Cornell Electric Vehicles – Mechanical

Drivetrain

Fall 2022 Technical Report

Jamie Cashman
Nila Narayan
Ashley Miller
Alyssa Yoon
Adrienne Yoon
Evan Leong

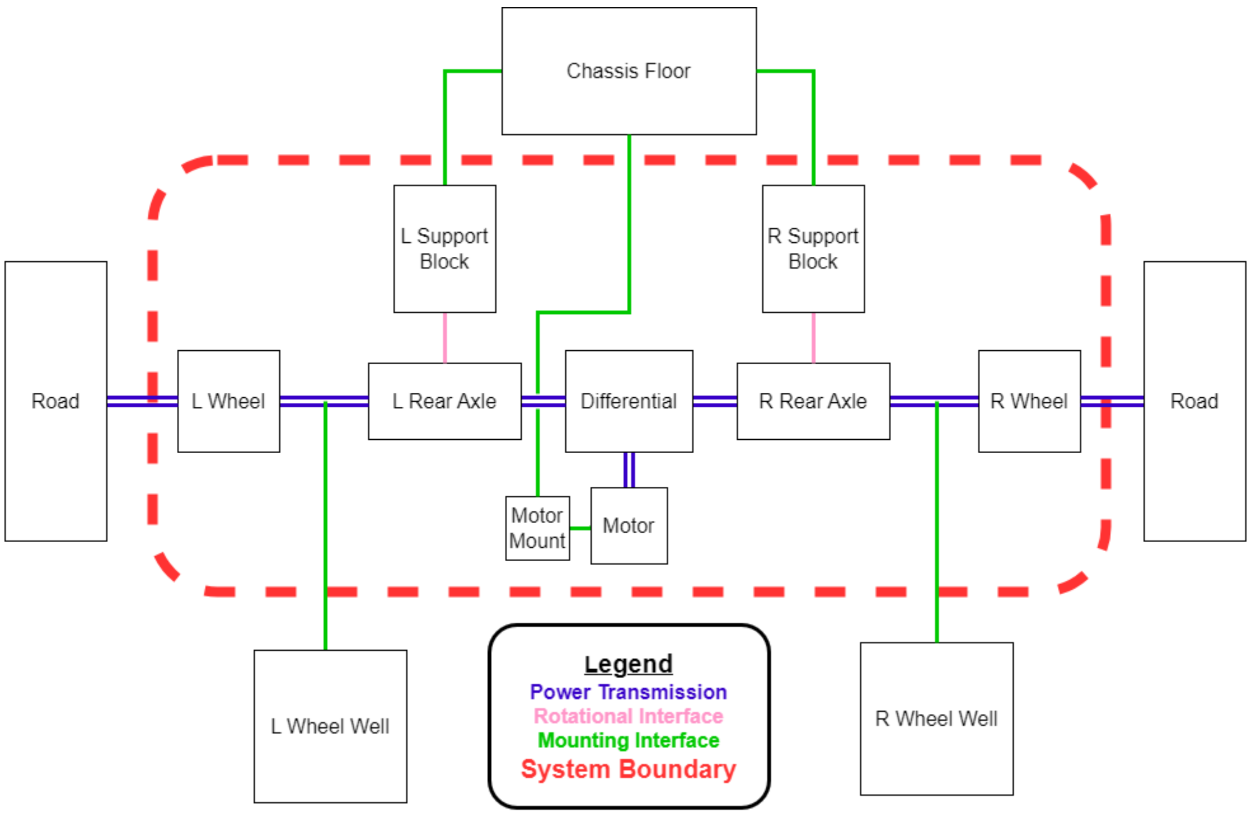
December 10, 2022

Table of Contents

Summary

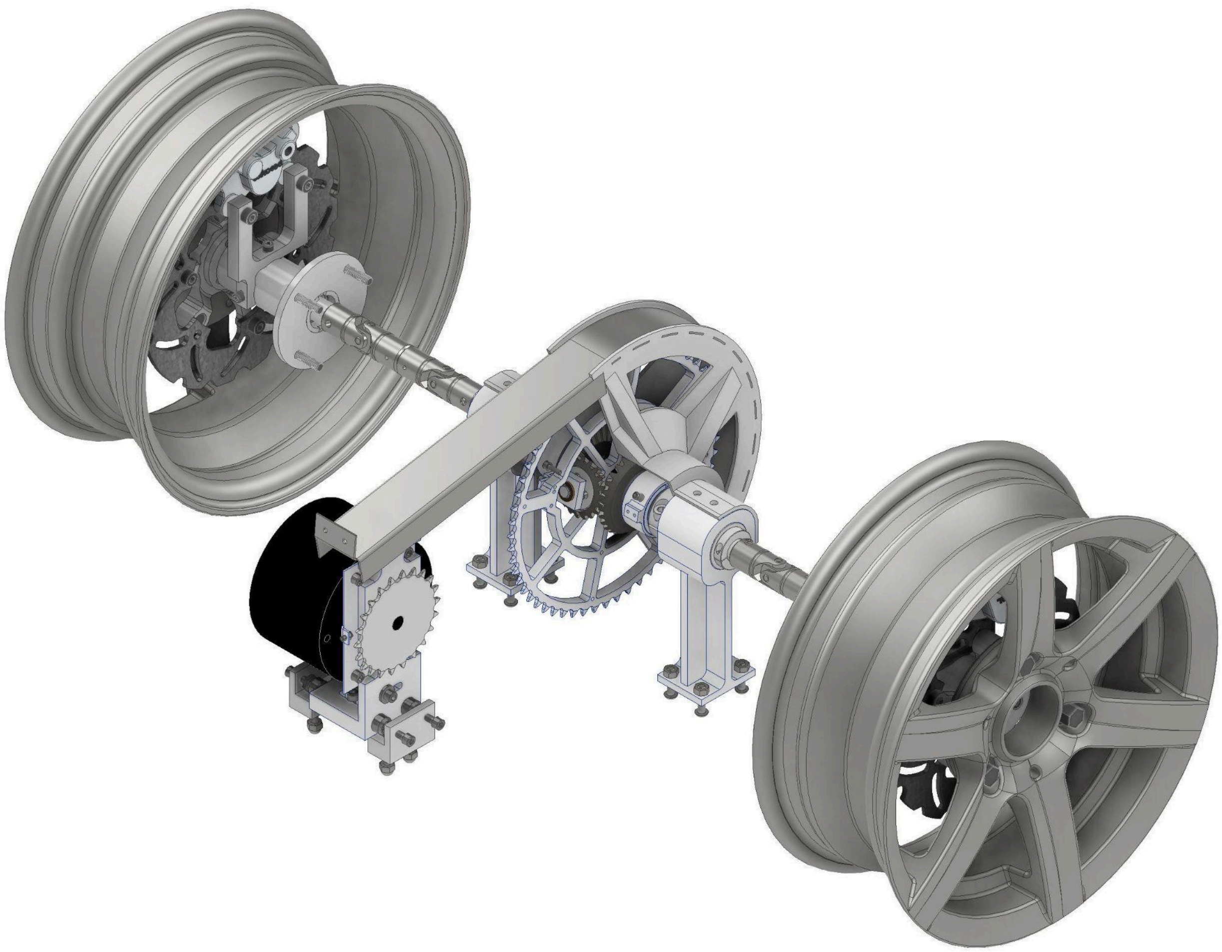
System Description

The drivetrain is the system within the car which provides power to the rear wheels, allowing it to be driven forward. The main components of the system are contained within the motor assembly, differential assembly, axle assemblies (L and R), and rear hub assemblies (L and R).



Drivetrain System Boundary Diagram

The motor mount, axle support blocks, and outer wheel hubs are rigidly mounted to the chassis of the vehicle. Power is transmitted from the motor, powered by 24V and 20A continuous current, to the differential via a sprocket and chain assembly with a gear ratio of 2.63. Torque is then transmitted via the axles from the differential to the wheels, which provides force at the road to propel the car forward. The open differential design allows the wheels to turn at different speeds as prescribed by the radius of a turn to prevent the wheels from slipping along the road surface and decreasing efficiency. One-way bearings at the axle-differential interfaces also allow for coasting while the motor is idle.



Full Drivetrain Assembly (Chain not pictured)

Terminology

Differential: A system that averages rotational speed (and torque) across two output shafts. It is used to prevent slipping of wheels during turning by allowing wheels to turn at different rates.

Spider gear: The intermediate gears within the differential that average rotational motion between the two output shafts. These are connected directly to the big sprocket.

Side gear: The gears that are driven by the spider gears and directly connect to the output shafts of the differential. These are connected to the axles.

Double U-joint: A joint that connects two shafts and allows rotational motion to be transmitted between them with an angular offset between the axes. A double U-joint in this system allows for easier alignment of axle support blocks and rear hubs.

RBM: Stands for- Rear Brake Mount. The machined part that mounts the brake calipers to the outer hubs, which hold the calipers in position over the brake rotors.

Research and Requirements

Background Research

The following resources can be used to better understand many general concepts that govern the drivetrain.

This video explains the concept of a differential.

This presentation by former Drivetrain lead, Alex Zhu, discusses different gear types.

This document lists all possible motors provided by Koford Motor. We will be using the 129H42A model.

Design Requirements

Because much of the design was already done last semester, the main requirements to consider this semester were related to the motor. The Electrical team provided the following constraints in regards to the motor:

- Must operate on 24V
- Can operate at a maximum of 20A continuous current
- Preferred: purchase from Koford Motors, the same manufacturer of the motor on the Cuckoo Caravan

Design Requirements Derived from Rules

There are several rules prescribed by the Shell Eco Marathon which impact the design of the Drivetrain to varying degrees. See References section for a link to the rulebook itself.

According to Article 27, the bulkhead must separate the vehicle’s propulsion system from the driver for safety purposes. This rule allows the Drivetrain to be designed with the bulkhead as a mounting part for some parts, such as the chain guard.

According to Article 49, the tire/rim assembly must have a width of greater than or equal to 80 mm from tire sidewall to tire sidewall. This rule impacts tire choice, and prevents using bike tires as was done for the Prototype Class in 2020-2022. However, this choice would have been made regardless of this rule being in place, as such small tires would never have supported the weight of the car.

According to Article 51, the vehicle must have a four-brake hydraulic braking system, where either the front and rear brakes are controlled separately, or an X-pattern is followed (front left and rear right, front right and rear left). A single master cylinder can be used as long as it has dual circuits, and a parking brake which can supply at least 50N of force must be designed. These parameters dictated the choice of master cylinder, pedal design, and parking brake design, all of which was done in conjunction with Steering.

The design of the Drivetrain is most directly impacted by Article 34 of the 2023 Shell Eco-Marathon rules, which states that for battery electric vehicles:

- All vehicle propulsion must be achieved only through the friction between the wheels and the road.
- Urban Concept ICE vehicles are required to have idling capabilities. This means the vehicle must be able to remain stationary while the engine is running.
- The car's electric starter cannot provide any forward propulsion (Article 64b, which is referenced by Article 34)
- Guards for transmission chains and/or belts are mandatory. This is why the aforementioned chain guard was designed.

Key Parameters

The drivetrain can be split into several key components/subsystems, as briefly described below:

Motor/Motor Mount:

- The motor, as described above, is limited to 24V and 20A continuous current
- The motor must also meet specific power requirements in order to finish the race. The calculations to determine this will be described in depth later in the report.
- The motor mount should have six degrees of freedom, for easy adjustment during assembly. However it should also have ways to secure it once the positioning is decided upon.

Differential:

- The big sprocket has a maximum diameter, prescribed by the distance between the rear axle and the floor of the chassis.
- The little sprocket must have a minimum of 12 teeth, and reliably clamp onto the motor shaft.
- The big and small sprocket should have a ratio of 2.63. The calculations to support this are described in the Analysis section of this report
- The differential, when assembled, must have as little slop/as high tolerance as can be reasonably manufactured.

Axle Assembly:

- Must be able to support the axles, and thus the differential.
- When assembling, must be as close to in line with another as possible, so as to have the drivetrain aligned in a straight line.

Rear Hubs:

- The positioning and mounting of the rear hubs is parameterized by the dimensions of the chassis.
- The measurements and tolerances for the machined parts in the rear hub assembly(inner hub, outer hub, RBM) are all determined by the dimensions of the off-the shelf/ pre-bought parts of the wheel hubs, brake calipers and brake rotor.

Brakes:

- The brakes are most tightly constrained by the Steering system. The master cylinder and pedal must be located in such a way that they are accessible to the driver without putting the steering assembly, specifically the tie rods, at risk of misalignment or damage.

Potential Designs

See the Spring 2021 Technical Report for an in-depth description of potential designs for the drivetrain.

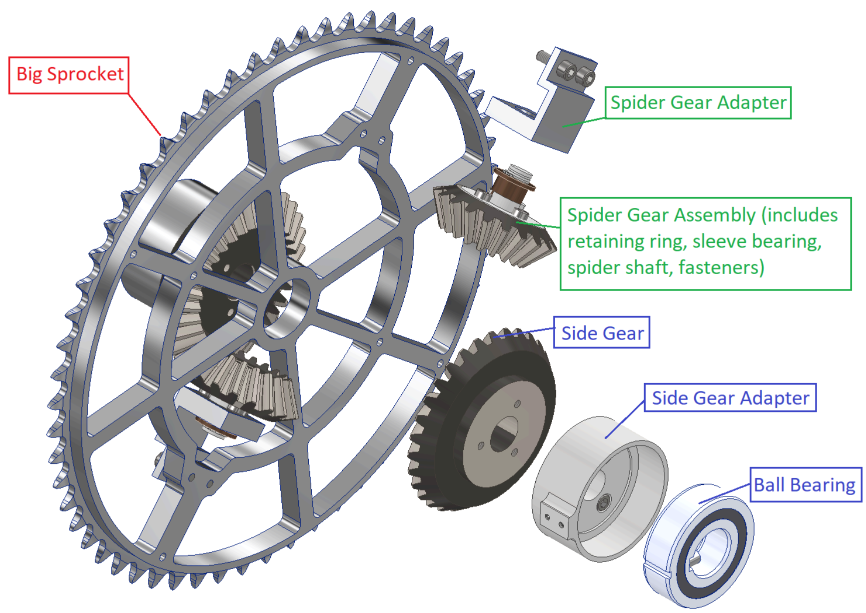
Selected Design and Justification

See the Spring 2021 Technical Report for a more in-depth justification behind certain large-scale design choices. Beyond this, the Detailed CAD section below also goes in depth as to why these designs were chosen, as well as changes made to previous design choices from 2021 and Spring 2022..

Detailed Design (CAD)

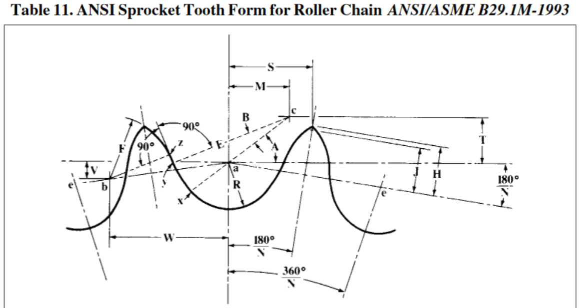
Differential

The primary purpose of the differential is to allow for turning without slip. This can be conceptualized by considering a track; the runners in the innermost and outermost lanes will turn different amounts, and they start at different positions along the track. Similarly, when a car turns, the left and right wheels must spin at a different rate as prescribed by the radius of the turn itself, otherwise slippage will occur at the wheels. The differential in the urban concept follows an open differential design, consisting of four bevel gears. Each side gear interfaces with a different axle, which allows the two wheels to rotate at different rates.



Sprockets

Table 11. ANSI Sprocket Tooth Form for Roller Chain ***ANSI/ASME B29.1M-1993***



Seating Curve Data—Inches

P	D_r	Min. R	Min. D_s	D_s Tol.[a]	P	D_r	Min. R	Min. D_s	D_s Tol.[a]
0.250	0.130	0.0670	0.134	0.0055	1.250	0.750	0.3785	0.757	0.0070
0.375	0.200	0.1020	0.204	0.0055	1.500	0.875	0.4410	0.882	0.0075
0.500	0.306	0.1585	0.317	0.0060	1.750	1.000	0.5040	1.008	0.0080
0.500	0.312	0.1585	0.317	0.0060	2.000	1.125	0.5670	1.134	0.0085
0.625	0.400	0.2025	0.405	0.0060	2.250	1.406	0.7080	1.416	0.0090
0.750	0.469	0.2370	0.474	0.0065	2.500	1.562	0.7870	1.573	0.0095
1.000	0.625	0.3155	0.631	0.0070	3.000	1.875	0.9435	1.887	0.0105

The big and little sprocket transmit power from the motor to the rest of the system via a chain connection, with a gear ratio of 2.63. The small sprocket has 22 teeth, and the big sprocket has 58 teeth, with the tooth following ANSI/ASME standards, shown on the left. The dimensions of the teeth are ultimately most dependent on the pitch, number of teeth, and nominal roller diameter from which all other parameters are derived. Definitions of all parameters, as well as a labeled diagram, are shown to the left.

[a] Plus tolerance only.

$P = \text{pitch } (ae)$

$N = \text{number of teeth} \quad D_r = \text{nominal roller diameter}$

$D_s = \text{seating curve diameter} = 1.005\,D_r + 0.003 \text{ (in inches)}$

$R = \tfrac{1}{2} D_s$ (D_s has only plus tolerance)

$A = 35° + (60° \div N) \quad B = 18° - (56° \div N) \quad ac = 0.8\,D_r$

$M = 0.8\,D_r \cos(35° + (60° \div N))$

$T = 0.8\,D_r \sin(35° + (60° \div N))$

$E = 1.3025\,D_r + 0.0015 \text{ (in inches)}$

$Chord\ xy = (2.605\,D_r + 0.003)\sin(9° - (28° \div N)) \text{ (in inches)}$

$yz = D_r[1.4\sin(17° - (64° \div N)) - 0.8\sin(18° - (56° \div N))]$

Length of a line between a and $b = 1.4\,D_r$

$W = 1.4\,D_r \cos(180° \div N)$; $V = 1.4\,D_r \sin(180° \div N)$

$F = D_r[0.8\cos(18° - (56° \div N)) + 1.4\cos(17° - (64° \div N)) - 1.3025] - 0.0015 \text{ inch}$

$H = \sqrt{F^2 - (1.4D_r - 0.5P)^2}$

$S = 0.5\,P\cos(180° \div N) + H\sin(180° \div N)$

Approximate O.D. of sprocket when J is $0.3\,P = P[0.6 + \cot(180° \div N)]$

O.D. of sprocket when tooth is pointed $+ P\cot(180° \div N) + \cos(180° \div N)(D_s - D_r) + 2H$

Pressure angle for new chain $= xab = 35° - (120° \div N)$

Minimum pressure angle $= xab - B = 17° - (64° \div N)$;

Average pressure angle $= 26° - (92° \div N)$

The small sprocket will interface with the motor via a clamping mechanism., and two M8 set screws.

The big sprocket interfaces with the differential via the spider and side gear assemblies, as well as the chain guard, all of which are outlined below. Both sprockets will be manufactured via a mix of milling and CNCing.

A one-way bearing inside of the big-sprocket will both align the part radially and only allow the transmission of torque in one direction, supporting the team's "burn and coast" strategy and maintaining efficiency.

Spider Gear Assembly

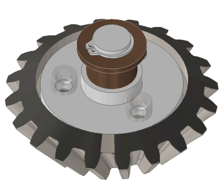
The Spider Gear Assembly consists of the spider (bevel) gears, which connects to the spider shaft and an oil-embedded sleeve bearing, to be purchased from McMaster. This oil-embedded bearing will serve to minimize losses due to heat and friction. The spider shaft and spider gears are attached to each other via three bolts, and threaded holes in the spider shaft. The spider gears interface with the big sprocket via an adapter piece and two M4 screws.

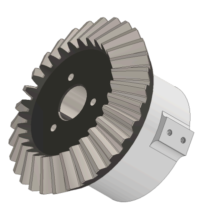
Side Gear Assembly

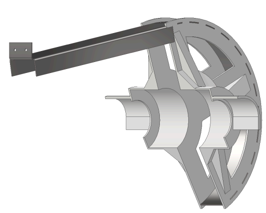
The Side Gear Assembly consists of the side (bevel) gears, and the side gear adapters, which contain a ball bearing, set screws, and a key to interface with the axles. It is worth noting that all four bevel gears (two spider gears, two side gears) are purchased from the shelf, as even with CNC machining such complex geometry is difficult (near impossible with our experience and equipment) to achieve with the required tolerances.

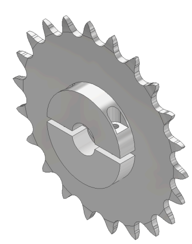
Chain Guard

The chain guard, to be made of a combination of PLA and sheet metal, interfaces with the large sprocket and the axle support blocks. It also interfaces directly with the chassis via two holes in the bulkhead. It is required by the shell eco-marathon.



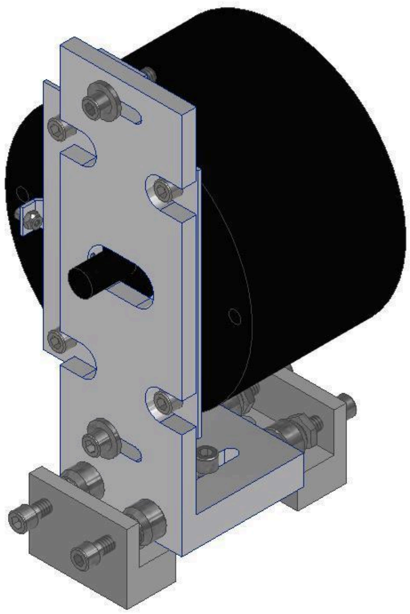






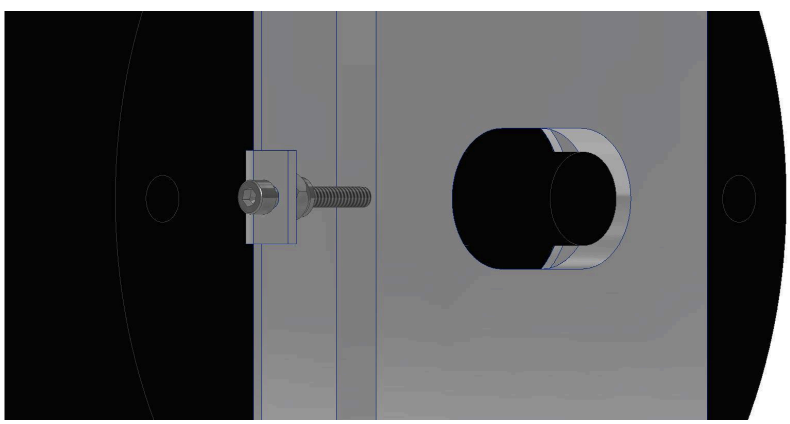
Pictured, left to right: spider gear assembly, side gear assembly, chain guard, and little sprocket

Motor Mount



The motor mount is designed to allow for six degrees of freedom. This makes chain alignment and tensioner much easier during assembly, as the little sprocket will be able to pivot and move back and forth relative to the big sprocket.

There are three main components of the motor mount: the L plate, the flat plate, and the mounting plate. The mounting plate is attached to the chassis floor via four bolts and is intended to be a permanent structure. The L plate connects to the mounting plate via two bolts in two slots. These slots allow for back-and-forth movement as well as axial rotation. The four toggle pads pressed to the L plate assist in fine adjustment of the L plate position. The flat plate connects the motor to the rest of the assembly. It is made out of steel sheet metal and will need to be bent and cut multiple times. The motor is attached via four bolts, and the flat plate is attached to the L plate via two bolts, one at the top and bottom of the flat plate.



The single 4-40 bolt in the assembly is secured to a flange on the flat plate through a clearance hole and lock nut. The other end of the bolt goes through a threaded hole in the L plate. This allows for fine back-and-forth adjustment of the flat plate and motor relative to the L plate. This provides up to slightly more than 14 mm in distance to allow for chain tensioning.

Axle Assembly

Short Axle

The short axle interfaces with the double U-joint and side gear and goes through the support block.

Long Axle

The long axle has an additional purpose that the short axle does not: it locates the differential. Like the short axle, the long axle interfaces with the double U-joint and the side gear. It interfaces with these parts using a coiled spring pin. However, the long axle additionally interfaces with the big sprocket with a bearing between the two. The long axle is supported by the support block.

Axle-Double u joint interfacing

Originally, both long and short axles had keyways for set screws for the axle-double u joint interface. The keyways were changed into a through hole through the shaft, as this was changed to a coiled spring pin assembly. The advantage of using coiled spring pins over set screws is that torque transmission is more efficient because a coiled spring pin is forced into a hole that is slightly smaller than the pin whereas a set screw may slip. It will also be easier in assembling/disassembling than set screws as long as the holes are made with higher tolerance, as it can easily be inserted and removed with a hammer.

The coiled spring pins (stainless steel of breaking strength 4300 lbs, ¼'' diameter, 1'' length) were ordered off the shelf. The length of the pins is the same as the length of the hole in the double-u joint so that the pins would not stick out of the hole in case it interferes with other systems.

- Max shear force on axle: 371.676 lbf
    - Max torque at wheel = max motor torque * sprocket ratio
    - 185.838 lbf*in = 70.5 lbf*in * 2.636
    - Max shear force = max torque at wheel / axle radius
    - 371.676 Ibf = 185.838 lbf*in / 0.5 in
- 4300 lbs breaking strength
    - FOS: 11.6

Pins with a factor of safety of 11.6 were chosen even though it is an overshoot, as pins with reasonably high breaking strengths only came in bulks of hundreds.

Support Blocks

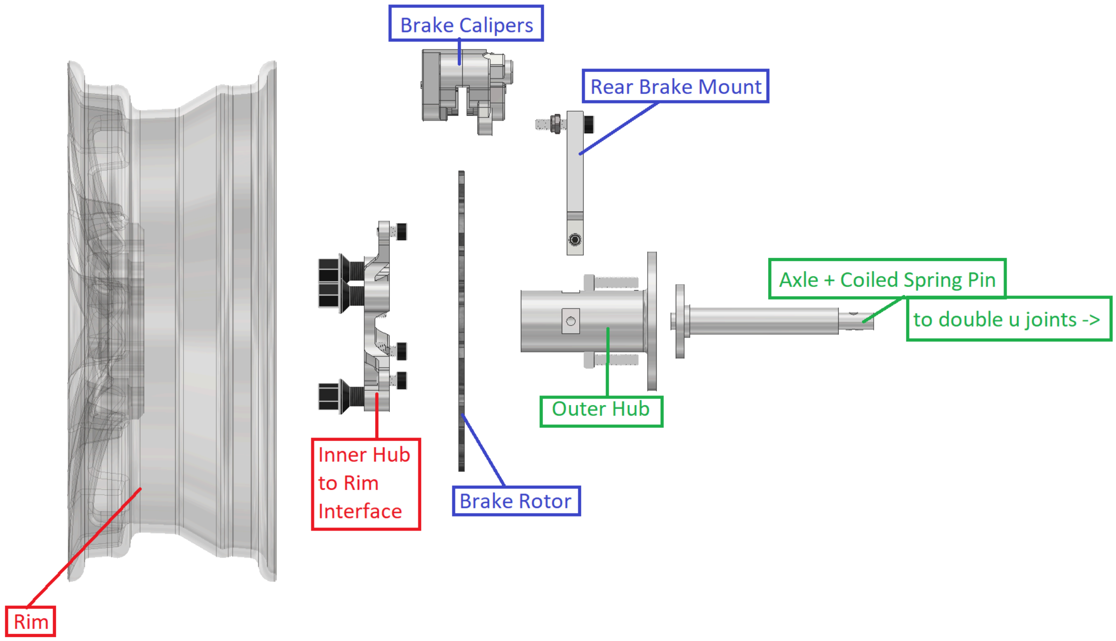
The support blocks have not been changed significantly this semester.

The inner diameter of the bores of the support blocks has been changed from press-fit to transition-fit, as a tight fit is no longer needed due to the shaft collars that constrain the axles from moving outwards. The diameter was increased slightly so that it has the exact same diameter as the outer diameter of the bearing that is to be pushed in the bore.

Shaft Collars

Last semester, calculations were done regarding the shaft collar selection. The drivetrain subteam considered several different sizes and had settled on an 8mm and 18mm that would be placed on either side of both of the supports. However, this semester the team ran into the issue that the practical width of the chassis was slightly smaller than the predicted width, meaning that the width of the drivetrain had to be reduced. So, it was decided to switch the 18mm shaft collar to another 8mm one on each support block so that the drivetrain would fit in the actual chassis. Calculations confirmed that this choice was okay since the safety factor for the holding force of the 8mm shaft collars is 2.7.

Rear Tire and Hub Assembly



Inner Hub

The Inner Hub connects the double u-joints to the rim/hub interface, which is subsequently connected to the wheel rims and the rest of the wheel assembly. This hub also transmits torque provided by the drivetrain, which is transferred to the double u-joints, to the rims of the wheel. The inner hub is held in place in the outer hub through a spacer and 2 ball bearings. The justification behind this is to allow the inner hub to rotate freely/ independently in the stationary outer hub.

Outer Hub

The Outer Hub is what constrains the Inner Hub axially and mounts the rear wheel assembly to the chassis at either end. It is also used as an attachment point for the Rear Brake mount.

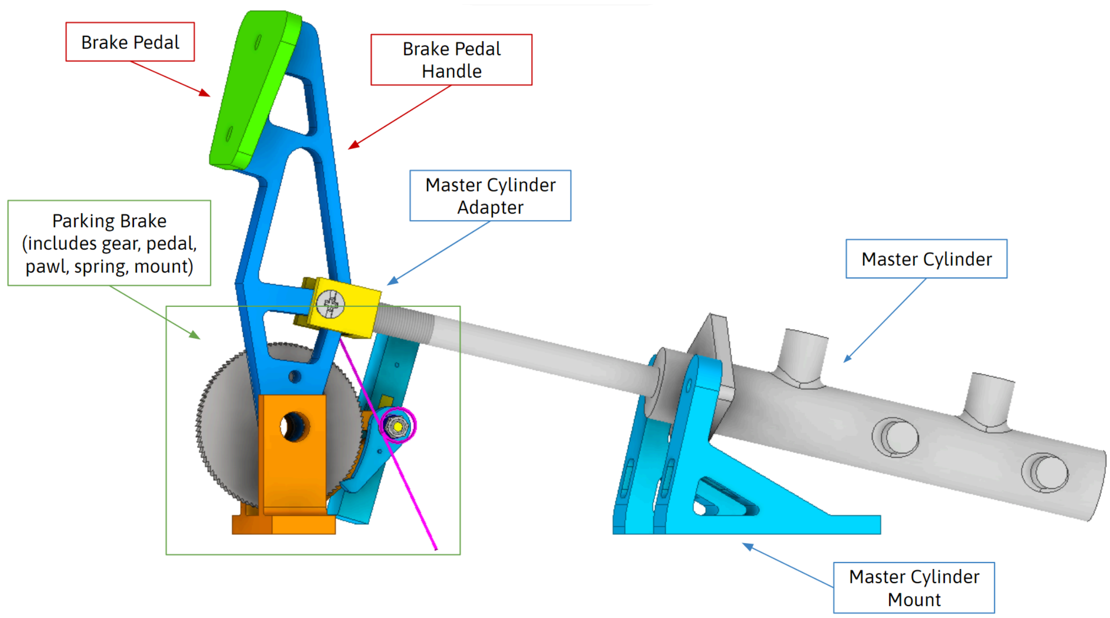
Brake Mount

The Rear Brake Mount is connected to the brake calipers and is what holds the calipers in place so that they can clamp over the brake rotor and provide a resistive force to the motion of the wheels.

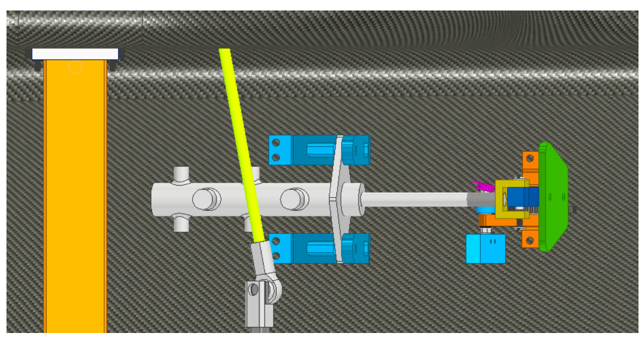
Rim/Hub Interface

The Rim/Hub interface connects the inner hubs to the rims themselves. It is also bolted to the brake rotor. The justification behind this being so that if the brake rotors are clamped down on by the calipers, the stopping power from the calipers can be transferred to the rotor, then to the rims in order to brake the car.

Brake and Parking Brake



Brake



The brake system is interfaced with chassis (base plate, side part) and steering (tie rods). In order for the master cylinder to fit under the tie rods, the master cylinder mounts have been lowered and angled down instead of being parallel with the ground. The master cylinder mounts and the pedal mount will be bolted to the base plate, and the brake fluid reservoirs will be attached to the wheel hub of the side part via velcro or bolts. There is a tube connecting the master cylinder and brake fluid reservoirs, which had to be taken into consideration when deciding where to mount the reservoirs so that they would not interfere with the tie rod.

Another factor that had to be considered when choosing the location of the brake system was that the base plate is curved at the head of the vehicle which would complicate the mounting process. Thus, the system was placed closer to the middle of the vehicle which is only slightly curved so that where the mounts will be bolted can be easily leveled using material such as epoxy without interfering with brake assembly.

Additionally, this placement only leaves around 30 inches between the brake pedal and the bulkhead which is a cramped and uncomfortable space for the legs, even for someone short. Thus, the driver's seat will be raised with padding for comfort in the legs and also to raise their eye level in order to be able to see out the windshield.

Brake lines

The routing of brake lines (as well as the brake fluid tubes and lines for sensors etc) are to be decided towards the end of manufacturing when all additional information required (such as how many lines will be needed and where they will be placed) are known.

Parking brake

The parking brake is interfaced with the brake pedal and uses a pawl and ratchet assembly to prevent the brake pedal from moving when parked. When it is engaged, the pawl connected to the parking brake pedal by a spring locks the ratchet which is connected to the brake pedal so that it will no longer rotate.

The problem with the current design is that the engagement of the parking brake pedal relies completely on friction due to rubber washers between the pedal and the pedal mount. When engaged, the spring connecting the parking brake pedal to the pawl is compressed and the only force that opposes this and holds the pedal in position is friction, which is not very reliable. Some proposed ideas to this problem were to:

I. Implement a mechanical latching mechanism like ones in a ball-pointed knock pen
II. Get rid of the spring completely and use a simple lock operated by hand
III. Use a magnetic latching system

Using idea I would make it easier for the driver to use the parking brake because all that needs to be done is to be pushed further down and released to disengage. The problem with this idea is that it may be hard to manufacture because the part is complex for its small size.

Idea II is much simpler, but the problem with this method is that it is not very appealing in terms of accessibility. The driver would have to reach the lock by hand every time trying to push or pull it out, and there needs to be a secure space for the lock to be placed while driving when it is not in use.

Idea III is also quite simple as magnetic latches are available on McMaster, but it is hard to test if it is strong enough to hold the pedal down and the parts may not be in the dimensions needed.

For now, using idea I seems the least problematic if it is possible to modify a pre-existing CAD of a pen latching mechanism so that it fits the parking brake assembly. However, it is still not guaranteed that this method may work and therefore alternative ideas will be needed in case it does not work out.

Analysis

Hand Calculations

Several calculations of load cases on the drivetrain were done. Some calculations, such as load cases for the support blocks and calculations that led to the selection of the shaft collars, were done last year. Please see previous FDR's and technical reports for more information on calculations done prior to Fall 2022. Some important calculations completed this semester include motor and gear ratio selection calculations, drivetrain efficiency estimate, and load cases for sprockets, coiled spring pins, axles, and the inner and outer hubs. The purpose of these calculations was to gain insight when it came to part selection, material selection, and design validation. Completing more thorough and well-documented calculations was a major goal of the drivetrain subsystem this semester.

The first major calculations were done to select the motor and determine the desired gear ratio so the sprocket tooth number could be chosen. These calculations were done by referencing the Shell rules (length of lap, necessary time to finish race, etc.) to determine the required torque and RPM of the motor with a given gear ratio. The calculations were done using this spreadsheet. After determining the required torque and RPM at the wheels, the desired gear ratio for the goal torque and RPM was found and then those ratios were averaged to select the ideal gear ratio. Next, since number of teeth is an integer, several combinations were tested and an ideal number of teeth was found: 22 and 58 teeth for little and big sprockets, respectively. Using the data provided by Koford, efficiencies of various motors were graphed at the desired torque and RPM and the motor was selected. The 129H42A motor will be used.

Another complicated calculation done was an efficiency estimate of the drivetrain. Since the description of how to calculate efficiency is extensive and complicated, for more information on the details of efficiency analysis see the spreadsheet above as well as this information sheet. The total efficiency of the drivetrain is the product of the efficiencies of each component. The components it is possible to estimate the theoretical efficiency of include the chain and sprocket system, all of the bearings in the drivetrain system, the gears making up the differential, the motor, and the wheels. The efficiency calculated here is the theoretical efficiency. The experimental efficiency would be determined after the car is assembled. Most likely, the largest efficiency losses we cannot account for in theory will be due to imperfect alignment when assembling the drivetrain. The final results found are:

Component	**Efficiency (using worst case if a range is given)**	**Source**
Motor	0.8989	Motor, Gear Ratio, and Sprocket Calculations - FA22 Updated
Sprockets / Chain	0.97-.99	Efficiency Estimation of Roller Chain Power Transmission System
Bevel Gears (4)	.98-.99 per gear	Bevel Gears | KHK Gear Manufacturer

Axles +U-joints + Coiled Spring Pins	1	Drivetrain Efficiency Estimate Info
Bearings (9)	.99 per bearing	Drivetrain losses (efficiency) – x-engineer.org
Wheels (4)	.8267837197 per wheel	Motor, Gear Ratio, and Sprocket Calculations - FA22 Updated
Theoretical Total Efficiency: $\eta_{drivetrain} = \prod_{components} \eta_{component}$	**.3433**	

For context on the efficiency result, most internal combustion cars convert about 12-30% of the energy from the engine to power at the wheels and most electric vehicles convert over

Another important load case that was analyzed was for the coiled spring pins. It was found that the maximum shear force on the axle would be 371.676 lbf. This was calculated using the maximum torque at the wheel divided by the axle radius. The coiled spring pins we will be using stainless steel, ¼''diameter, 1'' length spring pins with 4300 lbf breaking force. The safety factor is 11.6 so these coiled spring pins will be very effective and safe.

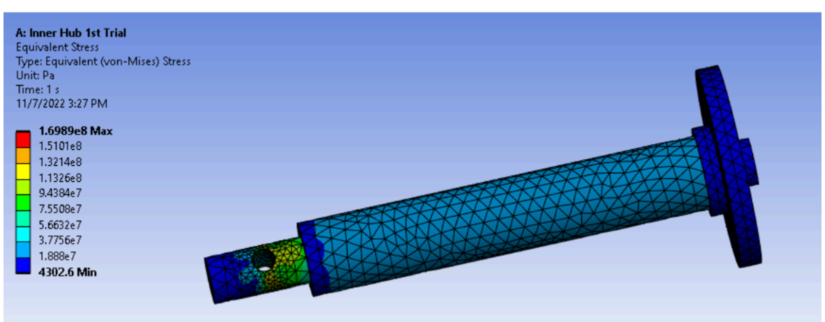
ANSYS

Rear Hubs ANSYS

Ansys simulations from last year were updated with additional considerations like bump and cornering load cases. Mesh refinements were included on areas prone to failure on the inner and outer hubs, and all the considerations and loads taken into account have been noted below in the ANSYS section as well as being present in the updated rear hub ANSYS folder shared on GrabCAD.

Inner Hub:

Revised Ansys Fall 2022 (Mesh Shown In Photo)



Min. Factor of Safety (6061 Aluminum: 1.57)

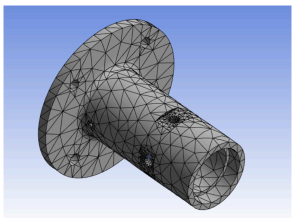
Boundary Conditions: Cylindrical supports placed at interface between inner hub and wheel, replicating the bolts connecting the inner hub to the hub/rim interface.

Load Cases: 21Nm torque placed at coiled spring pin interface replicating a stalled wheel, and a motor applying max torque at the u-joint end, shearing the coiled spring pin.

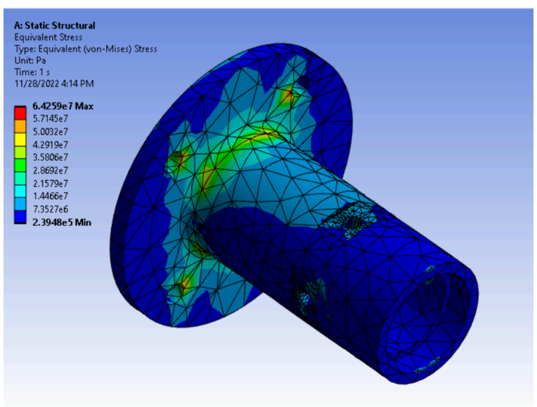
Changes: Updated for new max motor torque and sprocket ratio.

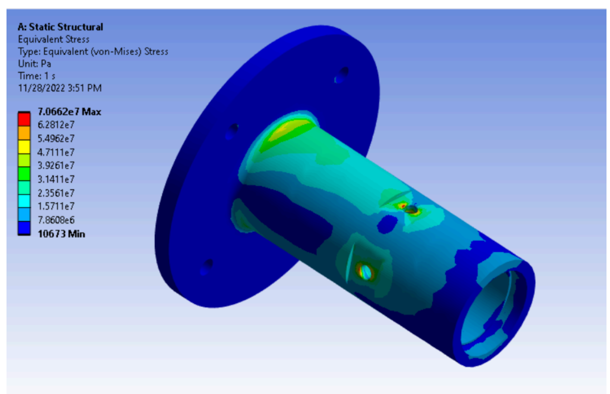
Outer Hub:

Revised Mesh



Revised Ansys Fall 2022 | **(Old) Ansys Spring 2022**





Min. FOS (6061 Aluminum): 4.037 | **Min. FOS(6061 Aluminum): 3.67**

Boundary Conditions: Cylindrical supports placed at the Outer Hub to Chassis attachment point, replicating the 4 mounting bolts.

Load Cases: 1200N force accounting for the weight of the chassis on the hub and its mounting bolts. 21Nm moment from wheel torque applied by the rear brake mount on the outer hub during a braking scenario.

Changes:

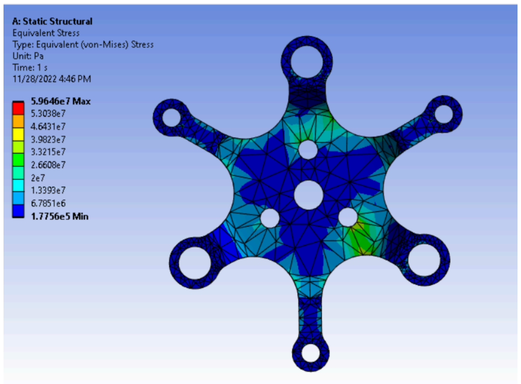
-Additional Mesh Refinements to RBM Holes

-Recalculated moment caused by max motor torque and new sprocket ratio

-Accounts for decreased Outer Hub Length from the Spring semester.

Rim to Hub Interface:

Revised ANSYS Fall 2022 (Mesh in Photo)



Min. F.O.S = 4.35

Boundary Conditions: Cylindrical supports placed at the 3 bolt holes where the wheel rim is mounted to the interface.

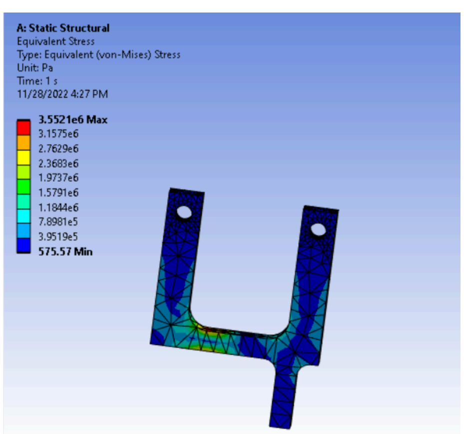
Load Cases: 1200N weight of car downward at mounting holes + Cornering moment and forces applied at the center of the rim interface that would work to bend the rim interface.

Additional Changes:

-Refined mesh around supports and connections to brake rotor. (areas of interest)

Rear Brake Mounts:

Revised Ansys Fall 2022



Min. F.O.S= 15

Boundary Conditions: Fixed cylindrical supports at Outer Hub mounting holes.

Load Case: Tangential Force applied by brake calipers on the two caliper mounting holes due to the reaction force caused by a stalling wheel case and max motor torque applied to the rotor.

Sprocket Ansys

When analyzing the sprockets, two main cases were considered: back tension due to the chain, which decays over time as the chain makes less contact with each consecutive tooth; and secondly, a buckling load case wherein pre-tensioning of the chain due to torque at the motor was considered.

In order to do Ansys, load cases were calculated for several parts that will be manufactured. When analyzing the loads on the sprockets, back tension force was defined as

$T_{tooth} = T_{chain} * \left(\frac{sin(\phi)}{sin(\phi+2\beta)}\right)^{k-1}$ where

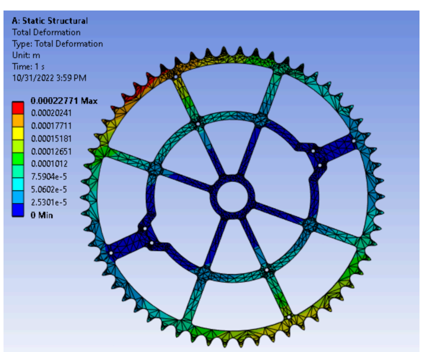
$\phi = sprocket\ minimum\ pressure\ angle = 17 - \frac{64}{number\ of\ teeth}$ in degrees,

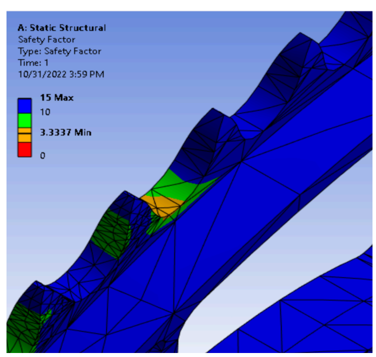
$2\beta = sprocket\ tooth\ angle = \frac{360}{number\ of\ teeth}$ in degrees, and k=number of engaged teeth.

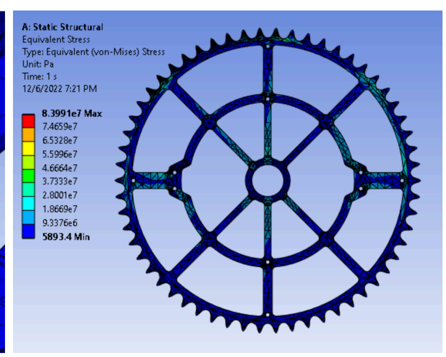
The force always becomes negligible after the first ten engaged teeth. Both the big and little sprocket have these tension forces on the first ten engaged teeth. These tooth forces were applied as loads in Ansys as well as a buckling load, which is related to the torque caused by pre-tensioning in the chain. The little sprocket additionally has the clamping mechanism on it that allows it to attach to the motor shaft. The clamping force is equal to the bolt rating times the cross sectional area.

In both cases, meshes were refined at the impacted teeth in order to judge wear over time. For the big sprocket, the mesh was also refined at the mounting points to the spider gear assembly.

Big Sprocket:





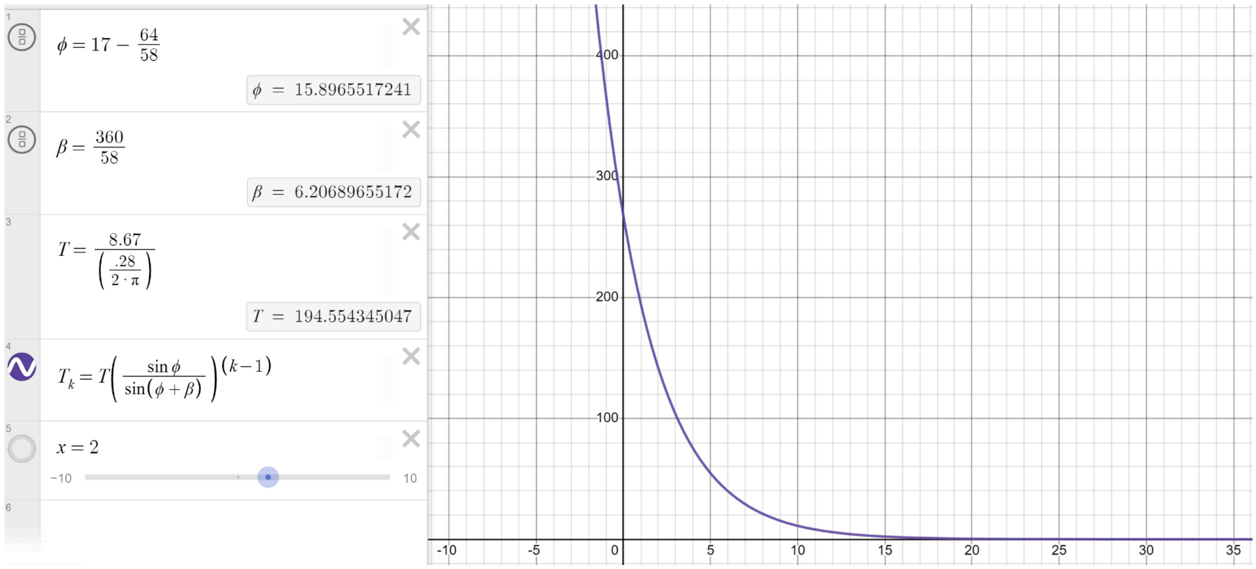


Min factor of safety: 3.334

Max deflection: 2.277e-4

Analysis of the big sprocket shows that, as would be expected, the point of greatest stress is the tooth which, at any given moment, is making the most contact with the chain. Even then, the safety factor is greater than 3, implying that wear is not of dire concern.

Below shows the plot for tooth tension as a function of *N=58* teeth for the big sprocket, though the curve is roughly the same for the small sprocket as well (but to a smaller scale)

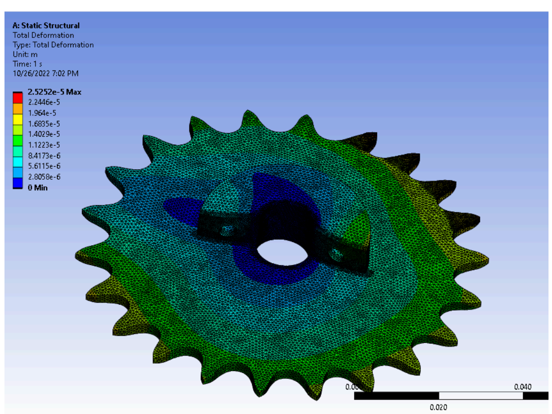


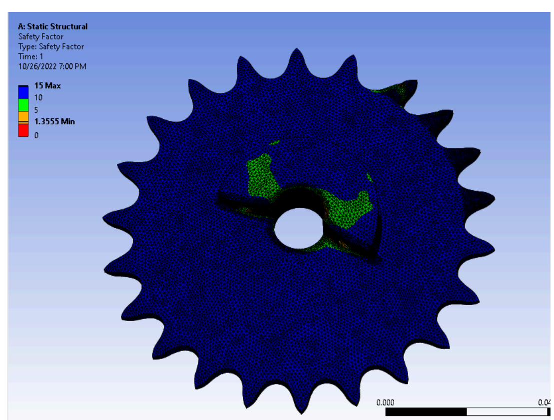
Little Sprocket:

Similar to the big sprocket, the small sprocket experiences the highest stress at the tooth most tightly engaged with the chain. Due to the clamping force, safety factor for this mechanism is slightly less than that of the big sprocket. However, it will still still withstand the applied forces with considerable safety.

Min factor of safety: 1.356

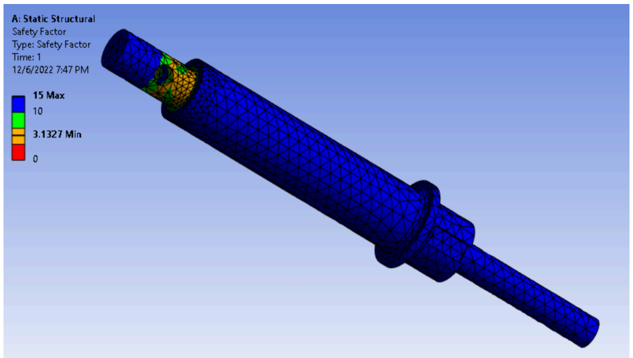
Max deflection: 2.57e-5

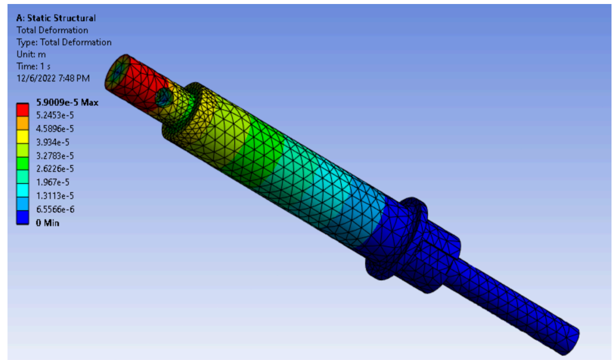


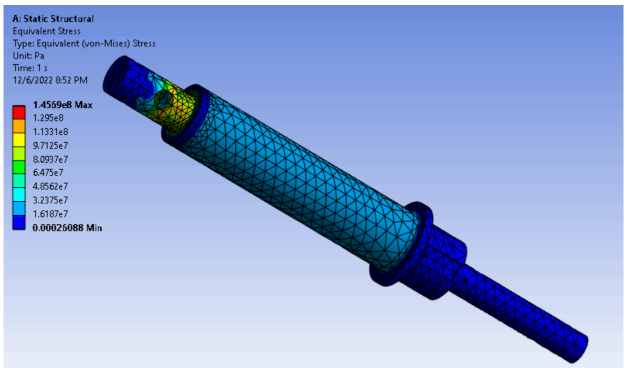


Axles Ansys

Long Axle







To calculate the worst case load on the axles, the maximum stall torque was applied to the axle as a moment transferred via the spring pin hole. This ended up being a force of approximately 21 N*m. For aluminum 6061, the safety factor was below 2. For steel, the safety factor was 2.8. For aluminum 7075, the safety factor was 3.1327. Since aluminum 7075 is lighter, it was chosen to be the material for the axles despite it being more expensive, because of how crucial the axles are to the structural integrity of the drivetrain.

Short Axle → same loads and results as the long axle

Brake Ansys

Master Cylinder Mount:

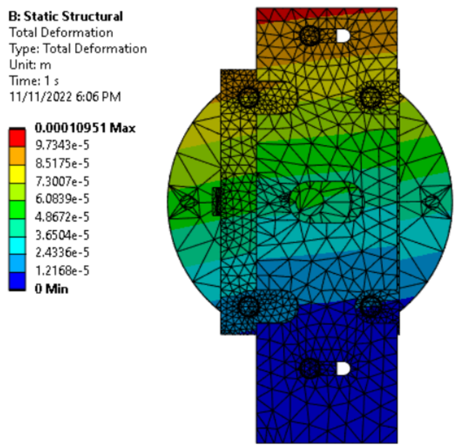
Changes made include a reduction in height and slanting the vertical part of the mount.

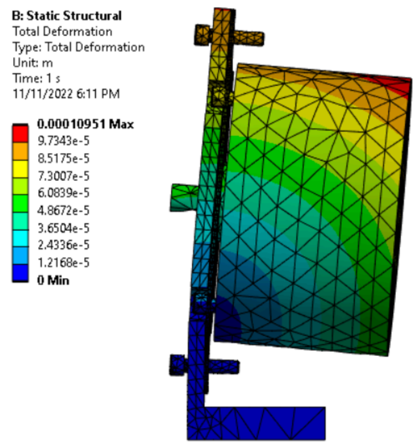
Boundary Conditions: Fixed supports are placed at the bottom holes where the mount is to be bolted to the base plate.

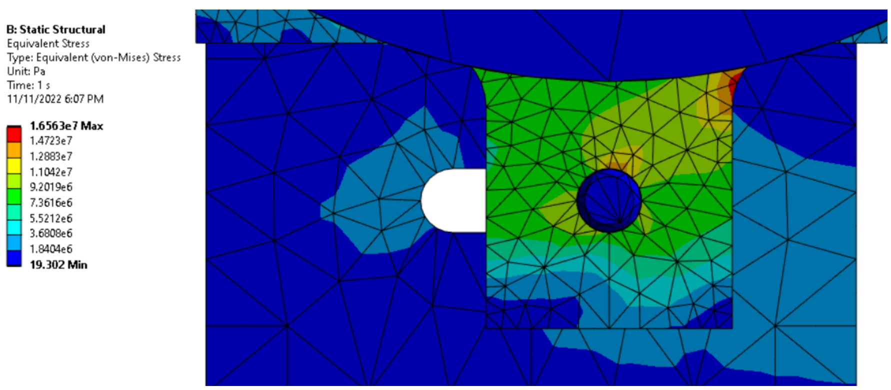
Load cases: For a sitting person with a solid backrest the maximum horizontal force acting on a pedal is typically 1000 N. 1000 N load is applied to the hole where the master cylinder is to be bolted.

Min. Safety Factor: 3.0967

Motor Mount







Boundary Conditions: Fixed bottom of L plate. Frictional contact sets at bolts and between components.

Load Cases: Motor Torque of 2.5 Nm, roughly equivalent to max motor torque when accelerating. Motor weight 7 pounds.

Safety factor above 15 for all components.

Bill of Materials

Click [here]() for a link to the Drivetrain BOM.

Manufacturing Plan

Assembly Context	Part	Material	Qty	Operations	Special Tool/Notes
Differential	Big sprocket	Al 6061 T6	1	CNC	Need high tolerance on all differential parts
	Little sprocket	Al 6061 T6	1	CNC	
	Spider Shaft	Al 6061 T6	2	Mill, Lathe	
	Spider Adapter	Al 6061 T6	2	Mill	
	Side Gear Adapter	Al 6061 T6	2	CNC	
	Side Gear	Al 6061 T6	2	Mill (facing)	Modifying an off-the-shelf part
	Spider Gear	Al 6061 T6	2	Mill (facing)	Modifying an off-the-shelf part
Rear Hubs	Inner hub	Al 6061 T6	2	CNC	Started by Alex in SP22, need to redo CAM
	Outer hub		2	Lathe	
	Rear brake mount		2	Mill	
Rear Axles	Short Axle	Al 7075	1	Lathe, CNC	
	Long axle	Al 7075	1	Lathe, CNC	
Motor Mount	L plate	Al 6061	1	Bandsaw, Mill	
	Mountin g plate	Al 6061	1	Mill	

	Flat plate		1	Mill, Sheet Metal bending	Talk to Joe about bending. Tools available in Emerson. The flange must be much longer than what is currently shown in CAD to be bent properly. This piece will need to be milled, bent, and milled again.
Parking Brake	Parking Brake Pedal	Al 6061	1	Mill	
Brake	Brake Pedal	Al 6061	1	CNC	
	Master Cylinder Mounts	Al 6061	2	CNC	
	Brake Pedal Handle	Al 6061	1	CNC	
	Master Cylinder Adapter	Al 6061	1	Mill	

Assembly Plan

The most sensible approach to assembling the drivetrain is to go from outside in. The most important goal of assembly is to maintain alignment both in the width and length directions of the car: the holes through the chassis must be placed so that the drivetrain maintains a straight line across the chassis from one rear wheel to the other; the support blocks and motor mount must be mounted to the floor of the chassis such that the chain is in a straight line from the little to big sprocket. For the alignment in the width direction of the car, the goal is that any deviations in the placement of each hole through the side of the chassis will be small enough so that the double u-joints provide enough tolerance for the drivetrain to not lose efficiency due to misalignment. For the alignment between the sprockets, the goal is that any deviations are small enough so that the motor mount provides enough tolerance to adjust and achieve alignment easily.

First, the holes on the sides of the chassis will be made. Some kind of alignment jig in order to keep the left and right sides aligned with each other is to be designed in order to minimize the effects of tolerance stack-ups during assembly. The alignment of holes on the left and right side of the chassis, after all, is one of the most crucial components towards the success and efficiency of the drivetrain.

Beyond this, the differential should be assembled outside of the vehicle again via some sort of testing/alignment jig, rather than assembling it entirely in the car. This will also serve as a way to judge slop in the system, and work to minimize it before the entire assembly comes together.

Both of the aforementioned assembly jigs are still to be designed, but should be fairly simple. A simple assembly jig for the differential, for example, can consist of an 80-20 “cage” of sorts which simulates the axle support blocks and allows the big sprocket and spider/side gear assembly to be mounted on some kind of pseudo-axle.

Testing Plan

As discussed in the Assembly Plan section, the differential will first be assembled outside of the vehicle. This provides opportunity to determine if tolerancing on the side and spider gear assemblies are sufficient, before potentially increasing tolerance stackups when mounting the axles. Essentially, assembly jigging will double as rudimentary testing. However, no explicit data is to be measured as of yet.

Project Reflection and Future Work

This semester, a large emphasis was places on documentation. Many parts of the project were delayed due to a lack of documentation. For example, the motor calculations last semester were not very detailed and the estimations and assumptions were not very accurate. So, this semester the drivetrain subteam made an effort to try to document calculations with spreadsheets and keep everything up to date on the google drive. If we had done this earlier, things would not have had to be redone and the design could have been finalized earlier.

Additionally, many seemingly small problems were left to the end as a last priority. Nonetheless, the small problems accumulated and ended up causing a time crunch to get several things done including fixing the fasteners in the CAD.

This semester, the subsystem as a whole put a lot of faith into previously made designs, even before taking the steps to re-evaluate and re-validate said designs. While the old designs and words of advice were not illegitimate by any means, it meant that decisions were being blindly trusted, lessening everyone's overall understanding of the subsystem. As the semester progressed, and especially after Final Design Reviews, this issue was brought to light, and a greater emphasis was placed on one's ability to understand and support design decisions even if they were not made by oneself.

References

- Shell Eco Marathon 2023 Rulebook
- ANSI/ASME B29.1M Sprocket and Chain/Roller Standards