MY BACKGROUND

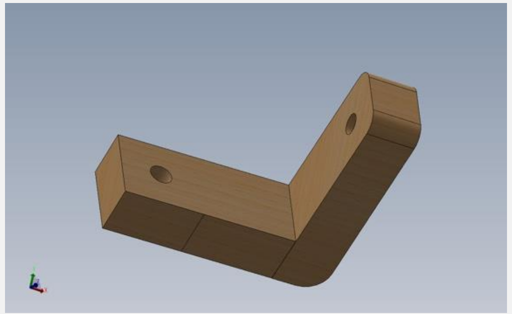
FSAE Suspension Engineer for Formula Buckeyes for 2 years.
Helped design, test, and manufacture mounts, uprights, and hardpoints using Mill and Lathe.
Helped design, test, and manufacture uprights and anti roll bar using SolidWorks and CNC Mill.
Secured 3rd Overall at Michigan FSAE (ICE) after 22 years.

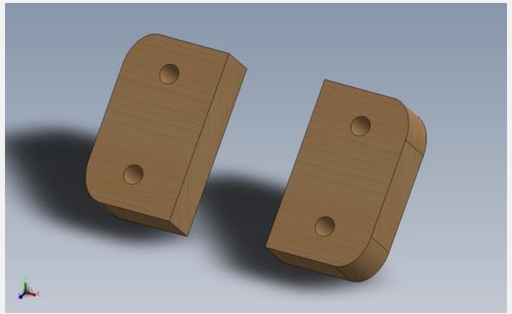
R&D Mechanical Engineering Intern at Allegion (Summer 2024)
Worked on the residential NPD locks division.

Undergrad Research:
AI Driven Topology Optimization for Multi-functional Materials and Intelligent Design Laboratory.
Commercial Polymer Gear Testing at Gear & Power Transmission Laboratory.

FORMULA BUCKEYES FSAE

Some of the hard points (Birch Plywood) I designed and manufactured using Waterjet, Sander, and the Mill





FORMULA BUCKEYES: ANTI-ROLL BAR

Goals:

1. Adjust the oversteer/understeer characteristics of the vehicle.
2. Want a wide enough adjustment range to achieve either at any time according to driver preference.
3. Keep weight to a minimum.

Design Scope:

1. Design anti roll bars (ARBs) around existing pushrod – rocker suspension setup.
2. Continue with 4 corner spring setup.
3. Roll center location and CG height changes not considered.
4. Overall chassis shape was treated as fixed.

Previous Stiffness Target was defined as 200 Nm/deg and the old bar did not meet this target. Did not know if this was accurate, the team decided to start from scratch.

AR BAR: SETTING SPECIFIC TARGETS

1. Tire is sensitive to load.
   - Lower COF at higher normal load.

2. ARBs function by changing the Lateral Load Transfer Distribution (LLTD)
   - Stiffer front roll rate = more load transfer on the front axle = more understeer.

3. Want the LLTD to be centered around CG location and have adjustment to be well in front or behind the CG.

Overall Load Transfer Equation: $\Delta W = \frac{W \cdot A_y \cdot H_{cg}}{TrackWidth}$

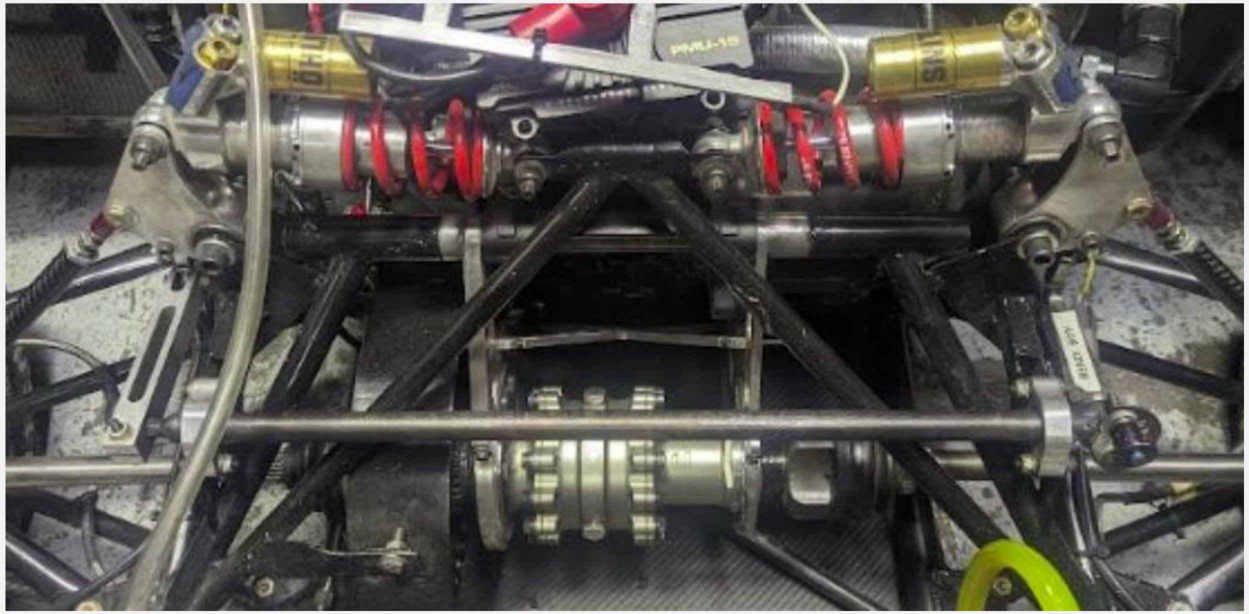
4. Changing roll stiffness of one axle relative to the other changes LLTD.

   - Stiffer front roll stiffness = more understeer.
   - CG location ~ 0.525 (52.5% rear)

Roll Moment Load Transfer Equation: $\Delta W = Total\ LT \cdot \frac{K_{\theta F} + \left((l-x) \cdot W \cdot \frac{h}{l}\right)}{K_{\theta F} + K_{\theta R} - W \cdot h}$

Front ARB Stiff Setting	50	Nm/deg
Rear Anti-Roll Bar Stiff Setting:	30	Nm/deg
Front Roll Rate with Anti-Roll Bar:	503.59	Nm/deg
Rear Roll Rate with Anti-Roll Bar:	521.91	Nm/deg
Front Roll LT w Bar	325.20	N
Rear Roll LT w bar	351.25	N
LLTD no Bar	0.543	Rear
LLTD with Rear Bar	0.566	Rear
LLTD with Front Bar	0.505	Rear

AR BAR: HITTING TARGETS



Left: T-Bar Setup in the front to achieve longer moment arm to twist section.
Right: Traditional twist bar setup in the rear.

Packaging constraints led to different solutions for the front and rear bars in order to hit stiffness targets and keep and acceptable F.O.S.

AR BAR: HITTING THOSE TARGETS

Determining required bar stiffness in the context of the motion ratio and moment arm length.

STEPS:

1. Motion Ratio – Input from Rocker Kinematics
2. Wheel Movement = sin(1°) * track width
3. Bar Movement = Wheel Movement * Motion Ratio
4. Bar Rotation = $tan^{-1}(\frac{bar\ movement}{Moment\ Arm\ Length})$
5. Bar Stiffness = $\frac{Target\ Stiffness}{Bar\ Rotation\ per\ degree\ of\ roll}$

Front Bar (t bar)		
Motion Ratio	0.58	
Wheel Movement per deg of roll	20.94	mm
Bar Movement	12.15	mm
Moment Arm Length	125.00	mm
Bar Rotation	5.57	deg
Bar Stiffness Required	8.98	Nm/deg

AR BAR: HITTING THOSE TARGETS

Focusing on front ARB since it was a bigger challenge (T-bar):

First choice of material was steel – easy to work with and acceptable fatigue properties.

STEPS:

1. $J = Area\ MOI = \frac{\pi}{32}(OD^4 - ID^4)$

2. G = Shear Modulus of Elasticity

3. $K\left(\frac{\text{Nm}}{\text{deg}}\right) = \frac{G \cdot J}{L} \cdot 1°\ rotation$

4. $Max\ Stress = \tau_{max} = \frac{(\text{Stiffness} \cdot \theta_{\text{max}}) \cdot r}{J}$

S_{ty} = *Tensile Yield Strength*
S_{sy} = *Shear Yield Strength = 0.5 * S_{ty} (Shirley's ME Design Approximation)*

Tube OD	0.008255	m
Tube ID	0	m
J	4.56E-10	m^4
G	8.00E+10	Pa
Length	0.07	m
Stiffness	9.11	Nm/deg
Tau-max	4.58E+08	Pa
Sty(4130)	4.35E+08	Pa
Ssy(4130)	2.18E+08	Pa
F.O.S	0.47	

FOS – Not Acceptable

AR BAR: HITTING THOSE TARGETS

Focusing on front ARB since it was a bigger challenge (T-bar):

Titanium has lower shear modulus (less stiff) but higher shear yield strength.

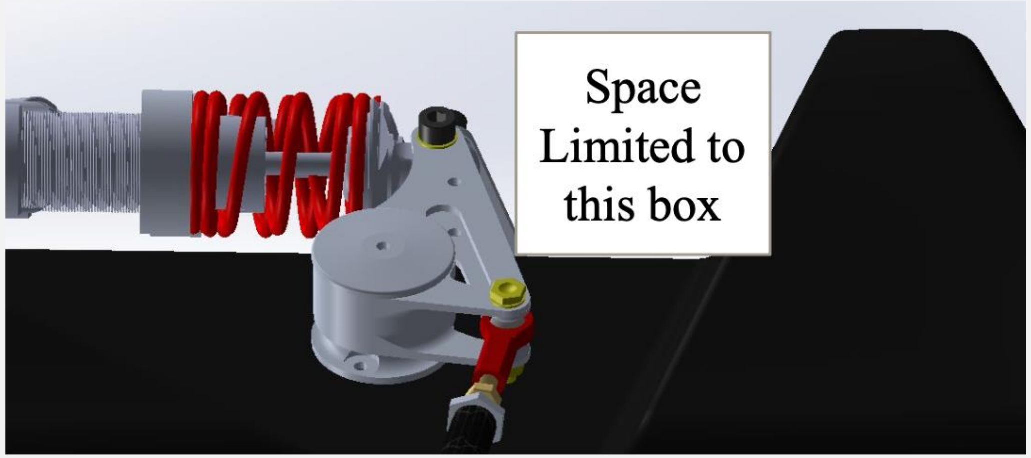
Thicker bar compared to steel to meet targets.

Lower F.O.S than we would typically accept but ARB is not a critical component in terms of damages to the car in the event of failure.)

Tube OD	9.53E-03	m
Tube ID	0.00E+00	m
J	8.08E-10	m^4
G	4.50E+10	Pa
Length	0.07	m
Stiffness	9.09	Nm/deg
Tau-max	2.97E+08	Pa
Syt (6Al-4V)	1.10E+09	Pa
Ssy (6Al-4V)	5.50E+08	Pa
F.O.S	1.85	

AR BAR: HITTING THOSE TARGETS

- Using a traditional twist bar set up in the from leads to a small moment arm.
- This increases the deflection of the twist bar, lower F.O.S.
- Allows for longer length of twist bar, however this is overpowered by the reduction in moment arm length.
- Wanted to keep height below the top of the chassis.



Front Bar (twist bar)		
Motion Ratio	0.58	
Wheel Movement per deg of roll	20.94	mm
Bar Movement	12.15	mm
Moment Arm Length	25.00	mm
Bar Rotation	25.91	deg
Bar Stiffness Required	1.93	Nm/deg
Tube OD	9.02E-03	m
Tube ID	0.00E+00	m
J	6.49E-10	m^4
G	4.50E+10	Pa
Length	0.25	m
Stiffness	2.04	Nm/deg
Tau-max	3.67E+08	Pa
Syt (6Al-4V)	1.10E+09	Pa
Ssy (6Al-4V)	5.50E+08	Pa
F.O.S	1.50	

AR BAR: HITTING THOSE TARGETS

- Adjustment blade rotates 90° to stiffen or soften the bar.
- Used Fixed end Cantilever Beam Equations

$K = \frac{3EI}{LL^3}$ and $I = \frac{bh^3}{12}$

$$\sigma_{max} = \frac{\left(\frac{OD}{2}\right) \cdot L \cdot F}{I}$$

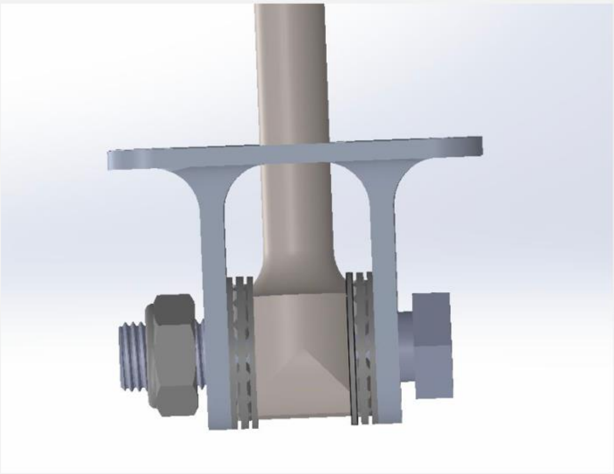
$$\text{Deflection ratio} = \frac{\text{Deflection of Twist bar}}{\text{Deflection of Adjustment bar}}$$

Adjustment Blade		
Length	0.125	m
OD	0.0127	m
Flat Thickness	0.004699	m
Force at End	36.36	N
E	1.14E+11	Pa
I stiff	8.02E-10	m^4
I soft	1.10E-10	m^4
K stiff	1.40E+05	N/m
K soft	1.92E+04	N/m
K ratio	7.30	
Twist Bar Deflection	2.18E-03	m
Blade Deflection Stiff	2.59E-04	m
Blade Deflection Soft	1.89E-03	m
Deflection Ratio Stiff	8.43	
Deflection Ratio Soft	1.15	
Sigma max stiff	7.20E+07	Pa
Sigma max soft	5.26E+08	Pa
Syt (6Al-4V)	1.10E+09	Pa
F.O.S stiff	15.29	
F.O.S soft	2.09	

AR BAR: CHASSIS INTEGRATION

- If chassis torsional stiffness is not high enough to transmit load between axles, ARBs will be ineffective.
- Set 70% effectiveness as the goal for chassis stiffness.
- Front roll stiffness, chassis stiffness, and rear roll stiffness are 3 springs in series.
- Applied a torque to the front suspension and found roll distribution for varied chassis stiffness.
- Target was set at 1350 Nm/deg.

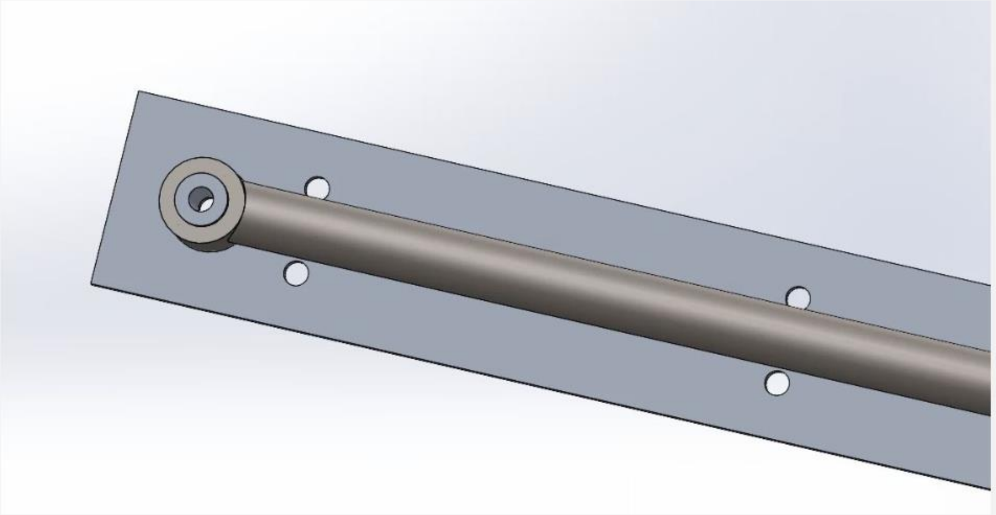
- Bolts are in shear, decided on 4 bolts on the mount to spread out the load.
- Cylindrical hardpoints in the chassis layup for each bolt.
- Provided maximum torque from bar on the chassis.



AR BAR: MANUFACTURING

- This was my first time working with CNC lathe, so I mostly shadowed the Suspension Lead to learn how to use the machine for making these parts.

- For the Rear ARB which was welded, I worked on designing and water jetting the locating plate.



AR BAR: TESTING

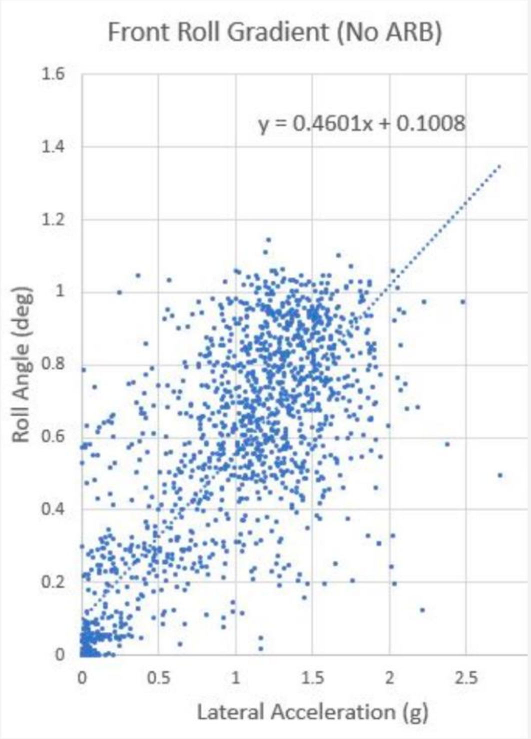
- Changes in roll distribution for different ARB positions show that the roll stiffnesses are changing.

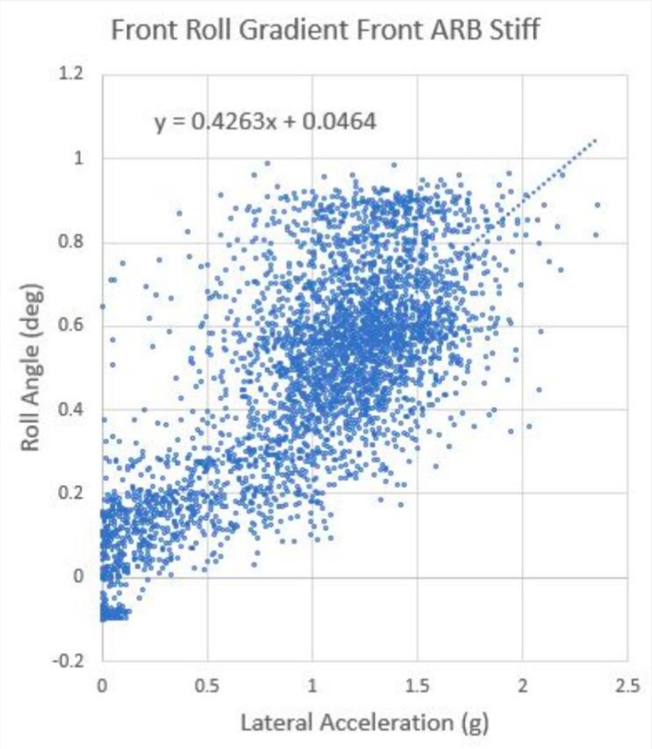
- Measured with linear potentiometers on each spring and damper unit.

- Theoretical Values:

Roll Distribution No Bar:	0.480	rear
Roll Distribution With Rear Bar:	0.467	rear
Roll Distribution with front Bar	0.506	rear

- Roll Gradient used to find total Roll Stiffness for various ARB settings.





$$Roll\ Stiffness\left(\frac{Nm}{deg}\right) = \frac{Mass\ (kg) \cdot moment\ arm\ (m) \cdot weightdistribution}{Roll\ Gradient\left(\frac{deg}{g}\right)/9.81}$$

Theoretical No ARB (excluding tire) = 702
Measured: 738
Theoretical with front ARB = 752
Measured = 796.85