Brinell Limit Testing Machine-Final Design Report

Prepared for:  Eric Polcuch
               Parker Aerospace - Control Systems Division
               Irvine, CA

Prepared by:  Joe Cloutier –  jcloutie@calpoly.edu
               Joshua Kessler –  jcklessle@calpoly.edu
               Mike Jaskulsky –  mike@jaskulsky.net

California Polytechnic State University, San Luis Obispo
Senior Project
Academic Advisor: Dr. Mase
June 04, 2010
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 The detailed drawings and assembly and procedure instructions of this project are of a confidential nature and will not be published at this time.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.
# Table of Contents

Executive Summary ............................................................................................................. 7

Introduction .......................................................................................................................... 8
  - Sponsor Background and Needs ................................................................................... 8
  - Formal Problem Definition ....................................................................................... 8
  - Objectives .................................................................................................................. 8
  - Management Plan ..................................................................................................... 9

Background ......................................................................................................................... 9

Design Development ......................................................................................................... 12
  - Discussion of Conceptual Designs ........................................................................... 12
  - Design Selection ....................................................................................................... 17

Description of the Final Design ......................................................................................... 20

Detailed Design Description ............................................................................................. 20

Analysis ............................................................................................................................... 22
  - Loading and Deformation Calculations .................................................................. 22
  - Sizing of Structural Members ................................................................................... 24
  - Validation of Structural Stability of Apparatus ......................................................... 25

Safety Considerations ....................................................................................................... 26

Material Selection ............................................................................................................. 27

Maintenance and Repair Considerations .......................................................................... 28

Cost Analysis .................................................................................................................... 28

Product Realization .......................................................................................................... 29
  - Manufacturing .......................................................................................................... 29
  - Assembly ................................................................................................................... 31

Design Verification ........................................................................................................... 31

Conclusions and Recommendations ................................................................................ 32

Appendix A ......................................................................................................................... 36
  - Analysis Support ........................................................................................................ 36
    - Cal Poly Brinelling Calcs 091206 ......................................................................... 36
    - Roark and Young Stress and Deflection Hand Calculations ............................. 38
    - Column Deflection ................................................................................................. 42
<table>
<thead>
<tr>
<th>Section</th>
<th>Page</th>
</tr>
</thead>
<tbody>
<tr>
<td>Excel Column Rod Calculations</td>
<td>44</td>
</tr>
<tr>
<td>Excel Spreadsheet of Calculations of Rectangular Tubing</td>
<td>45</td>
</tr>
<tr>
<td>Excel Spreadsheet of Calculations of Flat Plate as Horizontal Member</td>
<td>46</td>
</tr>
<tr>
<td>Excel Spreadsheet of Buckling Calculations for Crush Sleeves</td>
<td>47</td>
</tr>
<tr>
<td>Thread Calculations</td>
<td>48</td>
</tr>
<tr>
<td>Excel Spreadsheet Comparing Chosen Crush Sleeve Strength to Actual Load</td>
<td>49</td>
</tr>
<tr>
<td>Hand Calculations Verifying Stability of Final Design</td>
<td>50</td>
</tr>
<tr>
<td>Appendix B</td>
<td>53</td>
</tr>
<tr>
<td>Appendix C</td>
<td>55</td>
</tr>
<tr>
<td>Detailed Drawings</td>
<td>55</td>
</tr>
<tr>
<td>Appendix D</td>
<td>56</td>
</tr>
<tr>
<td>Assembly Instructions</td>
<td>56</td>
</tr>
<tr>
<td>Appendix E</td>
<td>56</td>
</tr>
<tr>
<td>Vendors</td>
<td>56</td>
</tr>
<tr>
<td>Acuity Laser</td>
<td>56</td>
</tr>
<tr>
<td>Appendix F</td>
<td>58</td>
</tr>
<tr>
<td>Vendor Literature</td>
<td>58</td>
</tr>
<tr>
<td>Boeckeler</td>
<td>59</td>
</tr>
<tr>
<td>Chicago Dial Indicators</td>
<td>61</td>
</tr>
<tr>
<td>Futek</td>
<td>62</td>
</tr>
<tr>
<td>Pacific-Bearing</td>
<td>63</td>
</tr>
<tr>
<td>Nook Industries</td>
<td>66</td>
</tr>
<tr>
<td>Appendix G</td>
<td>67</td>
</tr>
<tr>
<td>Senior Project Timeline</td>
<td>67</td>
</tr>
<tr>
<td>Ghantt Chart Tasks</td>
<td>67</td>
</tr>
<tr>
<td>Ghantt Chart for Fall Quarter 2009</td>
<td>68</td>
</tr>
<tr>
<td>Ghantt Chart for Winter Quarter 2010</td>
<td>69</td>
</tr>
<tr>
<td>Ghantt Chart for Spring Quarter 2010</td>
<td>70</td>
</tr>
<tr>
<td>Appendix H</td>
<td>71</td>
</tr>
<tr>
<td>Quality Function Deployment</td>
<td>71</td>
</tr>
<tr>
<td>References</td>
<td>72</td>
</tr>
</tbody>
</table>
Table of Figures
Figure 1. 3000 BLD Brinell Tester from Wilson Instruments..............................................10
Figure 2. CLB3 Hardness Tester from Wilson Instruments..................................................11
Figure 4. Load cell above jack ram...................................................................................13
Figure 3. Original design provided by Parker.......................................................................13
Figure 5. Load cell on base and jack on top.........................................................................14
Figure 6. Example of peg-board system..............................................................................14
Figure 7. Flange-mount linear bearing drawing..................................................................15
Figure 8. Lever Arm system for displacement measuring....................................................16
Figure 9. Dial Indicator displacement measuring.................................................................16
Figure 10. Design Concept 1 – Four post design with single sliding track.........................17
Figure 11. Design Concept 2 – Two-post design with peg-board sample plate locator........18
Figure 12. Design Concept 3 – Two post design with modified peg-board sample plate locator..........................................................................................................................19
Figure 13. Overall apparatus...............................................................................................20
Figure 14. Close-up of sliding plate system and load cell area............................................21
Figure 15. Spalling due to excessive load on the bearings....................................................27
Figure 16. Milling of the track system..................................................................................30
Figure 17. Dial indicator plate being machined on a CNC mill and base plate after welding..........................................................................................................................30
Figure 18. Final assembly of brinell limit test machine..........................................................31
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 Gantt 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.](image)

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.\(^3\)
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.](image)

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.](image)
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.](image)

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.](image)

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.

![Flange-mount linear bearing drawing](image)

**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.

![Design Concept 1](image)

**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 4th concept 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.](image)

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.](image)

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 Hertziann 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 Shigley's, Hertz found these two values to be related by the following expressions,

\[ F = 2\pi a^2 p_{\text{max}} \]  \hspace{2cm} (1)

and

\[ a = \left( \frac{3F}{8E} \right)^{1/3} \left( \frac{1-v_1^2}{d_1} + \frac{1-v_2^2}{d_2} \right) \]  \hspace{2cm} (2)

where \( F \) is the force between the two bodies and \( a \) is the radius of the resulting contact area. Also \( p_{\text{max}} \) is the maximum stress to brinelling, or permanent surface deformation of the material, and \( v \) 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 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 Brinellling 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,

\[
F = \frac{nhl p_{\text{max}}}{2}
\]  
(3)

and

\[
a = \left( \frac{nhp_{\text{max}}}{2} \right) \left( \frac{1 - \nu_1^2}{E_1} + \frac{1 - \nu_2^2}{E_2} \right) \left( \frac{1}{d_1} + \frac{1}{d_2} \right)
\]  
(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

\[
\delta_{\text{max}} = \left( \frac{g}{2} \right) \left( \frac{1 - \nu_1^2}{E_1} \right)^2 F \left( \frac{1}{d_1} \right)^2
\]  
(5)
and for the cylinder on a flat plate, we calculated a deformation of 0.0011223 inches using

$$\delta_{\text{max}} = 2 \frac{F}{l} \left( \frac{1 - \nu^2}{\pi E_i} \right) \left( 1 + \ln \left( \frac{l^2}{\frac{1 - \nu^2}{\pi E_i} l F d_1} \right) \right)$$

(S6)

**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\[^4\]

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

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.

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.

![Milling of the track system](image16.jpg)

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.

![Dial indicator plate being machined on a CNC mill and base plate after welding.](image17.jpg)

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.

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 500 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

\[ p_{\text{max}} = 570 [\psi] \cdot 10^3 \]  \hspace{1cm} (1)

Poisson’s Ratio; Shigley’s Table

\[ \nu_1 = 0.3 \]  \hspace{1cm} (2)

\[ \nu_2 = 0.3 \]  \hspace{1cm} (3)

Modulus of Elasticity; Shigley’s Table

\[ E_1 = 30.5 [\psi] \cdot 10^6 \]  \hspace{1cm} (4)

\[ E_2 = 30.5 [\psi] \cdot 10^6 \]  \hspace{1cm} (5)

Diameter of steel ball

\[ d_1 = 0.5 \text{ [in]} \]  \hspace{1cm} (6)

Diameter of opposing contact surface; Infinity for flat plate

\[ d_2 = 1 \text{ [in]} \cdot 10^{20} \]  \hspace{1cm} (7)

Length of roller bearings

\[ l = 0.5 \text{ [in]} \]  \hspace{1cm} (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

\[ F_a = 2 \cdot \pi \cdot a^2 \cdot p_{\text{max}} / 3 \]  \hspace{1cm} (9)

Radius of contact area; Shigley’s pg 117

\[ a = \left( \frac{3 \cdot F_a}{8} \cdot \left( \frac{1 - \nu_1^2}{E_1} + \frac{1 - \nu_2^2}{E_2} \right) \right)^{1/3} \]  \hspace{1cm} (10)
Determining Maximum Deflection of ball

**Governing Equation**

\[
\delta_{\text{max}} = \frac{a^2}{d_1/2}
\]

\[\delta_{\text{max}2} = (9/2)^{1/3} \cdot \left( \frac{1-\nu_1}{E_1} \right)^{2/3} \cdot F_a^{2/3} \cdot (1/d_1)^{1/3}\]  

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

\[
F_b = \frac{\pi \cdot b \cdot l \cdot p_{\text{max}}}{2}
\]

Radius of contact area; Shigley's pg 118

\[
b = \left( \frac{2 \cdot \pi \cdot I \cdot p_{\text{max}}/2}{\pi \cdot l} \cdot \left( \frac{1-x_1^2}{E_1} + \frac{1-x_2^2}{E_2} \right) \right)^{1/3}
\]

**Solution**

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

\[ \sigma_{\text{max}} = 1.616 \sqrt{\frac{PE^2}{K_d^2}} \]

since \( K_d = D_a \)

\[ \sigma_{\text{max}} = 1.616 \sqrt{\frac{PE^2}{D_a^2}} \]

Solve for \( P = \text{total load} \)

\[ \left( \frac{\sigma_{\text{max}}}{1.616} \right)^2 \left( \frac{PE^2}{D_a^2} \right) \]

\[ \left( \frac{\sigma_{\text{max}}}{1.616} \right)^3 = \frac{PE^2}{D_a^2} \]

\[ \left( \frac{\sigma_{\text{max}}}{1.616} \right)^3 D_a^2 = P \]

\[ \left( \frac{570,000 \%}{1.616} \right)^3 \left( \frac{1.12}{30,500,000 \%^2} \right) = P \]

\( P \) for 1" Dia ball = 851.69 lb

\( P \) for 1.5" Dia ball = 212.9 lb

\[ a = 0.981 \sqrt{\frac{PK_e}{E}} \]

since \( K_e = D_e \)

\[ a = 0.991 \sqrt{\frac{PD_a}{E}} \]

\[ a = 0.991 \sqrt{\left( \frac{12.9}{30,500,000 \%} \right)} \]

\( a \) for 1.5" Dia ball = 0.01336 in

\( a \) for 2.5" Dia ball

\[ y = 1.55 \sqrt[3]{\frac{D^2}{E^2 K_b}} \]

Since \( K_b = D_2 \)

\[ y = 1.55 \sqrt[3]{\frac{D^2}{E^2 D_2}} \]

\[ y = 1.55 \sqrt[3]{\frac{212.9}{(30,500,000)^2}} \] (1.5)

\[ y \text{ for } 0.5'' \text{ Dia. ball} = 0.000713 \text{ in} \]
2) \( T_{\text{emin}} = 0.591 \sqrt{\frac{P \cdot E}{k_e}} \)

\( P = \frac{L}{E} \quad \text{and} \quad k_e = D_e \)

\( T_{\text{emin}} = 0.591 \sqrt{\frac{P \cdot E}{L \cdot D_e}} \)

\( \frac{T_{\text{emin}}}{0.591} = \sqrt{\frac{P \cdot E}{L \cdot D_e}} \)

\( \left( \frac{T_{\text{emin}}}{0.591} \right)^2 = \frac{P \cdot E}{L \cdot D_e} \)

\( \frac{P \cdot E}{L \cdot D_e} = \frac{370,000 \cdot (6.5)(5)}{30500000} = P \)

\( P \) for .5" Dia./.5" Length = 7624.56 lb

\( P \) for .25" Dia./.25" Length = 1906.14 lb

\( b = 2.15 \sqrt{\frac{P \cdot k_e}{E}} \)

\( P = \frac{L}{E} \quad \text{and} \quad k_e = D_e \)

\( b = 2.15 \sqrt{\frac{P \cdot D_e}{L \cdot E}} \)

\( b = 2.15 \sqrt{\frac{(7624.56)(5)(5)}{(5)(30500000)}} \)

\( b \) for .5" Dia./.5" L = .03399 in

\( b \) for .25" Dia./.25" L = .01699 in
\[\Delta D_L = \frac{4 \rho (1 - \nu^2)}{\pi E} \left( \frac{1}{3} + \ln \frac{\Delta D}{b} \right)\]

\[\varphi = \frac{E}{L}\]

\[\Delta D_L = \frac{4 \rho (1 - \nu^2)}{\pi E L} \left( \frac{1}{3} + \ln \frac{2D}{b} \right)\]

Assuming \(D = D_L\)

\[\Delta D_L = \frac{4 (764.36)(1 - 5^3)}{\pi (50,000,000)(15)} \left( \frac{1}{3} + \ln \frac{2(5)}{.02394} \right)\]

\[\Delta D_L \text{ for } .5'' \text{ Dia} / .5'' \text{ L} = (579.19056 \times 10^{-6})(3.715) = .00215 \text{ in}\]

\[\Delta D_L \text{ for } .25'' \text{ Dia} / .35'' \text{ L} = (2.89453813 \times 10^{-6})(3.715) = .001076 \text{ in}\]
Equations

Calculation of Stresses, Strains, and Deflections of a Steel Tube and a Solid Steel Cylinder
Prepared by Joshua Kessler, Mike Jaskulsky, and Joe Cloutier, Cal Poly, SLO
For Eric Polcuch, Parker Aerospace
November 1, 2009

Parameter List

Modulus of Elasticity
\[ E = 30 \cdot 10^6 \text{ [lb/in}^2\text{]} \] (1)

Diameter of Solid Steel Cylinder
\[ D_s = 0.75 \text{ [in]} \] (2)

Outer and Inner Diameter of Steel Tube
\[ D_o = 1.5 \text{ [in]} \] (3)
\[ D_i = 1.25 \text{ [in]} \] (4)

Length of Steel Cylinder/Tube
\[ L = 12 \text{ [in]} \] (5)

Load Applied - 1000lb distributed over 4 Steel Cylinders/Tubes
\[ P = \frac{F}{2} \text{ [lb]} \] (6)
\[ F = 1000 \text{ [lb]} \] (7)

Stress of Steel Tube
\[ \sigma_t = \frac{P}{A_t} \] (8)

Cross Sectional Area of Steel Tube
\[ A_t = \frac{\pi}{4} \cdot (D_o^2 - D_i^2) \] (9)

Stress of Solid Steel Cylinder
\[ \sigma_s = \frac{P}{A_s} \] (10)

Cross Sectional Area of Solid Steel Cylinder
\[ A_s = \frac{\pi}{4} \cdot D_s^2 \] (11)

Strain of Steel Tube
\[ \epsilon_t = \frac{\sigma_t}{E} \] (12)
Strain of Solid Steel Cylinder

\[ \epsilon_s = \frac{\sigma_s}{E} \]  \hspace{1cm} (13)

Deflection of Steel Tube

\[ \delta_t = \epsilon_t \cdot L \]  \hspace{1cm} (14)

Deflection of Solid Steel Cylinder

\[ \delta_s = \epsilon_s \cdot L \]  \hspace{1cm} (15)

Crossbar Analysis

\[ L_{cb} = 14 \text{ [in]} \]  \hspace{1cm} (16)

\[ h = 1 \text{ [in]} \]  \hspace{1cm} (17)

\[ b = 8 \text{ [in]} \]  \hspace{1cm} (18)

\[ I = \frac{(1/12) \cdot b \cdot h^3}{48 \cdot E \cdot I} \]  \hspace{1cm} (19)

\[ y_{max} = \frac{F \cdot \epsilon_s^3}{48 \cdot E \cdot I} \]  \hspace{1cm} (20)

Solution

\begin{array}{ll}
A_s = 0.4418 & A_t = 0.54 \\
b = 8 \text{ [in]} & \delta_s = 0.0004527 \\
\delta_t = 0.0003704 & D_t = 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
\end{array}
Column Rods

<table>
<thead>
<tr>
<th>Parameters</th>
</tr>
</thead>
<tbody>
<tr>
<td>( P ) [lb]</td>
</tr>
<tr>
<td>( L ) [in]</td>
</tr>
<tr>
<td>( E ) [ps]</td>
</tr>
</tbody>
</table>

\[
\sigma = \frac{P}{A} \\
\varepsilon = \frac{\sigma}{E} \\
\Delta = \frac{\Delta}{L} \\
k = \frac{dF}{d\Delta}
\]

<table>
<thead>
<tr>
<th>Rod Diameter</th>
<th>Stress [ps]</th>
<th>Strain ( \varepsilon/E )</th>
<th>Elongation ( \Delta/L )</th>
<th>Stiffness [lb/in]</th>
<th>Weight [lb/shift]</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.5</td>
<td>5093</td>
<td>0.00017</td>
<td>0.00289</td>
<td>346499</td>
<td>1.02</td>
</tr>
<tr>
<td>0.625</td>
<td>3259</td>
<td>0.00011</td>
<td>0.00185</td>
<td>541405</td>
<td>1.53</td>
</tr>
<tr>
<td>0.75</td>
<td>2264</td>
<td>0.00008</td>
<td>0.00128</td>
<td>779623</td>
<td>2.21</td>
</tr>
<tr>
<td>1</td>
<td>1273</td>
<td>0.00004</td>
<td>0.00072</td>
<td>1385997</td>
<td>3.74</td>
</tr>
<tr>
<td>1.25</td>
<td>815</td>
<td>0.00003</td>
<td>0.00046</td>
<td>2165620</td>
<td>5.95</td>
</tr>
</tbody>
</table>
Cross Braces

<table>
<thead>
<tr>
<th>Parameters</th>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>P [lb]</td>
<td>2000</td>
<td></td>
<td></td>
</tr>
<tr>
<td>L [in]</td>
<td>12</td>
<td></td>
<td></td>
</tr>
<tr>
<td>E [psi]</td>
<td>30000000</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Excel Spreadsheet of Calculations of Rectangular Tubing

\[ I = \left(\frac{1}{12}\right) \times W \times H^3 \]

\[ I_{lower} = I_{upper} - I_{inner} \]

\[ \sigma = \frac{P \times L \times H}{4 \times I_{lower}} \]

\[ \delta = \frac{P \times L^3}{48 \times E \times I} \]

\[ W = A \times L \times \rho \]

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

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

| Rectangular Cross Section | | | | | | |
|---|---|---|---|---|---|
| 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 | 6.9535 | 1724 | 0.00034 | 2.87 | 9.77 |
| 2 | 3 | 0.25 | 2.5469 | 3534 | 0.00094 | 2.25 | 7.67 |
| 3 | 3 | 0.25 | 3.4848 | 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 | | | | | |
|---|---|---|---|---|
| 2 | 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 |
|-----------|-------|-------|--------------------------|---------------------|------------------------|------------|------------|
|           | 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   | 0.0511                   | 28800               | 0.04608                | 2.50       | 8.52       |
|           | 5     | 1     | 0.4167                   | 7200                | 0.00576                | 5.00       | 17.04      |
|           | 5     | 1.5   | 1.4063                   | 3200                | 0.01171                | 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

<table>
<thead>
<tr>
<th>Parameters</th>
<th>P [lb]</th>
<th>E [ksi]</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>2000</td>
<td>3000000</td>
</tr>
</tbody>
</table>

| Sy | 30000 |
| C  | 4     |

### Pipe Stock from Tubularsteel.com

<table>
<thead>
<tr>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>0.095</td>
<td>2.624</td>
<td>0.405</td>
<td>0.2150</td>
<td>0.0012</td>
<td>0.115</td>
<td>0.093</td>
<td>23</td>
<td>209124</td>
<td>0.059</td>
<td>2766.387</td>
<td>0.0689473</td>
</tr>
<tr>
<td></td>
<td>2.5</td>
<td>0.405</td>
<td>0.2150</td>
<td>0.0012</td>
<td>0.115</td>
<td>0.093</td>
<td>22</td>
<td>230384</td>
<td>0.066</td>
<td>2767.237</td>
<td>0.0656891</td>
</tr>
<tr>
<td></td>
<td>3.624</td>
<td>0.405</td>
<td>0.2150</td>
<td>0.0012</td>
<td>0.115</td>
<td>0.093</td>
<td>32</td>
<td>109637</td>
<td>0.095</td>
<td>2758.030</td>
<td>0.095223</td>
</tr>
<tr>
<td></td>
<td>3.5</td>
<td>0.405</td>
<td>0.2150</td>
<td>0.0012</td>
<td>0.115</td>
<td>0.093</td>
<td>31</td>
<td>117543</td>
<td>0.092</td>
<td>2759.212</td>
<td>0.0919648</td>
</tr>
<tr>
<td>0.119</td>
<td>2.624</td>
<td>0.54</td>
<td>0.3020</td>
<td>0.0038</td>
<td>0.155</td>
<td>0.157</td>
<td>17</td>
<td>647722</td>
<td>0.117</td>
<td>4713.114</td>
<td>0.11729</td>
</tr>
<tr>
<td></td>
<td>2.5</td>
<td>0.54</td>
<td>0.3020</td>
<td>0.0038</td>
<td>0.155</td>
<td>0.157</td>
<td>16</td>
<td>713570</td>
<td>0.112</td>
<td>4713.909</td>
<td>0.1117474</td>
</tr>
<tr>
<td></td>
<td>3.624</td>
<td>0.54</td>
<td>0.3020</td>
<td>0.0038</td>
<td>0.155</td>
<td>0.157</td>
<td>23</td>
<td>339578</td>
<td>0.152</td>
<td>4705.306</td>
<td>0.161989</td>
</tr>
<tr>
<td></td>
<td>3.5</td>
<td>0.54</td>
<td>0.3020</td>
<td>0.0038</td>
<td>0.155</td>
<td>0.157</td>
<td>23</td>
<td>364066</td>
<td>0.156</td>
<td>4706.410</td>
<td>0.1564646</td>
</tr>
<tr>
<td>0.091</td>
<td>2.624</td>
<td>0.675</td>
<td>0.4930</td>
<td>0.0073</td>
<td>0.209</td>
<td>0.167</td>
<td>13</td>
<td>1254942</td>
<td>0.124</td>
<td>5003.703</td>
<td>0.1244189</td>
</tr>
<tr>
<td></td>
<td>2.5</td>
<td>0.675</td>
<td>0.4930</td>
<td>0.0073</td>
<td>0.209</td>
<td>0.167</td>
<td>12</td>
<td>1381528</td>
<td>0.119</td>
<td>5004.164</td>
<td>0.1185393</td>
</tr>
<tr>
<td></td>
<td>3.624</td>
<td>0.675</td>
<td>0.4930</td>
<td>0.0073</td>
<td>0.209</td>
<td>0.167</td>
<td>13</td>
<td>657451</td>
<td>0.172</td>
<td>4999.164</td>
<td>0.1718346</td>
</tr>
<tr>
<td></td>
<td>3.5</td>
<td>0.675</td>
<td>0.4930</td>
<td>0.0073</td>
<td>0.209</td>
<td>0.167</td>
<td>17</td>
<td>704861</td>
<td>0.166</td>
<td>4999.806</td>
<td>0.1659551</td>
</tr>
<tr>
<td>0.126</td>
<td>2.624</td>
<td>0.675</td>
<td>0.4230</td>
<td>0.0086</td>
<td>0.199</td>
<td>0.217</td>
<td>13</td>
<td>1482500</td>
<td>0.152</td>
<td>6512.328</td>
<td>0.1619478</td>
</tr>
<tr>
<td></td>
<td>2.5</td>
<td>0.675</td>
<td>0.4230</td>
<td>0.0086</td>
<td>0.199</td>
<td>0.217</td>
<td>13</td>
<td>1633211</td>
<td>0.154</td>
<td>6512.990</td>
<td>0.1542947</td>
</tr>
<tr>
<td></td>
<td>3.624</td>
<td>0.675</td>
<td>0.4230</td>
<td>0.0086</td>
<td>0.199</td>
<td>0.217</td>
<td>18</td>
<td>777724</td>
<td>0.224</td>
<td>6505.824</td>
<td>0.2236656</td>
</tr>
<tr>
<td></td>
<td>3.5</td>
<td>0.675</td>
<td>0.4230</td>
<td>0.0086</td>
<td>0.199</td>
<td>0.217</td>
<td>18</td>
<td>833271</td>
<td>0.216</td>
<td>6506.744</td>
<td>0.2160126</td>
</tr>
</tbody>
</table>
Thread Calculations

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

<table>
<thead>
<tr>
<th>Constants</th>
</tr>
</thead>
<tbody>
<tr>
<td>K</td>
</tr>
<tr>
<td>b</td>
</tr>
<tr>
<td>m</td>
</tr>
<tr>
<td>b</td>
</tr>
<tr>
<td>m</td>
</tr>
<tr>
<td>Sp(5)</td>
</tr>
<tr>
<td>Sp(8)</td>
</tr>
<tr>
<td>St(5)</td>
</tr>
<tr>
<td>St(8)</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Bolt Dimensions</th>
<th>SAE GRADE 5</th>
</tr>
</thead>
<tbody>
<tr>
<td>10</td>
<td>24</td>
</tr>
<tr>
<td>10</td>
<td>32</td>
</tr>
<tr>
<td>12</td>
<td>24</td>
</tr>
<tr>
<td>12</td>
<td>28</td>
</tr>
<tr>
<td>0.25</td>
<td>20</td>
</tr>
<tr>
<td>0.25</td>
<td>28</td>
</tr>
<tr>
<td>0.3125</td>
<td>18</td>
</tr>
<tr>
<td>0.3125</td>
<td>24</td>
</tr>
<tr>
<td>0.375</td>
<td>16</td>
</tr>
<tr>
<td>0.375</td>
<td>24</td>
</tr>
<tr>
<td>0.4375</td>
<td>14</td>
</tr>
<tr>
<td>0.4375</td>
<td>20</td>
</tr>
</tbody>
</table>
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

<table>
<thead>
<tr>
<th>Bolt Size</th>
<th>TPI</th>
<th>At [in²]</th>
<th>Breakthrough load in 4 lb</th>
<th>Total load [lbs]</th>
<th>Smallest crush sleeve</th>
<th>Margin [lbs]</th>
</tr>
</thead>
<tbody>
<tr>
<td>10</td>
<td>24</td>
<td>0.0175</td>
<td>2100</td>
<td>8400</td>
<td>10400</td>
<td>11065.5</td>
</tr>
<tr>
<td>12</td>
<td>24</td>
<td>0.0292</td>
<td>2304</td>
<td>11616</td>
<td>13616</td>
<td>11065.5</td>
</tr>
<tr>
<td>12</td>
<td>28</td>
<td>0.0258</td>
<td>3096</td>
<td>12384</td>
<td>14384</td>
<td>11065.5</td>
</tr>
<tr>
<td>0.25</td>
<td>20</td>
<td>0.0318</td>
<td>3816</td>
<td>15264</td>
<td>17264</td>
<td>11065.5</td>
</tr>
<tr>
<td>0.25</td>
<td>28</td>
<td>0.0364</td>
<td>4368</td>
<td>17472</td>
<td>19472</td>
<td>11065.5</td>
</tr>
<tr>
<td>0.3125</td>
<td>18</td>
<td>0.0524</td>
<td>6288</td>
<td>25152</td>
<td>27152</td>
<td>11065.5</td>
</tr>
<tr>
<td>0.3125</td>
<td>24</td>
<td>0.0584</td>
<td>6960</td>
<td>27840</td>
<td>29840</td>
<td>11065.5</td>
</tr>
<tr>
<td>0.375</td>
<td>16</td>
<td>0.0775</td>
<td>9300</td>
<td>37200</td>
<td>39200</td>
<td>11065.5</td>
</tr>
<tr>
<td>0.375</td>
<td>24</td>
<td>0.0878</td>
<td>10536</td>
<td>42144</td>
<td>44144</td>
<td>11065.5</td>
</tr>
<tr>
<td>0.4375</td>
<td>14</td>
<td>0.1063</td>
<td>12756</td>
<td>51024</td>
<td>53024</td>
<td>11065.5</td>
</tr>
<tr>
<td>0.4375</td>
<td>20</td>
<td>0.1187</td>
<td>14244</td>
<td>56976</td>
<td>58976</td>
<td>11065.5</td>
</tr>
</tbody>
</table>
Hand Calculations Verifying Stability of Final Design

**Solution:**

\[
E = F \cdot (23.52 \text{ in}) \cdot (-1) = mg \cdot (6.00 \text{ in}) = 0
\]

\[
F = \frac{mg(6.00 \text{ in})}{23.52} = \frac{80.4(6.0)}{23.52} = 20.41 \text{ lbf}
\]

**Increasing Base Width Increases At Force**

<table>
<thead>
<tr>
<th>Base Width</th>
<th>Force to Tip</th>
</tr>
</thead>
<tbody>
<tr>
<td>14°</td>
<td>23.81 lbf</td>
</tr>
<tr>
<td>16°</td>
<td>27.21 lbf</td>
</tr>
<tr>
<td>18°</td>
<td>30.44 lbf</td>
</tr>
</tbody>
</table>

**Additional Consideration**

**Recon:** Determine deflection of apparatus under \( F = 20.41 \text{ lbf} \)

**Eqn:**

\[
S_{max} = \frac{F}{3E} I \text{ s/ft}^3, \text{ from Nautec.com for AISI 904A Stainless Steel E = 30\times10^6 \text{ psi}}
\]

\[
F = 20.41 \text{ lbf} = 0.25 \times 10^{-6} \text{ MN}
\]

\[
S_{max} = \frac{10.25 \times 10^{-6} (23.52 \text{ in})^3}{3(250 \times 10^6) (0.05 \text{ ft})} = 0.00826 \text{ in}
\]

\[
S_{max} = 0.0983 \text{ in}
\]
Given: Half-model of test apparatus. Component weights and height of center of gravity are obtained from solid modeling program.

Reqd: Find natural frequency of test apparatus.

Schematic:

Assumptions: Vertical section rigidly attached to base.

Solution:

Determine mass equivalent and spring of equivalent on half-model:

\[ k_e = \frac{m_e}{l} \]

For cantilever beam under transverse shear load:

\[ C = \frac{P}{Z} \]

For circular cross-section:

\[ k_e = \frac{FL}{I} \frac{2EI}{Z} \]

\[ k_e = \frac{\pi (0.375 \text{ in})^3}{2 (25.52 \text{ in})} \]

\[ k_e = 35.3 \times 10^3 \text{ lb in} \]

Real = 2.74 lb in

2.94 lb in
[Continuation of finding natural frequency]

**Elastic Model**

\[ FBD \]

\[ M_{eq} = m_{e}L \sin \theta - k_{e} \theta = I \ddot{\theta} \]

\[ m_{e}L \theta - k_{e} \theta = m_{e}L \theta \]

\[ \omega = \left[ \frac{m_{e}L - k_{e}}{m_{e}L} \right]^{\frac{1}{2}} \]

\[ = \left[ \frac{(0.28 \text{ lb}) (25.52 \text{ in}) - (35.3 \times 10^{-3} \text{ ft}^{3})}{0.03 \text{ ft}} \right]^{\frac{1}{2}} \]

\[ = \left[ \frac{35.02 \times 10^{-3} \text{ in} \cdot \text{ft}}{222.8 \text{ in}^{2} \text{ lbs} \cdot \text{sec}^{-2}} \cdot \text{sec} \right] \]

\[ = \left( 35.02 \times 10^{-3} \text{ in} \cdot \text{ft} \right) \]

\[ \omega = (1890 \text{ rad/sec})^{\frac{1}{2}} \]

\[ \omega = 13.48 \text{ rad/sec} \]

\[ f = \frac{\omega}{2\pi} \]

\[ f = 6.92 \text{ cycles/sec} \]
# Appendix B

## BOM

<table>
<thead>
<tr>
<th>Part</th>
<th>Sizing/Description</th>
<th>Part No.</th>
<th>Qty</th>
<th>Cost</th>
<th>Purchase From</th>
</tr>
</thead>
<tbody>
<tr>
<td>Bottom Plate</td>
<td>Mild Steel A36 Hot Rolled Rectangle 1&quot; x 5&quot;, Cut to 12&quot;</td>
<td>N/A</td>
<td>1</td>
<td>29.30</td>
<td>Online Metals</td>
</tr>
<tr>
<td>H Beam</td>
<td>1 x 1 x 11 GA (.120 wall) AS13 Steel Structural Square Tube, 2' Length</td>
<td>T11111</td>
<td>1</td>
<td>4.32</td>
<td>Metals Depot</td>
</tr>
<tr>
<td>Sliding Beam</td>
<td>Mild Steel A36 Hot Rolled Rectangle 3&quot; x 5&quot; x 0.12&quot;, Cut to 12&quot;</td>
<td>N/A</td>
<td>1</td>
<td>21.06</td>
<td>Online Metals</td>
</tr>
<tr>
<td>Top Plate Rectangular Tubing</td>
<td>3 x 3 x 3/16 wall AS50 Steel Structural Square Tube, 1' Length</td>
<td>T133316</td>
<td>1</td>
<td>9.50</td>
<td>Metals Depot</td>
</tr>
<tr>
<td>Column Rod</td>
<td>N112SS, 25.2 Overall Length. 1/2-13 external thread both ends. 5&quot; of thread one end, 2&quot; opposite end</td>
<td>N/A</td>
<td>2</td>
<td>232.86</td>
<td>PBC Linear</td>
</tr>
<tr>
<td>Flange Linear Bearing</td>
<td>3/4 inch rod, Linear ball bearing</td>
<td>JP112G</td>
<td>4</td>
<td>51.08</td>
<td>PBC Linear</td>
</tr>
<tr>
<td>Futek 250 lb Load Cell</td>
<td>250 lb pancake style load cell</td>
<td>FSH00150</td>
<td>1</td>
<td>695.00</td>
<td>Futek</td>
</tr>
<tr>
<td>Futek 2000 lb Load Cell</td>
<td>2000 lb pancake style load cell</td>
<td>FSH00126</td>
<td>1</td>
<td>695.00</td>
<td>Futek</td>
</tr>
<tr>
<td>Nook Jack</td>
<td>1 ton ball screw jack</td>
<td>1-BSJ-UR</td>
<td>1</td>
<td>800.00</td>
<td>Nook Industries</td>
</tr>
<tr>
<td>Hand Wheel</td>
<td>Spoked Die Cast Zinc Dished Hand Wheel Revolving Handle, 4-1/2&quot; Wheel Dia</td>
<td>6035K71</td>
<td>1</td>
<td>15.96</td>
<td>McMaster-Carr</td>
</tr>
<tr>
<td>Quick Pin 2&quot;</td>
<td>18-8 SS Ring-Grip Quick Release Pin 1/4&quot; Diameter, 2.0&quot; Usable Length</td>
<td>98404A960</td>
<td>2</td>
<td>3.90</td>
<td>McMaster-Carr</td>
</tr>
<tr>
<td>Spacer Block for Load Cell</td>
<td>Aluminum 6061 - T6 Bare Extruded Round 5.5&quot;, Cut to 2&quot;</td>
<td>N/A</td>
<td>1</td>
<td>19.39</td>
<td>Online Metals</td>
</tr>
<tr>
<td>Load Cell Button</td>
<td>Aluminum 6061 - T6 Bare Extruded Round 3&quot;, Cut to 2&quot;</td>
<td>N/A</td>
<td>2</td>
<td>5.88</td>
<td>Online Metals</td>
</tr>
<tr>
<td>Top Sample Plate</td>
<td>52100 Steel</td>
<td>N/A</td>
<td>4</td>
<td></td>
<td>Sourced from Parker</td>
</tr>
<tr>
<td>Bottom Sample Plate</td>
<td>52100 Steel</td>
<td>N/A</td>
<td>4</td>
<td></td>
<td>Sourced from Parker</td>
</tr>
<tr>
<td>Dial Indicator Plates</td>
<td>Multipurpose Aluminum (alloy 6061), 3/8&quot; Thick, 12&quot; x 12&quot;</td>
<td>9246K23</td>
<td>2</td>
<td>57.02</td>
<td>McMaster-Carr</td>
</tr>
<tr>
<td>Sample Plate Guide Pins</td>
<td>18-8 Stainless Steel Shoulder Screw, 1/4&quot; Shoulder Diameter, 2&quot; Shoulder Length, 10-24 Thread</td>
<td>9029B4544</td>
<td>14</td>
<td>38.64</td>
<td>McMaster-Carr</td>
</tr>
<tr>
<td>Jack Crush Sleeve</td>
<td>4142 Alloy Steel Tubing 2-1/2&quot; OD, 1-1/2&quot; ID, 1/2&quot; Wall Thickness, 1&quot; Length</td>
<td>6920T111</td>
<td>1</td>
<td>57.51</td>
<td>McMaster-Carr</td>
</tr>
<tr>
<td>Load Cell Crush Sleeves</td>
<td>5/8 OD x .156 wall x .313 ID 1020 DOM Structural Round Steel Tube, 2&quot; Length</td>
<td>T258156</td>
<td>1</td>
<td>18.74</td>
<td>Metals Depot</td>
</tr>
<tr>
<td>Top Column Crush Sleeves</td>
<td>Low-Carbon Steel Tubing, 1.25&quot; OD, 0.75&quot; ID, .250&quot; Wall Thickness, 1&quot; Length</td>
<td>77617T691</td>
<td>1</td>
<td>19.45</td>
<td>McMaster-Carr</td>
</tr>
<tr>
<td>Bottom Sample Plate Track</td>
<td>Aluminum 6061 - T6 Bare Extruded Rectangle 0.75&quot; x 6&quot;, Cut to 2.5&quot;</td>
<td>N/A</td>
<td>2</td>
<td>9.36</td>
<td>Online Metals</td>
</tr>
<tr>
<td>Middle Sample Plate Track</td>
<td>Aluminum 6061 - T6 Bare Extruded Rectangle 1.25&quot; x 3.5&quot;, Cut to 2.8&quot;</td>
<td>N/A</td>
<td>2</td>
<td>10.18</td>
<td>Online Metals</td>
</tr>
<tr>
<td>Top Sample Plate Track</td>
<td>Aluminum 6061 - T6 Bare Extruded Rectangle 1&quot; x 6&quot;, Cut to 3.5&quot;</td>
<td>N/A</td>
<td>2</td>
<td>17.46</td>
<td>Online Metals</td>
</tr>
<tr>
<td>Magnetic Arm - 1/8&quot; Ball (Don't need)</td>
<td>01 Tool Steel Hardened Rod 7/64&quot; Diameter, 1&quot; Length</td>
<td>4340T34</td>
<td>1</td>
<td>14.01</td>
<td>McMaster-Carr</td>
</tr>
<tr>
<td>Magnetic Arm - 1/2&quot; Ball</td>
<td>W1 Tool Steel Rod .2031&quot; Diameter, Trade Size 13/64&quot;, 3&quot; Length</td>
<td>8890K20</td>
<td>1</td>
<td>2.63</td>
<td>McMaster-Carr</td>
</tr>
<tr>
<td>Magnetic Arm - 1/4&quot; Ball</td>
<td>W1 Tool Steel Rod .2031&quot; Diameter, Trade Size 13/64&quot;, 3&quot; Length</td>
<td>8890K20</td>
<td>1</td>
<td>2.63</td>
<td>McMaster-Carr</td>
</tr>
<tr>
<td>Magnetic Arm - 1/8&quot; Ball</td>
<td>W1 Tool Steel Rod .2031&quot; Diameter, Trade Size 13/64&quot;, 3&quot; Length</td>
<td>8890K20</td>
<td>1</td>
<td>2.63</td>
<td>McMaster-Carr</td>
</tr>
<tr>
<td>Flange Sleeve for Magnetic Arm</td>
<td>Multipurpose Aluminum (Alloy 6061) Tube 2-1/2&quot; OD, 1-1/2&quot; ID, .500&quot; Wall Thick, 1&quot; Length</td>
<td>9056K91</td>
<td>1</td>
<td>130.08</td>
<td>McMaster-Carr</td>
</tr>
<tr>
<td>Magnet</td>
<td>Machinable Neodymium Disc Magnet, .070&quot; Diameter, .394&quot; Thick, .6 Pull lbs</td>
<td>59024K22</td>
<td>3</td>
<td>5.31</td>
<td>McMaster-Carr</td>
</tr>
<tr>
<td>Nylock Nut #10-24</td>
<td>-</td>
<td>-</td>
<td>16</td>
<td>16.00</td>
<td>Ace Hardware</td>
</tr>
<tr>
<td>Nylock Nut 1/2-13</td>
<td>-</td>
<td>-</td>
<td>4</td>
<td>1.00</td>
<td>Home Depot</td>
</tr>
<tr>
<td>Nylock Nut 3/8-16</td>
<td>-</td>
<td>-</td>
<td>2</td>
<td>2.00</td>
<td>Ace Hardware</td>
</tr>
<tr>
<td>Part</td>
<td>Sizing/Description</td>
<td>Part No.</td>
<td>Qty</td>
<td>Cost</td>
<td>Purchase From</td>
</tr>
<tr>
<td>---------------------------</td>
<td>--------------------------------------------------------------------------------------------</td>
<td>----------</td>
<td>-----</td>
<td>--------</td>
<td>----------------</td>
</tr>
<tr>
<td>Nut 1/4-20</td>
<td>-</td>
<td>-</td>
<td>8</td>
<td>0.48</td>
<td>Home Depot</td>
</tr>
<tr>
<td>Nylock Nut #4-40</td>
<td>-</td>
<td>-</td>
<td>4</td>
<td>4.00</td>
<td>Ace Hardware</td>
</tr>
<tr>
<td>Hex Nut 4-40</td>
<td>-</td>
<td>-</td>
<td>2</td>
<td>2.00</td>
<td>Ace Hardware</td>
</tr>
<tr>
<td>Flat Head Socket Cap Screw #4-40 x 5/8</td>
<td>-</td>
<td>-</td>
<td>4</td>
<td>4.00</td>
<td>Ace Hardware</td>
</tr>
<tr>
<td>Hex Bolt 1/4-20 x 3.5</td>
<td>-</td>
<td>-</td>
<td>8</td>
<td>1.68</td>
<td>Home Depot</td>
</tr>
<tr>
<td>Hex Bolt 1/4-20 x 3</td>
<td>-</td>
<td>-</td>
<td>16</td>
<td>2.20</td>
<td>Home Depot</td>
</tr>
<tr>
<td>Hex Bolt #10-24 x .75</td>
<td>-</td>
<td>-</td>
<td>2</td>
<td>2.00</td>
<td>Ace Hardware</td>
</tr>
<tr>
<td>Hex Bolt 3/8-16 x 2</td>
<td>-</td>
<td>-</td>
<td>4</td>
<td>4.00</td>
<td>Ace Hardware</td>
</tr>
<tr>
<td>Nut 1/4-20</td>
<td>18-8 SS Key-Locking Thin Wall Threaded Insert 10-24 Int Thread, 5/16&quot;-18 Ext Thread, 5/16&quot; L</td>
<td>93340A405</td>
<td>8</td>
<td>23.44</td>
<td>McMaster-Carr</td>
</tr>
<tr>
<td>Captive Nut 4-40 Thread</td>
<td>18-8 Stainless Steel Broach Style Captive Nut, 4-40 Thread Size, .060&quot; Minimum Panel Thickness (Pack of 25)</td>
<td>94648A320</td>
<td>1</td>
<td>7.30</td>
<td>McMaster-Carr</td>
</tr>
<tr>
<td>Eyebolt</td>
<td>Galvanized Steel Eyebolt with Shoulder &amp; Nut, 1/4&quot;-20 Thread, 500# Work Load Limit, 4&quot; Shank, 2&quot; Thread Length</td>
<td>3018T34</td>
<td>1</td>
<td>5.11</td>
<td>McMaster-Carr</td>
</tr>
<tr>
<td>Contact Point for Dial Indicator</td>
<td>Electronic and Dial Indicator Accessory, 3/16&quot; x 1/4&quot; Regular Contact Point for Dial Indicator</td>
<td>20625A651</td>
<td>3</td>
<td>4.41</td>
<td>McMaster-Carr</td>
</tr>
<tr>
<td>Disc Magnet (Not Used)</td>
<td>Ultra-High-Pull Neodymium Disc Magnet, 1/8&quot; Diameter, 1/8&quot; Thick, .7 Pull lbs</td>
<td>5862K51</td>
<td>3</td>
<td>5.10</td>
<td>McMaster-Carr</td>
</tr>
<tr>
<td>Disc Magnet (Not Used)</td>
<td>Ultra-High-Pull Neodymium Disc Magnet, 3/8&quot; Diameter, 1/8&quot; Thick, 3.6 Pull lbs</td>
<td>5862K95</td>
<td>2</td>
<td>7.20</td>
<td>McMaster-Carr</td>
</tr>
<tr>
<td>Washers</td>
<td>1/2&quot; Ball Bearings</td>
<td>N/A</td>
<td>8</td>
<td>0.50</td>
<td>Home Depot</td>
</tr>
<tr>
<td>Ball Bearings</td>
<td>1/2&quot; Ball Bearings</td>
<td>N/A</td>
<td>4</td>
<td>2.04</td>
<td>Central Coast Bearings</td>
</tr>
<tr>
<td>Socket Set Screw</td>
<td>Nylon Tip Alloy Steel Socket Set Screw, 8-32 Thread, 5/8&quot; Length (Pack of 10)</td>
<td>94115A196</td>
<td>1</td>
<td>6.02</td>
<td>McMaster-Carr</td>
</tr>
<tr>
<td>Carbide Drill Bit</td>
<td>Super-Duty Solid Carbide Jobbers' Drill Bit, Size L, 3-1/2&quot; Overall Lg, 1.7&quot; Drill DP, 118 Degree Point</td>
<td>20195A43</td>
<td>1</td>
<td>24.23</td>
<td>McMaster-Carr</td>
</tr>
<tr>
<td>Springs (Replaced)</td>
<td>Music Wire Precision Compression Spring, Zinc-Plated, 7/8&quot; Length, .30&quot; OD, .022&quot; Wire (Pack of 5)</td>
<td>9434K53</td>
<td>1</td>
<td>4.05</td>
<td>McMaster-Carr</td>
</tr>
<tr>
<td>Compression Spring</td>
<td>Brass Compression Spring, 25/32&quot; L, .042&quot; Wire (Packs of 12)</td>
<td>9657K86</td>
<td>1</td>
<td>7.85</td>
<td>McMaster-Carr</td>
</tr>
<tr>
<td>Conical Springs</td>
<td>Type 302 Stainless Steel Conical Compression Spring, 1&quot; L, .60&quot; OD, .343&quot; Small OD, .026&quot; Wire Diameter (Packs of 3)</td>
<td>1692K31</td>
<td>2</td>
<td>27.42</td>
<td>McMaster-Carr</td>
</tr>
<tr>
<td>Springs</td>
<td>Type 302 Stainless Steel Conical Compression Spring, 1/2&quot; L, .60&quot; OD, .375&quot; Small OD, .032&quot; Wire Diameter (Packs of 3)</td>
<td>1692K19</td>
<td>2</td>
<td>29.40</td>
<td>McMaster-Carr</td>
</tr>
<tr>
<td>Futek software/Hardware</td>
<td>USB connectivity kit for 250 lb and 2000 lb load cells, manufacturer calibrated</td>
<td>N/A</td>
<td>1000.00</td>
<td>Futek</td>
<td></td>
</tr>
<tr>
<td>Chicago Dial Indicator</td>
<td>Recommended LP3600</td>
<td>N/A</td>
<td>1677.00</td>
<td>Chicago Dial Indicators</td>
<td></td>
</tr>
</tbody>
</table>

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
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
Digital Dial Indicators

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.

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.

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 +/- 0.0002 in (+/- 0.005 mm).
• Resolution to 0.0001 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.

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 hardnosed stool load 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.
Chicago Dial Indicators

LOGIC SERIES GROUP 2 CASE PROBE
TRAVEL UP TO .600

SCALE - FULL

DIM. "A"

1/4 DIA. HOLE

CABLE, REMOTE PROBE

2-3/64 MINIMUM POSITION

3/16 DIA.

CONTACT POINT

TRAVEL  "A" DIMENSION

| .600  | 2-13/32 |
| .500  | 2-1/16  |
| .125  | 1-15/16 |
| .002  | 1-7/6   |
FUTEK MODEL LCF450 (L2900)

LOW PROFILE UNIVERSAL PANCAKE LOAD CELL

+OUTPUT
(TENSION)

8 x [0.0128 [37.1] THRU EQUALLY SPACED ON 3.50 [88.9] B.C.D."

BENDIX RECEPTACLE
PT02A-10-6P
MATING CONNECTOR
PT06A-10-65-SSR
(NOT INCLUDED)

THREA D UNF-2B THRU

LOAD SURFACE
DO NOT CONTACT

1.37 [34.8]

1.25 [31.8]

DO NOT APPLY LOAD ON COVER

SUPPORT OUTER RING

REFER TO LCF455 FOR TENSION BASE VERSION

SPECIFICATIONS:

RATED OUTPUT: 2 mV/V non-

SAFE OVERLOAD: 150% of R.O.

ZERO BALANCE: ±0.25% of R.O.

EXCITATION VCC OR VAC: ±25 MAX

BRIDGE RESISTANCE: 75 Ω

NONLINEARITY: ±0.1% of R.O.

HYSERESIS: ±0.2% of R.O. / CONTACT FACTORY

NONREPEATABILITY: ±0.2% of R.O.

CREEP: ±0.12% of LOAD / 12 MONTHS OF R.O.

TEMP. SHIFT ZERO: ±0.001% of R.O./ºC (MIN. 3% of R.O.)/ºC

TEMP. SHIFT SPAN: ±0.05% of LOAD/ºC (MIN. 3% of LOAD/ºC)

COMPENSATED TEMP: -40 to 85ºC [-40 to 185ºF]

OPERATING TEMP: -65 to 100ºC [0 to 212ºF]

DEFL exion: ±0.0082 to ±0.025 [3.5 to 1.5]

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

ACCESSORIES AND RELATED INSTRUMENTS AVAILABLE

CALIBRATION [STANDARD]: 0.1% 5% COMPRESSION: 100kΩ SHUNT CAL. VALUE

CALIBRATION TEST EXCITATION: 10 VDC

OTHER AVAILABLE RECEPTACLES & MATING CONNECTORS

BENDIX RECEPTACLE (PC024E-10-6P)
BENDIX MATING CONNECTOR (PC06E-10-65-SSR)
BENDIX RECEPTACLE (PT02H-10-6P)
BENDIX MATING CONNECTOR (PT06A-10-65-SSR)

10 THOMAS
IRVINE, CA 92618 USA
1-800-23-FUTEK (38695)

INTERNET:
http://www.futek.com

FUTEK

This drawing is solely for transformation and reference and excludes the original address. It is not to be changed in whole or in part, by any order or individual without written permission from FUTEK.
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

Type:
IP - Inch dimension series
CP - ISO metric dimension series
JP - JIS metric dimension series
KMP - 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

Modification:
Symbol Specification
A - Adaptable Type
OP - Open Type
W - Wire

Retainer Material:
Symbol Specification
6 - Polymer cage

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.

The data and specifications in this publication have been carefully compiled and are believed to be accurate and correct. However, it is the responsibility of the user to determine and ensure the suitability of Pacific Bearing's products for a specific application. Pacific Bearing's only obligation will be to repair or replace without charge, any defective components if returned promptly. No liability is assumed beyond such replacement. *Consult www.pacific-bearing.com for the latest technical updates.
<table>
<thead>
<tr>
<th>Nominal Shaft Diameter (inch/(\text{mm}))</th>
<th>Polymer Case</th>
<th>Part Number</th>
<th>Weight (oz/(\text{g}))</th>
<th>Major Dimensions &amp; Tolerances</th>
</tr>
</thead>
<tbody>
<tr>
<td>1/4</td>
<td>IPKC4G</td>
<td>33</td>
<td>0.2500 0.6350</td>
<td>D     0.5000 12.000 0.0005 0.013 1.3750 34.925 0.5801 1.2600 14.684 31.750</td>
</tr>
<tr>
<td>3/8</td>
<td>IPKC6G</td>
<td>45</td>
<td>0.3750 9.525</td>
<td>D     0.6250 15.875 0.004 0.010 1.5938 40.481 0.6719 1.0000 17.066 38.100</td>
</tr>
<tr>
<td>1/2</td>
<td>IPKC8G</td>
<td>106</td>
<td>0.5000 12.700</td>
<td>D     0.8750 22.250 0.006 0.016 2.3750 50.235 0.825 1.3500 26.988 44.450</td>
</tr>
<tr>
<td>5/8</td>
<td>IPKC10G</td>
<td>200</td>
<td>0.6250 15.875</td>
<td>D     1.1250 29.575 0.006 0.016 2.8125 71.438 1.2813 2.0000 32.544 50.800</td>
</tr>
<tr>
<td>3/4</td>
<td>IPKC11G</td>
<td>5</td>
<td>0.7500 19.050</td>
<td>D     1.2500 31.750 0.009 0.019 3.0937 75.581 1.3986 2.1975 35.22 55.563</td>
</tr>
<tr>
<td>1/2</td>
<td>IPKC16G</td>
<td>6</td>
<td>1.0000 25.400</td>
<td>D     1.5625 39.688 0.012 0.024 4.2913 108.744 1.9800 2.9900 50.463 63.500</td>
</tr>
<tr>
<td>1/16</td>
<td>IPKC20G</td>
<td>93</td>
<td>1.2500 31.750</td>
<td>D     2.0000 50.800 0.009 0.022 5.0000 127.000 2.3125 3.2800 58.738 79.375</td>
</tr>
<tr>
<td>1/32</td>
<td>IPKC21G</td>
<td>1.460</td>
<td>1.5000 38.100</td>
<td>D     2.3750 60.325 0.006 0.015 5.6875 144.463 2.9538 3.5000 68.862 90.250</td>
</tr>
<tr>
<td>2/32</td>
<td>IPKC22G</td>
<td>2.620</td>
<td>2.0000 50.800</td>
<td>D     3.0000 76.200 0.010 0.025 7.7500 196.850 3.6250 4.0750 92.675 111.325</td>
</tr>
</tbody>
</table>
RC60 Steel Shafting

- RC60 case hardened steel
- Polished for optimum surface finish

RC60 Steel Shafting

<table>
<thead>
<tr>
<th>PART NUMBER</th>
<th>NOMINAL SIZE</th>
<th>DIAMETER TOLERANCE</th>
<th>MAX. LENGTH</th>
<th>MIN HARDNESS DEPTH</th>
<th>WEIGHT PER INCH</th>
</tr>
</thead>
<tbody>
<tr>
<td>NIP08-xx</td>
<td>1/8&quot;</td>
<td>0.124</td>
<td>252</td>
<td>N/A</td>
<td>0.004</td>
</tr>
<tr>
<td>NIP95-xx</td>
<td>3/16&quot;</td>
<td>0.186</td>
<td>252</td>
<td>N/A</td>
<td>0.008</td>
</tr>
<tr>
<td>NIP16-xx</td>
<td>1/4&quot;</td>
<td>0.249</td>
<td>252</td>
<td>N/A</td>
<td>0.014</td>
</tr>
<tr>
<td>NIP22-xx</td>
<td>3/8&quot;</td>
<td>0.374</td>
<td>252</td>
<td>0.040</td>
<td>0.031</td>
</tr>
</tbody>
</table>

PRE-DRILLED & TAPPED RC60 STEEL SHAFTING*

- Suitable for Simplicity® bearings and linear ball bearings
- Available cut-to-length or in full random lengths

<table>
<thead>
<tr>
<th>PART NUMBER</th>
<th>NOMINAL SIZE</th>
<th>DIAMETER TOLERANCE</th>
<th>HOLE SPACING</th>
<th>MAX LENGTH</th>
<th>WEIGHT</th>
</tr>
</thead>
<tbody>
<tr>
<td>NIP08-xx</td>
<td>1/8&quot;</td>
<td>0.4995</td>
<td>3.00</td>
<td>8-32</td>
<td>0.055</td>
</tr>
<tr>
<td>NIP15-xx</td>
<td>3/8&quot;</td>
<td>0.6245</td>
<td>8-32</td>
<td>116</td>
<td>0.086</td>
</tr>
<tr>
<td>NIP12-xx</td>
<td>1/4&quot;</td>
<td>0.7495</td>
<td>10-32</td>
<td>160</td>
<td>0.125</td>
</tr>
<tr>
<td>NIP16-xx</td>
<td>3/16&quot;</td>
<td>0.9995</td>
<td>1/4-20</td>
<td>225</td>
<td>0.222</td>
</tr>
<tr>
<td>NIP18-xx</td>
<td>1/8&quot;</td>
<td>1.166</td>
<td>5/16-18</td>
<td>192</td>
<td>0.348</td>
</tr>
<tr>
<td>NIP24-xx</td>
<td>3/8&quot;</td>
<td>1.399</td>
<td>3/8-12</td>
<td>500</td>
<td>0.500</td>
</tr>
<tr>
<td>NIP32-xx</td>
<td>1/4&quot;</td>
<td>1.699</td>
<td>1/2-2</td>
<td>800</td>
<td>0.890</td>
</tr>
</tbody>
</table>

NOTES:
- Spacing length in part number using inches.
- For random lengths, add “R” to the part number.
- Example: for 1/2" shafting total length 13" = NIP08-13
- Customer specifies "y" dimension.

RC60 Steel Shafting...

440 Stainless Steel Shafting...

Aluminum Pre-drilled Support Rails...

Aluminum End Support Blocks...

Aluminum Non-drilled Support Rails...

Steel Shaft Rail Assembly...

Feather Shafting...

Feather Rails...

Inch Self-lubricated Bearings...

Inch Pillow Blocks...

Inch Flange Mounted Bearings...

Inch Ball Bearings...

Inch Ball Bearing Pillow Blocks...

ISO Metric Shafting...
# Appendix G

## Senior Project Timeline

### Ghantt Chart Tasks

<table>
<thead>
<tr>
<th>Task Description</th>
<th>Start Date</th>
<th>End Date</th>
<th>Duration</th>
<th>Notes</th>
</tr>
</thead>
<tbody>
<tr>
<td>Hertz Load Analysis</td>
<td>Mon 10/11/09</td>
<td>Mon 10/19/09</td>
<td>8 days</td>
<td>Joe, Josh, Mike</td>
</tr>
<tr>
<td>Preliminary List of Sensors</td>
<td>Mon 10/12/09</td>
<td>Mon 10/19/09</td>
<td>8 days</td>
<td>Joe, Josh, Mike</td>
</tr>
<tr>
<td>Project Proposal</td>
<td>Tue 10/13/09</td>
<td>Thu 10/22/09</td>
<td>8 days</td>
<td>Joe, Josh, Mike</td>
</tr>
<tr>
<td>Peer Design Review</td>
<td>Tue 10/20/09</td>
<td>Thu 10/29/09</td>
<td>8 days</td>
<td>Joe</td>
</tr>
<tr>
<td>Teleconference</td>
<td>Tue 10/20/09</td>
<td>Tue 10/20/09</td>
<td>1 hr</td>
<td>Eric, Joe, Josh, Mike</td>
</tr>
<tr>
<td>Preliminary Designs</td>
<td>Tue 10/20/09</td>
<td>Thu 11/5/09</td>
<td>13 days</td>
<td>Joe, Josh, Mike</td>
</tr>
<tr>
<td>Stress and Deflection Analysis</td>
<td>Tue 10/20/09</td>
<td>Thu 11/5/09</td>
<td>13 days</td>
<td>Joe, Josh, Mike</td>
</tr>
<tr>
<td>Detailed list of sensors and equipment</td>
<td>Tue 10/20/09</td>
<td>Thu 11/5/09</td>
<td>13 days</td>
<td>Joe, Josh, Mike</td>
</tr>
<tr>
<td>Lead Cell Research</td>
<td>Tue 10/20/09</td>
<td>Tue 11/3/09</td>
<td>11 days</td>
<td>Joe</td>
</tr>
<tr>
<td>Deflection Measurement Research</td>
<td>Tue 10/20/09</td>
<td>Tue 11/3/09</td>
<td>11 days</td>
<td>Mike</td>
</tr>
<tr>
<td>Datalogging Equipment Research</td>
<td>Tue 10/20/09</td>
<td>Tue 11/3/09</td>
<td>11 days</td>
<td>Josh</td>
</tr>
<tr>
<td>Design Review Meeting in Irvine</td>
<td>Fri 11/20/09</td>
<td>Fri 11/20/09</td>
<td>1 day</td>
<td>Eric, Joe, Josh, Mike</td>
</tr>
<tr>
<td>Secondary Designs, BOM’s, Costs for</td>
<td>Mon 11/23/09</td>
<td>Fri 12/4/09</td>
<td>10 days</td>
<td>12 Joe, Josh, Mike</td>
</tr>
<tr>
<td>Design Report (End of Quarter 1)</td>
<td>Mon 11/30/09</td>
<td>Fri 12/4/09</td>
<td>5 days</td>
<td>Joe</td>
</tr>
<tr>
<td>Finals Week</td>
<td>Mon 12/7/09</td>
<td>Fri 12/11/09</td>
<td>5 days</td>
<td>Joe</td>
</tr>
<tr>
<td>Winter Break</td>
<td>Mon 12/14/09</td>
<td>Fri 12/18/09</td>
<td>15 days</td>
<td>Joe</td>
</tr>
<tr>
<td>Review Detailed Drawings and BOM</td>
<td>Mon 1/4/10</td>
<td>Thu 1/7/10</td>
<td>4 days</td>
<td>Joe, Josh, Mike</td>
</tr>
<tr>
<td>Verify Req’d Tooling Available</td>
<td>Tue 1/5/10</td>
<td>Wed 1/6/10</td>
<td>2 days</td>
<td>Joe, Josh, Mike</td>
</tr>
<tr>
<td>Final Design Review Meeting in Irvine</td>
<td>Fri 1/8/10</td>
<td>Fri 1/8/10</td>
<td>1 day</td>
<td>Eric, Joe, Josh, Mike</td>
</tr>
<tr>
<td>Final Design Report</td>
<td>Mon 1/11/10</td>
<td>Fri 1/22/10</td>
<td>10 days</td>
<td>Joe, Josh, Mike</td>
</tr>
<tr>
<td>Incorporate Changes From Sponsor, Dr</td>
<td>Mon 1/11/10</td>
<td>Tue 1/20/10</td>
<td>7 days</td>
<td>Joe, Josh, Mike</td>
</tr>
<tr>
<td>Finals Week</td>
<td>Mon 3/8/10</td>
<td>Fri 3/12/10</td>
<td>5 days</td>
<td>Joe</td>
</tr>
<tr>
<td>Spring Break</td>
<td>Mon 3/15/10</td>
<td>Fri 3/19/10</td>
<td>10 days</td>
<td>Joe</td>
</tr>
<tr>
<td>Review Detailed Drawings and BOM</td>
<td>Mon 3/29/10</td>
<td>Fri 4/2/10</td>
<td>5 days</td>
<td>Joe, Josh, Mike</td>
</tr>
<tr>
<td>Submit Parts and Materials Order</td>
<td>Mon 4/5/10</td>
<td>Fri 4/9/10</td>
<td>5 days</td>
<td>Joe, Josh, Mike</td>
</tr>
<tr>
<td>Final Test Plan</td>
<td>Mon 4/12/10</td>
<td>Fri 4/16/10</td>
<td>5 days</td>
<td>Joe</td>
</tr>
<tr>
<td>Machining</td>
<td>Thu 4/13/10</td>
<td>Sat 4/17/10</td>
<td>3 days</td>
<td>Joe, Josh, Mike</td>
</tr>
<tr>
<td>Assembly</td>
<td>Tue 5/11/10</td>
<td>Sat 5/22/10</td>
<td>10 days</td>
<td>Joe, Josh, Mike</td>
</tr>
</tbody>
</table>

**Notes:**
- *Done:* Tasks marked with a check are completed.
- *Notes:* Any additional notes or comments for each task are included in the "Notes" column.
Ghantt Chart for Fall Quarter 2009

- **Hertz Load Analysis**
  - Joe, Josh, Mike
- **Preliminary List of Sensors**
  - Joe, Josh, Mike
- **Project Proposal**
  - Joe, Josh, Mike
- **Peer Design Review**
  - Joe
- **Teleconference**
  - Eric, Joe, Josh, Mike
- **Preliminary Designs**
  - Joe, Josh, Mike
- **Stress and Deflection Analysis**
  - Joe, Josh, Mike
- **Detailed list of sensors, and equipment**
  - Joe, Josh, Mike
- **Load Cell Research**
  - Joe
- **Deflection Measurement Research**
  - Mike
- **Data Logging Equipment Research**
  - Josh
- **Design Review Meeting in Irvine**
  - Eric, Joe, Josh, Mike
- **Secondary Designs, BOM’s, Costs for Final Design Review**
  - Joe, Josh, Mike
- **Design Report: (End of Quarter 1)**
- **Final’s Week**
- **Winter Break**
## Appendix H

### Quality Function Deployment

<table>
<thead>
<tr>
<th>Customer Requirements</th>
<th>Engineering Requirements</th>
<th>Design</th>
</tr>
</thead>
<tbody>
<tr>
<td>Load cell to measure forces*</td>
<td>6</td>
<td>6</td>
</tr>
<tr>
<td>Measure distance of compression*</td>
<td>4</td>
<td>5</td>
</tr>
<tr>
<td>Log data to laptop</td>
<td>3</td>
<td>3</td>
</tr>
<tr>
<td>Test up to 0.25&quot; roller bearings</td>
<td>9</td>
<td>9</td>
</tr>
<tr>
<td>Test up to 0.5&quot; ball bearings</td>
<td>9</td>
<td>9</td>
</tr>
<tr>
<td>Usable in varying thermal environments</td>
<td>4</td>
<td>6</td>
</tr>
<tr>
<td>Safety to operator</td>
<td>4</td>
<td>6</td>
</tr>
<tr>
<td>Easy to use</td>
<td>3</td>
<td>6</td>
</tr>
<tr>
<td>Easy to use</td>
<td>3</td>
<td>6</td>
</tr>
<tr>
<td>Fit on table top / workstation*</td>
<td>4</td>
<td>6</td>
</tr>
<tr>
<td>Mobility*</td>
<td>3</td>
<td>6</td>
</tr>
<tr>
<td>Accurate apparatus stiffness*</td>
<td>5</td>
<td>6</td>
</tr>
<tr>
<td>Reliability</td>
<td>5</td>
<td>6</td>
</tr>
<tr>
<td>Validate machine accuracy with known sample data*</td>
<td>5</td>
<td>6</td>
</tr>
<tr>
<td>Manufacturing Concerns</td>
<td>5</td>
<td>6</td>
</tr>
<tr>
<td>Safety</td>
<td>5</td>
<td>6</td>
</tr>
<tr>
<td>Machining</td>
<td>6</td>
<td>6</td>
</tr>
<tr>
<td>Assembly</td>
<td>6</td>
<td>6</td>
</tr>
<tr>
<td>Concept #1</td>
<td>60</td>
<td>60</td>
</tr>
<tr>
<td>Concept #2</td>
<td>60</td>
<td>60</td>
</tr>
<tr>
<td>Concept #3</td>
<td>50</td>
<td>50</td>
</tr>
<tr>
<td>Concept #4</td>
<td>40</td>
<td>40</td>
</tr>
<tr>
<td>Total</td>
<td>150</td>
<td>150</td>
</tr>
</tbody>
</table>

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

---

Grinding Limit Test Machine
Parker Aerospace
POC: Eric Polouch
Customers:
1. Shop Techs
2. Engineers

---

Faulty life of 10,000 cycles
Lack of ability to provide load with an accuracy of 1 lb
Durability of apparatus through temperature 90°F
Lack of motor to operate sensors through temperature range of XX to XX

---

Composite machine with the ability to rest on tabletop
Incorporate in final design
Incorporate in final design
Incorporate in final design
Incorporate in final design
Incorporate in final design
References


