Design of Drivetrain System for RGP007

Rose-Hulman Institute of Technology Grand Prix Engineering

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1. Loads on the Drivetrain System

By: Thaddeus Hughes
Date: 6/5/2017

Loads on the drivetrain come from three component limits: engine, wheels, and brakes. To a much lesser extent, loads from auxiliary components mounted to the drivetrain (exhaust, jacking, cooling) must also be supported. These are typically of much lesser magnitude, but still need to be analyzed.

Engine loads:

The engine produces an output torque at its' output shaft which must be calculated by working backwards from the crank torque.

Reported values from engine team are (typically) average values. Remember that the engine is a reciprocating process; this means that power is produced for only a small period of time.

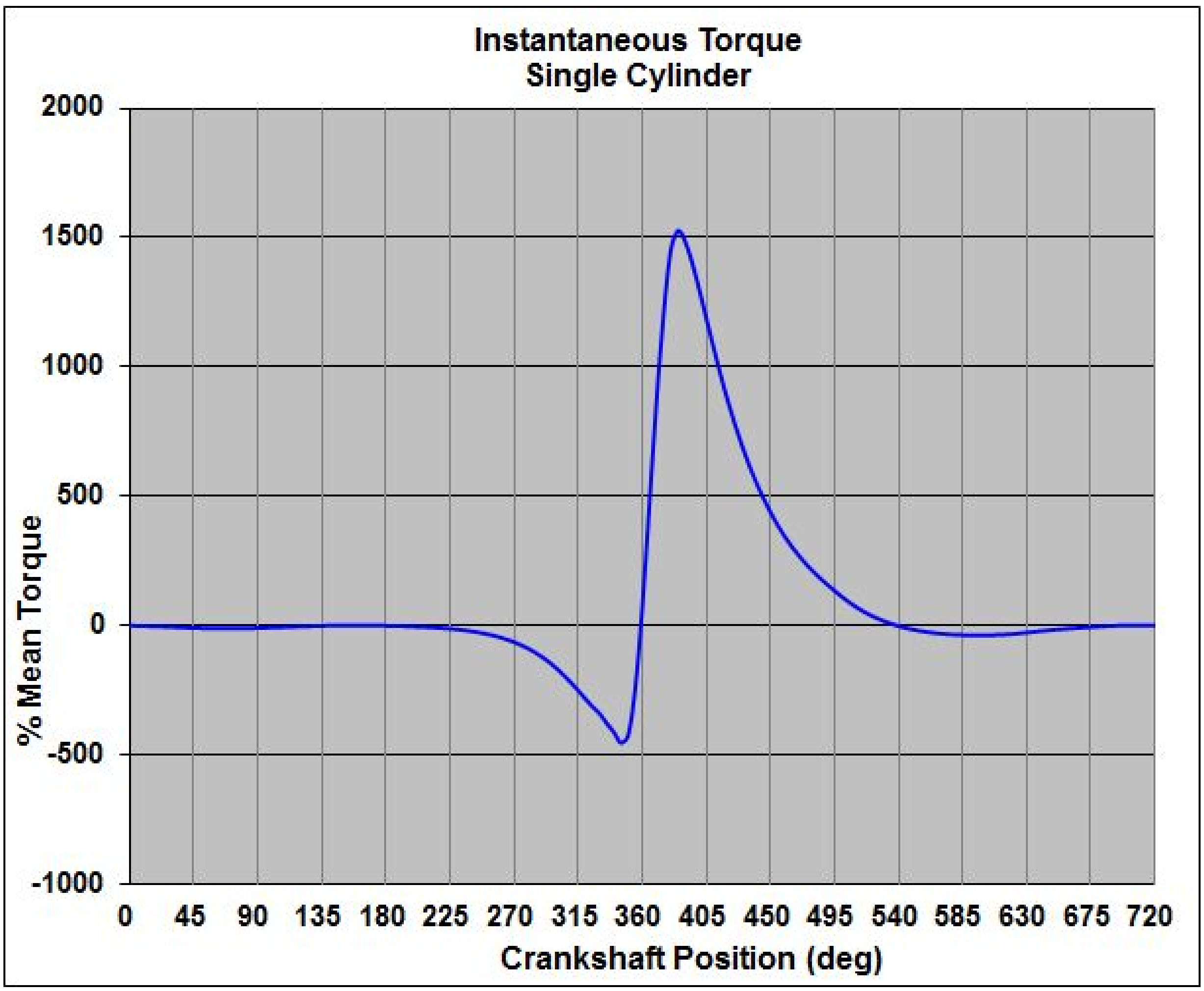


Figure 1: Characteristic torque pulse curve of a single-cylinder engine such as the YFZ450R. Source: http://www.epi-eng.com/piston_engine_technology/torsional_excitation_from_piston_engines.htm

It is important to understand that this is an un-damped response. There are many engineers who work to smooth this response curve. The addition of flywheels and the transmission serves to provide damping to the system and lessen the effect of torque pulses.
However, we will assume worst-case loading. To account for these pulses, a torque multiplier PF with a value of 16 for the YFZ is introduced.

However, this torque is not felt directly by the differential. Rather, the chain tension is actually what is felt.

While it is obvious that the engine may be used to accelerate the car, it can also be used to slow it down by backdriving the engine; this is called ‘engine braking’ and is employed routinely even if not intended by the driver. In such a situation, the worst-case loading is to treat the engine like a compressor.

Consider the change from bottom dead-center to top dead-center with valves closed. This will cause a volume change represented by the compression ratio (CR). Additionally, assuming air is adiabatic, we get the following relationship:

$$P_1 V_1^{\gamma} = P\,V^{\gamma} \;\Rightarrow\; P = \left(\frac{V_1}{V}\right)^{\gamma} P_1 \;,$$

where γ is the ratio of specific heats (about 1.4 for air). Kinematically, this creates a torque on the crankshaft.

$$M_{crank} = P A\, r \sin\theta \left[1 + \frac{r\cos\theta}{\sqrt{L^2 - r^2 \sin^2\theta}}\right]$$

Solving these equations for M and then maximizing M is done in EngBrake.mw because no analytic solution exists; Maple is used to generate a numeric solution.

Braking:

The caliper, mounted to the differential mount, must exert a moment on the brake disc that is capable of locking up the tires. This makes it clear that a reaction force is produced on the differential mount it is mounted to, but in actuality, the offset of the caliper produces loads much more complicated than this. Braking produces the following loads:

- A force from the bearings in both left and right differential mounts
- A couple from the bolts mounting the caliper
- A force from the bolts mounting the caliper Fbrake

Braking loads are smooth in comparison to the engine because they are simply friction induced. Thus, making the assumption that maximum braking load occurs under maximum tire grip is valid.

Tires:

These loads, however, are to some extent limited by the tires of the car. If we assume components in the drivetrain are lightweight, it is clear that the engine cannot provide torque greater in magnitude than the tires can resist. However, components in the drivetrain (including wheels) are not lightweight and do have gaps. Shock loads can be as high as any component in the system makes them.

However, we will still calculate the maximum tire-limited torque that can be applied to the differential. In normal operation, this is simply the torque from both tires combined, so we will lump the rear tires together.

In any load case, the reaction forces on the tires is calculated in the same fashion.

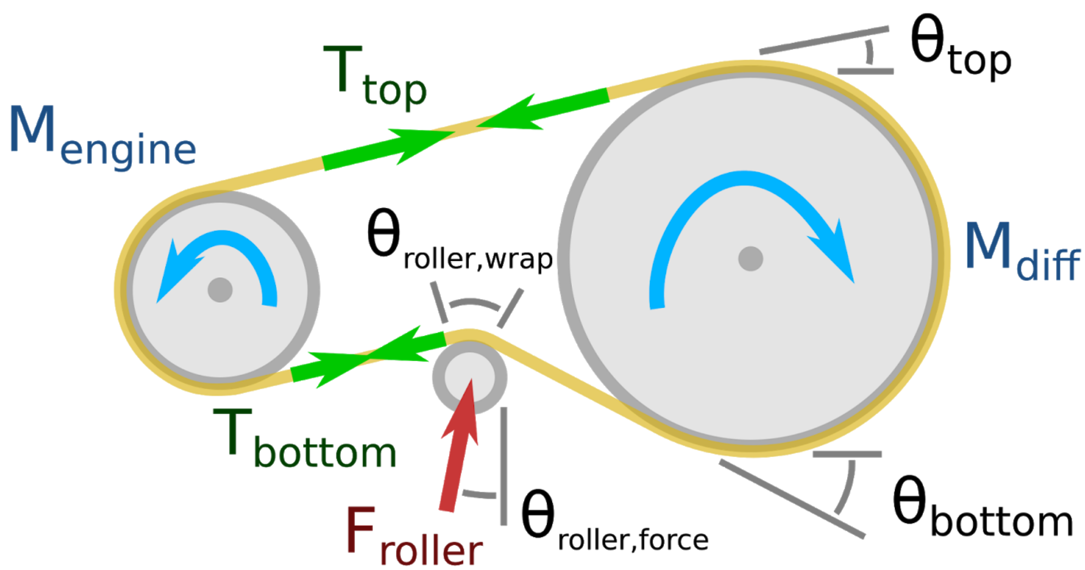


Figure 2: Schematic diagram showing key dimensions and internal forces of the chain drive

Roller reaction force:

The moments due to the bearing, losses, and inertia of the roller itself are neglected.

The sum of forces in the direction of F_{roller} leads to the following:

$$F_{roller} = 2\, T_{bottom}\, sin(\frac{\theta_{wrap}}{2})$$

These are broken down into X and Y components for ease of FEA analysis.

$$F_{roller,x} = F_{roller} sin(\theta_{roller,force})$$

$$F_{roller,y} = F_{roller} cos(\theta_{roller,force})$$

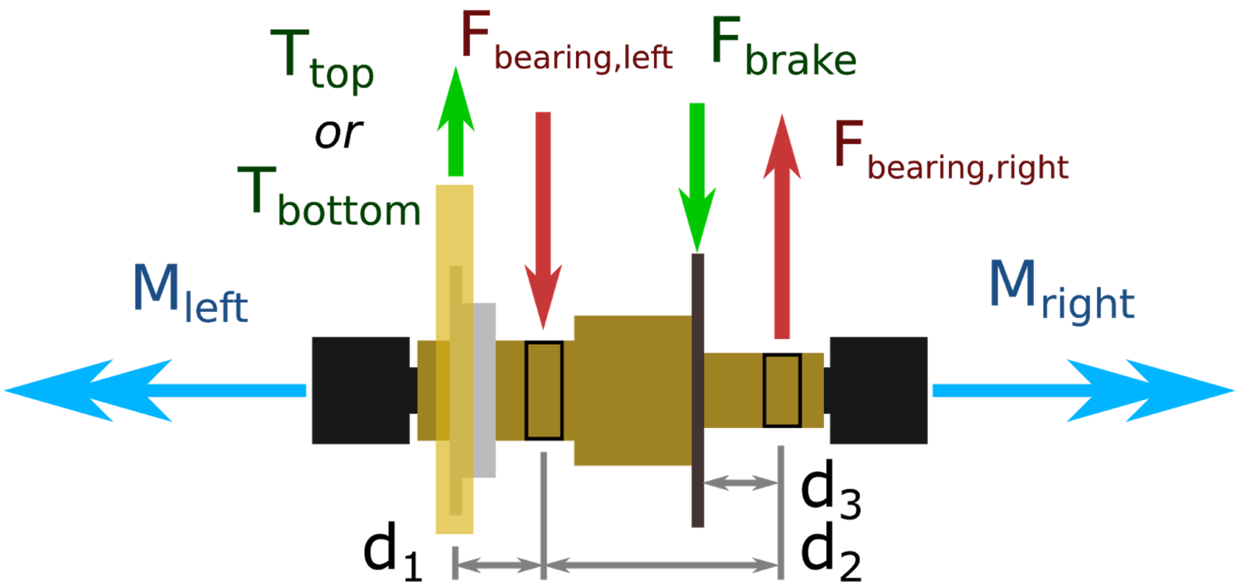


Figure 3: Free-body diagram of the differential. Note the plane being shown coincides with the line of action of the tension being considered, so reaction forces are parallel with the relevant chain tension

The sum of moments about the left bearing are zero for a lightweight system:

$$0 = d_2 F_{bearing,\ right} - d_1 F_{bearing,\ left}$$

The sum of forces in the direction of chain tension are zero for a lightweight system:

$$0 = T - F_{bearing,\ left} + F_{bearing,\ right}$$

This results in the following reaction forces:

$$F_{bearing,left} = T \frac{d_2}{d_2 - d_1}$$

$$F_{bearing,right} = T \frac{d_1}{d_2 - d_1}$$

During braking, the sum of moments about the right bearing are zero for a lightweight system:

$$0 = d_2 F_{bearing,left} + d_3 F_{brake}$$

The sum of forces in the direction of braking force are zero for a lightweight system:

$$0 = -F_{brake} - F_{bearing,\ left} + F_{bearing,\ right}$$

This results in the following reaction forces:

$$F_{bearing,left} = -F_{brake} \frac{d_3}{d_2 + d_3}$$

$$F_{bearing,right} = F_{brake} \frac{d_3}{d_2 + d_3}$$

The reaction forces can be broken down into x and y components to make FEA analysis easier. Refer to the Drivetrain Loads spreadsheet.

These cases all occur with fatigue. Let us consider how many cycles, then, each condition will happen. A typical FSAE endurance or autocross track has about 30-50 turns, and about 15-25 braking/acceleration events- these acceleration events usually come with 2-4 gearshifts. So, there are potentially 100 chain load cycles per lap! Per competition this means up to 5000 chain loading cycles between each dynamic event, practice, and especially endurance. For two competitions and practice time in the range of 50 times that of a competition, we're up to 500000 chain load cycles.

2. Loads placed onto Chassis by Drivetrain and Engine

By: Thaddeus Hughes
Date: 8/3/2017

From EngineLoadsCalculation.m we can calculate the loads that the engine and drivetrain place on the chassis.

Based on the input in figure 5.1, these are calculated to be as shown in figure 5.2.

```
% Dimensions updated and approved by Thad Hughes, 03 AUG 2017

%theta_tension = 5 *pi/180 % rad, -z from the -x; the angle of the top of the chain

theta_front_constraint = 10 *pi/180 % rad, -z from the +y; the angle that the front m

radius_eng_sprocket = 13*5/8/2/pi
radius_drive_sprocket = 34*5/8/2/pi

% Differential and chain drive planes
z_sprocket    = -4.44
z_diff_left   = z_sprocket + 1.75
z_diff_right = z_diff_left + 5.375

f_tension = 2000 % lbf



r_eng_sprocket_center = [-55.58,7.61,z_sprocket]
r_diff_sprocket_center = [-64,9.75,z_sprocket]

diff = r_diff_sprocket_center-r_eng_sprocket_center
phi = atan2(diff(2),-diff(1))
psi = asin((radius_drive_sprocket-radius_eng_sprocket)/norm(diff))

theta_tension = phi+psi
% should get 28.2 (based on 03 AUG 2017)

r_engine_rear = [-58.377,7.5,(-1.31+1.206)/2]
r_engine_front_left = [-41.6,3.95,-7.3]
r_engine_front_right = r_engine_front_left
r_engine_front_right(3) = -r_engine_front_right(3);

% Locations on the differential mount
r_diff_bearing_left = [-64,9.75,z_diff_left]
r_diff_top_left = [-57.75,11.286,z_diff_left]
r_diff_bot_left = [-59.4,5,z_diff_left]
r_diff_bearing_right = [-64,9.75,z_diff_right]
r_diff_top_right = [-57.75,11.286,z_diff_right]
r_diff_bot_right = [-59.4,5,z_diff_right]
```

Figure 5.1: Inputs to load calculation .m file

```
--- RIGHT DIFF LOADS ON TO CHASSIS ---

F_diff_top_right =

 -195.3196   51.2691         0


F_diff_bot_right =

 -417.1938  109.5084         0


V_diff_right =

   38.4963  146.6591         0

--- ENGINE LOADS ON TO CHASSIS ---

F_engine_rear =

   1.0e+03 *

   -1.8180    0.6328    0.4865


F_engine_front_left =

   56.8424  322.3690         0


F_engine_front_right =

   -1.9134  -10.8516         0


F_engine_lateral =

         0         0 -486.5194
```

Figure 5.2: Results from Load Calculation. Forces in lbf and oriented in standard forward-up-right X-Y-Z coordinates.

3. Gear Ratio Requirements

By: Thaddeus Hughes

Date: 7/26/2017

Based on the RoseLap - Summer Report document, we find the following:

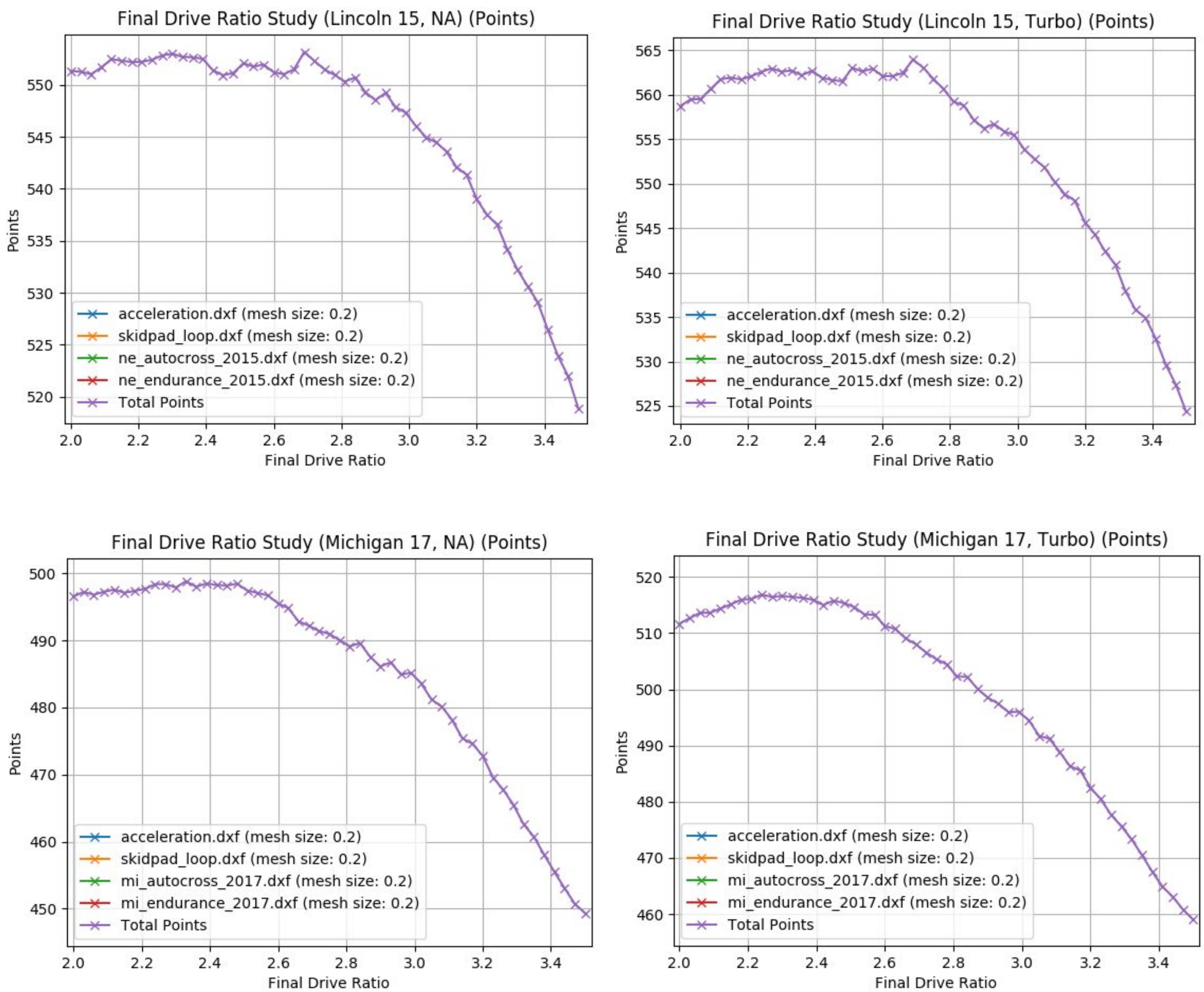


Figure 3.1: Overall points for turbocharger and without versus final drive ratio at both competitions.

This gives us the idea that a gear ratio between 2.2 and 2.4 will give us optimal performance. However, looking at the details shows a different picture. Take autocross/endurance performance compared against acceleration. Ideally, autocross courses should be ran with a tall ratio. Acceleration, though, does have a best gear ratio. If we could, we would pick the lowest ratio that allowed launch for autocross and endurance, and then the shortest ratio for acceleration and skidpad. This is not permitted under the rules, however.

If not, we will run a 13:30 at Michigan and a 13:35 at Lincoln. If we can, run a 13:36 on accel, and as tall as possible while still being able to launch on everything else.

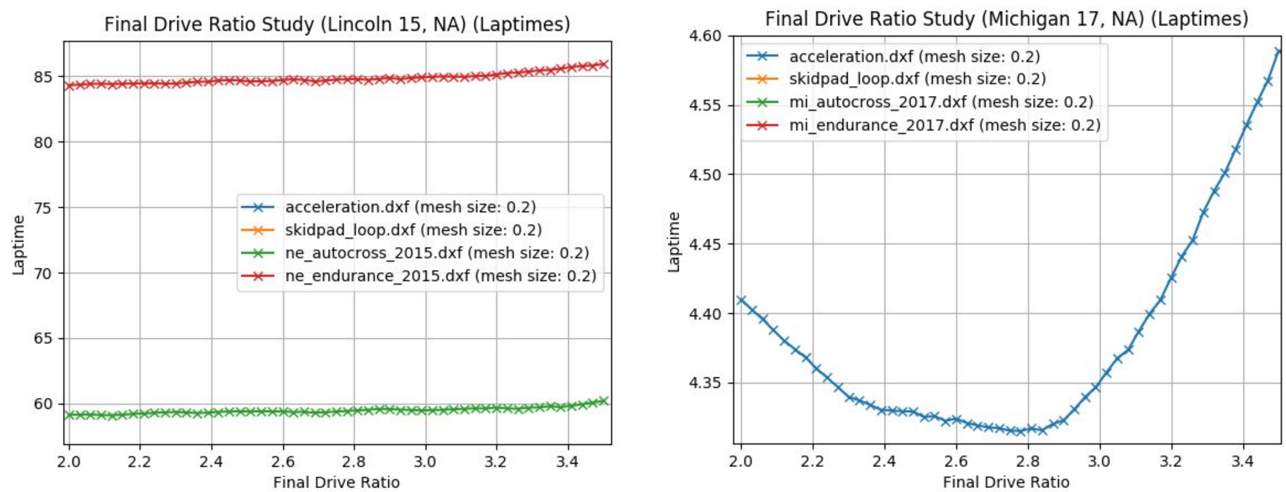
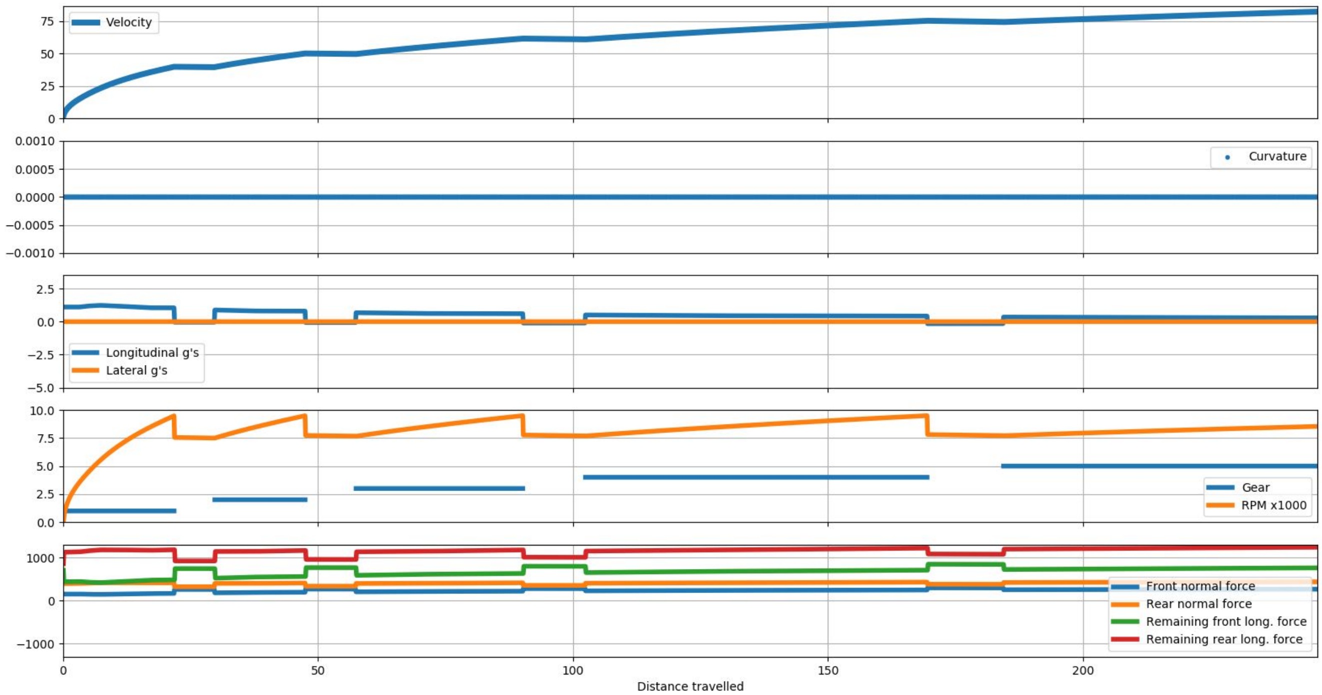


Figure 3.2: Comparison of final drive ratio effects on acceleration and autocross/endurance performance

Engine Limited Top Speed in MPH							
(RPM*60/(Extra Internal) * Engine/Diff * TireCircumference /InternalReduction							
Sprocket			Shifting Reduction (YFZ)				
Engine (Higher # means lower chain tension)	Diff	Ratio	*2.416*	*1.92*	*1.562*	*1.277*	*1.05*
14	*28*	*2*	**35.50**	**44.66**	**54.90**	**67.15**	**81.67**
14	*30*	*2.142857143*	**33.13**	**41.69**	**51.24**	**62.68**	**76.23**
14	*32*	*2.285714286*	**31.06**	**39.08**	**48.04**	**58.76**	**71.46**
14	*34*	*2.428571429*	**29.23**	**36.78**	**45.21**	**55.30**	**67.26**
14	*36*	*2.571428571*	**27.61**	**34.74**	**42.70**	**52.23**	**63.52**
~~*14*~~	~~*38*~~	*2.714285714*	~~26.15~~	~~32.91~~	~~40.45~~	~~49.48~~	~~60.18~~
~~*14*~~	~~*40*~~	*2.857142857*	~~24.85~~	~~31.27~~	~~38.43~~	~~47.01~~	~~57.17~~
~~*14*~~	~~*42*~~	*3*	~~23.66~~	~~29.78~~	~~36.60~~	~~44.77~~	~~54.45~~
13	*28*	*2.153846154*	**32.96**	**41.47**	**50.98**	**62.36**	**75.84**
13	*30*	*2.307692308*	**30.76**	**38.71**	**47.58**	**58.20**	**70.78**
13	*32*	*2.461538462*	**28.84**	**36.29**	**44.61**	**54.56**	**66.36**
13	*34*	*2.615384615*	**27.14**	**34.16**	**41.98**	**51.35**	**62.46**
13	*36*	*2.769230769*	**25.64**	**32.26**	**39.65**	**48.50**	**58.99**
~~*13*~~	~~*38*~~	*2.923076923*	~~24.29~~	~~30.56~~	~~37.56~~	~~45.95~~	~~55.88~~
~~*13*~~	~~*40*~~	*3.076923077*	~~23.07~~	~~29.03~~	~~35.69~~	~~43.65~~	~~53.09~~
~~*13*~~	~~*42*~~	*3.230769231*	~~21.97~~	~~27.65~~	~~33.99~~	~~41.57~~	~~50.56~~

Figure 3.3: Comparison of different gear ratios. Good ratios for overall points highlighted green, good ratios for acceleration highlighted blue.

FYI, this is what an acceleration run looks like. This is a time of 4.5s (a very feasible time- best times are around 4.1s), and a maximum velocity of 56 MPH.



4. Bearing Selection

By: Thaddeus Hughes
Date: 7/27/2017

Bearings on the drivetrain see only radial forces. We do need radial retention, however. NSK has graciously sponsored our teams' bearings on RGP006, and we are looking to do this again- this time, looking at all bearing possibilities, such as the ultra-thin section bearings. The Drexler differential has a 50mm bearing bore on the right side and 55mm on the left. NSK offers the following thin-section bearings:

Boundary Dimensions (mm)				Basic Load Ratings (N)		
d	D	B	r min	C_r	C_{0r}	Open
50	62	6	0.3	5 550	5 750	**NB 710**
	65	7	0.3	6 400	6 200	**6810**
	72	12	0.6	14 500	11 700	**6910**
55	68	7	0.3	6 850	7 100	**NB 711A**
	72	9	0.3	8 800	8 500	**6811**
	80	13	1	16 000	13 300	**6911**

We need a bearing to hold up for 600 miles of vehicle operation, and approximately a third will be spent under acceleration and a sixth under braking. Thus, 200 miles of acceleration (3000 lbf on the left bearing) and 100 miles of braking (1500 lbf on the right bearing).

We will now perform the necessary lifetime calculations to determine which bearings will be suitable for our application. Refer to http://www.astbearings.com/radial-ball-bearings-life-and-load-ratings.html for a more in-detail explanation of these calculations.

The lifetime of a ball bearing in revolutions is given as:

$$L_{na} = a_1 \times a_2 \times a_3 \times (Cr/P)^3 \times 10^6$$

Where Cr is the basic load rating, and P is the equivalent loading. a_1, a_2, a_3 are constants to derate/prorate life for operating conditions, reliability, and material.

Or, in miles:

$$L_{mi} = L_{na} \times r \times 2\pi / 5280$$

When a reliability of over 90% is required, the corresponding factor should be selected from the following table:

Reliability	90	91	92	93	94	95	96	97	98	99	99.6	99.9
a_1	1	0.92	0.84	0.77	0.64	0.62	0.53	0.44	0.33	0.21	0.1	0.037

We will target 98% reliability, so a1=0.33

a2 is built-in to the basic load rating of the bearing and should be left as 1. a3 will also be left at 1 because operating conditions for the bearings are good and normal.

Bearings undergo almost entirely radial forces, so P is simply the radial force.

Bearing #		NB711A	6811	6911	NB710A	6810	6910
Mass	kg	0.044	0.081	0.189	0.033	0.05	0.135
Loading		Tire-Limited Accel			Braking		
Cr	N	6850	8800	16000	5550	6400	14500
Cr	lbf	1539.94165	1978.3192	3596.944	1247.68995	1438.7776	3259.7305

a1		0.33	0.33	0.33	0.33	0.33	0.33
a2		1	1	1	1	1	1
a3		1	1	1	1	1	1
P	lbf	3000	3000	3000	1500	1500	1500
L_revolutions	rev	44634	94632	568789	189915	291220	3386766
L_miles	mi	39.84	84.46	507.64	169.50	259.91	3022.68

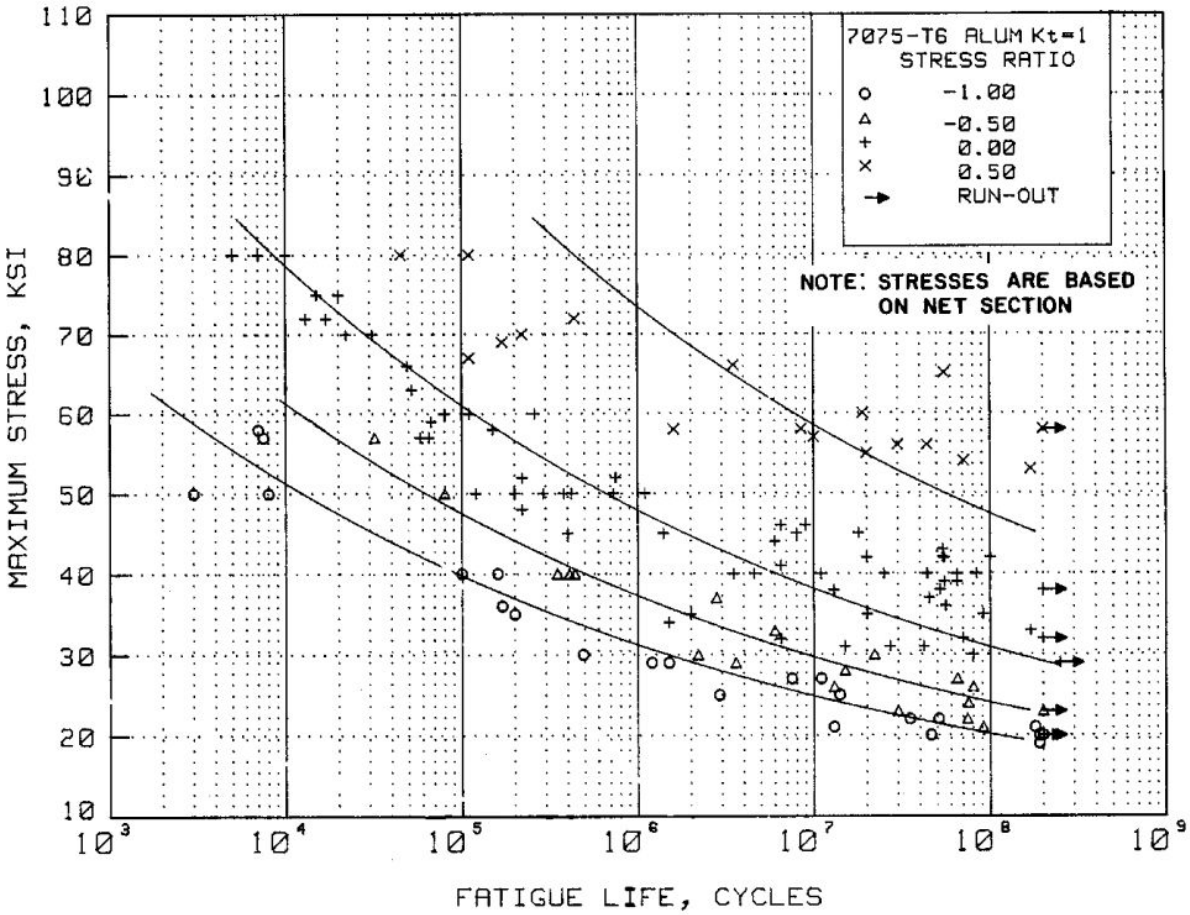
It becomes clear that the 6911 should be selected for the chain-drive side, but either the NB710A or 6810 would be acceptable for the other side. The weight difference is negligible (0.03 lb) but the lifetime increases by a significant amount, so we will err on the side of safety and use a 6810 for the right side. Additionally, we can't source a NB710A without a seal and the validity of fatigue calculations with these ultra-thin section bearings is uncertain.

6. Fatigue Calculations

Load cases all occur with fatigue. Let us consider how many cycles, then, each condition will happen. A typical FSAE endurance or autocross track has about 30-50 turns, and about 15-25 braking/acceleration events. Per competition this means about 600-1000 turns and 300-500 braking/acceleration events. Testing necessitates much more time than a competition, and these events can actually be very complicated and jerky rather than smooth, and so for safety, we will multiply by 500: 500,000 corners and 250,000 acceleration events.

An autocross or endurance course is usually around ¾ mile. The endurance event is to last approximately 13.66 miles. Assuming skidpad and autocross take minimal time, and the driver gets about 10 laps of practice time per competition, this brings the distance driven to ~25 miles per competition. Hopefully our drivers get in 9 times as much drive time on the car prior to competition, bringing the total distance autocrossing for two competitions and practice up to 600 miles.

From MIL-HDBK-5H, we find fatigue data for Aluminum 7075-T6 (unnotched) on page 3-382.



Luckily for us, lifetime has already been fitted in terms of maximum stress and R-ratio. R-squared = 81%.

$$log(N) \ = \ 18.21 \ - \ 7.73 \ log(S_{max}(1-R)^{0.62} - 10)$$

Most load cases we will look at are zero-based; that is, R = 0. Some will be fully-reversed; that is, R=-1.

$$S \ = \ \frac{10\ N^{0.129366}+226.859}{N^{0.129366}(1-R)^{31/50}}$$

7075-T6 Aluminum	Strength (ksi)			
	Zero-Based	with FOS =	Fully Reversed	with FOS =
R-ratio	0	1.5	-1	1.5
100,000	61.159	40.773	39.795	26.530
200,000	56.772	37.848	36.940	24.626
250,000	**55.441**	**36.961**	**36.074**	**24.049**
500,000	51.543	34.362	33.538	22.359
1,000,000	47.980	31.987	31.219	20.813

6. Halfshaft Selection / Design

The driveshafts must not fail under full load. The Taylor Race engineering halfshafts are made of 4340 steel, hardened to Rockwell 50 C

$$\tau = \frac{T\,d}{2\,J} \text{ where } J = \frac{\pi}{32}(d_{outer} - d_{inner})^4 \text{ for an annulus}$$

Minimizing torque steer is fairly important, and for this reason, torsional stiffness will be calculated. The angular deflection of a shaft is

$$\Theta = \frac{T\,L}{G\,J}$$

The results for each shaft under tire limitations are, then:

	T (in*lbf)	L (in)	G (psi)	Outer Diameter	Inner Diameter	Stress	Theta (deg)	dx at tire (in)	
Left Stub Axle	3,150	3.98	1.15E+07	1.1	0.65	13,726.81	0.49	0.08	
Left Halfshaft	3,150	13.56	1.15E+07	0.8	0.5	36,975.67	6.25	0.98	
Left Shaft							6.74	1.06	
Right Stub Axle	3,150	2.75	1.15E+07	1.1	0.65	13,726.81	0.34	0.05	
Right Halfshaft	3,150	15.133	1.15E+07	0.8	0.5	36,975.67	6.97	1.09	
Right Shaft							7.31	1.15	
Difference							0.57	0.090	8.13%

There is an 8% difference in displacements between the two shafts, which means an 8% difference in torque, assuming the tires are also infinitely stiff.

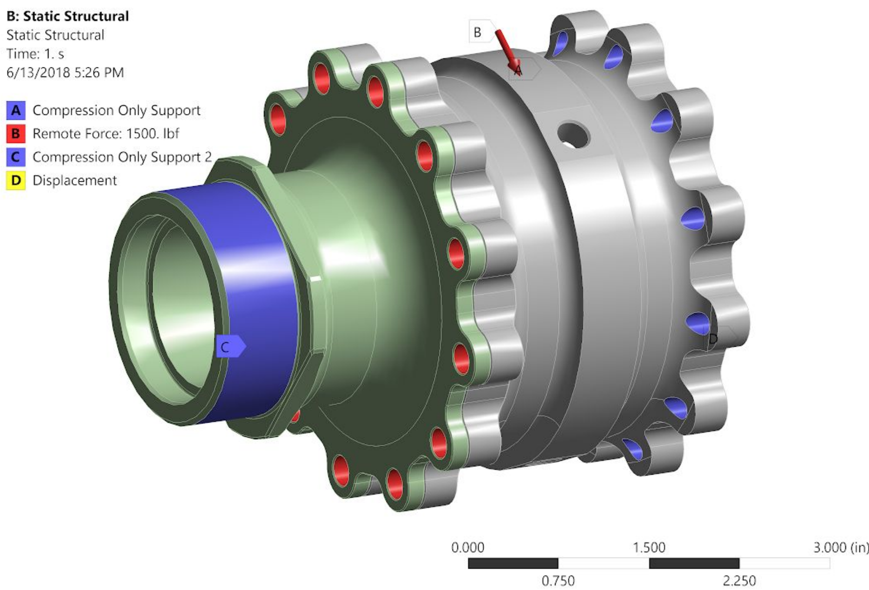
This could be combated with some fairly obvious modifications.

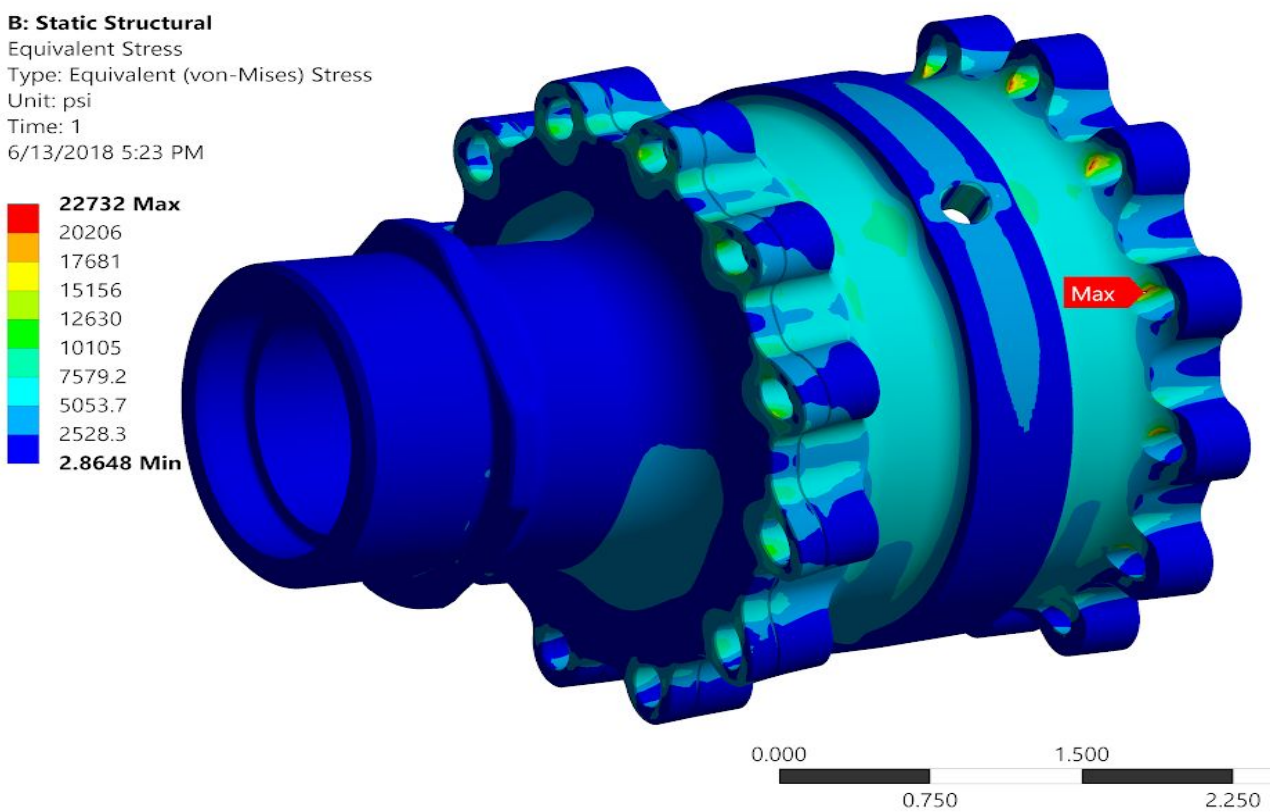
The shear stress seen is 37ksi (von-Mises: 64 ksi). The yield strength of the shafts is 200 ksi. This means there is a factor of safety of 3.13 against yield. From MIL-HDBK-5H, data on page 2-61, and an alternating stress ratio of -1 (braking being the same magnitude as acceleration), and 500,000 cycles, the fatigue strength is 110 ksi, bringing the factor of safety against fatigue to 1.71.

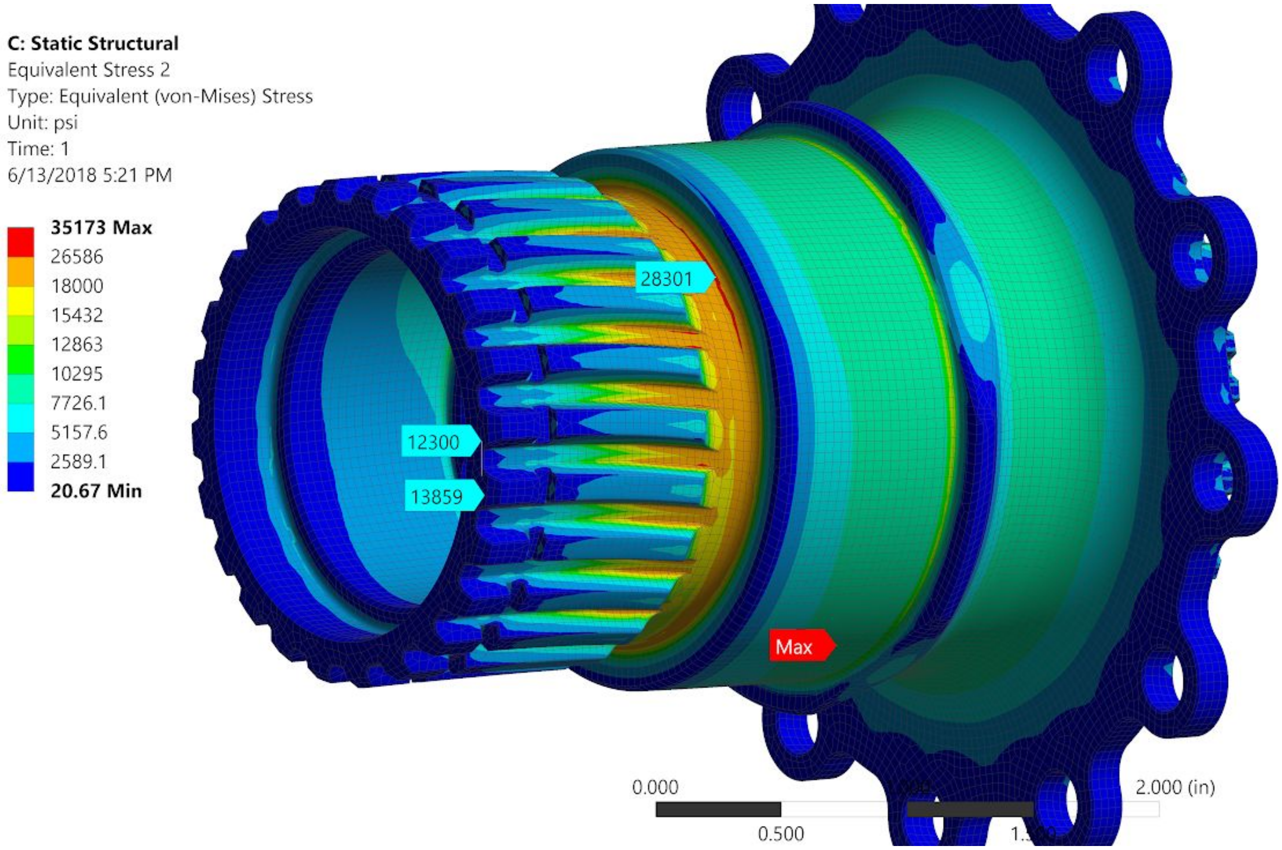
FYI 4340 is ridiculously strong.

7. Differential Casing Checks

The differential casing is checked for strength from both braking and acceleration fatigue (500,000 revolutions).







The failure point on the drive side is of some concern and probably worth doing some stress relief. Investigating this point reveals the region is in compression, so this is less of a concern.

8. Differential Carrier Design

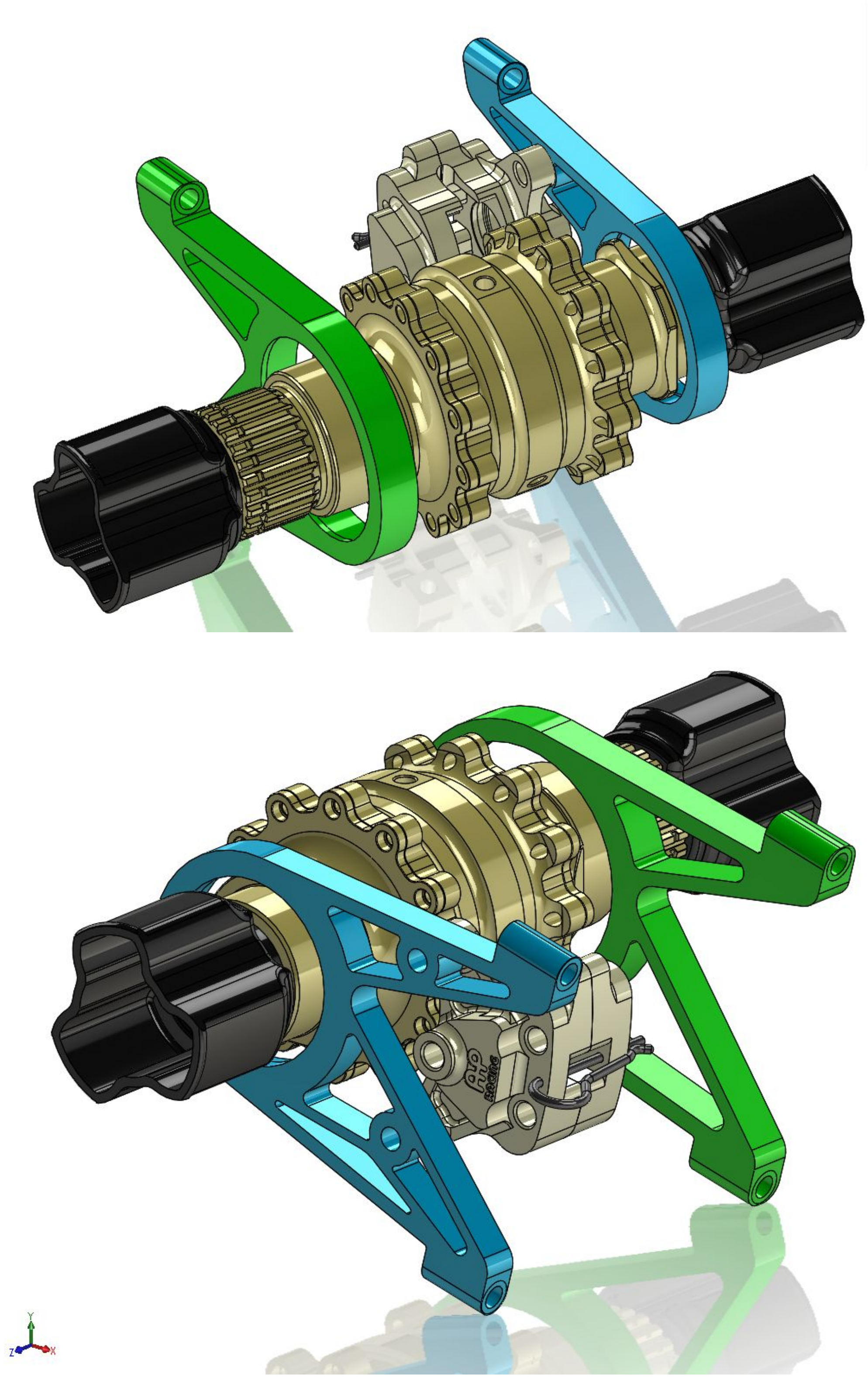


Figure 5.1: Initial shim-tensioned drivetrain design. Limited hand calculations amounting to chassis loads has been ran.

The left-side differential mount is placed into ANSYS to analyze for stress and fatigue. There are two main points of stress: around the lower bolt hole and around one of the fillets.

First, mesh convergence is studied. The criterion will be von-Mises stress for the entire component.

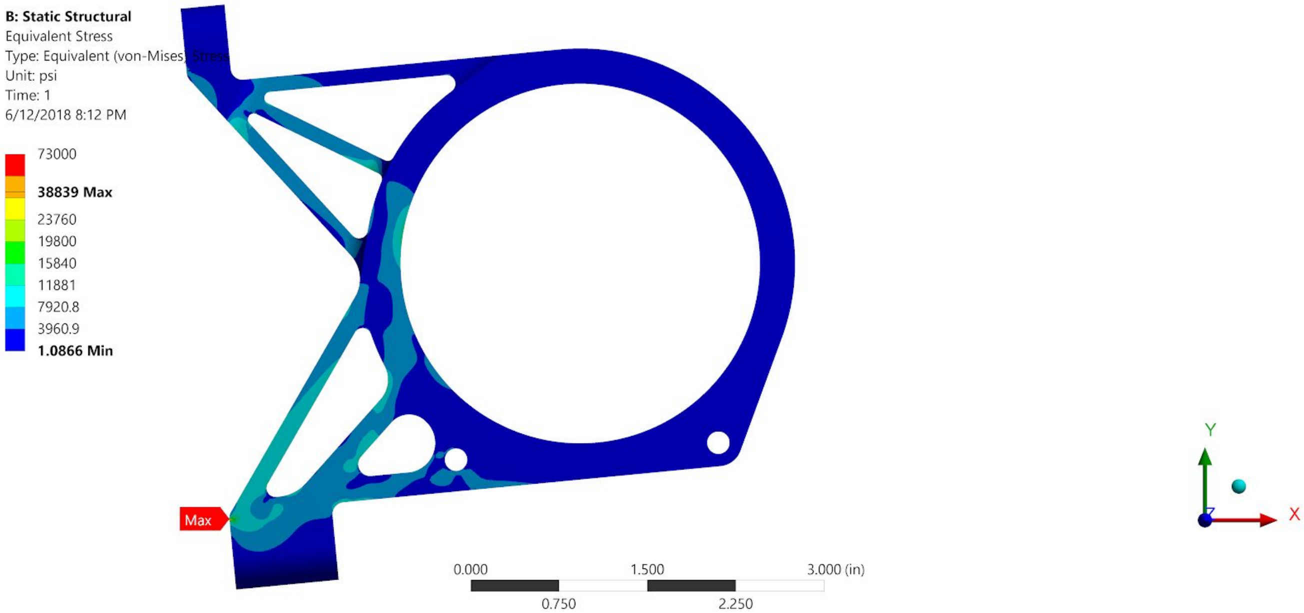


Figure 5.4: Results from left mount mesh convergence, design point 5.

Body Mesh Size (in)	von-Mises Stress (psi)
0.1	22538.65
0.08	24002.83
0.06	23910.63
0.05	23513.78
0.04	24470.66
0.03	24460
0.025	24226

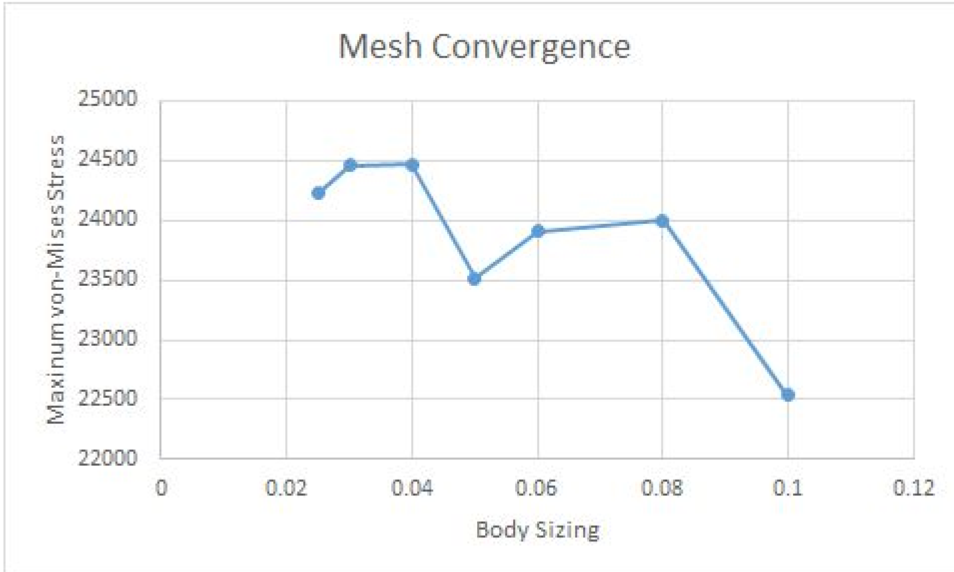


Figure 5.5: Results from left mount mesh convergence study

When the mesh size hits 0.025 inches, a singularity occurs at the base support. Thus, we will use the mesh sizing of 0.03 inches to balance speed and accuracy.

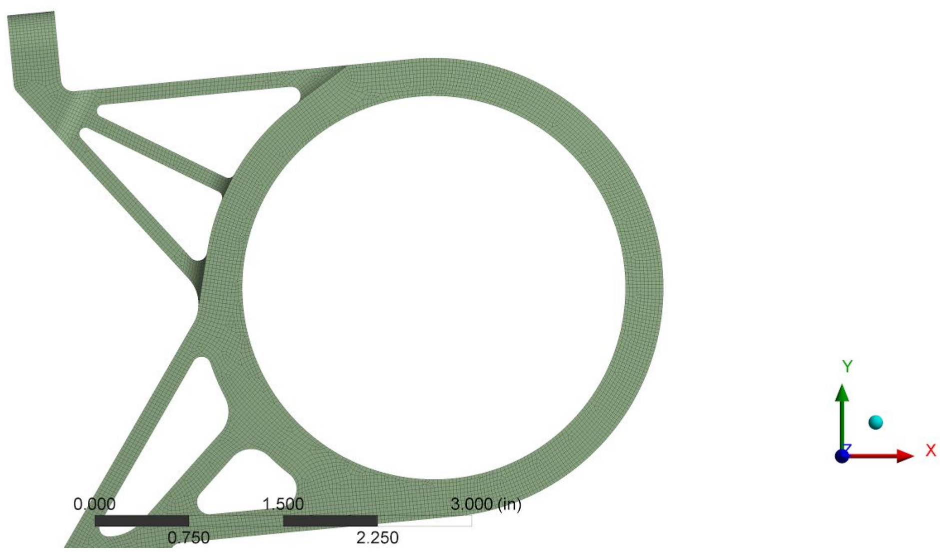


Figure 5.6: Mesh used in Analysis

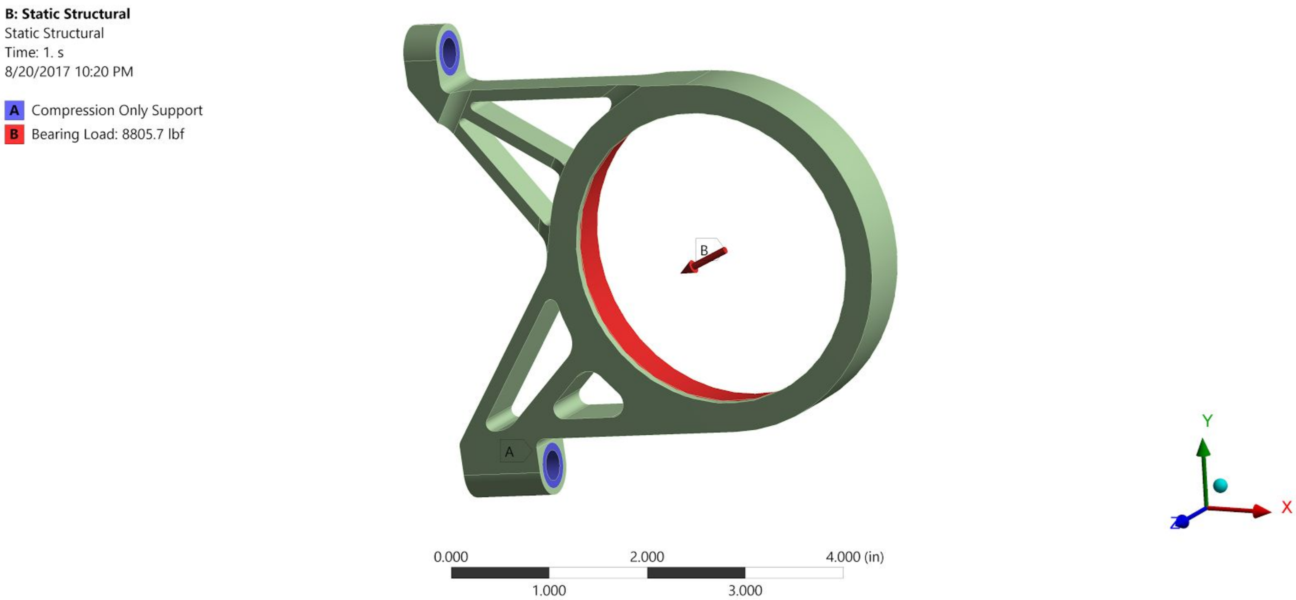


Figure 5.7: Setup for analysis

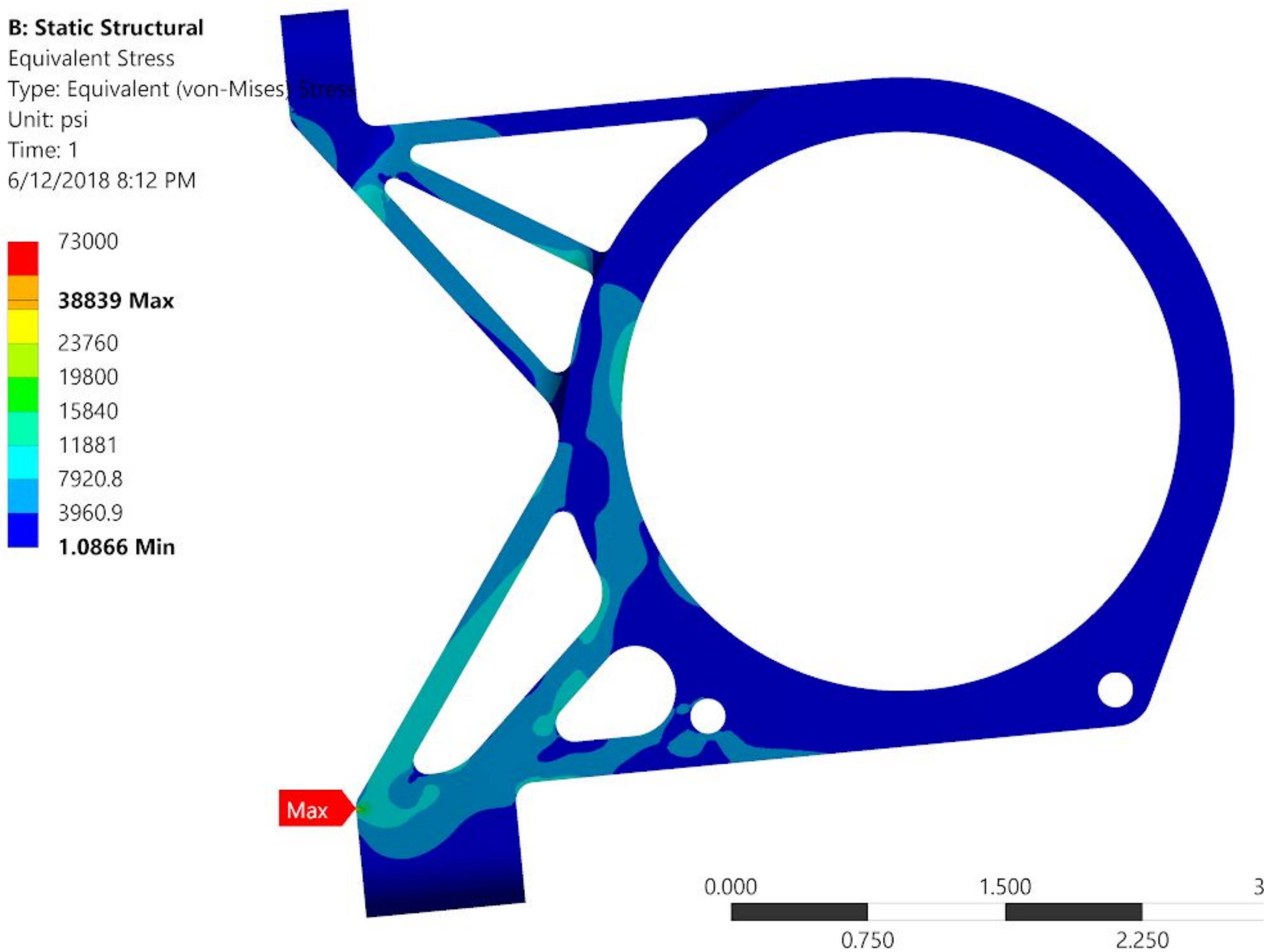
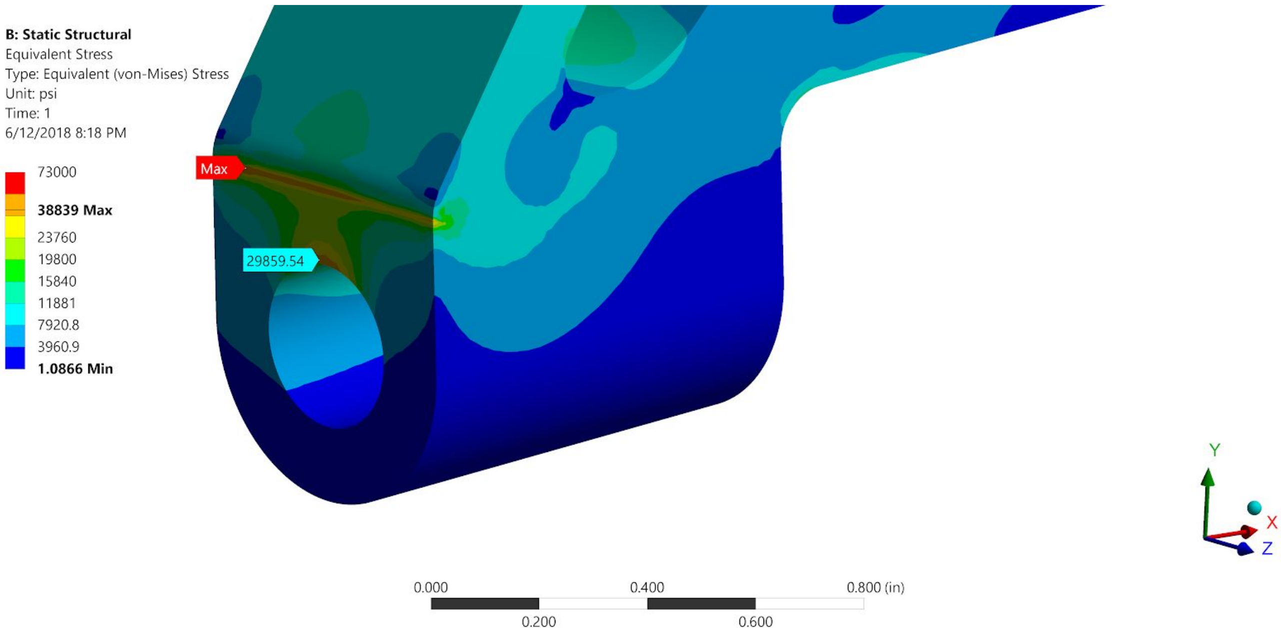
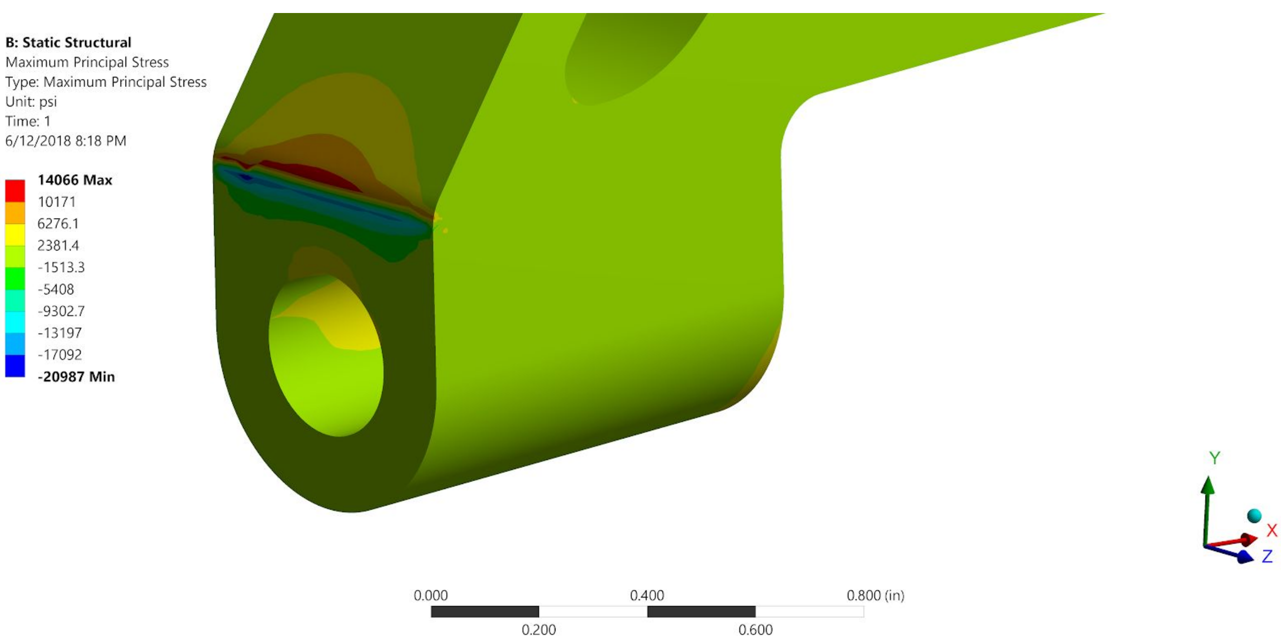


Figure 5.9: Stress under tire limited acceleration. There is a small singularity, but otherwise, everything is beneath the maximum allowable stress.



This region has a geometric discontinuity which causes the singularity.



The maximum principal stress is very negative, however- meaning this region is not subject to fatigue.

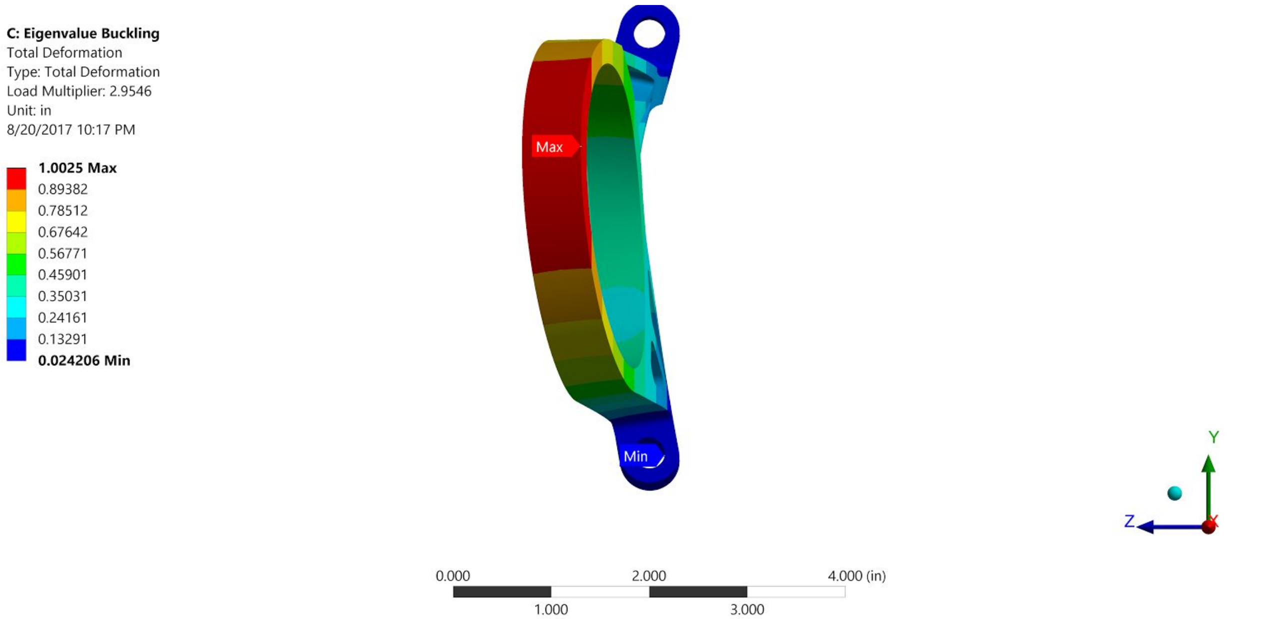


Figure 5.10: First buckling mode (FOS 2.9546)

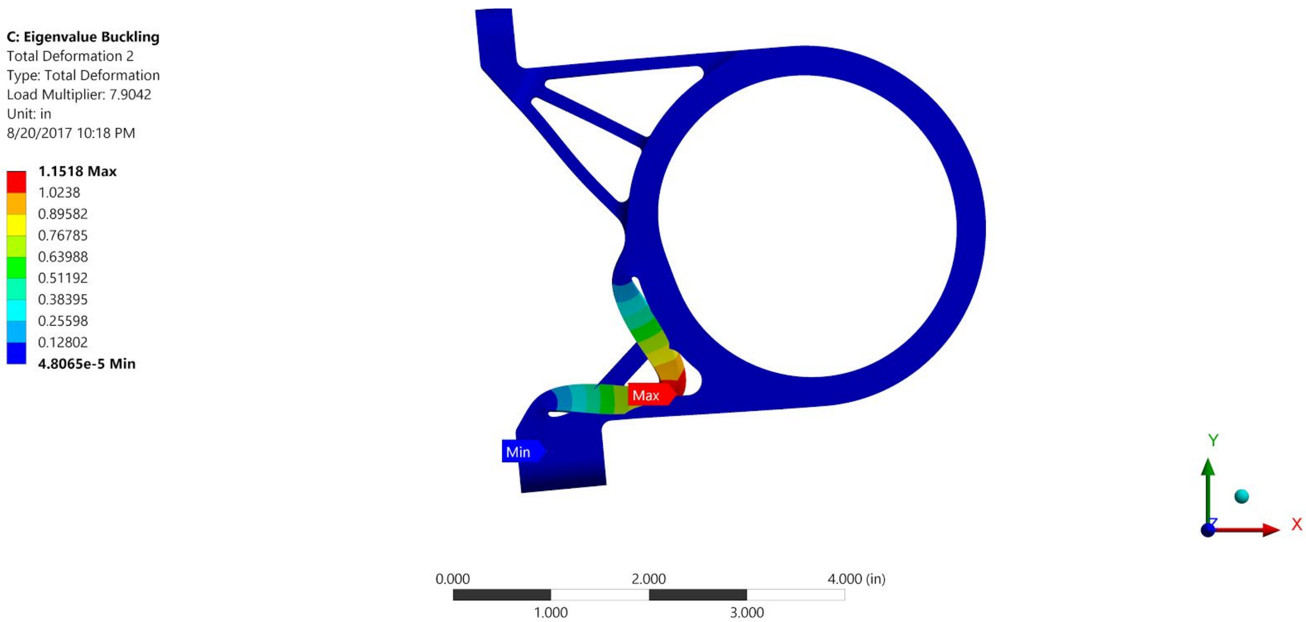


Figure 5.11: Second Buckling Mode (FOS 7.9042)

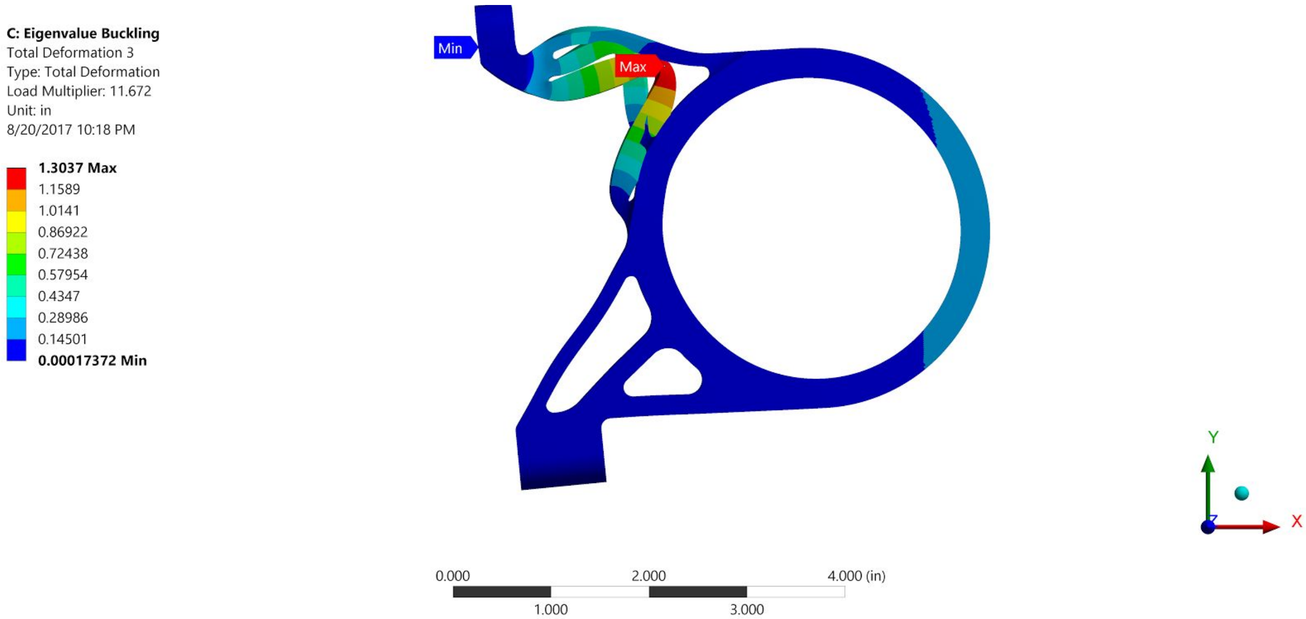


Figure 5.12: Third Buckling Mode (FOS 11.672)

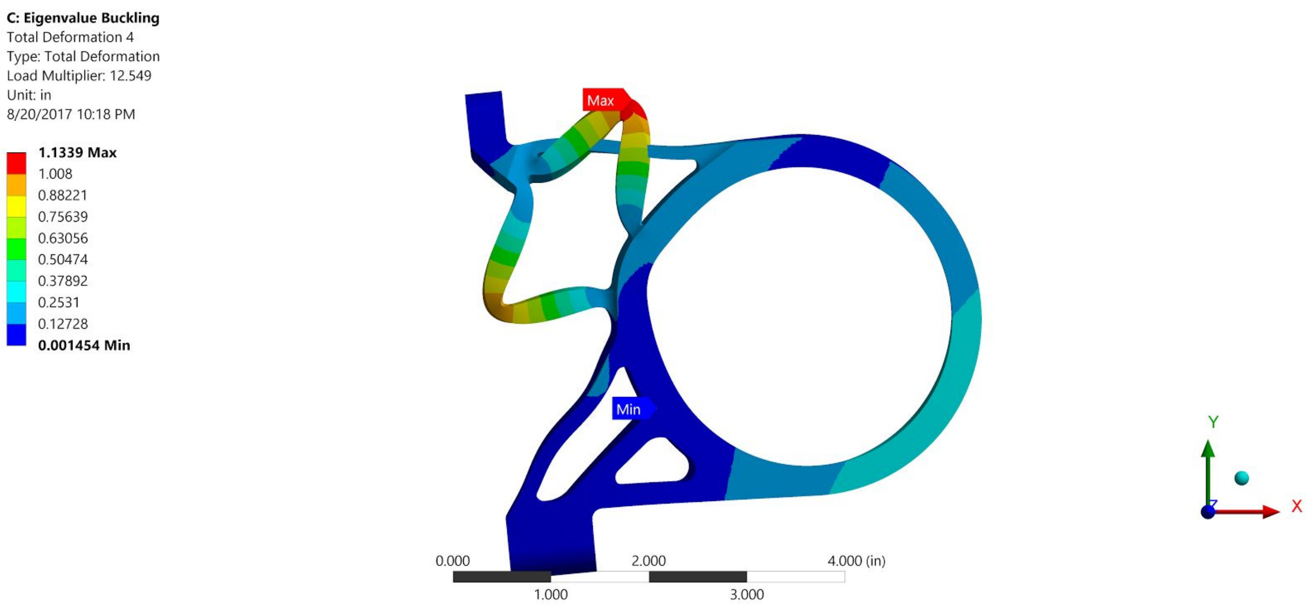


Figure 5.13: Fourth Buckling Mode (12.549)

9. Chassis Tube Design

Tubeframes like tubes in axial and bending loads. Torsion loads suck. Previous engine mounts have placed the tubes in- you guessed it- torsion. We also have done very minimal analysis of the tubes historically.

We need to make a mount that:

- Allows assembly (namely, the bolt that holds the engine to the mount)
- Survives engine loads (FOS of 2.0 against yield during tire-limited acceleration)
- Survives braking loads (FOS of 2.0 against yield during tire-limited braking)
- Connects the differential mounts
- Allows for chain clearance

- Minimizes weight

The FOS of 2 against yield will also ensure fatigue life is acceptable (considering the rule of thumb that steels have a endurance limit around half of their UTS)

Analysis is conducted with the following set-up:
- Engine mount tubes are cut from the rest of the chassis and held fixed at their ends
- Mounts are connected with revolute + translational joints. This means the analysis is not valid for stresses around bolt holes.
- Loads are calculated from the section above and applied in appropriate locations on the drivetrain.
- A joint is added between the two engine mount bolts.

After a few different designs, Thad has come up with the following design: tubes with tabs to an aluminum engine/diff hanger mount. This means tubes take loads mostly in axial and bending loads, since the tabs are rotated to transmit compressive/tensile forces rather than shear, and they cannot transmit moments. By making tabs that spread out the force, we can use 0.035" tubing (maybe even thinner if we are adventurous) in the rear tubes, aside from the lower crossmember being 0.065". The tubes are also coped so the crossmember goes 'through' both joints- providing greater weld joint area and better stress flow. Tabs are gusseted to maximize area through which forces are applied.

The weight of the mount is broken down as follows:
- Tabs: xx lbs
- Upgrading bottom tube from 0.035" to 0.065": 0.31 lbs
- Aluminum mounts: 0.6 lbs (could go down with more careful analysis)
- Weight savings from improved drivetrain design: approx. -0.75 lbs

Compare to RGP006:
- Tabs: xx lbs
- Upgrading upper tube from 0.035" to 0.095": 0.6 lbs
- Steel mounts: 0.35 lbs

The ANSYS analysis can be found on the COMPENSATOR under D:\ANSYS Analysis\Chassis-007\POWERTRAIN-CHASSIS.wbpj

Body Sizing	**Equivalent VM stress (psi)**	**Aluminum Fatigue FOS**
0.1	34908.7	1.423912
0.08	37047.63	1.439803
0.06	37880.13	1.363153
0.05	37619.1	1.325528
0.04	38612.62	1.354575

0.035	38729.33	1.332482
0.03	40191.32	1.334002

Figure 6.1: Mesh convergence study

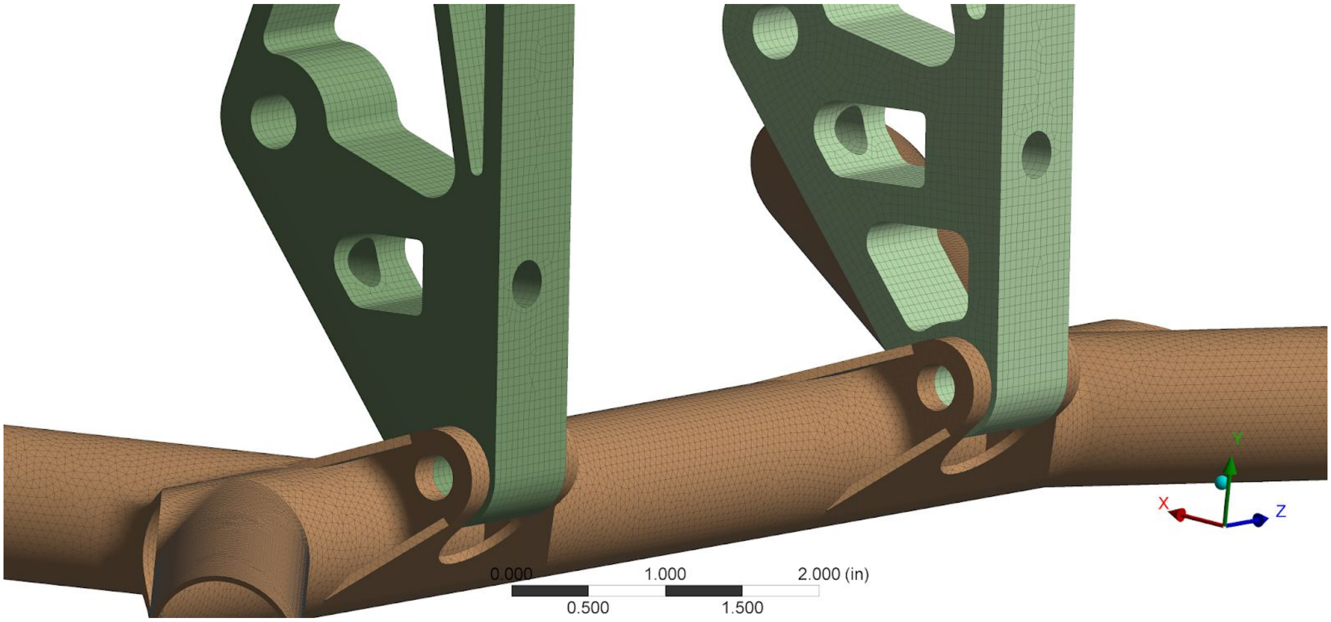


Figure 6.2: Mesh used in analysis. Quads/tris for hangers, tris for steel tubes

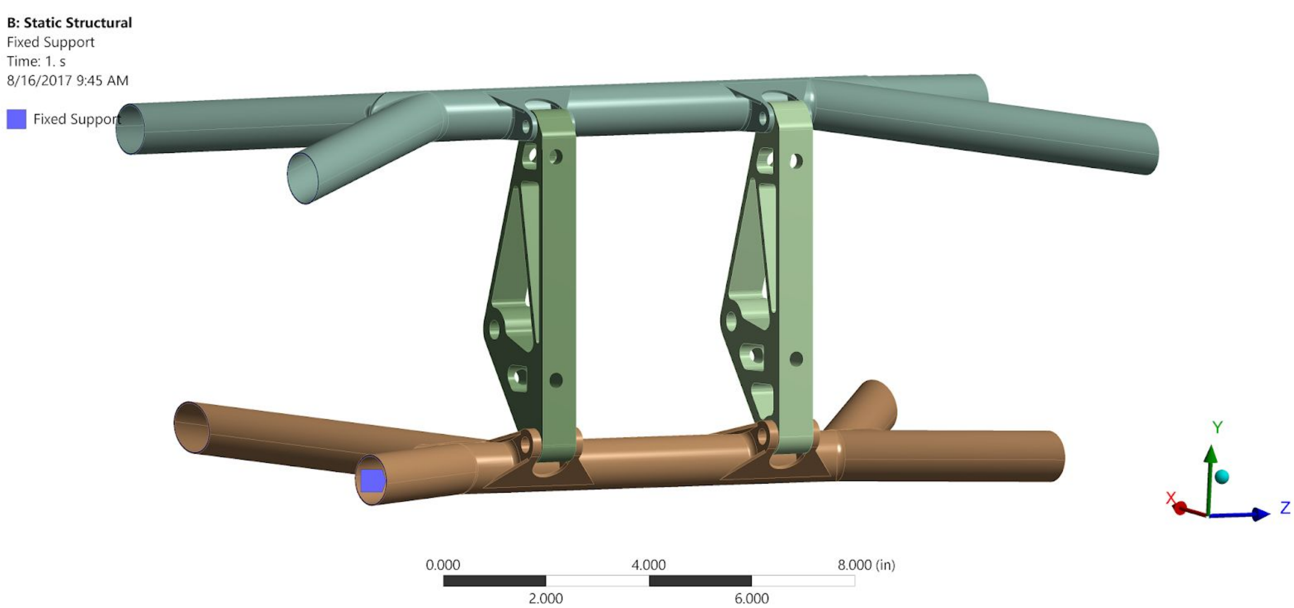


Figure 6.3: Fixed supports applied to 8 ends of tubes

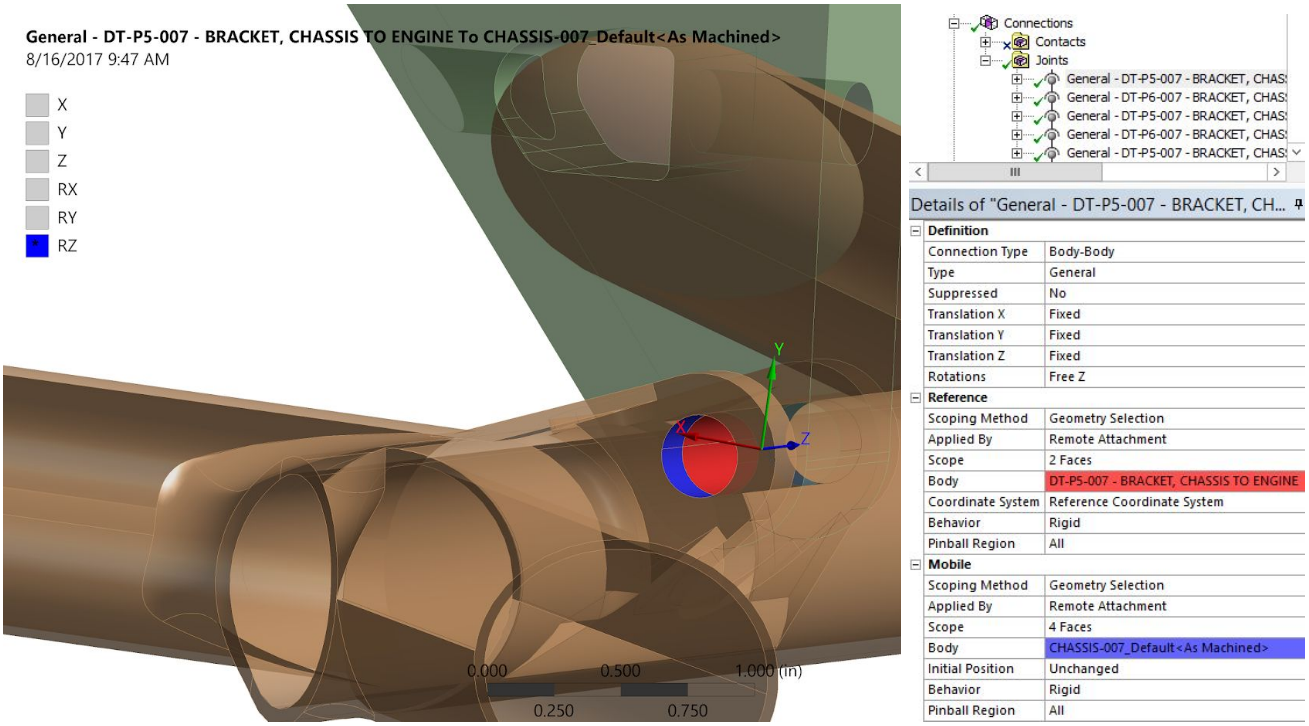


Figure 6.4: Sample joint used to connect steel tabs to aluminum hangers (1 of 4)

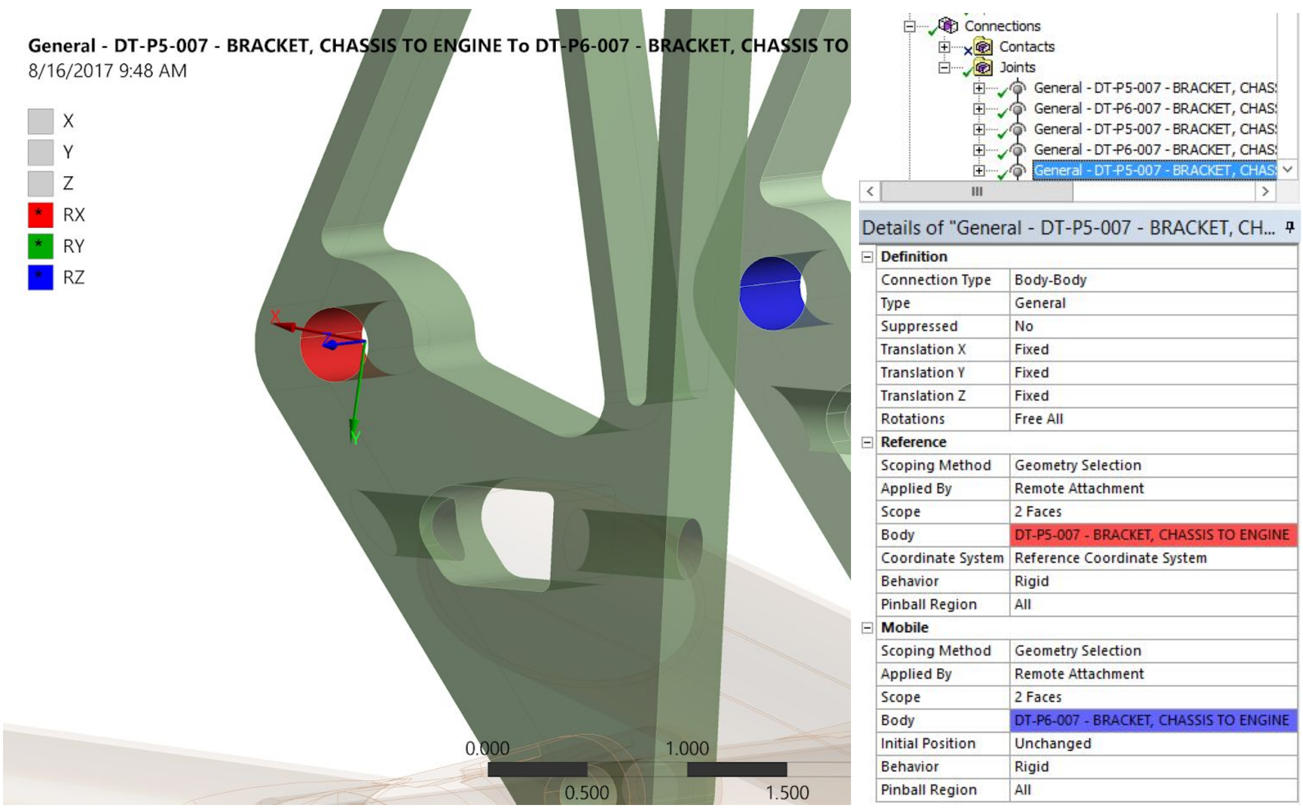


Figure 6.5: Joint used to connect sides of the engine mount

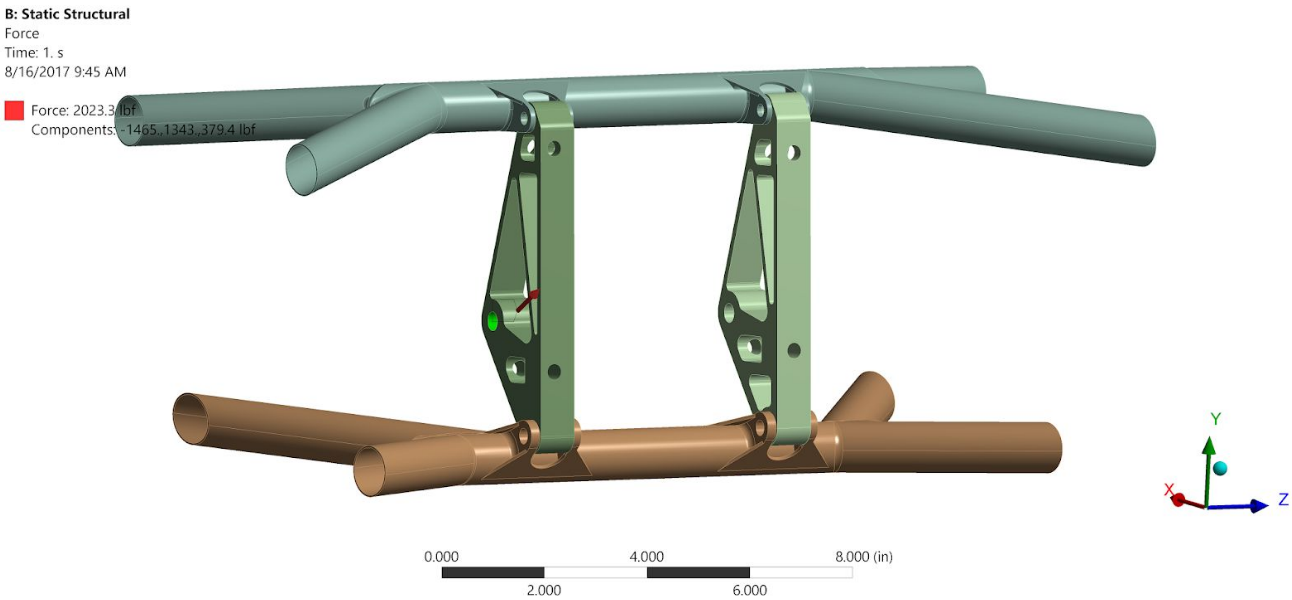


Figure 6.6: Engine force applied to aluminum hangers

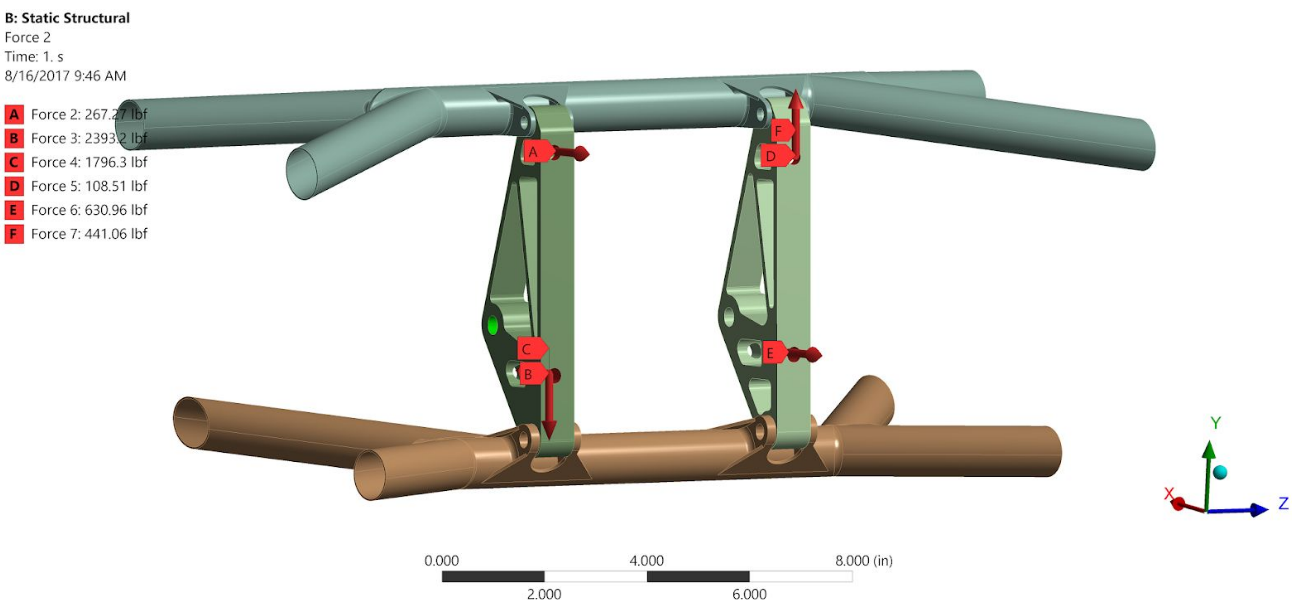


Figure 6.7: Drivetrain force applied to aluminum hangers

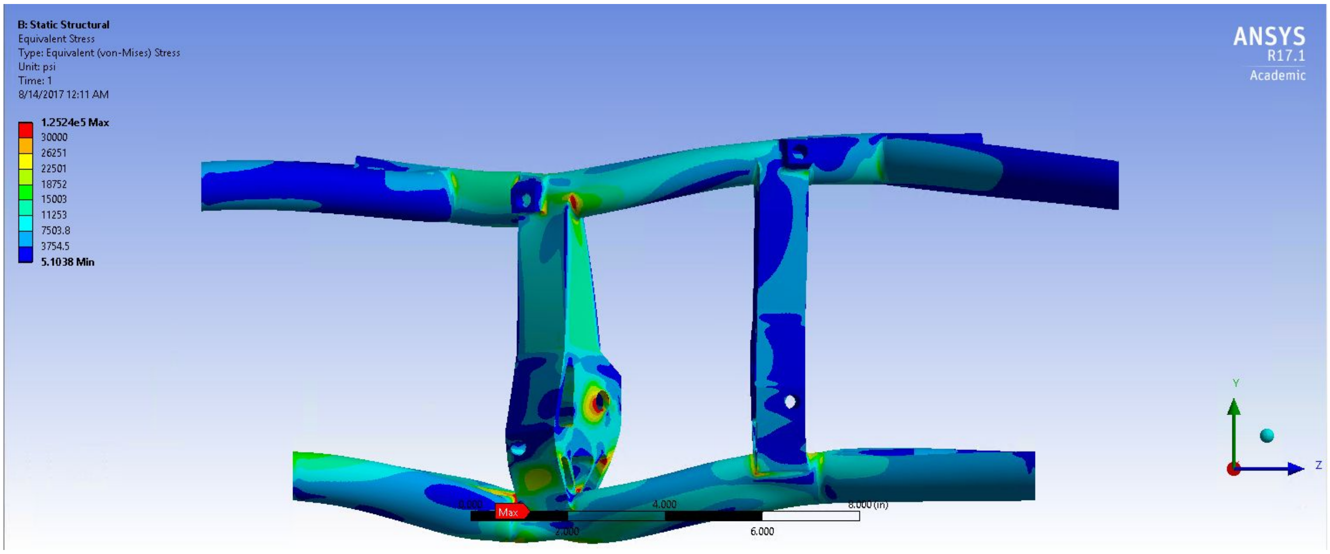


Figure 6.8: Welded steel engine mount concept, von-Mises stress

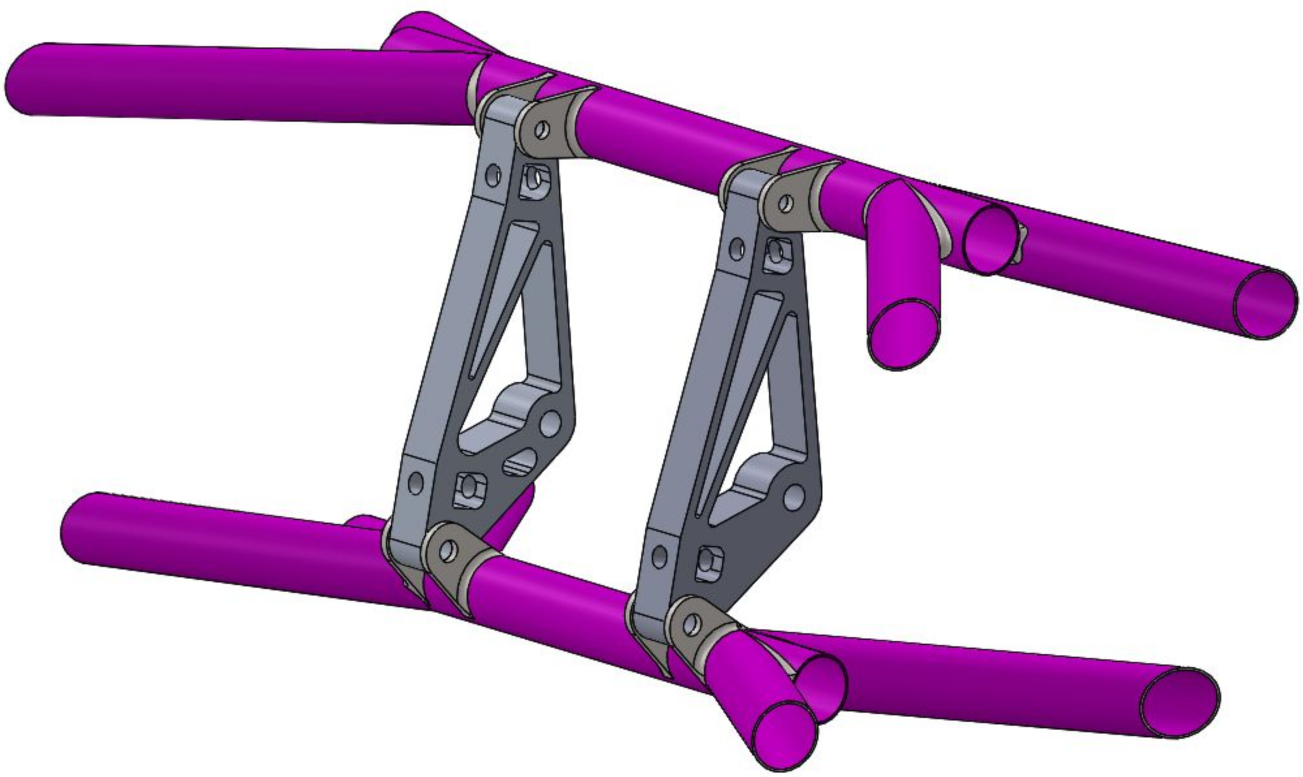


Figure 6.9: Initial concept of aluminum engine mount

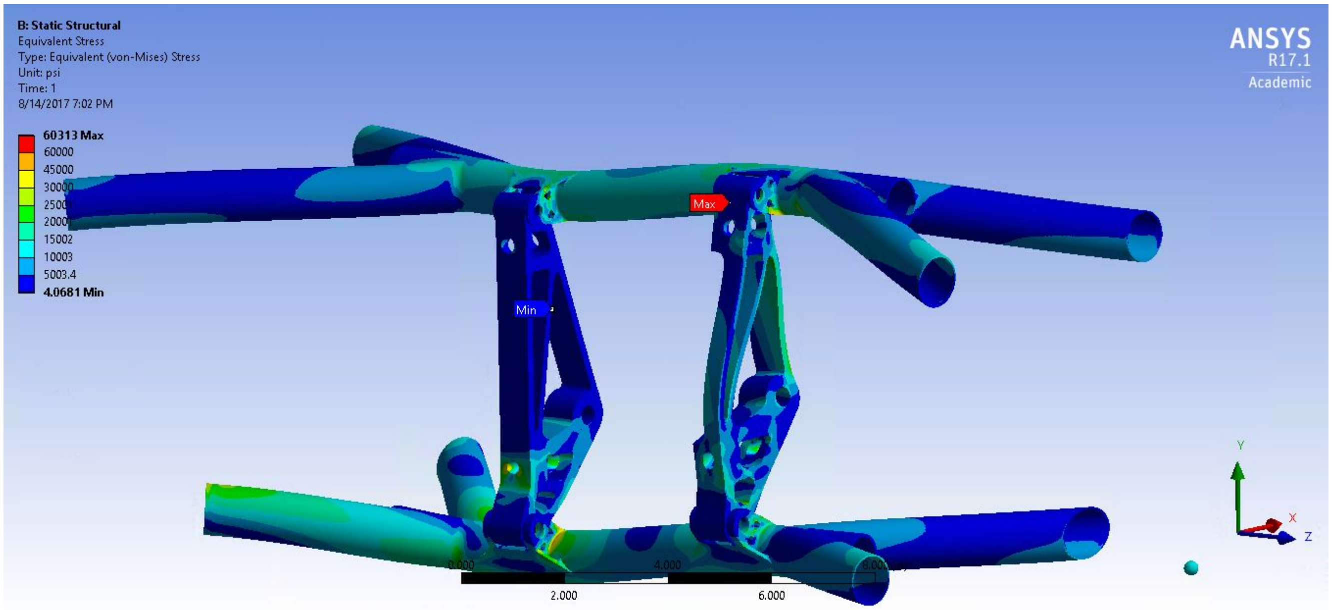


Figure 6.10: Initial concept of aluminum engine mount, von-Mises stress

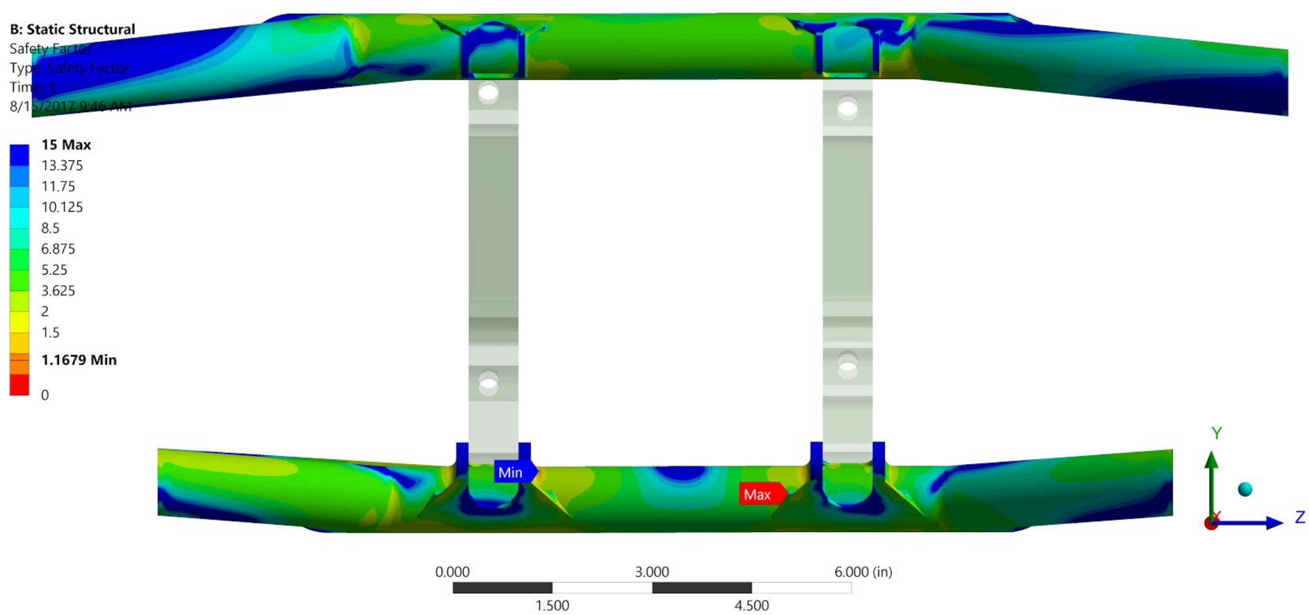


Figure 6.11: Nearly finished engine mount tubes, FOS against yield. Issues are the bottom inner tab fillets.

Validity of the stress around the area of the fillets is definitely questionable. The joint is not fully bonded, and the fillet is in a heat affected zone (HAZ). Still, we can make some cheap modifications to the tabs to better distribute the force.

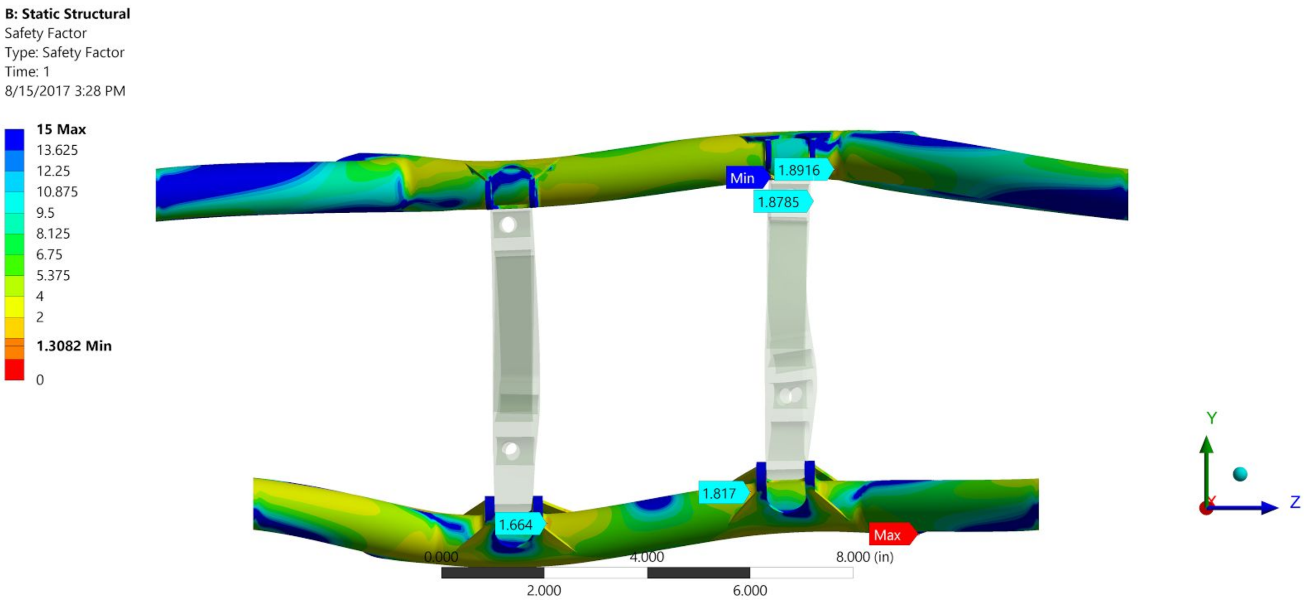


Figure 6.12: Nearly finished engine mount tubes, FOS against yield. Issues are around many weld fillets for tabs, and one tube joint near the top. Yellow and above are above a FOS of 2.

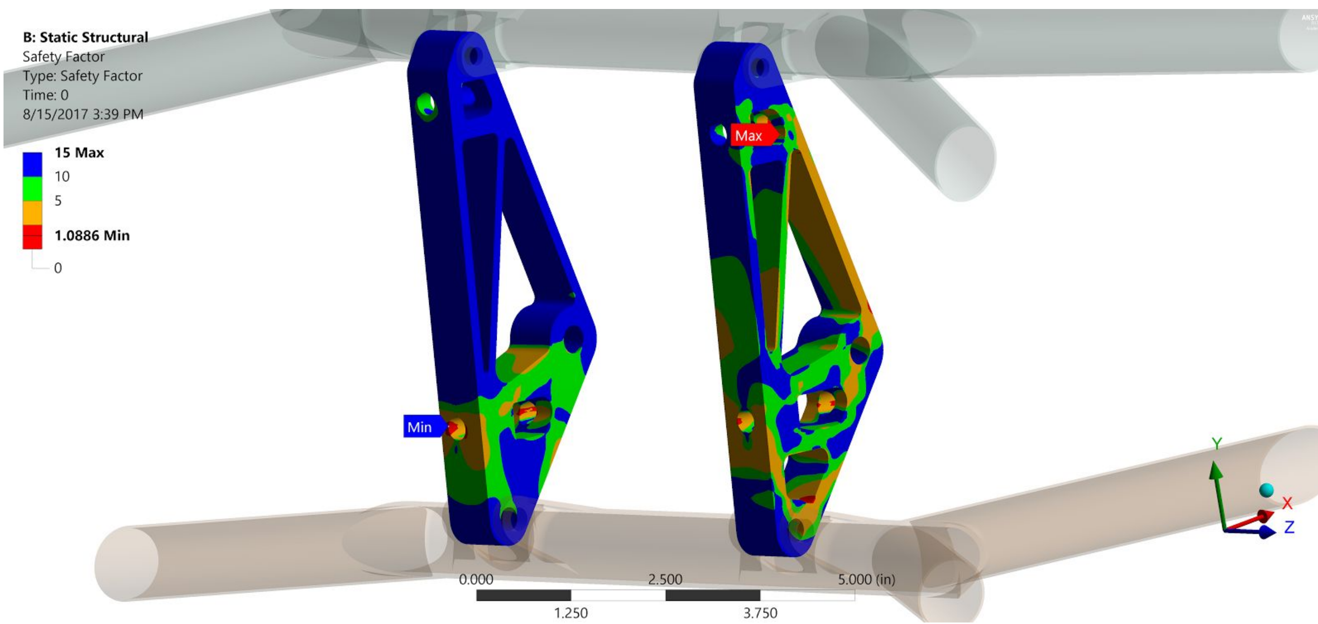


Figure 6.13: Hanger fatigue FOS. Red is below 2. Results around holes are not accurate (may be under or over conservative) because of the way loads are applied. Significant room for weight reduction.

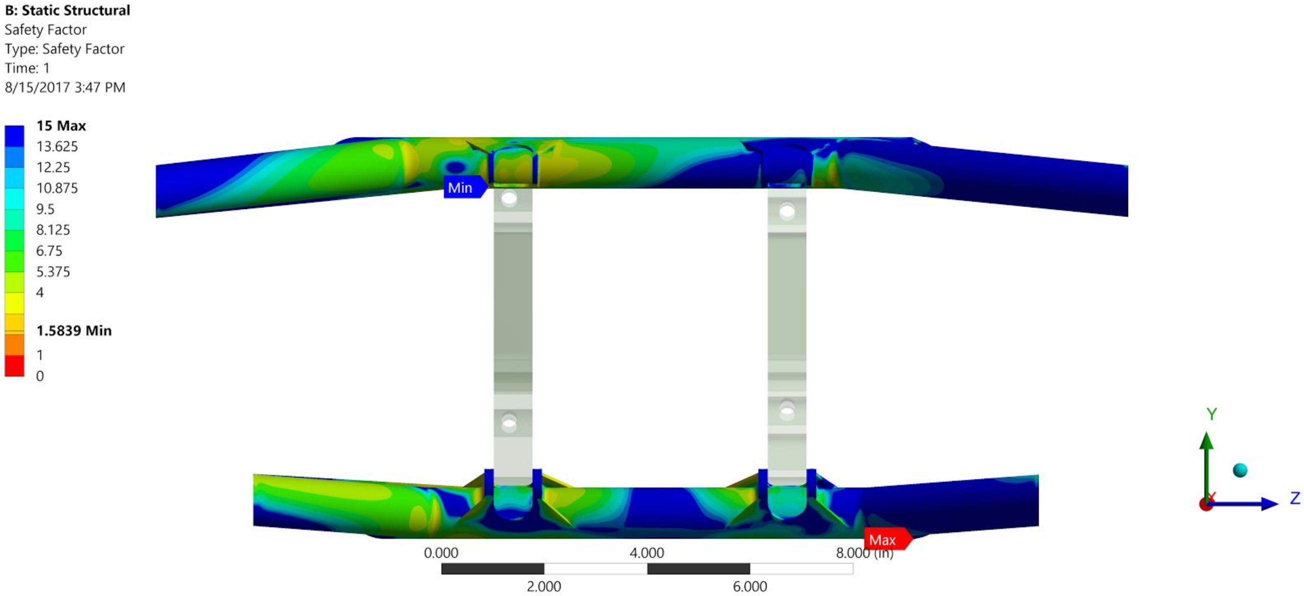


Figure 6.14: Nearly finished engine mount tubes, but with engine loads only applied to the left side hanger (and joint left intact). In this, the only issue is around the upper tab where a tab web was forgotten. Adding this web would easily put the design in safe operating region under this loading condition.

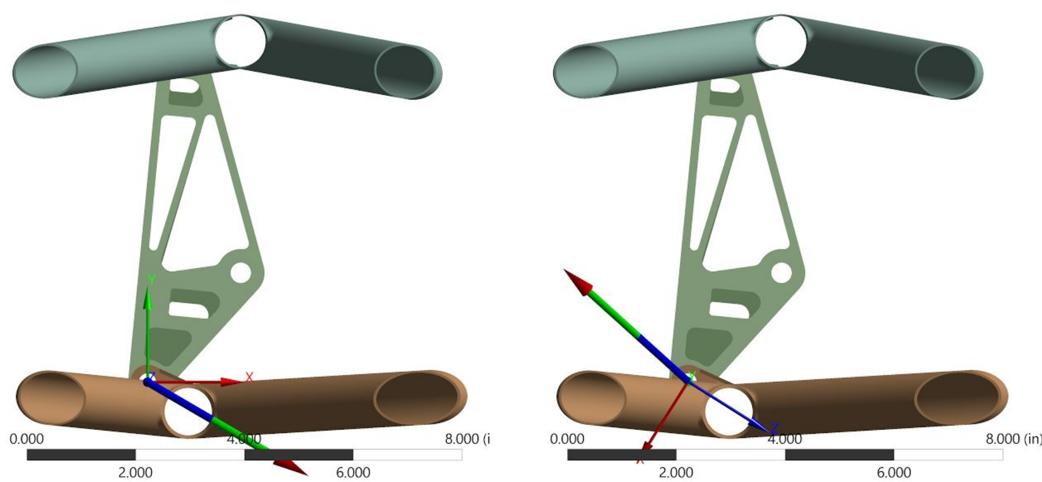


Figure 6.15: Joint forces, bottom left and bottom right respectively. Notice the lines of action intersect with the neutral axis of bottom cross tube, placing the tube in nearly pure bending/shear, not torsion. Left joint force: 1166 lbf. Right joint force: 1758 lbf. Upper joint forces are around 500 lbf.

We could, potentially, drop the bolt size from 5/16 to ¼ on the bottom and from ¼ to #10 on the top since the joint forces are small enough. We will wait on braking force analysis, though.

The variation in engine loading gives wavering confidence in this design. With loads applied only to the left side, the design will surely hold up (once a web is added to the upper tabs). Without this assumption, the design is mostly safe. I believe that the aluminum hangers should be analyzed further to bring down the weight, but that this design is the way to go.

10. Engine/Differential Carrier Bracket Design

The left-side engine carrier

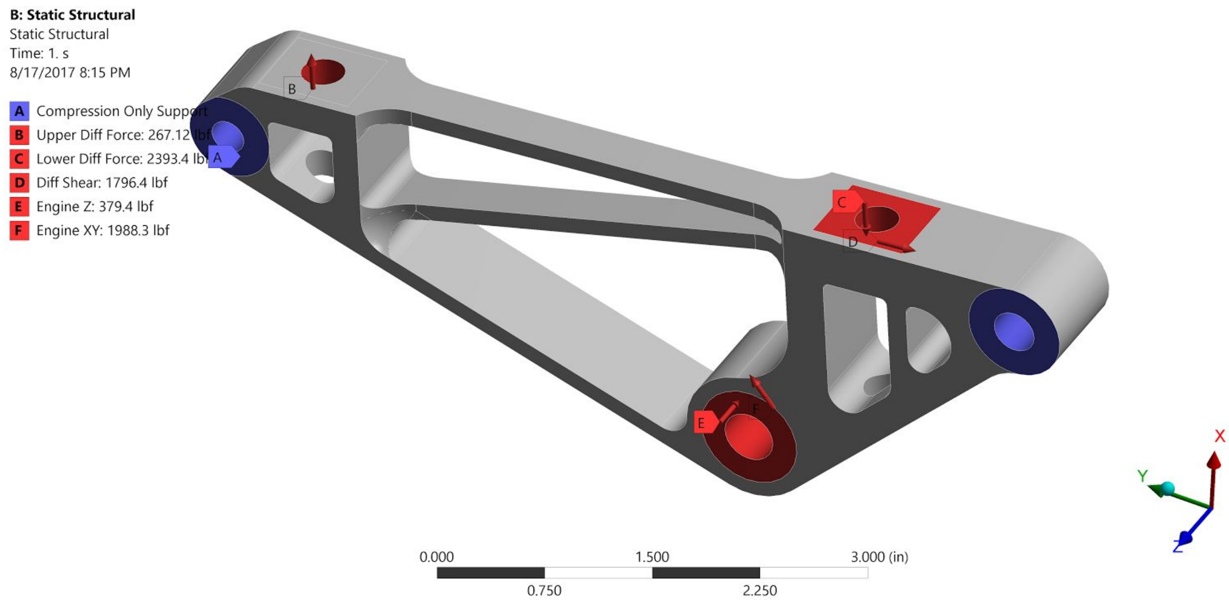


Figure 7.1: Rigid set-up

This set-up assumes the chassis is infinitely rigid but gives a good representation of the stresses around the mounting bolts.

Figure 7.2: Loose set-up

An alternative set-up assumes the chassis has no rigidity in the statically indeterminate direction but gives a poor representation of the stresses around the mounting bolts.

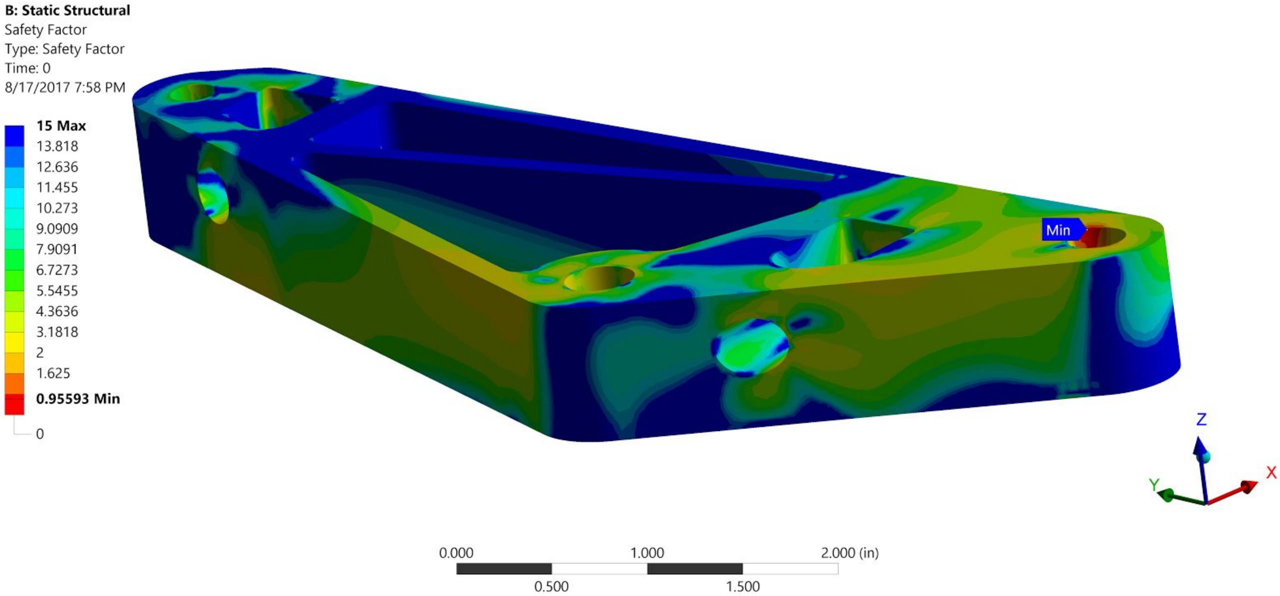


Figure 7.3: Initial carrier fatigue FOS. Weight: ~0.4 lbs

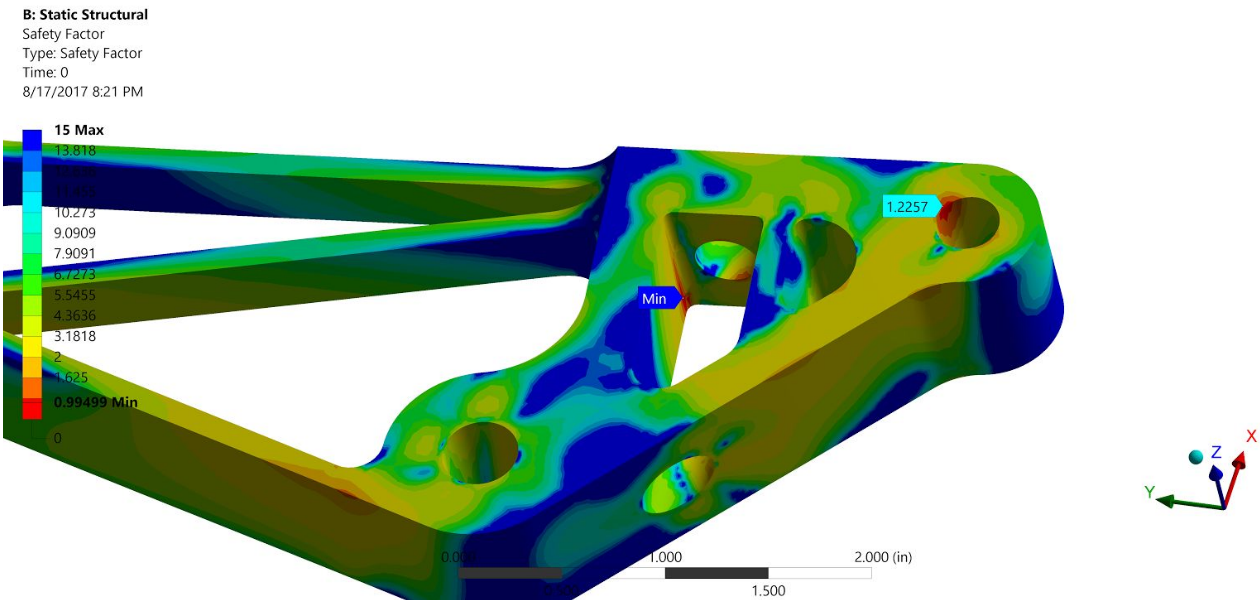


Figure 7.4: Revision 1-C fatigue FOS. Weight: ~0.35 lbs

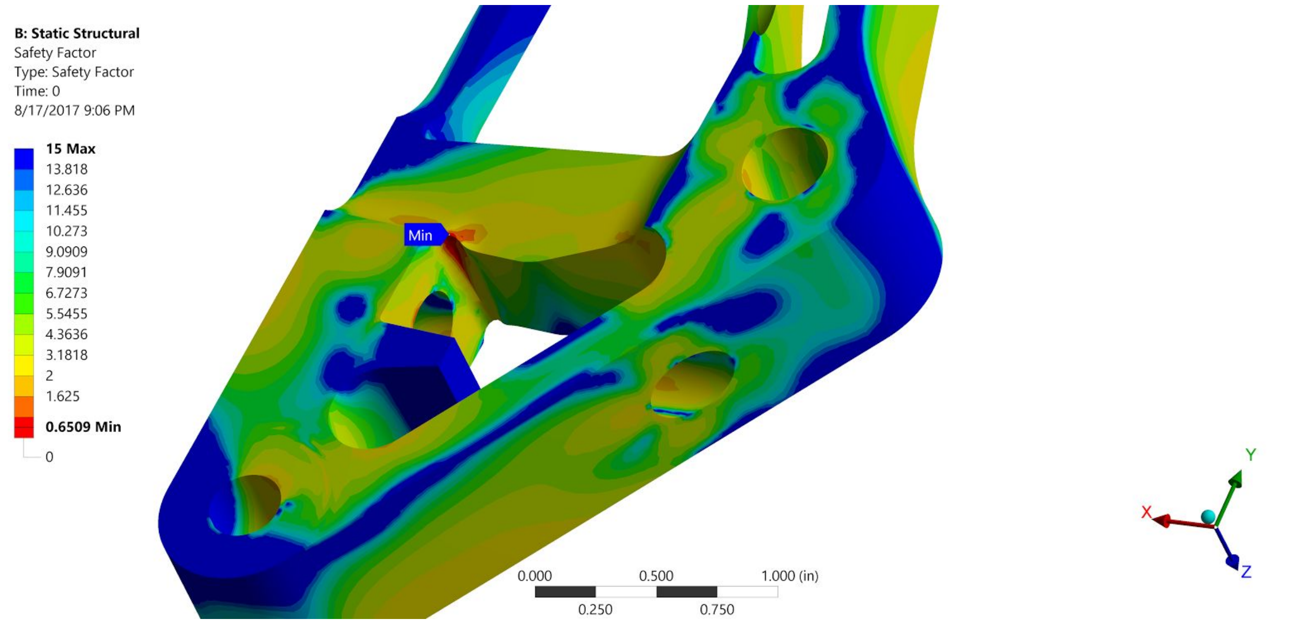


Figure 7.5: Revision 1-E fatigue FOS. Note extra pocket added between lower hole and engine lug

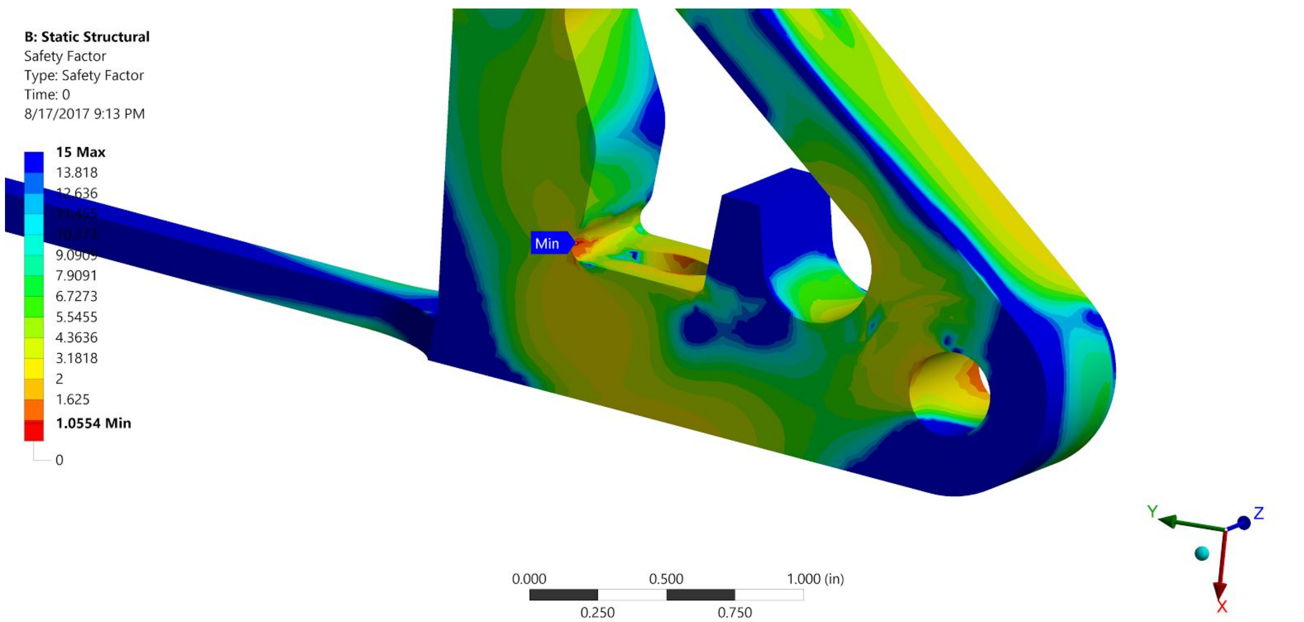


Figure 7.6: Revision 1-F fatigue FOS. Pesky pesky concentration

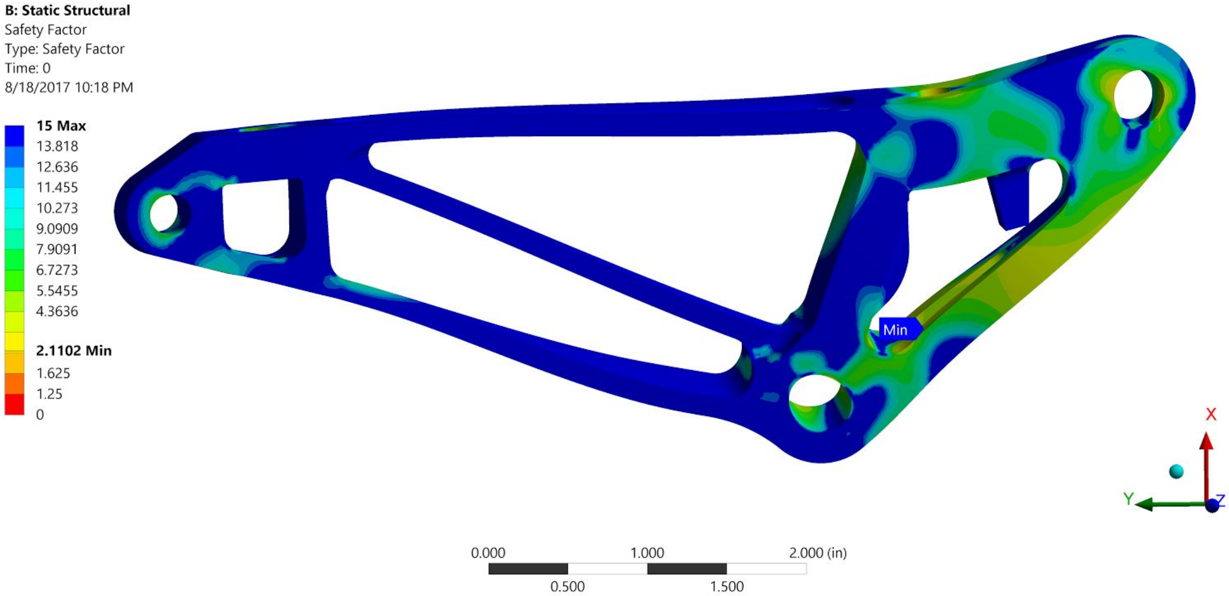


Figure 7.7: Revision 1-O fatigue FOS, stiff model. Weight: 0.296 lbs

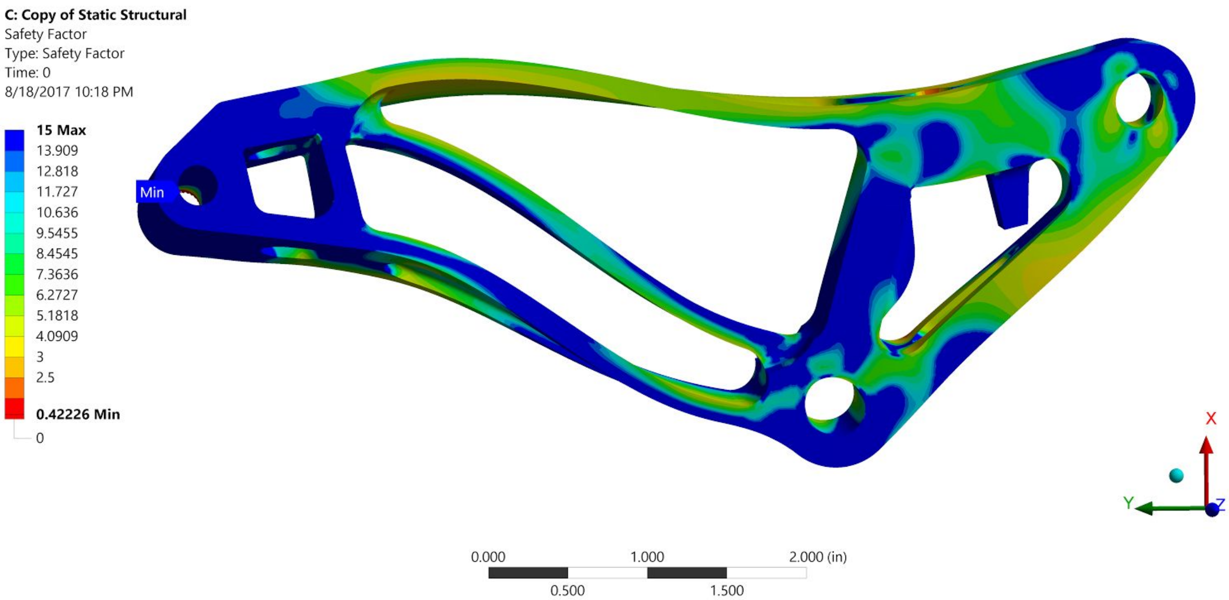


Figure 7.8: Revision 1-O fatigue FOS, unstiff model. Weight: 0.296 lbs. All areas but a singularity around a mounting lug have a FOS of greater than 2.

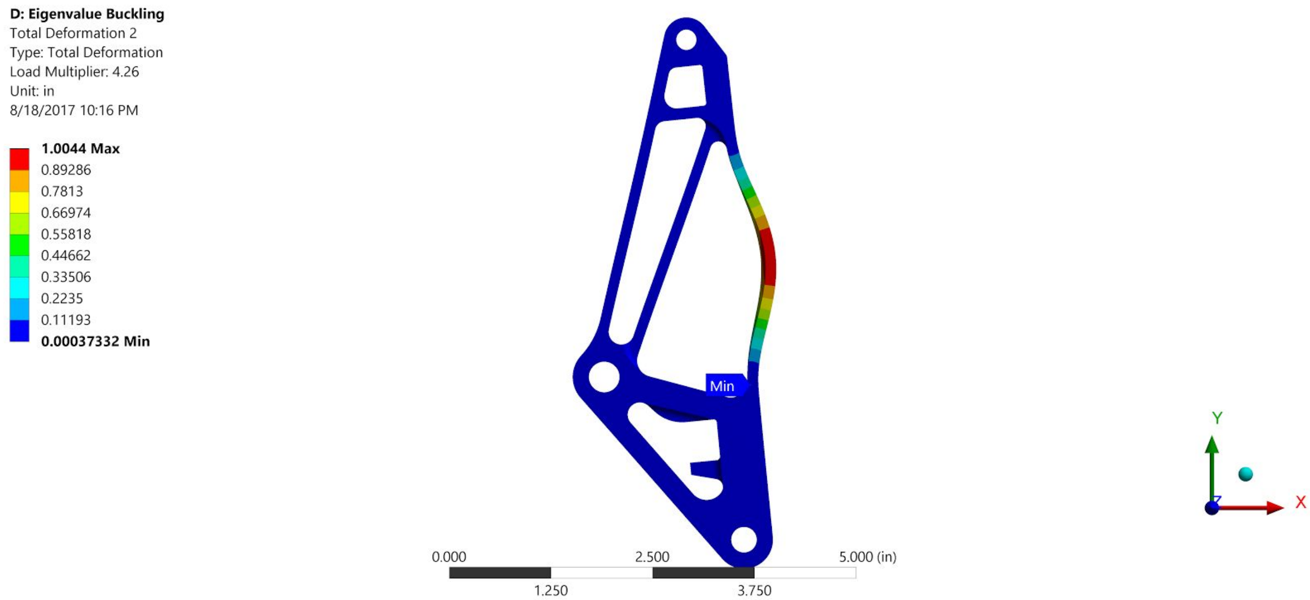


Figure 7.9: Revision 1-O buckling mode, lowest FOS (of 4.26).

Recall that buckling analysis is underconservative. However, the differential bracket itself will provide some support against buckling, as will the chassis. Regardless, this result isn't appealing and can be mitigated.

Yup, now at a mass of 0.292 lbs overall. Could perhaps go further on some struts but already feels pretty close as-is. The same analysis is repeated for the right side. We also need to analyze for braking loads.

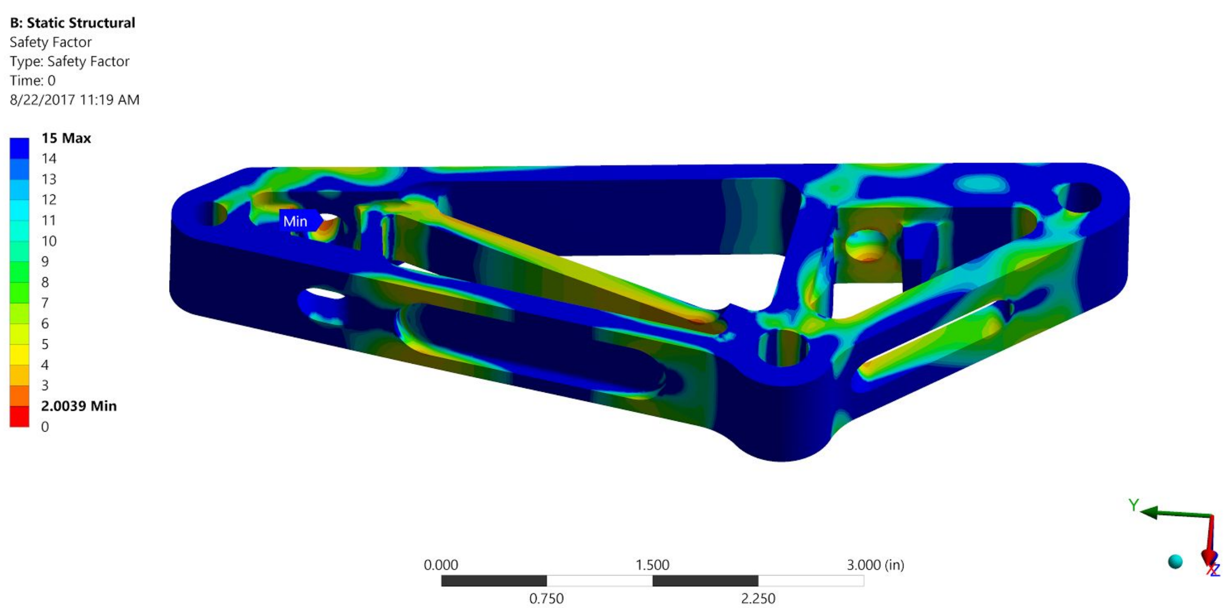


Figure 7.10: Right Side Fatigue FOS from Braking

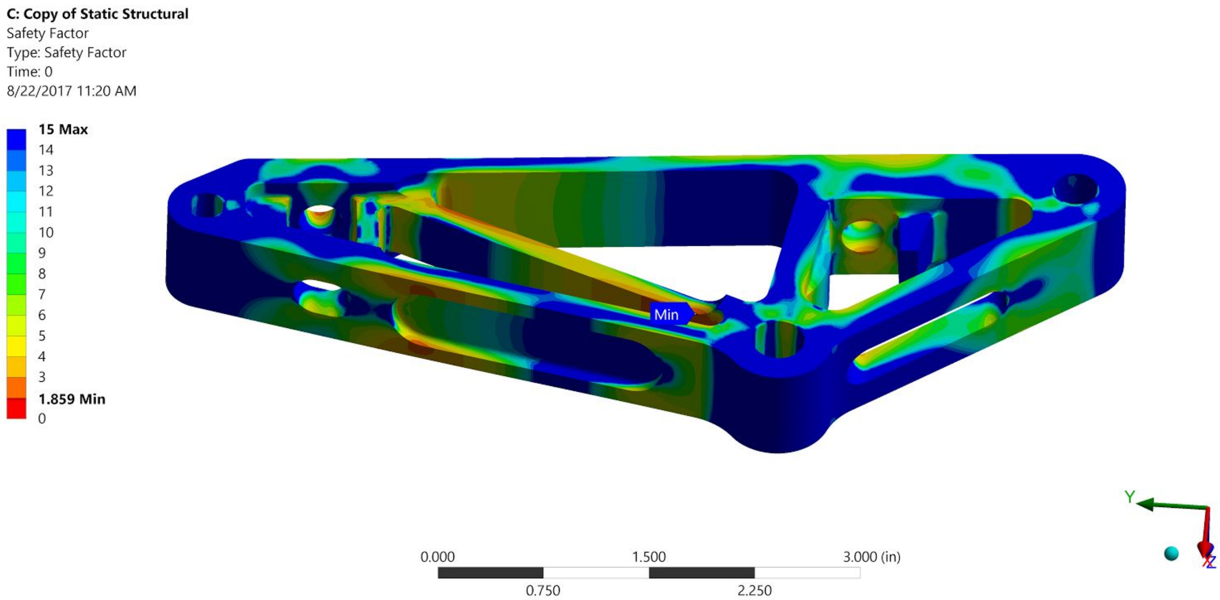


Figure 7.11: Right Side Fatigue FOS from Braking

However, let's feed this back into the chassis analysis which best models the stiffness of the tabs.

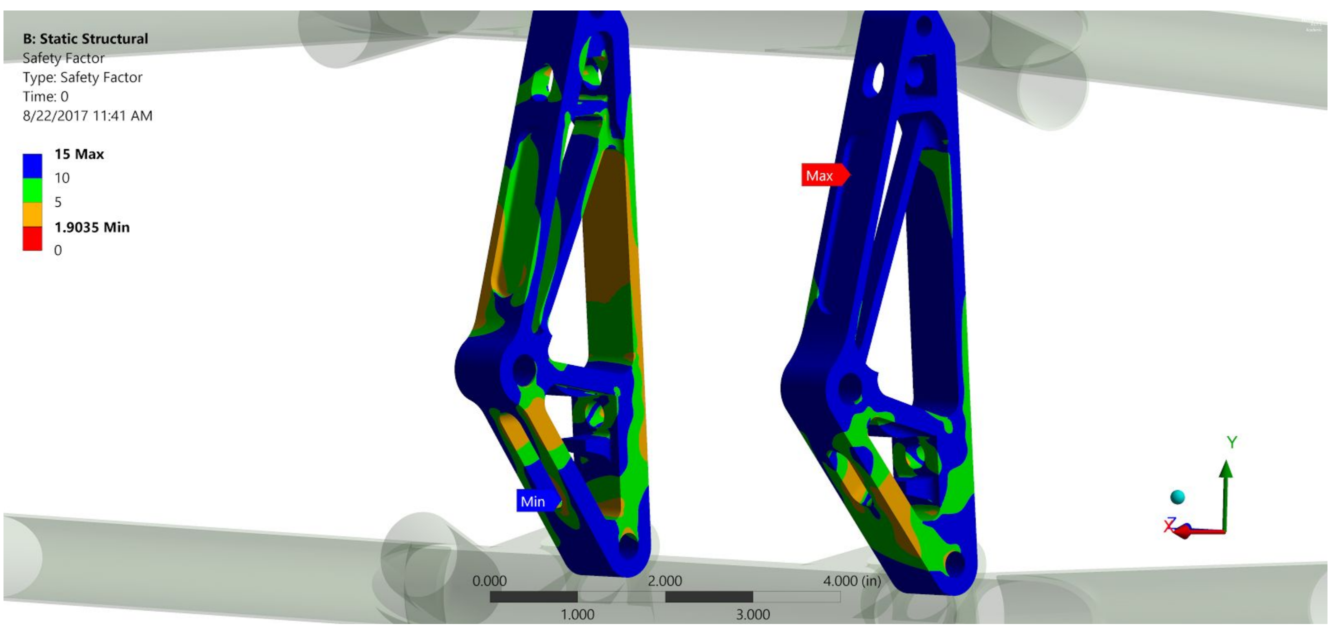


Figure 7.12

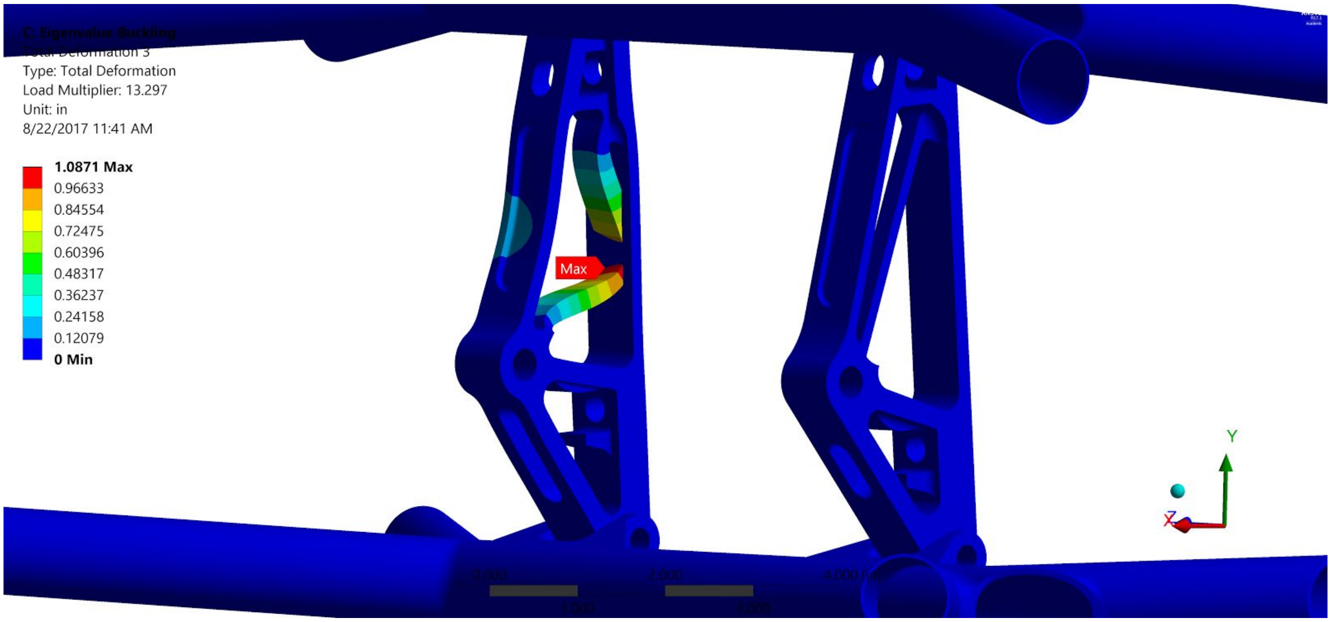


Figure 7.13

The final iterations are completed. The left side has holes added for the jacking bar added- and these do not affect the strength of the part. Analyzing braking is not as simple- the jacking bar supports some load.

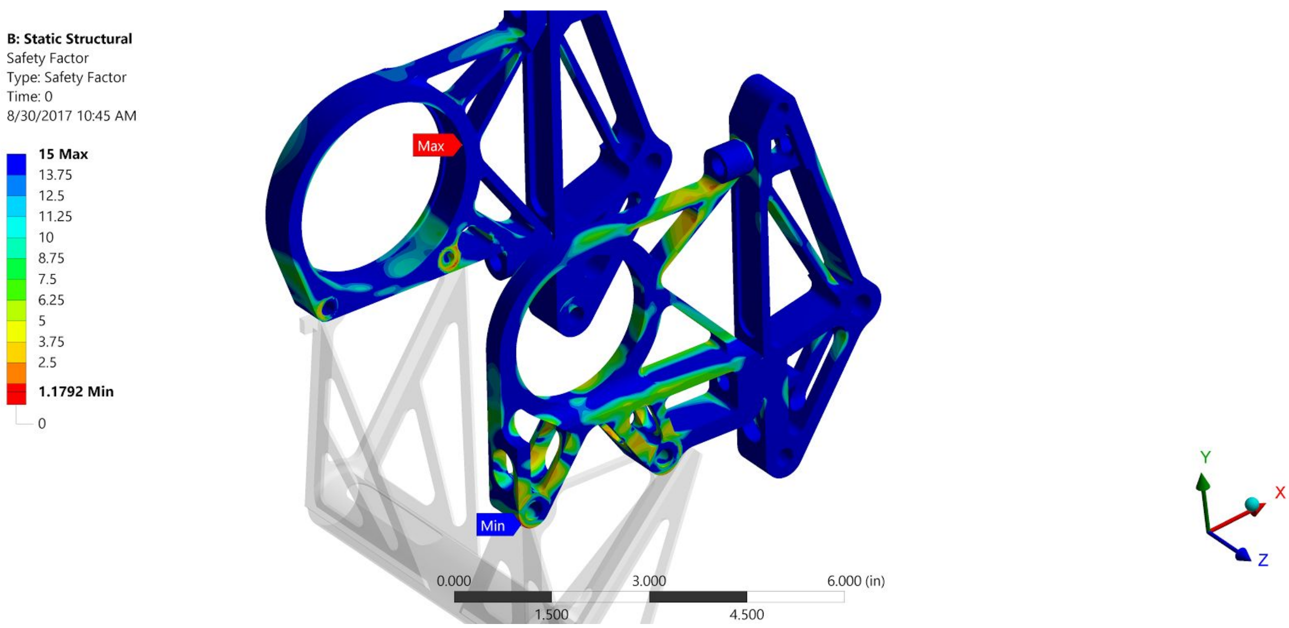


Figure 7.14: Fatigue FOS for aluminum components under braking loads. All under 2 except for contact areas which have typical concentrations from bonded contacts.

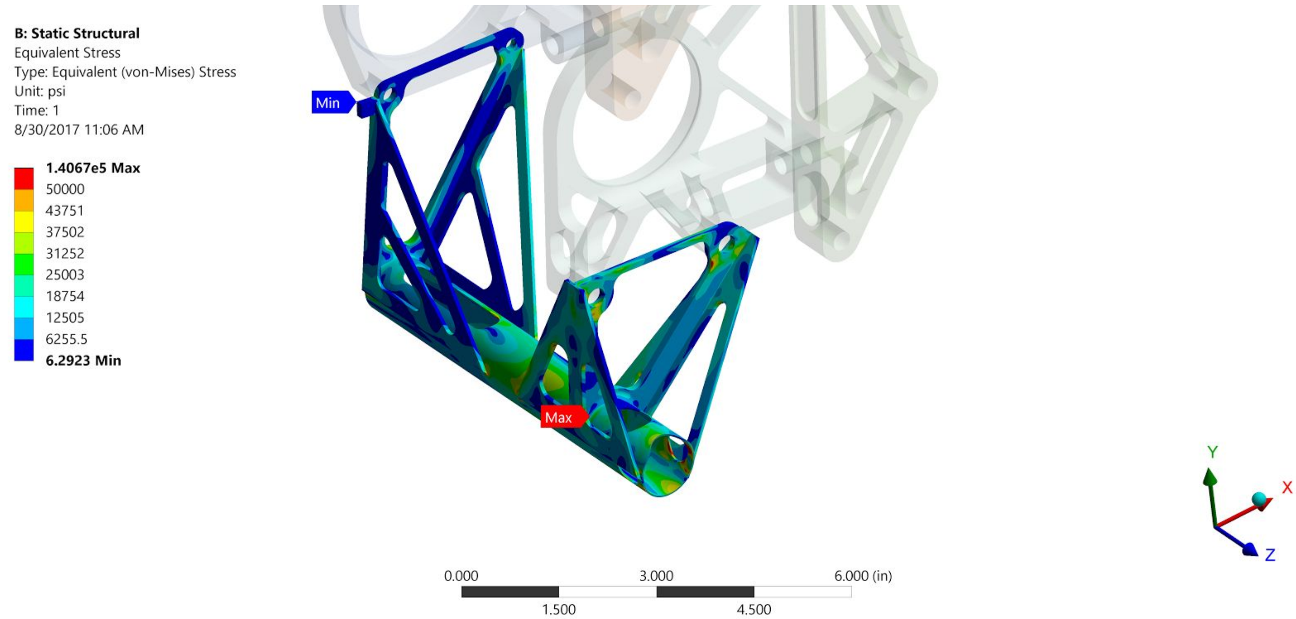


Figure 7.15: von-Mises stress for steel jacking truss under braking load. Most regions are safe, there are some singularities from sharp corners that will not be manufactured, but a few fillets do need increased in size.

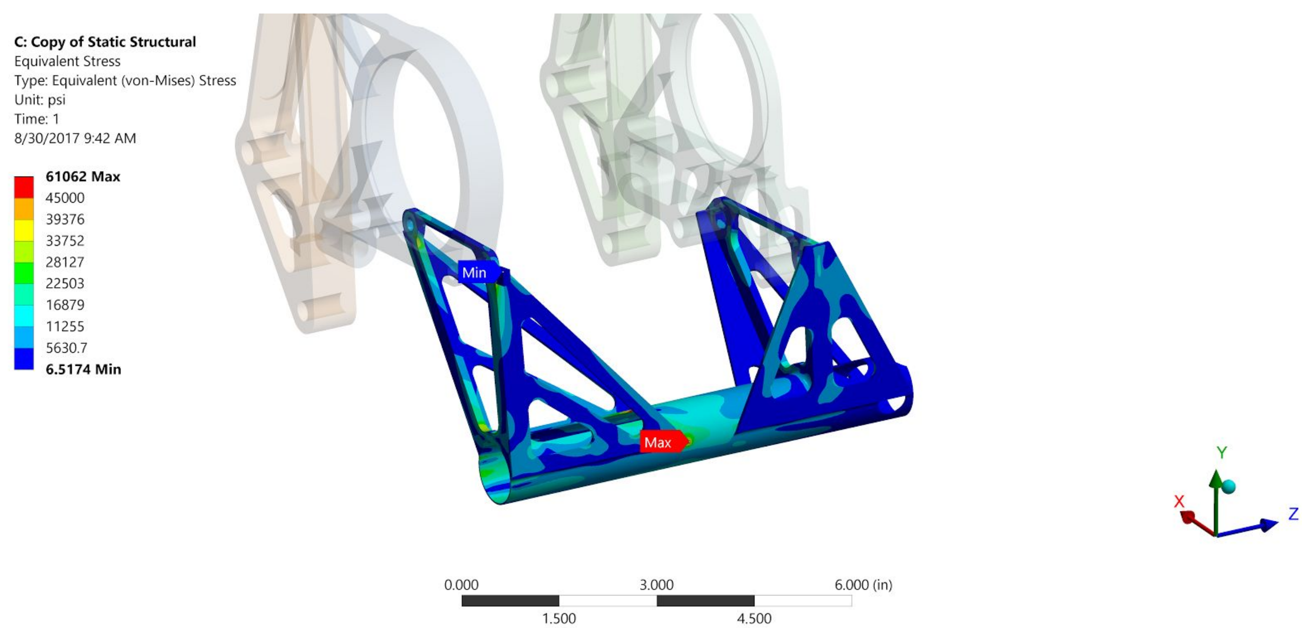


Figure 7.16: von-Mises stress for steel jacking truss under jacking forces. High stress comes from a sharp corner that will not be manufactured.

11. Sprocket Hub Design

The sprocket hub (which connects the Drexler differential input spline to the sprocket via 6 5/16" bolts) has been analyzed with ANSYS.

The first setup used applys a force and a moment to the input "spline" which has been removed because of the complexity required to model the spline. This can be added later- right now it just makes analysis time larger. A displacement support is added in the z-direction for the sake of convergence.

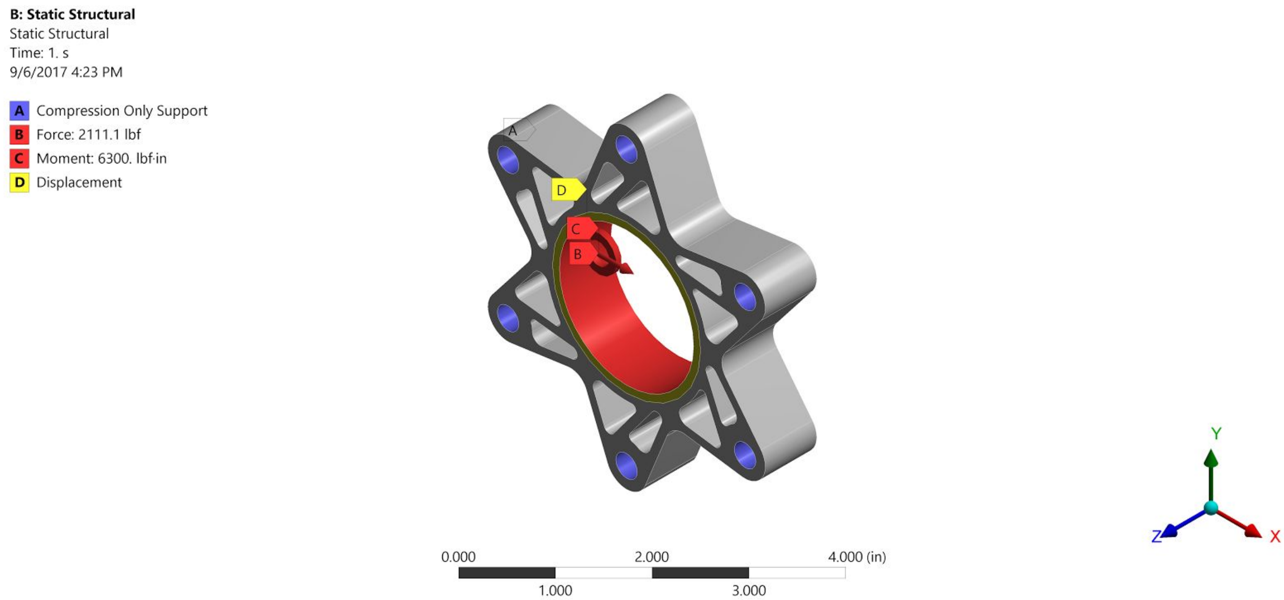


Figure 8-1: First analysis setup

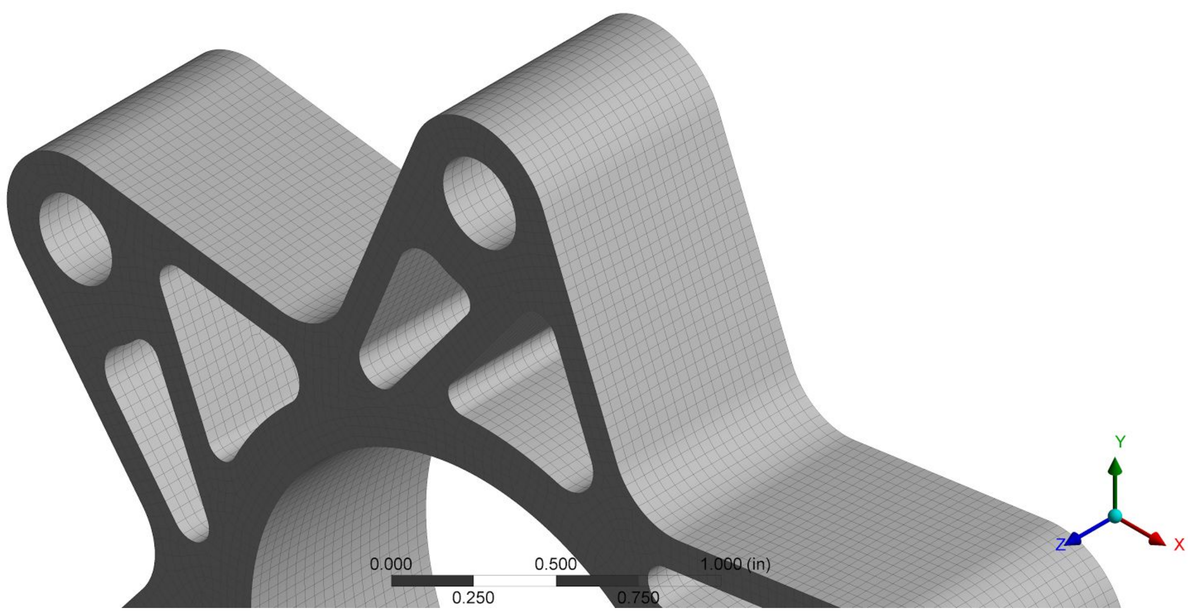


Figure 8-2: Mesh used in first analysis

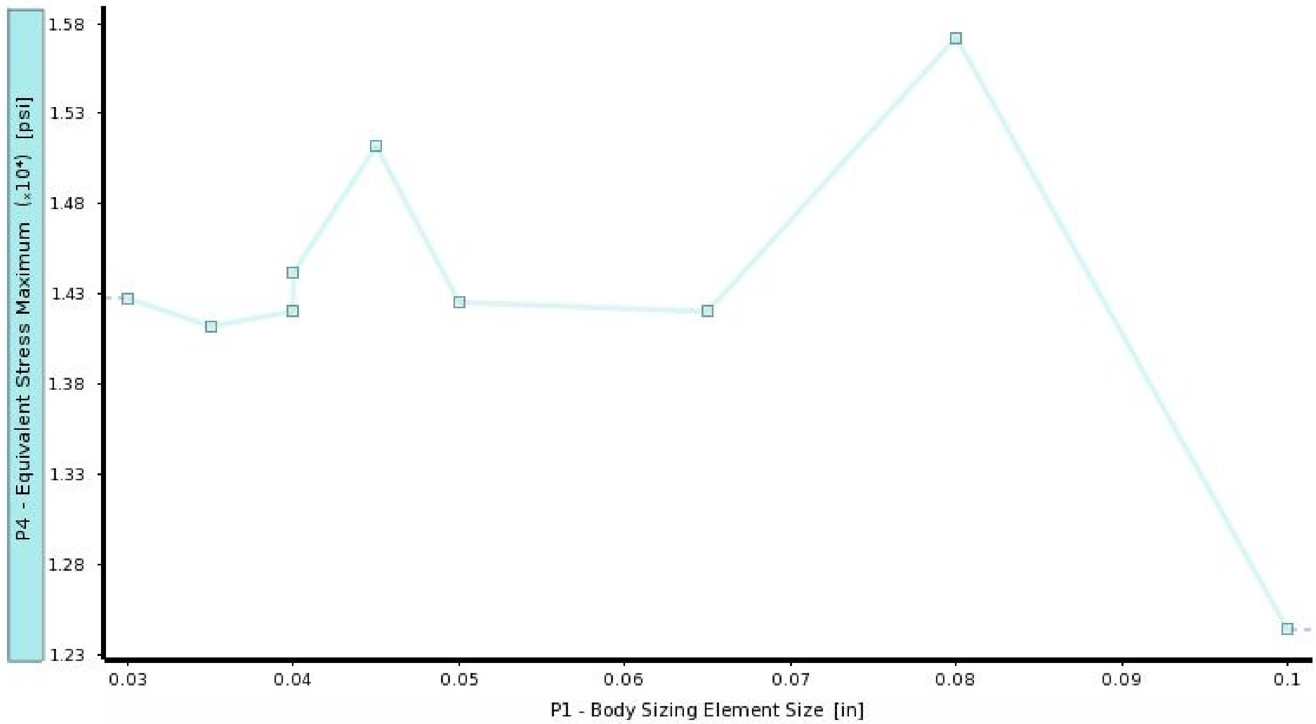


Figure 8-3: Mesh convergence for first setup

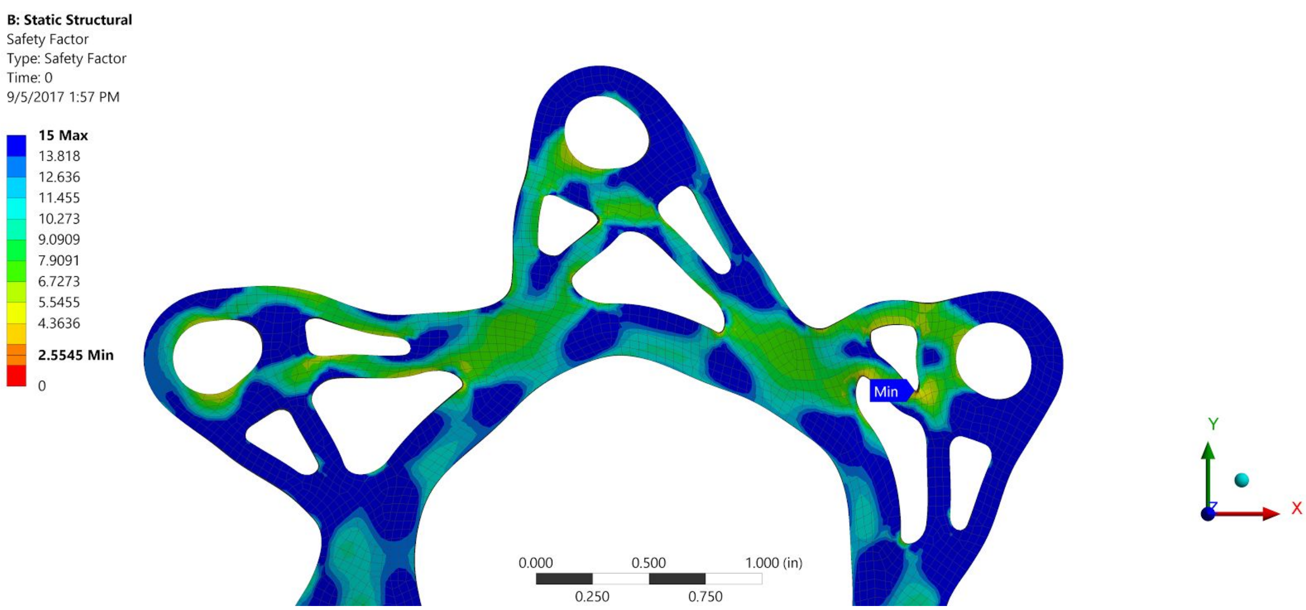


Figure 8-4: Revision 2-D results

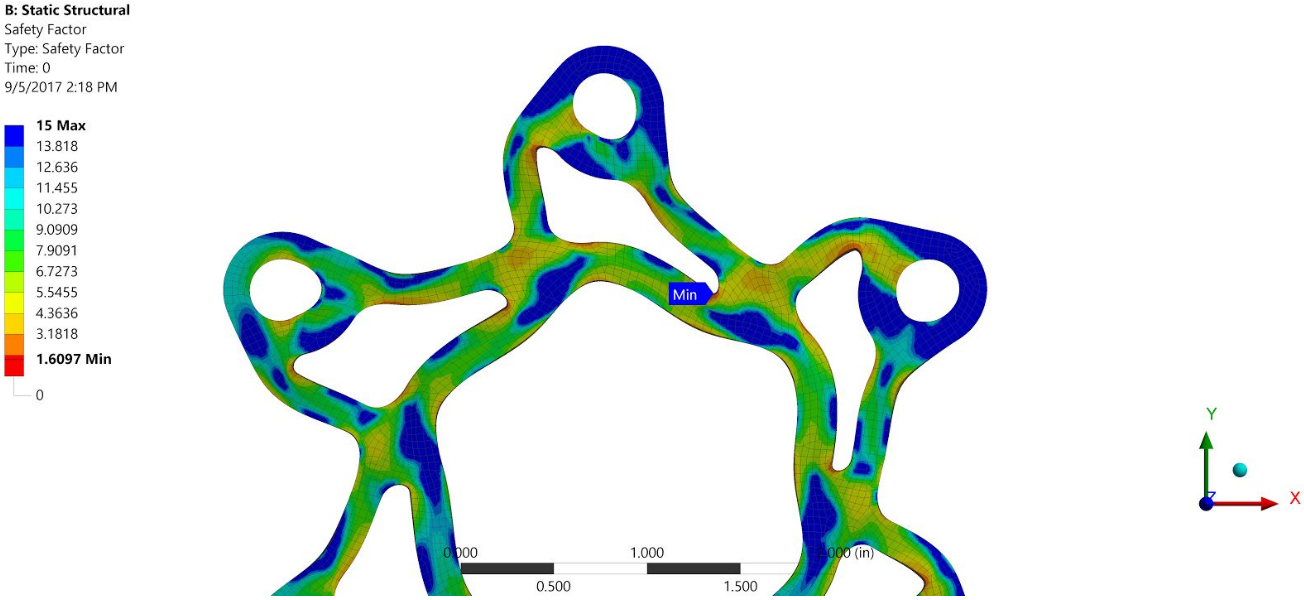


Figure 8-5: Revision 2-E results

An additional setup is introduced with the same mesh size and geometry, reversing which parts are loaded and which are constrained

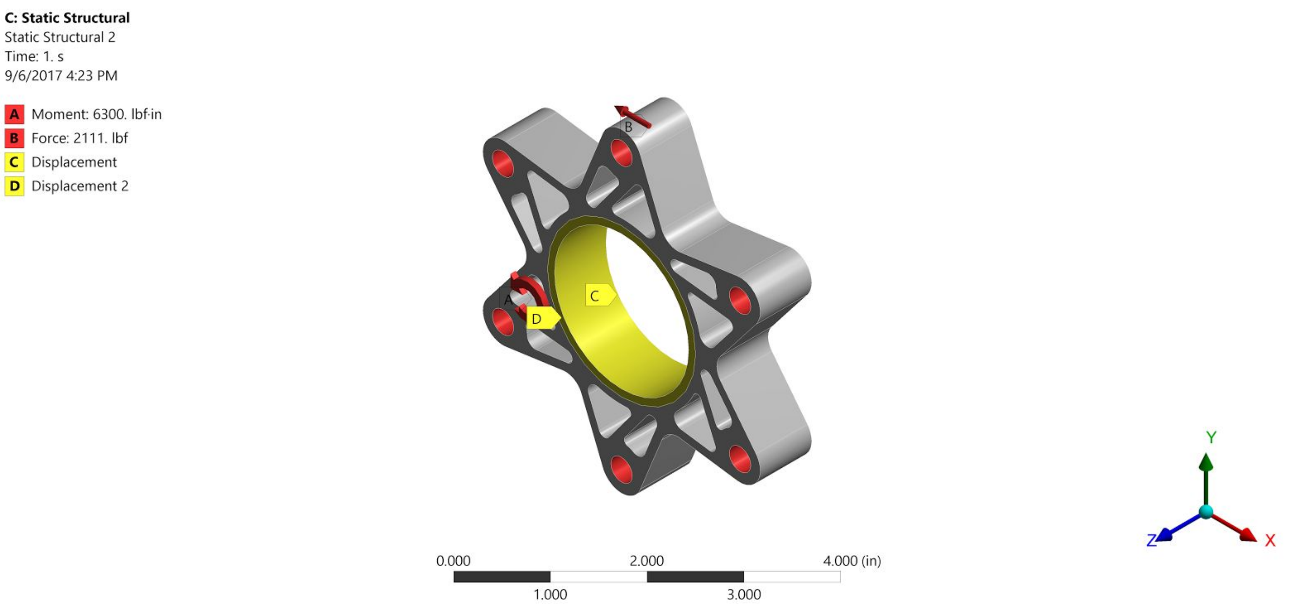


Figure 8-6: Second setup

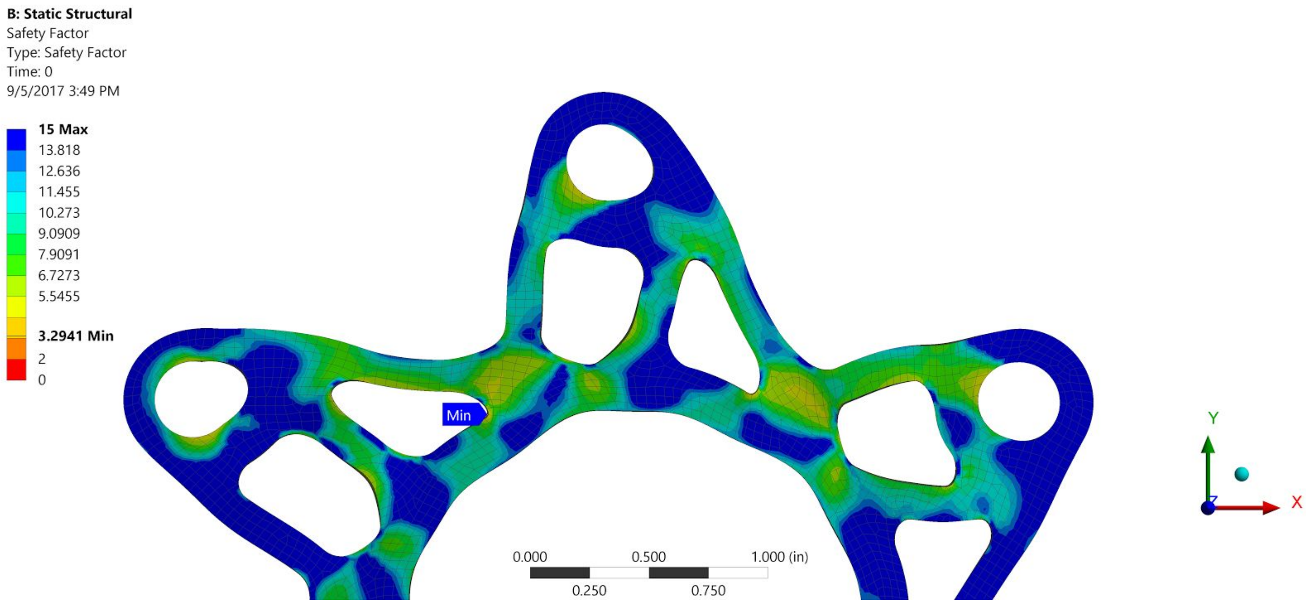


Figure 8-7: Revision 2-F results from setup 1

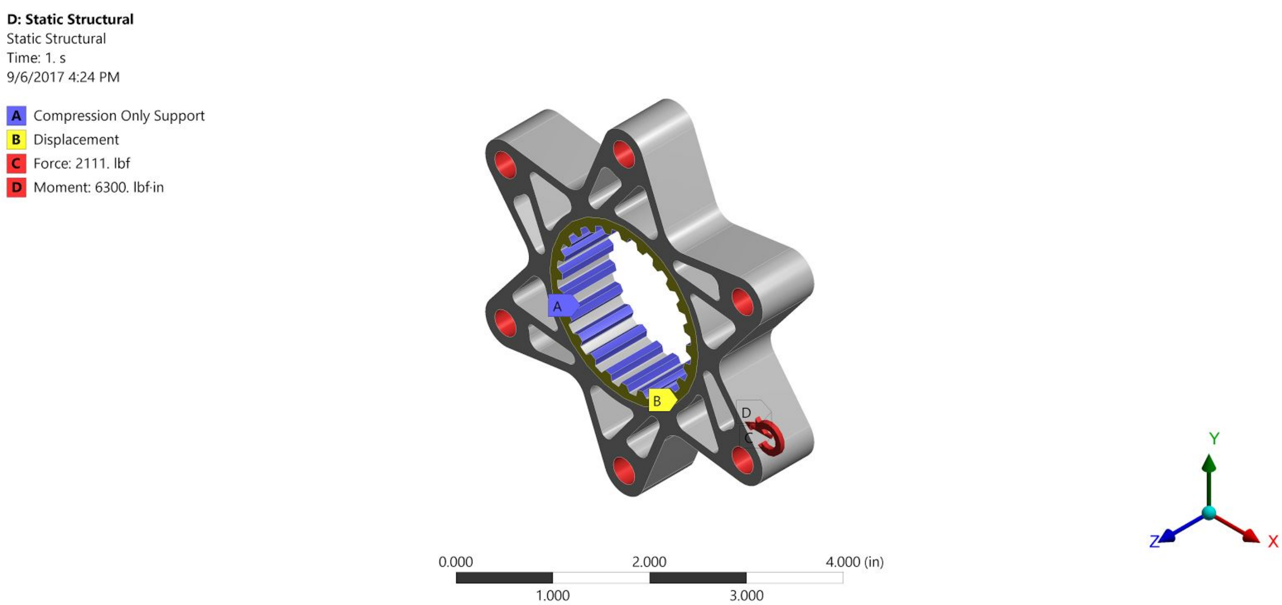


Figure 8-8: Third setup

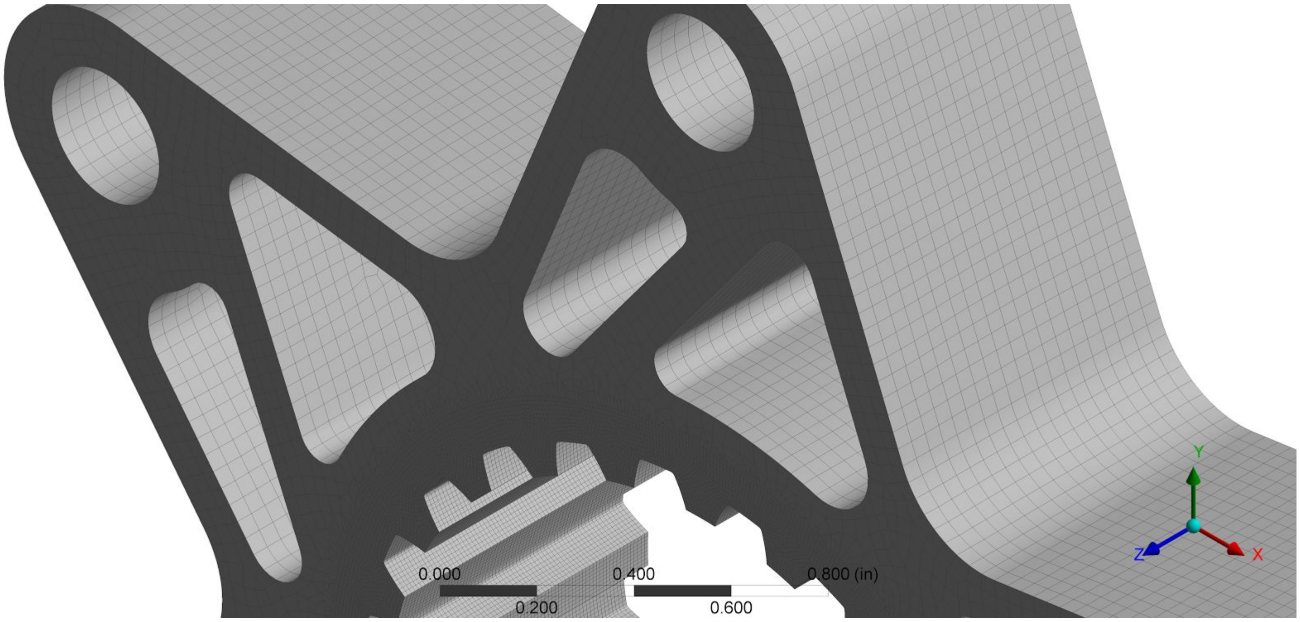


Figure 8-9: Third mesh used. Spline region is refined to 0.01”. Quads are used almost entirely, still.

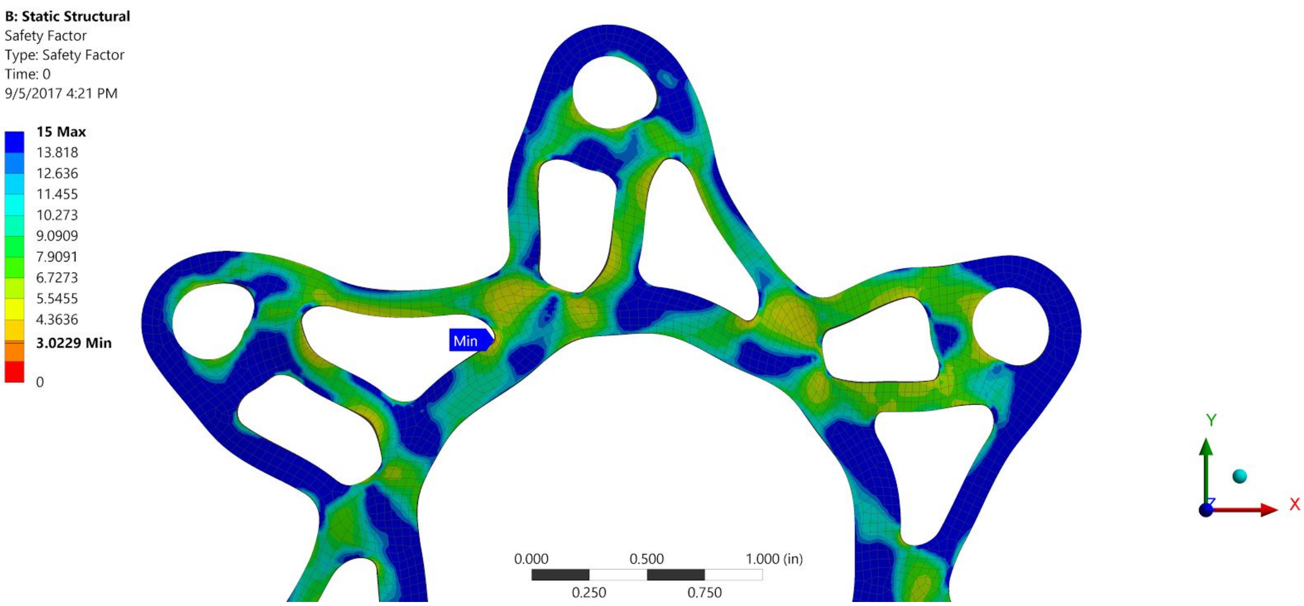


Figure 8-10: Revision 2-G under first setup. Fatigue FOS = 3.0229

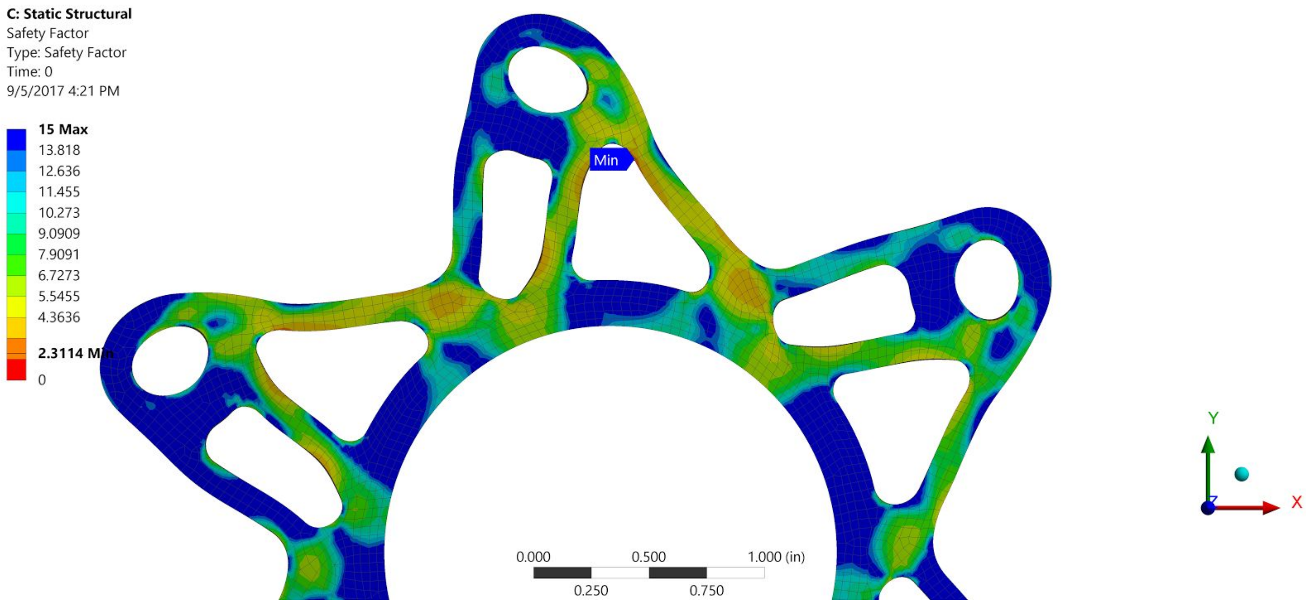


Figure 8-11: Revision 2-G under second setup. Fatigue FOS = 2.3114

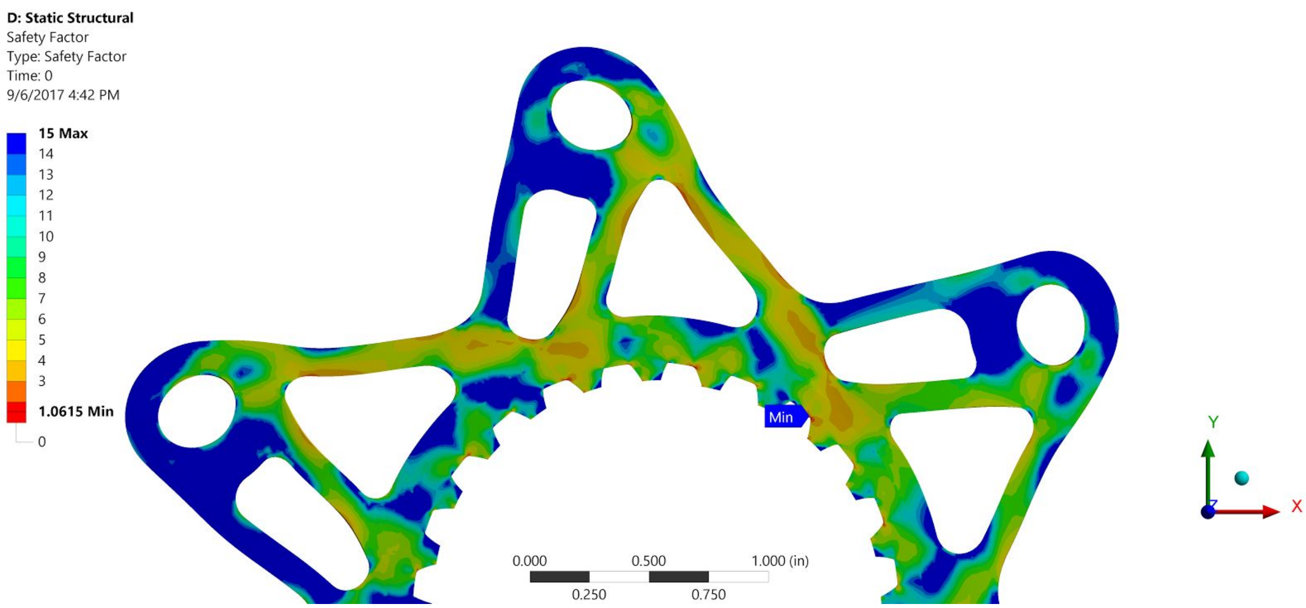


Figure 8-12: Revision 2-G under third setup. Fatigue FOS = 1.0615

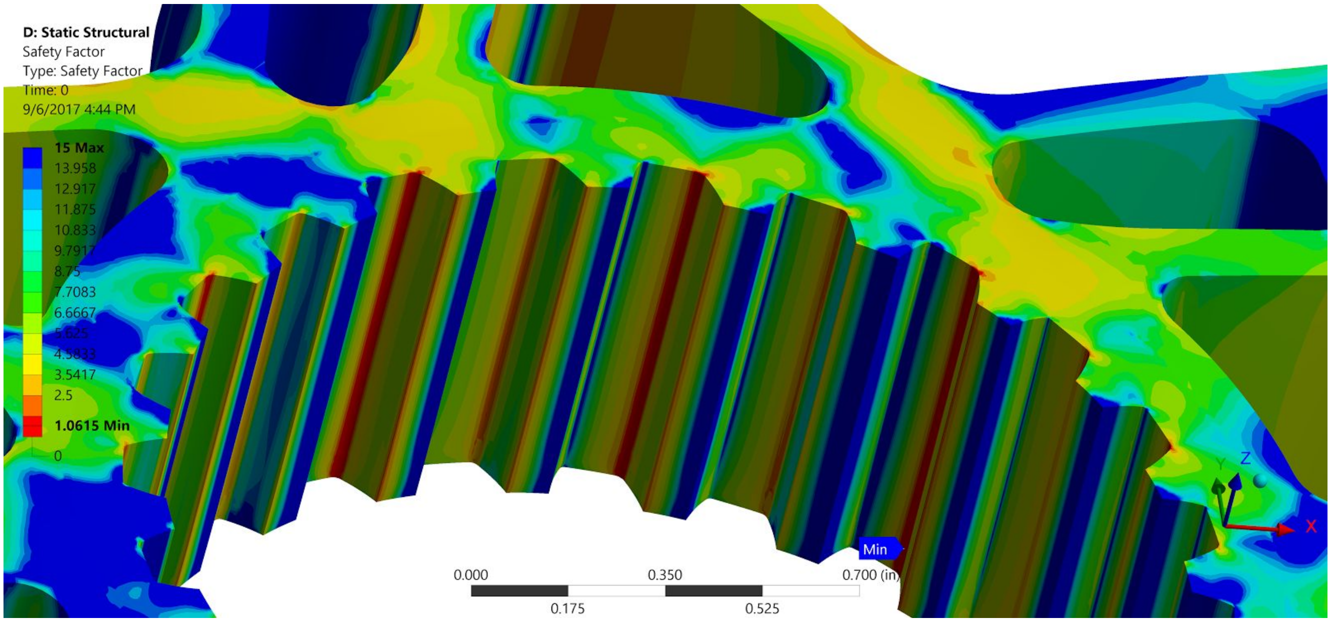


Figure 8-13: Revision 2-G under third setup, closeup around spline teeth. Fatigue FOS = 1.0615

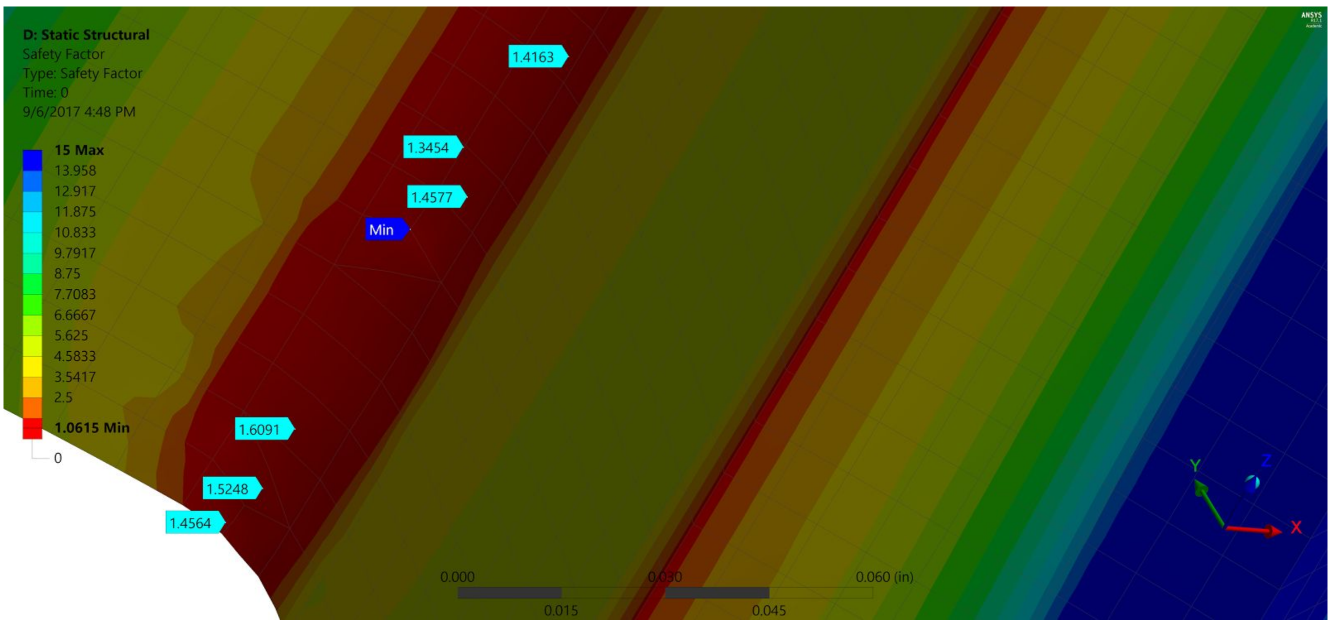


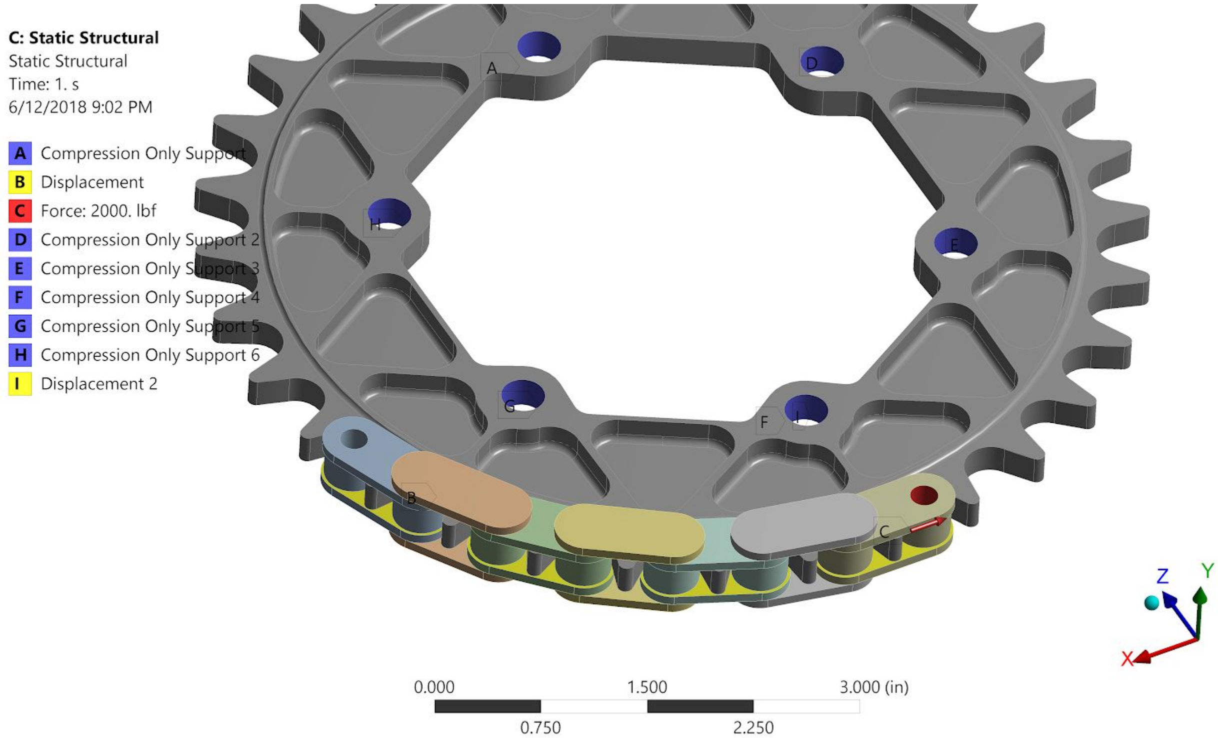
Figure 8-14: Revision 2-G under third setup, closeup at the minimum stress. Fatigue FOS = 1.0615. Note that this low FOS is at a mesh discontinuity.

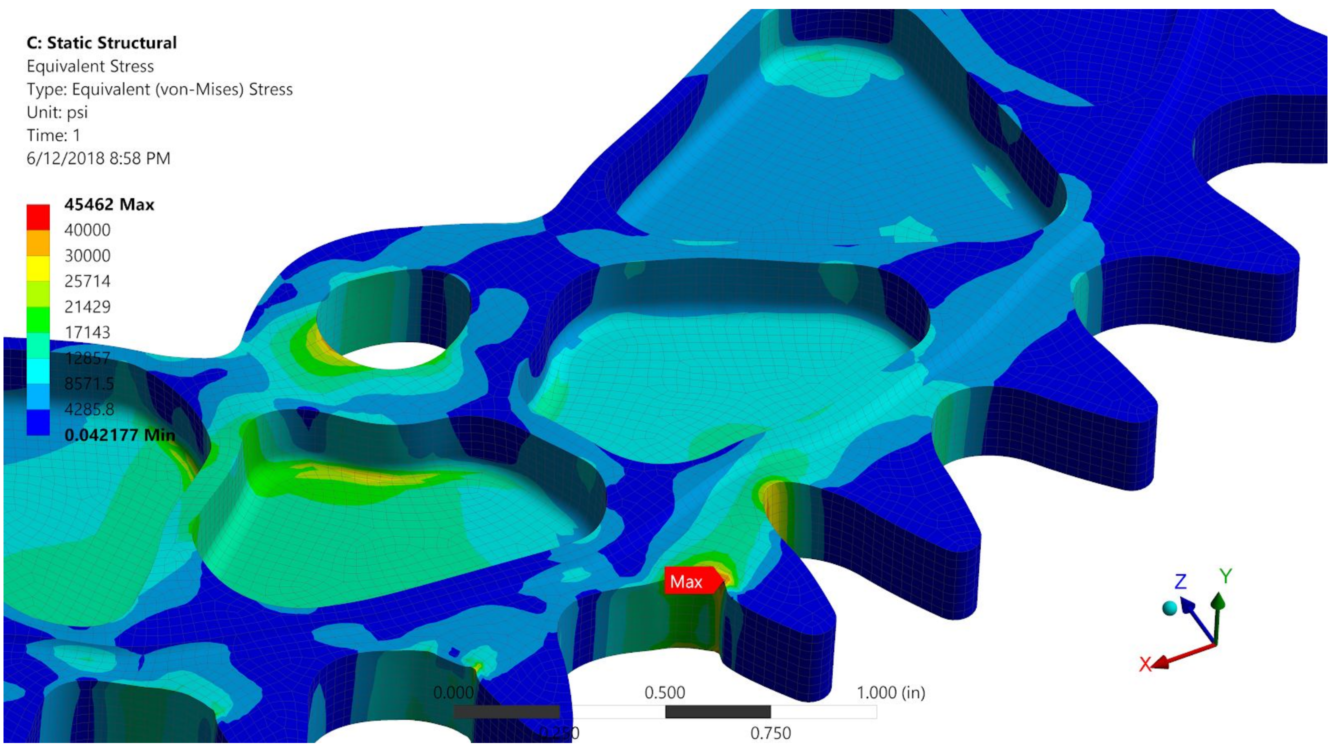
This fatigue FOS is acceptable around the spline teeth because:

1. This small FOS is at a mesh discontinuity, and is larger (>1.4) everywhere else.
2. This interface is designed for by Drexler
3. This interface has been used in the past
4. There are multiple teeth to bear the load (redundancy)

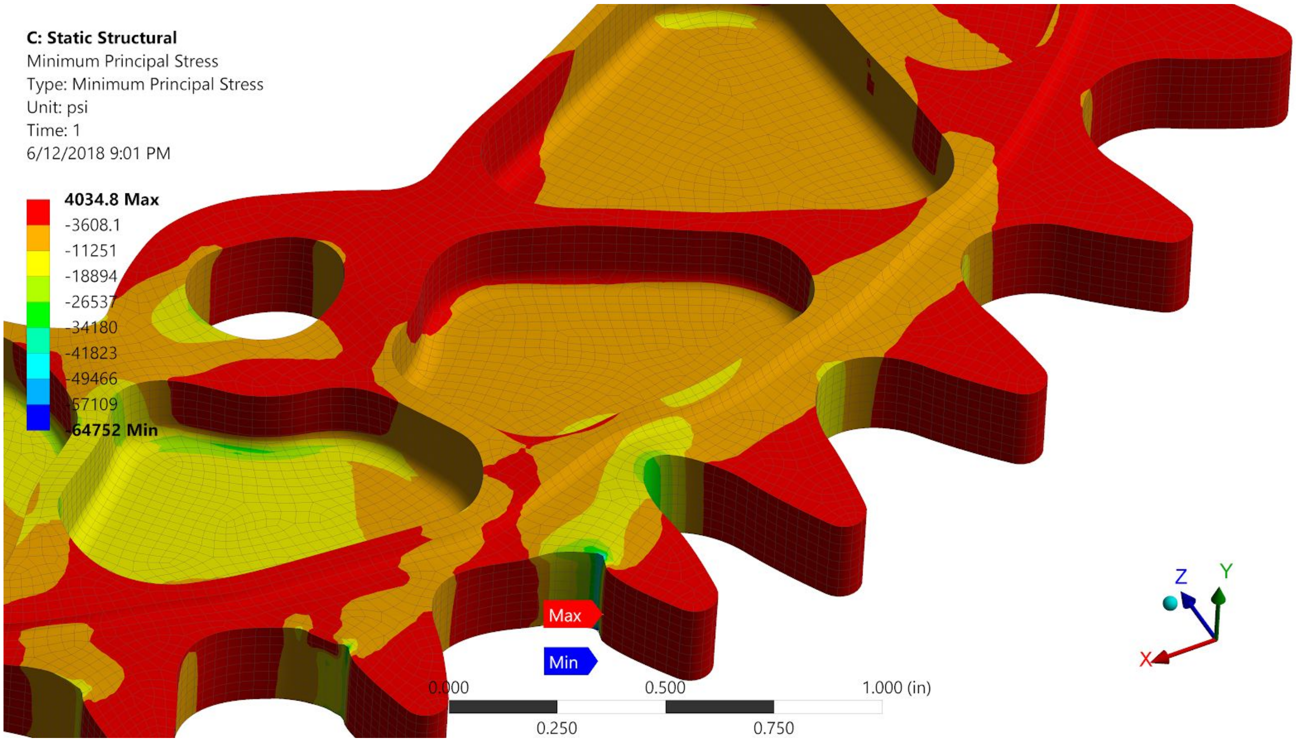
Thus, revision 2-G of this component is **good to manufacture.**

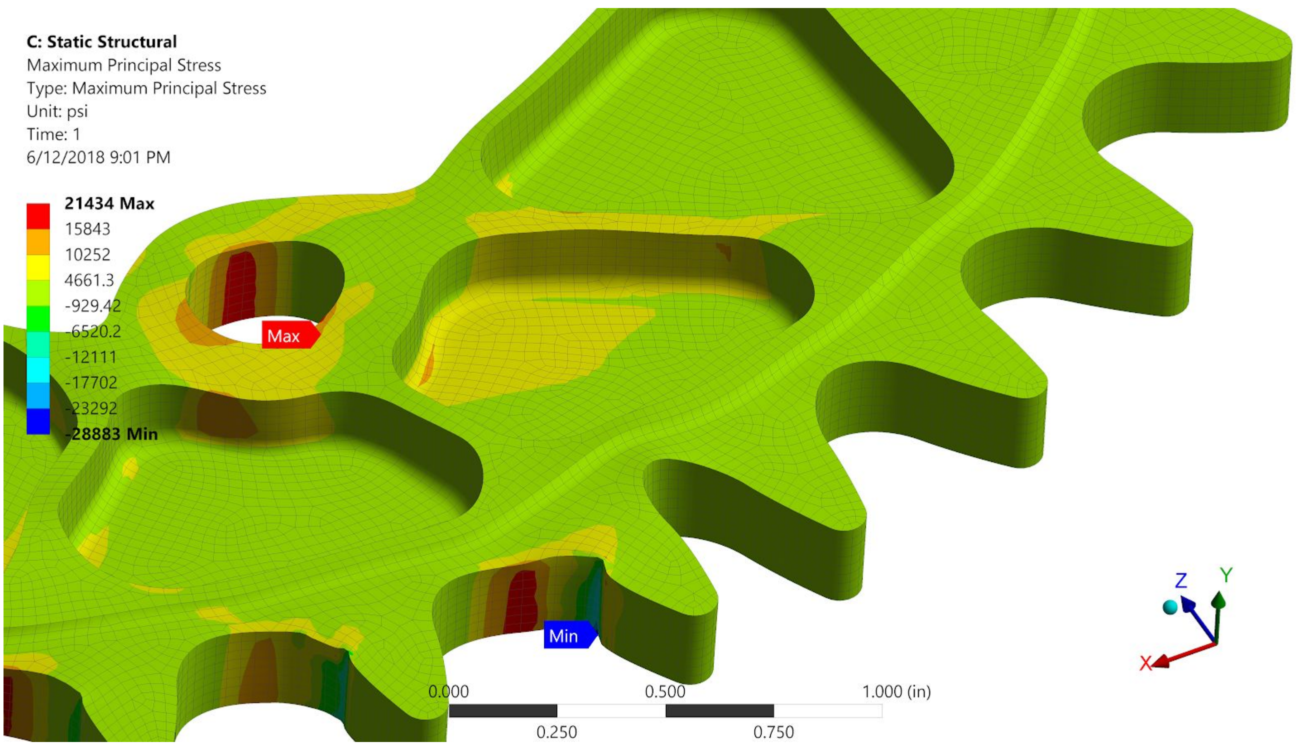
12. Sprocket Design





There are some regions which fall above the maximum stress criteria of 30 ksi, but this is only a problem if they are not in compression.





Indeed, the maximum principal stress has lower magnitude than the minimum (which is negative). This means the region is in compression, and not (as) subject to fatigue.