ME 430 Final Report

JumpSport Project 23: Trampoline and Bungee Fatigue-test Machine

December 8, 2013

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December 8, 2013

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Kevin Charles
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Dear Mr. Charles:

Included is the Final Report for the Bungee and Trampoline Fatigue-test Machine. Attached is our empirical analysis of your data, analytical models, annotated decision matrix, preliminary lead concept sketches, final design description, annotated parts drawings, and test results.

Sincerely,

Andrew Brock
Christopher D'Elia
Ryan Murphy

Distribution:
Professor Mohammad Noori: 1 copy

Abstract
JumpSport, a Saratoga based recreational and fitness trampoline company, sponsored this senior project team towards the development of a fatigue-testing machine. The goal of this project was to iterate the previous machine JumpSport used for bungee cord, spring, and trampoline fatigue testing. Since minimal study had been conducted on these specific bungee cords, the fixture was designed to confirm the theoretical correlations for rubber and perform the final tests according to a strict protocol. Operating with a specific test protocol in mind will allow the data from multiple tests to be combined and analyzed statistically. With a design concept in hand, the tasks were broken down into mechanism, power system, frame/fixture, and control. This project contains three main stages of development: design, fabrication, and verification. This report marks the end of the verification phase.

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Introduction

JumpSport, a Saratoga based recreational and fitness trampoline company, sponsored this senior project team towards the development of a fatigue-testing machine. The machine tests spring and bungee cords used on various trampoline models, as well as fully assembled trampolines. The company's previous testing apparatus required upwards of three weeks to run a test, exceeded the city's municipal noise rating, and had a limited test capacity. JumpSport requested that a new machine be created to overcome the previous machine's limitations.

Background

The goal of this project was to iterate the previous machine JumpSport used for bungee cord and spring-fatigue testing. The bungee cords are made from natural latex rubber coated in a talc powder for minimal friction and enclosed in a dual cloth outer sheath intended for protection and pretension. These elastic members are installed on trampoline frames and connected to a center mat providing the rebounding surface. The first machine was built by Jon Hylbert using two worm drive AC motors and a series of load cells. The load cell data feed was used to determine the number of strokes and min/max force reading for each stroke. A secondary part of the data reduction reported points at a specified frequency such that the data could be viewed in Excel.

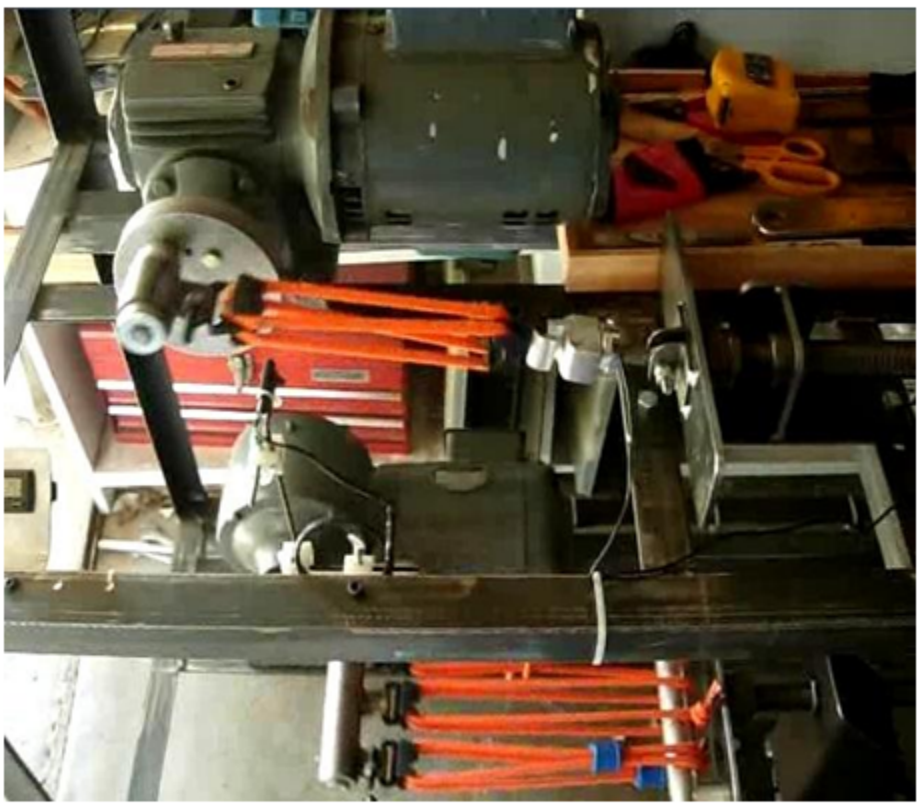

Figure 1: Current Test Machine

Various cords, fixturing methods and knot configurations were tested on this machine. These tests have been summarized in an Excel spreadsheet. However, none of the tests were conducted identically, so minimal conclusions have been drawn from the data as whole. Also, this summary indicates that consistent failure criteria has not been developed or applied. A brief analysis of the completed test samples reveals that some have failed aesthetically, some by fatigue or fracture, and some potentially by creep. Many of these tests have taken close to 3 weeks to complete, significantly limiting the designers' ability to quickly develop and test new solutions.

Iterating the test machine involved reducing noise and testing time, and increasing capacity, stroke, frequency, and the range of potential testing parameters. To decrease the test time, the parameters of an accelerated fatigue test must be better understood. Research of the fatigue characteristics of natural latex rubber revealed strong correlations with strain, strain rate, stress, composition, and cycles. [1] Other texts implied that a relationship may be established linearly between the maximum strain and the number of cycles. [2] Since minimal study was conducted on these specific bungee cords, the fixture needed to be designed to confirm these theoretical correlations for rubber and perform the final tests according to a strict protocol. The basis for this protocol can be derived from these articles as well as some ASTM standards including D4482-11 Standard Test Method for Rubber Property—Extension Cycling Fatigue. [3]

Objectives

The central objective of this project was to iterate the current JumpSport fatigue-test machine. The new test machine must be capable of standard life tests as well as accelerated life tests of all their spring and elastic cord products, as well as their trampolines. Standard life tests are run at the nominal operating conditions whereas the accelerated life tests will be run at extraneous conditions. To achieve these various conditions, the machine must be capable of variable frequencies, variable strokes, and comprehensive force and position measurement. The machine must also operate quietly and be able to accommodate the full range of JumpSport's trampoline products. Preliminary designs were evaluated analytically and economically prior to further development and fabrication.

A data acquisition system will be implemented to record the force and position in real time. An algorithm will be developed to reduce the noise and record important values to a secondary file. Along with the machine and data acquisition system, a test protocol will be developed for the standard life and accelerated life tests. The test protocol will include system setup parameters and failure criteria for each test.

A summary and preliminary weighting of these requirements has been provided in the quality function deployment matrix. Appendix A and B contain the matrix and detailed descriptions of each requirement. Since JumpSport provided the team with a series of customer requirements that are heavily influenced by their engineering team, the engineering requirement section was used to develop these more specifically. In Appendix C, an annotated decision matrix shows each of the three top concepts and how they meet the requirements.

Design Development
Preliminary testing and analysis was conducted to determine how the cords are loaded during each cycle. Analysis began with a test conducted by JumpSport. The data came from measuring the output of a series of accelerometers attached to a person bouncing on a trampoline. Using this acceleration data, the displacement as a function of time was estimated for the cords. The fixture was designed to replicate this motion. The results of this simulation are summarized below.

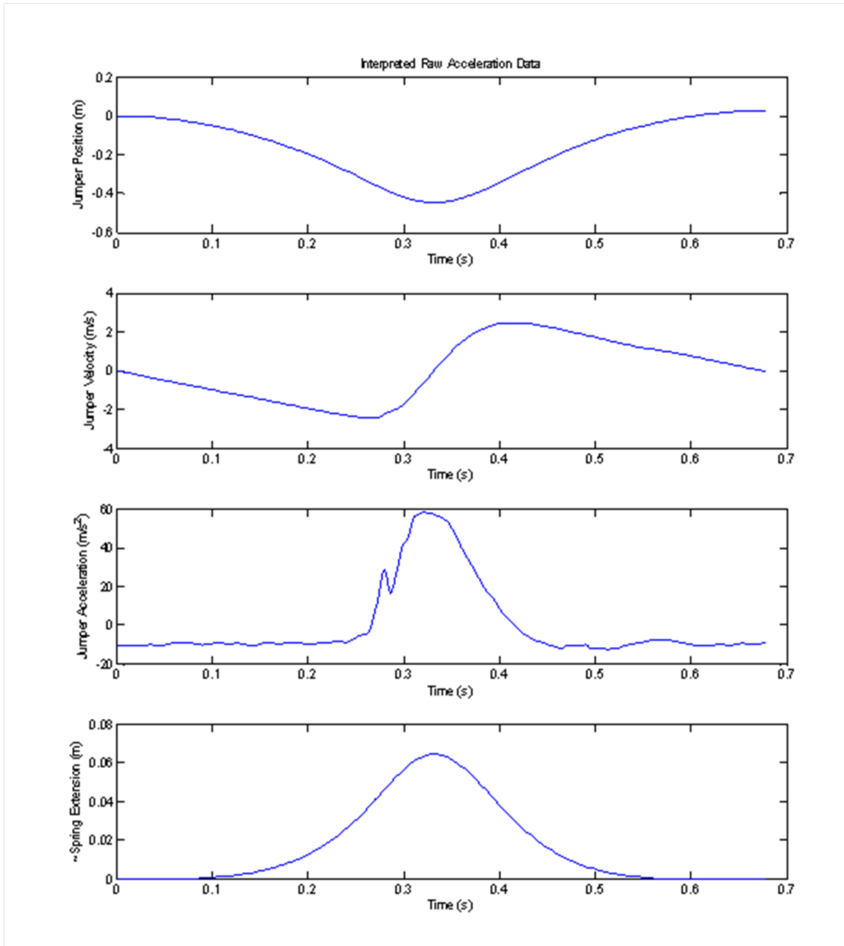

Figure 2: Empirical Trampoline Acceleration Data Analysis

To confirm the validity of these tests and acquire pure analytical data, a simulation of the trampoline was created. The model represents the trampoline as a mass spring damper system and the jumper as a weight impacting the trampoline. The model is given a jumping height for the jumper, trampoline stiffness, and trampoline damping; at the output, the model provides the displacement, velocity, and acceleration as a function of time. A trigonometric model of the trampoline mat yields the corrected displacement as a function of the time.

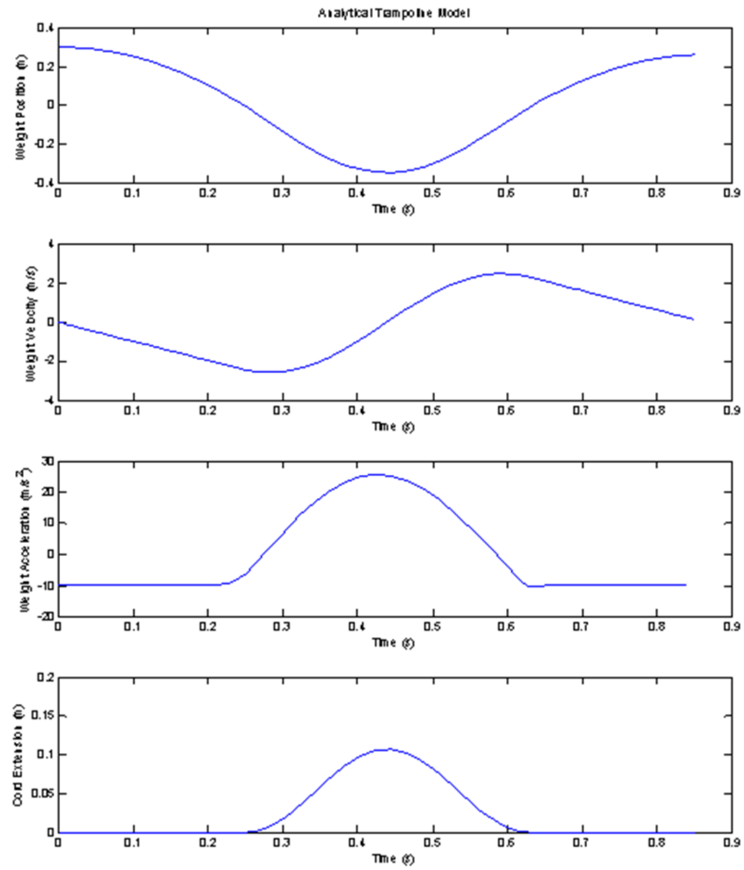

Figure 3: Analytical Trampoline Model

Using PID control and appropriate actuators, mechanisms similar to the ball screw and servo-hydraulic can be driven to this corrected displacement function. Individual simulations were conducted for each of these mechanisms, assuming no error in the control, to determine the necessary power input. However, other mechanisms (including the slider crank) have a characteristic displacement function. A slider crank test fixture simulation was created to compare the displacement function to the analytical model as well as estimate the necessary power input.

Distinct failure conditions must be defined and implemented such that tests can be conducted and terminated algorithmically. Operating with a specific test protocol in mind will allow the data from multiple tests to be combined and analyzed statistically. Statistical analysis will ensure that the data accurately describes the range of field performance. This distribution can then be compared with JumpSport's quality standards.

Three separate design solutions were analyzed and compared against the customer and engineering requirements outlined in the QFD. The first concept is a slider crank mechanism with an adjustable linkage and tailstock connected directly to the sample. The second concept is a ball-screw actuator attached directly to the sample which is secured to an adjustable tailstock. The third concept uses servo-hydraulics to indirectly actuate the sample via a series of linkages.

Slider Crank

The slider crank mechanism consists of a direct drive gear motor, adjustable crank arm, connecting rod, and linear bearing. The direct drive gear motor is operated by a variable speed controller with position feedback. The adjustable crank arm allows the controller to continually increase the stroke during a fatigue life test to maintain a consistent peak force. Adjusting the stroke to maintain a consistent peak force will replicate the average user as they continually wear out the cords out and thus displace the trampoline further. The connecting rod will attach the crank arm output to the sliding carriage. To limit frictional losses, the connecting rod will be mounted on bearings at either end. The connecting rod should be two or three times as long as the crank arm to most accurately replicate the motion of the trampoline without making the mechanism too large. The linear bearing or carriage is attached to a static supported shaft. As the mechanism reciprocates, the linear bearing ensures that the two ends of the specimen remain aligned. The tailstock, supporting the opposite end of the specimen as the slider crank, also employs a linear actuator to maintain constant preload and allow for automated setup and sample calibration.

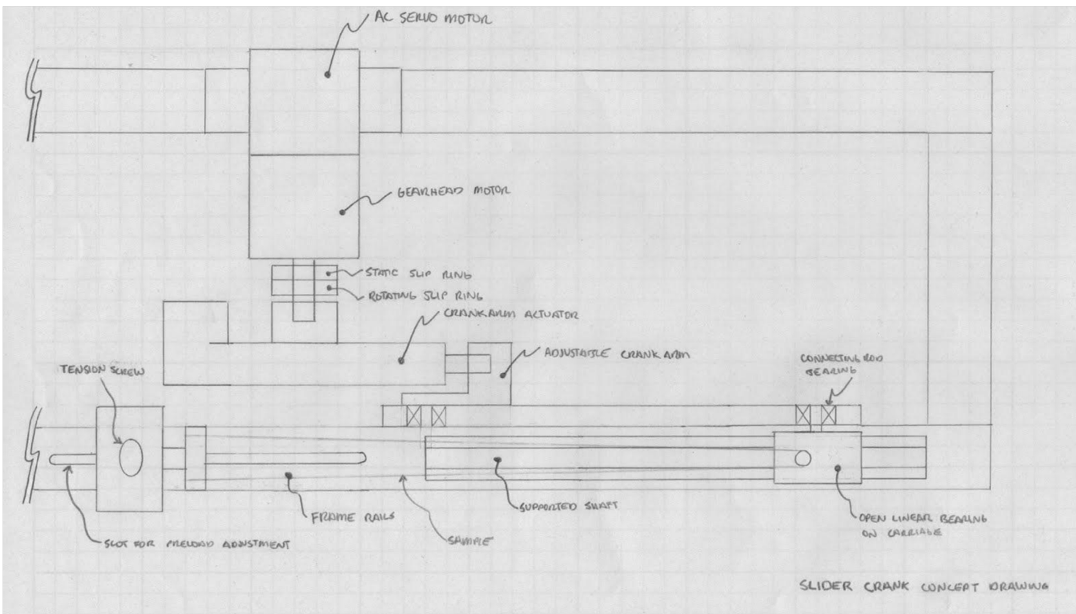

Figure 4: Slider Crank Concept Sketch

The slider crank linkages, surrounding framework, and fixtures can be designed for infinite life. Other components, including the motor and bearings, can be selected to withstand the required 100 test cycles prior to servicing. With a continuously rotating motor, the slider crank control algorithm can be simplified to maintain the motor speed. A simple controller limits the cost and noise involved in a continually reversing system. The motor controller will be capable of variable speeds (effectively delivering variable output frequency) and position feedback for data acquisition. In order to conduct tests at constant max strain as well as constant max force, the slider crank is designed with a linear actuator on the crank arm and on the tailstock. These actuators automate the mechanism's variable stroke and adjustable capacity. The surrounding frame and fixturing can interface with the existing load cells.

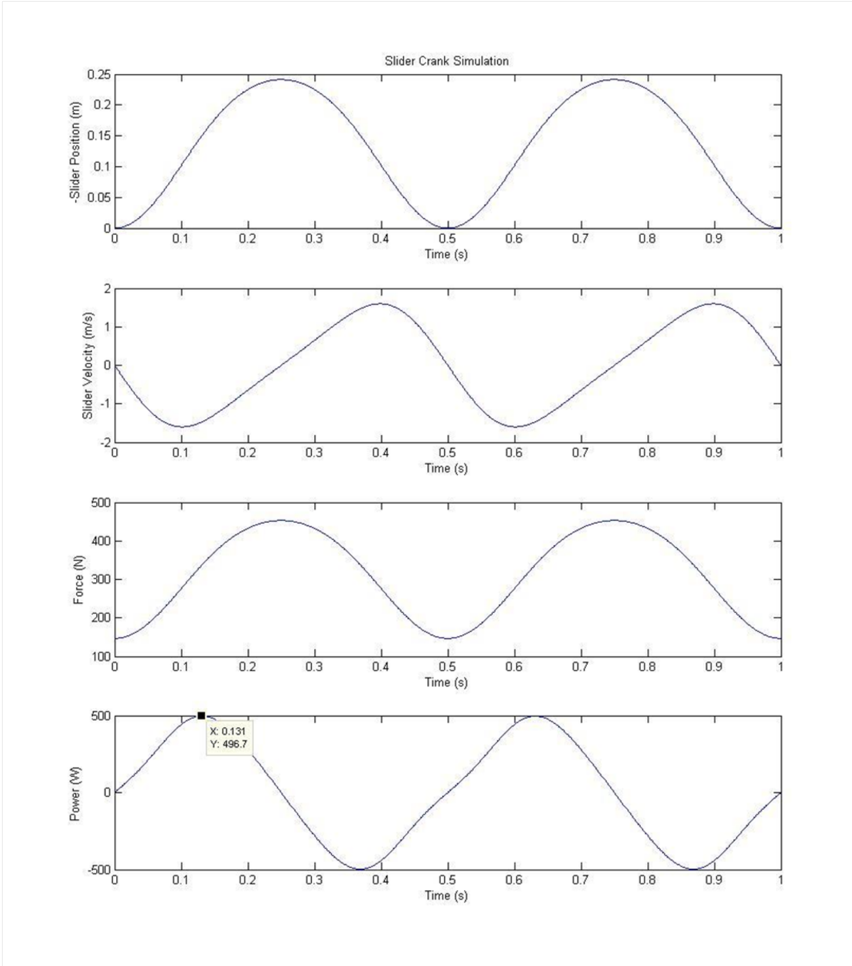

Figure 5: Slider Crank Model

Ball Screw

The ball screw concept design consists of a servo motor connected to a threaded shaft either directly or indirectly via a gear train. The threaded shaft provides a helical raceway for ball bearings which act as a precision screw. Ball bearings are contained within a ball assembly that can be fastened to a carriage to provide linear motion. The ball screw and motor combination will be fastened to a frame designed to sustain the fatigue loading during the test period. A vertically oriented machine will produce the smallest footprint. Fixturing will be designed for both the actuating and static ends of the machine. The static end will consist of steel tubing of the same diameter as the tubing on the trampoline, which will be connected to a load cell. The carriage fixturing will seek to emulate the trampoline mat so that test conditions accurately replicate actual operation. The mat material will be connected to the same plastic connectors used to affix the bungees. The machine will be controlled by a PID controller capable of controlling position or force based on desired input.

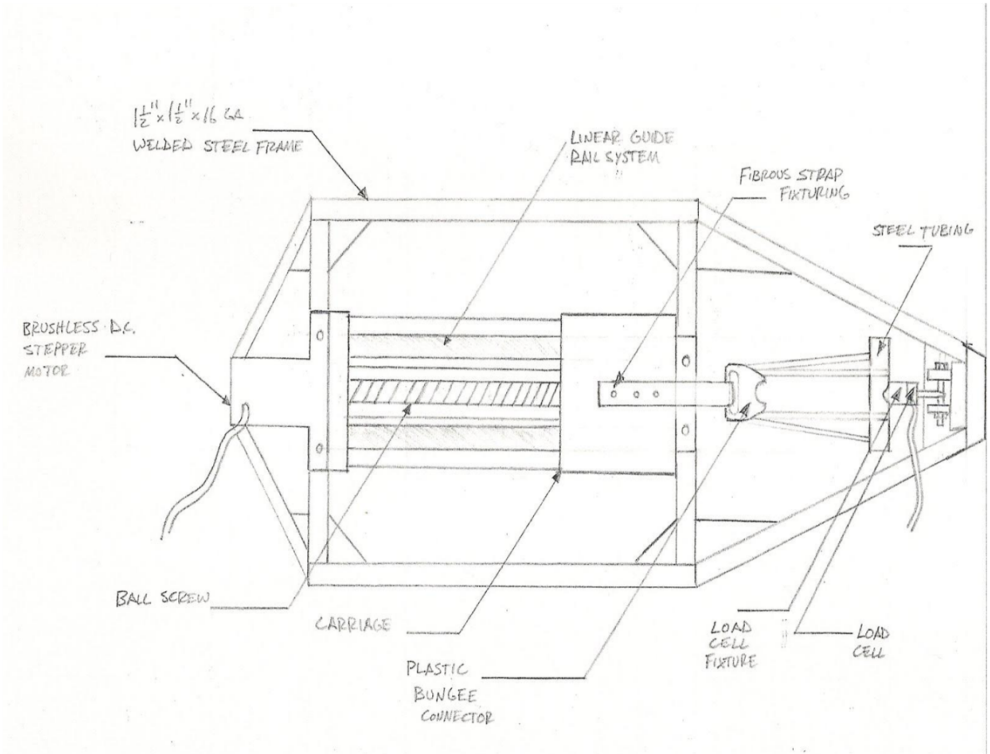

Figure 6: Ball Screw Sketch

The ball screw concept is inspired by existing fatigue test mechanisms that use a similar actuation method. To verify the viability of the ball screw concept, analysis was performed by hand and in MATLAB to derive the system's power requirements.

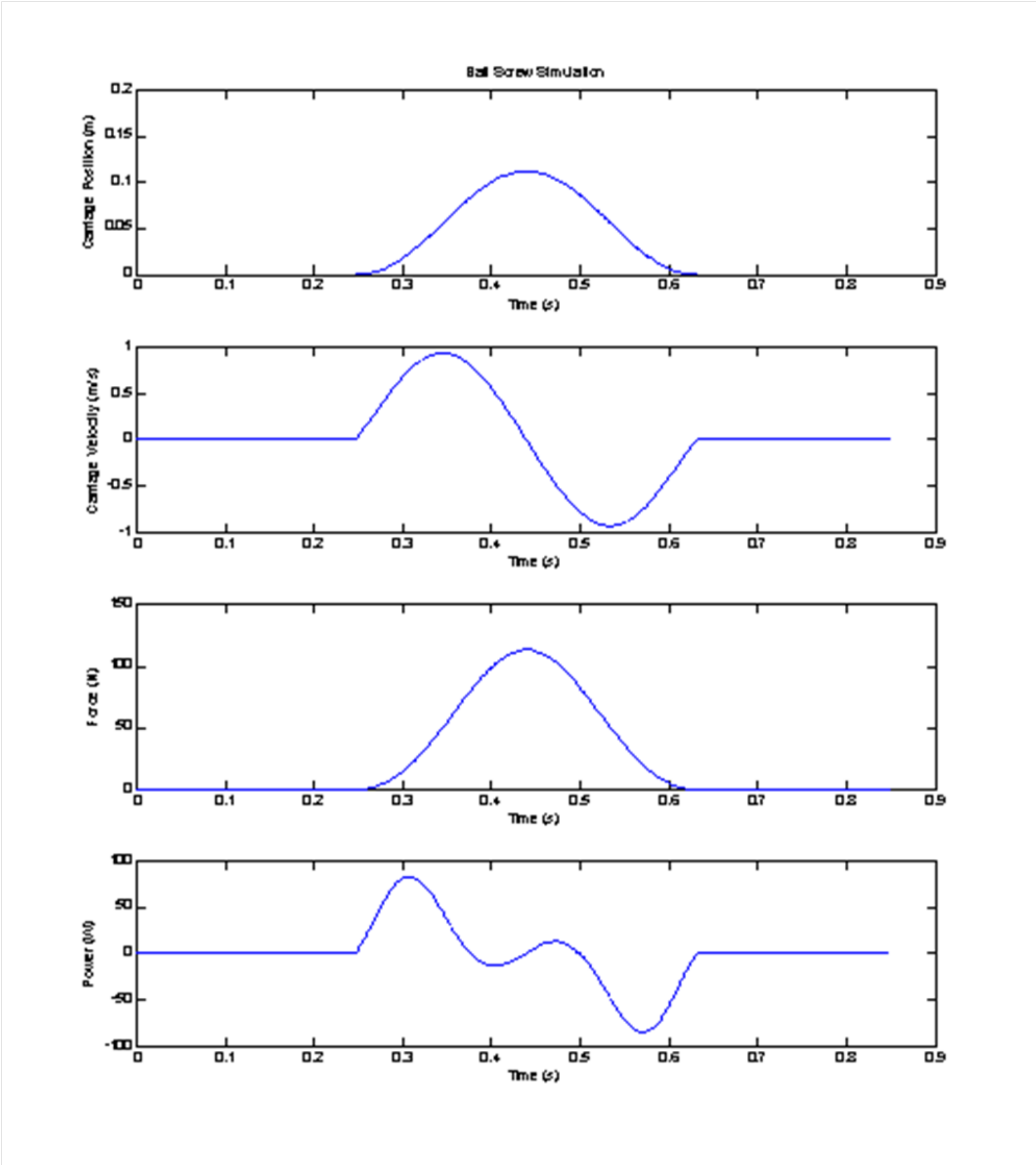

Figure 7: Ball Screw Model

Typical ball screws operating at maximum speed (3000 rpm) produce 58 – 62 dB. [4] The "Noise Element" report from Saratoga, California suggests that the outdoor evening noise not exceed 50 dB. [5] Since the machine will be running continuously it must be capable of producing 50 dB or less. The machine will run at speeds closer to 900 rpm which should produce less noise.

The test needs to be able to run up to a maximum frequency of 2 Hz. Ball screw actuators are capable of running frequencies from 0.001 Hz to 15 Hz, but higher frequencies limit the stroke. At 2 Hz, the design will be capable of producing the stroke necessary. Continually reversing the motor at these frequencies will limit the life of the ball screw and motor combination. The peak force may need to be kept constant throughout the test because the trampoline user's weight will not change while jumping on the trampoline. This feature can be implemented in the control system by allowing the load cell to relay information to the controller about changes in peak force. The ball screw will adjust the stroke accordingly to achieve the original force.

Servo-hydraulics

The servo-hydraulic mechanism comprises a linearly actuating hydraulic cylinder pin connected to a lever arm. The carriage, which holds the bungees, is connected to the top of the lever arm via a pin-slot connection, allowing the carriage to travel linearly along its rails as the lever arm rotates. The test specimens are affixed to the carriage using the plastic connectors used on the trampolines.

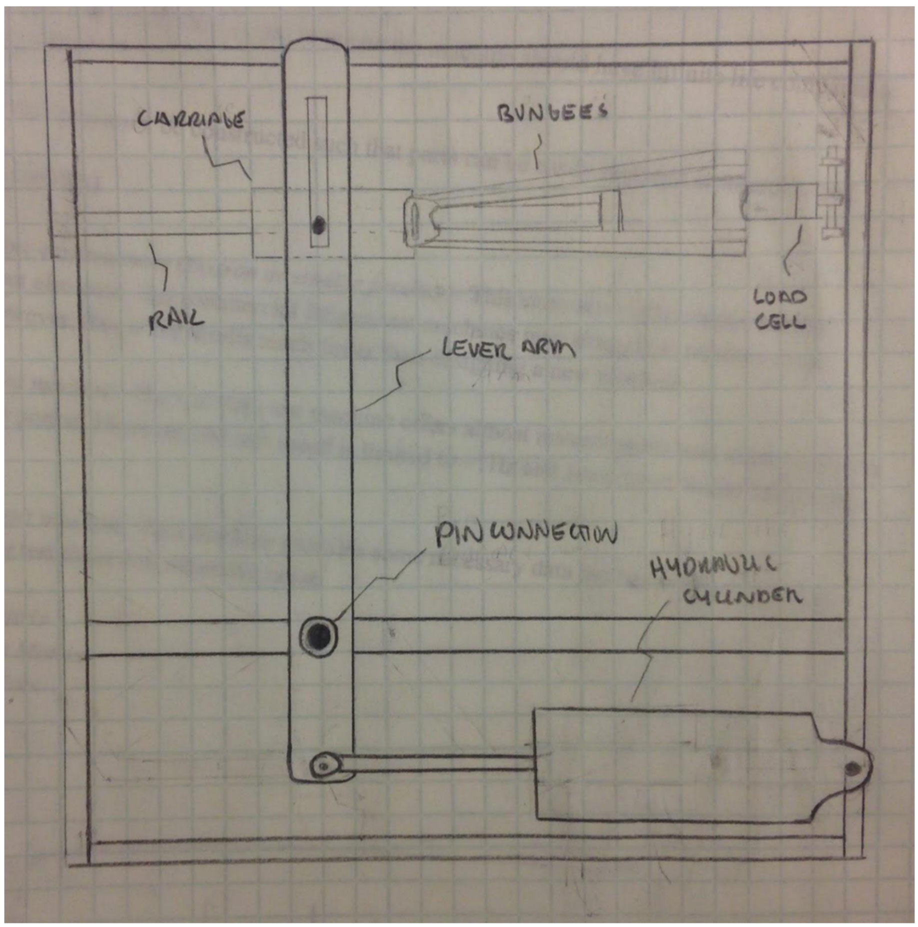

Figure 8: Servo-Hydraulic Sketch

Ideally, the cylinder would drive the carriage directly. However, hydraulics are better suited to high loads at low frequencies, and the combined requirement of 12in stroke at 2Hz would necessitate an excessively expensive hydraulic cylinder and accompanying fluidic system. The lever arm mechanism acts as an ideal energy transducer, magnifying the cylinder's output displacement and allowing it to operate at a lower stroke range. Using simple classical techniques (primarily PID), the hydraulic system's position can be controlled, allowing for accurate replication of the displacement curve.

The primary disadvantages of the hydraulic system are cost and noise. Even a 2" hydraulic ram moving 20lbs at 1Hz produces close to 85 dB noise—presumably, actuating a greater load at twice that frequency would result in even more noise. Additionally, hydraulic components possess less fatigue life than strictly mechanical devices and are typically not designed for continuous duty. The potential for fluid leaks and other necessary maintenance , coupled with reduced machine life, drive the total cost of operation of the hydraulic system upward.

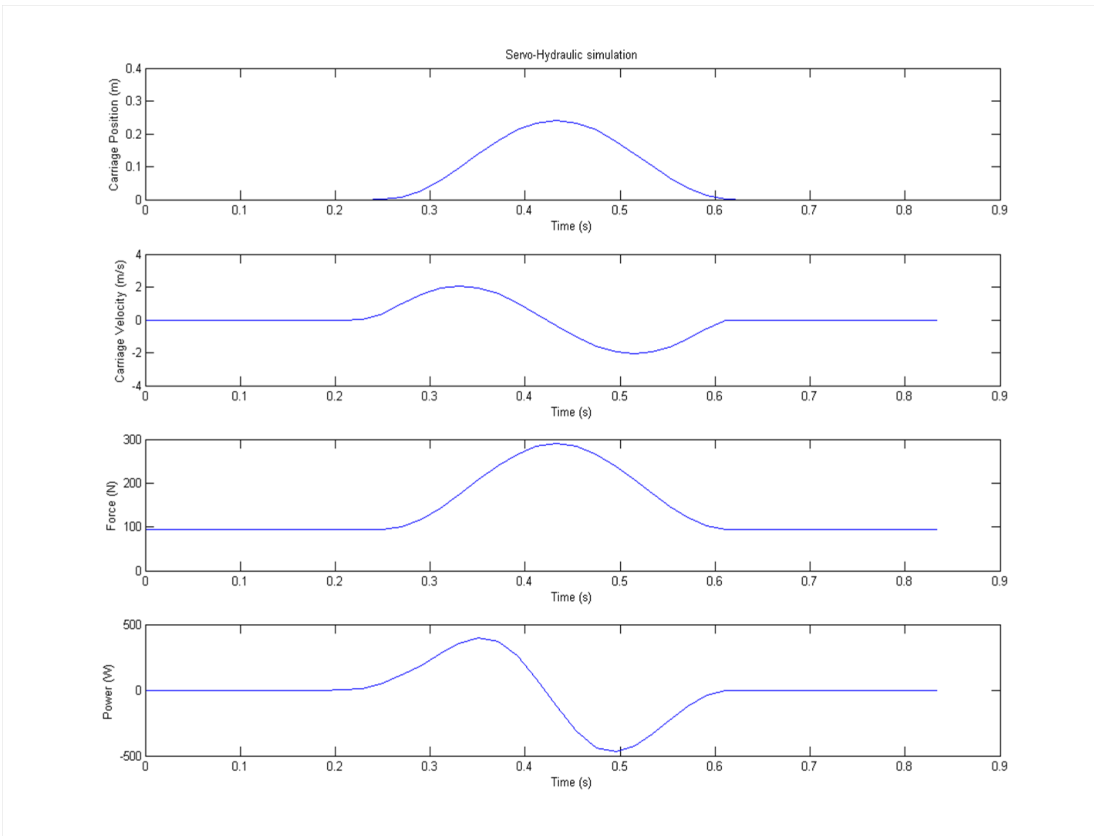

Figure 9: Servo-Hydraulic Model

Testing Outline

Three main tests are desired when analyzing the performance of cord designs. The quasi-static extension test is desired to quantify the stiffness of the full variety of JumpSport's elastic products, including all knot configurations. The standard life test is desired to continuously test the elastic constructs under standard ambient conditions until failure. The accelerated life test is desired to achieve the results of the standard life test in significantly less time.

Quasi-static Extension Test
This test requires the elastic construct to be extended slowly while the force and displacement are recorded. From this raw data, the stiffness of the construct is determined, measuring force as a function of strain or displacement. This test can be conducted at various points throughout the life of each construct to determine when failure has occurred.

Standard Life Test
This test is expected to run continuously at standard operating conditions until failure. Frequencies between 1 Hz and 2 Hz are expected at displacements given by the standard operating conditions. Based on the previous standard life tests conducted by JumpSport, ~2 Million cycles can be expected before a failure. At the specified frequency, these tests could run between 10 and 20 days.

Accelerated Life Test
Without detailed material properties available, a series of initial tests will need to be conducted to form a baseline for comparison when conducting accelerated life tests. These tests will be conducted on a single cord setup in a standardized fixture, likely reflecting JumpSport's latest trampoline design. Then, a series of tests should be run to failure at various maximum strains. The frequency could remain constant rather than the strain rate. These strains should be selected between 0 and 2. It is expected that the relationship between the number of cycles to failure and the strain will be linearly related. If this is not the case, it may be possible to normalize by the strain rate.

Regardless, this relationship can be used to identify designs that exceed the performance of the base design. These designs would continuously fall above the trend line that describes the baseline design. Since promising designs could be identified anywhere on this curve, it is possible to run preliminary tests at significantly higher strains for significantly fewer cycles.

Failure Criteria
Based on previous fatigue life tests conducted by JumpSport, it is not expected that the cords will fail catastrophically. Rather, it is expected that certain strands of the cord may fracture or the stiffness may degrade over time. Failure will be determined if the cords permitted the trampoline to bottom out under standard operating conditions. Software will constantly evaluate the stiffness of the cords during a fatigue test by periodically running the quasi-static extension test. The parameters of the failure determination procedure could be established largely using an algorithm. The user could specify the estimated cycles to failure and the software could anticipate an interval for the quasi-static extension test. If 100 extension tests were expected throughout each test, the cycles to failure could be detected within 1%.

If the stiffness of the cords could be reduced to a linear relationship, the intercept could be used to determine if the system stiffness has been reduced to the point of bottoming out. This assumes the failure mode is crack propagation leading to individual cord strands fracturing and thus the stiffness would degrade linearly.

Standard Operating Conditions
The standard operating conditions must be representative of the average user. The user weight and bouncing frequency would indicate certain displacements functions of time. A specific operating temperature will need to be selected and maintained within the tolerance of the ASTM standards.

T = 70 °F
Max Strain = 50%
Frequency = 1 Hz
Displacement Curve

Management

With a design concept in hand, the tasks were broken down into three subsystems linkage, motor and frame/fixturing. Responsibilities within each stage of development were divided according to the subsystems. Each team member was primarily responsible for one subsystem, but also design collaboratively to ensure consistency. The subsystems for the design phase were assigned to Christopher D'Elia, Andy Brock, and Ryan Murphy, respectively. As team leader, Christopher D'Elia helped direct the team's efforts and expertise. Christopher D'Elia was specifically responsible for motor selection and control, research, analytical modeling, and data acquisition. Andy Brock was specifically responsible for design of the linkage and cable carrier, and communication with the project sponsor and advisor. Ryan Murphy was responsible for frame/fixturing, materials selection, and cost estimates. The design required extensive iteration and collaboration between each team member.

Once designs were considered complete by each individual team member, the complete concept design went through an internal design review. Upon successful iteration after the internal review, the design entered the external design review. After the design was approved, it entered a design freeze wherein any design changes were prohibited or subject to critical review. The team continued with a division of fabrication. This will involve components purchasing, frame fabrication, linkage and fixture machining and design and build of the DAQ and control system. The team continued with a regroup for assembly, verification testing and material testing. The team followed the Gantt Chart provided in the Appendix to meet all deadlines and goals for the project.

By the end of spring quarter, the fatigue-testing apparatus was mostly constructed, awaiting the arrival of the motor, drive, and gearbox. Upon integration of the drivetrain in the fall, final manufacture and assembly were completed, and the machine prepared for testing and final delivery. Prior to the motor arrival the team developed and presented the plan for testing the apparatus itself, though time constraints disallowed thorough testing of the machine. Verification testing will need to be conducted on each of the measurement aspects of this fixture to ensure that the results are not directly driven by the test protocol. This will involve eliminating stress concentrations induced by fixturing that is not representative of the installed cords. Also, measurements taken dynamically must be verified to ensure noise and incomplete relaxation is compensated for.

The complete machine and report were presented at both the spring and fall senior project expos, before delivery to JumpSport in December of 2013.

Lead Concept

Arriving at a top concept involved evaluating the three lead concepts against customer and engineering requirements. To summarize and objectify this analysis, an annotated matrix was created with each of the three top concepts and the requirements. Each team member was responsible for one concept design and provided annotations for each requirement with respect to that concept. Finally, the concepts were ranked against each other and weighted using the customer requirement weights and engineering requirement weights from the QFD. Overall, the slider-crank scored highest, followed closely by the ball screw; ultimately, the slider-crank was chosen.

Motor System

The motor subsystem must be designed to power the mechanism when cycling during the most aggressive tests and during the most controlled tests. The most aggressive tests would involve the stiffest cords, 10mm diameter, the largest strain, 200%, and the highest frequency, 2 Hz. During the most controlled tests, the system must cycle the stiffest cords, 10 mm diameters, to the largest strain, 200% at a low frequency, 0.5 Hz. The controller must output the position while recording the force throughout a single revolution to produce the cord stiffness plot.

Since the aggressive setup dictates the maximum motor output power, a simulation of these conditions was conducted in MATLAB. The results indicate 500 W maximum power delivered to the cord. The simulation also confirms that the load will be constantly reversing. To avoid noise or excessive wear, low backlash precise servo motors and gear reducers should be used with a servo controller.

Linkage System

The slider crank mechanism includes two linkages: a crank arm and a connecting rod. The crank arm originally included a linear actuator, to allow dynamic stroke adjustment, enabling constant force tests by accounting for the reduction in stiffness that occurs as the specimens are fatigued.

The change in crank arm length will be determined by studying the failure criteria. The actuator needs to be able to increase the stroke of the mechanism such that the crank can continue to cycle the bungees to constant force as the bungees fatigue. Failure is characterized by a bungee displacement that corresponds to the trampoline bottoming out.

Because the crank arm will be constantly rotating, power will need to be delivered to the linear actuator by a slip ring or a similar mechanism. The asymmetric weight distribution of the linear actuator also means that the crank will need to be dynamically balanced to avoid excessive vibration.

There will be a bracket affixed to the end of the crank arm so that the second linkage can be connected at various points along the crank arm, as shown in the figure below. The purpose of this bracket is to adjust the range of strain which the specimens will be cycled.

The connecting rod length will be adjusted based on the sample length and desired stroke. The arm will have multiple bolt holes in order to increase adjustability and accommodate the range of potential strokes by maintaining constant proportionality with the crank arm.

Fixturing and Frame System

Fixturing for the bungees and springs will be designed for both the static and dynamic ends of the machine. The static end will consist of a linear actuator that will be used for initial setup and stroke adjustment. The user will be able to enter the dimensions of the bungee or spring to be tested and the actuator will set up the test accordingly. Attached to the linear actuator will be steel tubing, of the same diameter and finish as the trampoline's structural ring, which will be connected to a load cell. The dynamic end will consist of a carriage which attaches to the connecting rod of the slider crank mechanism. The carriage will glide with minimal friction using ball bearings and a specially outfitted guide rail system. The carriage will support fixturing similar to the mat on the trampoline. The mat material will be connected to the same plastic connectors used on the trampoline.

The frame will be made out of steel tubing with an area moment of inertia that is capable of supporting an alternating compression force of 100 lb. The design will involve a bench top style welded frame that minimizes the footprint while containing the mechanism. Although preliminary calculations regarding structural integrity are established, the frame will be fully designed after all mechanisms are selected so that dimensions are properly established for part position, size and placement of holes and covers for moving components.

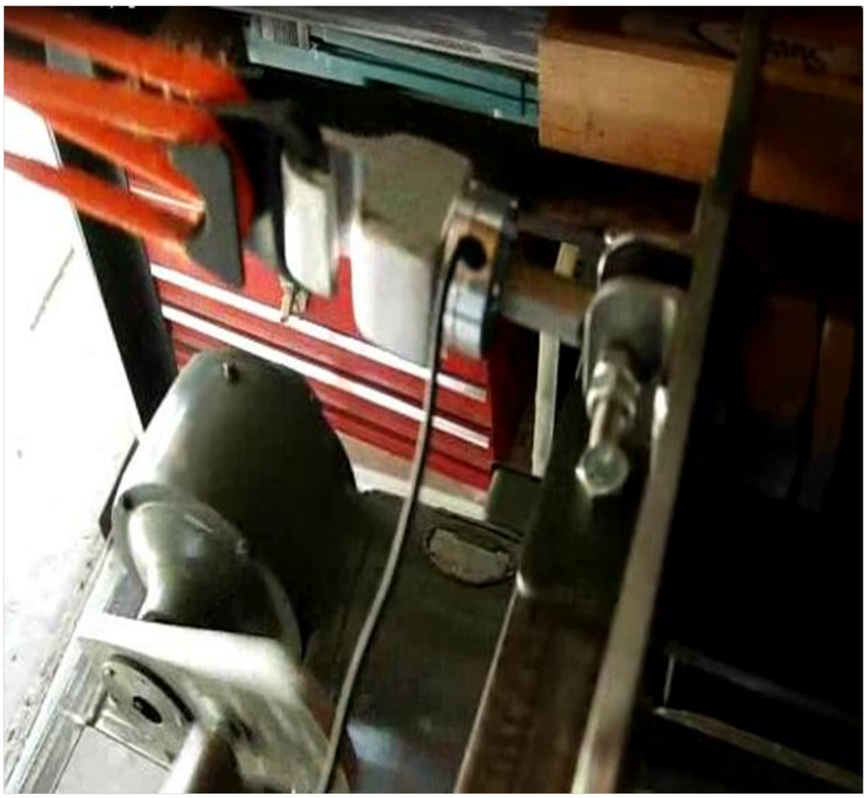

Figure 10: Current Load Cell Fixture

Final Design

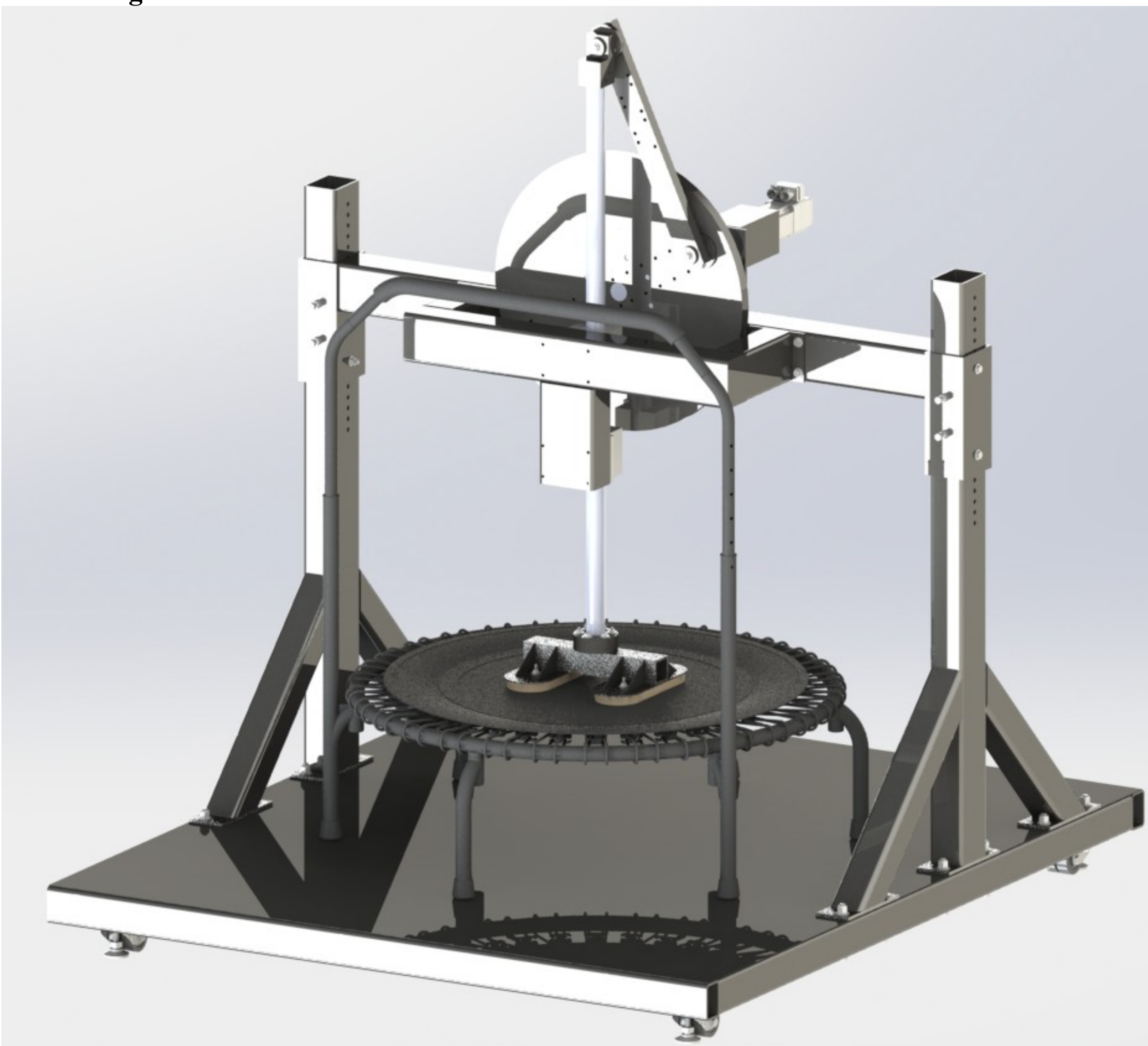

Figure 11: Final Fatigue Test Machine

In April 2013, the scope of the project was modified to include fatigue testing of trampolines in addition to bungees and springs. The fatigue testing of trampolines was originally a senior project assigned to Caroline Reeves, Ethan Flory, and Will Robertson, collectively known as the Bounce Test Trio, who began their project in the Fall of 2012. The change of scope unified the objective of the two projects, and thusly the Bounce Test Trio joined the current project team to aid in the completion of the final design. The results of their work are available in their senior project report, and are not included here.

Functional Description

Motor

The requirements for the motor and drivetrain arise from a system model of the trampoline test machine. The model includes an electric motor, system mass in three locations, and a spring damper array representing the trampoline. The electric motor model is complete with armature inertia and is coupled to a speed reducer. On the output of the speed reducer is a large flywheel inertia containing the crank rod. The connecting rod then attaches the crank rod to the slider. At the end of the slider is a ram mass, which contacts the trampoline spring damper array. The top of the slider stroke can be adjusted relative to the trampoline mat and all the system masses and inertias can be tuned.

By tuning the system to the stiffness of the trampoline and desired frequency, the power required can be significantly reduced. Initially, the system was setup for the max loading condition: 12in trampoline stroke at 2hz. The system operated in steady state, drawing about 500 watts at the peak of each stroke. Since the load is constantly alternating and varying in magnitude, a PI controller was necessary to prevent the system from cycling out of control. The system model, outlined by the block diagram, is shown in Appendix K. From this model, various outputs are available including position of the impactor and motor voltage in time. These two particular outputs have been reproduced in Figure 11 and Figure 12, respectively. Each of these plots shows the system response to a step input in desired velocity. The slight asymmetry in the transient response demonstrates the value of the controller in quickly stabilizing the system. Eliminating the control and driving at a constant voltage yields an unsteady response since the motor will attempt to match its speed to the load, which is constantly changing direction and magnitude. A rather large back electromotive force voltage can be seen in Figure 12; the voltage generated by the motor on the return stroke must be dissipated through the controller.

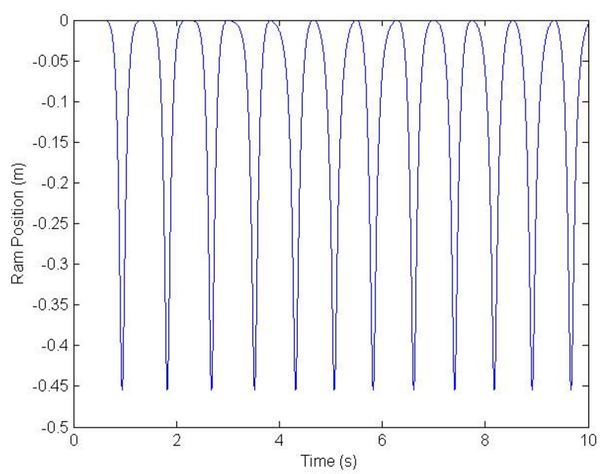

Figure 12: System model slider position response to a step input in velocity of 2 Hz.

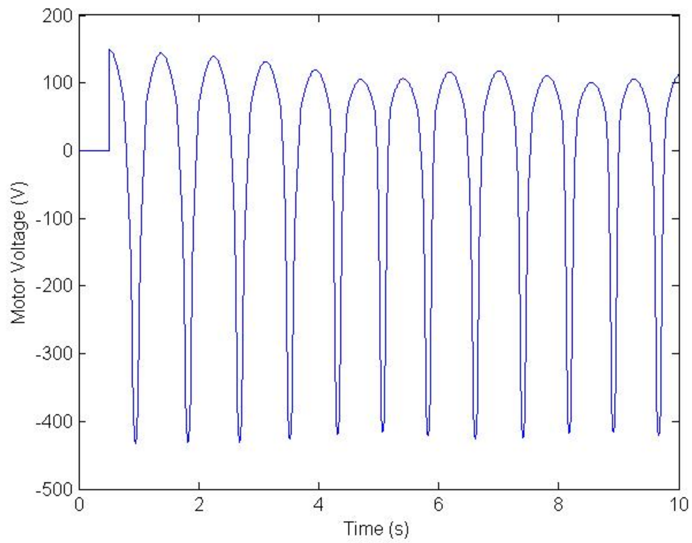

Figure 13: System model motor voltage response to a step input in velocity of 2 Hz.

Designing the slider crank mechanism properly and tuning the system mass to the stiffness of the trampoline can considerably reduce the input power. Rather than all the energy being provided by the motor, most of the energy is recovered into the mass and returned to the trampoline on the next stroke.

With an understanding of the motor power requirements, various motor types and manufactures were considered. Recalling from the QFD, quiet operation is an important requirement; since backlash in the first generation cord testing machine resulted in most of the excessive noise, eliminating backlash is a necessary part of the new design.

Standard AC gear motors were considered initially; however, the desire for precise speed control and limited backlash eliminated them as a candidate. Since DC servo motors are currently being phased out of automation, AC servo motors were considered since they offer precise speed control and regulation as well as compatibility with anti-backlash gear reducers.

Since Baldor motor parameters were put into the system model for initial sizing, one of these motors was placed in the SolidWorks model initially a with an appropriate gear reducer. However, a unique connection with Buckles-Smith, an industrial automation supplier, provided the best option for purchasing the motor and drive components. A sales engineer was contacted and provided with the system description. Using an array of Rockwell automation components, Buckles-Smith has outlined the necessary drive and control components. These components can be seen in the quote provided by Buckles-Smith in Appendix J. The sales engineer has also reviewed the previously outlined back EMF voltage concern and suggested a capable controller.

Motor Mount

The motor mount is a welded construct consisting of flat plate steel and box tubing. A 9.921" long 3x5x¼" piece of box tubing is centered on a 4x8x¼" plate and welded. The 4x8x¼" plate is mounted to the gantry using 3/8"-16 UNC, 5" long, grade 8 alloy steel hex head cap screws and 3/8"-16 UNC nuts. A 7.75x5x½" steel plate is welded to the front of the box tubing which provides the mounting surface for the Micron Motioneering planetary gear box. The mounting surface consists of 5 thru holes, 4 of which allow M8 thread black oxide socket head cap screws to attach to 4 M8 nuts with lock washers on the nut side and flat washers against the screw head. The center thru hole is used to concentric locate the gearbox therefore establishing concentricity for the motor, gearbox, shafts and flywheel. The mounting surface is reinforced with two welded 45° angle 3/8" gussets. Welded atop the box tubing and against gantry mounting plate is a 4x5x½" steel plate with 4 ¼"-20 UNC threaded holes for attaching the pillow block. The pillow block mounting plate and box tubing configuration were designed to accommodate the diameter of the LoveJoy shaft coupling while ensuring that there was minimal deflection with the high alternating loads during testing.

Linkage

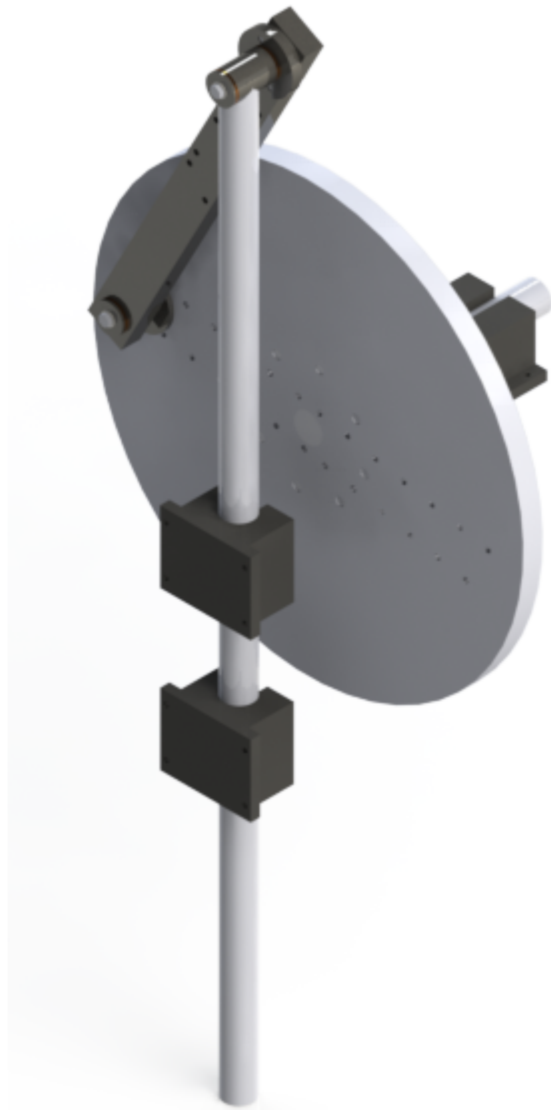

Figure 14: Linkage Subassembly

The linkage subassembly is a vertically oriented slider crank. The linkage design was driven by the need to deliver the prescribed range of actuating strokes to the trampoline (from 0 to 12") with a resolution of ½", and the need for infinite fatigue life. It must not cause or allow for excessive vibrations or audible noise, and must not pose a threat to the operator while in normal use.

The flywheel is a circular piece of 1" steel plate, with an outer diameter of 22.5". There are 26 holes cut for a 5/16"-24 tap, 8 cut for a ½"-13 tap, and a 1 ½" center bore. The 5/16-24 tapped holes are placed in the hole pattern shown in Figure 12, while the ½"-13 tapped holes are evenly spaced at a 3" radius.

The flywheel hole pattern is implemented as a means of changing the effective length of the crank arm to allow for adjustable stroke. The pattern minimizes the number of holes needed to affix the connecting stud by allowing the stud to "walk" back and forth between the holes to achieve each position, eliminating the need for separate holes for each position. The pattern maximizes space between the holes to reduce the likelihood of material failure.

The flywheel design is tuned to the stiffness of the system to allow for an optimal exchange of energy into its mass. Because of the high likelihood of dynamic imbalance, the flywheel will be sent to a specialist for dynamic balancing to reduce the likelihood of vibration and noise.

The piece that forms the interface between the flywheel and connecting rod is called the connecting stud. It is machined from a piece of 2 ⅞" 1018 round bar steel, with two clearance holes for 5/16" bolts drilled at a 1 1/8" radius, and a 1" blind center hole tapped for a 5/16"-24 bolt. This member was designed for fatigue life; it must be rated for infinite life under the heaviest loading conditions. It must also be stiff enough not to deflect significantly under these conditions, in which case it could potentially cause binding in the bushings. It is geometrically constrained by the need for clearance between the fasteners and the connecting stud, and by the available area for interfacing with the connecting rod, and designed to meet this need.

The second arm of the slider linkage is called the connecting rod. The connecting rod is cut from 1" by 2" 6061 Aluminum flat bar, and drilled with 14 5/16"-24 holes as well as with a 1" clearance hole for the connecting stud. At the interface between the flywheel's connecting stud and the connecting rod are two abutting ½" flanged Oilite bushings, which were selected because of their low cost/life ratio. While these bushings will not have infinite life due to wear, replacement is simple and cheap, and a more effective design solution than ball or roller bearings, which will not have much longer life.

The connecting rod was designed to meet the necessary fatigue life criteria without compromising the dynamics of the system by being too heavy. The connecting rod's effective length is adjustable in order to maintain proportionality with the crank arm. To this end, a series of holes are placed on the upper half of the connecting rod, to which another connecting stud can be fastened. This hole pattern was designed similarly to the hole pattern in the flywheel; it maximizes the spacing between holes and reduces the total number of holes necessary by allowing the upper connecting stud to "walk" between holes. The normal distance between each hole is the same as on the flywheel so that the holes on the two connecting studs can be placed in identical locations.

The connecting stud that interfaces between the connecting rod and the slider linkage is called the second connecting stud. It is machined from the same piece of 2 ⅞" round bar steel as the other connecting stud, but will not have the same step. This stud is designed to resist deflection and have infinite fatigue life under the heaviest loading conditions, though periodic examination is recommended in case of unexpected loading or wear.

To prevent the load cell signal cable from interfering with the linkage and to improve its fatigue life, a cable carrier assembly, consisting of an impactor-mounted bracket and IGUS energy chain, was designed. The cable carrier consists of 30 links of the Z14 zipper style energy chain. The cable carrier bracket is comprised of a 1/8" circular plate of radius 45mm, with 6 equally spaced 8.5mm holes placed at a 33mm radius, and a central hole of diameter 46.990mm. It has an extrusion that comes 280.645mm from the center of the circular plate, gently sloping from the edges of the circular plate to its 14mm width, which protrudes 167.650mm from the edge of the curve to the tip of the extrusion. The slope that connects the extrusion to the circular plate is 80mm long and is based on a radius of 162.984mm. Two 6mm holes are placed 120.650mm and 134.55mm horizontally from the center hole of the circular plate, in the center of the extrusion. The metric dimensions are necessitated by the metric load cell dimensions.

Similar pillow block bushings have been implemented on the main drive shaft and slider shaft. To support the slider shaft, two tight tolerance open PCB Linear bushings are used. These bushings are designed specifically for the 1½" ground shaft, which has tight cylindrical and radial tolerances. To avoid excessive distance between the slider shaft and connecting rod, the open bearings are used since they have a smaller overall thickness. The bushings also offer a self-aligning feature, which will provide easy alignment and have some allowance in case the slider linkage deflects. These bushings were selected based on their low cost and ability to withstand the required 50 tests before servicing. Should servicing be necessary, these pillow blocks have easily replaceable bushings.

On the main drive shaft, two pillow block bushings were also used since calculations demonstrated that bearings offered a limited gain in life. These bushings are fully enclosed since no vertical clearance issues exist. They interface a similar ground shaft to the slider shaft that is connected between the motor and flywheel. One end of this shaft is coupled to the motor via a LoveJoy shaft coupling. The other end of the drive shaft interfaces the flywheel via a drive flange that is welded to the main shaft. The center of the drive shaft passes through the drive flange and into the flywheel in order to locate the shaft concentric with the flywheel. A series of bolts pass through the drive flange and thread into the flywheel to hold the construct together. The drive flange can be welded to the main shaft and then faced flat to ensure that the flywheel bolts perpendicular to the drive shaft in final assembly. Once assembled, this flywheel, drive flange, and main shaft can be professionally balanced and installed on the test machine.

<u>Frame/Gantry</u>

Figure 15: Frame Subassembly

The base is a steel box tubing structure outfitted with adjustable leveling casters which provide for ease of transportation and can lock into place for fixity during testing. The framework is covered by a piece of 10 gauge sheet steel that serves as a rigid platform for the trampoline and frame subsystem. The attachment points for the gantry uprights accommodate 3 fastening plates, each with 2 holes, on either side of the base. The casters are connected by plates welded into each corner of the framework. Conducting tests on this structurally integrated platform allows the dynamic reaction forces to be contained by the structure. This eliminates the need to bolt the structure to the floor (which is still possible if concrete anchors are implemented) without compromising structural rigidity.

The frame was broken down into three distinct sections: base, uprights, and gantry. These sections were designed as separate welded constructs that are bolted together to form the final frame assembly. This promotes ease of manufacturing, as well as assembly/disassembly and transportation.

The base is designed to provide a platform for the trampoline to sit upon during testing. It also incorporates caster mounts and an attachment for the gantry. The caster mounts include a caster plate that is welded in each corner of the framework. The attachment points for the gantry accommodate 3 fastening plates, each with 2 holes, on either side of the base. The framework will be covered by a piece of 14 gauge sheet metal that will serve as the platform surface. Conducting tests on this structurally integrated platform allows the dynamic reaction forces to be contained by the structure. This eliminates the need to bolt the structure to the floor, and thus directly contributes to making the system more mobile.

The uprights have been designed to achieve the necessary adjustments associated with variable stroke and linkage proportionality. The hole series along the gantry upright permits the gantry to be raised to the correct height for each stroke. Once at the correct height, the gantry is through-bolted to the uprights in two locations and clamped by two additional bolts perpendicular to the through bolts. The through bolts are spaced such that they correspond to the hole series along the gantry. These bolts support the bulk of the direct shear load as well as permanently locate the gantry height. The additional bolts perpendicular to the through bolts can be tightened against the gantry upright to remove any play between the gantry and the gantry upright.

The gantry consists of three sub-sections: the gantry crossbar, the slider bearing mount, and the motor mount. The gantry crossbar includes a 4” x 4” x 3/16” main box tubing member that is supported by two pieces of custom box tubing. The custom box tubing is sized to fit around the gantry uprights and made from 4 individual lengths of ¼” plate welded along the edge. For alignment and potential future adjustment, the slider bearing mount is a separate construct bolted to the gantry. This mount supports the linear bearings in the center of the flywheel and spans around the flywheel to connect back to the gantry crossbar. For a similar reason, the motor mount is also a separate bolted construct.

Bungee Fixture

The bungee/spring fixture was designed such that the test specimens would be tested similar to real life circumstances. The top fixture consists of ½” plate that fastens to the load cell and slider linkage. Nylon strap material, similar to the trampoline mat, is held in between a 1/8” plate that fastens to the upper 1/2” plate with 3/8” grade 8 alloy steel hex head cap screws. The nylon strap is looped around the original trampoline clips which connect the mat to the bungees. The other end of the bungee cords wrap around 4” long, 1 1/8” ID steel tubing. An aluminum threaded shaft is press-fit inside the tubing where the Loadstar™ load cell can be fastened. A machined aluminum spacer is inserted in between the load cell and tubing to ensure secure fixturing. The lower end of the load cells are fastened to high strength steel ball joint rod ends. The rod ends are held by 1/4” grade 8 alloy steel hex head cap screws that mount with thru holes to two 1 1/2” steel angle iron and held by 1/4” nuts. The angle iron is fastened to the base of the frame using two 3/8” grade 8 alloy steel hex head cap screws. Steel springs can be fastened in a similar fashion.

Supporting Analysis

Linkage
Every member of the linkage assembly was designed for maximum fatigue life under the heaviest loading conditions, and analyzed to ensure that angular and positional deflections were not large enough to cause binding or critical interference. The complete analysis is available in the corresponding appendices for each part.

Frame/Gantry
Every member of the linkage gantry was analyzed to ensure structural stability and fatigue life. The complete analysis is available in the corresponding appendices for each part.

Motor/Drivetrain

The analysis pertaining to the motor and drivetrain is primarily described in the design and system modeling sections. The results of this analysis were used to select the motor components.

Safety Considerations
The potential for damage or injury inherent to high-load high-speed actuation necessitated extensive consideration of safety hazards. 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 address this concern, a protective cage will be built around the flywheel, connecting rod, and slider linkage. Acrylic or polycarbonate sheets mounted in a metal framework will be attached to the gantry, extending down to cover the flywheel and pillow blocks and extending 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, but protects the observer in the course of normal operation by prohibiting foreign intrusions, and in the event of catastrophic failure by containing any projectiles.

As with any high-energy system, the flywheel is a potential source of dynamic instability that might cause premature or unexpected machine failure, and in the catastrophic case is likely the single most dangerous component. The nearest subsystems have been accordingly designed to reduce this danger. The drive flange is affixed to the flywheel with 8 grade eight half-inch bolts and the slider linkage pillowblock mount spans around the flywheel, partially surrounding it. The flywheel itself is symmetrically designed precisely cut by waterjet, reducing the potential imbalance due to manufacturing mistakes.

The gantry was designed at infinite life to insure the machine would be able to withstand dynamic loads during fatigue testing. The analysis consisted of examining shear forces, bending moments, torques and the subsequent stresses. Indeterminate loading was used to incorporate a worst case scenario that would indemnify the best possible safety factor. For the gantry, a 4x4x3/16" steel box tubing was chosen which resulted in safety factor of 1.3.

Material Selection

Linkage

The flywheel is manufactured from 1018 carbon steel. This grade of steel provides sufficient strength and stiffness to withstand the heaviest loading conditions without deflecting or yielding, and is dense enough that the flywheel geometry can be made to fit within the confines of the frame and gantry while still being massive enough to fit the system specifications.

The connecting rod is made from 6061 aluminum flat bar. This grade of aluminum is strong enough to withstand the necessary fatigue loading and provide near-infinite life (the S-N curves for aluminum do not level off, so infinite life cannot be approximated as it can be for steel) but is lighter than steel. The weight of the connecting rod is important to the dynamics of the system—because of its radial distance from the center of the flywheel, any weight added to this reciprocating member increases the inertia of the system by a significantly greater factor than added weight to other members, such as the driveshaft. Though aluminum is less stiff and typically weaker than most steels, fatigue analysis showed near-infinite life under the heaviest loading condition, and beam analysis showed no significant deflection or yielding. Work hardening at the interface between the flywheel and the connecting stud (the hole through which the connecting stud passes) may prove to be the cause of failure in fatigue.

The slider linkage is a 1.5" OD, 1" ID hollow ground shaft. This member is primarily in compression, and according to stress analysis will not suffer significant deflection or undergo buckling even under the heaviest loading conditions.

The connecting stud that interfaces between the flywheel and connecting rod is machined from 2 7/8" diameter free machining steel. This steel was chosen for its balance of strength and manufacturability. Fatigue analysis shows that for the grade of steel chosen, infinite life is possible even under the heaviest loading conditions. Additionally, because of the complexity of the part, high machinability (as rated by AISI standards) was desirable to reduce production time and difficulty. The part was originally sized to be 3" in diameter, a dimension for which round bar stock was not readily available. The 2 7/8" diameter, which was determined to be sufficiently large to meet the design specifications as limited by the radial distance between the two clearance holes for the fasteners, thusly became the round bar stock dimension of choice.

The second connecting studs are machined from the same material as the first. Though the second connecting stud is not as complex, lacking the step present in the other connecting stud, manufacturing time and complexity were reduced by making two parts from the same stock.

The tube into which the connecting rod-slider linkage interface slides is DOM round tube, mitred to the top of the hollow ground linkage and connected to the second connecting stud via a pair of bronze flanged bushings.

Frame/Gantry

The base framework, which supports the machine, is designed using 2" x 3" x 1/8" box tubing and has ample room for the trampoline to be positioned in multiple loading zones.

The base framework braces were analyzed for fatigue loading with a distributed load from the trampoline via the sheet metal floor. These members provide a safety factor of 3.15 and maximum deflection of -0.048"es under max loading conditions.

The gantry that supports the motor and slider linkage pillow blocks is made of 4” x 4” x 3/16” box tubing. Fatigue analysis showed that the gantry was over designed with a safety factor of 3.0. Deflection calculations showed that the gantry would have a maximum deflection of 0.025”es with maximum loading conditions.

The gantry uprights are made from 3” x 4” x 3/16”. Fatigue analysis showed that the gantry uprights had a safety factor of 5.1. Deflection of the gantry upright resulted in -0.0112”. All mounting plates are made from ¼” cold rolled steel flat bar. Cold rolled steel was chosen for its clean surface finish and machinability.

The pillow block to gantry support resulted in a fatigue safety factor of 25.1 and a maximum deflection of -0.000609”. The pillow arm support resulted in fatigue safety factor of 12.5 and a maximum deflection of - 0.0049”.

The gantry upright braces are made from 3” x 3” x 3/16” box steel. These braces were chosen as they matched up perfectly with the 3” section of the 3” x 4” x3/16” box tubing. The caster plates were pre-manufactured out of 3/16” flat bar. The detailed design calculations and drawings for the frame and gantry members can be found in Appendix G.

Fabrication and Assembly

Linkage

The profile, center bore, and conservative tap drill holes for the flywheel were cut by H_2O Precision, Inc, a waterjet cutting service located in Hayward, CA. The center bore was precisely machined to fit the drive shaft. The tap drill holes were bored to their correct tap drill diameters, and then tapped with ½”-13 taps for the flange hub fasteners and 5/16”-24 pitch taps for the connecting stud fasteners. If deemed necessary, the flywheel will be balanced dynamically to account for any eccentricity or imbalances that might cause system vibrations or instability.

The connecting rod was cut from bar stock and sanded to its final length. The clearance hole for the flywheel connecting stud is bored to a 1” diameter. The smaller holes for the fasteners for the slider linkage connecting stud are bored and tapped for 5/16” -24 pitch fasteners.

The connecting studs were first cut with a band saw from the round bar stock, then faced flat on a lathe. The connecting stud that interfaces between the flywheel and the connecting rod was turned to its nominal dimensions in two operations—one to turn the diameter of the part to the first diameter, and a second for the other diameter. The second connecting stud was turned to its nominal dimensions in a single turning operation. The two clearance holes for the threaded fasteners on the base of each part were bored to 3/8” diameter. Both parts have a 5/16”-24 pitch tap drill bored 1” blind in their center. Each piece is machined to obtain a fine surface finish, reducing asperities and improving the wear-life of the bronze bushings that interface them. The connection between the connecting rod and slider linkage is cut from DOM round tube and mitered to the linkage.

Frame/Gantry

All members were cut with enough material left over to face to length. All frame and gantry members were TIG welded due to the strength and aesthetic appeal of the welds. 3/16” pre-manufactured steel caster plates were welded on the bottom of the frame. Casters were fastened to the caster plates with 7/16” grade 8 nuts and bolts. A 10 gauge cold rolled steel plate was welded to the top of the frame, providing the surface for the trampoline to be placed on. The gantry uprights are TIG welded to the 45 degree angle braces. Flange mounts are TIG welded to the bottom surface of the uprights and braces. Coradial holes at 17/32” were drilled through the flange mounts, sheetmetal floor and the frame so that the gantry upright can be fastened to the frame with ½” grade 8 nuts and bolts. The end of the gantry is welded to a box slider made from ¼” steel plate. Holes are drilled in the gantry slider so that the gantry may be bolted to the uprights at the necessary height. The gantry uprights encompass 22 holes drilled at ⅜ of an inch spaced ⅞” apart. The gantry was fastened to the uprights with ⅜” grade 8 steel nuts and bolts. 3 x 4 box tubing is TIG welded to 3 x 5 box tubing and flange mounts. 13/32” holes were drilled in the flange mounts. ⅜” grade 8 steel nuts and bolts fasten the flange mounts to the gantry. The box tubing was drilled with the appropriate holes needed to support the pillow blocks. On the opposite side, 3” x 5” box tubing was welded to the flange mount which is fastened to the gantry. ¼” flat bar is welded to the cantilevered end of the 3 x 5” box tubing, which in turn supports the motor.

Assembly Instructions

Frame

1. Slide the 12 ½"-13 bolts (with washers) into their through holes on the base, with the threads facing upwards. It may be necessary to lift one edge of the base several inches. Locate and affix each upright with the corresponding nuts.

2. On each upright, place a pin in the hole that corresponds to the lower edge of the gantry sliders. This pin provides a catch should the gantry fall, and is used to locate the gantry. Lift the gantry onto the uprights and slide it to the desired vertical position. Refer to the provided table to determine the appropriate gantry position for a given stroke. Bolt the gantry into place using the four 3/8"-16 bolts, taking care not to strip the permanently affixed nuts.

3. Using the four 3/8"-16 bolts, attach the motor mount to the gantry. Slide the driveshaft into the bearings and check for perpendicularity with the gantry. Align the two horizontal pillowblocks and bolt them into place using the eight ¼"-20 bolts.

4. Affix the slider bearing mount to the gantry using the eight 3/8"-16 bolts. If the vertical pillowblocks are not attached, affix them.

Linkage and Motor

1. Lift the flywheel above the slider bearing mount, then slide it onto the driveshaft. Align the eight drive flange bolt holes and ensure that the flywheel and drive flange are flush. Thread in the eight ½"-20 drive flange bolts. Bolt the connecting stud with two steps into its appropriate radial location on the flywheel using the two 5/16"-24 bolts.

2. Fit the slider linkage into its vertical bearings, taking care not to let it drop. Place the two larger bronze bushings into the horizontal tube at the top of the slider linkage. Insert the connecting stud with only one step into the tube.

3. Install the connecting rod by first turning the flywheel to a convenient position such that the first connecting stud is accessible. With the two smaller bronze bushings in place, slide the lower end of the connecting rod onto the first connecting stud. Align the upper connecting stud with the holes corresponding to the appropriate location on the connecting rod and bolt it into place using the two 5/16"-24 bolts. Install the 5/16"-18 bolts and the large 5/16" fender washers in the central threaded connecting stud holes.

4. Affix the impactor or bungee fixture to the end of the slider linkage, beneath the load cell and cable carrier bracket, using the ½"-13 socket head cap screw. Check all components to ensure correct alignment, especially the cable carrier bracket, which can be noisy if misaligned.

5. Affix the motor, gearbox, and to the motor mount using the four M8 socket head cap screws, taking care to ensure both vertical and horizontal alignment. Manually rotate the fully assembled linkage, checking for binding and interference. Apply lubrication as necessary.

Maintenance and Repair

Motor
While the linkage bushings are expected to complete ~20 million cycles or ~10 tests, the motor and drive components are expected to last substantially longer since their service is more complicated. The motor and drive bearings have been selected to sustain ~100 million cycles or 50 tests. A component of Buckles-Smith motor and drive selection process involves selecting components capable of achieving the desired life.

Linkage
The structural members of the linkage are expected to sustain at least ~100 million cycles, or 50 tests, without wear or fatigue, and should not require maintenance. The smaller parts, such as bushings and fasteners, will require inspection after every test to ensure integrity. The connecting studs for the connecting rod should be frequently inspected as well. The bushings are expected to last for ~20 million cycles or ~10 tests, after which they may need to be replaced to prevent excessive wear.

Frame/Gantry
Under normal operating conditions the frame will not require maintenance or repair. The wheels of the casters, however, will wear out if the machine is be moved frequently as they are made of hard rubber, but can be replaced. The nylon lock nuts are reusable, but will deteriorate withfrequent disassembly and therefore should be replaced after visual inspection determines nylon failure.

Bungee Fixture

The structural members of the bungee fixture should not require maintenance in the normal course of operation, though periodic inspection for fatigue is recommended. The base mounted load cells will likely need to be replaced before any structural members, and should also be tested periodically to ensure results fidelity.

DVPR

The design verification plan and report will involve testing the machine's ability to achieve the specified requirements. The requirements that will require testing include quiet operation, drivetrain/linkage life, variable frequency, and stroke variability.

Quiet operation will be influenced mainly by the motor and linkage noise. A servo motor and backlash free gear box should produce a minimal amount of noise. Since the exact system deflection and joint tolerance will not be determined until the machine is complete and several tests run, an unknown amount of noise will be produced by the joints. Design and manufacturing considerations have been made to stiffen these constructs and eliminate as much slop as possible. This should reduce the component wear and limit 'clicking' caused by backlash.

Drive train and linkage life tests will be conducted in parallel with the initial trampoline tests. Throughout each test, the components with finite life will be inspected for excessive wear and the components with expected infinite life will be checked for signs of fatigue. These inspections will be conducted over 500,000 cycles throughout an expected 2 million cycle test. In the event of excessive wear or unexpected fatigue, redesign will be conducted to improve the machine life.

Variable speed and frequency verification will also be conducted in parallel with the initial trampoline tests. These will involve tuning the controller to operate the trampoline at a range of strokes and frequencies. The impactor overshoot must be selected such that the system does not jerk as the mechanism kinematics force the reciprocating mass to change direction.

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Appendices

Appendix A: Engineering requirement definitions (HOWS/WHATS)

Adjustable frequency: In order to run the test in nominal and higher frequencies, the machine must have adjustable cycle speed.

Short Test Time: In order to turn over a significant number of tests, the machine must be able to run accelerated life tests.

Position Control: The machine must be capable of repeatedly and accurately actuating the specimen to a desired position.

Valid Data Reduction: The data acquisition system must be able to reduce noise during testing and report enough data for future reference.

Valid Dynamic Measurement: In order to accurately measure force and deflection, the machine must compensate for errors native to recording force in a continuously moving system.

Valid Static Measurement: The machine should be able to perform quasi-static extension tests (i.e. at slow speeds).

Temperature Control/Monitoring: In order to comply with ASTM standards of testing, tests must be conducted in a temperature-controlled environment, within two degrees of the desired temperature.

Efficient Operation/Setup: The machine must not require an excessive amount of setup time.

Safe Operation: The machine must not endanger the health or life of the technician.

True Strain Measurement: The data acquisition system must be capable of measuring the true strain of the device, if necessary.

Position Measurement/Capacity: The current position of the machine must be accurately logged for the entire stroke along with any force data; also, the machine must be capable of holding the entire range of JumpSport's specimens.

Force measurement and recording: The current force must be accurately logged over the entire stroke and only reduced during subsequent analysis.

Failure Determination: In order to establish the fatigue life of the test specimens, the machine should be able to relay information regarding failures, and to detect failure.

Valid Accelerated Life: Conditions must be established under which long term performance of specimens can be extrapolated from shorter-term tests.

Overload Protection: In the event of failure or improper installation resulting in an overload, the machine must be capable of shutting down automatically.

Infinite Machine Life: The components used to create the machine should have infinite life compared to the life of the specimens.

Serviceable: The machine must be constructed such that parts can be easily replaced or interchanged.

Current Solutions (NOWS)

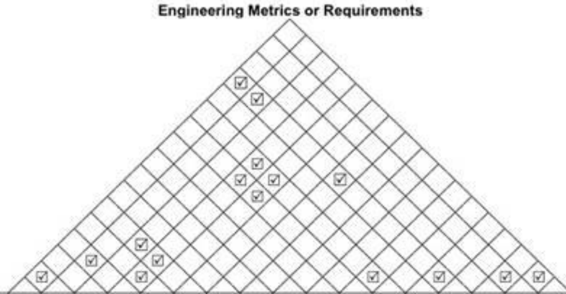
Purchasing a fatigue-test machine (Instron or similar product): This alternative offers significantly higher reliability but also cost. The commercial fatigue-test machines may struggle to produce rates exceeding ~Hz; however, they offer results much faster than designing a new machine.

Use the Cal Poly test machine: The Cal Poly test machine offers almost instant results and access over the entire senior project period. However, the test speed is limited to ~1Hz and JumpSport would like to own the machine.

The current JumpSport machine: This machine provides some necessary data but has not been verified and suffers from long test times and excessive noise.

Appendix B: QFD Matrix

Engineering Metrics or Requirements



Customer Attributes, Needs, Requirements, or Demanded Quality	Relative Importance or Weight	Adjustable Frequency	Short Test Time	Position Control	Valid Data Reduction	Valid Dynamic Measurement	Valid Static Measurement	Temperature Control/Monitoring	Efficient Operation/Setup	Safe Operation	True Strain Measurement	Position Measurement/Capacity	Force Measurement (Sample Rate)	Failure Determination	Valid Accelerated Life	Overload Protection	~Infinite Machine Life	Serviceable	Now: Purchase Instron	Now: Use CP Machines	Now: First Generation JumpSport
Quiet Operation/Minimal Vibration	7	3	1																5	5	1
Low Cost	8	9	3	3				1			3	3	3		3		9	1	1	5	5
Adjustable Stroke	11	9	3	3		1	1								9				5	5	3
Adjustable Capacity	11								3			1							4	5	4
Multiple Specimens	5	1	9		1	1	1		1		1	1	3		1		1		2	1	2
Accurate Force Measurement	8				3	9	9						9	1		3			5	5	2
Export to CSV/Excel	6				1														5	5	5
Plugs Into 110V Outlet	6																		4	5	5
Implement Existing Load Cells	3				1	3	3						9	1		1			3	1	5
Complies with Test Protocol	11	3	3	3	3	3	3	3		3	1	3	3	3	3				5	5	1
Benchtop Size	4	1	1					1										1	3	5	4
Frequency > 1Hz	9	9	9			1									9				3	1	1
Feedback from Load Cell for Realtime Control	11	3	1	9	1	9						3			1				5	5	1
Measurement Units		Hz	hours	+/- in	[illegible]	+/-lbs	+/-lbs	+/-F	hours		+/- in/in	in	lbs Hz		hours [illegible]	lbs max	machine life		50	53	39
Difficulty		2	5	2	2	2	2	3	2	1	3	3	4	4	5	2	3	1			
Target Values		0-3	<168	0.03	100	0.005	0.005	2	1		0.05	0-12	100 300		0.3	100+					
Assignment (IN-X, CP-O, JS-+) 1		+	+	+	+	+	+	+		+	+	+			+	+					
2			O						+					+	O		+	+			
3		O						X O	O		X		+								
4			X											X O	X						
5		X		X O	X O	X O	X O		X	X O	O	X O	X O			X O	X O	X O			
Absolute		348	238	189	82	238	130	45	38	33	40	106	171	44	253	27	77	12			
Relative		8	7	7	8	7	7	3	3	4	3	7	7	7	6	4	8	4			
Rank		1	3	5	9	3	7	11	14	15	13	8	6	12	2	16	10	17			

Figure 16: QFD

Appendix C: Decision Matrix

Customer Requirements	Weight	Slider Crank	Rank	Ball Screw	Rank	Servo-Hydraulics	Rank
Quiet Operation (<60dB)	7	The motor is likely the loudest component in the slider crank mechanism. If a brushed or brushless DC motor were selected, the noise would be negligable; however, more research would be necessary if a stepper motor were selected.	3	Ball Screws running at 3000 RPM produce 58 - 62 dB. 60 dB is the maximum outdoor daytime decibel rating.	2	There's likely going to be a LOT of vibrations and kickback; repeatedly running and reversing hydraulic cylinder can be about equivalent to a moderate shouting noise. We'd likely be fighting to reduce that through the entire design process.	1
Low Cost (~$2500)	8	The motor and drive components are likely the most expensive part of the system as the other components would be manufacutred by the team.	3	Ball screws have proven to be of high cost. One consideration may be designing a ball screw actuation system by choosing a motor, transmission and ball screw.	2	Here, you get what you pay for. We'd have a bigger budget sinice we'd be combined with the other team, but in general we'd still be looking at the lower end of the price scale. Still, a hydraulic system will likely cost somewhere between a slider crank and a ball screw, with the potential for costing more than the ball screw.	1
Adjustable Stroke (0-10 inches)	11	By varying the crank length either by replacing the crank or providing an adjustable link, the stroke could be varied when the test is stopped.	1	Ball screws have an inherent amount of adjustability. To ensure adjustability, choosing an actuator with more than the required stroke would be necessary.	3	The machine's stroke would be easily adjustable; within whatever range it's designed to, it should be able to hit any stroke with somewhere around 1/10th inch resolution/accuracy	2
Adjustable Capacity	11	The capacity could be adjusted by positioning the fixed end of the fixture along the axis.	1	The capacity will be accomodated within the oversized stroke of the ballscrew.	1	The machine's stroke would be oversized to allow for adjustable capacity.	1
Multiple Specimens	5	Multiple specimens could be attached to the sliding carriage or multiple fixtures could be produced to run simultaneus tests.	1	Fixtures could be designed for both ends of the machine with the capacity for multiple specimens.	1	The carriage could be designed so that multiple specimens can be loaded into the machine for a single test.	1
Accurate Force Measurement	8	Load cells on the static end could be installed to accurately measure the force both dynamically and in quasi-static conditions.	1	The current load cells could be implemented within the fixturing at the static end of the machine.	1	A controlled hydraulic system can be extremely precise (though perhaps less so at higher frequencies) so with a good control system it should be pssible to accurately log force and displacement.	1
Export to CSV/Excel	6	The data acqusition system can be setup to export data directly to a CSV file.	1	Data will be exported directly to a CSV file using the data acquisition system	1	The DAQ could export data directly to to a CSV file.	1
Plugs into 110V Outlet	6	The motor and drive components can be selected to accept 110V power.	1	Motors are available in both 110 V as well as 220 V. A motor capable of producing the force, speed and stroke needed will use 110 V.	1	The hydraulic system could be set to accept power from a 110v outlet, though a more relevant concern would be the overall electric properties/requirements of the system, such as power distribution, etcetera.	1
Implement Existing Load Cells	3	The non-cycling end of the fixture can be adaped to accept the existing load cells. Software will need to be developed to integrate the load cells and motor control systems.	1	The existing load cells could be placed at the stationary end of the machine.	1	The load cells we currently have would be affixed to the stationary end of the fixture, so this should not be an issue.	1
Complies with Test Protocol	11	The slider crank is capable of the three major tests: quasi static extension, standard life test, accelerated life tests.	2	The ball screw is capable of producing the following three valuble tests: quasi static extension, standard life tests and accelerated life tests.	3	The hydraulic system is capable of the three major tests: static extension, standard life, and accelerated life.	1
Benchtop Size	4	The slider crank can be designed similiar to other benchtop sized cyclic testing machines.	3	A system could be designed in the vertical position to allow for a smaller footprint. The machine will be able to fit in the confines of a room within a house and or shed.	2	The system can be designed to fit within the necessary confines: in this case, Jumpsport's insulated shed. The design can be oriented horizontally or vertically, depending on what the space restrictions necessitate.	1
Frequency (~2 Hz)	9	The input crank speed can be adjusted to achieve any speed within the motor's capability.	3	Ball screws have the capability of performing in the range of 0.001 Hz to 15 Hz. Higher frequencies most often require shorter strokes, but in the 1-3 Hz range a stroke within the requirements is feasible.	2	Hydraulics are generally built to handle lower frequencies at higher strokes (or high frequencies at low stroke) but the system could incorporate several other elements to enable delivery the necessary frequency and stroke.	1
Feedback from Load Cell for Realtime Control	11	The load cells can be interfaced to the controller to assist in determining failures and adjusting the preload to maintain a constant force test.	1	The control system can be implemented in such a way that the force is kept constant during the test. If the force changes, load cells will relay information to the driver which will cause the ball screw to adjust stroke thus increasing or decreasing force.	3	With a PID controller, the load cells could be implemented in a feedback system that maintains the desired output.	2
Engineering Requirements							
Adjustable frequency: In order to run the test in nominal and higher frequencies, the machine must have adjustable cycle speed.	348	The slider-crank mechanism can be driven by a constantly rotating motor at virtually any speed.	3	The ball screw actuator can be driven with a stepper motor which would be driven with a control system to produce adjustable frequency within the machine.	2	Adjustability should be easily accomplished, though running the apparatus at a sufficiently high frequency will require a bit of controls design to prevent overshoot (even a tiny overshoot adds up over time) and most likely will require a secondary lever arm to enable the machine to achieve sufficiently large stroke.	1
Short Test Time: In order to turn over a significant number of tests, the machine must be able to run accelerated tests.	238	The slider-crank mechanism can be driven at various frequencies and the stroke can be adjusted to shorten the test time.	3	A ball screw actuator system can be driven at various frequencies and the stroke can be adjusted to shorten test time.	2	Any machine that can achieve the necessary stroke and load condtions can carry out an accelerated life test.	1
Position Control: The machine must be capable of repeatedly actuating the specimen to a desired position.	189	Various sensors on either the slider or motor shaft can be used to monitor and control the position. Using certain stepper motors, the system can be moved and stopped incrementally. Using brushed/blushless motors, the system can be continuously driven and stopped with limited precision.	1	A PID will be chosen and programmed according to thew test specifications where force and position will be able to be controlled. The system will be modeled analytically before the PID is fully implemented.	1	Using PID Control, we could probably control position accurately, with only a small bit of overshoot and settle time. Everything could be easily configured with software, meaning little to no need for mechanical changes to the hardware.	1
Valid Data Reduction: The data acquisition system must be able to reduce noise during testing and report enough data for future reference.	82	The valid data reduction is an aspect of the data acquisition system that will be implemented in software following the mechanical design. Hardware will be selected to limit noise in the collections and processing steps.	1	The valid data reduction is an aspect of the data acquisition system that will be implemented in software following the mechanical design.	1	Because of the nature of the system, there will most likely be a lot of noise that the software of the DAQ would need to reduce, though the hardware will still be designed with the intent of minimizing this noise.	1
Valid Dynamic Measurement: In order to accurately measure force and deflection, the machine must compensate for errors native to recording force in a continuously moving system.	238	The valid dynamic measurement is an aspect of the data acquisition system that will be implemented in software following the mechanical design.	1	Valid dynamic measurement will be employed through the DAQ following the mechanical design of the machine.	1	Similar to valid data reduction; however, because of the controls on the machine, the system could be engineered (with some clever use of PID control) to compensate for native errors.	1
Valid Static Measurement: The machine should be able to form extension tests, either at pseudo-static conditions i.e. slow speeds.	130	The valid static measurement may be implemented through a pseudo-static or static means depending on the mechanical drive selected for the slider-crank.	1	Valid Static measurement will involve the correct design of the controller.	2	The machine will be able to produce sufficient stroke to perform the static extension tests, as well as having software protocols in place in order to correctly perform the test.	2
Temperature Control/Monitoring: In order to comply with ASTM standards of testing, tests must be conducted in a temperature-controlled environment, within two degrees of the desired temperature.	45	The temperature control is outside of the mechanical fixture.	1	The ball screw system will operate in nominal conditions. The temperature should be kept within plus or minus 2 degrees during testing although this is outside the scope of the test machine. A thermostat must be set inside the testing room.	1	Hydraulics would have no problem operating in a temperature-controlled environment, and should not produce too much heat (efficiency upwards of 80%).	1
Efficient Operation/Setup	38	The fixturing mechanism, stroke adjustment, and frequency adjustment can be designed to efficiently setup tests.	1	The fixturing mechanism, stroke adjustment, and frequency adjustment can be designed to efficiently setup tests.	1	The machine will likely require minimal operator iinput. Generally, only selecting the relevant test parameters via software, and affixing the test specimens would be necessary. However, software operation may require some initial setup, depending on how the UI is designed.	1
Safe Operation	33	Rapid moving parts including the crank arm and connecting rod can be enclosed to avoid user contact. Other safety features can be implemented in the mechanical and software design to prevent user error and harm.	1	The machine could be designed with an enclosure to protect the technician from rotating parts and/or a failure of the testing material or the machine.	1	The machine will be designed with covers or enclosurse over moving parts and pinch points, as well as labeled with clear safety warnings. The machine will be designed so that even if the test specimens fail catastrophically, it will not result in a risk of harm.	1
True Strain Measurement: The data acquisition system must be capable of measuring the true strain of the device, if necessary.	40	The true strain measurement device would be implemented on the specimen following the design of the drive mechanism.	1	The true strain measurement device would be implemented on the specimen following the design of the control system.	1	The design of the machine likely would not affect the DAQ's ability to handle this.	1
Position Measurement/Capacity: The current position of the machine must be accurately logged for the entire stroke along with any force data; also, the machine must be capable of holding all of JumpSport's specimens.	106	The position could be measured indirectly with a rotary encoder or directly with a linear encoder. Using a rotary encoder, the position could be determined regardless of the stroke or frequency.	1	The position could be measured indirectly with a rotary encoder or directly with a linear encoder. Using a rotary encoder, the position could be determined regardless of the stroke or frequency.	1	A linear encoder or string potentiometer could be used to measure position directly for any range of stroke or frequency.	1
Force Measurement Capacity: The force measurement devices must be capable of accurately reading forces in the ranges prescribed by the material and test conditions.	171	The slider crank mechanism can be designed to withstand the maximum forces associated with the cords stretched to maximum strain. The measurement and recording strategy will be implemented with the software development. The precision of the drive mechanism will need to permit enough position resolution to capture the cord stiffness.	1	With correct implementation of the load cells and control system, force measurement will be read accurately within the limits of the load cell. The force and position will be measured simultaneously in increments that produce viable data. This will involve the precise movement of the ball screw as well as implementation of the load cells. Ball screws are capable of precise movement as long as backlash has been limited.	1	With a good control system and sensor in place, hydraulics can be extremely precise. The speed of response necessitated by the high frequency at which the machine will be run will reduce this accurately somewhat, but not significantly.	1
Failure Determination: In order to establish the fatigue life of the test specimens, the machine should be able to relay information regarding failures.	44	Software can be developed to using the quasi-static extension test to determine when a cord has failed.	1	Within the DAQ, a failure determination will be implemented in order to effectively relay information rgarding failure.	1	If the machine can accurately measure force and position, then so long as failure criteria are loaded into the system, the apparatus should have no trouble detecting failure and reporting the conditions under which it occurred.	1
Valid Accelerated Life: Conditions must be established under which long term performance of specimens can be extrapolated from shorter-term tests.	253	The testing parameters, including frequency and stroke, will be variable such that valid accelerated life data can be gathered.	3	The machine will be capable of variable stroke and frequency which will be crucial to developing accelerated life tests.	2	So long as accelerated life conditions can be established, the machine should be able to easily run accelerated life tests at a higher strain rate, so long as the system is correctly designed.	1
Overload Protection: In the event of failure or improper installation resulting in an overload, the machine must be capable of shutting down automatically.	27	Software features can be implements to detect and stop in overloaded situations.	1	The machine can be designed to recognize an overload wherein the motor and or power will shut off.	1	Again, so long as the machine can calculate failure accurately, it should be able to deactivate upon failure detection in order to prevent overload.	1
Infinite Machine Life: The components used to create the machine should have infinite life compared to the life of the specimens.	77	The links of the slider crank can be designed to have infinite life; however, the bearings will likely have finite life that should be on the order of 100x the test duration.	3	The ball screws machine life is limited by the balls within the races although the use of a linear guide system might increase the life. Ball screw actuators are avilable on the market with high finite lifes, but the cost may prove to be outside the scope of the budget.	2	Having multiple components means more potential points of failure. Figures vary on hydraulic machine life but seem to be dependent on individual products. Infinite life, seems unlikely. Leaky valves and all sorts of problems can cause failure in the in the range of products we can obtain, but this is still most likely a by-the-product deal. Additionally, running the machine at such high frequencies may end up reducing life significantly, since they're designed for high loads applied slowly, not really high speed applications.	1
Serviceable: The machine must be constructed such that parts can be easily replaced or interchanged.	12	The bearings will likely need servicing prior to any other component as they will be designed to withstand ~100 test cycles. The mechanism can be designed such that bearing service is simple	1	The ball screw will likely need servicing before any other components as it will be designed to withstand ~ 80 test cycles. It would be difficult to design the system such that ball screw service is simple.	1	The machine should be made for serviceability regardless of its method of actuation, though because of their complex nature, servicing hydraulics may be difficult.	1

Appendix D: Gantt Chart

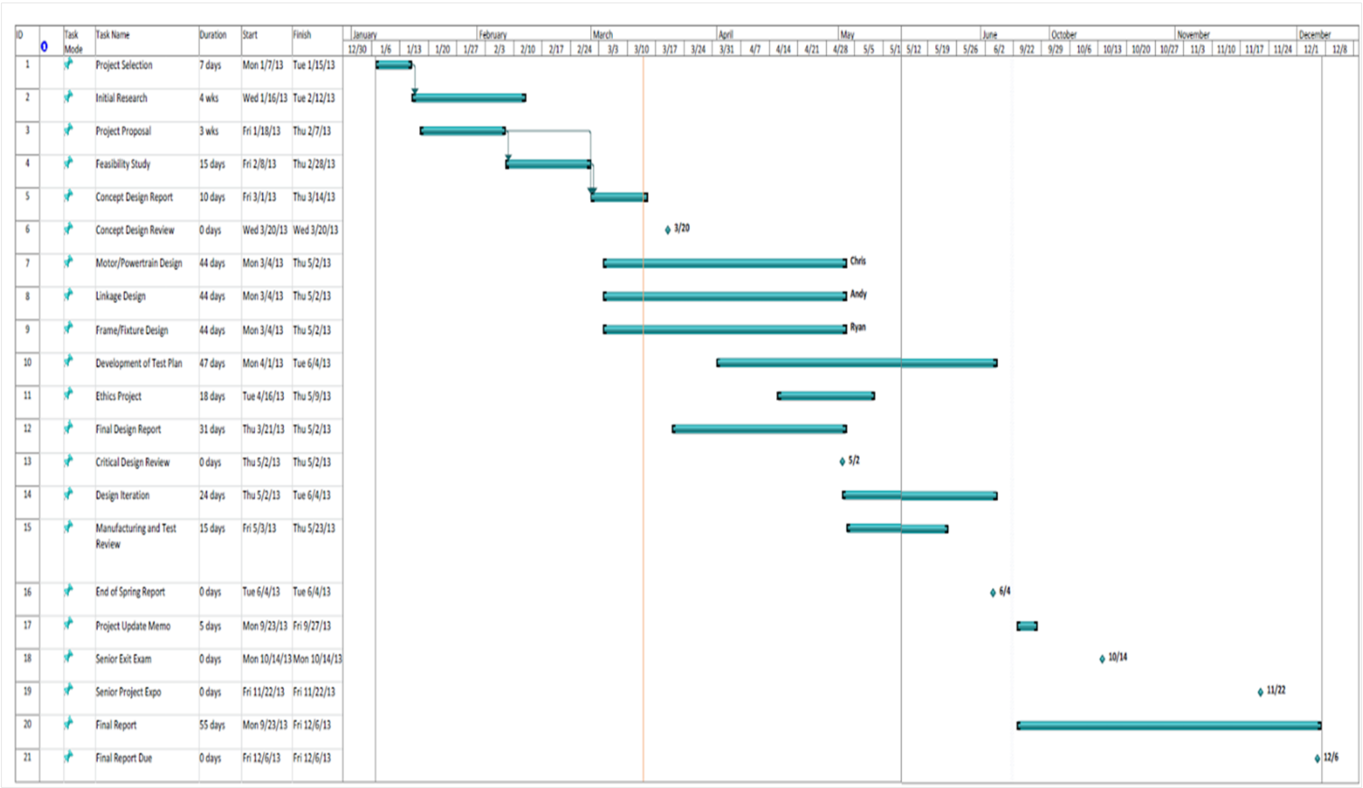


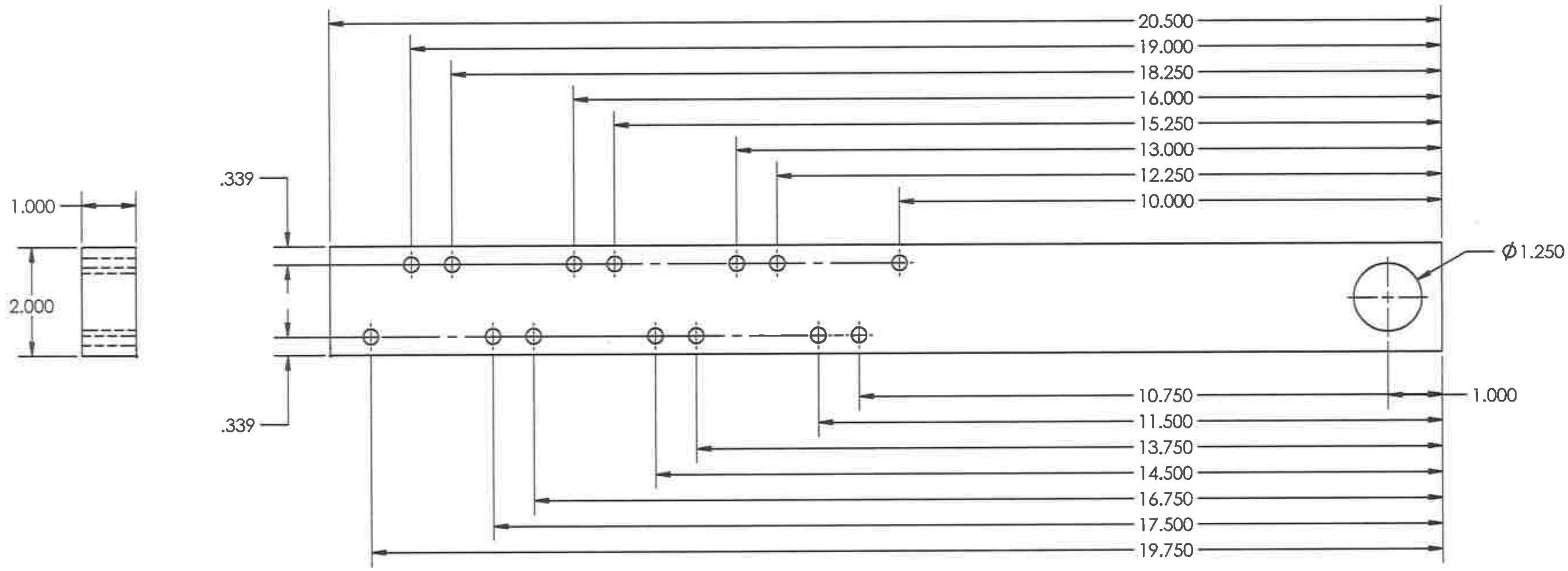
Figure 17: Gantt Chart

Appendix E: Sample Lengths

Table 1: Sample Lengths

JumpSport Bungee Lengths			
	Length [in]	Wrapped Length [in]	Length at Max Strain [in]
8 mm 2knot	19.75	4.9375	14.8125
8 mm 3knot	19.75	4.9375	14.8125
8 mm 4knot	19.5	4.875	14.625
10 mm 4 knot	19	4.75	14.25
Spring	8.625	N/A	N/A

Appendix F: Linkage Drawings and Supporting Analysis



PROPRIETARY AND CONFIDENTIAL
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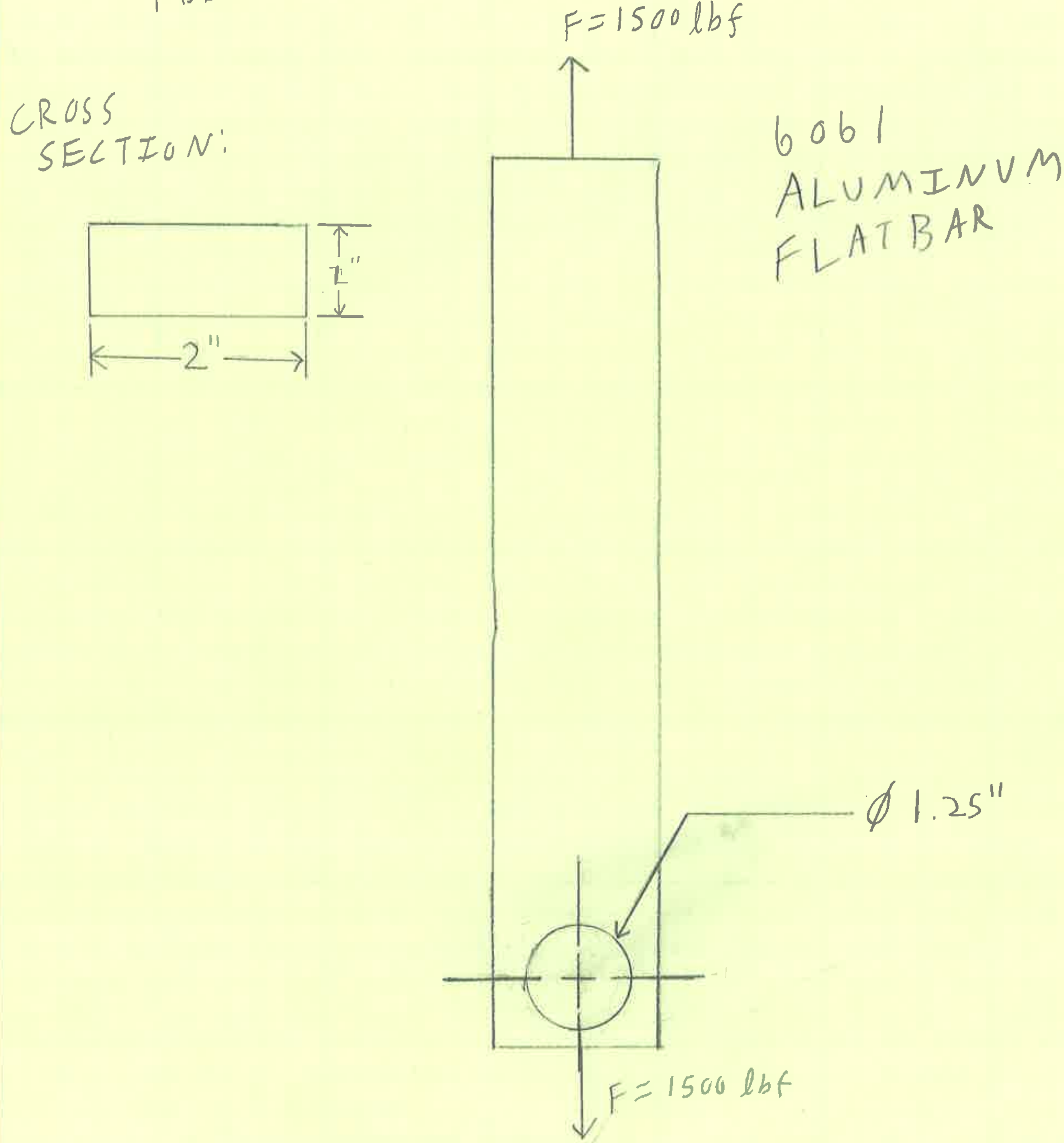
UNLESS OTHERWISE SPECIFIED:		NAME	DATE
DIMENSIONS ARE IN INCHES TOLERANCES: FRACTIONAL ± ANGULAR: MACH ± BEND ± TWO PLACE DECIMAL ± THREE PLACE DECIMAL ±	DRAWN	A. BROCK	5/10/2013
	CHECKED		
	ENG APPR.		
	MFG APPR.		
INTERPRET GEOMETRIC TOLERANCING PER:	Q.A.		
MATERIAL 6061 ALUMINUM	COMMENTS:		
FINISH			
NEXT ASSY / USED ON — APPLICATION	DO NOT SCALE DRAWING		

TITLE: CONNECTING ROD

SIZE	DWG. NO.	REV
B	2	A
SCALE: 1:2	WEIGHT: ? lbs	SHEET 1 OF 1

CONNECTING ROD

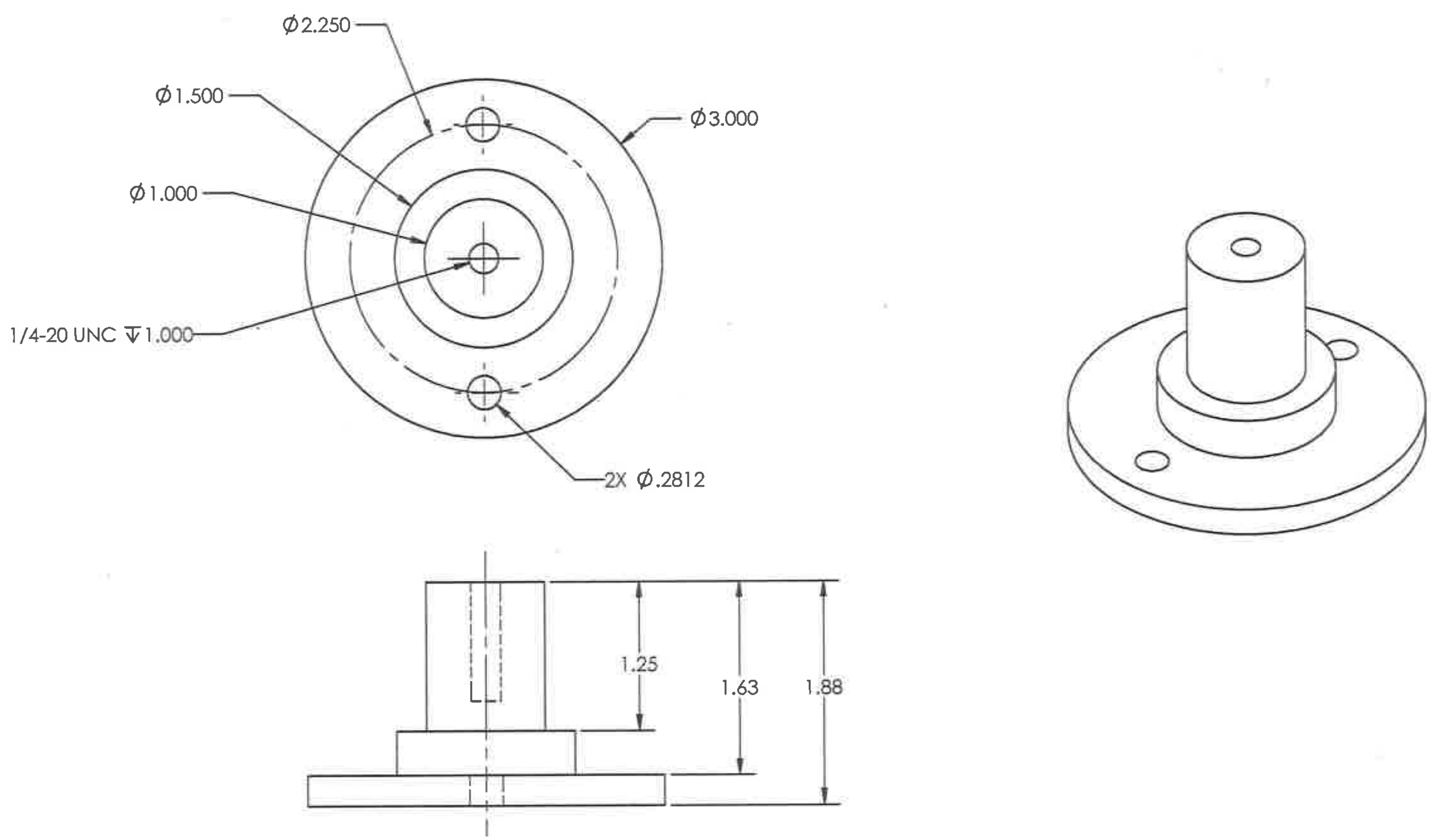
FBD



AXIAL STRESS AT THE HOLE:

$$\sigma = \frac{F}{A} = \frac{1500\ lbf}{(2\,in - 1.25\,in)(1\,in)} = 2000\ psi\ \text{TENSION}$$

FOR 6061 ALUMINUM AT 2000 PSI REVERSED TENSION, THE S-N DIAGRAM INDICATES AT LEAST $5 \cdot 10^8$ CYCLES FATIGUE LIFE, WHICH SATISFIES THE REQUISITE FATIGUE CRITERION.



PROPRIETARY AND CONFIDENTIAL

LINKAGE ASSY	
NEXT ASSY	USED ON
APPLICATION	

UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN INCHES
TOLERANCES: ±0.005
FRACTIONAL ±
ANGULAR: MACH ± BEND ±
TWO PLACE DECIMAL ±
THREE PLACE DECIMAL ±

INTERPRET GEOMETRIC TOLERANCING PER:

MATERIAL CARBON STEEL

FINISH

DO NOT SCALE DRAWING

	NAME	DATE
DRAWN	E. FLORY	5/2/13
CHECKED		
ENG APPR.		
MFG APPR.		
Q.A.		

COMMENTS:

TITLE:

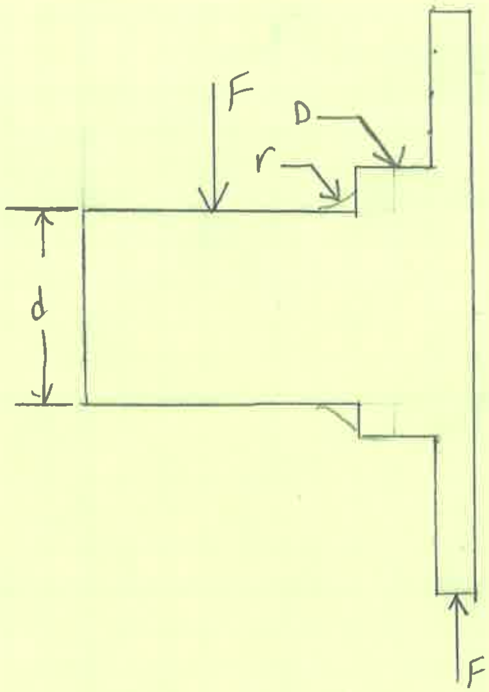
CONNECTING STUD FOR FLYWHEEL

SIZE	DWG. NO.	REV
B	41	A

SCALE: 1:1 WEIGHT: 0.94 lbs SHEET 1 OF 1

CONNECTING STUD

FBD



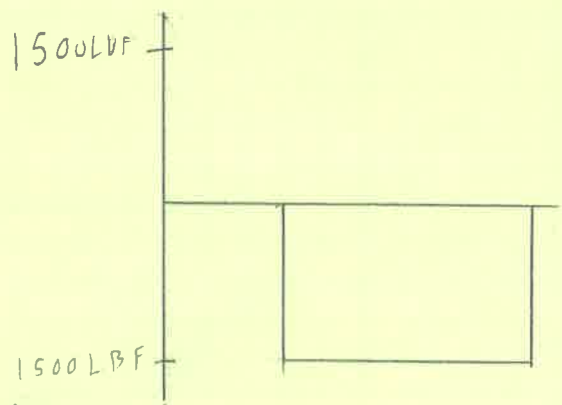
$F = 1500$ LBF

$R = \frac{1}{16}$ IN

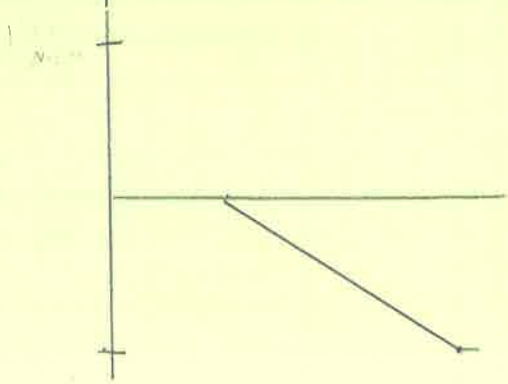
$D = 1.5$ IN

$d = 1$ IN

SHEAR
DIAGRAM



BENDING
DIAGRAM



CONNECTING STUD

THE CRITICAL STRESS OCCURS AT THE FILLET DUE TO LOCAL STRESS CONCENTRATION.

THE NOMINAL BENDING STRESS DUE TO THE APPLIED FORCE, F, IS

$$\sigma_m = \frac{Mc}{I}$$

$M = 1\text{in} \cdot 1500\,\text{lbf}$ (F IS APPLIED 1in FROM FILLET)

$c = 0.5\,\text{in}$

$$I = \frac{\pi (1\text{in})^4}{64} = 0.0491\ \text{in}^4$$

$$\sigma_m = \frac{1500\ \text{in}\cdot\text{lbf} \cdot 0.5\ \text{in}}{0.0491\ \text{in}^4}$$

$\sigma_m = 15279\ \text{psi}$

STRESS CONCENTRATION:

$K_t = f(r, d, D)$ [SHIGLEY]

$\frac{r}{D} = 0.0625$

$\frac{D}{d} = 1.25$

$K_t = 1.74$

NOTCH SENSITIVITY:

$q = f(r, S_{UT})$

$S_{UT} = 69\ \text{Ksi}$

$q = 0.70$

$K_f = 1 + q(K_t - 1) = 1.518$

$\sigma = K_f \sigma_m = 23193\ \text{psi}$

CONNECTING STUD

FATIGUE LIFE

$S_{UT} = 69$ KSi

$S'_e = \frac{1}{2} S_{UT} = 34.5$

$S_e = K_a K_b K_c K_d K_e K_f S'_e$

MARIN FACTORS

K_a: SURFACE FACTOR

$K_a = a\, S_{UT}^{\ b}$

MACHINED SURFACE: $a = 2.70$, $b = -0.265$

$K_a = 2.70 \quad ^{-0.265} = 0.879$

K_b: SIZE FACTOR

$K_b = 0.879\, d^{-0.107}$

$K_b = 0.879\,(1)^{-0.107} = 0.879$

K_c: LOADING FACTOR

BENDING: $K_c = 1$

K_d: TEMPERATURE FACTOR

ROOM TEMPERATURE: $K_d = 1$

K_e: RELIABILITY FACTOR

$K_e = 1 - 0.08 z_a$

RELIABILITY: 99 % $\qquad z_a$: 2.326

$K_e = 0.81$[illegible]

K_f: MISC FACTORS

$K_f = 1$

CONNECTING STUD

ENDURANCE STRENGTH, MODIFIED:

$$S_e = (0.879)(0.879)(1)(1)(0.814)(1)(34.5)$$

$S_e = 21.7$ ksi

FATIGUE FACTOR OF SAFETY

MOD. GOODMAN GIVES

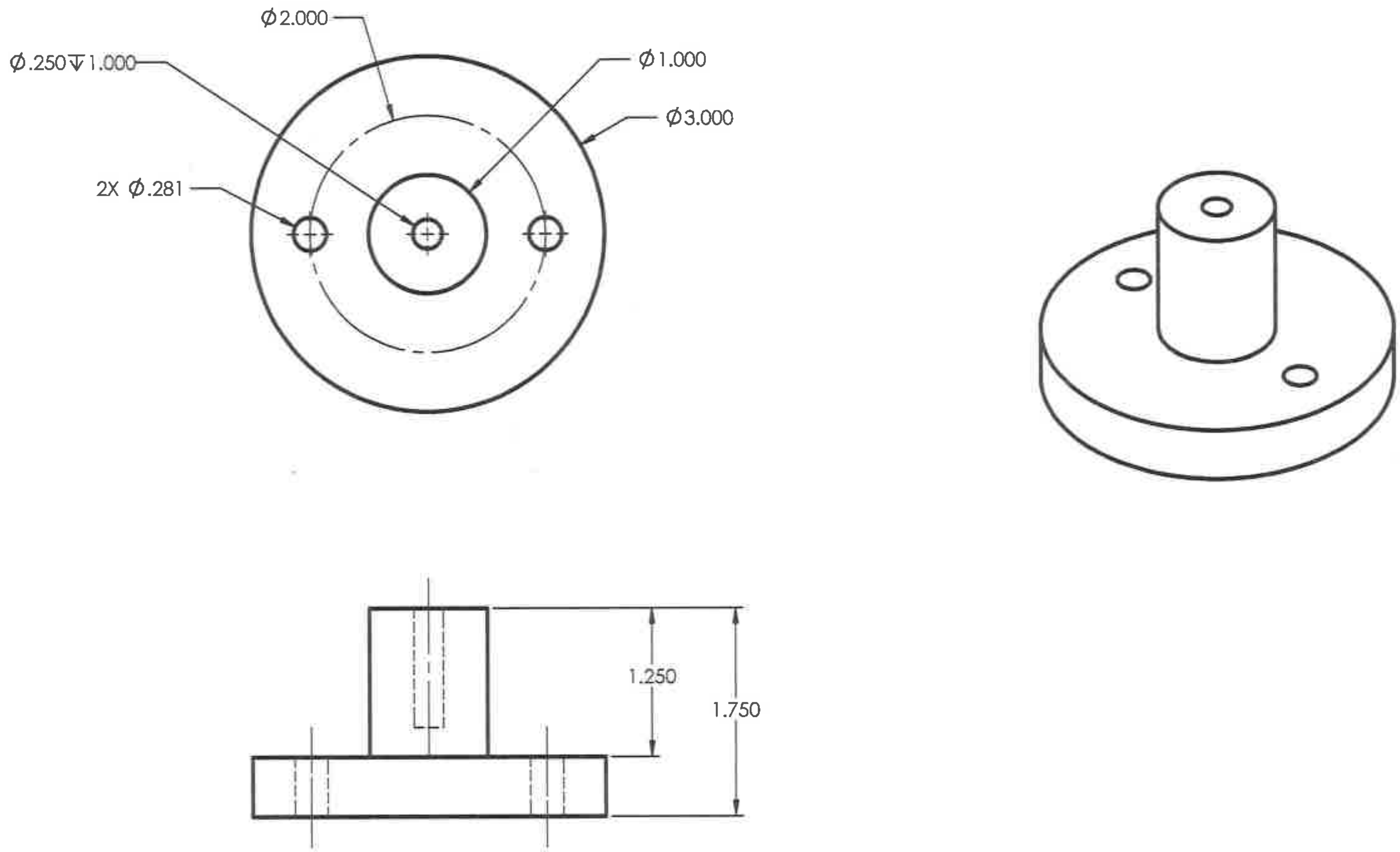
$$n_f = \frac{1}{\frac{\sigma_a}{S_e} + \frac{\sigma_m}{S_{UT}}}$$

STRESS CYCLES FROM 0 TO σ;

$$\sigma_a = \sigma_m = \frac{\sigma}{2} = \frac{1}{2}(23.193 \text{ ksi}) = 11.597 \text{ ksi}$$

$$n_f = \frac{1}{\frac{11.1 \text{ ksi}}{21.7 \text{ ksi}} + \frac{11.6 \text{ ksi}}{69 \text{ ksi}}}$$

$$n_f = 1.42$$



LINKAGE ASSY		
NEXT ASSY	USED ON	
APPLICATION		

UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN INCHES
TOLERANCES: ±0.005
FRACTIONAL ±
ANGULAR: MACH ± BEND ±
TWO PLACE DECIMAL ±
THREE PLACE DECIMAL ±
INTERPRET GEOMETRIC TOLERANCING PER:
MATERIAL CARBON STEEL
FINISH
DO NOT SCALE DRAWING

	NAME	DATE
DRAWN	E. FLORY	5/2/13
CHECKED		
ENG APPR.		
MFG APPR.		
Q.A.		
COMMENTS:		

TITLE:
CONNECTING STUD FOR SLIDER CRANK

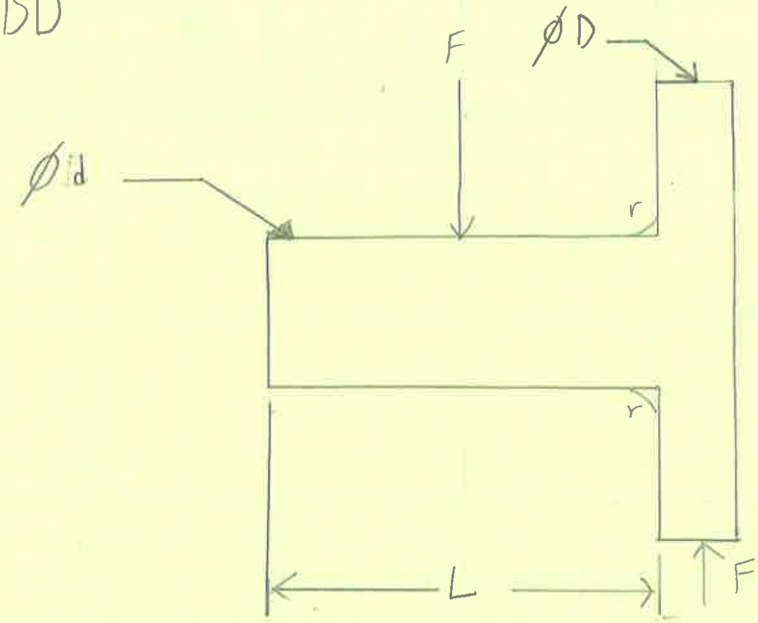
SIZE	DWG. NO.	REV
B	42	A
SCALE: 1:1	WEIGHT: 1.24 lbs	SHEET 1 OF 1

CONNECTING STUD 2

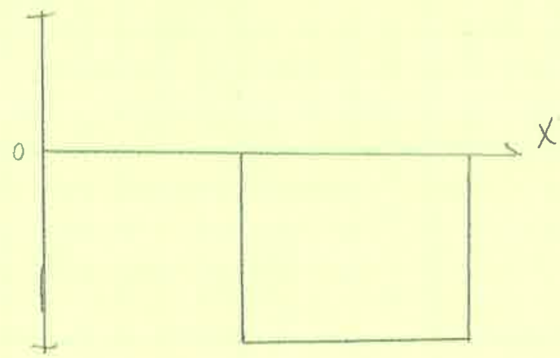
FATIGUE ANALYSIS

$D = 2.875$ IN
$d = 1.25$ IN
$L = 2.25$ IN
$F = 1500$ LBF
$r = \frac{1}{16}$ IN

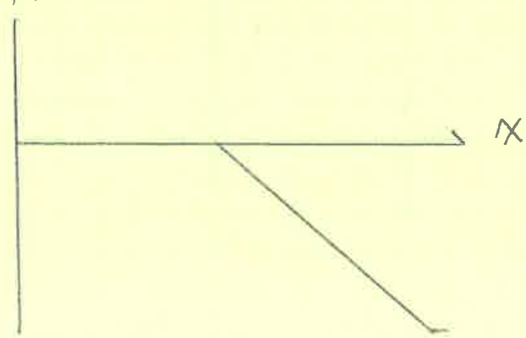
FBD



SHEAR DIAGRAM



BENDING DIAGRAM



CONNECTING STUD 2

THE CRITICAL STRESS OCCURS AT THE FILLET.

NOMINAL BENDING STRESS:

$$\sigma_m = \frac{Mc}{I}$$

$M = 1500$ in.lbf

$c = 0.5$ in

$$I = \frac{\pi (1\,in)^4}{64} = 0.0491\ in^4$$

$$\sigma_m = \frac{in.lbf \cdot 0.5\ in}{0.0491\ in^4}$$

$\sigma_m = 15279$ psi

STRESS CONCENTRATION:

$K_t = f(r, d, D)$ (SHIGLEY)

$\frac{r}{d} = .0625$

$\frac{D}{d} = 2.875$

$K_t = 2.0$

NOTCH SENSITIVITY:

$q = f(r, S_{UT})$

$S_{UT} = 69$

$q = 0.7$

$K_f = 1 + q(K_t - 1) =$ 1.[illegible]

$\sigma = K_f\ \sigma_m = 25974.3$ psi

CONNECTING STUD 2

ENDURANCE STRENGTH, MODIFIED:

$S_e =$ THE SAME AS FOR STUD 1

$S_e = 21.7$ ksi

FATIGUE FACTOR OF SAFETY:

MOD. GOODMAN GIVES

$$n_f = \frac{1}{\frac{\sigma_a}{S_e} + \frac{\sigma_m}{S_{UT}}}$$

STRESS CYCLES FROM 0 TO σ

$$\sigma_a = \sigma_m = \frac{\sigma}{2} = \frac{25.974 \text{ ksi}}{2} = 12.99 \text{ ksi}$$

$$n_f = \frac{1}{\frac{[illegible] \text{ ksi}}{21.7 \text{ ksi}} + \frac{13 \text{ ksi}}{69 \text{ ksi}}}$$

$$n_f = 1.27$$

BUSHING

THE LIFE OF THE BUSHING, IN CYCLES

CYCLES = Nt

$N = 120 \frac{rev}{min}$

$t = \frac{\pi D L \omega}{4 f_1 f_2 K V F}$

$D = 1$ in
$L = 1$ in
ω = WEAR ALLOWANCE =

P = PRESSURE = $\frac{F}{DL} = \frac{1500\, lbf}{1 in \cdot 1 in} = 1500\, psi$

$V = \frac{\pi D N}{12} = \frac{\pi \cdot 1 in \cdot 120 RPM}{12} = 31.4 \frac{in}{min} = 2.62 \frac{ft}{min}$

f_1, FROM SHIGLEY: $f_1 = 1.5$

f_2: NO FOREIGN MATTER, AMBIENT $T < 140°F$
$f_2 = 1.0$

WEAR FACTOR, K: FOR OILITE
$K = 3 \cdot 10^{-10}$

$t = \frac{\pi \cdot 1 in \cdot 1 in \cdot .00\ in}{4 \cdot 1.5 \cdot 1.0 \cdot 3 \cdot 10^{-10} \cdot 2.62 \frac{ft}{min} \cdot 1500\, lbf}$

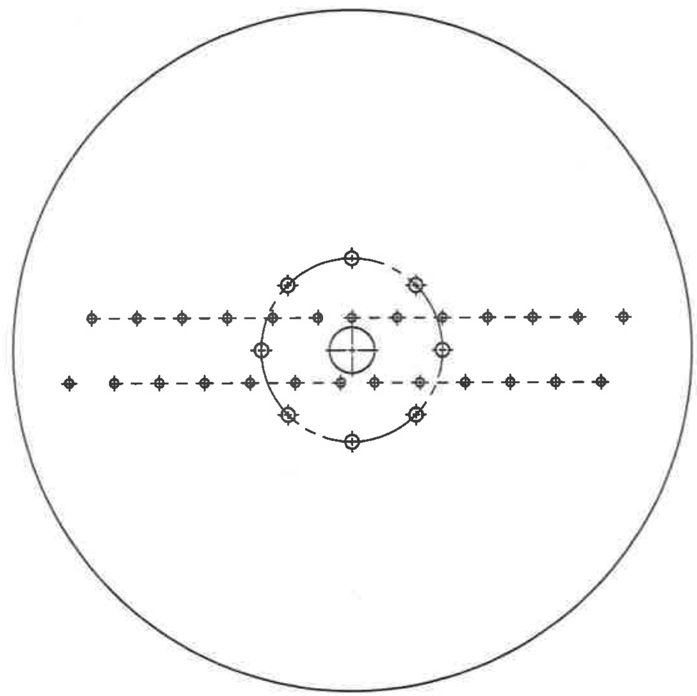
$t =$

CYCLES = $120 \frac{rev}{min} \cdot$

CYCLES =

$PV_{LIMITING} = 18000 \frac{psi \cdot ft}{min}$

$PV = 1500\, psi \cdot 2.62 \frac{ft}{min} = 3930 \frac{psi \cdot ft}{min} < 18000 \frac{psi \cdot ft}{min}$ ✓



NOTE: PART TO BE CUT BY WATERJET
USING EXPORTED DWG FILE.

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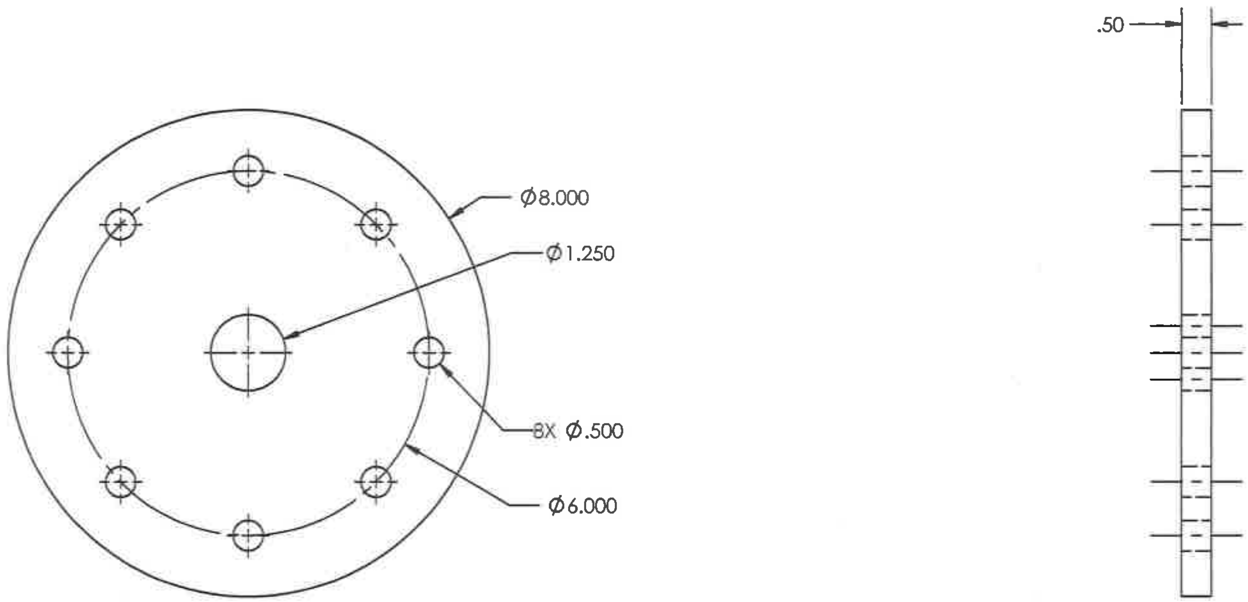
DIMENSIONS ARE IN INCHES TOLERANCES: FRACTIONAL± ANGULAR: MACH± BEND ± TWO PLACE DECIMAL ± THREE PLACE DECIMAL ±	DRAWN	E.FLORY	5/10/13	TITLE: FLYWHEEL
	CHECKED			
	ENG APPR.			
	MFG APPR.			
INTERPRET GEOMETRIC TOLERANCING PER:	Q.A.			
MATERIAL CARBON STEEL	COMMENTS:			SIZE B · DWG. NO. · REV A
FINISH				
DO NOT SCALE DRAWING				SCALE: 1:4 · WEIGHT: 111 LB · SHEET 1 OF 1

NEXT ASSY	USED ON
APPLICATION	

MECHANICAL ENGINEERING CAL POLY



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

INTERPRET GEOMETRIC TOLERANCING PER:

MATERIAL CARBON STEEL

FINISH

LINKAGE ASSY

NEXT ASSY | USED ON

APPLICATION

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	NAME	DATE
DRAWN	E. FLORY	5/2/13
CHECKED		
ENG APPR.		
MFG APPR.		
Q.A.		

COMMENTS:

TITLE:

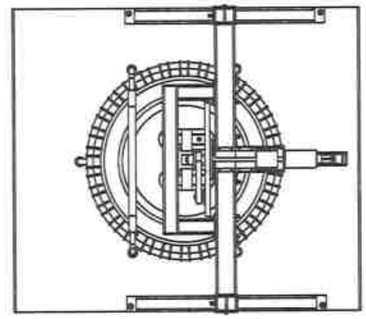
FLYWHEEL FLANGE HUB

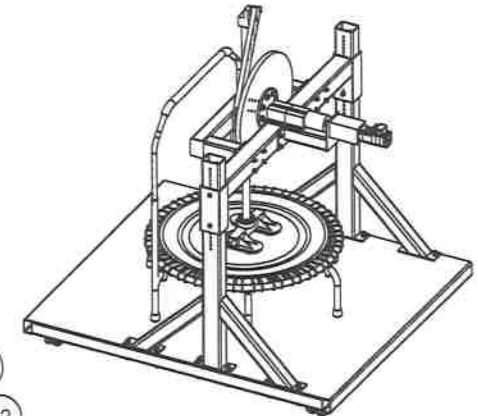
SIZE B | DWG. NO. 44 | REV A

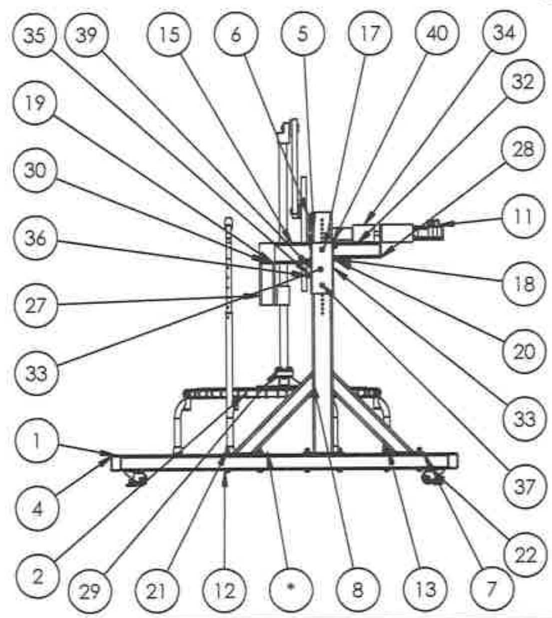
SCALE: 1:2 | WEIGHT: 6.69 lbs | SHEET 1 OF 1

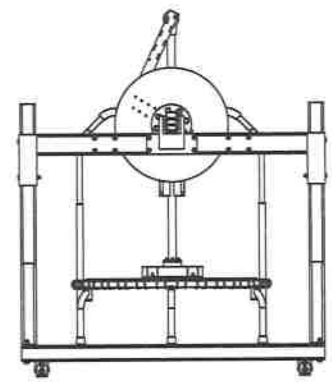
Appendix G: Frame Drawings and Supporting Analysis

ITEM NO.	PART NUMBER	QTY.
1	Frame_Base	1
2	Adult Tramp Asm	1
3	Caster Mount	4
4	Frame_Floor	1
5	Frame_Base_Upright	2
6	linkageASSMvirt	1
7	Flange_Mount	6
8	Base_Gusset	4
9	Gantry	1
10	Floor_Brace	2
11	MotorGearReducer	1
12	98023A033	24
13	91257A719	12
14	Floor_Cross	1
15	pillow_togantry	2
16	pillow_togantry_mount	2
17	98180A130	30
18	91257A644	12
19	pillow_arm	1
20	97135A230	12
21	97135A250	12
22	9854T51	4
23	90126A029	32
24	92620A540	16
25	93827A211	16
26	Floor_Support	2
27	pillow_support	1
28	motor_mount	1
29	Impactor Assy_pancake load cell_short center beam bolt	1
30	cover_plate	2
31	motor_love_togantry	1
32	motor_love	1
33	gantrySlider	2
34	Flexible_Spider_Shaft_Coupling	1
35	93827A225	8
36	92620A624	4
37	91257A651	4
38	Gantry_Slider_Outer	3
39	Gantry_Slider_Front	2
40	Gantry_Slider_Back	2









NEXT ASSY	USED ON	
APPLICATION		DO NOT SCALE DRAWING

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DIMENSIONS ARE IN INCHES
TOLERANCES:
FRACTIONAL ±
ANGULAR: MACH ± BEND ±
TWO PLACE DECIMAL ±
THREE PLACE DECIMAL ±
INTERPRET GEOMETRIC TOLERANCING PER:
MATERIAL
MATL
FINISH

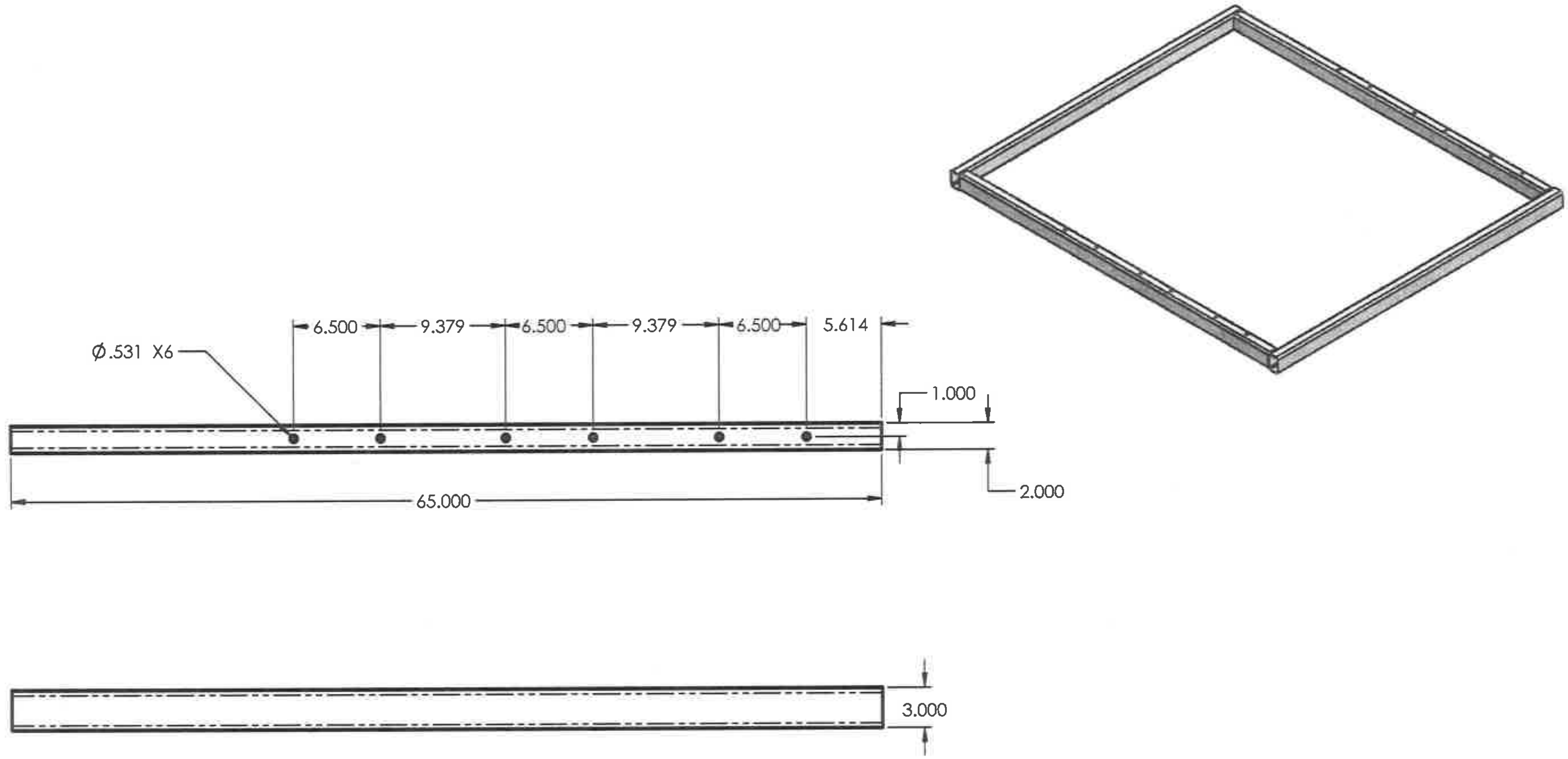
	NAME	DATE
DRAWN	WR	5/9/13
CHECKED		
ENG APPR.		
MFG APPR.		
Q.A.		

COMMENTS:

TITLE:
Frame

SIZE B | DWG. NO. | REV A

SCALE: 1:24 | WEIGHT: ? lbs | SHEET 1 OF 1



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

FINISH

NEXT ASSY | USED ON
APPLICATION | DO NOT SCALE DRAWING

	NAME	DATE
DRAWN	WR	5/9/13
CHECKED		
ENG APPR.		
MFG APPR.		
Q.A.		

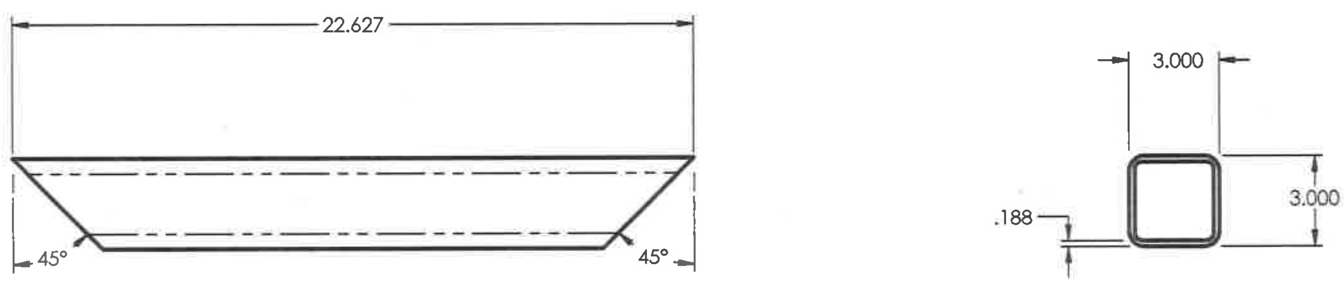
COMMENTS:

TITLE:

Frame_Base

SIZE B | DWG. NO. | REV

SCALE: 1:8 | WEIGHT: | SHEET 1 OF 1



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

INTERPRET GEOMETRIC TOLERANCING PER:
MATERIAL
FINISH

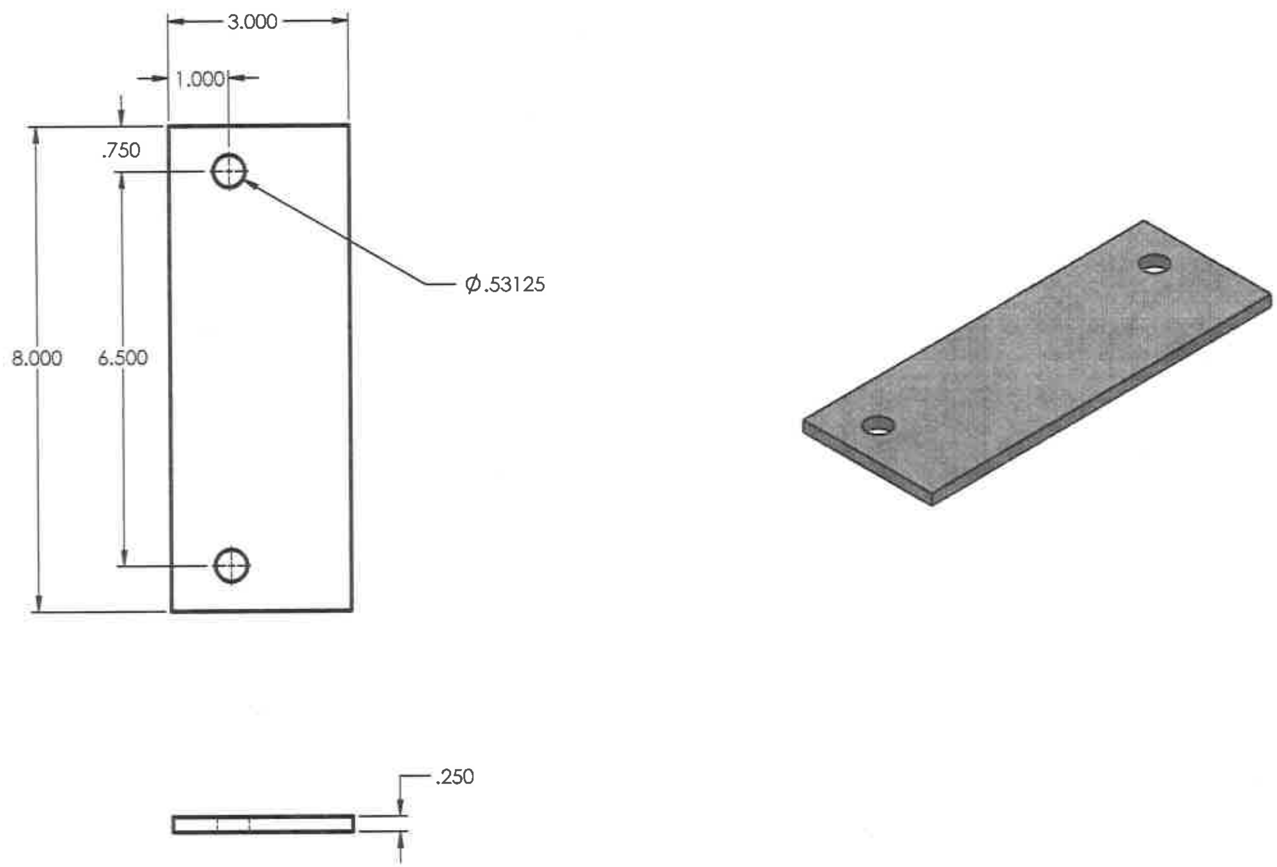
NEXT ASSY | USED ON
APPLICATION | DO NOT SCALE DRAWING

	NAME	DATE
DRAWN	R.MURPHY	5/7/2013
CHECKED		
ENG APPR.		
MFG APPR.		
Q.A.		

COMMENTS:

TITLE:
Base_Gusset

SIZE B | DWG. NO. | REV
SCALE: 1:4 | WEIGHT: | SHEET 1 OF 1



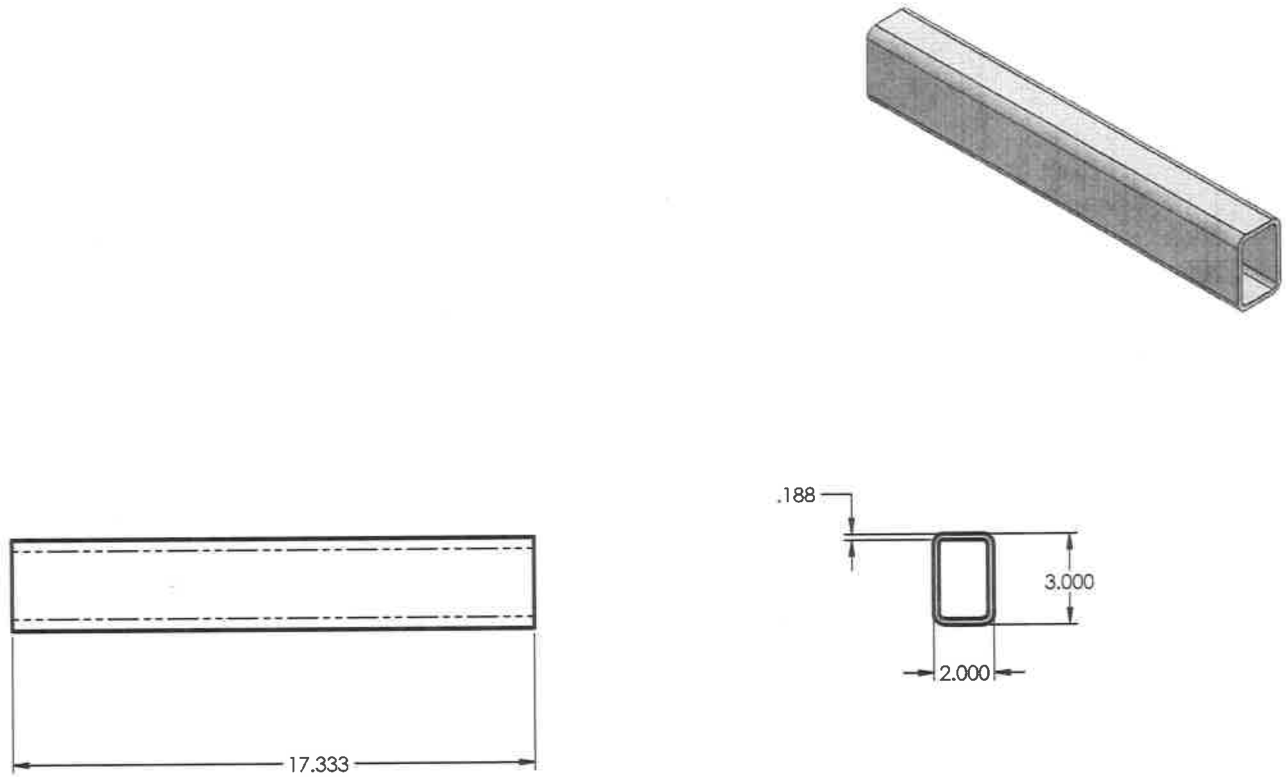
NOTE: Holes are clearnace holes
sized at 17/32" for a 1/2" bolt

UNLESS OTHERWISE SPECIFIED:	NAME	DATE
DIMENSIONS ARE IN INCHES TOLERANCES: ±0.005 FRACTIONAL ± ANGULAR: MACH ± BEND ± TWO PLACE DECIMAL ± THREE PLACE DECIMAL ±	DRAWN	4/28/13
	CHECKED	
	ENG APPR.	
	MFG APPR.	
INTERPRET GEOMETRIC TOLERANCING PER:	Q.A.	
MATERIAL	COMMENTS:	
FINISH		
NEXT ASSY / USED ON / APPLICATION	DO NOT SCALE DRAWING	

TITLE:

Flange_Mount

SIZE	DWG. NO.	REV
B		
SCALE: 1:2	WEIGHT:	SHEET 1 OF 1

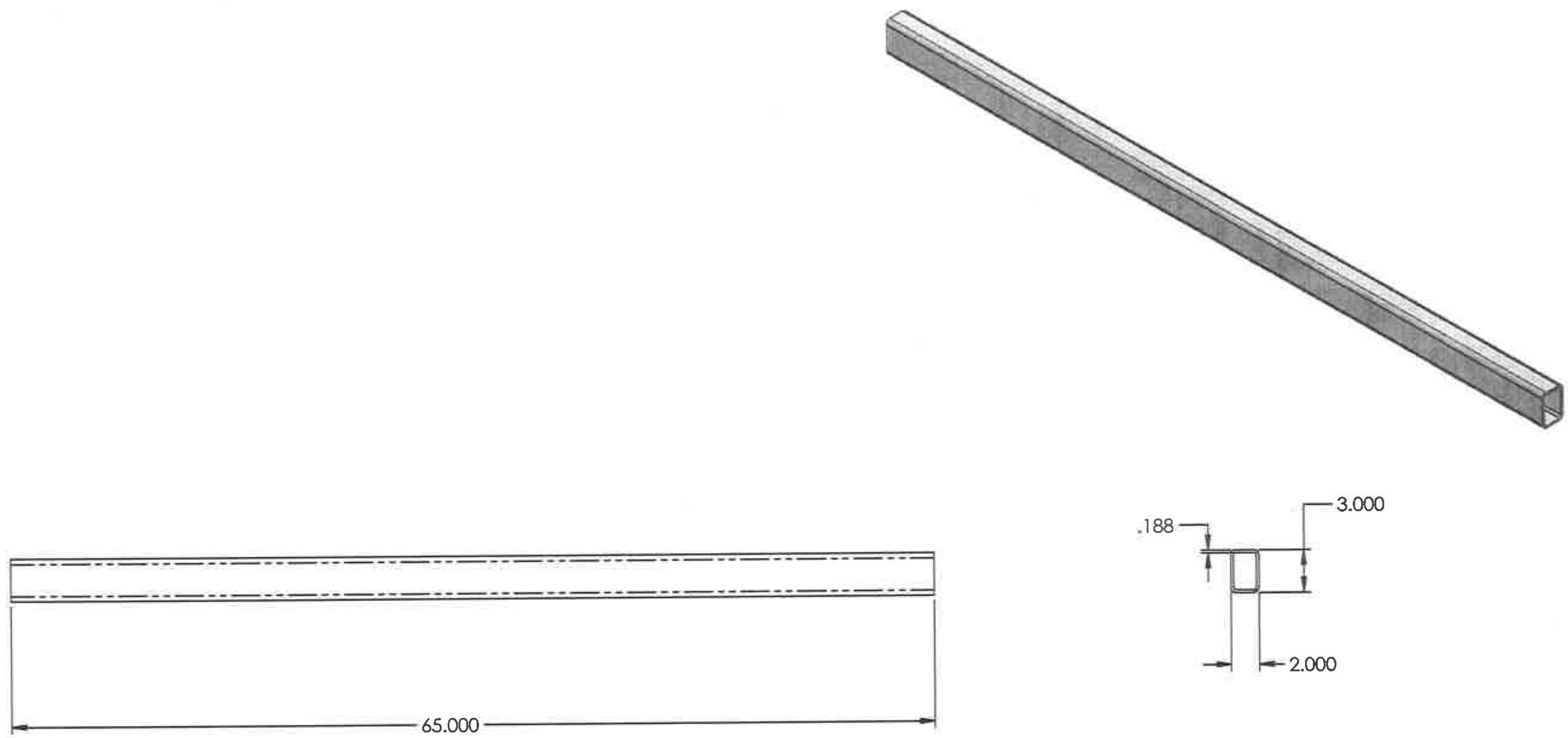


		UNLESS OTHERWISE SPECIFIED:		NAME	DATE
		DIMENSIONS ARE IN INCHES TOLERANCES:±0.010 FRACTIONAL ± ANGULAR: MACH ± BEND ± TWO PLACE DECIMAL ± THREE PLACE DECIMAL ±	DRAWN	R.MURPHY	5/7/2013
			CHECKED		
			ENG APPR.		
			MFG APPR.		
		INTERPRET GEOMETRIC TOLERANCING PER:	Q.A.		
		MATERIAL	COMMENTS:		
NEXT ASSY	USED ON	FINISH			
APPLICATION		DO NOT SCALE DRAWING			

TITLE:

Floor_Support

SIZE	DWG. NO.	REV
B		
SCALE: 1:4	WEIGHT:	SHEET 1 OF 1



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
FINISH
DO NOT SCALE DRAWING

NEXT ASSY | USED ON
APPLICATION

	NAME	DATE
DRAWN	R.MURPHY	5/7/2013
CHECKED		
ENG APPR.		
MFG APPR.		
Q.A.		

COMMENTS:

TITLE:

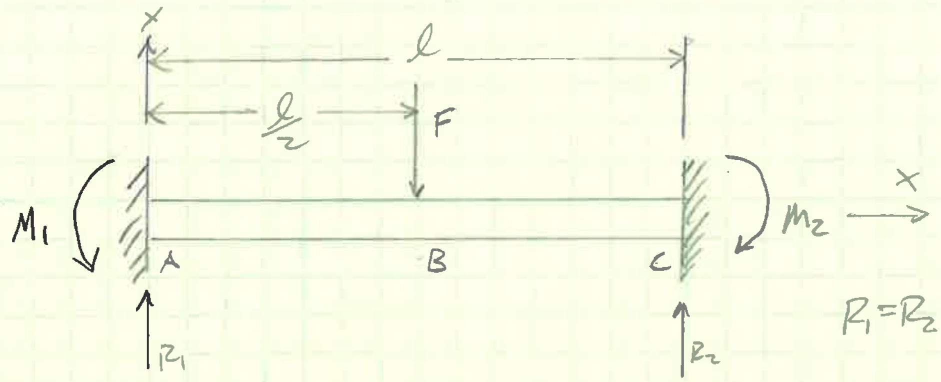
Floor_Brace

SIZE B | DWG. NO. | REV

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

MECHANICAL CAL POLY ENGINEERING

FLOOR BRACE HAND CALCULATIONS



$R_1 = R_2 = \frac{F}{2}$

$M_1 = M_2 = \frac{Fl}{8}$

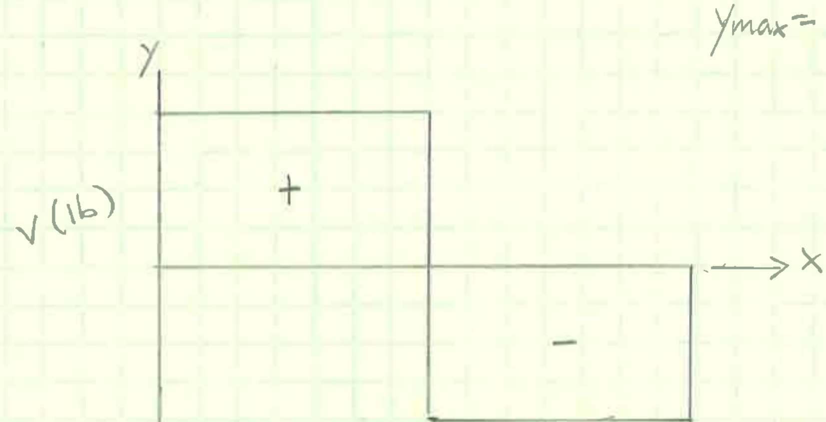
$F = 750\ lb$
$l = 65\ in$
$E = 30(10)^6\ psi$
$I = 1.857\ in^4$

$V_{AB} = -V_{BC} = \frac{F}{2}$

$M_{AB} = \frac{F}{8}(4x - l) \quad M_{BC} = \frac{F}{8}(3l - 4x)$

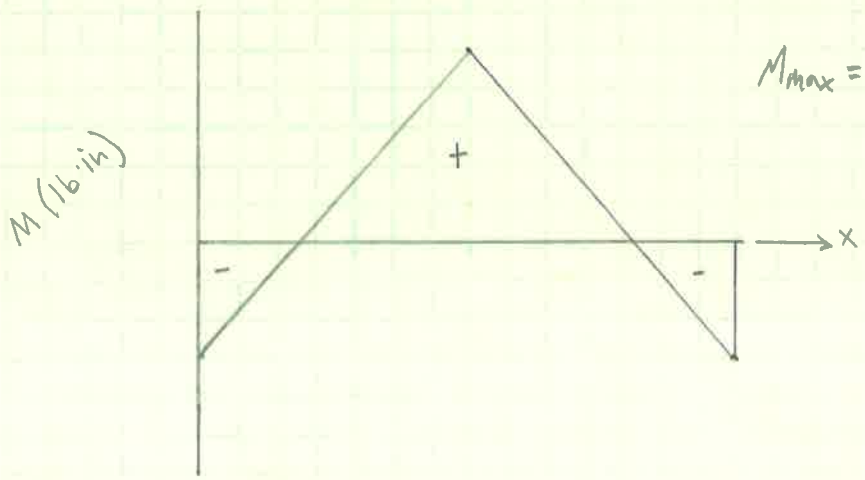
$y_{AB} = Fx^2(4x - 3l)$

$y_{max} = \frac{-Fl^3}{192EI}$



SHIGLEY'S TA-9-13
8th ED.

$M_{max} = \frac{Fl}{4}$



FLOOR BRACE HAND CALCS CONT.

MOD-GOODMAN FATIGUE

$\sigma_a = 4920$

$\sigma_m = 4920$

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

$S_e' = 0.5\, S_{ut}$

$S_e' = 0.5(57.99\ \text{kpsi})$

$S_e' = 28.996$

$k_a = a\, S_{ut}^{-b}$

$a = 14.4$ TABLE 6-2 SHIGLEY'S (HOT ROLLED)

$b = -0.718$ " " " "

$K_a = 14.4\,(57.99)^{-0.718}$

$k_a = 0.7803$

$k_b = 0.879\, d^{-0.107}$ $0.11 \le d \le 2\ in$ 6-20 SHIGLEYS

$k_b = 0.91\, d^{-0.157}$ $2 < d \le 10\ in$

$d_e = 0.808\sqrt{hb}$ TABLE 6-3 SHIGLEY'S

$d_e = 0.808\sqrt{3(2)\ in^2}$

$d_e = 1.979$

$k_b = 0.879(1.979)^{-0.107}$

$k_b = 0.817$

FLOOR BRACE HAND CALCULATIONS CONT.

$$M_{max} = \frac{F}{4}\left(4\left(\frac{\ell}{2}\right) - \ell\right)$$
$$= \frac{F\ell}{4}$$
$$= \frac{750\,lb\,(65\,in)}{4}$$

$$M_{max} = 12187.5\ lb\cdot in$$

$$y_{max} = \frac{-F\ell^3}{192EI}$$
$$= \frac{-750\,lb\,(65\,in)^3}{192(30\times10^6\,psi)(1.857\,in^4)}$$

$$y_{max} = -0.0193\ in.$$

$$\sigma_{max} = \frac{Mc}{I}$$
$$= \frac{12187.5\,lb\,in\,(1.5\,in)}{1.857\,in^4}$$

$$\sigma_{max} = 9840\ psi$$

YIELD

$$n_y > \frac{S_y}{\sigma_{max}}$$
$$> \frac{32000\,psi}{9840}$$
$$n_y > 1.625$$

MOD-GOODMAN FATIGUE

$$\left(\frac{\sigma_a'}{S_e} + \frac{\sigma_m'}{S_{ut}}\right) = \frac{1}{n}$$

$$\sigma_a' = \left(\sigma_a^2 + 3\tau_a^2\right)^{1/2}$$

$$\sigma_m' = \left(\sigma_m^2 + 3\tau_m^2\right)^{1/2}$$

$$\sigma_a = \frac{\sigma_{max} - \sigma_{min}}{2}$$

$$\sigma_m = \frac{\sigma_{max} + \sigma_{min}}{2}$$

FLOOR BRACE HAND CALCS CONT

$k_c = 1$ (bending) (6-26) SHIGLEYS

$k_d = 1$ ROOM TEMP

$k_e = 0.814$ 99% RELIABILITY TABLE 6-5 SHIGLEYS

$k_f = 1$ MISC

$S_e = (0.7803)(0.817)(1)(1)(0.814)(1)(28.996\,kpsi)$

$S_e = 15.05\,kpsi$

$$n = \left(\frac{\sigma_a'}{S_e} + \frac{\sigma_m'}{S_{ut}}\right)^{-1}$$

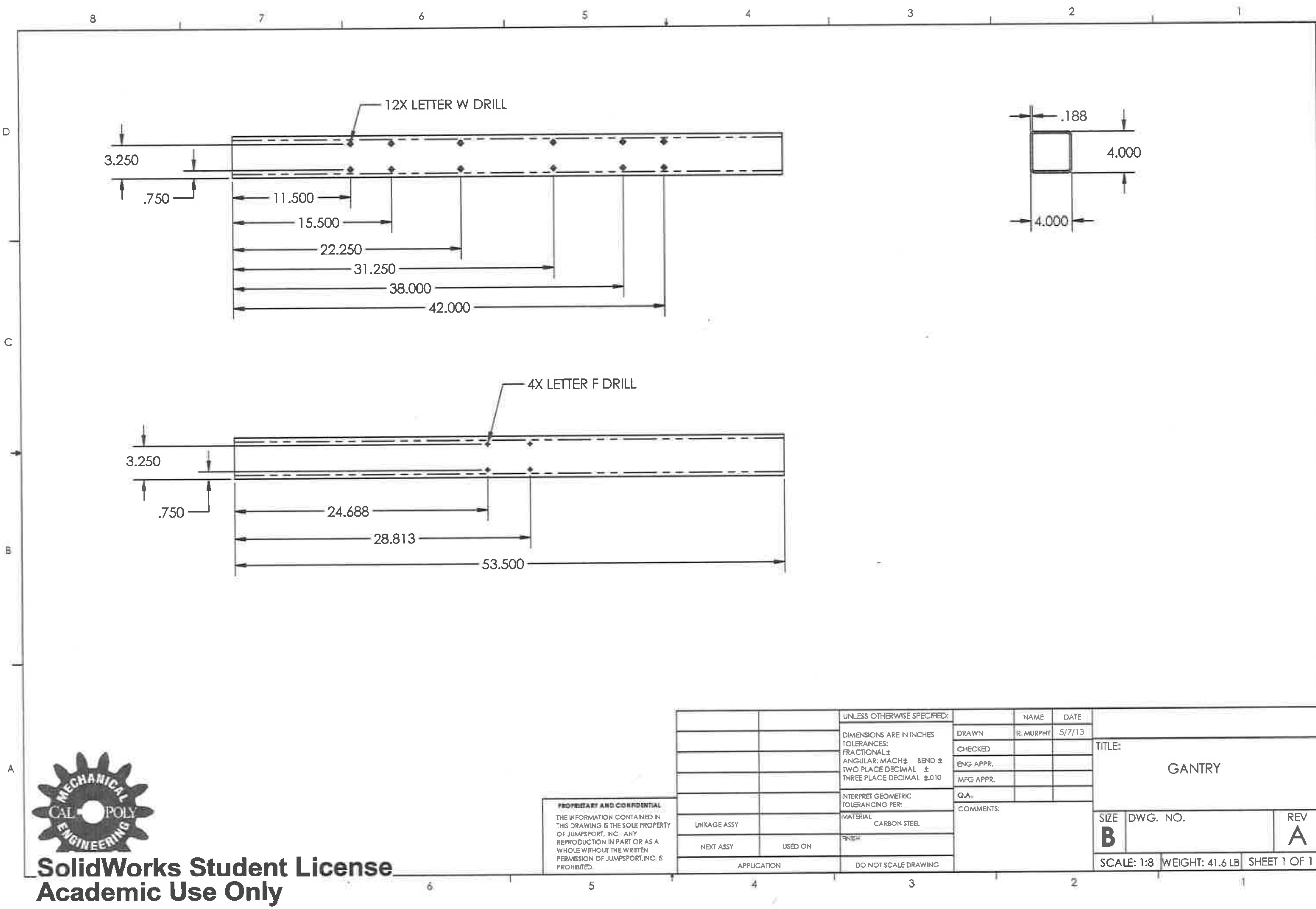
$$= \left(\frac{4920}{15050\,psi} + \frac{4920}{57990}\right)^{-1}$$

$n = 2.42$

ANALYSIS CONSTANTS

Max Force (lbs)	1500
Ult Str Steel (kpsi)	57.99

Description	Geometry			Moment of Inertia	Max Bending stress				Endurance Limit								SF
Floor Brace	L (in)	h (in)	b (in)	I (in^4)	c (in)	Moment (lb in)	Torque T(lb/in)	Max 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)	ymax (in)	
2"X3"X.120"	65	3.	2.	1.857	1.5	12187.5	0	9.84	28.995	0.637	1	1.979	0.818	0.817	15.11	-0.0193	2.4



12X LETTER W DRILL
3.250
.750
11.500
15.500
22.250
31.250
38.000
42.000
4X LETTER F DRILL
3.250
.750
24.688
28.813
53.500
.188
4.000
4.000
UNLESS OTHERWISE SPECIFIED:
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TOLERANCES:
FRACTIONAL ±
ANGULAR: MACH ± BEND ±
TWO PLACE DECIMAL ±
THREE PLACE DECIMAL ±.010
INTERPRET GEOMETRIC TOLERANCING PER:
MATERIAL CARBON STEEL
FINISH
DO NOT SCALE DRAWING
NAME DATE
DRAWN R. MURPHY 5/7/13
CHECKED
ENG APPR.
MFG APPR.
Q.A.
COMMENTS:
TITLE:
GANTRY
SIZE B
DWG. NO.
REV A
SCALE: 1:8
WEIGHT: 41.6 LB
SHEET 1 OF 1
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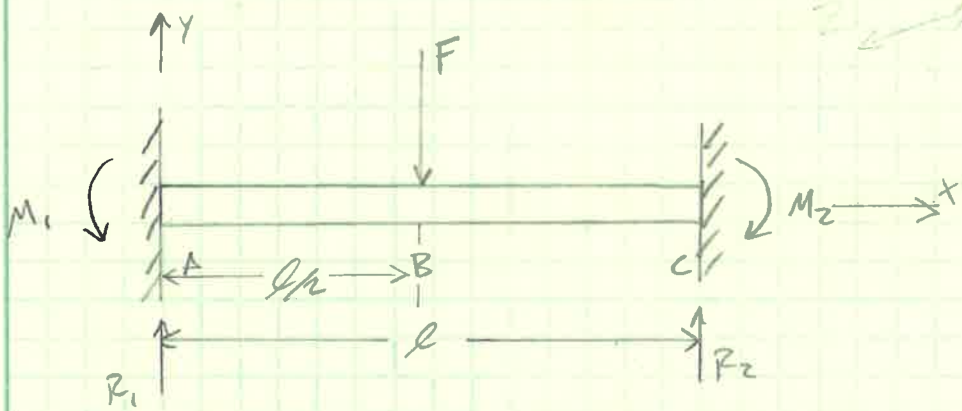
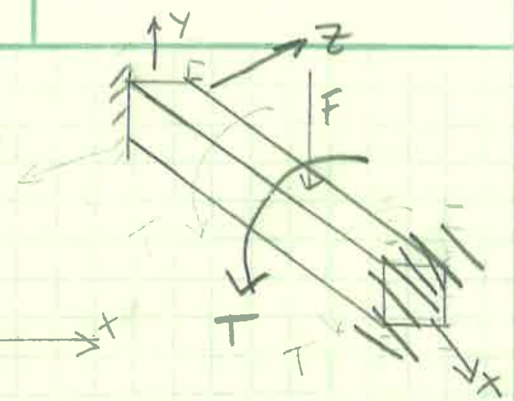
GANTRY HAND CALCULATIONS





$F = 1500\,lb$

$l = 53.5\,in$

$E = 30(10)^6\,psi$

$I = 6.960\,in^4$

$R_1 = R_2 = F/2$

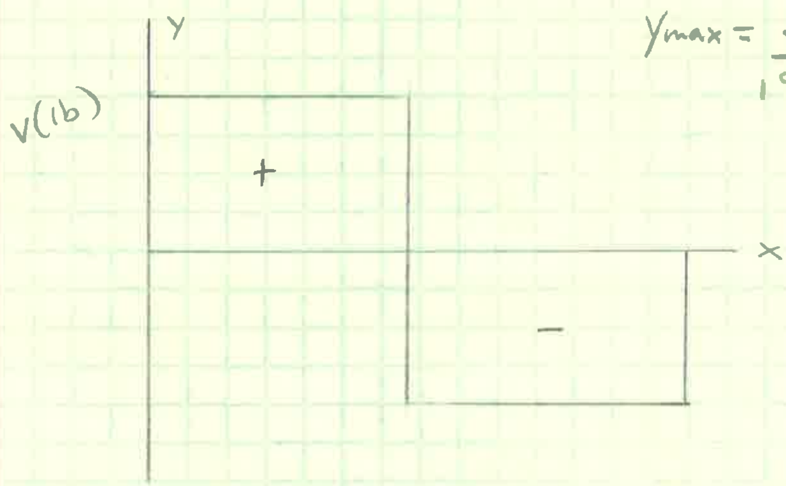
$M_1 = M_2 = Fl/8$

$V_{AB} = -V_{BC} = F/2$

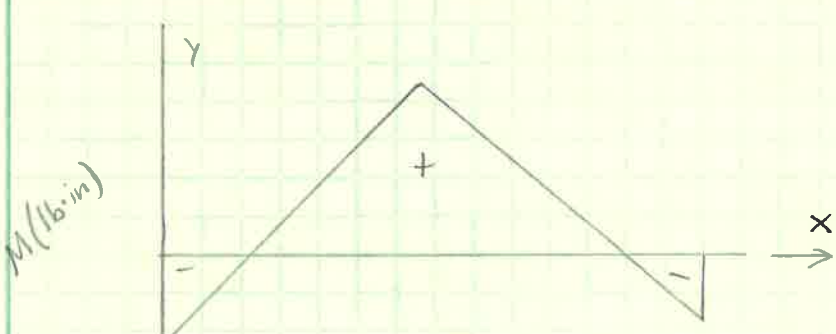
$M_{AB} = \frac{F}{8}(4x - l)\ ;\ M_{BC} = \frac{F}{8}(3l - 4x)$

$y_{AB} = Fx^2\,(4x - 3l)$

$y_{max} = \dfrac{-Fl^3}{192EI}$



SHIGLEY'S TABLE A9-3



$M_{max} = \dfrac{Fl}{4}$

$y_{max} = \dfrac{-Fl^3}{192EI}$

$= \dfrac{-(1500\,lb)(53.5\,in)^3}{192(30(10^6)psi)(6.96\ldots}$

$y_{max} = -0.00573\,in$

GANTRY HAND CALCULATIONS CONT.

$$M_{zz} = \frac{Fl}{4}$$

$$= \frac{1500\,lb(53.5\,in)}{4}$$

$$M_{zz} = 20062.5$$

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

$$= \frac{20062.5\ lb\,in\,(2\,in)}{6.960\,in^4}$$

$$\sigma_{max} = 5770\ psi$$

BY SYMMETRY TORQUE

$$T = F r_p$$

$$T = 1500\,lb\,(7.258\,in)$$

$$T = 10887\ lb{\cdot}in$$

$$\tau_{max} = \frac{Tr}{J} = \frac{Tr}{2I}$$

$$= \frac{10887\,(lb{\cdot}in)\,(2\,in)}{2\,(6.960\,in^4)}$$

$$\tau_{max} = 1564.22$$

FATIGUE

MOD-GOODMAN

$$\frac{\sigma_a'}{S_e} + \cancel{\frac{\sigma_m}{S_{ut}}}^{\,0} = \frac{1}{n}$$

ONLY ALTERNATING STRESS ON GANTRY

$$\sigma_a' = (\sigma_a^2 + 3\tau_a^2)^{1/2}$$

$$\sigma_a = \frac{\sigma_{max} - \sigma_{min}}{2}$$

GANTRY HAND CALCULATIONS CONT.

$\sigma_a = 5770$

$\tau_a = \dfrac{\tau_{max} - \tau_{min}}{2}$ psi

$= \dfrac{1564.22 - 0}{2}$ psi

$\tau_a = 782.11$ psi

$\sigma_a' = (5770^2 + 3(782.11)^2)^{1/2}$

$\sigma_a' = 16370$ psi

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

$S_e' = 0.5 S_{ut} =$

$= 0.5(57.99\,kpsi)$

$S_e' = 28.996\,kpsi$

$k_a = a S_{ut}^{-b}$

$a = 14.4$ (HOT ROLLED SHIGLEY'S TABLE 6-2)

$b = 0.718$

$k_a = 14.4(57.99)^{-0.718}$

$k_a = 0.7803$

$k_b = 0.879 d^{-0.107}$ $0.11 \le d \le 2in$ (6-20 SHIGLEYS)

$k_b = 0.91 d^{-0.157}$ $2 < d \le 10in$

$d_e = 0.808\sqrt{hb}$

$d_e = 3.232$

$k_b = 0.91(3.232)^{-0.157}$

$k_b = 0.757$

$k_c = 0.59$ (TORSION) (6-26 SHIGLEY'S)

GANTRY HAND CALCULATIONS CONT.

$K_d = 1$ RT

$K_e = 0.814$ 99% RELIABILITY (TABLE 6-5 SHIGLEYS)

$K_f = 1$ MISC.

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

$= (0.7803)(0.757)(0.59)(1)(0.814)(1) 28.996$ kpsi

$S_e = 8.226$ kpsi

$n = \frac{S_e'}{\sigma_a'}$

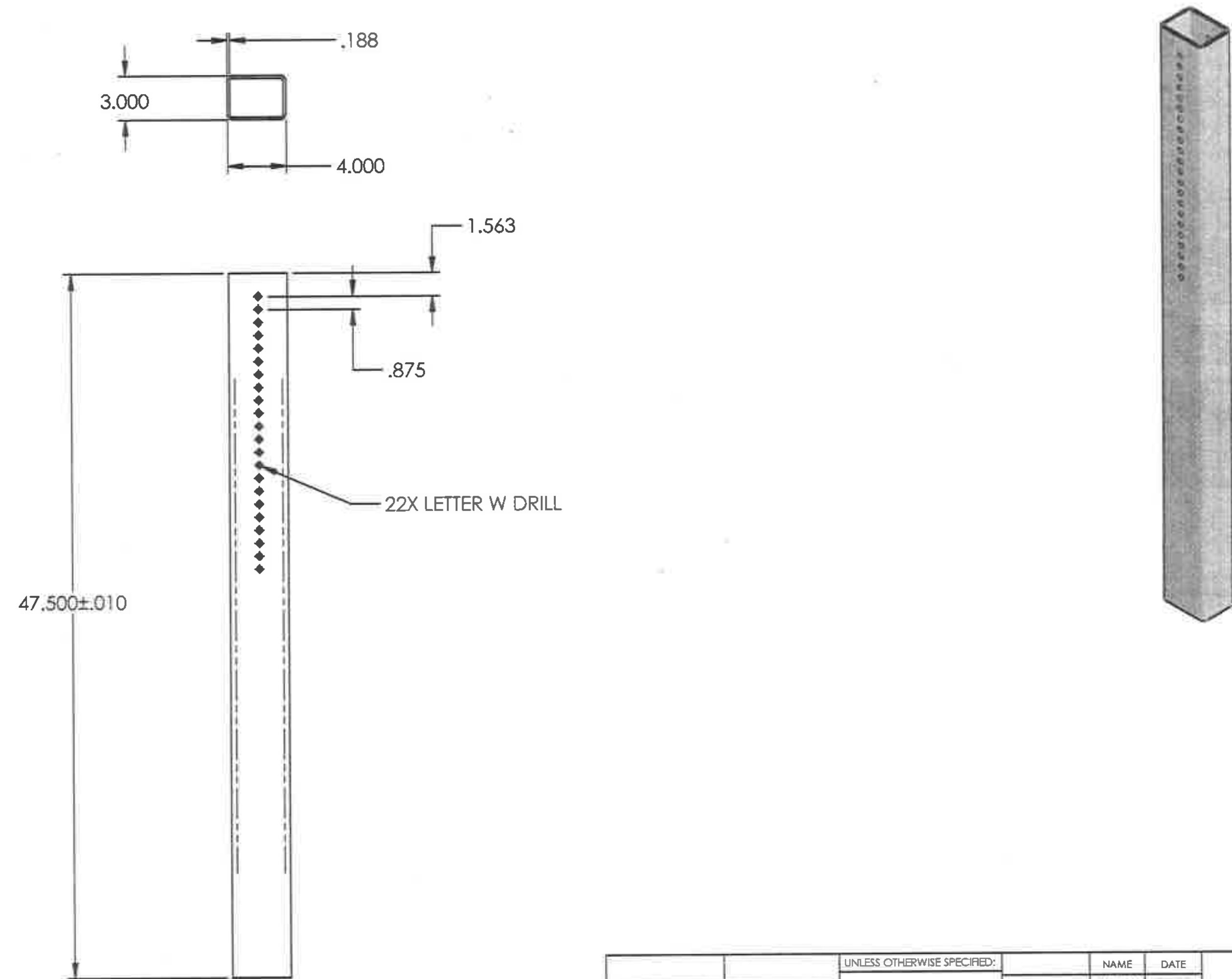
$= \frac{8226 \text{ psi}}{6320 \text{ psi}}$

$n = 1.30$

ANALYSIS CONSTANTS

Max Force (lbs)	1500
Ult Str Steel (kpsi)	57.99

Description	Geometry			Moment of Inertia	Max Bending stress						Endurance Limit								SF
Gantry	L (in)	h (in)	b (in)	I (in^4)	c (in)	Moment (lb in)	Torque T(lb/in)	Max Stress (kpsi)	Max Shear Stress (kpsi)	Stress Prime	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)	ymax (in)	
4"X4"X.1875"	53.5	4.	4.	6.960	2.	20062.5	10887	5.77	1.56	6.37	28.995	0.637	0.59	3.232	0.757	0.775	8.25	-0.0057	1.3



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TOLERANCES:
FRACTIONAL ±
ANGULAR: MACH ± BEND ±
TWO PLACE DECIMAL ±
THREE PLACE DECIMAL ±.005

INTERPRET GEOMETRIC TOLERANCING PER:

MATERIAL CARBON STEEL

FINISH

DO NOT SCALE DRAWING

	NAME	DATE
DRAWN	R. MURPHY	5/9/13
CHECKED		
ENG APPR.		
MFG APPR.		
Q.A.		

COMMENTS:

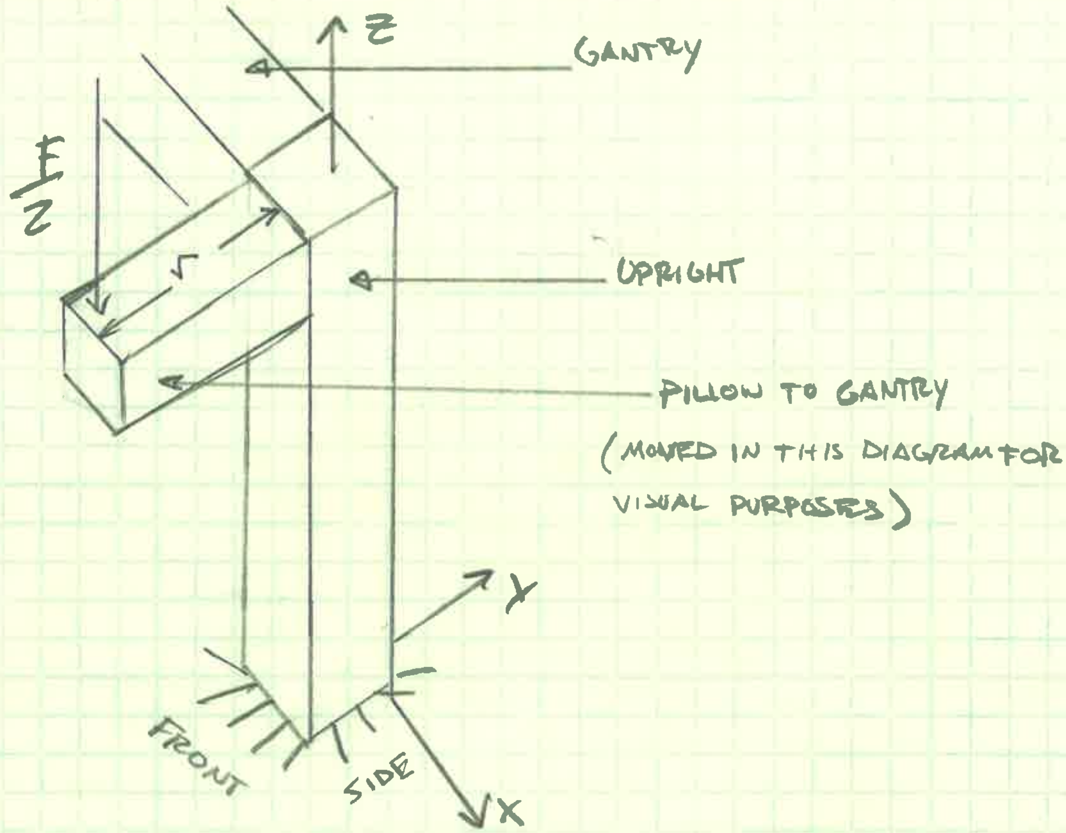
TITLE:

FRAME BASE UPRIGHT

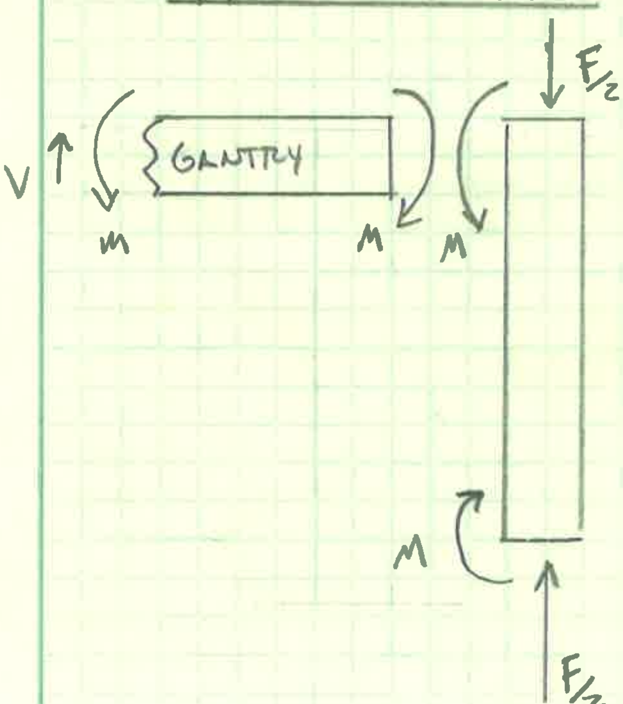
SIZE	DWG. NO.	REV
B		A

SCALE: 1:8 | WEIGHT: 31.8 LB | SHEET 1 OF 1

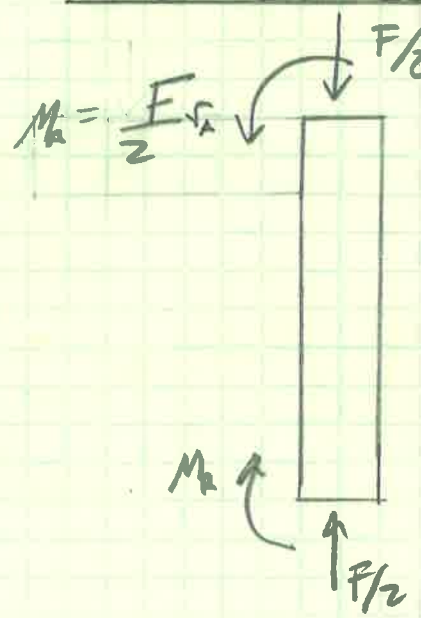
GANTRY UPRIGHT HAND CALCULATIONS



FRONT VIEW FBD



SIDE VIEW FBD



GANTRY UPRIGHT HAND CALCULATIONS

FRONT VIEW

M: CALCULATED FROM GANTRY CALCULATIONS

$M = 10031.25 \text{ lb}\cdot\text{in}$

SIDE VIEW

$M_A = \frac{F}{2} r_A$

$= \frac{1500}{2}(7.258\text{in})$

$M_A = 5443.5 \text{ (lb}\cdot\text{in)}$

$M_{max} = \sqrt{M^2 + M_A^2}$

$= \sqrt{(10031.25i)^2 + (5443.5)^2}$

$M_{max} = 11413.05 \text{ lb}\cdot\text{in}$

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

$= \frac{11413.05(\text{lb}\cdot\text{in})(2\text{in})}{(5.230\text{in}^4)}$

$\sigma_{max} = 4364.45 \text{ psi}$

$\sigma_{max\ axial} = \frac{F}{2A}$

$= \frac{1500\text{lb}}{2(2.394\text{in}^2)}$

$\sigma_{max\ axial} = 313.3 \text{ psi}$

GANTRY UPRIGHT HAND CALCULATIONS

FRONT

$$y_{max} = -\frac{M\ell^2}{2EI}$$

$$y_{max} = -\frac{10031.25(16in)(36in)^2}{2(30(10^6)psi)(5.23in^4)}$$

$$y_{max} = -0.04143\ in$$

SIDE

$$y_{max} = -\frac{M_a \ell^2}{2EI}$$

$$y_{max} = -\frac{5443.5(16in)(36in)^2}{2(30(10^6)psi)(5.23in^4)}$$

$$= -0.0225\ in$$

$$y_{total} = \sqrt{y^2 + y_a^2}$$

$$= \sqrt{(0.04143)^2 + (0.0225)^2}$$

$$y_{total} = 0.04715\ in$$

MOD- GOODMAN FATIGUE

$$\frac{\sigma_a'}{S_e} + \cancel{\frac{\sigma_m}{S_{ut}}}^{\,0} = \frac{1}{n}$$

ASSUMING ONLY ALTERNATING STRESS

$$\sigma_a' = \left[\left(\sigma_{a\,bend} + \frac{\sigma_{a\,axial}}{0.85}\right)^2 + \cancel{3\tau_a^2}^{\,0}\right]^{1/2}$$

$$\sigma_a' = (4364.45 + 313.3)\ psi$$

$$\sigma_a' = 4680\ psi$$

GANTRY UPRIGHT HAND CALCS CONT.

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

$S_e' = 0.5\, S_{ut}^{-b}$

$k_a = a\, S_{ut}^{-b}$

$a = 14.4$ (HR. SHIG. T. 6-2)
$b = 0.718$

$k_a = 14.4(57.99)^{-0.718}$

$k_a = 0.7803$

$k_b = 0.879 d^{-0.107}$ $\qquad \le d \le 2$ in

$k_b = 0.91 d^{-0.157}$ $\qquad 2 < d \le 10$ in

$d_e = 0.808\sqrt{hb}$

$= 2.799$

$k_b = 0.757$

$k_c = 0.85$ (bending) (6-26 SHIG.)

$k_d = 1$ R.T.

$k_e = 0.814$ 99% REL. T 6-5 SHIG.

$k_f = 1$ MISC.

$S_e = (0.7803)(0.757)(0.85)(1)(0.814)(1)(28{,}996\text{ kpsi})$

$S_e =$ [illegible]

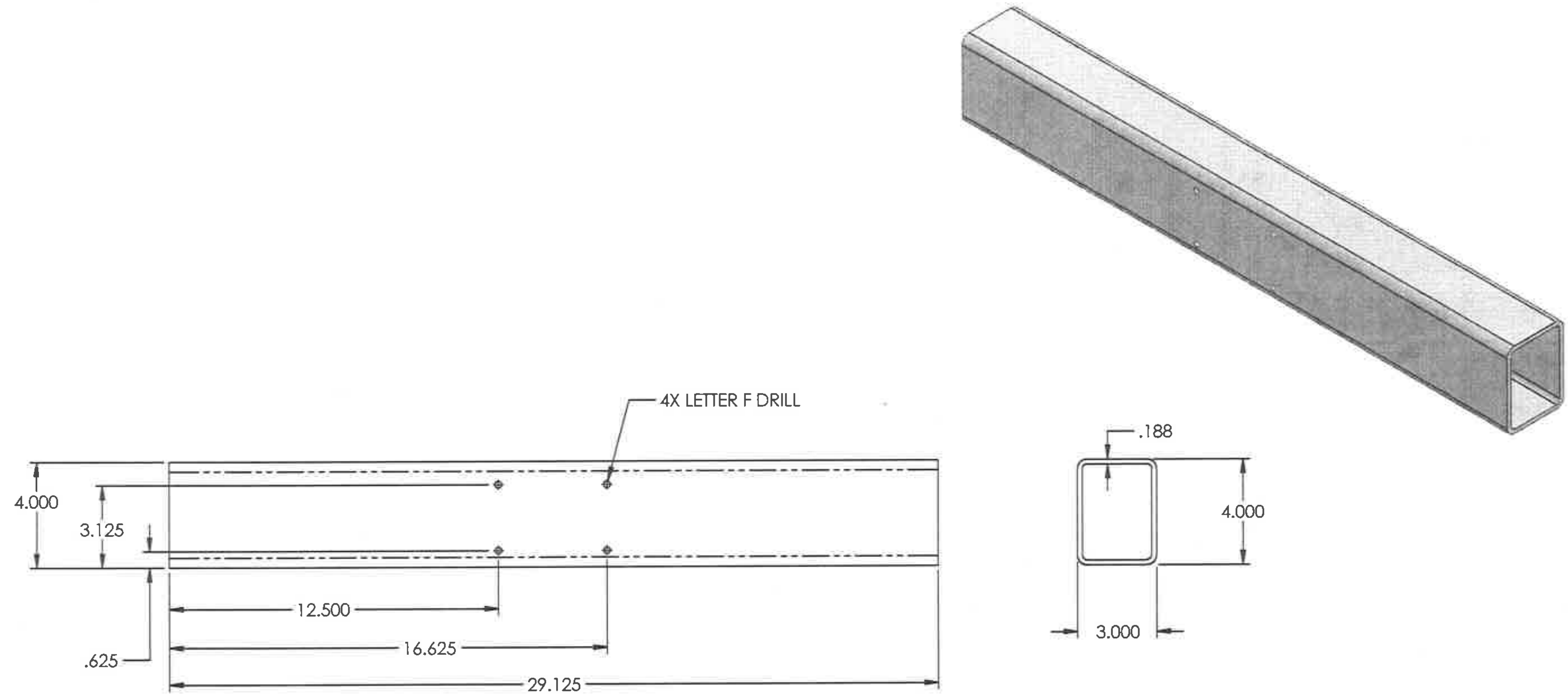
$n = \dfrac{15930\text{ psi}}{4680\text{ psi}}$

$n = 3.40$

ANALYSIS CONSTANTS

Max Force (lbs)	1500
Ult Str Steel (kpsi)	57.99

Description	Geometry						Moment of Inertia	Max Bending stress				Endurance Limit								SF
Gantry Upright	L (in)	h (in)	b (in)	t (in)	h_{inner} (in)	b_{inner} (in)	I (in^4)	c (in)	Moment (lb.in)	Torque T(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)	ymax (in)	
3"X4"X0.188"	36	4.	3.	0.188	3.624	2.624	5.230	2.	11413	0	4.68	28.995	0.710	1	2.799	0.774	0.787	15.93	-0.0112	3.4



UNLESS OTHERWISE SPECIFIED:	
DIMENSIONS ARE IN INCHES TOLERANCES: FRACTIONAL ± ANGULAR: MACH ± BEND ± TWO PLACE DECIMAL ± THREE PLACE DECIMAL ±.010	
INTERPRET GEOMETRIC TOLERANCING PER:	
MATERIAL	CARBON STEEL
FINISH	
DO NOT SCALE DRAWING	

NEXT ASSY	USED ON
APPLICATION	

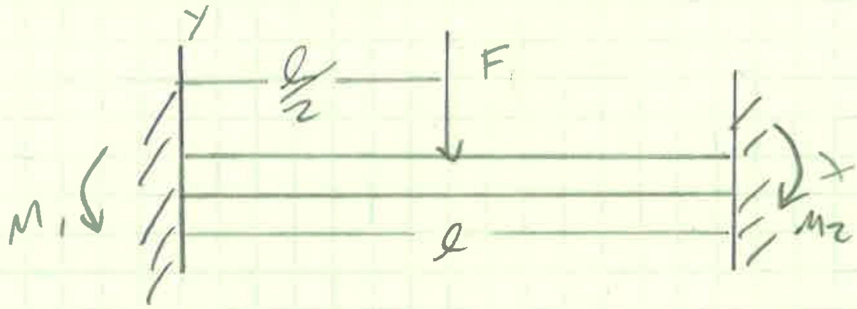
	NAME	DATE
DRAWN	R.MURPHY	5/7/13
CHECKED		
ENG APPR.		
MFG APPR.		
Q.A.		
COMMENTS:		

TITLE:

PILLOW ARM

SIZE	DWG. NO.	REV
B		A
SCALE: 1:2	WEIGHT: 19.6 LB	SHEET 1 OF 1

PILLOW ARM HAND CALCS



$R_1 = R_2 = F/2$

$M_1 = M_2 = F\ell/8$

$V_{AB} = -V_{BC} = F/2$

$M_{AB} = \frac{F}{8}(4x - \ell)$; $M_{BC} = \frac{F}{8}(3\ell - 4$

$M_{max} = \frac{F\ell}{4}$

$Y_{AB} = Fx^2(4x - 3\ell)$

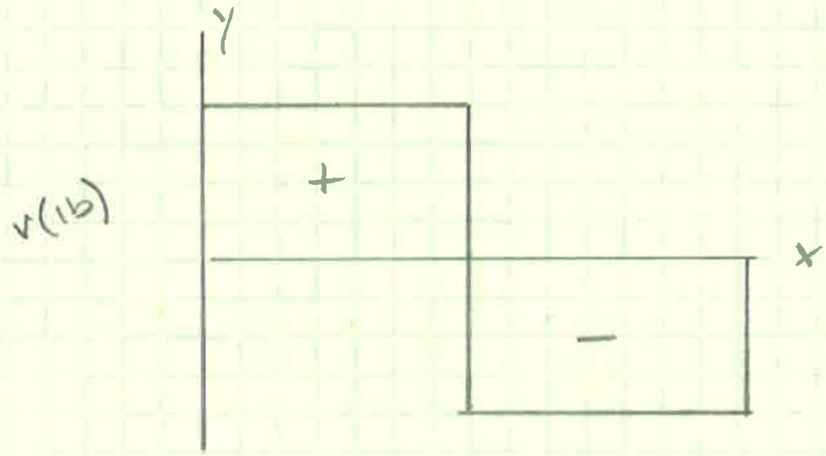
$Y_{max} = \frac{-F\ell^3}{192EI}$

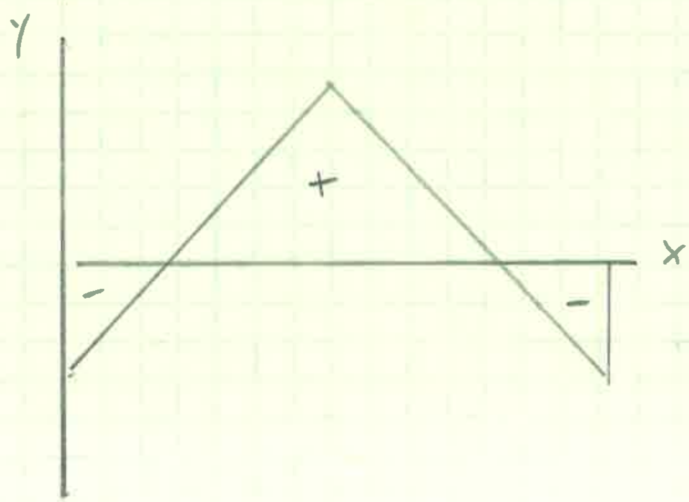
$F = 1500\,lb$

$\ell = 29.13\ in$

$E = 30(10)^6\ psi$

$I = 5.23\ in^4$





PILLOW ARM HAND CALCS CONT.

$$M_{max} = \frac{Fl}{4}$$

$$= \frac{1500lb(29.13in)}{4}$$

$$M_{max} = 10923.75 \text{ lbin}$$

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

$$= \frac{10923.75 \text{ lbin}(1.5in)}{5.23 in^4}$$

$$\sigma_{max} = 3130$$

YIELD

$$n_y > \frac{S_y}{\sigma_{max}}$$

$$> \frac{32000psi}{3130}$$

$$n_y > 10.22$$

MOD-GOODMAN FATIGUE

$$\left(\frac{\sigma_a'}{S_e} + \frac{\sigma_m'}{S_{ut}}\right) = \frac{1}{n}$$

$$\sigma_a' = (\sigma_a^2 + 3\tau_a^2)^{1/2}$$

$$\cancel{\sigma_m'}^{\,0} = (\sigma_m^2 + 3\tau_m^2)^{1/2}$$

$$\sigma_a = \frac{\sigma_{max} - \sigma_{min}}{2}$$

$$y_{max} = \frac{-Fl^3}{192EI}$$

$$= -\frac{1500lb(29.13in)^3}{192(30\times10^6psi)(5.23i}$$

$$y_{max} = -0.0012in$$

PILLOW ARM CONT.

MOD-GOODMAN FATIGUE

$\sigma_a = 3130$ psi

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

$S_e' = 0.5 S_{ut}$

$S_e' = 0.5(57.99\ kpsi)$

$S_e' = 28.996\ kpsi$

$k_a = a S_{ut}^{-b}$

$a = 14.4$ TABLE 6-2 SHIGLEY'S (HR)

$b = 0.718$

$k_a = 14.4(57.99)^{-0.718}$

$k_a = 0.7803$

$k_b = 0.879\ d^{-0.107}$ $0.11 \le d \le 2 in$ 6-20 SHIGLEYS

$k_b = 0.91 d^{-0.157}$ $2 < d \le 10 in$

$d_e = 0.808\sqrt{hb}$

$= 0.808\sqrt{3(4) in^2}$

$= 2.799$

$k_b = 0.91(2.799)^{-0.157}$

$k_b = 0.744$

$k_c = 1$ bending (6-26) SHIGLEY'S

$k_d = 1$ R.T.

$k_e = 0.814$ 99% Reliability

$k_f = 1$ Misc.

PILLOW ARM CONT.

$$S_e = (0.7803)(0.774)(1)(1)(0.814)(1)(28.996)$$

$$\underline{S_e = 15.93 \text{ ksi}}$$

$$n = \left(\frac{\sigma_a'}{S_e} + \cancelto{0}{\frac{\sigma_m'}{S_{ut}}}\right)^{-1}$$

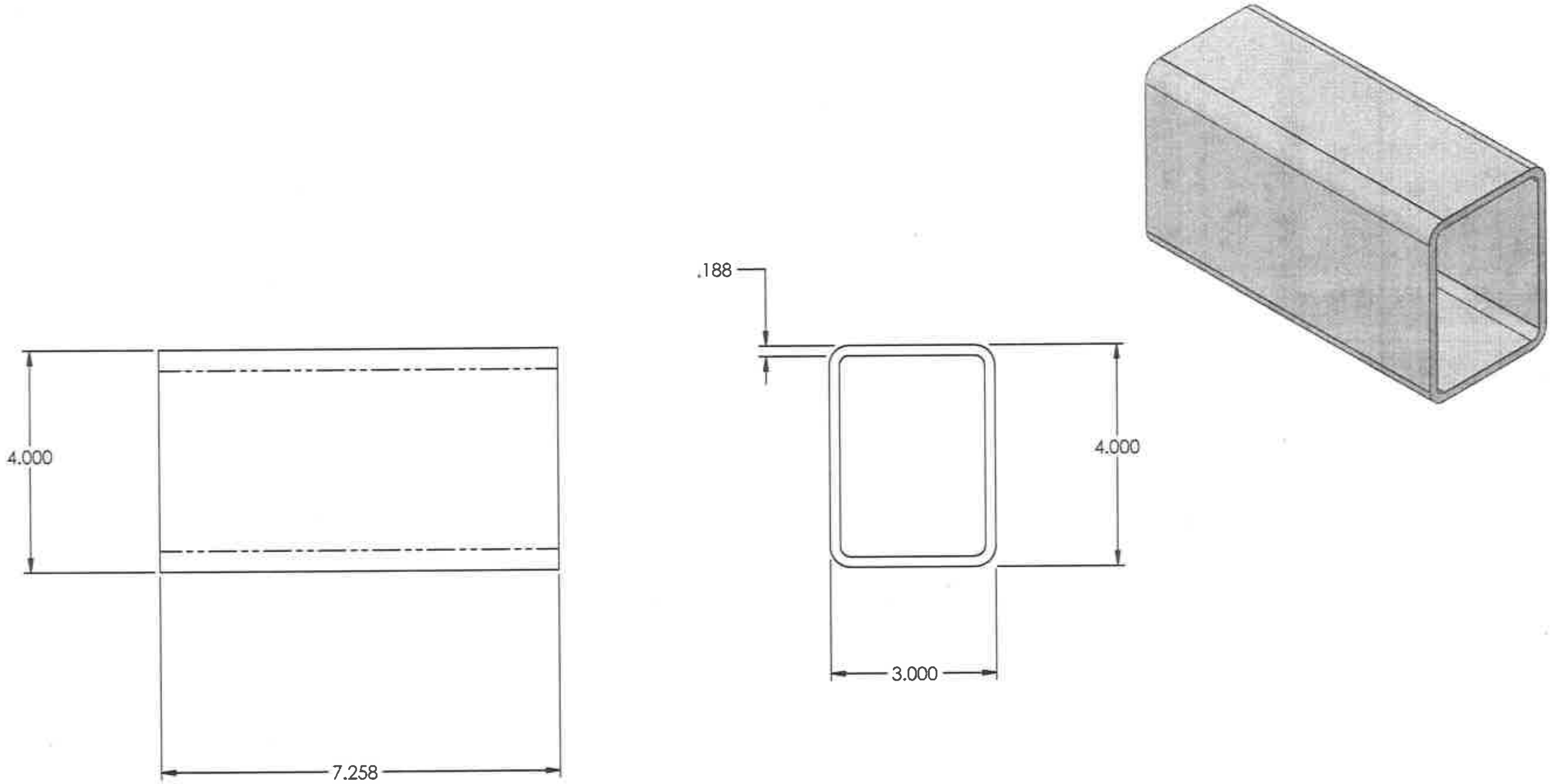
$$= \left(\frac{1565 \text{ psi}}{15930}\right)^{-1}$$

$$\underline{n = 5.1}$$

ANALYSIS CONSTANTS

Max Force (lbs)	1500
Ult Str Steel (kpsi)	57.99

Description	Geometry						Moment of Inertia	Max Bending stress				Endurance Limit								SF
Pillow Arm	L (in)	h (in)	b (in)	t (in)	h_{inner} (in)	b_{inner} (in)	I (in^4)	c (in)	Moment (lb in)	Torque T(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)	ymax (in)	
3"X4"X.188"	29.13	3.	4.	0.188	3.624	2.624	5.230	1.5	10923.75	0	3.13	28.995	0.710	1	2.799	0.774	0.787	15.93	-0.0012	5.1



UNLESS OTHERWISE SPECIFIED:	NAME	DATE
DIMENSIONS ARE IN INCHES TOLERANCES: FRACTIONAL± ANGULAR: MACH± BEND ± TWO PLACE DECIMAL ± THREE PLACE DECIMAL ±.010	DRAWN R.MURPHY	5/7/13
	CHECKED	
	ENG APPR.	
	MFG APPR.	
INTERPRET GEOMETRIC TOLERANCING PER:	Q.A.	
MATERIAL CARBON STEEL	COMMENTS:	
FINISH		
DO NOT SCALE DRAWING		

NEXT ASSY	USED ON
APPLICATION	

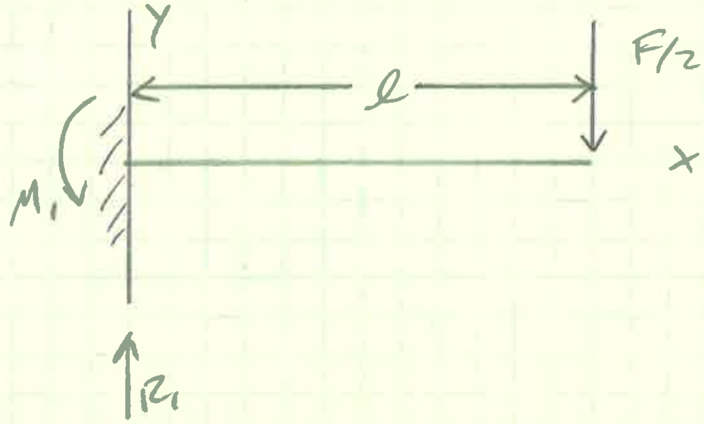
TITLE:

PILLOW TO GANTRY

SIZE B | DWG. NO. | REV A

SCALE: 1:2 | WEIGHT: 4.90 LB | SHEET 1 OF 1

PILLOW TO GANTRY SUPPORT



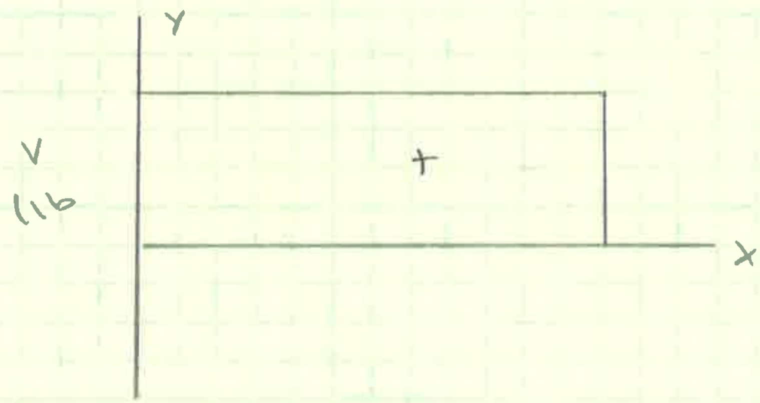
$R_1 = V = F$

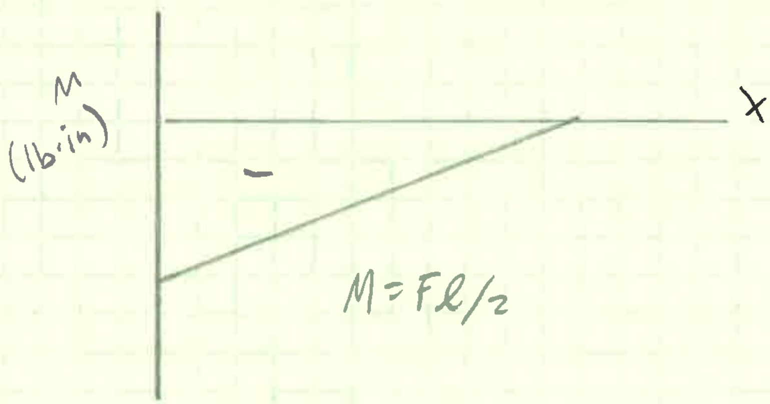
$M_1 = F\ell$

$M = F(x-\ell)$

$$y = \frac{Fx^2}{6EI}(x-3\ell)$$

$$y_{max} = -\frac{F\ell^3}{3EI}$$





PILLOW TO GANTRY CONT.

MOD. GOODMAN FATIGUE

$$\frac{\sigma_a'}{S_e} + \frac{\sigma_m'}{S_{ut}} = \frac{1}{n}$$

$$\sigma_a' = (\sigma_a^2 + 3\cancel{\tau_a^2}^{\,0})^{1/2}$$

$$\sigma_m' = (\sigma_m^2 + 3\cancel{\tau_m^2}^{\,0})^{1/2}$$

$$\sigma_a = \frac{Fl/2\, c}{I}$$

$$\sigma_m = 0$$

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

$$S_e' = 0.5 S_{ut}$$

$$k_a = a S_{ut}^{-b}$$

$$k_b = 0.91\, d_e^{-0.157}$$ $\qquad d_e = 0.808\sqrt{hb}$

$$= 0.91(2.799)^{-0.157}$$ $\qquad = 2.799$

$$k_b = 0.774$$

$$k_c = 0.85$$

$$k_d = 1$$

$$k_e = 0.62$$

$$k_f = 1$$

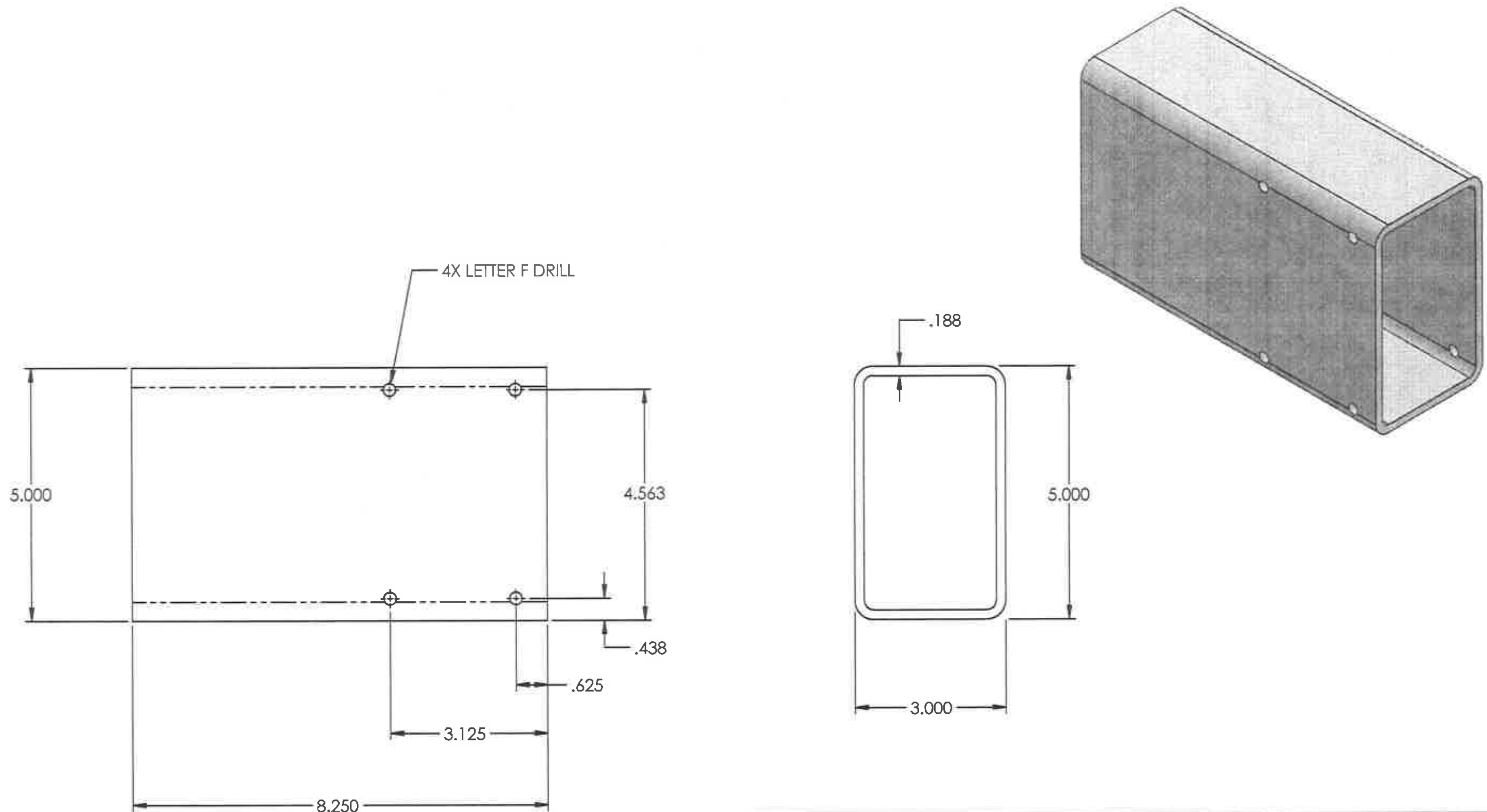
FROM EXCEL

$$\boxed{\begin{array}{l} n = 20.4 \\ y_{max} = -0.000609\ in \end{array}}$$

ANALYSIS CONSTANTS

Max Force (lbs)	1500
Ult Str Steel (kpsi)	57.99

Description	Geometry						Moment of inertia	Max Bending stress				Endurance Limit								SF
Pillow to Gantry Support	L (in)	h (in)	b (in)	t (in)	h_{inner} (in)	b_{inner} (in)	I (in^4)	c (in)	Moment (lb in)	Torque T(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)	ymax (in)	
3"X4"X.188"	7.258	3.	4.	0.188	3.624	2.624	5.230	1.5	2721.75	0	0.78	28.995	0.710	1	2.799	0.774	0.787	15.93	-0.0006	20.4



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	CHECKED		
	ENG APPR.		
	MFG APPR.		
INTERPRET GEOMETRIC TOLERANCING PER:	Q.A.		
MATERIAL CARBON STEEL	COMMENTS:		
FINISH			
DO NOT SCALE DRAWING			

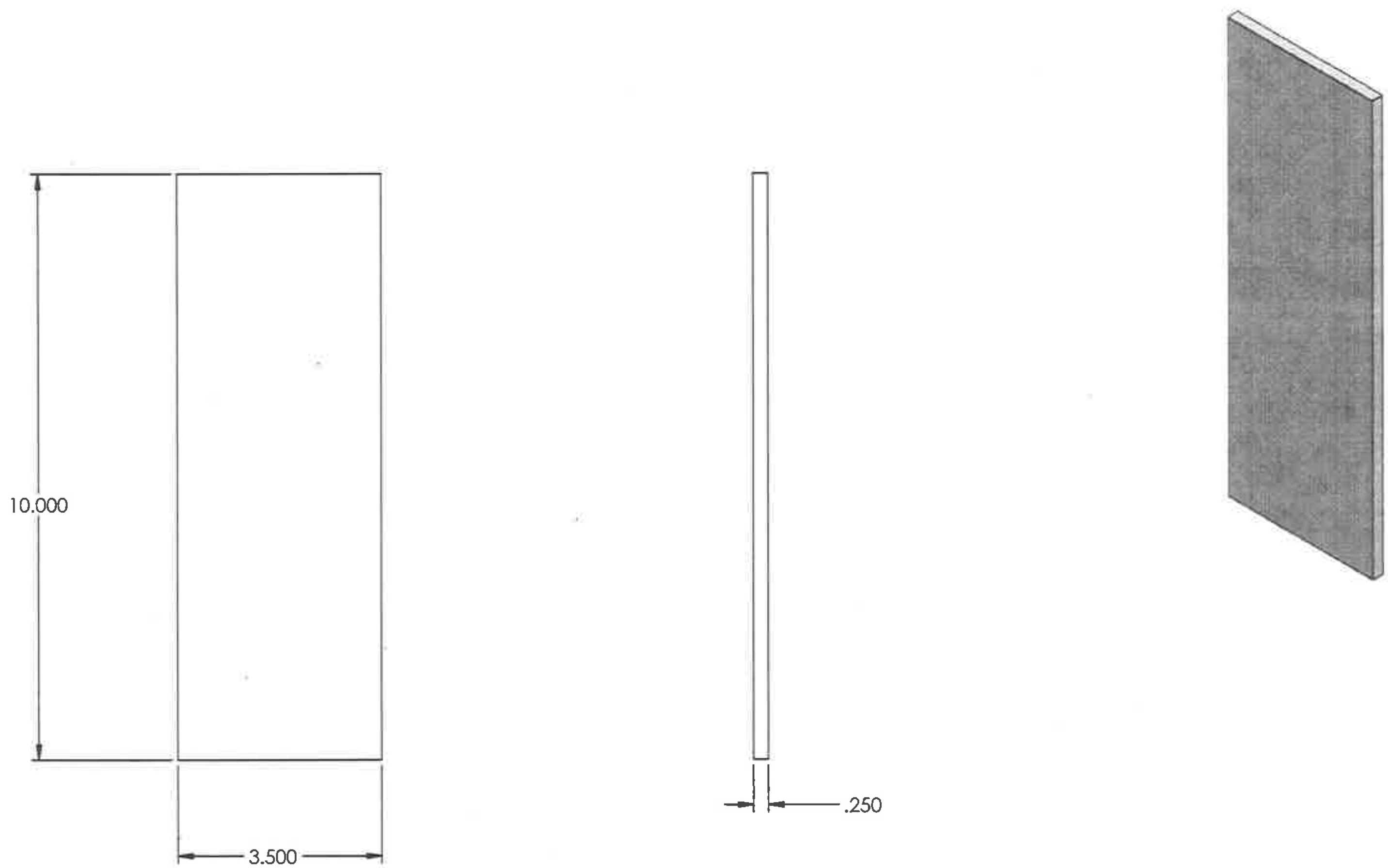
NEXT ASSY	USED ON
APPLICATION	

TITLE:

PILLOW SUPPORT

SIZE	DWG. NO.	REV
B		A
SCALE: 1:2	WEIGHT: 6.42 LB	SHEET 1 OF 1

MECHANICAL CAL POLY ENGINEERING



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DIMENSIONS ARE IN INCHES
TOLERANCES:
FRACTIONAL ±
ANGULAR: MACH ± BEND ±
TWO PLACE DECIMAL ±
THREE PLACE DECIMAL ±.010

INTERPRET GEOMETRIC TOLERANCING PER:

MATERIAL CARBON STEEL

FINISH

NEXT ASSY | USED ON

APPLICATION

DO NOT SCALE DRAWING

	NAME	DATE
DRAWN	R.MURPHY	5/7/13
CHECKED		
ENG APPR.		
MFG APPR.		
Q.A.		

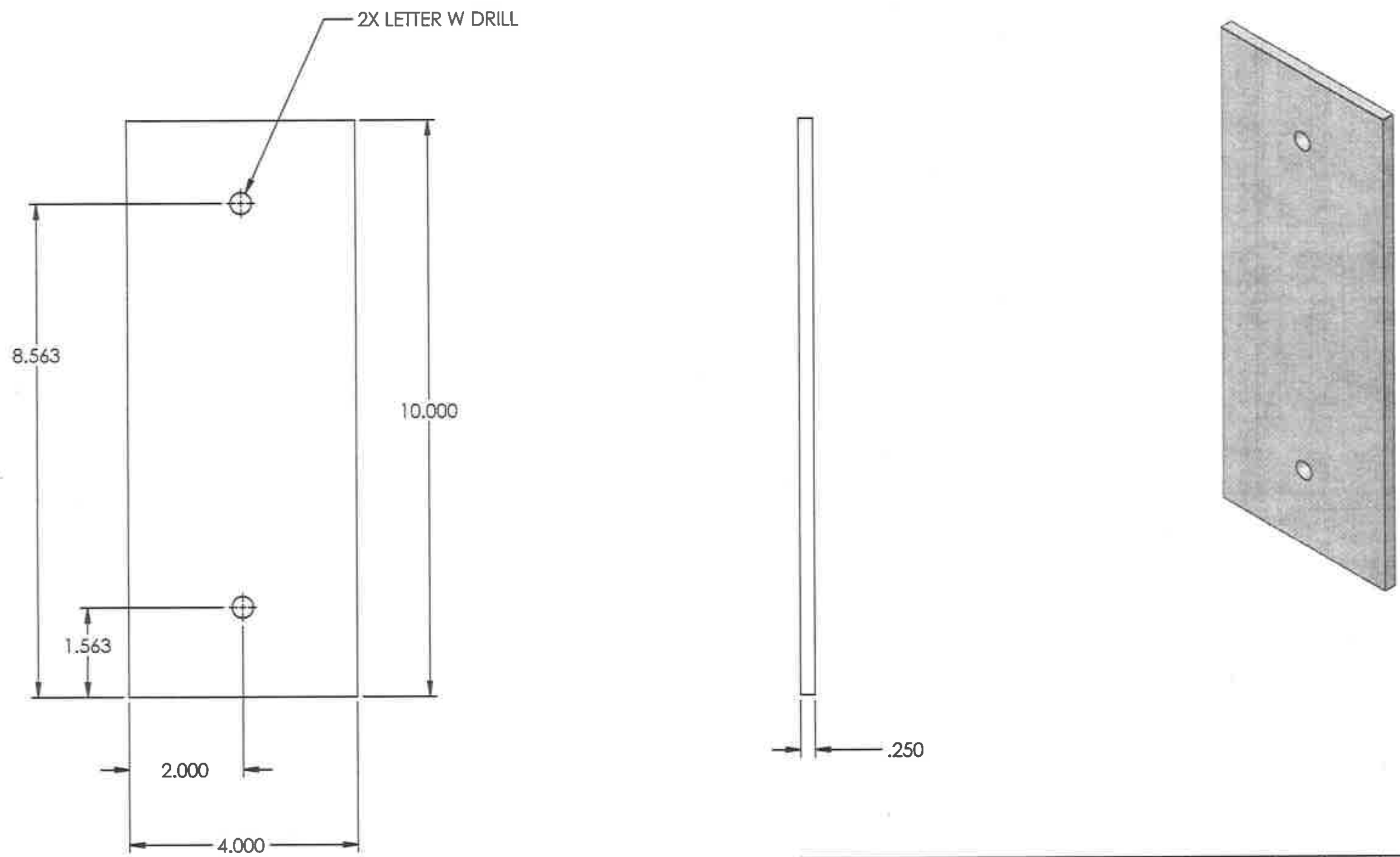
COMMENTS:

TITLE:

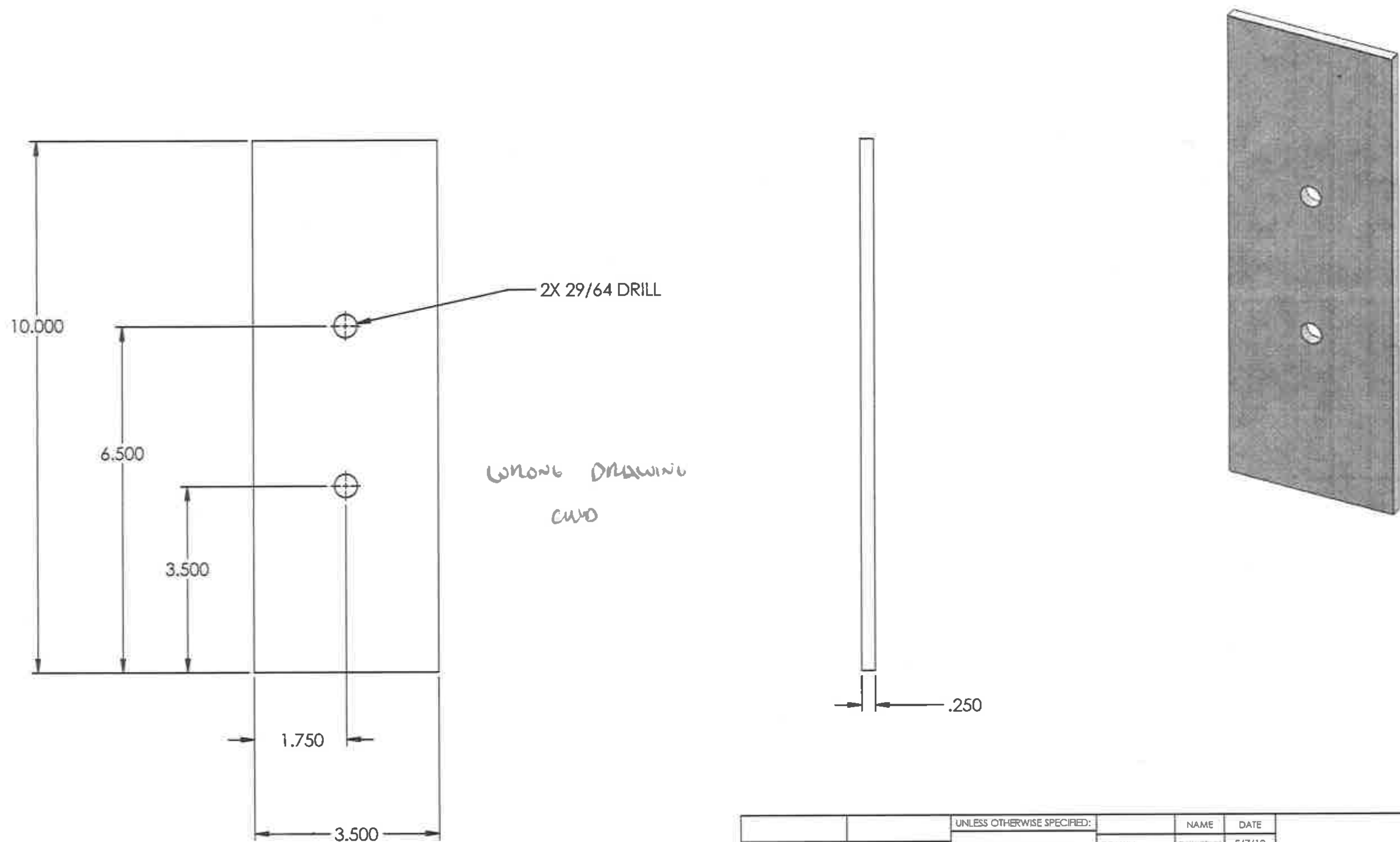
GANTRY SLIDER BACK

SIZE B | DWG. NO. | REV A

SCALE: 1:2 | WEIGHT: 2.47 LB | SHEET 1 OF 1



		UNLESS OTHERWISE SPECIFIED:		NAME	DATE	
		DIMENSIONS ARE IN INCHES TOLERANCES: FRACTIONAL ± ANGULAR: MACH ± BEND ± TWO PLACE DECIMAL ± THREE PLACE DECIMAL ±.010	DRAWN	R. MURPHY	4/28/13	TITLE:
			CHECKED			GANTRY SLIDER OUTER
			ENG APPR.			
			MFG APPR.			
		INTERPRET GEOMETRIC TOLERANCING PER:	Q.A.			
		MATERIAL CARBON STEEL	COMMENTS:			SIZE B — DWG. NO. — REV A
NEXT ASSY	USED ON	FINISH				
APPLICATION		DO NOT SCALE DRAWING				SCALE: 1:2 — WEIGHT: 2.80 LB — SHEET 1 OF 1



WRONG DRAWING
CWD

UNLESS OTHERWISE SPECIFIED:	
DIMENSIONS ARE IN INCHES TOLERANCES: FRACTIONAL ± ANGULAR: MACH ± BEND ± TWO PLACE DECIMAL ± THREE PLACE DECIMAL ±.010	
INTERPRET GEOMETRIC TOLERANCING PER:	
MATERIAL	CARBON STEEL
FINISH	
DO NOT SCALE DRAWING	

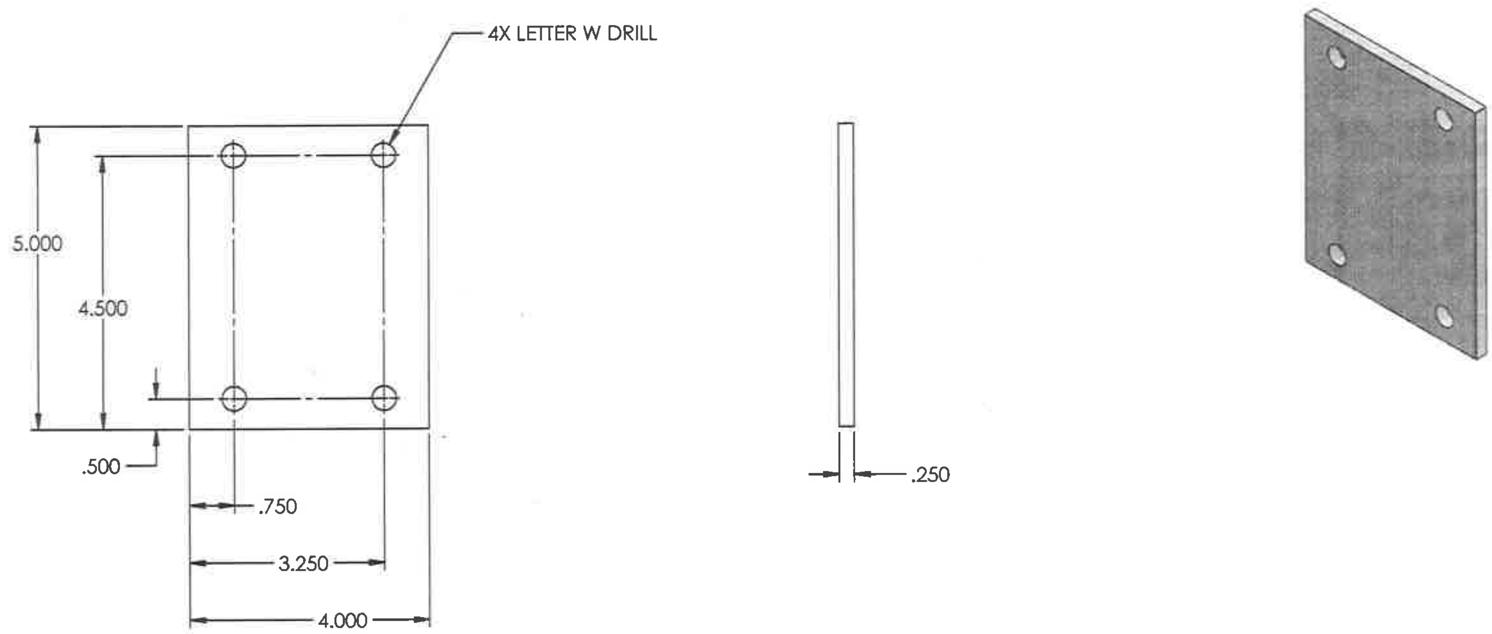
	NAME	DATE
DRAWN	R.MURPHY	5/7/13
CHECKED		
ENG APPR.		
MFG APPR.		
Q.A.		
COMMENTS:		

NEXT ASSY	USED ON
APPLICATION	

TITLE:

GANTRY SLIDER FRONT

SIZE	DWG. NO.	REV
B		A
SCALE: 1:2	WEIGHT: 2.44 LB	SHEET 1 OF 1



UNLESS OTHERWISE SPECIFIED:		NAME	DATE
DIMENSIONS ARE IN INCHES TOLERANCES: FRACTIONAL ± ANGULAR: MACH ± BEND ± TWO PLACE DECIMAL ± THREE PLACE DECIMAL ±.010	DRAWN	R.MURPHY	5/7/13
	CHECKED		
	ENG APPR.		
	MFG APPR.		
INTERPRET GEOMETRIC TOLERANCING PER:	Q.A.		
MATERIAL CARBON STEEL	COMMENTS:		
FINISH			
DO NOT SCALE DRAWING			

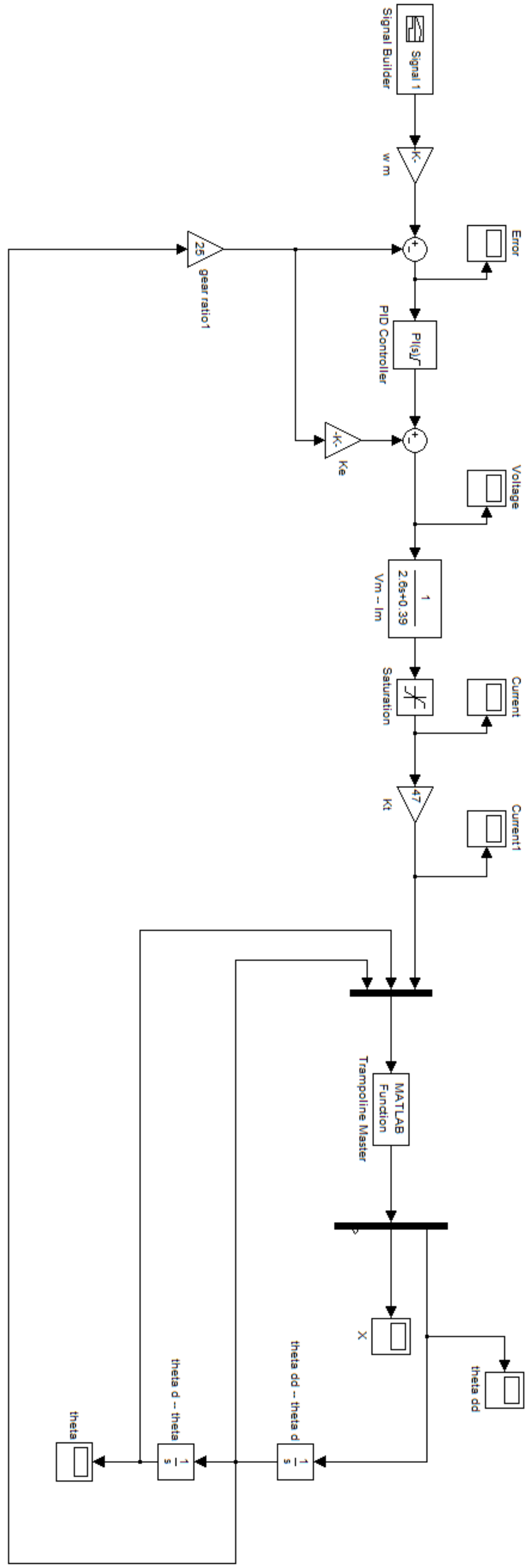
NEXT ASSY	USED ON
APPLICATION	

TITLE:

PILLOW TO GANTRY MOUNT

SIZE	DWG. NO.	REV
B		A
SCALE: 1:2	WEIGHT: 1.37 LB	SHEET 1 OF 1

Appendix K: System Model



Appendix L: Stroke Selection Table

Desired Stroke (in)	Lower Gantry Hole	Linkage Set	Crank Radius (in)	CR Position (in)
0.5	47.625	Small	0.375	0.75
1	46.75	Small	0.75	1.5
1.5	45.875	Small	1.125	2.25
2	45	Small	1.5	3
2.5	44.125	Small	1.875	3.75
3	43.25	Small	2.25	4.5
3.5	42.375	Small	2.625	5.25
4	41.5	Small	3	6
4.5	40.625	Small	3.375	6.75
5	39.75	Small	3.75	7.5
5.5	38.875	Small	4.125	8.25
6	38	Small	4.5	9
6.5	47.625	Large	4.875	9.75
7	46.75	Large	5.25	10.5
7.5	45.875	Large	5.625	11.25
8	45	Large	6	12
8.5	44.125	Large	6.375	12.75
9	43.25	Large	6.75	13.5
9.5	42.375	Large	7.125	14.25
10	41.5	Large	7.5	15
10.5	40.625	Large	7.875	15.75
11	39.75	Large	8.25	16.5
11.5	38.875	Large	8.625	17.25
12	38	Large	9	18