BRONCO MOTORSPORTS

California State Polytechnic University, Pomona
Formula SAE

2023-2024 Design Binder

Rockers & Pushrods

Isaac Haynes

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Abstract

The Cal Poly Pomona Formula SAE team (Bronco Motorsports) is a student design team that designs and builds an open wheel, formula-style race car for international competitions held by the Society of Automotive Engineers (SAE). The purpose of this project was to use data collected through physical testing, analytical calculations, and computer simulation to design and manufacture the rockers and pushrods portion of our vehicle's suspension system. Topics involving generative design, fatigue analysis, and system lightweighting will be covered in great detail specific to our FSAE application. Additionally, the project will provide an in-depth overview of manufacturing techniques and on-track testing leading up to 2024 competition and during competition as well.

BM-24 Full Vehicle Objectives

After the suffering of a mechanical DNF in our 2023 Michigan Competition, reliability and serviceability have been a main concern for future competition. We struggled with cooling, shifting, and electrical problems leading up to the competition, and because of this, our testing time was reduced. This year we started with top-down vehicle goals regarding weight, power, downforce, and Tire Sensitivity analysis, to understand how certain changes can affect lap time for all 4 different dynamic events at our specific competition. With that being said, taking these lessons learning and further development of the car in terms of reliability, without sacrificing innovation towards performance has been a fine line; however, a line that has guided our design for the 2024 Car.

Purpose of Subsystem

The rockers and pushrods are integral components of the vehicle's suspension—*suspending the chassis off the ground*. With basic understanding of a coupled F/R pushrod suspension design, the rockers interface with the in-board mounted dampers and pushrods. The pushrods interface with the rockers and a-arm ends located in the outboard assembly. These components serve as the primary factor in achieving a desired motion ratio (Spring disp./Wheel disp.) and cohering to the team's overarching vehicle dynamic goals for the year via a unified set of maximum loading cases.

FSAE 2024 Rules

V.3 SUSPENSION AND STEERING

V.3.1 Suspension

V.3.1.1 The vehicle must have a fully operational suspension system with shock absorbers, front and rear, with usable minimum wheel travel of 50 mm, with a driver seated.

V.3.1.2 Officials may disqualify vehicles which do not represent a serious attempt at an operational suspension system, or which demonstrate handling inappropriate for an autocross circuit.

V.3.1.3 All suspension mounting points must be visible at Technical Inspection by direct view or by removing any covers.

V.3.1.4 Fasteners in the Suspension system are **Critical Fasteners**, see **T.8.2**

V.3.1.5 All spherical rod ends and spherical bearings on the suspension and steering must be one of:

- Mounted in double shear
- Captured by having a screw/bolt head or washer with an outside diameter that is larger than spherical bearing housing inside diameter.

T.8 FASTENERS

T.8.1 Critical Fasteners

A fastener (bolt, screw, pin, etc) used in a location specified in the applicable rule

T.8.2 Critical Fastener Requirements

T.8.2.1 Any Critical Fastener must meet, at minimum, one of the following:

a. SAE Grade 5

b. Metric Class 8.8

c. AN/MS Specifications

d. Equivalent to or better than above, as approved by a Rules Question or at Technical Inspection

T.8.2.2 All threaded Critical Fasteners must be one of the following:

- Hex head
- Hexagonal recessed drive (Socket Head Cap Screws or Allen screws/bolts)

T.8.2.3 All Critical Fasteners must be secured from unintentional loosening with **Positive Locking Mechanisms** see **T.8.3**

T.8.2.4 A minimum of two full threads must project from any lock nut.

T.8.2.5 Some Critical Fastener applications have additional requirements that are provided in the applicable section.

T.8.3 Positive Locking Mechanisms

T.8.3.1 Positive Locking Mechanisms are defined as those which:

a. Technical Inspectors / team members can see that the device/system is in place (visible).

b. Do not rely on the clamping force to apply the locking or anti vibration feature.

Meaning If the fastener begins to loosen, the locking device still prevents the fastener coming completely loose

T.8.3.2 Examples of acceptable Positive Locking Mechanisms include, but are not limited to:

a. Correctly installed safety wiring

b. Cotter pins

c. Nylon lock nuts (where temperature does not exceed 80°C)

d. Prevailing torque lock nuts

Lock washers, bolts with nylon patches and thread locking compounds (Loctite®), DO NOT meet the positive locking requirement.

T.8.4 Requirements for All Fasteners

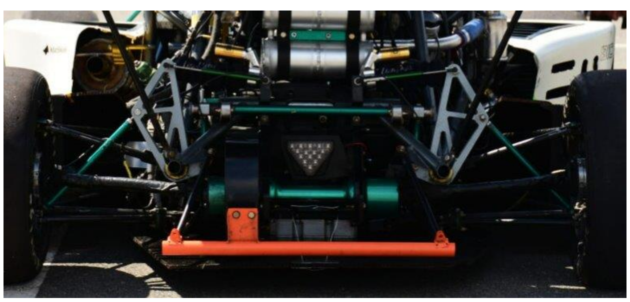
Adjustable tie rod ends must be constrained with a jam nut to prevent loosening.

Background & Theory

A basic design process of our suspension

A race car suspension holds the primary purpose of controlling the vehicle's time-transient tire contact patch while on track. By regulating the tire behavior through a well-tuned suspension, desired handling targets can be achieved for the vehicle and driver. If everything is related back to the tire, how can structural components such as rockers and pushrods be designed with this in mind? This question ought to be answered in two modes of engineering theory: Mechanical motion *without* forces (kinematics) and mechanical motion *with* forces (kinetics).

Within kinematics, we can dive into the points at which every suspension member is connected to. We call these Pick-up Points (PUPs for short). The 3D location of these points is crucial to how the car will behave on-track. For this year, we've decided on a coupled F/R suspension with in-board mounted dampers (in-board = connected directly to chassis). Since this design necessitates a rocker and pushrod, the PUPs for these components can be quantified into what is known as a *motion ratio* or spring displacement / wheel displacement. When designing the rocker PUPs, one needs to consider how their motion ratio may vary as the suspension actuates in compression or rebound. The engineer has the choice of a falling rate, steady-state, or rising rate motion ratio. Falling (wheel) rate rocker PUPs decrease leverage into compression, steady-state rocker PUPs maintain the same leverage (same motion ratio) though compression and rising (wheel) rate rocker PUPs increase leverage into compression. The overarching principle here is simply how much and the rate of applied leverage a.k.a mechanical advantage is allowed into the damper through actuation.



Within kinetics, obtaining accurate force values and compliance minimization is the name of the game. Properly understanding the transient forces each suspension member experiences is critical to designing reliable parts. Strain gauges, mathematical models, or a combination of both can greatly assist in this. Additionally, with the knowledge that quite literally, everything is a spring, attaining theoretical tire goals can become far less attainable with unaccounted structural compliance. Though there are many ways of reducing compliance, some of the most common are adhering to tolerances in manufacturing and setting adequate stiffness-to-weight targets when designing structural parts.

Here are the 6 basic formulas I utilized to perform all rocker and pushrod calculations.

Eq. 1 – Force

$$F = mass \cdot acceleration$$

Eq. 2 & 3 – Load Transfer

$$LongLT = F_{long} \cdot \left(\frac{CG_{Height}}{Wheelbase} \right)$$

$$LatLT = F_{lat} \cdot \left(\frac{CG_{Height}}{Trackwidth} \right)$$

Eq. 4,5,6 – Damping Force

$$F = kx + cv$$

$$WR = SR \cdot (MR)^2$$

$$\zeta = \frac{c}{2\sqrt{k \cdot m}}$$

$k = springstiffness$

$x = springdisplacement$

$c = dampingcoefficient$

$v = dampervelocity$

$WR = wheelrate$

$SR = springrate$

$\zeta = dampingratio$

$k = wheelrate$

$m = F\&Rsprungmass$

$MR = motionratio$

Eq. 7,8 – Pushrod Stress and Strain

$$\sigma = \frac{F}{A} = E \cdot \varepsilon$$

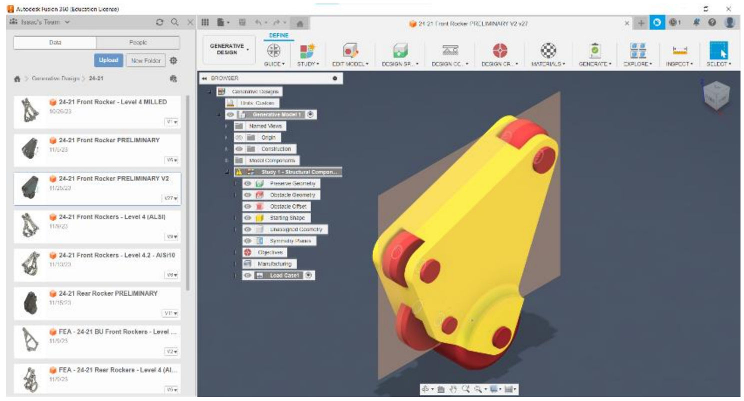
$$F = axialforce$$

$$A = crosssectionalarea$$

$$E = Young'sModulus$$

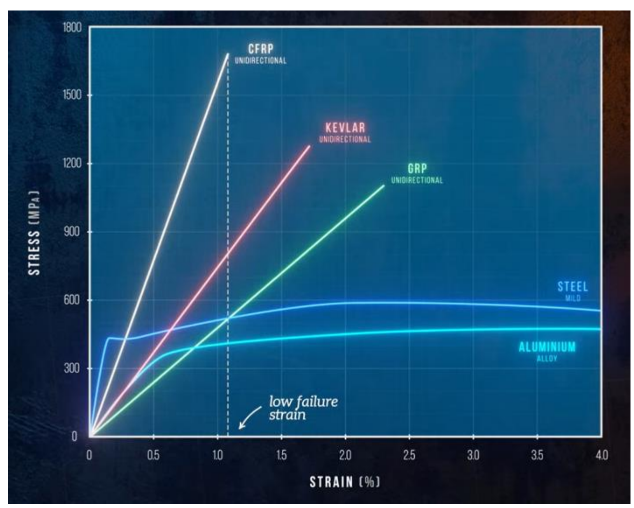
$$\varepsilon = strain = \frac{\Delta L}{L_{initial}}$$

How generative design works



One of the cornerstones of this year's rocker design is the usage of an up-and-coming AI technology known as *generative design.* This tool is used to iteratively generate stress-optimized 3D geometries based on several different constraints and boundary conditions. Bluntly, material is only grown (not removed) where it is needed. Within the Autodesk Fusion360 Generative Design Extension, the entire design process begins with a starting shape, preserve geometry, and obstacle geometry. They represent exactly what the names suggest. Once these entities are known in the model, the engineer inputs the appropriate loading conditions and the AI-based algorithm runs to remove material from the starting shape to optimally output a final product while abiding by a target FOS, displacement, and weight. The product is an organic, bone-like geometry optimal for the given static loading condition. This new-age tool allows for enhanced and creative design solutions for almost any type of structural part. Nevertheless, this goes to say that the AI outputs are only as good as the engineers' inputs. This is where the bulk of my analysis was catered. This video outlines my workflow quite well.

The basics on carbon fiber composites



As a technology ubiquitous in all of motorsport, carbon fiber composites are also implemented into our vehicle's suspension system for their ability to remain amazingly stiff at a fraction of the weight of its steel counterpart. Carbon fiber is a type of composite material made of a fiber (woven carbon fiber) and matrix (epoxy resin). Separate, these components are very weak. Together, they allow for an incredibly strong and stiff material depending on the desired weave, resin, and loading conditions. For our application of an FSAE suspension design, we require our members to have adequate strength and stiffness in tension, compression, and bending. These design parameters will dictate our chosen carbon weave for suspension tubing. Finally, the strength-to-weight ratio of carbon fiber is perhaps the most important factor into why it is extremely popular in motorsports. While it may not be quite as stiff or strong than steel, the strength-to-weight benefits often outweigh the value of a slightly more elastic final part. This happens to be the case for our carbon fiber pushrods.

Design Procedure

Now that we have the tools, let's put them to good use.

Step 1. – Analyze past on-track lin-pot data from previous competitions

Taking a look at Michigan 2023 Endurance data and other testing outings' lin-pot data, we can determine a max damper displacement for the run. This will directly correlate to the reaction force in the rocker at the damper. I'll detail this calculation later on.

Step 2. – Choose your pick-up points (PUPs) and motion ratio based on desired kinematic curves.

BM-24 Pickup Point Locations (24-11)				
Front (in)				
		x	y	z
Upper A-arm	Outboard	0.237	21.609	11.492
Upper A-arm	Aft	5.8387	10.68	8.387
Upper A-arm	Fore	-1.592	10.874	8.6757
Lower A-Arm	Outboard	-0.237	22	5.05
Lower A-Arm	Aft	7.49403	7.4461	4.1813
Lower A-Arm	Fore	-2.7089	7.52055	4.0419
Pushrod	Inboard	0.41	10.9536	23.79448
Pushrod	Outboard	0.41	20.32661	11.84186
Toelink	Inboard	-0.242915	8.3875	5.90157
Toelink	Outboard	-3.2536	22.5925	8.783831
Damper	Inboard	0.41	1.2345	22.83833
Damper	Outboard	0.41	7.50129	26.118
Front ARB				
Droplink	Rocker	0.41	10.1991	22.891
Droplink	Blade	0.41	10.1991	20.96
Blade	Tube	3.995	10.1991	20.96
Tube	Chassis	3.995	7.2423	20.96
Rocker Pivot		0.41	8.75202	22.10892
WC		0	24.4929	7.989

I was not the one who chose BM-24 rocker PUPs. Alex Block chose the revised points the year before when we were still racing BM22 2.0. The 2024 PUPs were basically designed in 2023 before I took on the subsystem. Knowing this going into the year, I decided to take more of an analysis and prototyping approach to the subsystem. Using the existing PUPs, I developed free body diagrams to take a look at how the reaction forces in the rocker vary per point. The same goes for motion ratio. Kept it at 1.2 front, 1.0 rear with a lack of reason to change.

Generally speaking, a rising rate stiffens up the suspension in high-speed situations while also providing low speed benefits which maximize damper travel thus shock absorption. This is the primary reason to run this setup. It allows for a wide range of high and low speed stability purely from a motion ratio standpoint.

Step 3. – Derive reaction forces in all PUPs for all required on-track loading cases.

With 15 different loading conditions needing proper analysis, I created this table to compile all of the rocker forces in terms of lateral, longitudinal, and vertical degrees of freedom. The meat here really is the vertical force logic that allows the calculation of reaction forces based on a combination of known/targeted values for corner weights, Long LT, Lat LT, and Bump G's (Fz). Make sure to be 100% confident in your inputs before populating the table. I've attached this excel file in the 2024 OneDrive.

Max Loading Forces - 2024 Rockers & Push-Rods

23-24 Max Loading Cases

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Scenario	Lat	Long	Vertical
Vertical Bump	0	0	3
Acceleration	0	1.2	1
Braking	0	-1.5	1
Cornering	2.4	0	1
Cornering + Acceleration	2	0.8	1
Cornering + Acceleration + Bump	0.7	0.5	1.2

*all values in Gs

Inputs

Vehicle Weight w/ Driver (lbf)	604	
CG Height (in)	12.3	
Wheelbase (in)	60.24	
	Front	Rear
Weight Distribution (%)	47	53
Static Corner Weight (lbf)	141.94	160.06
Trackwidth (in)	49.02	47.99

Scenario (Front corner, units: lbf)	Lat. LT	Long. LT	Vertical	Vertical Force Logic (F_z)
Vertical Bump	0	0	425.82	Front Corner Weight * 3G
Acceleration	0	-69.18	72.76	Front Corner Weight - Long. LT
Braking	0	92.41	234.35	Front Corner Weight + Long. LT
Cornering	170.09	0	312.03	Front Corner Weight + Lat. LT
Cornering + Acceleration	132.66	-46.12	228.48	Front Corner Weight + Lat. LT - Long. LT
Cornering + Acceleration + Bump	46.43	-28.83	245.588	(Front Corner Weight*Bump Gs) + Lat. LT - Long. LT

Scenario (Rear corner, units: lbf)	Lat. LT	Long. LT	Vertical	Vertical Force Logic (F_z)
Vertical Bump	0	0	480.18	Rear Corner Weight * 3G
Acceleration	0	69.18	229.24	Rear Corner Weight + Long. LT
Braking	0	-92.41	67.65	Rear Corner Weight - Long. LT
Cornering	185.47	0	345.53	Rear Corner Weight + Lat. LT
Cornering + Acceleration	144.65	46.12	350.83	Rear Corner Weight + Lat. LT + Long. LT
Cornering + Acceleration + Bump	50.63	28.83	271.532	(Rear Corner Weight*Bump Gs) + Lat. LT + Long. LT

Final Rocker Forces (lbf)	Front	Rear	Force Logic
Push-rod	425.82	480.18	Max F_z seen for a single given scenario -> Vertical Bump
Damper	371.76	474.08	F=vc+kx
ARB	291.61	468.071	F=kx (hard)
(F/R Corner Weight*3Gs) + Lat/Lo	595.91	665.65	

Notes:
All cornering scenarios involve outside tire forces
MAXs
Update

Longitudinal Load Transfer

Inputs

Vehicle weight: 604 lbs 273.97 kg

% weight dist: 47% F ⇒ 283.88 lbs 128.77 kg

53% R 320.12 lbs 137.04 kg

CG Height: 12.30 in 0.312 m

Wheelbase: 60.24 in 1.530 m

23-24 Max Loading Cases

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Scenario	Lat	Long	Vertical
Vertical Bump	0	0	3
~~Acceleration~~	0	1.2	1
~~Braking~~	0	-1.5	1
Cornering	2.4	0	1
~~Cornering + Acceleration~~	2	0.8	1
~~Cornering + Acceleration + Bump~~	0.7	0.5	1.2

*all values in Gs

Theory

*Derived from Optimum G slides

long. force = mass · long. accel

$$LLT = \text{long. force} \cdot \left(\frac{\text{CG Height}}{\text{wheelbase}}\right)$$

Acceleration [1.2 G]

1.2 G's = 11.016 m/s^2

long Force = 273.97 (11.016)

= 3018.05 N [total rear]

$LLT = (3018.05)\left(\frac{0.312}{1.530}\right)$

= 615.45 N $\left(\frac{1}{2}\right)$ [total rear]

= 307.72 N [each rear tire]

LLT = 69.18 lb_f [each rear tire]

Braking [-1.5 G]

-1.5 G's = -14.715 m/s^2

long Force = 273.97(-14.715)

= -4031.47 N [total front]

$LLT = (-4031.47)\left(\frac{0.312}{1.530}\right)$

= -822.10 N $\left(\frac{1}{2}\right)$ [total front]

= -411.05 N [each front tire]

LLT = -92.41 lb_f [each front tire]

Cornering + Acceleration [0.8 G]

↳ **LLT = 46.12 lb_f** [each rear tire]

Cornering + Acceleration [0.5 G] + bump

↳ **LLT = 28.83 lb_f** [each rear tire]

$V = V_0 + at$

$F = \mu_k F_n$

$= 1.5(70.72)$

$= 106$ N

Lateral Load Transfer

Inputs

Vehicle weight: 604 lbs 273.97 kg

% weight dist: 47% F ⇒ 283.88 lbs 128.77 kg

53% R ⇒ 320.12 lbs 137.04 kg

CG Height: 12.30 in 0.312 m

F/R Track: 49.02 in F 1.25 m

47.99 in R 1.22 m

23-24 Max Loading Cases

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Scenario	Lat	Long	Vertical
Vertical Bump	0	0	3
Acceleration	0	1.2	1
Braking	0	-1.5	1
~~Cornering~~	2.4	0	1
~~Cornering + Acceleration~~	2	0.8	1
~~Cornering + Acceleration + Bump~~	0.7	0.5	1.2

*all values in Gs

Theory

*Derived from Optimum G slides

Lateral force = mass · lat. accel

$$LLT = \text{lateral force} \cdot \left(\frac{\text{CG Height}}{\text{track}}\right)$$

Cornering

$2.4\ Gs = 23.54\ m/s^2$

Front:

Lat. force = 128.77 · 23.54

= 3031.25 N

$$LLT = 3031.25 \left(\frac{0.312}{1.25}\right)$$

= 756.60 N

LLT = 170.09 lb$_f$

Rear:

Lat. force = 137.04 · 23.54

= 3225.92 N

$$LLT = 3225.92 \left(\frac{0.312}{1.22}\right)$$

= 824.99 N

LLT = 185.47 lb$_f$

Cornering [2 G] + acceleration

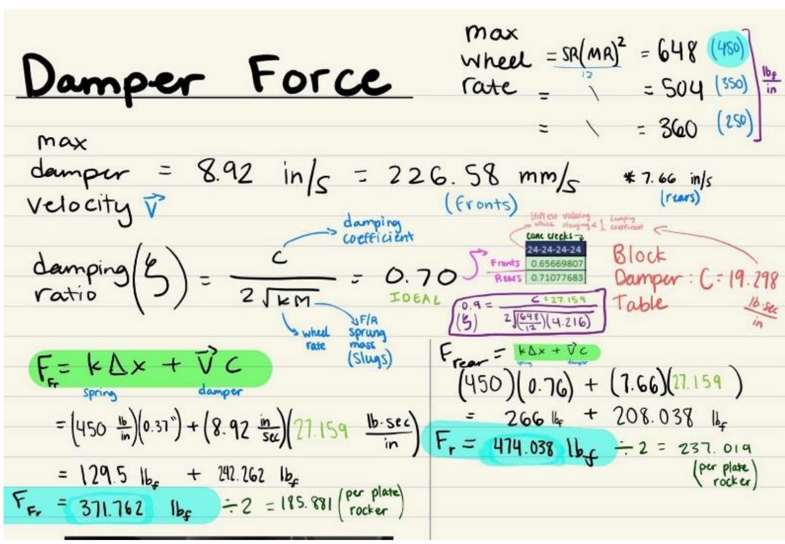
- Front: LLT = 132.66 lb$_f$
- Rear: LLT = 144.65 lb$_f$

Cornering [0.7 G] + acceleration + bump

- Front: LLT = 46.43 lb$_f$
- Rear: LLT = 50.63 lb$_f$

In order to do this, we need to know a few things: basic vehicle specs, lateral load transfer, longitudinal load transfer, corner weights, and max loading Gs per driving scenario. Here's a listed order to find these values so you can do this in a more organized fashion than I did. Only after you calculate the reaction forces can you start using generative design to output initial geometries.

1. Vehicle weight
2. % weight distribution
3. Corner weights
4. CG Height
5. Front and Rear trackwidth
6. Max loading cases (Gs)
7. Lateral Load Transfer
8. Longitudinal Load Transfer
9. Vertical Reaction Force (Fz)
10. Damper Reaction Force
11. ARB Reaction Force



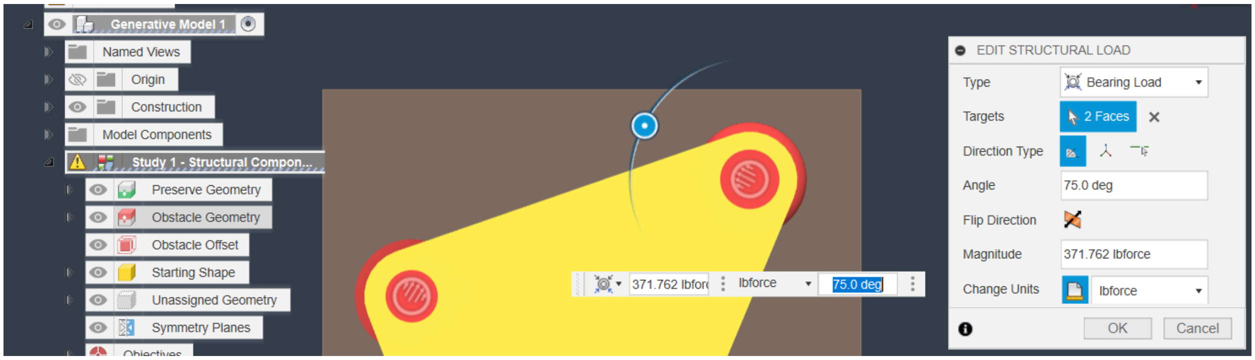
The damper reaction force takes a bit more work to obtain accurate values to use in generative design given that the force is dynamic and varies as a function of spring stiffness, spring displacement, damper velocity, and damping coefficient. Working with Alex Block, I was able to develop a relatively accurate damper force output based on a few characteristics of our dampers F/R shown in the accompanying calculation.

ARB Blade Max Force Value (lb)			
Front		Rear	
Soft	149.925	Soft	133.955
Medium	203.333	Medium	227.303
Hard	291.610	Hard	468.071

*Calculated using 0.8 degrees of roll

Finally, for the ARB reaction force, obtain these values from your ARBs captain. They'll be working on creating a table like this one which is dependent on max roll gradient data which we will hopefully get from Michigan 2024 comp this year.

Step 4. – Iteratively input forces into Fusion360 Generative Extension to obtain rocker geometry



1. Model starting shape, preserve geometry, and obstacle in SolidWorks as separate bodies.
2. Export as a STEP file and import into Fusion360
3. Assign each body to its respective role (Yellow = Starting shape, Red = Obstacle, Green = Preserve)
4. Apply forces as a bearing load to each bolt hole ID
    a. Obtain force angles from a 2-D sketch with a protractor
5. Apply a fixed geometry condition to the rocker pivot hole ID
6. Assign a material to the part
7. Define your targets
    a. FOS
    b. Displacement
    c. Weight
8. Define your manufacturing process
    a. 3-Axis
    b. 5-axis
    c. Additive
    d. CNC Router
9. Once you have the green mark you are ready to run the generative solver

I repeated this process until I had a design which did not have any questionable discontinuities or areas of thin cross-section. That happened to be the 22nd revision. I'd also be happy to run you through this workflow when you're at this point in design.

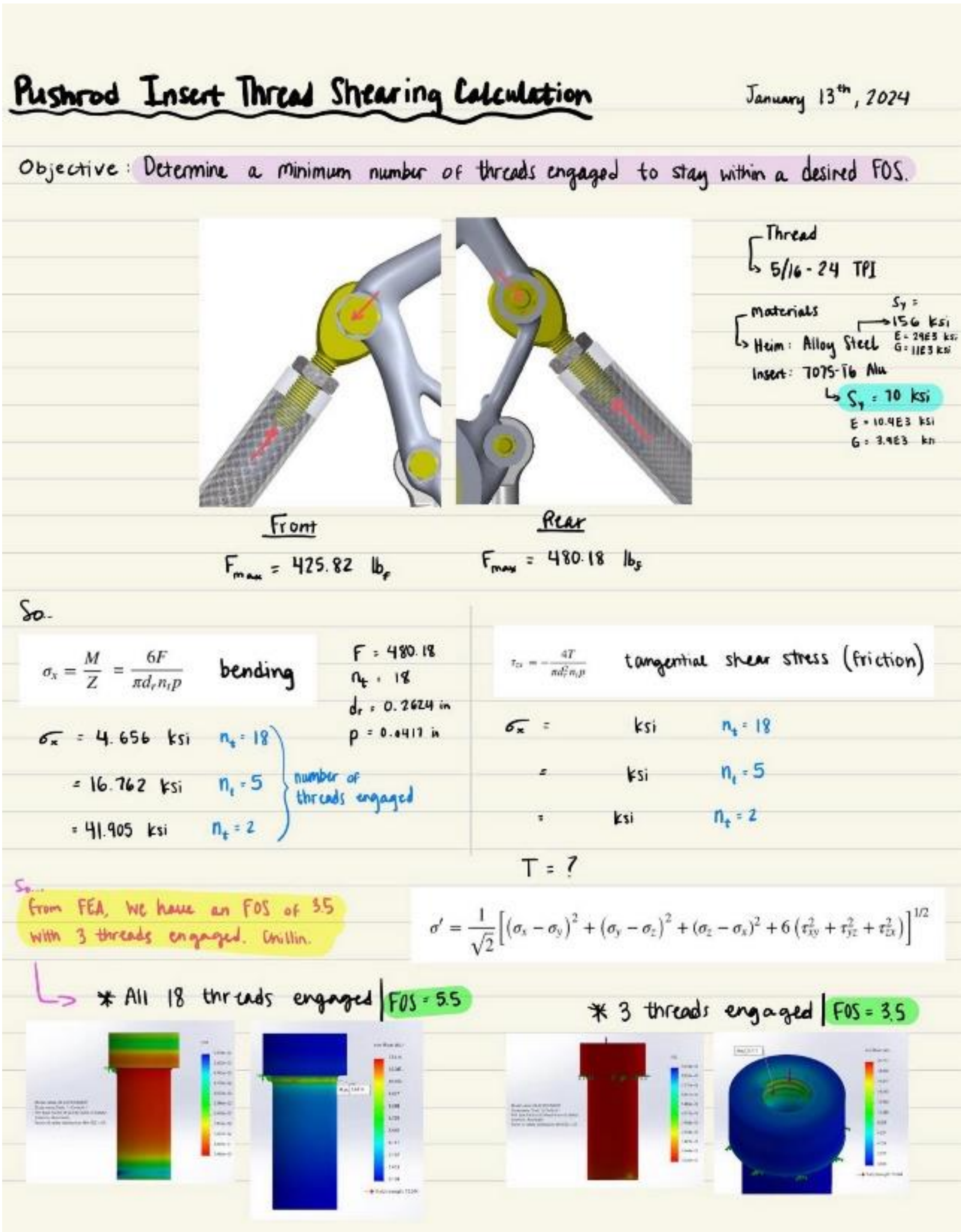
Step 5. – Test carbon tubes in axial stress to validate geometry, material, and bonding procedure

Now, we head into the 2nd aspect of the design: the Pushrods. I knew from the start that I wanted to run carbon fiber tubing for the pushrods simply because of how much weight could be saved while maintaining an adequate amount of stiffness. Many other teams out in Europe served as inspiration for my endeavor into this area of uncharted territory for the FSAE team here at CPP. Hopefully the following step-by-step guide will help you to produce carbon tubing at a much faster rate.

1. Acquire the following materials
   a. 7075-T6 Aluminum Inserts
   b. Loctite E-120HP
   c. 50ml Epoxy Gun + mixing tips
   d. Bottle of Alodine Aluminum Etcher
   e. WD-40 Bike Degreaser
   f. 120 grit sandpaper
   g. 32g (0.008”) OD copper wire
   h. Blue tape
   i. 2 epoxy mixing cups
   j. Snips
   k. Needle nose pliers
   l. Safety glasses, vapor mask, gloves, PPE, etc…
   m. Carbon tubes cut to length
   n. Microfiber towel
   o. White paper towels
   p. Timer
   q. Small strip of cardboard
2. Cut carbon tubes to length with handsaw then finely sand to get within 10thou using the height gauge in the shop (do not lathe or grind. It will fray very quickly this way)
3. Sand down inserts with 120grit
4. Degrease inserts and tube ID with degreaser
5. Pour about a 1.5-inch-deep amount of Alodine into a mixing cup. Do the same thing for water right next to it. Now fully submerge the degreased inserts into the Alodine etcher for 5 minutes.
6. Remove them from the Alodine with needle nose pliers and let them dilute in the cup of water for a minute. Then, take them out and air dry until completely dry.
7. While letting them dry, start lining the carbon tube with 3 strands of copper wire and tape them to the tube with blue tape. See the accompanying picture for proper installation. The whole point of this part is to keep the insert concentric with the carbon tube while it cures.
8. Apply epoxy adhesive within 10 minutes of acid removal to the entire bond surface on the insert AND in the tube. We broke a rear toe link due to improper bonding.

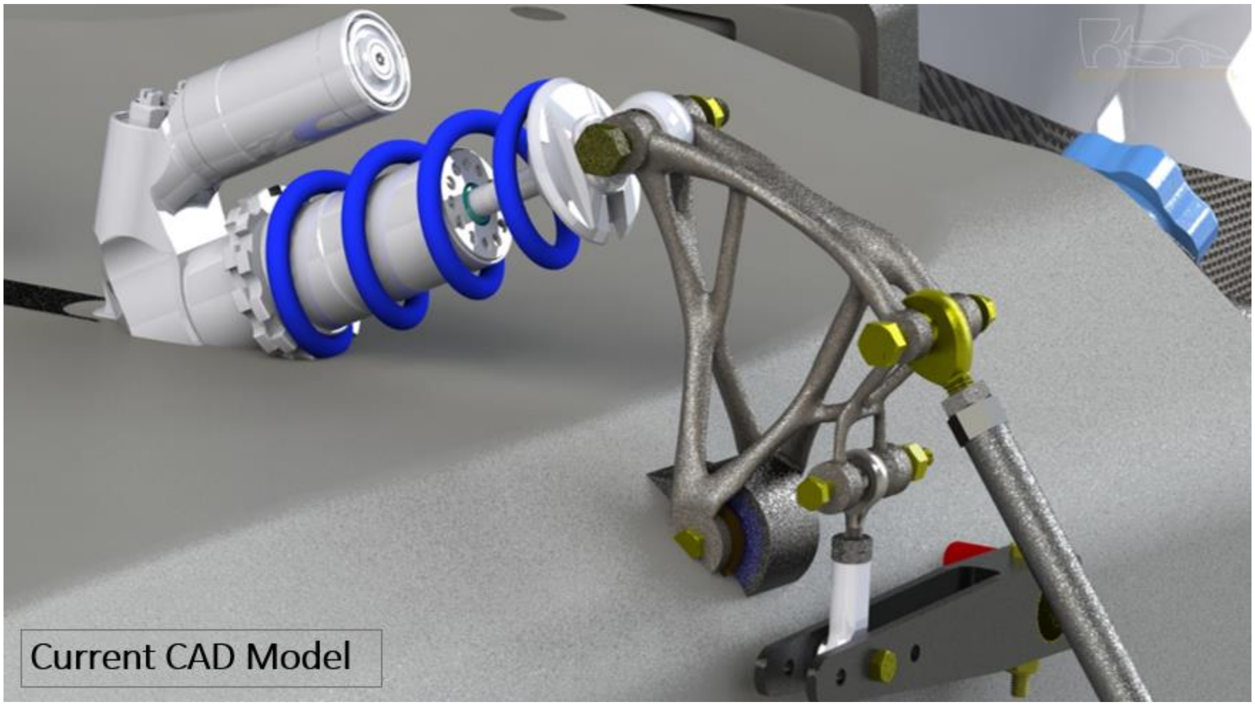
9. Once both mating surfaces are covered in epoxy, slowly insert 1 at a time while softly spinning the insert to distribute the epoxy evenly. Press into the tube until it bottoms out and epoxy squeezes out.
10. Now, pull the insert back out only about a quarter inch and use snips to cut off the exposed copper wire. We need to do this to allow an even mating surface to the OD of the carbon tube. Otherwise, the copper wire won't allow a clean bond and may potentially induce a stress concentration in the epoxy.
11. Once this is done, repeat step 9.
12. Now, clean up the leftover epoxy on the tube by spinning the tube while your free thumb and index finger cover the end of it. The final product should be a nice clean ring a epoxy covering the bond line.
13. Set the tubes on a paint can on its side to cure untouched for 24 hours.
14. Repeat the above steps for all tube ends.
15. Show all your friends the amazing and light carbon tubes you made for the car.

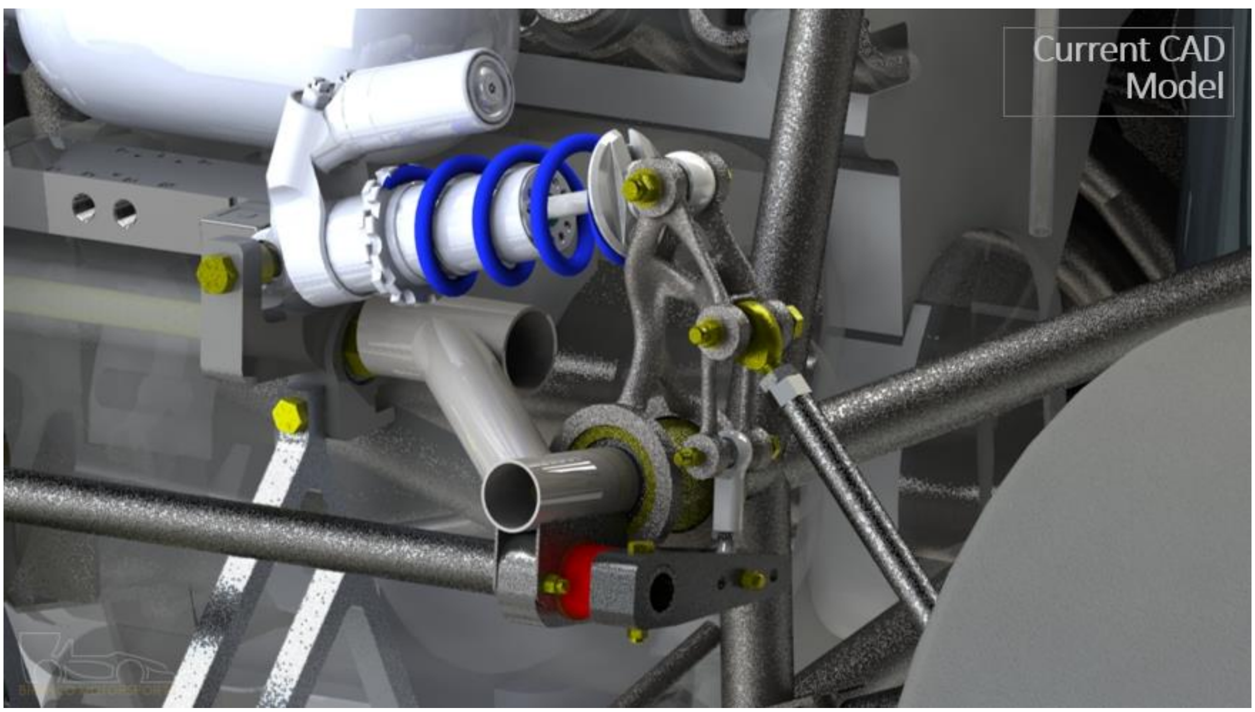
One of the failure modes I definitely wanted to avoid was the shearing of aluminum internal threads in the pushrod insert. Essentially, I focused one single thread and modeled it as a cantilever beam experiencing bending and transverse shear. With 7075 Aluminum (the strongest by a considerable margin), I was able to obtain FOSs for a couple different number of threads engaged. I wasn't too worried to begin with, but now that we have numbers to back up the assumption that these parts won't fail under our known loading conditions, we are in a much better spot from any engineer's perspective.



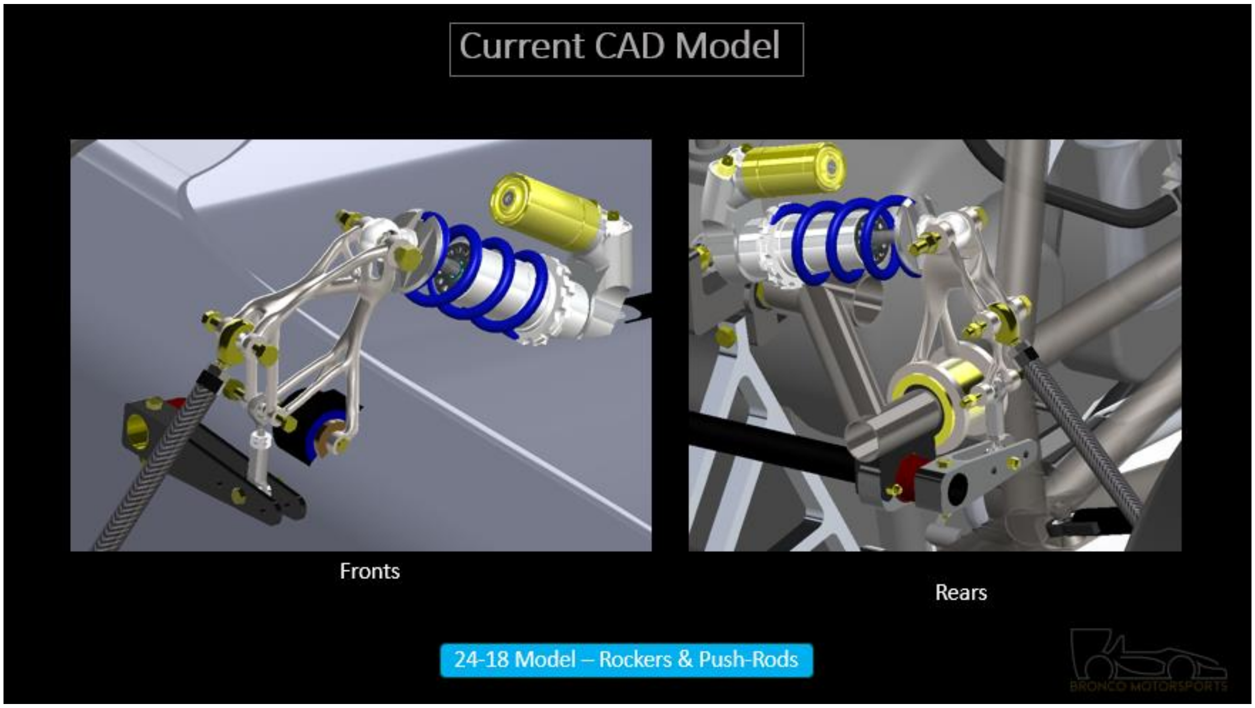
Design Revisions

24-17 – Mission Concept Review | August 2023

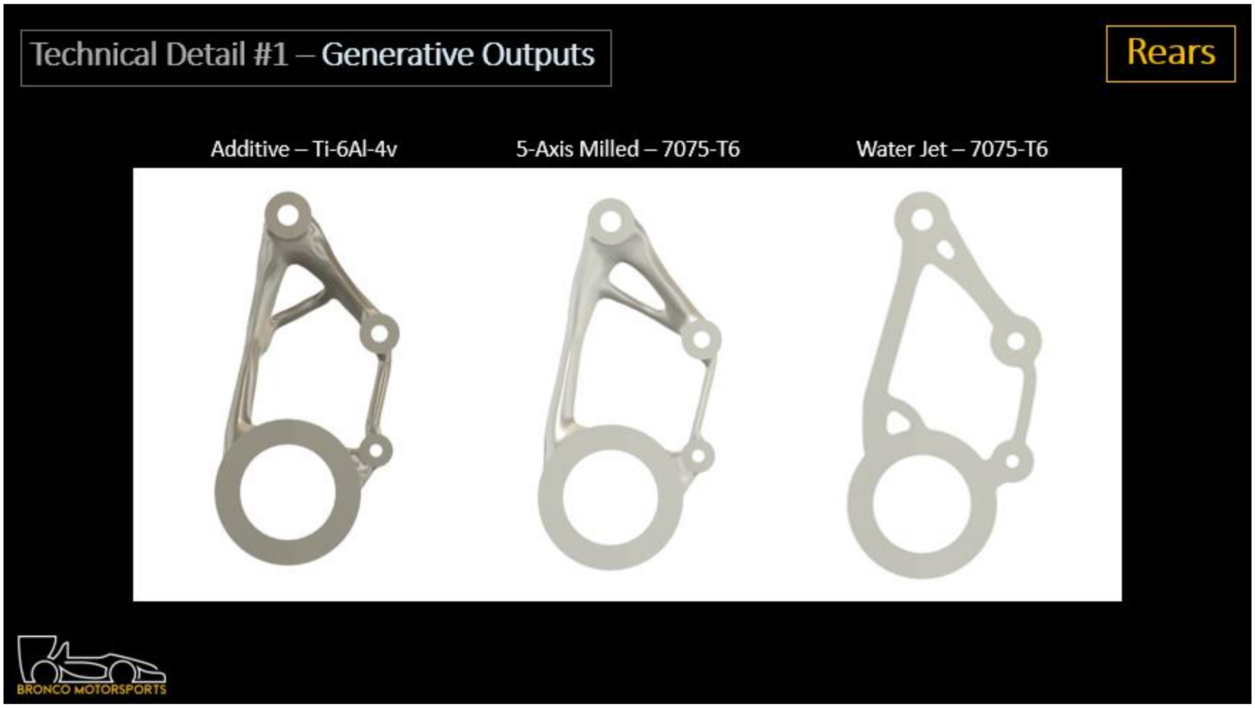




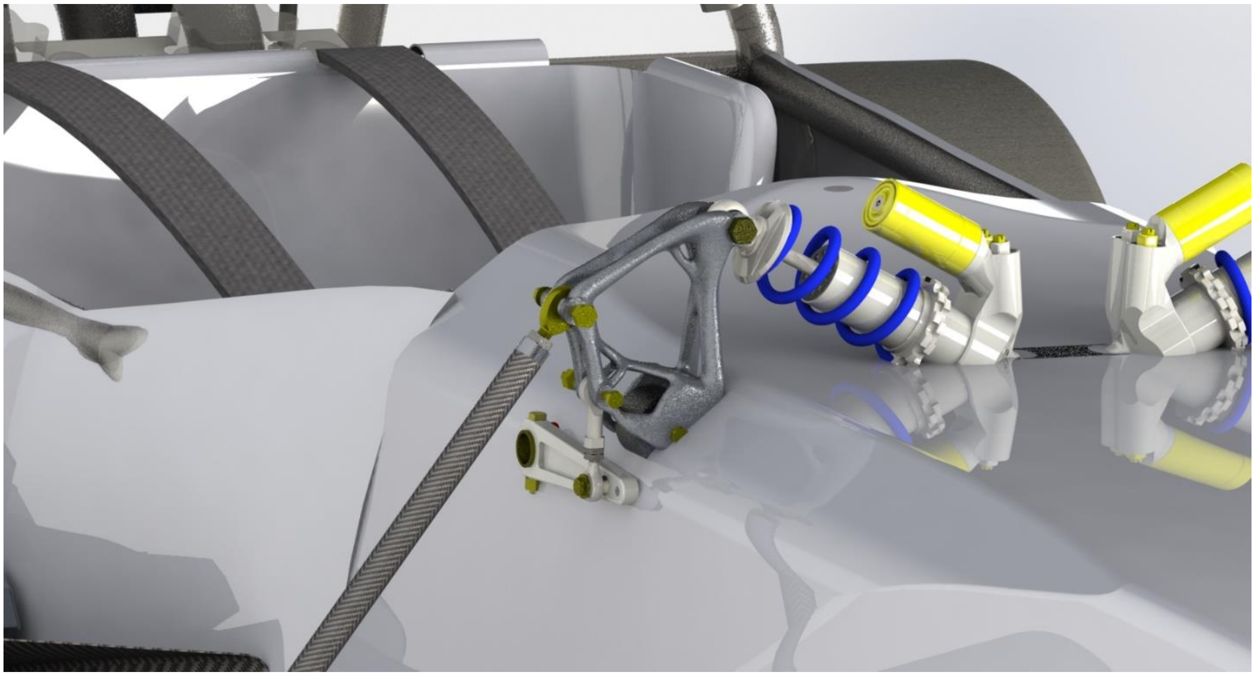
24-19 – Preliminary Design Review | September 2023

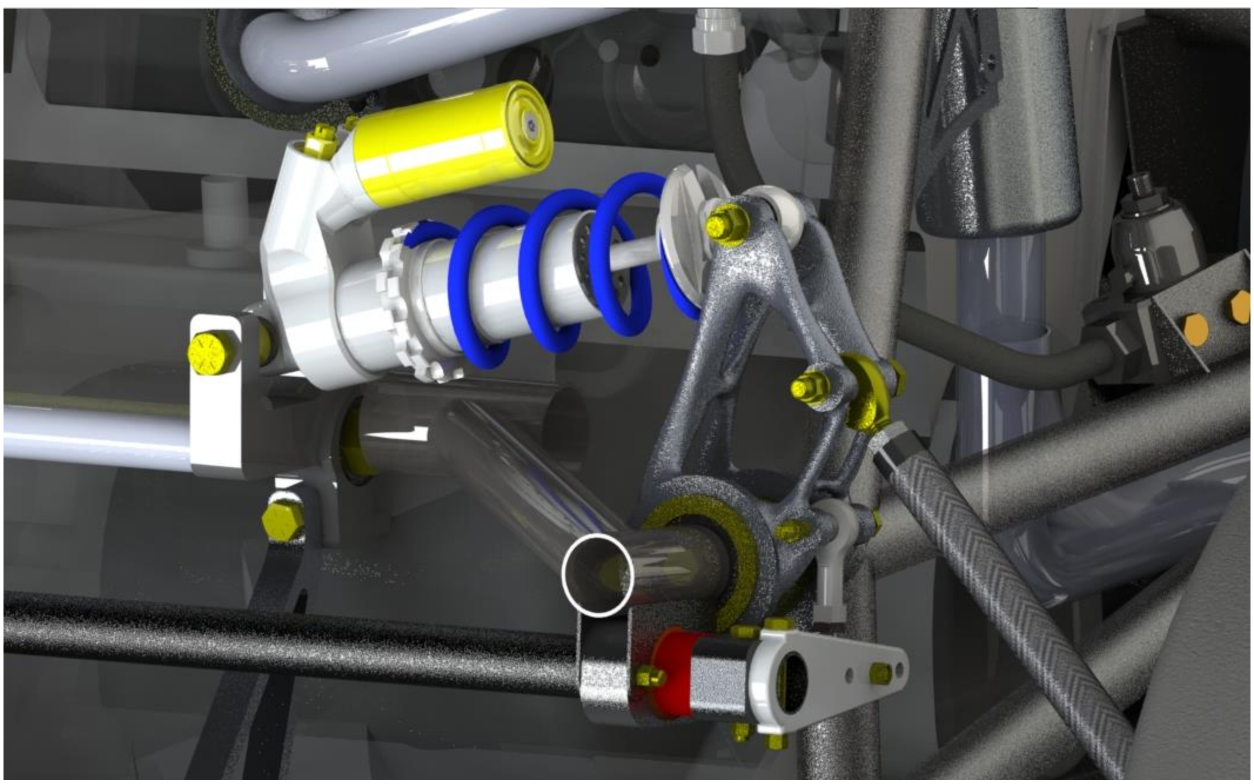


24-21 – Critical Design Review | October 2023

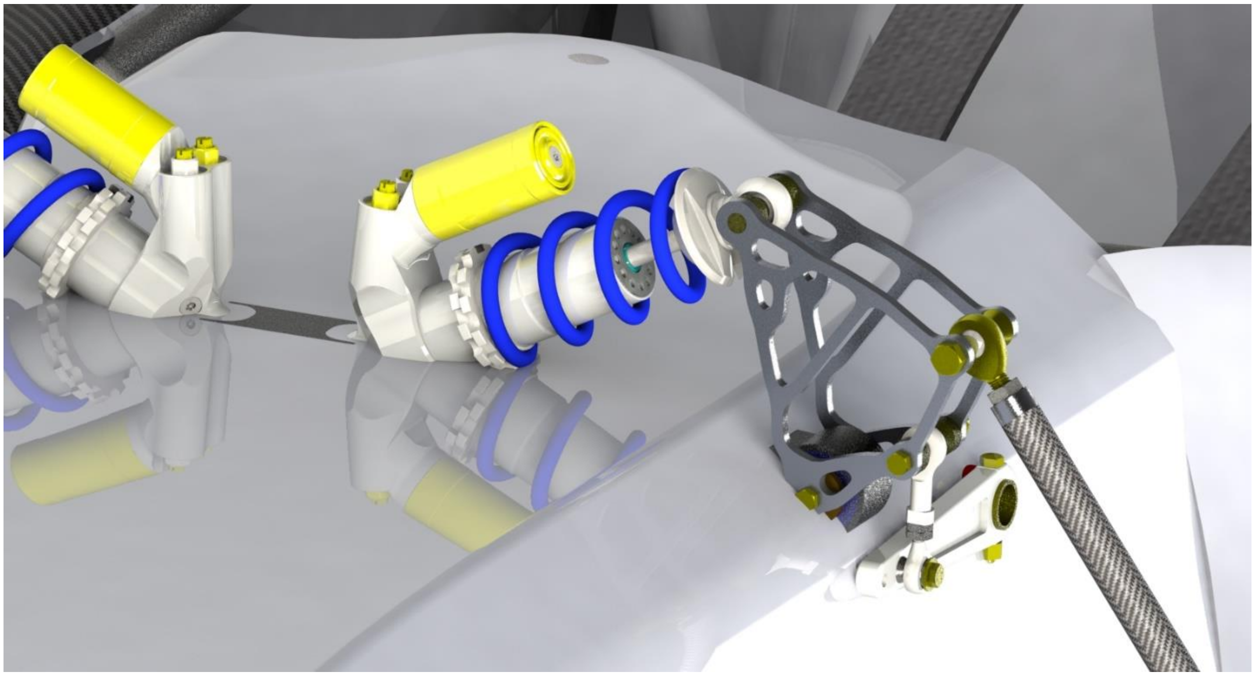


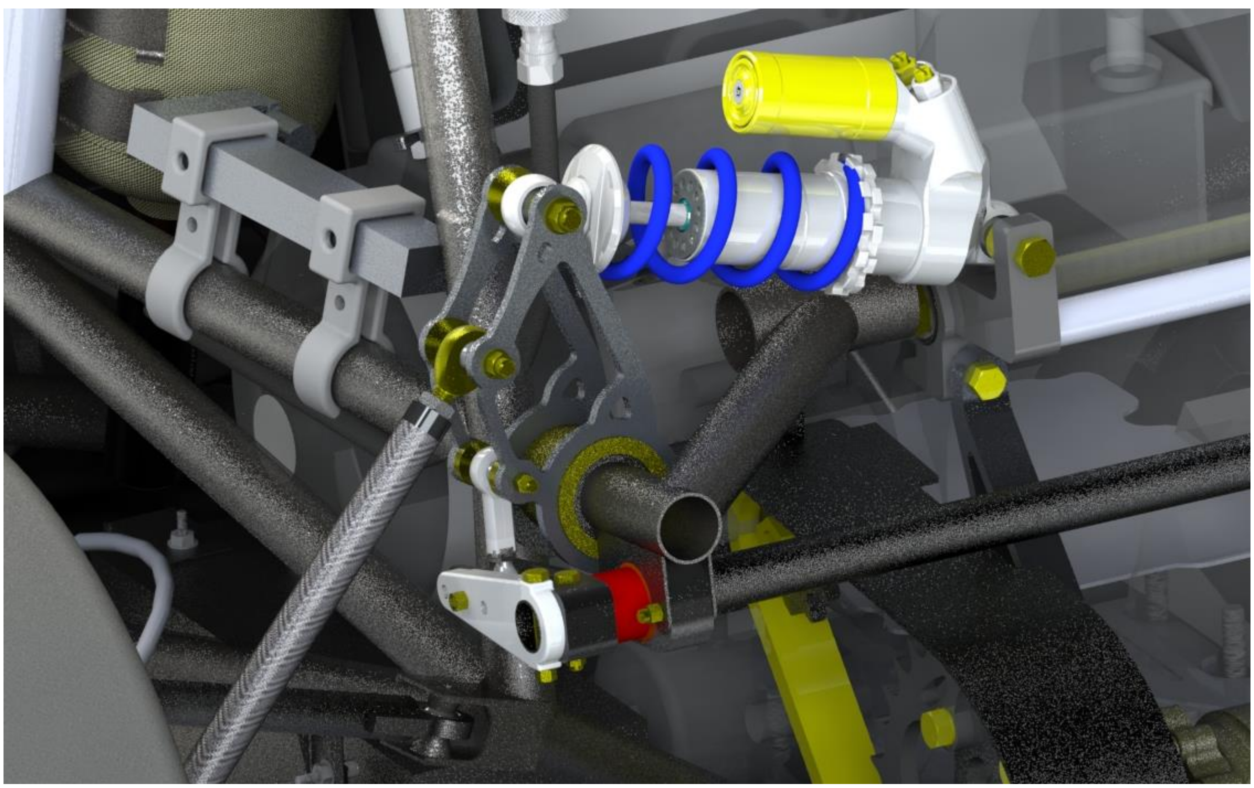
Final CAD Model | 24-22



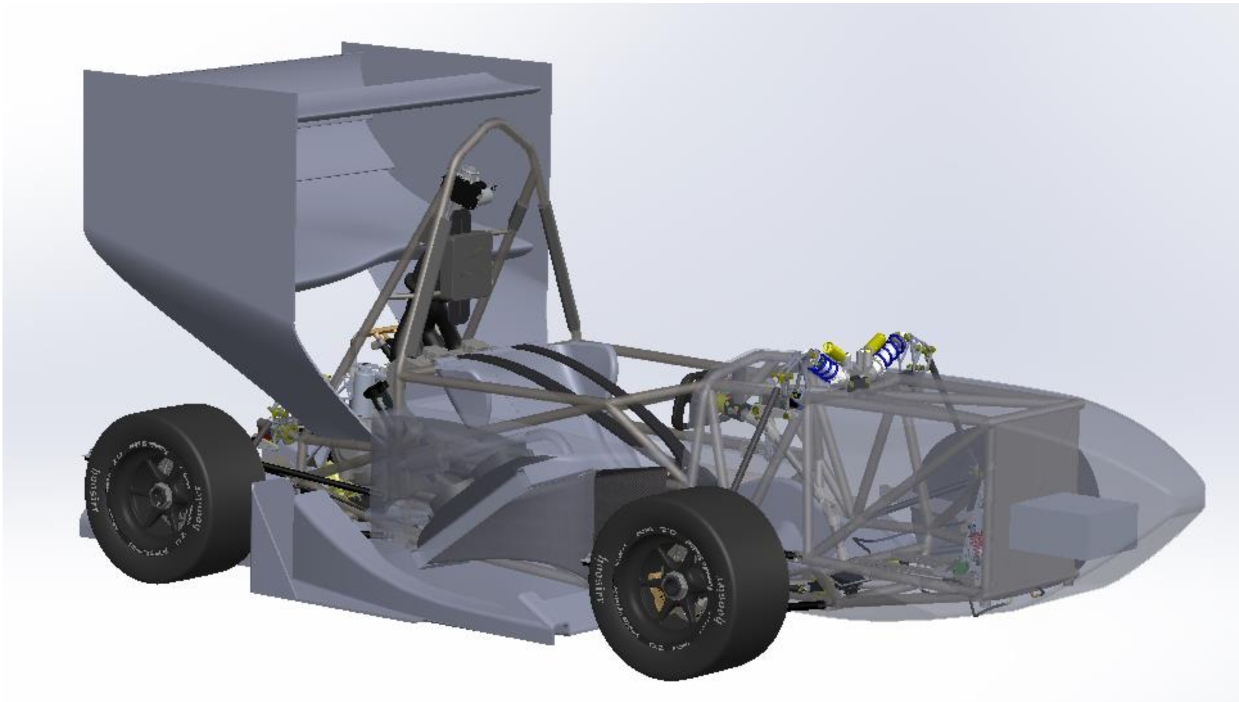


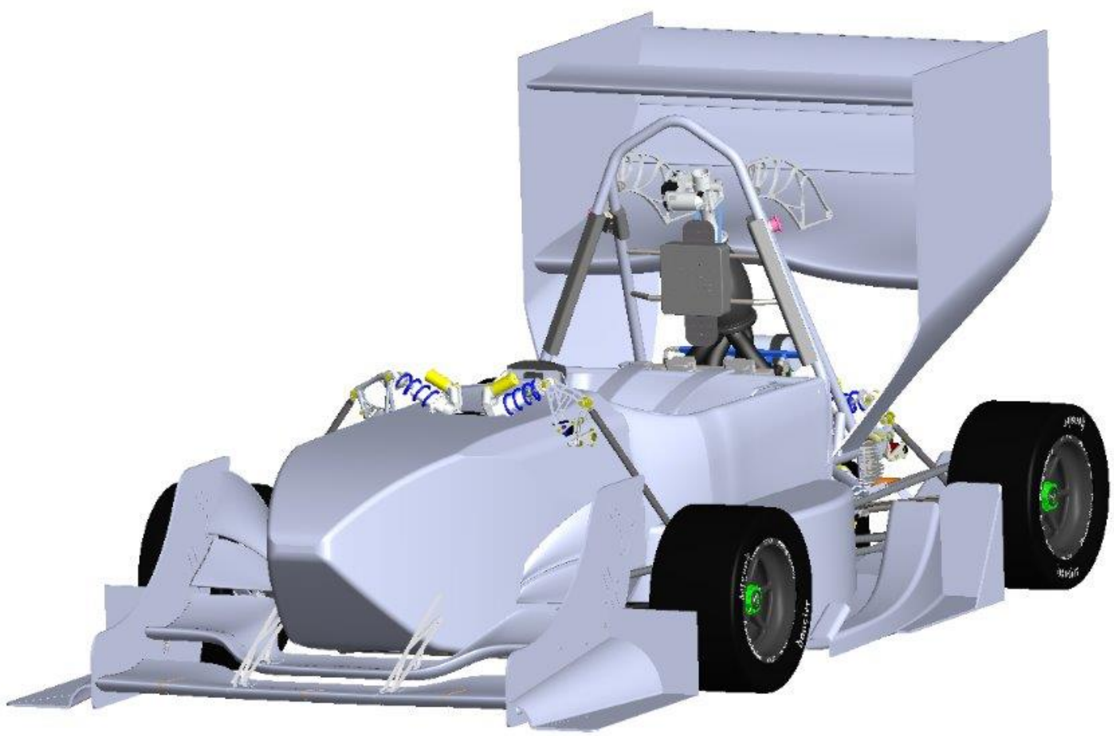
Final CAD Model | 24-22 | *BACK-UP*





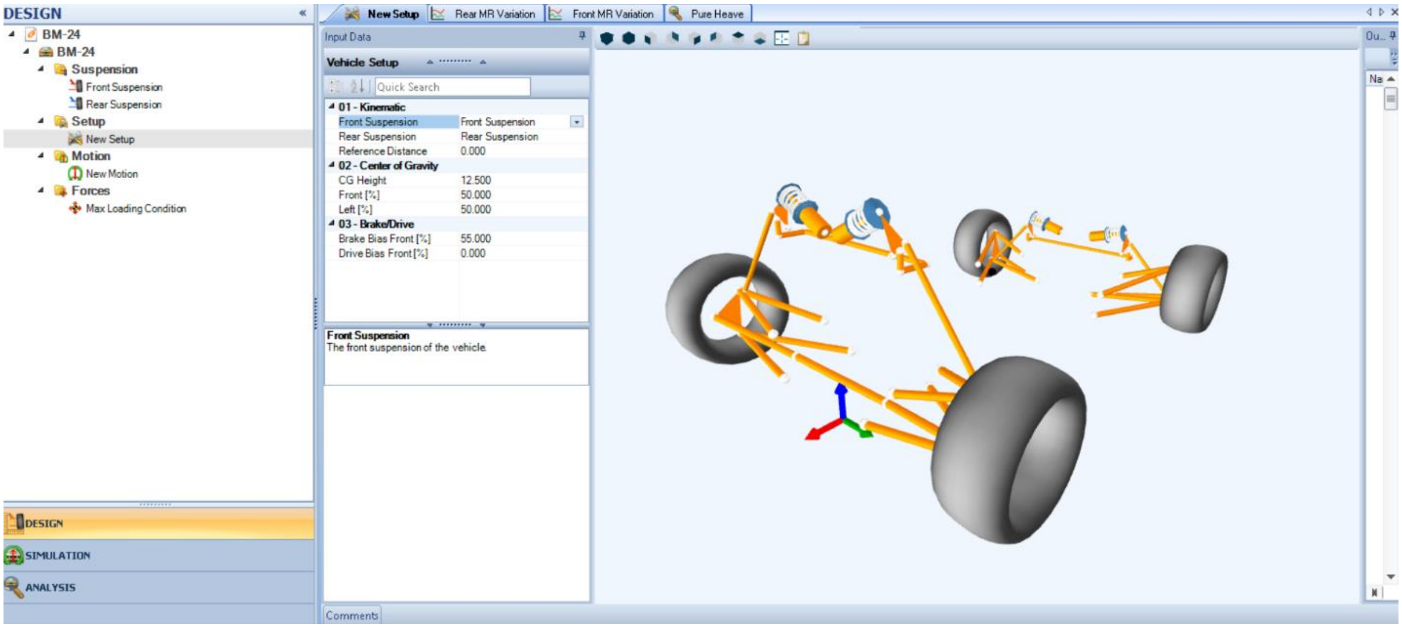
24-22 Full Vehicle Model



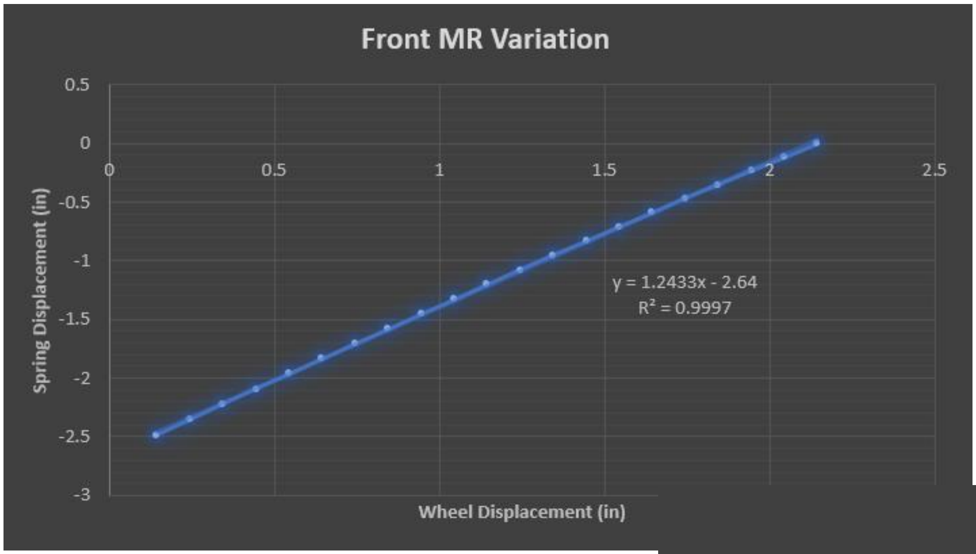


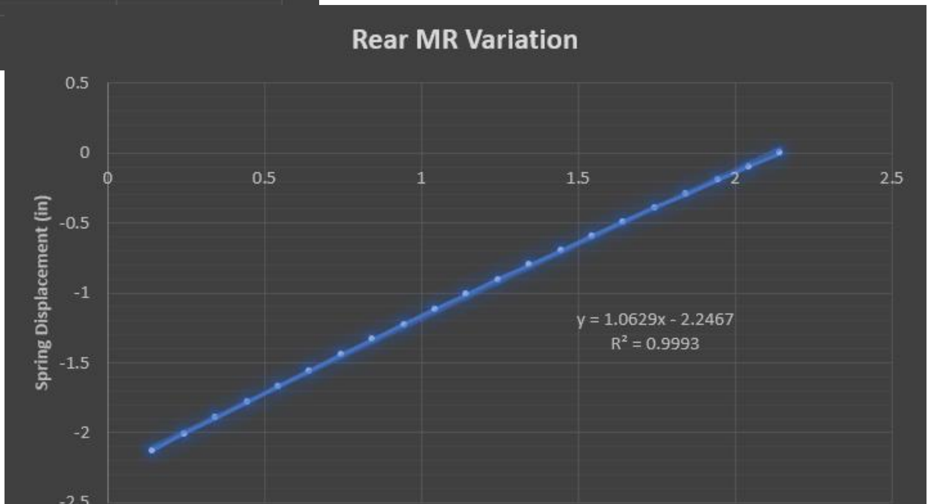
Simulations

Optimum Kinematics model of suspension PUPs



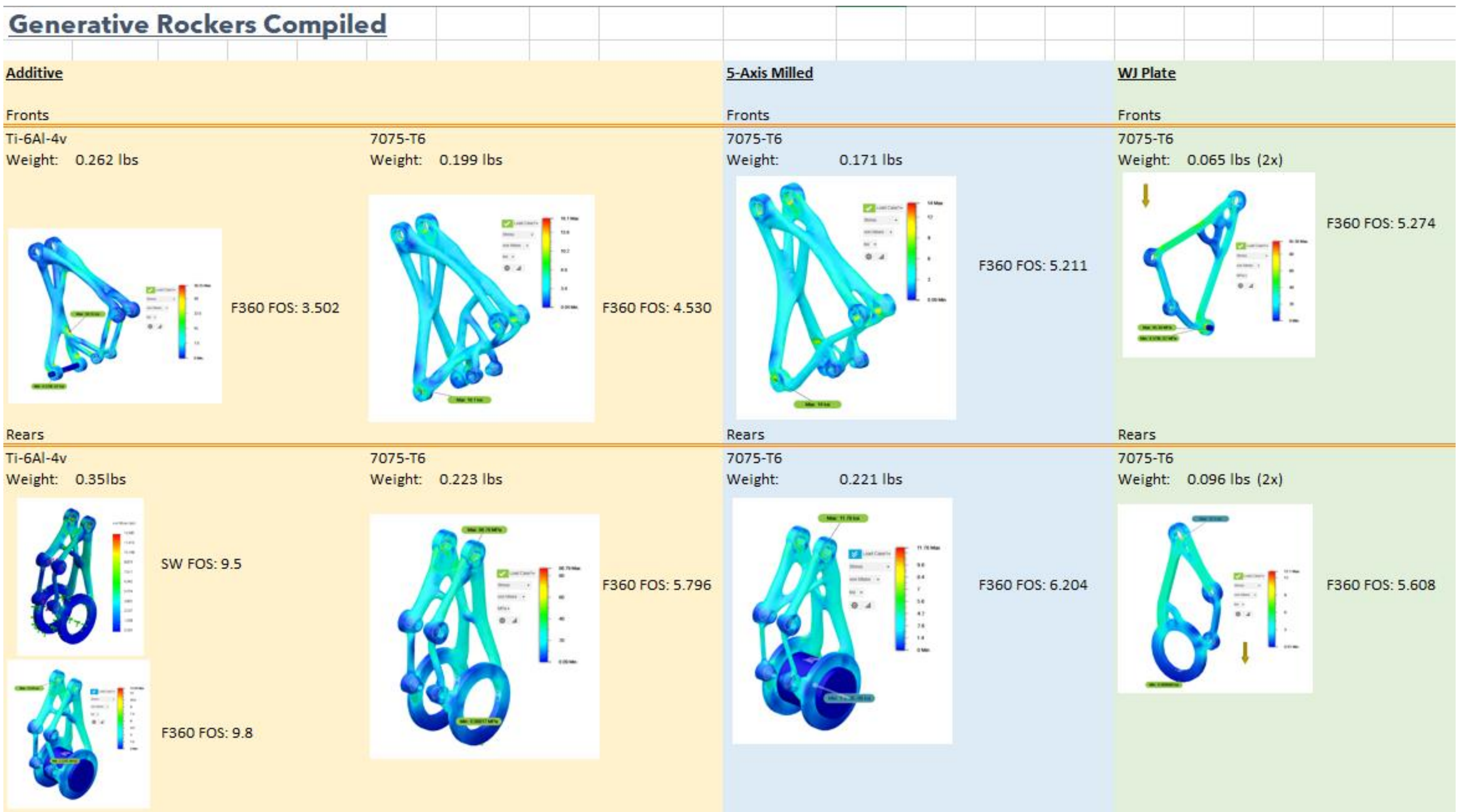
Motion ratio VS Ride height F/R

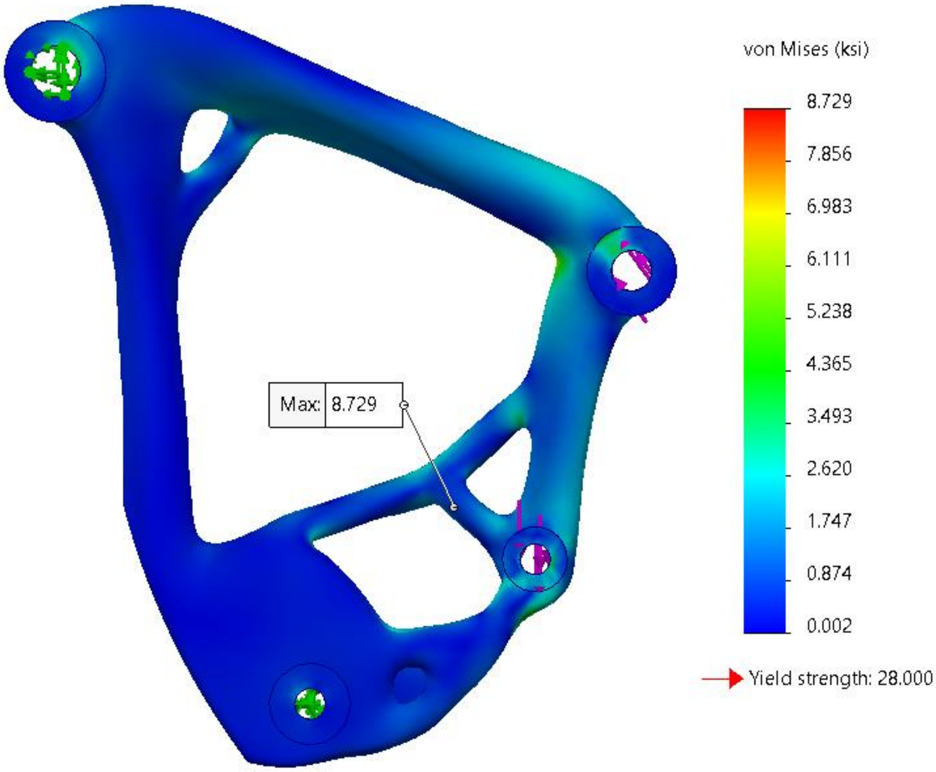


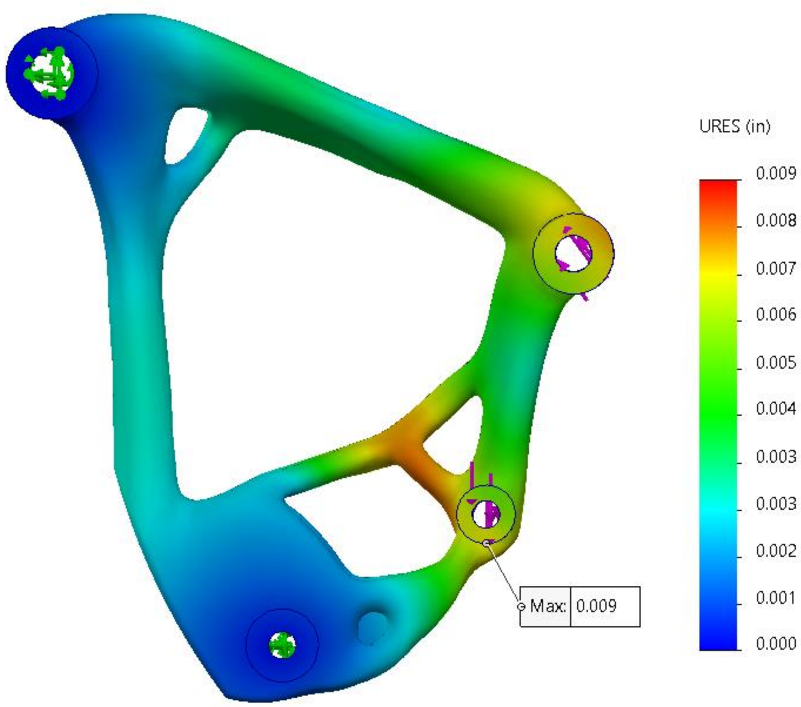


Rocker FEA

Shown in the first graphic is a compilation of several different rocker geometries based on the way they are intended to be manufactured. One of the other tests I was running was to see how the final factors of safety would differ between SolidWorks FEA and Fusion360 FEA. The difference ended up having Fusion360 FEA, on average, estimate 0.5 higher of an FOS per model. SolidWorks is slightly more conservative. I've displayed the final outputs for the rocker geometries currently on the car on the next page for reference. Given the current results, there is definitely room to slim down the current design for a lower FOS but only once we've created an S-N curve for AlSi10Mg or any other metal printed alloy. I've attached this file in the 2024 OneDrive.







Front Rocker

Pushrod force = 425.82 lbf

ARB force = 291.61 lbf (Hard)

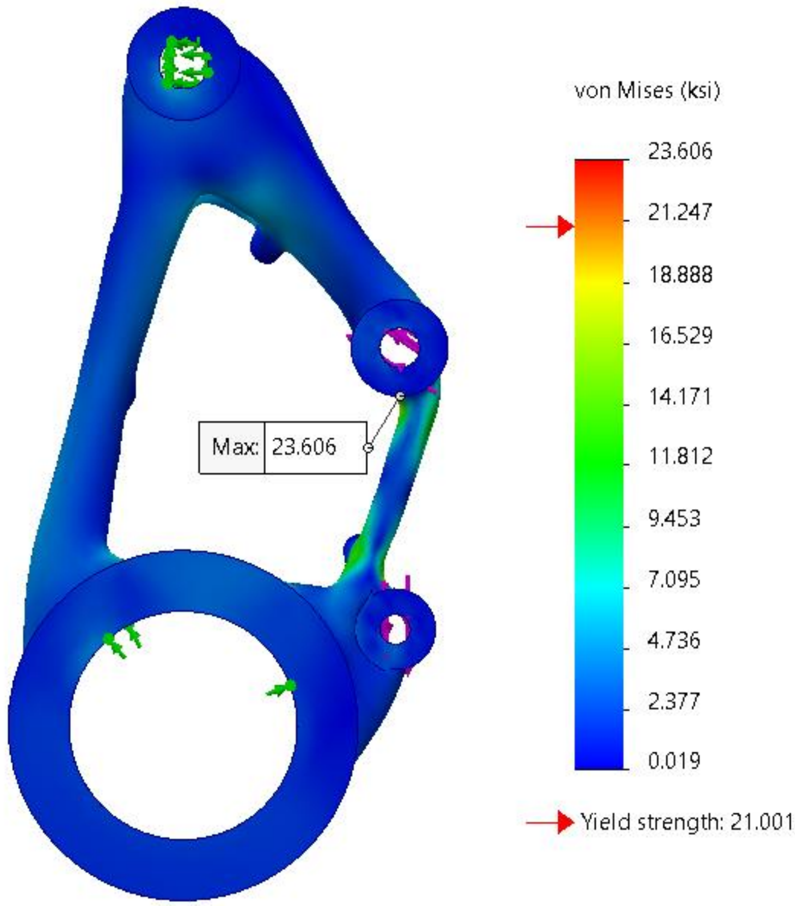
Max Stress = 8.729 ksi

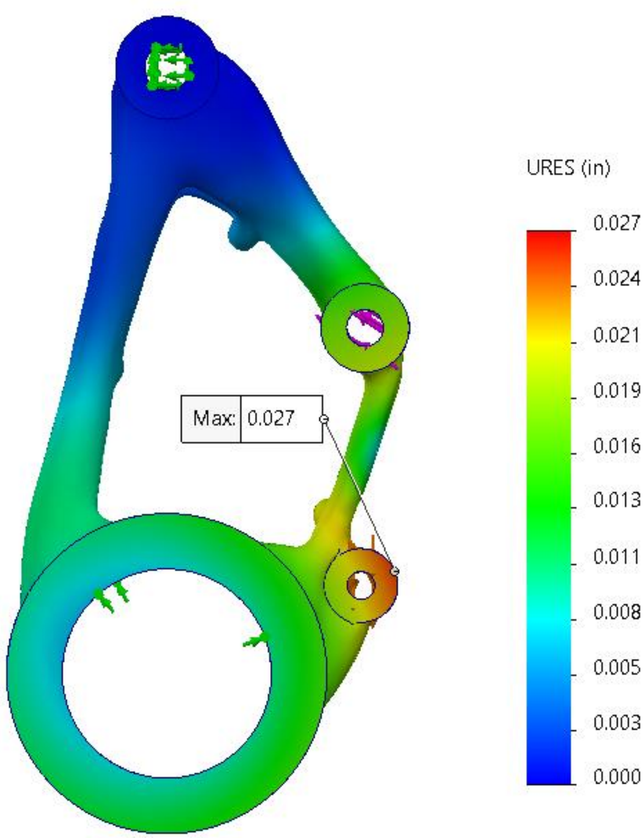
Max Displacement = 0.009”

Min FOS = 3.2

Notes: Rocker Pivot ID and Damper ID is fixed

An FOS of 1.2 is really all you need but given the lack of material data in fatigue, I chose to output a more conservation final design. I’ll touch more on the current fatigue situation later on.





Rear Rocker

Pushrod force = 480.18 lbf

ARB force = 468.07 lbf (Hard)

Max Stress = 23.606 ksi

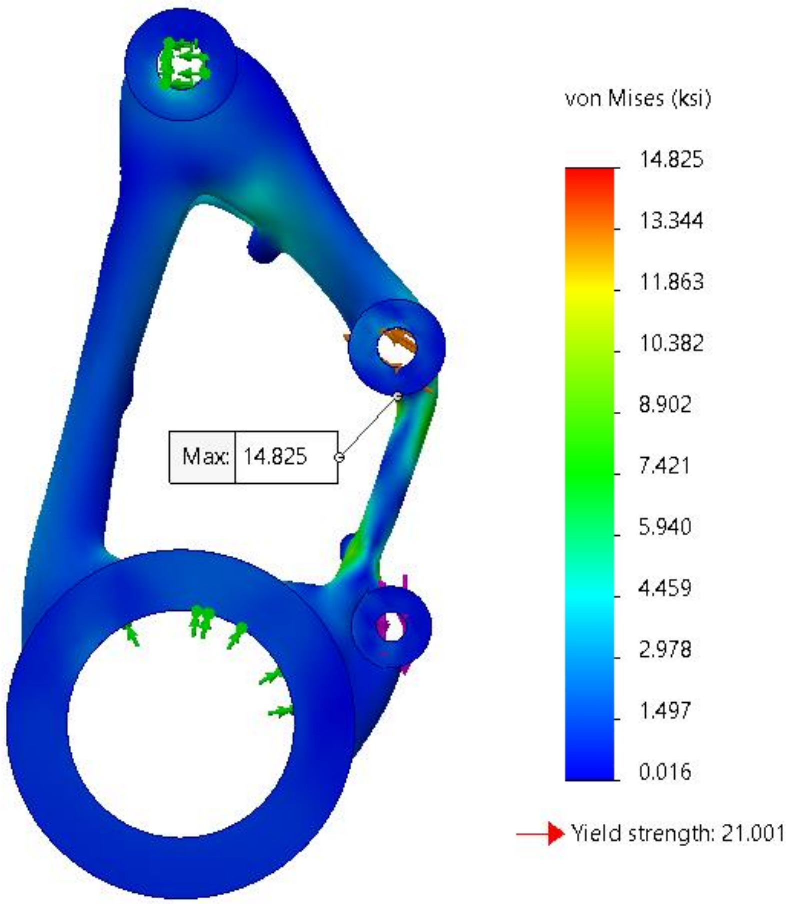
Max Displacement = 0.027"

Min FOS = 0.89 SW, 1.75 F360

Notes:

Rocker Pivot ID and Damper ID is fixed to obtain a static loading condition.

We're getting a 0.89 FOS due to the new value in Yield Strength we obtained from our in-person stress testing (21 ksi). We've been designing for 28 ksi at yield. This leads us to believe it may not be the best option to run the hard setting on rear ARB for comp or at all.



Rear Rocker

Pushrod force = 480.18 lbf

ARB force = 227.30 lbf (Medium)

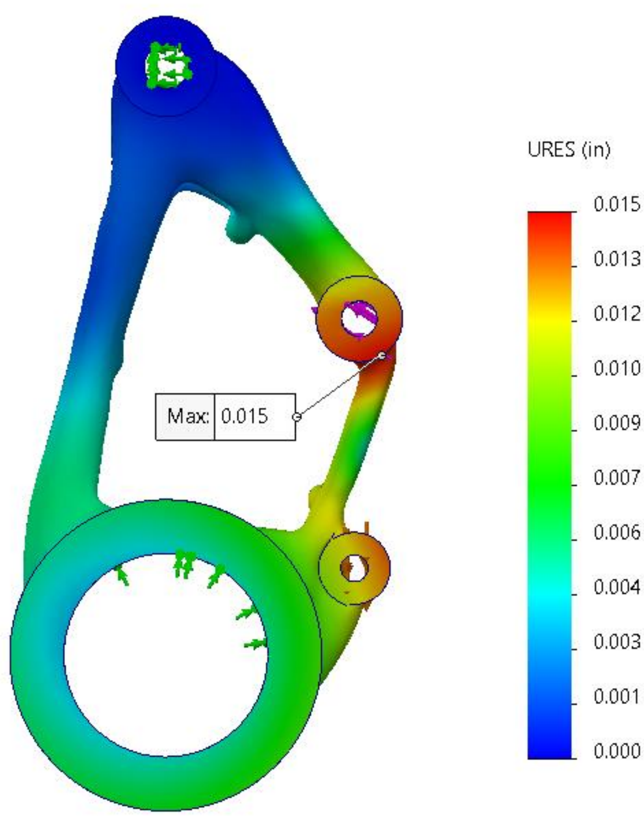
Max Stress = 14.825 ksi

Max Displacement = 0.015”

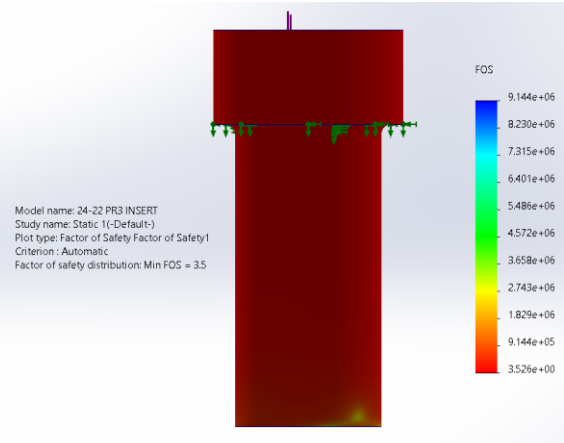
Min FOS = 1.4 SW

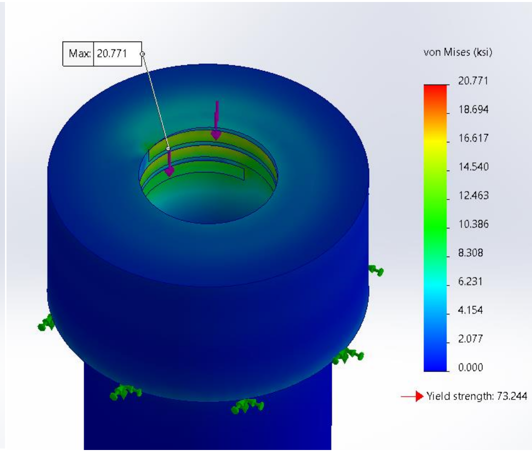
Rocker Pivot ID and Damper ID is fixed to obtain a static loading condition.

This is the current setup we will be running at comp with rear ARB on Medium setting.

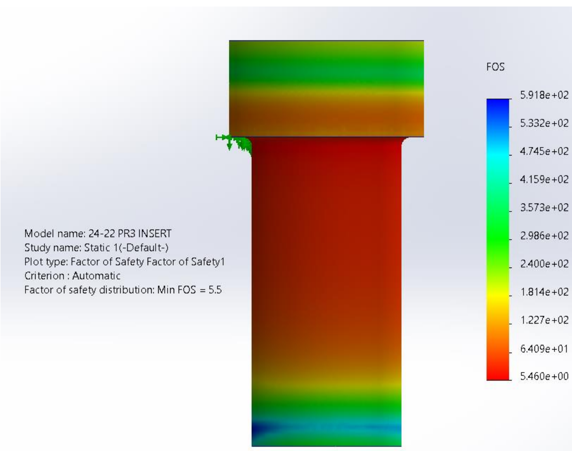


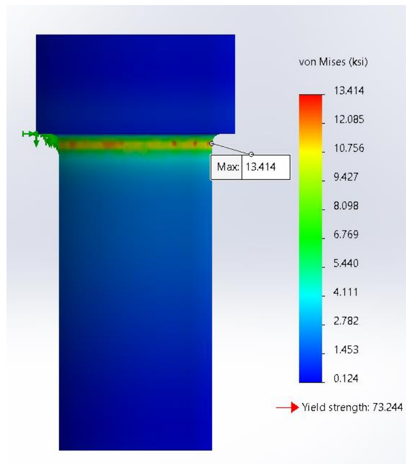
<u>FEA on Pushrod Inserts</u>





Even on only 3 engaged threads at max loading (480.18lbsf), FOS = 3.5



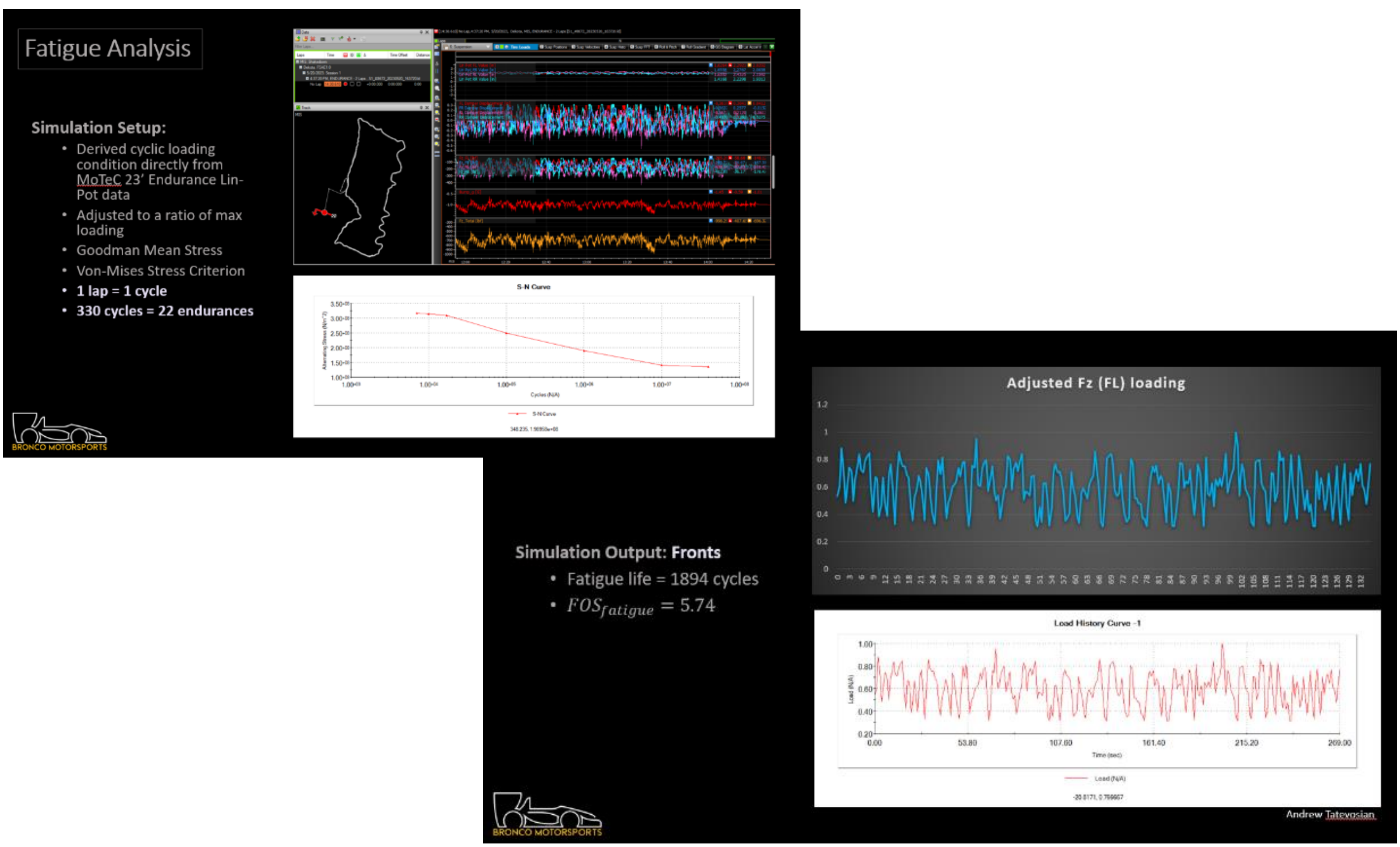


18 engaged threads (Heims bottomed out) at max loading (480.18lbsf), FOS = 5.5

<u>Rocker Fatigue Simulation</u>

One of the more difficult goals I had for rockers this year was to create my own S-N curve for AlSI10Mg (the powder alloy of the rockers) to be able to accurately run fatigue simulations on the metal printed rockers. This graph is the missing piece of the puzzle because it is unique to the characteristics of the material itself at several different number of cycles to failure. It is a graph of Fatigue Stress as a function of Number of Cycles to failure. You'd then input this into SW Fatigue or ANSYS Fatigue and run the solver with a highly specialized and accurate material metric applied. In order to do this, you would need a dedicated axial fatigue tester. CPP Aero department actually has one to test wings, but it is currently out of commission to my knowledge. If I had another year on the subsystem, I would definitely try to work with Orestis to get the machine running again.

Nevertheless, I ended up still running the SW Fatigue simulations with a known SN-curve for 7075 Aluminum. One interesting part about this study was how Andrew T. was able to convert lin-pot data in i2Pro and use that time transient data as a load ratio (history data) to accurately cycle the SW model through its unique fatigue solver known as the Rain flow counting method to solve for damage and cycles to failure. This setup has much merit if done under the right conditions. Unfortunately, this means very little for our rockers since they're metal printed and lack the necessary SN-curve to obtain reasonably accurate fatigue data. Shown below is the setup without proper outputs. I'd highly recommend you take a read through the fatigue portion of Andrew's Hubs and Uprights Design Binder for the proper execution of this type of fatigue analysis.



Manufacturing

Total Spent: $506.96

Notes: Reused the 5/16-24 Heims for Pushrods (245$ per Heim if you plan on getting new ones)

BOM		
Component	Description	Quantity
Front Rocker	Additive, Milled, Water Jet	2
Rear Rocker	Additive, Milled, Water Jet	2
Front Push-Rod	Carbon Tube	2
Rear Push-Rod	Carbon Tube	2
Front Rocker Pivot	Steel Round Stock - Lathed	2
Rear Rocker Pivot	Steel Round Stock - Lathed	2
SKF 61804 Deep Groove Ball Bearing	Housed within rocker pivot	4
Teflon Bushings	Between bolt and Ball Bearing	4
Threaded Inserts for PRs	7075-T6 Alu	8
Mil-Spec Spherical Bearing Rod Ends (SAE	Steel, ⍰5/16" ID, 5/16-24 Thread	8
Rear rocker sleeves	Contrain rear rockers axially	6
Front Damper Bungs	7075-T6 Alu - Lathed In-House	2
Rear Damper Bungs	7075-T6 Alu - Lathed In-House	2
Rocker Plates	3/16 7075-T6	4
Retaining nuts	Steel	8
Plate bungs	6061-T6 (1" stock)	
Steel PR inserts (hex stock)	4130 steel	8
steel PR tubing	0.056 wall thickness	4

Process demonstrated in chronological order

Pushrods

1. Cut tube to size (F/R) and refine on sanding wheel
2. CNC lathe inserts
3. Tap inserts
4. Begin the bonding process
    a. See attached step by step tutorial
5. Let cure for 24 hours
6. Wrap with matte clear tape for protection
7. Mount on car

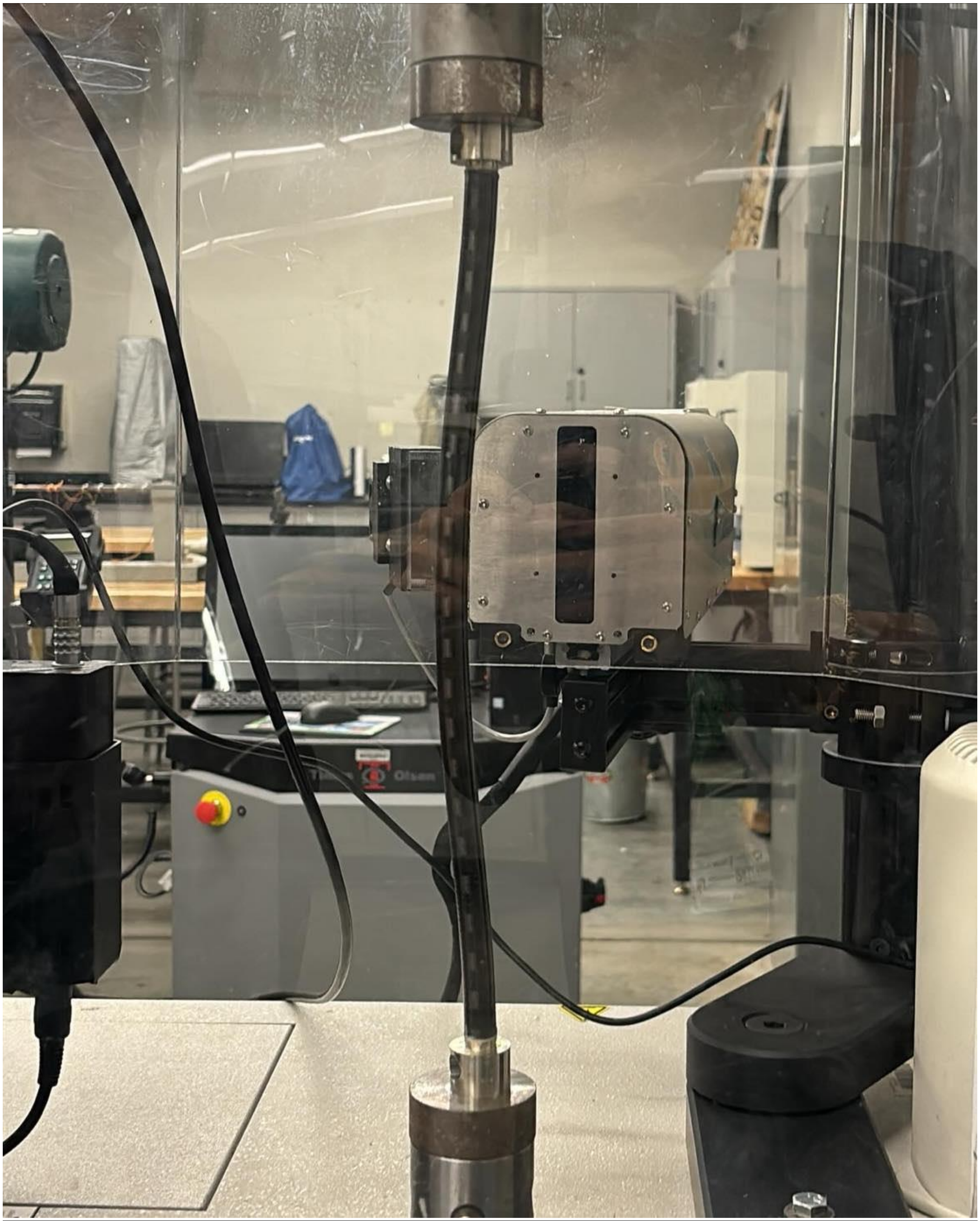
Rockers

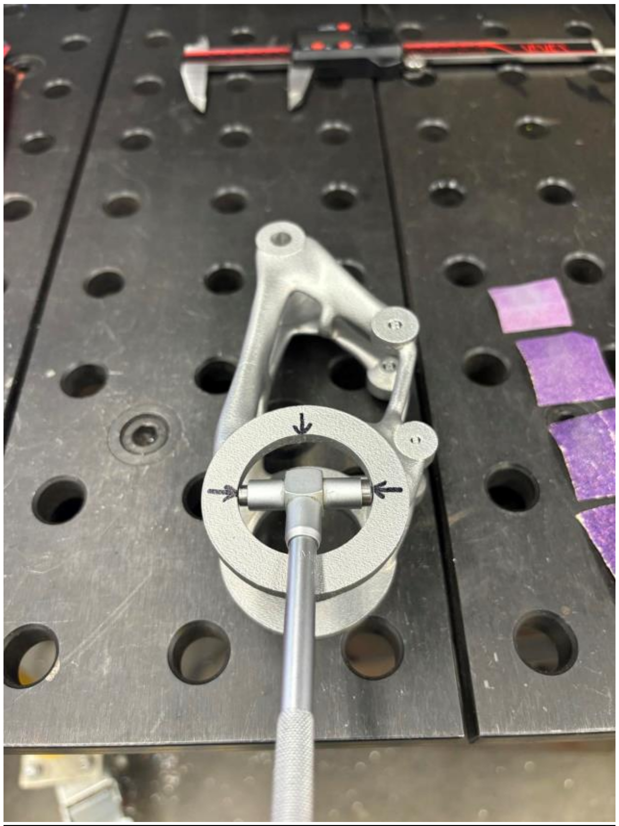
1. Receive printed rockers from MIMOTechnik
2. Lathe
    a. Front
        i. Rocker bearing housings (2x)
        ii. Teflon Bushings (2x)
    b. Rear
        i. Rocker Pivots (2x)
        ii. Rocker Bearing sleeves (4x)
        iii. Retaining Sleeves (2x)
3. Notch square tubing connection to chassis and weld before powder coat
4. Model Jigs to hold rockers horizontally in vice
5. Drill and ream all bolt holes, test for fitment
6. Degrease and apply a 3-layer coat of satin clear spray paint, blue tape on inside bolt faces
7. Send Rear rocker bearing sleeves and retaining sleeves out the Danco Anodizing
8. Press in the rear rocker bearing sleeves with a dead blow mallet
9. Apply Loctite 648 Retaining Compound for any gap between bearings (61804) and shaft
    a. Let cure for 24 hours
10. Mount on car

**NOTE: Backup manufacturing involves the use of plate rockers in double sheer w/ bungs and steel pushrods.*

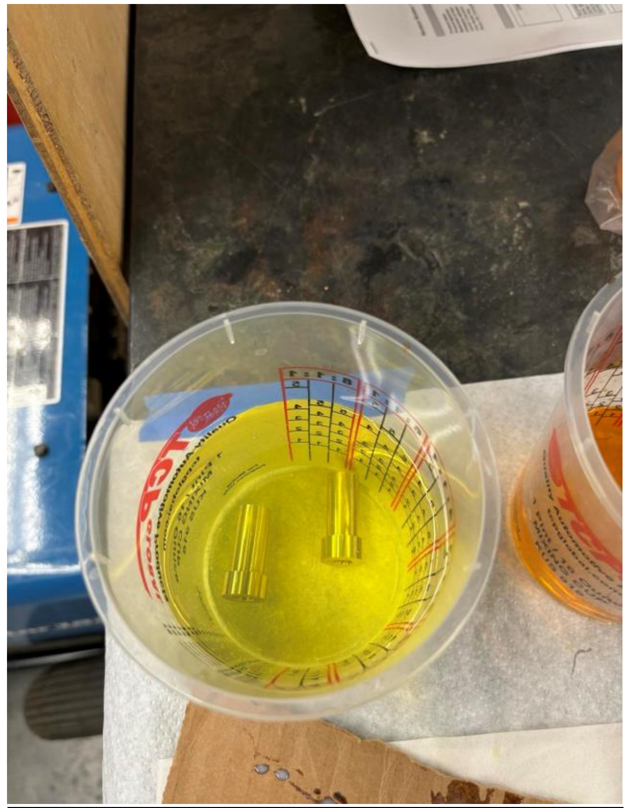

1

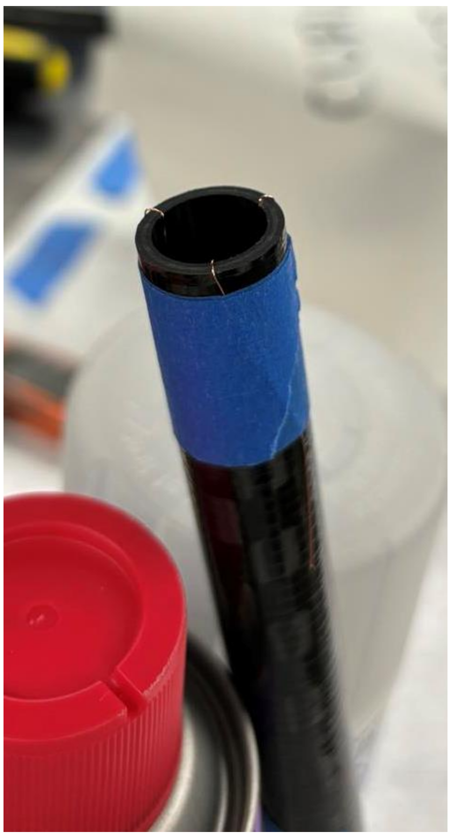

MIMO
EUROSENSOR


Olsen



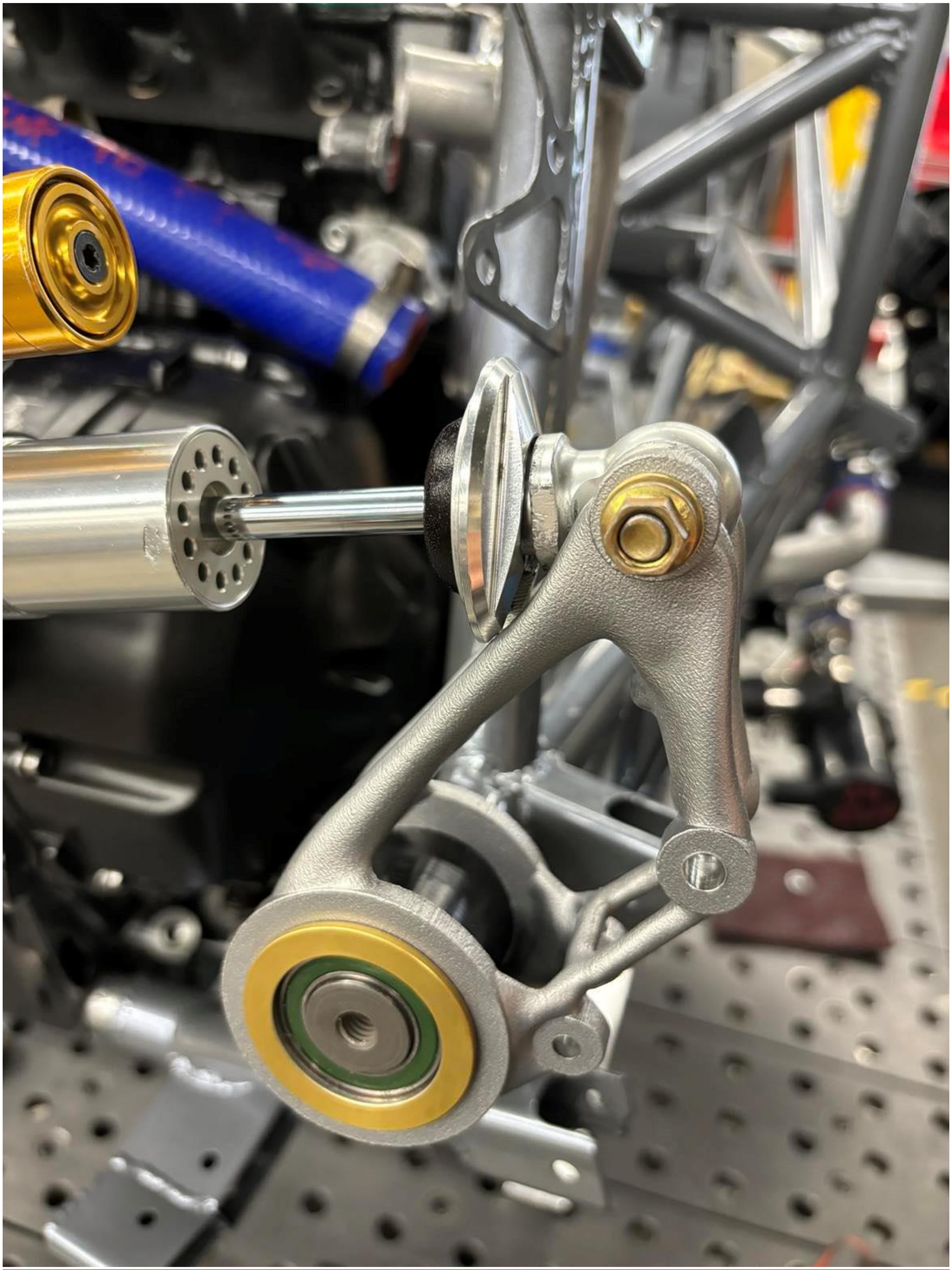

BOEING
COASTAL

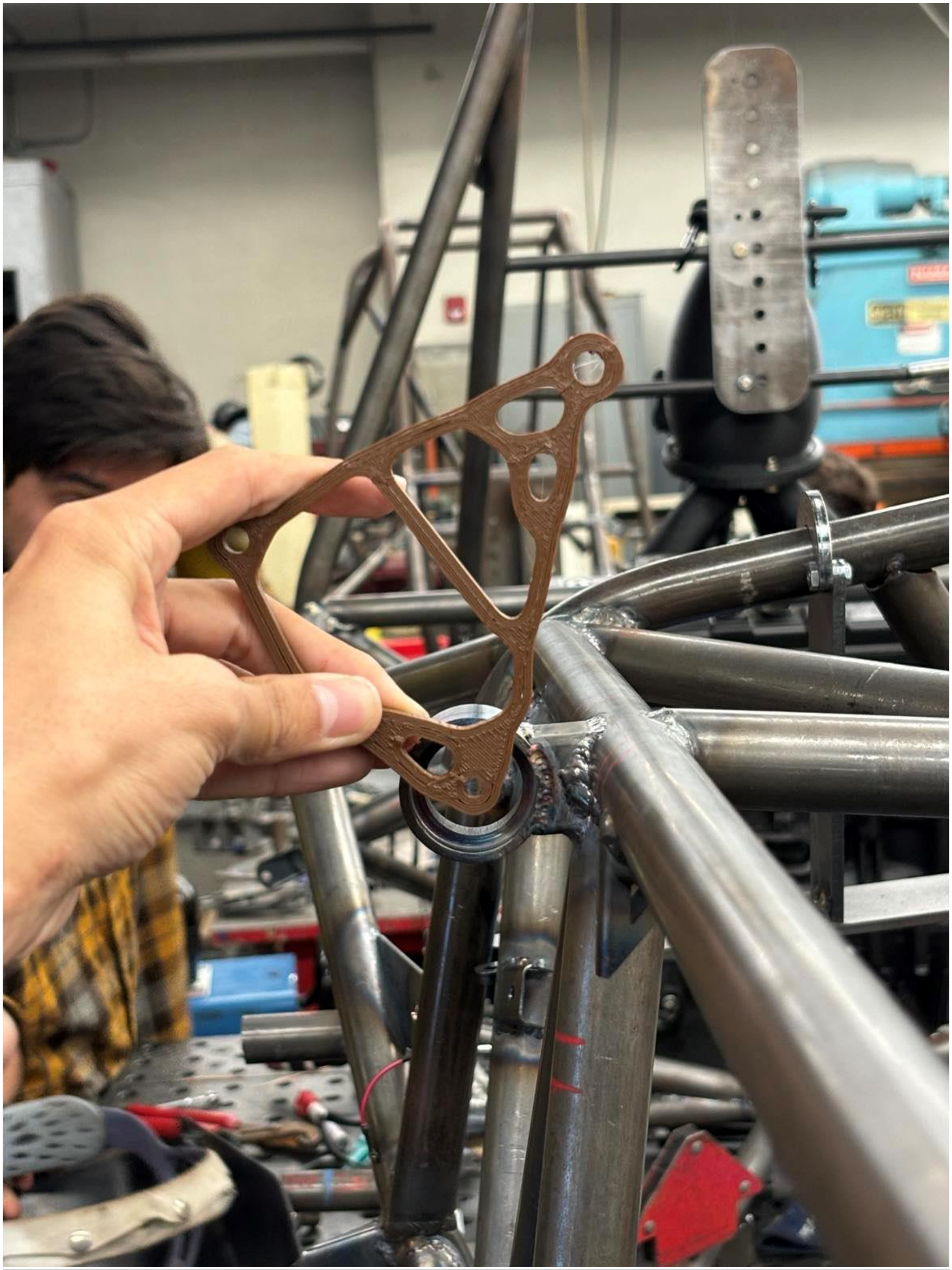


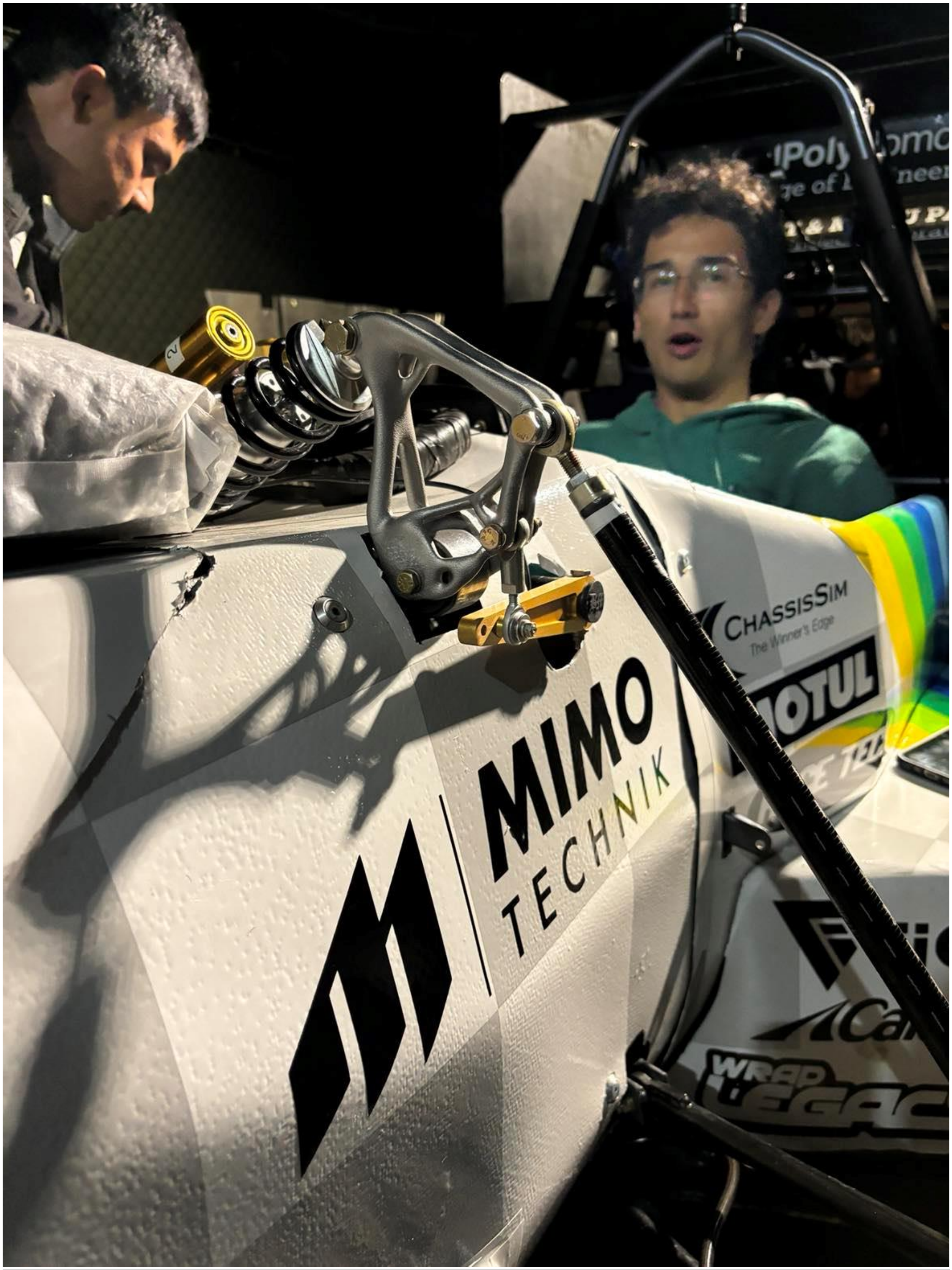



142
TARE
UNIT
HOLD
TAYLOR


121
TARE
UNIT
HOLD
TAYLOR



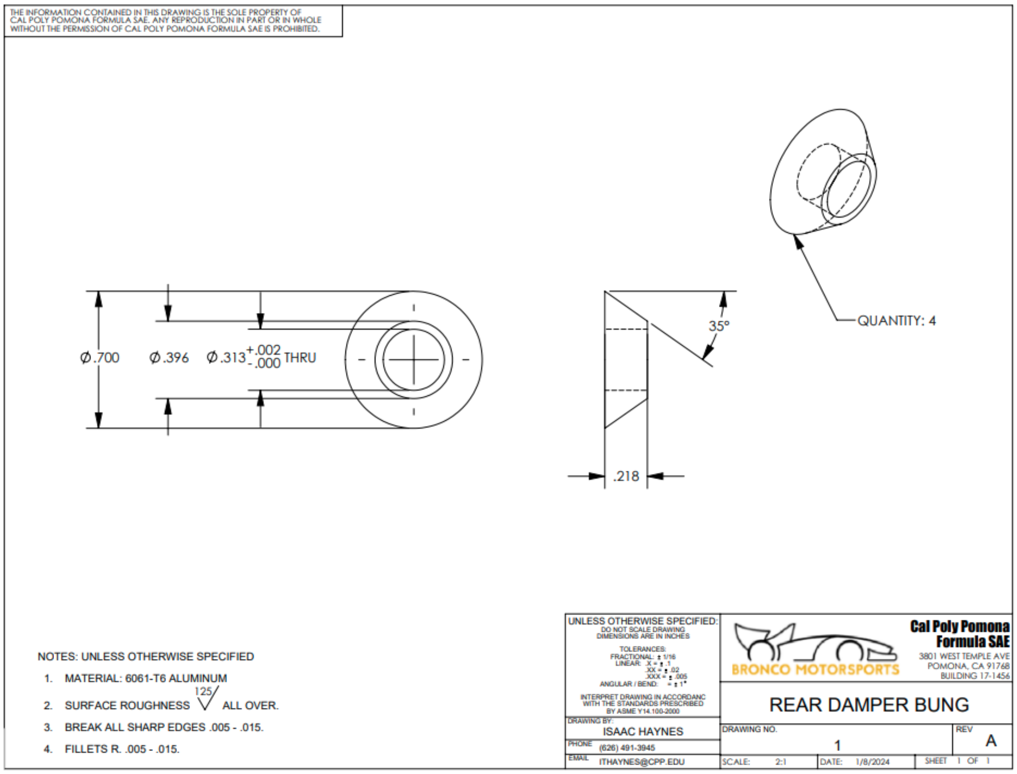


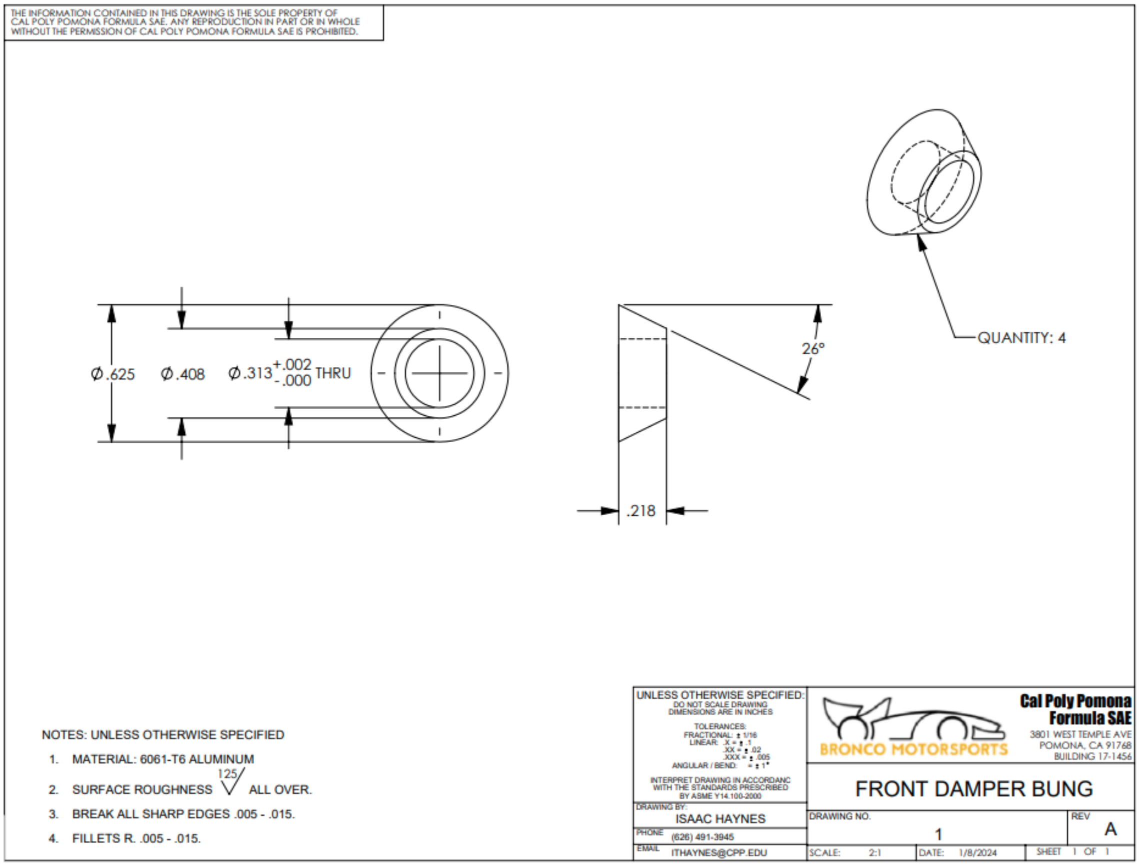

CHASSISSIM
The Winner's Edge
MOTUL
MIMO
TECHNIK

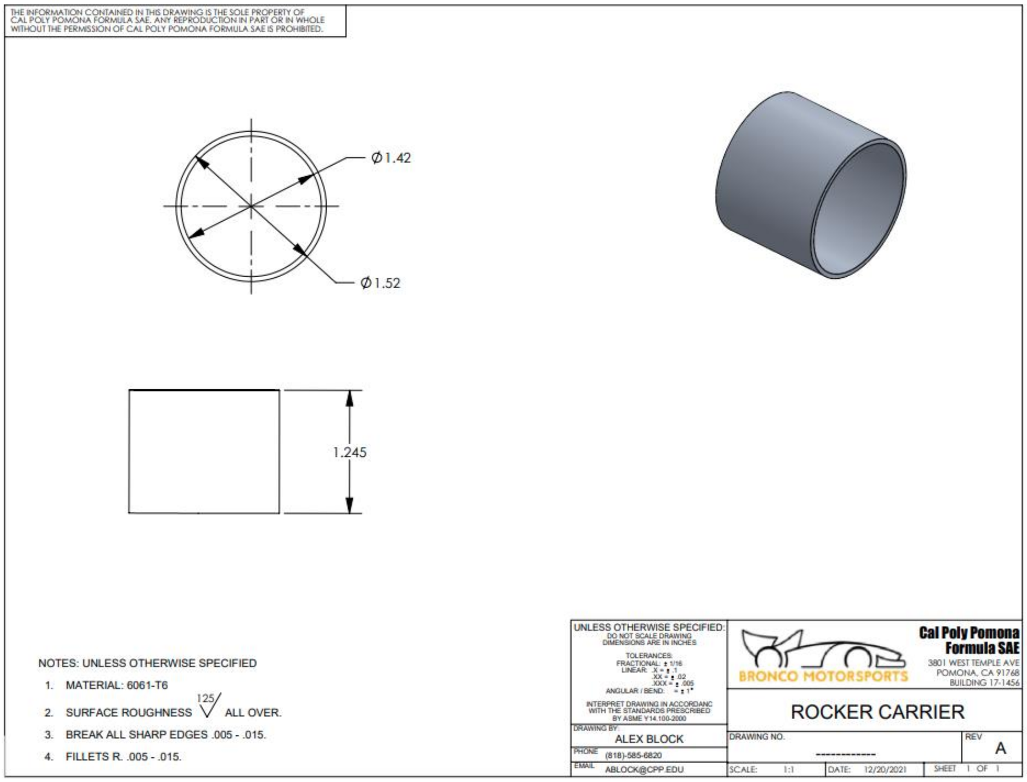

MODE

<u>Engineering Drawings</u>

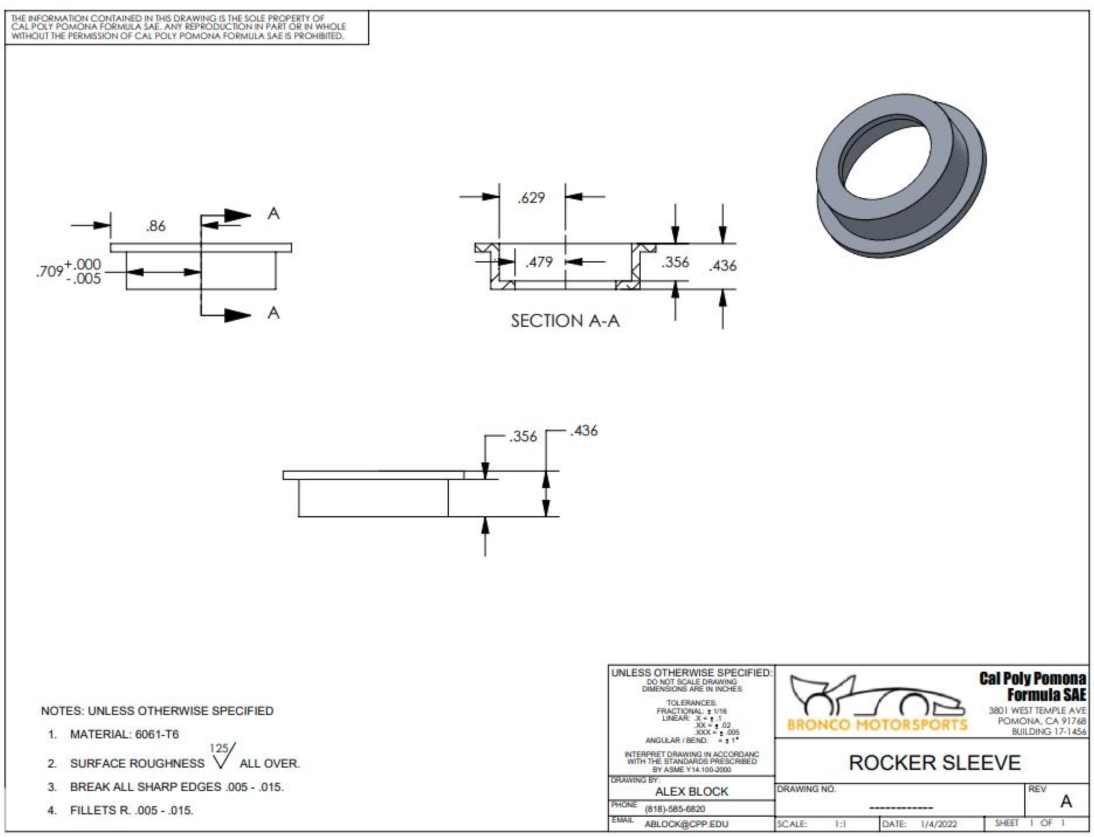
*Not all components shown; only components shown are newly machined for 2024 year



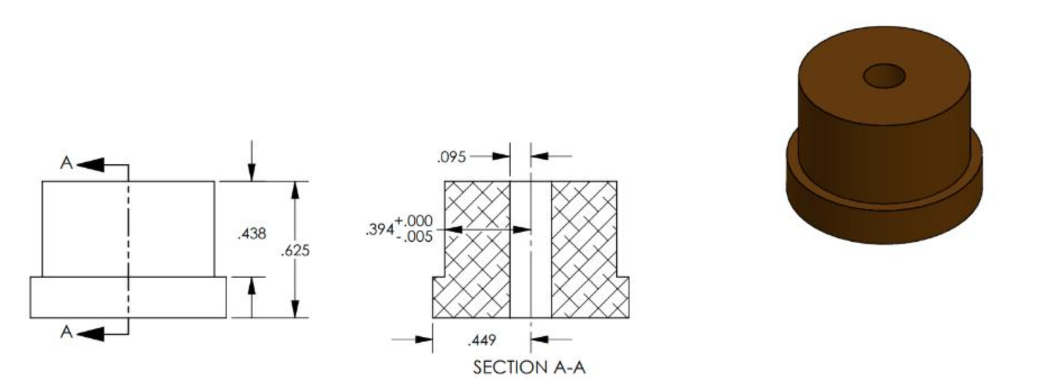




SOLIDWORKS Educational Product. For Instructional Use Only.



SOLIDWORKS Educational Product. For Instructional Use Only.



NOTES: UNLESS OTHERWISE SPECIFIED

1. MATERIAL: TEFLON
2. SURFACE ROUGHNESS 125 ALL OVER.
3. BREAK ALL SHARP EDGES .005 - .015.
4. FILLETS R. .005 - .015.

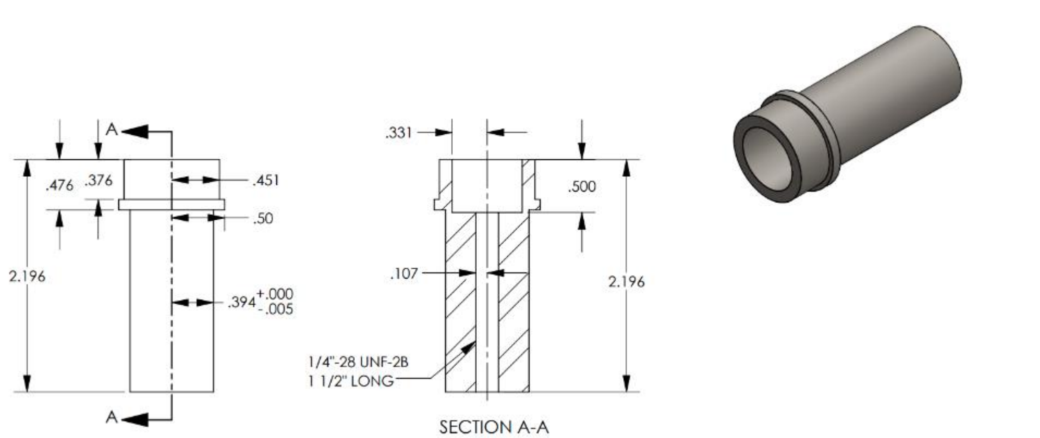
UNLESS OTHERWISE SPECIFIED:
DO NOT SCALE DRAWING
DIMENSIONS ARE IN INCHES
TOLERANCES:
FRACTIONAL: ± 1/16
LINEAR: X = ± .1
.XX = ± .02
.XXX = ± .005
ANGULAR / BEND: = ± 1°
INTERPRET DRAWING IN ACCORDANC WITH THE STANDARDS PRESCRIBED BY ASME Y14.100-2000

BRONCO MOTORSPORTS

Cal Poly Pomona Formula SAE
3801 WEST TEMPLE AVE
POMONA, CA 91768
BUILDING 17-1456

FRONT TEFLON INSERT

DRAWING BY:	ALEX BLOCK	DRAWING NO.	------------	REV	A
PHONE	(818)-585-6820				
EMAIL	ABLOCK@CPP.EDU	SCALE: 2:1	DATE: 1/4/2022	SHEET 1 OF 1	



NOTES: UNLESS OTHERWISE SPECIFIED

1. MATERIAL: 4130 STEEL
2. SURFACE ROUGHNESS 125 ALL OVER.
3. BREAK ALL SHARP EDGES .005 - .015.
4. FILLETS R. .005 - .015.

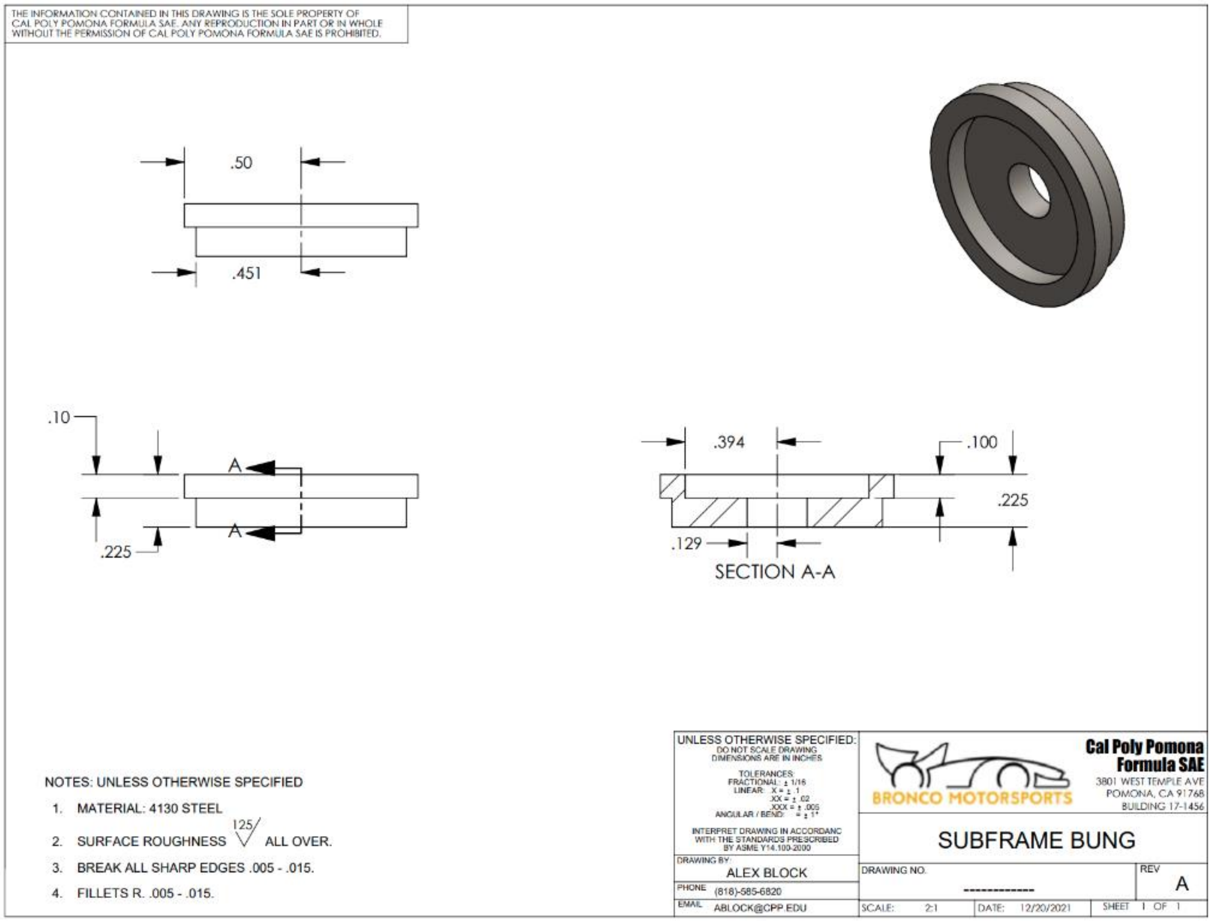
UNLESS OTHERWISE SPECIFIED:
DO NOT SCALE DRAWING
DIMENSIONS ARE IN INCHES
TOLERANCES:
FRACTIONAL: ± 1/16
LINEAR: X = ± .1
.XX = ± .02
.XXX = ± .005
ANGULAR / BEND: = ± 1°
INTERPRET DRAWING IN ACCORDANC WITH THE STANDARDS PRESCRIBED BY ASME Y14.100-2000

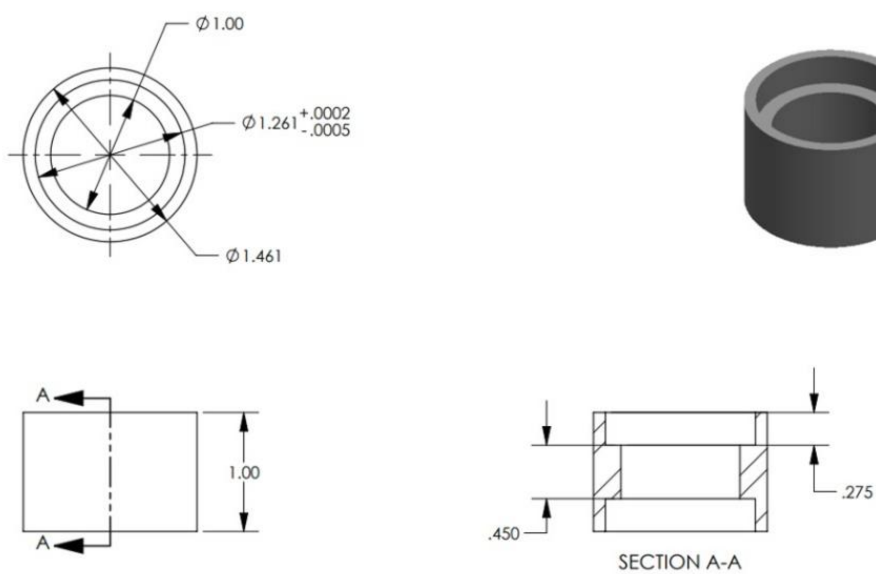
BRONCO MOTORSPORTS

Cal Poly Pomona Formula SAE
3801 WEST TEMPLE AVE
POMONA, CA 91768
BUILDING 17-1456

ROCKER PIVOT

DRAWING BY:	ALEX BLOCK	DRAWING NO.	------------	REV	A
PHONE	(818)-585-6820				
EMAIL	ABLOCK@CPP.EDU	SCALE: 1:1	DATE: 1/4/2022	SHEET 1 OF 1	



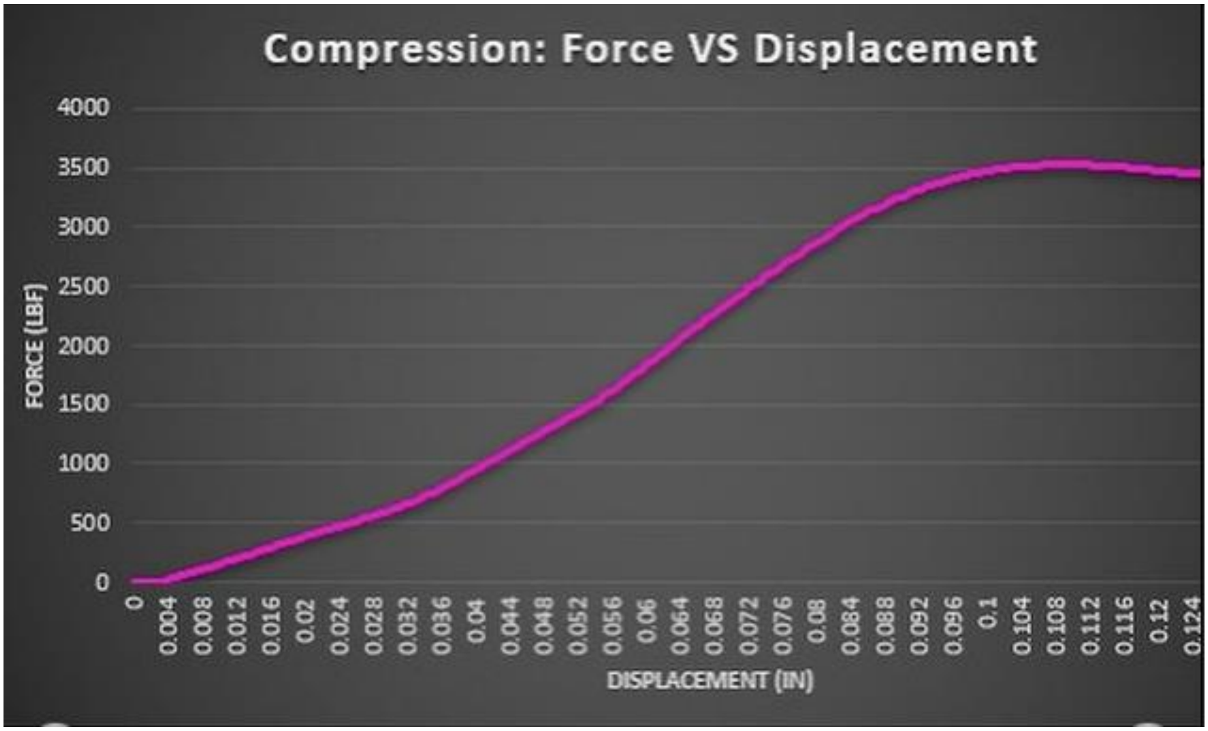


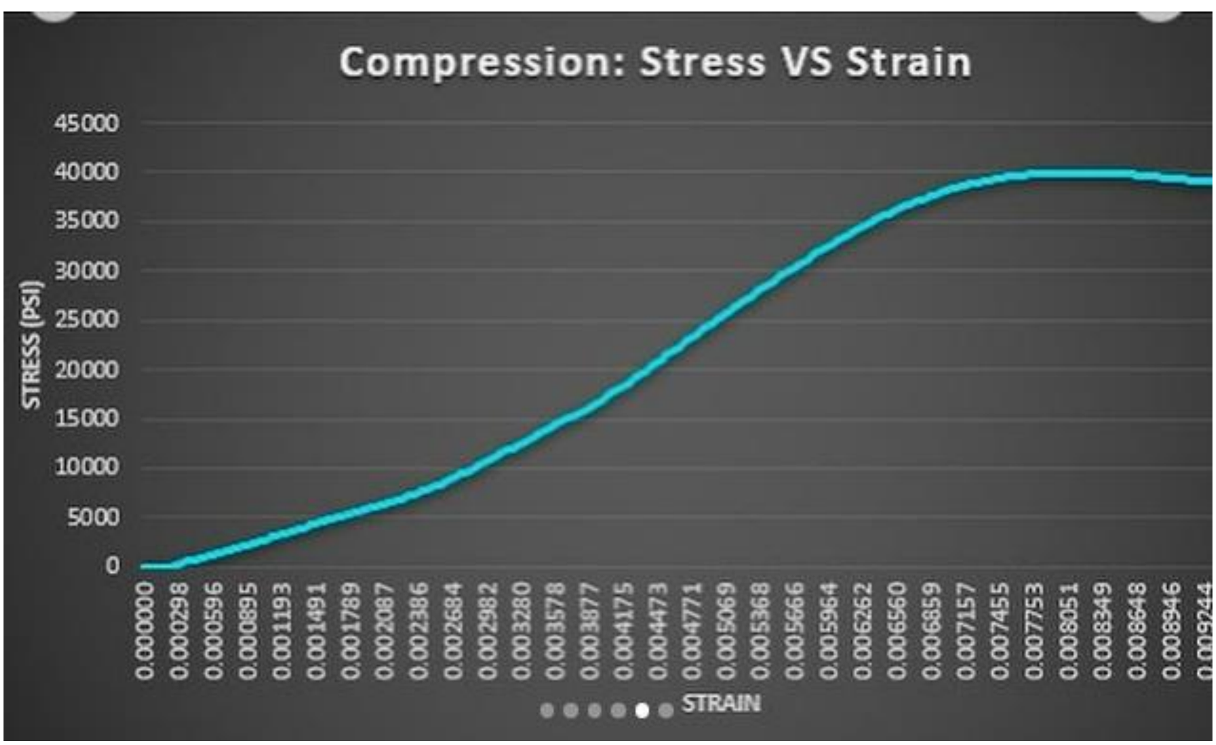
Rocker Bearing Housing

Testing

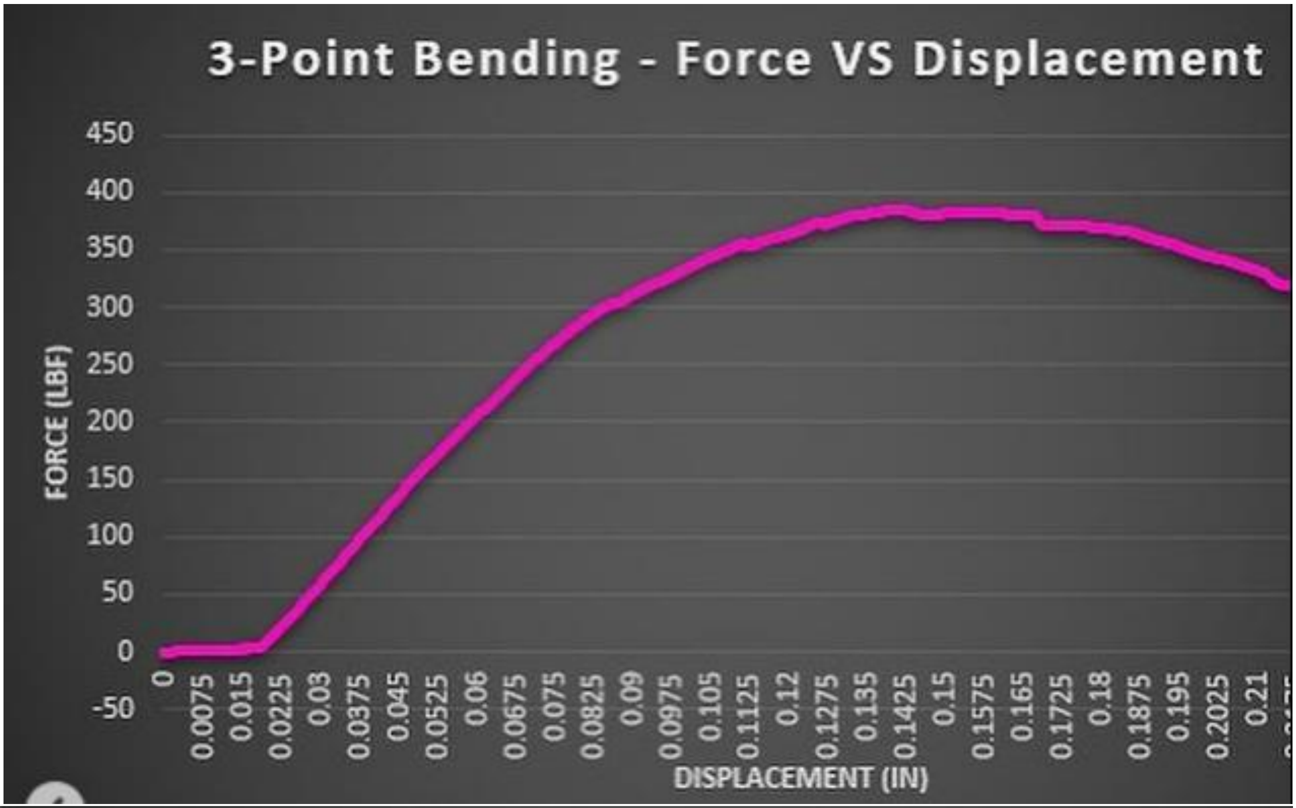
Pushrods

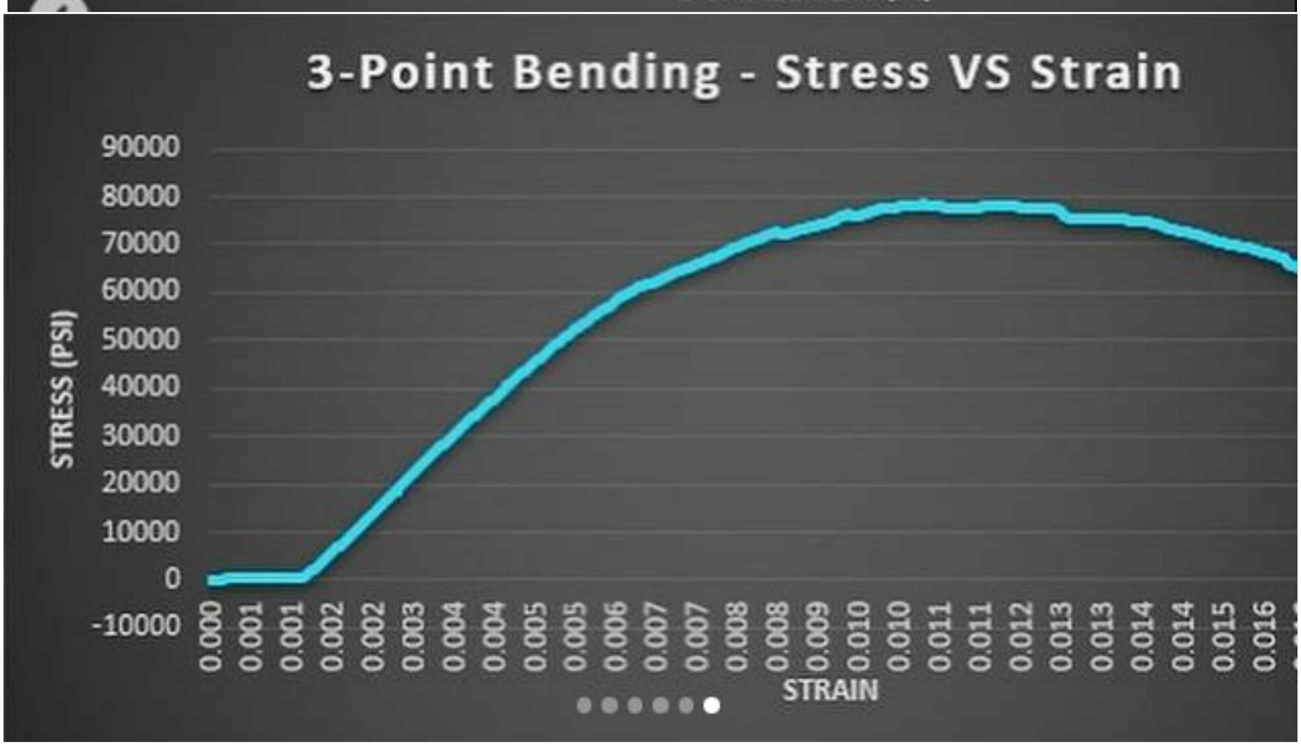
1. Axial Compression



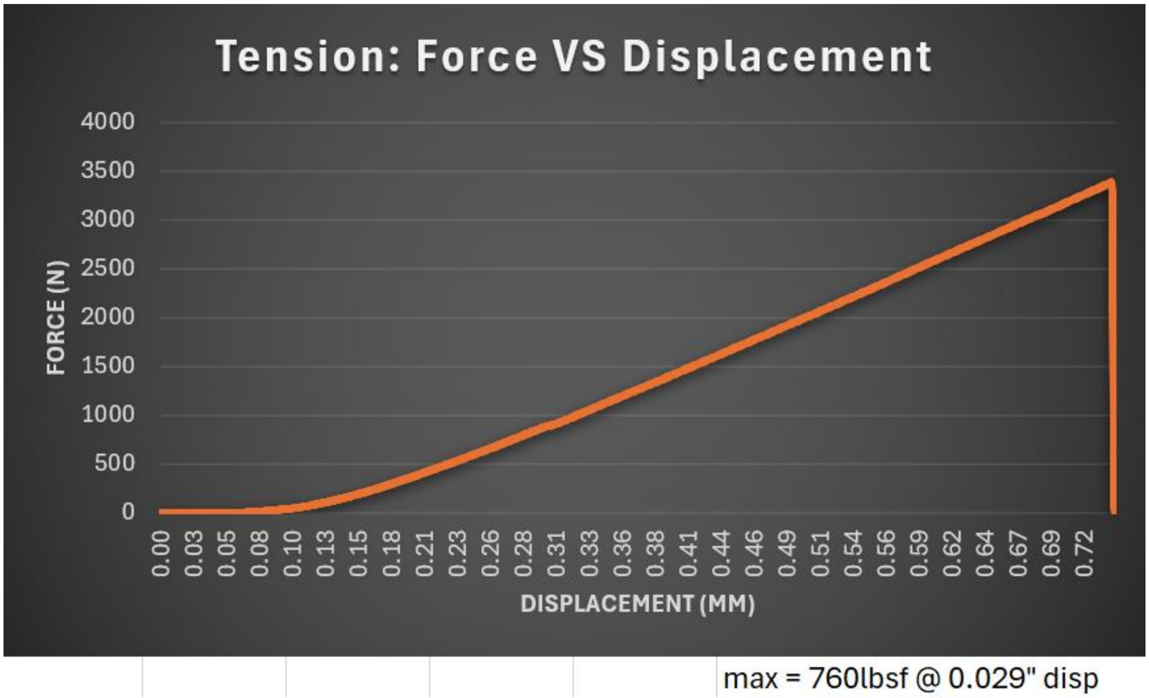


2. 3-Point Bending





3. Axial Tension

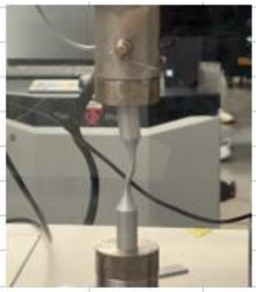


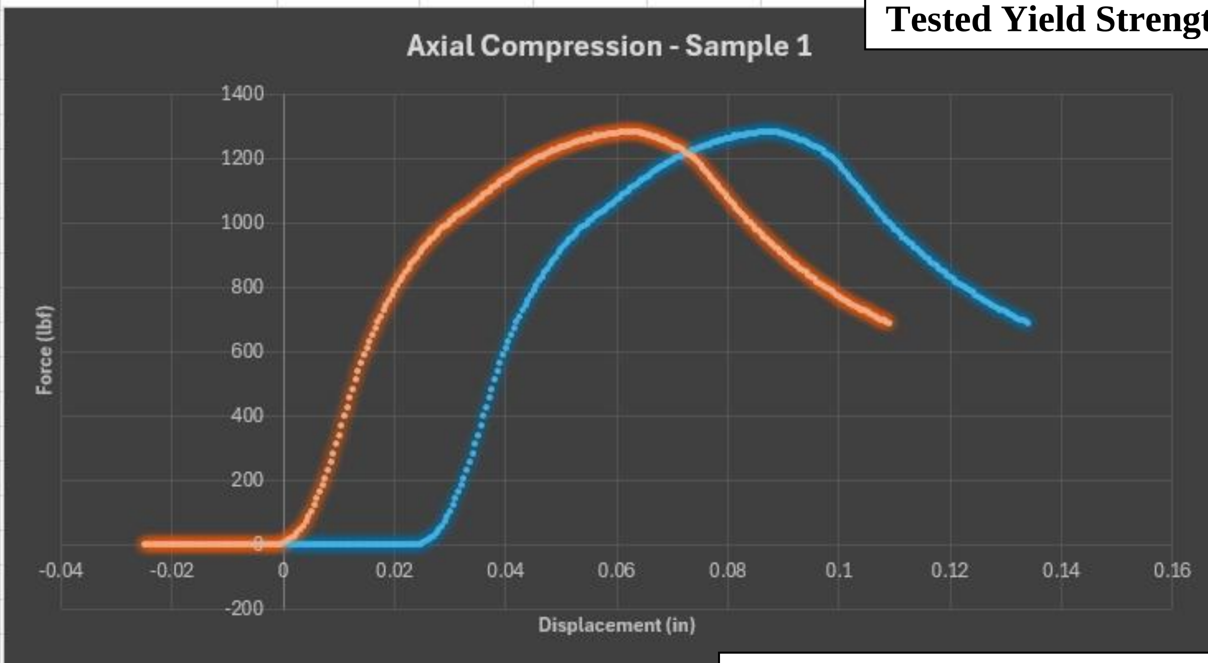
Rocker Material Testing – AlSi10Mg

Rated Yield Strength: 28ksi

Tested Yield Strength: 21ksi

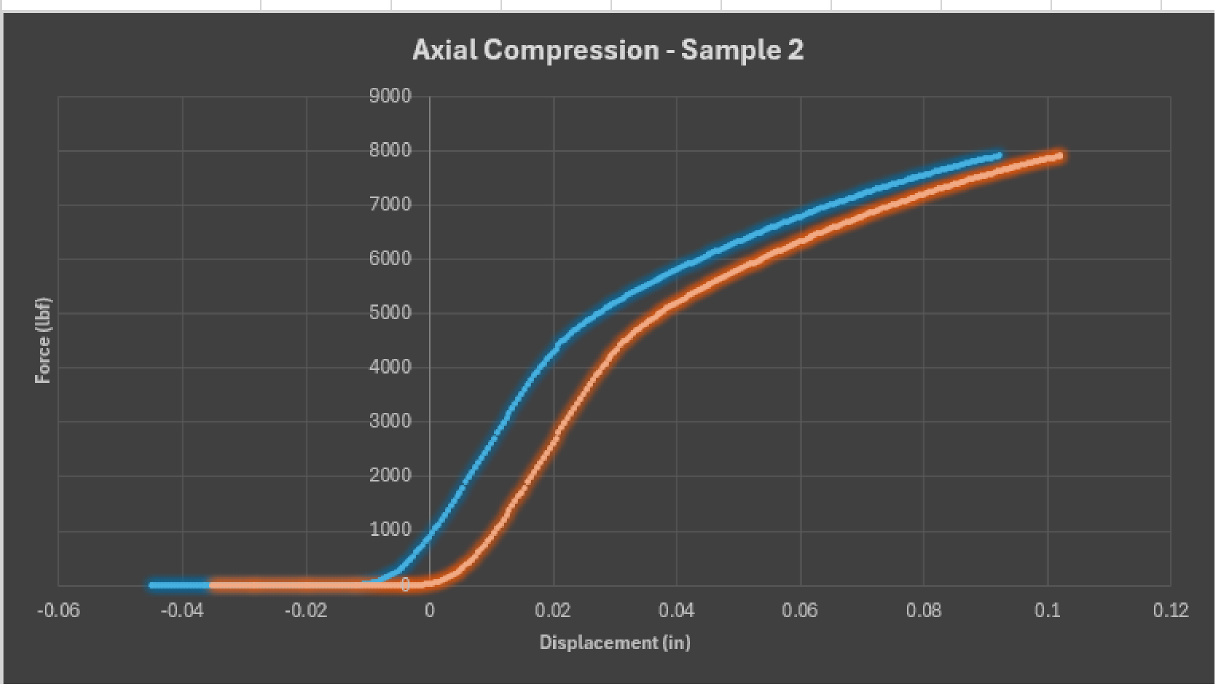
X-Sec Area (in^2)	0.02984963
Max force @yield (lbf)	632.117
Yield Strength (ksi)	21.177
Ultimate Strength (ksi)	43.032
Stiffness (lbf/in)	
Young's Modulus (ksi)	
Original Length (in)	
Disp @ Yield (in)	0.0155





X-Sec Area (in^2)	0.1846516
Max force @yield (lbf)	7904.929
Yield Strength (ksi)	21.328
Ultimate Strength (ksi)	42.810
Stiffness (lbf/in)	
Young's Modulus (ksi)	
Original Length (in)	
Disp @ Yield (in)	0.0274

*3D-printed rocker testing can really only be achieved indirectly through load cells or strain gauges on the rocker or pushrod itself. This may be in the works over the summer, but I'd highly recommend having this properly implemented onto the rockers for next year. Otherwise, we have little to no true validation that any of our hand calculations or simulations are reasonably correct.



Conclusion

Reflecting on an eventful year as Rockers and Pushrods Captain, there have been huge strides made this year with how we manufacture the subsystem. With the immense support of Mimotechnik, we were able to 3D-metal print the rockers and gain much testing with the material they are made out of. Additionally, this has been the first year we're running carbon tubing for our pushrods and toe links. The ability of the carbon tubing to be 60% the weight of its steel counterpart and still be sufficiently stiff is a highly desirable characteristic and one that I hope continues in the future for the team. Overall, I'm very satisfied with how I've gone about the year in terms of design and manufacturing of my subsystem. I plan on truly finalizing this design binder after summer competitions. In the future, I'd highly recommend to the next captain to accomplish a few things that I was not able to get to.

1. Conduct proper fatigue analysis after developing a unique SN-curve for 3D-printed metal with Andrew T.'s approach to analysis
2. Expand on carbon tube testing. There are several other parameters to test that can drastically increased bond strength and reduce weight
3. Look into manufacturing the rockers out of Scallmoy (a 3D-printed metal alloy with the same Yield strength as 7075 Aluminum)
4. Redesign and justify newly chosen rocker pick-up points
5. Look into mounting the front pushrods to the bottom a-arm end instead of the top.
6. Implement load cells on pushrods and strain gauges on rockers F/R to validate all hand calcs

As some final words, truly enjoy the time you have on the Formula team here at CPP. You've got so much at your fingertips in this program so make the most out of every Design Review, Shop Day, team outing, and embrace the grind. Most importantly, be kind, hardworking, humble, and if you think there's a chance on new concept, take that risk and make it a reality.

Sincerely,

Isaac Haynes

Auxiliary

@haynesengineering

Advice to the next captain

1. Understand WHY you chose the PUPs. I wish I had done more of this analysis when I started out.
2. Metal-print again and stay in good relationship with MIMO
    a. If these geometries and pushrods get through a full year w/ no problems, look to lighten even more.
3. Make you own IG engineering account to motivate you to get the job done while enjoying it along the way.

Works Cited

[1] – Alban, Cobi, MIT, Design of a Carbon Fiber Suspension System for FSAE Applications

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