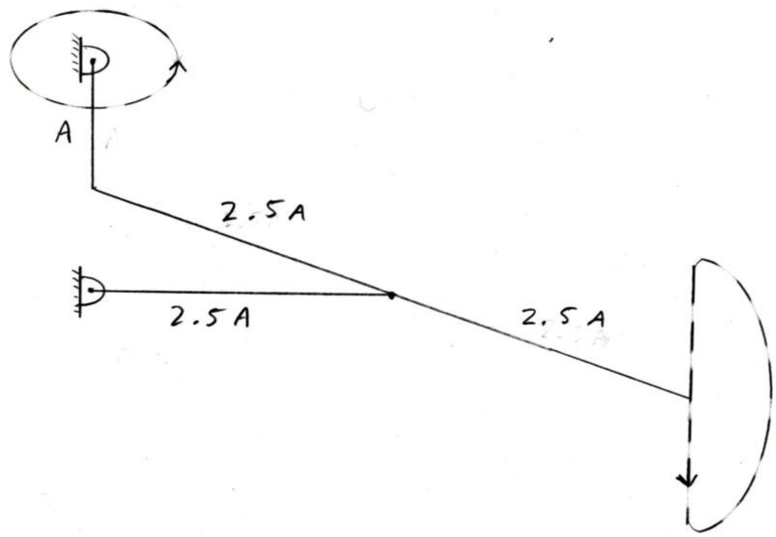

Figure 11. Hoeken's Linkage

A structure that uses Hoeken's linkage in combination with the rotational motor to linearly impact the trampoline is shown in Figure 12. A cord and pulley is attached to the handlebar system to allow for deflection of the handlebars.

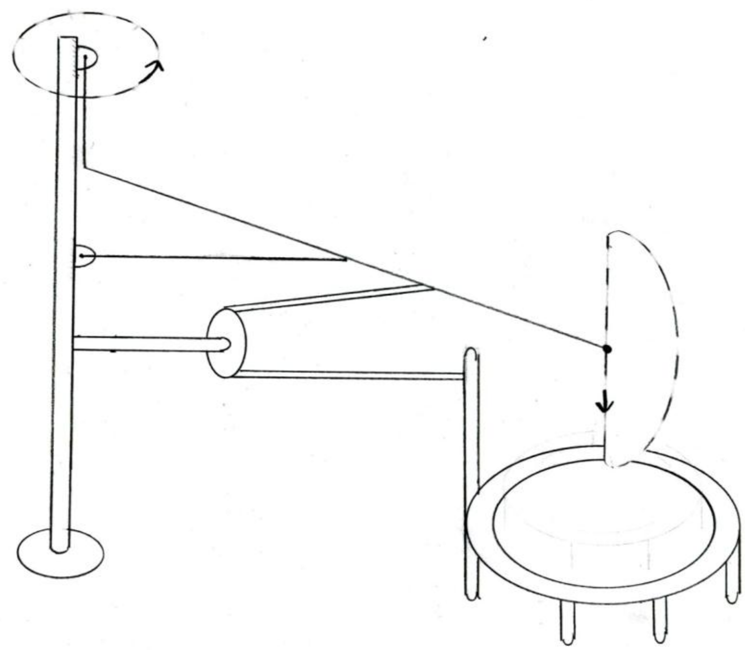

Figure 12. Hoeken's Linkage Structure

Rotational Motor Specification Satisfaction

This concept meets engineering specifications and requirements. The life cycle requirement is met, although bearings may need to be replaced throughout testing. The design easily handles a rate of 150 cycles per minute and has the ability to have an adjustable jump rate by adjusting the motor's rpm. A stroke that would correlate to a 250 pound person jumping is also achievable.

Rotational Motor Pros and Cons

One of the strengths of the rotational motor design is its ability to sustain a high impacting frequency throughout the duration of a test. The induction motor does not have to stop and reverse direction to power the trampoline because a slider-crank mechanism would be utilized. With the addition of a variable frequency drive, the induction motor could power tests of different impact frequencies corresponding to the desired test parameters. An induction motor with a linkage can be sized to handle the high impact loads at the required speed while still being financially competitive.

Some of the drawbacks of this design include manufacturing complexity and the possibility of a higher power input. The design and fabrication of the linkage is integral to the completion of this concept; in contrast, the linear motor and dribbler concepts do not require fabrication of powertrain components. Detailed design and fabrication of the linkage will likely be more complicated than the frame, therefore the rotational motor concept may be more difficult to manufacture. Since the stroke of the linkage is a fixed cycle and the motor is driving the linkage throughout the whole cycle, the motor may not be utilizing the elasticity of the trampoline to its full potential, possibly resulting in a higher power input than the other concept.

Selection within Rotational Motor Concept

The single crankshaft design best mimics the forces that a human would be exerting on the trampoline as the handlebar connecting rod moves up and down with the impactor. In doing so, the angle of force that the handlebar connecting rod applies to handlebars is constantly being varied in the same way that a humans would.

Dribbler Concept

The dribbler concept differs from the rotational to translational concept in that 1) it does not rely on electrical power to supply the energy to apply the desired force and 2) it does not rely on a prescribed path as a function of time to apply the desired force. This concept was inspired by the law of conservation of energy, which states that energy cannot be created nor destroyed within an isolated system. Therefore, the dribbler attempts to create as isolated system that has no external exchange. However, there are unavoidably some energy losses to the surroundings in the form of heat due to friction and collision. As a result, this small amount of energy is added back into the system by an external source: a small linear motor.

The testing is initiated by creating a large amount of gravitational potential energy; the large mass and impactor are raised a height above the trampoline. The mass is released and the energy is converted into kinetic energy as it falls, then into potential spring energy as it compresses the trampoline mat, then into kinetic energy as the mass rebounds off of the mat, and finally back into gravitational potential energy as the mass ends up a height above the mat. The dribbler concept was condensed into two main ideas: a horizontal dribbler and a vertical dribbler.

The Horizontal Dribbler Concept

The horizontal dribbler is so named because of the long horizontal arm that attaches the impactor and weight assembly to the main structure of the test rig. The arm moves radially about the support structure causing the impactor on the end of the arm to collide with the trampoline mat and bounce up and down. Any energy losses are added back into the system by a small linear motor located on the opposite side of the impactor. The linear motor adds kinetic energy into the system by pushing upward on the short side of the arm, therefore increasing the downward velocity of the impactor.

Figure 13. Horizontal Dribbler Concept Isometric View

The impactor below the weight platform is in the shape of two feet and each foot is adjustable laterally. To avoid interference between the impactor arm and the trampoline's circular structural frame, the feet extend below the weight platform by about a foot. This allows the trampoline mat to be fully compressed while the arm never descends below the plane of the trampoline structure. The weights are barbell gym weights that can be simply removed or added from a vertical cylinder in order to vary the weight. Force is applied on the handle bars by a spring that connects the top of the weight platform to the handle bars. As the weight platform moves up, the spring compresses and pushes the handle bars away from the apparatus. As the weight platform moves down, the spring elongates and pulls the handle bars closer to the apparatus.

Horizontal Dribbler Pros and Cons

The horizontal dribbler idea is advantageous because the ground rather than the test structure sees the reactive force from the linear motor.

This idea is unfavorable because the large mass on the end of the impactor arm creates large moments on the arm and the arm joint. Additionally, the test structure takes up a lot of space. Finally, because the impactor travels on a radial path, the impactor strokes through the mat at an angle rather than the vertical impact that is more characteristic of a human jumping on the trampoline.

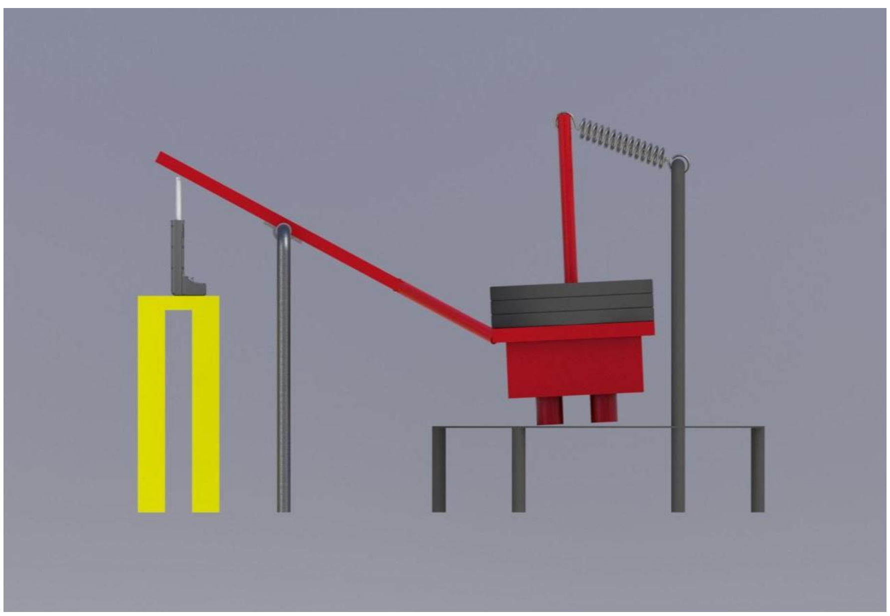

Figure 14. Horizontal Dribbler Concept Side View

The Vertical Dribbler Concept

The vertical dribbler is a permutation of the horizontal dribbler that defines the path of the free-falling weight by utilizing a vertical track instead of a horizontal arm. The carriage, consisting of the impactor and weight, has four bars with rollers extending out laterally from the weight platform that fit into the vertical track. The linear motor that adds energy back into the system is mounted a distance directly above the carriage. It presses downward on the weights when the carriage reaches its peak height.

The impactor's shape and size, the weight configuration, and the handle bar impactor are the same in the vertical dribbler as the horizontal dribbler.

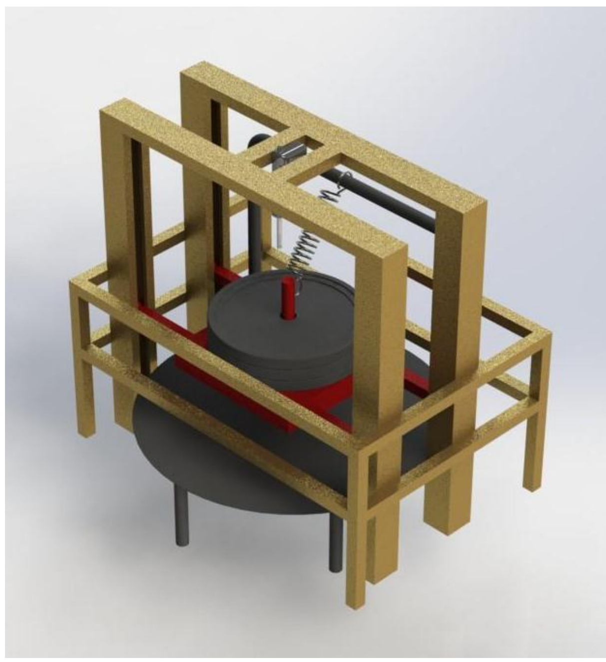

Figure 15. Vertical Dribbler Concept Isometric View

Note: Linear motor is mounted on bottom side of top crossbar

Vertical Dribbler Pros and Cons

The vertical impactor is beneficial because, contrary to the horizontal impactor, it applies a directly vertical force more similar to the path of a human jumping on the trampoline. Additionally, it is more compact with the structure fitting closely around the diameter of the trampoline. Finally, there are negligible moments on structure from impactor weight.

Drawbacks to this idea include difficulty in aligning the vertical impactor guide tracks, which could create additional friction losses between the carriage and the tracks. Additionally, the elevated motor location will provide additional stress on the structure.

The vertical impactor is the favored idea between the two dribbler ideas because of its similarity to human jumping, its negligible moments on the test apparatus, and the limited external energy needed to run the system.

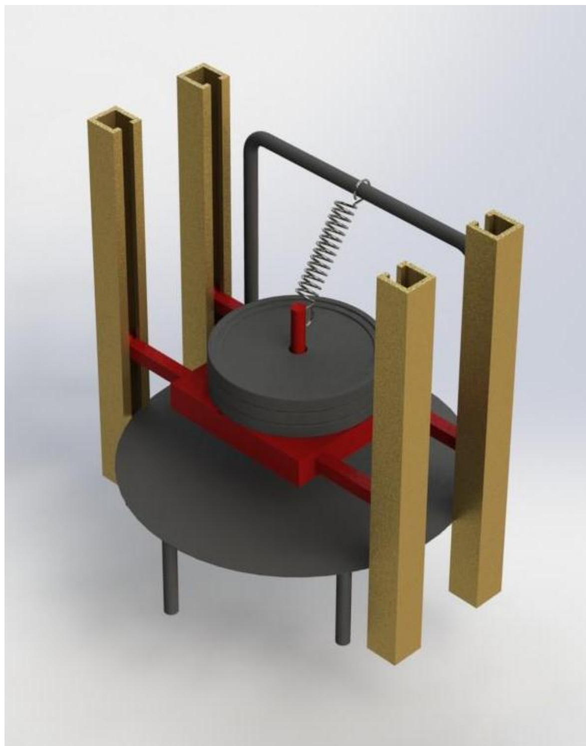

Figure 16. Vertical Dribbler Concept Isometric Section View

Note: Outer structure and linear motor removed for better track and impactor carriage visibility

Dribbler Pros and Cons

Overall, the dribbler concept has many advantages and disadvantages when compared to the other designs. The concept is ideal because of the lower power input. Additionally, the system closely mimics human jumping. Negatives are that there are safety concerns with the large mass bouncing up and down at such a high frequency. The system is more complex than the rotational to translational system in that there must be precise timing about when the linear motor acts on the mass-impactor system. Finally, starting the testing is more complex than just pushing a button like the other two concepts. Instead, the weight must be lifted a distance off of the mat.

Dribbler Specification Satisfaction

The dribbler concept meets all of JumpSport's specifications. The jump rate of at least 150 cycles per minute will be challenging to get the correct timing of the linear motor, but with a control system the linear motor can be timed to output a rate of 150 cycles per minute.

Linear Motor Concept

The basic premise behind the linear motor concept is utilizing a motor that outputs linear motion, the desired motion of the mat and handlebar impactors. Because the motor outputs linear motion, the impactor can be directly mounted onto the motor, eliminating the need for intermediate steps.

This concept uses two different linear motors, one for the trampoline mat and one for the handlebars, to conduct fatigue testing on the both the children's trampoline and the fitness rebounder. Linear motors are essentially metal tracks with a sliding forcer that can be precisely electronically controlled. The impactor is mounted directly to the forcer, and the motor is mounted to a metal (likely steel) frame.

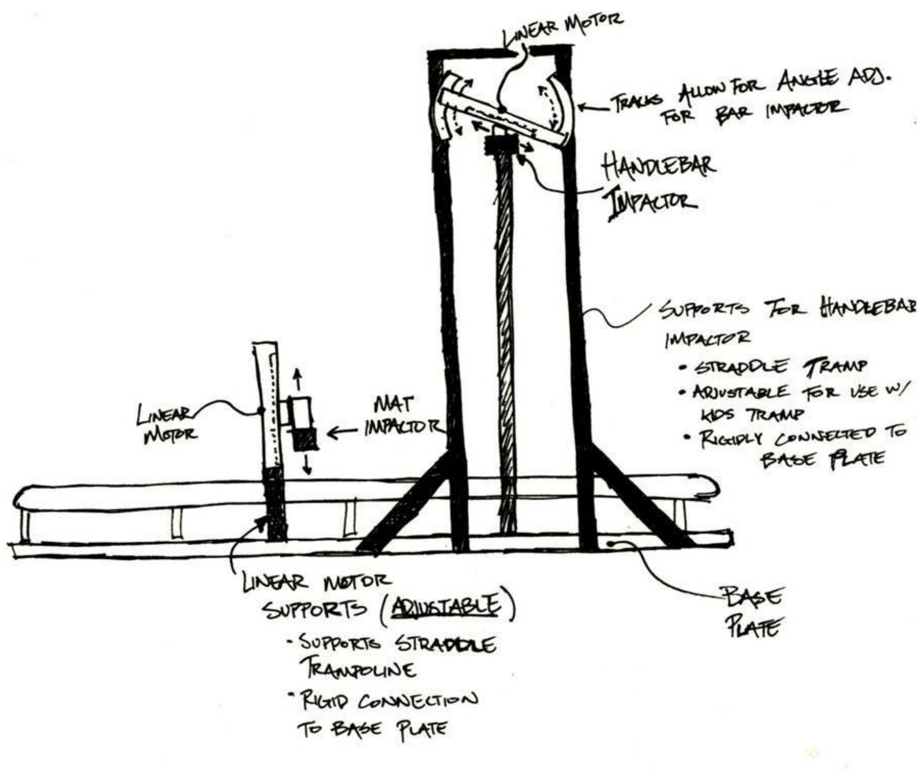

Figure 17. Initial Sketch of the Linear Motor Concept

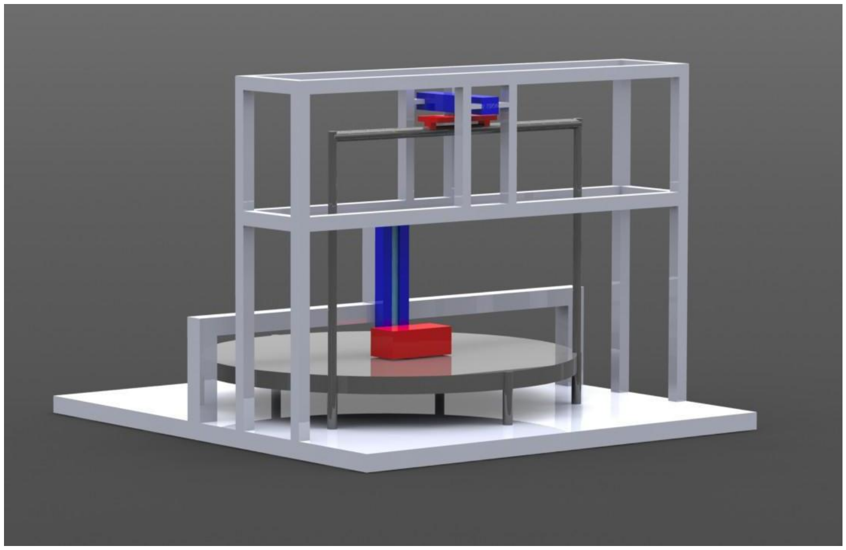

Figure 18. Linear Motor Concept Isometric Front View

Note: The structure is gray, the linear motors are blue, and the impactors are red.

For the mat impactor, the motor is mounted in a vertical orientation so that the impactor is centered on the mat. The impactor stroke will be approximately 36 inches (24 inches above the mat, and 12 inches from the mat to the floor). From testing, JumpSport and this senior project team determined that the average jumper will see a maximum force on their body between 3-6 times their weight. For a 250 lb jumper, the maximum weight recommended for the fitness rebounder, the largest force exerted on the trampoline would be 1500 lbs. Therefore, the linear motor for the mat impactor must be able to exert a peak force of 1500 lbs.

The mat impact linear motor will be mounted to a horizontal support beam that spans the diameter of the trampoline and is supported by two vertical members. The mat impactor support structure will be rigidly attached to the base plate, on which the trampoline will sit during testing. Since both impactor structures will be attached to the base plate, all of the forces seen during each impact cycle will be contained within the structure. No impact forces from the mat impactor will be directed to the ground, thereby eliminating any tendency for the test machine to move around due to the mat impact.

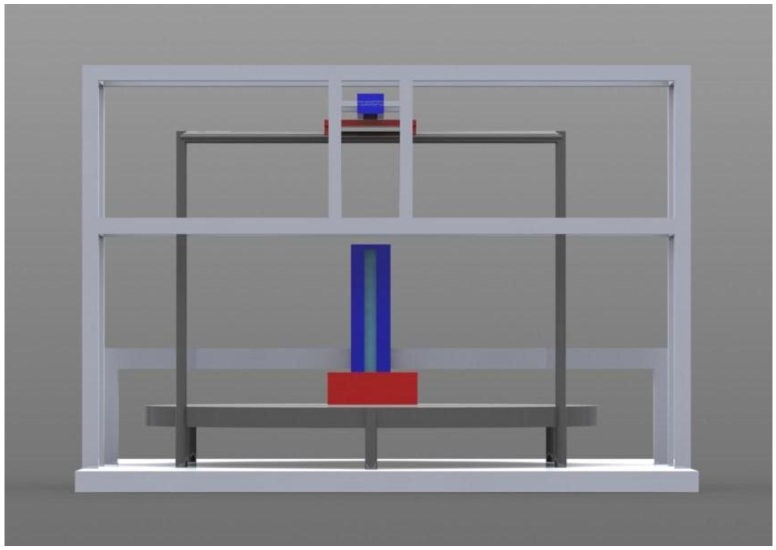

Figure 19. Front View of the Linear Motor Concept

The handlebar impactor will also be driven by a linear motor. This motor will be mounted above the handlebars, with a frame surrounding and supporting it. For rigidity of the structure, it will most likely be connected to the mat impactor support as well. The linear motor will be mounted in such a way as to allow the angle of impact on the handlebar to be adjusted. This will allow JumpSport to vary the angle of handlebar impact between tests if they so desire. The idea for this angle adjustment mechanism came from the spring-loaded pin adjustments commonly seen on fitness machines in weight rooms.

This senior project team observed that, under various jumping conditions, the handlebar deflection on the fitness rebounder was small, on the order of inches. This means the linear motor for the handlebar impactor can be much smaller than the motor driving the mat impactor, needing a stroke around 6 inches or less. The actual attachment to the handlebars will be compatible with both the fitness rebounder and the children's trampoline. As both are shaped and oriented differently, the attachment mechanism will either be adjustable or have different attachments for the two trampolines. The mechanism will be designed to attach to the handlebars in two places, simulating the two-hand grip that a jumper would have.

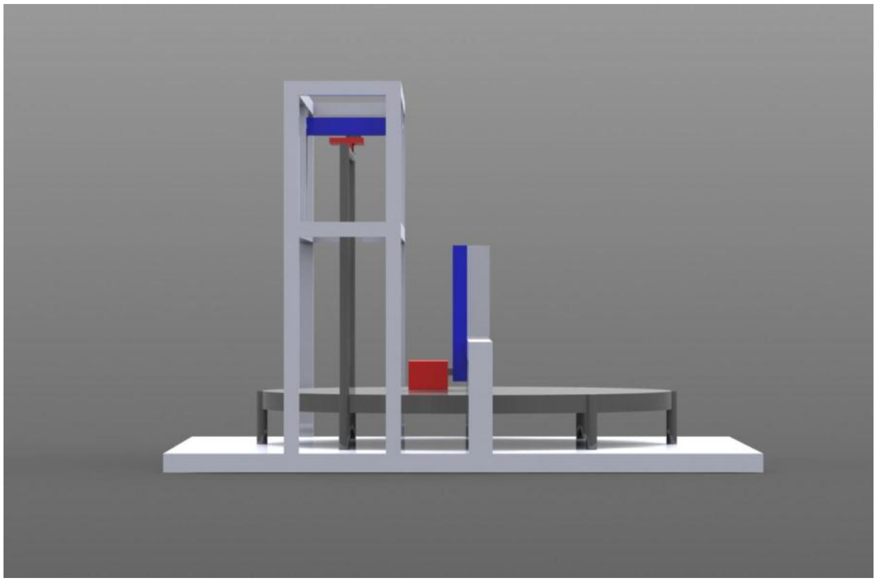

Figure 20. Side View of Linear Motor Concept

This system adds electronic complexity compared to the other designs, as the linear motors will need an amplifier, a controller, and software to complete the system. However, the design allows for more precise control of the impactors, as well as easily variable output.

In order for this design to work well and be easy to use, the general functions of a few other aspects of the system were considered. During testing, the trampoline needs to remain in one position. JumpSport specified that they do not want the trampoline to be rigidly secured to the structure, however, as that would not directly replicate a typical consumer environment. To keep the trampoline stable without over constraining it, the base plate will have indentations for the trampoline legs that will allow a little movement, but not too much. Also, the structure needs to allow for easy installation of a trampoline, either the fitness rebounder or the children's trampoline. The handlebar structure will need to be easily adjustable so that the trampoline and handlebars can slide directly into place without taking anything apart.

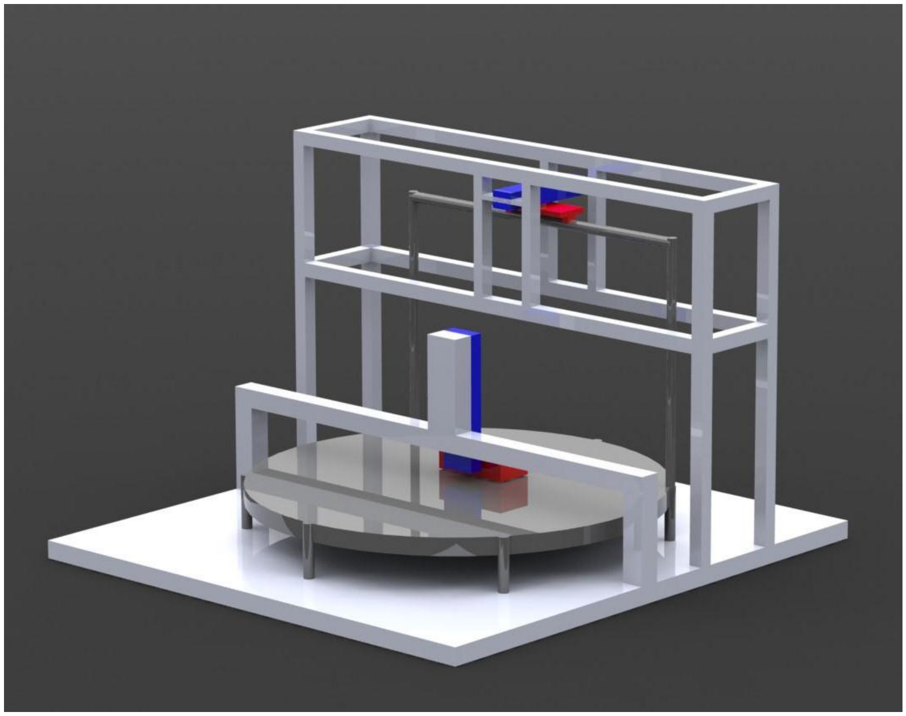

Figure 21. Rear Isometric View of Linear Motor Concept

Linear Motor Pros and Cons

Using linear motors as the power system for this machine allows for greater precision and infinite adjustability of the impact test sequence. Because the linear motors are computer controlled, the stroke and the speed are easily adjustable via changing the parameters in the motors' controlling software. This allows the machine to easily switch from tests on the adult trampolines to tests on the kid's trampoline models. Furthermore, the impact pattern is adjustable as well. If desired, the impact stroke and speed can be programmed to follow a sequence rather than remain constant.

Another benefit of using linear motors is that the power system is not exposed. Electronics and circuitry will be enclosed, leaving only a guide track and the sliding forcer. This eliminates the need for mechanical power transmission via a larger structure as the motor directly manipulates the impactor. Practically, this equates to safety because the moving components are confined to small spaces that can easily be enclosed by further protection. Even though linear motors are a more complex power system than a simple induction motor, the linear motor's added complexity is offset by the more simple structure needed to support it. With a linear motor, the most complicated pieces are purchased and there is no need to fabricate or structurally support complex linkage components.

The drawbacks of the linear motor concept are clear: cost and software complexity. Linear motors are much more expensive than other forms of linear or rotary actuation. Since cost is key, the inability to secure a linear motor that falls within budget would cause the cessation of the consideration of this concept. Another limitation is the possible complexity of software setup. Although it imparts more test flexibility, the more complicated software for controlling the linear motors could prove to be not worth the added benefits. Until a specific linear motor model is located, this concern cannot be adequately evaluated.

Linear Motor Specification Satisfaction

With the right detailed design, this concept is capable of satisfying all of the specifications that were derived from JumpSport requirements, with one possible exception. A correctly sized linear motor would be able to cycle up to 150 cycles per minute as required, impacting the mat with the force of a 250 lb jumper experiencing 6 g's of acceleration. This concept allows for implementation of the handlebar impactor as well – the same software used to operate the mat impactor would be used to control the handlebar impactor as well. With the right sensors and software, it would be fairly straightforward to implement data acquisition and automatic shut-off in the case of failure.

The biggest concern with this concept is cost. A properly specified linear motor can meet the other requirements, but may be way out of the price range (roughly $5000). Research into linear motors sized for the mat impactor has yielded quotes from $600 (From Misumi, Inc. Motor only), to a ballpark $20,000 when talking to a Parker distributor.

A secondary concern is whether a linear motor operating near its capacity will last for hundreds of millions of cycles likely to be seen by the fatigue test machine. This is unknown, but the issue is second to cost.

Supporting Preliminary Analysis

The main analysis performed during the conceptual design stage was theoretical calculation and physical validation of the maximum force that a jumper would exert on the trampoline during the impact cycle. In order to develop correct power system requirements, the trampoline loading conditions needed to be better understood. It was clear that, because the mat impact cycle had a much higher peak force than the handlebar impact cycle, the mat impact stroke requirements would likely drive the system's cost.

JumpSport had conducted various tests consisting of dropping a weight onto the trampoline mat from certain height and measuring the maximum acceleration. During these tests, they measured

a peak acceleration of 6 times the acceleration of gravity. Using this result, along with the 250 lb. weight limit of their fitness rebounder, the maximum force of the impact stroke was estimated to be 1500 lbs. This value corresponds to the peak force that a jumper at the maximum weight limit of the trampoline could exert upon the trampoline. After this initial calculation using JumpSport's data, it was clear that an independent verification of the results was necessary.

Through preliminary testing using an iPhone accelerometer application, a maximum acceleration of 3.5 times the acceleration due to gravity was measured. For a 250 lb jumper, this means that the maximum force they exert on the trampoline would be 875 lbs. Because of the unverifiable nature of the instrumentation of this preliminary test, another test was conducted using a calibrated accelerometer, an oscilloscope, and two jumpers of different weights. Utilizing an accelerometer and oscilloscope from the vibrations lab, the acceleration profiles were obtained during actual jumping. This method of testing allowed all of the variables present during normal trampoline use to be included in the test as well. Two different jumpers, one weighing 155 lbs and the other weighing 250 lbs (weights strapped to the second jumper were used to reach a total weight of 250 lbs), jumped on the trampoline while acceleration data was recorded on the oscilloscope. The resulting data showed that, for normal jumping, the maximum acceleration was about 3 g's for the 250 lb jumper and about 3.5 g's for the 155 lb jumper. When the jumpers were told to jump as hard as they could, the 155 lb jumper saw a peak of 5.5-6 g's and the 250 lb jumper saw a peak of 4-4.5 g's. During the second portion of the test, the 155 lb jumper was able to just nearly bottom out the trampoline mat, while the 250 lb jumper was able to bottom out the mat.

From these tests, it was determined that the trampoline mat bottomed out (corresponding to a mat deflection of 12 inches) with a maximum force of around 1200 lbs. Based on these tests, the specifications for the power system were set at delivering 1500 lbs maximum force at 150 cycles per minute. In order to add a bit of a factor of safety on the powertrain sizing, 1500 lbs was used as a design criteria instead of the 1200 lbs needed to bottom out the mat.

Conceptual Design Conclusion

The linear motor concept is a versatile platform that allows for accurate, variable testing of the trampoline system. With proper software, the test sequence is easily adaptable to numerous different tests that JumpSport may run. In purchasing a linear motor, JumpSport can be assured of an efficient, adaptable, and compact powertrain system. A linear motor could be implemented to have a fixed-stroke cycle, similar to a slider-crank mechanism, or a cycle where power is input intermittently as in the dribbler design. The drawback with the linear motor design is cost. Unfortunately, after quotes were solicited for linear motors, the cost was indeed found to be extremely prohibitive at the required levels of force, stroke, and speed.

Intermediate Design Development

During the conceptual design review with JumpSport, before linear motors were found to be financially unfeasible, it was clear that the advantages of the linear motor design were appealing. Because cost was suspected to be an issue, a secondary design was proposed that would utilize hydraulic cylinders instead of linear motors. Hydraulic cylinders would be able to provide the necessary load at a more reasonable price. Furthermore, hydraulics would allow for handlebar testing to be integrated easily into the system. At the Fall 2012 Senior Project Expo, JumpSport came across a machine that used hydraulic cylinders to perform life cycle analysis on ball joints. The hydraulic power system was similar to the proposed linear motor design and inspired research into using hydraulic cylinders in place of the linear motors.

At the start of winter quarter, numerous companies were contacted with the intent of fully understanding the capabilities of a hydraulic-powered impact machine. If a suitable setup was found to be financially reasonable, the hydraulic system would be specified and purchased. Concurrently to this research, the detailed design of the supporting structure commenced.

Preliminary finite element analysis was performed on a steel plate with a 1500 lb downward point force in the center and reaction force split evenly at each of the four corners. The reaction loads were initially modeled as points and then modeled as box tubing supported by the ground. The plate thickness and box tubing sizes varied. Results revealed large maximum deflections in the center of the plate from 0.9 to 3.0 inches.

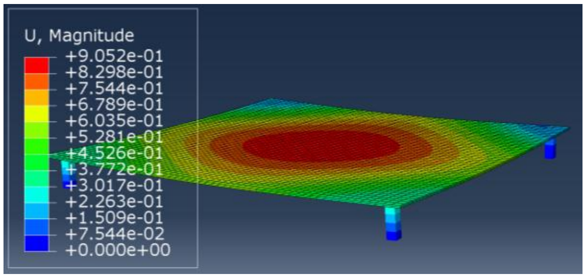

Figure 22. Preliminary FEA deflection analysis on machine base plate.

The purpose of this initial analysis was firstly to act as a tutorial to learn how to operate Abaqus, an FEA tool, and secondly to provide a backbone for further analysis.

Further analysis was abandoned when the power system became the immediate priority. Further analysis would be focused on distributing the 1500 lb force into a more realistic six points since the trampoline has six points of contact with the steel plate. If this yielded unsatisfactory results, the plate would be supported by beams under the plate; this would be modeled similarly to the preliminary models, but with a much smaller area contained within the steel framing.

The initial stages of structure design were difficult because of the uncertainty in the power system. With the size, number of components, and mounting requirements undecided, detailed design of the structure progressed slowly.

The three major companies that advised on hydraulic systems were JMR Manufacturing of Creston, CA, Motion & Flow Control Products of Portland, OR, and Zemarc, a Parker distributor located in Fresno, CA. Contact information for JMR Manufacturing was obtained from the senior project group that designed the ball joint tester. Both companies assured that hydraulics could meet the requirements of the trampoline fatigue test system and began working on specifying components for such an application. After some weeks, the hydraulic contacts began to admit that, while hydraulic cylinders could meet system requirements, the required impact frequency (150 cycles per minute or 2.5 Hz) was near the limit of their effective operation range. Both JMR and Motion and Flow suggested the possibility of using a hydraulic motor in conjunction with a slider-crank linkage. With the feasibility of hydraulic cylinders in question, Professor Widmann was contacted and recommended against using hydraulics of any kind for this senior project's application. He confirmed that hydraulic cylinders would not be able to simultaneously meet speed, stroke, and force requirements and instead recommended a slider-crank mechanism and electric motor.

At this point it became clear that the hydraulic cylinder option was not feasible and that a hydraulic motor with a slider-crank was unnecessarily complex. Effort was focused to design a slider-crank linkage powered by an electric motor.

Before an appropriate induction motor could be selected, the force and stroke characteristics needed to be translated to the torque load on the motor. This calculation required the linkage dimensions to be designed. To this end, the linkage was statically analyzed using the angle of rotation of the crank arm clockwise from top dead center. The reaction force resulting from impact with the trampoline mat was estimated by modeling the trampoline as a linear spring. At maximum extension of the impactor, the instantaneous torque on the motor would be zero due to the crank arm and connecting rod being collinear. Therefore the maximum torque on the motor would not correspond to the maximum reaction force from impact. An equation for torque applied to the motor with respect to the angle of the crank arm was statically derived and used in conjunction with the linear spring model of the trampoline to identify the characteristics of the applied torque. This equation was incorporated into an Excel spreadsheet in order to calculate torque throughout the entire impact cycle. It was discovered that the maximum torque on the motor was produced with the crank arm at approximately 130° clockwise from top dead center. Additional functionality was built into the Excel spreadsheet so that, in addition to the applied torque, the required power, average torque, and average power were also calculated. The linkage dimensions were used as variables, allowing different linkage proportions to be compared. The lowest torque and power requirements were found with linkage dimensions as follows: a crank

arm of 6 inches and a connecting rod (the member attached to the impactor) of 16 inches. Using 1200 lbs as the maximum force required to depress the trampoline mat, the maximum torque necessary was calculated to be 4226 in-lbs. and the maximum power necessary was calculated to be 10 hp. The full spreadsheet with detailed results is included in Appendix E and graphical representation of those calculations is shown in Figures 23 and 24. With this tool in hand, the search began for a motor capable of supplying the required torque and power needed to operate the slider-crank linkage.

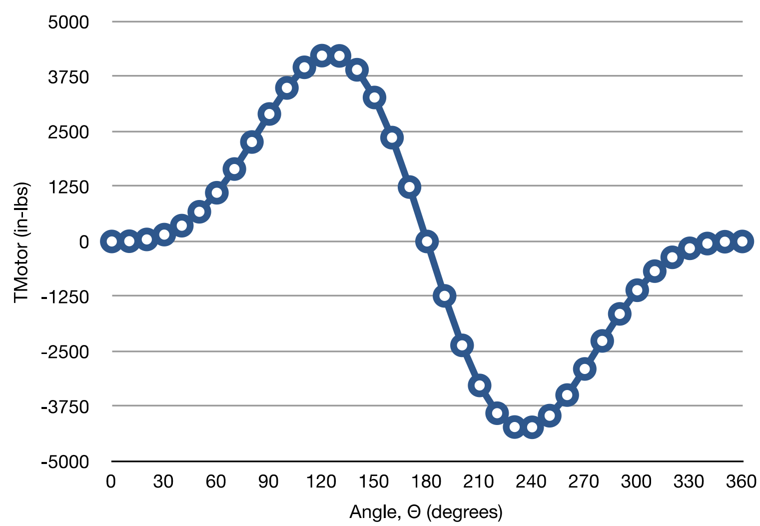

Figure 23. Torque applied to the motor during one impact cycle. Angle of the crank arm is zero at top dead center and progresses clockwise.

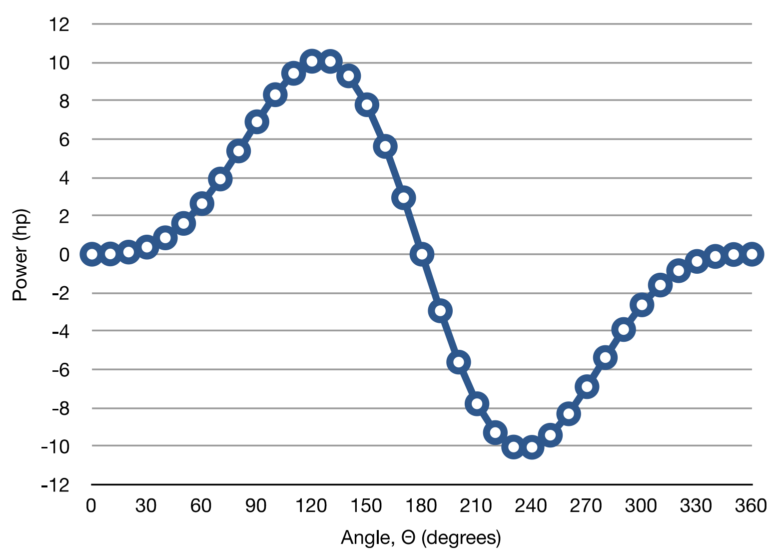

Figure 24. Power required by the motor during one impact cycle. Angle of the crank arm is zero at top dead center and progresses clockwise. Maximum power required is shown to be 10hp.

The search for an electric motor began with a brief survey of different classes of motors. Initial efforts were focused on three phase induction motors, as they are the most accessible and cost effective option. JumpSport's headquarters does not have access to three-phase power, necessitating a phase converter if a three phase induction motor was to be used. SEW Eurodrive, a respected motor supplier, was recommended by Professor Widmann as a potential source for a high-quality gearmotor. For this application, a gearmotor is desirable because the speed reducer is designed specifically for the motor. The motor-speed reducer unit can be easily integrated into the system as a single unit without any concerns that might arise from sourcing them separately and coupling them together. Using SEW Eurodrive's online motor selection tool, quotes were obtained for various motor system configurations. When 1500 lbs. was used as the value for the maximum force on the trampoline mat, the resulting motor configuration, sized to provide the maximum torque as calculated using the aforementioned Excel spreadsheet, had a final cost of $4720 (for actual quote, see Appendix C). This price included a three phase induction motor with an integrated speed reducer, as well as a variable frequency drive (VFD). The VFD allows continuous variation of the motor's speed from stationary up to full load speed, allowing the motor to be capable of a gradual start and the trampoline testing parameters to be more adjustable. A secondary quote was obtained using a maximum force of 1200 lbs. instead of 1500 lbs. For this configuration, the gearmotor and VFD combination was quoted at $3910.

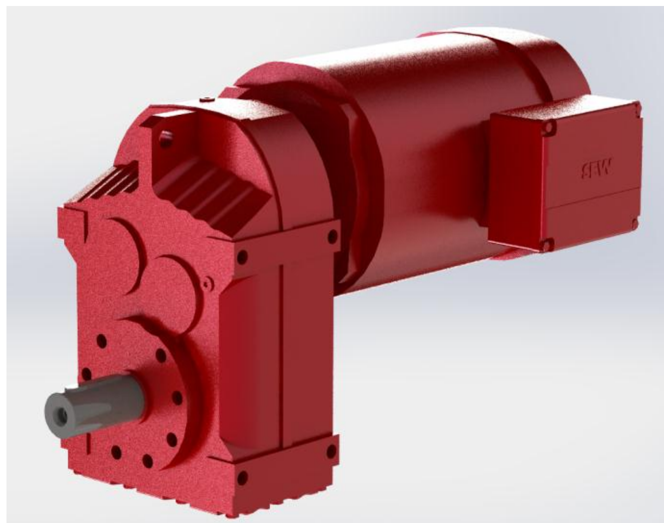

Figure 25. SEW Eurodrive parallel shaft helical gearmotor, F series

After consulting JumpSport, it was concluded that their current facilities would not be able to supply power to a 10 hp three phase motor. JumpSport's testing facility was limited to providing single phase power at 220 V on a line rated at 50 A (40 A available for continuous use). Therefore, the motor chosen must either be single phase or consume power at a level low enough to be compatible with a phase converter. The gearmotors specified from SEW Eurodrive were three phase machines and would not be able to power the system if run through a phase converter because of the power loss inherent in the phase conversion process. At the conclusion of this discussion, it was decided that a less expensive, single phase induction motor would be preferred over a more sophisticated, more expensive gearmotor.

Figure 26. Leeson 10hp farm duty motor sold on Ebay.

Ebay offered a small selection of cheaper single phase 10hp motors. The cheapest motor that was found that met all requirements was the Leeson 140706 farm duty motor (Figure 26). The 1740 rpm motor operates at 230V, 40Amp full load, and 60Hz. The motor was designed for high breakdown torque because it was specifically designed for air compressors, pumps and fan and blower applications. The motor is valued at $721.05, over six and a half times cheaper than the SEW Eurodrive motor. Note that unlike the SEW Eurodrive gearmotor, the Leeson price does not include a gear reducer. However, speed reducers are available on Ebay that could accommodate the Leeson motor. Averaging a couple hundred dollars and rarely exceeding one thousand dollars, the additional cost of the speed reducer would be 2.7 to 5.6 times cheaper than the gearmotor depending on the speed reducer used. A particular reducer was not specified as design efforts were rerouted to pursuing a less powerful system that would in turn pull less electrical power.

After consulting JumpSport, it was determined that a less powerful motor was desirable. In order to retain the same amount of energy through the linkage to the trampoline, the use of a flywheel was investigated. The energy gained by the flywheel inertia would make up for lost power in a motor with less horsepower.

A flywheel energy savings calculation was used to see if the needed power could be reduced. The flywheel study focused on the use of the flywheel as the crank arm member of the linkage. The only dimension of the flywheel that was varied was the thickness. The diameter of the flywheel was locked in from the stroke length requirements. As flywheel thickness increased, the calculated weight also increased. The weight and dimensions of the flywheel were inputs to the equation that calculated the power that could be saved using the flywheel versus having one of negligible weight. It was found that it would be most beneficial to have a flywheel weighing 50 pounds. It was found that the 50 pound flywheel would save 90 inch-pounds-force of torque if a 10% allowable speed drop during loading was assumed (see Appendix E). This was not very significant compared to the final mean needed torque value of a continuous 1900 inch-pounds-force.

Continuing with the flywheel energy savings concept, research was done on an external flywheel. The use of an external flywheel was studied in relation to a hydraulic punch press. The punch press gets a heavy external flywheel spinning at a higher rate than the system's linkage. The external flywheel is connected to the system through belts. The large amount of energy contained within the spinning material could be utilized by the punch press when needed. This was a potential solution to reducing the needed power but was going to be too much of a safety hazard. If any of the fasteners or parts supporting the external flywheel failed, it would have enough rotational energy stored to potentially take out a person, wall, or vehicle.

Senior Project Team Merger

It was evident to the group and sponsor that the power system needed to impact the trampoline properly was going to be too expensive for an individual group's budget. JumpSport brought to attention that they sponsored another senior project group that would need a similar motor and linkage. Each team had a budget of approximately $5000 and it was decided it would be in both group's best interests to collaborate and join budgets. With $10,000, the groups would have a large enough budget to properly power the system and impact the trampoline with the 250 pounds of force needed (max weight restriction for the rebounder). When the two groups came together, the project was split into two phases. The first phase was completing a machine that would fatigue test the trampolines. The second phase was completing the additional parts and fabrication needed to use the trampoline fatigue test machine to test the fatigue life of individual bungee cords and springs. When the groups came together, collaboration was needed often to ensure both group's needs were met. Adjustable frequency and adjustable stroke were newly added design criteria.

Chapter 4: Final Design

Figure 27. Final Fatigue Test Machine

Functional Description

Motor

The motor used to power the trampoline fatigue test machine will be installed as part of the second phase of this project. The senior project team consisting of Ryan Murphy, Chris D'Elia, and Andrew Brock will see the motor delivery, installation, and testing through to completion. In order to maximize precise control of the machine and consume less power, an approximately two horsepower servomotor will be used. Final motor specifications resulted from detailed system simulations conducted by Chris D'Elia. The simulation results and motor specifications will be presented upon the completion of the second phase of this project (Fall 2013).

Linkage

The slider crank linkage converts the rotary motion of the motor into the purely translational vertical movement of the impactor. The linkage itself is made of several components, namely the crank arm (flywheel), the connecting rod, and the crank slider (ramrod). The trampoline mat impactor, along with the load cell, is mounted on the end of the ramrod. In order to reduce the peak power required by the motor, the crank arm was designed as a flywheel. This allows the motor input power to maintain the flywheel's speed rather than directly overcoming the peak inputs required to impact the mat at 2.5 Hz. As the trampoline mat is depressed, the energy stored in the flywheel is used, and upon rebound of the impactor and the trampoline mat, energy is transferred back to the flywheel. This promotes system stability and lowers the necessary power input. The flywheel is attached to the drive shaft via a flange coupling, and is supported by two pillow blocks located on the drive shaft. The ramrod is supported by two pillow blocks as well.

One of the main features of the linkage is the ability to adjust its stroke. This feature is essential for the trampoline fatigue test machine to accommodate both trampoline and bungee fatigue tests. The hole pattern on the flywheel and connecting rod allows the entire linkage to be adjustable to accommodate both testing objectives of the machine. While the initial completion effort was focused on finish the first phase of the project – the portion of the machine capable of fatigue testing an entire trampoline – it was necessary to design the linkage with the design parameters of the bungee cord tests as well. For the trampoline system tests, it was essential to have a linkage capable of a maximum stroke of 18 inches. This allows for 12 inches of travel in contact with the mat and 6 inches of rebound off of the trampoline mat. The details of the linkage design were conducted by Ryan Murphy, Chris D'Elia, and Andrew Brock, and will be presented at the completion of their senior project in Fall 2013.

Frame/Gantry

The gantry provides support for the linkage, impactor, and motor subsystems, and provides the platform on which the trampoline sits. The gantry consists of the upright support members and the horizontal cross bar; together these components support the motor, linkage, and pillow blocks. The gantry uprights are bolted to the base plate, which is attached to the base frame. The base frame supports the entire structure and is itself mounted on four leveling casters. The casters allow the entire machine to be jacked down and easily wheeled around, then jacked back up into its fixed state for testing. The gantry was also designed to be adjustable in order to accommodate the range of linkage strokes.

Impactor

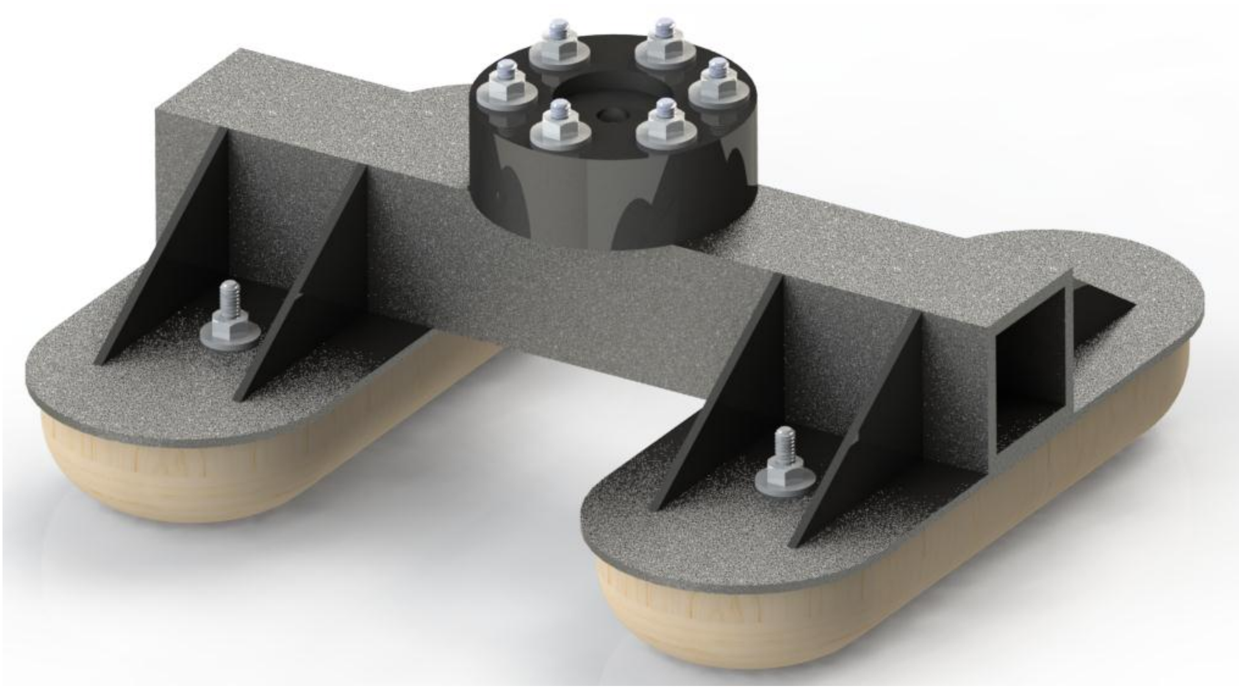

Figure 28. Impactor Subassembly

The impactor portion of the trampoline fatigue test machine is critical because it serves as both the interface between the machine and the trampoline mat as well as the platform for data acquisition. The impactor was designed to be light, have infinite life for fatigue, mimic the footprint of a human jumper, and incorporate a load cell for impact force measurement. In order to accommodate a variety of tests, the interface between the impactor, load cell, and slider rod must also allow for interchangeability of various attachments. It is also important that the impactor design remain as simple as possible while achieving these goals in order to minimize cost and manufacturing complexity.

The impactor subassembly was designed to have two main sections: the load cell and the impactor structure. The RSB6 Low Profile Pancake Load Cell from Loadstar Sensors is a short and wide cylinder, measuring 90 mm (3.54”) in diameter and 25 mm (0.98”) thick. It has six through holes on a 66 mm (2.60”) diameter circle for one side’s attachment, and one threaded M12 hole through the center for the other side’s attachment. This load cell was chosen for its compact dimensions and ease of integration into the impactor-slider rod system. The load cell attaches to the flange plate at the end of the slider rod by six ¼”-20 bolts in a circular pattern. The bolts go up through the bottom of the counter-bored through-holes provided in the load cell and are secured to the flange plate by nuts and washers. The impactor assembly attaches to the load cell by a M12 screw that threads into the mounting hole provided on the underside of the load cell.

The structure of the impactor is simple, with a single square tube 1 foot long serving as a horizontal member to which the two impactor feet attach. Each impactor foot consists of a wood piece that physically impacts the mat and is supported by ⅛” thick steel plate. The wood foot piece is rounded to reduce abrasion and tearing of the trampoline mat, and has two counter-bored through-holes for attachment to the steel support plate. Each support plate has corresponding through holes so that the wood foot piece can be attached to the support plate by two flange-head ¼” bolts secured with nuts and washers on the plate side. The counter-bores in the wood allow for the flange bolts to sit below the surface of the wood, which is important so that the bolts do not tear through the trampoline mat. Because each is mounted to a support plate, a wood foot piece only experiences compressive loading. This design allows the entire bending load to be taken by the foot support plate and the supporting gussets rather than the wood foot piece. This is critical for meeting the infinite fatigue life specification. Each foot support plate is made of ⅛” thick steel and welded to the base of the square tube that serves as the horizontal member. Fatigue analysis was done on the foot support plate in order to ensure it would not fail due to bending fatigue. Due to the need for a light impactor, a support plate thickness of ⅛” was chosen. In order to ensure the integrity of the plate over millions of loading cycles, four gussets (two on either side) were added per impactor foot. The gussets extend three inches from the edge of the horizontal member and ensure that the bending load does not cause failure of the support plate due to fatigue. Also, two gussets per side of the support plate allows for both balanced support of the plate and clearance for tightening the nut on the foot piece attachment bolt. The foot support plate is also rounded to match the curvature of the wood foot piece. This ensures that no sharp metal corners interfere with the mat surface during the mat's depression.

The horizontal base member was designed with two main considerations: weight and infinite bending fatigue life. The impactor orientation, a spread stance with two points of impact, needed a structural element that would span the stance and attach to the load cell on the end of the slider rod. The square tube section chosen marked a balance between substantial rigidity to avoid fatigue failure due to the bending load and the weight of the member. Square tubing is also inexpensive and the flat sides allow for simple attachment of the other impactor components by welding. The square tube section chosen was 2” by 2” with a 0.188” wall thickness. In order for the impactor to attach to the load cell, a hex head M12 screw is installed up through the bottom of the impactor horizontal member and into the threaded hole at the center of the load cell. The screw passes through only one side of the horizontal member so that the preload tension in the screw does not compress the square tubing. A socket clearance hole in the bottom face of the horizontal member allows for installation and tightening of the M12 screw through the bottom of horizontal member. This M12 screw does not take any of the compressive load upon trampoline impact; those forces are transmitted directly from the horizontal member to the bottom face of the load cell. The screw, then, merely serves to keep the system fastened together when the impactor leaves the mat. The M12 screw can easily handle any tensile load between the load cell and horizontal member. Preload in the screw will also be sufficient to prevent the impactor from

slowly loosening over the course of millions of cycles. Blue Loctite will also be utilized to prevent loosening or rotation of the interface.

This overall impactor design allows for easy interchangeability of different test fixtures. Particularly, when single bungee tests are desired, a specific bungee test fixture can be mounted to the end of the load cell using the same M12 screw. This feature imparts versatility to the machine; in conjunction with the range of system strokes, the fixture interchangeability allows for different types of tests to be run. This way, the machine's testing capability is not limited by the single impactor.

Safety Cover

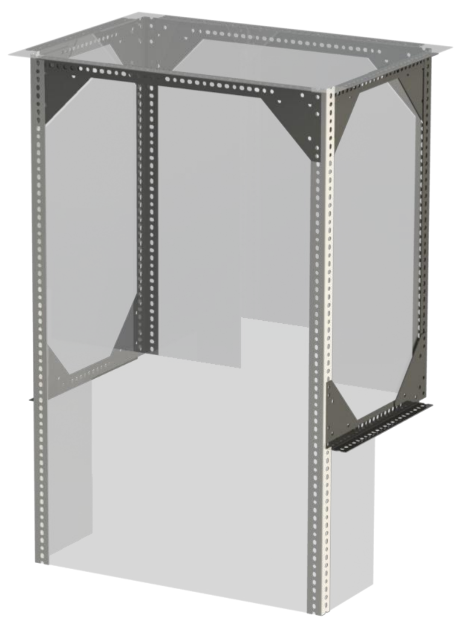

Figure 29. Safety Cover

The safety cover will be made of impact-resistant polycarbonate sheets mounted in a steel plating framework and will be attached to the gantry. The cover will extend down to cover the flywheel and pillow blocks and will extend up to cover the top of the linkage at its top dead center position. The shielding structure will also cover the rear portion of the flywheel that faces the motor. The transparent plastic structure allows for observation during the machine's operation while protecting the observer if anything were to break. If a failure occurred, any projectiles would be contained within this protective structure. In addition, the structure prevents foreign intrusions into the rotating and sliding components of the linkage. The attachment of the safety cover will be designed and the safety cover will be built in Phase 2.

Supporting Analysis

Linkage

Linkage analysis was mainly dominated by concerns of fatigue. The complete analysis of the linkage components was conducted by Andrew Brock, Ryan Murphy, and Chris D'Elia, the results of which will be reported upon completion of their senior project in Fall 2013.

Frame/Gantry

The component members of the frame and the gantry were designed to ensure sufficient stiffness and fatigue life. These components were analyzed in detail by Andrew Brock, Ryan Murphy, and Chris D'Elia. The results of the analysis will be reported upon completion of their senior project in Fall 2013.

Impactor

The horizontal beam geometry was analyzed based on the endurance limit since fatigue was the failure mode of greatest concern. A safety factor of 2 or greater was chosen. The box tubing geometry of 2"x2"x0.188" was chosen since it resulted in a safety factor of 2.3 and it weighed almost half as much of the flat bar geometry with equivalent safety factor (See Appendix E for Endurance limit analysis).

The load cell/impactor interface is connected by a single M12 blind screw. This connection was analyzed to determine whether the design preload on the bolt provided enough frictional force to keep the fastener from unscrewing and detaching from the impactor. Based on the pressure contact area, the 5700 lb bolt preload, and the friction coefficient of 0.15, the required torque to unscrew the joint is 86 ft-lb. This value provides ample torque to keep the impactor from detaching from the load cell. Theoretically, the force from the trampoline will apply no torque on the load cell as the force will be completely vertical. In reality, the impactor may contact the trampoline at a slight angle resulting in a minor component of horizontal force. However, 23% of the max force from the trampoline must be translated into a horizontal force at the maximum distance away from the load cell in order to provide the necessary torque to unfasten the impactor. It is unreasonable to resolve that such a large percentage of the maximum force would be translated by slight inconsistencies of the impactor angle. Note that this torque value was calculated with a factor of safety of 1.5, which allows for the possibility that the bolt may exceed its preload value when impacting the trampoline (See Appendix E for load cell interface torque analysis).

Fatigue analysis was done on the foot support plate in order to ensure it would not fail due to bending fatigue. With a ⅛" thickness, the plate by itself had a safety factor for infinite life of 0.2. This necessitated adding gussets, two per side of a single impactor support plate. The gussets

effectively eliminate the cantilevered beam loading condition of the support plate, and therefore adequately address fatigue concerns due to bending.

Cost Breakdown

All raw materials and service costs are detailed in the Bill of Materials in Appendix C. The total cost for Phase 1 of the project is $3,119.30 (excluding some shipping costs paid by JumpSport directly). This overall cost excludes the motor and the safety cover which will be purchased during Phase 2. The motor will be the most expensive component of the system, and though it is not yet completely specified, an initial quote from Buckles-Smith estimates pricing just under $5,000. Regarding the safety cover, initial cost analysis estimates $220.09 for all raw materials from McMaster-Carr. Details are recorded in Appendix C.

The linkage is the most costly subassembly of Phase 1 at $1,210.55. Because the linkage includes most of the moving components, it relies heavily on light-weight, strong material and on tight tolerances. Such requirements demand higher priced components.

The load cell is the next most costly subassembly at $798.00. This includes the calibration, the analog to digital interface, and the load cell itself. It is one of the most important subassemblies as it provides JumpSport with force data to analyze for the trampoline tests.

The remaining costs are accounted for by the following subassemblies: gantry ($170.18), uprights ($124.32), base ($410.28), and impactor ($39.76). All fasteners are documented separately from their subassemblies, costing a total of $215.13. Note that listed prices do not include additional costs such as sales tax and shipping. See Appendix C for complete cost analysis.

Safety Considerations

Analysis of the potential safety hazards is critical because the trampoline test machine cycles at speeds up to 2.5 Hz and maximum impact forces up to 1500 lbs. One of the primary concerns is the potential for interference in the motor, linkage, or impactor by a person or foreign object. Although unlikely, any such contact would prove damaging to both the source of the intrusion and the machine itself. To ensure maximum safety is addressed, a protective cage will be built around the flywheel, connecting rod and slider linkage. The cover will create a barrier to both keep foreign objects out and to keep the components within the confines of the cover in case of component failure.

The crank arm was designed to work as a flywheel. The flywheel has a large rotational inertia and will have a lot of stored energy when it is rotating at maximum velocity. If poorly manufactured or designed, the flywheel would be unstable and would induce vibration in the entire system, which could cause catastrophic failure. As such, care has been taken in designing the flywheel and shaft interface, the rotational bearings, and the shaft coupling. The flywheel will also be precision cut and dynamically balanced to ensure the integrity of the system.

The impactor assembly is attached to the slider rod by a single M12 screw, so it is an important safety consideration to make sure that there is no chance of this fastener coming loose. If this one bolt was to come loose during testing, the impactor assembly would separate from the machine and could cause damage to other components or nearby objects. Based on torque and preload calculations, the force of friction on the joint is sufficient and the connection will not become unfastened during testing.

Material Selection

Linkage

Material selection is critical on the linkage components because of the high stress cyclic loading and motion. Because the linkage is moving very quickly, the weight of each component drastically affects the dynamic stability of the system. As such, the ramrod was designed to be a hollow hardened steel shaft in order to easily take the axial load and to resist buckling. The large accelerations seen by the connecting rod made aluminum the best material choice. Even though aluminum does not have an endurance limit, it was still sized to provide long life. The flywheel is made from steel in order to have the high moment of inertia necessary for energy storage while maintaining cost effectiveness.

Frame/Gantry

The gantry and frame are constructed from steel box tubing, giving the structure a strong, stiff, and relatively inexpensive composition. Box tubing allows for a rigid structure with less material; the hollow rectangular cross section resists deflection due to a larger moment of inertia about the centroidal axes. Steel plate, supported by a box tubing frame underneath, provides a platform on which the trampoline sits during testing. Also not to be overlooked, the all-steel frame and gantry are easy to machine and weld. Lighter performance metals, in addition to being more expensive, are also much harder to machine. The combination of simplicity of fabrication, material availability, and strength make steel the obvious choice for the frame and gantry.

Impactor

The impactor assembly consists of a pancake load cell, one piece of box tubing, two foot plates, eight stabilizing gussets, two wooden foot pieces, and multiple kinds of fasteners. The pancake load cell was manufactured by Loadstar and is mainly alloy steel. No material decisions were made related to the load cell because it was only available in one composition.

The steel box tubing used for the impactor is 2'' x 2" x 0.188'' and is 1' long. Fatigue analysis showed this cross section to have the lowest weight while maintaining a safety factor on infinite life of at least 2 (see Appendix E). All analysis was done with carbon steel because it is critical that the impactor's horizontal beam have infinite life. Two carbon steel plates were used for the footplates of the impactor. Steel was needed for strength in these plates, as the ⅛" plate still required reinforcing gussets to maintain structural integrity. Since the horizontal member and the foot plates are joined together, similar metals are necessary to avoid corrosion and to allow for welding.

The impactor feet will be the material that actually contacts and impacts the trampoline mat surface. The foot material must be lightweight and non-abrasive so that the surface of the trampoline mat is not damaged during testing. Red oak was chosen since it is lightweight and has smooth, fine grain providing a non-abrasive material. The feet also need to withstand the compression of impacting the mat. The maximum force that each wood foot piece will see is approximately 750 lbs (half of the 1500 total max load). With a foot piece area of 36.25 in^2, the compressive axial stress on the foot is only 21.3 psi, which is easily handled by the wood. Red oak has an allowable compressive stress of 6,760 psi. In order to make the impactor feet a more realistic representation of an actual jumper, a pair shoes was attached to another pair of wooden impactor feet. It is of interest to try rubber soles as the impacting material because the friction between the rubber soles and the trampoline mat will be different than that between the red oak and the mat. The test results of both impactor foot materials could be compared and contrasted, providing potentially useful mat design information.

Maintenance and Repair

Impactor

The horizontal impactor beam was analyzed based on the endurance limit since fatigue was the failure mode of greatest concern. A parametric table was created in order to compare a variety of box tubing and flat bar geometries. The box tubing geometry of 2"x2"x0.188" was chosen since it resulted in a safety factor of 2.1, slightly exceeding the design safety factor or 2. This box tubing was chosen over flat bar since it weighed almost half as much of the flat bar geometry

with equivalent safety factor (see Appendix E for endurance limit analysis).

The load cell impactor interface is connected by a single M12 blind screw. This connection was analyzed to determine whether the design preload on the bolt provided enough frictional force to keep the fastener from unscrewing and detaching from the impactor. Based on the pressure contact area, the 5700 lb bolt preload, and the friction coefficient of 0.15, the required torque to unscrew the joint is 86 ft-lb. This was calculated with a 1.5 factor of safety which allows for the fact that the bolt may exceed its preload value when impacting the trampoline (see Appendix E for load cell interface torque analysis). The entire impactor subassembly can also be replaced with a new footprint configuration if JumpSport desires to adapt the machine for other applications or to test other stances.

Remaining Subassemblies

Maintenance and repair considerations for the motor, linkage, frame, and gantry are an extremely important aspect of the trampoline fatigue test machine design because of the number of loading cycles the machine must endure. Fatigue tests on the entire trampoline will likely last on the order of 2-4 million cycles, and this machine has been designed to complete many tests. The bushings used do not have infinite life, and must be replaced periodically to ensure a properly functioning linkage. The fabricated components have been designed to have an infinite life, and should not need any replacements. One component that has the potential for a more expensive failure is the motor. While it was specified with longevity in mind, proper maintenance in accordance with the manufacturer's instructions will be critical for proper functioning over the lifetime of the fatigue test machine. Detailed repair and regular maintenance instructions will be presented by Chris D'Elia, Andrew Brock, and Ryan Murphy at the completion of Phase 2 of the project in Fall 2013.

Chapter 5: Product Realization

Manufacturing Processes Employed

Gantry and Frame

Manufacturing the gantry and frame consisted mostly of cutting and drilling flat bar and box tubing followed by welding the pieces together.

The flat bar was cut to length with a single operation on the vertical band saw in the Cal Poly IME shop. This band saw was used for its accuracy to within 3 thousandths on an inch. Holes were then drilled into the flat bar for the flange plates, caster plates, and the upright sleeves. A mill was used for these holes for location accuracy to the thousandth of an inch.

The much longer pieces of box tubing were not cut to length in one operation on Cal Poly campus because the larger band saw in the ME shop often cut tapers instead of vertical, perpendicular cuts. Instead, the box tubing was cut a few inches longer than the designed length with a rough cut. One end of the box tubing was then machined orthogonally by facing the part in the mill.

Due to the adjustable design of the frame and gantry, many members required hole patterns. An edge finder was used to locate this faced edge and the holes were located and drilled in reference to this edge. The holes were made in four drilling operations: center-drill hole, pilot hole, a larger second pilot hole, and finally the specified hole. The travel of the mill did not cover the faced edge and all of the holes. As a result, the box tubing had a second setup where the part was shifted down the mill table and the remaining holes were referenced from the last drilled hole. The final operation for the box tubing machining was completed at Chris D'Ellia's father's machine shop. There, a larger and accurate band saw was used to cut the box tubing to length.

All holes were deburred with a countersink by way of either drill press or hand drill.

Finally, after all the members were machined, they were aligned and welded by a certified welder at a shop belonging to Chris D'Elia's father. All frame and gantry members were TIG welded due to its strength and aesthetic appeal.

The squareness of the parts and the accuracy of both the lengths and the hole locations were extremely important to this design because of the extensive welding specified by the design. These qualities ensured that each joint would fit properly and provided an accurate reference plane for the placement of the various hole patterns necessary in different gantry or frame members.

Impactor

Every part of the impactor was fabricated except for the pancake load cell, the purchased fasteners, and the Vans tennis shoes.

The first fabrication was the steel box tubing. Both ends were cut off in order to leave the steel box tubing approximately 12.25 inches long. Next, the disk sander and the right angles were used to sand the box tubing to an exact length of 12 inches, squared and even. To make the two large holes in the box tubing, drill press and the mill were used. First, pilot holes were drilled on the mill so that they would be concentric through the two sides of the box tubing. The drill press was used to make the 1.25 inch diameter hole. The drilled holes start with a small drill bit size and slowly worked up to the larger drill bit sizes. The belt sander was used to sharpen the larger drill bits before use. The drill press in combination with the vice and some securing tools worked perfectly for the 1.25 inch hole. The smaller hole was drilled through the opposite side of the tubing using the mill.

The footplates were the most challenging part of the fabrication of the impactor because the footplates required making rounded edges. First, the 1/8 inch thick steel plate was cut into two rectangles approximately 11.25 inches by 4.25 inches. These cuts were made with the vertical band saw. Then the disk sander and the right angle tool were used to sand the plates to exactly 11 inches by 4 inches each, with well squared edges. A full scale drawing of the rounded footplate was printed and cut out so it could be traced on the metal to ensure properly rounded corner shape. Lead was used to mark the rounds. Using the vertical band saw, all corners of the plates were cut off. The difficult part in making the round was using the disk sander to remove material in the right place and at the right rate. With some practice, the rounded edges came out looking good and the shapes of the footplates perfectly matched the drawings.

The fabrication of the eight gussets was tedious. First, eight right triangles were cut, each 3.25 inches by 2.25 inches on the orthogonal sides. These triangles were cut from 1/8 inch thick steel plate using the vertical band saw. These gussets were shaped on the disk sander until they were each 3 inches by 2 inches and squared off. The foot shaping was practiced on cheap 2x4 boards and once the shaping was perfected, the red oak feet were made.

Vans authentic tennis shoes, size 14, were taken apart and wrapped around wooden, shaped 2x4 feet. The size 14 fit perfectly onto the finalized, specified dimensions for the wooden impactor feet. The wooden feet had to be faced to a smaller thickness in order to make up for the additional rubber sole material. The thickness of the impactor assembly was crucial to make for an exact stroke length.

The assembly of the impactor consisted of tightening down bolts and making steel on steel welds. First, the footplates were welded to the box tubing in their proper locations. Next, each of

the eight gussets in between the box tubing and the footplates were welded. Two gussets were welded to each side of each individual footplate. Once the welds had time to cool and were cleaned up, assembly of the impactor parts began. The pancake load cell was attached to the impacting rod that is attached to the structure above it. The pancake load cell was fastened to the impacting rod using six steel bolts. The impactor assembly was fastened to the pancake load cell with one large M12 steel bolt. After shaping, the two wooden feet were attached to the bottom of the footplates using four steel bolts, two per foot. At this point, the impactor assembly was ready to be attached to the pancake load cell. The impactor was held in place while inserting the M12 bolt through the bottom of the box tubing. This bolt was tightened into the pancake load cell's threaded center. The impactor was firmly attached to the trampoline fatigue test machine and ready for continuous testing.

Recommendations for Future Manufacturing

When using the mill, it is recommended to double check the x and y locations before each hole is drilled. Some of the mills that were used for this project had insufficient x and y locks allowing for slop in the table location. It was discovered the table would sometimes move locations from one drilling operation to the next. As a result, the holes in the base lowers did not line up with the holes in the upright flanges. However, the holes were sized for a clearance fit and the bolts tightly fit within the outer limits of the holes and were able to fasten the lowers to the flanges.

Following operations are made easier if the box tubing is cut initially on a precise band saw. This will avoid the necessity of the additional operation of facing each piece of box tubing to counter the taper from the band saw.

For drilling holes in gantry or frame members, use a mill where its entire travel covers the length of the box tubing plus twice the diameter of the end mill. This will reduce the number of setups and will result in more accuracy since the drilled holes will be referenced from a single location instead of two locations.

When making the supporting plates for the wooden feet of the impactor assembly, a certain order of operations proved to make for the best rounded edges on the steel plates. First, print out a full scale drawing of the part and cut it out. Trace the exact arc shapes onto the raw steel. On the squared off edges, make a single cut with the vertical band saw tangential to the 90 degree arc traced onto the corner of the steel. The closer the cut is to being tangential and in contact with the arc, the easier it proved to be to sand the remaining material off and make a well-defined arc shape that follows the traced shaped.

A noncritical recommendation is to use the same drill bits for mirrored holes of the same diameter. The drills used for this project varied. Some were slightly dull and drilled slightly

larger holes in the members. This however did not affect the manufactured outcome of the gantry or frame members because the clearance holes were not oversized by more than a few thousandths of an inch. This concept of using the same tool also gives the manufacturer the advantage of knowing exactly how the tool will operate since it has been used before.

Chapter 6: Design Verification

Phase 1 testing was mainly comprised of making sure that the assembly of the machine was possible and that all the tolerance fits worked with each other and allowed for proper system dynamics. After the final welding of the members was complete, the assembly of the machine began. The base and uprights of the machine attached to all comprising parts and each other properly and proved sturdy. The gantry was assembled and all fasteners proved to support the gantry properly. The flywheel/linkage assembly fit together properly but in attempting to run the impacting rod through the pillow blocks that support it, it was found that the welding that occurred on both ends of the rod was not taken into account. In order to fit the tightly toleranced rod through its supporting pillow blocks, the rod was cut and an internal screw and nut was machined into the rod to hold the rod together. This fix worked well and we were able to attach the linkage to the machine. Once the linkage was attached, the dynamics of the linkage was tested. The linkage flywheel spun properly and the impactor assembly and impacting rod moved up and down properly. The testing of the machine assembly and simple dynamics proved successful for Phase 1.

Future assembly and testing of the other machine components will be completed by the Phase 2 team (Andy Brock, Chris Delia, and Ryan Murphy) in Fall 2013.

Figure 30. Base welding completed and ready for assembly verification

Figure 31. Base assembly verification

Figure 32. Base, upright, and gantry assembly verification complete

Figure 33. Impact foot assembly final verification

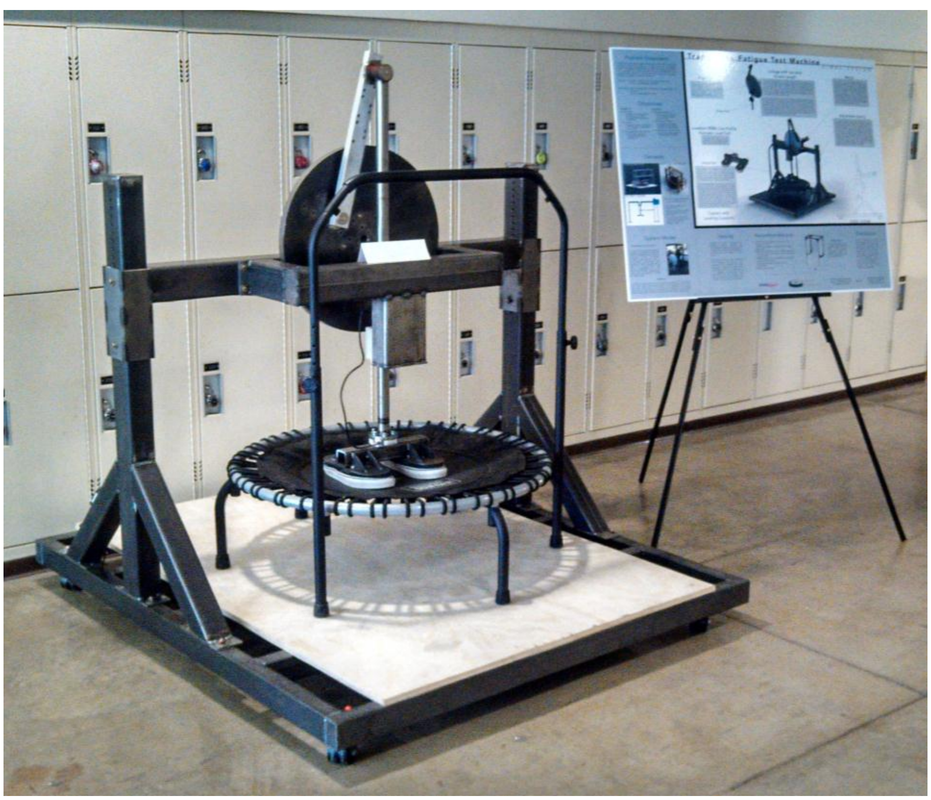

Figure 34. Completed Phase 1 assembly at expo

Chapter 7: Conclusions and Recommendations

Because the machine is not completely built and does not have a power system attached, extensive dynamic testing was not conducted. There was no finalized design or manufacturing completed for the safety cover or the cable carrier for the pancake load cell interface cable.

Cable Carrier

The first recommendation for the future tasks to be accomplished would be finalizing the design on the cable carrier. Next, the cable carrier should be purchased. A plan should be put together to complete any machining needed to enable the attachment of the cable carrier. Test that the cable carrier attaches to the machine properly and the cable fits and is routed properly through the cable carrier. Check that the cable carrier can move through the entire stroke that is necessary. Route the entire cable and interface box to the chosen final location and attach the interface to the structure. Machining might need to be done in order to attach the load cell interface box depending on the chosen location.

Safety Cover

The next task to complete would be the verification of the final safety cover design. Once the safety cover design is verified, purchase parts, cut the acrylic to the proper size in the machine shop, and assemble the safety cover. Attach the safety cover to the gantry and make sure that the safety cover fits snugly and that there are no small spaces that anything could fit through.

Power System

A motor needs to be chosen and purchased. The coupling for the motor and the drive shaft needs to be selected and purchased. A support system for the motor needs to be designed, manufactured, and attached to the machine in between the base and the motor. The steel base plate needs to be attached firmly to the structure, most likely welded, so that the motor support can potentially be welded to the base plate if needed.

The previous tasks will complete the manufacturing and assembly process of the machine. The next recommendations are related to the testing of the machine dynamics and testing of the trampolines.

Testing

The first recommendation would be to turn the motor on at low speed to make sure that the linkage flywheel and impacting rod move in the correct manner through a full rotation of the flywheel. Verify that the entire linkage assembly proves to be operational. Speed the motor up to the desired speed to test the full system dynamics. Once the machine proves to be stable and properly working at higher speeds, turn the system off. Once the system stops moving, insert a trampoline under the impactor. Repeat the motor testing process for the trampoline, starting at low speeds and working up to testing higher speeds. Tuning of the motor controller will likely be necessary to accomplish the proper impact force and stroke.

Two things to keep a close watch on when the machine is running with a trampoline are the deflection of the steel base plate and the movement of the trampoline on the base plate. If either of these happens during testing, some extra design might need to occur to keep the base plate from deflecting or to keep the trampoline from moving on the base plate.

Impactor

Two sets of impacting feet were made. Although the feet with tennis shoes molded around them proved sturdy and to be the finalized design, raw red oak shaped and sanded feet were also supplied. Testing is advised to compare the impacting force and friction characteristics related to the rubber soles versus the wooden impacting surface.

Handlebar Force Application

Lastly, if the machine proves to successfully fatigue test trampoline mats, it is recommended to attach a support and power system to apply the specified amount of 40 pounds of maximum force horizontally to the handlebars on the exercise rebounders.

References

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Appendix A: QFD Matrix

Rebounder Fatigue Test Machine

JumpSport Requirements	Weighting (1-5)	Surrounding Structure Width	Jump Rate	Life Cycles	Force Applied (Adult Trampoline)	Force Applied (Kid Trampoline)	Height of Contact	Width of Impactor	Applies Force to Both Mat and Handle Bars	Low Cost	Low Noise Level	Auto Shut-off Switch
		Engineering Requirements										
Adjustable for Diameter	5	9						3				
Variable Cycles Per Minute	5		9	9								
Stability	4		3		3	3	3				1	
Long endurance (Life)	5		3	9	3	3						
Data Acquisition	3			9	9	9						
"Craig's List" Parts	3			1						9		
Low Noise	2		3	1	1	1					9	
Adjustable Height	4				1	1	9					
Tests Both Mat and Handle Bars	2								9			
Plyo Tilt	1			1	3	3	3					
Uneven Surfaces	1			1	1	1	3					
Mimicks Human Jumping	3		9		9	9	3	3	9		3	
Shut Off Conditions for Trampoline Failure	4											9
Units		inches	cycles per minute	cycles	lbs	lbs	inches	inches		dollars	dBA	
Targets		52	120 to 150	800 million	250	175 (static) 6*g (dynamic)	6-13	30-Dec	yes	5000	51	yes
Weighted Importance		45	105	124	91	91	63	24	45	27	31	36

9	Strong Correlation
3	Medium Correlation
1	Small Correlation

Appendix B: Final Drawings

Impactor Assembly with Bill of Materials

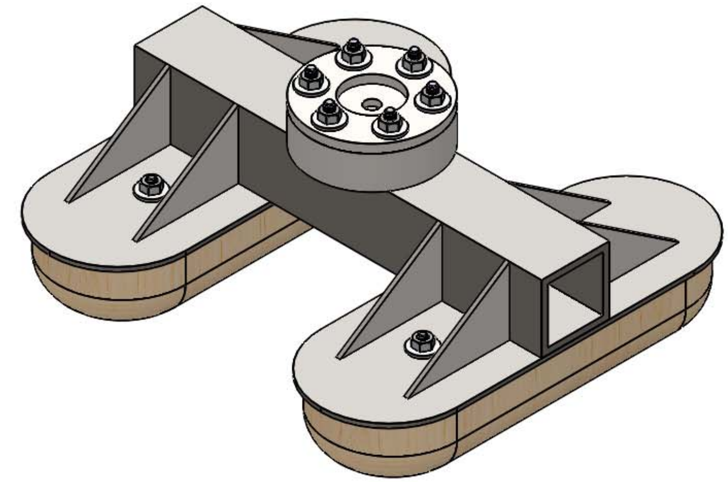
Horizontal Beam

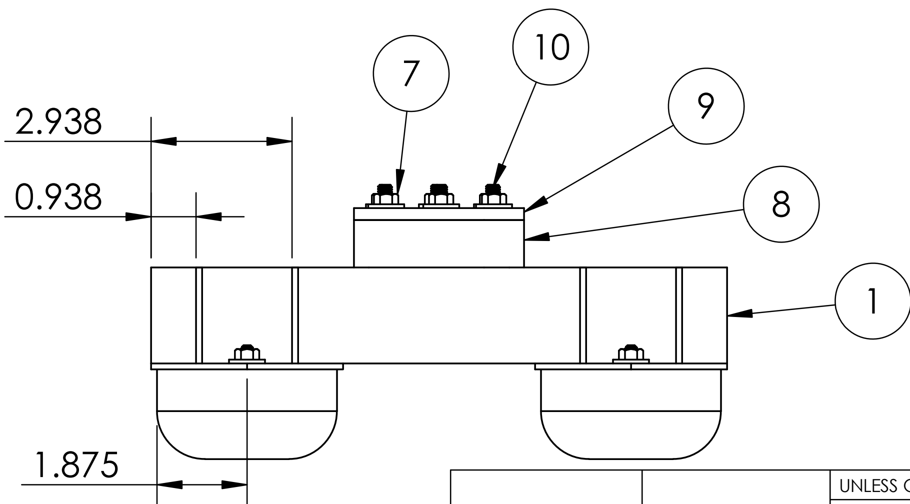
Foot Support Plate

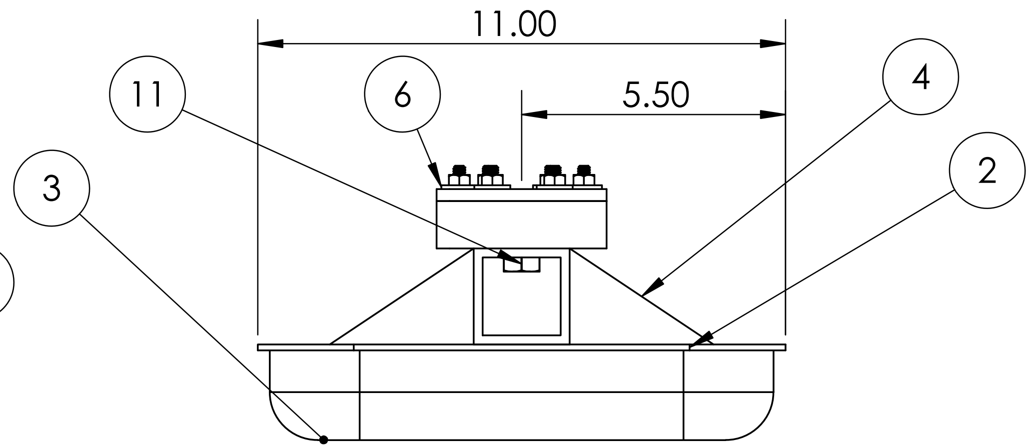
Foot Support Plate Gusset

Impactor Foot

ITEM NO.	PART NUMBER	DESCRIPTION	QTY.
1		Impactor Horizontal Beam	1
2		Foot Support Plate	2
3		Foot Piece	2
4		Foot Support Plate Gusset	8
5	92979A138	1/4"-20 Hex Flange Cap Screw	4
6	93852A102	Flat Washer (1/4" screw size)	10
7	93827A211	1/4"-20 Hex Nut	10
8		RSB6 Pancake Load Cell	1
9		Load Cell Flange	1
10	92620A546	1/4"-20 Hex Head Cap Screw	6
11	91310A709	M12 Hex Head Cap Screw	1







PROPRIETARY AND CONFIDENTIAL

SLIDER CRANK?	FINAL ASSY
NEXT ASSY	USED ON
APPLICATION	

UNLESS OTHERWISE SPECIFIED:

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

INTERPRET GEOMETRIC TOLERANCING PER:

MATERIAL : SEE BOM

FINISH

DO NOT SCALE DRAWING

	NAME	DATE
DRAWN	E. FLORY	4/26/13
CHECKED		
ENG APPR.		
MFG APPR.		
Q.A.		

COMMENTS:

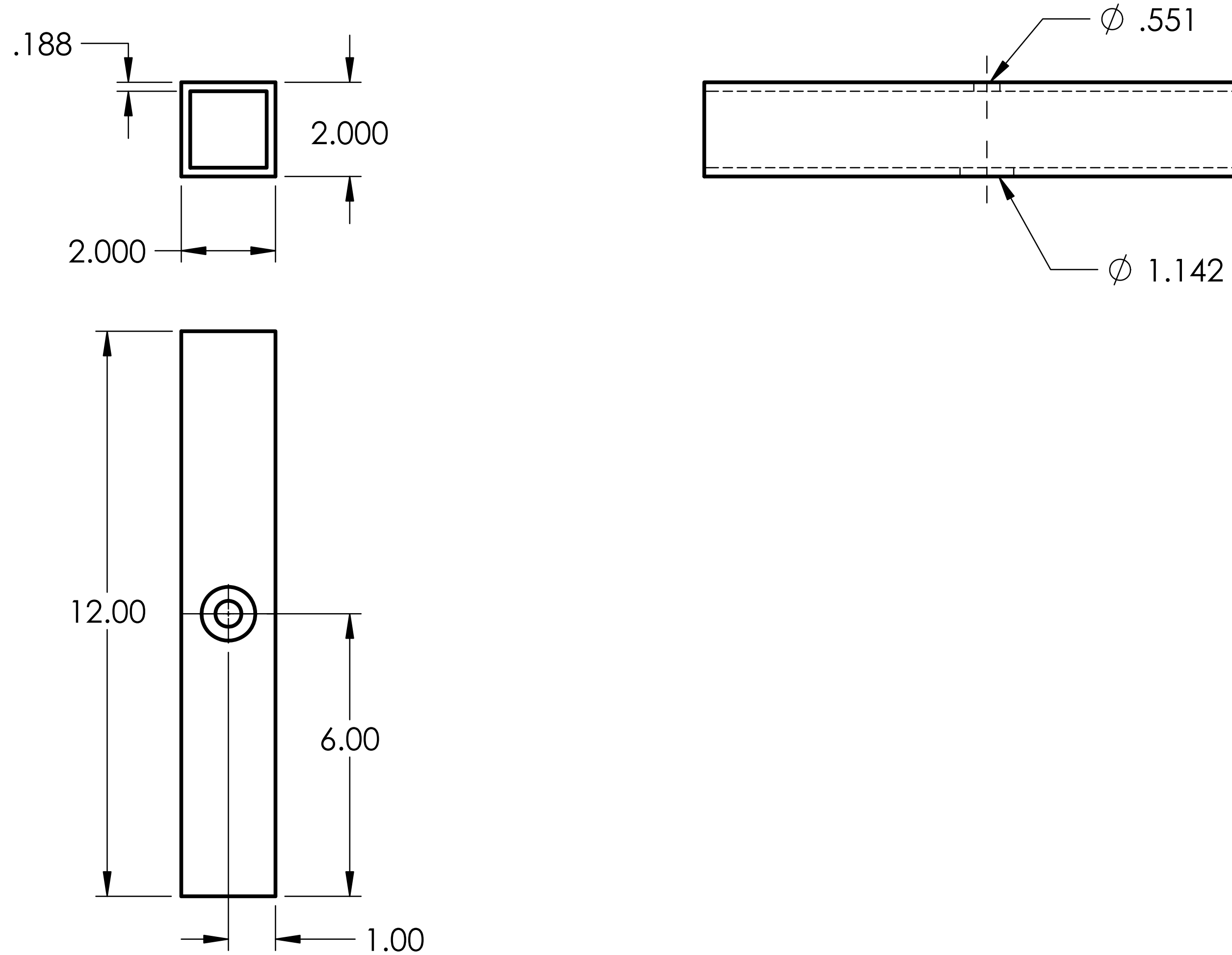
TITLE:

IMPACTOR ASSEMBLY

SIZE	DWG. NO.	REV
A		A

SCALE: 1:4 | WEIGHT: 14.4 lb | SHEET 1 OF 1

5 4 3 2 1



IMPACTOR ASSY	
NEXT ASSY	USED ON
APPLICATION	

UNLESS OTHERWISE SPECIFIED:

DIMENSIONS ARE IN INCHES
TOLERANCES: ±0.010
FRACTIONAL ±
ANGULAR: MACH ± BEND ±
TWO PLACE DECIMAL ±
THREE PLACE DECIMAL ±

INTERPRET GEOMETRIC TOLERANCING PER:

MATERIAL : AISI 1015 CD STEEL

FINISH

DO NOT SCALE DRAWING

	NAME	DATE
DRAWN	E. FLORY	4/25/13
CHECKED		
ENG APPR.		
MFG APPR.		
Q.A.		

COMMENTS:

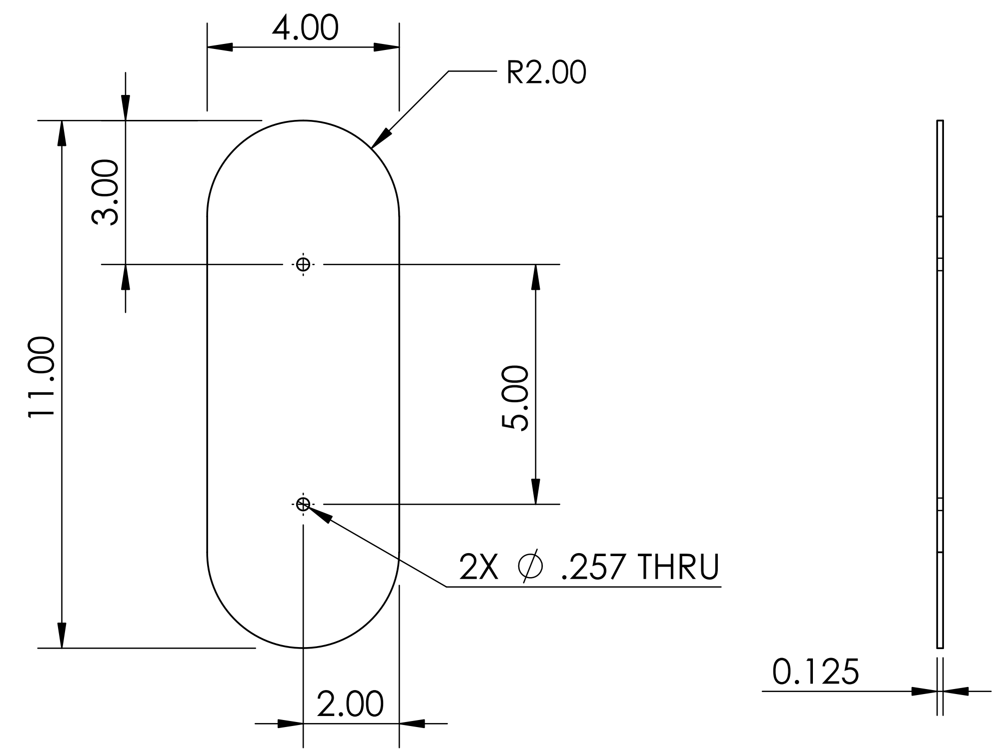
TITLE:

IMPACTOR HORIZONTAL BEAM

SIZE	DWG. NO.	REV
A	1	A

SCALE: 1:4 WEIGHT: 4.58 LB SHEET 1 OF 1

5 4 3 2 1



NOTE: Use drill F for holes.
Holes are clearance close
fit for 1/4" bolt.

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IMPACTOR ASSY	
NEXT ASSY	USED ON
APPLICATION	

UNLESS OTHERWISE SPECIFIED:

DIMENSIONS ARE IN INCHES
TOLERANCES: ±0.010
FRACTIONAL ±
ANGULAR: MACH ± BEND ±
TWO PLACE DECIMAL ±
THREE PLACE DECIMAL ±

INTERPRET GEOMETRIC TOLERANCING PER:

MATERIAL : AISI 1015 CD STEEL

FINISH

DO NOT SCALE DRAWING

	NAME	DATE
DRAWN	E. FLORY	4/26/13
CHECKED		
ENG APPR.		
MFG APPR.		
Q.A.		

COMMENTS:

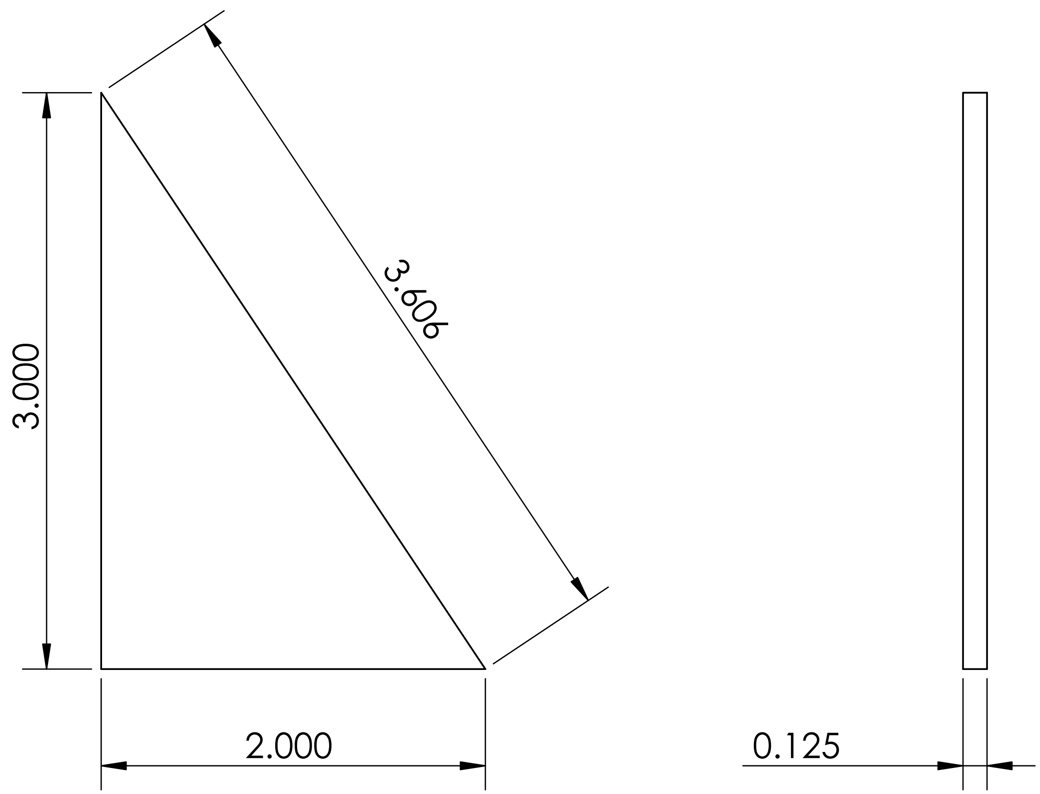
TITLE:

FOOT SUPPORT PLATE

SIZE	DWG. NO.	REV
A	2	A

SCALE: 1:4 WEIGHT: 1.44 LB SHEET 1 OF 1

5 4 3 2 1



		UNLESS OTHERWISE SPECIFIED:		NAME	DATE
		DIMENSIONS ARE IN INCHES TOLERANCES: ±0.010 FRACTIONAL ± ANGULAR: MACH ± BEND ± TWO PLACE DECIMAL ± THREE PLACE DECIMAL ±	DRAWN	E. FLORY	4/26/13
			CHECKED		
			ENG APPR.		
		INTERPRET GEOMETRIC TOLERANCING PER:	MFG APPR.		
			Q.A.		
IMPACTOR ASSY		MATERIAL : AISI 1015 CD STEEL	COMMENTS:		
NEXT ASSY	USED ON	FINISH			
APPLICATION		DO NOT SCALE DRAWING			

PROPRIETARY AND CONFIDENTIAL

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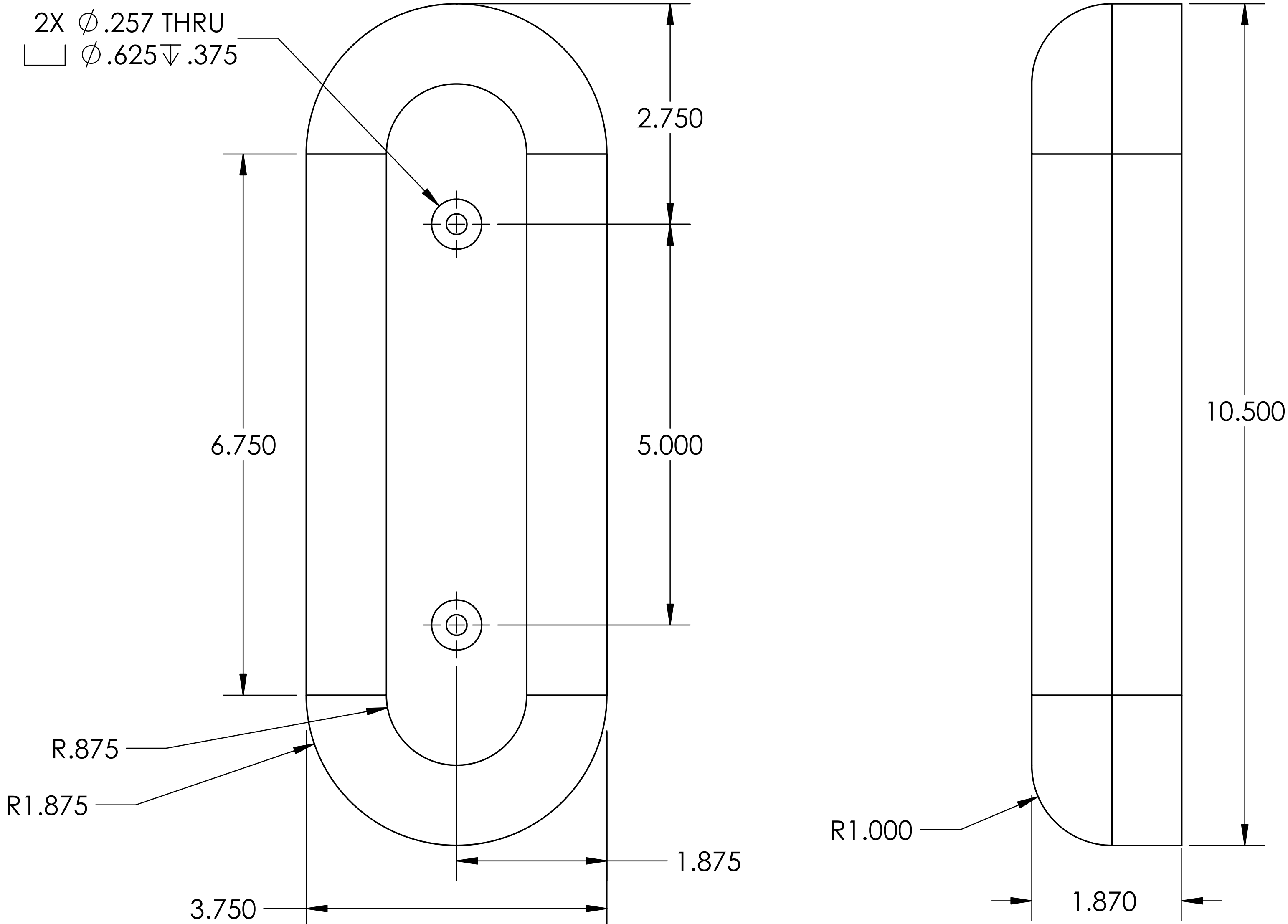
TITLE:

FOOT SUPPORT PLATE GUSSET

SIZE	DWG. NO.	REV
A	11	A
SCALE: 1:1	WEIGHT: 0.11 LB	SHEET 1 OF 1

5 4 3 2 1



NOTE:
F drill for thru hole (clearance close fit for 1/4" bolt).
For bore, use drill closest to 0.625".

IMPACTOR ASSY	
NEXT ASSY	USED ON
APPLICATION	

UNLESS OTHERWISE SPECIFIED:

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

INTERPRET GEOMETRIC TOLERANCING PER:

MATERIAL : DOUGLAS FIR

FINISH

DO NOT SCALE DRAWING

	NAME	DATE
DRAWN	E. FLORY	4/26/13
CHECKED		
ENG APPR.		
MFG APPR.		
Q.A.		

COMMENTS:

TITLE:

IMPACTOR FOOT

SIZE	DWG. NO.	REV
A	4	A

SCALE: 1:2 WEIGHT: 0.87 LB SHEET 1 OF 1

5 4 3 2 1

Appendix C: Cost and Vendors

List of Vendors

Overall Cost Analysis

Safety Cover Cost Analysis

Hydraulic Motor System Motion and Flow Quote

SEW Gearmotor Quote

LIST OF VENDORS

Vendor	Website	Telephone	Fax	Email
McCarthy Steel	http://www.slohomepage.com	(805) 543-1760		
Onlinemetals.com	http://www.onlinemetals.com/	800-704-2157	(206) 285-7836	sales@onlinemetals.com
McMaster-Carr	http://www.mcmaster.com/#	(562) 692-5911	(562) 695-2323	la.sales@mcmaster.com
Load Star	http://www.loadstarsensors.com/	(510) 274-1872	(510) 952-3700	
Pacific Coast Lumber	http://pacificcoastlumber.com/	(805) 543-5533		

COST ANALYSIS

LINKAGE SUBASSEMBLY

Item	Details	Description	Qty	Price Per Unit	Price Per Ft	Price	Source	Part Num.
Flywheel	Includes Material and Water Jet Cutting	Flywheel	1			$ 250.00	H20 Systems (Hayward)	
Aluminum Extruded Rectangle	1"x2" ; 6061 - T6511	Connecting Rod	2 ft	$ 20.31		$ 20.31	OnlineMetals.com	
Tubular Shaft (Unhardened)	1.5" OD 1" ID	Slider Shaft	4 ft	$ 233.47		$ 233.47	McMaster-Carr	8929T24
Pillow-Block Linear Sleeve Bearings	Closed for 1.5" shaft	Bearing for Ram Rod	2	$ 124.62		$ 249.24	McMaster-Carr	6374K141
Pillow-Block Linear Sleeve Bearings	Open for 1.5" shaft	Bearing for Ram Rod	2	$ 146.45		$ 292.90	McMaster-Carr	6374K327
Hardened Drive Shaft	1 1/2" dia; 1' in length	Motor Flywheel Shaft				$ 56.23	McMaster-Carr	1144K56
Steel Round (Cold Rolled)	2 - 7/8 " diameter	Connecting Studs				$ 38.62	McCarthy Steel	91264A798
Bronze Bushings	1" shaft dia; 1/2" flange SAE841; 1 1/8 OD		4	$ 1.96		$ 7.84	McMaster-Carr	6338K429
Flexible Spider Shaft Coupling		Motor Shaft Coupling	2	$ 30.97		$ 61.94	McMaster-Carr	5906K518
					Subtotal	$ 1,210.55		

GANTRY SUBASSEMBLY

Item	Details	Description	Qty	Price Per Unit	Price Per Ft	Price	Source	Part Num.
Mild Steel 1018 Rectangle (Cold Finish)	0.25"X3.5"	Gantry Upright	4 ft	$ 33.63		$ 33.63	OnlineMetals.com	
Steel Square Tubing	4"x4"x3/16"	Gantry Crossbar	4.5 ft		$ 10.78	$ 48.06	McCarthy Steel	
Steel Rectangular Tubing	4"x2"x3/16"	Linear Bearing Support	4 ft		$ 10.30	$ 41.20	McCarthy Steel	
Mild Steel 1018 Rectangle (Cold Finish)	0.25"X4"	Mounting Plates; Uprights	5 ft	$ 40.93		$ 40.93	OnlineMetals.com	
Mild Steel 1018 Rectangle (Cold Finish)	0.1875"X3"	End Caps for Gantry	1 ft	$ 6.36		$ 6.36	OnlineMetals.com	
					Subtotal	$ 170.18		

UPRIGHT SUBASSEMBLY

Item	Details	Description	Qty	Price Per Unit	Price Per Ft	Price	Source	Part Num.
Steel Rectangular Tubing	3"x4"x 3/16"	Corner Braces	7 ft		$ 6.30	$ 44.10	McCarthy Steel	
Steel Rectangular Tubing	3.5"x3"x 3/16"	Uprights	8 ft		$ 6.30	$ 50.40	McCarthy Steel	
Mild Steel 1018 Rectangle (Cold Finish)	3"x0.25"	Flange Mounts	5 ft	$ 29.82		$ 29.82	OnlineMetals.com	
					Subtotal	$ 124.32		

BASE SUBASSEMBLY

Item	Details	Description	Qty	Price Per Unit	Price Per Ft	Price	Source	Part Num.
Leveling Threaded Stem Casters		Casters	4	$ 73.09		$ 292.36	McMaster-Carr	9854T510
Flat Bar (Cold Finish)	1/4" x 4" x 4'	Caster Mounting Plates	4	$ 4.25		$ 17.00	McCarthy Steel	
Steel Rectangluar Tubing	3"x 2" x 0.120	Base Framework	10 ft		$ 4.46	$ 44.60	McCarthy Steel	
Steel Sheet (Hot Rolled)	14ga half sheet	Base Plate	1	$ 56.32		$ 56.32	McCarthy Steel	
					Subtotal	$ 410.28		

LOAD CELL SUBASSEMBLY

Item	Details		Qty	Price Per Unit	Price Per Ft	Price	Source	Part Num.
Pancake Load Cell	LoadStar RSB6-01KM-S Low Profile		1			$ 499.00	Load Star	RSB6-01KM-005-S
Digital Wired Load Cell Interface	Millivolt to USB converter; 16 bit; 6 ft cable		1			$ 199.00	Load Star	DI-100U
Digital Calibration in Compression			1			$ 100.00	Load Star	DCAL*C01
					Subtotal	$ 798.00		

IMPACTOR SUBASSEMBLY

Item	Details	Description	Qty	Price Per Unit	Price Per Ft	Price	Source	Part Num.
Red Oak	12"x12"x2"	Foot	1	$ 10.00		$ 20.00	Pacific Coast Lumber	
Square Tubing	2"X2"x3/16"	Foot Support	1 ft		$ 6.50	$ 6.50	McCarthy Steel	
Steel Flat Bar	4"x1/8"	Foot Support Plate	2 ft		$ 6.63	$ 13.26	McCarthy Steel	
Steel Flat Bar	2"x 1/8"	Foot Flanges	1 ft				McCarthy Steel	
Steel Plate	3.54" dia; 1/4" thickness	Load cell flange	1			Incl. w/ flywheel price	H20 Systems (Hayward)	
					Subtotal	$ 39.76		

FASTENERS

Item	Details	Description	Qty	Price Per Pkg	Price Per Unit	Price	Source	Part Num.
1/2"-13 x 4-1/4" HHCS	Grade 8 Alloy Steel	Frame	12	$ 3.53		$ 42.36	McMaster-Carr	91257A719
1/2"-20 x 1-1/2" HHCS	Grade 8 Alloy Steel; Fully Thrd	Linkage	8	$ 7.65		$ 7.65	McMaster-Carr	92620A746
3/8"-16 x 5" HHCS	Grade 8 Alloy Steel	Frame	12	$ 10.31		$ 30.93	McMaster-Carr	91257A644
3/8"-16 x 4-1/4" HHCS	Grade 8 Alloy Steel	Frame	4	$ 8.59		$ 8.59	McMaster-Carr	91257A651
3/8"-16 x 1" HHCS	Grade 8 Alloy Steel; Fully Thrd	Frame	4		$ 0.55	$ 2.20	Miner's Ace Hardware	
5/16"-24 x 1-1/4" HHCS	Grade 8 Alloy Steel; Fully Thrd	Linkage	4		$ 0.50	$ 2.00	Miner's Ace Hardware	
5/16"-24 x 1-1/2" HHCS	Grade 8 Alloy Steel; Fully Thrd	Linkage	4	$ 14.70		$ 14.70	McMaster-Carr	92620A611
1/4"-20 x 1-1/2" HHCS	Grade 8 Alloy Steel; Fully Thrd	Impactor- flange/load cell	6	$ 10.28		$ 10.28	McMaster-Carr	92620A546
1/4" - 20 x 1" HHCS	Grade 8 Alloy Steel; Fully Thrd	Linkage	2				Miner's Ace Hardware	
1/4"-20 x 3/4" HHCS	Grade 8 Alloy Steel; Fully Thrd	Frame	16	$ 11.65		$ 11.65	McMaster-Carr	92620A540
1/4"-20 x 2" HHCS	Grade 5 Zinc-Plated Steel	Impactor - foot	4		$ 1.23	$ 4.92	Miner's Ace Hardware	
M12 x 25mm HHCS	Class 10.9; 1.75mm pitch; Fully Thrd	Impactor- beam/load cell	1				Miner's Ace Hardware	
Flat Washer (1/2" screw size)	Grade 8 Steel; 1-1/16" OD; .09"-.18" Thk	Frame and Linkage	32		$ 0.50	$ 16.00	Miner's Ace Hardware	
Flat Washer (3/8" screw size)	Grade 8 Steel; 13/16" OD; .05"-.08" Thk	Frame	30		$ 0.23	$ 6.90	Miner's Ace Hardware	
Flat Washer (5/16" screw size)	Steel; 11/16" OD; .05"-.08" Thk	Linkage	6		$ 0.23	$ 1.38	Miner's Ace Hardware	
Flat Washer (1/4" screw size)	Steel ; 5/8" OD; .05"-.08" Thk	Frame and Impactor	38		$ 0.16	$ 6.08	Miner's Ace Hardware	
Flat Washer (1/4" screw size)	18-8 Black; 3/4" OD; .05"-.08" Thick	Impactor	4		$ 0.16	$ 0.64	Miner's Ace Hardware	
1/4"-20 Hex Nut	Grade 8 Steel; 7/16" W; 7/32" H	Frame; Impactor	26		$ 0.20	$ 5.20	Miner's Ace Hardware	
3/8"-16 Hex Nut	Grade 8 Steel; 9/16" W; 21/64" H	Frame	8		$ 0.40	$ 3.20	Miner's Ace Hardware	
1/2"-13 Nylon Insert Hex Locknut	Grade 8 Steel; 3/4" W; 19/32" H	Frame	12		$ 0.95	$ 11.40	Miner's Ace Hardware	
3/8"-16 Nylon Insert Hex Locknut	Grade 8 Steel; 9/16" W; 29/64" H	Frame	12		$ 0.45	$ 5.40	Miner's Ace Hardware	
Large Dia. Washer (for 1/4" screw)	Large Dia 1-1/2" OD, .05"-.08" Thk	Linkage	2	$ 7.51		$ 7.51	McMaster-Carr	91525A128
Flanged-Sleeve Bearing (1-1/4" shaft)	SAE 841 Bronze; 1-1/2" OD; 1" Length	Linkage	2		$ 5.31	$ 10.62	McMaster-Carr	6338K441
Flanged-Sleeve Bearing (1" shaft)	SAE 841 Bronze; 1-1/4" OD; 3/4" Length	Linkage	2		$ 2.76	$ 5.52	McMaster-Carr	633K436
					Subtotal	$ 215.13		

ADDITIONAL COSTS

Tax, Shipping, Cutting Charges, Discount	
McCarthy Sales Tax	$ 25.72
McCarthy Cutting Charge	$ 15.00
McMaster-Carr Sales Tax	$ 82.06
McMaster-Carr Shipping	$ 6.77
Miner's Ace Hardware Sales Tax	$ 5.23
OnlineMetals.com Sales Tax	$ 13.64
Load Star 10% Student Discount	$ 79.80
Load Star Sales Tax	$ 57.46
Load Star Shipping	$ 25.00
Subtotal	$ 151.07

NOTE:
Some shipping costs are not accounted for as JumpSport paid for some orders directly.

Total	$ 3,119.30

SAFETY COVER SUBASSEMBLY

Item	Details	Qty	Price Per Unit	Price	Source	Part Num.
Impact-Resistant Polycarbonate	Current design: need approx 33 ft^2 ; 4'X4'; 1/16" thk	2	$ 41.87	$ 83.74	McMaster-Carr	8574K81
	3'X3'; 1/16" thk	1	$ 38.71	$ 38.71	McMaster-Carr	8574K246
	2'X3'; 1/16" thk	1	$ 21.98	$ 21.98	McMaster-Carr	8574K245
Zinc Plated Steel Framing	1-1/2" X 1-1/2" ; 8 ft length	3	$ 16.32	$ 48.96	McMaster-Carr	4664T18
Corner Plates	6" X 6"	8	$ 2.41	$ 19.28	McMaster-Carr	4664T22
Fasteners	50 each: 5/16" x 3/4" bolts, nuts, washers	1	$ 7.42	$ 7.42	McMaster-Carr	4664T61
			Subtotal	$ 220.09		

fluid connector products, dba
MFCP MOTION & FLOW CONTROL PRODUCTS, INC.

CUSTOMER QUOTE

Account Number	Quote Number	Quote Date
97001	P2507960	02/22/13

SHIPPED FROM PORTLAND BRANCH
2929 N.W. 31ST AVE.
PORTLAND, OREGON 97210
PHONE 503 228-0190
800 479-0191
FAX 503 228-5331

Terms: **VISA**

BILL TO	SHIP TO
WILL ROBERTSON CAL POLY FATIGUE TESTER SAN LUIS OBISPO CA 93407	TBD

Customer Ref Number	Shipped Via	Expiration	Taken by	Quoted to	Salesman
		02/27/13	TYSONN		Portland House

Line	Qty Quoted	Part Number and Description	Unit Price	Extended Amount
1	1	YMP-200-2-C-S-B HYDRAULIC MOTOR	162.42	162.42
2	1	H29.6NOPO H-PAK POWER UNIT	3,923.25	3,923.25
3	1	FC51-10SAE PRESSURE COMPENSATING FLOW CONTROL 10SAE	73.75	73.75
4	1	AOC-22-2-30-1PH THERM TRANS HYD COOLER	818.58	818.58
5	1	SS-3A1D PRINCE SELECTOR VALVE	102.00	102.00
6	1	N620S NEEDLE VALVE	39.60	39.60
7	1	9-1649-04 CYL WELDED 1 BORE X 4 STROKE X 1/2 ROD	128.25	128.25
8	1	MB-2/1S-SV/18L/E-8 YOULI VALVE 12GPM 3POS 8SAE PORTS	91.67	91.67
9	2	RDBA-LSN SUN RELIEF CARTRIDGE VALVE	69.50	139.00
10	1	AYI SUN CROSS PORT BODY #6 SAE	65.83	65.83

Sub Total	Sales Tax	Total
$ 5,544.35	$ .00	$ 5,544.35

A charge of 1.5% per month or 18% per year will be added to all past due accounts.

All returns subject to a 25% restocking charge only as authorized.

SEW-EURODRIVE

PT Pilot® Quotation

Date 03/12/2013 **Quotation # 20130312-000111**

From	SEW-Western Region (CA) SEW-Eurodrive, Inc. 30599 San Antonio Street Hayward CA, 94544	**Phone** (510) 487-3560 **Fax** (510) 487-6433	
Email	cshayward@seweurodrive.com		
To	JumpSport, Inc. Building 13, Room 254 San Luis Obispo CA, 93407	**Phone** (510) 501-2177 **Fax**	
Attn	Ethan Flory		

Purchase Order N/A **Tag/Reference**

Model	F77DRE160M4			**Base Price**	$3,427 List
Gear Ratio	12.20				
Prim/Sec Stages	2 / 0	**Shaft Location**	--	**Cable Entry Loc**	Unsure
Output Speed	145 RPM	**Output Shaft Dia**	2.0"	**Overhung Load**	3,780 lbs
Motor Power	12.5 HP	**Output Shaft Style**	Solid Shaft	**Output Torque**	5,360 lb-in
Motor Voltage	230 VAC	**Input Shaft Dia**	--	**Torque Capacity**	13,280 lb-in
Frequency	60 Hz	**Flange Diameter**	--	**Nameplate S.F.**	2.50
Brake Voltage	--	**Flange Location**	--	**Load S.F.**	2.51
Mtg. Position	M1	**Conduit Box Loc**	Unsure	**Weight** (includes oil)	308 lbs

GEAR OPTIONS (List)

2.0" Shaft [$0], 5001 Green/Blue Paint [$0], Shell Omala S2 G 220 mineral oil (standard) [$0]

MOTOR AND BRAKE OPTIONS (List)

IP54 Enclosure [$0], 5001 Green Blue Paint [$0], Standard Aluminum Conduit Box - NPT holes [$0]

ELECTRONIC OPTIONS / MECHANICAL VARIABLE SPEED (List)

MC07B0110-203-4-00 [$3111]

COMMENTS / SPECIAL INSTRUCTIONS

Load Torque: 5300 lb-in. Ambient = 25C (77F) or lower.

OFFICE USE ONLY

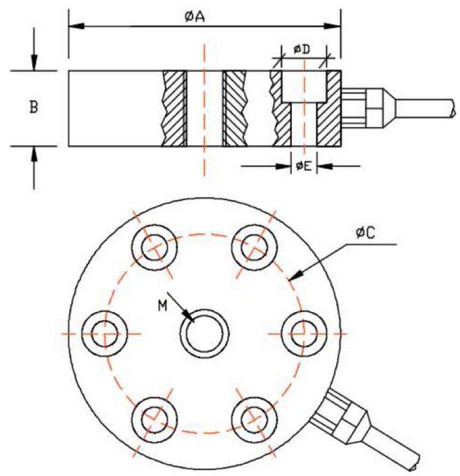
DELIVERY

Need by N/A
Ship Via N/A
FOB SEW Eurodrive, Inc.

Qty	1	List Ea	$6,538	Disc	0.722	Net Adders Ea	$0	Net Ea	$4,720.44	TOTAL	$4,720.44

Appendix D: Load Cell Pancake Data Sheet

RSB6 LOW PROFILE PANCAKE LOAD CELL



Capacity	250 - 1000 kg	2500 kg	5000 kg
A	90	90	140
B	25	25	45
C	66	66	115
D	15	15	18
E	8.5	8.5	11
M	M12	M12	M24x3

(All dimensions in mm)

Accuracy Specifications

Wiring Information

Cable Color Code	
Red	+ Excitation
Black	- Excitation
Green	+ Signal
White	- Signal

Load Cell Specifications

Zero Balance	2% of Full Scale
Safe Overload	150% of Full Scale
Ultimate Overload	300% of Full Scale
Connections	ϕ 5 x 2000 mm (6.5ft.)
Input Impedance	385 ± 15 Ω
Output Impedance	350 ± 4 Ω
Insulation	≥ 2000 M Ω / 50 V DC
Recommended Excitation Voltage	10 V DC
Compensated Temperature Range	14 to 100 °F (-10 to 40 °c)
Temperature Effect on Zero	0.02% of F.S / 10 °C
Temperature Effect on Span	0.02% of F.S / 10 °C

Ordering Information

Capacity	Part No.
250 Kg	RSB6-250M-S
500 Kg	RSB6-500M-S
1000 Kg	RSB6-01KM-S
2500 Kg	RSB6-2HKM-S
5000 Kg	RSB6-05KM-S

Calibration Options

*Co1	Compression
*Co2	Tension
*Co3	Universal

Recommended Interfaces

DI-100/DI-1000U
Digital Load Cell Interface

DI-1000Z
Wireless Load Cell Interface

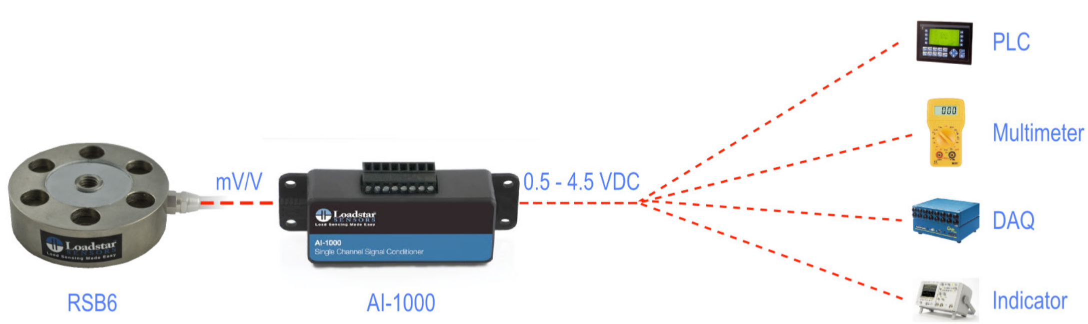
AI-1000
Signal Conditioner

RD-1000
Resistive Load Cell Display

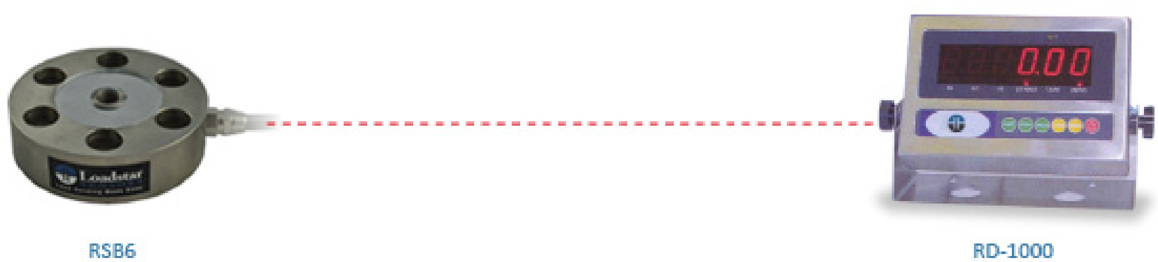
Wireless Load Cell Configuration

USB Load Cell Configuration

Analog Load Cell Configuration



Load Cell with LED Display



Appendix E: Detailed Supporting Analysis

Impactor Horizontal Beam: Endurance Limit Calculations

Linkage: Torque and Power Calculations

Load Cell/Impactor Interface: Torque Calculations

Hand Calculations: Fatigue Analysis for Foot Support Plate

Hand Calculations: Impactor Horizontal Beam Fatigue

Hand Calculations: Load Cell/ Impactor Interface

Hand Calculations: Flywheel Energy

ANALYSIS APPENDIX:
IMPACTOR HORIZONTAL BEAM: Parametric Table of Endurance

Model:
Max force split evenly between two point forces located at the extreme ends of the beam
Bolt holes accounted for in: moment of inertia by parallel axis theorem
Endurance limit: 99% reliability factor
Safety factor goal: 2

NOTE:
Based on eqns. from Shigley's
Mechanical Engineering Design

ANALYSIS CONSTANTS

Max Force (lbs)	1500
Min Tensile Str (kpsi)	63.8

FLAT BAR WITH HOLE

Description	Geometry			Moment of Inertia		Max Bending stress			Endurance Limit							SF
	L (in)	h (in)	b (in)	I_{NoHole} (in^4)	$I_{WithHoles}$ (in^4)	c (in)	Moment (lb in)	stress (kpsi)	Se prime (ksi)	ka*ke	kc*kd*kf	de	kb size factor (for 2<de<10)	kb size factor (for <2 in de)	Endurance Limit (ksi)	
2.5"x2.5"	12	2.5	2.5	3.255	2.538	1.25	4500	2.22	31.9	0.731	1	2.0	0.815	0.815	18.99	8.6
2.25"x2"	12	2.25	2.	1.898	1.375	1.125	4500	3.68	31.9	0.731	1	1.7	0.836	0.830	19.34	5.3
2"x2"	12	2.	2.	1.333	0.966	1.	4500	4.66	31.9	0.731	1	1.6	0.844	0.835	19.46	4.2
1.75"x1.75"	12	1.75	2.	0.893	0.647	0.875	4500	6.08	31.9	0.731	1	1.5	0.853	0.841	19.60	3.2
1.5"x1.5"	12	1.5	1.5	0.422	0.267	0.75	4500	12.64	31.9	0.731	1	1.2	0.883	0.861	20.07	1.6
1.375"x2"	12	1.375	2.	0.433	0.314	0.688	4500	9.86	31.9	0.731	1	1.3	0.869	0.852	19.86	2.0
1.25"x2"	12	1.25	2.	0.326	0.236	0.625	4500	11.93	31.9	0.731	1	1.3	0.876	0.856	19.96	1.7
1.125"x2"	12	1.125	2.	0.237	0.172	0.563	4500	14.72	31.9	0.731	1	1.2	0.883	0.861	20.07	1.4
1"x2"	12	1.	2.	0.167	0.121	0.5	4500	18.63	31.9	0.731	1	1.1	0.891	0.867	20.20	1.1
0.875"x2"	12	0.875	2.	0.112	0.081	0.438	4500	24.34	31.9	0.731	1	1.1	0.901	0.873	20.34	0.8
0.75"x2"	12	0.75	2.	0.070	0.051	0.375	4500	33.13	31.9	0.731	1	1.0	0.911	0.880	20.51	0.6
0.625"x2"	12	0.625	2.	0.041	0.029	0.313	4500	47.70	31.9	0.731	1	0.9	0.925	0.889	20.71	0.4
0.5"x2"	12	0.5	2.	0.021	0.015	0.25	4500	74.53	31.9	0.731	1	0.8	0.941	0.899	20.96	0.3
0.4375"x2"	12	0.4375	2.	0.014	0.010	0.219	4500	97.35	31.9	0.731	1	0.8	0.951	0.906	21.11	0.2
0.375"x2"	12	0.375	2.	0.009	0.006	0.188	4500	132.51	31.9	0.731	1	0.7	0.962	0.913	21.29	0.2
0.3125"x2"	12	0.3125	2.	0.005	0.004	0.156	4500	190.81	31.9	0.731	1	0.6	0.976	0.922	21.49	0.1
0.25"x2"	12	0.25	2.	0.003	0.002	0.125	4500	298.14	31.9	0.731	1	0.6	0.994	0.933	21.75	0.1
0.1875"x2"	12	0.1875	2.	0.001	0.001	0.094	4500	530.02	31.9	0.731	1	0.5	1.016	0.948	22.09	0.0
0.125"x2"	12	0.125	2.	0.000	0.000	0.063	4500	1192.55	31.9	0.731	1	0.4	1.049	0.969	22.57	0.0

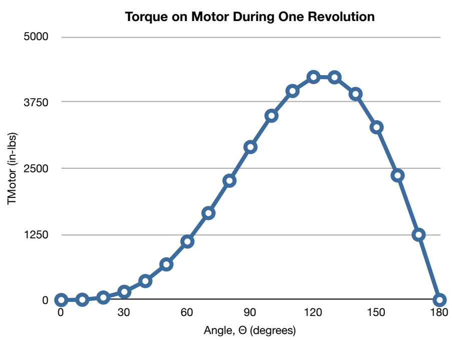
BOX TUBING WITH HOLES

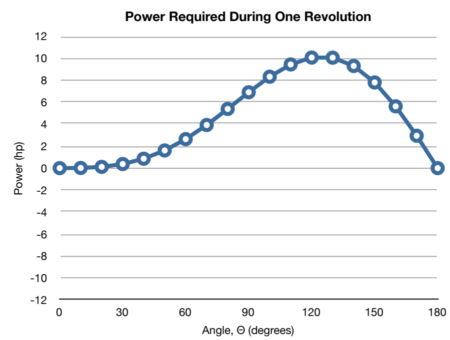
Description	Geometry						Moment of Inertia		Max Bending stress			Endurance Limit							SF
	L (in)	h (in)	b (in)	t (in)	h_{inner} (in)	b_{inner} (in)	I_{NoHole} (in^4)	$I_{WithHoles}$ (in^4)	c (in)	Moment (lb in)	stress (kpsi)	Se prime (ksi)	ka*ke	kc*kd*kf	de	kb size factor (for 2<de<10)	kb size factor (for <2 in de)	Endurance Limit (ksi)	
2"x2" x 0.25"	12	2	2	0.250	1.5	1.5	0.911	0.585	1.	4500	7.69	31.9	0.731	1	1.6	0.844	0.835	19.46	2.5
2"x2 "x 0.188"	12	2	2	0.188	1.624	1.624	0.754	0.491	1.	4500	9.16	31.9	0.731	1	1.6	0.844	0.835	19.46	2.1
2"x2 "x 0.083"	12	2	2	0.083	1.834	1.834	0.391	0.261	1.	4500	17.22	31.9	0.731	1	1.6	0.844	0.835	19.46	1.1
2"x1" x 0.188"	12	2	1	0.188	1.624	0.624	0.444	0.182	1.	4500	24.76	31.9	0.731	1	1.1	0.891	0.867	20.20	0.8

Hole: d_{top} (in)	0.551
Hole: d_{bottom} (in)	1.142

1		h (in)	c (in)	R (in)															
Cycles per Minute	150	10	6	16															
Time, t (sec)	0.0	0.011	0.022	0.033	0.044	0.056	0.067	0.078	0.089	0.10	0.11	0.12	0.13	0.14	0.16	0.17	0.18	0.19	0.20
θ (deg)	0	10	20	30	40	50	60	70	80	90	100	110	120	130	140	150	160	170	180
T_{Motor} (in-lbs)	0	6	47	156	359	675	1108	1647	2261	2899	3493	3961	4226	4220	3905	3274	2361	1237	0
Power (hp)	0	0.01	0.11	0.37	0.86	1.61	2.64	3.92	5.38	6.90	8.31	9.43	10.06	10.04	9.29	7.79	5.62	2.94	0
$T_{Motor\text{-}AVG}$ (in-lbs)		2.97951847	26.54798	101.580645	257.726917	517.284022	891.67294	1377.71493	1954.28566	2580.39347	3196.02973	3726.93961	4093.57836	4223.17262	4062.59193	3589.2266	2817.44407	1799.20194	618.615776
Area (in-lbs-s)		0.03310576	0.29497756	1.12867383	2.86363241	5.74760024	9.90747711	15.3079437	21.7142851	28.6710386	35.5114414	41.4104402	45.484204	46.9241402	45.1399104	39.8802955	31.3049341	19.9911326	6.87350863
$Area_{Max}$ (in-lbs-s)	398.18874136																		
$T_{Motor\text{ - }Max}$ (in-lbs)	4226																		
$T_{Motor\text{ - }Avg}$ (in-lbs)	1886																		
$P_{Motor\text{ - }Max}$ (hp)	10.1																		
$P_{Motor\text{ - }Avg}$ (hp)	4.5																		

This calculation assumes the trampoline is a linear spring with spring constant 100 lb/in, which corresponds to a 12 in deflection at 1200 lbs.





Flywheel Size

Flywheel Inertia Required (in-lbs-s²)	Mass (lbs)	Flywheel Diameter (ft)
190	20	0.73
	40	0.51
	80	0.36
	120	0.30
	200	0.23
	500	0.15
180	20	0.71
	40	0.50
	80	0.35
	120	0.29
	200	0.22
	500	0.14

Flywheel Design

Min Flywheel Speed (rpm)	Max Flywheel Speed (rpm)	Avg. Flywheel Speed (rpm)	Coefficient of Fluctuation, k	Flywheel Inertia Required (in-lbs-s²)
130	170	150	0.267	95.061
135	165	150	0.200	126.747
140	160	150	0.133	190.121
145	155	150	0.067	380.242
147	153	150	0.040	633.737

This table calculates the necessary inertia of a flywheel that would decrease from the max speed to the min speed if loaded by the torque in the first half of the impactor cycle.

2		h (in)	c (in)	R (in)																			
		12	6	18																			
θ (deg)	5	10	20	30	40	50	60	70	80	90	100	110	120	130	140	150	160	170	172	174	176	178	180
T_{Motor} (in-lbs)	0.797	6.353	50.171	165.626	380.159	710.840	1160.658	1715.164	2340.273	2982.338	3571.546	4029.199	4278.350	4256.148	3925.467	3283.318	2364.206	1237.625	995.552	749.858	501.431	251.174	0.000
$T_{Motor\text{ - }Max}$ (in-lbs)	4278																						
$T_{Motor\text{ - }Avg}$ (in-lbs)	1694				Power (hp)	4.03																	

3		h (in)	c (in)	R (in)																			
		14	6	20																			
θ (deg)	5	10	20	30	40	50	60	70	80	90	100	110	120	130	140	150	160	170	172	174	176	178	180
T_{Motor} (in-lbs)	0.836	6.668	52.593	173.270	396.607	739.065	1201.978	1768.526	2402.137	3047.270	3633.409	4082.561	4319.670	4284.373	3941.915	3290.962	2366.628	1237.940	995.714	749.927	501.452	251.177	0.000
$T_{Motor\text{ - }Max}$ (in-lbs)	4320																						
$T_{Motor\text{ - }Avg}$ (in-lbs)	1715				Power (hp)	4.08																	

APPENDIX: IMPACTOR: Load Cell Interface Torque Analysis

OBJECTIVE:
Determine necessary torque to unscrew load cell from impactor based on a desired preload.

NOTE:
Equations and statistical data from Shigley's

Statistical data based on 10 tests for M12x1.25 bolts torqued to 90 Nm

CALCULATING PRELOAD

Proof Load with SF of 1.5 ($F_p = A_t*S_p/ sf$)	3.39E+04 N
Preload (F_i)	25398 N

Safety Factor	1.5

BOLT PROPERTIES and DIMENSIONS

Tensile Strength (S_p)	8.30E+08 Pa
Tensile Stress Area (A_t)	6.12E-05 m^2
Bolt Diameter (d)	0.012 m
Bolt Condition (K) (Zinc-plated)	0.2

TORQUE TO UNSCREW

Contact Area	0.006185 m^2
Pressure (P=Fi/A)	4106379 Pa
Unscrewing Torque	117 Nm
Unscrewing Torque	86 ft lbs

CONTACT PRESSURE DIMENSIONS

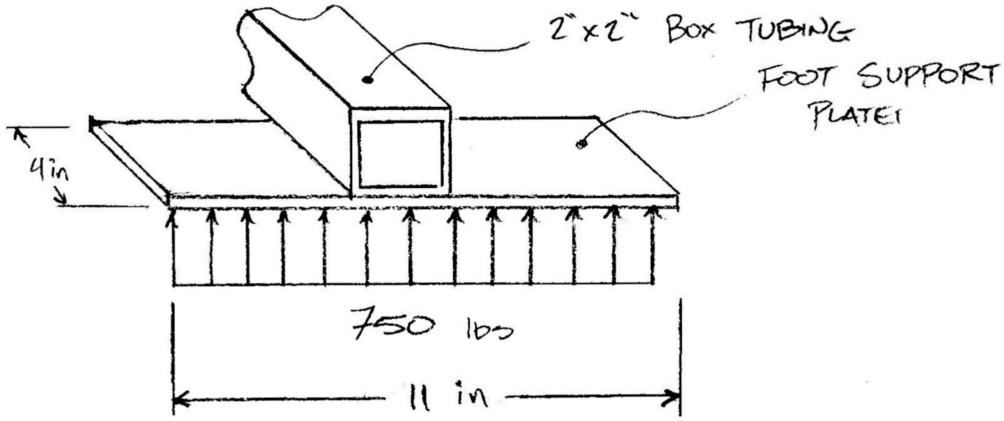
Outer Contact radius	0.045 m
Inner Contact radius	0.0075 m

TORQUE NECESSARY FOR DESIRED PRELOAD

Friction Coefficient (Ms)	0.15
Torque ($T=K*F_i*d$)	61 Nm
Torque	45 ft lbs

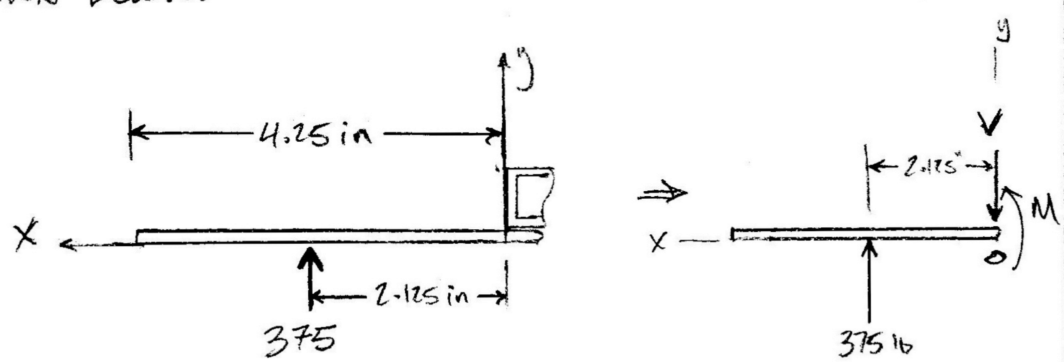
FATIGUE ANALYSIS FOR

FOOT SUPPORT PLATE



TO ANALYZE THE ENDURANCE LIMIT OF THE FOOT SUPPORT PLATE, BENDING FATIGUE ANALYSIS WAS CONDUCTED WITH A FEW SIMPLIFYING ASSUMPTIONS:

- WHILE THE OVERALL PLATE DIMENSIONS ARE 4" WIDE BY 11" LONG, THE IMPACTOR FOOT SURFACE IN CONTACT WITH THE TRAMPOLINE MAT IS ONLY 10.5" LONG. THERE FOR, THE LENGTH OF THE CANTILEVERED BEAM LOADED IN BENDING IS BASED OFF OF THE 10.5" OVERALL LENGTH INSTEAD OF 11".
- THE PEAK IMPACT FORCE EXPECTED IS 1500 lbs, THEREFORE THE LOAD ON EACH IMPACTOR FOOT ASSUMED TO BE 750 lbs AND UNIFORMLY DISTRIBUTED.
- THE SUPPORT PLATE IS MODELED AS TWO SEPARATE CANTILEVERED BEAMS WITH EACH SEEING A 375 lb DISTRIBUTED LOAD.
- THE DISTRIBUTED LOAD ON EACH HALF PLATE IS LUMPED INTO A POINT LOAD IN THE CENTER OF THE SPAN. THIS PLATE IS EVALUATED FOR FATIGUE DUE TO THE BENDING LOAD. THE FBD IS SHOWN BELOW.



$$\Sigma M_o = 0$$
$$M_o = 375\,\text{lb}\,(2.125\text{ in})$$

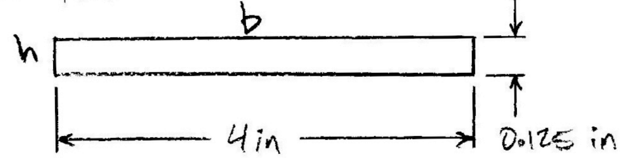
$$\boxed{M_o = 797\text{ in-lb}}$$

THEREFORE,

$$0 \leq M_o \leq 797 \text{ in-lb} \qquad \text{SINCE } 0 \leq F \leq 1500 \text{ lb}$$

CROSS SECTION OF PLATE:

FROM SHIGLEY'S PG. 281



$$I = \frac{bh^3}{12}$$

$$C = \frac{h}{2}$$

$$\sigma_a = \frac{Mc}{I} = \frac{M\left(\frac{h}{2}\right)}{\frac{bh^3}{12}} = \frac{6M}{bh^2}$$

$$\sigma_a = \frac{(797 \text{ in-lb})\,6}{(4 \text{ in})(0.125 \text{ in})^2} \rightarrow \boxed{\sigma_a = 76.5 \text{ ksi}}$$

WITH THESE DIMENSIONS, THE STRESS IS GREATER THAN THE ULTIMATE STRENGTH OF MOST MILD CARBON STEELS. MORE SUPPORT IS NEEDED TO RESIST THE BENDING STRESS.

FATIGUE ANALYSIS ALSO CONDUCTED TO SEE THE ENDURANCE LIMIT OF THIS CONDITION

$$S_e = k_a k_b k_c k_d k_e k_f S_e'$$

$$S_e' = \tfrac{1}{2}(S_{UT})$$

$$S_e' = \tfrac{1}{2}(70 \text{ ksi}) \qquad \text{FOR 1018 CD STEEL}$$

$$S_e' = 35 \text{ ksi}$$

$\underline{k_a} = a\,S_{UT}^{\,b}$
$= (2.70)(70 \text{ ksi})^{-.265}$

$k_a = .876$

$\underline{K_b}$ $.808\sqrt{hb} = d_e$
$= .808\sqrt{(.125 \text{ in})(4 \text{ in})}$

$d_e = .571$ in

$k_b = .879(.571)^{-.107} = .93$

$\underline{k_c} = 1$ (BENDING)

$\underline{k_d} = 1$ (ASSUMING 70°F)

$\underline{k_e} = .814$ (99% RELIABILITY)

$\underline{k_f} = 1$ (MISC FACTOR)

$$\therefore S_e = (.93)(.876)(1)(1)(.814)(1)(35 \text{ ksi})$$

$$S_e = 23.3 \text{ ksi}$$

$$SF = \frac{S_e}{\sigma} = \frac{23.3 \text{ ksi}}{76.5 \text{ ksi}} \longrightarrow \boxed{SF = .30}$$

IMPACTOR HORIZONTAL BEAM - FATIGUE CALCULATIONS

CALCULATE ENDURANCE LIMIT (S_e)

(SAMPLE CALCS FOR 2"x2" x 0.188" BOX TUBING WITH 0.551" HOLE ON TOP OF BOX TUBING, 1.142" HOLE ON BOTTOM OF BOX TUBING

- APPLY MODIFYING FACTORS TO MINIMUM TENSILE STRENGTH:

$$S_e = (k_a)(k_b)(k_c)(k_d)(k_e)(k_f) S_e$$

S_e': ENDURANCE LIMIT

$S_{ut} \leq 200$ kpsi

∴ ESTIMATE ENDURANCE LIMIT AS: $S_e' = 0.5 S_{ut}$

S_{ut} OF 1018 STEEL = 63.8 kpsi

$$S_e' = 0.5(63.8\text{ kpsi}) = 31.9\text{ kpsi}$$

MODIFYING FACTORS

k_a: SURFACE FACTOR

FOR COLD DRAWN, $a = 2.70$, $b = -.265$

$$k_a = a S_{ut}^{b} = 2.70\,(63.8\text{kpsi})^{-.265} = 0.898$$

k_b: SIZE FACTOR

CALCULATE AS SOLID RECTANGLE - SAME VALUE REGARDLESS OF WHETHER HOLLOW OR NOT.

EFFECTIVE DIAMETER:

$$d_e = 0.808\sqrt{hb} = 0.808\sqrt{(2)(2)} = 1.616\text{ in}$$

FOR $d_e < 2"$,

$$k_b = 0.91\,(d_e)^{-.157} = 0.91(1.616\text{in})^{-.157} = 0.835\text{ in}$$

k_c: LOADING FACTOR

FOR BENDING,

$$k_c = 1$$

k_d: TEMPERATURE FACTOR

FOR Temp = 70°C,

$$k_d = 1$$

k_e: RELIABILITY FACTOR

FOR 99% RELIABILITY,

$$k_e = 0.814$$

$\underline{k_f}$: MISCELLANEOUS FACTOR
NO MISCELLANEOUS FACTORS

$k_f = 1$

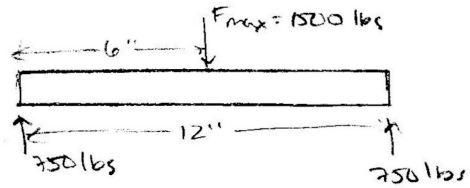
PLUG IN MODIFYING FACTORS:

$$S_e = (0.898)(0.835)(1)(1)(0.814)(1)(31.9 \text{ kpsi})$$

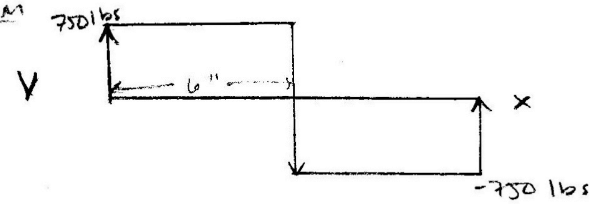
$$\boxed{S_e = 19.47 \text{ kpsi}}$$

<u>COMPARE ENDURANCE LIMIT TO MAX BENDING STRESS.</u>
• CALCULATE MAX BENDING STRESS.

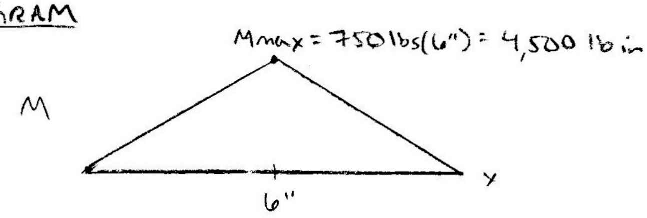
<u>FBD</u> ON BEAM. MODELED AS MAX FORCE DIVIDED UP EVENLY BETWEEN 2 POINTS FORCES ON EXTREME OPPOSITE ENDS.



<u>SHEAR DIAGRAM</u>



<u>MOMENT DIAGRAM</u>



$M_{max} = 4{,}500$ lb-in

BENDING STRESS

$$\sigma_{bending} = \frac{Mc}{I}$$

$M_{max} = 4{,}500$ lb-in
$C = h/2 = 2''/2 = 1''$

MOMENT OF INERTIA FOR BOX TUBING:

$$I = \frac{1}{12} bh^3 - \frac{1}{12} b_{inner} h_{inner}^3$$
$$= \frac{1}{12}(2'')(2'')^3 - \frac{1}{12}(2-(2(.188))(2-2(.188))^3$$
$$= 0.754 \text{ in}^4$$

SUBTRACT OFF INERTIA FROM HOLES USING PARALLEL AXIS THEOREM:

$$I_{holes} = I_o + Ay^2$$
$$= \frac{1}{12}(d_{bottom}+d_{top})\,t^3 + (d_{bottom}+d_{top})(t)\left(\frac{h}{2}-\frac{t}{2}\right)^2$$
$$= \frac{1}{12}(1.142''+0.551'')(.188^3) + (1.142''+0.551'')(.188)\left(\frac{2}{2}-\frac{.188}{2}\right)^2$$
$$= 0.2622 \text{ in}^4$$

$$I_{total} = I - I_{holes} = (0.754 - 0.2622)\text{ in}^4 = 0.492 \text{ in}^4$$

PLUG IN VALUES:

$$\sigma_{bending} = (4{,}500 \text{ lb-in})(1 \text{ in}) / (0.492 \text{ in}^4) = 9.15 \text{ kpsi}$$

$$\boxed{\sigma_{bending} = 9.15 \text{ kpsi}}$$

FACTOR OF SAFETY

$$F.S. = \frac{S_e}{\sigma_{bending}} = \frac{19.47 \text{ kpsi}}{9.15 \text{ kpsi}} = 2.1$$

$$\boxed{F.S. = 2.1}$$

LOAD CELL / IMACTOR INTERFACE ANALYSIS

CALCULATE PRELOAD:

PROOF LOAD. F_p

GIVEN: TENSILE STRESS AREA $(A_t) = 6.12 \times 10^{-5}\,m^2$
AND TENSILE STRENGTH $(S_p) = 8.3 \times 10^8\,Pa$
AND SAFETY FACTOR OF 1.5

$$F_p = (A_t)(S_p)/(sf) = (6.12 \times 10^{-5}\,m^2)(8.3 \times 10^8\,Pa)/1.5 = 3.39 \times 10^4\,N$$

PRELOAD: F_i

$$F_i = (0.75)(F_p) = (0.75)(3.39 \times 10^4\,N) = 25{,}398\,N$$

CALCULATE TORQUE REQUIRED TO UNSCREW LOAD CELL.

DERIVE TORQUE FROM PRELOAD:

$$F_i = PA = Pr\,dr\,d\theta = 2\pi Pr\,dr$$

$$T = F_{of}\,r = F_i \mu_s r = 2\pi P r^2 dr\, \mu_s$$

$$T = \int_{r_i}^{r_o} 2\pi \mu_s P r^2 dr = \frac{2\pi}{3}\mu_s P r^3 \Big]_{r_i}^{r_o} = \frac{2\pi}{3}\mu_s P (r_o^3 - r_i^3)$$

GIVEN: $\mu_s = 0.15$; $r_o = \frac{90}{(2)(1000)}\,m = 0.045\,m$; $r_i = \frac{15}{2(1000)}\,m = 0.0075\,m$

CONTACT AREA

$$A = \pi r_o^2 - \pi r_i^2 = \pi(0.045^2 - 0.0075^2)\,m^2 = 0.006185\,m^2$$

PRESSURE

$$P = F_i/A = 25{,}398\,N / 0.006185\,m^2 = 4.106\,MPa$$

TORQUE:

$$T = \frac{2\pi}{3}\mu_s P (r_o^3 - r_i^3)$$

$$= \frac{2\pi}{3}(0.15)(4.106 \times 10^6\,\tfrac{N}{m^2})(0.045^3 - 0.0075^3)\,m^3 = 117\,Nm$$

$$\boxed{T = 86\ ft\text{-}lbs}$$

ADEQUATE TORQUE?

$$F_{applied\,min} = \frac{T}{d_{max\,foot}} = \frac{86\,ft\text{-}lbs}{0.5\,ft} = 172\,lbs$$

PORTION OF MAX LOADING NEEDED AS A HORIZONTAL FORCE COMPONENT

· HORIZONTAL FORCE % = $F_{applied\,min} / F_{tramp\,max}$ = 172 lbs / 750 lbs

$\boxed{\text{HORIZONTAL FORCE \%} = 23\%}$

ALL FORCE FROM TRAMPOLINE SHOULD THEORETICALLY BE VERTICAL ∴ IT IS UNREASONABLE TO RESOLVE THAT 23% OF THAT MAX FORCE COULD BE TRANSFERED TO HORIZONTAL COMPONENT FORCE

Flywheel Design :

$I = \frac{W}{g}\left(\frac{k^2}{2}\right)$

I = inertia (in-lb-s²)

W = flywheel weight

$g = 32.2\,ft/s^2 = 386.4\,in/s^2$

k = radius gyration

k = 7.5 in (half diam.)

steel [.2836 lb/in²] ← Shigley's Design 7th ed.

$A = \pi r^2$

$A = \pi(7.5)^2$

$A = 176.7\,in$

$W = (.2836\,lb/in^2)A$

$W = (.2836)(176.7)$

$W = 50.12\,lbs$

$I = \frac{50.12}{386.4}\left(\frac{7.5}{2}\right)^2$

$I = 3.65\,in\text{-}lbf\text{-}s^2$

$\omega = 2.5\,Hz = 150\,rpm$

$I = \frac{\Delta E}{C_s \omega^2}$ ← Shigleys Design 7th ed.

Cs = coefficient speed fluctuation (allowable speed drop)

Cs = 10% or .1 ← industry standard

Flywheel Design (cont'd):

ΔE = energy made up by flywheel

$\Delta E = I \, C_s \, \omega^2$

$\Delta E = (3.65)(0.1)(15.7)^2$

$\Delta E = 90$ in-lbf

Appendix F: JumpSport Drop Test Data

# of cords on unit	Knot length [mm]	97 lbs					142 lbs					187 lbs				
		Peak Acceleration [g]	Peak Jerk [ft/s^3]	Static Spring Constant [lbf/in]	Dynamic Spring Constant [lbf/in]	Damping Ratio	Peak Acceleration [g]	Peak Jerk [ft/s^3]	Static Spring Constant [lbf/in]	Dynamic Spring Constant [lbf/in]	Damping Ratio	Peak Acceleration [g]	Peak Jerk [ft/s^3]	Static Spring Constant [lbf/in]	Dynamic Spring Constant [lbf/in]	Damping Ratio
30	533	6.01	78.8	37.0	25.5	0.008	5.52	64.9	40.6	36.4	0.010	5.30	54.7	44.7	44.5	0.011
30	508	5.89	74.6	42.0	27.1	0.010	5.67	68.5	46.4	35.0	0.009	5.62	62.3	49.9	44.4	0.011
30	483	6.02	80.0	42.0	25.7	0.008	5.69	68.0	42.9	36.5	0.010	5.55	61.5	48.3	45.5	0.011
30	545	6.16	85.3	39.8	29.7	0.010	5.74	70.6	45.4	39.9	0.011	5.56	61.9	49.9	48.0	0.011
30	520	6.16	84.1	45.7	27.1	0.008	5.78	72.6	51.6	37.4	0.009	5.56	62.5	55.4	46.2	0.010
30	495	6.20	85.1	48.5	26.7	0.008	5.86	73.9	59.8	36.6	0.008	5.64	64.5	57.5	47.8	0.008
36	545	6.33	89.7	59.1	31.3	0.010	5.91	101.7	63.1	41.7	0.010	5.69	68.0	66.1	51.9	0.011
36	520	6.40	96.3	64.0	29.6	0.009	5.98	78.8	68.9	39.9	0.009	5.72	68.7	89.4	49.8	0.009
36	495	0.00	0.0	0.0	0.0	0.000	6.04	82.3	75.7	39.4	0.008	5.80	102.9	78.0	49.3	0.009

# of cords on unit	Knot length [mm]	222 lbs					257 lbs				
		Peak Acceleration [g]	Peak Jerk [ft/s^3]	Static Spring Constant [lbf/in]	Dynamic Spring Constant [lbf/in]	Damping Ratio	Peak Acceleration [g]	Peak Jerk [ft/s^3]	Static Spring Constant [lbf/in]	Dynamic Spring Constant [lbf/in]	Damping Ratio
30	533	4.12	80.2	45.5	52.8	0.006					
30	508	5.72	63.8	50.7	51.2	0.013	4.5	110.2	52.7	59.5	0.008
30	483	5.66	165.1	49.3	52.1	0.013					
30	545	5.46	57.6	51.5	54.2	0.012	5.2	88.1	53.4	62.1	0.012
30	520	5.45	57.0	55.5	52.2	0.011	5.7	78.0	57.9	59.0	0.013
30	495	5.52	59.9	56.4	51.7	0.010	5.5	59.8	59.6	59.2	0.011
36	545	5.56	65.5	68.2	58.5	0.011	5.49	60.6	69.6	64.5	0.011
36	520	5.61	65.4	75.3	56.7	0.010	5.50	74.3	77.3	64.5	0.011
36	495	5.70	72.6	82.2	56.1	0.009	5.64	66.7	81.9	62.9	0.010