Design of Unsprung Mass for RGP007

Rose-Hulman Institute of Technology Grand Prix Engineering

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1. Objectives

By: Thaddeus Hughes
Date: 6/20/2017

Unsprung Mass refers to everything connecting the suspension to the tires, namely the upright, hub, wheel center, and rim. To increase the responsiveness of the suspension, this system should be made very lightweight. To make sure the suspension performs properly and predictably, this system should be made very stiff. These two factors generally oppose each other, which additional stiffness necessitating additional mass. The system must, of course, also hold up under all loads seen- generally, these components are fatigue components.

2. Load Cases

By: Thaddeus Hughes
Date: 6/20/2017

The wheel sees four different loading cases:

- Static/Cruising (only tire normal force)
- Left Turn (normal and lateral tire force)
- Right Turn (normal and lateral tire force)
- Acceleration (normal and longitudinal tire force)
- Braking (normal and longitudinal tire force)

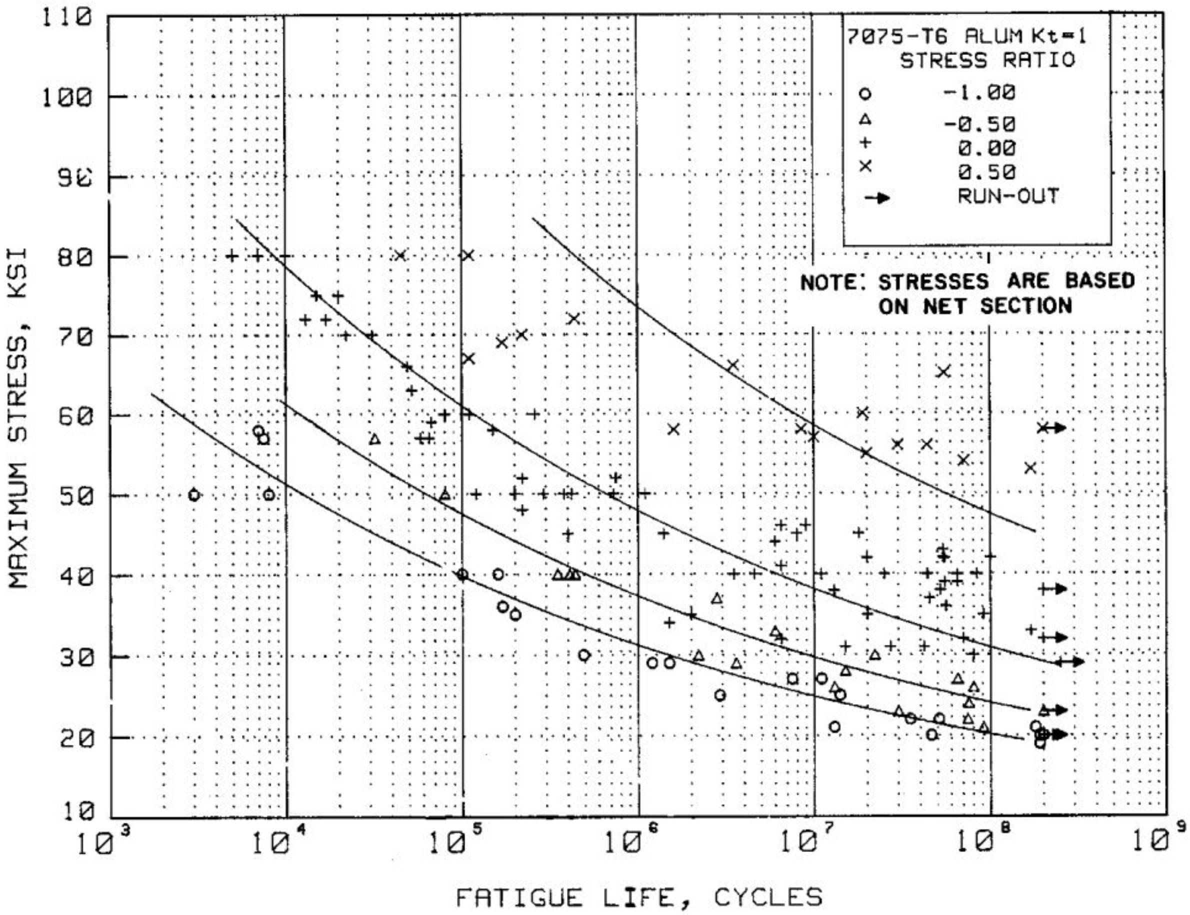
These can be superimposed to some degree, but the majority of events come from purely these five scenarios. For the sake of simplicity, we will only consider the left side unsprung mass; the right side will be a mirror image in every fashion.

By: Thaddeus Hughes
Date: 7/25/2017

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. 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.

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.

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

	Strength (ksi)	
7075-T6 Aluminum	Zero-Based	Fully Reversed
Cycles	0	-1
100,000	61.16	39.79
200,000	56.77	36.94
250,000	55.44	36.07
500,000	51.54	33.54
1,000,000	47.98	31.22

3. Rear hand-calculation analysis

By: Thaddeus Hughes
Date: 7/2/2017

We will consider traction-limited loads, so will start by analyzing loads on the wheel as shown in Figure 3.1.

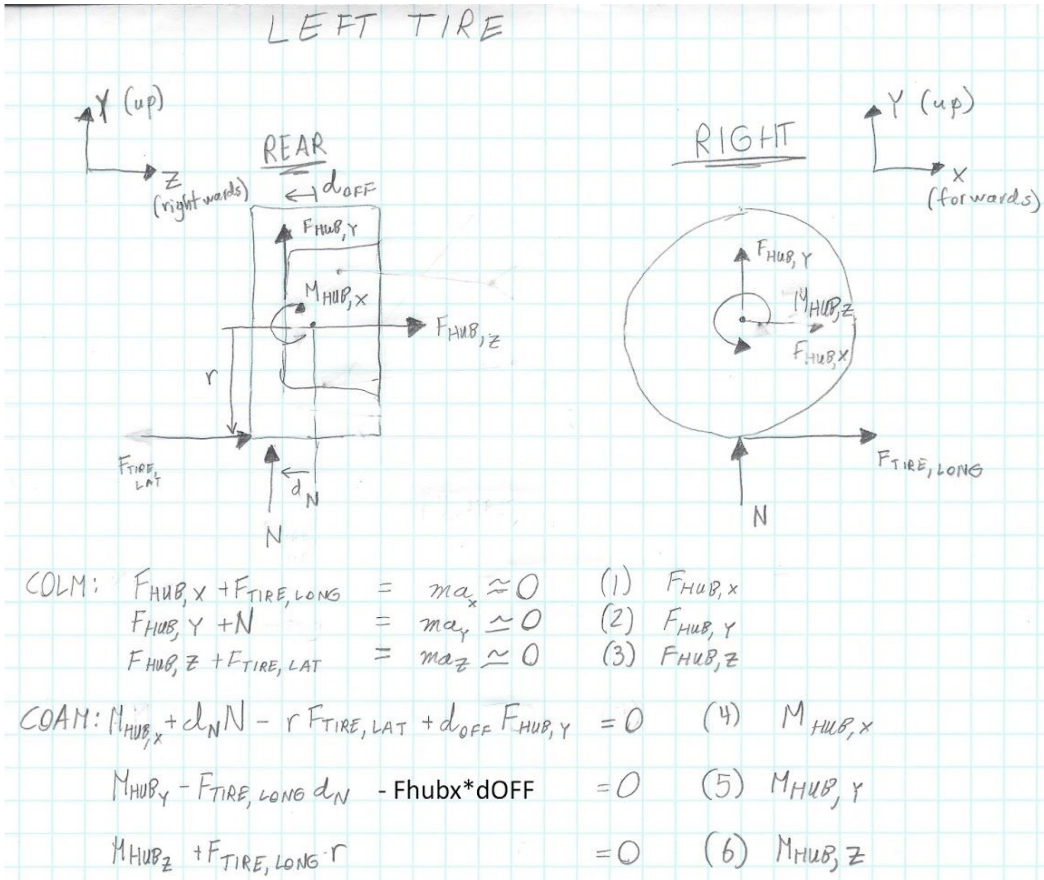


Figure 3.1: Analysis of tire under generalized loads

We consider that the joint between hub and wheel center is perfectly fixed. This is acceptable, as when we go to analyze the stress at this interface, we will model the wheel center along with the hub.

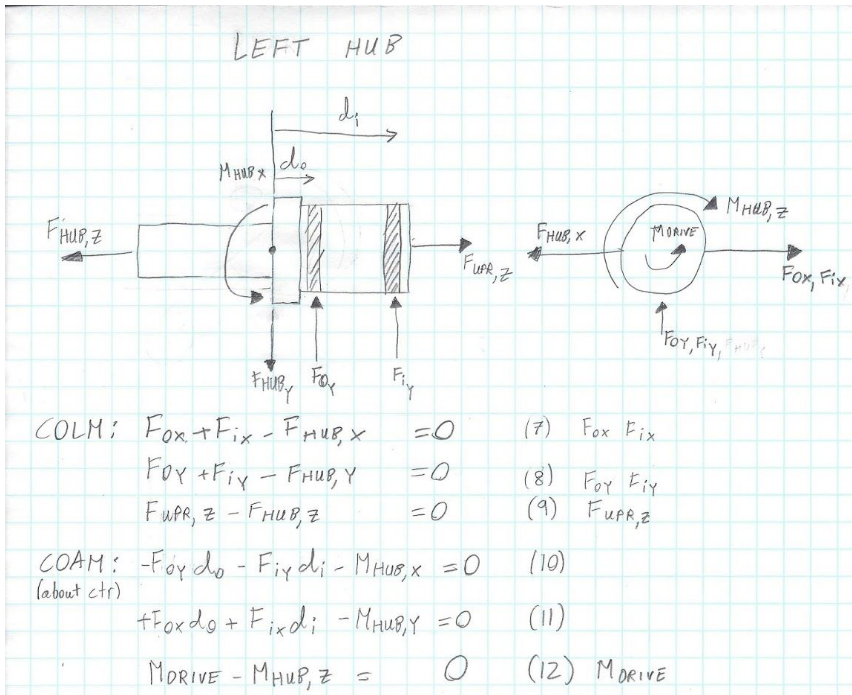


Figure 3.2: Analysis of hub under generalized loads

We consider the bearing supports as radial and thrust. The thrust component will be distributed between both bearing.

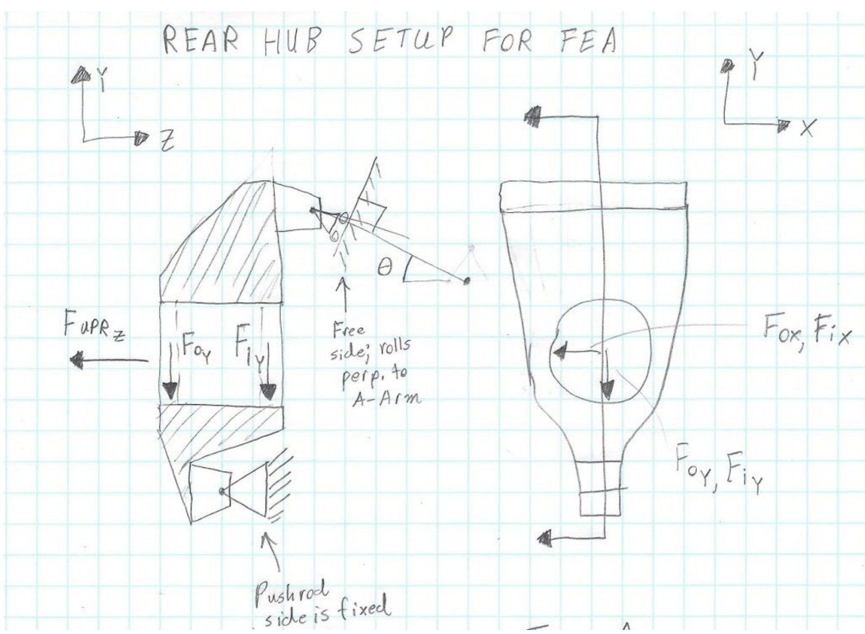


Figure 3.3: Constraints for the rear hub in FEA

For modeling the rear hub, we will consider the bottom mounting point for the A arm to be fixed in all directions (but not rotation). This is because the damper mounts to this bottom A arm, defining its vertical displacement. The upper A arm can be modeled as a roller constraint in a local coordinate system. The A arm mount point will be fixed in the plane of the A arm but free to move out of plane. While not a perfect representation of the pivot, this is a good application of a small-angle assumption. The toe adjustment tie rod mount point will be constrained only in the direction of the toe adjustment tie rod using a similar method.

Refer to the "Rear Load Calculations.mw" Maple Sheet for the solutions to these static equations.

Refer to the "Unsprung Loads" Spreadsheet for the loads applied to the rear upright.

4. Bearing Selection

By: Thaddeus Hughes
Date: 7/25/2017

Bearings on the unsprung mass system see primarily radial forces, although thrust definitely occurs. While tapered needle roller bearings are standard in larger vehicles, on lighter ones, deep groove ball bearings (DGBB) have shown to be a good way to reduce mass. 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. From NSK's catalog (e1153), we get the following specifications for a bearing which would allow the CV-in-hub design:

Boundary Dimensions (mm)				Basic Load Ratings (N)		Bearing Numbers					
d	D	B	r min	C_r	C_{0r}	Open	Shielded	Sealed		With Snap Ring Groove	With Snap Ring
60	75	7	0.3	7 000	7 700	**NB 712A**	—	—	—	—	—
	78	10	0.3	11 500	10 900	**6812**	**ZZ**	**VV**	**DD**	**N**	**NR**
	85	13	1	19 400	16 300	**6912**	**ZZ**	**VV**	**DDU**	**N**	**NR**
65	80	7	0.3	7 200	8 300	**NB 713A**	—	—	—	—	—
	85	10	0.6	11 900	12 100	**6813**	**ZZ**	**VV**	**DD**	**N**	**NR**
	90	13	1	17 400	16 100	**6913**	**ZZ**	**VV**	**DDU**	**N**	**NR**

We will now perform the necessary lifetime calculations to determine if this bearing (NB 713A) 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.

Putting this together:

```
> restart
Basic Load Rating in lbf
> Cr:=7200*0.22
                              Cr := 1584.00000                         (1)
Reliability factor
> a1:= 0.33
                              a1 := 0.33000                            (2)
Material factor
> a2:=1
                              a2 := 1                                  (3)
Lubrication factor
> a3:= 1
                              a3 := 1                                  (4)
Load applied
> P:=1600
                              P := 1600                                (5)
> L_revolutions := a1*a2*a3*(Cr/P)^3*1e6
                              L_revolutions := 3.20199 10^5            (6)
> L_miles := L_revolutions * 2*Pi*0.75 / 5280
                              L_miles := 285.77664                     (7)
```

This bearing will hold up with a reliability of 98% for 285 miles of cornering- which is more than half of the lifetime target for the vehicle. Thus, these bearings will be suitable.

By: Thaddeus Hughes
Date: 7/27/2017

http://www.kaydonbearings.com/RealiSlim_sealed_bearings_JB_typeC.htm We may need to use JB025CP0 bearings as they come with seals. The load rating is nearly the same (1549 lbf) if the ISO rating is indeed correct. If not, though, these are not suitable for our purposes and the 6813 bearings should be adopted instead. The effects of compliance should also be considered.

By: Thaddeus Hughes
Date: 10/1/2017

Kaydon isn't pulling through. We're going to revert to 6813 bearings in the rear and upgrade to 6810s in the front.

5. Rear Upright Design

By: Thaddeus Hughes
Date: 7/25/2017

To design a component, it is important to first understand the function of every bit of it.

The center bearing boss must be designed to hold bearings securely and without binding.

The bottom mount point to the suspension must carry vertical and horizontal loads, but is a pivot point; there is no moment support.

The upper mount points to the suspension must carry horizontal loads, and are only a pivot point in the X-direction; they must carry moments about the Y axis. Moments about the X and Z axes are taken care of by the horizontal components created by the upper and lower pivot points.

6 degrees of freedom are defined, so the upright is statically determinate with respect to external forces. This will make thinking about stress flows easier. We can surmise that while the loads on the bottom mount are smaller, it will be easier to design this point to be stiff because it only carries forces, not moments. The upper mount, however, could be more difficult.

Section	Parameter	Units	Stationary	Left Corner	Right Corner	Acceleration	Braking
Upright Forces (rectified)	Thrust (+Z onto upright)	lbf	0.00	-220.00	331.10	0.00	0.00
	Inner X (+ onto upright)	lbf	0.00	-6.00	-6.00	-105.00	169.20
	Inner Y (+ onto upright)	lbf	-0.51	-1173.84	1764.99	-0.86	-1.39
	Outer X (+ onto upright)	lbf	0.00	26.00	26.00	455.00	-733.20
	Outer Y (+ onto upright)	lbf	138.01	1311.34	-1528.49	234.20	377.39

By: Thaddeus Hughes
Date: 7/17/2017

RGP006's rear uprights weigh .7471 lbs

The upright initially built looks like this. The green component weighs .7140 lbs.

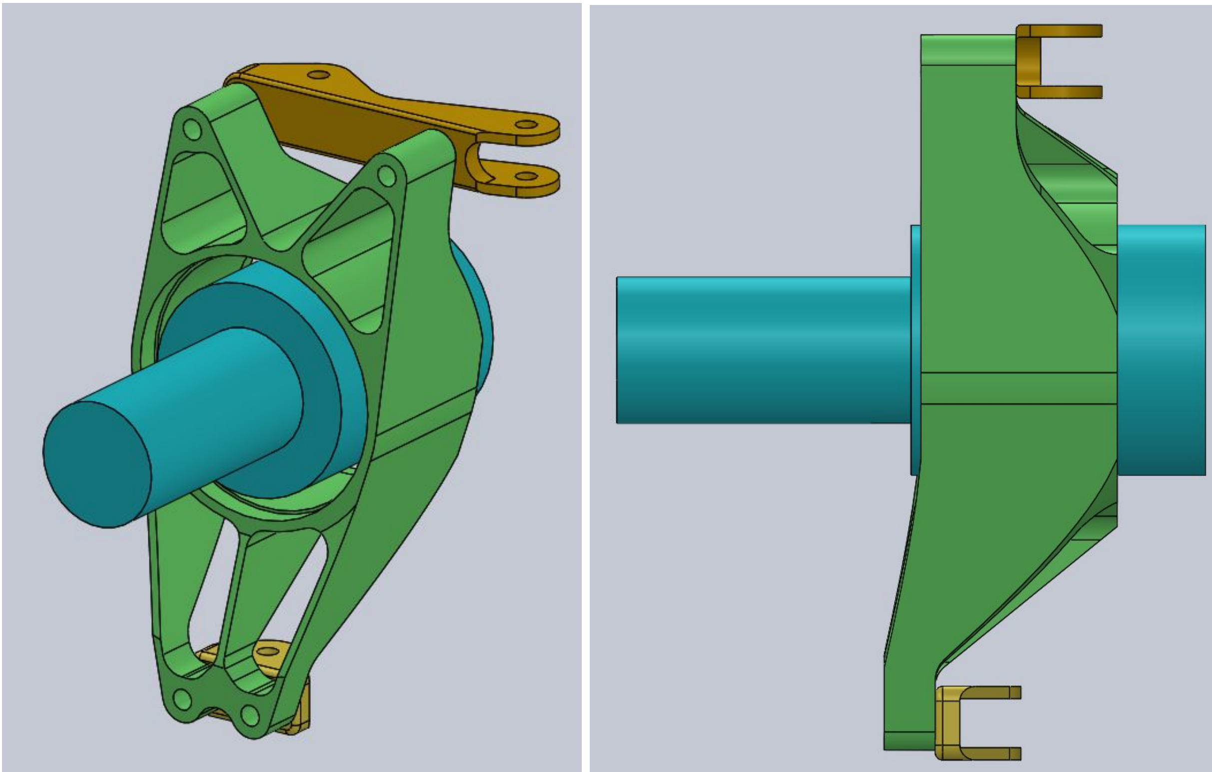


Figure 5.1: Initial upright as of 7/17/2017

After some slicing up of the legs of the upright and clever mesh controls, a mesh is generated in ANSYS which is composed completely of quads. This mesh takes a significant amount of time to produce- almost as much as it does to solve! This may seem insane, then, but this means doing sweeps over multiple load cases is cheaper, and when we get to contact analysis the improved mesh will decrease runtimes, fully justifying it. Additionally, the long struts will solve more accurately with quads rather than tris.

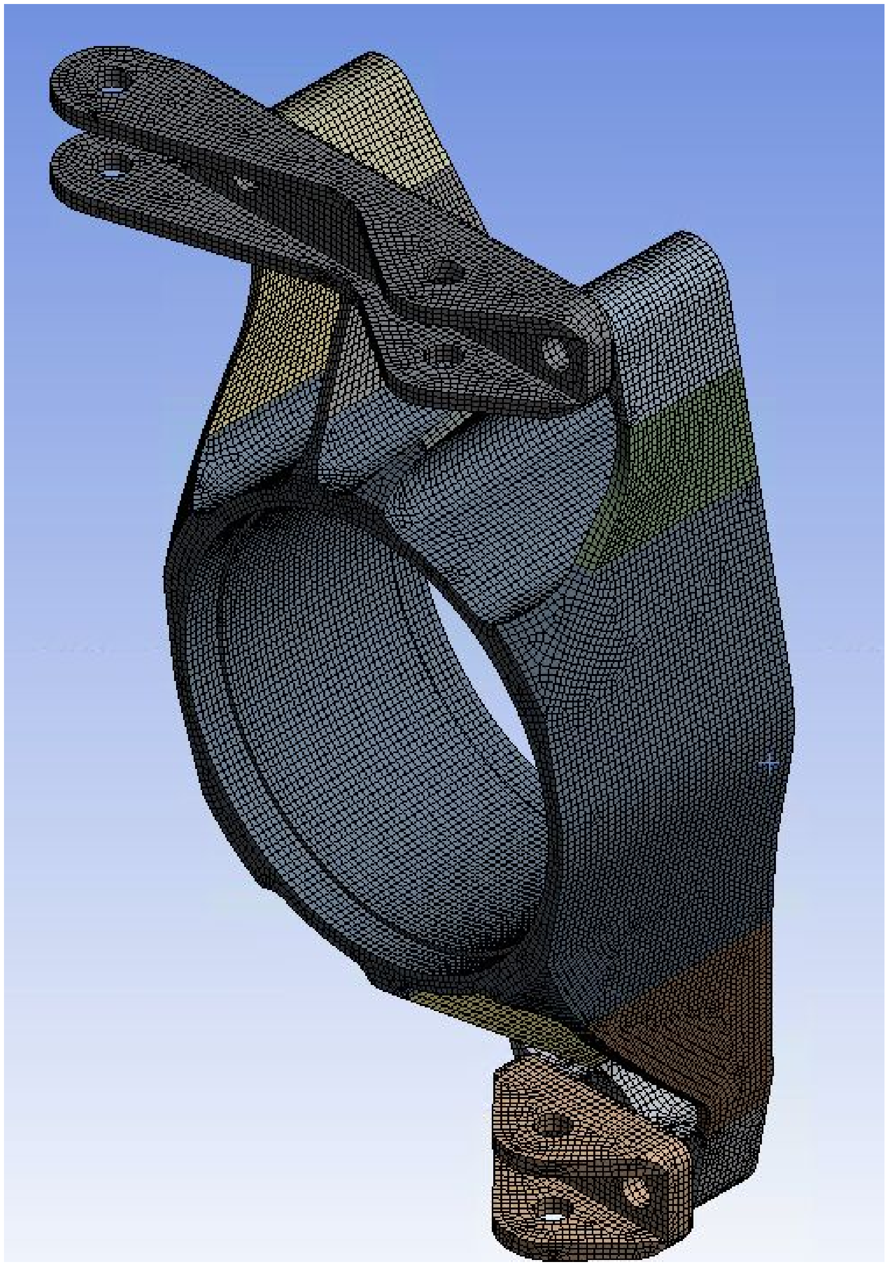


Figure 5.1: Initial upright mesh

FEA setup is done with 7 key commands:

- 2 bearing loads for the inner and outer bearings (radial bearing forces)
- 1 Z-direction load on the bearing surfaces (thrust bearing forces)
- A toe-rod coordinate system set-up at the top (figure 5.2)
- Remote displacement for the bottom two bolt holes (in X, Y, and Z directions)
- Remote displacement support for the upper A arm bolt holes (in X and Z directions in the toe-rod coordinate system)
- Remote displacement support for the toe rod bolt holes (in Z direction in the toe-rod coordinate system)

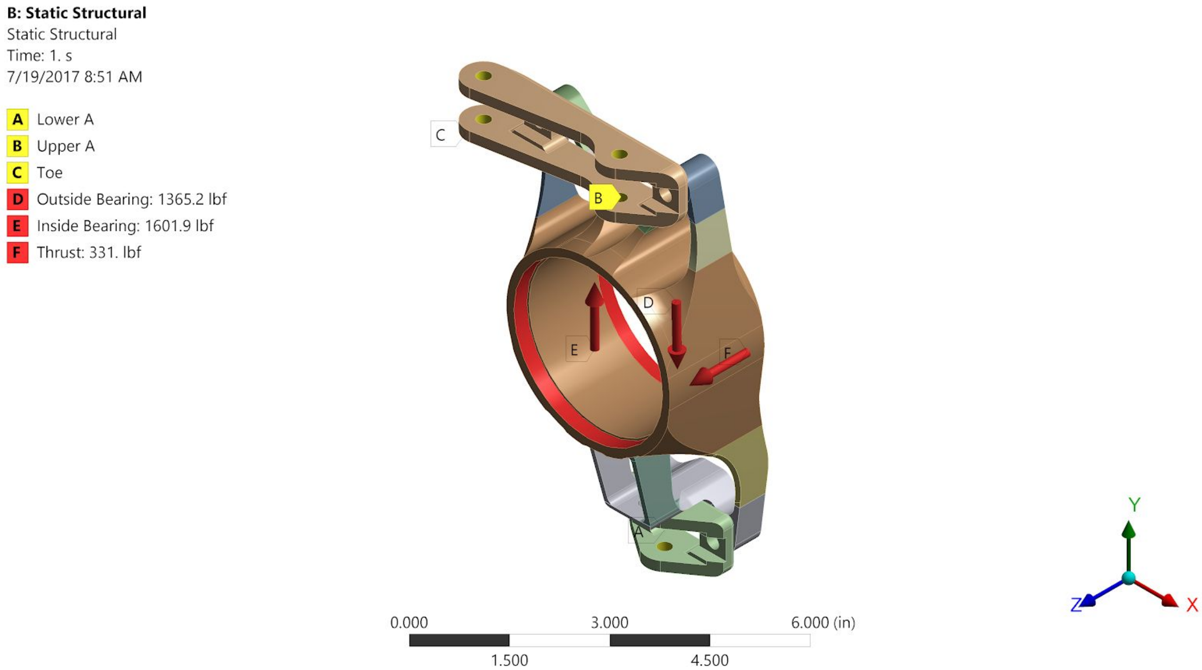


Figure 5.2: Upright FEA set-up. B&C in this image are not in the correct position for the remote displacement supports; location was not updated.

Contacts are bonded between all bodies for simulation speed and stability. Later, the true contact nature of fastener connections will be assessed. For right now, we will leverage Saint Venant's principle to justify this, disregarding stresses at the contact regions and rapidly designing the overall shape of the upright. Later, we will do the contact analysis in the smaller area of contact to ensure our design is valid. This approach should minimize computational time.

Thus, the outline for analysis looks like so:

- FEA1: Rear upright assembly
    - Analyze for fatigue under cornering, static, and acceleration events
    - FEA1_2: Rear upright brackets
        - Same conditions as FEA1
- FEA2: Rear hub to wheel center assembly (contact analysis)
    - Analyze for fatigue under cornering, static, and acceleration events
- FEA3: Full unsprung mass assembly (upright, brackets, etc...) (likely bonded analysis)
    - Analyze for toe compliance
    - Analyze for camber compliance

Under the Right Corner conditions we get the following result, in which many portions of the upright are above yield!

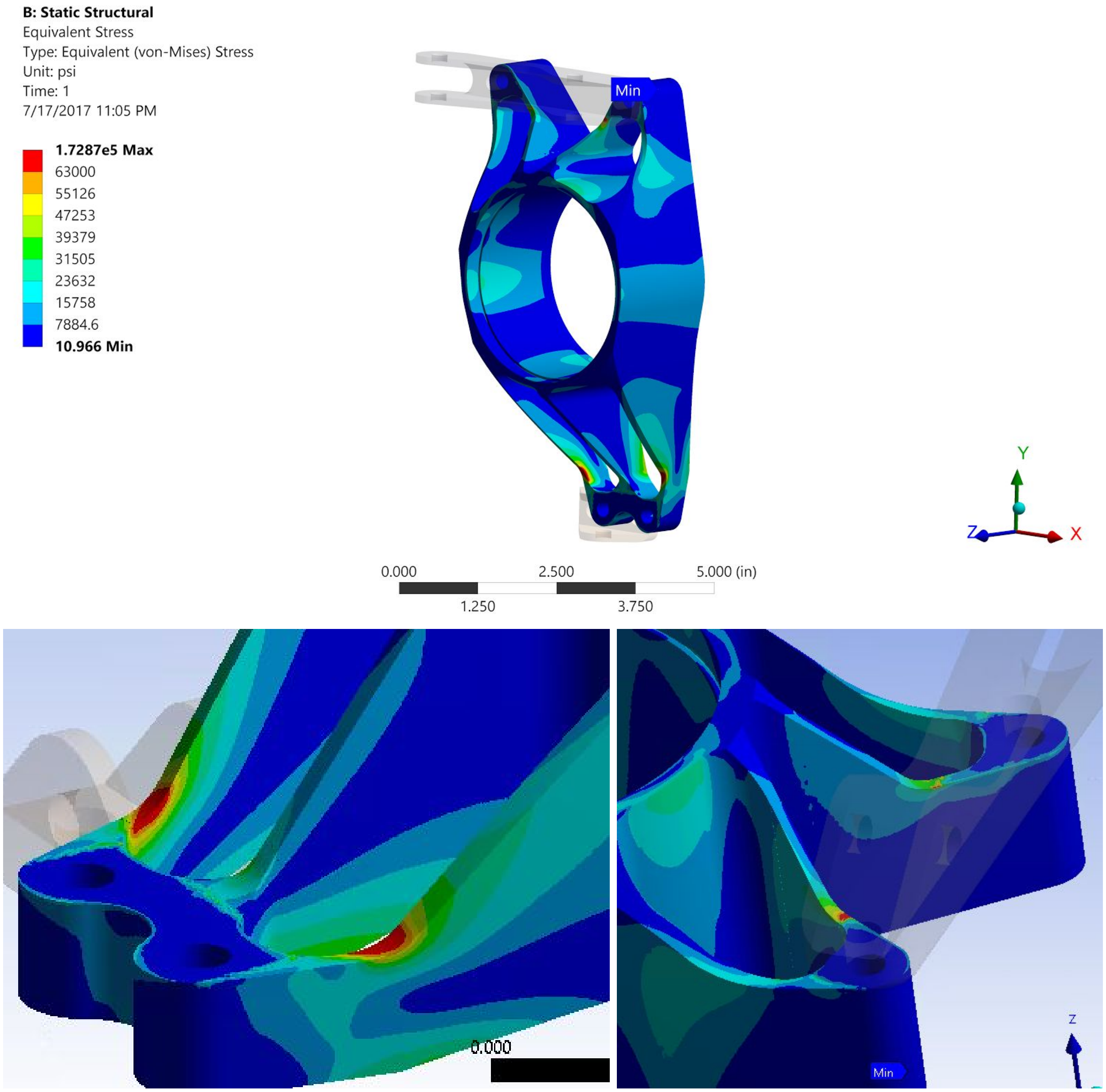


Figure 5.4: Initial upright FE stress results. Note locations of higher stress occur at the transition point between the 'legs' and the brackets, and the lower legs have a stress pattern that suggests the members are in bending (low stress in center, high stress at edges)

First, we will do the following to decrease stress:

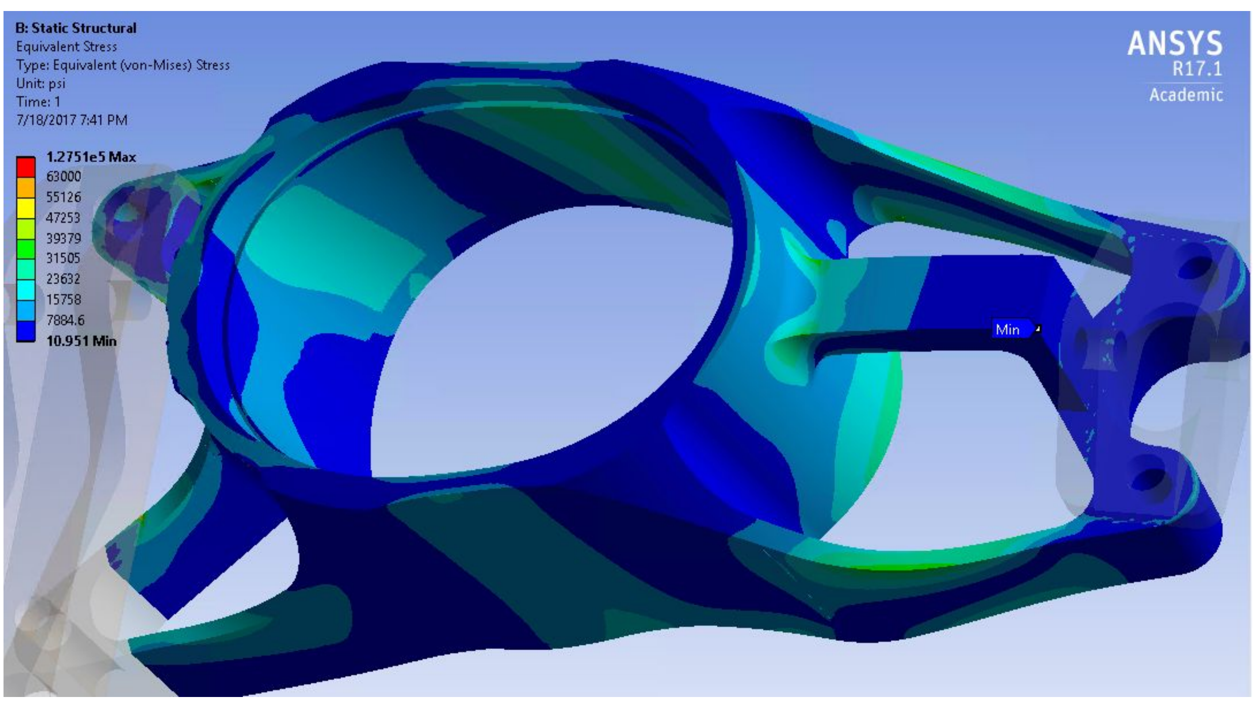
- Increase bottom leg thickness from 0.1" to 0.15"
- Increase bearing boss thickness from 0.125" to 0.15"
- Decrease the z-direction width of the bottom bracket so as to allow more lug thickness and reduce the moment produced by vertical reaction forces.
- Increase the cutout for all legs and create a larger fillet transitioning from the leg to mounting lugs

By: Thaddeus Hughes
Date: 7/18/2017

A potentially good way to cut down on the stress for the bottom mount point would be to better support the bracket as shown below in Figure 5.5. This would transfer vertical loads directly into the legs, rather than transmitting them as a combined moment and vertical load from the bracket, as well as reducing compliance, as the bracket is now properly triangulated! This would not be nearly as useful at the top, though, as the loads here are almost entirely horizontal and so bolt preload along with the wider base is enough to transfer the moment. This same mounting system should be employed with the front uprights as well, but on the top mount where the pullrod goes to.

Figure 5.5: Potential improvement for push/pullrod A-arm mounting bracket

Although this isn't stressed much, it does quite a bit to relieve stress. It does look ugly, though.



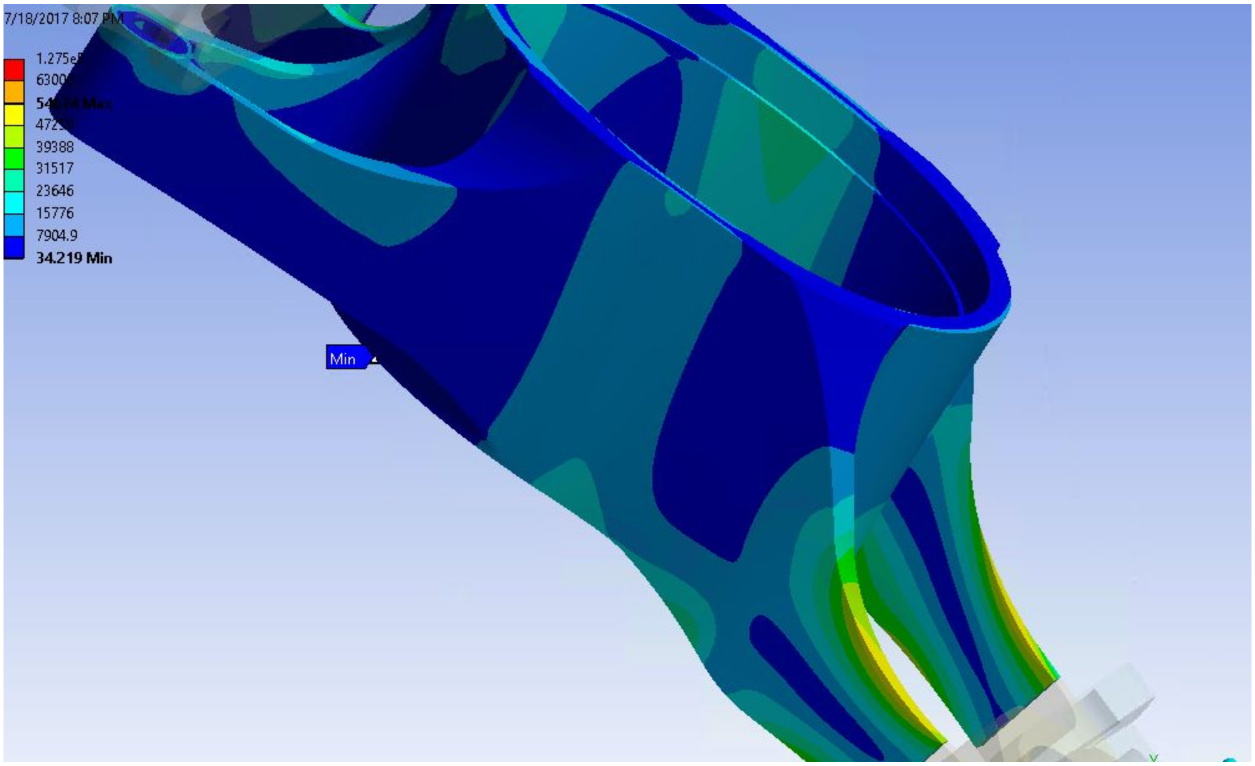


Figure 5.6: Revised bottom central strut versus without strut

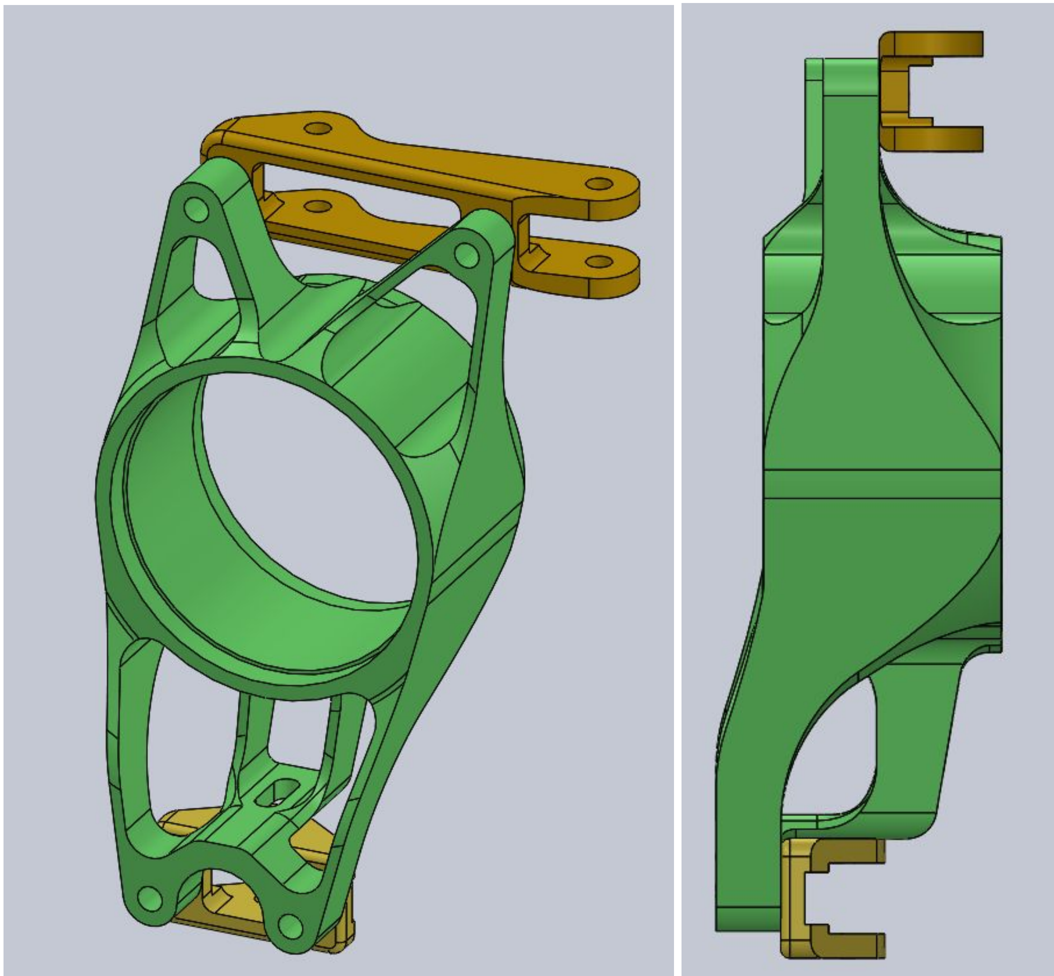


Figure 5.6: Upright revision 2A

After making the strut system prettier and doing a few iterations, we arrive on a good result with a few stress concentrations to deal with. The FOS used for these parts is against zero-based fatigue for an endurance limit of 1e+8 cycles. There are points where further mass could be removed, but this design shows a significant improvement in strength with some weight reduction (only considering the upright itself, not bearings). The load cases which show the most stress are right corners and acceleration events.

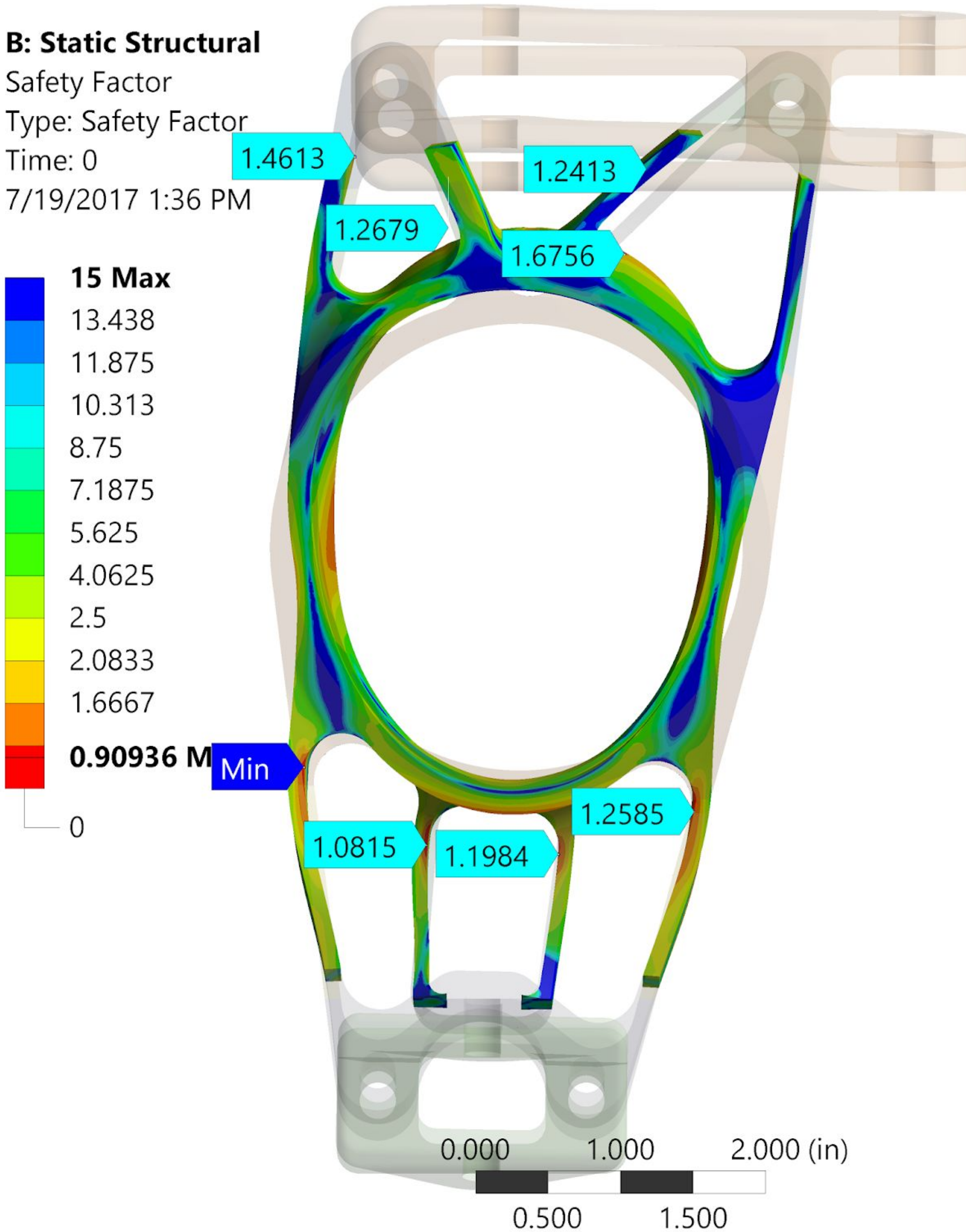


Figure 5.7: Upright revision 2A, safety factor against 1e+8 zero-based right corner cycles scoped to valid areas. Note minimums appear mostly in lower struts.

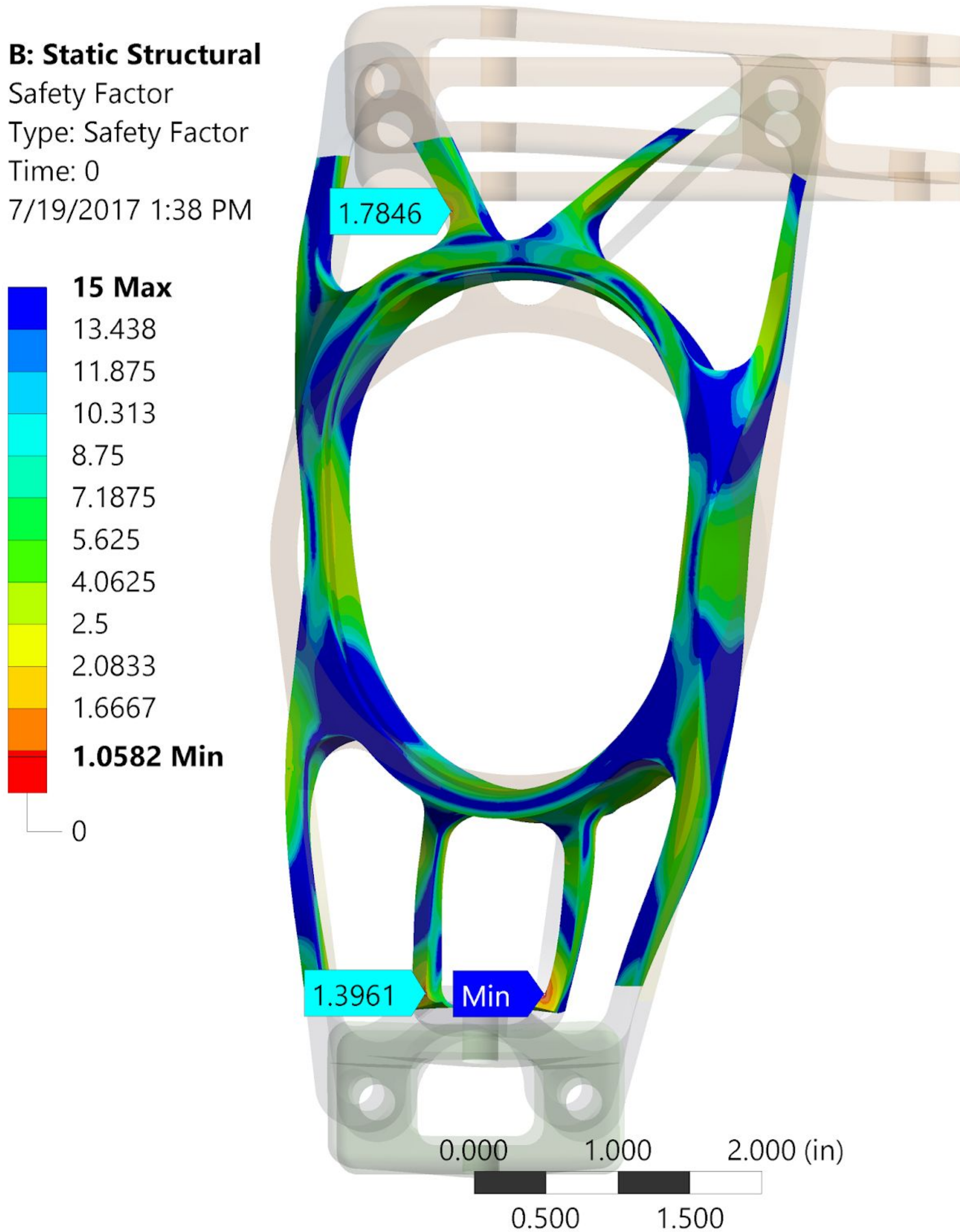


Figure 5.7: Upright revision 2, safety factor against 1e+8 zero-based left corner cycles scoped to valid areas. Note minimums appear mostly in lower struts.

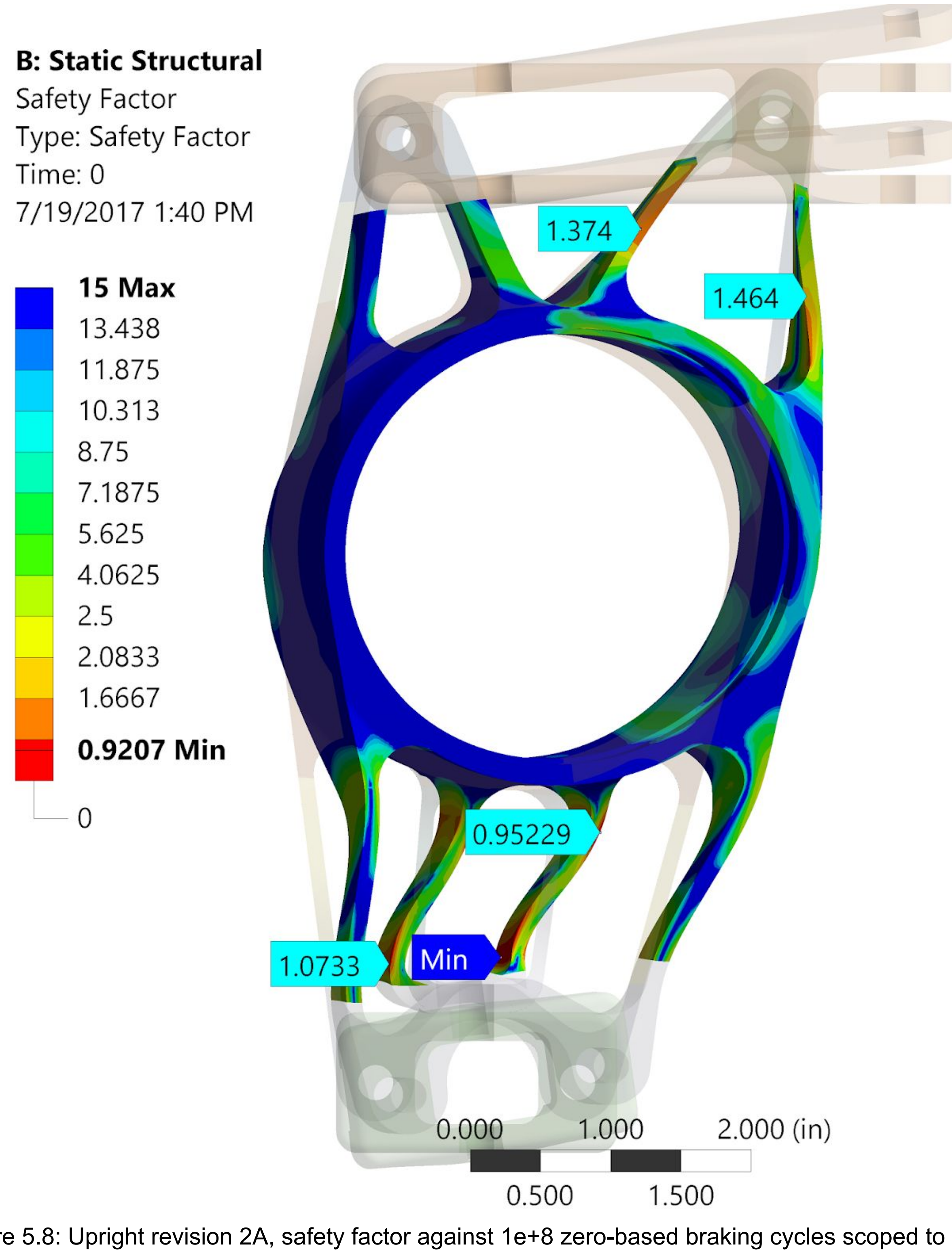


Figure 5.8: Upright revision 2A, safety factor against 1e+8 zero-based braking cycles scoped to valid areas. Note minimums appear mostly in lower struts, because these are not truly triangulated and are just thin beams in bending!

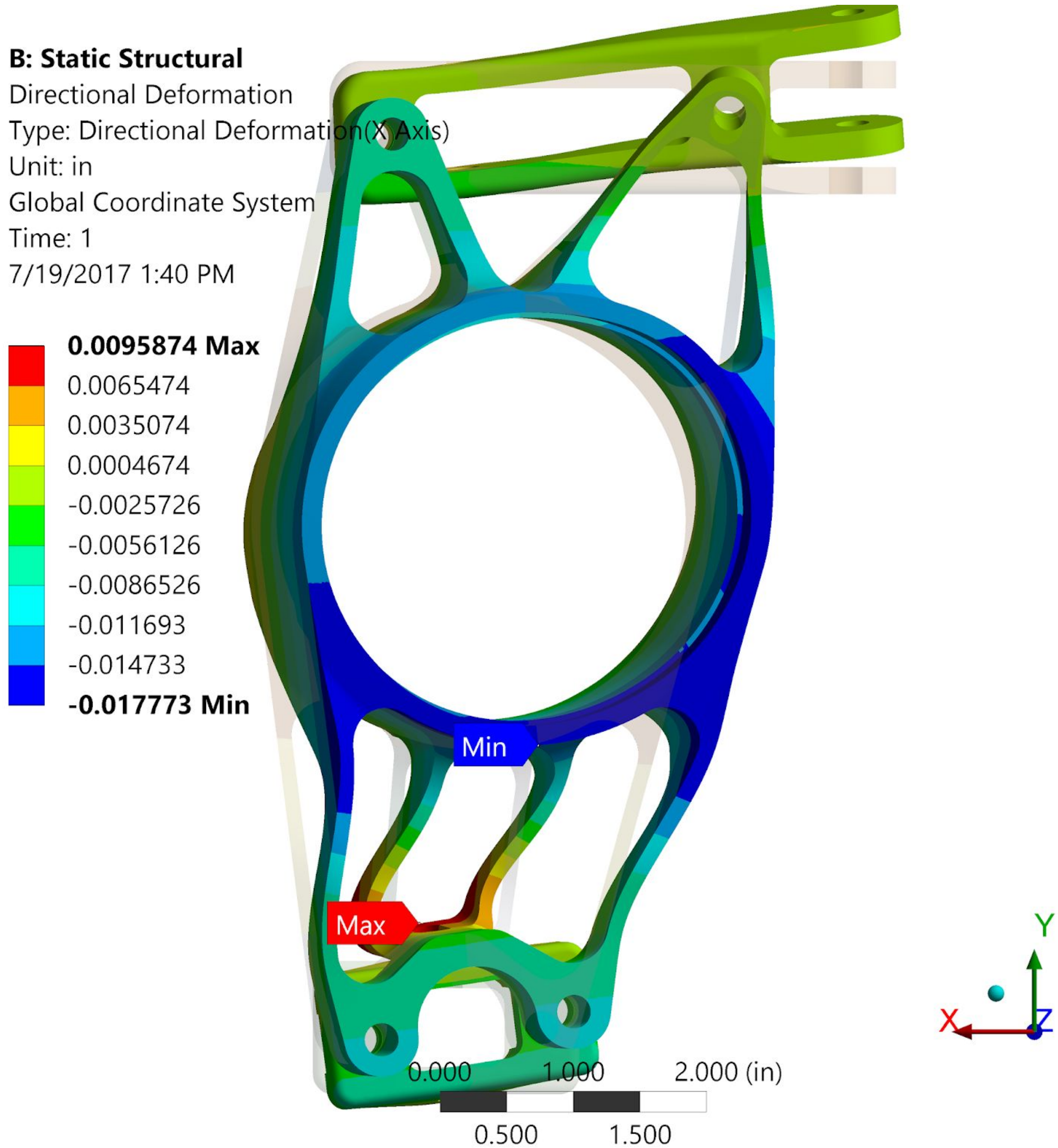


Figure 5.9: Upright revision 2A, X-axis deformation under a braking cycle. This demonstrates significant toe compliance and ought to be rectified.

Name	P1 - Thrust Z Compon...	P2 - Inside Bearing X Component	P3 - Inside Bearing Y Component	P4 - Outside Bearing X Component	P5 - Outside Bearing Y Component	P6 - Equivalent Stress Maximum	P7 - Life Minimum	P8 - Safety Factor Minimum
Units	lbf	lbf	lbf	lbf	lbf	psi		
DP 0	331	-5.24	1601.9	25.24	-1365	43797	4.7302E+07	0.90936
DP 1	-220	-5.24	-1062	25.24	1199.9	37636	1E+08	1.0582
DP 2	1	0	0.92	0	136.58	4996.8	1E+08	7.9705
DP 3	1	-91.63	1.57	441.63	231.77	30007	1E+08	1.3273
DP 4 (Current)	1	147.66	2.52	-711.66	373.48	43258	5.3716E+07	0.9207

Figure 5.10: Upright revision 2A, results from multiple loading conditions. Note worst case loadings. (Thrust component of 1 lbf is due to ANSYS complaining about load not being defined if it is zero, so a negligible value is used)

Design points:

DP 0: Left corner

DP 1: Right corner
DP 2: Resting
DP 3: Engine
DP 4: Braking

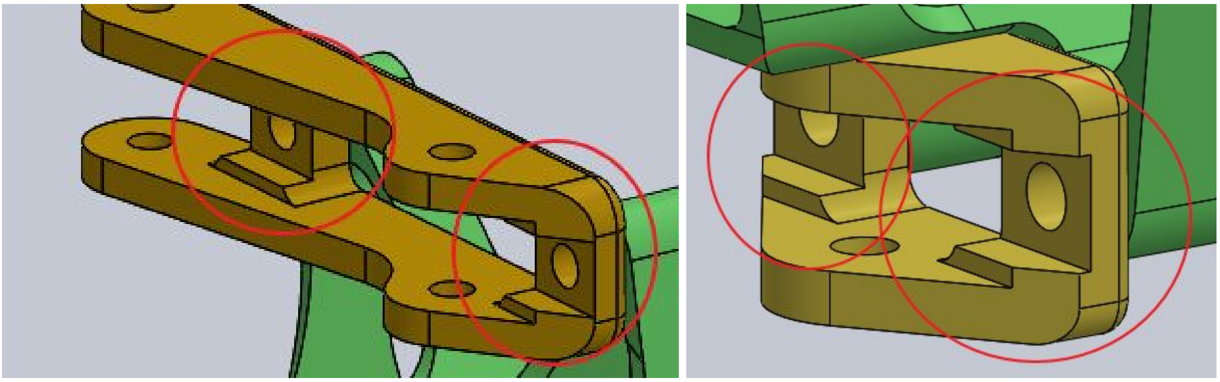


Figure 5.11: Wrench slots added to brackets

Work can quickly be done to bring the part to a FOS of 1.5 in the regions we are concerned with before proceeding to analyze with camber plates installed at the bottom and then contact analysis. Analyzing with camber plates at the bottom is important, as adding these plates changes where the support load is applied, thus changing the load distribution. It is not as important to add these plates to the top, since constraints and forces are applied in line with such adjustment. Additionally, clearance needs to be allowed for a hall effect sensor disk to be mounted to the hub-wheel interface. It may be beneficial to add ribs supporting the bearing boss on the front and rear, as it currently deforms severely from the bearing load.

The first course of action, then, is:

- Thicken up the lower struts
- Move apart the middle lower struts to provide triangulation for the bottom
- Add space for hall effect sensor disc
- Adjust fillets around stress concentrations

Then,

- Add a rib around the sides of the upright to stiffen the bearing boss

By: Thaddeus Hughes
Date: 7/19/2017

With the lower struts thickened, spread, space allocated for a hall effect sensor disc, and fillets adjusted, another run is made of the V2 upright, V2_B.

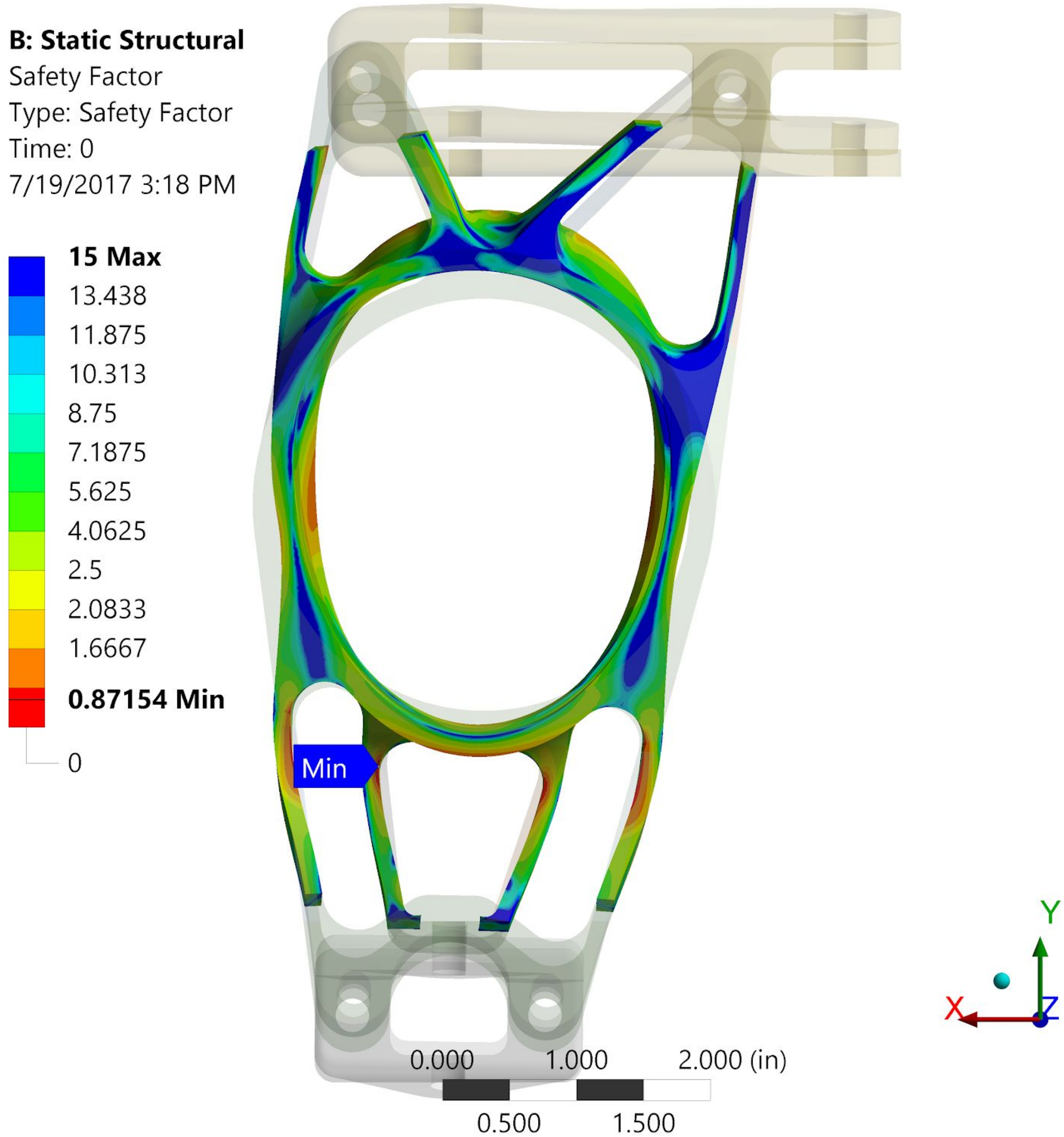


Figure 5.12: Upright revision 2B, FOS for fatigue during a left turn

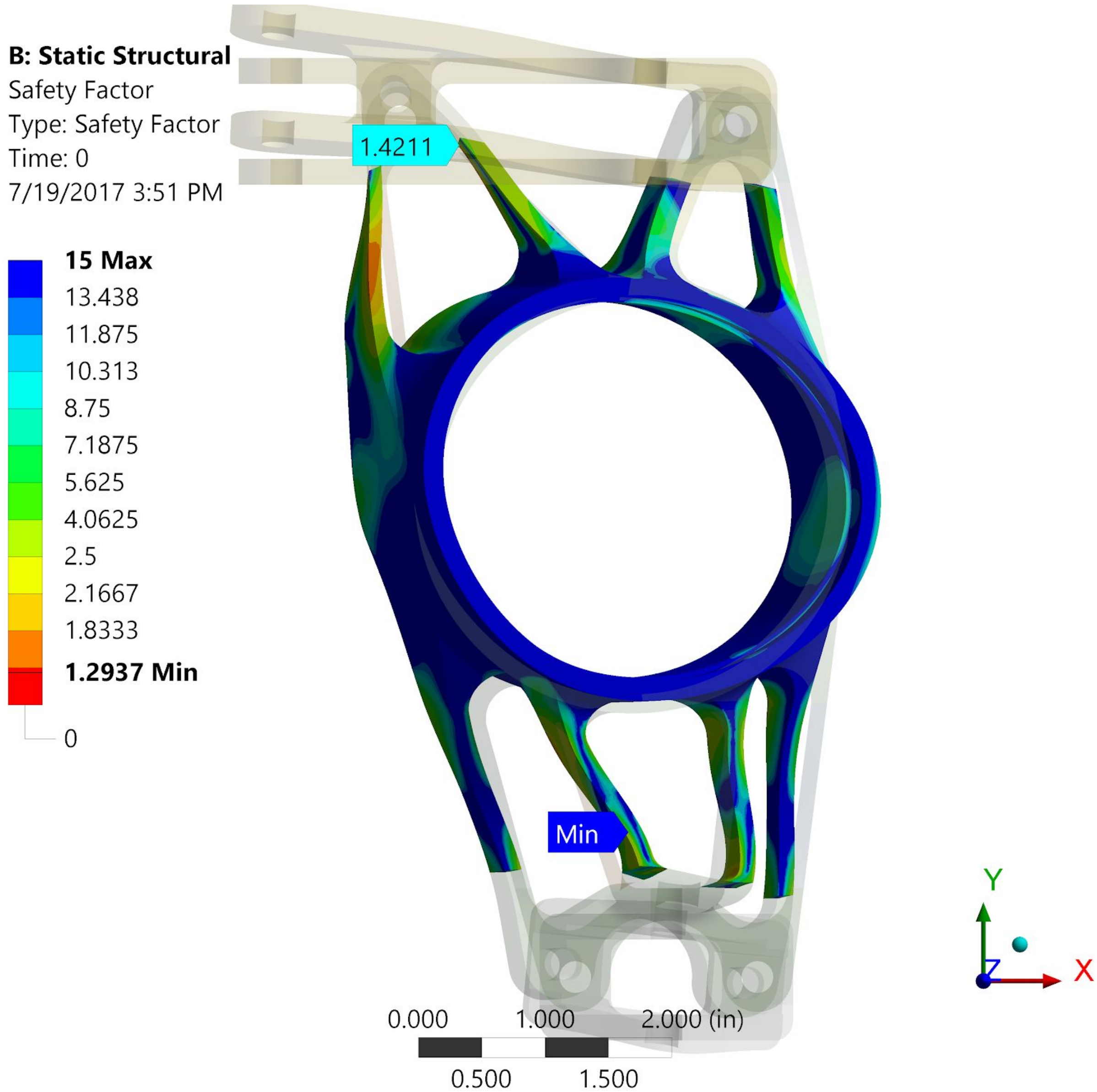


Figure 5.13: Upright revision 2B, Fatigue FOS during a braking event

Name	P1 - Thrust Z Compon...	P2 - Inside Bearing X Component	P3 - Inside Bearing Y Component	P4 - Outside Bearing X Component	P5 - Outside Bearing Y Component	P6 - Equivalent Stress Maximum	P7 - Life Minimum	P8 - Safety Factor Minimum	Retain
Units	lbf	lbf	lbf	lbf	lbf	psi			
DP 0 (Current)	331	-5.24	1601.9	25.24	-1365	45697	3.0229E+07	0.87154	☑
DP 1	-220	-5.24	-1062	25.24	1199.9	27581	1E+08	1.444	☑
DP 2	1	0	0.92	0	136.58	4700.1	1E+08	8.4737	☑
DP 3	1	-91.63	1.57	441.63	231.77	20248	1E+08	1.967	☑
DP 4	1	147.66	2.52	-711.66	373.48	30786	1E+08	1.2937	☑

Figure 5.14: Upright revision 2B, results from multiple loading conditions.

The upright is getting stronger in all design points except for left turns, where it slightly decreases. We'll make the bottom middle struts deeper and adjust fillets, and give it another go in version 2B!

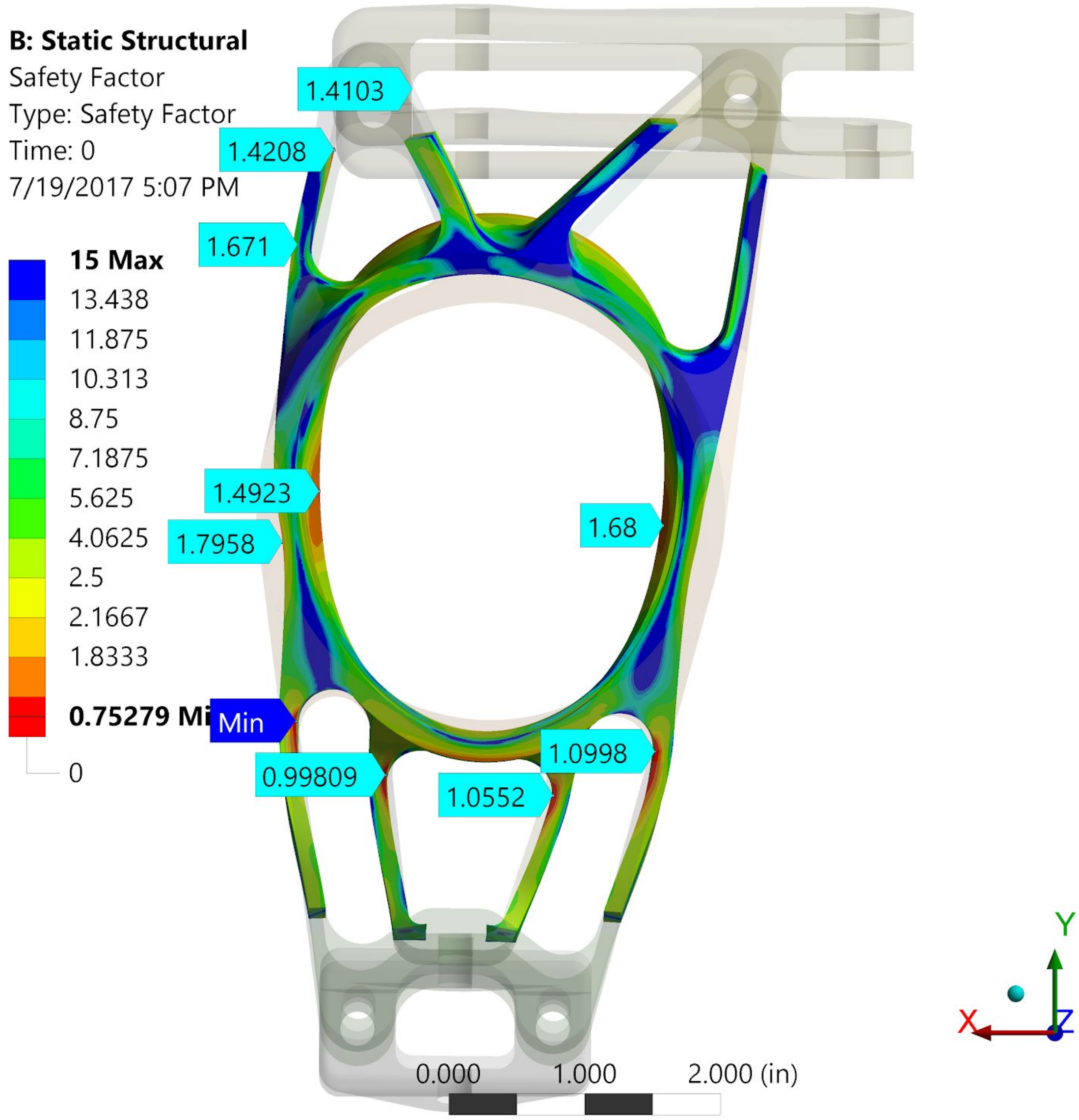


Figure 5.15: Upright revision 2C, Fatigue FOS for Left Corner Events.

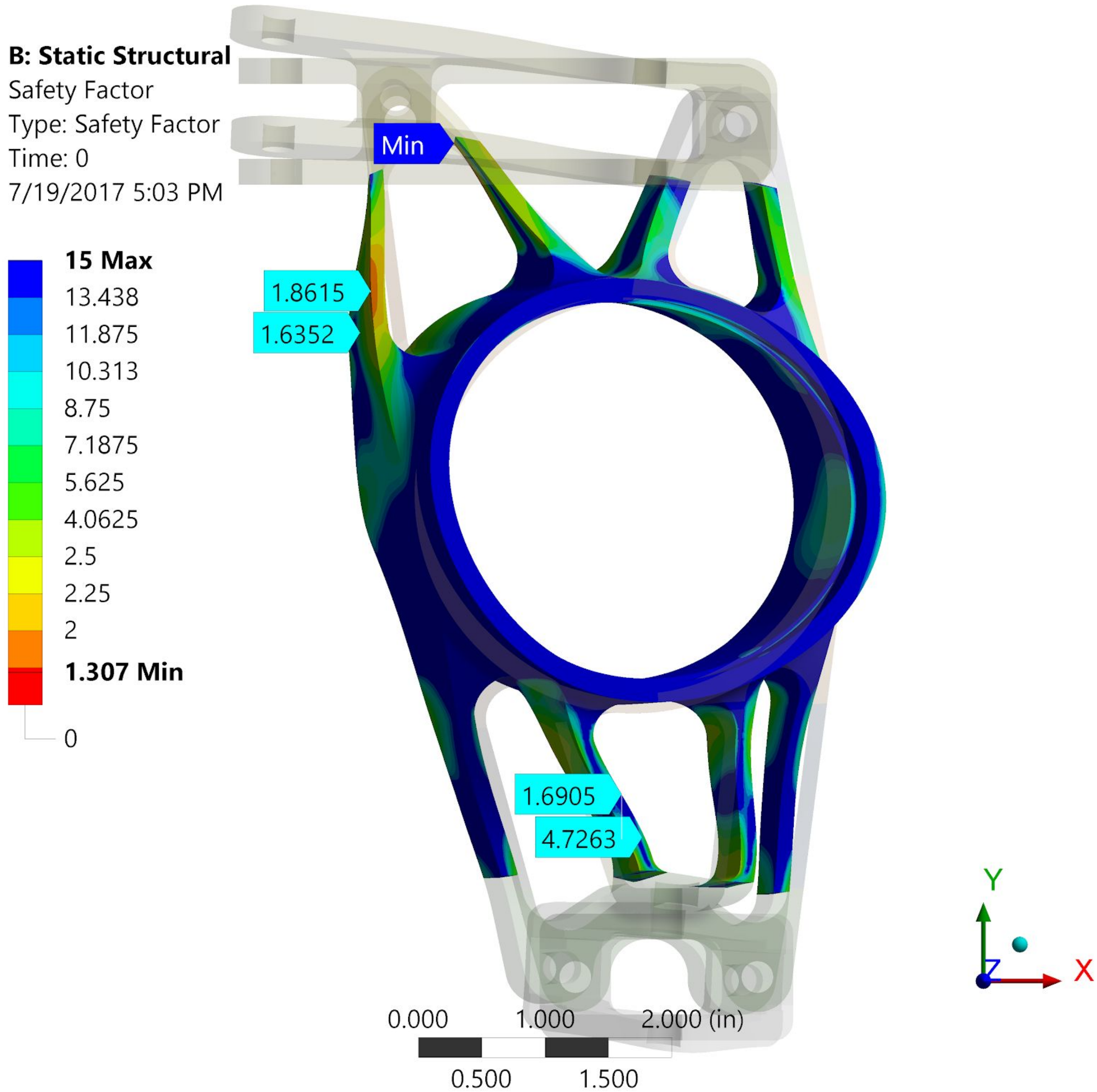


Figure 5.16: Upright revision 2C, Fatigue FOS for Braking Events.

Name	P1 - Thrust Z Compon...	P2 - Inside Bearing X Component	P3 - Inside Bearing Y Component	P4 - Outside Bearing X Component	P5 - Outside Bearing Y Component	P6 - Equivalent Stress Maximum	P7 - Life Minimum	P8 - Safety Factor Minimum
Units	lbf	lbf	lbf	lbf	lbf	psi		
DP 0	331	-5.24	1601.9	25.24	-1365	52906	5.5302E+06	0.75279
DP 1	-220	-5.24	-1062	25.24	1199.9	28132	1E+08	1.4157
DP 2	1	0	0.92	0	136.58	4216.9	1E+08	9.4448
DP 3	1	-91.63	1.57	441.63	231.77	21079	1E+08	1.8894
DP 4 (Current)	1	147.66	2.52	-711.66	373.48	30473	1E+08	1.307

Figure 5.17: Upright revision 2C, results from multiple loading conditions.

Fillet adjustment didn't work out so well for the left turn case. The FOS has dropped to 0.75! We'll readjust in rev 2D. We'll also add some extra support on the outside of the bearing boss- not sure how that will fare but it is indeed under a FOS of 1.5.

By: Thaddeus Hughes

Date: 7/22/2017

Much quick iteration was done to get to revision 5D. Namely, fillets have been adjusted, leg positioning on the top adjusted, strut size changed, the outside ribs added, and the topology of the bottom outer struts has been changed to make the struts straighter- helping speed sensor disc packaging as well as making stress flow smoothly.

A	B	C	D	E	F	G	H	I	J
Name	P1 - Thrust Z Compon...	P2 - Inside Bearing X Component	P3 - Inside Bearing Y Component	P4 - Outside Bearing X Component	P5 - Outside Bearing Y Component	P6 - Equivalent Stress Maximum	P7 - Life Minimum	P8 - Safety Factor Minimum	P9 - Total Deformation Maximum
Units	lbf	lbf	lbf	lbf	lbf	psi			in
DP 0 (Current)	331	-5.24	1601.9	25.24	-1365	26336	1E+08	1.5123	0.024723
DP 1	-220	-5.24	-1062	25.24	1199.9	17055	1E+08	2.3352	0.0091595
DP 2	1	0	0.92	0	136.58	2186.6	1E+08	15	0.0015305
DP 3	1	-91.63	1.57	441.63	231.77	12512	1E+08	3.1831	0.0080431
DP 4	1	147.66	2.52	-711.66	373.48	16305	1E+08	2.4426	0.010159

Figure 5.18: Upright revision 5D, results from multiple loading conditions

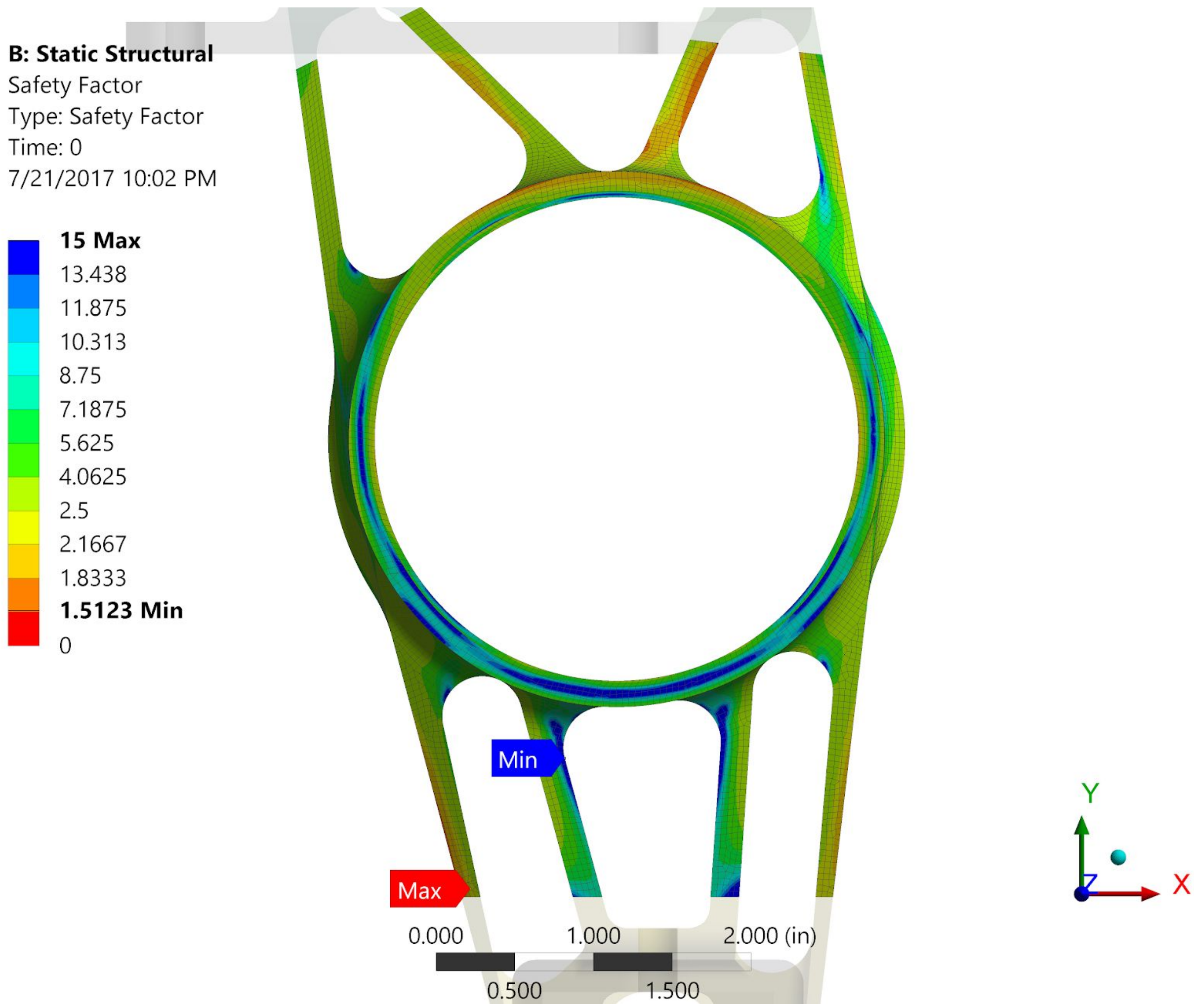


Figure 5.19: Upright revision 5D, Fatigue FOS during a left-hand corner

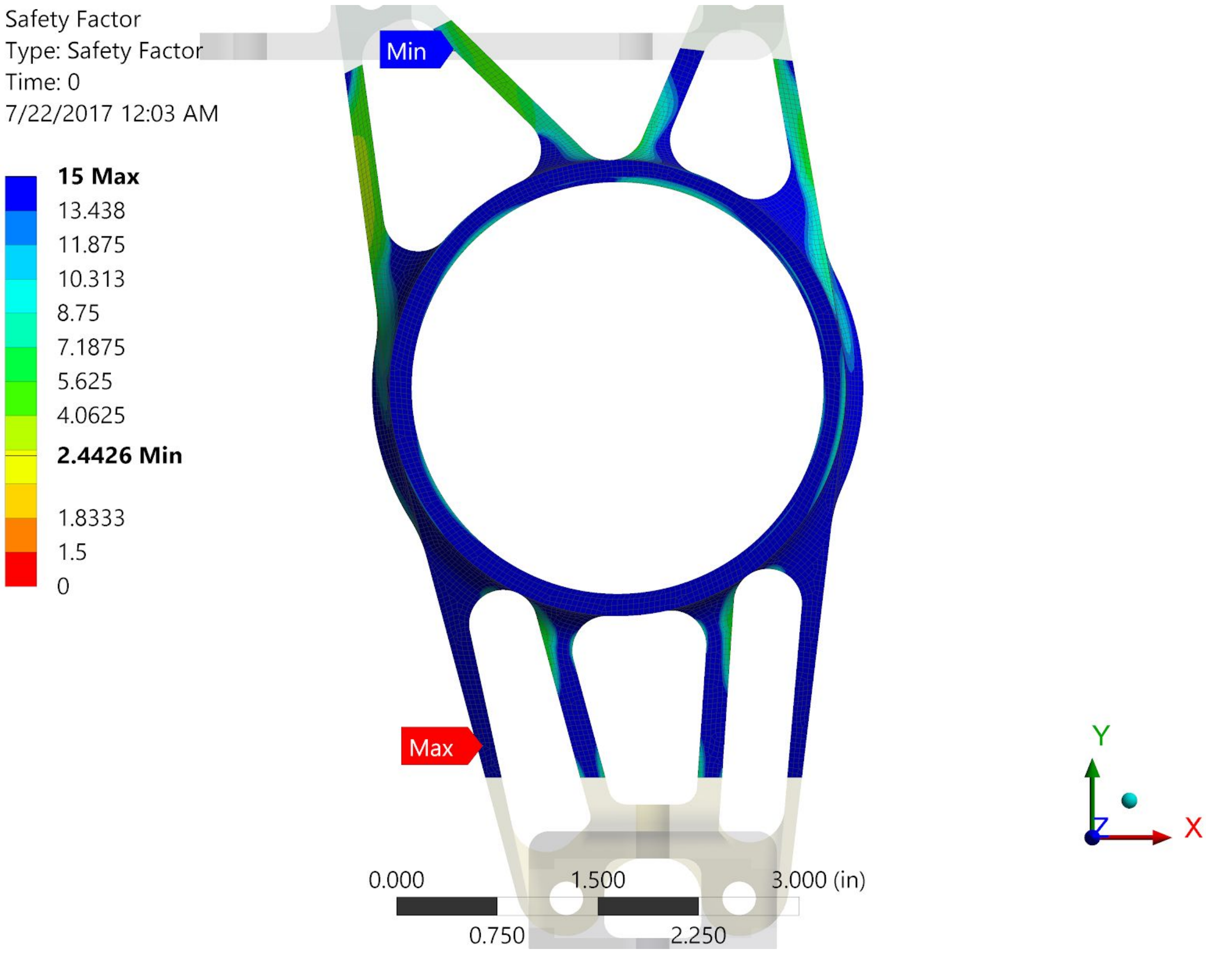


Figure 5.20: Upright revision 5D, Fatigue FOS under braking

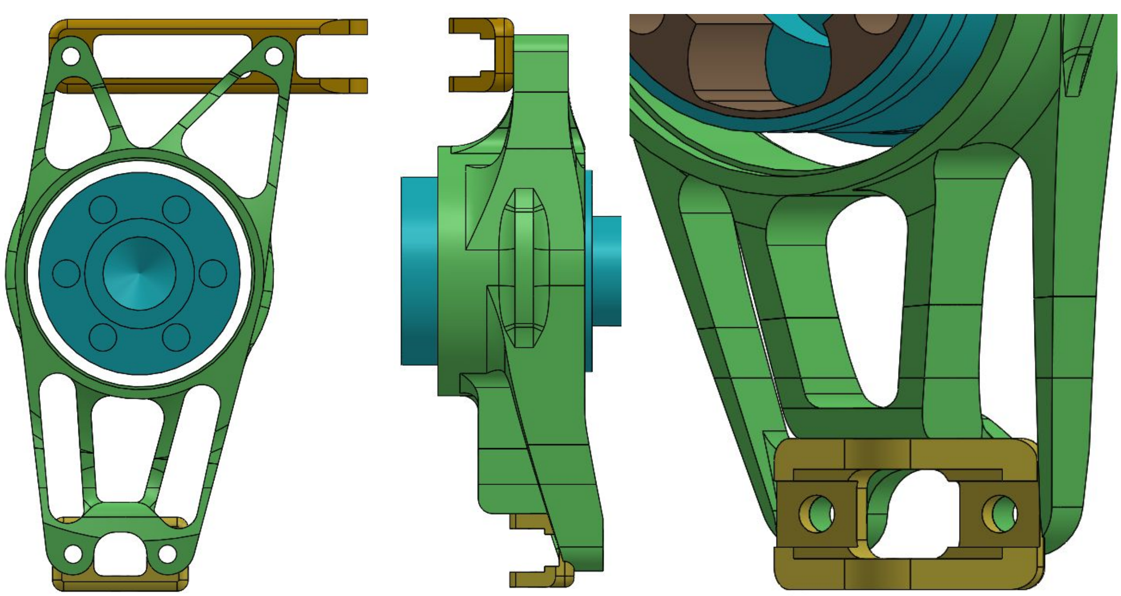


Figure 5.21: Upright revision 5D. Upright mass: 0.731 lb

By: Thaddeus Hughes
Date: 7/31/2017

Efforts to make good contact analyses were done but to little success. Thus, the old model was reloaded, the contacts scoped tighter, mesh refined, and the following resulted. The upright bearing boss was extended (providing better bearing support) and the lower struts simplified.

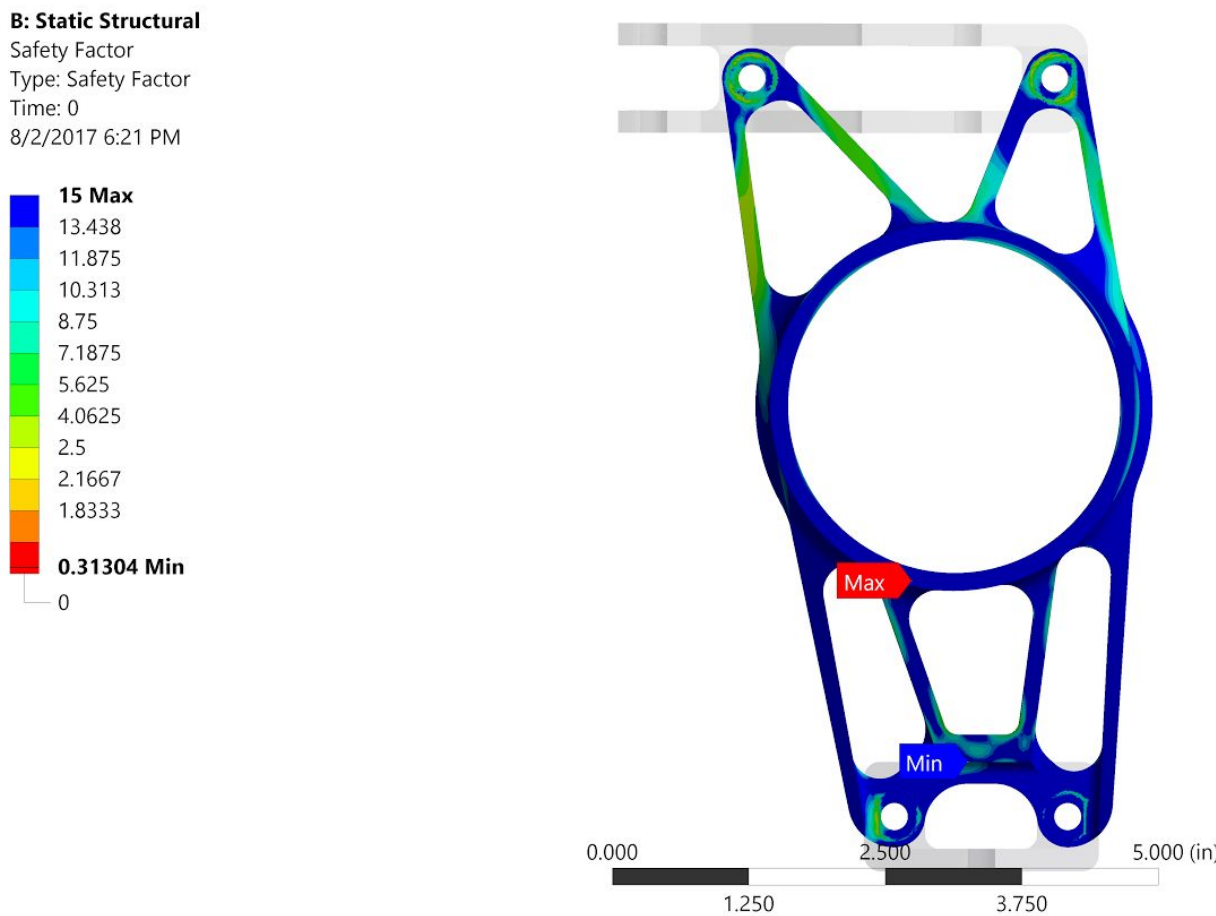


Figure 5.21: Upright revision 5N.

By: Thaddeus Hughes
Date: 8/3/2017

Version 5R+ passes all strength simulation with flying colors.

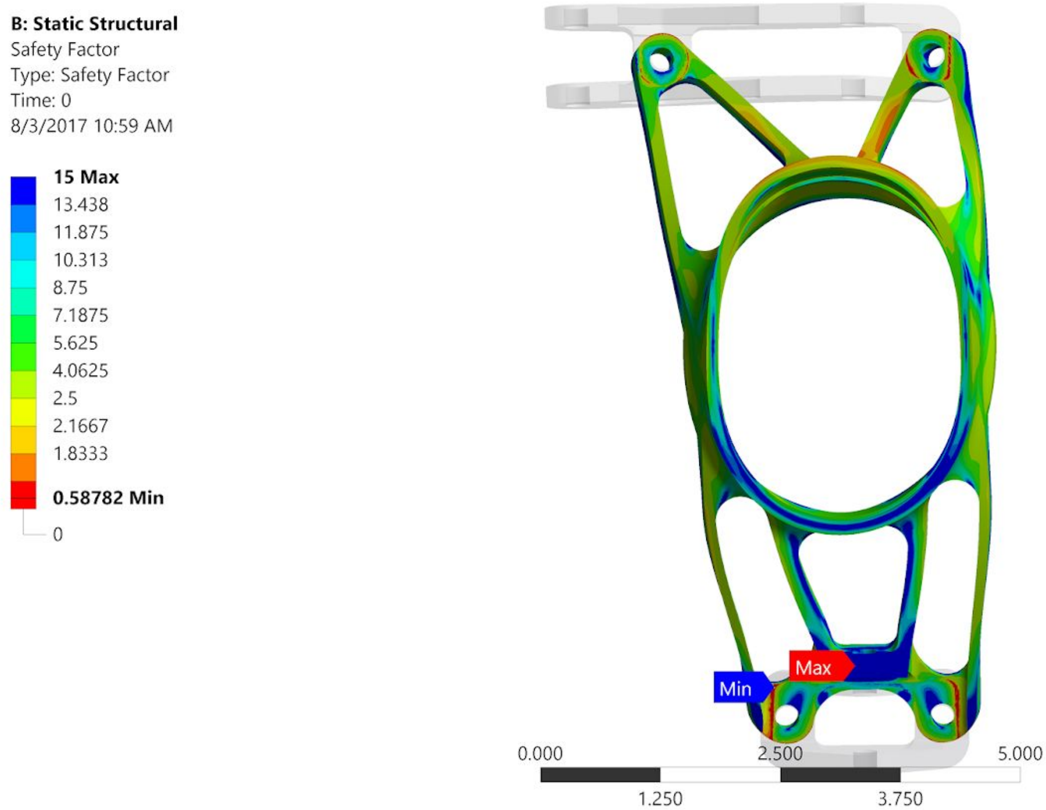


Figure 5.22: Upright revision 5R+, fatigue FOS under a left corner. All non-contact regions are above a FOS of 1.5.

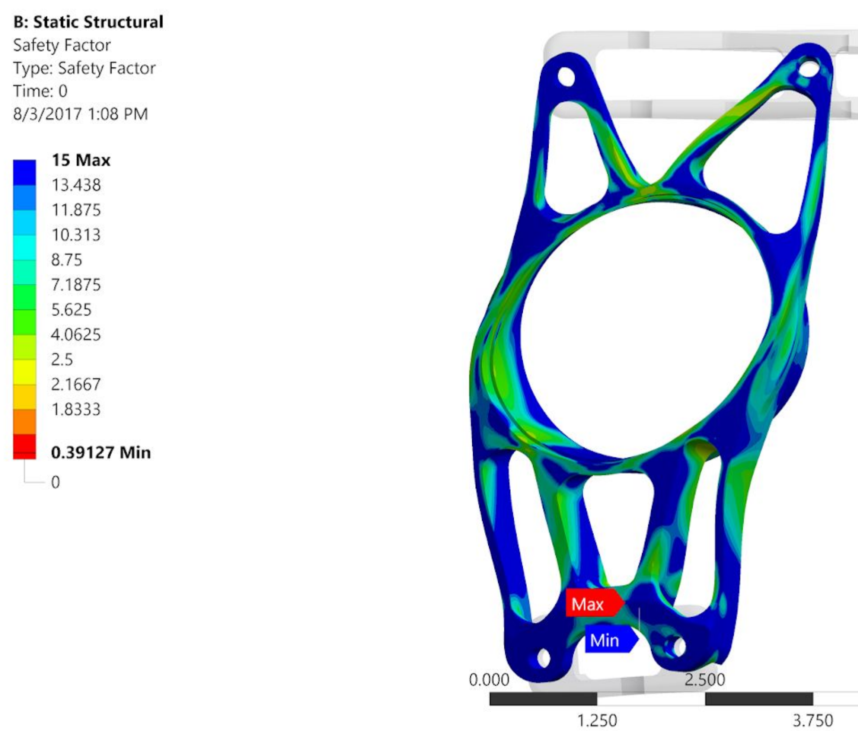


Figure 5.23: Upright revision 5R+, fatigue FOS under an acceleration event. All non-contact regions are above a FOS of 1.5.

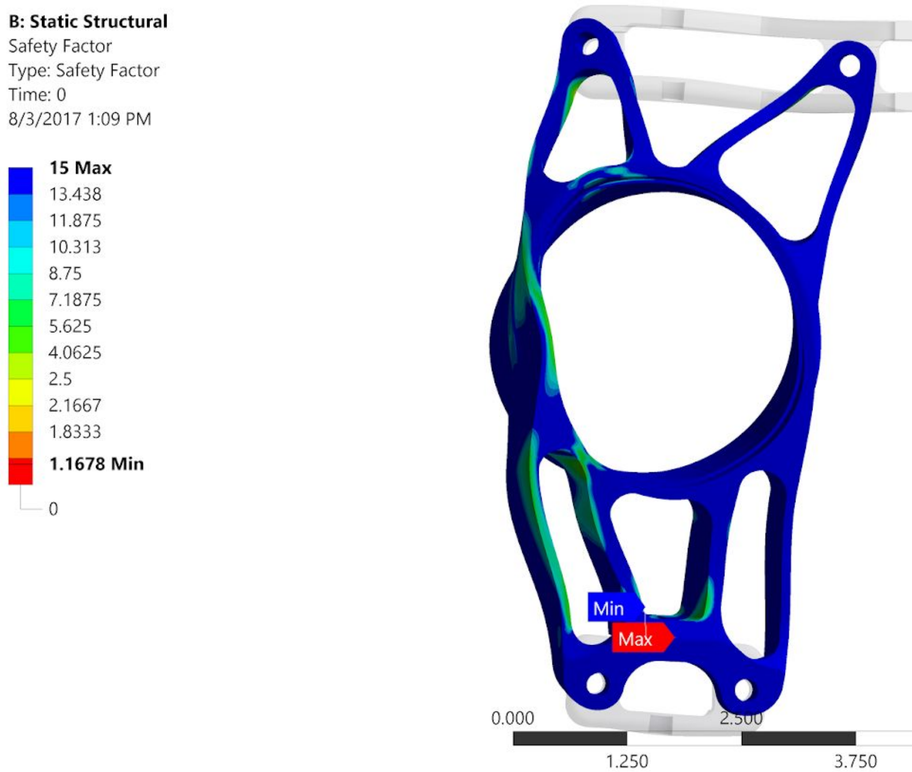


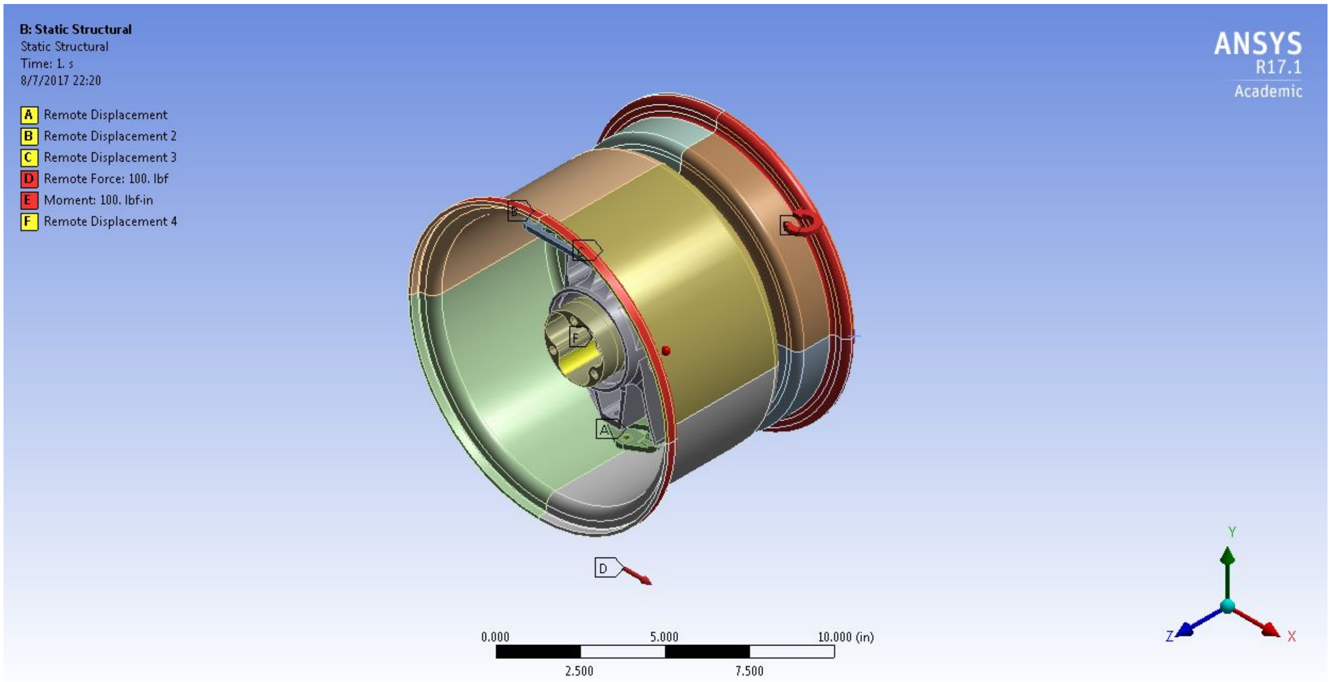
Figure 5.24: Upright revision 5R+, fatigue FOS under a braking event. All non-contact regions are above a FOS of 1.5.

Rear stiffness simulation was done with the same set-up used for the upright, and revolute joints in place of bearings, on a full upright-hub-wheel assembly.

Mesh convergence was first ran:

	Load						Inside						Outside					
	X	Y	Z	Moment	Upright & Hub Size	Wheel Shell Size	Xmin	Xmax	Ymin	Ymax	Zmin	Zmax	Xmin	Xmax	Ymin	Ymax	Zmin	Zmax
Name	P1	P2	P3	P4	P5	P6	P7	P8	P9	P10	P11	P12	P13	P14	P15	P16	P17	P18
DP 0	100	0	0	0.0001	0.2	0.1	-2.37E-03	5.54E-03	-3.95E-03	4.03E-03	-6.80E-04	7.69E-04	-2.92E-03	4.85E-03	-3.80E-03	4.03E-03	-1.89E-04	3.39E-04
DP 3	100	0	0	0.0001	0.175	0.1	-2.40E-03	5.57E-03	-3.98E-03	4.06E-03	-6.81E-04	7.70E-04	-2.95E-03	4.88E-03	-3.82E-03	4.06E-03	-1.88E-04	3.41E-04
DP 1	100	0	0	0.0001	0.15	0.075	-2.39E-03	5.63E-03	-4.00E-03	4.09E-03	-7.06E-04	7.97E-04	-2.95E-03	4.91E-03	-3.85E-03	4.09E-03	-1.96E-04	3.53E-04
DP 4	100	0	0	0.0001	0.125	0.075	-2.40E-03	5.64E-03	-4.01E-03	4.10E-03	-7.07E-04	7.97E-04	-2.96E-03	4.92E-03	-3.85E-03	4.10E-03	-1.96E-04	3.52E-04
DP 2	100	0	0	0.0001	0.1	0.05	-2.40E-03	5.66E-03	-4.02E-03	4.11E-03	-7.11E-04	8.02E-04	-2.97E-03	4.94E-03	-3.86E-03	4.11E-03	-2.01E-04	3.57E-04
DP 5	100	0	0	0.0001	0.075	0.04	-2.38E-03	5.69E-03	-4.03E-03	4.11E-03	-7.34E-04	8.26E-04	-2.97E-03	4.95E-03	-3.87E-03	4.11E-03	-2.01E-04	3.60E-04

The mesh in DP1 is used to balance speed and precision. Stiffness simulations will seemingly always benefit from further refinement. The displacements are measured at the edges of the rims.



	Loading				Toe (deg per unit)			Camber (deg per unit)		
	X	Y	Z	Moment	Inner	Outer	Average	Inner	Outer	Average
RGP007 REAR	100	0	0	0.001	5.09E-04	3.90E-04	4.50E-04	1.05E-05	1.05E-05	1.05E-05
	0	100	0	0.001	2.01E-05	2.01E-05	2.01E-05	1.91E-04	8.78E-05	1.40E-04
	0	0	100	0.001	2.03E-04	2.02E-04	2.02E-04	8.76E-04	7.49E-04	8.12E-04
	0.001	0	0	100	4.66E-04	4.49E-04	4.57E-04	2.54E-05	2.53E-05	2.54E-05
RGP006 REAR	100	0	0	0.001	8.81E-04	7.65E-04	8.23E-04	4.35E-06	4.29E-06	4.32E-06
	0	100	0	0.001	1.99E-05	1.97E-05	1.98E-05	1.28E-04	2.94E-05	7.88E-05
	0	0	100	0.001	1.87E-04	1.86E-04	1.87E-04	1.14E-03	1.01E-03	1.08E-03
	0.001	0	0	100	1.02E-03	9.99E-04	1.01E-03	1.71E-06	1.72E-06	1.72E-06
% Improvement	X Load				54%	65%	59%	-83%	-84%	-83%
	Y Load				-1%	-2%	-1%	-40%	-100%	-56%
	Z Load				-8%	-8%	-8%	27%	30%	28%
	Moment				75%	76%	75%	-175%	-174%	-175%

The stiffness is improved:

- By 59% for toe due to X loads (Important)
- By 75% for toe due to moments (Important)
- By 28% for camber due to Z loads (Important)

The stiffness is decreased:

- By 56% for camber due to normal force (Stiffness is high anyways)
- By 175% for camber due to moments (Stiffness very is high anyways)
- By 83% for camber due to X loads (Stiffness is high anyways)

Thus, we have made the system significantly stiffer with no weight penalty- exactly the direction we need to go!

6. Rear Hub Design

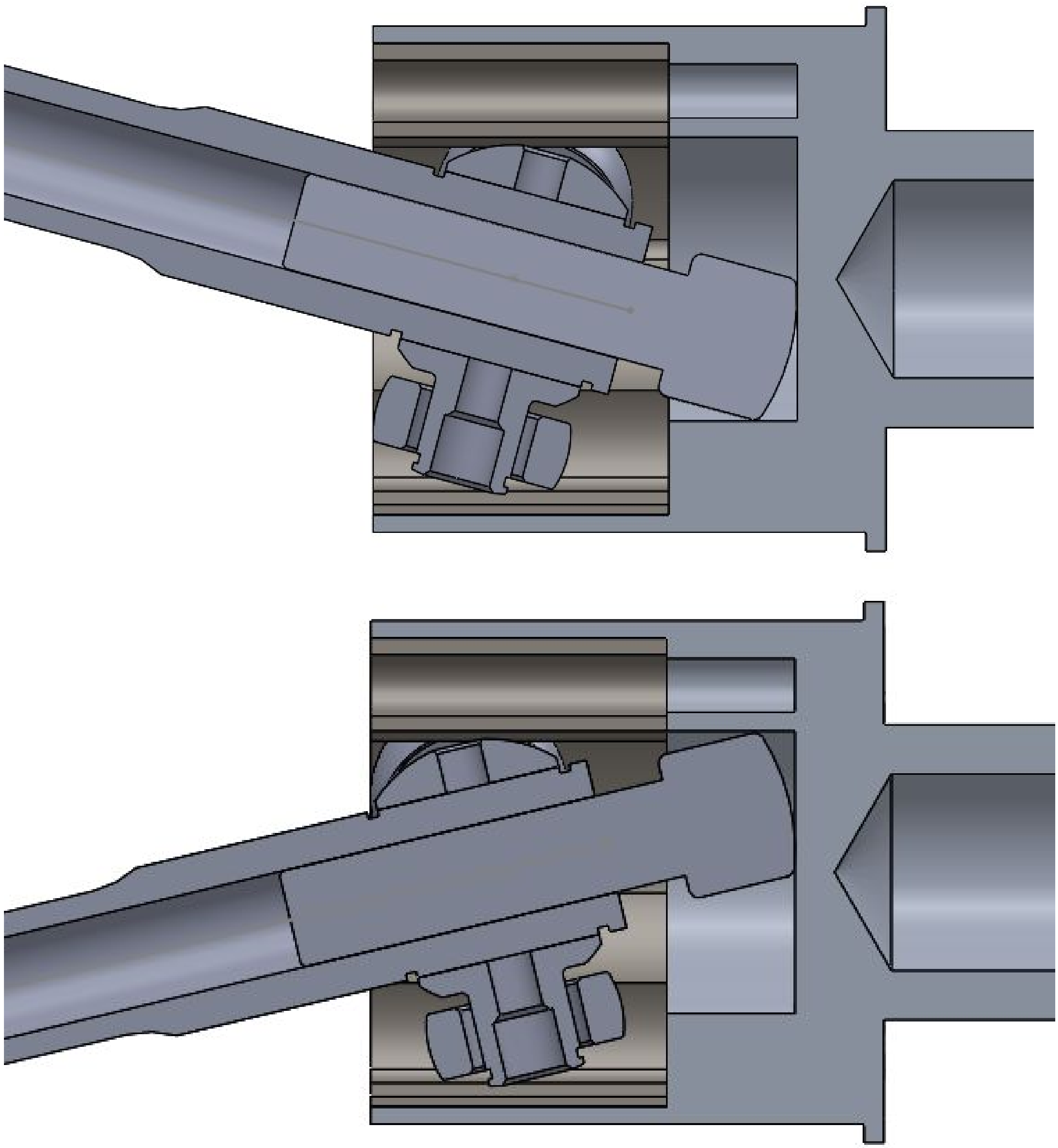
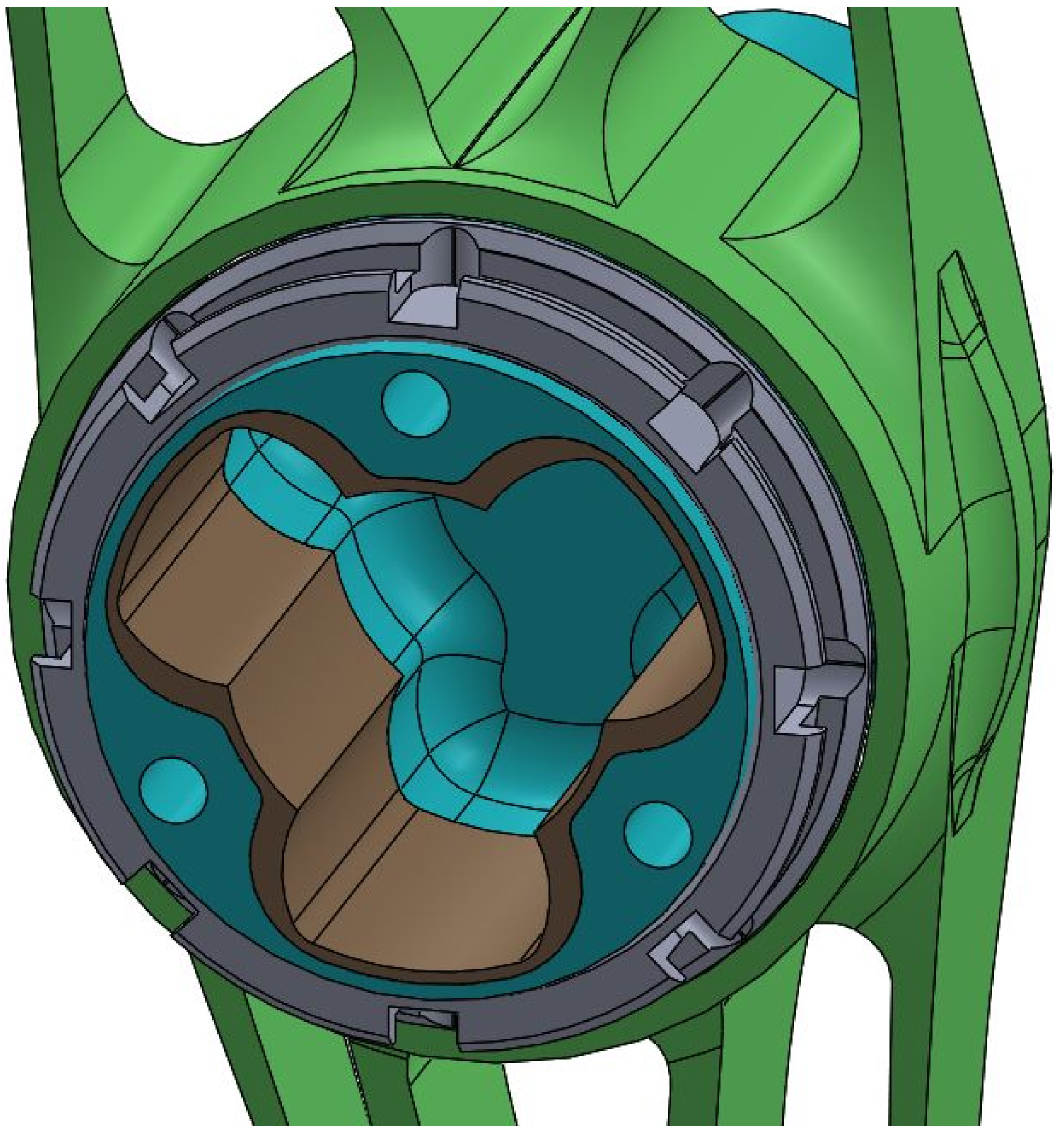


Figure 5.1: Hub design with CV joint and plunging system at +15 and -13.5 degrees of inclination

The hubs can easily accommodate a plunging system with 10 degrees of axle travel in this geometry which is nearly the same as RGP006. Currently, I'm researching to see if ditching the plunging system altogether is viable. This could easily be tested and omitted if desired.

If that is the case, the joint could be downsized, reducing mass and reducing the need for an external nut (the wheel speed disc, retaining plate, and nut would all serve as one part).

Spacer material:
https://www.onlinemetals.com/merchant.cfm?pid=22860&step=4&showunits=inches&id=71&top_cat=0
https://www.profabrication.com/2-625-od-straight-tubing-304-065-per-inch.html
https://www.profabrication.com/2-625-od-straight-tubing-ms-065-per-inch.html



The geometry for the hub insert could be mixed up. Why? Less steel, of course- meaning less mass. The insert is already EDM cut to a pretty high tolerance, why not do all of it that way? This also makes the aluminum threads stronger and the bolts don't need to go all the way through. Easy ways to drop some mass. The weight of the insert and hub is 1.045 lb as shown.

7. Unsprung Mass Testing Rig

By: Thaddeus Hughes
Date: 7/28/2017

This would be an excellent project for a freshman to head up and look very good in design review.

To validate that the new system does reduce compliance, and to measure compliance due to backlash, a rig for stiffness testing should be made. This would apply some remote loading to the wheel center, holding the upright constrained as described in the FEA set-up. If the load is variable, compliance due to backlash and stiffness can be measured separately by performing a linear fit- the initial backlash being the y-intercept and the stiffness contribution being the slope.

This would be a lot of data.

We would like to test:

- RGP004's uprights
- RGP005's uprights
    - With old bearings
    - With new bearings
    - Rears- with new thin-section bearings (make adapters)
    - With various preloads applied to bearings
- **RGP006's uprights (Most important- mostly interchangeable with 005 for these purposes)**
    - **With old bearings**
    - **With new bearings**
    - **Rears- with new thin-section bearings (make adapters)**
    - **With various preloads applied to bearings**
- **RGP007's uprights (when manufactured) (Most important)**
- RGP007 boxed uprights (if this is something we end up doing)

8. Preload Calculations

By: Thaddeus Hughes
Date: 7/29/2017

This document is used to find the nut torque in which the drive pins do not transfer torque and the contact faces of the rim and hub do need seperate (critical for stiffness calcs)

$$ro := \frac{2.5}{2}$$

$$1.25000 \tag{1}$$

$$ri := \frac{1.5}{2}$$

$$0.75000 \tag{2}$$

$$rii := \frac{1.}{2}$$

$$0.50000 \tag{3}$$

$$Ih := evalf\left(\frac{\mathrm{Pi} \cdot ro^4}{4} - \frac{\mathrm{Pi} \cdot rii^4}{4}\right)$$

$$1.86839 \tag{4}$$

$$area_{outer} := evalf\left(\mathrm{Pi} \cdot \left(ro^2 - ri^2\right)\right)$$

$$3.14159 \tag{5}$$

$$area_{inner} := evalf\left(\mathrm{Pi} \cdot \left(ri^2 - rii^2\right)\right)$$

$$0.98175 \tag{6}$$

Distance from neutral axis

$y := ro$

$$1.25000 \tag{7}$$

Moment exerted by cornering(?) forces

$M := 450 \cdot 9$

$$4050 \tag{8}$$

The stress at the furthest distance- without preload

$stress_a := \frac{M \cdot y}{Ih}$

$$2709.55411 \tag{9}$$

$FoS := 1.2$

$$1.20000 \tag{10}$$

Required preload is the amount required to drive the contact stress to zero rather than negative (which can't actually happen) under the cornering force.

$stress_{preload} := stress_a \cdot FoS$

$$3251.46493 \tag{11}$$

$F_{preload} := area_{outer} \cdot stress_{preload}$

$$10214.77833$$

$stress_{inner} := \frac{F_{preload}}{area_{inner}}$

$$10404.68777 \tag{13}$$

Factor to calculate required torque

$k := .2$

$$0.20000 \tag{14}$$

$Torque_{inlbf} := k \cdot F_{preload} \cdot (ro \cdot 2)$

$$5107.38916 \tag{15}$$

$Torque_{ftlbf} := \frac{Torque_{inlbf}}{12}$

$$425.61576 \tag{16}$$

$\mu := .3;$

$$0.30000 \qquad (17)$$

Aluminum on aluminum:1.2 if dry, .3 if lubricated.

$tpi := 12$

$$12 \qquad (18)$$

$$pitch := \frac{1.0}{tpi}$$

$$0.08333 \qquad (19)$$

$$Torque_2 := evalf\left(F_{preload} \cdot \left(\frac{pitch}{2 \cdot \mathrm{Pi}} + \frac{ri \cdot 2 \cdot \mu}{2 \cdot \cos\left(\frac{30 \cdot \mathrm{Pi}}{180}\right)} + \frac{2 \cdot (ri + .125) \cdot \mu}{2}\right)\right)$$

$$5470.73428 \qquad (20)$$

$$Torque_{2\,ftlbf} := \frac{Torque_2}{12}$$

$$455.89452 \qquad (21)$$

For torque cases:

$\mu_{alum-steel} := .61\ (assumes\ dry)$

$$0.61000$$

$$assumes\ dry \tag{22}$$

$Torq_{engine} := 55 \cdot 12$

$$660 \tag{23}$$

$highgear := 5.61$

$$5.61000 \tag{24}$$

$Torq_{enginemax} := Torq_{engine} \cdot highgear$ (assumes all torq goes to one wheel and we dont break traction)

$$3702.60000 \tag{25}$$

$Torq_{brakemax} := 252 \cdot 12$

$$3024 \tag{26}$$

$ro := \frac{2.5}{2}$

$$1.25000 \tag{27}$$

$ri := \frac{1.5}{2}$

$A_{tot} := \mathrm{Pi} \cdot \left(ro^2 - ri^2\right)$

$$3.14159 \tag{28}$$

$FOS := 2$

$$2 \tag{29}$$

Use Torq engine max

$$F_{preload} := evalf\left(\frac{FOS \cdot Torq_{enginemax} \cdot A_{tot}}{\mu_{alum-steel} \cdot \int_0^{2 \cdot \mathrm{Pi}} \int_{ri}^{ro} r^2 \, dr \, d\theta}\right)$$

$$11891.92372 \tag{30}$$

$\mu_{alum} := .3$

$$0.30000 \tag{31}$$

$tpi := 12$

$$12 \tag{32}$$

$pitch := \frac{1.0}{tpi}$

$$0.08333 \tag{33}$$

$$Torque := evalf\left(F_{preload} \cdot \left(\frac{pitch}{2 \cdot \mathrm{Pi}} + \frac{ri \cdot 2 \cdot \mu_{alum}}{2 \cdot \cos\left(\frac{30 \cdot \mathrm{Pi}}{180}\right)} + \frac{2 \cdot (ri + .125) \cdot \mu_{alum}}{2}\right)\right)$$

$$6368.96393 \tag{34}$$

$Torque_{2\,ftlbf} := \frac{Torque}{12}$

$$530.74699 \tag{35}$$

In summary:

- We need 10000 lbf of preload (thus 455 ft*lb) to ensure full contact during a corner
- We need 12000 lbf of preload (thus 530 ft*lb) to ensure that the aluminum surfaces, not drive pins transfer load to the wheels under acceleration from the engine.

9. *Bracket Analysis*

By: Thaddeus Hughes
Date: 8/3/2017

The brackets used to attach the uprights to the suspension have been analyzed. Mesh convergence was ran with respect to stress- a mesh size of 0.05 in for the body and 0.02 in for the contacting areas was selected. Stress singularities occur at the boundary between the compression-only support and the filleted area; this is a common type of stress singularity to see and is disregarded in this analysis. Material used is 7075-T651 aluminum. Running this analysis with 6061-T6 shows that it would work just as well. The brackets are most definitely suitable for use, and could be pocketed further.

Combined toe and A arm bracket:

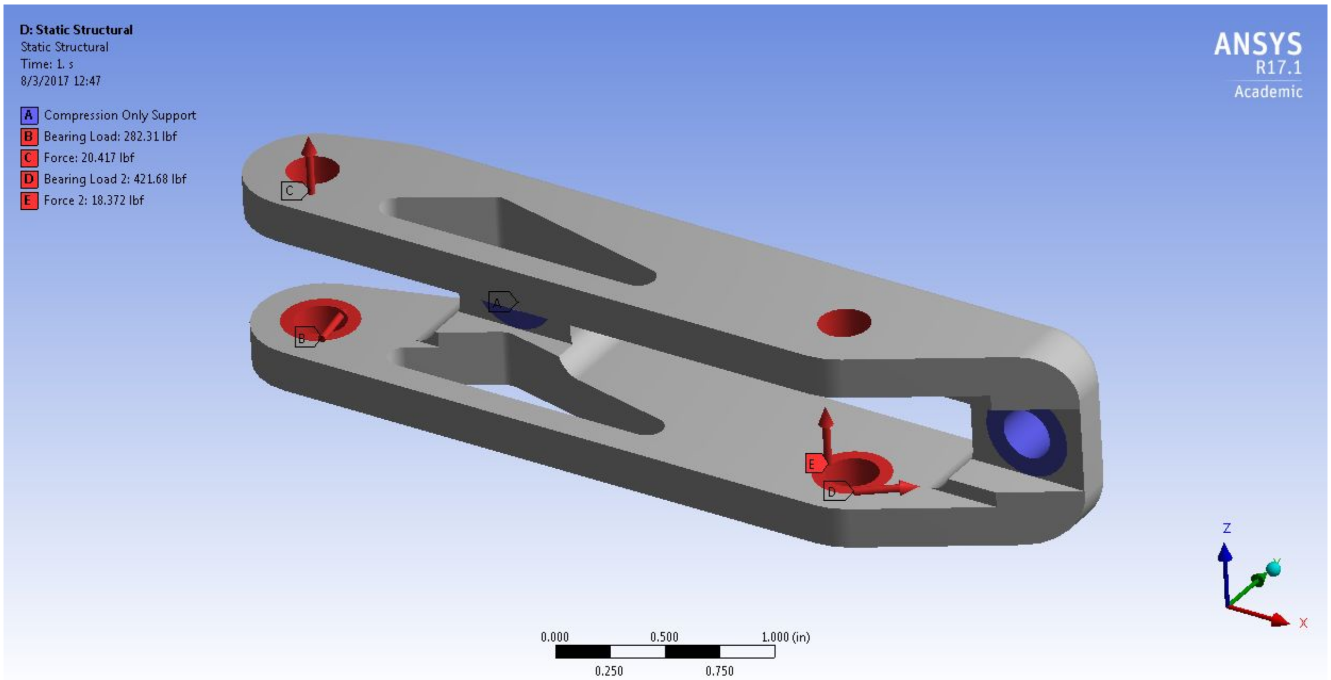


Figure 9.1: ANSYS set-up for the toe and A arm bracket

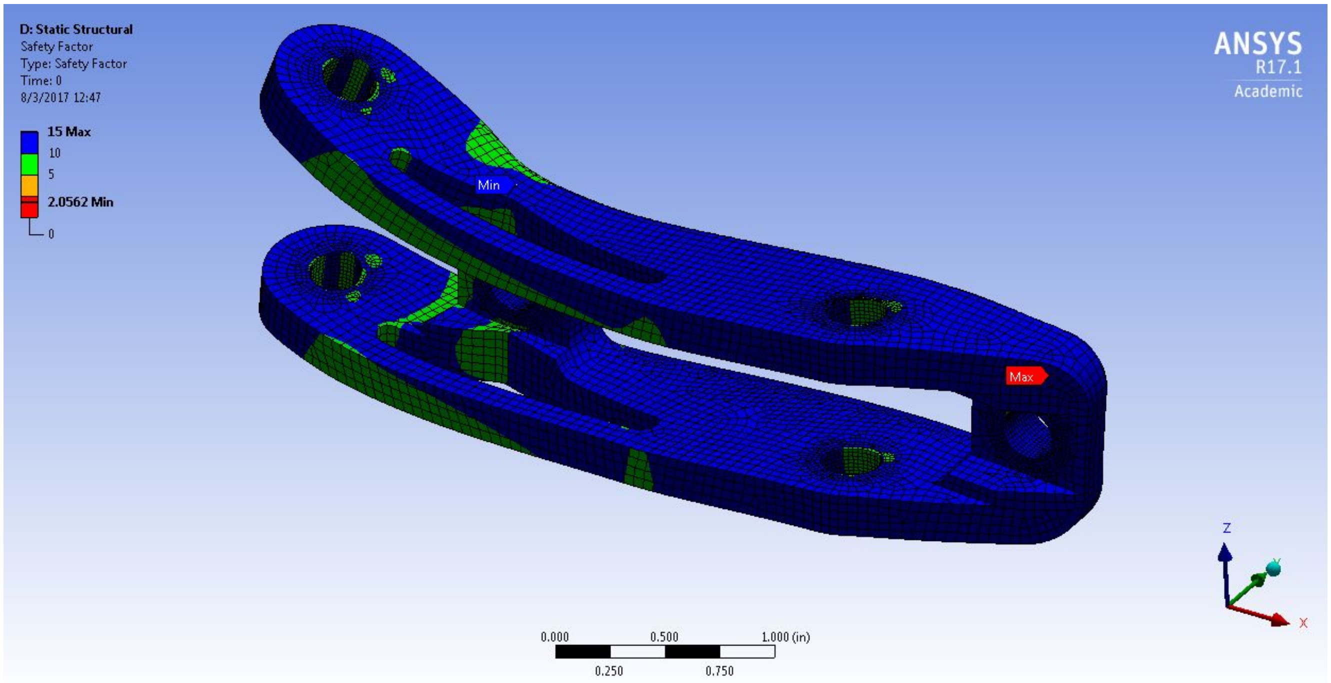


Figure 9.2: ANSYS results for the toe and A-arm bracket. FOS against 1e8 zero-based cycles: 2.0562

Toe Z	Toe Y	Toe X	A Y	A X	A Z	FOS
-0.13818	-1.88708	0.29921	-12.4	-5.59	-0.912	15
2.01	27.45	-4.35	262.8	-16.6409	19.2437	9.830804
-2.8699	-39.19	6.2145	-217.024	-21.847	-15.8916	3.025012
-13.1451	-179.516	28.4637	-197.064	-229.25	-14.43	2.904824
20.41692	278.825	-44.2098	250.8938	338.9199	18.3717	2.056236

Figure 9.3: ANSYS results for toe and A-arm bracket under various load conditions, tabulated.

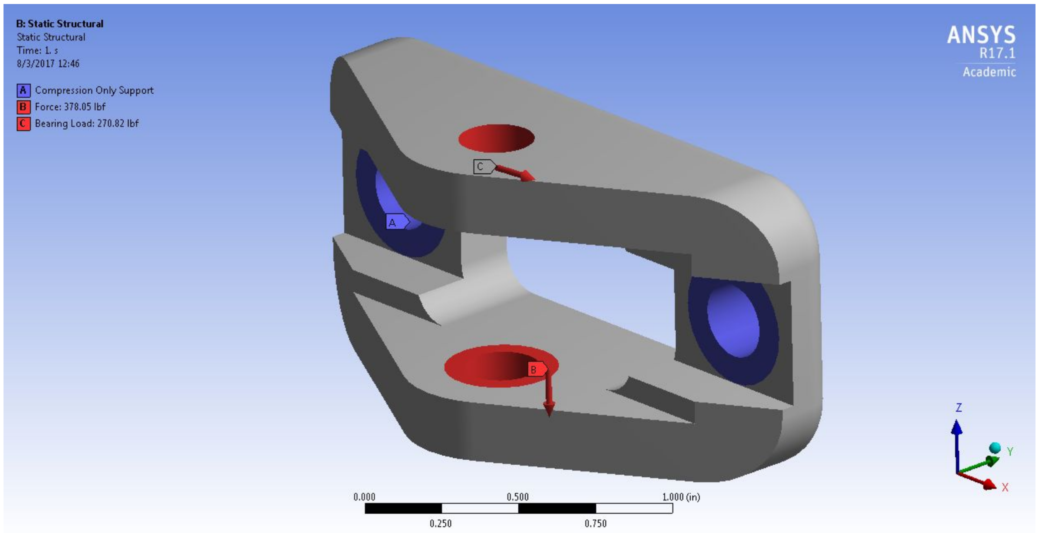


Figure 9.4: ANSYS set-up for the single A-arm bracket.

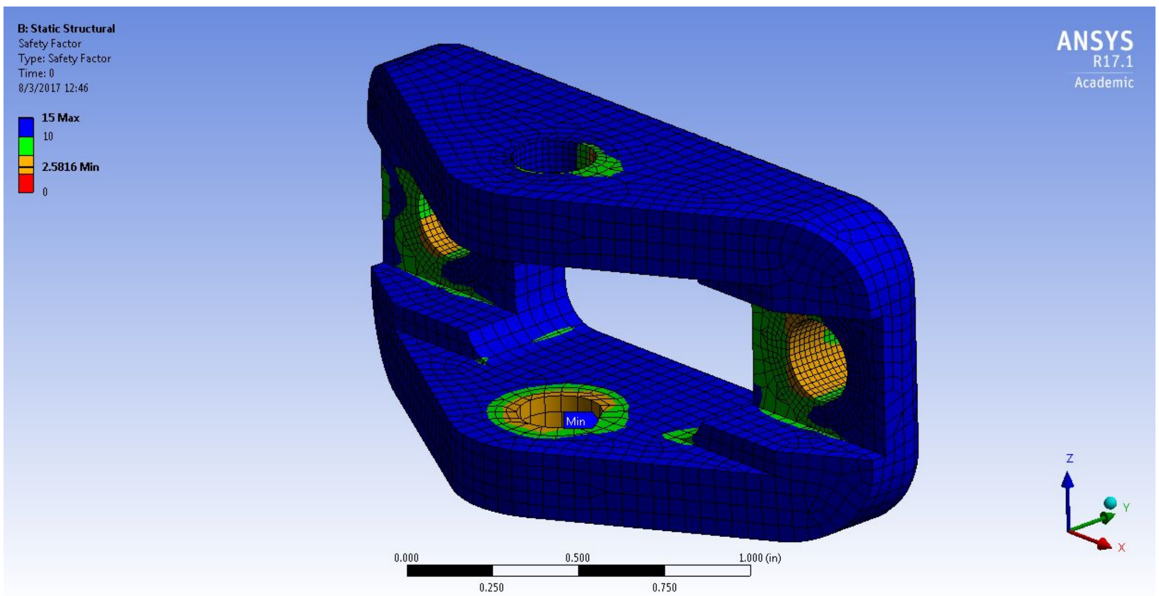


Figure 9.5: ANSYS results for the single A-arm bracket.

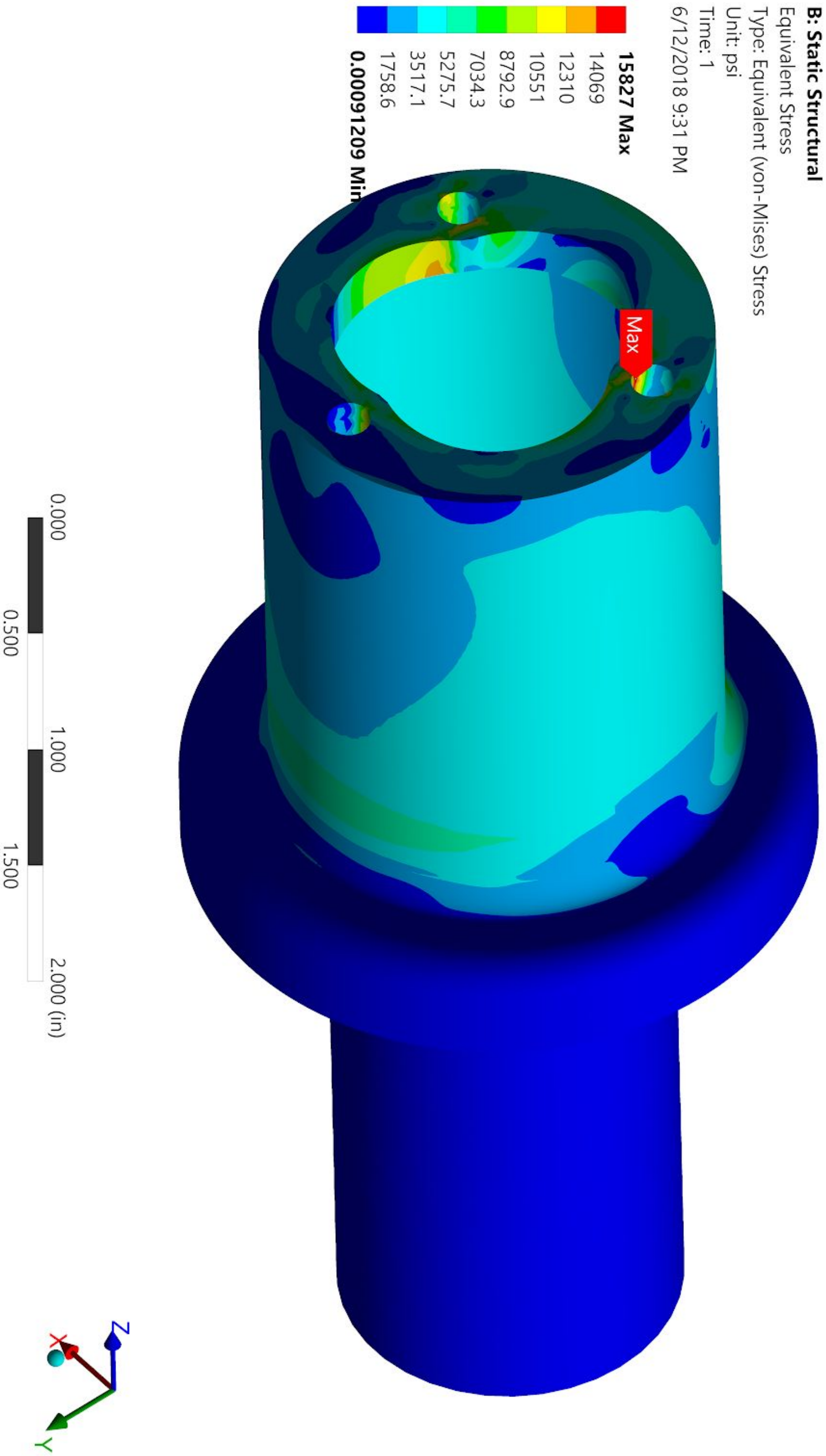
X (lbf)	Y (lbf)	Z (lbf)	FOS
5	-138	10	7.74
1	-219	566	2.71
-4	-151	398	3.85
-149	-235	17	4.34
269	-378	27	2.58

Figure 9.6: ANSYS results for single A-arm bracket under various loading conditions, tabulated.

Requirement to turn steering wheel with one hand:

Steering wheel diameter: 4 in
Thereforce, input torque: 900 in*lbf
Rack radius: 0.75 in

Therefore, force in tie rod: 1200 lbf
Steering lever arm: 3 in
Thereforce, torque on assy: 200*2*7 in*lbf
Bearing c-c: 2.12 in
Therefore, force on each bearing: 1700 lbf



B: Static Structural
Equivalent Stress
Type: Equivalent (von-Mises) Stress
Unit: psi
Time: 1
6/12/2018 9:31 PM
15827 Max
14069
12310
10551
8792.9
7034.3
5275.7
3517.1
1758.6
0.00091209 Min
Max
0.000
0.500
1.000
1.500
2.000 (in)
Z
X
Y