SUSPENSION

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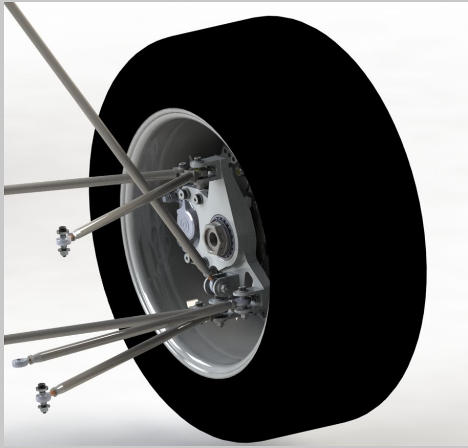
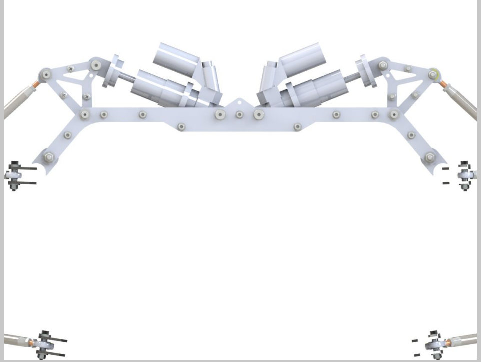


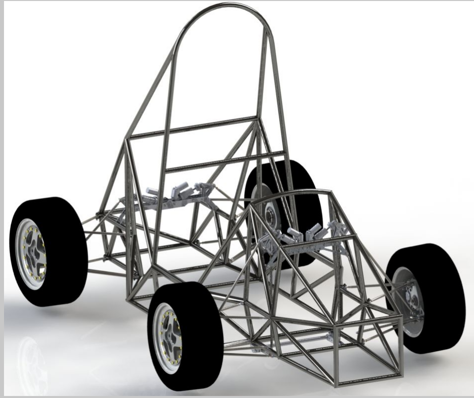


Overview

Design Methodology

Functional requirements
1. Make the car easy to control and handle
2. Control the tires to create maximum grip to the track
3. Dampen vibrations induced by acceleration, braking, turning, and bumps

2020-2021 season goals
- Simplify manufacturing
- Drastically reduce cost
- Reduce unnecessary unsprung weight
- Defend design decisions with hand calculations and simulations
- Improve system adjustability for tuning and driver preferences



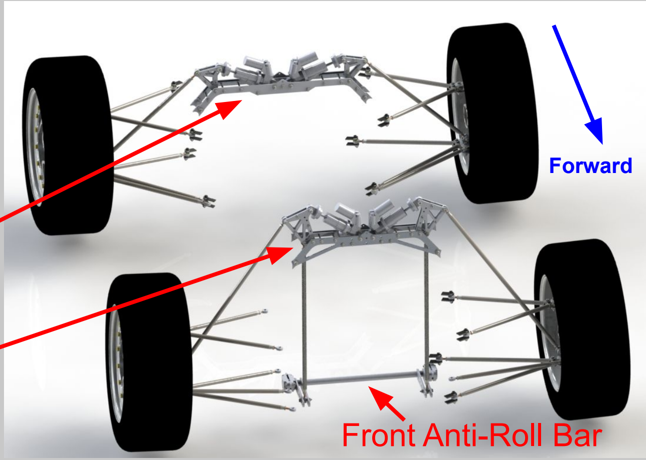
Specifications

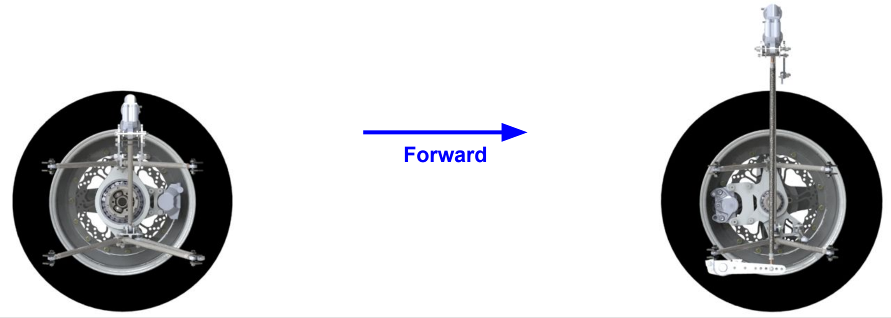
Property	Value	
	Front	Rear
Weight	47 lbs	55 lbs
Wheelbase	62 inch	
Track	52.3 inch	50.8 inch
Suspension Type	Double wishbone, unequal length, push rod	Double wishbone, unequal length, push rod
Motion Ratio	0.725	0.789
Anti-Roll Bar Stiffness	92.7 Nm/deg	N/A

Design Overview

Rear damper assembly

Front damper assembly





Rear wheel assembly

Front wheel assembly

Lap Simulator

Design Overview

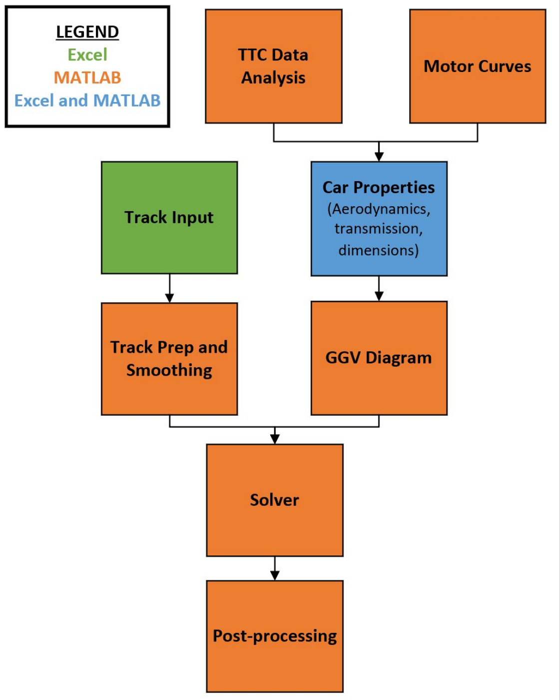
Goals:

- Observe trends in vehicle setup
- Inform initial design decisions for tire selection, motor selection, and battery sizing

Assumptions:

- Steady state
- Point mass
- Perfect driver
- No yaw inertia
- Constant slip angle and slip ratio

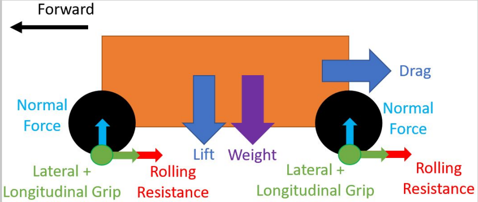
Program Structure:



Model

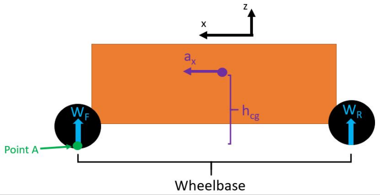
Steady state weight distribution on tires:

- Lift
- Weight
- Drag

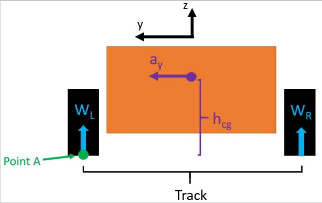


Steady state weight transfer:

- Longitudinal acceleration



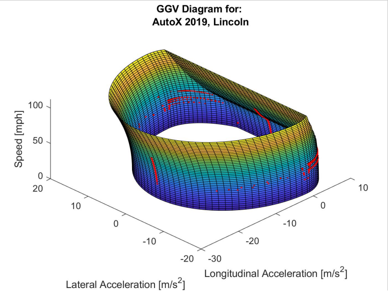
- Lateral acceleration



Results

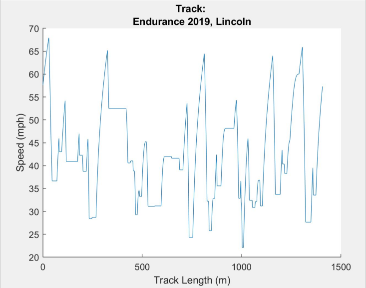
Simulator produces estimated GGV diagram for vehicle setup

- Plots simulator data on diagram
- Provides range of speeds and accelerations for each track



Simulator produces estimate of vehicle speed throughout each track

- Used to estimate lap times and energy consumption
- Have tires "compete" each other using FSAE scoring system



Tires

Selection Criteria

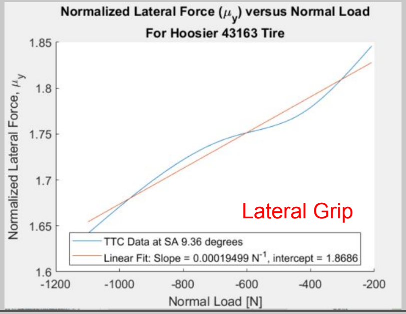
Consideration	Reason
Performance	Impacts the car's grip in cornering and braking scenarios
Weight	Extra weight limits the car's acceleration, increases unsprung mass
Trail	Impacts driver feedback and handling
Cost	Tire selection may impact cost of components in the wheel package
Packaging	Size of tire impacts how components are packaged inside the wheel

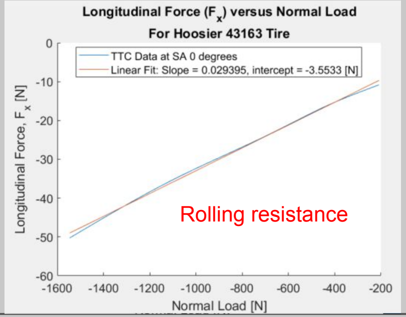
Linear Model

*Data provided by Formula SAE Tire Test Consortium (FSAE TTC) and Calspan Tire Research Facility (TIRF)

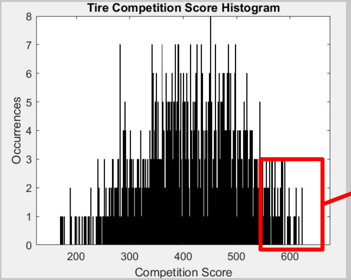
Linear model used to describe each tire's performance

- Helpful for initial tire selection
- Selected at max load slip angle or slip ratio



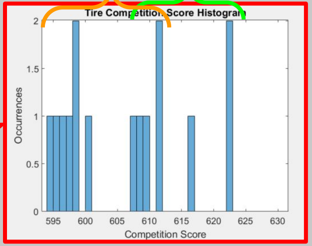


Performance Scoring



Hoosier 43105

Hoosier 43163



Tires are scored by "competing" in FSAE competition:

- Each setup races endurance, autocross, acceleration, and skidpad
- Weighted score generated using each event's scoring rules, provided in 2021 rules

Top scorers include Hoosier 43163 13" tire and Hoosier 43105 10" tire

Tradeoff Table

	Hoosier 43163 (13" OD Wheel)	Hoosier 43105 (10" OD Wheel)
Performance	About equal, Hoosier 43163	About equal, Hoosier 43105
Mass/Inertia	Slightly more (11 lbs tire)	Slightly less (10 lbs tire)
Cost	Guaranteed $2,300	At least $1,400, plus risky costs (wheel hub, brake rotor)
Packaging Difficulty	VERY easy	Difficult, even with custom components

Both tires provide similar magnitudes of performance and inertia

- Hoosier 43163 tire leads to cheaper and more open wheel package
- To increase testing time and decrease manufacturing cost, **Hoosier 43163, 20.5 x 7.0-13 R25B** tire selected

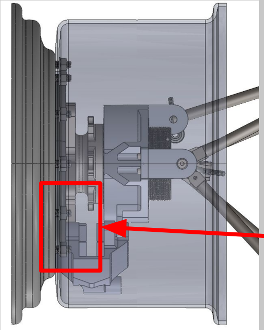
Wheel

Wheel Selection Overview

Requirement	Reason
Packaged parts must fit inside wheel package	Simplifies design and maintenance
Wheel must be of lightweight material	Reduces unsprung weight and rotational inertia
Matching hub must be simple	Reduce manufacturing complexity
Wheel must place bearings close to centerplane	Lowers bearing loads
Wheel must be cheap or reused from past season	Less funding to purchase expensive wheel options
Wheel must not produce excessive camber compliance	Need to control camber of each tire within reason

Packaging

10" OD wheel

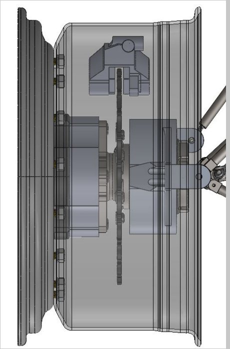


Caliper interference with hub and wheel

Difficult to package caliper, upright, and off-the-shelf wheel hub in a 10"

- Caliper interferes with wheel hub flange and wheel center

13" OD wheel



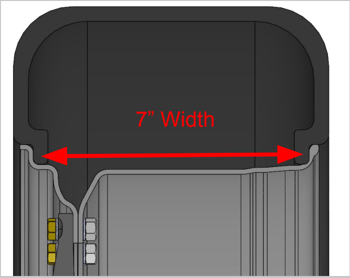
13" wheel greatly simplifies packaging

- Caliper, upright, and off-the-shelf wheel hub easily fit

Integration

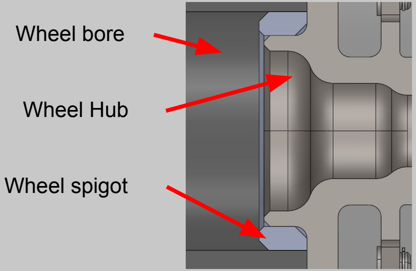
Tire data indicates 7" width provides optimal performance

- 7" wheel width chosen



Must align with wheel hub to reduce eccentric rotation

- Delrin wheel spigot used for mounting alignment



Wheel Decision Matrix

Wheel	Packaging	Mass	Hub Complexity	Wheel Bearing Loads	Cost	Compliance	Total
Weighting	10	7	10	3	4	6	
Keizer CL10	5	6	2	5	4	4	167
Keizer 10i	2	6	3	4	5	5	154
Keizer Kosmo 13"	8	3	7	4	7	3	229
OZ Mags 10"	2	8	2	3	3	3	135

Based on this season's priorities, we selected the **Keizer Kosmo 13" wheel**

- 4" backspacing → reduces upright and wheel bearing loads
- 7" width → preferable tire performance for Hoosier 43163

Geometry

Design Overview

Requirement	Reason
Front and rear track must provide sufficient rollover stability	Prevent rollover for high lateral accelerations
Wheelbase must be near minimum allowed	Keep turn radius low for FSAE-style courses
Geometry must implement unequal length, non parallel double wishbone	Provides more tuning range, additional weight savings, and easier analysis
Tire camber must be controlled between 0° and -3° during pitch and roll	Tire performance varies with camber angle
Tire must provide feedback to driver	Indicates max potential of tires, increasing performance
Settings must be adjustable	Provides range for tuning and driver preferences

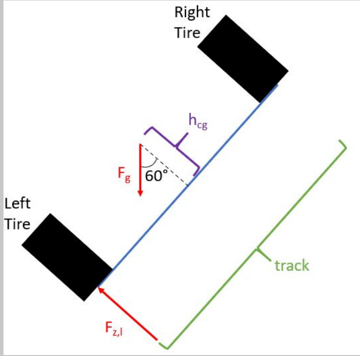
Wheelbase and Track

Wheelbase (1.575 meters)

- 2 inch margin accounts for manufacturing tolerances

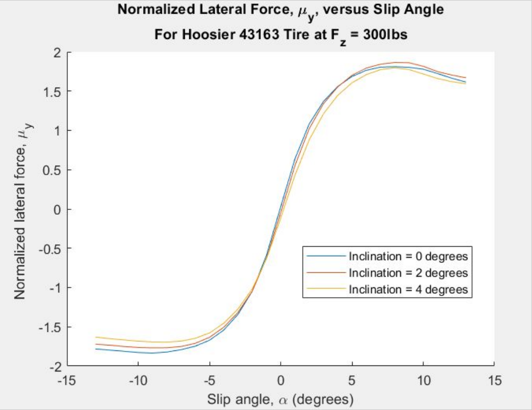
Track (Front = 1.328 meters, Rear = 1.290 meters)

- Provides rollover stability in 1.7g cornering with 344 mm CG height
- Margin is added:
  - Front margin: 7%
  - Rear margin: 5%
- From experience, rear track is smaller to reduce cone contact
- Car will fit in standard U-Haul 10 foot truck
  - Easy transportation



Camber

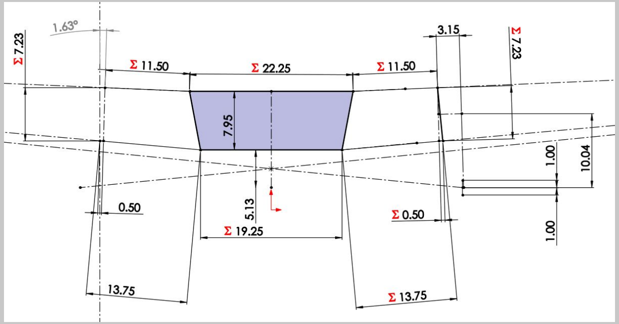
*Data provided by Formula SAE Tire Test Consortium (FSAE TTC) and Calspan Tire Research Facility (TIRF)



Using TTC data, Hoosier 43163 tire should be kept near -2° camber angle for optimal lateral grip

- Small decrease in performance for 0° camber angle at positive slip angle
- Significant decrease in performance for -4° camber angle at all slip angles

During expected loads, tires are constrained to keep camber angle between 0° and -3° for pitch and roll



Front and rear geometry initially set in 2D sketches

- Further modified to achieve desired camber change with roll and pitch

Load Cases

CALTECH RACING

Load Case Description

Load Case	Explanation or Description
Acceleration	Increases normal load on rear tires
Braking	Greatly increases normal load on front tires
Steady State	Provides a reference for frequent loading on each tire
Cornering	Load greatly increases moment on uprights
Braking, Hard Cornering	Checks effect of large lateral force with smaller braking force
Hard Braking, Cornering	Checks effect of large braking force with smaller lateral force
Acceleration and Cornering	Mix of two conditions increases loading on rear tires
3G Bump	If vehicle is dropped or hits bump, normal load greatly increases
Reverse Brake	If vehicle spins, normal load greatly increases on rear tires

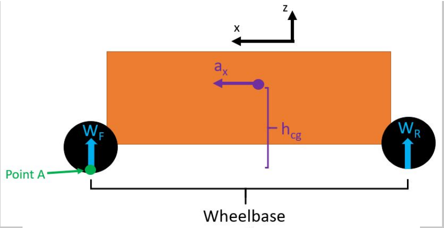
Analysis

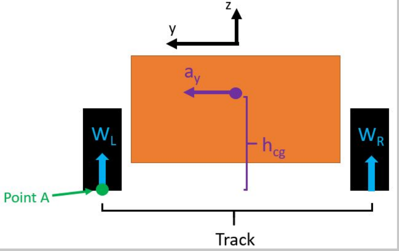
Lap simulator provides estimate of the car's accelerations in different driving scenarios.

These estimates use data from the tires, aerodynamics, and motor. These provide an overestimate, which we prefer before we validate our design.

Longitudinal and lateral accelerations are used to estimate weight transfer on each tire assuming static equilibrium.

Tire data then used to estimate lateral and longitudinal forces on each tire.





Results

Load Case		Acceleration	Braking	Steady State	Cornering	Braking, Hard Cornering	Hard Braking, Cornering	Acceleration and Cornering	3G Bump	Reverse Brake
Car Acceleration	Lon [G]	0.88	-2.46	0	0	-1.04	-1.9	0.76	0	2.46
	Lat [G]	0	0	0	1.97	1.78	1.08	1.35	0	0
Front Tire	F_x [N]*	0	-2000	0	0	-1521	-1356	0	0	211
	F_y [N]*	0	0	0	2460	2601	1346	1147	0	0
	F_z [N]*	-384	-1333	-622	-1296	-1213	-1288	-879	-1866	-87
Rear Tire	F_x [N]*	1188	-211	0	0	-1112	-899	1029	0	2000
	F_y [N]*	0	0	0	2754	1901	892	1885	0	0
	F_z [N]*	-1014	-87	-777	-1451	-1106	-854	-1446	-2331	-1333

*Coordinate system defined by SAE J670

Braking, Hard Cornering: Worst case for front upright, wheel hub, and bearings

- Large combined tire force, including large F_y

Cornering: Worst case for rear upright, wheel hub, and bearings

- F_y is applied at greater distance (compared to other tire forces), so larger moment

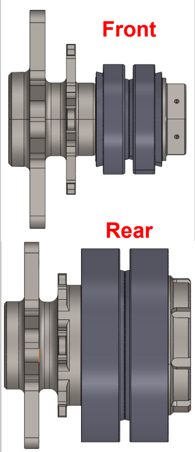
Braking and Reverse Brake: Worst case for rod ends, spherical bearings, and control arms

- Large F_x and F_z cause large forces on lower A-arm

Wheel Hub Assembly

Assembly Overview

Part	Requirement	Reason
Wheel Hub	Off-the-shelf component	Reduce manufacturing cost and complexity
	Must mount brake rotor	Interface with brake team
	Must have 4x100mm stud pattern	Mounts to wheel
Wheel Bearings	Must have seals	Prevent contamination and seize
	L10 life must exceed 1000 km at max speed (30m/s)	Max expected lifetime (about 50 endurance races of testing)
	Must support radial and axial loads	Lateral loads from cornering
	Bore must match wheel hub OD	Interface with wheel hub



Bearing Analysis and Selection

Front	Case: Braking, Hard Cornering	Outboard L10 Safety Factor: 6.7	Inboard L10 Safety Factor: 7.6
Rear	Case: Cornering	Outboard L10 Safety Factor: 13.1	Inboard L10 Safety Factor: 14.9

Front Bearing Life			
Variable	Value	Units	Notes
Weibull Parameter	4.84		99.5% reliability
Bearing 1, Equivalent Load	1811.20	N	
Bearing 2, Equivalent Load	1600.87	N	
Safety Factor, Bearing 1	6.68		
Safety Factor, Bearing 2	7.56		

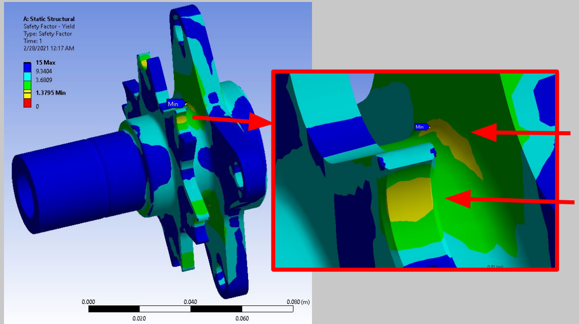
Rear Bearing Life			
Variable	Value	Units	Notes
Weibull Parameter	4.84		99.5% reliability
Bearing 1, Equivalent Load	1958	N	
Bearing 2, Equivalent Load	1722	N	
Safety Factor, Bearing 1	13.1		
Safety Factor, Bearing 2	14.9		

Component	Details
Front Wheel Bearing	Tapered roller bearing, Timken, LM67048/LM67010, sealed
Rear Wheel Bearing	Tapered roller bearing, Timken, 390A/394, sealed

Stress Analysis

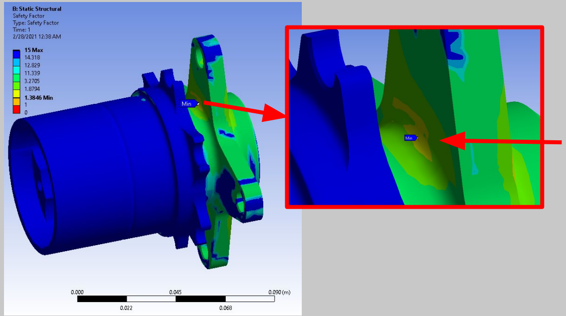
Load Case: Braking, Hard Cornering	Solver: Ansys	Material: Linear	Geometry: Linear
Safety Factor Yield: 1.4	Safety Factor Ultimate: 2.0	Temperature: 22°C	Notes: Front hub

Load Case: Cornering	Solver: Ansys	Material: Linear	Geometry: Linear
Safety Factor Yield: 1.4	Safety Factor Ultimate: 2.0	Temperature: 22°C	Notes: Rear hub



Bending behind hub flange (large moment with cornering)

High contact stresses on rotor mounts from max braking



Contact stresses from tripod CV joint are minimal

Bending behind hub flange (large moment with cornering)

Upright

Design Overview

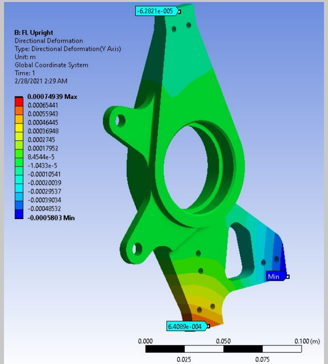
Requirement	Reason
Must be manufacturable on a 3-axis CNC mill	Reduce manufacturing cost and complexity
Yield safety factor > 1.4 for worst case loading	Strong for worst case loading with margin
Must produce less than 0.25° of camber angle compliance during worst case loading	Tire performance varies with camber angle
Must produce less than 0.75° of steering angle compliance during worst case loading	Controlling steering angle improves handling
Must have pockets to reduce unnecessary mass	Reduce unsprung mass
Must integrate with selected wheel bearings	Upright houses wheel bearings

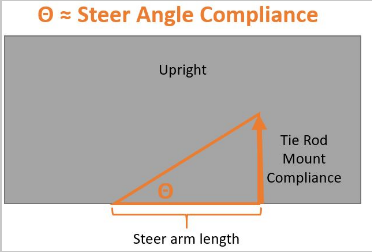
Front upright mass: 384 grams

Rear upright mass: 962 grams

Front Steer Compliance

Case: Hard braking, cornering	Solver: Ansys	Material: Linear	Geometry: Linear	Temperature: 22°C

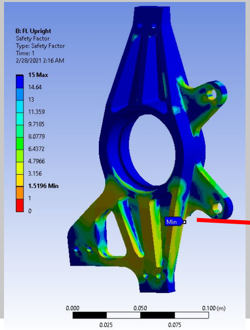


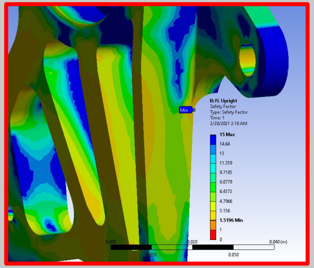


Front upright steer angle compliance ≈ 0.56°

Stress Analysis - Front

Load Case: Braking, Hard Cornering	Solver: Ansys	Material: Linear	Geometry: Linear
Safety Factor Yield: 1.5	Safety Factor Ultimate: 1.8	Temperature: 22°C	Notes: None



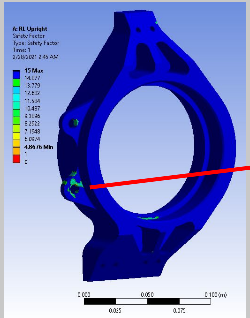


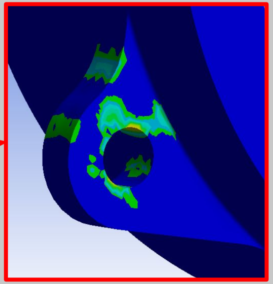
Minimum safety factor due to large bending stress in bottom half

Front upright to be machined out of 7075-T651 to attain high strength to weight ratio needed

Stress Analysis - Rear

Load Case: Cornering	Solver: Ansys	Material: Linear	Geometry: Linear
Safety Factor Yield: 4.9	Safety Factor Ultimate: 5.6	Temperature: 22°C	Notes: None





Minimum safety factor due to caliper mount bearing stress

Rear upright to be machined out of 7075-T6 to maintain high margins with lower mass

Upright (Continued)

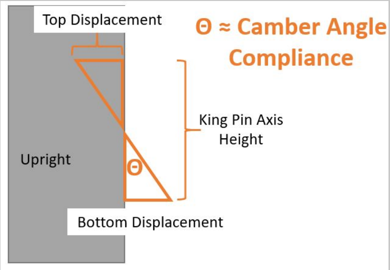
Camber Compliance Analysis

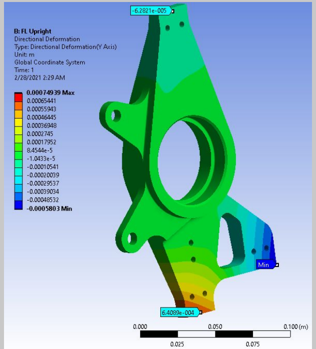
Front

Case: Hard braking, cornering	Solver: Ansys	Material: Linear	Geometry: Linear	Temperature: 22°C

Rear

Case: Cornering	Solver: Ansys	Material: Linear	Geometry: Linear	Temperature: 22°C



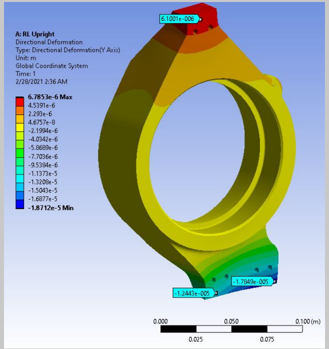


	Front	Rear
Top Displacement [mm]	0.063	0.0061
Bottom Displacement [mm]	0.64	0.012
King Pin Axis Height [mm]	184	184
Total Displacement [mm]	0.70	0.02
Camber Angle Compliance [°]	0.22	0.01

Legend
Inputs
Intermediate
Results

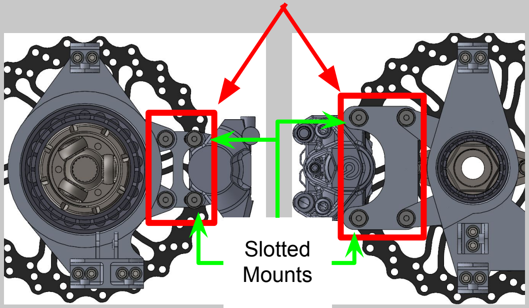
Front upright camber angle compliance ≈ 0.22°

Rear upright camber angle compliance ≈ 0.01°



Caliper Mounting

Each caliper is fastened using a separate mount



Rear

Front

These mounts:

- Use slots to provide caliper adjustment
- Are designed to fail in yielding before upright, to protect this costly component

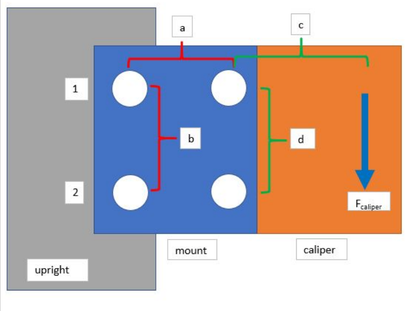
Caliper Shear Calculations

Caliper mount fasteners loaded in shear during braking

8mm diameter shoulder bolts

- Set by selected caliper

Fastener shear safety factor > 9.8 for all fasteners



	Fastener	Shear Stress [Pa]	Safety Factor
Front	Caliper/Mount	4.36E+07	13.3
	Mount/Upright	5.91E+07	9.8
Rear	Caliper/Mount	2.21E+07	26.3
	Mount/Upright	2.95E+07	19.7

Damper Selection

Category Breakdown

To organize our priorities, we developed a weighted decision matrix. 6 dampers in total were considered, with the top 3 shown on the right.
Final Choice: Öhlins TTX25 50mm dampers.

We plan to ensure minimal damper hysteresis by graphing the displacement of the bellcrank over various loads using a linear potentiometer (see ‘Testing’ slide later).

We used the following categories to facilitate our selection, listed from highest to lowest priority:

Category	Reason
Cost	Low cost needed to ensure budget for rest of system.
Adjust-ability	Damper tuning capabilities are crucial to optimizing performance while testing.
Mass	Lower mass improves vehicle efficiency.
Extended Length	Compact packaging helps reduce change of system interference.
Type	We prioritized coil-over dampers as they were more familiar and we didn’t have resources to commit learning another damper type.
Diameter	A larger piston allows for lower internal pressures to achieve the same force, reducing friction.

Weighted Damper Decision Matrix

Damper	Cost	Type	Diameter	Extended Length	Mass	Adjust-ability	Total
Weighting	10	5	5	5	7	10	
Öhlins TTX25 MkII 50mm	1.5	3	1	3	3	3	86
	$650	Spring	15mm	200mm	349g	4-way	
DBair IL	2	1	0*	3	3	3	76
	$475	Air	N/A*	200mm	397g	4-way	
Penske Quarter Midget Shocks	2.5	3	3	1	0*	1	65
	$200	Spring	41mm	270mm	N/A*	None	

* Scores which could not be determined were automatically given a zero for surety. There were lower ranking dampers with all categories accounted for.

Damper System Setup

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Design Overview

We set the following guidelines for the damper system setup:

- Avoid interference with all other systems.
- Minimize compliance
- Preferably, keep in-plane/2D.
  - This simplifies motion ratio calculation, as shown in the next page.
- Allow for motion ratios (MR) near 1.0.
- Try to keep weight low.
  - Ultimately, a lower priority.

Ultimately, we ended up with an in-plane push-rod configuration, mounted onto the chassis with brackets fastened around spacers with bellcranks mounted on corners to reduce compliance.

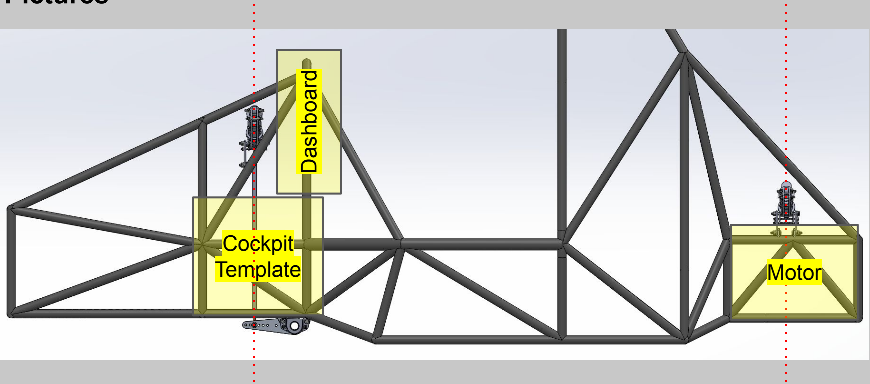
Overall Damper System Weight: 10.90 lbs.

Decision Matrix

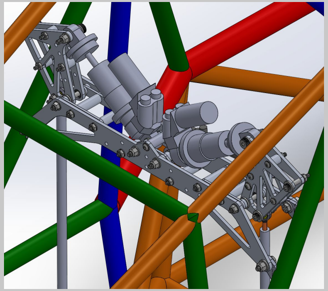
To ameliorate the cons listed with the in-plane push-rod set-up, we moved the position of the damper control on the upright slightly forward so that planarity could be maintained while clearing the motor shaft, and the width of the brackets was decreased by .25in to reduce weight.

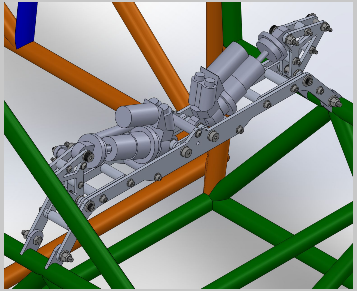
Options	Pros	Cons
Direct Actuation	- Less manufacturing/weight	- Force directly on frame beam (lots of compliance)
Pull Rod	- Won't interfere with motor shaft in rear	- Less accessible - MR ~ 0.5
Out-Of-Plane Push Rod	- MR ~ 1.0 - Accessible	- Out-of-plane forces on bell crank - Difficult to simulate/calculate
In-Plane Push Rod	- MR ~ 1.0 - Accessible	- Hits motor shaft in rear - Adds weight

Pictures



Note how the push-rod set-up allows for easy avoidance of system interference.





Front set-up on left, rear set-up on right.

Rear is mounted at nodes for minimal compliance. Front is mounted on diagonal beam which should reduce compliance in exchange for shear, hence we okay-ed the set-up, though it remains untested/not simulated.

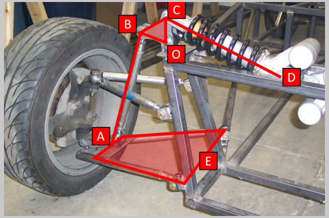
Motion Ratio Analysis

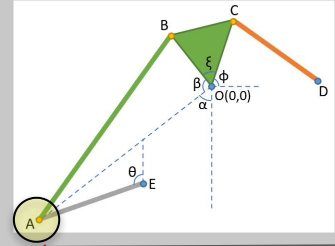
Design Overview

By setting the damper system in-plane, we're able to represent the damper geometry in 2D. Using MATLAB, we can quickly calculate for the overall motion ratio as well as how it changes over the wheel's travel. This quick calculator was quite useful during design as the car layout kept changing and new interferences arose.

We had 3 initial goals:

1) Avoid all system interference.
2) A motion ratio (MR) of 1.
    - Goal changed after further design.
3) Linear/steady motion ratio.
    - Unable to obtain this, we settled for a slightly increasing MR, indicating increasing stiffness into a turn.



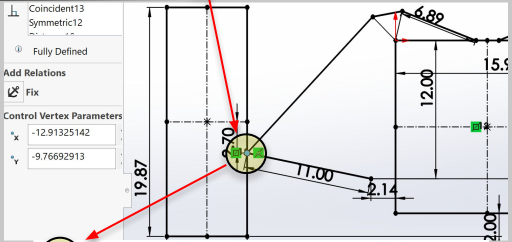


Calculations

From a SolidWorks sketch of the ride-height geometry, the MATLAB script takes the points and desired minimum jounce and rebound (1" above and below) as inputs and outputs:

- Overall motion ratio
- Instantaneous motion ratio over wheel displacement*
- Maximum jounce and rebound (corresponding with shock bounds)
- Shock length over time*
- Suspension geometry at minimum desired jounce and rebound*
-

(*) = Graph/figure output

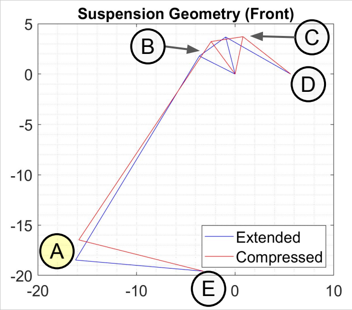


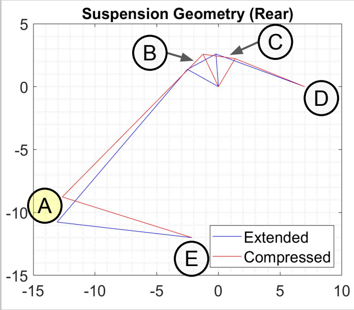

```
A = [-12.91325142 -9.76692913];
B = [-1.99443341 2.05427208];
C = [.57802889 2.53949463];
D = [6.98270 0];
E = [-2.1423 -12];
frontorrear = "Front";
wheel_height_ext = -1; %Damper ful
wheel_height_com = 1; %Damper ful
```

Analysis

	Front	Rear
Overall Motion Ratio	.7253	.7887
Maximum Jounce (in)	1.516	1.478
Maximum Rebound (in)	1.549	1.346

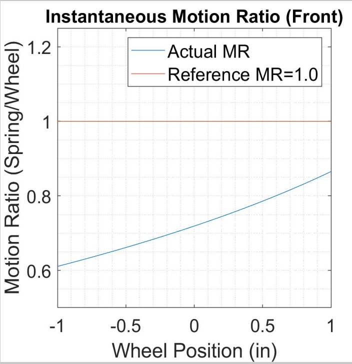
Suspension geometry labels correspond with figure to left.

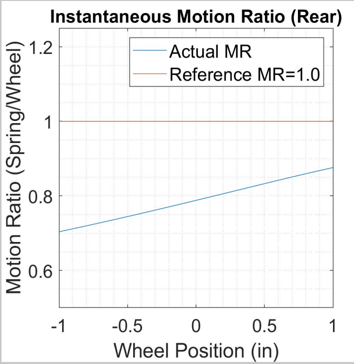




Both front and rear graphs show progressive motion ratios, which stiffen as the wheel center rises.

Average slope of MR increase:
- Front: .126
- Rear: .086





Spring Analysis

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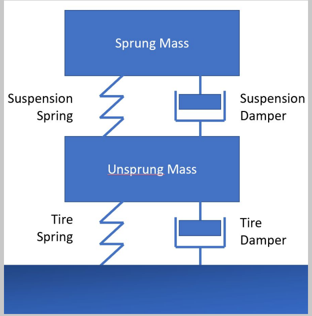
Design Overview

To solve for spring rates, we set 2 requirements for their ride frequencies:

1) Lie between 2.5-3.5 Hz.
2) Converge in 1.5-2 cycles.
    - Time difference in 1.75 cycles < 10% of front wheel ride period.
    - Minimizes pitch after a bump.

Knowing the component masses' and vehicle centroid, we calculated quarter-mass approximations for each wheel-system, which then could be modeled as 2 springs-in-series, letting us solve for desired spring rates using online tire data and guided by our requirements.

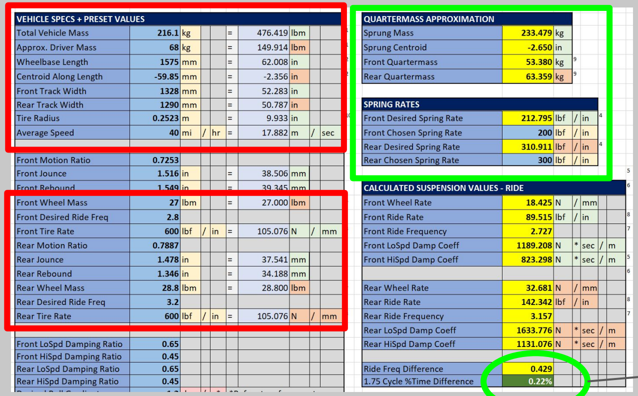


Calculations

Our main inputs are boxed in red, while our main outputs are boxed in green.

This Excel sheet also outputs corresponding damping coefficients given desired damping ratios.

A MATLAB script graphing how the front and rear frequencies converge over a bump accompanies this analysis.



VEHICLE SPECS + PRESET VALUES									
Total Vehicle Mass	216.1	kg			=	476.419	lbm		
Approx. Driver Mass	68	kg			=	149.914	lbm		
Wheelbase Length	1575	mm			=	62.008	in		
Centroid Along Length	-59.85	mm			=	-2.356	in		
Front Track Width	1328	mm			=	52.283	in		
Rear Track Width	1290	mm			=	50.787	in		
Tire Radius	0.2523	m			=	9.933	in		
Average Speed	40	mi	/	hr	=	17.882	m	/	sec
Front Motion Ratio	0.7253								
Front Jounce	1.516	in			=	38.506	mm		
Front Rebound	1.549	in			=	39.345	mm		
Front Wheel Mass	27	lbm			=	27.000	lbm		
Front Desired Ride Freq	2.8								
Front Tire Rate	600	lbf	/	in	=	105.076	N	/	mm
Rear Motion Ratio	0.7887								
Rear Jounce	1.478	in			=	37.541	mm		
Rear Rebound	1.346	in			=	34.188	mm		
Rear Wheel Mass	28.8	lbm			=	28.800	lbm		
Rear Desired Ride Freq	3.2								
Rear Tire Rate	600	lbf	/	in	=	105.076	N	/	mm
Front LoSpd Damping Ratio	0.65								
Front HiSpd Damping Ratio	0.45								
Rear LoSpd Damping Ratio	0.65								
Rear HiSpd Damping Ratio	0.45								

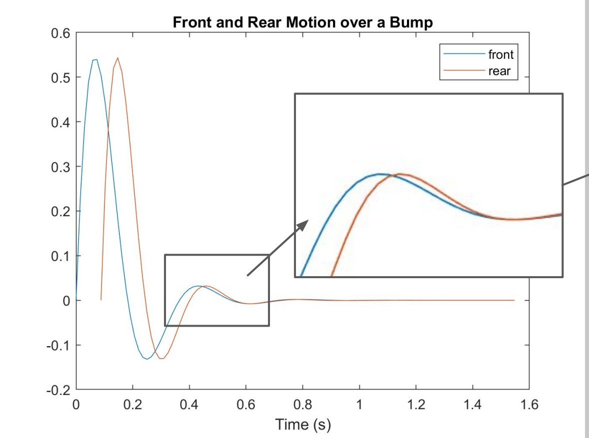
QUARTERMASS APPROXIMATION		
Sprung Mass	233.479	kg
Sprung Centroid	-2.650	in
Front Quartermass	53.380	kg
Rear Quartermass	63.359	kg

SPRING RATES				
Front Desired Spring Rate	212.795	lbf	/	in
Front Chosen Spring Rate	200	lbf	/	in
Rear Desired Spring Rate	310.911	lbf	/	in
Rear Chosen Spring Rate	300	lbf	/	in

CALCULATED SUSPENSION VALUES - RIDE						
Front Wheel Rate	18.425	N	/	mm		
Front Ride Rate	89.515	lbf	/	in		
Front Ride Frequency	2.727					
Front LoSpd Damp Coeff	1189.208	N	*	sec	/	m
Front HiSpd Damp Coeff	823.298	N	*	sec	/	m
Rear Wheel Rate	32.681	N	/	mm		
Rear Ride Rate	142.342	lbf	/	in		
Rear Ride Frequency	3.157					
Rear LoSpd Damp Coeff	1633.776	N	*	sec	/	m
Rear HiSpd Damp Coeff	1131.076	N	*	sec	/	m
Ride Freq Difference	0.429					
1.75 Cycle %Time Difference	0.22%					

Analysis

	Front	Rear
Allowed Spring Rates (lb/in)	175-225	300-350
Wheel Rate (N/mm)	16.12-20.73	32.68-38.13
Ride Frequency (Hz)	2.58-2.87	3.16-3.34
Low Speed Damping Coefficients (Ns/m)	1123-1250	1634-1731
High Speed Damping Coefficients (Ns/m)	777-865	1131-1198



Graphed left are front and rear spring rates of 200 and 300 lb/in, respectively. Can visually see tight convergence of front and rear ride frequencies.

0.22% time difference after 1.75 cycles, Indicating minimal pitch.

Anti-Roll Bar (ARB)

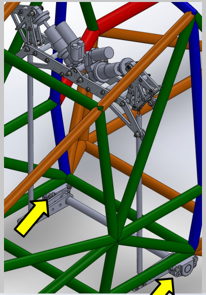
Design Overview

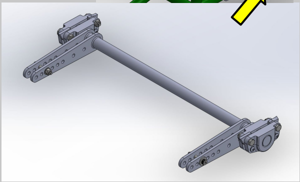
While the spring rates were tuned to minimize pitch, we still needed to account for roll.

To determine the need for anti-roll bars, we set the following a goal of a roll gradient ≤1.2 deg/g. We found the front spring rates weren't stiff enough to provide an adequate roll rate to meet this goal, and hence added an ARB.

For ease of manufacturing, the ARB consists of a 0.75" aluminum 6061-T6 shaft with welded plates serving as cantilevers, and is axially constrained by two bushings fastened onto tabs at nodes, minimizing compliance. Holes along the lever arm allow for adjustability.

ARB Weight with Bushing Attachment: 1.50 lbs.

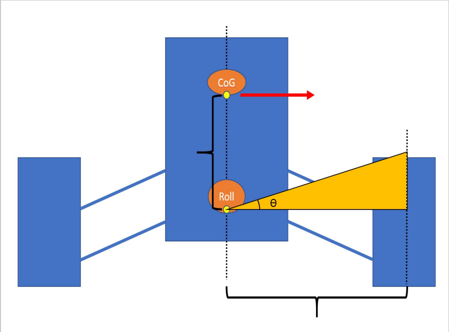




Model

For the front and rear separately, we modeled the load transfer by setting moment equilibrium between the lateral acceleration of the sprung mass about the roll axis and the spring force reacting to it applied at the wheel, as shown in the diagram to the right.

From there, we used Excel to calculate the need for an anti-roll bar, as well as its contribution to the roll rate given its material properties, dimensions, and installation ratio.



Calculations

Inputting ARB properties to determine contribution to front roll-rate:

Dimensional inputs setting up model

Desired Roll Gradient	1.2	deg	/	g*	*Refers to g-force, not grams			
Vehicle CoG Along Height	343	mm		=	13.504	in		10
Driver CoG Along Height	20.5	in		=	20.500	in		10
Sprung CoG Along Height	414.419	mm						
Front Roll Axis Height	65.3	mm		=	2.571	in		11
Front Roll Lever Length	13.745	in						12
Rear Roll Axis Height	95.5	mm		=	3.760	in		11
Rear Roll Lever Length	12.556	in						12

Highlight condition indicating whether an ARB is needed.

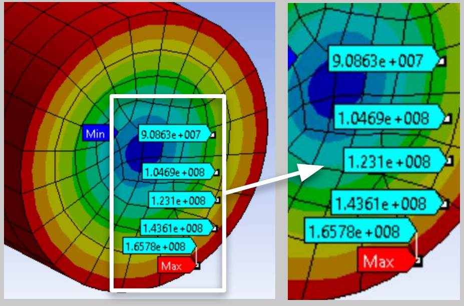
CALCULATED SUSPENSION VALUES - ROLL							13
Front Wheel Disp w/ Roll Gradient	0.5476	in					13
Front Desired Roll Rate	304.595	N	*	m	/	deg	13
Front Spring Roll Rate	234.617	N	*	m	/	deg	
Front Remaining Roll Rate (ARB)	69.978	N		m	/	deg	
							13
Rear Wheel Disp w/ Roll Gradient	0.532	in					13
Rear Desired Roll Rate	330.264	N	*	m	/	deg	13
Rear Spring Roll Rate	361.100	N	*	m	/	deg	
Rear Remaining Roll Rate	-30.836	N		m		deg	

Need Front ARB?	YES
Need Rear ARB?	NO

FRONT ANTI-ROLL BAR							
Attach. Disp Over Total Wheel Disp	0.6555	in					
Installation Ratio	0.214						
Torsion Bar Length	14.625	in					
Cantilever Length	4.6145	in					
Torsion Bar Outer Diameter	0.75	in					
Torsion Bar Inner Diameter	0	in					
Polar Moment of Inertia	0.000	m	^4				
Wheel Disp @1deg roll	0.456	in					
Angle of Twist @1deg roll	2.424	deg					
Required Min. Twist Rate	11.918	N	*	m	/	deg	14
Total Torque on Bar	28.884	N	*	m			14
Min. Desired Shear Modulus	19.619	GN	/	m	^2		
Shear Modulus of Chosen Material	26	GN	/	m	^2		
Twist Rate	15.794	N	*	m	/	deg	
ARB Roll Rate	92.740	N	*	m	/	deg	15
Total Front Roll Rate	327.356	N	*	m	/	deg	15
Excess Roll Rate	22.761	N	*	m	/	deg	

Stress Analysis Simulation

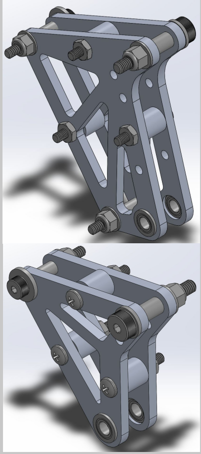
Load Case: 129 Nm	Solver: Ansys	Material: Aluminum 6061-T6
Max Stress = 166 MPa		Safety Factor Yield= 1.62



Simulation was set to "worst case-scenario," AKA the maximum possible torque given the shock lengths. Since this case cannot be attained, the lower safety factor is less concerning.

Bellcranks

Design Overview



To transfer wheel load to the shocks, we needed to design bellcranks strong enough to withstand the following loads::

- From the control arm attached to wheel upright
- From the spring and shock
- Bearing load from the pivot attachment
- From the anti-roll bar

All above loads were set in a simulation, with the control arm loads at peak values calculated by the lap simulator. The profiles were iteratively thinned to reduce mass, with a safety factor ≥2.0 in mind.

Each bellcrank consists of two waterjetted 6061-T6 3/16" plates separated by spacers, with shoulder bolts at each load point to prevent fastener shear and reduce rotational friction.

Front Bellcrank Assembly Weight: 0.29 lbs.

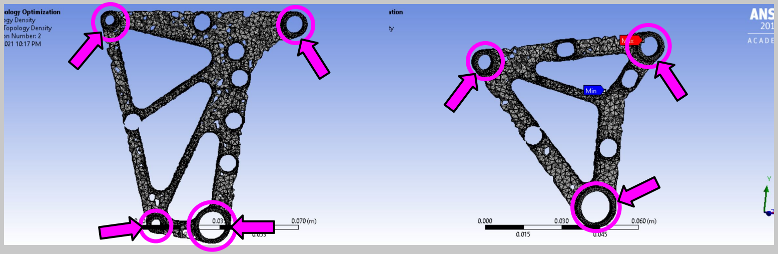
Rear Bellcrank Assembly Weight: 0.18 lbs.

Analysis Set-Up and Results

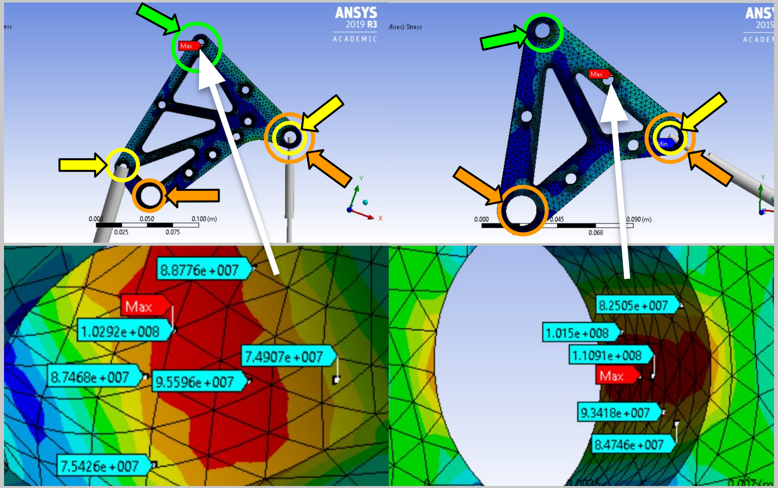
Solver: Ansys	Material: Aluminum 6061-T6	
	Front:	Rear:
Connections	2 Spring Body-to-Ground	1 Spring Body-to-Ground
Loads/Supports	2 Cylindrical, 1 Bearing Load	
Topology Optimization Settings	Minimize compliance, 40% mass, 4 exclusion regions	Minimize compliance, 40% mass, 3 exclusion regions
Max Stress	**103 MPa**	**115 MPa**
Safety Factor Yield	**2.62**	**2.35**
Safety Factor Ultimate	**3.01**	**2.70**
Mass Reduction from Original	**42.0%**	**42.4%**

Pictures

Front: Rear:



Notice how topology optimization only thinned out the profile without changing its shape.

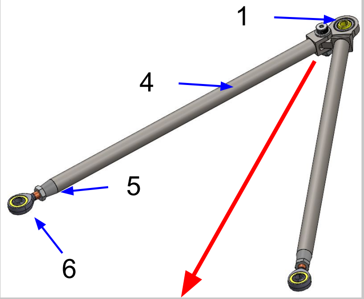


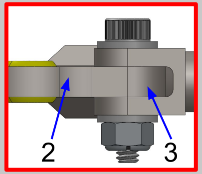
CALTECH RACING

Control Arms

Design Overview

1. COM-4T spherical bearing
2. 4130 A-arm housing
3. Adjustable lug joint
4. 4130 tube, ½” OD, .058” wall thickness
5. 4130 welded tube ends for rod end threads
6. AM-4T rod ends for geometry adjustment



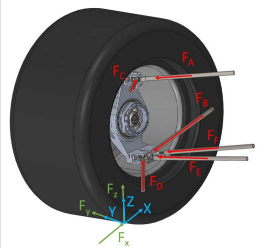


Adjustable lug joint:

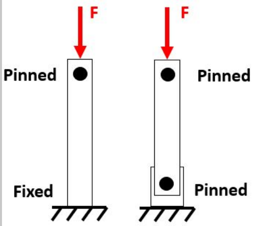
- Allows larger caster adjustments during testing
- Reduces rod end off-axis loading
- Relaxes welding tolerances

Calculations

Estimate axial force in each arm using 3D statics



Estimate critical buckling load for arms using Euler/Johnson solutions



Safety factor on critical buckling load

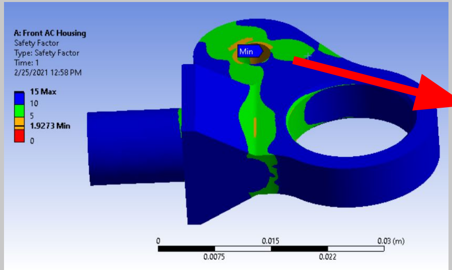
Arm		Buckling Safety Factor
Front	A	6.1
	B	3.5
	C	36.6
	D	2.1
	E	6.4
	F	46.2
Rear	A	14.5
	B	6.4
	C	6.9
	D	3.5
	E	5.9
	F	2.8

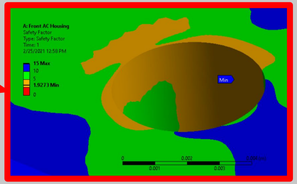
Minimum buckling safety factor = 2.1 for front D control arm

- Expected during hard braking, cornering

Stress Analysis - Housing

Load Case: Braking, Hard Cornering	Solver: Ansys	Material: Linear	Geometry: Linear
Safety Factor Yield: 1.9	Safety Factor Ultimate: 2.6	Temperature: 22°C	Notes: Front top housing





Bearing stress to yield is lowest safety factor failure mode

Manufacturing

Control arm welding fixture:

- Control arms mounted on rigid board
- Spherical bearings and rod ends are offset by spacers
- Different mounting holes on the same board to save material and cost

Fasteners and Bearings

Requirements and Selection

Requirement	Reason
Spherical bearing ultimate radial safety factor > 2	Must handle expected loads with margin
Bearing misalignment > 2 x (design misalignment)	Provides margin for geometry adjustment
PTFE lubrication on bearings	Simplifies maintenance of spherical bearings
Bolt shear safety factor > 2	Mounting bolts must be strong in shear

With these requirements, the following hardware is chosen:

Component	Details
Rod End	Aurora AM-4T, ¼” bore, ¼-28 thread, max 16 degree misalignment
Spherical Bearing	Aurora COM-4T, ¼” bore, max 13.5 degree misalignment
Bearing Bolts	Shoulder bolt, ¼” shoulder diameter, 10-24 thread

Rod End Loads

		Ultimate Radial Safety Factors
Front	A	4.7
	B	8.6
	C	6.3
	D	2.3
	E	3.1
	F	4.5
Rear	A	11.3
	B	6.6
	C	6.6
	D	2.6
	E	5.9
	F	3.1

Control arm forces are used to estimate loads into each rod end

- Axial loads are compared to bearing’s ultimate radial load
- Selected AM-4T, with ultimate radial load of 5260 lbs

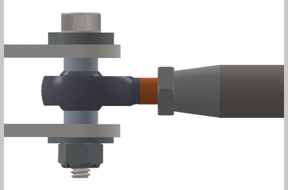
Margins are currently large

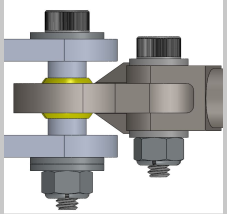
- Data will be used to validate control arm loads
- If loads are lower than expected, can use lighter and cheaper components

Lug Joints

Rod end joint

- Attached to frame





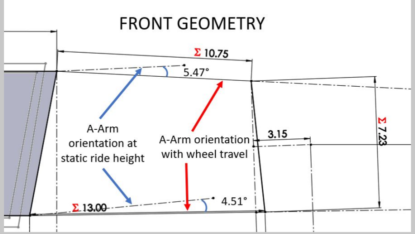
Spherical bearing joint

- Attached to upright

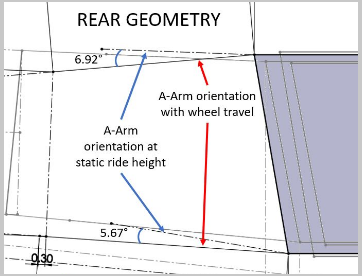
		Max shear stress [MPa]	Shear Ultimate Safety Factor
Front	A	65.4	8.9
	B	47.6	12.2
	C	46.9	12.4
	D	129.3	4.5
	E	99.6	5.8
	F	65.8	8.8
Rear	A	32.6	17.7
	B	58.1	10.0
	C	55.8	10.4
	D	140.8	4.1
	E	62.6	9.3
	F	118.4	4.9

Rod ends and spherical bearings captured in double shear

- Spacers used to increase allowed misalignment
- Double shear meets critical fastener rule (V.3.1.5)
- Double shear increases shear safety factor



Region	Max rod end misalignment (degrees)	Max spherical bearing misalignment (degrees)
Front	5.47	3.28
Rear	6.92	6.15



Selection of AM-4T and COM-4T provide misalignment margins of 131% and 120%, respectively

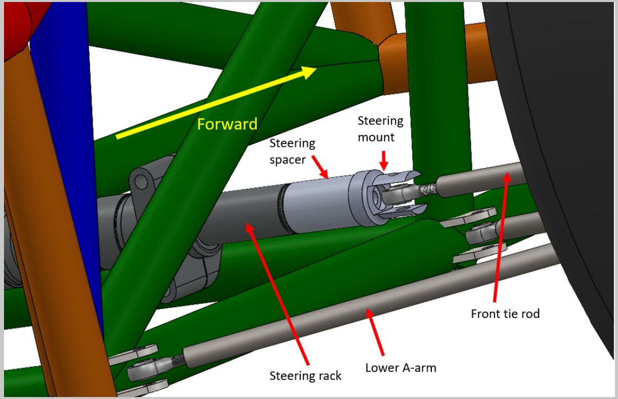
Large margins provide room for geometry adjustment

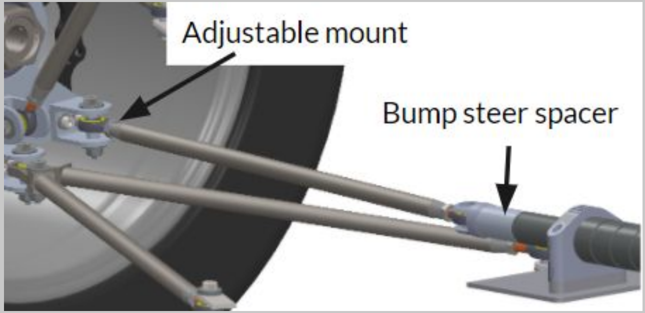
Integration

Steering

Steering spacer used to set initial bump steer

- Washers and spacers added or removed to adjust front bump steer





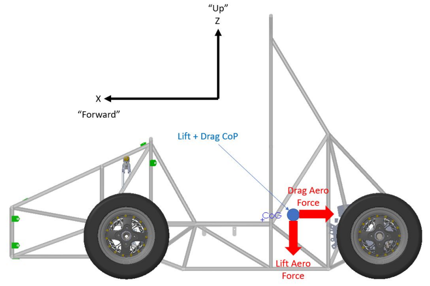
Adjustable mount provides:

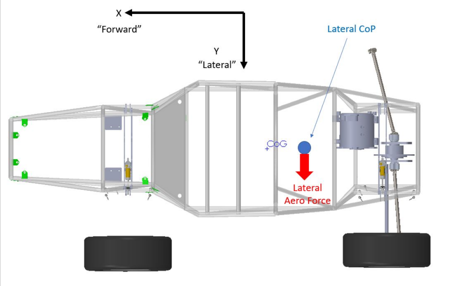
- Front toe adjustment (shims)
- Small changes to steer arm length (slots)

Aerodynamics

Define aerodynamic system requirements for handling:

- Center of lift 2 inches behind CG: limit “bite” on front tires at high speed
- Center of drag placed to balance pitch moment
- Lateral center of pressure 2 inches behind CG: cornering stability

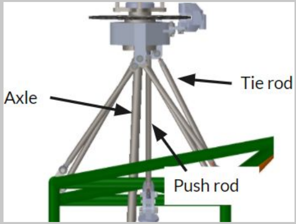




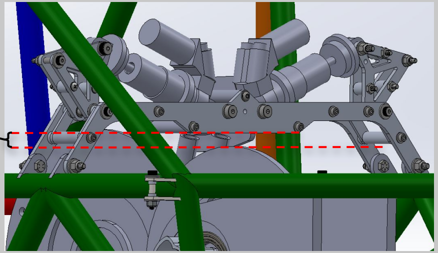
Drivetrain

Rear damper assembly moved forward to avoid rear axle

- Keeps push rod in-plane with damper assembly
- Also raises tie rod buckling safety factor



Rear damper assembly raised to avoid rear differential



Testing

Testing Overview

To validate the suspension design, several quantities must be measured

- Structural Loads
- Geometry Kinematics and Compliance Validation
- Damper Movement

Several other parameters can be measured in tests to inform small adjustments

- Infrared thermometer to measure tire temperature distribution
- Pressure gauge to measure tire pressure changes

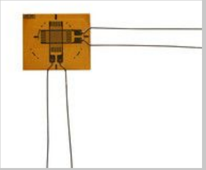
Structural Loads

Need to validate control arm load model

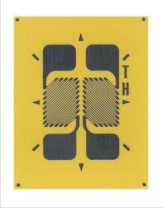
- T rosette to measure control arm strain
- Assume axial stresses
- In future designs, could work with lower safety factors (loads would be characterized)

Need to validate ARB section sizing

- Torque gauge to monitor ARB torsion
- Can also suggest new ARB settings



https://micro-measurements.com/pca/detail/030wt



https://micro-measurements.com/pca/detail/062th

T rosette	Torque gauge
Quantity: 24	Quantity: 2
Sample Frequency: 100 Hz	Sample Frequency: 100 Hz
Resistance: 350Ω	Resistance: 350Ω

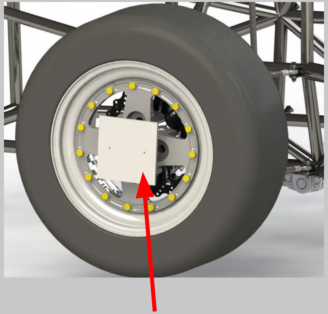
Geometry Kinematics and Compliance

Track kinematic travel of each quarter-car assembly using dial indicators and wheel reference plate

- Measures change in angles with wheel travel
- Checks setup with manufacturing inaccuracies
- In response, can modify setup using shims and rod ends

Make similar measurements to estimate quarter-car compliance

- Replace dampers with steel tubes
- Apply static weight to one wheel
- Observe deflection at wheel
- Compare to expected values to improve model or motivate design change



Wheel reference plate

Damper Movement

Can verify motion ratio and ride frequency calculations using linear potentiometers.

- Use MR calculator to equate potentiometer displacement to bellcrank displacement
- Verify MR calculations by manually moving wheel
- Verify ride frequencies in front and rear using 10 data points in period of oscillation (about 20 Hz)
- Check roll stiffness using cornering data

Adjust springs and ARBs according to testing outcomes.

- Increase/decrease ride frequency => Change spring for a higher/lower spring rate
- Increase/decrease roll stiffness => Shorten/lengthen ARB lever arm or change spring for higher/lower spring rate