Nathan Chan

Mechanical Engineering Portfolio

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About Me

I am a mechanical engineering graduate who is passionate about innovation and pushing the boundaries of technology. I enjoy all aspect of mechanical engineering, but I am particularly fond of dynamical systems, control systems and testing. While studying, I spend most of my free time as a member of the Concordia Formula Racing student team where we design and build race cars to compete in international collegiate events. This experience combined with my internship experiences have increased my love and appreciation for engineering and I hope to apply what I learnt in my career.

Education

Université Concordia University

B.Eng Mechanical Engineering

Internships

Pratt & Whitney — 4 months Internship

ABB — 4 months Internship

Concordia Formula Racing

- Member since 2021
- Powertrain Lead 2022-2023
- Vehicle Dynamics Co-lead 2023-2024

Engineering Support Intern

At ABB I worked on industrial electrical distribution cabinets, acting as the liaison between the team's mechanical engineer and the production staff. My responsibilities included ensuring all of production's questions are answered, creating and publishing drawing packages for production and client, and ensuring all production documents are up to date with revisions.

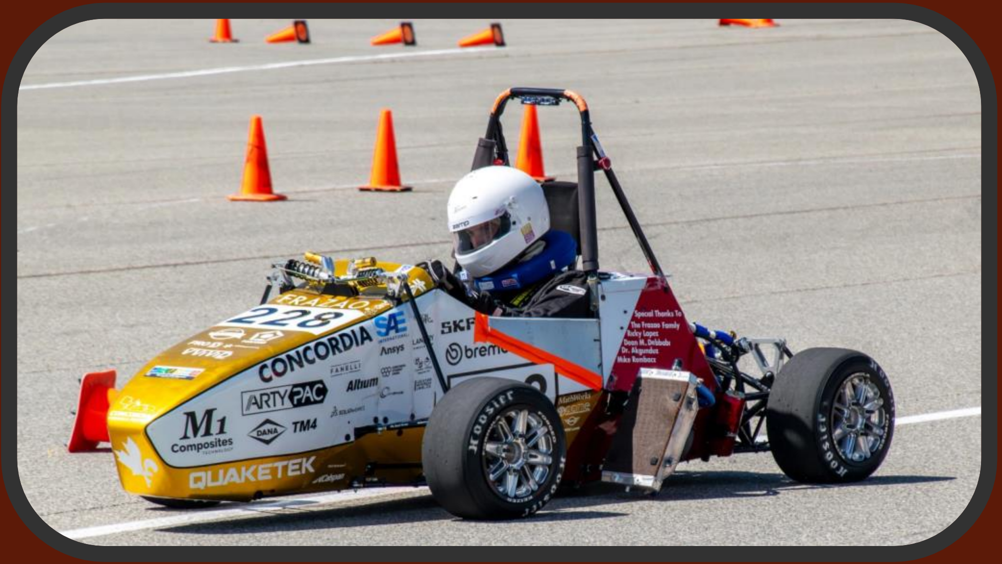
Hybrid System R&D Intern

I was part of the Advanced design and R&D team working on R&D of hybrid system test benches using Pratt & Whitney Canada’s turboshaft engines. My responsibilities included creating Simulink models of conceptual designs to investigate various configurations, integrating custom models with the pre-existing turboshaft engine models, and have meetings with shareholders to understand their needs and present our concepts.

Concordia Formula Racing

I joined Concordia Formula Racing because I needed some way to spend my free time and doing practical engineering seems to be the perfect way to do so. Throughout the 3 years I’ve been on the team I have learnt a lot of new skills, made a lot of new friends and had a blast competing against other likeminded students. I was also fortunate to be part of the team that built CFR’s first running electric car as well as their first competing electric car.

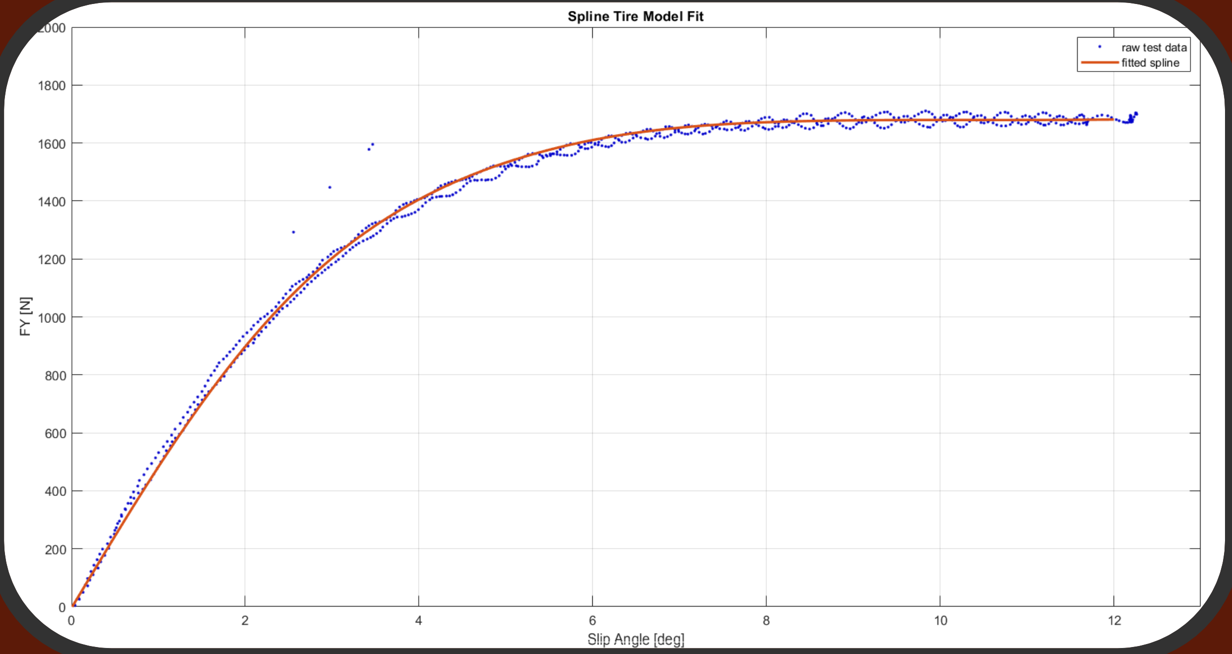
The following section highlights what I did and learnt during my time on the team.

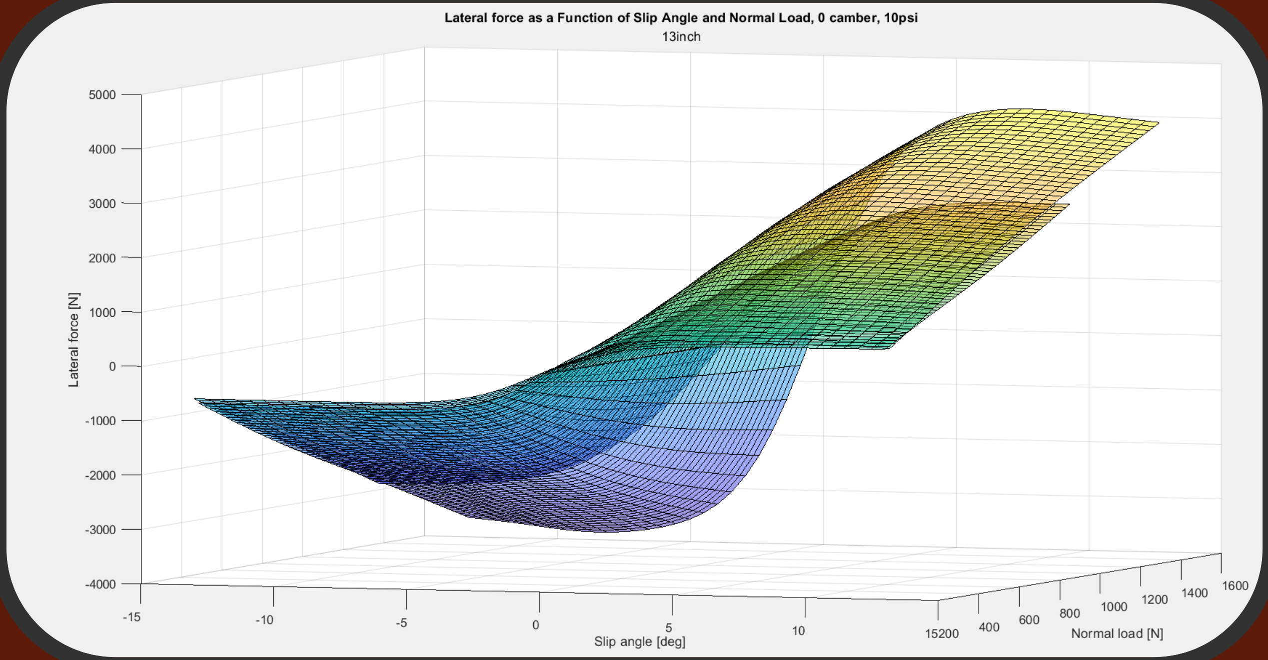


Parameter	Variable	Units	Value Front	Value Rear	Formula	Desciprtion
Vehicle Mass Properties						
Track	t_f, t_r	mm	**1219.2**	**1168.4**		measured
		in	48.00	46.00		
Wheel base	l	mm	**1575.0**		-	measured
		in	62.0			
Mass (no driver)	M_c	kg	**218**		-	measured
		lbf	480.7			
Mass (with 68kg driver)	M	kg	286		M_c + 68	
		lbf	630.6			
Center of gravity height	cgh	mm	**300.0**		-	measured
		in	11.8			
Unsprung mass (per corner)	M_{usf}, M_{usr}	kg	**10.0**	**10.0**	-	measured
		lbf	22.1	22.1		
Sprung mass (with driver)	M_s	kg	246.0		M - 2*M_{usf} - 2*M_{usr}	
		lbf	542.4			
Static weight distribution (front)		unitless	**0.480**			
Static weight distribution (left)		unitless	**0.500**			
Static weight distribution	fr	rr	0.24	0.26	-	
	fl	rl	0.24	0.26		
Static weight on tires	wfr, wrr	N	673	729	M*fl, M*fr	front right, rear right
		lbf	151.4	164.0		
	wfl,wrl	N	673	729	M*rl, M*rr	front left, rear left
		lbf	151.4	164.0		
cross weight percentage	cwp	unitless	0.5		-	
Unsprung mass CG height	h_{uf}, h_{ur}	mm	203.3	203.3		
		in	**8**	**8**		
Spruing mass CG height	hs	mm	315.7			
		in	12.43			
Vehicle Mass Properties						
Roll						
Roll stiffness	K_F, K_R	Nm/rad	13710.5	14794.5		
		lbf-in/rad	121348	130942.22		
Roll stiffness distribution	-	unitless	0.48	0.52		
Roll center height	Z_f, Z_r	mm	**10.09**	**13.99**		
		in	0.40	0.55		
Roll axis height at CG	Zcg	mm	11.962			
		in	0.47			
Unsprung mass load transfer per g		lbf	7.35	7.67		
Load transfer per g, rigid chassis	ΔF_{f0}, ΔF_{r0}	N	337.85	368.07		
		lbf	75.95	82.75		
LLTD, rigid chassis	-	unitless	0.479	0.521		
Chassis torsional stiffness	K_c	Nm/rad	**97402.8**			1699.999561
Unsprung mass load transfer per g,		lbf	7.35	7.67		
Load transfer per g, with chassis stiffness	ΔF_{fc}, ΔF_{rc}	N	337.49	368.43		
		lbf	75.88	82.83		
LLTD, with chassis stiffness	-	unitless	0.478	0.522		
Polar Moment Of Area	J	m^4	1.12487E-09	1.12487E-09		
Anti Roll bar torsional stiffness	K_a	Nm/rad	**200.0**	**0.0**		
		lbf-in/rad	1770.2	0.0		
ARB motion ratio	MR_{arb}	unitless	**2.000**	**2.000**		Wheel displacement/ARB attach point displacement
ARB lever arm	L_{arb}	mm	**100.0**	**70.0**		
		in	3.9	2.8		
ARB roll stiffness	$K_{\phi ARB}$	Nm/rad	7432.2	0.0		**ONLY VALID FOR TYPICAL U-TYPE ARB**
		lbf-in/rad	65782.8	0.0		
Total roll stiffness	K_F, K_R	Nm/rad	21142.7	14794.5		ARB+springs
		lbf-in/rad	187130.7	130942.2		
Total roll gradient	K_ϕ	deg/g	1.169			
Unsprung mass load transfer per g,		lbf	7.35	7.67		
Load transfer per g, rigid chassis, with ARB	ΔF_{f0}, ΔF_{r0}	N	403.80	302.12		
		lbf	90.78	67.92		
with chassis stiffness, with ARB		lbf	7.35	7.67		
Load transfer per g, With chassis stiffness, with ARB	ΔF_{fc}, ΔF_{rc}	N	397.97	307.95		
		lbf	89.47	69.23		
LLTD, with chassis stiffness, with ARB	-	unitless	0.564	0.436		
Roll						

I created a spreadsheet that can quickly calculate steady state suspension behavior. The calculations are completely derived from first principles and serve as a great way to study the trade-offs between different suspension parameters.

Although this spreadsheet does not take into account the dynamics of the suspension, the steady state behaviour is often very indicative of the dynamic behaviour, and our team uses this as the high-level suspension design tool that informs the rest of the design.





I developed Matlab scripts for the purpose of modeling tires performance. This script parses tire testing data from the Tire Testing Consortium (TTC) and fit them to a cubic spline.

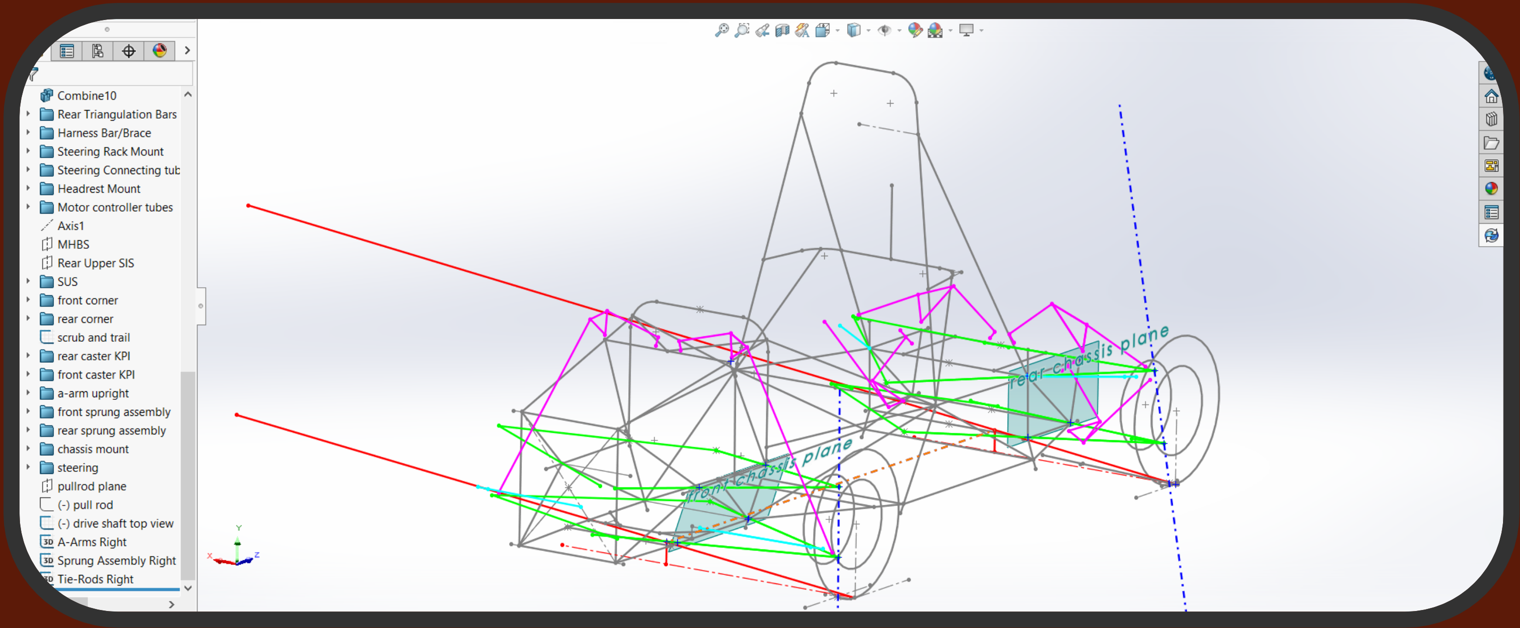
This tool was used in the selection of tires, which led to our choice of switching from 13" wheels to 10" wheels for the first time in the team's history.

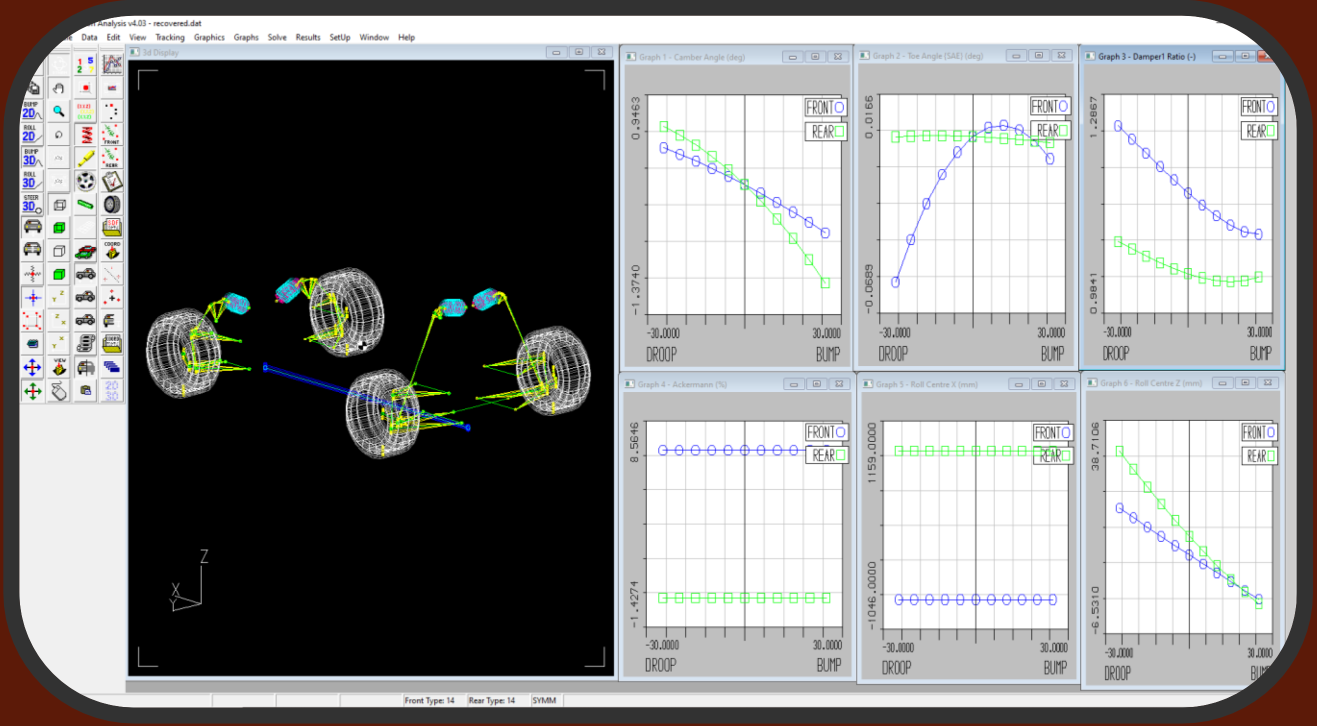
The tire models are also used in in-house vehicle dynamics simulations.

Details of the script can be found on my Github.

I developed a Solidworks tool that can be used to parametrically design the suspension simply by inputting parameters such as track width, roll center height, static camber angle etc.

This tool helped us quickly iterate through our suspension design, used in conjunction with Lotus Shark to obtain the desired suspension kinematics.



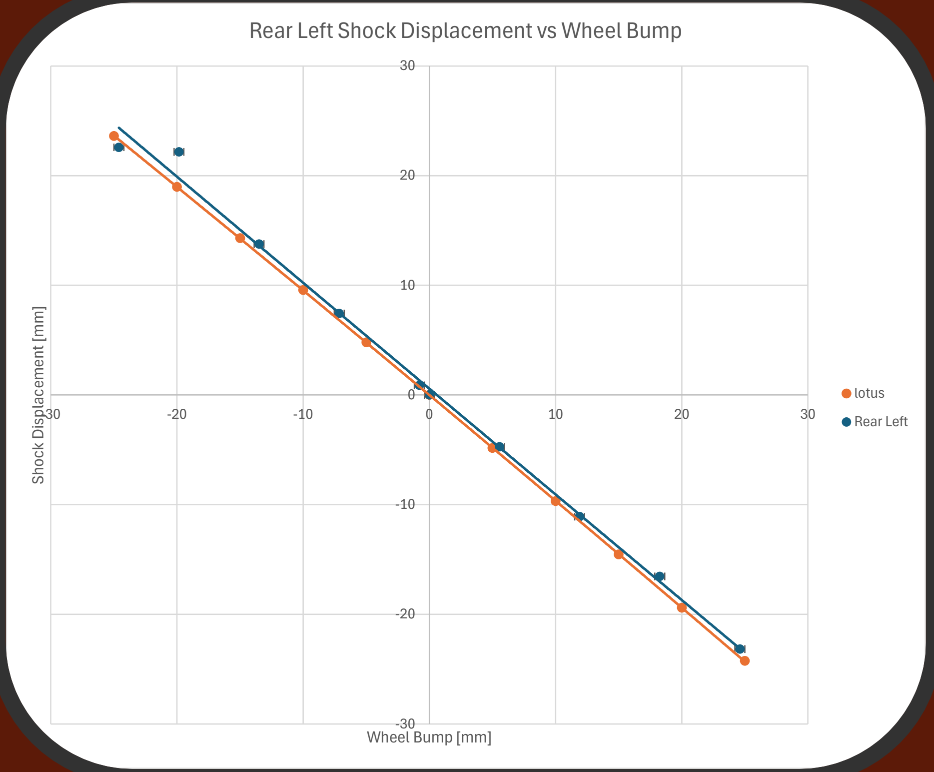


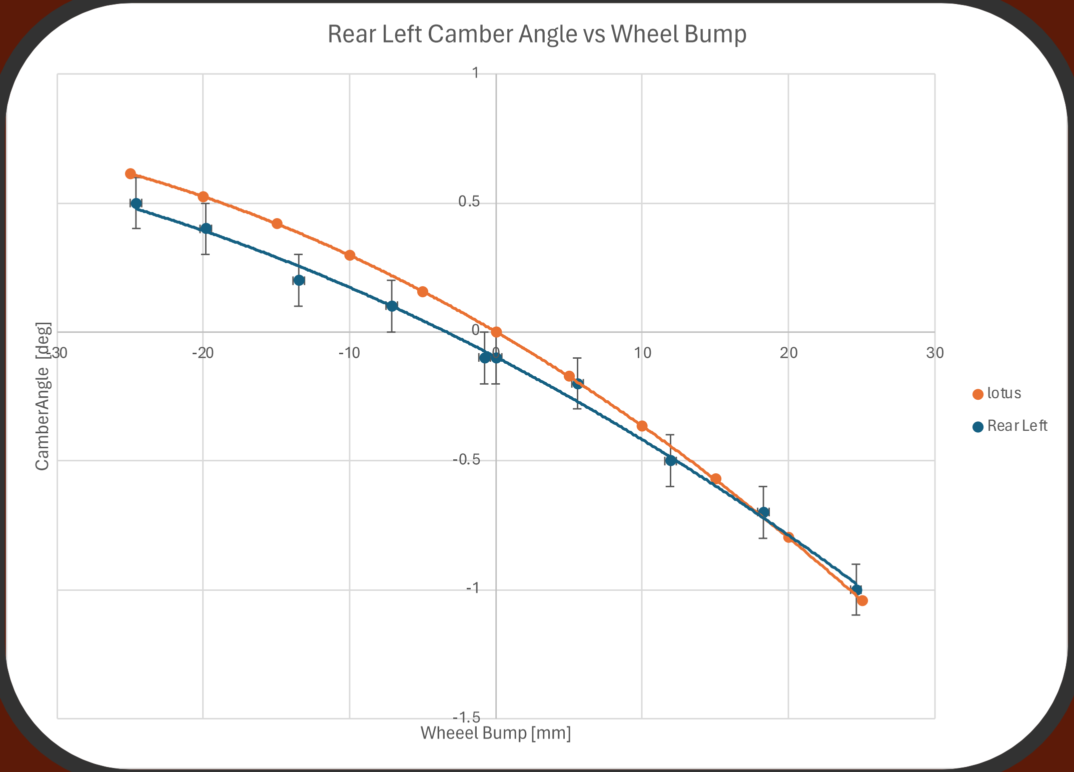
A second version of this tool was developed which worked the opposite way. It allowed the user to move suspension points around in space and have suspension parameters as an output.

This was done in response to the difficulty in using the first version where sometimes changing a parameter might create unpredictable outputs.

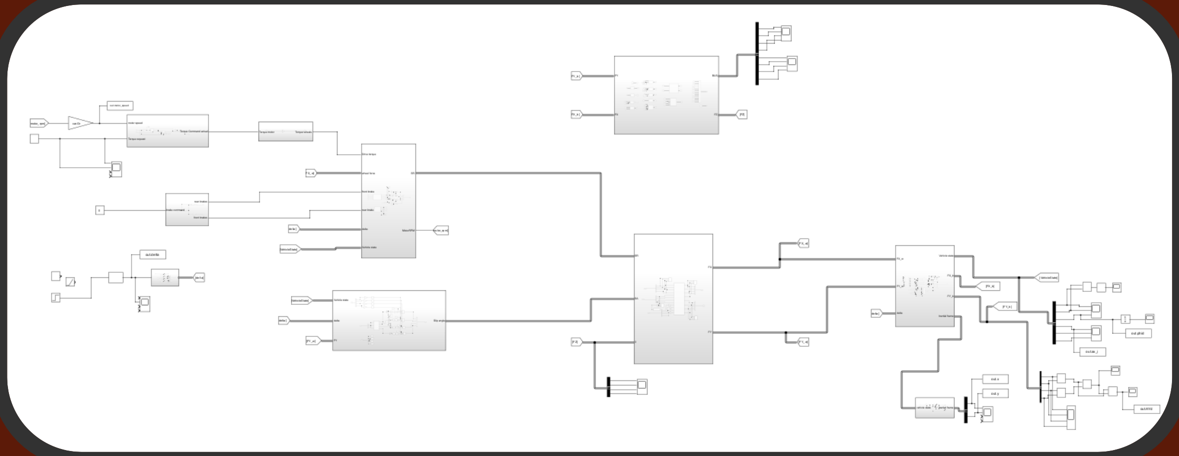
We measured the suspension kinematics physically to ensure the manufactured parts reflect our designs and to validate the kinematic models that was used during the design process. We measured the camber angle, castor angle change and spring travel as a function of the wheel travel.

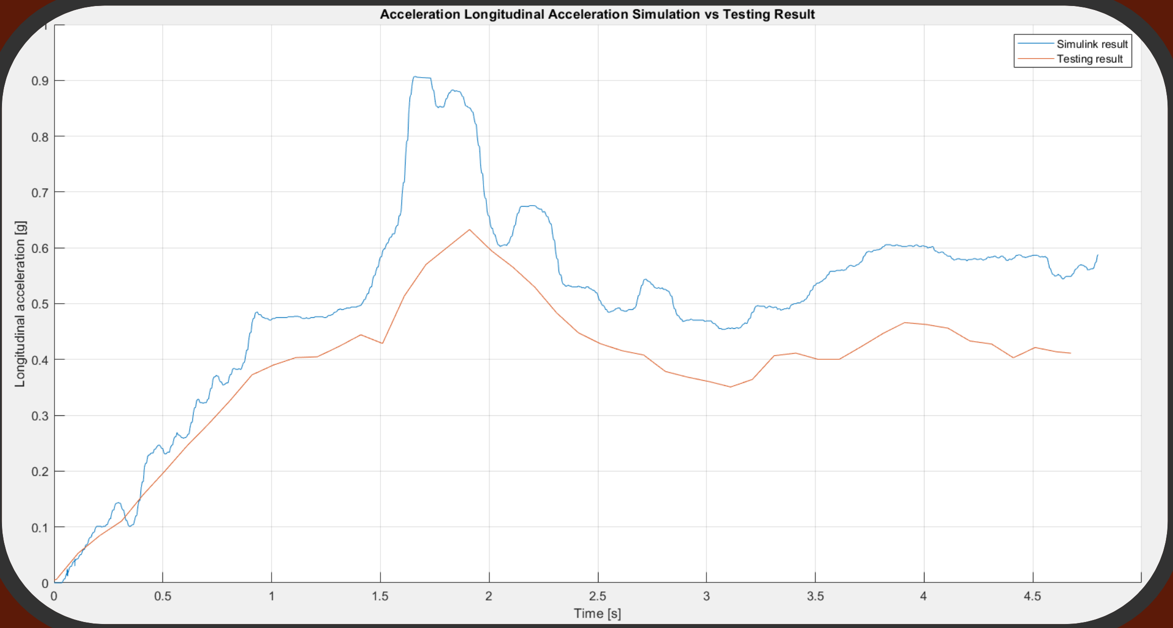
All the measurement gave results that correlated with the expected kinematics very well, meaning the manufacturing quality was good and the welding jigs we designed did its job. It also increased our confident in Lotus Shark and its ability to represent suspension kinematics.

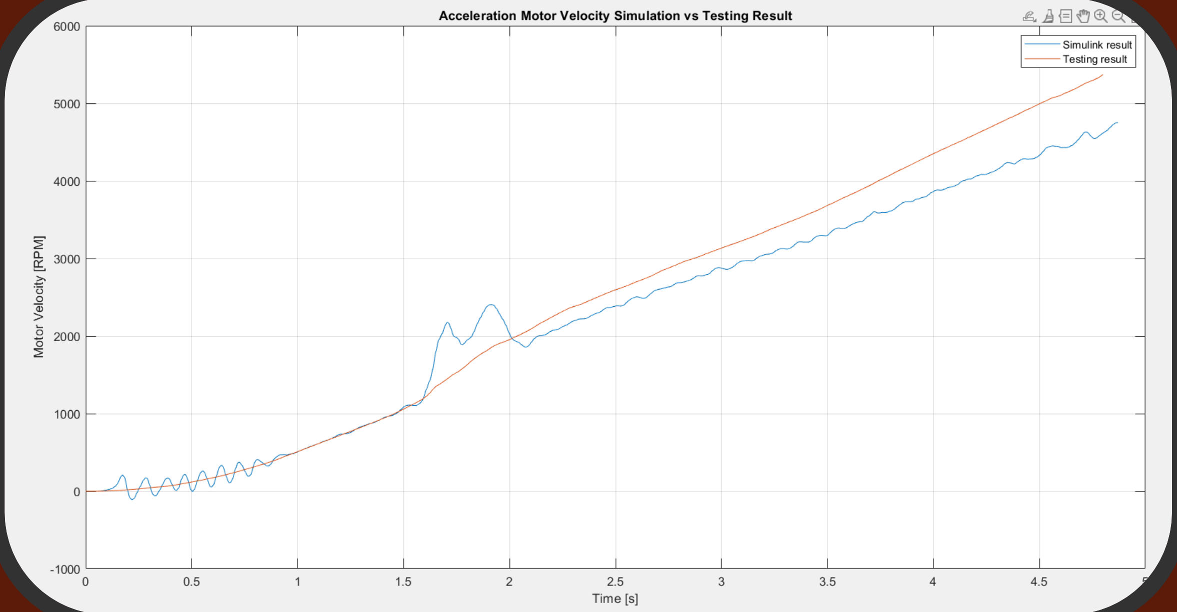




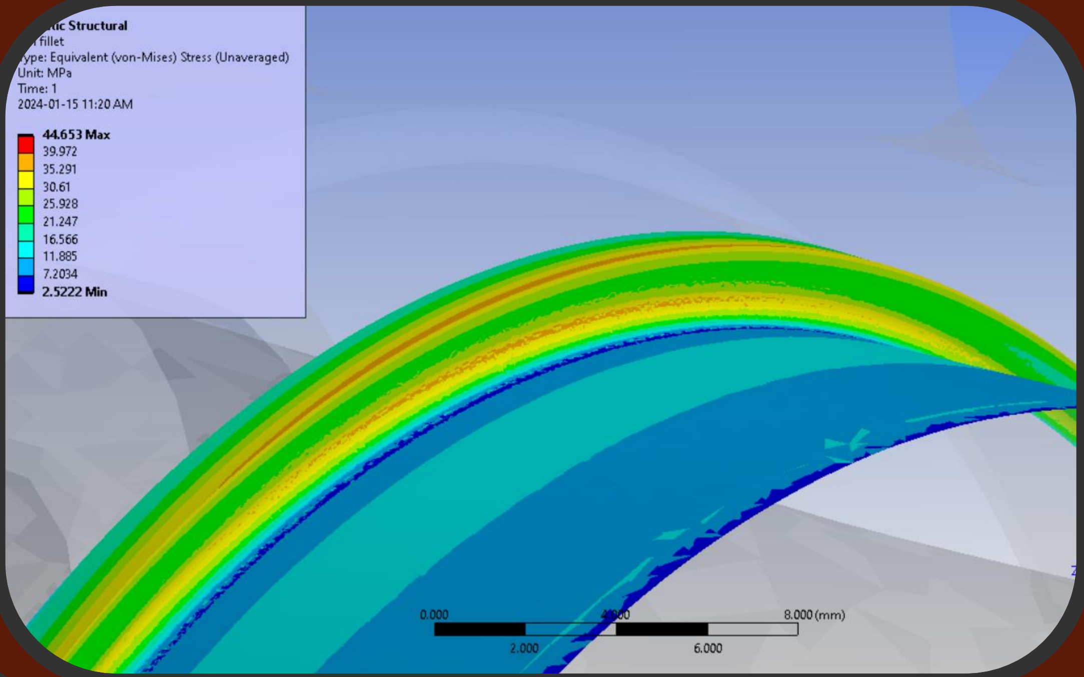
I also developed a 15dof planar vehicle model in Simulink as a tool to simulate and learn vehicle dynamics. However, from the limited testing and data acquisition we had it is obvious that the model is not quite representative yet, and a lot of work is still required for it to become a good simulator tool. Specifically, system identification of physical parameters is a main point of weakness of the team.







I performed structural analysis on the suspension components using various methods. For large bulk bodies such as the upright and hubs, we used Ansys finite element analysis. We faced many hardware limit while using Ansys and unfortunately, we were not able to perform any assembly FEAs.

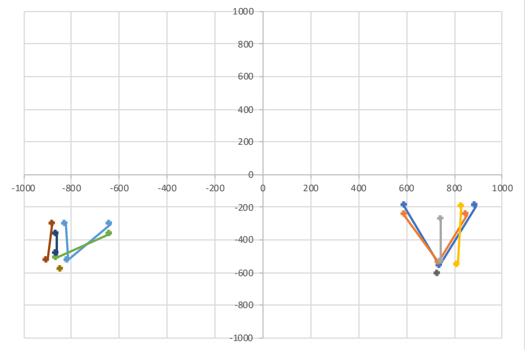


	x	y	z
wer a-arm fore	890.24	-192.2431	105.88
ower a-arm aft	590.54	-192.2431	105.88
lower a-arm upright	739.52	-559	105.88
upper a-arm fore	850.18	-246.4807	270
upper a-arm aft	590.54	-246.4807	270
upper a-arm upright	735	-540	295.63
pushrod outboard	743.3558	-536.3306	129.3417
pushrod bellcrank	743.3558	-275.49	664.39
tie rod Outboard	813.1659	-553.2197	163.6072
tie rod inboard	830	-195.01	153.69
shocks chassis	743.3558	-34.79	643.71
shock bellcrank	743.3558	-194.82	706.77
wheel axis	731	-610.6	203.2
wheel center	731	-609.6	203.2
bellcrank pivot	743.3558	-208.32	628.19
bellcrank axis	742.3558	-208.32	628.19

rear	x	y	z
lower a-arm fore	-639.0029	-303	108
lower a-arm aft	-824.0217	-303	108
lower a-arm upright	-814.6799	-523.2056	108
upper a-arm fore	-639.65	-365	270
upper a-arm aft	-863.4865	-365	270
upper a-arm upright	-862.46	-510.41	292
pushrod outboard	-862.46	-484.315	302.015
pushrod bellcrank	-862.46	-365	425
tie rod Outboard	-902.4919	-523.2056	108
tie rod inboard	-880	-303	108
shocks chassis	-862.46	-99.68	355
shock bellcrank	-862.46	-260	430
wheel axis	-844	-585.2	203.2
wheel center	-844	-584.2	203.2
ellcrank pivot	-862.46	-332.5126	328.9
lcrank axis	-863.46	-332.5126	328.9

front	Load [N]	OD [m]	ID [m]	Young's modulus [GPa]	yield strength [MPa]	Critical stress [MPa]	Actual stress [MPa]
lower a-arm fore	-291.43	0.0127	0.007938	205	435	180.395	-3.775
lower a-arm aft	3704.98	0.0127	0.007938	205	435	71.649	47.996
upper a-arm fore	-1961.75	0.0127	0.007938	205	435	89.657	-25.413
upper a-arm aft	2421.75	0.0127	0.007938	205	435	86.434	31.372
push rod	-1720.01	0.0127	0.007938	205	435	47.649	-22.282
tie rod	-2759.44	0.0127	0.007938	205	435	79.062	-35.747
rear							
lower a-arm fore	-3352.44	0.0127	0.007938	205	435	100.686	-43.429
lower a-arm aft	344.56	0.0127	0.007938	205	435	128.686	4.464
upper a-arm fore	-1922.53	0.0127	0.007938	205	435	106.241	-24.905
upper a-arm aft	1701.78	0.0127	0.007938	205	435	192.855	22.046
push rod	-1955.96	0.0127	0.007938	205	435	165.525	-25.338
tie rod	3018.53	0.0127	0.007938	205	435	128.135	39.103

Loads	
FZ [N]	1500
FX [N]	3415
FY [N]	0
Mz [Nm]	0
Mx [Nm]	0
My [Nm]	0

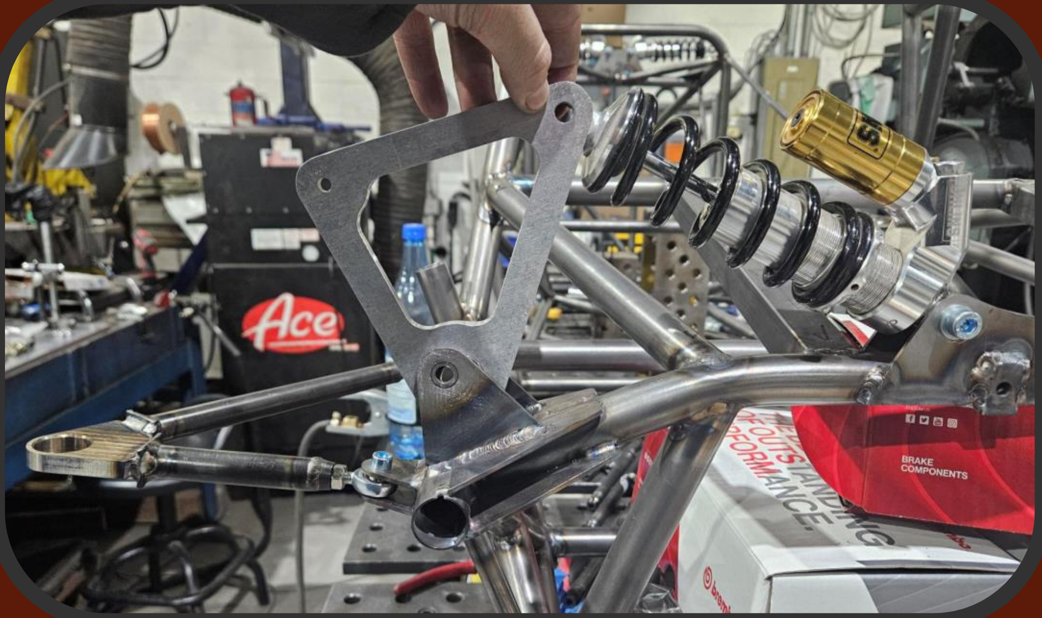


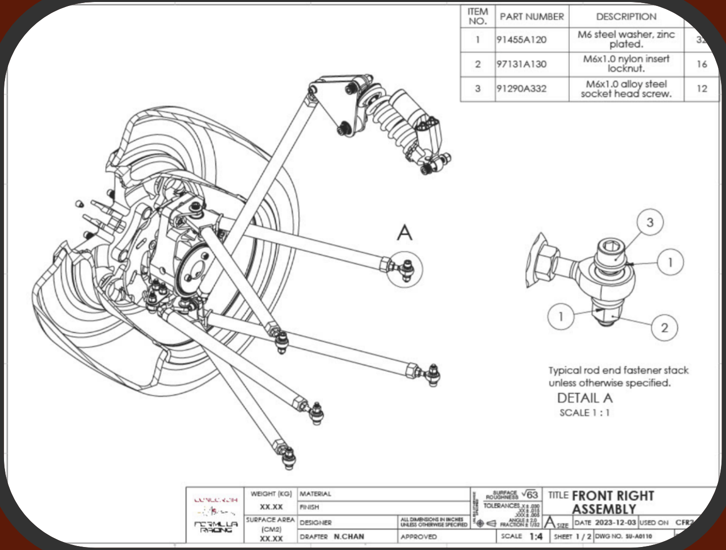
Loads from tire forces	
FZ [N]	1500.00
FX [N]	0.00
FY [N]	3862.41
FX combined [N]	0.00
FY combined [N]	3862.41
Mz [Nm]	33.89
Mx [Nm]	59.23
My [Nm]	0.00

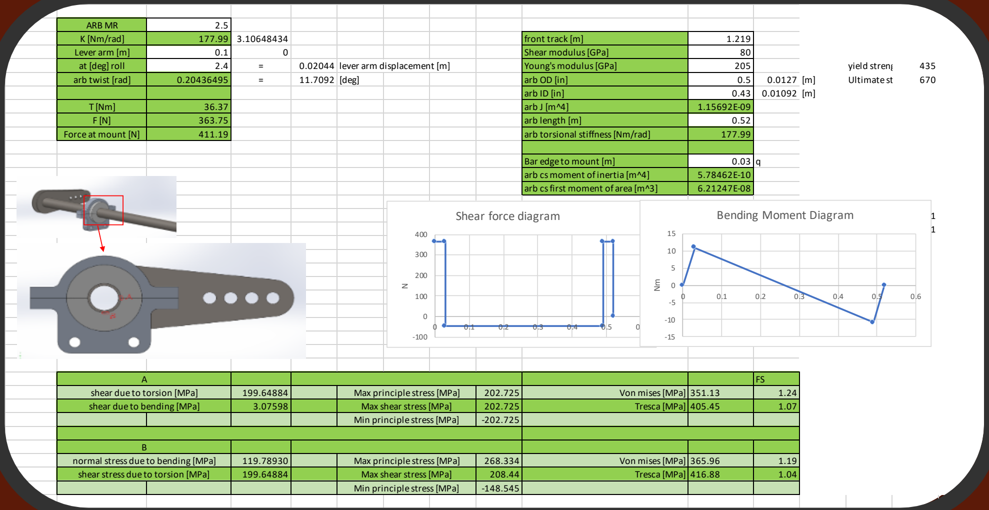
I also developed a spreadsheet to calculate the loads on the suspension linkages. This calculation assumes that all links are two force members. Under these assumptions the tensile/compressive loads can be calculated algebraically since there are 6 links in 6 dof.

This allowed us to quickly calculate the loads on the linkages without having to use large FEA assemblies.

I was responsible of the mechanical design of many components including bell-cranks, wishbones, welding jigs and many more. I also manufactured the components using various techniques including manual and CNC machining.







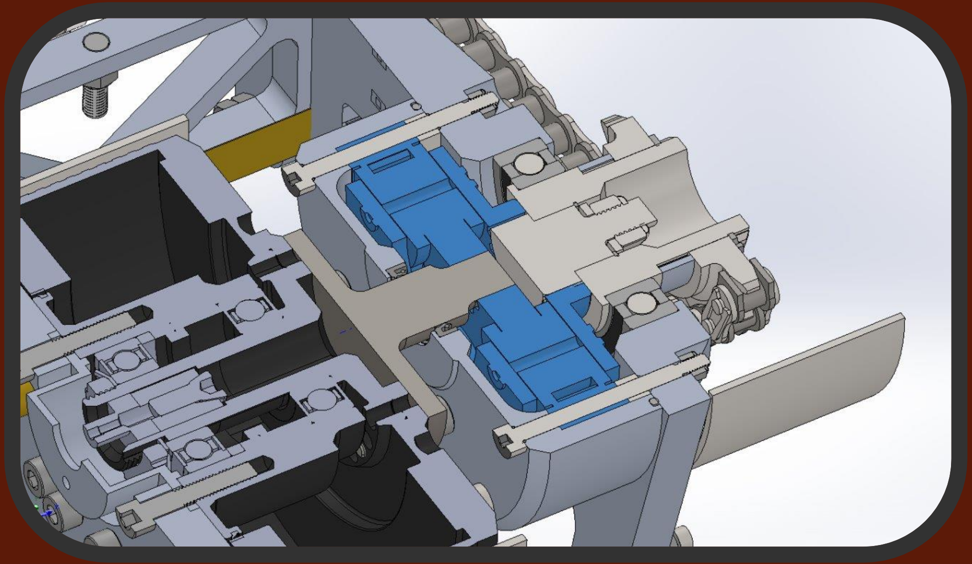
ARB MR	2.5			
K [Nm/rad]	177.99	3.10648434		
Lever arm [m]	0.1	0		
at [deg] roll	2.4	=	0.02044	lever arm displacement [m]
arb twist [rad]	0.20436495	=	11.7092	[deg]
T [Nm]	36.37			
F [N]	363.75			
Force at mount [N]	411.19			

front track [m]	1.219				
Shear modulus [GPa]	80				
Young's modulus [GPa]	205			yield streng	435
arb OD [in]	0.5	0.0127	[m]	Ultimate st	670
arb ID [in]	0.43	0.01092	[m]		
arb J [m^4]	1.15692E-09				
arb length [m]	0.52				
arb torsional stiffness [Nm/rad]	177.99				
Bar edge to mount [m]	0.03	q			
arb cs moment of inertia [m^4]	5.78462E-10				
arb cs first moment of area [m^3]	6.21247E-08				

A						FS
shear due to torsion [MPa]	199.64884	Max principle stress [MPa]	202.725	Von mises [MPa]	351.13	1.24
shear due to bending [MPa]	3.07598	Max shear stress [MPa]	202.725	Tresca [MPa]	405.45	1.07
		Min principle stress [MPa]	-202.725			
B						
normal stress due to bending [MPa]	119.78930	Max principle stress [MPa]	268.334	Von mises [MPa]	365.96	1.19
shear stress due to torsion [MPa]	199.64884	Max shear stress [MPa]	208.44	Tresca [MPa]	416.88	1.04
		Min principle stress [MPa]	-148.545			

I designed and conducted fuse testing to size the battery cell fusible links. These links are designed to blow in the event of a short circuit or failure that causes high current between cells. To test this, we setup a load in series with the fuse and a large truck battery, and iteratively decrease the fuse cutout size until it reaches the desired blow time.

One of the most interesting moment as powertrain lead came when we experienced a gearbox failure during testing in which the gearbox seized as the car was exiting a corner, in the process breaking the chain and 2 sprocket teeth as well.



There were gear marks on the spacer of the ring gear, and there were heat affected zones on the carrier where the rollers of the planet gears are.

The root cause of this failure was that the carrier was not constrained axially, allowing parts to shift and rub, ultimately seizing.

Changes we implemented included: reduce the complexities by using spur gears instead and implementing gear lubrication check more rigorously.

Personal Project: Motor controller

As an ongoing personal project, I am working on custom motor controller firmware for small BLDC drone motors. This project is mainly for me to learn about the math and workings of motors and motor controllers, which I plan on implementing in my future projects.

To supplement this project, I also made a simple motor simulator in Matlab. Details can be found in my Github.