Brinell Limit Testing Machine-Final Design Report

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Ball Busters

California Polytechnic State University, San Luis Obispo

Senior Project

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Statement of Confidentiality

The complete senior project report was submitted to the project advisor and sponsor. The detailed drawings and assembly and procedure instructions of this project are of a confidential nature and will not be published at this time.

Statement of Disclaimer

Since this project is a result of a class assignment, it has been graded and accepted as fulfillment of the course requirements. Acceptance does not imply technical accuracy or reliability. Any use of information in this report is done at the risk of the user. These risks may include catastrophic failure of the device or infringement of patent or copyright laws. California Polytechnic State University at San Luis Obispo and its staff cannot be held liable for any use or misuse of the project.

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Executive Summary

In keeping with the California Polytechnic State University motto of "Learn by Doing", this project was performed by Mechanical Engineering students Joe Cloutier, Josh Kessler, and Mike Jaskulsky II as their senior project. Starting in the Fall 2009 quarter and reaching completion with the end of the Spring 2010 quarter, this project provided these students with experience in application of a formal engineering design process in the solving of an open-ended engineering design problem, in developing and maintaining an engineering project schedule, as well as providing further experience working on an engineering team.

As the engineers of Parker Aerospace seek to use different metals in their high performance bearing applications than have traditionally been used in the past, often the data does not exist for them to be able to accurately design against brinelling. To provide their engineers with this data, Parker Aerospace proposed the following as a senior project to Cal Poly's seniors. They requested that a team of engineering students would design, fabricate, assemble, and validate through testing a machine that would determine the loads at the onset of brinelling for different metals and would allow for multiple measurements to be taken from each set of sample materials tested. Some of the secondary design requirements were for the test fixture to be portable, small enough to be used as a desktop unit, be able to accommodate a thermal chamber around the test area, and also provide measurements of the total deformation of the sample materials when under load. Also, time allowing, Parker Aerospace requested that the senior project team devote the last part of the last quarter to using the machine to provide data for a number of materials that they will provide.

The loads that the test machine would need to deliver to test all material samples to the onset of brinelling were determined through hertzian contact stress analysis. These calculated loads were then used to determine the deflection of the sample materials, allowing for the sizing of structural components and selection of necessary sensors.

The design for the fixture was developed around the initial design concept displayed in the Project Proposal by Parker Aerospace. After developing a number of different designs and variations of specific components of the fixture, the best of these design variations were presented to a panel of Parker Aerospace's engineers during a Preliminary Design Review. From these designs, a final design was selected and various modifications were made as suggested by Parker. A final design was decided on and the rest of the project was completed by the end of the Spring quarter.

Introduction

Sponsor Background and Needs

As Parker Aerospace works to develop bearings to meet their high performance requirements, they have been seeking to push the materials they use for their bearing races to the very extent of their loading limits. While a wealth of data is available for standard bearing materials, Parker engineers will need to test the brinelling limits of new materials so they may be utilized. Brinelling limits are found to be functions of the type of bearings, the material and heat treatment, and the operating temperatures of the bearing applications. This project will help Parker Aerospace test the loading limits of various metals in order to choose the lightest and most durable materials that will endure higher loads before Brinelling occurs.

Formal Problem Definition

The goals for this project were to design and build a test fixture, test the fixture against materials with known load limits, and determine the load limits of new material samples. The test fixture needed to be small enough to be portable and fit on a table top. Also it had to accommodate the addition of a thermal chamber. This required the instrumentation to be located such that they will not be affected by the thermal changes of the test area.

Objectives

The goal for this project was to design, build, and validate through testing, a portable, desktop test fixture that will determine the load limits of new material samples at the onset of brinelling. Validation was performed by determining load limits from samples of AISI 52100, a well-documented metal, and comparing the fixture's output values to documented values. Purchasing and machining selections were made such that our senior project team members were able to perform all the required machining for the components not purchased, detailed drawings are provided to Parker Aerospace so they can easily reproduce, or fix, the machine. A QFD was developed to help determine the design choice that best meets our project requirements. Some highlighted requirements include:

- A load cell to measure the forces exerted on the material
- A way to measure the distance of compression with 1/10000" resolution
- A way to easily log the data onto a laptop
- Test 1/8" to ½" ball bearing and 1/8" to ¼" roller bearing samples between two sample plates
- Enclosure for the test samples to be heated or cooled, from -60°F to 400°F, while leaving the load cell and displacement measurement devices open to ambient temperatures

- Prefer 110V electrical outlet power source
- Stiff apparatus that is able to sustain the maximum Brinelling loads without significant deformation
- Easily transportable
- Sample trays
- Validate the machine by testing a known material of AISI 52100

Management Plan

Everyone was responsible for contributing to the research, calculations, design ideas, production, testing, and reporting. Joe was responsible for keeping track of all the material, electronically and on paper. A Ghantt Chart with a complete breakdown of the foreseen tasks and milestones for the project is provided in Appendix G.

Having completed all background research and preliminary design calculations as well as the detailed design calculations used to size the components and select materials and having completed formulation of a number of designs, our project team was able to successfully enter the Preliminary Design Review with Parker Aerospace. The PDR with our sponsor and a panel of engineers was held on November 20, 2009, during which a final design was decided on. On leaving the PDR, we set the goal of finishing the detail drawings for the apparatus and selecting sensors in preparation for the beginning of fabrication and assembly during the Winter quarter.

After addressing the action items from the PDR, our Critical Design Review with our sponsor and their panel of engineers was conducted during the first week of Winter quarter, on January 8, 2010. During the weeks immediately after the CDR, our attentions were focused on addressing the action items that arose during the meeting. Five weeks after that meeting with our sponsor, that is six weeks into the next quarter, we are planning to begin machining the parts not purchased. Completing the assembly of the test fixture and compiling a complete Final Test Plan took place during the Spring quarter.

Background

Background research has provided us with knowledge of common test machines and practices. While each of these machines, by the intent of their design, could be used to determine Brinelling limits, none of the machines viewed in our research would be able to meet all the requirements for this project. The first and most significant issue that arose was that only one sample piece could be tested at a time. The goal for this new machine will be to test two sample plates during each load phase. Market research has indicated that all of the machines readily available function by indenting the sample material with a carbide ball. Depending on the price range, the indenting force can be provided by hand, electronically, or hydraulically. Also the quality of load indicator will fluctuate with price range. Another feature of the higher priced units is an optical scope used to measure the indent, whereas lower priced models do not provide such units, requiring additional equipment to be provided by the user.

Figure 1. 3000 BLD Brinell Tester from Wilson Instruments.

The 3000 BLD Brinell Hardness Tester, seen in Figure 1, is ideal for a wide range of Brinell loads from 187.5 – 3000 kgf. It was designed with a rugged construction to withstand harsh environments, and it combines high rigidity and close-loop load cell technology to ensure accurate and safe load applications. This model also uses an external microscope, which allows the operator to measure the diagonals and enter into the built-in keypad calculator for quick Brinell hardness value display. It was designed with an easy-to-use operator interface that allows for quick and easy set-up and operation. The menu is displayed on a large LCD, which shows test parameters, Brinell hardness, statistics, and conversion to ASTM and ISO.[3]

Figure 2.CLB3 Hardness Tester from Wilson Instruments.

The CLB3 Brinell Hardness Tester, seen in Figure 2, is a closed-loop Brinell Hardness Testing instrument and is a unique testing solution for accurate, high-capacity brinell testing. This model utilizes load cell technology and a proven Instron tension/compression frame to deliver an unlimited load range from 32.5 – 3000 kgf. It also includes a user-friendly control panel for method set up, start and stop, and a return functionality, as well as a 10 mm carbide ball indenter and two brinell test blocks.[2]

Nearly all metals can be tested with a brinell test by varying the test force and material sizes. Common loads and sizes range from 500 to 3000 kg and 5mm to 10mm carbide balls respectively. A large drawback is the need to measure the size of the indent, which needs to be very accurate, in order to calculate the brinell hardness.[1] While the scope of this project was only to be able to visually determine the load at which brinelling occurs, excluding the need to determine the brinell hardness by optically measuring the indent size, Parker's lab facility does have access to the necessary equipment to perform these measurements should these values become needed in the future. Another drawback is that brinell test machines do not directly provide the load values corresponding to the onset of surface deformation of the materials.

The American Society for Testing & Materials provides codes and standards for Brinell Testing. ASTM E-10 is a standard test for determining the brinell hardness of metallic materials.[2] There is also an ISO 6506 standard defining the brinell test method.

Design Development

Discussion of Conceptual Designs

While it was established early on through discussions with Parker Aerospace that AISI 52100 would be the benchmark material, it was also concluded that it would most likely be the hardest material the fixture would have to test. With this maximum hardness set, this allowed us to use calculations based on known values from AISI 52100 to determine the maximum load that the fixture would be required to provide. From this maximum load calculation, analysis for selecting the necessary jack, load cell and deflection measurement sensors, and sizing of structural components were performed.

In order to calculate the maximum load at which samples would brinell, we used hertzian contact stress relationships. At the time of our first meeting with Parker Aerospace, they provided us with data related to the brinelling of a 1” diameter ball of AISI 52100 steel. While we will only be testing ball bearings up to ½” in diameter, we performed the hertzian calculations for a 1” ball to first confirm our calculations would produce the same values Parker’s data indicated. As we would also be entering these calculations into an EES (Engineering Equation Solver) program for simplicity of future manipulation of parameters, this initial calculation would allow us to verify the output of the program. The amount of load to brinell a 1” diameter ball of AISI 52100 was found, by hand calculations and by the output of our EES program, to be about 850 lbs, thus agreeing with Parker’s data.

With our hertzian calculations and computer program verified, we altered the input parameters of the program in order to repeat the calculations for a ½” ball bearing of AISI 52100 steel, which produced a load to brinelling of 213 lbs. We modified our program and repeated these calculations to find the brinelling limit for a ½” diameter, ½” long cylindrical bearing of the same material. This produced a much larger value of 7613 lbs. As this far exceeded the load we anticipated for spherical bearings, it would likely require two different loading systems to provide the accuracy desired in both loading ranges. After consulting with Parker engineers, we decided to reduce the maximum size of roller bearings we would test to ¼” diameter and ¼” long. After modifying these parameters in our EES code, a maximum load to brinelling of 1903 lbs was calculated. From these calculations we decided to set the maximum load our test machine would provide to 2000 lbs.

To determine the expected deformation of the samples, we utilized equations published in the National Standards Laboratory Technical Paper Number 25 for both the condition of a sphere between two plates and a cylinder between two plates. Our calculations on the deflection of the ¼” diameter roller bearing show a compression distance of about 0.0013”, so Parker directed us to find a displacement measurement device with a resolution of $\pm$0.00001". See Appendix A for loading and deflection calculations.

There are many companies that provide ranges of load cells in different styles. A pancake style load cell will work best with our application. The most cost effective jack would be to use a manual bottle jack, like when jacking up an automobile. Other jacking options were determined to be hydraulic or pneumatic, or either a manually or electrically powered ball screw jack. We also checked multiple building materials and sizes to choose the best design for the frame of the system. Strength, stiffness, and deflections calculations helped us ensure our machine was designed appropriately and safely.

In this and the following paragraphs, a number of initial designs for the overall system and for individual components are presented. Figure 3 shows the initial design presented by Parker, which seems to be the best layout for this machine.

Figure 3. Original design provided by Parker.

The jack is located on the bottom, and presses a slide, with the sample plate and ball, into another sample plate. The load cell records the applied force. In this design, the load cell would be initially in tension so that brings to question if a compression only load cell can be used, or if a tension and compression load cell is needed.

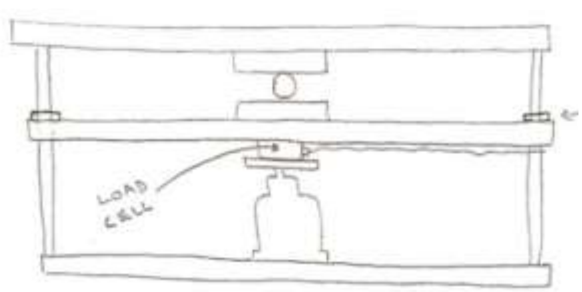

Figure 4. Load cell above jack ram.

The system depicted in Figure 4 is a modification of the original design provided by Parker. A key difference is that the load cell has been placed below the sliding table. This allows for both the top plate and the slide to have a “Peg Board” layout so that the sample plates can be moved for each load and the ball can stay loaded in line with the jack and load cell.

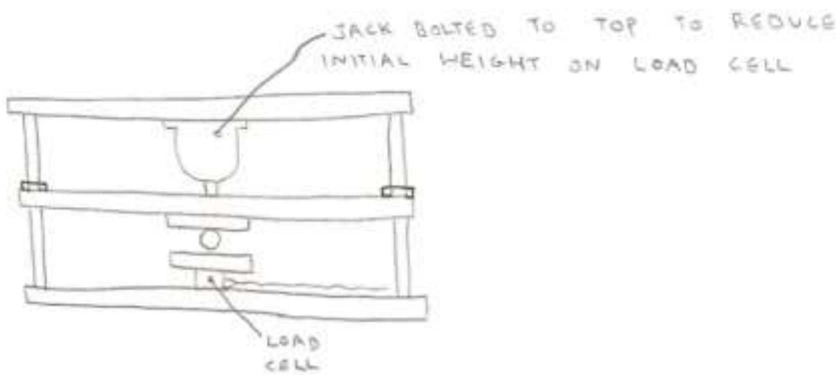

Figure 5. Load cell on base and jack on top.

The design sketch in Figure 5 has the system flipped, with the jack placed on top of the sliding plate. If the jack could somehow be attached to the sliding plate, then the load cell would have only a couple of pounds initially recorded from the weight of only one sample plate. The “Peg Board” layout, mentioned in the previous paragraph and appearing in detail in the following figure, Figure 6, would also be used in this design.

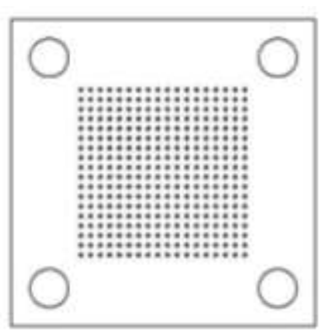

Figure 6. Example of peg-board system.

To provide explanation of some finer details, more exact portions of the system need to be addressed. Stickers can be made that can be applied to the sample plates that have the test coordinates permanently marked for future use. This would allow for consistent tracking of load values with respect to the position within the sample plate at which that load was applied. The sample plates can have tapped, or drilled, holes in the corners to allow the sample to be moved and located around in the peg board system among tests.

Another design that could be considered if the analysis can be worked out would be to use a two-post system, instead of four. This would mean that less material would be needed to purchase. Also, there would be two less linear bearings to purchase.

Flange-Mount Linear Bearings were considered early on as they would provide the best linearity for the system, but they also come at an increased cost. A competing consideration was that a simple bushing between the guides and the slide plate might perform equally well in this function. After discussions with Parker's engineers it was decided to only consider using flange-mount linear bearings. Figure 7 below shows the flange-mount linear bearing as it would be installed.

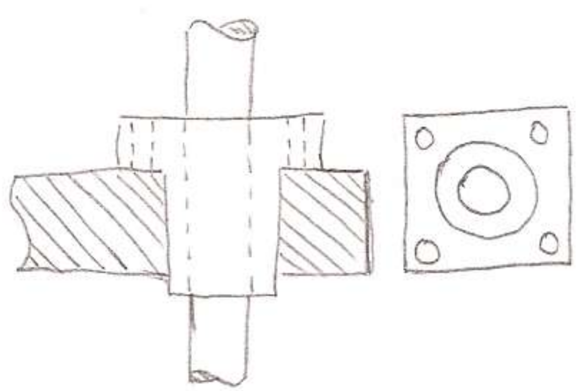

Figure 7. Flange-mount linear bearing drawing.

There are multiple displacement measuring devices readily available. Some instruments we have considered are: strain gage, digital dial indicator, linear actuator, laser, and lever arm system. The strain gages are cheap, but require many man-hours to setup on the system, and we would also need to find a way to have it correctly measure the distance between the plates. The laser system has a high level of accuracy, but comes at a high (but not prohibitive) cost. Many companies sell digital dial indicators and linear position sensors that easily output data to a laptop, while keeping a reasonable cost.

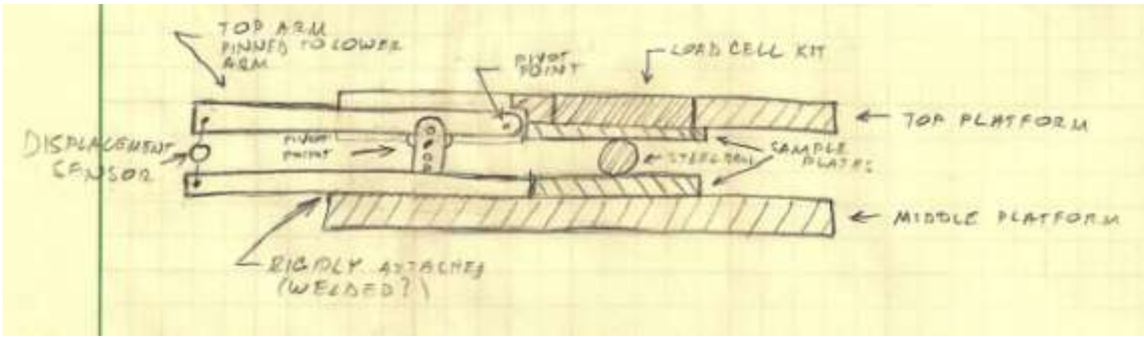

Figure 8. Lever Arm system for displacement measuring.

Figure 8 shows a design for utilizing lever arms to magnify the displacement between the two sample plates. This would help reduce the level of resolution necessary for our final measuring device. This system is very complicated, and requires very precise machining and installation. It will be much easier to have extra material on the sample plates where an indicator of some sort can poke through one sample plate, and measure the distance to the other sample plate. This is shown below in Figure 9.

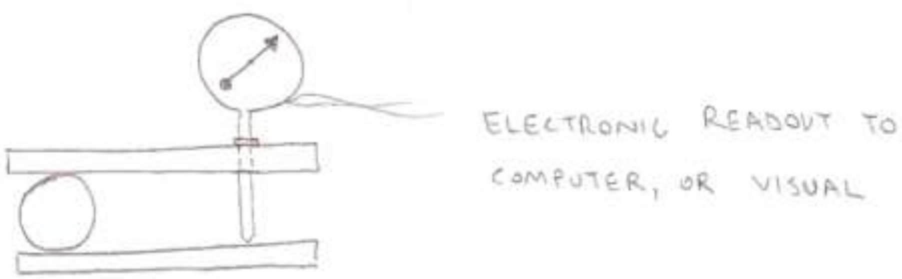

Figure 9. Dial Indicator displacement measuring.

After developing four complete designs, we held a Preliminary Design Review with our sponsor on Friday, November 20, 2009. The action items from this meeting consisted of contacting specific vendors for pricing and product delivery lead times and the performing of a handful of calculations. These calculations were thread stress and bolt torque calculations. A final action item from this meeting was to complete detailed drawings of each component. After each of these action items had been met, we held a Critical Design Review with Parker and a panel of the company's engineers on Friday, January 8, 2010. A list of action items were generated from this meeting as well. This time the list consisted of the modification of the design of a handful of components, the changing of the materials that certain components would be machined from, the completion of detailed drawings of each part with any necessary geometric tolerances, and the performing of a couple calculations. These calculations were to determine the overall apparatus' stability against tipping or excessive wobbling. With a few weeks devoted to addressing the issues that arose during the meeting, we were able to order commercial components and raw materials so that we could start machining and building our test machine. Once the machine was completed, we were able to begin validation of the text fixture and provide test data, along with a complete report, to Parker at the end of the Spring quarter.

Design Selection

We all came up with our own designs that we thought would meet the design criteria. The following paragraphs will give basic descriptions of the concept designs we generated for presentation to Parker Aerospace during our Preliminary Design Review.

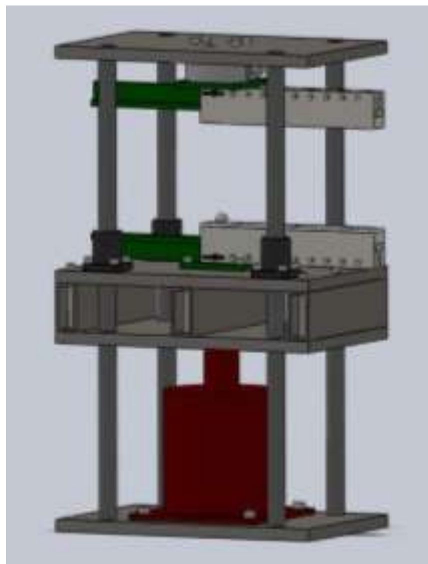

Figure 10. Design Concept 1 – Four post design with single sliding track.

The design in Figure 10 shown above consists of a four-post system, which has a top and bottom plate that is fixed. The purpose of the bottom plate being a large rectangular piece of steel is to prevent the machine from tipping over when applied with a significant load to the top of the machine. The middle sliding plate is able to move freely up and down with the motion of the jack shown in red. This sliding plate is composed of three rectangular tubes that are sandwiched by two large rectangular plates. The purpose of this sandwich structure is to add extra stiffness, since the rectangular tubing is stiffer in the shown orientation, to be sure that it will not fail under the loading conditions desired. The parts in dark grey are linear ball bearing flanges that will allow the sliding plate to slide in the vertical plane with low friction, as well as with low tolerances to minimize the angularity between the two column rods. The key component in this design is the sliding mechanism, which can be seen by the green and light grey parts. The green part is the sliding rails, which is directly mounted to the sliding plate as well as to the load cell, which is mounted to the top plate. The light grey parts are the sample plates, which are guided by the slide rails and have a series of holes where a quick release locking pin can be used to lock the sample plate in place at each sample location. This configuration will allow for the sample plate to be guided unidirectionally, providing simplicity and ease of operation. The quick-release locking pin will allow for the sample locations to be directly in the load path, which will reduce the risk of any eccentric loading. This design focuses on the sliding mechanism, and leaves the jack and load cell capabilities up for preference.

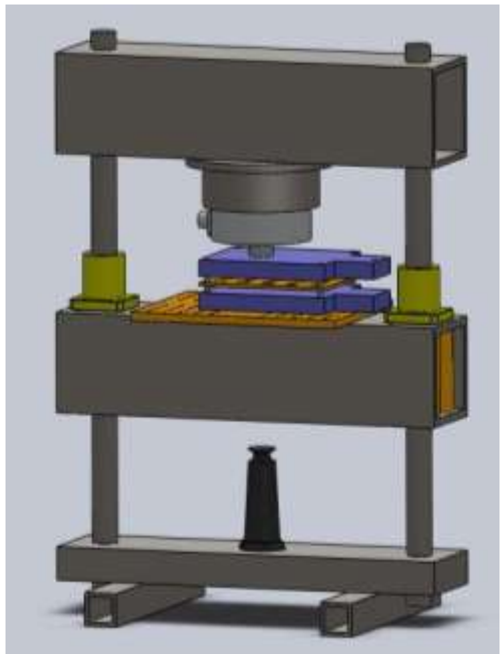

Figure 11. Design Concept 2 – Two-post design with peg-board sample plate locator.

The design shown in Figure 11 above consists of a simple two-post system, where the top and bottom plates are fixed, and the middle plate slides up and down as the jack loads or unloads the system. The jack is a miniature screw jack that can supply a 3-ton load, which can provide more than the required 2,000 lbs. Two lengths of square tubing are used to help give a wider and more stable footprint. There are two linear ball bearing flanges that will help guide the sliding beam up and down the vertical rods with minimal friction and angularity. Hanging from the top beam is a spacer plate, which provides clearance for the load cell so everything is not cluttered in a tight area at the top. This will provide more room for the operators' hands to work the machine.

The orange and purple components consist of the main operations for the sample area. The bottom orange plate consists of a peg board system, which is used to move the sample plates, so when a sample bearing is placed between the plates the load path will always be concentric. This reduces the opportunity for any eccentric loading, resulting in misleading measurements. There are holes in the corners of the purple sample plates, which allow a locating pin to keep the sample plates in position on the peg board system. A sample tray, the orange part between the two purple sample plates, will help keep the sample bearing in place. The extension on one of the sides of both sample plates allows a measurement recording device to measure the displacement between the approaching surfaces. Not shown, is a grid label will be affixed to the surfaces in test on the sample plates to keep records of which test loading was applied at those locations.

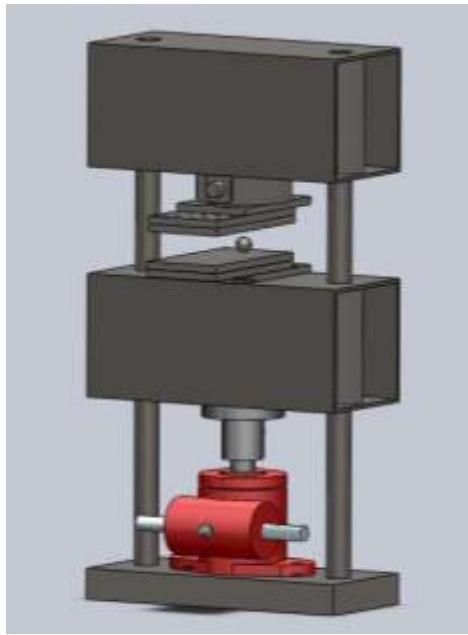

Figure 12. Design Concept 3 – Two post design with modified peg-board sample plate locator.

The design shown in Figure 12 above, like the preceding design, consists of a simple two-post system, where the top and bottom plates are fixed, and the middle plate slides up and down under the influence of a jack. The jack for this design is a two-ton ball-screw jack. Two linear bearings, modeled within the middle tubing, reduce friction in sliding and help alleviate any angularity that may arise during loading of the system. A spacer plate separates the load cell from the top tubing, providing clearance for positioning of displacement sensors without interfering with the motion of the apparatus. Eccentricity of the load path is maintained by positioning the bearing under the load cell for each sample taken. The specific positioning of the sample plates is achieved by numerous mating concave and convex holes machined into each sample plate.

Concepts from each of these three designs were combined together, resulting in the fourth design concept, described in the following section. A QFD house, shown in Appendix G, was set-up and through little knowledge of setting up one, we were able to see that overall our 4thconcept was the best choice. We had our Preliminary Design Review at Parker with five employees before Thanksgiving break. During this three-hour discussion, we learned that the engineers we worked with do not often use QFD's for their design selection. Instead we relied on in-depth discussions during the presentation to collaborate on the best part. By the end of the PDR, while a list of necessary action items had been generated, it was unanimously agreed that we should move forward with finalizing the fourth design.

Description of the Final Design

Detailed Design Description

Combining some of the initial concepts, we were able to obtain a design to meet all of the goals for the project. An image of the final design appears in Figure 13, below and detailed drawings of each of the components that will need to be manufactured for this apparatus can be found in Appendix C.

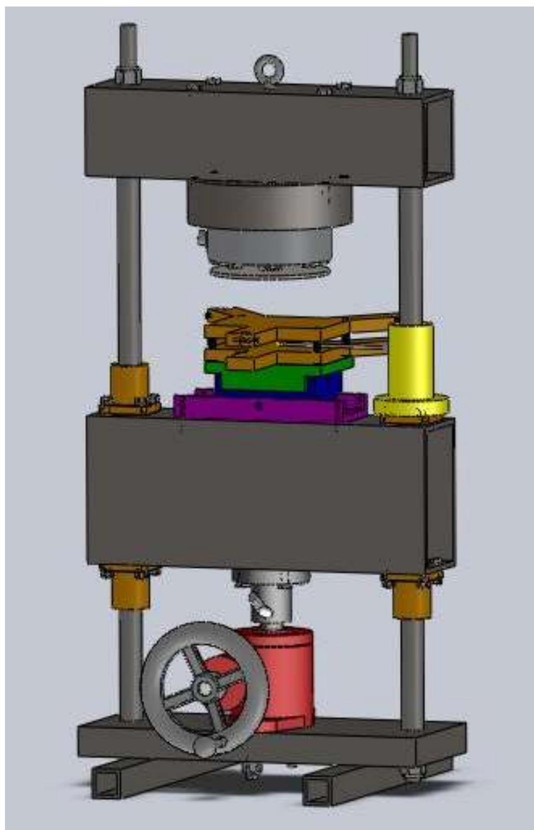

Figure 13. Overall apparatus.

The system has stiff rectangular tubing that will take bending loads with minimal stress and deflection. There is a spacer block to keep the load cell a certain distance down from the top rectangular plate to prevent it from interfering with the top rectangular plate. This allows more space to locate our measurement devices. The sample plates were designed with three extensions to adapt for three mechanical dial indicators. Before we decided to invest in highly accurate, expensive dial indicators, we tested for angularity of the sample plates by using less accurate dial indicators provided by Parker. We added the use of center mount flange ball bearings instead of the single edge mount ball bearings. This helped keep the system aligned better, and allowed it to carry more of a moment in case there is any eccentric loading. The ball screw jack from Nook Industries helps us obtain the required loads we sought with minimal friction and resistance. We also incorporated a lifting eye on the top rectangular tube which is to be used in helping transport the machine, which is a standard for anything over 50 lbs.

The main point to discuss is the sliding plate system, shown in Figure 14, to move the sample plates around and keep the sample ball concentrically loaded with the jack and load cell. We have two sets of tracks arranged 90 degrees from each other that allows us to move left to right as well as front to back. For illustrative purposes in Figure 14, the tracks allowing left to right movement are colored purple and blue, while the tracks allowing front to back movement are colored blue and green. The gold colored plates are the sample plates being tested. There are holes located along the tracks that allow us to lock the location of the sample plate, with a quick release pin, as the loading occurs.

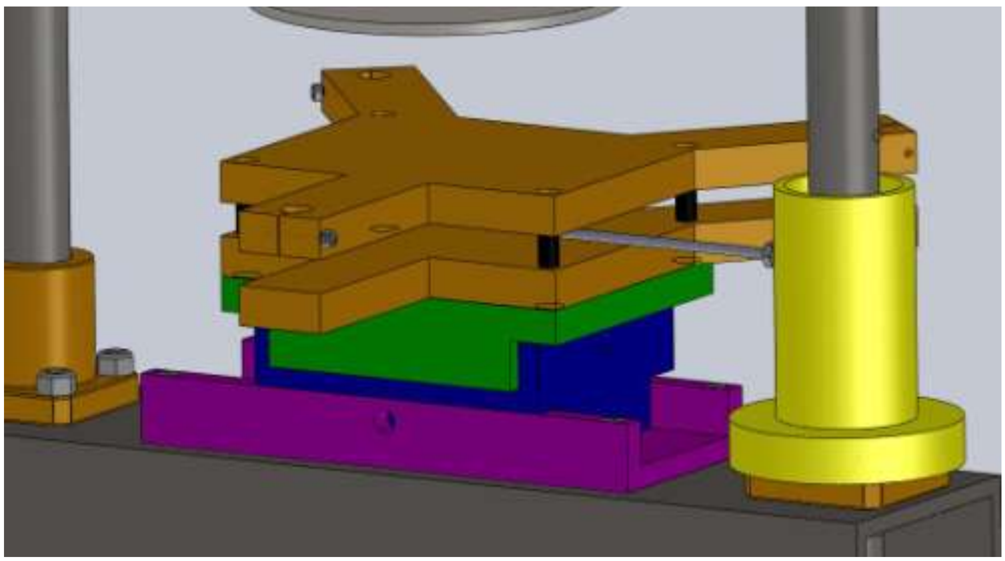

Figure 14. Close-up of sliding plate system and load cell area.

Another feature displayed in the preceding figure which is worth pointing out is the assembly designed to retain the sample ball or sample cylinder between the two sample plates. The purpose of this assembly is to ensure that the ball or cylinder remains concentric with the load path during positioning and repositioning of the sample plates via the track system. This assembly consists of the yellow flanged cylinder on the right and the connected silver colored arm. The silver arm is tipped with a strong magnet to which the sample ball or sample cylinder is attached. This part of the arm is hidden by the upper sample plate. The opposite end of the arm is threaded into the yellow flanged cylinder which is through tapped. The flange is positioned by pins (not visible) press-fit into rectangular tubing to ensure that the magnetic arm will properly secure the sample ball (or cylinder) in the load path.

Analysis

Loading and Deformation Calculations

In order to determine the load that the test fixture needs to supply, we performed analysis on the two cases of a sphere pressed into a flat plate (representing the ball from a ball bearing) and a cylinder pressed into a flat plate (representing a roller from a roller bearing) using Hertzian contact stress equations. When two bodies are in contact, Hertz theorized that the point force causes deformation of the two bodies and that the resulting area of contact can be related back to the deforming force. As reported in Shigleys[4], Hertz found these two values to be related by the following expressions,

$$F = 2\pi a^2 \frac{p_{max}}{3} \tag{1}$$

and

$$a = \left(\frac{3F}{8} \left(\frac{\frac{1-\nu_1^2}{E_1} + \frac{1-\nu_2^2}{E_2}}{\frac{1}{d_1} + \frac{1}{d_2}} \right) \right) \tag{2}$$

where F is the force between the two bodies and a is the radius of the resulting contact area. Also p_{max} is the maximum stress to brinelling, or permanent surface deformation of the material, and ν is Possion's ratio, E is the Modulus of Elasticity and d is the body diameter, for both bodies.

These values were checked by performing analysis on the same two cases using Roark and Young's equations for stresses between contact bodies. The actual value of the maximum load that would be needed was determined by following the specifications from Parker Aerospace that the hardest materials tested would be of AISI 52100 steel and that the largest ball diameter would be 0.5 inches and the largest cylinder dimensions would be diameters and lengths of 0.25 inches.

The equations for Hertzian contact stress analysis were compiled in Engineering Equation Solver, the formatted output of which is provided in Appendix A. The output from the Hertzian analysis in EES was checked by hand calculations using the Hertzian equations and again using the Roark and Young's equations, with photocopies of the Roark and Young equations also provided in Appendix A. The load calculated for the ball bearing on a flat plate through both methods was 213 lbs and for the cylinder on a flat plate was calculated by Herzian equations to be 1903 lbs and by Roark and Young equations[17] to be 1906 lbs.

During the initial weeks of this project, the Hertzian contact stress calculations were performed for only a ball on a flat plate with the ball having a diameter of 1.0 inch. This was to allow us to compare our calculated load value against published values for

AISI 52100 that were being referenced by Parker Engineers. When checking these first load calculations by Hertzian equations as entered in Microsoft Excel and checked by hand, with Parker engineers, they initially indicated that our calculations had to be incorrect because they were less than their published values by an approximate factor of ten. After this was communicated, we repeated our hand calculations, entered the Hertzian equations into EES and performed roughly two weeks of research, reading through numerous technical papers on methods of determining, specifically, the Brinelling limits of ball bearings or, generally, the deformation of contacting bodies. After these two weeks of being in a veritable design freeze as we tried to determine why we continued to get load values that remained in the same range of values regardless of the methods used, we checked back with Parker's engineers. During this follow-up teleconference, Parker's engineers communicated that they had realized an error in the published values they had been referencing and that they now agreed that the load value initially reported through Hertzian analysis had been correct. With our calculation methods validated, we proceeded to repeat our calculations for the loads required to bring a 0.5 inch ball and a 0.5 inch diameter and 0.5 inch length cylinder to the onset of Brinelling and further checked these values against the loads we calculated through Roark and Young analysis.

Using Hertzian contact stress analysis for the cylinders, the equations changed a bit from those used for the ball on a flat plate. Again from Shigley's [4], the equations are as follows,

$$F = \frac{\pi b l\, p_{max}}{2} \tag{3}$$

and

$$a = \left(\frac{2\frac{\pi l\, p_{max}}{2}}{\pi l} \left(\frac{\frac{1-\nu_1^2}{E_1}+\frac{1-\nu_2^2}{E_2}}{\frac{1}{d_1}+\frac{1}{d_2}} \right) \right) \tag{4}$$

With the load values known, we calculated the total deformation of the materials for both cases. This allowed us to determine the resolution of deformation sensors and provided a guide in determining required stiffness of the apparatus, as any appreciable deformation of the structural members of the fixture could not be allowed to occur in such a manner that they would affect the accuracy of the deformation measurements. For the calculations of the deformations, equations presented in National Standards Laboratory Technical Paper No. 25[13] were used. For the ball on a flat plate, we calculated a deformation of 0.0007136 inches using

$$\delta_{max} = \left(\frac{9}{2}\right)^{\frac{1}{3}} \left(\frac{1-\nu_1^2}{E_1}\right)^{\frac{2}{3}} F^{\frac{2}{3}} \left(\frac{1}{d_1}\right)^{\frac{1}{3}} \tag{5}$$

and for the cylinder on a flat plate, we calculated a deformation of 0.0011223 inches using

$$\delta_{max} = 2\frac{F}{l}\left(\frac{1-\nu_1^2}{\pi E_1}\right)\left(1+\ln\left(\frac{l^2}{\left(\frac{1-\nu_1^2}{\pi E_1}\right)\frac{F}{l}d_1}\right)\right) \tag{6}$$

Sizing of Structural Members

With the maximum required load values and deformation values determined, we were able to begin sizing the structural components. Since the load to brinell the cylinder was significantly higher than for the ball, this was the value chosen to size the fixture's structural members against. With this value of 1906 lbs as calculated through Hertzian equations, we decided to increase the required value to a round 2000 pound value and make all design calculations off of that value.

For the two vertical column rods, given a design constraint of concentric loading of the sample materials, no axial loads will be experienced by these members. Therefore, only normal stress calculations were performed on various sizes of steel solid rods using the equations for normal stress as presented in Shigley's[4]

$$\sigma = \frac{F}{A} \tag{7}$$

where it is worth noting that the force, F, is one-half the applied load of 2000 pounds in each rod. The initial EES program compiled and the eventual Excel spreadsheet used to quickly calculate the stresses and strengths of a number of different rod sizes are both located in Appendix A.

For the horizontal members, the fixed top beam to which the load cell was attached and the sliding middle member which was positioned by the jack, more in-depth efforts needed to be taken to ensure proper sizing. Modeling the rectangular tubing lengths as simply supported beams under point loads, the top member was able to be sized to support the loads that would be transmitted from the jack, through the sample materials and into the rectangular tubing. This method was also used for the middle member, however an additional step was taken to determine if the bottom section of the tubing would buckle under the applied load from the jack. By modeling the bottom section of the middle tubing as a flat beam with a length equal to the width of the tubing and a width equal to the tubing's length and a thickness of only the tubing material thickness and repeating the bending calculations, no reasonable beam thickness was found to prevent significant bending in the lower section of this member. The Excel spread sheets used to expedite these calculations can be found in Appendix A.

In order to support the transmission of the load from the jack through the rectangular tubing, a crush tube was designed to be placed inside of the rectangular tubing, between the jack and the sample plates. This crush sleeve was analyzed using J. B. Johnson's equations for short member buckling. Our initial intention for securing the linear bearings to the middle tubing was to run the bolts the entire depth of the tubing. To prevent buckling of the tubing due to the tension in the bolts, similar crush tubing members were designed. Upon further evaluation, two design changes were made. First it was decided to only secure the linear bearings with bolts short enough to clear the linear bearing flange and the tubing material thickness, and a bolt head, plus enough extra threads to conform to standard shop practices. The replacement of long bolts with short bolts eliminated the need of the crush sleeves. The second change was to use a total of four linear bearings, two secured to the top of the middle tubing and two bolted to the bottom of the middle tubing. While the tubing designed for the linear bearing bolts was no longer needed, we found it to be sufficient to support the total load being transmitted across the top rectangular tubing. This load is the sum of the preload in the bolts that fasten the load cell in the top rectangular tubing the transmitted load through the sample materials.

The bolts for fastening the linear bearings to the sliding rectangular tubing were designed concurrently with the crush sleeves. A bolt was needed to fasten the bearings, but no significant loads would be placed on them such that the joined bearings and tubes would be pulled apart. The bolts also needed to fit in the through-holes machined into the bearings and the load cells. By picking bolt sizes that would fit these holes and comparing the total preload from tightening these bolts to the total load under which crush sleeves of various sizes would not buckle, we were able to determine the optimum sizing for both the bolts and the crush sleeves. The spreadsheets used to determine the necessary strength and sizing of the crush sleeves against short member buckling, to determine the bolt strengths, and to confirm that the bolts and crush sleeves designed would also function well when used to fasten the load cell can be found in the near the final parts of Appendix A.

Validation of Structural Stability of Apparatus

One area of concern that was brought up during our CDR with Parker Aerospace was uncertainty of the stability of the test apparatus against light jostling. This concern was addressed through two separate calculations. This calculation can be found in their entirety as the last 3 pages of Appendix A. The first calculation performed was to model the entire apparatus as a rigid body, simply supported at the base. Then a force was applied to the top of the apparatus, horizontally and in the direction of the shortest dimension of the apparatus' footprint. Using a simple free-body diagram and a summation of moments equation, the greatest force that the model could support against tipping, as currently designed, is 20.41 lbs. This value was reported back to the engineers at Parker who stated that 20 lbs was sufficient, therefore no modifications were made to the machine.

The second calculation that was performed to verify the stability of the apparatus was to determine its natural frequency. This would give us insight into the likelihood of the apparatus being excited to dangerous oscillatory amplitudes due to human interaction. For this calculation, the oscillation of the half-model was found in the plane that the force from the previous calculation was applied in. The half-model was modeled as a vertical mass-less rod pinned to the ground with a torsional spring on its base and a point mass on the opposing free end. The torsional spring has a spring constant equivalent to the spring constant of the stainless steel vertical rod of the apparatus. The point mass has a value equal to the equivalent mass of the half-model. From these calculations, the natural frequency was found to be 6.9 cycles per second. While this frequency is not very fast, it should be fast enough for the apparatus to be stable under normal interactions.

Safety Considerations

While designing this machine, we had to make many considerations to be sure that it would be safe for the operator as well as any bystanders. We have completed a thorough analysis on every member in the machine to be sure that it will be stable and not fail in any manner. One of the main concerns with the machine during operation is whether there would be any repercussions if the ball bearing were to be eccentrically loaded or if the bearing would fail.

If the bearing would happen to be eccentrically loaded then there is a chance that the machine could send it shooting out the side of the machine in any direction based on how it was eccentrically loaded. To be sure that eccentric loading does not occur, we will be able to position the ball in the direct line of the load path to avoid any eccentric loading. We have also considered the analysis of eccentric loading on the bearing and have found that there would be minimal risks of the ball shooting based on the angularity.

The other concern with the possibility of the bearing failing was whether it would shatter or have any projectiles that would compromise the safety of the operator. In researching this issue, we found that the failure modes of the bearings would be due to excessive loading or overheating. Neither of these conditions are expected to occur as the loading will be low relative to what would be required to shatter a bearing and would be static, eliminating the possibility of heating, much less overheating.

In the case of excessive loading if the bearing is loaded passed the rated capacity then it will lead to spalling in the bearings. Spalling describes the process of surface failure in which spall, flakes of a material that are broken off a larger solid body, is shed as shown in Figure 15.

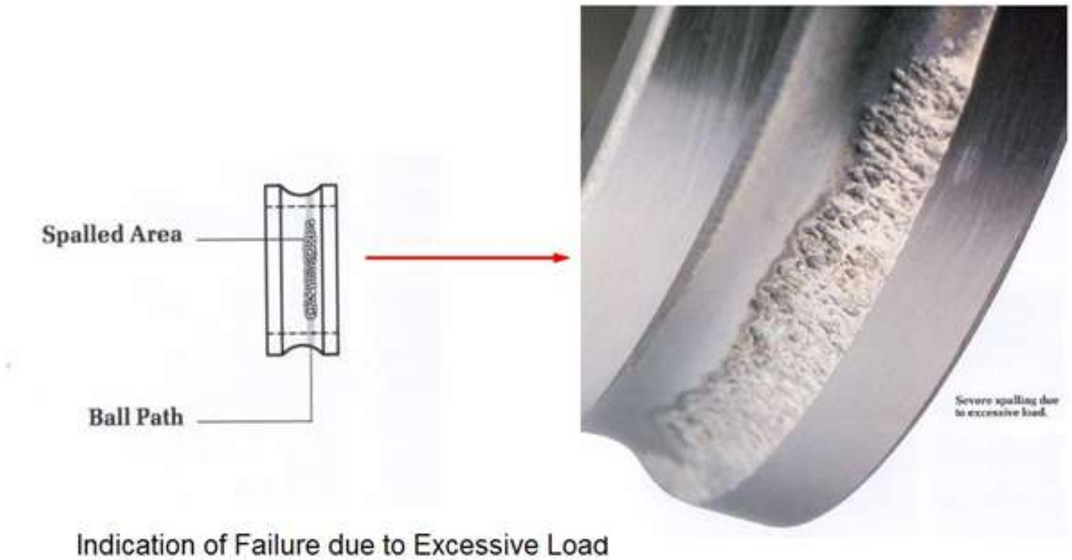

Figure 15. Spalling due to excessive load on the bearings.[8]

In the case of excessive heating of the bearing, it will tend to anneal (deprive of its hardness) the bearing when it operates around 400 degrees Fahrenheit. Once the ball loses its hardness, its load carrying capability also reduces because of the softness of the ball. Since a thermal chamber is a goal for this project we will be testing up to 400 degrees Fahrenheit where we might encounter the discussed overheating problem. We feel that the thermal chamber can in a way also provide as a safety enclosure that will prevent any form of projectile that could endanger the operator.

Material Selection

In considering materials to be used for the Brinell Limit testing machine, we tried to keep in mind cost and material properties. The material chosen will be used for the base plate, sliding plate, top plate, and columns. We tried to figure out the best geometry of the plates and columns to obtain a secure machine. We found that standard steel stock will be sufficient enough to give us the material properties that we need. Although standard steel would work for the columns, Parker engineers requested we go with a 440C stainless steel stock for its resistance to corrosion. Using the material properties of steel, we analyzed each part of the system to allow us to choose what geometries would be sufficient. We decided that we could use aluminum for the flange spacer, sliding tracks, and the spacer plate since this would lead to greater ease in the machining process. We found that instead of using just flat plates we could use a rectangular tube that would give us a better stiffness. To prevent any buckling of these tubes, we will be using crush sleeves inside to reinforce it. We have also analyzed the columns for buckling and found that a 0.75” diameter rod will suffice for the structure of our machine. All of the supporting analysis can be seen in Appendix A. Due to the calculations shown in Appendix A, it can be seen that SAE grade 5 hardware will be strong enough to fasten the machine.

Maintenance and Repair Considerations

We will be providing Parker Aerospace with detailed drawings, presented in Appendix C, as well as the Bill of Materials, in Appendix B, so that they will be able to make any repairs to the machine. They will have to do visual inspections of the overall structure of the machine to maintain a safe operating environment. Based on the frequency of operation, the bolts and nuts will need to be checked for proper torque every couple months. They will have to do calibrations of the load cell and displacement measurement device based on the manufacturer’s specifications. If we have extra material in the end of the fabricating process we can use the excess to make extra parts so that they can simply swap them out if one fails.

Cost Analysis

Parker Aerospace allocated a budget of $10,000 to this project. Our goal was to build the machine with the highest quality of components, while keeping cost to a minimum so that we, and our main contact, stay well under budget. Very little of our overall costs came from material and hardware selection. We also machined all of the non-commercially available components ourselves to save labor costs, except for the sample plates for which Parker offered their machining resources.

As seen in the Bill of Materials (BOM) in Appendix B, the main cost drivers were the sensor components. The load cells were purchased from Futek, which is local to Irvine. We compared Futek’s load cells to Transducer Techniques load cells, which were about $300 less than Futek. Both companies had pros and cons such as geometry, cost, and setup. We purchased and used two different load cells because the cylinder and sphere loads vary greatly, and the accuracies of the load cells come from a percentage of the total rated load. Transducer Techniques load cells had two different bolt patterns, while Futek had the same bolt pattern between the load cells. This eased the cost of machining, as well as lowers confusion of interchanging load cells while trying to line up the bolt holes. Futek has a higher overall cost for everything to obtain data to be logged onto a laptop, but they only have one simple USB plug instead of having to buy a couple of extra cables and an external readout. A description of the Futek sensor and related company literature can be found in Appendix F.

The displacement measuring devices were provided by Parker Aerospace. After our PDR, it was agreed that we would be unable to predict the required resolution for measuring the change in the distance between the plates until we had actually tested. Also, while our design goal was to keep the sample ball or cylinder concentric to the load path, the slightest eccentricity may cause angularity in the sample plates, with would affect the recorded measurements. Because of this possibility, we designed the displacement section to accept mechanical dial indicators with a resolution of .0001”, supplied by Parker. The initial tests were intended to incorporate three dial indicators, separated by 120°, to allow for averaging if slight angularity occurs. After the initial tests, it was planned that we would decide if three indicators or fewer will be necessary, and then what type of resolution will be needed in future testing. A list of various appropriate sensors is compiled in the BOM should different resolutions be deemed desirable.

The only other reasonably high cost item was the jack. There were many types of jacks available, but it was ultimately decided that a Nook Industries Ball Screw Jack with a cost of just under $750 would be purchased. Its gear ratio requires 25 turns to raise the jack 1” which allows the operator to easily narrow in on specific load values. It only takes about 10 in-lbs to supply the 2,000 lb load so a mill-like handle was adapted for the apparatus to help turn the worm gear driven jack. We stayed away from hydraulic jacks because of previous experiences with similar testing; it was too hard to obtain specific load values. We have also stayed away from pneumatic jacks because they require additional parts that can fail, they would produce high air pressure safety concerns, and the ease of portability would be hindered due to the necessity of air line connections.

All parts and the quantities needed and associated costs can be found in the Bill of Materials located in Appendix B. The total project cost includes all costs for commercial components, raw materials for all machined parts, and all hardware components. Shipping costs and tax have been included as well, resulting in a final cost of $5,339.27. This value is well below our budget limit of $10,000.

Product Realization

Manufacturing

Our project team performed nearly all of the machining using the capabilities available on campus. This was done to save on labor costs and to provide each team member with the opportunity to gain experience in machining practices. While Parker offered to machine any parts with specifications beyond our immediate capabilities, we only utilized this option for the machining of the sample plates as the hardness of the materials to be used for the apparatus verification would have proven significantly difficult and time consuming. The jack, linear bearings, hardware, sensors and instrumentation were purchased from various vendors based on their capability to accommodate the specific accuracies that this project requires, while the raw materials were purchased from various vendors based on price and availability of material types and sizes.

Various shop practices were learned and utilized by our team to properly and safely manufacture the components needed for this project. The primary tools used throughout the entire manufacturing process were a mill and a lathe. With the mill, parts were fly-cut to length, channels were cut using end mills, and holes were drilled along linear as well as circular patterns.

Figure 16. Milling of the track system

For parts requiring holes in circular patterns, the part was often secured in a rotary chuck to increase the accuracy and speed of the hole placement. On the mill, parts were parted or faced to length, turned down to specified diameters, and chamfers. Other various tools were used throughout the manufacturing process. For gross material separation, metal chop saws, and vertical and horizontal band saws were used. The base of the apparatus required some welding and the campus CNC machine was utilized to machine the geometrically complex dial indicator plates. Grinding wheels, dremels, and files were also utilized for various tasks.

Figure 17. Dial indicator plate being machined on a CNC mill and base plate after welding.

It was of the utmost importance to adhere to all tolerances indicated in the detailed drawings developed. This ensured that all components aligned properly, in turn ensuring that the load path would be kept concentric. The assembly process for this machine was fairly simple. In order to ensure that all components are properly aligned, assembly began with the base components with each additional component being added on top of the previous.

Assembly

For detailed assembly instruction, refer to Appendix D.

Figure 188. Final assembly of brinell limit test machine.

Design Verification

Validation was intended to be achieved by loading 1/2" balls and ¼" rollers between AISI 52100 steel plates as well as between 440C steel and comparing the loads at which visual deformation was detected to the loads calculated. All plates were heat treated to a hardness value of approximately 60 and ground to a smooth finish.

The basic testing procedure for this apparatus was to insert test plates and either a ball or roller into the sample area and attach all measuring equipment. Then the ball screw jack was raised up until the load cell indicated the desired load had been reached. After obtaining all required measurements, the jack would be lowered until the sample area was no longer under load, and then the samples would be moved along the tracks until the next sample location was concentric to the load path. Then the jack would be raised again until the next load was obtained and the process continued until all empty sample locations had been filled.

The original test procedure developed by our team for Parker's use was compiled with the intent of determining the load limit as precisely as possible. To achieve this goal, the top row would be tested, incrementing from a load well below the expected brinelling limit for the material at coarse increments to a value above the expected limit. The sample plate would then be removed and each test site would be visually inspected for brinelling. The next row would then be tested at finer increments about the value at which brinelling was found in the previous row. This process would be repeated until the last row, whereupon the increment value would be small enough to precisely determine the brinelling limit.

While this was the intended test procedure, the first full sample plates were tested at equal increments, starting significantly below the expected brinelling limit in the first row, first column, and ending significantly above the expected brinelling limit in the last row, last column. The ball was tested over a range from 50 to 575 lbs, loaded at 25 pound increments, with an expected Brinelling load of 213 pounds. The roller was tested over a range from 1050 to 2000 lbs, loaded at 50 pound increments, with an expected Brinelling load of 1975 pounds.

Both ball and roller test results resulted in the appearance of visual deformation for nearly all load values. Both the ball and roller tests exhibited visual deformation for all load values over 50 pounds. As this did not match our initial expectations, rather than trying to narrow in precisely on the loads when deformation first began, subsequent testing was performed in a similar manner. That is, the plates were loaded at coarse increments, and rough approximations of the brinelling limit were visually determined.

With the only definition for the Brinelling load limit being "the onset of deformation", the presence of visible deformation for loads significantly below the calculated values was concerning at first. In order to prove that the machine still provides values consistent with theory, the resulting marks were assumed to be the total area of contact during loading. Since this area of contact was initially calculated, we measured the diameter of the resulting mark to compare with the calculated diameter. Since the area of contact produced by the machine matched the calculated areas of contact, it can be concluded that the machine does provide loading data consistent with theory. Further explanation for the discrepancy that arose in the visual appearance of brinelling is presented in the conclusion.

Conclusions and Recommendations

The goal of the project was to design, fabricate, assemble, and validate through testing a machine to determine the loads at the onset of Brinelling for different metals and would allow for multiple measurements to be taken from each set of sample materials tested. Some of the secondary design requirements were for the test fixture to be portable, small enough to be used as a desktop unit, be able to accommodate a thermal chamber around the test area, and also provide measurements of the total deformation of the sample materials when under load. Finally, time allowing, Parker Aerospace requested that the senior project team devote the last part of the last quarter to using the machine to provide data for a number of materials that they will provide.

To meet these requirements, our team invested the first few weeks of the first quarter to researching current market Brinell test machines and studied currently accepted theory on the calculation, testing and detection of the onset of Brinelling. After the research phase, each team member independently developed concepts for a machine that would meet these requirements. After the individual concepts were developed, the team collaborated together, incorporating the strong aspects of each design into one master design, which was approved in a design review with Parker engineers during the first quarter and modified to better meet the project criteria over several meetings and teleconferences throughout the second quarter.

After the concept had been approved, all main components were designed using extensive calculations to ensure longevity of the machine and safety against sudden failure during the first quarter and the beginning of the second quarter. The entire design was built into a solid model and drawings for each component were developed. The majority of the second quarter was spent modifying these drawings to Parker's specifications, using geometric dimensioning and tolerancing. Also during the second quarter, vendors were established for the provision of the raw materials that were to be machined during the third and last term of the project.

With the majority of the drawings approved by Parker engineers, our team ordered materials at the start of the third term and began machining on campus. Over the course of the term, drawings were further updated and modified, either by direction from Parker, or as issues were encountered during the manufacturing. In nearly all cases, the parts were machined in order from the base up, allowing for the apparatus to be assembled in tandem with manufacturing. This resulted in being able to start testing as soon as manufacturing had been completed.

During testing, brinelling was perceived to occur for both ball and roller tests at all load values over 50 pounds. From discussions of our test data with Parker engineers, it was determined that the appearance of surface deformation was due to the rough surface finish of the sample plates. Since the plates had only been ground to a mildly smooth finish, the perceived deformations at low load values were concluded to be due to the ball or roller deforming the microscopic peaks of the finish. A repeated visual inspection of the sample plates resulted in our team agreeing that a faint distinction could be seen between load sites where only the roughness of the surface finish had been deformed and where full brinelling had occurred. Parker engineers believe that running the tests again on sample plates with a well polished finish will provide better results. This will allow for the original test plan to be followed and the actual brinelling load limit to be precisely determined

After working closely with our sponsor, we feel that we have thoroughly considered and adequately met all of the main specifications required in this project and are providing a quality end product. Additionally, the final product meets a handful of the secondary design specifications. The apparatus, while heavy, has been outfitted with an eye-hook, allowing for the entire unit to be safely lifted and transferred via a portable jack, meeting the secondary requirement of making the test apparatus portable. With a 12 inch by 10 inch footprint and a height of approximately 25 inches, the test apparatus easily meets the secondary requirement of being usable as a desktop unit. Also, given the small body, the addition of a thermal chamber to the apparatus would require minimal time and effort from Parker engineers, meeting yet another of the secondary requirements. With all of the analysis that went into the various components of our project, our team is confident that the machine will be stable and safe for the operator at the maximum load cases possible with plenty of margin for error. A complete Bill of Materials has been developed, consisting of all apparatus components and associated pricing. The apparatus and all relevant detailed part drawings are prepared and ready for delivery to Parker Aerospace.

Only two of the listed project requirements remain unmet by the efforts of this team. We have been unable to correlate the recorded displacement measurement data to the total deformation of the sample materials under load. Also we were unable to run any tests of additional materials. It is worth noting that the first unmet requirement is a secondary requirement, and does not affect the completion of the main requirements as set forth by Parker Aerospace. Additionally, the second requirement not met was intended to only be performed if time allowed. Parker engineers are currently in discussions with Cal Poly's engineering department to continue testing and analysis as another senior project. Since we were unable to perform the mentioned additional tests due to time constraints, this unmet requirement again does not detract from a finished project.

Some recommendations from our team to better improve the design and function of this machine are included below.

- Devise correlation between measured dial indicator locations and actual compression distance at the sample location. Once an accurate means of performing this correlation is established, purchasing of digital dial indicators with USB connectors for logging of data directly to a computer. This team considered Chicago Dial Indicators LP3600 suitable for this application.

- Develop a locking mechanism for the jack wheel to hold loaded apparatus stationary. Another option is to research other jacks or jack screws that are not back-drivable.
- After purchasing electronic dial indicators, a program (such as WinWedge) along with a USB junction box would aid in real-time data logging. This will allow for real-time analysis of compression distances as a function of the exact load being applied.
- Purchase of a permanent marker, or labels, for labeling the Test Plates and locating sample brinell markings to prevent smudges and losing information.

Appendix A

Analysis Support

Cal Poly Brinelling Calcs 091206

Equations

Calculation of applied force required for Brinelling of single 1 inch

diameter AISI 52100 steel ball on a flat steel plate

Prepared by Mike Jaskulsky, Joe Cloutier, and Josh Kessler, Cal Poly, SLO

For Eric Polcuch, Parker Aerospace

November 6, 2009

Parameter List

Maximum Hertz contact stress; quoted by Parker

$$p_{max} = 570\ [\psi] \cdot 10^3 \tag{1}$$

Poisson's Ratio; Shigley's Table ,,,,

$$\nu_1 = 0.3 \tag{2}$$

$$\nu_2 = 0.3 \tag{3}$$

Modulus of Elasticity; Shigley's Table ,,,,

$$E_1 = 30.5\ [\psi] \cdot 10^6 \tag{4}$$

$$E_2 = 30.5\ [\psi] \cdot 10^6 \tag{5}$$

Diameter of steel ball

$$d_1 = 0.5\ [\text{in}] \tag{6}$$

Diameter of opposing contact surface; Infinity for flat plate

$$d_2 = 1\ [\text{in}] \cdot 10^{20} \tag{7}$$

Length of roller bearings

$$l = 0.5\ [\text{in}] \tag{8}$$

Determining applied force required for Brinelling single ball on flat plate

Governing Equations

Maximum pressure at center of contact area; solved for applied force; Shigley's pg 117

$$F_a = 2 \cdot \pi \cdot a^2 \cdot p_{max}/3 \tag{9}$$

Radius of contact area; Shigley's pg 117

$$a = \left((3 \cdot F_a/8) \cdot \left(\frac{\frac{1-\nu_1^2}{E_1} + \frac{1-\nu_2^2}{E_2}}{1/d_1 + 1/d_2} \right) \right)^{1/3} \tag{10}$$

Determining Maximum Deflection of ball

Governing Equation

$$\delta_{max} = \frac{a^2}{d_1/2} \tag{11}$$

$$\delta_{max2} = (9/2)^{1/3} \cdot \left(\frac{1-\nu_1}{E_1}\right)^{2/3} \cdot F_a^{2/3} \cdot (1/d_1)^{1/3} \tag{12}$$

Determining applied force required for Brinelling single cylinder on flat plate

Governing Equations

Maximum pressure at center of contact area; solved for applied force; Shigley's pg 119

$$F_b = \frac{\pi \cdot b \cdot l \cdot p_{max}}{2} \tag{13}$$

Radius of contact area; Shigley's pg 118

$$b = \left(\frac{2 \cdot \pi \cdot l \cdot p_{max}/2}{\pi \cdot l} \cdot \left(\frac{\frac{1-\nu_1^2}{E_1} + \frac{1-\nu_2^2}{E_2}}{1/d_1 + 1/d_2}\right)\right) \tag{14}$$

Solution

$a = 0.01336$ [in]	$b = 0.01701$ [in]	$\delta_{max} = 0.0007136$ [in]	$\delta_{max2} = 0.0005991$ [in]
$d_1 = 0.5$ [in]	$d_2 = 1.000 \times 10^{20}$ [in]	$E_1 = 3.050 \times 10^7$ [lbf/in^2]	$E_2 = 3.050 \times 10^7$ [lbf/in^2]
$F_a = 213$ [lbf]	$F_b = 7613$ [lbf]	$l = 0.5$ [in]	$\nu_1 = 0.3$
$\nu_2 = 0.3$	$p_{max} = 570000$ [lbf/in^2]		

Sphere

1) $\sigma_{c\,max} = .616\sqrt[3]{\frac{PE^2}{K_D^2}}$

since $K_D = D_2$

$$\sigma_{c\,max} = .616\sqrt[3]{\frac{PE^2}{D_2^2}}$$

solve for P = total Load

$$\left(\frac{\sigma_{c\,max}}{.616}\right)^3 = \left(\sqrt[3]{\frac{PE^2}{D_2^2}}\right)^3$$

$$\left(\frac{\sigma_{c\,max}}{.616}\right)^3 = \frac{PE^2}{D_2^2}$$

$$\left(\frac{\sigma_{c\,max}}{.616}\right)^3 \frac{D_2^2}{E_2^2} = P$$

$$\left(\frac{570{,}000\ \frac{lb}{in^2}}{.616}\right)^3 \frac{(1\,in)^2}{(30{,}500{,}000\ \frac{lb}{in^2})^2} = P$$

P for 1" Dia ball = 851.69 lb
P for .5" Dia ball = 212.9 lb

$$a = .881\sqrt[3]{\frac{PK_D}{E}}$$

since $K_D = D_2$

$$a = .881\sqrt[3]{\frac{PD_2}{E}}$$

$$a = .881\sqrt[3]{\frac{(212.9)(.5)}{30{,}500{,}000}}$$

a for .5" Dia ball = .01336 in
a for .25" Dia ball

$$y = 1{,}55 \sqrt[3]{\frac{P^2}{E^2 K_D}}$$

since $K_D = D_2$

$$y = 1{,}55 \sqrt[3]{\frac{P^2}{E^2 D_2}}$$

$$y = 1{,}55 \sqrt[3]{\frac{(212{,}9)^2}{(30{,}500{,}000)^2(.5)}}$$

y for .5" Dia ball $= .000713$ in

Cylinder

2) $\sigma_{cmax} = .591\sqrt{\frac{pE}{K_D}}$

$p = \frac{P}{L}$ and $K_D = D_2$

$\sigma_{cmax} = .591\sqrt{\frac{PE}{LD_2}}$

$\frac{\sigma_{cmax}}{.591} = \sqrt{\frac{PE}{LD_2}}$

$\left(\frac{\sigma_{cmax}}{.591}\right)^2 = \frac{PE}{LD_2}$

$\left(\frac{\sigma_{cmax}}{.591}\right)^2 \frac{LD_2}{E} = P$

$\left(\frac{570{,}000}{.591}\right)^2 \left(\frac{(.5)(.5)}{30500000}\right) = P$

P for .5" Dia/.5" Length = 7624.56 lb
P for .25" Dia/.25" Length = 1906.14 lb

$b = 2.15\sqrt{\frac{pK_D}{E}}$

$p = \frac{P}{L}$ and $K_D = D_2$

$b = 2.15\sqrt{\frac{PD_2}{LE}}$

$b = 2.15\sqrt{\frac{(7624.56)(.5)}{(.5)(30{,}500{,}000)}}$

b for .5" Dia/.5" L = .03399 in
b for .25" Dia/.25" L = .01699 in

$$\Delta D_2 = \frac{4p(1-\nu^2)}{\pi E}\left(\frac{1}{3} + \ln\frac{2D}{b}\right)$$

$$p = \frac{P}{L}$$

$$\Delta D_2 = \frac{4P(1-\nu^2)}{\pi E L}\left(\frac{1}{3} + \ln\frac{2D}{b}\right)$$

assuming $D = D_2$

$$\Delta D_2 = \frac{4(7624.56)(1-.3^2)}{\pi(30{,}500{,}000)(.5)}\left(\frac{1}{3} + \ln\frac{2(.5)}{.03399}\right)$$

ΔD_2 for .5" Dia / .5" L $= (5.7929056 \times 10^{-4})(3.715) = .00215$ in

ΔD_2 for .25" Dia / .25" L $= (2.896452815 \times 10^{-4})(3.715) = .001076$ in

Column Deflection

Equations

Calculation of Stresses, Strains, and Deflections of a Steel Tube and a Solid Steel Cylinder
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Parameter List

Modulus of Elasticity

$$E = 30 \cdot 10^6 \ [\mathrm{lb/in^2}] \tag{1}$$

Diameter of Solid Steel Cylinder

$$D_s = 0.75 \ [\mathrm{in}] \tag{2}$$

Outer and Inner Diameter of Steel Tube

$$D_o = 1.5 \ [\mathrm{in}] \tag{3}$$

$$D_i = 1.25 \ [\mathrm{in}] \tag{4}$$

Length of Steel Cyclinder/Tube

$$L = 12 \ [\mathrm{in}] \tag{5}$$

Load Applied - 1000lb distributed over 4 Steel Cylinders/Tubes

$$P = \frac{F}{2 \ [\mathrm{lb}]} \tag{6}$$

$$F = 1000 \ [\mathrm{lb}] \tag{7}$$

Stress of Steel Tube

$$\sigma_t = P/A_t \tag{8}$$

Cross Sectional Area of Steel Tube

$$A_t = (\pi/4) \cdot (D_o^2 - D_i^2) \tag{9}$$

Stress of Solid Steel Cylinder

$$\sigma_s = P/A_s \tag{10}$$

Cross Sectional Area of Solid Steel Cylinder

$$A_s = (\pi/4) \cdot D_s^2 \tag{11}$$

Strain of Steel Tube

$$\epsilon_t = \sigma_t/E \tag{12}$$

Strain of Solid Steel Cylinder

$$\epsilon_s = \sigma_s / E \tag{13}$$

Deflection of Steel Tube

$$\delta_t = \epsilon_t \cdot L \tag{14}$$

Deflection of Solid Steel Cylinder

$$\delta_s = \epsilon_s \cdot L \tag{15}$$

Crossbar Analysis

$$L_{cb} = 14 \text{ [in]} \tag{16}$$

$$h = 1 \text{ [in]} \tag{17}$$

$$b = 8 \text{ [in]} \tag{18}$$

$$I = (1/12) \cdot b \cdot h^3 \tag{19}$$

$$y_{max} = \frac{F \cdot l_{cb}^3}{48 \cdot E \cdot I} \tag{20}$$

Solution

$A_s = 0.4418$	$A_t = 0.54$
$b = 8\,\text{[in]}$	$\delta_s = 0.0004527$
$\delta_t = 0.0003704$	$D_i = 1.25\,\text{[in]}$
$D_o = 1.5\,\text{[in]}$	$D_s = 0.75\,\text{[in]}$
$E = 3.000 \times 10^7$	$\epsilon_s = 0.00003773$
$\epsilon_t = 0.00003087$	$F = 1000\,\text{[lb]}$
$h = 1\,\text{[in]}$	$I = 0.6667$
$L = 12\,\text{[in]}$	$L_{cb} = 14\,\text{[in]}$
$P = 500\,\text{[lb]}$	$\sigma_s = 1132$
$\sigma_t = 926$	$y_{max} = 0.002858$

Excel Column Rod Calculations

Column Rods

Parameters	
P [lb]	2000
L [in]	17
E [psi]	30000000

	σ=P/A	σ = E × =σ/E	=Δ/L Δ = × L	k = dP/ dΔ k = P/ Δ	
Rod Diameter	Stress [psi]	Strain	Elongation [in]	Stiffness [lb/in]	Weight [lb/shft]
0.5	5093	0.00017	0.00289	346499	1.02
0.625	3259	0.00011	0.00185	541405	1.53
0.75	2264	0.00008	0.00128	779623	2.21
1	1273	0.00004	0.00072	1385997	3.74
1.25	815	0.00003	0.00046	2165620	5.95

Excel Spreadsheet of Calculations of Rectangular Tubing

Cross Braces

Parameters	
P [lb]	2000
L [in]	12
E [psi]	30000000

*Simply Supported Beam - Center Load Table A-9 Shigleys

**ASTM A500 Density [lb/in^3]= 0.284

$I = (^1/_{12}) \times W \times H^3$

$I_{tube} = I_{outer} - I_{inner}$ | $\sigma = (P \times L \times H/2)/(4 \times I_{tube})$ | $\delta = (P \times L^3)/(48 \times E \times I)$ | $W = A \times L \times \rho$

Rectangular Cross Section							
W [in]	H [in]	t [in]	Moment of Inertia [in^4]	Center Stress [psi]	Center Deflection [in]	Area [in2]	Weight [lb]
2	3	0.188	2.0549	4380	0.00117	1.74	5.93
3	3	0.188	2.7993	3215	0.00086	2.11	7.21
3	4	0.188	5.5925	2146	0.00043	2.49	8.49
4	4	0.188	6.9595	1724	0.00034	2.87	9.77
2	3	0.25	2.5469	3534	0.00094	2.25	7.67
3	3	0.25	3.4948	2575	0.00069	2.75	9.37
3	4	0.25	7.0677	1698	0.00034	3.25	11.08
4	4	0.25	8.8281	1359	0.00027	3.75	12.78

Horizontal Edge between Verticals					
L [in]	H [in]	W [in]	Moment of Inertia [in^4]	Center Stress [psi]	Center Deflection [in]
2	0.188	12	0.0066	84880	0.3612
3	0.188	12	0.0066	84880	0.3612
3	0.188	12	0.0066	84880	0.3612
4	0.188	12	0.0066	84880	0.3612
2	0.25	12	0.0156	48000	0.1536
3	0.25	12	0.0156	48000	0.1536
3	0.25	12	0.0156	48000	0.1536
4	0.25	12	0.0156	48000	0.1536

Excel Spreadsheet of Calculations of Flat Plate as Horizontal Member

Flat Plate							
W [in]	H [in]		Moment of Inertia [in^4]	Center Stress [psi]	Center Deflection [in]	Area [in2]	Weight [lb]
3	0.5		0.0313	48000	0.07680	1.50	5.11
3	1		0.2500	12000	0.00960	3.00	10.22
3	1.5		0.8438	5333	0.00284	4.50	15.34
4	0.5		0.0417	36000	0.05760	2.00	6.82
4	1		0.3333	9000	0.00720	4.00	13.63
4	1.5		1.1250	4000	0.00213	6.00	20.45
5	0.5		0.0521	28800	0.04608	2.50	8.52
5	1		0.4167	7200	0.00576	5.00	17.04
5	1.5		1.4063	3200	0.00171	7.50	25.56
6	0.5		0.0625	24000	0.03840	3.00	10.22
6	1		0.5000	6000	0.00480	6.00	20.45
6	1.5		1.6875	2667	0.00142	9.00	30.67

Excel Spreadsheet of Buckling Calculations for Crush Sleeves

Crush Sleeves

Parameters	
P [lb]	2000
E [psi]	30000000
Sy	30000
C	4

	Pipe Stock from Tubularsteel.com							Shigleys		J.B. Johnson	
Wall [in]	L [in]	OD [in]	ID [in]	MOI	Radius of Gyration	Area [in^2]	Slenderness	Pcr [lb]	Weight [lb]	Pcr [lb]	Weight [lb]
0.095	2.624	0.405	0.2150	0.0012	0.115	0.093	23	209124	0.069	2766.387	0.0689473
	2.5	0.405	0.2150	0.0012	0.115	0.093	22	230384	0.066	2767.237	0.0656891
	3.624	0.405	0.2150	0.0012	0.115	0.093	32	109637	0.095	2758.030	0.095223
	3.5	0.405	0.2150	0.0012	0.115	0.093	31	117543	0.092	2759.212	0.0919648
0.119	2.624	0.54	0.3020	0.0038	0.155	0.157	17	647722	0.117	4713.114	0.11729
	2.5	0.54	0.3020	0.0038	0.155	0.157	16	713570	0.112	4713.909	0.1117474
	3.624	0.54	0.3020	0.0038	0.155	0.157	23	339578	0.162	4705.306	0.161989
	3.5	0.54	0.3020	0.0038	0.155	0.157	23	364066	0.156	4706.410	0.1564463
0.091	2.624	0.675	0.4930	0.0073	0.209	0.167	13	1254042	0.124	5003.703	0.1244189
	2.5	0.675	0.4930	0.0073	0.209	0.167	12	1381528	0.119	5004.164	0.1185393
	3.624	0.675	0.4930	0.0073	0.209	0.167	17	657451	0.172	4999.164	0.1718346
	3.5	0.675	0.4930	0.0073	0.209	0.167	17	704861	0.166	4999.806	0.1659551
0.126	2.624	0.675	0.4230	0.0086	0.199	0.217	13	1482500	0.162	6512.328	0.1619478
	2.5	0.675	0.4230	0.0086	0.199	0.217	13	1633211	0.154	6512.990	0.1542947
	3.624	0.675	0.4230	0.0086	0.199	0.217	18	777224	0.224	6505.824	0.2236656
	3.5	0.675	0.4230	0.0086	0.199	0.217	18	833271	0.216	6506.744	0.2160126

Thread Calculations

Wrench Torque (T1=10^b+m*logd)
Torque to get Preload (T2=K*fi*d)
Preload for Reusable Connections (Fi=.75*At*Sp)
Load to Break Threads (P=St*At)

Constants		
K	0.2	
b	2.759	
m	2.965	SAE Grade 5
b	2.983	
m	3.095	SAE Grade 8
Sp(5)	85000	
Sp(8)	120000	
St(5)	120000	
St(8)	150000	

Bolt Dimensions			SAE GRADE 5		
Bolt Size	TPI	At [in^2]	Preload for Reusable Conn. [lbs]	Wrench Torque [ft-lbs]	Load to Break Threads [lbs]
10	24	0.0175	1116	4	2100
10	32	0.0200	1275	4	2400
12	24	0.0242	1543	6	2904
12	28	0.0258	1645	6	3096
0.25	20	0.0318	2027	9	3816
0.25	28	0.0364	2321	9	4368
0.3125	18	0.0524	3341	18	6288
0.3125	24	0.058	3698	18	6960
0.375	16	0.0775	4941	31	9300
0.375	24	0.0878	5597	31	10536
0.4375	14	0.1063	6777	49	12756
0.4375	20	0.1187	7567	49	14244

Excel Spreadsheet Comparing Chosen Crush Sleeve Strength to Actual Load

Check to confirm that load cell crush sleeves can support bolt preload and applied load

Bolt Dimensions			SAE GRADE 5				
Bolt Size	TPI	At [in^2]	Break Threa	eload in 4 b	Total load [lbs]	r smallest crush sleev	Margin [lbs]
10	24	0.0175	2100	8400	10400	11065.5	665.5
10	32	0.0200	2400	9600	11600	11065.5	-534.5
12	24	0.0242	2904	11616	13616	11065.5	-2550.5
12	28	0.0258	3096	12384	14384	11065.5	-3318.5
0.25	20	0.0318	3816	15264	17264	11065.5	-6198.5
0.25	28	0.0364	4368	17472	19472	11065.5	-8406.5
0.3125	18	0.0524	6288	25152	27152	11065.5	-16086.5
0.3125	24	0.058	6960	27840	29840	11065.5	-18774.5
0.375	16	0.0775	9300	37200	39200	11065.5	-28134.5
0.375	24	0.0878	10536	42144	44144	11065.5	-33078.5
0.4375	14	0.1063	12756	51024	53024	11065.5	-41958.5
0.4375	20	0.1187	14244	56976	58976	11065.5	-47910.5

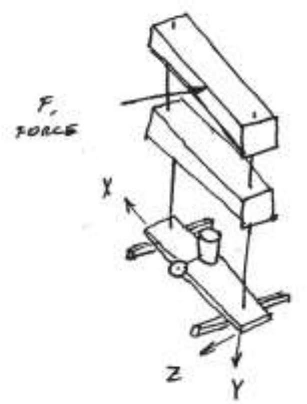
Hand Calculations Verifying Stability of Final Design

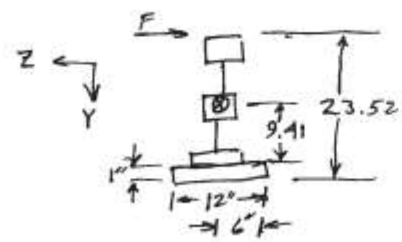
GIVEN: TEST APPARATUS WITH CENTER OF MASS AT COORDINATES $X = 5.97$, $Y = -9.41$, $Z = 2.48$ BY COORDINATE SYSTEM SHOWN ON FIGURE BELOW. (CENTER OF MASS COORDINATES COMPUTED BY "MASS PROPERTIES" FUNCTION IS SOLIDWORKS.) UNKNOWN FORCE TO BE APPLIED HORIZONTALLY AT TOP OF TEST APPARATUS, AS SHOWN BELOW.

REQUIRED: DETERMINE ~~IF APPARA~~ VALUE OF FORCE AT WHICH APPARATUS WILL START TO TIP.

ASSUMPTIONS: APPARATUS IS RIGID
WEIGHT IS 80 lbs
MODEL AS SIMPLY SUPPORTED

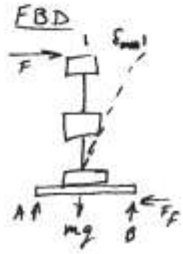
SCHEMATIC:





SOLUTION:

FBD



$A \to 0$ AT START OF TIPPING

$$\Sigma M_B = -F(23.52\,in) - A(12\,in) + mg(6.00\,in) = 0$$

$$F = \frac{mg(6.0\,in)}{23.52\,in} \Rightarrow F = \frac{80\,lb\,(6.0)}{23.52} \Rightarrow \boxed{F = 20.41\ lbs}$$

HORIZONTAL FORCE REQUIRED TO START ROCKING APPARATUS

INCREASING BASE WIDTH INCREASES ~~AN~~ FORCE

BASE WIDTH	FORCE TO TIP
14"	23.81 lbs
16"	27.21 lbs
18"	30.61 lbs

ADDITIONAL CONSIDERATION

REQ'D: DETERMINE DEFLECTION OF APPARATUS UNDER $F = 20.41$ lbs

SOL'N: $\delta_{MAX} = \frac{FL^3}{3EI}$, $I = \frac{\pi}{4} r^4$, FROM matweb.com FOR AISI 440A STAINLESS STEEL $E = 29 \times 10^6$ psi

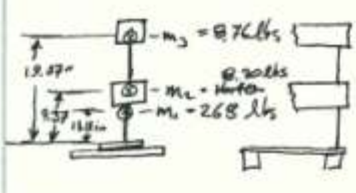
$r = \frac{0.75"}{2}$, $r = 0.375"$, SINCE TWO RODS $F = 10.205$ lbs $\therefore \delta_{max} = \frac{10.205\,lbs\,(23.52\,in)^3}{3(29 \times 10^6 \frac{lbs}{in^2})(\frac{\pi}{4})(0.375\,in)^4}$

$\delta_{max} = 0.09826 \Rightarrow \boxed{\delta_{max} = 0.0983\,in}$

GIVEN: HALF-MODEL OF TEST APPARATUS.
COMPONENT WEIGHTS AND HEIGHTS OF CENTER OF GRAVITY ARE DETERMINED FROM SOLID MODELING PROGRAM

REQ'D: FIND NATURAL FREQUENCY OF TEST APPARATUS

SCHEMATIC:



ASSUMPTIONS: VERTICAL SECTION RIGIDLY ATTACHED TO BASE

SOLUTION:

DETERMINE MASS EQUIVALENT & SPRING ~~OF~~ EQUIVALENT OF HALF-MODEL

CROSSED OUT SINCE $M_O = k_e\theta$

$$m_1 \overset{\theta}{\sin\theta}\, l_1 + m_2 \overset{\theta}{\sin\theta}\, l_2 + m_3 \overset{\theta}{\sin\theta}\, l_3 - \cancel{M_o} = \cancel{-k_e\theta} + m_e L \overset{\theta}{\sin\theta}$$

$$2 \quad m_e = \frac{m_1 l_1 + m_2 l_2 + m_3 l_3}{L}$$

$$= \frac{(268\,lbs)(11.11in) + 8.20\,lbs(9.37in) + 8.76\,lbs(19.37in)}{25.52\ in}$$

$\cancel{m_e = 10.83\ lbs}$ CALCULATOR ERROR

$$\underline{m_e = 10.99\ lbs}$$

$$k_e = \frac{M_o}{\theta} = \frac{FL}{\theta}$$

FOR CANTILEVER BEAM UNDER TRANSVERSE ~~LOAD~~ END LOAD

$$\theta = \frac{FL^2}{2EI}$$

FOR CIRCULAR CROSS-SECTION: $I = \frac{\pi}{4} r^4$

$$k_e = \frac{FL}{FL^2/2EI} = \frac{2EI}{L} \Rightarrow k_e = \frac{\pi E r^4}{2L}$$

FROM matweb.com, FOR AISI 440A STAINLESS STEEL, $E = 29 \times 10^6$ psi
$r = 0.75''/2 = 0.375$ in

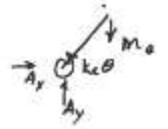
$$k_e = \frac{\pi (29\times10^6 \frac{lbs}{in^2})(0.375 in)^4}{2(25.52 in)}$$

$$\underline{k_e = 35.3\times10^3 \frac{lb\cdot in}{rad} = \cancel{2.94}\ \frac{ft\cdot lbs}{rad}\ 2.94\times10^3 \frac{ft\cdot lbs}{rad}}$$

[CONTINUATION OF FINDING NATURAL FREQUENCY]

EQUIVALENT MODEL

FBD



MAD

FOR POINT MASS
$I = m L^2$

$\ddot{\theta} = \omega^2 \theta$

$$\Sigma M_A = m_e L \overset{\theta}{\cancel{\sin\theta}} - k_e \theta = I \ddot{\theta}$$

$$m_e L \cancel{\theta} - k_e \cancel{\theta} = \frac{m_e}{g} L^2 \omega^2 \cancel{\theta}$$

$$\omega = \left[\frac{m_e L - k_e}{\frac{m_e L^2}{g}}\right]^{1/2}$$

$$= \left[\frac{(10.99\ \text{lbs})(25.52\ \text{in}) - (35.3\times10^3\ \frac{\text{lb}\cdot\text{in}}{\text{rad}})}{\frac{(10.99\ \text{lbs})(25.52\ \text{in})^2}{32.2\ \frac{\text{ft}}{\text{s}^2}}}\right]^{1/2}$$

$$= \left[\frac{35.02\times10^3\ \text{in lbs}}{222.28\ \frac{\text{in}^2\ \text{lbs}\cdot\text{s}^2}{\text{ft}}\cdot\frac{1\ \text{ft}}{12\ \text{in}}}\right]^{1/2}$$

$$= \left(\frac{35.02\times10^3\ \text{in lbs}}{18.52\ \text{in lbs}\cdot\text{s}^2}\right)^{1/2}$$

$$\omega = \left(1890\ \frac{\text{rad}^2}{\text{s}^2}\right)^{1/2}$$

$$\underline{\omega = 43.48\ \frac{\text{rad}}{\text{s}}}$$

$$f = \frac{43.48\ \frac{\text{rad}}{\text{s}}}{2\pi\ \frac{\text{rad}}{\text{cycle}}}$$

$$\underline{f = 6.92\ \frac{\text{cycles}}{\text{sec}}}$$

Appendix B

BOM

Part	Sizing/Description	Part No.	Qty	Cost	Purchase From
Bottom Plate	Mild Steel A36 Hot Rolled Rectangle 1" x 5", Cut to 12"	N/A	1	29.30	Online Metals
H Beam	1 x 1 x 11 GA (.120 wall) A513 Steel Structural Square Tube, 2' Length	T11111	1	4.32	Metals Depot
Sliding Beam	Mild Steel A36 Hot Rolled Rectangle Tube 3" x 5" x 0.12", Cut to 12"	N/A	1	21.06	Online Metals
Top Plate Rectangular Tubing	3 x 3 x 3/16 wall A500 Steel Structural Square Tube, 1' Length	T133316	1	9.50	Metals Depot
Column Rod	NIL12SS, 25.2" Overall Length. 1/2-13 external thread both ends. 5" of thread one end, 2" opposite end	N/A	2	232.86	PBC Linear
Flange Linear Bearing	3/4 inch rod, Linear ball bearing	IPK12G	4	51.08	PBC Linear
Futek 250 lb Load Cell	250 lb pancake style load cell	FSH00130	1	695.00	Futek
Futek 2000 lb Load Cell	2000 lb pancake style load cell	FSH00126	1	695.00	Futek
Nook Jack	1 ton ball screw jack	1-BSJ-UR	1	800.00	Nook Industries
Hand Wheel	Spoked Die Cast Zinc Dished Hand Wheel Revolving Handle, 4-1/2" Wheel Die, 1/2" Hole Dia	6033K71	1	15.96	McMaster-Carr
Quick Pin 2"	18-8 SS Ring-Grip Quick Release Pin 1/4" Diameter, 2.0" Usable Length	98404A960	2	3.90	McMaster-Carr
Spacer Block for Load Cell	Aluminum 6061 - T6 Bare Extruded Round 5.5", Cut to 2"	N/A	1	19.39	Online Metals
Load Cell Button	Aluminum 6061 - T6 Bare Extruded Round 3", Cut to 1"	N/A	2	5.88	Online Metals
Top Sample Plate	52100 Steel	N/A	4		Sourced from Parker
Bottom Sample Plate	52100 Steel	N/A	4		Sourced from Parker
Dial Indicator Plates	Multipurpose Aluminum (alloy 6061), 3/8" Thick, 12" x 12"	9246K23	2	57.02	McMaster-Carr
Sample Plate Guide Pins	18-8 Stainless Steel Shoulder Screw, 1/4" Shoulder Diameter, 2" Shoulder Length, 10-24 Thread	90298A548	14	38.64	McMaster-Carr
Jack Crush Sleeve	4142 Alloy Steel Tubing 2-1/2" OD, 1-1/2" ID, 1/2" Wall Thickness, 1' Length	6920T111	1	57.51	McMaster-Carr
Load Cell Crush Sleeves	5/8 OD x .156 wall x .313 ID 1020 DOM Structural Round Steel Tube, 2' Length	T258156	1	18.74	Metals Depot
Top Column Crush Sleeves	Low-Carbon Steel Tubing, 1.25" OD, 0.75" ID, .250" Wall Thickness, 1' Length	7767T691	1	19.45	McMaster-Carr
Bottom Sample Plate Track	Aluminum 6061 - T6 Bare Extruded Rectangle 0.75" x 6", Cut to 2.5"	N/A	2	9.36	Online Metals
Middle Sample Plate Track	Aluminum 6061 - T6 Bare Extruded Rectangle 1.25" x 3.5", Cut to 2.8"	N/A	2	10.18	Online Metals
Top Sample Plate Track	Aluminum 6061 - T6 Bare Extruded Rectangle 1" x 6", Cut to 3.5"	N/A	2	17.46	Online Metals
Magnetic Arm - 1/8" Ball (Don't need)	O1 Tool Steel Hardened Rod 7/64" Diameter, 1' Length	4347T34	1	14.01	McMaster-Carr
Magnetic Arm - 1/2" Ball	W1 Tool Steel Rod .2031" Diameter, Trade Size 13/64", 3' Length	8890K28	1	2.63	McMaster-Carr
Magnetic Arm - 1/4" Ball	W1 Tool Steel Rod .2031" Diameter, Trade Size 13/64", 3' Length	8890K28	1	2.63	McMaster-Carr
Magnetic Arm - 1/8" Ball	W1 Tool Steel Rod .2031" Diameter, Trade Size 13/64", 3' Length	8890K28	1	2.63	McMaster-Carr
Flange Sleeve for Magnetic Arm	Multipurpose Aluminum (Alloy 6061) Tube 2-1/2" OD, 1-1/2" ID, .500" Wall Thk, 1'Length	9056K191	1	130.08	McMaster-Carr
Magnet	Machinable Neodymium Disc Magnet, .078" Diameter, .394" Thick, .6 Pull lbs	5902K42	3	5.31	McMaster-Carr
Nylock Nut #10-24	-	-	16	16.00	Ace Hardware
Nylock Nut 1/2-13	-	-	4	1.00	Home Depot
Nylock Nut 3/8-16	-	-	2	2.00	Ace Hardware

Part	Sizing/Description	Part No.	Qty	Cost	Purchase From
Nut 1/4-20	-	-	8	0.48	Home Depot
Nylock Nut #4-40	-	-	4	4.00	Ace Hardware
Hex Nut 4-40	-	-	2	2.00	Ace Hardware
Flat Head Socket Cap Screw #4-40 x 5/8	-	-	4	4.00	Ace Hardware
Hex Bolt 1/4-20 x 3.5	-	-	4	0.92	Home Depot
Hex Bolt 1/4-20 x 3	-	-	8	1.68	Home Depot
Hex Bolt #10-24 x .75	-	-	16	2.20	Home Depot
Hex Bolt 3/8-16 x 2	-	-	2	2.00	Ace Hardware
Hex Bolt 1/4-20 x 1.25	-	-	4	4.00	Ace Hardware
Keysert	18-8 SS Key-Locking Thin Wall Threaded Insert 10-24 Int Thread, 5/16"-18 Ext Thread, 5/16' L	93340A405	8	23.44	McMaster-Carr
Captive Nut 4-40 Thread	18-8 Stainless Steel Broach Style Captive Nut, 4-40 Thread Size, .060" Minimum Panel Thickness (Pack of 25)	94648A320	1	7.30	McMaster-Carr
Eyebolt	Galvanized Steel Eyebolt with Shoulder & Nut, 1/4"-20 Thread, 500# Work Load Limit, 4" Shank, 2" Thread Length	3018T34	1	5.11	McMaster-Carr
Contact Point for Dial Indicator	Electronic and Dial Indicator Accessory, 3/16" x 1/4" Regular Contact Point for Dial Indicator	20625A651	3	4.41	McMaster-Carr
Contact Point for Dial Indicator	Electronic and Dial Indicator Accessory, 3/16" x 1/2" Regular Contact Point for Dial Indicator	20625A652	3	5.25	McMaster-Carr
Disc Magnet (Not Used)	Ultra-High-Pull Neodymium Disc Magnet, 1/8" Diameter, 1/8" Thick, .7 Pull lbs	5862K51	3	5.10	McMaster-Carr
Disc Magnet (Not Used)	Ultra-High-Pull Neodymium Disc Magnet, 3/8" Diameter, 1/8" Thick, 3.6 Pull lbs	5862K95	2	7.20	McMaster-Carr
Washers	1/2"	N/A	8	0.50	Home Depot
Ball Bearings	1/2" Ball Bearings	N/A	4	2.04	Central Coast Bearings
Socket Set Screw	Nylon Tip Alloy Steel Socket Set Screw, 8-32 Thread, 5/8" Length (Pack of 10)	94115A196	1	6.02	McMaster-Carr
Carbide Drill Bit	Super-Duty Solid Carbide Jobbers' Drill Bit, Size L, 3-1/2" Overall Lg, 1.7" Drill DP, 118 Degree Point	29195A43	1	24.23	McMaster-Carr
Springs (Replaced)	Music Wire Precision Compression Spring, Zinc-Plated, 7/8" Length, .30" OD, .022" Wire (Pack of 5)	9434K53	1	4.95	McMaster-Carr
Compression Spring	Brass Compression Spring, 25/32" L, .042" Wire (Packs of 12)	9657K86	1	7.85	McMaster-Carr
Conical Springs	Type 302 Stainless Steel Conical Compression Spring, 1" L, .60" L OD, .343" Small OD, .026" Wire Diameter (Packs of 3)	1692K31	2	27.42	McMaster-Carr
Springs	Type 302 Stainless Steel Conical Compression Spring, 1/2" L, .60" L OD, .375" Small OD, .032" Wire Diameter (Packs of 3)	1692K19	2	29.40	McMaster-Carr
Futek software/Hardware	USB connectivity kit for 250 lb and 2000 lb load cells, manufacturer calibrated	N/A		1000.00	Futek
Chicago Dial Indicator Software/Hardware	Recommended LP3600	N/A		1677.00	Chicago Dial Indicators

Total Sales Tax	35.31
Total Shipping Charges	77.71
Traveling Expenses	1056.85
Total Cost	7016.27

Appendix C

Detailed Drawings

Detailed Drawings Have Been Deleted From This Library Copy Per Confidentiality Agreement With Parker Aerospace

Appendix D

Assembly Instructions

Assembly Instructions Have Been Deleted From This Library Copy Per Confidentiality Agreement With Parker Aerospace

Appendix E

Vendors

Acuity Laser
Phone: 1-503-227-5178

Address: 2765 NW Nicolai Street
Portland, OR 97210

Boeckeler
Phone: 1-800-552-2262
Address: 4650 S. Butterfield Dr.
Tucson, AZ 85714

Chicago Dial Indicators
Phone: 1-800-344-GAGE
Address: 1372 Redeker Rd
Des Plaines, IL 60016

Futek
Phone: 1-800-23-FUTEK
Address: 10 Thomas
Irvine, CA 92618

Pacific-Bearing
Phone: 1-800-962-8979
Address: 6402 Rockton Rd
Roscoe , IL 61073

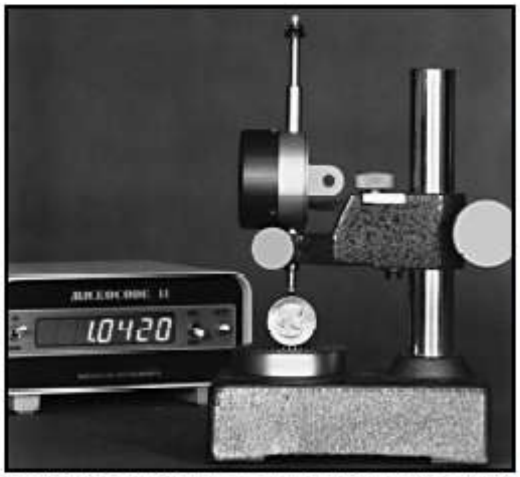
Nook Industries
Phone: 1-216-271-7900
Address: 4950 East 49th Street
Cleveland, OH 44125-1016

Appendix F

Vendor Literature

Boeckeler

Digital Dial Indicators



Boeckeler digital dial indicators are available with a variety of mounting backs. The standard back is the vertical mount. Also pictured is a one axis Microcode II digital readout.

APPLICATIONS

A Boeckeler® digital dial indicator can be used as a direct replacement for most mechanical dial indicators in many applications, including the following:

- **Production inspection.**
- **Z-axis measurement from microscope stands.**
- **Out-of-roundness measurements.**
- **Many other applications.**

FEATURES

Boeckeler's digital dial indicators offer many advantages over conventional dial indicators, such as:

- Encoded dial indicators have greater resolution than conventional indicators.
- Accuracy to +/- 0.000 2 in (+/- 0.005 mm).
- Resolution to 0.000 01 in (0.001 mm).
- Measurement range to 4 in (100 mm).
- Rapid, error-free measurement.
- Proven, trouble-free operation.
- Conformance to American Gage Design (AGD) tolerances.
- Electronics which are well insulated from shock, dirt, dust, chips, etc.
- Compatible with digital readouts which provide large, high contrast LED readings.
- A variety of mounting options: horizontal lug back, flat back, and adjustable back.
- Accessories available include shock tip, roller tip, lift lever, extensions, and extension tips to serve a variety of applications.

DIGITAL READOUTS

The Microcode II™ digital readout provides many useful features when used with digital dial indicators. The relative or absolute zero switch allows a temporary zero without losing an absolute zero reference. Inch or metric readings are selectable at any time. The direction of the count (plus/minus) sign can be reversed, and an optional RS-232 data port is available to provide data output to a printer or computer. Other options include: Min/Max/Diff for measuring "out-of-roundness," Offset for adding dimensions of gage rods and blocks, Averaging RS-232 output, and a Dual Voltage power supply. The Microcode II digital readout is a state-of-the-art microprocessor and can be tailored to your application.

Micrometer and Positioner Heads

APPLICATIONS

- **Measuring standard** in calibration laboratory.
- **Small parts inspection.**
- **Industrial quality control microscope measurement** such as: magnetic tape head production, integrated circuit mask measurement, fuel cell dimensioning and semiconductor inspection.
- **Biomedical microscope measurement** such as cell measurement and culture growth measurement.
- **Optical comparator measurement.**
- **Supermike retrofit.**

Displayed are two 1300 series digital micrometer heads and a digital readout.

FEATURES

- **Smooth, precise operation.**
- **Anti-backlash design.**
- **Non-rotating spindle.**
- **Constant measurement performance.**
- **NIST traceable**
- **Remote zero reset on positioner heads.**

PRODUCT DESCRIPTION

Boeckeler® micrometer and positioner heads achieve consistent measurement performance. Digital micrometer and positioner heads can be connected to a Boeckeler digital readout which provides instant inch/metric conversion, zero reset, and absolute/relative measurement as standard features.

Boeckeler micrometer heads are carefully balanced over the full range of measurement in order to maintain the famous Boeckeler performance. The thimble and barrel are made of high tensile strength aluminum alloy. They have an anodized finish with engraved vernier graduations in contrasting colors. No vernier graduations are provided on positioner heads. Each hardened steel lead screw and carbide tip is individually calibrated to specified tolerances.

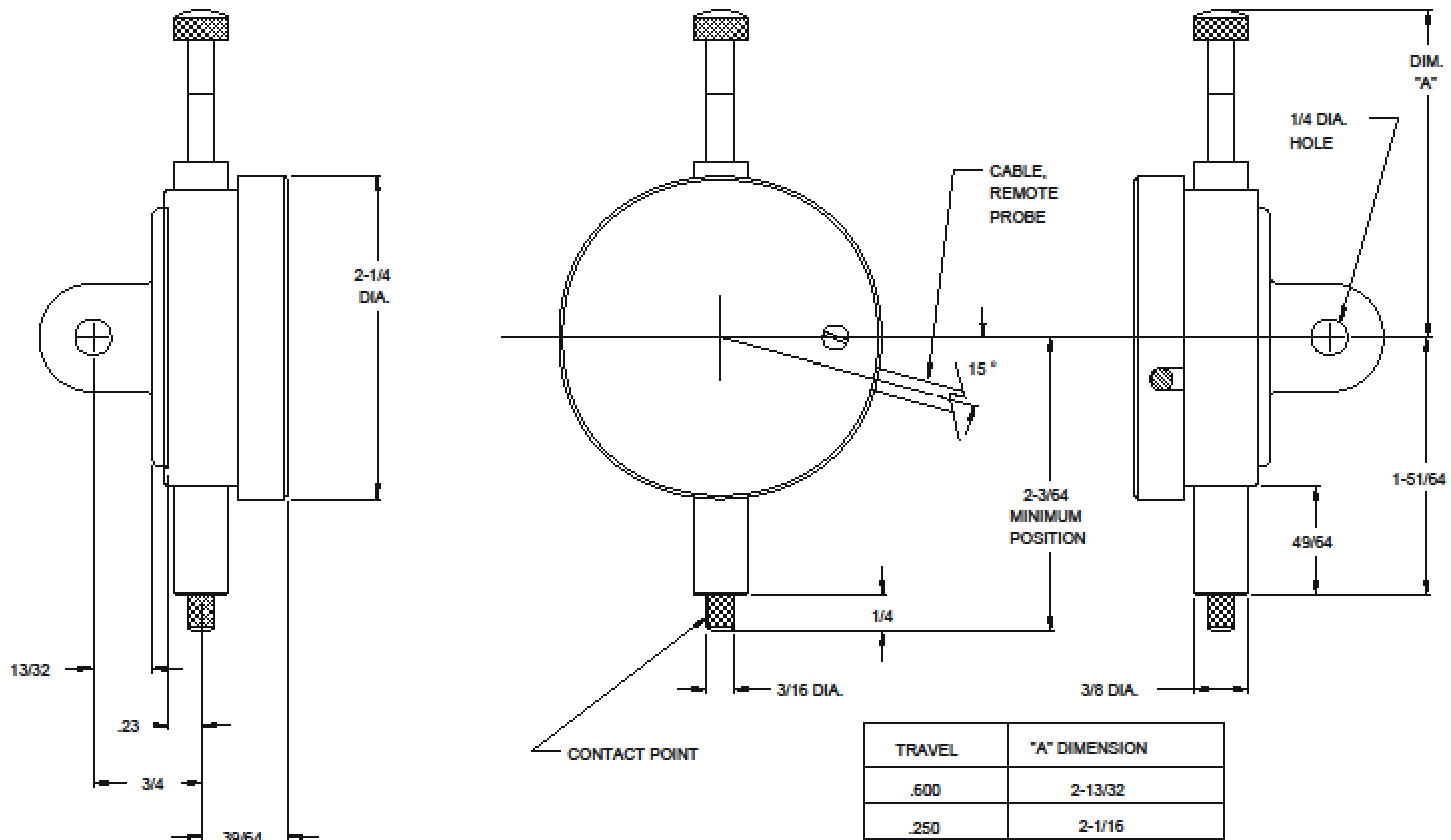
Every Boeckeler micrometer head is equipped with a non-rotating spindle which allows for consistent measurements while the thimble is being rotated. A flat carbide tip is standard. Spindles can be drilled and tapped on custom orders to accommodate the attachment of specially designed probes or anvils. Further stabilization of the spindle can be implemented by special ordering a captive spindle.

BOECKELER®

Chicago Dial Indicators

LOGIC SERIES GROUP 2 CASE PROBE
TRAVEL UP TO .600

SCALE = FULL



TRAVEL	"A" DIMENSION
.600	2-13/32
.250	2-1/16
.125	1-15/16
.062	1-7/8

Futek

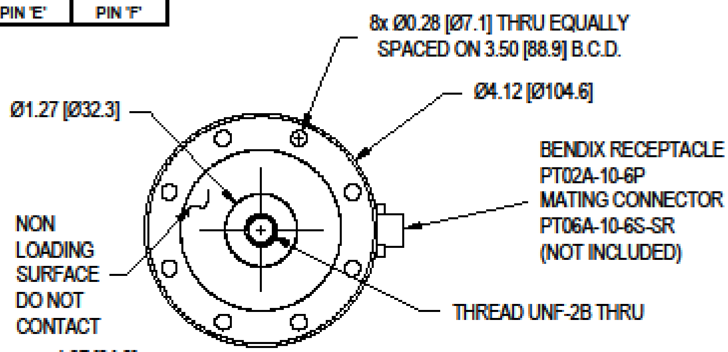
FUTEK MODEL LCF450 (L2900)

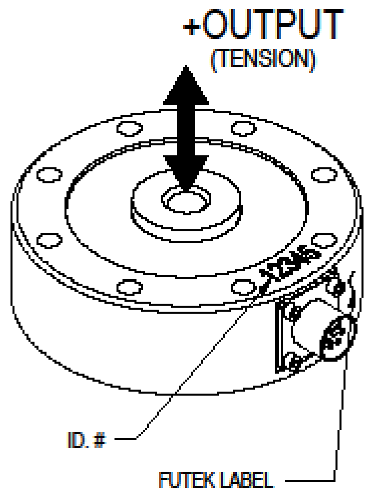
LOW PROFILE UNIVERSAL PANCAKE LOAD CELL

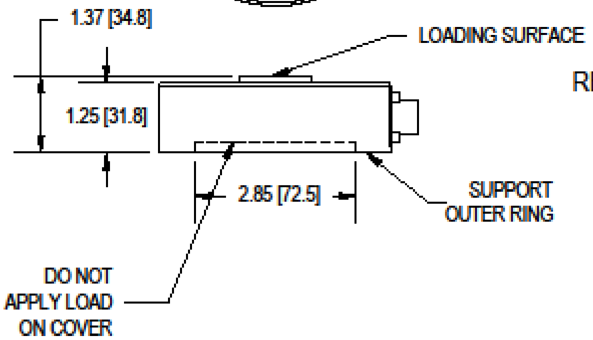
Drawing Number: FI1011

INCH [mm] | R.O.= Rated Output

WIRING CODE (CC1)			
+Excitation	-Excitation	+Signal	-Signal
RED PIN 'A'	BLACK PIN 'B'	GREEN PIN 'C'	WHITE PIN 'D'
+Sense	-Sense		
ORANGE PIN 'E'	BLUE PIN 'F'		







REFER TO LCF455 FOR TENSION BASE VERSION

Stock #	CAPACITY lb	CAPACITY N	THREAD	MATERIAL
FSH00053	300	1334	5/8-18	17-4PH ALLOY STEEL
FSH02050			M16 X 2	
FSH00128	500	2224	5/8-18	ANODIZED ALUMINUM
FSH00132			M16 X 2	
FSH00129	1000	4448	5/8-18	
FSH00133			M16 X 2	
FSH00130	2000	8896	5/8-18	
FSH00134			M16 X 2	
FSH00137	5000	22240	5/8-18	17-4PH ALLOY STEEL
FSH00141			M16 X 2	
FSH00138	10000	44480	5/8-18	
FSH00142			M16 X 2	

SPECIFICATIONS:

RATED OUTPUT	2 mV/V nom.
SAFE OVERLOAD	150% of R.O.
ZERO BALANCE	±1% of R.O.
EXCITATION (VDC OR VAC)	20 MAX
BRIDGE RESISTANCE	700 Ω
NONLINEARITY	±0.1% of R.O. } HIGHER SPECIFICATIONS OF 0.05% AVAILABLE CONTACT FACTORY
HYSTERESIS	±0.2% of R.O.
NONREPEATABILITY	±0.02% of R.O.
CREEP	±0.02% of LOAD
TEMP. SHIFT ZERO	±0.001% of R.O./°F [0.0018% of R.O./°C]
TEMP. SHIFT SPAN	±0.002% of LOAD/°F [0.0036% of LOAD/°C]
COMPENSATED TEMP.	60 to 160°F [16 to 72°C]
OPERATING TEMP.	-60 to 200°F [-50 to 93°C]
WEIGHT	ANODIZED ALUMINUM: 1.3lb [0.6kg], 17-4PH ALLOY STEEL: 3.5lb [1.6kg]
DEFLECTION	0.002 to 0.005 [0.05 to 0.13] mm.

CONNECTOR: 6 Pin BENDIX Receptacle (PT02A-10-6P)

ACCESSORIES AND RELATED INSTRUMENTS AVAILABLE

CALIBRATION (STD)	5 pt. COMPRESSION; 100K Ω SHUNT CAL. VALUE
CALIBRATION TEST EXCITATION	10 VDC

OTHER AVAILABLE RECEPTACLES & MATING CONNECTORS

BENDIX RECEPTACLE (PC04E-10-6P)
BENDIX MATING CONNECTOR (PC06E-10-6S-SR)

BENDIX RECEPTACLE (PT02H-10-6P)
BENDIX MATING CONNECTOR (PT06A-10-6S-SR)

FUTEK ADVANCED SENSOR TECHNOLOGY, INC.

10 THOMAS
IRVINE, CA 92618 USA
1-800-23-FUTEK (38835)

INTERNET:
http://www.futek.com

Simplicity® Ball Bearings
Technical Information

PRODUCT OVERVIEW

- The Simplicity® ball bearing consists of an outer cylinder, ball retainer, balls and two end rings. The ball retainer which holds the balls in the recirculating tracks is held inside the outer cylinder by end rings.
- Those parts are assembled to optimize their required functions.
- The outer shell is heat treated to ensure long life.
- The ball retainer is molded from a durable polymer to ensure smooth motion.

FEATURES

High Precision and Rigidity -
The Simplicity® ball bearing is produced from a solid steel outer cylinder and incorporates an industrial strength polymer retainer.

Ease of Assembly -
The standard type of Simplicity® ball bearing can be loaded from any direction. Precision control is possible using only the shaft supporter, and the mounting surface can be machined easily.

Ease of Replacement -
Simplicity® ball bearing of each type are completely interchangeable because of their standardized dimensions and strict precision control. Replacement because of wear or damage is therefore easy and accurate.

Variety of Types -
PBC offers a full line of Simplicity® ball bearings: the standard, integral single-retainer closed type, the clearance adjustable type and the open types. The user can choose from among these according to the application requirements to be met.

ORDERING INFORMATION

IP | F | C | 10 | G | W | OP

Type
IP: Inch dimension series
EP: ISO metric dimension series
JP: JIS metric dimension series
KHP: ISO metric thin wall dimension series

Flange Type
F - Round flange
K - Square flange

Flange Type Center
FC - Round flange center mount
KC - Square flange center mount

Nominal Shaft Diameter

Retainer Material

Symbol	Specification
G	Polymer Cage

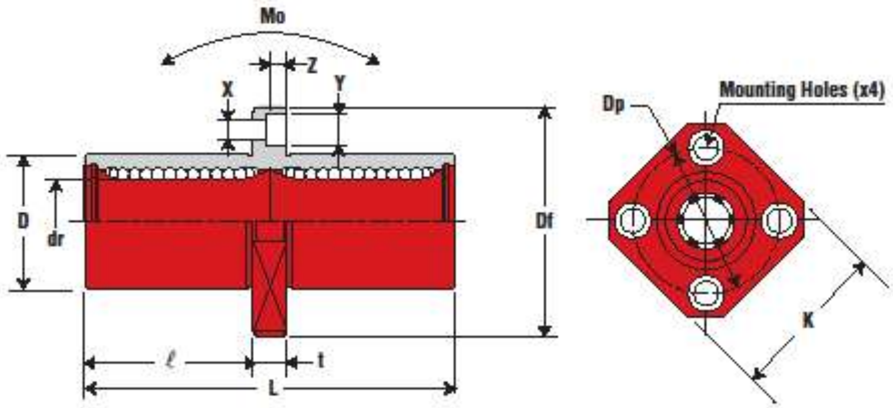
Symbol	Specification
W	Double wide

Modification

Symbol	Specification
No Entry	Standard Type
AJ	Adustable Type
OP	Open Type

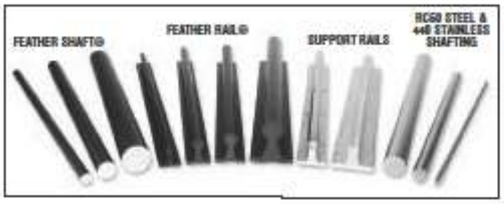
NOTE: Precision of inscribed circle diameters and outside diameters for the clearance adjustable type (…-AJ) and the open type (…-OP) indicates the value obtained before the corresponding type is subjected to cutting process.

Square Flange - Center Mount - IPKC
Linear Ball Bearings



IPKC - DIMENSIONAL INFORMATION

	PART NUMBER	MAJOR DIMENSIONS & TOLERANCES									
NOMINAL SHAFT DIAMETER (inch/ mm)	POLYMER CAGE	BALL CIRCUIT	WEIGHT (g)	dr (inch/ mm)	TOLERANCE (inch/ µm)	D (inch/ mm)	TOLERANCE (inch/ µm)	L (inch/ mm)	TOLERANCE (inch/ mm)	ℓ (inch/ mm)	Df (inch/ mm)
1/4 6.350	IPKC4G	4	33	0.2500 6.350	0/-.0004 0/-10	0.5000 12.700	0/-.0005 0/-13	1.3750 34.925	±0.012 ±0.3	0.5781 14.684	1.2500 31.750
3/8 9.525	IPKC6G		45	0.3750 9.525		0.6250 15.875	0/-.00065 0/-16	1.5938 40.481		0.6719 17.066	1.5000 38.100
1/2 12.700	IPKC8G		106	0.5000 12.700		0.8750 22.225		2.3750 60.235		1.0625 26.988	1.7500 44.450
5/8 15.875	IPKC10G		200	0.6250 15.875		1.1250 28.575		2.8125 71.438		1.2813 32.544	2.0000 50.800
3/4 19.050	IPKC12G	5	240	0.7500 19.050	0/-.0005 0/-12	1.2500 31.750	0/-.00075 0/-19	3.0937 78.581		1.3906 35.322	2.1875 55.563
1 25.400	IPKC16G	6	470	1.0000 25.400		1.5625 39.688		4.2813 108.744		1.9800 50.403	2.5000 63.500
1-1/4 31.750	IPKC20G		935	1.2500 31.750	0/-.0006 0/-15	2.0000 50.800	0/-.0009 0/-22	5.0000 127.000		2.3125 58.738	3.1250 79.375
1-1/2 38.100	IPKC24G		1,460	1.5000 38.100		2.3750 60.325		5.6875 144.463		2.5938 65.882	3.7500 95.250
2 50.800	IPKC32G		2,620	2.0000 50.800		3.0000 76.200	0/-.0010 0/-25	7.7500 196.850		3.6250 92.075	4.3750 111.125



COMPLETE PRODUCT OFFERING

- RC60 Steel
- 440 Stainless Steel
- End Blocks
- Cut-to-Length
- Joinable for longer lengths
- Custom Options
- Support Rails
- Random Lengths
- Pre-drilled and Tapped Shafting

RC60 STEEL SHAFTING

- RC60 case hardened steel
- Polished for optimum surface finish
- Suitable for Simplicity® bearings and linear ball bearings
- Available cut-to-length or in full random lengths

SMALL DIAMETER RC60 STEEL SHAFTING*

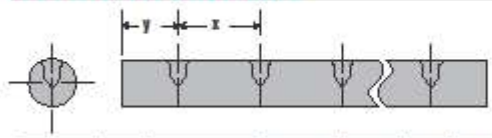
PART NUMBER	NOMINAL SIZE (inches)	DIAMETER TOLERANCE CLASS "L" MIN.	DIAMETER TOLERANCE CLASS "L" MAX.	MAX. LENGTH (in.)	MIN. HARDNESS DEPTH MIN.	WEIGHT PER INCH (lbs.)
NIL02-xx	1/8"	0.1240	0.1245	252	N / A	0.004
NIL03-xx	3/16"	0.1865	0.1870	252	N / A	0.008
NIL04-xx	1/4"	0.2490	0.2495	252	N / A	0.014
NIL06-xx	3/8"	0.3740	0.3745	252	0.040	0.031

RC60 STEEL SHAFTING*

PART NUMBER	NOMINAL SIZE (inches)	DIAMETER TOLERANCE CLASS "L" MIN.	DIAMETER TOLERANCE CLASS "L" MAX.	MAX. LENGTH (in.)	MIN. HARDNESS DEPTH MIN.	WEIGHT PER INCH (lbs.)
NIL02-xx	1/8"	0.1240	0.1245	252	N / A	0.004
NIL03-xx	3/16"	0.1865	0.1870	252	N / A	0.008
NIL04-xx	1/4"	0.2490	0.2495	252	0.030	0.014
NIL06-xx	3/8"	0.3740	0.3745	252	0.030	0.031
NIL08-xx	1/2"	0.4990	0.4995	180	0.060	0.055
NIL10-xx	5/8"	0.6240	0.6245	180	0.060	0.086
NIL12-xx	3/4"	0.7490	0.7495	180	0.060	0.125
NIL16-xx	1"	0.9990	0.9995	204	0.080	0.222
NIL20-xx	1-1/4"	1.2490	1.2495	204	0.080	0.348
NIL24-xx	1-1/2"	1.4989	1.4994	204	0.080	0.500
NIL32-xx	2"	1.9987	1.9994	204	0.100	0.890
NIL40-xx	2-1/2"	2.4985	2.4993	204	0.100	1.391
NIL48-xx	3"	2.9983	2.9992	204	0.100	2.003
NIL64-xx	4"	3.9976	3.9988	204	0.100	3.560

*NOTES: Specify length in part number using inches.
Example: for 1/2" shafting total length 15" = NIL08-15
Surface finish 8 Ra Max.
Surface finish 4 Ra Max. - Instrument Shafting

PRE-DRILLED & TAPPED RC60 STEEL SHAFTING*

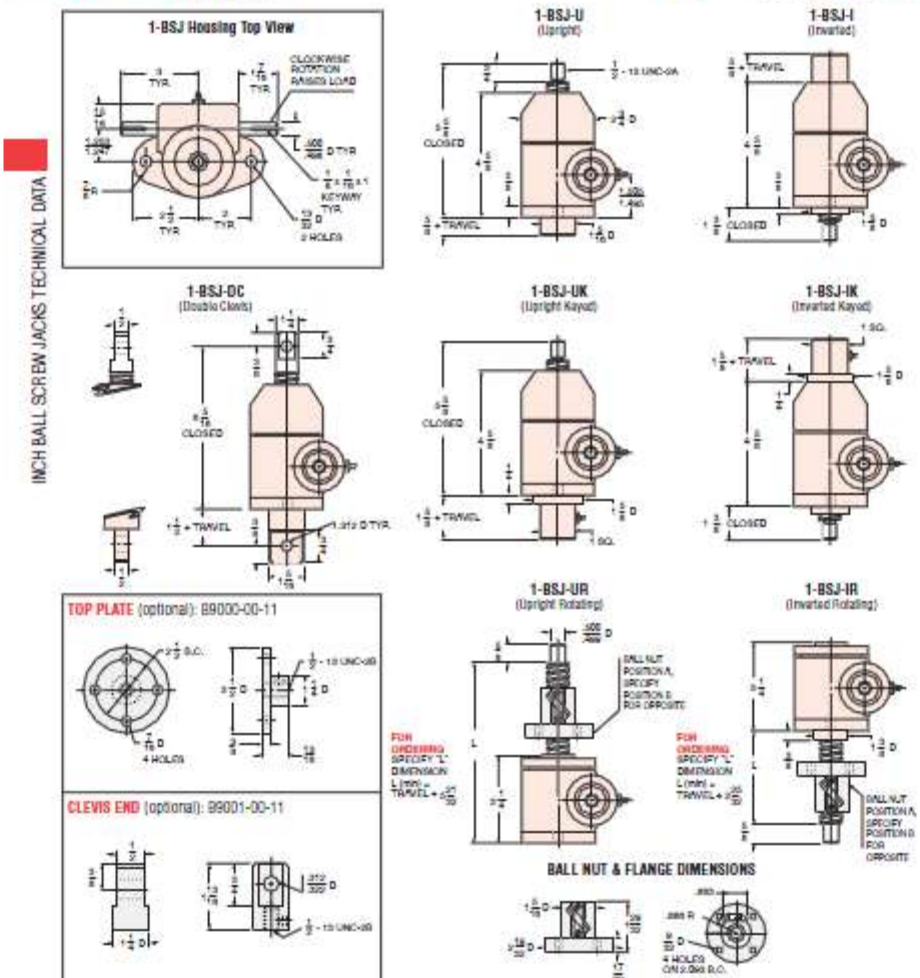


PART NUMBER	NOMINAL SIZE (inches)	DIAMETER TOLERANCE CLASS "L" MIN.	DIAMETER TOLERANCE CLASS "L" MAX.	HOLE SPACING X	HOLE SPACING Y	THREAD	MAX. LENGTH (in.)	WEIGHT PER INCH (lbs.)
NIPDL08-xx	1/2"	0.4990	0.4995	4.00	2.00	6-32	156	0.055
NIPDL10-xx	5/8"	0.6240	0.6245			8-32		0.086
NIPDL12-xx	3/4"	0.7490	0.7495	6.00	3.00	10-32		0.125
NIPDL16-xx	1"	0.9990	0.9995			1/4-20	192	0.222
NIPDL20-xx	1-1/4"	1.2490	1.2495			5/16-18		0.348
NIPDL24-xx	1-1/2"	1.4989	1.4994	8.00	4.00	3/8-16		0.500
NIPDL32-xx	2"	1.9987	1.9994			1/2-13		0.890

NOTES: Specify length in part number using inches.
For random lengths, add "R" to the part number.
Example: for 1/2" shafting total length 13" = NIPDL08-13
Customer specifies "y" dimension.

Nook Industries

INCH BALL SCREW JACKS **1-BSJ**



1-BSJ STANDARD SCREW

SCREW: 0750-0200
ROOT DIAMETER: 0.602
DRAG TORQUE: 3 IN.-LB.
START TORQUE: 2 x Running Torque
WEIGHT (Approx. in Pounds)
"0" TRAVEL: 8
PER INCH TRAVEL: .24
GREASE: .1

RATIO	TURNS OF WORM PER INCH TRAVEL	TORQUE TO RAISE ONE LB.		MAX. HP	MAX. WORM SPEED AT RATED LOAD		MAX. LOAD AT 1750 RPM	
		NON-KEYED	KEYED		NON-KEYED	KEYED	NON-KEYED	KEYED
5:1	25	.0095 in.-lbs.	.0104 in.-lbs.	1/3	1660 rpm	1515 rpm	1895 lbs.	1731 lbs.
20:1	100	.0045 in.-lbs.	.0049 in.-lbs.	1/4	1750 rpm	1608 rpm	2000 lbs.	1837 lbs.

CAUTION! JACK IS SELF-LOWERING. LIFTING SCREW OR NUT MUST BE SECURED TO PREVENT ROTATION FOR NON-KEYED UNITS.

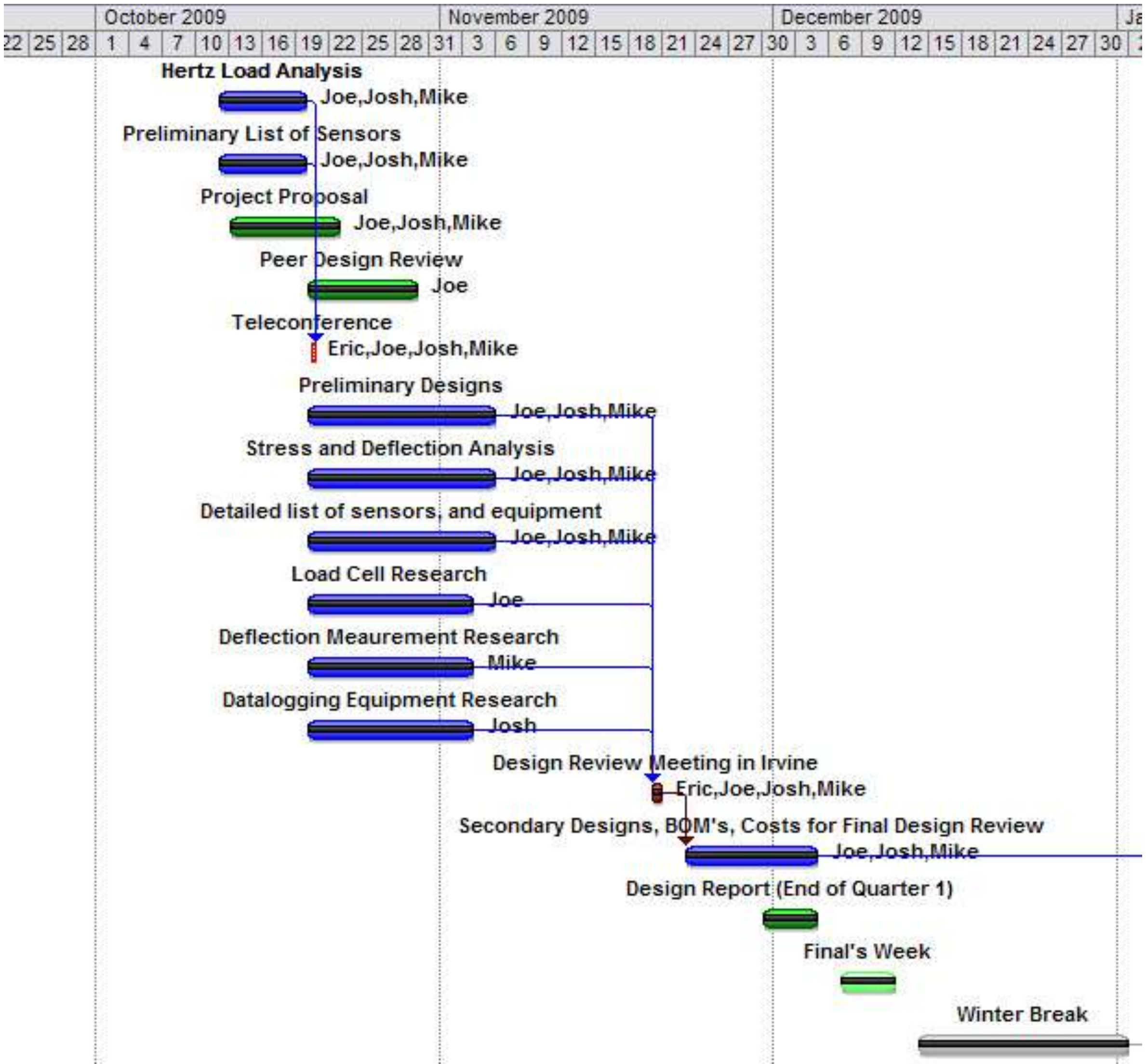
Appendix G

Senior Project Timeline

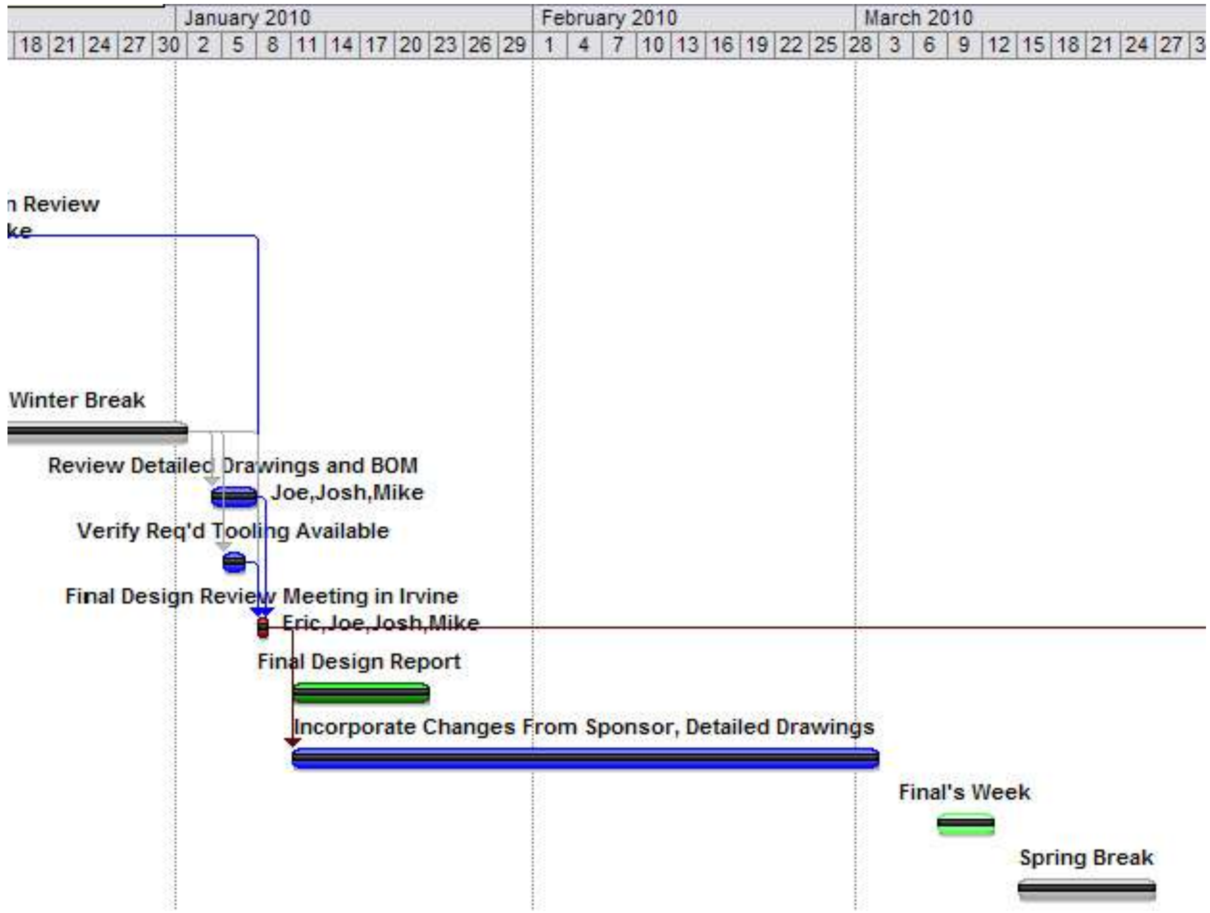
Ghantt Chart Tasks

1	✓	Hertz Load Analysis	6 days	Mon 10/12/09	Mon 10/19/09		Joe,Josh,Mike
2	✓	Preliminary List of Sensors	6 days	Mon 10/12/09	Mon 10/19/09		Joe,Josh,Mike
3	✓	Project Proposal	8 days	Tue 10/13/09	Thu 10/22/09		Joe,Josh,Mike
4	✓	Peer Design Review	8 days	Tue 10/20/09	Thu 10/29/09		Joe
5		Teleconference	1 hr	Tue 10/20/09	Tue 10/20/09	2,1	Eric,Joe,Josh,Mike
6	✓	Preliminary Designs	13 days	Tue 10/20/09	Thu 11/5/09		Joe,Josh,Mike
7	✓	Stress and Deflection Analysis	13 days	Tue 10/20/09	Thu 11/5/09		Joe,Josh,Mike
8	✓	Detailed list of sensors, and equipment	13 days	Tue 10/20/09	Thu 11/5/09		Joe,Josh,Mike
9	✓	Load Cell Research	11 days	Tue 10/20/09	Tue 11/3/09		Joe
10	✓	Deflection Meaurement Research	11 days	Tue 10/20/09	Tue 11/3/09		Mike
11	✓	Datalogging Equipment Research	11 days	Tue 10/20/09	Tue 11/3/09		Josh
12	✓	Design Review Meeting in Irvine	1 day	Fri 11/20/09	Fri 11/20/09	6,7	Eric,Joe,Josh,Mike
13	✓	Secondary Designs, BOM's, Costs for	10 days	Mon 11/23/09	Fri 12/4/09	12	Joe,Josh,Mike
14	✓	Design Report (End of Quarter 1)	5 days	Mon 11/30/09	Fri 12/4/09		
15	✓	Final's Week	5 days?	Mon 12/7/09	Fri 12/11/09		
16	✓	Winter Break	15 days?	Mon 12/14/09	Fri 1/1/10		
17	✓	Review Detailed Drawings and BOM	4 days	Mon 1/4/10	Thu 1/7/10	16	Joe,Josh,Mike
18	✓	Verify Req'd Tooling Available	2 days	Tue 1/5/10	Wed 1/6/10	16	
19	✓	Final Design Review Meeting in Irvine	1 day	Fri 1/8/10	Fri 1/8/10	13,	Eric,Joe,Josh,Mike
20	✓	Final Design Report	10 days	Mon 1/11/10	Fri 1/22/10		
21	✓	Incorporate Changes From Sponsor, D	37 days	Mon 1/11/10	Tue 3/2/10	19	
22	✓	Final's Week	5 days?	Mon 3/8/10	Fri 3/12/10		
23	✓	Spring Break	10 days?	Mon 3/15/10	Fri 3/26/10		
24	✓	Review Detailed Drawings and BOM	5 days	Mon 3/29/10	Fri 4/2/10		Joe,Josh,Mike
25	✓	Submit Parts and Materials Orders	5 days	Mon 4/5/10	Fri 4/9/10	19	
26	✓	Final Test Plan	5 days?	Mon 4/12/10	Fri 4/16/10		
27	✓	Machining	30 days	Tue 4/13/10	Sat 5/22/10		
28	✓	Assembly	10 days?	Tue 5/11/10	Sat 5/22/10		
29		Test of Fixture on AISI 52100	2 days?	Sat 5/22/10	Mon 5/24/10		
30		Data Report of AISI 52100 Testing Res	5 days?	Mon 5/24/10	Fri 5/28/10		
31		HARDWARE DONE	6 days?	Thu 3/18/10	Wed 5/12/10		
32		Prepare Final Report	10 days?	Mon 5/17/10	Thu 5/27/10		
33		Expo and Final Report	5 days?	Mon 5/31/10	Fri 6/4/10	32	

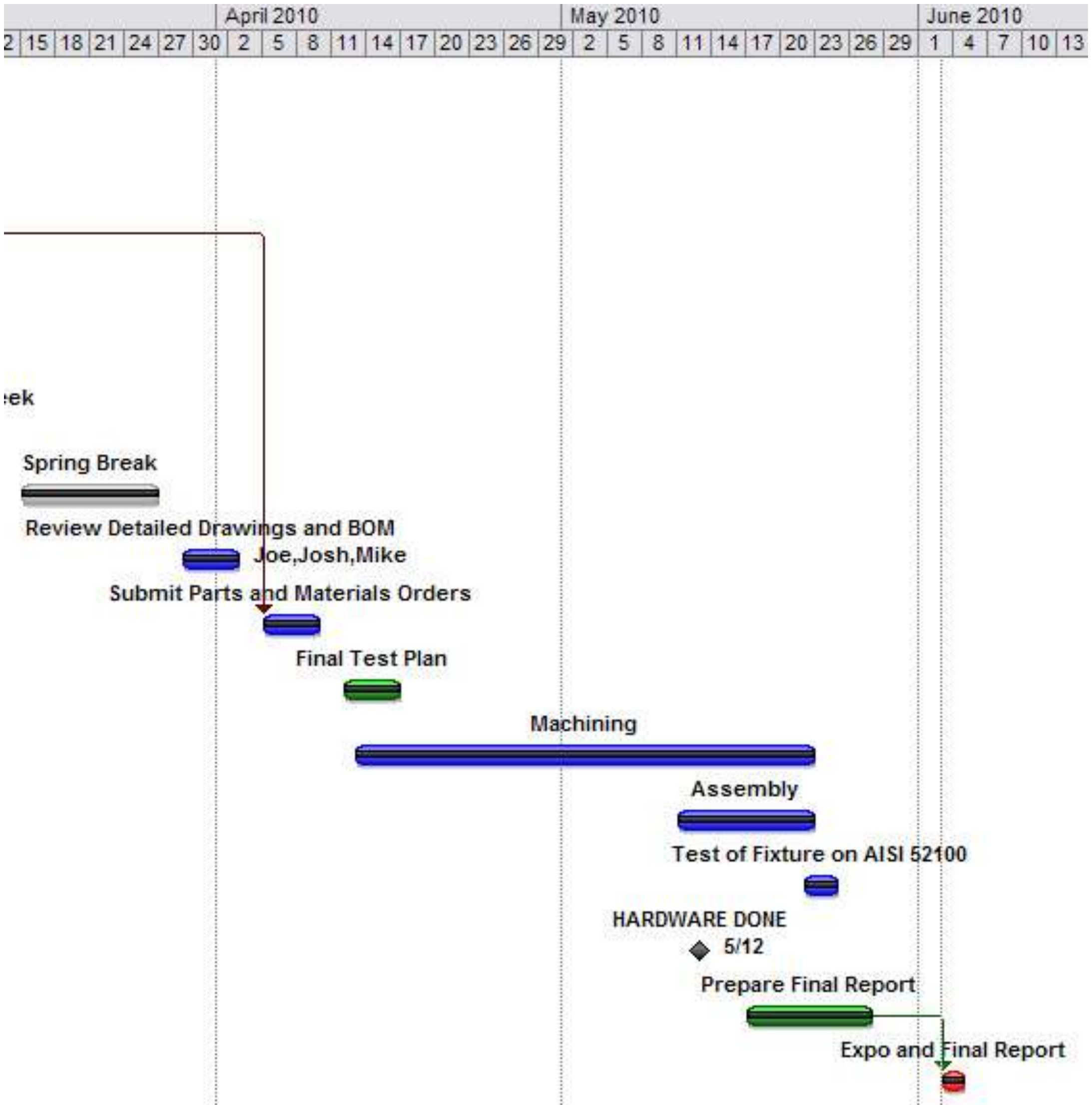
Ghantt Chart for Fall Quarter 2009



Ghantt Chart for Winter Quarter 2010



Ghantt Chart for Spring Quarter 2010



Appendix H

Quality Function Deployment

Brinelling Limit Test Machine
Parker Aerospace
POC: Eric Polcuch
Customers:
1. Shop Techs
2. Engineers

Engineering Requirements / Designs

Customer Requirements	Importance	Load Cell capable of reading up to 250 lbs	Load Cell capable of reading up to 2000 lbs	Compression measurement readout to 0.0001 inches	Fatigue Life of ∞ cycles	Jack able to provide load with an accuracy of 1 lb	Durability of apparatus through temperature cycles	Reliability of instrumentation through temperature range of XX F to XX F	Sensors capable of outputting measurements to a laptop	Accept sample plates with multiple sample locations	Jack motor powered by hand or 110 V electric power source	Sensors powered by 110 V electric power source	Safety enclosure	Thermal enclosure	Compact machine with the ability to rest on tabletop	Easily Transportable	Maintain concentric load path	Concept 1 - Josh's Design	Concept 2 - Joe's Design	Concept 3 - Mike's Design	Concept 4 - Combination Design
Functional Performance																					
Load cell to measure forces*	5	9	9														6	5	5	5	5
Measure distance of compression*	4			6														1	5	1	5
Log data to laptop	3								9									5	5	5	5
Test up to 0.25" roller bearings	5		9	9		6				6								5	5	5	5
Test up to 0.5" ball bearings	5	9		9		9				6								5	5	5	5
Usable in varying thermal environments	4						6							9				1	3	3	3
Human Interaction																					
Safety to operator	4				6								9	6		6		4	4	4	5
Fast to use	3								6	9								5	3	3	4
Easy to use	3								6	6	6	6						5	3	3	4
Easy to understand controls	2										6							5	3	2	4
Physical Requirements																					
Fits on table top / workstation*	4														9	9		5	5	5	5
Mobility*	3										6	6			9	9		4	5	5	5
Adaquate apparatus stiffness*	5			6	6													5	4	4	4
Reliability																					
Validate machine accuracy with known sample data*	5			9		9		6										5	5	5	5
Manufacturing Concerns																					
Safety	3				6		6						9	6		6		5	5	5	5
Machining	4									6					6	6	6	2	3	2	3
Assembly	4														6	6	9	4	4	4	4
Concept #1		90	90	171	84	120	36	30	105	147	84	54	81	63	117	171	78	1521			
Concept #2		90	90	189	78	120	48	30	81	123	66	48	81	81	132	186	84	1527			
Concept #3		90	90	165	78	120	48	30	81	117	60	48	81	81	126	180	78	1473			
Concept #4		90	90	189	84	120	48	30	93	138	78	54	90	87	132	192	84	1599			
																		Total			

9	Strong Correlation
6	Medium Correlation
3	Small Correlation
Blank	No Correlation

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