

BASEBALL BAT CONTOUR ANALYSIS

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ABSTRACT

This project was about designing a portable device that can measure the contour of a baseball bat precisely to understand the connection between on field performance and different curvature profiles. Right now, baseball organizations have limited information on the specific contour details of bats. We built a portable, non-damaging system that measures the contour of all acceptable MLB bat sizes.

Initial testing has been completed, and the results indicate that the data does not yet fall within the specified parameters outlined in the project requirements. The team is currently working to improve both data processing methods and the mechanical performance of the system. Vibrations generated by the servo motor while rotating the lead screw are negatively affecting the accuracy of data collection, so solutions to reduce vibration are actively being developed and tested.

PROBLEM DEFINITION

Recent innovations in baseball bat design have introduced increasingly complex and performance driven curvature profiles, redefining how athletes optimize swing dynamics and contact precision. To fully unlock the performance potential of these designs, rapid and highly accurate curvature modeling is essential for advanced analysis, iteration, and data-driven decision making. However, many existing measurement solutions are costly, time-intensive, or lack portability, limiting their practical use in dynamic team environments. This lack of data causes teams to be at a disadvantage compared to their opponents when it comes to deciding the best curvature for their players. The Houston Astros seek to pioneer a fast, portable, and precise method for capturing and recording the 2D contour of a baseball bat's length and cup enabling real-time insights and streamlined evaluation.

REQUIREMENTS, SPECIFICATIONS, DELIVERABLES

- Deliverables:
 - A functioning device to measure the contour of a bat and its cup
 - A CSV file of contour data

- Instruction documentation (Theory of Operations)
- Test data set containing example contour data
- Requirements:
 - The device must measure the contour of a baseball bat as a function of length
 - The device must measure the contour of the cup of a baseball bat, if it is present
 - The device must be able to be moved by one person
 - The device must be able to be transported between cities
 - The device must take quick measurements
- Specifications:
 - Measurements of the baseball bat must be completed in under 10 minutes
 - Measurements of the contour must be accurate within 0.005 inches
 - The length of the device must not exceed 4.5 feet
 - The width of the device must not exceed 2.5 feet
 - The device must measure a bat of length between 30"-36"

CONCEPTS

Each group member created a concept for the top-level system for the contour measuring device. Primary differences focused on the fixturing device to center the bat with respect to the laser sensor. Key considerations during the concept generation were if the fixture would be primarily attached to the knob or cup end of the bat, how to apply downward force to the bat to constrain it, and how to constrain it without impeding our measurement device. The measurement device being considered for each concept is a time-of-flight laser sensor, specifically the GP2Y0E02B sensor, which can determine the distance of objects a given distance away from the sensor. This sensor, along with the preliminary testing set up can be seen in Figure A1. This sensor would be moved via linear guide rail with a lead screw propelling it along the length of the bat, while the sensor is simultaneously collecting data. Combining the previously mentioned distance data collected from the sensor with its position along the bat, which is known from the lead screw

rotational speed, would provide sufficient data to create a contour of the bat.

Concept 1, seen in Figure B1, uses a conical feature in a block with slits cut out to allow for the laser sensor to have full visibility to the bat within the fixture. This fixture centers the bat by making three points of contact on the outer perimeter of the cup end of the bat. The conical geometry also allows for a variety of bat diameters to be centered within this fixture, due to the conical geometry tapering inwards the further down the fixture the bat rests. This fixture would be paired with a matching, smaller, conical fixture at the knob end, which would be attached to a linear guide on slide rails. This would allow a downward force to be applied to the knob, centering the bat between the two conical fixture pieces.

Concept 2, seen in Figure B2, utilizes a triangular shaped metal bracket, that has set screws to lock down the knob of the bat. This design also only constrains the knob of the bat, leaving the cup end free for measurement. The springs that connect to the bracket on this design apply an evenly distributed load to the base of the bat, ensuring that it aligns vertically for the measuring process.

Concept 3, seen in Figure B3, uses free rolling metal balls to establish 3 points of contact with the outside of the bat. The balls would sit in a pre-determined cylindrical cut out, allowing the bat to go into the cut out while contacting the balls. Bats of varying sizes, to an extent, could touch different portions of the balls curved surfaces, allowing for centering of different diameter bats.

Concept 4, seen in Figure B4, uses a “pressure plate” to determine when load from the bat is beginning to be applied, then uses “petal” like appendages to grab the barrel of the bat. This concept benefits from having the appendages center the bat, not requiring a force on the knob end to keep it stable. There is an increase in manufacturing complexity with this design, due to the increase in components required to make this design.

Each of these concepts were compared against each other, using Concept 3, the metal balls in a cylindrical cut out concept, as the baseline design to which all the others were compared to. A Pugh matrix was used during this decision process, as seen in Table B1.

The items under criteria were identified as the most relevant considerations for the project. Ease of manufacturing is an estimate of how difficult a design would be to manufacture given available resources. The cost criterion considers the estimated price of materials/services for each concept. Component complexity criterion compares the number of components required for each concept, fewer components would lead to lower complexity and therefore a better score for this criterion. Ease of testing setup defines the estimated time and difficulty setting the bat up in the fixture design when a bat needs to be measured. Robustness to different bat contours/sizes criteria estimates the ability of the fixture to accommodate a variety of bat sizes or the ability to make the fixture adjustable to different bat diameters.

From this table, the cup facing conical fixture design was identified as the version to take forward. An initial CAD concept

of this fixture with the baseball bat is shown in Figure C1. An updated/finalized version of this original concept is seen in Figure C2. Having limited individual components in the design, the ability to accommodate a variety of bat diameters with its tapered conical design, and access to CNC machines to create the conical geometry made this concept stand out from the rest.

MECHANICAL ANALYSIS

Tolerance Analysis/Issue

A primary tolerance issue encountered early in the design process was the straightness tolerance on the stanchions/alignment rods that held the knob end conical fixture. Significant straightness deviations of these stanchions could result in tilting of the bat, as illustrated in Figure D1. Our group originally was intending on using aluminum extrusions with a linear guide that would have a conical fixture attached to it. This guide would then slide down and rest on the knob of the baseball bat, using its own weight to center it with the bottom conical fixture on the barrel end of the bat. Testing was done on a piece of aluminum extrusion that was potentially going to be used as the rail system for the knob end conical fixture using a FARO Arm scanner. Results of this testing can be seen in the figure below.

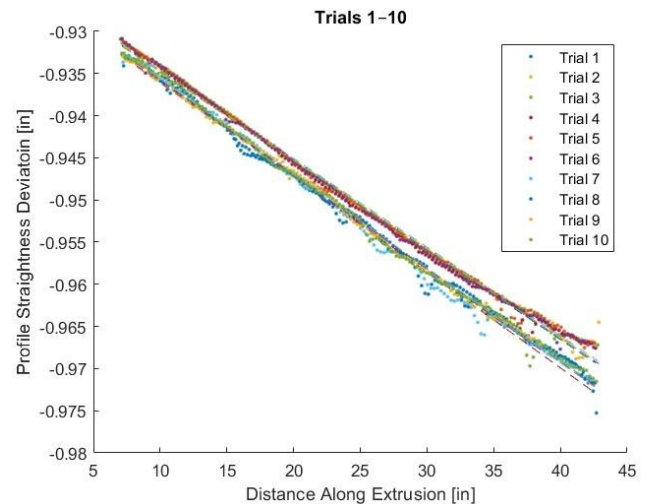


Figure 1: FARO Arm straightness test data of aluminum extrusion.

From this data, statistical analysis was done on the slope of the best fit lines for each trial set of data. Specifically, a t-distribution was used (less than 30 trials were conducted) with the following equation:

$$\mu = \bar{x} \pm t_{\alpha/2} \times \frac{s}{\sqrt{n}} \quad (1)$$

In this specific case our t-value would be 2.262, based on us having 10 trials and wanting a 95% confidence level, the standard deviation (S) for the slope values would be 3.4328e-05”, and n would be the same as the number of trials, 10 [1]. From the trials and the statistical analysis, the extrusion would have a straightness deviation of 0.00109” ± 2.46e-05” per inch

of extrusion length with 95% confidence. This would lead to approximately 0.040" straightness tolerance once you reach the end of the 37 inches needed for the design, as shown in D1. The team decided this value was significantly too large for our applications, so we looked for alternatives.

Thomson rods, specifically 0.75" diameter self-supporting rods with part number 8350T93 from McMaster Carr, were chosen for our design. These rods have a 0.002" per 1 foot of length straightness tolerance, which is much tighter controlled when compared to the aluminum extrusion previously being discussed. This would lead to a total straightness tolerance of 0.006" at the end of the rods, a large reduction from the 0.040" using the aluminum extrusion. However, the longest self-supporting Thomson rods the team could find were 25 inches long. Intermediate stanchion members that were created on the lathe, using 6061 Aluminum round stock. From the vendor, these bars have straightness tolerance of 0.0125" per 1 foot of length, comparable to that of the aluminum extrusions previously mentioned. When combining these two components, the total straightness tolerance would be approximately 0.0167" by the time you reach the end of the rods. This is still a greater than 50% reduction in straightness tolerance across the rail system.

Fatigue Analysis/Issue

The PLA vertical Servo motor mount used to stop the motion of the horizontal lead screw and allow for reversed motion undergoes a repeated displacement as the horizontal carriage is in motion.

An NX finite element simulation was run shown in Figure D2 to show the stress within the PLA mount when induced with a 0.1" displacement. The maximum stress value of 7.63 Mpa from the simulation was compared to a high cycle fatigue test of PLA shown in Figure D3. This comparison versus the S-N curve placed the PLA at more than 10^6 cycles before the part experiences failure.

For the referenced S-N curve, the fatigue life limit (S_e) of the failed PLA can be calculated using the equation below:

$$S_e = S_0 + \Delta S \times \frac{N_{FB}}{N_R} \quad (2)$$

Where S_0 represents the maximum stress level at which the PLA passed 10^6 cycles (without failure), ΔS indicates the amount of stress step, N_{FB} shows the number of the cycle that the PLA tolerates at failure stress, and N_R is the defined cyclic life [2].

Fastener Torque Calculation

The fasteners that were determined to be the most critical were the M6 fasteners that connected the base plates to the aluminum extrusion frame/base. These fasteners would have to support the largest load in the identified worst-case scenario set up. In this case the worst-case scenario would be if the entire fixture assembly were to be turned upside down, so the entire weight of the fixture would be loaded onto the 12 fasteners that

held the two base plates down. The total weight of the fixture components was determined to be 30.0441 lbf, as shown in Figure D4, when divided among the 12 fasteners that would be 2.504 lbf/fastener acting in tension.

To begin, the analysis/calculation a preload of 75% of the proof load of the fastener was used, considering the group wanted nonpermanent connections on these fasteners so it could be disassembled as necessary [2]. Due to the low loads expected on these fasteners, there was an expectation that the bolt preload would not be anywhere close to the preload required to reach 75% of the proof load. For this reason, the total load per fastener was considered in the comparison to ensure that we had enough fasteners and plate separation would not occur. This is opposed to determining the portion of the load that goes through the fastener directly and the member directly. For these purposes, the stiffness of the bolt and the stiffness of the members in the bolted connection were not required, due to the group using a more conservative approach by using the total tensile load at each bolted joint. This would result in a conservative comparison number, since the load we would be comparing too would be greater than the actual load in the fastener. The preload required to reach 75% of the proof strength for the M6 fasteners was 3292.08 lbf per bolt, which is much higher than the total tensile load per bolted connection, which was 2.504 lbf. This means that we are nowhere near plate separation occurring. For the torque calculation itself, the equation below was used:

$$T = K \times D \times F_i \quad (3)$$

Where T is the fastener torque, K is the nut factor, D is the nominal fastener diameter, and F_i is the preload. In this case a nut factor of 0.30 was used, which is common for black finished fasteners [3]. This results in a fastener torque requirement of 229.3 lbf-in, as shown in Figure D4.

Material Selection

To further guide the design direction of our fixturing mechanism, a structural analysis was performed to compare the displacement of the conical fixture (Concept 1) between two materials. Initially, the fixture was designed to be printed out of ABS plastic. After discussion of the precision of the 3D printing process as well as the team's desire to use the fixture interfaces to position other components, aluminum was also considered. A finite element model was created to analyze the displacement of the conical fixture. The load used in the analysis was approximately the weight of one baseball bat at about 30 oz or 1.875 lbf, applied in a ring at the maximum diameter of the barrel (2.61 in), see Figure B1 for the fixture concept set up. The fixture was fully constrained in the x, y, and z at the bolt holes, and it was also constrained in the z along the bottom surface representing the raised platform of the fixture (see Figure D5).

Figure D5 shows the FEM and simulation set up as well as results comparing the ABS plastic and Aluminum fixtures. In

comparing the displacement, the fixture made of aluminum 6061 had approximately 100x less displacement (magnitude) than the ABS fixture. This supports the team's decision to create both conical fixtures for the bat out of aluminum for structural reasons. The material choice and manufacturing process also provides more control over the tolerances of fixture features that mate with other components in the apparatus.

The solution used for this analysis was 101 Linear Statics. The model functions under the assumption that material behavior can be characterized as linear, meaning that as the load increases, the displacement can be scaled linearly. For the ABS, if the fixture were to be loaded with 10 times the weight of the bat, which may happen in the case of misalignment or improper use, the fixture could see a displacement on the order of magnitude of allowable tolerance on the bat contour measurement. With the aluminum, a 10x load of the bat weight will not have the same effect and therefore it is the material choice that the team will move forward with.

Computer-Based Analysis/Issue

The analysis chosen for this section was a structural finite element model of the Thomson rod mounting beam. The structure had the weight of two Thomson rods applied at their respective mounting locations on the beam. The weight of the rods was applied over the area of the nut and the bolt holes were represented using an RBE2 as well as fixed in all translational directions to represent fastening and clamping force. See Figure D6 in the annex for the FEM set up and results. The maximum displacement in the x direction is approximately one order of magnitude less than our accuracy specification. From this result, the model shows that the displacement in the x direction, corresponding to the distance between the sensor and the bat, makes a negligible contribution to the error in the system.

Fundamental Mechanical Analysis/Issue

The fundamental mechanical analysis completed for this project was determining the load required to hold down the plate. The statics problem utilized the geometry of the part as well as the weight of the lead screw as a point load. Summing the forces and moments, the reaction forces or the clamping force required to hold the part down as to not deform and support the aluminum beam as a load is being applied. See Figure D7 in the Annex for the calculations as well as the results which could, in future analysis, be used for a bolt torque calculation or used as a starting value for the deflection calculated in the previous section of this report.

MANUFACTURING

All components that were manufactured by the team were made of 6061 Aluminum. The primary reasoning for choosing 6061 Aluminum as opposed to other metals, such as steel, was weight considerations and ease of manufacturability. One of our

key requirements involved ease of transportability of our system, weight being a key consideration in this. There are also strict time deadlines, making softer material such as aluminum ideal from a machining perspective, allowing for faster feed rates and deeper passes on manual machines.

Many of our components were manufactured using a manual mill and lathe. This includes the base cone riser blocks, as shown in Figure E1, the smaller plate riser blocks, the laser carriages, and a few features on other components. Before this, the use of the horizontal bandsaw was necessary as seen in Figure E2, to cut our stock material down to appropriate sizes prior to milling. For the milling procedures, fly-cutters were used to face off our larger riser block components. End mills were used to bring them down to the proper height (height in this case referring to direction normal to faces that would be fastened to higher level assembly). Ensuring that every face was faced off ensured a smooth finish and a clean metal aesthetic. This can be seen in Figure E3. Another feature seen in Figure E3 is the hole/slot features for precision alignment. The design relies on being able to precisely orient the baseball bat in the center of fixture. This requires components to be located precisely with respect to one another. To accomplish this, dowel pins were press fit into components that had undersized reamed holes, to achieve an FN1 press fit. The complimentary parts that would mate with these dowel pins had slip fit holes and slots that were made on the mill. The slip fit holes used a 0.001" oversized reamer. The slot, which allows for manufacturing variability in one direction while maintaining a tight width tolerance for precision alignment, was made using an undersized end mill. The slot was gradually widened by .001" increments until the desired fit was achieved.

As well as using the mill, the manual lathe was also used for some of our circular components. The intermediate stanchion components, between the base conical fixture and the free-standing Thomson Rods, as well as the custom motor couplers were done on the lathe. The intermediate stanchions were brought down to size, then had either end tapped so the Thomson Rods could be fastened to them. The couplers were brought down to size, including the flange feature, and then the interior hole where the lead screw feeds through it was made. Once this was complete, the couplers were brought over to the mill, where they were fixed with v-blocks and the bolt hole pattern was created so it could be attached to the servo motor. A hole was milled and tapped on the side of the coupler as well, so a cone tipped set screw could be used to retain the lead screw. The completed couplers can be seen in Figure E4.

Aside from manual manufacturing operations, a ProtoTRAK mill was used for a few of our components with geometry that would be very difficult to create through manual means. Those parts include the upper alignment board that the 0.375" diameter Thomson Rods/lead screw feed through, both conical fixtures, and our base plates. Using the ProtoTRAK for the alignment board allowed for precision diameter holes for the press fit bushings/bearing without needing a large reamer. For the conical fixtures, the inner cone profile that centers the bat could not easily be done by hand, and the team did not want to

use a 4-axis Haas CNC machine we had axis too, primarily because many teams were attempting to use it at once, which could have resulted in a bottleneck in our production. The use of the ProtoTRAK with the bottom conical fixture can be seen in Figure E5. The bottom alignment plates were made on the ProtoTRAK primarily to save time, so the dozens of holes did not need to be cut out by hand.

The costs for manufacturing labor and hardware/materials can be found in the table below:

Table 1: Manufacturing cost estimate including materials and labor.

	Cost	Hours	Rate (Dollars/Hr)
Hardware/Material	\$ 1,014.00	N/A	N/A
Samuel Zeltman	\$ 3,200.00	32	\$ 100.00
Jack Todd	\$ 3,350.00	33.5	\$ 100.00
Melanie Landis	\$ 3,000.00	30	\$ 100.00
Natalie Shank	\$ 3,550.00	35.5	\$ 100.00
Total	\$14,114.00	131	N/A

Table 1 contains hours exclusive to manufacturing time. For a more detailed breakdown of the hardware/material costs see Table G1. A B.O.M. for the individual parts of our completed assembly can be seen in Figure F1, F2, F3, and F4 installed in their respective assemblies with component part numbers attached. A breakdown of development time/hours can be found in Table 2 below:

Table 2: Development/design cost and time spent.

	Cost	Hours	Rate (Dollars/Hr)
Samuel Zeltman	\$ 5,000.00	50	\$ 100.00
Jack Todd	\$ 6,350.00	63.5	\$ 100.00
Melanie Landis	\$ 6,500.00	65	\$ 100.00
Natalie Shank	\$ 5,150.00	51.5	\$ 100.00
Total	\$23,000.00	230	N/A

This is a tracker of all time spent on the project excluding manufacturing time. This includes concept generation, CAD modeling, electronics design, ordering/sourcing of materials/components, and drafting.

If we were to scale this system to a theoretical 1000 system, faster manufacturing techniques would be needed. This could include using a CNC machine for precision dowel pin slot/hole manufacturing, the complex cone geometry on the conical fixtures and more. Material savings would also have to be improved; alternative riser block designs could be utilized that save on material and weight.

TEST PLAN AND RESULTS

Specifications	Pass/Fail
Measurements of the baseball bat must be completed in under 10 minutes	Pass
Measurements of the contour must be accurate within 0.005 inches	Fail
The length of the device must not exceed 4.5 feet	Pass
The width of the device must not exceed 2.5 feet	Pass
The device must measure a bat of length between 30"-36"	Pass

To verify system measurement time, the longest of the four tested bats was placed in the fixture. Measuring the full length from barrel base to knob took approximately 7 minutes and 15 seconds. A maximum barrel diameter of 2.61 inches requires an additional 1.5 minutes for horizontal measurement, resulting in a total lead screw movement time of about 8 minutes and 45 seconds, which meets the measurement time specification.

To evaluate contour accuracy, a ground cylinder (0.25-inch diameter) was measured. Due to vibrations from lead screw oscillations, a clear contour could not be obtained. These oscillations are shown in Figure 2. The data was smoothed using MATLAB's *movmean* function to approximate a bat-like contour; however, this reduced measurement resolution as it included the points from noise and vibration.

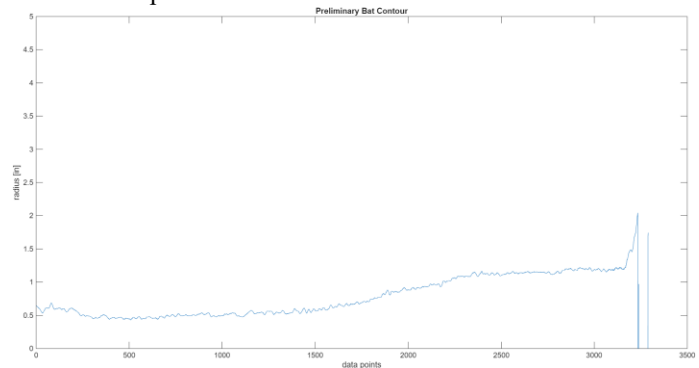


Figure 2: Preliminary Baseball Bat Scan with Time of Flight sensor; this data shows noise caused by oscillations in the lead screw through rotations in the lead mechanism.

The dimensions of the device correspond to the portability requirement for the project. After assembly, the device was verified to be in the design envelope (also used in the CAD) defined by the design requirements.

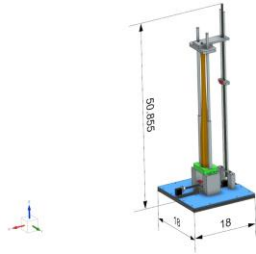


Figure 3: Baseball Contour Measurement Device design envelope

The design of the fixture has an adjustable conical fixture which allows for lifting and lowering of the fixture which accommodates the 30"-36" specification, answering the requirement for accommodating bats of different lengths.

Due to the vibrations caused by the lead screw mechanism, the device does not meet the most important specification of accuracy. Currently adjustments are being made to fixturing of the motors that power the lead screw mechanism to reduce oscillations and gyration of the lead screw. Furthermore, testing has shown that the orientation of the laser in the device's laser mount currently affects the resolution of the measurement. Figure A2 in the annex shows a calibration curve used to convert the bit output of the laser to a distance. In this calibration, the laser sender and receiver were placed perpendicular to the object being measured. The laser specification sheet details a recommended orientation for the laser in relation to direction of motion, the mount on the original device has been adjusted accordingly and will now reflect the new orientation in any future testing.

INTELLECTUAL PROPERTY

The design and process created during this project are likely not eligible for patent, as the methods being used are not necessarily novel or non-obvious. The use of lead screws to control horizontal and vertical motion, a carriage-mounted laser sensor, and an Arduino for control and data output all fall within well-established areas of patented technology. Existing patents such as US9658335B2 [4] and US20170184703 [5] already utilize the mounting of laser scanners to moving stages to capture surface profiles, which makes the core concept of this project not novel. Companies such as Hexagon AB, FARO Technologies, and Keyence have done extensive work in the area of laser scanning, holding many of the patents for key ideas related to this project. Due to the high volume of work and research done in the area, it is unlikely that this design would qualify for a patent.

SOCIETAL AND ENVIRONMENTAL IMPLICATIONS

This project has several impacts on public health, safety, and welfare by improving quality control in sports equipment manufacturing. More accurate and consistent measurements help ensure that equipment performs as intended and meets established standards which player safety and fairness in competition. Because the device is portable, it increases

accessibility by allowing testing to be performed directly at manufacturing sites, sporting events, or laboratories rather than requiring specialized off-site facilities. Additionally, the project has educational and research value by providing a platform for learning in mechanisms, manufacturing inspection, and automation.

The ethical and social concerns associated within this project pertain to fairness in sports, as improved testing can unveil previously illegal equipment leading to stricter manufacturing standards and potentially higher product costs. This may reduce accessibility for some athletes and organizations with limited budgets. User training is another consideration, as the system must be easy to operate and provide understandable results to avoid incorrect data interpretation. Safety risks also exist from direct laser exposure and open mechanical components. Proper safety labeling, shielding, and enclosure of moving mechanisms would be necessary to reduce risks to users and to the equipment.

From an environmental perspective, the project offers benefits through relatively low energy consumption and reduced material waste. More accurate measurements allow defects to be identified earlier in production, preventing defective products from progressing further in manufacturing. The portable design can also reduce the need to transport equipment. Concerns remain regarding electronic waste from sensors, motors, and control components as they reach the end of their life spans. The life span of metal and plastic structural components must also be considered.

RECOMMENDATIONS FOR FUTURE WORK

Future work for this project should focus on improving the accuracy and reliability of the measurement system. The most critical area for improvement is reducing the measurement noise cause by vibration in the lead screw mechanism and limitations of the current sensing approach. This could be addressed through both mechanical and controls improvements, such as implementing a more stable motion system (e.g, belt-driven stage or precision ball screw). Linear encoders can be implemented as part of the servo motors to provide precise position feedback. Updating the sensor system could also assist in providing the necessary measurement resolution that aligns with the ± 0.005 -inch accuracy specification.

To support these improvements, a comprehensive validation and calibration protocol should be developed including repeatability testing, and a full system uncertainty quantification.

Finally, adjusting the data collection process would help to increase the practical value of the device. If automated data processing and improved filtering techniques could be integrated into the system, this would not require data manipulation by discretion of the user.

ACKNOWLEDGMENTS

We would like to thank Professor Muir, Adam Brodie, and JJ Ruby for their time, knowledge, and efforts dedicated to the success of this project. We would also like to thank Samantha Kriegsman, Jim Alkins, Bill Mildenberger, Chris Pratt, and Kyle Christensen for their invaluable assistance throughout this project.

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- [4] Siercks, K., and Metzler, B., 2017, "Measurement system with a measuring device and a scanning module."
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ANNEX A
TESTING PROCEDURE



Figure A1: Preliminary testing of the time-of-flight sensor.

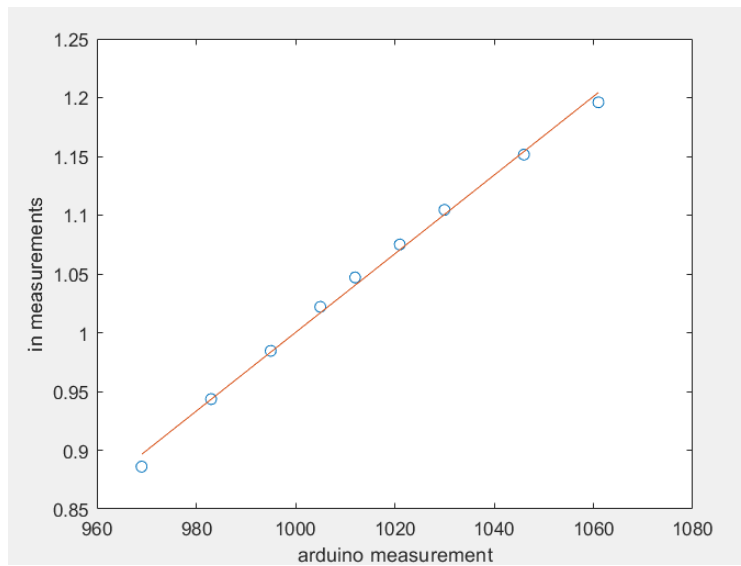


Figure A2: Initial Laser Scanning Test Data for Calibration Curve

ANNEX B
DESIGN/FIXTURE CONCEPTS

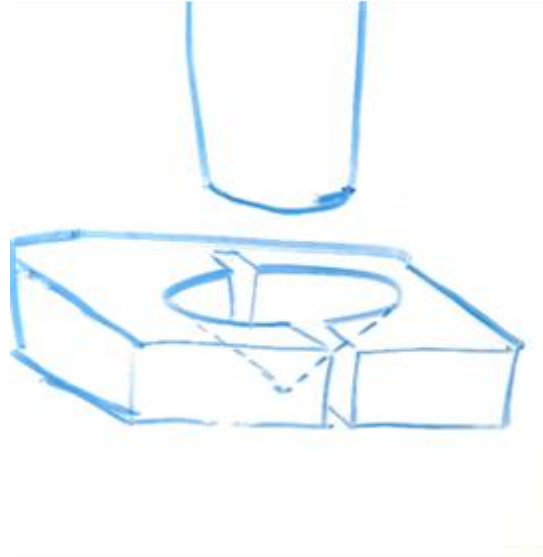


Figure B1: Cup facing conical fixture.

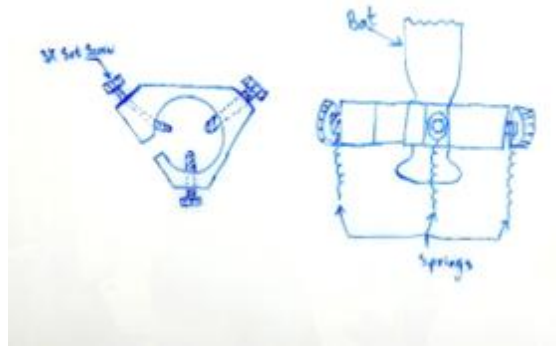


Figure B2: Knob facing set screw fixture with spring load.

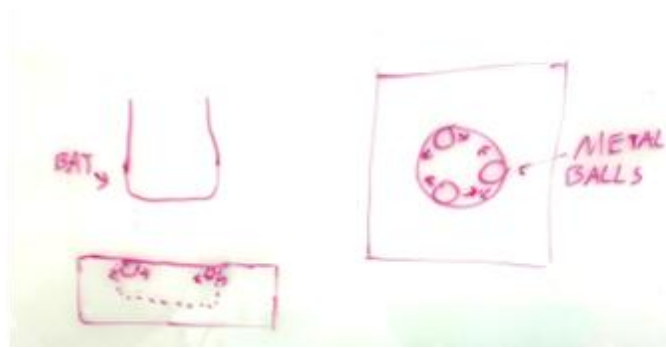


Figure B3: Cup facing cylindrical fixture.

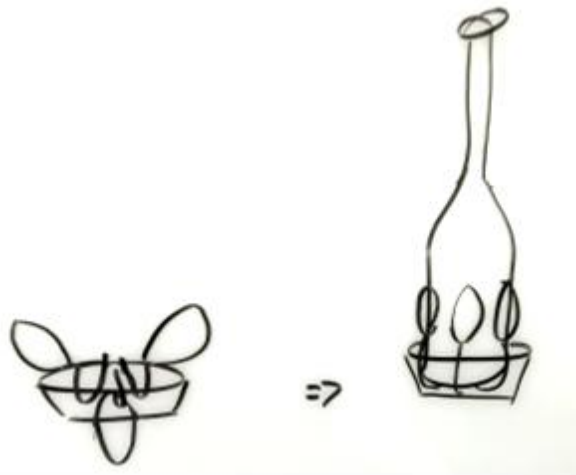


Figure B4: Cup facing lotus fixture.

Table B1: Concept selection matrix.

Criteria	Baseline (Cup Facing Cylindrical Fixture)	Cup Facing Lotus Fixture	Knob Facing Spring Loaded Set Screw Fixture	Cup Facing Conical Fixture
Ease of Manufacturing	0	-1	-1	0
Cost	0	-1	-1	1
Component Complexity	0	-1	-1	1
Ease of Testing Setup	0	1	-1	1
Robustness to Different Bat Contours/Sizes	0	1	1	1
Total	0	-1	-3	4

ANNEX C

CAD

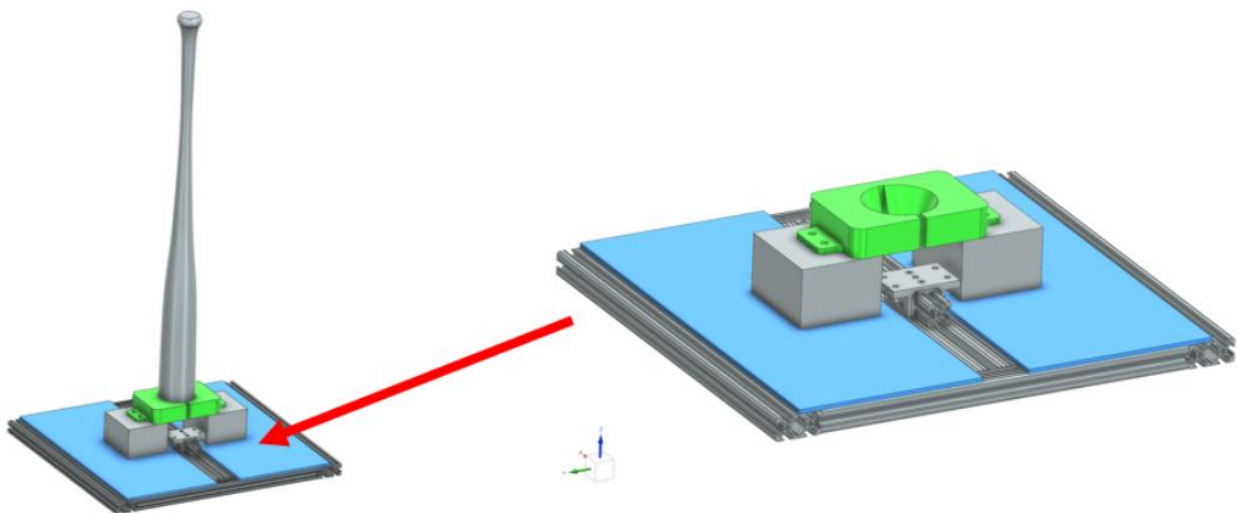


Figure C1: CAD for initial conical base fixture concept.



Figure C2: Final CAD Assembly

ANNEX D

MECHANICAL ANALYSIS

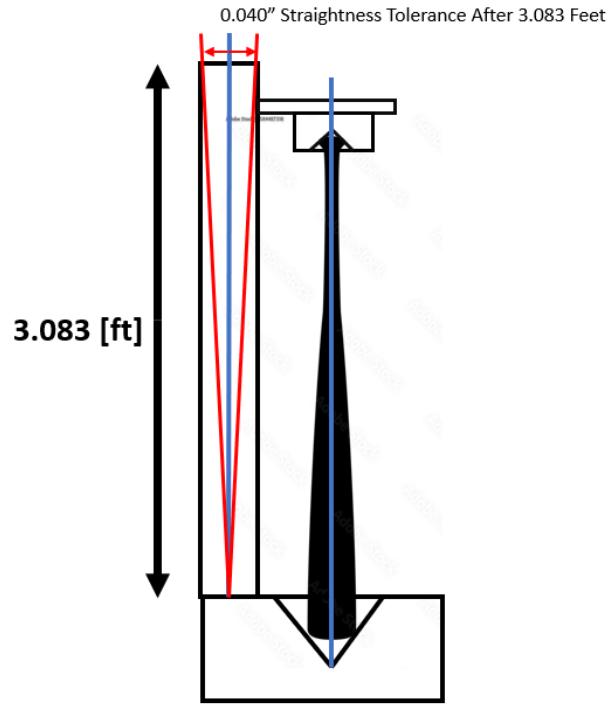


Figure D1: Diagram of straightness tolerance propagation as you move down aluminum extrusion long enough for baseball bat centering fixture.

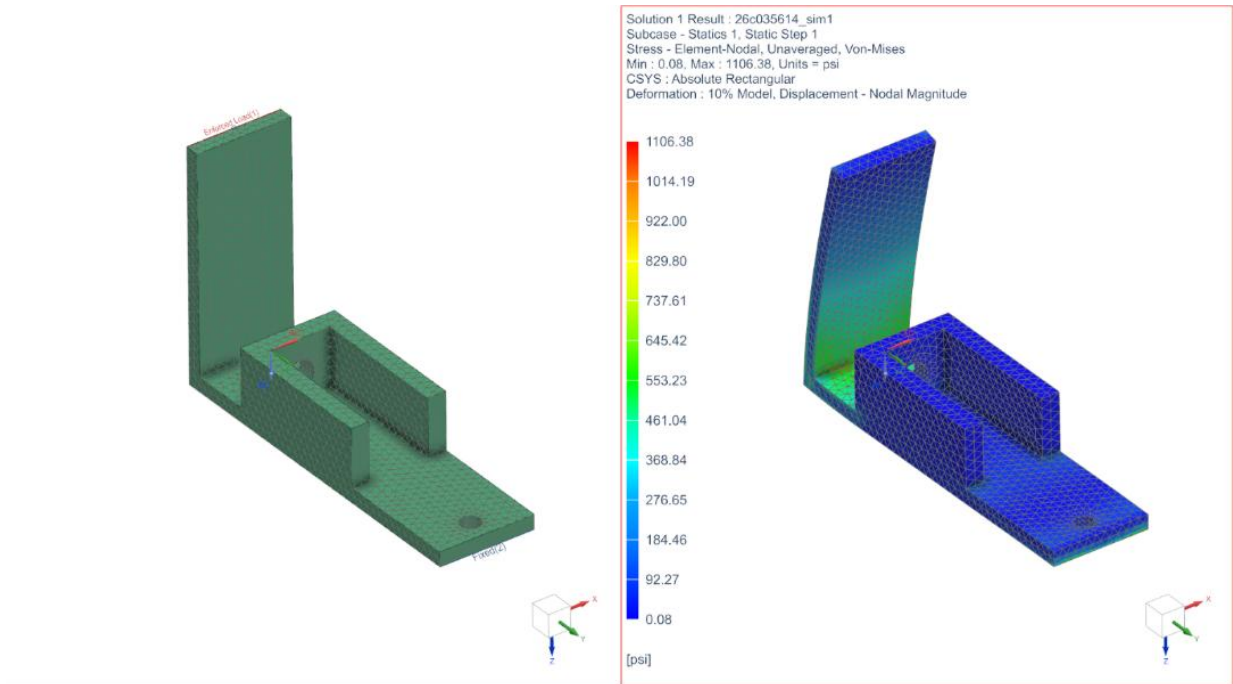


Figure D2: Averaged Von Mises Stress of PLA Servo Mount from Lead Screw Induced Displacement

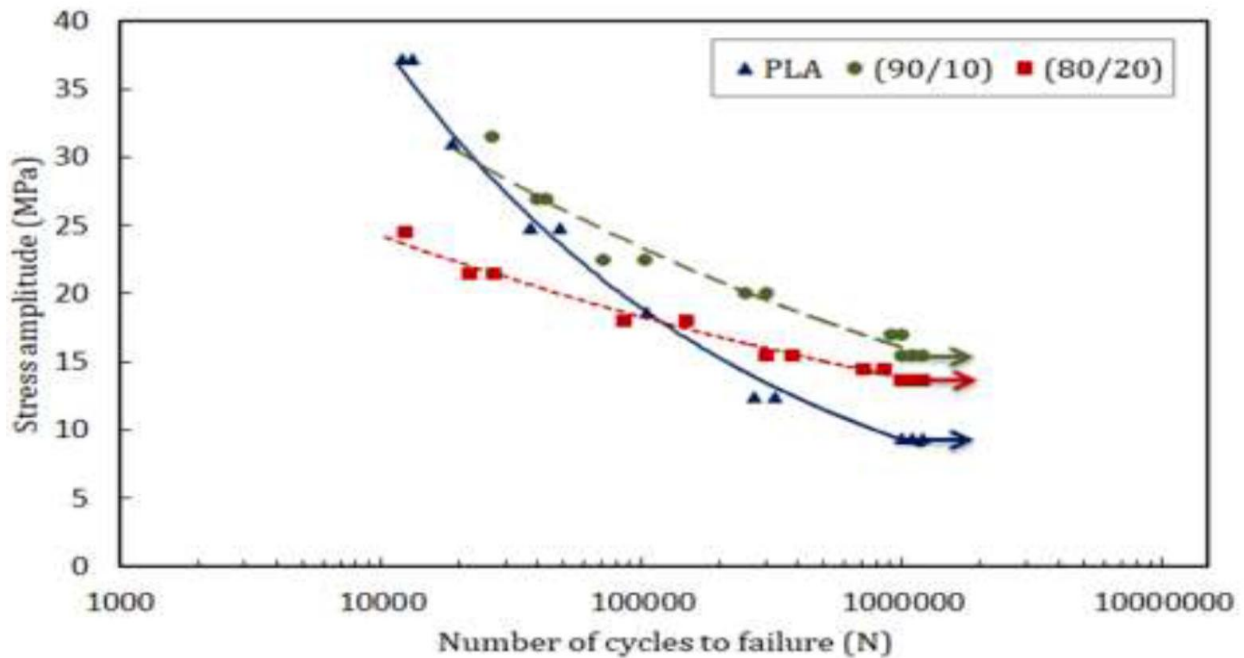


Figure D3: S-N curve for fatigue lifecycle of PLA

Baseball
ME 205
Gate D – Fastener Torque Calculation
4/25/2026

Material Assignments:

- NX Library Steel
- NX Library Al 6061

Goal: Determine torque for M6 fasteners that connect base plates to extrusion base.

Setup: Will use worst case loading on fasteners to determine required torque and preload. This will be if the entire fixture system were to be turned upside down and entire weight of system was acting in tension against the fasteners being considered.

Expected Tensile Load per Fastener = Worst Case Loading/# of Fasteners = Weight of Fixture/# of Fasteners = 30.0441 lbf / 12 fasteners = 2.504 lbf/fastener

Torque Equation: $T = K \times D \times F_i$

F_i = Preload = $0.75 \times F_p$ (nonpermanent connection) = 3292.08 lbf

F_p = Proof Force = $A_t \times S_p$ = 4389.45 lbf

A_t = Tensile Stress Area = $20.1 \text{ mm}^2 = 0.0312 \text{ in}^2$

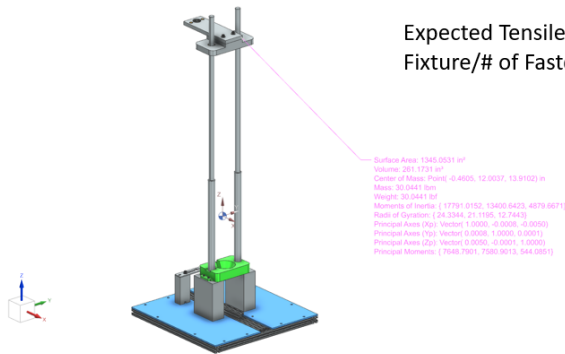
S_p = Proof strength = 970 MPa (Class 12.9) = 140687 psi

K = Nut Factor = 0.30 (for black finish)

D = Nominal Diameter of Fastener = 6 mm = 0.2322 inches

Since $F_i >$ Expected Tensile Load per Fastener, do not expect any plate separation/fatigue issues.

T = Fastener Torque Required = 229.3 lbf-in



Weight of fixture in NX

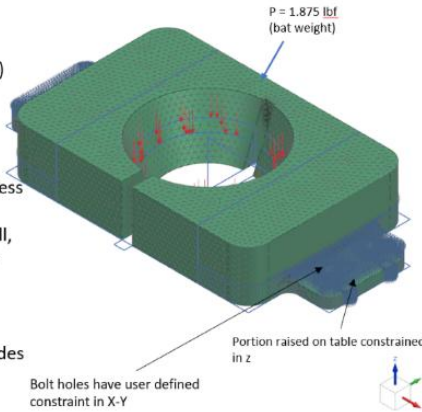
Figure D4: Fastener torque calculation.

Project 2 Gate B Analysis
Baseball Team

Analysis Goal: Compare displacement of bat fixture between aluminum and ABS 3D print

FEM Set-Up:
Solution 101 Linear Statics
Material 1: ABS Plastic (NX)
Material 2: Aluminum 6061 (NX)
TET10 (0.125" element size)

Results:
Aluminum results in 100 times less displacement. At higher loads, above the weight of the baseball, such as the weight of an arm or improper device loading, deformation in 3D printed part may affect contour reading. Machining aluminum also provides more accuracy for locating features



Results – Displacement Magnitude

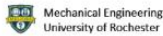
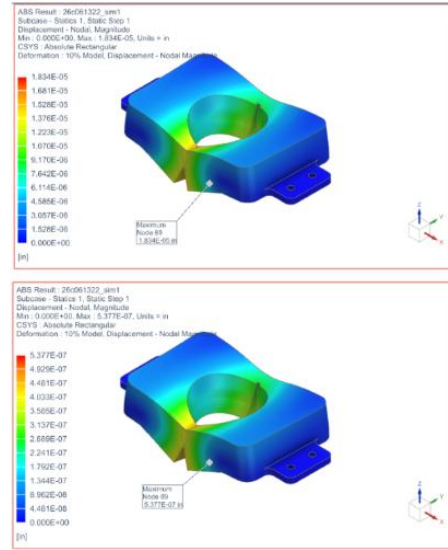


Figure D5: Analysis of the base conical bat fixture discussed in concept 1. Results show that Aluminum 6061 has approximately 100 times less displacement than ABS.

Project 2 Computer Based Analysis
Structural FEA of Thomson Rod Mounting Beam
Baseball Team

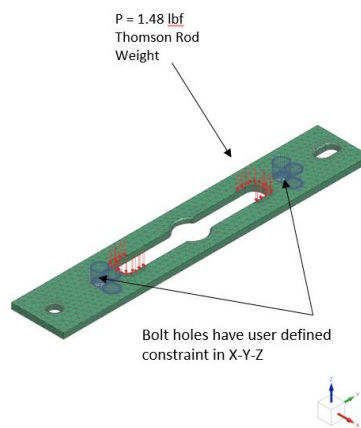
Analysis Goal: Determine the displacement in the x-axis due to weight of Thomson rods on the Thomson rod mounting beam.

FEM Set-Up:
Solution 101 Linear Statics
Material: Aluminum 6061 (NX)

TET10 (0.125" element size)
Weight of Thomson rods (1.48 lbf) are applied in circular area corresponding to 3/8" nut area. Bolt holes are constraining all 6 degrees of freedom of the object

Results:
The maximum x displacement (which affects the distance between sensor and object being sensed) is 6.07×10^{-6} in, which is much one order of magnitude less than resolution desired by laser measurement

FEM Set Up



Results – Displacement X

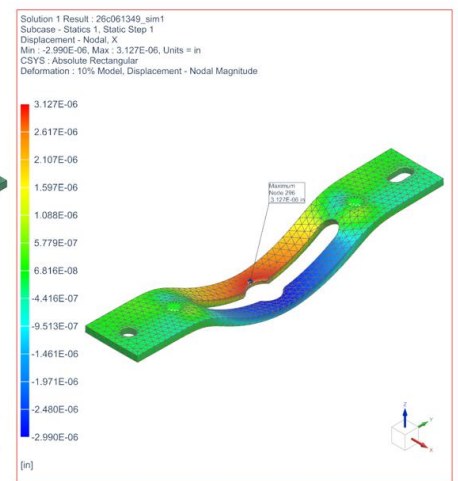


Figure D6: Computer Based Analysis.

Project 2 Statics Calculation
 Hand Calculation of Load on Thomson
 Rod Mounting Beam
 Baseball Team

Analysis Goal: Determine the load required to hold down the Thomson rod
 mounting beam

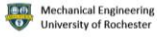
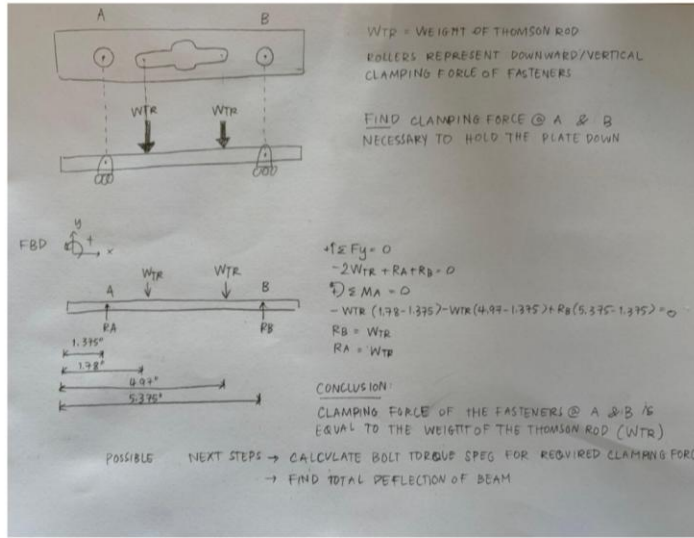


Figure D7: Statics Calculation

ANNEX E
MANUFACTURING



Figure E1: Riser cone being faced off on manual mill.



Figure E2: Stock piece being cut by horizontal bandsaw.

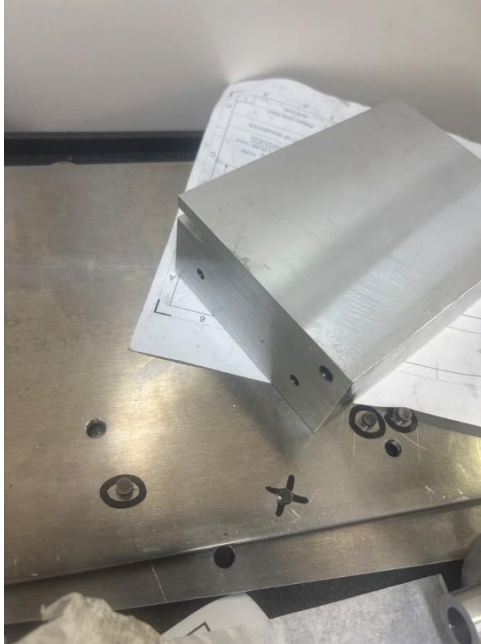


Figure E3: Completed cone riser block, with all features completed as well as sides faced off.



Figure E4: Custom couplers that transfer rotational motion from a servo motor to a lead screw.



Figure E5: Bottom conical fixture being made on ProtoTRAK mill.

ANNEX F

ASSEMBLY DRAWINGS

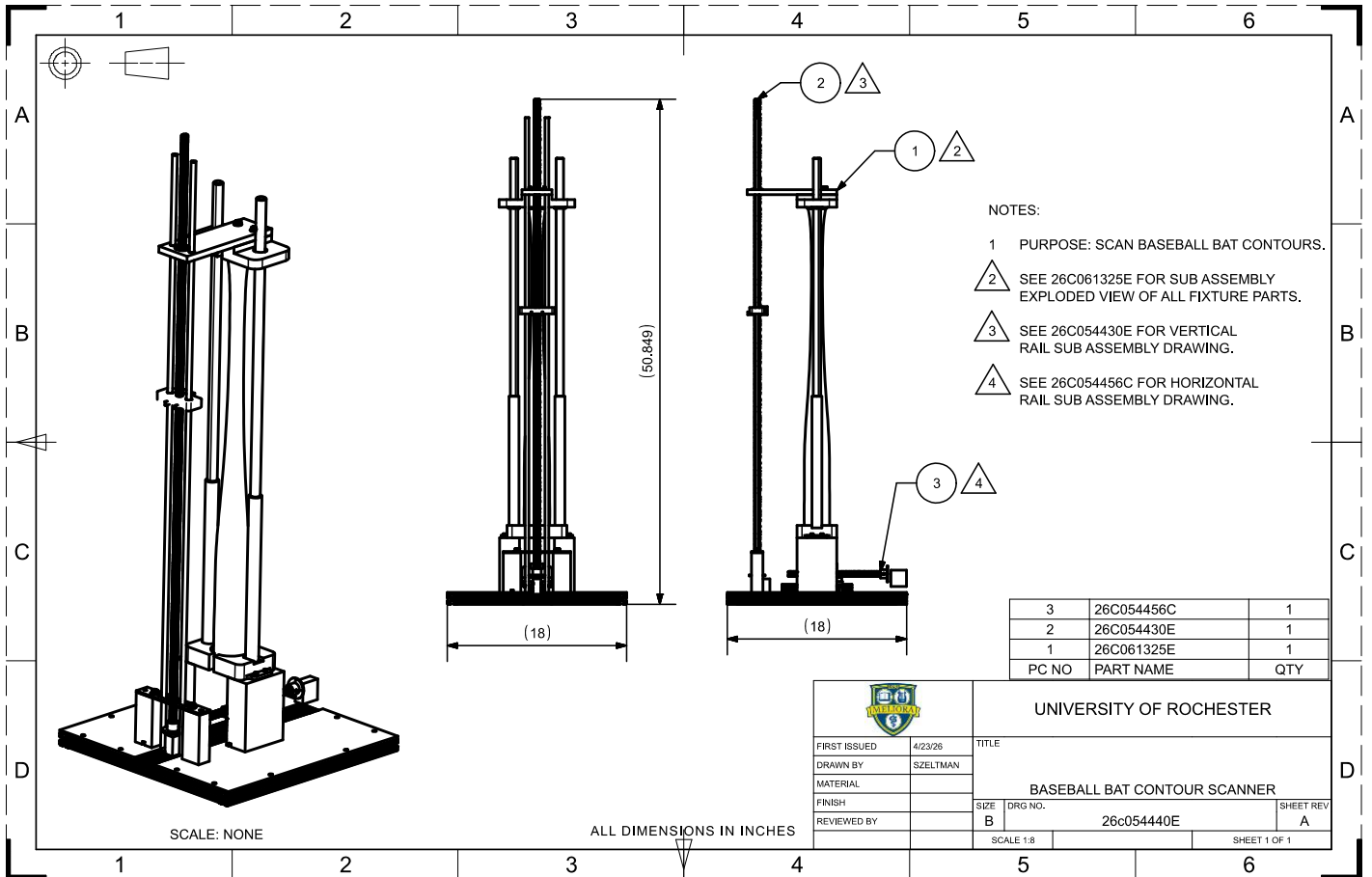


Figure F1: Top level assembly drawing containing 3 sub-assemblies.

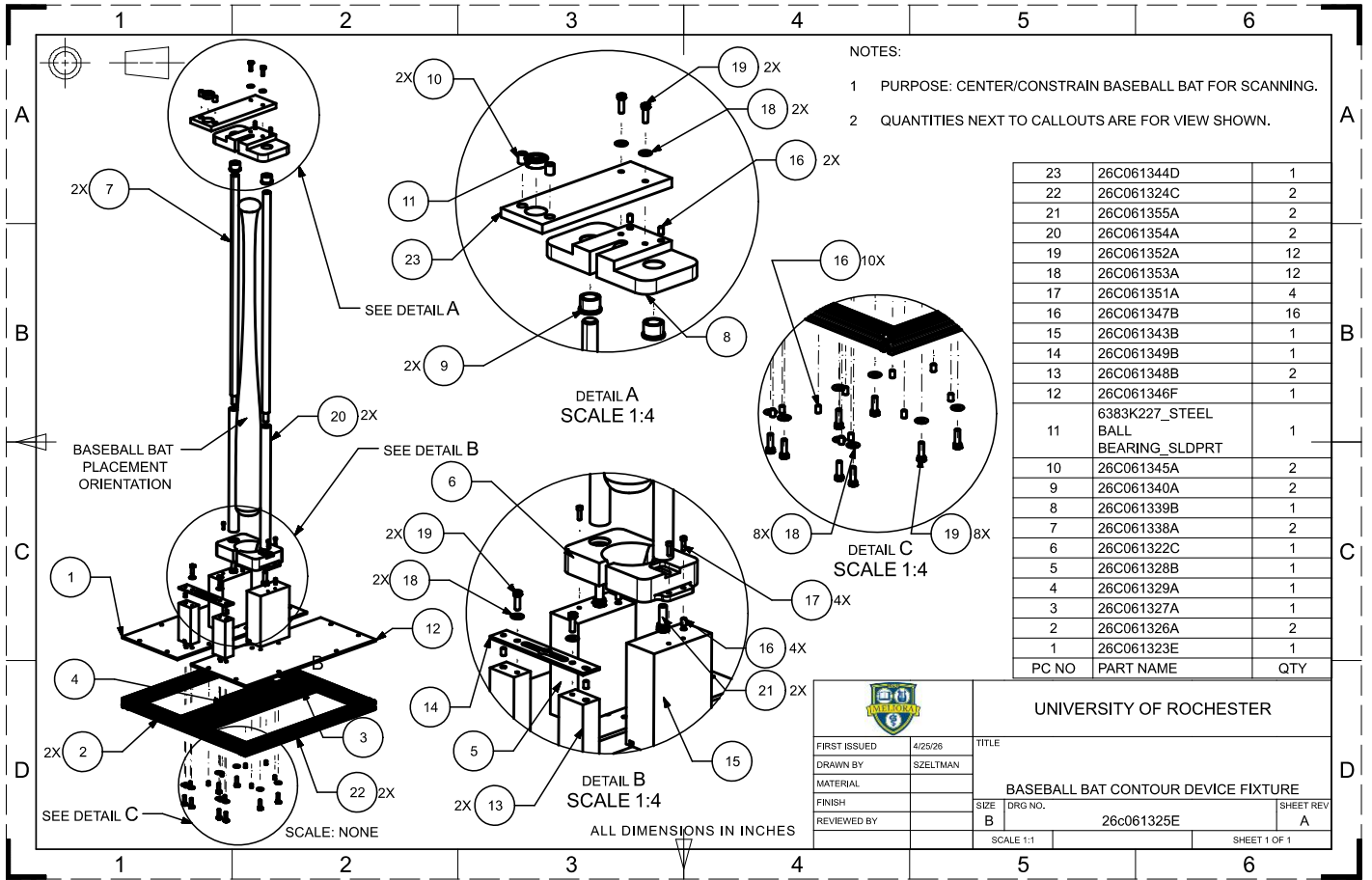


Figure F2: Fixture sub-assembly drawing with B.O.M.

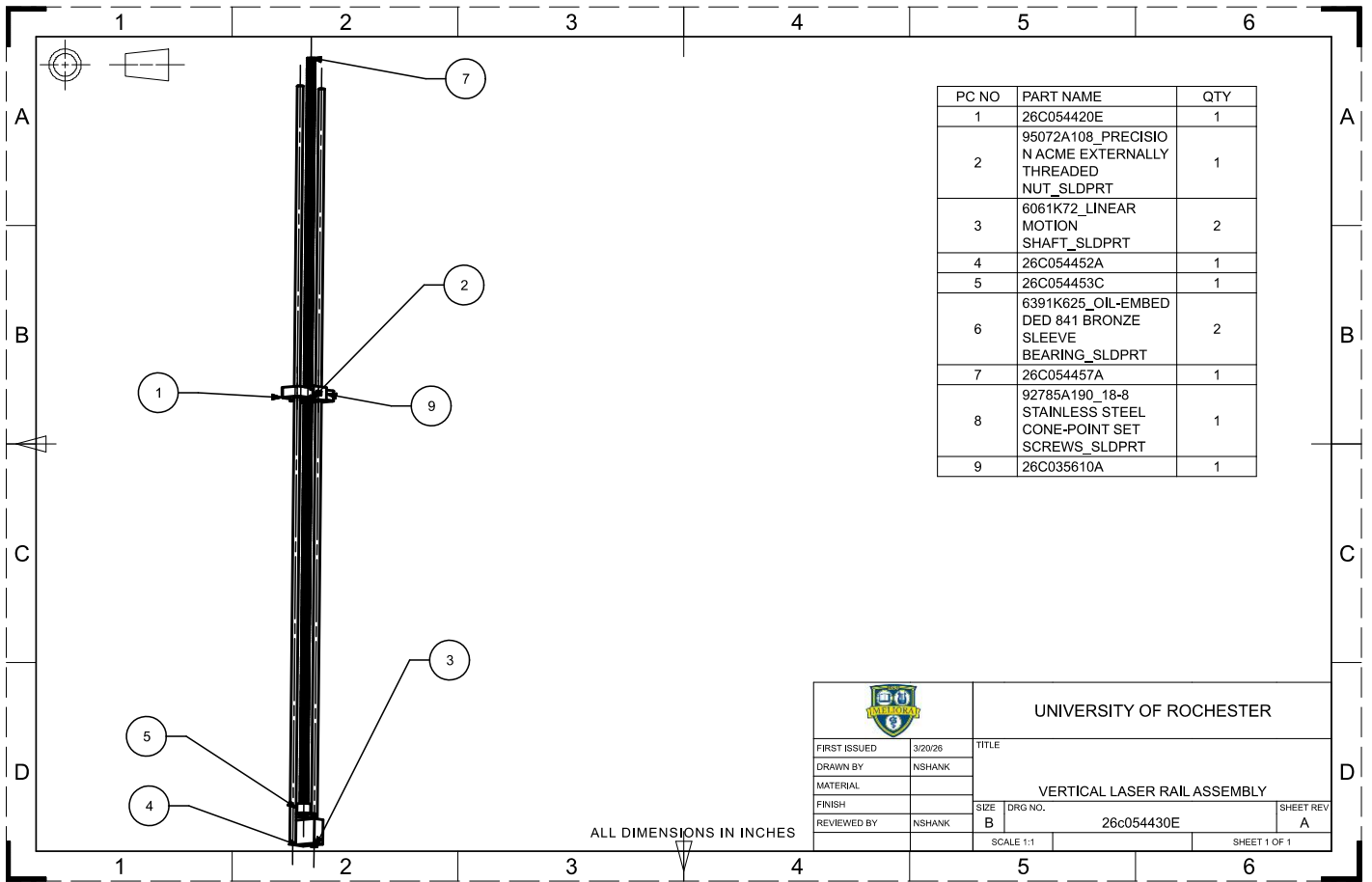


Figure F3: Vertical rail sub-assembly drawing.

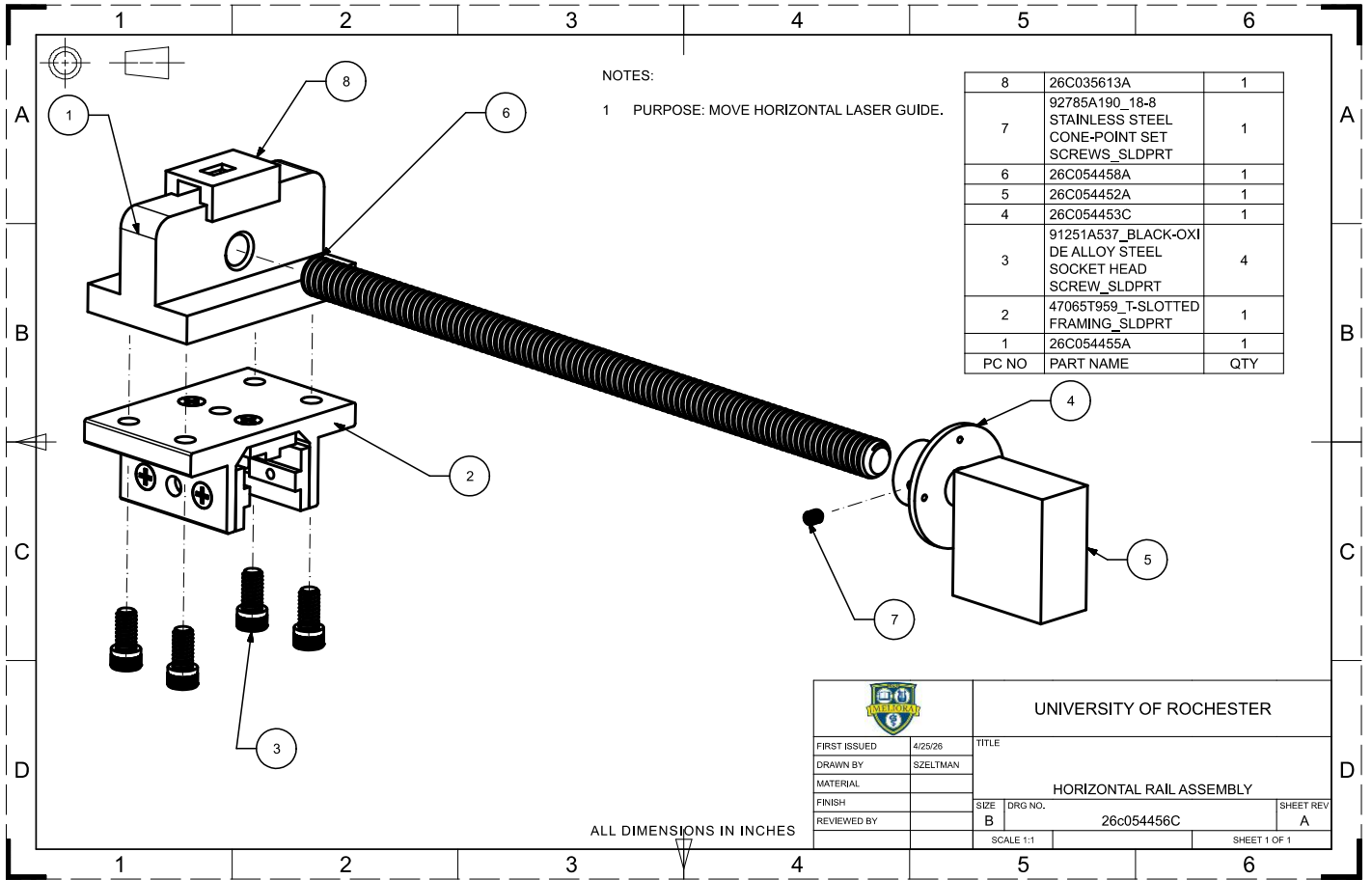


Figure F4: Horizontal rail sub-assembly drawing.

ANNEX G

BILL OF MATERIALS

Table G1: Bill of Materials

Bill of Materials						
item	Source	Part #	Description	Quantity	\$/Each	Total Cost
1	Adafruit	191	Arduino Mega 2560	1	\$39.95	\$39.95
2	Adafruit	4397	STEMMA QT / Qwiic JST SH 4-pin Cable with Premium Female Sockets - 150mm Long	1	\$0.95	\$0.95
3	Amazon		30 Set Black 1"x1" .1010 Series Aluminum Extrusion Profile Connector, 30pcs Corner Bracket with 60pcs M6 T-Slot Sliding Nuts for 10 Series Aluminum Rail Accessories	1	\$23.99	\$23.99
4	Amazon	B0CLH64R4S	RC Servo, High Torque Double Bearing Robot Servo 360-Degree Rotation	2	\$22.12	\$44.24
5	Amazon	B01EV70C78	Multicolored Dupont Wire 40pin Male to Female, 40pin Male to Male, 40pin Female to Female Breadboard Jumper Ribbon Cables	1	\$6.98	\$6.98
6	Amazon	B07ZYNWJ1S	ElectroCookie Prototype PCB Solderable Breadboard	1	\$8.79	\$8.79
7	DigiKey	102-5939-ND	CBL-UA-MB-1	1	\$3.47	\$3.47
8	Grangier	2HXH7	THOMSON Linear Shaft: Non-Hardened Solid Shaft, 3/8 in Dia, 48 in Lg, Steel	2	\$51.61	\$103.22
9	McMaster	47065T959	T-slotted framing	1	\$71.98	\$71.98
10	McMaster	8350T93	1050 Carbon steel threaded shaft	2	\$84.95	\$169.90
11	McMaster	98381A470	Dowel Pin Alloy Steel, 1/4" Diameter, 3/8" Long	2	\$5.50	\$11.00
12	McMaster	99030A105	1018 Carbon Steel Precision Acme Lead Screw	1	\$35.68	\$35.68
13	McMaster	92785A190	Stainless Steel Cone-Point Set Screws 8-32 Thread, 1/4" Long	1	\$0.00	\$-
14	McMaster	6391K625	Oil-Embedded 841 Bronze Sleeve Bearing for 3/8" Shaft Diameter and 7/16" Housing ID, 1/2" Long	4	\$1.42	\$5.68
15	McMaster	6338K581	Oil-Embedded 841 Bronze Flanged Sleeve Bearing	2	\$5.31	\$10.62
16	McMaster	91864A033	Black-Oxide Alloy Steel Socket Head Screw 8-32 X 9/16" L	1	\$13.25	\$13.25
17	McMaster	92141A029	18-8 Stainless Steel Washer 1/4" Screw Size, General Purpose, 0.281" ID, 0.625" OD	1	\$5.50	\$5.50
18	McMaster	91251A540	Black-Oxide Alloy Steel Socket Head Screw 1/4"-20 Thread Size, 3/4" Long	1	\$11.38	\$11.38
19	McMaster	47065T101	T-Slotted Framing Single Four Slot Rail, Silver, 1" High x 1" Wide, Solid	1	\$44.32	\$44.32
20	McMaster	47065T511	T-Slotted Framing Triple Rail, Silver, 3" High x 1" Wide x 24" Long, Solid	1	\$31.61	\$31.61
21	McMaster	1610T15	Multipurpose 6061 Aluminum 2" Diameter Disc	1	\$15.97	\$15.97
22	McMaster	60355K173	Steel Ball Bearing Open, Trade Number R8, for 1/2" Shaft Diameter	1	\$7.72	\$7.72
23	McMaster	89015K231	Multipurpose 6061 Aluminum Sheet 1/8" Thick, 2" x 24"	1	\$13.41	\$13.41
24	McMaster	89015K284	Multipurpose 6061 Aluminum Sheet 0.190" Thick, 18" x 18"	1	\$80.07	\$80.07
25	McMaster	8975K513	Multipurpose 6061 Aluminum Bar 1/2" Thick x 3" Wide x 12" Long	1	\$24.25	\$24.25
26	McMaster	91251A624	Black-Oxide Alloy Steel Socket Head Screw 3/8"-16 Thread Size, 1" Long	1	\$12.48	\$12.48
27	Mouser	GP2Y0E02B	Distance measurement sensor	2	\$11.24	\$22.48
28	SMC	01292	1-1/2 x 4 x 15.5 Aluminum 6061	1	\$72.00	\$72.00
29	SMC	01318	2" x 4" x 12" Aluminum 6061 Bar	1	\$70.00	\$70.00
30	SMC	01266	1-1/4"x2"x9" Aluminum 6061 Bar	1	\$18.00	\$18.00
31	SMC	01284	1-1/2"x2"x8" Aluminum 6061 Bar	1	\$18.00	\$18.00
32	SMC	01458	1" DIA x 27" Aluminum 6061 Round Stock	1	\$18.00	\$18.00
Total number parts				41	Total Costs	\$1,014.89