Cockpit & Controls Engineering Design Notebook

SR21 (F.S. 2019-2021)

Shocker Racing Formula SAE, Wichita State University, Wichita KS

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Start Date: August 2, 2019

Completion Date: TBA

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08/02/19

Preamble:

The purpose of this notebook is to document all important and relevant information related to the design, analysis, construction, and testing of a Formula SAE racecar for use at competition. This specific notebook will cover the area of design of the racecar cockpit and subsequent ancillary subsystems related to the operation of the racecar. All important design decisions, criteria, constraints, problems and solutions shall be listed and documented in chronological order with the aim of providing a guide to future sub team design leaders to aid in their own designs by providing them with relevant information from their predecessors. In short, the goal of this document is to provide direction and insight to the future design leaders by documenting the processes of building a racecar.

Introduction:

The Project shall be called SR21. The purpose of the project shall be to design, build, and ultimately compete in an open wheel, open cockpit, formula-style racecar homologated to the Formula SAE 2021 rules and regulations to compete in the 2021 FSAE completion season. This project, which shall be called “the car” from here on, is the direct successor to SR19. Many concepts/ideas may be transferred from SR19 to the new car, SR21.

This particular notebook shall be used to help design and document the Cockpit and Controls (abbreviated to CC) sub-team of Shocker Racing. The current Cockpit and Controls Lead Design Engineer is Justin Wierman, and this notebook shall be kept by him.

Goals:

<u>Car Goals:</u>

- Dynamic
  - Run a full endurance course at least twice before competition.
  - Acceleration time under 4.6 seconds
  - At least one wet-weather testing session
  - 5.5 sec left and right skidpad
  - First and second place raw time at an SCCA event
- General
  - No fires
  - Have the car ready by Oct. 1 2020
  - Overall weight under 470 lbs. sans driver
  - Clutch paddle shift system
  - Documentation of everything before manufacturing
  - CAD models of everything before manufacturing
  - Design for manufacturability

<u>Sub-team Goals:</u>

- Cockpit and Controls
  - Make in-house steering wheel with data logger mounted in it

- Implement paddle shifter
- Easier Adjustable pedals
- Increase belt/tab comfort with-in cockpit

10/04/19

Brake System:

Today, I'll begin work on designing the brake pedal and system for SR21.
One of the first things I need to determine is how long the pedal should be, in addition to the material it will need to use. This will depend on the forces that will be occurring at the pedal. According to Rule T.3.1.10: "The brake pedal must be:

a. Fabricated from steel or aluminum OR machined from steel, aluminum, or titanium.
b. Designed to withstand a force of 2000N without any failure of the brake system or pedal box.
This may be tested by pressing the pedal with the maximum force that can be exerted force that can be exerted by any official when seated normally."

This rule is what I will design the pedal to, using FEA simulation to do the analysis. Also, from reading the cockpit and Controls design review/report from SR16, the pedal had a pedal ratio of 5.013:1 for an input (driver) load of 75lbs. The brakes in SR16 have been extremely reliable, having been proved so though many hours of testing. Therefore, it is safe to assume that these are good numbers to work from.

In addition to the above rule, the brakes will need to be able to lock all four wheels simultaneously, which SR16 (a car of very similar weight) is able to do quite easily.

10/05/19

Continuing from yesterday, I will need to further define what I require from the design so that I will have a clear understanding and objectives with which I can begin to design and construct the brake pedal. I'll begin with a list of the criteria and constraints that I will be working with, and then I will add some notes on what I wish to have the pedal do or be. I can then assess my brake pedal designs based off of that information. This will be an initial list and will be subject to change.

Criteria: (*Not in any particular order).
- Be comfortable to use by all drivers.
- Be adjustable to accommodate drivers of different builds.
- Have a captive toe cup.
- Have a captive heel cup.
- Be light weight, yet strong.
- Incorporate an adjustable brake balance bar.
- Have a properly calibrated pedal ratio.
- Use an appropriately sized master cylinder bore.

Constraints: (*Not in any particular order).
- Pedal must be fabricated from either steel or aluminum.
- OR can be machined from steel, aluminum or titanium.

- Must be able to withstand at least 2000N (450lbs of force applied to the pedal by the drive.
- Feature dual circuit plumbing.
- Be capable of locking all four wheels simultaneously.
- Have a brake over travel switch (BOTS) *Rule T.3.2.

10/09/19

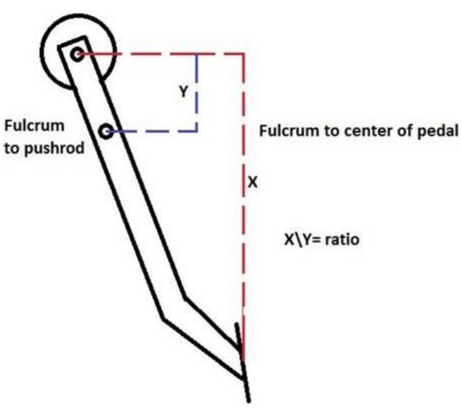
As mentioned before, SR16s brake pedal assumed an average of 75lbs of force input by the driver. Where this information is coming from, I don't know. It's probably a safe assumption, but I would like to do some actual human performance testing on some of our drivers/potential drivers to determine if 75lbs is a good number, or if it needs to be adjusted. I would like to do this by taking a group to the Heskett Center, which is our campus gym, and have each subject do leg presses and calf raises to determine the maximum amount of weight they can comfortably push without exerting themselves. I don't want the drivers to be exerted since they will need to push the pedal over and over during endurance. I will do this by having them adjust the weight and record their feedback, then tabulate it. Then a proper number can be verified. This is assuming that there is adequate time to do so.

The optimum pedal input will affect the overall pedal ratio for the brakes, so the sooner I get that performance data the sooner I can start on the pedal. That being said, an online article from Master Power Brakes (https://techtalk.mpbrakes.com/how-to-series/calculating-pedal-ratio) says that the ideal pedal ratio for a car without power brakes or any other assists should is anywhere between 5.01-7.01:1. A brake pedal with too little ratio becomes very hard and fatiguing to operate, while one with too large of a pedal ratio becomes too soft and delicate to properly modulate.

10/10/19

The same goes for the master cylinder. A master cylinder with a bore that is too large is going to feel soft and accidental lock ups will be an easy mistake to make. On the flip side, a cylinder with too small a bore diameter will make the brake stiff and hard to use, and will fatigue the driver quickly. According to NAPA Know How blog (http://knowhow.napaonline.com/how-to-measure-brake-pedal-ratio/), for a car without power brakes or assists, a master cylinder should not be any larger than 1" diameter bore.

Now, onto the slightly more technical side. I have mentioned the pedal ratio several times without explaining it. The pedal ratio is simply the ratio between the distance between the fulcrum and the pushrod (for the master cylinders) and the distance between the fulcrum and the end of the pedal where the force is applied. As illustrated:



The pedal ratio essentially multiplies the forces being applied through the pedal just as a lever would. For example, if the pedal ratio is 5:1, and the driver inputs 100lbs of force at the end of the pedal, there will be a 500lb force applied to the pushrods.

This exemplified why the pedal ratio plays a role in braking force and why it needs to be carefully considered. The master cylinder will be the same way. My decision on pedal ratio/cylinder bore will have to wait until after I collect my human performance data. (*Picture from Napa Know How Blog*).

12/20/19

Long needed update: After a month and a half of grueling classes and other such things I'm finally back to working on the brake pedal design. My time over the past few months has been spent on trying to pass my classes (which I did fortunately) and trying to get SR19 running and driving, as well as repairing the broken SR16 car, which snapped a bolt in the right rear upright and broke some other stuff along the way. It's finally fixed, and now we need to find time to take 16 out to give it a shakedown run to make sure we fixed it properly before taking the car back to autocross to train new drivers, and to get practice for myself. Also, a couple of weeks ago (sometime in mid-November) I was elected team co-captain along with Felipe Escalante, so my time will also be split trying to manage the team. Hopefully this won't hamper my progress too much. More to come soon.

12/28/19

I took a much needed break for Christmas to spend time with family and friends. Now that that is over it's time to return to the pedal design. I haven't made nearly as much progress as I should have over the past semester, so my goal for this semester is to buckle down and crank out some design work. As I mentioned before, part of the time last semester was spend trying to get SR16 working, and SR19 firing. Still finishing the details on that one. I'm writing all this down so that whomever may be reading this will have an understanding of what is going on beyond the context of engineering design, and where I am coming from as a designer. Managing school and formula is a difficult balance to achieve. Having said all that it's time to get back to the more formal work of engineering design.

As I wrote on the previous page, I wanted to collect some data from our drivers to find out the ideal pedal ratio. But it's now winter break, most people are gone, and the gym is closed anyway. With that in mind, I'll need to choose the pedal ratio based off of old designs and test it later when I get a chance. It's not an ideal scenario, but it's what I've got.

I just spend the past hour reading a short paper written by an FSAE team in India about their brake pedal design. I didn't learn much from it but I'll link it just in case I want to reference it again (https://ijret.org/volumes/2017v06/i10/IJRET20170610005.pdf). The only thing that might have been valuable in terms of information was the bit about their car needing to produce 1.65 G of braking deceleration to lock all four wheels. I have no idea where they got that number from or how they may have calculated it. I've also been reading Carroll Smith's *Engineer to Win* book. It was written in 1984 so the info is a little dated, but still decent. I haven't finished the section on brakes yet, but so far I haven't learned anything pertinent to me yet. Ok, I just finished it, and there wasn't any useful information in it. Joy. I'm moving on to reading the SpeedPro Series on suspension and brakes. We'll see if it's worth my time.

Ok, so I've been reading a bit of it, and it does actually have some useful recommendations. On page 81 it mentions that the master cylinder that is used should be the smallest it can be: "The question many enthusiasts face s=is what size of master cylinder, or cylinders, should be used? The answer is without a doubt the smallest that can be used in the particular installation, but one, or ones, that can displace sufficient brake fluid conducive to low brake pedal movement, and can achieve the required amount of line pressure of 1100-1200 psi or so as to operate the brakes at maximum efficiency." I can't figure out if the 1100-1200 PSI is accurate for only large race cars, or also for small formula cars like ours. I'll have to find out later. To summarize what the author said about too large or too small of master cylinders, basically if the cylinders are too small, and the pedal ratio too large, the pedal will be soft and could bottom out before reaching the required line pressures. On the flip side, if the cylinders are too large and the pedal ratio too small you won't meet the line requirements anyway. This is because an amount of brake fluid needs to be displaced before the pads make contact with the rotor, but won't increase line pressure, which will require a fair amount of pedal travel and fluid displacement. Line pressure will only increase when the pads make contact and the brake "goes hard". Fun balancing act, no? Fun note I found in here was about "residual line pressure valves". Apparently what they do is they keep a small amount of line pressure so as to keep the pads next to the rotors without affecting forward motion of the car. This apparently helps with pedal travel. Something to look further into.

12/30/19

New day and I'm back in the shop. It's almost a new decade. Holy cow. Ok, back to reading. The next section in the SpeedPro books is the part about pedal ratio. According to this book, the smaller the brake pedal ratio is, the less actuation will be needed to engage the brakes, and the quicker it will be, but it will also make the driver work harder to push the pedal. The recommended pedal ratios seem odd though. It suggests an absolute minimum of 2.5:1 while 5:1 is the higher end, with 7:1 being generally the biggest, and 10:1 being the absolute maximum. NAPA didn't seem to think 5:1 was that high. I might google some other FSAE teams and try to see what their pedal ratios are. One good point it did provide that I've thought about loosely what about where the foot should be on the pedal. The book states that the most force is going to be applied with the ball of the foot on the pedal face. This means that the pedal should be sized so that the pedal ratio is calculated up to the ball of the foot, not the toe. That's something important to remember. *The force will imparted at the center of the ball of the foot.*

Brake System project will continue on page 7.

1/2/20

Cockpit Sizing:

For now I'm doing a little bit on the cockpit dimensions so that the Chassis lead can continue on his model. This will happen periodically where multiple projects are occurring simultaneously. I am keeping this journal in chronological order, so I will sometimes leave off a project to write a section on a different topic, then return to the one I was working on before. Be aware that this will happen and simply look at the end of the section to know where it continues.

It's a new decade but old arguments still go on. For some time the Chassis Lead Lieth and I have been going back and forth on how big the cockpit should (or should not) be. Several weeks of this and still no conclusion. The new chassis for SR21 is going to be based off (an iteration of) SR19. Thus, the cockpit will be similar. And hence I've spent the last two hours trying to work out with him what needs to change. We've reached a point where we can't agree. (Note to reader, this happens a lot in engineering. Handle it well and calmly and it will be resolved. Don't get angry or frustrated or nothing good will come of it. Take it from an experienced member.) Here is what is being proposed for changes. **1)** Length of the front (knee) roll hoop to the front bulkhead. From drivers seated in SR19 all complained it was too short to fit the driver's legs. The knees were bent so much that many of their thighs we contacting the steering wheel causing major difficulties steering even when the car was on jacks. I'm proposing extending this length by 1-2" in order to better accommodate the drivers legs. Lieth is in agreement on this one. **2)** The size of the front (knee) roll hoop needs to be altered slightly. I want an extra inch at the crown of the hoop to accommodate a higher steering wheel. Because in Rule V.3.3.1: "In any angular position, the top of the Steering Wheel must be no higher than the top-most surface of the Front Hoop. See figure following F.5.8.6". Thus, if we raise the crown of the front hoop, we can raise the steering wheel, eliminating this issue. Lieth also agrees here. However, he wants to make the front hoop wider. His reasoning is that we need more space to accommodate the driver's legs and possibly more components. I disagree. I think the spacing is fine the way that it is. As a driver myself I'm comfortable with it. However, I will note that I am not the only driver, so I can't rely solely on my input. That being said, some drivers will simply not fit. I want to select drivers who are close to my size to space the chassis. We need to be moving to a smaller chassis to save weight as our car is already a chunky lady. Thus smaller drivers like myself will need to be used. I'm not trying to be selfish by sizing the car to me, but I'm one of the smallest and lightest people on the team who is capable of driving the car. (For reference I'm 6' 150lbs, which is comparable to an F1 driver). I'm willing to take a hit on the cockpit being small in judging if we can win back the points in the dynamic events. Besides, if the car will fit Percy, we pass the 95th percentile requirement anyway, which SR19 currently does. Therefore, I am against this purposed widening, as my legs fit with some clearance. Also, the shifter and clutch are in an awkward spot on the car. This is because of the bar that is in the cockpit as bracing.

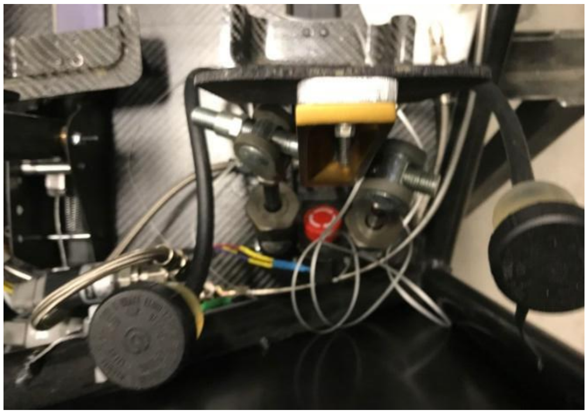


As you can see from the three images above, the shifter is fairly far forward in the cockpit. This can be seen especially in the first image. In the second image you can see how having it that far forward encroaches into the driver knee space. Not having that shifter there would make it far more comfortable to use the shifter, and would create plenty of knee space. It's so far forward because as you can see in the third image, it is the only place you can actually get your hand around the handle. Any further back and the bar would make it impossible to use the shifter. I put it there because I had to, not because I wanted to. I'm trying to work with Lieth to see if the bar can be removed, which would solve many problems, including not needing to make the knee hoop any wider, thus saving weight and space. This is the biggest disagreement we are having.

4) I believe we can bring the top bars of the cockpit (the ones next to the driver's shoulders) closer to the driver. They aren't bad now, but I think they could be better. Lieth does not want to do this. This is something I'm willing to concede to, however, I don't want it any larger than it currently is. **5)** I want to narrow the front bulkhead to taper the nose of the car, which should also help with aerodynamics as well as weight. The front where the pedals are is actually quite wide for just two pedals and can be reduced, so long as it is within the rules. Lieth is sorta Ok with that so long as the attenuator still fits. He says that it will increase the CG since the attenuator will have to be flipped to fit, but it's so light I don't see this as an issue.

Anyway, we've taken a break from the debate and will return to it later.

Also, with the cockpit space, specifically in the pedal area, if I can make the pedals with the underfoot configuration for the master cylinders, it should also help give more space in the cockpit as the master cylinders won't be protruding like they do in the current configuration:



Note the difference in the two. The left is the design by Ryan Hoadly for SR19 (originally SR17), assembled by Justin Wierman (me). The one on the right is an underfoot style pedal arrangement for their premade pedals. Notice how the master cylinders in Ryan's design stick out past the pedal face. This significantly limits how far forward the pedals can be, while in the underfoot design, the master cylinders are tucked up underneath the drivers heels (hence the name) to where they remain out of the way. The only limit to how far forward they can go is the length of the pedal travel before it hits the bulkhead. All this is to say that with this pedal arrangement, I can better utilize the space available to me, which will also for greater space in the cockpit for the driver without actually making it bigger. Often things come down to efficiency, and this case is no expectation.

Also, while I'm on the subject, I'd like to eventually take a look at the clutch and shifter we have been using since 2016. It's been a good design, but a little bulky. I'd like to see if I can make it a bit smaller and cut down on the amount of room that it takes up. Obviously, like the goals stated on page 2, we are trying to implement paddle shifter. But if they don't work out (and there is fairly good chance they won't) we will have a viable back up to use. But since I'm planning on only having this as a backup, I won't spend much time trying to improve it.

I think that's all for now. I'll probably get back to designing the brake system.

Cockpit Sizing project continues on page --.

1/3/20

Brake System: (Continued from Page 5).

After taking a short break to try to work out the size for the cockpit, I'll be getting back to my main focus of working on the brake design. I need to keep reading the Speedpro book to research designs, then come up with an initial design. The author notes on page 83 that with a larger pedal ratio there will be little excessive pedal travel, even if the overall travel of the pedal is longer. He recommends that you use the smallest bore master cylinder that provides a solid pedal feel with no more than 1-1.5" of pedal travel. That seems to be all for the brakes part of the book that is useful for my design. I'm now going to check online to get some ideas of what other teams are running. Mostly to get an idea of what I should use for our particular application, but also to verify the accuracy of the information I just read.

Just found a useful paper written by MIT in 2018 on braking systems for an FSAE car (linked here: https://dspace.mit.edu/bitstream/handle/1721.1/119947/1080312700-MIT.pdf?sequence=1&isAllowed=y). For their vehicle, they calculated a braking torque (something I need to learn how to do) of 384 Nm in the front, and 145 Nm in the rear. Based off of this they selected a 1" cylinder for the front, and 5/8" for the rear. Interestingly enough, MIT used a 3.14:1 ratio for their car. Their car weighed 507 lbs. For reference, SR16 weighed 478lbs (w/o driver) and has a reported pedal ratio of 5.013:1. The pedal felt pretty light to be honest, so a lower ratio is possible. The cylinders it uses are 5/8" for both front and rear.

Actually, this is a really useful document I'll have to read more about.

Reading more into the report, I've found some calculations would could prove to be useful in designing the brake system. The first is for braking torque required:

The braking torque requirement for the front and rear axles can be calculated from the following equation given that the normal force, F_z, at each of the axles and the tire radius, r_{tire}, is known during a braking event.

$$\tau_{braking,f} = r_{tire} \cdot F_{z,f} \cdot \mu_{tire} \tag{1}$$

$$\tau_{braking,r} = r_{tire} \cdot F_{z,r} \cdot \mu_{tire} \tag{2}$$

Note that these equations assume a constant coefficient of friction, μ_{tire}, for the tires. In reality the coefficient of friction is sensitive to normal force acting on the tire, but for the purposes of designing the hydraulic braking system, it is acceptable to use an average coefficient of friction for the tires.

Figure # (Image credit to MIT Motorsports).

**Side note on braking torque, Engineers Edge (https://www.engineersedge.com/mechanics_machines/braking_torque_13635.htm)describes braking torque as "…the force applied at the brake wheel to stop the motion of the moving equipment". Furthermore they claim that "Assuming the operating conditions for the equipment are constant, a brake having a retarding torque equal to the full load torque of the motor to which it is applied is usually satisfactory."*

For the above equation, I would need to calculate the $F_{z,f}$ and $F_{z,r}$ during a braking event, meaning that because of the momentum of the car, I can't simply use a static FBD and simple algebra to find the normal forces. Instead I'll have to rely on my knowledge of dynamics in order to determine the weight transfer during braking. This is because the conservation of momentum of the car will "push" more of its weight onto the front tires instead of the rear. In technical jargon this is called "heave" and is something suspension tries to account and compensate for. I'll probably need the CG for the car in order to calculate this, which I don't currently have. I suppose that I could use the CG figures from SR19 since this car will be an iteration of that car. Not perfect but it might get me close enough.

1/6/20

So, I've had some time over the weekend to think about how I'm going to solve for the normal forces on the tires during maximum braking. And I think I have a solution: I'll use this video from Purdue University (https://www.youtube.com/watch?v=Ccnv8voKx9Y) as a guide to solving for the forces. Then, for the variables, I can use the data logged from SR16's accelerometer to find the peak G load under braking as the acceleration, and use the CG and wheel base data from SR19. Once I have solved for the normal force, I can then use the tire data sheet from the tires we are going to use for the new car to find the average coefficient of friction at normal operating temperature to find the braking torque.

The reason I can make these assumptions and approximations like this is because SR16's weight will be very similar to the weight of the new car, as such the peak deceleration will be nearly the same. Since SR21 will be an iteration of SR19 using its wheel base and CG will also be a close approximation. These assumptions don't need to be extraordinarily accurate. I figure that a 10% error will be acceptable, for the reason that all of those numbers will ultimately affect how much force the driver will need to apply to the pedal, which is fairly easy to control. For example, if the target driver force input is 100lbs of force applied to the pedal face, and there is a 10% error, that's only 10lbs of force that either needs to be added or removed to have optimal braking. So the max braking force would be 110lbs and the minimum would be 90lbs. Not too awful for a driver to control themselves, right?

Once I have figured this out, and once I have the target driver input force (abbreviated from here on as TDIF, or DIF for driver input force) I can then use the next equation from the MIT paper to find the pedal ratio and master cylinder size:

The hydraulic braking system is a hydrostatic pressure system, with an incompressible fluid, meaning that the calculation of forces and pressures is very simple. Recall that the brake pedal is attached to two independent master cylinders with a bias bar that distributes force from the pedal to each master cylinder. The master cylinder output pressure is proportional to the force input to the master cylinder. There is no pressure drop in any of the brake lines between the master cylinders and the brake calipers, since there is no fluid flow or any kind of pressure regulator device. Pressure from the brake lines applies force on the pistons in the brake calipers, applying a clamping force on the brake pads and rotors. The friction force of the brake pads acts at an effective brake rotor radius and results in braking torque. The following equations map driver input force at the pedal, to a resulting braking torque.

$$F_{driver} \cdot PR \cdot BB_f = \frac{\tau_{braking,f} \cdot A_{MC,f}}{2 \cdot r_{rotor,effective} \cdot \mu_{pad-rotor} \cdot A_{C,f}} \quad (3)$$

$$F_{driver} \cdot PR \cdot BB_r = \frac{\tau_{braking,r} \cdot A_{MC,r}}{2 \cdot r_{rotor,effective} \cdot \mu_{pad-rotor} \cdot A_{C,r}} \quad (4)$$

(Image credit to MIT Motorsports)

F_{driver} is the TDIF, *PR* is the pedal ratio, BB_f is the front brake bias percentage, BB_r is the rear brake bias percentage, $\tau_{braking}$ is the braking torque (calculated from the first equations), $A_{MC,f}$ is the master cylinder area of the cylinder connected to the front wheels, $A_{MC,r}$ is the area of the master cylinder routed to the rear wheels. $A_{C,f}$ is the area of the front caliper piston, and $A_{C,r}$ is the area of the rear caliper piston. $r_{rotor,effective}$ is the radius of effective radius of the brake rotor***, and $\mu_{pad\text{-}rotor}$ is the coefficient of friction between the brake pads and the brake rotors.

****effective rotor radius is essentially the radius of the center of the brake pads, as seen in this diagram:*

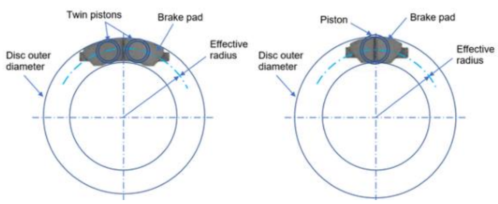


Figure # (Image credit unknown)

I'll need to get the coefficient of friction from the suspension lead (Luke in this case) as well as the effective rotor radius, and area of the caliper piston, since these are things that he will be picking out. I can then try several permutations of calculations using various pedal radios and master cylinder diameters to find out the best combination, then use a decision matrix to determine the best one based off of various criteria. It's a lot of numbers, but it should all be worth it.

After that is completed, I can use the following equation to calculate the maximum hydraulic pressure in the system to verify that the brakes lines I buy or that we already have an support the forces it will be subjected to. It's not terribly important because brake lines are pretty strong, but more validations usually doesn't hurt. Plus, more data for judging.

$$P_{hydraulic,f} = \frac{F_{driver} \cdot PR \cdot BB_f}{A_{MC,f}} \quad (5)$$

$$P_{hydraulic,r} = \frac{F_{driver} \cdot PR \cdot BB_r}{A_{MC,r}} \quad (6)$$

Figure # (Image credit to MIT Motorsports)

1/7/20

Just contacted the suspension and brakes lead, he says he hasn't picked out anything yet, so I don't have any tire data, no brake pad or rotor data. So I'll probably need to base my design entirely off of SR19's current configuration, save for the peak deceleration under max breaking as the car isn't running yet. (We are getting awful close though). It should be a good enough approximation to do the job. Like I said yesterday, the approximations don't need to be terribly close.

I'll start collecting data and putting it in the data table below:

Variable:	Data:	Measured how?
Overall weight:	628 lbs	Physically from SR16 + my weight
Front weight:	314.5 lbs	Physically from SR16
Rear weight:	313.5 lbs	Physically from SR16
Peak brake deceleration:	1.2 G	SR16’s data logger (average).
Tire radius:	~8.75”	SW and physically from SR19
Tire coefficient of friction:	2.32 (@ 12 psi)	Calculated from Calspan tire data
Rotor-pad coefficient of friction:	.35	MIT Motorsports data
Effective rotor radius:	~2.8”	Approximated from Solidworks
Front caliper piston area:	1.23 in^2 (per piston)	Wilwood data sheet
Rear caliper piston area:	1.23 in^2 (per piston)	Wilwood data sheet
Wheel base:	61”	Solidworks
CG height (measured from bottom of tire):	~10”	Estimated
Horizontal distance from CG to front wheels:	~30.5”	Estimated from weight
Horizontal distance from CG to back wheels:	~30.5”	Estimated from weight

Turns out I can’t use SR19’s weight since it was measured when the car was neither completed, nor had fluids in it, thus the measurement would be too far off, so I can’t use it. Back to SR16 I go. Speaking of missing data, I’m going to have to look up all the variables myself since none of the other leads know the information. It’s a little irritating if I admit it.

It’s been 30 minutes and I’m still trying to fill this thing out. Turns out it’s not nearly as simply as I thought it would be… Understatement of the century. The master assembly for SR19 does not have any mass properties to it, so the computer has no clue where the CG is. Adding to my woes, Hoosier tire is apparently run by morons, since they evidently have no clue what the coefficient of friction is for their own compounds. Because, you know, that’s asking too much. (Should have bought Pirellis or something). Anyhow, I’m trying to do some research to find any tire data I can, or even just where to find it. By the way, the tire compounds SR19 is using are Hoosier 18x7.5-10 R25Bs.

Ok, so I actually found out that the team purchased tire data from Calspan on these exact tires. And that the coefficients of friction that I need are in there some were. So after an hour of searching for the documents and spending the next half hours ~~drowning~~ swimming through a literal ocean of data and spread sheets I think I have finally found the information that I need. I haven’t found it yet, but I’m going to keep looking tomorrow. Just so that I don’t forget where the files are, here’s the “address”:

› This PC › engr_SAE_share (\\ad.wichita.edu\engr) (X:) › Formula SAE › Engineering › Suspension Design & Reference Files › Tire Data › Round 6 Data › Hoosier 18.0 x 7.5 -10 R25B

1/8/20

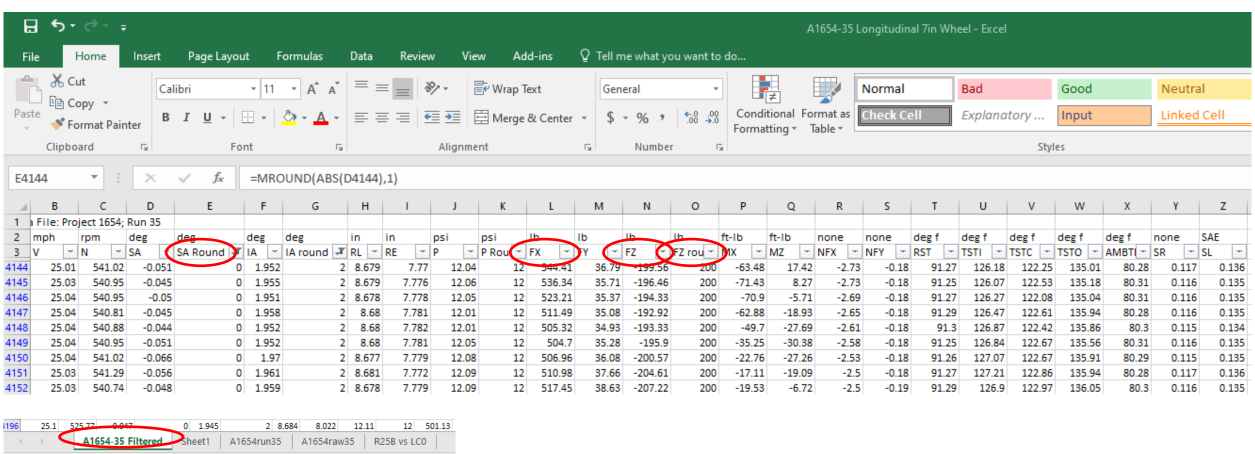
I’M still working on trying to locate the particular tire data I need. I’ve just contacted some older members (the ones who got the data—Ryan mostly), but I haven’t gotten a response yet. Rather than twiddle my thumbs and not be productive I’m going to research how to find the CG of the car, as well as try to find the data for the rotor-pad coefficient of friction. For future reference, the brake calipers SR19 is using are Wilwood GP200 calipers ()https://www.wilwood.com/calipers/caliperlist?subname=gp200%20caliper, and the pads are Wilwood CM high performance brake pads (https://www.wilwood.com/brakepads/brakepadsprod?itemno=150-12128k).

Also, it would appear that finding the coefficient of the pads on the rotors is also not a simple matter, since Wilwood won't provide any data. Besides that, brake pads are evidently sensitive to heat, so I couldn't just find a basic number and run with that. The way MIT did it was to build their own dynamometer using their own equipment to test their brakes. The calipers they used are the same that we are using, so that is slightly helpful. I don't know yet if this is relevant, but they determined that the minimum coefficient of friction was .32. And oh good (no sarcasm) they are using the same brake pads as us. MIT are using 4130 steel alloy for their rotor material. MIT seems to be rather forth coming with their data, so I might try contacting them tomorrow to see if I can ascertain their dynamometer data. We'll wait to see what they say. Also, before I go, Ryan, the old team member who purchased the Calspan data just got back to me, and he said he'd hunt around for what I'm looking for. (P.S. As of right now I'm still filling out the data table above).

1/9/20

Right, so last night Ryan got into contact with me. He said that he had looked over the copy of the data he had on his PC and he told me that I would need to look at run 1654-35, and calculate the coefficient of friction from the FZ and FX numbers at zero slip angle and the pressure that we are running. He told me I can calculate that with the following equation: $\frac{|FX|}{|FZ|} = \mu$. He also noted that the number should be greater than 1 since tires create force, and that it will vary by slip ratio, and to just average them all. I'll go ahead and try to start doing that now, probably in Excel, but I haven't used Excel in a while, so I may have to retrain myself.

Once again this was not as simply as I had hoped it would be. So I talked with Ryan some more and got some more things cleared up. The data I need to be looking at is the A1654-35 Longitudinal Run for 7in wheels, on the "filtered" tab. The columns I need to look at are SA Round (slip angle rounded) for SA Round=0, and the FX and FZ column (for FZ either the rounded or unrounded, depending on the accuracy desired). I'll drop a screenshot below for easier reference:



	B	C	D	E	F	G	H	I	J	K	L	M	N	O	P	Q	R	S	T	U	V	W	X	Y	Z
1	a File: Project 1654; Run 35																								
2	mph	rpm	deg	deg	deg	deg	in	in	psi	psi	lb	lb	lb	lb	ft-lb	ft-lb	none	none	deg f	deg f	deg f	deg f	deg f	none	SAE
3	V	N	SA	SA Round	IA	IA round	RL	RE	P	P Rou	FX	FY	FZ	FZ rou	MX	MZ	NFX	NFY	RST	TSTI	TSTC	TSTO	AMBTI	SR	SL
4144	25.01	541.02	-0.051	0	1.952	2	8.679	7.77	12.04	12	544.41	36.79	-199.56	200	-63.48	17.42	-2.73	-0.18	91.27	126.18	122.25	135.01	80.28	0.117	0.136
4145	25.03	540.95	-0.045	0	1.955	2	8.679	7.776	12.06	12	536.34	35.71	-196.46	200	-71.43	8.27	-2.73	-0.18	91.25	126.07	122.53	135.18	80.31	0.116	0.135
4146	25.04	540.95	-0.05	0	1.951	2	8.678	7.778	12.05	12	523.21	35.37	-194.33	200	-70.9	-5.71	-2.69	-0.18	91.27	126.27	122.08	135.04	80.31	0.116	0.135
4147	25.04	540.81	-0.045	0	1.958	2	8.68	7.781	12.01	12	511.49	35.08	-192.92	200	-62.88	-18.93	-2.65	-0.18	91.29	126.47	122.61	135.94	80.28	0.116	0.135
4148	25.04	540.88	-0.044	0	1.952	2	8.68	7.782	12.01	12	505.32	34.93	-193.33	200	-49.7	-27.69	-2.61	-0.18	91.3	126.87	122.42	135.86	80.3	0.115	0.134
4149	25.04	540.95	-0.051	0	1.952	2	8.68	7.781	12.05	12	504.7	35.28	-195.9	200	-35.25	-30.38	-2.58	-0.18	91.25	126.84	122.67	135.56	80.31	0.116	0.135
4150	25.04	541.02	-0.066	0	1.97	2	8.677	7.779	12.08	12	506.96	36.08	-200.57	200	-22.76	-27.26	-2.53	-0.18	91.26	127.07	122.67	135.91	80.29	0.115	0.135
4151	25.03	541.29	-0.056	0	1.961	2	8.681	7.772	12.09	12	510.98	37.66	-204.61	200	-17.11	-19.09	-2.5	-0.18	91.27	127.21	122.86	135.94	80.28	0.117	0.136
4152	25.03	540.74	-0.048	0	1.959	2	8.678	7.779	12.09	12	517.45	38.63	-207.22	200	-19.53	-6.72	-2.5	-0.19	91.29	126.9	122.97	136.05	80.3	0.116	0.135
4196	25.1	525.77	0.047	0	1.945	2	8.684	8.022	12.11	12	501.13														

1/10/20

I spend most of yesterday crunching the numbers on the tire data, and I have finally found some concrete numbers. I'll also leave screen shots but just for brevity these are the numbers:

- At 14 psi, μ=1.983175
- At 12 psi, μ=2.316526
- At 10 psi μ=5.023276

	A	B	C	D	E	F	G	H
1	psi	FX	FZ	FX ABS.	FZ ABS.	μ (tire-road)		μ avg.
2	14	444.73	-190.95	444.73	190.95	2.329039015		1.983175
3	14	442.42	-197.81	442.42	197.81	2.236590668		
4	14	448.73	-202.01	448.73	202.01	2.221325677		
5	14	459.88	-202.49	459.88	202.49	2.2711245		
6	14	468.5	-201.32	468.5	201.32	2.32714087		
7	14	471.49	-198.48	471.49	198.48	2.375503829		
8	14	466.81	-194.88	466.81	194.88	2.395371511		
9	14	456.78	-191.6	456.78	191.6	2.384029228		
10	14	449.79	-187.15	449.79	187.15	2.403366284		
11	14	450.21	-185.56	450.21	185.56	2.426223324		
12	14	452.53	-187.43	452.53	187.43	2.414394707		
13	14	451.48	-192.33	451.48	192.33	2.347423699		
14	14	445.7	-197.36	445.7	197.36	2.258309688		
15	14	441.7	-200.89	441.7	200.89	2.198715715		

	A	B	C	D	E	F	G	H
1	PSI	FX	FZ	FX ABS:	FZ ABS.	μ (tire-road)		μ avg.
2	12	544.41	-199.56	544.41	199.56	2.728051714		2.316526
3	12	536.34	-196.46	536.34	196.46	2.730021378		
4	12	523.21	-194.33	523.21	194.33	2.692378943		
5	12	511.49	-192.92	511.49	192.92	2.651306241		
6	12	505.32	-193.33	505.32	193.33	2.613769203		
7	12	504.7	-195.9	504.7	195.9	2.576314446		
8	12	506.96	-200.57	506.96	200.57	2.52759635		
9	12	510.98	-204.61	510.98	204.61	2.497336396		
10	12	517.45	-207.22	517.45	207.22	2.497104527		
11	12	526.36	-206.73	526.36	206.73	2.546122962		
12	12	534.21	-204.78	534.21	204.78	2.608702022		
13	12	537.45	-199.92	537.45	199.92	2.68832533		
14	12	533.02	-195.81	533.02	195.81	2.722128594		
15	12	522.22	-193.44	522.22	193.44	2.69964847		

	A	B	C	D	E	F	G	H
1	psi	FX	FZ	FX ABS.	FZ ABS	μ (tire-road)		μ avg.
2	10	526.17	-199.67	526.17	199.67	2.635198077		5.023276
3	10	526.17	-195.86	526.17	195.86	2.686459716		
4	10	526.17	-192.79	526.17	192.79	2.729239068		
5	10	526.17	-192.11	526.17	192.11	2.738899589		
6	10	526.17	-193.28	526.17	193.28	2.72231995		
7	10	526.17	-196.96	526.17	196.96	2.671456133		
8	10	526.17	-201.1	526.17	201.1	2.616459473		
9	10	526.17	-204.72	526.17	204.72	2.570193435		
10	10	526.17	-206.07	526.17	206.07	2.553355656		
11	10	526.17	-205.31	526.17	205.31	2.562807462		
12	10	526.17	-200.74	526.17	200.74	2.621151739		
13	10	526.17	-194.89	526.17	194.89	2.699830674		
14	10	526.17	-189.43	526.17	189.43	2.777648736		
15	10	526.17	-187.19	526.17	187.19	2.810887334		

The file location for these excel documents are here:

› This PC › engr_SAE_share (\\ad.wichita.edu\engr) (X:) › Formula SAE › Engineering › FSAE 2019-2021 › SR21-CC-Cockpit and Controls › SR21-CC-A000-Research and Technical Documents

So now for some analysis, as you can see tire pressure greatly matters in terms of coefficient of friction for various tire pressures. After talking with some other old members, they confirmed that tire coefficients change pretty drastically when the tire pressure is low. I also confirmed with them that the numbers made sense, so I evidently didn't screw up too much. But this leaves me with a problem: since the pressures vary so greater for slightly different tire pressure, which one do I use in my calculations for the brakes? Since the last suspension and brakes lead doesn't know what we're running at, nor has the current S&B lead chosen any if their design requirements, this makes my job doubly difficult. Hence, I'm choosing to use the data from the 12 psi run, since it is the middle pressure, and presumably the one we are most likely to use. Again, I could be wrong, but even if I'm wrong, I can always change the master cylinder size to compensate. Also, I can partly counteract any inaccuracies by making the pedal variable in terms of ratio, and fortunately I have a decent idea about how to do this. But anyway, I hope this adequately explains my reasoning.

Also, I don't need to contact MIT because they actually have the data I need posted later in their document. I'll post a screenshot below:

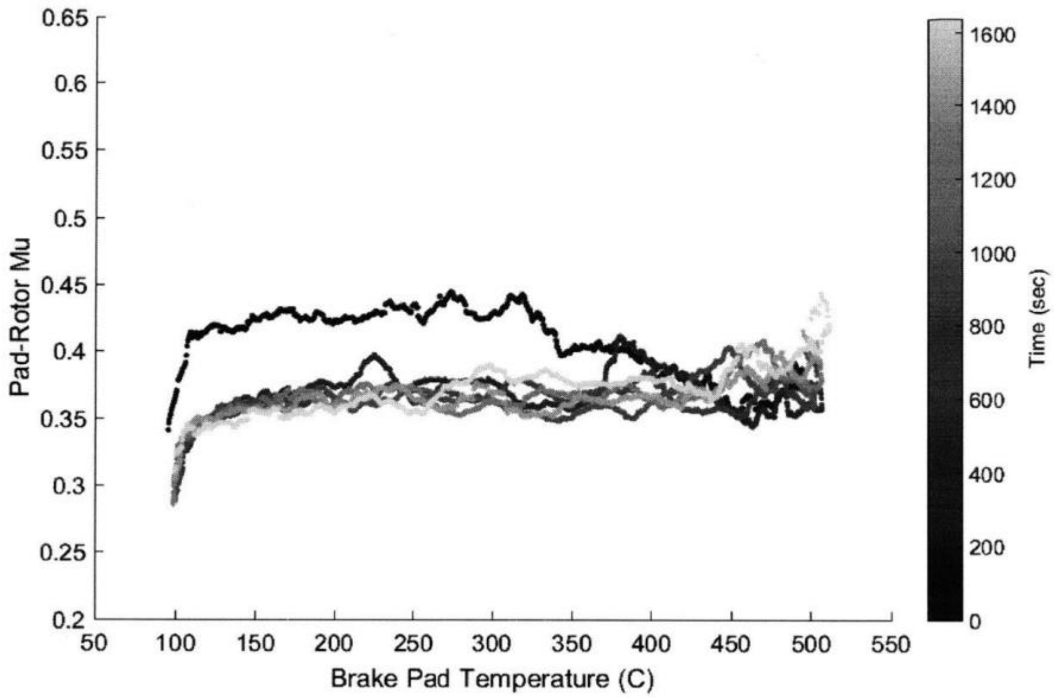


Figure 8. Mu vs temperature for 4130 alloy steel rotors and CM compound brake pads. Shown in the colorbar is time, showing how the data changes from run to run. Note how the first run is an outlier in the data, indicating that there is a "run in" period before the brakes become more consistent. Subsequent runs are very consistent with each other, indicating that this material and brake pad combination is very resilient to brake pad wear.

Image credit to MIT Motorsports

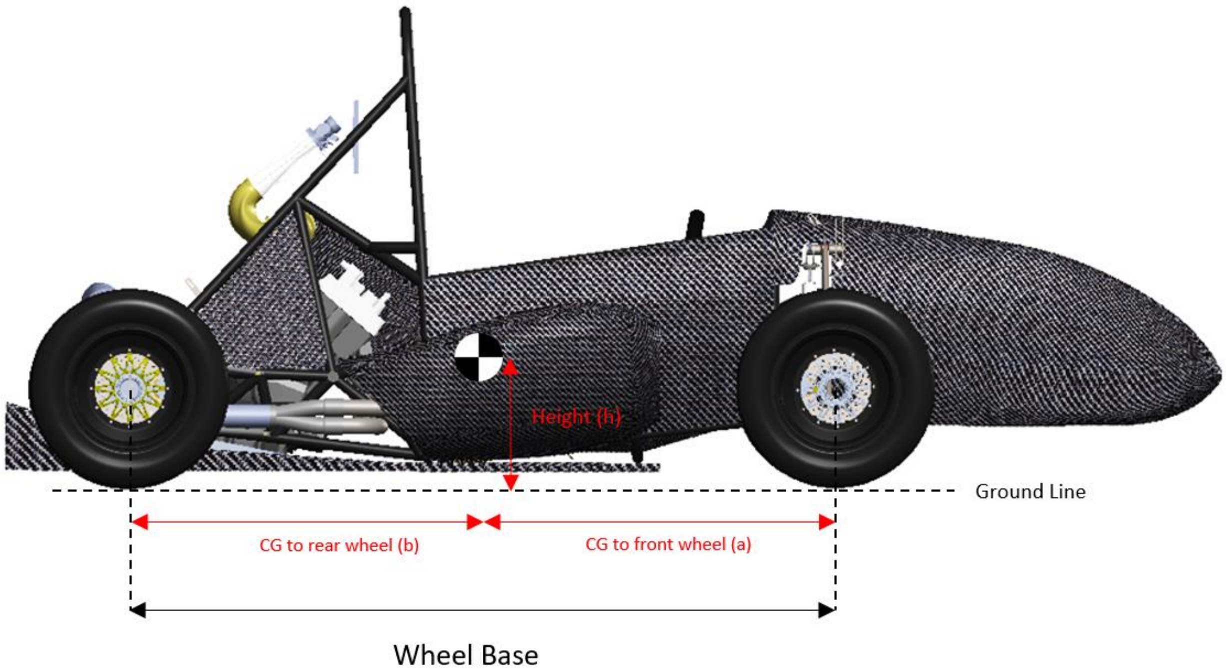
So now I'll have to find the average temperature for the brakes and use this table to find out what the coefficient should be. I might also speak to the S&B lead about using 4130 alloy for the brake rotor material since we have the data for it right here. But, it would appear that .35 is the average for most of the temperature ranges, so that's what I'll use.

The last bit of information I need is the CG data. From some cursory glances at a few articles it would appear that you can measure the cg of a car by tipping it and using some trig to determine where the CG center is. But, we don't have the equipment to do that. I think I can make some assumptions and approximations for the CG, but I'll have to read more into it. I think this will be all for tonight. Hopefully tomorrow I can get the CG data and start making my calculations. Then I can select the appropriate cylinders and ratios.

1/11/20

Ok, so there is no easy and quick way to measure car CG (especially since we don't really have the equipment), so I'll just use approximations from data found from other teams. After scanning the somewhat unhelpful FSAE forum and the somewhat more helpful FSAE reddit posts, the average CG height for most FSAE teams is ~10-11" from the ground. 11" especially with a driver in the car, so that's what we'll go with.

For the CG of the car w.r.t. the horizontal distance between the front and rear wheels, I haven't found out anything from researching the web, but the more I think about it, the more I realize that it's so simple I don't need to look it up. Since the car is so well balanced that it has near 50/50 weight distribution, it stands to reason that the CG would be right in the center of the wheel base. I.e. half way between the front and rear wheels, equidistant to each. I'll put a diagram below for clarity:



In this case: Wheel base= 61", and a= b, a= (Wheel base)/2, and h=11".

As of right now, the table of page 11 is completed to the best accuracy available to me. I could *try* to find out an accurate number for the CG, but I feel pretty confident in my estimations. Plus, I've spent enough time on calculating this stuff for the brakes. I have to remember that there are other things I need to do, and I *am* on a time constraint. The sooner I can get everything designed, the sooner the car can be built. With that in mind, I'm going to end here. I'll start crunching the numbers soon and post my results. For now I need to start looking into some of the more expensive components for the cockpit since our budget hearing is coming up.

Brake System project continues on page 19.

Cockpit Ancillaries (Initial Considerations):

I'm going to try to put together a short list of some of the biggest ancillary components the cockpit will need (so that we can purchase them fairly soon) and compare similar products. That way I can hammer out more of what I'll be working with further down the road.

The two most expensive components that the cockpit will need to purchase (as opposed to building) is the steering rack and the dash display. The steering rack *could* be built by us, but cutting the rack and pinion would be difficult, require tools we don't own, require skills we haven't acquired, and time we don't have. Therefore I as lead elect to purchase a commercially available rack for these reasons. The dash display couldn't be built by us anyway, so the idea of making one is out. There are a number of different ways that I could display the vital data to a driver and team (such as a motorcycle tachometer that has been used in cars about a decade ago), but the only thing that really makes sense is to have an electronic LCD or LED display somewhere in the cockpit. And if we take a short view back to the past (on page 1 to be specific), the sub-team goals stated that I wish to incorporate a display in an in-house made steering wheel. This is still my goal. This does limit my selection of possible models, however, as the wheel can realistically only be so big, meaning that I will have to source an appropriately sized model to fit in the wheel.

I'll begin with a comparison of the steering racks, as there are less of those, making the decision a little simpler. I fear the dash display will be more difficult to decide on.


Image credit to Kaz Technologies

The first to be considered is the Kaz Technologies FSAE steering rack. According to the company (https://www.kaztechnologies.com/fsae/steering-rack/) it is a steering rack designed especially for FSAE teams. It weighs 3 pounds (according to the company) and costs $670. It does not include the spline coupler, but it does include the gold retaining clamps that holds it in position. This is the rack we have used at least twice in the past. It's a decent rack and I don't really have any complaints about it except that it is heavy. 3 lbs is quite a bit of heft. The body is made of aluminum with steel rack and pinion for strength. It has a rack travel of 3.25 in stop-to-stop. It also has an adjustable pre-load, making it easier or more resistive to changes in steering direction. It can also include an optional built in steering sensor. One of the more useful features it has is multiple grooves in the arms of its body, which allows the retaining clamps to be positioned in multiple configurations. It does include some Solidworks files for the parts, but they're not very good because you can't manipulate the model well. I ended up making a better model for in about an hour, though it is fairly basic. A couple more hours and it would be pretty spot on.

The next rack I'm looking at is a rack from a company called Formula 7 (http://www.formula-seven.com/shop-products/steering-racks-w-rotative-sensor/). There are several sizes for this rack (400mm, 450mm, 500mm) and each weigh 870g (1.92 lb), 940g (2.07 lb), and 1010g (2.23 lb) respectively. It costs anywhere from €1,000-€1,250, ($1112-$1390) depending on the size and features chosen. The body is made from aluminum and carbon fiber, which contributes to it's lightness. It also has machined steel rack and pinion components. It does not include any fastener hardware, unlike the Kaz Tech rack, meaning we would need to machine our own. It also does not have adjustable positioning, unlike the Kaz Tech rack. It has a stop-to-stop travel of 72mm (2.83 in), and is rated to take 1.88kN of axial load. It also has an optional built in Hall-Effect sensor built in.

1/14/20

The last rack that I'm considering is this one. It is the "NARRCO" rack from fsaeparts.com (https://fsaeparts.com/products/narrco-racks?variant=15279234220075). The racks weighs just 1.1 lbs (in its smallest version) and costs $600 and comes in various sizes. It doesn't give much information about the total rack travel. It appears to be made from almost all aluminum (hence the weight savings). It comes with the rubbery/plastic-y looking racks stops and requires one to purchase separate mounting hardware. I have several reservations about this rack however. First off is the lack of a splined coupler. It's smooth. I have no idea how you are supposed to attach a coupler to it. Second, I have never heard of this company before, and did not see it being used in any of the FSAE cars I saw during Lincoln 2019. It's pretty obscure, which makes me raise my eye brows. Also, I have heard that aluminum internal components on a steering rack tend to wear out quickly, leaking to excessive slop by the end of endurance. Not ideal.

Below I have created a decision matrix to help to determine which steering rack is the best. I have compared each product by the different categories written in the top row. I then rated them in each category by giving the most desirable one a 3, and the least desirable a 1. I will then simply add the "points" each item scored overall, and the one with the most amount of "points" is considered the best (mathematically anyway).

Product:	Weight:		Total Travel:		Rack Speed:		Gear Material:		Sensor		Cost:	
Kaz Tech:	3lb	1	3.25 in	3	4.71"/rev	3	Steel	2	Optional	2	$670	2
Formula 7:	2.23lb	2	2.83 in	2	3.14"/rev	1	Steel	2	Optional	2	$1120	1
NARRCO:	1.1lb	3	2.5 in	1	3.46"/rev	2	Aluminum	1	N/A	1	$600	3

Results: Kaz Tech= 13. Formula 7= 10. NARRCO= 11. Thus by decision matrix, the Kaz Tech rack is the best option available. Which I would concur with. It's been reliable and robust, and we know how to use it, mount it, and how it feels and behaves. I'll give these results in the next engineering brief.

1/15/20

Next, it's time to decide which dash display I'll be using. I obviously need to work with the electrical lead on this part (which is currently James Lott) on determining what the dash needs to have to connect to our data logger/ecu. Obviously if something I'm considering doesn't have something we need it to, we can't use it.

The first dash I'll be looking at is this NT Dash from XAP Technology (https://store.xap.fr/en/21/10035-dash-nt5-110-standard-version.html). It has a 5" TFT LCD screen with 16 programmable RGB indicator LEDs. It weighs 550 grams (1.21 lb.) and costs €1,150 ($1283). It uses CAN communication and is compatible with most major data logging systems (AIM, Motec, etc.). The screen comes with ~30 premade screen configurations, but with a €60 ($67) software, one can make any type of configuration they wish. Probably not necessary though. The unit has an aluminum body with an apparently removable carbon fiber bezel. As this unit is not a dash/data logger unit, it is thinner than some of the other units I'll consider, making it more ideal to place in a custom made wheel. Also, the placement of the electrical output won't be in the way of the quick release I'll need to fit by the rules. The downsides to this unit are that it is a relatively unknown manufacturer in the FSAE world, though the company does have a considerable pedigree. It is a French electronics company specializing in race

car electronics and produces the steering wheels for the gen 2 Formula 2 and Formula e cars. So it is going to be a bit of a grey area in terms of advice/ support/ information from other teams, it does seem to be a solid and helpful company. That being said, it may be better to use a dash that we are already familiar with and know how it works (such as a Motec or AIM unit).

Image credit to Motec

The next dash I'm looking at is the one we currently are using on SR19. This is the C125 dash/data logger from Motec. It costs northwards of $1,500 (depending on the discount given), and it weighs 342 grams (0.754 lb.). Like the XAP, it has a 5" TFT screen with programmable LEDS. It does not have the separate lights on the side like the XAP. Also the shape of the dash makes it inconvenient to fit into a steering wheel, especially with the curved LED top. It is also thicker than the XAP model and its connector placement makes it difficult to have a quick release, which is a must. The benefit to this dash is that we have used it before, so there won't be any major electrical changes to be made will be minimal. Again, the LEDs are programmable, as well as the screen configuration. Also, this model has built in data logging. Which isn't very helpful since having the accelerometer in the wheel throws off the data. In my opinion it's better to have a split dash and logger for this reason.

The next dash I'll consider is the AIM MSX 1.2 dash (https://www.aim-sportline.com/en/products/mxs-1.2/index.htm). Like the other dashes, the screen is a 5" TFT with integrated shift and warning lights. This unit is by far the biggest at nearly 6.5" wide. It weighs 530 grams (1.16 lb.) and costs more than $1,140. And just like the others it has a configurable display. It is also CAN compatible, as well as being from the same manufacturer of the separate data logger, which should ensure that there are no connectivity issues. This is probably the biggest benefit to this dash. The downsides include the very large size, thickness, and bad placement of the electrical connectors. This unit is too big to fit in a practical steering wheel, without making the wheel so big it takes up a lot of cockpit leg room. In addition, it is a dash/logger system as well, meaning that the accelerometer would, again, be useless in a moving wheel.

The final dash I'll consider is this AIM MXm (https://www.aim-sportline.com/en/products/mxm/index.htm). This is the simplest dash available. It weighs 330g (.73 lb.) and costs $1,100. This screen is not as configurable, as it is an old school style LCD with a color-changeable background. 5 small LED shifts lights on the top, which 2 warming lights in the corner. Same compatibility benefit as the previous AIM system. But it is not quite as small as it looks. It's about 5.4" in width and rather thick, meaning it uses a lot of space that could be used for things like buttons. The high

costs and low tech makes it one of the least desirable choices, but I'll run it throw the matrix and see what happens.

These are of course, all things that need to be considered when choosing the appropriate dash. I'll make another decision matrix to help sort out which is the best one. The same scoring system from before still applies:

Product:	Cost:		Weight:		Size (wide):		Size (thickness):		Warning Lights:		Data Logging:		Configurable:		Integrateable:
NT Dash	$1,283	2	1.21 lb	1	6"	2	1.25"	2	Yes	2	No	2	Yes	2	4
C125	$1,500	1	0.754 lb	3	5.3"	4	1.11"	4	No	1	Yes	1	Yes	2	3
MXS	$1,140	3	1.16 lb	2	6.5"	1	1.34"	1	Yes	2	Yes	1	Yes	2	1
MXm	$1,110	4	0.73 lb	4	5.4"	3	1.12"	3	Yes	2	No	2	Sorta	1	2

Here's the results: NT Dash= 17, C125= 19, MXS= 13, MXm=21.

The results are actually kinda interesting. The C125 and the MXm have the most points, even though those are my least favorite ones. Part of this is the fact that none of these categories are weighted. In other words some of the categories are more important than others, but the matrix doesn't account for that. One example is cost. They are actually pretty similar in price, but again, the matrix doesn't account for that. The weight is also not nearly as important as how big the unit is and how well it can be integrated into the wheel. Bearing that in mind, I'm going to go with the NT Dash from XAP Tech. The size is a bit large, but a very similar fits in their premade wheel I'll show below:


Image credit to XAP tech

This is the Formula 2 wheel I mentioned earlier. The screen is the same size. The only difference is the lack of the warning lights on the side of the car, which have been relocated to the top of the wheel in a triangular cluster. This is the type of wheel I'm aiming to make. We can't use this one however, because of the indentations in the circumference of the wheel. The wheel might be a bit too big anyway. I'll just have to check with James to ensure the screen will work for him. Then I can work on ordering it. My fall back will be the Motec C125 and I'll work around the difficulty.

1/16/20

Brake System: (Continued from page 15).

I just finished calculating the normal force on the front and rear tires during maximum braking. I used the numbers I posted on the chart on page 15, and used the equations found in the Purdue video I posted earlier to calculate this:

Braking Torque:

$$\tau_{Braking,f} = r_{tire} \cdot F_{z,f} \cdot \mu_{tire} \quad (1)$$

$$\tau_{Braking,r} = r_{tire} \cdot F_{z,r} \cdot \mu_{tire} \quad (2)$$

$$r_{tire} = (0.73\ ft)$$

$$F_{z,f} = 437.13\ lb$$

$$F_{z,r} = 190.87\ lb$$

$$\mu_{tire} = 2.32$$

(1) Front:

$$\tau_{Braking,f} = (0.73\ ft)\cdot(437.13\ lb)\cdot(2.32)$$

$$\boxed{= 740.32\ lb\cdot ft}$$

(2) Rear:

$$\tau_{Braking,r} = (0.73\ ft)\cdot(190.87\ lb)\cdot(2.32)$$

$$\boxed{= 323.26\ lb\cdot ft}$$

$$\boxed{F_1 = 437.13\ lb}$$
(front)

$$F_2 = W - F_1 = (628\ lb) - 437.13\ lb$$

$$\boxed{F_2 = 190.87\ lb}$$
(rear)

As you can see from the calculations, during maximum braking the normal force on the front wheels is 437.13 lb, and on the rear it's 190.87 lb. ***Note these numbers are representative of both wheels for the front and back, not individual tires.** I can now use this to calculate the braking toque. However, there is an ice storm headed towards Wichita, and I need to get home before it hits, so I'll have to finish this tomorrow.

1/17/20

Well, I'm back at it again. Last night when I was driving home I started to realize that I didn't know what the best ratio for the balance bar to be set at was. I've heard somewhere that 60/40 front to rear is the general set up most people start off with. But then I realized I could calculate the percent of front to back braking percentage from the numbers I have just calculated last night. So I divided the braking normal forces by the original (static) normal forces to find the percentages:

$$\frac{437.5}{628} = .697 \approx 70\% \qquad \frac{190.87}{628} = .303 \approx 30\%$$

So if this calculation is correct, then it would indicate that we should have 70% of the distribution to the front, and 30% to the rear. But just now after I have done these calculations, I've started to think that it wouldn't depend on normal force for the distribution for the brake bias, but rather on the normal force distribution percentage. So instead I'll see how those numbers turn out and then I can reevaluate.

Above are the braking torque calculations. The front two wheels require 740.32 lb-ft of torque (370.16 lb-ft each) to achieve maximum braking, and the rear needs 323.26 lb-ft of torque (161.63 lb-ft each).

As mentioned just above, I'll try calculating the brake bias percentage based off of this, and it should give the correct number:

$$\frac{740.32}{1063.58} = .707 \approx 70\% \qquad\qquad \frac{323.26}{1063.58} = .303 \approx 30\%$$

So it turns out that it doesn't matter if you use normal force or braking torque to calculate the ideal brake bias ratio. Which makes sense if you think about it.

Now that I have the braking torque calculated, I can now use the next set of equations (on page 9) to determine the ideal pedal ratio. But before I can do that I'll need to get some volunteers to head to the gym (either the Heskett center or the newly opened YMCA) to do some human testing to determine the TDIF. I'll need to get that information and average it to put in the correct data. It's Friday so we do have the weekend, but we're finishing up 19 and are planning on taking 16 out for a test run after getting her fixed. So that's probably out. Monday is MLK Day, which might work, and school starts the next day, so it might be best to get it done before the doom of classes resumes. I'll probably work on some other stuff while waiting.

Brake System project will continue on page 24.

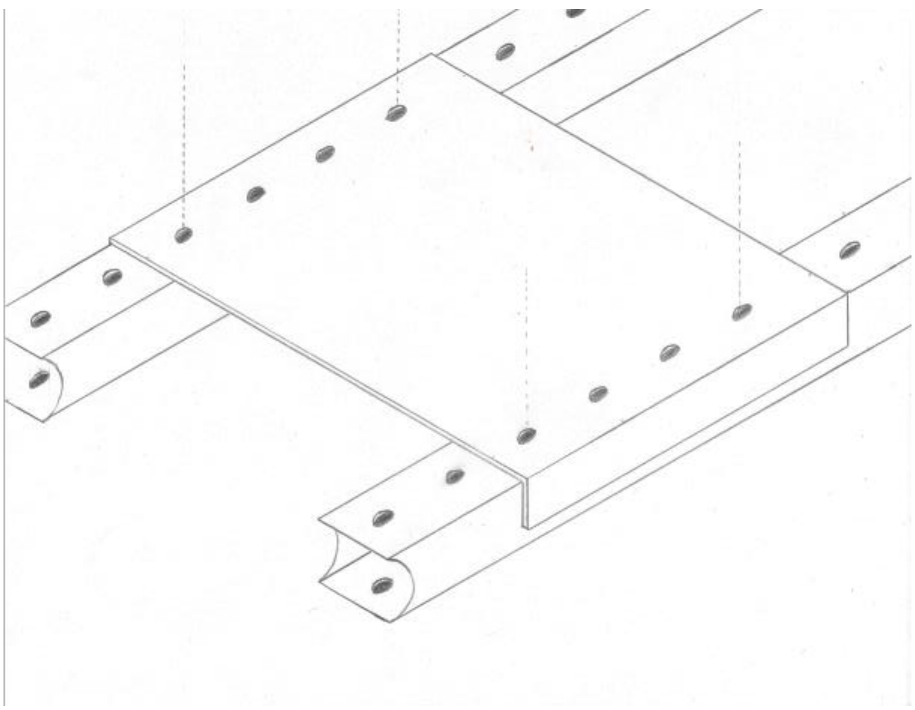
Pedal Mounting Plate (Initial Design Ideas):

While I'm waiting for the TDIF info, I thought I'd work on some ideas for mounting the pedals to the car. Our traditional mounting style has included two separate rails to which the pedals are independently mounted to by bolts. In SR16 they used aluminum aircraft seat sliders, but those turned out to be incredibly inconvenient to adjust. I should know since my very first assignment on this team was to try and adjust them. It took me awhile (and some help), I'll admit. SR19 used two separate tube rails. The brake rail was made of steel and welded to the car. The other was made from aluminum and was bolted onto mounts. The wholes for adjustment were on the side of the rails, underneath the forward cockpit closeout panel (floor). This, combined with some warpage from welding, making installing and removing the pedals a nightmare. So the question becomes: *how do I do it better?* I've got some ideas. My main one is something I saw and liked at competition:

In the left photo: This is the pedal mounting solution I liked the best. It's a plate to which both pedals are attached, and the plated is slide back and both on two rails on the side. They are attached using two quick release pins on either side of the car. These can be seen on the close up on the right. I've checked the rules out, and they say you can actually use they fasteners. I'll drop a close up pic of them below:

These meet the requirements for positive locking fasteners in the rules. Thus, we should be able to use them, which are a lot easier to use than bolts. This particular one pictured is available from Grainger: (https://www.grainger.com/product/3JCX5?gclid=Cj0KCQiA04XxBRD5ARIsAGFygj_pJEjjniey9tsb7nMa_fu97OJCPUAT6d_o2-Rok9x1Iu48PaDOfVUaAkZ9EALw_wcB&cm_mmc=PPC:+Google+PLA&ef_id=Cj0KCQiA04XxBRD5ARIsAGF

ygj_pJEjjniey9tsb7nMa_fu97OJCPUAT6d_o2-Rok9x1Iu48PaDOfVUaAkZ9EALw_wcB:G:s&s_kwcid=AL!2966!3!281698275558!!!g!470847567040!)



I also like the movable plate idea. It will allow both pedals to be stationary mounted, while still allowing adjustability. I'd still like to use a similar rail system like in SR19, but modified. Below is a crude sketch of what I'm thinking of:

In the depiction above you can see the two rails I mentioned. The wholes for the fasteners have been moved from the horizontal position in SR19 to the vertical position. This will allow easier removal of the pins I'm going to use. The plate that rides on the rails is where the brake and throttle pedal will mount. I'll have to look into different materials to make it out of, and also look into removing material for weight savings. Most likely I'll have to try several with FEA to determine whether they'll be strong enough.

1/20/20

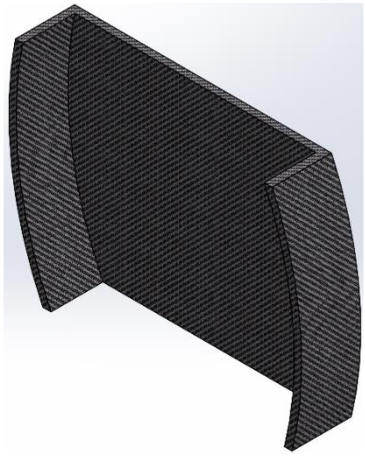
Quick update: I finally got the stuff I needed to create the custom length throttle cable. But when I put it all together, I've found a few problems. **1**: the way the throttle pedal was designed, the bottom does not have enough swing to pull the throttle cable enough, meaning we aren't getting full throttle, even at max throttle input. This is obviously a big problem. **2**: The return springs used to help the throttle quickly shut are contacting the cable hammer (the thing that actually pulls the cable) when you get to about half throttle input, making it harder to push than need be. **3**: the carbon fiber end of the cable that holds the little aluminum piece that in turn holds the end of the throttle cable jacket is flexing under throttle input (see below) and is making the first problem a little worse.

In reality, this pedal was a bit of an example of what not to do. It's a combination of bad initial design (not my design, I'm actually not sure who designed it) and poor implementation (my fault). The throttle never had a cable pull system because they had initially intended to have an electronic throttle body, which would have used position sensors, therefore it didn't need to have a throttle hammer to begin with. And I, without realizing this, didn't think to redesign it. Instead when I realized why they had done what they had done, I made some modifications to convert it to mechanical. Much to my chagrin they evidently didn't work. I'll have to keep working at the problems and see what I can do about it. I can take some consolation, as I am *definitely* not the only one who made some mistakes. In any case, this needs to be addressed in future designs. I think I'll need to calculate how much the throttle cable needs to be pulled, then determine the size of the pedal and placement of the fulcrum to allow adequate swing to the throttle hammer.

One last note for today: classes resume tomorrow, and this semester will be a heavy load. Hopefully I can balance everything I need to do. Expect some changes to the flow and schedule of the notes.

1/25/20

Update: Not much design progress to report. I've been busy settling back into the rhythm of classes and homework. Beyond that I have been working hard on trying to make the clutch cable work. Which hasn't been easy. The ends keep wanting to not solder to the cable. It turns out that material



choice makes a big difference. Stainless steel will NOT stick to regular solder. Only silver solder which isn't cheap, or easily available. Only non stainless steel cable or cable ends will stick. And it has to be carefully done so that the solder actually gets into the whole and sticks. I thought I had it finished yesterday, only to test the clutch and it pop out. Which is frustrating. To say the least. I'm almost too disheartened to keep designing. I've also been working on some of the graphics/art for the t-shirt/promotional items, so there's something. I've just got those final few things to do and my tasks on SR19 and I'm finished.

Classes shouldn't be too bad this semester so hopefully I can get a lot done.

1/29/20

Brake System: (Continued from page 21).

Quick update, I've been working on the captive heel and toe cups for the pedals, and this is what I have come up with so far:

These are the captive cups I have come up with so far. As one could probably tell from the name "captive" the point of these pedal cups is to provide a surface to help hold the shoe in place during hard cornering. Having the walls on both sides of the pedal face will allow the heel and toe of each shoe to remain firmly in place no matter how much cornering force the car will generate. I've had that problem in SR16 when I was driving it. The pedal face is entirely flat, so my foot would occasionally slip off it during a hard corner, making me loose time trying to find the pedals before I could accelerate again. This is annoying when it happens on the throttle, but if it happened during braking, that would be really bad. Thus this solves that issue and makes the car just a tad safer to drive.

The design is pretty simple, but then again, it really don't need to be. They were designed after Formula 1 pedals I found online. I liked the idea and traced a pattern of the largest driver shoe were would be using and loaded the trace into Solidworks, then make the part with the trace as an outline. As

you can tell from the picture, They're to be made from carbon fiber. Either pre-preg or infusion, it doesn't matter too much which. The molds for these should be fairly simple to make from either aluminum or foam. Both should probably be machined. But that's later on down the road. *not the cups in the picture are for the right foot and thus actually meant for the throttle. The brakes will be the same, but mirrored.*

I haven't added anywhere to place fastening hardware yet because I'm waiting to make a 3D printed physical prototype first to check that the design works and I don't need to make any huge changes before I invest time into making the fastening device. Also, although it isn't modeled (yet) the surface of the pedal will have some grip tape, just to help out. Also, I wanted to have it adjustable to have different pedal ratios.

2/5/20

Update: I've been busy with school and meetings for the team. Not much progress on the pedals. I'll need that driver TDIF data by the end of the week. If I can't get any volunteers, I'll just use myself as a data point.

Beyond that I have bad news. After a meeting with our lab manager Steve over various things including shop safety, we broached the topic of purchasing from overseas… Basically we can't. Not without months of paper work and meetings that is. As co-captain I'll need to meet with the General Counsel (the people who approve these foreign purchases) anyway, so I can find out more then. But I'm pretty limited now.

I bring this up because it does impact my designs for the wheel, shifter and a few other things. Specifically the dash display. The one I want is from XAP Tech, a French company. With no US based vendors. Meaning if I really want the part, I'll have to go through a lengthy process which may not work anyway, because CG needs to approve the purchase, and do a bunch of paper work and agree to all the terms and conditions. Which might not happen. In any case, if I want it, I'd better get started soon.

It also effects my shifter. As you know from reading the team goals on pages 1 and 2 the team are wanting to implement paddle shifters. This red tape will make that a lot more difficult. I'll get into why specifically when I get to the shifter section.

Now, more on the brakes: I've been researching what metals I want to use to make the brake pedal body from. The two most obvious choices are 6061 T-6 and 7075 T-6 series aluminum, as they are extremely light weight and yet very strong. I'm leaning towards using 7075 T-6 in the pedal body because it has double the tensile strength and 1.5 times the shear strength of 6065 T-6 aluminum. The downside is that it is a harder metal to machine, cannot be welded, and is generally more expensive. This should be too much of a problem however, as I shouldn't need to weld it. The fact that it is pretty much as strong as steel for way less weight is good enough reason.

2/7/20

Update: I just got the TIF raw data tonight. We ended up going to the YMCA on campus and using the leg press. I'll put the raw data below. I forgot to take photos, so I'll try to do that at a later date. Basically what happened was we each did about 10-20 reps with varying weight, pressing mostly with the balls of our feet to see what was a comfortable weight to push, which simulated the force required to make the car achieve maximum braking. Below is each student, and the weight they felt they could push continually during a race.

Student Name:	Comfortable Force (in pounds)
Justin W	70lb.
Ben R	70 lb.
Daniel C	72 lb.
Jarrett B	45 lb.
Alan C	50 lb.
Connor P	35 lb.
Angela ?	10 lb.

Based on the average (excluding outlier data) the average force appears to be ~60 lb. Comparing this to SR16's brake pedal, which is 75 lb., this seems rather light. And SR16 isn't a heavy car. Which means I need to make a decision on whether I accept this value, or I use my intuition to tweak the number. I suspect the people who haven't driven SR16 don't realize how little pressure it actually takes to stop the car, and how you don't want a super light pedal. Based on that, the average would be closer to 70lbs, a number I'm more comfortable with. So I'll make the decision as the cockpit designer and go with 70lbs as the TDIF, and base my calculations off of that.

Speaking of calculations, I can now estimate the brake bias using this equation from page 9:

TDIF=F_{driver}

Just to recap, $\tau_{braking,f}$= 740.32 lb*ft and $\tau_{braking,r}$= 323.26 lb*ft (data from page 21). TDIF=70 lb. All of the other necessary data can be found in the blue table on page 11, except for the area of the front and rear brake master cylinders (A_{mc}). That I will have to find when I decide which master cylinders I will need to use. I think I'll use ¾" master cylinder diameter on the front and rear wheels. This is the middle size of the range Tilton offers, which means I can still adjust my pedal sensitivity even if I calculated it wrong. In other words, if the pedal is too light I can add a smaller cylinder, and if it's too heavy I can get a bigger one, all without having to redesign the pedal/pedal ratio. Thus, the area if the master cylinder is $A = \frac{\pi D^2}{4}$. A=.442 in^2 (0.00307 ft^2).

3/14/20

Brakes System will continue on page 29.

Major Update:

Normally, this is the part where I'd write "Quick update:" and proceed to fill you, dear reader, in on what has been happening since my last journal entry. However, I fear this update will be anything but quick. So let's start off with the basics. Shortly after the last journal entry over a month ago I proceeded to take the info I'd been gathering since early January and plugged it into the equations I'd found from MIT. And discovered my numbers were *waaaaaaaay* off. Like, it told me the pedal ratio should be ~30:1. Which is really far off from what it should be. A pedal ratio for a car like ours should be somewhere around 5:1. So the calculated pedal ratio is over 6 times larger than it should be. And I can't figure out where I went so wrong. So let me list out some possible sources of error: (keep in mind I'm writing this about a month after I did my calculations).

1. My approximations were off. I had to use several approximations during the course of gathering the relevant information to calculate the pedal ratio. It is entirely possible that I made a false assumption or a mistake in calculations. This is not the most likely case however, as I used sound

$$F_{driver} \cdot PR \cdot BB_f = \frac{\tau_{braking,f} \cdot A_{MC,f}}{2 \cdot r_{rotor,effective} \cdot \mu_{pad-rotor} \cdot A_{C,f}} \quad (3)$$

$$F_{driver} \cdot PR \cdot BB_r = \frac{\tau_{braking,r} \cdot A_{MC,r}}{2 \cdot r_{rotor,effective} \cdot \mu_{pad-rotor} \cdot A_{C,r}} \quad (4)$$

reasoning and logic to come to these conclusions as well as verifying them as best I could.

2. The tire data assumption was wrong. This is similar to the first case, but related only to the tires. The tire data from Calspan had data sets from 3 separate runs, each run using a different tire inflation pressure (10psi, 12psi and 14psi). The coefficient of friction of the tire from each run varied considerably. This is because tire inflation is related to tire contact patch. A more inflated tire will be more rigid, deflecting less and creating a smaller contact patch, and a less inflated tire will deflect more and have larger contact patch. Contact patches have an impact on the amount of grip. I had decided to use the coefficient of friction from the 12psi set as 12psi was the median pressure. However, when I really think about it, why would we *not* inflate the tires to the highest coefficient of friction available? To put it more clearly, why wouldn't I use the highest coefficient of friction the tires could have to calculate maximum braking? The coefficient of friction for 12psi is around 2, and the coefficient of friction for 10psi is around 5, which is over twice a high. This in turn will change the braking torque. I believe this is one the sources of error in my calculations.
3. The other source of error is in the TDIF. According to MIT, their drivers can comfortably produce 900-1050 N of force on the pedal over and over again. This means 200-235 lbf. Which is way higher than most of our drivers wanted to put into the brakes. This is also one of the biggest sources of error in my calculations. I'll need to consider this further.
4. The master cylinder size may need to change. The MC size does have an effect on the braking TDIF and it is possible I need to recalculate a few times using these using various master cylinder sizes and see if it helps. I've been taking numerical methods and I might be able to use Octave to help do the calculations. Or maybe not. I haven't decided. If I do, I'll of course link to it like I have with all my other reference material.

The good news about all of this is that if I comb through the calculations, I should be able to find out where exactly I went wrong, using the above possible sources of error. The bad news is that I really don't have time to do so. I had basically rage quite after the calculations came out super wrong and decided to come back to it the next week to figure it out. However, the school semester kicked in hard and I suddenly had more tests, quizzes and papers than I knew what to do with. I got so busy I had to take a two-week break from the team just to have enough time to keep up. This is part of the reason I'm only now writing down all this. That's the next part of my update is that I'm currently trying to get back into the swing of things after basically not getting anything done for a full month. I had a design review on the 5th of march that I'll have to go over in detail here, but for now I'll just finish up my update.

Which brings me to the next part of my update: the entire team situation. The last time I wrote a journal entry in here I had not yet even heard of the Coronavirus (now named COVID-19). Fast forward to Thursday the 12th and the university puts out word that all in-person classes will be cancelled for the remainder of the semester and everything will be moved online. No in-person office hours, tutoring, labs, nothing. This also meant essentially evicting all students who did not have good reason to stay out of their dorms or on-campus apartments. What this means for us is that half of the team is now gone and will have to attend meeting electronically and work remotely on their projects. The good news is that we have a Discount channel set up, and VPN access for all of the sub-team leads. Having this already in place will make transitioning easier.

The one big unknown right now is the situation with competition. If universities are cancelling in-person classes and sending everyone away in order to produce adequate social distancing, how can competition proceed? We are not sure yet and SAE International has not put word forth. We'll have to wait and hear what they have to say.

In the meantime, we have been given an extra week of spring break in order for professors and grad students time to figure out how to make their classes work online. This should be very useful for us, as we can get more work done on our projects.

3/21/20

Another update: Yesterday (Friday the 20th) we found out that competition was planning on holding competition online by doing virtual static events. All dynamic events have been cancelled as we can't risk meeting to drive the cars. The situation sucks a lot, but there is also nothing we can do about it. I've been waiting a long time to drive at competition, but that won't happen for another year at least. But as a nugget of wisdom for whomever may be reading this, anyone can handle success and good times, but what really tests you as a person and as an engineer is how you handle the bad times and failure, even when events are out of your control. Keep doing your absolute best and don't quit.

Now, for a further update on the COVID-19 situation, it keeps getting worse by the day. As of writing this, over 26,000 people have been infected by the virus, with lots more to come. I'm mentioning this because the situation is changing every day, in ways that could eventually affect this team and the progress that we make. For example, several cities and even states have issued orders to stay home expect for the most essential things, like buying groceries and going to the pharmacy. In other words, we may not be able to get back into the shop for several weeks, or possibly months. That will have a huge impact on what we are able to do on the cars if all we can do is work remotely. I don't know how the whole situation will pan out. Whoever is reading this is lucky enough to have the benefit of hindsight—I don't. We'll make the best we can of a bad situation.

Also, side note, Alan Cecenas is now the Chassis Lead.

Now that that update is complete I need to cover what was talked about at my latest design review.

3/5/2020 Design Review Debrief:

So again, I'm writing this a little over two weeks after it happened because I was so overwhelmed with schoolwork, and I've forgotten some of the specifics that were brought up. My bad. Anyway, if you want to see it, it is under the shared Google drive under the following file path:

FSAE Shared > SR21 FY2019-20 > Design Reviews > Design Review 03/05/2020 > Copy of C&C Design Review

I won't go over everything in there because you can easily go look for yourself, but I may add a few more things in this journal for clarity later. The feedback I got was ok, most people seemed to like what I had, though there were a few questions and some debates. There was a question about reliability with the steering rack (Lieth), but I answered by saying we had been using that same rack for years now. Also, that's why I added the steel/aluminum category in the decision matrix. The debates were mostly about the paddle shifters. The chief engineer (Gus) thinks we ought to build the paddle shift system and ignore the rules, then make it rules compliant at a later point. I, respectfully, think that is a terrible idea.

We'd be wasting our time working on a system we can't even use for competition, instead of designing it to be rules compliant to begin with. The latter of which I think is a far better idea. The reasoning behind this is that it is much harder to try to modify something to be rules compliant than it is the be designed that way to begin with, not including the possible safety hazards the rules are in place to prevent. Willfully ignoring rules is never a good idea in my opinion.

One other thing that was mentioned was the size of the steering wheel. They wanted to know how big it was going to be. I answered by saying as small as I could make it and still package everything it needs to have. The bigger the wheel the lower we would have to mount it in the cockpit to the rule about the wheel needing to be below the horizontal line of the font hoop in any position. That rule in addition to the one about cutouts on the wheel are a major pain in the butt to me.

These are really the only points I can really remember being brought up. Like I said, I'll cover more of the topics in the slides in more detail later.

3/22/2020

Brakes System (Continued from Page 27):

Here, I'm going to cover in more detail the progress I made on the brakes that I outlined in my 3/5/2020 Design Review. I had been working on it a bit but was too busy to write any of it down in this journal, instead writing it on my slides. So here we go:

I've probably already mentioned this is a previous journal entry, but there are several design flaws with SR19's brakes. The first being that the architecture of the pedal makes it take up a lot of space in the footwell of the cockpit. This isn't good because it means there is less room for adjustability. We can't slide it forward or backward because it takes up too much room. The second flaw is that is there a pretty large amount of slop in the balance bar, so much so that it actually contacts the back of the pedal face and has chipped away at the carbon fiber on the back of the pedal face. I even had to put a space in between the pedal body and the face so that the pedal could move at all. This, obviously needs to be fixed. The third flaw is that the pedals (both of them in this case) are difficult to remove from the pedal rail. This is because they were designed to be bolted in with lock nuts, with the holes on the sides of the pedal rail, meaning that they are underneath the front closeout panel, and you have to get below with a wrench and socket to be able to adjust them. From personal experience this process takes over 15 minutes to do and is a major pain in the butt. It is not something that could be easily done at competition or autocross to make the car suit each driver better. (*Note a possible quick and dirty fix for this could be to use the quick release pins mentioned on page 22 in place of the bolts. However, since the pedals really can't move anyway this would be a moot point in the case of SR19). The final design flaw these pedals have is that the pedal faces themselves are flat themselves are flat. The reason this is a problem is because when cornering hard or violently, the driver's feet can easily slip off the pedal. This means that the driver has to reposition his or her foot back onto the pedal, which takes time. This is inconvenient in the case of slipping off the throttle because the driver loses time repositioning his/her foot before they can accelerate. But it is outright dangerous if the foot slips off the brake pedal, as any delay in emergency braking could be seriously hazardous. The grip tape (similar to a skateboard) is an ineffective means of keeping the driver's foot in place.

For the first design flaw, I am going to attempt to solve it by working with the chassis lead to lengthen the distance from the knee hoop to the front bulkhead by about 2 inches to facilitate greater room for adjustment. I am also going to solve the problem by making a more compact pedal design, which is somewhat similar to 19's but improved. I have decided to move away from the 'underfoot' pedal configuration I had initially planned on, simply because it wouldn't be appropriate for the layout of the cockpit. The style is much better suited to GT cars with a more upright seating position. It also would make it difficult to adjust the braking bias. I have also mentioned previously about a central sliding plate to which the brake and throttle would be mounted. I am still planning on implementing this plate.

For the second design flaw, the one about balance bar slop, I have decided upon using a new style of balance bar made by Tilton Engineering.


Image credit to Tilton Engineering

This is a much more compact unit than some of the other styles of balance bars out there. This gives it an advantage because it can be packaged better. The narrow width will mean the balance bar clevises will not be able to reach the pedal faces unlike in 19's design, eliminating that issue. It also solves another issue: space on the pedal. Most balance bars use a disk-like bearing that slides inside if a steel sleeve. This is how bias is adjusted. The difficulty this creates is that the sleeve requires that it be mounted inside of the body of the brake pedal. This means that a hole must be bored through the middle of the side of the brake pedal body. And the hole is typically large, over an inch or greater. (Below is a picture of the type of bar I'm talking about).


Image credit to Tilton Engineering

This places restrictions on where the bar can be placed, as well as partially dictating the geometry of the pedal body. But the new design of the balance bar allows it to be mounted from the outside, via the holes on the side of the body. This also means it could potentially be adjusted if need be. It will allow more flexibility in the design of the brake pedal, which is obviously a good thing.

The all-in-one body design will also make it easier to handle and eliminate the slop of previous styles.

Brakes System will continue on page 33.

6/15/2020

Another Major Update (and 2020 Competition Briefing):

I have not been very good at keeping this journal up to date in the past few months. I haven't had much to report on since working from home has made a lot of complications in our lives. I'm going to take some time to update on what's been happening lately and what that means for the future. Over the past 3 months I've been working a tad bit on the steering wheel design (more on that to come in a later entry). But my main focus has been on where exactly I was wrong with my brake calculations and why I was getting such weird numbers from. I have found out my numerous errors and have some numbers that make me much happier. I'll fill in the details after a bit. Other than that, since school let out last month I have been preparing for the online, Virtual Formula SAE Competition. That has included making slides and trying to prepare a presentation for the judges. (For those interested the location of that design presentation is on the Google Drive under the following file path: **FSAE Shared > SR21 FY2019-2020 > Michigan 2020 Design Presentations > Cockpit and Controls**. The audio recording of a recap between myself and Lieth Jaradat explaining how the Design Event went and the questions that were asked can also be found next to the C&C slides. The other member's presentations and recaps are also there). The other thing I had to do was to get ready for the Cost Event, which I was put in charge of for some reason. Part of why I'm writing this update is to recap on paper how those presentations went and what all happened. I'll begin that now:

(Side note: for clarity, we were using SR19 for this competition, so non of the current work we have done for SR21 was used. Instead we presented for the old work on SR19. I'm still putting it in this journal as I think the experience and what we learned is useful).

First, we had the cost event in the early afternoon of Monday, the 9th of June. Unfortunately, SAE was very disorganized with this switch to online format, and most of the dates were only roughly announced a few weeks before they actually happened, and we only found out that we would be presenting that Monday a week before it happened. The very Monday I'd be traveling back from a vacation in Texas I'd planned a month or so before, and couldn't reschedule. The place I was at for the weekend had neither Wi-Fi nor cell service, which I didn't know before hand. A memo sent by the SAE organizers made it sound like we had to present our cost report on the fuel system. Turns out this wasn't the case, and the team had decided to write a different cost scenario and had forgot to tell me until Sunday night. I ended up doing the call in the business center of a hotel in Dallas with several people around. All this is to say I wasn't very prepared, nor in a good location to have a meeting. And it turns out I was expected to do most of the talking… Actually all of it. And I didn't know they wanted an actual presentation. Basically, I ended up telling them what we planned to do AS I WAS READING IT MYSELF. But it ended well because I managed to bullshit myself through the presentation and got a score of 19/20. Not too shabby. (To be fair everyone else wasn't prepared either, so it didn't really matter who ended up talking).

Humorous anecdotes aside, the two judges were very nice and understanding. They asked us to go over our plan (which can be found here: **FSAE Shared > SR21 FY2019-2020 > Cost Event Virtual 2020**, as well as the audio recording of the meeting. Please don't listen to it because I think my voice might have shook a bit) and asked a few questions. Didn't take long fortunately.

As for the Design Presentation, I was a little more prepared. It went pretty smoothly too. We started out as a main group in one channel, then after some brief introductions we split into our various sub-teams and got pared with the Judges for each role. The judge I was pared with was Benjamin Dean, the brakes and ergonomics judge. I went over the presentation in the call Lieth and I did, but I'll put some of the main points here: One of the things he really liked was the design matrix. He was very happy to see this, but suggested I highlight or use a different color or font to show which product I decided upon (in this case the steering rack). But he told me it was only a nitpick and not a big deal. It's still good to know. He wanted numbers on the target input force for the steering torque, which I didn't have. He said it would be good to have a number for this, and suggested I start by reading some books for articles about human performance and do some actual testing of our own to see what we could comfortably do. I agree with this. When I got to the brakes, he asked me if I had a system for figuring out where I can save the most weight. His point was that I could spend a lot of time trying to optimize the weight for each component, but I would be wasting a lot of it. Instead, he suggested that I weight each component, either IRL or in SolidWorks and find the heaviest pieces and cut down the weight on those and an iterative process. This would save more time and it would be more methodical. "Finding the low hanging fruits" as he put it. He also wanted to know how we had come up with how much adjustment was needed in the pedal. I told him we had used an ergo jig to figure it out, which was true since the old team had made one when SR19 was still SR17. The other question he asked was how we decided on 7075 T-6 series aluminum instead of 6061 T-6 aluminum. I told him that the high strength to weight ratio justified the more difficult machining. Of course, I didn't make the decision to use 7075 at the time, but it's a reasonable justification. I spoke a bit on using 7075 on page 26, but I'll probably talk more about it later. He then asked how easy it was to adjust the pedals and I truthfully answered not easily. I then mentioned I hoped having a quick release pin would help.

I then moved on to the rotors and calipers. I gave the argument for the 4130 steel rotor, citing the smoothness of the mu curve over a wide range of temperatures and run times. My justification is that having a smooth curve results in predictable braking feel, allowing a driver to get accustomed to it. It might be lower than the cast iron, but having better predictability will make the lap times drop. He said it was probably one of the best arguments for 4130 rotors he's heard. I was careful to mention my source of the data, to avoid any sort of plagiarism. The only thing he nitpicked on was he wanted the rotors and the calipers to have their own sides to make the presentation more polished. He really didn't have too much to comment on the throttle pedal.

Moving to the shifter, he seemed pleased with the visuals I had provided for how the shifter worked. When we got to the data display, he was oddly specific in asking me why exactly we had gone with the Motec unit. I told him we wanted to have a color display and he was curious why we had specified *color*. I don't remember everything else, but there's more detail in the phone conversation with Lieth. Finally we got to ergonomics which the judge was particularly interested in, given it was his specialty. One of the bigger questions he asked was why we had gone with a smaller chassis, and I told him weight savings. He then quizzed me on the benefits of weight savings versus the smaller and more cramped cockpit and asked how we quantified the benefits vs the downsides. This was a bit more difficult to answer. He ended up telling me that in his experience, a few more inches of the cockpit didn't hurt the weight much, but did have an effect on how comfortable the driver was, which is something to keep in mind. He then asked me about the curved floor close out panels and why they were riveted. He wanted to know the weight savings of the rivets versus the bolts, which I obviously didn't have an answer for, other than that the aluminum pop rivets were lighter. As for the curved floor boards, I told him that they held curve along with the body making it more comfortable, and also lowering the driver slightly to help lower the CG of the driver. He seemed intrigued by this. Finally he gave some feedback on the carbon panel for the starter and off switched, cautioning us to be careful about making sure they were properly grounded to ensure the car didn't have any electrical problems.

Other than that he didn't have much more to say. Overall it went well, and it turned out to be a more 2 sided conversation rather than the usual judging routine. It helped give us some direction that we needed and have a conversation with a profession to get some feed back. In the end, I got 2 take a ways from it: have numbers, but have the reasons behind the numbers, and be a bit more methodical in the design choices.

I think that about sums up this major update. I'll restart more regular entries tomorrow.

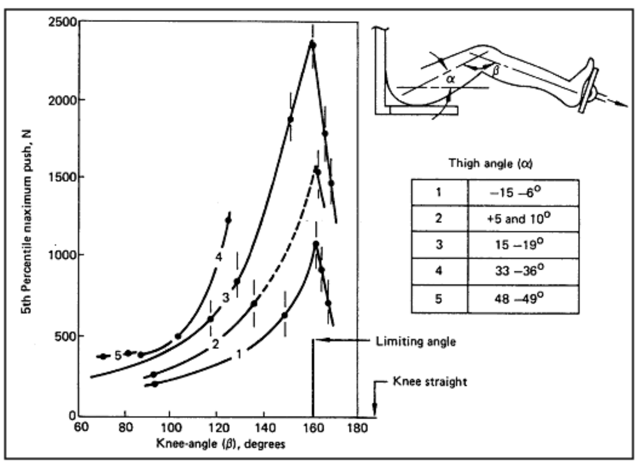
6/16/2020

Brakes System (Continued from Page 31):

As I mentioned in yesterday's big update, during our forced hiatus I had been working on finding the errors in my brake calculations. I have now found where those bugs are. On pages 27 and 28 I had made some suppositions as to where the sources of error lay. The first was that my approximations were off. They were not. The bits of data I had to infer are accurate enough to where they are not significant sources of error. There likely is some error in there, but not enough to have a major effect. My second supposition was that I used the wrong coefficient of friction for the tires in my brake torque calculations. My reasoning at that time was that I should have used the highest coefficient of friction available. But I think the numbers I used are representative enough. 12 psi is the median pressure, so that's likely where the tires will be at. No, the true sources of error were suppositions 3 and 4. There IS an error in the TDIF and master cylinder size.

For the TDIF, my mistake was in the way I measured how much force a driver could put into the brake pedal comfortably. I used a leg press machine to do it. This was not a good representation of the way a driver presses the brakes. I found the reason for this while doing some research on how MIT got their TDIF numbers from. I ended up finding a NASA research document on human performance capabilities. And as it turns out, the way the test subjects legs are angled relative to the plane of the horizontal floor has a massive impact on how much that person can push with their legs. That article can be found here:

Figure 4.9.3-7 Leg Strength at Various Knee and Thigh Angles (5th Percentile Male Data)



Reference: 1, p. 115; NASA-STD-3000 206

https://msis.jsc.nasa.gov/sections/section04.htm. For simplicity I'll also leave a screen shot of the part I'm referencing:

As you can see, when the thigh is at 15-19 deg. from the horizontal, and the calf line is 160 deg. relative to the thigh line, an average person can push up to ~<2500 N. This translates to about 550 lbs. Granted, it doesn't say weather that is both legs or just one. I'm assuming both. But even 225 lbs per leg is not bad. And now you can see why using a leg press machine is useless as a test for TDIF. The legs were be positioned closer to the 4th or 5th data line on this graph, which shows that in this orientation, the legs can't push nearly as much.

Below the graph is a picture of a machine very similar to the one I used to measure our TDIF (picture source unknown). Notice the orientation of the legs? Not very conducive to force production, if the NASA data is accurate (hint: it is). This also makes sense when you think about it in a different way: if you can only push between 70-100 pounds with one leg, how on earth could anybody possibly stand on one leg and raise themselves on tip toes? I weight 150 lbs. and I can do it easily. I'm one the lightest

members of the team, so if I can do at least 150 on one leg, everyone else should be able to do the same or more. Suddenly, 200 lbs. doesn't sound that undoable. To test this assumption further I put 50 lbs. of textbooks in a backpack and put it on. I weighed myself to be sure: 203.2 lbs. I then stood on one leg and raised up on my toes several times. It wasn't too difficult, but it did get tiring after a while. However, this does not worry me as my legs will be slightly bent in the cockpit and that will help out with the force generation. This isn't the best method to test TDIF, but it's what is available to me right now.

One way I could improve this for the future is to use a calf raise machine (picture of what I mean on the left) instead of the leg press and get some volunteers to try it out to gather data over a wide range of people, not just myself. But due to COVID-19 that isn't really an option because the team is scattered, and we still haven't returned to in person meetings. Sufficed to say 200 lbs. is a good estimate for TDIF.

My 4th supposition was that the master cylinder size might be wrong. This is true, but not the in the way I was thinking. Throughout my research I had known that master cylinders do affect the pressure in the brake lines, which in turn contributes to how much force the brake calipers can apply. This is true. However, I had thought that a bigger master cylinder meant more pressure. Intuition would have bigger *master cylinder = more braking force*. Nope. Turns out it's the other way around. The smaller a master cylinder diameter is the larger the force it produces. The trade off is that it requires more displacement in the cylinder to move the pads the same amount. Here's the article where I found this information:
https://www.markwilliams.com/braketech.html#:~:text=While%20a%20larger%20master%20cylinder,create%20the%20same%20system%20pressure.

The final reason the calculations were off wasn't in my suppositions on pages 27-28. I had realized that in part of the formula, it required the area of the area of the caliper piston, which I had found and wrote down in the data table on page 11 as 1.23 in^2 per piston. And that's where I screwed up on that part: *per piston.* I had only put 1.23 in^2 and didn't multiply it by 2, which I needed to do.

6/29/20

Quick update: new year elections have been held. I am no longer Captain. I am now Chief Engineer. Big fun.

6/30/20

Last night I finished and wrote down the calculations for the pedal ratio and rear master cylinder size. Only to double check Tilton's website to find out that the 0.5" master cylinder I used in my calculations doesn't exist. The smallest they offer is 5/8" or 0.625. This may not seem like much, but it moved the pedal ratio from 2.5 to 4. I checked AP Racing as well, but their smallest size is only 0.05" smaller, so it wouldn't matter. After some quick scratch paper sketches (taking into account the size of the bias bar) that the pedal footprint (the lateral space it takes up in the cockpit would be about 8". This is pretty big. It may in fact be bigger than SR19s, which was *not* my goal. However, *if* my brake calculations are right, SR19s pedal ratio is too small, and won't be effective. I've put solid engineering into my pedal design, and it's the numbers I have to work with. Alan added about 2 inches to the front bulkhead distance, so it should be alright. Perhaps during testing I'll find out the pedal ratio could have been smaller, but I'd rather not chance it. It's better to make it bigger than necessary in this case, since it is such a critical component. Hopefully when I get some feedback I can decide if the ratio can be smaller. I'll drop the new brake calculations down below, then I need to get started on actually designing the thing. Calculations have taken too long to complete. Blame COVID I guess.

*(Calculations on next page)

Pedal Ratio Calculations:

- Original Equations:

(Front): $F_{driver} \cdot PR \cdot BB_f = \dfrac{\tau_{braking,F} \cdot A_{mc,F}}{2 \cdot r_{rotor,eff} \cdot \mu_{pad\text{-}rotor} \cdot A_{c,F}}$

(Rear): $F_{driver} \cdot PR \cdot BB_r = \dfrac{\tau_{braking,r} \cdot A_{mc,r}}{2 \cdot r_{rotor,eff} \cdot \mu_{pad\text{-}rotor} \cdot A_{c,r}}$

- Reworked Equations:

(1) $PR = \dfrac{\tau_{braking,F} \cdot A_{mc,F}}{2 \cdot r_{rotor,eff} \cdot \mu_{pad\text{-}rotor} \cdot A_{c,F} \cdot F_{driver} \cdot BB_f}$

(2) $A_{mc,r} = \dfrac{F_{driver} \cdot PR \cdot BB_r \cdot 2 \cdot r_{rotor,eff} \cdot \mu_{pad\text{-}rotor} \cdot A_{c,r}}{\tau_{braking,r}}$

- Knowns:

$\cdot F_{driver} = 200\,lb$ $\cdot BB_f = 0.7$ $\cdot BB_r = 0.3$ $\cdot \mu_{pad\text{-}rotor} = 0.35$

$\cdot \tau_{B,F} = 740.32\,lb \cdot ft$ $\cdot \tau_{B,r} = 323.26\,lb \cdot ft$ $\cdot r_{rotor,eff} = 2.8\,in$

$\cdot A_{c,F,r} = 1.23\,in^2 \cdot 2$ $\cdot A_{mc,F} = 0.625\,in.$

- Substituting & Solving:

$$PR = \frac{(740.32\,lb \cdot ft \cdot 12\,\frac{in}{ft})(\frac{\pi (0.625)^2}{4})}{2 \cdot (2.8\,in)(0.35)(1.23\,in^2 \cdot 2)(200\,lb)(0.7)}$$

$$\boxed{PR = 4:1}$$

- Plugging into 2nd Equation:

$$A_{mc,r} = \frac{(200\,lb)(4)(0.3)(2)(2.8\,in)(0.35)(1.23\,in^2 \cdot 2)}{(323.26\,lb \cdot ft \cdot 12\,\frac{in}{ft})}$$

$$A_{mc,r} = 0.298\,in^2$$

- Converting:

$$A_{cir} = \frac{\pi D^2}{4} \iff D = \sqrt{\frac{4 \cdot A}{\pi}}$$

$$D = \sqrt{\frac{4(0.298\,in^2)}{\pi}} \Rightarrow D = 0.62\,in \approx \boxed{0.625\,in}$$

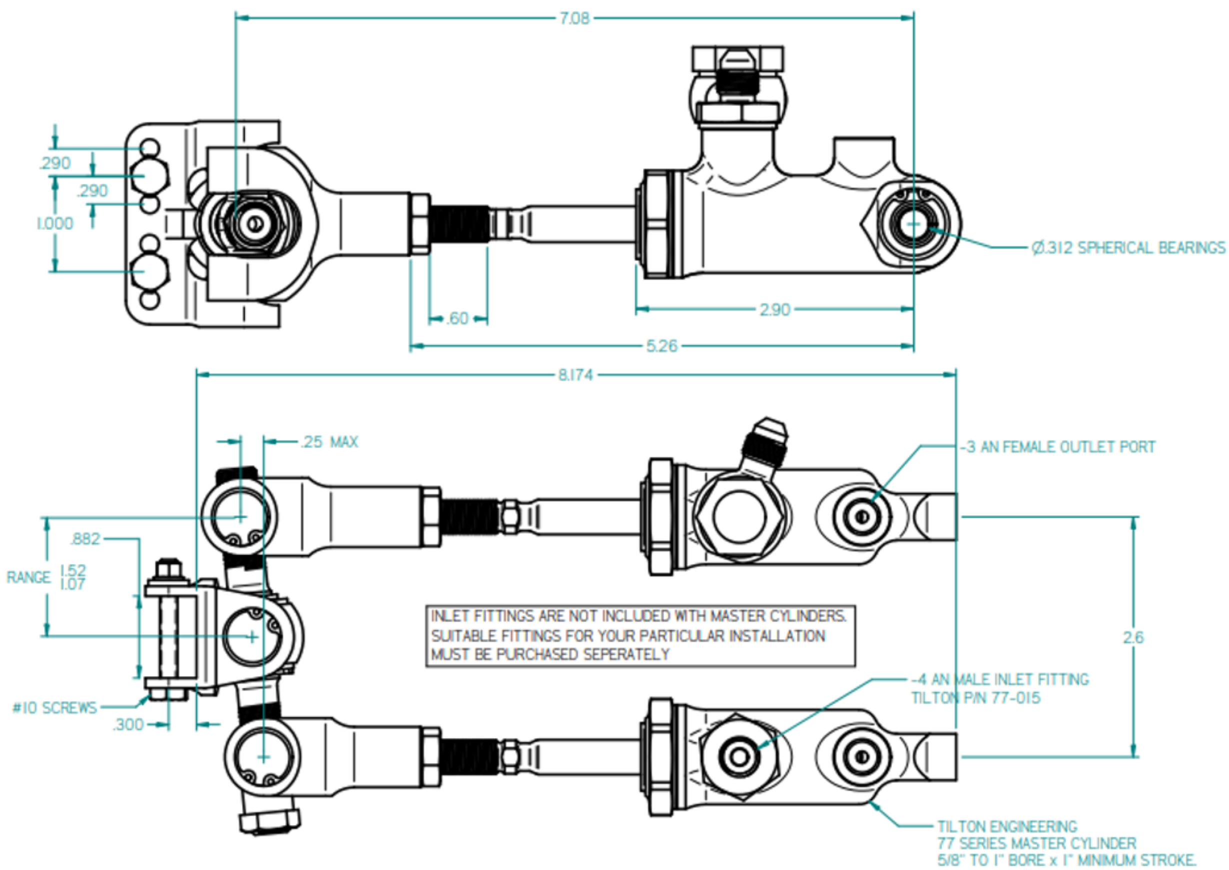
Checking my calculations with a computer algebra system:

$$\frac{(740.32 \cdot 12)\left(\frac{\pi 0.625^2}{4}\right)}{2(2.8)(0.35)(1.23 \cdot 2)(200)(0.7)} = 4.03767\ldots$$

$$\frac{(200)(4)(0.3)(2)(2.8)(0.35)(1.23 \cdot 2)}{323.26 \cdot 12} = 0.29831\ldots$$

Thus, from these calculations we can see that the pedal ratio is indeed 4:1 and the size of both master cylinders should be 0.625" or 5/8" in diameter. At least that should make ordering them easier.

Next, I'll work on creating the actual CAD model for the pedal. Part of the dimensions of the pedal will be created from the pedal ratio, but it will also be constrained by the dimensions of the balance bar. Below is the diagram provided by Tilton in their installation instructions:



I can use this for quick reference when I'm designing the pedal body.

7/1/20

One of the biggest challenges now is going to be optimizing the pedal and validating that the pedal will be strong enough to meet Rule T.3.1.10b that states: "The brake pedal must be:

b. Designed to withstand a force of 2000N without any failure of the brake system or pedal box. *This may be tested by pressing the pedal with the maximum force that can be exerted force that can be exerted by any official when seated normally."*

Essentially, my next task will be to create the pedal and attempt to optimize it such that it can withstand the 2000 N (or ~400 lbs.) required to meet the rules, while also making the pedal as light as I can. This will definitely require some FEA (finite element analysis), but it's been a few years since I've done the Solidworks tutorial on FEA, and I've never used it on a real part before. I really don't know much about it right now, so I'll probably spend a few days researching it and finding tutorials online and some resources. If I find any I'll link the useful ones in here so whoever is reading this in the future can easily find them too.

After watching a couple of videos on how to do FEA on Solidworks, I've realized I'll need to pick the material in order to do the analysis. I've more or less decided upon 7075 T-6 aluminum, but following the advise from the competition judge last month, I'll go ahead and compare the two most obvious metal choices and do a design matrix to have an addition reinforcement in my decision. Thus, I'll be comparing 6061 T-6 aluminum alloy and 7075 T-6 aluminum. Before I get into the comparison of these metals, here is some reasoning behind why I chose these metals to begin with. Both 6061 and 7075 series aluminum are very strong alloys and are the most commonly available aircraft grade aluminum. They are both lightweight and very strong and are commonly used in all types of industries. However, due to the difference in alloying metals, they have different material properties. For a more comprehensive discussion on these two metals and to see the various alloying metals, check out this article I used to come up with the design matrix: https://www.thomasnet.com/articles/metals-metal-products/6061-aluminum-vs-7075-aluminum/. Another difference is the cost of the metals, which I'll factor into the matrix. I got the metal prices from Midwest Steel and Aluminum. One of the reasons I am not considering steel is because while it is very strong, it is also heavier. 7075 T-6 is almost as strong as steel and is lighter. The other thing to consider is the weld on a steel pedal would need to be pretty decent considering the cyclic loads the pedal will be under, and the fact that the pedal is the primary safety device of any vehicle. In some places steel is more appropriate than aluminum, but in this case, I think aluminum is the better choice. The reason I am not considering titanium is because it is expensive, hard to get, and hard to machine. It is probably a better material than aluminum, but these factors I've just listed make them unsuitable for the application. One final word on 7075 T-6 aluminum is that since the primary alloying material is zinc, it is essentially impossible to weld. It will simply crack at the slightest stressed. There *is* a way to weld 7075, and it was invented a few years ago. It uses nano particle coated filler rod to prevent cracking, but since it so new and is proprietary technology, it's still basically impossible for all intents and purposes. 6061 is very weldable however, but the pedal body won't need to be welded, so I won't consider that in my matrix.

Now for the matrix itself:

Material:	Yield Strength: (x 1)	Young's Modulus: (x 1)	Hardness: (x .075)	Machinability: (x 0.75)	Corrosion Resistance: (x 0.25)	Cost: (x 0.25)	Total:
6061 T-6	276 MPa 1	68.9 GPa 1	95 BHN 0.75	"Good" 1.5	"Good" 0.5	0.196 $/in^3 0.5	5.25
7075 T-6	503 Mpa 2	71.7 GPa 2	150 BHN 1.5	"Fair" 0.75	"Fair" 0.25	0.263 $/in^3 0.25	6.75

As we can see from the above matrix, the 7075 is indeed the better material choice for the body of the pedal. I should also explain my new system of weighting each category. Since some categories are more important than others, I decided to use a multiplier system to weight the categories. I used 4 “levels” of importance and assign a multiplier between 0-1 accordingly. So, the most important categories will receive a “(x 1)” multiplier, and the least important will receive a “(x 0.25)” multiplier. Then I rank each material in each category between 1 and the number of categories. In this case, there are only two, so it goes between 1 and 2. But if there were four materials to rank the possible scores would be as follows: 1,2,3,4. Before recording the scores in each cell however, I multiply them by their weighted multiplier, the record them in the box. Then, simply add the scores across the rows and which ever material has the biggest score wins. It’s a little hard to describe in words, but the example above should make it clearer.

For clarity I’ll put the definitions for yield strength, young’s modulus, and hardness below so anyone unfamiliar with the terms can better understand why these are important factors:

Yield Strength: “The *yield strength* is defined as the stress at which a predetermined amount of permanent deformation occurs”. (*Source: Engineers Edge. Read more here:* https://www.engineersedge.com/material_science/yield_strength.htm).

Young’s Modulus (also called Modulus of Elasticity): “The modulus of elasticity of a material is a measure of its stiffness. It is equal to the stress applied to it divided by the resulting elastic strain.” *(Source: Collins Dictionary. Video explain more here:* https://www.youtube.com/watch?v=DLE-ieOVFjI).

Hardness: “Hardness is the property of a material that enables it to resist plastic deformation, penetration, indentation, and scratching. Therefore, hardness is important from an engineering standpoint because resistance to wear either by friction or erosion by steam, oil, and water generally increases with hardness.” *(Source: Engineers Edge. Read more here:* https://www.engineersedge.com/material_science/hardness.htm).

With that explanation out of the way, I can now move on to studying more of how to do FEA to optimize the pedal design. Like I said earlier, I’ll put links to whatever I find useful.

7/2/20

Ok, here are some videos and articles that may prove useful in doing my FEA analysis:

1) https://www.youtube.com/watch?v=7fQ0gxq3bMs

2) https://www.youtube.com/watch?v=2LDSQMCeBBs

3) https://www.youtube.com/watch?v=4pbIAQQ9tGc

4) https://www.simscale.com/blog/2016/11/learn-finite-element-analysis-fea/#:~:text=The%20finite%20element%20analysis%20is,to%20develop%20better%20products%2C%20faster.

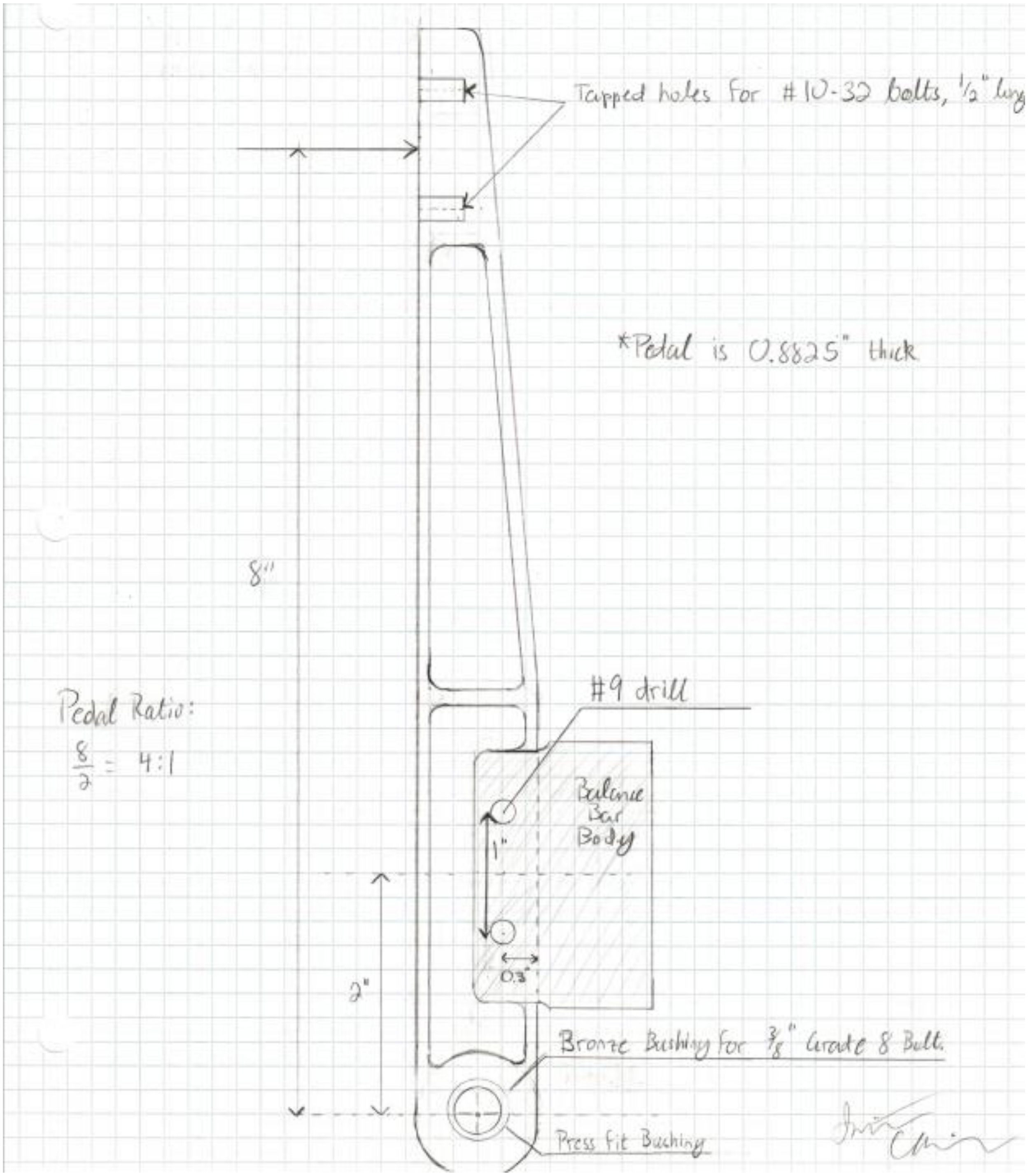
5) https://www.designnews.com/automation-motion-control/6-things-all-engineers-should-know-using-fea

6) https://www.designworldonline.com/using-fea-to-study-a-mechanical-part/

Before I can begin any analysis of the pedal however, I need to actually have a pedal design. This, like I said earlier is going to be challenging because I don't really know a good method to figure out the best geometries to make the pedal strong but light.

7/9/20

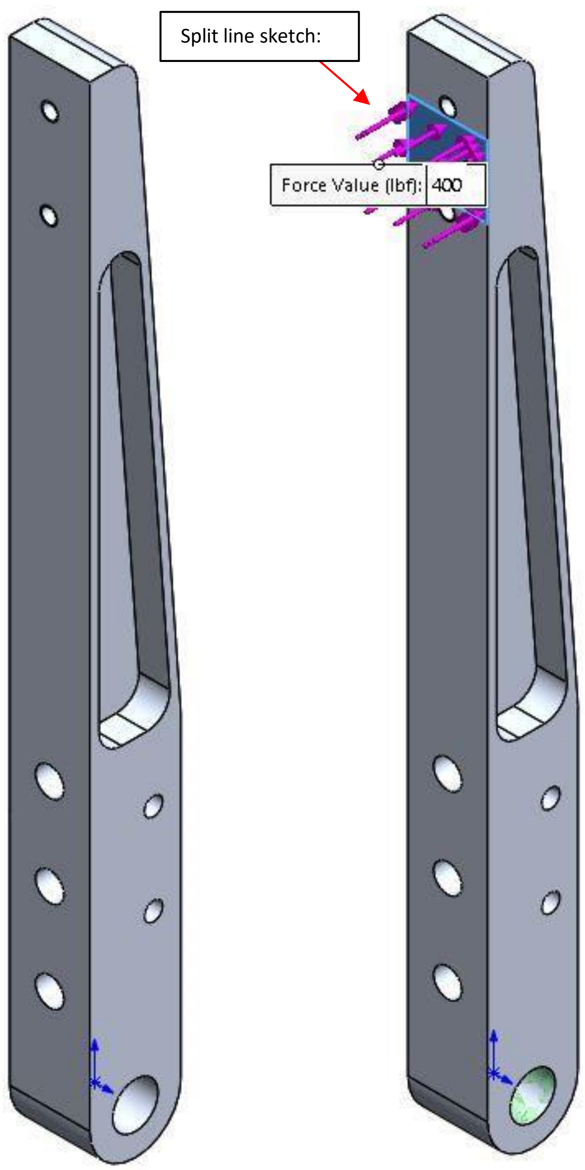
Ok, so I've been pretty busy helping people with some projects over the past week and editing the Bylaws for our upcoming start of year meeting this weekend, but I did manage to sketch out a pedal design on some graph paper. But as I was going through the initial design, I realized just how long that 900 Series bias bar was going to make the whole pedal assembly. It's probably too long. So I took a look at Tilton's 600 Series bias bar to see what kind of space I'd be saving. It would be approximately 2 inches, which would be a good improvement, though not a perfect one. It's the pedal ratio that needs to come down, but I can't do much to change that for this year's design. It's something that can be improved in future years. But getting back to the pedal design, I also noticed that Tilton says it is for fixed master cylinders, not the kind we are going to be using. I spend last night thinking about that and remembered that not only does SR19's pedal use it, but several other teams use it as well. So it should hopefully be fine. I think I'll draw a new pedal design (or modified to be more specific), put them both in CAD and figure out from there which one I want. For now I'll put the pedal design I already have down below. I'll need to draw it in CAD, then analyze it with FEA.



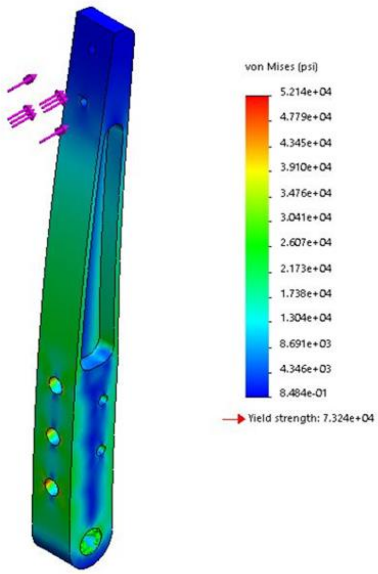
(Above): 1st iteration of the brake pedal design

7/23/20

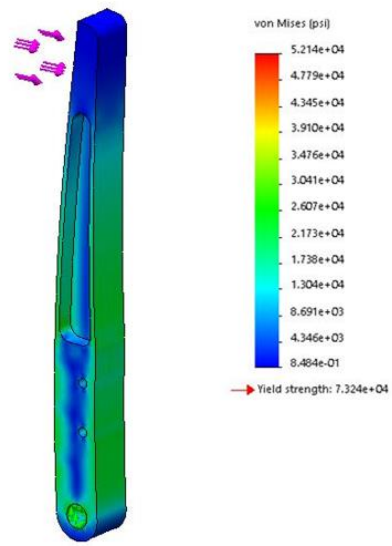
Update: Been working for a few days on other projects and had some family stuff to take care of, so I'm just now getting back to work on the pedal design. During the past two weeks I realized that the pedal design above needed to be modified because it was too short to accommodate the size 12 race boot I was using for a reference on where to place the holes for the inserts for the heel cups. So I ended up making a new pedal design 1" taller than the previous design. It is saved as **SR21-CC-A003-S001-X001-P001.2 Brake Pedal Body Rev 1**. This replaces the original design now found in **SR21-CC-A003-S001-X004**. I also added the wholes needed for said inserts. I just finished running the FEA on it. Here is the new pedal body, along with the set up I did for it to begin the analysis:



On the left hand side, is the new pedal revision which is now 10" tall, compared with the 9" it was previously. Below on the front face is the holes that will be used to except press fit steel inserts to hold the #10-32 flathead bolt that will be used to attach the heel cup. I elected to go with steel inserts over simple tapped aluminum because I was worried about stripping the aluminum over time. As you can see there are three holes, which correspond to two different positions for the wheel cup. It'll be either higher or lower. The lower setting is for larger shoes (U.S. men's 10-12) and the higher is smaller sizes, like U.S. men's 9 and under. This keeps the balls of the feet directly between the two bolt holes in the upper part of the pedal, where the input force was designed to be. Otherwise, drivers with smaller feet would have less pedal ratio than those with larger feet. For the FEA analysis, I had to create a square sketch on the space I just mentioned so that the simulation software would know where to put the force. Otherwise it thought that it should distribute the force over the entire front of the pedal. Instead, it is localized in the area seen on the right side. Here is a video explaining how to do that: https://www.youtube.com/watch?v=ZU5sNsDkWPE. I set the input force to be the 400 pounds specified by the rules, ensured that the material was specified to be 7075-T6 aluminum and ran the simulation. The links I gave a few pages ago will show you how to do that. Here are the results of the simulation study:

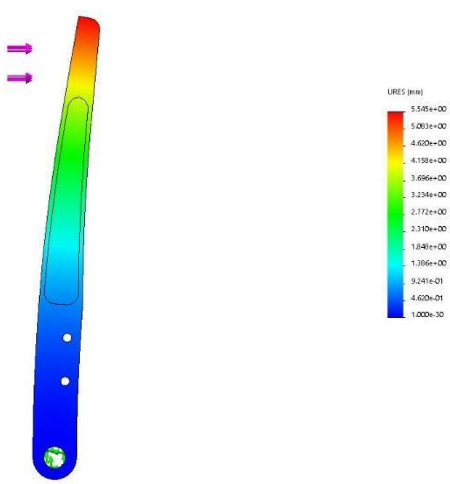


Von Mises stresses on front side of brake pedal.

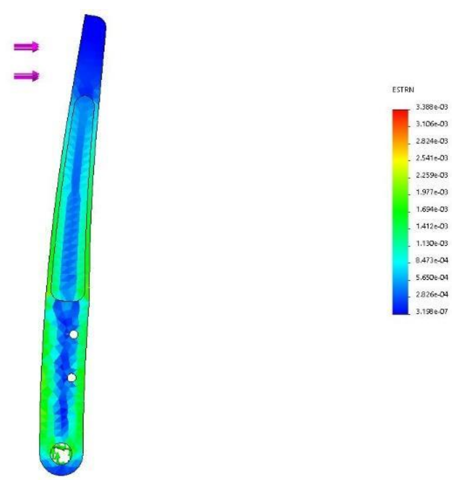


Von Mises stresses on back side of brake pedal.

Above are the Von Mises stresses for this pedal iteration. As we can see from the stress map, at no point does the stress get anywhere near the yield strength of the 7075-T6 material, thus we know the pedal is safe. Actually it can be lightened because we don't need that much material that we have currently. But this project has gone on long enough, and I need to start cutting my time spent on this project down, so I may end up leaving it for future Cockpit Leads to work on the design. Here is the results for the displacement and strain:



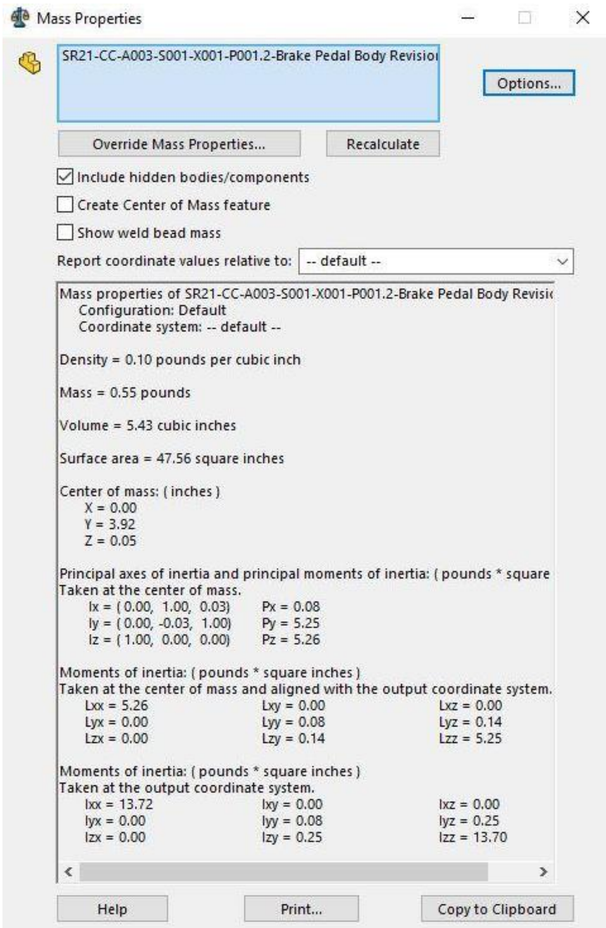
Displacement of the pedal from the applied force.



Strain on the pedal from the applied force.

Not too bad. I'll also put the projected mass properties down below as well:

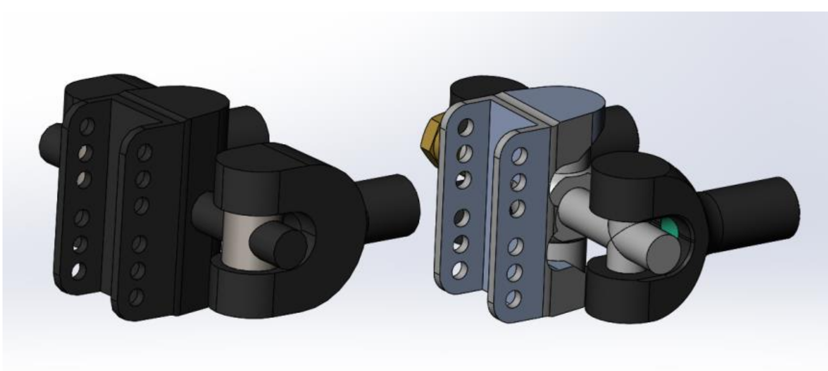
As we can see it's supposed to weight half a pound, which isn't great, but it's what we've got. I need to move on to more important stuff. On that note, I sent Tilton Engineering an email asking for a better model of their 900 series bias bars, but they declined to give me one. So my next task is to quickly design a rudimentary one for modeling purposes so I can position the master cylinders where they need



to go on the mounting bracket for the brake pedal. I'll use the Solidworks model they did provide and take enough measurements to get a rough model.

7/24/20

Ok, so I'm finished with the bias bar model. It wasn't crazy hard. Only took me a few hours to make. Unfortunately, some of the parts where not easy to measure off of the model Tilton did provide, so some of it was kind of flubbed so it looks close enough to the original. The holes and other major/important parts are roughly where they need to be. I'd say there is maybe a +/- .05" tolerance on everything they didn't explicitly dimension on their installation on page 37. But again, it does the job. Side note, on the bias bar and on the model given to me, the clevises are unequal length. This is intentional for later fitment and adjustment once assembled with the master cylinders. I chose not to do so on my model because it would make it a lot easier to position in the pedal assembly. I am fully aware there will be a slight difference once the real thing is made. Actually you can see it on the diagram on page 37. The bar is slightly angled relative to the vertical. This is for biasing the bar, but again, for my purposes I won't worry about it. Here is the assembly of the bias bar I made (left) versus the Tilton model (right):



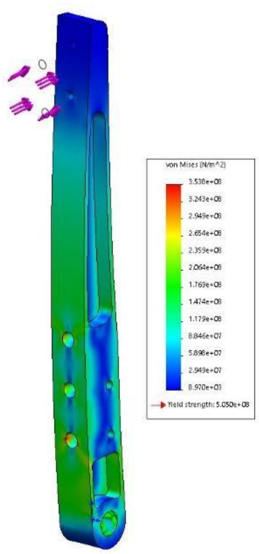
But I digress…

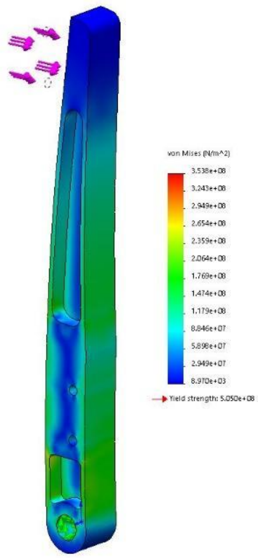
I'll begin the assembly of the pedal body and its components now. I'll put a picture of it when I'm finished.

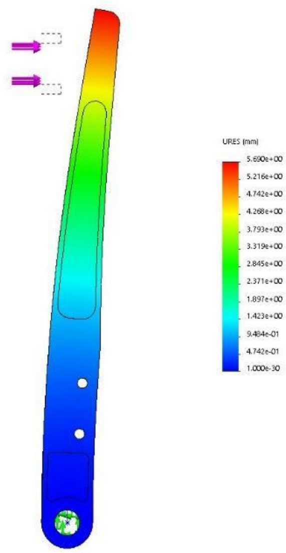
Update: AAAAAAAAAAAAAARRRRRRRRRRRRGGGGGGH! The pedal body once again need to be altered. AGAIN. The heel cup is still going to be making contact with the frickin pedal mount!

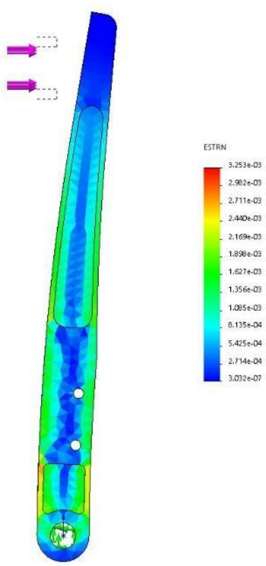
7/25/20

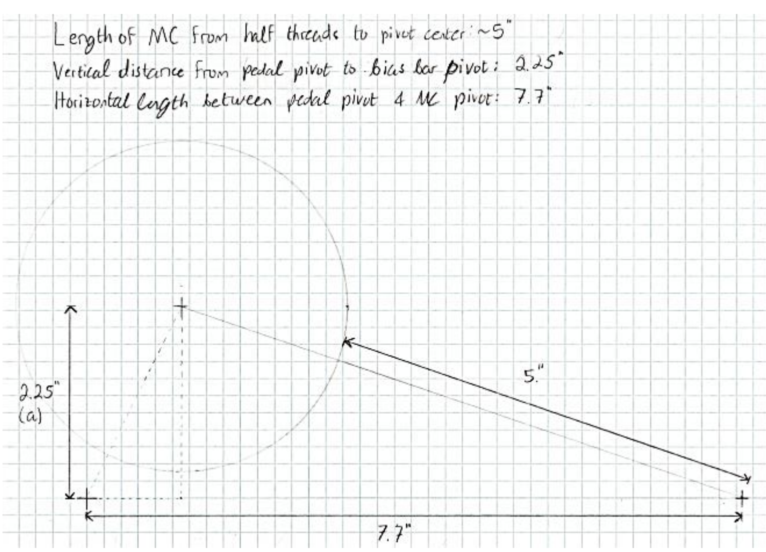
Update: never mind on making a new pedal, I just forgot to properly constrain the holes for the heel cups, which meant they were in the wrong place. I fixed that, and now they should be fine. I also added a cutout beneath the mounting holes for the bias bar to shed a bit of weight. To be on the safe side, I redid the FEA on it to make sure it was going to be solid. Here's the results:







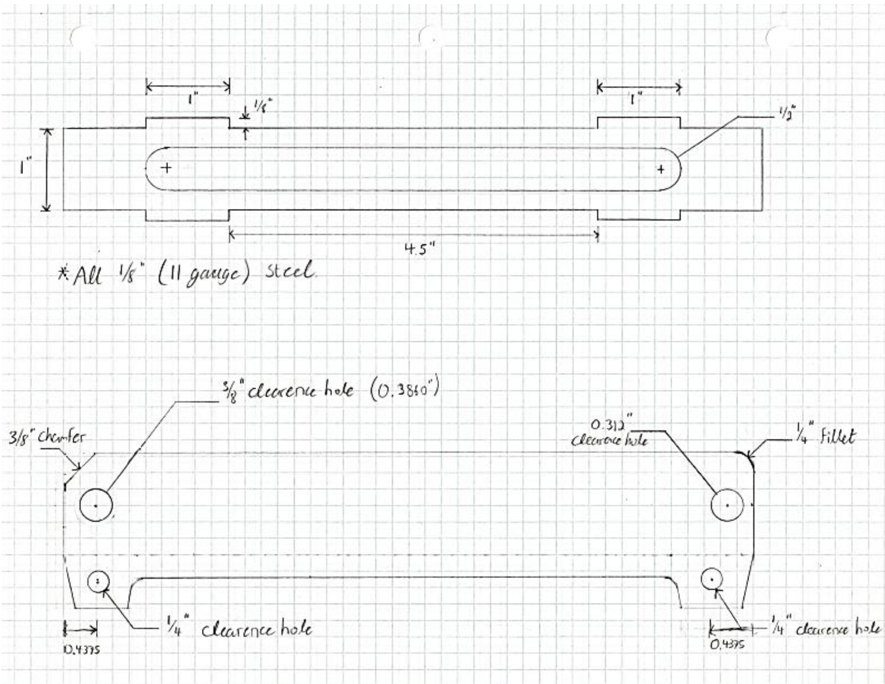




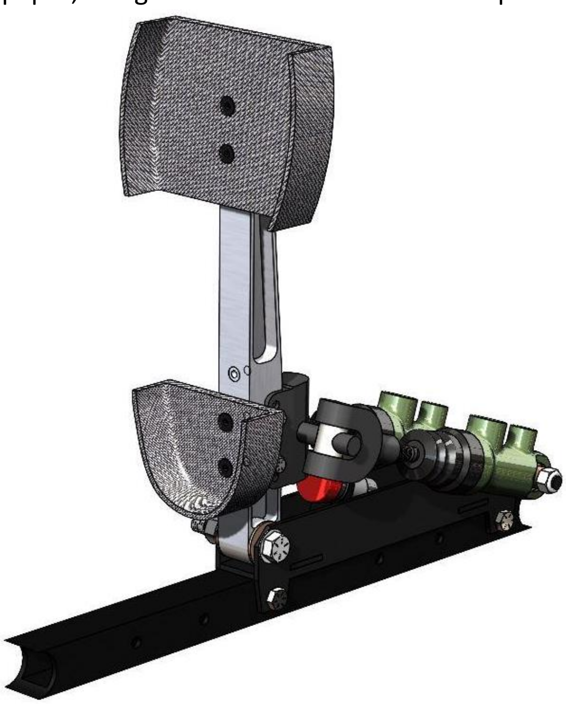
Not bad again. The mass went up from 0.55 lb. to 0.552 lb. so really no change. I can once again work on the assembly of the pedal and parts.

8/1/20

Last night I finished all the parts and assembly for the brake pedal. It's fully done now I can finally move on to other projects needing my attention. I haven't been updating this journal over the past week or so because I wanted to focus on actually working on the design and assembly



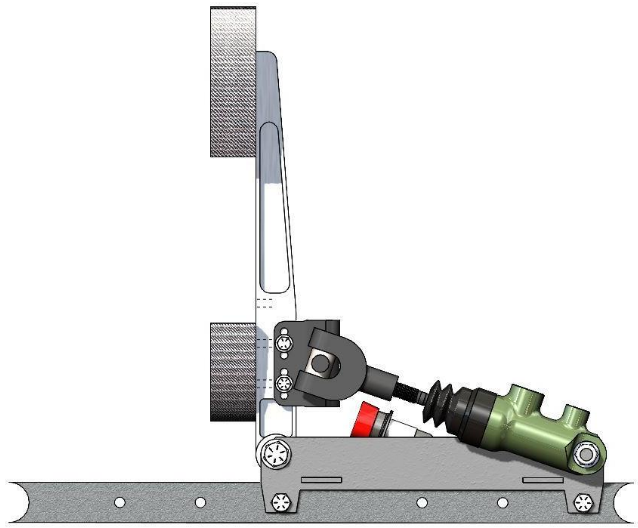
as there were a lot of small things that I needed to do to finish it. Part of what I did was to design the base base/bracket and the slider bar it will be anchored to. I did that by drawing a 1:1 scale version on a piece of graph paper, then transferring the measurements and design into the computer. I ended up needing to change around a few things once it was in the assembly, but the drawings really helped. But before I could even draw it, I needed to know where the holes for mounting the master cylinders were. So I drew a simply diagram, again on a piece of paper, using the measurements from the pedal and bias bar where, then used a compass and a ruler to draw where the master cylinders would end up, and simply measured the lateral distance. It'll make more sense when you see the drawings. Here they are:



Here you can see what I'm talking about. I drew the center of the pedal pivot (bottom left) and measured the height of the center of the bar where the clevises attach. Then I drew a circle with a radius equal to the distance of the end of the clevis from the center of the bar. I knew that the distance from center of the bearings in the MCs to where they would thread into the clevises was 5", so I used a ruler and found where on the circle it would "fit". Then measured the lateral distance.

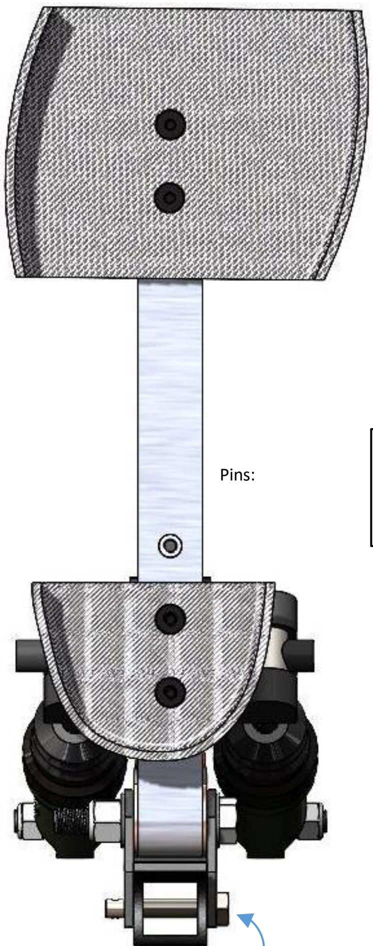
And here is how I designed the pedal base. I drew the holes for the pedal pivot and the master cylinders then drew everything else around it. It's not quite the exact same as the one in the computer as I obviously had to change some dimensions a bit to make it fit. But this is an excellent way to design a part. Much easier to do even a quick sketch on paper to transfer it to the computer rather than simply designing it in the computer alone.

With that, here is the final assembly of the brake pedal:



Here it is. Honestly, not too bad. It's definitely a lot longer than I was hoping that it would be, but that's the way the cookie crumbles. I ended up measuring the distance the from the pedal face to the front roll hoop was in SR19, then measured how far it would be in SR21 and there was only about a 1" difference. So asked Alan to extend the chassis' front bulkhead an additional 1.5" to make a bit more space. I designed the slider bar (again on a piece of graph paper with a ruler) to have holes in it 2" apart.

So overall, the pedal will have 4" of adjustability. Not that I'm expecting to need it, but at least very short drivers can drive it. It should net more points at competition for having adjustability for many different drivers. For attaching the pedal base to the slider bar, I considered using quick release pins that I mentioned on page ** but in the end decided not to use them. This is for one simple reason: I don't know if they could up to the amount of forces being exerted on the pedal. I don't have enough time to try and analysis it, so for safety's sake, I can't assume they won't fail. In its place I have deiced to use grade 8 bolts that I will cut the threads off of and bore a small in the side of it to make my own quick release pins. They will certainly be strong enough and won't be hard to make. I'll simply put a cotter in the hole to hold it into place. PS, yes this meets the rules in the rule book. I don't know if they are



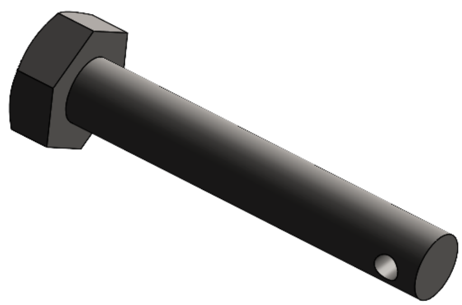
considered critical fasteners, but even if they are they meet both Rule T.8.2 and T.8.3 so it's all good. Here's what I'm talking about:

Some more thoughts: This pedal isn't perfect. The pedal body designed needs refinement and weight reduction. I need to try and fix the pedal ratio for future cars. The goal should be to lower it to 3:1. This will make it so there is less lateral space being taken up in the cockpit foot well.

Grade 8 ¼-20 bolt (2.5" long) with the threads cut off and a hole for a cotter in bored into it. These will be used to attached and anchor the pedal base to the slider bar.

Cotter pin that will be put in the bored hole in the pin to act as a positive locking mechanism to prevent the pin from falling out.

the TFID and the rotors. I really need more driver data for the TDIF, but I



am out of time. And since it is Luke's first year as suspension lead, I don't want to overload him with trying to redesign the rotors and analyze the additional torque on the uprights, so this is as good as it is going to get for this year. I'm honestly proud of what I did, even if it wasn't perfect. The design is solidly rooted in good engineering design and practices. It is a good base for future cockpit designers to build off of. Speaking of which, here are the areas that need focus that I didn't get to: The pedal ratio, the force analysis on the pedal base and the reaction forces on the pedal body from the master cylinders. For the pedal base, I chose 1/8 (11 gauge) steel because I felt that would be strong enough. But there isn't a lot of math or physics behind it. It probably needs some analysis, but I don't have enough time. This is definitely one of the big areas I haven't touched enough on. They could probably be lighter if you were to do some proper FEA on them. Again, a project for the future.

8/20/20

For all intents and purposes this is the end of the brakes system. Obviously, there will be more details to add later, but I'm moving on. Next big thing I'll write about is probably actually plumbing the brake lines. I'm looking into how to do that right now and have been looking for a few weeks. School has started which will dull my progress but oh well. I'm also working on an eBOM for the brake pedal which I will upload when I am finished with it. For now though, I'm done with the brake pedal.

**Brakes System will continue on page 97.*

Throttle System:

And thus it begins. The journey of a thousand steps begins with a single mile. Or something like that… actually that's not a bad slogan for this team if you think about. Philosophy aside, it's time to put my foot down. Both figuratively and literally. Because it is time to work on the noisy pedal. I've had a bit of time to think about how I wanted to make it and how it should work. But first I should lay out what I need to do or not do. First, I'll need to reread the rules on the throttle to ensure it will be rules compliant.

Criteria:

Constraints:

- "The vehicle must be equipped with a carburetor or throttle body" (Rule IC.3.1.1)
    - "The carburetor or throttle body may be of any size or design" (Rule IC.3.1.1a)
- "The foot pedal must return to its original position when not actuated" (IC.3.1.2)
- "A positive pedal stop must be incorporated on the throttle pedal to prevent over stressing the throttle cable or actuation method" (Rule IC.3.1.3)
- "If the throttle system contains any mechanism that could become jammed, for example a gear mechanism, then this must be covered to prevent ingress of any debris"

Throttle System will continue on page 52.

8/22/20

Headrest:

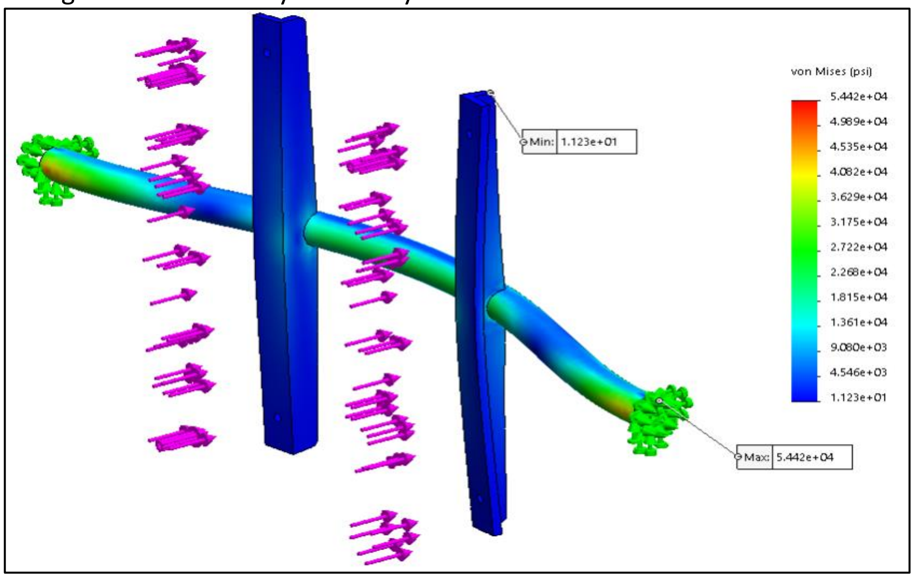
I was going to go over the head rest at a later point, but when I was scanning the rules I noticed that there was a rule about the head rest and the force it had to withstand. It's Rule T.2.8.4. It states:

"The Head Restraint, attachment and, mounting must be strong enough to withstand a force of:

a. 900N [200 lbf] applied in a rearward direction
b. 300N [67 lbf] applied in a lateral or vertical direction"

So, since Alan is finishing up his chassis and is going to be sending it off to VR3 soon, I wanted to make sure we were rules compliant before we sent if off to be cut and welded. With that in mind I tried doing a simple FEA on the headrest back plate, brackets and bar. But I ran into a problem. Solidworks doesn't have material data for composites. Things like yield strength and modulus of elasticity are missing from the list of known characteristics. I tried looking up the material characteristics of the soric core carbon fiber, but there are none available. My guess is that that is because composite material characteristics vary heavily depending on how they are layed up and things like resin ratio and a myriad of other factors. So I said “f*ck it” and just did the FEA without the carbon back plate. I’m very confident it can handle the loads. It’s carbon fiber after all. Here are the results:

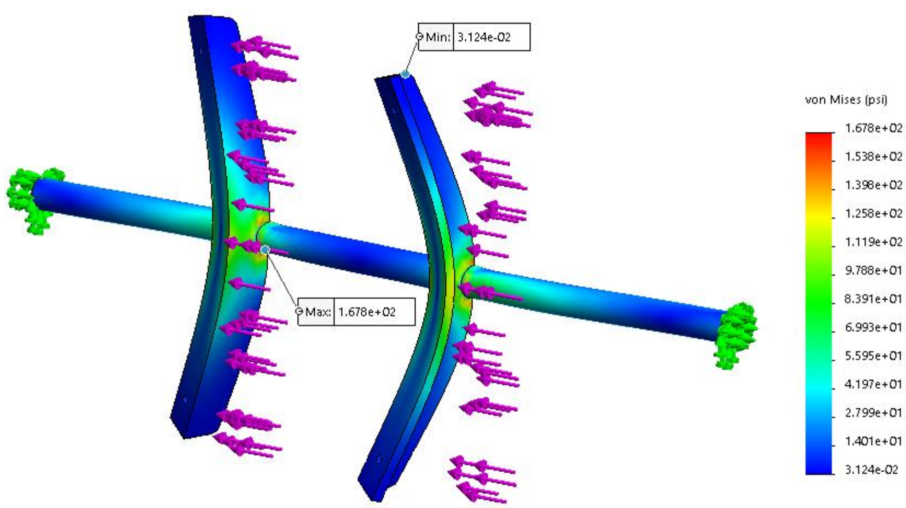
Using the yield strength for 4130 steel (which the tubes and brackets are made of) is 63.1 ksi. Dividing this by the max stress the bar will experience, which is 5.442*10^4 psi gives you the factor of safety, which ends up being: 63100psi/5.442*10^4=1.16. Another way to look at it is the system is 16% stronger than absolutely necessary.



This is the 200 lbf force put on the brackets and bar in the rearward direction. It passes this test.

Again, as above we take the yield strength and divide it by the max stress. 63100psi/1.678*10^2psi=376. So it is a lot stronger than necessary. The reason it is so high is because the load is 1/3 of the 900N load on the first FEA and it is being loaded parallel to the axis of the tube, putting it effectively in tension and compression. Tubes and beams are much stronger when loaded in this manner.

This is the FEA with the 300N applied to it. Again the design passes inspection.



Now that that is finished, I’ll briefly touch on how the headrest will be made, then I’ll move back onto the throttle pedal. The head rest bar will be made at VR3 along with the rest of the chassis and will be welded in-house. The upright steel brackets will be water jetted out at innovation hub, then welded together, but NOT immediately welded on the bar. Instead, the back plate, which will be made out of the soric core carbon that is to be hand cut out and drilled (as it is very easy to do), will be riveted to the uprights, and then the brackets can be welded or at least tacked to the bar. There is a good reason for this. The reason being that if you miss align the holes or the uprights warp when hot the holes for the rivets will no longer be in the same stop making it hard to install the back plate. I know this because it happened on SR19. Live and learn. Finally, the foam pad (which is a special, rules-required foam called Conform) will be cut out and covered in the faux leather we have with a small ~20mm hole in the back for inspection to check we are using the appropriate foam. I will likely try to embroider the head rest cloth cover, but we'll see. That pretty much covers it for the head rest. I'll begin working on the throttle shortly.

10/22/20

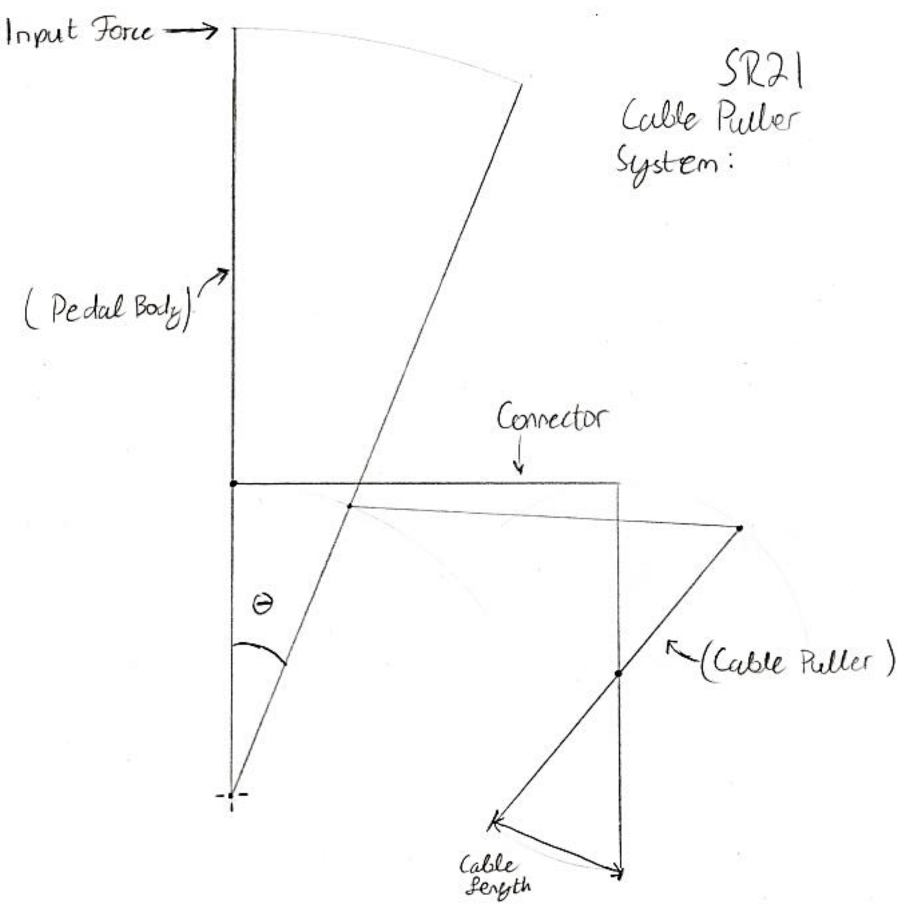
"I'll begin working on the throttle shortly." LOL *psych*. Yeah, that obviously didn't happen. Two months later and it still isn't close to being done. School and personal life have been hella crazy the past 2 months, combined with difficulties being chief engineer. But I'll see what I can do with the time that I have.

For now I need to give an update on what has been happening with the team and specially with my sub team. For the team, we made the decision a few weeks ago to decommission SR19. Both poor design and the pandemic finally killed it. We no longer felt it was an asset worth trying to save. Instead, the more expensive components (ECU, PDM, battery, sensors, wheels and tires, etc) will be reused for SR21. This will save us about $6,000 in parts, not to mention not having to go through the process of ordering them. We also decided to retire SR16. The amount of parts that are close to breaking, as well as all the things that we can't see are very high. Whenever we drive the car, it seems to break very soon after driving it. Repairs are time consuming, expensive and demoralizing. More recently the brake disks in the front have warped requiring us to make new ones. Purchasing the material took 6 weeks, then it turns out someone ordered the wrong type of steel. We got 316 steel instead of 4130 normalized steel, which likely won't work well. We also noticed that the bearings in the front right hub have gone very, very bad. Daniel, the member of the drive train sub team who specializing in bearings informed us the others were likely to go bad soon too. Replacements would cost over $1000. With the likelihood of a catastrophic failure of a component while one of us was driving is fairly high, the older members decided we should retire the car for good. This was voted upon during our meeting unanimously. Both cars will be stored in the main glass window in the back of our shop to display what we build in our workshop. To replace our main training vehicle, the team has decided to buy a KZ class shifter go kart to be used as a training vehicle for current drivers and new drivers. The money saved from SR19 will be used to purchase the kart (this one: https://www.musgraveracing.com/shop/2014-praga-tacho-chassis-w-complete-honda-cr125-engine-package/). We are currently in the process of doing this.

In addition, Alan's chassis was finished a few weeks ago as well and was sent off to VR3 to be cut and bent. It will arrive shortly. We will start the jigging and welding process soon thereafter. As for my sub team, I have been struggling to get anything done with the volume of other responsibilities I have. I did however manage to get the new racing harness ordered. We ended up getting a fully sponsored, FSAE specific harness from HMS Motorsport, which is pretty awesome. That arrived yesterday. In terms of design, what I need to do next is to design the throttle pedal and fix the issues I'm having with the fire wall. More on that later.

Throttle System (Continued from page 49):

The biggest design challenge I'm facing right now is trying to match the pedal input ratio to the output ratio of the throttle cable puller to ensure the throttle will open up sufficiently. In other words, I need to make the pedal move enough that it can pull the cable to open the throttle all the way without having to extend the pedal very far. SR16’s pedal has a fairly long throw, but only moves through an angle of about 22°. The length of the pedal is about 8”. The length of the pedal for SR21 will be 10” to match the length of the brake pedal. This means the driver’s foot will need to extend even further than in SR16, so I want 22° to be the limit. The angle the disk on the throttle body needs to move is 72° and the diameter is 1.25”. I’ll put a diagram below for clarity:



In order to figure out how much the cable puller on the throttle will need to pull the throttle cable, I first need to calculate how much length of cable needs to be pulled. This can be simply calculated using the following expression: $L_{Cable}=2\pi(\frac{D}{2})(\frac{\theta}{360})$. Plugging in the variables: $L_{Cable}=2\pi(\frac{1.25"}{2})\left(\frac{72^o}{360}\right)=\frac{\pi}{4}=.78539"$. This means the cable needs to be pulled a total length of 0.785” by the cable puller.

10/23/20

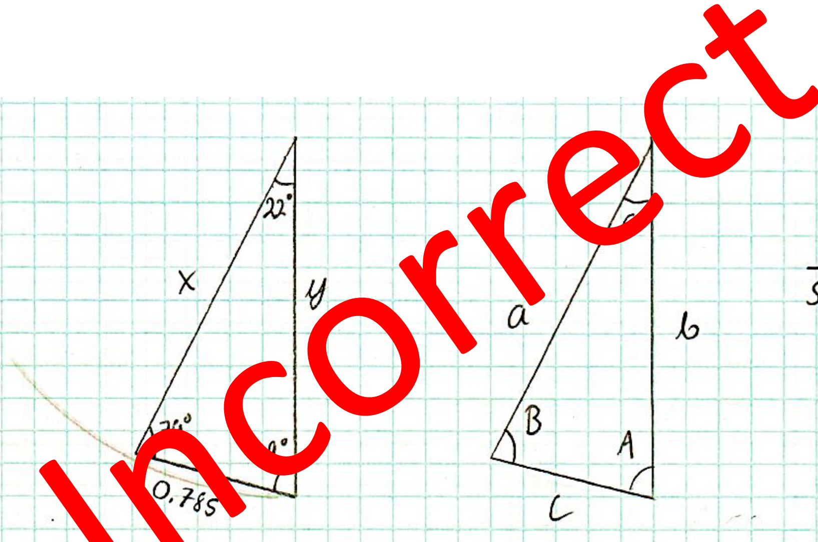
The way I’m planning on designing the pedal is similar to a system I became quite familiar with last semester in Design of Machinery: a Grashof 4-Bar Double Rocker mechanism. One of the bars with be the pedal body, the other being the cable puller. The base will be the pedal base, and the final bar with be the connector between the pedal body and the cable puller. Below is a basic diagram for what I’m thinking:

Obviously, this isn't exactly how it will work, but it's a close approximation to it. The pedal body input will cause the cable puller to pull back. The difficulty will be in determining the lengths of the cable puller, and the location of it's pivot relative to that of the pedal body. This is what has been holding me up for awhile since I don't know how to get past it. I'll try again tomorrow.

10/30/20

So after a week or so of mental gymnastics, I've figured out that I that I need to find the length of the cable puller arm from the end to the center pivot. That way I can make sure the cable puller sweeps through enough of an arc to pull the cable far enough. I did that with the following math/geometry:

I decided to us 22° as the input angle. I did this so it matches in the input angle of the throttle pedal. This should make the rest of the geometry easier to determine. The other two angles I determined by using the fact that the interior angles of a triangle add up to 180°. I've already figured out the 22° and other two will be the same by geometry. Then, I used x and y as the variables, but these are obviously equal, as it is the same arm at two different positions. I then used the Law of Sines to find the



$$\frac{a}{\sin(A)} = \frac{b}{\sin(B)} = \frac{c}{\sin(C)}$$

$$x = y$$

$$\frac{180° - 22°}{2} = 79°$$

$$\frac{x}{\sin(79°)} = \frac{0.785''}{\sin(22°)} \Rightarrow x = \frac{0.785''}{\sin(22°)} \cdot \sin(79°)$$

$$x = 2.057''$$

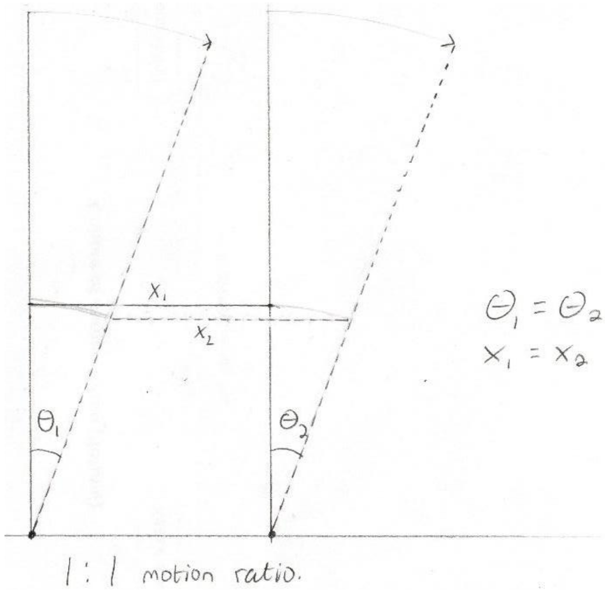
Find the corrected version on page 58.

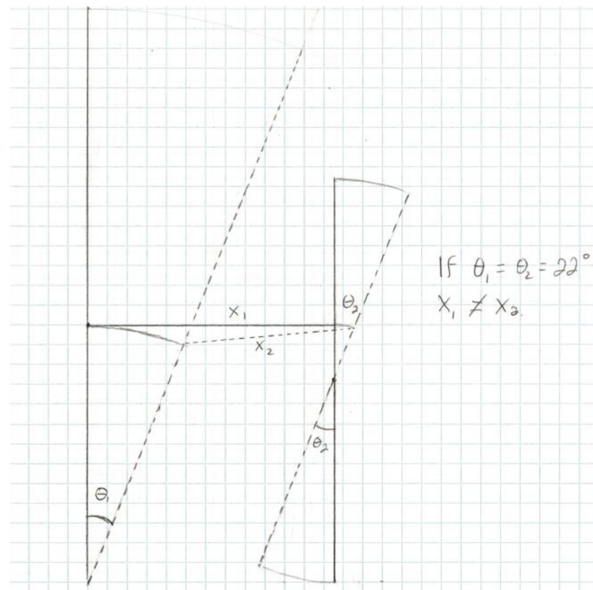
length of the arm. 0.785” is the length the cable puller needs to be displaced. In the end I calculated that the length needs to be 2.057” for an input of 22°.

11/5/20

The only thing left for the cable puller is to decide where I want the pivot point to be, but this should be relatively simple. Once that has been decided I can do a quick motion position sketch to ensure that parts will work as expected. I want the bottom of the puller to be near the floor of the car when the pedal is installed in the car with the closeout panel also installed.

Ahhhhg. I messed up. I forgot that if you want to have a 1:1 motion ratio like I was trying to do, you have to have the two pivot points on the same horizontal plane. In other words, the way it’s set up, I won’t get 22° of output from the cable puller if I use 22° of input from the throttle pedal. It’s difficult to explain in words so I’ll try to draw it:



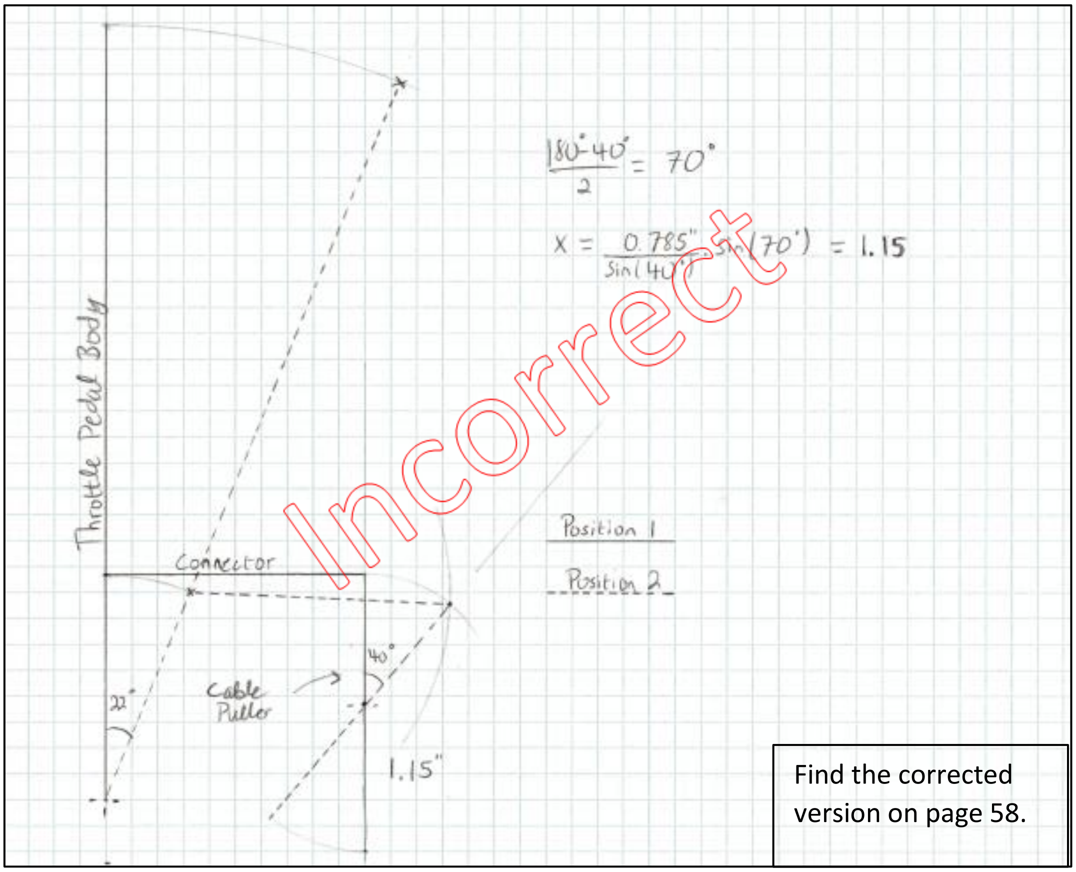


You can see from the first photo what I mean by a 1:1 motion ratio. Theta 1 is the same as theta two, meaning that the second rocker arm will move the same amount as the input. The connector denoted as X1 and X2 (same connector at different moments in time and space) must always remain the same. But you will notice that the pivot points are at the same horizontal location. If we take a look at the second picture, we can see that if we try to the theta 1 and theta 2 to be the same with the pivot centers at different places, we simply can't because the connector would become shorter and shorter as the pedal travels through its input motion. Thus X2 is shorter than X1, and this cannot happen.

11/6/20

I've spent the past while trying to figure out what I was doing wrong with the design, and how to make the 4-bar cable puller work. It doesn't help it's been a while since I took Design of Machinery with Dr. Lankrani. That sort of material is relevant to what I'm trying to do.

I think I've figured it out now. I went about trying to design it the wrong way. I'll try to explain how I came up with the following drawing:



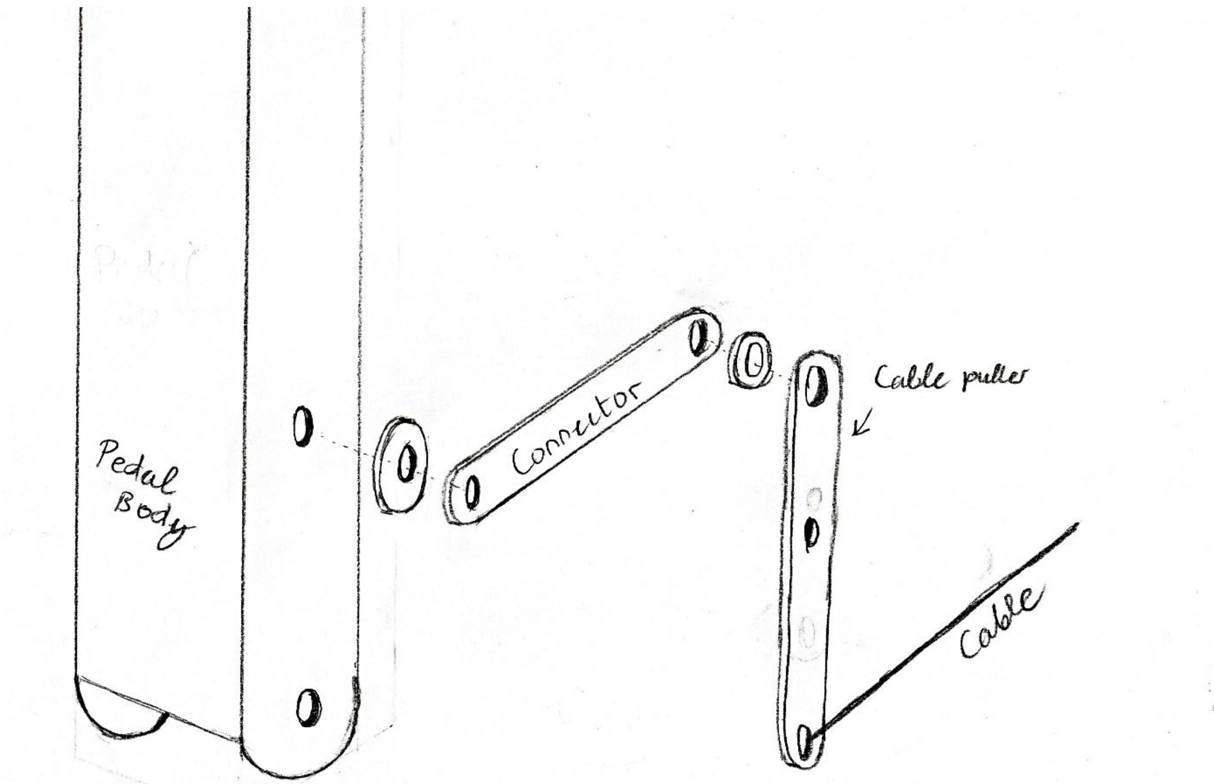
Find the corrected version on page 58.

I started by drawing the throttle pedal line (at position 1). I then drew it at position 2. The dashed lines represent the same bars are their maximum position. From there I figured out how high I wanted the connecter to be, then drew it to the desired length. In this case it is 1.75” high, and 2” long. I then drew a long line downwards from that point vertically towards the bottom of the page. This will serve as a temporary cable puller bar. I then used my compass to draw an arc from the connector pivot point on the throttle body to its equivalent point on the position 2 throttle. I then put the point of the compass on that new point and swept out an arc with a radius equal to the length of the connector. I then had to choose an arbitrary length on the cable puller bar to put its pivot point. In this case I those 1” down from the connector attachment point. I chose this because it seemed to me an appropriate length to use. Once again I used my compass to sweep another arc from the point until it intersected the bigger arc I had previously drawn. That demarcated the point at which the cable puller would end at position 2. I then draw a line from the point to the original cable puller pivot point. I then measured the interior angle this created, which turned out to be 40° then did the same trig I had done a few pages ago to find the puller length. I then marked this length, which is 1.15” on the extended line I drew at the beginning, and erased the rest. Then once more I used the compass to sweep out and find the end point at position 2. Last but not least I drew the connector at position 2 and verified it was the same length it was when at position 1. It is and the design will fit within the space I need it to on the throttle, so this portion of the design is done.

11/21/20

Update: I've been gone since Friday a couple of weeks ago. I had to quarantine due to COVID for about a week, then finals earlier this week, then I had a funeral of a family member to attend. I have just now had a chance to be back and get some things done in the shop. I've forgotten some of the things that I was thinking about last week in terms of the design, so I'll spend the next while looking over my notes from my last few entries, and catch up with the other members to see what I've missed. I'll continue to work on the throttle design after that.

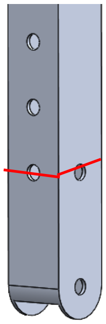
Ok, so I've done that, and I'm ready to move ahead a bit with the design of the throttle. I've come up with a quick sketch of what I'm intending to do with the cable puller. Here it is:



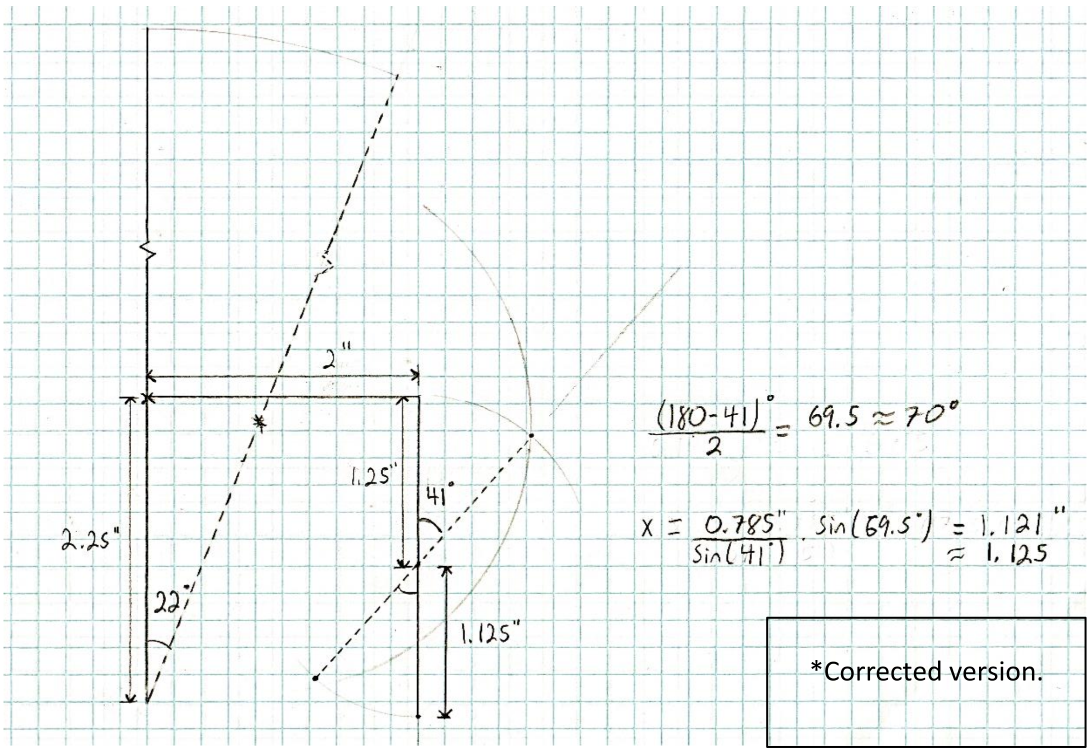
On the left is the pedal body, then the low friction washer, the connector link, another low friction washer, the cable puller link, and a representation of the orientation of the cable. A method of attachment will be created later. I'm planning on putting a #10-32 thread rivnut in the hole in the pedal body, thus avoiding the issue of putting a nut on the inside of the aluminum tube pedal body. I'll just use a normal bolt and nut arrangement for the connector and cable puller links.

12/4/20

Once again I discovered a problem with the cable puller design (*eye roll*). Turns out the place where I put the hole for the connector link to attach was right alongside the hole for the heel cup. The reason this is a problem is because either the bolts or the rivnuts will contact and interfere with each other.



This is shown in the screen cap to the left. You can see where the bolts would interfere with each other were they to be threaded in at the same time, which (presumably) they would. I'll therefore have to move the hole location for the cable puller higher up. This will of course mean that I will have to redo the geometry for the cable puller, but this shouldn't be too difficult. Here it is below:



The new lengths are slightly different than the previous iteration. The link length of the pedal body has been increased by .5", and the angles and lengths of the cable puller link has also changed. Also, in the calculations I rounded up slightly on the final number. This will make it a more standard dimension (1 1/8"), and the difference is only 0.004" which is about the thickness of a human hair. It isn't large enough to care about, so I ended up rounding it. A quick exanimation of the drawing confirms the 0.785" distance is met. This should work fine now. I'll go update Solidworks and continue.

Throttle System will continue on page 77.

1/8/21

Steering Wheel:

I'm taking a break from finishing the pedals to work on the steering wheel. Partly because I need a change of pace, but also partly because Alan (chassis lead) needs to place the steering column support in the SolidWorks model so he can submit the SES document on January 20th. I've been working on it here and there for a few months now, mostly toying with some ideas. I've finally gotten enough done to where I can actually write down some stuff. From as early as this time last year I was trying to figure out what I wanted to do for the steering wheel, looking at different data display options. I eventually decided upon the Motec D153 unit for various reasons, but the most important of which was being able to put the screen in the steering wheel of the car. This has been my goal since Summer of 2019 when we began the very beginnings of building this car. At the competition that summer (the one where we didn't even have a working car) I got my *** reamed out by the judge for the placement of the data display on SR19. I didn't want to put it there, but I didn't have much choice in the matter, and I got roasted for it. So it's been one of my biggest goals to make something better with the display already in the dash, much like a modern formula wheel. The reason we couldn't simply buy a premade unit from a professional company is because of the rule about no concave sections in the steering wheel (V.3.3.4). "The Steering Wheel must have a continuous perimeter that is near circular or near oval. The outer perimeter profile may have some straight sections, but no concave sections. "H", "Figure 8", or cutout wheels are not allowed." Most, if not all good options were illegal by this rule. This could be why the rule exists, because I can't think of any other good reason for it. In any case the following wheels, which all would have been good choices, were disqualified from being used as a result:

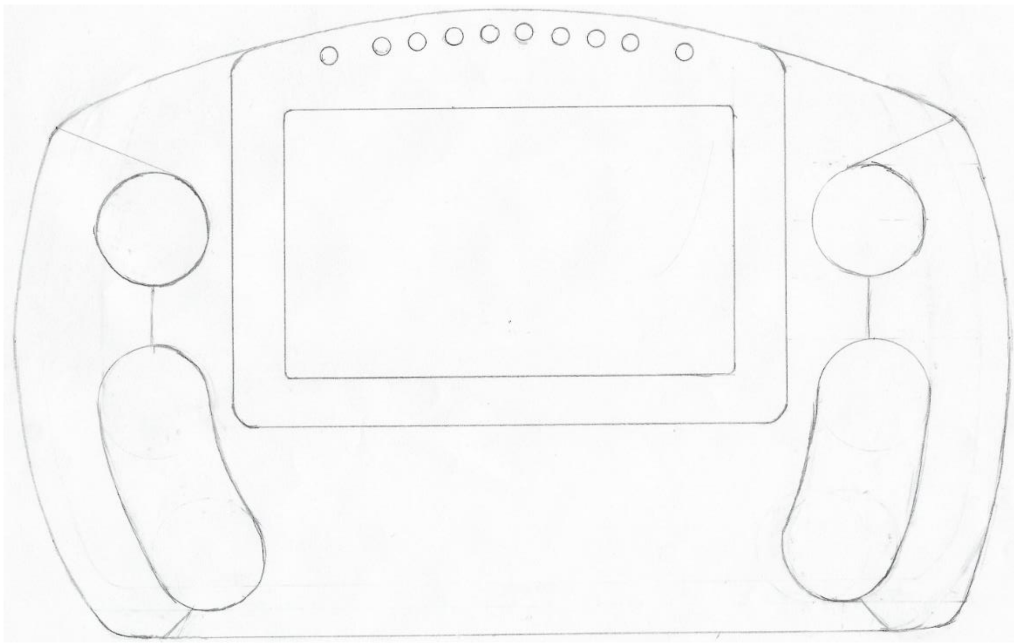
Image from AIM Technology

Image from XAP Technology


Image from XAP Technology

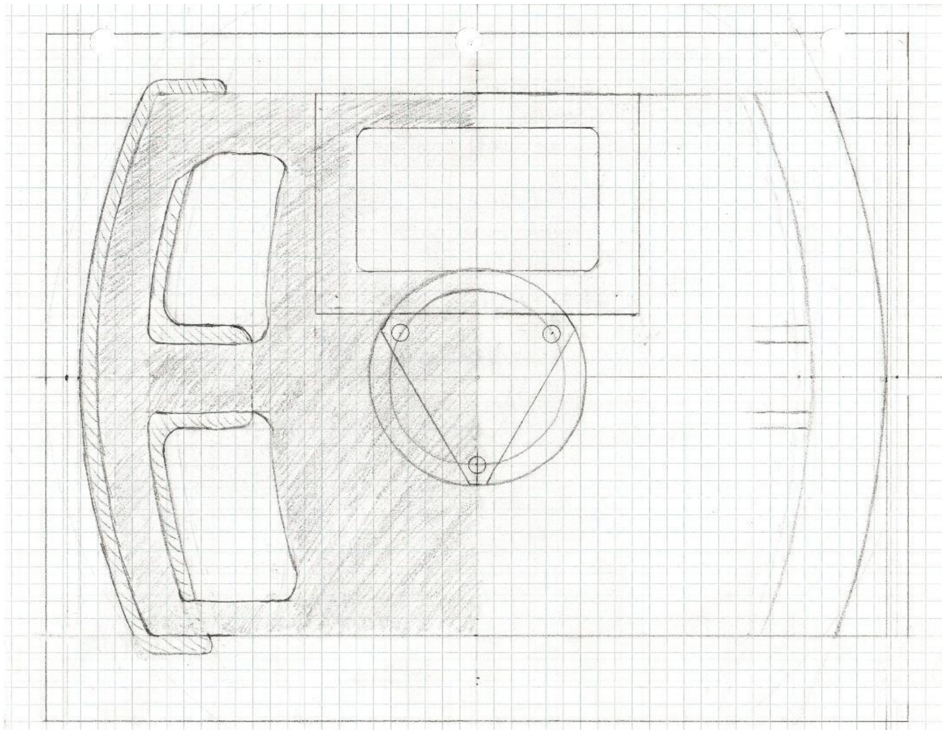
This basically leaves me with the two options of using the old steering wheel styles from SR16 and SR19, or making an in house steering wheel. I've obviously decided upon the latter. When I was looking for inspiration I came across this image of another formula SAE team's steering wheel:

This wheel is from the University of Victoria in British Columbia. The earliest example of it appears on their 2016 challenger. It is probably the best looking student designed wheel I've seen to date. This is was the wheel I wished to imitate in my design. I tried to figure out how to do it. I made an initial sketch some time back, perhaps in April:

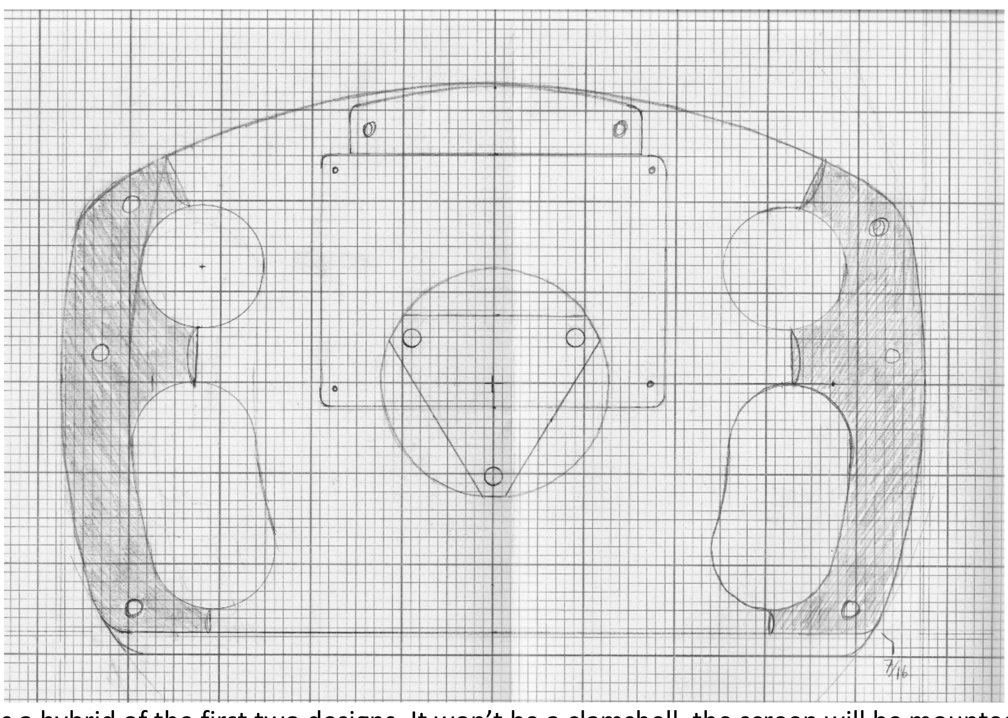


This was back when I was still considering using the Motec C125 data display from previous years, hence the much larger screen. This one was to be a clamshell design like the UVic wheel with the display nestled in between the structural metal plate and the shell enclosure. I ended up shelving the design for a while whilst I was busy with other tasks and projects.

Recently, I revisited this design when I started trying to actually design the wheel. I tried using a different wheel style that would be easier to create and that featured the D153 screen. It was to have the screen behind the structural plate with a small cutout for it, much like the second XAP wheel on the last page. This meant I would have to keep the screen above the area where the quick release would need to be mounted. It ended up making it a bit taller. The partially finished drawing I made for it is here:



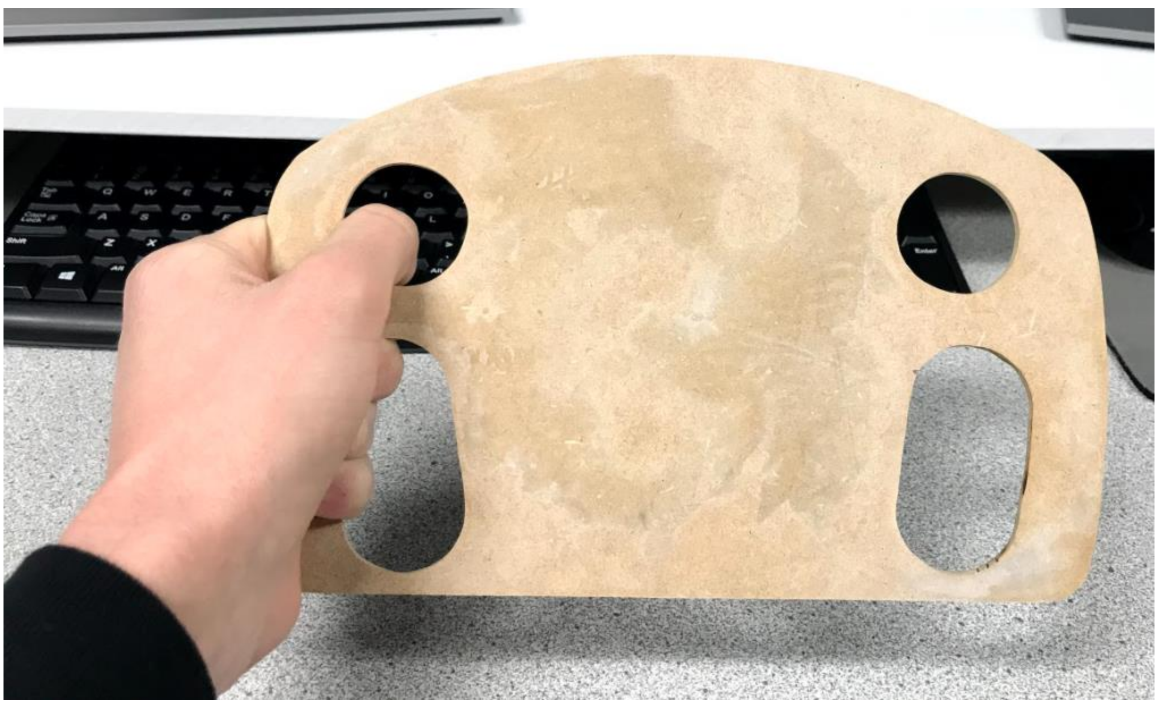
The drawing is only half finished because I thought it was so big and ugly that I didn't want to finish it. Instead I came up with this over the last few days:



It is a hybrid of the first two designs. It won't be a clamshell, the screen will be mounted to the front of the metal plate, and there will be two grips for the hands, but there won't be anything covering them. This is to make manufacturing easier. Could I design a better one? Of course. Could I do it in time? No.

For the wheel design, one of the more important things is the diameter of the wheel. The wheel must not be mounted so that any part of the wheel in any orientation may rise above the top of the knee hoop, according to the rules. V3.3.1: "In any angular position, the top of the Steering Wheel must be no higher than the top-most surface of the Front Hoop." What this means is that no matter how the wheel is turned, it can't be higher than the top of the knee hoop. This means I had to pay special attention to the maximum radius of the wheel. I couldn't let it get too big or I'd have to mount the wheel lower to compensate and encroach on the driver's legs. Many of the larger team members complained to me about SR19's wheel position because it was so close to their legs they couldn't turn it. While I tried then and will try now to mitigate that, it must also be understood that not everyone will be able to fit in the car, no matter how much they was to. It's unfair and it sucks, but this is a competition. We could make the car big enough to fit everyone, but we'd lose a lot to weight and bulk of the car. I personally never had much trouble moving the wheel. It's tight, but that's how formula cars are. For reference I am 6' 150lbs.

Bearing all that in mind, I have been working on the wheel design, and as of tonight I have the beginning of it in Solidworks. I didn't do much in SW tonight because I decided to make a physical mock up the wheel to ensure it was big enough to fit our hands comfortable. I ended up making it out of some spare wood sheeting we had from a previous order's packing material. Here it is:

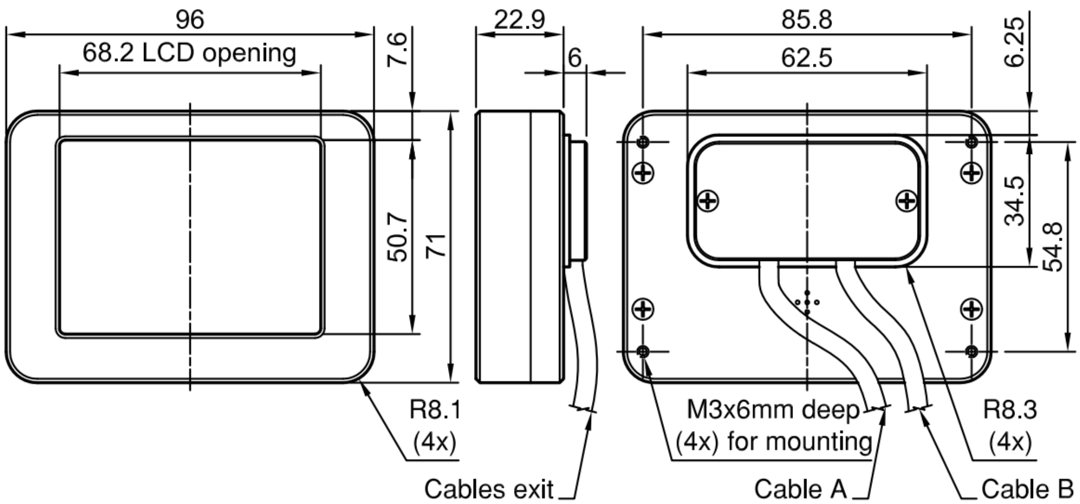


The wheel obviously doesn't have any grips, which makes it a tad uncomfortable to hold, but the sizing seems decent. I have fairly large hands and mine fit comfortable in the openings I designed, even with gloves on. I had a few other team members try it out to see what they think and they seem to be fine with the shape and size. I do have a small concern about the size though. The area between the thumb and finger holes it a bit small, and I may have to make it larger if the FEA tells me it will be too weak. It shouldn't be a huge deal though. Since it seems to be a good design I'll press on with the design. This is probably all I'll do tonight though, since it's late and I'm getting tired. Tomorrow I might make some tweaks and keep working on the SW model.

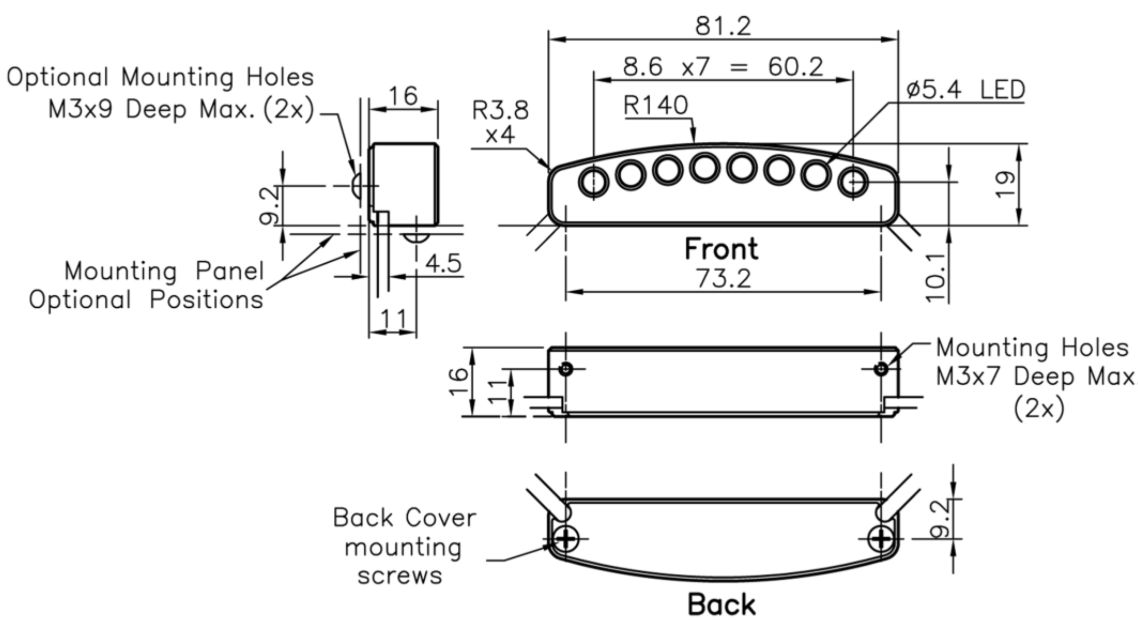
1/9/21

Today I'll begin working on finishing up the wheel. I thought about it some after I went home yesterday and I think that the space between the thumb and finger holes I mentioned last night will be fine. If the FEA comes back to say they need to be stronger, I can make the material thicker without much trouble. For now I hope to have the holes for the data display and shift light array in by the end of tonight, and since I'm feeling optimistic, I might be able to have parts of the hand grips done as well. No time like the present!

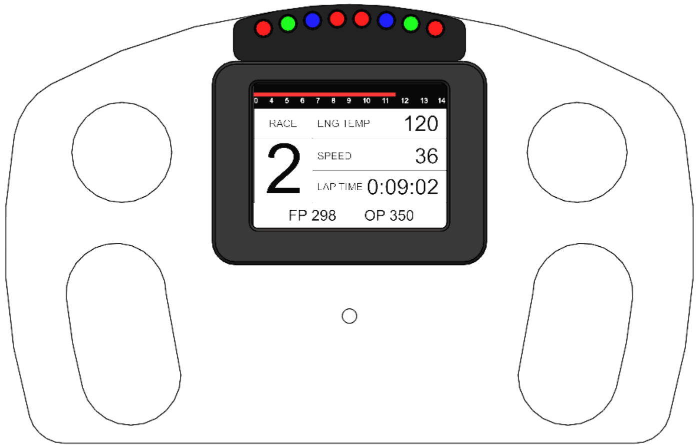
Also to make it easier and for my own future reference I'll put the technical drawings for each below:34.5

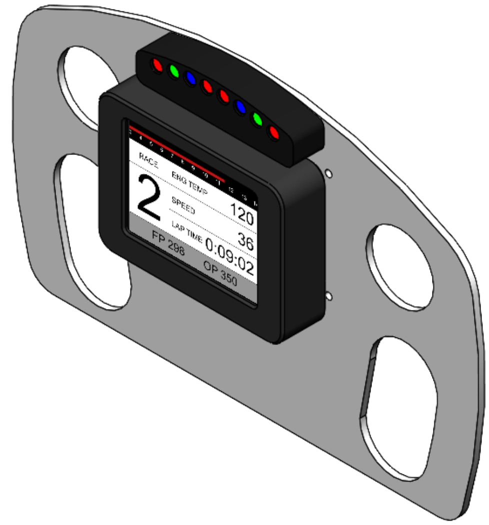


All dimensions in mm.

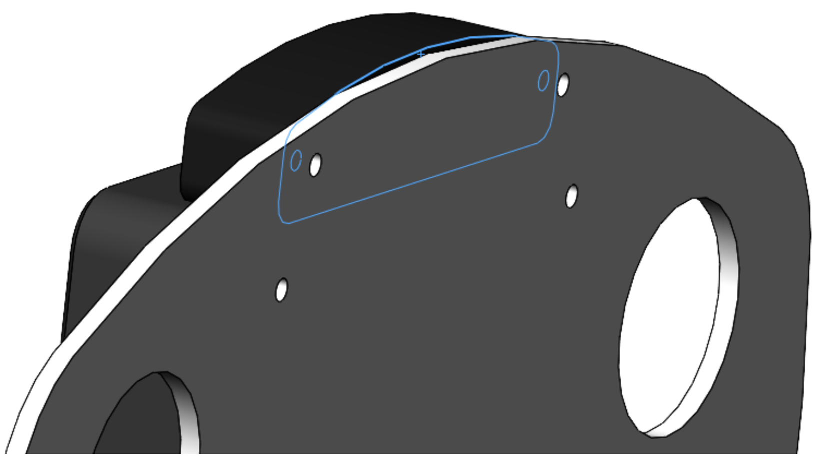


I finished placing the holes for the shift lights and the data display. Here it is:





Above: to check that I had gotten the holes in the right position, I made a quick assembly of the pieces I have made so far. They seem to fit pretty well, and should work well when they are installed. I will have to make a spacer between the wheel body so I can properly bolt the quick release on. (Only cap head bolts are accepted. I learned that at tech inspection with SR19). But this is a good start. There is one thing slightly concerning to me though:

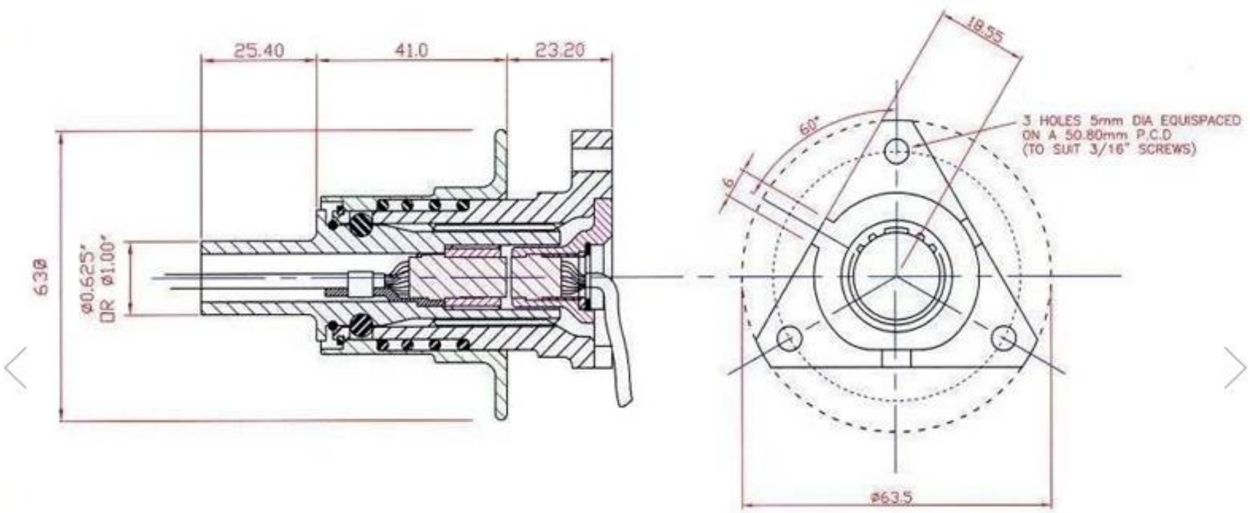


When we look at the back of the wheel, we can see the top of the lights sticks out a bit over the edge of the base plate. This was not intended to be that way. However, since this won't be a clamshell wheel it isn't a big deal and I'll leave it for now. When I measure the overhang it comes out to be .03in, which isn't much.

I think that'll be all for tonight as it is getting late and another ice storm is supposedly headed here. I'll try to work on it more tomorrow.

1/13/2021

I'll need to keep using this for further reference, so here is the drawing for the SPA Technique quick release that we used on SR16 and SR19. The one we are using this year is a BG model (BGR700), but is very similar:

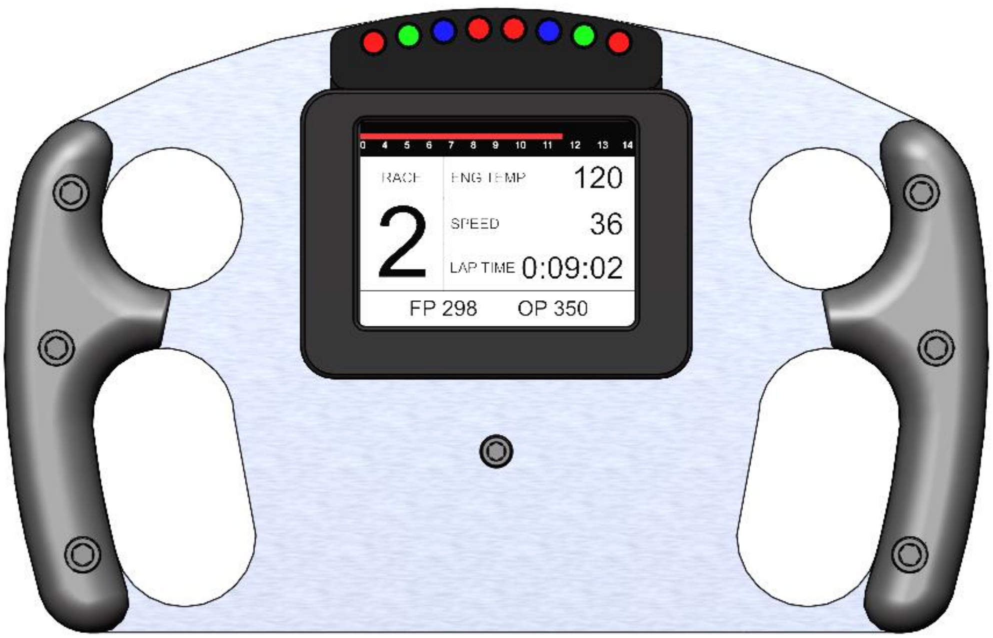


Update: I've finished the riser for the steering wheel, and have most of it assembled. The grips are the only thing that are left to do beyond putting a few nuts and bolts into the assembly. The grips will be a pain in the butt, so I may end up doing them tomorrow. But the wheel is almost done:

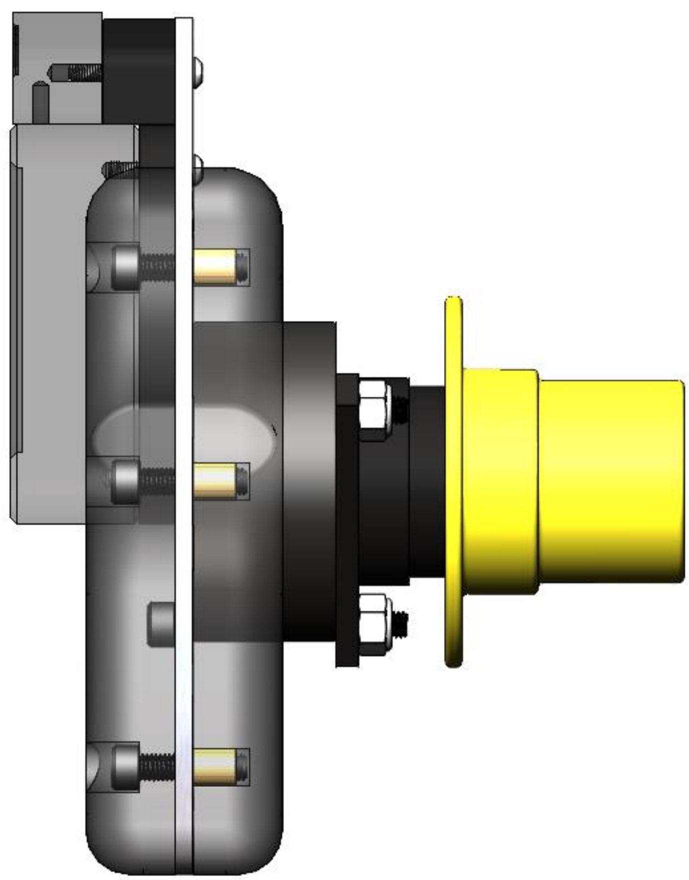
I also added a spacer on the quick release. There are a handful of companies that make them in various thicknesses, so if we decide during testing that we want the wheel in a slightly different direction, we have options to do so. I think this will be all for tonight. Oh, and I think I may have made a slight mistake in creating the shift light model which would explain why the it has that overhang I talked about. I'll try to fix it when I get time.

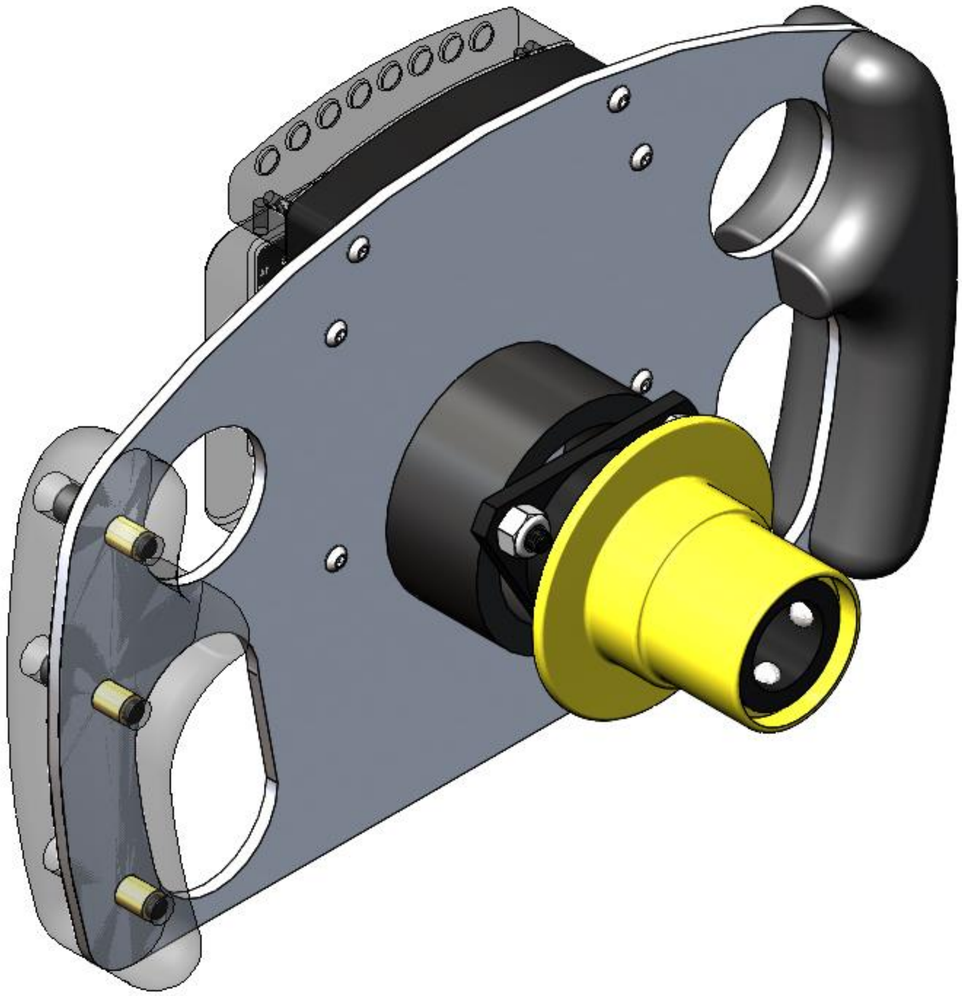
1/14/21

I've finished the steering wheel tonight. It took a while but it Is done now and turned out not too bad:




RACE
ENG TEMP
120
2
SPEED
36
LAP TIME
0:09:02
FP 298
OP 350





There is a 1” spacer ring on the back I can get from a few different suppliers. The quick release like I mentioned earlier is BG, but very nearly identical to the SPA Technique one, so the model is adequate. The bolts holding it on are 10-32 x1.625 black oxide socket head bolts with nylock nuts on the other side. I still haven’t determined the material the base plate will be made out of, but it’s either going to be steel or 7075-T6 aluminum for strength. In either case I’ll have them water jetted out, putting only very small holes for the bolts as pilot holes, then finishing them out with precision bits and tapping where necessary. The grips on the side are slightly contoured for driver comfort, and will be bolted together using 10-32 bolts 0.75” long, fully threaded. They are to be black oxide like the rest of the bolts we use. These should be strong enough. The grips themselves will be 3D printed out of a strong material that is yet to be determined (likely ABS). I’ll probably rubber coat them black with Plasti-dip (that’s what the stuff was originally meant for) to give it good grip. The small bronze fittings are metal inserts that are heat pressed into the plastic to allow the two sides to be bolted together. The D153 data display and the shift lights are both held on with M3 button head bolts, the ones on the data display being 15mm long, and the ones on the shift lights being 22mm long, because of the extra thickness of the riser I made. The riser will also be 3D printed. There will have to be a channel put into the riser to accommodate the cables from both the data display and the shift lights, but since I don’t have either right now, I’ll figure out cable routing later.

1/15/20

While the wheel is basically finished (I even tried to do a render on it because I had some time to kill, which didn't turn out very well at all), I still have a bit more to do on it. Namely the FEA on the base of the wheel to ensure it is going to be strong enough to handle the forces being applied to it (no pun intended). However, I'm still waiting on the calculations for the torsional forces from Suspension. I'm also waiting to have the grips for the wheel 3D printed so I can tape them to the wooden wheel cutout I made a week ago so I can test the ergonomics. But the guy I got to print them is feeling pretty sick, so I won't get them for a while. Till then, I'll have to work on other stuff.

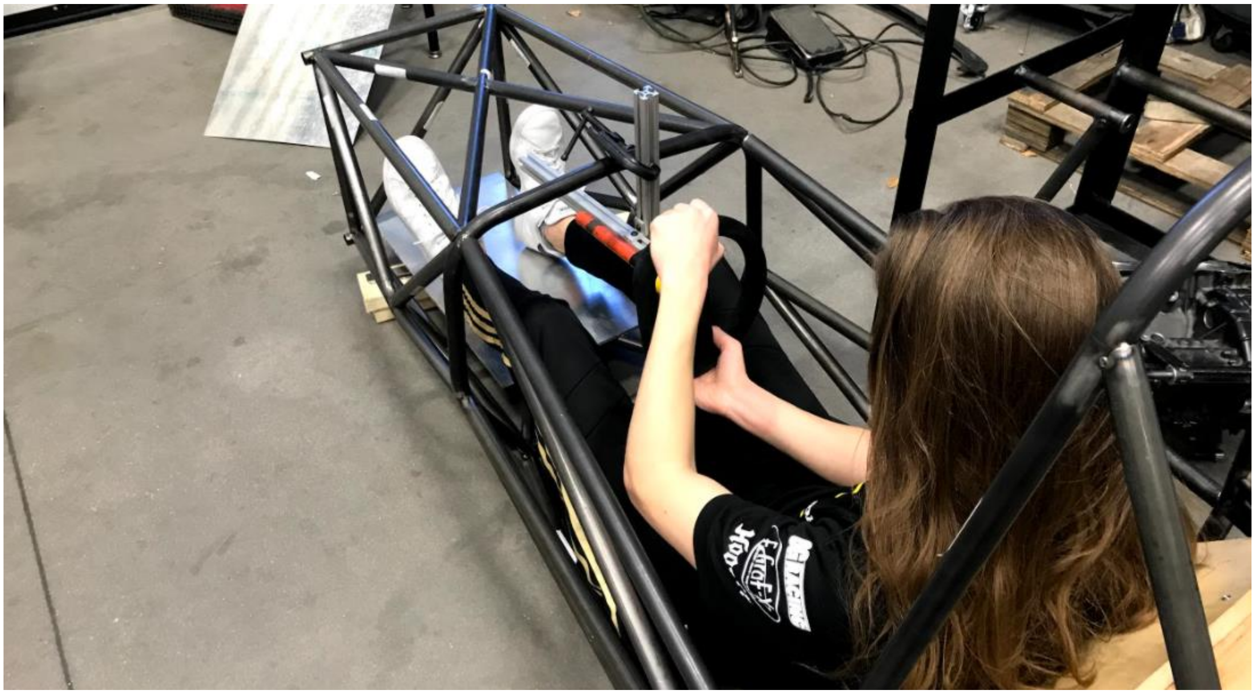
Steering Wheel project will continue on page 72.

Steering Column:

This could have fit in with the steering wheel project, but it affects the steering column much more, so I'll put it here. A few days ago the chassis lead and I worked together to place the steering wheel inside the chassis so that he could place the column support tubes in the chassis and finish the SES document. To do this he had already tacked the chassis together so that it didn't need to be jigged the whole time. Therefore, I was actually able to get inside it and see how it felt. It's a lot bigger than SR19's cockpit for sure, but a lot of us couldn't fit in the car very well aside from me. At any rate we ended up using the steering wheel from SR16 to figure out how far away from the knee hoop to place the wheel. We did this by using a very simple 80-20 jig clamped onto the chassis:

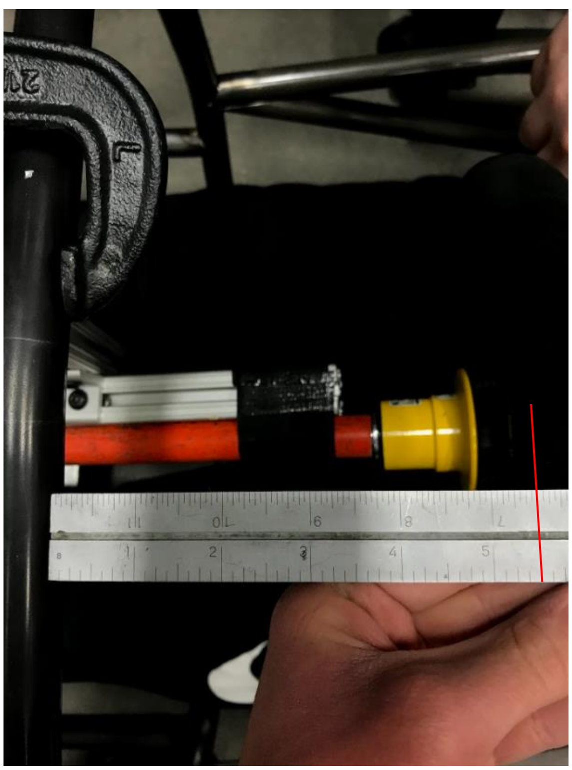
To help simulate the driving position we used a piece of MDF to act as the firewall and a flat piece of aluminum to place out feet. We had the steering wheel mounted on a scrap piece of steel that we tapped to the 80-20 to represent the steering column and help us find out angle. From SR19 I know that the steering rack must be placed 20 degrees from vertical to allow the steering rack to work. I also know from SR19 that the tube connecting the to the steering wheel must be at a 10 degree angle from horizontal. When I go into detail about the steering column later I'll explain why.

We ended up taping a digital angle finder onto the 80-20 so we could get an accurate measurement and simply adjusted the set screws on the 80-20 till we got the wheel were we wanted it.



(Yes, this is a guy in the picture. In fact, it's me. I just really long hair.)

Once we roughed out where it needed to go I jumped in and played around with it for a bit. I ended up scooting it up closer to the knee hoop because I was having to hold the wheel like a T-rex, which would tire my arms out much sooner. Once we found a good position we left it overnight and had more people try it out the next day. They liked the way it was so that's how it's going to be.



Looking at the photo on the left we can see the measurement at 5.5 inches from the rear of the knee hoop to the front of the Steering wheel grip (it is a little hard to see). The maximum length this can be in the rules in 9.8" (250mm), so we are well within compliance. The top of the wheel in any orientation is also below the line on the top of the knee hoop, so that is also rules compliant. Overall, the seating position is fairly comfortable, and even larger drivers shouldn't have any reason not to be able to fit in this chassis. It is definitely bigger than it needs to be, but I'd rather have a larger chassis to begin with and start putting it on a diet as the iterations progress over the next few years. This will also help cut weight as well without having to go through an arduous process of tube reconfiguration. Again, this is a foundation car so it needs to be a solid base.

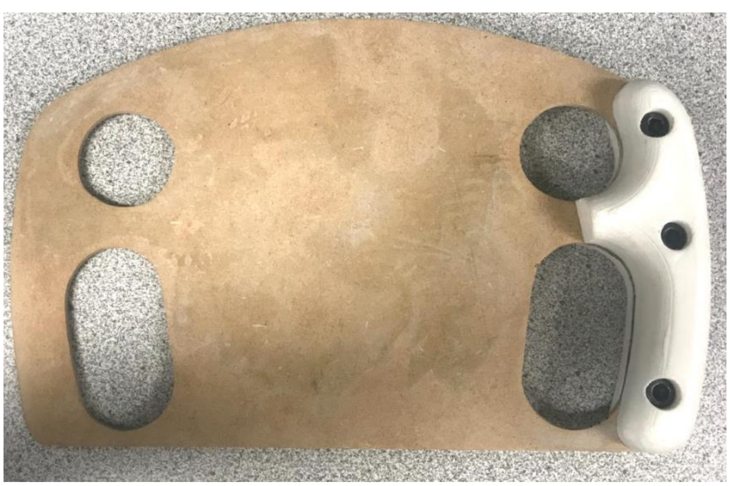
As we can see from the photo above, this placement should give plenty of leg room (there is more than there appears in the photo. It's a bit of an optical illusion). I'm happy with it, so Alan make a quick mockup of a central tube and tacked it onto the chassis so we could once again test the placement and his idea for the hanger.

Steering Column project will continue on page 88.

1/20/21

Steering Wheel (Continued from page 69):

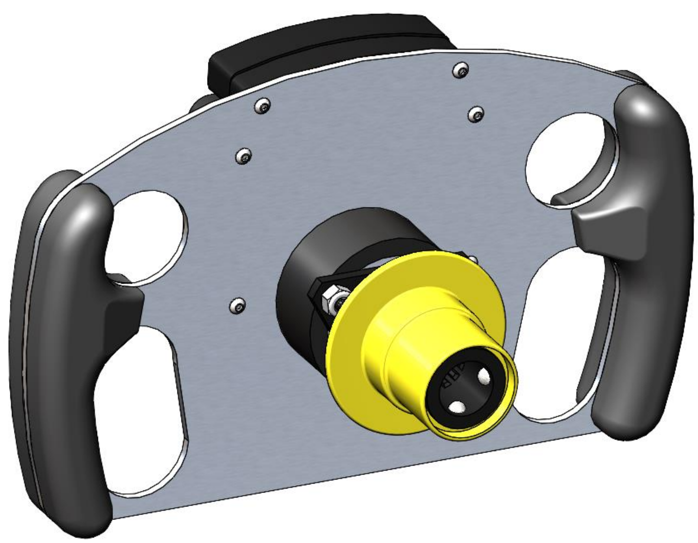
Update: I've had the grips on the right side of the wheel printed (thanks to Felipe) and installed them on the wood mock-up I made a few weeks ago:



I had only the right side made since it was to be an initial design and not the final one, and I wanted them as soon as possible. Unfortunately, Felipe fell sick with Covid, so he wasn't able to deliver them until today, or I'd have had both sides of them made. Oh well. The grips feel good to hold for the most part. The thickness and width are comfortable, and the bolt holes in the front don't bother the hands that hold them. The curvature feels natural, and the fillets on the edges feel good as well. Overall the design is fine, but there are a few I will made to fine-tune the design.

The front part of the grips are fine, the bolts sit low enough that they are below where the hand will rest, and like I said earlier are comfortable. But on the back side where the fingers will rest is a little different. In my original design (like the ones printed out), I had design hexagonal holes for nylock nuts to sit to accept the bolts that go through the front to hold the whole thing together.

But in the later revision (like the one shown on pages 66-68) I had decided to use heated rivserts to accept the smaller bolts (1/4-20 to #10-32). So the holes at the back will be gone. In addition, after holding the grips today, I decided that the backside of the grips is not quite as comfortable as it could be. The way the wheel of designed to save space, it meant that the index finger would wrap around the wheel rather than go through one of the holes. This means that it will rest on the raised grip portion. However, as the picture on the left shows, this isn't very natural and slightly uncomfortable. To remedy this, I've cut out a triangular section front top to bottom on this portion of the grips, and it should help:



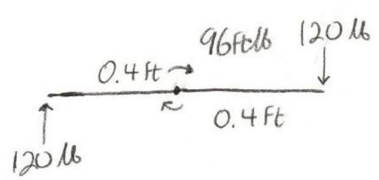
As you can see, the profile of the rear grips is now more ergonomic and should be easier to grip. I'm having this final iteration of grips 3D printed right now and will have them installed as soon as I get them. I'm still waiting on suspension to finish their calculations on the forces the wheel and column, and until I have those numbers I'm hesitant to perform any FEA on them. In the meantime, I'll try to finish up the throttle pedal and get that out of the way, and I'll try researching what the maximal load on the steering components should be. I remember that there is an article about it floating around somewhere, I just can't remember where. When I find it, I'll try to link to it as usual.

1/22/21

Ok, after a bit of searching I found the article I was looking for. It's *Cockpit Control Forces or How Robust Do Driver Controls Really Need To Be?* by Steven Fox (2010) (http://www.fsaeonline.com/content/Cockpit%20Control%20Forces%20SI%20SAE.pdf). The paper is also in the Research and Technical Documents folder if necessary. At any rate, in the article Steven explains that the steering system should be able to withstand a minimum of 100 N-m (75 lb-ft) of force applied at the wheel. This will help me in doing my FEA for both the steering wheel and steering column. Now, I have to figure an appropriate factor of safety. The article says that 100 N-m should be the minimum it is designed to, but not the maximum. Therefore, I'll still have to find an appropriate factor of safety to implement for the FEA. Based off of an article from The Engineering Toolbox.com (https://www.engineeringtoolbox.com/factors-safety-fos-d_1624.html), an appropriate factor of safety for general applications where the environment is not severe and weight is an important consideration is 1.3-1.5. In other words, 30-50% bigger/stronger than it needs to be. Therefore, an appropriate loading I need to use should be between 130 N-m and 150 N-m.

Applications	Factor of Safety - *FOS* -
For use with highly reliable materials where loading and environmental conditions are not severe and where weight is an important consideration	1.3 - 1.5
For use with reliable materials where loading and environmental conditions are not severe	1.5 - 2
For use with ordinary materials where loading and environmental conditions are not severe	2 - 2.5
For use with less tried and for brittle materials where loading and environmental conditions are not severe	2.5 - 3
For use with materials where properties are not reliable and where loading and environmental conditions are not severe, or where reliable materials are used under difficult and environmental conditions	3 - 4

Since lightness is a concern, I'll choose 1.3 as my factor of safety (FOS). I'll now need to perform FEA on the baseplate of the steering wheel. Therefore, I need to use 130 N-m (96 ft-lb) as my max input torque.



9.5 in ÷ 2 = 4.75 in → 0.4 ft

96 ft-lb ÷ 0.4 ft = 240 lb

240 lb ÷ 2 = 120 lb

Checking:

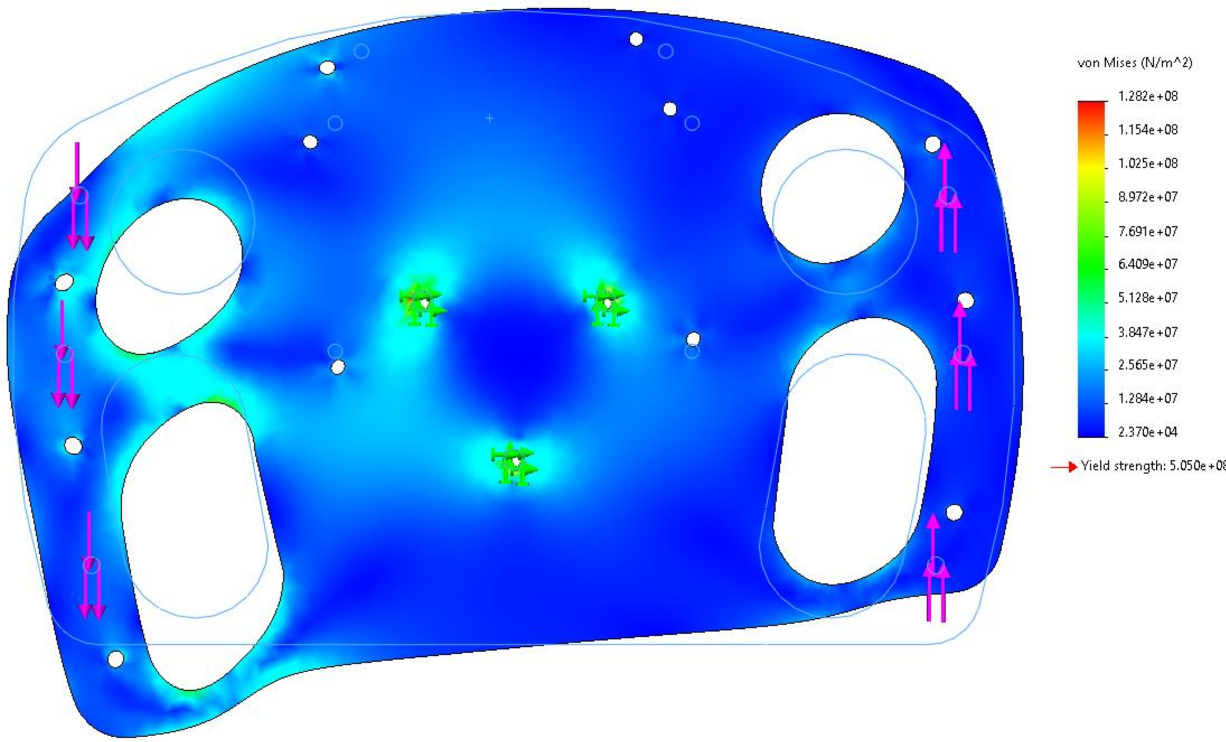
120 lb · 0.4 ft = 48 ft-lb · 2 = 96 ft-lb

Since the diameter of the steering wheel is 9.5 I did the quick math to the left to figure out that 120 lbf would be needed on either side of the wheel to create the moment of 96 ft-lb. I then used this in the FEA simulation to determine the adequacy of my design.

First I had to determine a material for the wheel. Normally, I would have done this as one of the first steps, but I did it a little differently this time, determining the geometry first, then selecting a material later. I did however, have a few material choices in the back of my mind when designing of course. Either a strong aluminum or a fairly strong steel. I decided that I'd test a few materials until I found one that would fit the bill so to speak, so I used 7075-T6 series aluminum to start with. As it turned out, this was a good choice, as the material combined with the wheel geometry is more than adequate. Sure, I could test other materials and be more diligent in my selection, but I'm running out of time. Something to improve upon next time.

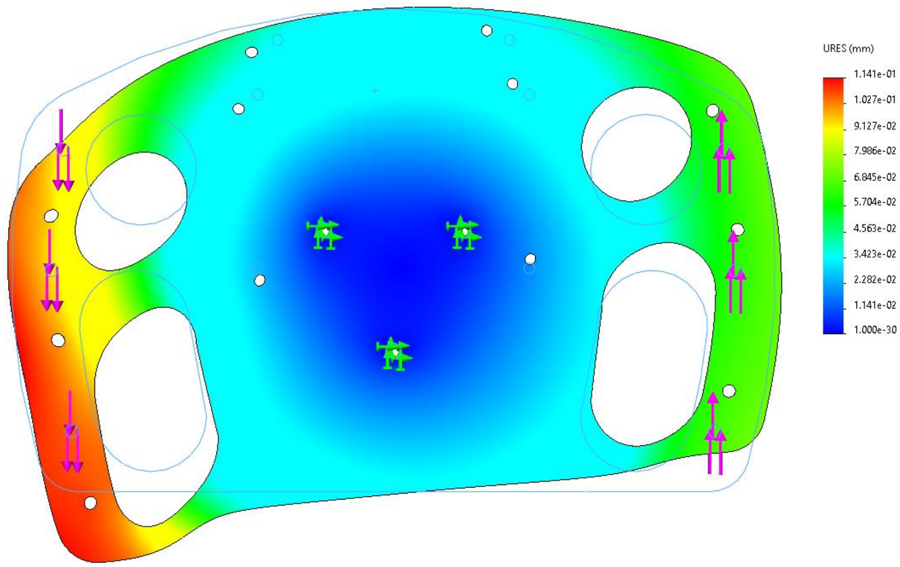
After determining the material, I then used three fixed geometry anchors at the center holes for the quick release. This will allow me to see if the wheel can withstand the forces when the wheel is turned fully on it's lock and it can no longer turn. I then applied the 120 lbf force on each side equally on each of the bolt holes on either side. I applied the force on either end opposite to each other to create the moment arm, instead of a bending moment. The reason I applied the forces to the bolt holes is because that is where the vast majority of the force will be transmitted, not as a distributed force across the wheel surface. The driver's hands will apply the force to the grips, which will then transfer the force to the bolts holding it on, then into the wheel through the bolt holes. Hope that makes sense.

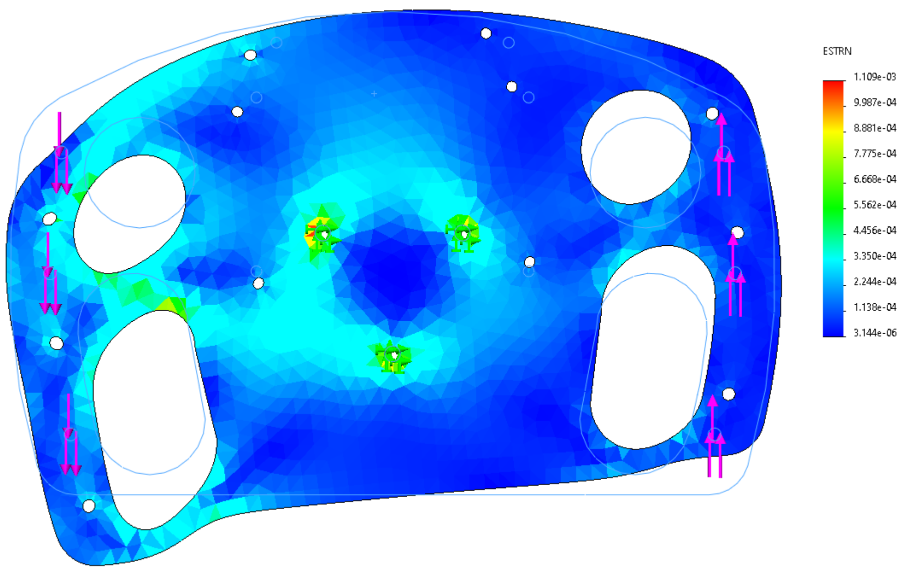
Here is the result of the simulation:



As you can see from above, the highest stresses the wheel experiences is nowhere near the yield strength of the material, meaning this is a good and safe design. In fact, it might be a bit stronger (read: heavier) than it needs to be, but that is fine. I know this wheel won't be perfect, but it is a good start. Also, like it mention's in Steven Fox's article, never underestimate the strength of a scared driver, and the wheel should be one of the last thing in the car to break. Lightening can be done when there is more time.

Here is the displacement and strain from the simulation as well:





I overlaid the results with the original outline of the wheel for reference. It is important however, to remember that Solidworks over exaggerates the displacement of the material for visualization purposes. The wheel will not deform this much even under the 96 ft-lb torque. What I find interesting about the simulation results is that the wheel will deform on one side much more than the other will. The left side in this case is displaced much more than the right side. While this makes sense since the bottom side of the wheel has less material, so when the force in normal to that direction it will have less material to resist deformation, it was never the less something I wasn't expecting. This is why using simulation is very important, to catch things like this. It isn't a big deal in this case, but it is interesting to note.

Note that I have not done FEA on the grips or on the display riser. In the latter case, it isn't necessary. In the former, I can't think of a good way to accurately analyze the forces going through the grips, so anything I do is likely going to be misleading. I think they will be fine however, so long as they are made from a very strong plastic like ABS and printed will 100% infill. If not I'll find out later…

Steering Wheel project will continue on page 78.

1/23/21

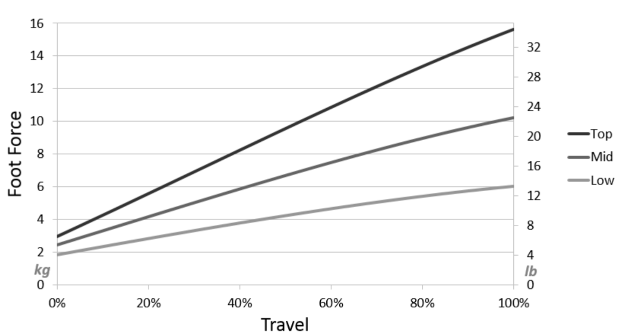
Throttle System (Continued from page 58):

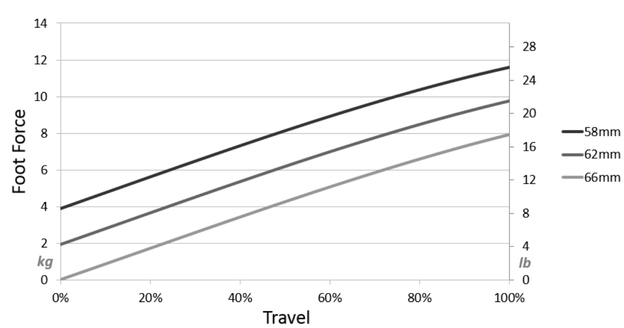
Picking back up on the throttle which has been on the back burner for a while, the next challenge I'm facing after designing the basics of the cable puller is the spring return mechanism. This doesn't sound hard to do on paper, but I am faced with a few difficulties in designing it. Chief among which is how strong the return spring should be. In other words, what spring rate should I choose for the pedal? Too light and the throttle will be touchy. Too stiff and the car will be too uncomfortable to drive. As with most things in engineering, I have to find the Goldilocks zone for everyone to be happy. The problem with this though is what is too light and what is too stiff? What is even comfortable? Most of us have experience using something with a spring, and most of us have as good idea of what would be comfortable pressure to apply to a throttle pedal. But since springs usually don't have their spring rate written down anywhere, how are we to relate what we know would feel comfortable with a numeric value for spring constant? Put more simply, what value or number of spring rate is comfortable? How stiff is a 15 N/m in the real world, for example? It's like asking someone how much something weighs in kilograms when they have never even heard of the metric system. We know how it *feels*, but what *number* is associated with that? This is the puzzle I'm trying to solve. And I can't simply move past it to something else, because a spring's size is usually somewhat dependent on the spring rate, meaning I can't finish any of the return mechanism geometry until I know how big the spring will be.

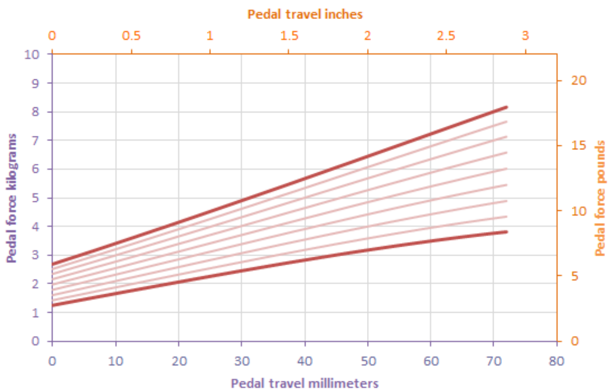
1/25/21

I've had a chance to do some research on spring rates for throttle pedals. There is someone online who make his own sim pedal set (https://chippedwood.wordpress.com/2015/09/03/diy-hydraulic-load-cell-sim-racing-pedals/) and used a Century C838 spring for the throttle, which he says provides 2.45 N/mm spring rate stiffness, which converts to 14 lbf/in. According to Ajax Springs (retailer) the overall length is 3.5 in with an OD of 0.75", which should probably work for my needs if I can't find another.

Another place I looked was Heusinkveld Engineering. They make very high end sim pedals for pro sim racers and Formula One drivers alike. They have two slightly different models for their throttle pedal, and published the following force curves for their pedals:







The two black and white graphs are for the *Sprint* model pedal, and the colorized one on the left is the *Ultimate* model. The left black and white graph is position versus force, and the other one gives data on the spring preload for the Sprint pedal. The colorized one is also force vs. position.

Looking at the graphs and analyzing, we can see the inputs range from initial inputs of 1-3 lbs to a full travel input of 8-35 lbs. The upper numbers do seem a bit stiff, but that's the range given. Bear in mind when I say "in puts" I do not mean how much force the driver must use to actuate the pedal, I mean the force applied to the spring through leverage via the pedal arm. The actual amount the driver will have to push will be lower because of the mechanical advantage.

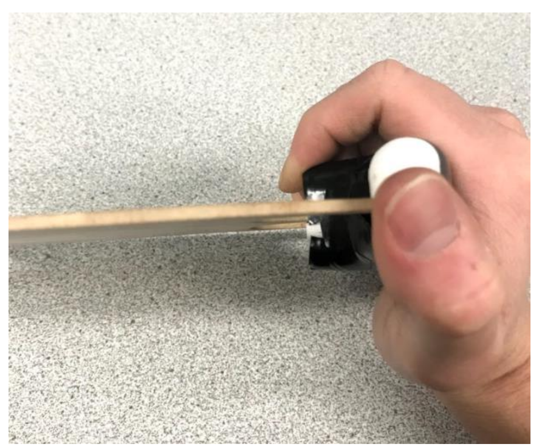
Throttle System will continue on page 78.

1/29/21

Steering Wheel (Continued from page 76):

This should be the last update from the steering wheel until manufacturing. Felipe brought the new 3D printed grips in today, and I had a change to install those heated nut inserts into the plastic grips. This wasn't very hard to do. Just a matter of heating up the brass with a soldering iron and pressing them into place. Then I was able to sand down the grips and put them on the wheel. I put the new ones on the left side and the old ones I kept on the right for comparison sake. After having a change to hold the prototype wheel and compare the grips, I can immediately tell the chamfer I put on the back helped immensely. It's now a lot more comfortable to wrap your finger around and hold. I passed it around a bit, and everyone seemed to agree that it helps out. So this design is good, and I shouldn't need to do anything different to it other than put the channels for the wires coming out the back of the data display and shift lights.

New



Old

I am actually fairly surprised at how small of a change it was versus how much it helped. That's ergonomics for you. Like I said, until I add the channels for the wires into the riser, and until manufacturing, this is the last of the steering wheel project.

Steering Wheel project will continue on page 79.

Throttle System (Continued from page 78):

Time to begin again on the throttle system. I'd really like to have this done and out of the way soon. Seems like I've been saying it will be done soon for a while now. I haven't been able to work on it for a few day because of the design review we had tonight. But now I can try to restart it in earnest, because school will start again in just a couple of days, and I'll soon be busy.

After looking at the Heusinkveld, as well as that homemade pedal set, it seems like a spring constant of between 12-15 lbf/in will suffice. If I were to mount the center of the spring 2.5 inches from the centerline of the pedal pivot, then it would yield a pedal ration of 3.6:1. At a point of that distance, the pedal arm would travel a horizontal length of 1.1 inches. If a spring of 14.5 lbf/in were used in that instance, than the force needed to move the 1.1 inches would be 13.18 lbf. Dividing that by the pedal ratio of 3.6, that would mean the driver would input (at maximum, but without spring preload) a grand total of 3.66 lbf. Definitely on the lower end, but I think in this case, I'd rather have a softer pedal than one too firm. If necessary I could try to improve the design for later years if I find it is too light.

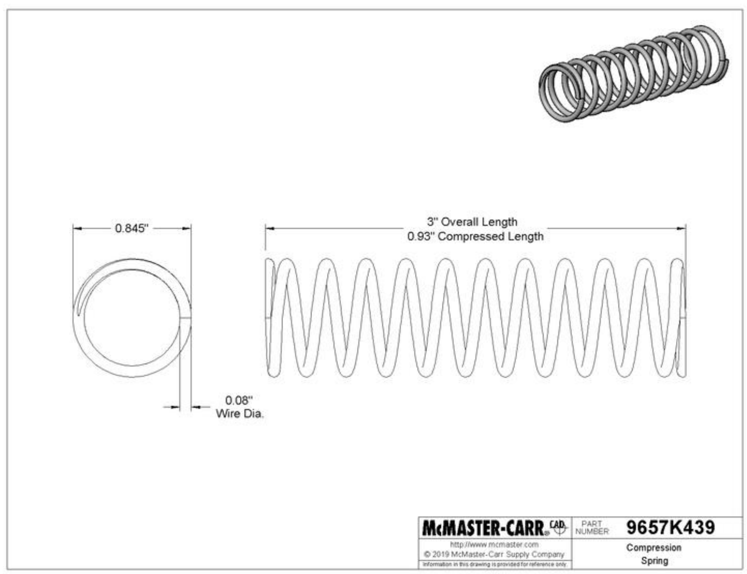
I used the 14.5 lfb/in as an example, because I've found a spring on McMaster that could work for my application that has that specific spring rate:

Packs of 6 In stock
$9.68 per pack of 6
9657K439

ADD TO ORDER

CAD 3-D Solidworks Download

Spring Type	Compression
System of Measurement	Inch
Length	3"
OD	0.845"
ID	0.685"
Wire Diameter	0.08"
Compressed Length @ Maximum Load	0.93"
Maximum Load	30 lbs.
Rate	14.5 lbs./in.
Material	Zinc-Plated Music-Wire Steel
End Type	Closed and Ground
Rate Tolerance	Not Rated
OD Tolerance	Not Rated
RoHS	RoHS 3 (2015/863/EU) Compliant
REACH	REACH (EC 1907/2006) (06/25/2020, 209 SVHC) Compliant
DFARS	Specialty Metals COTS-Exempt
Country of Origin	Germany, India, Mexico, Peoples Republic of China, United Kingdom, or United States



Throttle System will continue on page 81.

2/2/21

Steering Wheel (Continued from page 78):

Quick update on the wheel, I decided to get some rubberized spray paint from O'Rielly's to test them out to see if we could rubber coat the grips. I got Dupli-Color Rubberized Undercoat and Dupli-

Color Custom Wrap (plasti-dip), both in black. After cleaning up the old iteration grips with some sandpaper and soapy water (or soapy wooder if your name is Chris Fix). I dried them off and sprayed

one side with the coatings, the front with the undercoat, the back with the plasti-dip.

At first, the undercoat (right picture) looked like the better option. It sprayed on thickly and evenly, and gave a nice rubbery looking texture (the photos were taken soon after they were sprayed). The only imperfection came from when I moved it and touched it to see if it was dry yet. Whereas the plasti-dip looked thin and like it would tear off easily. I left them both to dry for about 48 hours.

When I came back to pick them both up and see the results, I thought the undercoat still looked good. Until I handled it.

Even after 48 hours of dry time it hasn't adhered to the surface and big parts of it came away at low pressure. It left a big stain on my fingers as well, which I needed to wash to get off. Seeing as we will be wearing gloves when using these, having it flake off and staining the gloves is not a good option. I was honestly disappointed at the results. They were pretty crap. Perhaps I needed to seal the plastic with a primer, or use adhesion promoter. Perhaps because the humidity was high because it was raining outside.

The off brand plasti-dip did much better:

While it went on much thinner (this was after two coats) it held on much better and didn't flake or rub off even with from abrasion applied by my hand to see if it would wear off. I'll end up using this stuff then. It's a bit more expensive than the undercoat ($16.93 list price vs $9.31, we paid less), but it is the better product. It gives good grip to the surface of the parts and it comes in many different colors, so we could change it up if we wanted to. When I do this to the real grips, I'll use some kind of sandable primer so I can remove the striations in the plastic and to seal it, then spray several coats over the surface of the parts. If I spray light even coats, it should come out very smooth.

Felipe is also trying out printing some TPU on top of the ABS grips, but I have some doubts as to how well that will work out. If nothing else the plasti-dip is a good fall back. In either case it's better than the electrical tape grips or sports tape or any of the such like. Although batter's tape would be good if literally all else fails.

All for the this update. I'll get back to the throttle.

Steering Wheel project will continue on page 94.

3/4/21

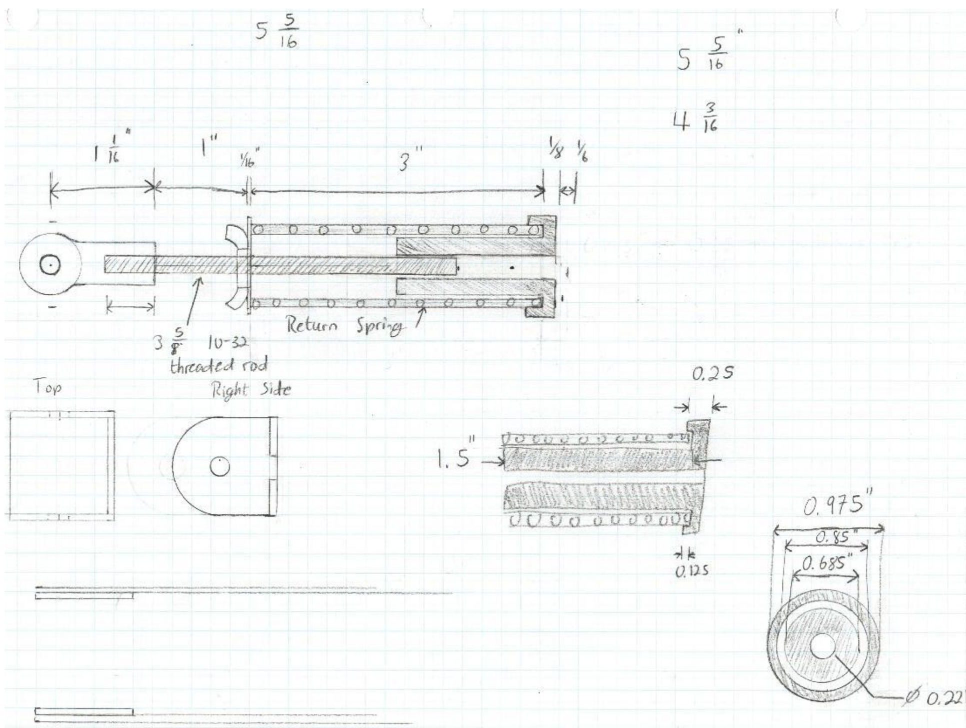
Throttle System (Continued from page 79):

Ok, so I've got a lot to update on the throttle. I've spent the past week or so working on documentation for the Design portion of the FSAE online competition's Knowledge Event. Since I'm currently the chief it fell to me to ensure the documentation was completed and sent in. I had to make and format a ton of slides and help reformate others, so that took almost all of last weeks' time. I did however finish designing the throttle pedal, pedal rails, and the front closeout floor panel. I'll go over their design in more detail below:

The last thing I had done on the throttle was a little over a month ago when I had tried to figure out what spring constant to use in my design. I ended up choosing the 14.5 lbf/in spring from McMaster. 14.5 lbf/in is approximately the right spring constant to use for this application –based off of the Heusinkvelt data , and the physical dimensions of the spring itself will fit nicely into the design idea I had. Speaking of Heusinkvelt, I ended up basing a lot of my design off of their sim throttle pedal. It's one of the best designs in the sim pedal industry, and I knew it would be fairly easy to incorporate my cable puller design into it.

What I like about this pedal design in particular is the adjustable throttle spring I mentioned before. By tightening the ribbed thumb nut, it's possible to "pre-load" the spring and make the motion threshold higher, without changing out the spring or changing the pedal geometry. Basically, it means that the pedal will be harder to push when the nut is tightened down. The more it's tightened, the more initial force will be required to move the pedal. I basically copied that in my design, with a few revisions. This will make the pedal more versatile to use and easy to adjust for multiple driver's preferences. For example, I prefer a stiffer pedal as it allows me to modulate it more precisely, but another driver might want to have it easier to press. With this set-up a few twists of the nut, and the change is made.

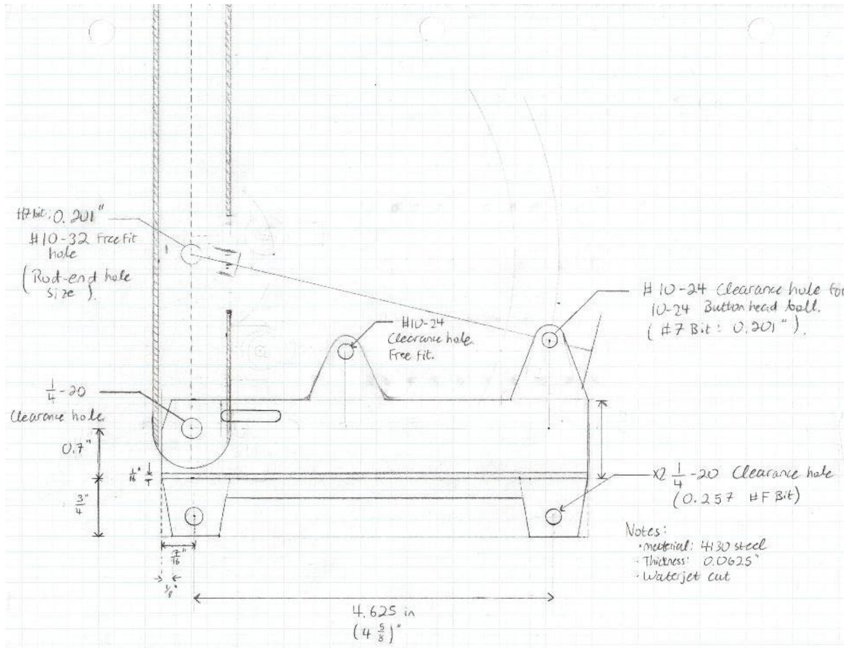
There were a few other things I wanted to incorporate into the pedal design. I wanted it to match the brake pedal in terms of adjustability of the heel and toe cup, and in the pedal rail. The two also needed to be the same in height. Having two different sized pedals would be confusing and awkward to use in the car. Essentially they needed to be twins. Since the brake pedal had already been designed, this portion of the throttle design was basically already done. The cable puller geometry had already been done too, so at that point it was a matter of arranging all the components, giving them a proper layout and designing the pedal base that everything would attach to. Like usual, I did this portion of design on graph paper using a standard ruler, engineer's scale, a compass, protractor, and several sized shape templates. While this isn't exactly "drafting" like they teach in high school, many of the principles are the same. And yes, this is a fairly old fashioned approach to design, but I honestly feel it's the best way to do it. By making a physical sketch on paper, you can easily draw where components will be and measure distances and angles that could prove tricky in Solidworks. Changes can easily be made



as well, then when the design is done, it is very easy to transfer the design to Solidworks, which is what I did with the components for this pedal:

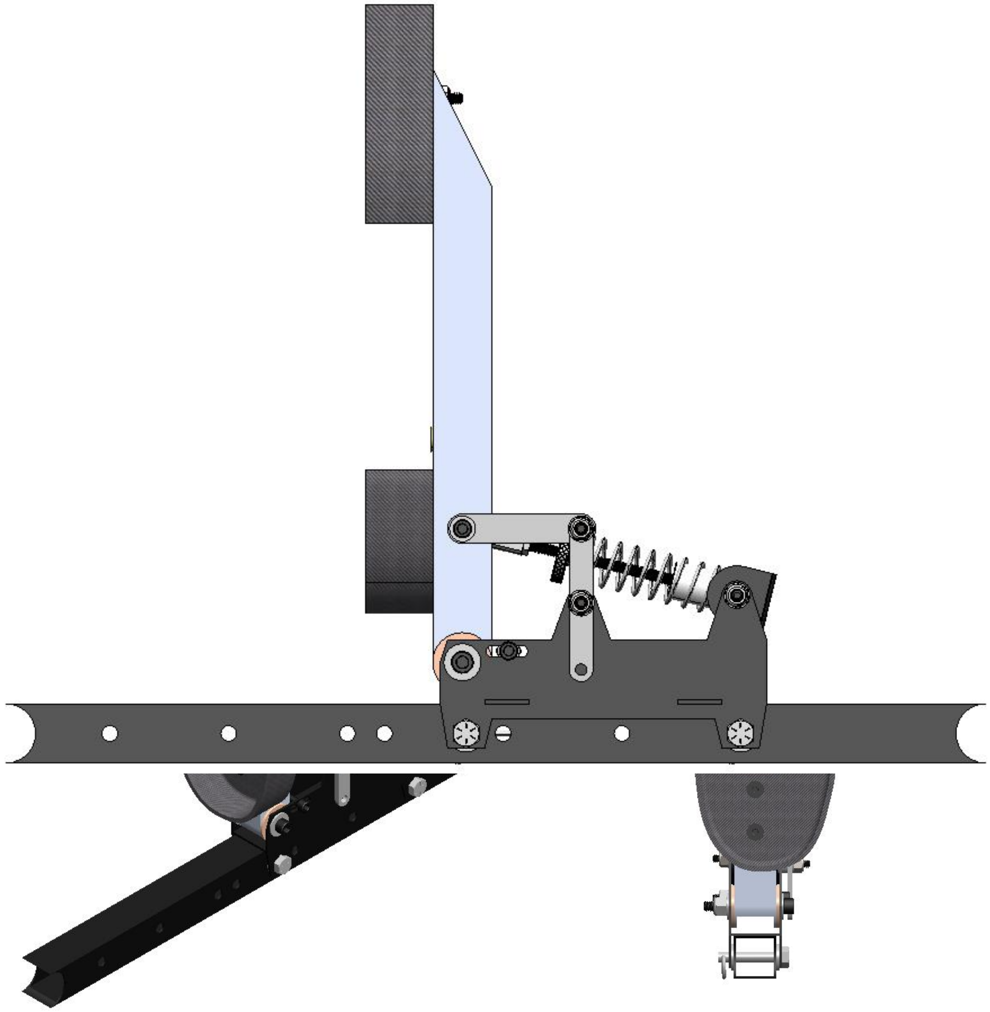
Above is the rough layout sketch for the pedal return spring with the rod end, the guide rod that helps keep the spring from buckling, and the sleeve for both the guide rod and return spring, that will both hold them in place, as a few other components. Since the rod will slide in and out of this sleeve, and because of its shape, I'm calling it the butter churn because I have a penchant for stupid names (also it kinda describes what it does pretty well).

As you can see from the sketch, unlike the Heusinkvelt design, the guide rod that I designed doesn't go all the way to the end of the so called butter churn. That is because otherwise, it would poke out the other of the churn and that could contact something I don't want it to, so I cut it off short. Once I had drawn these things as they would be assembled, I was able to measure the length of the whole thing, which then let me design the layout for the pedal base:

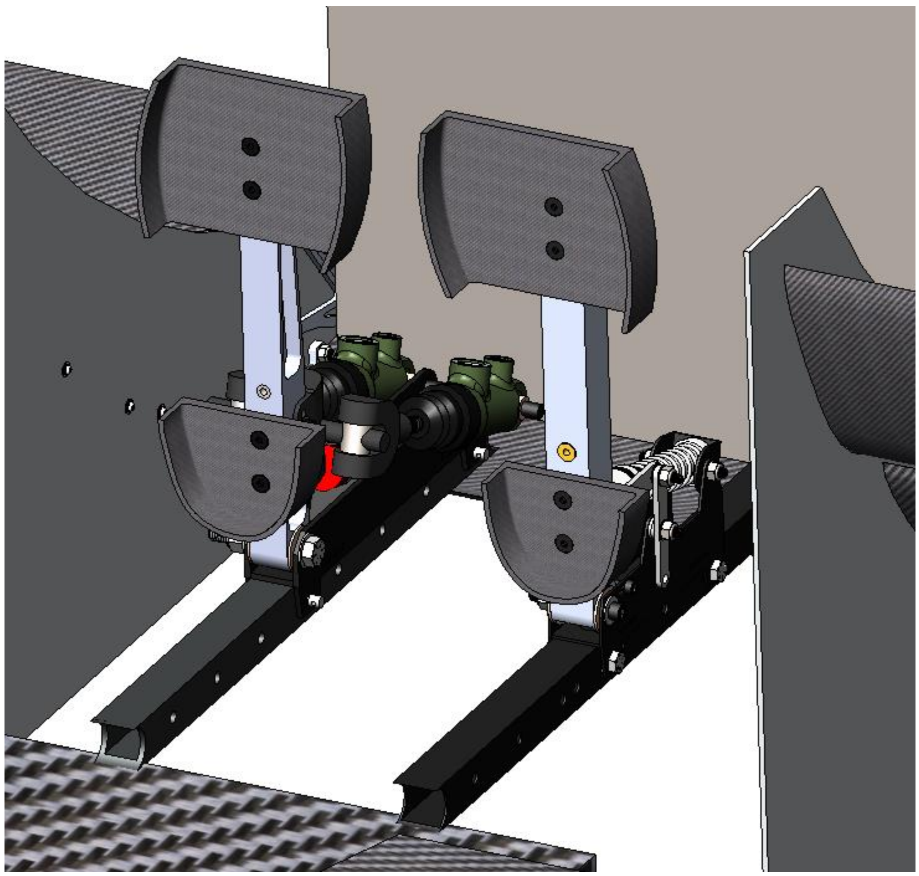


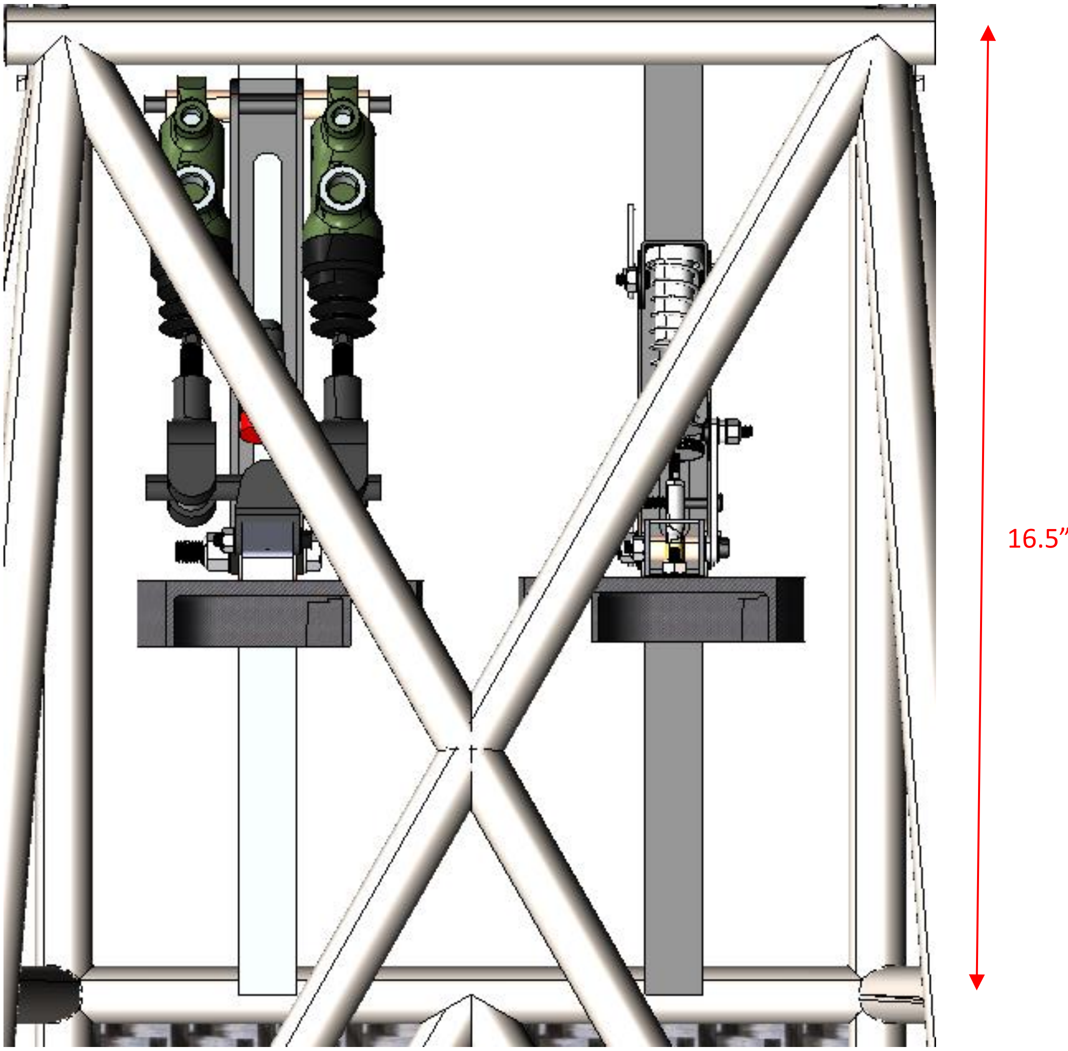
The sketch above has most of everything in it, the pedal body, the base with all the mounting holes and geometry requirements. The return spring geometry is represented roughly by the long line extending from the 10-32 hole on the pedal body to the 10-24 hole on the top far right hole. The placement of the 10-24 hole was found by using a compass to sweep an arc with the radius equal to the length of the spring assembly (5 5/16”) with the center line at the 10-32 hole. You can faintly see the lines left over from this. I then used a ruler to pick a spot on that arc that would coincide with a standard horizontal distance (not a crazy decimal like 4.763) from the pedal pivot hole (it came out to be 4.625 as you can see from the picture). Once I had all the holes I needed placed, I was able to draw the rest of

the pedal
body and
figure out their

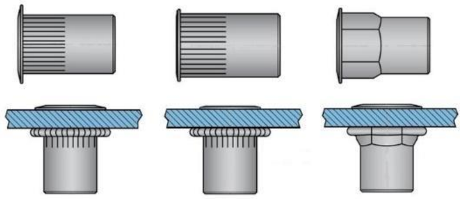


dimensions as well. Once this was completed I could design all the parts in Solidworks and assemble them together. And this is the result:



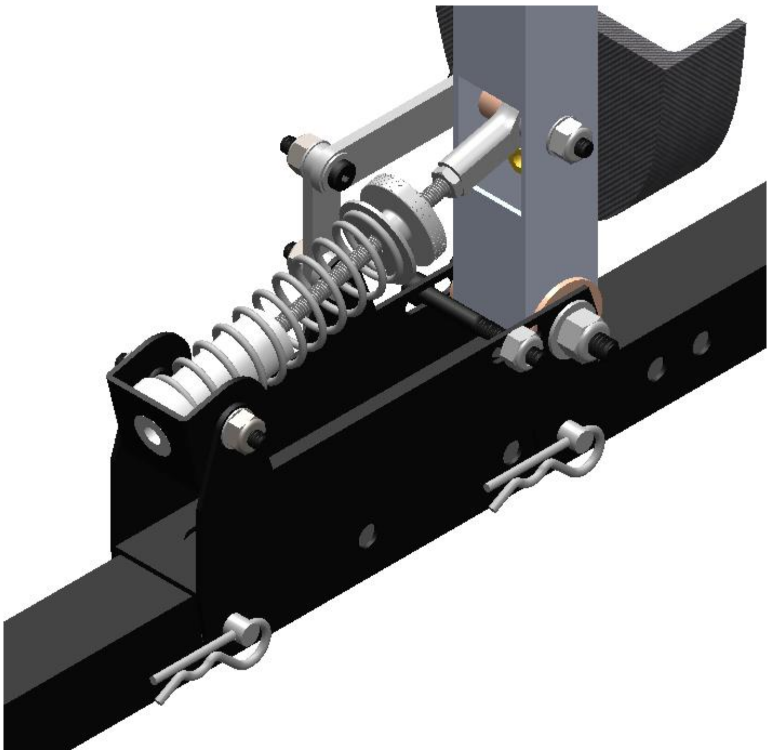


Overall, the design came out quite well. In total, the max length of the pedal is ~ 7in, several inches short than the brake pedal, meaning fitment isn’t much of an issue. Like the brake pedal, the pedal body is 11” tall from the pedal pivot the top of the toe cup. Originally the brake and throttle pedal had 4” of forward and backward (longitudinal) adjustment in the car, but when I put them both in the master assembly, I had realized I had been slightly off in my measurement (perhaps the chassis dimension changes and I didn’t remember it?), so length of the pedal rails came out to be 16.5” instead of just 16”. This allowed me to reconfigure the pedal rails, so that now there is 6” of adjustability. In this case, more is better, so that’s a good thing to gain. A few of the special features of this throttle pedal are the attachment hardware for the heel cups. While the toe cup uses a conventional flathead bolt and locknut combination, the heel cup screws into rivserts nuts instead. I decided to use these instead of regular nuts because I want the heel cups to be able to be moved quickly and easily. Since these rivnuts are threaded, the bolts don’t need an additional nut on the end, meaning they can be taken off quite easily using nothing more than an Allen wrench. This is similar to how the heel cups bolt onto the brake pedal, but on the brake pedal they are press fit inserts. Here’s what a rivnut looks like for illustration:



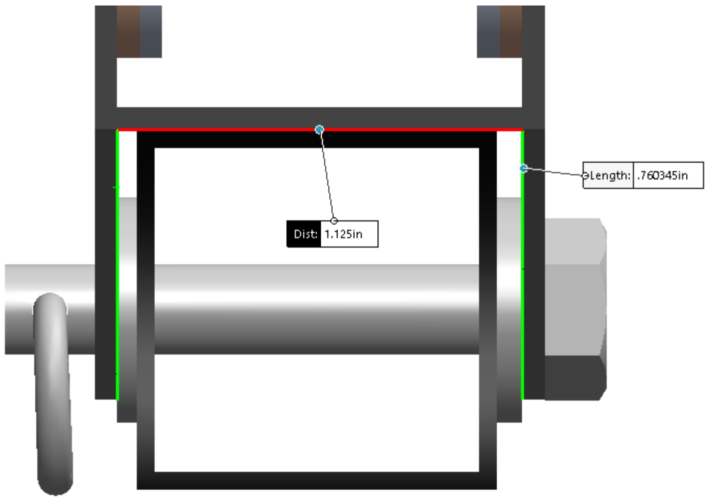
When installed on a flat surface, the bottom portion crushes, which seals it in place.

There are a few other key points I'd like to call attention to:



The back of the pedal body has a square hole cut out of it. The reason for this is it was an easy way to find an attachment point for the bolt that goes through it. Had the back remained flat and uncut, I'd have to make some sort of tab to attached the bolt and nut to. Not only would that be a pain in the butt, but it also makes the pedal longer by about an inch and a half. We want to try and save as much space in the cockpit as we can, so this turned out to be the best solution. Another thing to point out is the preload adjustor on the return spring assembly. It its comprised of an oversized #10 washer, and a special knurled 10-32 nut from McMaster. It's sized so that it can easily be adjusted by hand. The butter churn I mentioned earlier can be seen here too. I'll have it made out of 3D printed ABS. I'm not too worried about strength here, since it will always be under tension, which 3D prints handle better than tension. It will be held in place in the pivot basket by having a small raised key that fits into a hole designed in the basket. The pivot basket is there to allow the spring assembly to change angle, since it will have to slightly as the pedal body travels through it's arc. The pivot basket is held in by two ultralow profile head 10-24 bolts and nylock nuts. Looking at the pedal base where it meets the pedal rail, we can see the same pin and cotter pin arrangement as the brake pedal. I'll use the same hardware to save on the amount of random parts we need to order. They're grade 8 hardware, which is definitely over kill on a throttle, but like I said, we'll be ordering the parts for the brake pedal anyway, so we may as well keep them the same.

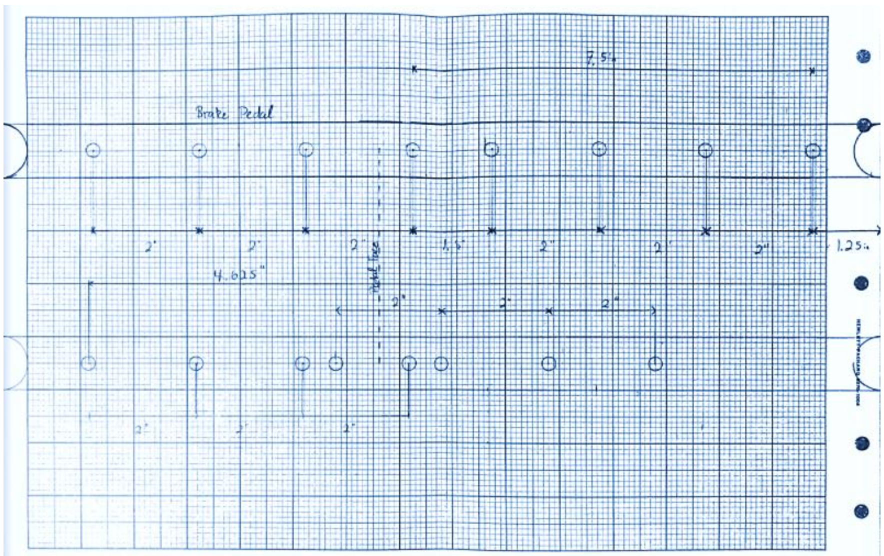
Another thing to note is about the space between the pedal base and the rail. On both the throttle and the brake pedal, the rail is a 1"x1"x16.5" steel square tube. I decided to use steel for both, even though I could definitely make it lighter with aluminum. Again, it's about not having to over complicate the design more than I already have and to not have to order a ton of materials. It's heavier, but again, this is a base car with lots of room for improvement. But back to the sizing. The 1"1" OD of the tube was dictated when I designed the brake pedal. When I designed the throttle pedal however, I had to make it a bit wider than the brake pedal. The throttle pedal body is 1” wide, as opposed to the brake pedal which is 0.8825” (to match up with the bias bar). All in all, there is a slight gap between the throttle pedal base and the rail, since I can’t order 1.25”x1.25” steel bar. So, to fill this gap I’m using washers I’’ either weld or bond on. Probably bond because it will be easier. Again, strength won’t be an issue since they are just preventing lateral movement. The forces will all be compressive, which means



that the force won’t be transferred through the washers at all.

While I’m on the subject of the pedal rail, now might be a good time for a quick overview on how I designed the rail for the pedals. I won’t go over most of it since it is fairly self-explanatory, but the placement of the holes is important. Again, the driver dimension was the length of the brake pedal. I had designed the rail so that in the rear most position (pedals closest to the front bulkhead) the end of the master cylinders would have a small gap to the bulkhead for clearance and fitment purposes. I also wanted two inch increments of adjustability. So once again, I made a hand drawing using graphing paper, and placed the holes where they needed to be. Later, when I did the layout for the throttle, I knew I wanted to the front of the throttle to always be lined up with the brake pedal, so some quick measuring and math later I had it all laid out. Then I simply transferred it to Solidworks.

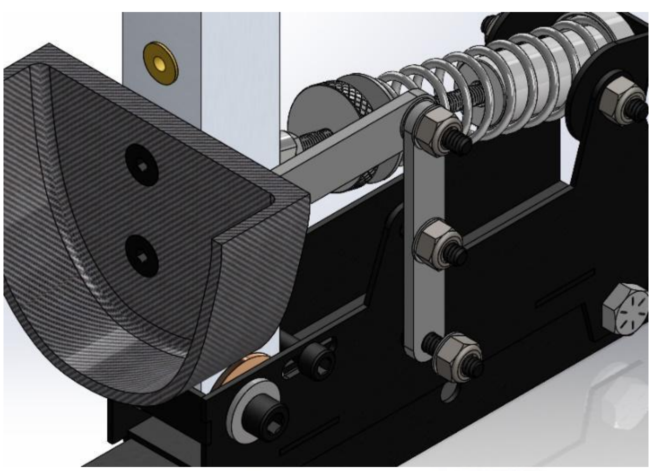
Here's the layout sketch. It's not very clear because I had to scan it with my phone since it is a 1:1 scale, and wouldn't fit in the scanner. But I think it shows it clearly enough.



The last thing to mention is on the throttle cable, and cable puller assembly.

4/2/21

Ok, long break for school and the cost report over, let's get back to the cable puller assembly. As I have already shown in previous sections, the cable puller uses two linkages to create a mechanism that pulls the throttle cable back. I've been having some trouble coming up with a way to connect the actual cable to the puller, but I believe I have a good solution.



At the bottom of the puller I have put a nut and bolt, but intentionally didn't model the nut tightened down all the way. I am planning on using this as an attachment point for an eye-end cable crimp:

This will crimp go over the steel throttle cable, and the eye at the end will go onto the bolt at the bottom of the cable puller. From there it can be adjusted as necessary. This will help with the adjustability issues that have plagued us in the past.

Here are a few other examples of the cable end that could work.

Throttle Pedal project will continue on page 97.

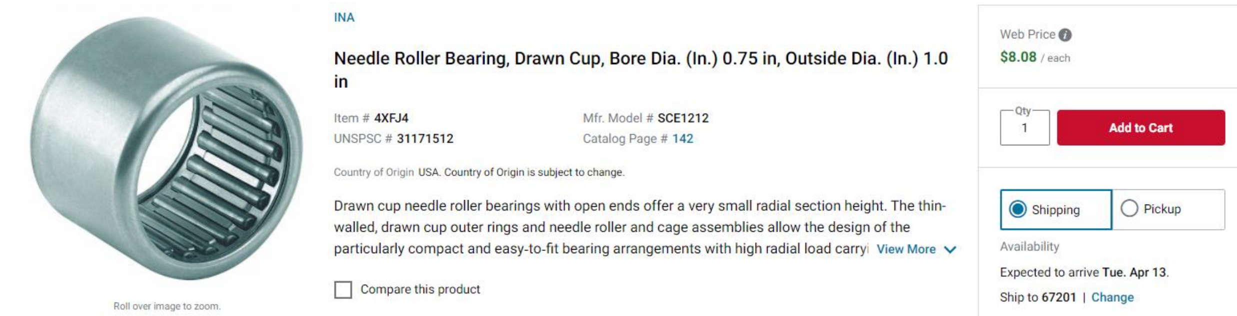
4/8/21

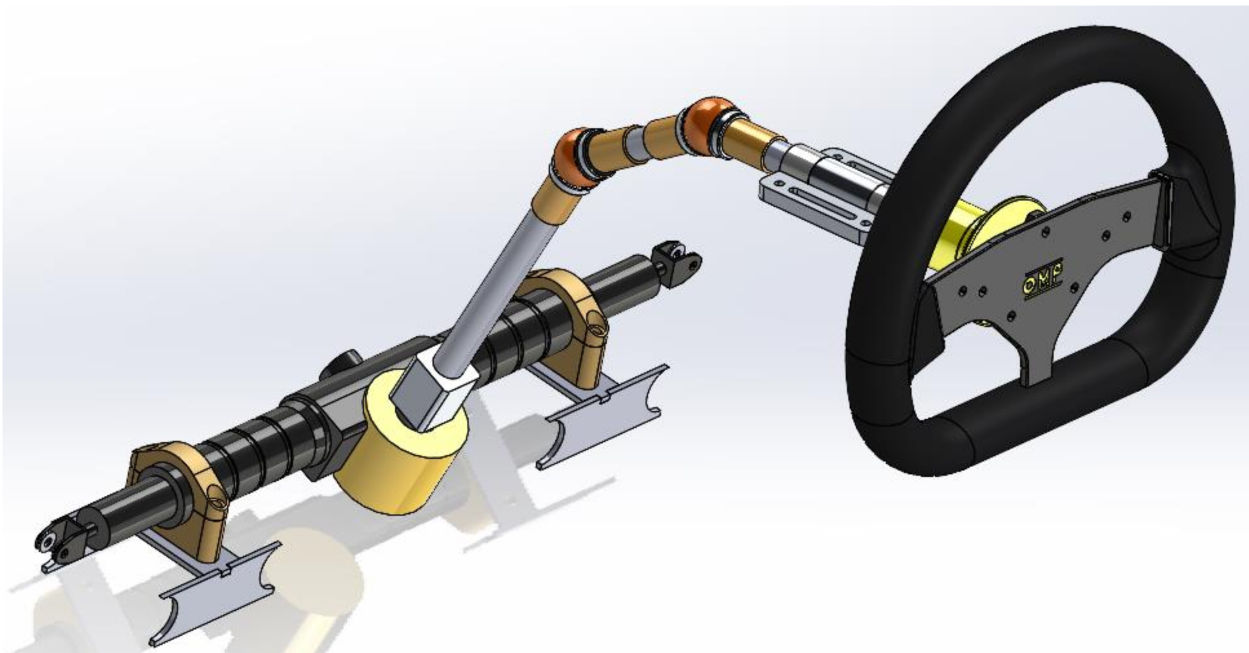
Steering Column (Continued from page 72):

I'll now be continuing onto the steering column project, as aside from the shifter, the cockpit is basically done after the column is finished. I've been looking for a splined coupler to connect the steering rack to the column, but unfortunately, the specific one we have used in the past is no longer available. The one we had used for SR19 was a 5/8-36 spline on one side and a 5/8 bore on the other made by a company called Stiletto. They still make other versions, with the 5/8-36 spline, but with a ¾" bore. This means that I'll have to change the design of the other components from previous years to reflect this diameter change, including the bearings and attachment brackets on the chassis.

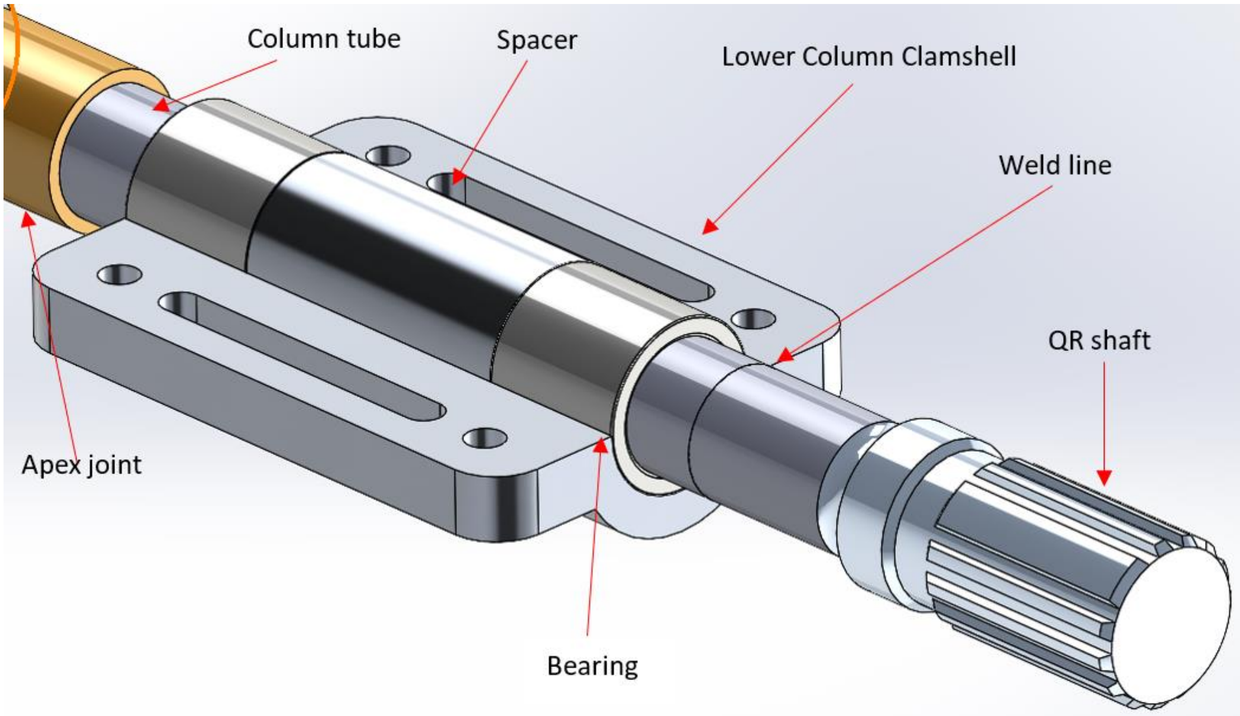
There are a few other designs of coupler out there, but this one seems to be the only one I like and know will fit with the Kaz Tech rack. The pass-through bolt is what helps clamp down on the splines, and the bolt shaft helps hold the column in place.

Going with this size of coupler means I'll need to find bigger bearings as well to fit the larger diameter of the steering column. Luckily, I was able to quickly find a bearing from Grainger that should fit the bill pretty well:



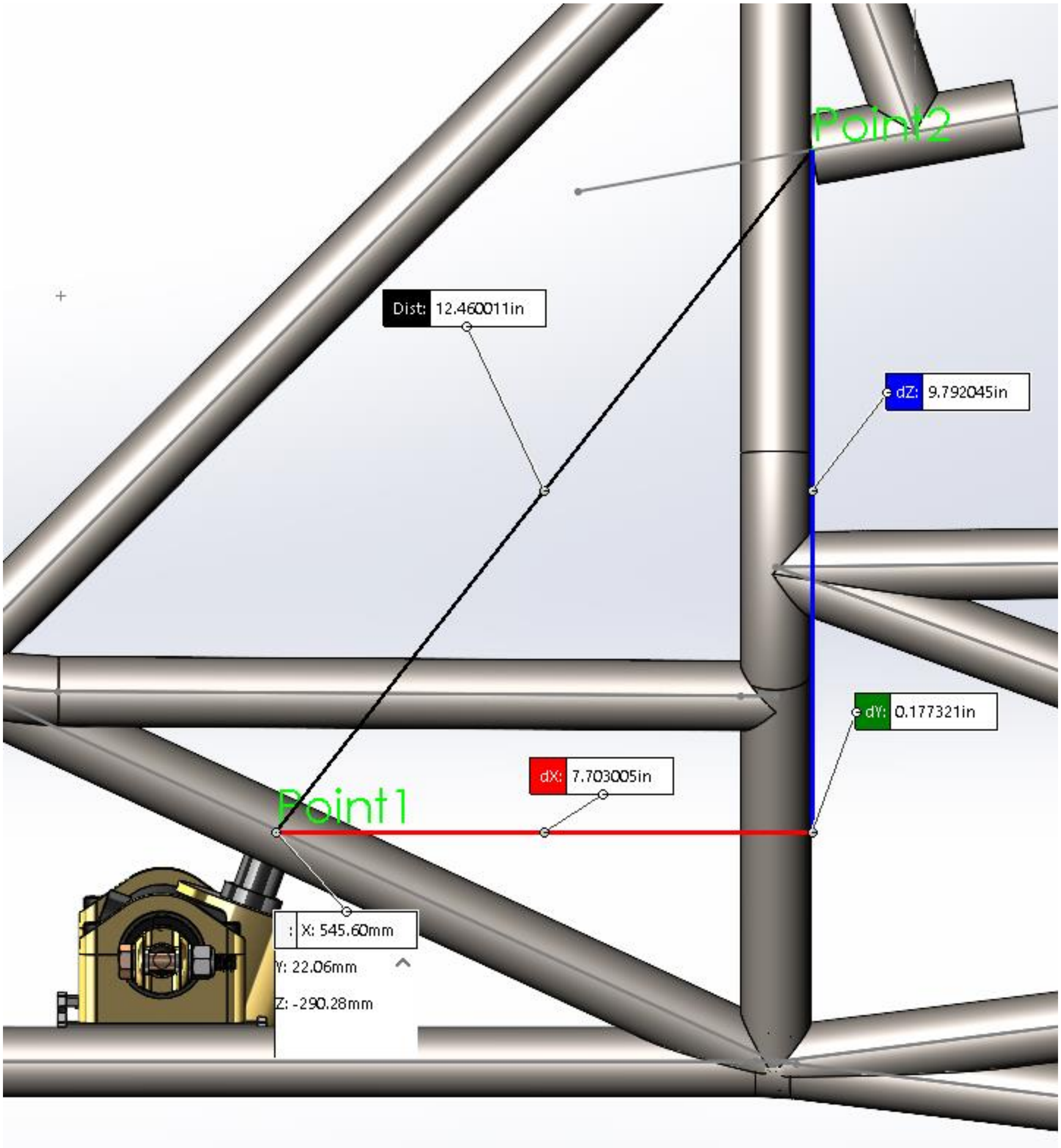


Above is the assembly from SR19 with the wheel, rack and column. On the column is the coupler I was refereeing to (rough model), the column tubes, the Apex joints, the bearings, spacer between the bearings, and the clam shell. Here is a close up image of the bearing and the lower column clamshell. I'll have to redesign the clamshells to fit the larger diameter bearings as well.



What I need to do now is take a bunch of reference measurements of the chassis and steering wheel geometry so I can design where the tubes and the angles need to be. This will be the tricky part. For SR19's steering rack I did a 1:1 scale layout on paper and had to hand draw the angles and everything, then transfer it to the computer. It took a long time and still needed tweaking, but it eventually worked, so I think I'll do this the same way.

The knee hoop is 12" horizontally distant from the bar in front of the steering rack. The center of the top of the knee hoop is ~19" above the center of the same bar. These measurements will help me layout the 1:1 sketch and place everything accurately. The wheel sketch that Alan did is a little larger than the wheel diameter that I designed, and it is a little lower than the top surface of the knee hoop (rules compliance), but that's fine. It's not much and it adds some margin for error and/or adjustment later on.



Steering Column project will continue on page 103.

5/13/21

2021 Competition Review:

I haven't updated the journal for a while due to a variety of factors. Namely preparing for competition, school, work, and other commitments. I'll try to give an update on the current state of affairs, but before that it is necessary to do a debrief and note all the key things that were talked about and brought up during the online competition. The videos from the one-on-one judging are on the google drive under **Shared>FSAE Shared>SR21-FY2020-2021>Comp Presentations>Recordings.** They are recorded on zoom; you can go view them to see for yourself.

First of all, I think the presentations went pretty smoothly without any major hiccups. We didn't spend too long on the main car/full team part of the presentation, mostly just introduced ourselves, the team and explained our current situation to the judges, so they would know where we were coming from in terms of a design standpoint and why we didn't have a running car. Or one that was built at all… All of this took about 10-15 minutes to do.

Next, we went into our breakout rooms through Zoom. (Technically it was Pathable, which is literally just "professional" Zoom, so that part of it was easy. I introduced myself again to the Design Judge, whom turned out to be a senior manufacturing engineer at Blue Origin, which was equal parts terrifying and awesome at the same time. But he seemed like a pretty decent guy, and overall, while I did get grilled, I thought it went well. We disagreed on a few things, but there were definitely some areas he pointed out to me that need more work/improvement, etc. And a small handful of things he considered were wrong, but there weren't too many of those. I'll try to break down each part of my presentation and the discussion I had with him in the parts they were presented in:

- Brakes:

- Throttle:

-

7/30/21

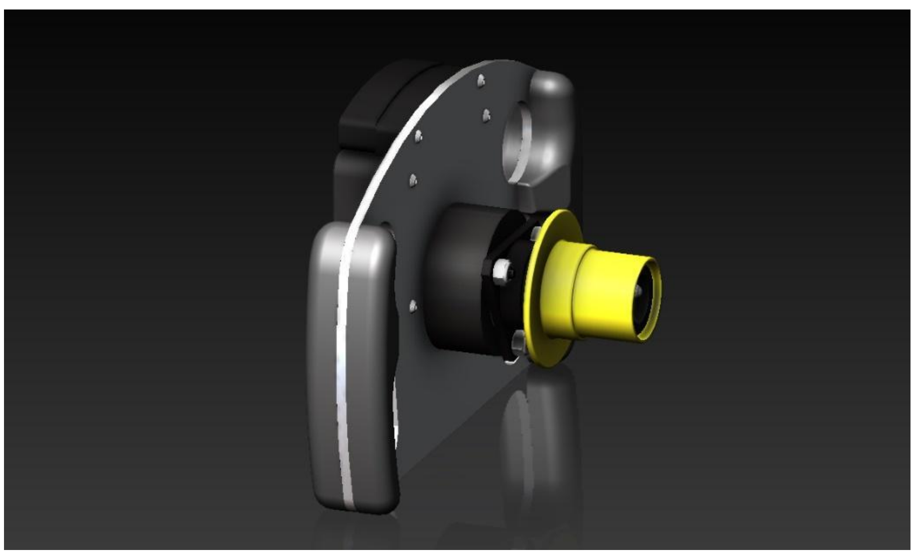
Steering Wheel (Continued from page 81):

I haven't updated the journal in a while, mostly due to work, so I'll try to update the progress I've made so far. For the steering system, several months ago I received steering wheel spacers

from Speedway Motors. They were the "JOES Steering Wheel Spacer, Black Anodized Aluminum":

They come in various lengths to accommodate various sized drivers. For SR19, a lot of drivers complained that they could barely fit their legs into the cockpit and still turn the wheel due to the wheel being so close to the knee hoop of the chassis. To fix this we had planned to add spacers to push the wheel further out, but never got around to it.

Anticipating that issue, I designed the wheel to have the spacer already:

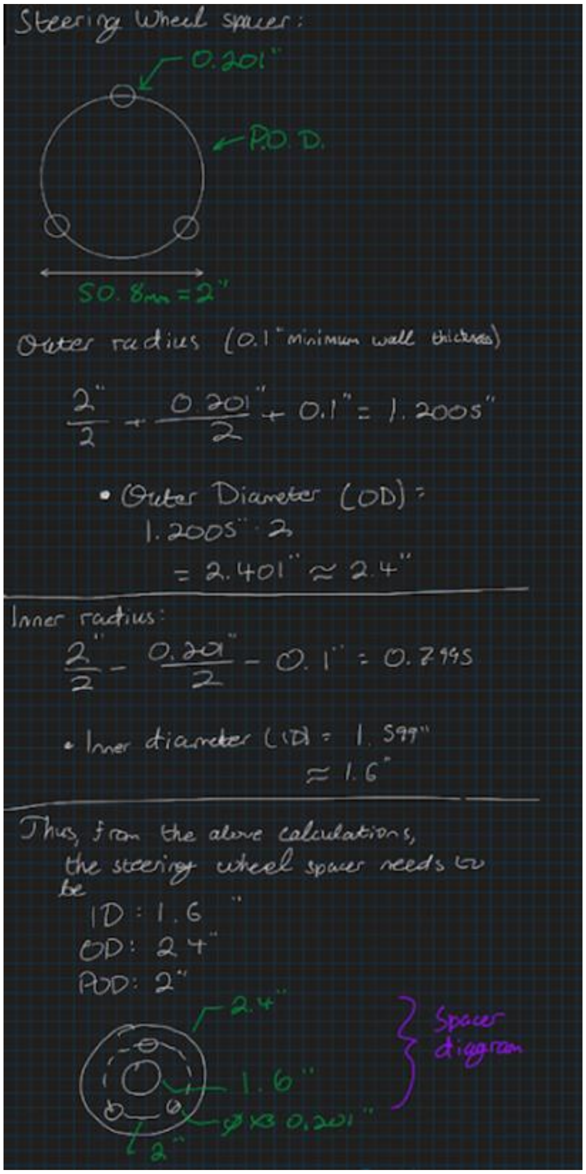


I was going to buy several spacers, then add the 1" one as standard to the car so I compensate for any manufacturing error afterwards. However, when I received them, it turns out they don't fit. They were advertised as universal fit for a 3 hole quick release base. But as it turns out, there is a difference between the European quick releases (SPA, BG, etc) and American quick releases. At least I think. All I know is that the bolt holes didn't line up.

Either way, these motor couplers won't fit. So I needed to get a little more creative with my approach. I decided that I could probably turn the blanks on the lathe, cut the blanks to length, then get a drilling jig to drill out the holes. Therefore, I took another look at the drawing for the SPA technique quick release on page 65 as a reference, and chose a wall thickness as a guess, then figured out the dimensions I would need as drew it up in Solidworks.

I estimated the desired wall thickness by using my calipers and using my engineering sense of intuition to determine that wall thickness to be around 0.1 inches thick. Obviously, this is just a guess, and will need to be checked through FEA later on. The material will be aluminum for lightness and strength, and based off of my previous research the alloy will be either 6061, 6063, or 7071 T-6.

The P.C.D (pitch circle diameter, the diameter of the centers of the bolt holes) is 50.8mm from the drawing, which equals 2 inches. The bolt holes are for #10-24 bolts, so by looking at the chart on Little Machine Shop https://littlemachineshop.com/reference/tapdrill.php, I know that the drill hole dimeter is a #7 bit, or 0.201 for a free fit clearance hole. Therefore, to find both the outer and inner diameters, I took the radius of the P.C.D. (1 inch), and added half the bolt hole diameter (0.1005"), then added the wall thickness. This gace me the outer radius of 1.2005. I multiplied it by 2 to find the

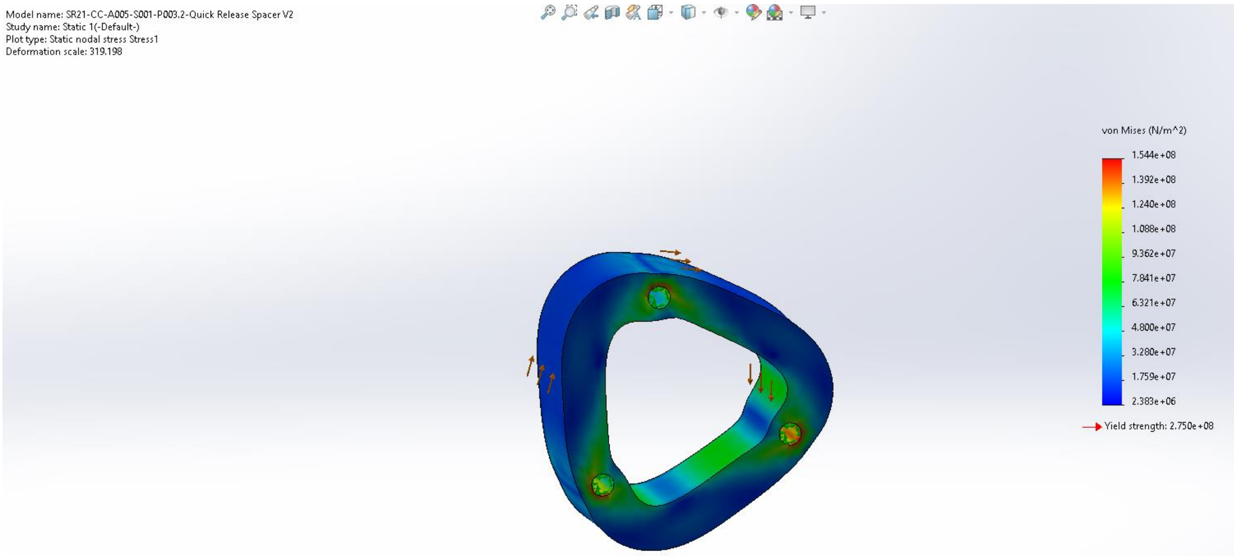


diameter, which was 2.401”, which can be roughly rounded to 2.4”. To find the inner radius, I subtracted instead of added to find the inner diameter of 1.6”.

Using these measurement, I put them into Solidworks:

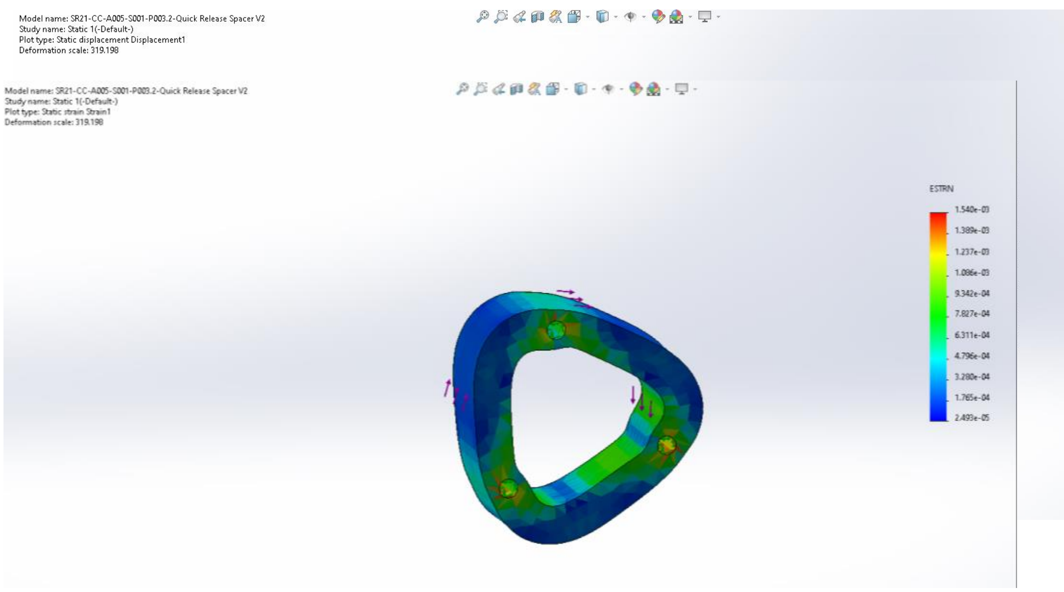
The next thing to do was to test the FEA to ensure this design would remain strong enough to handle the loads being applied to it. Remember that the loads being applied to the steering system were calculated back on page74 to be 96 ft-lb with a factor of safety of 1.3. Now, here is an area that can be approved. I don’t know how to add torques to FEA very well. I ended up added the 96 ft-lb torque to the exterior and interior walls of the spacer, however, this is inaccurate because the forces will be applied through the bolt holes, not the surfaces of the cylinder. I think it can be assumed to be reasonable fine, but better FEA is never a bad thing. It’s something that if I get the time I’d like to go back and revisit, because I’m sure the judges won’t be happy with that kind of FEA.

But, I don’t have much time, so this will have to suffice:

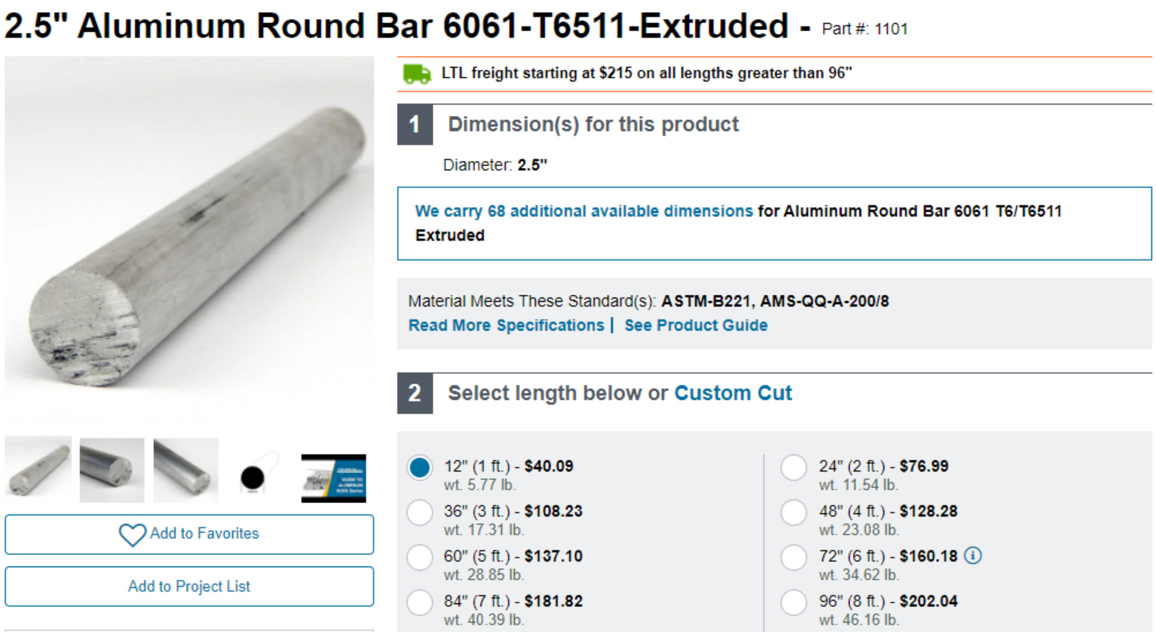


Considering the yield strength is 2.75 Mpa (according to Solidworks. Checking with ASM, it is 2.76 MPa), and the max stress experienced is 1.544 MPa, I think I can be reasonably certain of the safety of this spacer, and can ethically use it on the car.

The other FEA results are shown below and have been explained on previous sections.



I looked on Online metals that they have an aluminum blank that should work for what I need. https://www.onlinemetals.com/en/buy/aluminum/2-5-aluminum-round-bar-6061-t6511-extruded/pid/1101



It is a bit on the pricey side, but I can always use it for more projects in the future, and I will be needing to make more than one spacer anything (at least, that's likely the case). Since our new mini lathe is 7"x12" (7 inch diameter, 12 inch length max size), I can easily turn it on that, which will save some time.

Until I can get that ordered and delivered however, that is it for the steering wheel updates.

Steering Wheel project will continue on page --.

Throttle System, and Brakes System (Continued from pages 79, and 49, Respectively):

This is a bit of an odd section, as it involves both the throttle and brake pedals at the same time, where as before I had kept them separate. The reason for this is because I had an idea for the heel & toe cups, which affects both pedals.

Originally, I had planned to make the cups out of carbon fiber, with a soric core sandwich, like the previous pedals had been made from:

This is *probably* the best way to make them, however since our composites leads keep coming and going like clockwork, and because nobody bothers writing anything down (something everyone needs to work on), our current composites leader is still learning the ropes. Therefore, I think it would be in both her interest and mine to not overload her with work as she's got enough to do. For that reason, I had decided to switch to a 3D printed facia with an aluminum backing for strength. The problem is 3D printed plastic isn't terribly strong due to the nonuniform cooling caused by the layering process of most printers. Essentially, the individual layers put down by the printer cool at a different rate, sometimes causing delamination of the print, and weakens the overall part, unlike if the entire part had cooled at nearly the same rate, like in injection molding. Anyway, I was worried about the strength especially around the areas of would surround the driver's feet, as these would have shear forces put on them by the driver's feet pressing on them during hard cornering. To solve this issue, I took a que from the old powertrain lead Felipe, and decided to make a mold of the 3D printed pedal cups, mold them in silicone, and pour recasts of them in a high strength casting resin. I had seen the process before, but had never considered it for use in the cockpit. Here is a video example of what I am talking about: https://www.youtube.com/watch?v=Pu1RTkWHA3o.

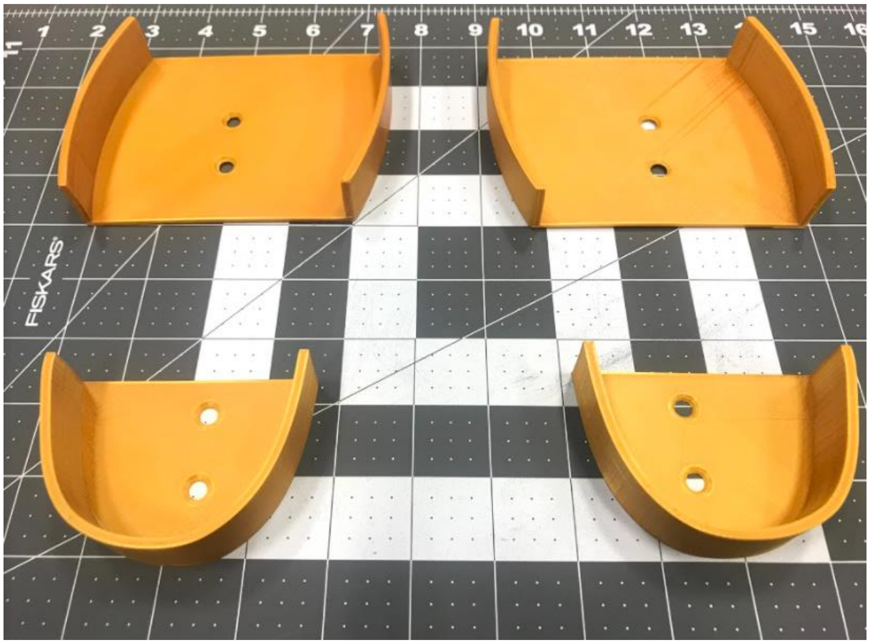
Here, he's making a video about making a replacement light for a car, but the process will be similar. The video is from a company called Smooth On who specialize in casting products. I decided to use their Mold Star 30 casting silicone, as it has a shore hardness of 30A, meaning it's about as hard as a pencil eraser (it ended up being a bit more squishy than that but not by a lot). I also decided to use their epoxAcast 690 casting epoxy, as it has a high compression strength, which is what I'll need for the pedals. They're both available from amazon: https://www.amazon.com/Mold-Star-Silicone-Making-Rubber/dp/B00ETAY8RI/ref=sr_1_1_sspa?dchild=1&keywords=mold+star+30&qid=1627765910&sr=8-1-spons&psc=1&smid=A3VDBPE82S43CG&spLa=ZW5jcnlwdGVkUXVhbGlmaWVyPUE2TVJUTVlOTktSS1QmZW5jcnlwdGVkSWQ9QTA1MDk5ODkzWVBZV1hYWTI3OFAmZW5jcnlwdGVkQWRJZD1BMDg2NTM1Nkw4OExCRzhKWFo2TiZ3aWRnZXROYW1lPXNwX2F0ZiZhY3Rpb249Y2xpY2tSZWRpcmVjdCZkb05vdExvZ0NsaWNrPXRydWU=

https://www.amazon.com/EpoxAcast-Clear-Casting-Epoxy-Resin/dp/B00IRC42PC/ref=sr_1_2?dchild=1&keywords=epoxacast+650&qid=1626216068&sr=8-2

I also decided to buy an additional item from amazon that I could use to add to the epoxy before it cured and give it an interesting effect: https://www.amazon.com/Glow-Dark-Pigment-Powder-Fluorescent/dp/B01K081NCA/ref=sr_1_1_sspa?dchild=1&keywords=smooth%2Bon%2Bignite&qid=1626216704&sr=8-1-spons&spLa=ZW5jcnlwdGVkUXVhbGlmaWVyPUExTUlZUFg1STdPNzJRJmVuY3J5cHRlZElkPUEwMTc0MzQ4SE1GSFowSTg4TkJMJmVuY3J5cHRlZEFkSWQ9QTA2MzYyOTRDWDJOUUNENlczWkkmd2lkZ2V0TmFtZT1zcF9hdGYmYWN0aW9uPWNsaWNrUmVkaXJlY3QmZG9Ob3RMb2dDbGljaz10cnVl&th=1

It's a glow in the dark pigment that can be mixed in and cured. I thought it might be a cool thing to try adding to some test parts just to see it's effect. It'd be pretty cool to have markings on both the throttle and brake pedals that glowed in the dark environment of the foot well. I was intending to cover the whole thing with the rubber spray coating from the hand grips for extra grip as well, so it might be fun to mask off an area of both the pedals so that a design can be put onto it. Maybe a 'play' and 'stop' button on each pedal. I bought fluorescent green, but if it works out well, I may have to get other colors as well. Since this was an Amazon purchase, we couldn't go through normal means to buy it. I had to get the Captain to request a P-card, then purchase it that way.

While waiting for the products to arrive I got started on making the parts to be molded. I got Felipe to print them out in PLA with the highest quality print settings.

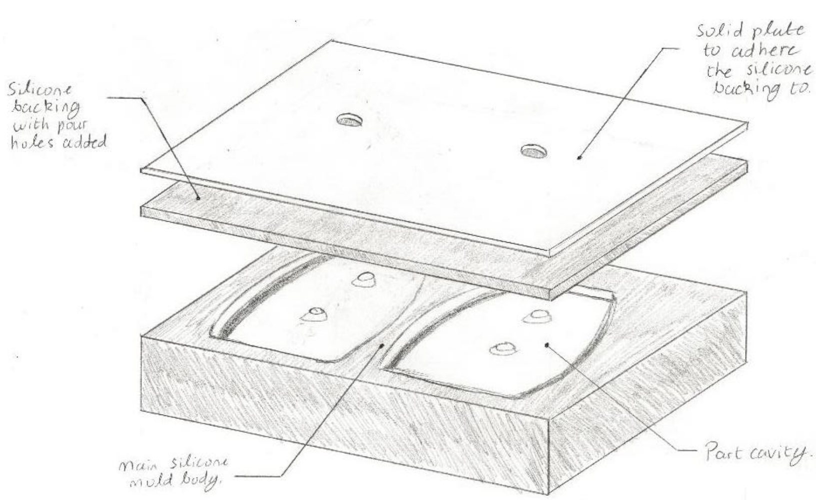


While they come out very smooth from the printer, they inevitably had the striations from the print process. I didn't want those to transfer to the final product, so I sanded them down smooth with various grits of sand paper till I got most of the striations out. After that I cleaned them with alcohol, then sprayed them with several coats of high build primer to coat the rest of the scratches and striations I couldn't get with the sand paper.

Once I had several layers built up, I sanded back them of the spots and smoothed them out. There were a few deeper lines that I couldn't get out with the primer however, so I filled those spots in with wood filler, sanded them back smooth and flush with the rest of the parts, then resprayed them with primer (hint: using a very small amount of water on the wood filler helps lower the viscosity and makes it much easier to spread around smoothly). After that coat had cured, I sprayed several more coats of primer sealer to give a smooth sealed finish to the primer in preparation of casting. The next task was to built the box for the parts to be cast in. I ended up using a different process to what was shown in the video above. I ended up taking a piece of MDF and squared the edges, then cut some polycarbonate we had lying around the shop and cut it to cover the top of the MDF (to create a smooth surface to cast on) and to make the side pieces to hold the silicone in. I used hot glue to bold the edges to the bottom and to seal the cracks on the inside, much like silicone caulking. Side note: don't use silicone caulk to seal the edges. One of the biggest advantages of silicone to cast is that it only sticks to one thing: more silicone. So, if you use caulking or a silicone spatula to mix/pour the silicone, it will ruin it.

The next challenge was getting the parts to be sealed to the bottom. The way I had planned to make the mold was to seal the bottom side of the cups to the bottom of the mold, then pour the mold over it. Then the parts would pop out the back. I would then make a "back plate" so to speak from a thin pour of silicone with some pour holes cast into it, the adhered to a stiff piece of material like wood or metal plate. I would then place this backing plate onto the other half of the mold and pour the epoxy through the pour holes into the main cavity of the mold. This would eliminate the need for key holes like what was shown in the demo video above. I realize that sounds a bit unclear, so I'll draw a quick diagram.

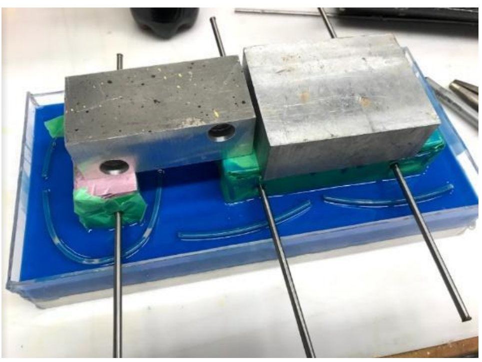
The silicone backing is used because like I mentioned earlier, silicone doesn't stick to anything but other silicone, so the epoxy won't stick to it. I used a backplate without the silicone, the epoxy would bond to it and make getting the parts out impossible.



Getting back to adhering the parts to the bottom of the mold, I tried using purple glue stick and hair spray (commonly used on print beds to help parts stick), but they were very underwhelming to say the least. This is partly due to the fact that the 3D printed pieces are bowed at the bottom where they slightly lifted from the print bed. Getting these to stick hard enough was the challenge because it just wanted to peel back up. I then tried using some fast-dry wood glue we had laying around, figuring it wouldn't be too bad. It never set because the pieces being bonded were non-porous or moist (like wood would be), so it never dried. After that I covered the bottom of the parts with packing tape, and covered the bottom of the mold box in packing tape as well. I then used Super 77 spray adhesive on the bottom of the parts and the bottom of the mold, waited for the surfaces to become tacky, then carefully pressed them onto each other, and added pieces of scrap aluminum as weights to make sure the parts were fully pressed down. I then left these to dry overnight.

The purpose of the packing tape is to help release the parts when I demold the mold thing. Had I sprayed directly onto the surfaces, they would have been glued strongly to the bottom of the mold. But having that sticky backed packing tape means that I can peel the parts off the tape, and peel the tape of the bottom without damage. When I came back after leaving it overnight the parts seemed solidly stuck together, so I was satisfied with that.

By that time, the silicone had arrived, as well as the casting epoxy and glow in the dark pigment powder, so I was able to start making the molds. The Mold Star silicone is a 1:1 ratio by volume, so I filled a small disposable cup with water, marked the level with a sharpie, then poured the water into another cup, and marked it as well. I did that to make sure I had the same volume of part A and B in each cup, since we don't have graduated measuring cups. The directions said to stir each component before mixing, so Parth (new member) and I did that. We then poured each part into the aforementioned cups, mixed some more, then mixed the two parts in a larger cup and stirred it together. It's important to make sure the two parts are thoroughly mixed, as the silicone cures through a chemical reaction. If there are parts that are not mixed very well, the cast won't be as strong since there will be sections of the mold that either didn't cure, or didn't cure strongly. Since the two parts are different colors, it's easy to see if it hasn't mixed well. If there is any marbling, you need to stir a lot more. Once the two parts were mixed well, we poured it into the mold box. The problem was that I didn't end up ordering enough silicone to fill the mold up. To be fair I didn't know how much was in the kit, and I hadn't made the mold box, so I wouldn't have known how much I needed anyway. I tried using some foam blocks wrapped in painters' tape (it ended up preventing some of the silicone to cure, so it was still sticky around the sticky side of the tape when I demolded it) to fill up unused space in the mold, but still didn't have enough. I needed about half an inch more. The pot life of the silicone is about 45 minutes, and the demold time is 6 hours.



I've ordered some more silicone and am waiting for it to arrive. Fortunately, like I had mentioned earlier silicone will stick to itself, so as long as I clean up the surface of the mold to remove the oils from my hand, I could be able to pour silicone over it and it will be one solid piece. Other than that, the mold came out really well. There are virtually no bubbles, even though I wasn't super careful about it, or used a degassing chamber. All I did was tap on the sides of the mold a lot to knock the bubbles free. Also, it comes out as a very nice lapis blue.

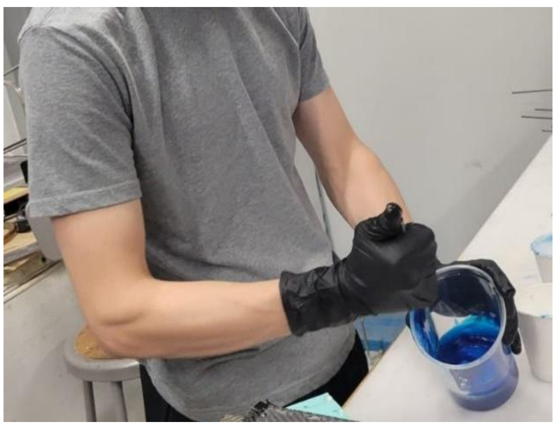
8/2/21

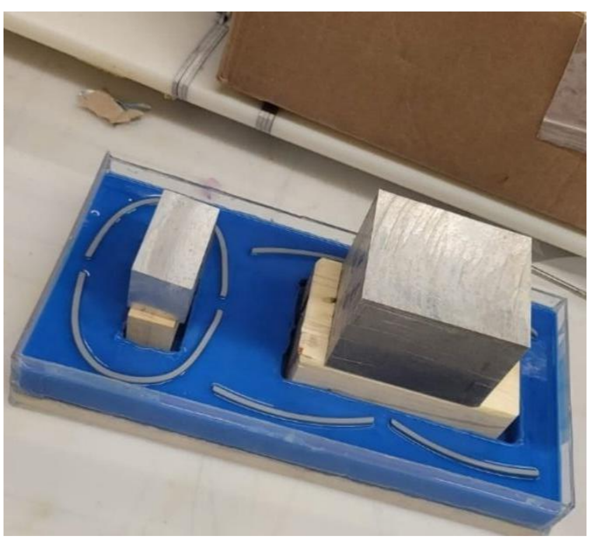
Alright, I got more silicone in tonight, so we'll try filling the rest of the mold. I'll try to grab some more pictures to show the process as well.

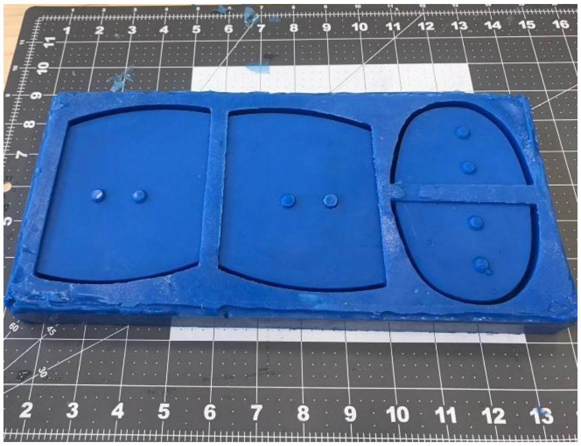
8/5/21

The cast ended up being pretty easy. I poured in more silicone in one corner and let it self-level. Again, very few bubbles even though I didn't try very hard to get them out.







The silicone ended up sticking to itself really well. I did clean off the surface well with rubbing alcohol before hand to remove the finger prints/skin oils to help it adhere better. It's like it was cast as one solid piece all along.

The only problem is that with the wood and foam blocks I used the silicone ended up being too thin around the toe cup section, so I'll have to fill in the area with a little bit more silicone, then use a piece of MDF to fill the rest and give it strength. Otherwise the weight of the epoxy will cause the mold to deform and the part will come out wrong.

8/17/21

Even though I added more silicone, I was still going to add some MDF pieces to fill out the rest of the mold just to make sure. However, while looking at some playdoh we had lying around from someone's project last year, I realized I could make a custom infill piece for the two caps in the mold with the clay. I did that a little bit ago and am waiting for the playdoh to set. I want these pieces to remain removable to help with demolding of the final parts when I eventually get to casting. We'll see how the playdoh comes out.

Throttle and Brakes System project will continue on page --.

Steering Column Project (Continued from page 92):

It's been a while since I've worked on the column, but it needs to get done soon. School has started and I'm running out of time.

To recap for myself, the outer diameter of the column is increasing from 5/8" to ¾" due to the rack coupler I had been using not being available. This won't be a big deal, just something to note. The outer diameter of the quick release shaft is 5/8". In the previous year's cars, we had used the lathe to shave down the shaft to the inner diameter of the column tube to make it fit. This won't be necessary any more.

Looking at online suppliers, 4130 (what the chassis currently uses) is the only material available for the size that I would need for this application. 4130 is a strong steel, so I feel confident in using it for the steering column. Never the less, it's still always good to check out the material properties and applications from trusted online data bases to ensure it will be right for our needs.

Alloy Steel 4130 (Chromoly)

Grade Summary: A chromium molybdenum alloy steel. Alloy 4130 is a widely used Aircraft Steel due to its weldability, ease of fabrication and mild harden ability. Alloy 4130 is also vacuum degassed to meet the magnetic particle inspection standards of AMS 2301.

Typical Applications: Aircraft seating, Roll cages, Motorsport applications

Available products: Round Bar, Rectangular Bar, Round Tube and Plate.

This item is not available online, please contact your local store for pricing and size availability

Typical Chemical Analysis:

1. *C – .28/.33 Max. *Mn – .40/.60 *P – .035 Max.
2. *S – .04 Max. *Si – .15/.35 *Cr – .80/1.10 *Mo – .15/.25

Typical Mechanical Properties**

Tensile Strength (PSI) 97,000
Yield Point (PSI) 52,000
Elongation*** 28
Brinell Hardness 217
* Chemical Analysis will vary on each heat number

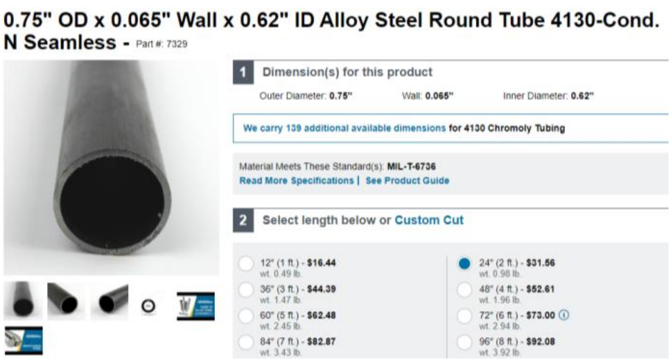
**All values are averaged values and are representative.

*** % in 2"

Source: https://www.metalsupermarkets.com/metals/alloy-steel/alloy-steel-4130/

As it says to the left, it is a good material for structural applications, such as aircraft and motorsports.

With that in mind, I can find the right tubing I need. Again, I'll use online metals, as I've had some luck with them before. Although TW metals is a sponsor, we have been having poor interactions with them and are effectively no longer using their business after they took forever to ship us the wrong metal. I only note that for future reference. Perhaps we could get online metals to sponsor us and drop TW.



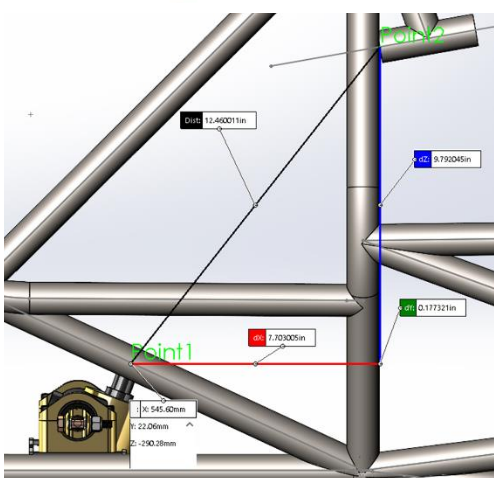
The steel I found is on the left and can be found here: https://www.onlinemetals.com/en/buy/alloy-steel/0-75-od-x-0-065-wall-x-0-62-id-alloy-steel-round-tube-4130-cond-n-seamless/pid/7329

It's $31.56 for 2 feet and $44.39 for 3. We shouldn't need much more than that.

Getting back to the design, I can only make the U joints bend to a 30 degrees interior angle before they begin to bind:

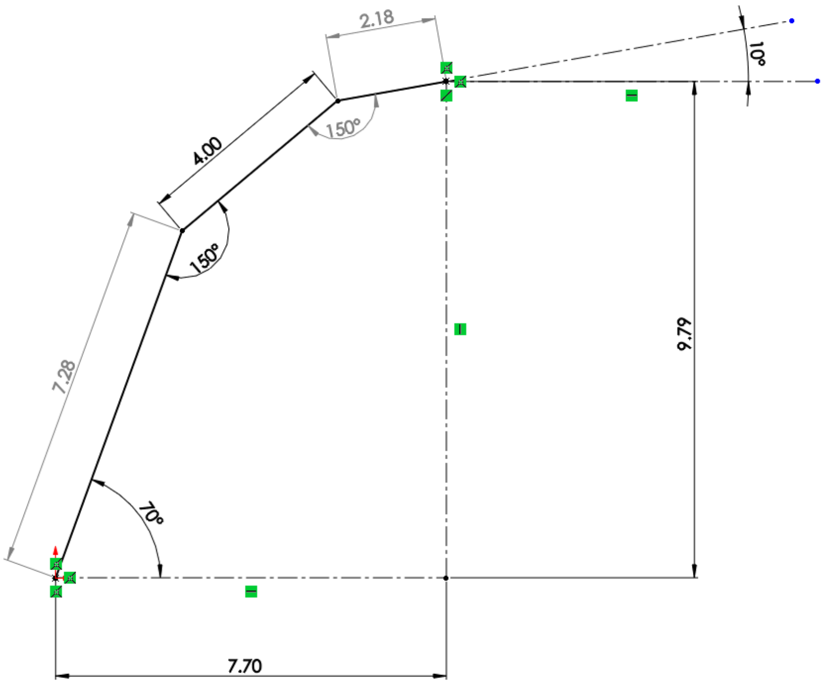
This means I have limited space to work with, making this more challenging.

To recap, these are the points I need to connect with the column:



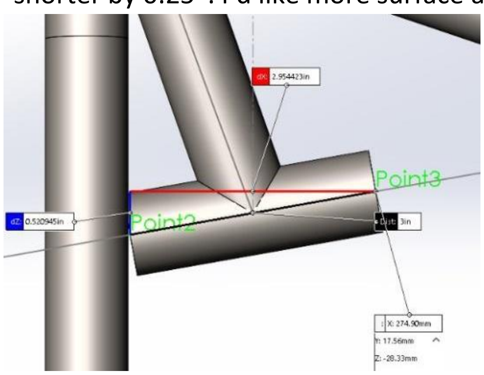
The steering rack is already set at a 20 deg angle relative to the vertical, meaning 70 deg angle relative to the bottom tube and ground plane. The column holder (represented currently as a solid tube) is at a 10 deg angle relative to the horizontal.

Using this information, as well as the distances obtained from the measurement in the picture on the left, I drew a layout sketch in Solidworks to find the lengths of the tubes. Previously I had said I'd do a hand sketch, but for this type of work where precision is paramount, Solidworks was the much better and faster choice. The results of the sketch are as follows:

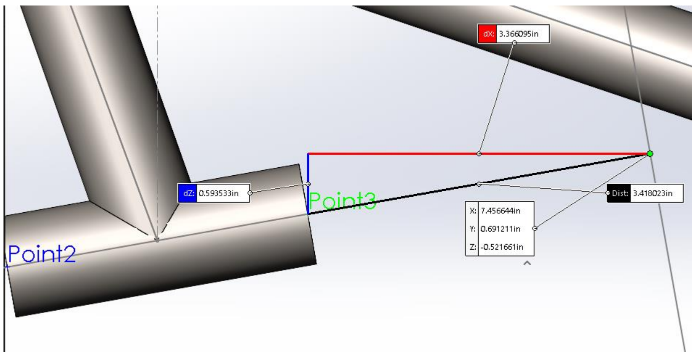


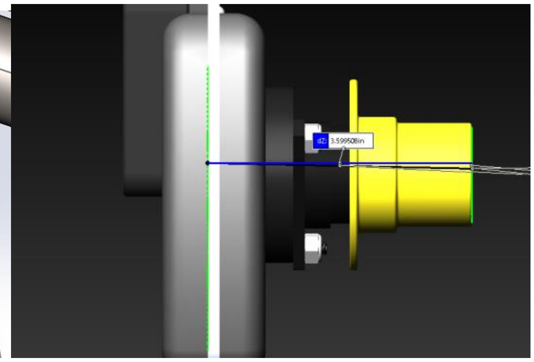
The location of this sketch is: **X:\Formula SAE\Engineering\FSAE 2019-2021\Cockpit and Controls\SR21-CC-A001-Reference Models\SR21-CC-A001-S005-Steering Column Reference**. The greyed-out constraints are over constrained and are there just for measurement purposes. The length of the middle tube is variable to a certain extent. I played around with the angles and distances where I could, but the spacing is still going to be hella tight in the car. Each corner vertex is where the U joint center is going to be. The U joints are ~~1.875” (1 7/8)~~ from center to end. Meaning that I’ll only have 0.3” of space left. I still need space to weld a spacer onto the column where it meets the tube to prevent it from moving around, which will eat up even more room. I need more space. To solve this, I plan on making the column holder shorter by 0.5” (0.25” on each end, more on this later), and I also plan on making the tube at the end of the U joint shorter by 0.25”. I’d like more surface area to be able to make more rosette welds onto, but in this case, I need the space. That will give me 0.805” of space to work with. That should be enough to do what I need to do. The other apex U joints will also be a tight fit as well, but not as bad as this one.

(It’s actually 1.75. More on pg. 106)



The reason I need more space on both sides of the column hanger (the 0.25” of space I mentioned above) is because I’m having spacing issues on that side too. Because of the rule about the top of the steering wheel not being allowed to extend above the top most portion of the knee hoop in any direction (to explain this in plain English: the steering wheel is not a perfect circle, so it is wider than it is tall. This means when the wheel is turned 90 degrees to either side, it is sticking up higher than it is when the wheels are straight. When the wheel is in this orientation, it is not allowed to stick out of the cockpit any higher than the very top most surface of the front [knee] hoop. This is so that if the car rolls over, it will not bend the wheel into the driver’s legs), I am limited to how far backwards (towards the driver) I can extend the steering wheel. The sketch below depicts the line of the steering column, and the front face of the steering wheel. By measuring the distance, I have about 3.418” of space from the front of the column hanger to the front face of the wheel. Here’s the problem: the wheel (the way it’s currently designed) is 3.5725” long when measured from the front face of the wheel to the back of the quick release. (You’ll see in the model that the measured number is 5.599, but this is because the model is slightly inaccurate. The 3.5725” was measured from the actual part we’ll be using).





Obviously, you can’t fit a part that is 3.5725” long is a space only 3.418” long. So more space needs to be made. Hence, the reason for the 0.25” reduction in length of the column hanger. That will give me a bit more room to work with. The other area I am making room with is the spacer on the back of the steering wheel. Recall that purpose of adding the spacer is to allow the driver some degree of adjustability when the steering column is installed. If I made a design or manufacturing error and the column was too long or too short (meaning the wheel would be in the wrong place for the driver and thus make it uncomfortable/impossible to sit in the car) then the driver could request a different spacer to get the placement just right. That’s why I had gone with the somewhat arbitrary 1” figure. You could make it bigger or smaller. But I need the space anyway, so I’m going to shorten the spacer to just half an inch. It still allows us some adjustability, but gives us more space to work with.

By doing that I now have a 0.433” gap. Again, not much, but enough to work with. Now that I have the rough dimensions I need, I can start to make the actual CAD model for the steering column.

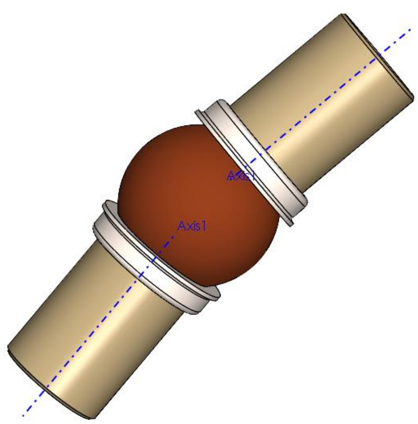
8/28/21

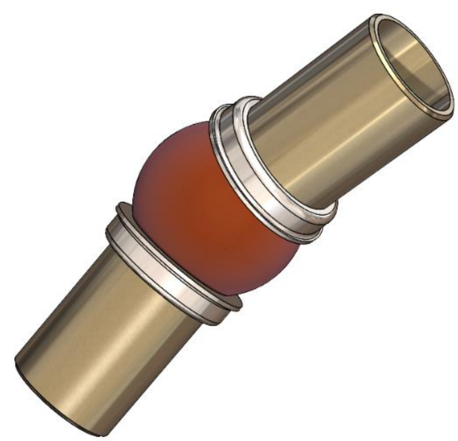
****Note:** Turns out I accidentally measured the Apex joint wrong, and it is slightly shorter than what I had though. It’s actually 3 5/8 in total length, not 3 7/8, meaning I have an extra 1/8” inch on each of the sides. It’s also listed as that in the data sheet from Pegasus Motorsports (https://www.pegasusautoracing.com/productselection.asp?Product=1490). It doesn’t really change anything design-wise, but it will help with the space issue. I may not even need to trim the Apex joint, but we’ll see.

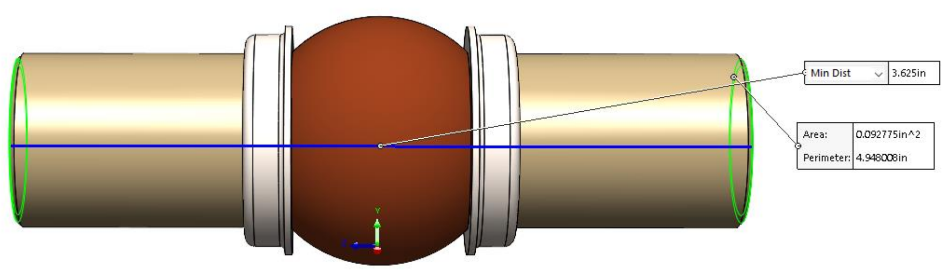
I’ve started working on the design of the steering rack. The first thing to do is to design the Apex joints I talked about above. There is an older model I made, but I decided to make a slightly improved version of it, partly because I needed to update the sizing anyway (from 5/8 to ¾), but also because I wanted to improve the model’s mates. In Solidworks, there are several different types of assembly mates. There are simpler “Standard Mates” that we use almost exclusively use here at Formula, mostly because they do most of what we need them to do. I don’t think anyone else on this team has used anything other than the standard mates before. The problem is they are static mates. In other words, once mated, they can’t move. This can be a problem when you’re trying to design a part that has at least

1 degree of freedom, such as a U joint. Once mated, I can't get it to bend if I need it to. I ran into this problem when designing the steering column for SR19. When I made modeled everything and had gone to install it in the Master Assembly model, I couldn't get it to mate right. It's difficult to explain through text why, but basically, because I couldn't *perfectly* measure every length and angle I needed, the model was just *slightly* off, just enough to not allow me to mate it properly in the model. And I couldn't adjust it because the model of the Apex joint was fixed; I couldn't bend it into place.

To fix that issue, I used one of the other types of mate: "Mechanical Mates". These are supposed to be used to make things like hinger work. They constrain the model's parts, but still allow them to move like they would in real life. I used the Universal Joint mechanical mate, which means that I can now move the Apex joint model around a bit like I could in real life, which should solve the assembly problem later. The parts of the model are fairly simple: I took measurements from the actual part and made two separate pieces. The first is the metal bit that we attach to the column tube. The second is a simple ball that represents the rubber boot that covers that actual U joint part of the mechanism. It would be waaaay to hard to try and model this accurately and I don't have time for that, so a simple ball will work just fine. I assembled two of the metal parts together with the ball and used the "Universal Joint" mate on the two reference axes of the metal parts to make them be able to move.



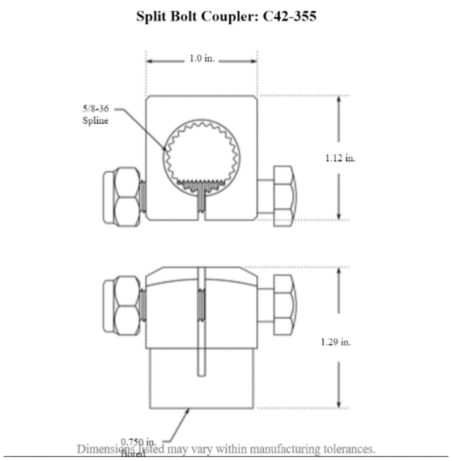




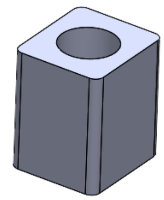
The next thing I need to model is the Splined Coupler for the column end where it meets the steering rack.

8/31/21

I found the diagram for the coupler I need here: https://www.pro-werks.com/partdetail/C42-355-B/

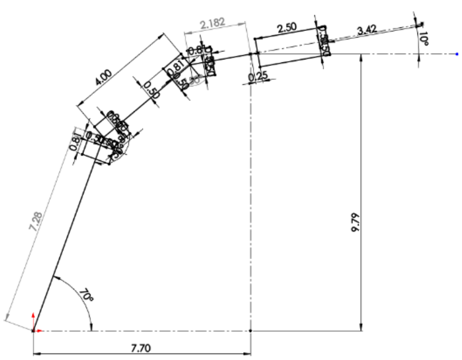


Of course, it's not super useful. The diagram doesn't show the secondary dimensions which means I can't model an accurate version of it in Solidworks, so for the time being I made a crappy model of it that basically just shows the height, which is really what I need anyway. It'll suffice till I get the actual part and can measure and model it correctly.



9/3/21

With that done, I can take measurements from the sketch I did in Solidworks:



I found (through some arithmetic that the tube lengths need to be the following:

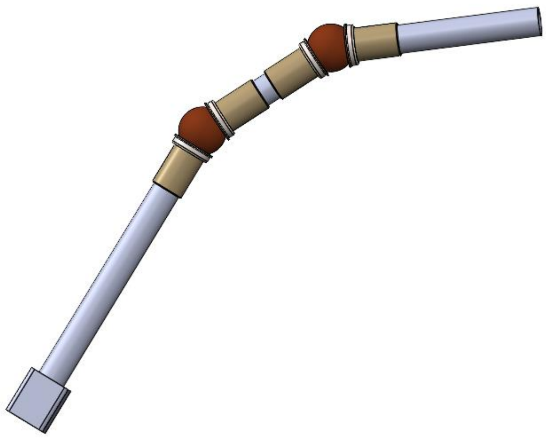
Bottom tube - 6.9686"

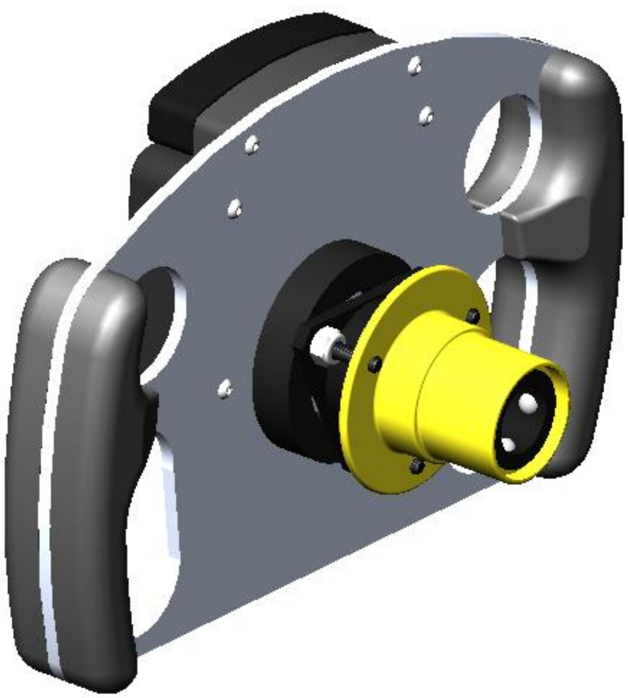
Middle tube - 2.375"

Top tube - 4.7147"

I can't get a super accurate number for the bottom tube since I don't have the Stiletto coupler, but it's close enough that I can change it later. This being the case, the total tube length is 14.0583". I'll order a lot more so we have extra if I make a mistake, but it gives us a ballpark number. Now, finally, we can create the CAD and get assembling.

That wasn't too hard. Probably the easiest part of the whole process. Couple of things to note: I made the bottom tube the fixed component so I could rotate the Apex joints. Second, I didn't put a





distance mate on the simple splined coupler model, as I am planning on measuring the offset when it's installed in the master assembly.

What will be a bit more tough is making the clamshell for the column to hold it in place.

To do that there are a couple of things I need to do first. One is fix the length of the steering wheel spacer, and the second is to make the quick release shaft that connects to the quick release on the steering wheel. I forgot I didn't have a model of that. Fortunately, I do have the physical part, so I can make a halfway decent model of it.

Once I