Course Project - Custom Wrench

Design of Mechanical Components 313

Michael Carlone, Michael Grady, Calvin Lareau, Dinor Nallbani

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Introduction & Requirements

Extra-Ultralight Biking LLC specializes in providing high-performance, ultralight biking gear for long-distance riders. The company's customer base prioritizes reducing weight in every aspect of its equipment. Extra-Ultralight Biking has hired our team with the task of designing a lightweight, functional wrench. The wrench must meet the needs of these riders while offering a balance of performance, weight, and additional versatility.

The constraints of this project are to create a wrench out of AISI6061-T6 aluminum that weighs no more than 1.25 oz (~35.4 g) while ensuring it remains robust enough for basic equipment repairs during expeditions. The wrench must therefore incorporate at least one additional functionality beyond basic torque application to minimize the necessity of carrying extra tools. We will fulfill this by machining a flat-head screwdriver as part of the handle. The handle of the wrench must be at least 6 inches long, provide a flat surface for testing, and offer a secure grip for the user under normal use conditions. The jaws of the wrench will be designed to accommodate a fixed bolt size and required torque load, which will be determined by given specifications. To ensure the longevity of the wrench, our team has required that the factor of safety with which we are designing the wrench must be ≥ 2.

In this report, we will outline the concept generation, analysis, manufacturing, and design process, along with the final test results. Emphasis will be placed on optimizing weight while maintaining the necessary durability and usability to pass required performance tests. The functional test evaluates the ability of the wrench to loosen a ⅜-16 bolt tightened to 170in-lb. The stiffness test evaluates the strength of the handle through a three-point bend test with a 50 N load applied to the center of the handle.

Concept Generation

Each team member independently generated one concept; then, we came together to select which aspects and features to carry into our design and analysis process. We took into consideration all design requirements before selecting the design we wanted to continue developing and improving to fulfill project requirements and maximize performance. Each of the designs developed took a different approach to the wrench, and more distinctively, the added functionality outlined in the project requirements.

Concept 1: Developed by Michael Carlone

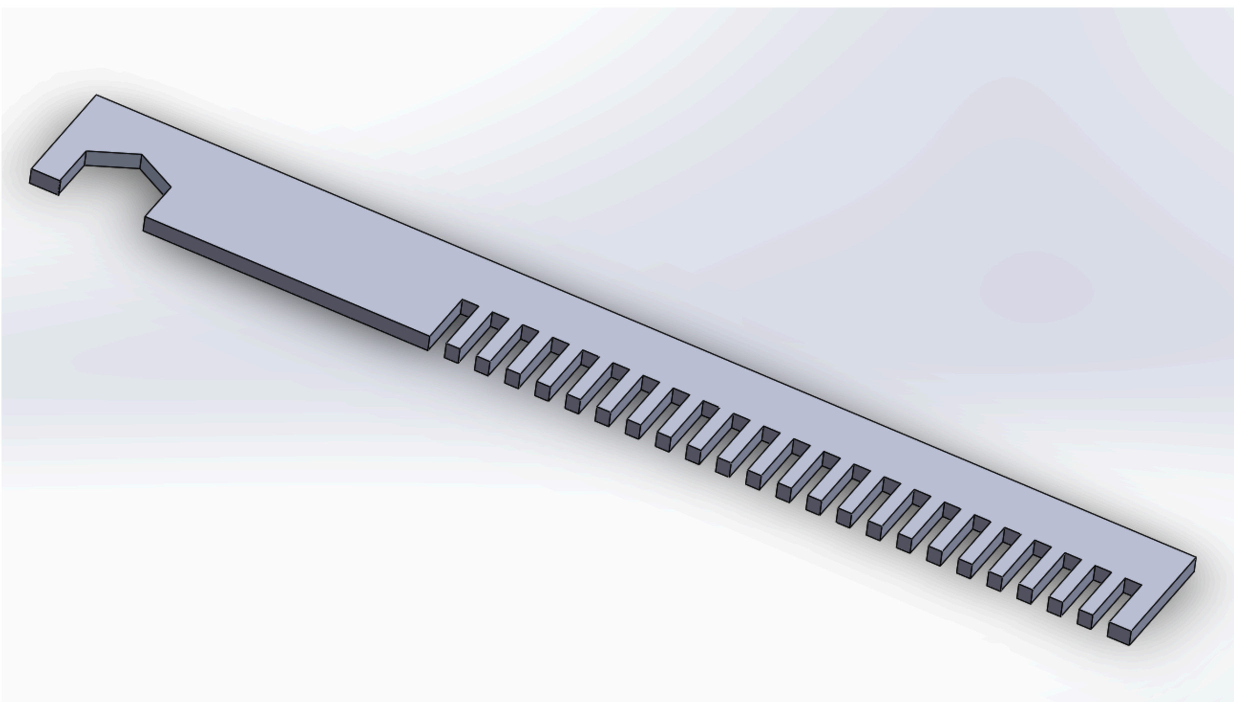

Figure 1. Preliminary wrench concept generated by Michael Carlone. Additional functionality: Comb

For this preliminary design, Michael aimed to keep the overall structure as simple and machinable as possible. He focused on utilizing straight edges and minimizing the number of angled turns, streamlining the manufacturing process. This approach ensures that the wrench can be produced quickly and effectively while maintaining the necessary strength and durability for its intended use.

For the added functionality, a comb feature was integrated into the design. This comb is located on one side of the wrench, positioned strategically to allow for its use without interfering with the wrench’s primary function. The design of the comb ensures that it does not add unnecessary complexity to the tool, being simple to manufacture while still providing the additional utility. The back side of the wrench is left smooth and unimpeded, making it easy for the user to grip and apply torque when using the wrench itself.

Concept 2: Developed by Michael Grady

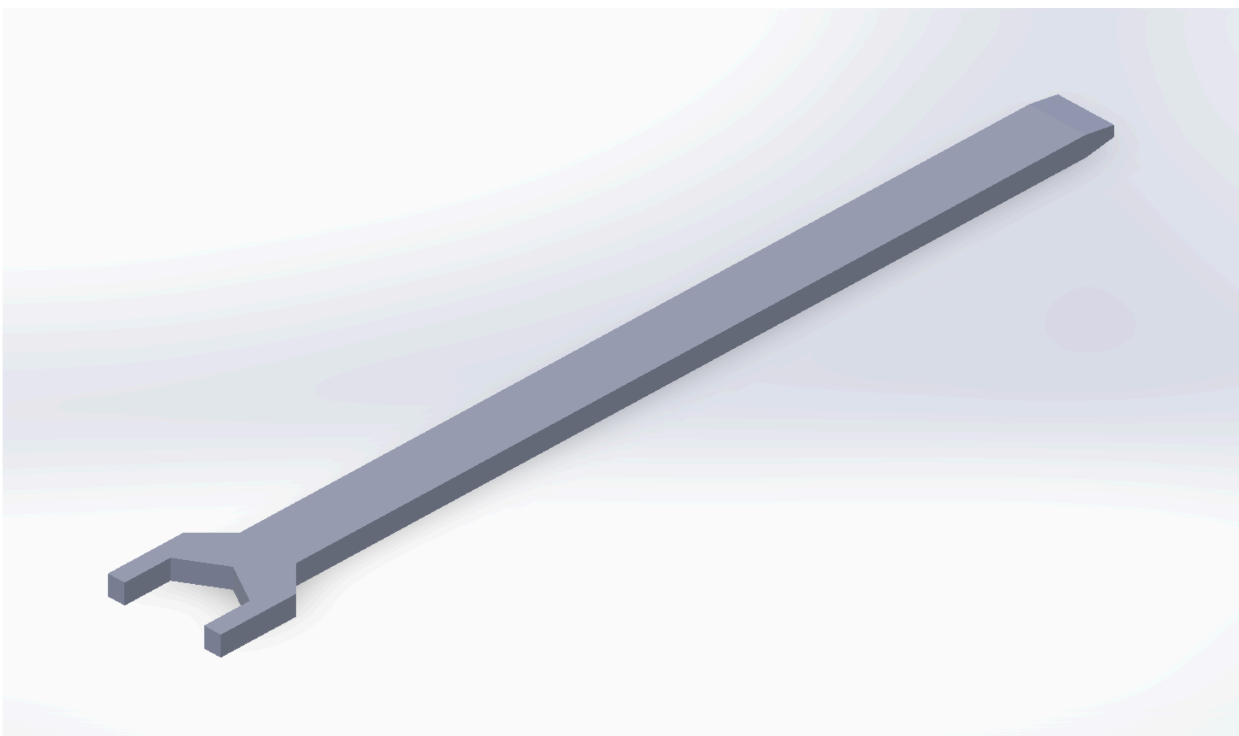

Figure 2. Preliminary wrench concept generated by Michael Grady. Additional functionality: Flathead Screwdriver

For this design, Michael chose to implement a front-facing bolt cavity with a flathead screwdriver machined into the rear of the wrench as an additional functionality. The front-facing cavity aims to create a wrench that is easier to use while keeping relatively simple machining cuts. The dimensions of the wrench are thin to reduce the weight at the potential cost of strength.

The flathead screwdriver was chosen as the additional functionality due to the ease of machinability and the weight-to-functionality ratio that it provides. The screwdriver encompasses a small portion of the design, weighs very little, and does not interfere greatly with the ability of the wrench while providing an important use.

Concept 3: Developed by Calvin Lareau

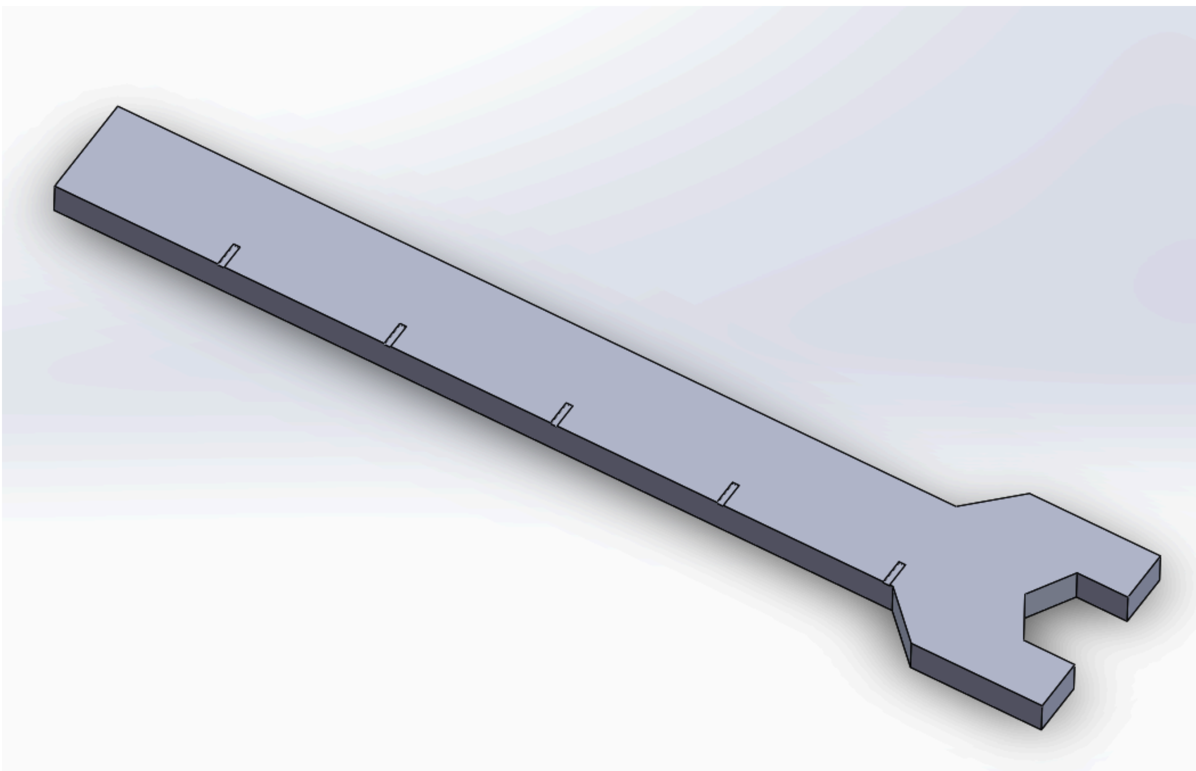

Figure 3. Preliminary wrench concept generated by Calvin Lareau. Additional functionality: Ruler

Calvin's preliminary design prioritized the strength and functionality of the wrench highly. Similarly to the previous design, a front-facing, hexagonal bolt cavity was used. This front-facing design had a larger head and thicker prongs to increase their strength and the load the wrench could sustain before deforming. A ruler was integrated into the design by etching hash marks at 1-inch increments along the handle. This simple-to-implement feature would not interfere with the use of the wrench, compromise strength, add excess weight, or complicate machining; yet, it provides a useful secondary function for the rider.

Concept 4: Developed by Dinor Nallbani

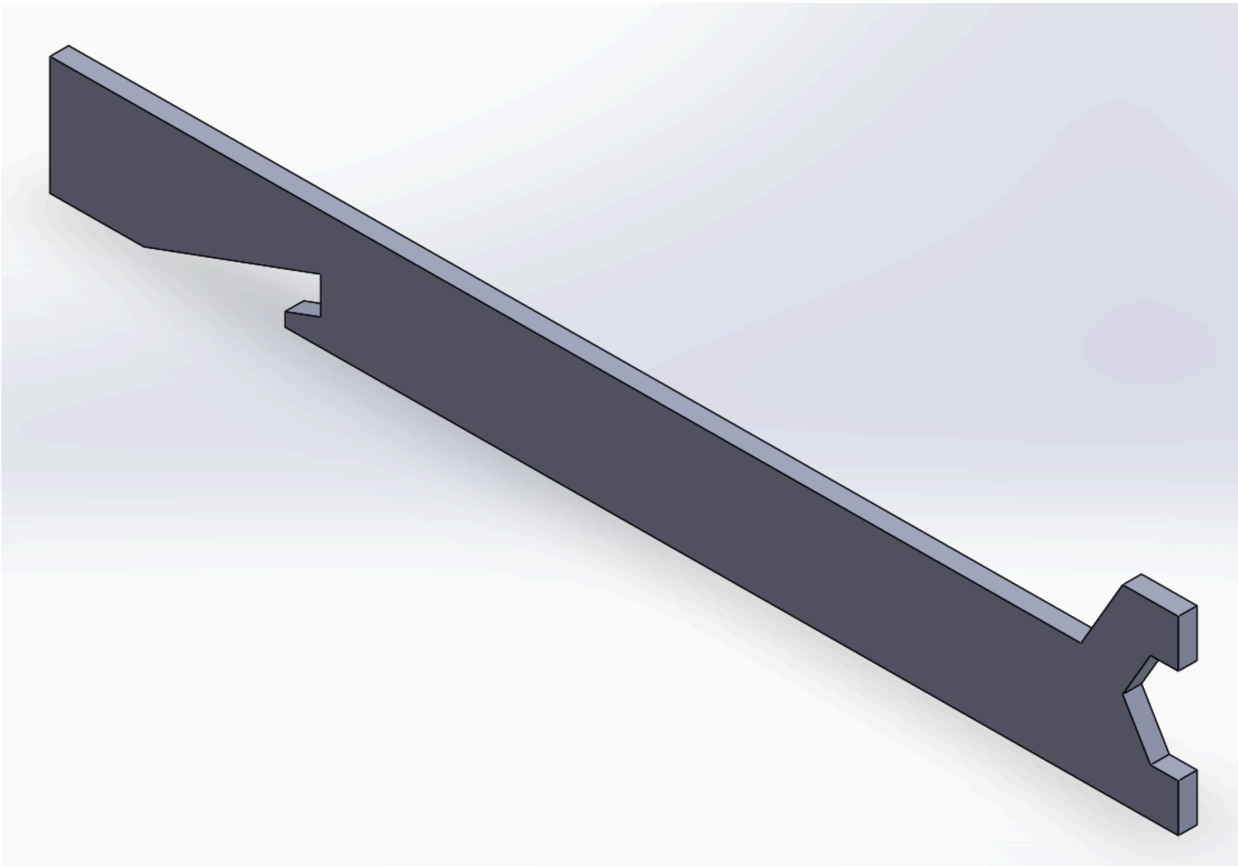

Figure 4. Preliminary wrench concept generated by Dinor Nallbani. Additional functionality: Bottle Opener

As Dinor's first concept, he wanted to do something a little unconventional. There was an attempt to ensure machinability with the whole wrench being able to be manufactured using straight-line cuts. Utilizing this approach allows for the wrench to be easily manufactured with fewer hiccups than if it were to be made with a more complicated design

The idea behind the wrench cut-out being on only one side of the wrench is that for a rectangular cross-section, the height matters much more than the width. Since we were given the requirement to make the wrench as light as possible, making the wrench stronger in the required direction is far more important than the direction perpendicular to it. That is why this concept also has the wrench quite thin, as the majority of the bending that needs to be resisted will be in the horizontal direction.

Each of the provided concepts featured an open jaw located on or near the end of the wrench, so we chose to maintain this feature in our subsequent designs. We elected to place the bolt cavity in the middle of the wrench, rather than on the side, to avoid concentrating stress on a given side, as we felt this would increase the likelihood of failure at that point. We further selected the flathead screwdriver as our additional feature, as we felt it was the most relevant in the context of emergency bike repairs, making it most synergistic with the wrench's primary function. Additionally, its high utility did not interfere with machinability, strength, or weight compared to other features.

By-hand Analysis Set-up

Our by-hand analysis was performed by analyzing the two loading conditions of the torque/functional test and 3-point bending test separately, utilizing the Euler-Bernoulli beam theory. This method predicts lower stress values than a Von Mises equivalent stress analysis, consequently suggesting smaller required dimensions for the wrench. However, after discussing this approach with Professor Olson, we determined it was an appropriate process for identifying preliminary dimensions to inform our finite element analysis (FEA). In addition to the loading conditions, we employed the weight constraint of 1.25 oz imposed on the wrench to derive a third, competing dimensional constraint.

In our analysis, we modeled the wrench as a uniform bar with a constant cross-sectional area with dimensions of the *base (b) x height (h)*, where the base dimension is parallel to the ground in the jaw-lying-flat orientation. Further, we selected a total length of 7 inches for the bar, which consisted of the 6 inch required handle length and a 1 inch allotment for the jaw. We assumed all Euler-Bernoulli beam theory assumptions (constant C.S., away from loaded ends, homogeneity, small deformations, etc). In choosing to ignore head geometry we did not consider stress concentrators resulting from chamfers or cuts. We also did not consider transverse shear as a significant factor in driving failure. We utilized a factor of safety of 2 in our derivations and obtained the material property values of AL 6061-T6 from Matweb (σ_y= 276MPa = 40ksi, ϱ = 0.0975 lb/in^3).

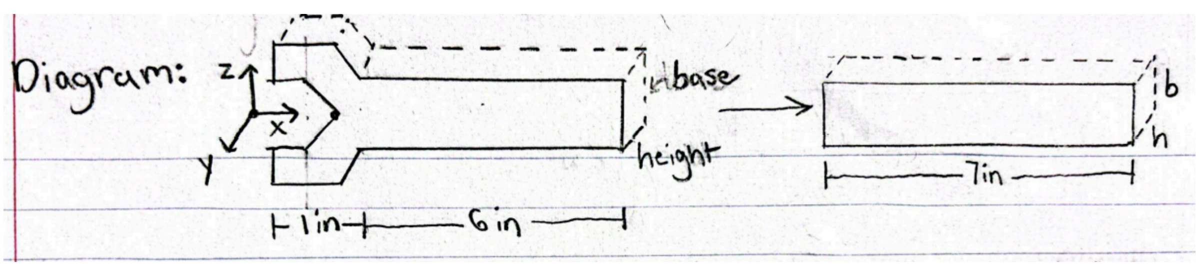

Figure 5. Shows simplification of geometry, as well as the wrench's orientation in the functional test condition

For the 3-point bend test, we modeled the Instron testing apparatus' supports as simple supports, with the first located 1in along the bar (at the start of the handle) and the second 6in away at the bar's edge. The 50N testing force was modeled as a point force in the exact center of the handle, 4in along the bar. The shear force, bending moment, and maximum allowable bending stress were calculated. These values were then employed to derive a constraint for the cross-sectional dimensions, as illustrated in Figure 6 below.

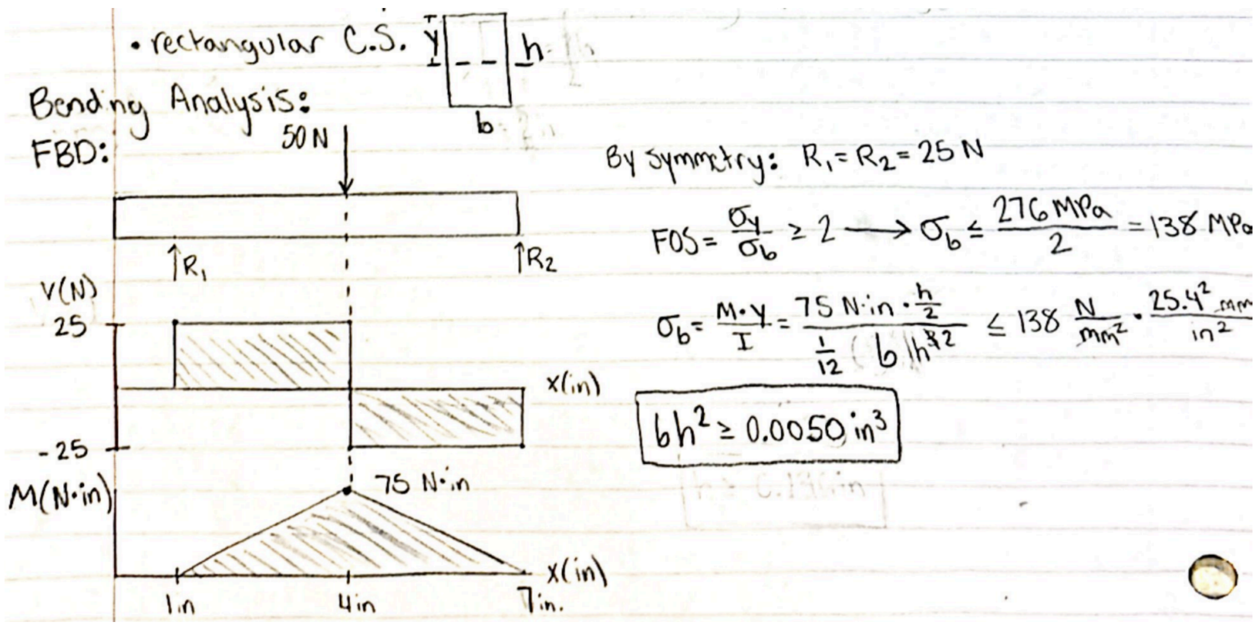

Figure 6. Free body, shear, and moment diagrams; calculation of maximum allowable bending stress; and derivation of cross-sectional dimension constraint for 3-point bend test.

For the functional test, we modeled the bolt as a fixed support at the jaw-end of the bar (x=0) which produced a 170 in-lb bending moment reaction (equal to our required torque). Then, we modeled the force applied by the user in turning the bolt as an evenly distributed force along the back 3in of the bar (4≤x≤7) with an intensity of "w" lb/in. Three inches was selected based on the team member's average hand width across the knuckles of ~3.5in, as the effective width contributing to force application would be slightly less due to hand curvature and grip. Notably, the wrench's orientation in this test is different from that of the 3-point bend test, as the jaws are upright rather than flat (figure 5); thus, the "base" and "height" are switched, which is very important in calculating the moment of inertia (figure 7). To determine the value of the intensity "w," a moment balance about the origin of the bar (x=0) was performed by converting the distributed load to its equivalent concentrated load of w*3 lb applied at the midpoint of distribution, x=5.5 (figure 7). The vertical reaction force R_y at the fixed support was calculated and shear and moment diagrams were rendered. Finally, the resulting bending stress was calculated and compared to the maximum allowable bending stress to derive a constraint for the C.S. dimensions. Diagrams and the complete derivation are illustrated below in Figure 7.

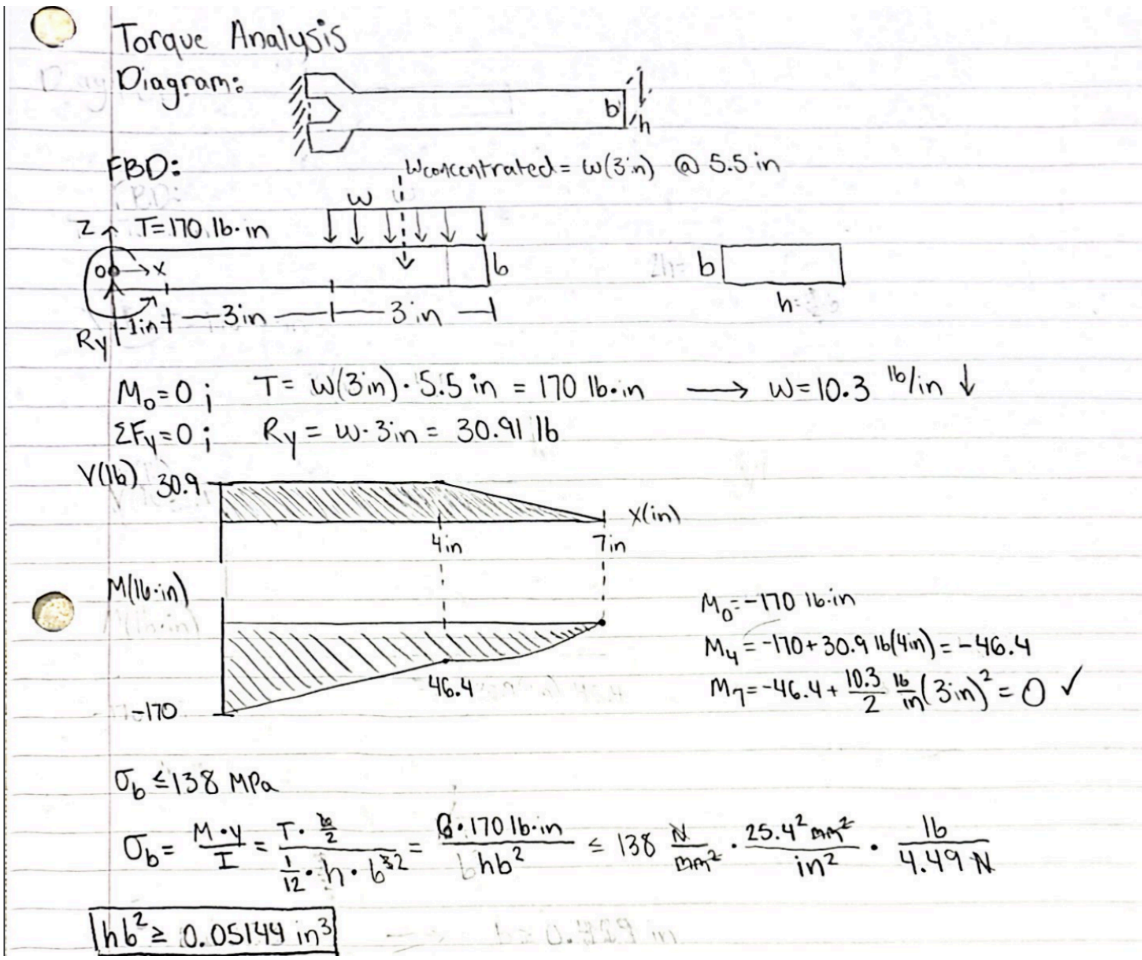

Figure 7. Free body, shear, and moment diagrams; calculation of distributed load intensity, and reaction force; and derivation of dimension constraint

Finally, we derived a constraint for the cross-sectional dimensions to account for the weight limit of ~35.4g imposed on the wrench. We created an expression for the bar's weight by multiplying our rod's base, height, and length by the density of Al6061-T6 and set this value to be less than or equal to the weight limit. The result of this derivation is illustrated in Figure 8.

Weight Analysis: $W \leq 1.25\ oz$

$$W = V \cdot \rho = b \cdot h \cdot L \cdot \rho = bh \cdot 7in \cdot 0.0975\ lb/in^3 \cdot \frac{16\ oz}{lb} \leq 1.25\ oz$$

$$\boxed{bh \leq 0.1145\ in^2}$$

Figure 8. Derivation of weight's constraint on cross-sectional dimensions

We utilized these constraints by inputting different ratios of height and base (i.e b=2h) to determine values at which all three competing constraints could be met. This numerical process is illustrated in the Design Process section as specified.

Finite Element Analysis Set-up

To ensure that each new iteration was improving in either strength or weight to the previous design, finite element analysis was done to determine the characteristics of the design given the required loading conditions. The same (or very similar) steps were followed for each design in set-up to ensure comparability between each design. Ansys was utilized as the primary FEA software. To begin, the completed CAD model was uploaded and given the necessary material requirements. Because the wrench was to be made of AISI6061-T6 Aluminum, we supplied Ansys's engineering data with the relevant properties: an isotropic elasticity model with a Young's Modulus of 68.9 GPA and a Poisson's Ratio of .33 (sourced from MatWeb). From there, we determined a reasonable mesh size given the overall size of the wrench. A body sizing for both bodies (note the split in the middle of the handle) was employed with a 2 mm element size. The resulting mesh for Version 1 is shown below.

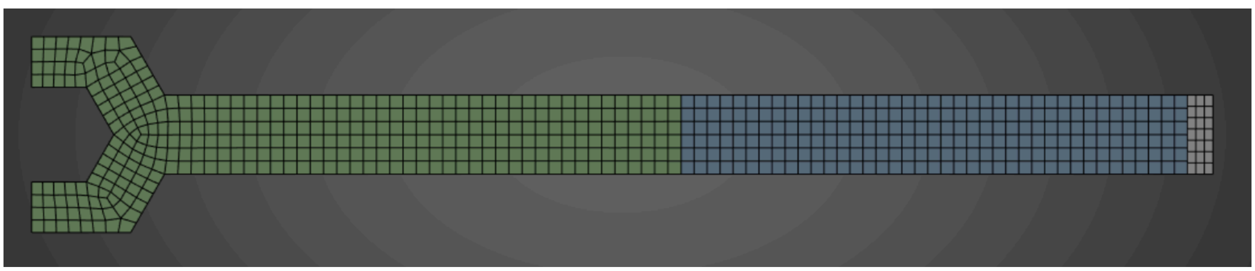

Figure 9. Top view of body sizing mesh with 2 mm element size.

In the case of both the combined loading condition (explained later) and just the 170 lb-in torque, fixed supports were applied to both faces on the inside of either head prong. This was done with the assumption that the bolt head would not be in contact with the chamfer/fillet that some of the iterations have within the head shape. Because of how a wrench is used, this support setup was deemed reasonable for the location and method for "fixing" the areas where the bolt head comes into contact. This is shown in Figure 12A.

For the loads in either case, the 170 lb-in torque (determined to be 30.9 lbf distributed across 3 inches - see hand calculations for reasoning) is applied to the back 3 inches on the side of the wrench, simulating a hand pushing and turning it appropriately. This is shown in Figure

12B. As previously mentioned, a split was created in the CAD model 3 inches from the chamfer of the screwdriver.

Due to the “unsteadiness” of the turning motion, some perpendicular force will accompany the torque when the wrench is in use. This defines the combined loading condition. We estimated that 24 N (5.4 lbf) was reasonable (if anything, too strict) for the perpendicular force that will also be applied to the back 3 inches of the handle. This is shown in Figure 12C. This results in a combined loading scenario that mirrors the real-life method of rotating a wrench about a bolt head. Fixed supports are where the bolt applies its resistive torque, the handle torque represents the hand doing the turning, and the perpendicular force represents the unsteadiness/flexibility of the hand during the movement. This is shown in Figure 13.

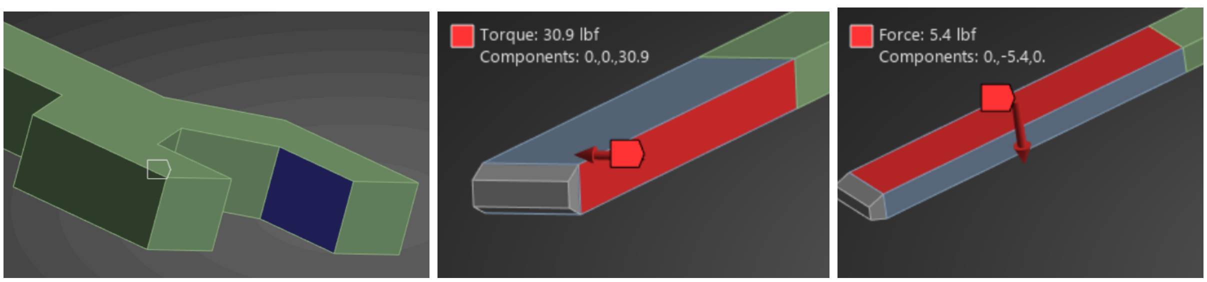

Figure 12. A: Fixed supports on prongs *Figure B: 30.9 lbf torque* *Figure C: Perpendicular 5.4 lbf force*

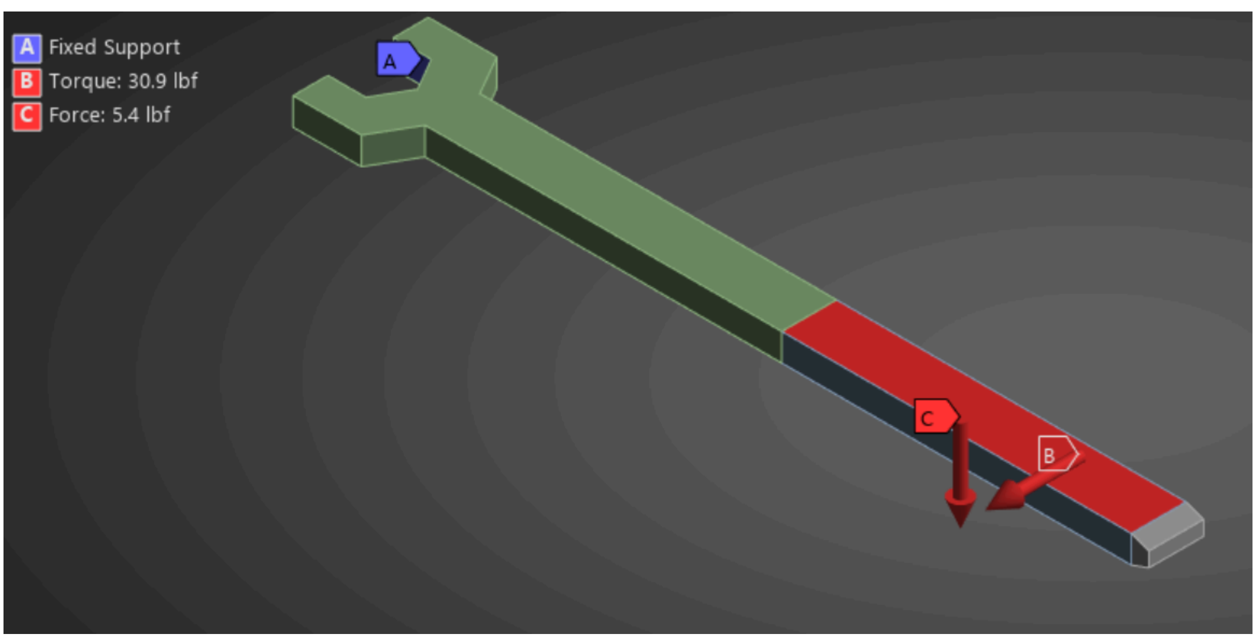

Figure 13: Combined loading condition

The combined loading condition was the primary driver of each iteration - i.e. we ensured that the Von Mises equivalent stress for this scenario was below the 40,000 psi yield strength of the aluminum stock (sourced from Matweb) before we moved to the next version. For each iteration, this scenario was done in place of just the 170 lb-in case, as we assumed that if the wrench could survive the more strenuous test, it should have no problem when the 24N perpendicular load is not applied.

For the 50N bending test, fixed supports were applied to the bottom of the wrench. One support was placed on the edge of the handle, where the chamfer for the screwdriver begins. The other support was placed on a split edge made so that six inches of the "handle" remained before the chamfer for the screwdriver began. The 50 N load was applied to the top face of the wrench pointing downwards in the middle of the two support locations, mimicking the location and maximum load of the Instron's head applying the force. Here, we assumed that the wrench would not be fixed to the machine in any manner. We utilized fixed supports to maintain the wrench's position, but we assumed that those locations would remain in contact with the testing device. Regardless, reasonable values for the equivalent stress were found, as will be explained and shown later, so no justifiable evidence would have influenced switching the way conditions were applied. The bending test, with both fixed supports and the 50N load, is pictured in Figure 14.

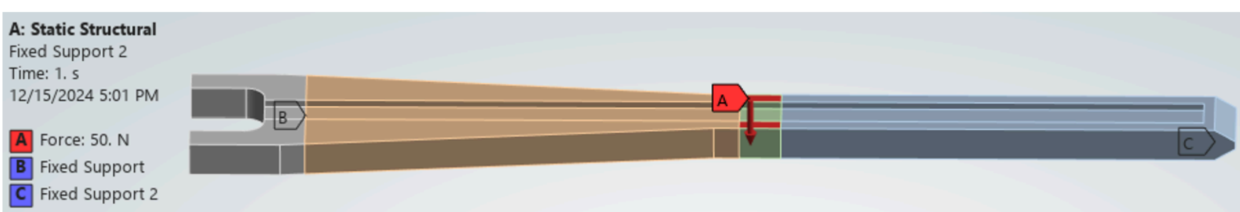

Figure 14. Bending test with both fixed supports and 50N load

Design Process

As described in the by-hand analysis section, we utilized the constraints derived from the functional test, bending test, and weight limit to determine our preliminary dimensions for the wrench by inputting different ratios of height and base until all three constraints could be met, as shown below. This led us to select the ratio of b=2*h and dimensions of height=0.235in, b=0.47in to serve as the starting point for designs.

From bending: $bh^2 \geq 0.005\ in^3$ 1)
torque: $hb^2 \geq 0.05144\ in^3$ 2)
weight: $bh \leq 0.1145\ in^2$ 3)

if $b = \frac{1}{2}h$: 1) $h^3 \geq 0.01\ in^3 \rightarrow h \geq 0.216$ in
2) $\frac{1}{4}h^3 \geq 0.05144\ in^3 \rightarrow h \geq 0.59$ in not possible
3) $\frac{1}{2}h^2 \leq .1145\ in^2 \rightarrow h \leq 0.48$ in

if $b = h$: 1) $h^3 \geq 0.005\ in^3 \rightarrow h \geq 0.17$ in
2) $h^3 \geq 0.05144\ in^3 \rightarrow h \geq 0.372$ in not possible
3) $h^2 \leq .1145\ in^2 \rightarrow h \leq 0.338$ in

if $b = 2h$: 1) $2h^3 \geq .005\ in^3 \rightarrow h \geq 0.136$ in
2) $4h^3 \geq .05144\ in^3 \rightarrow h \geq 0.234$ in ✓ constraints possible
3) $2h^2 \leq .1145\ in^2 \rightarrow h \leq 0.239$ in

Final/selected design: AL6061-T6 bar w/ h=0.235in, b=0.47in

Figure 15. Final design dimensions from by-hand analysis

We employed these dimensions when designing our first iteration, maintaining a constant height of .235 inches throughout and a base of .47 inches in the handle. Of course, this width was deviated from in the geometry of the head/jaws. Throughout the iterative design process, we utilized Ansys to model how each iteration performed under the combined loading condition and used the results to drive further changes. We continuously took note of how our changes affected the maximum stress endured by the wrench and what Factor of Safety (FOS) this yielded in comparison to the material's 40 ksi yield strength. Given the importance of lightweighting, we further examined the FOS-to-weight ratio of the wrench. This ratio was calculated as FOS/grams, where a larger number indicated a better relationship between performance and weight.

All photos of Ansys FEA shown below (apart from the bending test case for V8) are shown with a true (1.0) deformation scale, meaning every image is as close to real-life performance as possible. The photos depict the combined loading condition, where the wrench is subjected to both the 170 in-lb torque of the bolt and a 24 N incidental force, as described in the ANSYS set-up section.

Version 1

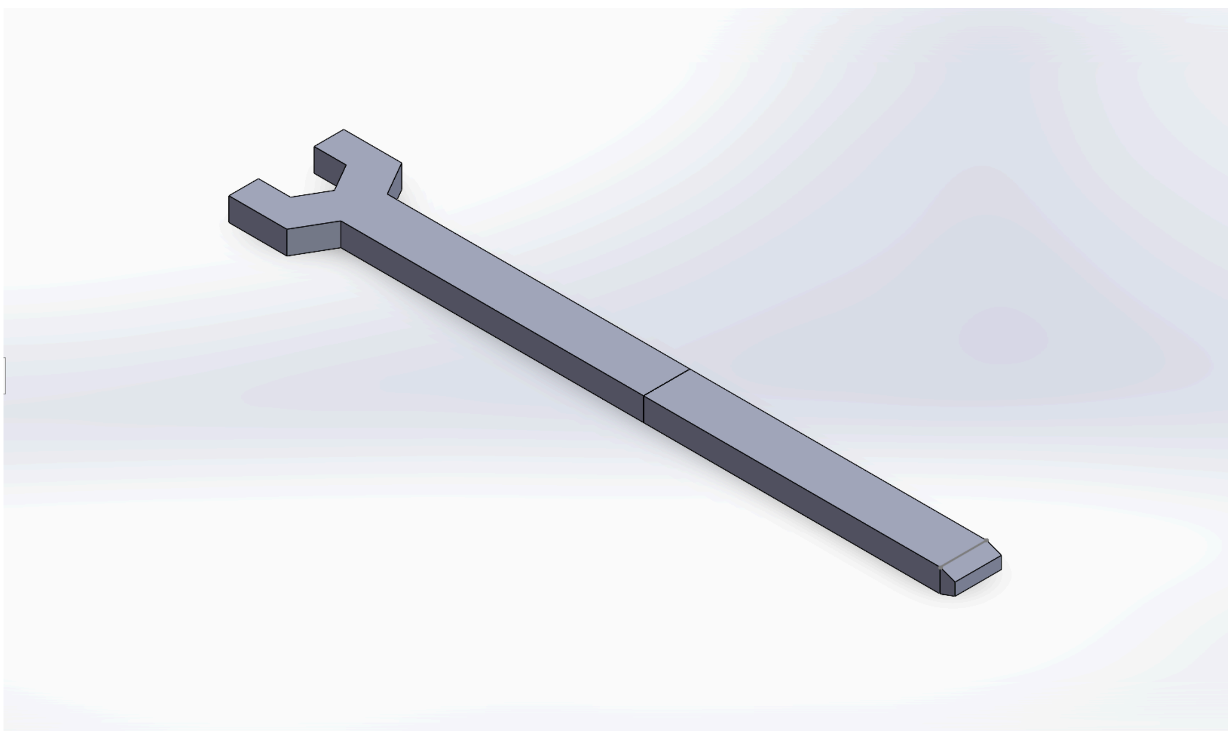

Figure 16. Solidworks of Version 1

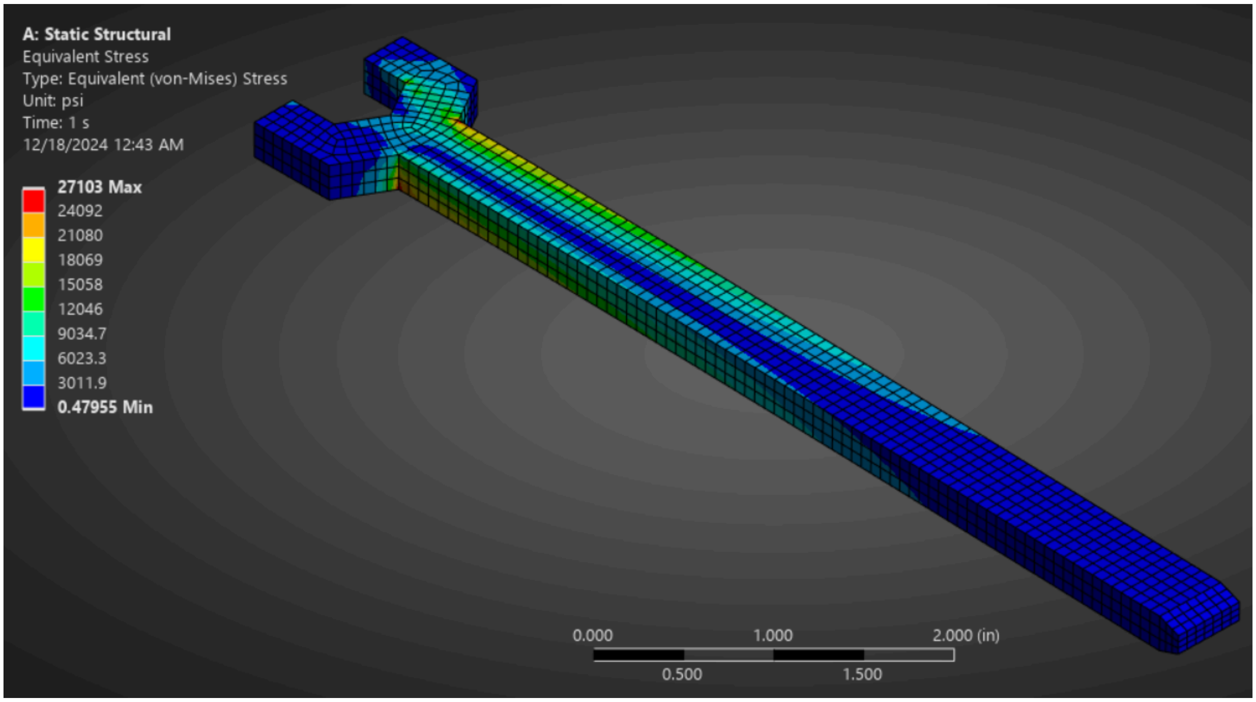

Figure 17. ANSYS Von Mises Stress for Version 1

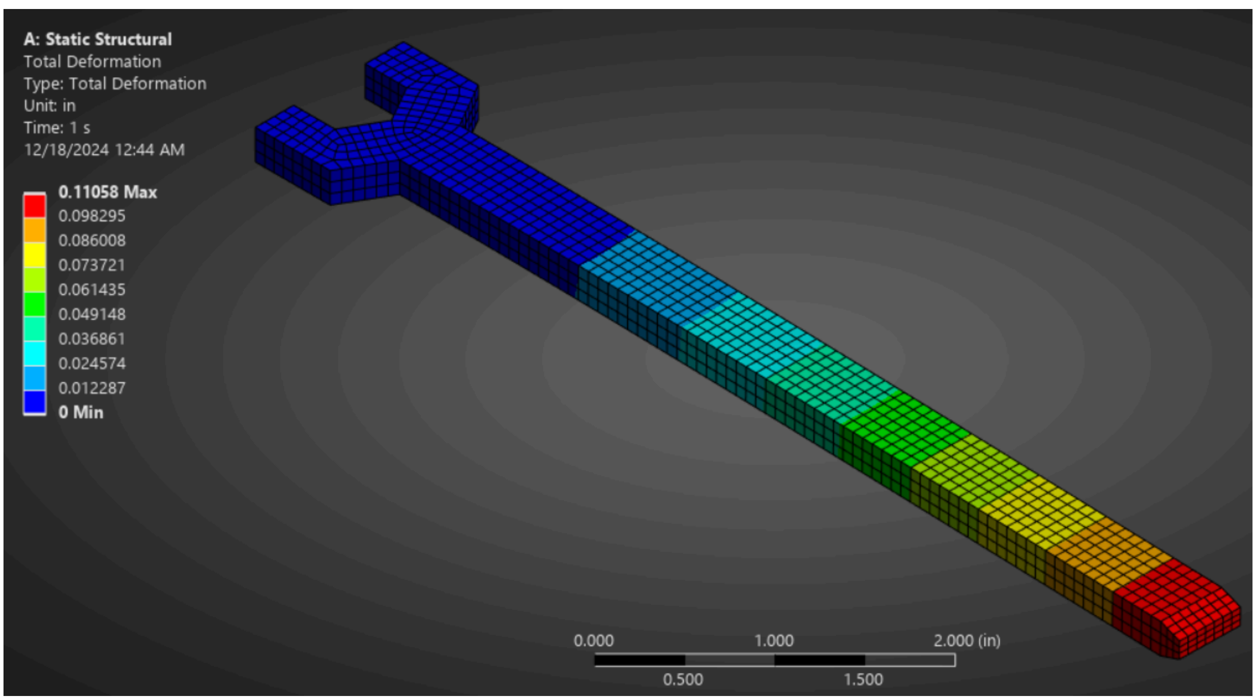

Figure 18. ANSYS Total Deformation for Version 1

Our very first design, as mentioned before, started off with a wrench height of 0.235 inches and a handle width of 0.47 inches. Using Solidworks' mass properties tool, the mass was calculated at 36.59 grams, which was above the limit, but fairly close for a first design. The chamfer connecting the head of the wrench to the handle was set to sixty degrees, following the angle of the bolt head.

Once we ran it through Ansys, we found the estimated stress to be about 27.1 ksi. This was great news; even though our mass was a little over the maximum allowed 35.4 grams, the maximum stress found in the wrench was below the material's yield strength. Given this preliminary success, the following iterations were designed with a perspective of refining the design rather than completely innovating it. V1's factor of safety equaled 1.48, below our desired FOS of 2, but we decided to address the issue of weight first. The FOS/g was 0.0403, which was a starting point to try and build off of.

Version 2

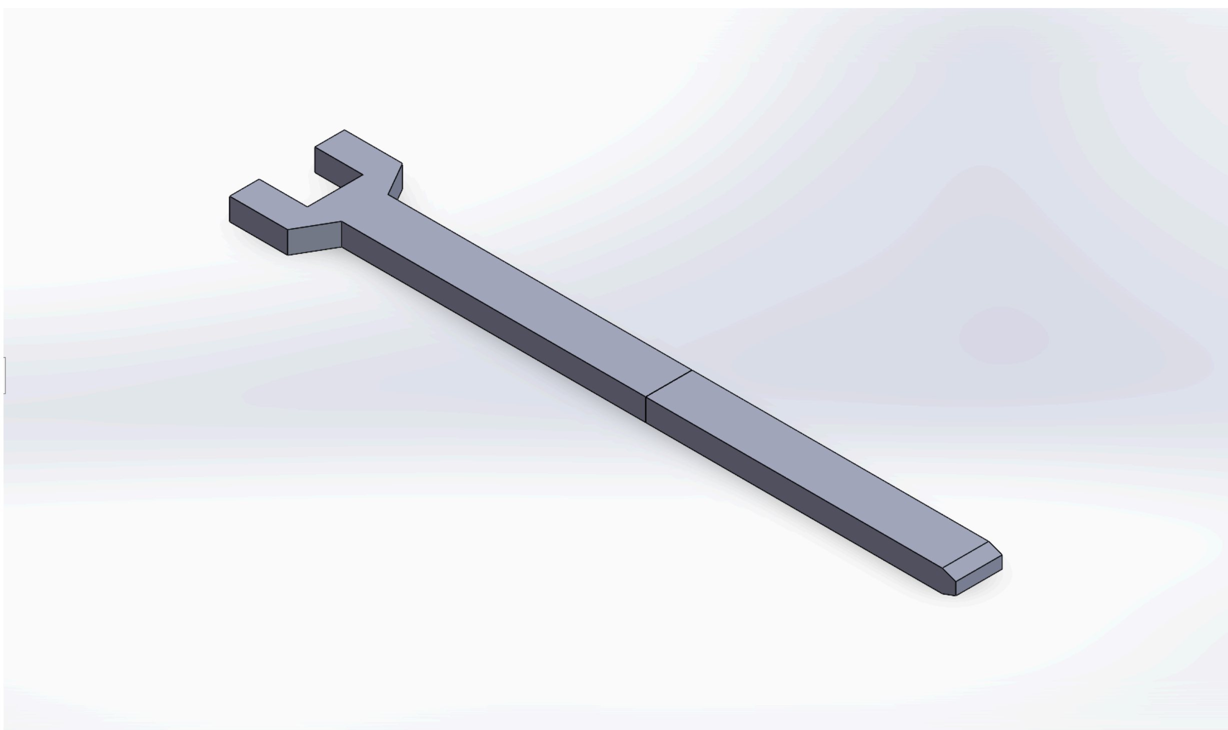

Figure 19. Solidworks for Version 2

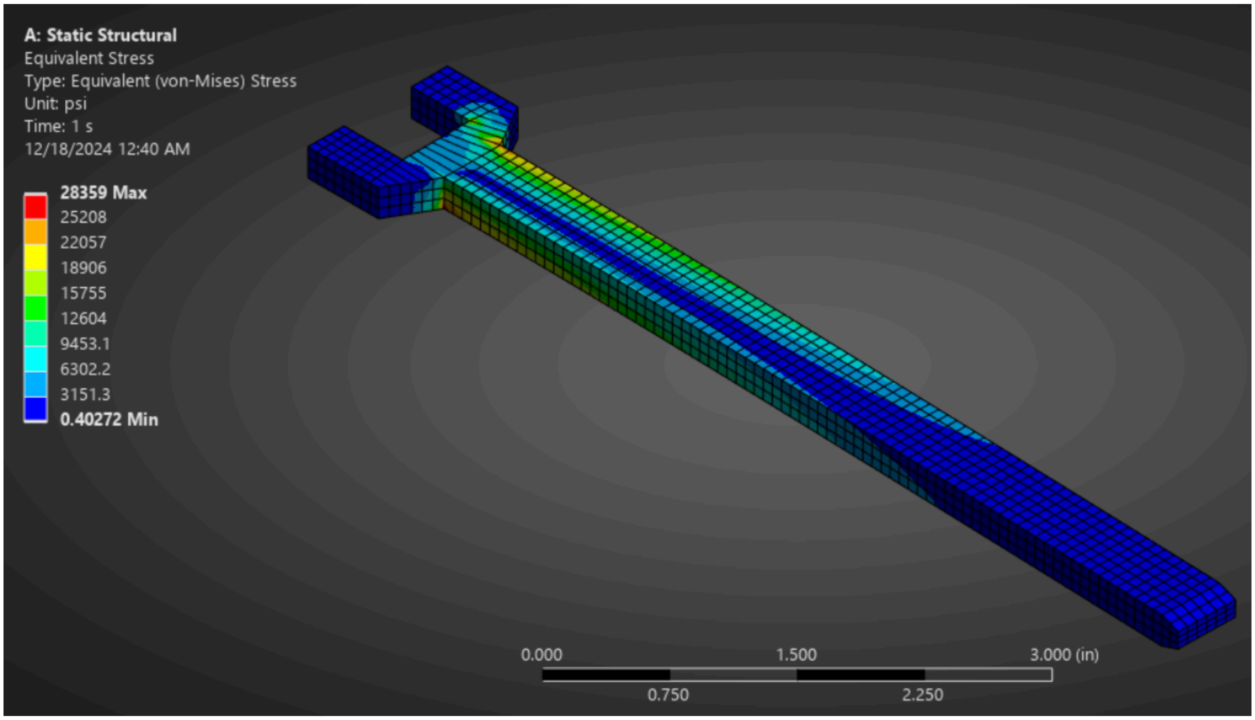

Figure 20. ANSYS Von Mises Stress for Version 2

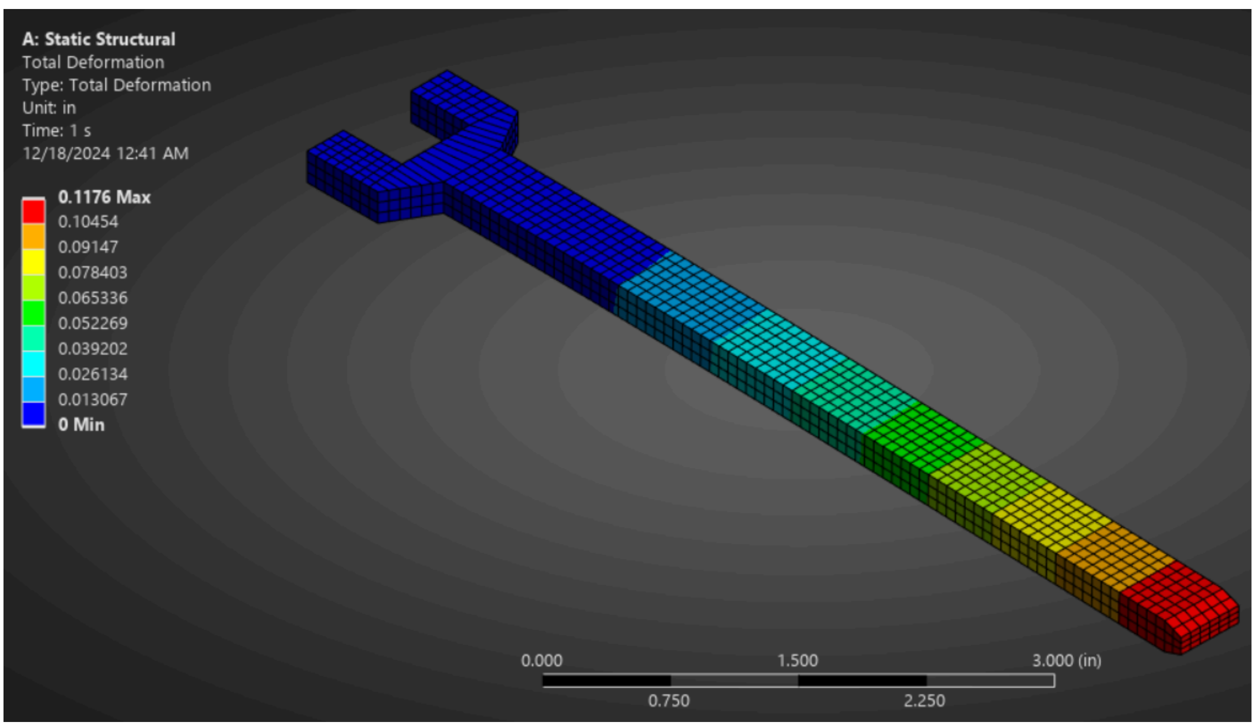

Figure 21. ANSYS Total Deformation for Version 2

After our first iteration, we sought to decrease the wrench's weight. Thus, in the second iteration, we deviated from the hand calculation values and slimmed down the wrench handle. This decision was based on the Von Mises stress distribution, which depicted that the handle's body was plenty strong enough to withstand the forces applied, while most of the stress was concentrated where the chamfer of the jaws met the handle. As a result, we decreased the height of the entire wrench from 0.235 inches to 0.225 inches, and the width of the handle from 0.47 inches to 0.45. Another factor we improved in this iteration was manufacturability. The jaws were changed from a hexagonal shape that would perfectly fit the bolt into a square which would be far easier to machine.

The combined changes increased the maximum stress to 28.3 ksi, still below the materials yield stress of 40 ksi. As a result, the FOS for version two was lower at 1.41, but also the weight of the wrench decreased to 33.33 grams. Combining that information provided us with an FOS/g of 0.042, an increase from version one.

Version 3

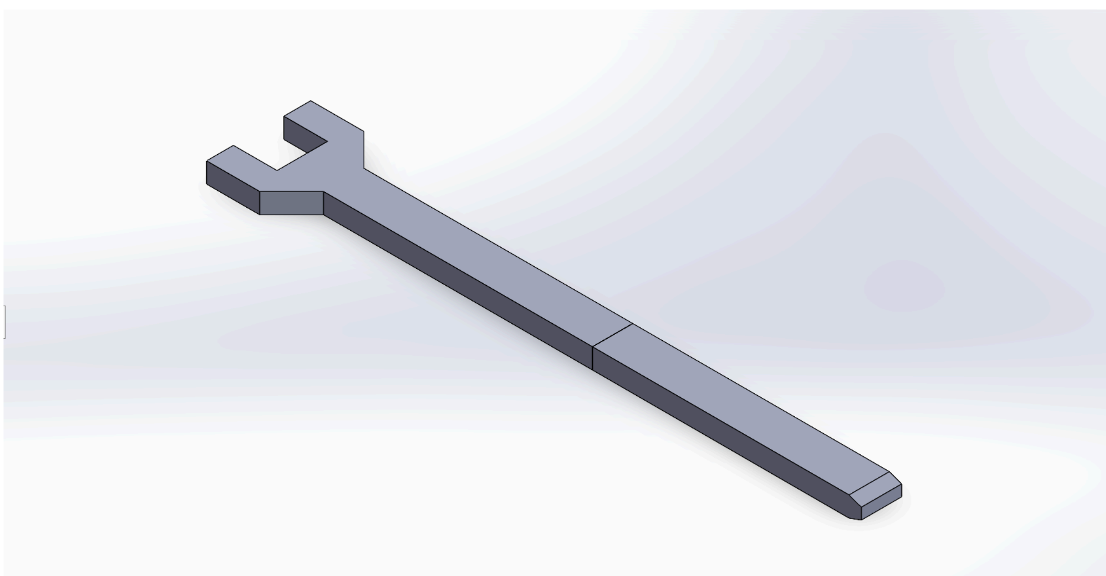

Figure 22. Solidworks for Version 3

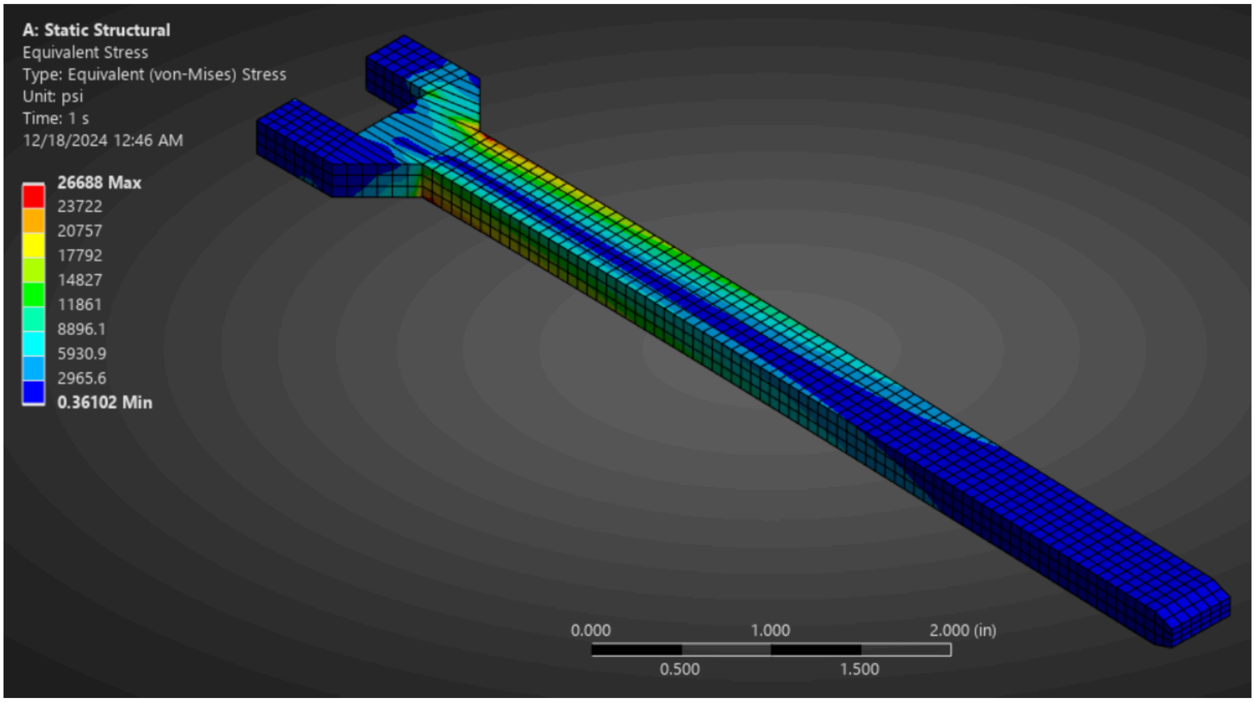

Figure 23. ANSYS Von Mises Stress for Version 3

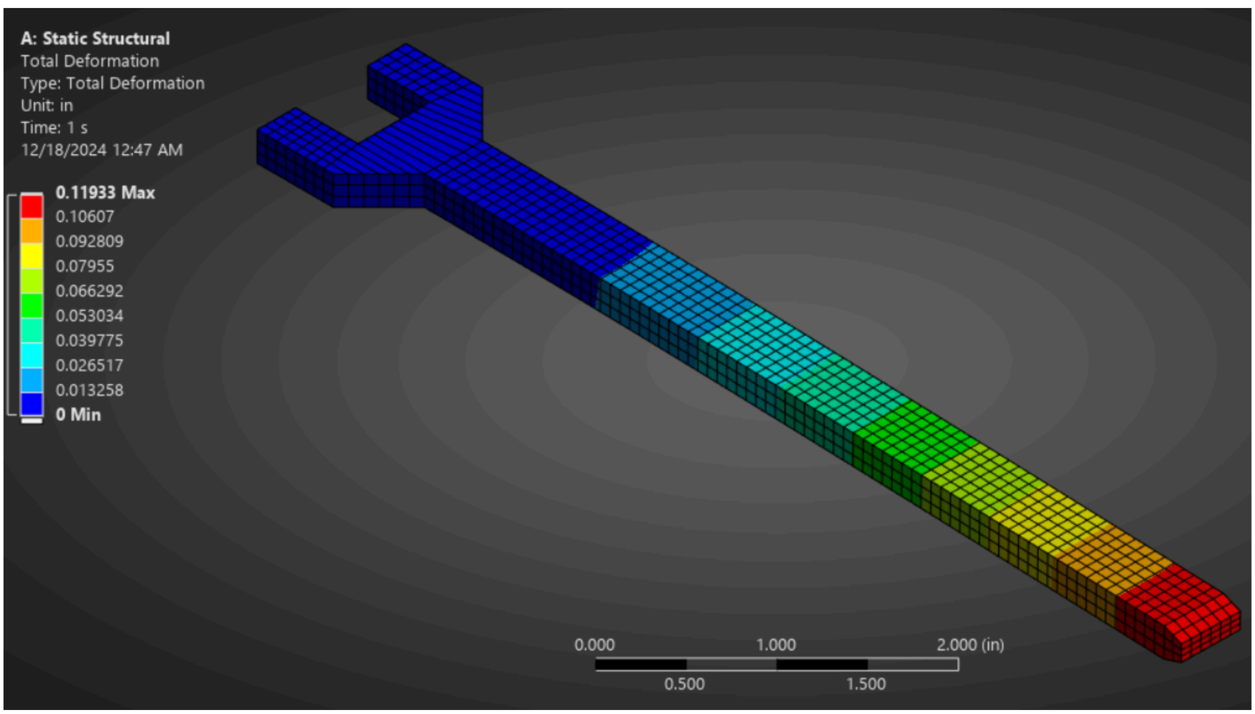

Figure 24. ANSYS Total Deformation for Version 3

The next change we felt was important to maximize the FOS/g value was to lengthen the head chamfer from a 60 degree angle to a 45 degree angle in order to lengthen it and reduce stress concentration at the joint. This resulted in a lower maximum Von-Mises stress of 26.7 ksi, and a slightly higher FOS of 1.49. On top of that, the FOS/g was 0.044, a slight increase from the last iteration despite the mass increasing marginally to 33.73g. While this iteration was minor, it provided significant insight on how to reduce stress concentration at the joint, and we built upon this knowledge in future iterations.

Version 4

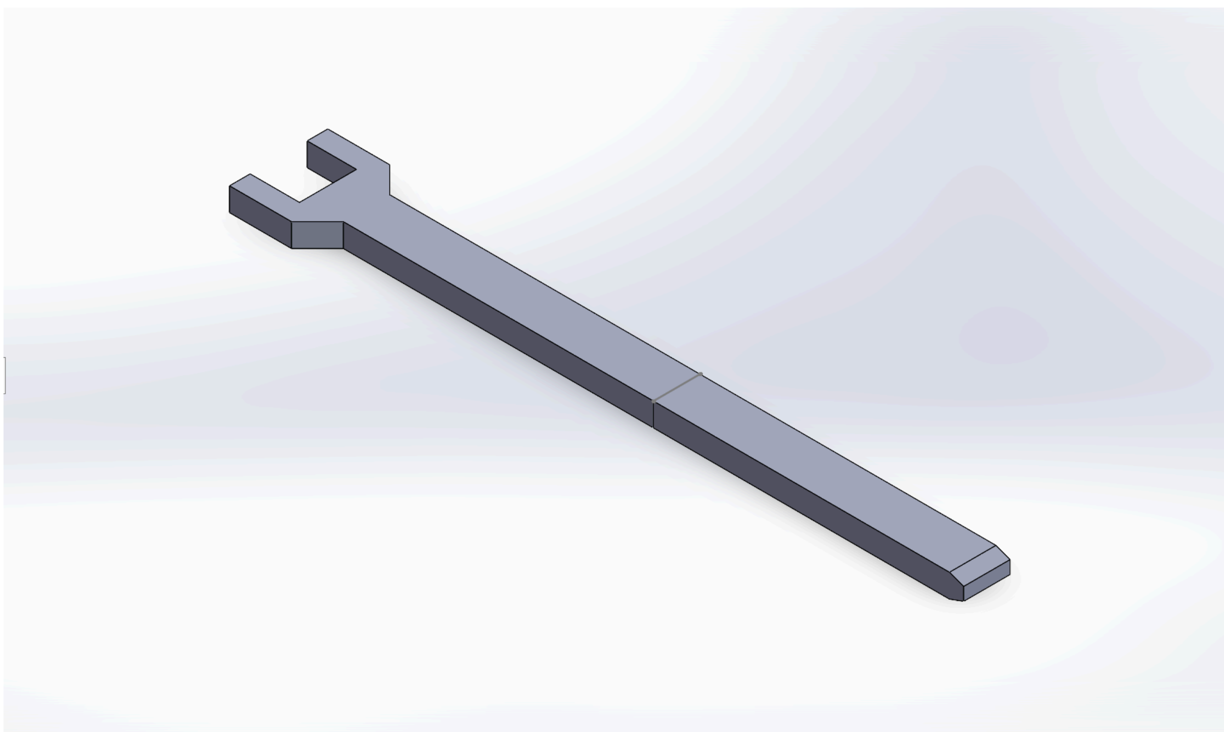

Figure 25. Solidworks for Version 4

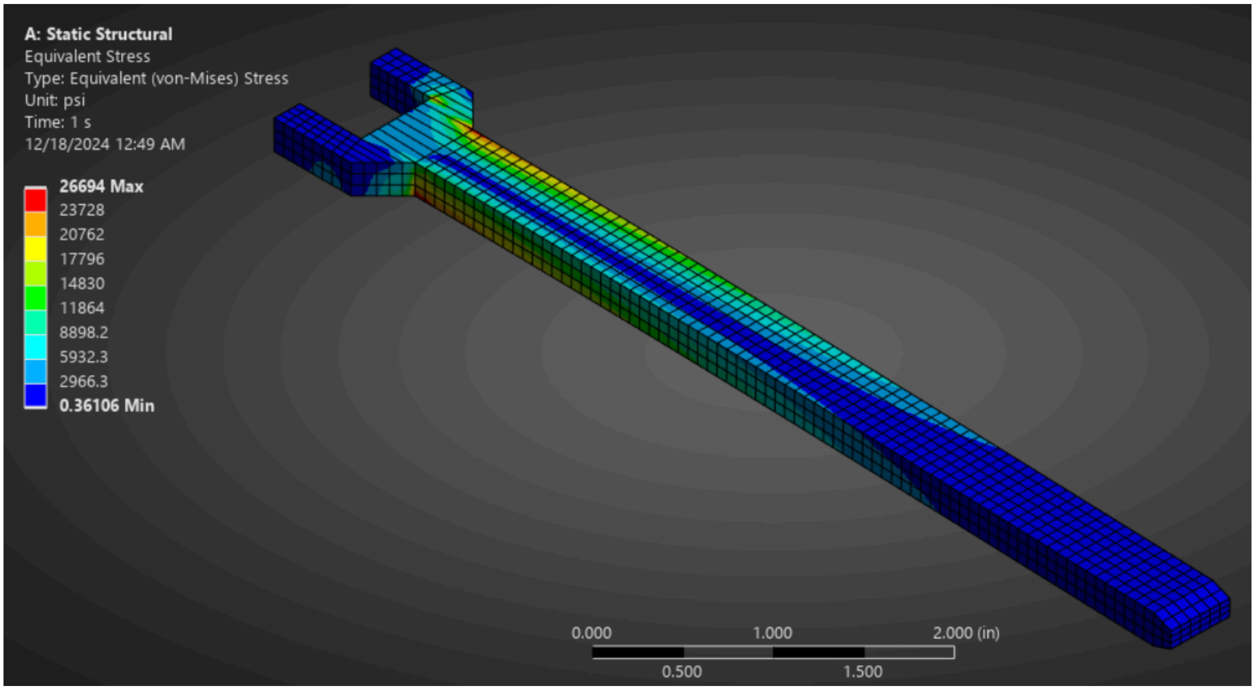

Figure 26. ANSYS Von Mises Stress for Version 4

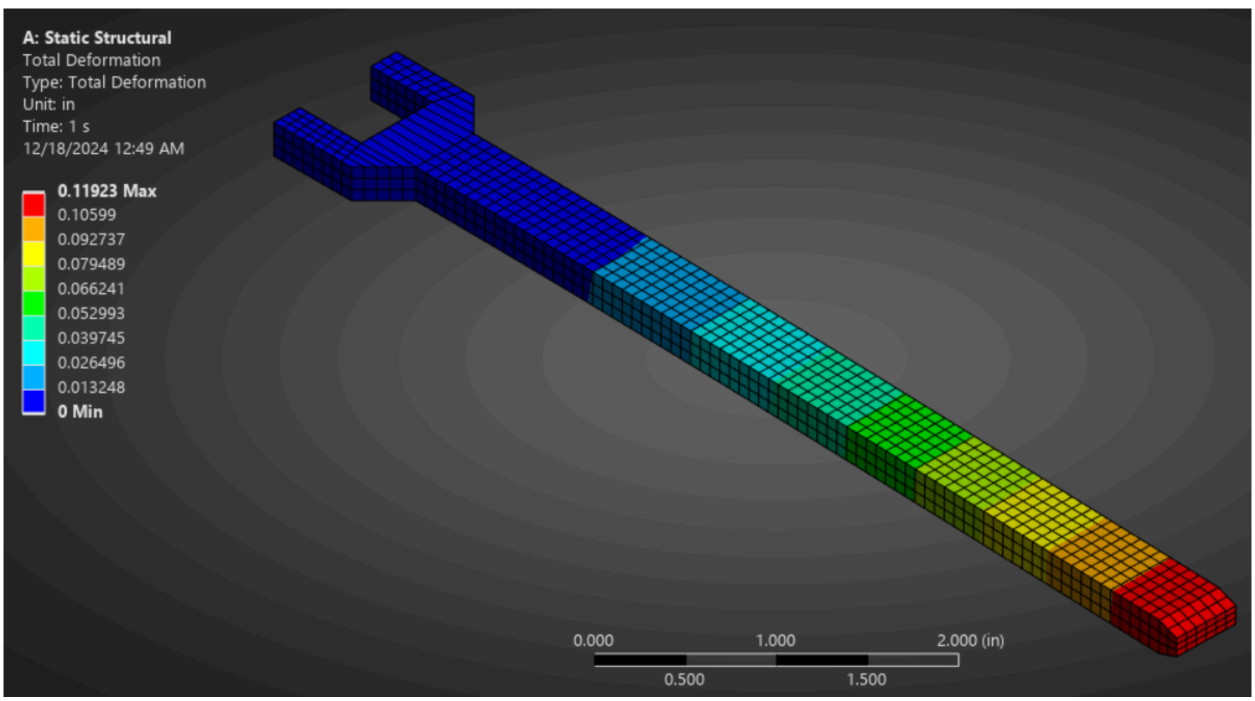

Figure 27. ANSYS Total Deformation for Version 4

As we saw the mass increase in the previous iteration, version four sought to counteract this change. When examining the results of prior simulations, we saw that the prongs of the wrench's jaw did not exhibit meaningful stress or deformation. Thus, we felt slimming down the prongs was a viable mode of eliminating weight, and we decreased their width from 0.3 inches to 0.25 inches.

This change barely affected the maximum stress, which was still approximately 26.7 ksi (figure 26), but it decreased the mass a fair amount, bringing it down to 31.54 grams. The FOS of this iteration is 1.499, similar to the previous, but the design shows its merit through the FOS/g value of 0.0475, the highest yet.

Version 5

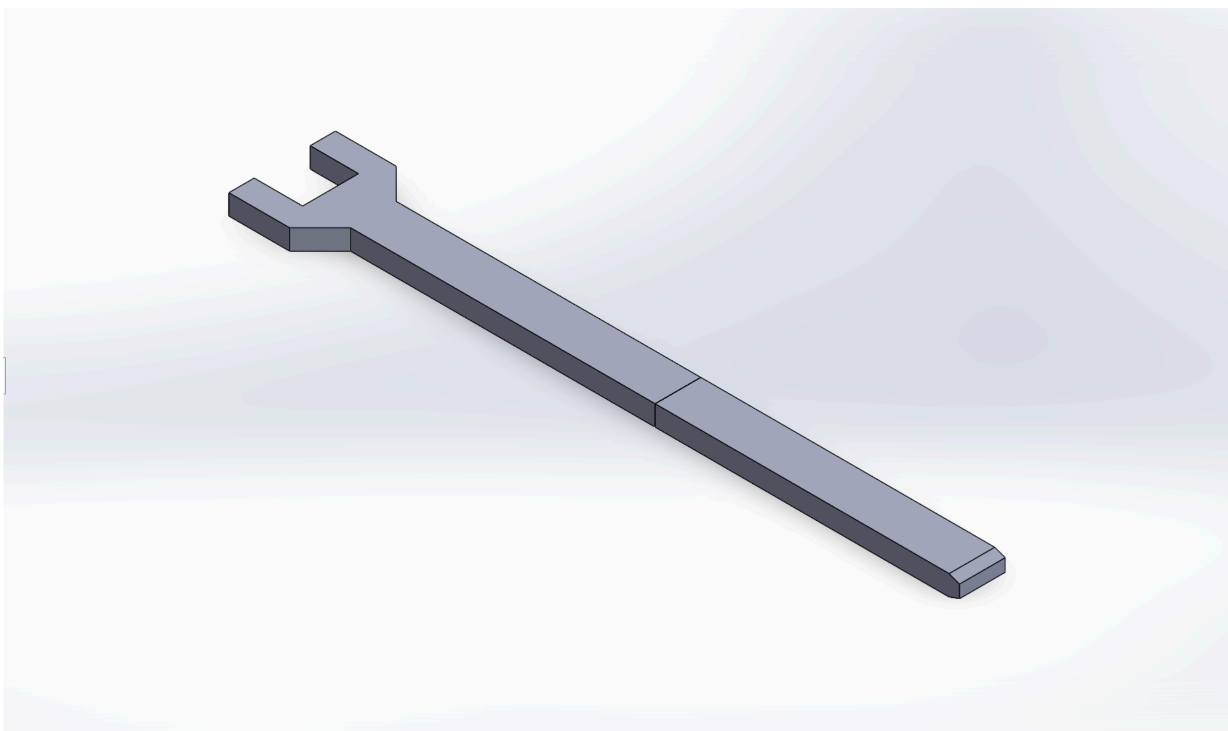

Figure 28. Solidworks for Version 5

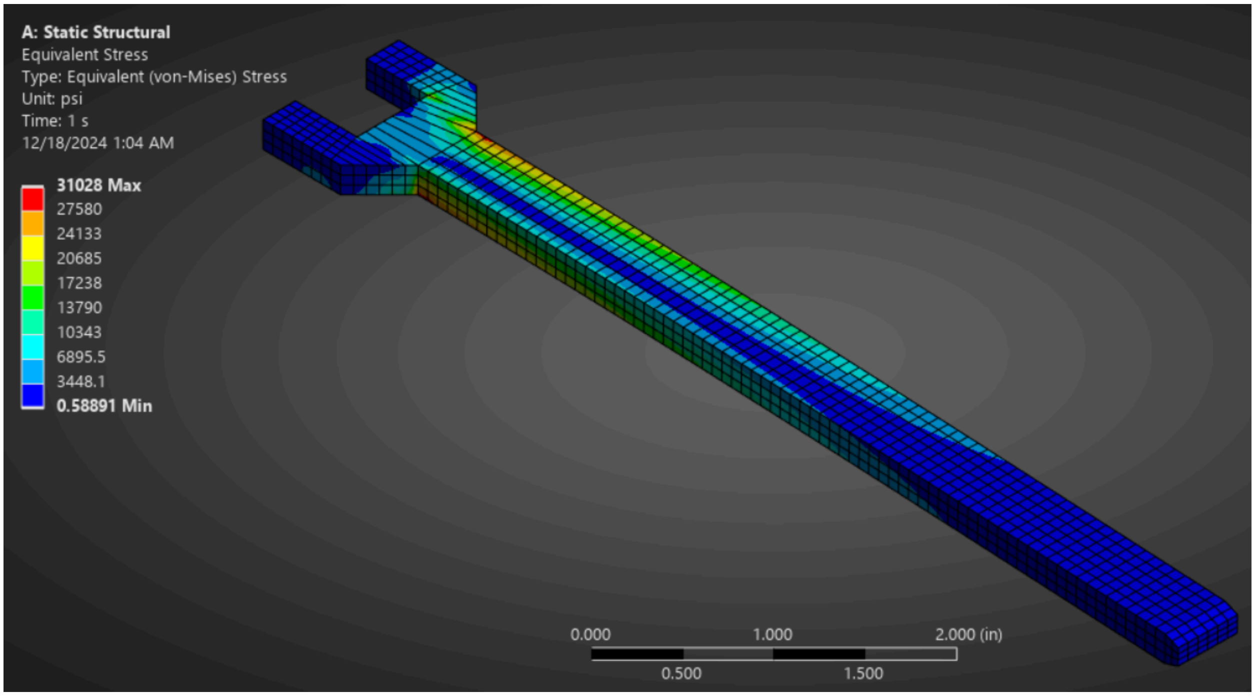

Figure 29. ANSYS Von Mises Stress for Version 5

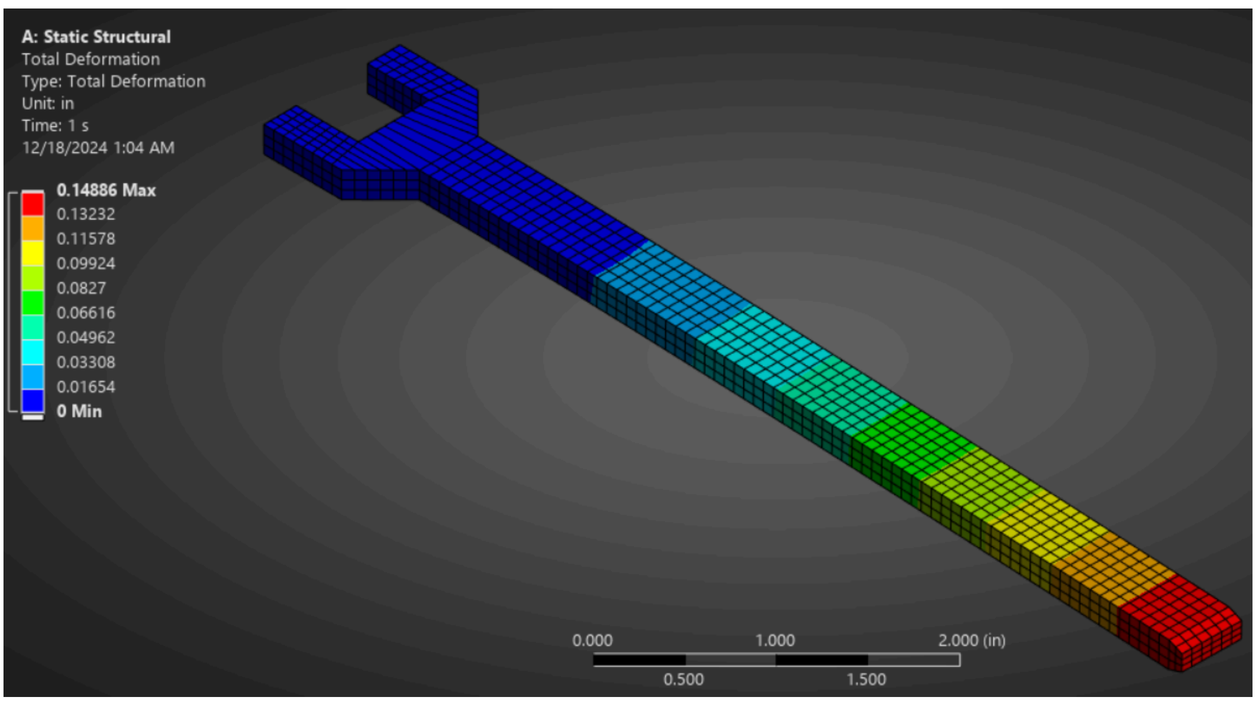

Figure 30. ANSYS Total Deformation for Version 5

From previous iterations, we deduced that arm width and height were not limiting factors in the strength of the wrench as altering them didn't significantly increase the von-Mises stress.

Therefore, in the fifth iteration, we further decreased these dimensions. We dropped the width of the arms down to .2 inches from .25 inches, and the height of the wrench down to .2 inches from .225 inches. These changes successfully brought our mass down to 29 grams but decreased the factor of safety from 1.499 to 1.29 and the FOS/g from 0.0475 to 0.044. The group felt this was a reasonable step, but may have overreached in attempting to reduce the wrench's thickness, so we decided to revert the height to .225 inches.

Version 6

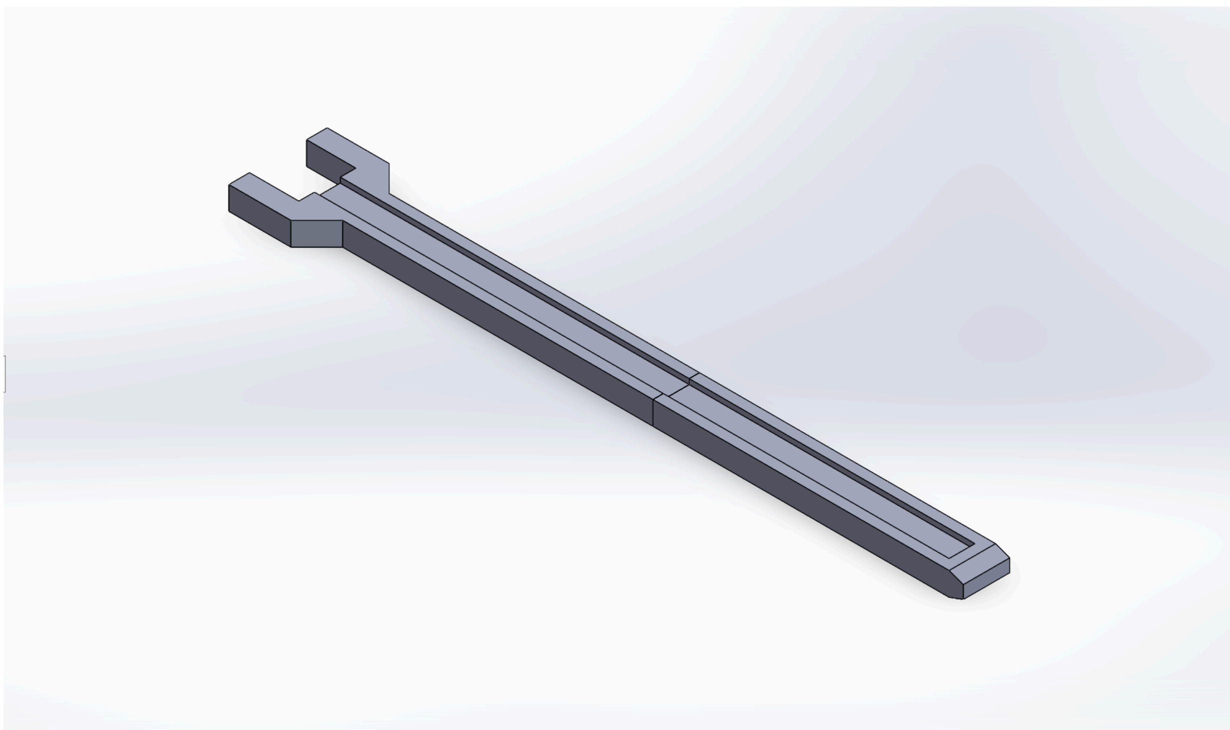

Figure 31. Solidworks for Version 6

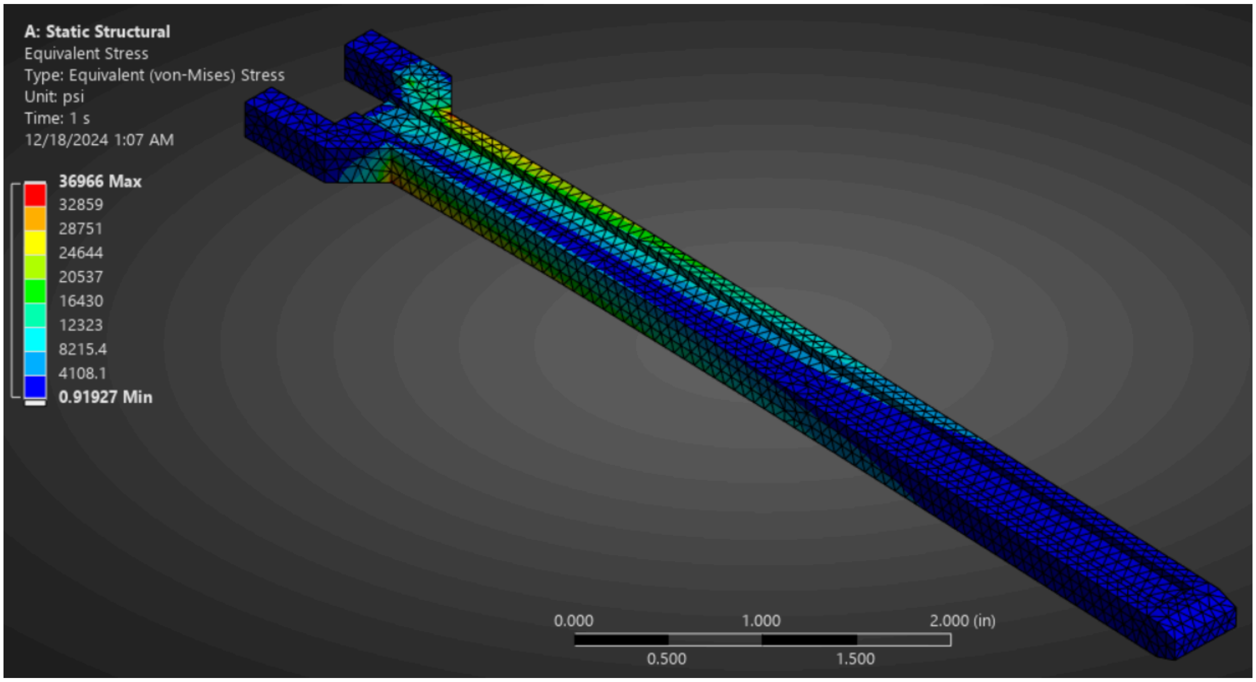

Figure 32. ANSYS Von Mises Stress for Version 6

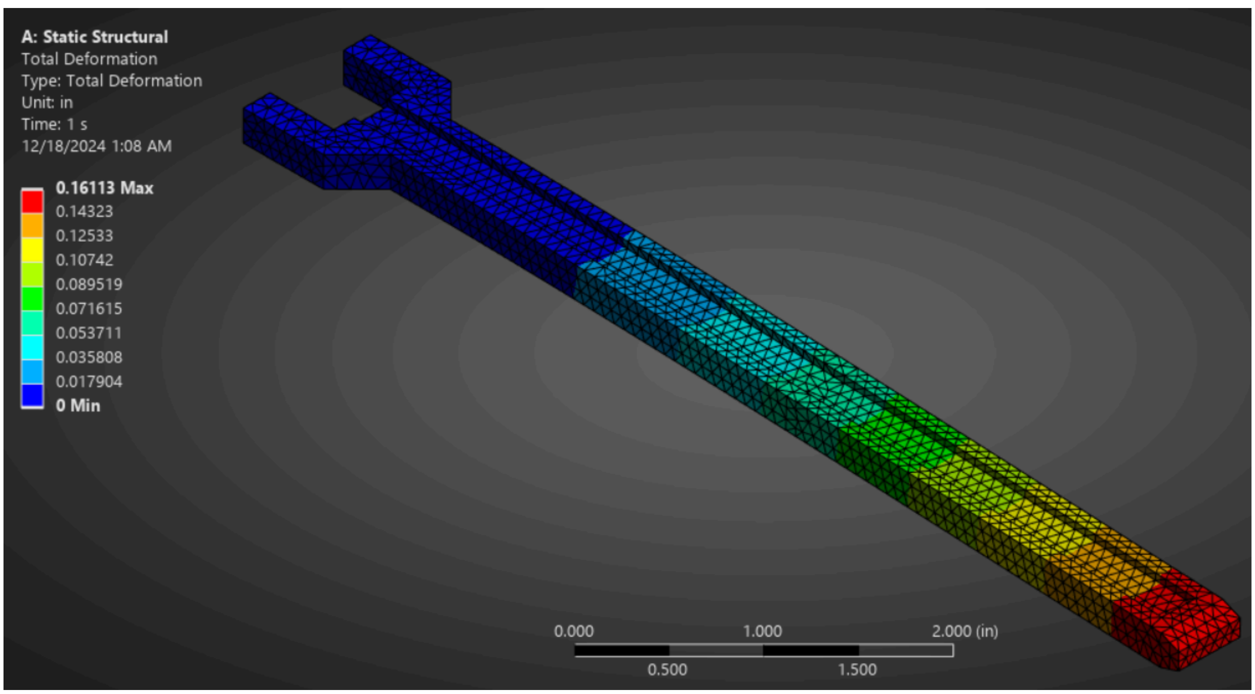

Figure 33. ANSYS Total Deformation for Version 6

After 5 iterations of incremental changes, we had effected only a slight improvement in the FOS-to-weight ratio; thus, we realized we had to take a bolder step to achieve a breakthrough. We implemented a large change in Version 6 to achieve our lightest wrench. After reverting the wrench's overall height to .225 inches, we decided to cut into the top and bottom faces of the handle to create an I-beam inspired shape. We hoped this shape would maintain strength in the direction the torque was applied while eliminating unnecessary volume. We believed the incidental force wouldn't pose a major issue, as previous results didn't illustrate significant deformation in that direction. This iteration of the wrench weighed in at an impressive 24.75 grams.

Unfortunately, the maximum stress spiked to around 37 ksi, leading to a factor of safety of 1.081, which was unacceptable. However, due to the dramatic weight reduction, the FOS/g was the same as the previous iteration, 0.044. This prevented us from reverting completely; rather, we sought to reduce the maximum stress while maintaining some of this lightweighting.

Version 7

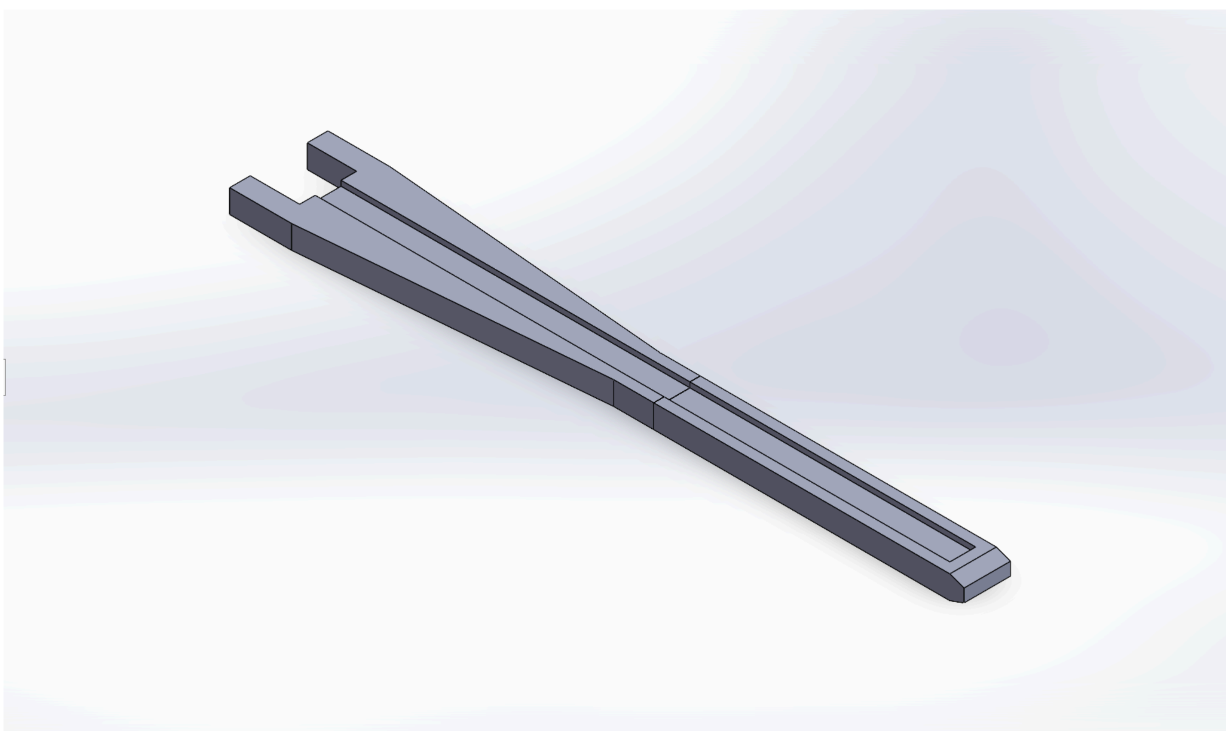

Figure 34. Solidworks for Version 7

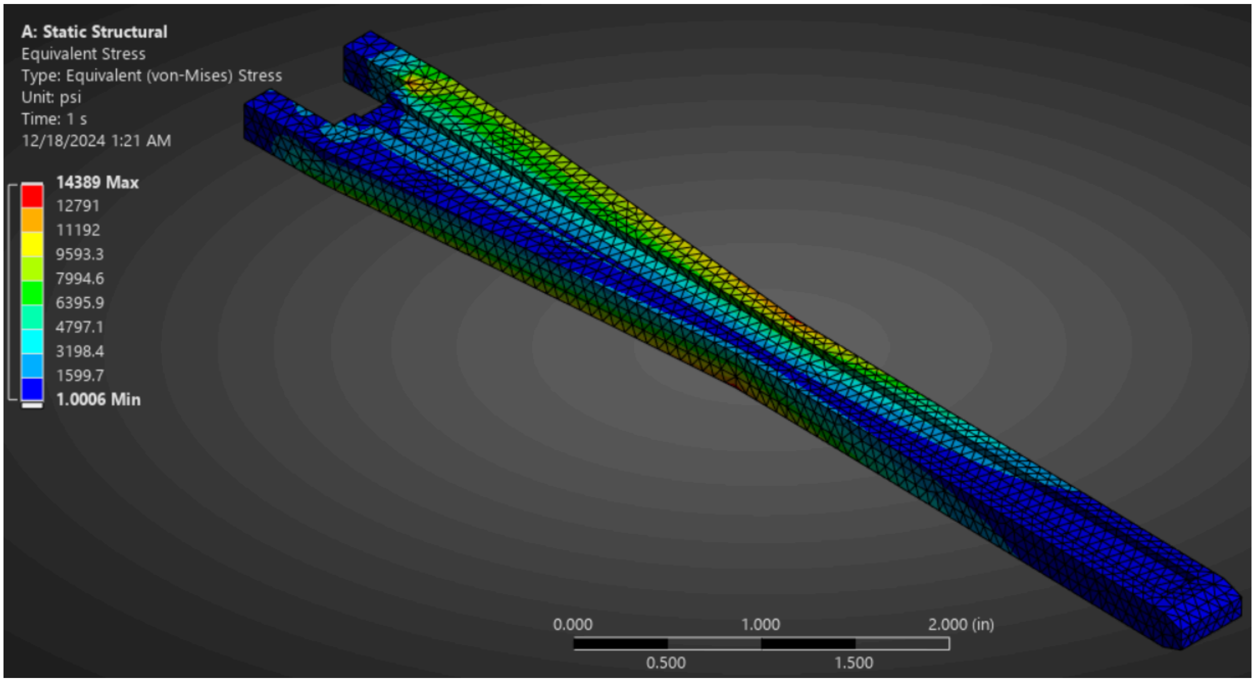

Figure 35. ANSYS Von Mises Stress for Version 7

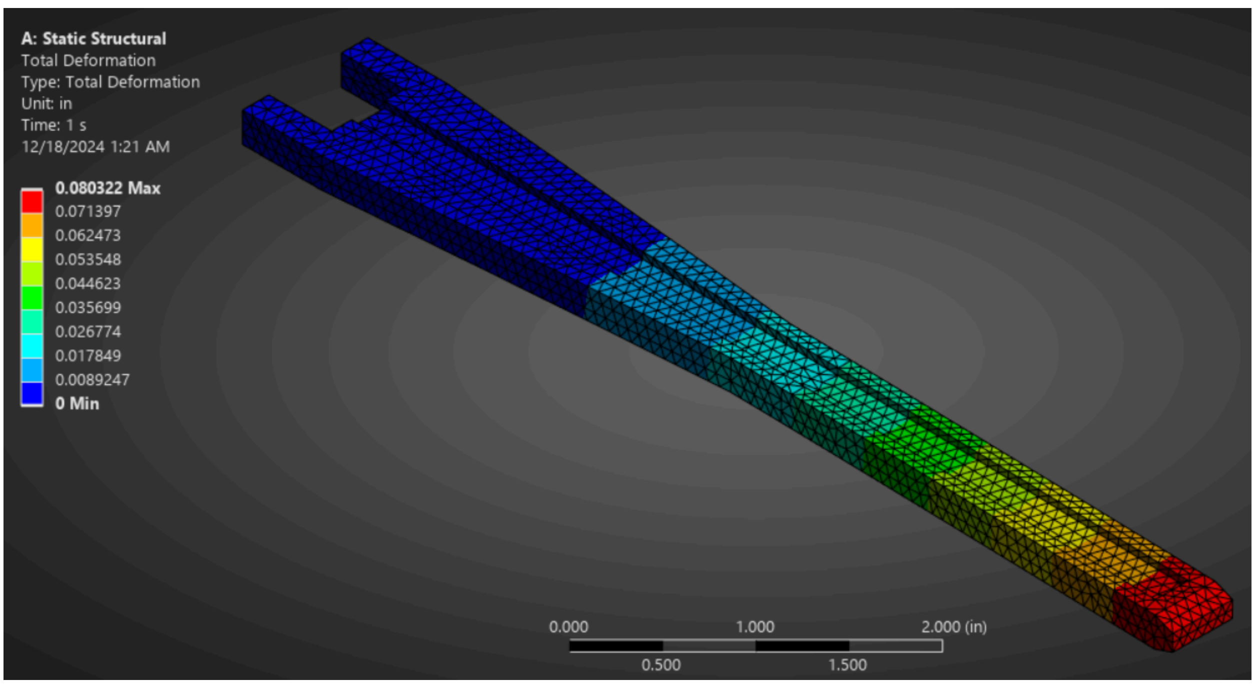

Figure 36. ANSYS Total Deformation for Version 7

In version 7, we switched gears from reducing weight to improving the factor of safety to surpass our requirement of ≥2, and we utilized the knowledge we gained from prior iterations to do so. We noticed that the stress was still highly concentrated at the joint between the chamfer and handle. In iteration 3, we decreased the angle of this chamfer to lengthen it, successfully reducing this concentration. Thus, we repeated this process and decreased the angle from 45 degrees to a mere 5 degrees. This alteration had a major impact; although it weighed significantly more at 31.27 grams, it also had the highest factor of safety of 2.778, well above our minimum of 2. As a result, it also had the highest FOS/g of 0.089, almost double the value of the next highest design.

Version 8 (Final Design)

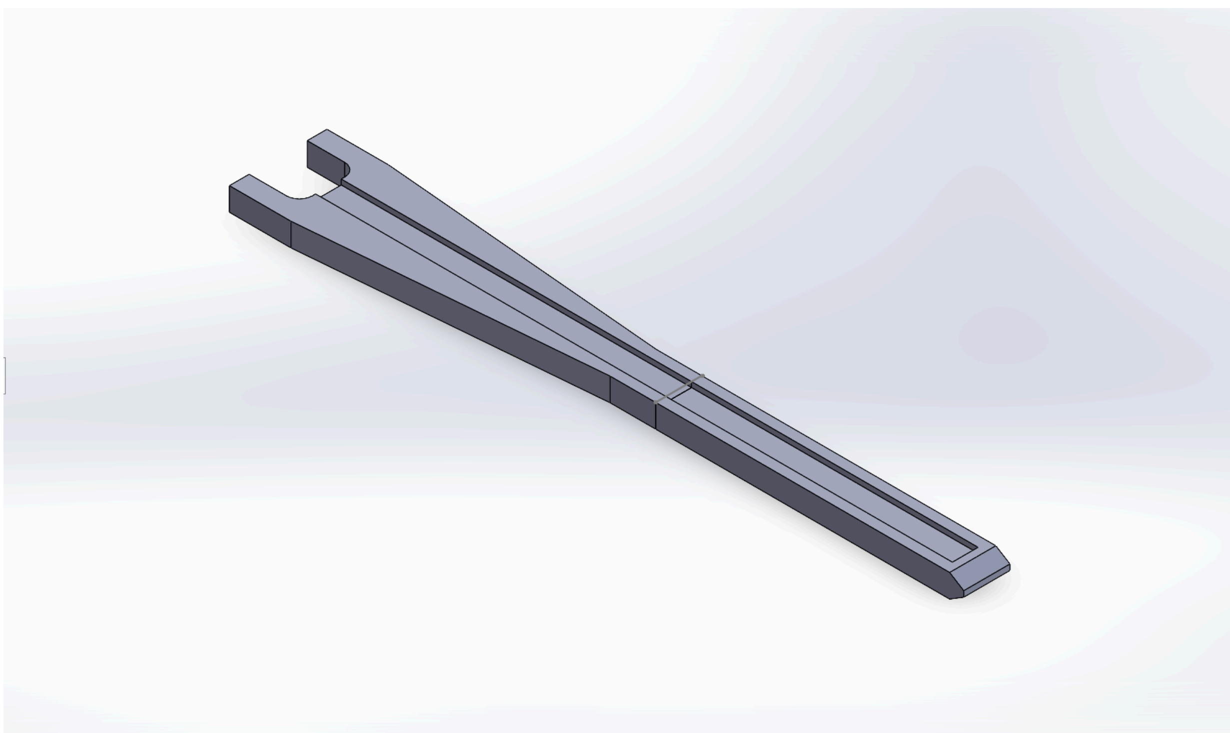

Figure 37. Solidworks for Version 8

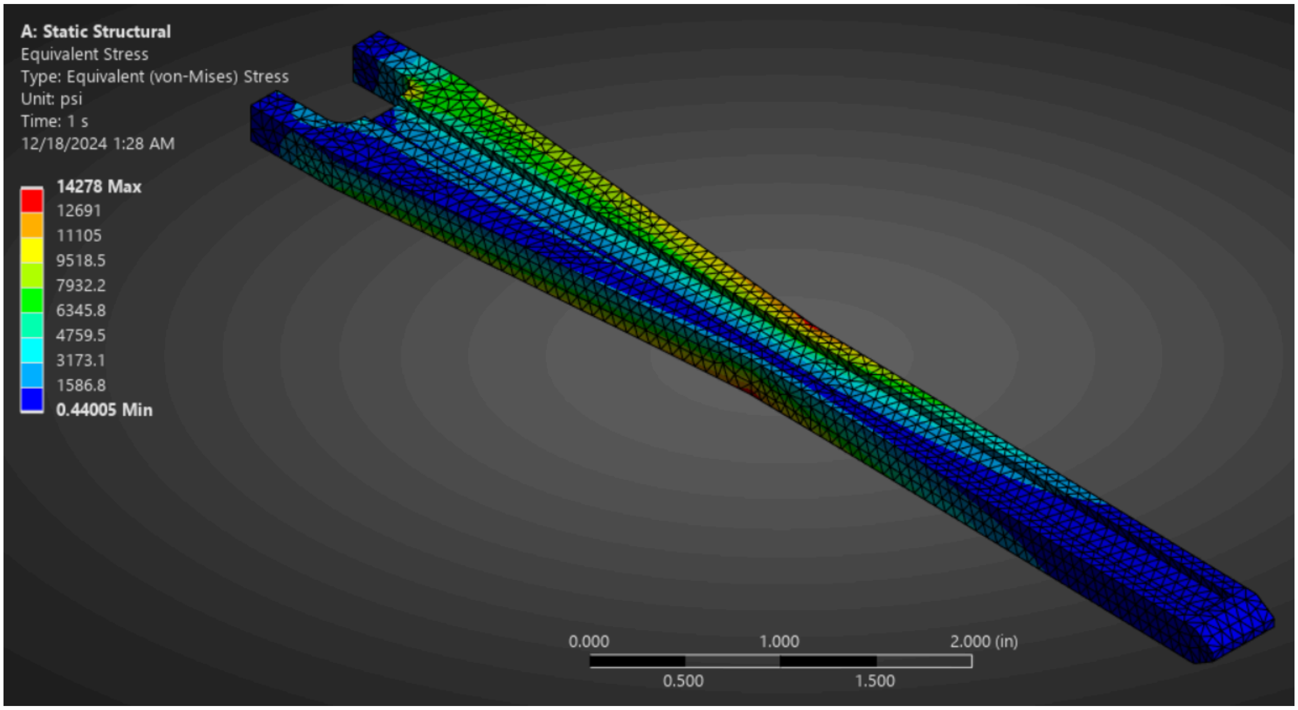

Figure 38. ANSYS Von Mises Stress for Version 8

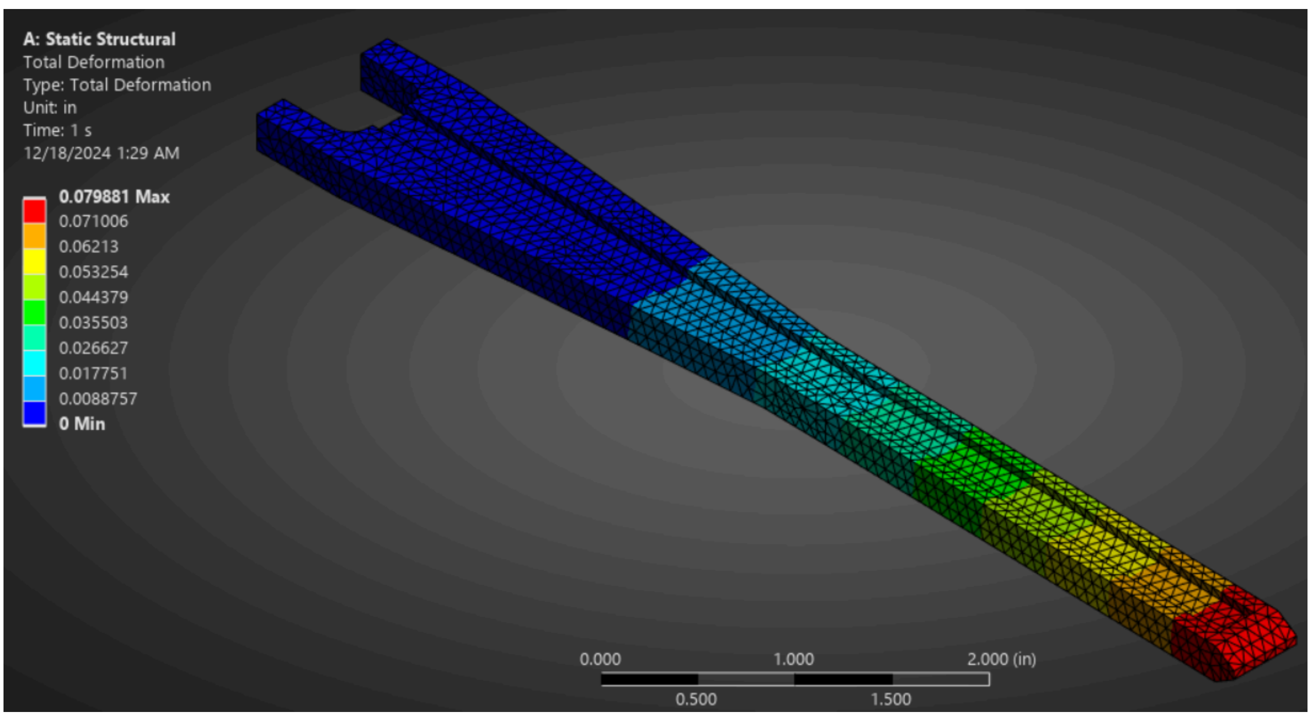

Figure 39. ANSYS Total Deformation for Version 8 Combined Loading Case

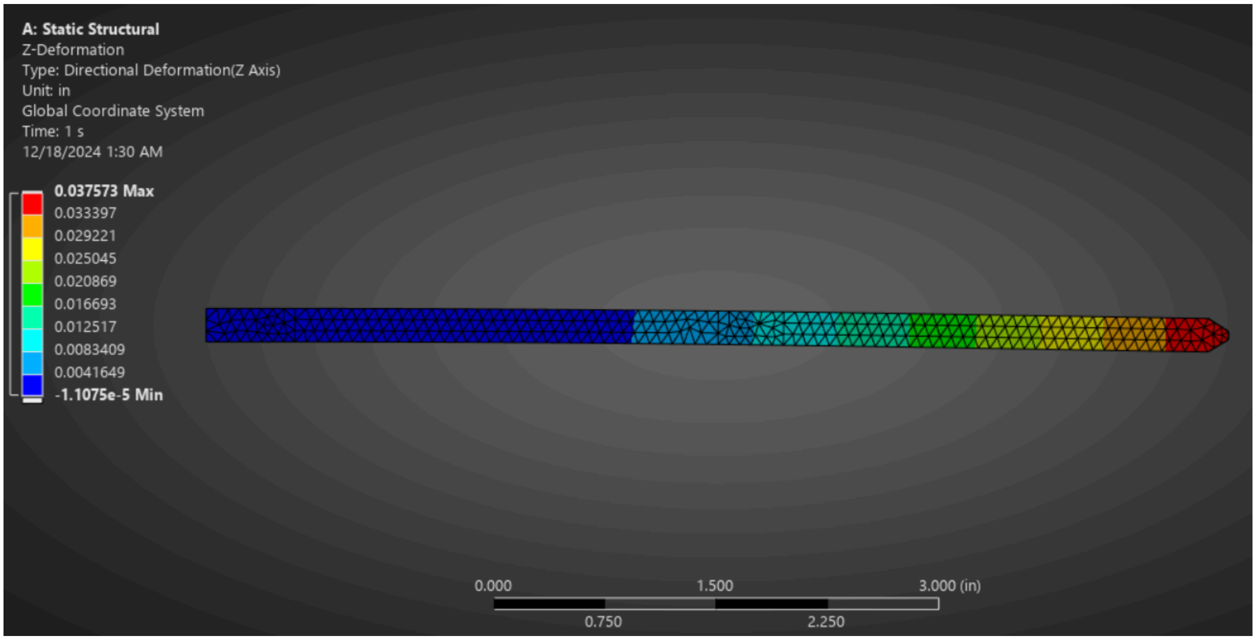

Figure 40. ANSYS Directional Deformation (Z axis) for Version 8 Combined Loading Case

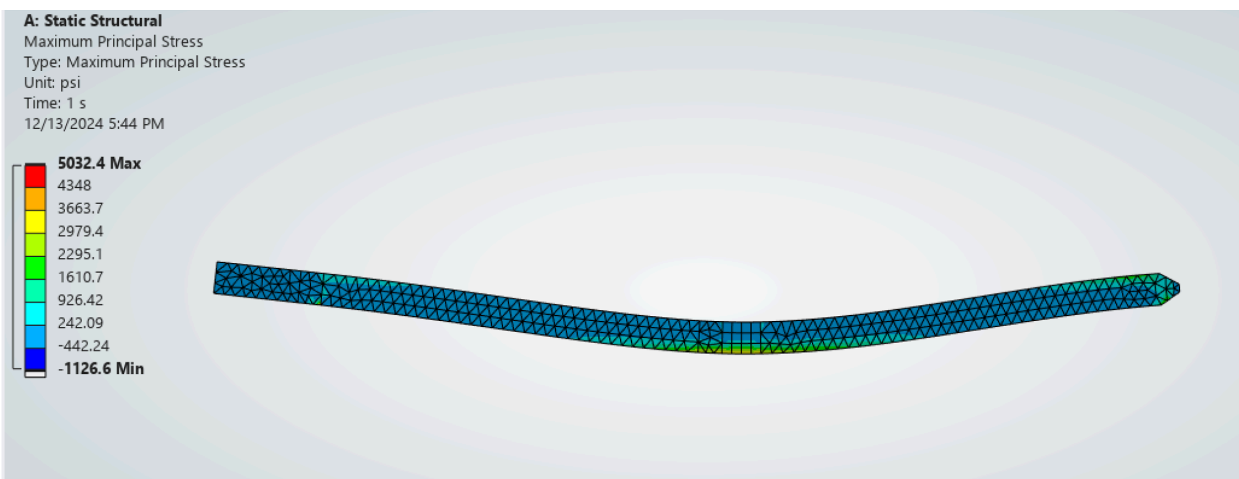

Figure 41. ANSYS Maximum Principal Stress for Version 8 Bending Test

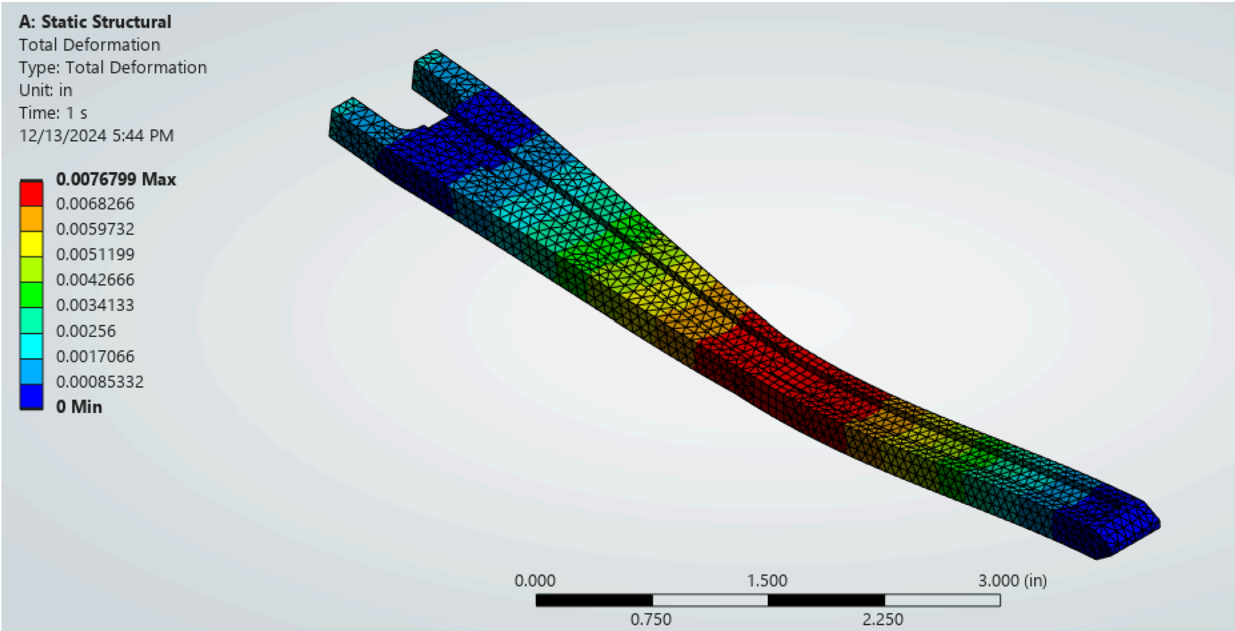

Figure 42. ANSYS Total Deformation for Version 8 Bending Test

Our final design, version 8, is quite similar to version 7. We were happy with the strength and weight of the wrench based on Ansys simulations, but we needed to improve machinability and adjust select features. One major change was increasing the size of the bolt cutout to provide clearance with the bolt, an oversight in previous designs. In addressing this oversight, the prongs of the wrench became even thinner (0.19in), though Ansys continued to show that this wouldn't be an issue. On top of that, we changed the flathead screwdriver from ⅛ inch to 3/50 inches so it would be compatible with a ¼ in slotted screw. Finally, we added the fillet which results from the circular end mill used in machining to the inside of the wrench's prongs to more accurately model the machined part.

Our final iteration had a maximum von Mises stress of 14.3ksi in the combined loading case, the lowest of all the iterations, which we believed to be due to the internal fillet reducing stress concentration. The mass calculated in Solidworks was 31.22 grams; as a result, the final FOS was 2.79, and the final FOS/g was 0.089. The total deformation was .08in, most of which occurred in the Z-direction (.04 inches) due to the torque of unscrewing the bolts, which aligned with our expectations. Even so, this is quite small and negligible to the human eye. Additionally, for the final iteration, we analyzed the 3-point bending test case, which yielded a maximum principle stress of only 5ksi and a minimal deformation of under 0.008in.

Test Results

To test the final wrench, the tool was subjected to two tests, functional and stiffness, as well as a weigh-in to see how successful our design was. The results of our weigh-in are shown below.

Figure 43. Final weight of wrench (32.7 grams)

As can be seen, our wrench successfully weighed in less than the constraint weight of 35.4g by 2.7g. This is slightly heavier than our predicted weight from our Solidworks evaluation, and this discrepancy is likely due to geometric differences resulting from machining tolerances.

For the functional test, the bolt was tightened to the required specification (170 in-lb) and our wrench successfully unscrewed the bolt although there was visible deformation of the wrench head (shown below) during the test. The wrench did not break, but the deformation may impact its future performance for unscrewing bolts

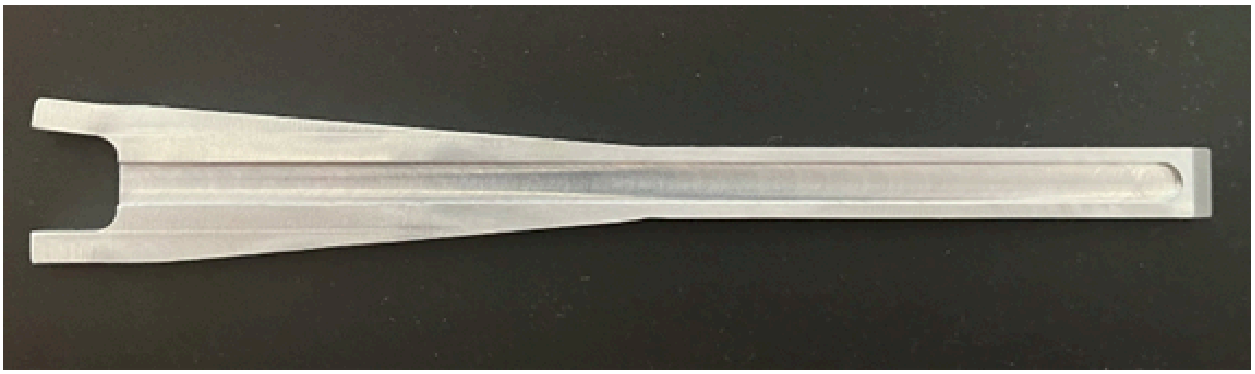

Figure 44. Wrench after functional and stiffness tests

For the three-point bending test, the wrench deformed very little, an amount indiscernible by the human eye. The machine used to perform the test, reported a deformation of 0.465 millimeters when the maximum load was applied as shown below. The deformation was nearly all elastic, although there was a permanent deformation of .032 millimeters.

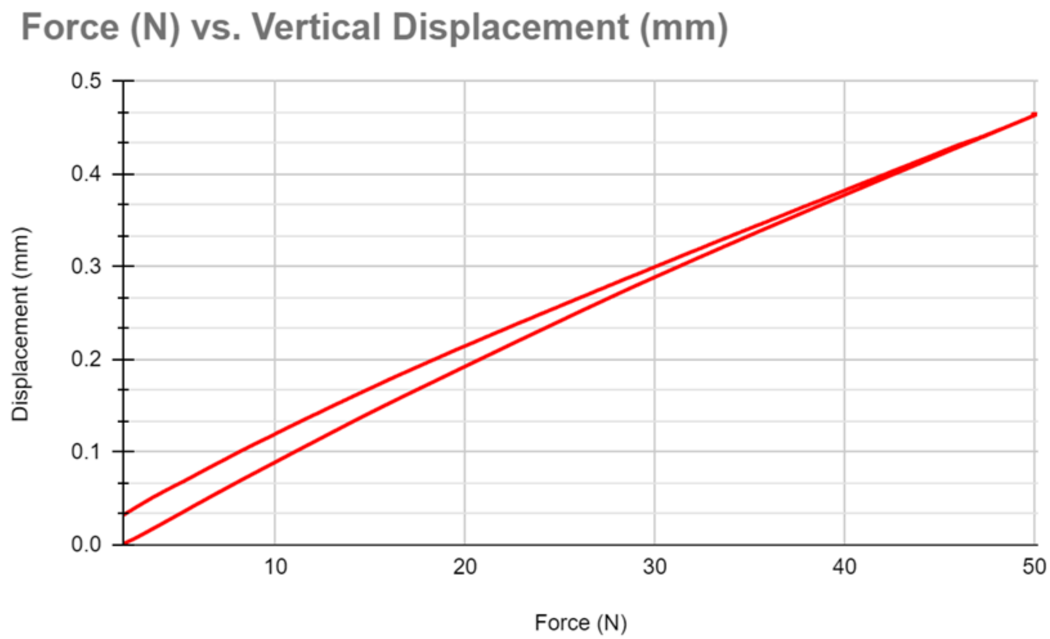

Figure 45. Force v.s. Displacement graph of wrench during stiffness test

The added functionality of a screwdriver was also tested independently of the torque and three-point bend test, and the attachment successfully unscrewed a ¼in diameter slotted screw that was hand-tightened an unspecified amount.

Discussion and Conclusion

Our group was tasked by Extra-Ultralight Biking LLC to design a lightweight, functional wrench for their target audience of long-distance bike riders. This wrench was to be made of AISI6061-T6 aluminum, weighing no more than 1.25 oz or 35.4 g. While staying under the weight limit, our designed wrench must have the strength to loosen a ⅜-16 bolt tightened to 170in-lb and withstand a 3-point bend test on the handle, which must be greater than or equal to 6 inches, with a 50N force directed downwards in the middle of the handle. To better assist the biker that will be using our product, the wrench was also required to incorporate an additional functionality other than the wrench itself. To combat continued use failure, our team has selected a factor of safety greater than or equal to 2 for the wrench.

For preliminary design, each member of the group created a wrench with various geometries and additional functionalities. These functionalities included a comb, ruler, bottle opener, and a flathead screwdriver. We chose to focus on the flathead screwdriver due to its simple machinability, light weight, and usefulness in real-world applications. Through critical by-hand calculations, our group was able to construct a base design to work off. This design, when run through simulations, slightly exceeded the weight requirements but performed well in terms of strength. The design was modified slightly through five iterations to lower the weight while keeping a similar level of strength. In our smaller iterations, we did not achieve large changes in stress or weight; however, they allowed us to identify patterns that we later used to inform our larger, more impactful changes. We noticed that changing the height had a lesser impact on FOS than changing the base and lengthening the chamfer reduced stress concentrations. This led to the major design changes in versions six and seven, with the final version, number 8, consisting of minor measurement changes to fit functionality restraints and reduce machining complexity.

After testing our wrench, we ended with a final weight of 32.7 grams, 2.7 grams below the limit. During the functional test, our wrench successfully loosened the bolt but was left with minor plastic deformation to one of the bolt prongs. During the stiffness test, the wrench completed the test with a maximum deflection of 0.465 millimeters and a plastic deformation of 0.032 millimeters. As shown by these tests, our wrench performed well in the stiffness test and poorly in the functionality test. To combat the bending of the wrench prongs, there are two avenues that we could pursue. First, we could simply increase the size of the prongs to increase their strength and resistance to failure. However, we believe it would be more beneficial to instead close off the prong section to have a bolt hole. By closing the prong section into one square cut-out for the desired bolt size, we can shift a majority of the stress from strictly the thin prongs to the combined closed strength of the cut-out. To account for the increase in weight due to adding this extra section of aluminum, we plan to reduce the size of the handle due to the wrench's exceptional performance in the stiffness test. This could be achieved by increasing the size of the handle indent or reducing the overall height of the handle. These changes should produce a stronger and more reliable wrench.

Appendix

Detailed Drawings

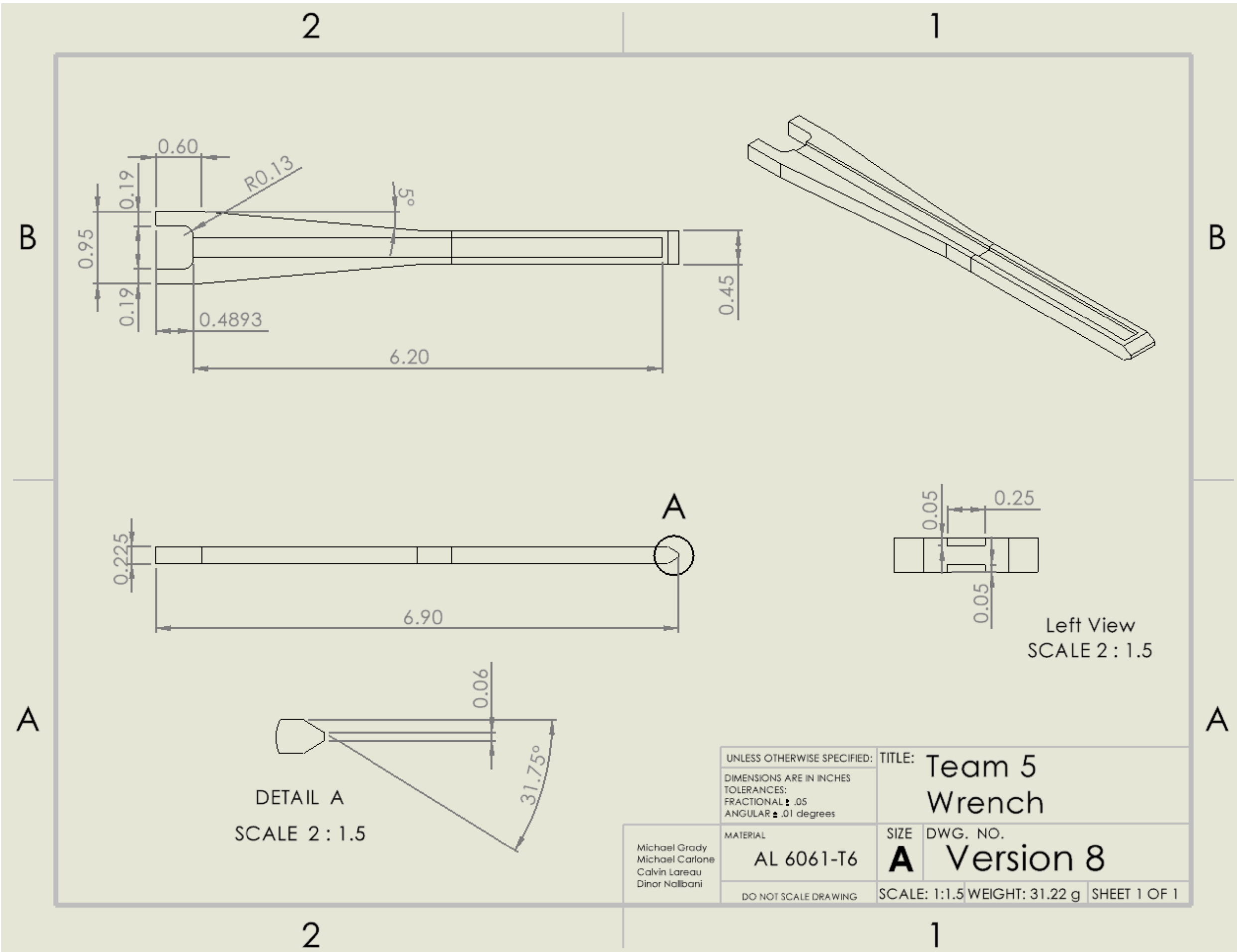

Figure 46. Solidworks Drawing of Final Design (Version 8)

Manufacturing Plan

Machining and manufacturability were at the forefront of our minds while designing this wrench. We wanted to ensure that the design could be easily created using basic metalworking machinery that can be found in a machine shop. The design started with a piece of AL6061-T6 stock that was 1.5 inches wide and 0.25 inches tall.

1. Cut stock to 7 ⅛ inches

We started by cutting the raw stock material to a length of 7 ⅛ inches. This is the initial step to prepare the material for further processing. We ensured that the cut is clean and precise, as this will form the basis for all subsequent adjustments and operations.

2. Smooth sharp edges with a deburring tool (This step is repeated after each cut)

After each cut, we used a deburring tool to smooth any sharp edges, burrs, or other related markings left from the cutting process. This is essential to avoid injury and ensure that the part can be handled safely. We repeated this step after each cut to maintain a clean finish on the part.

3. Trim the total height from 0.25 inches to 0.225 inches

For the rest of the steps, a milling machine was used with various-sized end mills. For these steps, we ensured that the machine was spinning at 1500 RPM when cutting the aluminum piece. We zeroed the Z axis of the end mill to the top of the “wrench”. Cuts were then made going back and forth, cutting .025 off the top until the entire piece measured .225 inches tall.

4. Trim total side length from 7 ⅛ inch to 6.9 inches

Similar to the previous step, we trimmed the length to be our desired 6.9 inches. We established a zero point at the end of the piece with an edge finder. In our zero calculation, we had to account for the diameter of the edge finder. Once zeroed, cuts were made going back and forth, cutting 0.35 inches off of the side in 0.05-inch Z increments.

5. Trim total width from 1.5 inches to 0.95 inches

Similar to the previous two steps, we trimmed the width to be our desired 0.95 inches. For this cut, we set our wrench up with the thinner side facing upwards instead of the larger side. We established a zero point at the end of the piece with an edge finder. In our zero calculation, we had to account for the diameter of the edge finder. Once zeroed, cuts were made going back and forth, cutting down the entire length in 0.1-inch Z increments.

6. Place zero point at the center of the end of a wrench

For the next two steps, we created a zero point at the center of the end of the piece. To achieve this, we had to zero the X direction, which was similar to the process done in the previous steps, remembering to account for the edge finder diameter. To zero the Y, we edge found the top edge and moved the edge finder to be positioned halfway down the piece, being 0.95/2 inches, once again accounting for the diameter of the tool.

7. Cut center indent out of wrench

With the zero point found in step 6, we cut 6.552 inches in the X direction 0.05 inches deep to create the middle vacancy. This cut stops just before the chamfering of the screwdriver head at the end of our wrench. We flipped the wrench over, zeroed again to ensure precision, and repeated this step.

8. Cut a hole in the wrench for the bolt to fit

For this step, we repeated step 6 to create a starting point for this cut. Once positioned in the center of the end of the wrench, we cut 0.351 inches into the wrench in the X direction, followed by 0.15 inches upwards in the Y direction, followed by cutting back left fully out of the wrench in the X direction. We repeated this process in 0.15-inch increments downwards in the Z direction until the entire cavity was cut out.

9. Trim handle section to create desired dimensions and set up for chamfer

Similar to the previous trimming sections, we edge found the X and Z directions to position the mill at the end of the wrench. We cut 3.30 inches in the X direction, ending with cutting up and down in the Y direction to clear any debris at the end. This was repeated in 0.05-inch increments in the Z direction 4 times for a total of 0.20-inch depth. This process was repeated for the other side of the wrench.

Figure 47. Wrench after Step 9 in the machining process

10. Create long 5-degree chamfers from head to handle

To create the angled chamfer, we angled the wrench in the clamps and used a digital protractor until it showed the 5-degree angle. We then cut straight across the length of the wrench until we cut through due to the 5-degree shift. This process was repeated with a decreasing Z position of 0.05 until we cut to the end of the handle. This process was repeated on the other side.

11. Cut flathead screwdriver chamfer

We used a similar process to step 10. We angled the wrench in the clamps and used a digital protractor until it showed a 31.75-degree angle. We used 31.80 because the protractor did not extend to the hundredth place for the decimals. Once we zeroed the Z on the tip of the edge of the wrench, we cut back and forth in the Y direction .015 inches down in the Z to create the proper-sized screwdriver head. We repeated this process on the other side to complete the wrench. All sharp edges and corners were deburred afterward.