BRONCO MOTORSPORTS

California State Polytechnic University, Pomona
Formula SAE

2024-2025 Design Binder

Hubs and Uprights

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BM 25 Full Vehicle Objectives

Overall Vehicle Goals:

- 668 points minimum to be top 10.
- Reliability is a top priority. A low performing car that is reliable can be improved to increase performance, but a low reliability car that is high performing is much harder to "fix".
    - Finish endurance race
- Simplicity is also a must. Considering the timeline, resources, and green-ness of the team, simplicity will make the goal the most achievable while inducing lesser risk.
- Rules passing car is necessary. Each year, not passing tech sets us back tremendously. This leaves out the chance to use the test track, make setup changes, and double check components and hardware.

Vehicle Dynamics Goals:

- Lateral acceleration
    - 1.9 lateral Gs (transient)
    - 1.36 Gs (steady state)
    - Top 10 in skidpad event
- Achieve given Gs on a G-G diagram consistently

Purpose of Subsystem

Hubs:

- Establish a fixed point for the wheel, brake rotor, and wheel speed sensor gear
- Connects the wheel to the suspension assembly and drivetrain
    - Wheel bearing allows wheel to rotate relative to the car
- Wheel nut provides axial constraint of the wheel during cornering

Uprights:

- Houses the hub assembly
    - Constrains wheel bearings
- Connects hub to the suspension control arms and steering tie rod by serving as an intermediate mounting point to the wheel
    - Transfers forces acting on the wheel to the shock
    - Allows for wheels to turn
- Dictates key suspension alignment and steering parameters
    - Toe
    - Camber
    - Caster
    - Ackermann
- Has tabs to mount tire temperature, brake temperature, and wheel speed sensor

FSAE 2025 Rules

V.3.2.3 Steering systems must use a rigid mechanical linkage capable of tension and compression loads for operation

V.3.2.4 The steering system must have positive steering stops that prevent the steering linkages from locking up (the inversion of a four bar linkage at one of the pivots). The stops:

a. Must prevent the wheels and tires from contacting suspension, bodywork, or Chassis during the track events

b. May be placed on the uprights or on the rack

V.4.2 Wheel Attachment

V.4.2.1 Any wheel mounting system that uses a single retaining nut must incorporate a device to retain the nut and the wheel if the nut loosens.

A second nut (jam nut) does not meet this requirement

V.4.2.2 Teams using modified lug bolts or custom designs must provide proof that Good Engineering Practices have been followed in their design.

V.4.2.3 If used, aluminum wheel nuts must be hard anodized and in pristine condition.

Background/Theory

The fundamental theory and principles that provide the basis of analysis for the subsystem is vehicle dynamics. This systematic exploration of forces, with a focus on lateral and longitudinal load transfers, is undertaken to optimize the design and performance of the upright subsystem within the BM25 project. As an integral component of the BM25 Vehicle Dynamics system, the system lead has provided maximum loading scenarios, which have been utilized by each subsystem in designing their components. These loading conditions have been derived through an extensive examination of testing data, utilizing Motec i2. Subsequent revisions consider data from various testing days, events, and competitions, culminating in the selection of data from the 2024 Endurance competition event.

Vehicle Parameters		
Long. Accel	-1.5	g
Center of Gravity Height	12.63	in
Lateral Accel	0	g
Wheelbase	60.23	in
b	28.31	in
a	31.92	in
Load Transfer(- +)	**-217.03553**	**lb**
Vertical Load On Rear	**141.76447**	**lb**
Vertical Load On Front	**531.33553**	**lb**

Vehicle Parameters		
Long. Accel	1.60	g
Center of Gravity Height	12.63	in
Lateral Accel	2.00	g
Wheelbase	60.23	in
b	28.31	in
a	31.92	in
Front end Deflection	0.684	in
Rear end Deflection	-0.386	in
Pitch	1.017	deg
Pitch Gradient	0.636	deg/g
Load Transfer(- +)	**231.50**	**lb**
Vertical Load On Rear	**590.30**	**lb**
Vertical Load On Front	**82.80**	**lb**

Lateral Load Transfer Calculations

The analysis begins by focusing on lateral load transfer, a crucial aspect of understanding vehicle dynamics. Lateral load transfer refers to the redistribution of forces across the vehicle during cornering or maneuvering, which affects the weight distribution on each wheel and, consequently, the stability and handling characteristics of the vehicle.

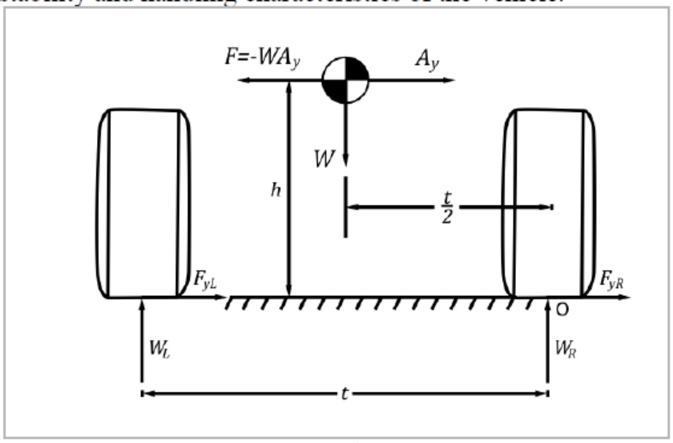


Figure 3. Free body diagram of a car, rear view. (MILLIKEN & MILLIKEN, Race Car Vehicle Dynamics – Adapted)

Using Figure 1, we're able to calculate the theoretical lateral load transfer under a specified amount of lateral acceleration[in G's]. Applying a moment about point O, we were able to come up with the equation.

$$WA_y h + W\left(\frac{t}{2}\right) - W_L(t) = 0$$
$$\Rightarrow \; W_L = \frac{W}{2} + WA_y\left(\frac{h}{t}\right)$$

We can use this simple method to determine the vertical load on the either right or left side tires. Where W is the weight of the car, Ay is the lateral acceleration in G's, h is the CG height and t is the track width. However, for our application, there are a bit more parameters we must consider when analyzing this form of lateral load transfer. The other parameters that we must consider are front roll stiffness, front roll axis CG, rear roll stiffness, front roll center height, etc. The application of these parameters lies in the analysis of Roll Load Transfer Geometry.

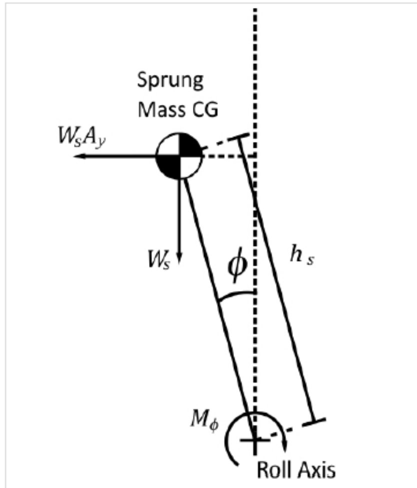


Figure 4. Roll Load Transfer Geometry

The derivation of the applicable formulas are based on key definitions such as the moment equilibrium about the roll axis:

$$M\emptyset - W_s h_s sin\emptyset - W_s A_y h_s cos\emptyset = 0$$

Where M is the roll resistance moment, and $W_s A_y h_s cos\emptyset$ is the roll moment. The term $W_s h_s sin\emptyset$ is a gravity component that arises due to the sprung CG being shifted to the side when the chassis rolls. Solving for and dividing by Ay we obtain the *roll sensitivity* to lateral acceleration of the car, i.e. the amount of body roll per unit of lateral acceleration:

$$\frac{\emptyset}{A_y} = \frac{W_s h_s}{K_{\emptyset F} + K_{\emptyset R} - W_s h_s}$$

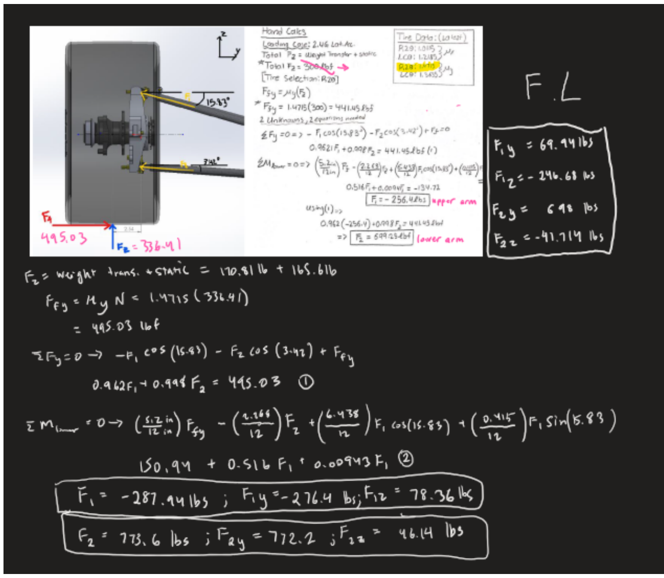
Where KF is the roll stiffness of the front and KR is the rear roll stiffness. To simplify the extent of this derivation, I will be skipping ahead to the final derivation of the lateral load transfer.

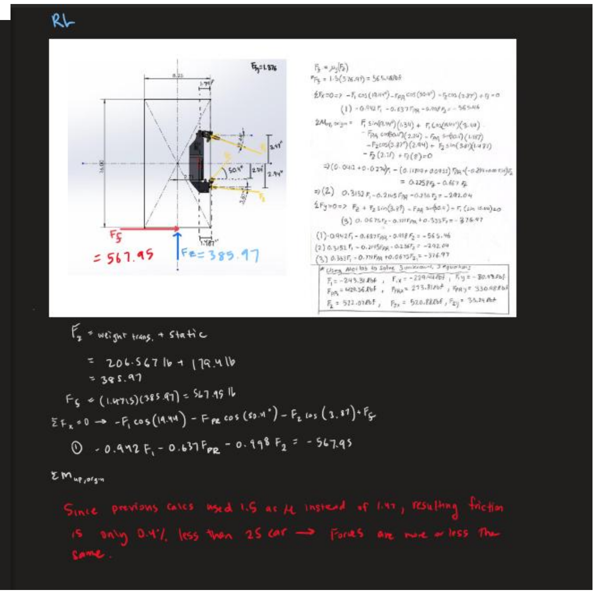
$$Weight\ Transfer = \left[\frac{W}{t_f} * \frac{Z_{RC} * K_{\emptyset F}}{K_{\emptyset F} + K_{\emptyset R}} + \frac{t_f}{2 * Wheelbase} * Z_{rc}\right] * Lateral\ Acc$$

FBD Diagrams + Hand Calculations for Upright Forces

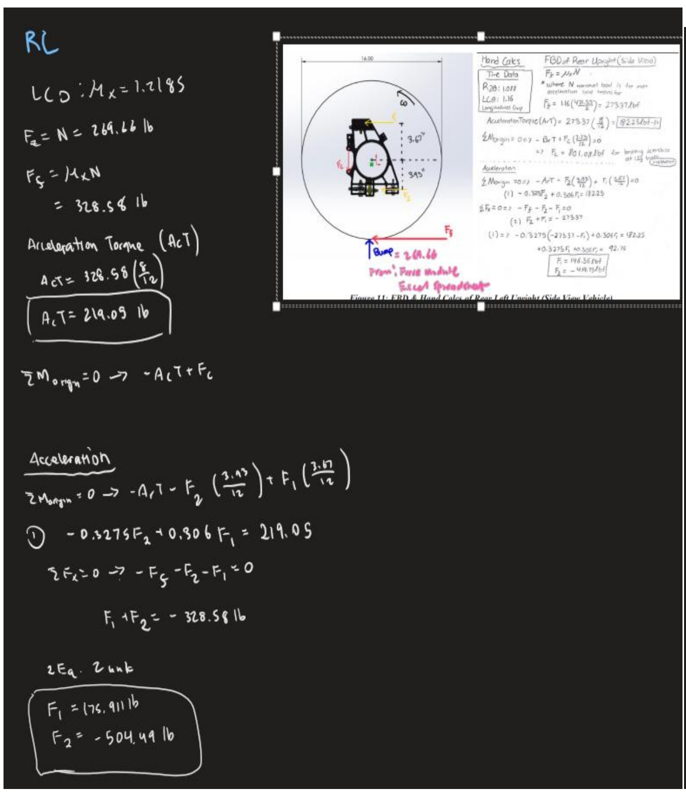
Although the IMU gets forces for us based on the vehicle’s acceleration; to double check these values, we could perform vector statics calculations to verify the IMU measurements are correct/within the ballpark of the actual values.

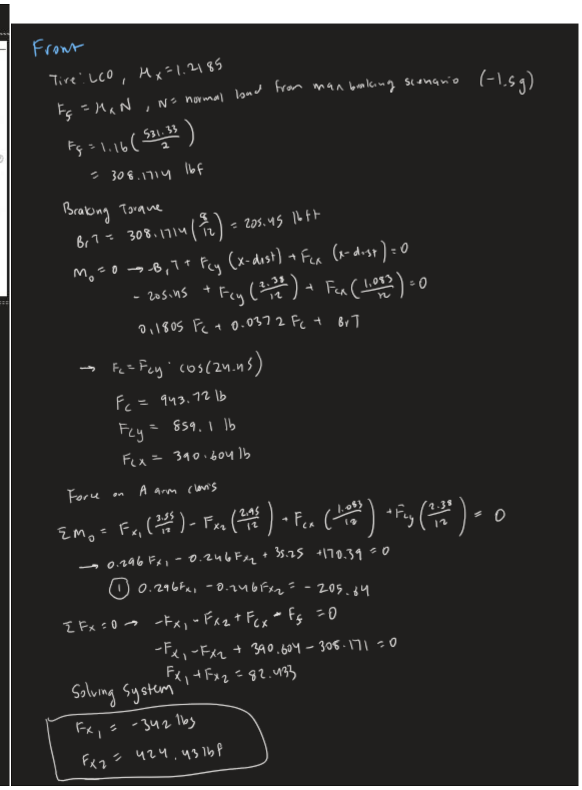
Pure cornering load case:





Pure Acceleration/Braking





Combined Braking/Acceleration + Cornering

FR, (R20: μ_y = 1.4715, μ_x : 1.0115)

F_z = weight trans. + Static

$= +256.23_{\text{LAT WT.}} + \frac{430.5}{2}$ long. WT + Static

$F_z = 471.48$ lbf

$F_{fy} = 1.4715(471.48) = 693.782$

$F_{fx} = 1.0115(471.48) = 476.902$

Eq

① $\Sigma F_x = 0 \rightarrow R_{ux} + R_{lx} + F_{fx} = 0$

② $\Sigma F_y = 0 \rightarrow R_{uy} + R_{ly} - F_{fy} - R_s = 0$

③ $\Sigma F_z = R_z + F_z = 0$

$\Sigma M_{Lx} = 0 \rightarrow -(5.2)F_{fy} + (2.764)F_z - (6.438)(R_{uy}) - (0.415)R_z - 2.95\,R_y = 0$

④ $\rightarrow -6.438R_{uy} - 2.95R_s = 1543.8$

⑤ $\Sigma M_{Ly} = 0 \rightarrow -(5.2)F_{fx} + 0.283F_z + 6.438(R_{ux}) = 0$

$\Sigma M_{Lz} = 0 \rightarrow -2.871(R_s) + 0.283(F_{fy}) - 2.264(F_{fx}) + 0.415(R_{ux}) - 0.48(R_{uy})$

⑥ $\rightarrow -0.4796R_{cy} - 2.871R_s = 574.574$

$R_{Lx} = -541.41$ lbf $R_{uy} = -88.2326$ lbf

$R_{ux} = 364.51$ lbf $R_s = -330.77$ lbf

$R_{ly} = 782.015$ lbf $R_z = -471.48$ lbf

Brake Torque

Brake = BrT

$BrT = (\frac{d}{2})(F_{fx})$

$BrT = 317.935$

$\Sigma M_o = -B_rT + F_{cy}(x\,dist) + F_{cx}(x.dist) = 0$

$-317.935 + 0.1805F_c + 0.0372F_c = 0$

$F_c = 1460.43$ lbf

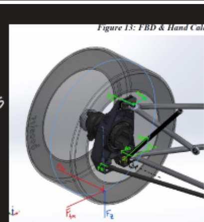
$F_{cx} = 604.476$ lbf

$F_{cy} = 1329.162$ lbf

RL

Acceleration + cornering

Tire: R20 $\rightarrow \mu_y = 1.4715$; $\mu_x = 1.0115$; LC0: $\mu_x = 1.2185$, $\mu_y = 1.3635$



$LLT = 159$ lb

$Fz\_st = 162$ lb

$LongLT = 474.55/2$

$F_y = \mu N$

$F_x = \mu_x N$

F_z = weight trans + static

$= 159 + 179 + \frac{474.55}{2}$

$= 575.275$

$F_{fy} = 1.3635(575.275)$

$= 784.387$ lbf

$F_{fx} = 1.2185(575.275)$

$= 700.973$

Equilibrium

A-Arm: lower, upper; Push rod

$\Sigma F_x = 0 \rightarrow F_{fx} + R_{Lx} + R_{ux} = 0$

$\Sigma F_y = 0 \rightarrow -F_{fy} + R_s + R_{uy} + R_{ly} - R_{PR} \cdot \cos(50.4) = 0$

$\Sigma F_z = 0 \rightarrow F_z + R_{PR}\sin(50.4) = 0$

$\Sigma M_{Lx} = 0 \rightarrow -(5.25)(F_{fy}) + (3.15)(F_z) - 6.617(R_{uy})$

$\Sigma M_{Ly} = 0 \rightarrow (6.369)R_{ux} - (1.304)F_z - (5.25)(F_{fx}) + (1.291)R_{PR}(\sin 50.4) = 0$

$\Sigma M_{Lz} = 0 \rightarrow [R_{PR}\cos(50.4)(1.291)] - R_s(3.697) + (0.5538)R_{uy}$

$-F_{fx}(3.15) + F_{fy}(1.304) = 0$

6 Eq. 6 unknowns

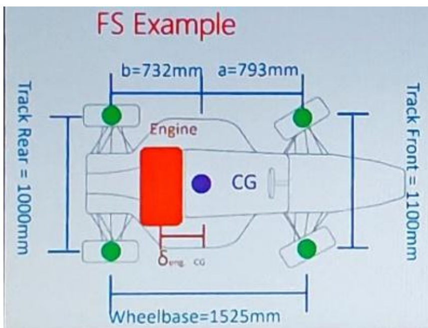
$R_{lx} = -1513.1$ lbf $R_{uy} = -348.5$ lbf

$R_{ux} = 812.2$ lbf $R_s = -538.8$ lbf

$R_{ly} = 1196.4$ lbf $R_{pr} = -746.1$ lbf

Yaw inertia Calculations

Inertia is the resistance to changing direction of motion. A higher yaw inertia would mean that the vehicle would change direction slower which is a negative affect on vehicle handling. The uprights contribute a lot to the vehicle's moment of inertia since they are located at some of the furthest points away from the vehicle's center of gravity.



δeng. - CG	200.0	mm
NSM	9.0	kg
Engine Mass	50.0	kg
Distance To Front NSM	1.0	m
Distance To Rear NSM	0.9	m

Contribution to the Yaw Inertia of 1 Front NSM	8.4	kg*m^2
Contribution to the Yaw Inertia of 1 Rear NSM	7.1	kg*m^2
Contribution to the Yaw Inertia of ALL 4 NSMs	30.9	kg*m^2
Contribution to the Yaw Inertia of the Engine	2.0	kg*m^2

$a = 31.92$ in $\quad c_f = 49.986''/2 = 24.993''$

$b = 28.31$ in $\quad c_r = 47.92''/2 = 23.96''$

$d_f = \sqrt{a^2 + c_f^2}$

$d_r = \sqrt{b^2 + c_r^2}$

$d_f = \sqrt{31.92^2 + 24.993^2} = 40.54''$

$d_r = \sqrt{28.31^2 + 23.96^2} = 37.088''$

* m includes hub Assem.

$I_f = m_f d_f^2$

$= \left(\frac{3.2}{32.2} \text{ lbs}\right)(40.54'')^2$

$= 163.33$ lbm-in^2

$I_r = m_r d_r^2$

$= \left(\frac{4.47}{32.2}\right)(37.088)^2$

$= 190.95$ lbm-in^2

4 corners

$\Sigma I_{total} = 708.56$ lbm-in^2

don't divide by 32 ?

Generative Design

$I_f = \left(\frac{2.86}{32.2}\right)(40.54^2)$

$= 145.97$ lbm-in^2 / 4700.386 lbm-in^2

$I_r = \left(\frac{4.44}{32.2}\right)(37.088^2)$

$= 189.668$ lbm-in^2

$\Sigma I_{total} = 671.285$ lbm-in^2

708.56 vs. 671.289

= 5.6% increase in total yaw inertia

- For entire assembly (tires, hnu, wheels, brakes)

$m_f = 15.31$ lbs (regen.) $+ 1.227 = 16.537$ lbs

$m_r = 14.83$ lbs $+ 0.8944 = 15.7244$ lbs

Non-regen

$m_f = 16.877$ lbs

$m_r = 15.7544$ lbs

$$I_f = \left(\frac{16.877}{32.2}\right)(40.54)^2 = \boxed{861.404 \text{ lbm-in}^2} \quad \text{vs.} \quad I_f = \left(\frac{16.537}{32.2}\right)(40.54)^2 = \boxed{844.05 \text{ lbm-in}^2}$$

$$I_r = \left(\frac{15.75744}{32.2}\right)(37.088^2) = \boxed{673.126 \text{ lbm-in}^2} \quad \text{vs.} \quad I_r = \left(\frac{15.7244}{32.2}\right)(37.088)^2 = \boxed{671.715 \text{ lbm-in}^2}$$

I_f % increase: 2.056%

I_r % increase: 0.21%

$$\% \text{ increase} : \frac{\text{regen} - \text{nrml}}{\text{nrml}}$$

$$\left.\begin{aligned} I_{total}(\text{regen}) &= 3{,}031.53 \text{ lbm-in}^2 \\ I_{total}(\text{nrml}) &= 3069.06 \text{ lbm-in}^2 \end{aligned}\right\} \% \uparrow : \boxed{1.23799\%}$$

New weight v2

for entire outboard assembly

$m_f = 16.877 + 0.2 = 17.077$ lb

$(1.33 - 1.13)$ lb

outboard assembly mass total

$$I_f = \frac{17.077}{32.2}(40.54)^2$$

$$\boxed{I_f = 871.62 \text{ lb-in}^2}$$

$$\boxed{\% \uparrow = 1.186\%}$$

Wheel Nut Torque

Torque Calc

Thursday, December 26, 2024 11:21 PM

1. Determine the Required Preload (F_p):

- The preload is the axial force required to properly clamp the components together.
- It is typically specified as a percentage of the bolt's proof load (F_{proof}).

$$F_p = k \times F_{proof}$$

Where:

- k is a percentage (commonly 70–90%).
- F_{proof} is the proof load, calculated as:

$$F_{proof} = A_t \times \sigma_{proof}$$

A_t = Tensile stress area of the bolt (found in standard tables).
σ_{proof} = Proof strength of the bolt material (MPa or psi).

2. Calculate the Design Torque (T):

Use the torque-tension relationship:

$$T = k \times F_p \times d$$

Where:

- T = Torque (Nm or lb-ft).
- k = Torque coefficient (also called nut factor, typically 0.15 for lubricated threads or 0.2 for dry threads).
- F_p = Preload (N or lb).
- d = Nominal diameter of the bolt (m or in).

For 7075 T6 Al

Sigma_proof = [illegible] MPa 430 MPa

D = 45 mm --> 0.045 m 1.772 in
Pitch = 3 TPI
F_proof = 1.21269 x 10^3 kN
K = 0.2 (unlubricated)
F_p = 224.34 kN

Torque = 515 Nm
= 380 ft-lb

$$F_p = \frac{0.75 \cdot \sigma_p \cdot \pi \cdot (D - 0.9382P)^2}{4} \doteq 40{,}339.2 \text{ lbf}$$

$\sigma_p \approx 430$ MPa or 63000 psi

$D = .045$ m $\rightarrow$ 1.772 in

$p = 3$ TPI

$$\text{Torque} = kDF_p$$
$$= (0.2)(1.772 \text{ in})(40{,}339.2 \text{ lbf})$$
$$= 1{,}191.35 \text{ lbf} \cdot \text{ft}$$

(Rear)

$\mu_{tire} = 1.4715$

Lateral Force on Hub

$$\frac{\text{Nrml load} \cdot \mu \cdot r_{whl}}{r_{hub}}$$

$$\frac{319.52 \text{ lb} \cdot 1.4715 \cdot \left(\frac{10}{12}\right) \text{ ft}}{\left(\frac{1.77}{12}\right) \text{ ft}} = 2{,}656.35 \text{ lb}$$

Nut diameter: 1.77 in (nominal)

Pitch: 3 mm

Nut factor: 0.25

$$T = kDF$$
$$= (0.25)\left(\frac{1.77}{12}\right) \text{ ft} \, (2656.35 \text{ lb})$$

$$\boxed{\begin{array}{l} T = 97.84 \text{ ft-lb} \\ \text{FOS of } 1.2 \rightarrow 115 \text{ ft-lb} \end{array}}$$

Design Procedure

<u>Uprights</u>

1) Obtain vehicle target weight, CG location, IMU forces
2) Perform hand calculations with the given forces to determine acting forces on uprights
3) Run OptimumG simulation on vehicle forces, verify with hand calculations
4) Compare new vehicle goals with previous vehicle goals; analyze whether the old uprights design(s) meet the new goals
5) Model uprights that I believe better suits the new vehicle goals
6) Conduct FEAs, fatigue analysis to validate designs
7) Compare new and old designs objectively using a design matrix
8) Go forward with manufacturing using chosen design

<u>Hubs</u>

1) Do uprights procedure 1-4
2) Conduct FEAs and fatigue analysis simulations to verify whether old hub design needs to be changed
3) Simulations showed a high FOS and fatigue life, so old hub design could safely be used

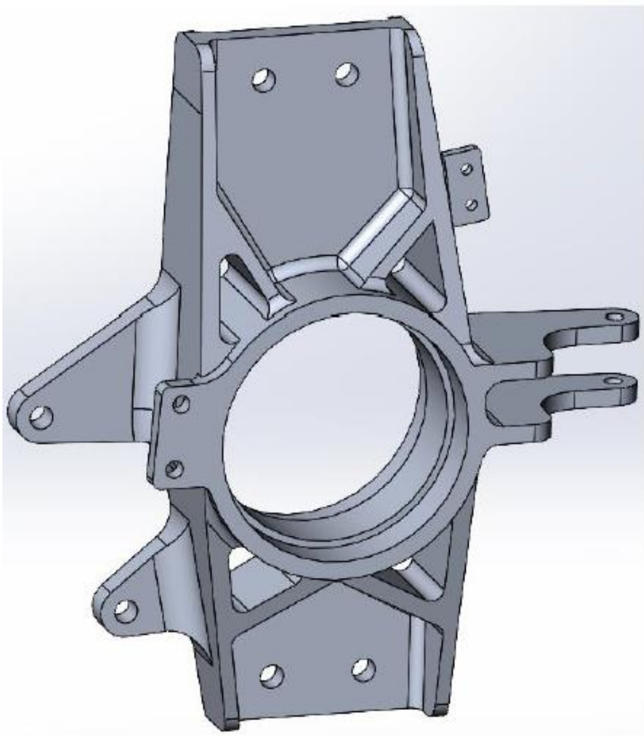
<u>Wheel Nuts</u>

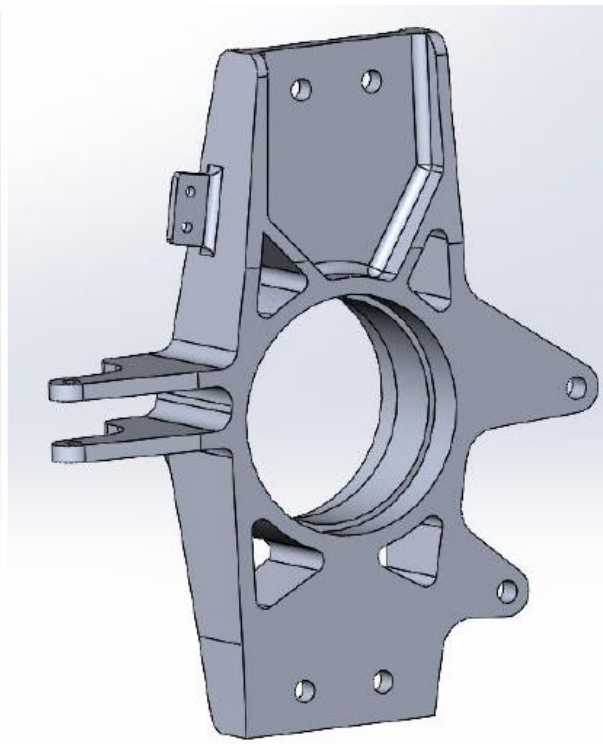
1) Identify issues of previous year's nuts
    a. Holes went past the hub holes when fully torqued
    b. Holes tended to not align when torqued down
2) Brainstorm and implement new changes to fix the issues
    a. Move holes farther outboard
    b. Increase hole diameter for higher chance of alignment
    c. Make holes ovular to increase window to alignment
    d. Used a number of holes on nut which was not a multiple of the number of holes on the hub

Design Revisions

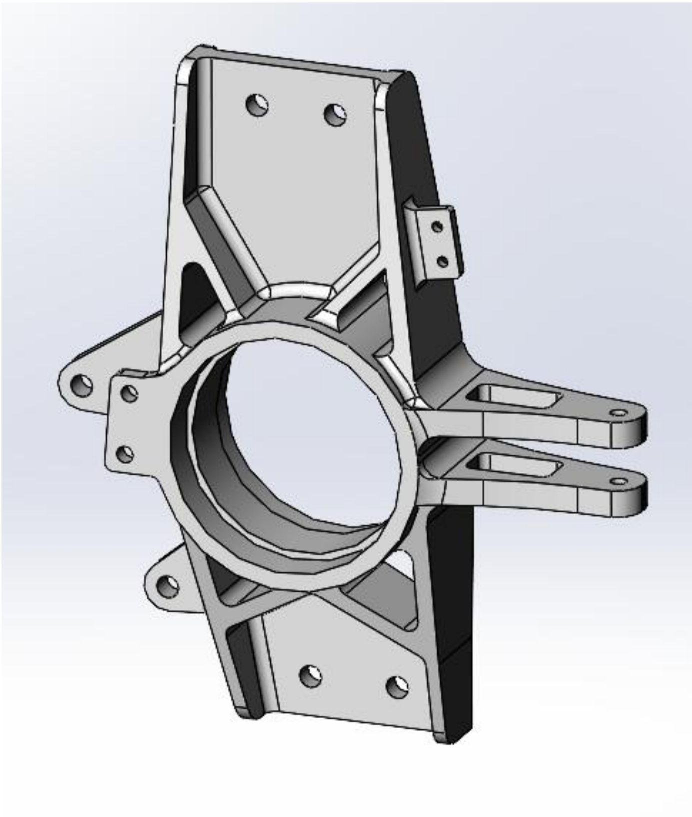
Front Upright

Original Design From Previous Year



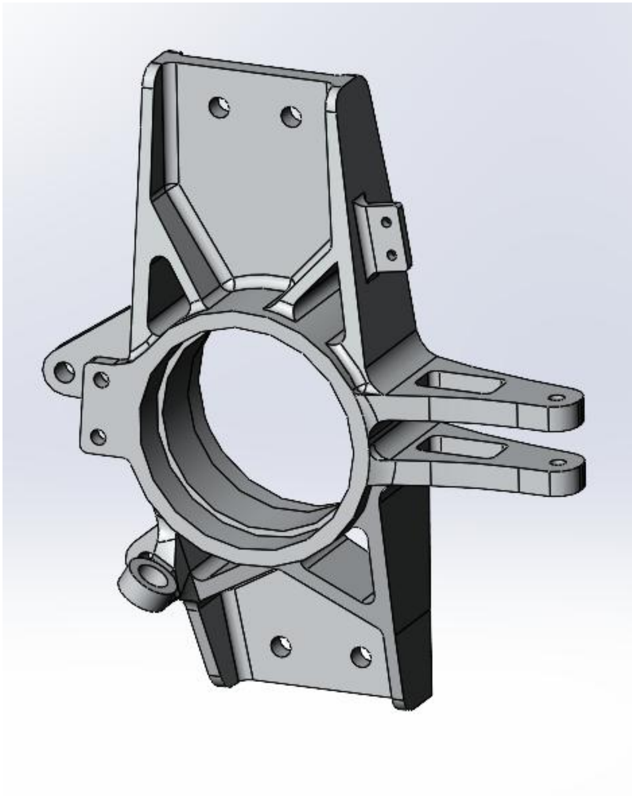
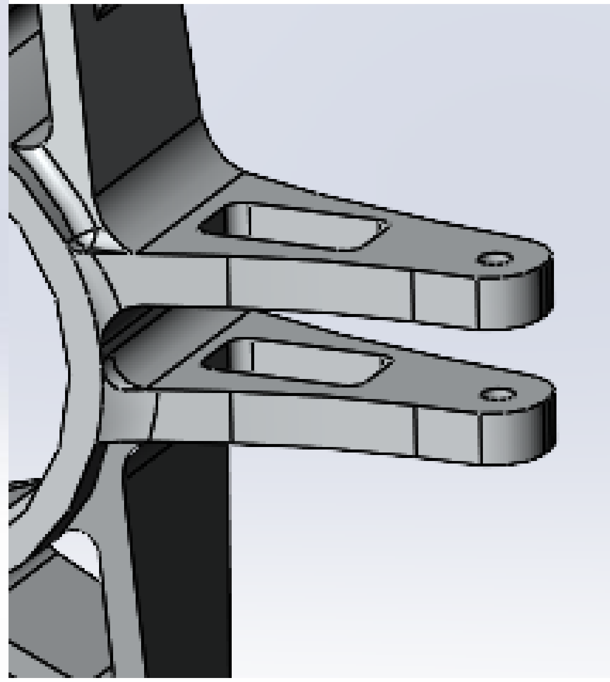


Revision 1: Updated Steering Pickup point location + Steering arm geometry



- Relocated Steering PuP to new location determined by steering captain
- Reshaped steering arm to allow for full steering wheel lock range
    - Previous shape had to be sanded down to allow for full lock to be achieved
- Did FEA to determine how much weight could be cut from the new arm with minimal change to previous deflection and factor of safety

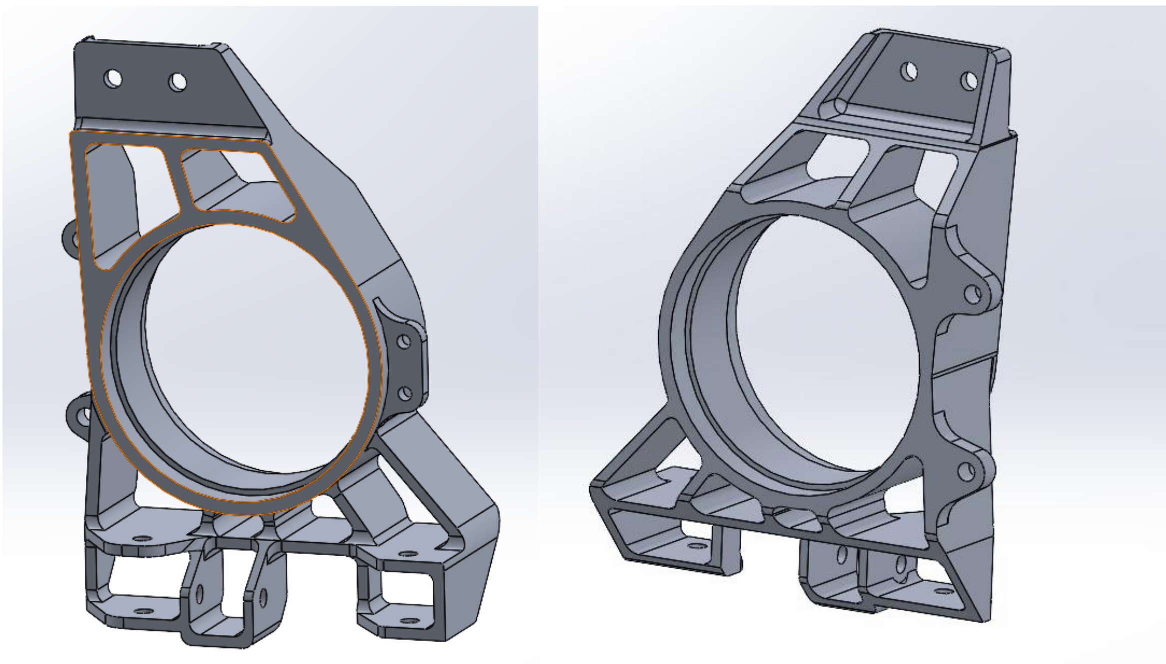
Revision 2: Added Wheel Speed Sensor Mount onto the Upright body

- Added wheel speed sensor mounting point to upright
- Allows for accurate and refined sensor placement, as opposed to welding a sensor mount onto the A-arm
- Needs 5-axis CNC to manufacture, which machining sponsor has

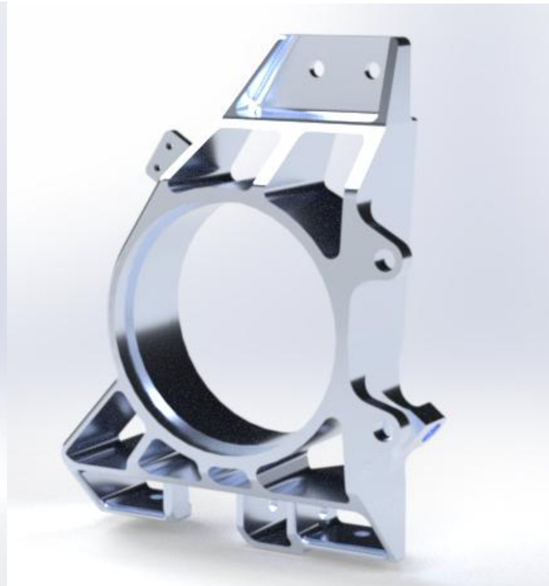
Rear Upright

Original Design



- Lacks sensor mounting points
- Had areas of high stress concentration according to FEA

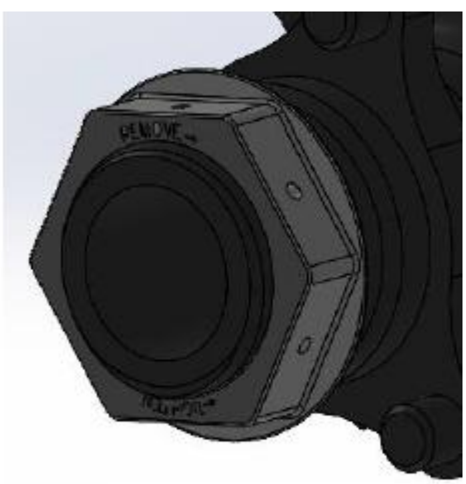
Final Revision

- Added wheel speed sensor mount directly on upright
- Added tire temperature sensor mount
- Made geometry smoother in areas with high stress concentration

Wheel Nut

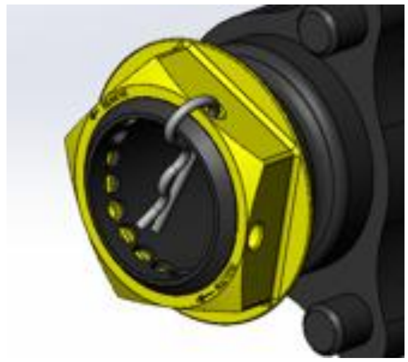
Original



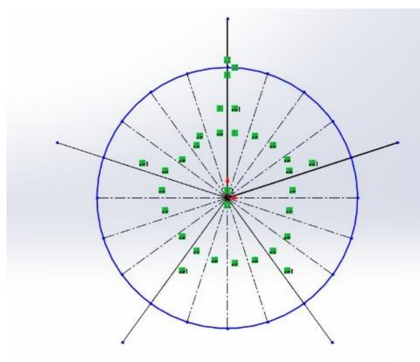
- Even (6) amount of safety pin holes on the nut meant two holes both align with one of the 20 holes on the hub simultaneously
    - 1/6 less chance of a hole aligning, which was a problem as no hole perfectly aligned at the proper torque

- Holes were also made too far in

Revision 1: Pentagon Nut

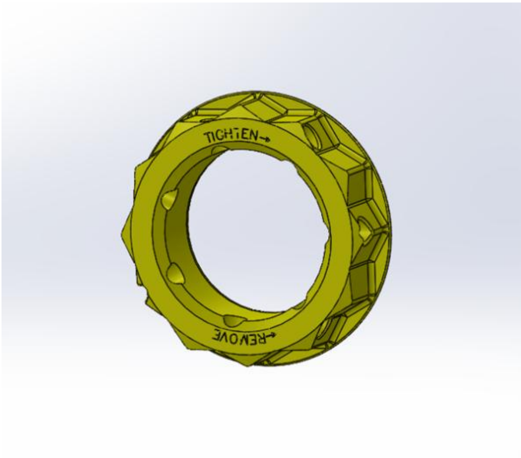


- Initially thought was an odd number of holes (5) would allow for only one hole to be perfectly aligned at a given time
    - Ended up being worse, all hole aligned at the same time as 5 is a direct multiple of 20



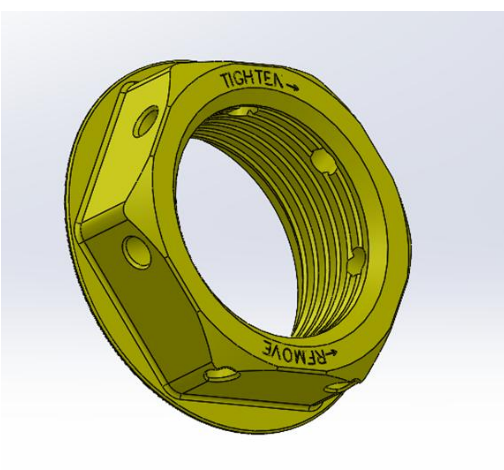
- Believed there were 5-point sockets with a compatible size
    - Realized there were not any, so the socket had to be specially made for this nut to work
- Adjusted hole depth more outwards to ensure hole aligned at torque

Revision 2: 12 Point Nut



- 12 Point nut with 7 holes
- Hole concept worked well, no holes aligned at the same time
- Concerns with holes on the small points of the nut

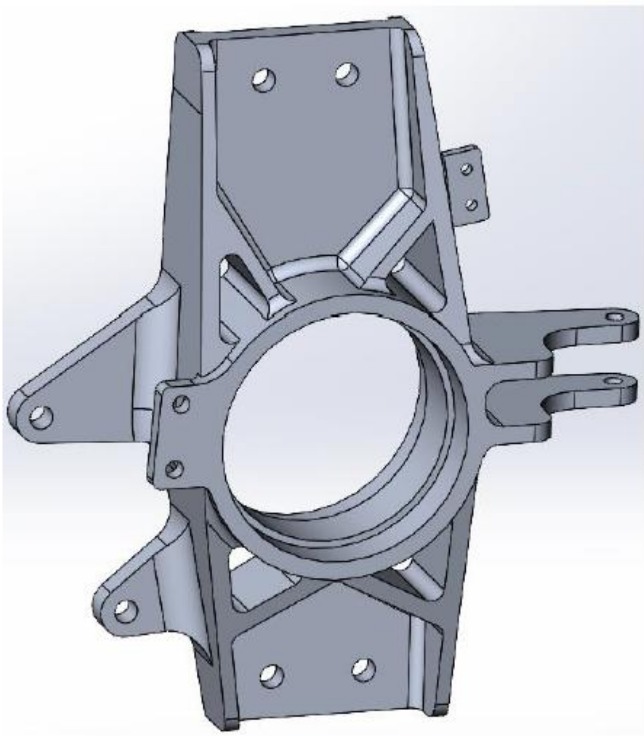
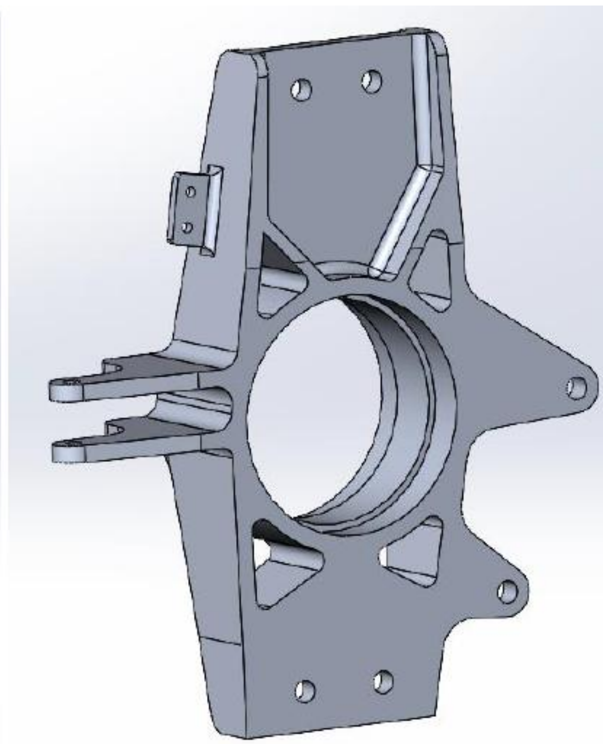
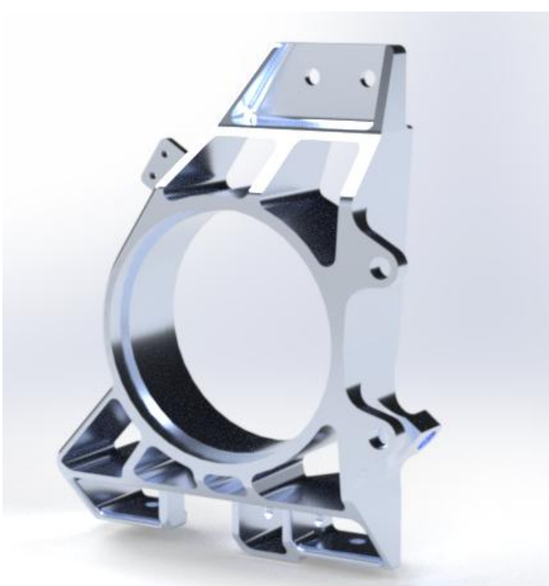
Revision 3: Hex with 7 holes

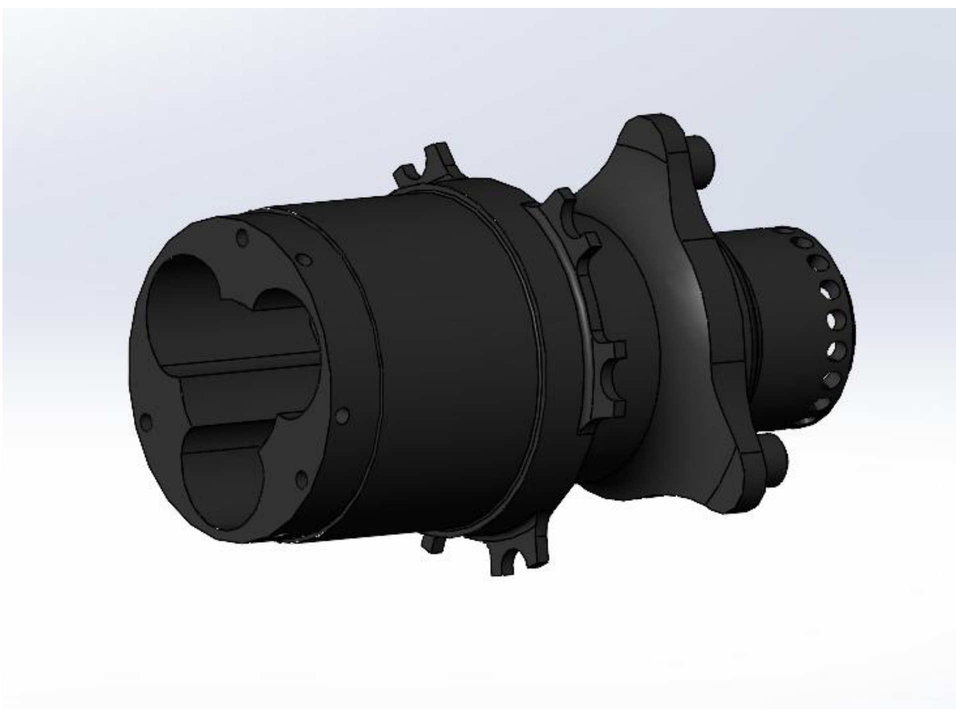


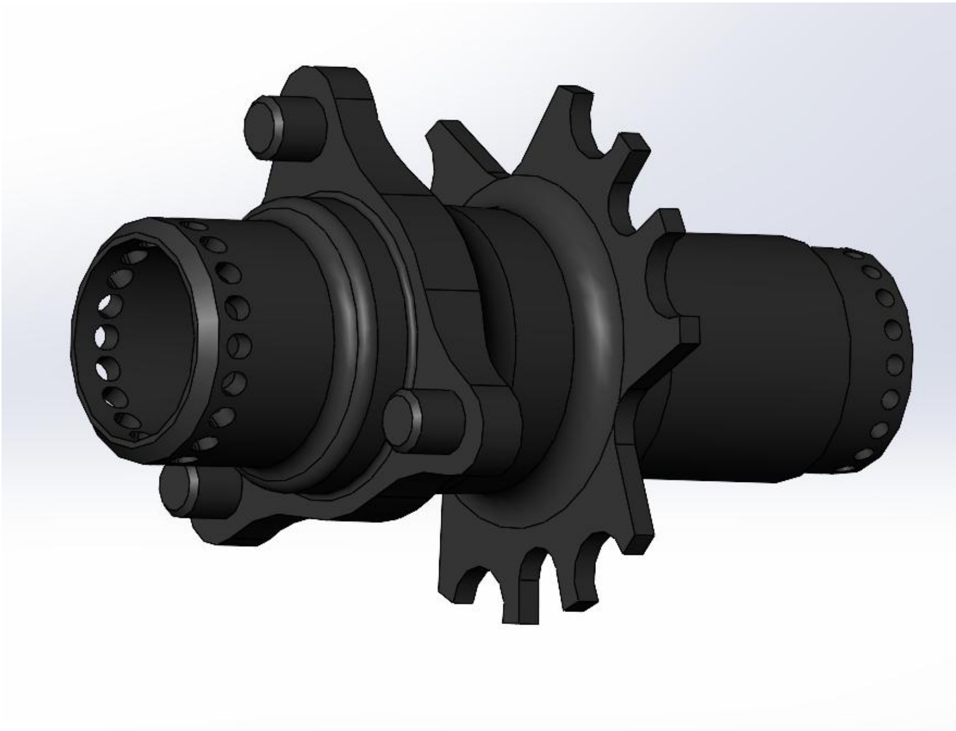
- Added the same 7 holes to a hexagon
- Increased pin hole diameter to .1" to increase alignment margin for error
- Can use the same socket previously used

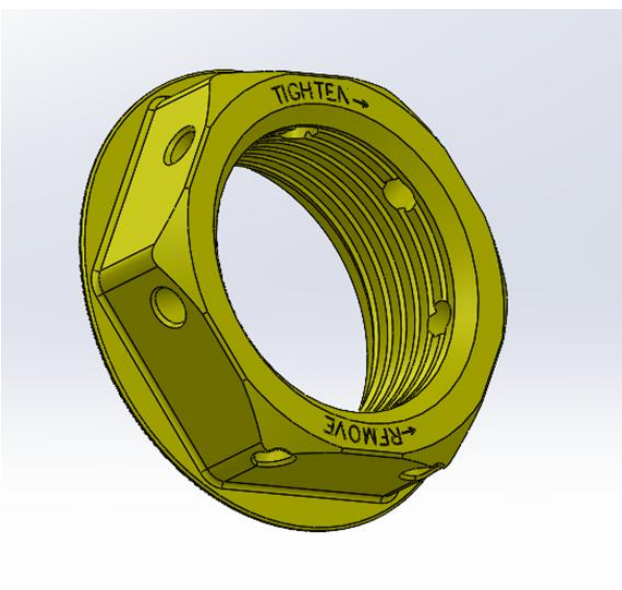
- Increased extrusion so that the larger holes would fit

CAD Models





Simulations

BM 24-25 Max Loading Cases

Set #	Name	Acceleration		
		Lateral(g)	Longitudinal(g)	Vertical(g)
0	Pure Cornering (Left Turn)	**2.4**	**0**	**1**
1	Pure Acceleration	**0**	**1.2**	**1**
2	Pure Braking	**0**	**-1.5**	**1**
3	Cone Strike	**0**	**0**	**3**
4	Combined Corner and Brake	**2**	**-0.8**	**1**
5	Combined Corner and Cone	**1.2**	**0**	**3**
6	Combined Brake and Cone	**0**	**-1**	**3**
7	Combined Corner and Accel	**2**	**0.8**	**1**
8	Combined Corner Brake and Cone	**0.7**	**-0.5**	**3**

Primary load case sets: 0, 2, 4, 7

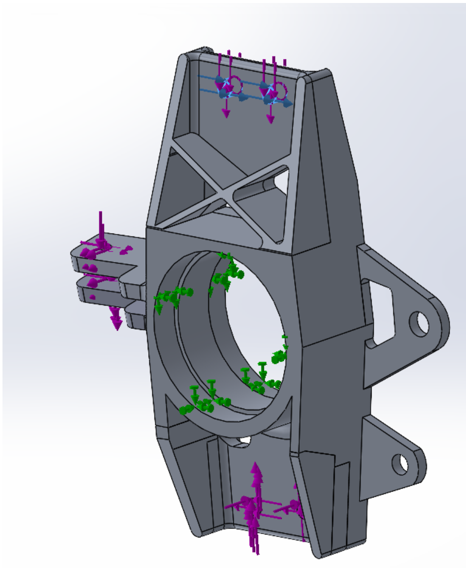
Resulting Forces:

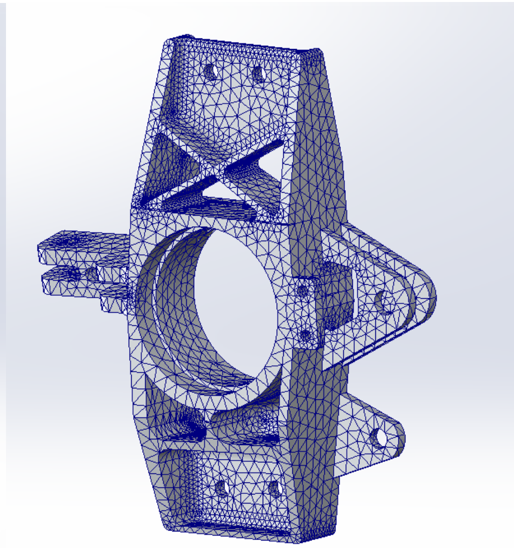
Force Results [Tires: LC0]										
		Front Left								
Set #	Name	Upper A-Arm Fx[N]	Upper A-Arm Fy [N]	Upper A-Arm Fz [N]	Lower A-Arm Fx [N]	Lower A-Arm Fy [N]	Lower A-Arm Fz [N]	Tierod Link Fx [N]	Tierod Link Fy [N]	Tierod Link Fz [N]
0	Pure Cornering (Left Turn)	71.81	-766.60	-1655.00	-7.15	3171.73	178.10	-64.67	-323.13	-50
1	Pure Acceleration	323.67	11.45	-384.00	-797.10	-315.00	8.14	65.14	303.54	57.1
2	Pure Braking	-1065.10	989.90	-1062.30	2734.71	-22.90	4.10	-202.55	-967.00	-137.59
3	Cone Strike(Front left)	67	861.43	-2152.45	-93.81	-988.60	-27.50	26.83	127.15	16.084
4	Combined Corner and Brake	-858.53	-397.11	-1642.30	2355.91	3935.35	187.77	-258.55	-1288.23	-195.57
5	Combined Corner and Cone	102.04	-50.50	-2623.31	-70.87	1737.63	79.00	-31.20	-154.5	22.4
6	Combined Brake and Cone	-928.94	1454.05	-2271.80	2361.96	-693.86	-94.50	-162.00	-760.2	-119.37
7	Combined Corner and Accel	1154.20	-1106.10	-1412.98	-2697.35	1857.51	142.42	159.88	796.5	135.33
8	Combined Corner Brake and Cone	-966.34	456.25	-2457.12	2566.80	2216.30	74.83	-239.34	-1149.03	-177.436

FEAs

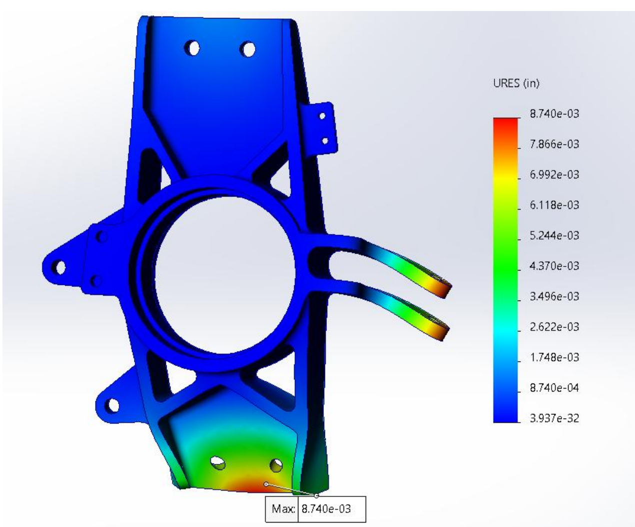
Assumptions:

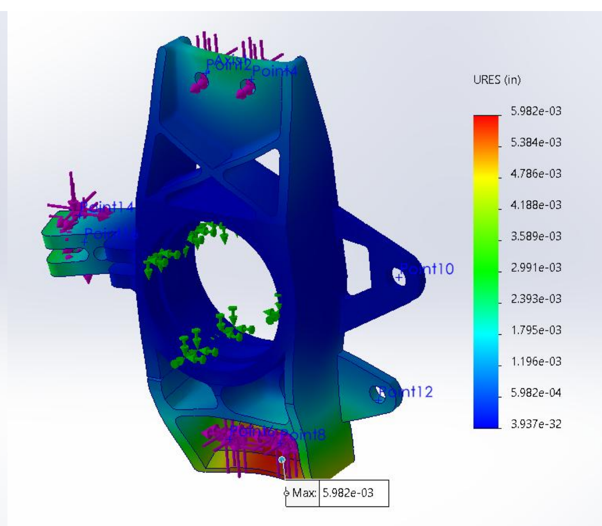
- Fixed at inner bearing race
- Max quality default mesh with mesh control in small fillets/corners
- Bearing loads applied to bolt holes
- Most important load case is case 4 since it is the most frequently happening in an endurance/autoX scenario



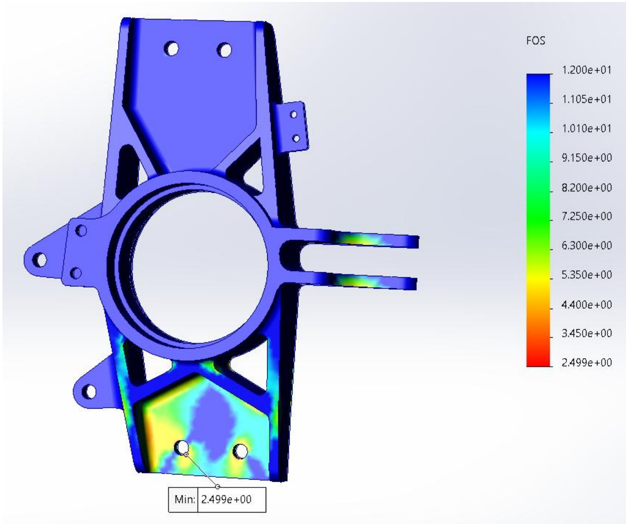


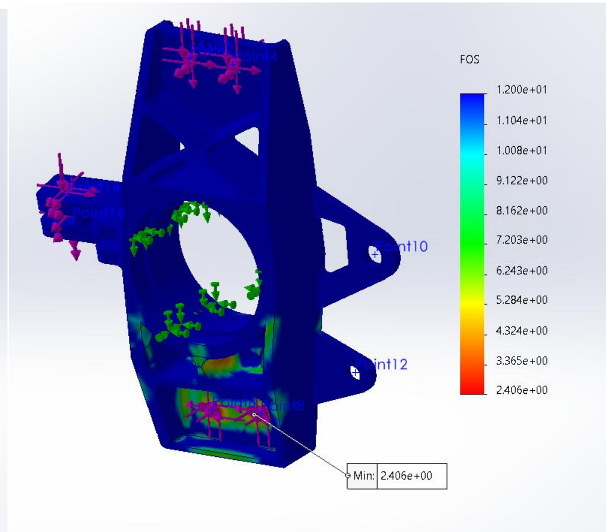
Deformation Case 4:



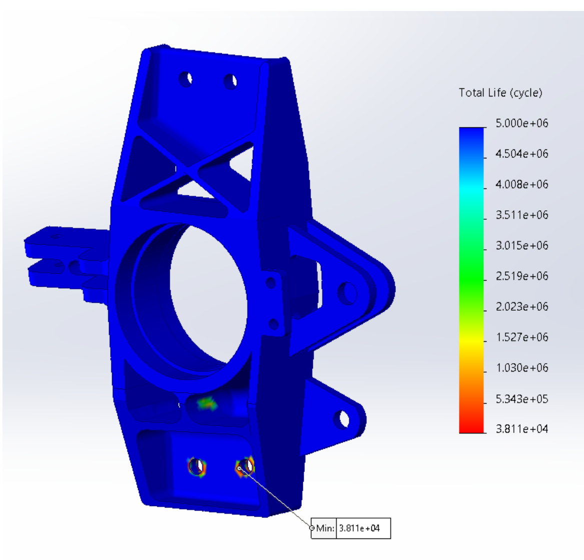


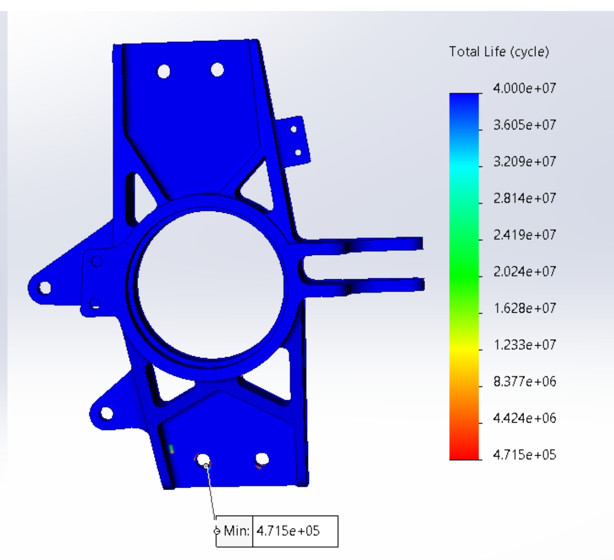
FoS Case 4:

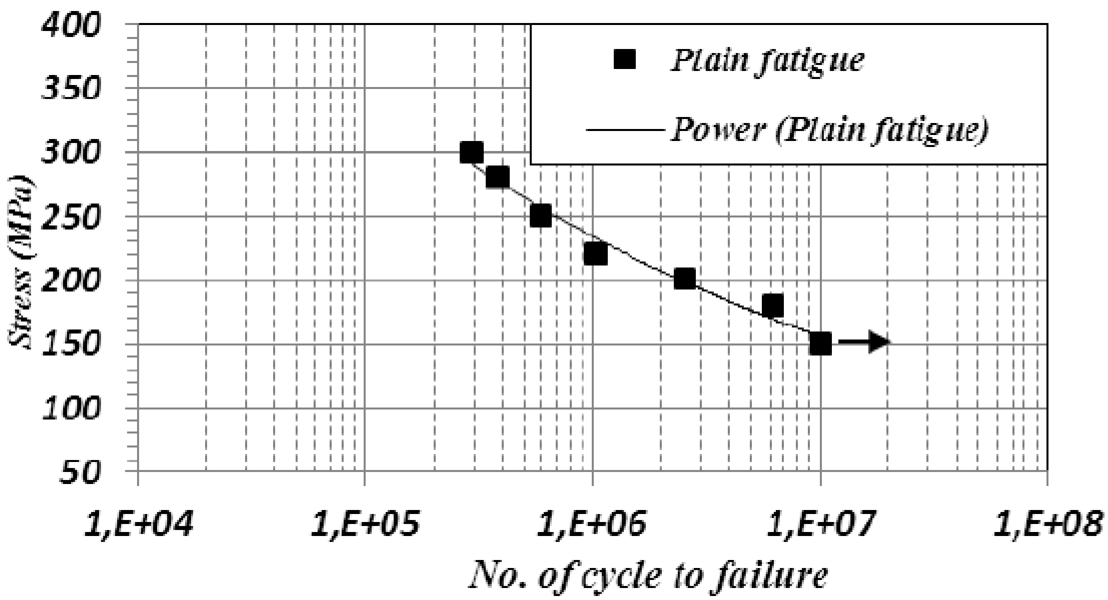




Fatigue Case 4

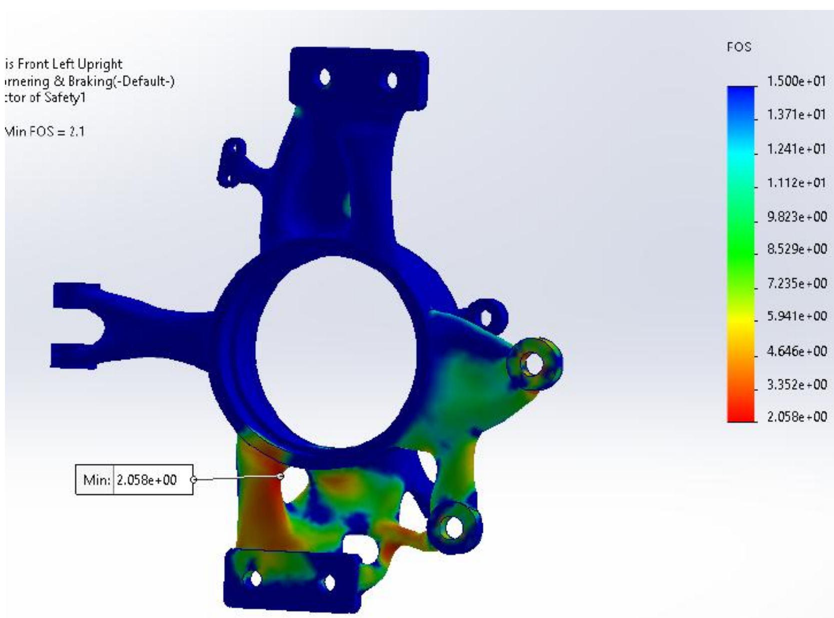






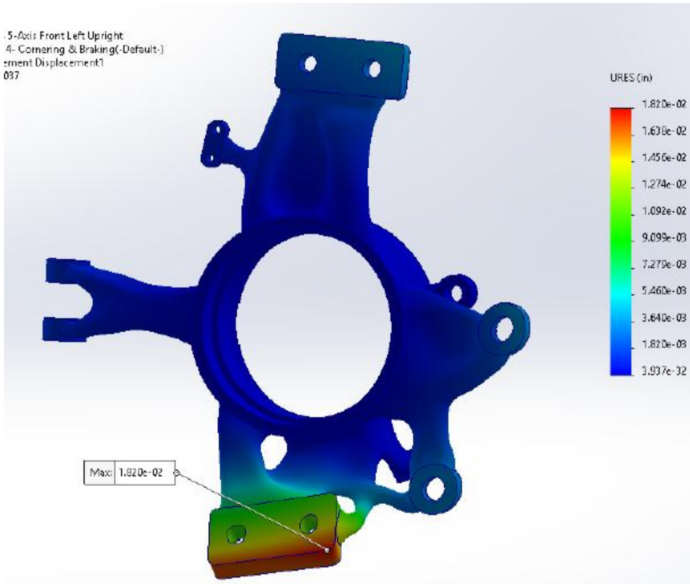
The S-N curve for 7075 Al shows an average stress of 245 MPa leads to just under 1,000,000 cycles.

Generative Design Front Upright



Min FOS: **2.1**

The previous year's car used a generatively designed upright that was machined with a 5 axis CNC mill. Although it is lighter and has a comparable FOS of 2.1 compared to the 2.5 of the traditional uprights, we did not choose to use this set of uprights as its Ackermann geometry is not as desirable as the traditional uprights according to driver feedback



Max Displacement: **0.0182”**

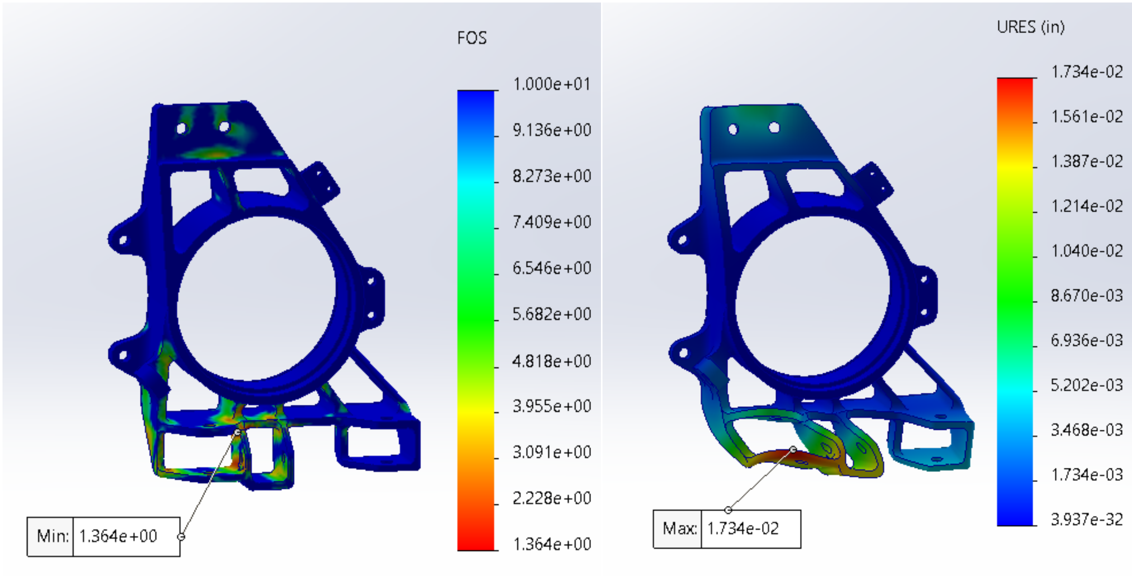
In terms of displacement, the increased deflection value of 0.0182” does not correlate linearly with tire performance due to unwanted camber change. This data comes from the tire captain’s analysis on how different cambers affect optimum tire performance.

Combined Cor	Stress (Von Mises) (psi)	Displacement (inches)	FOS (min)	Weight (l	Fatigue (Min Cycles)
Old (Bearing)	2.93E+04	8.74E-03	2.5	1.13	471500
Old (Force)	5.12E+04	1.74E-02	1.43		
New (Force)	5.70E-04	1.79E-02	1.278		
New (Bearing)	4.27E+04	1.03E-02	1.715	1.29	
New V2 Beefy	3.05E+04	5.98E-03	2.41	1.33	38100
Notes	Old is better in FOS and stress, but by relatively small margins (~4%). Displacement is 46.154% more on the old. However, this is arguably the most important load case				
Pure Cornering					
Old (Bearing)	4.19E+04	1.38E-02	1.749		
Old (Force)	4.03E+04	1.39E-02	1.819		
New (Force)	1.94E+04	8.89E-03	3.785		
New(Bearing)	3.16E+04	8.28E-03	2.317		
New V2 Beefy	2.64E+04	5.62E-03	2.771		
Notes	V2 is significantly better in every category, not close				
Pure Braking					
Old (Bearing)	1.68E+04	1.12E-02	4.372		
Old (Force)	2.33E+04	1.20E-02	3.151		
New (Force)	2.41E+04	7.57E-03	3.044		
New(Bearing)	1.85E+04	1.25E-02	3.954		
New V2 Beefy	1.64E+04	7.03E-03	4.471		
Notes	Results between V2 and old are very close for FOS and stress, but max displacement sees a 37% decrease going to the V2				
*Comparing Bearing Load Sims					

Column1	Column2	Column3	Column4
KPI	Point Weight	Old 3 (Datum	New 3 V2
Weight	3	0	-1.504
Max Stress	1	0	-0.41
FOS (Combined)	4	0	-0.36
FOS (Corner)	2		5.843
FOS(Brake)	2		0.226
Max Deflection(Comb)	3	0	3.158
Max Deflection(Corner)	1.5	0	5.928
Max Deflection(Brake)	1.5	0	3.723
Fatigue Resistance	4	0	-0.5
Total		0	27.7265
Point System: Percentage x 10			

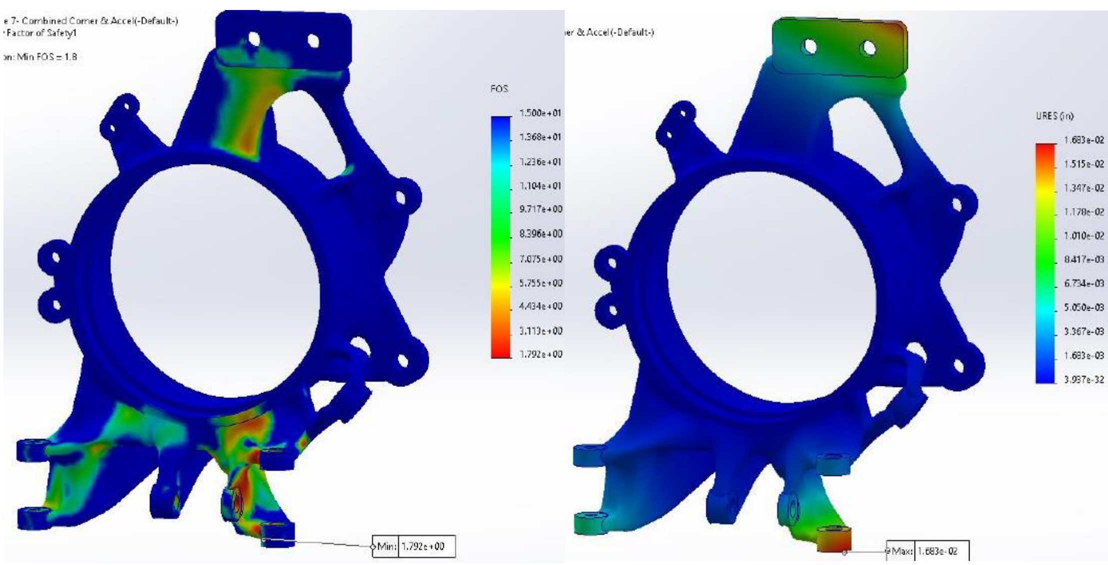
Although the new uprights do perform better than the old uprights in many categories and score higher in the design matrix, in reality, they are overkill. Tire simulations have shown that the amount less deflection of the new uprights does not linearly correlate with tire performance as there are diminishing returns as the deflection decreases. On top of that, having improved factors of safety of 2.771 and 4.471 don’t mean much in terms of raw percent increase. This is a high-performance racecar that is designed to not break during proper racing scenarios. The loads given to the upright are very reliable and representative of what the upright experiences as proven in past years. So, increasing factors of safety when they are already excessively high ends up lowering performance, as it is adding unnecessarily weight.

Rear Uprights



FOS and Displacement respectively

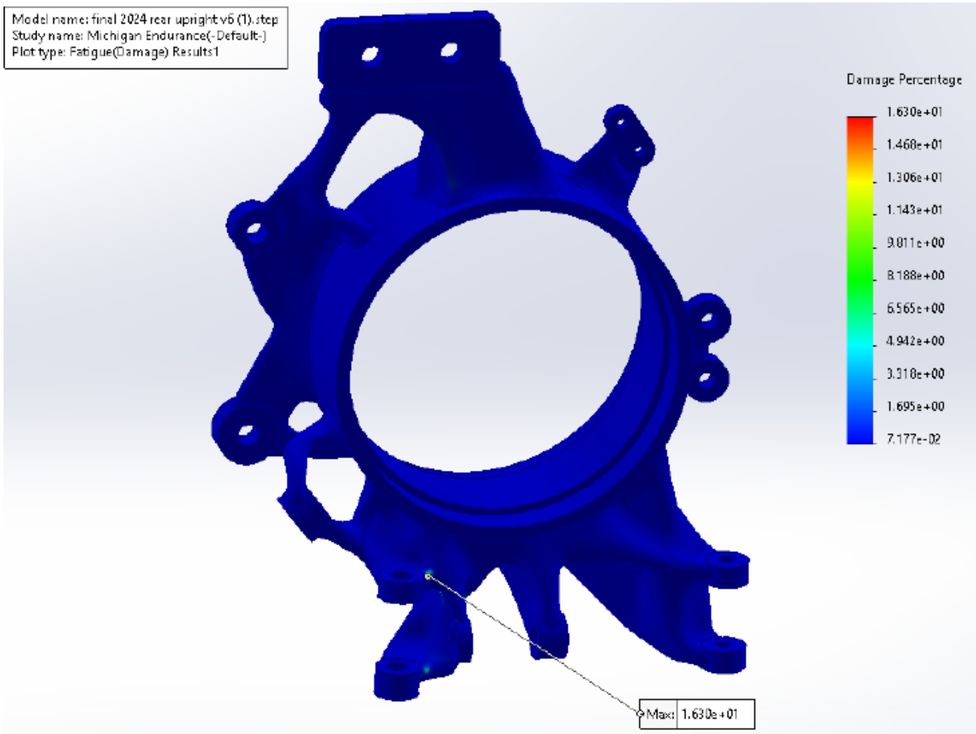
Previous Year's Upright Design



FOS and Displacement respectively

Despite the proposed new 3-axis friendly upright design being more "traditionally" manufactured, the FOS and displacement of 1.792 and .01683" respectively on the generatively designed uprights showed to perform better in terms of factor of safety, being 31.3% higher at its minimum point. This may be because the 3-axis uprights has many tight fillets which may induce stress concentrations, lowering the FOS. Due to this and the overall vehicle goal of reliability, I will be going with the 5-axis design for this car as it proves to be more reliable in FEA simulations and is also lighter.

Fatigue



The 5-axis upright has 16.7% damage across 330 laps which equates to a total of 22 endurance races. Even though these uprights were ran on the previous car, the previous car suffered with electrical and powertrain reliability issues which gave it a very low run-time. Including testing from this year, I have estimated that it had gone through 14 endurance cycles. With its damage only being at 16% for 22 endurance cycles, I believe that the upright will be used well within its endurance limit.

Lap Time Simulation for Front 3- Axis Front Uprights

OptimumG

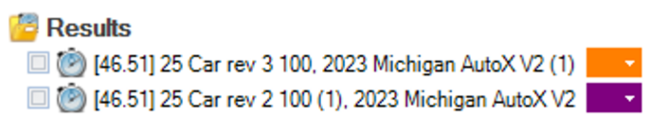
Endurance

Vehicle Weight (original)	599
Component Original Weight	2.26
Component New Weight	2.7
Weight Delta	0.44
Vehicle New Weight	599.44
Lap time (original)	126.318427
Lap time (new)	126.324382
Total lap time gain/loss	0.005954953

Comparing the total endurance lap times from the previous year's generatively designed uprights and this year's traditionally designed uprights, the increased mass of 0.44lb creates a difference of around 0.01 seconds over a lap. Over the course of around 22 laps, this amounts to 0.22 seconds which is relatively insignificant. A factor that could also determine the lap time would be driver skill, which could further negate this lap time differential.

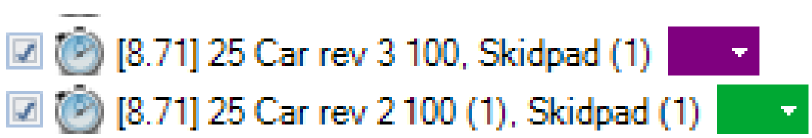
AutoCross

- No change over an autocross laps
- Rev 2 Mass: 278.306 kg
- Rev 3 Mass: 279.706 kg

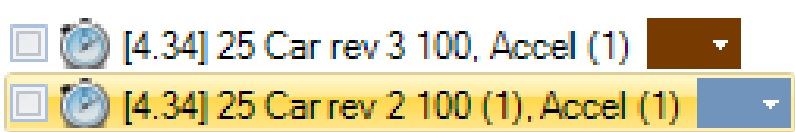


Like Endurance, there is minimal lap time change. Since AutoCross is significantly shorter than endurance, this lap time differential will be even more insignificant.

Skid Pad



Acceleration



There is no difference in times for Skid Pad and Acceleration events.

Manufacturing

Thanks to our team's sponsors Aether Machining and EEE Machining, the uprights were machined using 5-axis CNC machines and the wheel nuts were CNC lathed. Unfortunately, due to last minute unforeseen back orders, EEE machining was only able to manufacture one of the two front uprights. This news came only two weeks before the car was to leave for competition. Due to this, I was not able to find another manufacturer who could machine them in time before the car left for Michigan. So, we will end up running the backup option for the front uprights, which is last year's design with the old steering point and no wheel speed sensor gear.

The wheel nuts were then sent to be anodized. I chose a type III anodization (as per rules) with a layer thickness of 7-10 ten thousandths of an inch. This would make it on the thinner side of the type III specification. I chose this thickness to ensure that the threads of the nut stayed within tolerance of the hub threads.

Front upright made by EEE Machining

Rear Uprights made by Aether Machining

Testing

Unfortunately, due to a lack of time, I was not able to validate the upright design using real life data. However, if there was more time I would have used strain gauges on the uprights to measure the strain. The strain would then be used to calculate KPIs such as the maximum deflection and direction, and stresses.

$$\sigma = E\varepsilon$$

$$\frac{P}{A} = E\frac{\delta}{L} \rightarrow \boxed{\delta = \frac{PL}{EA}}$$

I would have then used dye penetrant in order to check for any micro-fractures or imperfections that may have occurred when the upright was under load during testing. Ideally there would be no imperfections since the testing time would be under the endurance limit of the uprights. However, there could be small stress concentrations that the FEA mesh did not pick up on.

Works Cited

Budynas, R.G., Nisbett, J. K., Shigley's Mechanical Engineering Design, 10th edition, McGraw-Hill, 2015.

Milliken, W. F., and D. L. Milliken. *Race Car Vehicle Dynamics*. 1995.