

Comprehensive Design Review

BAJA SAE 2025-2026

November 13, 2025



Paul M. Rady
Mechanical Engineering
UNIVERSITY OF COLORADO **BOULDER**



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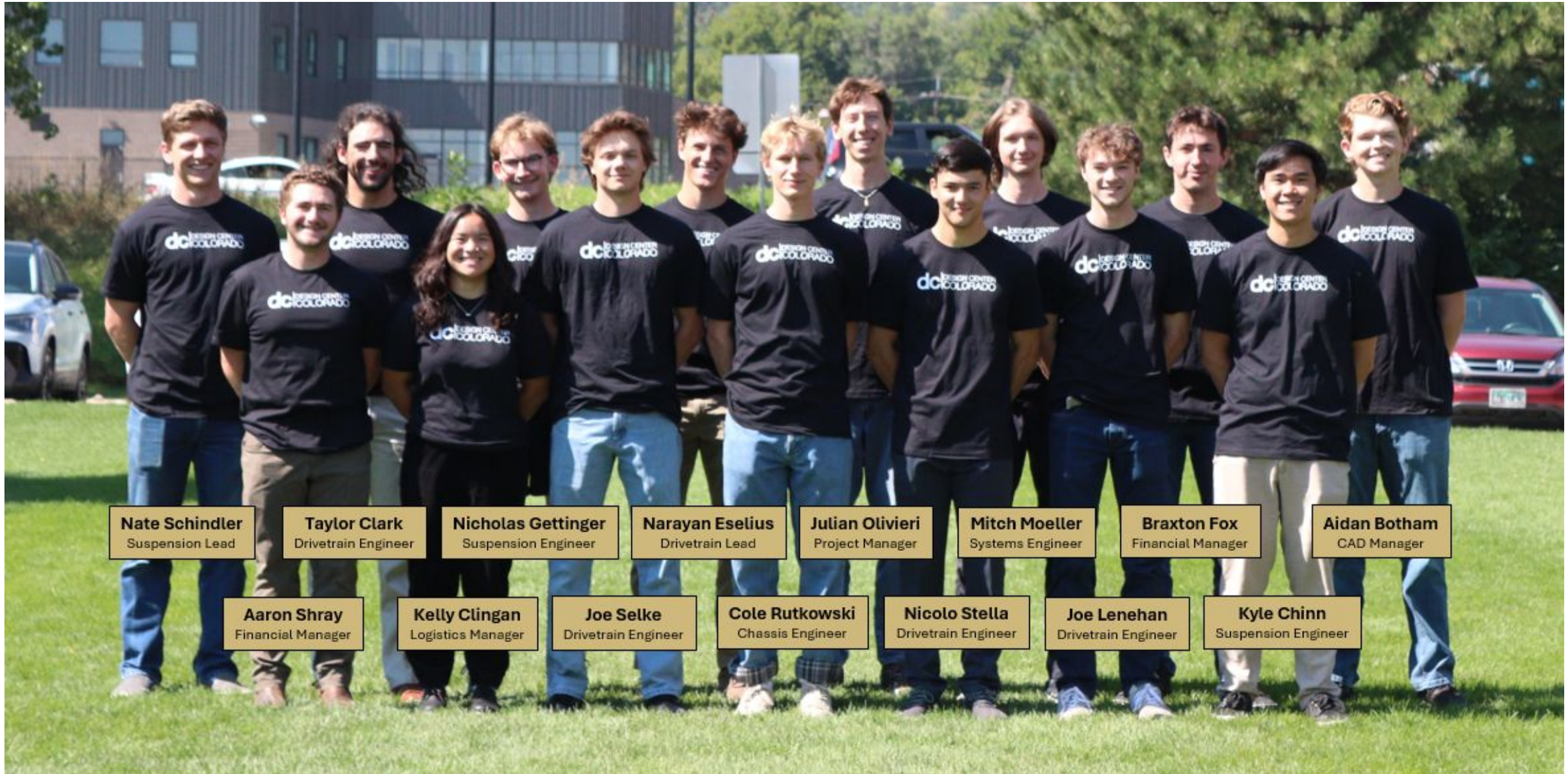


Taylor Clark
GALLERY

The Family, Friends, Advisors, and
Alumni of CU Boulder SAE Baja

Meet The Team

3



Results: Arizona 2025 Team Placement

Score Category	Result (Out Of 69 Teams)
Tech Inspection	Passed
Endurance Race	35th
Acceleration	40th
Maneuverability	DQ
Suspension/Traction	19th
Pilot Pull	4th

Results: Arizona 2025 Team Scoring	
Score Category	Result
Cost Report	9/15 pts
Cost Prototype	18/85 pts
Design Evaluation	30/150 pts
Business Presentation	15/70 pts

Timeline

CDR	November 13 th	About 90% of the car should not change after CDR
Manufacturing Review	December 5 th	Last Day of Classes
First Major Parts Order	December 6 th	Reading Day of Finals Week
Steel Order	December 6 th	
SAE Business Presentation Report	December 13 th	Can submit late for partial scoring
Start of Chassis	January 8 th	First day back in the spring
Rolling Chassis	February 6 th	Suspension and Chassis Need to be completed for rolling chassis
Roll Cage Documentation Package	February 16 th	Required SAE Deliverable
Written Cost Reduction Report	February 20 th	Required SAE Deliverable
Business Presentation	February	
Online Cost Prototype	March 20 th	Required SAE Deliverable
Design Review Briefing	April 2 nd	Required SAE Deliverable
Running Vehicle	April 10 th	Mid-semester Reading Day
Testing	April 11 th – April 25 th	Testing Stops Before Final Exams
Competition in Oregon	5/7 - 5/10	Grand Finale

Agenda

Item	Time	Hour
Overall Design	<5 Minutes	6:00 – 6:05
Budget Status	<5 Minutes	6:05 – 6:10
Suspension Design	10 Minutes	6:10 – 6:20
Controls Design	10 Minutes	6:20 – 6:30
Drivetrain Design	15 Minutes	6:30 – 6:45
Chassis Design	10 Minutes	6:45 – 6:55
Intermission	10 Minutes	6:55 – 7:05
Questions and Feedback	7:05 - End	

Complete Assembly

8



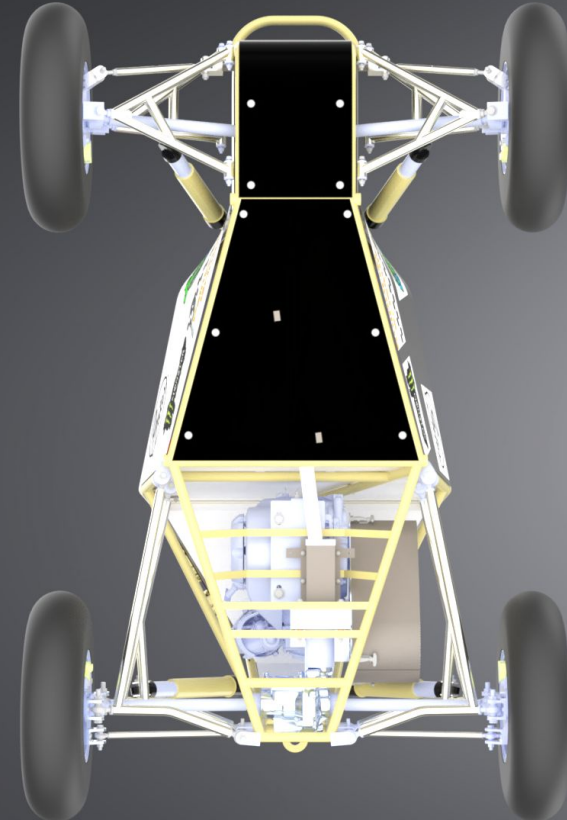
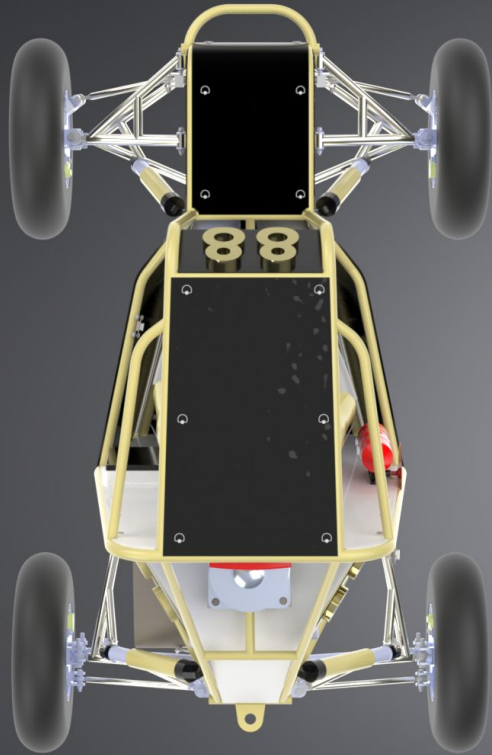
Complete Assembly

9



Complete Assembly

10



Complete Assembly

11



Overall Team Goals

Goal	Reason	Implementation
Reduce Vehicle Weight by 10% Missed (Measured in CAD with an extra 70 lbs added for Tabs/Welds)	Improve acceleration of the vehicle which will help us perform better at competition	Shrink parts that have a High Factor of safety with medium to high confidence of failure condition
Finish Endurance without Failure In Progress	We want to place high in endurance	Build a robust vehicle with about 50% proven reliability (reused parts)
Increase Serviceability In Progress	Increased serviceability will allow for more rapid testing and modifying during our test phase throughout April which will also enable us to have a better performing vehicle	Work on mounting in parallel with the design of all components, instead of it being an afterthought

Finance

Total Budget Estimate Overview

Category	Description	Estimated Cost
Housing Travel Registration	Housing, Transport, and Competition Registration Costs	~\$2,000 (Rental Vans) ~\$2,500 (Housing) -\$1,850 (Registration) ~\$2,000 (Gas)
Chassis	Assorted 4130 Tubing Thicknesses and Lengths, Body Panels, Fasteners	~\$3,000
Front/ Rear Suspension	4130 Tubing, Fasteners, Linkages	~\$2,100 total
Controls /Brakes	Steering rack, Assorted Electronic Components	~\$400
Drivetrain	Transfer Case Materials, New CV Axles, U Joints	~\$2,200
*Miscellaneous	Brake Service Kits, External Shock Service, Generator Rental, Team Shirts, Car Decals, Etc	\$1,500

**CDR Total Estimate:
\$17,550***

Category	Cost
Travel / Competition	\$8,350
Construction	\$7,700
*Miscellaneous	\$1,500

***PDR Total
Estimate:
\$18,950**

Budget and Current Funds Overview

Sponsor	Contribution	Application of support
Funds from 2025 team	*\$500 as of 10/5	-\$250 Competition / Travel -\$250 Materials
F.F.A.A. of CU Boulder SAE Baja	*\$2500+	-\$1250 Competition / Travel -\$1250 Materials
Taylor Clark Gallery	*\$1000	-\$500 Competition / Travel -\$500 Materials
Painter's Grinding	*\$1500	-\$750 Competition / Travel -\$750 Materials
Jamsi Online / Nick Wolfrum	*\$1000	-\$500 Competition / Travel -\$500 Materials
Monster Energy	In - Kind Donation	-Competition / Team / Event Support
Design Center Colorado	\$2000	-\$1850 Competition Registration -\$150 Materials

Remaining Needs: \$9,050*

	Competition / travel	Construction	Miscellaneous
Raised	\$5,100	\$3,400	\$0
Gap	\$3,250	\$4,300	\$1,500
*Does not include pledged support			

*Reported Gift Account Balance (11/11/25):	~\$5,200
*Estimated Gift Account Donations:	~\$6,500

Funding Gap Plan

Pledged/Anticipated Support*		Planned Grants and Sponsorship Programs*		Volunteer Opportunities	
Sponsor	Support	Source	Support	Source	Support
Jamsi Online	\$1000 (spring donation)	EEF (Spring Award)	\$5,500	SAE Learn Twice Program	\$2500 (30 hrs collective volunteering)
Gates Belts	\$2000+ and CVT Belts	CEAS Travel	\$4,500	CU Athletics and Ticketing	\$50/volunteer (Per CUEC Event) *
O'Reilly Auto	In Kind / Financial	Gene Haas Foundation	\$2,500	*Up to 27 CUEC Basketball Events Per Academic Year	
Michigan Science Corporation	Wheel Force Transducer	SKF Sponsorship Program	Up to \$2000 of Sponsored Bearings	CU Athletics and Ticketing	\$100/Volunteer (Per Folsom Event) *
Futek	Sensor Packages	KHK Sponsorship program	Discount on Gears	*Up to 6 Football Games Per Academic Year	
Freewave	Radio/Telemetry Communication	HyGear Suspension	Discount of Shock Service		

*Is not a complete list

Final Remaining Gap:
\$6,050

Pledged Support:
\$3,000

Total Potential Fundraising Options:
\$16,000+

Volunteer Opportunities:
\$2,500+

Potential Grants:
\$13,500

Future Potential Partners (2026-2027)

Potential Partners*		Potential In-Kind or Discounted Support*	
Sponsor	Support	Source	Support
EarthRoamer	Financial	Century Springs	Springs
Matrix Technologies	Financial	G & C Design and Fabrication	In-Kind (Custom Gears and Gear Stock)
Wright EDM	Under Review		
Rousseau Metal	Under Review		
Accu-Precision	In-Kind / Financial		
SRAM	Financial / In Kind		

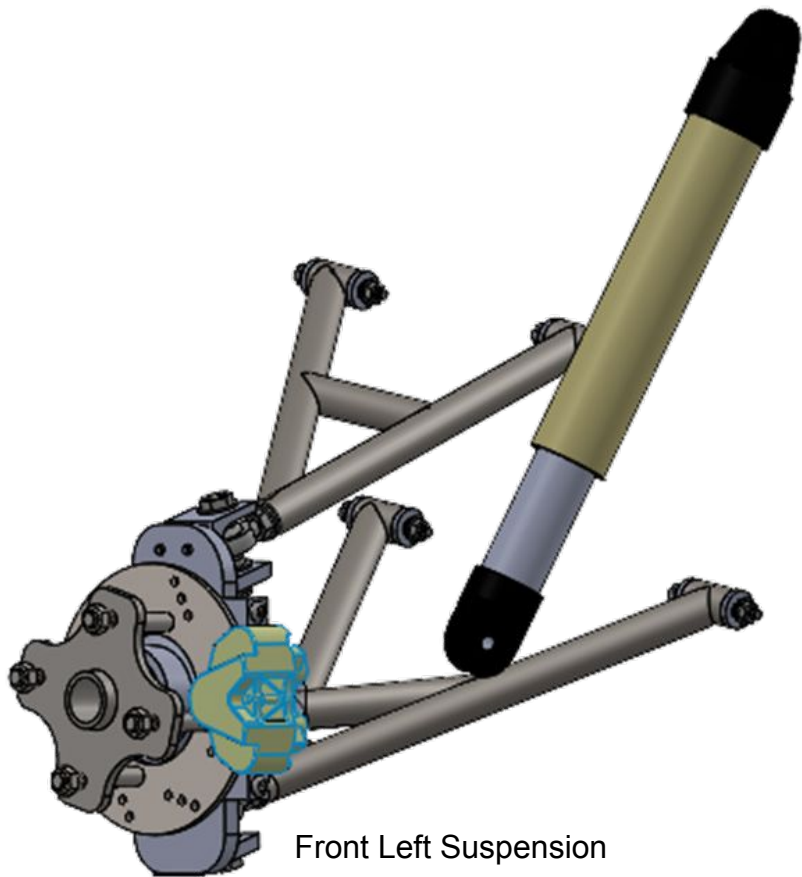
*Is not a complete list

Suspension

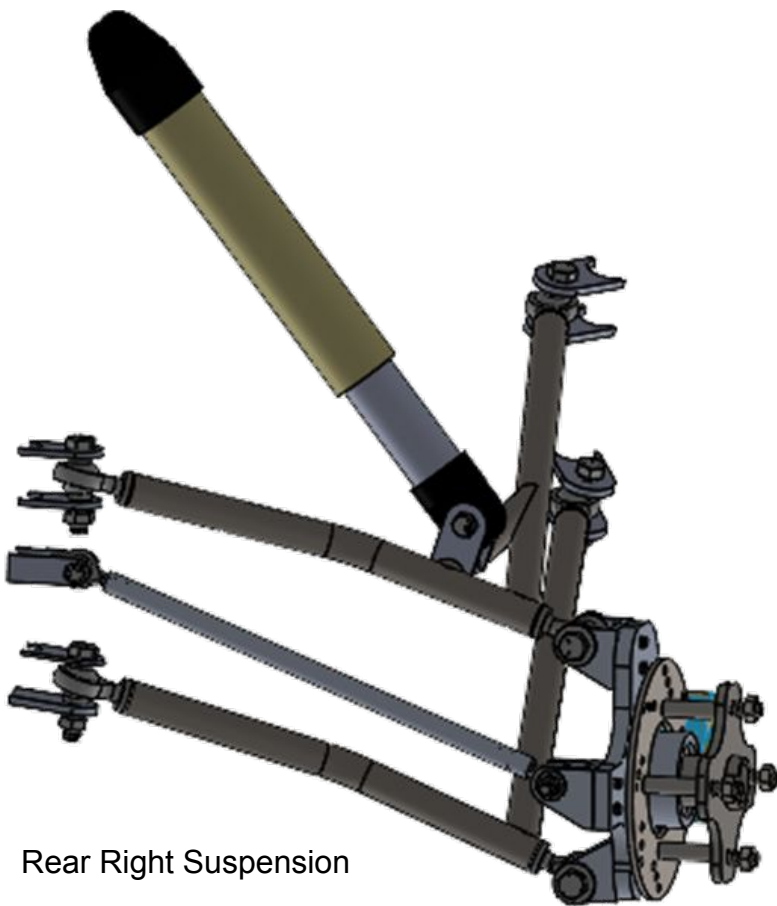
Suspension Goals

Goal	Reason	Implementation
Reduce body roll in cornering by 5% In Progress	Improving cornering speed and handling	Raise the roll center, bringing it closer to center of gravity
Have zero set-in from suspension travel (shocks return the vehicle to designed ride height and have 0 roll at steady state) In Progress	All travel constantly usable and the handling more uniform	Redesign bushings and increase angle shocks in the rear to increase progression through travel
Have frontal impact compliance Achieved	This will give absorption of forward impacts, improving ride quality	Tilt the suspension back 10°

Suspension Overall View



Front Left Suspension



Rear Right Suspension

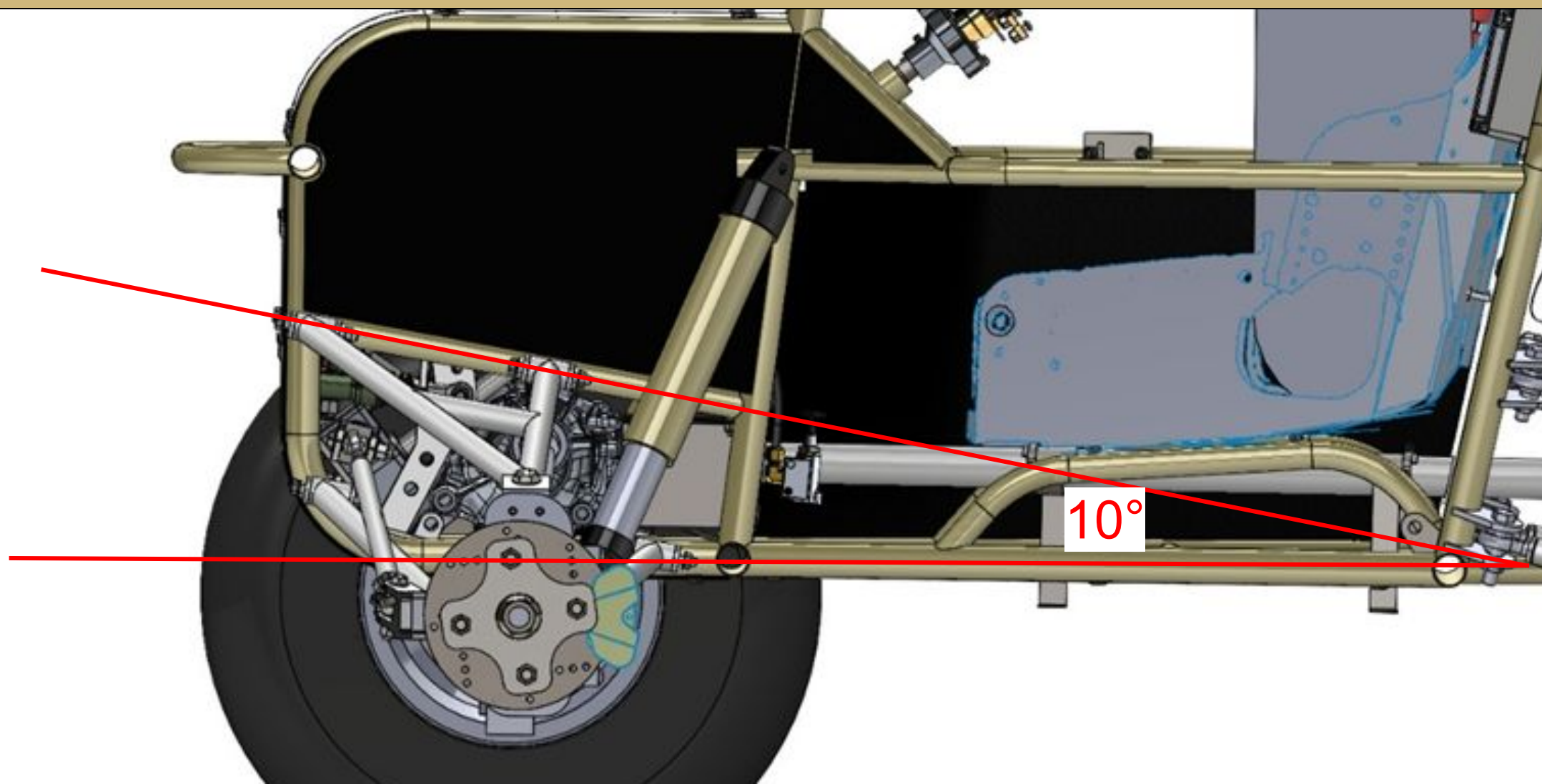
Front Left Subassembly Summary

Weight [lbs]	Cost (Excluding Shock)	Rules
This year: 14.0 Last year: 16.8	Anticipated: \$376	B.8.7.1 B.1.4.2

Rear Left Subassembly Summary

Weight [lbs]	Cost (Excluding Shock)	Rules
This year: 15.0 Last year: 19.8	Anticipated: \$348	B.1.4.2

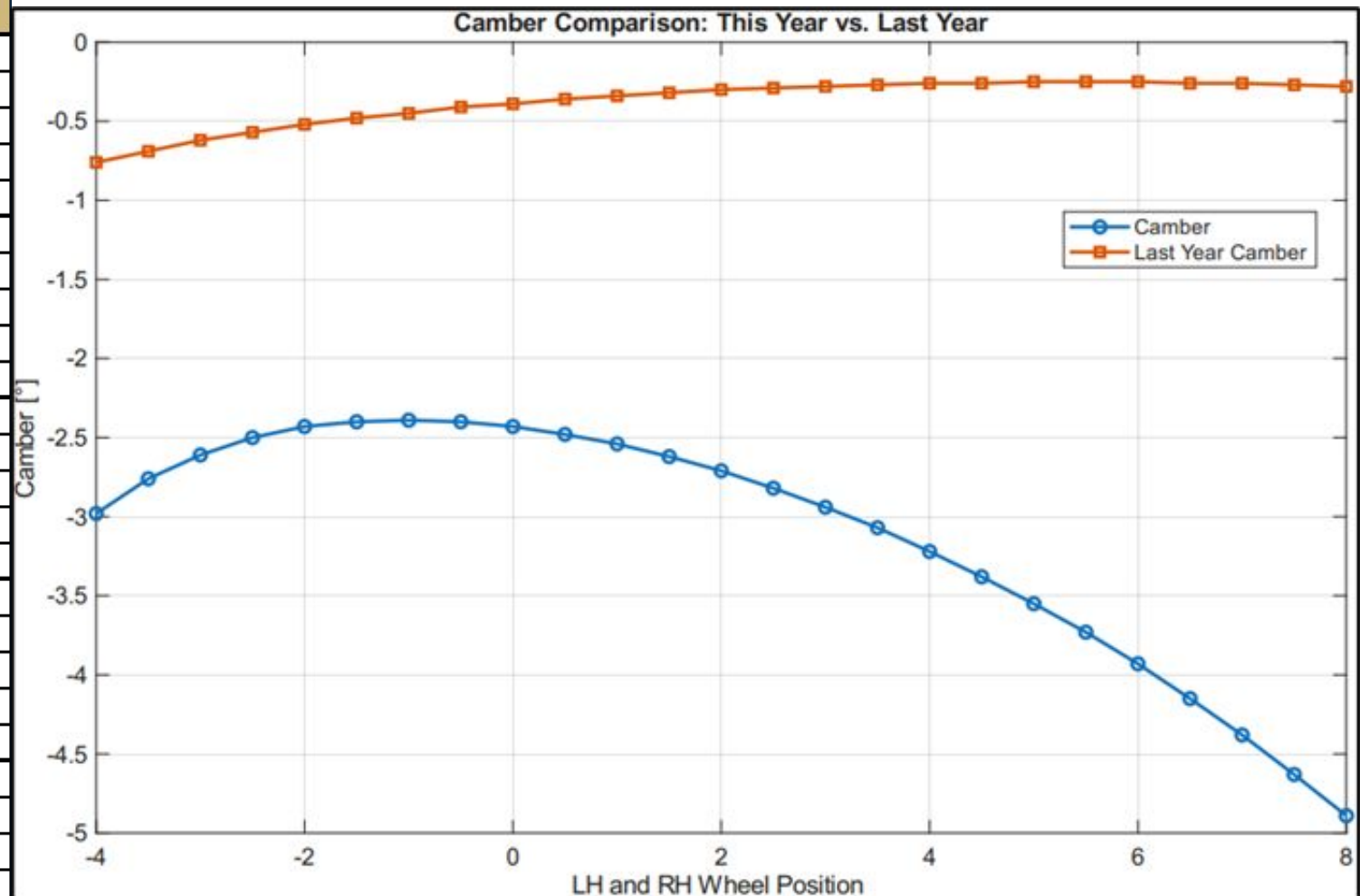
Front Suspension Rake



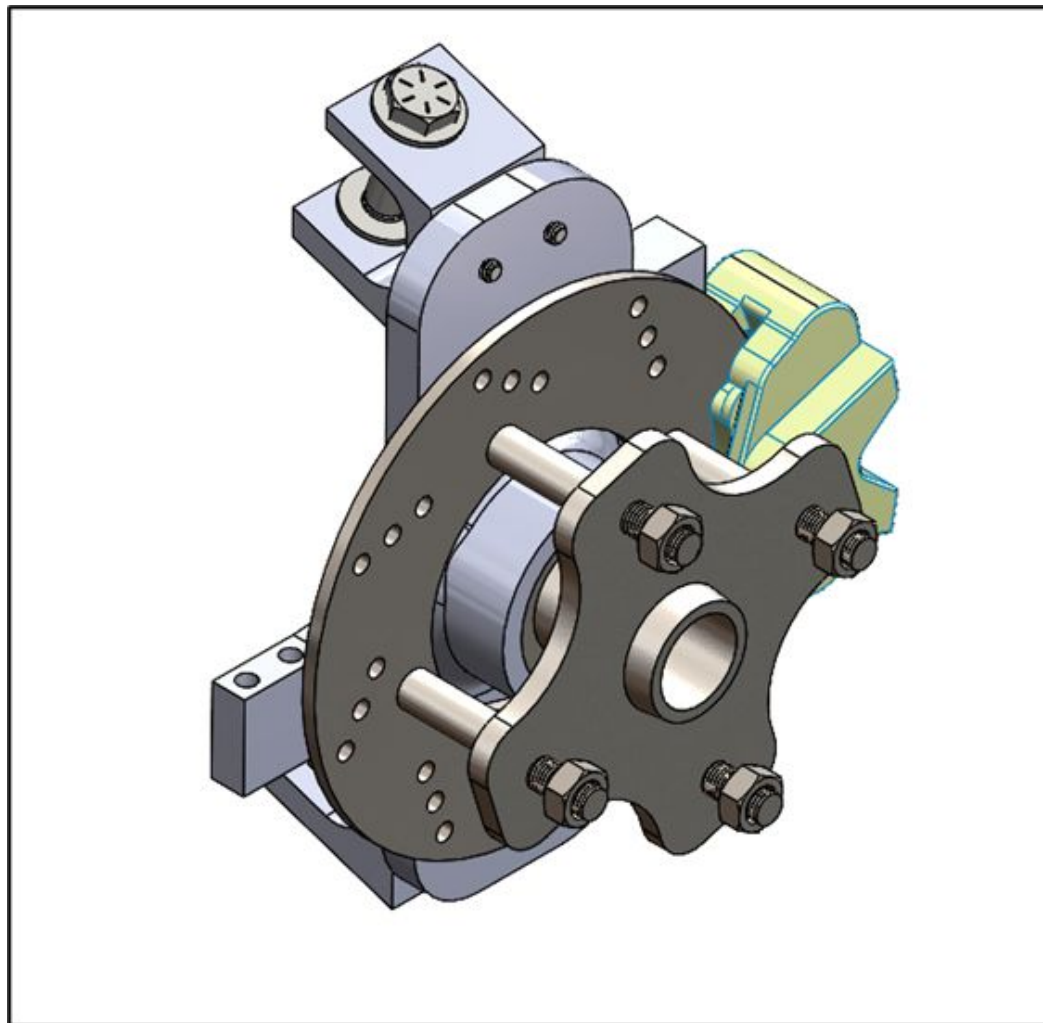
22 Front Suspension Analysis		
Spec	This Year	Last Year
Roll Center Below Frame	1.8 in	6 in
Total Width	58 in	54 in
Travel	11.5 in	12 in
Ground	12 in	12 in
Static Camber	-2°	-7°
Progressive Camber	-2 to -5°	0.48°
Shock angle	68.2°	74°
Suspension Rake	10°	0°
Caster	4°	0°

Front Suspension Kinematics

LH and RH wheel	Drive Shaft Length	Plunge [in]	Inner CV Joint Angle	Outer CV Joint Angle
8 bump	16.5	-0.44	11.32	6.34
7.5 bump	16.5	-0.456	9.67	5.03
7 bump	16.5	-0.469	8.04	3.79
6.5 bump	16.5	-0.478	6.43	2.73
6 bump	16.5	-0.484	4.88	2.17
5.5 bump	16.5	-0.486	3.46	2.5
5 bump	16.5	-0.486	2.42	3.51
4.5 bump	16.5	-0.482	2.35	4.79
4 bump	16.5	-0.475	3.31	6.21
3.5 bump	16.5	-0.464	4.72	7.68
3 bump	16.5	-0.451	6.29	9.21
2.5 bump	16.5	-0.433	7.94	10.78
2 bump	16.5	-0.412	9.62	12.38
1.5 bump	16.5	-0.387	11.34	14.01
1 bump	16.5	-0.358	13.08	15.68
0.5 bump	16.5	-0.323	14.85	17.38
Static	16.5	-0.284	16.65	19.13
0.5 droop	16.5	-0.239	18.47	20.92
1 droop	16.5	-0.188	20.32	22.76
1.5 droop	16.5	-0.13	22.2	24.65
2 droop	16.5	-0.063	24.12	26.61
2.5 droop	16.5	0.014	26.09	28.64
3 droop	16.5	0.102	28.1	30.77
3.5 droop	16.5	0.205	30.18	33
4 droop	16.5	0.326	32.33	35.37



Front Suspension Upright Structural Analysis

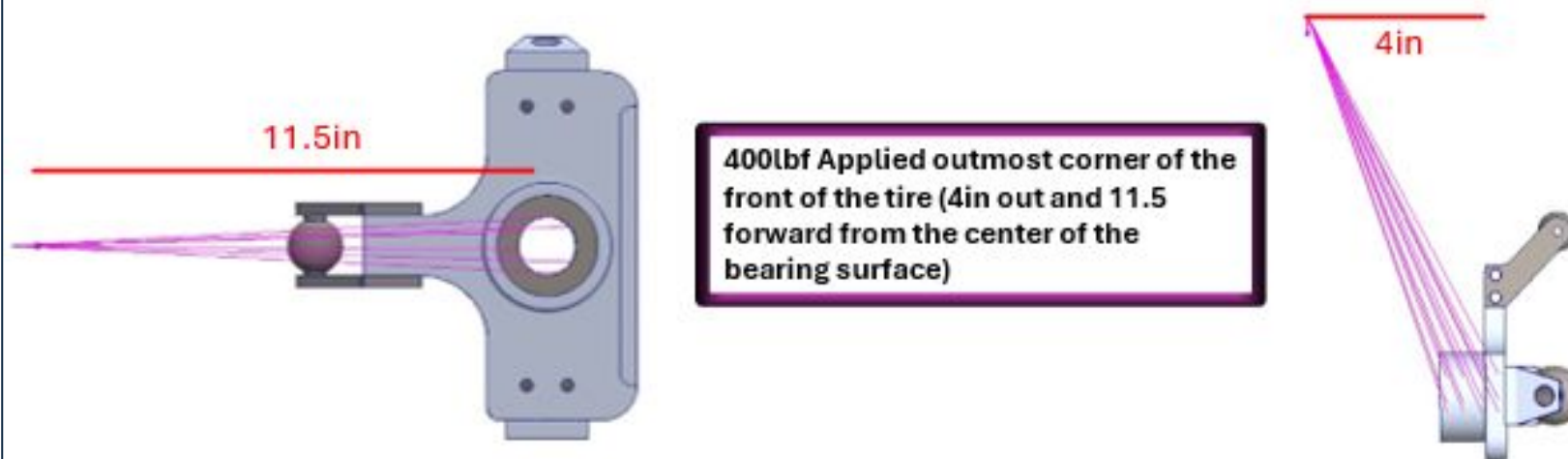


Last Year's Upright Failure Case

Failure Condition: Frontal impact on the outermost point of the front of 1 front tire stopping the vehicle.

Justification: We experienced a crash on our first testing day which resulted in this loading case and a plastic deformation 0.075 thou originating in the aluminum right after the tie rod bracket ended.

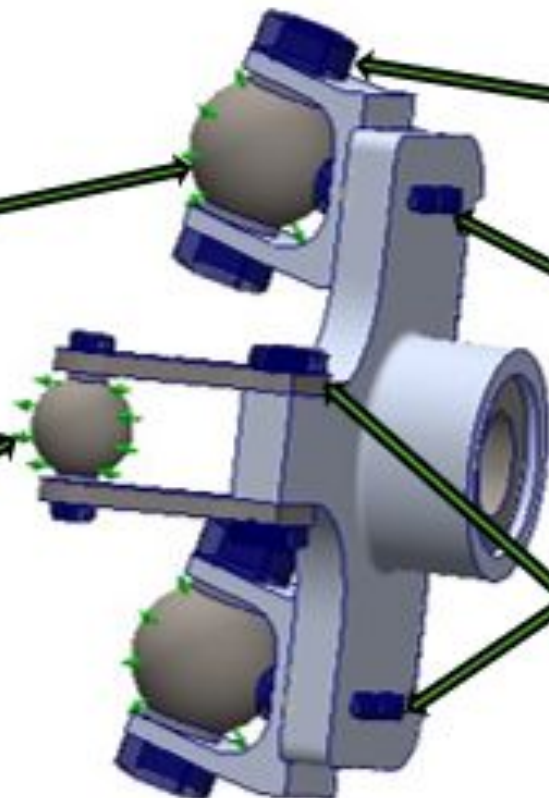
Load Diagram:



Constraints:

Radial Constraint acting on the outside face of the control Hiems. This locks all translation and is an over constraint for the top hiem as it does allow for up-down movement.

A translation constraint in the direction of the tie rod representing the 2 force member. This Acts on the outside face of the Hiem.



Bolt connections for $\frac{1}{2}$ -20 bolts with 144in-lbf and bond interaction between the Heim joint and the bracket. This is because of friction.

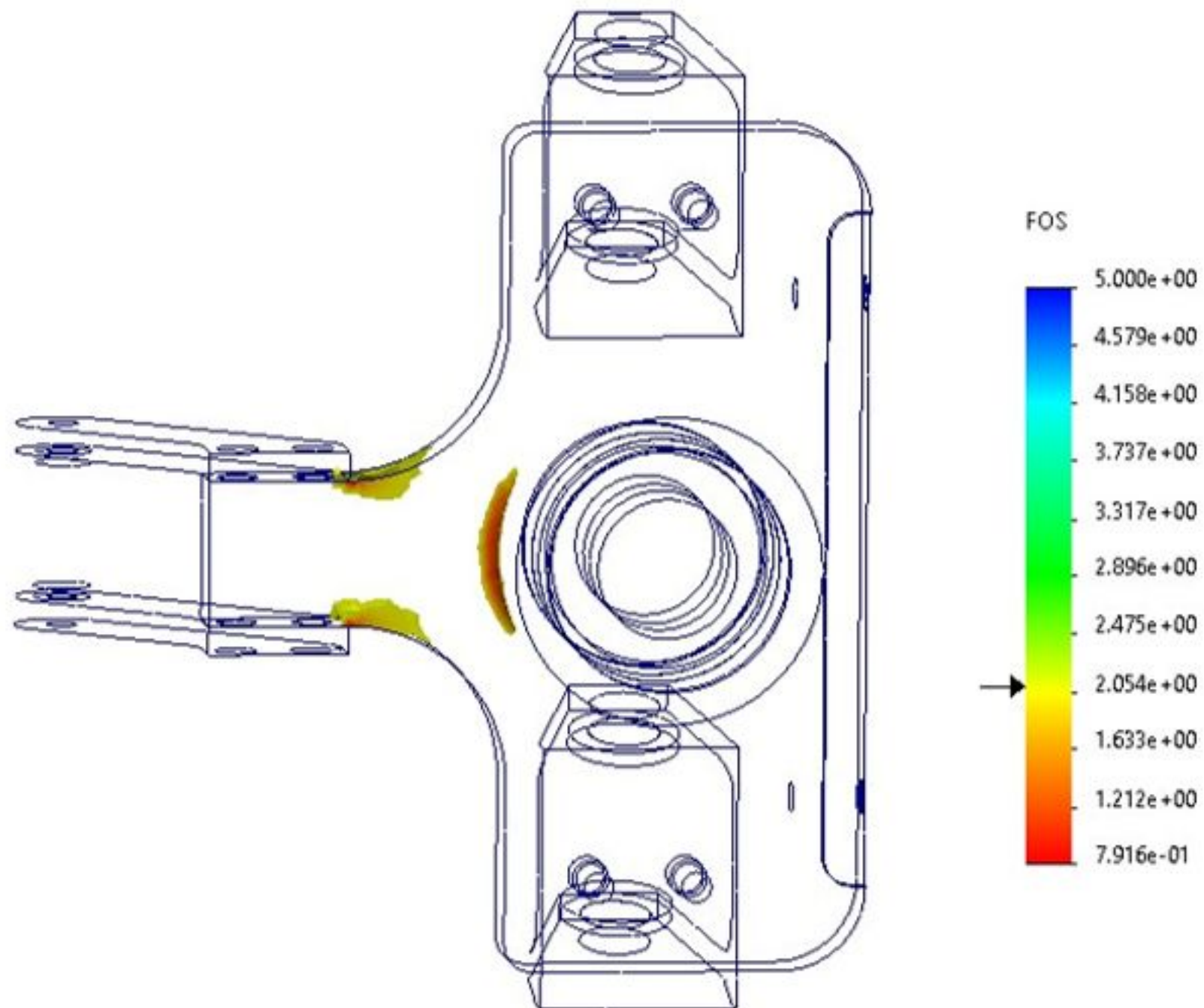
Bolt connections for $\frac{1}{4}$ -20 bolts with 144 in-lbf and local interaction between the back face of the bracket and upright

Bolt connections for 10-32 bolts with 50 in-lbf and local interaction between the back face of the bracket and upright

FOS: .8

Equations (If Applicable):

Confidence Level & Explanation:Medium: Due to SolidWorks limitations on Boundary conditions this model is over defined. For a front impact there is Little to no upward forces therefore this is an "OK" fix. Also Due to the 24-25 upright not failing at the base of the bearing housing, seen in the FEA, this model is close but not perfect.

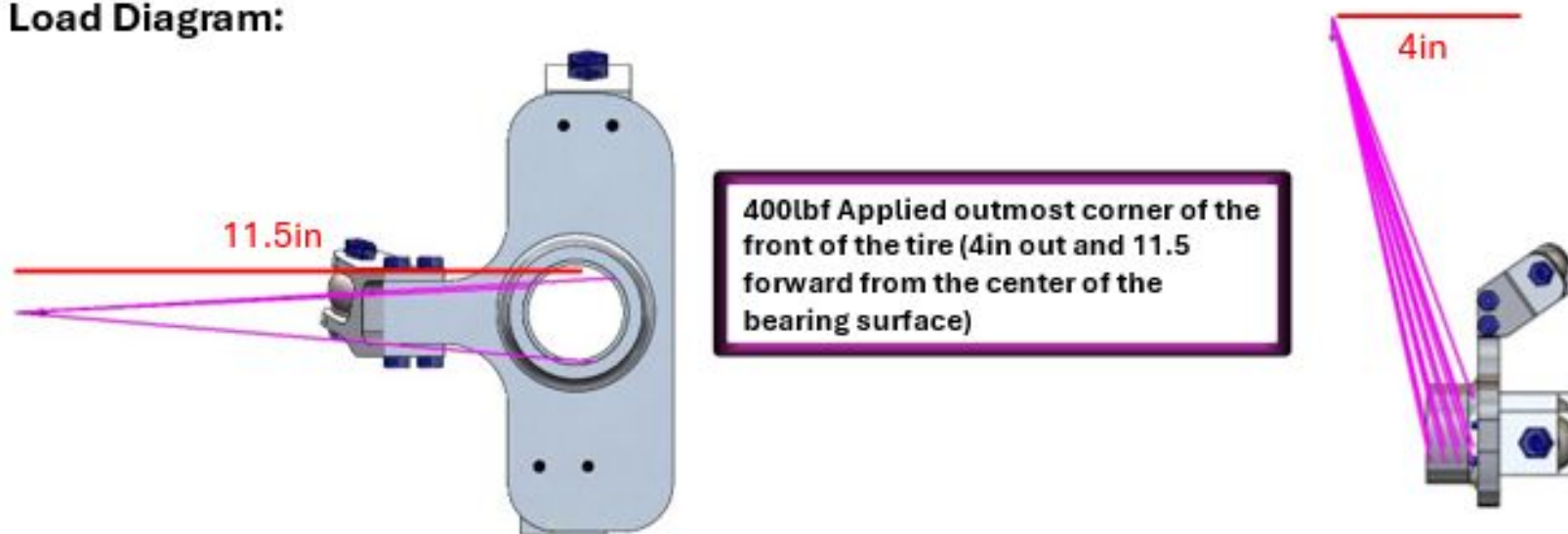


This Year's Upright Failure Case

Failure Condition: Frontal impact on the outermost point of the front of 1 front tire stopping the vehicle.

Justification: Due to the failure of last years upright the same loading case was assumed on this year model

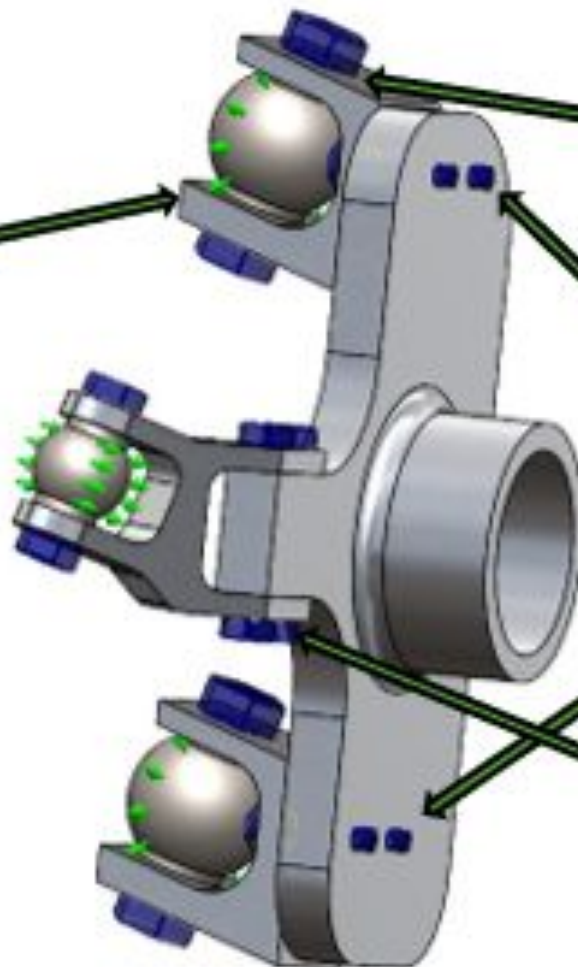
Load Diagram:



Constraints:

Radial Constraint acting on the outside face of the control Hiems. This locks all translation and is an over constraint for the top hiem as it does allow for up-down movement.

A translation constraint in the direction of the tie rod representing the 2 force member. This Acts on the outside face of the Hiem.



Bolt connections for $\frac{1}{2}$ -20 bolts with 144in-lbf and bond interaction between the Heim joint and the bracket. This is because of friction.

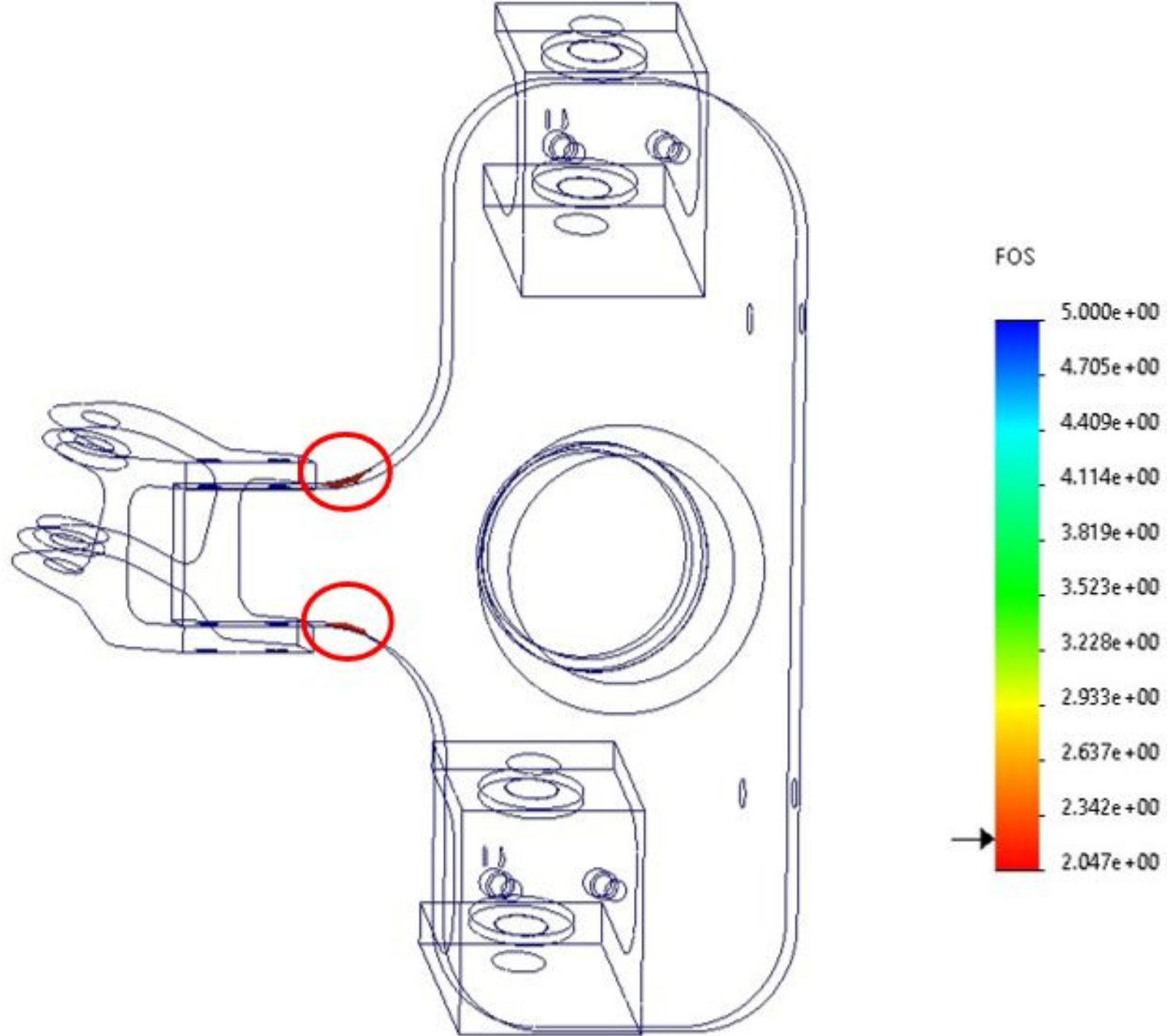
Bolt connections for $\frac{1}{4}$ -20 bolts with 144 in-lbf and local interaction between the back face of the bracket and upright

Bolt connections for 10-32 bolts with 50 in-lbf and local interaction between the back face of the bracket and upright

FOS: 2.05

Equations (If Applicable):

Confidence Level & Explanation:Medium: Numbers are gathered from previous FEA but lack supporting test data for loading force.

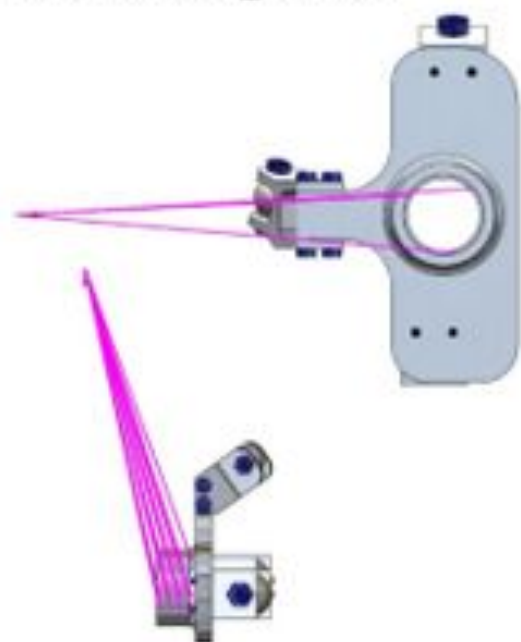


Failure Condition: Frontal impact on the outermost point of the front of 1 front tire stopping the vehicle with added braking.

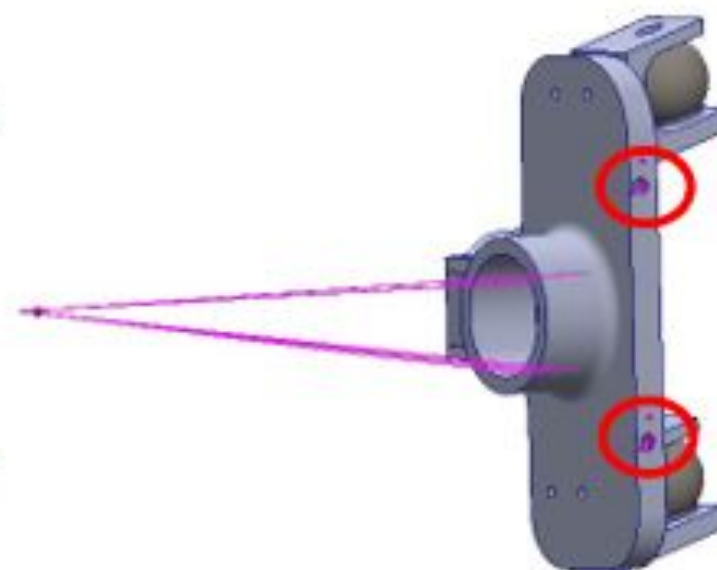
Material: 6061-T6 Aluminum upright (0.59" thick) and brackets (0.25" thick top and back with 0.375" thick bottom), Heim ball joints are 4130 Chromoly only input for BC's.

Justification: Frontal loading case derived from pervious FEA With the added condition of make braking torque.

Load Diagram:



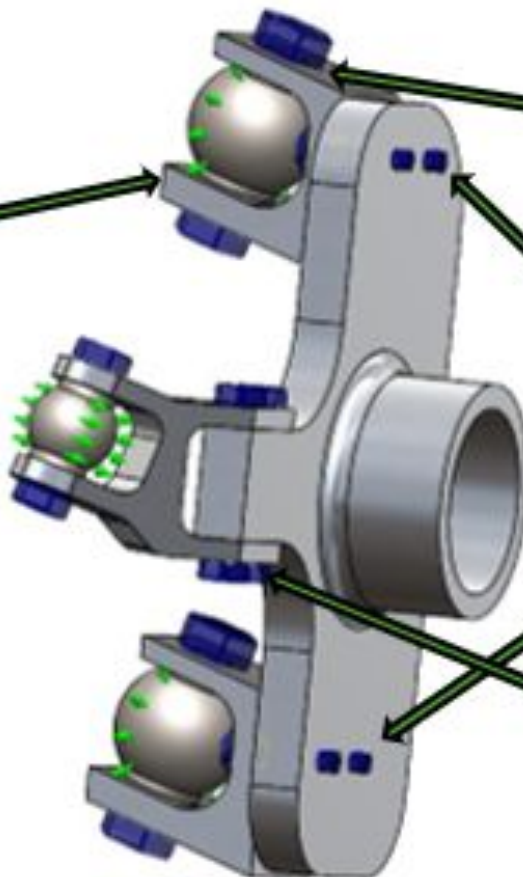
400lbf Applied outmost corner of the front of the tire (4in out and 11.5 forward from the center of the bearing surface). Also with 650lbf bearing force slit between the brake bracket bolts.



Constraints:

Radial Constraint acting on the outside face of the control Hiems. This locks all translation and is an over constraint for the top hiem as it does allow for up-down movement.

A translation constraint in the direction of the tie rod representing the 2 force member. This Acts on the outside face of the Hiem.



Bolt connections for $\frac{1}{2}$ -20 bolts with 144 in-lbf and bond interaction between the Heim joint and the bracket. This is because of friction.

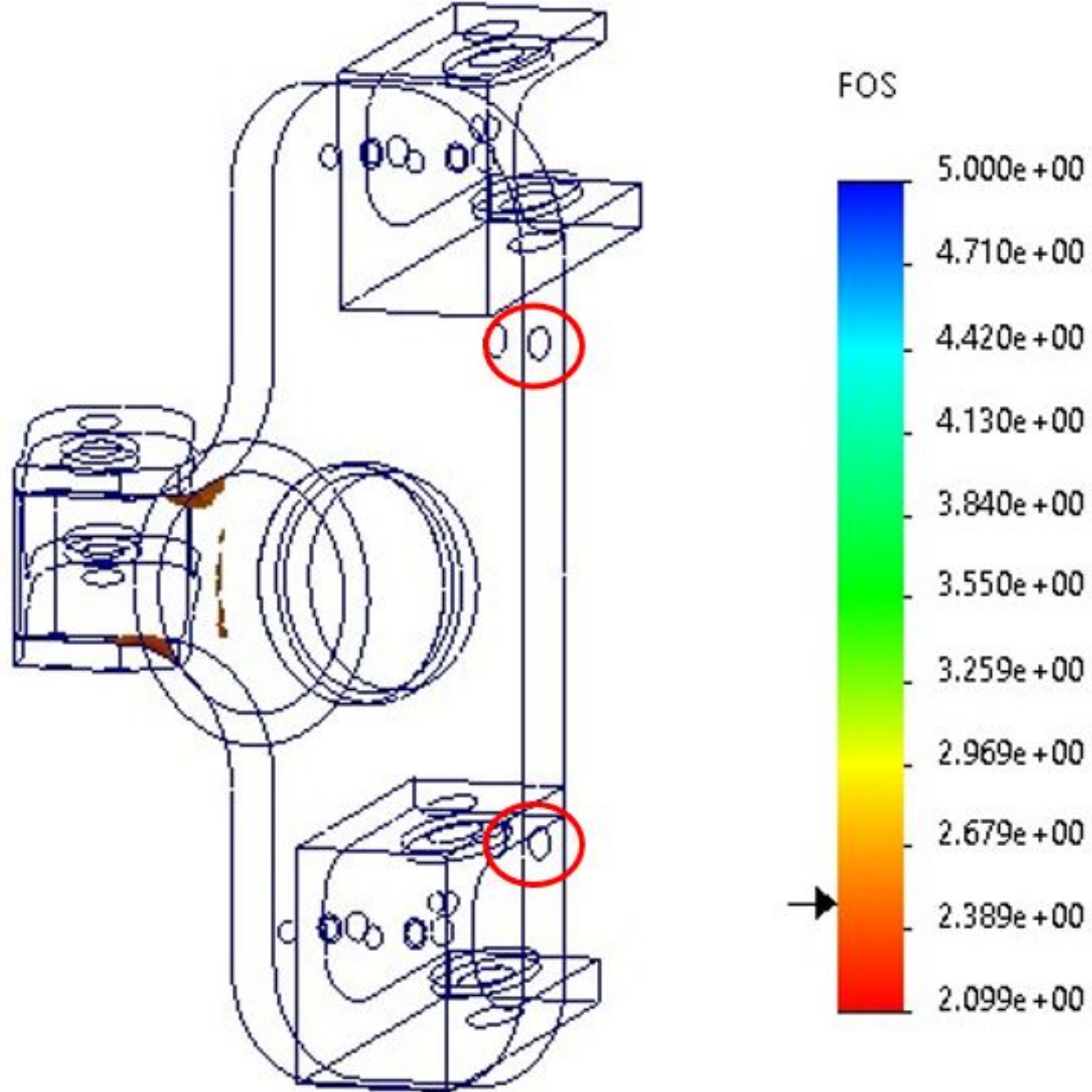
Bolt connections for $\frac{1}{4}$ -20 bolts with 144 in-lbf and local interaction between the back face of the bracket and upright

Bolt connections for 10-32 bolts with 50 in-lbf and local interaction between the back face of the bracket and upright

FOS: 2.1

Equations (If Applicable):

Confidence Level & Explanation: Medium: Numbers are gathered from previous FEA but lack supporting test data for loading force. Brake calculations are derived from max braking torque described in controls section 3.2.4.

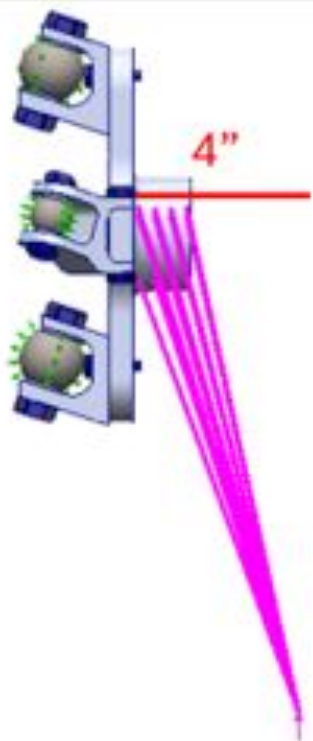


Failure Condition: Approximately a 2G load (1000lbf) loaded on the outmost corner of the bottom of the tire.

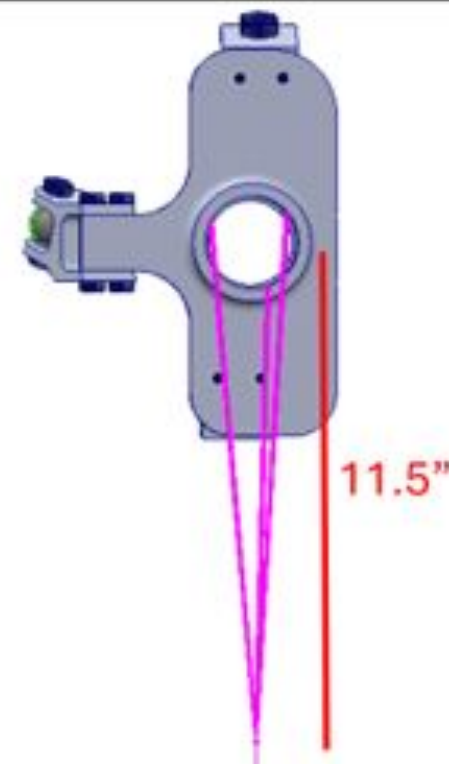
Material: 6061-T6 Aluminum upright (0.59" thick) and brackets (0.25" thick top and back with 0.375" thick bottom), Heim ball joints are 4130 Chromoly only input for BC's.

Justification: This is a worse case loading if the vehicle where to landed a jump on one wheel of the front wheels.

Load Diagram:



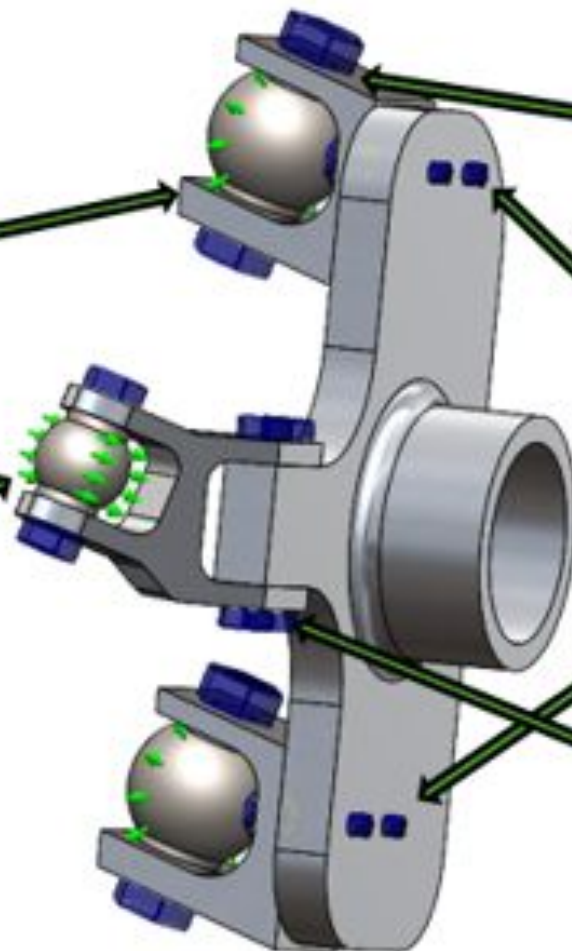
1000lbf Applied outmost corner of the bottom of the tire (4in out and 11.5 downward from the center of the bearing surface)



Constraints:

Radial Constraint acting on the outside face of the control Hiems. This locks all translation and is an over constraint for the top hiem as it does allow for up-down movement.

A translation constraint in the direction of the tie rod representing the 2 force member. This Acts on the outside face of the Hiem.



Bolt connections for $\frac{1}{2}$ -20 bolts with 144in-lbf and bond interaction between the Heim joint and the bracket. This is because of friction.

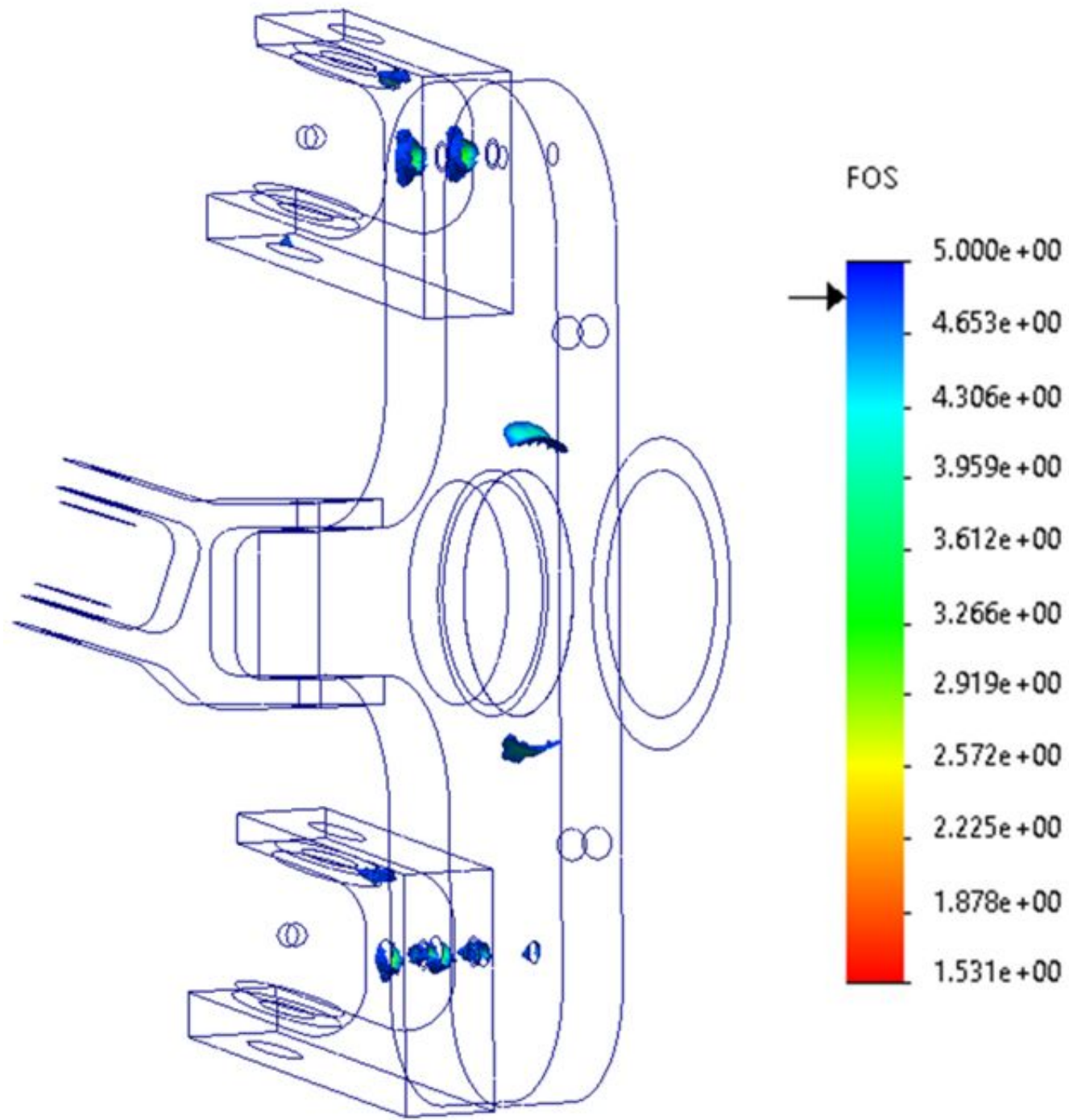
Bolt connections for $\frac{1}{4}$ -20 bolts with 144 in-lbf and local interaction between the back face of the bracket and upright

Bolt connections for 10-32 bolts with 50 in-lbf and local interaction between the back face of the bracket and upright

FOS: 1.5

Equations (If Applicable):

Confidence Level & Explanation: Low: Loads are not well known. Also Due to SolidWorks limitations the model is over defined in the top Hiem explained above.

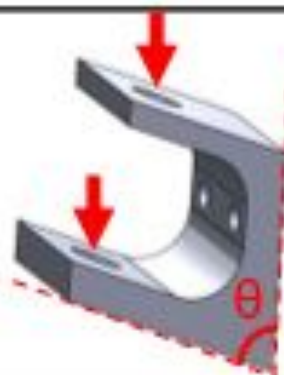


Failure Condition : Max force of shock acting on control arm translating to the bracket.

Material: 6061-T6 Aluminum (0.25" thick top and back with 0.375" thick bottom).

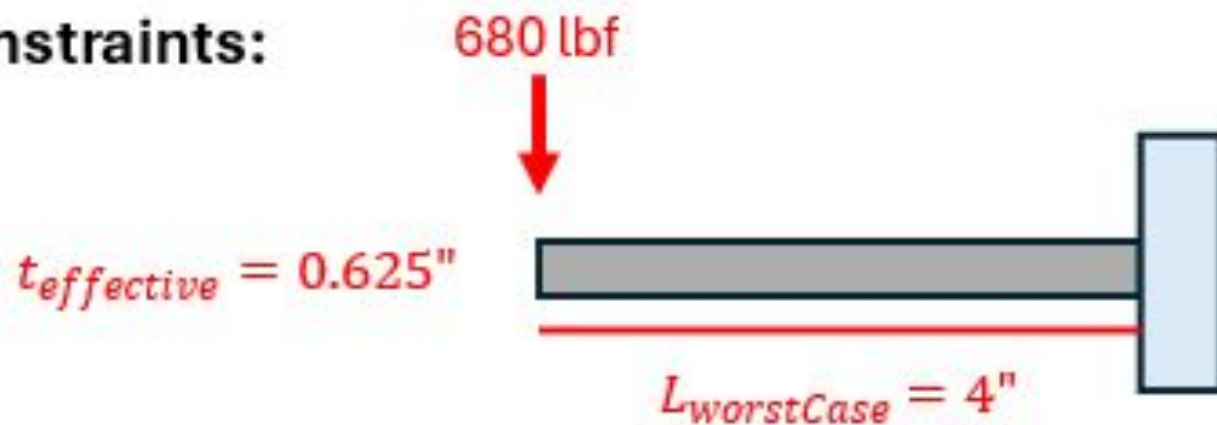
Justification: Compression loading from a jump or large bump acting on the bracket

Load Diagram:



680lbf total downward
perpendicular on the two the
bracket arms. This was derived from
a max force of the shock at 1600lbf

Constraints:



Simple beam calculation assume
worse case length, force, and no
filet.

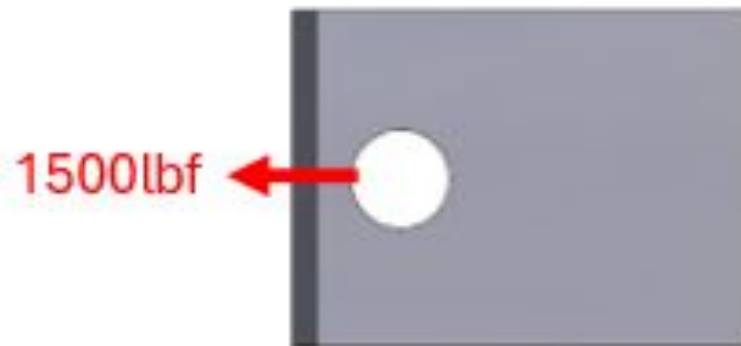
FOS: 5 Equations (If Applicable): $F_{\text{bracket}} = \frac{F_{\text{shock}} L_{\text{shock}} \cos(\theta)}{L_{\text{arm}}}$ $M = F_{\text{bracket}} \cdot L$ $I = \frac{bh^3}{12}$ $\sigma_b = \frac{Mc}{I}$ $FS = \frac{\sigma_y}{\sigma_b}$	Confidence Level & Explanation:medium, Deriving the maximum force from FOX shocks data and previous years using the same value without failure.
Calculations and/or FEA: --- Aluminum Bracket Calculations --- Moment of Inertia (I): 0.035604 in^4 Applied Moment: 1125.25 lb·in Bending Stress: 9876.44 psi Safety Factor (Bending): 3.54	

Failure Condition: a 3000lbf force split equally between both control arm brackets

Material: 6061-T6 Aluminum (0.25" thick top and back with 0.375" thick bottom).

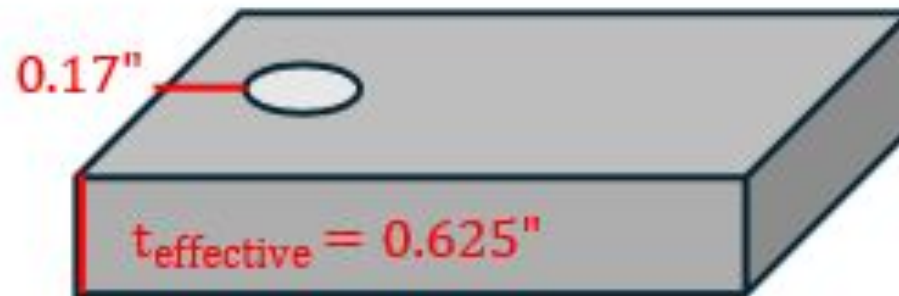
Justification: A Frontal impact translated directly to the bracket or side impact causing a moment on the upright and translated to the bracket.

Load Diagram:



3000lbf total load in the worst case direction

Constraints:



Combine Bracket thicken using the worsted case distance from edge

FOS: 5 Equations (If Applicable): $F = \frac{F_{\text{total}}}{n_{\text{brackets}}}$ $\sigma_{\text{tear}} = \frac{F}{A}$ $FS_{\text{tear}} = \frac{\sigma_y}{\sigma_{\text{tear}}}$	Confidence Level & Explanation:Medium, Using previous years numbers with 3000lbf frontal impact, they have never experienced failure.
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Calculations and/or FEA:

--- Aluminum Bracket Calculations ---

Tear-Out Area: 0.2125 in²

Force per Bracket Side: 1500.00 lb

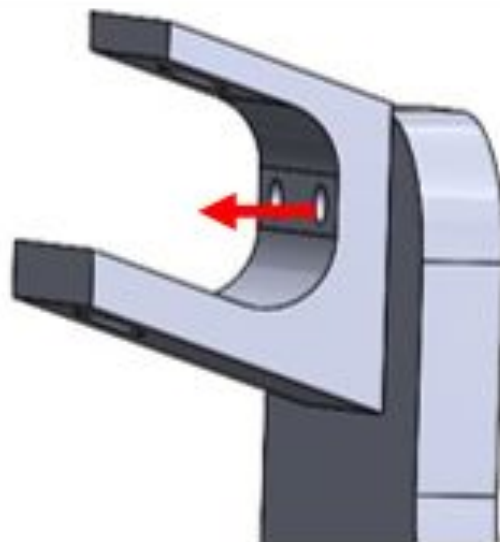
Tear-Out Stress: 7058.82 psi

Safety Factor (Tear-Out): 4.96

Failure Condition : A torque or side impact causing a moment pulling on the bolts
Material: 6061-T6 Aluminum (0.59").

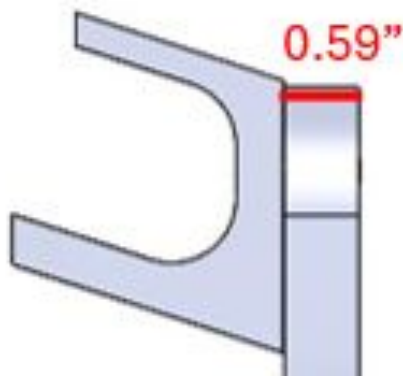
Justification: A Frontal impact translated directly to the bracket or side impact causing a moment on the upright and translated to the bracket.

Load Diagram:



No force is applied in this direction
instead we are calculating make
strength of thread pull out

Constraints:



The upright which is what we are
threading into is assumed to be
ridged and 0.59in thick.

FOS: N/A Equations (If Applicable): $A_{\text{shear}} = \pi D_{\text{avg}} L_{\text{eng}}$ $F_{\text{tear-out}} = \tau_{\text{allow}} \times A_{\text{shear}}$	Confidence Level & Explanation : High: All values of material and their strengths are well know and tested throughout industry.
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Calculations and/or FEA:

THREAD TEAR-OUT CALCULATION RESULTS:

Bolt Size:	1/4"-20 UNF
Major Diameter:	0.2500 in
Minor Diameter:	0.1887 in
Engagement Length:	0.590 in
Average Diameter:	0.2193 in
Shear Area:	0.4066 in ²
Tear-Out Strength:	9758 lb

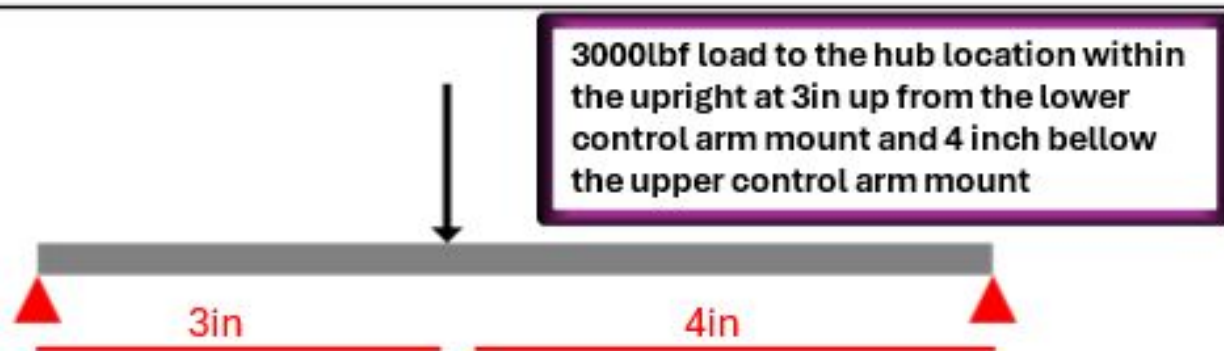
Front Suspension Control Arm Analysis



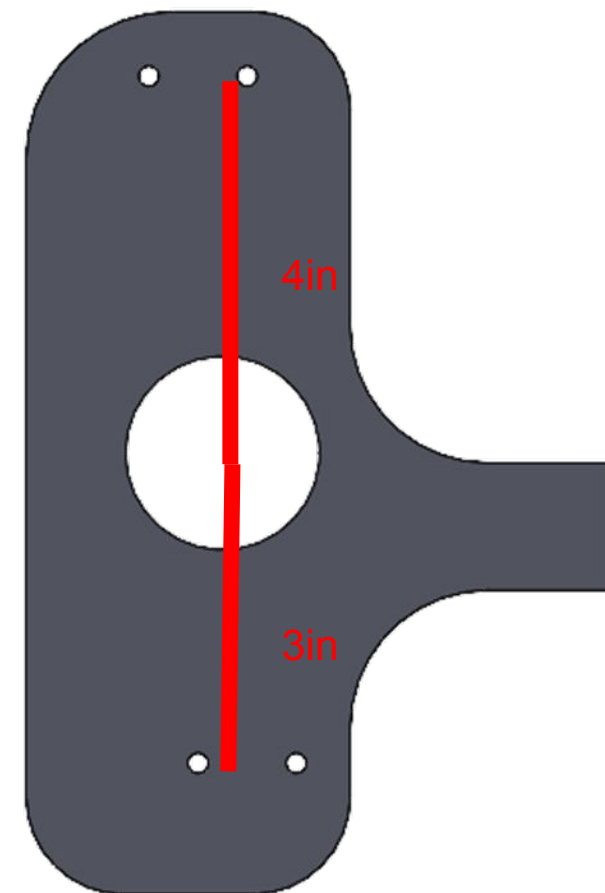
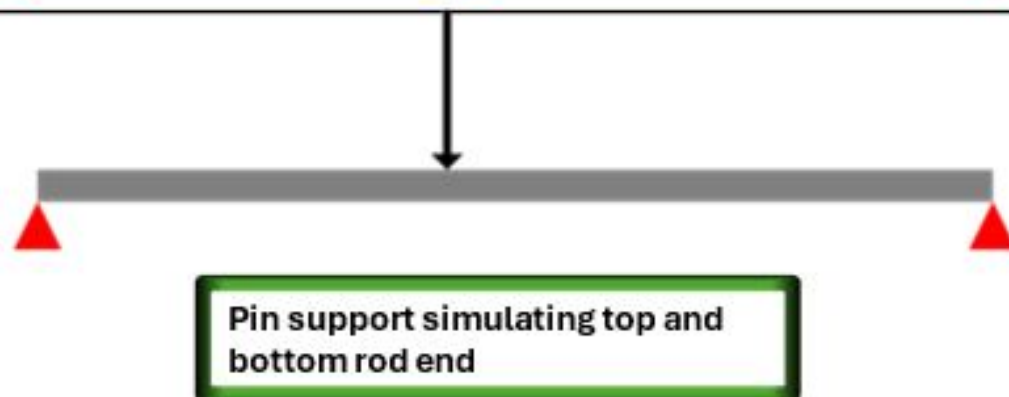
Failure Condition: Upright frontal load distribution to top and bottom control arms

Justification: with the hub not being centered between the upper and lower control arms the weight is not evenly distributed to each arm.

Load Diagram:



Constraints:



Front Left Upright

FOS: N/A

Equations (If Applicable):

Confidence Level & Explanation: Medium, **45**

Using previous years numbers with 3000lbf frontal impact, they have never experienced failure.

Calculations and/or FEA:

Shear Diagram

(Max +ve) Shear Load (lb): 1714.286,

Location (in): 0.000, 3.000

(Max -ve) Shear Load (lb): -1285.714,

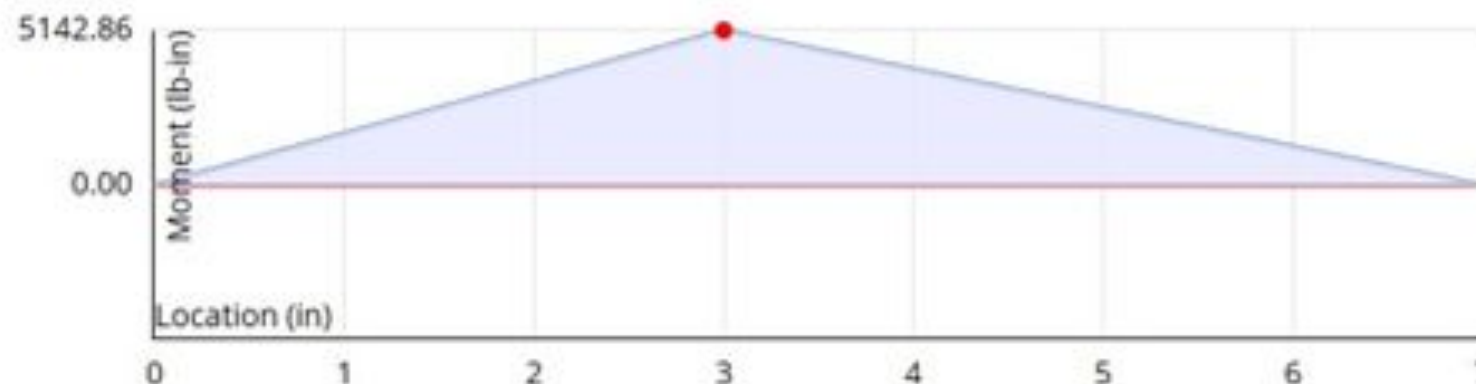
Location (in): 3.000, 7.000



Moment Diagram

(Max +ve) Moment Load (lb-in): 5142.857,

Location (in): 3.000



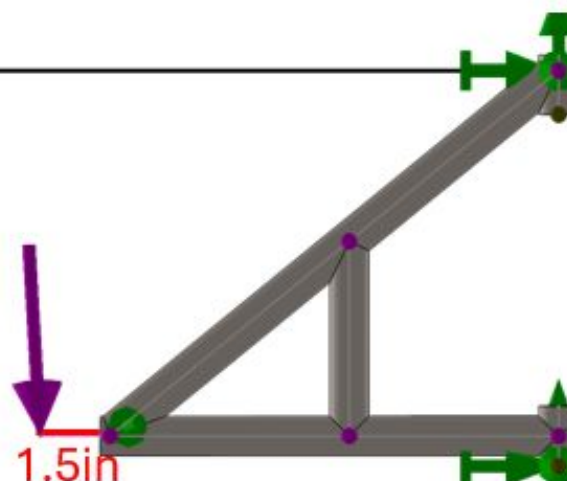
Failure Condition: Frontal impact on the front of the wheel

Material: Alloy Steel 4130 1" OD x 0.058 Wall

Justification: when we experience frontal obstacles at speed, part of the force will be translated into the control arms.

Load Diagram:

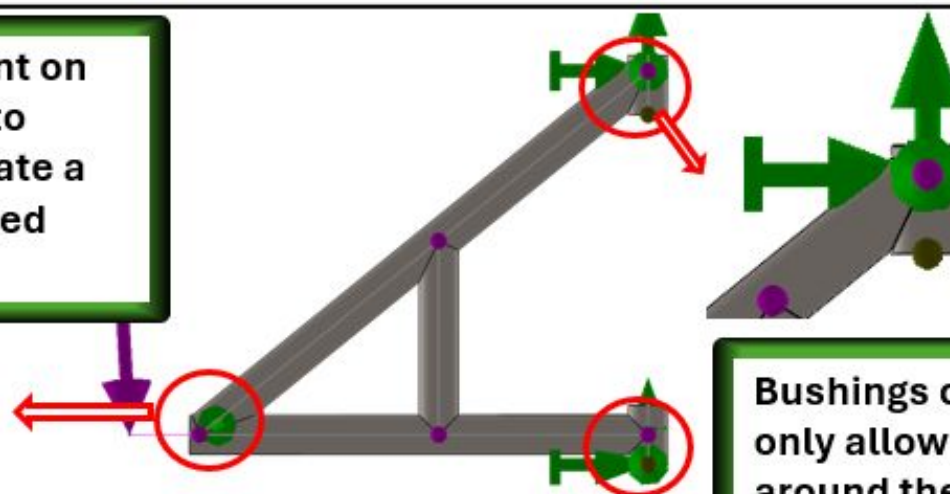
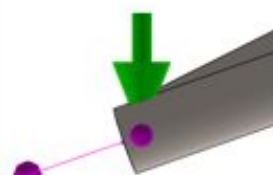
1350lbf applied to the outer most point of the upper control arm, 1.5 inches out where the center of the rod end would be.



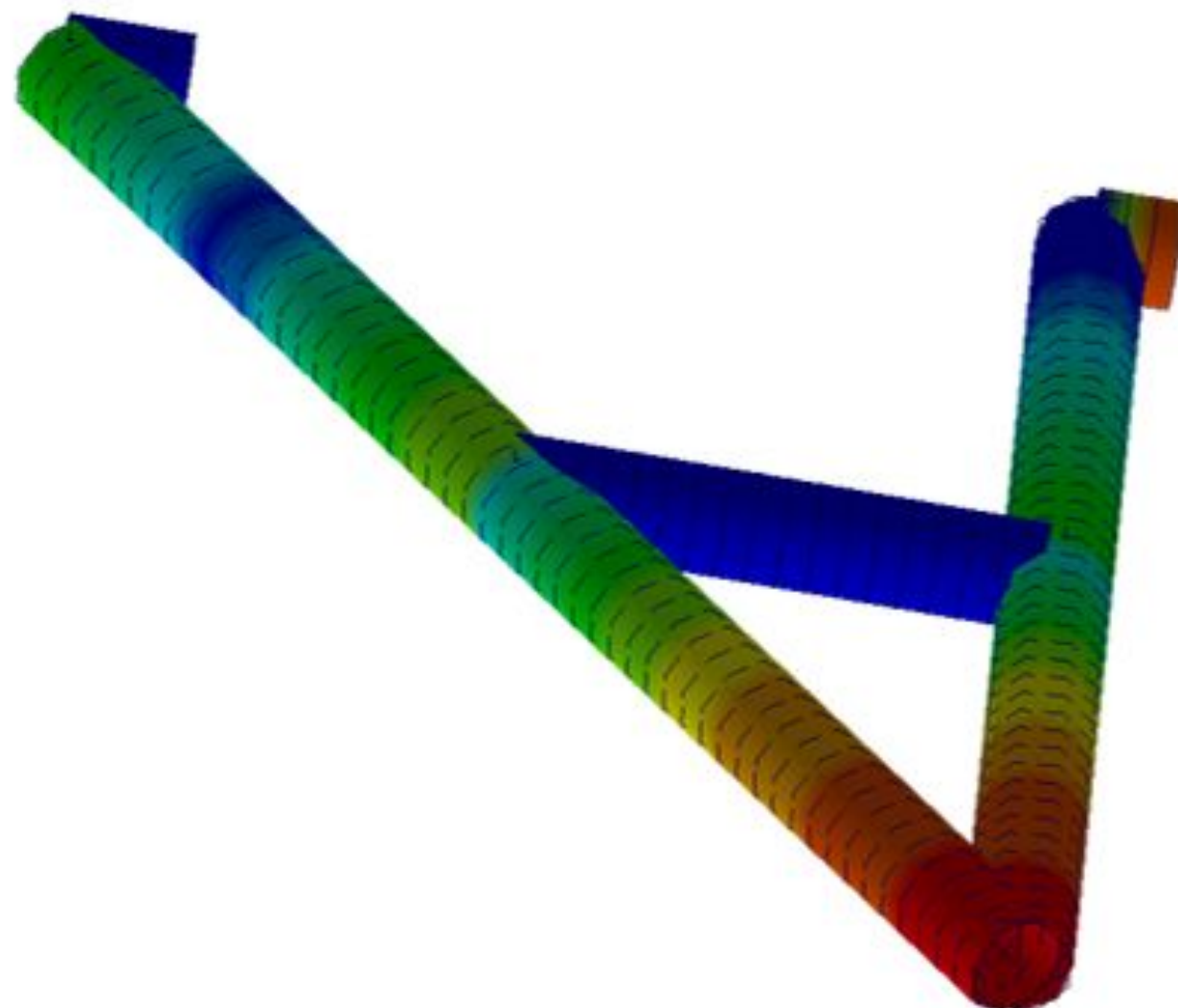
Constraints:

Single translation constraint on the end of the control arm to prevent rotation and simulate a ridged shock force translated through the upright

Rotated front view of Rod end connection



Bushings constrained to only allow rotation in axis around the bushing.

FOS: 1.921**Equations (If Applicable):****Confidence Level & Explanation:** Medium,
Using previous years numbers with 3000lbf
frontal impact, they have never experienced
failure.**Calculations and/or FEA:****FOS**

5.000e+00

4.672e+00

4.343e+00

4.015e+00

3.686e+00

3.358e+00

3.029e+00

2.701e+00

2.372e+00

2.044e+00

1.716e+00

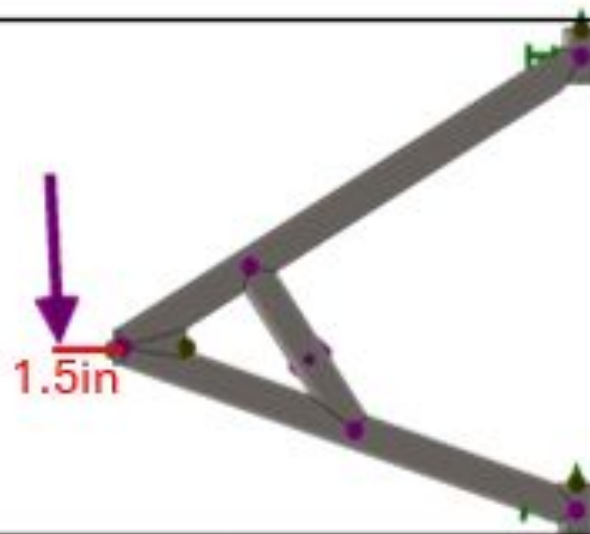
Failure Condition: Frontal impact on the front of the wheel

Material: Alloy Steel 4130 1"OD x 0.095"Wall, Crossmember 1"OD x 0.058"Wall

Justification: when we experience frontal obstacles at speed, part of the force will be translated into the control arms.

Load Diagram:

1750lbf applied to the outer most point of the upper control arm, 1.5 inches out where the center of the rod end would be.

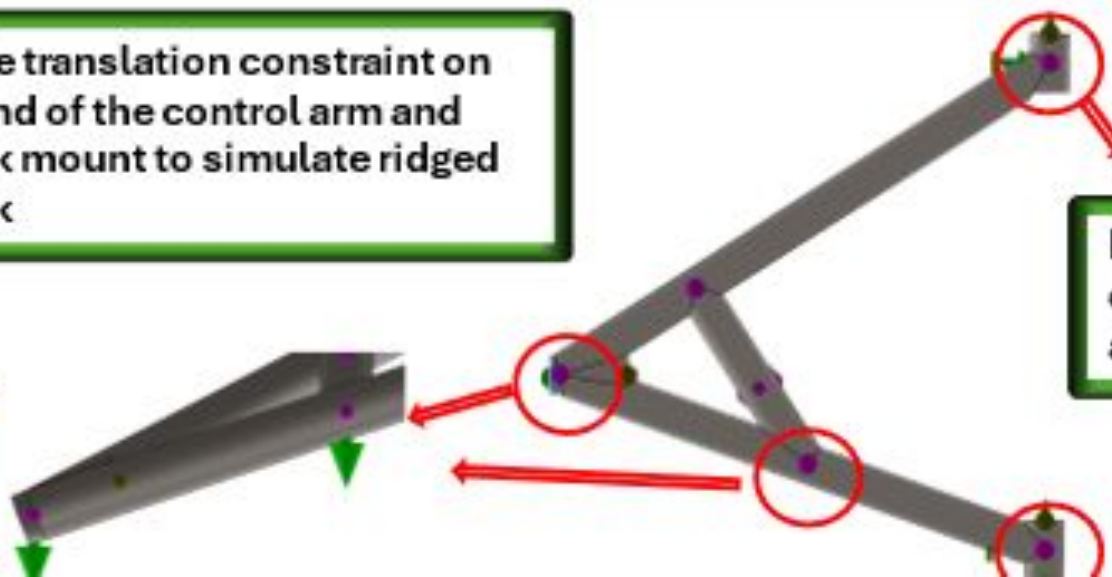


Constraints:

Single translation constraint on the end of the control arm and shock mount to simulate ridged shock

Bushings constrained to only allow rotation in axis around the bushing.

Rotated front view of Rod end connection



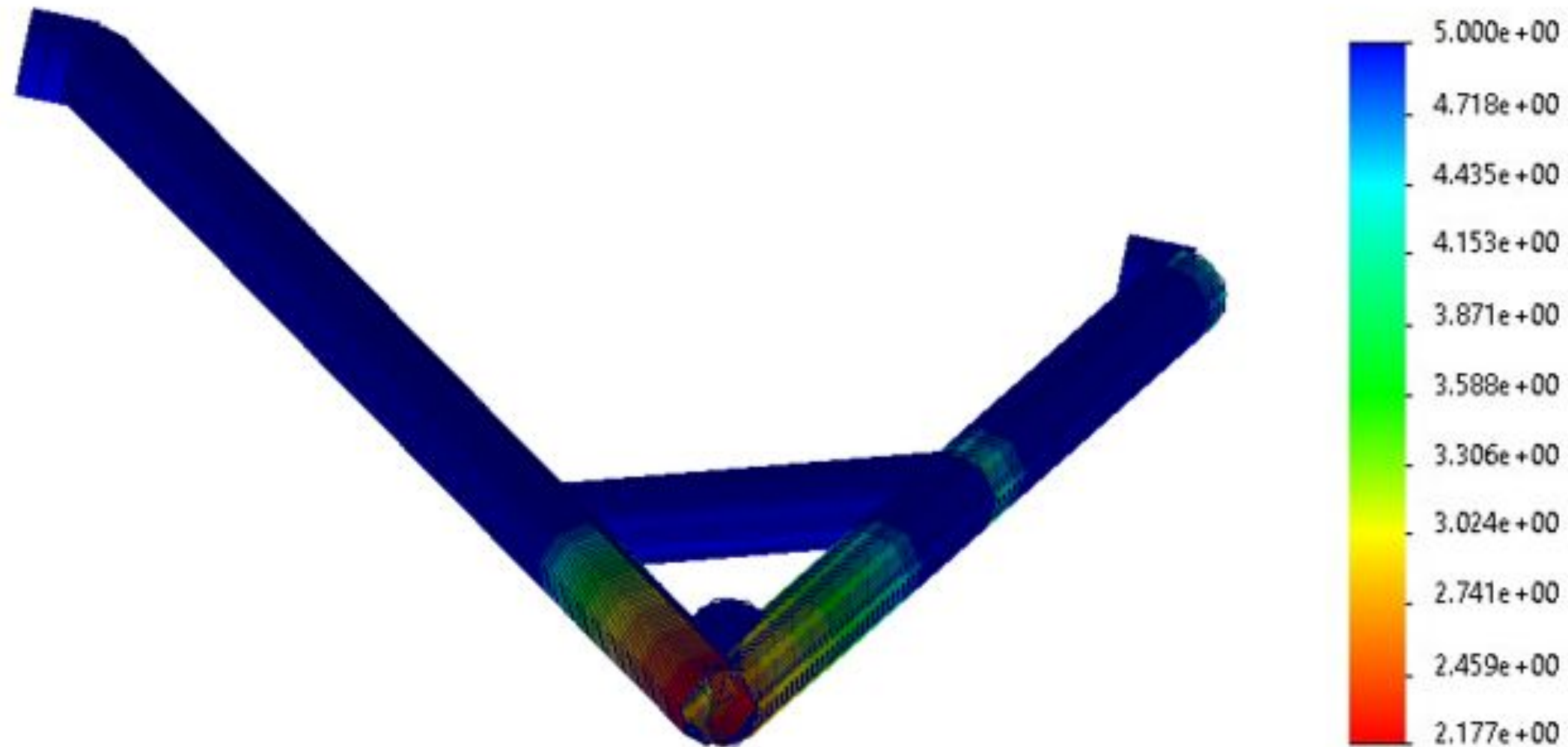
FOS: 2.177

Equations (If Applicable):

Confidence Level & Explanation: Medium,
Using previous years numbers with 3000lbf
frontal impact, they have never experienced
failure.

49

Calculations and/or FEA:



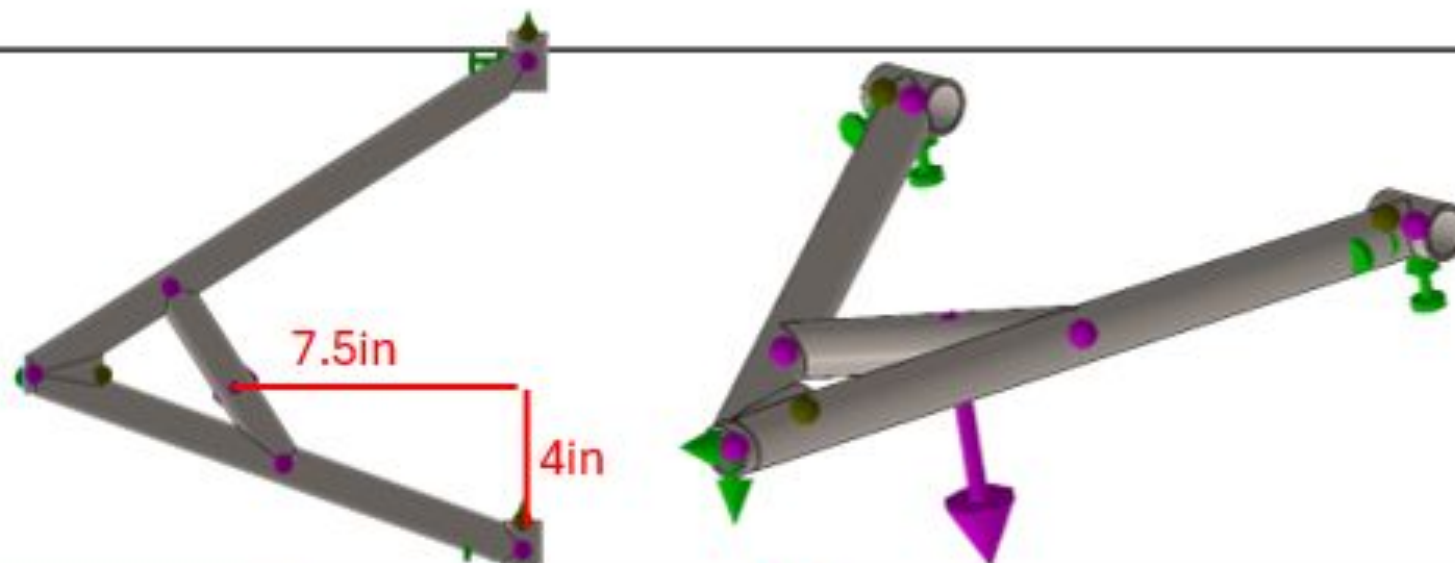
Failure Condition: Arm failure due to shock loading

Material: Alloy Steel 4130 1"OD x 0.095"Wall, Crossmember 1"OD x 0.058"Wall

Justification: using data form Fox Shocks we have derived that the maximum force exerted by the shock is 1600lbf.

Load Diagram:

1600lbf is applied perpendicular to the arm on the crossmember

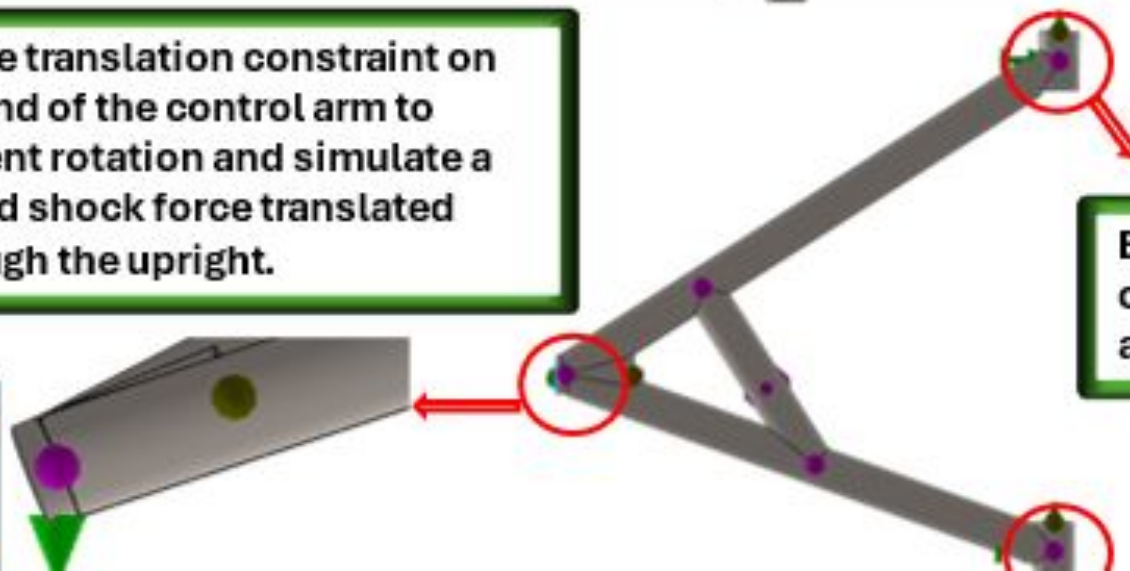


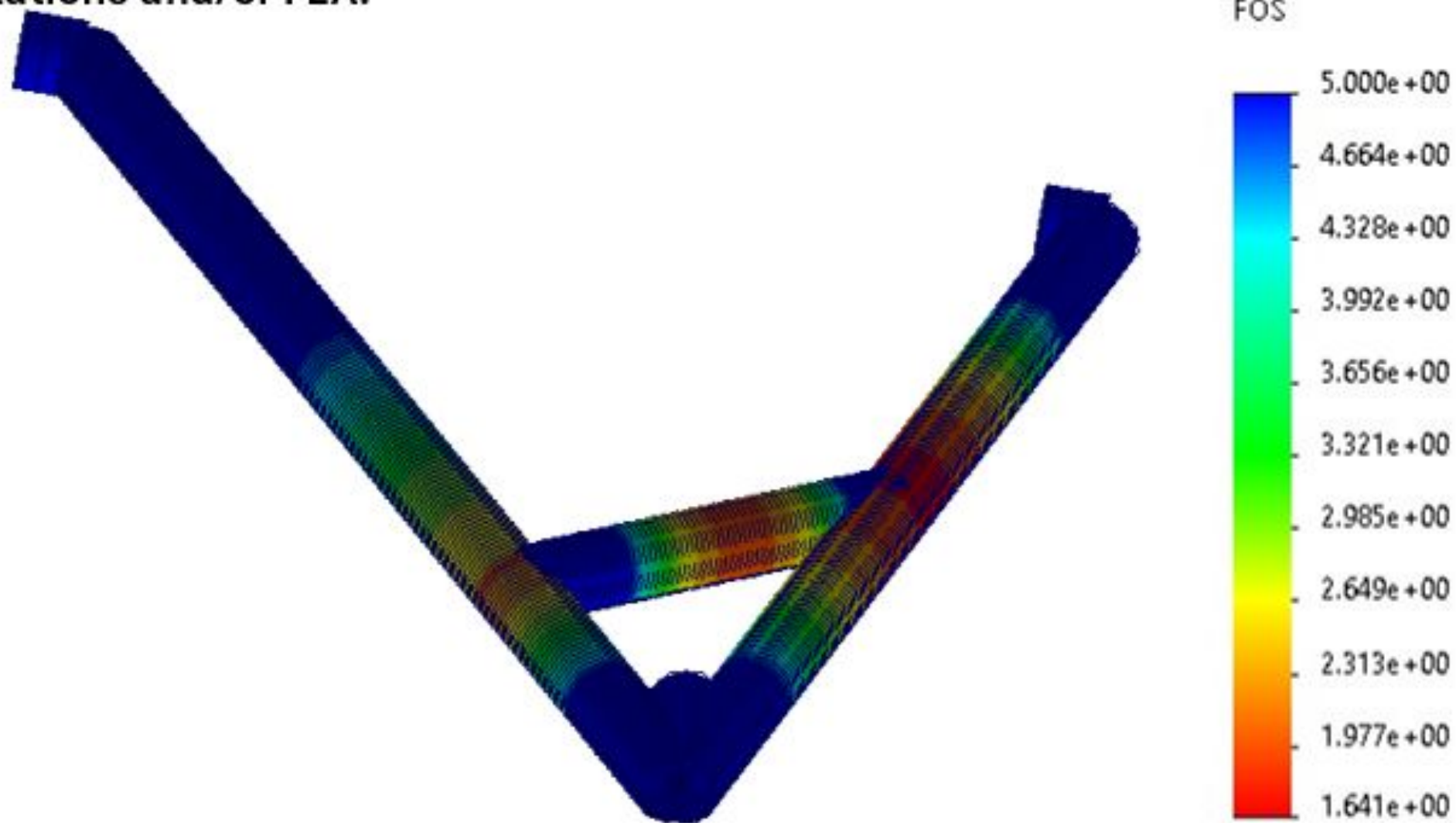
Constraints:

Single translation constraint on the end of the control arm to prevent rotation and simulate a rigid shock force translated through the upright.

Bushings constrained to only allow rotation in axis around the bushing.

Rotated front view of Rod end connection



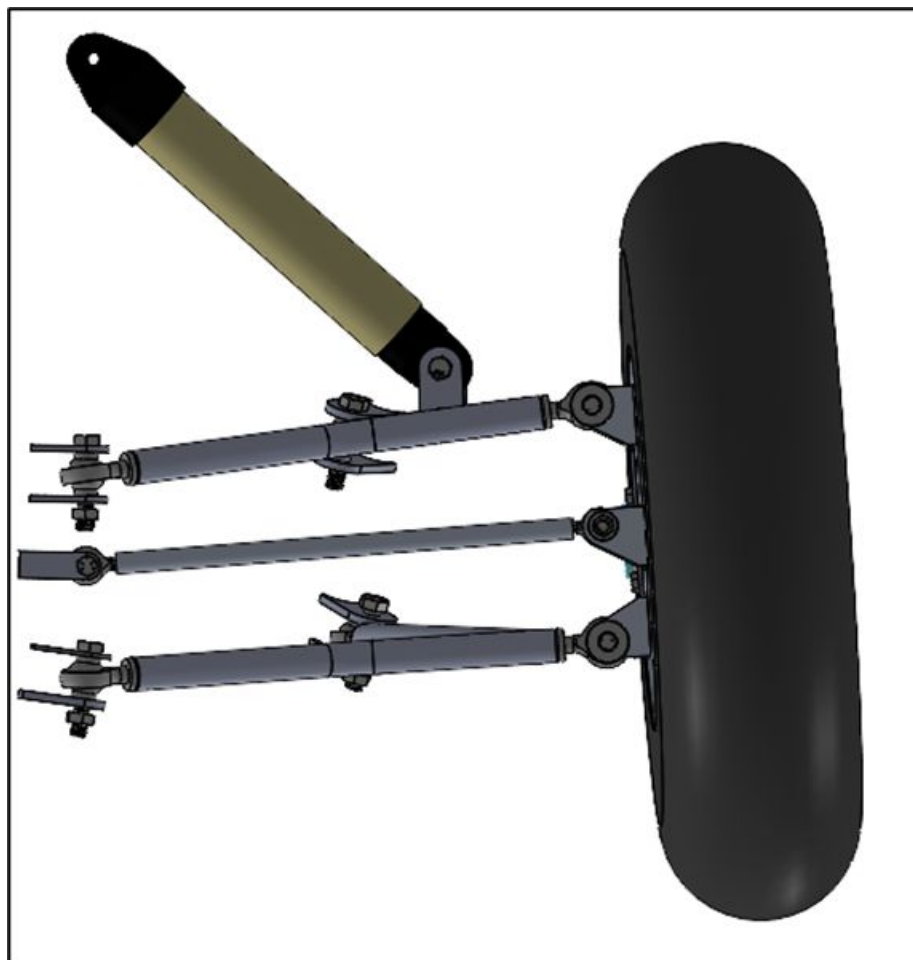
FOS: 1.641**Equations (If Applicable):****Confidence Level & Explanation:**medium,
Deriving the maximum force from FOX
shocks data and previous years using the
same value without failure.**Calculations and/or FEA:**

Rear Suspension Design

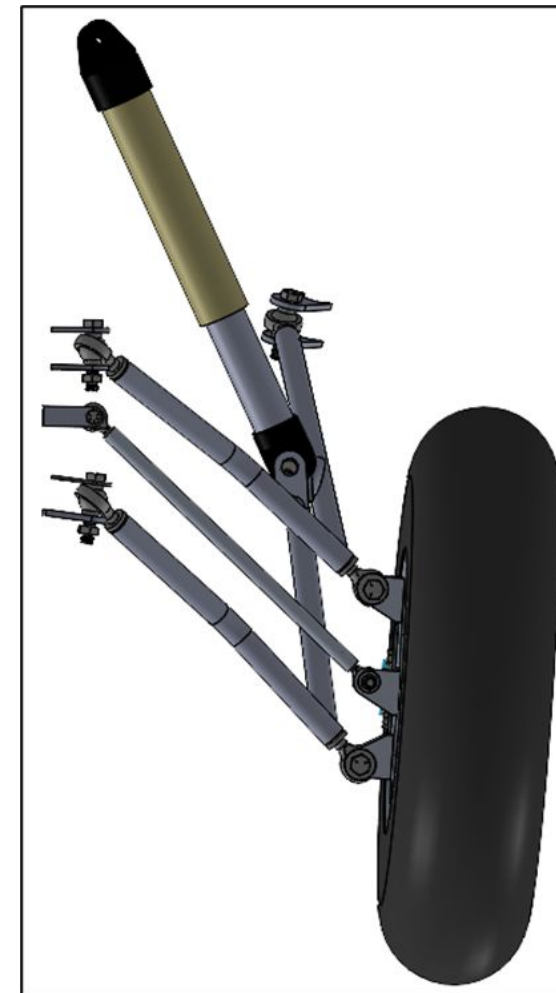
Rear Right Views from Back



Ride Height



Full Bump



Full Droop

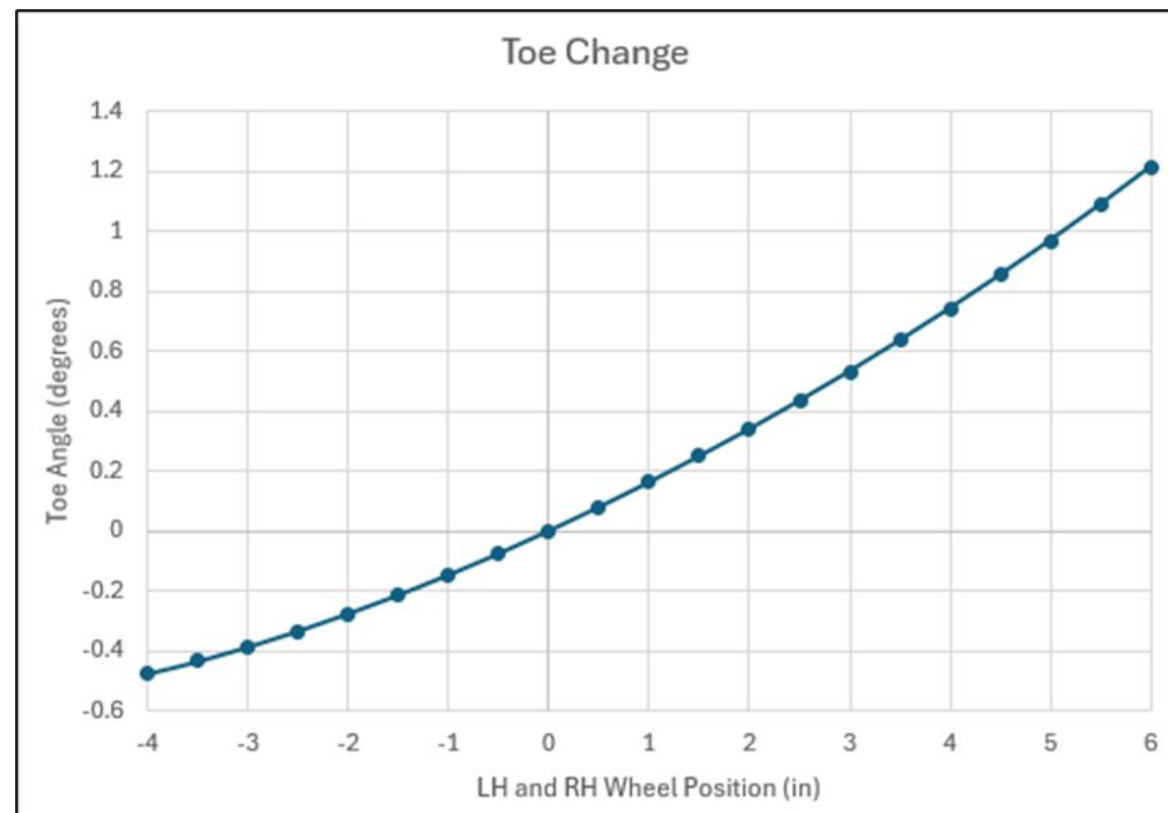
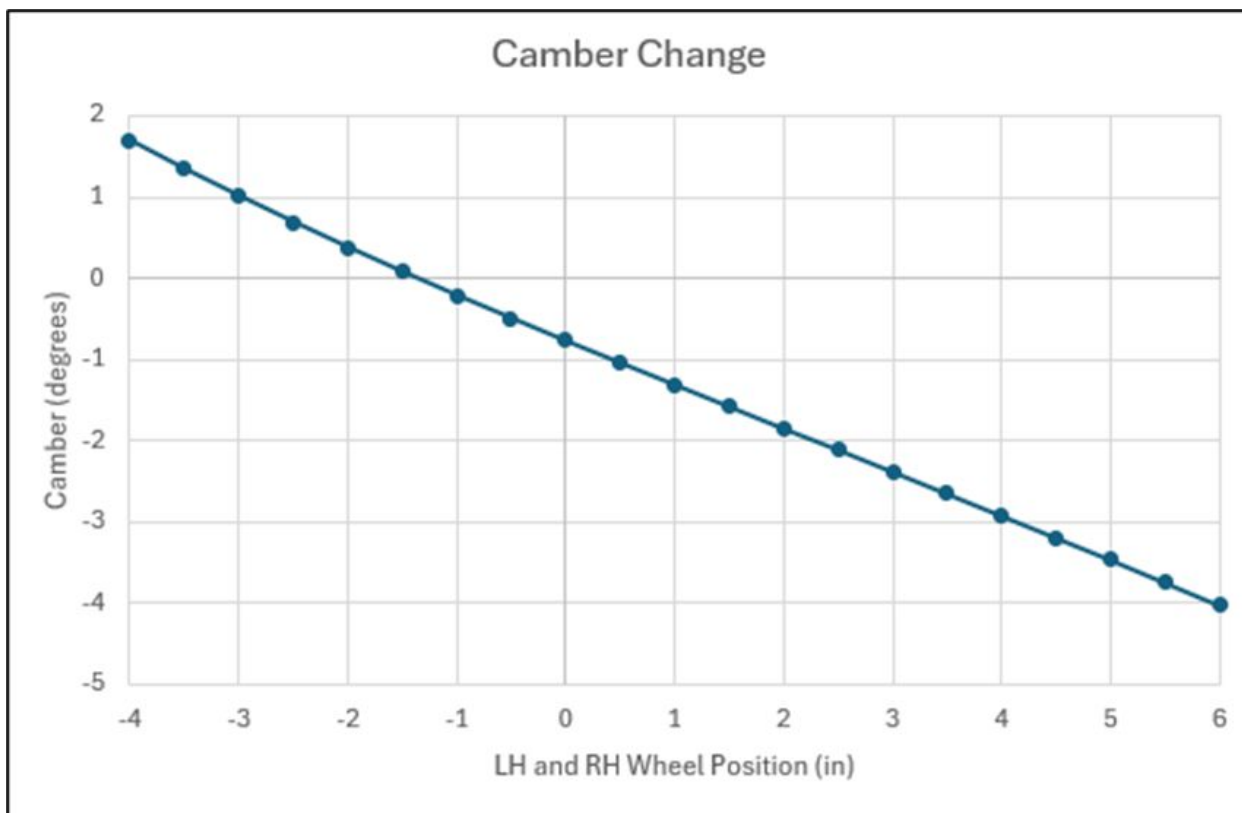
Rear Suspension Analysis

Spec	This Year	Last Year
Roll Center Below Frame	2.3 in	6 in
Total Width	58 in	54 in
Travel	10 in	12 in
Ground	12 in	12 in
Static Camber	-1°	-7°
Progressive Camber	1.7 to -4°	0.48°
Shock Angle	~52.4°	~90°
Toe Change	-0.46 to 1.22°	1.7°

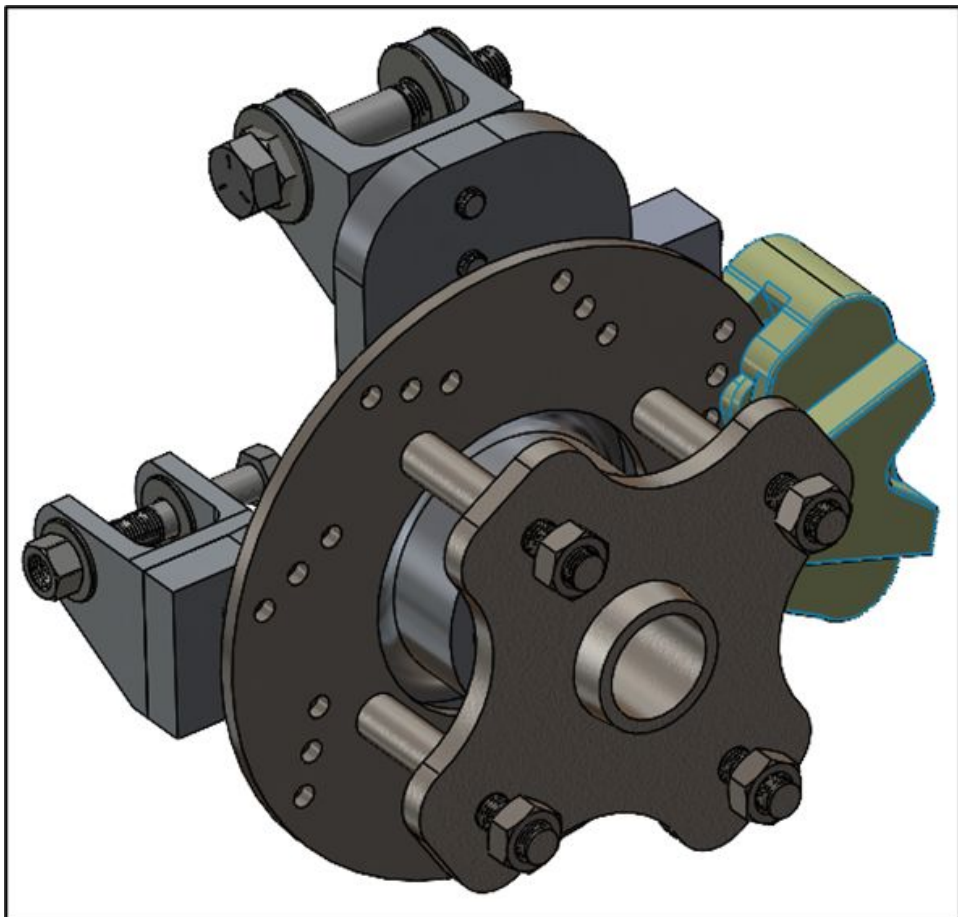
Rear Suspension Kinematics

LH and RH wheel	Drive Shaft Length	Plunge [in]	Inner CV Joint Angle	Outer CV Joint Angle
6 bump	17.03	-0.144	7.74	6.54
5.5 bump	17.03	-0.145	6.14	5.55
5 bump	17.03	-0.144	4.56	4.76
4.5 bump	17.03	-0.141	3.05	4.31
4 bump	17.03	-0.135	1.72	4.29
3.5 bump	17.03	-0.127	1.38	4.72
3 bump	17.03	-0.117	2.47	5.49
2.5 bump	17.03	-0.104	3.95	6.48
2 bump	17.03	-0.089	5.51	7.61
1.5 bump	17.03	-0.071	7.11	8.83
1 bump	17.03	-0.05	8.73	10.11
0.5 bump	17.03	-0.027	10.37	11.43
Static	17.03	0	12.02	12.79
0.5 droop	17.03	0.03	13.68	14.16
1 droop	17.03	0.063	15.36	15.56
1.5 droop	17.03	0.1	17.06	16.98
2 droop	17.03	0.141	18.77	18.41
2.5 droop	17.03	0.187	20.5	19.86
3 droop	17.03	0.238	22.26	21.32
3.5 droop	17.03	0.295	24.04	22.79
4 droop	17.03	0.358	25.85	24.28

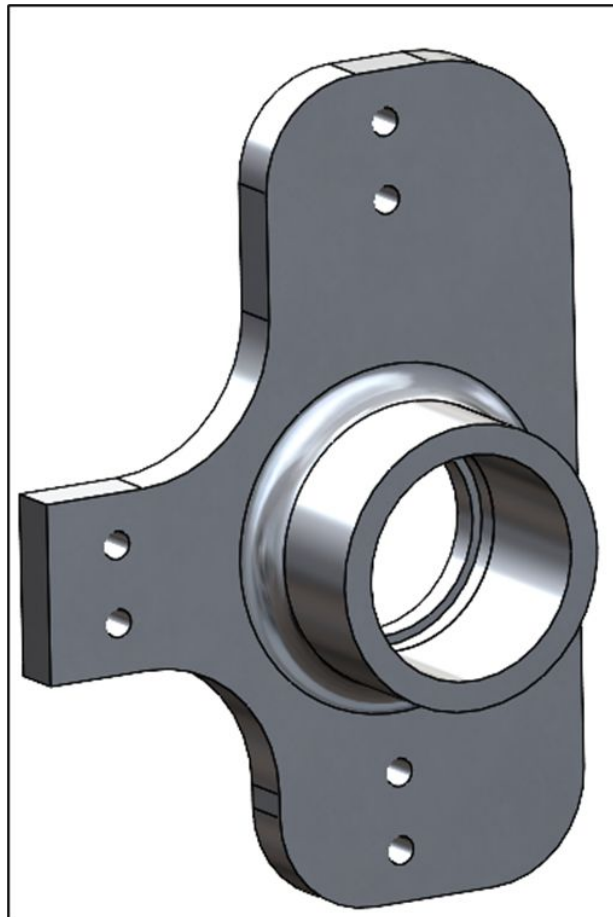
Rear Suspension Kinematics



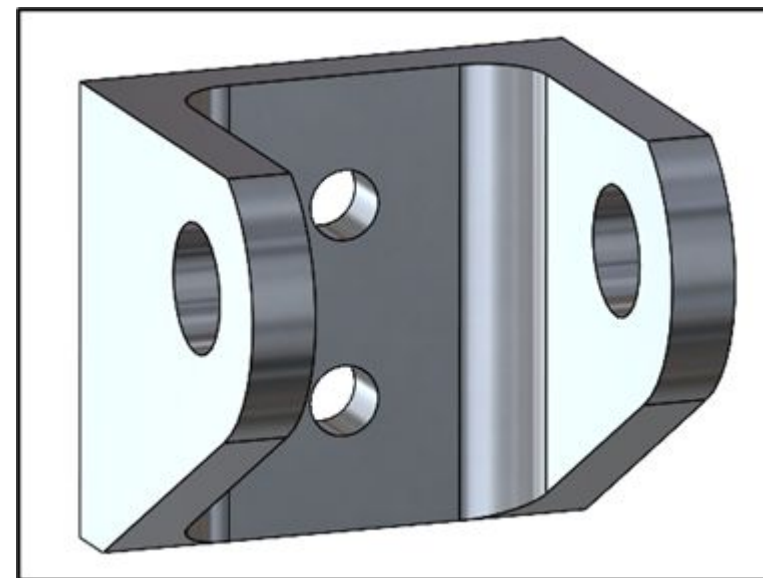
Rear Suspension Upright



Hub Assembly



Upright



Control Arm Bracket

Rear Suspension Upright FEA

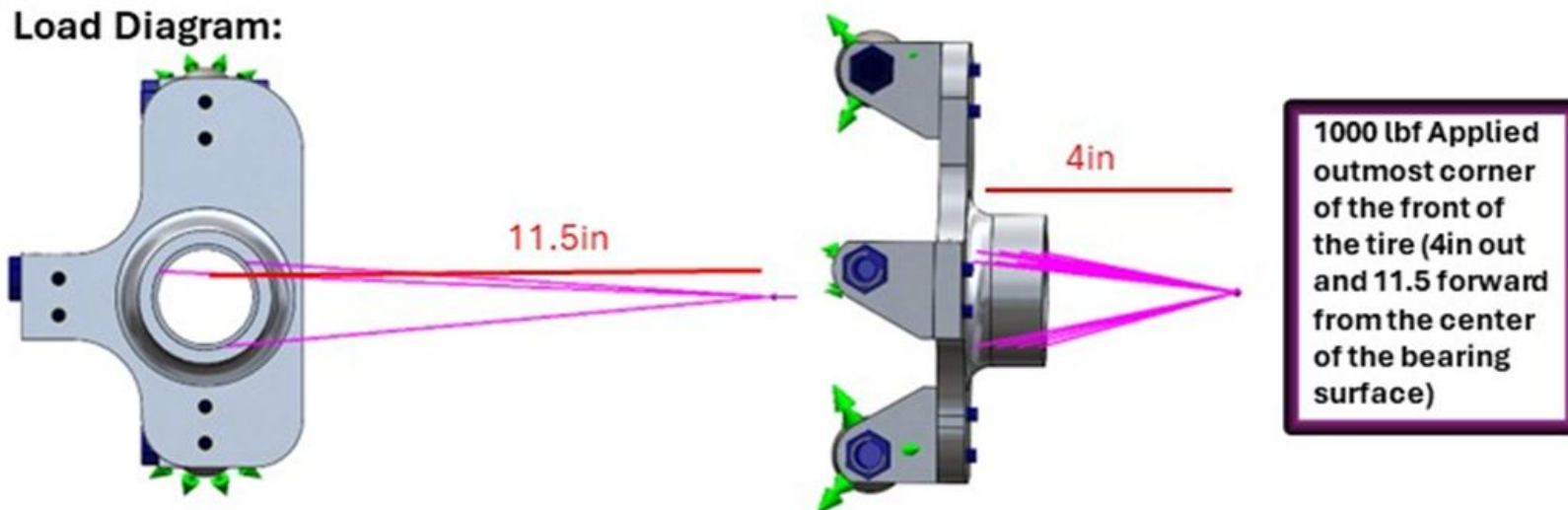
Failure Condition: Frontal impact on the outermost point of the front of 1 front tire stopping the vehicle.

Material: 6061-T6 Aluminum upright (0.5" thick) and brackets (0.25" thick), Heim ball joints are 4130 Chromoly only input for boundary condition simulations

Justification:

This is the upright experiencing approximately a 2g load from hitting an obstacle head on.

Load Diagram:

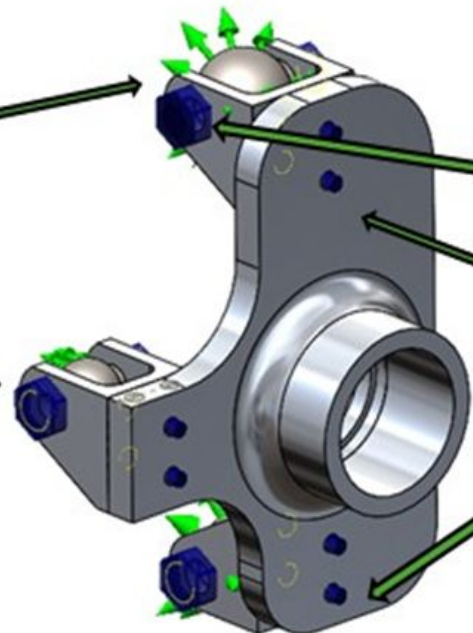


Rear Suspension Upright FEA

Constraints:

Radial constraint acting on the outside face of the control arm Hiems. This locks all translation and is an over constraint for the top hiem as it does allows for up-down movement.

A translation constraint in the direction of the toe rod representing the 2 force member. This acts on the outside face of the Hiem.



Bolt connections for $\frac{1}{2}$ -20 bolts with 144in-lbf and bond interaction between the Heim joint and the bracket due to friction

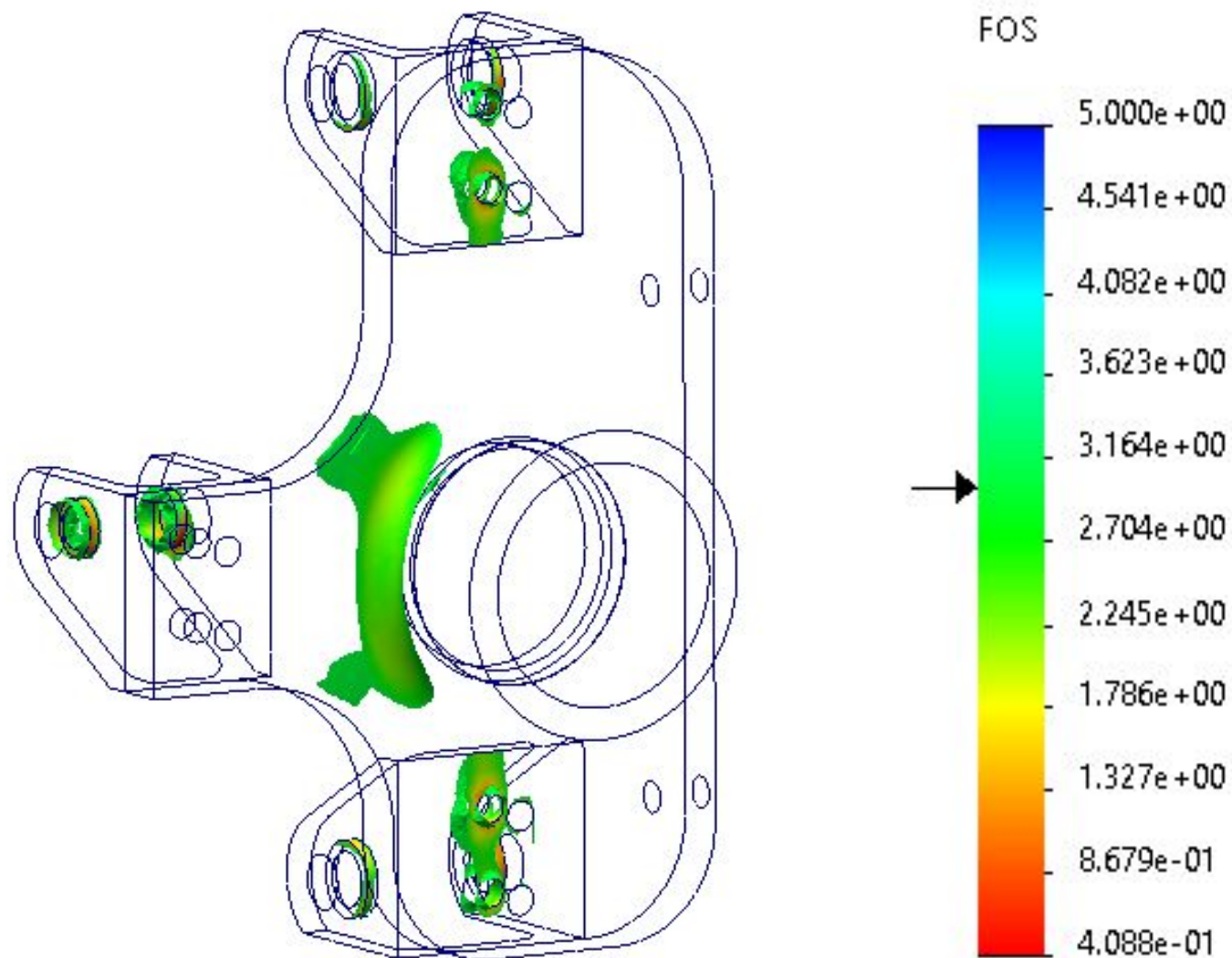
Bolt connections for $\frac{1}{4}$ -20 bolts with 144 in-lbf and local interaction between the back face of the bracket and upright

FOS: 0.52

Equations (If Applicable):

Confidence Level & Explanation: Low: Numbers are gathered from previous FEA but lack supporting test data for loading force. Heims are over constrained so results are not completely accurate. Failure occurs at brackets and results were disproven with tear out, bearing, and shear stress hand calculations.

Rear Suspension Upright FEA



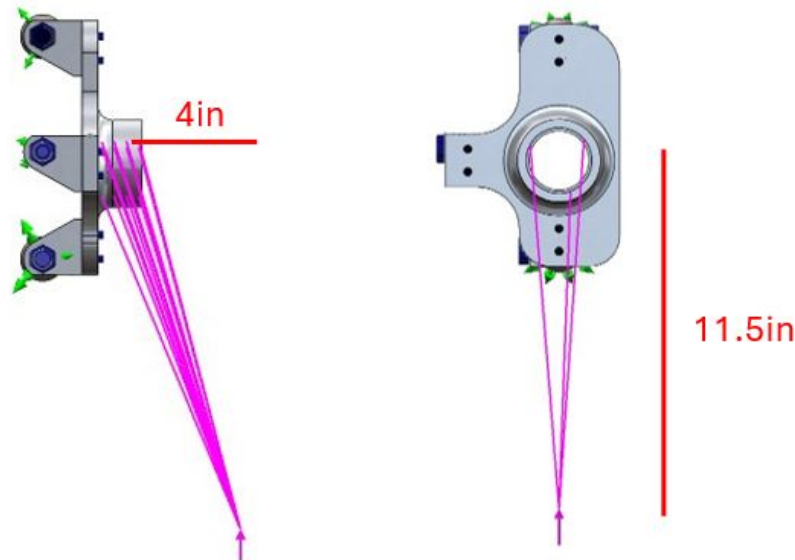
Rear Suspension Upright FEA

Failure Condition: Landing a jump on the outside bottom of one wheel under approximately a 2G load.

Material: 6061-T6 Aluminum upright (0.5" thick) and brackets (0.25" thick), Heim ball joints are 4130 Chromoly only input for boundary condition simulations

Justification: This is a worse case loading if the vehicle where to landed a jump on one wheel of the front wheels.

Load Diagram:



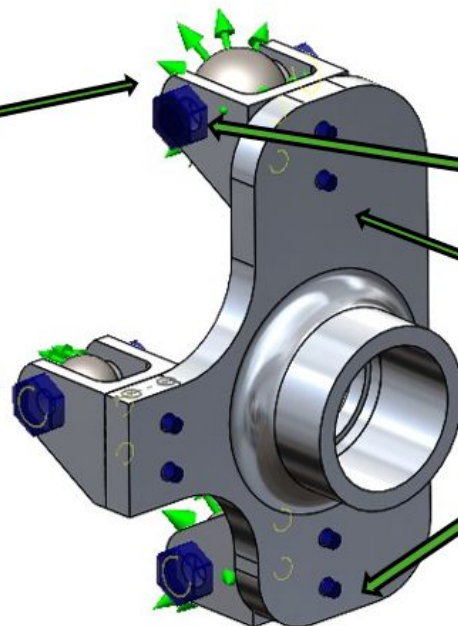
1000 lbf Applied
outmost corner of the
front of the tire (4in
out and 11.5 forward
from the center of the
bearing surface)

Rear Suspension Upright FEA

Constraints:

Radial constraint acting on the outside face of the control arm Hiems. This locks all translation and is an over constraint for the top hiem as it does allow for up-down movement.

A translation constraint in the direction of the toe rod representing the 2 force member. This acts on the outside face of the Hiem.



Bolt connections for $\frac{1}{2}$ -20 bolts with 144in-lbf and bond interaction between the Heim joint and the bracket due to friction

Bolt connections for $\frac{1}{4}$ -20 bolts with 144 in-lbf and local interaction between the back face of the bracket and upright

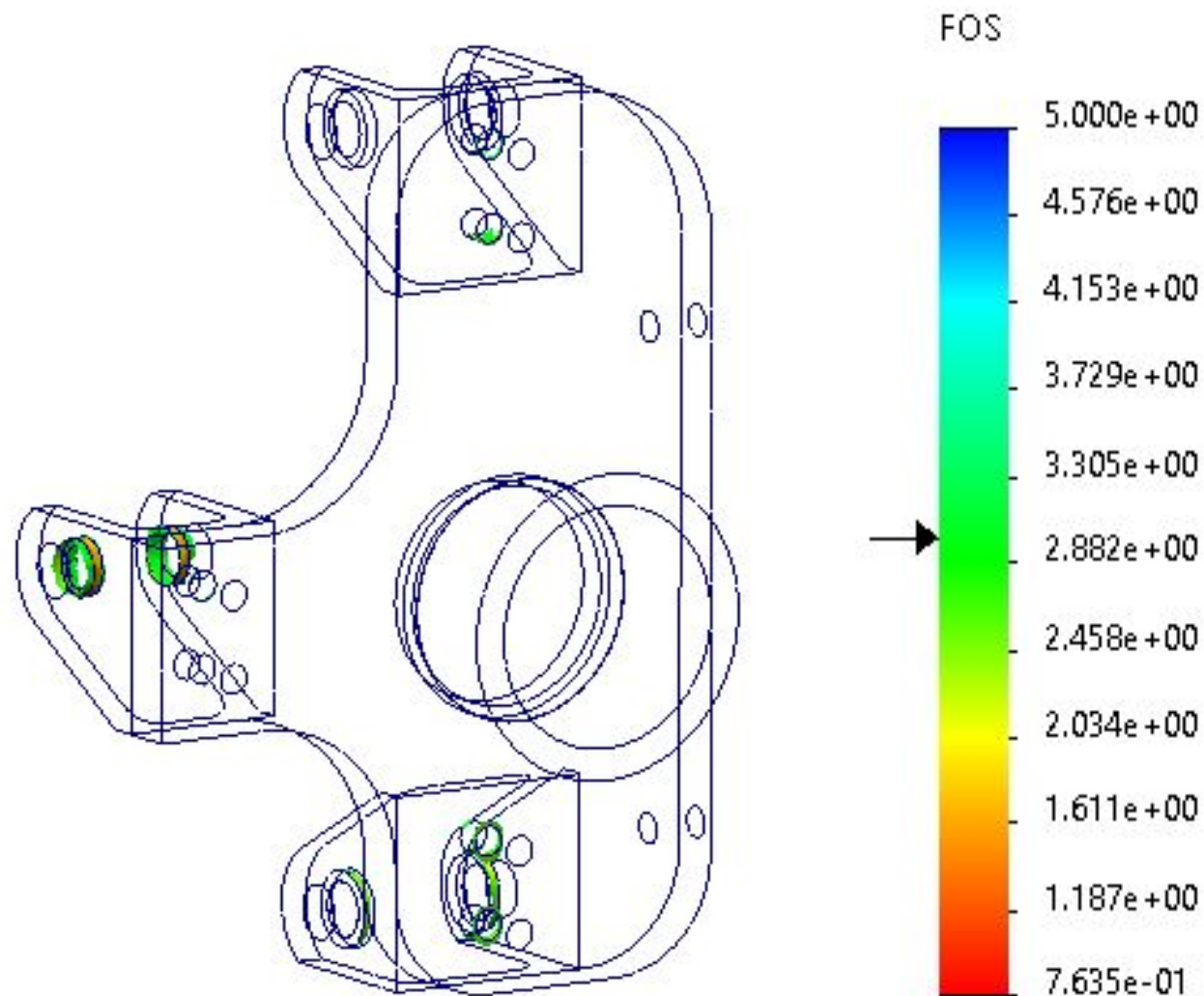
FOS: 1.0

Equations (If Applicable):

Confidence Level & Explanation:

Low: Loads are not well known. Due to SolidWorks limitations the model is over defined in the top Hiem explained above.

Rear Suspension Upright FEA

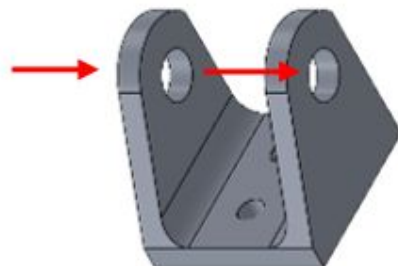


Rear Suspension Upright Brackets FEA

Failure Condition: bending of the bracket due to a frontail impact of 3000lbf

Justification: Last year experience serial crashes and collision therefore calculating worse-case scenario of a side impact which would create a moment and attempt to pull the bolt out of the bracket

Load Diagram:



3000 lbf split between top and bottom brackets resulting in 1500lbf per bracket

Constraints: 1500lbf



Simplified to a combine thickness of both bracket arms with worst case distance to edge.

Rear Suspension Upright Brackets FEA

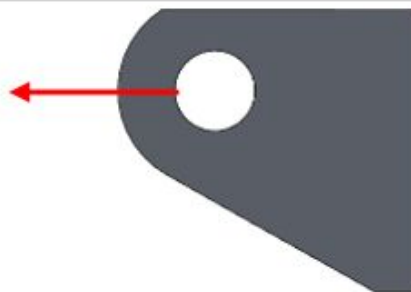
FOS:	Confidence Level & Explanation: Medium, Using previous years numbers with 3000lbf frontal impact, they have never experienced failure.
Equations (If Applicable): $I = \frac{bh^3}{12}$ $\sigma = \frac{Mc}{I}$ $FS_{\text{bend}} = \frac{\sigma_{\text{allow}}}{\sigma}$	
Calculations and/or FEA: --- Aluminum Bracket Calculations --- Moment of Inertia (I): 0.018229 in^4 Applied Moment: 1500.00 lb·in Bending Stress: 20571.43 psi Safety Factor (Bending): 1.70	

Rear Suspension Upright Brackets FEA

Failure Condition: bolt tear out due to max force of shock translated to the bracket the bracket of 2000lbf

Justification: Last year experience serial crashes and cloisons therefore calculating worse-case scenario of a side impact which would create a moment and attempt to pull the bolt out of the bracket

Load Diagram:



1200lbf applied to the Heim bolt which translate to a 1600lbf of the shock

Constraints:



Simplified to a combine thickness of both bracket arms with worst case distance to edge

Rear Suspension Upright Brackets FEA

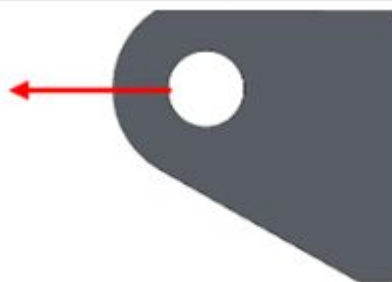
FOS:	Confidence Level & Explanation: medium, Deriving the maximum force from FOX shocks data and previous years using the same value without failure.
Equations (If Applicable): $F_{\text{bracket}} = F_{\text{shock}} \frac{L_{\text{shock}}}{L_{\text{arm}}}$ $\sigma = \frac{F_{\text{bracket}}}{A}$ $FS_{\text{tear}} = \frac{\sigma_{\text{allow}}}{\sigma}$	
Calculations and/or FEA: --- Aluminum Bracket Calculations --- Tear-Out Area: 0.1250 in^2 Force on Bracket: 1166.01 lb Tear-Out Stress: 9328.10 psi Safety Factor (Tear-Out): 3.75	

Rear Suspension Upright Brackets FEA

Failure Condition: bolt tear out due to side impact of the wheel creating a pulling on the bracket of 2000lbf

Justification: Last year experience serial crashes and cloisons therefore calculating worse-case scenario of a side impact which would create a moment and attempt to pull the bolt out of the bracket

Load Diagram:



2000lbf applied to the Heim bolt which translate to roughly a 1250lbf impact to the bottom side of the tire

Constraints:



Simplified to a combine thickness of both bracket arms with worst case distance to edge

Rear Suspension Upright Brackets FEA

FOS:

Equations (If Applicable):

$$F_{\text{wheel}} = 1500 \frac{L_{bt}}{L_{bw}}$$

$$\sigma = \frac{F}{A}$$

$$FS_{\text{tear}} = \frac{\sigma_{\text{allow}}}{\sigma}$$

Confidence Level & Explanation:Low:

Loads are not well known for this loading case. Previous years have done

Calculations and/or FEA:

--- Aluminum Bracket Calculations ---

Tear-Out Area: 0.5250 in²

Force per Bracket Side: 2000.00 lb

Force on Bottom Wheel: 1235.29 lb

Tear-Out Stress: 3809.52 psi

Safety Factor (Tear-Out): 9.19

Rear Suspension Upper Control Arm



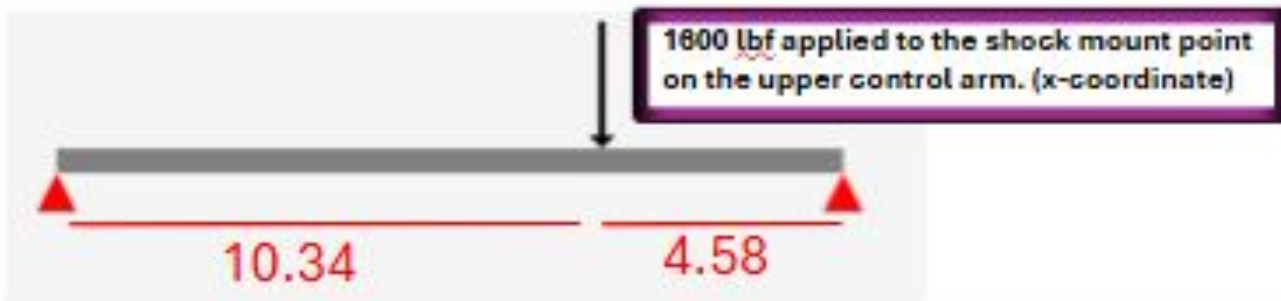
Front of Car (Right Side)

Upper Control Arm Beam Approximation

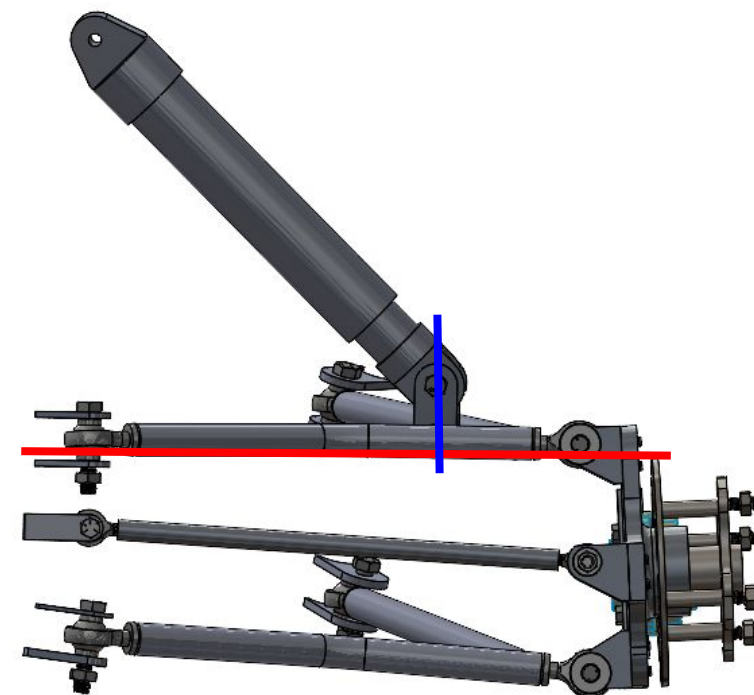
Failure Condition: 1600lbf load at shock mount (side to side beam) 1in OD X 0.095 4130

Justification: The shock is not equidistant between the chassis and upright.

Load Diagram:



Constraints:



Upper Control Arm Beam Approximation

FOS:

Equations (If Applicable):

$$I = \frac{\pi}{64} \cdot (\text{Tube}_{\text{diam,out}}^4 - \text{Tube}_{\text{diam,in}}^4)$$

$$C = \frac{\text{Tube}_{\text{diam,out}}}{2}$$

$$\sigma_{\text{max}} = M_{\text{max}} \cdot \frac{C}{I}$$

$$\text{FOS} = \frac{\sigma_{\text{Yield}}}{\sigma_{\text{max}}}$$

Confidence Level & Explanation:

Low

Our single dimension beam approximation shows that the structural member will be under the highest stress closest to the knuckle

Calculations and/or FEA:

Reactions

Location: 0 (in), Force Reaction = -491.1528 (lb)

Location: 14.92 (in), Force Reaction = -1108.8472 (lb)

Shear Diagram

(Max +ve)Shear Load (lb): 491.153,

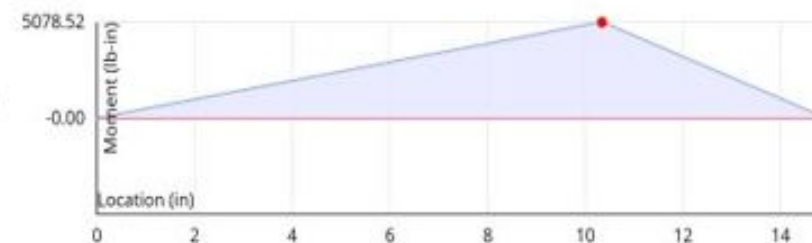
Location (in): 0.000, 10.340

(Max -ve)Shear Load (lb): -1108.847,

Moment Diagram

(Max +ve)Moment Load (lb-in): 5078.520,

Location (in): 10.340



Rear Suspension Upper Control Arm FEA

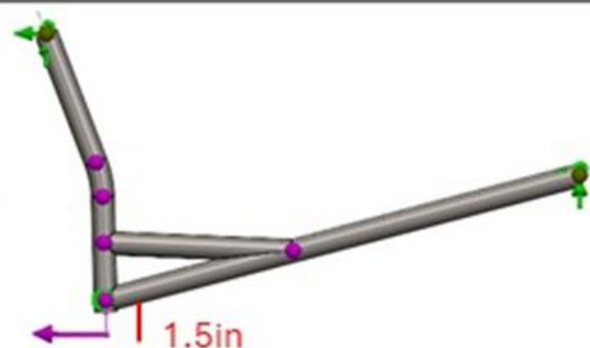
Failure Condition: Front impact on the front of the rear wheel (Upper Control Arm)

Material: Alloy Steel 4130 1" OD, forward member 0.065 Wall, rear and cross member 0.093" OD

Justification: When we experience frontal obstacles at speed, part of the force will be translated into the control arms.

Load Diagram:

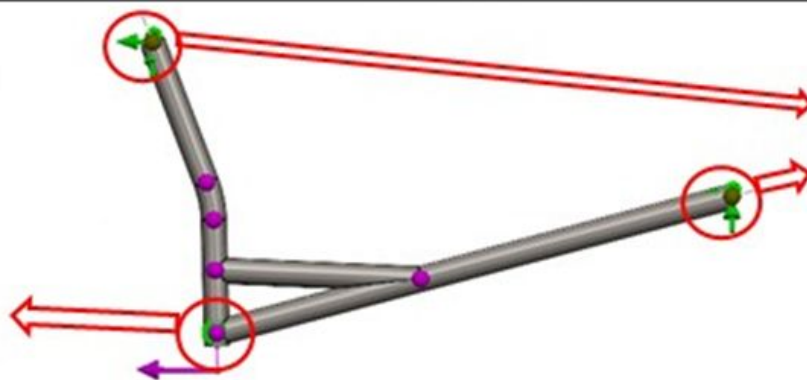
1350lbf applied to the outer most point of the upper control arm, 1.5 inches out where the center of the rod end would be.



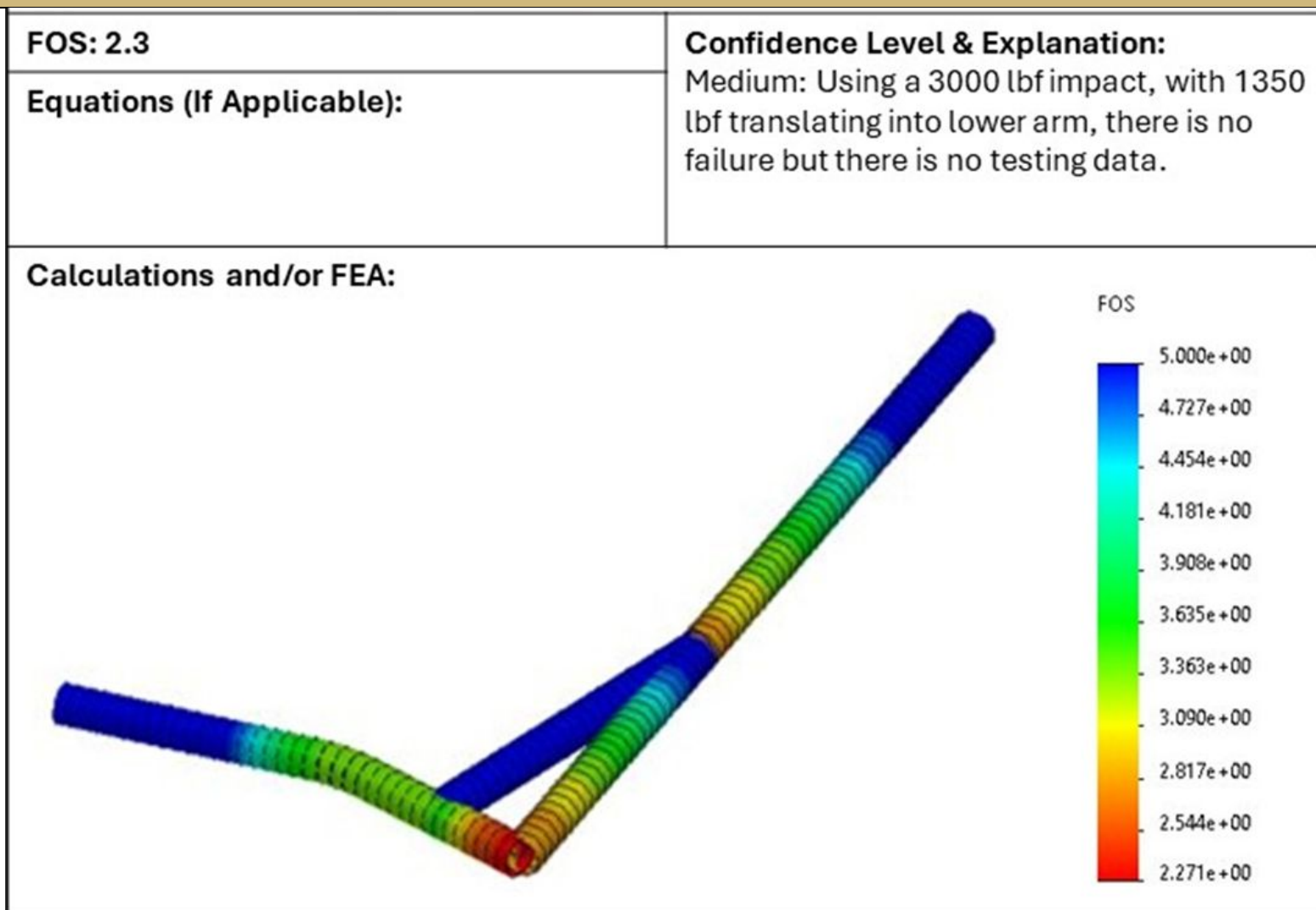
Constraints:

Single translation constraint on the end of the control arm to prevent rotation and simulate a ridged shock force translated through the upright

Heims constrained to only allow rotation



Rear Suspension Upper Control Arm FEA



Rear Suspension Upper Control Arm FEA

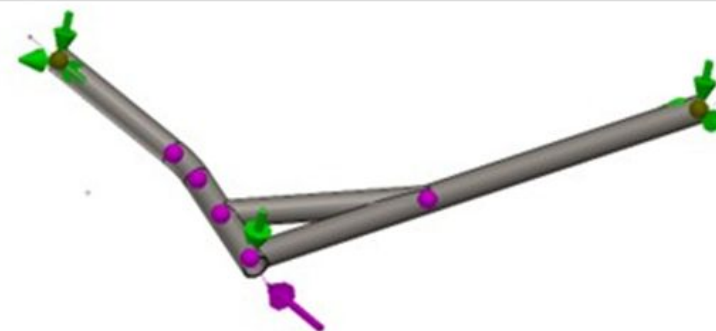
Failure Condition: Side impact on the center of the rear wheel (Upper Control Arm)

Material: Alloy Steel 4130 1" OD, forward member 0.065 Wall, rear and cross member 0.093" OD

Justification: The control arm will experience a force if the tire experiences an impact from the side.

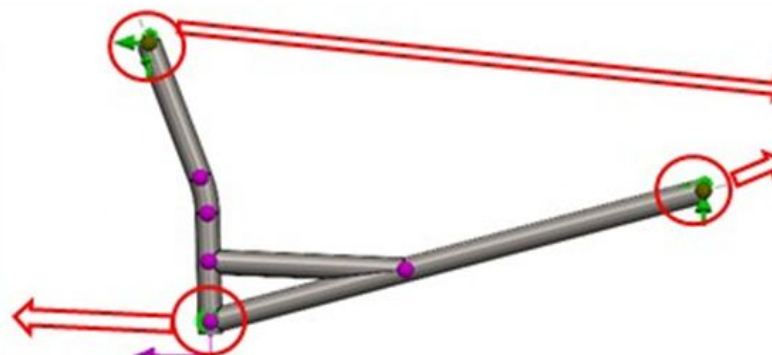
Load Diagram:

1350lbf applied to the outer most point of the upper control arm, 1.5 inches out where the center of the rod end would be.



Constraints:

Single translation constraint on the end of the control arm to prevent rotation and simulate a ridged shock force translated through the upright



Heims constrained to only allow rotation

Rear Suspension Upper Control Arm FEA

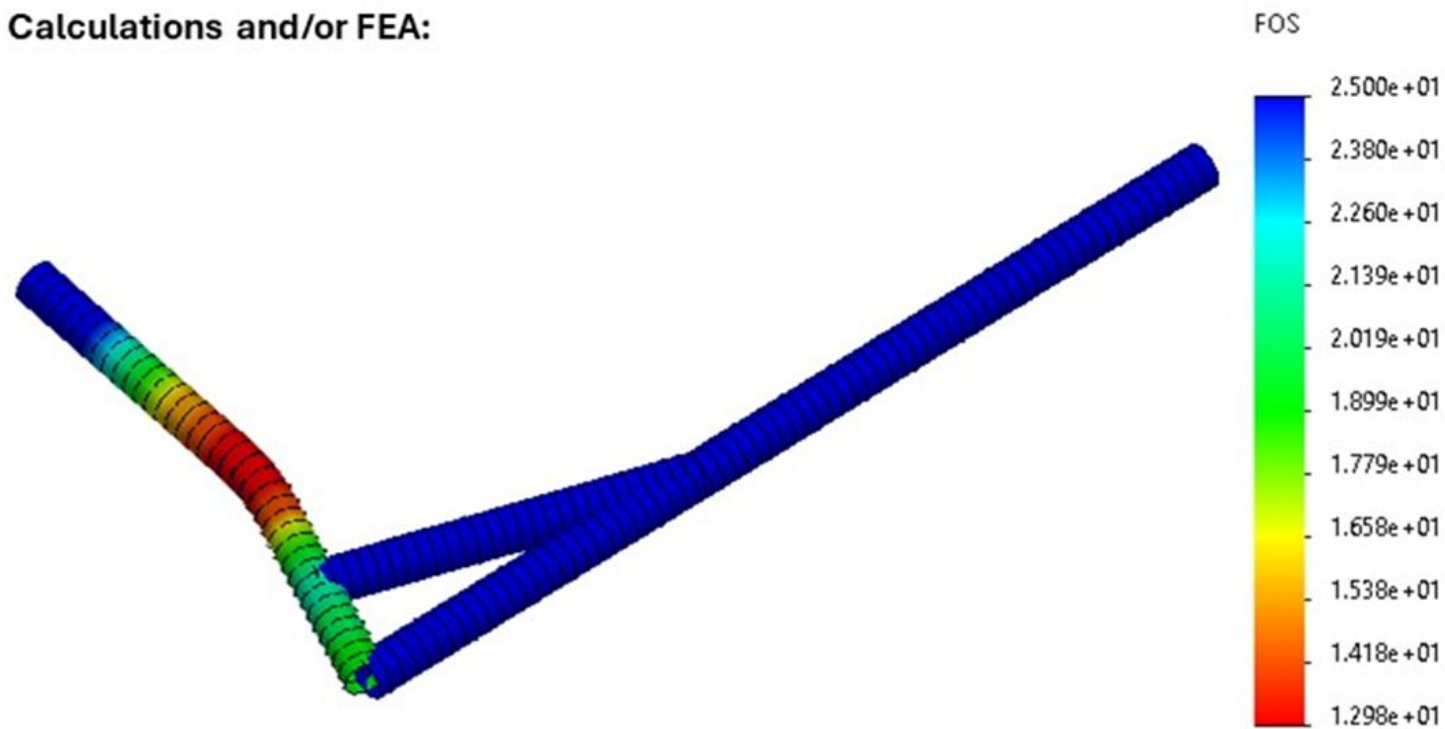
FOS: 13

Equations (If Applicable):

Confidence Level & Explanation:

Medium: Using a 3000 lbf impact, with 1350 lbf translating into lower arm, there is no failure but there is no testing data.

Calculations and/or FEA:



Rear Suspension Upper Control Arm FEA

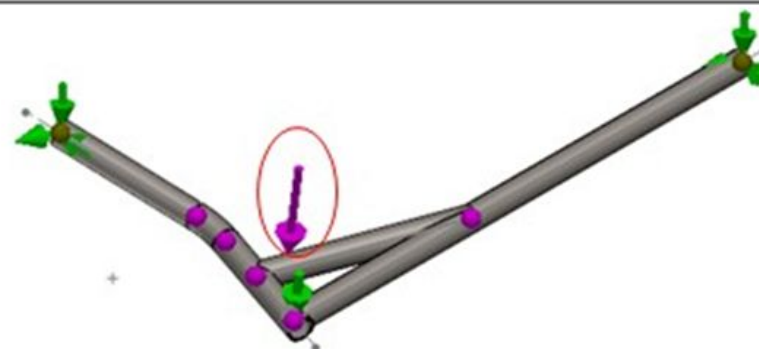
Failure Condition: Maximum shock force on upper rear control arm

Material: Alloy Steel 4130 1" OD, forward member 0.065 Wall, rear and cross member 0.093" OD

Justification: At maximum compression, the shock exerts a 1600 lbf force downwards on the upper control arm

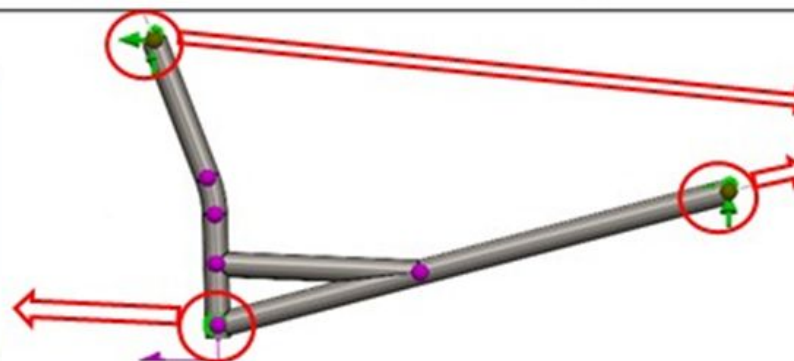
Load Diagram:

1600 lbf applied to the shock mount point on the upper control arm cross member.



Constraints:

Single translation constraint on the end of the control arm to prevent rotation and simulate a rigid shock force translated through the upright



Heims constrained to only allow rotation

Rear Suspension Upper Control Arm FEA

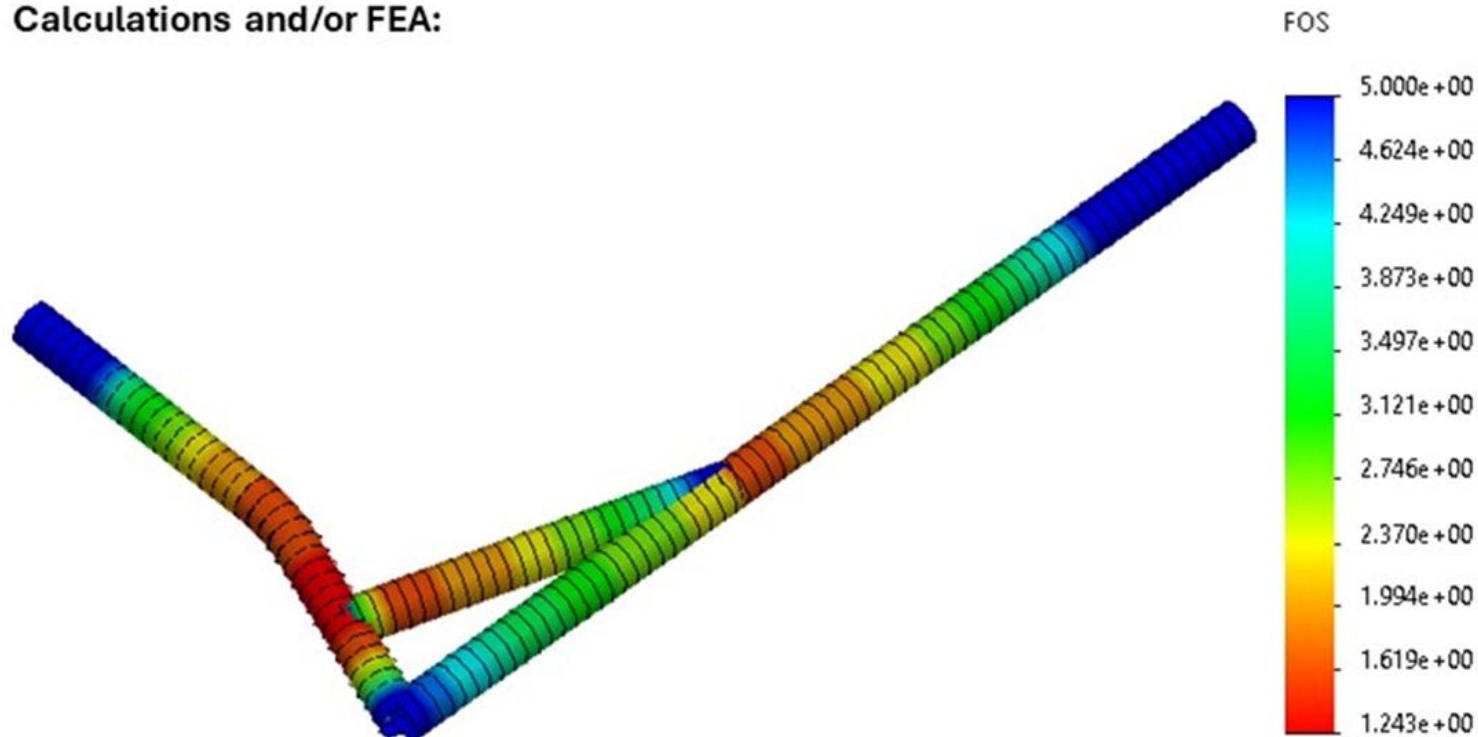
FOS: 1.2

Equations (If Applicable):

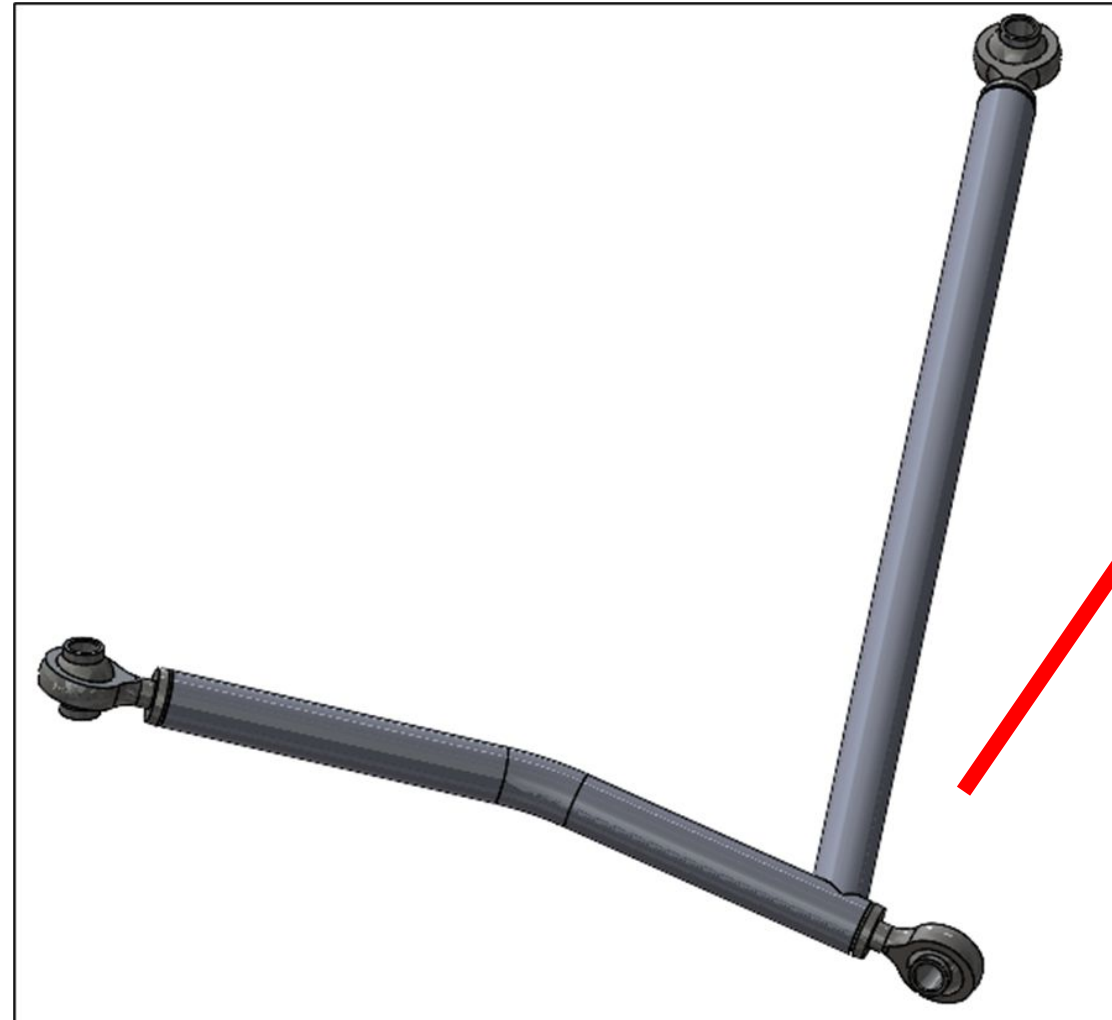
Confidence Level & Explanation:

Medium: Maximum force from FOX shock data was derived and has been used in previous years without failure.

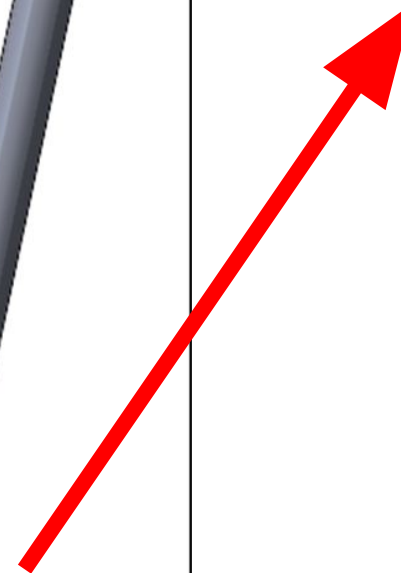
Calculations and/or FEA:



Rear Suspension Lower Control Arm



Front of Car (Right Side)



Rear Suspension Lower Control Arm

Failure Condition: Front impact on the front of the rear wheel (Lower Control Arm)

Material: Alloy Steel 4130 1" OD 0.083 Wall

Justification: When we impact frontal obstacles at speed, part of the force will be translated into the control arms.

Load Diagram:

1750lbf applied to the outer most point of the upper control arm, 1.5 inches out where the center of the rod end would be.



Constraints:

Single translation constraint on the end of the control arm to prevent rotation and simulate a ridged shock force translated through the upright

Heims constrained to only allow rotation



Rear Suspension Lower Control Arm

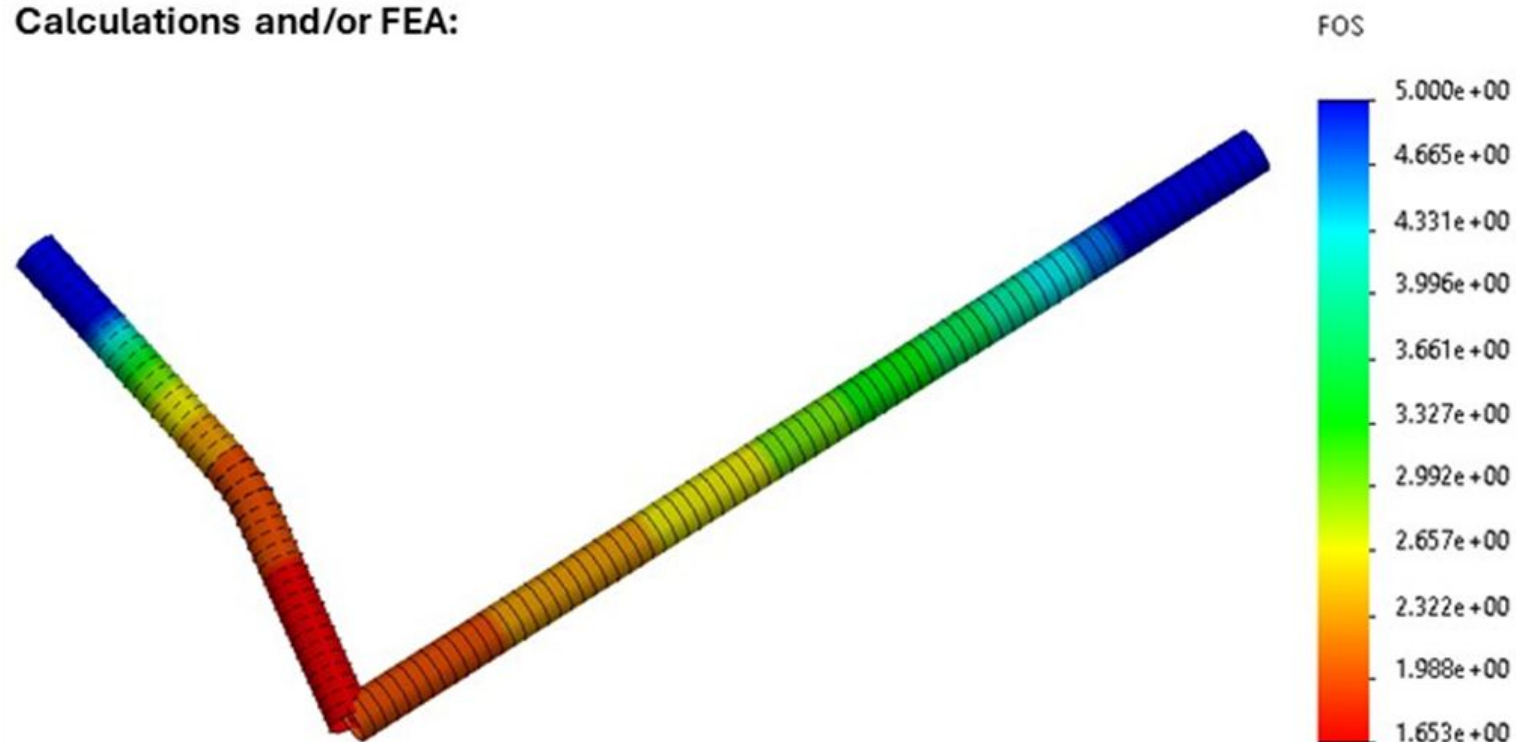
FOS: 1.7

Equations (If Applicable):

Confidence Level & Explanation:

Medium: Using a 3000 lbf impact, with 1750 lbf translating into lower arm, there is no failure but lack of testing data.

Calculations and/or FEA:



Rear Suspension Toe Rod

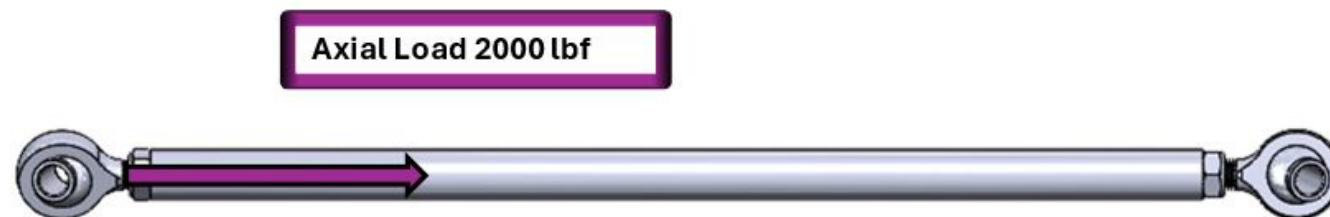


Rear Suspension Toe Rod Analysis

Failure Condition: Side impact or collision from an opposing vehicle, causing buckling in the toe rod.

Justification: Aluminum 6061 15" long tie rod experiencing side impact of 2000lbf
0.63 OD 0.13 thick

Load Diagram:



Constraints:



Rear Suspension Toe Rod Analysis

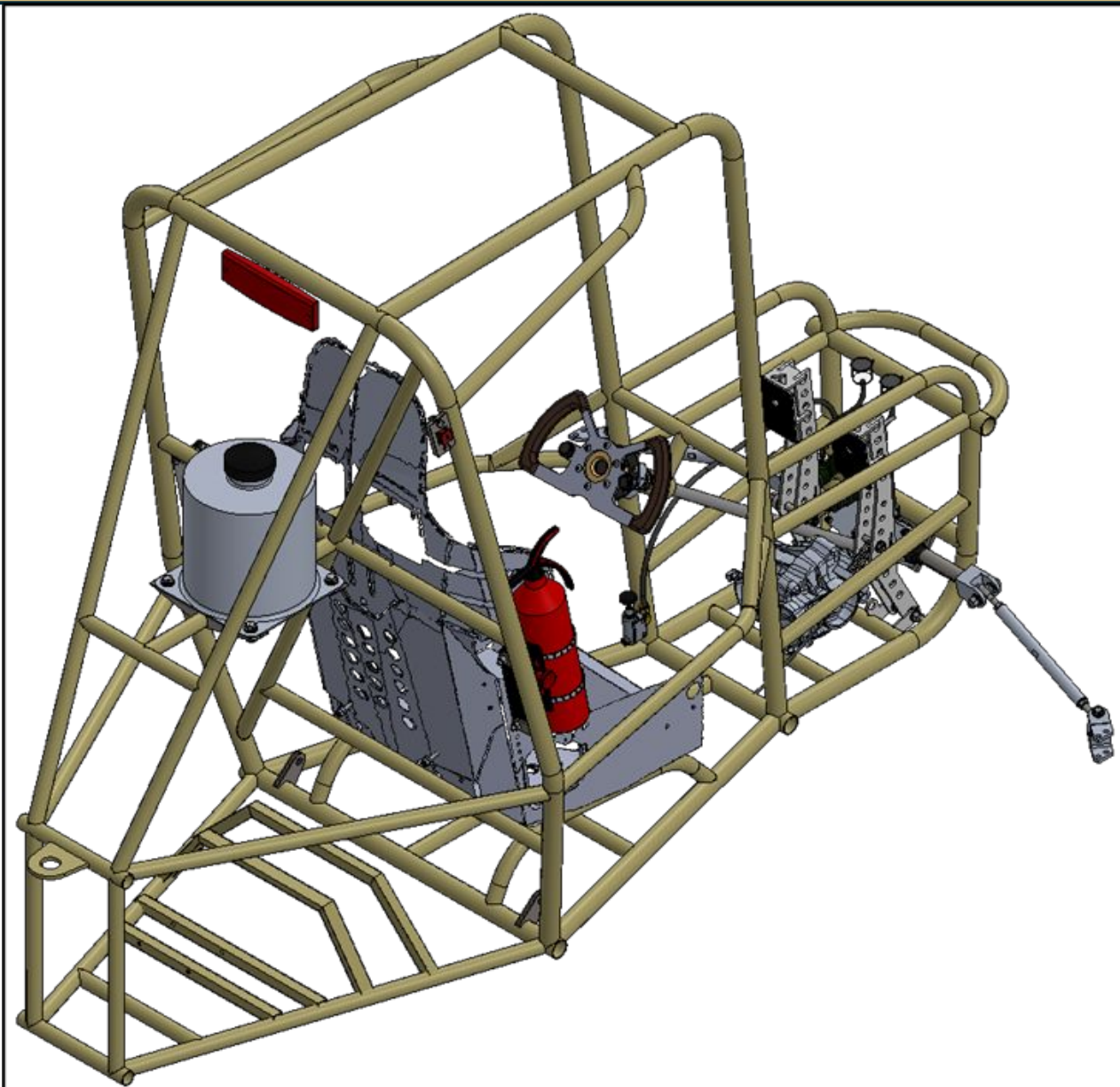
FOS: 1.991	Confidence Level & Explanation:
Equations: $I = \frac{\pi}{64} (D_{\text{outer}}^4 - D_{\text{inner}}^4)$ $r = \sqrt{\frac{I}{A}}$ $R_s = \frac{L}{r}$ $R_{\text{trans}} = \sqrt{\frac{2\pi^2 E}{K^2 S_y}}$ $\sigma_{\text{buckling}} = \frac{\pi^2 E}{(K R_s)^2}$ $F_{\text{buckling}} = \sigma_{\text{buckling}} A$ $\text{FOS} = \frac{F_{\text{buckling}}}{F_{\text{max}}}$	Medium: The tie rod material and expected load values are known from previous years
Calculations and/or FEA:	$E = 1.001 \times 10^7 \text{ [lbf/in}^2\text{]}$ $S_y = 19,943 \text{ [lbf/in}^2\text{]}$ $\sigma_{\text{buckling}} = 19,499 \text{ [lbf/in}^2\text{]}$ $F_{\text{buckling}} = \boxed{3,982 \text{ [lbf]}}$ $F_{\text{max}} = 2,000 \text{ [lbf]}$ $\text{FOS} = \boxed{1.991}$

Controls

Controls Goals

Goal	Reason	Implementation	Difficulties
Make electronic controls accessible Achieved	Current controls are in a dangerous spot to reach while driving	Design a new wheel that maintains strength and incorporates the differential switches	Usability with race conditions, sealing, and clear fail-safe without compromising serviceability
Create lighter brake pedal with more travel Missed. Almost Exactly the same as last years weight and travel	Save weight and improve driver ergonomics	Create a new pedal with adjustable geometry that uses water-jet steel arms	Maintain target line pressure and firm feel without the master cylinder bottoming and pedal flex/fatigue
Improve throttle pedal cable clamping reliability Achieved	Increases the adjustability of the throttle, provides a better feel, and won't lose tension if pressed hard during an impact or roll over	We will use cable barrel housing, add a spring at the pedal, and adding a spring at the throttle linkage	Prevent slip and strand damage under vibration with consistent clamp torque and corrosion resistance

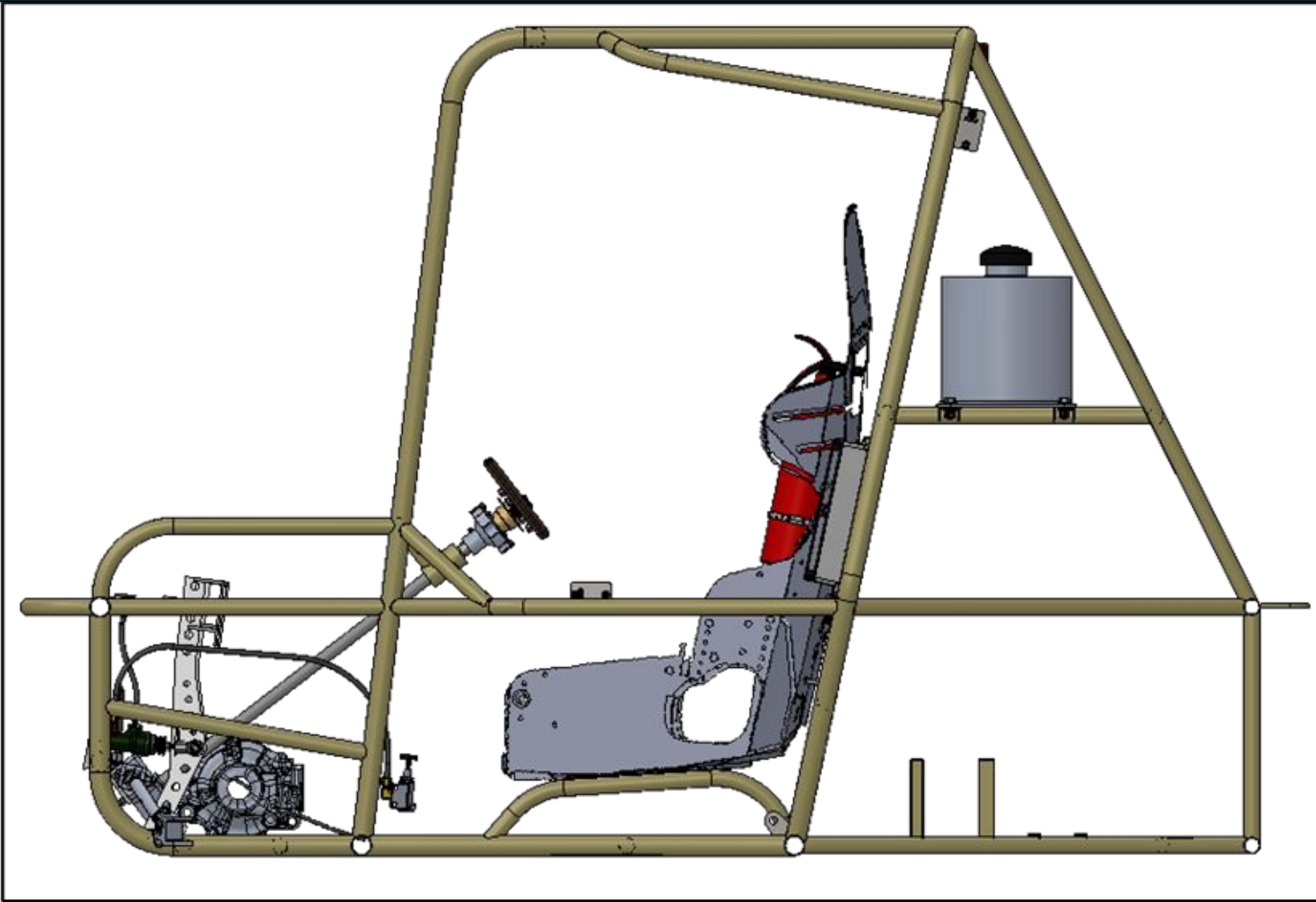
Controls Full Assembly



Controls Assembly	
Weight [lbs]	Cost
≈ 42	Anticipated: \$320

Controls Full Assembly

87



Steering Assembly Overview



Steering Assembly	
Weight [lbs]	Cost
This Year $\approx$ 13.2 (From CAD) Last Year $\approx$ 17 (Measured)	Anticipated: \$22 5

Steering Wheel



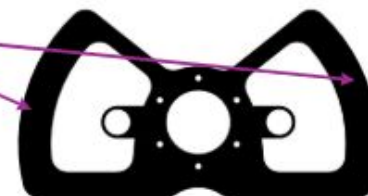
Steering Wheel Analysis

Failure Condition: If the driver slams into the wheel, should they brace need to themselves.

Justification: This would be worst Case Scenario. With the Driver strapped down, they should not be able to exert 75 lbs to the wheel.

Load Diagram:

Load of 75 lbs applied to each side, specifically where the hands will be



Constraints:

6 ¼ -20 bolts hold the wheel in place. The Wheel is also constrained to the flat plate of the quick release



Steering Wheel Analysis

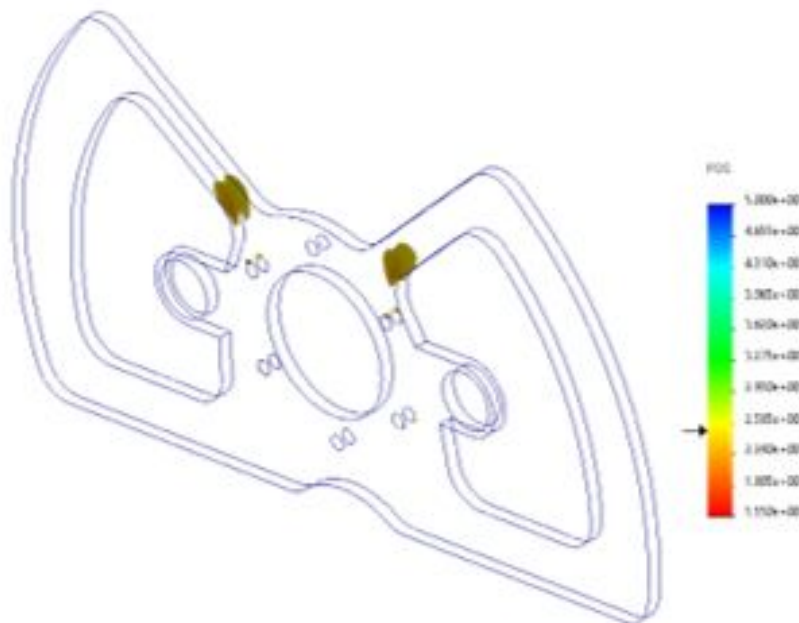
FOS: 1.6

Equations (If Applicable):

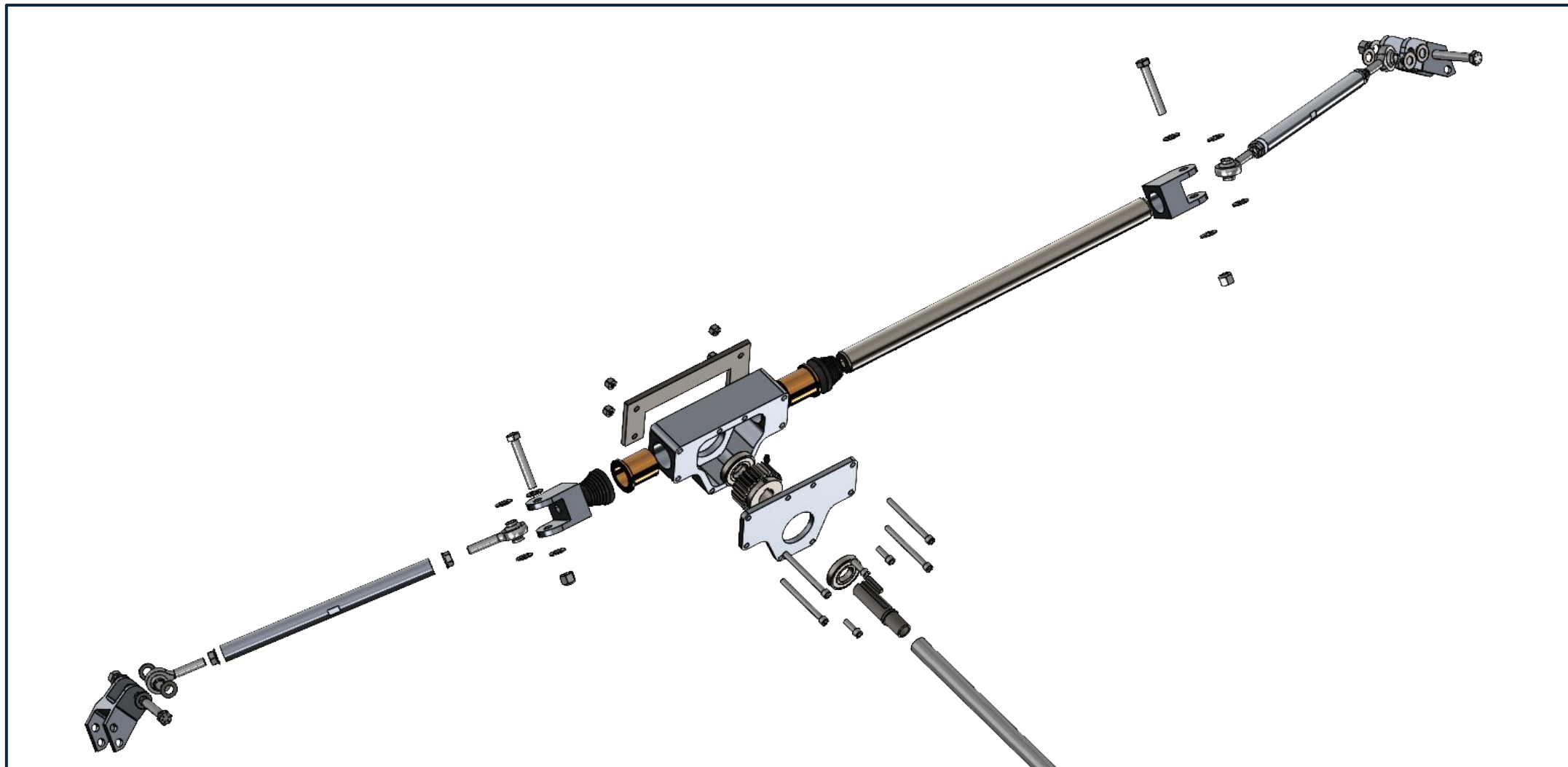
Confidence Level & Explanation: Medium.

While the driver may not be able to exert 75 lbs while strapped in, they may exert more while getting in or out of the car

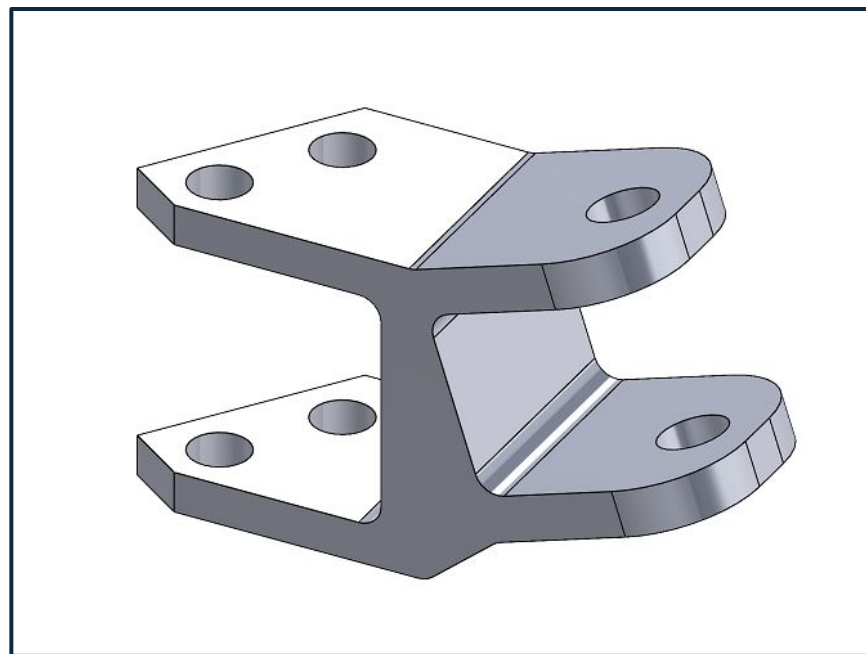
Calculations and/or FEA:



Lower Steering Assembly Overview

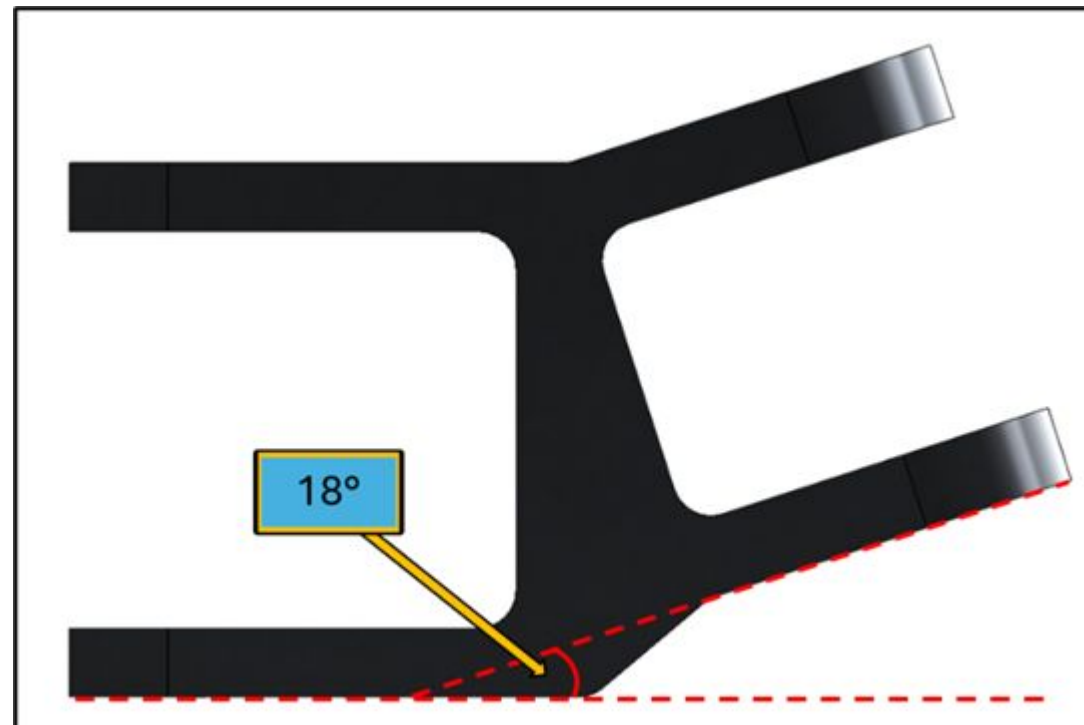
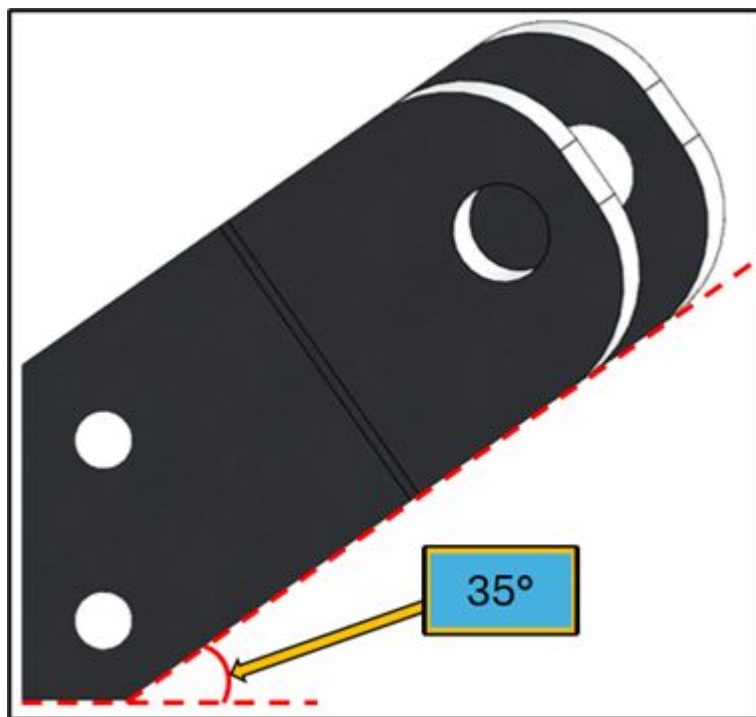


Steering Assembly Critical Parts

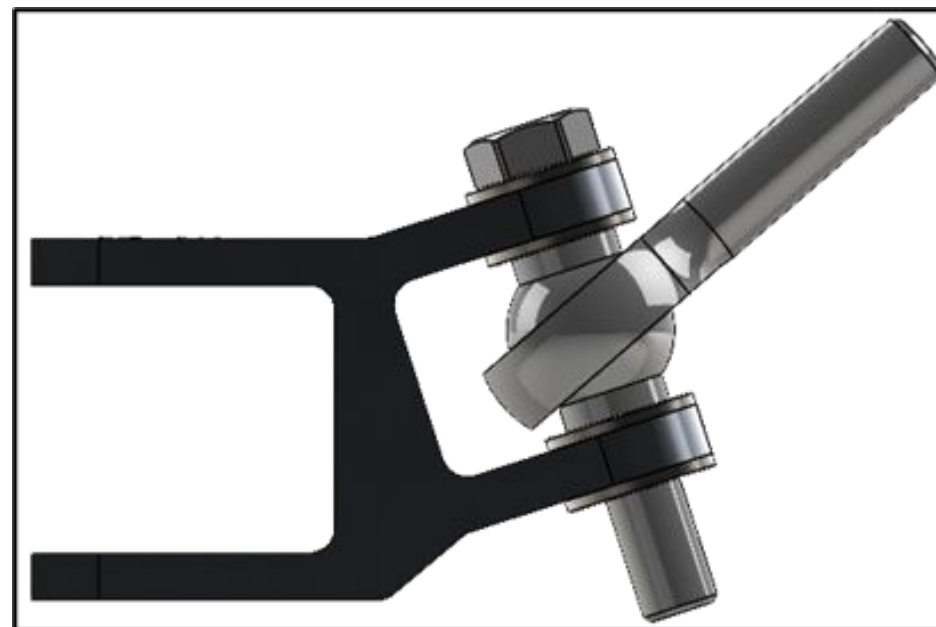
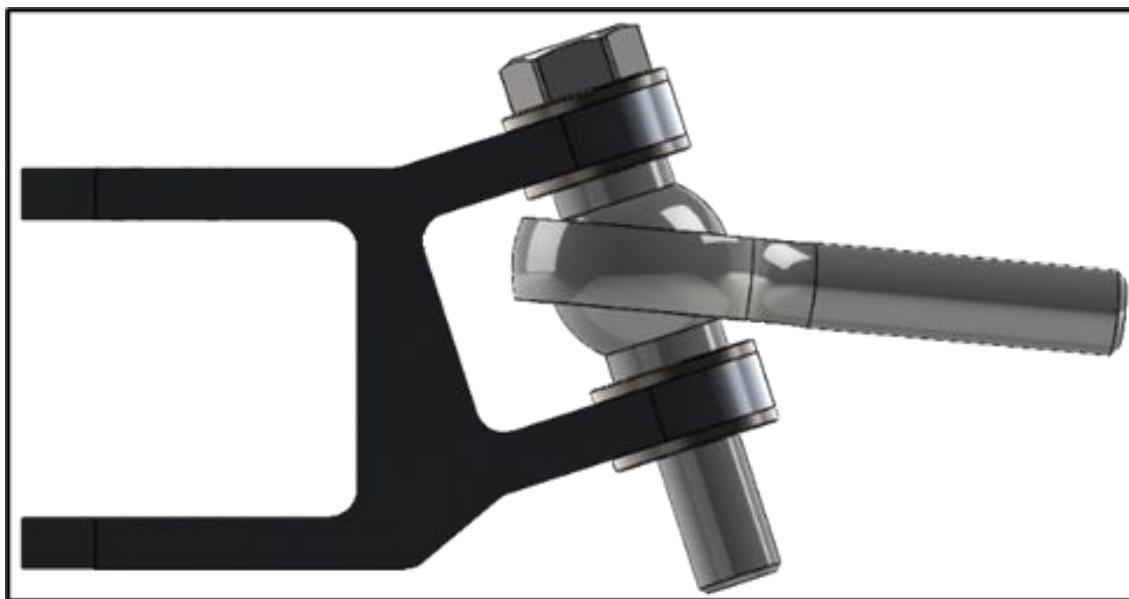


Tie Rod Hub Attachment Bracket

Steering Assembly Critical Parts



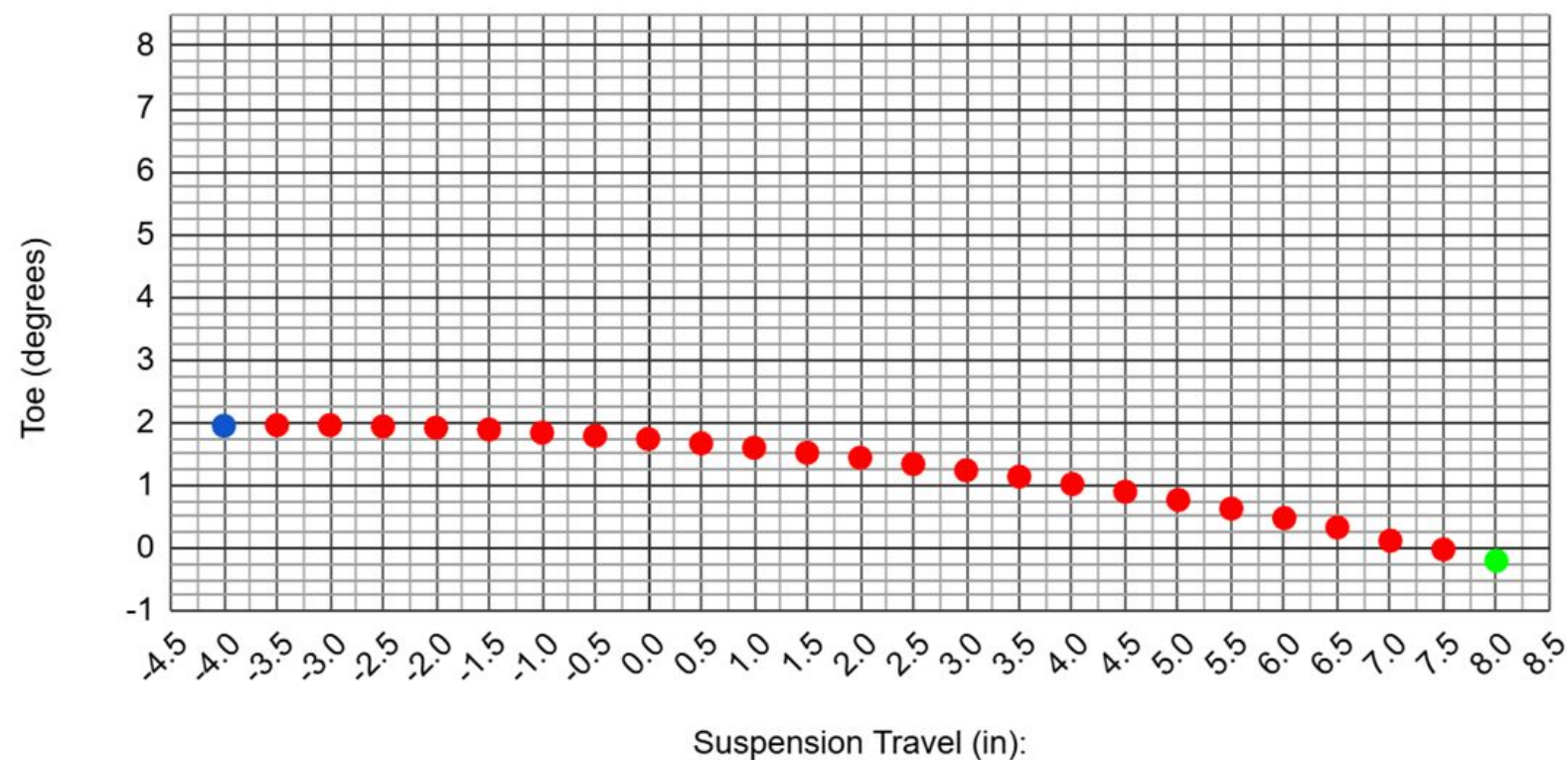
Steering Assembly Critical Parts



Steering Geometry

Absolute Toe (degrees): vs. Suspension Travel (in):

Negative is Toe Out, Positive is Toe In



Steering Angle

Approximate Steering Angles from Our Latest Assembly:

Max Inside Wheel Turn Angle $\approx 30.5^\circ$

Max Outside Wheel Turn Angle $\approx 35^\circ$

Tie Rod Buckling Analysis

Failure Condition: Side impact or collision from an opposing vehicle, causing buckling in the tie rod.

Justification: The calculated axial load of 5,000 lbf is significantly higher than a realistic side impact load of 2,000 lbf at competition. Therefore, with a FOS of 1.15 with an axial load of 5,000 lbf on an Aluminum 6061 tie rod, this suggests that the realistic impact of 2,000 lbf will not cause our tie rod to fail.

Load Diagram:



Constraints:



Tie Rod Buckling Analysis

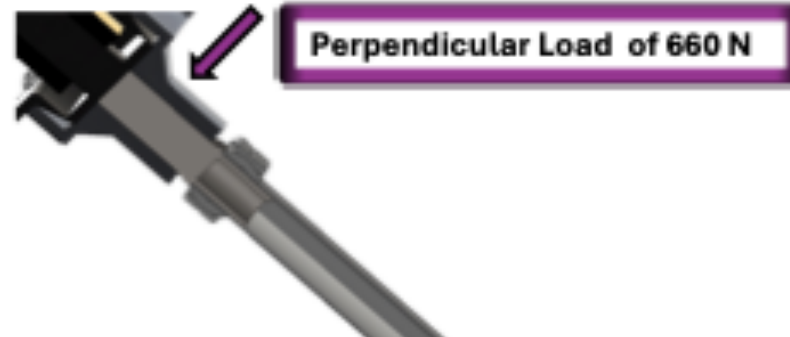
FOS: 1.15	Confidence Level & Explanation: Medium-High since the tie rod material and expected load values are known.	
Equations (If Applicable): See Below		
Calculations and/or FEA:		
Input/Output	Equations	Calculation
OD (in)	outer diameter	0.75
Thickness (in)	wall thickness	0.19
E (ksi)	Young's Modulus	9993.1
I (in^4)	area moment of inertia	0.014611579
P (lbf)	Applied axial compressive load	5000
A (in^2)	Cross sectional area	0.334265458
L (in)	Unsupported length	15.8
FOS		1.154550534

Steering Rod Insert Bending Analysis

Failure Condition: Frontal impact from a collision with an opposing car or obstacle, where the driver pulls down on the steering wheel to brace themselves.

Justification: The applied lateral force of 660 N comes from a study conducted by an FSAE Chief Design Judge to evaluate minimum driver applied forces for SAE competition. Thus, it is reasonable to use this as a baseline for the design of the steering assembly.

Load Diagram:



Constraints:



Steering Rod Insert Bending Analysis

FOS: 10.84		Confidence Level & Explanation: Medium since the expected perpendicular load and length of the shaft are known.
Equations (If Applicable): See Below		
Calculations and/or FEA:		
Input/Output	Notes	Calculations
Force (lbf)	perpendicular load	148.374
Length (in)	distance of shaft (assuming key is gone)	1.625
OD (in)	outer diameter	0.75
Thickness (in)	wall thickness	0.048
Yield Strength (ksi)	4130 tube	63.1
ID (in)	OD - 2*t	0
Moment (in lbf)	F*L	241.10775
y	OD/2	0.375
I (in^4)	area moment of inertia	0.015531555
Stress (ksi)	M*y/(I/1000) to get ksi	5.821400592
FOS	yield/stress	10.8393159

Steering Assembly Analysis - Steering Column Torsion

Failure Condition: The driver hits an obstacle or an obstruction, and the steering wheel jerks out of the driver's hands.

Justification: The applied torque value of 100 Nm comes from a study conducted by an FSAE Chief Design Judge to evaluate minimum driver applied forces for SAE competition. Therefore, it is reasonable to use this as a baseline for the design of our steering assembly.

Load Diagram:



Constraints:

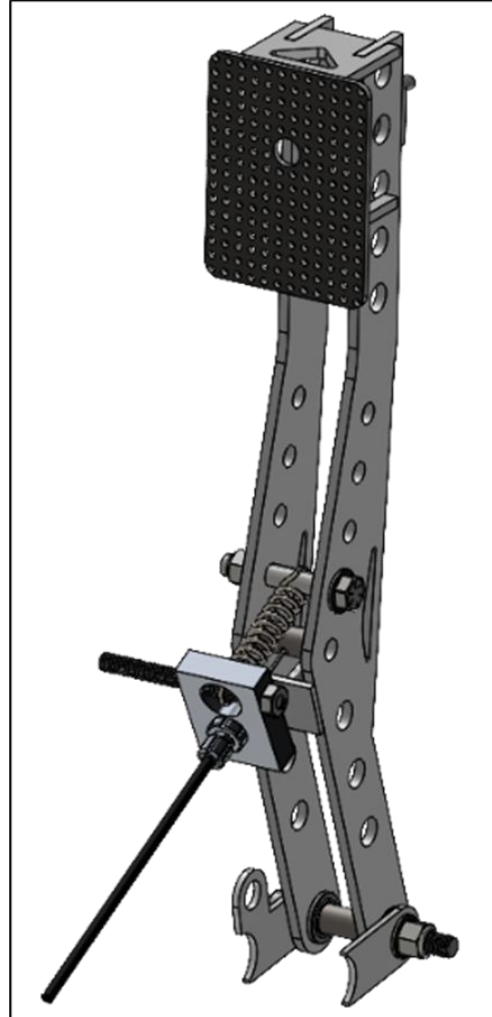
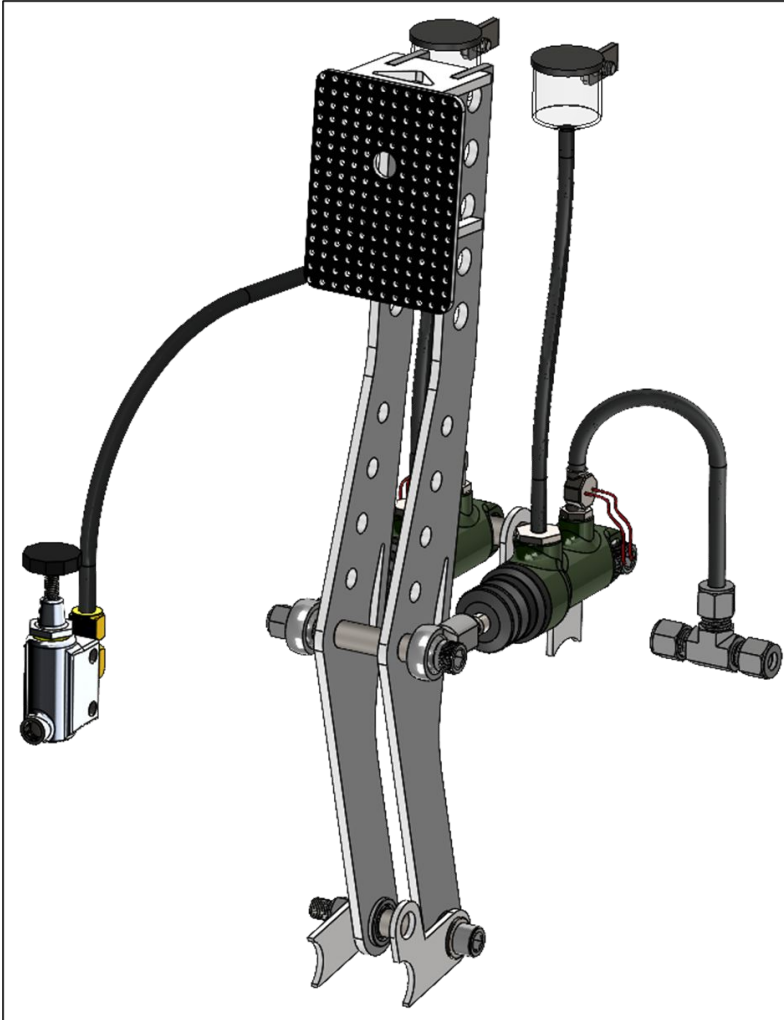
Fixed constraint where it attaches to steering rack on a pinion gear.



Steering Column Torsion

FOS: 1.88		Confidence Level & Explanation: Medium -High since applied torque is from FSAE Study. Steering column material properties are known.
Equations (If Applicable): See Below		
Calculations and/or FEA:		
Torque (in lbf)	applied torque	885.0745792
OD (in)	outer diameter	0.75
Thickness (in)	wall thickness	0.035
Shear Strength (ksi)	4130 tube	62
ID (in)	OD - 2*t	0.68
c (in)	OD/2	0.375
J (in^4)	polar moment	0.010071994
Shear Stress (ksi)	T*c/(J/1000)	32.953054
Modulus of Rigidity (GPa) Found on MatWeb		80
Length (in)	Length of shaft from CAD	25
Angle of Twist (degrees)	TL/GJ	10.84815893
FOS	Shear strength/shear stress	1.881464462

Foot Controls Overview



Brake and Throttle Pedal

Weight [lbs]

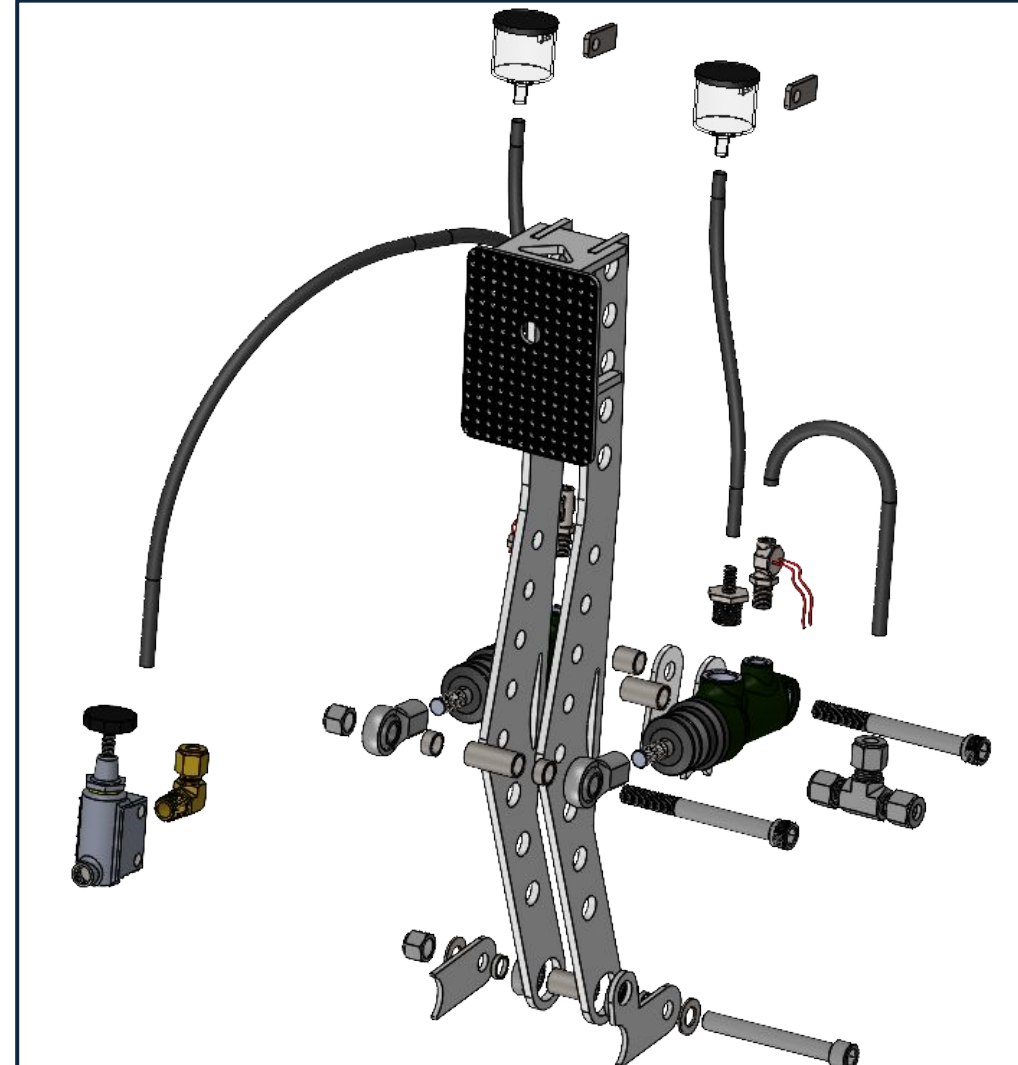
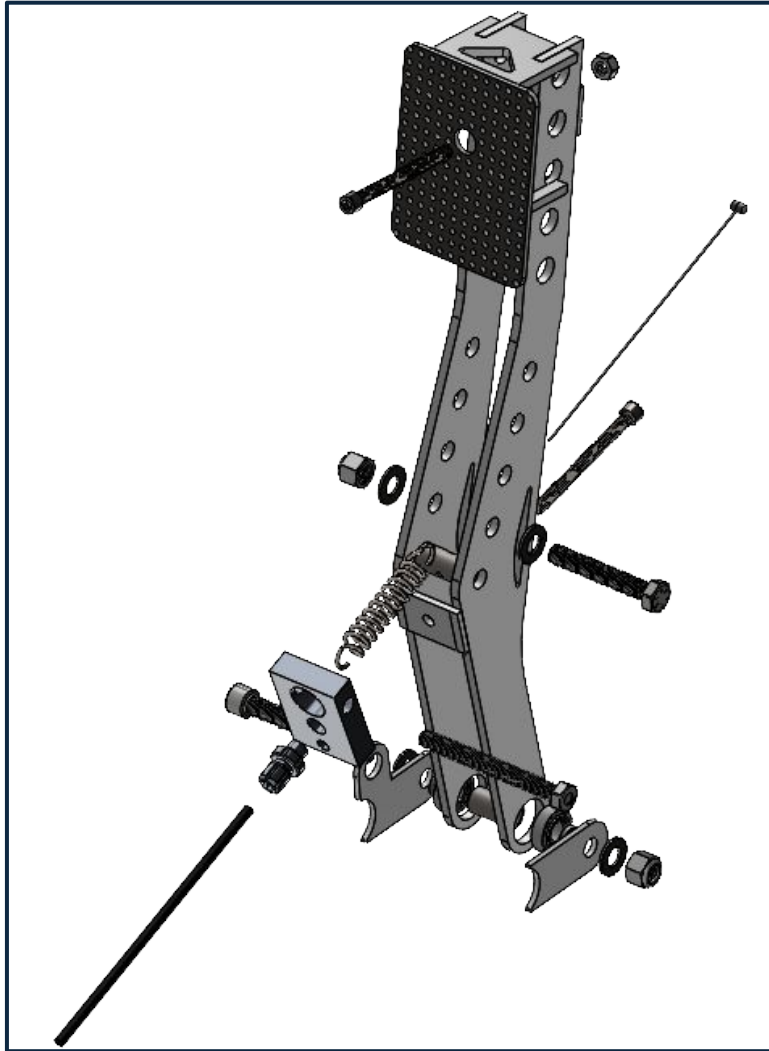
Last Year ≈ 7.5
(Measured)

This Year ≈ 7.5
(CAD)

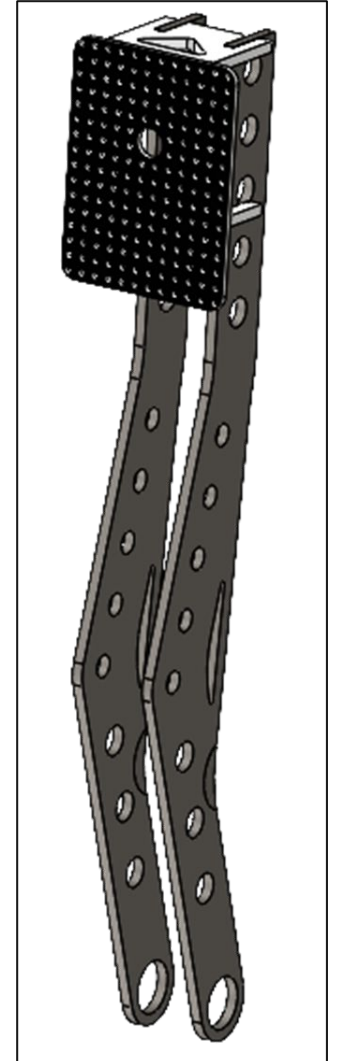
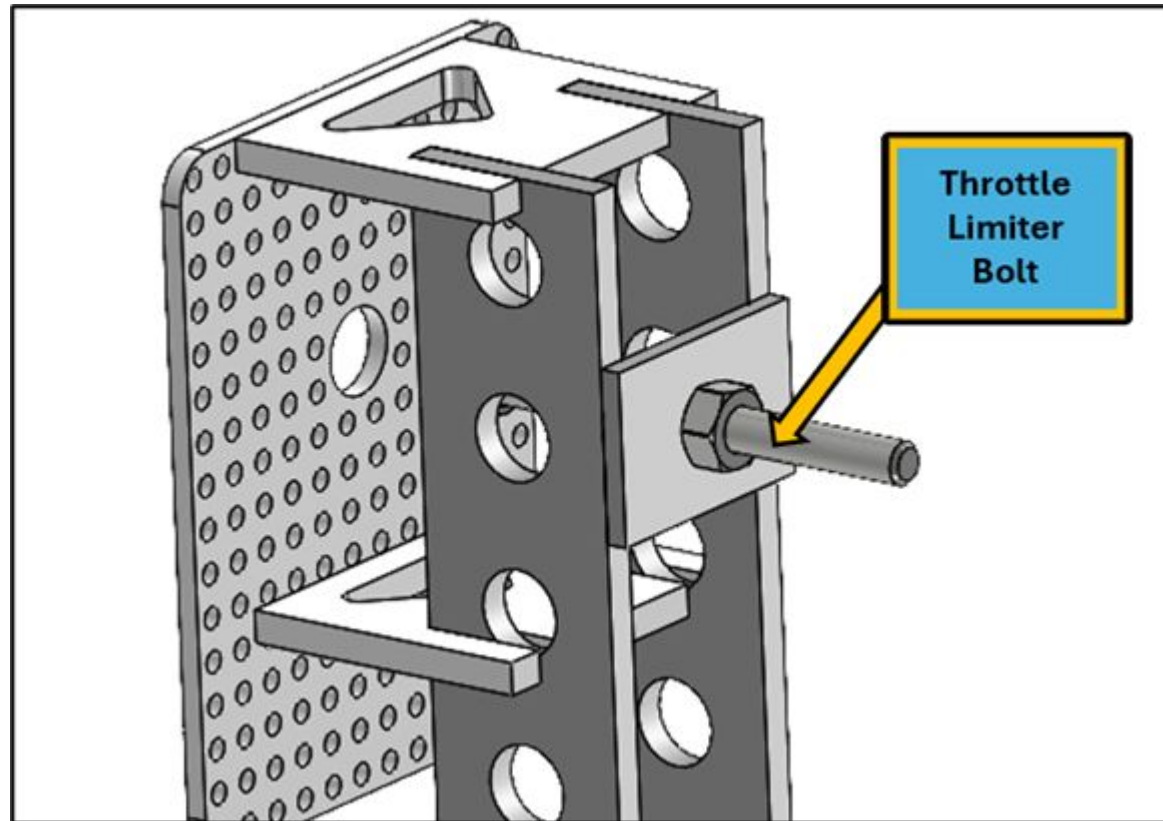
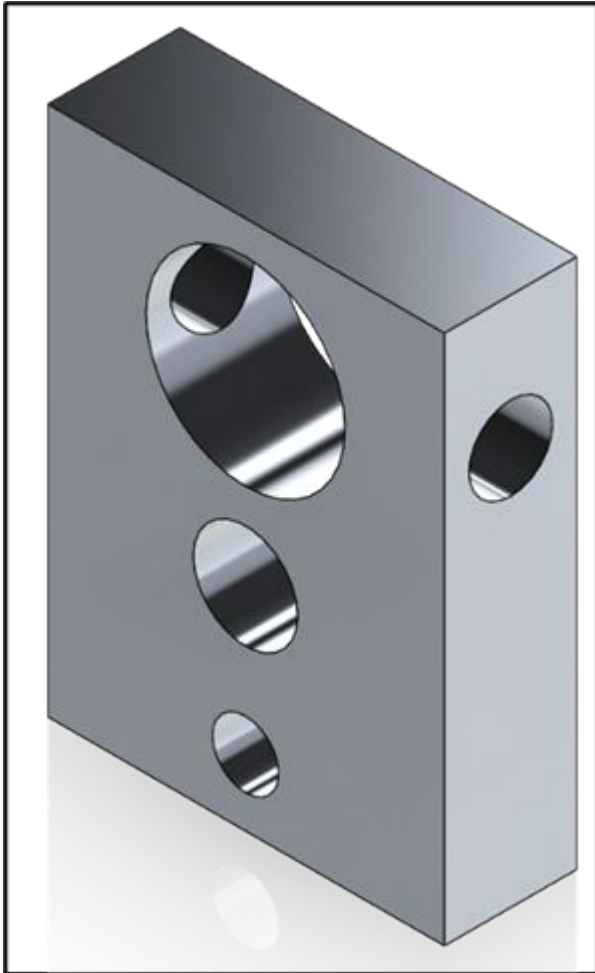
Pedal Comfort



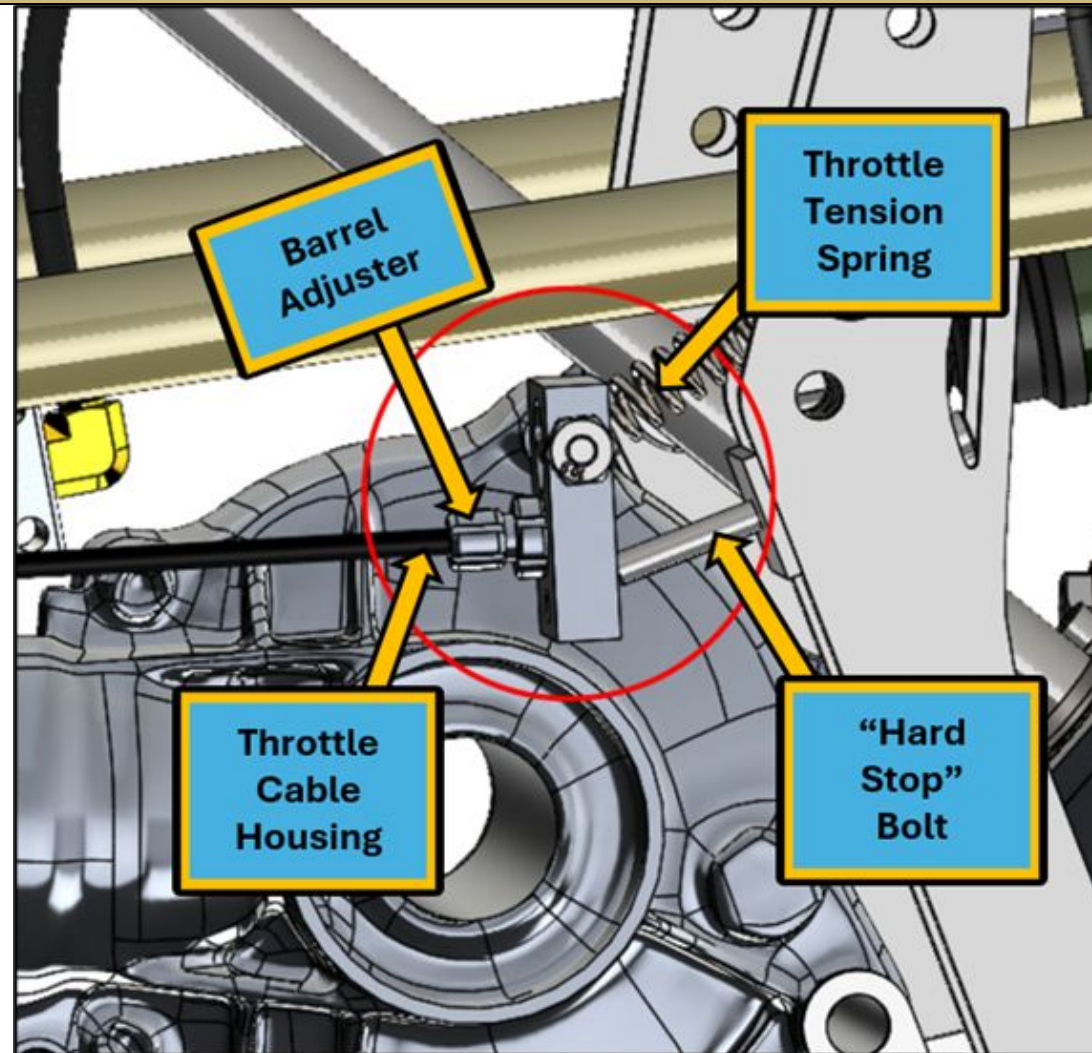
Pedal Exploded Views



Throttle Pedal Critical Parts



Pedal Mounting View

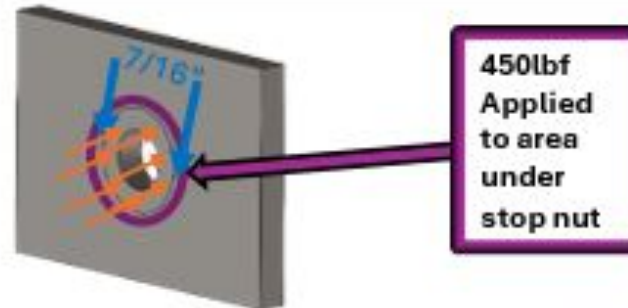


Analysis - Throttle Pedal Critical Part (1)

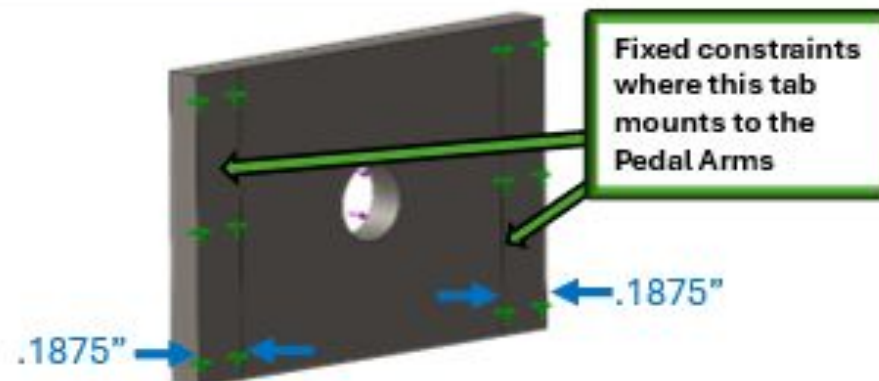
Failure Condition: Frontal impact/Crash where driver's full momentum is transferred into the pedals resulting in a 450lbf load on the Throttle Limiter Bolt Plate.

Justification: Baja SAE Ruleset requires that the brake pedal be able to sustain a 450lbf loading condition in the case of a crash or frontal impact. Therefore, it is safe to assume that the throttle pedal should be able to sustain the same force, for worst case scenario.

Load Diagram:



Constraints:



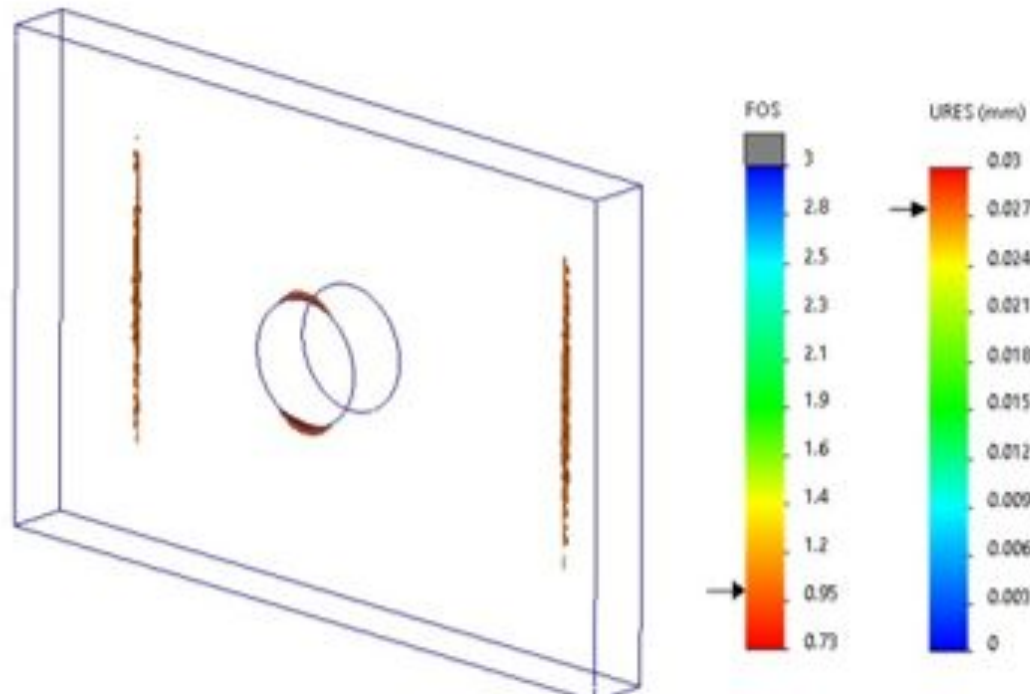
Analysis - Throttle Pedal Critical Part (2)

FOS: 0.73

Equations (If Applicable):

Confidence Level & Explanation: High. The loading force for this part is well known and the mesh/fixturing is simple.

Calculations and/or FEA:

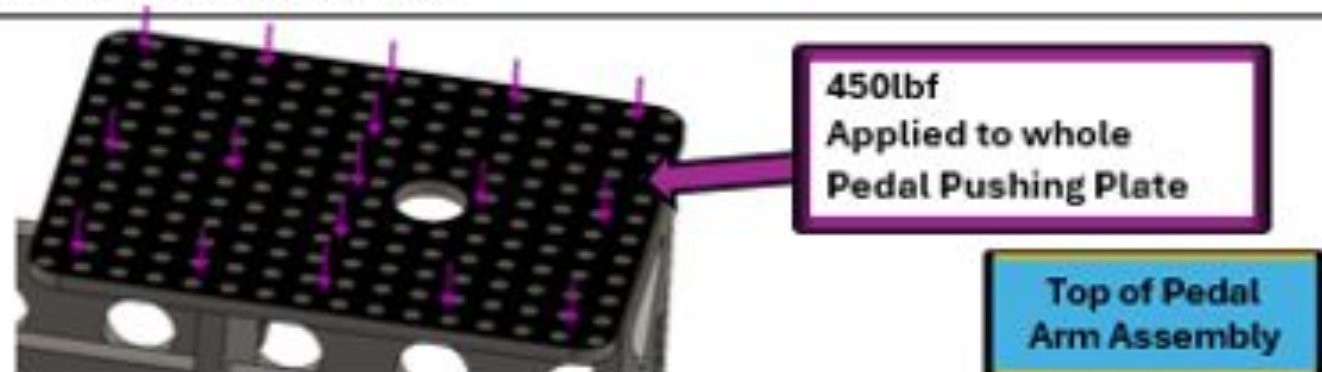


Brake Pedal Analysis - 450lbf Loading Condition (1)

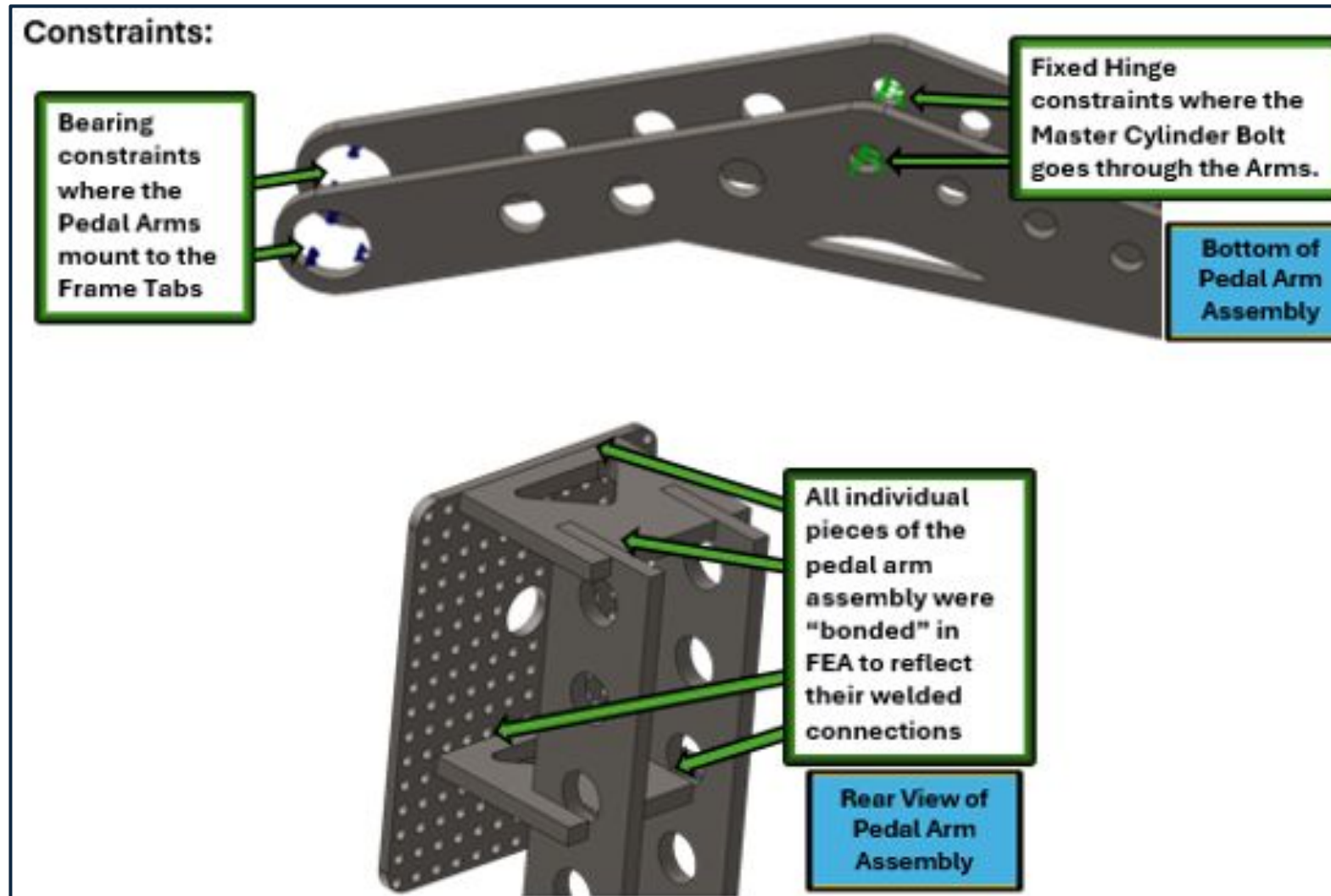
Failure Condition: Frontal impact/Crash where driver's full momentum is transferred into the pedals resulting in a 450lbf load on the pedal arms with locked out master cylinders.

Justification: Baja SAE Ruleset requires that the brake pedal be able to sustain a 450lbf loading condition in the case of a crash or frontal impact. In this condition, you would fully lock out the brake master cylinders and they would act as a rigid fulcrum transmitting load throughout the top and bottom of the pedal arms.

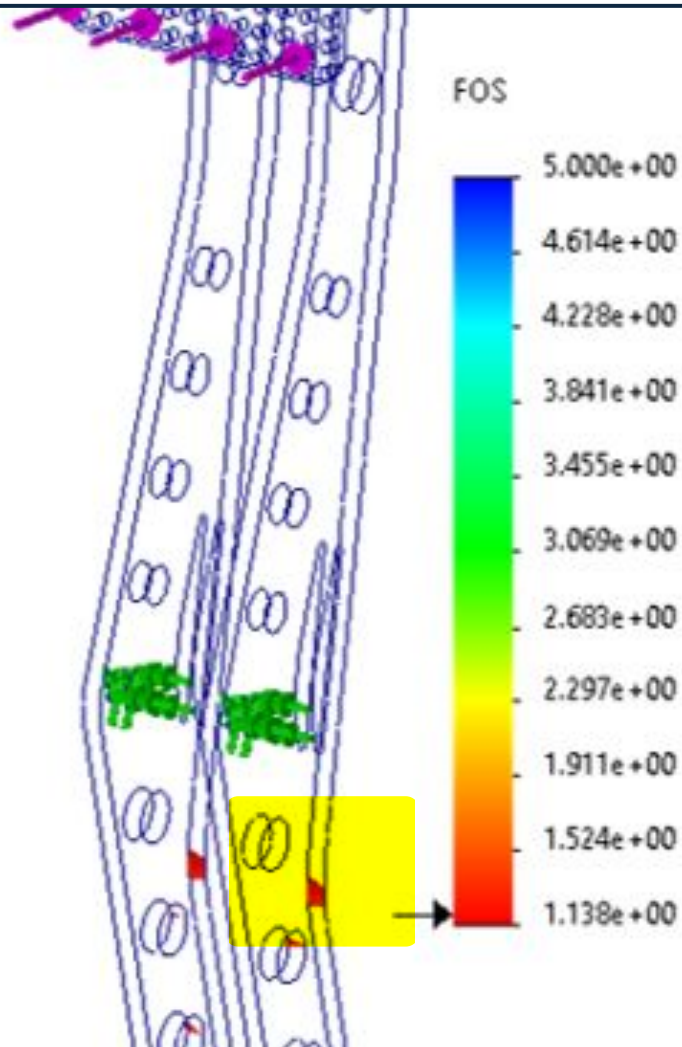
Load Diagram:



Brake Pedal Analysis - 450 lbf Loading Condition (2)



Brake Pedal Analysis - 450 lbf Loading Condition (3)



FOS: 1.138

Confidence level & Explanation:

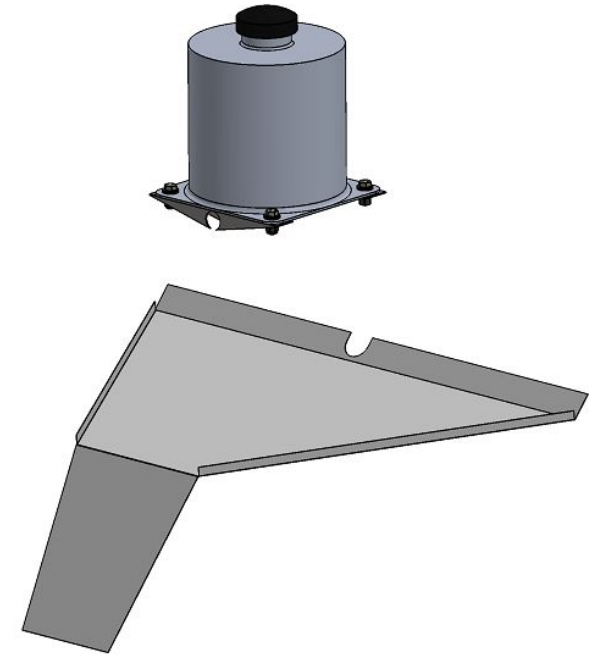
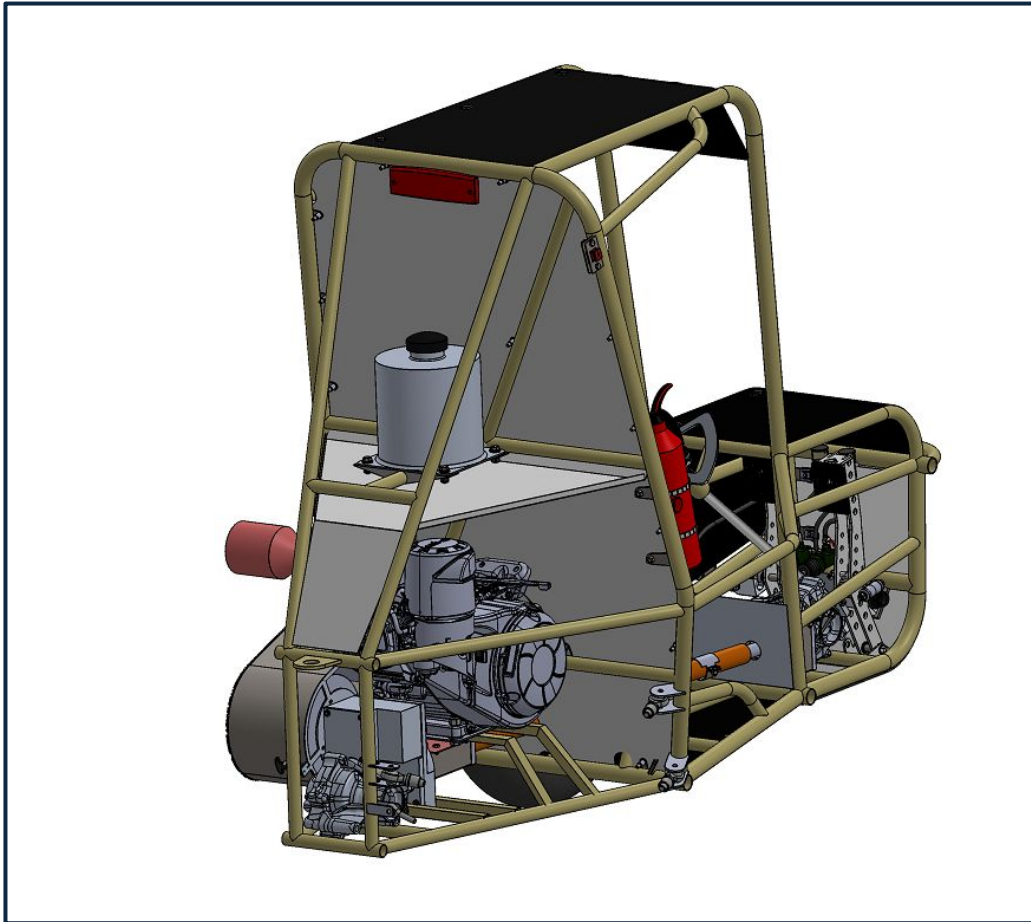
Medium-High Confidence because the loading application is defined in the rules but the constraints available in SolidWorks FEA do not perfectly represent the fixturing of the part

Brakes

- Dual Tilton 78 Series Master Cylinders, 5/8 Bore 1 Inch Stroke
- Weight: 0.4 lbs per master cylinder
- Steel braided lines
- Wilwood dual piston calipers
- Low pressure sensors for brake lights
- 120 lbs to lock all 4 wheels on asphalt
- 46 lbs to lock all 4 wheels on dirt



Fuel Tank and Drip Shield



Gas Tank And Drip Shield Summary

Weight [lbs]	Cost	Rules
~10	~\$5.60	B.6.7

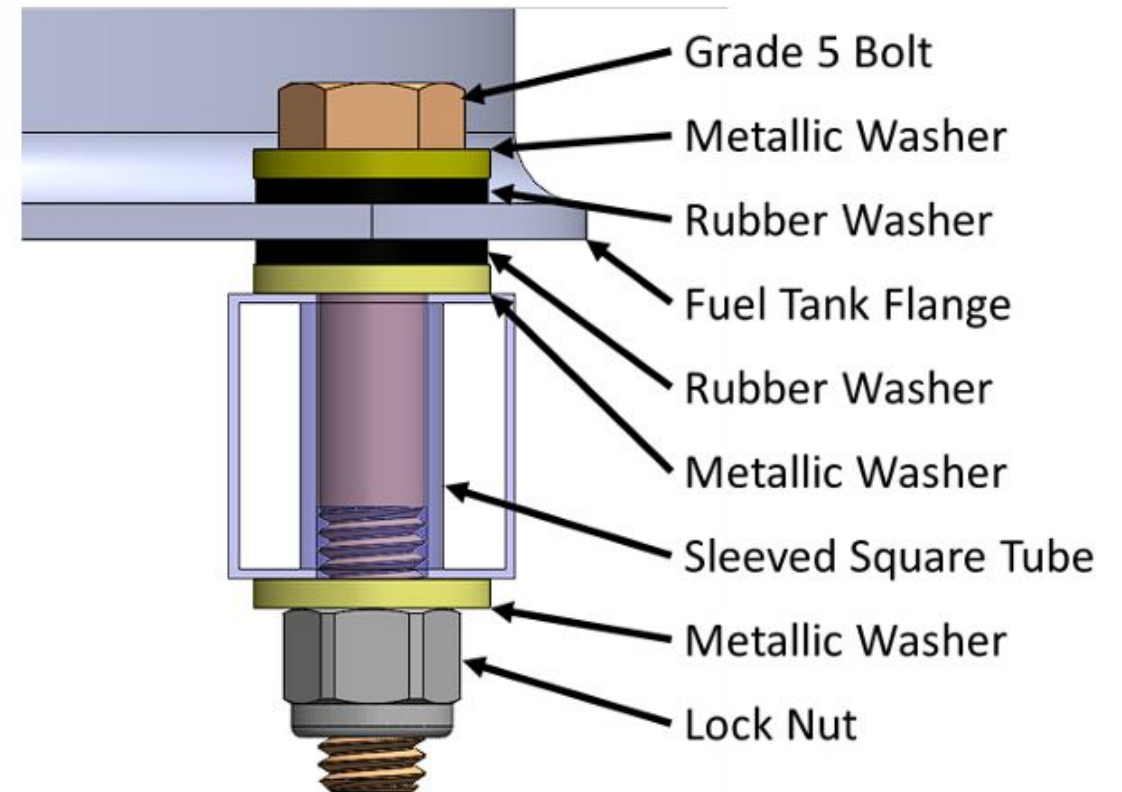
Fuel Tank and Drip Shield Cont.

The only permitted fuel tank is Pyrotec Part Number SFC1000

Mounting is compliant with B.6.5.1 mounting rules

4.8° slant aftward to allow fluids to run off away from the engine

Fluids drip 90° from the exhaust as per B.6.7

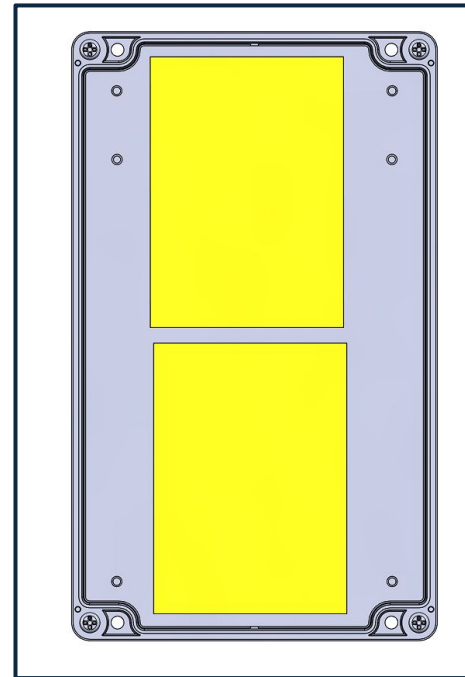


Differential Electronics

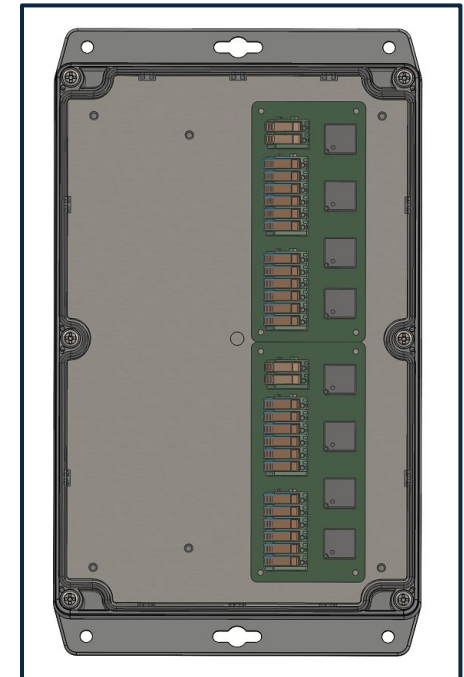
Reusing last years differentials, so we will be reusing last years electronics

Specifically reusing the two PCBs and Dewalt batteries and their respective cases.

Buttons will be mounted to the wheel instead of on the steering column support

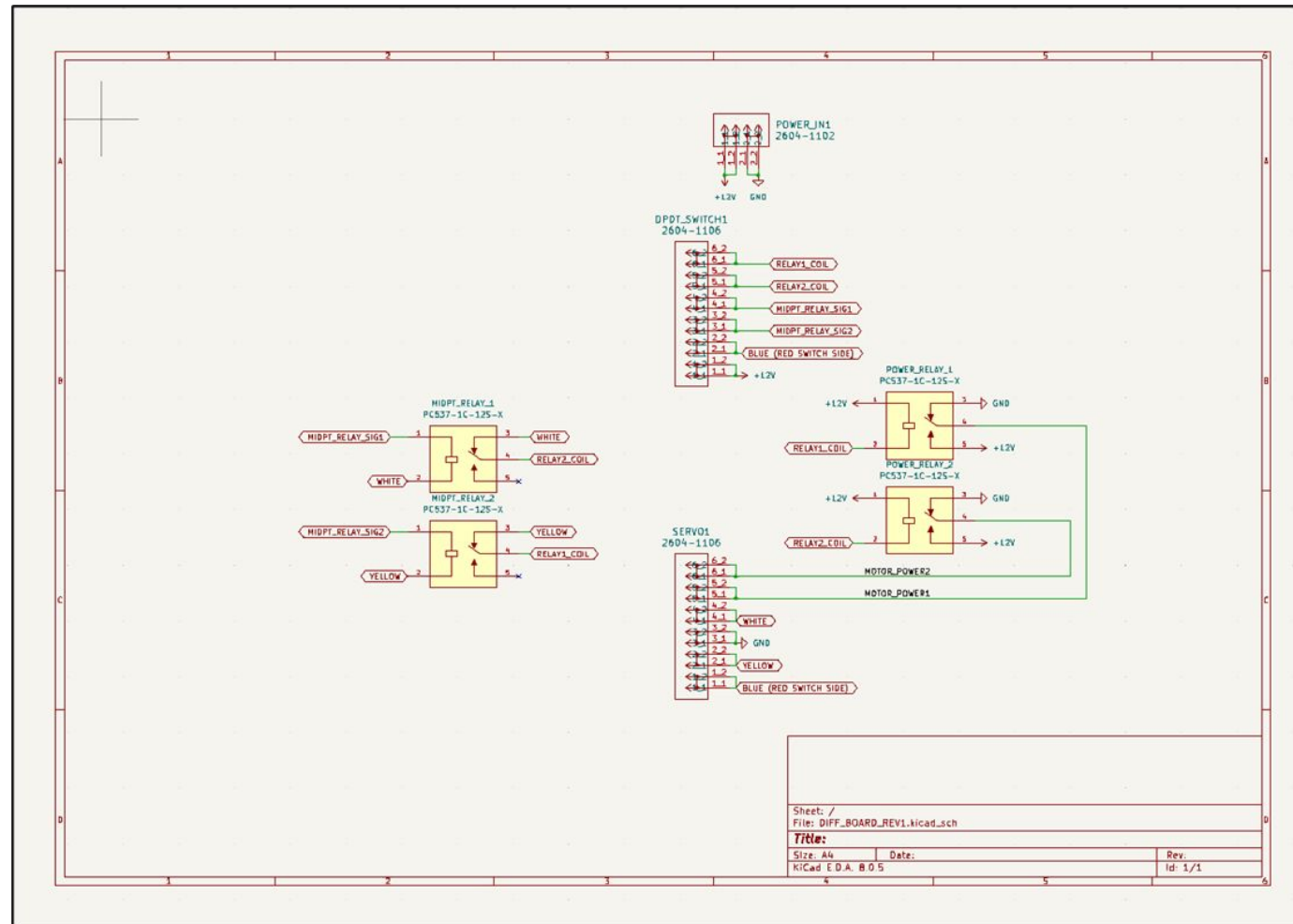


Dewalt Battery Box

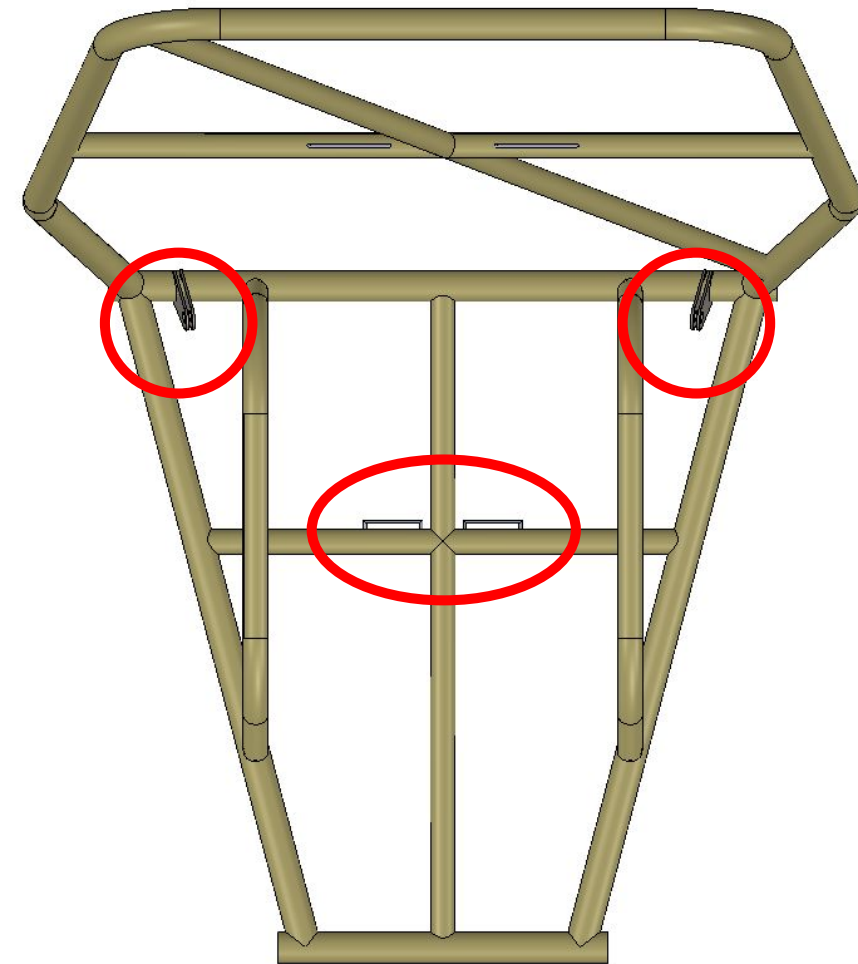
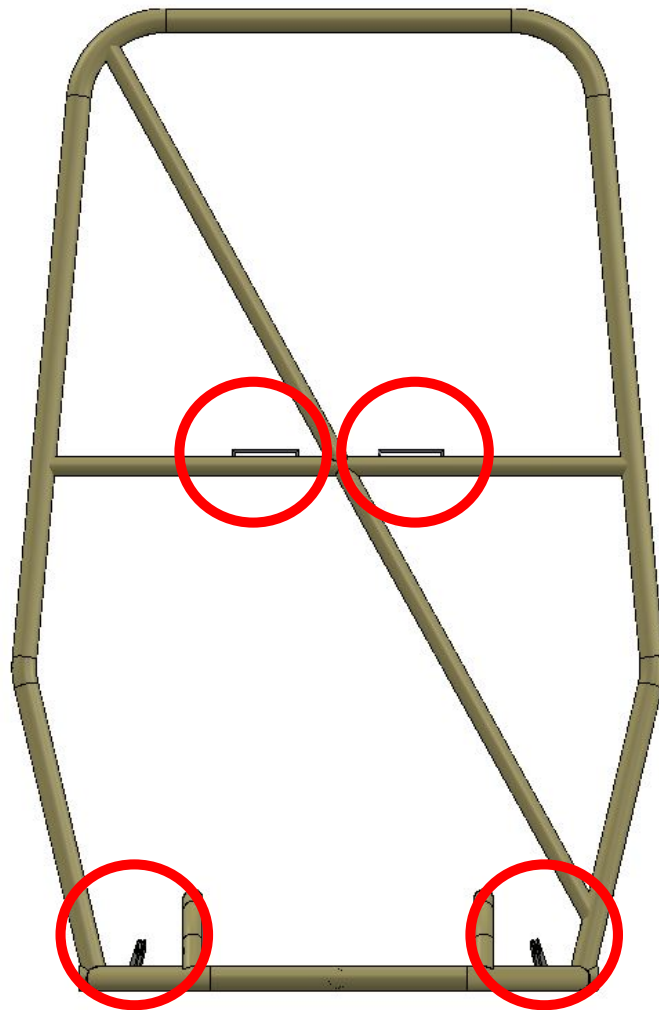
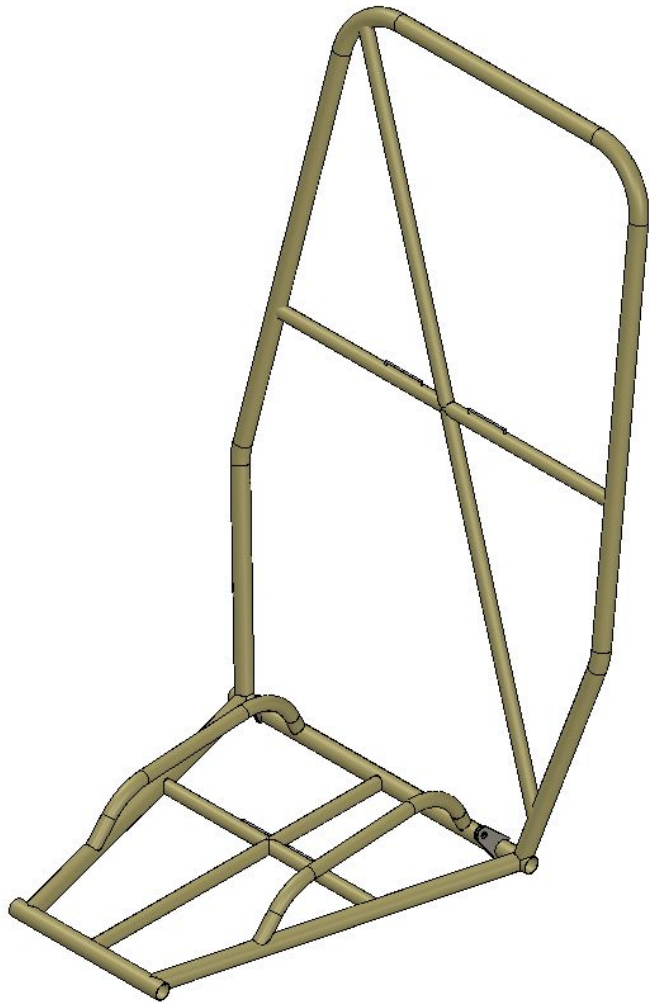


PCB Box

Differential Electronics

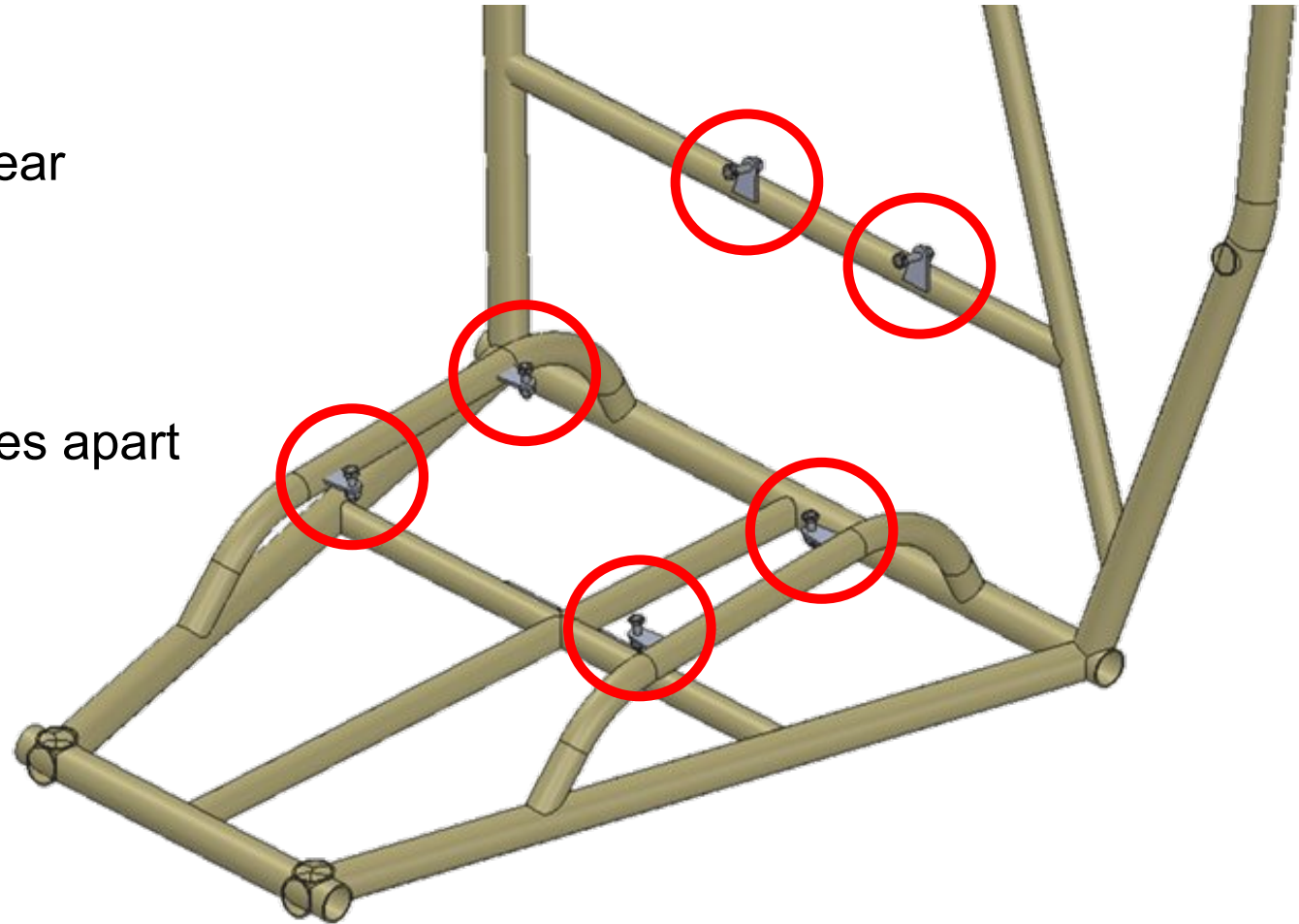


Harness Mounting



Seat Tabs Mounting

- Overall height of seat: 4.34 in.
 - More than 0.5 in. lower than previous year
- Rule B.4.5.3 – Seat Mounting Points
 - 0.17 mm of spacers for seat
 - Rear and bottom seat tabs are 6.5 inches apart
- Rule: B.12.2.1 - Dimensions
 - Bottom tabs: W - 0.78 in, H - 1 in.
 - Rear tabs: W - 1 in, H - 1 in.



Communication Systems Overview

Description	Cost
Cardo PackTalk Edge Headset - Duo Pack	On-hand

- Natural Voice Operation and Automatic volume adjustment instead of buttons/switches
- Unit-to-Unit Max Distance: 1 mile
- Directly addresses communication difficulties between driver and pit at competition from last year

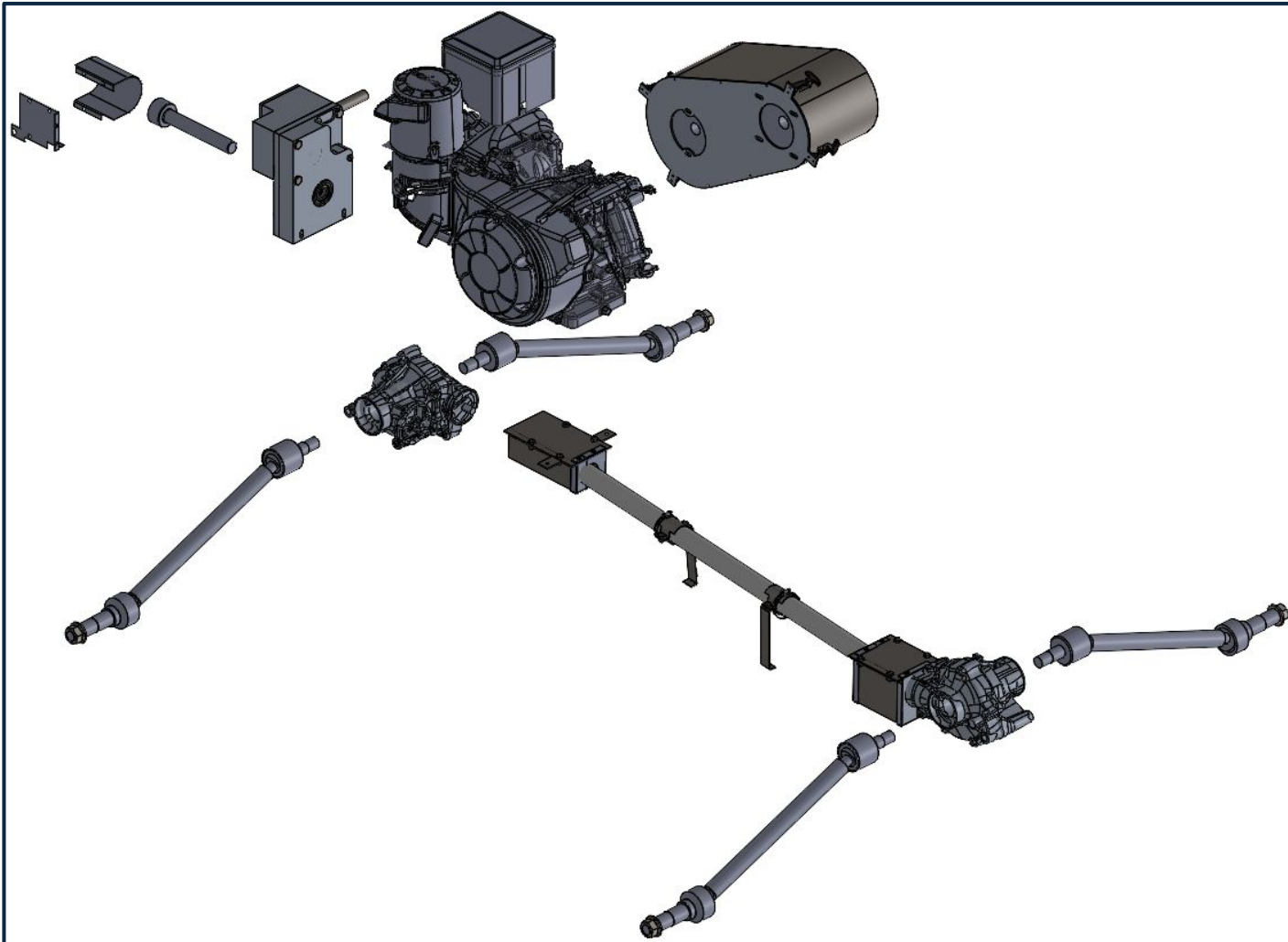


Drivetrain

Drivetrain Goals

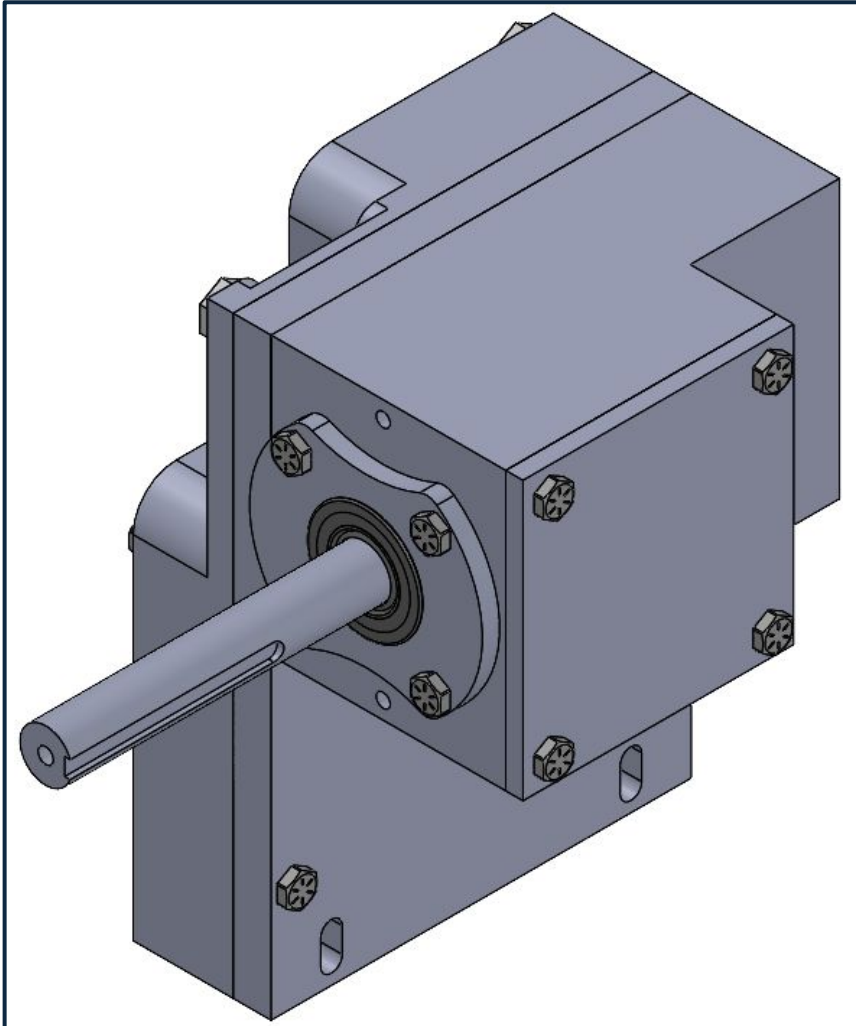
Goal	Reason	Implementation	Difficulties
Complete the endurance without failure In Progress	Failure of specific drivetrain components ends our race, and endurance is the majority of available points	Using worst case scenarios for loading conditions, minimal custom components	Balancing safety factors and design choices while maintaining performance goal capabilities
Increase maximum acceleration by 10% compared to last year In Progress	Acceleration, maneuverability, and a < 2 mile endurance loop are events prioritizing acceleration over top-speed. Analysis indicates poor drivetrain efficiency	Final drive choice, CVT choice, and testing sessions to tune the CVT/diagnose limiting performance factors	Determining main performance limitation
Custom drivetrain components will be removable in < 10 minutes In Progress	Maintenance, testing, and accessibility in the case that replaceable parts fail	Removable covers, accessible fasteners, order of assembly/disassembly	90° power transfer, guard requirements on rotating components
Lower the COG of the drivetrain by 10% compared to last year In Progress	Reduced roll when cornering and increased approach angle	Lowered engine and seat relative to the bottom most chassis member	Lowering the seat while maintaining driveshaft clearance, and transfer case placement

Drivetrain Overall Assembly



Drivetrain	
Weight [lbs]	Cost
This Year: ~185 (CAD)	\$2,200
Last Year: 195 (Actual)	

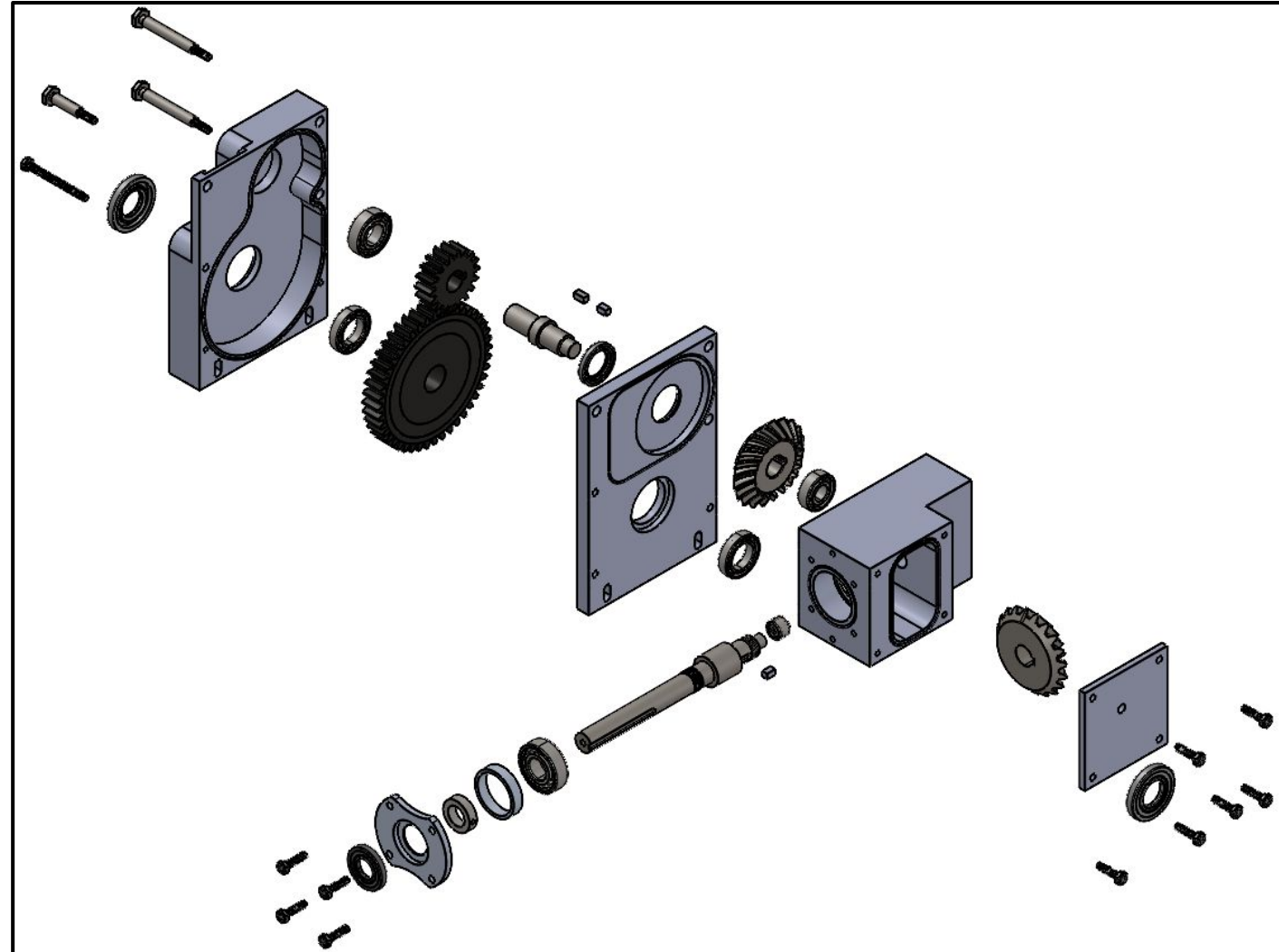
Franken-Box Overview



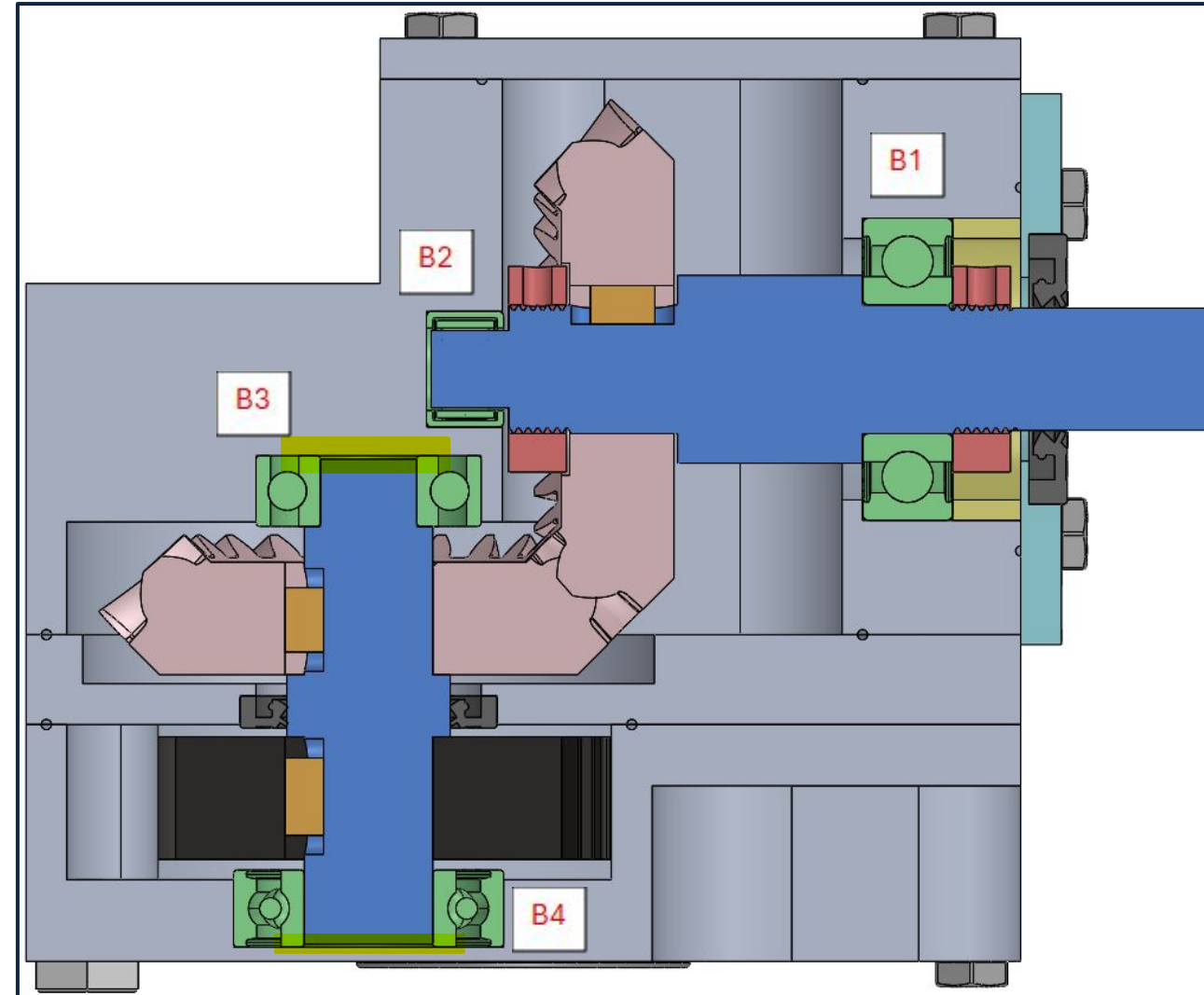
Franken-Box	
Weight [lbs]	Cost
~20	Anticipated: \$1,350

Franken-Box Design Choices

- Engine directly over driveshaft
- CVT kept inside of the frame
- Clamshell design to support both spur and spiral gears
- 2.44:1 gear ratio for final drive of 9:1



Franken-Box Internals



Franken-Box Spiral Gear Stress Analysis

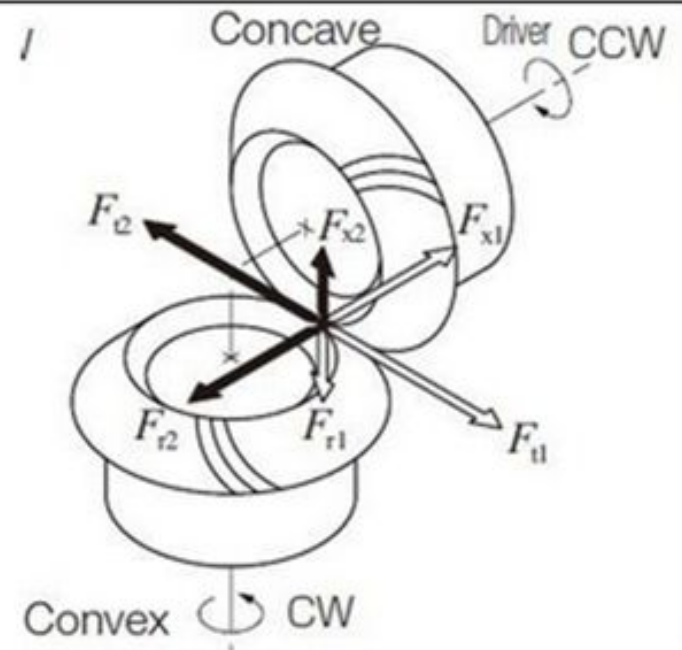
Failure Condition: Engine at peak torque while the front wheels are just off the ground and the Gage CVT is at its minimum gear ratio.

Justification: If this occurs, then the Spiral gears will experience the maximum realistic torque possible. Making sure the gears survive this will increase reliability

Load Diagram:

Ref: [KHK Gear Forces Reference](#)

Tangential, radial, and axial forces generated by the mesh of the spiral bevel gears.



Franken-Box Spiral Gear Stress Analysis

FOS: 1.45	Confidence Level & Explanation Medium confidence level. Accurate equations for torque loading and comparison to last years car.
Equations (If Applicable): $\sigma_F = F_{tm} \frac{Y_F Y_e Y_\beta Y_C}{0.85 \cos \beta_m m b} \frac{R_e}{R_e - 0.5 b} \left(\frac{K_M K_V K_O}{K_L K_{FX}} \right) K_R$	
Calculations and/or FEA: Safety factor under JGMA standards: 1.45 Last years safety factor under JGMA standards: 1.01 Safety factor under last years conditions (AGMA standard): 2.57 Last years safety factor under last years conditions (AGMA standard): 2.34	

AGMA:

$$\sigma_t = \underbrace{\left(\frac{W_t}{F J m} \right)}_{\text{Torque}} \cdot \underbrace{\left(\frac{K_a K_m}{K_v} \cdot K_S K_B K_I \right)}_{\text{Stress Factors}}$$

JIS:

$$\sigma_F = F_{tm} \frac{Y_F Y_\epsilon Y_\beta Y_C}{0.85 \cos \beta_m m b} \frac{R_e}{R_e - 0.5 b} \left(\frac{K_M K_V K_O}{K_L K_{FX}} \right) K_R$$

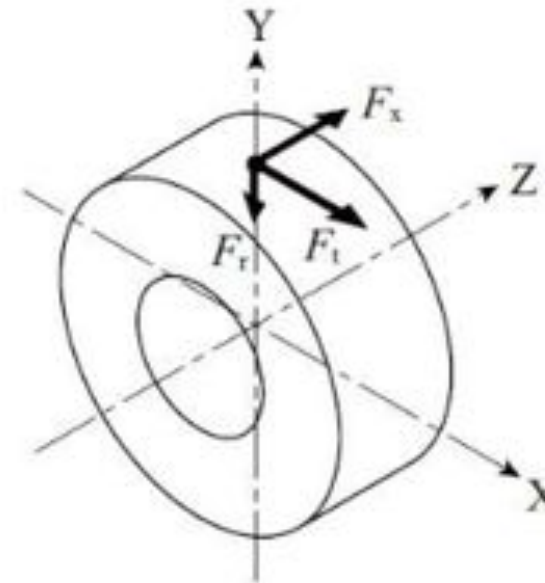
Franken-Box Spur Gear Stress Analysis

Failure Condition: Engine at peak torque while the front wheels are just off the ground and the Gage CVT is at its minimum gear ratio.

Justification: : If this occurs, then the spur gears will experience the maximum realistic torque possible. Making sure the gears survive this will increase reliability

Load Diagram:

Spur gears only have tangential and radial forces.



Franken-Box Spur Gear Stress Analysis

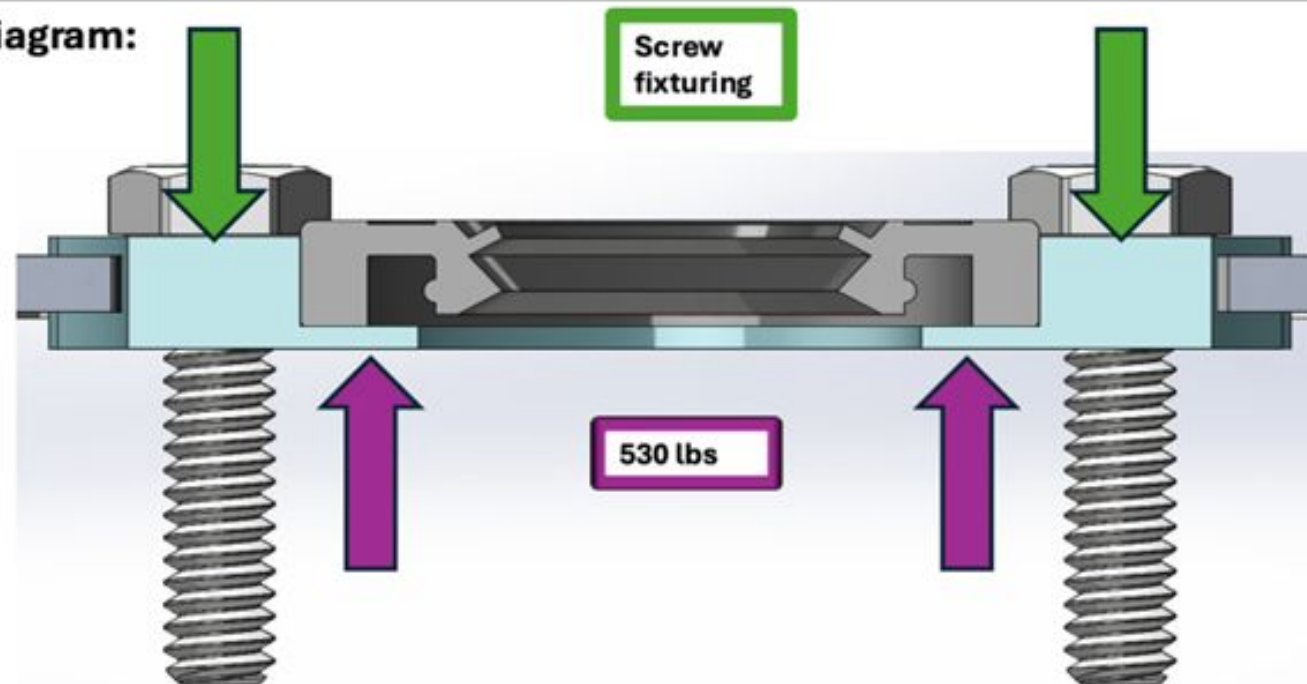
FOS: 2.8	Confidence Level & Explanation: Medium confidence. Equation well known for bending tooth strength, loads well known.
Equations (If Applicable): $\sigma_F = F_t \frac{Y_F Y_\epsilon Y_\beta}{m_n b} \left(\frac{K_V K_O}{K_L K_{FX}} \right)$	
Calculations and/or FEA: <pre>% Spur bending tooth strength Y_lewis = 2.9; S_Ye = 1/mp; S_Kv = 1.7; S_Kl = 1.1; S_t_one = S_t/mp; Obt_lewis = S_t*S_Ye*S_Kv*Y_lewis/(S_module*S_face_width*S_Kl); Lewis_SF = mp*Ob_tlim/Obt_lewis;</pre> Spur Pinion Safety Factor: 2.81	

Franken-Box Flange Tearout Analysis

Failure Condition: Full axial force from spiral bevel miter gears on B1, supported by the bearing flange plate

Justification: If the engine is at peak torque, and the front wheels are just off the ground, and the CVT is at the minimum gear ratio, the gears would experience the maximum possible realistic torque, placing the maximum axial force on the bearing plate

Load Diagram:



Franken-Box Flange Tearout Analysis

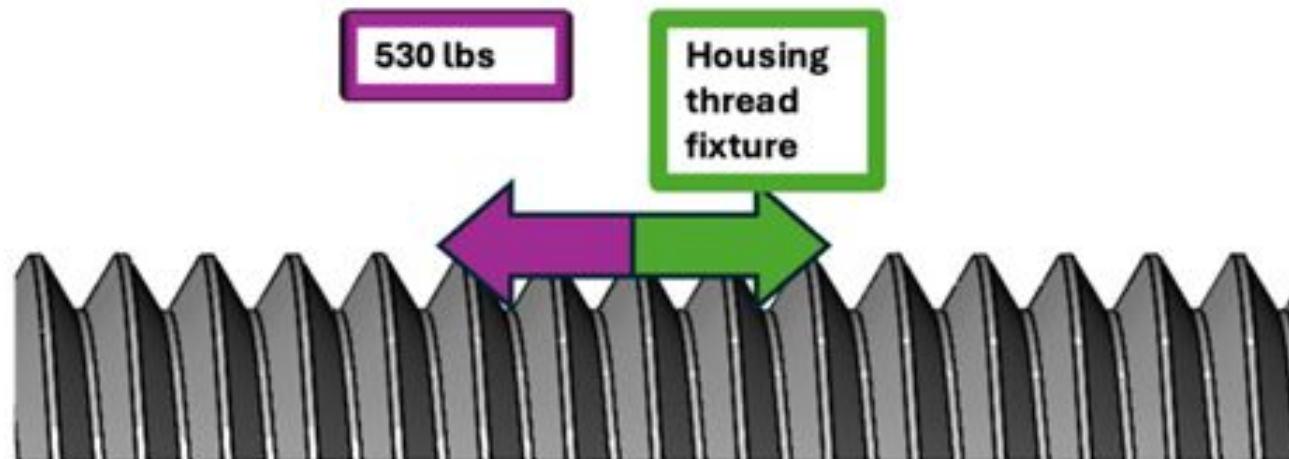
FOS: 14.7	Confidence Level & Explanation Medium confidence. Shear tear out calculation for the plate thickness.
Equations (If Applicable): $\tau = \frac{F}{A}$	
Calculations and/or FEA: <pre>%% Tearout stress on the bearing bore step % Aluminum shear yield strength SY_al = 30000; %psi A_shear_mid = pi*42*0.05/25.4; F_shear_mid = 530; %lbs S_shear_mid = F_shear_mid/A_shear_mid; SF_shear_mid = SY_al/S_shear_mid; Shear tearout Factor of safety for B1: 14.7</pre>	

Franken-Box Bolt Tearout Analysis

Failure Condition: Spiral Bevel gear at full axial load against B1 where three out of the four bolts have backed out of the bearing flange plate.

Justification: Excessive vibrations may cause the bolts to back out of the housing, and in the worse case, there would only be one bolt left to withstand the load.

Load Diagram:



Franken-Box Bolt Tearout Analysis

FOS: 20	Confidence Level & Explanation
Equations (If Applicable): $A = \pi(dr)(wi)(Tpi)$ $\tau = \frac{F}{A}$	Medium confidence. Equations known to be accurate and the failure criteria is extremely unlikely.
Calculations and/or FEA: <pre>% Shear stress to strip the threads wi = 0.8; Q20p = 1/20; Astrip = pi*dr*wi*Q20p; F_axial_input = (2.3553e+03)/4.4482216152605; %lbs % Shear strength Sys_q20 = 30000; %psi num_threads = 20; SF = num_threads*(Sys_q20*Astrip)/(F_axial_input); Safety factor with only one bolt: 20.00</pre>	

Franken-Box Gear Ratio Analysis

- Calculated final drive past the CVT
- Power limited acceleration
- Acceleration from drivetrain goals
- Assumed 60% efficiency

$$\frac{W + W_r}{g} a_x = \frac{T_e N \eta}{r} - R_r$$

$$N = 9.00$$

Franken-Box Shaft Stress Analysis

Failure Condition: Input shaft experiences yielding due to bending moments and torques greater than the yield strength of our material, normalized AISI 4140 Steel.

Justification: The major loads that would cause failure on this shaft are the bending moment and torque caused by the Secondary CVT, cantilevered against the bearings. Von Mises and Goodman fatigue are well recognized methods for calculating shaft failure.

Load Diagram:



FOS: 1.5

Equations (If Applicable):

$$d = \left\{ \frac{32N_f}{\pi} \left[\frac{\sqrt{(K_f M_a)^2 + \frac{3}{4}(K_{fs} T_a)^2}}{S_f} + \frac{\sqrt{(K_{fm} M_m)^2 + \frac{3}{4}(K_{fsm} T_m)^2}}{S_{at}} \right] \right\}^{\frac{1}{3}}$$

$$S_f = C_{load} C_{size} C_{surf} C_{temp} C_{reliab} S'_f$$

Confidence Level & Explanation:

Medium-High: Given that our shaft diameters are all larger than our minimum calculated diameter, and assuming a certain level of heat treatment, this should perform well.

Results:

- Input shaft minimum diameter: 0.71546 in
 - Used diameter: 0.75 in
- Pinion shaft minimum diameter: 0.62744 in
 - Used diameter: 0.75 in
- Output shaft minimum diameter: 0.75481 in
 - Used diameter: 1 in

Franken-Box Shaft Stress Analysis

Variable	Notes	Value	Variable	Notes	
K_f	Fatigue stress concentration factor	1	T_a	Alternating torque	5000 Nmm
K_{fm}	Mean stress fatigue concentration factor	1	T_m	Mean torque	100000 Nmm
K_{fs}	Shear stress concentration factor	1	S_f	Fatigue limit, found through correction factors	276.8344 Mpa
K_{fsm}	Mean stress fatigue concentration factor	1	S_{ut}	Ultimate tensile strength of material	1020 Mpa
M_a	Moment amplitude,	73417 Nmm	N_f	Safety factor	1.5
			S_f'	Uncorrected fatigue limit	510 MPa

Results:

- Input shaft minimum diameter: 0.71546 in
 - Used diameter: 0.75 in
- Pinion shaft minimum diameter: 0.62744 in
 - Used diameter: 0.75 in
- Output shaft minimum diameter: 0.75481 in
 - Used diameter: 1 in

Franken-Box Key Stress Analysis

Failure Condition: The keys will yield if they experience too much bearing or shear stress, binding up in the shaft and causing failure.

Justification: Bearing and shear stress are the two different ways keys can fail, with the forces coming from the torque of the shaft and gears.

Load Diagram:



FOS: Minimum of 1.5

Equations (If Applicable):

$$\sigma_x = \frac{F}{A_{bearing}} \quad \tau_{xy} = \frac{F}{A_{shear}}$$

Confidence Level & Explanation:

High: According to calculations, every key is greater than our minimum factor of safety.

Results:

Material Properties for 1045 Steel with Brinell B87 Hardness
 Ultimate Shear Strength: 329 MPa
 Yield Strength: 450 MPa
 Ultimate Tensile Strength: 570 MPa
 Minimum Factor of Safety Goal for Keys: 1.5

 Key #1: Input Shaft Spiral Bevel Gear Key

Shear stress: 142.3873 MPa

about a 2.3106 FOS

Crush stress: 284.7745 MPa

about a 1.5802 FOS

 Key#2_1: Pinion Shaft Spiral Bevel Gear Key

Shear stress: 149.9813 MPa

about a 2.1936 FOS

Crush stress: 299.9625 MPa

about a 1.5002 FOS

 Key#2_2: Pinion Shaft Spur Gear Key

Shear stress: 124.9844 MPa

about a 2.6323 FOS

Crush stress: 249.9688 MPa

about a 1.8002 FOS

 Key#3: Output Shaft Spur Gear Key

Shear stress: 141.7146 MPa

about a 2.3216 FOS

Crush stress: 283.4291 MPa

about a 1.5877 FOS

Franken-Box Key Stress Analysis

Calculations and/or FEA:

$$\sigma_x = \frac{2T}{(H - \text{depth}) * l * D} * \frac{1}{10^6} = 284.8 \text{ MPa} (2.3 \text{ FoS})$$

$$\tau_{xy} = \frac{2T}{W * l * D} * \frac{1}{10^6} = 142.4 \text{ MPa} (1.6 \text{ FoS})$$

Variable	Notes	Value
W and H	Width and height of key	7 mm
D	Shaft Diameter	0.75 in
T	Torque (worse case)	100 Nm
L	Key length	10 mm
depth	Key slot depth	3.5 mm

Results:

Material Properties for 1045 Steel with Brinell B87 Hardness
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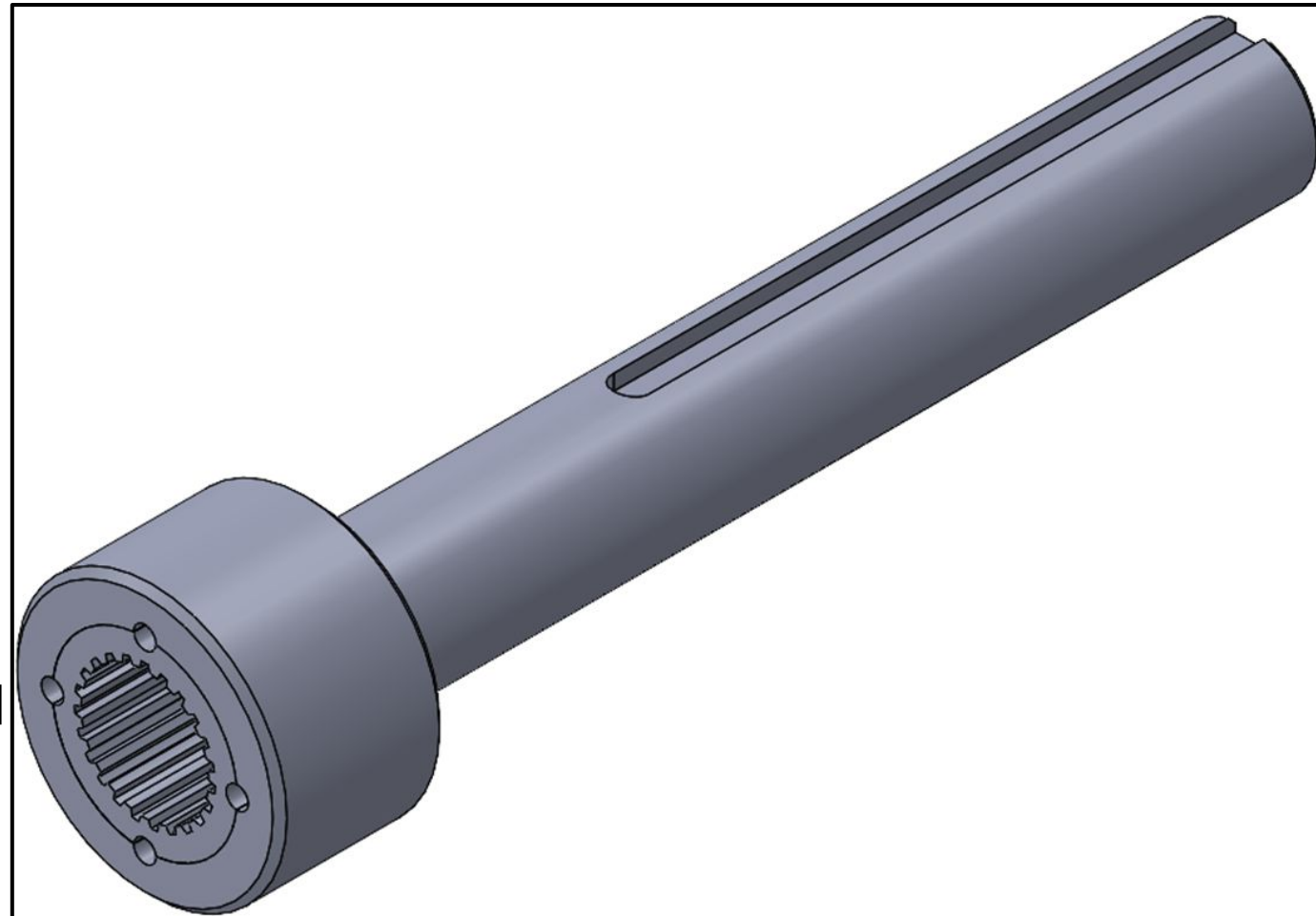
about a 2.3216 FOS

Crush stress: 283.4291 MPa

about a 1.5877 FOS

Franken-Box Output Shaft

- Connects transfer case to rear Yamaha differential
- OEM spline pressed in shaft and rotationally locked with roll pins
- Keyed slot interfaces with spur gear and driveshaft U-joints



Franken-Box Bearing Analysis

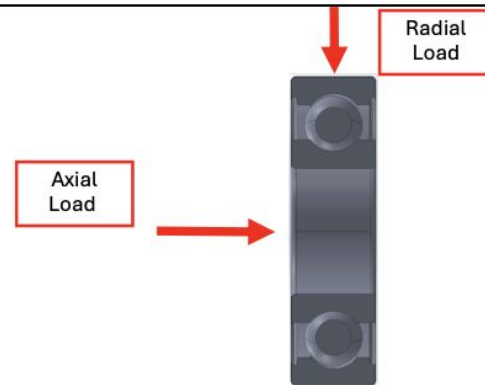
Failure Condition:

Bearing failure can result from excessive load, poor lubrication, contamination, or misalignment, leading to wear, overheating, and eventual seizure.

Justification:

Each bearing will experience either radial or axial loads, or both, due to the forces transmitted through each shaft as well as the tension from the CVT belt.

Load Diagram:



FOS (Based on a 50-hour life):

B1: 16.2
B2: 13.8
B3: 7.2
B4: 11.2
B5 & B6: 17.4

Equations:

$$P = XVF_r + YF_a$$

$$L_{10} = \left(\frac{C}{P}\right)^3$$

$$\begin{aligned} \Sigma F_x &= 0 \\ \Sigma F_y &= 0 \\ \Sigma M &= 0 \end{aligned}$$

Confidence Level & Explanation:

Medium to high. Bearings L_{10} values are much higher than we expect to be running the vehicle for.

Franken-Box Bearing Analysis

Equations:

$$P = XV F_r + Y F_a \quad \Sigma F_x = 0$$

$$L_{10} = \left(\frac{C}{P}\right)^3 \quad \Sigma F_y = 0$$

$$\quad \quad \quad \Sigma M = 0$$

Calculations:

All L_{10} were calculated (in hours) via SKF's bearing selector, which yielded:

B1: 809

B2: 692

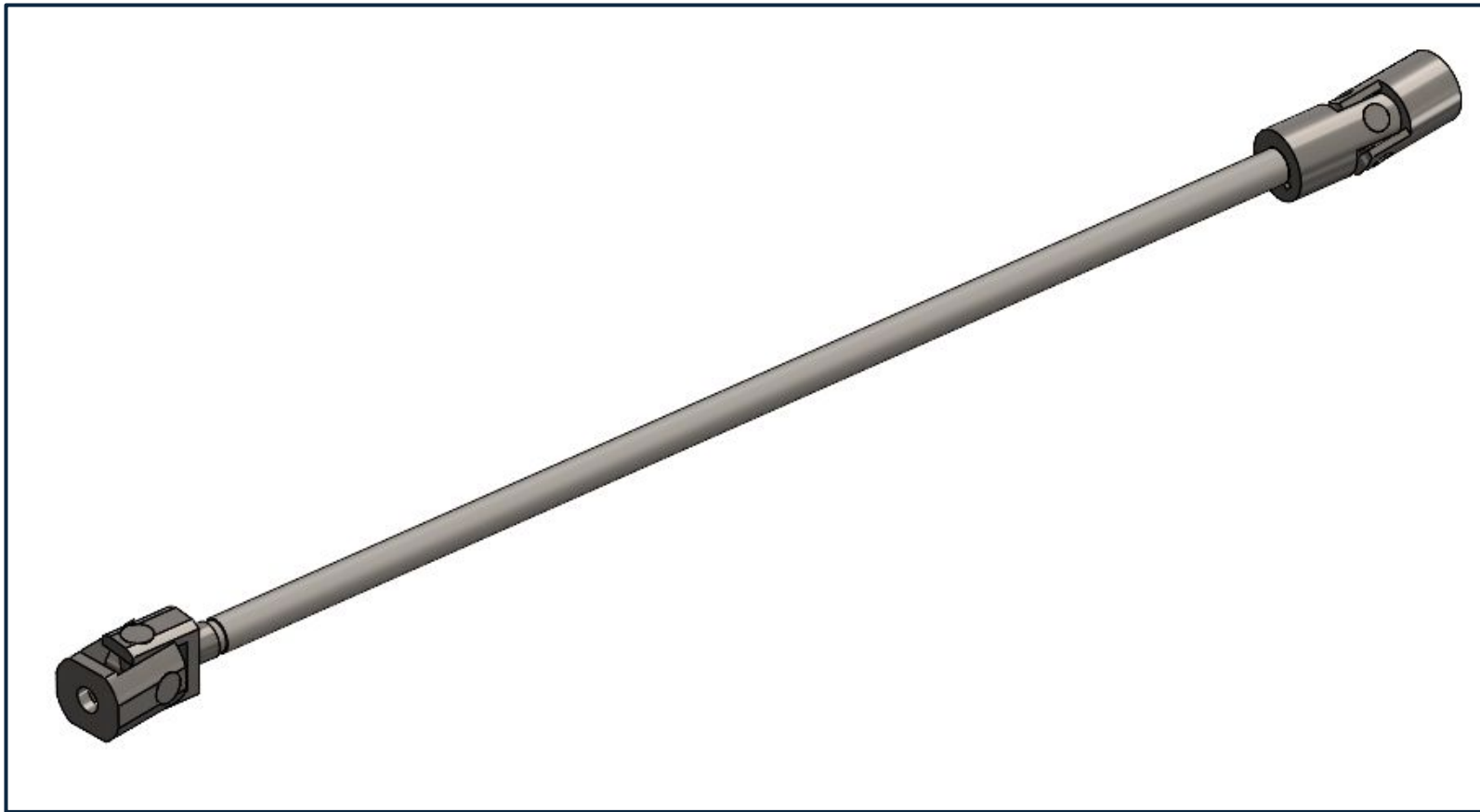
B3: 362

B4: 558

B5: 871

B6: 871

Driveshaft Overview



Driveshaft Summary		
Weight [lbs]	Cost	Rules
~8.3 (CAD)	Anticipated: \$170	B.9.2.2.2

Driveshaft Overview

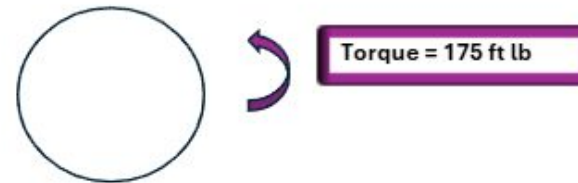


Driveshaft Calculations

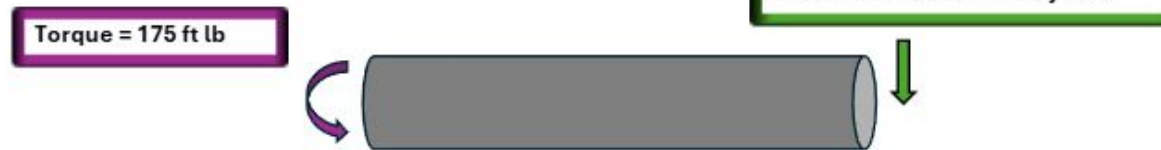
Failure Condition: The maximum CVT ratio produces too much torque for the driveshaft to withstand.

Justification: the maximum torque condition comes from calculating the torque on the driveshaft after the transfer case reduction and maximum possible CVT ratio, making it the highest torque we can possibly achieve.

Load Diagram:



Constraints:



FOS: 2

Confidence Level & Explanation:

Medium- Since factor of safety is calculated at the highest possible calculated torque but there is not real world measured torque values.

Driveshaft Calculations

Driveshaft Torque	Input/Output	Equations	Calculation
4130 Alloy Normalized	OD (in)	outer diameter	1
	Thickness (in)	wall thickness	0.049
	G (ksi)	Shear Modulus	11600
	J(in^4)	Polar moment of inertia	0.03319
	T (lb-ft)	Applied torque	175
	Y (Ksi)	Yield strength	63.1

$$\tau = \frac{Tr}{J}$$

Driveshaft Calculations

Failure Condition: Driveshaft failure from excessive angle of twist during initial torque impulse.

Justification: When the vehicle starts to move from a stationary position, the driveshaft will momentarily be experiencing full torque on one side and fixed position on the other side.

Load Diagram:



Constraints:



Angle of Twist: 11 Degrees

Equations (If Applicable):

$$\phi = \frac{TL}{JG}$$

Confidence Level & Explanation:

Medium- Since angle of twist is calculated at the highest possible calculated torque but there is not real world measured torque values.

Driveshaft Calculations

Driveshaft Angle of Deflection	Input/Output	Equations	Calculation
4130 Alloy Normalized	T (in-lb)	Torque	2000
	J(in^4)	Polar Moment of Inertia	0.03319
	G (ksi)	Shear Modulus	11600
	L (in)	Length	35

$$\phi = \frac{TL}{JG}$$

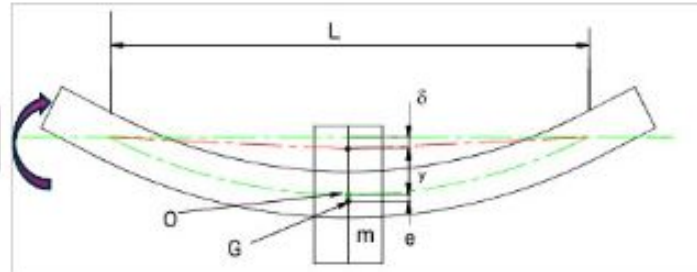
Driveshaft Calculations

Failure Condition: Driveshaft reaches its critical speed and starts whirling.

Justification: The driveshaft is a relatively thin and long metal tube, and the rpm we plan on running it at may be near the shafts critical speed.

Load Diagram:

Max speed= 1600 rpm



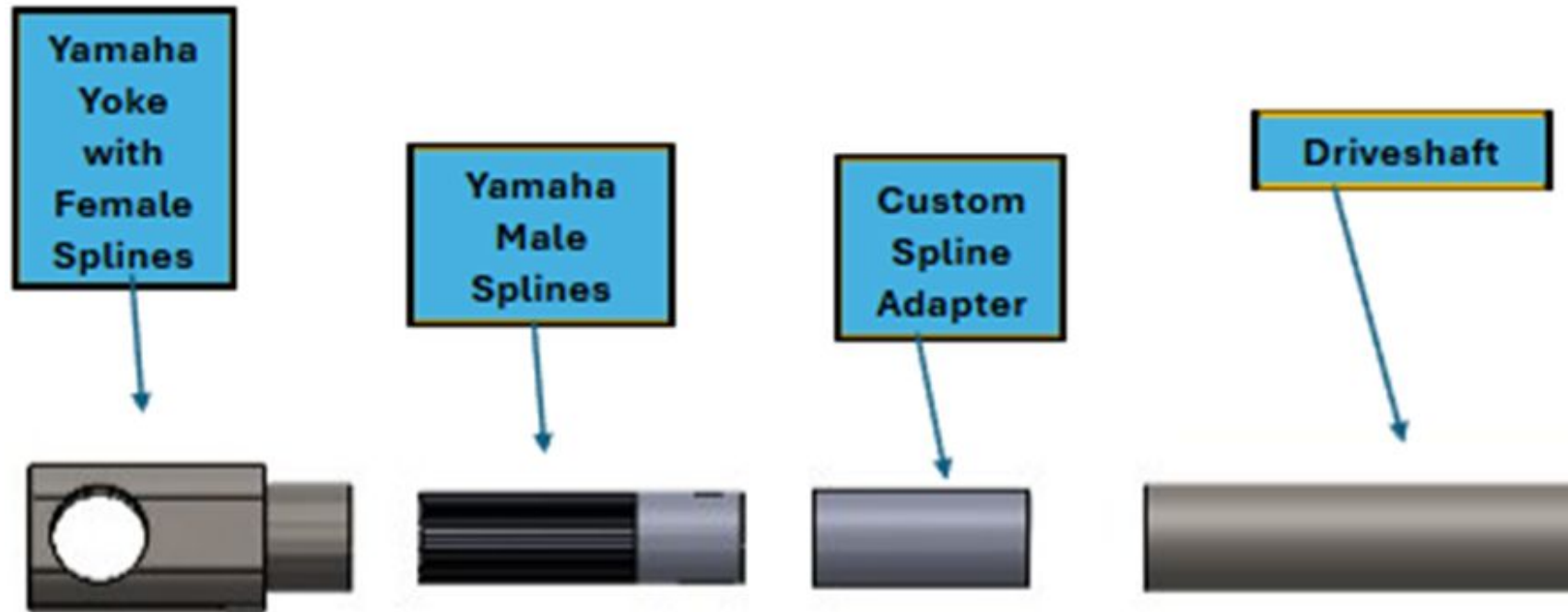
Critical speed: 3620 rpm

Driveshaft Calculations

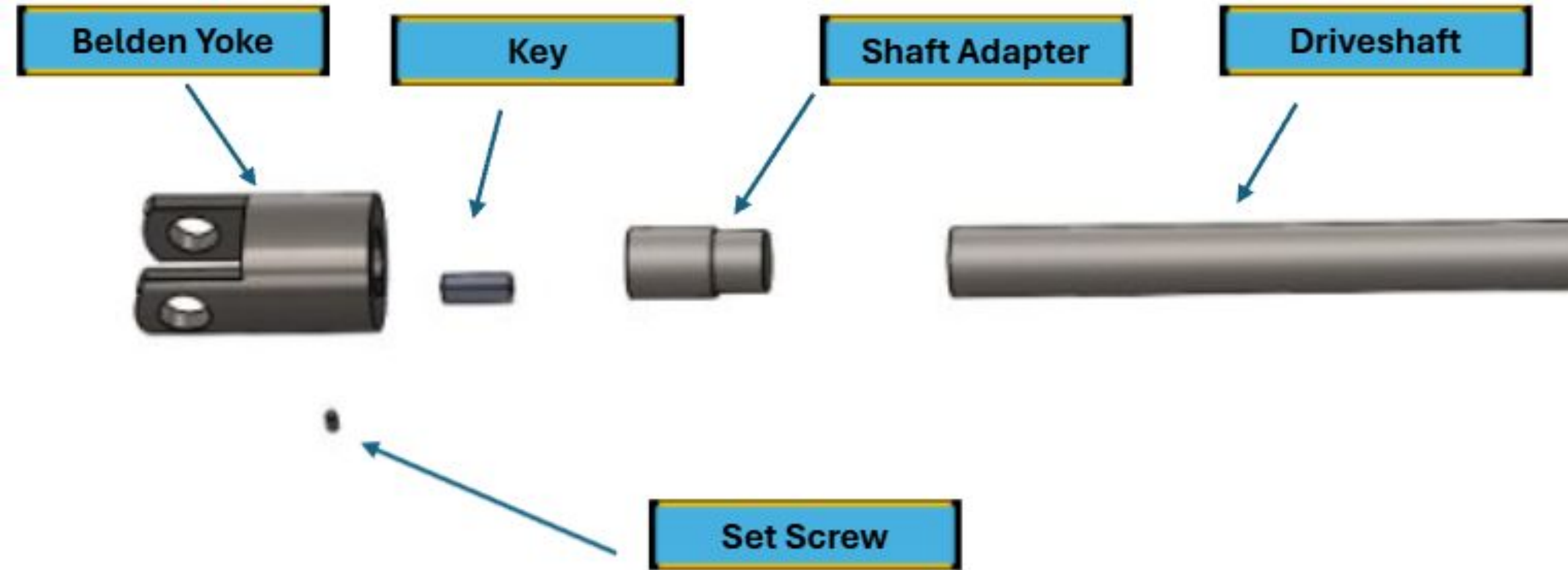
Driveshaft Critical Speed	Input/Output	Equations	Calculation
4130 Alloy Normalized	OD (in)	outer diameter	1
	Thickness (in)	wall thickness	0.049
	E (Ksi)	Modulus of elasticity	29700
	Density (lb/in^3)	Density	0.284
	L (in)	Length	35
	I (in^4)	Area Moment of Inertia	0.01659
	A (in^2)	Area	0.1464
	g (in/s^2)	acceleration due to Gravity	386

$$N_c = 9.55 \sqrt{\frac{48EIg}{\rho AL^4}}$$

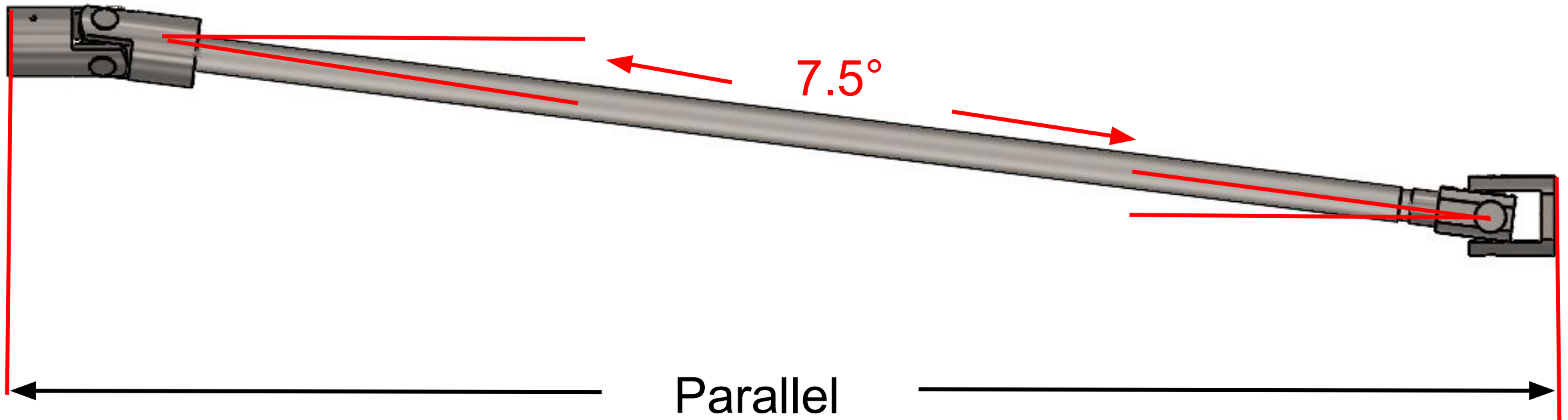
Driveshaft Connections



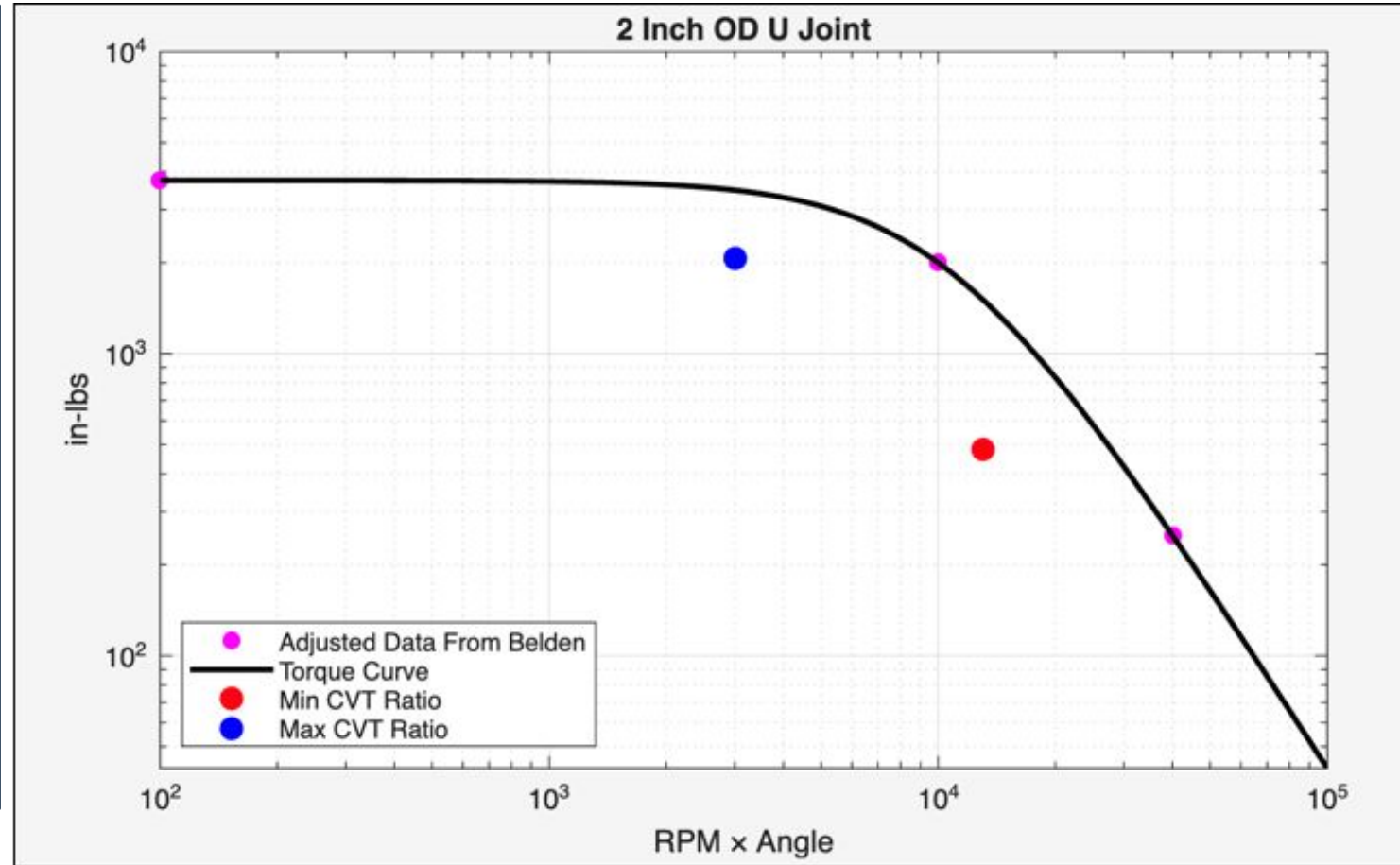
Driveshaft Connections



Driveshaft Phasing

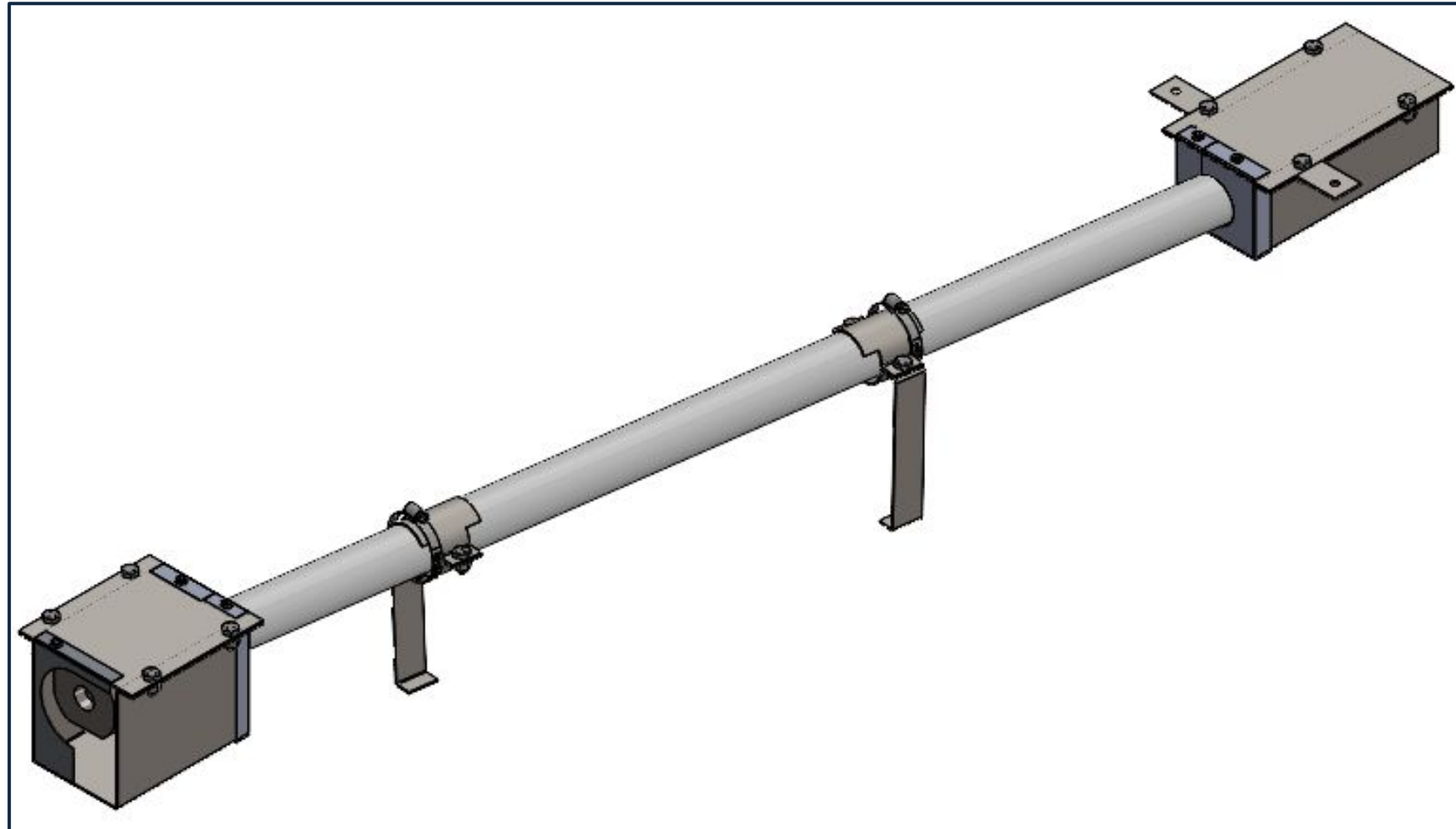


Universal Joint Selection



Graph of U-joint angle vs. RPM and Transmittable Torque

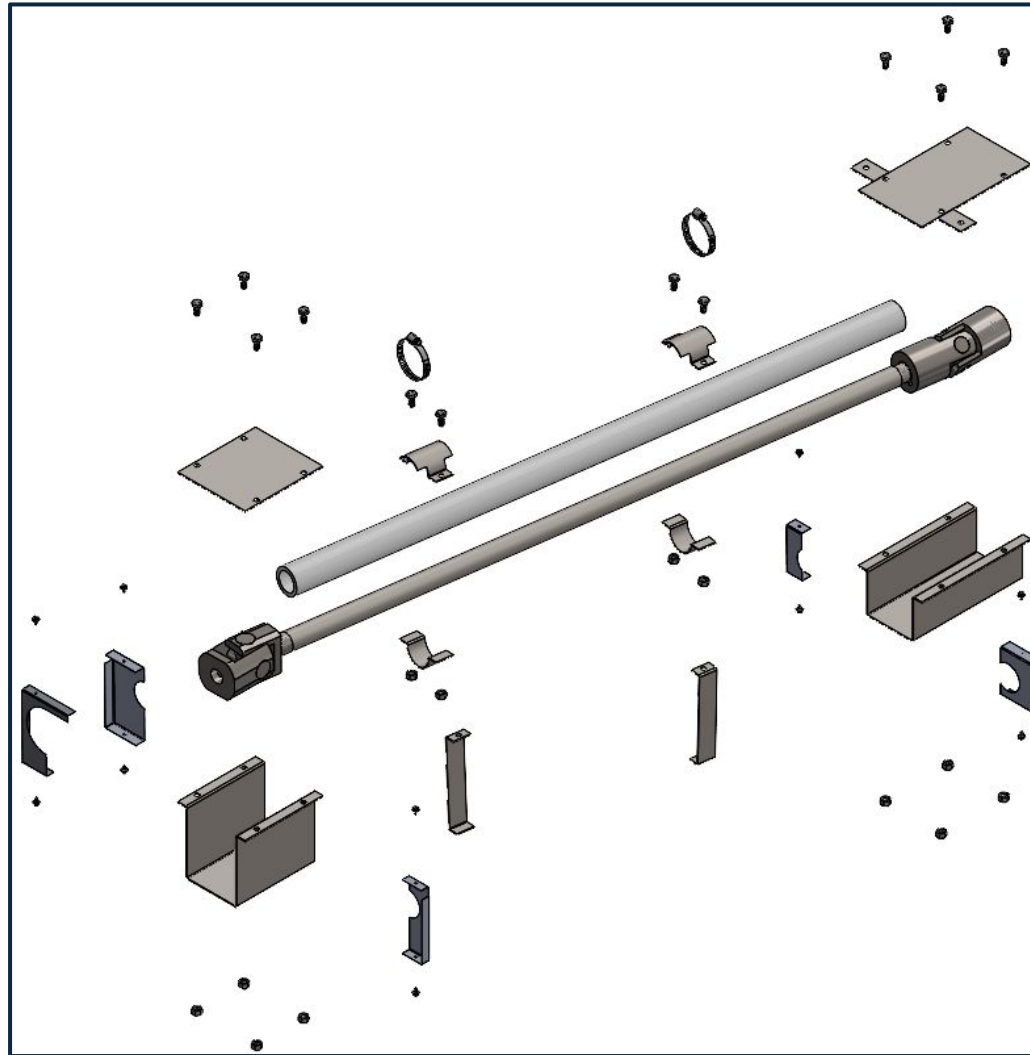
Driveshaft HROE



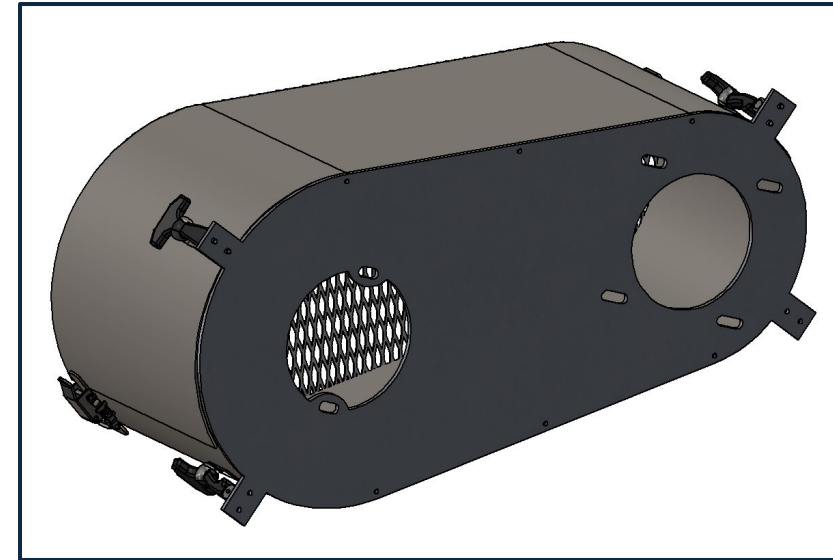
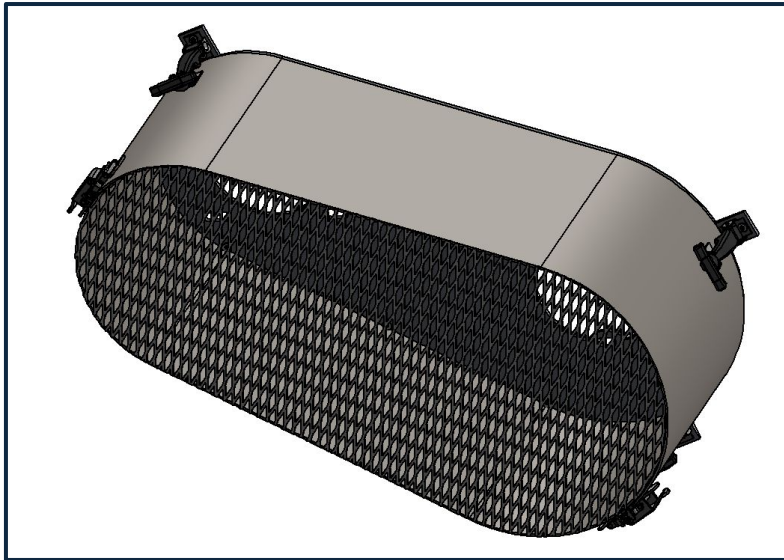
Driveshaft HROE Summary

Weight [lbs]	Cost	Rules
~16.5	Anticipated: \$311	B.9

Driveshaft HROE



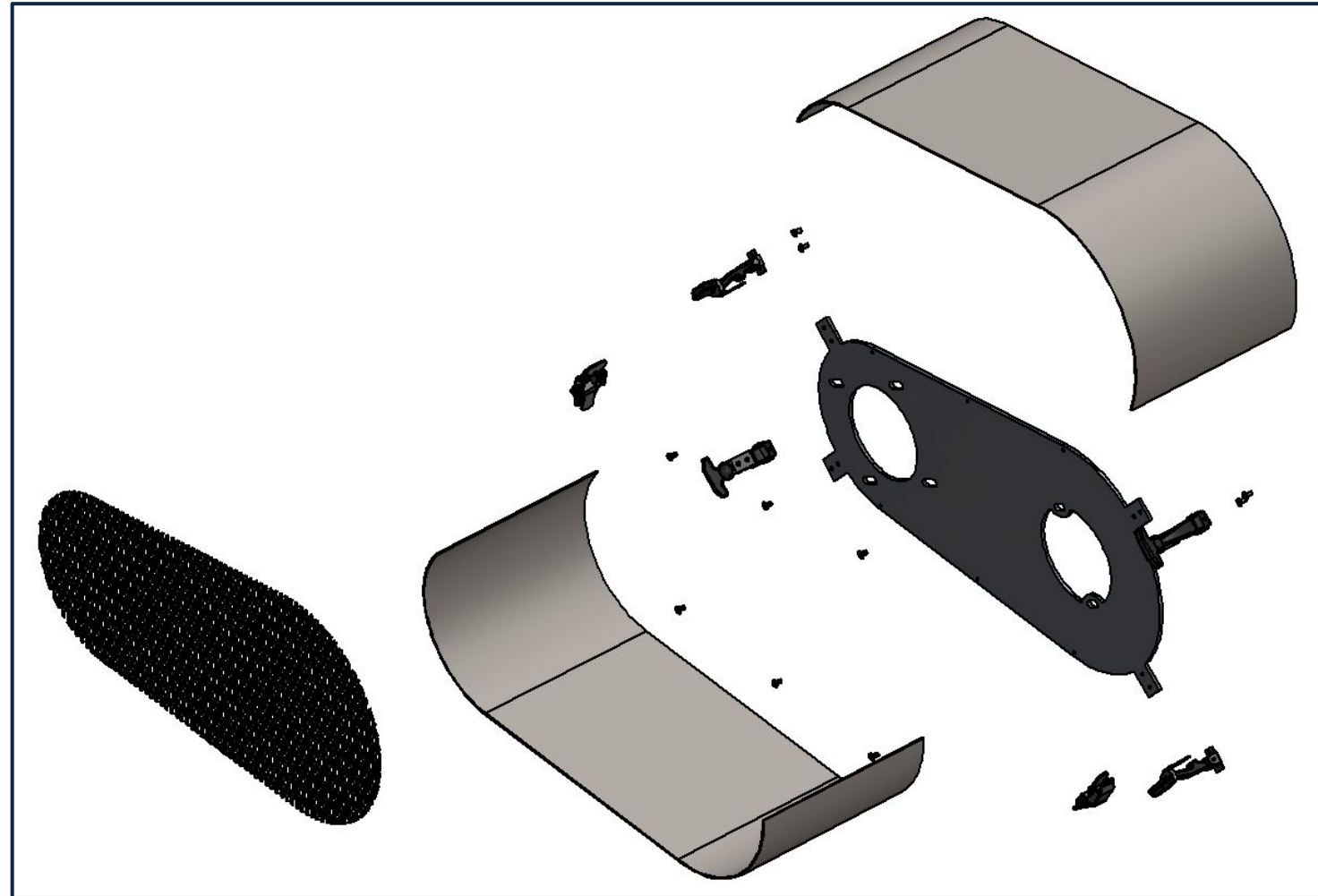
CVT Cover (HROE and PPAE)



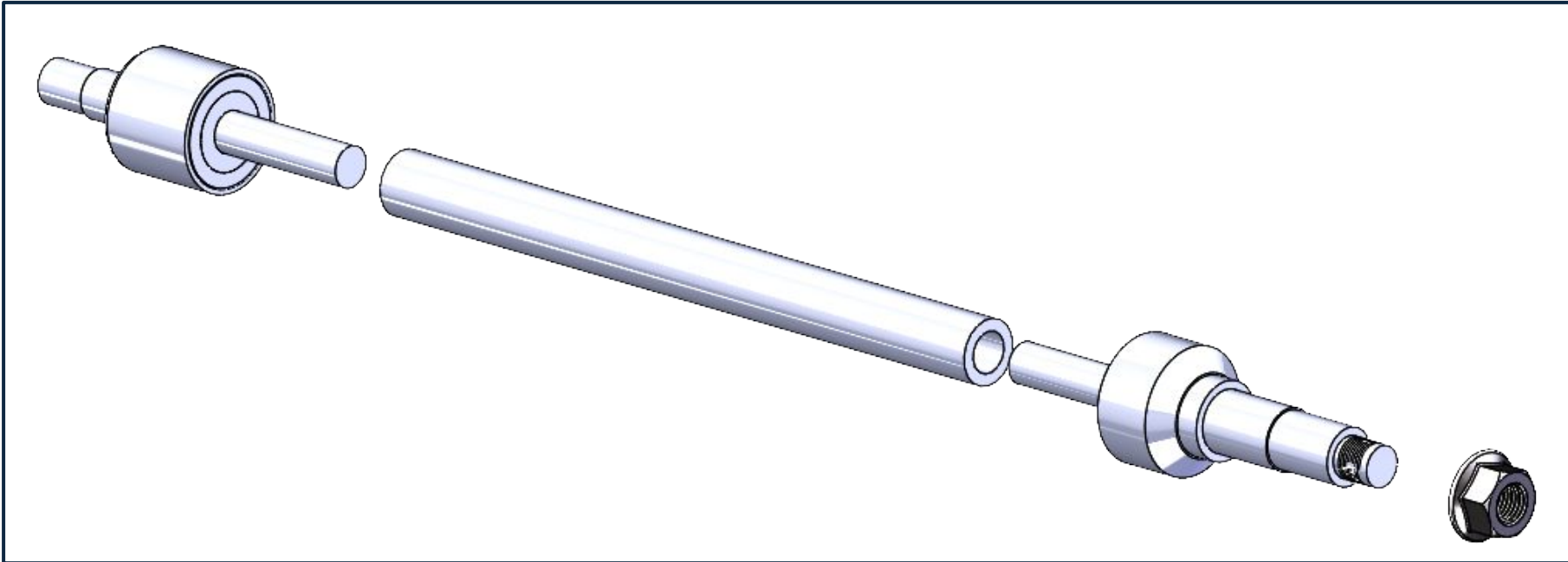
HROE Cover		
Weight [lbs]	Cost	Rules
7.5 (CAD)	\$175	B.9

CVT Cover (HROE and PPAAE)

- Two-piece HROE for removal
- Fastening:
 - HROE Top → HROE Bottom
 - Spring-loaded draw-latches
 - HROE → Backplate
 - Rubber draw-latches



CV Axles



CV Axle Summary		
Weight [lbs]	Cost	Rules
~16.5	Anticipated: \$228	B.9

- Extend OE CVs to proper length, and sleeve prior to welding for concentricity (Prior projects have proven this feasible)

CV Axle Center Shaft Calculation

Failure Condition: The applied torque on the CV Center Shaft is too large for the given size and material.

Justification: The 2024-2024 team used a similar wall thickness and material for their shafts.

Load Diagram:



FOS: 10.67

Equations (If Applicable): See Below

Confidence Level & Explanation:

Very high. A large safety factor combined with a known "worst case scenario" gives high confidence.

CV Axle Center Shaft Calculations

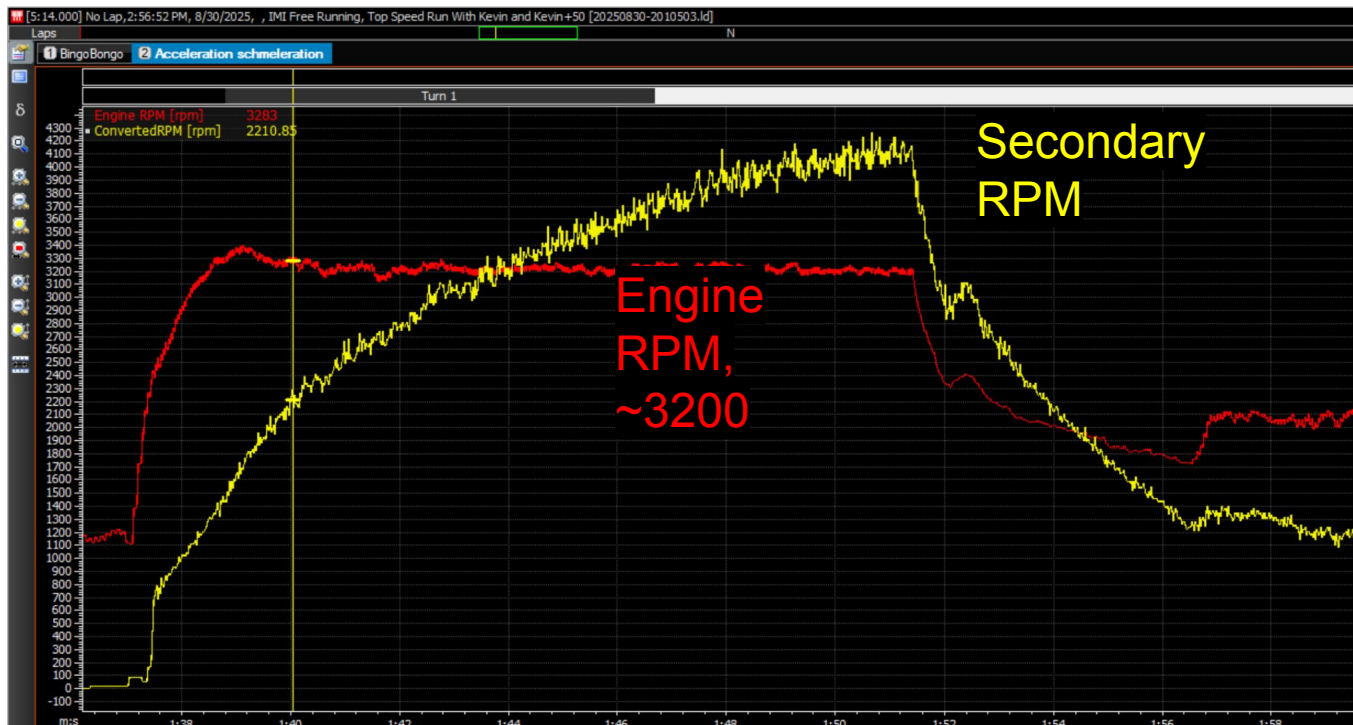
Calculations:

Torque (in lbf)	applied torque	1940.033333
OD (in)	outer diameter	1.25
Thickness (in)	wall thickness	0.25
Shear Strength (ksi)	4130 tube	62
ID (in)	$OD - 2*t$	0.75
c (in)	$OD/2$	0.625
J (in ⁴)	polar moment	0.208621387
Shear Stress (ksi)	$T*c/(J/1000)$	5.812063901
Modulus of Rigidity (GPa)	Found on MatWeb	80
Length (in)	Length of shaft from CAD	14.05
Angle of Twist (degrees)	TL/GJ	0.645176167
FOS	Shear strength/shear stress	10.66746702

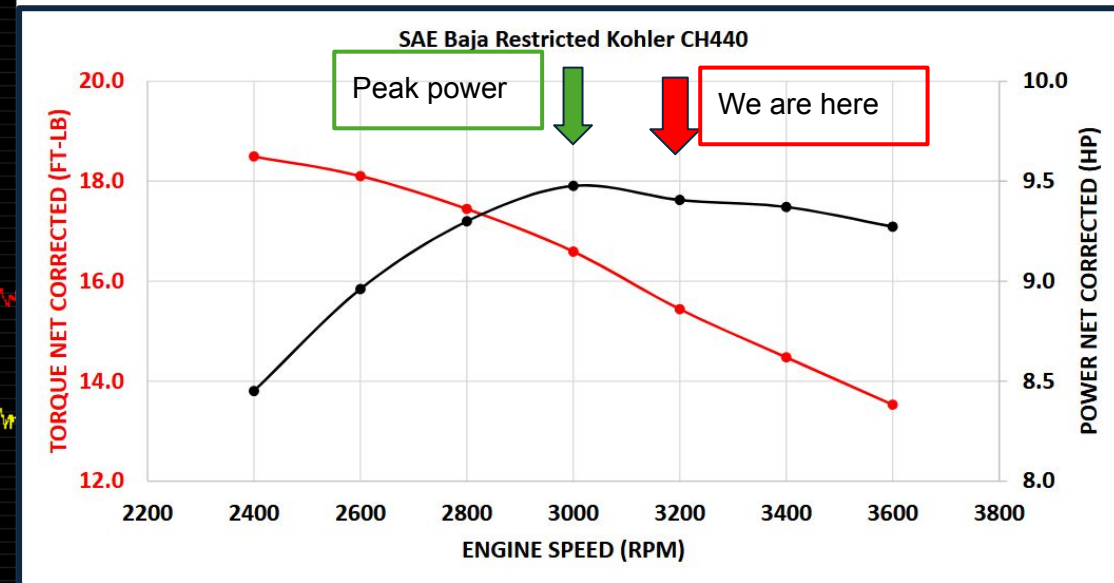
MOTEC Data Analysis

- Issues with previous CVT tuning:
 - Primary Engine RPM was 200+ RPM greater than peak power, losing torque
 - Possible solution: Adjust belt pretension to reduce engagement speed

RPM vs. Time



SAE Baja Restricted Kohler CH440				
Engine Speed (rpm)	3600	3400	3200	3000
Torque Net Corrected (ft-lb)	13.5	14.5	15.4	16.6
Power Net Corrected (HP)	9.3	9.4	9.4	9.5



MOTEC Data Analysis

- Issues with previous CVT tuning:
 - Acceleration was identical between 2 runs with significant weight difference, possibly geared too low with an 11:1 ratio
- Possible solutions:
 - Our new ratio of 9:1 may improve this
 - Increasing our top end shift speed by increasing engagement load



CVT Tuning

MOTEC allows for meaningful CVT Tuning.

Vital CVT tuning goals:

- Ensure Peak power is maintained throughout entire shift

Useful, time permitting goals:

- Ensure backshift rate is appropriate
- Ensure belt is appropriately loaded

Ideally, find out once and for all if CVT tuning is a worthy endeavor.



Chassis

Chassis Goals

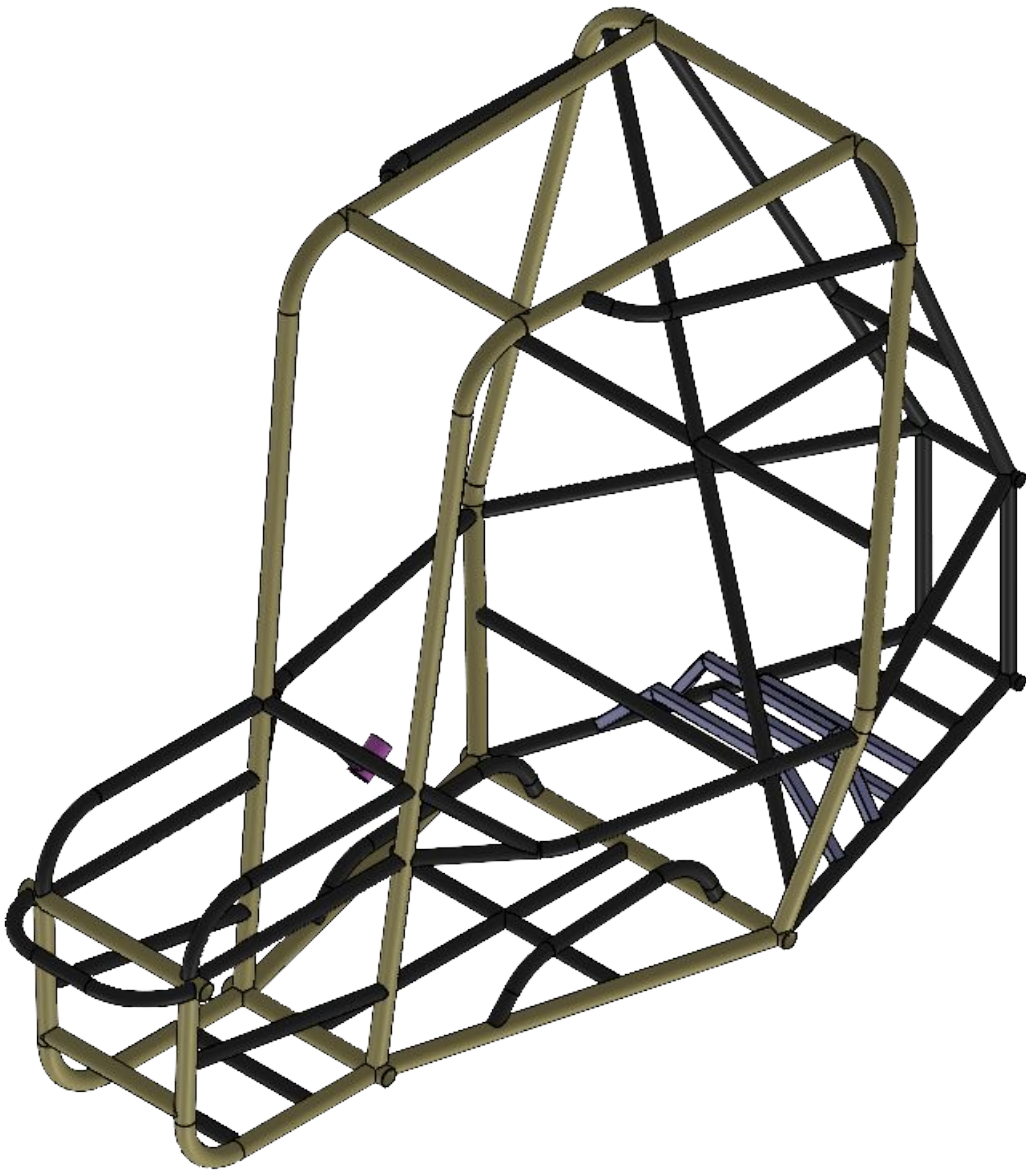
Goal	Reason	Implementation
Reduce weight of chassis by 10% Achieved (raw tube weight only, can't calculate welds/tabs accurately)	Improve Acceleration and Top Speed	Rear Supported Design Improve Packaging
Shorten Length by 10" Missed Actual: 8.9"	Decrease Turn Radius Reduce Weight	Reduce Secondary Mounting Tubes
Flat Floor Achieved	Raise Ground Clearance in Full Bump Easier manufacturing on fixture table	Improve Packaging Contain Drivetrain Components above Floor

Chassis Overall View

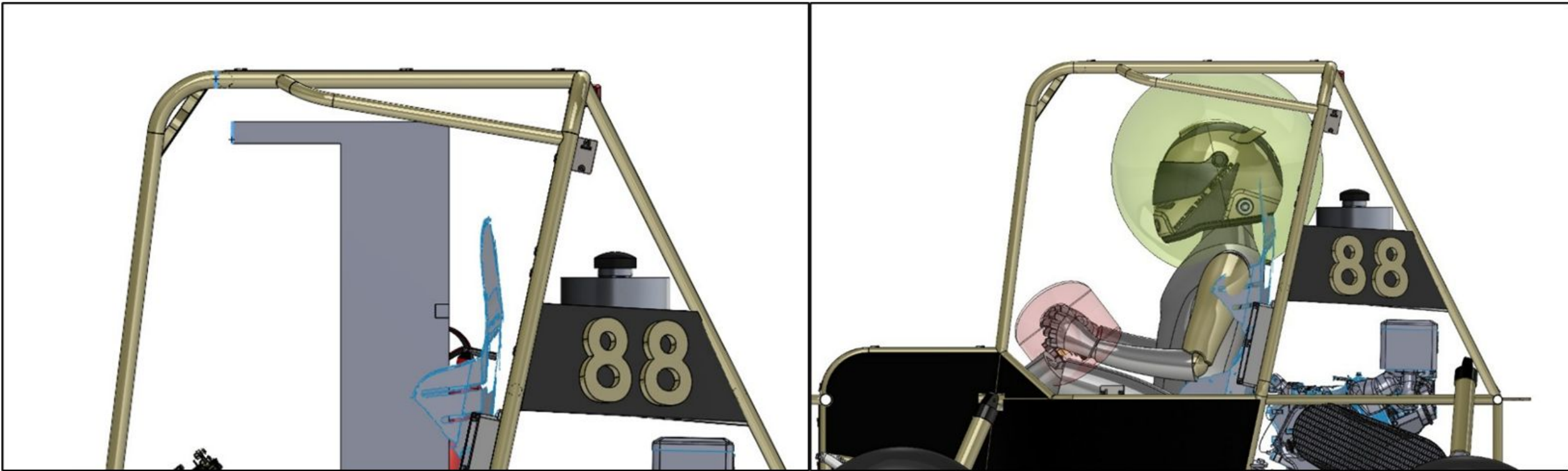
Primary	1.25" x .063
Secondary	1" x .049
Tertiary	.75" SQR x .049
Steering Column Support	1.5" x .049

Total Cost Estimate: \$3000
\$2650 in tubes, \$350 in panels and fasteners

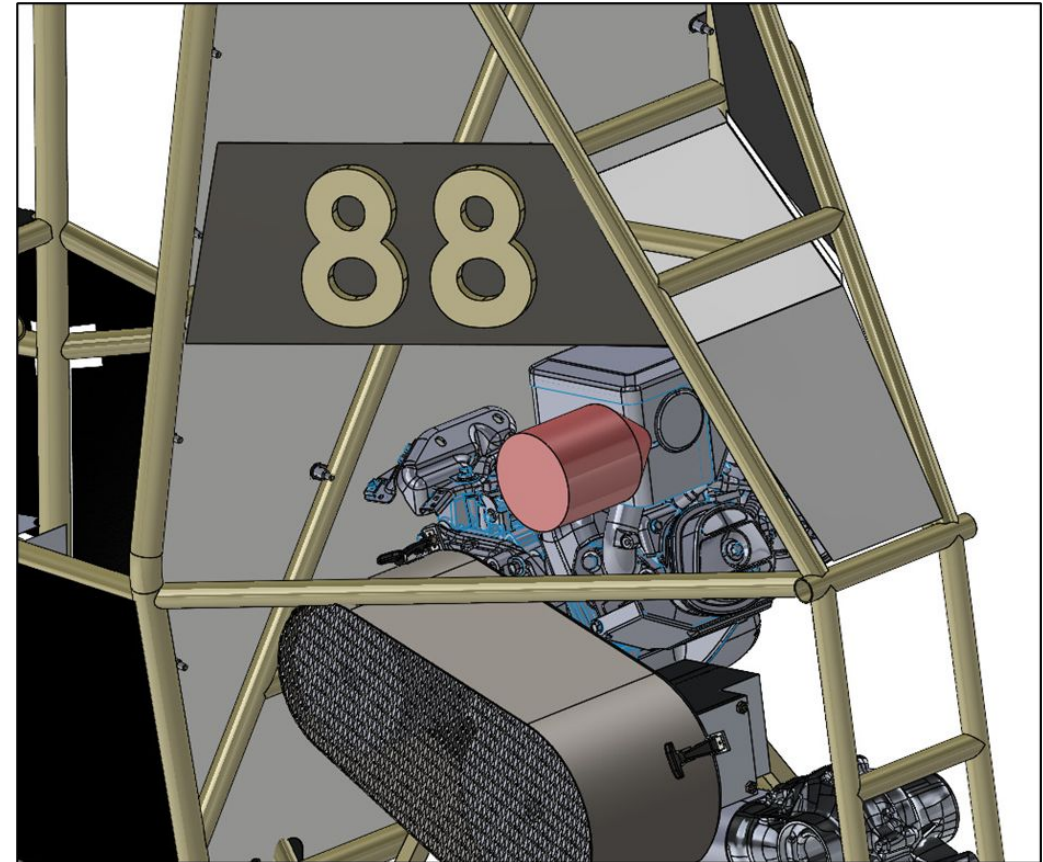
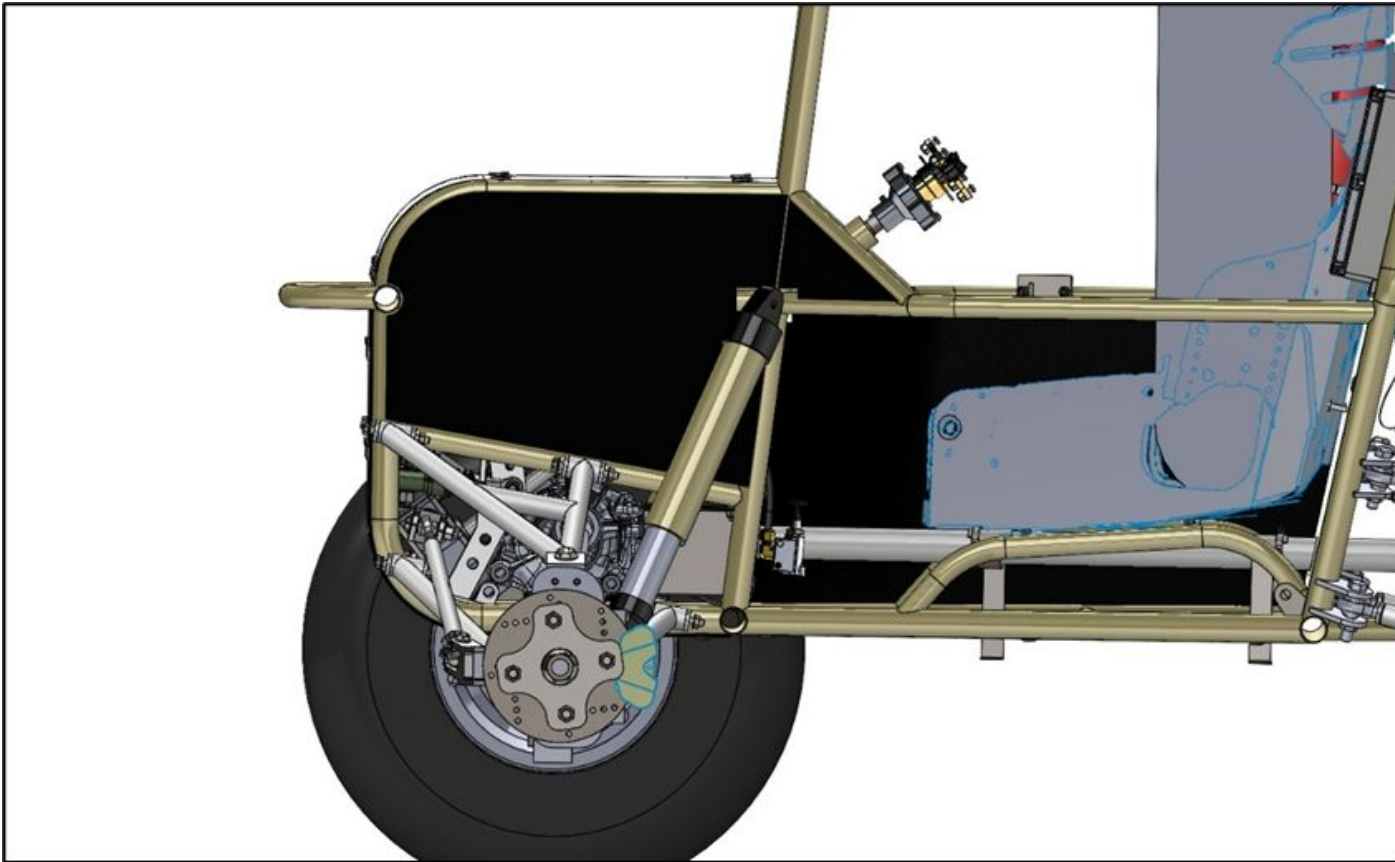
Spec	This Year	Last Year
Weight	66.8 lbs	83.7 lbs
Length	75.9 in	84.8 in
Width	33.6 in	33.3 in
Height	50.6 in	51.6 in



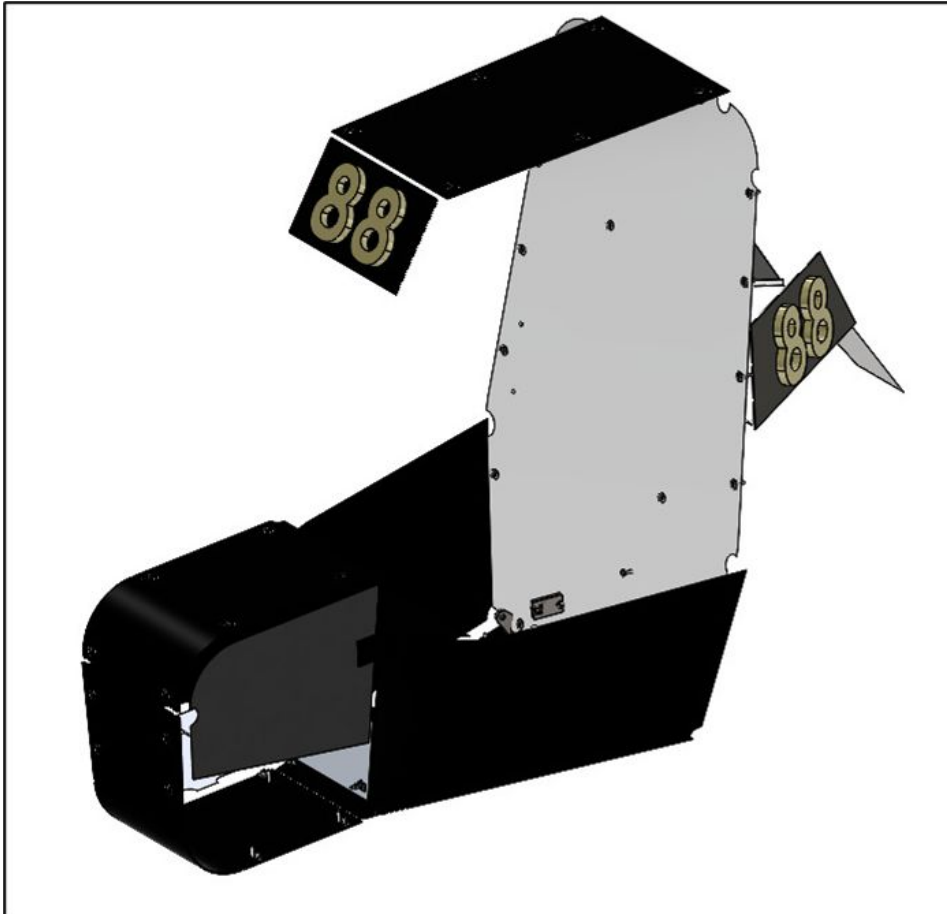
Driver Clearance



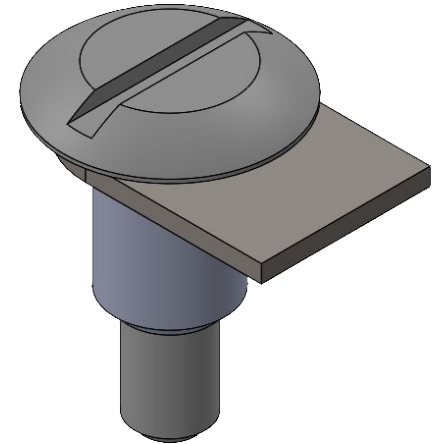
Integration



Panels and Mounting Tabs

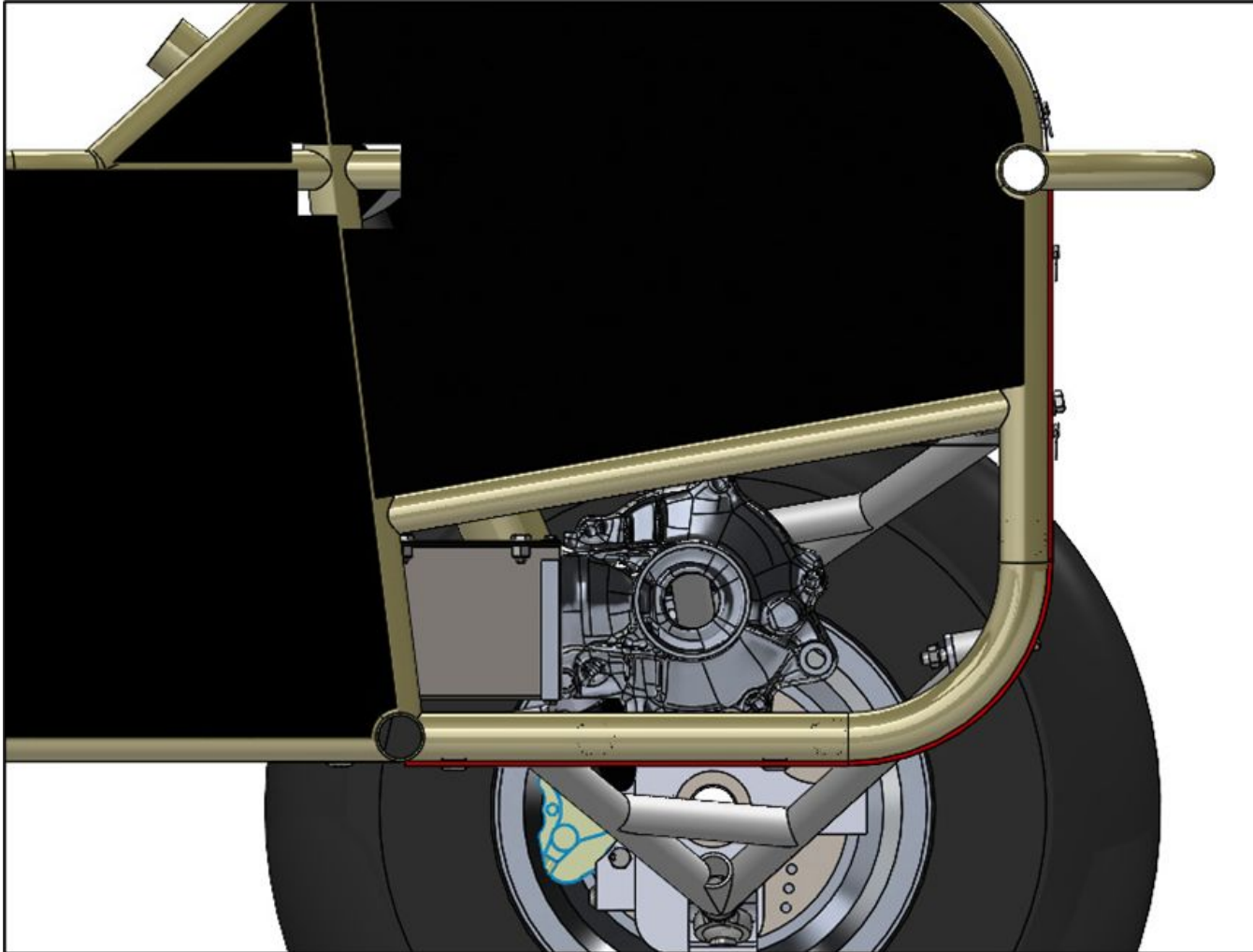


1/4-20 Quick
Release Fasteners



1/4-20 Rivet Nuts

Oregon Considerations

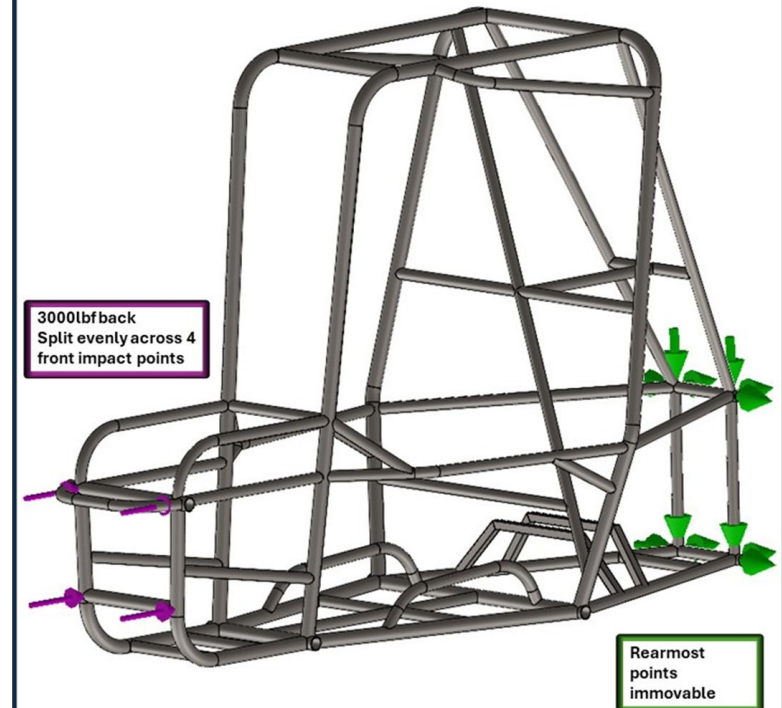


Failure Condition: Front Impact

Material: Alloy Steel 4130; 1.25" x .065 Primary, 1.0" x .049 Secondary

Justification: There is a good chance that the chassis experiences a head on collision during testing or competition, either with a tree, rock, or other solid object

Load Diagram & Constraints:



Thank You!



Paul M. Rady
Mechanical Engineering
UNIVERSITY OF COLORADO **BOULDER**



dc | DESIGN CENTER
COLORADO