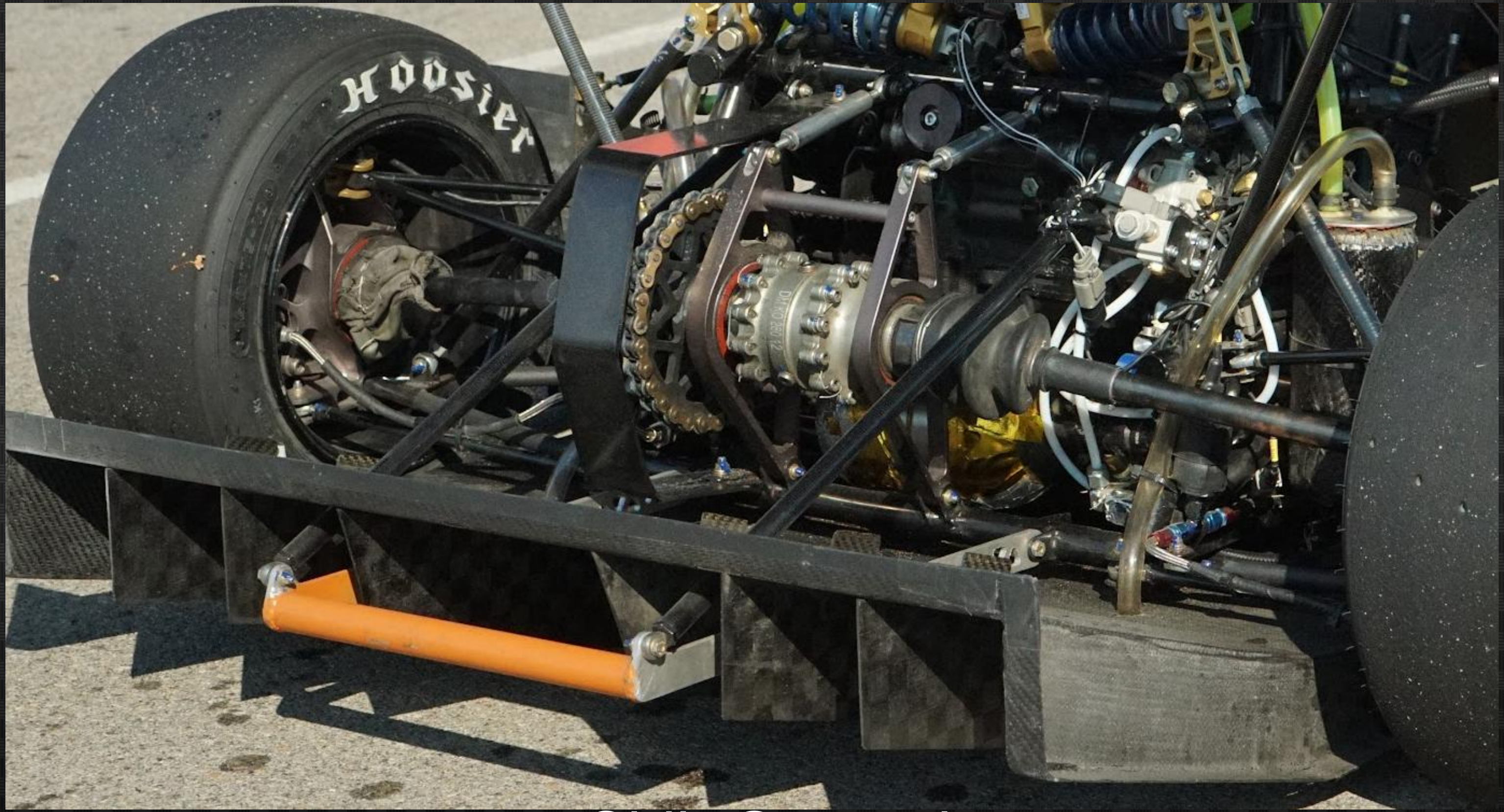
Purdue FSAE – Drivetrain CDR



Sidh Gurnani

10/02/2024

Table of Contents

Changes Since Last Review

Changes Since Last Review

- Changes since PDR:
  - CAD
    - Diff Carrier Cutouts
    - Modeled max chain stretch
      - Done by expression, where the top node, diff axis move when the turnbuckle is "extended"
    - Temporary Tabs in Layout
  - Analysis
    - Bolt + Lug Calcs
    - Final FEA → Sprocket, CV Inserts
      - WIP → Turnbuckles and Diff Carriers

Changes Since Last Review

- Changes since PDR (by component):
  - Final Drive Ratio
    - FDR set at 31/12
  - Differential
    - Differential moved 0.27 in forward to ensure an even number of links in chain
  - Sprocket
    - Revised boundary conditions in FEA
    - Mesh Convergence
    - Mass optimization
  - Diff Carriers
    - Ran individual FEA + combined FEA
    - Mesh Convergence in progress
    - Mass Optimization in progress

Changes Since Last Review

- Changes since PDR (by component):
  - CV Inserts
    - Material changed to Custom 465
    - FEA + Hand Calculations
    - Reoriented to work with Hubs CAD
  - Turnbuckles
    - Optimized angle to reduce loading
    - FEA + Hand Calculations
  - Chain Guard
    - No changes made since PDR due to allocation of my resources elsewhere

Changes Since Last Review

- Changes since PDR (by component):
  - Misc
    - Hardware in CAD
    - Tabs in CAD
    - Mass Properties in CAD
    - Loads appropriately sourced for bolt/lug calcs
      - Will dive later into this…

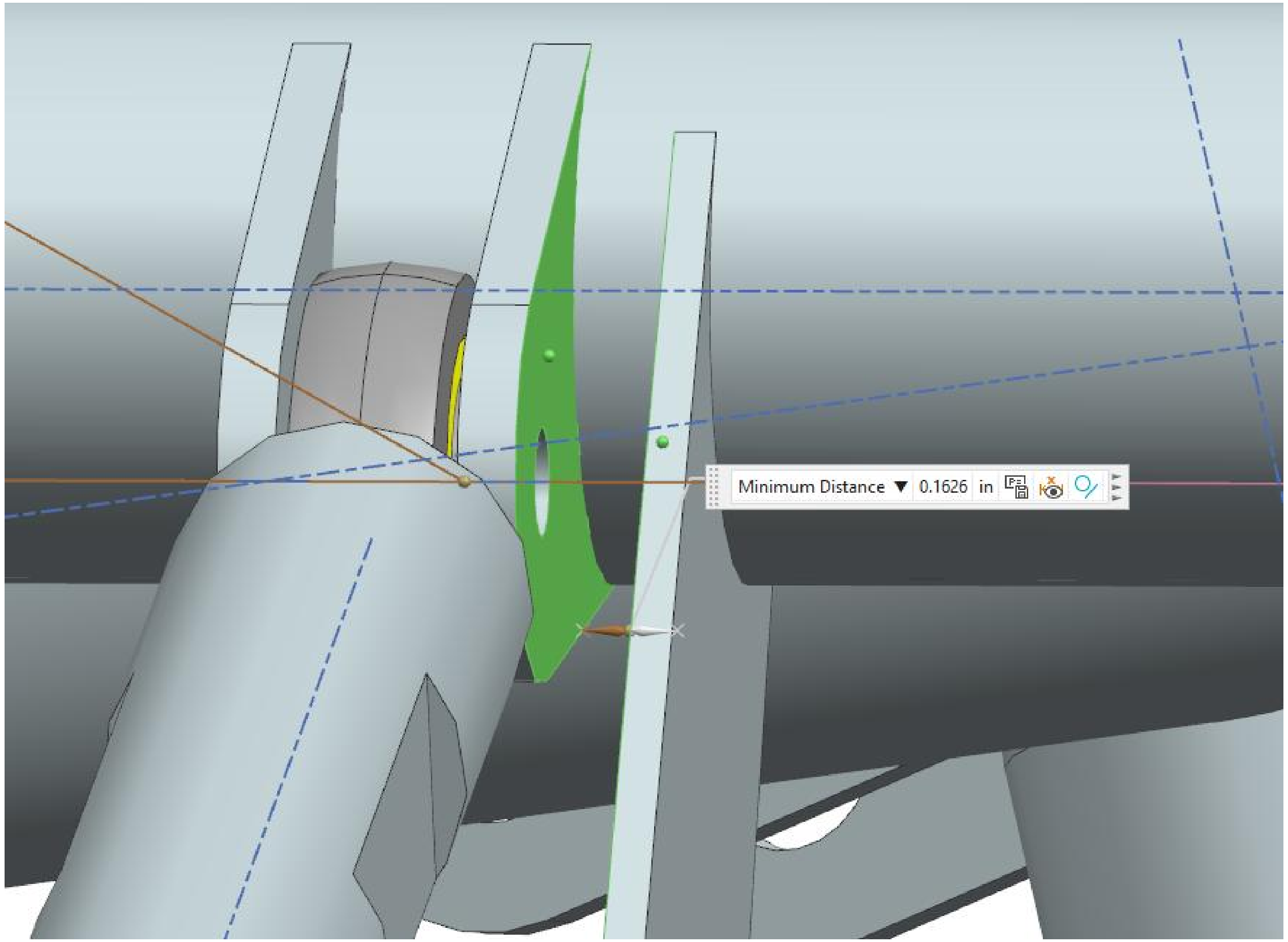
Changes Since Last Review

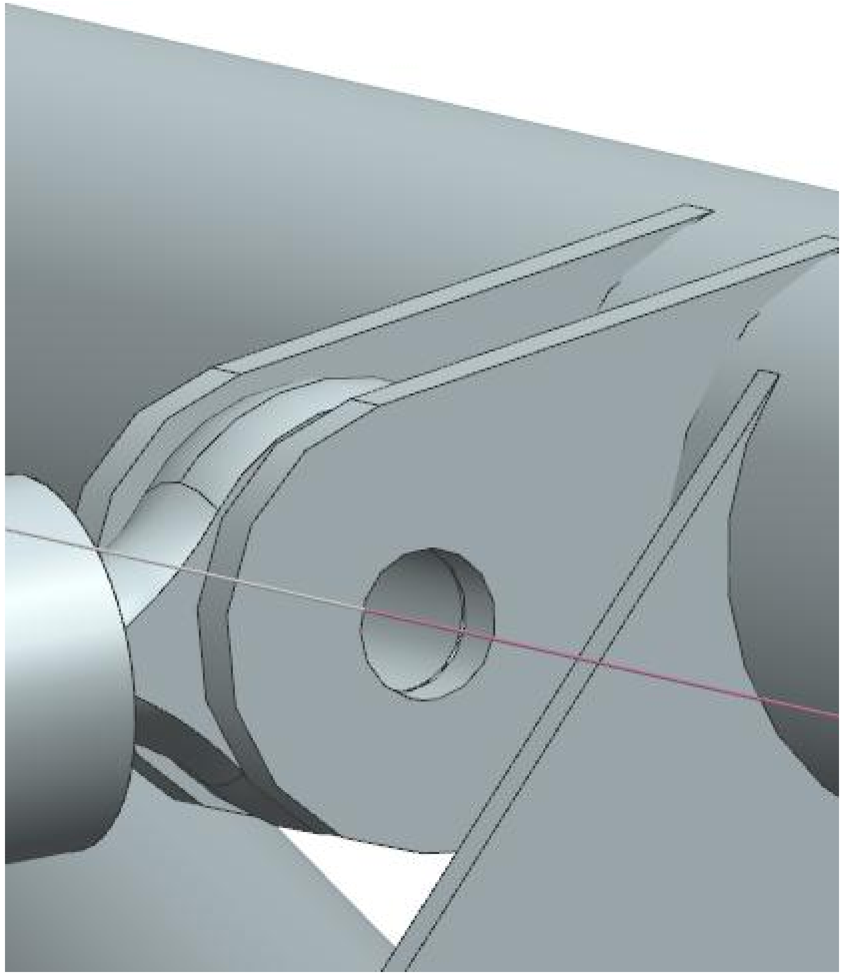
- PDR Feedback:
  - Check max accel g forces, maybe not 1.65
    - Sticking with 1.65, went back and forth about it while deriving CV Insert loading
  - Margin on sprocket is sus yield or ultimate?
    - Shear stress SF → ultimate
    - Compressive SF and Tensile Yield → yield
  - Margin reporting is confusing why is it this way?
    - This mainly has to do with an “allowable” value not being properly defined in some situations (i.e. spline stresses) thus SF is used instead
  - How to determine amount of press fit for bearings (shigleys tables)
    - For now I am sticking with 0.001 in accordance to what NTN recommends, but I believe it should be more than that, around 1.5 to 2 thou.

Changes Since Last Review

- PDR Feedback (cont'd):
  - Why are we using 1.2 on carriers
    - The SF for carriers can be lowered to 1.1 yield and 1.4 ultimate, but the sprocket is something I have designed to 1.4 ultimate (and 1.2 yield)
  - Engine and carrier tab is concerning
    - Should be resolved now after playing around with different tab angles
  - Rod ends?
    - Remains an open item for now as the most realistic option is to use the ones from McMaster Carr

Changes Since Last Review



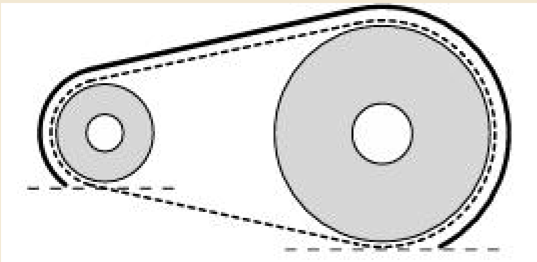


Rules

Rules

Rule	Exact Text	Requirements From Rule
T.5.1	Any transmission and drivetrain may be used.	Any configuration can be used as team sees fit. Must comply with rest of rules below.
T.5.2.1	Exposed high speed final drivetrain equipment such as Continuously Variable Transmissions (CVTs), sprockets, gears, pulleys, torque converters, clutches, belt drives, clutch drives and electric motors, must be fitted with scatter shields intended to contain drivetrain parts in case of radial failure.	Drivetrain system must contain "shield" (i.e. Chain Guard) to prevent exposed parts from scattering (i.e. Sprocket, Chain).

Rules

Rule	Exact Text	Requirements From Rule
T.5.2.2	The final drivetrain shield must: a. Be made with solid material (not perforated) b. Cover the chain or belt from the drive sprocket to the driven sprocket/chain wheel/belt or pulley c. Start and end no higher than parallel to the lowest point of the chain wheel/belt/pulley d. Cover the bottom of the chain or belt or rotating component when fuel, brake lines T.3.1.8, control, pressurized, electrical components are located below	Final guard must be made of a solid material and cover the sprocket-chain system as shown in the image. 
T.5.2.3	Body panels or other existing covers are acceptable when constructed per T.5.2.7 / T.5.2.8.	Body panels can be used as part of the guard system if constructed to rules specification.
T.5.2.4	Frame Members or existing components that exceed the scatter shield material requirements may be used as part of the shield.	Existing frame members and/or components can be used as long as they EXCEED material requirements

Rules

Rule	Exact Text	Requirements From Rule
T.5.2.5	Scatter shields may be composed of multiple pieces. Any gaps must be small (< 3 mm).	Guard system can be made of multiple components if gaps are small (i.e. less than 3 mm).
T.5.2.6	If equipped, the engine drive sprocket cover may be used as part of the scatter shield system.	OEM guards can be used.
T.5.2.7	Chain Drive - Scatter shields for chains must: a. Be made of 2.66 mm (0.105 inch) minimum thickness steel (no alternatives are allowed) b. Have a minimum width equal to three times the width of the chain c. Be centered on the center line of the chain d. Stay aligned with the chain under all conditions	For chain driven systems (what we historically run), the following must be met: a. Made of 0.105 inch minimum thick steel b. Overall width equal to 3 times width of chain c. Centered along center line of chain at all times d. Stay aligned with chain under all conditions
T.5.2.9	Attachment Fasteners - All fasteners attaching scatter shields and guards must be 6 mm or 1/4” minimum diameter Critical Fasteners, see T.8.2	All fasteners used in guard system must be in accordance to T.8.2

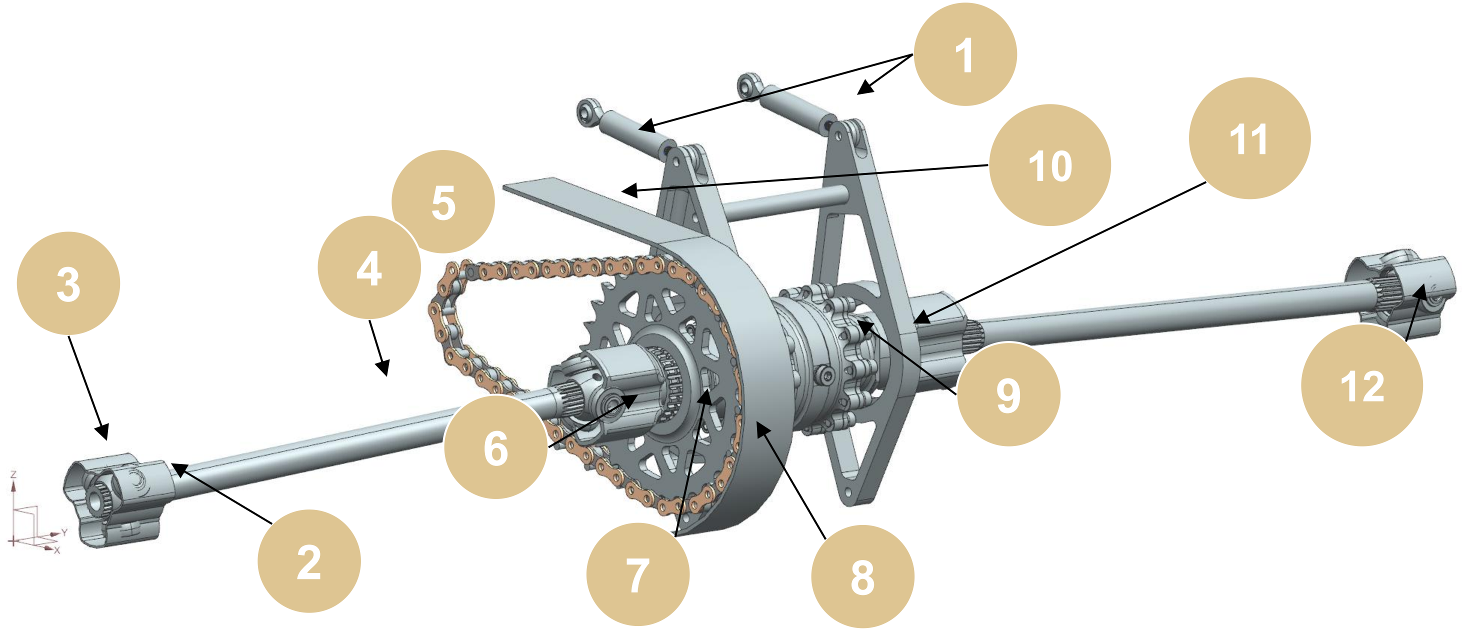
Rules

Rule	How Satisfied?
Chain Guard Width	

Work in Progress

CAD

CAD



1. Turnbuckles
2. CV Inserts
3. CV's (Tripods)
4. Halfshafts
5. Front Sprocket
6. Chain
7. Rear Sprocket
8. Differential Carriers
9. Differential
10. Chain Guard
11. Inboard Tripod Housing (Tulips)
12. Boots (not pictured)

Mass

Mass

Component	PF24 Masses (lbm)	PF25 Masses (lbm)
Left Carrier	0.6012	0.8080
Right Carrier	0.3445	0.5894
Turnbuckles	0.1853	0.2455 (body + rodends) x2
Stability Bar	0.0893	~0.0893
Sprocket	0.447	0.5093
Sprocket Spacer	0.069	~0.069
Chain	1.5742	~1.43 (blind guess)
Chain Guard	2	~2 (WIP)
Tripods	1.32	1.32
CV Inserts	0.3628	0.3628
Halfshafts	3.99	3.1055
Differential	10.85	10.85
Hardware	0.2624275	~0.25 (no towbar)
Tabs	1	~1
Total	[illegible]	[illegible]

Take with a grain of salt

Philosophy

Philosophy

- Driver Centered Design
  - Using a differential for ease of driving as well as predictability
  - Designed, to an extent, with serviceability to maximize track time
- Low, Light, Tight
  - Differential moved as far forward and down as possible without interference to the frame (from the halfshafts)
  - Mass optimization on sprocket, diff carriers carried out

Philosophy

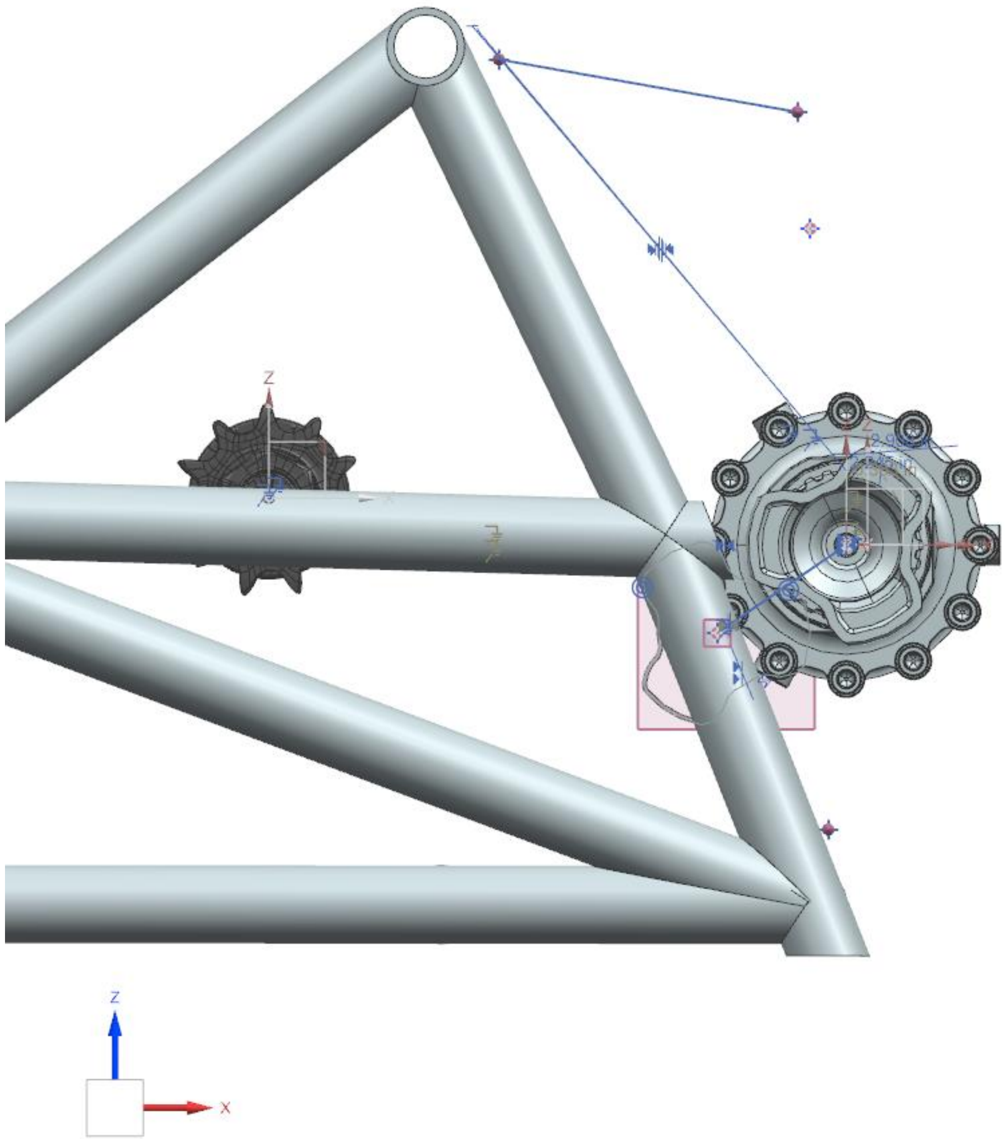
- Vehicle Goals:
  - Driver Safety
    - System designed to a factor of safety which is AT THE MINIMUM team standards, with conservative estimates and setups to add a buffer
  - System Reliability
    - Utilize combination of sprockets that minimizes wear on sprockets
    - Replace components as needed, determined by visual inspection
  - Max Comp Points
    - Mass reduced where possible, while ensuring safety standards
    - Accommodate FDR to allow team to be as competitive as possible
    - Positioning differential closer to CG
  - Driver Confidence/Comfort
    - No major failures during lifetime of the vehicle
    - Find and set up a tune for a differential that yields no driver complaints (and one that doesn't shred the hubs)
  - Serviceability
    - Utilize chain tensioning system to adjust chain tension in under two minutes (turnbuckles)
    - Accommodating space needed for appropriate tools for servicing

Trades

Trade #1 – Diff Placement

- Differential had to be moved post-IDR to achieve an even number of links due to final drive changes
  - Option #1 – Move diff towards front of car
    - Allows even number of links to be achieved
    - Tighter packaging (may have necessitated with architecture)
      - Has an effect on CHA clearances
    - Halfshaft angles go down (and subsequently a small amount of length)
  - Option #2 – Move diff towards rear of car
    - Allows even number of links to be achieved
    - Much looser packaging
      - Diff carriers would have to be upsized
      - Turnbuckles would be longer (slightly more prone to buckling)
    - Halfshaft angles go up (and subsequently a small increase in length)

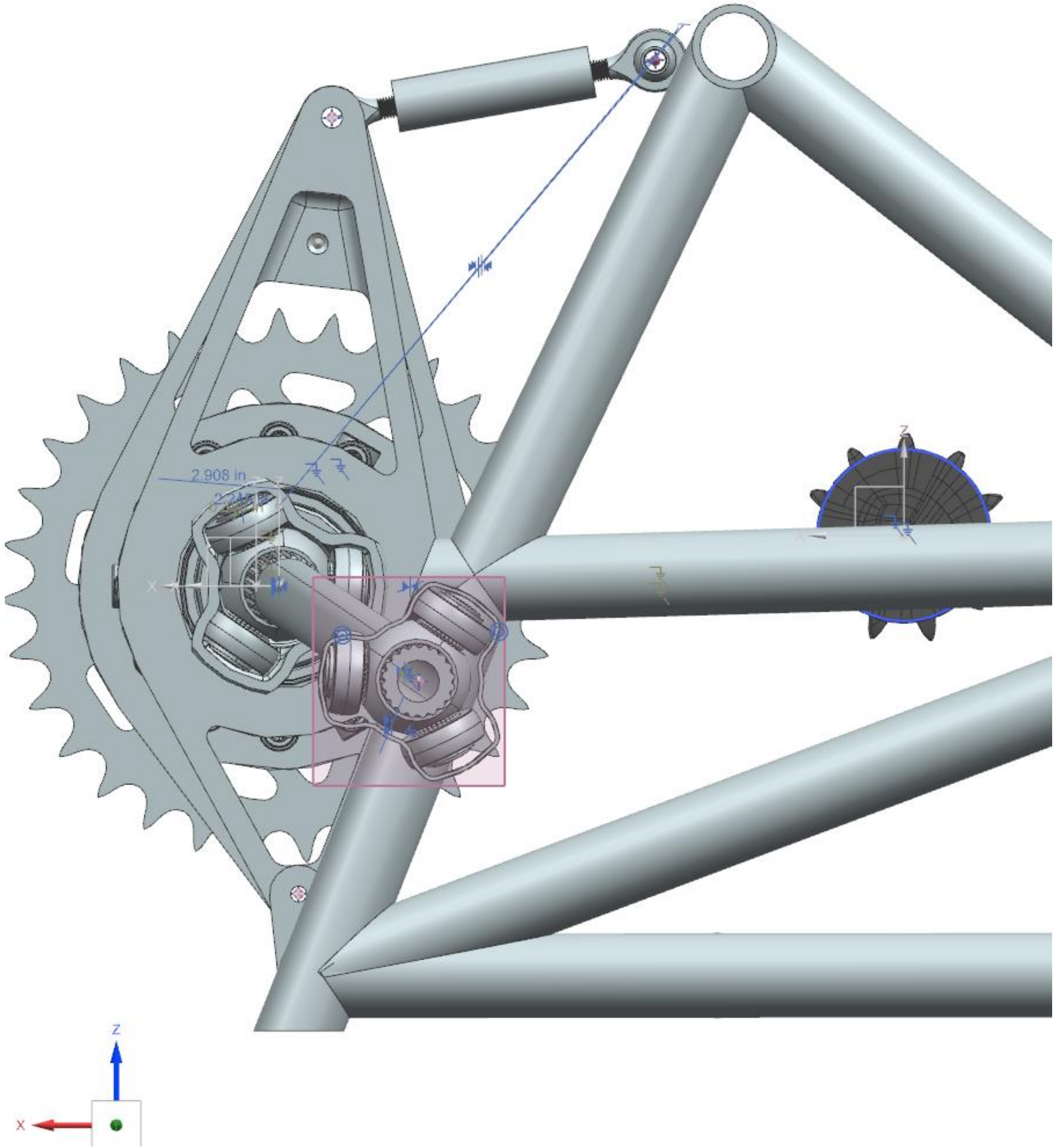
Trade #1 – Diff Placement



Trade #2 – Diff Carrier Top Node

- Placement of top node can have an affect on loading through turnbuckles and diff carriers (as well as any hardware) → study carried out on hand calcs
  - Option #1 – Move top node down
    - Smaller diff carrier → less mass optimization
    - Increases loading at the top node
  - Option #2 – Move top node up
    - Larger diff carrier → more mass to optimize
    - Decreases loading at the top node

Trade #2 – Diff Carrier Top Node



Design

Design

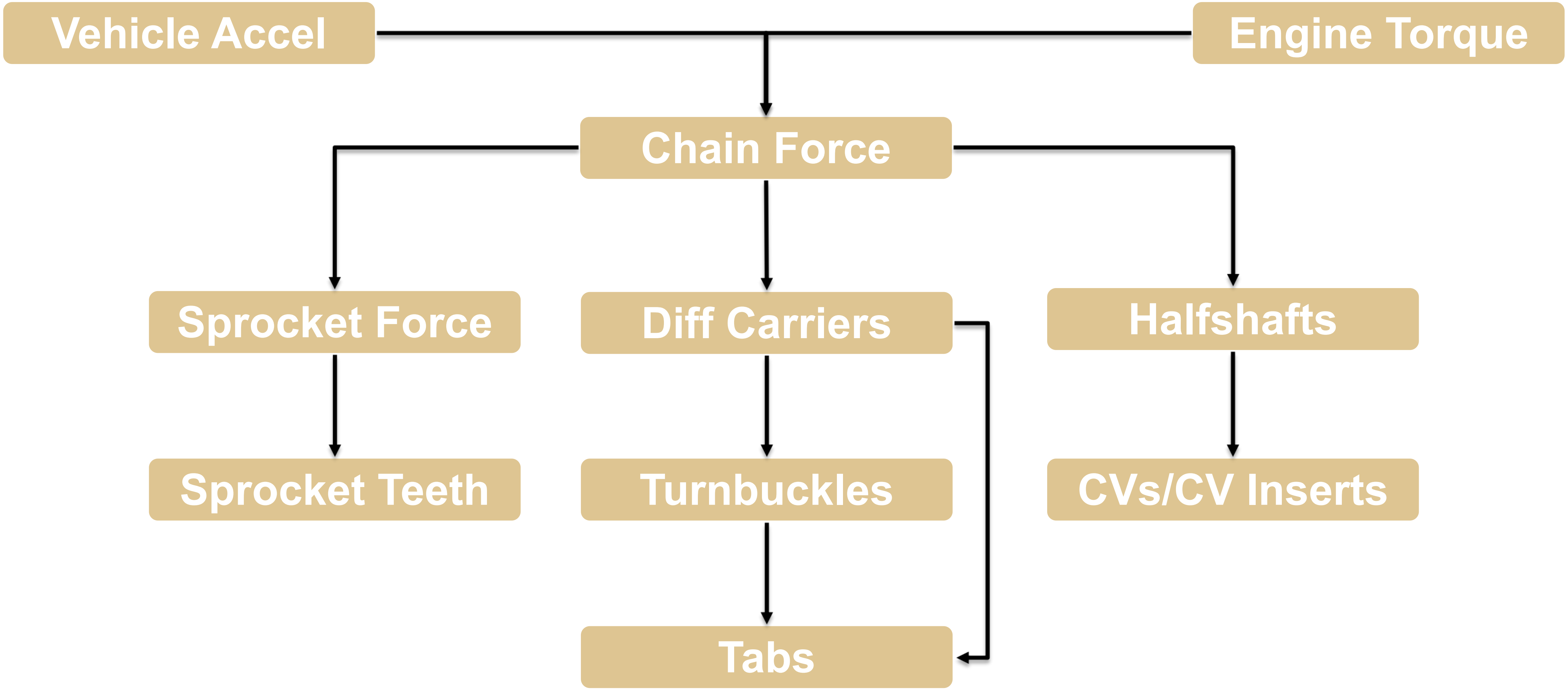
Architectural Item	Description (if needed)
Differential	Drexler Limited Slip Differential
Chain Tensioning System	Turnbuckles (custom made body with sourced rod ends) Goal is to accommodate about 3% of chain tension
CV Inserts	Custom, PF20 Special Material change to Custom 465 Steel
CV's (Tripods)	Purchased RCV
Halfshafts	Purchased RCV Lengths (L/R) – 15.95 in / 18.70 in Angles (L/R) – 7.6437 deg / 6.447 deg
Chain	DID 520 Roller Chain
Rear Sprocket	Custom made; This year modeled in CAD to resemble real life more
Differential Carriers	Custom made
Tulips	Using one short, one long and will slide into diff
Boots	Purchased (or higher fidelity custom)
Chain Guard	Custom; 0.105 in thick steel

Design

Architectural Item	Why?
Differential	Good for drivability; good history with using despite the weight it carries
Chain Tensioning System	Turnbuckles provides a simpler solution compared to the other alternatives (idler sprocket, eccentric diff carriers)
CV Inserts	Material change necessitated given new assumptions made about loading and failures from PF24
CV’s (Tripods)	Purchased RCV; used extensively
Halfshafts	Purchased from RCV (as we are using tripods from them)
Chain	Compatible with front sprockets at our disposal
Rear Sprocket	Made custom to work for a variety of situations (generally speaking) Al for machining and because of metallurgical compatibility with other metals
Differential Carriers	Made custom to work with our situation Al for machining (and budget since Arconic)
Tulips	Work with our current diff
Boots	Better sourcing avoids issues of PF24
Chain Guard	Made to rules spec

Analysis

Analysis – Flow Map



Analysis - Global Inputs

- Vehicle Parameters
  - Vehicle weight, tire diameter, max accel
- Sprocket Parameters
  - Number of teeth, FDR, pitch, etc
- Launch Parameters
  - 1st gear, 2nd gear

Analysis – Chain Force

- Three methods to determine chain force
  - Max of these three methods is taken as the load going through chain
    - PF25 → Tire Launch and 2nd Gear Inertia

Chain Load		
Method 1: Tire Launch and Average Inertia		
Tire Launch	703.828125	ft-lbf
Average Inertia	131.7441191	ft-lbf
Chain Load	3246.084167	lbf
Method 2: Tire Launch and 2nd Gear Inertia		
Tire Launch	703.828125	ft-lbf
2nd Gear Inertia	186.3839263	ft-lbf
Chain Load	3458.352363	lbf
Method 3: 1st Gear Inertia + 1st Gear Torque		
1st Gear Inertia	77.104	ft-lbf
1st Gear Torque	760.7935913	ft-lbf
Chain Load	3255.119035	lbf
Max Chain Load	**3458.352363**	**lbf**

Analysis – Sprocket Force

- Using equations (RC Binder Chain Calcs) and the input (chain force) the force on each tooth can be determined.

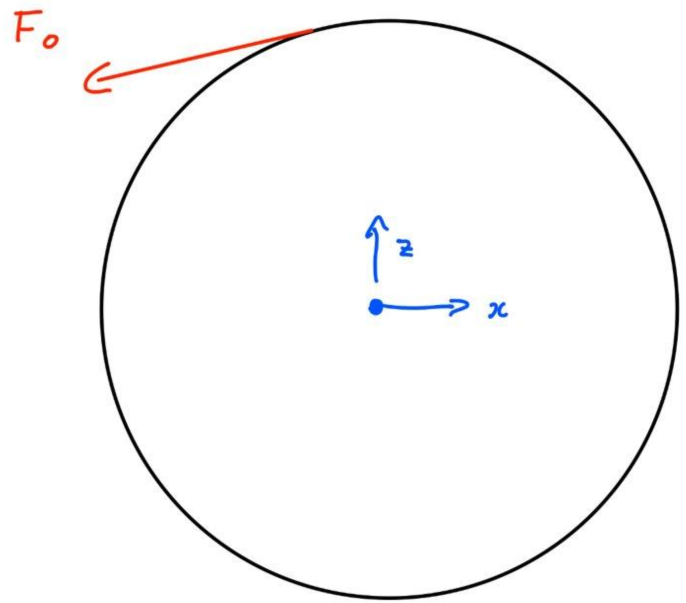
- Coordinate system provided below for reference

$$F_n = F_o * (\frac{sin(\phi))}{sin(\phi + \gamma)})^{n-1} * (\frac{sin(\gamma)}{sin\phi + \gamma})$$

$$Pressure\ Angle\ (deg) : \phi = 35 - \frac{120}{N}$$

$$Articulation\ Angle\ (deg) : \gamma = \frac{360}{N}$$

$N : Number\ of\ sprocket\ teeth$
$n : n_{th}\ tooth$
$F_o : Chain\ tension\ (lbf)$



Analysis – Sprocket Force

Tooth Table					
Tooth	Force Mag (lbf)	% Force	Pressure Angle	X Force (lbf)	Z Force (lbf)
1	1025.731396	29.65953983	-0.543	-878.0314531	-530.2694268
2	781.3046602	22.59181767	-0.341	-736.4167429	-261.0121698
3	595.1236105	17.20829887	-0.138	-589.4709618	-81.82968238
4	453.3085873	13.10764606	0.065	-452.3587718	29.32945458
5	345.2873852	9.984158609	0.267	-333.0135751	91.24328537
6	263.0071031	7.604982821	0.470	-234.4753459	119.1387781
7	200.3338067	5.792752898	0.673	-156.6774953	124.8430878
8	152.595248	4.412368433	0.875	-97.75688226	117.1703959
9	116.2325525	3.360922783	1.078	-54.97191401	102.4114003
10	88.53490808	2.560031448	1.281	-25.31250012	84.83930273
11	67.43747585	1.949988572	1.484	-5.877563297	67.18085589
12	51.36745773	1.485315906	1.686	5.915520551	51.02570265
13	39.12684573	1.131372446	1.889	12.23742032	37.16390185
14	29.80311124	0.861771969	2.092	14.8288396	25.8520977
15	22.70117672	0.656415956	2.294	15.02789087	17.01487351
16	17.29159819	0.499995269	2.497	13.82139333	10.39078701
17	13.17109557	0.380848861	2.700	11.9055384	5.63346375
18	10.03248841	0.290094454	2.902	9.746653823	2.377722227
19	7.64179587	0.220966376	3.105	7.636679346	0.27959379
20	5.820793578	0.168311177	3.308	5.740694225	-0.962324115

Analysis – Sprocket Force

- First 7 teeth put into ANSYS to a significant portion of the loading (defined as a percentage of the max chain load)
- Table below displays error between using first 7 teeth vs all teeth in ANSYS

Rear Sprocket Loading (from Tooth Table)		
Net X Force	-3380.444346	lbf
Net Z Force	-508.5566732	lbf
Net Force from Tooth Table	3418.484147	lbf

Net Force Validation		
Max Chain Load	3458.352363	lbf
Force from Tooth Table	3418.484147	lbf
Error	1.152809521	%

Analysis – Sprocket Force

- Top image shows the inputs in spline stress calculator
  - Sprocket from last time has been thickened from 0.467in to 0.5in
- Bottom image shows safety factors
  - Shear Stress SF → Yield
  - Compressive SF → Ultimate
  - Tensile Yield SF → Yield
- Reminder this is worst case torque (760 lbf-ft)

Material Parameters	Alloy	Al 7055 -T7651	
	Tensile Yield	80	ksi
	Compressive Yield	87	ksi
	Shear Ultimate	47	ksi
Spline Parameters	Major Diameter (D_ri)	1.9528	in
	Minor Diameter (inner)	1.7953	in
	Tooth Thickness (t)	0.1149	in
	Number of Teeth (N)	24	teeth
	Spline Length (Le)	0.5	in
	Spline Type	Flexible	
	Max RPM	1754	RPM
	Pressure Angle (ϕ)	30	deg
	Spline Sleeve Dia (D_oi)	2.6	in
	Lewis Form Factor (Y)	1.5	unitless
	Torque Applied (T)	760.7936	lbf

Shear Stress SF	1.18772
Compressive SF	5.97648
Tensile Yield SF	1.462591

Analysis – Diff Carriers Force

- Beam calculation is carried out on the sprocket and carrier system to find loads
- Statics on 3 node model to get forces at the nodes
  - I personally think these loads are not super representative since the load applied is a point load as opposed to a bearing load → used ANSYS + probe to get loads on bolts and tabs

Uprights Load Calculator

Inputs		
Sprocket Force:	3458.35236	
Dist Sprocket to Upleft:	1.32	in
Dist Upleft to Upright:	5.36	in
Upright Force:	851.683791	lbf
Upleft Force:	-4310.0362	lbf
Chain Angle:	9.7011	deg
Turnbuckle Angle:	10	deg

Coordinates		
Node	X	Z
Center	30.9	9.125
Top	30.2742	14.6621
Bot	30.668	5.4892

Upright		
Upright X Force:	-839.50491	lbf
Upright Z Force:	143.515791	lbf
Dist Bot to Center X	-0.232	in
Dist Bot to Center Z	3.6358	in
Dist Bot to Top X	0.3938	in
Dist Bot to Top Z	9.1729	in
Top Node Resultant Force:	331.641921	lbf
Upright Top Node X:	326.603535	lbf
Upright Top Node Z:	57.5890153	lbf
Upright Bottom Node X:	512.901377	lbf
Upright Bottom Node Z:	-201.10481	lbf

Upleft		
Upright X Force:	4248.40365	lbf
Upright Z Force:	-726.27688	lbf
Dist Bot to Center X	-0.232	in
Dist Bot to Center Z	3.6358	in
Dist Bot to Top X	0.3938	in
Dist Bot to Top Z	9.1729	in
Top Node Resultant Force:	1704.13569	lbf
Upright Top Node X:	1678.24604	lbf
Upright Top Node Z:	295.920056	lbf
Upright Bottom Node X:	-5926.6497	lbf
Upright Bottom Node Z:	430.356825	lbf

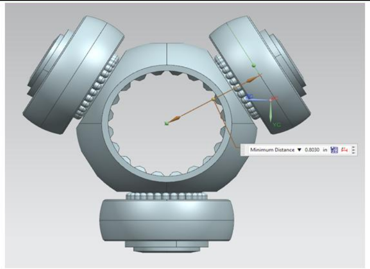
Analysis – Bolt/Lug Calcs

- Unfortunately, I am blocked by ANSYS at this point
  - I do not want to use values in tables, since they are not 100% representative of what is actually occurring (in my mind)
  - Without a combined case (will explain later) I am not able to properly source bolt and lug loads

Analysis – CV Insert Force

- Unlike previous few seasons, contact patch is assumed to be 2/3 rollers at launch
  - Load has gone up significantly since that time, hence necessitating the switch to a much stronger type of steel

CV Insert Loads



Global Inputs		
Vehicle Weight	650	lbm
Max Proj Accel	1.65	G
Tire Diameter	15.75	in
Launch Torque	55	ft-lbs
CV Roller Radial Distance	0.803	in

Engine Gearing Parameters		
Primary	2.073	out:in
1	2.583	out:in
2	2	out:in
3	1.667	out:in
4	1.444	out:in
5	1.286	out:in
6	1.15	out:in
Final Drive	2.5833333	out:in

Differential Parameters	
Ramp Angle (deg)	Lock up %
30	88
40	60
45	51
50	42
60	29
From Drexler manual	

Vehicle Weight Method		
Vehicle Acceleration Force	1072.5	lbf
Rear Axle Total Torque	8445.9375	in-lbs
Acceleration Ramp Angle	45	deg
Associated Accel Bias Case	51	%
Maximum Single Wheel Torque	4307.4281	in-lbs
Radial Roller Bearing Force	2682.0848	lbf

Engine Torque Method		
Chosen Launch Gear	1	-
Adjusted Engine Torque	760.793591	ft-lbs
Acceleration Ramp Angle	45	deg
Associated Accel Bias Case	51	%
Maximum Single Wheel Torque	4656.05678	in-lbs
Radial Roller Bearing Force	2899.16362	lbf

Analysis – CV Insert Force

- With the new steel being used, much higher SF is achieved (yield)
  - Worst case SF for CV Insert is ~1.199 at a location 0.008 into the insert

Shape	Cylinder	4340-AMS-6414	
Poisson's Ratio	0.32		
Elastic Modulus	29000	ksi	
Yield Strength	235	ksi	
Diameter of Object	1.003	inch	
Shape	Plane	Carpenter Technology Custom 465 (H1000)	
Poisson's Ratio	0.27		
Elastic Modulus	28400	ksi	
Yield Strength	200	ksi	
Diameter of Object	1.012	inch	
Length of Contact	0.33	in	
Applied Load	2899.16	lbf	
d1	1.003	in	
d2	10000000000	in	
	Considering Roller as Cylinder		
b	0.018887963	in	
p_max	296.1104809	ksi	
z	0.0082	in	
Sigma_x of CV Roller	-124.3254334	ksi	
Sigma_x of CV Insert	-104.8995844	ksi	
Sigma_y	-116.8990328	ksi	
Sigma_z	-271.6179465	ksi	
T_max	83.35918105	ksi	
CV Roller safety factor	**1.410**		
CV Insert safety factor	**1.200**		

Analysis – Turnbuckles

- Ideally would use values from ANSYS, but used highest force for no margin
  - If ANSYS loads exceed this value, then there could be an issue (however, trade study was carried out to reduce forces)
- Input load was 2000 lbf
  - Tensile/Compressive doesn't matter

Euler Buckling MS	12.72
Johnson Buckling MS	3.18
Tube Tensile Yield MS	11964.050
Tube Compressive Yield MS	4.801
Weld tensile yield MS	5165.726
Tube tensile ultimate MS	10824.522
Weld tensile ultimate MS	4984.438
Rod end MS	0.000
Buckling MS	3.179
Female thread shear MS	8.553
Insert Bond MS	1000000.000
Female thread fails before rod end	False
Buckling Type	Johnson
Minimum MS	0.000
Driving Failure	Rod end MS

FEA

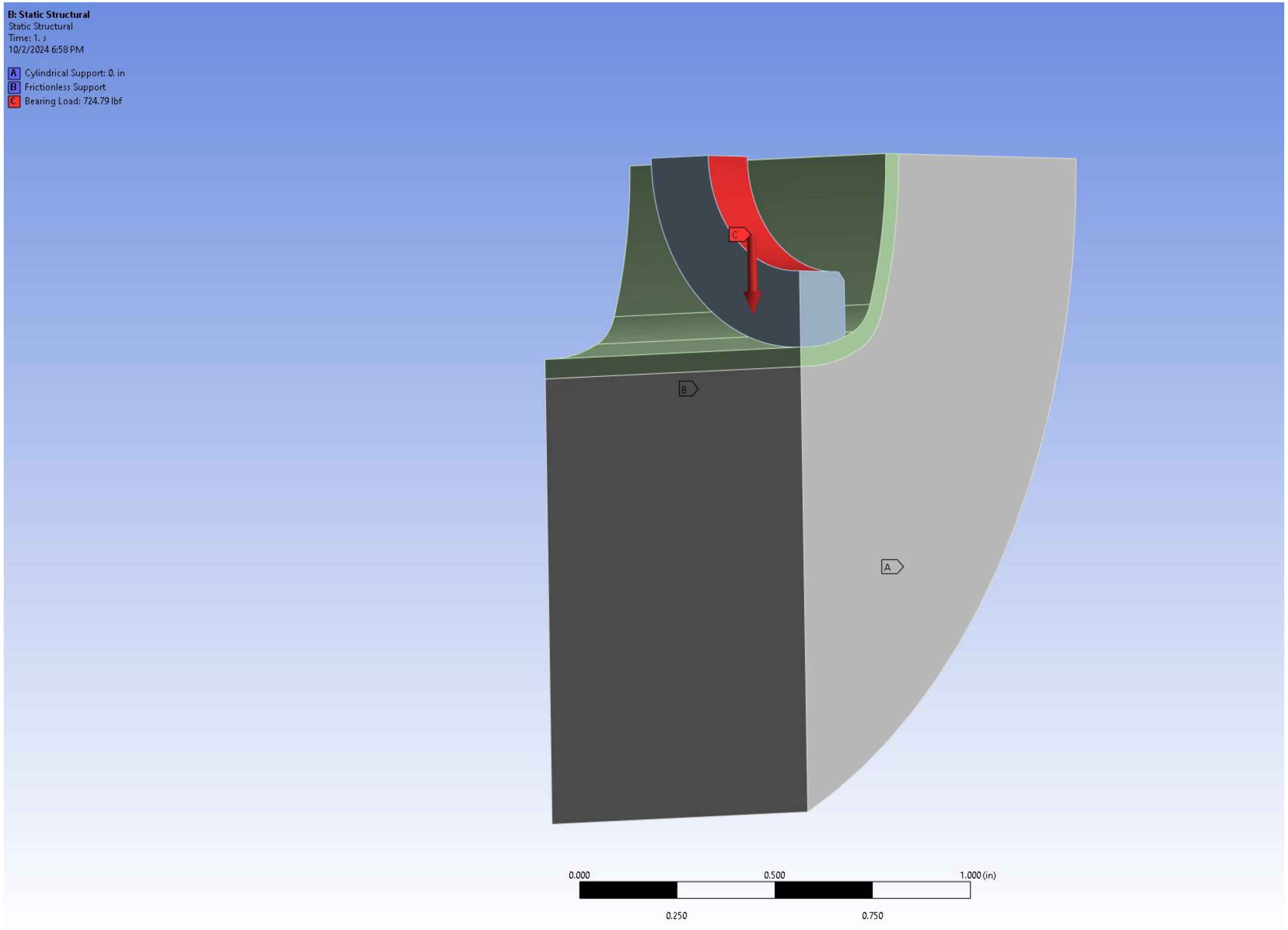
FEA – CV Inserts

- Notes
  - FEA will display a stress level near infinite due to the nature of contact stresses for a cylinder and a plane converging at a line
  - FEA can be used to find the max depth of stress and compare to hand calculations performed

FEA – CV Inserts

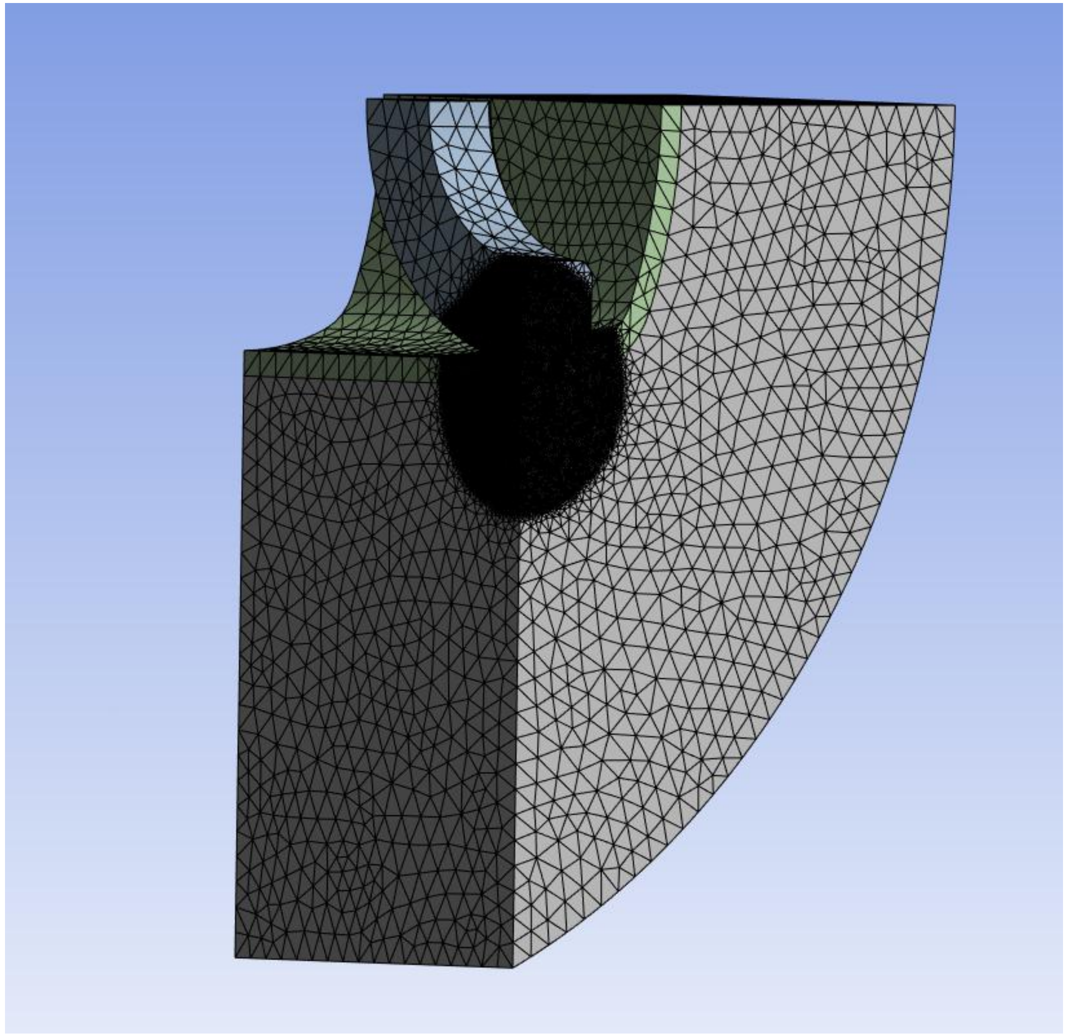
- Setup:
  - Quarter tire model
    - A quarter of roller, CV housing, and mock hub are modeled (to ease meshing and solve time)
  - Support
    - Cylindrical Support and Frictionless Support used to support the “hub” in place
  - Loads
    - ¼ of the load seen according to CV Insert Loads calculator is applied to the quarter roller

FEA – CV Inserts



FEA – CV Inserts

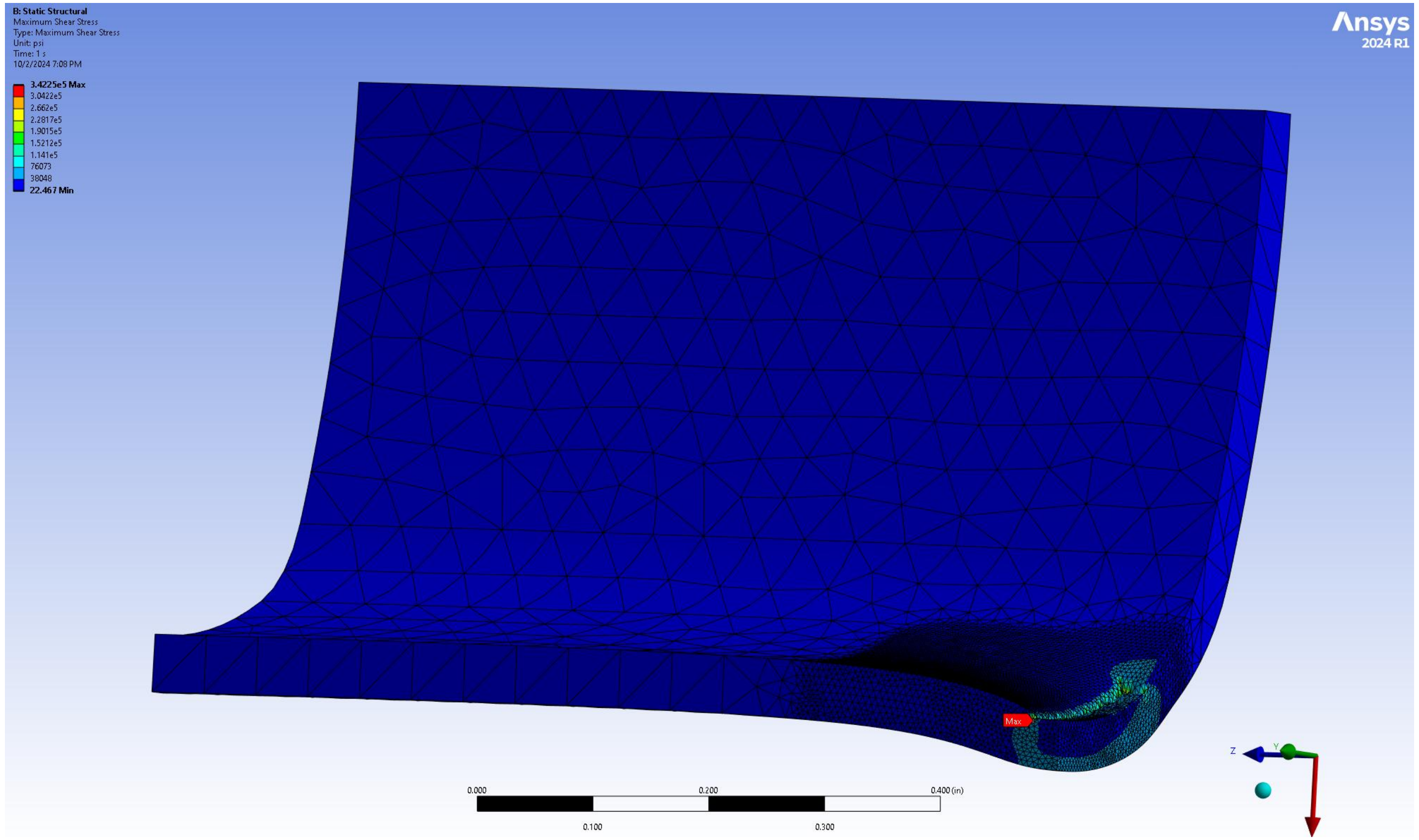
- Mesh:
  - General Parameters
    - Element Size = 0.05 in
    - Quadratic Element Order
  - Sphere of Influence and Contact Sizing
    - Better mesh on contact surfaces

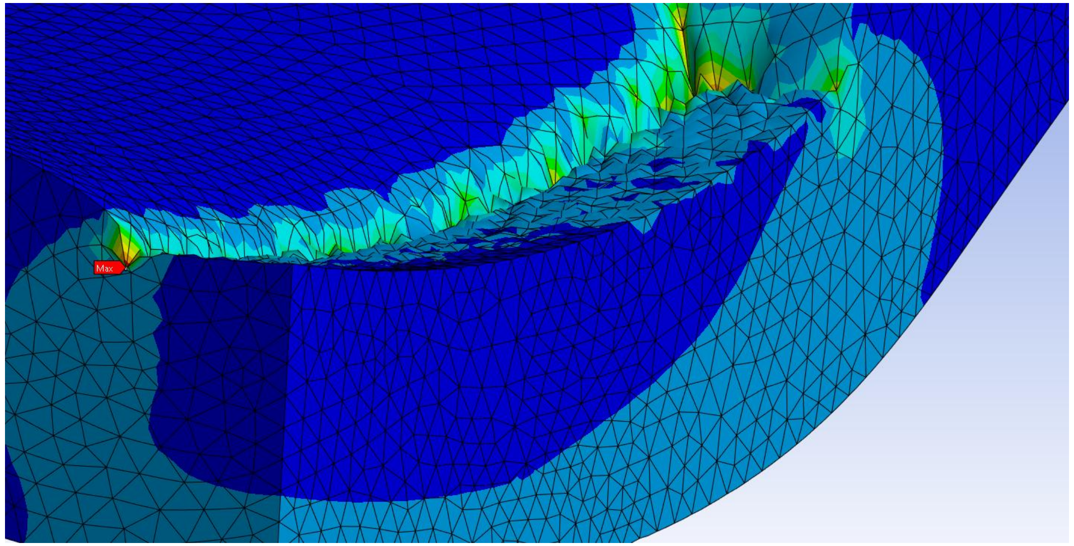


FEA – CV Inserts

Load Case: Launch	Solver: Ansys	Material: Custom 465	Geometry: Linear
Safety Factor Ultimate: 1.4	Safety Factor Yield: 1.1	Temperature: 71.6 F	Notes:

Section	Max Stress	Failure Mode	Material	Yield Allowable (with knockdowns)	Ultimate Allowable (with knockdowns)	MS_Yield	MS_Ultimate	Jacobian Ratio (Gauss Points) Avg	Aspect Ratio Avg
			Steel	181 ksi				0.99987	1.7889



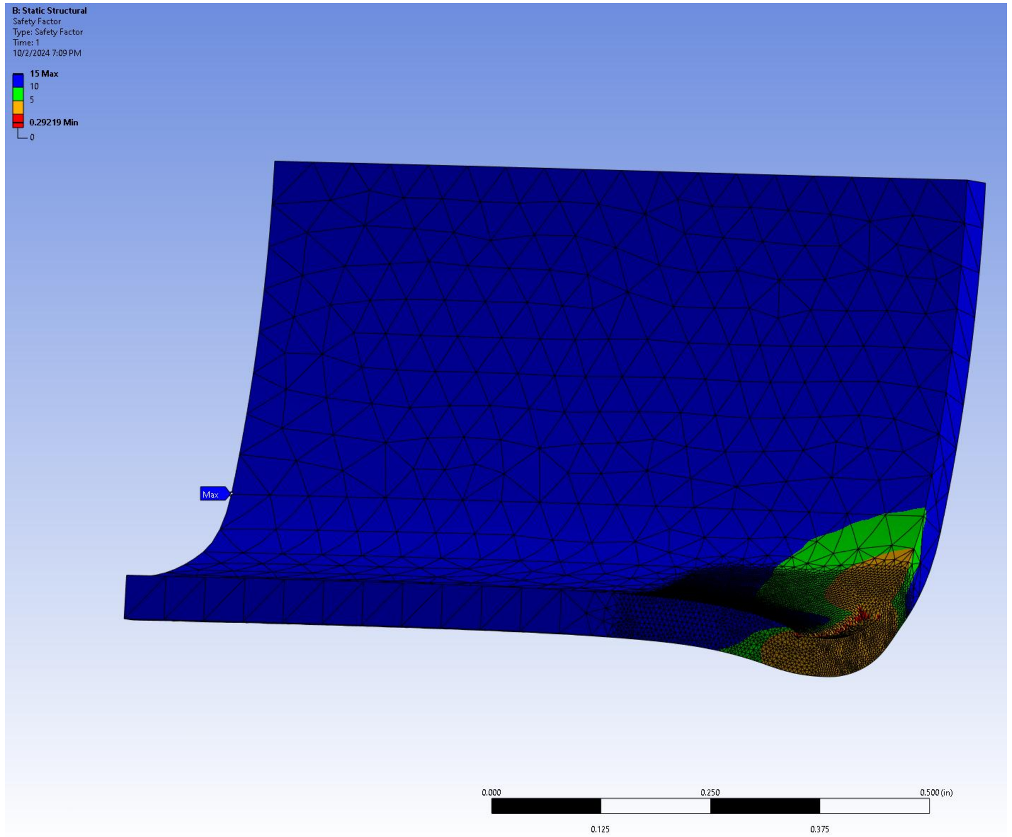


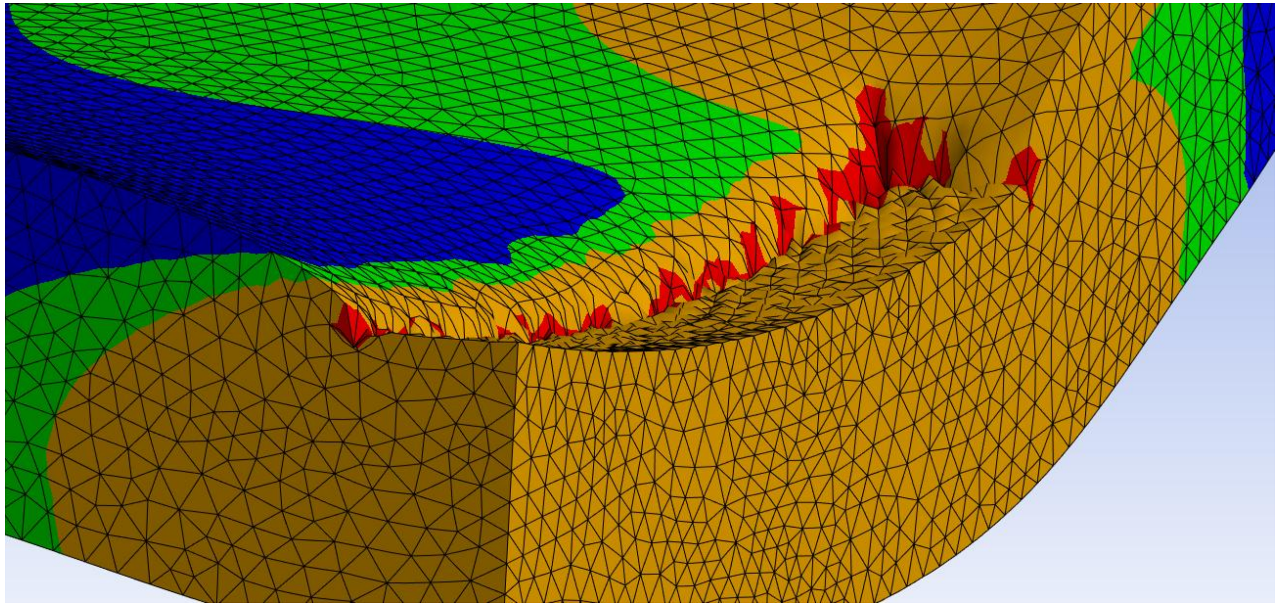
FEA – CV Inserts

PURDUE FORMULA SAE™

Load Case: Launch	Solver: Ansys	Material: Custom 465	Geometry: Linear
Safety Factor Ultimate: 1.4	Safety Factor Yield: 1.1	Temperature: 71.6 F	Notes:

Section	Max Stress	Failure Mode	Material	Yield Allowable (with knockdowns)	Ultimate Allowable (with knockdowns)	MS_Yield	MS_Ultimate	Jacobian Ratio (Gauss Points) Avg	Aspect Ratio Avg
			Steel	181 ksi				0.99987	1.7889

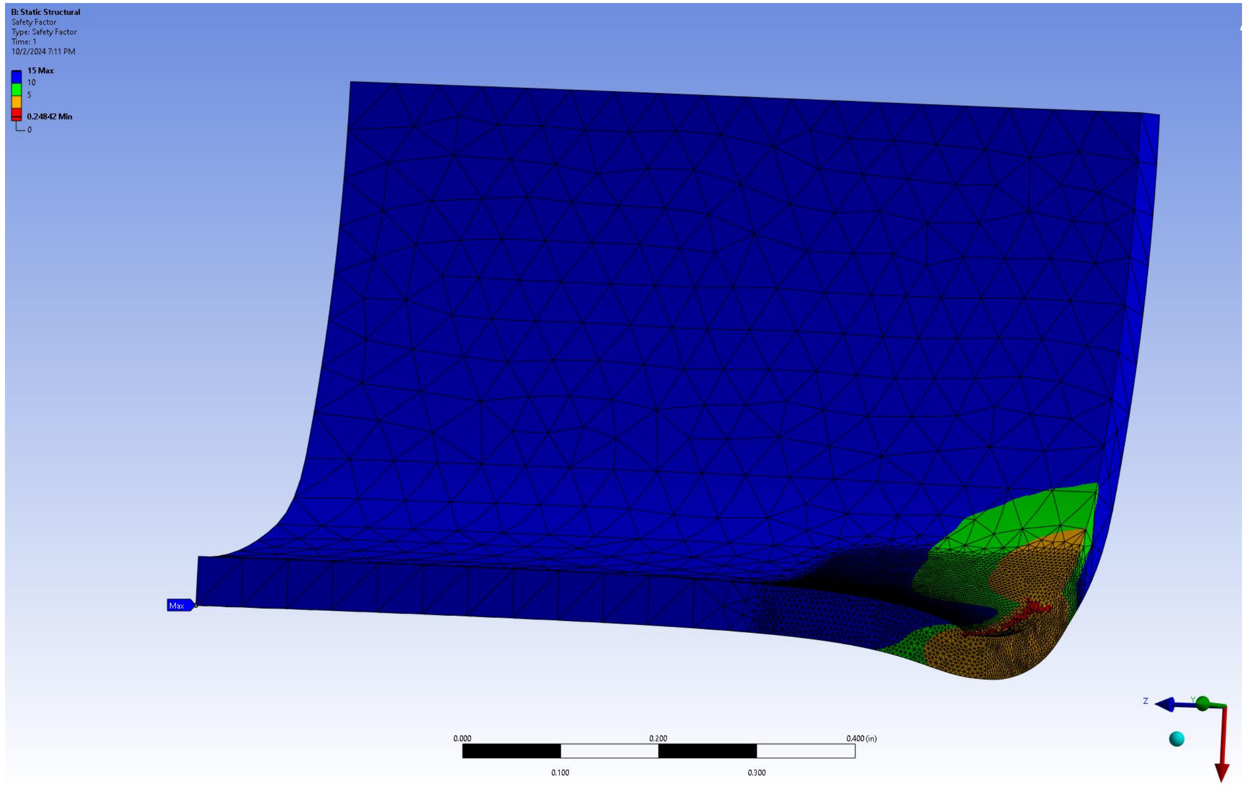


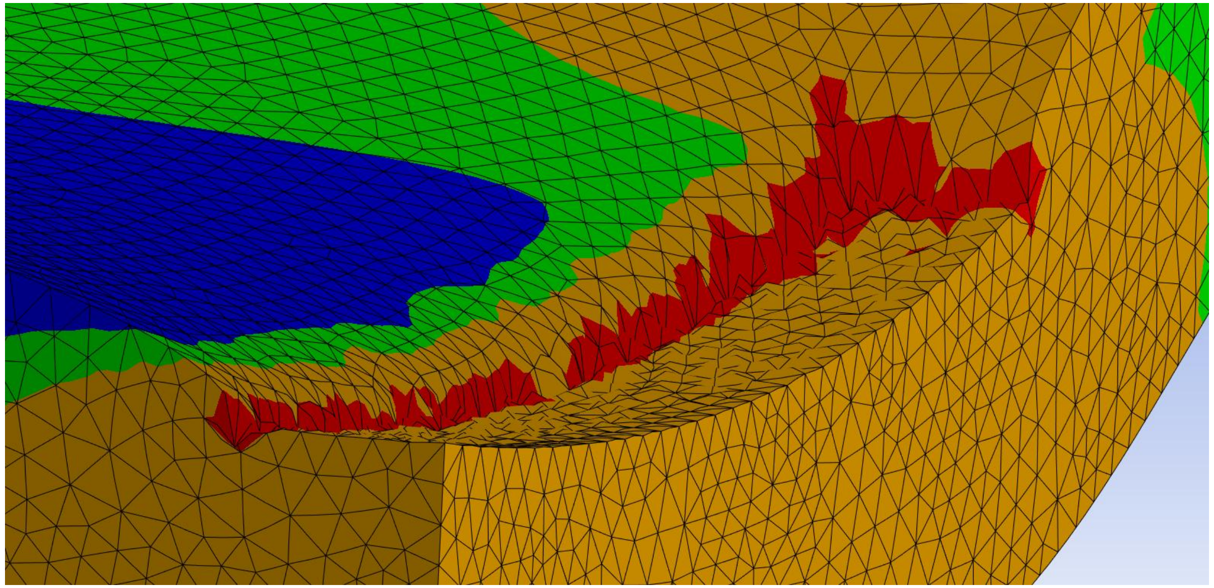


FEA – CV Inserts

Load Case: Launch	Solver: Ansys	Material: Custom 465	Geometry: Linear
Safety Factor Ultimate: 1.4	Safety Factor Yield: 1.1	Temperature: 71.6 F	Notes:

Section	Max Stress	Failure Mode	Material	Yield Allowable (with knockdowns)	Ultimate Allowable (with knockdowns)	MS_Yield	MS_Ultimate	Jacobian Ratio (Gauss Points) Avg	Aspect Ratio Avg
			Steel	181 ksi				0.99987	1.7889

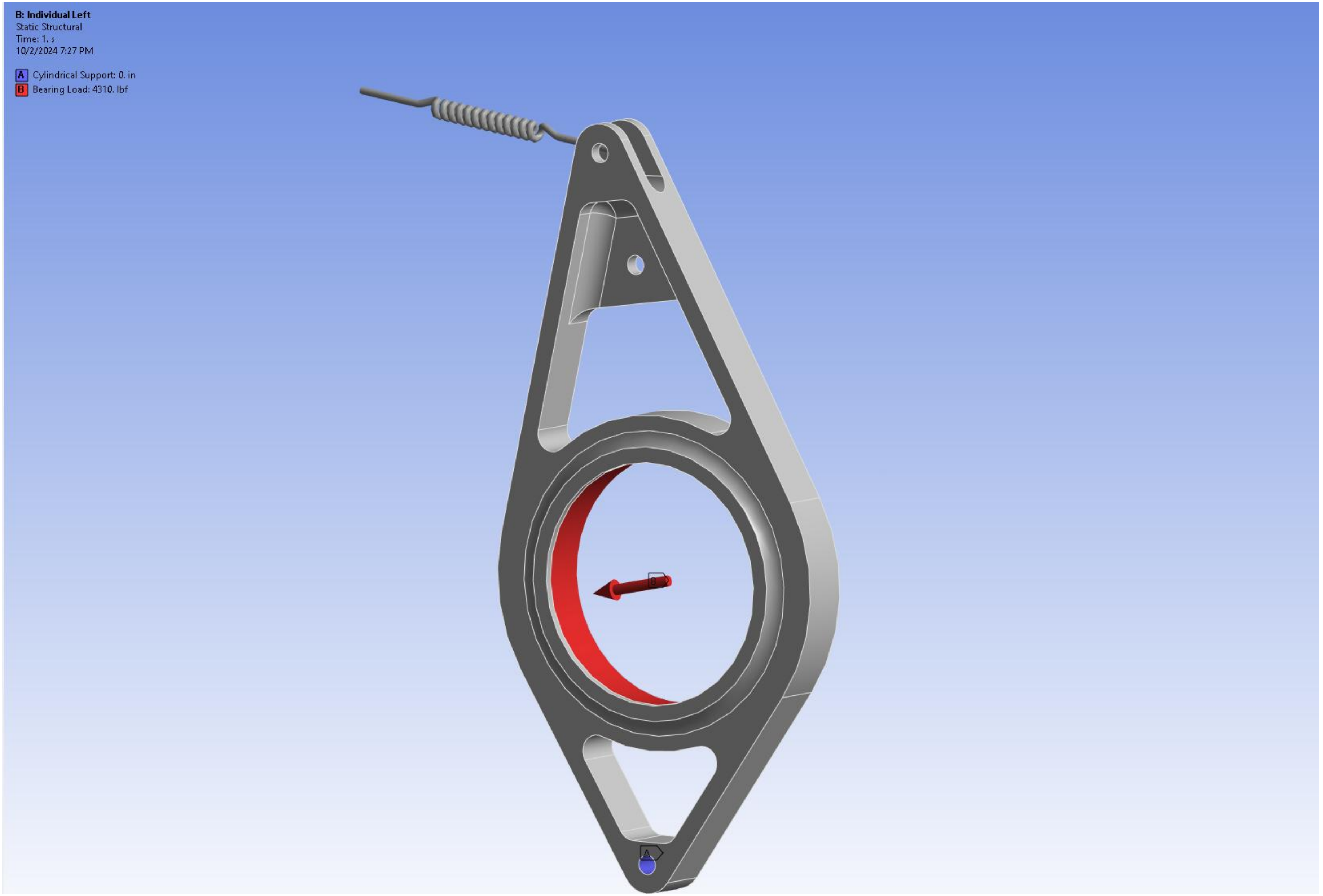




FEA – Diff Carriers (Individual)

- Setup:
  - Turnbuckle modelled as spring with stiffness calculated
  - Cylindrical Support on Bottom Node
  - Bearing Load
    - Load applied onto a mock bearing face to simulate a bearing load going through the part
  - Testing for equivalent stress, shear stress, deformation, and eigenvalue buckling

FEA – Diff Carriers (Individual)



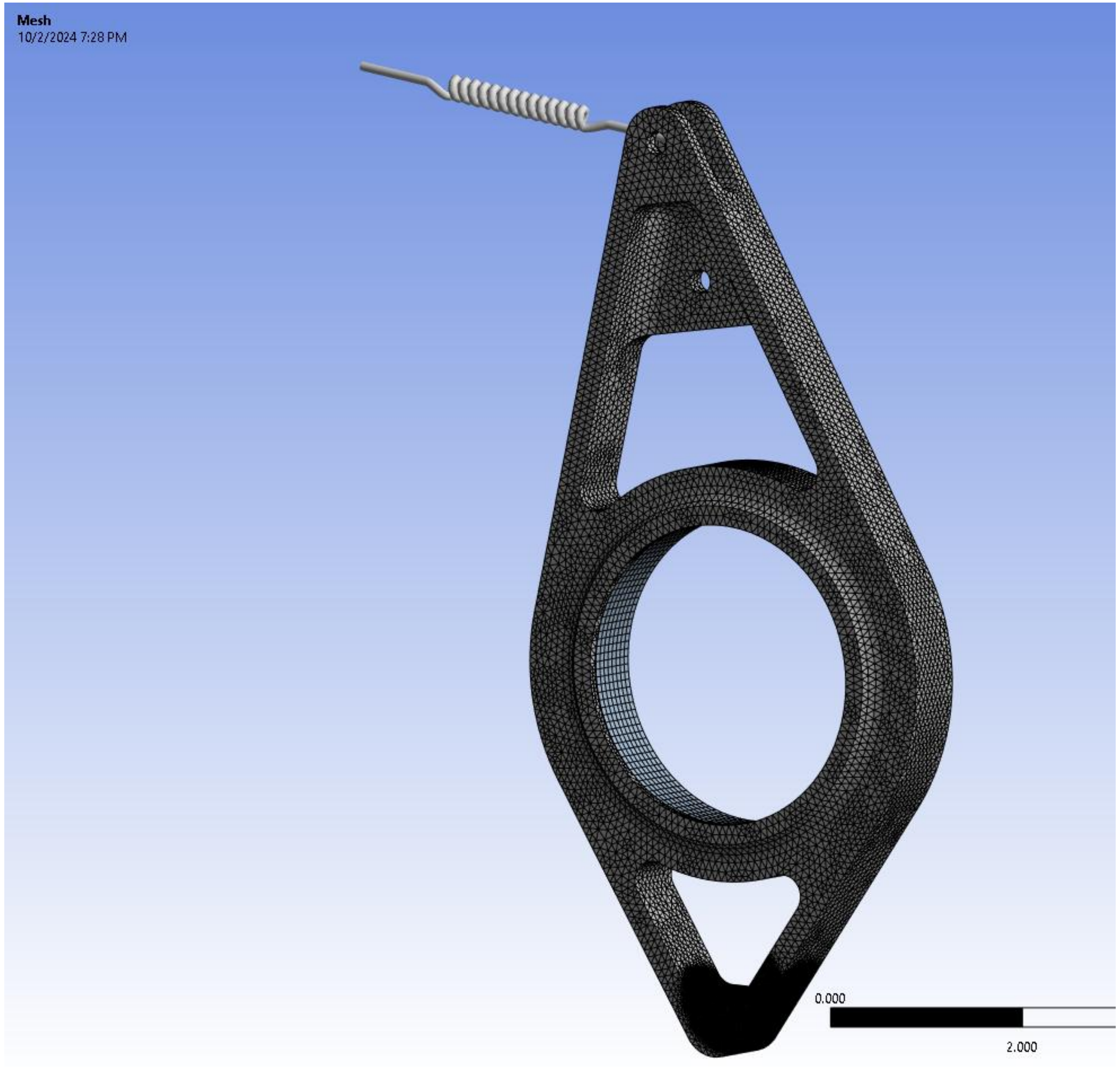
FEA – Diff Carriers

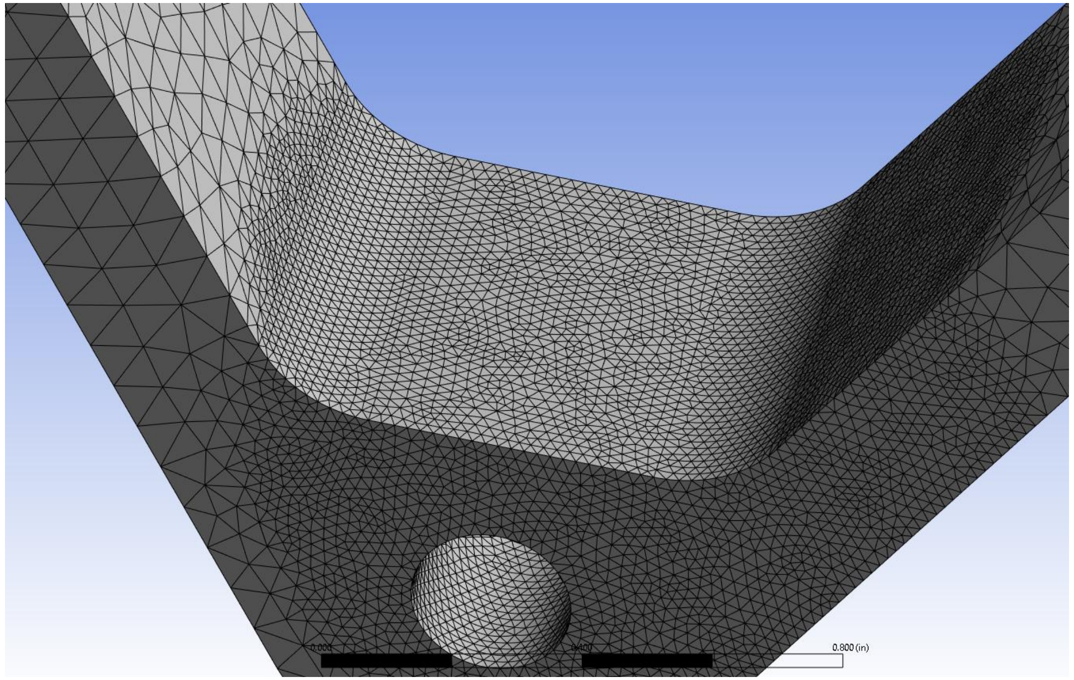
- Mesh:
  - Mesh convergence not carried out for this one
    - Peak stresses remain in similar spots so it is better to run a coarser mesh all around and a sphere of influence where there is peak stress (usually the bottom fillet)
  - Settings:
    - General element size at 0.1 in
    - SOI element size at 0.02 in
      - Ran at the bottom fillet for both
- Note:
  - Mass is not fully optimized
  - Eigenvalue buckling not ran due to length of time running on lab computers
  - Combined setup not run due to issues with individual right case
    - Ran once, went to change a relatively normal parameter, sends carrier's stress to basically infinity… something I have to investigate before I can run a combined case (I did both carriers exact same way, so any input would help here)

FEA – Diff Carriers (Combined)

- Setup:
  - Turnbuckles modelled as spring with stiffness calculated
  - Cylindrical Support on Bottom Nodes
  - Bearing Loads
    - Load applied onto a mock bearing face to simulate a bearing load going through the part
  - Testing for overall system stresses and reaction probes to get bolt and lug calcs

FEA – Diff Carriers

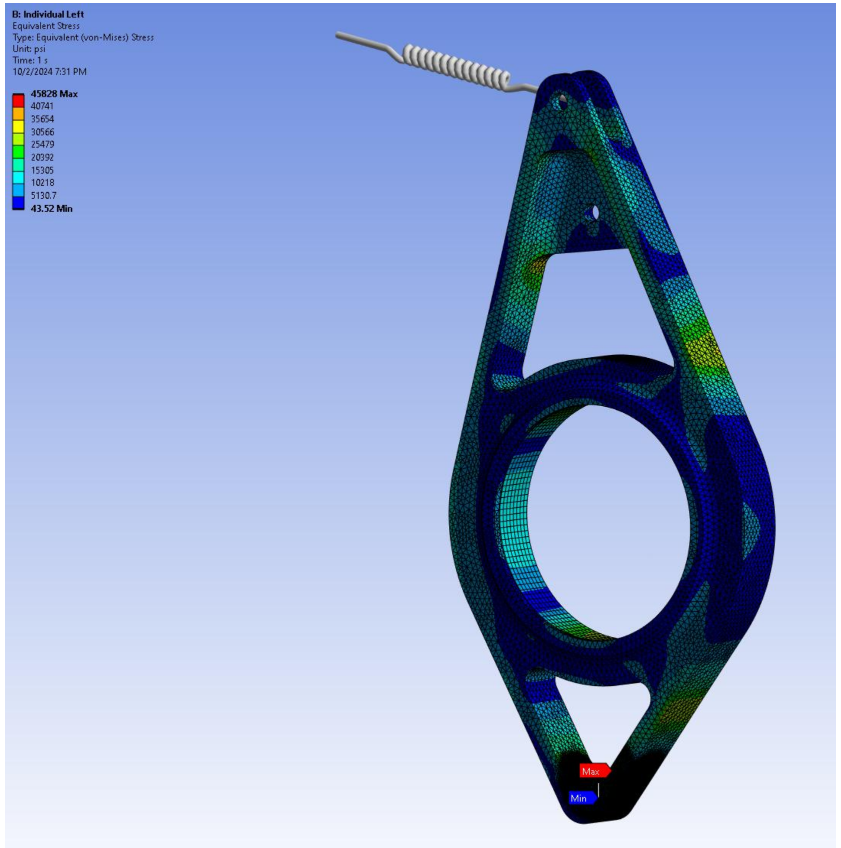


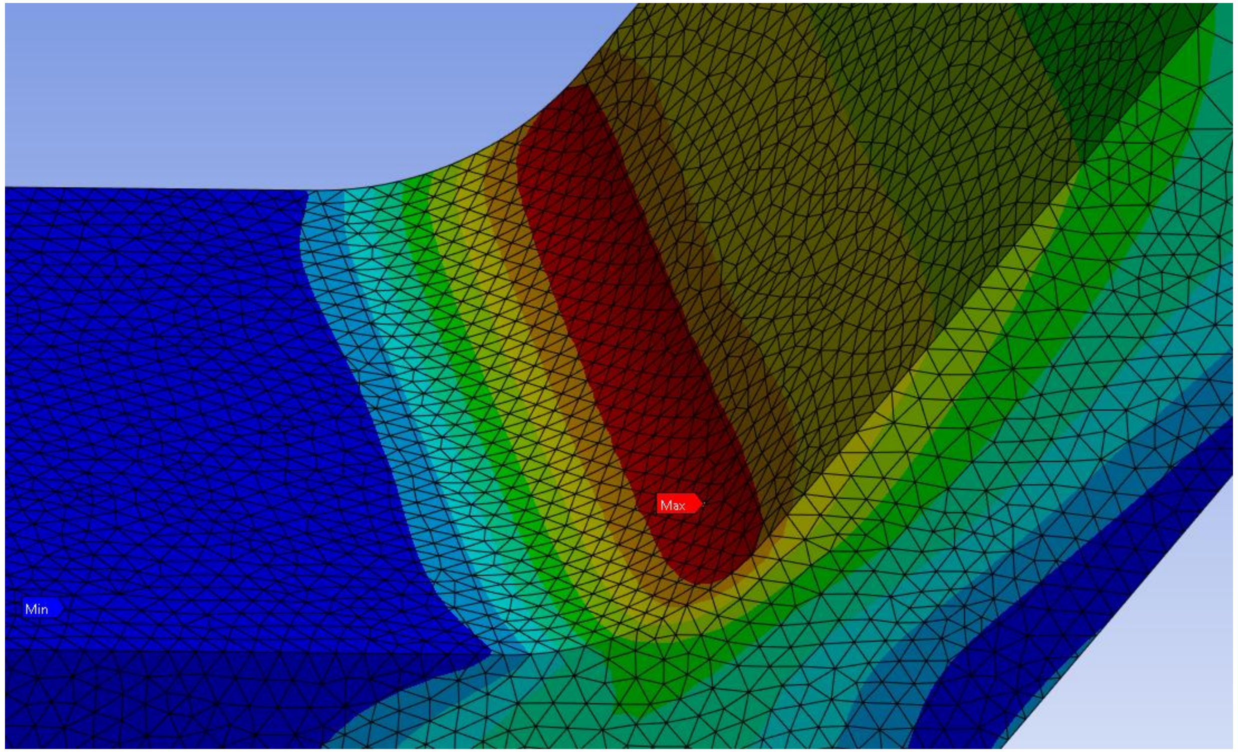


FEA – Left Carrier (Individual)

Load Case: LC	Solver: Ansys	Material: Al 7055-T77511	Geometry: Linear
Safety Factor Ultimate: 1.4	Safety Factor Yield: 1.2	Temperature: 71.6 F	Notes:

Section	Max Stress	Failure Mode	Material	Yield Allowable (with knockdowns)	Ultimate Allowable (with knockdowns)	MS_Yield	MS_Ultimate	Jacobian Ratio (Gauss Points) Avg	Aspect Ratio Avg
			Al	61600 psi	57000 psi			0.99854	1.8052





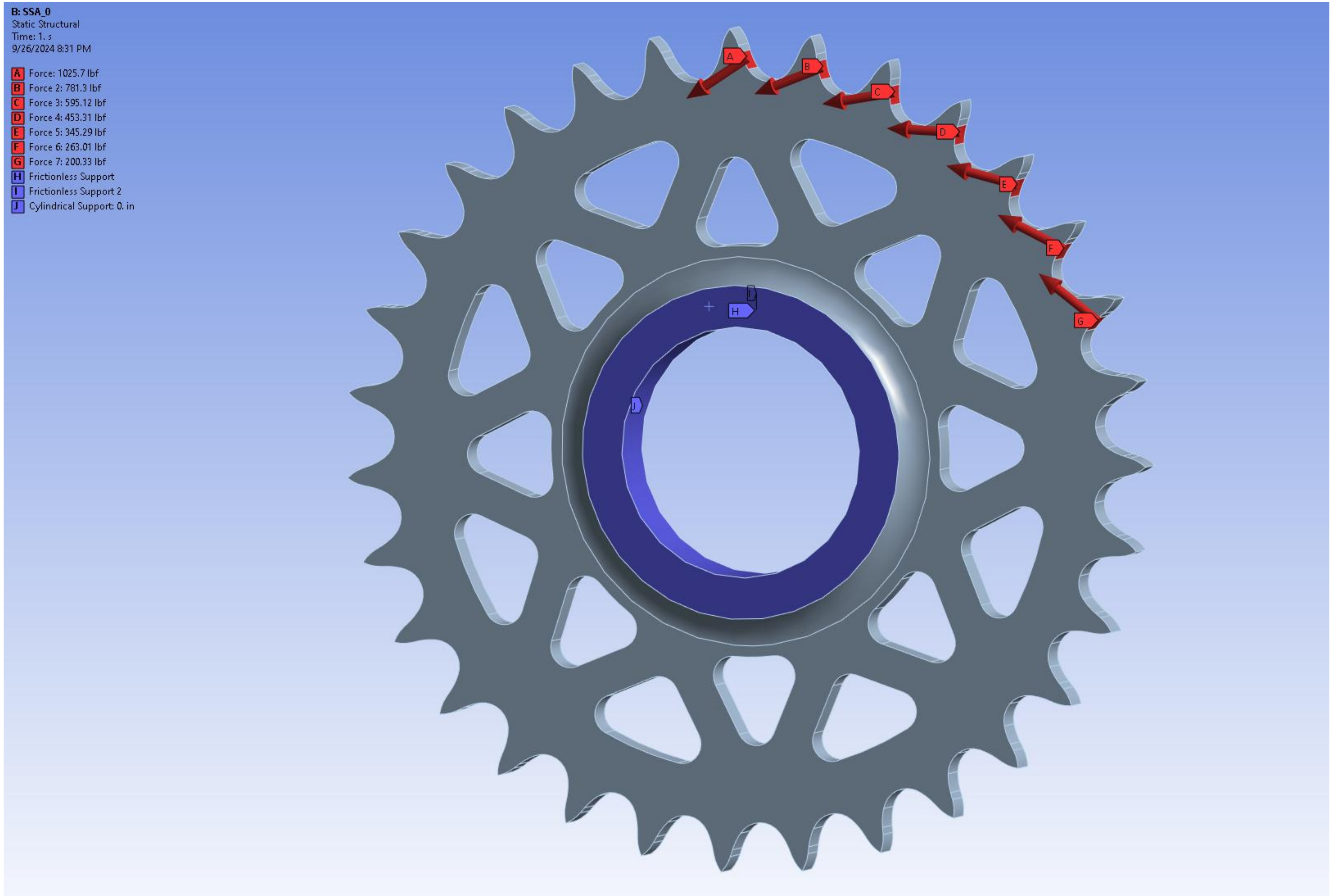
FEA – Sprocket

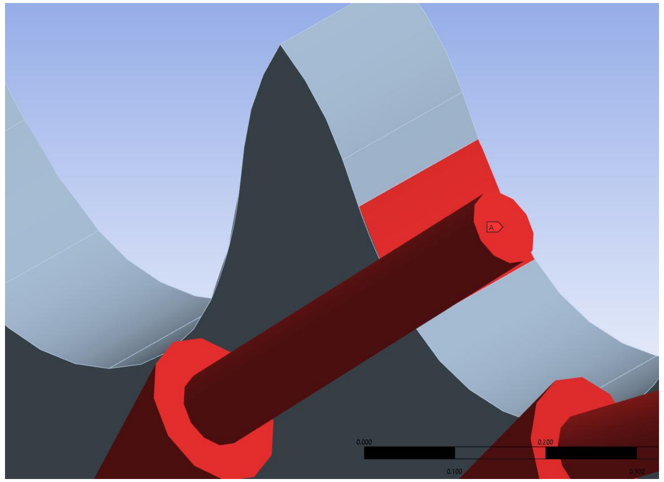
- Notes
    - FEA ran without splined section to ease meshing
        - Calculations carried out on spline stress calculator as previously defined
    - Loading from tooth table applied onto first 7 teeth
        - Represented under 5% of the max load that is seen
            - Ran sim using first 13 teeth, but proved to be no more useful than 7 in terms of peak stresses obtained

FEA – Sprocket

- Setup:
  - First seven tooth forced modeled as point loads on surface of tooth
  - Frictionless supports on the sides
  - Cylindrical support where spline would be
    - Changed from PDR (based on IDR feedback)
    - Sprocket is fixed relative to the differential, but the differential is free to rotate. Thus, a cylindrical support was recommended

FEA – Sprocket

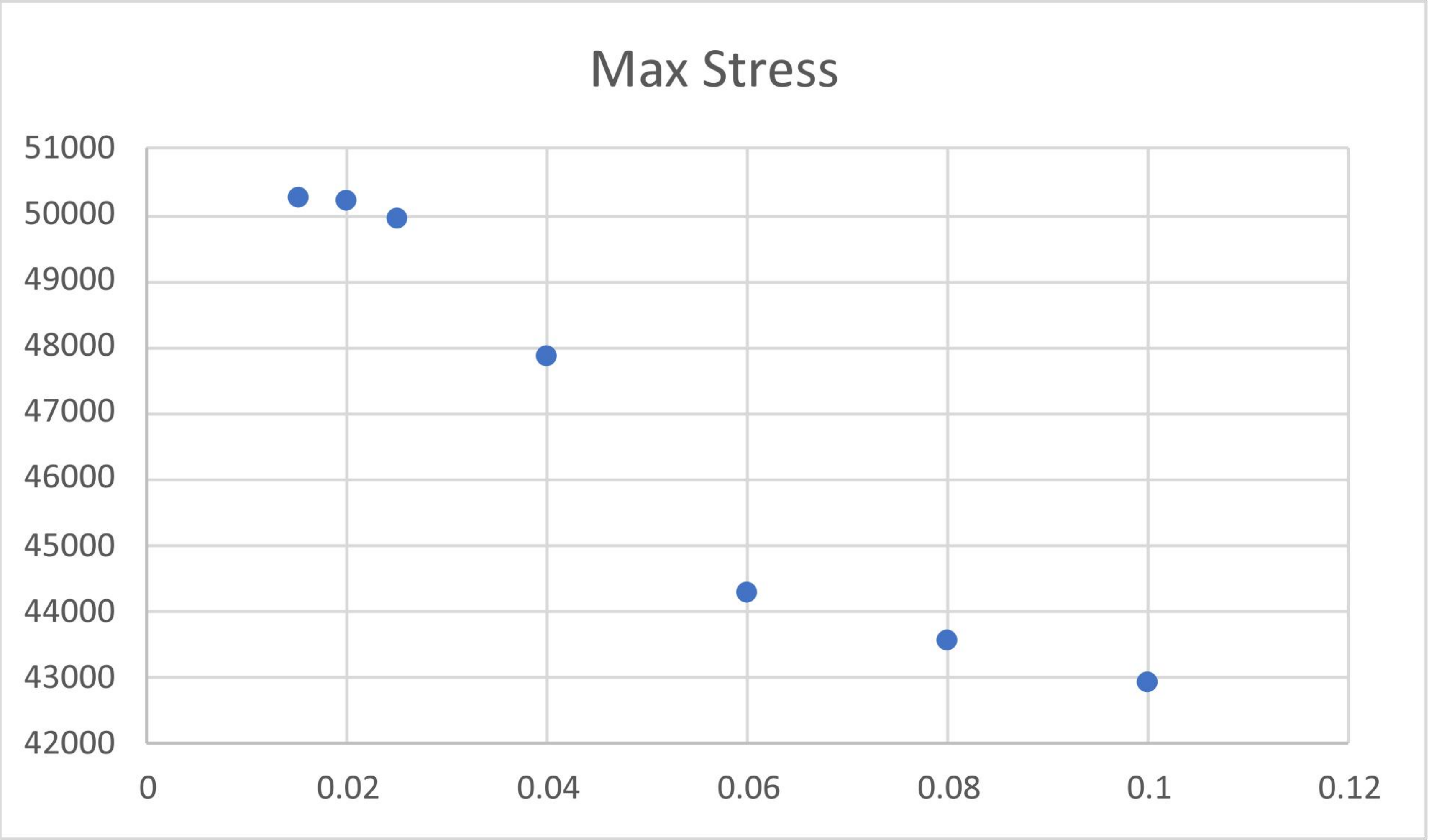




FEA – Sprocket

- Mesh Convergence Study:
  - Element size changed to perform study

Element Size	Max Stress
0.1	42946
0.08	43621
0.06	44311
0.04	47911
0.025	49960
0.02	50242
0.015	50283



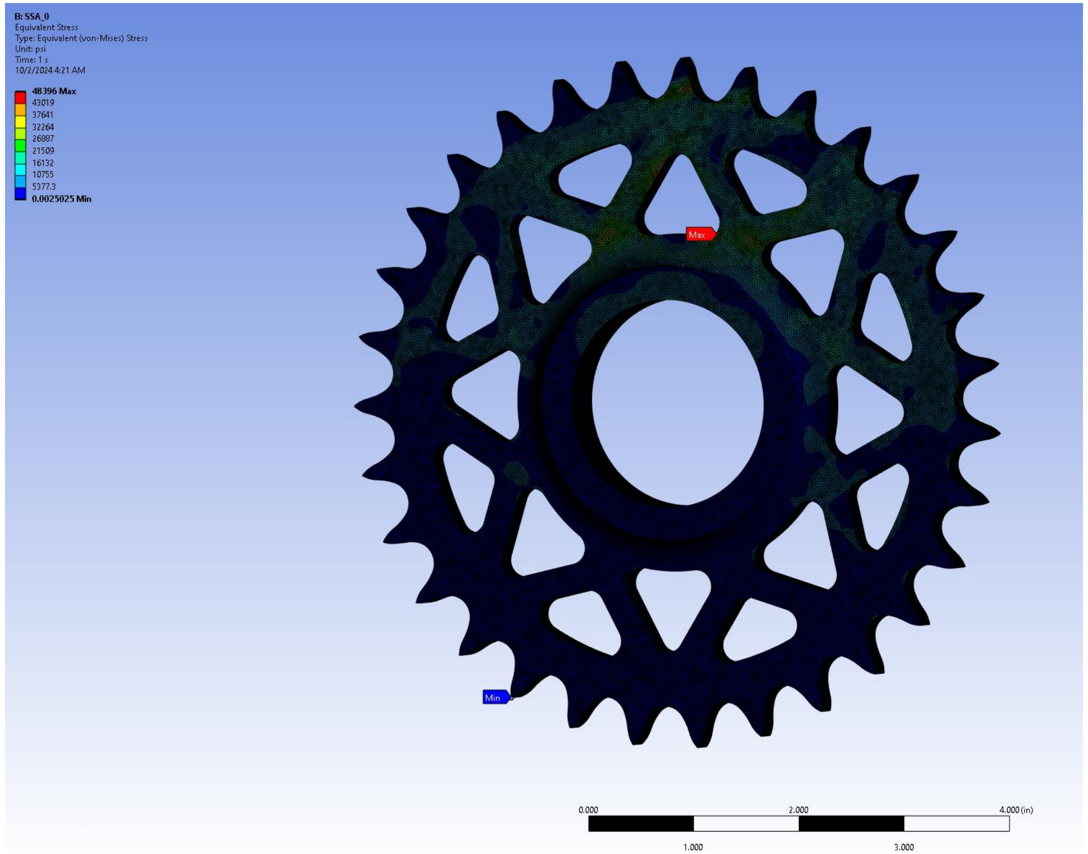
FEA – Sprocket

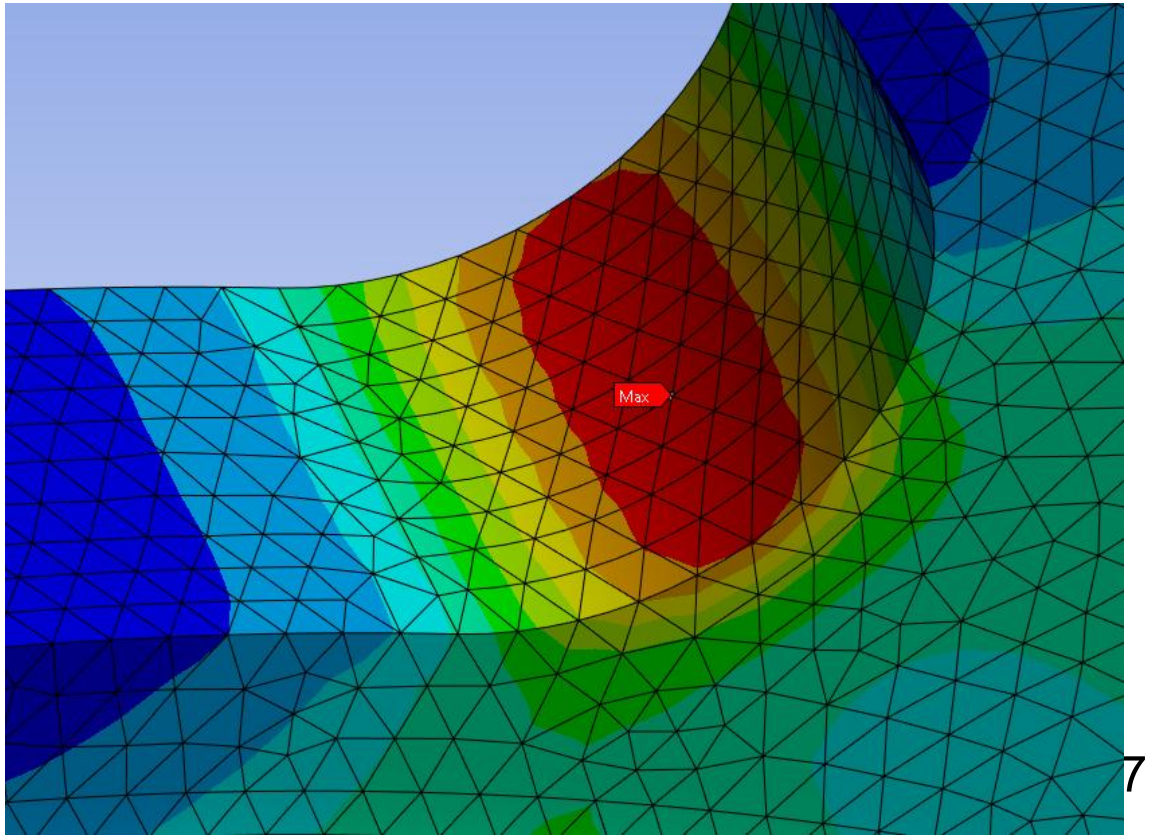
- Final Mesh Parameters:
  - General:
    - Element Size – 0.025 in
      - Ran SOI where needed
      - Decided on upon results from mesh convergence study
    - Element Order – Program Controlled
  - Patch Conforming Method
    - Tetrahedrons
    - Patch Conforming Algorithm
    - Quadratic Element Order

FEA – Sprocket SSA_0

Load Case: SSA_0	Solver: Ansys	Material: Al 7055-T77511	Geometry: Linear
Safety Factor Ultimate: 1.4	Safety Factor Yield: 1.2	Temperature: 71.6 F	Notes:

Section	Max Stress	Failure Mode	Material	Yield Allowable (with knockdowns)	Ultimate Allowable (with knockdowns)	MS_Yield	MS_Ultimate	Jacobian Ratio (Gauss Points) Avg	Aspect Ratio Avg
	43896 psi		Al	61600 psi	57000 psi			0.99844	1.8946

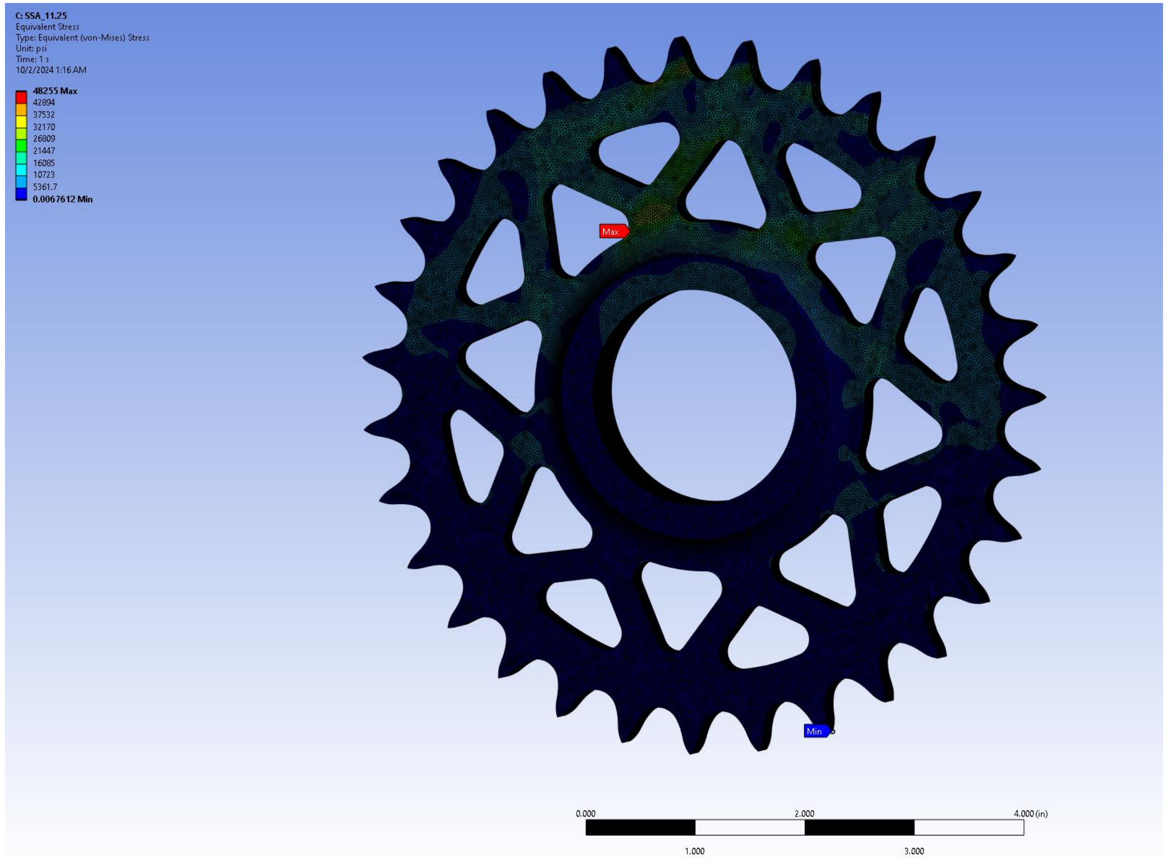


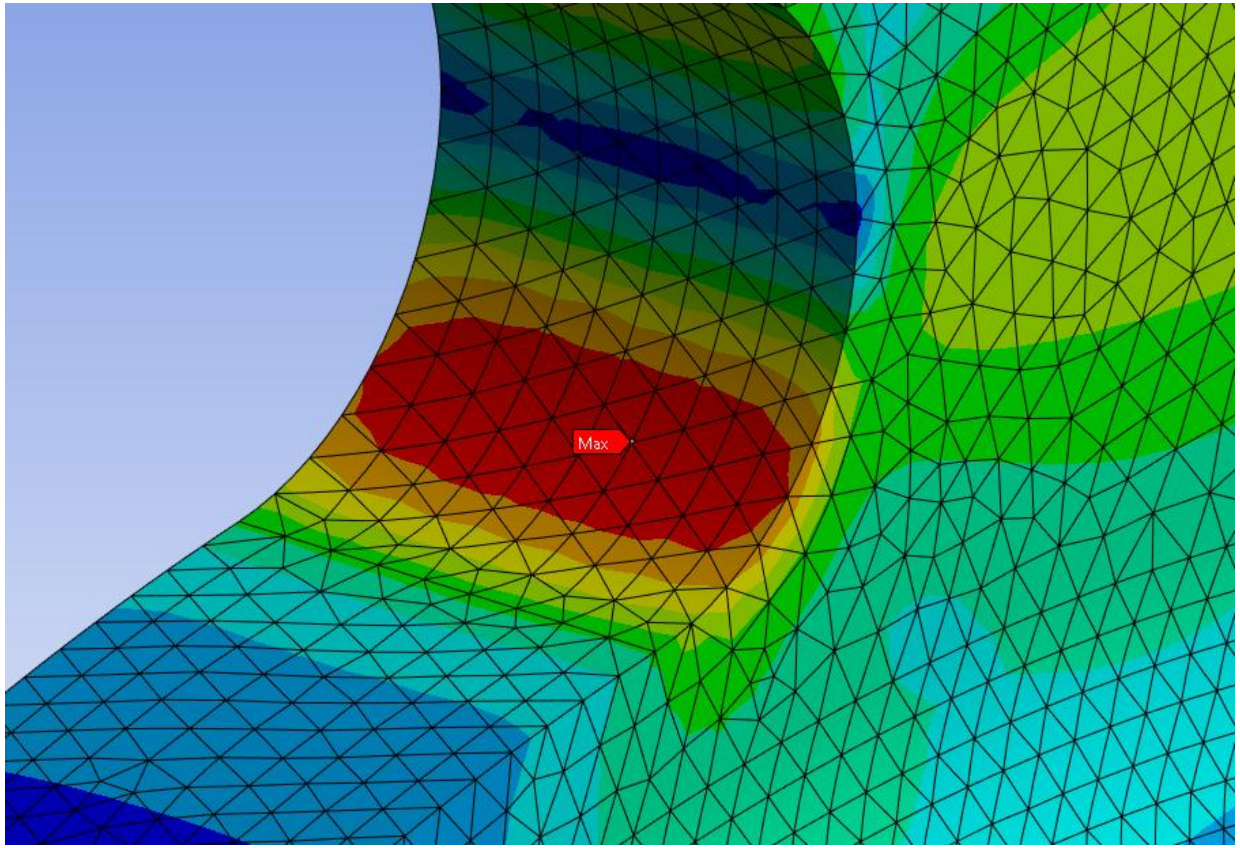


FEA – Sprocket SSA_11.25

Load Case: SSA_11.25	Solver: Ansys	Material: Al 7055-T77511	Geometry: Linear
Safety Factor Ultimate: 1.4	Safety Factor Yield: 1.2	Temperature: 71.6 F	Notes:

Section	Max Stress	Failure Mode	Material	Yield Allowable (with knockdowns)	Ultimate Allowable (with knockdowns)	MS_Yield	MS_Ultimate	Jacobian Ratio (Gauss Points) Avg	Aspect Ratio Avg
	48255 psi		Al	61600 psi	57000 psi			0.99843	1.8941

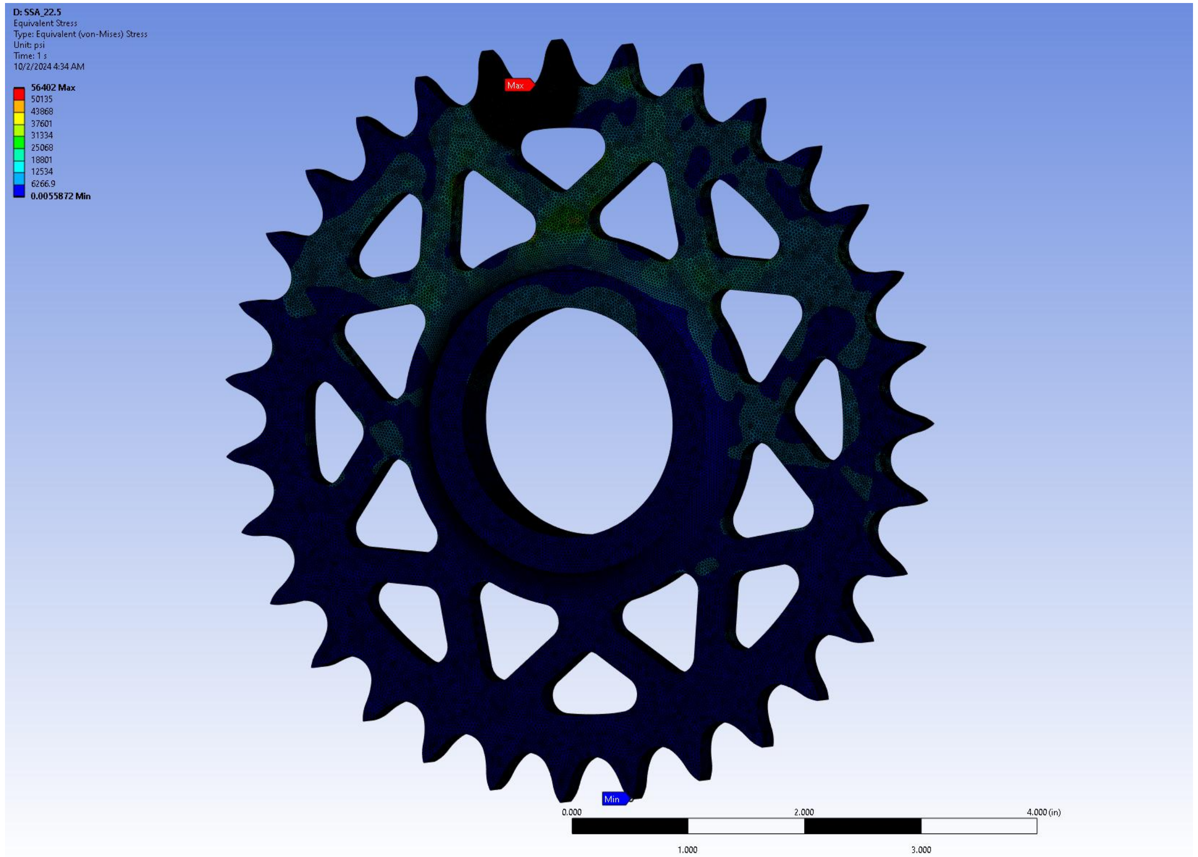


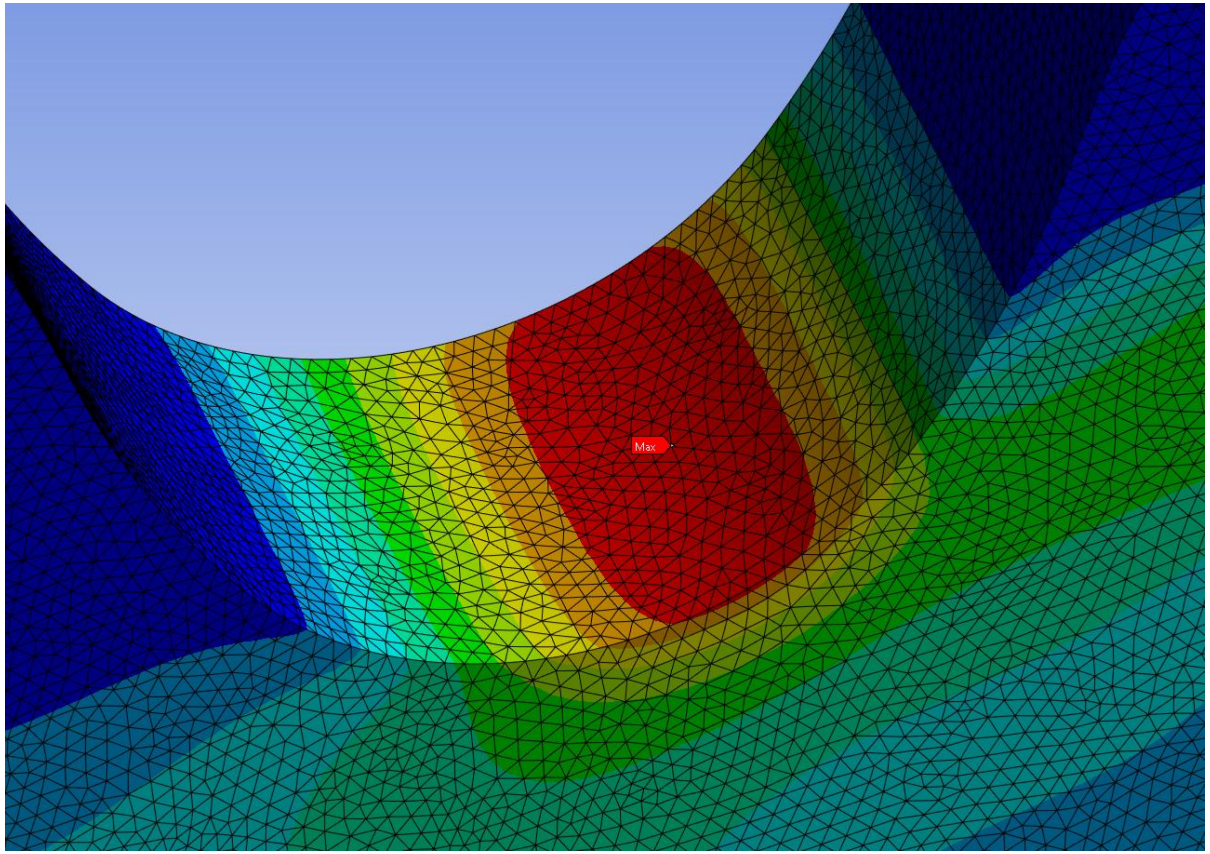


FEA – Sprocket SSA_22.5

Load Case: SSA_22.5	Solver: Ansys	Material: Al 7055-T77511	Geometry: Linear
Safety Factor Ultimate: 1.4	Safety Factor Yield: 1.2	Temperature: 71.6 F	Notes: Ran SOI

Section	Max Stress	Failure Mode	Material	Yield Allowable (with knockdowns)	Ultimate Allowable (with knockdowns)	MS_Yield	MS_Ultimate	Jacobian Ratio (Gauss Points) Avg	Aspect Ratio Avg
	56402 psi		Al	61600 psi	57000 psi			0.99905	1.8477

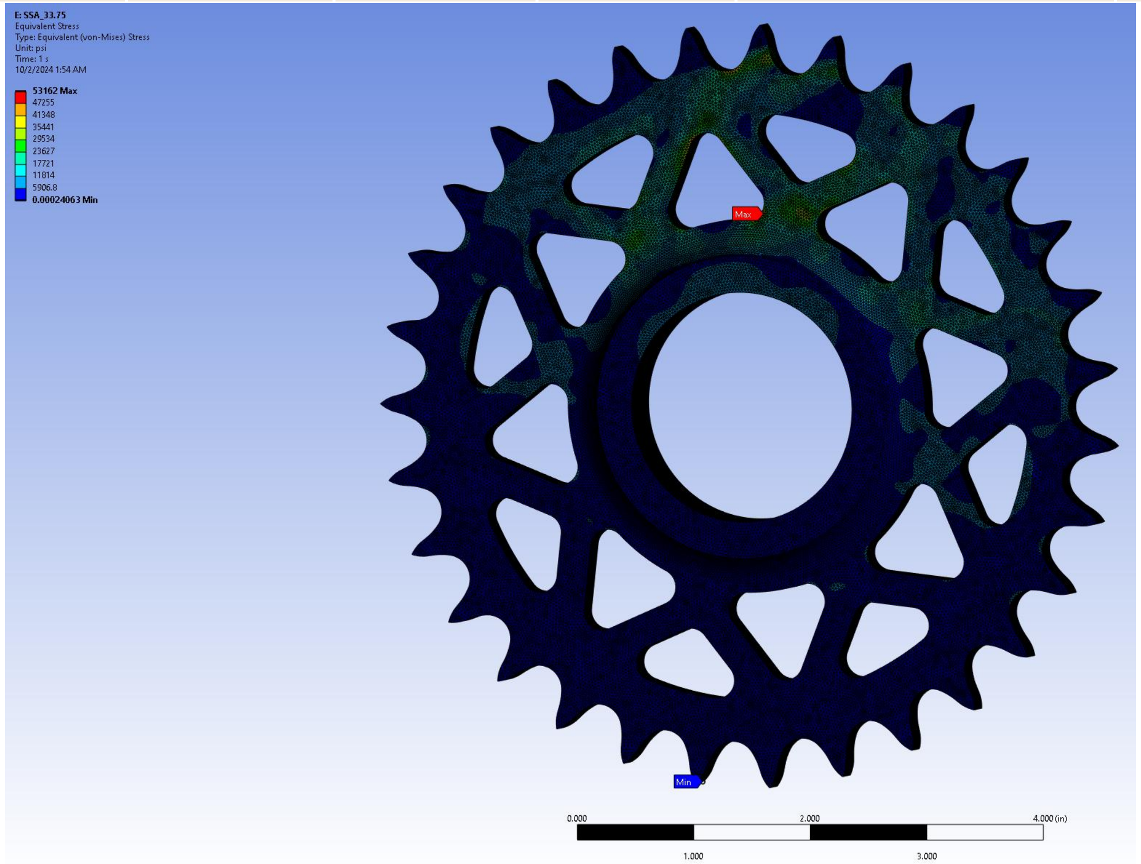


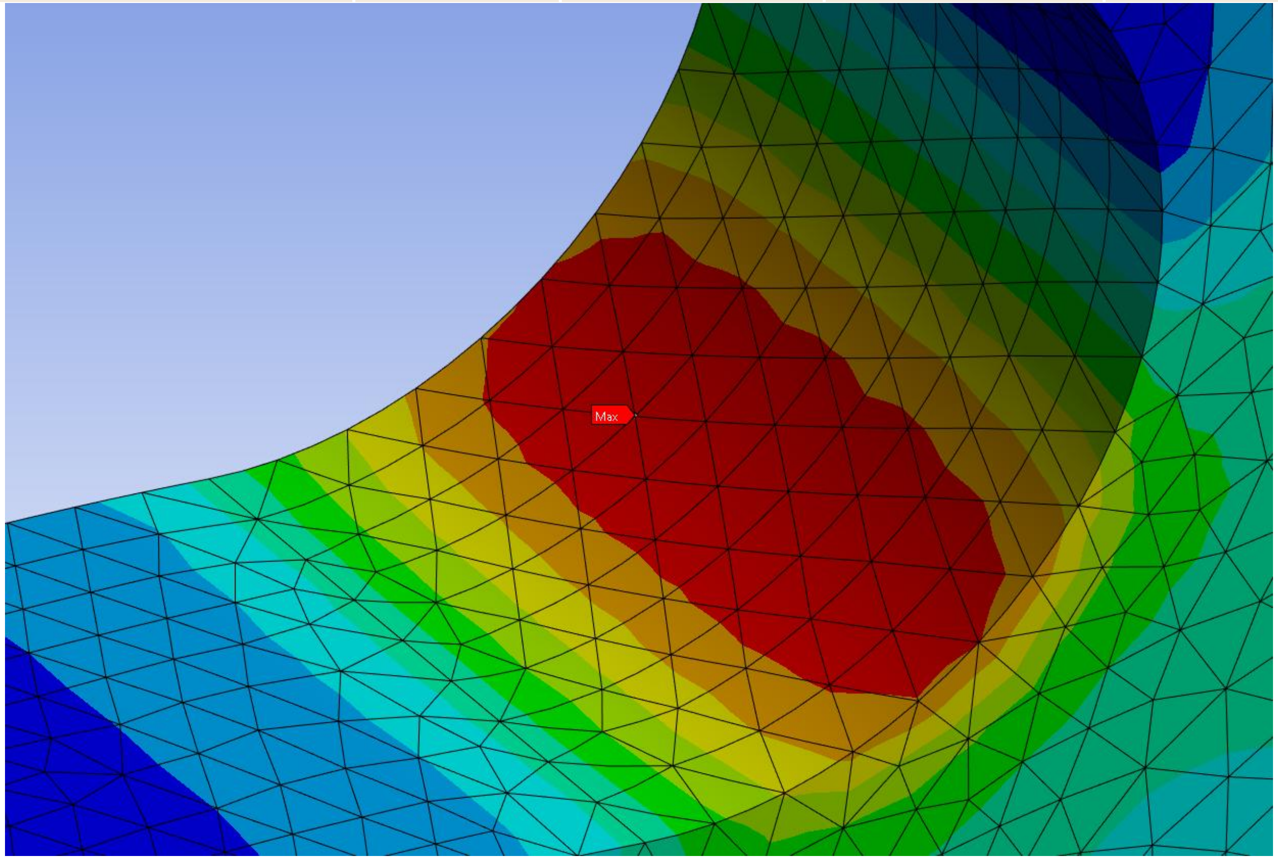


FEA – Sprocket SSA_33.75

Load Case: SSA_33.75	Solver: Ansys	Material: Al 7055-T77511	Geometry: Linear
Safety Factor Ultimate: 1.4	Safety Factor Yield: 1.2	Temperature: 71.6 F	Notes:

Section	Max Stress	Failure Mode	Material	Yield Allowable (with knockdowns)	Ultimate Allowable (with knockdowns)	MS_Yield	MS_Ultimate	Jacobian Ratio (Gauss Points) Avg	Aspect Ratio Avg
	53162 psi		Al	61600 psi	57000 psi			0.99844	1.894





Manufacturing

Manufacturing - Sprocket

- Stock Prep
  - Cut to round ~ 7” OD
  - Thickness = 0.625”
- Wire EDM
  - Spline section
- Operation 1 – Manual Lathe
  - Machine down OD to ~0.2” past the tip of the teeth
  - Machine down thickness to 0.5”
  - Machine down part to make basic profile with no teeth or cutouts
- Operation 2 – CNC Lathe
  - Cut teeth and cutouts
- Operation 3 – Post Processing
  - Filing
- Type II Hard Ano

Manufacturing - Carriers

- Stock Prep
  - Cut to size
- Operation 1 – CNC
- Type II Hard Ano
- Operation 2 – Cut rodend slots
  - Mill to appropriately sized dimension and tolerancing (0.001) for a tight fit
- Operation 3 – Boring
- Operation 4 – Press fit bearings
  - Exact fits are a bit tbd at the moment

Manufacturing – Turnbuckles

- Acquire stock
- Operation 1 – Cut to size
  - Use lathe and cut to size based on CAD dimensions
- Operation 2 – Tap threaded holes

Manufacturing – CV Inserts

- Acquire stock
- Operation 1 – Cut to size/face if needed
- Operation 2 – Wire EDM
  - HT not needed on PF25 as it will come heat treated

Manufacturing – Chain Guard

- Acquire stock
- Operation 1 – Laser cut to appropriate size
- Operation 2 – Bend
  - Either use a CNC roller or hammer and torch

“Manufacturing” - Halfshaft

- Acquire Halfshaft from RCV
- Measure lengths on car
- Operation 1 – Cut to Length
    - Cut to as measured length on car (not necessarily same as CAD)
- Operation 2 – Cut Grooves for O-rings

Cost

Cost

Major Component	Owned/Sponsored/Purchased	Estimated Cost + Misc Notes
Drexler Salisbury LSD	Owned	$3000
Halfshafts	Purchased	$300 x 2
Tripods	Purchased	$115 x 4
Diff Carrier Bearings	Sponsored (SKF, NTN)	$150 x 4
Inboard CV (Tulip) Housings	Owned	$750
Aluminum Stock	Sponsored (Arconic)	$1000
Chain	Purchased	$100 x 2-3
Hardware	Purchased	$250

Total Estimated Cost (excluding owned and sponsored parts)	$1610

Note: This is preliminary as I am still working through design and analysis. This is a rough estimate.