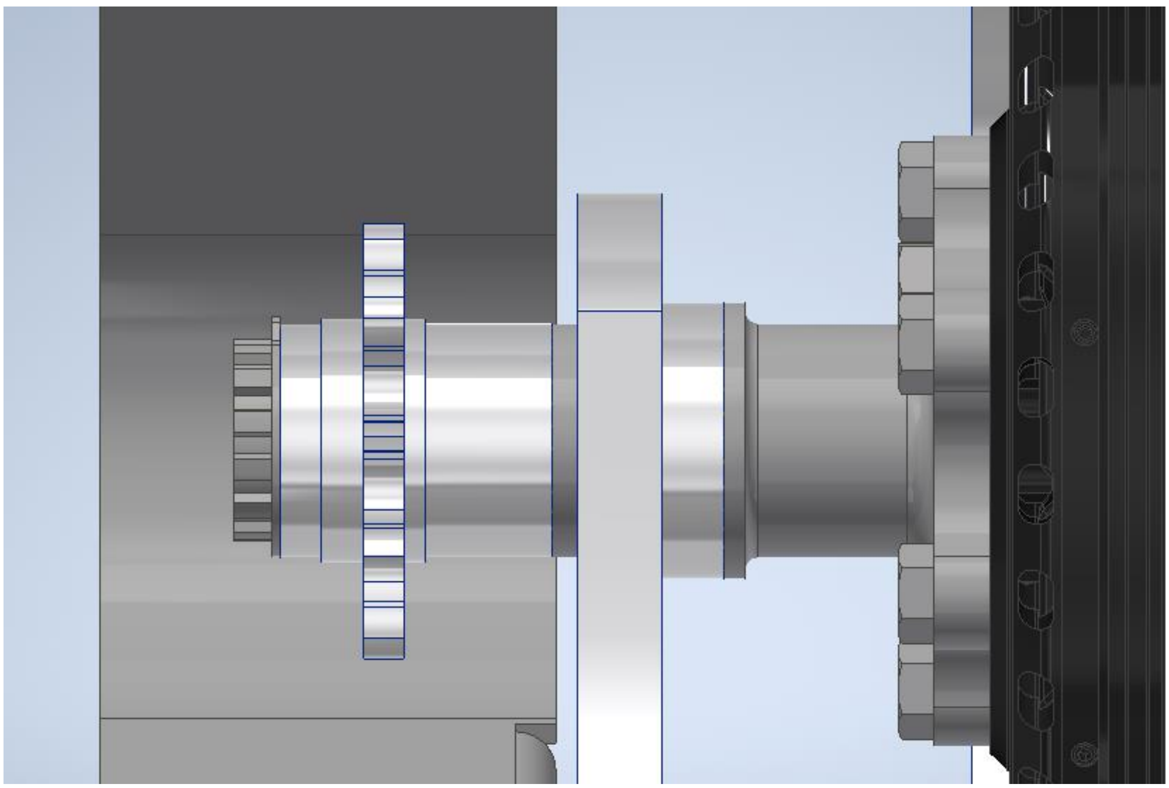
ARG23 Fall Technical Report

Due 11:59pm on Wednesday, December 14, 2022

Gearing and Sprockets
ARG23 Fall Technical Report
December 14, 2022
Jeff Wilcox – jtw88



Subteam: Drivetrain
Subteam Lead: Maddy Zackeo
Team: Chassis
Technical Team Lead: Diego Pierce

0. Introduction (10)

The sprocket and gearing within the drivetrain system is everything that directly transmits power from the motor to the differential. Although this is centered around the chain, this also includes the connection points of the chain (the sprockets) as well as the motor shaft and some but not al alignment components.

It is necessary because without my system, the wheels would not receive power. The performance is optimized by matching parameters at the wheels while also matching parameters for the motor, specifically accounting for coefficient of friction between the wheels and the ground, maximum torque of the motor, and other factors internal and external to the car.

Applicable Rules

- F.11.2.4: Non Crushable Items mounted behind the Rear Impact structure must not be able to come through the Rear Impact structure
- T.5.1: Any transmission and drivetrain may be used

1. Technical Overview (don't explain your design yet, and do not carry out any analytical methods) (20)

The most important value of this system is the FDR (final drive ratio), which is the ratio of the number of teeth in the rear sprocket to the number of teeth in the front sprocket. An FDR smaller than 1 will increase the angular velocity but decrease the torque while an FDR larger than 1 will increase the torque but decrease the angular velocity. To determine the necessary FDR for the vehicle, a static analysis of the rear wheels should be used to identify the necessary slipping torque as a function of the FDR as well as the translational acceleration, velocity, and displacement as a function of known variables. Extreme values (max and min) for the FDR can be found using the max torque and max angular velocity of the motor. This gives you a range of values for which the FDR is optimized for the chassis and motor parameters. To select a value within this range, using a fixed front sprocket (or for each front sprocket being considered), iterate through the rear sprocket tooth values that fit within the range of FDRs calculated before and determine which rear sprocket size has the optimal chain slack for the separation distance of the motor and the differential. Once the optimal sprocket pair(s) has/have been selected, you have your FDR(s).

For the Motor Shaft, the most critical part of the design is to ensure it can withstand the applied motor torque. To determine this value, I used the following shearing stress formula for splines[1]:

$$S_s = \frac{16TK_a}{\pi D_{re}^3 K_f} \text{ for a solid shaft}$$

$$S_s = \frac{16TD_{re}K_a}{\pi (D_{re}^4 - D_h^4) K_f} \text{ for a hollow shaft}$$

For the alignment of sprockets, spacers are needed that go over the motor shaft splines. The lengths of these can be found with simple geometry.

A rough chain length estimation can be found using geometry of the choice of sprockets as well as the center distance of the motor and the differential.

2. History of Past Designs (don't explain your design yet) (10)

- Begins with a broad team history of your part/system (i.e., the general evolution of the system, mentioning the unique approaches used in the past, with citation).
    - On ARG12, the rear sprocket used a 2 piece sprocket with an aluminum inside made from the aluminum Drexler blank with premade splines and a steel 4140 outside to withstand the forces on the teeth which were deemed too large for aluminum. On ARG13, the rear sprocket was one piece which was deemed easier to design and lighter. ARG14 ANSYS revealed that truss design was weaker with trusses directed in the direction of motion, against a hypothesis from the previous year. The ARG14 sprocket was Aluminum 7075-T6 and lighter than ARG13 with a one piece truss design. ARG15 switched to 4140 Steel but maintained a similar design. ARG16 introduced a sprocket with circular weight saving holes rather than a truss design due to fear of sprocket failure within the truss design. ARG17 followed this circular pattern which each car since has followed in using.
- Next, summarize the recent evolution of your part/system (which will pertain more to ARG23's interests). This should be a bit more specific, spending some amount of time to describe the large changes, and the results of how those parts/systems performed.
    - The last time the sprocket was taken on as a design part was on ARG20. Since then the car has changed quite a bit and we want to focus on optimizing the drivetrain that is on the car, not on the car 3 years ago. Significant changes include the change in number of teeth on the rear sprocket, a redesign of the motor shaft, and accommodation for two different motors depending on which motor ends up being used (there have been issues with Cornell's approval as well as with shipping. Overall, the recent evolution of the part is non-existent so this year was used to bring the sprocket and gearing back into the scope of the team's focus.
- Discuss why the component selection has changed. This should create a coherent recent evolution path. Include pictures/CAD images and citations.

Figure 1: ARG20 Motor Shaft

Above is the ARG20 motor shaft which was designed to remove stress concentrations, it is unclear how the sprockets were aligned in this setup. There is also no way of ensuring concentricity with the motor for this shaft. Below is the rear sprocket from ARG20. Reevaluating the final drive ratio for this car drove a desire for a redesigned sprocket.

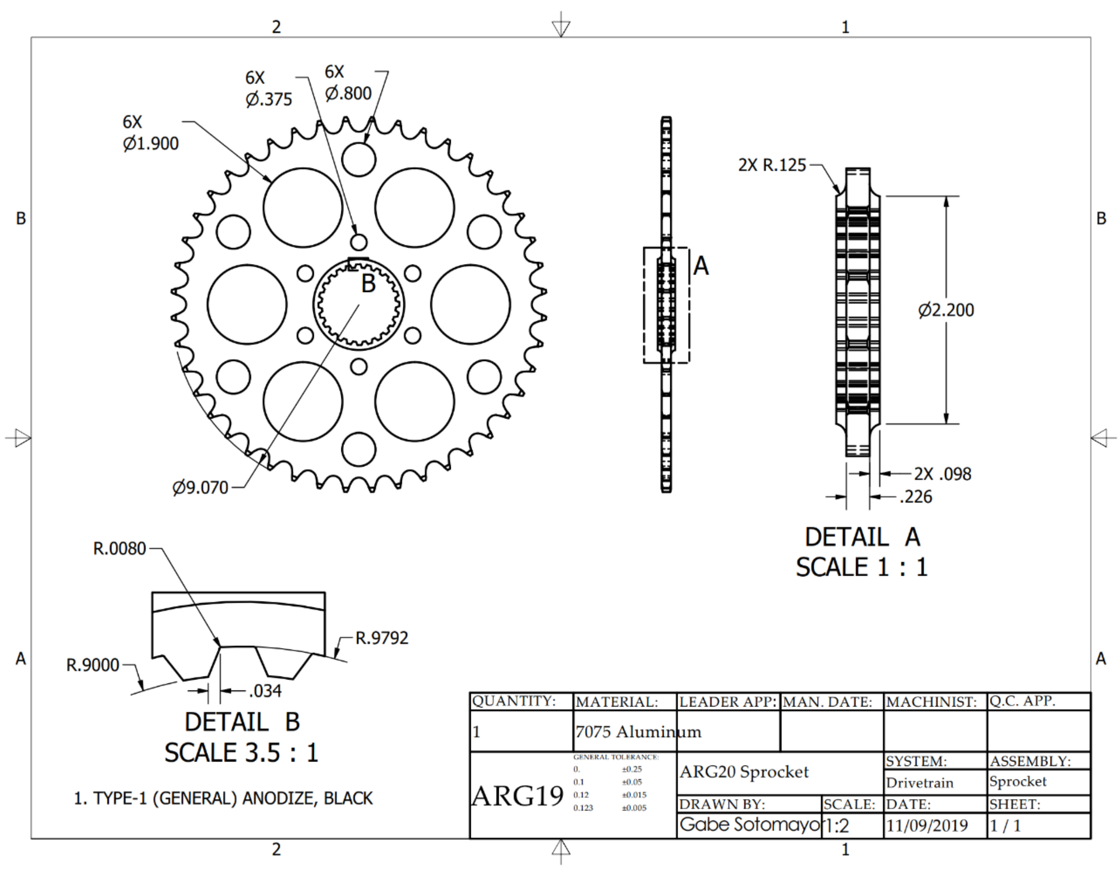


Figure 2: ARG20 Rear Sprocket

- Again, this is (hopefully) a compilation of the summer prep, and should also have been part of your PDR, but this should require a little more explanation/citation.
- **As for PDR, if your part is completely new for this year, mention other teams' design implementations.**
    -

3. Data (10)

- The data needed to set design criteria was the approximate total car mass, the center of mass height, the wheelbase, the coefficient of friction between the tires and the ground, the moment of inertia about the (imaginary) rear axle, and the tire diameter.
- The data needed to validate design of the system is the acceleration time which is what was optimized within this gearing design.
- Accel run times, chain slack, and vibrational noise will influence tuning within the drivetrain system. Originally, we were planning to have a load cell within the drivetrain but that was removed due to the already complex nature of the system.

(1) (Should be performed) To test the efficiency of the drivetrain we should be running accel and timing the result. Comparing this with the theoretical tells us whether or not the actual drivetrain is behaving how we expect it to theoretically. If it is not behaving as we thought, we can run the analysis backwards and see what the applied torque for the actual case should be and compare this with what torque is being output according to the motor. If these are similar, there is a motor issue. If these are similar, there is a drivetrain efficiency issue, which will lead to tuning. (Response variable being recorded is time)

- (i) Run accel. Setup a 75 m straight and send it full throttle while someone times the start and finish. Run a few trials to get sufficient data.
- (ii) If the data is inconsistent, try to identify something that is not working correctly. To check the efficiency, we should be seeing reliable similar runs, regardless of how fast they are.
- (iii) Compute the mean/median of the data and put that through the calculations backwards to identify the average torque produced by the motor.
- (iv) Take accel run data from the motor and compare the average torque to the calculated average torque. Tune the car as needed to improve the efficiency.

(2) (Done on ARG22) COG Test, find the COG by balancing the car. I was not there for this test, but it is outlined on the S:Drive under S:Drive>Cars>ARG23>Chassis>Chassis System Parameters [2] (Response variable being recorded is CG height).

- Because we are using the same monocoque mold as last year this seems like reasonable data to reuse although if weight moved around in the car for different subteams this could change a bit. Even with different distribution of weight for different subteams, this should not change a whole lot from last year's car.

- (3) (Continually checked for chain health) Measuring chain slack is important for making sure the drivetrain is not facing vibrational issues, a large issue in the past. We found this year that it is good to have about half a link of slack in the chain. Testing for this is still being developed, but from what I understand this involves taking one side of the chain and measuring the vertical travel distance of one pin [3].

4. ARG23 Design(50)

This year we are using a single strand 520 chain with an 11 tooth front sprocket and 43 tooth rear sprocket (both purchased) with a 4340 steel motor shaft wire EDM'd to fit the splines of the front sprocket. The motor shaft works on both the Emrax 208 and Emrax 228, either of which may be used depending on what we have access to. The rear sprocket is being mounted onto an aluminum 7075 mount which is made from the Drexler differential sprocket blank.

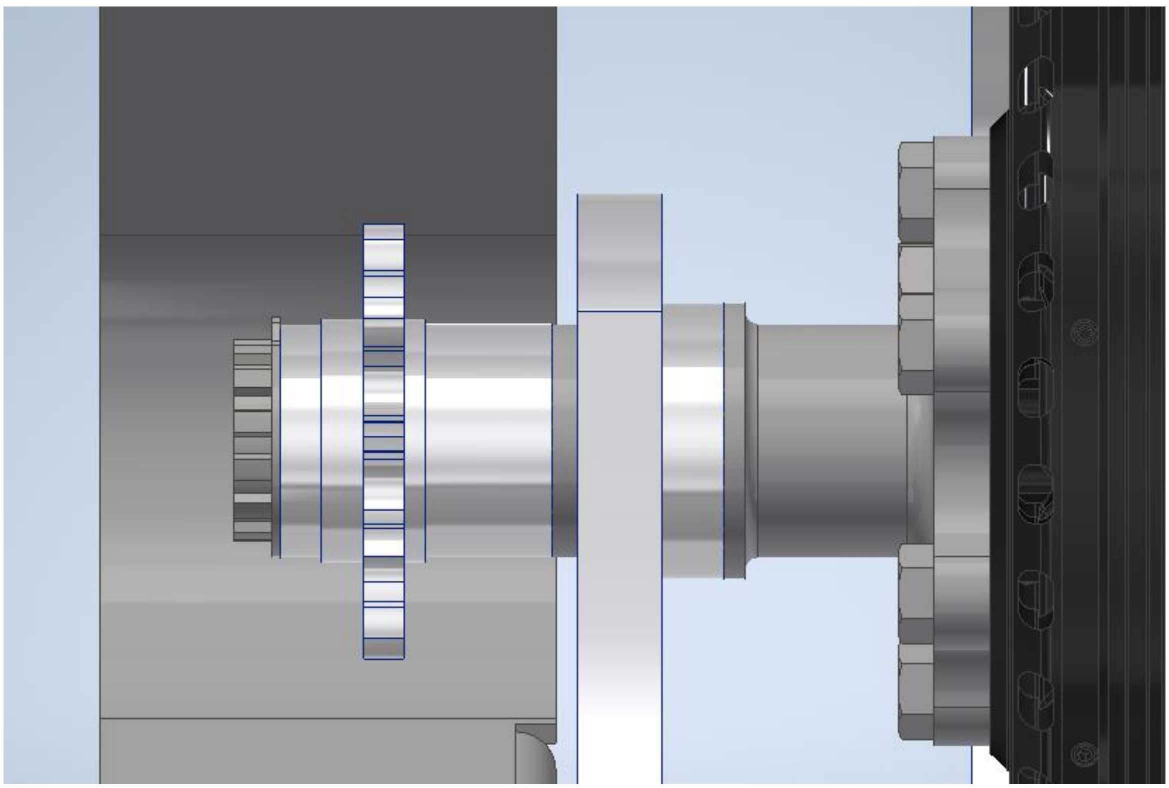


Figure 3: ARG23 Motor Shaft on Emrax 208

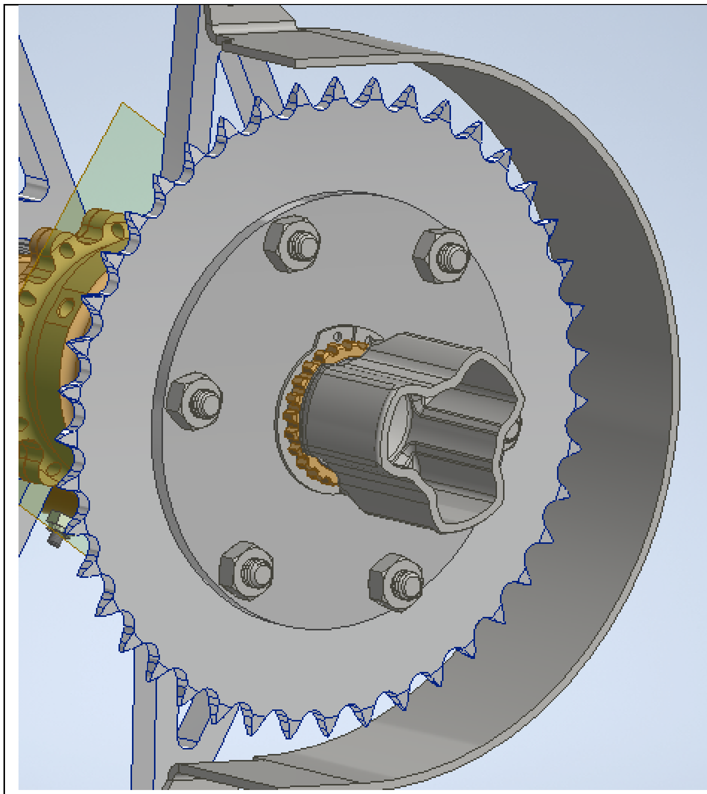


Figure 4: ARG23 Rear Sprocket Setup Driver's Left

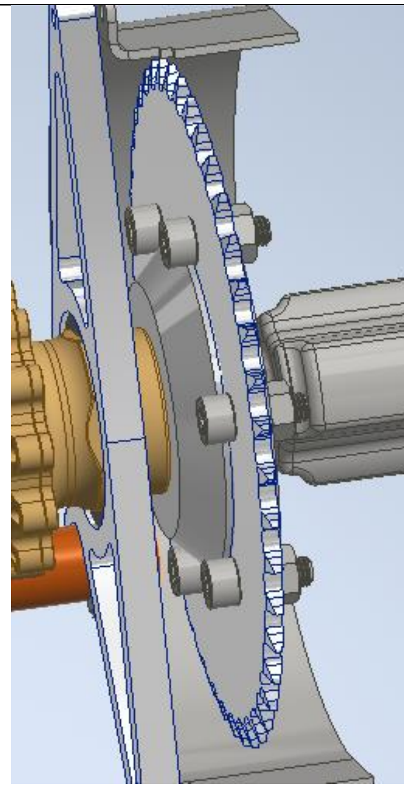


Figure 5: ARG23 Rear Sprocket Setup Driver's Right

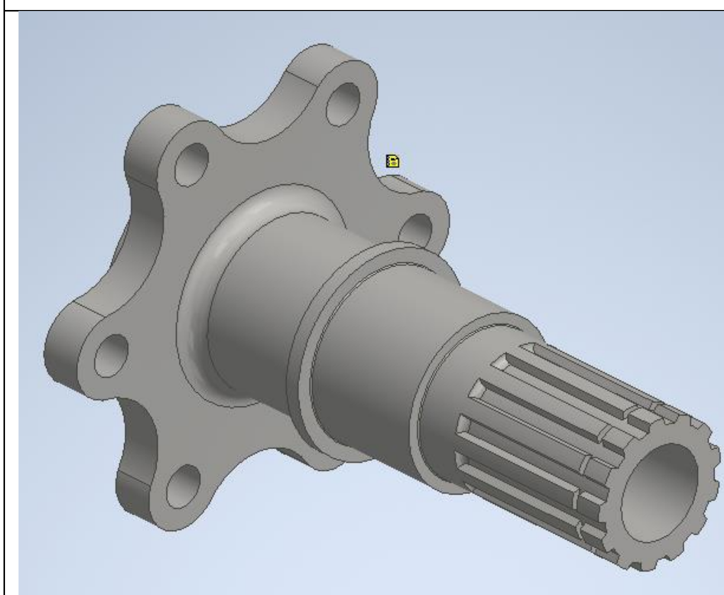


Figure 6: ARG23 Motor Shaft

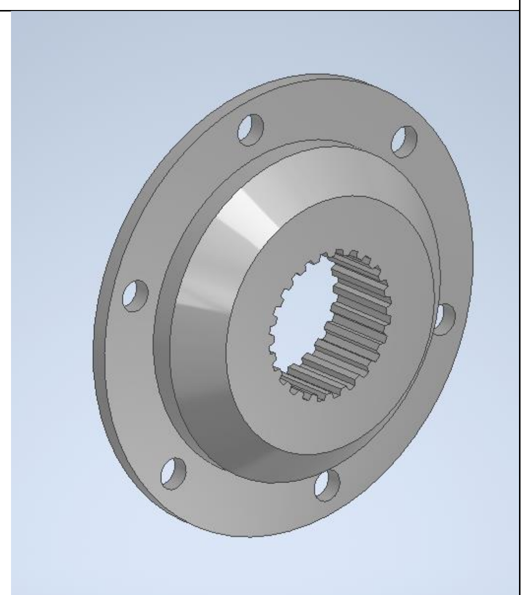


Figure 7: ARG23 Rear Sprocket Mount

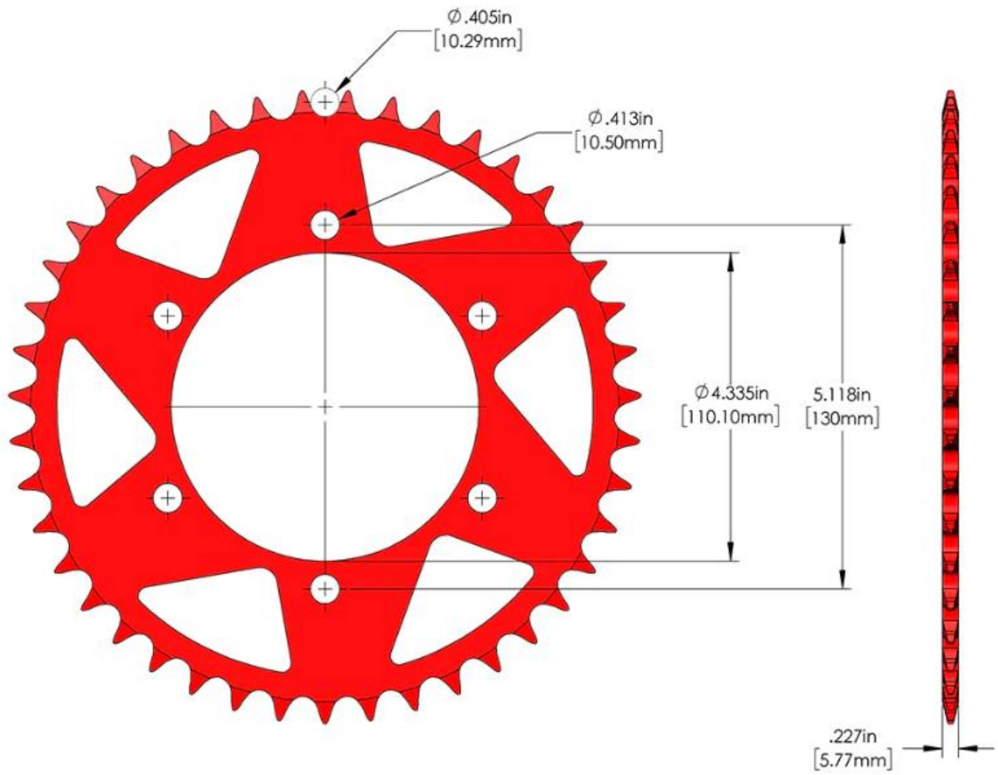


Figure 8: Sprocket Drawing

I decided to continue using a chain drive from last year because that can be modified to fit the separation distance between the motor and the differential rather than a gear system which needs to be fixed. Because the car is electric, there is no need for a gearbox. Originally, I started designing a chain drive system with a 520 single strand chain system in mind as well as Honda front 11 tooth sprocket in the front. A new rule was introduced this year which minimizes how much of the rear monocoque face can be removed for the drivetrain which had two large impacts: two piece bearing carriers which were taken on by Maddy and minimized chain interference. Because of this desired minimal chain interference, I looked into using smaller chain but to withstand the same loads, we would need to use triple strand at a minimum which would cause more interference and introduce more inefficiencies. Due to this, we stuck with 520 chain. The motor shaft probably took the most thought, as it needed to allow alignment of the sprockets, withstand the load from the motor, ensure motor concentricity, and have access to a stator bolt through the center.

ARG22 faced lots of vibrational and efficiency issues within the drivetrain so lots of the supports had to be redesigned (motor mounts, bearing carriers, clevises). This required changes within the motor shaft for alignment.

I see this year's team goal as passing tech day 1 at competition so we can participate in dynamic events this year (last year we made it to competition and passed all inspections on the last possible day which led to us passing 2 of the 3 static events and not participating in any of the dynamic events). This design stays within the rules and should allow us to make good time in the accel event at comp.

Analytical Models

To start, the range of FDR needs to be determined as outlined in the technical overview. A dynamic analysis of the wheel with vertical constraint was used to determine the behavior of the vehicle as shown below:

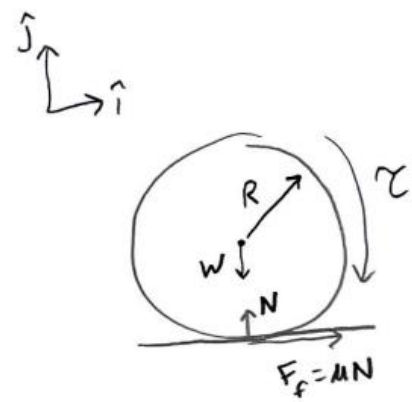


Figure 9: Wheel FBD

$$\sum \vec{F} \cdot \hat{y} : N - W = 0$$
$$N = W$$
$$\sum \vec{F} \cdot \hat{x} : \mu N = ma$$
$$a = \frac{\mu N}{m}$$
$$\sum \vec{M} \cdot \hat{z} : -\tau + \mu RN = 0 \text{ (Slipping Torque)}$$
$$\tau = \mu N R$$
$$G\tau_m = \mu NR$$
$$\tau_m = \frac{\mu NR}{G} \text{ (Slipping Torque from Motor)}$$

The above values for the motor torque and the vehicle acceleration are important for determining the motor limits and the optimal FDR for accel. In doing these calculations, I looked at analysis both with and without weight transfer. I will go through the dynamics of the vehicle without weight transfer, then with transfer, but first we a justification why slipping dynamics are applicable.

Slipping Dynamics or No Slipping Dynamics?

To determine whether slipping dynamics can or cannot be used we can look at the kinetic energies related to the car. There will be translational kinetic energy due to the velocity of the car moving and there will be rotational kinetic energy due to the wheels and the unsprung mass rotating. The following analysis was evaluated with a translational speed of 20 m/s (45 mph), assuming the car mass with a driver is 286 kg, and using the tire radius of 0.2286 m. The moment of inertia of the wheels and unsprung where taken from last year's corner assemblies in CAD with $I = 0.172 \text{ kg m}^2$.

$$KE_T = \frac{1}{2}mv^2$$
$$KE_T = \frac{1}{2}(286 \text{ kg})(20 \text{ m/s})^2$$
$$KE_T = 57200 \text{ J}$$
$$KE_R = \frac{1}{2}I\omega^2$$

$$KE_R = \frac{1}{2}(0.172 \text{ kg m}^2)\left(\frac{20 \text{ m/s}}{0.2286 \text{ m}}\right)^2$$
$$KE_R = 659.038 \text{ J}$$
$$\frac{KE_R}{KE_R + KE_T} = \frac{659.038 \text{ J}}{57200 + 659.038 \text{ J}} = 0.011$$

Comparing the rotational energy to the total kinetic energy we see that it is approximately 1 percent of the total kinetic energy so it can be considered negligible for our calculations.

Without Weight Transfer

In this scenario, we can assume the weight on the rear wheels is the half of the total gravitational force on the car due to the rear weight bias. From there basic dynamics can be applied.

$$N = \frac{mg}{2}$$
$$a = \frac{\mu N}{m} = \frac{\mu g}{2}$$
$$v = \int_0^t a dt = \frac{\mu g}{2} t \text{ (Start from rest)}$$
$$\Delta x = \int_0^t \frac{\mu g}{2} t dt = \frac{\mu g}{4} t^2$$
$$t = \sqrt{\frac{4\Delta x}{\mu g}} \text{ (Time for Accel)}$$

For competition, accel is 75 meters, which gives a time of 5.05 seconds and a max velocity of 29.70 m/s without weight transfer.

With Weight Transfer

In this scenario, there is additional downforce on the wheels from the weight transfer. This additional force has the following formula [4]:

$$F_{wt} = \frac{mah}{L}$$

We can use this to solve for the dynamics of the vehicle with weight transfer.

$$N = \frac{mg}{2} + \frac{mah}{L}$$
$$a = \frac{\mu N}{m} = \frac{\mu g}{2} + \frac{\mu ah}{L}$$
$$a = \frac{\mu g}{2(1 - \mu h/L)}$$
$$v = \int_0^t a dt = \frac{\mu g}{2(1 - \mu h/L)} t$$
$$\Delta x = \int_0^t v dt = \frac{\mu g}{4(1 - \mu h/L)} t^2$$

$$t = \sqrt{\frac{4(1 - \mu h/L)\Delta x}{\mu g}}$$

As noted in the previous section, accel is 75 meters, which gives a time of 4.37 seconds and a max velocity of 34.33 m/s with weight transfer.

FDR Limits

To make sure the gear ratio does not push beyond the limits of the motors angular velocity, the max angular velocity of the accel run needs to be compared with the maximum angular velocity of the motor. Below is a diagram of the chain system connecting the motor to the diff:

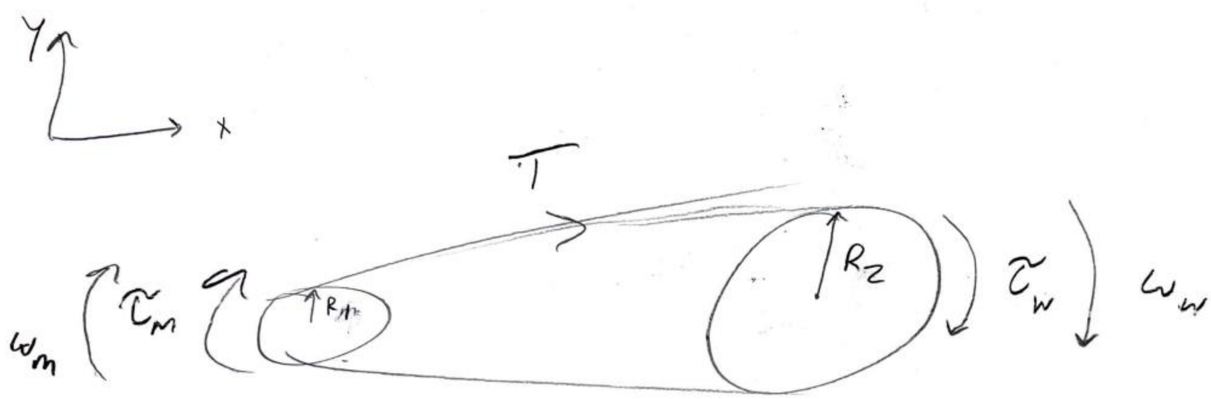


Figure 10: Chain Transmission FBD

Assuming continuous tension within the chain, $T = \frac{\tau_m}{R_1} = \frac{\tau_w}{R_2}$
Assuming unity power efficiency, $P_m = P_w \rightarrow \tau_m\omega_m = \tau_w\omega_w$
With these assumptions, we can derive a relation between the angular velocity of the motor and the angular velocity of the wheels.

$$\frac{\tau_m}{R_1} = \frac{\tau_w}{R_2}$$
$$\frac{\tau_m}{\tau_w} = \frac{R_1}{R_2}$$
$$\frac{\omega_w}{\omega_m} = \frac{R_1}{R_2}$$

At this point, we will be using the wheel angular velocity to solve an inequality related to the motor angular velocity so we need to know more about R_1 and R_2. In sprockets, the pitch (P) gives information about how the teeth and the pins in the chain are spaced. Knowing this, we can solve for the circumference and use this to solve for the radius. I will use N_1 to denote the number of teeth on sprocket 1 and N_2 to denote the number of teeth on sprocket 2.

$$C_1 = N_1 P = 2\pi R_1$$

$$R_1 = \frac{N_1 P}{2\pi} \rightarrow R_2 = \frac{N_2 P}{2\pi}$$
$$\frac{\omega_w}{\omega_m} = \frac{2\pi N_1 P}{2\pi N_2 P}$$
$$\frac{\omega_w}{\omega_m} = \frac{N_1}{N_2} = \frac{1}{G}$$
$$\omega_m = G\omega_w$$

I used G to denote ratio of number of teeth on the rear sprocket to the number of teeth on the front sprocket. This is the FDR! With this knowledge we can now solve the inequality to learn something about the FDR:

$$\omega_m \leq 6000\text{rpm} \equiv 628.319 \text{ rad/s}$$
$$\omega_w = \frac{v}{R}$$
$$G\omega_w \leq 628.319 \text{ rad/s}$$

The lower limit of the FDR from this equation will be at the highest wheel angular velocity. The highest angular velocity can be found using the highest translational velocity from the dynamics calculations which would be the 34.33 m/s from the weight transfer case.

$$G\frac{34.33 \text{ m/s}}{0.2286 \text{ m}} \leq 628.319 \text{ rad/s}$$
$$G \leq 4.18$$

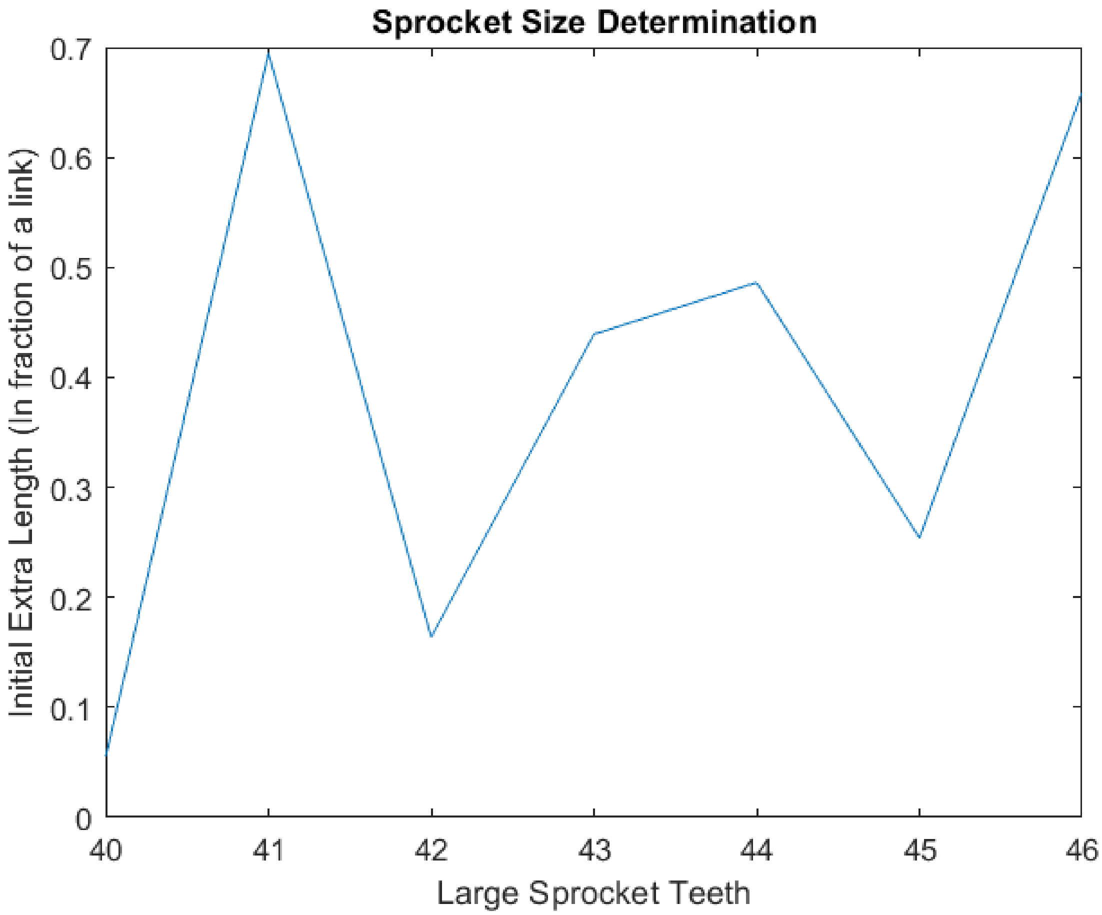
Therefore, the max FDR should be 4.18 to prevent the motor from limiting speed due to angular velocity limits. Another limit on the FDR will be from the motor's torque limit. To identify this limit we will need to look at the slipping torque.

$$\tau_m = \frac{\mu N R}{G}$$

This can be rearranged for G in terms of a bunch of variables. We can then solve for G at the max motor torque (the minimum FDR) by plugging in what we know about the normal force from previous analysis. Because the normal force N was greater with weight transfer we will use the weight transfer scenario to find the upper bound of this lower limit. (If we go below the upper bound it could be bad, whereas if we go above the lower bound it would not be a problem).

$$G = \frac{\mu N R}{\tau_m}$$
$$G = \frac{\mu R}{\tau_m}\left(\frac{mg}{2} + \frac{mah}{L}\right)$$
$$G = \frac{\mu R}{\tau_m}\left(\frac{mg}{2} + \frac{\mu mgh}{2L(1 - \mu h/L)}\right)$$
$$G \geq \frac{(1.2)(0.2286 \text{ m})}{140 \text{ Nm}}\left(\frac{(286 \text{ kg})(9.8 \text{ m/s}^2)}{2} + \frac{(1.2)(286 \text{ kg})(9.8 \text{ m/s}^2)(0.3302 \text{ m})}{2\big((1.5748 \text{ m}) - (1.2)(0.3302 \text{ m})\big)}\right)$$
$$G \geq 3.67$$

This provides us with a range of $3.67 \leq G \leq 4.18$ which is pretty big with the front sprocket we are running. With the 11 tooth at the front, we could run anything between a 41 and a 45 tooth sprocket. I used the matlab code found in the folder associated with this report to calculate which sprocket sizes would minimize the excess chain. The 43 tooth sprocket minimized the excess chain slack. (See the last section of the script, the rest of it is a bit of a mess and I never got the time to clean it up. Apologies for that.)



Motor Shaft Strength Calculations

As talked about in the technical overview, the motor shaft needs to be strong enough to withstand the loads presented by the motor. The material selection for the motor shaft was Steel 4340 because the motor shaft experiences high torques and galling is a possible concern. I do not know much about galling, but 4340 was recommended as a good preventative material for galling. Beyond this selection, there are a few stress calculations from the Machinery's Handbook which are important for defining the motor shaft and selecting hardening types. The first is the Shear Stress Under Roots of External Teeth given by the following formula (specifically for hollow shafts):

$$S_S = \frac{16TD_{re}K_a}{\pi(D_{re}^4 - D_h^4)K_f}$$

Where T is the transmitted torque, K_a is the spline application factor, D_{re} is the root diameter (the smaller major diameter of the splines), D_h is the hole diameter of the shaft, and K_f is the fatigue life factor. For the shaft, I used the max motor torque $T = 1240$ lb in, $K_a = 1.6$, $D_{re} = 1.0625$ in, $D_h = 0.75$ in, and $K_f = 1$. These k values represent an application between intermittent and heavy shocks for application and fatigue after 10,000 cycles. These values produce a shear stress of 11206 psi at the root which gives a safety factor of approximately 3.57 for surface hardened steel.

The next stress to evaluate within the motor shaft is the Shear Stress at the Pitch Diameter of the Teeth which is given by the following formula:

$$S_S = \frac{6TK_aK_m}{DNL_etK_f}$$

Where all values from before are the same. $K_m = 1$ is a load distribution factor, $N = 13$ is the number of teeth, $L_e = 0.4$ inches is the effective length of the splines (the length in contact with the sprocket), $t = 0.4$ inches is the tooth thickness. For the current spline design, this shear stress value was calculated to be 5087 psi which has a safety factor of 7.86 for surface hardened steel.

Lastly, the Compressive Stresses on Sides of the Spline Teeth were evaluated with the following formula from the Machinery's Handbook:

$$S_C = \frac{2TK_mK_a}{DNL_ehK_w}$$

Where the only new variable introduced is $h = 0.1$ inches which is the depth of engagement of the teeth (equivalent to 1/P where P is the diametral pitch). It is important to note that spline (diametral) pitch is defined as the number of teeth per inch of pitch diameter, the same way diametral pitch is defined in gears. In sprockets, the pitch is defined as the half the distance between 3 consecutive pins. I found this easy to get mixed up. The value of this compressive stress came to 4239 psi which corresponds to a safety factor of 0.94 for surface hardened steel.

I misinterpreted what the difference between flexible splines and fixed splines was when I first went through these calculations so I will restate the difference as explained in Machinery's Handbook: "A fixed spline is one which is either shrink fitted or loosely fitted but piloted with rings at each end to prevent rocking of the spline which results in small axial movements that cause wear. A flexible spline permits some rocking motion such as occurs when the shafts are not perfectly aligned. This flexing or rocking motion causes axial movement and consequently wear of teeth. Straight toothed flexible splines can accommodate only small angular misalignments (less than 1 degree) before wear becomes a serious problem." In the future, I would recommend looking into a hardening the motor shaft beyond the surface hardening to increase this compressive safety factor. There is not much that can be done to increase the manufacturing precision since this is an outsourced part that we do not have the capability to manufacture (cannot wire EDM in house).

Looking at the things that could be modified, the rear sprocket mount will work but it could use some work. It is a large chunk of metal with no weight saving holes so another design with less weight could be beneficial for the team. This is something Nick Thomas (team lead) recommended working on next semester.

There is nothing that currently stands out to me as being nice to have in terms of testing that has not been done yet or is not scheduled to happen next semester.

Looking to the future, the motor shaft is pretty well thought out besides the hardening, I would recommend modifying the rear sprocket setup, possibly designing a one piece rear sprocket to reduce weight. We went with a premade sprocket that attaches to a sprocket mount made in house because that allows for adjustability for different motors, but this could possibly be done with multiple rear sprockets in the future (past cars have had multiple sprockets designs).

5. Components Overview

- Motor Shaft Stock
    - 4340 steel, 1 ft long, 4” diameter rod (enough for two motor shafts)
    - Yield Strength: 68 ksi
    - McMaster-Carr
- Motor Shaft Spacer Stock
    - 1 ft long, 1.5” OD, 1” ID Aluminum 6061 Round Tube (enough for 4 sprocket spacers)
    - 1 ft long, 2” OD, 1.25” ID Aluminum 6061 Round Tube (enough for 4 bearing spacers)
- Rear Sprocket Mount Stock
    - Drexler Differential Sprocket Stock (not much info given)
    - Aluminum 7075
- Rear Sprocket
    - 43 Tooth Red Vortex 520 Sprocket – 130 mm bolt circle
    - FSAEParts.com was vendor, unhelpful in getting more details about the part prior to purchase
- Retaining Rings (Diff)
    - External Retaining Ring for 1-31/32" OD, Black-Phosphate 1060-1090 Spring Steel (Pack of 10)
    - McMaster-Carr
    - 97633A225
- Retaining Rings (Motor Shaft)
    - External Retaining Ring for 1-3/16" OD, Black-Phosphate 1060-1090 Spring Steel (Pack of 15)
    - McMaster-Carr
    - 97633A330
- Chain
    - Fire Power - 520 O-Ring FPS Chain

FIRE POWER CHAIN SPECIFICATION CHART								
	Pin Diameter mm	Pin Length mm	Roller Diameter mm	Plate Thickness mm	Tensile Strength	Wear Index	Max. Rec. Displacement Street	Max. Rec. Displacement Dirt
420FPS	3.96	14.7	7.77	1.5	4,300	100	80	80
428FPS	4.44	16.3	8.51	1.5	4,400	100	100	80
520FPS	5.08	17.3	10.16	2	7,400	100	250	125
530FPS	5.08	20.5	10.16	2	7,400	100	350	N/A
420FPH	3.96	16.3	7.77	1.5	4,520	120	110	110
428FPH	4.5	17.4	8.51	1.5	4,900	120	125	125
520FPH	5.08	18.5	10.16	2	8,000	200	450	450
530FPH	5.08	21.3	10.16	2	8,000	200	550	550
520FPO	5.38	20.8	10.16	2.2	9,000	1,000	750	650
525FPO	5.38	22.35	10.2	2.2	9,000	1,000	900	750
530FPO	5.38	24	10.2	2.2	9,000	1,000	1000	750
520FPX	5.38	19.95	10.16	2	8,500	1,500	750	650
525FPX	5.38	22.35	10.2	2.2	9,000	1,500	1,000	750
530FPX	5.38	24	10.2	2.2	9,000	1,500	1,000	750

    - MOJOMotorSport.com
    - Also ordered Master Links
- Rear Sprocket Mount Bolts
    - M10x1.5, 10 mm shoulder length, 10 mm shoulder diameter, shoulder bolts (6x)
    - McMaster-Carr
    - 90782A128
- Rear Sprocket Hipro Nyloc Nuts
    - M10 Hexagon Nylon Locking Nuts (DIN 985) - Stainless Steel (A2)
    - Accu
    - HNN-M10-A2

6. References

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